OPERATION OF CAMS FOR THE VALVE - GEAR OF A WESTINGHOUSE GAS ENGINE

BY

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ARMOUR INSTITUTE OF TECHNOLOGY

1915

25

DESIGN, INSTALLATION AND OPERATION OF CAMS FOR THE VALVE-GEAR OF A THREE-CYLINDER, 8 x 10 IN. WESTINGHOUSE GAS ENGINE

A THESIS

PRESENTED BY

WILLIAM PATERSON
LUDWIG W. A. BUNGE

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

HAVING COMPLETED THE PRESCRIBED COURSE OF STUDY IN

MECHANICAL ENGINEERING

MAY 27, 1915

G. F. Gebhardt 5/26/15
H. M. Raymond
L. C. Monin

Preface

The work described in this paper was undertaken with a view to making a much more thorough analysis of the engine performance both before and after the installation of the new cams than is here presented. Very much to the regret however to the writers the limits set by the time available has made any complete thermodynamic study impossible. Many interesting problems present themselves in an investigation of this kind, any one of which might form the matter of an extended study. It is necessary therefore, to record the results in the present incomplete state in the hope that they might be useful as a starting point for future investigation.

The writers wish to acknowledge their gratitude to Professor Daniel Roesch, under whose direction the work was carried on, for his kind and attentive assistance. Acknowledgement is also due to the other members of the faculty and to those members of the student body whose help has made the work possible.

Chicago, Illinois
May 22, 1915.

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Chicago, Illinois
May [illegible], 1911.

C O N T E N T S

List of Illustrations and Diagrams

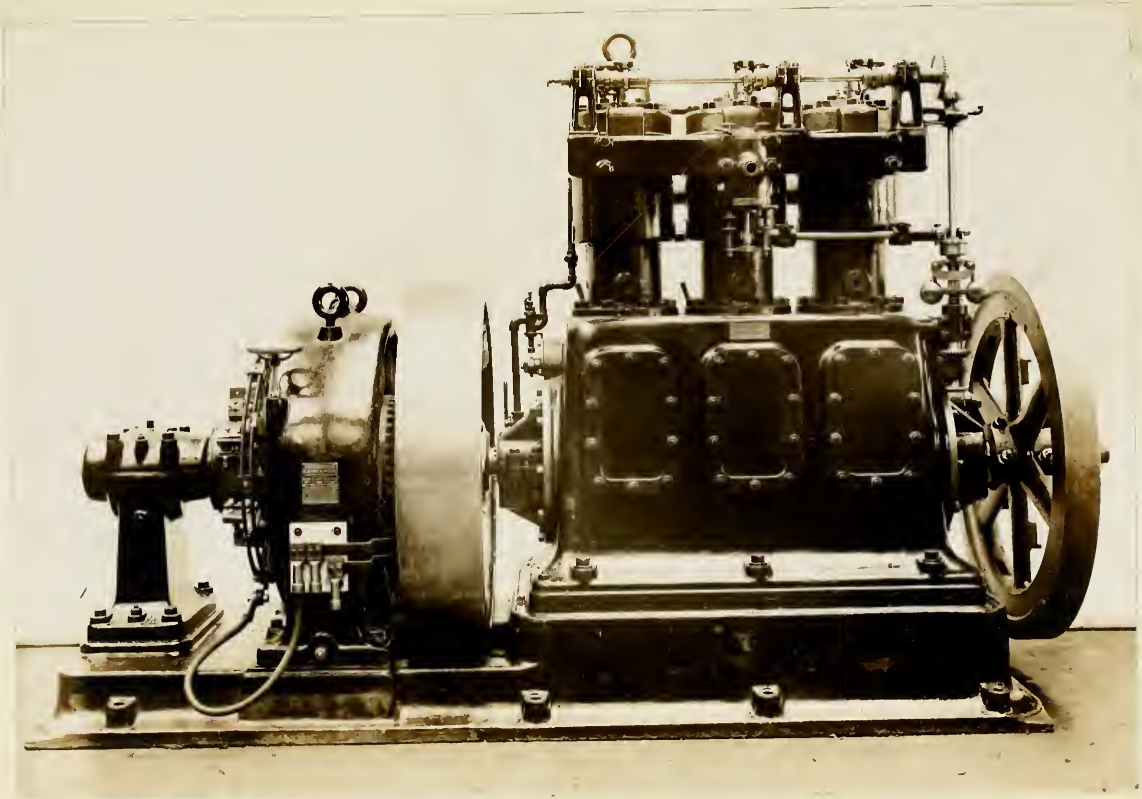

Figure 1

Introduction

The investigations described in these pages were undertaken with the purpose of studying the effect upon the performance of a Westinghouse gas engine, produced by making certain predetermined changes in the forms of the cams controlling the valve motion.

The engine used in the experiments is a three cylinder, 8" x 10", four cycle, constant volume, vertical machine, made by the Westinghouse Machine Company of East Pittsburgh, Pa. It is rated at 40 horse power and is capable of delivering about 45 brake horse power continuously. It was built about 1902 and owing to the fact that there has been a considerable advance in the matter of cam design since that time, it has been thought possible to increase the maximum output and economy by means of modifying the cams and valve motion. The considerations leading to this conclusion will be discussed in another part of this paper.

The Engine

The general appearance of the engine is shown in the illustration, Fig.1. During the tests, however, it was fitted with a water cooled Prony brake in place of the generator shown in the photograph. This and the other photographic illustrations were, very kindly, furnished by the manufacturers and although they represent a somewhat later design, embodying some minor changes, in all essentials the construction here shown is the same as that the engine used. Fig. 2 is a very good general representation of the internal construction, being a section through the left cylinder. In referring to the cylinders and corresponding parts, the words left, center and right will be used as indicating the parts in the order named, beginning with those at the governor end of the engine.

The construction of the machine is shown in greater detail in Figs. 3,4,and 5, representing plan and front and end elevation. The bed plate and crank case are cast in

