ELECTRIC CRANE CONSTRUCTION.

Frontispiece.

ELECTRIC

CRANE CONSTRUCTION.

BY

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With over 366 Figures and 23 Tables.

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PREFACE.

THIS book is based chiefly on notes and data compiled by the Author in the course of his work, and is intended primarily as a reference book for engineers engaged in the design and construction of electric cranes, and for those who require to instal, test, and use them. It is hoped, however, that it will also be of some interest to students, and that some of the chapters, such as those on structural steelwork, machinery, and toothed gear, will be of use to engineers engaged on other classes of work.

The first seven chapters each deal with one of the principal types of crane. Each of these chapters commences with a description of the construction and duty of the crane, not with the idea of imparting elementary information, but as a recapitulation of the purpose for which the crane is intended. The stresses and other points peculiar to the type of crane are next dealt with, and then drawings and descriptions of typical cranes are given, test figures being also added in several cases. Points common to all forms of cranes are dealt with in the succeeding chapters, and in a final chapter crane installations are discussed and described.

The description of the 160-ton revolving cantilever crane in Chapter vi. is taken partly from an article in *The Engineer* and partly from information supplied to the Author by the makers. Figures to which an asterisk is appended are from blocks supplied by *Engineering*.

In the chapter on Crane Structures, the design of struts and columns with lattice bracing has been dealt with somewhat fully, as compression members, such as jibs and towers, are an important feature in crane work, and it is hoped that Tables XX. to XXIII. giving the breaking loads of British Standard Steel Sections when employed as struts will be found of use. In dealing with girders the effect of rolling loads is emphasised with a view to showing that the maximum stress does not occur on all the members when the load is in the position which gives the maximum bending moment, and that it is, therefore, necessary to take out the stresses for different positions of the load.

In the chapter on Design of Machinery the question of deflection receives consideration, as the deflection of shafts or of the structure supporting them is liable to lead to excessive transmission losses and to noise and excessive wear in toothed gearing.

The question of deflection also requires to be borne in mind in cases where the use of high-strength steel is contemplated with a view to reduction of weight. Sheffield firms are now turning out steel having a tensile strength as high as 123 tons per square inch with an elastic limit of 116 tons per square inch. The modulus of elasticity of this steel is, however, about the same as that of mild steel. Consequently, in parts designed for a very small permissible deflection this steel offers no advantage over ordinary mild steel as it would require to be of the same dimensions.

In the chapter on Brakes the forms of brake mostly in use are described, together with calculations for their design. Some space has been given to showing in detail the method of ascertaining the pressure per square inch between the frictional surfaces of band brakes as this is an important point affecting their design.

In dealing with Toothed Gearing the involute form of tooth only has been treated, the cycloidal form being of no practical interest at the present time. An analysis of the tangential and radial forces in involute gear, both when friction is neglected and when it is taken into account, is given, and this leads up to a comparatively simple formula for the efficiency of involute gear.

In worm gear the length of the line of contact between the threads and teeth is a very important dimension, as it determines the load which the gear can transmit. Another point with which the Author has dealt is a method of ascertaining the increased load which worm gear will safely carry when working intermittently, as compared with the load which it will carry when working continuously.

The subject of the stress set up in the outer wires of wire ropes when bent and the energy lost in running them round sheaves is one on which there seems to be considerable difference of opinion among engineers.

Mr. R. W. Chapman's paper, from which extracts are given, is a valuable contribution to the subject and will no doubt help to put the matter on a definite basis.

Table No. XIV., which the Author has worked out from Mr. Chapman's formulæ, will save designers the trouble of making calculations for the more commonly used types and sizes of ropes.

The formulæ for calculating magnet windings will, the Author trusts, prove useful to designers, as they obviate the necessity of reference to tables of resistance, &c.

The question of the correct method of testing crane motors is being much discussed at present. The Author favours plain continuous test runs in preference to more complicated methods.

As, however, the rate of rise of temperature of motors decreases as their size increases, a uniform length of test run for crane motors irrespective of their size cannot give satisfactory results in practice.

In the chapter on Motors a method is shown for calculating approximately the length of continuous run at a given load which will give the same temperature rise as that attained ultimately by the motor when working intermittently on the same load with a given load factor, and this is sufficiently accurate for all practical purposes.

The formulæ for the calculation of Controller Resistances have been used by the Author for some years, and are of service not only for designing controllers, but for checking designs submitted by controller makers.

The planning of crane installations is a subject calling for broader treatment than the design of individual cranes. Before proceeding to design an installation it is necessary to make a careful study of the duty which the installation as a whole is to perform, and this point being settled the engineer can then proceed to determine the most suitable types of crane to be employed and their number, sizes and speeds, makers standard designs being utilised as far as possible.

The chapter on Crane Installations is therefore limited to dealing with the question of types of crane, employment of accumulators on crane circuits and so on, and in the descriptions which are given of installations of cranes, only those points which appear to be of general interest are brought out, smaller details being omitted.

The Author wishes to take this opportunity of expressing his thanks to the various firms mentioned in the book, and to the following gentlemen, some of whom have kindly assisted him with data, photographs, and drawings of cranes, &c., of their manufacture or in their employment, while others have been good enough to send him copies of papers contributed by them to various scientific societies, extracts from these papers being given in various parts of the book:—Admiral R. H. Bacon, D.S.O., C.V.O.; R. A. Bruce, Esq., A.M.Inst.C.E.; Basil T. Courtney, Esq.; F. W. Crawter, Esq., A.M.Inst.C.E.; F. W. Davis, Esq., A.M.Inst.C.E.; Lieut-Col. C. Q. Henriques, M.Inst.C.E.; Prof. W. E. Lilly, M.A., D.Sc.; R. Matthews, Esq., M.Inst.C.E.; Dr. Robert Pohl, A.M.I.E.E.; Vincent L. Raven, Esq., M.Inst.C.E.; W. L. Spence, Esq., A.M.Inst.C.E.; W. Stokes, Esq., M.Inst.C.E.; E. J. Taylor, Esq.; John Temperley, Esq.

The Author also desires to thank the Councils of the Institution of Civil Engineers and the Institution of Mechanical Engineers for the loan of blocks and for permission to make extracts from their Proceedings, the extracts being further acknowledged at those parts of the book where they are given.

C. W. HILL.

WESTMINSTER, *January*, 1911.

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LIST OF SYMBOLS.

$\mathbf{A}$	Current in amperes.
$\mathbf{A}_a$	„ in armature (amperes).
$\mathbf{A}_g$	Ampere turns required for air gap.
$\mathbf{A}_m$	Mean current taken by motor when starting.
$\mathbf{A}_o$	Armature current with motor standing.
$\mathbf{A}_t$	Ampere turns.
$\mathbf{A}_1$	Maximum current taken by motor when starting.
A	Sectional area in square inches.
A_c	Arc of action.
A_m	See Figs. 248 to 252.
A_x	„ „
a	Angle of incidence; also angle of face of rack teeth, and of worm threads on central section, see Fig. 276; also angle of rope with horizon at point of support, see Figs. 55, 56, 57, and 58.
a_c	Angle of leg with horizon, in sheer legs.
a_r	Rate of acceleration, f.p.s. per second.
a_t	Angle of tooth of helical wheel, Fig. 264; also angle of worm thread, Fig. 291.
a_1, a_2, a_3, a_4	See Fig. 276.
a_5	Angle of rope between carriage and nearer tower. See Figs. 58, 62, and 63.
a_6	„ „ „ further tower. „ „
a_7, a_8	See Fig. 63.
$\mathbf{B}$	Magnetic flux density in lines per sq. in.
B_1, B_2, B_3, etc.	Breadth in inches of parts numbered 1, 2, 3, etc.
$\mathbf{C}$	Constant for temperature.
C_t	Temperature rise C. in time t.
c	Constant for roller bearings, see Table X.; also watts per square inch in Chapter on Design of Magnets.
$\mathbf{D}$	Depth, in inches, of winding on coil.
D	Distance in feet.
D_g	Effective depth of girder.
D_i	Dip of imaginary cableway.
D_m	Distance in feet from point m.
D_o	„ „ o.
D_1	Dip in feet of unloaded rope. Fig. 55.
D_2	„ due to load. Fig. 56.
D_3	„ due to load plus weight of rope. Fig. 57.
D_4	„ on half span due to weight of rope, Fig. 57; also dip in feet due to weight of rope on portion of span, Fig. 63.
D_5	Distance in feet from carriage to nearer tower. Figs. 55, 62, and 63.
D_6	Dip in feet of upper half of hauling rope when loaded carriage is at centre of span.

D_7	Dip in feet at any distance D_5 due to weight of rope only. Fig. 55.
D_8	,, ,, D_5 due to load only. Fig. 62.
D_9	,, ,, D_5 due to load plus weight of rope. Fig. 63.
d	Diameter in inches, generally.
	Also diameter of pitch circle of wheel.
	,, ,, rope.
	,, ,, shaft or roller.
	,, mean diameter of conical roller.
d_r	Rate of deceleration, f.p.s. per second.
d_y	Least diameter of section, inches.
d_1	Distortion of centre line of worm thread. Fig. 276.
d_2	Half the difference of thickness of worm thread between top and bottom. Fig. 276.
d_3	Diameter of pitch circle of worm.
d_4	Distance in inches from centre of column to centre of load in an eccentrically loaded column.
E	Modulus of elasticity.
E_r	,, ,, of rope.
e	Efficiency.
	Also base of hyperbolic logarithms = 2·718.
e_m	Efficiency of motor.
e_y	Distance from neutral axis to extreme edge of section.
F	Force required to accelerate load moving in straight line horizontally or vertically, Chapter ix.; also tangential force at point of contact of driven wheel, Chapter xiii.; also tangential force at rim of brake drum, Chapter xii.
F_1	Total force required for acceleration—*i.e.*, linear *plus* rotary, Chapter ix. Also, tangential force at point of contact of driving wheel, Chapter xiii.
F_2	Radial thrust on driven wheel.
F_3	,, ,, driving ,,
F_4	Force normal to working surfaces = load on shaft due to driving force.
F_5	Tractive effort in lbs. per ton based on E.H.P. at crane terminals.
F_6	Ditto., to overcome mechanical losses only.
F_7	Force required in tons to pull carriage of cableway up incline.
f	Working stress in teeth of wheels.
	Also, breaking strength of rope, tons per square inch, Chapter vii.
	,, working stress, tons per square inch, in columns or beams, Chapter x.
	,, in some cases as shown by the context f represents the stress contemplated in the formula.
f_b	Stress, tons per square inch, caused by bending of strut.
f_c	Ultimate crushing strength, tons per square inch.
f_e	Elastic limit, tons per square inch.
f_l	Limit of stress, tons per square inch, on a strut of one wave length.
f_s	,, at extreme edge of section of bent strut.
f_t	Tensile strength, tons per square inch.
G	Length of air gap in inches.
G	Gauge of rails, feet.
g	Mean girth in inches.
g_1	External girth in inches.

H Horse-power.
h Thickness of tooth at weakest part. Fig. 273.

I Moment of inertia.

K Constant for strength of teeth, see Table XII.
K_1 „ load on worm wheels.
K_2 „ „ „ Fig. 290.
k_3 „ secondary flexure, of columns.

$\mathbf{L}$ Length of coil in inches.
L Distance in feet from centre of load to centre of balance weight. Fig. 21.
L_e Length of leg, in sheer legs.
L_p „ panel in plate girder.
L_r „ rope in feet when fully loaded. Fig. 57.
L_s „ „ hanging freely between two supports with no load except that due to its own weight. Fig. 55.
L_u Length of rope when under no stress whatever. When lying full length on the ground, for instance.
L_1 Load per wheel in tons.
L_2 Load in tons.
Also, load in tons on rivet.
L_3 Load in tons distributed.
l Virtual length of column in inches. See Figs. 86 to 89.
Also, length of cantilever, or half length of girder supported at both ends.
„ height of tooth from weakest part. See Fig. 273.
„ length of shaft or roller in inches.
„ in formulæ for deflection and centrifugal whirling of shafts, the distance between centres of bearings in inches.
l_1 Height of worm thread on central section. Fig. 276.
l_2 „ „ any given plane of section. Fig. 276.
l_3 Height of wheel tooth above pitch line. Fig. 283.
l_4 Length of contact line.

M Maximum temperature rise C.
M_b Bending moment.
M_{bA}, M_{bB}, M_{bC} etc. Bending moment at points A, B, C, etc., along girder.

$\mathbf{N}$ Total magnetic flux.
N_b, N_c, N_d, etc. Numbers of teeth in driven wheels on shafts b, c, d, etc.
N_r Number of rollers.
N_w „ wheels.
n „ teeth in wheel.
Also, number of steps in controller.
n_a, n_b, n_c, etc. Number of teeth in driving wheels on shafts a, b, c, etc.
n_r Number of rivets in panel of plate girder.
n_s „ friction surfaces in disc brake.
n_1 „ teeth in smaller wheel of pair.
n_2 „ „ in driving wheel.
n_3 „ „ in driven „
n_4 „ „ on which load is being taken.

P Axial thrust on screw.
Also, total radial pressure in lbs. on coil and band brakes.
„ pressure in lbs. on one block of block brakes. Fig. 232.
„ total end pressure in lbs. on disc brake.
P_r Pressure in tons per inch width of roller, in rings of live rollers.
P_1 Horizontal force at foot of back leg in sheer legs.
Also, horizontal force, in one panel, on flange of plate girder.
P_2 Vertical force at foot of back leg in sheer legs.
P_3 Pressure on rope in lbs. per linear foot due to wind.
P_4 Total resultant pressure on rope in tons.
P_5 Total wind pressure in tons on jib when vertical. In side view.
p Pitch in inches. Teeth of wheels, etc.
Also, pitch of rivets in inches.
p_m Maximum stress, tons per square inch in straight strut, consistent with stability. Euler.
p_n Ditto., when direct stress is allowed for. Rankine.
p_r Ditto., allowing also for secondary flexure. Prof. Lilly.
p_1 Lead of worm or screw in inches.
p_2 Wind pressure per square feet.

$\mathbf{R}$ Resistance in ohms.
$\mathbf{R}_a$ „ of armature and brush contacts.
$\mathbf{R}_e$ „ external to motor armature.
$\mathbf{R}_m$ „ of motor and connections to controller.
$\mathbf{R}_s$ „ in parallel with armature.
R Horizontal radius of jib in feet.
R_1 Mean radius of roller path in feet.
R_2 Radius of balance weight in feet. See $F\,E$ in Fig. 21, or $D\,B$ in Fig. 27a.
r „ in inches, unless otherwise stated.
r_e „ „ external.
r_i „ „ internal.
r_p „ of pitch circle of driving wheel.
r_q „ „ „ driven „
r_1, r_2, r_3, etc. Radius of parts 1, 2, 3, etc.; also
r_1 Radius of driving wheel to point of contact.
r_2 „ driven „ „
r_3 $= A\,R$ in Figs. 248 to 252.
r_4 $= R\,B$ „ „
r_5 Radius of worm to top of threads.
r_6 „ „ bottom of threads.
r_7 Mean frictional radius of screw.
r_8 „ „ thrust collar.
r_9 „ „ brake disc.

$\mathbf{S}$ Cooling surface in square inches.
S Speed in feet per minute.
S_l Half span of imaginary cableway.
S_m Speed f.p.m. at point m.
S_o „ „ o.
S_s Revolutions per minute of motor on slow speed step of controller.

S_1 Speed in feet per second.
S_2 Revolutions per minute.
S_3 Span in feet.
S_4 Surface in square feet of steel lattice structure taking both back and front latticing.
S_5 Shearing force in tons.
s Factor of safety.
Also, number of turns in coil brake.
" " coil of wire.
T Torque, in inch-lbs., unless otherwise stated.
Also, torque of driven wheel in chapter on Toothed Gearing.
T_e Equivalent torque.
T_1 Torque of driving wheel neglecting frictional losses in teeth, Chapter xiii.
Also, tension on coil or brake band on fast side, Chapter xii.
T_2 Torque of driving wheel, allowing for frictional losses in teeth, Chapter xiii.
Also, tension in coil or brake band on slack side, Chapter xii.
T_3 Time constant for heating curves.
T_4 " " cooling "
T_5 Torque on screw.
T_6 Tension in rope, in tons, at centre of span.
T_7 " " at point of support.
T_8 " " per square inch.
t Thickness of insulation on wire.
Also, time in seconds in chapter on Acceleration.
t_a Acceleration period in seconds.
t_b Steady running period in seconds.
t_c Deceleration period in seconds.
t_m Time in seconds from point m.
t_o " " o.
t_1 Thickness of web.
t_2 " flange.

$\mathbf{V}$ E.M.F. in volts.
$\mathbf{V}_1$ Back E.M.F. of motor at full speed and full load.
$\mathbf{V}_2$ Volts across brushes of motor.
$\mathbf{V}_3$ Back E.M.F. of one motor on last series step of series-parallel controller.
V Volume in cubic inches.
V_d Vertical distance in feet between centres of top and bottom bearings of mast of derrick crane.
v Leakage coefficient for magnets.

$\mathbf{W}$ Watts.
$\mathbf{W}_1$ Watts per cubic inch per 1° C. per second.
W Weight on hook in tons.
Also, load on teeth in lbs., Chapter xiii.
" weight on hook plus weight of carriage in chapter on Cableways.
" load in tons on column or beam, Chapter x.
" load on roller bearing in lbs., Chapter xi.
" load in tons on live ring of rollers, Chapter xi.
W_a Weight of armature in lbs.
W_b Load at point A plus half weight of jib. Jib cranes and derricks.

W_d Balance weight in tons.

W_b, W_c, W_d, etc. Weights of parts on shafts b, c, d, etc.

W_e Weight of leg in sheer legs.

W_j „ jib and attached parts.

W_m Maximum load in strut consistent with stability. Euler.

W_n Ditto., allowing for direct stress. Rankine.

W_p Breaking load in tons applied eccentrically to a column.

W_r Weight of rope in tons between points of support.

W_s Weight of rope in tons between carriage and further tower.

W_t Total equivalent weight at a given radius of gyration in chapter on Acceleration.
Also, total weight travelled, traversed, or slewed in chapter on Power for Driving Cranes.

W_x Equivalent weight referred to a given radius of gyration.

W_1, W_2, W_3, etc. Weight of parts 1, 2, 3, etc.

w Weight of cylinder in lbs.
Also, width of teeth, Fig. 273, and width of worm-wheel teeth, Fig. 276.
„ „ conical rollers for live rings. Fig. 227.
„ „ brake drum.

w_r Weight of rope in lbs. per foot.

w_s Weight per cubic inch.

w_1, w_2, w_3 Weight of parts 1, 2, 3, etc.

x, y Horizontal and vertical ordinates respectively of curve $C L$ in Fig. 276.

x_1 $= \frac{w}{p}$ in chapter on Toothed Gearing.

Z Modulus of section.

Z_t Modulus of section with respect to torsion.

α Constant for tensile strength of rope.

β „ weight of rope.

γ Space factor.

δ Deflection in inches.

θ Angle of twist of shaft in degrees.
Also, arc of embrace of band brake in degrees.

κ Radius of gyration in inches.

κ_a „ „ of armature.

κ_b, κ_c, κ_d, etc. „ „ of parts on shafts b, c, d, etc.

κ_1, κ_2, κ_3, etc. „ „ of parts 1, 2, 3, etc.

λ Load factor.

μ Coefficient of friction.

τ Time in years.

ϕ Angle of repose.

ELECTRIC CRANE CONSTRUCTION.

CHAPTER I.

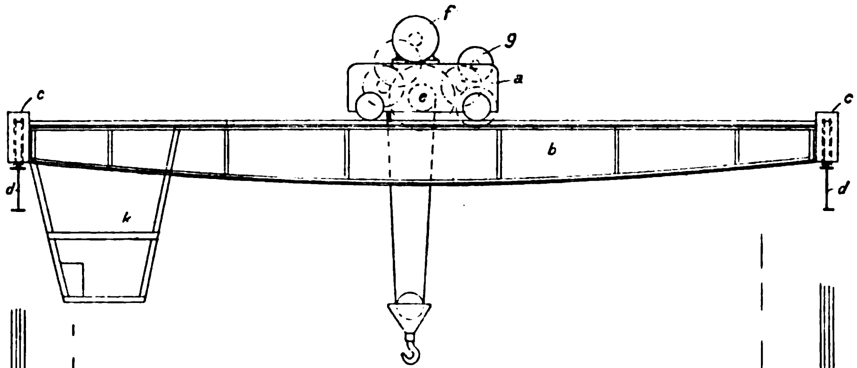
OVERHEAD CRANES.

THE duty of this type of crane, Figs. 1, 2, and 3, is to lift, travel, and deposit loads anywhere within the space shown in dotted lines, the length of the space being limited only by the length of the gantry, *d*, upon which the crane runs. In general construction the crane consists of the crab *a*, which carries the hoisting gear, and which runs on rails on the cross girders *bb*. These cross girders are supported on the end carriages *cc*, which run along the gantry *dd*. On the crab the lifting barrel *e* is driven through a train of gearing by the motor *f*, and one pair of the wheels on which the crab travels is driven by the motor *g*. On the end carriages one wheel or more of each is driven by the motor *h*, which is usually mounted at the centre of the cross girders, the motion being transmitted by the cross shaft *i*.

The controllers, one for each motion, are placed in the driver's cage *k*. Current is conveyed to the motors on the crab by means of bare trolley wires supported on insulators from the cross girders, the crab being provided with sliding contacts which run on the trolley wires. Similar trolley wires are laid along the gantry to convey current from the supply mains to the crane.

The stresses to which the crane structure is subject are of a simple character, the principal ones being the vertical bending and shearing stresses on the cross girders, due to the load on the crane hook, the weight of the crab and the weight of the girders themselves. In the case of cranes having a fast travelling motion, there is a bending stress on the cross girders in the horizontal plane, due to acceleration and retardation. This point is dealt with in the chapter on Starting Torque and Acceleration.

Fig. 1.



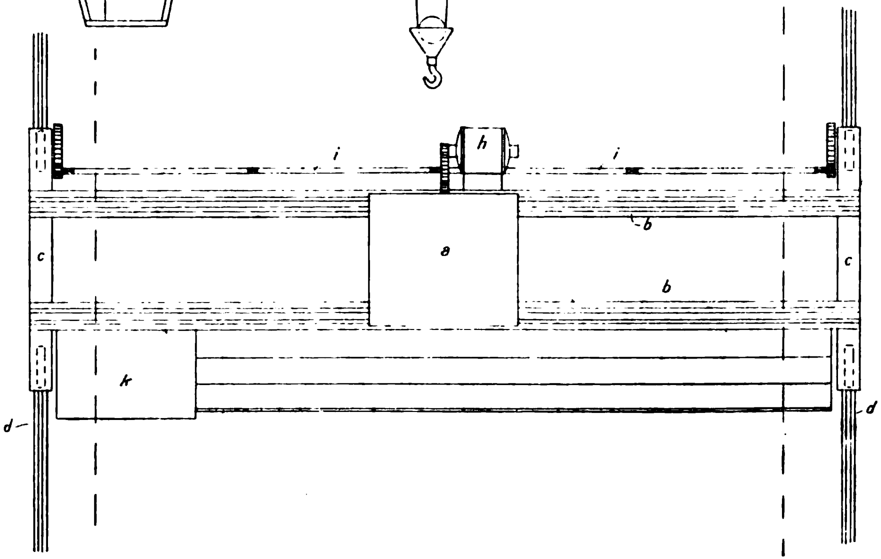
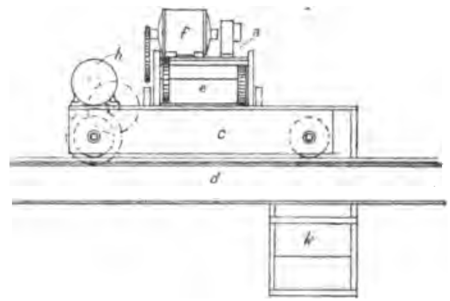


Fig. 3.

Fig. 2.



Overhead Crane.

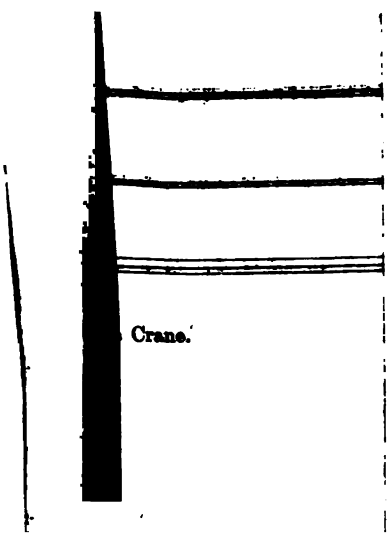

Crane.

Fig. 1.

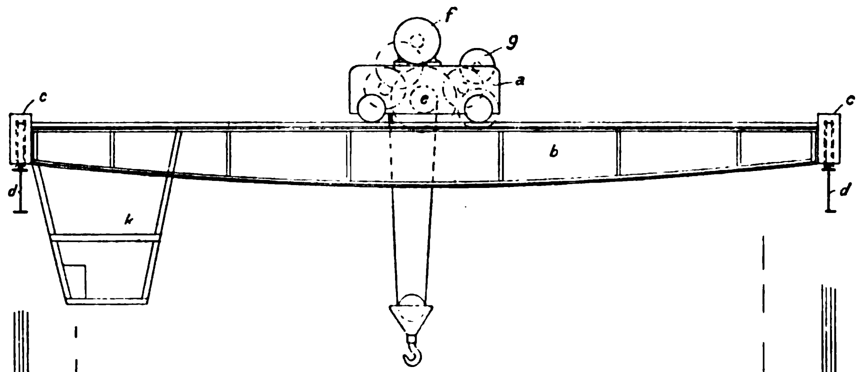
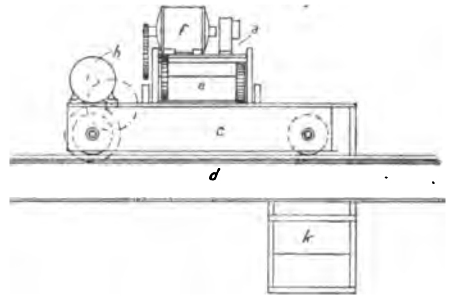


Fig. 2.



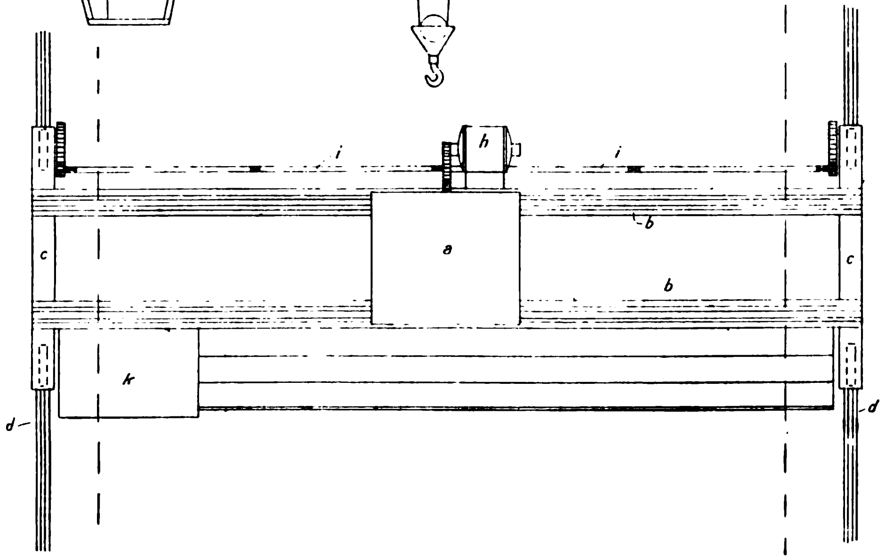


Fig. 3.

Overhead Crane.

Crane.

5-Ton Overhead Crane.—This is an example of the simplest type of electric overhead crane. It was constructed to the author's specification, for handling plates and bars in a large material yard, and has been at work about five years.

A general drawing of the crane is given in Fig. 4. The span of the crane is 60 feet. The speeds of the motions are, hoisting 12 feet per minute, cross traversing 125 feet per minute, travelling 250 feet per minute. Series-wound motors are used for each of the motions. When tested the power taken was—for hoisting, light 2·8 H.P., with 5 tons on hook 7·8 H.P.; cross traversing, light 2·8 H.P., with 5 tons 3·6 H.P.; travelling, light 5·6 H.P., with 5 tons 8·4 H.P. These powers are the electrical horse-powers taken at the crane terminals, and so include all losses. A diagram of the electric connections is given in Fig. 5, details of the collector gear in Fig. 6, and a sectional drawing of the electro-magnetic brake in Fig. 7. This form of brake was patented by the author some years ago, the patent having now lapsed. As shown in Fig. 7, the brake consists of two discs sliding on feather keys on the motor shaft. These discs are placed between plates, one of which forms part of the bracket which carries the shaft bearing, and the others swing on links, and are provided with stops to prevent rotation. The outer plate is acted on by a spring which presses it, the discs and the plates tightly together so that the motor shaft cannot turn. An electro-magnet is provided which, when energised, pulls back the outer plate and so releases the discs. The coil of the electro-magnet is connected in series with the hoisting motor so that when current is put on by the controller the brake is released, and when current is cut off the brake renews its grip. Thus the brake is always either entirely on or quite off, and no form of control is provided to regulate the speed of lowering, as, owing to the short distance through which the loads are lowered, there is no fear of the load attaining a dangerous speed when running down.

20-Ton Electric Overhead Crane.—This crane was constructed by Messrs. T. Broadbent & Sons, Ltd., of Huddersfield, and its general arrangement is shown in Fig. 8. The span of the crane is 57 feet 6 inches. The crab is fitted with two separate winding gears, each driven by a series-wound motor of 15 B.H.P. The heavy gear is capable of lifting 20 tons at a speed of 15 feet per minute, and the light gear 5 tons at 30 feet per minute. The crab traverses at a speed of 80 feet per minute, the traversing motion

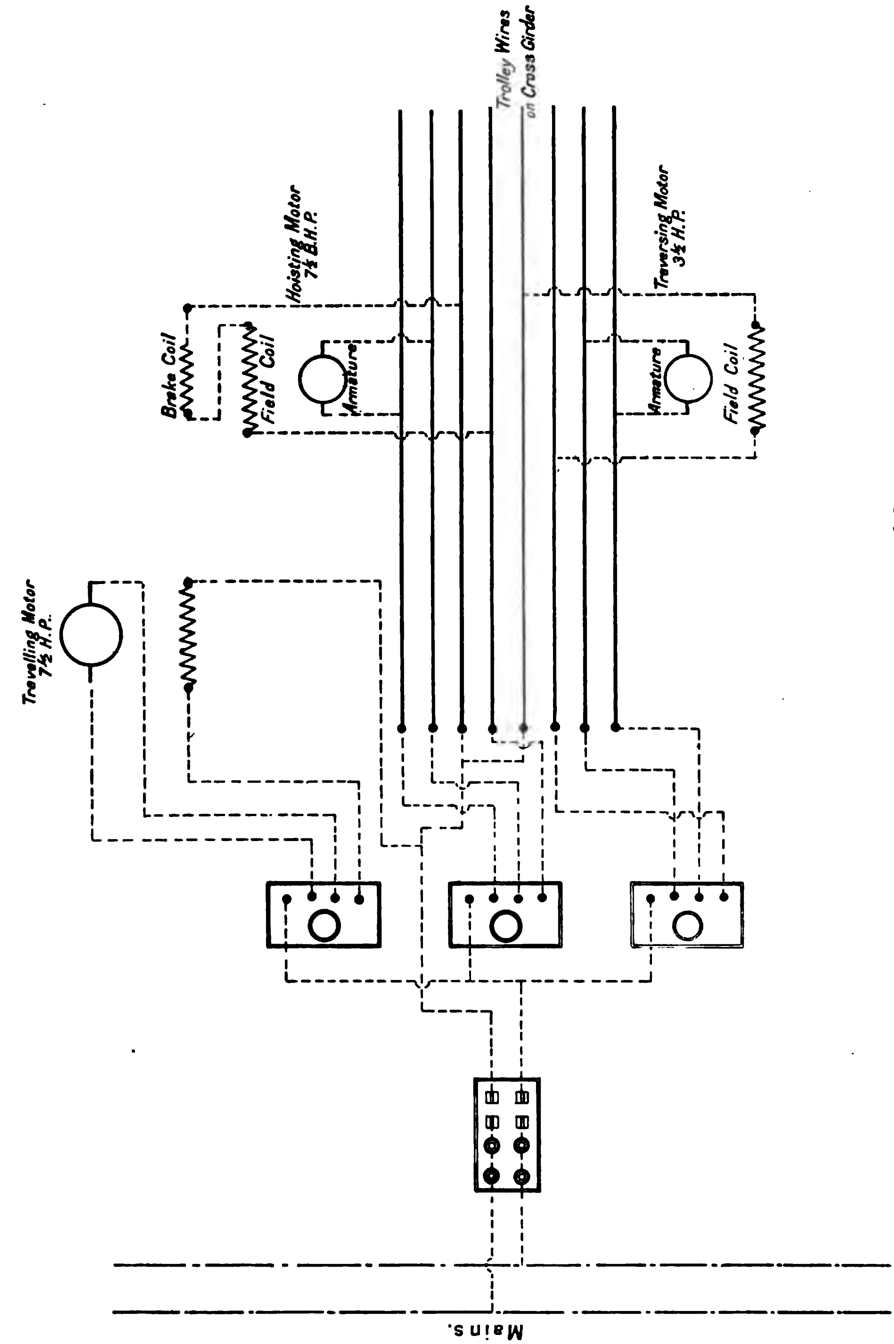


Fig. 5.—Connections of 5-ton Overhead Crane.

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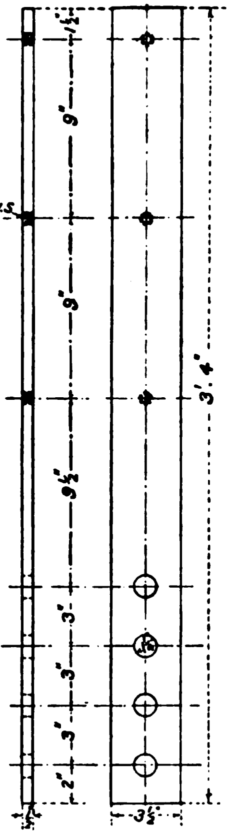

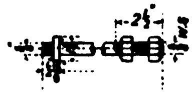

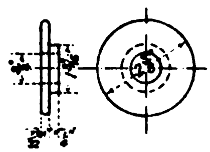

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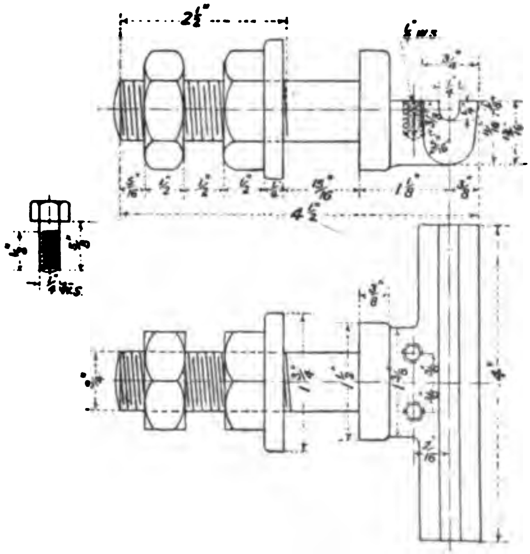


ad Crane.

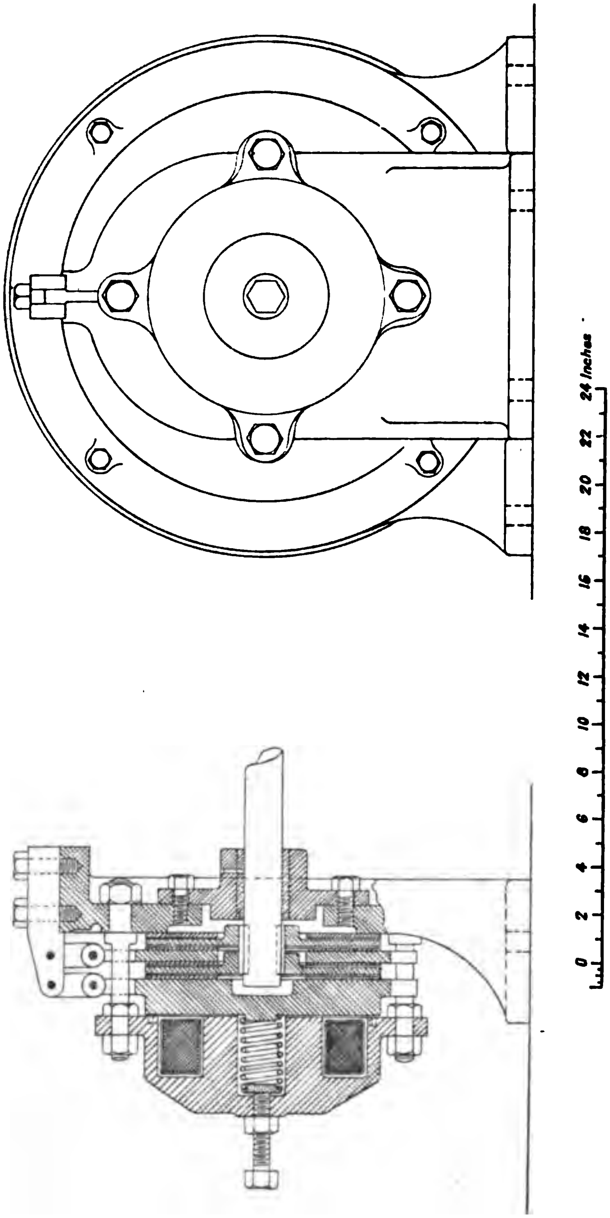
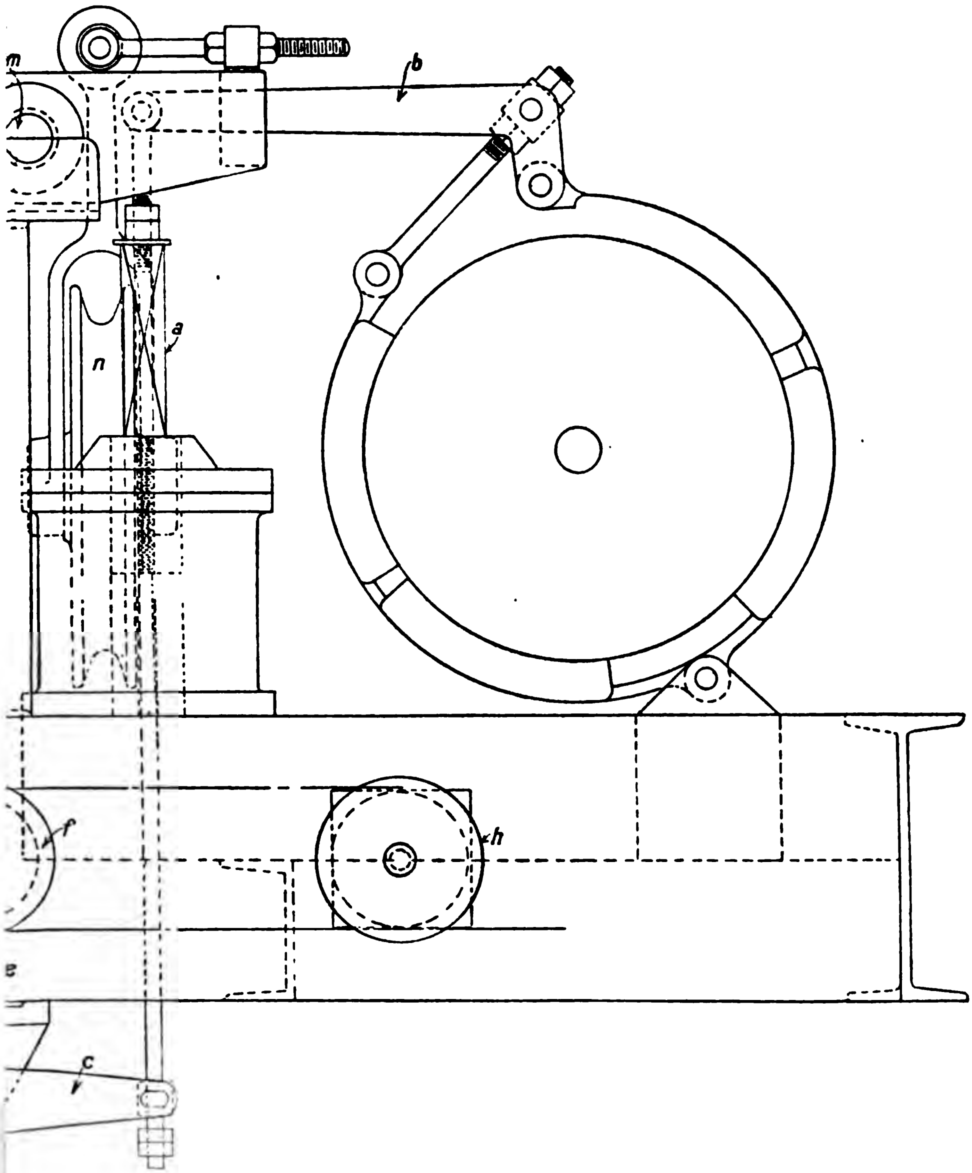


Fig. 7.—Electro-magnetic Disc Brake.

being driven by a motor of 5 B.H.P. The travelling motion is provided with a motor of 15 B.H.P., which drives the crane at

200 feet per minute. The arrangement of brakes and method of lowering adopted by Messrs. Broadbent are illustrated in Fig. 9. On the light hoisting gear a band brake is provided on the motor shaft. This brake is normally held on by the spring *a* pushing up the bent lever *b*. The brake is released by the solenoid shown, the plunger of which pulls down the lever *b*. The coil of this solenoid is in circuit with the light hoist motor, so that when current is switched on to the motor the brake is released, and when the current is switched off the brake is applied. In order to control the speed of the load when lowering, this brake is provided with an arrangement by which it can be released by hand instead of by the solenoid. Thus the load can be allowed to run down by its own weight with no current on the motor, the speed of lowering being kept in hand by regulating the pressure on the brake. In this arrangement the rod which connects the lever *b* to the solenoid plunger is extended downwards to the lever *c* mounted on the rocking shaft *d*. On this rocking shaft there is a second lever *e* carrying the grooved pulley *f*. On the end carriage of the crane a flexible wire rope is anchored at the point *g* (see Fig. 8). This rope is led from the anchorage round the pulley *f* to the idle pulley *h*, and thence round the guide pulley *i* (Fig. 8) to the lever *k* in the driver's cage. When the crab runs backwards and forwards on the cross girders the rope runs round the pulleys *f* and *h* without affecting their relative positions, but when the rope is pulled by the driver, moving lever *k* away from him, the pulley *f* with the lever to which it is attached is pulled towards *h*, so rocking the shaft *d*, and depressing levers *b* and *c*, thus releasing the brake. On the heavy lifting gear an exactly similar brake with hand release operated by a second rope and lever *k* is carried on the motor shaft, and, in addition, a mechanical brake is provided on the second motion shaft. This is applied by the lever *l*, which rocks on the pivot *m*. The equalising pulley *n* of the hoisting rope of the heavy lift gear hangs from the lever *l*, so that when there is a load on the crane hook the end of the lever to the right hand of *m* is pulled down, the opposite end being forced up, and so applying the brake through the link *o* and triangular piece *p*. Thus the force with which the brake is applied is proportional to the load on the hook. The brake drum is provided with a free-wheel arrangement, which permits it to revolve freely in the direction for lifting, but prevents it rotating in the reverse direction unless the brake is released (see chapter on *Brakes*).



m
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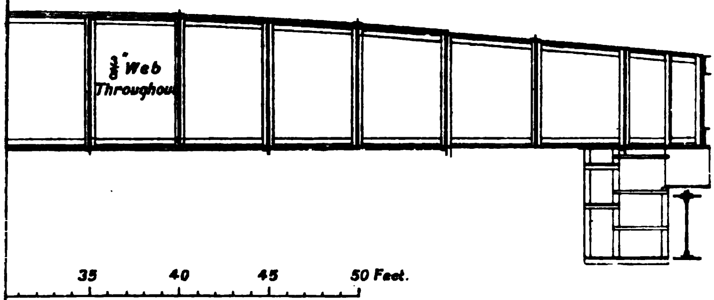

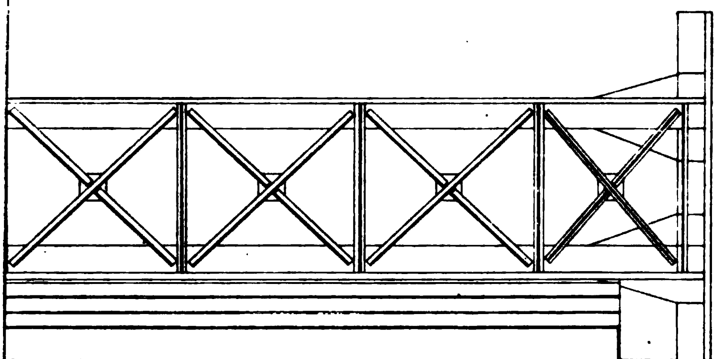

bracing.

Crane.

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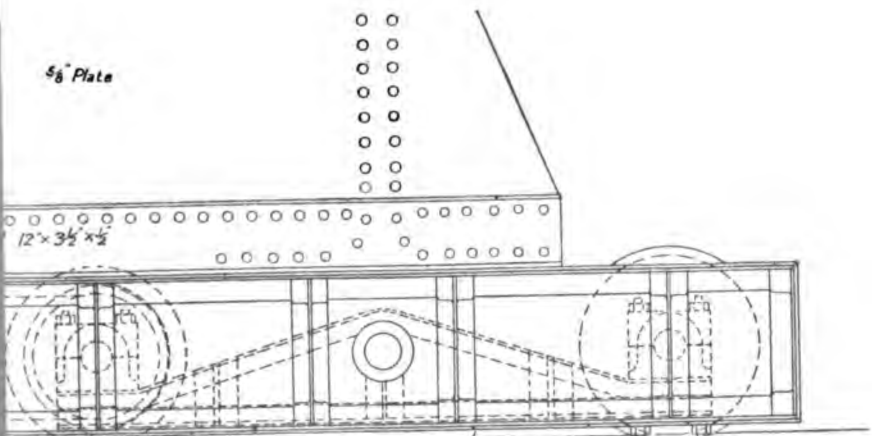


iew of 30-Ton Overhead Crane.

In order to release this brake by hand simultaneously with the solenoid brake on the motor shaft, the rocking shaft d is prolonged and carries an additional rocking lever and link clearly shown in the drawing, which, when the rope is pulled, pull down the lever l at the same time that the lever b is pulled down, so that both brakes are released by the driver moving the lever k. In Fig. 9 the parts of the solenoid brake magnet for the heavy lift are exactly the same as for the light lift, and so are lettered the same, but with a suffix to distinguish them, thus a_1, b_1, c_1, etc.

The two separate ropes and levers to operate the brakes on the heavy and light gears respectively are shown on the plan, Fig. 8.

30-Ton Electric Overhead Crane.*—This crane was constructed to the author's designs by the Cleveland Bridge Company, Ltd., for use in their yard at Darlington, and has been in regular use since its completion in June, 1903. A general view of the crane is given in Fig. 10. The span of the gantry rails is 87 feet, and the depth of the cross girders at the centre is 7 feet 6 inches. With the full load of 30 tons on the hook and the crab at the centre of the span, the stress in the flanges is about 5·5 tons per square inch. This stress is somewhat low, but was adopted to allow of the crane taking considerable overloads occasionally. In order to economise head room, the crab runs on the bottom flanges of the girders, and to give access to the crab the two centre panels of the girders are of lattice work, the remainder having a plate web.

The general details of construction of the crane are shown in Fig. 11. The crab frame was constructed of steel plates and angles with jaws at the ends to take axle boxes. The width of the crab is less than the distance between the bottom flanges of the girders, so that when the axle boxes are disconnected and the wheels are run out of the way the crab can be lowered to the ground, rolled joists of H section being provided at the centre of the girders to take blocks and tackle for the purpose. The gearing throughout the crane is of steel, and all teeth are cut with the exception of those on the wheel of the hoisting drum and on the pinion which drives it.

On the hoisting gear the load is taken on an eight-part rope winding two parts on opposite sides of the barrel, so that the load travels in a vertical line. Thus the purchase of the rope is 4 to 1, and the direct pull off the barrel with 30 tons on the hook is $7\frac{1}{2}$ tons. The rope is $\frac{7}{8}$ inch diameter, and has six strands with 37 wires per

* See "Electric Cranes." Hill, *Proc. Inst. C.E.*, vol. clx., p. 368.

strand, the breaking load being 24·75 tons. The pull on the rope due to the load on the hook is 3·75 tons, so giving a factor of safety of 6·6. The diameter of the hoisting barrel and sheaves is 24 inches, so being 27·5 times the diameter of the rope.

The shaft of the hoisting motor is provided with an electromagnetic brake similar in construction to Fig. 7, with two sliding discs on the motor shaft. As in the case of the 5-ton crane already described, this brake is always either on or off, its function being simply to hold the load when the motor is stopped. No hand release or other mechanical arrangement is used to control the speed of lowering, this being effected electrically by a method devised by the author, and differing from usual practice in that a shunt-wound motor is employed. For the driving of hoisting gears a shunt motor has two advantages over a series-wound motor. (1) Its speed can be regulated over a fairly wide range by inserting resistance in its magnet circuit, the variation so obtained being greater than that obtained with a series motor. (2) A shunt motor, if overhauled by the load commences to generate current in opposition to that supplied to it, thus automatically controlling the speed of lowering. Even with a shunt motor, however, there is a possibility of the speed becoming excessive in lowering unless the controller is specially designed to prevent it. When starting the motor, whether hoisting or lowering, resistance must be inserted in series with the armature, and gradually cut out as the speed rises, until the full voltage is obtained across the brushes. For lowering an additional arrangement is required for the following reason :—When the controller is full on, on the lowering side, and the load is driving the motor as a dynamo, the voltage generated will be in excess of the line voltage, and a current corresponding to the torque exerted on the armature shaft through the gearing will be forced into the circuit. The generated voltage will depend on the resistance of the motor armature, and as this is comparatively low, the excess of speed of the motor will be small, say, about 10 per cent. of the normal speed. When the controller handle is moved back, the conditions are altered as an appreciable resistance is now introduced, and as the motor must still generate the same current as before, it must run faster, and as the controller handle is moved further back the speed will continue to increase. To prevent this, the controller must be arranged to insert resistances in parallel with the armature on the lowering side, in addition to those in series with it. Thus, when the controller is on an intermediate step, if the motor is driving the load, the series resistances prevent excessive current passing through the armature, while, if the load is driving the motor, the parallel resistances prevent it running too fast. This arrangement has been applied to a number of cranes, and has been found to work very well.

The results of a series of tests made with this crane are given in Table I. From these it will be seen that when lifting a 30-ton load at a speed of 4½ feet per minute, the ratio of the useful H.P. at the hook to the electrical H.P. delivered at the crane terminals was 68·6 per cent., which is somewhat higher than the efficiency usually abtained, and is probably partly due to the sheaves and shafts of the hoisting train being fitted with roller bearings. The variation of speed obtainable by varying the strength of the magnetic field is from 4½ feet per minute with full load to 19 feet per minute with hook only. The maximum speed for any load is set by the shunt regulator, and when the controller is full on the motor runs at this speed, and if overhauled by the load when lowering it returns current to the circuit, and is prevented from going any faster.

Thus, with 30 tons on the hook and the shunt regulator set for 5 feet per minute the speed of lowering with the controller on the first step is 9 inches per minute, on the second step 3½ feet per minute, and when full on the speed is 5 feet per minute, and at this speed the motor returns 20 amperes to the circuit. The power available at the hook in this case is 10·27 H.P., and the power returned to the circuit is 5·82 H.P., giving a reverse efficiency—*i.e.*, the efficiency of the crab as a generating plant of 56·6 per cent. Similar regulation can be obtained for other loads. When lowering 5 tons at 10½ feet per minute, the crane neither absorbs nor gives out current, the needle of the recording ammeter moving steadily along the zero line, as shown in Fig. 13. The traversing motor is series wound, and has an ordinary drum controller. The shafts of the traversing train are supported in self-oiling bearings, and the crab axles have roller bearings. The weight of the crab complete is 11 tons. The traversing speed with a load of 30 tons is 182 to 187½ feet per minute, and the power taken at the main terminals is equivalent to a tractive effort of 16·8 to 18·5 lbs. per ton. When starting the rate of acceleration is limited by the tendency of the load to swing, and it is found that, for traversing, an acceleration of 0·6 to 0·7 foot per second per second cannot conveniently be exceeded. To provide sufficient retardation when stopping, the traversing motor is provided with an electric brake similar to that

TABLE I.—Tests of 30-Ton Crane. Taken January 31, 1904.

(See Diagrams, Fig. 13.)

Voltage at Crane Terminals 217 for all Tests.

No.	Work.	Load on Hook.	Total Load.	Speed, Feet per Minute.	Amperes.	H.P. at Hook.	E.H.P.	Efficiency.	Tractive Effort.	Acceleration Period in Seconds.	Rate of Acceleration, f.p.s. per sec.	Mean Current during Acceleration.	B.T. Units Consumed during Acceleration.	B.T. Units per 20 Feet when running Steadily.	Position of Controller.	Remarks.
		Tons.	Tons.						Lbs. per Ton							
1	Hoisting, .	30·1	30·1	4·4	45	8·99	13·1	68·6	...	...	...	...	...	...	Full on	
2	,, .	30·1	30·1	4·5	46	9·19	13·38	68·7	...	...	...	...	...	...	,,	
3	,, .	15·1	15·1	10·17	52	10·42	15·13	68·9	...	...	...	...	...	...	,,	
4	,, .	15·1	15·1	10·1	54	10·35	15·7	65·9	...	...	...	...	...	...	,,	
5	,, .	5	5	10·25	25	3·48	7·27	47·9	...	...	...	...	...	...	,,	
6	,, .	Hook only	...	18·9	32·5	0	9·46	...	...	...	...	...	...	...	,,	
7	,, .	,,	..	18·46	35	0	10·18	...	...	. .	...	...	...	...	,,	
8	Lowering, .	30·1	30·1	0·73	...	...	...	...	...	...	...	...	...	...	1st step	27 amps. in short-circuited armature
9	,, .	30·1	30·1	3·4	...	...	...	...	...	...	...	...	...	...	2nd ,,	25 amps. in short-circuited armature
10	,, .	30·1	30·1	5·03	-20	-10·27	-5·82	-56·6	...	...	...	...	...	...	Full on	
11	,, .	30·1	30·1	5·05	-20	-10·3	-5·82	-56·5	...	...	...	...	...	...	,,	
12	,, .	15·1	15·1	10·9	-21	-11·17	-6·11	-54·7	...	...	...	...	...	...	,,	
13	,, .	15·1	15·1	10·9	-21	-11·17	-6·11	-54·7	...	...	...	...	...	...	,,	
14	,, .	5	5	10·5	0	3·56	0	...	...	...	...	...	...	...	,,	
15	,, :	5	5	10·5	0	3·56	0	...	...	...	...	...	...	...	,,	
16	,, .	Hook only	...	17·4	27·5	0	8	...	...	...	...	...	...	...	,,	
17	,, .	,,	...	17·5	30·0	0	8·73	...	...	...	...	...	...	...	,,	
18	Traversing, .	30·1	41·1	187·5	13·5	...	3·93	...	16·83	5	·625	35·2	0·0106	0·0052	,,	
19	,, .	30·1	41·1	182	14·5	...	4·2	...	18·5	5	·605	...	...	...	,,	
20	,, .	15·1	26·1	182	12	...	3·49	...	24·25	5	·605	26·2	0·0079	0·0048	,,	
21	,, .	15·1	26·1	176·5	14	...	4·07	...	29·15	4	·735	...	...	...	,,	
22	,, .	5	16	182	12	...	3·49	...	39·5	5	·605	25·1	0·0075	0·0048	,,	
23	,, .	5	16	182	12·5	...	3·63	...	41·1	5	·605	...	...	...	,,	
24	,, .	0	11	207	12·5	...	3·63	...	52·6	5	·605	25·6	0·00772	0·0044	,,	
25	,, .	0	11	193·5	12·5	...	3·63	...	56·2	4·5	·715	...	...	...	,,	
26	Travelling, .	30·1	93·1	322·5	50	...	14·5	...	15·94	15	·36	102	0·092	0·0112	,,	
27	,, .	30·1	93·1	319	52·5	...	15·28	...	16·97	17	·31	...	...	...	,,	
28	,, .	15·1	78·1	322·5	50	...	14·5	...	19	16	·335	96	0 092	0·0112	,,	
29	,, .	5	68	322·5	45	...	13·1	...	19·7	15	·356	97	0·088	0·010	,,	
30	,, .	0	63	341	45	...	13·1	...	20·1	15	·378	89·5	0·081	0·0095	,,	

on the hoisting motor. The travelling motor is a series-wound machine, and, like the traversing motor, is provided with an electric brake to increase the retardation when stopping. The motor is placed at the centre of the girders, and drives the cross shaft through single-reduction gearing. The deflection of the girders with a 30-ton load at the centre is $\frac{7}{16}$ inch. If the cross shaft were mounted in ordinary bearings this amount of spring would cause it to bind, and would add to the power required to travel the crane. To avoid this, the shaft is carried in swivelling roller bearings, and the separate lengths of which it is made are joined together by flexible couplings, so that it turns freely under all conditions. Each of the end carriages has three wheels, an equalising lever being provided to give equal loads to each wheel. At first sight it is not quite clear how a single equalising lever can distribute the load upon three wheels. Reference to Fig. 12, which gives the distribution of the loads, shows that there are in reality two levers, the end carriage itself

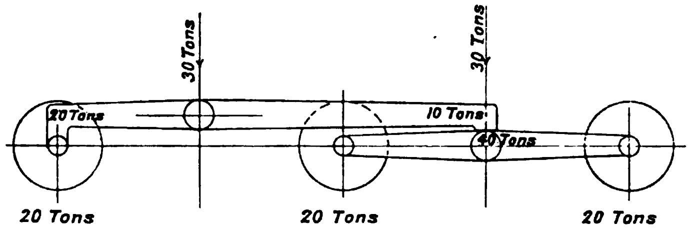


Fig. 12.—Distribution of Load on Travelling Wheels.

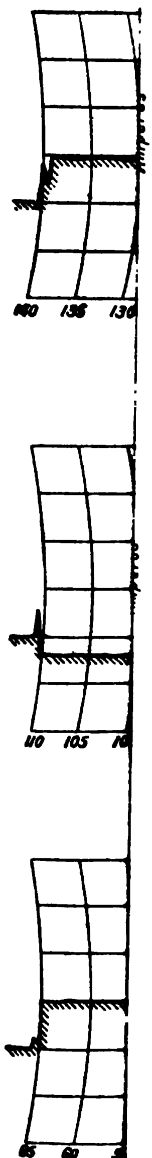
forming one of them. The loads shown are those obtained when the crab is traversed as far to one side as it will go, and with 30 tons on the hook. With the crab at the centre and the same load the weight on each wheel is 15·5 tons. In order to prevent slip, two wheels in each end carriage are driven, the first by gear from the cross shaft and the second by means of a chain from the first.

The total weight of the crane, including the crab, is 63 tons. With a load of 30 tons the speed of travelling is 322½ feet per minute. and the power taken at the crane terminals represents a tractive effort of 16 to 17 lbs. per ton. It has been found that, when starting. an acceleration of 0·3 to 0·4 foot per second per second cannot conveniently be exceeded. Recorder diagrams taken with this crane are shown in Fig. 13.

100-Ton Electric Overhead Crane.—Constructed by Messrs. J. Adamson & Co., Ltd., of Hyde, Cheshire. A general view of this crane is given in Fig. 14, and a cross-section to an enlarged scale in Fig. 15. The span is 70 feet 3 inches. The girders are of the box pattern, with internal bulkheads between the webs, and are carried on end carriages built up of steel plate. The bottom plates of these carriages are extended to form gussets, as shown in Fig. 14, thus giving extra stiffness to the crane. The cross girders are secured to these gussets, and the end carriages by turned bolts fitted into rymered holes.

The crane runs on four wheels in each end carriage; these are mounted in pairs in cast steel equalising levers, fitted with self-lubricating axle boxes lined with gun-metal bushes (see Fig. 16). The travelling wheels are 2 feet 6 inches diameter, and have double flanged rolled-steel tires, shrunk on cast-iron hubs. The total weight of the crane being 104 tons, the load on each wheel with the crab at the centre with a load of 100 tons is 25·4 tons. With the crab, the weight of which is 35 tons, moved as far as it will go to one side, the load on each of the wheels at that side is about 40 tons. The crab has double-plated steel sides rivetted together, with bosses between for the various shafts and axle bearings. It runs on four cast-steel double-flanged wheels 1 foot 8 inches diameter with turned treads. The longitudinal travelling motion is driven by a series-wound motor of 32 B.H.P. running at 400 revolutions per minute, the speed of travelling being 90 feet per minute. An electro-magnetic brake is fitted to the motion to bring the crane quickly to rest after the current is cut off. The main hoisting motion is driven by a 47 B.H.P. motor running at 375 revolutions per minute, the speed of lift for 100 tons being 4½ feet per minute, with an efficiency of 47 per cent. An auxiliary hoisting barrel is also fitted, to deal with loads up to 15 tons, which it hoists at a speed of 30 feet per minute, being driven by a second 47 B.H.P. motor running at 375 revolutions per minute. The efficiency of the auxiliary lift with 15 tons is 60 per cent. The gears for both these hoisting motors are of mild steel throughout, with machine-cut teeth, and they are fitted with electro-magnetic brakes, which sustain the load when the current is cut off. Mechanical brakes of the enclosed Weston type are also fitted, to prevent undue acceleration when lowering.

The electro-magnetic brake prevents the load moving unless the current is on, whilst the mechanical brake maintains control over the load whilst it is being lowered. The electro-magnetic brake consists essentially of a couple of hinged levers curved to fit



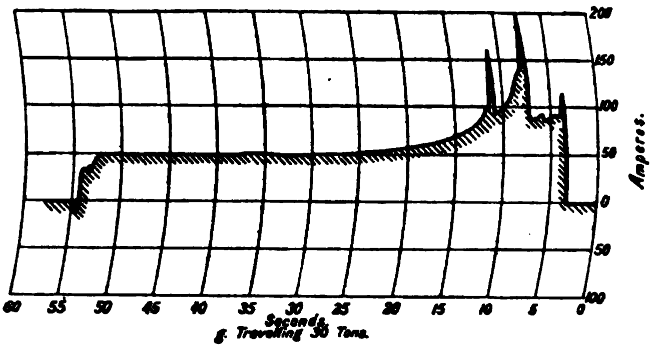
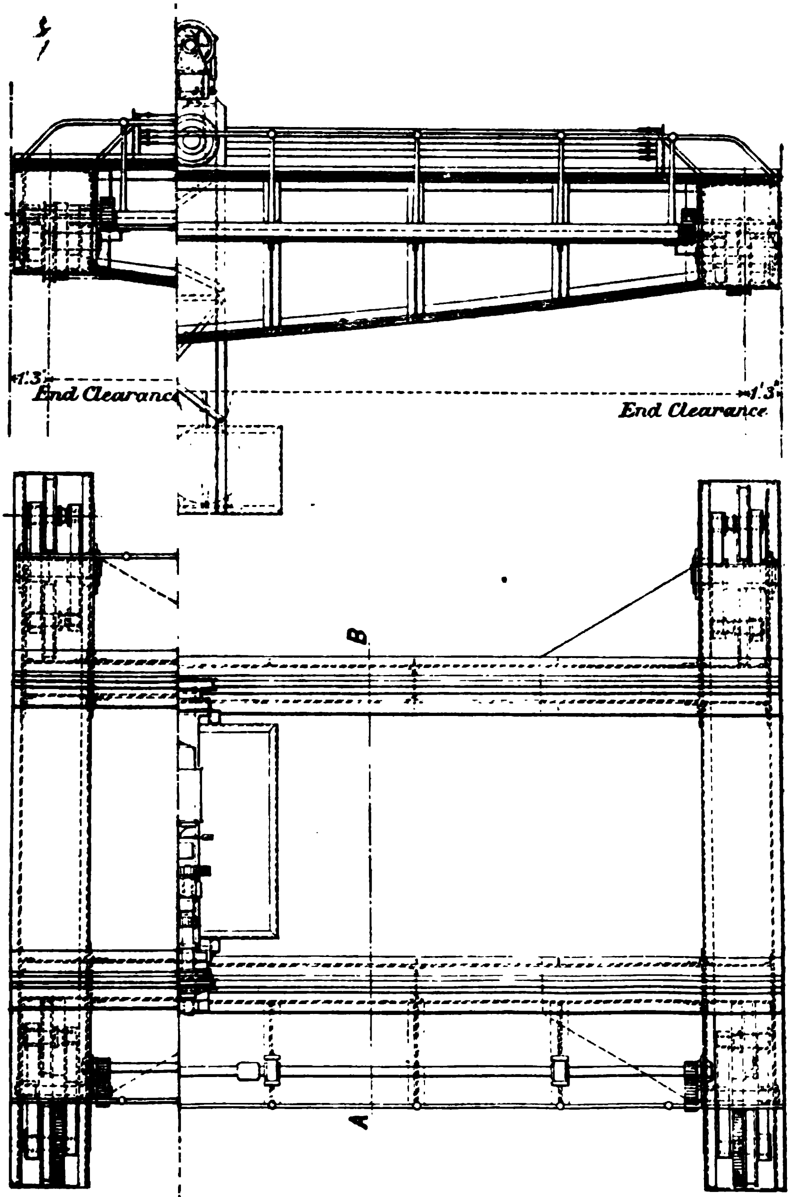


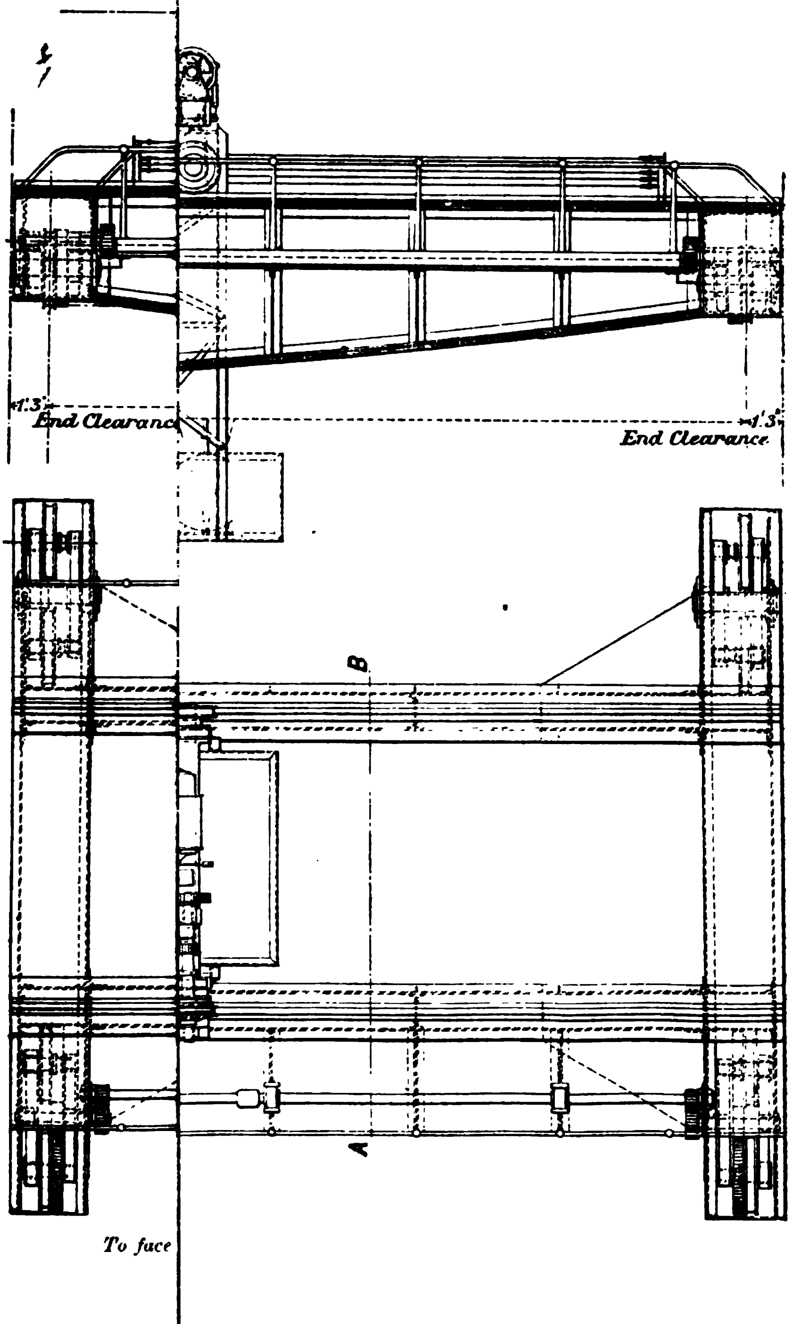
Fig. 13.—Ampere Meter Diagrams taken on 30-Ton Overhead Crane.

Hill's " Ele



To face

Hill's "Ele



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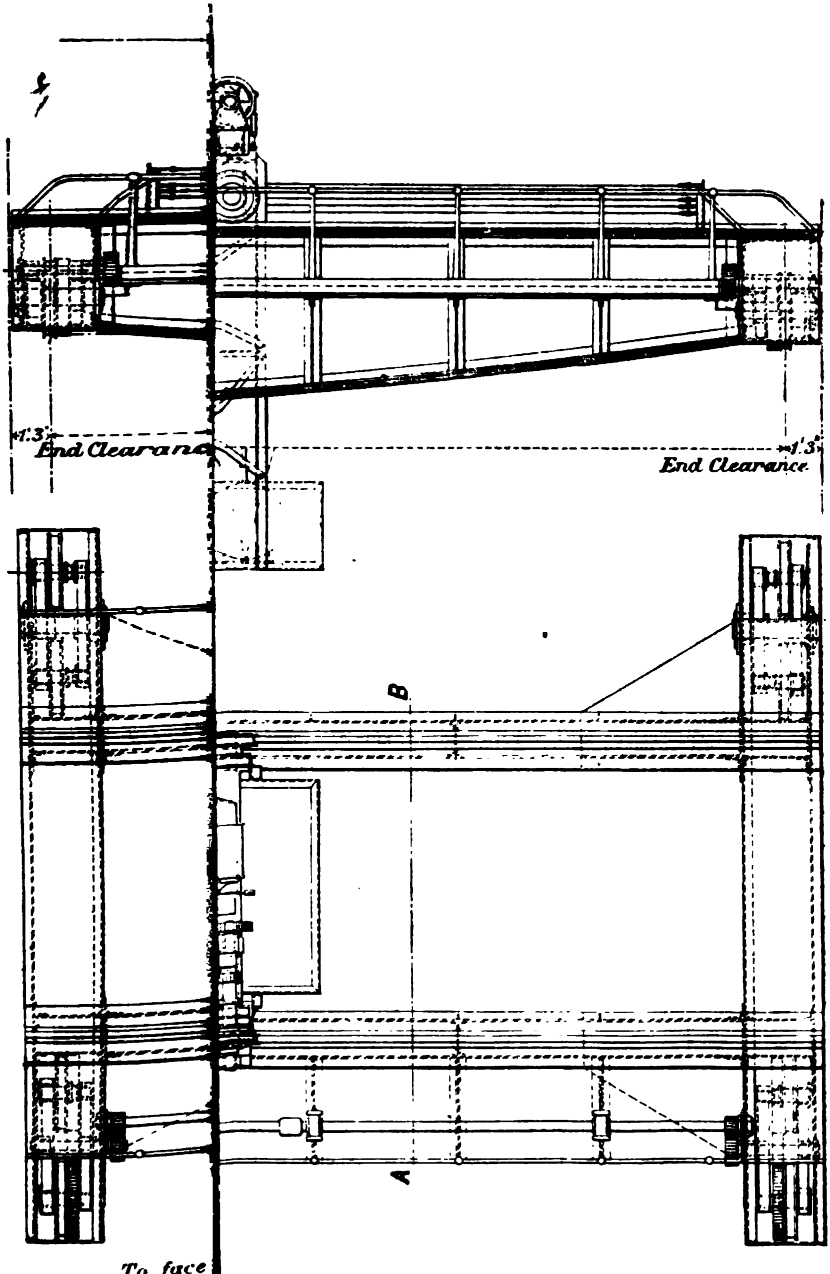

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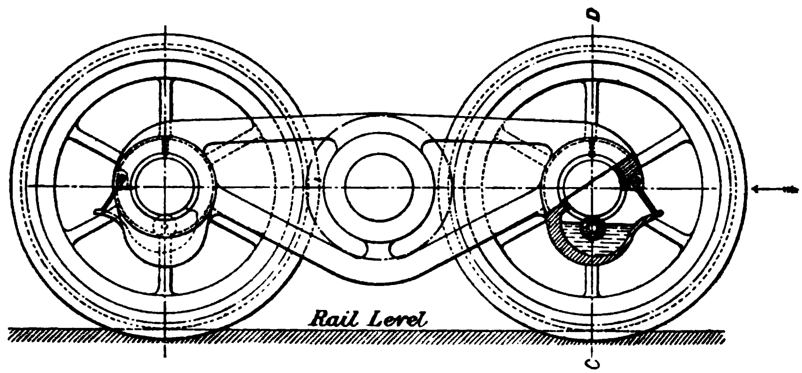
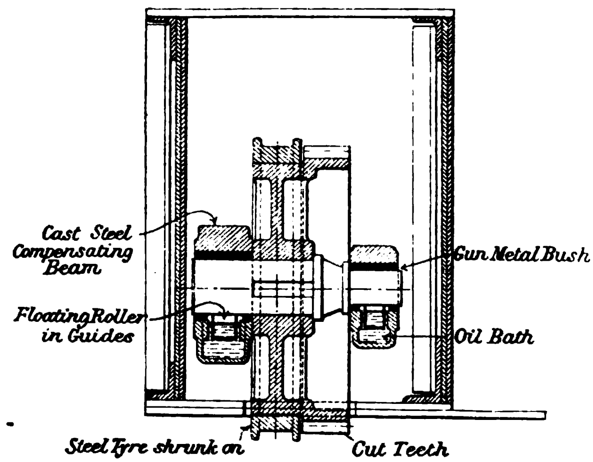
the circumference of the brake pulley, and lined with wood. These levers are suspended from the motor casing by their hinge, and are normally pressed into contact with the brake pulley by a system



Travelling Wheels.



SECTION C.D. LOOKING IN DIRECTION OF ARROW

Fig. 16.*—100-Ton Overhead Crane.

of clamping levers actuated by a weight. Under these conditions the brake pulley, which is grooved to increase the friction, is rigidly held so that the load cannot descend. The levers are adjustable, to allow for the gradual wear of the brake blocks. The clamping weight can, however, be lifted by means of a solenoid, and this solenoid, being in series with the lifting motor, the brake is held off when current passes through the latter, either in the direction for lifting or that for lowering. A cushion is provided within the solenoid casing to prevent "hammering" as the current comes on or off. It will be seen that this brake comes into action whenever the current through the motor ceases, even if this interruption is due to the blowing of a fuse or other accidental cause. To lower a load, current must be passed through the motor, as otherwise the electric brake is locked. This current, however, releases the brake, which being always, either fully on or completely off, provides no means of controlling the descent of a load. This control is accordingly supplied by the mechanical brake, which is a Weston type friction clutch, enclosed in an oil-tight casing, and flooded with a lubricant. The casing carries the loose discs of the clutch, other discs lying between these rotating with the brake shaft. Ratchet teeth are cut on the exterior of the casing, and a pawl engages with these. Friction rings sliding on the ratchet-casing bring a positive pressure on this pawl whenever the load begins to run down, thus forcing it into engagement. The end pressure, by which the friction plates are forced into contact so as to lock the clutch, is secured by means of a quick thread screw and nut, connected with one of the pinions of the hoisting motion. This end pressure is, therefore, proportional to the weight lifted, and the load is thus securely and automatically held in all positions. To lower the load, it is necessary to reverse the motor, which tends to unscrew the nut above referred to, relieving the pressure between the friction plates, and allowing the load to descend. The load, therefore, cannot descend unless the motor is revolved by power in the lowering direction. The discs being numerous, the pressure between them is slight per unit of area, and the surfaces being lubricated, the wear is small. Half the discs are of steel and the others of gun-metal. Illustrations of brakes of this type are given in the chapter on Brakes.

The hoisting barrels are of cast iron, the grooves for the rope being turned in the lathe, running right- and left-handedly from each end so as to give a truly vertical lift. The diameter of the barrel is 3 feet 6 inches, and is large enough to give a lift of 50 feet without overlapping. The main load is carried by eight plies of specially flexible steel wire rope of 5 inches circumference, and the auxiliary load by four plies of steel wire rope $2\frac{3}{4}$ inches circumference. The diameter of the auxiliary hoist barrel is 1 foot 6 inches. The compensating pulleys for these ropes are suspended from a substantial cross beam fixed to the crab sides. The diameter of the sheaves round which the main ropes are led is 2 feet 8 inches, and the sheaves on the auxiliary hoist are 1 foot 7 inches. The main rope has six strands with 37 wires per strand, and a breaking strength of 105 tons, while the auxiliary rope has six strands with 37 wires per strand, and a breaking strength of $31\frac{1}{2}$ tons. The hooks are made from selected forged scrap, and are suspended from the bottom block by steel plate side links and crossheads with turned ends. Ball bearings running in turned steel washers are fitted under the nuts on the shanks of the hooks, to facilitate the turning of the load without giving any twist to the ropes.

The cross traversing motion obtains its drive from a 16 B.H.P. motor running at 333 revolutions per minute, the speed of traverse being 50 feet per minute. The motions are all controlled from independent crane controllers placed in a cage suspended from the crab.

30-Ton Electric Goliath Crane.—The Goliath type of crane performs the same duty as the overhead type, and the author has, therefore, included it in this chapter.

The difference is merely one of constructional detail. Instead of, as in the overhead crane, running along a gantry, the end carriages run on rails laid on the ground and the cross girders are carried at the required height by means of columns mounted on the end carriages.

In cases where there is no objection to the crane running along the ground, the Goliath type is preferable as its adoption eliminates the cost of the gantry.

Fig. 17 gives a general view of a 30-ton electric Goliath constructed by Messrs. Stothert & Pitt, Ltd., of Bath, and a drawing of the crab to a larger scale is given in Fig. 18. The span from centre to centre of the rails is 60 feet. The end carriages are constructed of plates and angles, the columns are made of rolled joists of H section with bracings of channel section, and the cross girders have plate webs. The junction of the cross girders with the columns is braced with plate and angle corner brackets and horizontal gussets.

The frame of the crab is also constructed of plates and angles, and has two hoisting barrels, main and auxiliary. One lifting motor, series-wound, of 35 B.H.P. is provided, and by means of a clutch and change gear this can be connected to drive either of the hoisting barrels. On the main barrel the full load of 30 tons is hoisted at a speed of 10 feet per minute, while on the auxiliary lift barrel a load of 10 tons can be hoisted at 30 feet per minute. The main load is taken on a four-part rope of 4 inches circumference winding two parts on the barrel. The diameter of the barrel is 1 foot 8 inches, and of the sheaves 1 foot 10 inches. The auxiliary lift is taken on a two-part $\frac{7}{8}$-inch chain winding two parts on the barrel. On the second motion shaft of the hoisting gear a band brake is fitted. This is applied by a weight and lever, and is released by a solenoid connected in the circuit of the lifting motor. For lowering, this brake is provided with a hand release worked with a steel wire rope on the same general principle as on the overhead crane, Fig. 9, already described. The arrangement is shown clearly in Fig. 18.

The crab traversing motion is driven by a series-wound motor of 8 B.H.P. at a speed of 30 feet per minute. The crane travels at a speed of 80 feet per minute, the travelling motion being driven by a series-wound motor of 25 B.H.P. Each end carriage has four wheels, two wheels in each carriage being driven by means of pitch chains. In order to equalise the load on the eight wheels, the axles are provided with springs. The total weight of the crane is 61 tons, and the weight of the crab is 10 tons.

With 30 tons at the centre of the span the load per wheel is 11·4 tons, and with the same load, but with the crab traversed as far as it will go to one side the load is about 16 tons per wheel.

Overhead Crane with Revolving Jib.—Where the floor space of a building is obstructed by columns, the form of crane shown in Fig. 19 may be employed. This consists of an overhead crane, from the crab of which a revolving jib is suspended. The end of the jib can be travelled between the columns so as to reach all parts of the floor. The stability of the revolving arrangement and the stresses in it are determined on the same principles as for loco. jib cranes, while the cross girders are subjected to ordinary bending stresses. The crane illustrated was constructed by Messrs. Applebys, Ltd.

Golia

Ta

To face p. 16.

Fig 19.—Overhead Revolving Jib Crane.

CHAPTER II.

LOCOMOTIVE AND PORTABLE JIB CRANES.

THIS is a class of crane which, with various modifications of arrangement and details, is probably more widely used than any other form. It will lift, travel, and deposit loads anywhere within a space, the width of which is equal to twice the horizontal radius of the jib, the rails on which it runs being laid down the centre of this space. The height to which loads can be lifted depends on the length of the jib and the distance which they can be travelled is limited only by the length of track on which the crane runs.

One form of **Loco. Jib Crane** is shown in Fig. 20. The load hangs on the hook *a* attached to the rope *b*, which passes over a sheave at the head of the jib *c* to the hoisting drum *d*, which is driven through gearing by the motor *e*. A second rope, *f*, is fastened to the head of the jib, and passes over the guide pulley *g* to a second hoisting drum *h*, which is driven through gearing by the motor *i*. This second rope acts as a stay to hold the jib in position, and by winding in or paying out the rope by rotating the drum *h*, the angle of the jib may be varied, as shown in dotted lines. A clutch is provided by which the motor, *i*, can be geared either to the drum *h* for the purpose of raising or lowering the jib, or to a vertical shaft at the centre of the crane which leads down to bevel gear below the frame to drive the travelling wheels. Thus the motor *i* in this case is used to serve two purposes. A separate motor is, however, sometimes used to drive each motion in order to simplify the gearing. The upper portion of the crane is carried by rollers on a circular path, *k*, being held in position by a central pin on which it turns. Thus this part of the crane can be turned through a complete circle. To effect this the circular path *k* is provided with teeth on its internal periphery, and a pinion gearing into these teeth is driven through worm gear by the motor *l*.

At the point *m* cast-iron balance weights are fastened to the frame to counteract the load hanging on the jib, and so prevent the crane from overturning. In order to prevent the crane falling over forwards when loaded or backwards when unloaded, the centre of gravity of the whole mass, crane plus load, must under all conditions come within the points of support. That is to say, when the jib is lying fore and aft the centre of gravity must come within the centres of the wheel base, and when lying across it must come within the centres of the rails. The first condition can always be

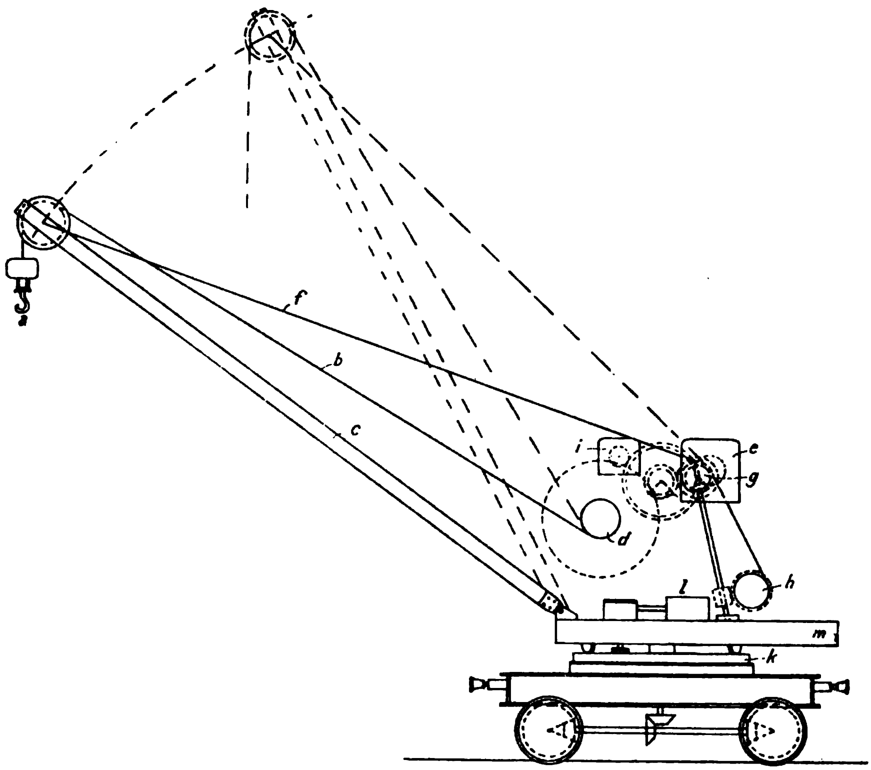


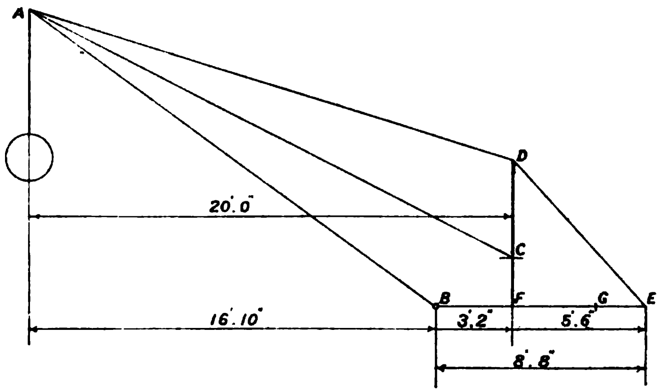
Fig. 20.—Loco. Jib Crane.

met by making the wheel base sufficiently long, but it is not always possible to fulfil the second condition, especially in the case of narrow gauge lines, and in such cases it is customary to increase the width of the crane by means of cross girders fastened to the framing, the ends of these girders being supported from the ground when the crane is working, or by fastening the crane to the track rails with clips. In the latter case it is necessary to see that the rails themselves are securely held down, or the clips may pull them up, and so allow the crane to fall over.

Current may be conveyed to the crane either by an overhead wire from which the current is collected by means of a trolley pole, or by a third rail laid on the ground. In cases where the crane does not need to travel when at work, connection to it may be made by a flexible cable from a junction box in the roadway.

Connection between the truck and upper portion of the crane is effected by circular sliding contacts, which permit of the crane slewing through a complete circle.

The principal stresses in the structure may be most conveniently determined by graphic methods. Referring to Fig. 21, the diagram shows the upper portion of a locomotive jib crane—that is to say,



Load hanging from A = 5·00 tons.
Half weight of jib = 0·75 „

5·75

Fig. 21.

the part which revolves. The stresses in this part may be determined in the following manner :—In diagram Fig. 22 set off to any convenient scale the vertical length da_1 equal to the load hanging on the hook, plus the weight of the hook and its attachments, plus the weight of rope hanging from the point A (Fig. 21), plus the weight of the sheave at A, and plus half the weight of the jib AB, the stay AD, and the rope AC. Generally speaking, the two latter are so light relatively to the jib that they may be neglected without introducing any appreciable error. From a_1 set off the line a_1b parallel to the lifting rope AC, and equal in length to the load on this rope. From b draw a line parallel to AB, and from d a line parallel to AD. These lines will meet in the point a, and the line ad will represent the tension in the stay AD, and ab the

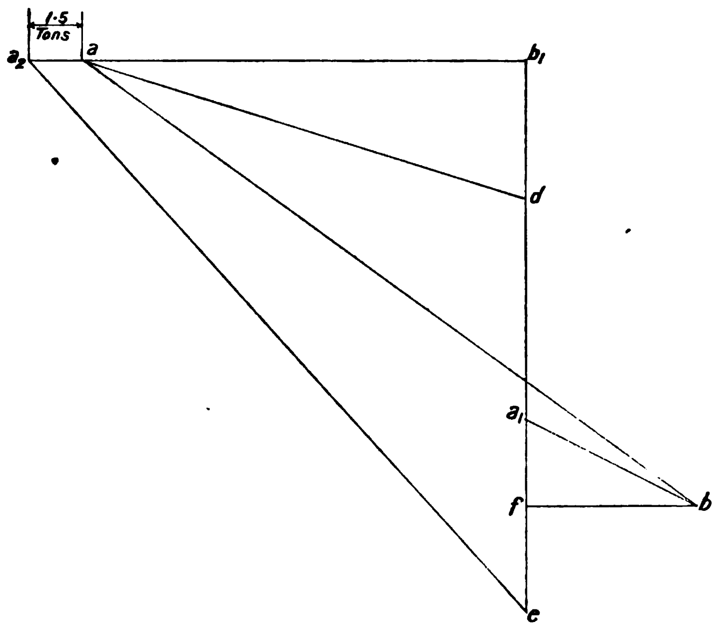


Fig. 22.

compression in the jib AB. The line da_1 requires to be produced at both ends. From point b draw the line bf at right angles to da_1 produced. The length fb then represents the horizontal pull of the lifting chain at the point C. The part DF is then equal to a girder supported at D and F, and loaded at C, the amount of load at D and F respectively depending on the position of C. In the case illustrated, the horizontal pull at C is 4·5 tons, and as C is at one-third of the length of DF from the bottom, there is a pull of 1·5 tons at D, and 3 tons at F. From point a draw ab_1 at right angles to da_1 produced. Continue this line to a_2, draw aa_2 equal to the horizontal pull at D. From a_2 draw a_2e parallel to DE, a_2e will then represent the tension in the back stay DE. The compression on DC is equal to de, while that on CF is equal to

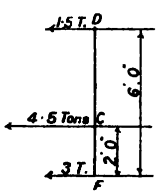


Fig. 22A.

$de - a_1f$. The upward pull at E, due to the tension in the stay DE, is equal to b_1e; but, in order to ascertain the amount of balance weight necessary at E to hold the crane in equilibrium when supported at the point B, we must take into account the downward force at F, thus the amount of balance weight required is

$$b_1e - \frac{(de - a_1f) \times BF}{BE}.$$

The downward force at the point B equals

$$b_1f + \frac{(de - a_1f) \times FE}{BE},$$

and this is also equal to the weight of the balance weight plus the load. Of course, to obtain the total weight at B we must add the weight of mechanism, and parts which do not enter into the stress diagram. As the greater portion of the weight of these parts lies to the right of B, it assists the balance weight at E, and so somewhat lessens the quantity of weight required at this point. The compression on EF is represented by a_2b_1, while on BF it is a_2b_1 plus the horizontal force at F due to the lifting chain. The compression on BF is also equal to the horizontal force due to the compression on the jib, which is represented by $ab_1 + fb$. The maximum bending moment on BE occurs at F, and is equal to

$$(de - a_1f) \times \frac{BF \times FE}{BE}.$$

On DF the maximum bending moment occurs at C, and is equal to

$$fb \times \frac{CF \times DC}{DF}.$$

As an illustration, the crane shown in Fig. 21 has been taken to be loaded with 5·75 tons, and the stresses have been worked out and tabulated in Table II.

In the above case it is assumed that the forward edge of the roller path is of such a diameter that the forward rollers come directly under B. In many cases they come, however, somewhere between B and F. This produces some modification of the stresses, but the principle on which they are determined remains the same. There is a further point to determine. When there is no load on the hook, the balance weight at E would cause the crane to fall over backwards if the point of support remained at B. We must, therefore, find a fresh point of support G. The distance

$$GE = \frac{L W_b}{W_b + W_a}.$$

In the case illustrated in Fig. 21 the load at A consisting of the crane hook and its attachments plus half the jib is 1 ton, and GE equals

$$\frac{25 \cdot 5 \times 1}{1 + 11 \cdot 2} = 2 \cdot 1 \text{ feet.}$$

If, then, the roller path is made of such diameter that the extreme points of support come at B and G respectively, the centre of gravity will be at B when the crane is fully loaded, and at G when unloaded, and at intermediate loads it will come between B and G.

TABLE II.—STRESSES IN LOCO. JIB CRANE. Figs. 21 and 22.

Tension + Compression −

Member. Fig. 21.	Ref. Fig. 22.	Tons.
Jib, AB, ...	ab	−19·7
Lifting rope, AC, ...	a_1b	+ 5·0
Stay, AD, ...	ad	+12·0
Stay, DE, ...	a_2e	+19·2
Vertical, DC, ...	de	−10·5
Vertical, CF, ...	$de - a_1f$	− 8·2
Upward pull at E due to stay DE, ...	b_1e	14·2
Downward force at E due to compression on $CF = \frac{8 \cdot 2 \times 3 \cdot 16}{8 \cdot 66} =$	...	3·0
Resultant upward pull at $E = 14 \cdot 2 - 3 =$	...	11·2
Balance weight at $E = \frac{5 \cdot 75 \times 16 \cdot 83}{8 \cdot 66} =$	...	11·2
Downward load at $B = b_1f + \frac{8 \cdot 2 \times 5 \cdot 5}{8 \cdot 66} =$	...	16·95
Downard check $= 11 \cdot 2 + 5 \cdot 75 =$	...	16·95
Compression on EF, ...	a_2b_1	−13·0
Compression on BF, ...	$a_2b_1 + 3$	−16·0
Horizontal force at B due to compression on jib, ...	$ab_1 + fb$	16·0
Bending moment at F on BE in ft.-tons $= \frac{8 \cdot 2 \times 3 \cdot 16 \times 5 \cdot 5}{8 \cdot 66} =$	...	16·25
Bending moment at C on DF in ft.-tons $= \frac{4 \cdot 5 \times 2 \times 4}{6} =$	...	6·0

So far, we have dealt with the upper portion only of the crane structure, but to ascertain the stability of the crane as a whole we must take the truck into account. In the illustration given it has been shown that the upper portion of the crane balances on the points B and G with no load and full load respectively. If, now, the truck is so designed that its wheel base and centres of rails come under the points B and G, the crane will not only be in balance at the two extremes, but will have a margin of stability equal to the weight of the truck, although a larger margin is usual. The roller race may be made of a less diameter than the distance BG without affecting the general stability. If, however, it is made less than BG tension will be thrown on the centre pin as the extremes of full load and no load are approached. The tension on the pin

$$= \frac{W_b(R - R_1)}{R_1} - \frac{W_d(R_2 + R_1)}{R_1}$$

at full load, and

$$= \frac{W_d(R_2 - R_1)}{R_1} - \frac{W_b(R + R_1)}{R_1}$$

at no load.