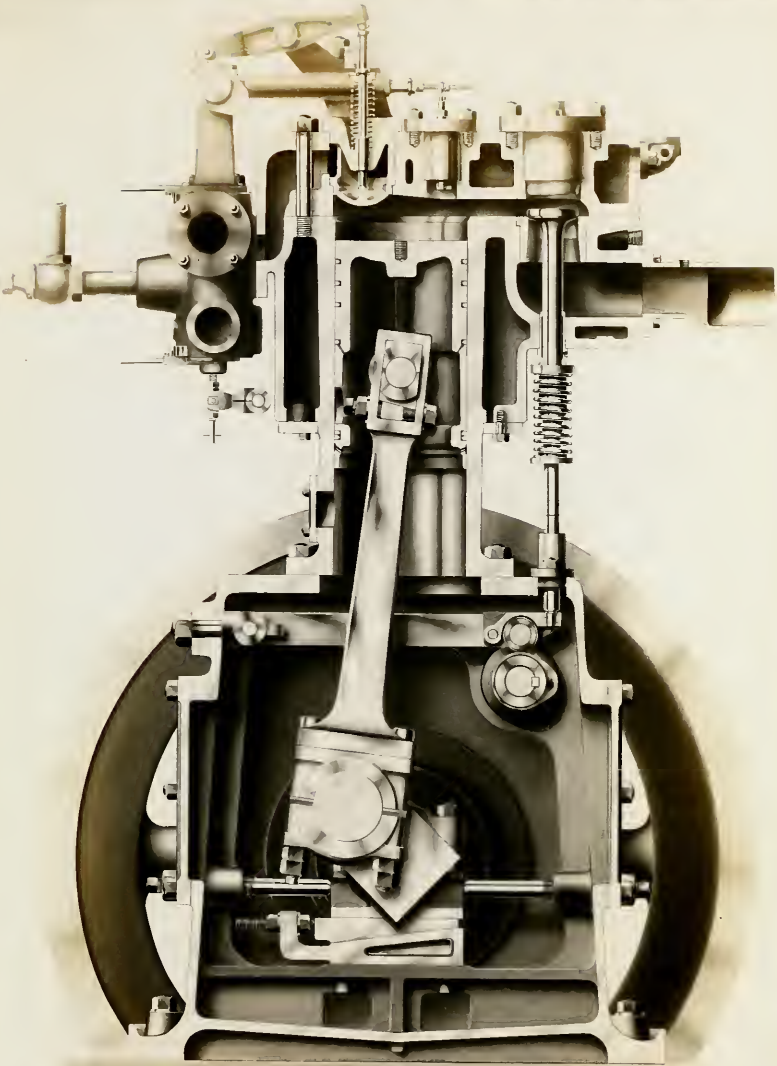

Figure 2

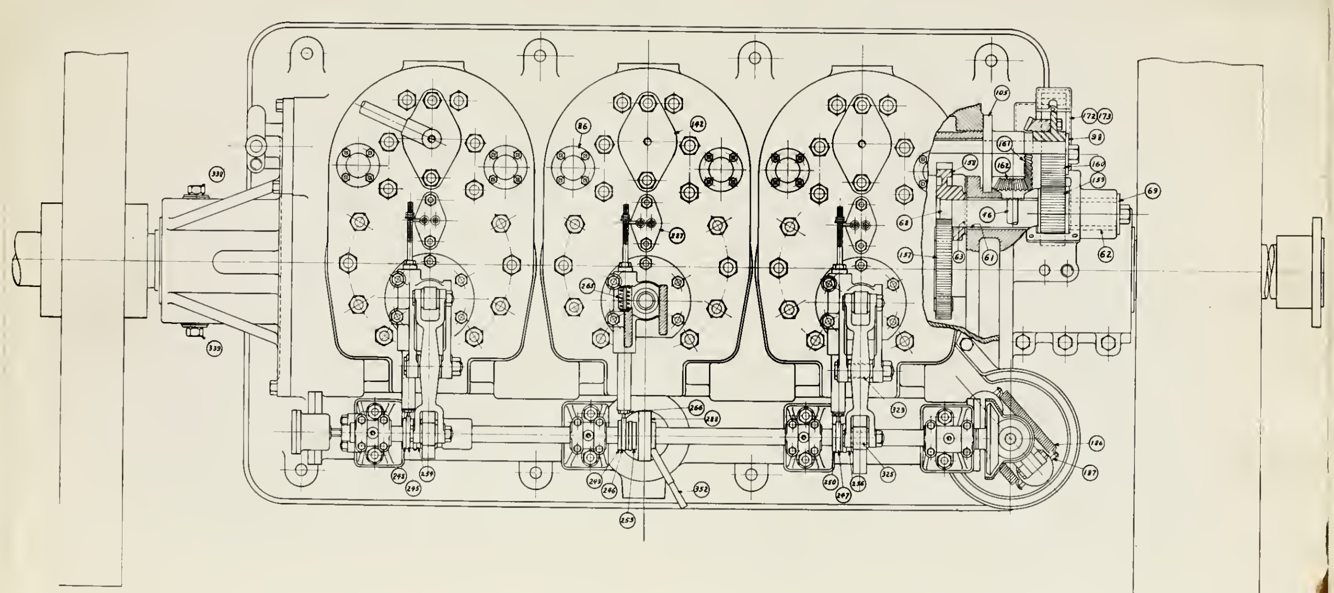

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Figure 3

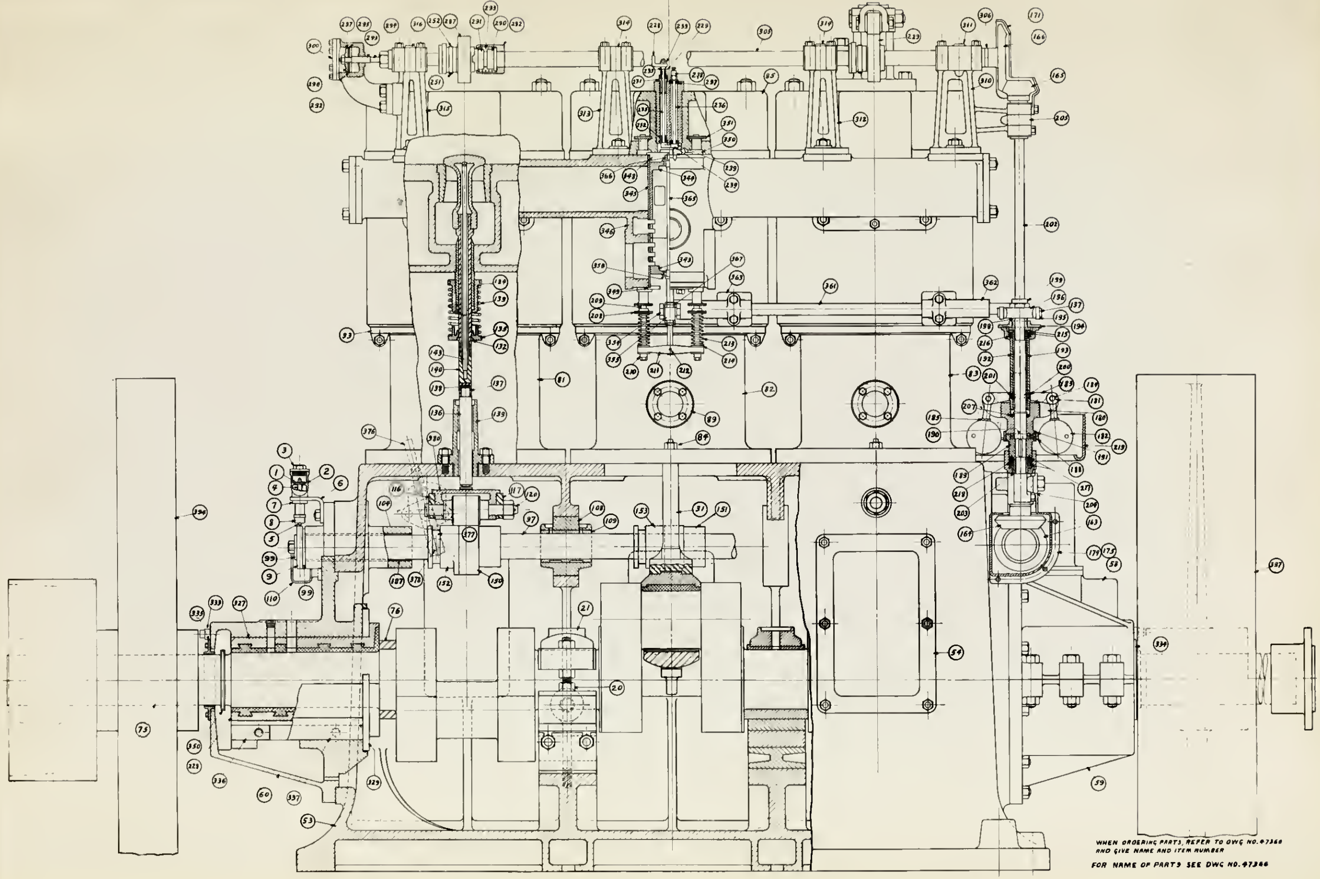
WHEN ORDERING PARTS, REFER TO DWG NO. 47368 AND GIVE NAME AND ITEM NUMBER
FOR NAME OF PARTS SEE DWG NO. 47366

Figure 4

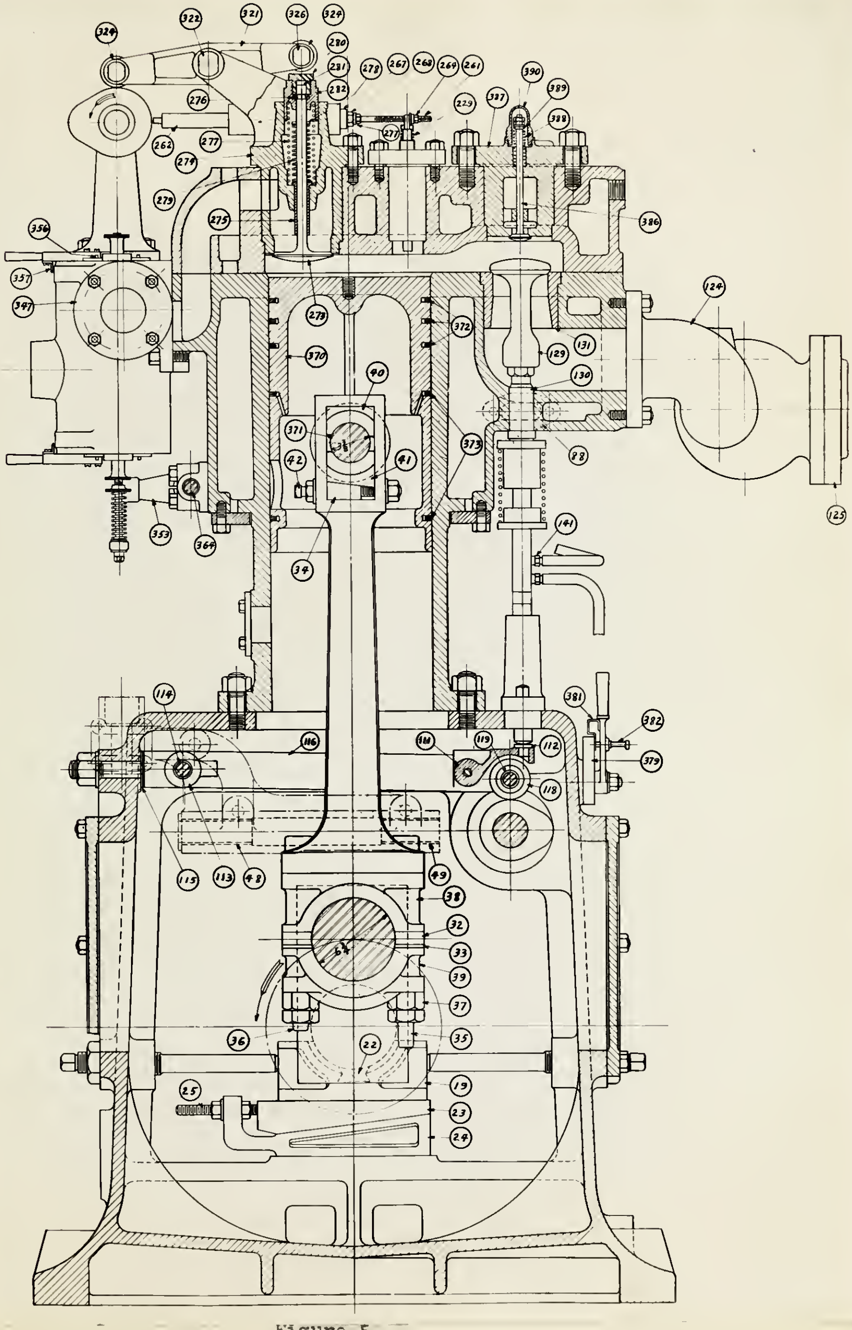
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Figure 5

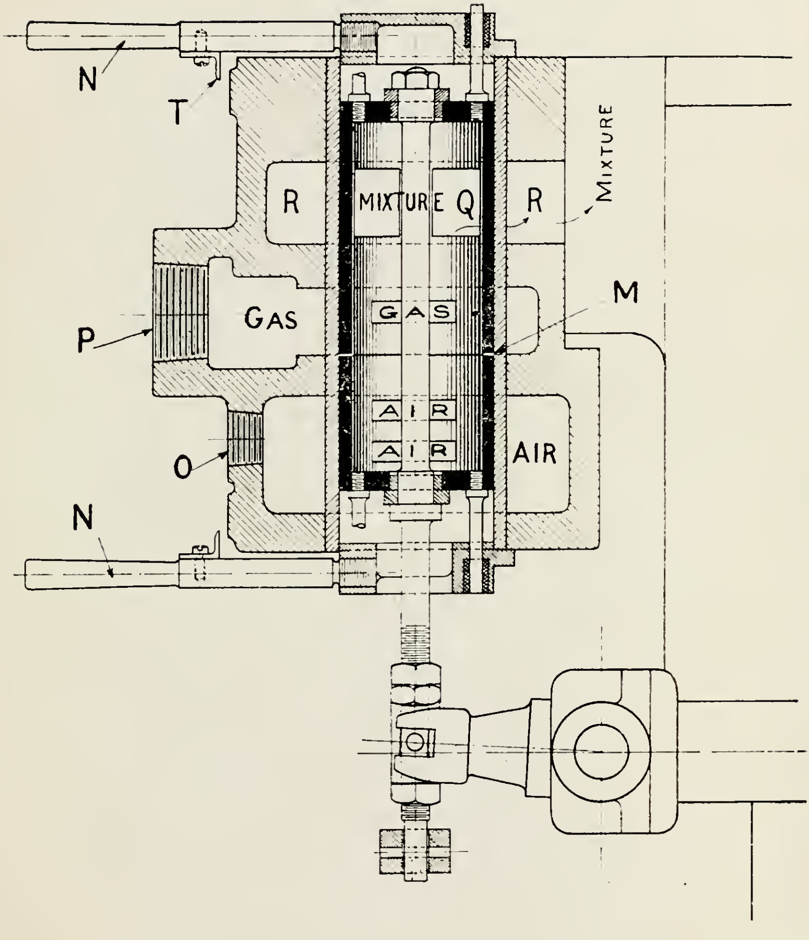


Figure 6

one piece and bolted to a concrete foundation. The three cylinders are separate castings, are water jacketed, and are bolted to the crank case. The valve chambers are bolted to the upper ends of the cylinders. The main shaft and cranks, which are set 120 degrees apart, operate in the crank case. Lubrication is effected by means of large oil cups on the main bearings. The drips from these bearings discharge into the crank case so that the internal working parts are kept flooded by the splash system.

The fuel gas is admitted through the mixing valve shown at 347, Fig. 5 which communicates with the passages connecting with the inlet valves. The details of the mixing valve are shown in Fig. 6. The air inlet is at O and the gas pipe connects at P. Gas and air pass into the valve through ports in the sleeve M, the relative amounts of opening of which are controlled by means of the levers N,N. From the valve the mixture passes into the annular space R and thence into the engine.

The inlet valves are shown at 273 and the exhaust at 129, Fig. 5. The engine is of the ell head type, the inlet valves being located directly above the cylinders and the exhaust in a passage forming an ell toward the back of the engine. Spring poppet valves are used. The inlet valves are operated by means of a set of cams 287, 288, and 289, Figs. 3 and 4, communicating a reciprocating motion to the valve stems 281, through the rocker arms 231, pivoted at 322 and fitted with rollers at 324. A means of timing the events is afforded in the adjustable plungers with lock nut at 326, Fig. 5, in place of the roller shown in the illustration. These plungers make it possible to vary the amount of clearance between the arm and the valve stem.

The exhaust valves are operated by a similar set of cams mounted on a shaft running through the crank case. One of the exhaust cams is shown at 150, Fig. 4 and an end view of it, unnumbered, in Fig. 5. The cam operates in contact with a roller 118 mounted on a lever 116 which is pivoted at 113. The motion is communicated through the shoe 112, Fig.5 and a plunger 136, Fig. 4, to the valve stem 130 and the valve 129, Fig. 5. Between the plunger and the valve stem proper is a short space in which may be inserted small circular discs [illegible] and [illegible] to a [illegible]. The [illegible] cylinders are [illegible], [illegible] are [illegible] to the [illegible]. [illegible] of the [illegible]. [illegible] are set 180 degrees apart, [illegible]. [illegible] is effected by means of [illegible]. The [illegible] the [illegible] cams [illegible], [illegible] in the exhaust system.

The [illegible] gas is admitted through the [illegible] valve shown at [illegible], Fig. 2 [illegible] with the [illegible] connecting with the inlet valves. The details of the [illegible] valve are shown in Fig. 5. The [illegible] the gas pipe [illegible]. The [illegible] through ports in the [illegible], the [illegible] of [illegible] of which are [illegible] by means of [illegible] Fig. 5. From the valve the mixture [illegible] the [illegible] and [illegible] into the engine.

The inlet valves are shown at 174 and the exhaust at 145, Fig. 1. The engine is of the [illegible] type, the inlet valves being located directly above the cylinders and the exhaust is a [illegible] to [illegible] toward the [illegible] of the engine. [illegible] valves are used. The inlet valves are operated by means of a set of cams 180, [illegible] 179, Figs. 3 and 4, communicating a [illegible] motion to the valve stems [illegible] through the rocker arms [illegible], pivoted at [illegible] and fitted with rollers at [illegible]. A means of [illegible] the events is afforded in the adjustable plungers [illegible] at 205, Fig. 3, in place of the roller shown in the illustration. These plungers make it possible to vary the amount of clearance between the arm and the valve stem.

The exhaust valves are operated by a similar set of cams mounted on a shaft running through the [illegible]. One of the exhaust cams is shown at 120, Fig. 1 and an end view of it, [illegible], in Fig. 4. The cam operates in contact with a roller [illegible] mounted on a lever [illegible] pivoted at 119. The motion is [illegible] through [illegible] and a plunger [illegible], [illegible] to the valve stem [illegible] and the valve 150, Fig. 7. Between the plunger and the valve stem [illegible] is a [illegible] space in which may be inserted [illegible] of [illegible] or shims of whatever total thickness the adjustment of clearance between the cam motion and the valve, when seated, requires.

To the end of exhaust cam shaft at 9, Fig. 4 is secured a small auxilliary cam which operates an air valve admitting compressed air to the right cylinder as a means of turning the engine over until it picks up its stroke in starting. In order to make this operation possible a set of three additional auxilliary cams is provided, seen at 152, Fig. 4. These cams are free to slide along the shaft on feathers and are controlled by the hand levers 376, Fig. 4. When in the position shown in the figure, they cause the exhaust valve to open toward the end of every stroke, allowing the air to operate the engine in the manner of a single acting steam engine. When the lever 376 is in a vertical position, they are free of the roller and revolve with the shaft idle.

The cam shafts are run by means of a train of gears from the main shaft, shown in Figs. 3 and 4, 157 to 166.

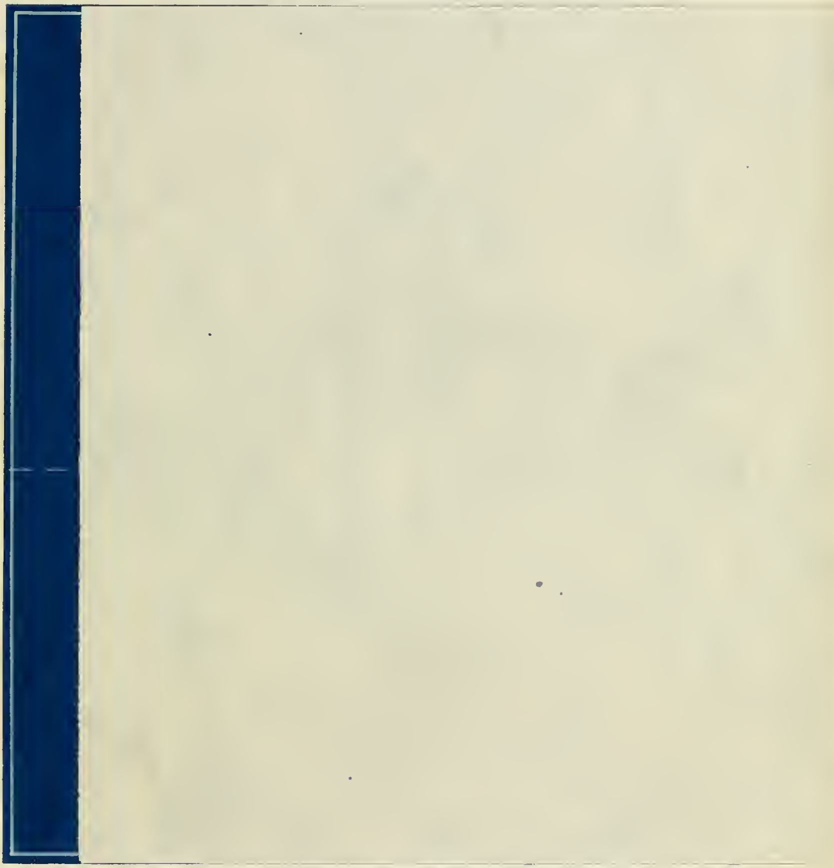
The governor, which is of the fly ball type, is secured to the vertical shaft 202, Fig. 4. The motion of the governor resulting from speed variations, is communicated through a yoke at 197 to a shaft 361 and thence to the mechanism shown at the lower end of the mixing valve, Fig. 6. An increase in speed results in lowering the valve stem and reducing the quantity of gass admitted. The fuel is ignited by means of the make and break system of electric spark. The sparking plugs are seen at 277, Fig. 3, and are operated by spring controlled plungers, 262, Fig. 5, in contact with adjustable cams on the inlet cam shaft.

The Original Cams

An outline of the shape of the admission cams originally furnished with the engine is shown in Fig. 7, and that of the original exhaust cams in Fig. 9. The considerations that led to the adoption of these designs need not be considered here. In Table I are shown the position and sequence of events in each of the three cylinders as operated by the original cams. The measurements were made by means of observing the positions of the cranks as marked in the fly wheel with respect to upper or lower dead center. In Table II are listed the amounts of valve lift in inches and the areas of valve opening in square inches,

FIGURE 7.

FIGURE 8



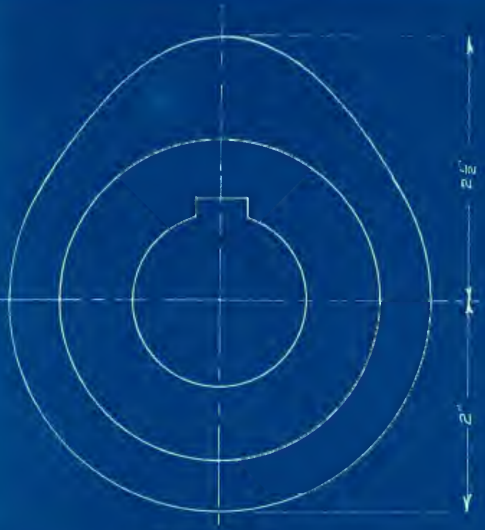


FIGURE 9.

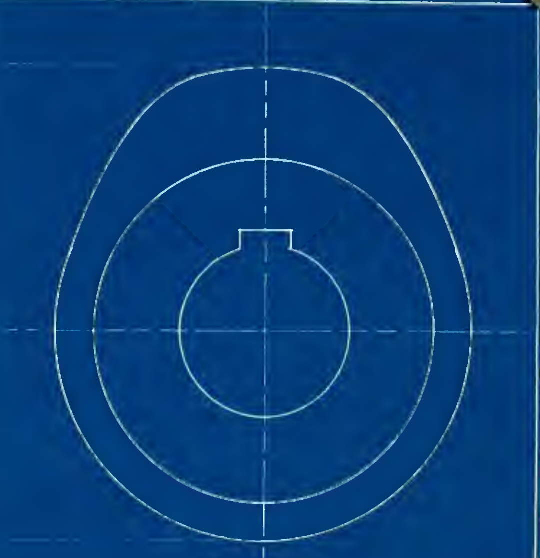


FIGURE 10

Table I

Events	Cylinders		
	Left	Center	Right
Inlet opens	11° early	9° early	14° early
Inlet closes	19° late	12° late	16° late
Ignition	18° early	13° early	16° early
Exhaust opens	22° early	31° early	29° early
Exhaust closes	2° early	5° late	1° early

for every 15 degrees of crank revolution during the time valves are in motion.

Preliminary Tests

In order to obtain information upon which to base the design of the new cams as well as to afford a basis of comparison in studying the effect of the new design, a series of tests was made with the original cams set as indicated in the foregoing description. The series consisted of five short runs at loads of approximately 10, 20, 30, 40 and 45 brake horse power. The first four were twenty minutes and the last a quarter of an hour in duration. For testing purposes the engine is equipped with Westinghouse "wet" gas meter reading in cubic feet, provided with a thermometer and an open end manometer for ascertaining the temperature and pressure of the gas supply. Thermometer cups are provided at suitable points in the inlet and outlet jacket water pipes for convenience in taking temperatures. The jacket water discharges through a two way cock into two weighing tanks.

Frequent cards were taken from all three cylinders during the tests and all other observations were made at intervals of five minutes. These observations are recorded in tabulated form in the printed logs 1 to 7 and the results as calculated are given in 8 and 9. Stop motion cards were taken at each load to show more clearly the characteristics of the lower loops. Sample cards from each run are submitted in Figs. 11 to 15. Figs. 16 to 20 are enlarged diagrams made from cards selected from each run as approximating most nearly the average mean effective pressure for the run.

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Test of Westinghouse three cyl. ***Gas Engine***	Brake Circumference, ft. 33.263
Type Four cycle, throttling	Dead Weight of Arm, lbs. 16.4
Size 8 x 10	On Scales, lbs. 15.9
Fuel Used Artificial gas	Total, lbs. 32.3

MECHANICAL ENGINEERING LABORATORY

ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse three cyl. *Gas Engine*

Type Four cycle, throttling

Size 8 x 10

Fuel Used artificial gas

Brake Circumference, ft. 33.263

Dead Weight of Arm, lbs 16.4

On Scales, lbs. 15.9

Total, lbs. 32.3

LOG OF TEST:

Barometer 29.60

Date December 28, 1914.

	Time	Revolution Counter	Explosions Per Minute	Revolutions Per Minute	Gas Meter	Cubic Feet Gas Per Hr.	Temperature—Degrees F: Entering Jacket Water	Issuing Jacket Water	Gas at Meter	Room	Exhaust	Pressure: Gas at Meter, In. Mer.	At Eng., In. Water	Weights: Jacket Water	Scale Reading	Brake Load, Net	Horse Pow'r: B.H.P.	I.P.H.
	11/17			331	8765		38	78	64	57		5"						
	22			333	8791	312	38	78	64									
	27			332	8817	312	38.5	78.5	64	57								
1	32			332	8844	324	38	78.5	64	57								
	37			332	8870	312	38.5	78.5	64.5	57								
	Average			332	Total 105	315	38.2	78.3	64.1	57		5"		911#	20 min	29.7#		
	Corrected					333.9												
	10/45			330	8568		37	78.5	65	50		4"						
	50			330	8595	324	37	81	64									
2	55			330	8632	444	37	81	63									
	11/00			330	8667	420	37	83	63									
	05			330	8701	408	37	82	63	54.5								
	Average			330	Total 133	399	37	81.1	63.6	52.25		4"		998#	20 min	61.7#		
	Corrected					422.9												
	9/25			330	7671		37	86	67	52.5		2"						
	30			326	7719	576	37	87	69									
3	35			328	7758	588	37	87.5	69									
	40			328	7816	576	37	87	69									
	45			329	7866	600	37	87	69									
	Average			328.2	195	585	37	86.9	68.6	52.5		2"		1099#	20 min	91.7#		
	Corrected					520.1												

REMARKS

Observer.

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Test of Westinghouse three cyl. ***Gas Engine***	Brake Circumference, ft. 33.263
Type Four cycle throttling	Dead Weight of Arm, lbs. 16.4
Size 8 x 10	On Scales, lbs. 15.9
Fuel Used Artificial gas	Total, lbs. 32.3

Mechanical Engineering Laboratory
Armour Institute of Technology

Test of Testinghouse three cyl. **Gas Engine**
Type Four cycle throttling
Size 8 x 10
Fuel Used Artificial gas

Brake Circumference, ft. 33.263
Dead Weight of Arm, lbs. 16.4
On Scales, lbs 15.9
Total, lbs. 32.3

LOG OF TEST: 2nd sheet Barometer 29.6 Date December 28, 1914.

Time	Revolution Counter	Explosions Per Minute	Revolutions Per Minute	Gas Meter	Cubic Feet Gas Per Hr.	Temperature—Degrees F: Entering Jacket Water	Issuing Jacket Water	Gas at Meter	Room	Exhaust	Pressure: Gas at Meter, In. Mer.	At Eng., In. Water	Weights: Jacket Water	Scale Reading	Brake Load, Net	Horse Pow'r: B.H.P.	I.P.H.
9/50			325	7921		37.5	90.5	69	53		1"						
55			322	7980	708	37	92	69									
4 10/00			322	8047	804	37	93.5	69									
05			322	8097	600	37	93.25	69									
10			322	8156	708	37	94	69									
Average			322.6	Total 235	705	37.1	92.65	69	53		1"		836.5# 15 min		132.7#		
Corrected					747.3												
10/15			318	8221		37	94	69	54		3"						
20			314	8290	828	37	94	69									
5 25			315	8360	840	37	95	69									
30			317	8426	792	37	95	69									
Average			316	Total 205	820	37	94.5	69	54		3"		877# 15 min		140.7#		
Corrected					869.2												

Remarks:

Observer

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Armour Institute of Technology

Test of Westinghouse three cyl. *Gas Engine.* *Date* December 28, 1914.

Mechanical Engineering Laboratory

ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse three cyl. ***Gas Engine.*** ***Date*** December 28, 1914.

Size 8" x 10" Type Four cycle Clearance Left 106.81 / Center 100.53 / Right 109.95 cubic inches.

Rating 40 H. P. at 325 R. P. M. Clearance Left 21.25 / Center 20 / Right 21.87% piston displacement.

Indicator Used Crosby Maximum Compression 35.2 lbs. per sq. in.

Scale Spring 80#

DATA FROM CARDS

No.	R. P. M.	Area	Length of Card	Mean Effective Pressure	I. H. P.	B. H. P.	Mechanical Efficiency
Left							
1		1.04	2.97				
2		.93	2.98				
3		1.08	2.94				
4		1.05	2.95				
5		1.00	2.97				
	Average	1.02	2.962	27.55	5.81		
Center							
1		.71	2.90				
2		.73	2.90				
3		.82	2.89				
4		.85	2.92				
5		.93	2.90				
	Average	.808	2.902	22.24	4.69		
Right							
1		.90	2.96				
2		.93	2.95				
3		.82	2.95				
4		.83	2.95				
5		.80	2.94				
	Average	.856	2.95	23.2	4.89		
					15.39	9.95	64.6%

Piston areas = 50.16 sq.in.

Remarks:

Observer.

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Test of Westinghouse, three cyl. *Gas Engine.* *Date* December 28, 1914.

Mechanical Engineering Laboratory

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Test of Westinghouse, three cyl. **Gas Engine.** **Date** December 23, 1914.

Size 8" x 10" Type Four cycle Clearance Left 106.81, Center 100.53, Right 109.95 cubic inches.

Rating 40 H.P. at 325 R. P. M. Clearance Left 21.25, Center 20, Right 21.87 % piston displacement.

Indicator Used Crosby Maximum Compression 51.2 lbs. per sq. in.

Scale Spring 80#

DATA FROM CARDS

No.	R. P. M.	Area	Length of Card	Mean Effective Pressure	I. H. P.	B. H. P.	Mechanical Efficiency
Left							
1		1.75	3.00				
2		1.94	2.99				
3		1.75	2.98				
	Average	1.813	2.99	48.48	10.14		
Center							
1		1.30	2.90				
2		1.63	2.91				
3		1.40	2.92				
4		1.24	2.90				
	Average	1.393	2.91	38.30	8.01		
Right							
1		1.52	2.96				
2		1.74	2.95				
3		1.54	2.97				
	Average	1.60	2.96	43.25	9.05		
					27.20	20.53	75.4

Remarks:

Observer.