The remaining stresses, in the truck, consist of simple bending moments, which may be easily determined.

3-Ton Electric Locomotive Jib Crane.—This crane was constructed by Messrs. Stothert & Pitt, Ltd., of Bath, and is illustrated in Fig. 23. It was constructed to handle a load of 3 tons at a maximum radius of 20 feet, the minimum radius of the jib being 13 feet, as shown in dotted lines on the drawing. The gauge of rails was 6 feet, the wheel base 7 feet, and the crane could handle its full load in all positions without requiring to be propped up on cross girders or clipped to the rails. The speed of lift for 3 tons was 100 feet per minute, and for 2 tons 150 feet per minute, the hoisting motor being of 30 B.H.P. The slewing motion was driven by a separate motor of 2¾ B.H.P., the speed of slewing with 3 tons at maximum radius being 130 feet per minute, and with 2 tons 200 feet per minute. One motor of 11 B.H.P. was provided to drive the derricking and travelling motions, clutches being provided to throw the motor into engagement with one or other of the motions as required. The speed of travelling with a 2-ton load was 200 feet per minute.

Current was collected from an overhead wire by means of a trolley pole as in electric tramway practice, the return current passing through the rails. The principal stresses with full load were approximately—Compression on jib at maximum radius

15·3 tons, at minimum radius 10·3 tons. Tension on stay 10·5 tons and 4·4 tons. Tension on lifting rope 3 tons. The stay rope was three part and 2½ inches circumference, with a breaking strain of 19·75 tons, the factor of safety being 5·65. The lifting rope was 2⅜ inches circumference, having a breaking strain of 17·85 tons, and factor of safety 5·95. The sheaves and lifting drum were about 21 times the diameter of the lifting rope.

The total weight of the crane was 20 tons.

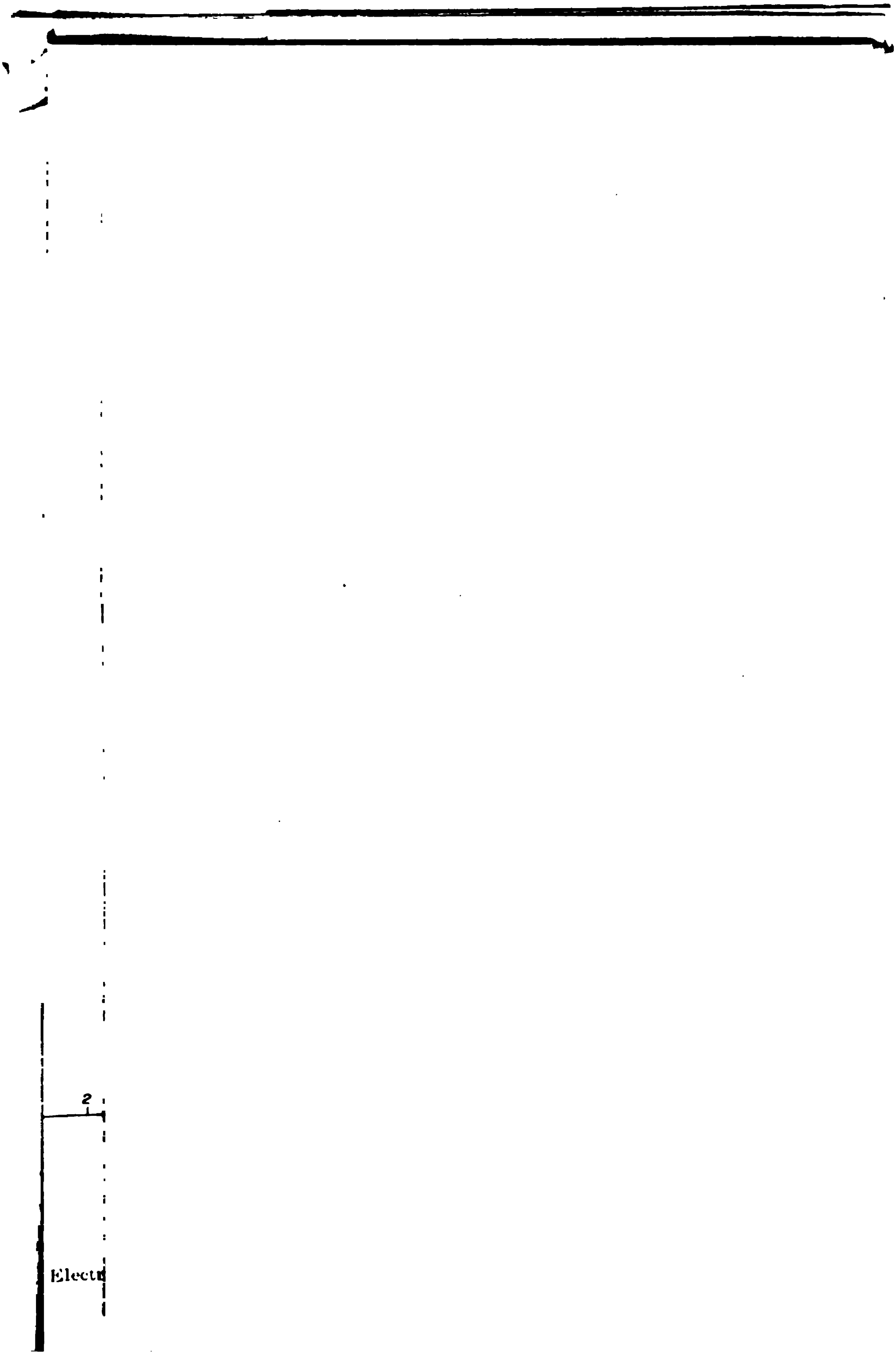
2½-Ton Portable Electric Jib Crane.—Made by Messrs. J. H. Wilson & Company, Ltd., of Seacombe, near Birkenhead (Fig. 24). This crane was driven by power in the lifting and slewing motions. It was provided with broad wheels, the front pair being arranged to swivel, to enable it to be drawn along ordinary roads, and the truck frame had extensions at the sides fitted with screws, so that the crane could be propped up level when working on an uneven surface.

The jib was cranked in order to increase its reach, and the jib stay was provided with a union with right and left-hand screws for varying the horizontal radius of the jib. The speed of lifting was 50 feet per minute for 2½ tons with double purchase rope, and 100 feet per minute with 1¼ tons and single purchase. The lifting motor was of 16 B.H.P., running at 370 revolutions per minute, and the lifting gear was single reduction, the motor pinion gearing directly into a large spur wheel on the lifting barrel shaft. This gearing ran in an oil-tight gear case with oil bath, so as to ensure easy and quiet running. At the head of the jib a limit switch was placed to cut off current in case of overwinding. The hoisting rope was 2 inches circumference, with a breaking strain of 13 tons, the factor of safety being 10·4. The diameter of sheave was 23·6 times the diameter of the rope, and the hoisting barrel 19 times.

The slewing gear was worked by a 5 B.H.P. motor running at 760 revolutions per minute, the speed reduction gear consisting of a steel-worm and phosphor-bronze worm-wheel, totally enclosed and running in oil. The speed of slewing was about 250 feet per minute. A junction box was placed on the crane truck, and current was brought to it from a junction box in the roadway by means of a flexible cable.

The weight of the crane complete was 8¾ tons.

Harbour Cranes.—For harbour work, for the loading and unloading of ships, locomotive jib cranes are frequently used having carriages of sufficient height and width to allow of the passage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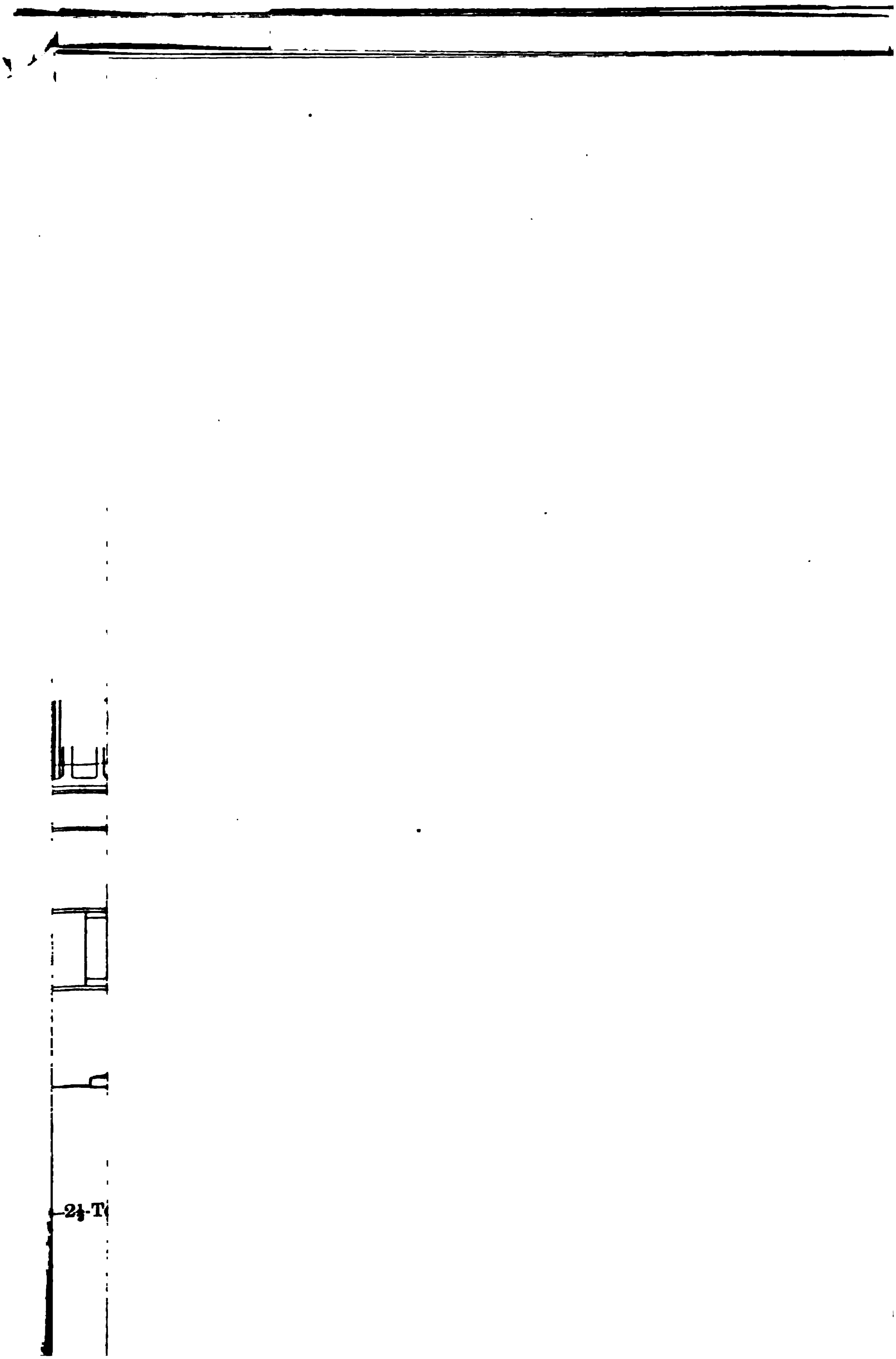


under them of railway trains and other traffic. A crane of this type, made by Messrs. J. H. Wilson & Company, Ltd., is shown in Fig. 25. This was one of a pair constructed for the Belfast Harbour Commissioners. The cranes were designed for a working load of 4 tons at a radius of 50 feet. The height of carriage is 14 feet from the rail level to allow for the passage of locomotives, etc., and the extreme outside width of the framework is about 14 feet. The hoisting motor is of the series-wound enclosed type of 50 B.H.P., giving a lifting speed of 160 to 180 feet per minute. The hoisting barrel is provided with a clutch, so that it can be run free on the shaft for lowering, the speed of descent being controlled by a brake. The slewing motor is of 10 B.H.P., and that for derricking 9 B.H.P. The slewing motor will cause the loaded crane to make a complete revolution in 30 seconds, and the rate of derricking is 20 feet in 60 seconds. The travelling motor is of 30 B.H.P., giving a speed of travel of 100 feet per minute. The cranes are each provided with a 35-cwt. coaling grab, made by Messrs. Priestman Brothers & Company, Ltd., of Hull. The weight of the grab is balanced by means of weights at the back of the crane frame, so that the hoisting motor has to lift only the weight of the contents of the grab, and not that of the grab itself. Other cranes of this type made by Messrs. Cowans, Sheldon & Company, Ltd., of Carlisle, are illustrated in the chapter on *Crane Installations*.

CHAPTER III.

DERRICK CRANES.

THE **Derrick Crane**, sometimes called the Scotch crane, is principally used in the erection of buildings and in work of an allied character, for which purpose it is found to be most convenient.

Generally the Derrick is used as a fixed crane, in which case its duty consists in lifting, traversing, and lowering loads anywhere within a space forming a portion of an annular ring. As the back stays which support the top of the mast are placed at an angle of 90° the jib can be swung through nearly 270°, so that the annular space is bounded by this angle, while its outer and inner radii correspond with the maximum and minimum horizontal radii of the jib.

Occasionally, in order to increase its range, the three points of support of the crane are mounted on trucks running on rails. It then becomes capable of performing the same duty as a locomotive jib crane, except that the jib cannot turn through a complete circle.

Fig. 26 shows diagrammatically the arrangement of framing and ropes of a Derrick crane. The load hangs on the rope *a*, which passes over a sheave at the head of the jib *b*, and is led over a guide pulley *c* to the hoisting barrel *d*. The stay rope *e* is attached to the top of the mast *f*, and is led round the guide pulleys *g* and *h* to the derricking barrel *i*. When loads are being hoisted and lowered the derricking barrel is held by a ratchet and pawl. When it is desired to raise or lower the jib the two barrels are clutched together, and the pawl is thrown out of engagement with the derricking barrel. The two barrels are arranged so that as the jib stay rope is hauled in the load rope is paid out, or *vice versâ*, the relative speeds of rope being such that as the jib is lifted or lowered the load travels along an approximately horizontal line. Thus the load may be travelled horizontally either by lifting or lowering the jib or by slewing it. For slewing the jib a large toothed-wheel *j* is bolted to the bottom framing *kk*. A pinion, not shown in the drawing, gearing into this wheel is carried by brackets attached to the mast *f*. On the pinion being rotated by a suitable motor, it runs round the large wheel carrying the mast and jib with it.

As the jib only swings backwards and forwards through a portion of a circle, electric connection between the fixed and moving parts may be made by means of flexible cable, sliding contacts not being necessary.

The maximum stresses occur in the framing when the jib is

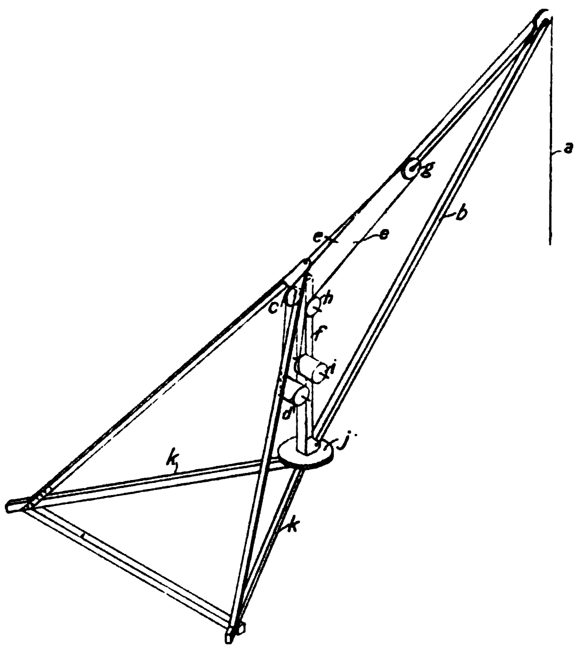
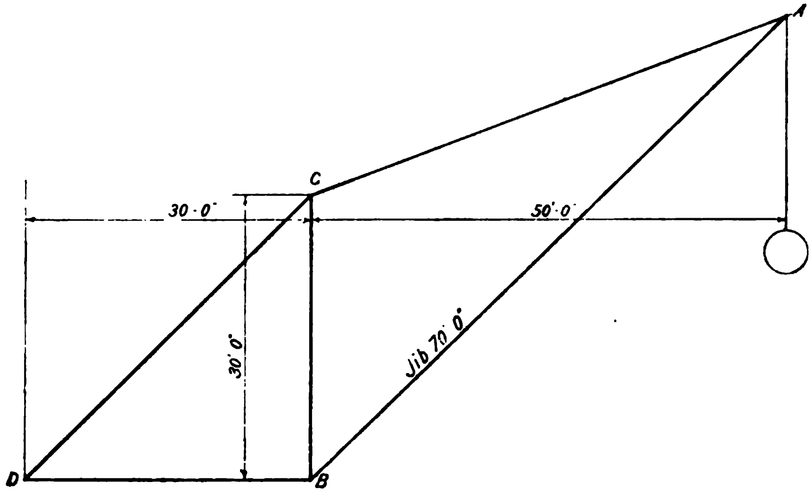


Fig. 26.—Derrick Crane.

directly in line with one of the back stays, as in Fig. 27*a*, and when it is swung round to its extreme position, and touching one or other of the back stays, as in Fig. 27*c*. In either of these positions all the stresses, except those due to wind pressure, come practically on the one back stay, while in any other position of the jib they are divided between the two.

To ascertain the stresses in the case shown in Fig. 27*a*, lay out the line *cb*, Fig. 27*b*, representing to any convenient scale the load hanging from *A* plus half the weight of the jib (see p. 19). Produce *cb* at both ends. From *b* draw a line parallel to *AB*, and from *c* a line parallel to *AC*. *ab* will then represent the



Load hanging from A = 5·0 tons.
Half weight of jib = 1·5 „

6·5 „

Fig. 27*a*.

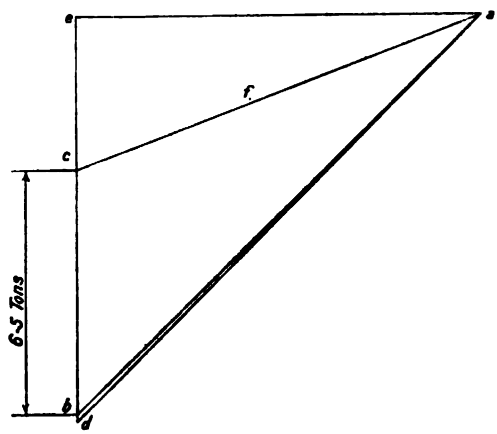


Fig. 27*b*.

compression on the jib, and *ac* the tension in the lifting rope and jib stay as these lie close together, and may for the purposes of the stress diagram be represented by one line. On the line *ca* lay off *cf* equal to the load hanging from *A*, then the tension on the jib

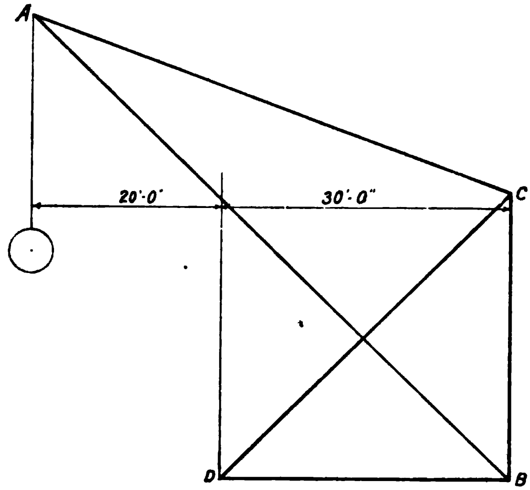


Fig. 27*c*.

stay is represented by *fa*. From *a* draw a line parallel to *CD*, cutting *cb* produced, at *d*. The compression on the mast *CB* is then represented by *cd*. The downward force at *B* equals *cd* plus *eb*. To obtain the total load at *B* and *D* we must take into account the weight of the machinery, and those parts of the structure not included in the stress diagram. The total load is, of course, the weight of the whole structure. The tension in the back stay *CD* equals

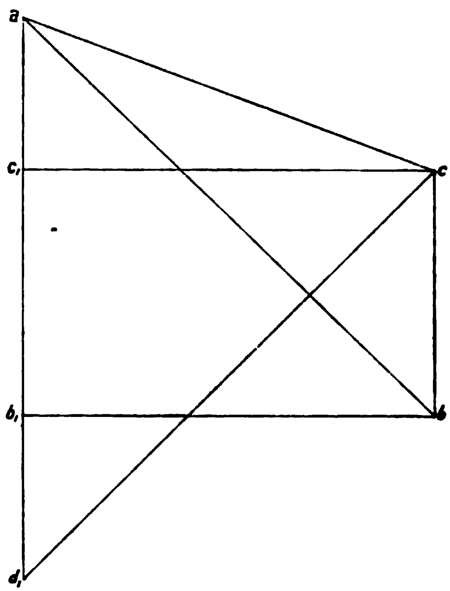


Fig. 27*d*.

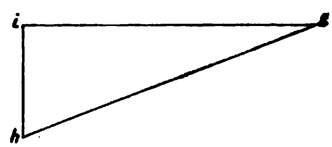


Fig. 27*e*.

ad, and the upward pull at *D*, due to this tension, equals *ed*. This upward pull is equal to the amount of balance weight required to hold the crane in equilibrium in this position. From *a* draw a horizontal line cutting *cb* produced at *e*. Then the compression on the member *DB* is represented by *ea*. As the jib stay rope and lifting rope pass over guide pulleys at the top of the mast on to barrels mounted lower down, there is an additional compression on the mast between these points due to the pull on these ropes.

The stay rope being in two parts, see Fig. 26, the pull on this rope equals 0·5 *fa*.

In Fig. 27*e* draw the line *gh* equal to *cf* plus 0·5 *fa* parallel to *AC*. From *h* and *g* draw vertical and horizontal lines meeting in *i*. Then the compression on the mast due to the pull of the ropes will be equal to *hg* minus *hi*. When the jib is swung round to its other position, as shown in Fig. 27*c*, the stresses in *AB* and *AC* remain as before. The stay *CD* is now in compression. This compression is represented by the line cd_1 in Fig. 27*d*, and is equal in amount to the previous tension. The mast is subjected to a tensile stress represented by the line ad_1, which is diminished between the guide pulleys and barrels by the amount of compression due to the pull on the ropes (see Fig. 27*c*). The upward pull at *B* equals b_1d_1.

For purposes of illustration, the stresses have been worked out for a 5-ton Derrick, with 70-foot jib, and are given in Table III.

TABLE III.—STRESSES IN DERRICK CRANE. (Figs. 27*a* to 27*e*).

Tension + Compression −

Member.	Fig. 27*b*.		Fig. 27*d*.	
		Tons.		Tons.
Jib, *AB*,	*ab*	−15·3	*ab*	−15·3
Lifting rope, *AC*,	*cf*	+ 5·0	...	+ 5·0
Jib stay, *AC*,	*fa*	+ 6·6	...	+ 6·6
Back stay, *CD*,	*ad*	+15·4	cd_1	−15·4
Bottom stay, *DB*,	*ea*	−10·8	b_1b	+10·8
Mast, *CB*,	*cd*	− 6·75	ad_1	+15·1
Upward force at *D*,	*ed*	10·8	...	...
Check $\frac{6\cdot5 \times 50}{30} =$	...	10·8	...	...
Upward force at *B*,	...	...	b_1d_1	4·3
Check $\frac{6\cdot5 \times 20}{30} =$	...	...	...	4·3
Downward force at *B*,	*eb* + *cd*	17·3	...	...
Check 10·8 + 6·5 =	...	17·3	...	...
Downward force at *D*,	...	...	c_1d_1	10·8
Check 6·5 + 4·3 =	...	...	...	10·8
Stress on mast, *CB*, allowing for pull of ropes. Fig. 27*e*. *hg* − *hi* = 5·4,	...	−12·15	...	+ 9·7

CHAPTER IV.

TRANSPORTERS.

The Temperley Transporter.—A special form of crane was invented by Mr. John Temperley in 1892 (patent No. 21,170), and was called by him a Transporter.

The special feature in which this transporter differs from an ordinary crane is that in it the operations of lifting, travelling, and lowering the load are all accomplished by the hauling in and paying out of a single rope, this being rendered possible by the use of an automatic travelling carriage, which forms the subject of Mr. Temperley's first patent.

In its simplest form the Temperley transporter consists of the straight beam *a*, Fig. 29, the automatic travelling carriage *b*, the rope *c*, and the winch *d*. The duty of the transporter is to transfer material from the point *A* to the point *B*. Its operation is as follows:—When the carriage is in the position shown, it is locked

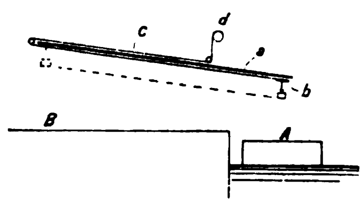


Fig. 29.—Transporter.

to the beam. The bucket being filled, the driver starts the winch, hauling in the rope and lifting the bucket. When it is lifted up to the carriage the lifting hook is locked to the carriage, so that the weight of the load is taken off the rope, and at the same instant the carriage is unlocked from the beam. As the winch is still hauling in the rope, the carriage runs along the beam. When it arrives at the upper end the driver reverses the winch, which now pays out the rope. This causes the carriage to lock itself to the beam and release the lifting hook, so that as the rope continues to pay out the bucket is lowered. When it has been lowered a sufficient distance it is emptied, and the driver again reverses, so lifting the bucket. When the lifting hook arrives at the carriage it is locked again, and the carriage is freed from the beam. The driver again reverses, and as the rope is paid out the carriage runs down the beam by gravity. On arriving at the bottom end, the carriage locks itself to the beam, and the lifting hook is freed. As the rope is still paying out the bucket is lowered down to *A* ready to be refilled, and the cycle of operations can be repeated.

As the material is transported from *A* to *B* simply by the movements of the one rope without requiring any movement of the crane

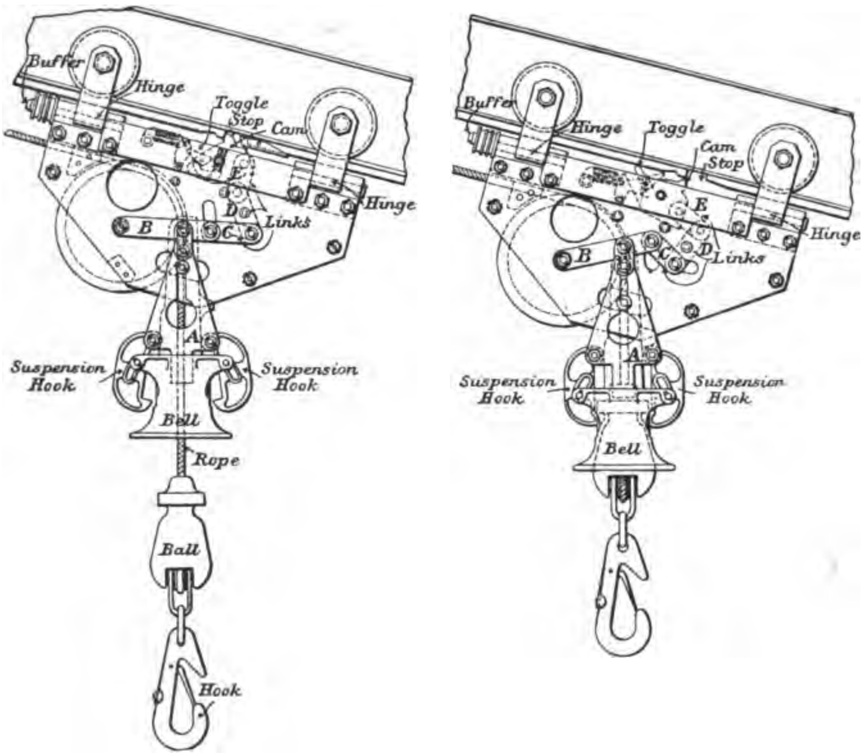


Fig. 29*a*.* Fig. 29*b*.*

Travelling Carriage on the Single-rope Toggle-cam System.

itself, such as slewing or travelling, the cycle of operations can be got through quickly, and large quantities of material can be transferred in comparatively short periods.

The construction of the automatic travelling carriage in its most recent form is shown in Figs. 29*a* and 29*b*. On the underside of the beam on which the carriage runs, stops, one of which is shown in the figures, are provided at intervals in positions where loads require to be lifted or lowered, these stops being for the purpose of locking the carriage to the beam. The locking is effected by the tooth of the cam, which is pivotted to the carriage frame, projecting into the gap in the stops, as shown in Fig. 29*a*. The lower portion of the carriage is hinged to the part which runs on the beam, so that it is free to swing laterally if a load when hooked on to the lifting rope is not directly under the beam. In Fig. 29*a* the carriage is locked to the beam, and the hook is free for lifting or lowering, while in Fig. 29*b* the hook is locked to the carriage, while the latter is free to run along the beam. The action of the mechanism is as follows :—The carriage being locked to the beam, and the lifting hook free as in Fig. 29*a*, when the hook is hoisted as far up as it will go, the ball which is suspended on the rope just above the hook enters the bell and striking the lower end of the link *A* drives it up. As the link goes up it closes the suspension hooks, causing them to grip the lip on the ball, as shown in Fig. 29*b*, so that the load is now suspended from the frame of the carriage, and its weight is

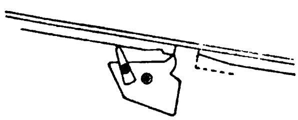


Fig. 29*c*.

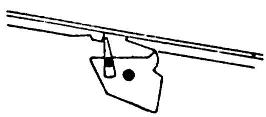


Fig. 29*d*.

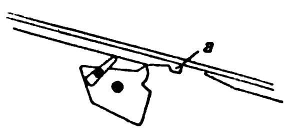


Fig. 29*e*.

taken off the lifting rope. The link *A* also pushes up the lever *B*, the left-hand end of which is pivotted to the carriage, while the right-hand end rises, pulling with it the link *C*, which causes the lever *D* to swing, its upper end moving to the right, so pulling the link, *E*, which draws down the cam until its tooth is clear of the stop, as shown in Fig. 29*b*. The position of the parts is now completely reversed, the hook being locked and the carriage free. On the cam there is pivotted a little piece called a toggle. At the moment the cam comes out of engagement with the stop this toggle comes in contact with the lower side of the beam, and slopes from left to right, as shown in Figs. 29*b* and 29*c*. As the carriage is drawn up along the beam the upper end of this toggle is caught by the first stop which the carriage passes, and its position is reversed so that it now slopes from right to left. The carriage is drawn a little way past the stop at which lowering is to take place, and the direction of the rope is reversed. The carriage now backs down the beam till the toggle strikes the point *a*, Fig. 29*e*. This causes the cam to swing up, so that its tooth re-enters the gap in the stop. At the same time, it pulls link *E* so that the lower end of lever *D* moves from left to right so, through link *C*, pulling down lever *B* and link *A*, and opening the suspension hooks so that the mechanism returns to the position shown in Fig. 29*a*. It will be noted that when the hook is hoisted up and the cam freed from the beam the toggle always slopes from left to right, as shown in Fig. 29*c*, so that it will allow the carriage to run down the beam without engaging any of the stops. To lower the hook at any particular stop the carriage must be run down past that stop, then drawn up the beam again past the stop, in order to reverse the toggle, and then backed down on to the stop. To save the time which would be taken by these three movements the stop at the lower end of the beam is provided with a projection, shown dotted in Fig. 29*c*, which catches the cam itself, and throws it into engagement with the stop. This projection is provided with a spring, not shown in the figure, to soften the blow, and is made movable, so that it can be attached at any point on the beam, where a number of lowerings are to be made.

The winch for operating the transporter does not call for any special description, as any good winch will serve the purpose.

The original simple design of beam has been greatly elaborated, and beams are now made to pivot horizontally for adjustment to various points for lifting and lowering, they are also made to hinge up to clear the rigging of ships, and are also mounted on travelling towers and various other arrangements.

Fig. 30 shows a transporter at work at the Glasgow Corporation Electric Generating Station, unloading coal from canal barges and delivering it into storage hoppers in the boiler house. The coal is lifted from the barges in skips carrying 15 cwts., provided with drop bottom doors, out of which the coal runs into the hopper on a movable weighing machine in the valley of the boiler-house roof. The transporter is worked by an electric winch, and is carried by a travelling tower running on rails having a gauge of 19 feet. As the waterside arm extends over a public road it is hinged, and when the transporter is not in use it is drawn up into a vertical position by means of a derrick barrel on the winch.

Fig. 31 shows a travelling tower transporter erected at the Poutiloff Works at St. Petersburg for unloading coal from barges and depositing it into railway trucks. It commands three lines, as shown in the figure. The stops on the underside of the beam are

To face page 36.

Fig. 30.—Temperley Travelling Tower Transporter.

situated immediately over the centre of each railway line, so that the bucket can only be lowered exactly over the truck. The transporter is designed for handling skips of 50 cubic feet capacity, holding about 25 cwts. of coal, and is capable of making 50 to 60 trips per hour. This transporter is fitted with Temperley's automatic dumping fall block and skip, which enable the driver to tip the load at will at any point, so that there is no need for a workman at the point of discharge. The principle on which the

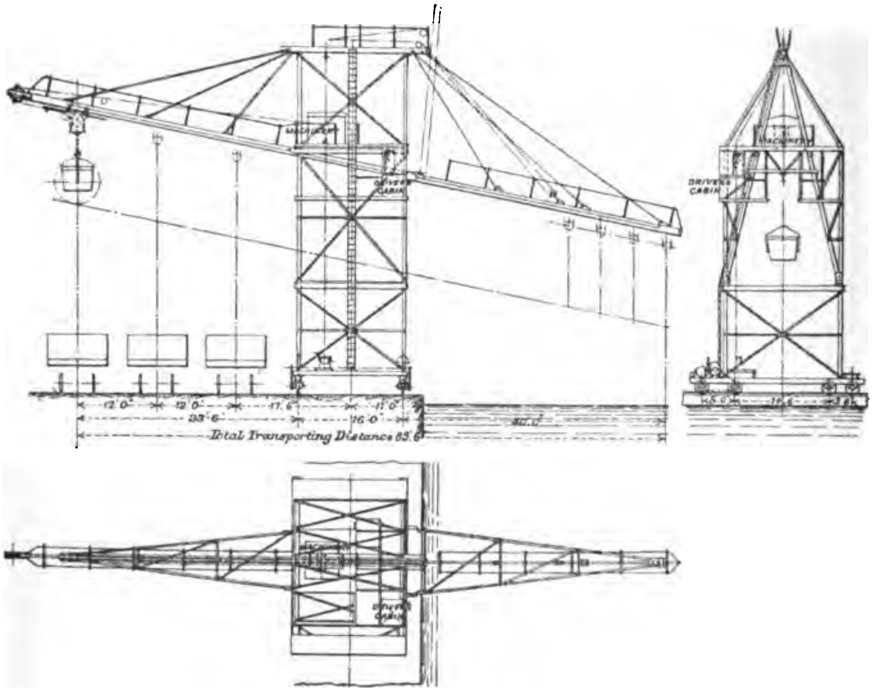


Fig. 31.*—Travelling Tower Transporter at the Poutilow Works, St. Petersburg.

automatic fall block and skip work is illustrated in Figs. 32*a*, 32*b*, and 32*c*.

The skip is held in the upright position by a latch at the side, which can be raised by a lifting bar which is pivotted to the bridle of the skip. The fall block has two hooks, one of which takes the bridle of the skip and holds the weight, while the other is employed to raise the lifting bar when it is desired to tip the contents of the bucket. The mechanism employed to lift the auxiliary hook is somewhat similar to that on the travelling carriage, the movement

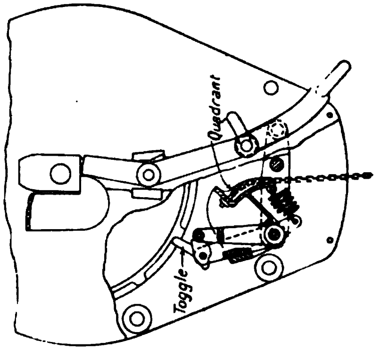


Fig. 32c.

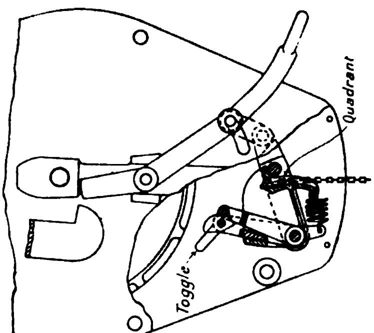


Fig. 32b.

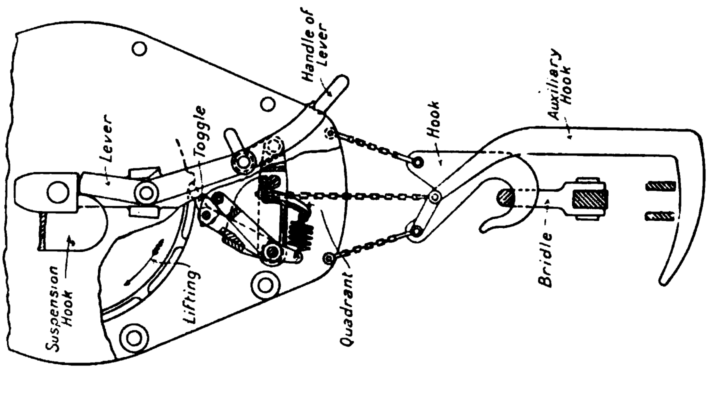


Fig. 32a.

being obtained by a small toggle which engages with projections on the rim of the fall block sheave. The operation is as follows:—The skip being full, the handle is pulled to the right hand, and the parts are thrown into the positions shown in Fig. 32*b*, in which the toggle is clear of the sheave. The skip is then hoisted, and when the fall block reaches the carriage the suspension hook (see Figs. 29*a* and 29*b*), of which there is in this case only one, as it engages with the fall block pushes the lever over, moving the parts into the positions shown in Fig. 32*a*, except that the toggle slopes from left to right, as in Fig. 32*b*. On lowering the skip one of the projections on the sheave catches the toggle and turns it over so that it is now cocked ready for tipping. When the skip has been lowered a sufficient distance the driver reverses, a projection on the sheave catches the toggle, and this moves the quadrant into the position shown in Fig. 32*c*. As the quadrant moves it lifts the central chain which is attached to the auxiliary hook, so lifting the latch and causing the bucket to tip.

In a later form of Temperley transporter for heavy loads and quick speeds the automatic travelling carriage is discarded, and a system of multiple ropes is used. The general arrangement of ropes when used for working a grab is shown in Fig. 33. The hoisting drum is divided into two parts connected by epicyclic gearing, the motor driving on to a central spur wheel. One-half of the rope is connected to the body of the grab, while the other half is led to the sheaves which open and close the grab.

When the two drums are driven through the epicyclic gearing the grab is lifted or lowered with the jaws closed. To open or close the jaws the left-hand drum is held by a brake, and on running the motor the right-hand rope hauls in or pays out as the case may be, and so operates the jaws of the grab. To lift or lower the grab with the jaws open the two drums are locked together. The epicyclic gear being now inoperative, the weight of the grab is taken on the left-hand rope, while there is no tension on the right-hand rope, and as this rope hauls in and pays out at the same speed as the left-hand one, there is no movement of the grab jaws. For travelling the carriage a separate winding drum and motor are used. If a brake be applied to prevent the hoisting motor from running and the travelling motor be started, the carriage will travel along and the hoisting drums will revolve in opposite directions, being driven by the hoisting rope, and the grab will travel in a horizontal line, being neither lifted nor lowered. If both motors run simultane-

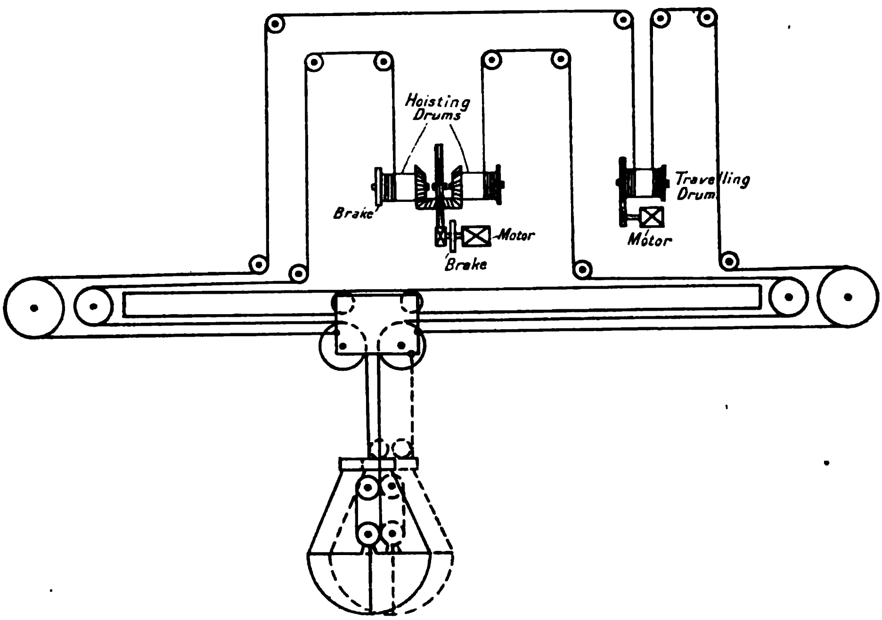


Fig. 33.

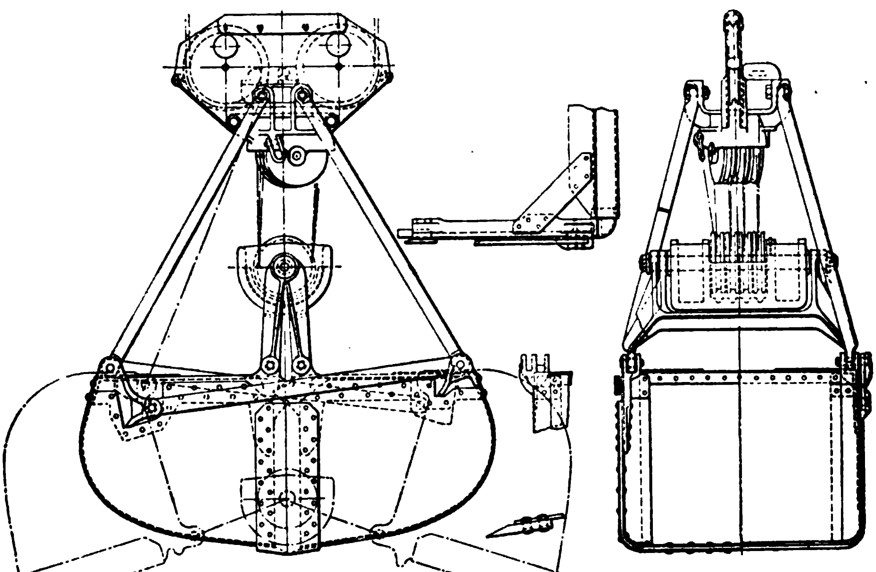
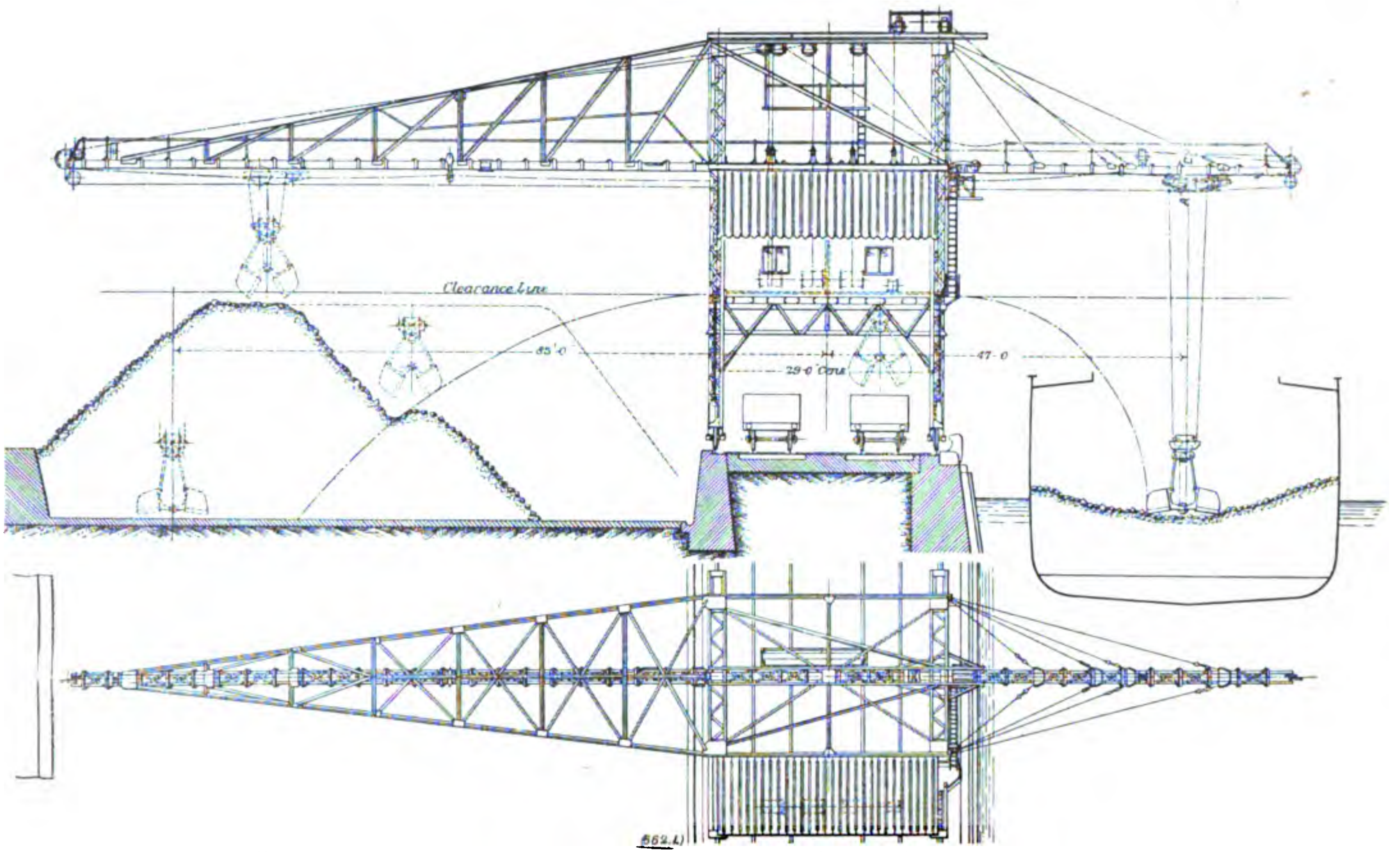


Fig. 34.*—Temperley Grab for Two-rope System.



Figs. 35 and 36.*—Travelling Tower Transporter on the Admiralty Pier, Portland Harbour.

ously, the load will be lifted or lowered, as well as travelled, and the grab will follow a diagonal path.

Fig. 34 shows clearly the construction of the grab. A special feature is the use of the pivotted side links by means of which a gradually decreasing speed and correspondingly increasing force is imparted to the jaws as they close.

Figs. 35 and 36 show a transporter at Portland working on the principle just described. This transporter consists of a travelling tower running on rails having a gauge of 29 feet. The overhead track for the carriage is 132 feet long, and is composed of two cantilevers. The inboard cantilever is fixed, and is a braced structure having an out-reach from the centre of the tower of 85 feet. The outboard cantilever is arranged to hinge up, its out-reach is 47 feet, and it is supported by wire ropes instead of steel tie bars.

The hoisting winch motor is of 60 brake horse-power, and the travelling motor 50 brake horse-power. A third motor of 15 brake horse-power is provided for lifting the hinged arm.

The transporter was guaranteed to discharge 200 tons of coal in four hours from the hold of a collier and distribute it over 75 to 85 feet from the centre of the tower. On actual trial the amount handled varied from 208 to 220 tons in the time stated.

2¼-Ton Transporter Crane.—Constructed by Messrs. Ransomes & Rapier, Ltd., of Ipswich. The general design of this crane is shown very fully in Fig. 37. A large carriage running on eight wheels supports a cantilever which can slew through a complete circle. A small carriage for handling the load runs along the bottom of the cantilever. A diagram of the arrangement of ropes is shown in Fig. 38. The load carriage is hauled backwards and forwards by the rope *a* operated by the drum *b*, and the load is hoisted and lowered by the rope *c* operated by the drum *d*.

It will be noted that by running the carriage inwards with the hoisting barrel standing, the load will be hoisted along a path having an angle of 45° to the horizontal, and running the carriage out lowers the load along the same path. By running both drums and regulating their speeds, the load may be caused to follow a path of any desired angle. The load can be hoisted and lowered vertically by running the hoisting motor only, the carriage in this case being held stationary by the hauling rope. The actual path taken by the load when working is shown in Fig. 37. Thus, when a load is being transported, instead of taking two separate straight courses, up and along, it follows a diagonal course from corner to

MAIN TRAVEL	4 HP Motor AT 380 R.P.M.	N	2	6 3/16"	14	[illegible]"	98.1 : 1	29.44 F.P.M.
		O	2	1'-1 3/8"	28	1 1/2"		
		P	2	4 5/16"	18	1 3/4"		
		Q	2	1'-8"	36	1 3/4"		
		R	2	10"	18	1 3/4"		
		S	7	2'-8 5/16"	58	1 3/4"		

corner. Its course is thereby shortened, and the process of transportation expedited.

The four motions of the crane are driven by separate motors. As the crane is used for transporter work, for transferring loads directly from point to point without intermediate motions, the travelling and slewing gears are only used for setting the crane in position over its work, and so are driven by comparatively small motors.

The lifting barrel is driven through single reduction gear by a motor of 50 brake horse-power, 220 volts, running at 315 revolutions per minute, giving a lifting speed of 241 feet per minute with $2\frac{1}{4}$ tons load. With half-load the speed of motor is 394 revolutions per minute, and the speed of lift 300 feet per minute. The barrel for traversing the carriage is also driven by a motor of 50 brake horse-power, and the traversing speeds are the same as those for lifting.

The slewing gear is driven by a motor of 7 brake horse-power,

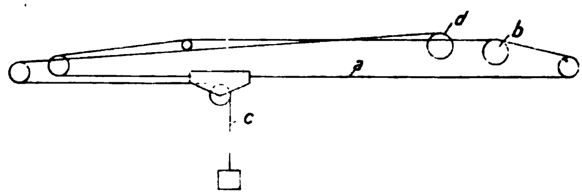


Fig. 38.

running at 380 revolutions per minute, and with full load the cantilever turns through a complete circle in about five minutes, the time of turning being less with lighter loads. The travelling gear is also driven by a motor of 7 brake horse-power, running at 380 revolutions per minute, the speed of travel being about 30 feet per minute.

Some recording ammeter diagrams taken by the author on this crane when unloading a barge are shown in Fig. 39. The material was not handled in bulk, but was packed in baskets, several of which were hung on the crane hook to make up each load. The loads were somewhat under the light load capacity of the crane, but the diagrams illustrate the cycles of operation practically as well as if loads of the full capacity of the crane had been handled. Referring to the diagrams, No. 11 cycle has been divided up into its separate periods, and these have been tabulated in Table IV. Owing to the fact that each load is made up of several pieces, each of which has to be hooked and unhooked by hand, the crane stands during a considerable portion of each cycle. Thus of the time taken for the twelve complete cycles shown on the diagram, 43·5 per cent. is spent in handling the load with the crane standing. The average working time of the hoisting motor was 33·7 per cent., and of the traversing motor 28·4 per cent. of the time the crane is in use. It will be noted that during a portion of each cycle both motors are running.

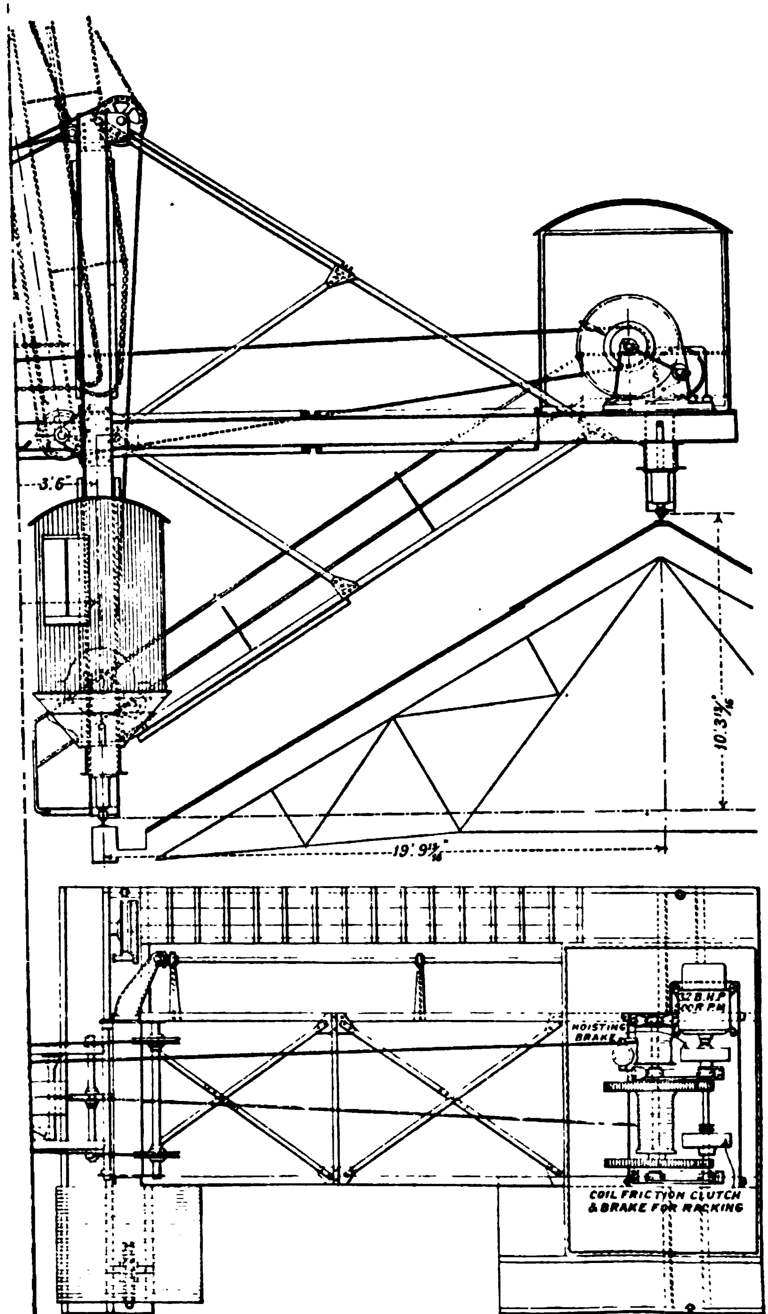
TABLE IV.—CYCLE NO. 11 OF FIG. 39.

No. of Period.	Description of Operation.
1	Hoisting motor lifting load vertically.
2	Hoisting and traversing motors both running and load rising obliquely.
3	Traversing motor only running, and load completing its journey upward at an angle of 45°.
4	Men steadying load and getting ready to guide it when lowered.
5	Lowering.
6	Unhooking load.
7	Hoisting up hook.
8	Running out.
9	Lowering hook to barge.
10	Hooking on some load.
11	Paying out rope to reach other load.
12	Hooking on remainder of load.

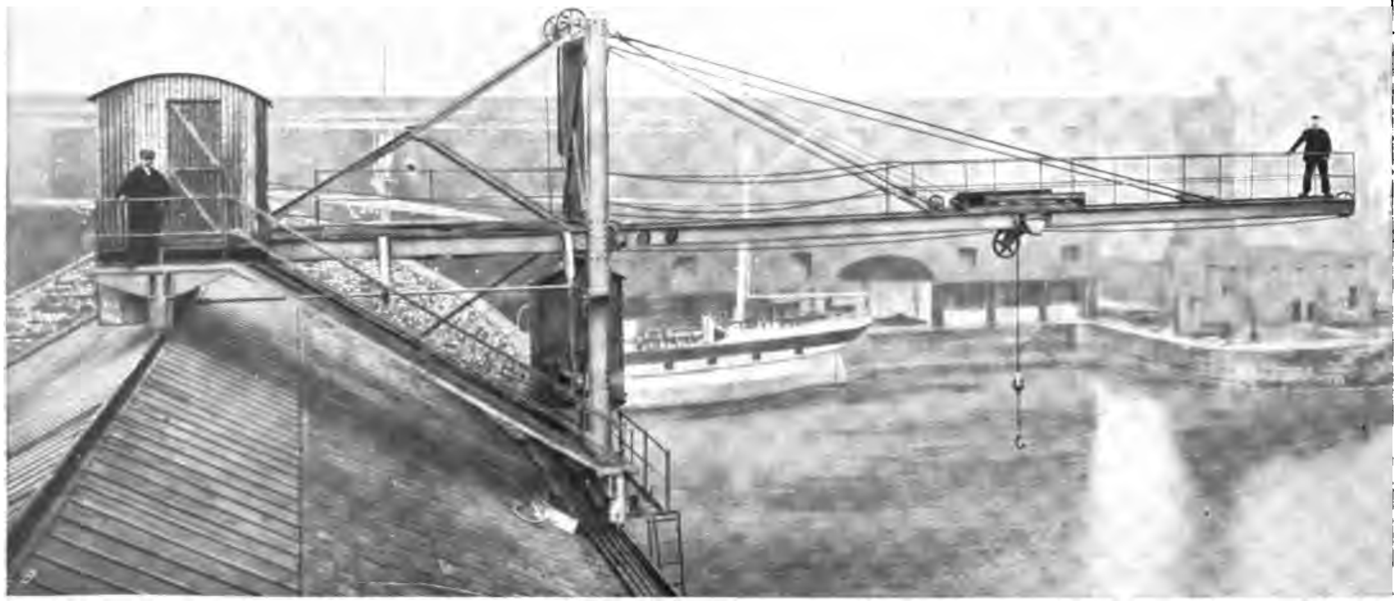
1½-Ton Electric Transporter Crane (Figs. 40, 41, and 42, constructed by Messrs. Applebys, Ltd., Leicester). This crane runs on the roof of a warehouse, and is used for loading and unloading vessels. In order to clear the rigging of ships the overhanging arm is arranged to lift up.

In this crane the arrangement of ropes is different to that in the last crane described. The traversing rope is fastened to the outer end of the travelling carriage, passes round a guide pulley at the end of the cantilever arm, and leads thence to the traversing barrel. It is only used for traversing the carriage outwards. The hoisting rope passes round a guide pulley on the carriage, round a second guide pulley at the inner end of the arm and to the hoisting barrel. This rope serves the double purpose of hoisting and lowering loads and traversing the carriage inwards. The two barrels are on one shaft, and may be run together or the hoisting barrel may be run while the traversing barrel remains at rest, clutches and brakes being provided for the purpose.

For traversing the carriage both barrels are run, while for



3' 6"
10' 3 1/16"
19' 9 13/16"
HOISTING BRAKE
COIL FRICTION CLUTCH & BRAKE FOR RACKING



To face p. 44.]

Fig 41.*—1½-Ton Transporter Crane.

To face p. 44

Fig. 42.*—1½-Ton Transporter Crane.

hoisting and lowering loads the traversing barrel is stopped, the hoisting barrel only being run.

It will be noted that when the crane is fully loaded the greater part of the weight of the crane and its load comes on the warehouse wall, but when the crane is unloaded, the weight of the balance weight comes on the ridge of the roof, and the roof principals must, of course, be strong enough to stand this.

The crane is capable of handling a load of 1½ tons with an outreach of 47 feet from the centre of the lower rail, the minimum outreach being 3 feet 6 inches. The vertical lifting range is 76 feet.

The lifting and traversing gear is driven by one motor of 32 brake horse-power running at 500 revolutions per minute, the speed of lift with 1½ tons being 250 feet per minute, and with no load 700 feet per minute. The speed of traversing with 1½ tons is 300 feet per minute, and with no load 400 feet. The crane travelling motion is driven by a motor of 5 brake horse-power, running at 600 revolutions per minute, the speed of travel being 60 feet per minute. The motion for hinging up the arm is driven by a motor of 3 brake horse-power, running at 1,000 revolutions per minute, which lifts the arm in two minutes.

CHAPTER V.

SHEER LEGS.

FOR handling very heavy weights, and more particularly for such work as lifting masts, boilers, guns, etc., into or out of ships alongside a wharf, sheer legs are largely used. A perspective diagram of sheer legs is given in Fig. 43. The load hangs on the rope *a*, which is led over a sheave at the top of the legs to the hoisting drum *b*. The front legs *cc* are pivotted at the points *dd*, and are held in position by the back leg *e*. The bottom of the back leg can be moved along the line *fg* by means of a screw, so giving horizontal travel to the load. When the legs are in the outer position, as shown in full lines, the legs *cc* are in compression and the leg *e* in tension, while when traversed to the inner position, as shown dotted, all three legs are in compression. It will be noted that it is only possible to move loads along the line *AB*, no side motion being provided.

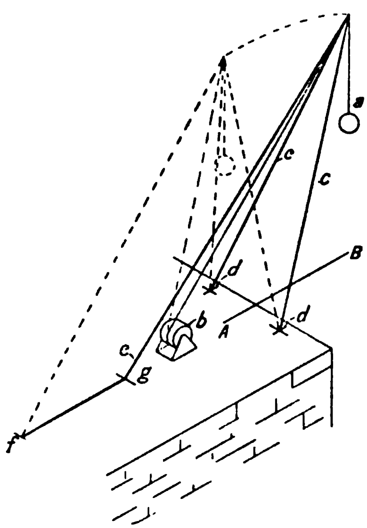


Fig. 43.

To ascertain the stresses when in the outer position (Fig. 44) lay off the vertical line *cd* in Fig. 45, representing to any convenient scale the load hanging from *A* plus half the weight of the three legs. From *d* draw *db* parallel to the lifting rope *AD*, and equal to the pull on this rope. As a multiple rope is always used for heavy sheers, the pull on this rope is only a fraction of the load on the hook. From *b* draw a line parallel to *AB*, and from *c* a line parallel to *AC*. The two lines meet in *a*, and *ab* gives the compression on the two front legs, and *ac* the tension on the back leg. To ascertain the load on each front leg, bisect *ab* in *e*, and draw a perpendicular at this point. From a draw a line parallel to AB' (Fig. 44). This line will meet the perpendicular in f, and af will be the load on each front leg. The horizontal force at c, due to the tension in the back leg, is given by ag, and the vertical force by cg.

The vertical load at the foot of each front leg equals 0·5 gh.

To obtain the stresses in the inner position, when all the legs are in compression, lay off the vertical line ab in Fig. 46, representing the load at A plus half the weight of the legs. From a draw the line ac parallel to the lifting rope, and equal to the pull on this rope. Also, from a draw lines parallel to AB and AC, and complete the parallelogram as shown. Produce these lines above a, and complete the parallelogram there also. Then the line de gives the com-

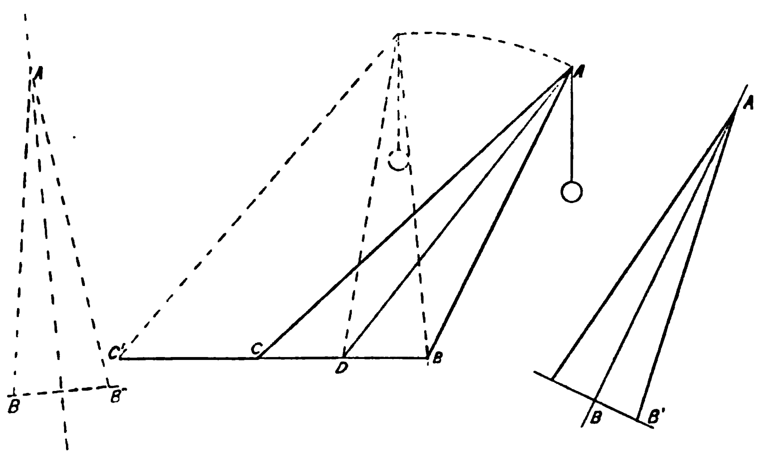


Fig. 44.

pression on the two front legs, and fg that on the back leg. To ascertain the load on each front leg draw a line from d parallel to AB'' (Fig. 44), bisect de in h, and erect a perpendicular at this point, then di is the compression in each front leg. To obtain the vertical load at the foot of each front leg draw a horizontal line from d, and a vertical from e meeting in k. Then the vertical load is 0·5 ek, and 0·5 dk the horizontal force at the foot of each front leg tending to push out the dock wall. To obtain the vertical and horizontal forces at the foot of the back leg draw a horizontal line from g and a vertical from f meeting in l, then gl equals the horizontal force, and fl the vertical. In addition to the stresses due to compression or tension the legs are subject to a bending moment due to their own weight. The maximum bending moment is at the middle of the leg, and

$$M_b = \frac{W_e L_e}{8} \text{cosin } a_e.$$

The nut at the foot of the back leg slides in guides, and the friction between the nut and guides affects the axial force on the screw. If the guides were frictionless this force would be equal to the horizontal force at the foot of the leg, as found at *ag* (Fig. 45) and *gl* (Fig. 46). When the horizontal force at the foot of the back leg is resisting

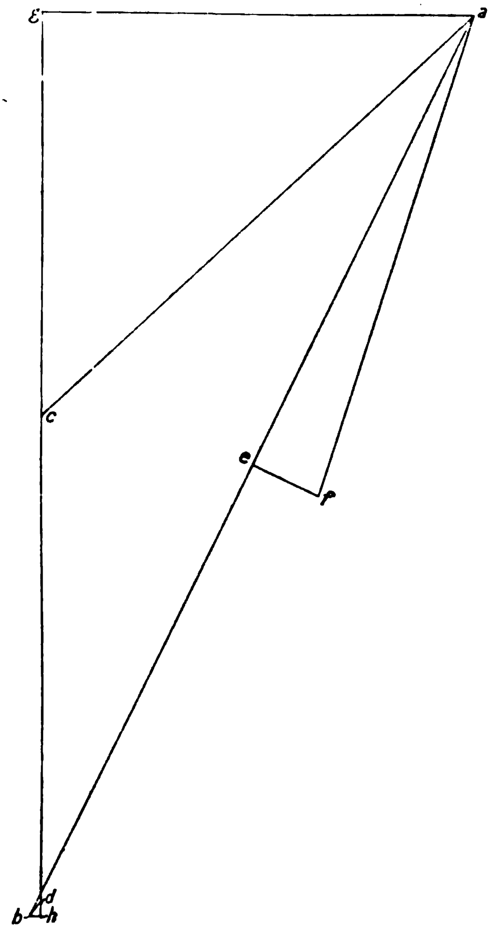


Fig. 45.

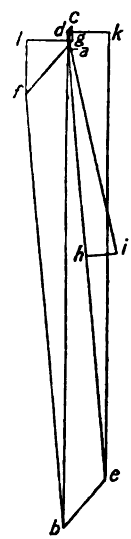


Fig. 46.

the motion of the nut, the screw has to exert an axial thrust P sufficient to overcome the horizontal force P_1 plus the force necessary to overcome the friction of the slides due to the vertical force P_2. In this case, therefore, $P = P_1 + \mu P_2$.

When the horizontal force tends to move the nut in the direction in which it is being driven, this force itself overcomes the slide friction, and exerts an axial thrust on the screw $= P = P_1 - \mu P_2$, which tends to revolve the screw in the direction in which it is being driven, so that the torque applied to it only requires to be sufficient to overcome the friction of the threads and thrust bearing.

The torque which requires to be applied to the screw to drive the leg along in the first case

$$T_5 = P\left\{\frac{2\mu\pi r_7 + p_1}{2\pi r_7 - \mu p_1} r_7 + \mu r_8\right\}.$$

In the second case—

$$T_5 = P\left\{\frac{2\mu\pi r_7 - p_1}{2\pi r_7 + \mu p_1} r_7 + \mu r_8\right\}.$$

In this latter case, if the coefficient of friction is sufficiently low or the lead of the screw sufficiently long, T_5 will have a negative sign showing that the horizontal force will cause the screw to revolve, so that instead of being driven by the motor it will require to be held in check by a brake.

The usual coefficient of friction, when running under full load, is 0·04 to 0·05, and when stopped it is about double this.

As the forces are considerable, it is convenient to take them in tons in the calculations, and the dimensions of the screw and thrust bearing in inches. T_5 is then in inch-tons, and the horse-power required to turn the screw $= \cdot 035\ T_5 S_2$. It is very important that means should be provided to prevent the driver accidentally traversing the nut hard up against the bearings at either end of the screw. Should this occur the nut may be screwed up so hard against the bearing that the full power which the motor can exert will be insufficient to unscrew it. To prevent this, it is usual to provide an arrangement of limit switches, such as that shown in Fig. 47. The arrangement consists of the solenoid main switch A, the two limit switches BB', and the contacts CC', etc., which are fitted on the controller barrel in addition to those required for controlling the motor current. The limit switches are placed a short distance from each end bearing, and are caught and opened by the nut as it travels towards the bearing, being closed by a spring when the nut travels in the opposite direction. The contacts of the solenoid main switch are placed between the main circuit terminals and the controller, so that the motor cannot start unless the solenoid switch is closed. The action of the arrangement is as follows :—To start the motor, the driver moves the controller handle from the off position to the step No. 1. As he does this the contact C closes the circuit through the coil of the solenoid, and the limit switch B, causing the plunger to pull up and close the motor circuit. As the handle passes to step 1 the contact C leaves the corresponding fixed contact, and C_1 now makes the circuit through the resistance R, thus reducing the current through the solenoid coil to an amount sufficient to hold the plunger up firmly, but not sufficient to pull it up should it from any cause, such as a momentary failure of current, fall to its lower position. The resistance R remains in circuit in all the other positions of the controller handle. The motor having started, the nut travels

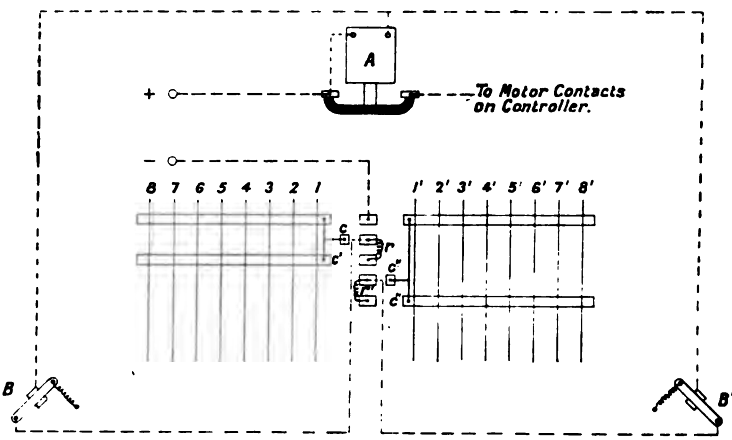
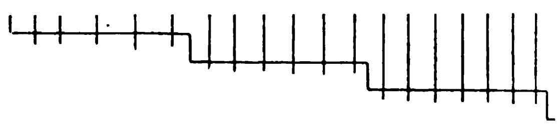


Fig. 47.—Limit Switches.

towards B, and should it be driven too far it pushes B open, so breaking the solenoid circuit, and causing the plunger to fall and break the motor circuit. The motor cannot now be driven further in the direction of B, but can be driven in the opposite direction by putting the controller handle to step No. 1′, etc., in which case the solenoid circuit is made through the limit switch B'. As the nut travels away from B the spring closes the switch, so that the motor can be again travelled towards B when required. This arrangement has the advantage that only quite small wires need be laid along the guides to the switches B and B' (which may also be quite small), as the current required by the solenoid is only from $\frac{1}{2}$ to 1 ampere. A further advantage is that it acts as a safety



To face p. 51.]

device to prevent the motor being injured by having full voltage switched on to it when it is standing with no resistance in circuit, for should current be cut off and then put on again with the controller on any running step, no current will pass to the motor, as the solenoid main switch will remain open, and can only be closed by bringing the controller handle to the off position, so introducing the full starting resistance into the motor circuit. The arrangement may be applied in any case where it is desired to prevent over winding, and has been used by the author with satisfactory results in a number of cases. It was originally devised by the Sturtevant (now the Adams Manufacturing) Company, Ltd.

150-Ton Electric Sheer Legs.—A set of electrically driven sheer legs by Messrs. Cowans, Sheldon & Company, of Carlisle, is shown in Fig. 48. For lifting the full load of 150 tons two separate winding gears are used operating together. Each gear is driven by a motor of 60 brake horse-power, running at 500 revolutions per minute, and when the two gears are running together they are capable of lifting the full load at a speed of 6 feet per minute. In each gear the rope is eight ply, so that the pull on each rope, allowing for weight of sheaves, etc., is 10 tons, and this is also the amount of direct pull off each barrel. The rope is 5 inches circumference, with a breaking strain of 88 tons. For a lift of 75 tons one motor only is used, the speed of lift being the same as for full load.

An auxiliary light lift gear is provided capable of lifting 30 tons at a speed of 20 feet per minute, this also being driven by a motor of 60 brake horse-power running at 500 revolutions per minute. The rope in this case is three ply, and the same size as on the main lift.

The inhaul winch is capable of pulling 5 tons off the barrel at a speed of 60 feet per minute, and is driven by a motor of 30 brake horse-power.

The horizontal screw gives a traverse of 73 feet 6 inches, and is 12-inch diameter and 2-inch pitch. It is driven by a motor of 60 brake horse-power, and the speed of traverse is 2 feet per minute.

The weight of each front leg is 60 tons, and that of the back leg 70 tons.

Alternative Designs of Sheer Legs. — In place of a horizontal screw, as in the example already shown, we may use an inclined screw, which comes directly in line with the back leg when the legs are in their extreme outward position. The screw may be either carried in a rigid inclined frame, in which case slides are necessary to guide the nut, as in the case of a horizontal screw, or the screw may be pivotted so as to follow the varying angularity

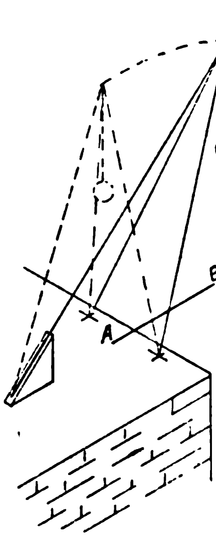


Fig. 49.

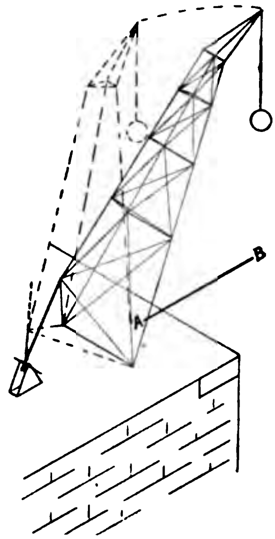


Fig. 50.

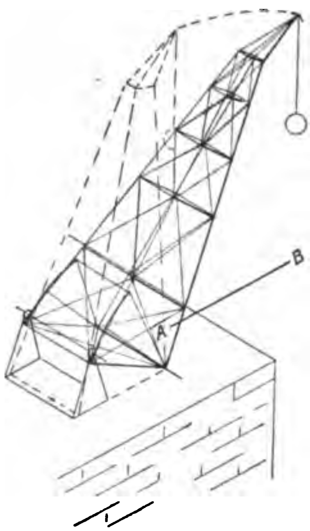


Fig. 51.

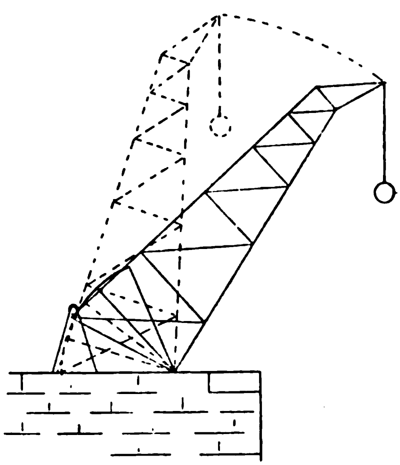


Fig. 52.

of the back leg, in which case no slides are necessary. The arrangement is shown in Fig. 49.

The advantage gained is in the shorter and less expensive screw. Where the screw is pivotted there is also a slight gain in the power required for traversing, owing to the absence of the slides with their accompanying friction. The general design of the sheer legs may be modified by bracing them together, as shown in Fig. 50. The structure being now a deep cantilever, the stresses due to its own weight are lessened, and a corresponding reduction in material is possible. Also, owing to the bracing, a further reduction of material is possible in the compression members. Thus, for the support of a given load, the braced structure is lighter than one in which the three separate legs are employed.

A stiffer and equally light construction may be obtained by using four members forming a cantilever of rectangular cross-section. Traversing of the load is in this case obtained by two screws connected to the two back legs and driven simultaneously, or the screws may be abolished altogether and movement obtained by attaching segments of spur wheels to the sides of the structure, these being driven by pinions, as shown in Figs. 51 and 52, which are a perspective view and a side elevation respectively. The frictional losses in the nut and thrust bearing are eliminated in the last design with a consequent gain in the efficiency of the traversing motion.

CHAPTER VI.

REVOLVING CANTILEVER CRANES.

A FORM of crane which is coming into use in docks and shipbuilding yards, and which in the future will probably to some extent take the place of sheer legs, is shown in Figs. 53 and 53*a*. In this a revolving structure forming two cantilevers is carried by a tower of any required height. The cantilevers are of unequal length, the longer one carrying a travelling carriage, which hoists, lowers, and traverses the load, while the shorter cantilever carries the winding gear and balance weight. As the superstructure can revolve through a complete circle, the crane can take up and deposit loads anywhere within an annular space bounded by the inner and outer radius of traverse of the carriage, so that it has a wider range of application than sheer legs, which can only take up and deposit loads along a straight line.

The design of the crane structure involves no special points. The superstructure consists of a couple of ordinary braced cantilevers. Stability is obtained by so proportioning the weights of the superstructure and balance weight that under all conditions of loading the centre of gravity comes within the diameter of the roller path. The four legs of the tower are never loaded equally, so it is necessary to determine the maximum load, which, under the extreme conditions of loading and wind pressure, may come upon one leg, and to make all the four legs of sufficient strength to carry this load. The spread of the legs should be such that with maximum wind pressure there is no upward pull on any of the foundation bolts.

Fig. 53 shows a 160-ton revolving cantilever crane made by Messrs. Cowans, Sheldon & Company, of Carlisle, for the Admiralty.

It is constructed for the following loads :—160 tons at 95 feet radius, 80 tons at 105 feet, and 50 tons at 128 feet, the test load being the very large one of 240 tons at 95 feet radius.

The cantilever girders are 220 feet long overall, 28 feet deep

at the centre, and are fixed 20 feet apart centre to centre. The bracing is of the Warren type, with vertical struts to support the upper flanges, so halving their unsupported length and reducing the bending moment in them. Each top flange has a double rail track, the gauge of rails being 2 feet 6 inches.

The roller path on which the superstructure revolves is 46 feet diameter, and has 72 forged steel rollers, 12 inches diameter by 9 inches long, each alternate one being connected to the centre casting by steel radial rods. The centre pin is of forged steel, 18 inches diameter.

The tower is 109 feet high above the level of the quay, and is 46 feet square to the centres of the main columns, which are themselves 2 feet 6 inches square.

The tops of the columns are connected by girders 18 feet deep, which serve to carry the cross girders which support the lower roller path.

The main lifting gear consists of two independent purchases of 80 tons each, this arrangement being adopted in order to enable long guns and shafts to be angled when passing through hatchways, etc. Each main block is carried by eight parts of rope winding one part on a barrel 6 feet 6 inches diameter by 10 feet long. The rope is $4\frac{3}{4}$ inches circumference, with a breaking strain of 79 tons, the factor of safety being thus 7·9.

Each of the main lift purchases is driven by a series-wound motor of 60 B.H.P., running at 550 revolutions per minute, the voltage of the circuit being 440. The two motors are worked off a three-drum controller, so arranged that the right or left-hand drums control the corresponding sets of gear in the ordinary way. When it is required to work the two sets of gear together, the two motors are controlled off the centre drum. An adjustable field rheostat is provided to ensure the two motors running at the same speed, even with unequal loads. Thus, if desired, a load can be adjusted to any desired angle, and then lifted at this angle throughout the full lift.

The main barrels are driven through three reductions of steel gear with cut teeth. An automatic solenoid brake is fitted on an extension of the motor shaft, and to control the lowering Weston screw type brakes with gun-metal and steel discs are fitted on the intermediate shaft. The speeds of lift are :—With the two gears, 160 tons at 6 feet per minute ; with one gear, 80 tons at 6 feet per minute : and loads up to 15 tons at 30 feet per minute. The light lift gear is of generally similar construction to the main lift gear, and is driven by a motor of 60 B.H.P., running at 550 revolutions per minute, with tramway type controller. This gear lifts 30 tons at 20 feet per minute, and loads up to 7½ tons at 75 feet per minute. The load is taken on four parts of rope, winding two parts on the barrel. The size of rope is 4¼ inches circumference, with a breaking strain of 65 tons, the factor of safety being 8·6.

The traversing gear consists of two barrels driven through steel-cut gearing by a motor of 30 B.H.P. running at 580 revolutions per minute. An automatic solenoid brake is provided on the intermediate shaft. The speeds of traversing are :—With 160 tons, 30 feet per minute ; with 80 tons, 60 feet per minute. The traversing ropes are 3 inches circumference, and 32 tons breaking strain.

The machinery for hoisting and traversing is carried in a steel framing on the back cantilever arm, and helps to balance the weight of the forward arm and load. To provide the additional weight necessary for balancing, a box is provided at the back end, filled with 160 tons of stone ballast.

The machinery is contained in a steel house provided with an overhead electric crane of 12 tons capacity, capable of lowering a barrel and wheel complete to the ground level.

The revolving motion is driven by a motor of 60 B.H.P. running at 550 revolutions per minute, and the motion is transmitted to the rack through worm, spur, and bevel gearing. A slipping device is fitted to relieve the gear or structure from undue shocks. The whole of the gear is machine-cut, with the exception of the rack. All the gear is of steel, with the exception of the worm wheel, which is phosphor-bronze. The speeds of slewing are :—160 tons, 1 revolution in eight minutes ; 80 tons, 1 revolution in six minutes ; no load, 1 revolution in six minutes. The revolving gear is arranged in duplicate, as shown in the illustration.

All the motors are series-wound, totally enclosed, and rated for a temperature rise of 70° F. on a full load run of one hour.

When tested the deflection of the cantilever with 160 tons at 95 feet radius was 4⅝ inches, and with 240 tons it was 7⅜ inches.

The weight of the revolving superstructure, exclusive of load, is 580 tons, and the weight of the entire crane, also exclusive of load, is 950 tons.

Fig. 53*a* shows a 100-ton revolving cantilever crane made by Messrs. Applebys, Ltd., for Earle's shipyard at Hull. The height of this crane from the ground to the top of the tower is 94 feet, and

to the top of the girder rail is 132 feet. The particulars of the loads are as follows :—100 tons at 70 feet radius, 50 tons at 100 feet, and 10 tons at 150 feet. The test load was 125 tons at 70 feet. The tower is 30 feet square to the centres of the columns, and the roller paths are 30 feet diameter.

The slewing gear is driven by a motor of 20 brake horse-power, and consists of two reductions of machine-cut steel spur gear and one of steel bevel gear, all mounted on a common bed plate. The bevel gear is connected by a vertical shaft to a cast-steel pinion, which engages with a steel rack on the bottom roller path. Change gears are provided to give two speeds of slewing, these being one revolution in six minutes with 100-ton loads, and one revolution in three minutes with 50-ton loads.

The main hoisting gear is placed at the end of the short arm. The rope passes from the winding barrel over sheaves fixed on the travelling trolley to the snatch block, and thence to the end of the long arm, where it is anchored. The main hoist is driven by a 60 brake horse-power series motor. The barrel is 5 feet 6 inches diameter, with a turned spiral groove to take the whole of the rope at a single lap. The weight of the barrel and its spur wheel is 15 tons. The speed of lift with 100 tons is 5 feet per minute, and 10 feet with 50 tons. The auxiliary hoisting gear has a motor of 30 B.H.P., and hoists a load of 10 tons at 30 feet per minute. The traversing gear is driven by a motor of 30 B.H.P., and the speed of traverse with full load is 36 feet per minute.

CHAPTER VII.

CABLEWAYS.

CABLEWAYS for the conveyance of materials from point to point are of very ancient date, but those which, in addition to conveying loads, can also hoist and lower them at any point, and which are generally known as Blondins, are a comparatively recent invention. It is only these latter which can properly be included in a work on cranes; they may, in fact, be regarded as overhead cranes of large span, in which the cross girders are replaced by a steel wire rope.

A form of Blondin which is very extensively used is shown in Fig. 54. The load hangs on the rope *a* which passes over guide

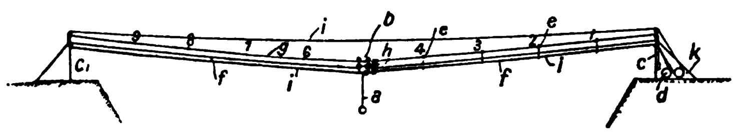


Fig. 54.—Cableway.

pulleys on the carriage *b* and at the top of the tower *c* and leads to the hoisting drum *d*. This rope is supported by the carriers *e*, which hang on the main rope *f* on which the carriage runs. The object of these carriers is to divide the hoisting rope into short spans when the carriage is distant from the head tower, so as to ensure the descent of the empty hook when the rope is paid out by the hoisting drum. If these carriers were not used the tension in the span between the carriage and the tower, due to the weight of the rope itself, would be so great that on paying out the rope the hook would not descend unless it were heavily loaded. The carriers are extended above the main rope, and have a vertical slot through which the button rope *g* passes. The width of the slot is different in each carrier, being narrowest in the one nearest the head tower, and being slightly wider in each succeeding carrier. The travelling carriage is provided with a horn, *h*, on which the carriers ride. The purpose of the button rope is to ensure the carriers taking positions which will divide the hoisting rope into the desired spans. This is accomplished by attaching to the rope *g* at the required positions, buttons of diameters varied to engage with the slots of successive carriers. Thus button No. 1 is of a diameter which will pass through the slots of all the carriers except No. 1 carrier, button No. 2 will pass through all the carriers except No. 2, and so on. As the carriage runs away from the head tower it consequently leaves a carrier at each button and picks them up again as it returns. The rope *i* for hauling the carriage backwards and forwards is fastened to one end of the carriage, passes round guide pulleys on the tail tower *c′*, and round a guide pulley on the head tower *c* to a capstan drum, *k*. From this drum it passes round another guide pulley on the head tower, and is fastened to the other end of the carriage. Thus the hauling rope holds the carriage positively in position at all times. When loads are being hoisted or lowered, the capstan drum is held by a brake and the hoisting drum is driven. To travel the carriage both drums are driven, being clutched together so that they run at the same speed. Then, as the carriage travels along the hoisting rope is hauled in or paid out at the same speed as the hauling rope, so that the load remains suspended at a constant distance below the main rope. This form of Cableway is generally associated with the name of Mr. Spencer Miller, who has taken out numerous patents in the United States, and to whom its success is largely due.

In some cases the towers, instead of being secured to the ground, are mounted on travelling carriages. This is for the purpose of moving the cableway from point to point, and not for travelling loads, although it would be quite feasible to travel the cableway bodily under load, provided suitable steps were taken to ensure the towers travelling at uniform speed. The arrangement would then be analogous to that of a Goliath crane.

In another form of Blondin, a motor is mounted on the carriage and two sets of gearing are provided, to either of which the motor can be clutched. One set of gearing drives the hoisting drum, and the other set drives the travelling wheels, so that the machinery is all self-contained on the carriage, on which the driver rides.

In the latter case, as the tractive effort passes through the travelling wheels, the wear of these wheels and of the rope is somewhat heavy, and to avoid this the arrangement has in some cases been adopted of hauling the carriage backwards and forwards by a rope driven by a separate capstan, while retaining the motor and hoisting drum on the carriage.

Both of these arrangements have the advantage of dispensing with the button rope and carriers first described, which are sometimes a source of trouble, especially in cableways of considerable span.

In the design of cableways, the most important points to be determined are the tensions in the ropes and the vertical loads on the towers.

The following calculations are based on the assumption that the curve in which the rope hangs is parabolic. This is not strictly correct, as the curve of the rope is really a catenary. Treating it as a parabola, however, greatly simplifies the calculations, while the error introduced is so trifling as to be negligible.

When a rope is hanging freely between two points of support,

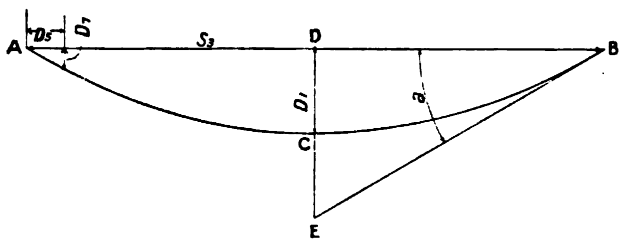


Fig. 55.

see Fig. 55, the tension due to its own weight is—

$$T_6 = \frac{W_r S_3}{8 D_1}$$

at the point **C**, where the rope is horizontal. The maximum tension occurs at the point of attachment to the support, and is—

$$T_7 = T_6 \text{ sec. } a.$$

The angle may be obtained graphically by laying out the tangent to the curve of the rope at the points **A** or **B**. At the centre of the span draw the vertical line **DE** equal to twice the dip. Lines joining **AE** or **EB** will be tangents to the curve at **A** or **B**. The angle may also be found by the equation—

$$\text{Tan } a = \frac{4 D_1}{S_3}.$$

When a weight is suspended on a cord, supported at two points, if the weight of the cord is negligible it takes the shape shown in Fig. 56, and the tension on the cord is—

$$T_6 = \frac{WS_3}{4D_2}.$$

Taking into account both the weight of the rope and the load

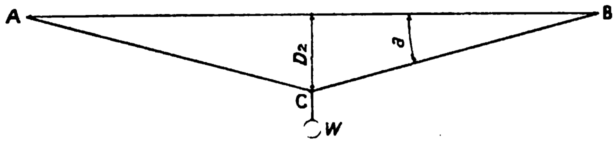


Fig. 56.

suspended on it, see Fig. 57, the tension is greatest when the load is at the centre of the span, and is—

$$T_6 = \frac{S_3}{D_3}\left(\frac{W_r}{8} + \frac{W}{4}\right)$$

at the point C. The maximum tension on the rope is, as before, at the point of attachment, and is—

$$T_7 = \frac{S_3}{D_3}\left(\frac{W_r}{8} + \frac{W}{4}\right) \text{ sec. } a.$$

The angle a may, in this case, be obtained approximately by

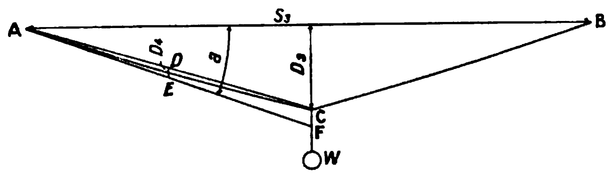


Fig. 57.

calculating the dip for the half-span, thus—

$$D_4 = \frac{W_r S_3}{32T_6}.$$

It should be mentioned here that this equation is really only correct for a rope, the ends of which are at the same level. In the cases with which we are dealing the inclination of the rope is, however, so small that it need not be taken into account in the calculation.

Draw the vertical line **DE** bisecting the half-span, touching the line **AC**, and having a length equal to twice D_4.

A line drawn from **A** to **E** and produced will cut the vertical centre line of the span at **F**, and will be a tangent to the curve of the rope at **A**. The distance **CF** will equal $4D_4$, and, therefore—

$$\text{Tan } a = \frac{D_1 + 4D_4}{\cdot 5S_3}.*$$

The tensile strength of a rope, and its weight for a given length, being both proportional to the square of the diameter, we may, provided we know the relations between the diameter and cross-section of the rope, calculate the diameter of rope to carry any required load with any required span and dip. The breaking strength of a rope, in tons, may be expressed as ad^2f, in which a includes the relation of the area of a circle to its diameter and the percentage of interstices in the rope, and f is the breaking strain in tons per square inch of the steel. The weight of the rope in lbs. per foot is βd^2, in which β includes the relation of the area of a circle to its diameter, the percentage of interstices in the rope, and the specific gravity of the steel.

For ropes from 1 inch to 5 inches circumference having six strands, with 24 wires per strand, a averages 0·32 and β 1·42, while for ropes from 2 to 12 inches circumference having six strands, with 37 wires per strand, a averages 0·345 and β 1·59.†

The safe working stress on the rope will be $\frac{ad^2f}{s}$, the weight of the rope in lbs. per linear foot will be βd^2, and taking the length of the rope as being equal to the span, which we may do in the present

* A more exact method, which allows for the inclination of the rope, is to assume that the portion of the rope **AC** forms part of the rope of an imaginary cableway of greater span and greater dip than in Fig. 57, and that in this imaginary cableway the rope hangs freely in a parabolic curve as in Fig. 55. Calling the half-span of the imaginary cableway S_i and the dip at the centre D_i,

$$S_i = \frac{\frac{T_6 D_3}{\cdot 5w} + (\cdot 5\, S_3)^2}{2S_3}$$

and

$$D_i = \frac{D_3}{1 - \frac{(S_i - \cdot 5S_3)^2}{S_i^2}}$$

and the tangent of the angle a at the point of support in Fig. 57 $= \frac{2D_i}{S_i}$.

† For 6/19 $a = 0{\cdot}45$ and $\beta = 1{\cdot}58$.

case without any appreciable error, T_6 will equal $\frac{\alpha d^2 f}{8}$ and W_r in tons $= \frac{\beta d^2 S_3}{2{,}240}$.

Then,

$$\frac{\alpha d^2 f}{8} = \frac{S_3}{D_3}\left(\frac{\beta d^2 S_3}{8 \times 2{,}240} + \frac{W}{4}\right),$$

and

$$d = \sqrt{\frac{WS_3}{4D_3\left(\frac{\alpha f}{8} - \frac{\beta S_3^2}{17{,}920 D_3}\right)}}.$$

As an example, we will take a cableway of 1,520 feet span, with a dip equal to $\frac{1}{20}$ the span, which is a value frequently adopted. The dip is then 76 feet. Total load, 10 tons. Cable, 7 strands, 37 wires per strand. $\alpha = 0{\cdot}345$. $\beta = 1{\cdot}59$. Breaking strength of cable, 110 tons per square inch. Factor of safety, 4.