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Test of Westinghouse three cyl. *Gas Engine.* *Date* December 28, 1914.

Mechanical Engineering Laboratory

Armour Institute of Technology

Test of Westinghouse three cyl. **Gas Engine.** **Date** December 2_, 1914.

Size 8" x 10" Type Four cycle Clearance Left 1_5.81, Center 100.53, Right 109.95 cubic inches

Rating 40 H.P. at 325 R. P. M. Clearance Left 21.25, Center 20._, Right 21.87 % piston displacement

Indicator Used Crosby Maximum Compression 6_.4 lbs. per sq. in.

Scale Spring 240

Data from Cards

No	R. P. M	Area	Length of Card	Mean Effective Pressure	I. H. P	B. H. P	Mechanical Efficiency
Left							
1		.73	3.00				
2		.71	3.01				
3		.71	2.98				
4		.76	3.03				
5		.77	3.02				
	Average	.736	3.008	58.8	12.24		
Center							
1		.73	2.90				
2		.76	2.89				
3		.68	2.89				
4		.79	2.89				
5		.71	2.90				
	Average	.734	2.894	60.50	12.68		
Right							
1		.71	2.97				
2		.71	2.96				
3		.71	3.00				
4		.76	2.99				
5		.73	2.99				
	Average	.724	2.982	58.30	12.13		
					37.05	30.35	81.9%

Remarks

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Test of Westinghouse three cyl. ***Gas Engine.*** ***Date*** December 28, 1914.

Mechanical Engineering Laboratory

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Test of Westinghouse three cyl. **Gas Engine.** **Date** December 18, 1914.

Size 8" x 10" Type Four cycle Clearance Left 106.81 Center 100.53 Right 109.95 cubic inches.

Rating 40 H.P. at 325 R. P. M. Clearance Left 21.25 Center 20 Right 31.87 % piston displacement.

Indicator Used Crosby Maximum Compression 61.5 lbs per sq. in.

Scale Spring 240

DATA FROM CARDS

No.	R. P. M.	Area	Length of Card	Mean Effective Pressure	I. H. P.	B. H. P.	Mechanical Efficiency
Left							
1		1.05	3.02				
2		1.04	3.02				
3		.91	3.02				
4		.98	3.02				
5		1.04	3.01				
	Average	1.004	3.[illegible]	79.68	16.29		
Center							
1		.82	2.89				
2		.86	2.89				
3		.94	2.89				
4		.93	2.88				
5		.94	2.90				
	Average	.898	2.89	74.57	15.25		
Right							
1		1.04	2.96				
2		1.00	2.95				
3		1.03	2.98				
4		.99	2.96				
5		1.06	2.98				
	Average	1.014	2.966	82.80	16.93		
					48.47	39.85	82.4%

Remarks

Observer.

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ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse three cyl. *Gas Engine.* *Date* December 28, 1914.

Mechanical Engineering Laboratory

ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse three cyl. ***Gas Engine.*** ***Date*** December 18, 1914.

Size 8" x 10" Type Four cycle Clearance Left 106.81, Center 100.53, Right 109.95 cubic inches.

Rating 40 H.P. at 325 R. P. M. Clearance Left 21.25, Center 20, Right 21.87 % piston displacement.

Indicator Used Crosby Maximum Compression 81.6 lbs. per sq. in.

Scale Spring 240

DATA FROM CARDS

No.	R. P. M.	Area	Length of Card	Mean Effective Pressure	I. H. P.	B. H. P.	Mechanical Efficiency
Left							
1		1.21	3.01				
2		1.24	3.03				
3		1.19	3.02				
4		1.16	3.01				
	Average	1.20	3.017	95.28	19.08		
Center							
1		.98	2.89				
2		.98	2.88				
3		.96	2.89				
4		.96	2.88				
	Average	.97	2.88	80.64	16.15		
Right							
1		1.14	2.98				
2		1.12	2.99				
3		.81	3.00				
4		.78	2.98				
	Average	.962	2.987	77.28	15.48		
					50.71	44.80	88.4

Remarks:

Observer.

MECHANICAL ENGINEERING LABORATORY

ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse 3 cyl.8 x 10 *Gas Engine* *Date* December 28, 1914.

Kind of Fuel Artificial gas

Mechanical Engineering Laboratory

Armour Institute of Technology

Test of Westinghouse 3 cyl.8 x 10 ***Gas Engine*** ***Date*** December 28, 1914.

Kind of Fuel Artificial gas

No. of Run		1	2	3	4
Duration of test,	hours.	1/3	1/3	1/3	
Gas consumed,	cu. ft.	111.2	141	206.5	
Air consumed,	cu. ft.				
*Calorific value of gas, total,	B. T. U. per cu. ft.	672.5	672.5	672.5	
Calorific value of gas, effective.	B. T. U. per cu. ft.	625	625	625	
Jacket water supplied,	lbs.	911	998	1099	
*Gas per hour,	cu. ft.	332.43	420.29	607.15	
*Air per hour,	cu. ft.				
Jacket water per hour,	lbs.	2733	2994	3297	
Gas at meter,	ins. mercury.	.38	.295	.15	
Barometer,	ins. mercury.	29.60	29.60	29.60	
Gas at engine,	ins. water.				
Jacket water { Inlet	deg. F.	38.2	37	37	
Jacket water { Outlet,	deg. F.	73.8	81.1	86.9	
Gas at meter,	deg. F.	64.1	63.6	68.6	
Air in room,	deg. F.	57	52.25	52.5	
Revolutions per minute,	rev.	332	330	328.2	
Explosions per minute,		166	165	164.1	
Pressure in lbs. per sq. in. above atmosphere					
(a) Maximum pressure,		74.8	145.6	206.4	
(b) Pressure just before ignition,		34.8	54.4	62.4	
(c) Pressure at end of expansion,		10.4	18.4	24.0	
(d) Exhaust pressure,		1.2	2.8	1.2	
Mean effective pressure,	lbs. per sq. in.				
Builders' rating,	H. P.	40	40	40	
Actual indicated H. P.,	H. P.	15.39	27.20	37.05	
Actual brake H. P.	H. P.	9.95	20.53	30.35	
Mechanical Efficiency	%	64.6	75.4	81.9	
B. T. U. per I. H. P. per hr.,		13500	9660	10240	
B. T. U. per B. H. P. per hr.,		20880	12795	12500	
*Cu. ft. gas per I. H. P.,		21.7	15.46	16.40	
*Cu. ft. gas per B. H. P.,		33.4	20.50	20.00	
Heat equivalent of I. H. P., efficiency	per cent.	19.9	26.4	24.9	
Heat rejected in jacket water	per cent.	52.8	52.3	43.4	
Heat rejected in exhaust and lost through radiation and incomplete combustion,	per cent.	27.3	21.3	31.	
Heat equivalent of B. H. P. efficiency	per cent.	12.2	19.9	20.2	

*All gas volumes reduced to 62° F and 30 inches mercury.

Observer

Mechanical Engineering Laboratory

Armour Institute of Technology

Test of Westinghouse 3 cyl. 8 x 10 *Gas Engine* *Date* December 28, 1914.

Kind of Fuel Artificial gas

2nd sheet

Mechanical Engineering Laboratory

ARMOUR INSTITUTE OF TECHNOLOGY

Test of Westinghouse 3 cyl. 8 x 10 ***Gas Engine*** ***Date*** December 28, 1914.

Kind of Fuel Artificial gas
2nd sheet

No. of Run		4	5
Duration of test,	hours.	1/3	1/4
Gas consumed,	cu. ft.	249.1	217.5
Air consumed,	cu. ft.		
*Calorific value of gas, total,	B. T. U. per cu. ft.	672.5	672.5
Calorific value of gas, effective.	B. T. U. per cu. ft.	625	625
Jacket water supplied,	lbs.	836.5 } 15 min	877
*Gas per hour,	cu. ft.	729.17	847.63
*Air per hour,	cu. ft.		
Jacket water per hour,	lbs.	3346	3508
Gas at meter,	ins. mercury.	.074	.055
Barometer,	ins. mercury.	29.60	29.60
Gas at engine,	ins. water.		
Jacket water { Inlet	deg. F.	37.1	37
Jacket water { Outlet,	deg. F.	92.65	94.5
Gas at meter,	deg. F.	69	69
Air in room,	deg. F.	53	54
Revolutions per minute,	rev.	322.6	316
Explosions per minute,		161.3	158
Pressure in lbs. per sq. in. above atmosphere			
(a) Maximum pressure,		273.6	436
(b) Pressure just before ignition,		75.8	81.6
(c) Pressure at end of expansion,		28.8	38.4
(d) Exhaust pressure,		1.2	1.2
Mean effective pressure,	lbs. per sq. in.		
Builders' rating,	H. P.	40	40
Actual indicated H P.,	H. P.	48.47	50.71
Actual brake H. P.	H. P.	39.95	44.80
Mechanical Efficiency	%	82.4	88.4
B. T U. per I. H. P. per hr.,		9400	10440
B. T. U. per B. H. P. per hr.,		11400	11820
*Cu. ft. gas per I. H. P.,		15.05	16.7
*Cu. ft. gas per B. H. P.,		18.25	18.8
Heat equivalent of I. H. P., efficiency	per cent.	27.1	24.4
Heat rejected in jacket water	per cent.	40.7	38.1
Heat rejected in exhaust and lost through radiation and incomplete combustion,	per cent.	32.2	37.5
Heat equivalent of B. H. P. efficiency	per cent.	22.35	21.55

*All gas volumes reduced to 62° F and 30 inches mercury

Observer

Form 6

MECHANICAL ENGINEERING LABORATORY

ARMOUR INSTITUTE OF TECHNOLOGY

DETERMINATION OF THE CALORIFIC VALUE OF GAS, BY JUNKER'S CALORIMETER.

Date December 28, 1914.

Form 6

Mechanical Engineering Laboratory

ARMOUR INSTITUTE OF TECHNOLOGY

DETERMINATION OF THE CALORIFIC VALUE OF GAS, BY JUNKER'S CALORIMETER.

Date December 28, 1914.

Kind of gas Artificial gas

Source City mains

NO. OF RUN	1	2	3	4	5
Time starting run,	11-21	27-05			
Time ending run,	14-42	31-20			
Duration of run, minutes,	3-21	4-15			
Temperature of chimney, ° F,	51	51.4 51.0			
Temperature of air, ° F,	59	59.4 59.6			
Temperature of gas at meter, ° F,	52	57.8 57.9			
Temperature of entering jacket water, ° C,	2.7	3.6			
Temperature of issuing jacket water, ° C,	31.7	24.1			
Range of temperature, °F	52.2	36.9			
Pressure of atmosphere, inches, mercury,	29.6	29.6			
Pressure of gas at meter, inches, water,	2.0	3.76 3.80			
Pressure of gas at burner, inches water,					
Weight of discharge tank and water, lbs.,	7.99	10.46			
Weight of discharge tank, empty, lbs.,	1.42	1.36			
Weight of jacket water, lbs.,	6.57	9.10			
Reading of meter, beginning,	0	0			
Reading of meter, end,	.5	.5			
Total gas consumed, cu. ft.,	.5	.5			
Total gas consumed reduced to 30″ Hg. and 62° F., cu. ft.,	.506	.503			
Cu. ft. gas per hour,	9.[illegible]6	7.10			
Lbs. steam condensed per cu. ft. gas,	11 c.c. .0478	10.7 c.c. .0469	Average		
Calorific value of Gas B. T. U., total,	678	667	672.5		
Calorific value of Gas B. T. U., effective,	630	620	625		

1 Gram = 0.0022046 lbs.

1 inch water = 0.0736 inches mercury.

Remarks:

Observer

Load 9.95 B.H.P.
Cyl. left.
Card no.1.
Spring 80 lb.
Length 2.97 in.
Area 1.04 sq. in.

Load 995 B.H.P.
Card Left Cyl.
Length 2.98 in.
Spring 20 lb.

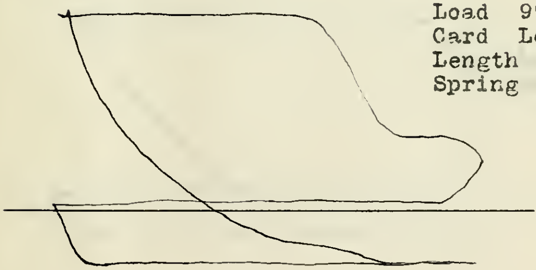
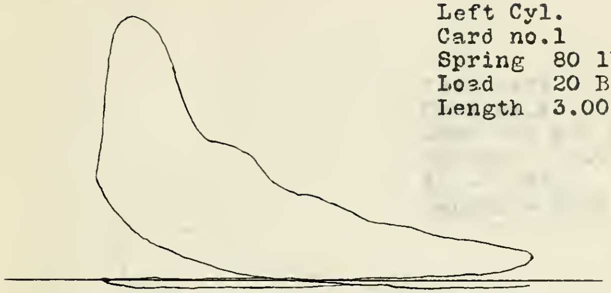
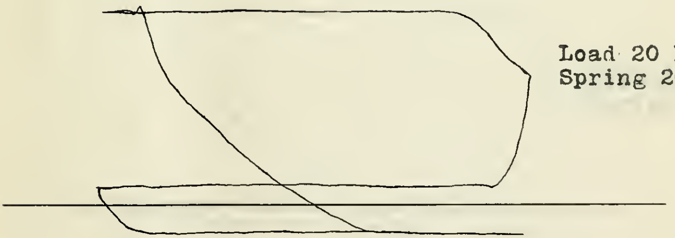


Figure 11



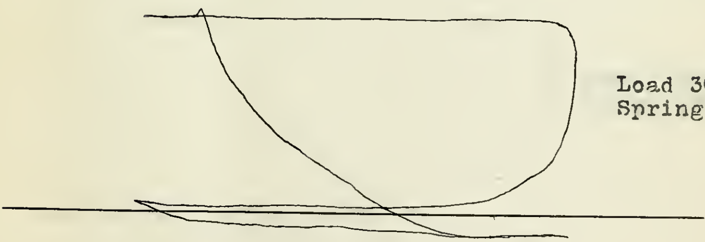
Left Cyl.
Card no.1
Spring 80 lb.
Load 20 B.H.P.
Length 3.00 in.



Load 20 B.H.P.
Spring 20 lb.

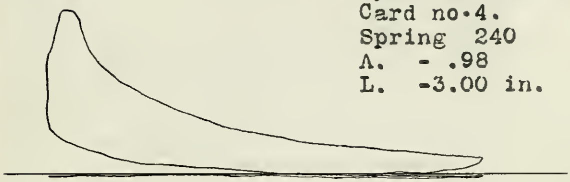
Figure 12.

Cyl. left
Card no.1
Load 30 B.H.P.
Spring 240
A. - .73
Length - 3.00 in.

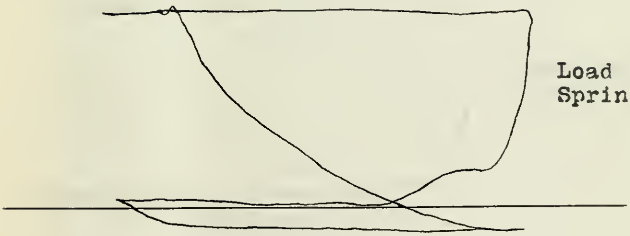


Load 30 H.P.
Spring 20 lb.

Figure 13.

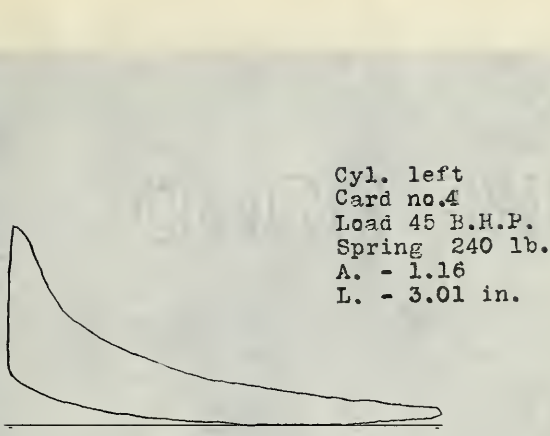


Load 40 B.H.P.
Cyl. left.
Card no.4.
Spring 240
A. - .98
L. -3.00 in.



Load 40 H.P.
Spring 20 lb.

Figure 14.


Cyl. left
Card no.4
Load 45 B.H.P.
Spring 240 lb.
A. - 1.16
L. - 3.01 in.


Load 45 B.H.P.
Spring 20 lb.

Figure 15.

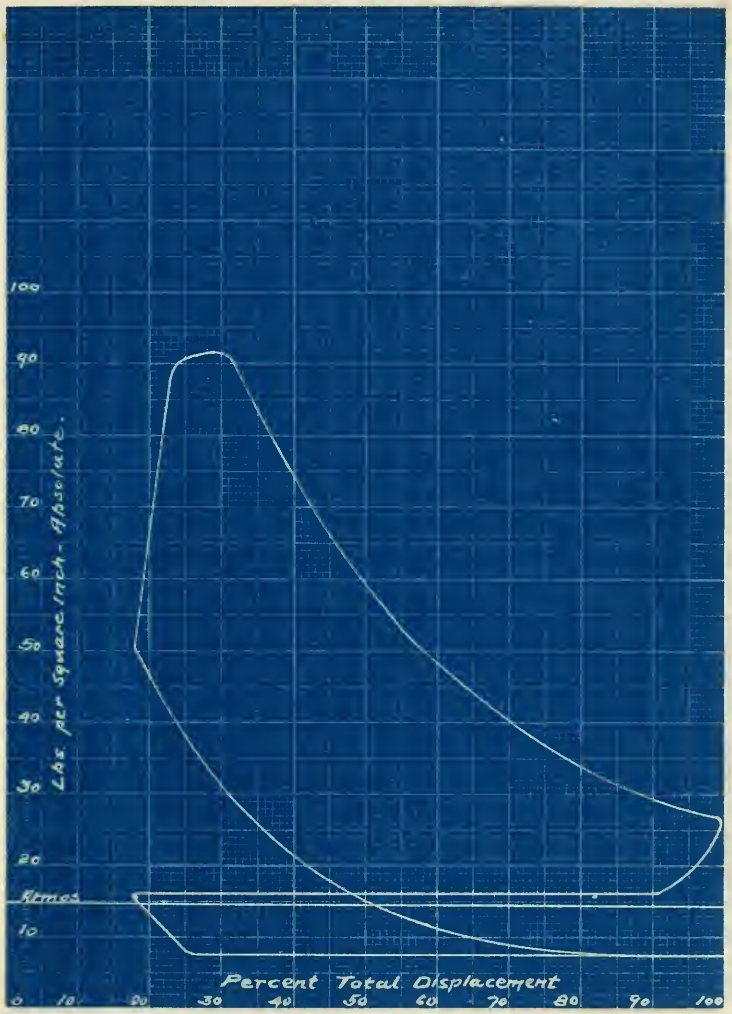
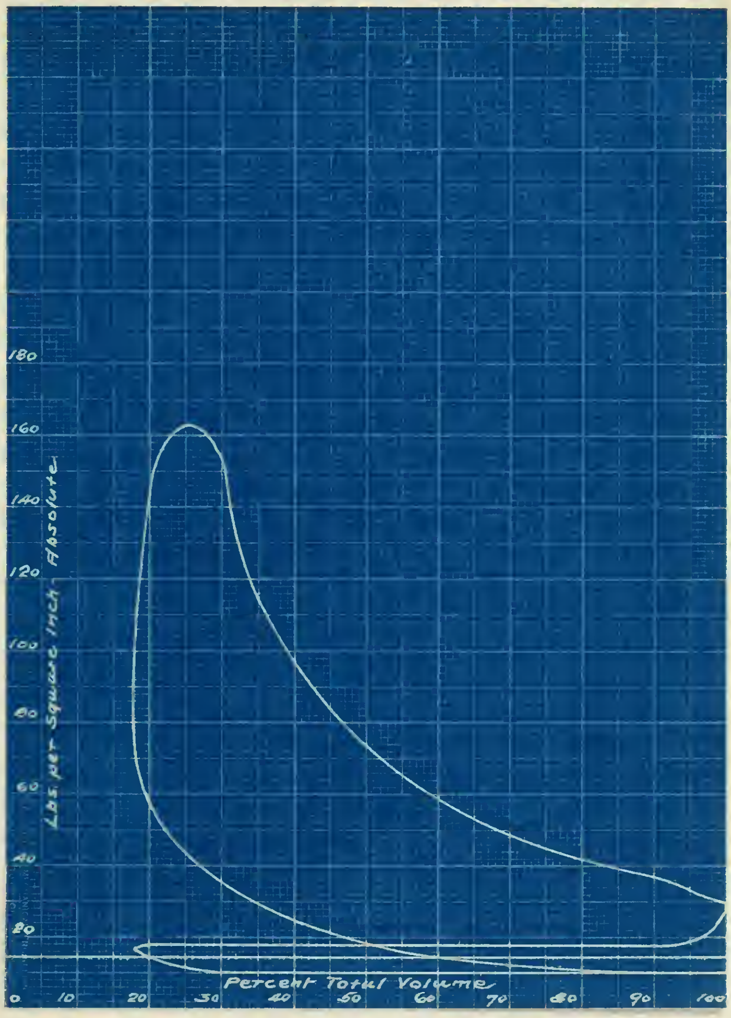

100
90
80
70
60
50
40
30
20
10
Lbs. per Square Inch. Absolute.
Percent Total Displacement
0
10
20
30
40
50
60
70
80
90
100

Figure 16



180
160
140
120
100
80
60
40
20
Lbs. per Square Inch Absolute
0
10
20
30
40
50
60
70
80
90
100
Percent Total Volume

Figure 17

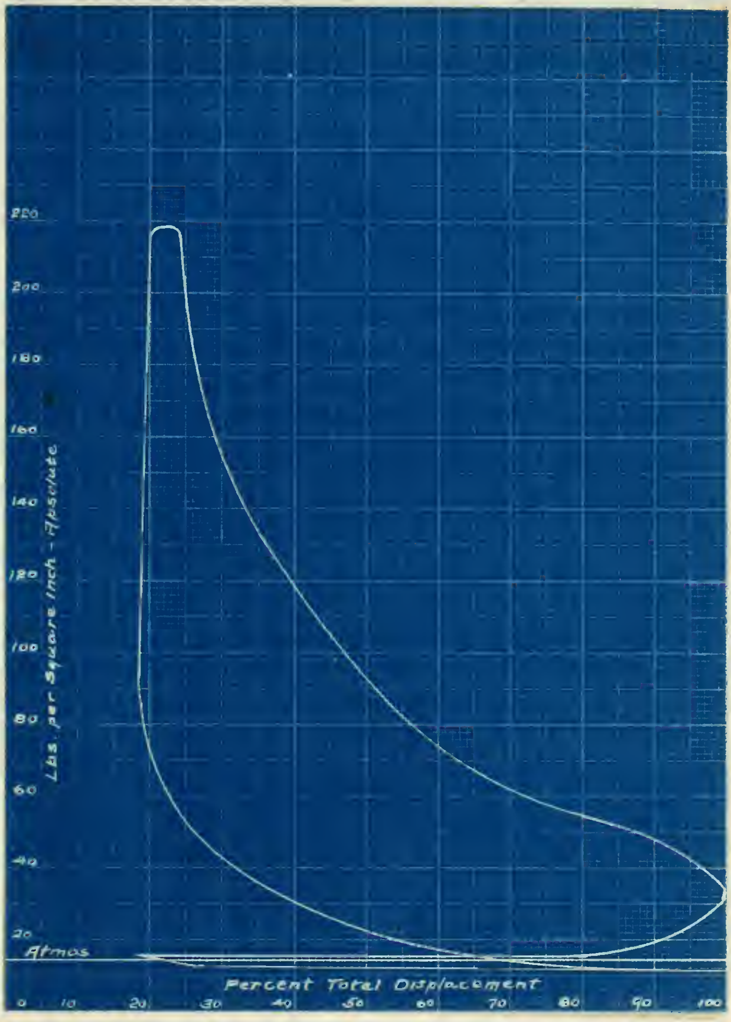
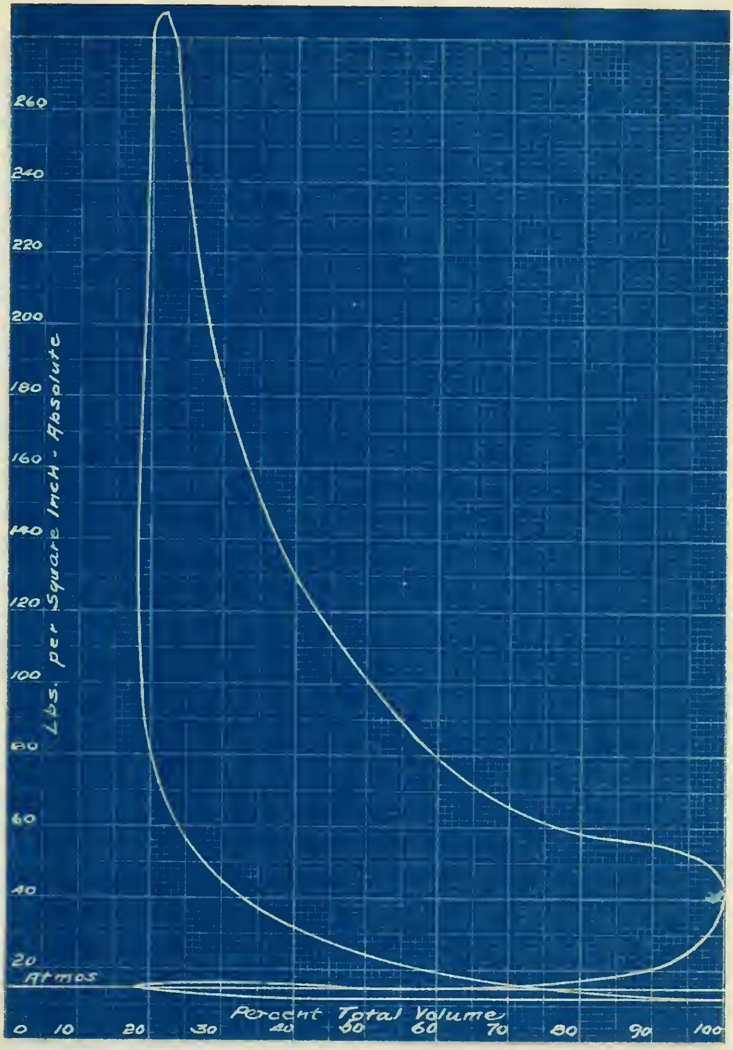