Diameter of rope—

$$d = \sqrt{\frac{10 \times 1{,}520}{4 \times 76 \times \left(\frac{\cdot 345 \times 110}{4} - \frac{1{\cdot}59 \times 1{,}520 \times 1{,}520}{1{,}7920 \times 76}\right)}} = 2{\cdot}71.$$

Weight of rope between points of support—

$$W_r = \frac{1{\cdot}59 \times 2{\cdot}71 \times 2{\cdot}71 \times 1{,}520}{2{,}240} = 7{\cdot}95 \text{ tons.}$$

Tension in rope at centre of span—

$$T_6 = \frac{1{,}520}{76} \times \left(\frac{10}{4} + \frac{7{\cdot}95}{8}\right) = 70 \text{ tons.}$$

Tension in rope per square inch—

$$T_8 = \frac{70}{\cdot 345 \times 2{\cdot}71 \times 2{\cdot}71} = 27{\cdot}5 \text{ tons.}$$

Angle of rope at point of support—

$$D_4 = \frac{1{,}520 \times 7{\cdot}95}{32 \times 70} = 5{\cdot}4 \text{ feet.}$$

$$\text{Tan } a = \frac{76 + (4 \times 5{\cdot}4)}{760} = 0{\cdot}1285 = \tan 7^\circ\ 19'.$$

Tension in rope at point of support—

$$T_7 = 70 \times \sec. 7^\circ\ 19' = 70{\cdot}56 \text{ tons.}$$

In the formula for the diameter of the rope its length was taken as being equal to the span, the error so introduced being practically negligible.

In erecting the cableway, however, it is necessary to know the dip of the rope when unloaded, and to obtain this the length of the rope when loaded must be ascertained.

Referring to Fig. 57, the length of the rope may be taken as equal to the sum of the two straight lines, **AC** and **CB**. The true length of the rope is, of course, the sum of the two curved lines joining **AC** and **CB**. The difference, however, in length between the straight line and the curve, even in the case of such a long span as 1,520 feet, amounts to less than 1 inch, and so need not be considered. The length of the rope when fully loaded at the centre of the span is then—

$$L_r = 2\sqrt{(\cdot 5S_3)^2 + D_3^2}.$$

The length of an unloaded rope hanging freely between two supports being approximately

$$= 2\sqrt{(\cdot 5S_3)^2 + 1\cdot 333\, D_1^2},^*$$

The dip equals—

$$D_1 = \sqrt{\cdot 1875(L_r^2 - S_3^2)}.$$

In calculating the dip of the unloaded rope we cannot neglect the stretch of the rope. Stretching is due to two causes. Firstly, when new there is a certain amount of looseness in the construction of the rope, and during the first few days of working this tightens up, so leading to a permanent increase in the length. This does not require to be taken into account in the calculations, but renders necessary the provision of adjusting gear to take up the slack as the

* The true length of rope in Fig. 55

$$= \frac{(\cdot 5S_3)^2}{2D_1}\left\{\tan a \,.\, \sec a + \text{hyp log}\,(\tan a + \sec a)\right\}$$

and in Fig. 57 it is

$$= \frac{S_t^2}{2D_t}\left\{\tan a \,.\, \sec a + \text{hyp log}\,(\tan a + \sec a) - \tan a_t \,.\, \sec a_t - \text{hyp log}\,(\tan a_t + \sec a_t)\right\}$$

$$\text{Tan } a_t = 2\,\frac{D_t - D_3}{S - \cdot 5S_3}$$

$$\text{Sec } a_t = \sqrt{1 + \tan^2 a_t}.$$

See footnote, p. 62.

rope stretches. Secondly, there is the stretch due to the elasticity of the rope. This is always present, so that the rope continually changes in length with every change of load. The amount of stretch is made up of two values—firstly, the elongation due to the direct elasticity of the material, which is the same as it would be if the rope were composed of a bundle of parallel wires; and, secondly, the elongation of the rope regarded as a helical spring. The latter varies with the structure of the rope, and is best found by experiment. The general result of experiments on the elasticity of ropes shows that the elastic stretch of a rope is about three times what it would be if made up of straight parallel wires. Thus, taking E as the direct modulus of elasticity of the steel, the modulus of elasticity of the rope $E_r = \cdot 36\,E$. This may be taken as a general value, but whenever possible it is advisable to ascertain by experiment the true value for the particular rope or class of rope which is being used.

The length of rope when fully loaded having been determined by the formula already given, the length when subject to no stress whatever, as it would be, for instance, when lying on the ground =

$$L_u = \frac{L_r}{1 + \dfrac{T_6}{E_r a d^2}}.$$

In the example with which we are dealing, taking E as 14,000 tons per square inch, $E_r = 5{,}040$, $L_r = 1{,}527\cdot 6$, and—

$$L_u = \frac{1{,}527\cdot 6}{1 + \dfrac{70}{5{,}040 \times \cdot 345 \times 2\cdot 71 \times 2\cdot 71}} = 1{,}519 \text{ feet.}$$

When the load is removed from the rope (when in position) it shortens and T_6 decreases; at the same time the dip lessens, and this tends to increase T_6. There is a certain dip at which these values come into balance, and this is the point at which the rope comes to rest when the load is removed. This point may be obtained by plotting two curves, one of T_6 and length of rope, and the other T_6 and D_1. The intersection of these curves gives the point required. In the present case the values so found are—dip 42 feet 6 inches, tension in rope 35·5 tons, and length of rope 1,523·2 feet.

The rope varies in length with change of temperature, the variation being 0·0000069 of its length per degree Fahrenheit. Taking the range met with in this country between the extremes of winter and summer as about 80°, the rope which we have taken as an example would vary 0·8 foot in length when hanging freely, this giving a variation in the dip of about 5 feet, and in the tension of rather over 3½ tons. It is advisable to take observations from time to time of the dip, and adjust the rope, letting it out in winter and taking it in in summer, so as to keep the dip fairly constant.

In the calculations given so far, no allowance has been made for wind pressure in increasing the stresses on the rope.

The force exerted by a given wind pressure upon a cylinder is 0·6 of the force, which it would exert on a plane surface of the same projected area. The pressure per linear foot due to wind pressure on a rope is then $P_3 = \cdot 05\ p_2 d$. The wind pressure acts horizontally, while the weight of the rope acts vertically, and the two combined form a resultant pressure on the rope in tons—

$$P_4 = \frac{\sqrt{(w_r^2 + P_3^2)}S_3}{2{,}240}.$$

The tension in the rope at the centre of the span now becomes—

$$T_6 = \frac{S_3}{D_3}\left(\frac{P_4}{8} + \frac{W}{4}\right).$$

As P_3 varies with the diameter of the rope, and w_r with the square of the diameter, the relative effect of wind pressure decreases as the size of the rope increases. In the case of a rope 2·71 inches diameter, weighing 11·67 lbs. per foot, a wind pressure of 40 lbs. per square foot produces a pressure per linear foot of 5·42 lbs., and the resultant pressure per linear foot is 12·9 lbs. In the case of the cableway of 1,520 foot-span and 10-ton load given previously this makes the total resultant load on the rope (P_4) 8·74 tons, so that the tension now becomes 71·85 tons instead of 70, the increase of stress in this case being very trifling.

In the case of smaller ropes the effect of wind pressure is more pronounced.

The hoisting rope being divided up into short spans by the carriers, it is necessary to calculate the tension in the rope due to its own weight in one of these spans, in order to ascertain what weight to attach to the hook to ensure its running down without load, when the rope is paid out. The size of the rope is determined simply by the weight which it has to lift, as in the case of the hoisting ropes of other forms of cranes.

The button rope is fastened to one tower, passes under a pulley on the top of the carriage, over a pulley at the top of the other tower, and has a weight hung on the end to keep it tight. Being held down by the carriage, its dip always coincides with that of the main rope.

The actual pull on the rope due to the buttons engaging with the carriers cannot be calculated, but experience shows that if the rope is from $\frac{5}{8}$ to $\frac{3}{4}$ inch diameter, according to the span of the cableway, it will give satisfactory results. In order that it may always keep tight the tension on it should be such that if detached from the carriage its dip would be slightly less than that of the main rope when unloaded. This tension is—

$$T_6 = \frac{\beta d^2 S_3^{\,2}}{16{,}125\, D_1},$$

and represents the amount of weight to be hung on the rope.

As the hoisting rope passes round a pulley on the carriage it always, when loaded, tends to pull the carriage towards the head tower with a force equal to the tension in the rope, and the pull is balanced by an equal tension in the hauling rope on the opposite end of the carriage. The greatest tension occurs on the hauling rope when it is pulling the loaded carriage up the incline towards the tail tower. The tension then consists of that required to balance the tension in the hoisting rope plus the tractive force required to pull the carriage along the main rope plus the force required to pull the carriage up the incline. It should be noted that, although the tension in the hoisting rope is transmitted to the hauling rope, it does not correspondingly increase the power required for hauling, as when the drums are clutched together for hauling the one tension balances the other through the drums. The tractive force may be taken at 150 lbs. per ton. In order to ascertain the force required to raise the carriage up the incline it is necessary to determine the angle of the main rope when the carriage is in its nearest position to the tower.

If the main rope were perfectly flexible and free from weight and elasticity the path of the loaded carriage, as it travelled along, would be an ellipse, the major axis of which would be equal to L_r, and the minor axis twice D_3. This, however, is not the case, owing to the fact that as the carriage approaches either tower the tension in the main rope diminishes, and on account of its elasticity it becomes shorter.

Taking an ellipse, the major axis of which is equal to the length of the rope when unloaded, the minor axis equal to twice D_1, and foci corresponding to the points of attachment of the rope to its supports, the angles of rope obtained for positions of the carriage near the towers agree, very nearly, with those found in practice, being generally a little on the safe side. The angles which the main rope makes with the horizon at each end of the carriage may be calculated thus (see Fig. 63)—

$$\text{Tan } a_5 = \frac{D_9}{D_5} \text{ and tan } a_7 = \frac{D_9}{S_3 - D_5}$$

$$D_9 = \frac{2D_1}{L_4} \sqrt{\{L_4(D_5 + x)\} - (D_5 + x)^2}$$

$$x = \cdot 5(L_4 - S_3).$$

The curvature of the rope between the carriage and the nearer tower need not be taken into account. To allow for the curvature between the carriage and the further tower, take W_8 as the weight of the rope in this part of the span, and D_4 as the dip at the centre of this distance, then—

$$D_4 = \frac{W_8(S_3 - D_5)}{8T_6}$$

$$\text{Tan } a_8 = \frac{4D_4}{S_3 - D_5}$$

$$a_6 = a_7 - a_8.$$

The angles having been calculated, the tension on the hauling rope required to pull the carriage up the incline may be obtained graphically, as shown in Fig. 58. Draw the vertical line *ab* representing the weight of the carriage plus load. From *b* draw lines at the respective angles a_5 and a_6, and complete the parallelogram, the lines meeting in *c* and *d*. Then the pull of the hauling rope is *cb-bd*.

Owing to the flatness of the curve, the angles of the main rope cannot be obtained accurately by graphic construction. When erecting the ropes, the dip of the hauling rope is made the same as that of the main rope when unloaded. Thus, when there is no load on the carriage the ropes all have the same dip, and hang in parallel curves.

When the carriage is at the centre of the span, and is loaded, the dip of the main rope increases, and that of the lower half of the hauling rope increases to the same extent, so that this rope is drawn tighter, and the dip in the upper half of the rope is decreased. The stress in this rope due to the decreased dip in the upper half is then—

$$f = \frac{\beta S_3^2}{17{,}920\,\alpha D_6} \text{ per square inch.}$$

If this is found to be excessive, the main rope should be altered

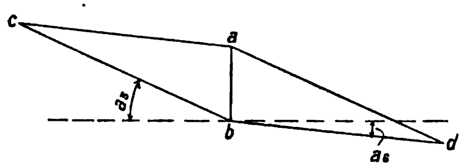


Fig. 58.

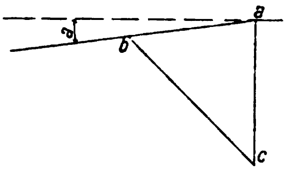


Fig. 59.

so as to lessen the variation in the dip between full load and no load.

The load on the towers is greatest when the loaded carriage is at the centre of the span, and may be found graphically, as in Fig. 59. Draw the line *ab*, the angle *a* of which is the same as that of the main rope at the point of attachment to the tower, and the length of which is equal to the sum of the tensions in the different ropes. From *a* draw a vertical line, and from *b* a line parallel to the back stay rope of the tower meeting in *c*. The vertical load on the tower is then represented by *ac*, and the tension in the stay by *bc*. The top of the tower is usually stayed sideways against wind pressure. The effect of the latter is two-fold, (1) it increases the vertical load on the tower owing to the pull of the stay, and (2)

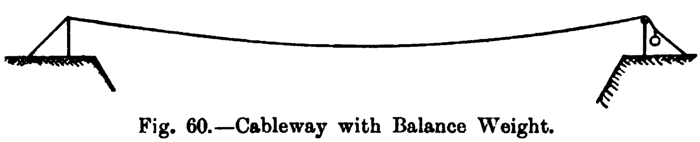


Fig. 60.—Cableway with Balance Weight.

it sets up a bending moment, the maximum of which is at a point half-way up the tower.

In some designs of cableway the main rope, instead of being fixed to each tower, is fixed to one tower and led over a pulley at the top of the other, a weight being hung on the end so as to maintain a constant tension. An arrangement which has been in use for many years in granite quarries in Scotland is shown in Fig. 60.

The main rope is fixed to the top of one tower, carried over a pulley on the other, and anchored to the ground, a weight being hung on the rope between the pulley and the ground.

In another form, patented by Brothers, and shown in Fig. 61, the rope is fastened to both towers. One tower is fixed, while the other rocks on a pin, and has a balance weight hung on it.

In designing cableways of this class it is not necessary to take into account the effects of elasticity or change of temperature, as the variation in length of rope due to these causes is automatically taken up by the balance weight.

It is preferable to arrange the design so that a constant tension is maintained on the rope. The diameter of the rope and the tension in it with a dip of, say, $\frac{1}{20}$ the span, and with the fully loaded carriage at the centre, are calculated, and the balance weight is arranged to give this tension on the rope.

As the tension does not now diminish as the carriage approaches the towers, the angle of inclination of the rope when the carriage is near a tower is less than in cableways having a fixed rope, and the

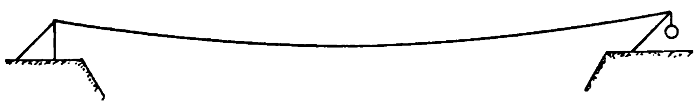


Fig. 61.—Brothers Cableway.

power required to travel the carriage is correspondingly reduced. The angles of the rope may be ascertained approximately as follows: —Assuming as before that the rope when unloaded hangs in a parabolic curve, the dip at any distance D_5 from the nearest tower (see Fig. 55) =

$$D_7 = D_1 - \frac{D_1\{(\cdot 5S_3) - D_5\}^2}{(\cdot 5S_3)^2}$$

$$D_1 = \frac{\beta d^2 S_3^2}{1{,}792\,T_6}.$$

Regarding the main rope as a cord, the dip due to a load hung near to one tower (see Fig. 62)—

$$D_8 = (S_3 - D_5)\tan a_6.$$

$$\text{Sin } a_6 = \frac{WD_5}{S_3 T_6}.$$

$$\text{Tan } a_5 = \frac{D_8}{D_5}.$$

The total dip due to the load and weight of rope (Fig. 63)—

$$D_9 = D_7 + D_8.$$

The curve in that portion of the rope between the carriage and the nearest tower may be neglected, so that

$$\text{Tan } a_5 = \frac{D_9}{D_5}.$$

In calculating a_6, the curve due to the weight of the rope between the carriage and the further tower should be taken into consideration.

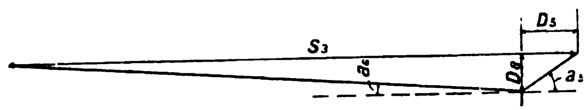


Fig. 62.

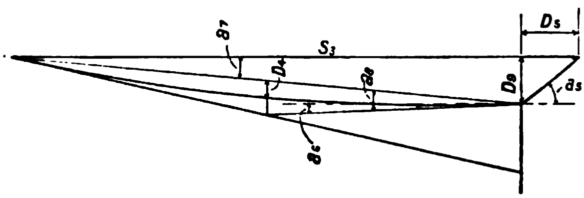


Fig. 63.

Taking W_s as the weight of the main rope between the carriage and the further tower, and D_4 as the dip at the centre of this distance

$$D_4 = \frac{W_s(S_3 - D_5)}{8T_6}.$$

$$\text{Tan } a_7 = \frac{D_9}{S_3 - D_5}.$$

$$\text{Tan } a_8 = \frac{4D_4}{S_3 - D_5}.$$

$$a_6 = a_7 - a_8.$$

The angles having been found, the force required to pull the carriage up the incline may be found graphically, as in Fig. 58.

As has already been stated, the calculations in this chapter are of an approximate character, but the results obtained with them are sufficiently accurate for the practical purposes of cableway construction.

10-Ton Cableway of the Fixed Rope Type.*—A general view of this cableway is shown in the Frontispiece. It was constructed by the Cleveland Bridge Company, Limited, of Darlington, for use in building the King Edward VII. Bridge at Newcastle-on-Tyne, for which they were the contractors. The general arrangement, machinery, and electrical equipment were designed by the author, while the towers and anchorages were designed by Mr. Max Am Ende, M.Inst.C.E.

The span from centre to centre of the towers was 1,520 feet, and loads up to 10 tons could be handled. The towers (see Fig. 64) were of light steel construction, rocking on pivots carried on concrete foundations. The height of the towers from the centre of the pivots to the centre of the main rope was 96 feet, and the vertical load on each tower with a load of 10 tons at the centre of the span was about 100 tons.

The back stay of each tower consisted of sixteen ropes, $1\frac{1}{4}$ inches diameter, having six strands, 19 wires per strand, the breaking strain of each rope being 61 tons. These ropes were attached to 32 long screws, $1\frac{3}{8}$ inches diameter, by means of which the length of the stays could be varied so as to rock the towers and adjust the dip of the main rope. On the Newcastle side the tower was mounted on the masonry of the bridge, to which the back stays were also anchored. The tower on the Gateshead side stood on a concrete foundation, and the back stays were anchored to a block of re-inforced concrete weighing 450 tons.

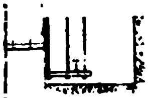
Counter anchorages were provided to prevent the towers falling in case of failure of the main rope. These consisted of four ropes to each tower, the ropes having six strands, 19 wires per strand. The diameter of the ropes was 0·95 inch, and the breaking strain 43·5 tons.

The towers were sufficiently extended sideways at the bottom to withstand wind pressure, so that no side stays were required.

The main rope was $3\frac{1}{8}$ inches diameter, having six strands, 37 wires per strand, the diameter of each wire being 0·143 inch. The breaking strain of the rope was 360 tons, and its weight $10\frac{1}{2}$ tons. It was made by Messrs. T. & W. Smith, Ltd., Newcastle-on-Tyne. The tension on the rope with the fully loaded carriage at the centre of the span was 90 tons, so giving a factor of safety of 4.

The hoisting rope was 0·71 inch diameter, six strands, 19 wires per strand, with a breaking strain of $15\frac{3}{4}$ tons. The hauling rope

* See *Proc. Inst. C.E.*, vol. clxxiv., p. 181.



was 0·875 inch diameter, six strands, 19 wires per strand, breaking strain 23·6 tons. The button rope was 0·78 inch diameter, six strands, 7 wires per strand, breaking strain 19$\frac{3}{4}$ tons. It was provided with fourteen buttons, so as to divide the hoisting rope into spans of about 100 feet. Owing to the large number of buttons, and in order to avoid getting to sizes which would be unwieldy,

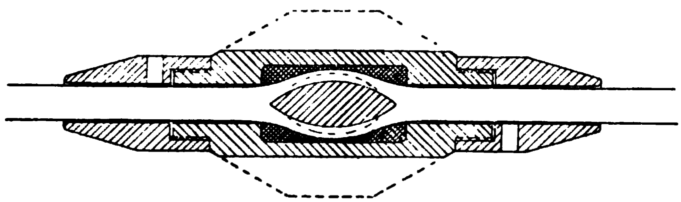


Fig. 65.—Section of Button.

the increase of size from button to button was only $\frac{1}{8}$ inch. Thus, No. 1 button nearest to the head tower at Gateshead was 2$\frac{1}{4}$ inches diameter, while No. 14 was 3$\frac{7}{8}$ inches. Owing to the small amount of wear thus allowed on the buttons, there was a possibility of their being worn out before the button rope required renewing, and the author, therefore, designed a form of button, the wearing part of which could be renewed without disturbing the button rope. One of these buttons is shown in Fig. 65. A brass spreader was first

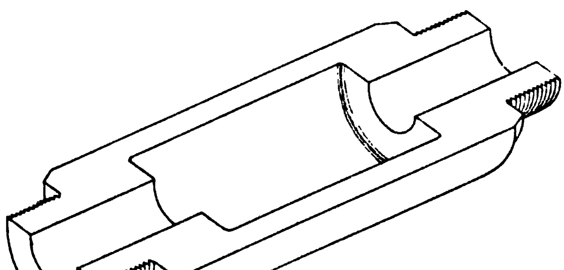


Fig. 66.—Isometrical Perspective View of Half Button.

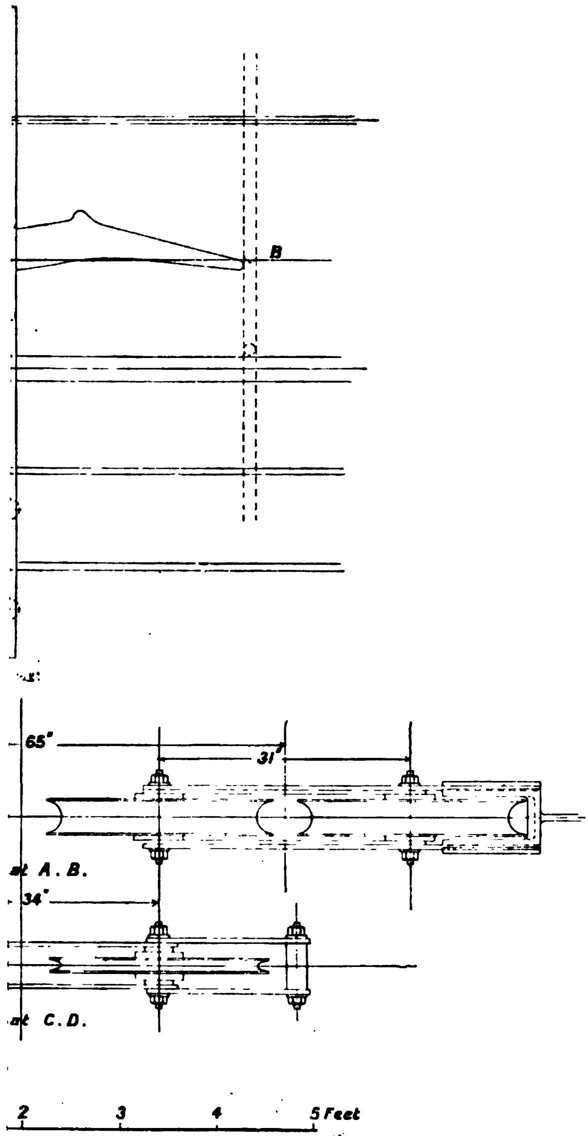
inserted in the rope, and a cylindrical white metal button was then cast on it. An aluminium-bronze button split longitudinally and threaded at the ends was then fitted on to the white metal button, and held together by two steel nuts. An isometrical perspective view of one of these half-buttons is shown in Fig. 66. When a button requires renewal the two nuts are unscrewed, the worn aluminium-bronze pieces are removed and replaced by new ones, and the nuts screwed on again. The slots in the hoisting rope carriers had sides made of plain pieces of tool steel, which could be easily renewed. The carriage (Fig. 67) consisted of a light framing of mild steel, having four travelling wheels, 2 feet diameter, the wheels being mounted in pairs on equalising levers. Two wheels, 12 inches diameter, were provided for holding the button rope in position. The sheaves for the hoisting rope were 2 feet diameter. The sheaves and travelling wheels were fitted with Macfarlane's patent self-oiling arrangement, in which the bosses are made hollow to contain oil; the gun-metal bushes have dovetail grooves running along the bore, these grooves being filled with leather strips and oil holes being drilled through the bushes behind the leather. Thus, the leather strips are kept constantly saturated with oil, which wipes off on to the pins, and keeps them in good condition. The carriage was provided with a platform on which a man could ride for the purpose of inspecting the ropes. The weight of the carriage complete was 35 cwts., so that with 10 tons on the hook the load per wheel was 2·94 tons.

For lubricating the main rope an ingenious arrangement was devised by Mr. Storr, the Cleveland Bridge Company's electrician on the contract. Briefly, this consisted of a cylinder surrounding the main rope. The ends of this had a rifled bore to suit the spiral of the rope, and the middle part of the cylinder was enlarged and contained oil. The arrangement was attached, through a swivel, to the carriage by a piece of twine, and served three purposes. As it moved along the front portion of the cylinder scraped the dirt off the rope so acting as a cleaner, the middle portion as it revolved supplied oil to the rope, superfluous oil being scraped off the rope back into the cylinder, while if a broken wire in the rope were encountered the twine broke and left the cylinder standing to mark the position of the damage.

The hoisting and hauling ropes were carried down the head tower, and led round guide pulleys to the winch, which was placed in a cabin about 100 feet distant. This cabin had two storeys, the winch being placed on the ground floor, while the controller and levers were placed on the upper floor, which had large windows to enable the driver to have a good view of the work.

The winch (Fig. 68) was made to the author's design by Messrs. Ransomes & Rapier, Ltd., of Ipswich.

The two drums were 5 feet diameter. The hoisting drum was


B
65"
31"
at A. B.
34"
at C. D.
2
3
4
5 Feet

Vin

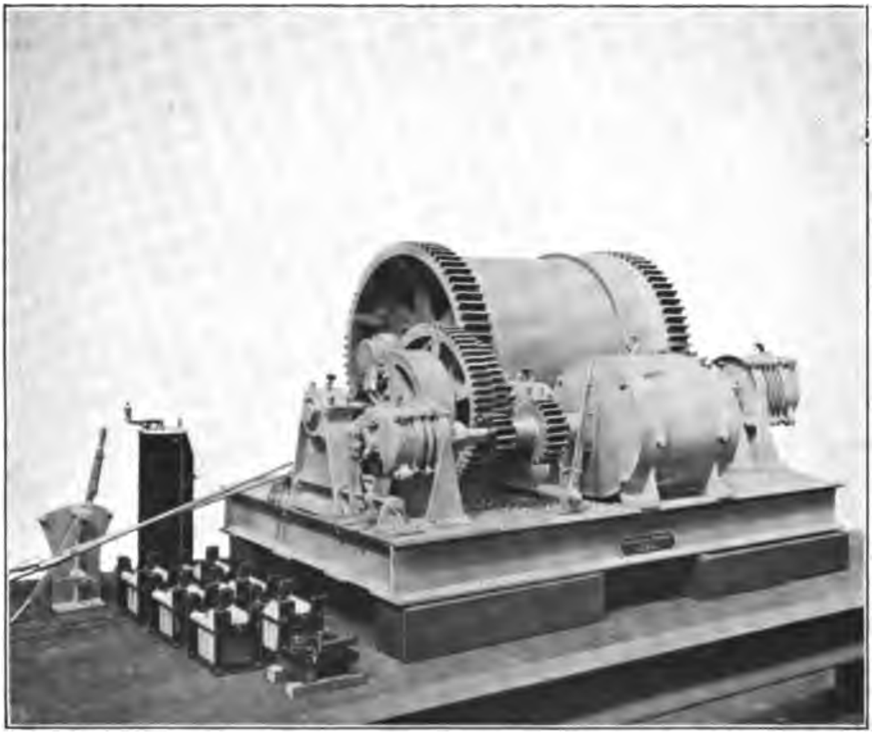


Fig. 69.—Winch for 10-Ton Cableway.

To face p. 74.]

permanently geared to the motor, while the capstan drum for hauling could be clutched in or out by a friction clutch on the second motion shaft. Each motion was fitted with an electro-magnetic disc brake. The hoisting brake was provided with hand release, so that loads could be lowered on the brake without putting current on the motor. The coil of the hoisting brake was permanently in the circuit, so that this brake always released when current was switched on to the motor. The coil of the brake for the hauling drum was in a loop circuit, which was only brought into the main circuit when loads were to be travelled, in which case this brake also released when current was switched on to the motor. The switch for bringing the coil of the hauling drum brake into the main circuit was connected to the lever of the friction clutch so that the movement of the one lever put the clutch in or out, and at the same time effected the necessary change in the electrical connections. The motor was series-wound 100 brake horse-power, 500 revolutions per minute. As the hoisting rope was four part, the speed of travelling was four times the speed of hoisting. The winch was provided with speed change gear, and the speeds were, in slow gear for 10-ton loads, hoisting 75 feet, and travelling 300 feet per minute; and in fast gear for 5-ton loads, hoisting 150 feet per minute, and travelling 600 feet. An illustration of the winch is given in Fig. 69.

10-Ton Electric Cableway with Balanced Rope.* — This cableway was on the Brothers system, and was used in the construction of the railway bridge over the Zambesi, near the Victoria Falls, for which Sir Douglas Fox & Partners were the engineers.

The span was 870 feet, and loads up to 10 tons could be handled, the speed of lifting being 20 feet per minute, and travelling 300 feet per minute.

The diameter of the rope was 2⅛ inches, and the dip with full load at the centre of the span was 50 feet, so that the tension was about 70 tons.

A general view of the cableway is shown in Fig. 70. Two carriages were made for this cableway, one by Messrs. Scott & Mountain, and one by Messrs. Ransomes & Rapier. Fig. 71 shows the carriage made by the latter firm. The hoisting drum was mounted on the carriage, which also carried an electric motor. Gearing and clutches were provided, so that the motor could drive either the hoisting drum or the travelling wheels. Thus the carriage was self-contained, and generally similar to the crab of an overhead

* See *Engineering*, 22nd April, 1904; also *Proc. Inst. C.E.*, vol. clviii., p. 216.

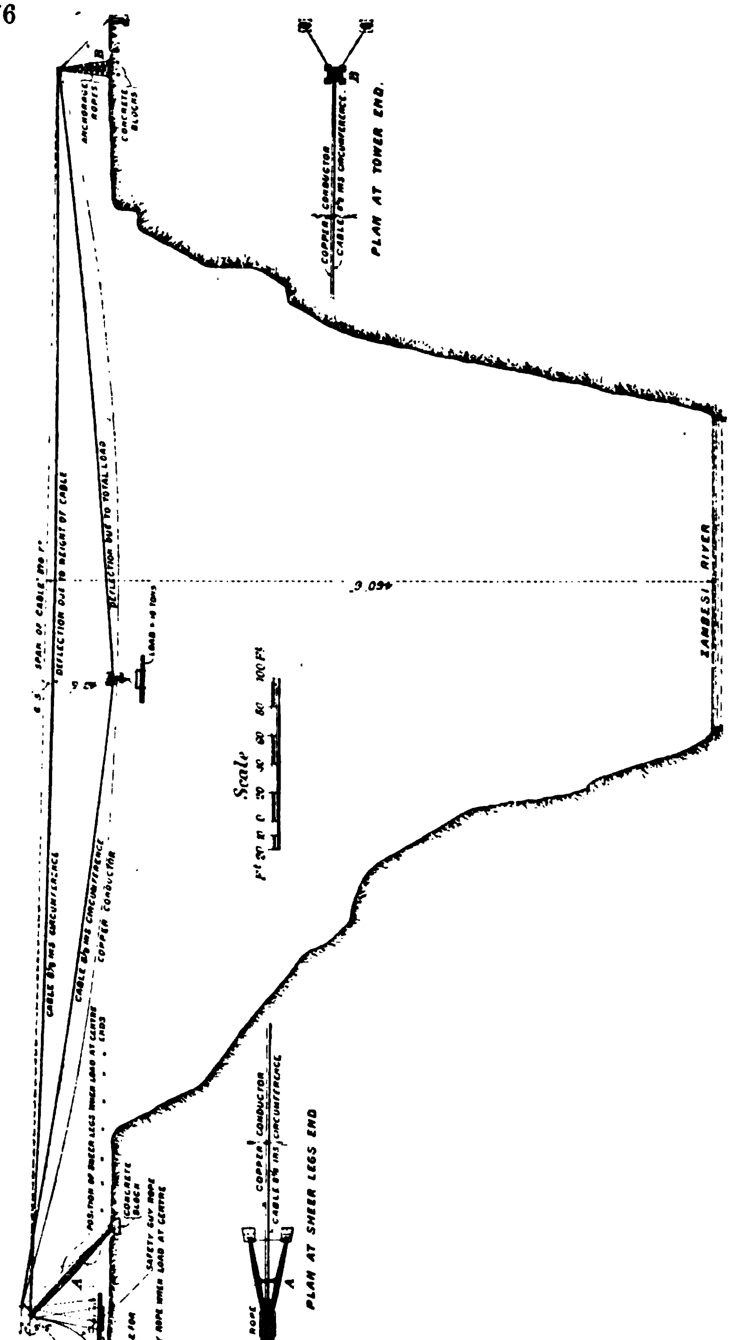
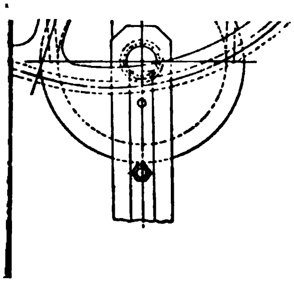


Fig. 70.*—10-Ton Cableway over the Zambesi.



Side View.

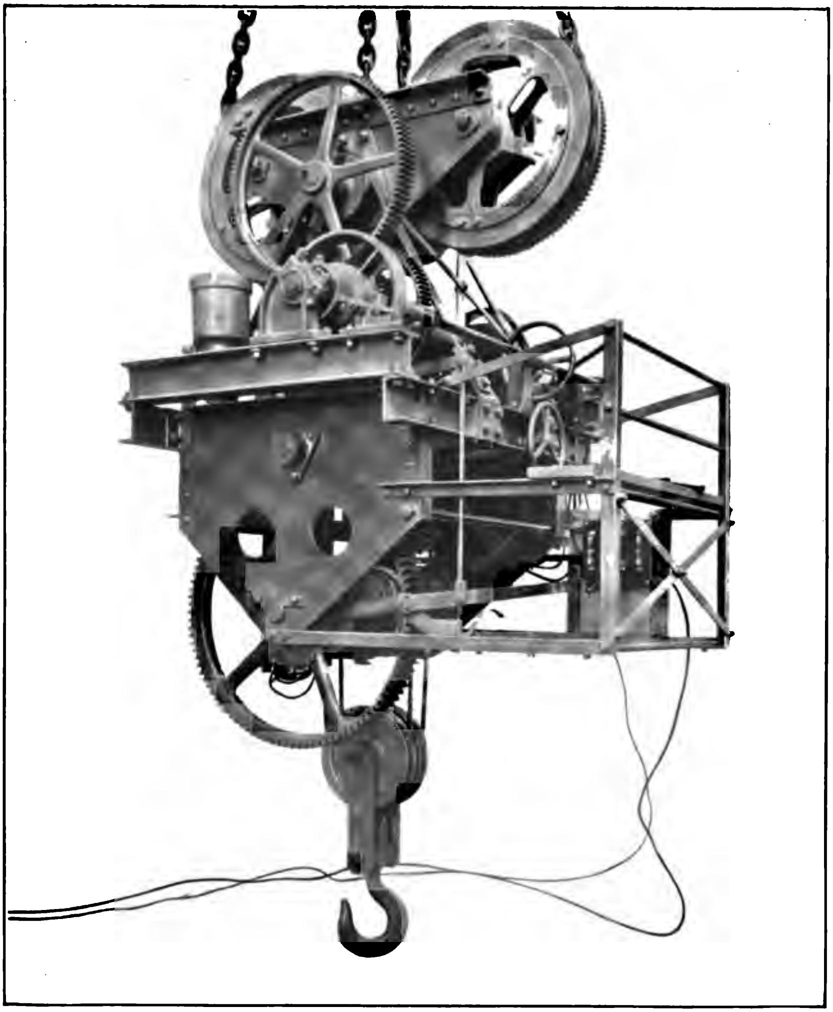


Fig. 72.—10-Ton Carriage for Zambesi Cableway.

To face p. 76.

crane, and no ropes were required from tower to tower except the main rope. The diameter of the travelling wheels was 2 feet 6 inches, and, as the weight of the carriage was 5 tons, the load per wheel with the full load of 10 tons was 7·5 tons. An illustration of the carriage is given in Fig. 72. The driver rode on the carriage, to which current was conveyed by a copper wire stretched from tower to tower, the return current passing through the main rope.

From some tests of this cableway made by the author it was found that to hoist 10 tons at 20 feet per minute required 27 E.H.P., and to travel this load at 300 feet per minute required 45 E.H.P.

At the first glance the cableway having a balanced main rope and self-contained carriage appears to offer considerable advantages, but closer examination shows that the advantages are not so great as they at first appear.

The advantage claimed for the self-contained carriage is that it abolishes all ropes except the main rope, and it renders hoisting rope carriers unnecessary. Owing, however, to the greater weight of the self-contained carriage, the main rope has to be of increased size, so that the amount of material in the rope is about the same as in the group of ropes in cableways having a travelling monkey, while the passing of the tractive effort through the wheels leads to rapid wear of the wheels and rope. It is an undoubted advantage to be able to dispense with the button rope and hoisting rope carriers, so that probably a satisfactory arrangement would be to use a self-contained carriage constructed to haul itself along. This could be done by fastening a rope to each tower, and passing it round a capstan drum on the carriage, so that it could haul itself along the main rope instead of driving the travelling wheels. For moderate loads and speeds this form of carriage could be used, but for heavy loads and high speeds the large power required would make the carriage too heavy to be practicable. The balanced main rope may be used either with the travelling monkey or the self-contained carriage, and is certainly an advantage, as the balance weight automatically compensates for changes of length due to elasticity and temperature, and by tightening the rope as the carriage approaches the towers it considerably lessens the power required to get up the incline.

CHAPTER VIII.

POWER REQUIRED FOR CRANE DRIVING.

THE power required to drive a hoisting motion is made up of two parts—(1) That which performs the work of lifting the load, and (2) that which is required to overcome the friction of the gear, etc. The first may be calculated quite simply, but the second can only be estimated with any approach to accuracy by reference to data based on previous experience.

In making calculations of a general character, the useful work is taken as a fraction of the whole, and its probable value is estimated from the test figures of the nearest previous parallel case. This fraction represents the efficiency e of the machinery, and the total power required is then—

$$H = \frac{2{,}240\ WS}{33{,}000\ e} = \frac{\cdot 06788\ WS}{e}.$$

If H is to represent the brake horse-power of the motor, the value taken for e is the efficiency of the mechanical parts only, but if H is to represent the total electrical horse-power taken at the crane terminals, then the value taken for e must allow for the motor losses in addition.

Although for general purposes records of the overall efficiency of previous cranes are useful, it is better for the designer whenever possible to analyse the test figures and ascertain the losses in the different parts of the machinery. By this means the effect of alterations in future designs may be more accurately estimated, and in addition the detail figures show where the greatest losses take place, and so indicate the lines on which improvements may be made.

The following are examples of analyses taken from test figures:

30-Ton Crane (Fig. 11).—The sources of loss in the hoisting gear were as follows :—Bearings, two 3¼ inches, two 4 inches, and two 5 inches. All roller bearings mounted in swivels. Ropes and Sheaves—Rope ⅞ inch diameter, $\frac{6}{37}$, winding two parts on 24-inch drum, and passing round six 24-inch sheaves fitted with roller bearings. Speed of rope four times the speed of load. Gearing—Pinion 17 teeth, 1⅛-inch pitch into wheel, 88 teeth ; pinion 21 teeth, 1⅝-inch pitch into wheel, 92 teeth ; pinion 15 teeth, 2-inch pitch into wheel, 76 teeth. The first two sets having cut teeth, and the last cast teeth.

Table No. 1, Test No. 1 (Fig. 13a).	H.P.	e
30·1 tons lifted at 4·4 f.p.m.,	8·99	·686
Hook and sheaves, ·6 ton at 4·4 f.p.m.,	·18	·014
Hoisting gear,	1·737	·133
Brake coil,	·145	·011
Motor losses,	2·048	·156
	13·100	1·000

Table No. 1, Test No. 3 (Fig. 13c).		
15·1 tons lifted at 10·17 f.p.m.,	10·42	·689
Hook and sheaves ·6 ton at 10·17 f.p.m.,	·414	·027
Hoisting gear,	2·744	·181
Brake coil,	·145	·010
Motor losses,	1·407	·093
	15·130	1·000

5-Ton Derrick Crane.—Hoisting motion driven by a shunt-wound motor, the speed of which was varied to suit different loads by varying the field strength. No electro-magnetic brake was fitted, a foot brake only being used. Sources of mechanical loss—Bearings, two 3 inches, and two 4 inches. Ordinary plummer blocks with gun-metal steps and Stauffer lubricators, bolted on to steel-work of mast. Rope and sheaves—Rope 0·96 inch diameter, $\frac{6}{37}$, winding one part on drum 18 inches diameter, and passing round one sheave 17 inches diameter and one sheave 21 inches diameter. Sheaves bushed with gun-metal and having ordinary lubrication. The rope being single, its speed is the same as that of the load. Gearing—Pinion 15 teeth, 1¼-inch pitch into wheel, 76 teeth-cut gear; and pinion 15 teeth, 1½-inch pitch into wheel, 101 teeth-cast gear.

Diagram (Fig. 73*a*).	H.P.	e
5 tons lifted at 44 f.p.m.,	14·93	·643
Hoisting gear,	4·52	·195
Motor losses,	3·75	·162
	23·20	1·000
Diagram (Fig. 73*c*).		
2·1 tons lifted at 90 f.p.m.,	12·83	·690
Hoisting gear,	3·49	·188
Motor losses,	2·28	·122
	18·60	1·000

The calculation of the power for the travelling and cross traversing motions is based on the assumption, from previous experience, of a tractive effort in lbs. per ton of gross weight. This tractive effort is frequently calculated on the E.H.P. absorbed at the crane terminals, as will be seen in the various test figures given in this work. In this case the equation for the horse-power is simply—

$$H = \cdot 06788\ W_t F_5 S.$$

A collection of values of the tractive effort for various sizes and types of cranes is useful for general purposes, but it is advisable wherever possible to analyse the various losses, and so ascertain how the total tractive effort is made up. It is not generally convenient to separate out the mechanical losses, but if the motor losses and mechanical losses are separated, and the tractive effort is calculated per ton per wheel on the mechanical losses only, the figure so obtained is more useful as a guide in future designs than a general figure based on the horse-power absorbed at the crane terminals. The formula for the brake horse-power of the motor is then—

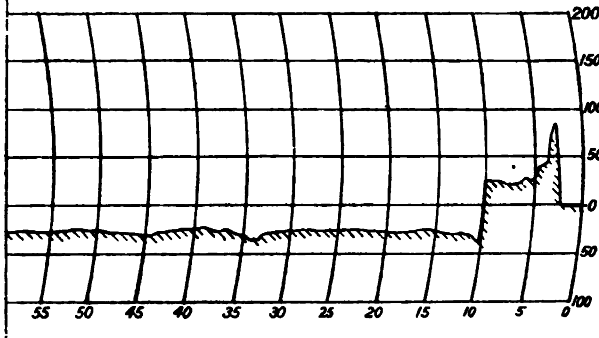
$$H = \cdot 06788\ L_1 N_w F_6 S,$$

while the electrical horse-power absorbed at the crane terminals is—

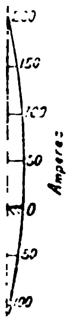
$$H = \frac{\cdot 06788\ L_1 N_w F_6 S}{e_m},$$

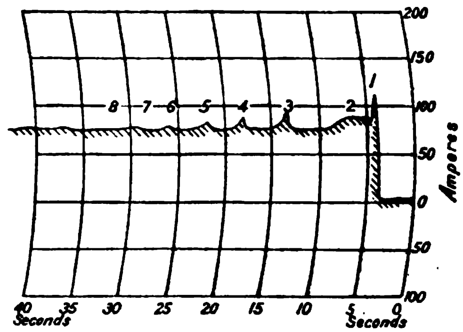
e_m being the efficiency of the motor.

Referring to Table No. I., the mechanical losses in the crab



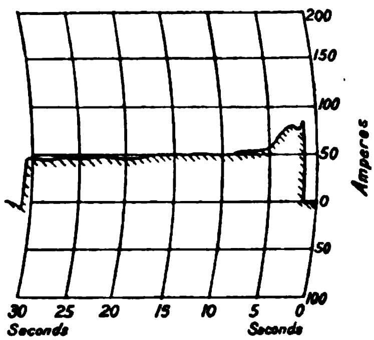
Lowering 5 tons at 50 feet per minute.
– 25 Amperes = – 7·4 E.H.P.
H.P. at Hook = 17, Reverse Eff. = 43½ per cent.





e. Lifting 5 tons.
Diagram showing current on successive steps
of Controller when starting.

cent.



h. Derricking 5 tons out 50 feet radius.
47·5 Amperes = 14 E.H.P.

Amperes

on 5-Ton Derrick Crane.

traversing motion were as follows :—Rolling friction of four wheels, 20 inches diameter. Bearings—Four axle bearings, 4¾ inches diameter, swivelling roller bearings. Two 2½ inches and four 2 inches, all gun-metal bushed, swivelling, and self-oiling. Gearing —Pinion 17 teeth, ⅞-inch pitch into wheel, 46 teeth ; pinion 19 teeth, 1-inch pitch into wheel, 50 teeth ; pinion 17 teeth, 1⅛-inch pitch into wheel, 50 teeth ; and pinion 21 teeth, 1½-inch pitch into wheel, 38 teeth, all cut gear.

Deducting electrical losses, the curve (Fig. 74) gives the values of F_6 corresponding to different wheel loads.

The mechanical losses in the travelling motion were—Rolling friction of six wheels, 30 inches diameter. Bearings—Twelve axle

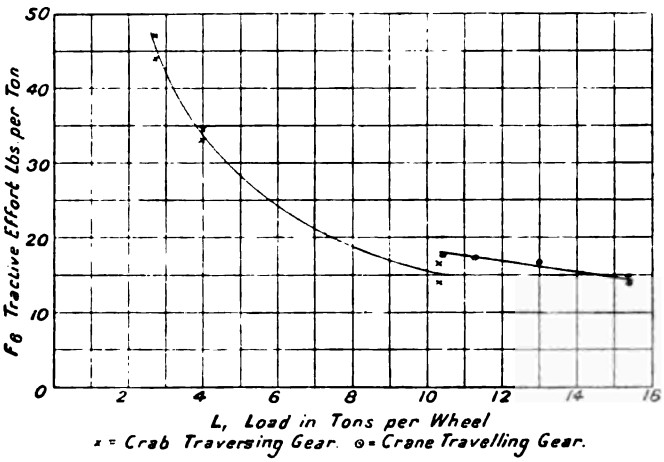


Fig. 74.—Curve of Tractive Effort.

bearings, 4¾ inches diameter, roller bearings. Six 3-inch bearings gun-metal bushed and fitted with lubricators, six 3-inch swivelling roller bearings. Gearing—Pinion 17 teeth, 1¼-inch pitch into wheel, 70 teeth, cut gear ; two pinions 15 teeth, 1⅝-inch pitch into wheels, 58 teeth, cast gear. Two driving chains (see Fig. 11, end view).

The roller bearings for the axles of the travelling motion did not turn out satisfactorily, but were in good order at the time the test was taken, which was when the crane had been at work six months, so that the figures form an interesting comparison with those of the crab traversing train, and show the greater tractive effort required in the travelling motion for a given wheel load, owing principally to the larger number of bearings in this motion, the losses in which more than counterbalance the gain due to the larger wheels.

The power required to travel the carriage on a cableway varies continually as the carriage travels along. In travelling from tower to tower the carriage runs down the incline freely, taking little or no power till it gets to the centre of the span. At this point it travels horizontally, and the power required is simply that necessary to overcome the tractive resistance, which averages about 150 lbs. per ton. After passing the centre, it commences to ascend the incline, and the power increases as it gets nearer to the tower. The force necessary to ascend the incline, found by the method shown in Fig. 58, being represented by F_7, the horse-power required is—

$$H = \cdot 06788\ S(W F_6 + F_7),$$

this being the brake horse-power of the motor.

In calculating the power required to lift the jib of locomotive jib cranes, in which the load lifts with the jib the total weight lifted is the weight of the load plus half the weight of the jib. The mechanical losses in the jib lifting gear only require to be allowed for as the hoisting gear remains at rest while the jib is being raised.

In Derrick cranes (Fig. 28), the gear being compensated, the load does not lift with the jib. In this case the weight lifted is simply half the weight of the jib, but as the hoisting gear runs as well as the jib lifting gear, as explained in the chapter on Derrick Cranes, the losses in both these motions have to be allowed for. As the end of the jib describes an arc of a circle, the power required to lift it is not uniform, being a maximum at the outer radius and a minimum at the inner. In the case of the crane shown in Fig. 28 the current taken for lifting the jib was 110 amperes at the outer radius and 15 amperes at the inner, while the current for lifting the load was 80 amperes.

The power required to drive a slewing motion will depend on whether the weight of the revolving structure is taken on a central pivot or on a ring of rollers. In a Derrick crane the weight is taken on a pivot, so that the power is made up of the frictional loss in the bottom pin as a pivot, and the frictional loss on the top and bottom pins, due to the horizontal pressure on them, to which must be added the frictional losses in the gearing, and the motor losses. The mechanical losses being of a very indeterminate character, a general formula for the power is probably most suitable, thus—

$$H = CS_2\left(W_t + \frac{W_b R}{V_d}\right).$$

C being a constant based on previous experiments, and varying with the form of transmission gear employed.

In the case of a 5-ton Derrick crane with steel lattice-work jib, 70 feet long, weighing 3 tons, an ampere-meter diagram is given in Fig. 73*g*, showing the current taken when slewing at maximum radius at full load. The particulars in this case were as follows :—Horse-power, 2·95 at crane terminals ; speed of slewing, 0·75 revolution per minute ; total weight slewed = W_t = 14 tons ; radius, 50 feet ; distance from centre to centre of top and bottom bearings, 30·75 feet. W_b = 6·5 tons. C = 0·16. In this case a separate motor was used for slewing, and it drove the slewing pinion through cut spur gear and bevels.

In another exactly similar crane one motor was used to drive all the motions, friction clutches being provided for putting the various motions in and out. In this crane the power taken to slew a 5-ton load at 50 feet radius was 8·5 horse-power, the extra power being all wasted in the gearing. The Derrick crane, shown in Fig. 28, took 10 amperes when slewing steadily under full load, the horse-power being 2·95. The speed of slewing was 0·5 revolution per minute. Total weight slewed, 12·5 tons. Radius, 30 feet. Distance from centre to centre of top and bottom bearings, 21·25 feet. Load, plus half jib, 5·5 tons. C = 0·29. A separate slewing motor was used, the power being transmitted to the slewing pinion through worm gear. The worm losses probably account for the higher value of C.

If a Derrick is to slew under wind pressure, the power required to work against this pressure must be added to the previous figures. As a rule, derricks cannot be worked with more than a light breeze blowing, as the swinging of the load makes them unmanageable. The additional power required for slewing against wind pressure—

$$H = \frac{\cdot 5 \times 12}{63{,}025} \times S_4 p_2 R S_2$$

$$= \cdot 0000952\ S_4 p_2 R S_2.$$

This gives the maximum power, which would only require to be exerted when the jib was in such a position as to swing directly against the wind, the power being less in all other positions. In any case, the additional power is not very great. Taking the 5-ton Derrick with 70-foot steel jib, the surface of this taking back and front latticing is 144 square feet. A wind velocity of 30 miles per hour, which is a fairly brisk gale, corresponds to a pressure of about 3 lbs. per square foot. Under these circumstances the additional horse-power required

$$= \cdot 0000952 \times 144 \times 3 \times 50 \times 0 \cdot 75 = 1 \cdot 54 \text{ horse-power.}$$

If the revolving portion of a crane is carried on a ring of rollers, the general formula of the power for slewing may be based upon the tractive effort required at the mean radius of the roller path. The horse-power is then—

$$H = C W_t R_1 S_2,$$

in which C is a constant depending on the nature of the transmission gear. As an example, the power required for slewing the transporter (Fig. 37) is shown among the recorder diagrams in Fig. 39. The weight of the revolving portion was 58 tons, and the mean radius to the centre of the roller path 8 feet 2 inches.

CHAPTER IX.

STARTING TORQUE AND ACCELERATION.

WHERE a crane is required to perform a definite cycle of operations, it is usual to specify the time which the different motions of the crane shall take to move the load through definite distances. In such a case, in order to allow for the time lost in starting and stopping, the full speed of the motion requires to be greater than the mean speed obtained by dividing the distance by the time.

During the movement of the load from one point to another the time is divided into three periods, t_a = the acceleration period, t_b = the period during which the crane runs steadily at its full-rated speed, and t_c = the deceleration period.

Taking D as the distance, and S as the full speed—

$$S = \frac{D}{\cdot 5t_a + t_b + \cdot 5t_c}.$$

These points are illustrated in the curves in Fig. 75, which represent the moving of a load from one point to another, 150 feet distant, in 30 seconds. The mean speed is 5 feet per second, while with t_a, t_b, and t_c each equal to 10 seconds, the full speed requires to be 7·5 feet per second.

By diminishing t_a and t_c we can bring the full speed nearer to the mean speed, but in doing this we increase the rate of acceleration and deceleration, and so increase the overload on the motor and gearing during the first period, and increase the brake pressure and consequent wear of machinery during the latter period. The proportions of the three periods are, therefore, a matter for the judgment of the designer in each particular case. There is an important difference in the percentage of overload on the motor during acceleration of hoisting motions as compared with travelling motions.

Thus, taking a load of 10 tons, moved at a rate of 7·5 feet per second, and getting up to full speed in ten seconds. The rate of acceleration will be 0·75 foot per second per second, and the pull on the rope required to give this acceleration will be 0·233 ton.

When hoisting, the pull on the rope when running steadily is 10 tons, and when accelerating 10·233, or an overload of 2·33 per cent., which is very much inside the limit to which we may reasonably go.

For travelling, a fair average tractive coefficient is 25 lbs. per ton, so that when running steadily the horizontal pull is 0·11 ton, and when accelerating 0·34 ton, or an overload of 209 per cent., which, except for special cases, is excessive.

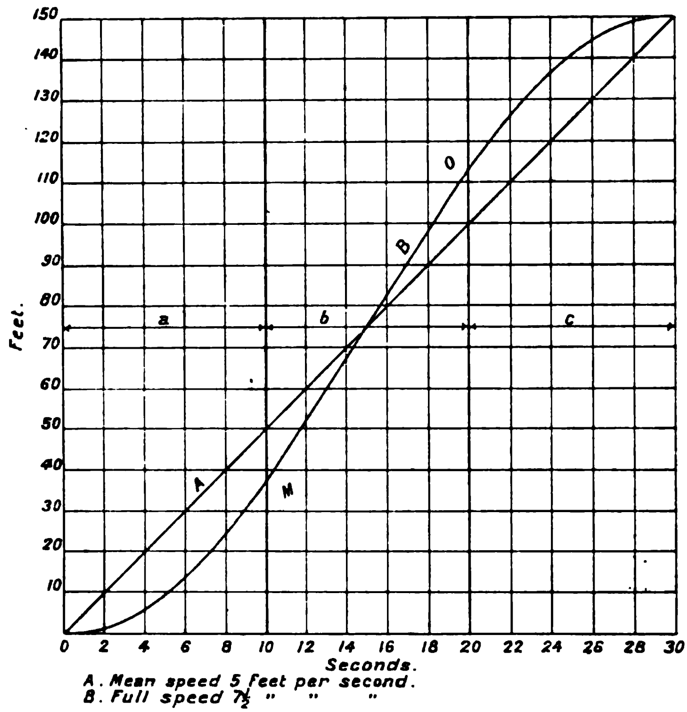


Fig. 75.

In order to obtain reasonable overload figures, it is consequently usual to adopt a shorter acceleration period for hoisting, and a longer one for travelling.

A further important point is the relationship of the force required to accelerate the rotary parts to that required for the linear acceleration of the load itself.

Dealing first with the hoisting motion, we will take the three crabs shown in Figs. 76, 77, and 78. Each of these performs the same amount of work—viz., 135 foot-tons per minute, but with different loads and speeds.

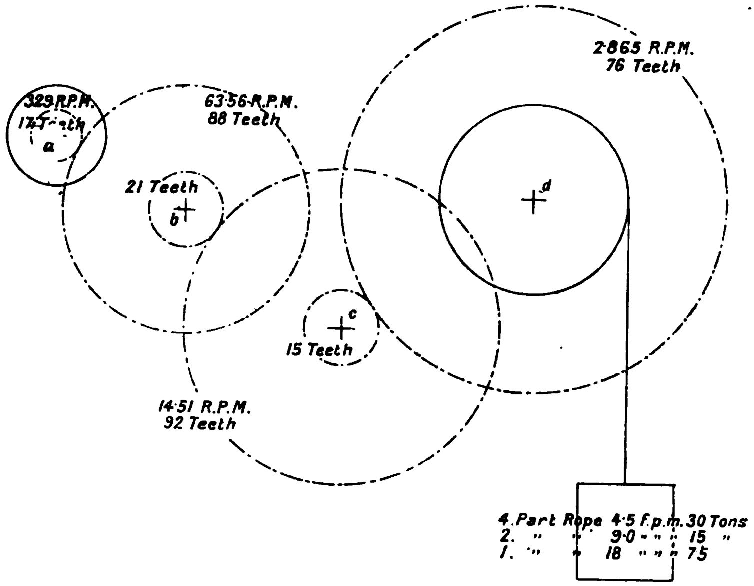


Fig. 76.

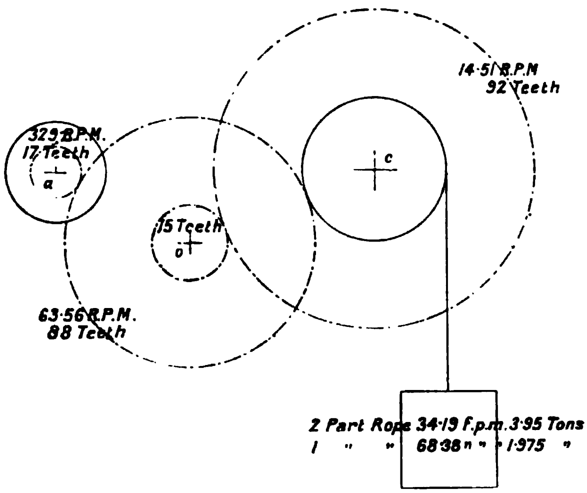


Fig. 77.

In Table V. are given the weights and radii of gyration of the revolving parts, and the equivalent weight of these parts at the centre of gyration of the motor armature. In this table weights are in lbs., radii of gyration in inches, κ_a = radius of gyration of armature, W_a = weight of armature, and W_t = total equivalent weight of all the revolving parts if assembled at the centre of gyration of the armature.

In determining the amount of torque which the motor must exert in order to impart the necessary acceleration to the revolving parts we may adopt either of two methods.

Taking each shaft in succession, we may calculate the force required at their centre of gyration, to accelerate the parts mounted on that shaft, and from this determine the force required at the centre of gyration of the armature, allowing, of course, for frictional losses. The sum of these forces, plus that required to accelerate the armature itself, represents the total force required to accelerate the revolving parts. This is a laborious process, involving as it does the calculation of the individual speed and rate of acceleration of each shaft.

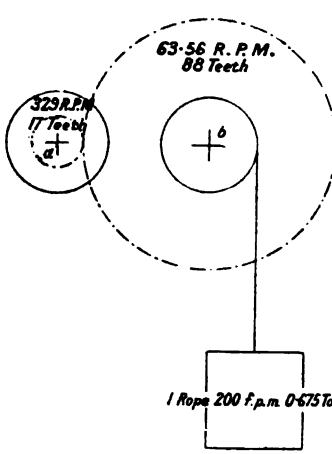


Fig. 78.

A more expeditious process is to calculate for each shaft the amount of weight which, at the centre of gyration of the armature, will require the same amount of accelerating force as would be required at the centre of gyration of the armature for the acceleration of the parts mounted on that shaft. The sum of these weights has been described above as the equivalent weight at the centre of gyration of the armature and represented by W_t.

The equivalent weight of one revolving part at the centre of gyration of another revolving part to which it is geared is inversely proportional to the squares of the respective velocities of the centres of gyration of the two parts, thus for shaft C (Fig. 76)—

$$W_x = \frac{W_c(\kappa_c n_a n_b)^2}{(\kappa_a N_b N_c)^2 e},$$

in which W_c and κ_c are the weight and radius of gyration respectively of the parts mounted on shaft C, n_a n_b and N_b N_c being the numbers of teeth on the driving and driven wheels respectively, and e a fraction representing the efficiency of the gear up to that point.

Substituting the actual figures, we get—

$$W_x = \frac{500 \times (12 \times 17 \times 21)^2}{(4\cdot5 \times 88 \times 92)^2 \times \cdot95} = 7\cdot3.$$

In Table V. there is also given the ratio which the total equivalent weight bears to the weight of the armature in the case of each crab illustrated. These figures are useful as indicating the percentage which may be added to the armature in cases where there is not time to go through the process of calculating the weights in detail.

From Table V. the figures in Table VI. have been calculated, which show the various accelerating forces required by the three crabs for various loads and speeds.

TABLE V.

Fig.	Shaft.	Weight.	Radius of Gyration.	W_x Equivalent Weight at x_a	$\frac{W_t}{W_a}$
76	A	504	4·5	504	$\frac{576\cdot6}{504} = 1\cdot14$
	B	330	10	62·4	
	C	500	12	7·3	
	D	4,168	13	2·9	
77	A	504	4·5	504	$\frac{582\cdot4}{504} = 1\cdot16$
	B	330	10	62·4	
	C	1,500	10·15	16	
78	A	504	4·5	504	$\frac{594\cdot7}{504} = 1\cdot18$
	B	655	8·45	90·7	

It will be noted that in all cases the accelerating force required for the revolving parts greatly exceeds that required for the load itself, and the figure $\frac{F_1}{F}$ will be a general guide to designers in cases where they have not time to make calculations.

In Fig. 75 the periods t_a or t_c are shown equal, and in horizontal motions this is usually the case.

In vertical motions, when hoisting, the load resists acceleration, and when lowering the reverse is the case.

Taking the crab in Fig. 76 with 30 tons load, we already have the accelerating force for hoisting given in Table VI. When stopping, force is required to pull up the revolving parts, and, neglecting for the moment the action of the automatic electric brake, we will assume that this force is provided by the action of gravity on the load.

TABLE VI.—DUTY OF CRANE 135 FT.-TONS PER MINUTE. ACCELERATION PERIOD $2\frac{1}{2}$ SECONDS.

		Fig. 76.			Fig. 77.		Fig. 78.
Load on hook,	Tons	30	15	7·5	3·95	1·975	·675
Speed of hook,	F.P.M.	4·5	9·0	18·0	34·25	68·5	200
,,	F.P.S.	·075	·15	·3	·57	1·14	3·32
Accel. of hook,	F.P.S. per sec.	·03	·06	·12	·23	·45	1·34
Speed of κ_a	F.P.M.	778	778	778	778	778	778
,,	F.P.S.	12·96	12·96	12·96	12·96	12·96	12·96
Accel. of κ_a	F.P.S. per sec.	5·2	5·2	5·2	5·2	5·2	5·2
Force required to accel. load. At load,	lbs.	62·5	62·5	62·5	62·5	62·5	62·5
Force required to accel. load. At κ_a (= F),	lbs.	·45	·85	1·2	3·1	6·1	17·3
Force required at κ_a to accel. arm. only,	lbs.	81	81	81	81	81	81
Force required at κ_a to accel. all rotary parts,	lbs.	93	93	93	94	94	96
Total force at κ_a for accel. = (F_1),	lbs.	93·45	93·85	94·2	97·1	100·1	113·3
Ratio $\frac{F_1}{F}$,	...	207	110	78·5	31·2	16·6	6·5
*Force at κ_a to lift load steadily,	lbs.	480	455	442	442	432	420
*Total force at κ_a when accel.,	lbs.	573·4	548·8	536	539	532	533
Overload torque on arm. when accel.,	per cent.	20	20	21	22	23	27
Force at pitch line of pinion a when lifting steadily,	lbs.	710	672	655	655	640	620
†Ditto when accelerating,	lbs.	730	695	675	680	670	670
Overload on pinion when accel.,	per cent.	3	3·5	3·5	4	5	8

We must now reverse our previous calculation, and ascertain the equivalent weight of the revolving parts at the point of suspension of the load, instead of at the centre of gyration of the motor armature. This amounts to 7,550 tons, so that when current is cut off we have the equivalent of 7,580 tons travelling at a speed of 0·075 foot per second, with a retarding force of 30 tons. The actual retarding force will be a little over 30 tons, owing to the friction of the gear, but the amount is negligible. The load will be

* Allowing for frictional losses in gearing.

† Force to accelerate armature being deducted, as it does not come on the pinion teeth.

stopped in 0·58 second, as shown on the curve (Fig. 79). So far as the time taken in stopping is concerned, there is evidently no need for the assistance of a brake. As the brake comes on the instant current is cut off the motor, its effect is to slightly shorten the deceleration period, but its great advantage is that by coming on while the machinery is still running it holds the load firmly the instant it comes to rest.

On starting to lower, we have an equivalent weight of 7,580 tons at the point of suspension accelerated by a force of 30 tons, assisted by a little current on the motor, the shape of curve being practically the same as when stopping the hoist, but reversed.

When stopping a load which is being lowered the conditions are again different. In this case the load is driving, and the re-

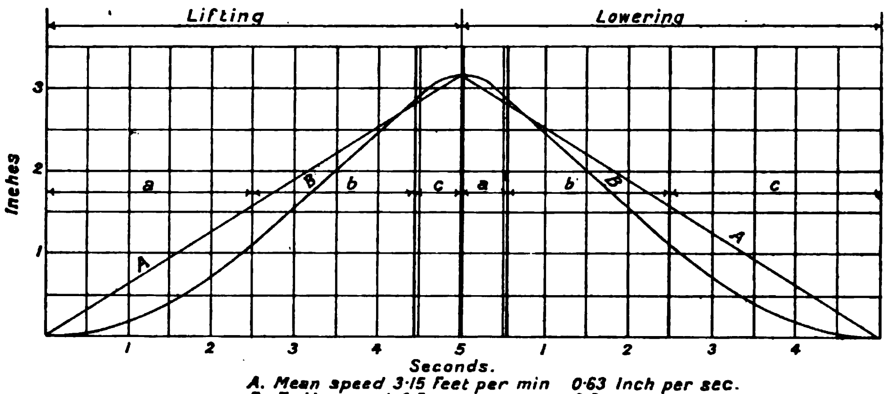


Fig. 79.

tarding force is provided partly by the brake and partly by the friction of the gear, the effect of which is, however, very trifling in the case of high efficiency gears. Allowing 2½ seconds for stopping, the retarding force required at the centre of gyration of the armature is 573 lbs., or a torque of 2,575 inch-lbs. on the brake. In this case the stress on the motor pinion teeth is about the same as when starting to hoist, and the shape of curve is the same, but reversed. The torque on the brake when holding the load of 30 tons at rest is 2,160 inch-lbs.

Fig. 80 is a recording ampere-meter diagram, showing the crab (Fig. 76) hoisting 30 tons at 4½ feet per minute with a starting period of 2½ seconds.

It will be seen from the foregoing figures and considerations that for the ordinary speeds of hoisting used in crane work the force required to accelerate the rotating parts greatly exceeds that

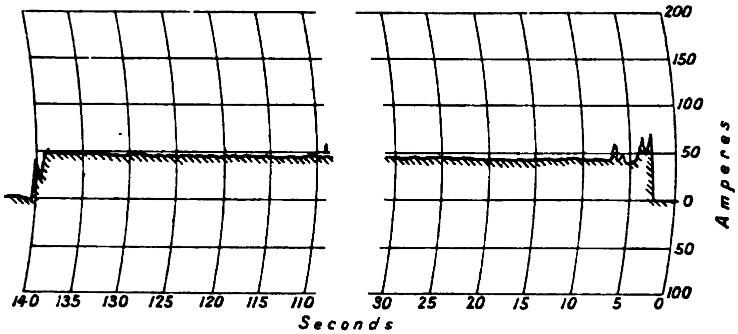


Fig. 80.

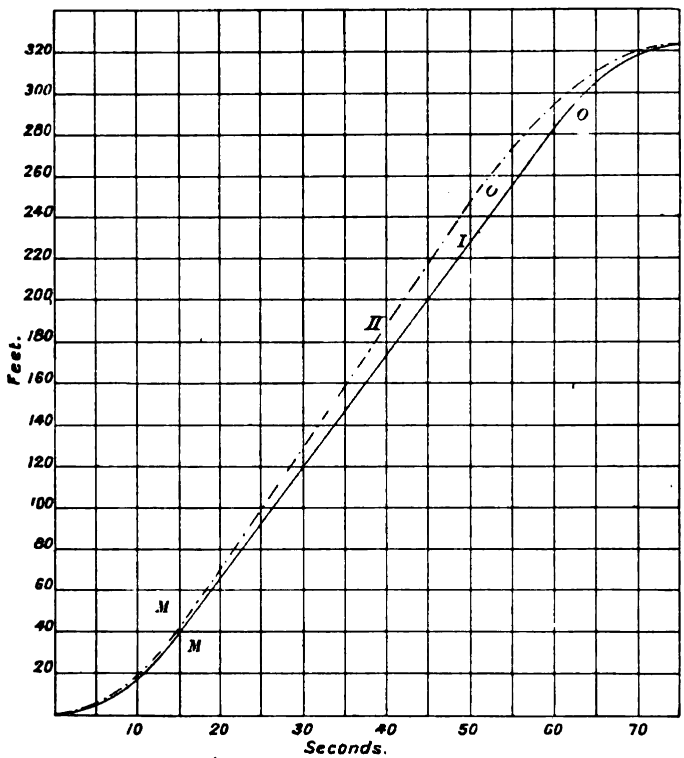


Fig. 81.

required to accelerate the load, and further, that by far the greater portion of the accelerating force is required for the motor armature, the weight and radius of gyration of which should, therefore, be kept as small as possible. Also, for the same duty in foot-tons per minute, the less the load and higher the speed the greater is the overload torque on the armature when starting.

Turning to the question of the travelling motion, Table VII. has been prepared giving the acceleration forces, etc., for two cranes—a 30-ton and a 10-ton. Both of these cranes are of considerable span, and their weight is consequently high compared to the loads which they handle.

TABLE VII.

		30-Ton Crane.		10-Ton Crane.	
Weight of crane,	Tons	63	63	45	45
,, load,	Tons	30	0	10	0
Total weight,	Tons	93	63	55	45
Speed,	F.P.M.	322·5	341	370	400
,,	F.P.S.	5·35	5·7	6·15	6·65
Acceleration period,	secs.	15	15	15	15
Rate of acceleration,	F.P.S. per sec.	·356	·38	·41	·442
Weight of motor armature,	lbs.	504	504	504	504
Radius of gyration κ_a	in.	4·5	4·5	4·5	4·5
Equivalent weight of other rotary parts at κ_a	lbs.	125	125	112	112
Speed of κ_a	F.P.M.	700	740	805	870
,,	F.P.S.	11·6	12·3	13·4	15·5
Rate of acceleration of κ_a	F.P.S. per sec.	·775	·82	·895	1·04
Force to accelerate load. At load,	lbs.	2300	1660	1530	1380
* ,, ,, ,, At κ_a (= F),	lbs.	1180	850	785	710
Force at κ_a to accelerate all rotary parts.	lbs.	15	15·9	17	19·8
Total force at κ_a for acceleration (= F_1),	lbs.	1195	865·9	802	729·8
† $\frac{F_1}{F}$,	...	1·013	1·019	1·022	1·028
*Tractive coefficient,	lbs. per ton	16	20	27·6	29
Force at κ_a when travelling steadily,	lbs.	688	585	700	600
Total mean force during acceleration,	lbs.	1883	1451	1502	1330
Overload torque on arm. when accel. taken on normal torque at full load,	per cent.	174	125	115	104

As the force required for rotary acceleration is so small, the percentage of overload torque on the motor pinion is practically the same as on the armature.

A comparison of this table with Table VI. shows us that for the travelling motion the accelerating force required for the rotating parts is quite insignificant compared to that required for the acceleration of the total weight in a horizontal direction. This point is brought out prominently by the ratios $\frac{F_1}{F}$ given in the two tables.

* Allowing for gearing friction. † Compare with Table vi.

It will be noted that the overload torque at starting greatly exceeds that required for hoisting, and that as very little of the accelerating force is absorbed by the motor armature practically the whole of the overload comes on the pinion teeth. Thus, in the case of a crane which required the same H.P. for the hoisting motion as for the travelling motion when running steadily, the motor and gearing for the travelling motion would require to be larger and

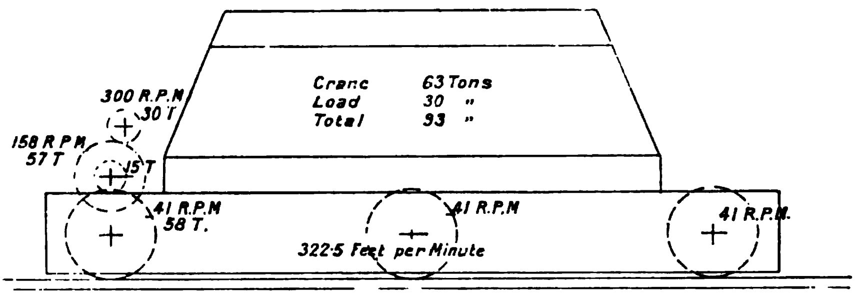


Fig. 82.

stronger than for the hoisting motion, owing to the greater stresses to which they would be subjected when starting. Fig. 82 shows an end view of the 30-ton crane, with the arrangement of gearing for the travelling motion, and Fig. 83 is an ampere-meter diagram,

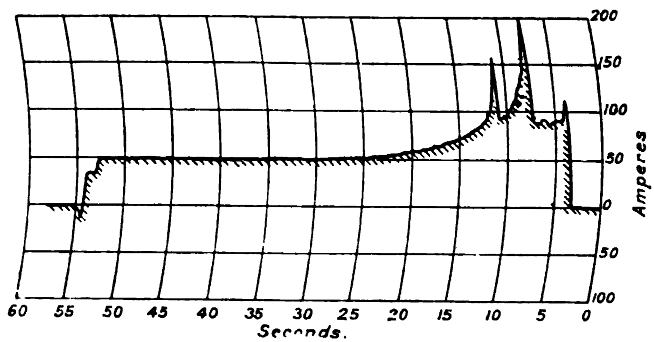


Fig. 83.

showing the current taken by this crane when accelerating and when running steadily with 30 tons' load. The currents are relatively 102 amperes (mean) and 50 amperes, showing an overload current of 104 per cent. The relation of current to torque is dealt with in the chapter on Motors.

In cases where the crane has to transport its load between two points in a given time, the amount of energy consumed per trip will depend on the relative proportions of the periods t_a, t_b, and t_c.

In Fig. 81, in which the load is moved from one point to another, 322·5 feet distant in 1¼ minutes two curves are given. One of these curves, No. I., is a symmetrical one, in which t_a and t_c are both 15 seconds. The rates of acceleration and deceleration are both alike—viz., 0·356 foot per second per second—and the speed during t_b is 322·5 feet per minute, the mean speed over the run being 260 feet per minute. Current is cut off at the point O, and the electro-magnetic brake mounted on the armature shaft comes into action, assisting the frictional resistance of the crane to bring it to rest.

The current being 102 amperes during t_a, and 50 amperes during t_b, with a voltage of 217 we get a total energy consumption of 227 watt-hours (0·227 B.T.U.).

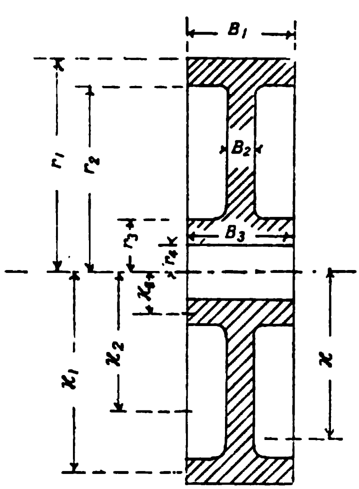


Fig. 84.

The frictional resistance of the crane being equal to 16 lbs. per ton of total weight will cause a rate of deceleration of 0·23 foot per second per second, and, as in the case of curve I., we require a rate of 0·356, the assistance of the brake is necessary. In order to bring the crane to rest in 15 seconds from a speed of 322·5 feet per minute it requires to exert a retarding torque of 1,700 inch-lbs.

As the energy dissipated by the brake has to come from the circuit in the first instance, a saving in current consumption may be effected by allowing the crane to come to rest under its own friction without the assistance of the brake. We then get the speed curve No. II., in which t_a is 15 seconds with a rate of acceleration of 0·38 foot per second per second, the speed during t_b is 345 feet per minute, and t_c is 24·75 seconds, with a rate of deceleration of 0·23 foot per second per second, the mean speed being, as before, 260 feet per minute.

The current during the period t_a is now 109·5 amperes, and during t_b 53·5 amperes, the voltage being as before, but as the current is cut off so much sooner, the total consumption is reduced to 212 watt-hours.

In order to avoid over-running, it is still advisable to retain the brake. On reaching the point *O* the driver moves the controller handle back to step No. 1. In this position current is cut off from the motor, but the brake magnet is still energised, so that the brake is held off and the crane runs freely. On moving the controller handle to the off position the brake comes into action and pulls the crane up.

During each starting period a considerable proportion of the electric energy is wasted in resistances, and, in order to show this point more clearly Table VIII. has been prepared.

TABLE VIII.—Energy Required, on Different Systems, to move a Total Weight of 93 Tons a distance of 322·5 Feet in $1\frac{1}{4}$ Minutes, including Starting and Stopping. (See Curves, Fig 81.)

System.		Curve I. Watt Hours.	Curve II. Watt Hours.
One motor, with ordinary controller and resistance,	Starting,	92	99
	Steady run,	135	113
	Total,	227	212
One motor, with no starting resistance losses,	Starting,	48	53
	Steady run,	135	113
	Total,	183	166
Two motors, with series parallel control,	Starting,	72	76
	Steady run,	135	113
	Total,	207	189

We see from this Table that if the starting resistance losses could be eliminated there would be a very considerable saving in consumption of energy. Unfortunately, at present the only means available for this purpose are too expensive in first cost, and too cumbersome to be practically applicable to crane work, the only really practicable arrangement being the series parallel, which, as shown in the third instance in the Table, does effect some economy over the single motor system. The series-parallel system, however, requiring, as it does, two motors and a somewhat special controller, is more expensive in first cost than the ordinary single motor arrangement, and so can only be used with advantage on cranes requiring considerable power.

In designing the framing of a crane the stresses due to acceleration should receive attention, as, if neglected, these stresses may be sufficient to cause failure of the structure. In the case of the 30-ton crane just dealt with, the force due to the acceleration of the travelling motion is equal to a load of 0·75 ton pressing sideways on the girders at the centre. As these girders are of massive design this amount of side pressure is of little importance, and a considerable increase in the rate of acceleration would cause no undue stress.

The circumstances are different, however, in the case of derrick and jib cranes in which the accelerating force for the slewing motion has to be transmitted by the joint at the foot of the jib.

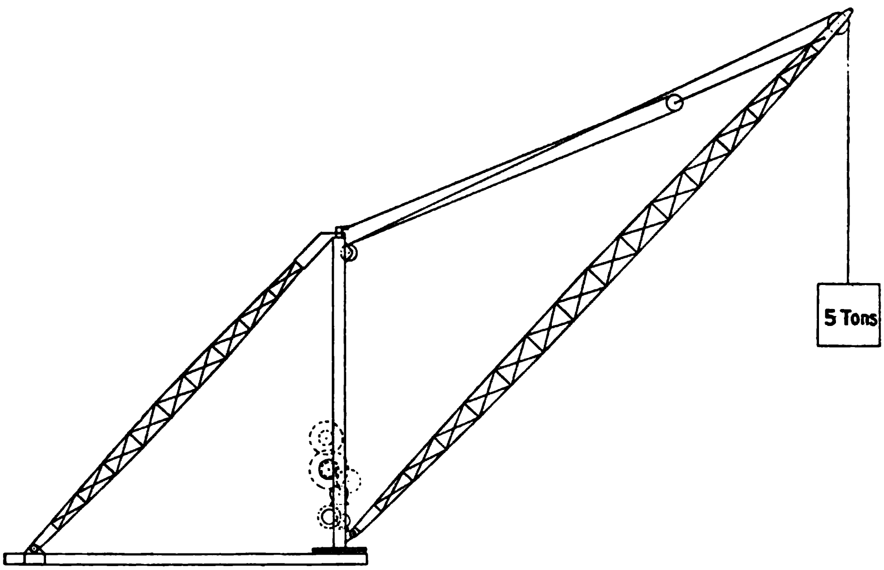


Fig. 85.—5-Ton Derrick Crane.

Fig. 85 is a diagram of a derrick crane having a jib 70 feet long, weighing 3 tons, and handling a load of 5 tons at a radius of 50 feet. The total load to be slewed is here 8 tons, and the mean radius of gyration is 41·6 feet.

The path of the load from guy to guy is about 225 feet, and the time allowed to move the load through this distance is one minute. With an acceleration period of 15 seconds, the speed during t_b requires to be 5 feet per second, and the force on each half of the joint is 1·2 tons (see Fig. 103), which is quite sufficient considering the already heavy load on these joints. If, however, the acceleration period is reduced to 5 seconds (in order to get quickly over short distances), the speed during t_b will be 4·2 feet per second, and the force during acceleration on each half of the joint will be 3 tons, which is altogether excessive for the usual proportions of joint. Cases have occurred in which the joint has been broken owing to the slewing motion being started too quickly. To prevent this, it is preferable to use a small separate motor for the slewing motion with its own circuit breaker to limit the starting current.

Where a heavy load is carried on a rivetted framing, such as in the case of a revolving cantilever crane, in which the back balance weight is carried on a lattice cantilever, the stresses due to acceleration and retardation of the balance weight must be provided for by horizontal bracing, or the rivetting will be liable to work loose in a comparatively short time.

FORMULÆ FOR REFERENCE.

Speed Curves (see Figs. 75, 79, and 81).

The Period t_b.—In order that the crane may move over a given distance D in time t ($= t_a + t_b + t_c$), including starting and stopping, the speed during t_b requires to be—

$$S_1 = \frac{D}{\cdot 5t_a + t_b + \cdot 5t_c}.$$

In some cases the crane may be allowed to run under its own momentum during this period. Current being cut off at the point M, the speed steadily falls from this point, at which acceleration ceases, to the point O, at which the brake is put on.

The mean speed is then that given by the formula above, while the speed at $M = S_m = S_1 + \cdot 5d_r t_m$, and the speed at $O = S_o = S_1 - \cdot 5d_r t_m$, in which d_r is the rate of deceleration in feet per second per second due to the frictional resistance of the crane, and t_m is the time in seconds from the point M.

The distance which the crane will run under its own momentum $= D_m = \dfrac{\cdot 5S_m^2}{d_r}$.

The Period t_a.—The rate of acceleration $= a_r = \dfrac{S_1 \text{ (or } S_m)}{t_a}$, and the points in the acceleration curve are found by the formula $D = \cdot 5\, a_r t^2$.

The Period t_c.—In this period the crane comes to rest from the point O under its own friction supplemented, if necessary, by the friction of a brake.

The time taken to stop is $t_o = \frac{S_o}{d_r}$, in which t_o is the time in seconds from point O and S_o is the speed at that point. The distance from point O, at which it stops, is—

$$D_o = \frac{\cdot 5 S_o^2}{d_r}.$$

The intermediate points in the deceleration curve, measured from point O are found by the formula—

$$D_e = t_o(S_o - \cdot 5 d_r t_e).$$

Radius of Gyration and Force for Acceleration.—Radius of Gyration of plain cylinder $= \kappa = \frac{r}{\sqrt{2}}$.

Radius of Gyration of hollow cylinder $= \kappa = \sqrt{\frac{r_e^2 + r_i^2}{2}}$, in which r_e is the external and r_i the internal radius.

Torque required for acceleration $= T = \cdot 00027 \frac{\kappa^2 S_2 w}{t_a}$, in which $w =$ the weight of the cylinder. Dimensions being in inches, weight in lbs., torque in inch-lbs., and time in seconds.

A wheel such as that shown in section in Fig. 84 requires to be treated as a series of separate hollow cylinders.

If its weight is not known, and so has to be calculated, the following formulæ are most convenient :—

The weight of each part is taken out separately, thus—

$$W_1 = w_s \pi B_1(r_1^2 - r_2^2), \text{ and so on.}$$

The radius of gyration is then taken out separately for each part, thus—

$$\kappa_1 = \sqrt{\frac{r_1^2 + r_2^2}{2}}, \text{ and so on.}$$

The torque for acceleration is then—

$$T = \cdot 00027 \frac{S_2}{t_a}(W_1\kappa_1^2 + W_2\kappa_2^2 + W_3\kappa_3^2).$$

If the mean radius of gyration of the wheel as a whole is required, it is found thus—

$$\kappa = \sqrt{\frac{W_1\kappa_1^2 + W_2\kappa_2^2 + W_3\kappa_3^2}{W_1 + W_2 + W_3}}.$$

If the weight of the wheel is already known, and we only want its mean radius of gyration, this may be calculated thus—

$$\kappa = \sqrt{\frac{\cdot 5\{B_1(r_1^4 - r_2^4) + B_2(r_2^4 - r_3^4) + B_3(r_3^4 - r_4^4)\}}{B_1(r_1^2 - r_2^2) + B_2(r_2^2 - r_3^2) + B_3(r_3^2 - r_4^2)}}.$$

As fourth powers are somewhat inconvenient, while squares can be obtained direct from a slide rule, and as $a^4 - b^4 = (a^2 - b^2)(a^2 + b^2)$, the calculation may be very simply made in the manner shown in the following numerical example, taking Fig. 84 with dimensions in inches :—

$$\kappa = \sqrt{\cdot 5 \times \frac{6 \times (12^4 - 10{\cdot}5^4) + 1{\cdot}5 \times (10{\cdot}5^4 - 3^4) + 6 \times (3^4 - 1{\cdot}5^4)}{6 \times (12^2 - 10{\cdot}5^2) + 1{\cdot}5 \times (10{\cdot}5^2 - 3^2) + 6 \times (3^2 - 1{\cdot}5^2)}} = 9{\cdot}4.$$

$$\begin{array}{rrcrcrl}
12^2 = & 144 & & & & 144 & \\
10{\cdot}5^2 = & 110 & & & & 110 & \\
\hline
& 34 & \times 6 & = 204 & \times & 254 & = 51{,}500 \\
10{\cdot}5^2 = & 110 & & & & 110 & \\
3^2 = & 9 & & & & 9 & \\
\hline
& 101 & \times 1{\cdot}5 & = 152 & \times & 119 & = 18{,}000 \\
3^2 = & 9 & & & & 9 & \\
1{\cdot}5^2 = & 2{\cdot}25 & & & & 2{\cdot}25 & \\
\hline
& 6{\cdot}75 & \times 6 & = 40{\cdot}5 & \times & 11{\cdot}25 & = 456 \\
\hline
& & & 396{\cdot}5 & & & 69{,}956
\end{array}$$

$$\sqrt{\frac{\cdot 5 \times 69{,}956}{396{\cdot}5}} = 9{\cdot}4.$$

Slewing Motion of Jib or Derrick Crane.—The radius of gyration κ of the weight W may be taken as the radius of the point of suspension, and half this distance may be taken as the radius of gyration of the jib $= \kappa_j$. These radii are more conveniently taken in feet, the weights in tons and speed in feet per second. The torque required for acceleration, in foot-tons, is then—

$$T = \frac{(Wx^2 + W_j x_j^2) S_1}{32 \cdot 2 \, x a_r}.$$

In all the preceding calculations the rate of acceleration is regarded as a constant quantity. In reality it does not instantly rise to its full value at the commencement of the period t_a, and drop instantly to zero at the end of this period. It varies continually as the controller handle is moved from point to point, so the value taken in the preceding calculations is in reality the mean value.

CHAPTER X.

DESIGN OF CRANE STRUCTURES.

THE methods of ascertaining the static loads on the principal parts of crane structures have been given in the chapters describing the different types of cranes.

In addition to the static loads, we have to take into consideration the loads due to the driving forces of the machinery, and the additional loads imposed by the forces required for acceleration, and those developed by the application of the brakes.

As the loads on a crane are intermittent and variable, and liable to be applied suddenly, their true values are, to a certain extent, indeterminate, and generally exceed those found by the usual methods of calculation.

A fairly high factor of safety should, therefore, be adopted, or, what amounts to the same thing, a value should be taken for the working stress in the material, less than would be allowable if the loads could be predetermined with certainty and were merely static. The value to be assigned to the working stress must, therefore, be based as far as possible on data gained by experience. As a general rule, it may be suggested that, taking the normal working load on the hook and adding the probable forces due to acceleration or retardation and wind pressure, the working stress allowed should not exceed 45 per cent. of the elastic limit of the material, so allowing for the test load, which is usually 50 per cent. above the normal working load, and for indeterminate stresses.

On those portions of the structure which carry the machinery the dimensions of the parts should be determined not upon the strength of the material, but upon the amount of deflection permissible, for if this be excessive, as it sometimes will be if the parts are proportioned on the working stress given above, the alignment of the shafts will be upset with consequent excessive wear in the bearings, and wear and noise in the faster running portions of the toothed gearing.

Struts and Columns.—In a strut which is very short in proportion to its diameter the material is subject to a direct compressive stress only, and failure of the strut occurs when it is loaded up to the ultimate strength in compression of the material.

The ultimate strength in compression of a material like mild steel is not an exactly determinate quantity. Mild steel does not, like cast iron, definitely crack and break up when a given load is reached. Instead, after the point of elastic limit is passed, it squeezes out like dough. The ultimate strength in compression is, therefore, rather a conventional quantity, and in most text-books it is taken as being equal to the ultimate strength in tension.

On a short strut the breaking load is then simply—

$$W = Af_c.$$

If a strut is very long in proportion to its diameter, it will fail by bending over and buckling, and in this case the amount of load which causes failure will depend upon the elasticity of the material, and not upon its crushing strength. If a long strut of elastic material be bent, as in Fig. 86, it will exert an end force in the effort to straighten itself, and the value of this force will be constant irrespective of the amount to which the strut is bent, provided it is not bent beyond the elastic limit of the material. If, now, the strut be loaded with a weight equal to this end force and the strut be bent by the application of a horizontal force at C, on the removal of this force the strut will remain bent, as the weight and the end force exerted by the bent strut balance each other. If the weight be slightly reduced, the strut will straighten itself, while if it be slightly increased the strut will bend right over and collapse. The weight is in this case the maximum load consistent with stability, and its value, as determined by Euler, is—

$$W_m = \frac{\pi^2 EI}{l^2}. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which I is the least moment of inertia of the section, and l the virtual length of the strut, the dimensions being taken in inches, and the load in tons.

The virtual length l varies with the arrangement of the strut. If both ends are free, but guided in the direction of the load (Fig. 86) $l = L$. If the strut is flanged at one end and bolted firmly down while the other end is quite free (Fig. 87) $l = 2L$. If one end is bolted and the other free, but guided in the direction of the load (Fig. 88) $l = \frac{L}{\sqrt{2}}$. If both ends are bolted and fixed in direction (Fig. 89) $l = \cdot 5L$.

The maximum load per square inch consistent with stability —

$$p_m = \frac{W_m}{A} = \frac{\pi^2 E x^2}{l^2}. \quad . \quad . \quad . \quad . \quad (2)$$

The statement made above that the end force exerted by the bent strut is constant irrespective of the amount to which it is bent, provided it is not carried beyond the elastic limit, may be very simply verified. Referring to Fig. 86, the bending moment $M_b = W_m \delta$. M_b also equals fZ, in which f is the stress produced at the extreme edge of the section when the strut is bent to the deflection

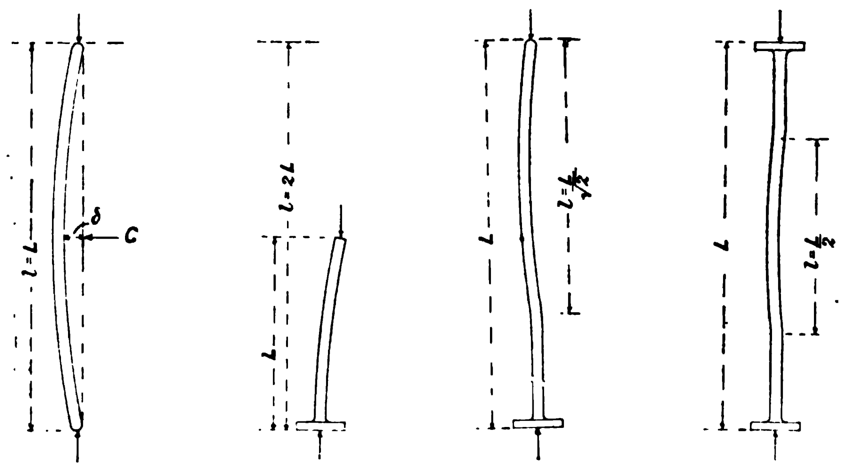


Fig. 86. Fig. 87. Fig. 88. Fig. 89.

δ and Z is the modulus of the section, which is equal to the moment of inertia divided by the distance e_y from the neutral axis to the extreme edge of the section.

As $W_m \delta = fZ$, $W_m = \frac{fZ}{\delta}$, and as the ratio $\frac{f}{\delta}$ is constant within the elastic limits of the material, the value of W_m is also evidently constant.*

* This can also be shown by a very simple experiment. Take an ordinary flexible steel rule, place it vertically on the desk, and hook a Salter's spring balance over the top. On pulling down the balance so as to bend the rule, the reading on the balance will be constant whether the rule be bent considerably or only slightly.

The author tried this with a rule, the average thickness of which was 0·033 inch, Euler's formula is not strictly correct, as it takes into account only the stress in the material due to bending, and omits the direct stress due to the load. Taking the latter into account, the end force which the bent strut will exert (and consequently the load W) will be less than that given by Euler, the difference being in the case of very long struts, so slight as to be negligible, but becoming appreciable as the length of strut is decreased.

When a strut is bent the stress due to its deflection $= f = \dfrac{W\delta}{Z}$, and the deflection $= \delta = \dfrac{M_b l^2}{\pi^2 EI}$.