220
200
180
160
140
120
100
80
60
40
20
Atmos
Lbs. per Square Inch - Absolute
Percent Total Displacement
0
10
20
30
40
50
60
70
80
90
100

Figure 18



260
240
220
200
180
160
140
120
100
80
60
40
20
Atmos
Lbs. per Square Inch - Absolute
0 10 20 30 40 50 60 70 80 90 100
Percent Total Volume

Figure 19

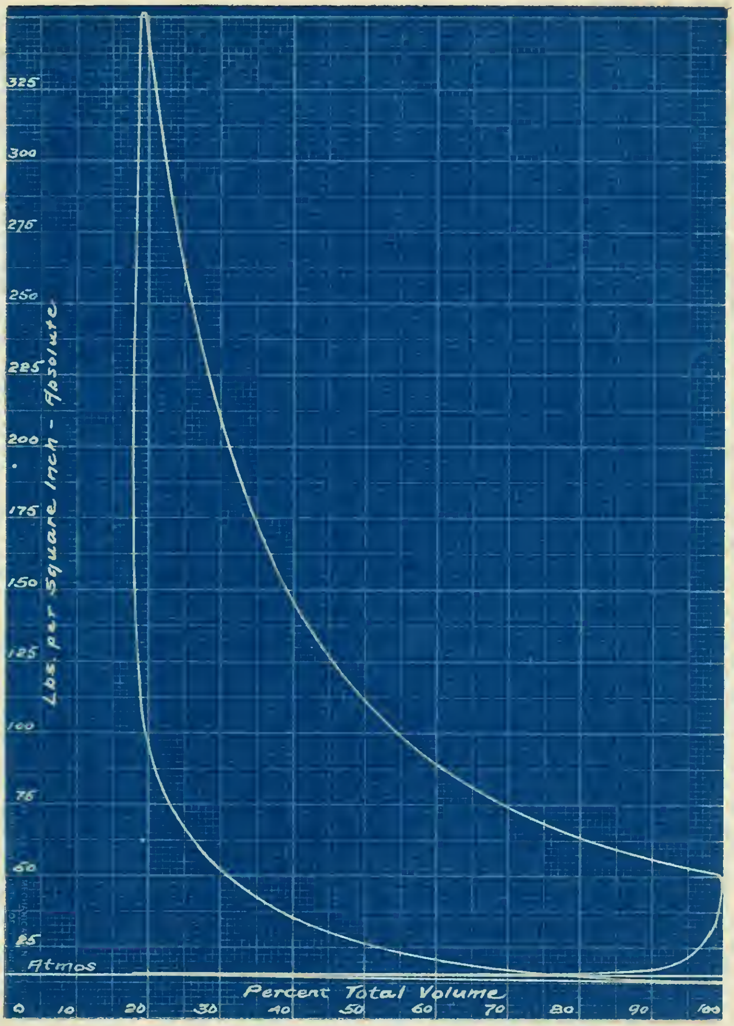

325
300
275
250
225
200
175
150
125
100
75
50
25
Atmos
Lbs. per Square Inch - Absolute
Percent Total Volume
0
10
20
30
40
50
60
70
80
90
100

Figure 20

The fuel used was artificial gas and its calorific value was determined by means of a Junker calorimeter before and after the engine tests. The results of these determinations are given in log 10 and the average effective value used in the calculations.

In a test made by Professor Roesch, the gas meter was found to register 6% below the actual quantity passed and corrections were made for this discrepancy. Corrections were also made reducing the gas to standard conditions of temperature and pressure, 30" mercury and 62° F, both in the engine tests and calorimetric determinations. Air and gas were admitted to the mixing valve in about the ratio of six to one by volume.

Changes Determined Upon

As already stated, it was thought the possible maximum power output of the engine might be increased by altering the valve motion as controlled by the cams. The possibilities in this respect are limited to changes in (1) the total period during which the valve is open (2) the periods of opening and closing (3) length of dwell (4) amount of valve lift.

It is, of course desirable to reduce the fluid friction as much as possible and this may be accomplished to some extent by making the valves open as quickly as possible and remain wide open during as great a part of the period between the beginning of opening and end of closing as possible. It is desirable however to have the action such that the valves work smoothly and quietly. To accomplish this a motion is required which proceeds gradually but is at the same time accomplished in the shortest possible time. A motion that proceeds at a constant rate of accelleration has been found in many cases satisfactorially to meet these requirements. The working parts must be so arranged that the motion is constantly accellerated during one half of the distance to be travelled and decellerated at the same rate during the other half. It was therefore decided to design the new cams to give a motion of this nature. The length of dwell is determined of necessity as a consequence of the decision as to the duration

of the constantly accellerated motion as it is the total period of port opening less the sum of the opening and closing periods. The possible improvement by increasing the amount of valve lift is limited by the size of the passages on either side of the valves and by the clearances of the working parts of the engine. It was recognized that the maximum lift must be enough to give a maximum port opening at least equal to the area of the passages through which the gases must pass. The determination of what these lifts should be will be discussed in a later paragraph.

The principal object in changing the total period during which the valve is off its seat is to allow of as great a period of dwell as can be obtained. There are of course certain limits within which the events must fall and the shorter the periods of opening and closing can be made the greater is the portion of the total period which can be allotted to the period of dwell. An examination of the Figs. 11 to 16 shows in the upturned toe that exhaust opening is later than it should be. Exhaust closing should occur a little after the upper dead center has been passed to obtain full advantage of the inertia of the exhaust gases. Inlet opening should occur as near the beginning of the stroke as possible and closing should occur some considerable interval after the crank has passed the lower dead center.

The sequence of events decided upon, for the experimental purposes of this investigation, as a basis for the design of the cams was as follows:

Inlet opening	0°
" closing	40° late
Exhaust opening	55° early
" closing	10° late

Reckoning in degrees from the upper dead center as zero, this gives inlet opening at zero and closing at 220° of angular displacement of the crank. As the cam shaft makes only one half a revolution to one of the crank, the corresponding inlet cam travel is 110°. Of this amount 40° each were allowed for opening and closing and the remaining 30° for dwell.

In the same way the total exhaust valve motion occurs in a period corresponding to 245° of angular crank displacement, and this is equal to 122.5° of the cam shaft. In this case opening and closing are to occur during 45° each and the period of dwell of the constantly accelerated motion as it is the total period of port opening less the sum of the opening and closing periods. The possible improvement by increasing the amount of valve lift is limited by the size of the passages on either side of the valves and by the clearances of the working parts of the engine. It was recognized that the maximum lift must be enough to give a maximum port opening at least equal to the area of the passages through which the gases must pass. The determination of what these lifts should be will be discussed in a later paragraph.

The principal object in changing the total period during which the valve is off its seat is to allow of as great a period of dwell as can be obtained. There are of course certain limits within which the events must fall and the shorter the periods of opening and closing can be made the greater is the portion of the total period which can be allotted to the period of dwell. An examination of the Figs. it is shown in the figures that exhaust opening is later than it should be. Exhaust closing should occur a little after the upper dead center has been passed to obtain full advantage of the inertia of the exhaust gases. Inlet opening should occur as near the beginning of the stroke as possible and closing should occur some considerable interval after the crank has passed the lower dead center.

The sequence of events decided upon, for the experimental purposes of this investigation, as a basis for the design of the cams was as follows:

Inlet opening	0°
" closing	40° late
Exhaust opening	50° early
" closing	10° late

Reckoning in degrees from the upper dead center as zero, this gives inlet opening at zero and closing at 220° of angular displacement of the crank. As the cam shaft makes only one half a revolution to one of the crank, the corresponding inlet cam travel is 110°. Of this amount 40° each were allowed for opening and closing and the remaining 30° for dwell.

In the same way the total exhaust valve motion occurs in a period corresponding to 240° of angular crank displacement, and this is equal to 120° of the cam shaft. In this case opening and closing are to occur during 45° each and the period of dwell will be 32.5°.

The application of these figures to the design will be found in the next section.

Design of New Cams

The periods of opening, dwell and closing having been decided upon, the design of the cams to produce these motions was largely a matter of kinematics. As it was thought desirable to increase the maximum valve lift if possible, some calculations were necessary to find what this might be made.

Considering first the inlet valves, an examination of them showed that there was sufficient clearance of all moving parts to allow for some considerable change in this respect. It was therefore necessary to find what maximum lift would give a port opening not less than the area of the passage around the valve stem. The minimum area about the valve stem, being an annular space, is the area within the cage less the sectional area of the stem. The internal diameter of the cage is 2 1-8 " and the diameter of the stem is 7-16", see Fig. 23. The area of the annular space is, therefore,

$$A_1 = \frac{\pi}{4}\left[\left(2\tfrac{1}{8}\right)^2 - \left(\tfrac{7}{16}\right)^2\right] = 3.396 \text{ sq. in.}$$

It remained to find a maximum port opening equal to this area. As the valves are conical in shape, making an angle of 45° with the axis of the stem, the port opening is given by the equation,

$$A_2 = \pi(0.707\,dh + 0.353\,h^2) \qquad (1)$$

where d = diameter of cage and h = valve lift. Equating $A_1 = A_2 =$ 3.396 sq. in.; substituting $d = 2$ 1-8" as above and solving for h, a value is found of $h = .63$". As the clearances between the moving parts were enough to allow of this amount of lift, it was decided upon as the maximum lift to be effected by the new inlet cams.

The derivation of this equation is given in "The Gasoline Automobile", by P. M. Heldt, on page 219, Vol. I and is as follows: Let d = the diameter of the bore of the valve seat and let h = the amount of valve lift in inches, as in Fig. 21. Also let x = the angle which the valve seat makes with a plane perpendicular to the axis of the valve stem. Now, if the valve and seat were flat, the

FIGURE 21.

passage when open would be in the form of a right circular cylinder but due to the conical shape of the valve it is actually the frustrum of a cone. The area of this truncated cone is therefore the area of port opening. Referring to Fig. 21 the line BC = B'C' is an element of the conical surface whose area is to be ascertained and is equal to h cos X, also CE = BC cos X = h cos X sin X. Therefore the diameter CC' is equal to d + 2h cos X sin X. The area of the conical surface is, $\pi \frac{CC' + BB'}{2} \times BC$

$$= \pi(d + h \cos X \sin X)\, h \cos X$$

$$= \pi(dh \cos X + h^2 \cos^2 X \sin X)$$

In this case the angle X is equal to 45°. Substituting, cos 45° = sin 45° = .707, the equation becomes,

$$A_2 = \pi(0.707\, dh + 0.353\, h^2) \qquad (1)$$

in which A_2 is the area of port opening.