Under the breaking load W_n the stress at the extreme edge of the section when bent =

$$f_s = \frac{W_n}{A} + \frac{W_n}{Z}$$

$$= W_n\left(\frac{1}{A} + \frac{\delta}{Z}\right)$$

$$= W_n\left(\frac{1}{A} + \frac{f_s l^2}{\pi^2 EI}\right).$$

$$W_n = \frac{f_s}{\dfrac{1}{A} + \dfrac{f_s l^2}{\pi^2 EI}}$$

$$I = Ax^2,$$

and the breaking load per square inch on the straight strut =

$$p_n = \frac{W_n}{A} = \frac{f_s}{\left(\dfrac{1}{A} + \dfrac{f_s l^2}{\pi^2 EAx^2}\right)A}.$$

$$= \frac{f_s}{1 + \dfrac{f_s l^2}{\pi^2 Ex^2}} \quad . \quad . \quad . \quad . \quad (3)$$

---

the width 1·08 inches, and the length over all 12·6 inches. The moment of inertia was, therefore,

$$\frac{1{\cdot}08 \times {\cdot}033^3}{12} = 0{\cdot}000003234,$$

and

$$W_m = \frac{3{\cdot}14^2 \times 30{,}000{,}000 \times 0{\cdot}000003234}{12{\cdot}6^2} = 6{\cdot}03 \text{ lbs.}$$

On bending it with the spring balance the reading was 6 lbs., irrespective of the amount to which the rule was bent.

This property may be utilised in cases where a spring is required to exert a constant pressure with varying deflection.

For any given material $\frac{f}{\pi^2 E}$ is a constant, so that the equation becomes—

$$p_n = \frac{f_e}{1 + c\left(\frac{l}{x}\right)^2}, \quad . \quad . \quad . \quad . \quad . \quad (4)$$

which is Rankine's formula for struts.

The point may be approached a little differently. When a loaded strut is straight, the material is subject to a direct stress $\frac{W}{A}$. If now the strut is bent until the fibres at the extreme edge of the section are stressed to the elastic limit, the difference between this and the direct stress is a measure of the force tending to straighten the strut. According to Euler's formula, when the straight strut is loaded with the breaking weight W_m the stress in the material is $p_m = \frac{\pi^2 E x^2}{l^2}$, and when the strut is bent to the elastic limit f_e, the end force exerted by it is still equal to W_m. But if we now take into account the direct stress, the total stress under these conditions will be $f_e + p_m$, which will not be possible, as it exceeds the elastic limit. We must, therefore, find a new value p_n for the load per square inch when the strut is straight such that when bent to the elastic limit, the total stress at the extreme edge of the section will be made up of two stresses having the same relationship as f_e and p_m above, but of such values that their total does not exceed the elastic limit of the material.

This total is then—

$$p_n = \frac{f_e \cdot \frac{\pi^2 E x^2}{l^2}}{f_e + \frac{\pi^2 E x^2}{l^2}}, \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which simplifies to the Rankine formula No. 4.

In cases where the Euler curve has been already worked out, Formula No. 5 is more convenient than No. 4, as the values of $\frac{\pi^2 E x^2}{l^2}$ can be taken direct from the Euler curve. As an example, take a strut in which $\frac{x}{l} = \frac{1}{250}$ and $E = 14{,}000$ tons per square inch. By formula No. 2 p_m will equal 2·211 tons per square inch, and this will also be the end force exerted by the strut when bent to the elastic limit if direct stress is neglected.

Taking the elastic limit at 14 tons per square inch, and allowing for direct stress, $p_n = \frac{14 \times 2{\cdot}211}{14 + 2{\cdot}211} = 1{\cdot}91$.

$14 - 1{\cdot}91 = 12{\cdot}09$, which is a measure of the end force exerted by the bent strut, this force being $\frac{2{\cdot}211 \times 12{\cdot}09}{14} = 1{\cdot}91$.

Gordon's formula for struts is of a similar form to Rankine's, but, apparently with the idea of saving the designer the trouble of working out the radius of gyration, he takes the least diameter of the section instead of the least radius of gyration. The construction of the formula is as follows :—

As before,
$$f_s = \frac{W_n}{A} + \frac{W_n \delta}{Z},$$

and
$$W_n = \frac{f_s}{\frac{1}{A} + \frac{f_s l^2}{\pi^2 EI}}$$

$$= \frac{f_s}{\frac{1}{A} + \frac{A f_s d_y}{{\cdot}5\,\pi^2 EZA} \cdot \frac{l^2}{d_y^2}}.$$

$$(I = Ze_y = {\cdot}5 Z d_y.)$$

$$p_n = \frac{W}{A} = \frac{f_s}{1 + \frac{A f_s d_y l^2}{{\cdot}5\pi^2 ZEd_y^2}}. \quad . \quad . \quad . \quad (6)$$

At this point the formula agrees exactly with Rankine's. Gordon, however, takes the value $\frac{A f_s d_y}{{\cdot}5\pi^2 EZ}$ as a constant depending on the material and shape of section, so that the formula becomes—

$$p_n = \frac{f_s}{1 + c\left(\frac{l}{d_y}\right)^2}. \quad . \quad . \quad . \quad . \quad (7)$$

The constant c is in this case made up of two values, $\frac{f_s}{\pi^2 E}$ being the constant for the material as in the Rankine formula, and $\frac{Ad_y}{{\cdot}5Z}$ being the constant for the shape of the section. In regard to the latter, it should be pointed out that this is only constant when the proportions of the cross-section are kept constant as well as the shape. Therefore, if this formula be applied, with the same constant, to sections having the same shape, but different proportions, the results obtained will be discordant.

Taking a steel having a modulus of elasticity of 14,000 tons per square inch, an elastic limit of 14 tons per square inch and a crushing strength of 30 tons per square inch, the following curves have been calculated.

Fig. 90 shows in the upper curve the value of p_m from Euler's formula No. 2 for struts up to a length of 250 times the radius of gyration. In this case the formula is not applicable to struts having a length less than about 100 times the radius of gyration, as at this point the load is equal to the elastic limit of the steel.

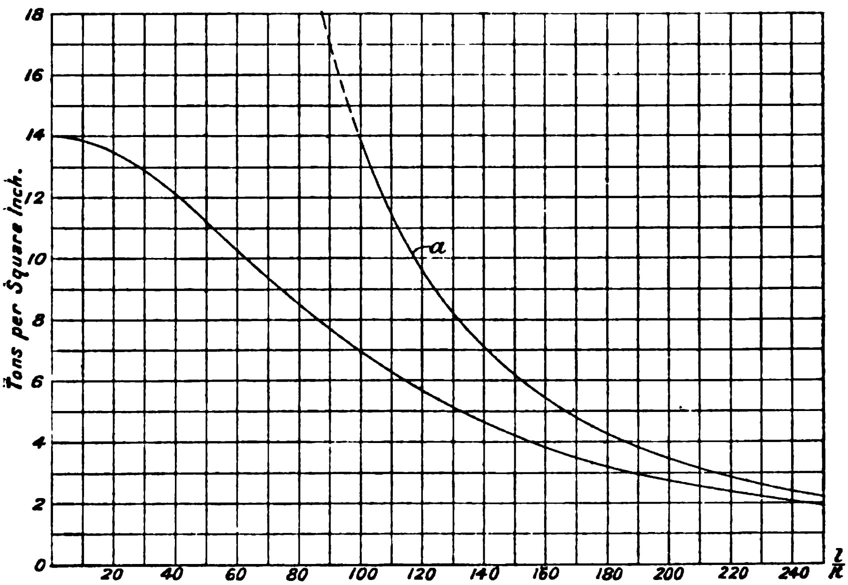


Fig. 90.—a = Euler.

The lower curve shows the load per square inch on the straight strut, which, when the strut is bent, will bring the material at the extreme edge of the section on the compression side up to the elastic limit. The values in this curve have been calculated from Rankine's formula No. 4, taking $f_s = f_e = 14$ tons per square inch, and $c = \frac{14}{\pi^2 \times 14{,}000} = \frac{1}{9{,}870}$.

This curve is not satisfactory. At the extreme right it gives, as with Euler, the maximum load consistent with stability—that is to say, the breaking load of the strut, while at the left hand it gives the load which will stress the material to the elastic limit, which, although it is practically the yield point of the strut, is not the load which corresponds with the crushing strength of the material. Therefore, we cannot use this curve with a constant factor of safety. For instance, taking a factor of safety of 5, the safe stress for a strut at the extreme right of the curve would be $\frac{1 \cdot 91}{5} = \cdot 38$, which is truly one-fifth of the breaking load, while at the extreme left of the curve it would be $\frac{14}{5} = 2 \cdot 8$. The breaking strength at this point is, however, usually taken as the crushing strength of the material, in this case 30 tons per square inch, so that if we take 2·8 as the safe load, we are really allowing a factor of safety of $\frac{30}{2 \cdot 8} = 10 \cdot 7$ instead of 5.

To get over this difficulty the Rankine formula is generally used with $f_s = f_c$, the crushing strength of the material instead of the elastic limit.

Substituting this, and taking it at 30 tons per square inch, but retaining the same value for c as before, as is frequently done, the curve in Fig. 91 is obtained. Euler's curve being also laid down for comparison. This curve is no better than the previous one, for while the curve is correct at the left hand, it gives a result about twice as high as Euler at the right hand, whereas we know that it must be something less than Euler. The reason for the discrepancy in this case is that by taking $f_s = 30$, while still retaining the same value for c, the assumption is introduced that at the moment of failure the material is stressed both to its elastic limit and its crushing strength simultaneously, which is obviously impossible.

Both curves are, therefore, equally unsatisfactory. We may obtain a rough practical solution of the difficulty by separating the direct and bending stresses at each point on the curve, and applying to each its own factor of safety.

Referring to Fig. 90, the stress at the commencement of the curve is 14 tons per square inch, whereas the crushing strength is 30 tons. With a factor of safety of 5 the safe load is 6 tons, and $\frac{14}{6} = 2 \cdot 33$. If, in conjunction with Fig. 90, we take a factor of safety of 2·33 for a strut, which is so short that there is no bending, and 5 for one so long that the direct stress is negligible, the load so obtained will be truly one-fifth of the breaking load of the strut for both extremes of the curve. The factor of safety at any intermediate point on the curve will be $S = \frac{2\cdot33\, p_n + 5f_b}{p_n + f_b}$, f_b in this case $= 14 - p_n$.

The curve (Fig. 92) gives the values found by this equation, these being the values to be used for the factor of safety at any

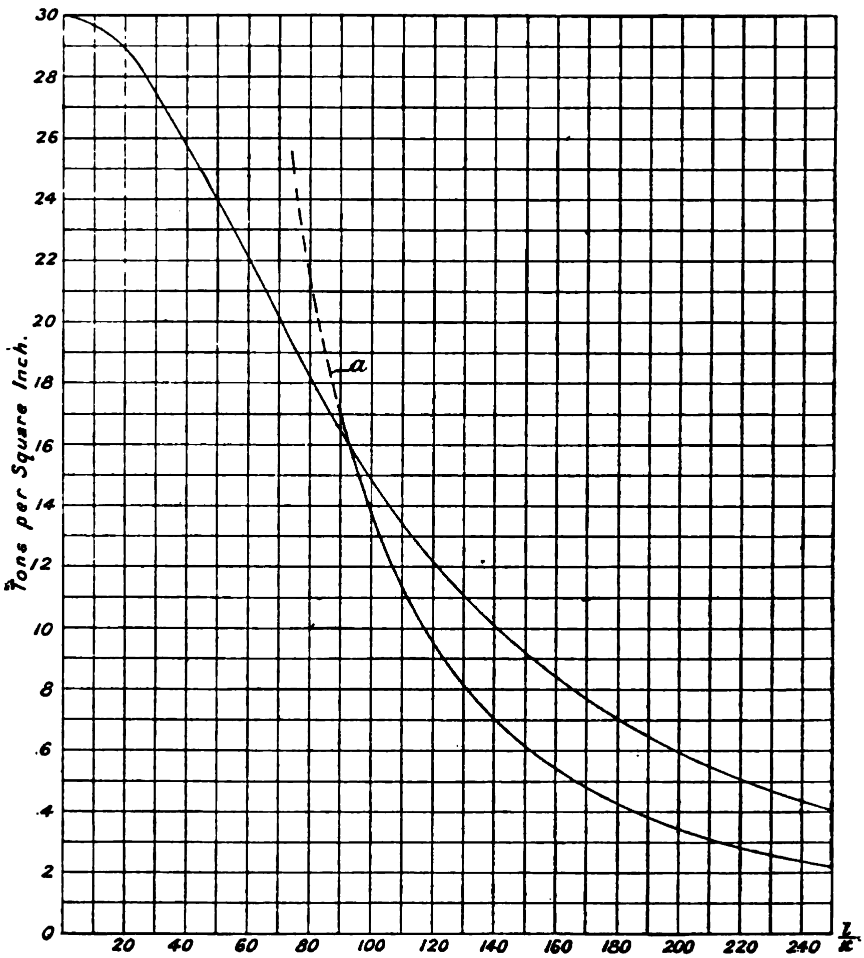


Fig. 91.—a = Euler.

point in the curve (Fig. 90), in order to find the load which will be one-fifth of the breaking load at that point. From these values the curve (Fig. 93) has been plotted, showing the safe loads for a factor of safety of 5, and the Euler curve $\div$ 5 has been plotted also.

From this the curve of breaking loads, Fig. 94, has been taken for comparison with Figs. 90 and 91.

If in equation No. 4 f_s is taken at 30 tons, and

$$c = \frac{30}{\pi^2 14,000} = \frac{1}{4,606},$$

the dotted curve in Fig. 94 is obtained

This curve is more satisfactory than Figs. 90 and 91. It is, however, not quite satisfactory, as it embodies the assumption that at the moment of failure, whether by direct stress or bending, the material is stressed up to its crushing strength. It will be

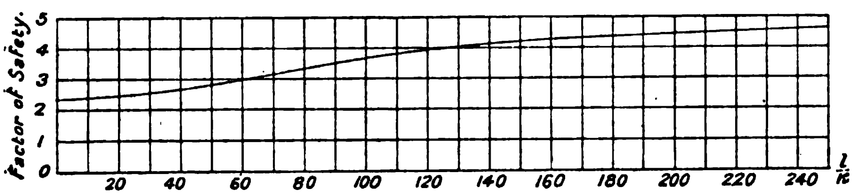


Fig. 92.

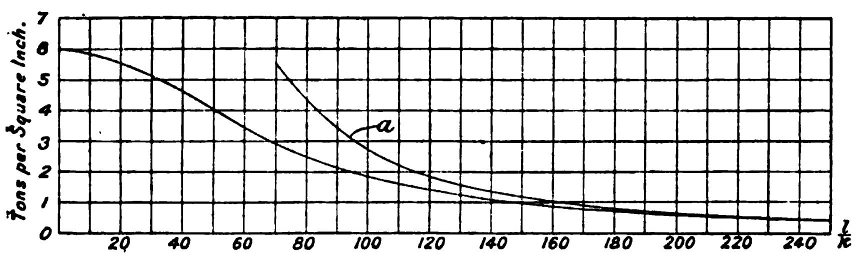


Fig. 93.—a = Euler.

noted, however, that the difference between this and the curve shown in full lines is very trifling, and its calculation involves much less work.

A built-up column with lattice bracing may fail from one of three causes. (1) The column, as a whole, may fail. (2) The corner members may fail between the points at which they are braced. (3) The metal in the members may, through insufficient thickness, wrinkle under the load to which they are subjected, and so lead to failure of the column. In each of these cases, therefore, the appropriate strut formula must be complied with, in order to ensure that the column throughout its parts is of sufficient strength for the load which it has to carry.

Cases (1) and (2) are easily dealt with, but in case (3) we are confronted by the difficulty of assigning values to l and the moment of inertia.

This point has been investigated very fully by Professor W. E. Lilly, of Trinity College, Dublin.

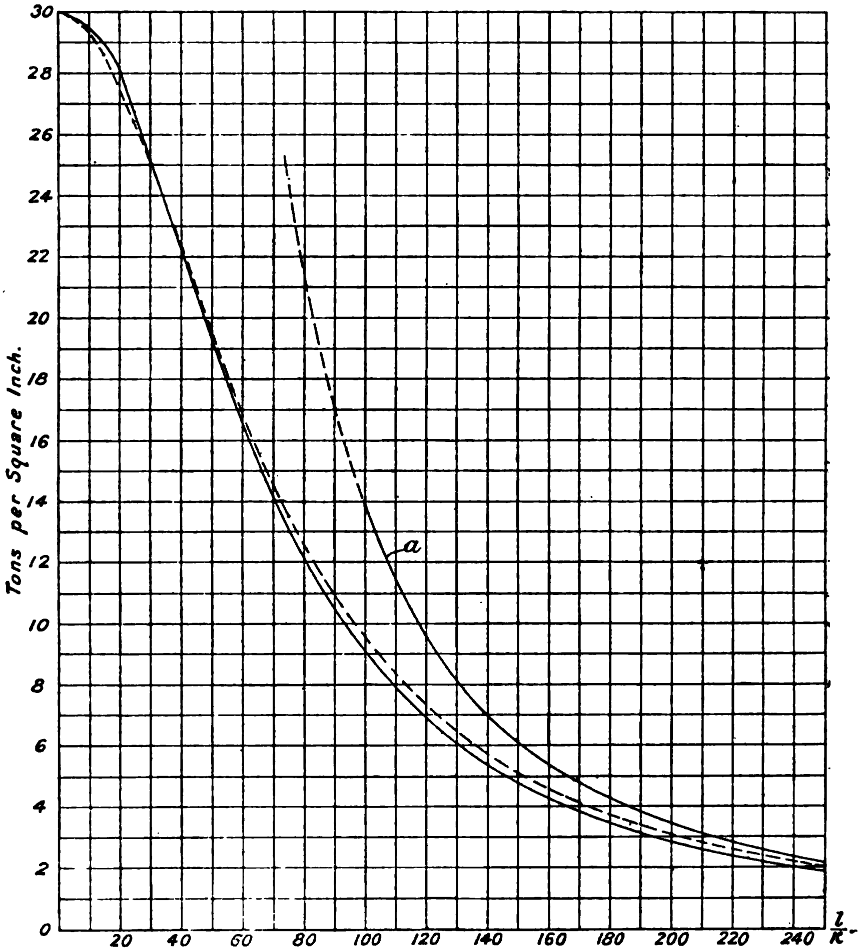


Fig. 94.—a = Euler.

Professor Lilly has carried out a very large number of experiments on the breaking loads of struts, and has published the results, together with formulæ based upon them.*

* *Design of Plate Girders and Columns*, by W. E. Lilly, M.A. (Chapman & Hall; also *Proceedings Inst. Mech. E.*, June 1905, "Strength of Columns," by W. E. Lilly.

The result of his researches may be summarised as follows :— When a short metal tube is subjected to direct thrust, it fails by wrinkling or secondary flexure, the tube breaking up into a series of waves. With a constant sectional area the larger the diameter of the tube and the smaller the thickness of metal the less is the load which produces failure. The equation for this load is—

$$f_l = \frac{f_c}{1 + k_3 \frac{x}{t}} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

in which f_l = the limiting load per square inch on a column of one wave length, k_3 = a constant which is further dealt with later, and t is the thickness of metal in the section.

If, now, in the calculation of a strut we use the Rankine formula No. 4 with $f_s = f_l$ and $c = \frac{f}{\pi^2 E}$, we shall obtain a figure for the breaking load of the strut which takes secondary flexure into account.

With a view of making one calculation instead of two, f_l can be embodied in the Rankine formula, thus—

$$p = \frac{f_l}{1 + \frac{f_l}{\pi^2 E} \cdot \left(\frac{l}{x}\right)^2}$$

$$= \frac{\dfrac{f_c}{1 + k_3 \frac{x}{t}}}{1 + \left(\dfrac{\dfrac{f_c}{1 + k_3 \frac{x}{t}}}{\pi^2 E}\right) \cdot \left(\frac{l}{x}\right)^2}$$

$$= \frac{f_c}{1 + k_3 \frac{x}{t} + \frac{f_c}{\pi^2 E} \cdot \left(\frac{l}{x}\right)^2}$$

$$= \frac{f_c}{1 + k_3 \frac{x}{t} + c \left(\frac{l}{x}\right)^2} \quad . \quad . \quad . \quad . \quad (9)$$

It will be noted that in equation No. 9 $c = \frac{f_c}{\pi^2 E}$.

In the case of I beams in which the flanges are thicker than the web, the formula is modified, thus—

$$p_r = \frac{f_c}{1 + \sqrt{\frac{t_2}{t_1} \cdot k_3 \frac{x}{t_2} + c\left(\frac{l}{x}\right)^2}} \quad . \quad . \quad . \quad (10)$$

In which t_2 and t_1 are the thickness of the flange and web respectively, as in the book of British Standard Sections.

The value of k_3 for different shapes of section is as follows :—

(1) For hollow circular cross-section $k_3 = \frac{50 f_c}{E}$.

(2) For hollow square cross-section $k_3 = \frac{60 f_c}{E}$.

(3) For I section . . . . $k_3 = \frac{70 f_c}{E}$.

It has been shown above that for a constant cross-sectional area an increase of diameter lessens the resistance to secondary flexure, while it increases the resistance to primary flexure. There is a certain proportion of thickness to diameter at which the two resistances are equal, and which gives the maximum strength for a given weight of strut.

The proportions for maximum strength may be ascertained in the following manner.

For round hollow sections in which the material is thin, the mean diameter $d = \kappa\sqrt{8}$ nearly, the sectional area $A = \sqrt{8}\pi\kappa t$, and $t = \frac{A}{\sqrt{8}\pi x}$.

Referring to equation No. 9, the value of p_r will be a maximum when $k_3\frac{x}{t} + c\frac{l^2}{x^2} = k_3\frac{\sqrt{8}\pi x^2}{A} + c\frac{l^2}{x^2}$ is a minimum. Calling this y and differentiating, the $dc = k_3\frac{\sqrt{8}\pi x}{A} - c\frac{2l^2}{x^3}$, and when y is a minimum—

$$x = \sqrt[3]{\frac{l^2 ct}{k_3}} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

For square hollow thin sections the mean diameter * $s = k\sqrt{6}$ nearly, and $A = \sqrt{6} \cdot 4kt$. The equation for the proportions of a square hollow strut of maximum strength is the same as No. 11.

None of the preceding formulæ produce a curve which agrees

* = Mean length of side.

exactly with the curve of breaking load as found by experiment, for those lengths of strut most commonly used in practice. The majority of the published results of experiments relate to struts of cast or wrought iron, and are, therefore, of little practical use to the designer of cranes, as struts of these materials do not enter into his work.

The author prefers, therefore, to rely on Professor Lilly's experiments, as, although made on comparatively small specimens, they were made on the material, mild steel, with which crane makers have to deal.

The particulars of the steel were, tensile strength, 32 tons per square inch; crushing strength taken as 35·7 tons; and modulus of elasticity 13,390 tons.

In Fig. 95 six diagrams are given. The first of these is for a solid column, and the remainder are for hollow columns having different thicknesses of material.

In diagram **A** two calculated curves are given, the Euler and the Rankine, while on the other diagrams the curve obtained from Professor Lilly's formula, No. 9, is also plotted.

In addition, there is laid down on each diagram the curve of breaking loads found by actual test.

A formula which would agree with the experimental breaking load curve would be too complicated to be of any practical service, and it is, therefore, preferable to use the formulæ already given, making such allowances as the experimental curves show to be necessary.

Fig. 96 gives a diagram of three dimensions, showing the actual breaking loads of solid and hollow columns of circular section.

This has been worked out by the author from the six experimental curves in Fig. 95, and enables the breaking load of a column to be read off direct without calculation. As an example, take the case of a column in which $\frac{l}{x} = 72$ and $\frac{x}{t} = 4{\cdot}4$. On Fig. 96 draw the line ab at $\frac{l}{x} = 72$, erect verticals at b and c, and join de. From point f at $\frac{x}{t} = 4{\cdot}4$ draw a vertical cutting de in g. The length fg then represents the breaking load on the vertical scale, in this case 16·75 tons per square inch.

For columns of other than circular section the breaking load may be calculated from that found from Fig. 96. Thus, taking a column of the same proportions but square, c, either for the

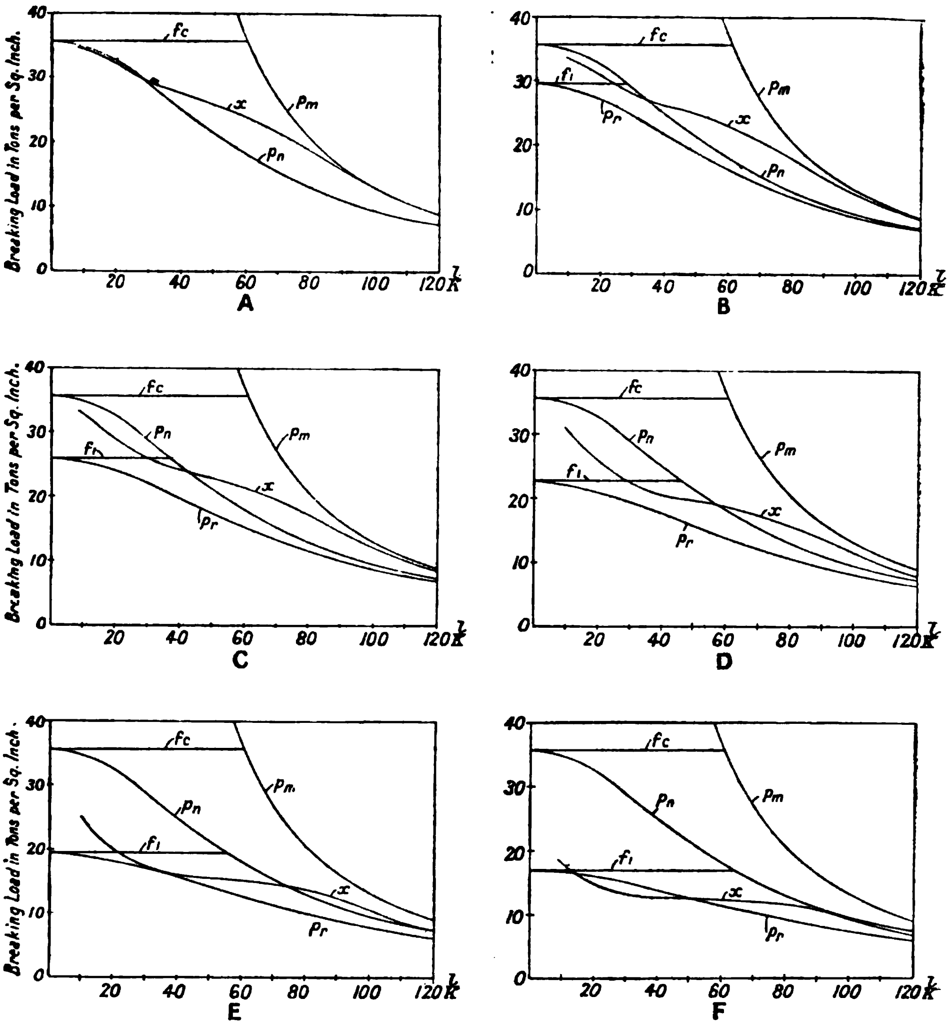


Fig. 95.—Breaking Loads of Round Columns.

E = 13,390 tons per sq. inch.

f_c = 35·7 tons (80,000 lbs.) per sq. inch.

$p_m = \pi^2 E\left(\frac{\kappa}{l}\right)^2$ = Euler's Equation No. 2.

$p_n = \frac{f_c}{1 + c\left(\frac{l}{\kappa}\right)^2}$ = Rankine's Equation No. 4.

$c = \frac{f_c}{\pi^2 E}$

$k_3 = \frac{50 f_c}{E}$.

$f_l = \frac{f_c}{1 + k_3\frac{\kappa}{t}}$ = Prof. Lilly's Equation No. 8.

$p_r = \frac{f_c}{1 + k_3\frac{\kappa}{t} + c\left(\frac{l}{\kappa}\right)^2}$ = Prof. Lilly's Equation No. 9.

x = Curve of actual breaking loads from tests.

A. solid column.

B. Hollow column in which $\frac{\kappa}{t} = 1·5$.

C. ,, ,, ,, = 2·778.

D. Hollow column in which $\frac{\kappa}{t} = 4·386$

E. ,, ,, ,, = 6·25.

F. ,, ,, ,, = 8·5.

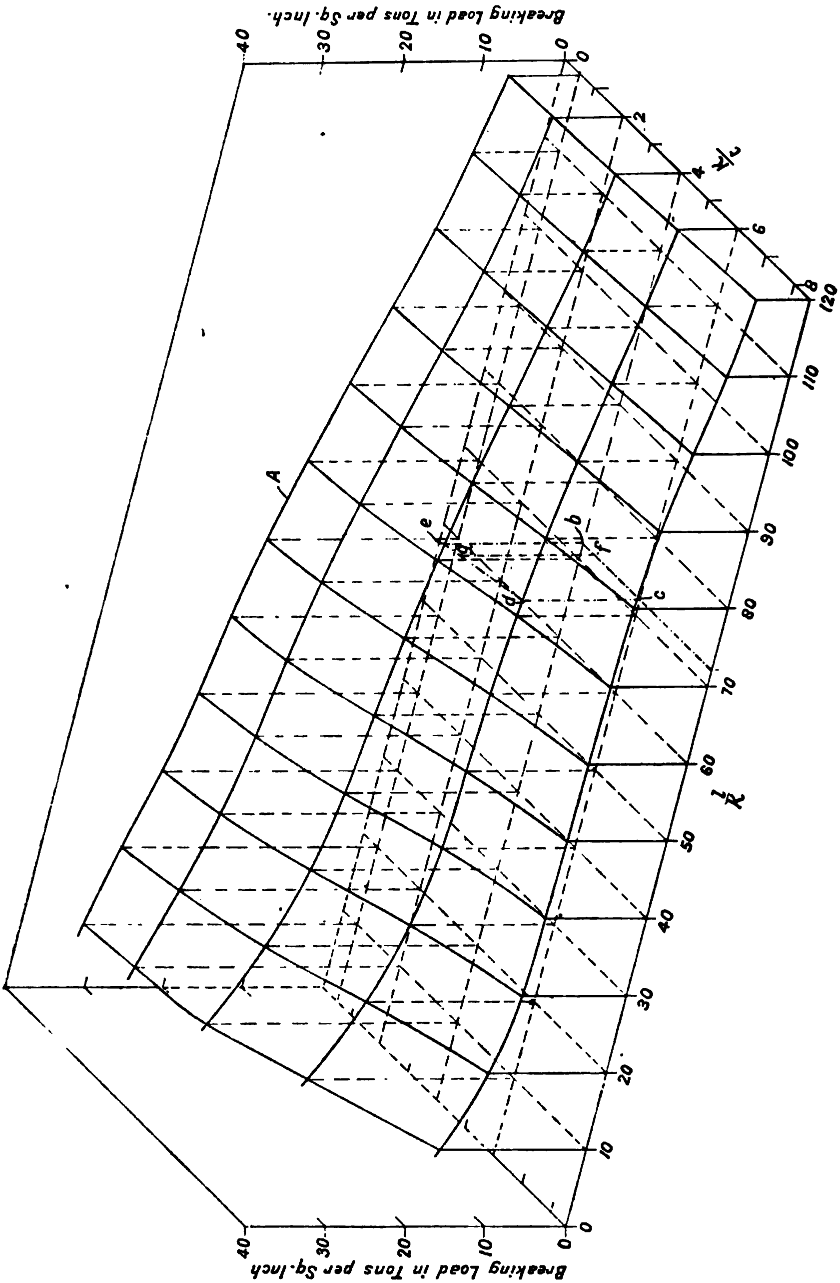


Fig. 93.—Breaking Loads of Round Columns—*A* = Solid Column.

round or square column $= \dfrac{35{\cdot}7}{\pi^2 \times 13{,}390} = \dfrac{1}{3{,}702}$, k_3 for the round column $= \dfrac{50 \times 35{\cdot}7}{13{,}390} = \dfrac{1}{7{\cdot}5}$, and for the square column $= \dfrac{60 \times 35{\cdot}7}{13{,}390}$ $= \dfrac{1}{6{\cdot}25}$.

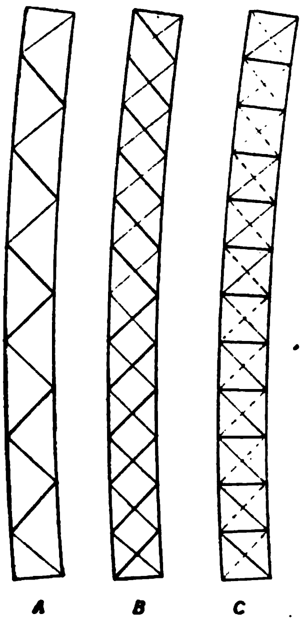


Fig. 97.

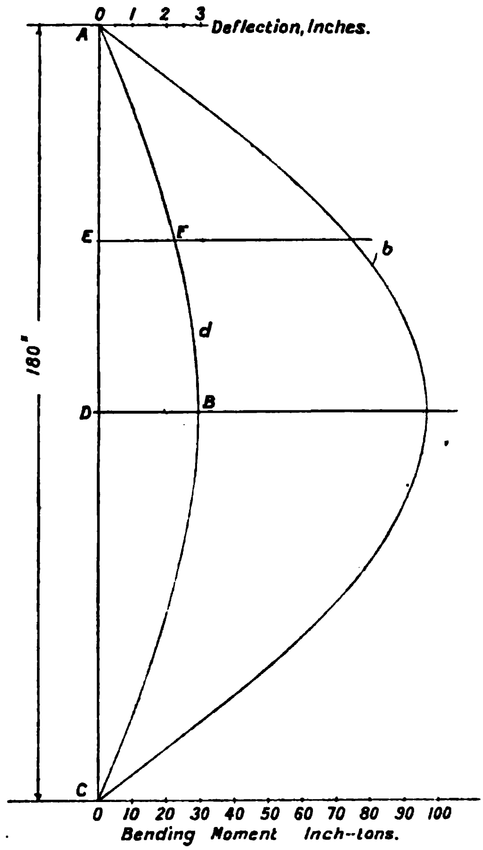


Fig. 98.—d = Deflection, b = Bending Moment.

The breaking load of the square column will then be—

$$16{\cdot}75 \times \frac{1 + \dfrac{4{\cdot}4}{7{\cdot}5} + \dfrac{72^2}{3{,}702}}{1 + \dfrac{4{\cdot}4}{6{\cdot}25} + \dfrac{72^2}{3{,}702}} = 16{\cdot}1 \text{ tons per sq. inch.}$$

Owing to the fact that the calculated curves do not coincide at all points with the observed curves, the equation No. 11 generally gives only approximate proportions for the maximum strength. It provides, however, a point for the designer to start from, and by obtaining points on each side of this, from Fig. 96, and plotting them, the proportions for maximum strength may be readily found.

The three systems of latticing most commonly used for columns are shown in Fig. 97. Diagram *A* shows a single Warren system in which the alternate diagonals are subject to tension and compression. Diagram *B* shows a double Warren system, in wh˙ch each set of diagonals takes half the load. In systems *A* and *B* the stresses in the diagonals are reversed if the column is bent the opposite way to that shown.

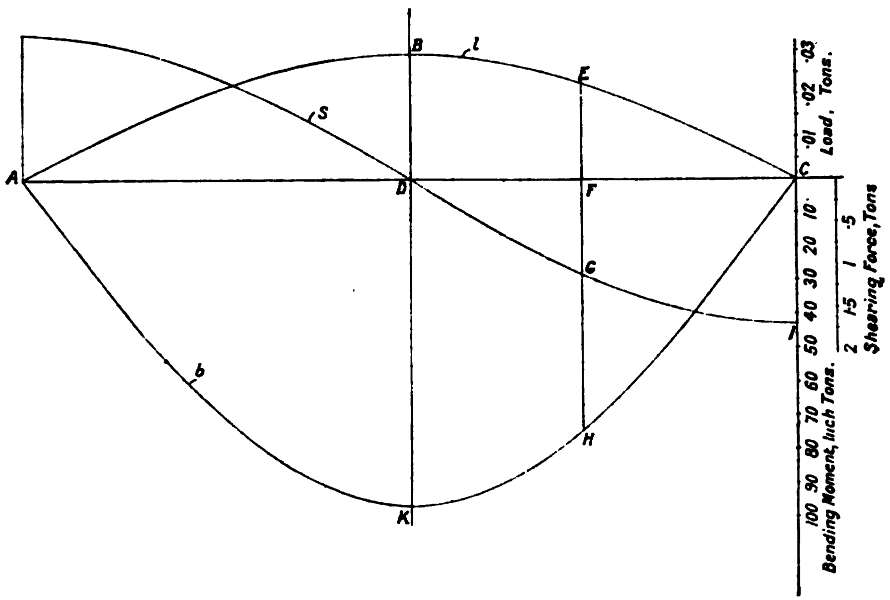


Fig. 99.—*b* = Bending moment, *s* = Shearing force, *l* = Load.

In diagram *C* the horizontal members are subject to compression whether the column is bent to the right or left. When the column is bent as shown, the ties in full lines come into action, while those shown dotted are idle, these conditions being reversed when the column is bent the other way.

When a column is straight and the load central, there can be no stress on the latticing, stress only occurring when the column is bent.

The dimensions of the lattice work have, therefore, to be determined from the curve of deflection of the column, and should be such that when the limiting stress is reached in the flanges it is also reached in the lattice bars. When a column is bent under the application of a load W the bending moment at the centre (see Fig. 98) is $W \times$ **BD**, and at any intermediate point such as **F** it is $W \times$ **EF**. The curve to which the column bends must be such that the stress at any point due to the bending moment is balanced by the stress at that point due to the deflection and the form of curve which satisfies this condition is the sine curve. The curve of deflection being a sine curve, the curve of bending moments will obviously be of the same form.

The sine curves may easily be laid out by dividing the half length of the column into 90 equal parts, regarding these as degrees, and multiplying the maximum deflection, which is at 90, by the sine of the respective divisions. Thus, at the forty-fifth division the height of the curve is $\cdot 707 \times$ **BD**. Having found the curves of deflection and bending moment due to the load, the diagram of the loading which, applied to the column as a beam (see Fig. 99), would bend it to the same deflection, may be determined. From this load diagram the stresses on the latticing may be ascertained in the same way as for an ordinary girder.

As all the curves are sine curves, it is only necessary to calculate their maximum values, the intermediate values being found by the aid of a table of sines.

The maximum values may be determined by the following equations :—

The deflection (Fig. 98)—

$$\mathbf{BD} = \delta = \frac{M_b l^2}{\pi^2 EI} \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$(M_b = fZ)$$

The shearing force (Fig. 99)—

$$\mathbf{CI} = S_5 = \frac{M_b \pi}{l} \quad . \quad . \quad . \quad . \quad . \quad (13)$$

The load (Fig. 99)—

$$\mathbf{BD} = L_2 = \frac{S_5 \pi}{l} \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The area of the shearing force diagram **CID** is equal to the bending moment **DK**, and the area of the load diagram **BCD** is equal to the shearing force **CI**. At any intermediate point, such as **F**, the area

BEDF of the load diagram is equal to the shearing force **FG**, and the area **FCIG** of the shearing force diagram is equal to the bending moment **FH**. The accuracy of the work may, therefore, be quickly checked with a planimeter.

As an example, we will take a column 180 inches long and of the cross-section shown in Fig. 100.*

It is assumed to be sufficiently stayed sideways to compel bending to take place in the plane **YY**. The properties of the section, the latticing being neglected, are—Sectional area, 2 square inches; moment of inertia, 15·16; radius of gyration, 2·753; and modulus of section, 5·05. By equation No. 2 the maximum stress consistent with stability would be—

$$f_m = \frac{3 \cdot 14^2 \times 13{,}390 \times 2 \cdot 753^2}{180^2} = 30 \cdot 91 \text{ tons per sq. inch.}$$

This, however, does not take into account direct stress. In order to allow for this imagine the column to be composed of two portions, the one resisting the direct stress, and being otherwise inert, while the other when bent exerts end force and conforms to Euler's equations. Further, as the close approximation of the full and dotted curves in Fig. 94 shows that only a negligible error is introduced by the assumption, it will be assumed that failure of the column takes place when it is bent to the crushing strength f_c of the material. Taking f_c at 35·7 tons per square inch, the area which takes the direct stress may be obtained from equation No. 5.

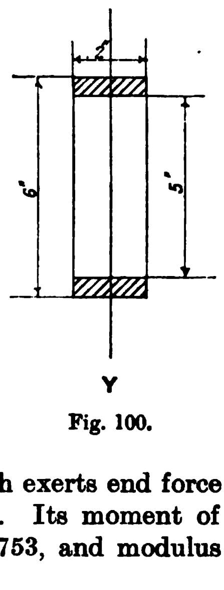


Fig. 100.

$$\frac{35 \cdot 7 \times 30 \cdot 91}{35 \cdot 7 + 30 \cdot 91} \times \frac{2}{35 \cdot 7} = 0 \cdot 9282 \text{ sq. inch.}$$

Then the area of that portion of the column which exerts end force when bent $= 2 - 0 \cdot 9282 = 1 \cdot 0718$ square inches. Its moment of inertia is 8·126, radius of gyration, as before, 2·753, and modulus of the section 2·71.

The dimensions of the latticing are based upon this portion of the column.

f_m will still be 30·91, and $W_m = 30 \cdot 91 \times 1 \cdot 0718 = 33 \cdot 12$ tons.

* This is merely a simple section for the purpose of illustrating the application of the formulæ.

This figure may be checked by means of equation 4, which applies to the whole column—

$$W_n = \frac{35{\cdot}7 \times 2}{1 + \dfrac{35{\cdot}7 \times 180^2}{3{\cdot}14^2 \times 13{,}390 \times 2{\cdot}753^2}} = 33{\cdot}12 \text{ tons,}$$

The bending moment will be $f_c Z = 35{\cdot}7 \times 2{\cdot}71 = 96{\cdot}76$, and the deflection—

$$\delta = \frac{96{\cdot}76 \times 180^2}{3{\cdot}14^2 \times 13{,}390 \times 8{\cdot}126} = 2{\cdot}92 \text{ inches.}$$

The curve of deflection and the curve of bending moments derived from it are shown in Fig. 98.

The shearing force =

$$S_5 = \frac{96{\cdot}76 \times 3{\cdot}14}{180} = 1{\cdot}688 \text{ tons.}$$

and the height of the load diagram at the middle of the column =

$$L_2 = \frac{1{\cdot}688 \times 3{\cdot}14}{180} = {\cdot}0295 \text{ ton.}$$

The curves of load and shearing forces are shown in Fig. 99.

To this load should be added the loads, if any, due to wind pressure or other causes, and if the column is not vertical the loads due to the weight of the column itself.* From the complete load diagram so found the dimensions of the latticing may be obtained by means of a stress diagram, the same as for an ordinary latticed girder. As the column is latticed on both sides, each set of latticing takes half the load.

On working out a column in detail in the above manner, it will be found that the dimensions of the lattice bars diminish towards the middle of the column. The usual practice, however, is to make the lattice bars equal throughout, and of the maximum size required at the ends, as the saving of material effected by diminishing them towards the middle of the column is more than counterbalanced by the extra cost of labour which would be involved by a departure from uniformity. In the case of small columns, the size of the lattice bars is generally calculated direct from the maximum shearing

* If there are such additional loads the area of the flanges must be increased proportionately.

force, without going to the refinement of making a stress diagram. The stress on the lattice bars is in this case equal to the shearing force $\times$ the secant of the angle which the bar makes with the direction of the shearing force. In calculating the size of the lattice bars in compression it should be noted that, as they are rivetted on the flat, they may be taken as struts fixed at both ends (see Fig. 89), their virtual length being then taken at half their actual length.

When the load on a column is not central, the breaking load is inversely proportional to the amount of eccentricity, the deflection at the breaking point being taken into consideration.

The breaking load is then—

$$W_p = \frac{W_n}{1 + \dfrac{d_4}{\dfrac{Z}{A} + \delta}} = \frac{f_c}{\dfrac{1}{A} + \dfrac{d_4 + \delta}{Z}} \quad . \quad . \quad . \quad (15)^*$$

* Equation 15 is based on failure of the column taking place when it is bent to a given compressive stress, f_c.

For a centrally loaded column $f_c = \dfrac{W_n}{A} + \dfrac{W_n\delta}{Z}$, and for the eccentrically loaded column $f_c = \dfrac{W_p}{A} + \dfrac{W_p(d_4 + \delta)}{Z}$.

Then
$$\frac{W_n}{A} + \frac{W_n\delta}{Z} = \frac{W_p}{A} + \frac{W_p(d_4 + \delta)}{Z}.$$

$$\frac{W_n}{W_p} = \frac{\dfrac{1}{A} + \dfrac{d_4 + \delta}{Z}}{\dfrac{1}{A} + \dfrac{\delta}{Z}} = \frac{Z + A\delta + Ad_4}{Z + A\delta} = 1 + \frac{Ad_4}{Z + A\delta}.$$

$$W_p = \frac{W_n}{1 + \dfrac{Ad_4}{Z + A\delta}} = \frac{W_n}{1 + \dfrac{d_4}{\dfrac{Z}{A} + \delta}}.$$

Also
$$W_p = \frac{f_c}{\dfrac{1}{A} + \dfrac{d_4 + \delta}{Z}}.$$

This assumes that the curve of deflection is the same in both cases, which is not strictly correct, but only introduces a small error. If the deflection is so small in comparison with the eccentricity as to be negligible,

$$f_c = \frac{W_p}{A} + \frac{W_p d_4}{Z}.$$

and

$$W_p = \frac{f_c}{\dfrac{1}{A} + \dfrac{d_4}{Z}}.$$

in which d_4 is the distance from the centre of the column to the centre of the load, in inches.

The most important compression member dealt with in crane work is the jib, in the design of which the two views require different treatment. In side view the jib should be treated as a column with rounded ends (Fig. 86), subject to compression and to bending stress due to wind pressure and its own weight, the maximum bending moment being at the middle. In plan the jib should be treated as a column fixed at the bottom and free at the top (Fig. 87), subject to compression and to bending stress due to wind pressure, and to the acceleration and retardation of the slewing motion, the maximum bending moment being at the foot of the jib. If the jib were to be lowered away till it became horizontal, the bending stress due to its own weight would be a maximum, and that due

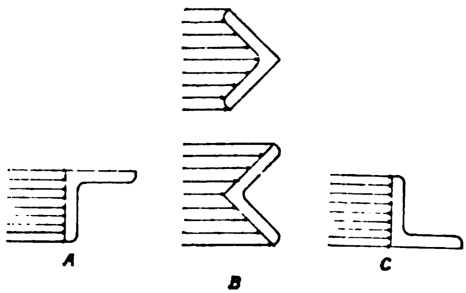


Fig. 101.

to wind pressure would be zero, while if heaved up to a vertical position these conditions would be reversed. At any intermediate position the bending stress due to its own weight would vary with the cosine of the angle. If the jib offered a plane surface of constant area to the wind, the bending stress due to wind pressure would vary with the square of the sine of the angle. Being latticed, however, the area which it offers is not constant at all angles. In Fig. 101 the area offered by an angle bar when the jib is horizontal, and when it is vertical, is shown at *A* and *C* respectively, while the area presented when the jib is at an angle of 45° is shown at *B*. The two angle bars shown at *B* both offer the same projected area of surface, but the upper angle bar would offer the greater resistance to the wind, as it has been found by experiment that a concave surface offers a greater resistance to the wind than a convex one. It is only a small portion of the latticing which offers this increased area.

In important cases the correct projected area may be worked out for several positions of the jib, and from these a curve may be laid out. In the generality of cases, however, the author considers that the effect of the slightly increased surface at intermediate angles may be allowed for by regarding the bending stress due to wind pressure as varying with the sine of the angle instead of the square of the sine. In order to show the extent of the allowance

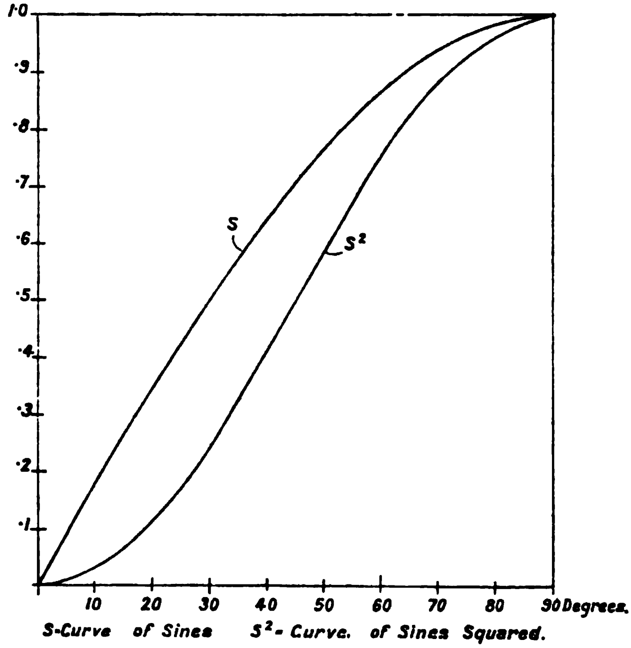


Fig. 102.

made by this assumption the two curves in Fig. 102 have been plotted.

The total distributed load in tons on the jib, in side view, due to the combined effects of wind pressure and its own weight is then—

$$L_3 = P_5 \sin a + W_j \operatorname{cosin} a.$$

in which P_5 is the total pressure in tons which would be exerted by the wind on the jib if vertical, W_j is the weight of the jib in tons, and a the angle which the jib makes with the horizon.

By differentiation the angle of the jib at which L_3 is a maximum is found by the equation—

$$\text{Tan } a = \frac{P_5}{W_j}.$$

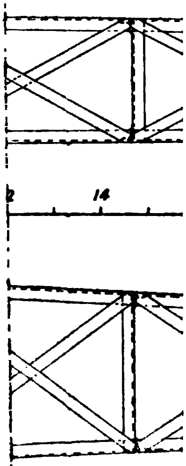
Fig. 103 gives an example of a 70-foot latticed steel jib for a 5-ton derrick crane. The weight of the jib is 3 tons. With 5 tons on the hook the compression on the jib is 15¼ tons. The area of one side, as shown in the upper view, is 72 square feet, so that the whole area exposed to side wind is 144 square feet. The area of the top latticing, as shown in the lower view, is 76 square feet, so that the total area exposed to end wind is 152 square feet.

Fig. 104 is an example of a steel tower for a cableway, the vertical load on which was about 80 tons.

Girders.—In cranes of the cantilever type, such as Fig. 37, the bracing of the cantilever is usually either on the Linville system (Fig. 105) or on the Warren system (Fig. 111).

In the former the length of the girder is divided into equal panels, the struts are vertical, and the ties inclined, their inclination increasing towards the point of support of the girder. The form of the stress diagrams is given in Figs. 106 and 107. The weight of the girder is taken as being supported at the points A, B, C, etc. In order to ascertain the maximum stress on any member it is advisable to make a separate diagram with the rolling load at each of the positions A, B, C, etc. The maximum stress on the top and bottom members occurs when the load is at A, the maximum on any tie occurs when the load is hanging at the foot of that tie, while the maximum on any strut occurs when the load is hanging at the foot of the next tie. Thus the strut No. 8 is subject to maximum stress when the load is hanging at the foot of No. 7 tie.

The diagram (Fig. 106) shows the stresses due to the weight of the girder itself. On the vertical line at the right of the diagram the distances A, B, C, etc., represent to any convenient scale the portion of the weight of the girder supported at the correspondingly lettered points on the girder diagram. No. 1 line is then drawn parallel to No. 1 member on the girder diagram (Fig. 105), and so on, the lengths of the lines on the stress diagram representing the loads on the correspondingly numbered members on the girder diagram.



Crane. All

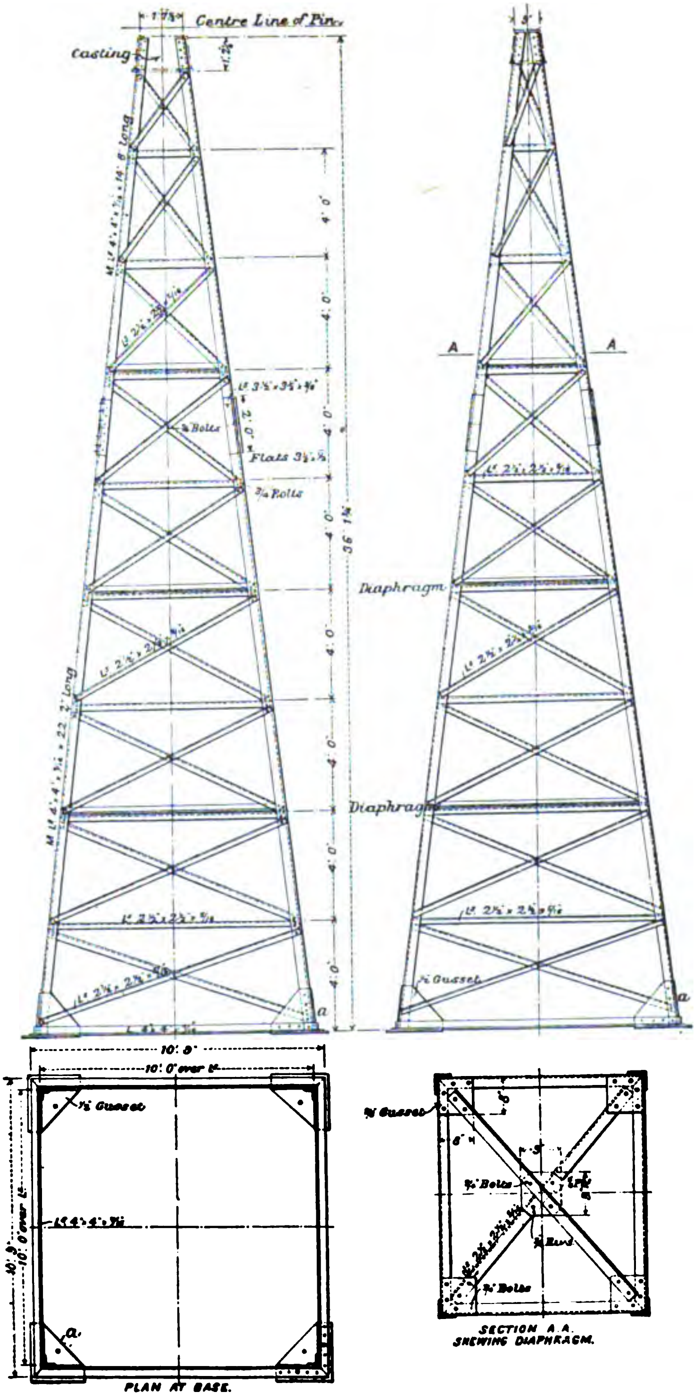


Fig. 104.*—Fixed Tower of Zambesi Cableway (See Fig. 70.)

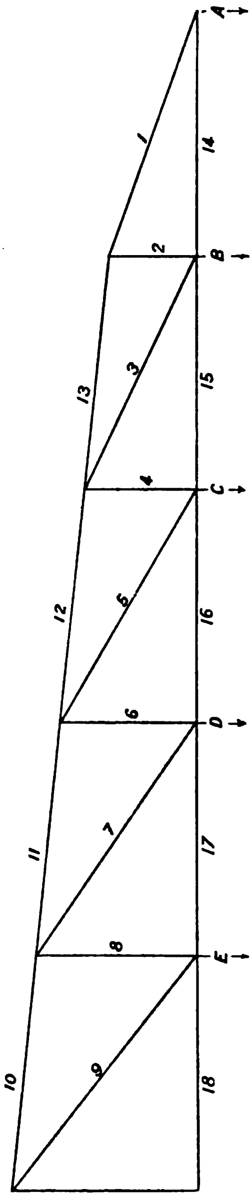


Fig. 105. Side Elevation of Cantilever with Linville Bracing.

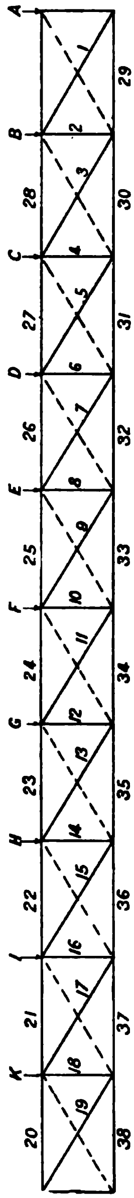


Fig. 108. Plan of Cantilever Showing Wind Bracing.

Fig. 107 shows the stresses with the load hanging at A, the weight of the load being added to the girder weight at this point.

It will be noted that on the stress diagram the lines representing the struts are drawn partly full and partly dotted. If each increment of the weight of girder were hung at the bottom of the strut the full line would represent the load on the strut, but as a portion of the weight comes at the top the true load is represented by the full line plus a portion (probably about one-third) of the dotted line. Commonly the whole length, full plus dotted, is taken as the load, and this gives a result on the safe side.

A plan of the wind bracing is given in Fig. 108. This is on the Linville system, the ties shown in full lines acting when the wind is blowing in the direction shown by the arrow heads, while the ties shown dotted come into action when the direction of the wind is reversed.

The area exposed to wind pressure is taken as twice the area of the girder, as seen in elevation, it being assumed that the effect of the wind is the same both on the back and front latticing, and the pressure is taken as acting at the points A, B, C, D, etc. The corresponding stress diagram is given in Fig. 109.

Usually in designing these cantilevers the stresses are taken out twice—(1) for the weight of girder only plus a wind pressure of 40 lbs. per square foot, and (2) for the weight of girder plus load, plus a wind pressure of 3 to 5 lbs. per square foot as the crane cannot generally be used with a stronger wind than this.

In commencing the design of the girder, its weight must be assumed from the nearest parallel case, and when the design is complete its weight should be calculated and the stresses checked, alterations being made in the design if the stresses so found do not agree with those required.

The cantilever (Fig. 110) having Warren bracing with the ties and struts at an angle of 45° is somewhat lighter than the previous type, for the same strength.

The angle of the ties and struts being fixed, the proportions of the girder are also fixed, and are expressed by the following equations, the symbols in which correspond with the lettering on the drawing, while n = the number of panels (four in Fig. 110).

$$H_1 = \frac{L \tan a}{\left(\frac{1 + \tan a}{1 - \tan a}\right)^n - 1}.$$

In this case L is the total length of the girder, and in Fig. 110 is represented by L_4.

$$H_2 = H_1\left(\frac{1 + \tan a}{1 - \tan a}\right)^n.$$

$$L = \frac{H_2 - H_1}{\tan a}$$

$$= H_2\frac{1 - \left(\frac{1 - \tan a}{1 + \tan a}\right)^n}{\tan a}.$$

For intermediate values of L, such as L_3, n represents the number of complete panels up to that point, thus—

$$L_3 = H_1\frac{\left(\frac{1 + \tan a}{1 - \tan a}\right)^3 - 1}{\tan a},$$

and the same is the case for the intermediate heights x, the equation for which is—

$$hx = H_1\frac{(1 + \tan a)^n}{(1 - \tan a)^{n+1}}.$$

thus—

$$x_4 = H_1\frac{(1 + \tan a)^3}{(1 - \tan a)^4}.$$

Fig. 111 is the girder diagram, and Figs. 112 to 120 give a complete set of stress diagrams with the rolling load at each of the points A, B, C, etc., and the stress diagram for the weight of the girder only.

The vertical members shown in Fig. 111 do not enter into the stress diagrams, as they take no part of the girder stresses proper. They serve a double purpose—(1) by providing additional points of support for the runway girder, they shorten the spans, and so enable a lighter girder to be used; (2) by shortening the unbraced lengths of the compression member, they render a smaller section possible, and so effect a saving of material in this member. They are subject to a tension equal to the weight of the rolling load as it passes the points to which they are attached. Figs. 121 to 153 show a cantilever girder with Warren bracing for a 5-ton crane, the overhang of the cantilever being 75 feet. An additional vertical member *DE* is provided here. This adds nothing to the direct strength of the girder, but by supporting the top tension member it prevents it sagging, so improving the appearance, and by providing means for cross bracing it adds slightly to the general stiffness.

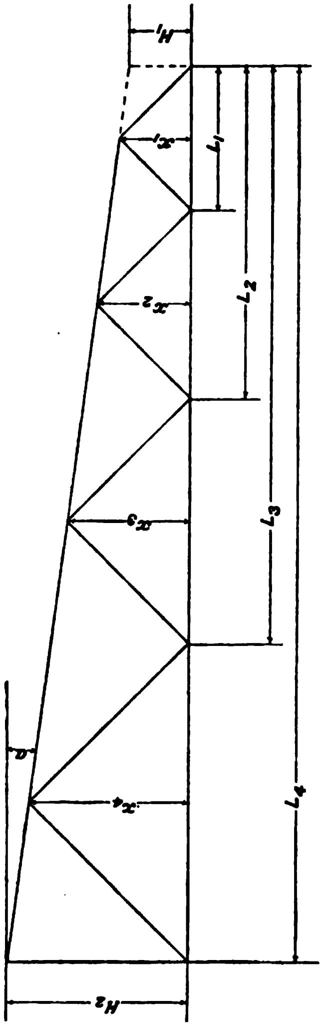


Fig. 110.

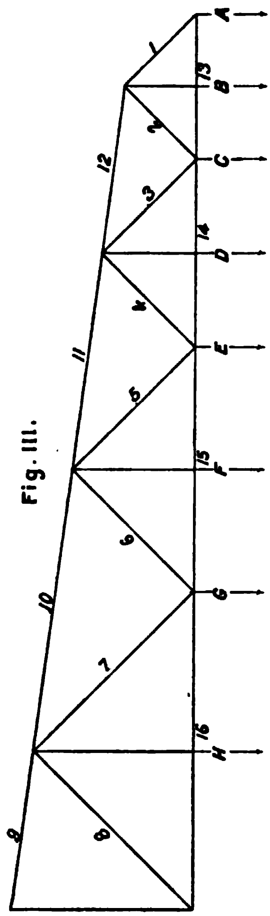


Fig. 111.

The provision of vertical members of this class is very much a matter of taste, some designers omitting them altogether. It will be noted that three of the diagonal struts are braced at the middle, so enabling lighter sections to be used. The weight of this girder is 7·9 tons.

Stress diagrams for the wind pressure are not shown, as they are of a generally similar character to that in Fig. 109. Separate stress diagrams should be taken out for the top and bottom wind bracing, the pressure on the top bracing being much lower than that on the bottom, owing to the lower part of the girder offering a larger area to the wind.

Figs. 154, 155, and 156 show diagrammatically the form of girder for cranes of the revolving cantilever type. In this case the rolling load runs on the top member of the long arm, and the two girders forming this arm cannot be braced across, owing to the hoisting rope hanging between them.

The bracing shown is on the Warren system, with the struts and ties at 45°, but with a strut at the outer end of the arm instead of a tie as in the girder last described. This slightly alters the equations for the proportions of the girder—

$$H_1 = \frac{L \tan a}{\left\{ \dfrac{(1 + \tan a)^n}{(1 - \tan a)^{n+1}} \right\} - 1}.$$

$$H_2 = H_1 \frac{(1 + \tan a)^n}{(1 - \tan a)^{n+1}}.$$

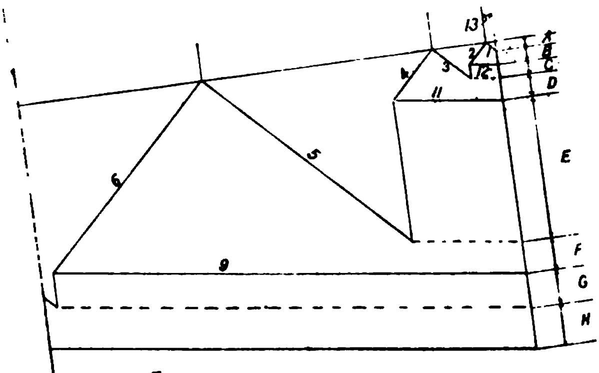
$$L = H_2 \frac{1 - \dfrac{(1 - \tan a)^{n+}}{(1 + \tan a)^n}}{\tan a}.$$

The equation for x remains as before.

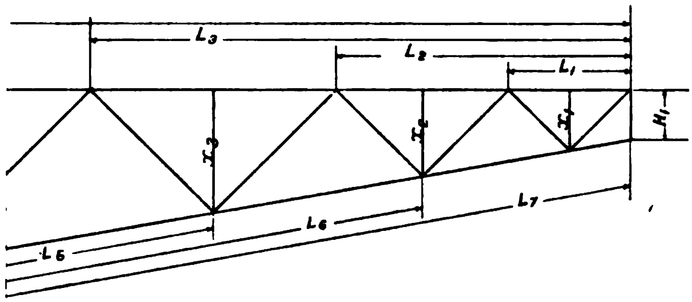
In laying out girders of this type, taking the overhang L of the long arm and the angle of the bottom member as primary dimensions H_1, and then H_2 are calculated. For the short arm H_2 must be the same as for the long arm, while the angle of its bottom member is generally greater. Taking these as the primaries L and H_1 are calculated. In Fig. 155 the members are numbered, and the corresponding stress diagram with the load hanging at A is given in Fig. 157.

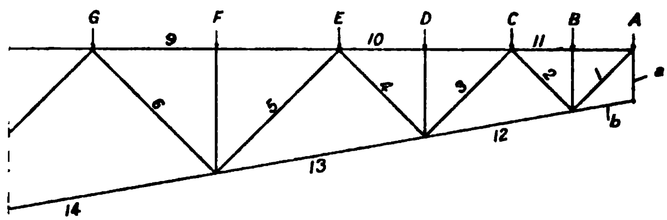
In this case the vertical members provide additional points of support for the girders on which the carriage runs, so shortening their span and reducing their weight.

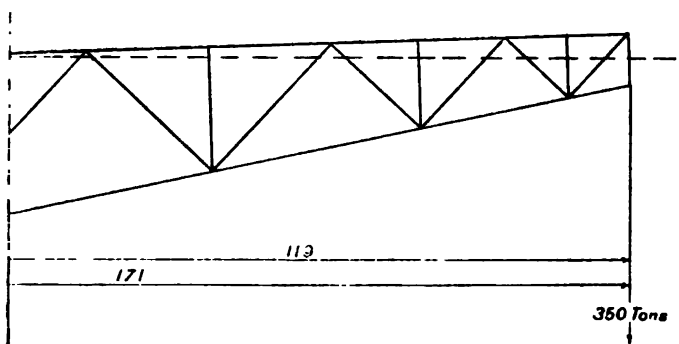
The parts *ab* are for the purpose of carrying a girder which connects the ends of the cantilever girders, so assisting to stiffen them.



Load at E.







lving Cantilever Cranes

The centre bracing at the support may consist of a strut and tie, as shown at Fig. 154, or two ties, as at Fig. 155. When the loads on the two arms are in balance the centre of gravity of the whole revolving structure coincides with the centre of the support, and there is no stress on the centre bracing.

If the loads are out of balance, but the centre of gravity still within the points of support, the structure will not tend to turn over bodily, but, if the centre bracing were omitted, would tend to distort, as shown in Fig. 156, the bracing being necessary to prevent this distortion.

Figs. 158 to 161 show the method of ascertaining diagrammatically the stresses on centre bracing of both types when the arms

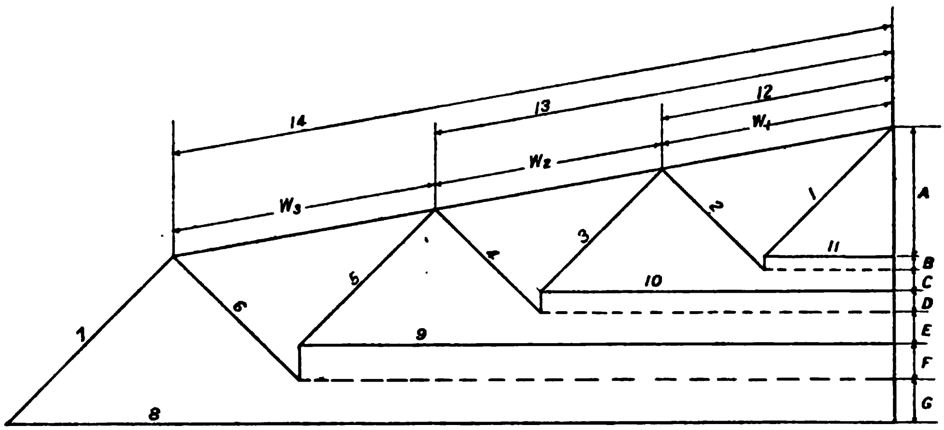


Fig. 157. Stress Diagram for Figs. 154 & 155 with Load at A.

are out of balance. The diagrams being fully figured are self explanatory.

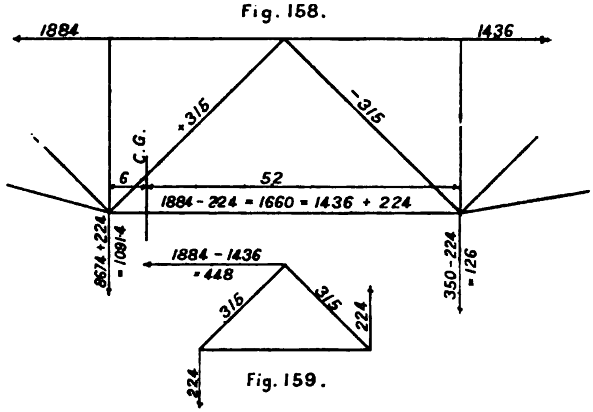
As the two girders forming the long arm cannot be braced across, the compression members must, in the horizontal plane, be sufficiently wide and stiff in themselves to withstand the compressive forces without the assistance of cross bracing.

Each compression member is then equal to a column fixed at the base and loaded at points along its length which correspond to the points of attachment of the vertical bracing, see Fig. 162. The section at the lower end should then conform to the following equation, which is based on Rankine's formula No. 4 :—

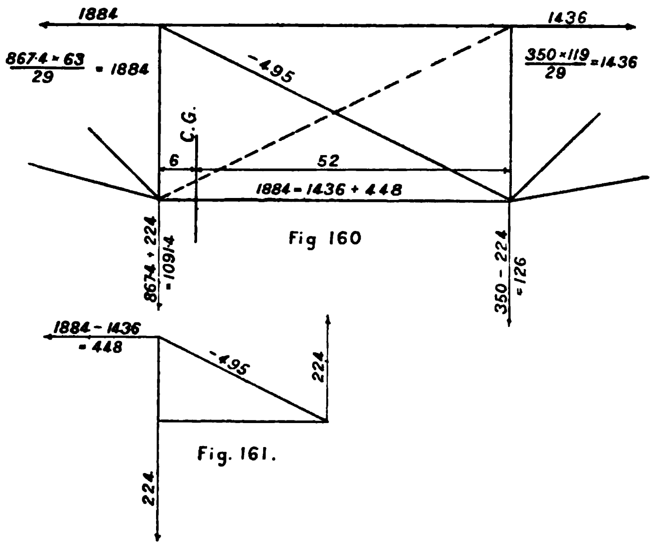
$$A = \frac{s}{f_c}\left\{ W_1\left(1 + c\frac{4L_7^2}{x^2}\right) + W_2\left(1 + c\frac{4L_6^2}{x^2}\right) + W_3\left(1 + c\frac{4L_5^2}{x^2}\right)\right\}, \quad (16)$$

in which s = factor of safety, f_c = compressive strength of material in tons per square inch. W_1, W_2, and W_3 are the loads in tons

Fig. 158.



Figs. 158 & 159. Stresses on Centre Bracing of Fig. 154.



Figs. 160 & 161. Stresses on Centre Bracing of Fig 155.

given on the stress diagram (Fig. 157). $c = \frac{f_c}{\pi^2 E}$. L_5, L_6, and L_7

are the lengths in inches shown on Fig. 154, and κ is the radius of gyration in inches.

For braced girders supported at both ends an economical form is the single Warren with verticals, as shown in Fig. 163, or the double Warren without verticals, as in Fig. 175, the struts and ties being at 45° in both cases. If possible the wheel base of the crab should be not less than the distance between the points of attachment of the bracing, and the crab should be so designed as to have equal wheel loads. Referring to Fig. 163, the weight of the girder is taken as being equal to a load of 0·3 ton at each panel point, and the rolling load is taken at 2·5 tons per wheel. A girder of this weight would really be strong enough for double this load, the 2·5 tons being adopted here merely for purposes of illustration, and to keep down the size of the stress diagrams. The diagram of the stresses on one-half of the girder due to its own weight is given in Fig. 164. The vertical distance Y represents to scale the reaction at the left hand abutment, and is made up of the loads A, B, C, and half the load at D. The vertical line Y having been drawn, line No. 1 parallel to member No. 1 is laid down, and the horizontal line 12 is drawn from the top of Y to meet it. The length of line 1 then represents the compression on No. 1 member, and line 12 the tension on No. 12 member. The shearing force which has to be resisted by member No. 2 being Y-B, the vertical line ab is drawn, and from the point b the line 2 is drawn meeting the horizontal line 13. Line 2 then represents the tension on No. 2 member, and line 13 the compression on No. 13 member, and so on.

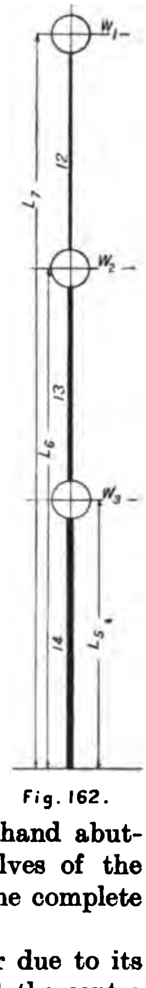


Fig. 162.

The diagram may be more neatly arranged, as in Fig. 165, which enables the vertical lines to be dispensed with. In Fig. 166 the reaction at the right-hand abutment Z is added, and the diagrams for the two halves of the girder are interlaced, so giving the stress diagram for the complete girder.

Fig. 167 gives the diagram of stresses on the girder due to its own weight, and to the rolling load when the crab is at the centre of the span, as shown in Fig. 163. Fig. 174 shows the shearing forces and bending moments under the same conditions. The bending moments due to each load are shown, and the curve *b* is the complete bending moment curve, as found by summing up the separate bending moments. The accuracy of the curve *b* may be checked from the shearing force diagram, as the area of the diagram from the abutment to any given point on the girder is equal to the bending moment at that point. Thus, the mean height of the diagram *cdefgh* is 3·4, and the length of the girder to the point *B* is 13 feet, so that the bending moment at this point is $3{\cdot}4 \times 13 = 44{\cdot}2$ foot-tons, and this corresponds with the length of the line *gi*. In Fig. 167 the vertical lines representing the loads agree with those on Fig. 174, which represent the shearing forces. Therefore, the load on any diagonal equals the shearing force, which that diagonal has to resist, multiplied by the secant of 45° (1·414). The horizontal force due to any diagonal will equal the shearing force multiplied by the tangent of 45°, and as the tangent of 45° is unity the horizontal forces at any point *A*, *B*, *C*, etc., will be equal to the sum of the shearing forces up to that point. If we take the bending moment, from curve *b*, at any point and divide it by the depth of the girder, the horizontal force so found will agree with the horizontal force at the same point as given in Fig. 167. For instance, the bending moment at the centre of the girder being 72·3 foot-tons and the depth of the girder 6·5 feet, the horizontal force at the centre is 11·15 tons, which agrees with the length of the line 14 on Fig. 167, which represents the compressive force on the member No. 14 in Fig. 163. The tensile force on the bottom of the girder at this point is, of course, the same, but is made up of 9·75 tons tension on member No. 11 plus 1·4 tons horizontal force due to member No. 4, these forces being balanced by equal and opposite forces on members 10 and 5.

In order to ascertain the maximum load to which any member may be subjected, it is necessary to take out the stresses for a number of positions of the rolling load. In Figs. 168 to 173 this has been done, with the wheels at *AB*, *BC*, *CD*, *DE*, *EF*, and *FG*. It will be noted that the position of the load affects the stresses on the horizontal members as well as the diagonals, thus with the wheels at *AB* (Fig. 168) the load on No. 12 is greater than when the crab is at the centre, as in Fig. 167. Variation of the position of the rolling load not only varies the stresses on the members, but also leads to reversal of stress in all the diagonals except the two end ones. It is, therefore, necessary to calculate the size of the diagonals both for the maximum tension and the maximum

compression to which they may be subjected, and adopt whichever size is the largest.

The series of diagrams (Figs. 167 to 173) might be completed by the addition of diagrams for the stresses produced with the wheels at *YA* and *GZ*, but the stresses in these cases are much smaller than in any of the others, so that the diagrams, although interesting as showing the reversal of stress on members 2 and 7, are of no particular value.

In dealing with the double Warren bracing, each set of bracing may be treated as a separate girder (see Figs. 176 and 177). The girders are lettered and loaded to correspond with Fig. 163. Fig. 178 is the stress diagram for half the girder shown in Fig. 176, and Fig. 179 is a similar diagram for Fig. 177. As in the case of the girder shown in Fig. 163, it is necessary in this case to work out stress diagrams for each position of the rolling load, both for Figs. 176 and 177. The complete set of diagrams is not given here, however, as they are merely on the same general lines as those already given.

It is not really strictly correct to treat the double Warren girder as though it were composed of two separate girders, owing to the two systems of bracing being connected together through the horizontal members. Thus, if the complete girder (Fig. 175) were loaded only with the loads and at the points shown in Fig. 177, all the load would apparently be on one set of bracing, the other set being unloaded. The deflection of the girder would, however, throw stress on to the second set of bracing, which would then help the first set to take the load, although not carrying any direct load itself. Thus, there is always an equalising tendency between the two sets of bracing, but in order to be on the safe side it is better to disregard this, and to treat each set separately. Figs. 186 to 201 show a structure for a 15-ton Goliath crane designed by Mr. Max Am Ende, M.I.C.E. This was specially designed for use in bridge construction, lightness being a consideration, as every additional ton of weight in the crane necessitates a proportionate increase in the weight and cost of the temporary staging on which the crane runs. The bracing of the cross girders was of the double Warren type without verticals, and it differed from that shown in Fig. 175 in having an odd number of panels, so that there was no central apex, and when the crab was at the centre of the span both sets of bracing were loaded equally. The top members of the cross girders were stayed by horizontal girders to prevent buckling, and these horizontal girders also supported the driver's platform.

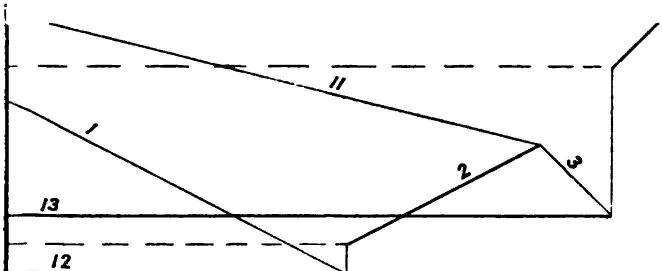
Where head room is a consideration, the bottom member of the girder may be curved, so that the girder can hang between the end carriages, instead of standing on them (see Figs. 4, 8, and 14).

Taking the bending moment curve b in Fig. 174, and dividing the bending moment at the points A, B, C, etc., by 11·15, we obtain the shape of girder shown in Fig. 180. In this the horizontal stresses in the top and bottom members are uniform, and amount to 11·15 tons. The shearing stresses are entirely taken up by the curved lower member, so that no bracing is required. In fact, the shape of the lower member is that which would be assumed by a flexible cable, subject to the same conditions of loading, and the upper member is simply a strut resisting the tension of the lower member. The verticals shown are merely to transmit the loads to the bottom member. The stress diagram, which is similar to that for a loaded cable, is given in Fig. 181.

The condition of uniform stress only obtains, however, so long as the crab is at the centre of the span. As soon as the load moves the stress exceeds 11·15 tons at some points, and is less at others, so that the equilibrium is upset, and bracing becomes necessary.

A more suitable shape for the curved girder is one in which, for any position of the rolling load, the horizontal stress in the flanges shall not exceed a given maximum. In order to determine this shape, it is necessary first to ascertain the maximum bending moments obtained with the rolling load at various positions. In Fig. 182 the curve a is a curve of maximum bending moments obtained by plotting the bending moment curves for various positions of the rolling load and joining the highest points. In the figure three of these curves are shown, b for the crab at the centre, the same as in Fig. 174, c for the wheels at AB, and d for the wheels at BC. It will be noted that each of these curves touches the curve a at one point. Curves for the remaining positions of the wheels would touch the curve a at the remaining points.

Taking, as before, a horizontal stress of 11·15 tons, and dividing the maximum bending moment at each point by this, so as to obtain the depth of girder, the form shown in Fig. 183 is obtained. The design may be simplified by omitting three of the verticals, so that the girder now takes the form shown in Fig. 184, the verticals at C and E being lengthened to 6 feet 6 inches, and those at A and G to 3 feet $7\frac{3}{8}$ inches.



ress Diagram for Fig 184.

The stress diagram for this girder with the crab at the centre is given in Fig. 185, and is on the lines of Figs. 164 and 181. In order to ascertain the maximum stresses on the different members, stress diagrams for the various positions of the rolling load require to be worked out. It does not appear necessary to give the complete set here, but, as an example, Fig. 185*a* is given, showing the stresses when the wheels are at *AB*.

A plate web girder is practically equivalent to a braced girder, in which the ties of the bracing are replaced by a continuous plate. The struts are still necessary, but in plate girders they are called stiffeners.

If the stiffeners are vertical, as is the usual practice, the construction becomes practically a Linville girder in which the inclined ties are replaced by sheets of metal. It is evident, then, that in parallel plate girders the horizontal stresses are taken by the top and bottom flanges, while the shearing forces are taken by the stiffeners and web.

Assuming a parallel plate girder of the same dimensions and weight and loaded in the same manner as the braced girder shown in Fig. 163, the diagram of shearing forces and bending moments in Fig. 174 would apply. Stiffeners would be placed at each of the points *A*, *B*, *C*, etc., and each of these would require to be of sufficient strength, as a strut, to carry a load equal to the shearing force at those points. The sectional area of the web in any panel, in the vertical plane, would require to be equal to the shearing force on that panel, divided by the safe working stress, in shear, of the material. Thus, the dimensions of the stiffeners and thickness of web would diminish towards the centre of the span precisely as the dimensions of the ties and struts diminish towards the centre of the span in braced girders. It should be borne in mind that in plate girders, as in braced girders, the stresses due to different positions of the rolling load must be taken into account.

Plate girders with curved flanges may have their depths determined on the same lines as the braced girders in Fig. 183, the flange, however, describing a curve instead of a polygon. The curved flange in this case, as in the case of braced girders, takes part of the shearing force, so that the stress on the web and stiffeners is correspondingly reduced.

Figs. 202 to 205 show a plate girder with curved lower flange and double web for a 5-ton overhead crane. The weight of the girder is $3\frac{3}{4}$ tons. The diagram of bending moments is given in Fig. 206, and the shearing forces in Fig. 207.

As the cross girders of overhead and Goliath cranes cannot be

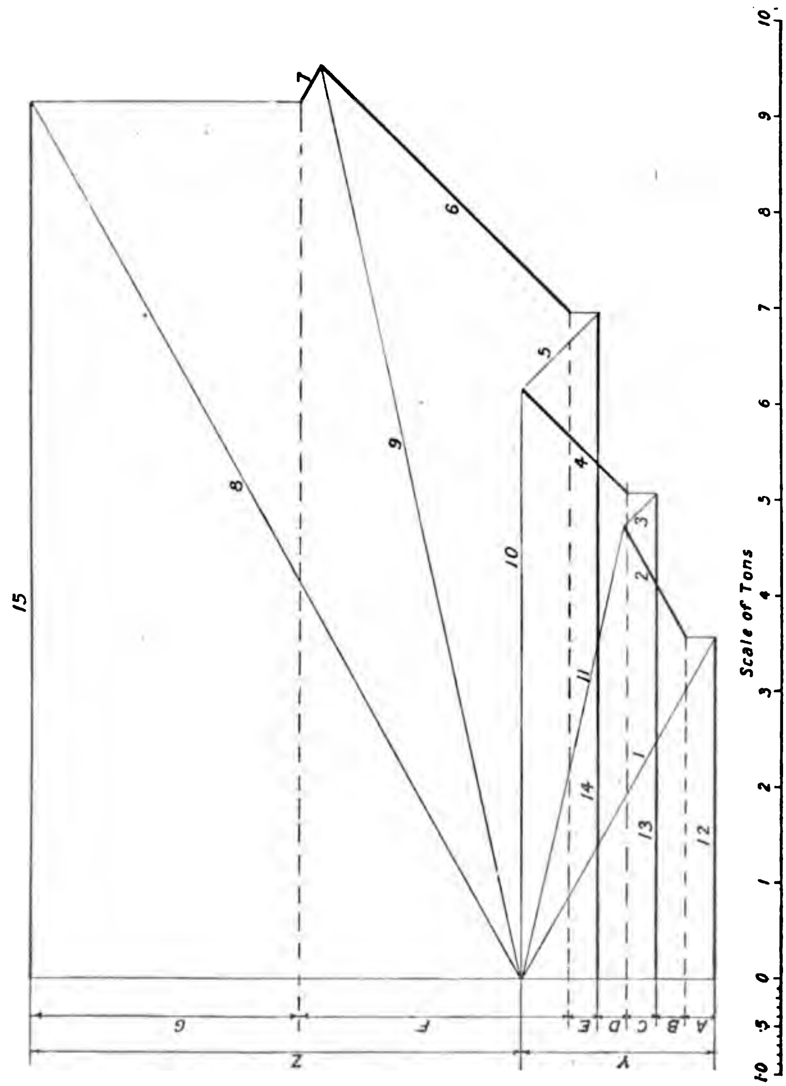
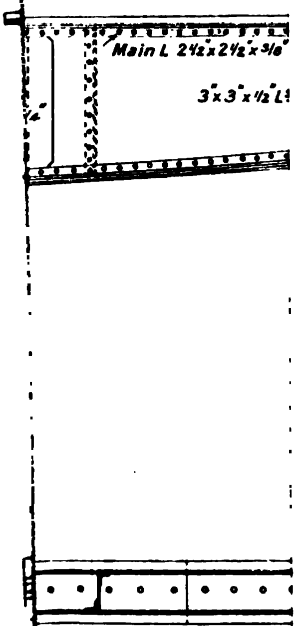


Fig. 185 a. Stress Diagram for Girder Fig. 184 with Wheels at A.B. Refer to Fig. 166.

connected by horizontal bracing, the top members must either be sufficiently wide and stiff horizontally, to avoid buckling due to the compressive forces, or they must be braced by horizontal girders,

s. 186 to 201.—Structures for 15-1



Half Sectiona

Ton Crane.

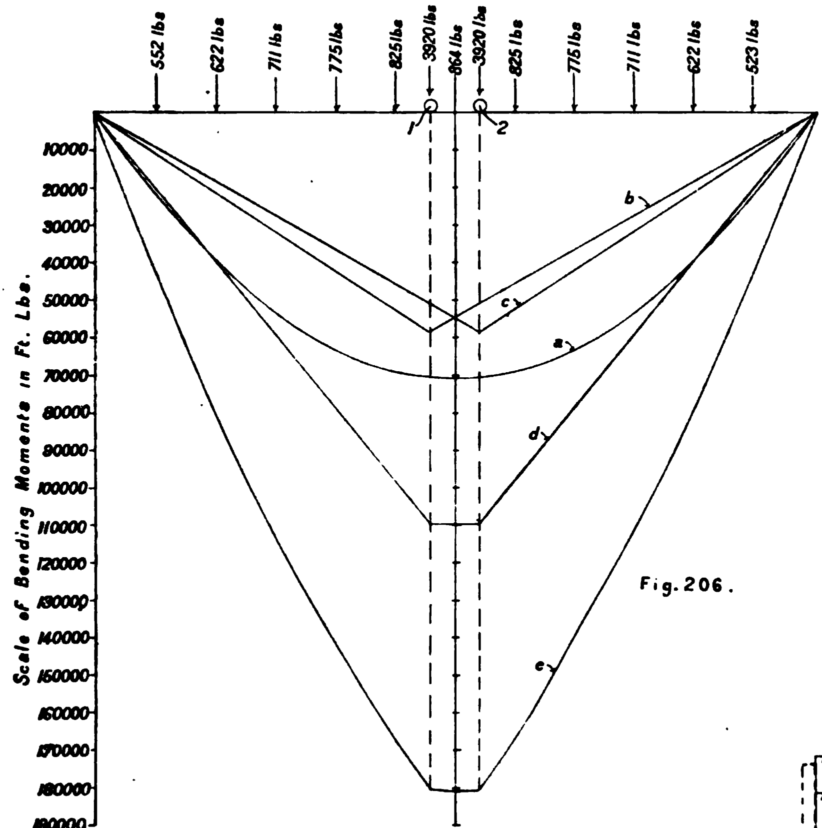
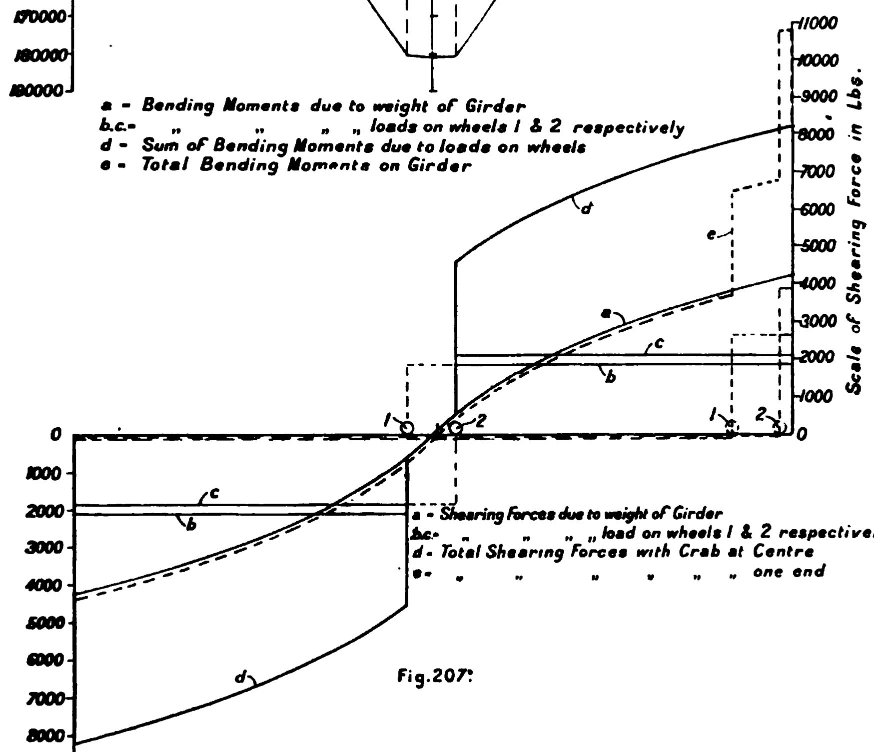


Fig. 206.

a - Bending Moments due to weight of Girder
b.c.- „ „ „ „ loads on wheels 1 & 2 respectively
d - Sum of Bending Moments due to loads on wheels
e - Total Bending Moments on Girder



a - Shearing Forces due to weight of Girder
b.c.- „ „ „ „ load on wheels 1 & 2 respectively
d - Total Shearing Forces with Crab at Centre
e - „ „ „ „ „ „ one end

Fig. 207.

the latter arrangement being preferable as the horizontal girders serve also to support the driver's platform. The dimensions of these girders may be determined on the same principles as the braced columns dealt with earlier in this chapter with reference to Fig. 99, addition being made to the dimensions so found, in order to give the girder sufficient strength to carry the driver's platform in addition to resisting the buckling tendency of the top member of the cross girder. As already mentioned, an example of the application of these horizontal girders is given in Figs. 186, etc.

The arrangement of the rivetting of the top and bottom flanges to the web in plate girders is dependent upon the distribution of the horizontal forces along these members.

Dividing the girder into any convenient number of panels, the horizontal force in each panel may be determined either from the curve of bending moments or the diagram of shearing forces. Calling the panel at the extreme left No. 1, the horizontal force on this panel will be—

$$P_1 = \frac{M_{bA}}{D_g}.$$

On No. 2 panel it will be—

$$P_1 = \frac{M_{bB} - M_{bA}}{D_g},$$

and so on.

The number of rivets per panel being $n_r = \frac{L_p}{p}$, the load per rivet will be $L_2 = \frac{P_1}{n_r}$.

Combining these equations, the load per rivet, taken from the curve of bending moments, will be—

$$L_2 = \frac{p(M_{bB} - M_{bA})}{D_g L_p},$$

and so on.

Taking the diagram of shearing forces—

$$P_1 = \frac{S_5 L_p}{D_g},$$

and

$$L_2 = \frac{S_5 p}{D_g}.$$

This last equation is the most convenient, as, S_5 being the shearing force at a given point, and D_g the effective depth of the girder at that point, it is applicable to girders with curved flanges, as well as parallel girders. The pitch of rivets being taken in inches, and load per rivet in tons, the other values should also be in inches and tons. The usual pitch of rivets $\frac{3}{4}$ or $\frac{7}{8}$ inch diameter is 4 inches.

In all plate girders the load on the rivetting is greatest in the panels at the two ends, this being more particularly the case in girders having curved flanges. To take an extreme case, the girder in Fig. 180 would require the whole of the rivetting at the extreme ends, none being required along the flanges, as no web would be necessary.

In cases where, with a pitch of 4 inches, the load per rivet becomes excessive, two rows of rivets may be used, each of 4 inches pitch, the rivets in one row being opposite the centres of the spaces in the other row.

Generally, for the sake of appearance, the rivetting is made uniform all along the girder, although in some cases a double row of rivets is placed in the first panel, the remainder being single.

Taking the safe working stress, in shear, on mild steel rivets at 5 tons per square inch, and the safe bearing load at 10 tons per square inch of projected area of rivet holes, Table No. IX. has been prepared, giving the safe loads on rivets of various diameters in plates of various thicknesses.

TABLE IX.—Safe Loads on Rivets in Tons.

Diameter of Rivet.	Safe Load in Shear.		Safe Bearing Load. Thicknesses of Plate.							
	Single.	Double.	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$
$\frac{1}{2}$ inch,	·98	1·96	1·25	1·56	1·87	2·18	...	...	...	...
$\frac{5}{8}$,,	1·53	3·06	1·56	1·95	2·3	2·7	3·1	...	...	...
$\frac{3}{4}$,,	2·2	4·4	1·87	2·34	2·81	3·28	3·75	4·21	4·68	...
$\frac{7}{8}$,,	3·0	6·0	2·18	2·73	3·28	3·82	4·37	4·92	5·46	6·01

The deflection of a girder, in which the area of the flanges is varied, so as to keep the stress per unit of flange area constant, is—

$$\delta = \frac{fl^2}{ED_g},$$

in which l is the length in the case of a cantilever and half the length in the case of a girder supported at both ends.

CHAPTER XI.

DESIGN OF MACHINERY.

In the previous chapters relating to types of cranes general arrangements of machinery have been illustrated, and it is, therefore, only proposed to deal with a few details in this chapter, the examples given being taken from the shop drawings of work carried out by the author.

In designing shafting, the diameter is usually calculated on the basis of the breaking strength in torsion of the material. In some cases, however, such as where a shaft is fitted with cut gear at some distance from a bearing, the diameter requires to be calculated from the permissible deflection in order to prevent chatter and excessive wear of the gear, and in the case of long shafts it is sometimes necessary to base the size upon the amount of permissible twist between the two ends.

The torque on a shaft in inch-lbs. is—

$$T = \frac{12 \times 33{,}000 \times H}{2\pi S_2} = \frac{63{,}025\,H}{S_2}.$$

The safe working torque on a shaft is $T = fZ_t$, in which f is the safe working stress in torsion, which for ordinary mild steel may be taken at 8,000 lbs. per square inch, and Z_t the modulus of the section with respect to torsion. For round shafts—

$$Z_t = \frac{\pi}{16} d^3 = \cdot 196\, d^3,$$

and the diameter of the shaft if subjected to torsion only is—

$$d = \sqrt[3]{\frac{T}{\cdot 196 f}}. \quad . \quad . \quad . \quad . \quad (1)$$

Axles of crabs, travelling carriages, etc., which have only to support a bending moment may have their dimensions determined on the same principles as beams.

The safe bending moment is then $M_b = fZ$. f in this case being the safe working stress obtained by dividing the modulus of rupture by a factor of safety. For mild steel f may be taken at 16,500 lbs. per square inch. The modulus of a circular section with respect to bending being $\frac{\pi}{32}d^3 = \cdot 0982\ d^3$, the diameter of the axle is—

$$d = \sqrt[3]{\frac{M_b}{\cdot 0982 f}}. \quad . \quad . \quad . \quad . \quad (2)$$

In the majority of cases a shaft is subject both to a torque due to the power transmitted and to bending moments due to weights which the shaft has to support, or to the forces on the teeth of gearing, etc.

The torque and bending moment having been determined, they may be reduced to an equivalent torque by the equation—

$$T_e = M_b + \sqrt{M_b^2 + T^2}. \quad . \quad . \quad . \quad . \quad (3)$$

The diameter of shaft can then be found from formula No. 1, substituting T_e for T.

The principal purpose for which the deflection of a shaft requires to be known is to make sure that there will not be sufficient deflection to cause chatter in the gearing. Generally in such a case the force producing the deflection is the load on the wheel teeth themselves, and the deflection required to be known is that at the point where the force is applied which is not necessarily the maximum deflection produced.

For a shaft supported between bearings as in Fig. 208 the deflection is—

$$\delta = \frac{Wa^2b^2}{3lEI}.$$

For a circular section $I = \frac{\pi}{64}d^4$ and

$$\delta = \frac{64\ Wa^2b^2}{3\pi lEd^4} = \frac{6\cdot 79\ Wa^2b^2}{lEd^4}. \quad . \quad . \quad . \quad . \quad (4)$$

Where the force is applied to an overhanging length of shaft, as in Fig. 209, the actual bend of the shaft may be found from equation No. 4, if for W we substitute the reaction at the support A, which is equal to $\frac{Wl}{a}$.

As the bearings AB are at the same level, the deflection at W will be the amount so found multiplied by $\frac{l}{a}$.

Then,

$$\delta = \frac{6{\cdot}79\ Wla^2b^2l}{laEd^4a} = \frac{6{\cdot}79\ Wlb^2}{Ed^4}. \qquad (5)$$

In a case where a long shaft drives a set of machinery at each end, the shaft itself being driven at one end, it is advisable to calculate the amount of twist in the shaft, in order to ascertain the extent to which the one set of machinery will lag behind the other.

Fig. 208.

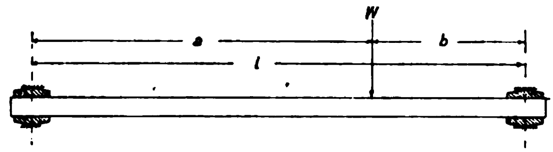
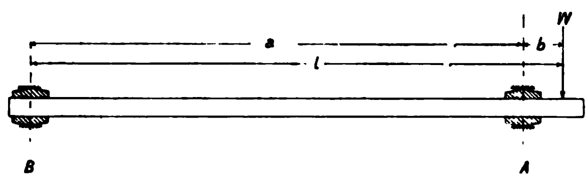


Fig. 209.



The twist of a shaft in circular measure is $\frac{2Tl}{E_tZ_td}$, and in degrees—

$$\theta = \frac{114{\cdot}7\ Tl}{E_tZ_td} = \frac{114{\cdot}7\, fl}{E_td}.$$

E_t is the modulus of transverse elasticity, which for steel is 12,000,000 lbs. per square inch.

The amount of twist in a steel shaft, in degrees, is then—

$$\theta = \frac{114{\cdot}7\, fl}{12{,}000{,}000\ d} = \frac{{\cdot}9558\, fl}{d} \times 10^{-5}, \qquad (6)$$

dimensions being in inches and stress in lbs. per square inch.

Where a length of unloaded shafting is supported between two bearings, the distance apart at which the bearings should be placed will depend, in the case of a slow-speed shaft, upon the permissible deflection of the shaft under its own weight, and in the case of high speed upon the tendency to centrifugal whirling of the shaft.

A mild steel shaft $3\frac{1}{4}$ inches diameter freely supported at two points, 24 feet apart, has a deflection at the centre, by actual measurement, of $\frac{5}{8}$ inch.

The deflection being proportional to the weight, and cube of the length, and inversely proportional to the fourth power of the diameter, the deflection for any other shaft, based upon this figure, would be—

$$\delta = \frac{\cdot625 \times d^2 \times \cdot7854 \times l \times \cdot288 \times l^3 \times 3\cdot25^4}{3\cdot25^2 \times \cdot7854 \times 24 \times 12 \times \cdot288 \times (24 \times 12)^3 \times d^4}$$

$$= \frac{\cdot9594\, l^4}{d^2} \times 10^{-9}, \qquad (7)$$

the dimensions being in inches.

The deflection of the $3\frac{1}{4}$-inch shaft with the bearings 10 feet apart would be 0·019 inch.

For a high-speed shaft, the distance between bearings, in inches, at which centrifugal whirling will occur is *—

$$l = \sqrt{\frac{4{,}732{,}000\, d}{S_2}}. \qquad (8)$$

For keys a good proportion is to make the key of square section, the side of the square being one-quarter the diameter of the shaft, and the key being let half into the shaft and half into the wheel boss. For large shafts, two keys of half the dimensions may be used.

In designing bearings, if the diameter of the shaft is sufficient for the work which it has to do, the bearing will have sufficient surface if for slow speeds the length of bearing is equal to the diameter of the shaft, for moderate speeds $1\frac{1}{2}$ times the diameter, and for higher speeds up to 500 revolutions per minute twice the diameter.

An example of a pedestal for a $2\frac{1}{2}$-inch shaft is shown in Figs. 210, 211, and 212. The pedestal body is of cast iron, and the steps are of gun-metal with Stauffer lubrication.

In bearings carrying a heavy load it is necessary that the bearing steps should be supported along their whole length. If made with projecting flanges, as in the light bearing just

* See paper by Prof. Dunkerley, *Transactions of the Royal Society*, vol. 185*a*.

described, the gun-metal will wear away in the body where it is supported by the pedestal, while the flanges, not being strong enough to stand up to the pressure, will be bent down and broken off. Figs. 213, 214, and 215 show a plummer block for a 15-inch shaft, in which the gun-metal steps are supported

Fig. 210.

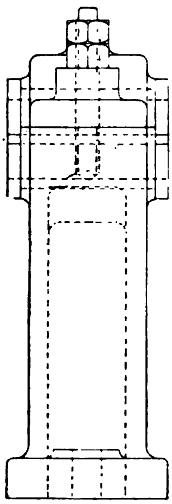
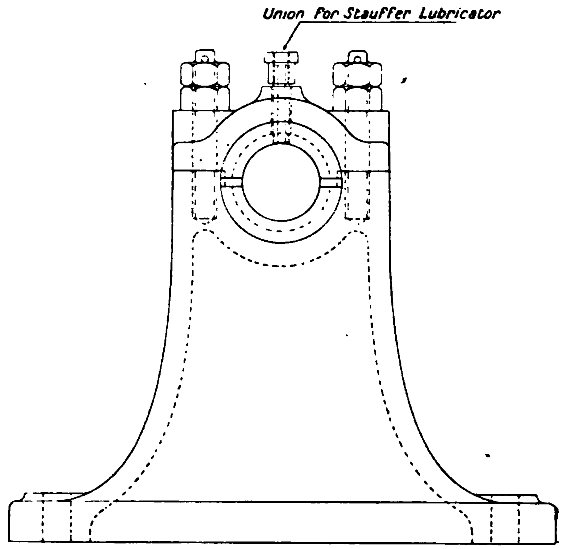
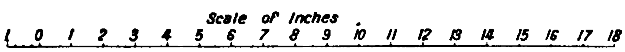


Fig. 211.





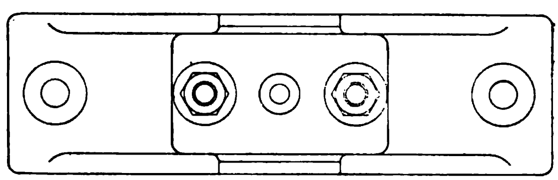


Fig. 212.

Pedestal for 2½-inch shaft.

along their whole length by the plummer block body, the bore in which is enlarged at the ends to take the flanges. Owing to the low height of centre, which was unavoidable, the body of this plummer block was cast in steel, and the cap was cast in steel to match it. Stauffer lubricators were used. The load on this bearing

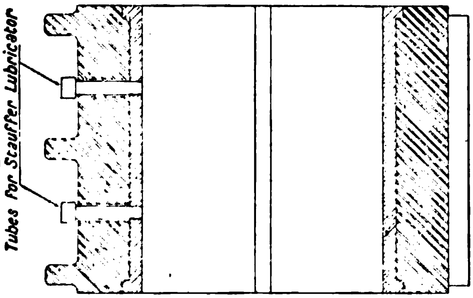


Fig. 214.

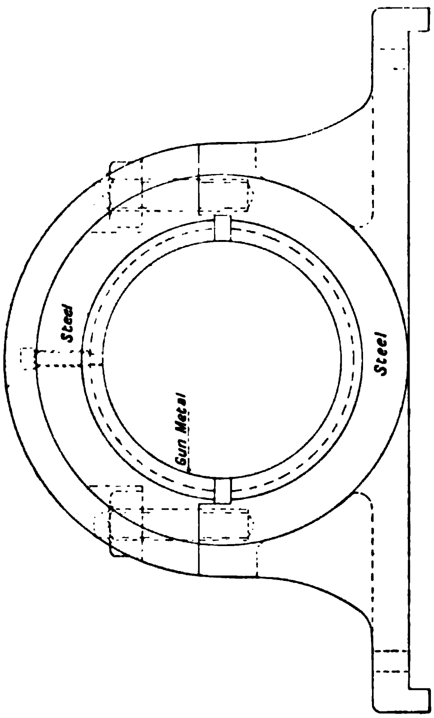
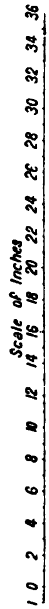


Fig. 213.



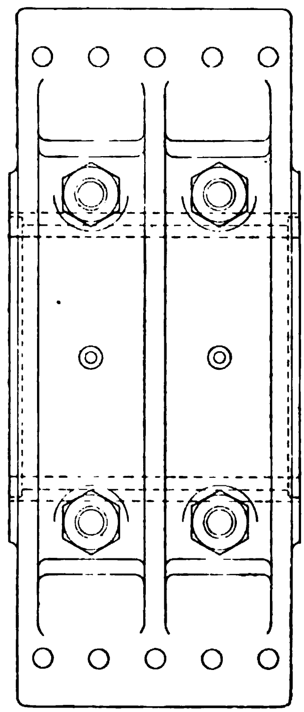


Fig. 215.

Plummer Block for 15-inch Shaft.

varies from 85 to 180 tons, the speed of shaft is 0·2 revolution per minute, and the running is intermittent with reversals every few

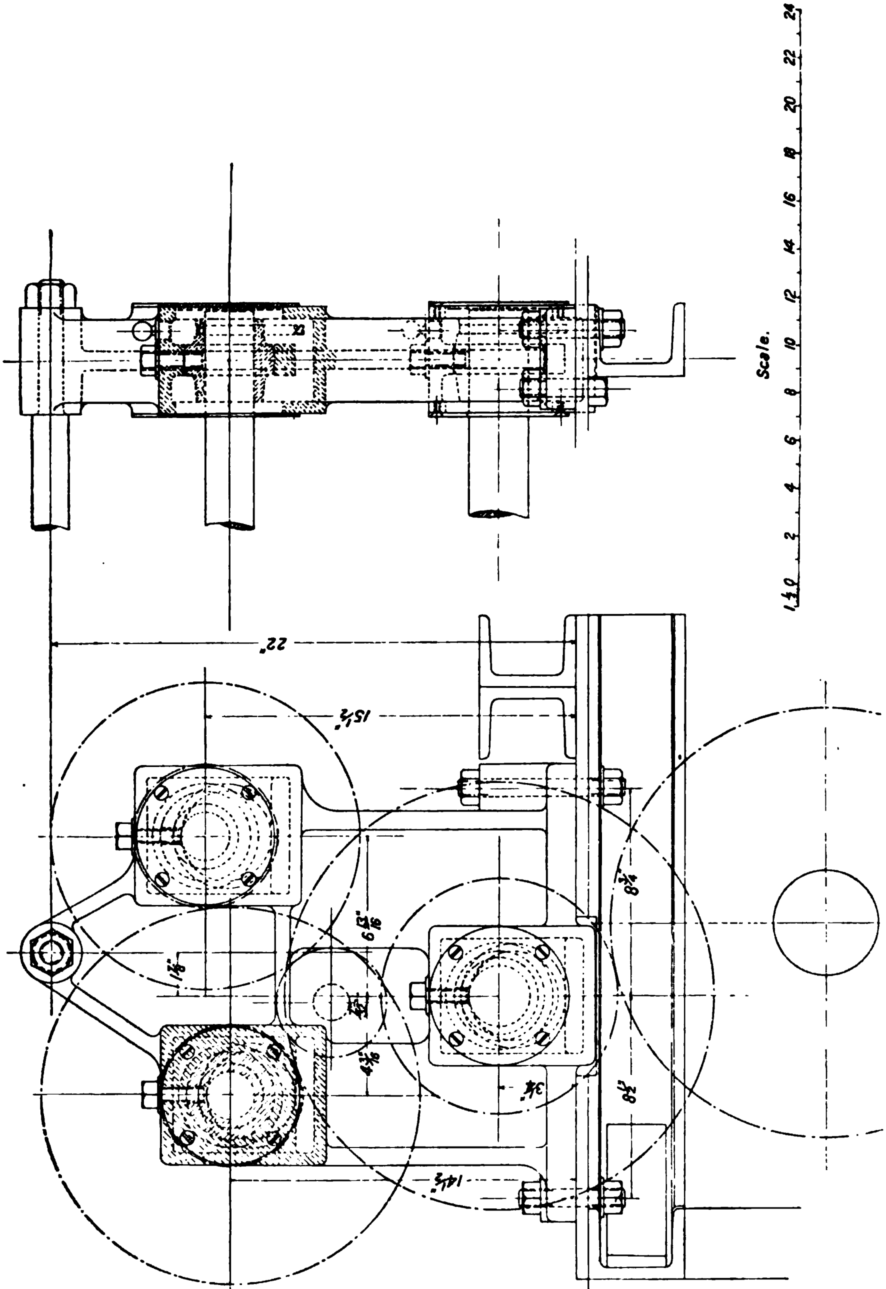


Fig. 216. Fig. 217.

Frame Side with Self-oiling Bearings.

minutes. There is a varying horizontal force on the bearing at right angles to the shaft, which occasionally rises to 70 tons, and to meet this the ends of the sole plate have lips which are planed to fit accurately on the seating. The block is planed all over the bottom, and fits on a steel seating, which is planed on top and at the ends to fit the overhanging lips. The weight of the block is 1 ton.

Figs. 216 and 217 illustrate a frame side with self-oiling bearings. These are of very simple construction, and are quite satisfactory with moderate loads and speeds.

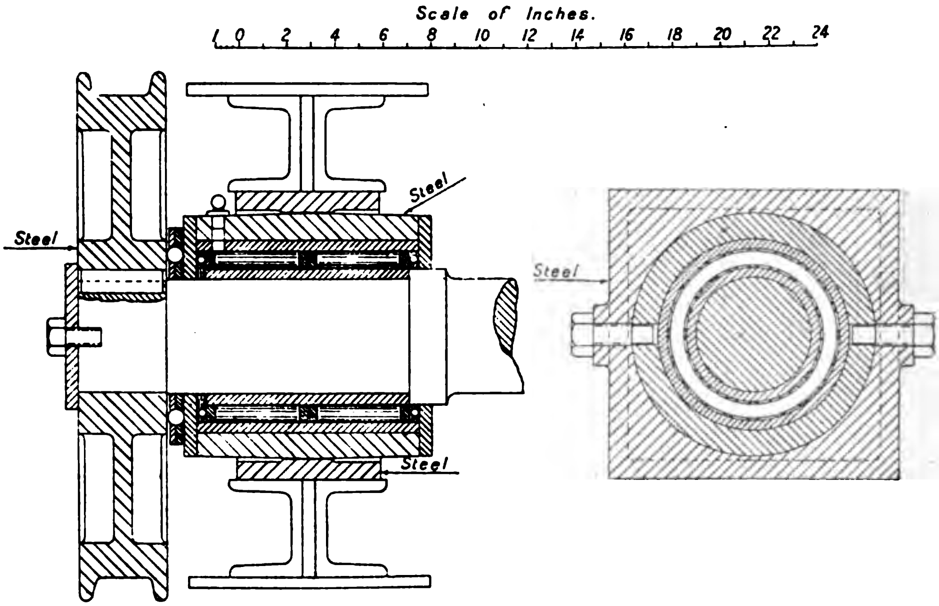


Fig. 218. Fig. 219.

Roller Bearing for $4\frac{3}{4}$-inch Axle.

A roller bearing for a crab axle $4\frac{3}{4}$ inches diameter is shown in Figs. 218 and 219. In this bearing the rollers were of steel hardened and ground, and they ran between hardened and ground steel races. Any tendency to end thrust of the rollers was taken up by ball thrust washers inside the casing, while end thrust on the axle was taken up by ball thrust washers fitted between the wheel boss and the end of the bearing. In order that they might adapt themselves better to any slight bend of the axle two sets of short rollers in separate gun-metal cages were used instead of one set of rollers the full length of the bearing. The rollers, cages, and hardened-steel races were made by the Auto-Machinery Company, of Coventry. This bearing is No. 1 in Table X.

The roller bearings on the barrel shaft in Fig. 226 had hardened and ground steel rollers running on the ordinary turned surface of the shaft, and in a steel bush bored but not ground, neither the shaft nor the bush being hardened. The rollers, cages, and steel bush were made by the Empire Roller Bearings Company, of London, and the bearing is No. 2 in Table X.

TABLE X.—Dimensions of Roller Bearings.

No.	Dia. of Inner Race.	Revs. per Min.	Peripheral Speed of Inner Race. F.P.M.	W	N_r	l	d	c	Remarks.
1	5·5	20	28·8	11,620	20	3·75	·75	1,033	Races hardened. Been running several years.
2	5	3	3·93	10,100	14	10	·875	411	Races soft. Been running several years.
3	$4\frac{7}{8}$	3	3·829	17,900	30	1·5	·5	3,940	Races hard. Been running several years.
4	6	38	59·68	18,000	16	12	·9375	500	Races soft. Been running several years.
5	5·5	40	57·6	22,400	20	3·75	·75	1,991	Races hard. Failed in six to nine months.

All the above bearings were for intermittent work.
All except No. 5 were in swivels.

To ensure satisfactory working, it is essential that roller bearings should be mounted in swivels, so that the bearing pressure shall be evenly distributed along the whole length of the roller.

A formula for the proportions of roller bearings, due to Prof. Stribeck,* is—

$$W = \frac{cldN_r}{5},$$

in which W is the load in lbs. on the bearing, c a constant representing the load in lbs. on a unit bearing, l the length and d the diameter of the rollers in inches, and N_r the number of rollers in the bearing.

For bearings with unhardened roller races working continuously under full load, Prof. Stribeck gives the value of c at 85 to 156.

* *Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 6, 1902. In the original the formula is in metrical measurements, and the value of c is then 6 to 11.

In Table X. the values of c are given for several roller bearings in actual use.

Prof. Stribeck's formula appears to be based on the strength of the rollers, and not on the area of contact surface between the roller and race due to their elasticity. Also, it does not take speed into account. Therefore, the value to be adopted for c will vary according to the speed, the value being apparently inversely proportional to the $\frac{2}{3}$ power of the peripheral speed, but with a maximum for intermittent work of 1,000 for unhardened surfaces and 4,000 for hardened ones. For continuous running under full load it would be advisable not to exceed one-sixth of these values. Referring to Table X., as bearing No. 4 was satisfactory, it is evident that No. 2 could have carried a heavier load safely, owing to its slower speed.

Bearings No. 5, which failed, had not such a high specific load as No. 3, and their failure is attributable either to their higher speed, or to the fact that they were not mounted in swivels. Possibly both causes contributed to the failure.

For light work, ball bearings give very satisfactory results. A number of crane motors fitted with ball bearings have now been at work several years, and probably the use of ball bearings for crane motors will eventually become universal.

The design of toothed gearing being fully treated in a subsequent chapter, a few examples only are given here of gears of different sizes for the purpose of illustrating their general proportions.

Figs. 220 and 221 show a cast-steel pinion and wheel with cast teeth having a pitch of 7 inches. The particulars of the gear are as follows :—Pinion, 9 teeth ; pitch circle (p), $20\frac{1}{16}$ inches diameter. Base circle of involutes (b), $9\frac{1}{16}$ inches radius. Thickness of tooth at pitch line, $3\frac{1}{16}$ inches ; width of space, $3\frac{5}{16}$ inches. Height of tooth above pitch line, 2 inches. Depth below pitch line, 1 inch. Wheel, 86 teeth. Pitch circle (p), 15 feet $11\frac{5}{8}$ inches diameter. Base circle of involutes (b), 7 feet $8\frac{3}{4}$ inches radius. Thickness of tooth at pitch line, $3\frac{3}{16}$ inches ; width of space, $3\frac{13}{16}$ inches. Height of tooth above pitch line, $\frac{11}{16}$ inch ; below pitch line, $2\frac{5}{16}$ inches.

The pinion having only 9 teeth, the whole load is taken on one tooth. As will be seen in Fig. 220, the line of contact CD slightly exceeds the pitch, so that a pair of teeth do not go out of contact until the succeeding pair have just entered into engagement. The tangential force at the pitch line of this gear is very variable, the maximum being 51 tons. The gear works inter-

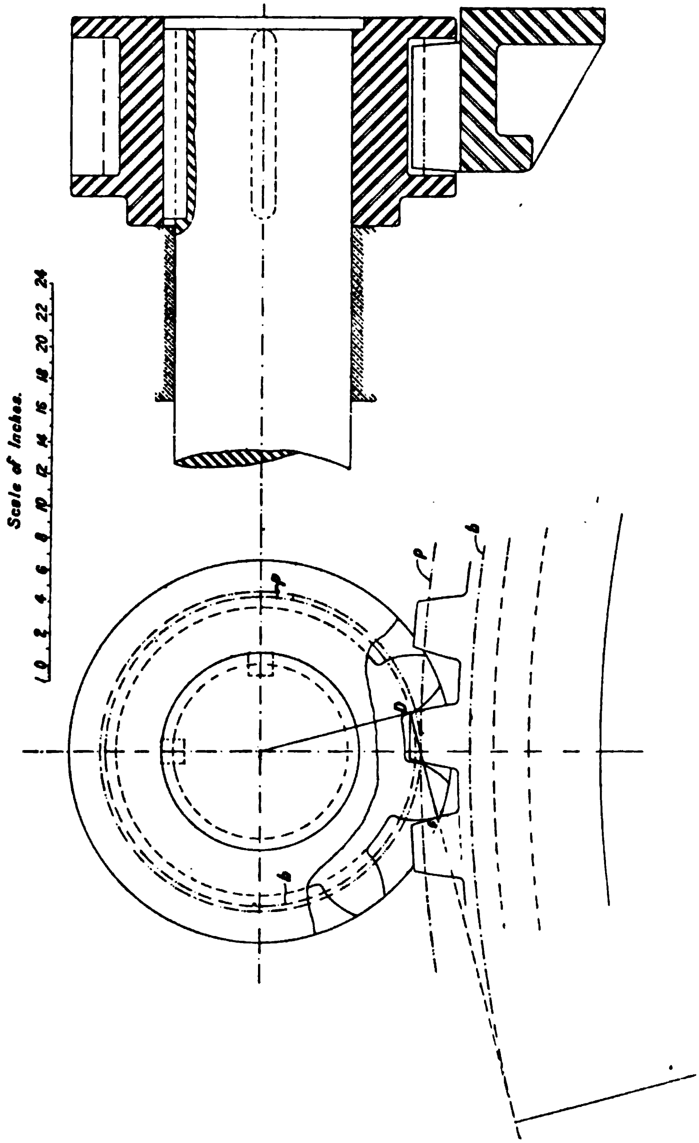


Fig. 221.

Fig. 220. Steel Gear, 7-inch Pitch.

mittently, the speed of the pinion being 2·0 revolutions per minute, with reversals every few minutes. If this gear had to transmit

the full load of 51 tons continuously, the width of the wheels would have to be increased very materially, or their wear would be extremely rapid. As, however, this heavy load only comes upon them occasionally, all that was necessary was that the teeth should have sufficient strength to bear this load without breaking and without bruising of their surfaces. The gear was set to work two years ago, and shows very little wear.

Figs. 222 and 223 show a cast-steel pinion and wheel $2\frac{3}{4}$-inch pitch, with cast teeth transmitting 9·1 tons at the pitch line. In this case, also, the load is taken on one tooth only, and the work is intermittent, the speed of pinion being 9 revolutions per minute. The following are the particulars:—Pinion, 14 teeth. Pitch circle (p) 12·25 inches diameter. Base circle of involutes (b), 5·9 inches radius. Thickness of tooth at pitch line, $1\frac{3}{8}$ inches; width of space, $1\frac{3}{8}$ inches. Height of tooth above pitch line, $\frac{1}{2}$ inch. Depth below pitch line, $\frac{5}{8}$ inch. Wheel, 106 teeth. Pitch circle (p), 92·8 inches diameter. Base circle of involutes (b), 44·8 inches radius. Thickness of tooth at pitch line, $1\frac{5}{16}$ inches; width of space, $1\frac{7}{16}$ inches. Height of tooth above pitch line, $\frac{7}{16}$ inch. Depth below pitch line, $\frac{11}{16}$ inch.

The gears in Figs. 220 to 223 being single-tooth gears, the teeth are designed on the principles for single-tooth gear, explained in the chapter on Toothed Gearing, the height of tooth being discontinued at the point where the useful surface ends, so that the whole surface from the involute base line to the top of the tooth is utilised.

Figs. 224 and 225 show a pair of steel wheels having cut teeth $1\frac{3}{8}$-inch pitch transmitting 43 H.P., the speed of the smaller wheel being 67 revolutions per minute. The particulars are:—Driving wheel, 56 teeth. Pitch circle (p), 24·51 inches diameter. Base circle of involutes (b), 11·84 inches radius. Driven wheel, 110 teeth. Pitch circle (p), 48·14 inches diameter. Base circle of involutes (b), 23·26 inches radius.

In this case the arc of action is equal to twice the pitch, so that the load is always taken on two teeth.

Fig. 226 shows a hoisting barrel carried on roller bearings, the direct pull off the barrel being 8 tons.

In designing live rings of rollers, such as those used on revolving cantilever cranes, the dimensions are generally based on the extent of the contact area between the roller and roller path, due to the elasticity of the material which is proportional to the square root

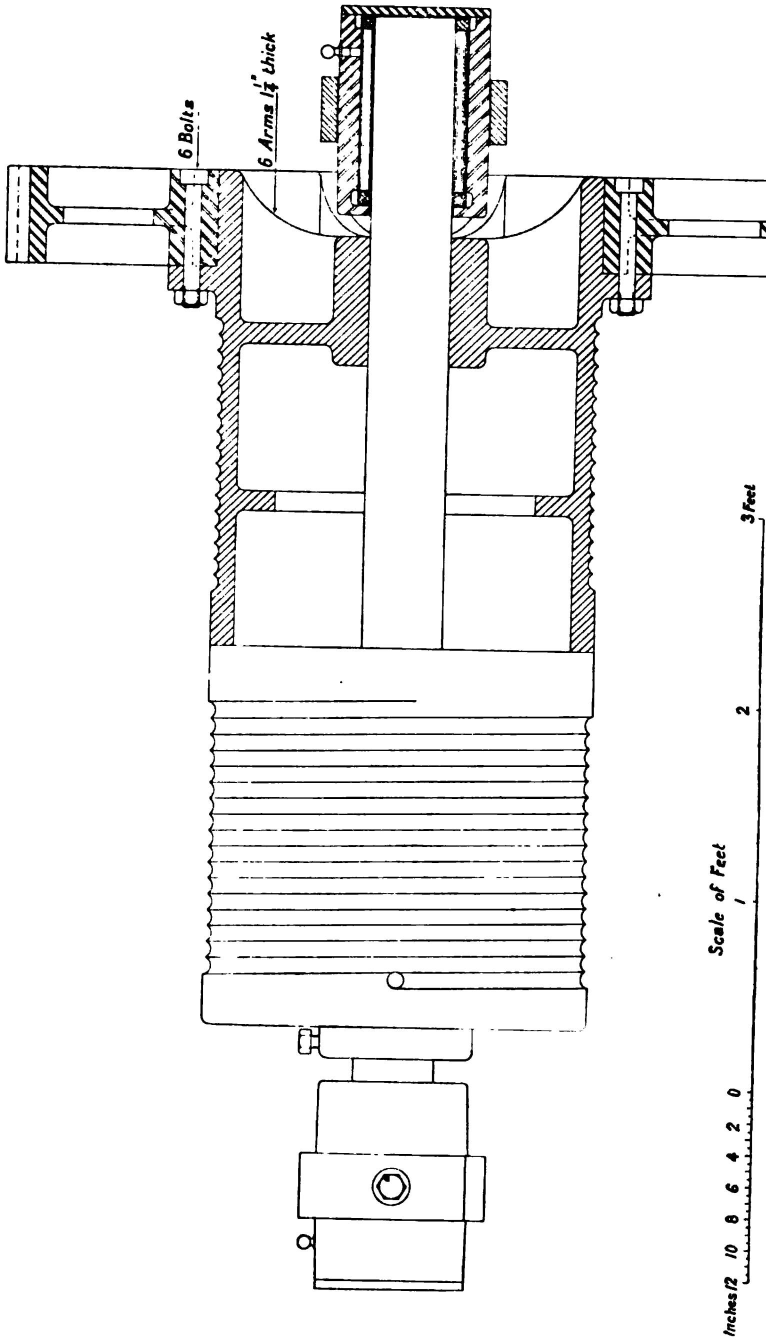


Fig. 226.—Hoisting Barrel with Roller Bearings.

of the roller diameter. The pressure in tons per inch width of roller is then—

$$P_r = c\sqrt{d},$$

c being a constant, and d the mean diameter of the roller in inches. A value for c for steel rollers and roller paths based upon satisfactory practice is 0·33.

This is lower than the value used for swing bridge work, the reason being that in a swing bridge the weights are constant, and can be so arranged that their centre of gravity coincides with the centre of the roller ring, the load being then always equally distributed over the whole of the rollers. In a crane the load continually varies, so that the centre of gravity of the revolving mass seldom coincides with the centre of the roller ring. Consequently the rollers are generally unequally loaded, and this has to be allowed for in the value adopted for c.

Making the pitch of the rollers 1·25 times their diameter, the number of rollers is—

$$N_r = \frac{12 \times 2\pi R_1}{1 \cdot 25\, d},$$

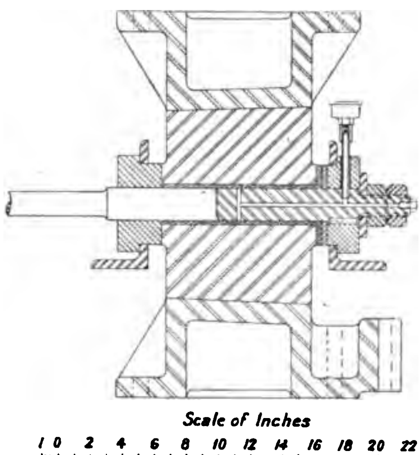


Fig. 227.—Section of Live Roller and Paths.

R_1 being the mean radius of roller path in feet.

The equation for the width of rollers is then—

$$w = \frac{1 \cdot 25\, W\sqrt{d}}{\cdot 33 \times 12 \times 2\pi R_1} = \frac{\cdot 05024\, W\sqrt{d}}{R_1},$$

W being the total weight in tons of the revolving structure plus load.

Fig. 227 is a section of a roller and path. The pin on which the roller runs is provided with a key to prevent it from turning, and so unscrewing the lock nuts, and it is fitted with a Stauffer lubricator.

Owing to the comparatively slow speed and intermittent working of cranes, the wheel loads adopted may be much higher than would be practicable for continuous high-speed work, such, for instance, as railway work. In connection with the latter, Mr. Mallock, in his paper on the action between wheel and rail at the Conference of the Institution of Civil Engineers in 1907, gave as the limit of wheel load for ordinary steel wheels and rails a pressure of 20 tons per square inch of surface contact between wheel and rail. Beyond this pressure indentation and cold-rolling of the material takes place, so that at high speeds the wheels and rails would be rapidly worn out.

In crane work this pressure is frequently exceeded, and cold-rolling of the surfaces takes place, as may be seen by examining the rails on any gantry on which heavy cranes are at work. At the same time the rate of wear is not unreasonable, owing to the slow speeds and intermittent nature of the work.

In order to lessen the wear, gantry rails of very hard steel are now being introduced, with wheels having centres of cast iron or ordinary steel and tyres of hard steel, to suit the rails. Where separate steel tyres are used, they should not only be hard enough to withstand the surface wear, but they should also be of ample thickness, otherwise they will be expanded by the rolling action, and will come off the wheels.

An example of an ordinary light cast-steel travelling wheel is given in Figs. 227*a* and 227*b*, the load on this wheel being 10 to 11 tons, and the linear speed 185 feet per minute. A heavier steel travelling wheel with a steel spur driving ring bolted to it is shown in Figs. 227*c* and 227*d*. The load on this wheel is about 20 tons, and the linear speed 325 feet per minute.

If no cold-rolling of the surfaces took place, the proportions of wheels for given loads might be calculated on similar lines to those of live rollers. Where cold-rolling takes place, however, the life of the wheel requires consideration, so that the speed should be taken into the calculation.

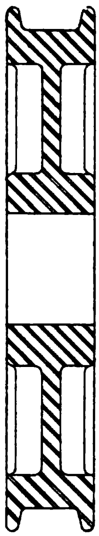
The equation for the dimensions of the wheel is then—

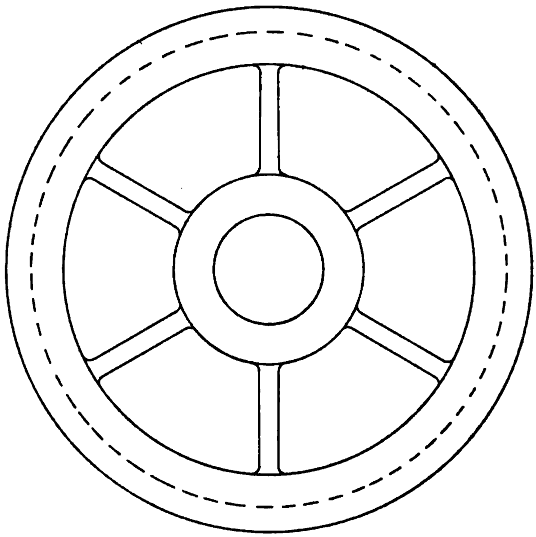
$$W = C_1 w \sqrt{d},$$

W being the load on the wheel in tons, w the width of tread in inches, and d the diameter in inches, C_1 being a constant. The time which the wheel will go in regular work before it requires re-turning will be approximately—

$$\tau = \frac{C_2 w \sqrt{d}}{SW},$$

τ being the time in years, and C_2 a constant.

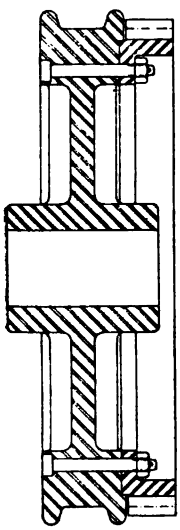


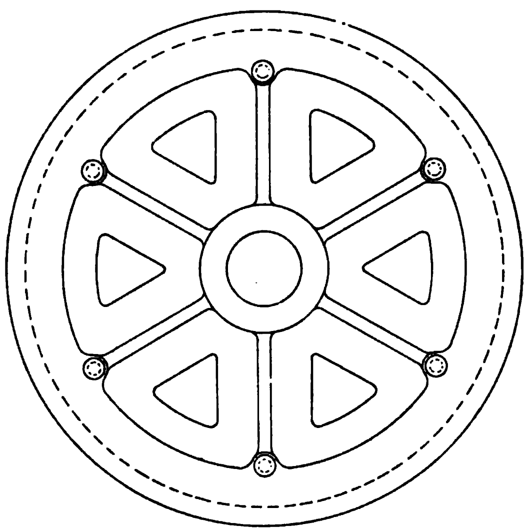


Scale 1½ Inches to a Foot.

Inches 12 9 6 3 0 1 Foot

Fig. 227*a*. Fig. 227*b*.





Scale 1 Inch to a Foot

Inches 12 9 6 3 0 1 2 3 Feet

Fig. 227*c*. Fig. 227*d*.

For the wheel shown in Figs. 227*c* and 227*d*, the value of C_1 is 1·62, and the value of C_2 is 3,694. For very heavy wheel loads it is more convenient to increase the width of tread than to increase the diameter of wheel. With wide treads, instead of a rail, a flat plate may be used, rivetted to the top of the girder on which the wheel runs.

CHAPTER XII.

BRAKES.

An extremely simple and efficient form of brake is that which may be seen in use on wells in the country. A piece of rope is tied to the winch frame and passed two or three times round the barrel in the direction in which it turns for lowering, the loose end being held in the hand. This is the coil brake in its simplest form, and it is remarkably efficient notwithstanding its simplicity.

The coil brake, in its more mechanical form, consisting of several turns of steel wire rope wound around a cast-iron or steel drum, has been used to some extent in crane work, while the coil principle applied in the form of the band brake is used more largely than any other.

The holding power of a coil or band brake may be ascertained by means of the following equations :—

The torque upon the brake drum due to the load hanging on the crane hook being T, the tangential force at the rim of the drum will be—

$$F = \frac{T}{r}, \qquad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

r being the radius of the drum in inches, F being in lbs., and T in inch-lbs. The total radial pressure required to be exerted by the coil or band in order to exactly balance this force =

$$P = \frac{F}{\mu}. \qquad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

When a band is passed round a drum, embracing any arc ABC, one end of the band being fastened to the framing of the machinery, and the other loaded with a weight (see Fig. 228), the tension T_1 in the rope on the side where it is fastened to the frame is equal to the tension T_2 on the other side due to the weight, plus the tangential force F due to the frictional grip of the band on the drum.

Thus the force $F = \frac{T}{r} = P\mu = T_1 - T_2$.

Referring to Fig. 229, and taking a small arc *DEF*, draw a tangential line perpendicular to *DO*, the length *ab* of which represents to any convenient scale the tension T_1 in the band at the point *D*, and a second tangential line *bc* representing the tension T_2 at the point *F*. On completing the parallelogram the inclined line *bd* represents the reaction at the surface of the drum, which balances the difference of the tensions in the band at the points *F* and *D*. The angle ϕ which the line *bd* makes with the radius *EO* is the angle of repose of the band on the rim of the drum, and the tangent of this angle is equal to the coefficient of friction μ. Drawing from

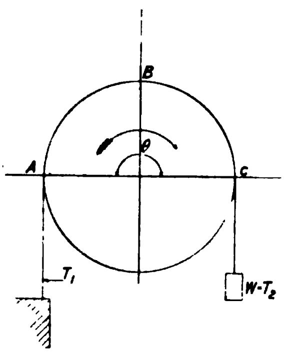


Fig. 228.

Arrow shows direction of rotation of drum when lowering.

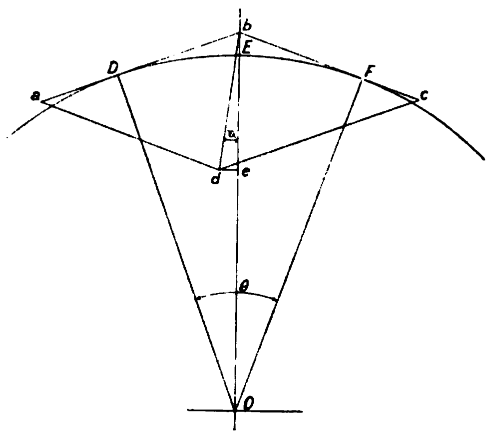


Fig. 229.

d the line *de* perpendicular to *EO*, the distance *be* represents the radial force exerted by the band upon the rim of the drum, and calling this P_1—

$$P_1 = \frac{T_1 - T_2}{\tan \varphi} = \frac{T_1 - T_2}{\mu}.$$

Taking the whole arc embraced by the coil, the ratio of the tensions in the band =

$$\frac{T_1}{T_2} = e^{\mu\theta}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which *e* is the base of the Napierian logarithms—2·718—and θ is the arc of embrace in circular measure. In dealing with a coil brake, it is more convenient to take the number of turns of rope around the drum than to take the arc in radians, and as one complete turn = 2π radians,

$$\frac{T_1}{T_2} = e^{2\pi\mu s}, \quad . \quad . \quad . \quad . \quad . \quad (4)$$

T_1 being equal to $T_2 + F$, the number of turns required in a coil brake is—

$$s = \frac{\log \frac{T_2 + F}{T_2}}{2\pi\mu \log e}. \quad . \quad . \quad . \quad . \quad . \quad (5)$$

As an example, take a coil brake having a drum 18 inches diameter to sustain a torque of 12,000 inch-lbs. Then $F = \frac{12{,}000}{9} =$ 1,333 lbs. Make the weight hanging on the rope, say, 56 lbs. This is then the value of T_2, and $T_1 = T_2 + F = 56 + 1{,}333 = 1{,}389$ lbs. Coefficient of friction, 0·15.

$$s = \frac{\log \frac{1{,}389}{56}}{2 \times 3{\cdot}14 \times {\cdot}15 \times \log 2{\cdot}718} = 3{\cdot}4.$$

Say, $3\frac{1}{2}$ turns.

To save time in making calculations, the curves shown in Figs. 230 and 231 have been worked out. Fig. 230 shows the relation between $\frac{T_1}{T_2}$ and the coefficient of friction for different numbers of turns on the coil, and Fig. 231 shows the relation between $\frac{T_1}{T_2}$ and the number of turns on the coil for different values of the coefficient of friction. Thus, in the example just given, $\frac{T_1}{T_2} = \frac{1{,}389}{56} = 24{\cdot}8$. Referring to Fig. 231, the number of turns corresponding to this value, with a coefficient of friction of ·15, is 3·4, which agrees with the above calculation.

In the calculations for band brakes it is more convenient to take the arc of embrace in degrees, so that the equation becomes—

$$\frac{T_1}{T_2} = e^{\mu\frac{\theta}{57 \cdot 29}} = e^{\cdot 017454\,\mu\theta}, \quad . \quad . \quad . \quad . \quad . \quad (6)$$

in which θ is the arc of embrace in degrees. In a band brake the arc of embrace is usually determined by considerations of mechanical

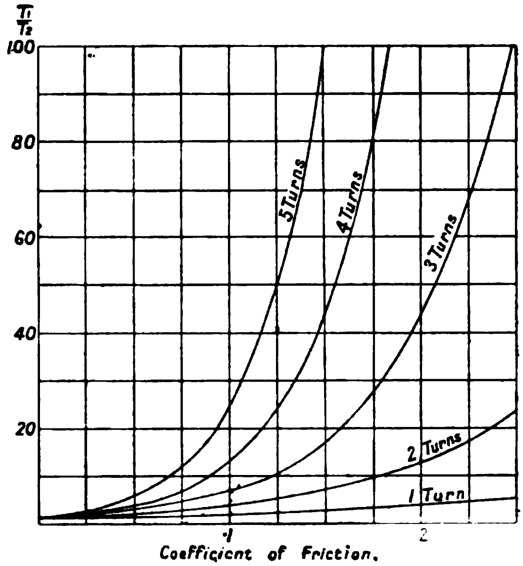


Fig. 230.

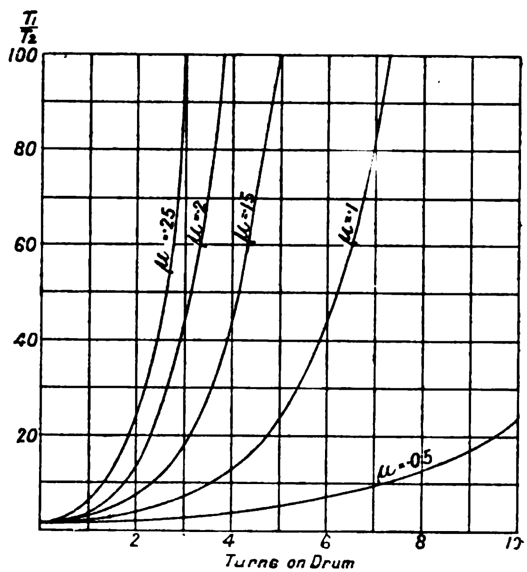


Fig. 231.

design, and the value of T_2 is calculated to suit, the equation being—

$$T_2 = \frac{F}{e^{\cdot 017454\,\mu\theta} - 1}. \qquad (7)$$

T_2 then represents the weight or force to be applied to the band in order that the holding power shall be equal to F. Band brakes, which are faced with leather, should be of such proportions as to avoid excessive pressure between the band and the drum. If the pressure is too high, the leather will be rapidly worn away, or it may become overheated and charred.

The tension in the band increases gradually from T_2 to T_1, the tension at any point being expressed by the equation—

$$T_y = T_2 e^{\cdot 017454\,\mu\theta_x},$$

T_y being the tension in the band at a point θ_x degrees from the point C (Fig. 228).

It was shown in Fig. 229 by graphic construction that the radial pressure exerted by the brake band at any point on the rim of the drum was proportional to the difference in tension over an extremely short length of the band at that point. This difference is equal to the rate of increase in the tensions, and may be found by differentiation, the dc being—

$$\frac{dT_y}{d\theta_x} = \cdot 017454\ T_2\mu e^{\cdot 017454\,\mu\theta_x} = \cdot 017454\,\mu T_y,$$

which is the same as $T_1 - T_2$ per degree at the point θ_x.

The radial pressure has already been shown to be $= \frac{T_1 - T_2}{\mu}$. So, taking P_y as the pressure at any point θ_x—

$$P_y = \cdot 017454\ T_2 e^{\cdot 017454\,\mu\theta_x} = \cdot 017454\ T_y.$$

This formula produces a curve, the integration of which gives a result which coincides exactly with the value found from equation No. 2.* The pressure per square inch =

$$P''_y = \frac{360\,P_y}{2\pi r w}$$

$$= \frac{T_2 e^{\cdot 017454\,\mu\theta_x}}{rw} = \frac{T_y}{rw}, \qquad (8)$$

* That is to say,

$$P = \int_0^\theta \cdot 017454\ T_2 e^{\cdot 017454\,\mu\theta_x} d\theta_x = \frac{T_2 e^{\cdot 017454\,\mu\theta}}{\mu} - \frac{T_2}{\mu} = \frac{T_1 - T_2}{\mu}.$$

in which r is the radius of the drum, and w its width, both being in inches.

If the curve of pressure per square inch be calculated by equation No. 8 and plotted, and the mean pressure ascertained, it will be found that this pressure coincides exactly with the mean pressure given by the equation—

$$P_{\text{mean}} = \frac{360\ P}{2\pi r w \theta}. \qquad (9)$$

As $T_1 = T_2 e^{\cdot 017454\,\mu\theta}$, the maximum pressure per square inch will be—

$$P_{\text{max}} = \frac{T_1}{rw}$$

$$= \frac{T}{r^2 w \left(1 - \frac{1}{e^{\cdot 017454\,\mu\theta}}\right)}, \qquad (10)$$

in which T, as before, represents the torque in inch-lbs. From this equation the dimensions of a brake drum for a given maximum pressure may be determined.

A maximum pressure of 65 lbs. per square inch has been found satisfactory in practice.

In band brakes lined with wood blocks the difference in radius between the drum and the band is considerable, and should be allowed for in the calculations.

The coefficient of friction of leather or wood on iron varies between ·3 and ·4. If in designing brakes the coefficient is taken at ·3, the dimensions of the brake will have a margin on the right side. Brakes on the coil principle are more particularly suited for cases in which the drum requires to be prevented from turning in one direction only, and in which the torque is moderate in amount.

Where the drum requires to be held against turning in either direction, and where the torque to be sustained is very heavy, block or disc brakes are preferable.

For sustaining extremely heavy loads, block brakes of the type shown in Fig. 233 are largely used, as their construction is simple and economical.

The brake shown, which was a comparatively small one for this type, sustained a torque of 968 inch-tons. The brake blocks were of hard wood, and the brake levers were a couple of 14 × 6 rolled-steel joists. The brake was for occasional use, and, owing

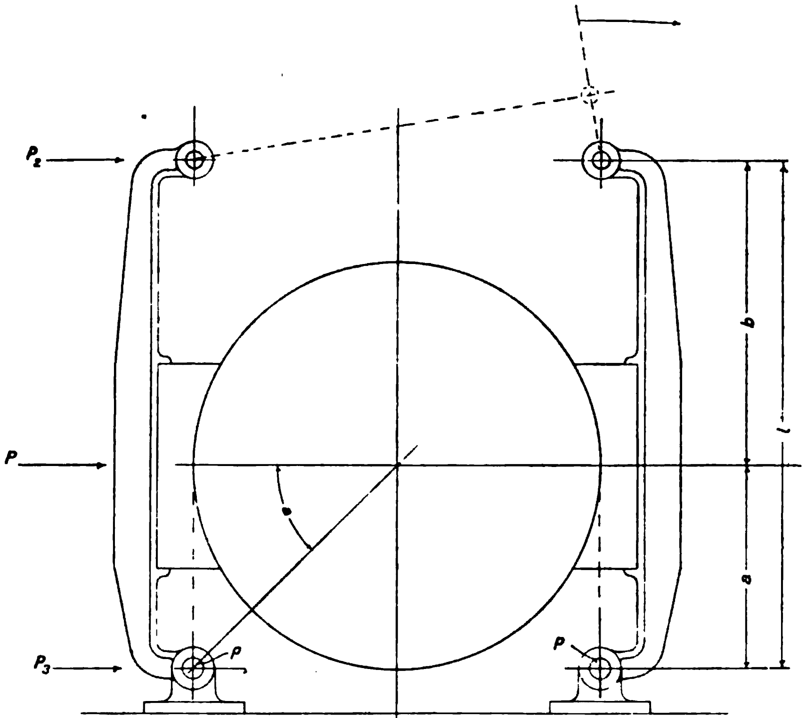


Fig. 232.

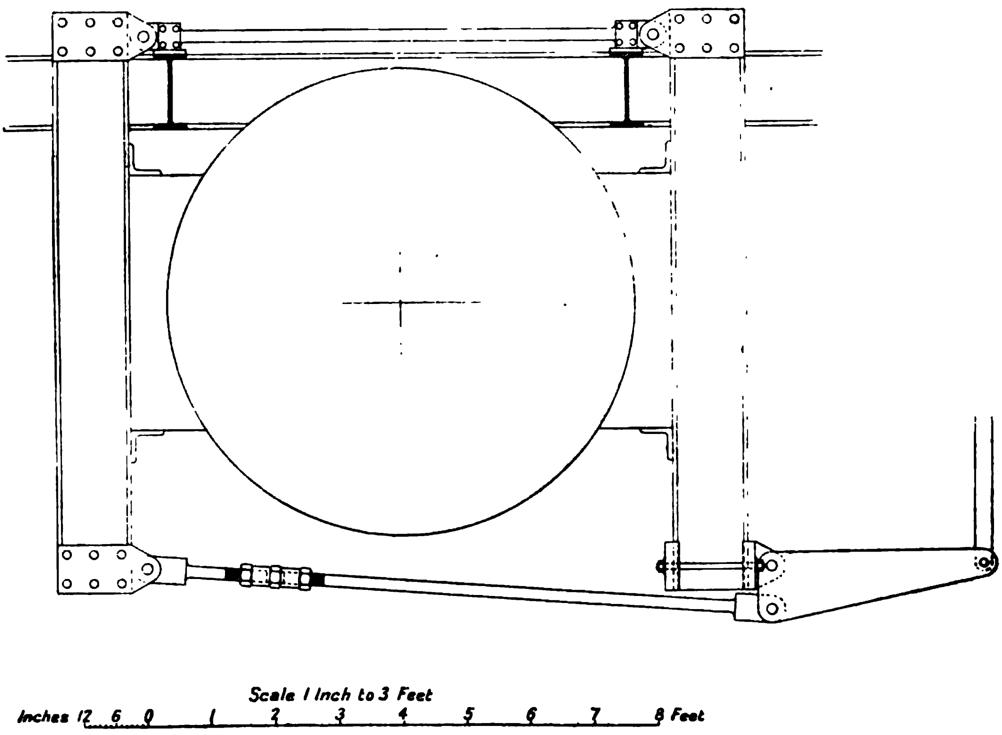


Fig. 233.

to the limitations of space, its dimensions were somewhat smaller than would be desirable for continuous work.

The following are the formulæ usually employed in the design of block brakes. For a brake of the type shown in Fig. 232, in which the centres of the pins *pp* are in line with the central point of contact of the block on the drum, the pressure required on each block, in order to sustain a given torque T is—

$$P = \frac{T}{2\mu r}. \qquad (11)$$

This equation is not strictly correct, as the drum in attempting to turn under the influence of the load, tends to carry the brake round with it, so leading to an increase of pressure on one block and a decrease on the other. The total holding power remains, however, about the same, so that the equation is sufficiently accurate for practical purposes. The force P_2 required at the top of the lever is—

$$P_2 = \frac{Pa}{l}, \qquad (12)$$

and the force P_3 on the pin p is—

$$P_3 = \frac{Pb}{l}. \qquad (13)$$

Owing to the tendency of the drum to carry the brake round with it, the force on one pin is increased, while on the other it is diminished. The force on each pin due to this tendency is—

$$F_1 = \frac{F}{2 \sec \alpha}, \qquad (14)$$

which requires to be added to P_3 for the one pin, and deducted from P_3 for the other, P_3 being first resolved into the same direction as F_1.

In equations 11, 12, 13, and 14, α, a, b, and l refer to the similarly lettered dimensions on Fig. 232.

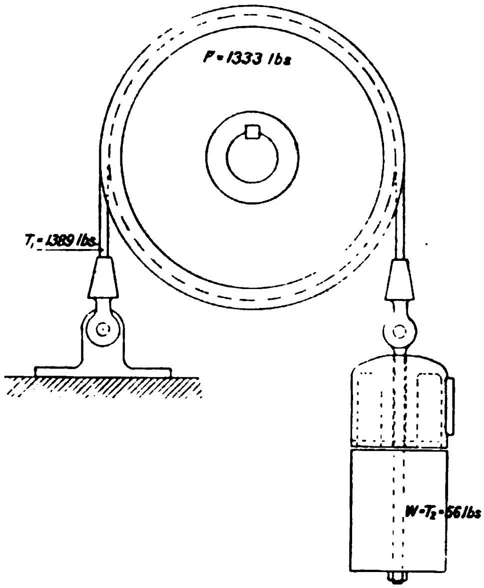
In disc brakes, such as that shown in Fig. 7, the end pressure required is—

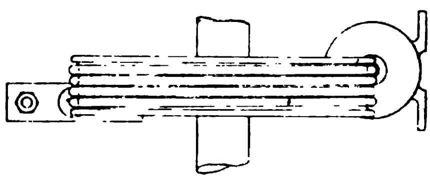
$$P = \frac{T}{r_9 \mu n_s}, \qquad (15)$$

in which r_9 is the mean frictional radius and n_s the number of the friction surfaces.

In most electric cranes, the load is sustained by an electro-magnetic brake, the action of which is automatic. The brake is normally held on by a weight or spring, and an electro-magnet is provided which, when energised, lifts the weight or pulls back the

Fig. 234.





Scale 1 inch to a Foot.

Inches 0 1 2 3 Feet

Fig. 235.

spring, and so releases the brake. The exciting coil of the electro-magnet is connected in the circuit of the hoisting motor, so that when current is switched on to the motor the brake is released, and when current is switched off the brake renews its hold. Thus, the brake is always either hard on or entirely off, and its only purpose is to sustain the load when the hoisting motor is not running. It cannot be used to control the speed of lowering unless fitted with an independent hand or foot release.

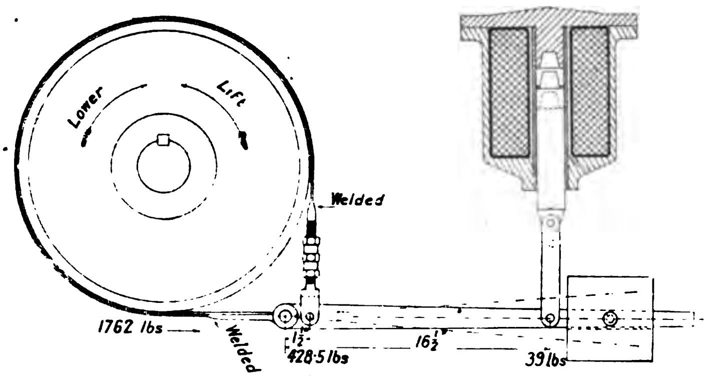


Fig. 236.—Flanges of brake drum not shown; drum 3 inches wide between flanges.

Figs. 234 to 241 illustrate a series of electro-magnetic brakes, all designed for a torque of 12,000 inch-lbs.

Figs. 234 and 235 show a coil brake in which an electro-magnet

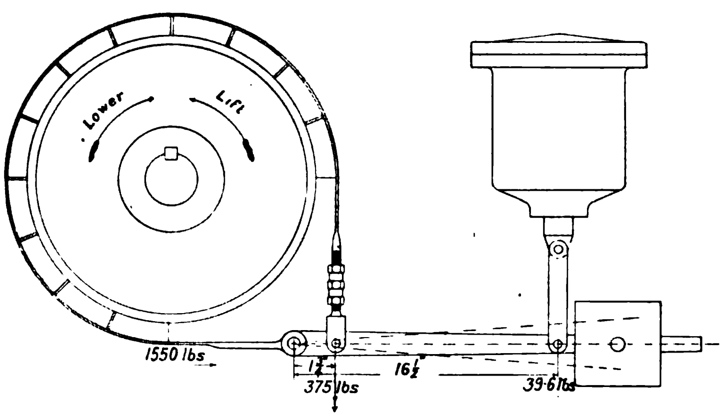


Fig. 237.—Figures give tension in strap.

is employed to lift the weight T_2 hanging on the rope, and so release the brake. In this case the rope is shown bearing directly on the drum, but in many cases the rope is fitted with brass wearing pieces placed fairly close together. These brass pieces bear on the drum, and so prevent wear of the rope. In order to allow for the elasticity of the rope, the lift of the electro-magnet in this case is fairly large—namely, $\frac{3}{8}$ inch. As the rope stretches, the magnetic gap increases, and can be re-adjusted by means of the nut at the bottom of the weight. With larger brakes of this type the weight

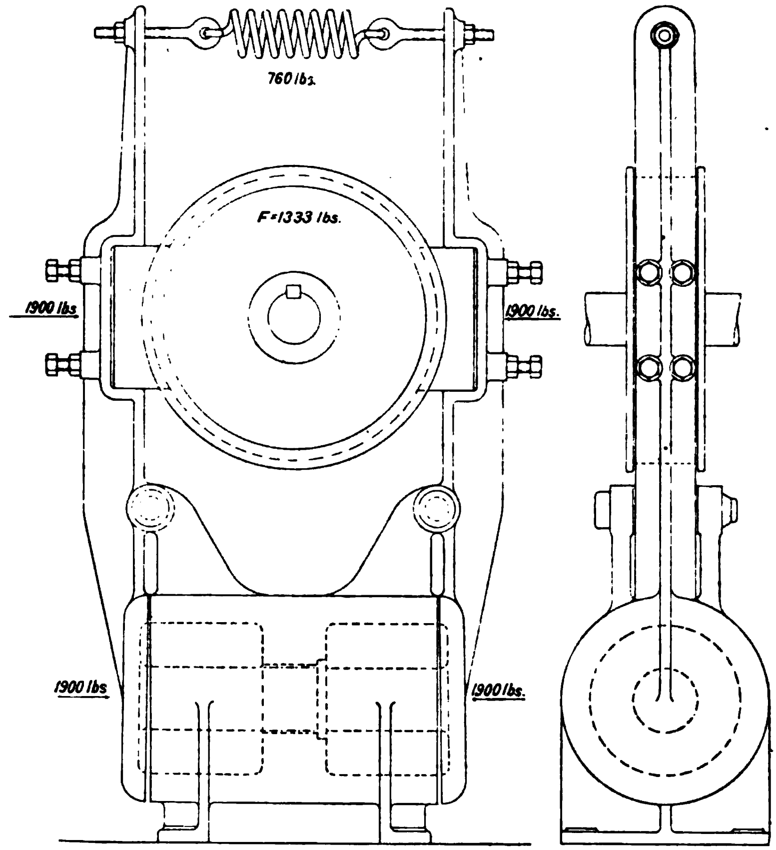


Fig. 238. Fig. 239.

requires a fairly long lift, and to obtain this it is preferable to use a solenoid plunger magnet instead of an ordinary electro-magnet.

Fig. 236 illustrates a leather-lined band brake. To allow for the slight compression of the leather, a movement at the end of the band in a brake of this size of at least ¼ inch should be provided, and to give this amount of movement the end of the lever requires to move through 2½ inches. Solenoid plunger magnets are, therefore, necessary for this type of brakes. The same brake lined with wood instead of leather is shown in Fig. 237. It will be noted that, owing to the greater radius of the band in this case, the forces are slightly modified.

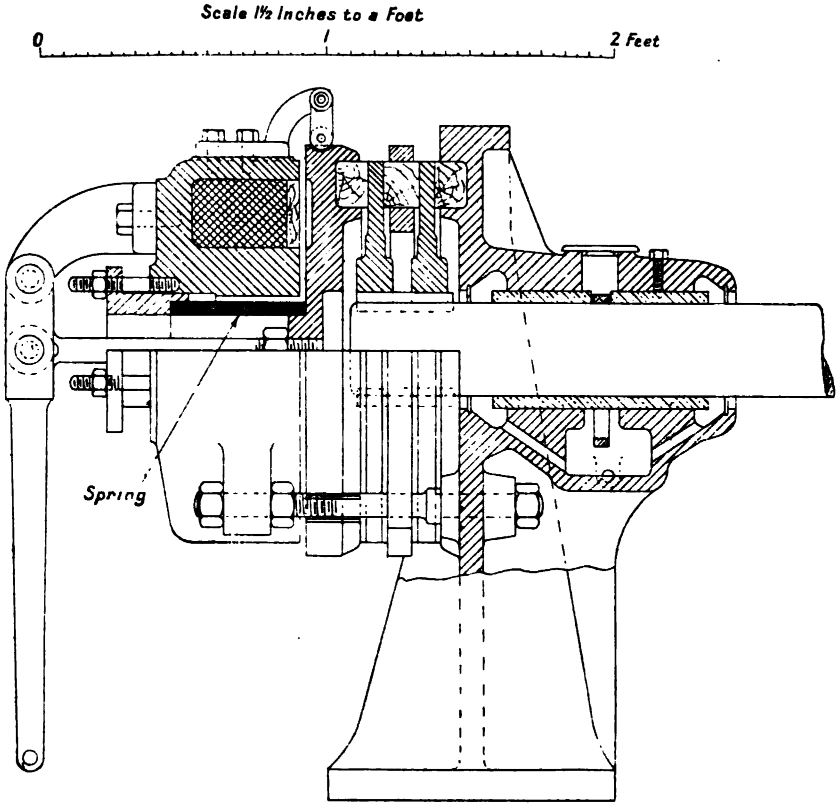


Fig. 240.

A form of block brake which has been extensively used is shown in Figs. 238 and 239. In this the brake levers are extended, and have enlarged ends, which form the armatures of an electro-magnet, which itself forms the framing to carry the brake. The blocks are pressed on to the drum by a helical spring, the tension in which can be adjusted, while the wear of the brake blocks can be taken up by means of the set screws shown on the drawing.

Fig. 240 is a section of the disc brake provided for the hoisting motion of the ropeway winding gear shown in Figs. 68 and 69. The brake was provided with a hand release, so that loads could be lowered without putting current on to the motor. The test figures of this brake are given in Fig. 241.

In Table XI. (p. 180) are given particulars of the magnet windings of the foregoing brakes, together with the watts required to release the brakes.

For brakes fitted with helical springs it is best to leave the final

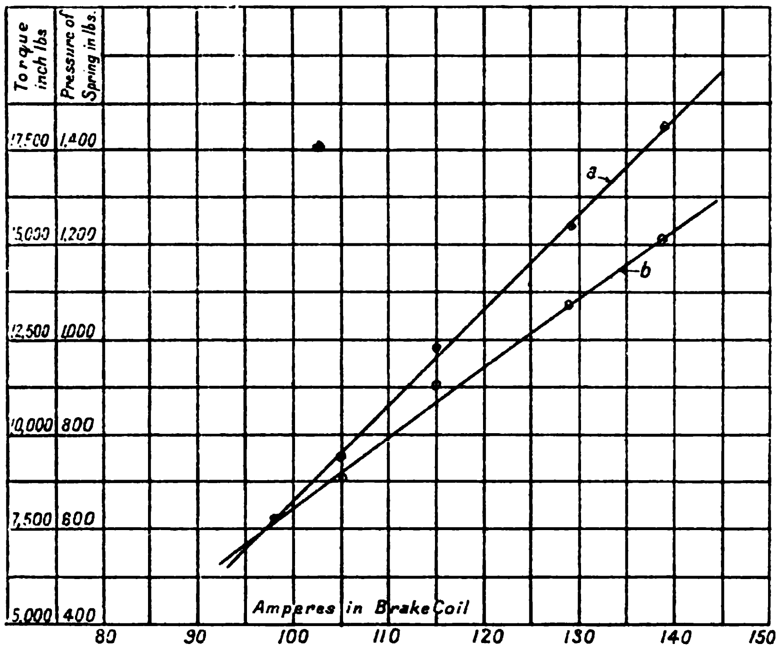


Fig. 241.

Air gap, 0·1 inch. Coil, 24 turns.

a = torque. b = spring pressure.

dimensions of the springs to the spring makers, simply specifying the duty which the springs are to perform. In laying out the design of the brake it is, however, advisable to ascertain the approximate size of the spring, in order to make sure of allowing sufficient space for it, and to avoid specifying something which is not possible.

Fig. 242 gives approximate proportions of springs such as would be suitable for use on the brakes described in this chapter.

The spring is subject to maximum load when the brake is held

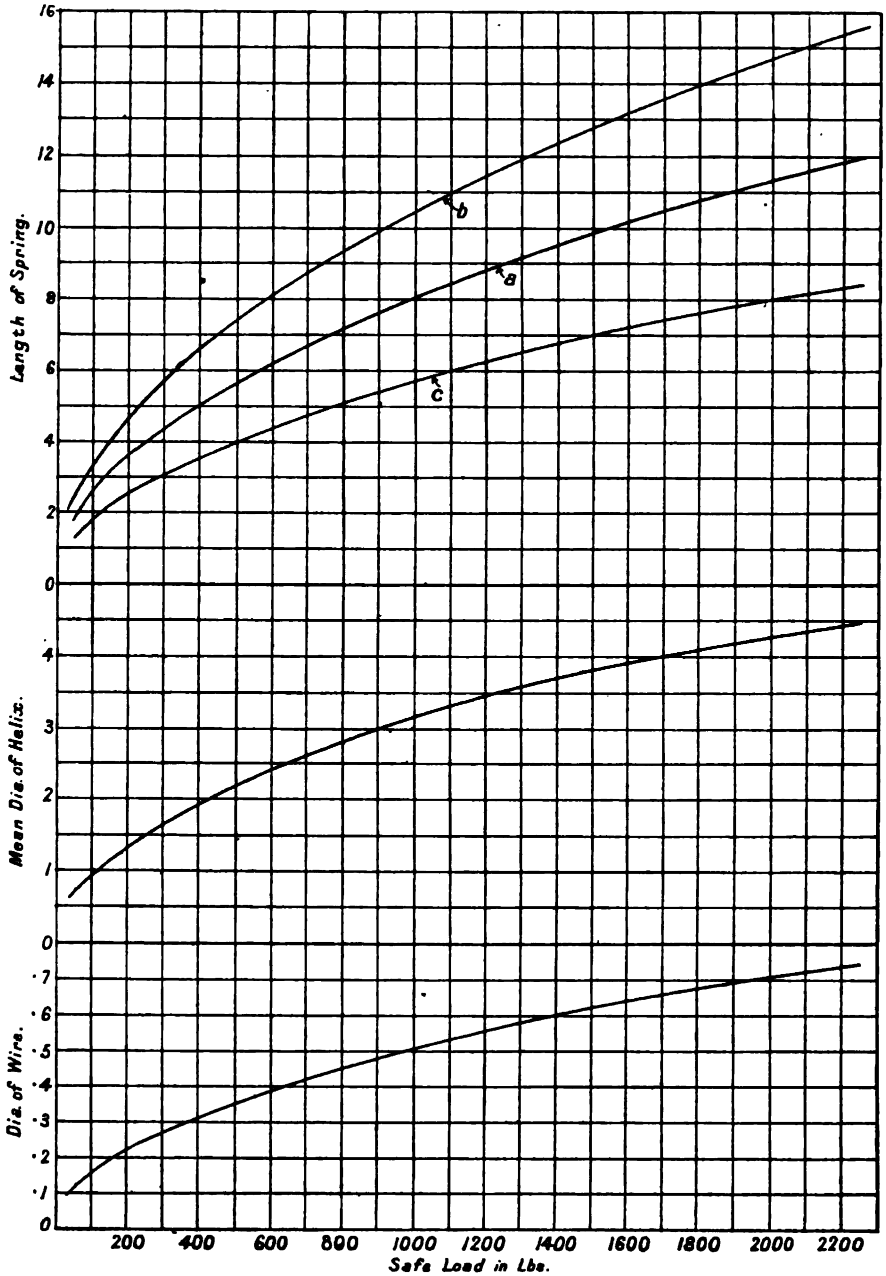


Fig. 242.

Each spring contains 8 turns of wire.
Pitch = twice diameter of wire.
a = length of spring unloaded.

b = length of tension spring fully loaded.
c = ,, ,, compression spring fully loaded.

All dimensions in inches.

off by the electro-magnet, and it should then have a margin of about 10 per cent. within the load given in Fig. 242. Thus, if the maximum load on a spring will be 1,000 lbs., the proportions to take from the diagram will be those corresponding to a load of 1,100 lbs. Where there is any doubt as to the amount of the maximum load a larger margin should be allowed.

Although the proportions given in Fig. 242 are sufficiently near for use in laying out the design of the brake they are not accurate enough for making the spring. As already mentioned, the actual dimensions of the spring should be left to the spring makers, the particulars specified with the order being (1) the maximum load, (2) the external diameter of the spring, (3) the length free, and (4) the length when subject to the maximum load.

Fig. 243.

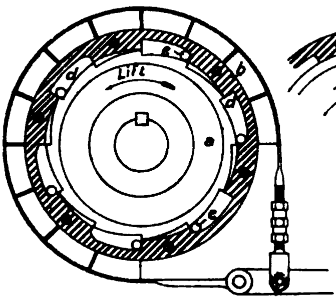


Fig. 244a.

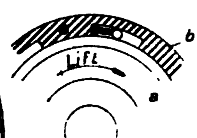
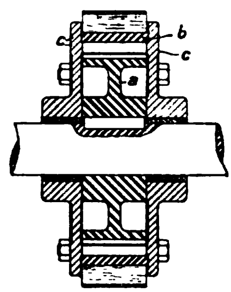
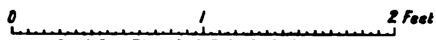


Fig. 244.





Scale for Figs. 236, 237, 243, 244, 245. & 246. 1 Inch to a Foot.

Automatic self-sustaining brakes are occasionally required, one form being shown in Figs. 243, 244, and 244*a*. The wheel *a* is keyed on the shaft. The ring *b* is carried by the discs *cc*, which are free on the shaft, but prevented from rotating by the band brake, which clasps the ring *b*. This brake is normally held on by a weight or screw. On the inner periphery of the ring *b* tapering recesses *d* are cut, and corresponding recesses are cut on the rim of the wheel *a*. Each recess in the ring contains a hardened steel roller *e*. When the wheel turns in the direction for hoisting, the rollers run to the large end of the recesses in the ring, and so permit the wheel to revolve. When the wheel tries to run in the reverse direction under the influence of the load, the rollers run to the opposite ends of the recesses, and act as keys to lock the wheel to the ring. To lower the load, the brake is eased off, and the wheel and ring then revolve together in the direction for lowering.

Sometimes the wheel is arranged with a plain surface, and the recesses have a very slight incline, so that when the wheel turns back the rollers jam the wheel and ring together, the rollers being pressed by springs towards the smaller end of the recesses, so as to ensure that they all grip equally. In this case the grip is frictional and there is a bursting force on the ring (see Fig. 244*a*).

The coil brake may be used as a simple non-return brake. Referring to Figs. 234 and 235, the brake shown exerts a torque of 12,000 inch-lbs. in the direction for lowering. If the electro-magnet be removed, and the weight of 56 lbs. be allowed to remain hanging on the rope, the resistance offered by the brake in the direction for hoisting cannot exceed 518 inch-lbs., which is negligible in comparison with the torque required for hoisting. For lowering a load, the brake is eased off by means of a hand or foot lever, which lifts the weight.

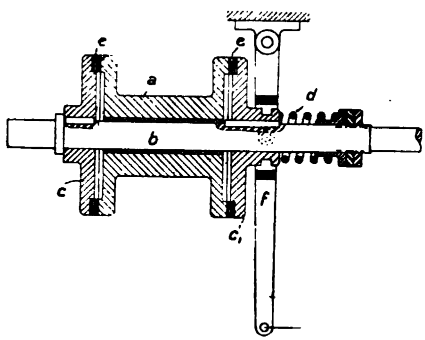


Fig. 245.

In cases where the hoisting gear is accessible to the driver, satisfactory control over the speed of lowering may be obtained by an ordinary hand or foot brake, by an electro-magnetic brake fitted with a hand or foot release arrangement, as in Figs. 68 and 241, or by a free-barrel arrangement, as shown in Fig. 245. This was a small winch for lifting about 5 cwts. The drum *a* was free to turn on the shaft *b*. At one end of the drum a disc *c* was secured to the shaft, and at the other end a second disc c_1 was arranged to slide on a feather key. The discs and drum were all pressed together by the spring *d*, friction rings of wood *e* being provided between the discs and flanges of the drum. The shaft *b* was driven by self-retaining worm gear, so that the load could not drive the winch back when the motor stopped. To lower the load the spring was drawn back by the lever *f*, so relieving the pressure on the friction surfaces, and allowing the drum to run round. Thus, the friction surfaces acted as a clutch for hoisting and as a brake for lowering.

Where the hoisting gear is not directly accessible to the driver, as, for instance, on the crabs of overhead cranes and Goliaths, the electro-magnetic brake may have a release arrangement operated by cords from the driver's cage, as in Figs. 8, 9, and 18, or the electro-magnetic brake may be supplemented by an automatic disc brake, as illustrated in Fig. 246. On the motor shaft *a* a quick thread screw *b* is provided. The screw engages with a nut, *c*, the outer end of which slides on feather keys on the hoisting shaft *d*. On the same keys there are friction discs *e*, and other friction discs *f* are arranged to slide on keys or projections on the inner periphery

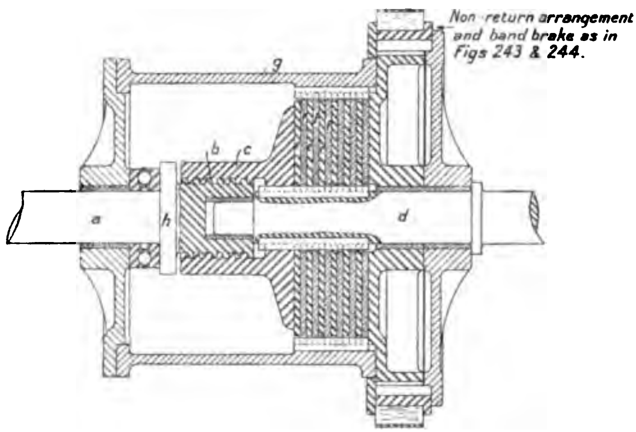


Fig. 246.—Automatic Lowering Brake.
12,000 inch-lbs.; screw, ¾-inch pitch, 4 thread, 3-inch lead.

of the case *g*, which is free on the shaft, but prevented from turning in the direction for lowering by a non-return arrangement and band brake similar to that shown in Figs. 243 and 244. The action of the brake when hoisting is as follows :—

The motor being started, the screw revolves, while the nut, being keyed to the hoisting motion shaft, is prevented from turning owing to the resistance offered by the load on the hook. The nut is, therefore, driven along by the screw, and presses upon the discs until it is tight enough to act as a coupling between the two shafts. The discs, casing and hoisting shaft, then all revolve together at the same speed as the motor shaft. When the motor stops the non-return arrangement prevents the load running back. When

12

the motor shaft is started in the opposite direction the brake cannot turn with it, as the non-return arrangement prevents it. Therefore, the screw draws the nut back, and relieves the pressure on the friction discs, so allowing the load to run down. As the nut is keyed to the hoisting shaft, if this shaft and the motor shaft are running at the same speed, the nut remains stationary longitudinally, and puts no pressure on the discs. If, however, the load tries to overtake the motor, the nut travels along the screw, and puts pressure on the discs, so checking the speed of the load. When the motor stops, the load causes the nut to press the discs up tight, and it comes to a stop. Thus the load when lowering follows the motions of the motor, starting when it starts, running fast when it runs fast, slow when it runs slow, and stopping when it stops, the principle being exactly similar to that of the valve motion of a steam steering gear. When there is no load on the hook a positive drive is generally required to send it down. In this case when the motor is started the nut runs back till it presses against the disc *h*, and it now acts as a coupling to the two shafts in the lowering direction.

In designing these brakes the angle of thread of the screw should be made not less than the angle of repose of the metal, so that it will unscrew easily. If the angle of thread is too small, the nut will be screwed up very tightly when hoisting, and when starting to lower a very large current will be required, in order to get the nut free, and this will cause the lowering to start with a violent jerk. The principal object of the ball thrust bearing is to reduce the force required to unscrew the nut, and so help to give a gentle start. The angle a_t of the screw thread at the mean radius (see Fig. 291) may be found from the equation—

$$\operatorname{Tan} a_t = \frac{p_1}{2\pi r_7}. \quad . \quad . \quad . \quad . \quad . \quad (16)$$

The end pressure on the screw when hoisting is—

$$P = \frac{T}{r_7} \cdot \frac{2\pi r_7 - \mu p_1}{2\mu\pi r_7 + p_1}. \quad . \quad . \quad . \quad . \quad . \quad (17)$$

And this is also the tension on the case and on the bolts which secure the ends of the case.

The end pressure required on the discs, in order to sustain a given torque, may be found from equation No. 15. In order to ensure good control over the lowering, this end pressure should be kept very low, say, $\frac{1}{10}$ to $\frac{1}{20}$ of the value of P found from formula No. 17. The number of friction surfaces required is then—

$$n_s = \frac{CT}{\mu r_0 P}, \quad . \quad . \quad . \quad . \quad . \quad (18)$$

in which the constant C may be taken at from 10 to 20. The brake casing should contain a quantity of oil, so that all parts will be thoroughly lubricated.

The object of the band brake is to provide an easy means of getting the load down in case the automatic brake should jam when hoisting a load. It is only provided in special cases, a ratchet wheel and pawl being generally considered sufficient.

Control of the speed of lowering may be effected electrically by taking advantage of the fact that a motor, when driven mechanically, generates current. When a series hoisting motor is used, the controller is so arranged that on the lowering side it alters the motor connections to those of a series dynamo, so that when driven by the descending load the motor generates current, and opposes a torque, which keeps the speed within the desired limits. The current generated is absorbed by resistance, and the system is consequently known as rheostatic braking. This system nearly always has a fascination for those who have newly taken up electric crane work. When a firm of crane builders commence making electric cranes, or a firm of switch makers commence making crane controllers, one of the first things they almost invariably do is to invent rheostatic control, and take out a patent for it. The system has, however, some disadvantages, which are dealt with in the chapter on Controllers.

Another system of electrically controlling the speed of lowering is the regenerative system, which differs from the rheostatic system in that a shunt wound motor is employed, and the current generated by the descending load is returned to the circuit instead of being passed through resistances. The author has employed this system in a number of cases, and test results are given in Table I. and in Figs. 13 and 73. The regenerative system offers the advantages of some slight economy in current consumption, owing to the return of current to the circuit, a very perfect control of the lowering speed, especially where extremely slow speeds are required for adjusting the load, and a saving in cost of brakes. There are cases, however, in which its use necessitates special arrangements.

TABLE XI.—MAGNET WINDINGS OF ELECTRO-MAGNETIC BRAKES.

Figure.	Current to Pull Off.	Full Load Current of Motor.	Size of Wire.	Turns.	Resistance. Ohms.	Watts with Full Load Current.	Watts per Sq. In. of Surface.	Weight of Wire, Lbs.	Pressure on Brake Surfaces.
234 and 235	185	380	19/13	24	·0014	202	2·88	10·2	195·8 lbs. per linear inch. Maximum. 44·9 lbs. ,, Mean. 65·2 lbs. per square inch. Maximum.
236 and 237	185	315	0·432 dia.	75	·0054	540	2·88	55·46	34·82 lbs. ,, Mean.
238 and 239	185	415	37/14	40	·0065	1120	2·88	102	52·8 lbs. ,,
240	185	370	19/12	24	·0037	505	2·88	44	30·5 lbs. ,,

Note.—All the above brakes are series wound, and are arranged to pull off with less than the full load current of the motor. (See remarks on series coils in chapter on Design of Magnets.)

If a few cranes are working on a circuit on which there are a number of motors working regularly, the regenerative system is very suitable, as the current fed to the circuit by the cranes when lowering helps to drive the other motors on the circuit, and relieves the generating plant to that extent.

If the circuit consists of cranes only, the regenerative current from a crane which is lowering will generally go to a crane which is hoisting, travelling, or traversing; but it may occur that a crane will be lowering when all the others are stopped, or it may even happen that all the cranes are lowering at once. In such case the only path for the regenerative current is through the armature of the generator. The generator then becomes a motor, and drives the engine. If the electric energy supplied to the generator does not exceed that necessary to overcome the mechanical losses of the combined plant, all that happens is that the speed accelerates slightly, the governor shuts off steam, and the dynamo drives the engine until the regenerative current ceases, when the normal conditions are restored. The regenerative energy may, however, exceed this amount. There is then a further acceleration of speed of the dynamo and engine, and as the back E.M.F. of the dynamo increases, the speed of the crane motor also increases, leading to a further increase of the dynamo speed. Thus the speed of the dynamo and engine and motor continually accelerates. As a rule, the time during which a load is being lowered is too short to allow of the speeds under these circumstances becoming dangerous, but control over the speed of lowering is lost, and the load runs down rapidly. This may be prevented by the provision, on one or two steps of the controller, of resistances into which the regenerative current may be diverted. When the driver finds the load going down too fast he draws back the controller handle to slow it, and in doing so breaks connection with the main circuit, and introduces resistance across the motor brushes. This checks the speed of the motor, and as the dynamo no longer receives the regenerative current, the speed of the dynamo and engine returns to the normal. A much better plan, however, is to employ accumulators on the circuit. Any regenerative current then goes to charge the accumulators, and no trouble will be experienced from variations of speed. The question of the employment of accumulators on crane circuits is dealt with in a subsequent chapter.

CHAPTER XIII.

TOOTHED GEARING.

In Fig. 247 we have an elementary toothed gear consisting of two levers turning on the centres *AB*, having, where they bear upon each other, surfaces so shaped that their point of contact travels along the line *CD*, and, notwithstanding the constant change in the length of the respective radii of the two levers at their point of contact, the angular velocity of the driven lever coincides with that of the driver from the point *C*, at which contact commences to the point *D*, at which it ends, the effect produced being the same as if the two circles *EF* (the pitch circles) rolled upon each other without slipping.

The curves of the contact surfaces are involutes, and their shape is obtained as follows :—The line *AB* is divided at a point *P* into two lengths *AP*, *BP* the respective dimensions of which are inversely proportional to the desired angular velocities of the two levers. Thus, if lever *A* were required to have half the angular velocity of *B*, the length *AP* would be twice *PB*.

In the present instance the angular velocity of the levers is identical, so that the point *P* bisects *AB*.

Through *P* is drawn the line *CD* at any chosen angle with *AB*, in the present case 70°. The complement of this angle, 20°, is known as the angle of incidence a of the teeth.

Two circles *GH* forming the base circles of the involutes, and to which the line *CD* is a tangent are then struck, and from these circles involute curves are drawn from the points *C* and *D* up to the circles *IJ*, which cut the points *CD*, and correspond with the tops of the teeth in a complete gear wheel. The arc A_c through which the levers turn in passing from *C* to *D* is known as the arc of action, and in a complete wheel should be a definite multiple of the pitch of teeth.

A sufficiently close approximation to the involute can be

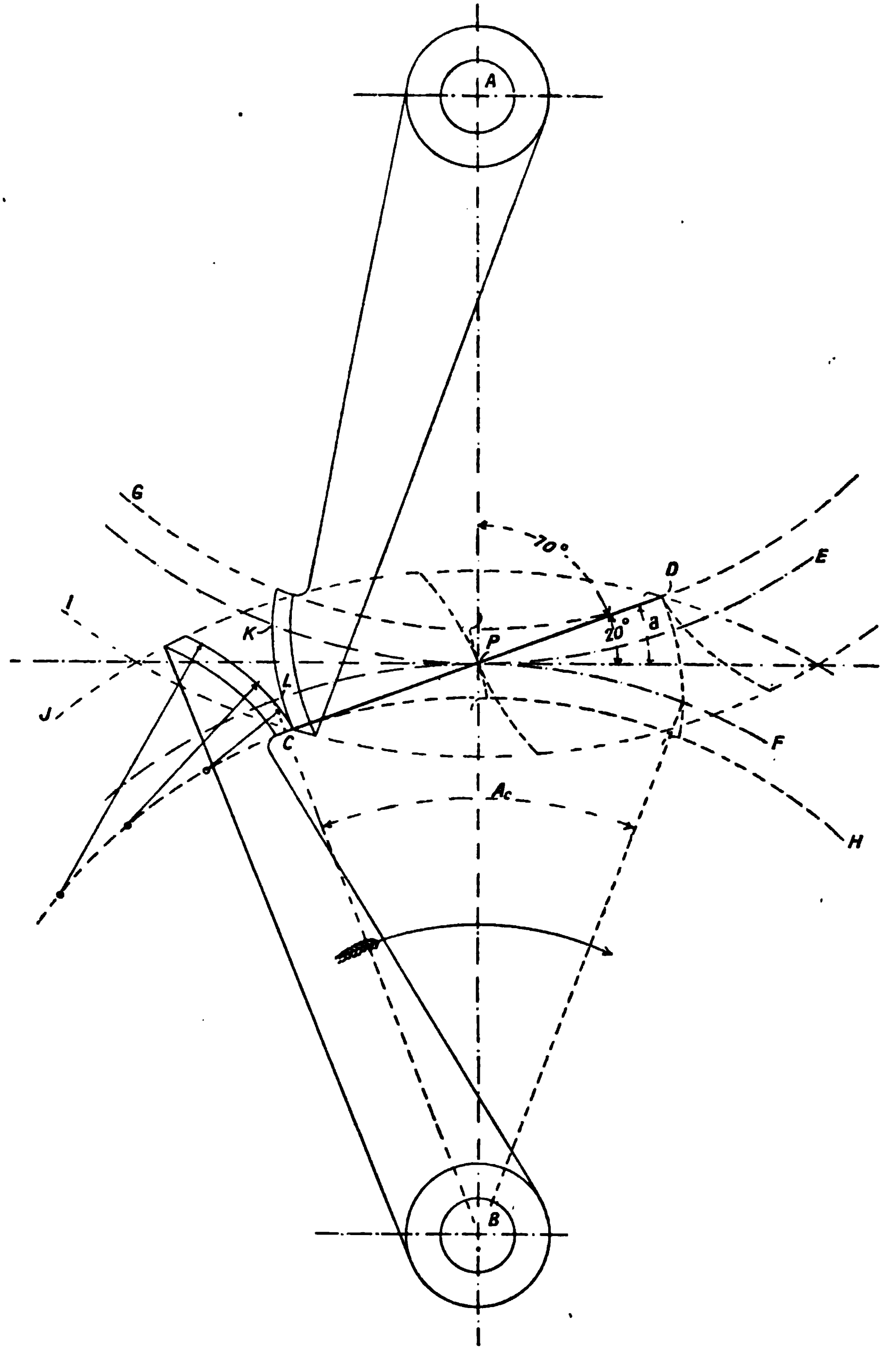


Fig. 247.

obtained by striking circles from two or three points on the base circles as shown.

Owing to the fact that the teeth bear upon each other at the angle a, there is in addition to the tangential driving force, a radial force tending to thrust the centres AB apart.

It is usually stated that in gear having involute curves the velocity of the driven wheel will be uniform with that of the driver, even if the distance between the centres A and B is varied. This statement, although true, requires qualification. If the distance AB is decreased, the base circles GH are brought nearer together, and the line CD being a tangent to these is correspondingly shortened. If the working surfaces bear upon each other beyond the points CD, the uniformity of the velocity will be destroyed. As they must, therefore, only bear along CD, the effect of shortening this line by lessening AB is to shorten the period during which uniformity of driving is obtained. If, on the other hand, the distance AB is increased, the base circles are moved further apart, and CD is lengthened. As, however, the levers are now further apart, their working surfaces will only bear along a portion of the distance CD.

Thus, the effect of either increasing or decreasing the distance apart of the centres is to shorten the period during which uniform driving is obtained.

This point requires to be taken into account when designing gears which are to work with varying centres.

Neglecting the effect of friction, the line CD which is always normal to the surfaces at their point of contact gives the direction of the resultant force between them.

As the radii of the levers at their point of contact constantly vary, we obtain with constant torque a variation of the tangential and radial forces with the varying positions of the levers, although the resultant remains constant.

This is shown in Figs. 248, 249, and 250, in which the varying forces and dimensions are figured, the torque of the driven lever being taken at 100 inch-lbs., and the radius of the pitch circles EF at 6 inches.

In Fig. 248 the working faces have just come into contact at the point C. In this position the radius r_1 of the driving lever at its point of contact is 5·638 inches, and the torque being 100 inch-lbs., the tangential force is 17·74 lbs. In this case the direction of the tangential force coincides with that of the line CD, so that in this

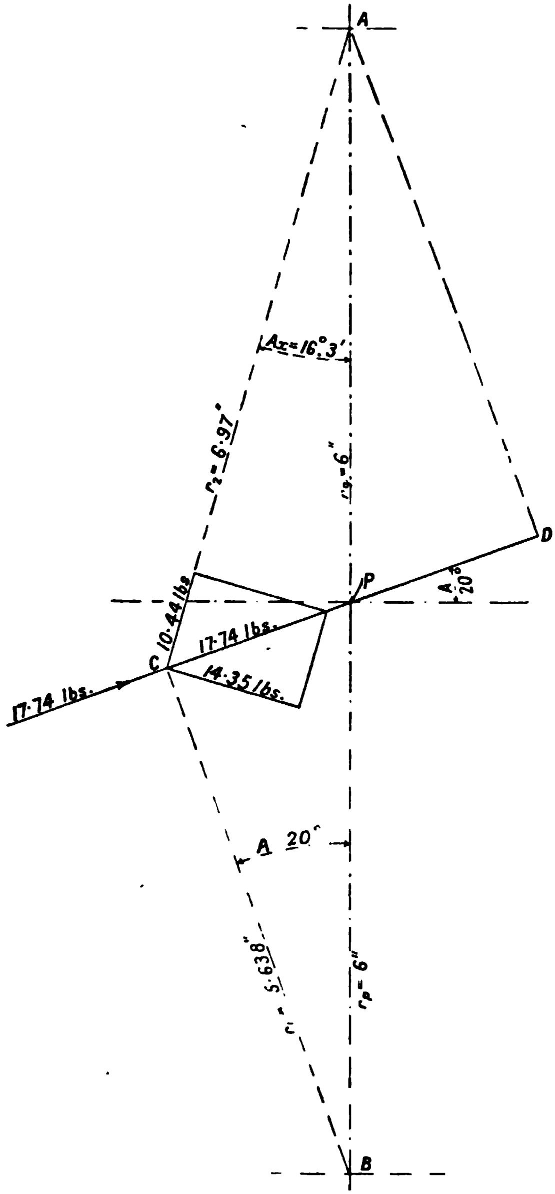


Fig. 248.

position there is no radial thrust on the driving lever. The radius of the driven lever r_2 being 6·972 inches, the tangential force is

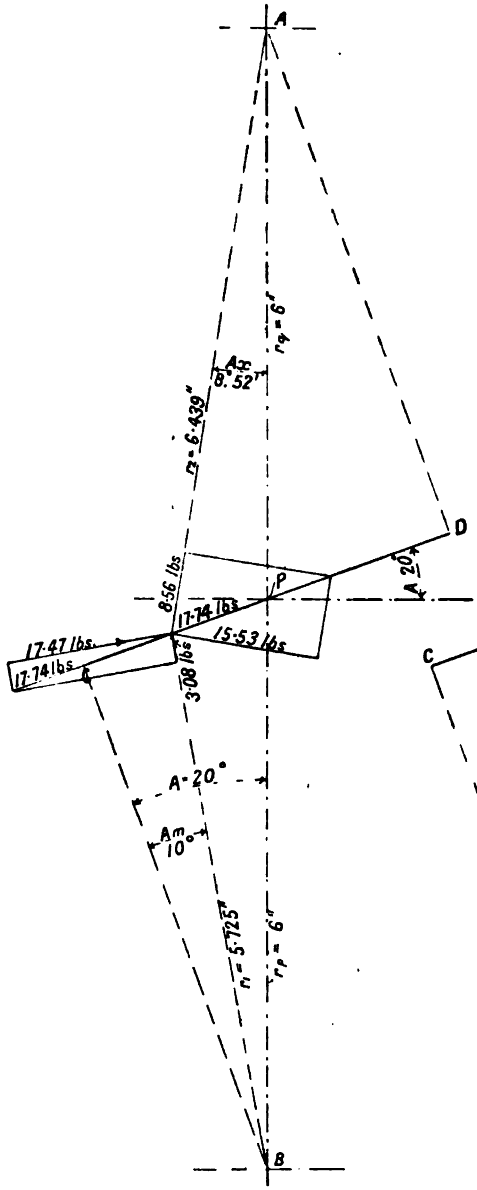


Fig. 249.