The method of determining the outline of a cylindrical cam to produce a motion of constant accelleration was taken from "The Gasoline Automobile", by E. M. Heldt, page 230, Vol.I. It is as follows:- Considering that the cam revolves at a constant angular velocity, if the vertical motion of a point in contact with the perimeter in a line through the axis of the cam is to proceed with a constant accelleration, then the displacement or lift, h, corresponding to the angular displacement of the cam through an angle A between any two positions will be proportional to the square of the time, t, required for the cam to revolve through this angle. This relation may be expressed by the equation,

$$h = Ct^2 \qquad (2)$$

in which C is the constant of proportionality; or, A being directly proportional to t,

$$h = C'A^2 \qquad (3)$$

as the motion of the valve is to have a constant accelleration during one half of the period of opening and an equal but opposite accelleration or decelleration during the other half, the value of the constant, C', is found by equating it to one half of the maximum lift divided by the square of the angle through which it is to turn, in order to produce this amount of lift. Thus the equation becomes,

$$C' = \frac{h}{2A^2} \qquad (4)$$

In this instance h = .63" and A = 20°, as the total period of opening covers the time required for the cam to turn through 40°. Sub-

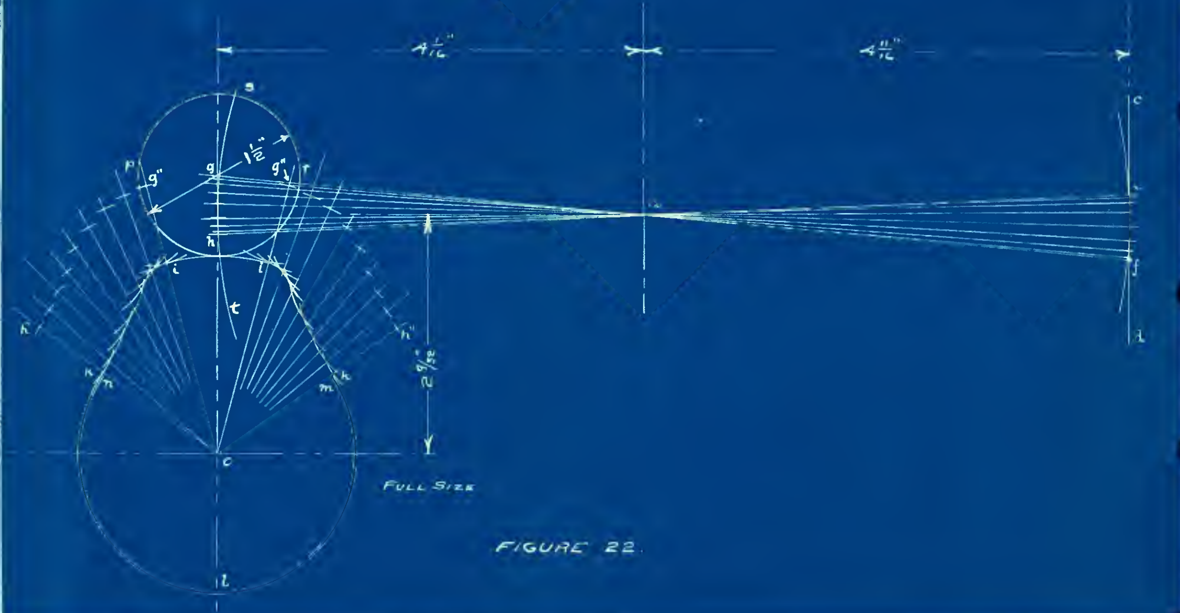


FIGURE 22.

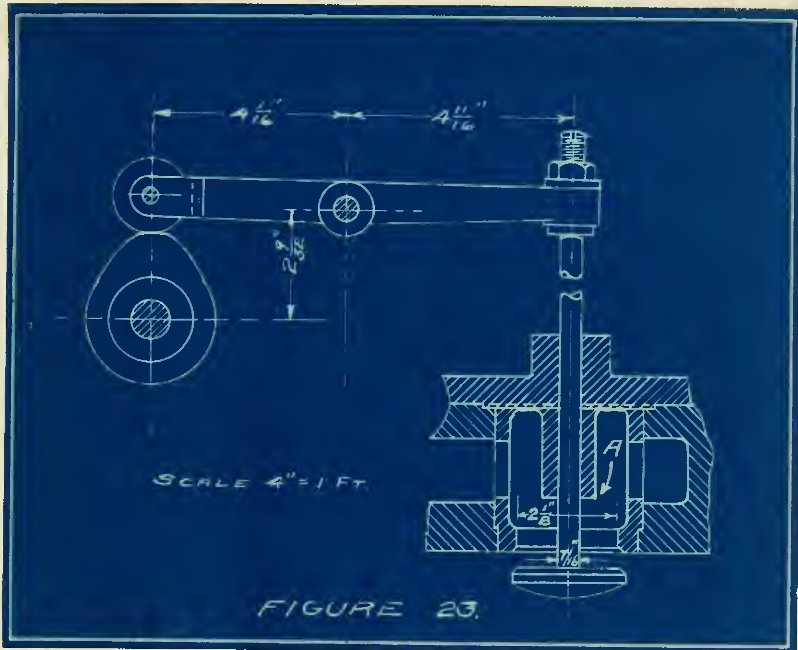


FIGURE 23.

of the roller pr. These points now represent the positions of the axis of the roller corresponding to the cam radii iO-- nO. Acrs were next drawn about O as a center and having radii equal to Og -- Oh and intersecting the cam radii at g" --- h". With these points as centers arcs were then drawn of radii equal to that of the roller and these arcs located the outline of the cam. The circle K K represents the cam radius corresponding to position C of the valve stem when the valve is closed and the circle, mln, or base circle, the actual outline of the cam in order to leave a small amount of clearance; in this case 1-32".

The same method was used in determining the outline of the exhaust cams. It was found however that no improvement could be effected here by means of increasing the maximum lift without more extensive alterations in the working parts of the engine than were thought warrantable on account of time limitations. This was due to the fact that although there is sufficient clearance above the valve to allow for greater lift, there is not enough clearance between the arm carrying the roller and the top of the crank case. What improvement was possible therefore in the exhaust valve motion was dependent on the periods of opening and closing and the length of dwell at maximum opening. The maximum lift was therefore taken the same as with the original-cams. This is half an inch, and by a similar set of calculations to those employed for the inlet valves, the amounts of lift corresponding to a number of cam positions were computed. The results are given in Table IV.

As the cam is to turn through an angle of 45° during period of opening, $22\frac{1}{2}^{\circ}$ while the valve remains wide open, and 45° during closing, it remained to determine the cam outline by means of a kinematic drawing. This is shown in Fig. 24 and will be readily interpreted taken together with Fig. 25. The point O represents the center of the cam and the lines bh and b'g the top of the lever working about the pivot at a. cd is the line of motion of the valve stem resting on the top of the lever. mn is the base circle of the cam and is 4" in diameter. K K represents the clearance circle, an allowance of 1-32" having been made for this purpose. The various cam positions are represented by the radial lines Og" to Oh". The circle pr is the roller against which the cam operates. This roller moves along the arc St. The arc ii is as determined above $32\frac{1}{2}^{\circ}$ and has a radius of 2.49", taken at $2\frac{1}{2}$". The various valve stem positions are laid off along cd as at e' --- f' and the intermediate points. This was done, for the sake of greater accuracy, by means of similar triangles, the line ef being drawn parallel to cd and at such a distance along bv that fv is ten times as great of the roller pin. These points may represent the positions of the axis of the roller corresponding to the cam [illegible]. Arcs were then drawn about C as a center and having radii equal to Og -- [illegible] intersecting the [illegible]. The three points so obtained [illegible] were then drawn of radii equal to that of the roller [illegible] the outline of the cam. The circle [illegible] represents the cam [illegible] corresponding to position [illegible] of the valve [illegible] the circle [illegible] the actual [illegible] of the cam in order to leave a small amount of clearance; in this case 1-[illegible]".

The same method was used in determining the lift of the exhaust cams. It was found however that no improvement could be effected by means of increasing the maximum lift without more extensive alterations in the working parts of the engine than were thought warrantable on account of time limitations. This was due to the fact that although there is sufficient clearance above the valve to allow for greater lift, there is not enough clearance between the rod carrying the roller and the top of the crank case. What improvement was possible therefore in the exhaust valve action was dependent on the position of opening and closing and the length of dwell at maximum opening. The maximum lift was therefore taken the same as with the original cams. This is built on [illegible] by a similar method and similar to those employed for the intake valves, the amounts of lift corresponding to a number of cam positions were computed. The results are given in Table IV.

As the cam is to turn through an angle of 45° during period of opening, 45° while the valve remains wide open, and 45° during closing, it remained to determine the cam outline by means of a kinematic drawing. This is shown in Fig. 24 and will be readily interpreted taken together with Fig. 23. The point C represents the center of the cam and the [illegible] the top of the lever working about the pivot at [illegible] of the [illegible] when resting on the top of the lever. [illegible] is the base circle of the cam and is 4" in diameter. [illegible] represents the clearance circle, an allowance of 1-32" being made for this purpose. The various positions are represented by the radial lines [illegible]". The circle [illegible] is the roller contact with the cam [illegible]. The roller moves along the [illegible]. The arc [illegible] is determined [illegible] and has a radius of [illegible]", taken at [illegible]. The various [illegible] positions are [illegible] points. This was done [illegible] by means of similar triangles, [illegible] of each [illegible] distance [illegible] as f'v. The valve lifts as given in Table IV were then multiplied by ten and these distances laid off along ef. The intersections of lines connecting these points to the point v, with the line cd therefore locate very accurately the positions of valve stem e'f', corresponding to the various angles. Lines through these points tangent to the circle wx and extended to some arc gh give at their intersections with gh a means of locating the various positions of the roller. This was done by describing the arcs gg' ------ hh' of equal radii to intersect the arc st at points g'---- h'. Then by describing arcs of radii equal to Og' ---- Oh' and intersecting the cam radii at g" ---- h", the positions of the center of the roller with respect to the cam are located. The intersections of arcs of radius equal to that of the roller with the cam radii give a series of points sufficient to mark the outline of the cam.

Table IV

Valve Lifts Corresponding to Cam Positions
Exhaust Cams

Angle Degrees	Lift Inches
0	0
5	0.0123
10	0.0485
15	0.1095
20	0.1940
$22\frac{1}{2}$	0.2245
25	0.2960
30	0.3809
35	0.4415
40	0.4777
45	0.4900

For purposes of comparison the outlines of the new cams are shown in Figs. 8 and 10 together with those of the old in Figs. 7 and 9. The relation between port opening and crank position as contemplated in the design is represented graphically by the solid line in Fig. 26. This diagram affords a direct comparison of the valve motion as produced by the old cams with that for which the new cams were designed. The dotted line shows the valve action

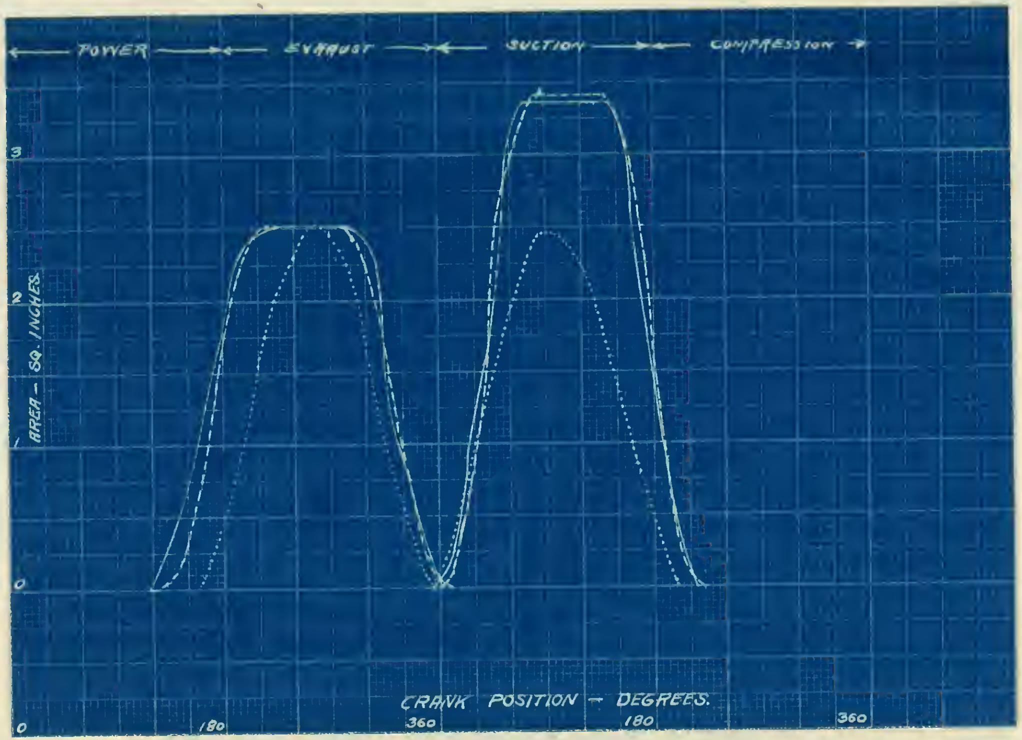

POWER
EXHAUST
SUCTION
COMPRESSION
3
2
AREA - SQ. INCHES
0
CRANK POSITION - DEGREES.
0
180
360
180
360

Figure 26

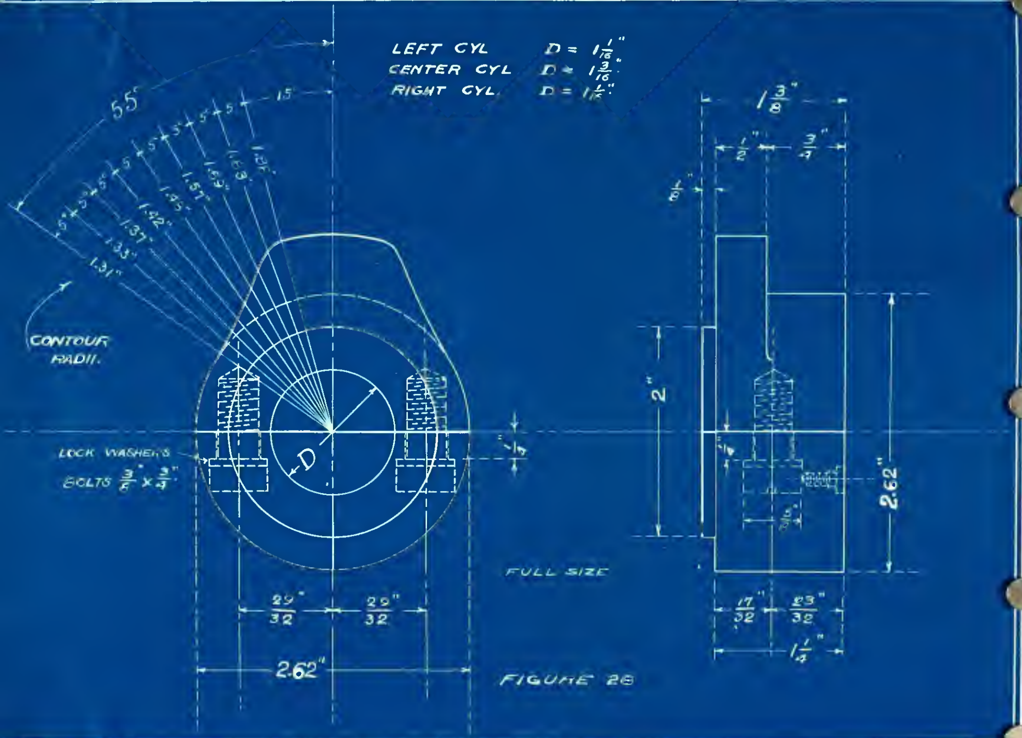
with the original cams. The solid line is plotted with values of port opening computed from the amount of valve lift in Tables III and IV and scheduled in Table V. The curve representing the old cam motion is plotted from Table II.