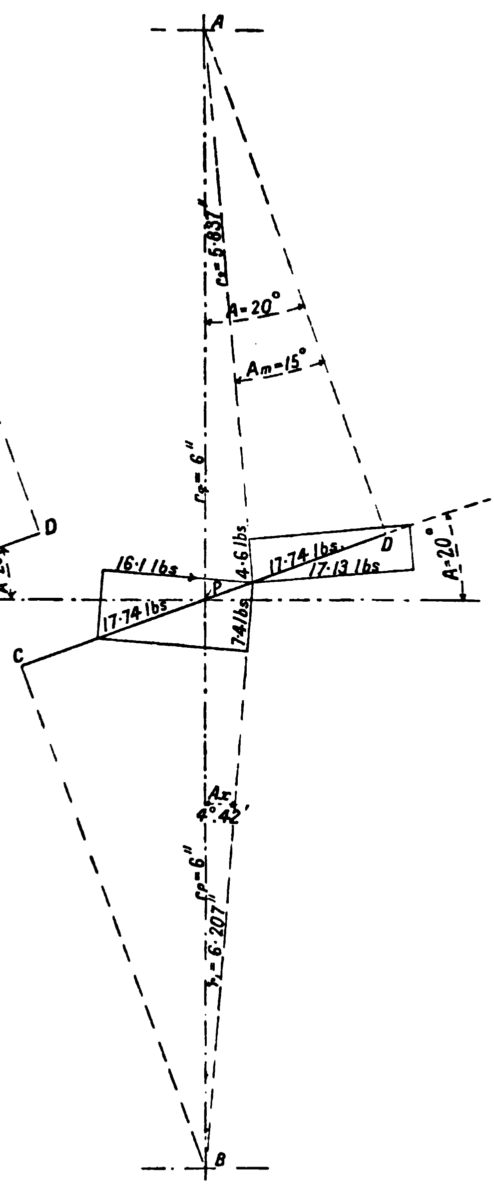


Fig. 250.

14·35 lbs. The radial thrust is 10·44 lbs., and the resultant along *CD* is 17·74 lbs.

In Fig. 249 the point of contact has moved 10° from its original position, and the new values of the various forces and dimensions are figured on the diagram.

In Fig. 250 the point of contact has moved 5° beyond the pitch point P, and the various values are again figured.

Although the forces may be ascertained by means of diagrams, it is usually more convenient to calculate them ; the formulæ being as follows :—

$$r_1 = \frac{r_p \sec A_m}{\sec A}.$$

$$\operatorname{Tan} A_x = \frac{r_1 \sin (A - A_m)}{r_p + r_q - \dfrac{r_1}{\sec (A - A_m)}}.$$

$$r_2 = \left(r_p + r_q - \frac{r_1}{\sec (A - A_m)}\right) \sec A_x.$$

Torque of driven lever $= T$. Torque of driving lever $= T_1$.
Tangential force, at point of contact, of driven lever $= F$.
Tangential force, at point of contact, of driving lever $= F_1$.

$$F = \frac{T}{r_2}. \qquad F_1 = \frac{T_1}{r_1}.$$

Radial force on driven lever $= F_2$.
Radial force on Driving lever $= F_3$.

$$F_2 = F \tan (A + A_x). \qquad F_3 = F_1 \tan A_m.$$

Resultant force normal to working surfaces $= F_4$.

$$F_4 = \frac{T}{r_q} \sec A.$$

F_4 also equals the load on the shaft due to the driving force.

The above formulæ apply during the period of approach from C to P. For the period of recess from P to D they require to be slightly altered, as the positions of the known angles are reversed.

$$r_2 = \frac{r_q \sec A_m}{\sec A}.$$

$$\operatorname{Tan} A_x = \frac{r_2 \sin (A - A_m)}{r_p + r_q - \dfrac{r_2}{\sin (A - A_m)}}.$$

$$r_1 = \left\{ r_p + r_q - \frac{r_2}{\sec (A - A_m)} \right\} \sec A_x.$$

$$F_2 = F \tan A_m. \qquad F_3 = F_1 \tan (A + A_x).$$

As the tangential speeds of the points of contact of the two levers cannot be the same except at the pitch point, there must be sliding and consequently frictional loss between the working surfaces and the length of slide is equal to the difference in length of these above and below the pitch line, which is again equal to the length KL measured along the curved surface. The speed of sliding is greatest at the points C and D, and is zero at P, at which point the efficiency is 100 per cent. In diagrams such as Figs. 248, 249, and 250 the effect of friction is the same as though the angle of incidence were increased by an amount equal to the angle of repose of the metal during the approach, and diminished by an equal amount during the recess. Taking an angle of repose of 5°, the Fig. 251 shows the alteration in the values of the forces when the levers are in the same position as in Fig. 249. The tangential force on the driven lever is, of course, still 15·53 lbs., but as the angle of incidence is now as given by the line MN, the radial thrust is 10·39 lbs., and the resultant 18·64 lbs. On the driving lever the radial force is 4·824 lbs., and the tangential 18·07 lbs., the efficiency being, therefore, 96·65 per cent.

The formulæ for the various forces which are affected by friction are now—

$$F_1 = \frac{F \sec (A + A_x + \varphi)}{\sec (A_m + \varphi)}.$$

$$F_2 = F \tan (A + A_x + \varphi).$$

$$F_3 = F_1 \tan (A_m + \varphi).$$

$$F_4 = F \sec (A + A_x + \varphi).$$

The above are for the period of approach.

In Fig. 252 the levers are in the same position as in Fig. 250, and a comparison of these diagrams shows the effect of friction during the period of recess.

The formulæ for this period are—

$$F_1 = \frac{F \sec (A_m - \varphi)}{\sec (A + A_x - \varphi)}.$$

$$F_2 = F \tan (A_m - \varphi).$$

$$F_3 = F_1 \tan (A + A_x - \varphi).$$

$$F_4 = F \sec (A_m - \varphi).$$

If we wish to ascertain the efficiency only, we can do this by a

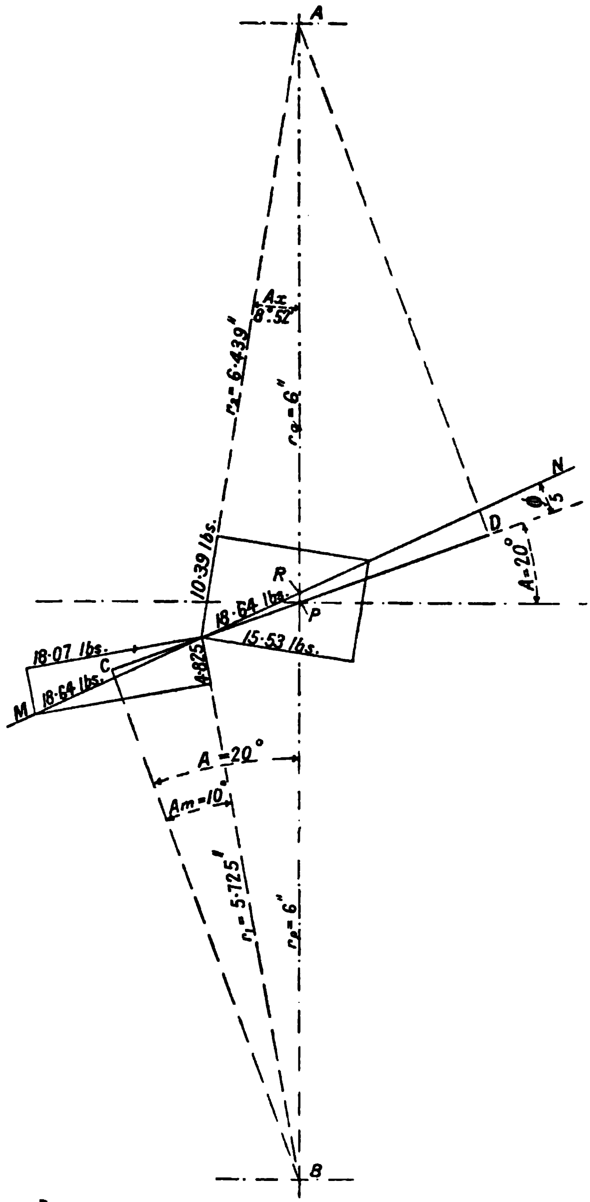


Fig. 251.

simple method, which does not necessitate the calculation of the various forces in detail.

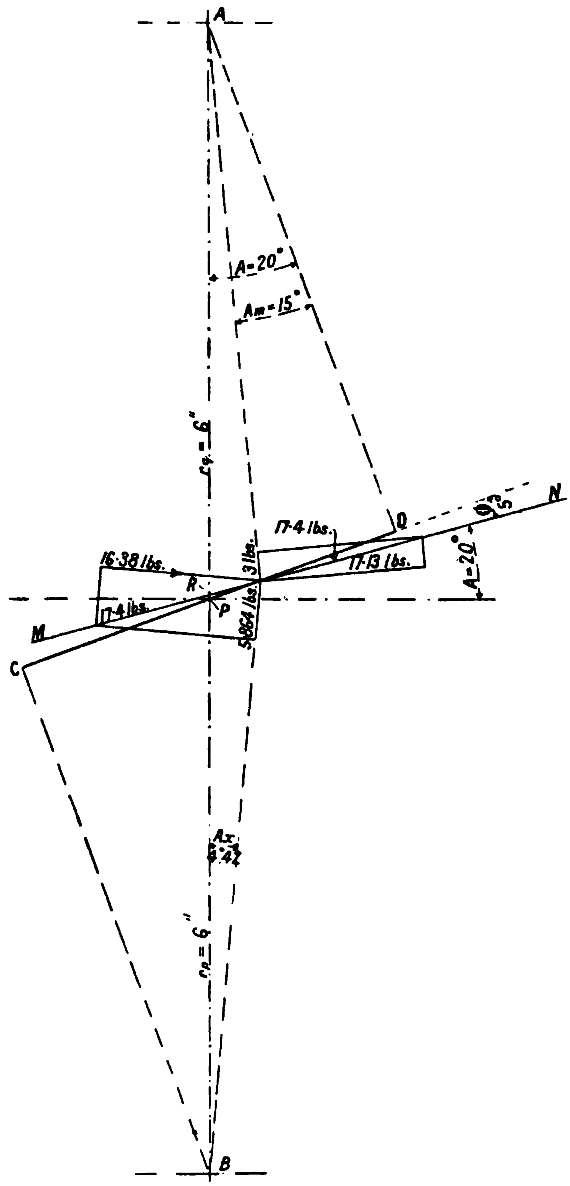


Fig. 252.

Referring to Figs. 251 and 252, the line MN, the inclination of which represents the angle of incidence as modified by friction, cuts the line AB at a point R above P.

The effect of friction in increasing the torque of the driving lever at any position is the same as though, at that position, the radii of the pitch circles were AR instead of AP, and RB instead of PB.

Thus in Figs. 249 and 251 the torque T_1 of the driving lever neglecting friction is $17{\cdot}47 \times 5{\cdot}725 = 100$, and the torque T_2 allowing for friction is $18{\cdot}07 \times 5{\cdot}725 = 103{\cdot}5$. In Fig. 251 the length AR ($= r_3$) $= 5{\cdot}898$, and RB ($= r_4$) $= 6{\cdot}102$.

$$\text{Then } T_2 = \frac{T_1 r_4 r_q}{r_p r_3} = \frac{100 \times 6{\cdot}102 \times 6}{6 \times 5{\cdot}898} = 103{\cdot}5 \text{ and } \frac{100}{103{\cdot}5} = {\cdot}9665,$$

which coincides with the torque and efficiency found by the detail calculation of the forces.

As efficiency $= \frac{T_1}{T_2}$, the formula for efficiency becomes—

$$\text{Eff.} = \frac{T_1}{\frac{T_1 r_4 r_q}{r_p r_3}} = \frac{r_p r_3}{r_q r_4}.$$

The lengths r_3 and r_4 may be found diagrammatically, as in Figs. 251 and 252, or, more accurately, by calculation, thus—

For the approach—

$$r_3 = r_q - x_1 + x.$$

$$r_4 = r_p + x_1 - x.$$

$$x = r_p \sec A_m \sin (A - A_m) \sin A.$$

$$x_1 = \frac{r_p \sec A_m \sin (A - A_m) \tan (A + \varphi)}{\sec A}.$$

And for the recess—

$$r_3 = r_q + x_1 - x.$$

$$r_4 = r_p - x_1 + x.$$

$$x = r_q \sec A_m \sin (A - A_m) \sin A.$$

$$x_1 = \frac{r_q \sec A_m \sin (A - A_m) \tan (A - \varphi)}{\sec A}$$

This formula for the efficiency is equally applicable to internal or external gears.

It illustrates an interesting point—namely, that in external gears of a given pitch the lowest efficiency is obtained with wheels of equal size, the efficiency increasing as the difference in the diameter of the wheels increases, and reaching its highest value with a rack, while with internal gear the reverse is the case, the efficiency increasing as the wheels approach each other in size. If we refer to Figs. 258, 259, and 260, we see that the length of the line *KL*, which is a measure of the frictional loss, is greatest in Fig. 258 where the wheels are of equal size, less in Fig. 259, where one wheel is twice the diameter of the other, and less again in the case of the rack in Fig. 260, while in the internal gear (Fig. 261) it is still further diminished.

So far, we have only dealt with the efficiency at given points,

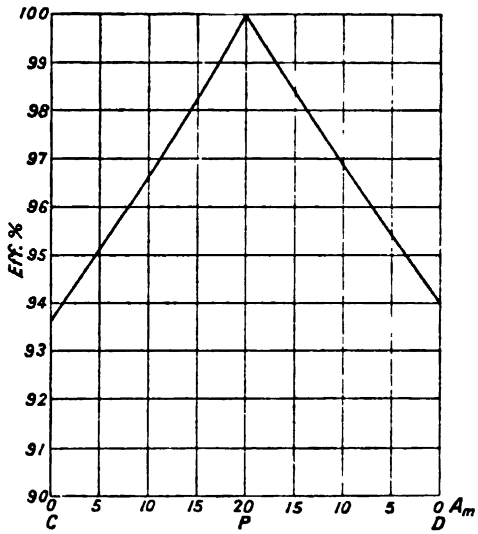


Fig. 253.

or the instantaneous efficiencies, but what we require to know in practice is the mean efficiency during the period from *C* to *D*. In order to ascertain this, the curve (Fig. 253) has been prepared. This shows the efficiencies of the two levers (Fig. 247) during the period from *C* to *D*, assuming an angle of repose as before of 5°, corresponding to a coefficient of friction of 0·087. This is a very high figure, and would only be obtained in practice under very bad conditions, but is adopted here in order to obtain substantial figures for purposes of illustration. The efficiency at point *C* is 93·6 per cent., and the mean efficiency during the approach is 96·7 per cent. At point *D* the efficiency is 94 per cent., and the mean during the recess is 96·9 per cent., the mean for the whole period being 96·8 per cent. It will be noted that during the recess the efficiency is a trifle higher than during the approach.

In actual work it is not necessary to calculate a number of instantaneous efficiencies, and then find their mean. Where the wheels are of equal size, the mean efficiency corresponds very closely with the instantaneous efficiency at the point in the approach at which the angle $A_m = \cdot 53a$, so that the one calculation is sufficient.

Where the wheels are of unequal size, it is necessary to calculate two values of the efficiency, one half-way along the arc of approach, and one half-way along the arc of recess. The mean of

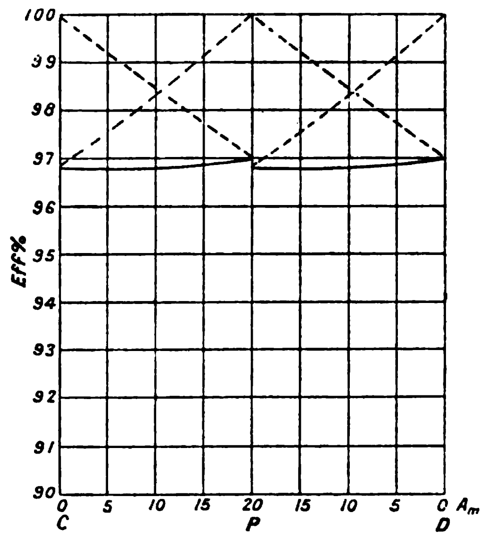


Fig. 254.

these values will be near enough to the true mean for all practical purposes.

In designing toothed wheels, we may adopt either of two methods. We may design the wheels specially in pairs which will work together with the best possible efficiency, but will not work with any other wheels, or we may design a universal range in which any wheel from the smallest size up to a rack will work with any other wheel of the range.

Special designs are best used for very large wheels which would be outside the limits of any ordinary range, and in cases where a very large number of similar sets of gear are required, as the cost of special cutters would then be justified.

To get the best results, it is desirable that the teeth should be in gear from the point C to the point D.

When this is the case, the arc of action is a little greater than twice the angle of incidence. If the load is to be taken on one tooth, as in the case of wheels having small numbers of teeth, the pitch should be equal to the arc of action, while in wheels having larger numbers of teeth, and in which the load can be taken on two teeth, the pitch should be equal to half the arc of action.

The exact relationship between the arc of action, the angle of incidence, and the pitch may be obtained by the method shown in Fig. 270. In this diagram, which is based on Fig. 247, the lines AD and BC are produced, and form the sides of rack teeth as shown shaded. It should be explained here that a rack being a wheel of infinite radius, the involute curve becomes a straight line at right angles to the line CD.

The distance $P_1 \ P_2$ is then the true length, measured on the straight, of the arc of action at the pitch line of the wheels, and is equal to the pitch, or to twice the pitch, according as the load is taken on one tooth or two teeth.

Taking always the smaller of a pair of wheels—

$$A_c \text{ in degrees } = \frac{360\, n_4}{n_1}.$$

In linear measurement—

$$A_e = P_1 P_2 = d \tan a = \frac{\pi d n_4}{n_1}.$$

Therefore,

$$\text{Tan } a = \frac{\pi n_4}{n_1}.*$$

In wheels having small numbers of teeth there is a sensible difference

* Referring to Figs. 224 and 225 it will be seen that in this gear the angle of incidence was 15 degrees, and the length of CD was limited by the height of teeth so as to bring the load on to two teeth. If calculated by the above formula the angle of incidence necessary to bring the load on to two teeth would have been

$$\text{Tan } a = \frac{2\pi}{56} = \cdot 1122 = \tan 6^\circ\ 24'.$$

This would give a slightly higher efficiency than the 15 degrees, but would necessitate special cutters.

between the length of the pitch and its chord. In laying out the teeth either with a rule or dividers, we lay out the chord, and we, therefore, require to know the true length of the chord in relation to the pitch.

The length of the chord $= d \sin \frac{180}{n}$.

The procedure then to be followed in designing special wheels is first to determine the pitch of teeth suitable for the load, find the number of teeth in the proposed wheel, and ascertain whether the load can be taken on two teeth or only on one. Then find the value of a, and proceed with the construction as in Fig. 247.

In Fig. 255 we have two wheels of equal size having 12 teeth. In this case the load comes on one tooth, and $\tan a = \frac{\pi}{12} = \cdot 2618 = \tan 14° \; 40'$, which is practically 14·5°, and may be taken as such.

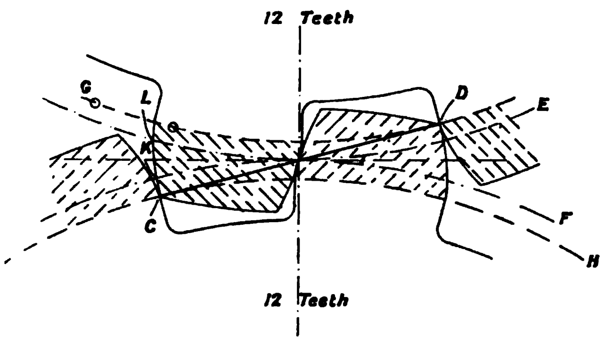


Fig. 255.

Taking $d = 12$ inches, the pitch $= 3 \cdot 14$ inches, and the chordal length $= 12 \times \sin \frac{180}{12} = 12 \times \cdot 2588 = 3 \cdot 105$ inches.

In Fig. 256 a wheel of 12 teeth gears with one of 25 teeth. In this case the line CD is not carried on to the true tangent point D', but is made the same length on both sides of the pitch point as for the smaller wheel. The reason is that if the line were carried on to D, and the teeth constructed accordingly, those of the smaller wheel would assume an undesirable (and, if there were considerable difference in the size of the wheels, an impossible) shape.

In Fig. 257 a wheel of 12 teeth gears into a rack. As already mentioned, a rack is equivalent to a wheel of infinite radius, and the involute then becomes a straight line at right angles to CD.

In Fig. 258 two wheels of 24 teeth gear together, the angle being again 14° 40′, which is again taken as 14·5°.

In Fig. 259 a 24 tooth-wheel gears into one having 49 teeth, and in Fig. 260 a 24 tooth-wheel gears into a rack.

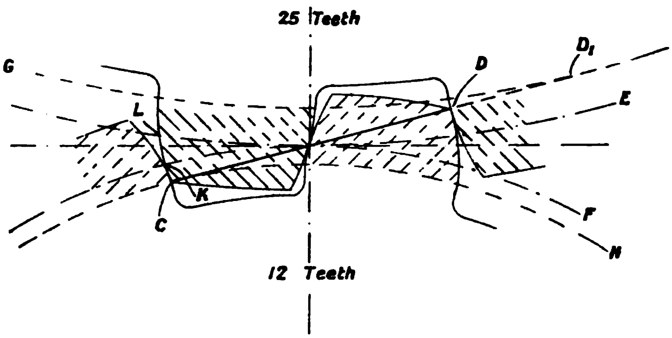


Fig. 256.

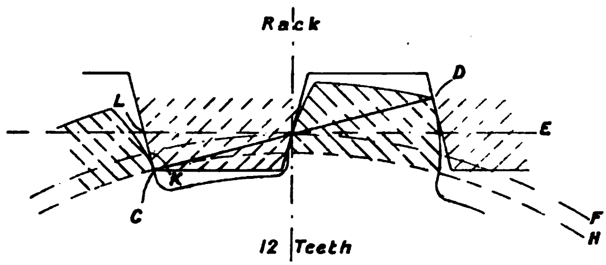


Fig. 257.

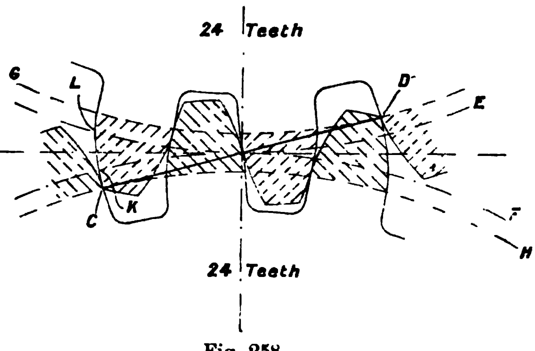


Fig. 258.

In Fig. 261 a wheel of 24 teeth gears into an internal wheel having 49 teeth. In this case the construction is as before, but it will be noted that the result is that the spaces of the internal wheel take the shape of teeth of an external wheel. In all the foregoing figures from 255 to 261 inclusive, the working part of the teeth is shown shaded, and it will be noted that in all cases its depth, measured from the top of the tooth, is the same. The unshaded portion is merely to provide a recess to clear the tops of the opposing teeth.

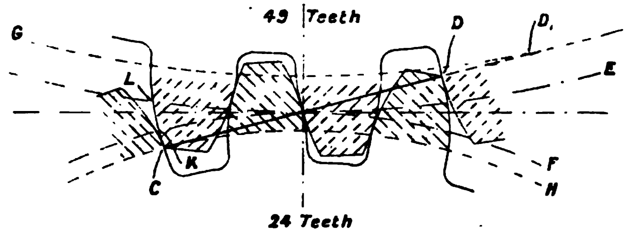


Fig. 259.

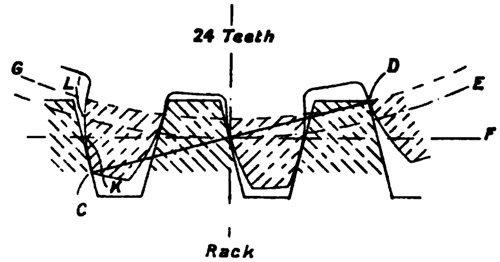


Fig. 260.

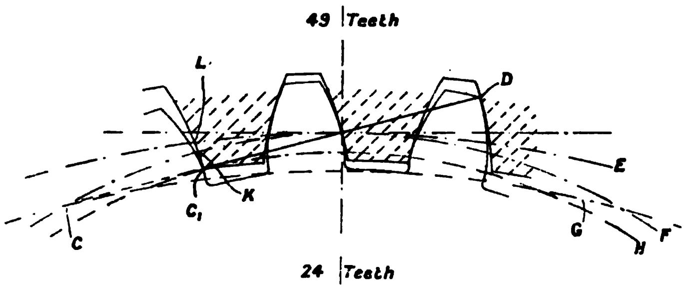


Fig. 261.

The formulæ which have been given relating to efficiency show that this increases as the angle of incidence is decreased, so that in special gear in which the angle is based on the number of teeth, the efficiency is in all cases the highest attainable. The arrangement of the two levers in Fig. 247 is analogous to that of a gear such as Fig. 255, in which the load is taken on one tooth, and the efficiency is as already shown in Fig. 253. If the levers were duplicated so as to produce an arrangement analogous to Fig. 258, in which the load is taken on two teeth, the efficiency would be as shown in Fig. 254.

In Fig. 253 there is no loss at P, and the maximum loss, which takes place at C and D, considerably exceeds the mean.

In Fig. 254, when one tooth is at C, and transmitting half the load with maximum loss, the other tooth is at P, and transmitting the other half of the load with no loss, so that the maximum loss is halved as compared with Fig. 253, but the mean loss throughout the period is the same in both cases. Thus, there is no gain in efficiency by taking the load on two teeth instead of one, but there is less fluctuation in load on the driver, and more regular wear on the teeth.

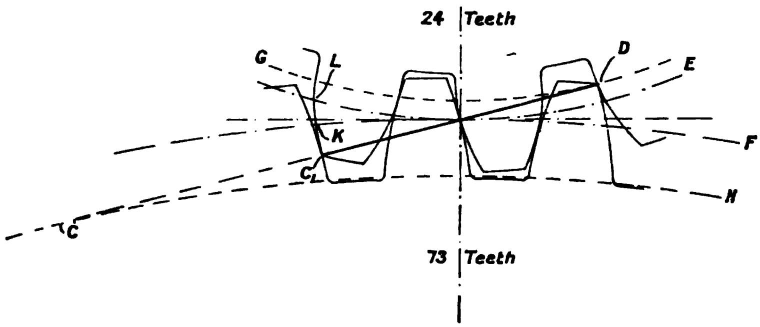


Fig. 262.

In the designs given so far, the thickness of tooth at the pitch line has been equal to half the pitch, no clearance being allowed nowadays for cut gear. This causes the thickness of tooth at the root to be less in the smaller of a pair of wheels than in the larger, and, as the strength of the gear as a whole depends upon the weaker wheel, the pitch, and consequently the angle of incidence, have to be larger than would be necessary if the teeth in both wheels were of the same strength.

In Fig. 262 an example is given in which the teeth are of equal thickness at the roots, so that the gear is stronger than if the teeth were of equal thickness at the pitch line.

In all the preceding designs the length of line CD has been such that the angle CBD or DAC (which ever corresponds to the smaller wheel) is equal to the angle of incidence of the teeth, so that the period during which uniform angular velocity may be obtained is utilised as far as practicable.

By cutting down the tops of the teeth so as to halve the contact periods *CP* and *PD*, the efficiency may be materially increased. Thus, if the working surfaces in Fig. 247 be cut shorter, so that contact begins and ends when a_m equals 10°, the mean efficiency will be raised from 96·8 to 98·3 per cent.

In Fig. 263 the teeth of the wheels shown in Fig. 258 have been cut down so as to halve the period of contact. The load is now

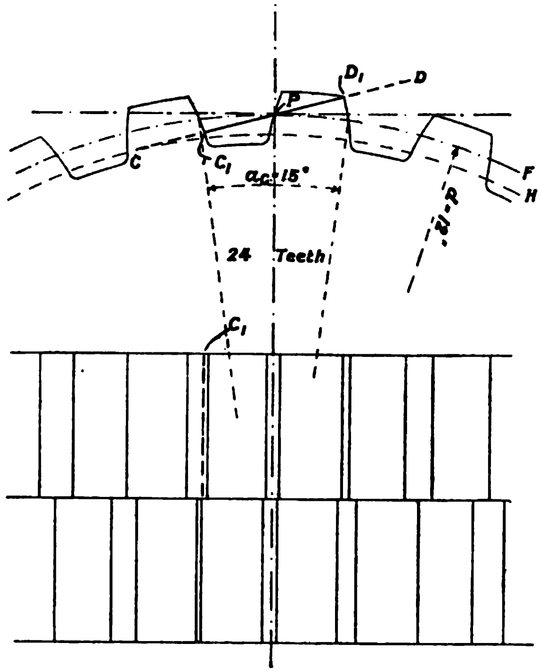


Fig. 263.

taken on one tooth, but as the pitch does not exceed the arc of contact the motion will still be transmitted quite steadily. If, however, we cut the wheel into two wheels of half the width and bolt the two halves together with the teeth of one facing the spaces of the other, we get the load taken on two teeth, and the arrangement becomes for all practical purposes similar to Fig. 258, except that the frictional losses are reduced by about 48 per cent. Instead of stepping the teeth in this way, we may set the teeth at an angle on the face of the wheel, so producing helical teeth. In Fig. 264 the gear in Fig. 263 is shown arranged as a helical gear. This would have the same efficiency as Fig. 263, and as the working surfaces come into action obliquely, it is claimed that they work more smoothly and with less noise than ordinary gear. Helical gear is not usually made with teeth of the shape shown in Fig. 263, the more usual form being that shown in Fig. 265. In this case there is no gain in efficiency over the ordinary gear (Fig. 258), the only advantage being the smoothness of working.

In ordinary gear the contact between two teeth is on a line

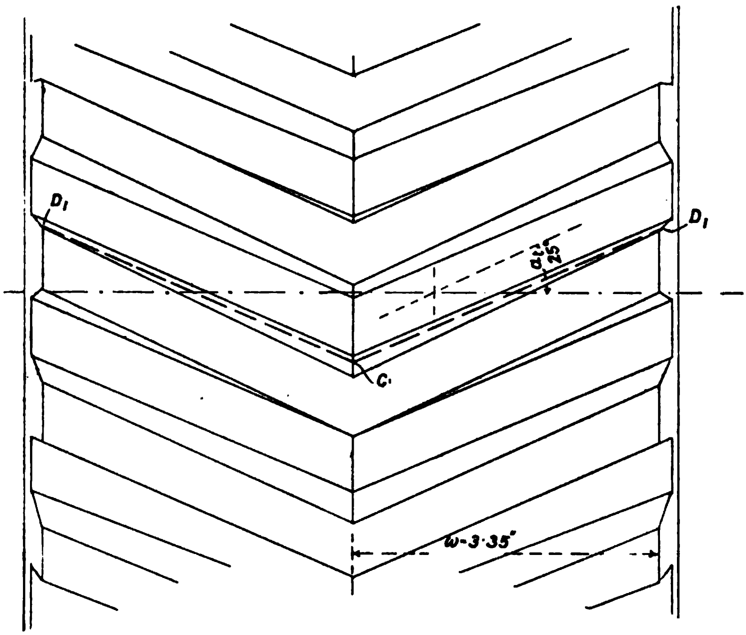


Fig. 264.

straight across the face of the teeth, and equal to their width (see Fig. 263, C_1).*

In helical gear the contact is on a line which lies obliquely across the face of the teeth, as shown dotted in Figs. 264 and 265. The length of the line of contact where the load is taken on one tooth will be equal to the width across one tooth, and where the

* In order to make a distinction the line *CD* may be described as the *Path of Contact* of the teeth, while the line along which the teeth bear upon each other across their faces may be described as the *Line of Contact*.

load is taken on two teeth will be equal to the width across two teeth, but the maximum width of any one tooth, taken parallel to the axis of the wheel, which is in action at any time, cannot exceed

$$\omega = \frac{\pi d a_c}{360 \tan a_t}.$$

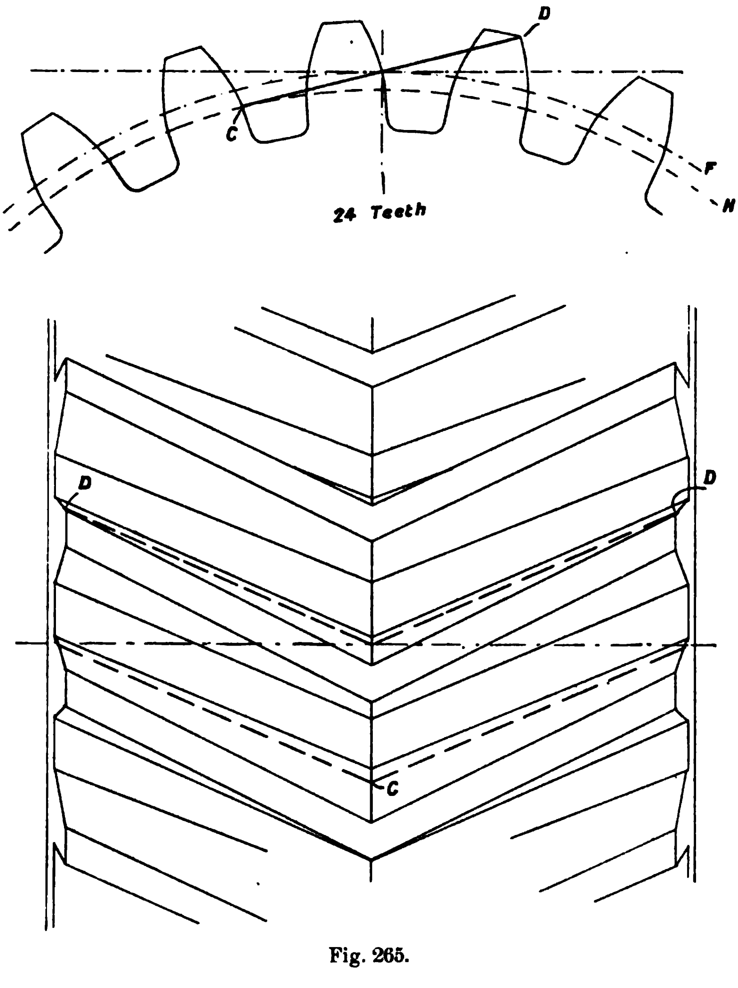


Fig. 265.

in which ω = width of tooth, d = diameter of pitch circle, a_c = arc of action, and a_t = angle of tooth across face of wheel.

In order to neutralise end thrust due to the obliquity of the teeth, it is usual to have two sets of teeth, set at opposite angles, and the smaller wheel of the pair is mounted free to slide on a feather key, so that it will set itself correctly with respect to the other wheel.

In a universal range of wheels any pair of which shall be capable of working together, the pitch, length of tooth above and below the pitch line, and the angle of incidence must be the same for all the wheels in the range.

It has already been shown that to get the best results the angle of incidence should vary with the number of teeth in the smaller wheel of a pair. In a universal range this is not possible, as the angle must be suitable for the smallest wheel in the range—that is to say, it must be such that in the smallest wheel the arc of action is not less than the pitch. The proportions most commonly adopted are those originated by Messrs. Brown & Sharpe.* In these the smallest wheel of a set has 12 teeth, the angle of incidence is 14·5°, the height of tooth above the pitch line is 0·318 of the pitch, and the depth below is 0·368 of the pitch.

Fig. 266 shows a pair of 12-tooth wheels working together, the shape of the teeth being that due to Messrs. Brown & Sharpe, and adopted by most engineers.

In these the angle of incidence and arc of action correspond with those of the 12-tooth wheels in Fig. 255. A comparison of these figures shows that the shaded working, and, therefore, really useful, portion of the teeth is identical in both cases, but in Fig. 266 there is a useless portion at the top of the tooth, and to clear this the depth of tooth below the pitch line has to be much greater than in Fig. 255, so that an altogether longer and weaker tooth is obtained. It will be noted that the surfaces of the teeth are cleared back slightly below the base line of the involute, so that there is no contact between them and the unshaded part at the tops of the opposing teeth. Very slight wear, however, brings them into contact, causing irregular driving and noise, especially at high speeds. In wheels having from 12 teeth up to 29 teeth, the load is taken on one tooth during some portion of the arc of action, so that wheels of 29 teeth or less must be of sufficient strength to take the load on one tooth.

In wheels having 30 teeth or more the load is always taken on at least two teeth, and their strength may be calculated on this basis.

* See *Practical Treatise on Gearing*, Brown & Sharpe.

This statement is apparently contradicted by Fig. 258, in which, with an angle of incidence of 14·5°, we get the load taken on two teeth when there are only 24 teeth in the wheels. The reason for the discrepancy is that in Fig. 258 the tops of the teeth correspond with the points C and D, the height of tooth above pitch line being

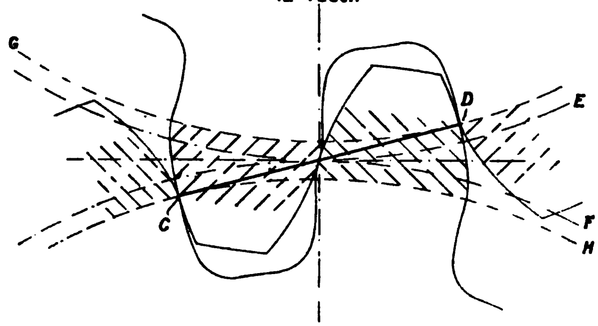


Fig. 266.

0·364 of the pitch, while in Brown & Sharpe's gears the height is only 0·318 of the pitch, and, as we have already seen, cutting down the teeth shortens the arc of action (see Fig. 275).

In Fig. 267 a 12-tooth wheel gears with a 30-tooth. In this

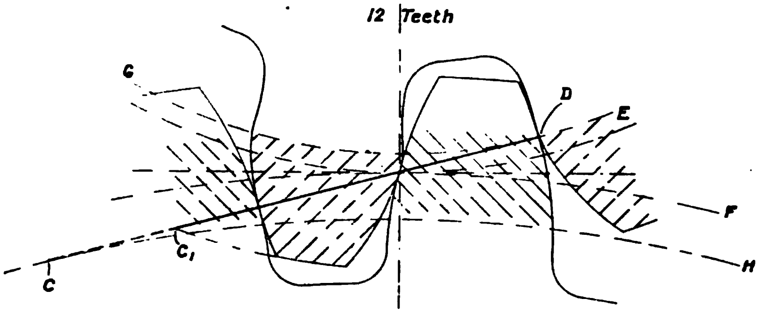


Fig. 267.

case the whole of the involute surface of the smaller wheel is utilised, but in the larger wheel a considerable portion of the upper part of the tooth is useless.

Fig. 268 shows a 12-tooth wheel gearing into a rack, and here again the upper parts of the rack teeth are not utilised. If the rack teeth were carried up straight to the top, as in Figs. 257 and 260, the edges would cut into the teeth of the wheel. To avoid this, the rack teeth are rounded off above the shaded part, the shape of the rounding being quite immaterial so long as it gives the necessary clearance. Thus, in a universal range of this type the larger wheel of a pair always has a useless portion at the top of its teeth, the size of this useless portion increasing with the difference in diameter of the pair. Where the difference is great, and in the case of racks, the teeth of the larger wheel or rack will cut into the teeth of the smaller wheel unless rounded away, and as the amount to be rounded off is variable, depending on the number of teeth in the opposing wheel, it cannot be done in the wheel-cutting machine in the first instance, but requires to be done specially afterwards.

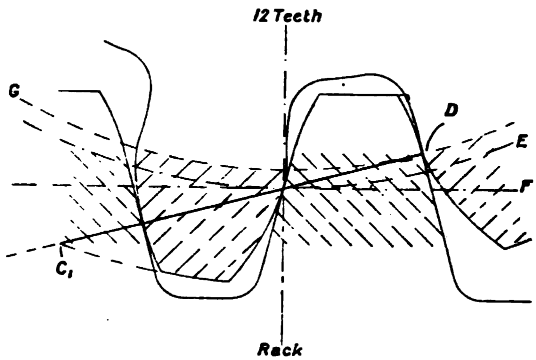


Fig. 268.

It has already been stated that, although the universal gears will run silently when new, as soon as they have worn a little the blank upper portions of the teeth come into contact with those portions of the opposing teeth which are below the base line of the involute, so causing irregular driving and noise, especially at high speeds. To prevent this, the teeth may be cut away below the base line, as shown in Fig. 269, the amount cut away being equal to the amount of wear allowable. The teeth will then run silently till the protruding portion is worn down, when they should be renewed. As the part cut away always commences at the involute base line and extends downwards, it is invariable, and can be done by machine in the first instance. Where this is done, no rounding off of the upper portions of teeth is necessary.

The points mentioned above are not special to Messrs. Brown & Sharpe's design of wheel, but are the necessary accompaniment of any attempt to design a universal range of wheels, and are due to the fact that the design must be a compromise.

Against these disadvantages universal wheels have the advantage that they can be cut in quantities and kept in the stores, and passed out to any job irrespective of the particular wheel with which they are to gear, so possessing the same advantages as standard screw threads.

Helical wheels can be designed in universal ranges, provided the angle a_t be made identical throughout the range.

The effect of wear upon the shape of the teeth depends upon whether the load is taken on one or two teeth during the arc of

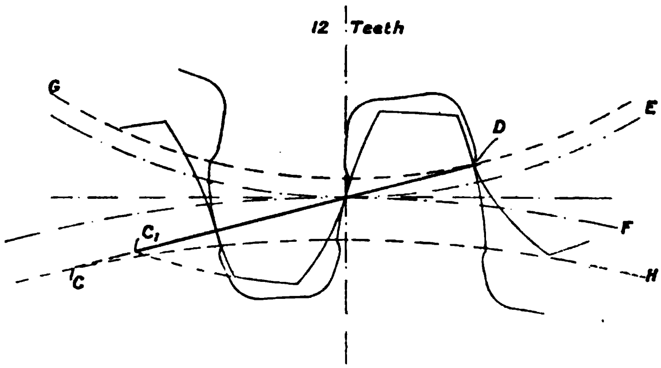


Fig. 269.

action. Referring to Fig. 247, which is typical of a one-tooth gear, it will be seen that in passing from C to P the face, or portion from pitch line to top of tooth, of A slides over the flank, or portion from pitch line to base line, of B, while in passing from P to D the face of B slides upon the flank of A. As the same amount will be worn off each, it follows that the metal will be worn away to a greater depth on the flanks than on the faces, and the involute shape will be destroyed. The result of this will be that during the period from C to P the pitch line speed of the driven wheel will be greater than that of the driver, while from P to D it will be less, the mean speed from C to D being, of course, the same as that of the driver.

Where the load is taken on two teeth throughout the period of

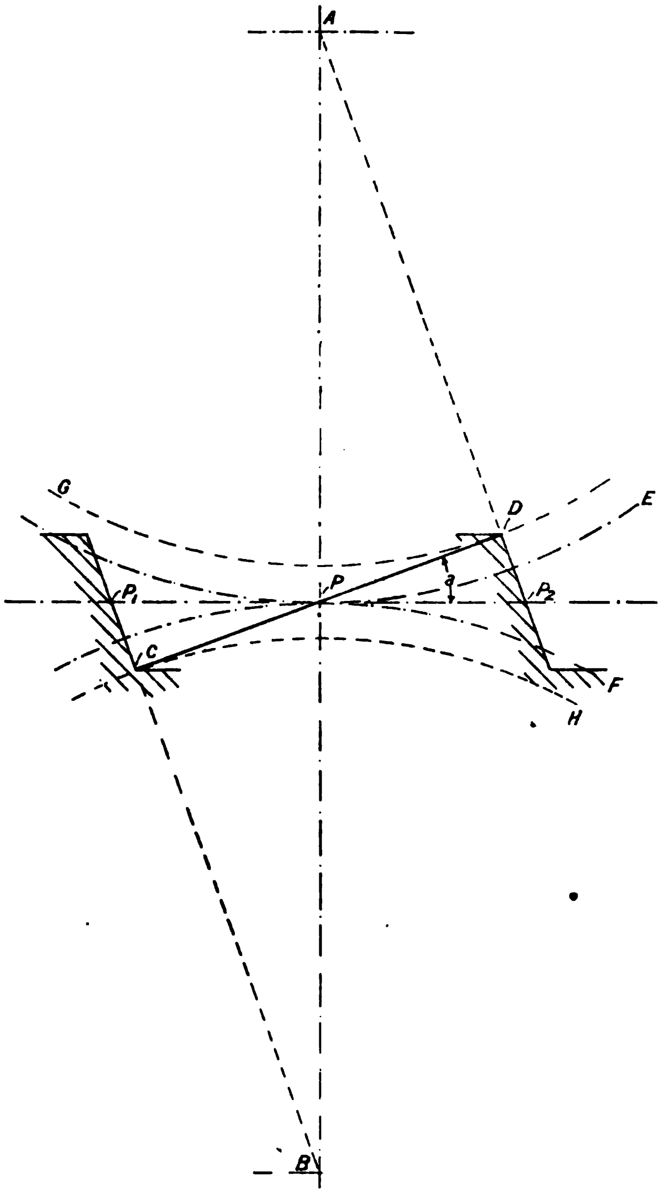


Fig. 270.

action this destruction of the involute curve does not take place. As contact between one pair of teeth takes place at a point where the rate of sliding is different to that at the point of contact of the other pair of teeth there is a tendency for the one pair of teeth to correct the wear of the other pair, so that if they are the correct shape when new, they will tend to preserve that shape. In cases where the number of teeth on which the load is taken varies during the arc of action the teeth will wear to some irregular shape.

The permissible limits of wear depend upon two considerations, (1) the amount of noise and backlash allowable, and (2) the weakening of the teeth.

Generally speaking, for the better classes of work the permissible limits of wear will be reached before any weakening of the teeth has taken place.

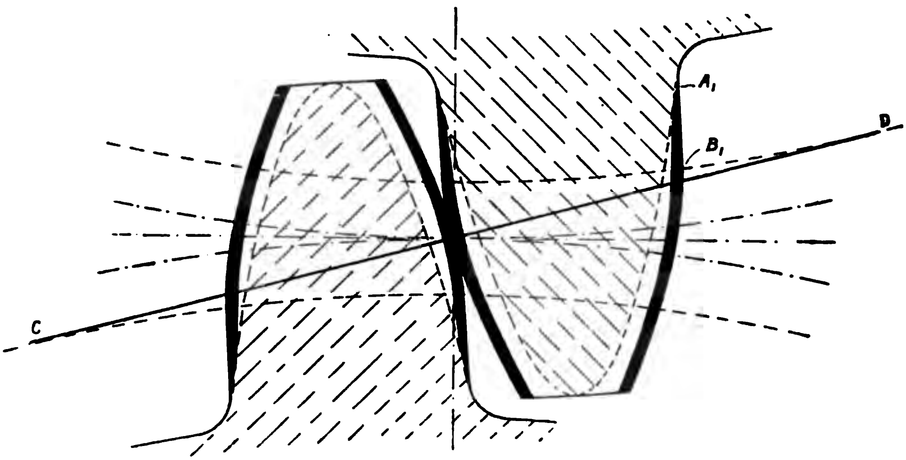


Fig. 271.

In Fig. 271 a pair of teeth are shown which form part of a gear on which the load is taken on two teeth, and in which the wear is, therefore, uniform. It is a reversing gear, so that wear takes place on both sides of the teeth. The dotted and shaded portion shows the shape of the tooth for uniform strength from the root upwards, and indicates the amount of wear which could take place before its strength was impaired. The black shaded portion shows the amount which can be worn off before encroaching upon any part of the dotted portion, and, therefore, before weakening of the tooth commences. In the case of the teeth shown, the amount is 10 per cent. of the original thickness of the teeth at the pitch line.

In Fig. 272 the same teeth are shown, but driving in one direction only, so that only one side of the teeth is worn. In this case the amount of wear is about 8 per cent. of the original thickness of the teeth at the pitch line.

The black shaded portion A_1B_1 below the base line of the involute is not properly speaking worn away, but is cut away by the top of the opposing tooth, and might as well be cut away when the teeth are being cut in the first instance, as shown in Fig. 269.

In calculating the strength of gearing, the teeth should be treated as cantilevers of rectangular section, the formula for which is—

$$W = \frac{fwh^2}{6\,l},$$

in which W = working load in lbs. at the top of the tooth, f = the safe working stress in lbs. per square inch at the weakest part of the tooth, which is a little below the commencement of the fillet at the line marked h in Fig. 273, and w, h, and l are dimensions in inches at the points shown in the figure.

In wheels working steadily at very slow speeds, the working load is the force corresponding to the H.P. transmitted, and is, therefore, easily found by the formula—

$$W = \frac{63{,}025\ H}{S_2 r_1 n_4},$$

H being the horse-power, S_2 = the speed in revolutions per minute, and r_1 = the radius to the extreme edge of tooth. In such cases f is taken as the usual working stress allowable in cantilever beams of the material of which the wheel is made.

With increase of speed, vibration and shocks are set up which introduce additional forces to that due to the power transmitted. These forces cannot be determined, and so have to be allowed for by an empirical decrease in the value of f as the speed increases. Further, with increase of speed, the pressure per inch width of tooth should be decreased in order to avoid excessive wear, and this requirement can be met by a further decrease in the value of f.

The curve (Fig. 274) gives values of f which make allowance for both conditions. This is not a theoretical curve, and probably does not agree with any definite formula. It was found by taking particulars of a number of gears which were working satisfactorily,

plotting the values of *f*, and taking the average. A considerable number of values of cast-iron gears were obtained. It was only possible to obtain a comparatively small number of values for steel,

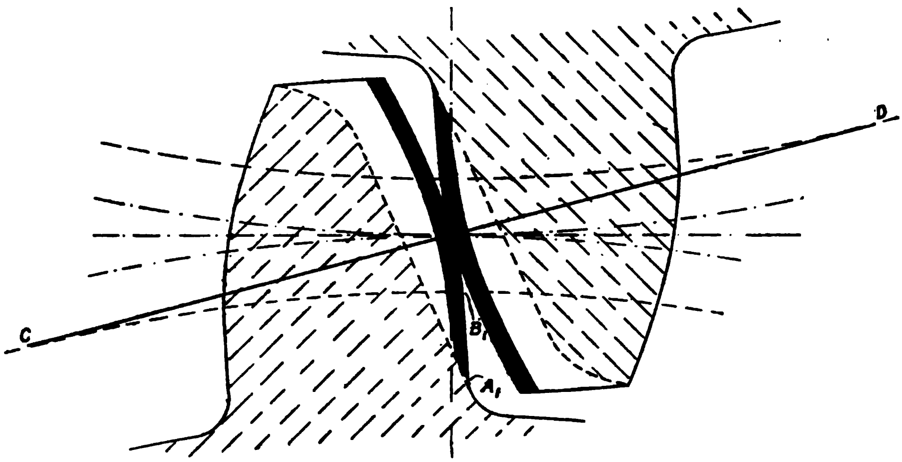


Fig. 272.

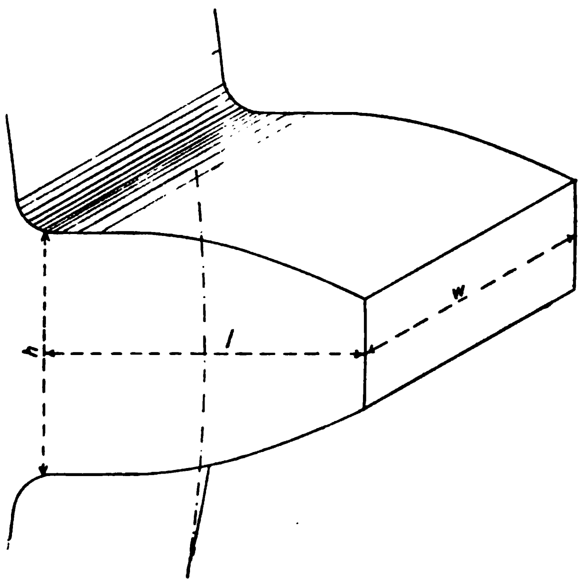


Fig. 273.

but these appeared to show that in general practice the value of *f* for steel is taken at twice that for cast iron.

When a wheel has been designed the strength may be checked by the above formulæ, but when commencing to design an approximate formula is necessary for the purpose of obtaining dimensions to start from.

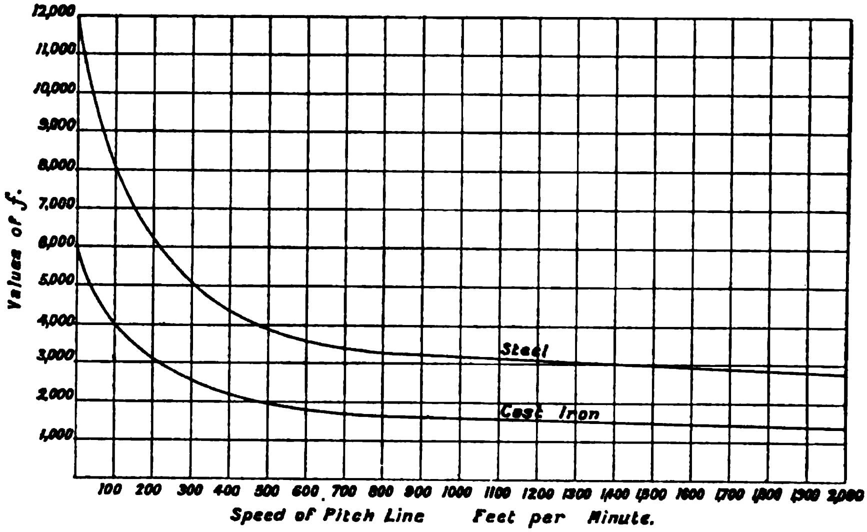


Fig. 274.

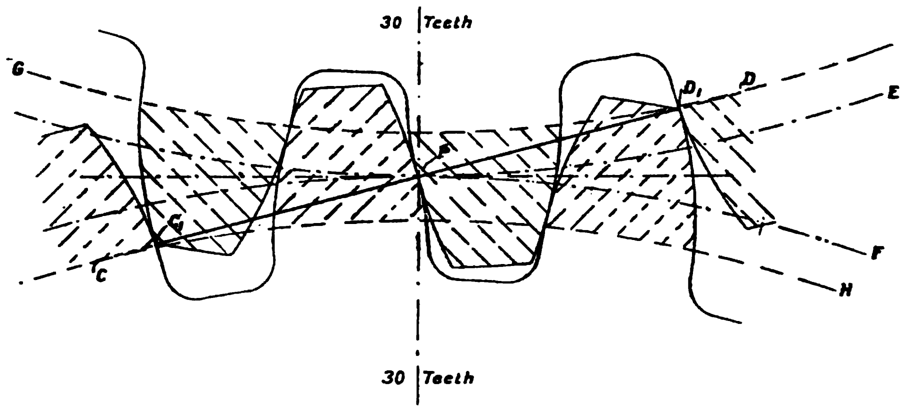


Fig. 275.

A convenient formula is—

$$p = \sqrt{\frac{KH}{S_2 d f x_1}},$$

in which p = the pitch in inches, K = a constant, which includes the number 63,025, the number of teeth on which the load is taken, and the relation of h and l to the pitch, and $x_1 = \frac{w}{p}$.

Values of K are given in Table XII.

TABLE XII.

No.	Description.	K
1	For 12-tooth wheels of the proportions shown in Fig. 255, in which load is taken on one tooth, ... Approximately correct up to 23 teeth.	1,160,000
2	For 24-tooth wheels of the proportions shown in Fig. 258, in which load is taken on two teeth, ... Approximately correct for wheels having more than 24 teeth.	945,000
3	For 12-tooth wheels of Browne & Sharpe's proportions, as shown in Fig. 266, in which load is taken on one tooth, Approximately correct up to 29 teeth.	2,110,000
4	For 30-tooth wheels of Browne & Sharpe's proportions, as shown in Fig. 270, in which load is taken on two teeth, Approximately correct for wheels having more than 30 teeth.	762,000

In calculating the strength of a pair of wheels, the smaller wheel is, of course, taken.

The ratio of the width of tooth to the pitch depends upon the pressure allowable per inch of width, and upon the accuracy and rigidity with which the gear can be mounted.

It has already been shown that wheels in which the load is taken on two teeth tend to retain their correct form when worn, so that it is advisable that the smaller wheel of a pair should have a sufficiently large number of teeth to take the load on two. Thus, if using Brown & Sharpe's proportions the smaller wheel of a pair should have not less than 30 teeth, or 24 teeth if of the proportions shown in Fig. 258. This, however, may in some cases lead to wheels of such a width in relation to the pitch that they will not work satisfactorily unless very accurately mounted and supported by a bearing on each side. In many cases the necessities of design compel the use of overhanging wheels, and in these the width should not exceed three times the pitch.

With regard to the allowable pressure per inch of width, this does not appear to have been fully investigated. For cast-iron wheels at ordinary speeds a pressure of about 600 lbs. has been given as the maximum allowable.* For high speeds the pressure should be less, and for slow speeds it may be greater, but the correct

* See *Machine Design*, W. C. Unwin.

ratio between speed and pressure has not been determined. It has been suggested that speed × pressure should be a constant and modifications of this suggestion have also been offered, but none of them seem applicable in practice.

The pressure on steel may be taken at double that on cast iron. In calculating the dimensions of teeth by the formulæ given, in conjunction with the curve (Fig. 274), if we take the relation x_1 of the width to the pitch at 3 we shall obtain proportions of teeth in which the linear pressure is reasonable, except in the extreme case of very large pitches, in which the value of x_1 should be increased. It will be noted, as previously mentioned, that for a given pitch and width the shape of the curve gives a diminishing value of the linear pressure as the speed increases. Teeth calculated by the foregoing formulæ will allow 10 per cent. of wear if reversing, or 8 per cent. if running in one direction, without impairment of strength. If a greater amount of wear is required, the extra amount should be added to the thickness of the teeth, as found by the formulæ.

An interesting example of high values is provided by a pair of wheels designed by Mr. J. Christie.* These wheels are 37·6 and 56·4 inches diameter respectively, the pitch is 4·92 inches, and width 24 inches. The wheels work with varying speeds and powers, the maximum figures being 260 revolutions per minute of the smaller wheel, 3,300 H.P., and a tangential force of 2,100 lbs. per inch of width. The stress at the roots of the teeth is about 2,300 lbs. per square inch.

The smaller wheel is of fluid compressed forged steel containing Carbon 0·86 per cent., Manganese 0·51 per cent., Silicon 0·27 per cent., Phosphorus and Sulphur both less than 0·03 per cent. The larger wheel is an annealed steel casting containing Carbon 0·47 per cent., Manganese 0·66 per cent., Phosphorus and Sulphur both less than 0·05 per cent. These wheels are stated to have run satisfactorily for several years.

In helical gear, as the contact lies on an oblique line across the face of the teeth, these are never loaded at the top right across the whole width as in straight gears. Thus, if we refer to Fig. 265, we see that the load on the leading tooth is distributed between the top of the tooth and the pitch line.

If the load were uniformly distributed over the whole surface between these limits the tooth would be capable of carrying about

* See *Engineering News*, New York, Feb. 28, 1901.

50 per cent. more load than a straight tooth, but as the load lies obliquely across the tooth the stress is greater on one side than the other, and its carrying capacity is, therefore, only about 20 to 25 per cent. greater than that of an ordinary tooth.

In designing the teeth of bevel wheels the same principles are followed as in straight wheels, the curves of the teeth being involutes, and the number of teeth being sufficient wherever possible to allow of the load being taken on two teeth. In determining the strength of the teeth, the dimensions at the smaller side should be taken, and the curve (Fig. 274) and its accompanying formulæ may be used.

In the cutting of gear teeth three methods are at present in use.

In the first the shape of tooth and space is laid out as accurately as possible by a draughtsman, and a template is made of the shape of the space. A milling cutter is then made to the template, and the gear is cut with this cutter in a wheel-cutting machine having dividing gear. This system was originated by Messrs. Sharp & Roberts, of Manchester, in the early part of last century, and some of their machines are still in use. For a given pitch it is necessary for absolute accuracy to have a separate cutter for each different number of teeth, but Messrs. Brown & Sharp state that in ranges of wheels from 12 teeth up to a rack it is sufficient for practical purposes to have eight cutters for each pitch.

In another system, the Fellowes, a single cutter only is required for each range of wheels. This cutter is exactly similar to the smallest wheel of the range, and cuts by a rectilinear motion similar to that of a shaping machine, the cutter and wheel being rotated at their correct relative speeds as the cutting takes place. No drawings or templates are required for making the cutters, the true involute form of which is generated by rolling the cutter past an emery wheel in true rack and pinion motion, the cutter representing the pinion, and the emery wheel one side of one tooth of the rack. This system is more accurate than the first, and is applicable to internal as well as external gears.

In a third method the previous system is reversed, a single cutter being used for each range, but being a replica of the rack instead of the smallest wheel of the set. The cutter is in the form of a hob, and it and the wheel revolve together at their correct relative speeds, the true involute form being originated on each tooth of the wheel as it is cut. The system is equally as accurate as the previous one, but is not applicable to internal gears.

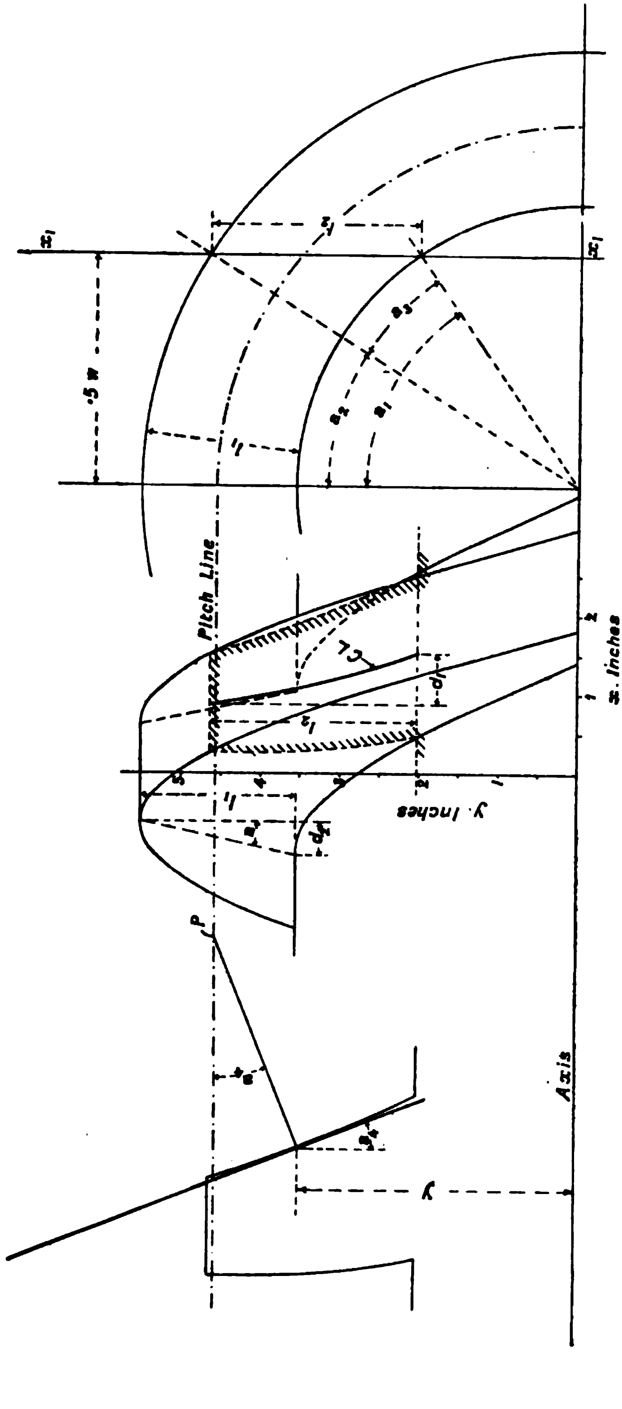


Fig. 276.

In cutting helical gears the cutter is set at the correct angle, and the relative movement of the cutter and wheel is such that an accurate spiral is traced across the face of the wheel.

Bevel wheels can only be accurately cut by reciprocating cutters, which trace the correct involute, while at the same time giving the necessary taper to the teeth.*

Worm Gear.—Taking a rack and wheel such as those shown in Fig. 260, if the rack be moved along slowly the wheel will revolve slowly, and the motion will be silent owing to the slow speed. The extent of the motion will, however, be limited by the length of the rack. If, now, we substitute for the rack a worm having threads the same shape in section as the rack teeth, we shall obtain the equivalent of a rack of infinite length, and on rotating the worm at a suitable speed the motion will be the same as that of the rack and wheel, except that it will be continuous.

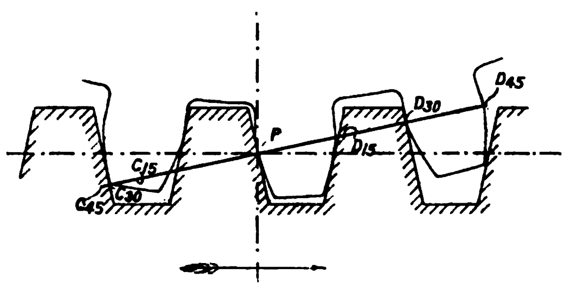


Fig. 277.

The threads of the worm are straight-sided, so that on their central section their shape is similar to that of a rack (see Figs. 276 and 277. At all other planes of section the shape of the thread is distorted, the faces becoming curved and their inclination being increased on one side and lessened on the other. The distortion becomes greater as the distance from the central plane increases. Thus, in Fig. 276 the section of the thread on plane X_1X_1 is shown shaded, and in Figs. 278 and 279 the sections on planes 1-1 and 2-2 (of Figs. 282, etc.) are shown.

The wheel teeth, on the central section, are of involute form, and on other planes of section they are conjugates of the worm threads.

In designing worm gear we should endeavour to obtain as long

* See paper by J. H. Gibson, *Engineering*, March 26, 1897.

a line of contact as possible, and to avoid as far as possible any variation in its length. Any such variation causes the pressure per linear inch on the teeth to vary as they pass through the arc of action, so tending to cause irregular wear of the thread and tooth faces, and unsteady driving.

In order to obtain a long contact line, the width of wheel should be as great as practicable. The limit of useful width is reached at that plane of section at which one face of the worm thread becomes vertical at the pitch line. Practically we may take the width w (see Figs. 277, etc.), as being the distance between those planes of

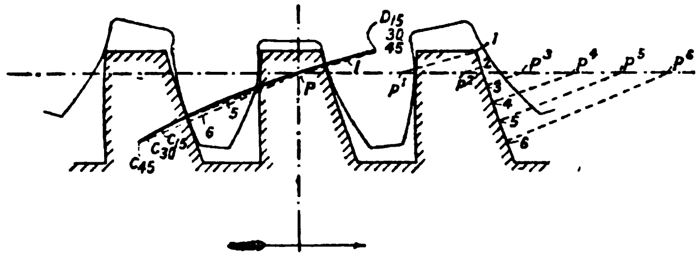


Fig. 278.

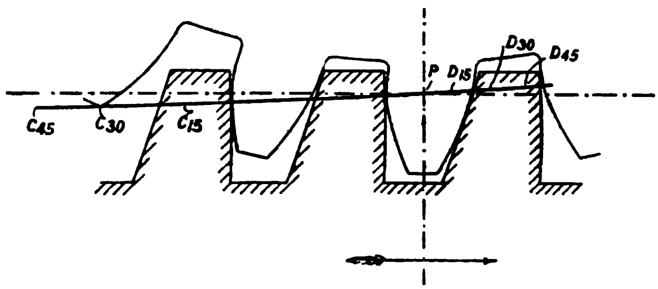


Fig. 279.

section at which the distortion of the centre line of the thread d_1 equals half the distance in thickness of the thread between top and bottom d_2 (see Fig. 276). In Figs. 278 and 279 $d_1 = d_2$, and it will be noted that one face of the thread is upright, but, owing to its curvature, it still has some inclination at the pitch line.

In Fig. 276 the section X_1X_1 is taken at a point at which $d_1 = 1{\cdot}5d_2$, in order to obtain more pronounced shapes for purposes of illustration.

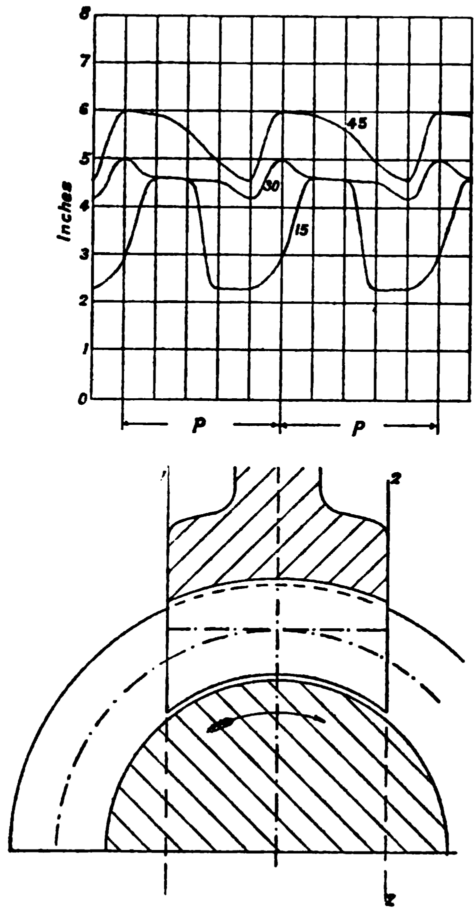
Limiting the value of w to that at which $d_1 = d_2$, w may be found as follows :—

$$a_3 = \frac{360\, l_1 \tan a}{p_1}.$$

$$\text{Tan } a_2 = \frac{r_6 \sin a_3}{r_5 - \dfrac{r_6}{\sec a_3}}.$$

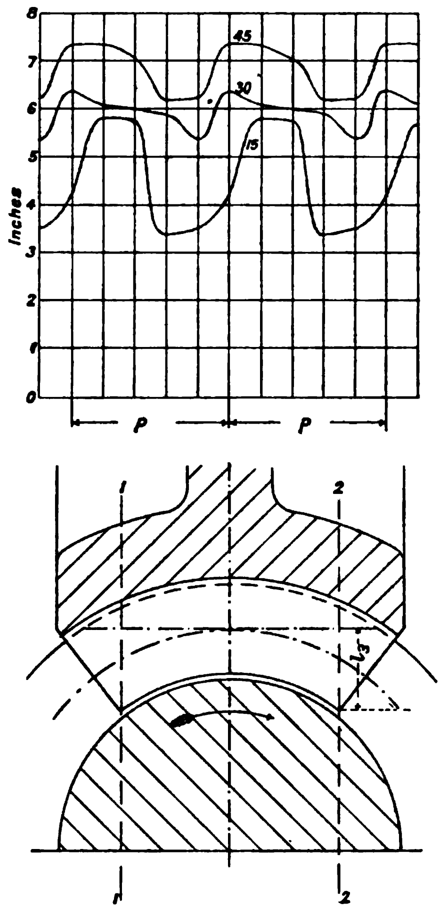
$$w = 2\, r_5 \sin a_2.$$

In cases where the lead of the worm is small in comparison with its diameter, the limit $d_1 = d_2$ cannot be reached.



Figs. 280 and 281.

Referring to Fig. 282, it will be seen that the height of wheel teeth above the pitch line l_3 increases with the distance from the central plane, a maximum being reached on planes 1-1 and 2-2. As this height increases the teeth become narrower at the top, in side view, the maximum possible height being reached when the



Figs. 282 and 283.

teeth become pointed. The extreme limiting value of w is then that at which the tops of the teeth become pointed. This varies with the number of teeth in the wheel, and merely as a general guide, Fig. 289 is given to show the relation in the case of the gear in Figs. 277, etc. To allow for wear, the actual value of w should be less than this, the metal at the top of the teeth being 0·1 to 0·15 of the pitch, instead of coming to a point. These limits of width are given as ordinary practical limits, they may be exceeded in special cases to meet extra heavy loads on the teeth, but it should be borne in mind that beyond the limits given only half the extra width of wheel is useful, this being on the side at which the worm threads enter the wheel.

In order to ascertain the length of contact line of any given wheel, we require first to determine the shape and length of the contact path *CD* at a number of planes of section. At the central plane (Fig. 277) the contact path is straight, as in the case of an ordinary rack and wheel, while at other planes it is curved and of different lengths. Thus, in Fig. 278, which shows the section on 1-1 (Fig. 282), at which the threads enter the wheel, the path is convex relatively to the pitch line, and is shorter than at the centre, while in Fig. 279, giving the section on 2-2, where the threads leave the wheel, the path is concave and longer than at the centre.

The construction of the shape of the threads at different planes of section is a matter of simple projection, while the method of obtaining the shape and length of the contact path is illustrated in Fig. 278. The lines 1, 2, 3, etc., are drawn normal to the surface of the thread at the points correspondingly numbered, and cut the pitch line at the points $p_1 p_2 p_3$, etc. These lines are then transferred to the pitch point *P*, and the line joining their extremities represents the contact path. The correct direction of these lines is most expeditiously found by calculation in the following manner.

Referring to Fig. 276, the line *CL* represents the centre line of the thread at the shaded section. This line forms a curve, the base lines of which are, vertically, the centre line of the thread at the central section, and, horizontally, the axis of the worm. x represents the advance, from the vertical base line, of any point on the centre line of the thread, y being the distance of that point from the axis of the worm. The higher value of x, which is at the root of the thread $= \frac{p_1 a_1}{360}$, and the lower value, at the top of the thread $= \frac{p_1 a_2}{360}$. Between these values the formula for the curve is—

$$y = \cdot 5\, w \,.\, \text{cotan}\, \frac{360\, x}{p_1}.$$

The dc of this $= \frac{w\pi}{p_1} \,.\, \text{cosec}^2\, \frac{360\, x}{p_1}$, and is the cotangent of the angle of inclination of the curve CL to the vertical at any point x. It is more convenient to have the tangent of this angle so inverting the formula—

$$x = \frac{p_1}{360} \cdot \cot^{-1} \frac{y}{\cdot 5\, w}.$$

The $dc = -\frac{1}{1 + \left(\frac{y}{\cdot 5\, w}\right)^2} \cdot \frac{p_1}{w\pi}$ = tangent of the angle of inclination of the curve CL to the vertical at any point y, and has the advantage of not requiring reference to a table.

The angle of inclination of either of the thread faces at any point y is then found, thus—

$$\operatorname{Tan} a_4 = \frac{l_1 \tan a}{l_2} \pm \frac{1}{1 + \left(\frac{y}{\cdot 5\, w}\right)^2} \cdot \frac{p_1}{w\pi},$$

w in this case being the width between the planes of section at which the thread is being taken.

TABLE XIII.

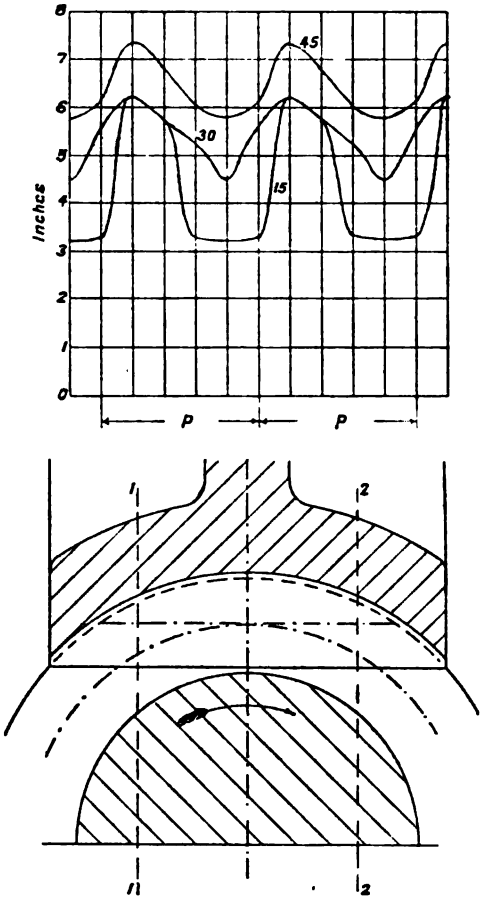
Wheel Section.	Curve.	No. of Teeth.	Mean Length of Contact Line.	Arc of Action / Pitch
Fig. 281,	Fig. 280,	15	3·34	1·42
,,	,,	30	4·6	2·0
,,	,,	45	5·46	2·54
Fig. 283,	Fig. 282,	15	4·6	1·42
,,	,,	30	5·96	2·0
,,	,,	45	6·72	2·54
Fig. 285,	Fig. 284,	15	4·4	1·42
,,	,,	30	5·52	2·0
,,	,,	45	6·6	2·54
Fig. 287,	Fig. 286,	15	3·24	1·42
,,	,,	30	4·28	2·0
,,	,,	45	4·82	2·54

The remainder of the construction can be done graphically with quite sufficient accuracy for practical purposes.

The point C at which contact commences is the point at which the circle joining the tops of the wheel teeth cuts the contact path, and the point D, at which it ends, is either the point at which the top of the thread cuts the contact path, or that point on the contact path which is just touched by a circle struck from the wheel centre, whichever is reached first.

Fig. 288 shows the approximate shape of the contact line across the face of the tooth in successive positions as it passes through the arc of action.

Figs. 277, 278, and 279 show a 30-tooth wheel in gear with the worm. The angle of the thread faces being 12°, the arc of action practically coincides with twice the pitch, so that with an ordinary rack and wheel the length of the contact line would be equal to the

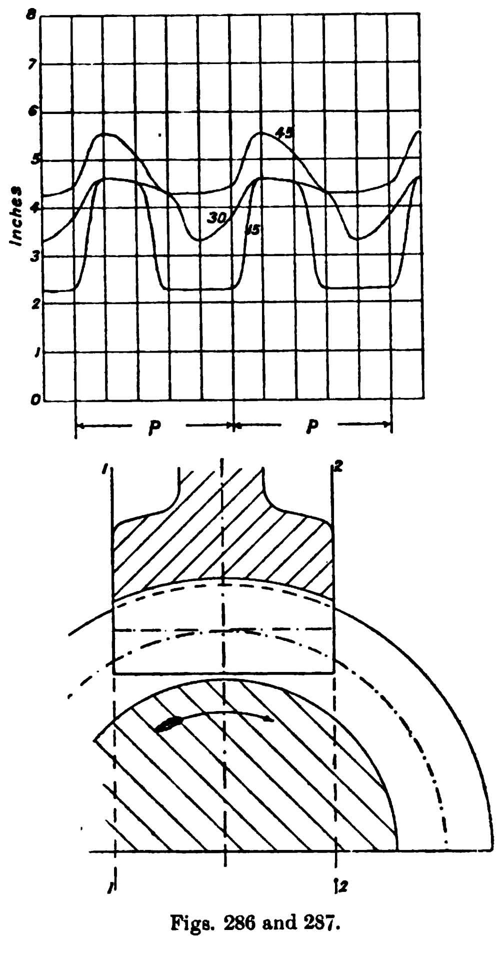


Figs. 284 and 285.

width of two teeth, and its length would be constant, as one tooth would come into gear just as another was going out. Owing to the spiral form of the worm gear, the length of the contact line varies as shown in Fig. 283. There is, however, less variation with the 30-tooth wheel, in which the arc of action equals twice the pitch.

than with the 15- or 45-tooth wheels, in which the relation of the arc of action to the pitch is not a whole number.

Figs. 281, 283, 285, and 287 show the variation in the length of the contact line per phase for wheels of 15, 30, and 45 teeth having different sections, the results being given in Table XIII.



Figs. 286 and 287.

It will be noted that the section of wheel shown in Fig. 282, which is the one most commonly used, gives the best results.

Owing to the rotation of the worm, the frictional loss in worm gear is necessarily much greater than in spur gear for the same coefficient of friction. Referring to Fig. 277, if this were an ordinary rack and wheel, the amount of sliding of the teeth in passing from C to P would be about $\frac{3}{16}$ inch, which we may call the slide of the teeth. The rotation of the worm causes a sliding motion at right angles to this, and its amount during the same period amounts to about $4\frac{3}{4}$ inches. This we may call the slide of the screw.

The screw friction is so large relatively to the tooth friction that the latter may be neglected in calculations of the efficiency and frictional losses.

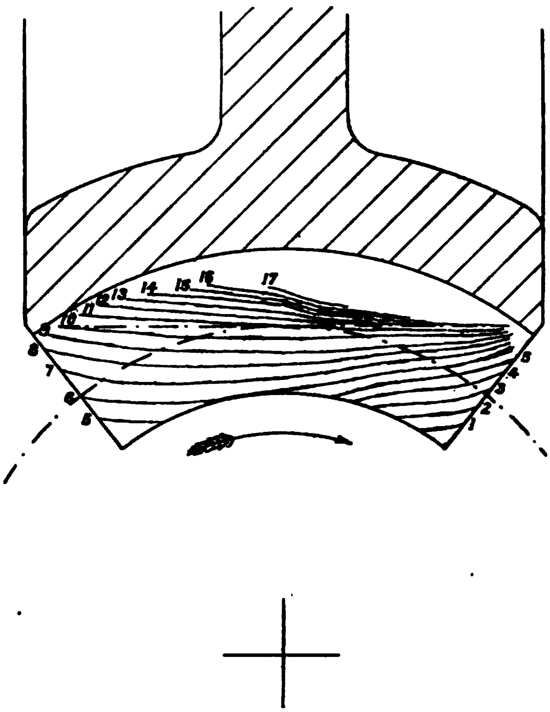


Fig. 288.

Owing to the large amount of slide, as compared with spur gear, continuous lubrication is necessary to reduce the coefficient of friction to a minimum, the gear being contained in a gear case having an oil bath.

The heat generated by the friction of the threads and teeth is principally taken up by the oil by which it is carried to the walls of the gear case, being conducted through them and radiated to the surrounding atmosphere.

The temperature becomes constant when the rate of radiation of heat from the gear case walls is equal to its rate of generation at the teeth.

The load W, which the wheel teeth may carry, apart from their mechancial strength, is thus governed by the temperature at which the lubricating oil may be worked, as an excess of temperature will so lower the viscosity of the oil that it will be squeezed out from between the teeth, so causing them to seize. It has been pointed out by Mr. R. A. Bruce in his very complete paper on "Worm Contact,"* that the load W is supported on a film of oil between the teeth, this film of oil having a definite breadth. Thus the load is taken, not on a line, but on an area, which is equal to the effective breadth of the oil film $\times$ the length of the contact

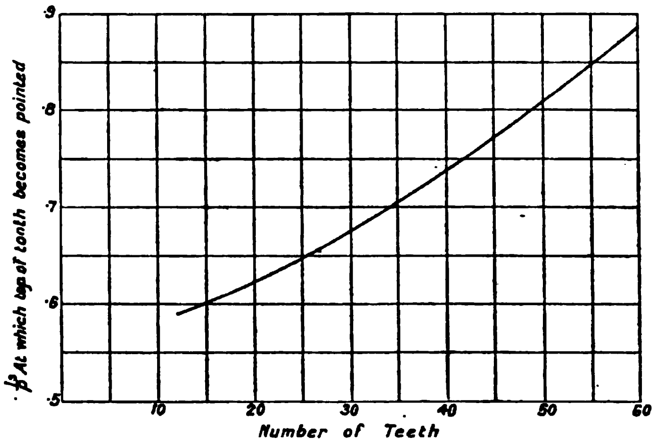


Fig. 289.

line. The effective breadth of the oil film is greater on the entering side of the wheel, where the curves of the threads and teeth are similar, and less on the receding side, where the threads and teeth are of opposite curvature, the average value of the effective breadth varying as $\sqrt{d}\,.\cos a$ at the pitch line.

The load W for a given temperature is then $= K_1 l_4 \sqrt{d} \cos a$, in which K_1 is a constant depending chiefly on the viscosity of the lubricant at the working temperature, and varying with the speed, radiating surface of the gear case, volume and nature of the oil, etc.

The gear in Figs. 277, 278, and 279, with a 30-tooth wheel of the section shown in Fig. 282, carried continuously a load of 640 lbs.

* *Proc. Inst. Mech. E.*, Jan. 19, 1906.

with very moderate heating. On the basis of the above formula, with a 45-tooth wheel it would have carried 890 lbs., and with a 15-tooth wheel 350 lbs. with the same pressure per unit of area. As it was subject to overloads, for very short periods, amounting to two or three times the normal load, the pitch had to be much coarser than would have been necessary had the normal load not been exceeded.

In order to avoid the expenditure of time necessary to take out the length l_4, an approximate formula, due to Mr. Bruce, may be used for preliminary designs. The length l_4 depends upon the length of the contact path CD and the width of the wheel. With

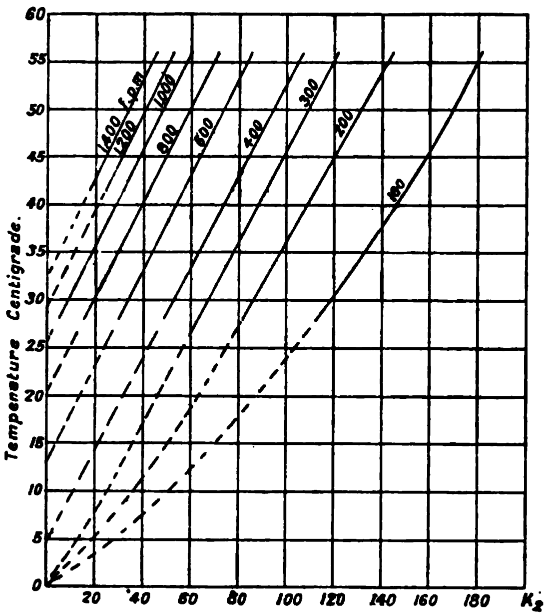


Fig. 290.

a constant angle of incidence, and large number of teeth, the length of contact line may be taken to vary as the width of the wheel, which is proportional to d_3 tan a_1. Also, with the angle of incidence constant, cos a may be omitted, the effective breadth of the oil film being proportional to $\sqrt{d}$. The formula then becomes

$$W = K_2 \cdot \sqrt{d} \cdot d_3 \cdot \tan a_1.$$

K_2 is a constant depending on the same conditions as K_1, but its actual value is different, owing to the different form of the equation.

The curves in Fig. 290 are based on some test figures given in the paper mentioned above, and show the values of K_2 for different speeds, loads, and temperatures, the speed of pitch line in feet per minute being marked on each curve. The particulars of the gear tested are as follows :—Soft steel worm 3-inch pitch diameter triple thread, 3-inch lead, angle of incidence 14·5°. Bronze wheel 9½-inch pitch diameter. Lubricant, heavy cylinder oil. Worm dipping in oil bath. Size of bath about three times volume of worm, with proportionate cooling surface.

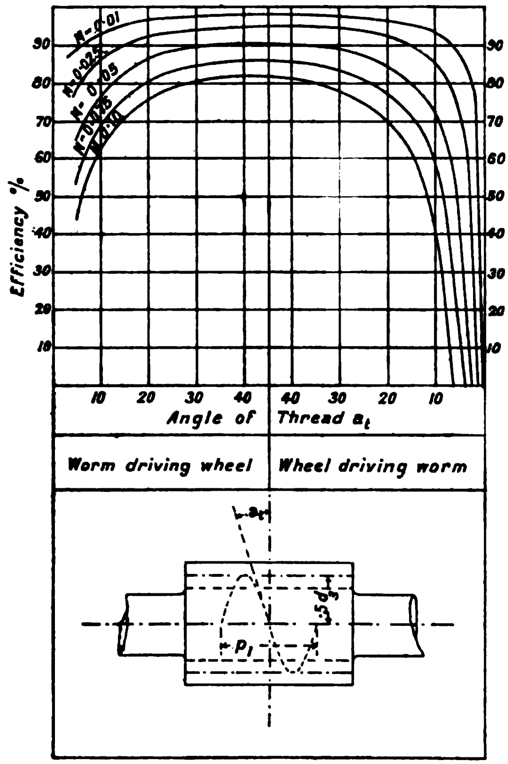


Fig. 291.

Owing to their curvature, the teeth of wheels of the section (Fig. 282) are slightly stronger than those of ordinary spur wheels of the same size. We may, however, for practical purposes, when calculating the strength use the same formulæ and values of K as have been given for spur wheels.

Some interesting examples of worm gear drives were given by Mr. W. L. Spence in his paper on Power Transmission from Electric

Motors,* from which it would appear that in calculating the strength of worm-wheel teeth we may take the strength of phosphor-bronze, in the above formulæ, as being the same as that of steel.

The formula for the efficiency of worm gear is given by Professor Unwin, in his "Machine Design," as—

$$\text{Eff.} = \frac{1 - \mu \frac{p_1}{d_3 \pi}}{1 + \mu \frac{d_3 \pi}{p_1}}.$$

The efficiency curves in Fig. 291 were calculated by the author from this formula, and are reproduced from his paper on "Electric Cranes,"† by permission of the Council of the Institution of Civil Engineers. The left-hand portion of this Fig. gives the efficiencies when the worm is driving the wheel, and the right-hand portion those obtained when the wheel is driving the worm, the points at which the curves cut the zero line giving the angle of thread at which the gear would be self-retaining if there were no other frictional losses in the mechanism. As there are always other frictional losses, these angles may be slightly exceeded without sacrificing the feature of self-retention, which, however, is seldom utilised in modern crane work, owing to the low efficiency which it entails when the worm is driving the wheel.

The curves of efficiency and coefficient of friction in Fig. 292 are based on some efficiency tests made by Mr. N. Westberg.‡ The particulars of the gear are as follows:—Worm 5 threads, 95 mm. pitch diameter, 185 mm. lead. Wheel 68 teeth, 37 mm. pitch, 801 mm. pitch diameter. Speed of motor 780 revolutions per minute. Worm wheel 57½ revolutions per minute. The worm was provided with ball thrust bearings, the shape of section of the wheel was similar to Fig. 284, and the gear was contained in a gear case with oil bath, the worm being above the wheel.

It has already been stated that the working load on the teeth is limited by the permissible rise of temperature.

Generally the machinery of cranes runs intermittently, working for short periods and then standing idle for short periods. When working in this manner, a greater load can be carried by the teeth for a given rise of temperature than would be the case if the gear

* *Trans. Inst. Engineers and Shipbuilders in Scotland*, vol. l., part v.

† *Proc. Inst. C.E.*, vol. clx., p. 368.

‡ *Zeitschrift des Vereins Deutscher Ingenieure*, June 21, 1902.

were running continuously. Each time the gear runs its temperature will rise, and each time it stands its temperature will fall. In the early periods, while the gear is cool, the radiation of heat will not be equal to the rise of temperature, so that as the work of the crane goes on the temperature of the gear will gradually increase. As it increases, the rate of radiation during the idle periods will increase, so that a point will eventually be reached at which the rise of temperature during the working period will be equalled by the fall of temperature when standing, and beyond this point the temperature will remain practically steady.

The ratio of the working time to the working time plus idle time

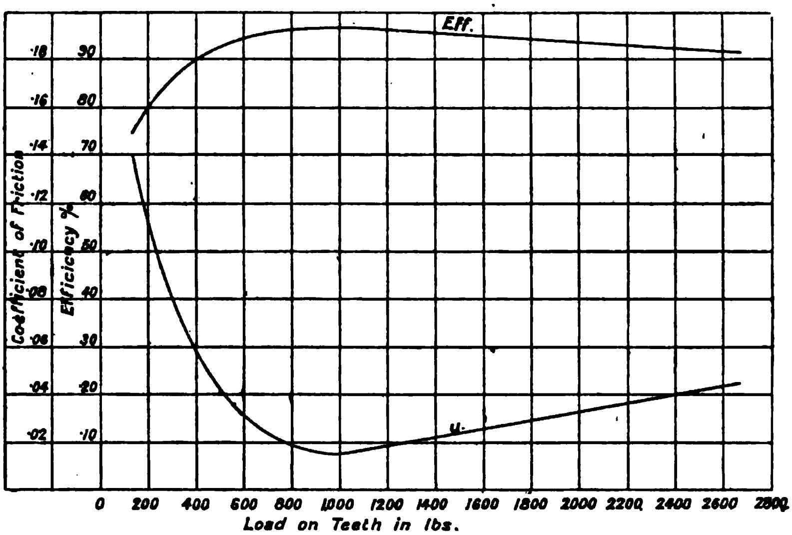


Fig. 292.

may be called the load factor of the gear. Thus, if the gear works for one minute and stands for three its load factor is 0·25, and so on.

The intermittent load which the gear will carry for a given final rise of temperature may be ascertained by a series of simple experiments.

Several runs should be made with different loads on the gear, the temperature being taken every few minutes, and the run continued until the temperature becomes constant. When the highest temperature has been reached the gear should be allowed to stand, and as it cools down the temperature should be taken at frequent intervals.

From the figures so obtained a series of curves such as those in Fig. 293 is laid out. These are logarithmic curves, the formulæ of which are, for the heating curves—

$$C = M \cdot \left(1 - e^{-\frac{t}{T_3}}\right),*$$

and for the cooling curves,

$$C = M \cdot e^{-\frac{t}{T_4}},$$

in which C_t is the temperature Centigrade at the time t in minutes from the commencement of the period, M the final rise of temperature on the heating curves, or the temperature from which the cooling starts on the cooling curve, T_3 is the time constant, and represents

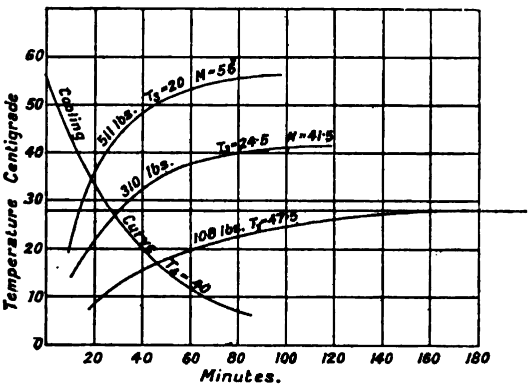


Fig. 293.

the time which the gear would take to attain the final temperature if there were no losses from radiation; for the present purpose it may be taken as the value which satisfies the equation, and e is the base of natural logarithms = 2·718.

Having decided on the permissible rise of temperature, we can by the following method determine the working load which, with a given load factor, will give this rise of temperature.

The rate of heating at any point on the heating curves $= \dfrac{M - C_t}{T_3}$, and the rate of cooling at any point on the cooling curve $= \dfrac{C_t}{T_4}$. The rates being in degrees per minute. When the temperature has become practically steady, the rates of heating and cooling

* See page 257.

are inversely proportional to the times of working and standing, so that for gear working on short periods the load factor is

$$= \frac{\dfrac{C_t}{T_4}}{\dfrac{M - C_t}{T_3} + \dfrac{C_t}{T_4}} = \frac{1}{\dfrac{T_4\left(\dfrac{M}{C_t} \times 1\right)}{T_3} + 1}.$$

The curve (Fig. 294) showing the relation between the load on the teeth and the load factor for a temperature rise of 28° C. has been calculated by this formula from the curves in Fig. 293. Thus, for a load of 511 lbs., with a load factor of 0·33 (working, say, for one minute and standing for two), the mean final rise of temperature of the gear will be 28° C. Referring to Fig. 293, it will be seen that

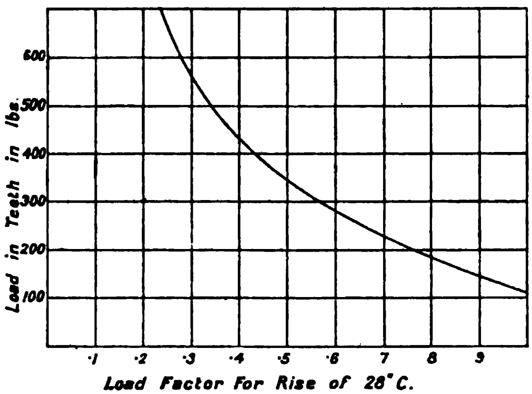


Fig. 294.

running continuously with a load of 511 lbs. a temperature rise of 28° C. will be attained in about 14 minutes. The relation which these curves establish between intermittent and continuous running is of valuable assistance in testing. It is much more convenient to test by means of a continuous run rather than an intermittent one, and so long as we know the relation between the two the continuous test is just as good. Thus if the gear is required to work with a load of 511 lbs., a load factor of 0·33, and a final temperature rise of 28° C., we know it will fulfil these conditions if, on a continuous run with this load, it attains this rise of temperature in 14 minutes.

The first machines for cutting worm wheels by means of a hob were patented by Sir J. Whitworth in 1835. In these machines the wheel blank was revolved at the correct speed, and the hob fed in towards the centre of the wheel. This method continued to be the standard till about ten years ago, when Mr. J. H. Gibson * introduced the improved method, in which the centre of the hob is placed at the same distance from the centre of the wheel as the worm will have from the finished wheel. The hob is then fed tangentially, so that it threads its way through the metal of the wheel rim, and as at the same time the wheel rotates at its correct speed a very perfect form of tooth is produced. Instead of a hob, a single fly cutter may be used, the cost of which is so small that it cannot be regarded as an objection to the use of special angles of thread, if such are otherwise desirable.

* See *Engineering*, March 26, 1897.

CHAPTER XIV.

HOOKS, LIFTING MAGNETS, ROPES, AND CHAINS.

In calculations for the design of crane hooks, the hook is taken as being loaded, with the weight W on its centre line (See Fig. 295), the principal stresses being a direct tensile stress on the stem and a compound stress due to combined tension and bending which reaches its maximum at the section ab. The tensile stress in the stem is simply $\frac{W}{A}$, A being the sectional area of the stem at the bottom of the screw threads. On the section at ab there is a tensile stress f_a at a, and a compressive stress f_b at b, their respective values being—

$$f_a = W\left(\frac{1}{A} + \frac{dy_a}{I}\right) \quad . \quad . \quad . \quad . \quad (1)^*$$

and

$$f_b = W\left(\frac{1}{A} - \frac{dy_b}{I}\right) \quad . \quad . \quad . \quad . \quad (2)^*$$

in which A is the sectional area at ab, I the moment of inertia, and d, y_a, and y_b the distances correspondingly lettered in the figure. The moment of inertia is approximately that of the trapezium $cdef$, which encloses the section, the effect of the rounded corners being neglected.

$$I = \frac{(w_a^2 + 4\,w_a w_b + w_b^2)h^3}{36\,(w_a + w_b)} \quad . \quad . \quad . \quad (3)$$

The symbols in this equation representing the correspondingly lettered dimensions in the figure. In actual work the hook is not always loaded directly on the centre line. The work is frequently

* These equations are really only correct for straight tension bars eccentrically loaded, and when applied to curved tension bars, such as hooks, empirical values of f must be employed. The correct method of calculating the stresses in hooks has been very fully investigated by Mr. E. S. Andrews and Professor Karl Pearson, and published by them in a paper "On a Theory of the Stresses in Crane Hooks, &c." Draper's Company Research Memoirs. Dulau & Co., 37 Soho Square, W.

held in slings which lie in the hook at a considerable angle, as shown in the figure, and tend to prize it open. Occasionally work is lifted on the point of the hook, by inserting it in a hole or under a flange, this tending to pull the hook out straight from the point, and sometimes the hook is used to drag work along the floor. Increased stress is also at times thrown on the hook through slipping of the slings, causing the load to surge. Although the effect of surges is greatly lessened by the present day practice of using wire ropes

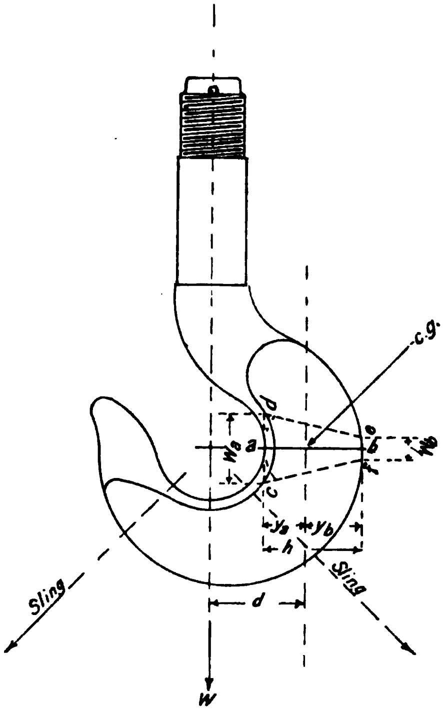


Fig. 295.

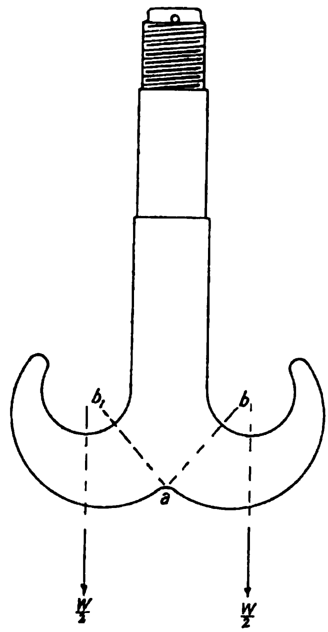


Fig. 296.

instead of chains, the elasticity of the ropes causing them to act as springs, still a certain increase of stress must take place. Consequently the stresses to which a hook may be subjected are somewhat indeterminate, and this uncertainty requires to be allowed for in the factor of safety. In designing a new hook for a given class of work, equations 1 and 2 may be used with values of f_a and f_b taken from hooks which have already given satisfaction in the same class of work.

In a range of hooks the dimensions of which are strictly proportional, their carrying capacity will be proportional to the square of their linear dimensions. Thus, if one hook is calculated out in

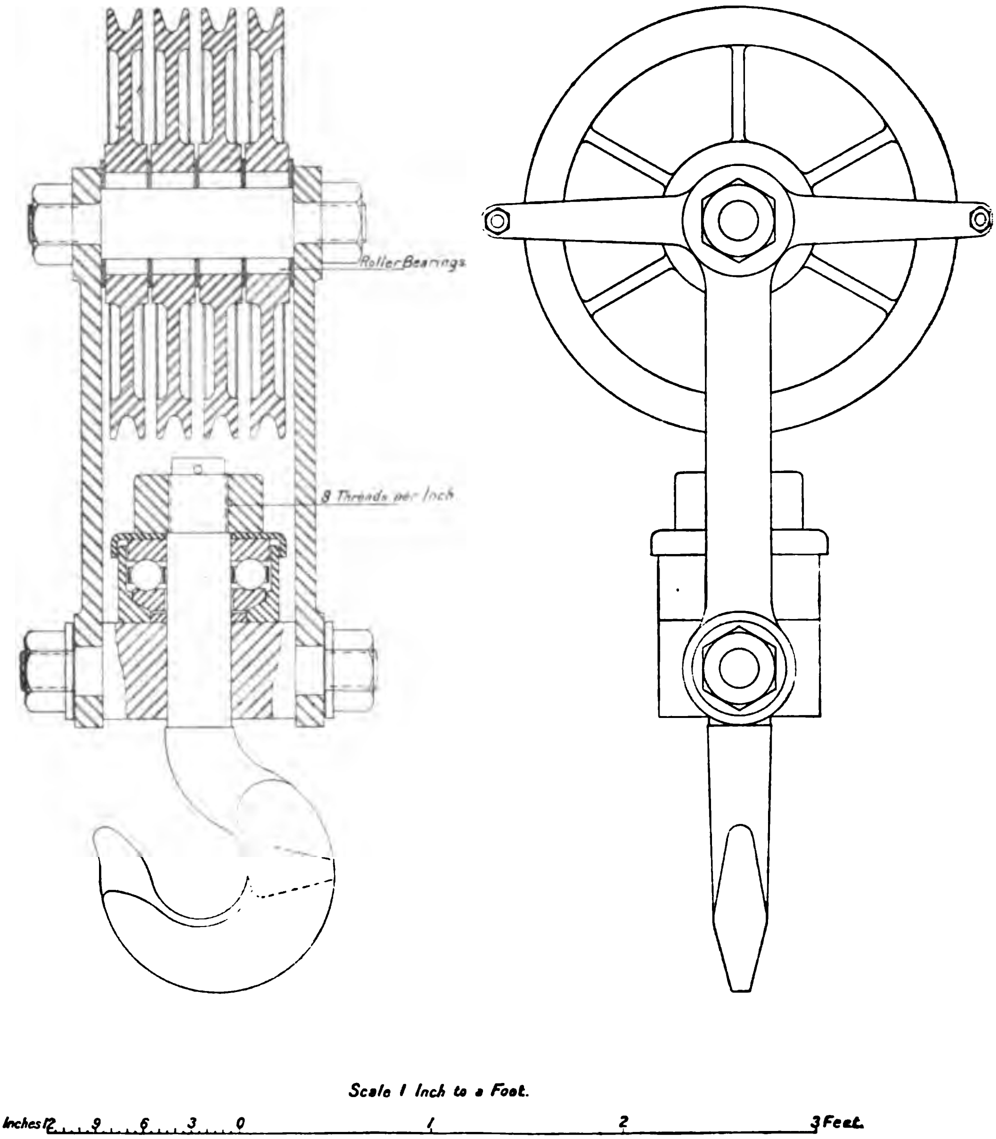


Fig. 297. Fig. 298.

Steel Hook for 30-Ton Crane.

detail, the dimensions of the others may be taken from it by simple proportion, the basis dimension being the diameter of the stem at the bottom of the screw thread.

In the Rams Horn type of hook (Fig. 296), the maximum stress is along the lines *ab* and ab_1. The leverage of the load being less than in hooks of the type shown in Fig. 295, the rams horn hook is, for a given load, lighter than the other, and so is specially suitable for heavy loads. With this form of hook the slings may be crossed, so avoiding the tendency to pull the hook open.

Figs. 297 and 298 show a steel hook for a 30-ton load. The diameter of the stem at the bottom of the thread is 3·465 inches, the area 9·43 square inches, and the tensile stress 3·2 tons per square inch. The sheaves are steel castings, and the pins, links, and hook are forged in steel. The ball bearing is by the Hoffmann Company, of Chelmsford, and is one of their heavy type.

For handling and stacking plates, bars, pipes, and other general materials, electro-magnets have been used to a considerable extent. The advantage which they offer is that as the magnet can be operated by a switch in the driver's cage no workmen are required at the point where the material is taken up, or at the point where it is deposited. The magnet is hung on the crane hook, and a twin flexible conductor is led to it.

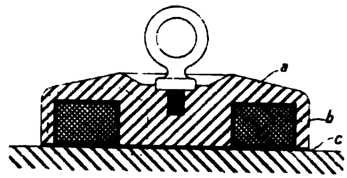


Fig. 299.
Scale same as Fig. 297.

In handling material the magnet is lowered down on to the piece to be lifted. Current is turned on, and the magnet takes hold of the piece. It is then lifted, traversed, and lowered into place. Current is then turned off, and the magnet lets go.

The magnets may be made with pole pieces shaped to suit the material they will be required to handle, a magnet suitable for material having a flat surface being shown in Fig. 299. The magnet consists of the steel casting *a*, the exciting coil *b*, and the facing *c* of hard brass, manganese steel or other non-magnetic material. An eye bolt is provided for hanging on the crane hook. The magnet shown has a tractive force, when laid on a flat piece of mild steel, of not less section than itself, of 6,828 lbs. If handling materials which will not be injured if they are accidentally dropped, this magnet may be used for loads up to 1½ tons. In cases where dropping of the load would cause damage, a larger margin is desirable, and the load should not exceed 1 ton. Electro-magnets should not be used to carry loads over spaces where men are at work, as any accidental failure of current would cause the magnet to drop the load. The magnet shown in Fig. 299 consumes 146 watts. The weight of the copper wire is 38¾ lbs., and the total weight of the magnet is 197 lbs.

For hoisting purposes in crane work steel wire ropes are now largely used in preference to chains.

Formulæ for the weight and breaking strength of steel wire ropes have been given in the chapter on Cableways. The formulæ may be used for the purpose of preliminary designs, but for final designs standard sizes of rope taken from makers' lists should be employed.

In settling the size of rope for a given load, it is usual to choose one having a breaking strength from 6 to 7 times the load which it will be required to sustain. This is not altogether a satisfactory procedure, as it takes no account of the size of sheaves over which the rope will run. Whether the rope is to run over a very small sheave or a very large one the factor of safety employed is the same, although the stress in the rope will be much greater in the first case than in the second.

The stresses in wire ropes due to bending have recently been very fully investigated by Mr. R. W. Chapman, who has given the results of his investigations in a paper contributed to the Australasian Institute of Mining Engineers.* When a rope is bent around a sheave the stress per square inch in the wires at the extreme edges of the section, in the plane of bending is—

$$f = \frac{Ed}{D} \cdot \cos^2 a \cdot \cos^2 b, \quad . \quad . \quad . \quad . \quad (4)$$

E being the modulus of elasticity, d the diameter of the individual wires in the rope, D the diameter of the sheave, a the angle of lay of the wires in the strands, and b the angle of lay of the strands in the rope.

Taking a rope having six strands, and 37 wires per strand, the wires being ·036 inch diameter, the diameter of the rope will be ·792 inch, and its sectional area ·2264 square inch. At 90 tons per square inch the breaking strength of the rope will be 20·4 tons. With a factor of safety of 7, the working load of the rope would be 2·9 tons. Using a sheave, the diameter of which is twenty

* See *Engineering Review*, October, 1908.

times that of the rope—that is, 15·84 inches diameter—and taking $a = b = 18°$, the stress due to bending will be—

$$f = \frac{28{,}500{,}000 \times \cdot 036}{15 \cdot 84 \times 2{,}240} \times \cdot 81 = 23 \cdot 42 \text{ tons per square inch.}$$

This stress not being liable to variation, it may be deducted from 90, leaving 66·58. The direct stress due to the load being $\frac{2 \cdot 9}{\cdot 2264} = 12 \cdot 8$ tons per square inch, the factor of safety, after allowing for the stress due to bending, is $\frac{66 \cdot 58}{12 \cdot 8} = 5 \cdot 2$. With a sheave of half the diameter, the stress due to bending would be doubled, and the factor of safety would be $\frac{90 - 46 \cdot 84}{12 \cdot 8} = 3 \cdot 37$.

The bending moment required to be applied to a rope to bend it to a radius r is—

$$M_b = nm \frac{EI \cos a \,.\, \cos b}{r} + n \frac{EI \cos a}{r}, \qquad (5)$$

in which M_b is the bending moment in inch-lbs., n is the number of strands, m the number of wires per strand exclusive of the central core wire, I the moment of inertia of the individual wires, and r the radius of the sheave. The right-hand portion of the equation refers to the central wire in each strand. In the case of ropes, such as $\frac{6}{24}$, in which each strand has a hemp core instead of a central wire, this portion of the equation is omitted.

The energy in foot-lbs. absorbed in running the rope round the pulley is—

$$\text{Ft.-lbs.} = \frac{M_b S}{r}, \qquad (6)$$

in which S is the speed in feet per minute. This includes the energy absorbed both in bending the rope and straightening it again.

Taking the same rope again, the moment of inertia of the wire ·036 diameter is $\cdot 036^4 \times \cdot 0491 = \cdot 8247 \times 10^{-7}$, and

$$M_b = \begin{cases} \dfrac{6 \times 36 \times 2 \cdot 85 \times \cdot 8247 \times \cdot 9025}{7 \cdot 92} = 57 \cdot 85 \\[2ex] \dfrac{6 \times 2 \cdot 85 \times \cdot 8247 \times \cdot 95}{7 \cdot 92} = 1 \cdot 69 \end{cases}$$

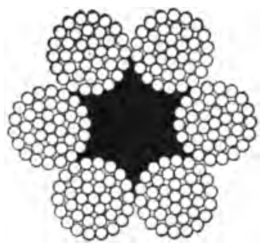
$$\overline{59 \cdot 54} \text{ inch-lbs.}$$

TABLE XIV.—PROPERTIES OF STEEL WIRE ROPES.

E = 28,500,000. $a = b = 18$ degrees. Breaking stress 90 tons per sq. in. Stress due to load $\frac{90}{7} = 12{\cdot}85$ tons per sq. in.
Stress due to bending = 25·7. Factor of safety after allowing for stress due to bending = 5.

No.	Constructions of Rope.								Diameter of Sheave. Inches.	Bending Moment. Inch Lbs.	H.P. absorbed in Bending Rope, per 100 ft. per minute.
	Strands.	Wires per Strand.	Diameter of Wires. Inch.	Diameter of Rope. Inches.	Circumference of Rope. Inches.	Sectional Area. Sq. Inch.	Breaking Load. Tons.	Breaking Load ÷ 7. Tons.			
1	6	19	·032	·448	1·407	·09176	8·258	1·18	12·83	23·59	·01114
2	6	19	·036	·504	1·583	·1163	10·47	1·496	14·44	33·59	·0141
3	6	19	·040	·560	1·759	·1436	12·92	1·846	16·03	46·11	·01744
4	6	37	·032	·704	2·211	·1787	16·08	2·298	12·83	45·87	·02167
5	6	37	·036	·792	2·488	·2264	20·37	2·91	14·44	65·31	·02741
6	6	37	·040	·880	2·764	·2797	25·17	3·597	16·03	89·64	·03389
7	6	37	·048	1·056	3·317	·4018	36·16	5·167	19·24	154·9	·0487
8	6	37	·056	1·232	3·87	·5461	49·15	7·022	22·45	245·9	·0664
9	6	37	·064	1·408	4·423	·7170	64·53	9·22	25·65	367·1	·08674
10	6	37	·072	1·584	4·976	·9035	81·37	11·61	28·86	522·6	·1098

TABLE XV.—Mild Plough Steel Wire Crane Ropes, Black.

Flexible Steel Wire Rope, 6 Strands, each 12 Wires.				Extra Flexible Steel Wire Rope, 6 Strands, each 24 Wires.		Special Extra Flexible Steel Wire Rope, 6 Strands, each 37 Wires.		
 A				 B		 C		
Size Circum.	Approximate Weight per Fathom.	Minimum diam. of Barrel or Sheave for Slow Speeds.	Guaranteed Breaking Strain.	Approximate Weight per Fathom.	Guaranteed Breaking Strain.	Approximate Weight per Fathom.	Guaranteed Breaking Strain.	Size Circum.
Inches	Lbs.	Inches	Tons	Lbs.	Tons	Lbs.	Tons	Inches
1	·63	6	1¾	·88	3¼	...	...	1
1¼	1·06	7½	2½	1·55	5	...	...	1¼
1½	1·44	9	4	1·88	7½	2·0	8	1½
1¾	2·0	10½	5½	2·68	9¾	2·88	11	1¾
2	2·44	12	7	3·78	13	4·0	14½	2
2¼	3·37	13½	9	4·75	16¼	5·2	17½	2¼
2½	4·19	15	12	5·31	20½	6·3	22	2½
2¾	5·25	16½	15	6·12	24	6·81	26½	2¾
3	6·25	18	18	8·0	28½	8·81	32¼	3
3¼	7·06	19½	22	9·37	34	10·38	36½	3¼
3½	8·25	21	26	10·75	39	11·9	43	3½
3¾	9·87	22½	29	12·19	45½	13·5	50	3¾
4	11·25	24	33	13·62	51½	15·3	56½	4
4¼	12·35	25½	36	15·69	59	17·12	65	4¼
4½	13·44	27	39	17·75	65	19·0	70½	4½
4¾	...	...	...	19·88	74	21·69	79	4¾
5	...	...	...	22·5	82½	24·38	88	5

Messrs. Bullivant & Co., Ltd., London.

Taking the speed of the rope at 100 feet per minute, the energy absorbed by the rope will be—

$$\frac{59{\cdot}54 \times 100}{7{\cdot}92} = 751{\cdot}8 \text{ ft.-lbs. per minute,}$$

which is practically ·023 H.P. On the basis of Mr. Chapman's

formulæ Nos. 4, 5, and 6, the author has worked out Table No. XIV. of the properties of steel wire ropes. Table No. XV., kindly furnished to the author by Messrs. Bullivant & Co., Ltd., London, gives the sizes and breaking strength of steel wire crane ropes.

TABLE XVI.—Safe Working Loads for Short Linked Chains.

Table showing the safe working loads for various sized chains from 2½ to 5 tons per square inch. In the last column but one is the *half Admiralty* test, which is usually given as a safe working load.

Size.	At 2½ Tons per Sq. Inch.	At 3 Tons per Sq. Inch.	At 3½ Tons per Sq. Inch.	At 4 Tons per Sq. Inch.	At 4½ Tons per Sq. Inch.	At 5 Tons per Sq. Inch.	Half the Admiralty Proof Test.	Size.
Inch.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Inch.
1/8	·05	·1	·1	·1	·1	·1	·1	1/8
3/16	·1	·2	·2	·2	·2	·2	·2	3/16
1/4	·2	·3	·3	·4	·4	·5	·3	1/4
5/16	·3	·4	·5	·6	·7	·8	·5	5/16
3/8	·5	·6	·7	·9	1·0	1·1	·8	3/8
7/16	·7	·9	1·0	1·2	1·3	1·5	1·1	7/16
1/2	1·0	1·1	1·3	1·5	1·7	1·9	1·4	1/2
9/16	1·2	1·4	1·7	1·9	2·1	2·4	1·8	9/16
5/8	1·5	1·8	2·1	2·4	2·7	3·0	2·3	5/8
11/16	1·8	2·2	2·5	2·9	3·3	3·7	2·8	11/16
3/4	2·2	2·6	3·0	3·5	3·9	4·4	3·4	3/4
13/16	2·5	3·0	3·6	4·1	4·6	5·1	3·9	13/16
7/8	3·0	3·6	4·2	4·8	5·4	6·0	4·5	7/8
15/16	3·4	4·1	4·8	5·5	6·2	6·9	5·2	15/16
1	3·9	4·6	5·4	6·2	7·0	7·8	6·0	1
1 1/16	4·4	5·2	6·2	7·0	7·9	8·8	6·7	1 1/16
1 1/8	5·0	5·9	6·9	7·9	8·9	9·9	7·5	1 1/8
1 3/16	5·5	6·6	7·7	8·8	9·9	11·0	8·4	1 3/16
1 1/4	6·1	7·3	8·5	9·7	10·9	12·2	9·3	1 1/4
1 5/16	6·7	8·1	9·4	10·8	12·1	13·5	10·3	1 5/16
1 3/8	7·4	8·9	10·3	11·8	13·3	14·8	11·3	1 3/8
1 7/16	8·1	9·7	11·3	12·9	14·5	16·2	12·4	1 7/16
1 1/2	8·8	10·5	12·3	14·0	15·8	17·6	13·5	1 1/2
1 9/16	9·5	11·4	13·3	15·2	17·1	19·1	14·6	1 9/16
1 5/8	10·3	12·4	14·4	16·5	18·6	20·7	15·8	1 5/8
1 11/16	11·1	13·3	15·6	17·8	20·1	22·3	17·0	1 11/16
1 3/4	12·0	14·4	16·8	19·2	21·6	24·0	18·2	1 3/4
1 13/16	12·9	15·4	18·0	20·6	23·2	25·8	19·5	1 13/16
1 7/8	13·8	16·5	19·3	22·0	24·8	27·6	20·8	1 7/8
1 15/16	14·7	17·6	20·6	23·5	26·5	29·4	22·4	1 15/16
2	15·7	18·8	21·9	25·1	28·1	31·4	24·0	2
2 1/16	16·7	20·0	23·3	26·7	30·0	33·4	25·3	2 1/16
2 1/8	17·7	21·2	24·8	28·3	31·9	35·4	26·8	2 1/8
2 3/16	18·7	22·3	26·2	30·0	33·8	37·5	28·3	2 3/16
2 1/4	19·8	23·8	27·7	31·8	35·7	39·7	30·0	2 1/4
2 5/16	21·0	25·2	29·4	33·6	37·8	42·0	31·8	2 5/16
2 3/8	22·1	26·5	30·9	35·4	39·8	44·2	33·6	2 3/8
2 7/16	23·3	27·9	32·6	37·3	41·9	46·6	35·5	2 7/16
2 1/2	24·4	29·3	34·2	39·1	44·0	48·9	37·5	2 1/2

Messrs. E. Baylie & Co., Ltd., Stourbridge.

Crane chains are of the short-link type, the links being 4½ diameters of the iron in length, and 3¼ diameters in width, overall, across the centre. They should be of wrought iron of the highest quality with very high ductility, the breaking stress in tension being 23 tons per square inch, with a reduction of area of not less than 50 per cent., and an elongation of 20 per cent. in 10 inches. In chain slings provided with rings, the rings, to be of equal strength with the chain, should be made from bar twice the diameter of that from which the chain links are made. Steel chains have not so far given satisfactory results as compared with wrought iron.

Table XVI. gives a table of safe loads on wrought-iron short-link crane chains. This is taken from a paper on chains read by Mr. E. J. Taylor, of the firm of Messrs. E. Baylie & Co., Ltd., Stourbridge, before the Manchester Association of Engineers, and is reproduced here by his kind permission.

The approximate weight of short-link chains in lbs. per fathom is $W = 70\ d^2$ in ½-inch chain to $62\ d^2$ in 2-inch chain, d being the diameter of bar from which the chain is made.

CHAPTER XV.

DESIGN OF MAGNETS.

THE tractive force W, in lbs., of a magnet is given by the equation—

$$W = \frac{A\,\mathbf{B}^2}{72{,}134{,}000} \text{ per pole,} \quad . \quad . \quad . \quad . \quad (1)$$

A being the area, in square inches, of the surface of the magnet pole, and $\mathbf{B}$ the density of the magnetic flux in lines per square inch.

The total magnetic flux through the air gap, as at c in Fig. 299, is then $\mathbf{N} = A\mathbf{B}$. The density of flux in the object which the magnet is lifting may be taken as equal to the flux in the air gap. There is a certain amount of magnetic leakage from pole to pole, the leakage flux not passing through the air gap, and, therefore, exerting no tractive force. The magnetic density in the body of the magnet is—

$$\mathbf{B}_m = \frac{v\,\mathbf{N}}{A_m}, \quad . \quad . \quad . \quad . \quad . \quad (2)$$

A_m being the cross sectional area of the magnet, $\mathbf{B}_m$ the density, and v the coefficient of leakage, which averages 1·25 for magnets of the type shown in Fig. 299. To determine the total ampere turns $\mathbf{A}_s$ required in the exciting coil it is necessary to calculate separately, and then add together the ampere turns required to force the magnetic flux through the gap, the material which the magnet is lifting, and the body of the magnet. The ampere turns required for the air gap are—

$$\mathbf{A}_g = {\cdot}3132\,\mathbf{BG}, \quad . \quad . \quad . \quad . \quad (3)$$

$\mathbf{G}$ being the length of the gap in inches.

The ampere turns required for the metallic portion of the magnetic circuit may be taken from the curves in Fig. 300, which show the relation between the density and the ampere turns required per inch length of magnetic circuit both for wrought iron, or magnet steel, and cast iron.

Taking the magnet in Fig. 299 as an example. The inner pole piece is 7 inches diameter = 38·48 square inches area, and the area of the outer pole piece is the same. The density is taken at 80,000 lines per square inch. As both poles exert a pull, the tractive force is—

$$W = \frac{2 \times 38{\cdot}48 \times 80{,}000^2}{72{,}134{,}000} = 6{,}828 \text{ lbs.}$$

The ampere turns required for the exciting coil are as follows :— The facing c is $\frac{1}{16}$ inch thick, and the flux has to pass twice through this. The ampere turns for the gap will then be—

$$\mathbf{A}_s = {\cdot}3132 \times 80{,}000 \times {\cdot}0625 \times 2 = 3{,}132.$$

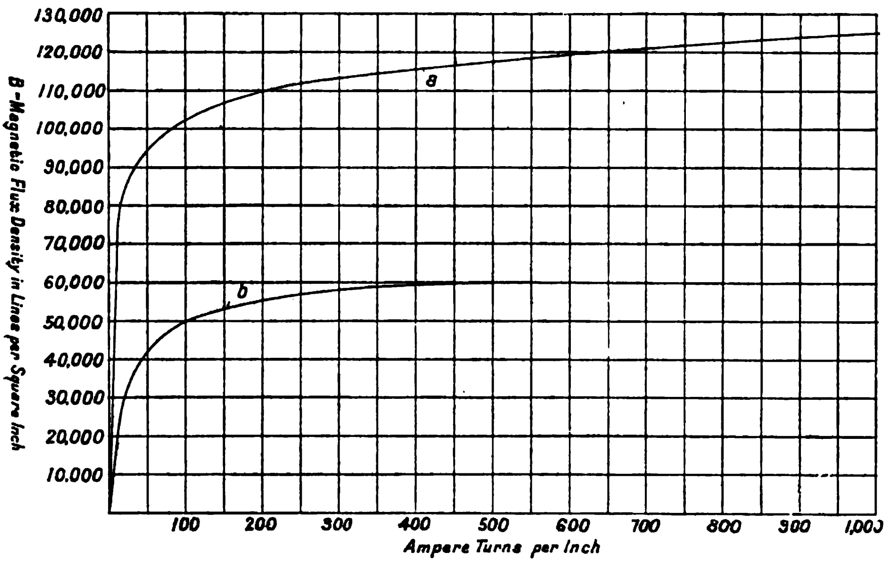


Fig. 300.

a = wrought iron or magnet steel.
b = cast iron.

The density in the steel plate may be taken at 80,000, and curve a (Fig. 300) shows that this requires 15·2 ampere turns per inch. The length of magnetic path, as shown in dotted lines, is 7 inches, so that the ampere turns are 15·2 × 7 = 106·4.

The cross sectional area of the magnet with respect to the flux is uniform, and is the same as that of the pole piece. The density in the magnet is then 80,000 × 1·25 = 100,000, and requires 79·5 ampere turns per inch. The length of magnetic path is 12 inches, and $79 \cdot 5 \times 12 = 950$ ampere turns. The total excitation required is then $3{,}132 + 107 + 950 = 4{,}189$ ampere turns.

In solenoid plunger magnets of the type shown in Fig. 236 the plunger is subject to two separate pulls. One is the direct magnetic pull exerted by the pole piece on the plunger, and is in accordance with equation No. 1, while the second is the solenoidal pull of the coil on the plunger. There is considerable friction between the plunger and the brass bush in which it slides, and it is usual to regard the solenoidal pull as being about equal to the frictional resistance, and so to neglect both in the calculations. In applying equation No. 1, A is the sectional area of the plunger, and $\mathbf{B}$ is the density in the air gap between the plunger and the pole piece. When the plunger is in its outer position the ampere turns required for the air gap, owing to its length, are so great relatively to those required for the remainder of the magnetic circuit that the latter may be neglected, so that the total ampere turns may be found from equation No. 3. Where a coned plunger is used, $\mathbf{G}$ may be taken as the distance between the nearest points of the plunger and pole piece. Referring to Fig. 236, the plunger is $1\frac{3}{4}$ inches diameter and $2 \cdot 405$ square inches sectional area. The length of stroke is $2\frac{1}{2}$ inches, but as the coned portion of the plunger is $1\frac{1}{4}$ inches long, $\mathbf{G}$ is about $1\frac{1}{4}$ inches. The pull required being 39 lbs.

$$\mathbf{B} = \sqrt{\frac{72{,}134{,}000 \times 39}{2 \cdot 405}} = 34{,}200.$$

And the ampere turns—

$$\mathbf{A}_s = \cdot 3132 \times 34{,}200 \times 1 \cdot 25 = 13{,}400.$$

When current is switched on, and the plunger is drawn towards the pole piece, the diminution of the length of the air gap causes an increase in the magnetic flux, and this leads to an increase in the pull, so that the plunger is drawn along with an increasing velocity, and is liable to strike the pole piece with considerable force. In order to lessen this force it is usual to arrange to trap some air between the plunger and the pole piece, and the compression of this air tends to soften the blow.

If the plunger were allowed to come into close contact with the pole piece the residual magnetism might be sufficient to hold up the plunger when current was switched off, and so prevent the brake from acting. A brass washer about $\frac{1}{16}$ inch thick is, therefore, fitted on the plunger, and is shown in the drawing. In calculating the force on the plunger when in its inner position the reluctances of all the parts of the magnetic circuit must be taken into account, as the gap being now extremely small its reluctance only forms a small portion of the total reluctance of the magnetic circuit.

Fig. 301 gives the characteristics of the solenoid plunger magnet shown in Fig. 236. The curve *a* shows the pull exerted at different positions of the plunger with constant excitation, as calculated on the reluctance of the air gap only, the remainder of the magnetic circuit being neglected. Curve *b* shows the correct curve of pull when the whole magnetic circuit is taken into account, and the difference between curves *a* and *b* represents the error introduced by taking the air gap only into account in the calculations. Curve *c* is a hyperbolic curve showing the total pressure exerted by the air trapped between the plunger and pole piece as the plunger is drawn inwards, provided that there is no leakage.

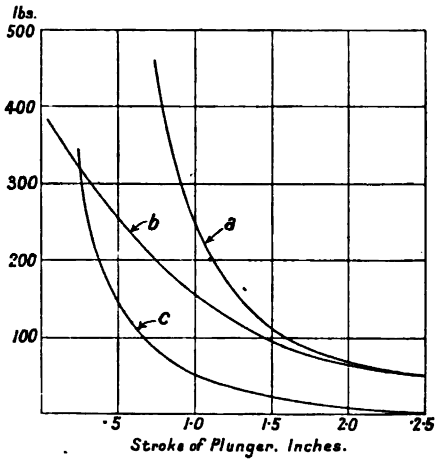


Fig. 301.

In proportioning the windings of magnet coils, the relation to be adopted between the watts absorbed by the coil and its cooling surface, in order to prevent undue heating, will depend upon the circumstances under which the coil is to work. In the case of a coil connected in series with a hoisting motor, the winding may be proportioned to give a loss as high as 3 watts per square inch, when taking the normal full-load current of the motor. The average loss is, of course, much less than this, as the motor has frequently to lift and lower the empty hook, and, except in transporter work, the loads themselves vary, so that the average current passing through the motor is less than the full-load current, and as the loss in the coil is proportional to the square of the current, the average loss will be much below the loss of 3 watts per square inch, which occurs when the motor is lifting the full load. On transporter work it is advisable not to exceed a loss of 2 watts per square inch.

Coils which are connected in shunt with a motor receive the full voltage of the circuit each time they are switched on, and the watts loss calculated on the full voltage should, therefore, be equal to the average loss obtained in a series coil. A figure which gives satisfactory results is 1·25 watts per square inch.

With regard to the cooling surface, the whole surface of the coil parts with heat to the metal casing, which conducts it away and radiates it. The outer cylindrical surface of the coil is, however, the most efficient, as it is in close contact with the thin wall of the casing through which the heat passes easily. In this chapter the value adopted for the cooling surface is then the area of the outer cylindrical surface of the coil. The remaining surface is roughly proportional to this, so that no great error is caused by adopting this figure.

For calculating the windings, the author has found the following formulæ very useful. They were originally given by him in an article in the *Electrical Review* in August, 1895 :—

The resistance of a 1-inch cube of copper at 15° C. being ·6756 × 10^{-6} ohms, the resistance of a coil is—

$$\mathbf{R} = \frac{\cdot 6756\, sg}{\frac{\pi}{4} d^2} \times 10^{-6} = \frac{\cdot 8602\, sg}{d^2} \times 10^{-6}, \quad . \quad . \quad (4)$$

R being the resistance in ohms, s the total number of turns in the coil, g the mean girth in inches, and d the diameter of the wire in inches. The ampere turns in a shunt coil are—

$$\mathbf{A}_s = \frac{\mathbf{V}s}{\mathbf{R}}, \quad . \quad . \quad . \quad . \quad . \quad (5)$$

V being the voltage across the coil. Combining equations 4 and 5, the diameter of wire required for a shunt coil is—

$$d = \sqrt{\frac{\cdot 8602\, \mathbf{A}_s g}{\mathbf{V}} \times 10^{-6}}. \quad . \quad . \quad . \quad . \quad (6)$$

The weight of a cubic inch of copper being ·3213 lb., the weight of wire in a coil is—

$$W = \cdot 3213\, sgd^2 \frac{\pi}{4} = \cdot 2523\, sgd^2. \quad . \quad . \quad . \quad (7)$$

In designing a magnet, the cross-sectional dimensions are usually fixed, and the length of the coil is then proportioned to give the amount of cooling surface required. In commencing the design the length has to be assumed in order to calculate the ampere turns. The ampere turns having been determined, and in the case of shunt coils the size of wire also, the length may be checked by the following formulæ, and revised if necessary :—

The cooling surface of the coil requires to be $= \frac{\text{watts}}{C}$.

C representing the number of watts allowable per square inch. The cooling surface itself $= \mathbf{L}g_1$, $\mathbf{L}$ being the length of the coil, and g_1 the external girth in inches.

Then,

$$\mathbf{L}g_1 = \frac{\mathbf{W}}{C}.$$

The watts loss in a coil =

$$\mathbf{W} = \mathbf{A}^2\mathbf{R} = \frac{\mathbf{A}_s^2\mathbf{R}}{s^2}$$

$$= \frac{\mathbf{A}_s^2}{s^2} \times \frac{\cdot 8602\, sg}{d^2} \times 10^{-6}$$

$$\left(s = \frac{\mathbf{LD}}{d_1 d_2}\right)$$

$$= \frac{\cdot 8602\, \mathbf{A}_s^2 d_1 d_2 g}{\mathbf{LD}d^2} \times 10^{-6},$$

in which $\mathbf{D}$ is the depth of winding on the coil, d_1 the diameter of the wire over the insulation, and d_2 the diameter of the wire over the insulation plus the thickness of paper insulation (if any) between the layers. The temperature rise of the coils will be about 30° to 40° C., so that their resistance when hot will be about 15 per cent. greater than when cold. Thus, in a shunt coil when hot both the ampere turns and the watts will be 15 per cent. less than when cold, while in a series coil the ampere turns will remain constant, and the watts will be 15 per cent. greater.

For a shunt coil—

$$\mathbf{L}g_1 = \frac{\cdot 8602\, \mathbf{A}_s^2 d_1 d_2 g}{1 \cdot 15\, C\mathbf{LD}d^2} \times 10^{-6},$$

and

$$\mathbf{L} = \sqrt{\frac{\cdot 8602\, \mathbf{A}_s^2 d_1 d_2 g}{1 \cdot 15\, C\mathbf{D}d^2 g_1} \times 10^{-6}}. \qquad (8)$$

For a series coil the size of wire cannot be determined till the size of the coil is settled, so in calculating the length we may eliminate $\frac{d_1 d_2}{d^2}$ and substitute the constant value 1·2.

The length of a series coil is then—

$$\mathbf{L} = \sqrt{\frac{\cdot 8602 \times 1\cdot 2 \times 1\cdot 15 \times \mathbf{A}_s^2 g}{C\mathbf{D}g_1}} \times 10^{-6}$$

$$= \sqrt{\frac{1\cdot 187\ \mathbf{A}_s^2 g}{C\mathbf{D}g_1}} \times 10^{-6}. \quad . \quad . \quad . \quad . \quad (9)$$

The diameter of the wire on a series coil is—

$$d = \sqrt{\frac{\mathbf{LDA}}{\mathbf{A}_s}} - t, \quad . \quad . \quad . \quad . \quad (10)$$

t being twice the thickness of the insulation on the wire, no paper insulation being required between the layers.

As an example, the coil of the magnet shown in Fig. 299 may be calculated by these formulæ. The ampere turns required have already been found to be 4,189. As this is a shunt coil, the ampere turns when cold must be 15 per cent. greater than this, otherwise they will be 15 per cent. too low when the coil is hot. 4,189 × 1·15 = 4,818. The mean diameter of the coil is 11 inches, and the mean girth 34·55 inches. Voltage of circuit, 220.

$$d = \sqrt{\frac{\cdot 8602 \times 4{,}818 \times 34\cdot 55}{220}} \times 10^{-6} = \cdot 02551.$$

The nearest even diameter being ·026, the ampere turns will then be $\frac{4{,}818 \times \cdot 026^2}{\cdot 02551^2} = 5{,}000$. The watts loss in the coil being taken at 1·25 watts per square inch of cooling surface, d_1 at ·034, d_2 at ·039, **D** being 3·5 inches, and the external diameter of coil 15 inches, and girth 47·12 inches.

$$\mathbf{L} = \sqrt{\frac{\cdot 8602 \times 5{,}000^2 \times 34 \times 39 \times 34\cdot 55}{1\cdot 15 \times 1\cdot 25 \times 3\cdot 5 \times 26^2 \times 47\cdot 12}} \times 10^{-6} = 2\cdot 479 \text{ inches.}$$

If this length did not agree with the length assumed in commencing the design, it would be necessary now to alter the drawing and calculate again. In this case, however, it agrees with the length on the drawing, so we may proceed to calculate the coil in detail.

The number of layers of wire will be—

$$\frac{\mathbf{D}}{d_2} = \frac{3 \cdot 5}{\cdot 039} = 89 \cdot 7, \text{ say } 90.$$

And the turns per layer—

$$\frac{\mathbf{L}}{d_1} = \frac{2 \cdot 479}{\cdot 034} = 72 \cdot 9, \text{ say } 73.$$

Total turns $= 73 \times 90 = 6{,}570$.

The resistance of the coil when cold =

$$\mathbf{R} = \frac{\cdot 8602 \times 6{,}570 \times 34 \cdot 55}{\cdot 026^2} \times 10^{-6} = 288 \cdot 8 \text{ ohms.}$$

And when hot, $288 \cdot 8 \times 1 \cdot 15 = 332 \cdot 1$ ohms.

The watts absorbed by the coil when hot are—

$$\mathbf{W} = \frac{220^2}{332 \cdot 1} = 145 \cdot 7.$$

The cooling surface of the coil is $2 \cdot 479 \times 47 \cdot 12 = 116 \cdot 8$ square inches.

The watts per square inch when hot are then—

$$\frac{145 \cdot 7}{116 \cdot 8} = 1 \cdot 248,$$

which practically agrees with the 1·25 watts per square inch intended.

The weight of wire in the coil is—

$$W = \cdot 2523 \times 6{,}570 \times 34 \cdot 55 \times \cdot 026^2 = 38 \cdot 71 \text{ lbs.}$$

This coil being for a lifting magnet, it is necessarily a shunt coil, but we will assume that a brake magnet of the same size is required to work in series with a motor, the full-load current of which is 125 amperes. The ampere turns being constant irrespective of temperature, are taken at 4,189. The watts loss may be 3 watts per square inch. The length of coil required will then be—

$$\mathbf{L} = \sqrt{\frac{1 \cdot 187 \times 4{,}189^2 \times 34 \cdot 55}{3 \times 3 \cdot 5 \times 47 \cdot 12} \times 10^{-6}} = 1 \cdot 206 \text{ inches.}$$

In this case the drawing could be altered and the coil shortened, so that the ampere turns would be reduced slightly. For the purposes of this calculation they may, however, be retained at their present value. The diameter of the wire would then be—

$$d = \sqrt{\frac{3\cdot5 \times 1\cdot206 \times 125}{4{,}189}} - \cdot025 = \cdot330 \text{ inch.}$$

The total turns required are $\frac{4{,}189}{125} = 33\cdot5$, say 34.

The diameter of the wire over the insulation being ·355 inch, the number of layers in the coil will be—

$$\frac{3\cdot5}{\cdot355} = 9\cdot86, \text{ say } 9.$$

And the turns per layer will be $\frac{1\cdot206}{\cdot355} = 3\cdot4$.

This is not sufficient, so the coil will require to be lengthened to take four turns, so making the length $4 \times \cdot355 = 1\cdot42$ inches, There will then be eight layers of 4 turns per layer, and one layer of 2 turns, making 34 turns total, the short layer being the inside one, and the space equal to 2 turns being filled up with wood. The resistance of the coil will be—

$$\mathbf{R} = \frac{\cdot8602 \times 34 \times 34\cdot55}{\cdot33^2} \times 10^{-6} = \cdot009278 \text{ ohm. cold,}$$

and

$$\cdot009278 \times 1\cdot15 = \cdot01066 \text{ ohm hot.}$$

The watts when hot will be—

$$\mathbf{W} = 125^2 \times \cdot01066 = 166\cdot5 \text{ watts.}$$

The cooling surface is $1\cdot42 \times 47\cdot12 = 66\cdot91$ square inch, and the watts per square inch $= \frac{166\cdot5}{66\cdot91} = 2\cdot49$, which is less than the 3 watts per square inch originally intended, this reduction being due to the lengthening of the coil from 1·2 to 1·42 inches. It occasionally happens that a series brake magnet coil is required to be capable of carrying the full-load current of the motor, and to have sufficient turns to be capable of pulling the brake off with less than the full-load current. In this case, in applying equation No. 9, the ampere turns must be increased in the proportion of the two currents. Thus, if the coil just dealt with were required to carry 125 amperes, and to pull off with 62·5 the ampere turns to be taken in equation No. 9 must be $\frac{4{,}189 \times 125}{62\cdot5} = 8{,}378$.

The length of coil required would then be—

$$L = \sqrt{\frac{1{\cdot}187 \times 8{,}378^2 \times 34{\cdot}55}{3 \times 3{\cdot}5 \times 47{\cdot}12}} \times 10^{-6} = 2{\cdot}413 \text{ inches.}$$

The remainder of the calculation is not given here, as it is on the same lines as the preceding one.

If, in designing a coil, the size of wire given by the calculations is an odd size, the nearest obtainable size of wire should be used. Many sizes are now drawn in addition to those of the standard wire gauge, and it is advisable for the designer to have by him a list of the diameters which the wire drawers can supply.

CHAPTER XVI.

MOTORS.

It is not proposed in this work to go into the general question of the design of electric motors, but only to deal with those points in their design which are specially affected by the conditions of crane work.

In the early days of electric crane work the motors used were the same as those used for continuous work. It was soon realised, however, that, as a crane motor is only loaded intermittently, and has frequent periods of rest during which it can cool down, motors rated for continuous work were unnecessarily large, and it then became customary to use motors of lower ratings. Thus, if a given motion on a crane required 20 horse-power, a motor capable of giving 10 horse-power continuously might be used. This, however, soon led to difficulties, as the mechanical parts of motors used under these conditions were insufficiently strong, and the commutating properties were not suitable for the large overloads to which they were subject. The leading firms of motor makers then commenced to construct motors specially designed to suit the requirements of crane work.

In such motors the commutating properties and strength of mechanical parts are the same as for motors of the same power working continuously, while the quantities of iron and copper are much less, as, owing to the frequent periods of rest, the losses allowed in relation to the cooling surface may be much greater than in machines for continuous work.

Series-wound motors are used almost exclusively for driving the different motions of cranes. For the tractive motions, such as travelling, traversing, etc., the arrangement is very simple, consisting of a series motor and a plain speed-regulating controller. For hoisting motions, the arrangement, as may be seen from the descriptions in previous chapters, is not quite so simple, the difficulty being not so much to hoist the load up as to get it down again safely. Of the various arrangements already described, the one which seems to find most favour at present is to use a series-wound motor and a controller having special slow-speed steps, in addition to the ordinary speed-regulating steps, and to use two brakes, an electro-magnet brake on an extension of the motor shaft at the commutator end, and an automatic disc brake of the type shown in Fig. 246 on the motor shaft at the driving end, or on one of the intermediate shafts of the hoisting motion.

One feature of the series motor, which causes it to be favourably regarded for crane work, is that under varying loads its speed varies in some inverse proportion to the load, so that when driving

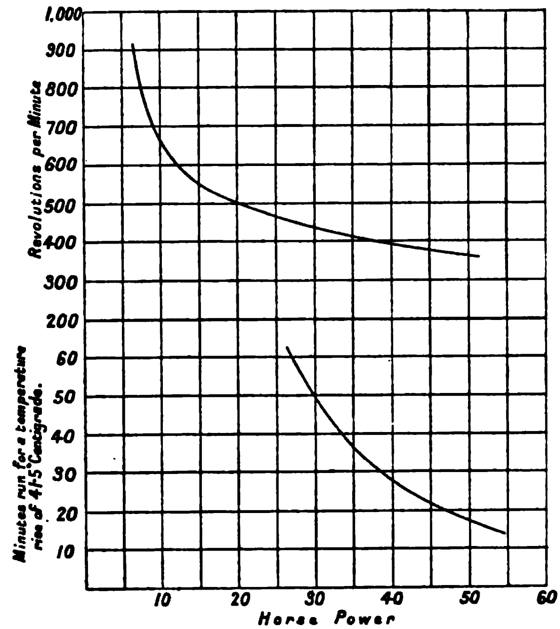


Fig. 302.

a hoisting motion it lifts the light loads at a faster speed than the heavy ones. The extent to which the motor does this depends upon its characteristic curve, and upon the horse-power at which it is rated. Fig. 302 shows a typical series motor characteristic, the upper curve giving the relation of speed and horse-power, and the lower the length of run at a given horse-power which will cause the temperature of the machine to rise 75° F. (= 41·5° C.) above the surrounding air. This machine may then be rated either at 53 B.H.P., 360 R.P.M., ¼ hour; 38½ B.H.P., 395 R.P.M., ½ hour;

or 27 B.H.P., 450 R.P.M., 1 hour, or, of course, at any intermediate values which may be read from the curves.

To illustrate the effect of rating on the variation of speed, take the case of a crab which, with no load on the hook, requires $\frac{1}{5}$ of the horse-power required at full load. Taking the $\frac{1}{4}$-hour motor, the power at full load would be 53 H.P., and at no load 10·6 H.P., and the speeds 360 R.P.M. and 640 R.P.M., or a speed ratio of 1·77. With the $\frac{1}{2}$-hour motor the powers would be $38\frac{1}{2}$ and 7·7, and the speeds 395 and 770, the ratio being 1·94. With the 1-hour motor the powers would be 27 and 5·4, the speeds 450 and 950, and the ratio 2·1. The speed ratio also depends on the efficiency of the crab, for the less the power which this absorbs the higher will

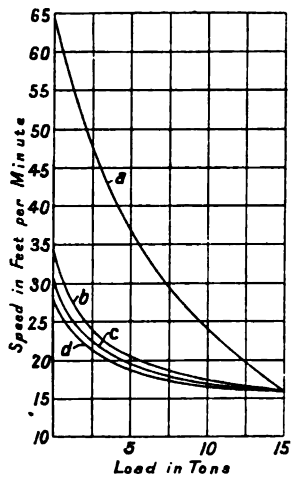


Fig. 303.

a = Shunt motor.
b = Series ,, 1 hour rating.
c = ,, ,, $\frac{1}{2}$,,
d = ,, ,, $\frac{1}{4}$,,

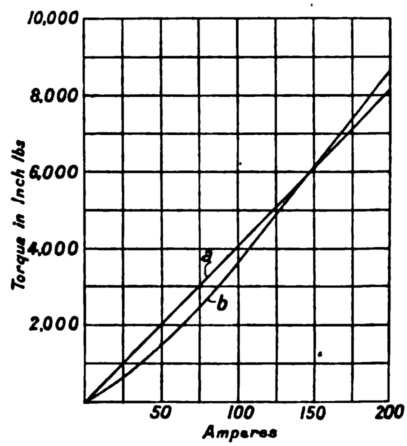


Fig. 304.

a = Shunt motor.
b = Series ,,

be the speed at no load. Fig. 303 shows the relative speed-load curves of a 15-ton crab when driven by a $\frac{1}{4}$, $\frac{1}{2}$, or 1 hour rated motor.

It is frequently stated, as a further point in favour of series motors, that for a given current they develop a greater torque than shunt-wound machines of similar size. This statement, however, requires qualification.

Fig. 304 gives curves of current and torque for the same machine as that from which the curves in Fig. 302 are taken, when wound as a series machine, and when wound as a shunt. It will be noted from this curve that below a certain current the shunt motor gives the greater torque, while above this current the conditions are reversed, the difference, however, not being very great in any case.

As has already been mentioned in the chapter on Brakes, shunt motors may be used to drive hoisting motions, provided that employment can be found for the regenerative currents which occur when lowering. With such motors a much more satisfactory relation between speed and load can be obtained than with series motors, as may be seen from the curve in Fig. 303.

The variation of speed for different loads is obtained by varying the strength of the motor field, and this may be effected automatically. One way in which this may be done is to hang the equalising sheave on a lever, as in Fig. 9, and employ this lever to move the arm of a field regulating resistance switch. An objection which formerly existed with regard to shunt machines was that the field coils being wound with wire of small diameter, the cotton insulation occupied a large amount of space relatively to the wire, and so necessitated the employment of a larger-sized carcase than that required for a series motor of the same power and speed. With the introduction of wires insulated with enamel this objection has been removed, and it is now possible to use the same size of carcase for either series or shunt winding.

The use of commutating poles has further made it possible to construct shunt machines, the speed of which may be varied through a wide range by variation of field strength without setting up any tendency to sparking.

The question of the correct method of testing and rating crane motors is one of considerable interest, and it has been dealt with in articles in the technical press and papers at the Institute of Electrical Engineers by E. Oelschläger, R. Goldschmidt, Dr. R. Pohl, and the author.*

The desirable way to test a motor is to subject it to exactly the same conditions as those which it will meet with in its regular work. On this principle a crane motor would be tested by running it intermittently, the periods of working and standing corresponding with those of the crane.

The inertia and friction of repose of the test load would require to be such as to cause the starting current, each time the motor

* E. Oelschläger, *Electrotechnische Zeitung*, Dec. 20, 1900; R. Goldschmidt, *Journ. Inst. E.E.*, March 9, 1905; C. W. Hill, *Journ. Inst. E.E.*, Feb. 22, 1906; R. Pohl, *Journ. Inst. E.E.*, March 16, 1910.

was switched on, to be the same as the starting current which would be taken in regular work, and the test would require to be continued until the motor temperature became steady.

Such a method of testing is not practicable. In the first place, in the general run of crane work it is impossible to say what the work of the crane will be, especially as in many cases it varies from day to day. Consequently an intermittent test for a motor which is to be used on general crane work can only be based on a guess as to the future working conditions, and its results are of no value as an indication of the behaviour of the motor in actual work.

Secondly, although in transporter work we know the cycles of operation beforehand, there is the difficulty when arranging an intermittent test of getting the starting currents right. Unless this can be done, the test is valueless, as, owing to the short periods of working the starting currents have a considerable effect upon the temperature rise.

Thirdly, the cost of making intermittent tests is very considerable, and adds to the cost of production of the motors without adding anything to their value.

Consequently it is necessary to settle upon some form of test, which can be carried out at a reasonable cost, and which will indicate reliably the suitability or otherwise of the motor for its work.

Electric traction engineers had to deal with a similar problem, and they long ago introduced the system of short time tests. The rated horse-power of a traction motor is that required to be exerted as a maximum, and the standard test conditions adopted by the Committee of Standardisation of the American Institute of Electrical Engineers are that when run continuously for one hour, at the rated horse-power the rise of temperature shall not exceed 75° C. Experience has shown that traction motors which conform to these figures give satisfaction in regular service.

Crane engineers, following the lead of the traction engineers, adopted a one-hour time run for motor testing. The rated horse-power was taken at that required to drive the fully loaded crane when running steadily, and not the power required at starting, and the temperature rise adopted was 75° F. instead of C. It was soon found that, except for motors of large sizes, machines conforming to this test were larger than was necessary. A move was made to the opposite extreme, and a time run of 15 minutes for the same horse-power and temperature rise was tried. A large number of these ¼-hour rated motors were made, but the results obtained with them were most unsatisfactory, many machines breaking down and burning out. A ½-hour time run was then adopted, and experience has proved that for the general run of crane work and ordinary sizes of motors the ½-hour rated machine gives quite satisfactory results, and is, if anything, a little on the safe side.

The maximum temperature rise M of an armature or magnet coil above the surrounding air is expressed by the formula—

$$M = \frac{\mathrm{CW}}{\mathrm{S}}, \quad . \quad . \quad . \quad . \quad . \quad (1)^*$$

and the formula of the curve of temperature rise is—

$$C_t = M\left(1 - e^{-\frac{t}{T_3}}\right), \quad . \quad . \quad . \quad . \quad . \quad (2)\dagger$$

in which t is the time from the commencement of the run, C_t the

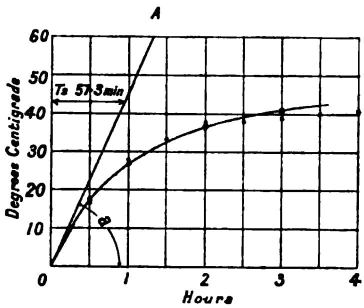


Fig. 305.

× = Observed points.

⊙ = Calculated ,,

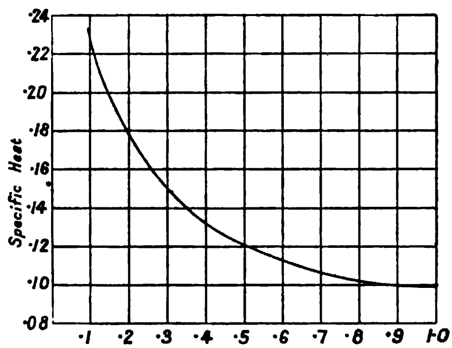


Fig 306.

temperature rise at that time, and T_3 is the time constant of the formula. The nature of this time constant is illustrated in Fig. 305, which gives the temperature curve of a field magnet coil.

If no dissipation of heat took place from the surface of the coil, the rise of temperature would follow the line OA, which is a tangent to the curve at the point of origin, and T_3 represents the time in which the final temperature M would be reached under these circumstances. T_3 is then equal to M cotan a.

The rate of rise of temperature represented by the line OA is dependent upon the specific heat and volume of the coil, and may be predetermined from their values.

* W. B. Esson, *Journ. Inst. E.E.*, vol. xix., p. 148.

† See p. 229.

Taking the specific heat of copper at 0·095, and that of cotton insulation at 0·38, the curve (Fig. 306) shows the relation between the mean specific heat of a coil and its space factor, the latter being $= \dfrac{\text{cross-section of copper}}{\text{total cross-section of coil}}$.

From Fig. 306 the further curve (Fig. 307) has been plotted showing the watts per cubic inch of coil giving a rise of 1° C. per second. It is simply a straight line, and its formula is—

$$\mathbf{W}_1 = 23{\cdot}1 + 35\,\gamma, \qquad \ldots \qquad (3)$$

γ being the space factor.

These curves do not take into account the variation of the specific heat with change of temperature. The variation is, however, very slight, the increase in specific heat for copper being

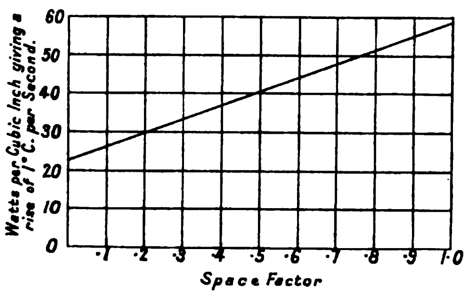


Fig. 307.

about 10 per cent. for a rise of 100° C., so that no great error is involved by the omission. The field magnet coil, of which the temperature curve is given in Fig. 305, was a compound-wound coil, the shunt consisting of 953 turns of wire 0·057 inch diameter, and the series 20 turns of wire 0·22 inch diameter, the wire being double cotton covered in both cases. The space factor of the coil was 0·383, the volume 278 cubic inches, and the cooling surface 148 square inches. The total watts in the coil were 127, and the final rise in temperature 43° C. From equation 1 the value of **C** was then =

$$\frac{148 \times 43}{127} = 50{\cdot}1,$$

this representing the temperature rise per watt of energy loss per square inch. The watts per cubic inch were $\frac{127}{278} = 0 \cdot 457$. The watts per cubic inch per 1° C. per second were—

$$\mathbf{W}_1 = 23 \cdot 1 + (35 \times 0 \cdot 383) = 36 \cdot 5,$$

and the rate of rise of temperature, if there were no dissipation of heat, would be—

$$\frac{\cdot 457 \times 60}{36 \cdot 5} = 0 \cdot 7512 \text{ degree per minute.}$$

On the diagram the relative scales being 20 and 60, the tangent of the angle a will be—

$$\text{Tan } a = \frac{\cdot 7512 \times 60}{20} = 2 \cdot 25 = \tan 66° \ 2',$$

and the time constant will be—

$$T_3 = \frac{60 \ M \text{ cotan } a}{20} = \frac{60 \times 43 \times \cdot 444}{20} = 57 \cdot 3 \text{ minutes.}$$

The formula of the curve (Fig. 305) is then—

$$C_t = 43 \times \left(1 - 2 \cdot 718^{-\frac{t}{57 \cdot 3}}\right).$$

From this the points shown in the figure have been calculated. The observed points taken by thermometer are also plotted in the figure. There is a very close agreement between the calculated and observed points, and it is probably closer than would generally be obtained.

From the foregoing example it will be seen that, for equal scales, the tangent of the angle a is expressed by the equation—

$$\text{Tan } a = \frac{60 \ \mathbf{W}}{(23 \cdot 1 + 35\gamma) V}, \quad . \quad . \quad . \quad . \quad (4)$$

in which V is the volume in cubic inches. The angle a depends then on the specific heat and volume, while the ultimate rise of temperature M, as shown by equation 1, depends on the heat dissipating property of the cooling surface (expressed by $\mathbf{C}$) and its extent ($\mathbf{S}$).

On this account it has been pointed out by Dr. Pohl that the system of short-time tests may tend to lead designers in wrong directions. As the rise of temperature on a short time run will depend almost entirely on the angle a, there will be a temptation to try and keep down the watts per cubic inch of volume rather than the watts per square inch of cooling surface, or, in other words, to endeavour to reduce the rate of rise of temperature rather than the final temperature itself.

Dr. Pohl, therefore, suggests that short-time runs should be abandoned, and that the machine should be run for a sufficient length of time for its temperature to become steady, the horse-power at which it is run being that at which the internal loss is equal to the mean internal loss which will occur when the machine is driving the crane in regular work. As we do not know what the regular work of a crane will be, we should, in order to adopt this proposal, have to determine upon a conventional load factor, the value of which could only be found after a few years of trial and error, as in the case of the time tests already described.

In the original definition of a crane motor load factor, it was assumed that the motor exerted its exact rated horse-power each time it ran, so that the load factor $= \frac{Ht}{H(t + t_1)} = \frac{t}{t + t_1}$, in which t was the time during which the motor ran, and t_1 the time during which it stood.

Thus, if a motor ran for one minute and stood for three it was said to have a ·25 load factor. Under these circumstances, the electric supply taken by the motor in regular work, as recorded by a meter, would be ·25 of the amount which would be recorded if the motor ran continuously at full load, while the mean internal losses in the motor would be ·25 of what they would be if full load were maintained continuously.

Assuming that the heat dissipating property of the cooling surface remains constant, the temperature rise would also be ·25 of the full load temperature rise. It will be seen from the various ampere meter diagrams given in this book that the motor does not develop the same horse-power each time it runs, nor is the power constant during the run. Consequently, a more complete expression for the load factor is necessary.

Fig. 308 shows a simple cycle in which the motor makes one run of 60 seconds and stands for 90. The normal current taken by the motor when running steadily is 100 amperes. The speed of an electricity meter, with constant voltage, being proportional to the current passing through it, a meter in the circuit of this motor would register 112·4 ampere minutes, as the time of the run is one minute and the mean current is 112·4 amperes. The amount of electricity registered by the meter would then be $\frac{112 \cdot 4 \times 60}{100 \times 150} = \cdot 45$ of the amount which would be registered if the motor were running continuously at its full load of 100 amperes. This may then be regarded as a load factor from the user's or supply company's point of view, and in the author's paper at the Institution of Electrical Engineers he called it the external load factor. The internal losses in the machine when working on this cycle will not be ·45 of those when working continuously on 100 amperes, as the internal losses are approximately proportional to the square of the current.

The root mean square value of the current shown in Fig. 308 is 114 amperes, and the internal losses when working on the cycle shown will be $\frac{114^2 \times 60}{100^2 \times 150} = \cdot 52$ of those when working continuously at 100 amperes. This is the figure upon which the temperature rise depends, and was called by the author the internal load factor, but in his paper was based on the square of the horse-power. As we are dealing with the temperature rise, the external load factor need not be further considered, and the internal load factor with which we have to deal may be called simply the load factor = λ.

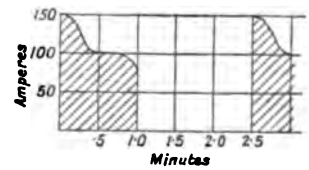


Fig. 308.

In actual work the case is a little more complicated. Taking Fig. 39, cycle No. 11, the length of the cycle is 78·5 seconds, and during this time the hoisting motor makes five separate runs. Assuming that a motor was used, the full load current of which was 100 amperes, the load factor would be—

$$\lambda = \frac{(106^2 \times 8 \cdot 75) + (80^2 \times 1 \cdot 7) + (110^2 \times 4 \cdot 8) + (64^2 \times 9 \cdot 5) + (128^2 \times 2 \cdot 2)}{100^2 \times 78 \cdot 5} = 0 \cdot 31.$$

In this calculation the quantities in brackets are the R.M.S. amperes, and the seconds for each of the five runs. If we wish to test the motor by giving it a continuous run until a final steady temperature is reached, the current at which it should be run is found thus—

At full load the watts loss would be $\mathbf{A}^2\mathbf{R}$, while when working on the cycle shown the loss would be $\lambda\mathbf{A}^2\mathbf{R}$, which would be equal to $x^2\mathbf{R}$, x being the current required for the continuous test run, then—

$$x^2\mathbf{R} = \lambda\mathbf{A}^2\mathbf{R},$$

and
$$x = \sqrt{\lambda\mathbf{A}^2} = \mathbf{A}\sqrt{\lambda},$$

in which **A** is the full load current. Thus for this machine the test current would be $100 \times \sqrt{\cdot 31} = 55\cdot 6$ amperes. Running with this current, the machine being series-wound, the copper losses would be the same as when working intermittently on the cycle shown in the figure.

We have still to consider the iron losses. Loss due to hysteresis is proportional to the speed and to the 1·6 power of **B**, while the eddy current loss is proportional to the square of the speed and the square of **B**. The voltage at the terminals of the machine being constant, while the current varies, the magnetic flux will vary, and the speed will be at all times inversely proportional to the flux. The eddy current loss will, therefore, remain constant, while with decrease of flux and corresponding increase of speed the hysteresis loss will be reduced. The variation in the hysteresis loss is not great, thus in the extreme case of half flux and double speed it is reduced to about $\frac{2}{3}$ the normal. Practically, we may say that the iron loss is constant during each run.

The mean iron loss during the whole cycle is then $\frac{26\cdot 95}{78\cdot 5} = \cdot 342$ of what it would be if the machine ran at full load continuously during the period.

For the continuous time run we then require to run with such a voltage that the relations of speed and **B** result in an iron loss which is ·342 of what it would be if the machine were running continuously at its full load. Dr. Pohl suggests, as an approximation, that the test voltage should be normal voltage $\times \sqrt{\lambda}$.

We have not yet considered the heat due to bearing friction, and to the commutator and brushes. The effect of rise of temperature in a machine is twofold.

Firstly, it increases the resistance of the conductors, and so slightly decreases the efficiency; and, secondly, if the rise is excessive, the insulating materials are injured. In connection with crane motors the latter effect is the only one with which we need concern ourselves in settling the conditions of a test dealing with the temperature of the windings. This temperature is mainly dependent on the copper and iron losses, and, although a little of the heat due to the bearings and commutator may get to the windings, its amount is so small that it may be disregarded.

A continuous time run with reduced load gives no information as to the mechanical behaviour or commutating properties of the machine under full load, so that it must in any case be supplemented by a run at the full-rated horse-power, and this run must be for a limited period in order to avoid excessive temperature rise. Thus we are brought back to the short-time run at full power as the final test. At the same time, the point which has been raised by Dr. Pohl is very important, and should be borne in mind when comparing tenders for crane motors. Any tendency for designers to go in the direction indicated may be easily counteracted, as the previous considerations point to the conclusion that in comparing offers of crane motors all based on the same temperature rise and short-time run, and of equal mechanical strength, the machine which is the lightest in proportion to its general dimensions will have the lowest final temperature rise in regular work, and will be the most desirable machine to employ.

On the same assumptions as those upon which the calculations for the continuous time run at reduced load were made, we may easily calculate the length of continuous run at full load which will give the same rise of temperature as that which will be attained on the ordinary intermittent work of the crane.

Referring again to cycle No. 11 in Fig. 39, it has already been shown that the mean watt loss during the cycle is the same as though the machine ran continuously at 55·6 amperes. We now require to know the heating curve of the machine, which should be ascertained by experiment. It is desirable to take several curves at different loads and speeds as the value of Ç in equation 1 is not constant for a machine, but varies with different speeds, owing to the variation in the air currents set up by the armature. These curves need only be taken once, and can then be used as a basis for future machines of the same size and type.

The heating curves having been taken, may be plotted in the form shown in Fig. 309, which is merely illustrative, and not actually experimental.

From the curve the value of M can be read directly, and the value of T_3 may be calculated. It is advisable to take the value of T_3 at several points on the curve, and then take the average of the value so obtained.

Thus at the point a the value of T_3 =

$$\frac{t \log e}{\log \dfrac{1}{1 - \dfrac{C_t}{M}}} = \frac{60 \times \log 2{\cdot}718}{\log \dfrac{1}{1 - \dfrac{19{\cdot}4}{41{\cdot}5}}} = 95 \text{ minutes.}$$

The maximum temperature rise of the machine being 41·5° C. when running continuously, with 55·6 amperes, the maximum temperature rise with 100 amperes would be in the proportion of the squares of the currents if **C** were constant. The machine being series-wound, its speed will be lower with 100 amperes than with 55·6, and **C** will be somewhat higher. It is on this account that experimental values of **C** are so necessary. Supposing that **C** is

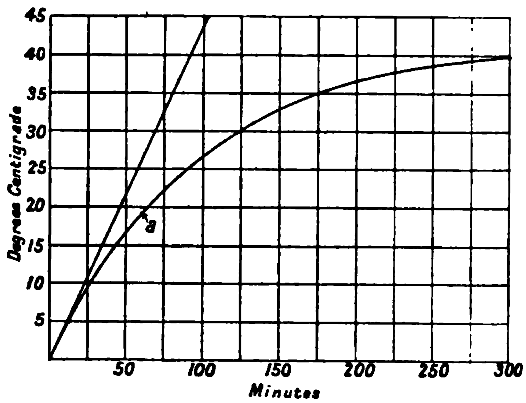


Fig. 309.

10 per cent. higher than at 100 amperes, the maximum temperature rise would be—

$$M = \frac{41{\cdot}5 \times 100^2 \times 1{\cdot}1}{55{\cdot}6^2} = 148 \text{ degrees.}$$

If **C** were constant, M would, as shown in equation 1, be proportional to the watts loss, while the cotan a, as shown by equation 4, would be inversely proportional. As $T_3 = M$ cotan a its value would remain constant. If, however, **C** varies, then T_3 varies in the same proportion. Thus at the 100 ampere load $T_3 = 95 \times 1{\cdot}1 = 104{\cdot}5$ minutes. Then the length of continuous run at 100 amperes, which will cause the machine to have a rise of 41·5° C., will be—

$$t = \frac{T_3 \log \dfrac{1}{1 - \dfrac{C_t}{M}}}{\log e} = \frac{104 \cdot 5 \times \log \dfrac{1}{1 - \dfrac{41 \cdot 5}{148}}}{\log 2 \cdot 718} = 34 \cdot 4 \text{ minutes.*}$$

Thus a machine constructed for a rise of temperature of 41·5° C., when run continuously with a load of 100 amperes for 34·4 minutes, will have the same rise of temperature when working regularly on the cycle shown in the figure.

In the case just dealt with, the cargo, owing to its nature, had to be put on and off the crane hook by hand, and there were consequently long periods during which the motors stood. If the load could have been handled automatically these long stoppages could have been eliminated, and the length of cycle reduced from 78·5 to 43·5 seconds.

The load factor would then be $\lambda = \frac{\cdot 31 \times 78 \cdot 5}{43 \cdot 5} = \cdot 56$, and the current for a continuous run with 41·5° rise would be $100 \times \sqrt{\cdot 56} = 75$ amperes. This evidently requires a larger machine, the temperature rise of which on a continuous run at 100 amperes would be—

$$\frac{41 \cdot 5 \times 100^2 \times 1 \cdot 1}{75^2} = 81 \text{ degrees.}$$

With a machine in which $T_3 = 125$, the length of a run at 100 amperes for a rise of 41·5° would be—

$$\frac{125 \times \log \dfrac{1}{1 - \dfrac{41 \cdot 5}{81}}}{\log 2 \cdot 718} = 90 \text{ minutes.}$$

We will now assume that the crane is being used on a material such as coal which can be grabbed, and that the arrangement of winding gear is the same as on the transporter crane (Figs. 40 to 42), and the cableway (Fig. 54, etc.), there being one motor, with a friction clutch between the hoisting and travelling drums. The

* The curve to equation 2 being asymptotic, the final temperature rise would only be reached when t = infinity. The curve reaches 95·5 per cent. of its final value, however, when $t = 3T_3$, so that the length of time run need not in any case, even for a load factor of unity, exceed three to four times the value of the time constant.

ampere-meter diagram of the motor would then be somewhat as shown in Fig. 310.

The load factor in this case is—

$$\lambda = \frac{(216^2 \times 10) + (112^2 \times 6) + (70^2 \times 1\cdot5) + (95^2 \times 2) + (112^2 \times 6) + (204^2 \times 5)}{200^2 \times 30\cdot5} = \cdot69.$$

The normal full load current of the motor being 200 amperes. For a maximum rise of 41·5°, when run continuously, the machine would require to be run at $200 \times \sqrt{\cdot69} = 166$ amperes. The maximum rise of the machine, if run continuously at 200 amperes, would be—

$$\frac{41\cdot5 \times 200^2 \times 1\cdot1}{166^2} = 66\cdot5 \text{ degrees.}$$

T_3 for a machine of this size would be about 135. Then the length of the test run at 200 amperes for a rise of 41·5° C. would be—

$$\frac{135 \times \log \dfrac{1}{1 - \dfrac{41\cdot5}{66\cdot5}}}{\log 2\cdot718} = 132 \text{ minutes.}$$

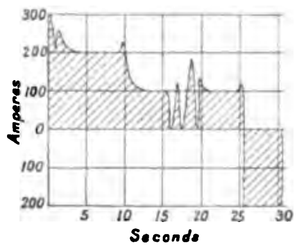


Fig. 310.

In this case a shunt machine is assumed, giving a regenerative current when lowering the grab at a high speed. As the machine is in almost constant motion, it would not be worth while to continually switch the shunt current on and off. It could be kept on continuously and resistance inserted at each lowering period. The load factor of the shunt circuit would then be a little below unity, but practically the shunt winding should be proportioned for continuous work. The regenerative current would reduce the supply of electricity taken by the crane, and the external or consumer's load factor would be ·455, while if there were no regenerative current, and the grab were simply lowered on the brake, it would be ·62.

In the calculations relating to the surface temperature of the motor windings a point to be borne in mind is the rise in temperature which takes place at the surface when the motor stops, due to stoppage of the currents of air set up by the armature when running. An example of this rise in temperature is shown in the curve in Fig. 311. This was taken by thermometer from a magnet coil of a semi-closed motor. It will be noted that after rising to 58° the temperature falls rapidly for an hour, after which it follows a cooling curve, which if produced, as shown in dotted lines, cuts the point *S*.

The effect of this rise of surface temperature is that during an intermittent run, each time the motor stops the surface temperature will rise, and when the motor starts again the rise of temperature will start from a higher point than that at which it stopped. Consequently, at the moment of stopping an intermittent run the surface temperature will be almost, if not quite, as high as that obtained a few minutes after stopping a corresponding continuous run.

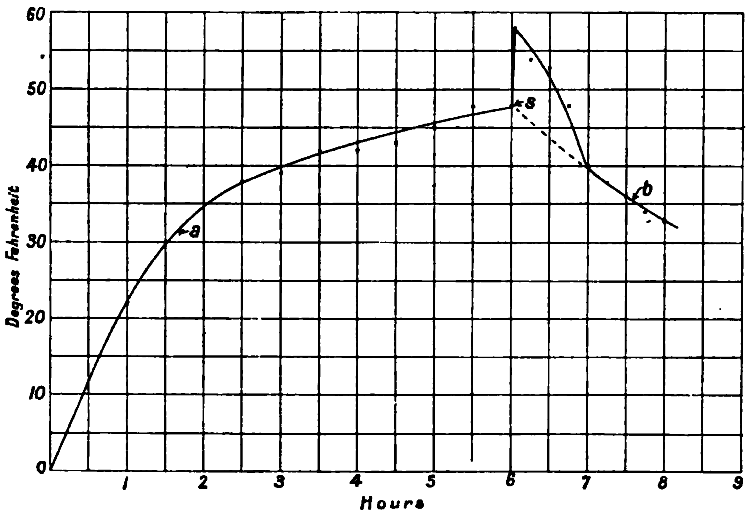


Fig. 311.

a = Heating curve.
b = Cooling ,,
s = Point at which run was stopped.

The considerations of temperature rise in this chapter apply, of course, equally to direct current and to alternating-current motors.

Alternating-current motors of the polyphase type, as at present made, are not so suitable for crane work as direct-current machines, principally on account of their want of flexibility in speed. The speed of a polyphase motor is settled by the number of its poles and the periodicity of the current, and cannot be economically varied, the only method of speed regulation available being by means of resistance in the rotor circuit. Thus the speed of this class of motor is the same whether lifting or lowering, and whether handling the full load or only the light hook, so that the work of the crane is slow and uneconomical compared with one fitted with direct-current motors. The polyphase motor has the same property as a shunt motor of generating current if overhauled by the load, and this property should be allowed for when designing cranes to run on a polyphase circuit. Where there are a number of cranes to be operated, and the supply is alternating, it is advisable to put in a rotary converter or motor generator to supply direct current to the cranes.

Single-phase commutator motors of the type used for electric traction have similar characteristics to series-wound direct-current motors, and are quite satisfactory for crane work where a single phase supply of suitable periodicity is available.

CHAPTER XVII.

CONTROLLERS AND COLLECTORS.

In dealing with controllers, as with motors, it is not intended in this book to go into the question of their general design, but only to consider the arrangement of controller connections and proportions of resistances suitable for crane work.

The two principal points to be observed in the design of a controller are that it shall be capable of bringing the machine to which it is connected to full speed in a specified time, and that its resistances shall be so proportioned that the peaks of current as the controller is moved from step to step shall be uniform in height, and shall not exceed a predetermined amount for which the circuit breakers can be set.

The drum type of controller is now almost universally used. It possesses the advantages that, the fixed contacts being all in one line, and the moving contacts arranged on the surface of a cylinder, any number of combinations may be easily arranged, and that as the breaking of the circuit takes place along one line, a single blow-out magnet can be used.

It would be impossible to illustrate here all the combinations which are in use, so a few typical examples are given. In the figures the fixed contacts are arranged vertically, and are numbered, and the two halves of the drum are shown opened out flat, and divided by vertical lines, which represent the position of the moving contacts on each step of the controller, and these lines are numbered to correspond with the points on the dial plate which is fitted on the top of the controller case.

Fig. 312 shows a simple controller for a series-wound motor. This is suitable for a travelling, traversing, or slewing motion, and may be used for hoisting in combination with a series connected electro-magnetic brake and automatic disc brake, or for short lifts the latter may be dispensed with, as in the crane (Fig. 4). Fig. 313 is a controller for a shunt-wound motor. In this a non-inductive resistance is provided, which is connected across the terminals of the shunt field coil, when the controller is in the off position, by means of contacts 16 and 17. For small machines the non-inductive

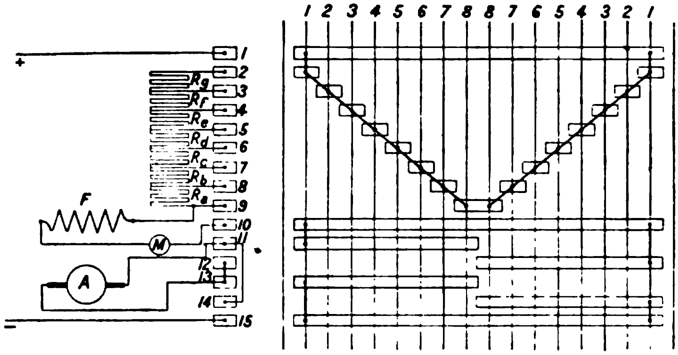


Fig. 312.
A, Armature; *F*, series field coil; *M*, blow-out magnet.

resistance may be connected permanently across the shunt and contact 17 can be dispensed with.

Fig. 314 is a controller arranged for rheostatic braking on the

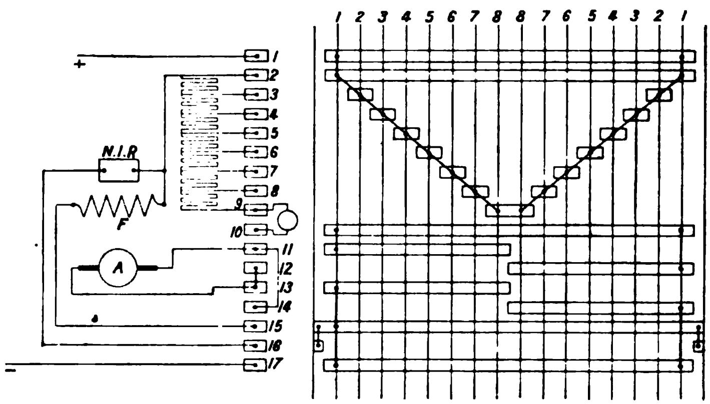


Fig. 313.
A, Armature; *F*, shunt field coil; *M*, blow-out magnet; *N.I.R.*, non-inductive resistance.

lowering side. In this case the motor is series-wound, and an electro-magnetic brake is provided, the coil of which is connected in shunt. It will be noted that the connection of this coil is single pole only. If made double pole an additional contact would be required.

On the lowering side, on steps 1 to 5, the motor is connected as a dynamo, and generates current when driven by the descending load, the speed being regulated by the varying amount of resistance on the various steps. In some cases an additional amount of resistance is necessary, and when this is the case it is inserted between steps 8 and 13, instead of the direct connection shown. On step 6 the brake is held off and the motor disconnected so that the load drops freely. For loads which are too light to run down of themselves, the driving steps 7 to 10 are provided.

A drawback to the use of this type of controller lies in the weakness of the residual field of the motor. When connected as a

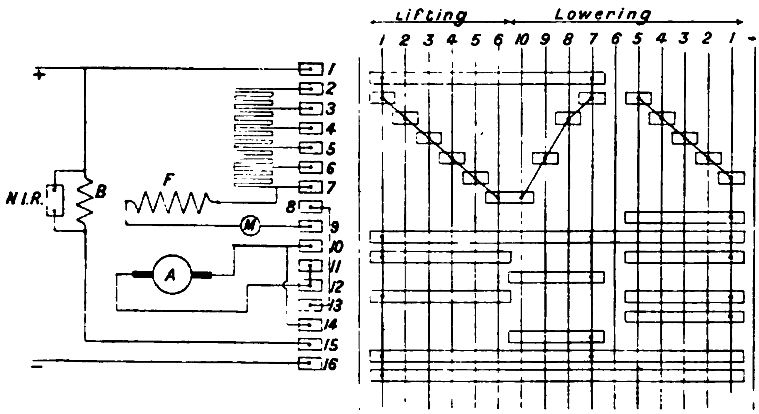


Fig. 314.

A, Armature; *F*, series coil; *B*, shunt brake coil; *N.I.R.*, non-inductive resistance; *M*, blow-out magnet.

dynamo for lowering a load the current generated has to pass through the sliding contacts on the trolley wire, and if there is a little dirt on these the feeble E.M.F. due to the residual field is not sufficient to overcome their resistance, and the machine fails to build up, so that the load drops. With clean contacts, heavy loads can be lowered satisfactorily, but lighter loads may give trouble. If, for instance, the weight of a load is just insufficient to overcome the friction of the crab when standing, the driver has to put on some current to start it. The friction of repose of the crab being now overcome, the load runs freely and accelerates rapidly. When the controller handle is moved back to check the speed, and comes on to the rheostatic steps, the motor builds up suddenly, and as it is running beyond its normal speed, it generates a very high E.M.F., which causes flashing around the commutator.

These difficulties have been overcome in some instances by arranging the controller so that on the rheostatic steps the armature and series field coil are connected into two separate circuits. The armature, as before, is short-circuited through resistances, while the series field is separately connected to the main circuit through other resistances, so that the machine becomes a separately excited dynamo.

This arrangement works well, but is somewhat expensive, owing to the large resistances required.

Fig. 315 shows a controller for a series motor and series brake provided with two slow-speed steps. On these steps resistance is connected in parallel with the armature as well as in series. This controller is very suitable for use with a series-wound hoisting motor and series-connected electro-magnetic brake in combination with an automatic brake of the type shown in Fig. 246.

In connection with this controller there is shown a solenoid main switch, which is an extremely useful appliance. It sometimes occurs that while a crane is running the current fails, the crane stops, and the driver forgets to put off the controller. When current comes on again, if the controller has been left full on, the circuit breaker will most probably go, but if it has been left half on there may be sufficient resistance in circuit to prevent the circuit breaker acting, and so the crane starts up by itself, and may cause a most serious accident. One function of the solenoid main switch is to prevent accidents from this cause. On the controller barrel there are fitted two contacts *aa*, which are placed between the off position and the first step. As the handle is being turned on, one or the other of these contacts, according to the direction in which the handle is being turned, puts full voltage on to the coil of the solenoid, so that the plunger is pulled up and the main circuit made for the controller. On the first and subsequent steps of the controller the resistance *b* or *c* is introduced into the circuit of the solenoid coil, the current in which is thus reduced to an amount which is sufficient to hold the plunger up firmly, but which is insufficient to pull it up if it is in its lower position. If, now, the crane is running, and a failure of current takes place, the plunger drops and breaks the circuit to the controller. When current comes on again, if the controller handle has been left on, it passes through the solenoid coil, but as it has also to pass through the resistance *b* or *c*, the amount of current is insufficient to pull up the plunger so that the motor is protected, and to start the crane the driver has to bring the controller handle back to the off position, and start afresh.

The solenoid main switch may also be used for making emergency stops. For this purpose, instead of directly connecting the points *de*, they may be connected through a circuit provided with push button switches at various points. These switches are held in the closed position by a spring, and are opened by pressing on

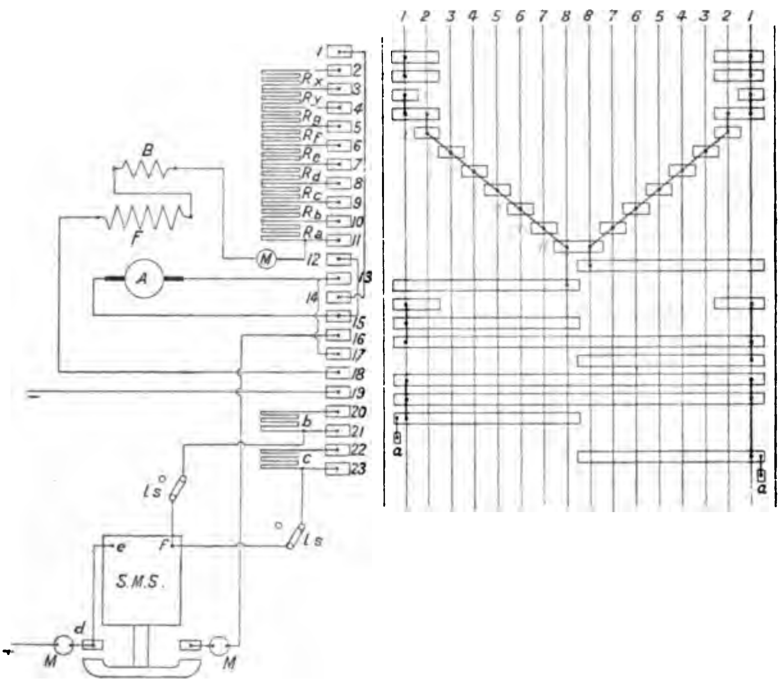


Fig. 315.

A, Armature; *F*, series field coil; *B*, series brake coil; *M*, blow-out magnet; *S.M.S.*, solenoid main switch.

a button. Thus when pressed they break the solenoid circuit, and stop the crane. When released they remake the solenoid circuit, but owing to the resistance *b* or *c* the plunger is not pulled up, and the crane does not restart.

Thus by this arrangement the crane can be stopped by anybody from any required point, but can only be started by the driver.

The arrangement may also be used to prevent over-running in either direction, as has already been explained in the chapter on *Sheer Legs* (see Fig. 47).

In Fig. 315 two circuits are led from the terminal *f* of the solenoid, one for each direction of running, with a limit switch *ls* in each. These switches are normally held in the closed position by a spring, and are so placed as to be pushed open by some moving piece of the crane just before it reaches the extreme limit of its travel. The limit switch in one direction having been opened, the crane cannot be driven further in that direction, but, as there is a separate circuit for the opposite direction, the crane can be reversed, and the limit switch is then reclosed by its spring.

Thus the action of the limit switches is quite automatic and independent of the driver, and may be employed to prevent over-running or over-winding.

The arrangement may also be used to prevent loads being lifted which are too heavy, and to prevent jib cranes being racked out beyond the radius which is safe for a given load. For the first purpose a lever arrangement similar to Fig. 9 may be used to operate a limit switch, while for the second purpose the stay rope of the jib may be passed round a snatch block mounted in a slide, and held in position by a spring. When the pull on the stay rope is excessive, the snatch block moves and opens the limit switch, so that the jib cannot be racked out further, but by reversing the controller it can be brought back. In the case of cantilever cranes the deflection of the cantilever may be employed to operate the limit switch.

In Fig. 316 a modification of the solenoid main switch connection is shown, by which only one resistance is required instead of two. In Fig. 317 the arrangement of connections is shown for a solenoid main switch, which is to be used only as a no-voltage release or as an emergency stop, but not as a limit switch for the two directions. Either of these solenoid main switch arrangements may, of course, be applied to any of the controllers shown.

Fig. 316 shows the arrangement of controller for a shunt-wound hoisting motor of large size, say 100 H.P. or more, with which slow-speed steps are required in both directions. On No. 1 step on the lowering side the field is excited and the brake held off, while the armature is short-circuited on to the first resistance. On this step the motor, therefore, acts as a separately excited dynamo. On the other slow-speed steps the armature is connected to the circuit with resistances both in series and parallel.

This controller arrangement, but without the slow-speed steps, would be suitable for the case shown in Fig. 310 of a shunt-wound motor for coal grabbing.

Fig. 317 is a controller arrangement for two series-wound motors

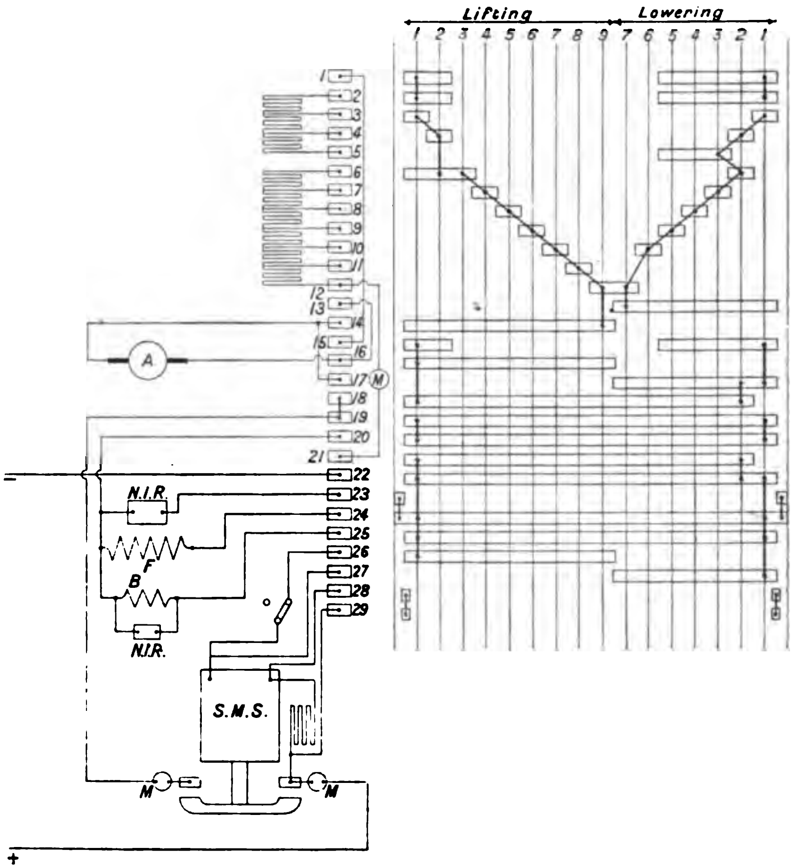


Fig. 316.

A, Armature; *F*, shunt field coil; *B*, shunt brake coil; *N.I.R.*, non-inductive resistances; *M*, blow-out magnets; *S.M.S.*, solenoid main switch.

working in series-parallel, such as would be suitable for the case mentioned in the chapter on *Starting Torque and Acceleration* with reference to Table VIII. Between steps 5 and 6 there are two transitional steps for effecting the change from series to parallel and *vice versâ.* The positions of these steps are not marked on the dial plate, nor are there any notches corresponding to them in the star wheel, so that the driver cannot recognise them, but passes direct from 5 to 6 or 6 to 5. On following out the connections, it will be noted that no interruption of current takes place in changing from series to parallel or the reverse. This is essential, as any interruption of current would cause the brakes to go on.

In order to calculate the resistance required for a controller,

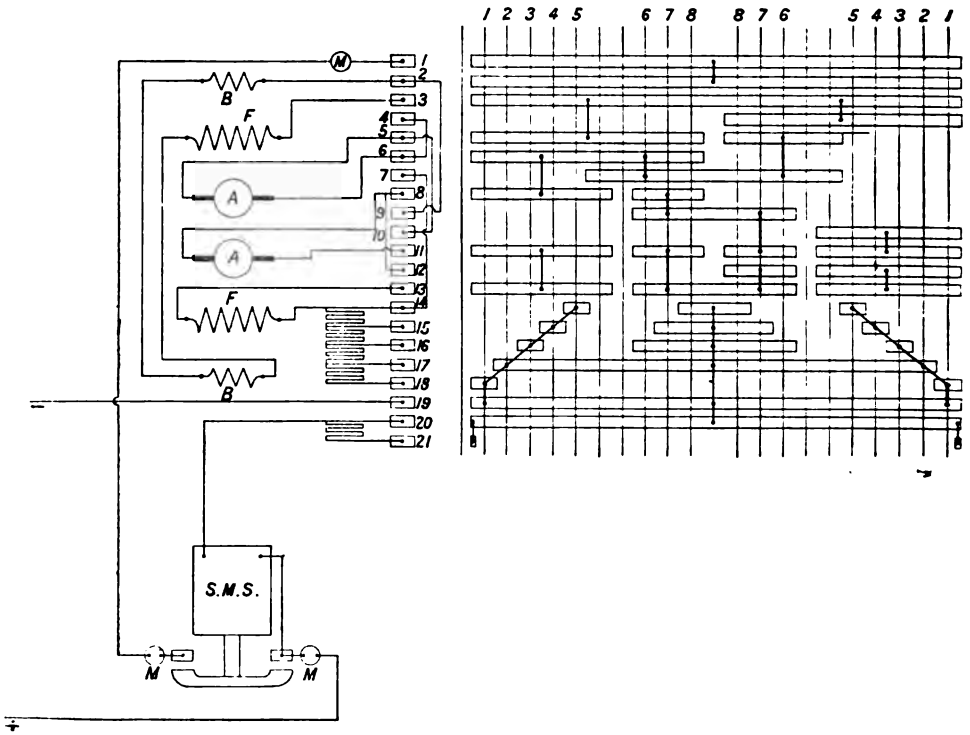


Fig. 317.

A, *A*, Armatures; *F*, *F*, series field coils; *B*, *B*, series brake coils; *M*, blow-out magnets; *S.M.S.*, solenoid main switch.

it is necessary first by the methods given in the chapter on *Starting Torque and Acceleration* to ascertain the torque which the motor must exert in order to get the crane motion up to full speed in the specified time.

From the characteristic curve of the motor the current corresponding to this torque may be ascertained. This will then be the mean current during the starting period.

The mean current being taken approximately as

$$\mathbf{A}_m = \mathbf{A} + \frac{\mathbf{A}_1 - \mathbf{A}}{3}.$$

The maximum current will be—

$$\mathbf{A}_1 = 3\mathbf{A}_m - 2\mathbf{A}, \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$\mathbf{A}$ being the full-load running current of the motor, and $\mathbf{A}_1$ the maximum current on each step of the controller.

The total resistance required will then be—

$$\mathbf{R} = \frac{\mathbf{V}}{\mathbf{A}_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

This including the resistance of the motor and the connections between the motor and controller. The number of steps in the controller will be—

$$n = \frac{\log \dfrac{\mathbf{R}}{\mathbf{R}_m}}{\log \dfrac{\mathbf{A}_1}{\mathbf{A}}} + 1, \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$\mathbf{R}_m$ being the resistance of the motor and its connections to the controller. The resistances in detail will then be as follows, referring to Fig. 312, and calling the last step No. 8.

On No. 8 the resistance is simply $\mathbf{R}_m$.

" 7, $\mathbf{R}_m \dfrac{\mathbf{A}_1}{\mathbf{A}} - \mathbf{R}_m$

" 6, $\mathbf{R}_m \left(\dfrac{\mathbf{A}_1}{\mathbf{A}}\right)^2 - \mathbf{R}_m \dfrac{\mathbf{A}_1}{\mathbf{A}}$,

and so on.

As an example, take a series motor of 26·5 B.H.P., 500 R.P.M., 220 volts, and an efficiency of 90 per cent., the resistance of the motor and connections being ·154 ohm. The normal full-load current is 100 amperes. It is required to attain full speed in ten seconds, and to do this a mean current of 113·1 amperes is required. The rate of acceleration is then 50 R.P.M. per second.

The maximum current will be—

$$\mathbf{A}_1 = 113{\cdot}1 \times 3 - 100 \times 2 = 139{\cdot}3 \text{ amperes.}$$

The total resistance—

$$\mathbf{R} = \frac{220}{139{\cdot}3} = 1{\cdot}575 \text{ ohms.}$$

and the number of steps—

$$n = \frac{\log \dfrac{1{\cdot}575}{{\cdot}154}}{\log \dfrac{139{\cdot}3}{100}} + 1 = 8 \text{ steps.}$$

The resistance in detail will be—

On No. 8 step, 0·154 ohm.
" 7 " $({\cdot}154 \times 1{\cdot}393) - {\cdot}154 = {\cdot}0607$ ohm.
" 6 " $({\cdot}154 \times 1{\cdot}393^2) - {\cdot}154 \times 1{\cdot}393 = {\cdot}0845$ ohm,

and so on, as given on the diagram (Fig. 318).

The back E.M.F. of the motor when running at full load and full speed being $220 - ({\cdot}154 \times 100) = 204{\cdot}6$ volts, the speed on any step of the controller will be—

$$S_2 = \frac{500 \times (220 - 100\,\mathbf{R})}{204{\cdot}6},$$

R being the resistance up to that step.

Thus on step No. 1 it is—

$$S_2 = \frac{500 \times (220 - 157{\cdot}5)}{204{\cdot}6} = 153 \text{ R.P.M.}$$

The number of seconds pause on each step is found by dividing the increment of speed on that step by the rate of acceleration in revolutions per second. Thus on step 4 the increment is 56 R.P.M., and this divided by 50 gives 1·12 seconds. In some cases it is not desirable to switch on the maximum current directly. Suppose that the current on the first step was not to exceed 75 amperes, all other conditions remaining the same. A ninth step would then be required, and the total resistance would be $\frac{220}{75} = 2{\cdot}94$ ohms. The resistance R_h would be $2{\cdot}94 - 1{\cdot}575 = 1{\cdot}365$ ohms. Sometimes a standard controller has to be brought in, in which case the resistance used has to conform to the particulars of the controller.

The equation for the resistance is then—

$$\mathbf{R} = \sqrt[n]{\mathbf{R}_m\left(\frac{\mathbf{V}}{\mathbf{A}}\right)^{n-1}}, \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which **V** is the voltage of the circuit.

Thus if, in the preceding case, an 8-step controller had to be used, but with a current of 75 amperes on the first step, there would only be seven useful steps, and the resistance would be—

$$\mathbf{R} = \sqrt[7]{\cdot154 \times \left(\frac{220}{100}\right)^6} = 1\cdot505 \text{ ohms.}$$

This would be the resistance for the seven steps, the total resistance

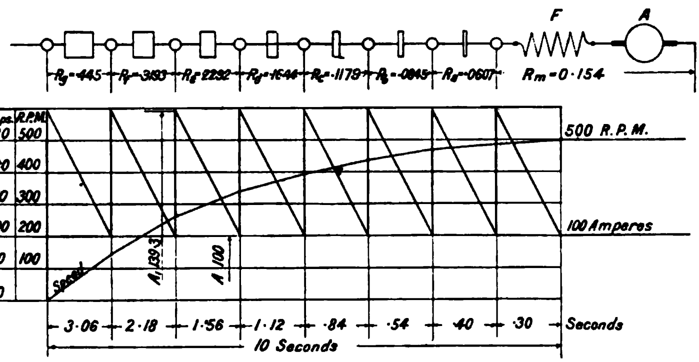


Fig. 318.

A, Armature; *F*, series field coil.

being as before 2·94 ohms, so that R_g would be—

$$2\cdot94 - 1\cdot505 = 1\cdot435 \text{ ohms.}$$

$$\mathbf{A}_1 \text{ would be } \frac{220}{1\cdot505} = 146 \text{ amperes.}$$

$$\mathbf{A}_m \text{ would be } 100 + \frac{146 - 100}{3} = 115\cdot3 \text{ amperes,}$$

and the rate of acceleration 51 R.P.M. per second, so that the starting period would be 9·8 seconds.

In designing series parallel controllers, equations 1 to 4 may be used, two separate sets of calculations being made, one for the series steps and one for the parallel ones. In calculating the series steps, $\mathbf{R}_m$ is twice the resistance of one motor and its connections, while for the parallel steps it is half the resistance of one motor and its connections. The ratio $\frac{\mathbf{A}_1}{\mathbf{A}}$ should be kept as nearly as possible the same on both sets of steps, so that the current peaks per motor will be uniform, so enabling the circuit breakers to be set fairly closely. It is a good plan to arrange the circuit breakers so that when one goes it trips the other, and so prevents the whole load coming on one motor, although this is not advisable in all cases. Sometimes it is better to risk the one motor running overloaded rather than have a stoppage. In calculating the parallel steps, instead of using $\mathbf{V}$ in equations 2 and 4, $\mathbf{V} - \mathbf{V}_3$ should be used, $\mathbf{V}_3$ being the back E.M.F. of one motor when running on the last series step.

As an example, take a series parallel controller for two series-wound motors of 50 B.H.P. each, 500 volts, 570 R.P.M., the resistance of each motor and its connections being 0·6 ohm. The full load running current of the motors is 87·5 amperes each.

$\mathbf{A}_1$ is required to be 131·2 amperes, so that $\mathbf{R} = \frac{500}{131{\cdot}2} = 3{\cdot}81$ ohms, and the number of steps required—

$$n = \frac{\log \frac{3{\cdot}81}{1{\cdot}2}}{\log \frac{131{\cdot}2}{87{\cdot}5}} + 1 = 3{\cdot}86, \text{ say 4 steps.}$$

As the steps do not come to an even figure, the resistance requires to be recalculated—

$$\mathbf{R} = \sqrt[4]{1{\cdot}2 \times \left(\frac{500}{87{\cdot}5}\right)^3} = 3{\cdot}866 \text{ ohms.}$$

$$\mathbf{A}_1 = \frac{500}{3{\cdot}866} = 129{\cdot}3 \text{ amperes,}$$

and

$$\frac{\mathbf{A}_1}{\mathbf{A}} = \frac{129{\cdot}3}{87{\cdot}5} = 1{\cdot}48.$$

On the last series step the back E.M.F. of one motor will be—

$$\mathbf{V}_3 = \frac{500 - (87{\cdot}5 \times 1{\cdot}2)}{2} = 197{\cdot}5 \text{ volts.}$$

Assuming $\mathbf{A}_1$ for the parallel steps to be $129{\cdot}3 \times 2 = 258{\cdot}6$ amperes,

R will be $\frac{500 - 197 \cdot 5}{258 \cdot 5} = 1 \cdot 17$ ohms. And $\mathbf{R}_m$ will be $\frac{\cdot 6}{2} = \cdot 3$ ohm. The number of steps will then be—

$$n = \frac{\log \frac{1 \cdot 17}{\cdot 3}}{\log \frac{258 \cdot 6}{175}} + 1 = 4 \cdot 47, \text{ say 4 steps.}$$

Recalculating the resistance—

$$\mathbf{R} = \sqrt[4]{\cdot 3 \times \left(\frac{500 - 197 \cdot 5}{175}\right)^3} = 1 \cdot 116 \text{ ohms.}$$

$\mathbf{A}_1$ is then $\frac{500 - 197 \cdot 5}{1 \cdot 116} = 271$ amperes.

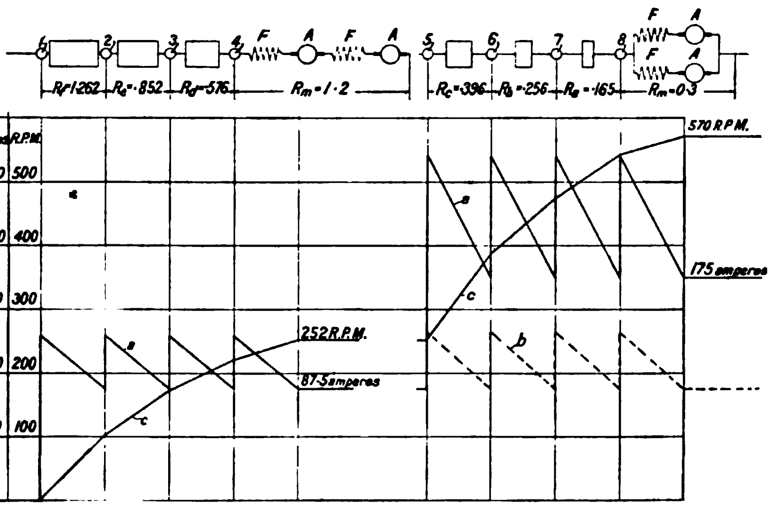


Fig. 319.

a, Total current; *b*, current per motor; *c*, speed; *A*, armature; *F*, series field coil.

The remaining particulars are shown on the diagram (Fig. 319). It will be noted that the current per motor is not absolutely identical on the series and parallel steps, but is very nearly so. In many cases the design of series parallel controllers has to be a compromise, the same resistances being used for both sets of steps, so that the currents cannot be kept uniform throughout. Two or three trial designs should then be made in order to ascertain the best arrangement.

In arranging slow-speed steps, as in Figs. 315 and 316, parallel resistances are used, as by their means slow speed can be obtained over a wide range of load with fewer steps and a smaller quantity of resistance than would be the case if series resistances only were employed. To illustrate this point, we will take the 26·5 B.H.P. series-wound motor already dealt with, the starting diagram of which is given in Fig. 318. If this were required to run on the first step at 10 per cent. of the full-load speed with full current, the speed required would be 50 R.P.M.

The back E.M.F. at full speed being 204·6, at 50 R.P.M. it will

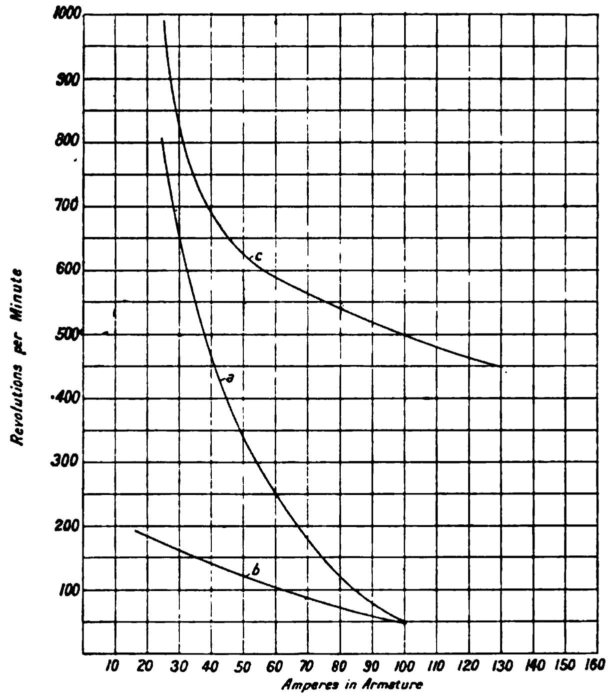


Fig. 320.

a, Speed on slow step with series resistance only.
b, ,, ,, ,, parallel and series resistance.
c, Speed of motor with controller full on.

be 20·46, so that the total resistance required will be—

$$\frac{220 - 20{\cdot}46}{100} = 1{\cdot}995 \text{ ohms.}$$

The resistance already calculated being 1·575, the extra resistance required will be 0·42 ohm. This will give 50 R.P.M. with 100 amperes, but with less loads the speeds will rise as shown on curve *a* in Fig. 320.

The resistance of 0·42 ohm has to carry 100 amperes, and so absorbs 4,200 watts.

If, instead of this, a resistance of 1·1 ohms is connected in parallel with the armature, the other resistances remaining as before, the speeds on this step with various armature currents will be as shown in the curve *b*. This resistance has to carry 29 amperes, and so absorbs 930 watts. To obtain the speeds shown in curve *b* by means of series resistances only would necessitate a large number of extra steps and extra resistances. For many purposes a single slow-speed step giving the speeds shown in curve *b* is sufficient, as there is generally no objection to light loads going faster. Where, however, very slow speed is required over a wide range of loads two slow-speed steps may be used, and it is very seldom necessary to have more. The parallel resistances for the slow-speed steps may be calculated by the following equations:—

For a given slow speed S_s the E.M.F. across the brushes of the motor requires to be—

$$\mathbf{V}_2 = \frac{S_s \mathbf{V}_1}{S_2} + \mathbf{A}_a \mathbf{R}_a, \quad . \quad . \quad . \quad . \quad (5)$$

$\mathbf{V}_1$ being the back E.M.F. of the motor at full load, $\mathbf{A}_a$ the current in the armature, and $\mathbf{R}_a$ the resistances of the armature and brush contacts. The total current will be—

$$\mathbf{A} = \frac{\mathbf{V} - \mathbf{V}_2}{\mathbf{R}_e}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$\mathbf{R}_e$ being the resistance external to the armature—that is, the series resistances and the resistance of the series field winding in the case of series-wound motors. The parallel resistance will then be—

$$\mathbf{R}_s = \frac{\mathbf{V}_2}{\mathbf{A} - \mathbf{A}_a}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

In dealing with shunt-wound machines, the shunt field coil being of very high resistance in comparison with $\mathbf{R}_s$, its effect may be neglected. Referring to Fig. 320, the calculation of the parallel resistance is as follows:—

The total resistance on the first step is 1·575 ohms, $\mathbf{R}_a$ being 0·114 ohm and $\mathbf{R}_s$ 1·461.

$$\mathbf{V}_2 = \frac{50 \times 204{\cdot}6}{500} + {\cdot}114 \times 100 = 31{\cdot}86 \text{ volts.}$$

$$\mathbf{A} = \frac{220 - 31{\cdot}86}{1{\cdot}461} = 128{\cdot}8 \text{ amperes.}$$

And

$$\mathbf{R}_s = \frac{31{\cdot}86}{128{\cdot}8 - 100} = 1{\cdot}106 \text{ ohms.}$$

For a series machine these figures are not absolutely correct, as the field strength is increased and the speed would come a little under 50 R.P.M., being in this case 47 R.P.M. Such a slight difference as this is generally negligible.

In order to ascertain the total current and speeds with a given arrangement of resistances for armature currents ranging from no load to full load, an equation may be constructed from two extreme cases—(1) with the machine standing so that there is no back E.M.F., and (2) with the machine running at such a speed that its back E.M.F. is equal to $\mathbf{V}_2$, so that there is no current in the armature. In case 1 the total current—

$$\mathbf{A}_o = \frac{\mathbf{V}}{\dfrac{1}{\dfrac{1}{\mathbf{R}_a} + \dfrac{1}{\mathbf{R}_s}} + \mathbf{R}_e}, \quad . \quad . \quad . \quad . \quad . \quad (8)$$

and the armature current—

$$\mathbf{A}_a = \frac{\mathbf{A}_o\,\mathbf{R}_s}{\mathbf{R}_a + \mathbf{R}_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

In case 2 the current—

$$\mathbf{A}_r = \frac{\mathbf{V}}{\mathbf{R}_s + \mathbf{R}_e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

If the intermediate values of the currents be worked out, it will be found that the values of the total currents lie along a straight line connecting the points $\mathbf{A}_o$ and $\mathbf{A}_r$. Consequently, for any given armature current $\mathbf{A}_x$ the value of the total current $\mathbf{A}_y$ will be given by the equation—

$$\mathbf{A}_y = \frac{(\mathbf{A}_o - \mathbf{A}_r)\mathbf{A}_x}{\mathbf{A}_a} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$\mathbf{A}_a$ in this case is the value found by equation 9.

From the values of $\mathbf{A}_y$ the ampere turns in the field coil may be found, and by means of the induction curve of the machine the speeds can be ascertained. The curve b in Fig. 320 has been calculated in this way, and it will be noted that it is practically a straight line. In the case of a shunt machine it would be quite straight. Assuming a straight line, the speed for any armature current may be obtained very simply.

In case 2 the speed is—

$$S_r = \frac{S_2 \mathbf{A}_r \mathbf{R}_s}{\mathbf{V}_1}. \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Then the speed S_y for any armature current $\mathbf{A}_x$ is—

$$S_y = S_r - \frac{S_r \mathbf{A}_x}{\mathbf{A}_a} = S_r\left(1 - \frac{\mathbf{A}_x}{\mathbf{A}_a}\right). \quad . \quad . \quad . \quad . \quad (13)$$

$\mathbf{A}_a$ being again the value found by equation 9.

The preceding equations are for determining the proportions of series and parallel resistances when the motor is driving the load.

When the load is driving the motor, as for instance when a load is being lowered with a shunt machine having resistances in series and in parallel with its armature, the following equations are applicable.

When a machine is working in this way either of three conditions may obtain. (1) The voltage at the brushes of the machine may be less than the circuit voltage, in which case the current in the parallel resistance will come partly from the armature and partly from the circuit. (2) The voltage at the brushes of the machine may be equal to the circuit voltage, in which case the current generated by the armature will flow in the parallel resistance, and there will be no current flowing in the circuit. (3) The voltage at the brushes of the machine may exceed the circuit voltage. In this case the current generated by the armature is divided, part flowing through the parallel resistance, and the remainder passing into the circuit.

Case 1.—With a given load on the crane, the current generated by the motor may be ascertained from the reverse efficiency of the hoisting gear, as shown in Table I. (Figs. 13*b*, 13*d*, and Figs. 73*b* and 73*d*).

With a given arrangement of resistances the circuit current and speed of motor may be calculated as follows :—

The current in the circuit—

$$\mathbf{A} = \frac{\mathbf{V} - \mathbf{A}_a\mathbf{R}_s}{\mathbf{R}_e + \mathbf{R}_s}. \qquad (14)$$

The voltage across the brushes of the motor—

$$\mathbf{V}_2 = \mathbf{R}_s(\mathbf{A} + \mathbf{A}_a), \qquad (15)$$

and the speed of the motor—

$$S = \frac{S_2(\mathbf{V}_2 + \mathbf{A}_a\mathbf{R}_a)}{\mathbf{V}_1}. \qquad (16)$$

If, on the other hand, we require to find the amount of parallel resistance in order to obtain a given speed, this may be calculated as follows :—

The voltage required across the brushes will be—

$$\mathbf{V}_2 = \frac{S_s\mathbf{V}_1}{S_2} - \mathbf{R}_a\mathbf{A}_a. \qquad (17)$$

The circuit current will be—

$$\mathbf{A} = \frac{\mathbf{V} - \mathbf{V}_2}{\mathbf{R}_e}, \qquad (18)$$

and the amount of parallel resistance required—

$$\mathbf{R}_s = \frac{\mathbf{V}_2}{\mathbf{A} + \mathbf{A}_a}. \qquad (19)$$

Case 2.—In this case there is no current in the circuit, and $\mathbf{V}_2 = \mathbf{V}$.

The amount of parallel resistance is—

$$\mathbf{R}_s = \frac{\mathbf{V}_2}{\mathbf{A}_a}. \qquad (20)$$

The speed may be found from equation 16.

Case 3.—As the voltage at the brushes in this case exceeds that of the circuit, the circuit current flows in the reverse direction, so that $\mathbf{A}$ has a negative sign.

With a given arrangement of resistances the circuit current and speed of motor are found by the following equations :—

The circuit current is

$$\mathbf{A} = \frac{\mathbf{R}_s\mathbf{A}_a - \mathbf{V}}{\mathbf{R}_s + \mathbf{R}_e}. \quad . \quad . \quad . \quad . \quad . \quad (21)$$

The voltage across the brushes is—

$$\mathbf{V}_2 = \mathbf{R}_s(\mathbf{A}_a - \mathbf{A}), \quad . \quad . \quad . \quad . \quad . \quad (22)$$

and the speed is given by equation 16.

If we require to ascertain the amount of parallel resistance to keep the motor to a given speed, it may be found thus. $\mathbf{V}_2$ being obtained from equation 17, the circuit current will be—

$$\mathbf{A} = \frac{\mathbf{V}_2 - \mathbf{V}}{\mathbf{R}_e}. \quad . \quad . \quad . \quad . \quad . \quad (23)$$

and the amount of parallel resistance—

$$\mathbf{R}_s = \frac{\mathbf{V}_2}{\mathbf{A}_a - \mathbf{A}}. \quad . \quad . \quad . \quad . \quad . \quad (24)$$

The following figures illustrate the effect of using parallel as well as series resistance when the load is driving the motor.

Assuming the same machine as before, but shunt wound, and taking the reverse efficiency of the hoisting gear at 50 per cent., the current generated by the armature will be 50 amperes.

The resistance of the armature is ·114 ohm, and the speed 500 R.P.M. when the E.M.F. generated by the armature is 204·6 volts, the voltage of the circuit being 220.

(1) With the controller full on the speed of the machine will be—

$$\frac{500 \times (220 + 50 \times \cdot 114)}{204 \cdot 6} = 551 \cdot 5 \text{ R.P.M.}$$

(2) With the controller half on, $\mathbf{R}_e$ being 0·6967 and no parallel resistance, the speed will be—

$$\frac{500 \times (220 + 50 \times \cdot 8107)}{204 \cdot 6} = 636 \cdot 5 \text{ R.P.M.}$$

This speed being excessive, a parallel resistance may be used to bring it down. Say it is required not to exceed 575 R.P.M., then the parallel resistance is calculated thus—

$$\mathbf{V}_2 = \frac{575 \times 204 \cdot 6}{500} - 50 \times \cdot 114 = 229 \cdot 6 \text{ volts.}$$

$$\mathbf{A} = \frac{229 \cdot 6 - 220}{\cdot 6967} = 13 \cdot 78 \text{ amperes,}$$

and the amount of parallel resistance—

$$\mathbf{R}_s = \frac{229 \cdot 6}{50 - 13 \cdot 78} = 6 \cdot 339 \text{ ohms.}$$

In this case the current of 50 amperes generated by the motor divides, 36·22 amperes going into the parallel resistance, and 13·78 amperes flowing back into the circuit.

(3) On the first step of the controller a speed of 55 R.P.M. is required, $\mathbf{R}_e$ in this case being 1·461 ohms. It is required to find the value of $\mathbf{R}_s$.

$$\mathbf{V}_2 = \frac{55 \times 204 \cdot 6}{500} - 50 \times \cdot 114 = 16 \cdot 81 \text{ volts.}$$

$$\mathbf{A} = \frac{220 - 16 \cdot 81}{1 \cdot 461} = 139 \cdot 1 \text{ amperes.}$$

and

$$\mathbf{R}_s = \frac{16 \cdot 81}{139 \cdot 1 + 50} = \cdot 0889 \text{ ohm.}$$

If the series resistance were cut out on this step and parallel resistance only employed, as on the first lowering step of Fig. 316, the amount of parallel resistance then required would be—

$$\mathbf{R}_s = \frac{16 \cdot 81}{50} = \cdot 3362 \text{ ohm.}$$

Collectors.—In overhead cranes the trolley wires are stretched along the cross girders, and current is taken from them by trolley wheels or sliders carried by an insulating arm fixed to the crab. Details of a set of collector gear for an overhead crane are given in Fig. 6.

For locomotive jib cranes the current may be taken from a third rail or an overhead trolley wire, and in either case standard electric traction fittings may be employed.

For movable cranes used about docks flexible conductors may be used, junction boxes being provided at suitable points, see Fig. 328.

For calculating the voltage loss, resistance and weight of trolley wires and conductors, the author has found the following equations useful :—

The loss in volts in a conductor is—

$$\mathbf{V} = \frac{\cdot 811 \, L\mathbf{A}}{A} \times 10^{-5}, \quad . \quad . \quad . \quad . \quad (25)$$

L being the total length of conductor in feet, and A its sectional area in square inches.

The resistance of a conductor is—

$$\mathbf{R} = \frac{\cdot 811\ L}{A} \times 10^{-5}, \qquad (26)$$

and the weight in lbs.—

$$W = 3{\cdot}855\ LA. \qquad (27)$$

CHAPTER XVIII.

CRANE INSTALLATIONS.

In planning an installation of cranes, it is first necessary for the engineer to have before him complete particulars of the duty which the installation as a whole is required to perform, in order that he may come to a correct decision as to the various types and sizes of crane which it will be most advantageous to employ.

In handling goods in bulk the general principle to be followed is to convey the load from point to point with as few changes of direction as possible. Thus, for this class of work, the transporter type of crane will for a given consumption of power transfer a greater weight of material in a given time than the jib type of crane, as in the latter there is time lost at every cycle in starting and stopping the slewing motion, and the difference in the working of the two types is still more marked if at every cycle the jib has to be lowered and raised. On the other hand, for dealing with loads of a general and variable character the flexibility of the jib type of crane renders it preferable to the transporter.

In some cases it is not convenient, and in others it is not even possible to convey current to cranes of the locomotive jib type, neither an overhead trolley line, nor a conductor rail laid under or along the ground being suitable. In these cases it has hitherto been customary to use steam cranes, but these have the disadvantage that an hour or so is required every morning for getting up steam, and steam has to be kept up all day, even though an occasional lift only may be required. To overcome the difficulty we may, instead of conveying electricity to the crane, generate current on the crane itself by means of a petrol engine and dynamo.

The petrol engine being always ready to start, the time lost in getting up steam with a steam crane is saved, and the engine can be stopped if the crane is not wanted for a time. By adopting electrical transmission, an amount of flexibility in driving is ensured which would not be obtained if the petrol engine were geared mechanically to the different motions.

Cranes handling heavy weights which require to be set down very gently may be provided with hoisting motions of the electro-hydraulic type. In these the lifting is accomplished by the ram of a hydraulic cylinder, which is supplied with water by a set of pumps driven by an electric motor. The exhaust pipe of the cylinder is led to a tank to which the suction pipe of the pumps is also connected, so that the same water may be used over and over again. The whole arrangement is self-contained, and may be mounted on the crab of an overhead crane, current for the motor being conveyed to the crab by means of trolley wires in the usual way. Overhead cranes working on the electro-hydraulic principle are employed at the Swindon Works of the Great Western Railway for lifting complete locomotives.

Where a number of cranes are working together on one circuit the current taken is very fluctuating. The generating plant from which they take their supply has to be sufficiently large to give the maximum current, and as this considerably exceeds the average the plant cannot be run in the most economical manner.

Fig. 321 is an ampere-meter diagram taken on a circuit containing a large number of cranes and similarly intermittent working machines. The extreme variation of the current in this case is

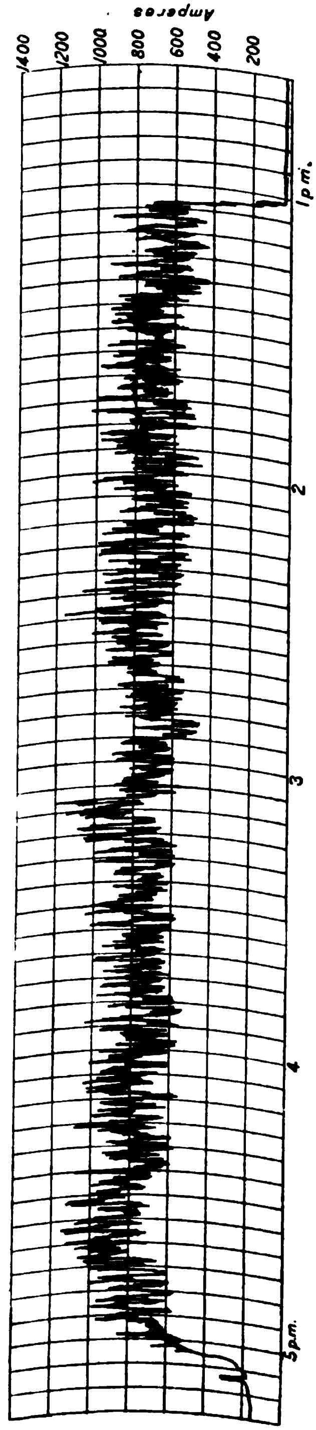


Fig. 321.

from 430 to 1,180 amperes, while the mean current taken over the whole period is 725 amperes, the voltage being 220.* To deal with this load a generating plant having an output of about 1,000 amperes at least would be required, and owing to the fluctuating nature of the load the consumption of fuel, lubricating oil, etc., would be much greater than if a plant could be installed suitable for running continuously at the average load, and designed to give its maximum efficiency when working on this load.

By employing accumulators on the circuit in combination with a reversible booster this may be rendered possible, the generating

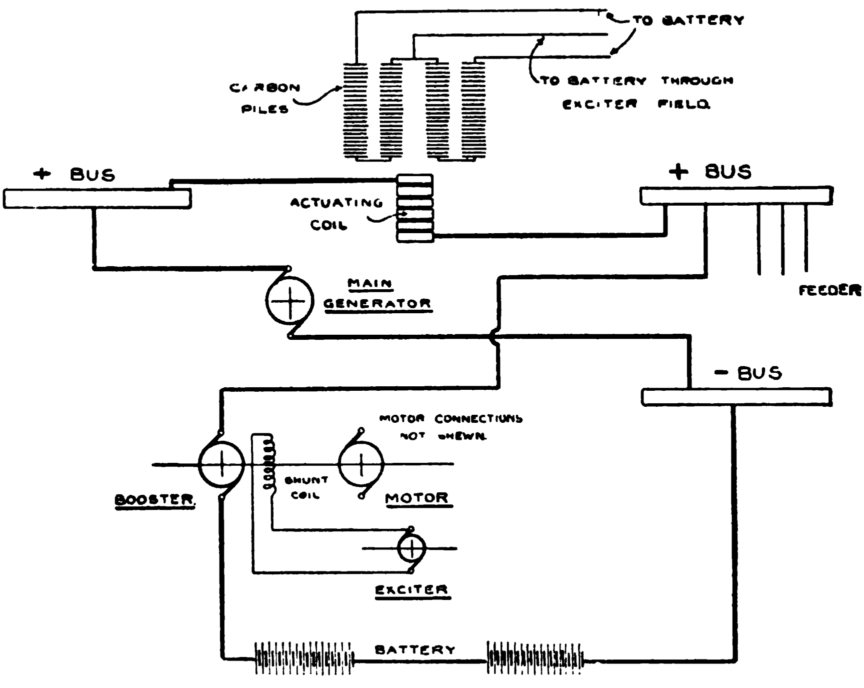


Fig. 322.

plant now giving a steady output, while the fluctuations are dealt with by the accumulators and booster. Thus, when the line current falls below the mean, the difference passes into the

* The total horse-power of motors on this circuit was 837, so that if the whole of the motors were working simultaneously at their full power, a current of about 3,300 amperes would be required. Thus the average working horse-power was 0·22 of the total horse-power installed, and the maximum working horse-power 0·362.

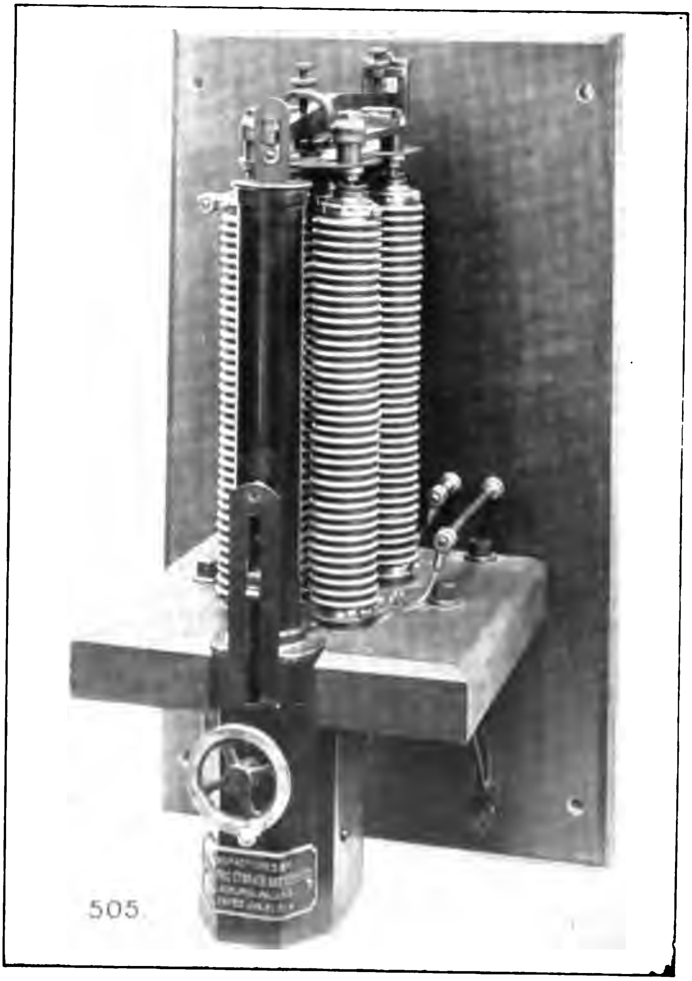


Fig. 323.

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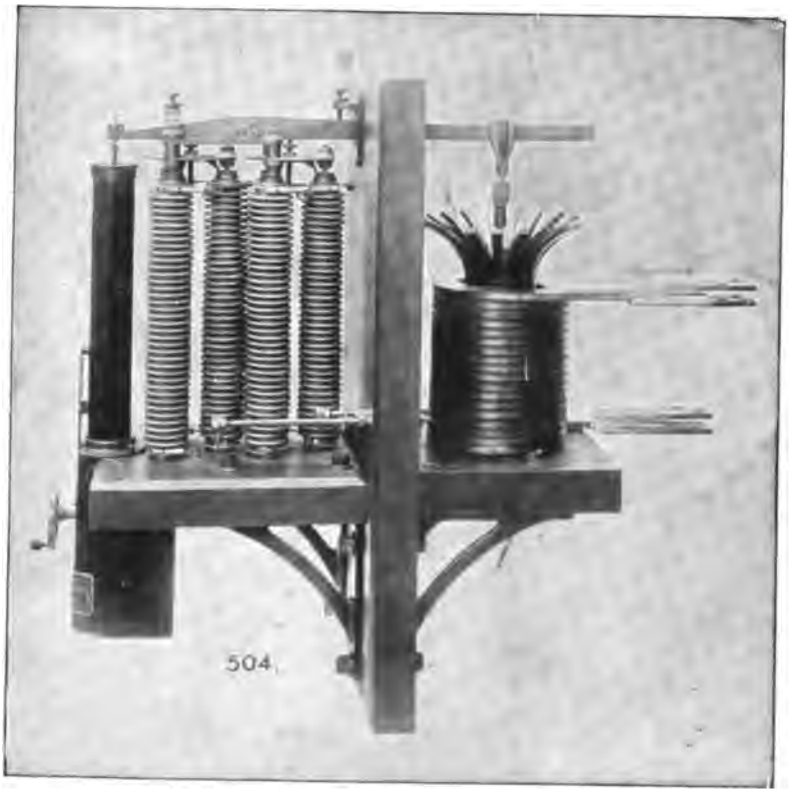


Fig. 324.

To face p. 292.]

accumulators to charge them while, when the line current rises above the mean, the extra current is supplied by the accumulator.

One form of reversible booster which has given very good results is the Entz, made by the Chloride Electrical Company, Ltd. As shown in the diagram of connections (Fig. 322) the booster consists of three machines running together and controlled by a regulator operated by a solenoid coil traversed by the generator current on its way to the external circuit. The machines consist of (1) a separately excited generator, which forms the actual booster; (2) a motor, which drives the two other machines; and (3) an exciter, the field winding of which is connected at one end to the centre of the battery of accumulators, and at the other to the centre of the field regulating resistance, so that its position is analogous to that of the galvanometer in a Wheatstone Bridge. The action of the arrangement is as follows:—When the current taken by the line is at its average value, the two arms of the resistance are equal, and no current flows in the field coil of the exciter; consequently the voltage of the booster is zero, and no current flows into or out of the battery. When the line current exceeds the average, the right-hand arm of the resistance is decreased in value, and this sets up a current in the field coil of the exciter of such direction and magnitude that the exciting current supplied to the booster causes it to generate E.M.F. in the same sense as the battery, so causing the latter to supply the line with an amount of current equal to that by which the line current exceeds the average.

When the line current falls below the average the reverse process takes place, the left-hand arm of the resistance being decreased, owing to the operation of the spring. The booster now generates E.M.F. in opposition to that of the battery, and causes an amount of current to pass into the battery equal to the difference between the line current and the average current.

The construction of the regulator is somewhat interesting, and is illustrated in Figs. 323 and 324.

The regulator consists of two sets of piles of carbon discs, over which is a balanced lever pivoted between the two sets. From one end of this lever is freely suspended the soft iron core of a solenoid, which carries the entire generator load, while to the other end is attached a helical spring, whose tension may be adjusted by hand to counterbalance the pull of the solenoid at any desired load on the machines. Slight variations of load above or below this amount will cause changes in pressure on the carbon piles, resulting in wide variations in their contact resistance.

Reverting to Fig. 321, the average line current is 725 amperes, and the mean variation above and below this figure amounts to 50 amperes with a maximum of 450.

For this a battery having a discharge rate of 375 amperes for three hours would be a suitable size. The booster should be a machine capable of carrying 50 amperes continuously, and of working up to 450 amperes without sparking or mechanical injury, and it should also be capable of generating any voltage from 0 to 66 by variation of its field current. The motor would require to be of sufficient size to drive the booster and exciter under maximum conditions, and the exciter would be a little machine generating current for the field circuit of the booster. In regular work the generator current would require to be equal to the mean line current plus the current lost in the battery plus the current taken by the motor.

The author does not propose to give any comparative figures of first cost or running expenses of this system here, as, although such figures might be correct at the time they were given, they would not be correct a few months later when market prices had altered. It is preferable to take each case as it arises, basing the figures of comparison upon current prices and upon current guarantees as to fuel consumption, rate of maintenance, etc., which manufacturers are prepared to give.

The first cost of combined plant might in some cases exceed that of a simple generating plant, the justification for the increased cost being the saving effected in the running expenses.

As an example of the economy effected by this system, it may be mentioned that its adoption on the Greenock tramways resulted in an immediate drop in the coal consumption from 6¼ lbs. to 5 lbs. per unit generated, and the shutting down of one generating unit.

One of the most important applications of electric crane work is in the loading and unloading of ships. An interesting installation of electric cranes for this purpose is that carried out at the Middlesbrough Docks by Mr. Vincent L. Raven, M.I.C.E., Chief Mechanical Engineer of the North-Eastern Railway, described by him in a paper before the Institution of Mechanical Engineers.*

* "Middlesbrough Dock Electric and Hydraulic Power Plant," V. L. Raven, Proc. Joint Meeting Inst. Mech. E. and Am. Soc. Mech. E., Chicago, June 2nd., 1904.

A general plan of the installation is shown in Fig. 325. There are nineteen 3-ton and five 10-ton cranes, all being jib cranes of the type shown in Fig. 326, and 26 electric capstans, giving a pull of one ton, constructed as shown in Fig. 327, all the cranes and capstans being made by Messrs. Cowans, Sheldon & Co., Ltd., and the electrical equipment by Messrs. Siemens Bros. & Co., Ltd. To supply current for the cranes and for the electric lighting of the docks there is a power station containing three direct-coupled sets, each having an output of 240 kilowatts at 430 volts at a speed of 380 R.P.M. The engines are compound, and on a six hours' test at full load with a steam chest pressure of 128 lbs., and a vacuum of 24 inches, had a steam consumption of 28 lbs. per kilowatt.

Each of the 3-ton cranes on being completed was tested in the following manner :—

A large iron tub (fitted with a valve at the bottom), weighing 1 ton net, was brought close to the quay wall, lowered into the dock, filled through the valve, there being a port in the side of the tub, so that the weight of water could not exceed 2 tons, or a total load of 3 tons. This was lifted 30 feet, and slued through a half-circle simultaneously, then lowered on the quay, where the valve on the bottom of the tank opened, and the water returned into the dock. The tub was again lifted empty, taken back, and the same process continued. Each crane was worked in this manner for three hours, making on an average 100 lifts in the stated time. The current used was :—

Lifting, ...	18·3 B.O.T. units.
Sluing, ...	6·4 ,,

or a total of 24·7 Board of Trade units, or 2·74 Board of Trade units per 1,000 foot-tons, or a total cost of 5·4d. per 1,000 foot-tons.

The 10-ton cranes are of the same general design as the 3-ton, but with two-speed gear to the lifting motion. The revolving motor is 12 B.H.P. at 1,000 R.P.M., and the lifting motor, which is of 60 B.H.P. in single gear, drives directly through a pinion on to the barrel shaft, the ratio of gearing being 8 to 1, and a double part of rope is used.

The double gear gives a multiplying power of 20 to 1. With the single gear loads up to 2 tons are lifted, and the double gear is

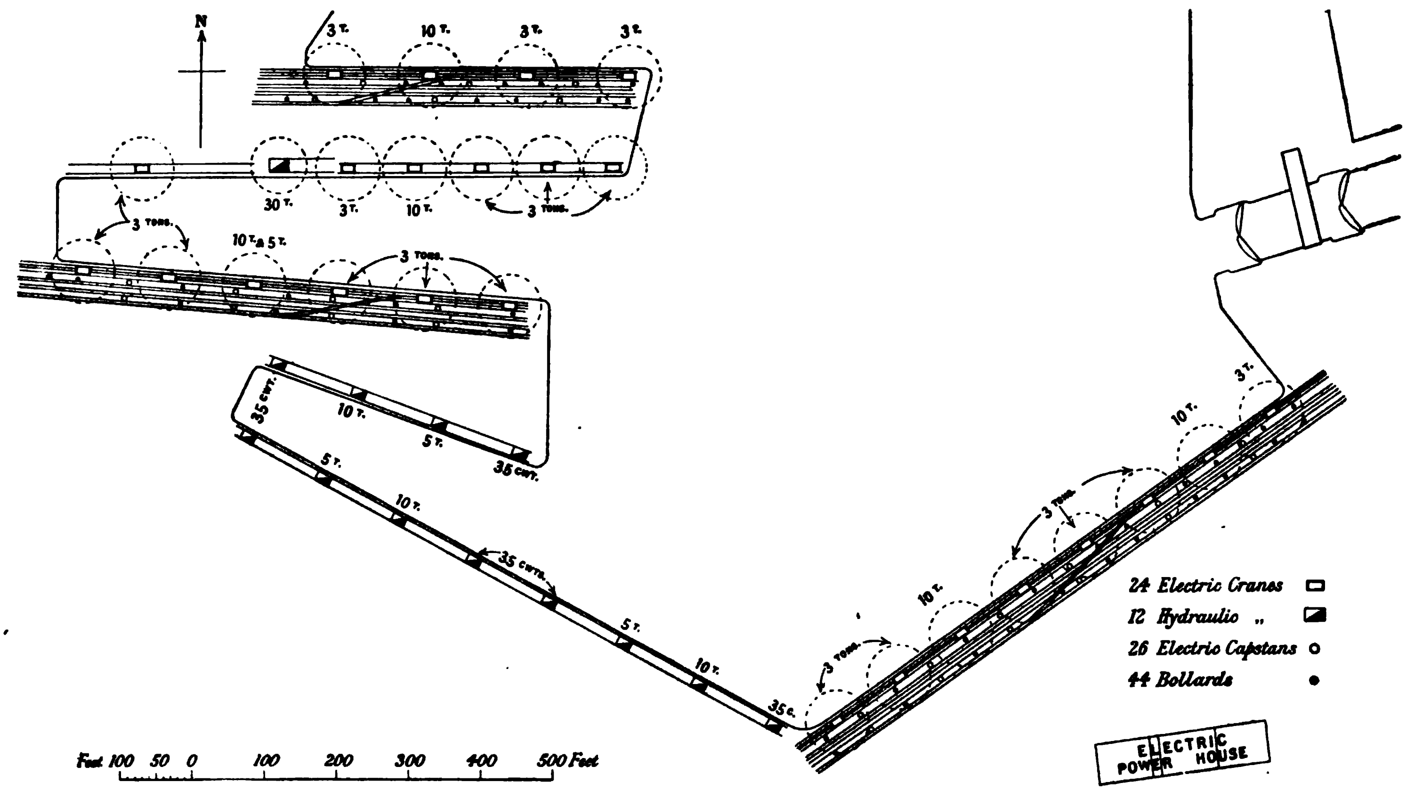
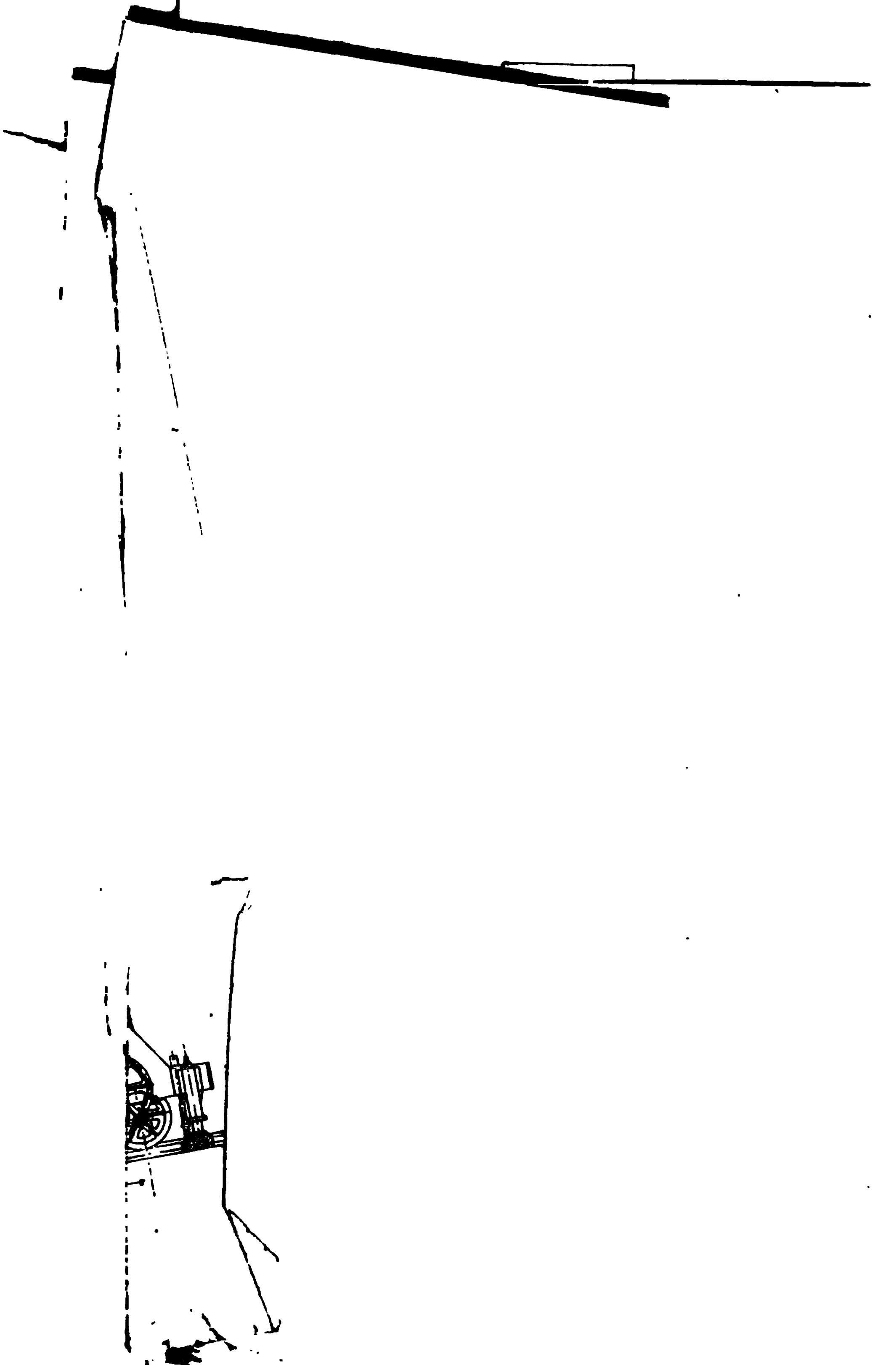
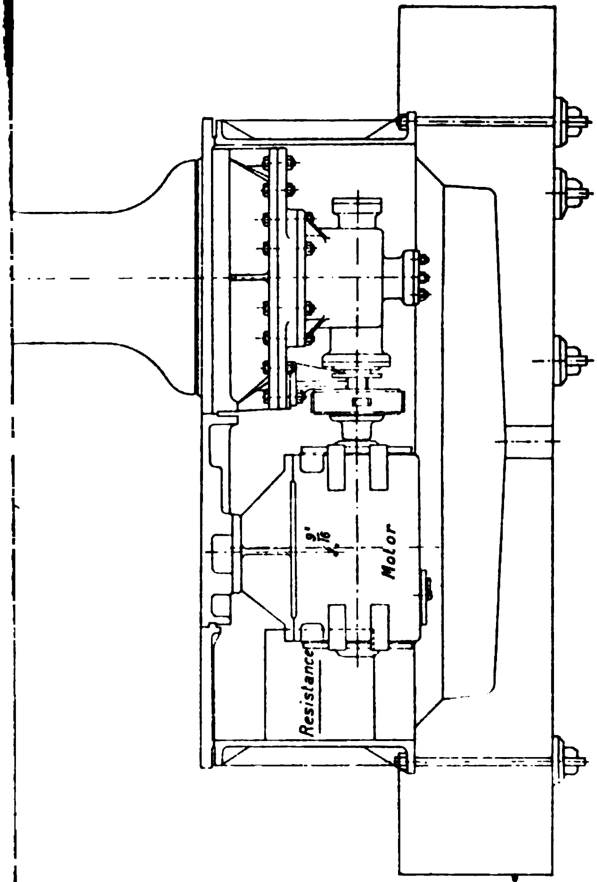


Fig. 325.—Middlesbrough Dock, N.E. Ry.





Letter	Description	Dia	Pitch	Teeth	Width	Bore	Remarks	Ratio
A	*Worm*	$3\frac{1}{2}''$	$4\frac{1}{2}''$	3			*Forged Solid with Shaft*	23·5 To 1
B	*Worm Wheel*	$8\frac{5}{8}''$	$1\frac{1}{2}''$	18	$3\frac{1}{4}''$	$2\frac{1}{2}''$	*Delta Metal Machine Cut Teeth*	
C	*Spur Pinion*	6·21"	$1\frac{1}{2}''$	13	$4\frac{1}{2}''$	$2\frac{1}{4}''$	*Wrot Steel " " "*	
D	*Spur Wheel*	24·35"	$1\frac{1}{2}''$	51	4"	$3\frac{3}{8}''$	*Cast Iron " " "*	

Inches 12 9 6 3 0 1 2 3 4 5 Feet

Scale $\frac{1}{2}$ inch to a Foot.

used for loads from 2 to 10 tons. With the 10-ton cranes the following tests were made :—

The first load was 2 tons, lifted through a height of 20 feet by the single gear, slued through 180°, and then lowered 20 feet ; the light chain hoisted 20 feet, and the crane slued back through 180°. This cycle of evolutions was performed five times. The load was then increased a further 2 tons each time, up to and including 10 tons, and the same number of evolutions made with each load as in the first case, but double gear being used for the lifting. The total current used being 8·1525 B.O.T. units, or 2·717 per 1,000 foot-tons, at a cost of 5·4d. per 1,000 foot-tons, equivalent to 358,710 foot-lbs. of energy per minute for the total work.

The electric capstans (Fig. 327) are 24 B.H.P. at 1,000 revolutions per minute, and are capable of exerting a steady pull of 1 ton at a speed of 200 feet per minute, or of hauling a load of 100 tons along a level road. The capstan-head is driven by a cast-iron spur-wheel, which is engaged with a steel pinion. The latter is keyed on the same shaft as a brass worm-wheel. The driving worm is cut out from a blank forged solid with its shaft, and is coupled direct to the motor spindle. The worm and wheel work in an oil bath. The motor runs at 1,000 revs. per minute, and is quite enclosed.

In construction and design it is similar to the crane motors, but is shunt-wound to avoid large variations in speed of the capstan head. Upon the motor shaft there is fixed an automatic mechanical brake, the principle of which provides that when the capstan-head is driven by the motor the brake releases itself automatically, but should the capstan tend to run back through being overhauled by the weight, the brake at once locks itself, and sustains the load. Such a brake is absolutely necessary upon an electric capstan, as the latter differs from a hydraulic capstan in one important respect —namely, that in the hydraulic capstan the water in the cylinders holds the load, whereas in the electric capstan there is nothing to sustain the load upon the current being cut off, and if the load is allowed to run back a serious accident may ensue. It is also essential that the brake should be automatic, so as to claim none of the driver's attention from his work.

The electrical switch gear consists of a controller with magnetic blow-out, and with overload release gear. The controller is worked by a pedal projecting above the capstan case by about 4 inches. This pedal is removed when the capstan is out of use. The pedal is connected to a dash-pot, which prevents the controller being

TABLE XVII.—ELECTRIC CAPSTAN TESTS, MIDDLESBROUGH DOCK, 19TH AND 20TH AUGUST, 1903.

Weight of Load.	Distance Hauled.	Time Occupied.	Speed per Minute.	Power Used in Units.	Power Required Theoretically, taking Tractional Force at 23 Lbs. per Ton.	Actual Power Used by Machines.	Efficiency of Machine.	Cost of Power.	Cost of 1,000 Foot-tons of Work.
Tons.	Feet.	Secs.	Feet.			Foot-lbs.	Per cent.	Pence.	Pence.
174	100	47½	126	0·51	400,200	1,354,764	30		
145	100	40	150	0·51	333,500	1,354,764	25		
116	100	37	162	0·31	266,800	823,484	32	4·1467	6·63
87	100	35	171	0·31	200,100	823,484	24		
58	100	33½	179	0·26	133,400	690,664	19		
29	100	33	182	0·21	66,700	557,844	12		

Power used by electric capstan running light for ten minutes = 0·5 unit or 1,328,200 foot-lbs. Value 0·985 penny.

Note.—These tests were made with ordinary dock wagons in bad weather. The wagons had the usual grease-boxes to axles. Lines very badly laid, but level.

operated too rapidly when a driver is starting the capstan, but at the same time, by means of valves in the plunger, allows the controller to return rapidly to the off position. The return to the off position is effected by means of a weight, which is lifted as the pedal is depressed. In the event of the capstan being pulled up by sudden overload, the release gear works instantaneously, and breaks the main circuit. Upon the pressure being removed from the pedal, the return of the controller to the off position automatically replaces the release switch, and the capstan is ready to start again. The whole of the gear is contained in a water-tight cast-iron case sunk in the ground, the top of which consists of chequered plates flush with the quay side. The trials of these capstans are shown on Table XVII.

The distribution of current is effected by a network of feeders and distribution cables laid below ground. These cables are fibre-insulated and lead-sheathed, they are laid in wooden troughs on the solid system, and are at a depth of about 2 feet from the surface. Water-tight junction boxes are provided at the junctions of the feeders.

The crane connection boxes made by Messrs. Siemens Bros. & Co., Ltd., are in two portions, the bottom of which contains the positive and negative distributing cables, and the top the connection apparatus for the cranes. The connection socket consists of a circular gun-metal hood, which accurately fills the aperture in the cover of the box, and is so shaped as to shield the joint from wet. The interior of this hood contains two gun-metal tongues, which meet the two clip contacts in the interior of the box. A stud and groove ensure that the socket shall enter the aperture only in the correct position. The flexible crane cable enters the hood of the socket through a trumpet mouth, and the whole apparatus is shaped so as to obstruct the quay-side as little as possible. One of these connection boxes is shown in Fig. 328.

In addition to the tests already mentioned, a further test was made with loads of rails worked as if they had been ordinary cargo into a ship. They were lifted to a height of 20 feet, slued through a distance of 103·5 feet, lowered down to empty waggons, the light chain lifted a height of 20 feet, slued through 103·5 feet, and lowered down again. The total weight dealt with was 1224·9 tons, the total B.O.T. units used being 174. The units used per 1,000 foot-tons of work were 7·1, and the cost 14·3d.

In addition to the electric cranes there are twelve hydraulic.

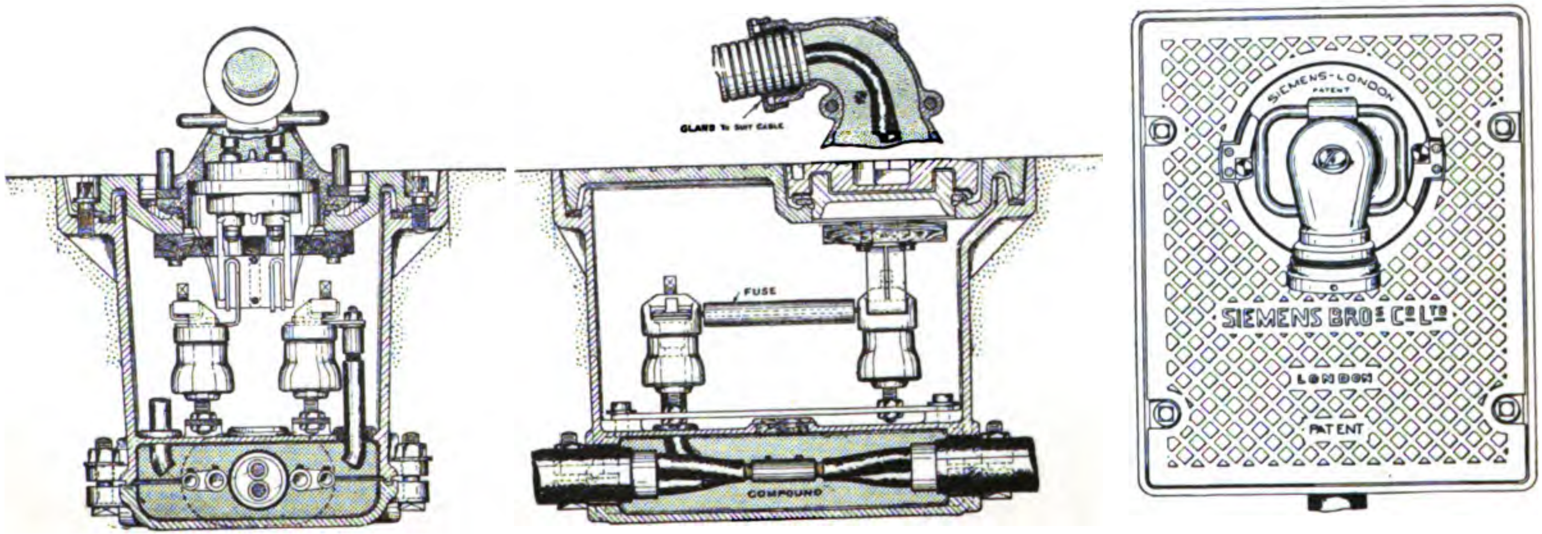

GLAND TO SUIT CABLE
FUSE
COMPOUND
SIEMENS-LONDON
PATENT
SIEMENS BROS Co Ltd
LONDON
PATENT

Fig. 328.

In 1903 the quantity of traffic dealt with at this dock was :—

Coal and coke exported, . .	259,746 tons.
Merchandise exported, . .	297,304 ,,
Merchandise imported, . .	33,696 ,,
Total, . .	590,746 ,,

To do this work, the quantity of coal burned at the power station, which supplies both the electric power, lighting, and hydraulic power water, was 3,428 tons 4 cwts., or the total merchandise worked per ton of coal burned was 172·3 tons.

At the works of Sir W. Armstrong, Whitworth & Co., in Manchester, there is an installation of overhead cranes, for serving the heavy tools, the working cost of which has been very carefully ascertained, and communicated by Mr. R. Matthews, M.I.C.E., in a paper to the Manchester Association of Engineers.

The shop in which these cranes are installed consists of seven bays of 50 feet span, 225 feet long, the length covered by machines being about 200 feet. Nos. 1, 2, and 7 bays have 70-ton cranes; Nos. 3 and 4, 30-ton cranes, and Nos. 5 and 6, 60-ton cranes. A careful record of the work of these cranes extending over a period of ten days was made, and from this record an estimate of the annual cost was prepared as shown in Table XVIII. It will be noted that in this table oil and drivers' wages are omitted, the reason being that the figures were originally obtained for comparison with the cost of rope driven cranes, and as these items would be the same in both cases, it was not necessary to include them.

The figures in the table are chiefly of interest as showing that in a heavy machine shop, the tools in which principally consist of heavy shafting, gun boring, and turning lathes, in which the jobs remain in the lathes for long periods, there is a great amount of time during which the cranes are idle, and consequently their first cost should receive due consideration. All the cranes are of the multiple motor type.

Inspection of Table XVIII. shows that the cost of the electricity taken by the cranes was quite an insignificant item in comparison with the annual charges on the first cost.

For work of this class, therefore, cranes of low first cost, provided they were reliable and quick, would be preferable, even

TABLE XVIII.—ANNUAL COST OF ELECTRIC CRANES.

Number of Bay.	1	2	3	4	5	6	7
Power of crane,	70 tons	70 tons	30 tons	30 tons	50 tons	50 tons	70 tons
Annual number of services,	1,710	1,995	1,184	627	456	598	655
Feet lifted per annum,	17,100	19,950	11,840	6,270	4,560	5,980	6,550
,, traversed per annum,	68,400	79,800	47,360	25,080	18,240	23,940	26,220
,, travelled ,,	136,800	159,600	94,720	50,160	36,480	47,880	52,440
Equivalent in hours worked,	61·45	72·3	50·5	26·725	21·2	27·85	23
Interest on capital,	£100 0 0	£100 0 0	£60 0 0	£60 0 0	£90 0 0	£90 0 0	£100 0 0
Depreciation,	100 0 0	100 0 0	60 0 0	60 0 0	90 0 0	90 0 0	100 0 0
Cost of repairs,	50 0 0	50 0 0	30 0 0	30 0 0	45 0 0	45 0 0	50 0 0
Cost of power,	2 7 0	3 3 0	2 5 0	1 4 7	0 12 4	0 18 1	0 18 0
Annual charge,	£252 7 0	£253 3 0	£152 5 0	£151 4 7	£225 12 4	£225 18 1	£250 18 0

Electricity charged at 1·5d. per unit.

though they might have a slightly lower efficiency, and consequently higher consumption of electricity. For instance, instead of multiple motor cranes, cranes having a single motor might be employed, in which the different motions are put in and out of gear by means of clutches. The clutches might be of the Weston disc type working in oil, and operated by electro-magnets, or they might be of the electric-induction type. In either case the controlling switches for the clutches could be operated by cords from the floor, so that no crane driver would be required. An additional control to the travelling motion might be provided, similar in principle to that of a push button lift, with a push button at each machine, so that when the machineman wanted the crane he would press the button and the crane, if not engaged, would come to him.

An interesting application of electric crane work was that employed in the construction of the King Edward VII. Bridge at Newcastle-on-Tyne for the North-Eastern Railway. The bridge was designed by Dr. C. A. Harrison, M.I.C.E., Chief Engineer of the North-Eastern Railway, and was built by the Cleveland Bridge Company, Ltd., one of their Directors—Mr. F. W. Davies, A.M.I.C.E.—taking charge of the work. The author acted as Consulting Engineer to the Contractors. Most of the machinery employed in the construction of the bridge was driven electrically, and for the supply of the necessary current a generating station was built containing two direct-coupled sets, each giving an output of 750 amperes at 240 volts and 400 R.P.M. for the power circuit, and three small sets for lighting. The aggregate H.P. of the motors employed was about 800.*

In sinking each caisson for the pier foundations three Derrick cranes were used. The hoisting motion of these cranes was fitted with two-speed gear. On the slow speed they were capable of hoisting 5 tons at 40 feet per minute, and were used for lifting the air-locks, girders, and other heavy parts into place; while on the quick speed, of 80 feet per minute, they were used to hoist the spoil from the excavating chamber at the bottom of the caisson. One of these cranes is described in detail in a previous chapter, and is shown in Fig. 28. In addition to these cranes there was in each lock a small winch for lifting the air-lock door into place, the construction of the winch being similar in principle to that shown

* The King Edward VII. Bridge, Newcastle-on-Tyne. F. W. Davis and C. R. S. Kirkpatrick, *Proc. Inst. C.E.*, vol. clxxiv., p. 158; also *Electrical Review*, August 10, 1906.

in Fig. 245. The general arrangement of the three cranes is shown in the sectional drawing of the caisson (Fig. 329), and the photograph (Fig. 330), while Fig. 331 is a recording ampere-meter diagram showing the current taken by the three cranes when they were all at work hoisting up spoil from the river bed, each load of about 30 cwts. including the bucket being hoisted up clear of the staging, and slued through half a circle to discharge into a barge alongside.

For setting the masonry on the piers several 5-ton derrick cranes with 70 feet jibs were employed, as shown in the photograph (Fig. 332). In order to obtain good control in setting the blocks, the hoisting motors of these cranes were shunt-wound, and the controllers were provided with slow-speed steps, as shown in the connection diagram (Fig. 333). For conveying materials and machinery across the river the cableway shown in the frontispiece and in Fig. 64 was employed. The cableway was at work about two years, and proved successful in every way. It handled about 23,625 tons of material, and was constantly employed in dismantling, removing, and building 7-ton steam travelling cranes, fixing centres, transporting ashlar, lifting and setting granite (where too heavy for the 5-ton cranes to deal with), and finally for putting up the river staging on which the girders were built. During this period none of the ropes required renewal, and the rims of the travelling wheels of the carriage were renewed once. For constructing the girders two Goliath cranes of the type shown in Figs. 186 and 334 were employed running on staging built up from the river.

At the Coventry Ordnance Works, which is one of the most recently built machine shops in this country, there is an installation of 54 electric overhead cranes, ranging in size from 5 tons to 100 tons. A list of the sizes is given in Table XIX., which is interesting as showing present practice with regard to the powers and speeds of the different motions.

It will be noted that some of the cranes are English and others German. In the German cranes the cross girders are latticed, of the type shown in Fig. 184, and the hoisting motion is provided with rheostatic braking, while most of the English cranes have plate girders and automatic mechanical brakes, as shown in Fig. 246. In the majority of the cranes the controllers for the traversing and travelling motions are coupled together, and worked with a universal handle.

An interesting example of steel structural work for carrying overhead cranes is afforded by the big gun shop at these works.

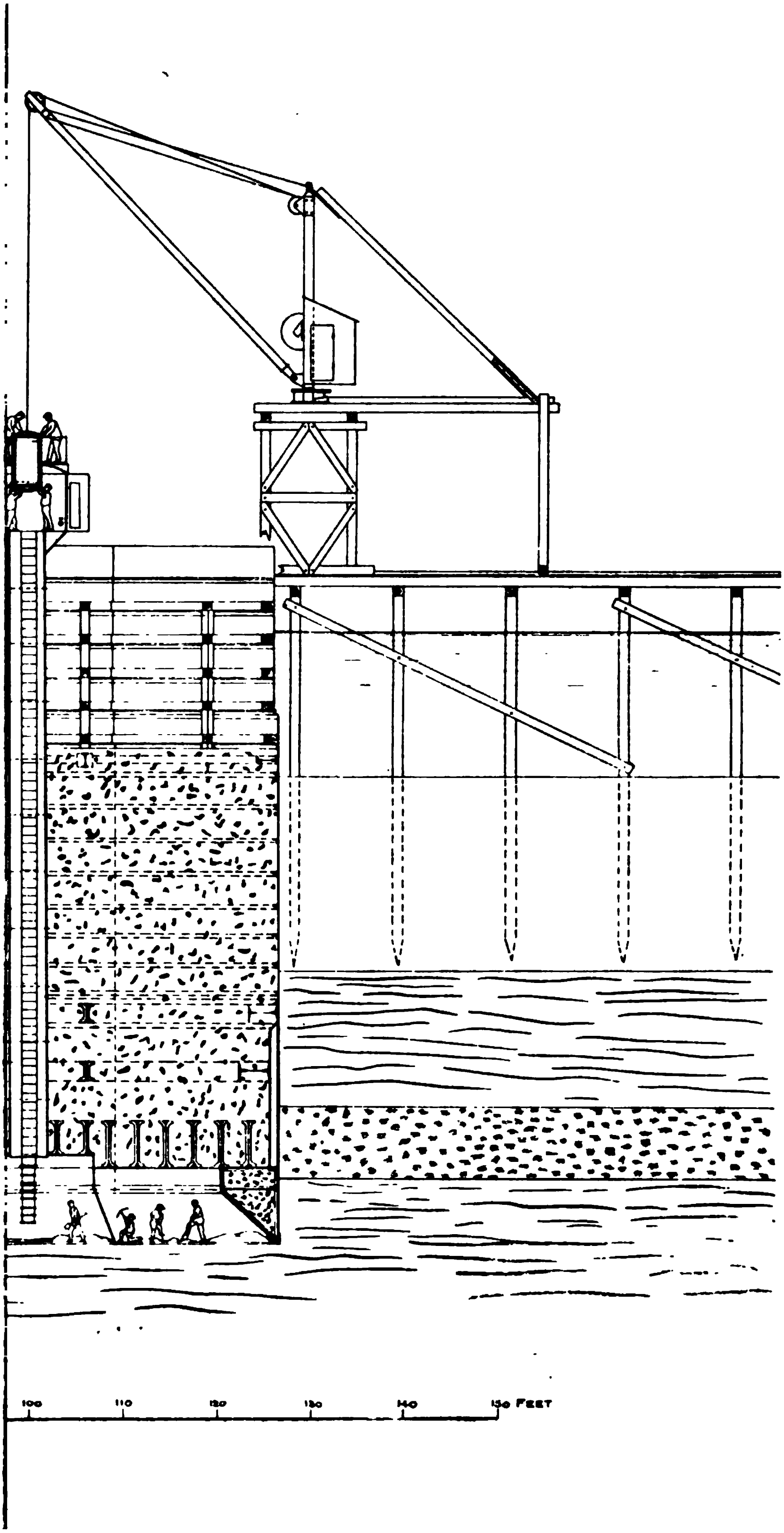
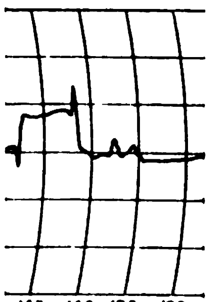

100
110
120
130
140
150 FEET

Fig. 330.—Derrick Cranes on Caisson.

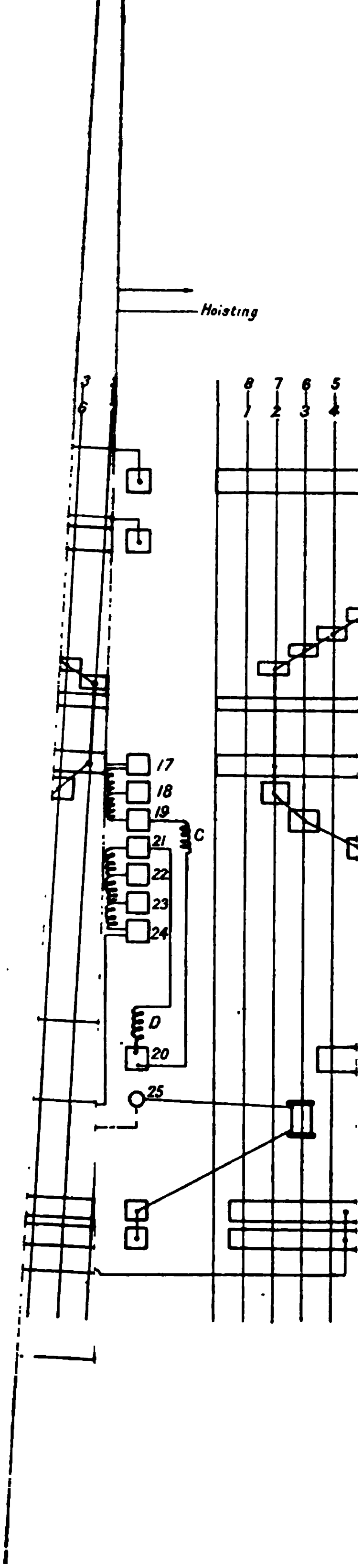
To face p. 304.]



'ent taken by Three

Fig. 332.—Derrick Cranes Erecting Piers.

To face p. 304.]



Hoisting
3
6
8
1
7
2
6
3
5
4
17
18
19
C
21
22
23
24
D
20
25

Fig. 334.—15-Ton Goliath Crane.

To face p. 304.]

un Shop.

This shop is 980 feet long, and about 200 feet wide, and is one of the largest machine shops in the world. A cross-section, which shows the arrangement of the steel work carrying the cranes and roof is given in Fig. 335. Longitudinally the columns are spaced

TABLE XIX.—List of Sizes of Electric Overhead Cranes at the Coventry Ordnance Works.

CRANES WITH DIRECT-CURRENT MOTORS ON 400-VOLT CIRCUIT.

Maker.	Capacity.	Hoist.	Long Travel.	Cross Travel.
Niles Tool Co., . .	5 tons	20 ft. P.M. 10 H.P.	325 ft. P.M. 7 H.P.	150 ft. P.M. 1½ H.P.
Benrather, . . .	5 tons	5 ft. P.M. 3 H.P.	150 ft. P.M. 3 H.P.	50 ft. P.M. 1 H.P.
Stuckenholz, . .	10 tons	13 ft. P.M. 15·5 H.P.	300 ft. P.M. 15·5 H.P.	66 ft. P.M. 3 H.P.
Broadbent, . . .	15 tons	6 ft. P.M. 10 H.P.	150 ft. P.M. 8 H.P.	60 ft. P.M. 4 H.P.
Adamson, . . .	15 tons	6 ft. P.M. 10 H.P.	150 ft. P.M. 8 H.P.	60 ft. P.M. 4 H.P.
Stuckenholz, . .	40 tons	5 ft. P.M. 28 H.P.	180 ft. P.M. 28 H.P.	66 ft. P.M. 9·5 H.P.
	Aux. 10 tons	13 ft. P.M. 15·5 H.P.		
Ransomes & Rapier, .	40 tons	6 ft. P.M. 25 H.P.	180 ft. P.M. 25 H.P.	75 ft. P.M. 12 H.P.
	Aux. 8 tons	16 ft. P.M. 12 H.P.		
Ransomes & Rapier, .	75 tons	5 ft. P.M. 40 H.P.	130 ft. P.M. 40 H.P.	70 ft. P.M. 25 H.P.
	Aux. 10 tons	16 ft. P.M. 25 H.P.		
Krupp, . . .	80 tons	40 ft. P.M. 170 H.P.	90 ft. P.M. 24 H.P.	45 ft. P.M. 10 H.P.

CRANES WITH ALTERNATING-CURRENT MOTORS (3-PHASE). 400 VOLTS. 25 PERIODS. ALL SLIP-RING MOTORS.

Maker.	Capacity.	Hoist.	Long Travel.	Cross Travel.
Stuckenholz, . .	10 tons	19 ft. P.M. 20 H.P.	210 ft. P.M. 20 H.P.	48 ft. P.M. 5 H.P.
Cowans & Sheldon, .	30 tons	16 ft. P.M. 45 H.P.	120 ft. P.M. 20 H.P.	48 ft. P.M. 6 H.P.
Ransomes & Rapier, .	60 tons	30 ft. P.M. 60 H.P.	120 ft. P.M. 45 H.P.	63 ft. P.M. 15 H.P.
	Aux. 10 tons	22 ft. P.M. 25 H.P.		
Stuckenholz, . .	100 tons	12 ft. P.M. 50 H.P.	90 ft. P.M. 50 H.P.	34 ft. P.M. 20 H.P.

20 feet centre to centre. In the left-hand bay there are two 40-ton cranes, in the centre bay one 75-ton and two 10-ton, and in the right-hand one 40-ton and one 10-ton.

The 75-ton crane has a total weight of 66 tons. It has four travelling wheels in each end carriage, so that with full load at the centre of the span the load per wheel is 17·6 tons. The wheel base is 14 feet 3½ inches. With full load and the crab traversed to one side, the load per wheel on that side is about 26½ tons.

The weight of the 40-ton crane is 37 tons. There are two travelling wheels in each carriage, so that with full load at the centre the load per wheel is 19¼ tons. The wheel base is 12 feet 9 inches. With full load and the crab traversed to one side, the load per wheel is about 28½ tons.

A general view of the central bay is shown in Fig. 336, which is reproduced from an illustration given in the *Engineer* in June, 1907, in an article giving a very full description of the Coventry Ordnance Works. At the time this photograph was taken the shop had not received its full complement of tools.

Fig. 337 shows the structural steel work to carry the cranes and roof over the gun pit. The cranes are carried in two tiers, the upper crane being of 80 tons capacity, and the lower one 30 tons.

The 80-ton crane is specially constructed for dipping guns in the oil bath. Full load of 80 tons is hoisted at a speed of 40 feet per minute, and 40 tons at 90 feet per minute. The latter load is lowered for oil dipping at a speed of 250 feet per minute, the load at this speed being controlled by automatic brakes running in water to keep cool. The last two gears in this crane are run in an oil bath.

The weight of the 80-ton crane is 88 tons, and there are two travelling wheels in each end carriage, the wheel base being 19 feet. With full load at the centre, the load per wheel is 42 tons, while with the crab traversed to one side it is 62 tons.

The dimensions of those portions of the steel work which carry the cranes are marked on the drawing. The whole of the buildings were erected by the Coventry Ordnance Works Company's own staff.

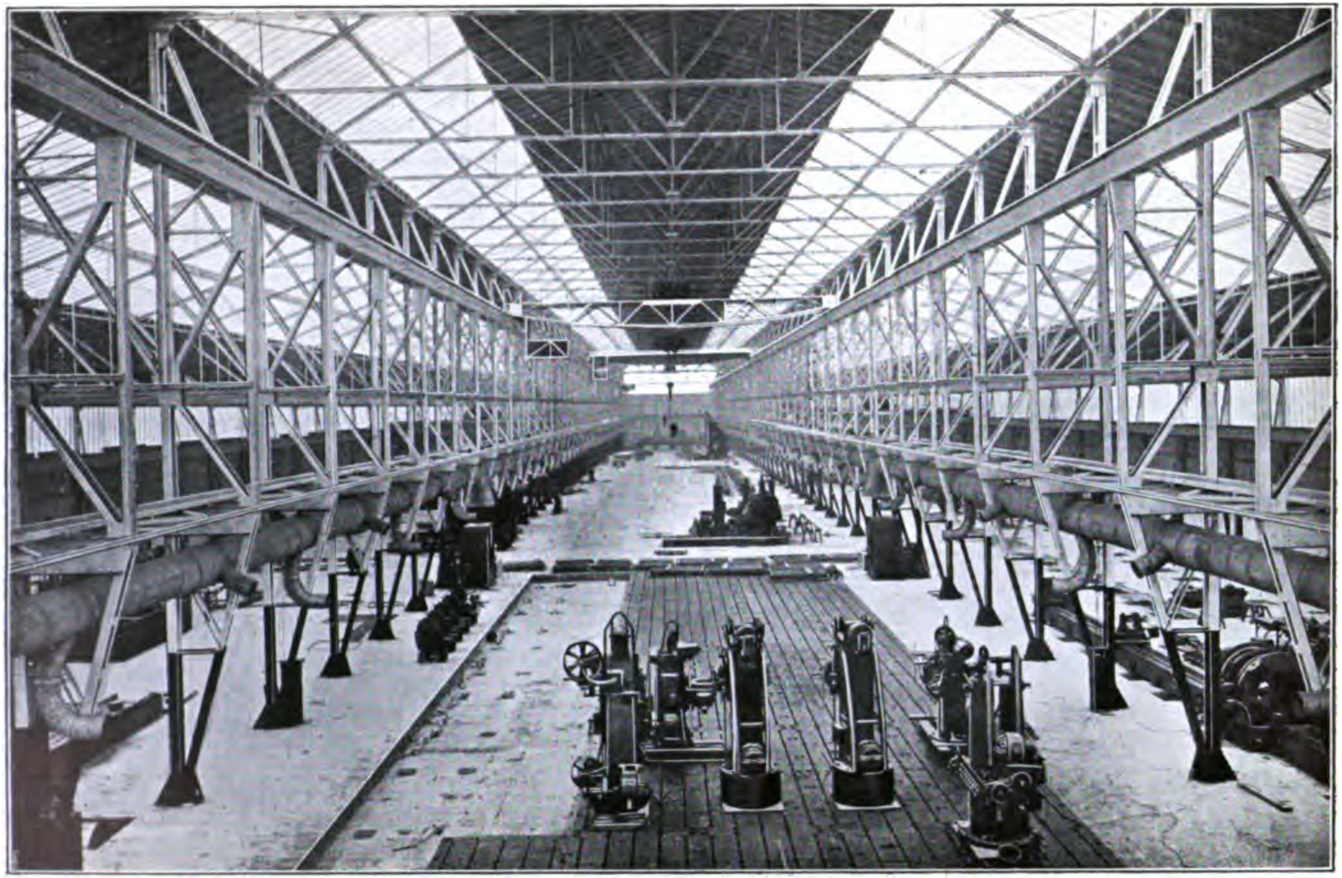


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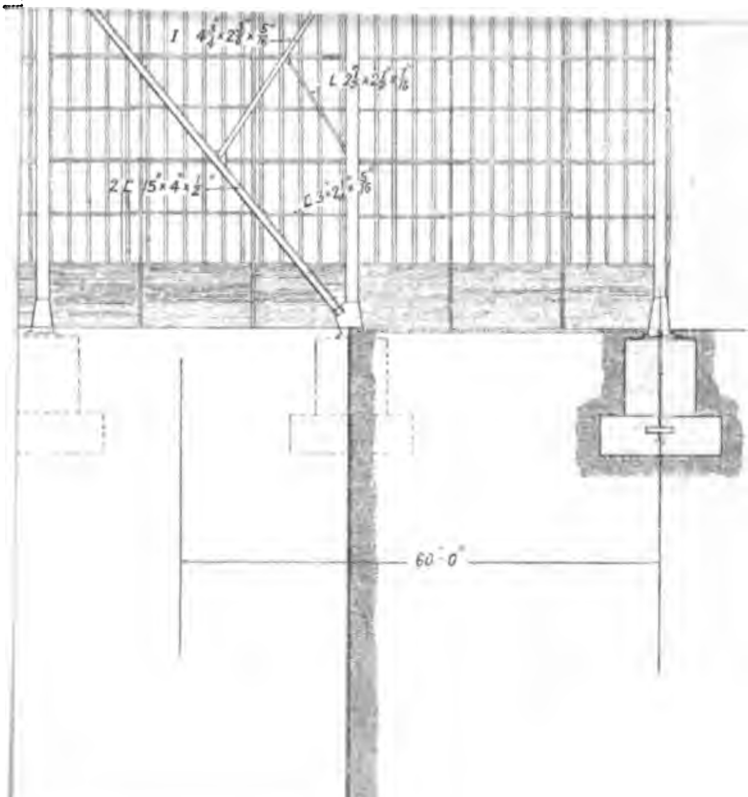
Fig. 336.—Coventry Ordnance Works, Central Bay of Big Gun Shop.


60'-0"

TABLES

OF

APPROXIMATE BREAKING LOADS OF BRITISH STANDARD STEEL SECTIONS WHEN EMPLOYED AS STRUTS.*

* For dimensions of these sections see Book No. 6, *British Standard Sections*, issued by the Engineering Standards Committee.

In these tables the breaking loads given are calculated on the basis of the Rankine formula No. 4 in Chapter X.

The sectional area being taken into account the formula becomes—

$$W_n = \frac{A f_c}{1 + c\left(\frac{l}{\kappa}\right)^2}.$$

f_c is taken at an average value of 30 tons per square inch. The breaking load of struts made of other steel will be approximately proportional to the ultimate strength in compression of the steel. Thus the breaking load of struts made of a 35-ton steel will be greater than that given in the table in the proportion of $\frac{3{\cdot}5}{3{\cdot}0}$ approximately. The value of c being $\frac{f_c}{\pi^2 E}$ is taken at $\frac{30}{\pi^2\,13{,}390} = \frac{1}{4{,}405}$.

The virtual length, l, of the struts is given instead of the actual length, so that the tables are applicable to struts of either of the forms shown in Figs. 86 to 89.

These tables are intended as a guide to the designer to assist him in finding suitable sections when commencing a design. Before finally deciding the dimensions of a strut the thickness of its material in relation to its radius of gyration should be considered, and a reference made to the curves in Fig. 95.

uts.

Size.	Column.			
	5′ 0″	5′ 6″	6′ 0″	7′ 0″
B.S.E.A.,	.. ..	.. ..	.. ..	.. ..
B.S.E.A.,	.. ..	.. ..	.. ..	.. ..
B.S.E.A.,	.. ..	.. ..	.. ..	.. ..
B.S.E.A.,	.. ..	.. ..	.. ..	.. ..
B.S.E.A.,	3·12 5·17	.. ..	.. ..	[illegible]
B.S.E.A.,	4·21 7·0	3·7 6·15	.. ..	[illegible]
B.S.E.A.,	[illegible]·00 [illegible]·6 [illegible]·0	6·85 10·0 12·8	5·95 8·65 11·2	[illegible]
B.S.E.A.,	[illegible]·2 [illegible]·9 [illegible]·5	8·85 13·0 16·8	7·7 11·3 14·6	[illegible]
B.S.E.A.,	[illegible]·8 [illegible]·8 [illegible]·4	11·2 16·4 21·4	10·0 14·6 19·0	[illegible]
B.S.E.A.,	[illegible]	19·5 27·1 31·5	17·3 24·0 28·0	[illegible]
B.S.E.A.,	[illegible]	26·6 37·0 43·0	23·8 33·2 38·5	[illegible]
B.S.E.A.,	[illegible]	42·8 56·5	38·5 50·8	[illegible]
B.S.E.,		53·8 71·0	49·0 64·5	[illegible]
B.S.E.		92·5 126·0	85·0 116·0	[illegible]
B.S.E.		133·0 177·0	124·0 165·0	[illegible]
B.S.E.		.. ..	173·0 234·0	[illegible]

used as Struts.

Length of Column.							
′ 0″	9′ 0″	10′ 0″	11′ 0″	12′ 0″	13′ 0″	14′ 0″	15′ 0″
..	..	..	..	..	..	..	..
8·75	..	..	..	..	..	..	..
0·3	..	..	..	..	..	..	..
0·2	16·7	13·9	..	..	..	..	..
1·5	17·7	14·8	..	..	..	..	..
6·1	30·3	25·7	22·1	19·0	..	..	..
9·7	33·3	28·2	24·1	20·8	..	..	..
5·2	47·0	40·3	34·9	30·4	26·6	..	..
2·1	35·4	29·8	25·5	22·05	..	..	..
1·3	52·2	45·7	38·6	33·6	30·0	..	..
6·3	21·7	..	..	..	..	..	..
5·5	38·0	32·2	27·6	23·6	..	..	..
7·2	57·1	48·8	42·2	36·7	32·3	..	..
2·2	80·0	69·1	60·3	52·8	46·7	41·5	36·9
3·2	36·0	31·0	25·9	..	..	..	..
6·3	56·3	48·1	41·7	36·2	31·7	..	..
4·0	62·8	53·7	46·4	40·2	35·2	..	..
0·4	86·6	75·0	65·2	57·2	50·5	45·5	40·0
9·0	58·6	50·0	43·2	37·5	32·8	..	..
2·0	95·0	81·0	70·0	60·6	52·8	..	..
4·8	90·2	77·9	68·0	59·4	53·0	46·5	41·4
2·8	70·0	59·8	51·4	44·6	38·7	..	..
3·1	96·5	84·0	72·9	63·8	56·2	49·8	44·3
3·8	62·5	53·3	45·9	39·8	34·9	..	..
3·7	74·8	63·7	54·7	47·4	..	..	..
2·0	104·5	90·2	79·0	68·6	60·3	53·3	47·4
3·9	114·2	98·4	85·1	74·4	65·3	57·7	..

15′ 0″	16′ 0″	17′ 0″	18′ 0″	19′ 0″	20′ 0″	21′ 0″	22′ 0″	23′ 0″	24′ 0″
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59·2	53·3	48·2	..	..	..	..	..	..	..
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138·1	125·2	114·3	104·5	95·8	88·0	81·2	..	..	..
..	..	..	..	..	..	..	..	..	..
4·7	67·3	60·9	..	..	..	..	..	..	..
·3	181·6	166·3	152·7	140·5	130·0	120·0	111·6	103·7	96·4
	..	..	..	..	..	..	..	..	..
	66·2	59·8	..	..	..	..	..	..	..
5	83·6	75·5	..	..	..	..	..	..	..
·2	65·2	..	..	..	..	..	..	..	..
·8	83·4	75·4	..	..	..	..	..	..	..
..	..	..	..	..	..	..	..	..	..
5·4	4·8	..	..	..	..	..	..	..	..
02·9	·4	..	..	..	..	..	..	..	..
64·3	8	121·0	110·2	..	..	..	..	..	..
213·3		158·8	144·8	132·4	121·7	..	..	..	..
231·0		171·2	156·0	142·3	..	..	..	..	..

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18·7	16·3	14·4	12·8	..	..	..	..	..
24·5	21·4	18·9	16·8	..	..	..	..	..
25·3	22·3	19·7	..	..	..	..	..	..
25·6	22·5	19·8	..	..	..	..	..	..
28·7	25·6	22·9	20·6	18·5	16·8	15·2	..	..
37·7	33·5	30·0	27·0	24·3	22·0	20·0	..	..
30·6	35·0	31·5	28·2	25·4	23·0	..	..	..
55·2	49·8	45·0	40·7	37·0	33·7	30·8	28·3	26·0

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