Table V

Valve Lift and Port Opening as per New Cam Design

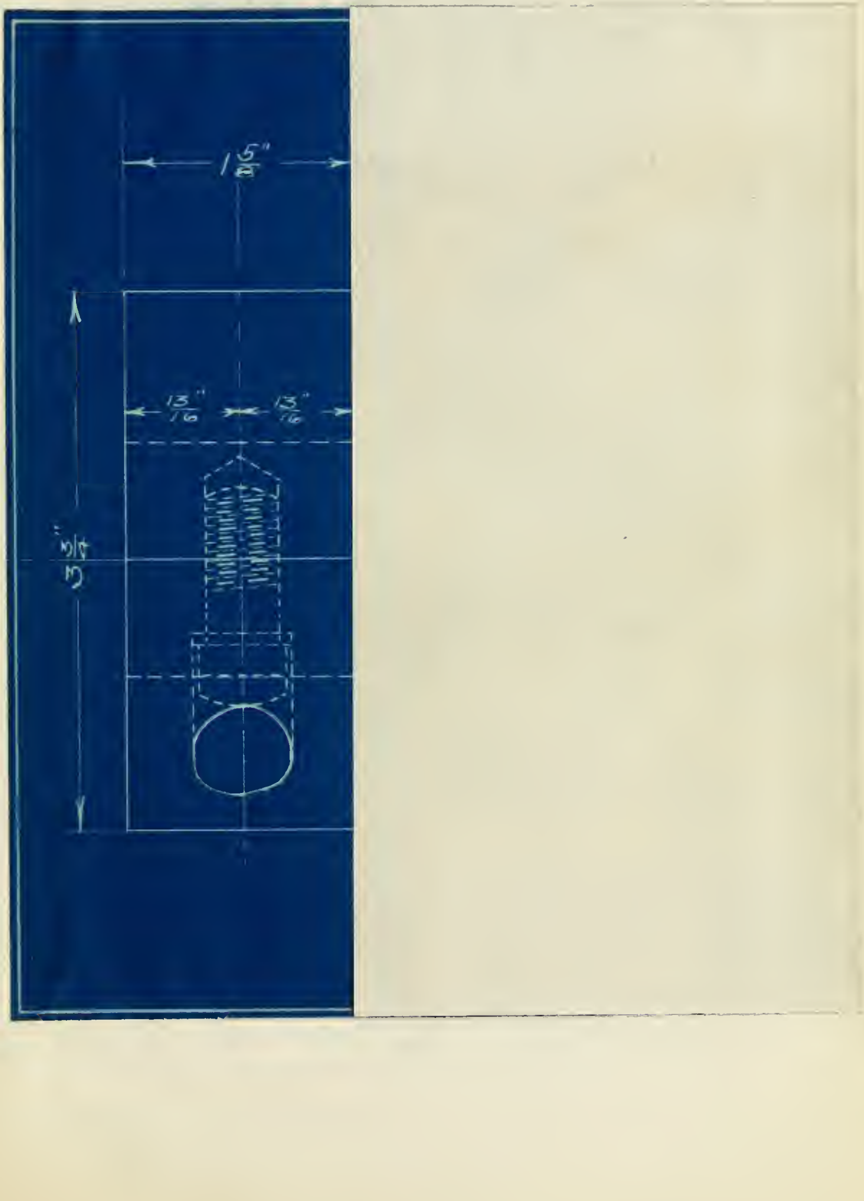
Inlet valve			Exhaust valve		
Crank Position Degrees	Valve Lift Inches	Port Opening Sq. In.	Crank Position Degrees	Valve Lift Inches	Port Opening Sq. In.
0	0	0	125	0	0
10	0.020	0.093	135	0.012	0.056
20	0.079	0.378	145	0.048	0.225
30	0.177	0.870	155	0.109	0.513
40	0.315	1.595	165	0.194	0.831
50	0.453	2.541	170	0.245	1.192
60	0.551	2.936	175	0.296	1.454
70	0.610	3.293	185	0.381	1.904
80	0.630	3.414	195	0.442	2.236
110	0.630	3.414	205	0.478	2.437
140	0.630	3.414	215	0.490	2.508
150	0.610	3.293	280	0.490	2.508
160	0.551	2.936	290	0.478	2.437
170	0.453	2.541	300	0.442	2.236
180	0.315	1.595	310	0.381	1.904
190	0.177	0.870	320	0.296	1.454
200	0.079	0.378	330	0.245	1.192
210	0.020	0.093	335	0.194	0.831
220	0	0	340	0.109	0.513
			350	0.048	0.225
			360	0.012	0.056
			370	0	0

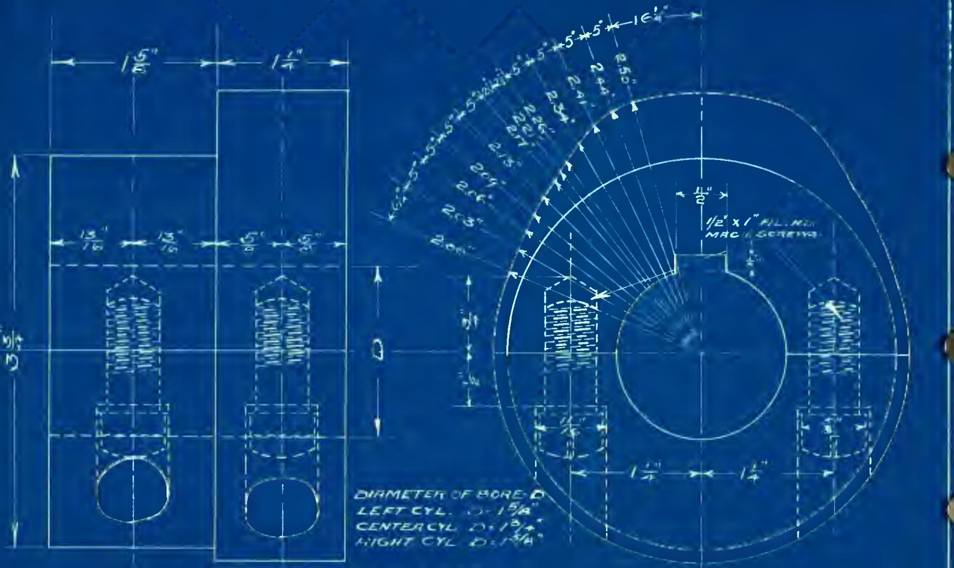
55°
5°
1.31"
1.33"
1.37"
1.42"
CONTOUR RADII.
LOCK WASHERS
BOLTS $\frac{3}{8}$" x $\frac{3}{4}$"



LEFT CYL D = 1 1/16"
CENTER CYL D = 1 3/16"
RIGHT CYL D = 1 1/16"
55°
15°
CONTOUR RADII.
LOCK WASHERS
BOLTS 3/8" x 3/4"
D
2"
2.62"
1 3/8"
1/2"
3/4"
29/32
17/32
23/32
1 1/4"
FULL SIZE

FIGURE 28


1 5/8"
13/16"
13/16"
3 3/4"



FULL SIZE

FIGURE 26

DETAIL OF
EXHAUST CAMS
THREE CYLINDER
WESTINGHOUSE GAS ENGINE

Installation and Adjustment

The keys or feathers on the cam shafts being set at angles of 120°, the new cams were fitted to the old keys so that the centers of their high points came at the same place relative to the other mechanism as the old ones. The adjustment is a matter of trial and was accomplished by turning the engine over and observing the positions of the cranks as marked on the fly wheel at the instants the events occurred. The occurrence of intake or exhaust valve events may be shifted forward or back with respect to the cranks by shifting the gears driving the cam shafts forward or back one or more teeth as required. This method has the defect that the amount of shift is restricted to a minimum corresponding to the pitch of the gear teeth. Further adjustment may be secured by changing the amount of clearance between cams and followers. An increase of clearance causes the interval between opening and closing to decrease and the opposite effect is produced by a decrease in clearance. After making a number of trials, shifting the gears and adjusting clearances, an adjustment was found which quite closely approximates the valve control contemplated in the design of the new cams. The sequence of events effected by this adjustment is given in Table VI

The actual port openings, as in the above adjustment corresponding to the various crank positions are given in Table VII and a graphical representation of this relation is superimposed, in broken line, in Fig. 26, on the corresponding curves for the old cams and the opening as contemplated in the design.

Table VI

Events	Dead Center	Cylinders		
		Left	Center	Right
Inlet opens	Upper	6° early	0	0
" closes	Lower	35° late	40° late	43° late
Ignition	Upper	12° early	18° early	18° early
Exhaust opens	Lower	59° early	63° early	60° early
" closes	Upper	8° late	6° late	6° late

Table VII

Actual Valve Lift and Port Openings as Controlled by the New Cams

Left cylinder

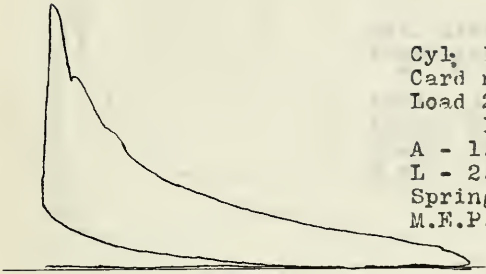
Inlet			Exhaust		
Crank Position Degrees	Valve Lift Inches	Port Opening Sq. In.	Crank Position Degrees	Valve Lift Inches	Port Opening Sq. In.
351	0	0	117	0	0
10	0.05	0.239	135	0.07	0.326
25	0.19	0.937	150	0.15	0.712
40	0.32	1.625	165	0.26	1.266
55	0.54	2.874	180	0.36	1.794
70	0.62	3.357	195	0.46	2.345
85	0.62	3.357	210	0.48	2.456
100	0.62	3.357	225	0.49	2.513
115	0.62	3.357	240	0.49	2.513
130	0.62	3.357	255	0.49	2.513
145	0.57	3.051	270	0.49	2.513
160	0.50	2.627	285	0.49	2.513
175	0.35	1.787	300	0.47	2.399
190	0.15	0.733	315	0.31	1.527
205	0.05	0.239	330	0.19	0.910
216	0	0	345	0.10	0.469
			360	0.04	0.185
			362	0	0

Conclusion

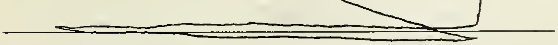
It was the original intention to make a series of final tests, after the installation of the new cams, similar to the preliminary ones. Lack of time unfortunately has made this impossible. The engine was run however under various loads enough to show that its operation was satisfactory and a few indicator cards were taken. These cards are submitted herewith, Figs. 30 to 35. Fig. 36 is an enlargement of Fig. 30. This card was taken with a net brake load of 172 pounds which corresponds at normal speed to about 54 developed horse power. The engine appeared capable of operating at this load continuously, but it probably is close to the maximum output without further alteration. Compared to the 45 horse power developed in the last run of the preliminary test, which was about all the engine could carry continuously, a gain has been made of about 20% in this respect.

Referring again to Fig. 26, this diagram serves to show at a glance the improvement effected. The diagram records the facts only as taken from the left cylinder. The area under the curve is, in the case of the inlet valve, a measure of the weight of combustible per charge and as the driving force increases with the weight the cards should show a greater mean effective pressure. In the case of the exhaust valve the area under the curve is a measure of the capacity for exhausting burned gases and is in consequence an indication of reduced exhaust velocity and back pressure.

The result of the work is epitomized in Figs. 20 and 36. A comparison of these cards bears out the conclusions drawn from the consideration of the port opening diagram. The total area of the full load diagram is greatly increased due to increased charge of combustible and consequently greater m.e.p. Both compression and maximum pressures are increased and the toe is rounded off downward. This latter modification reduces unnecessary back pressure and crank pin stresses at the lower dead center. And in addition and conclusion, in proportion to the power developed, the negative work as represented by the lower loop is reduced.

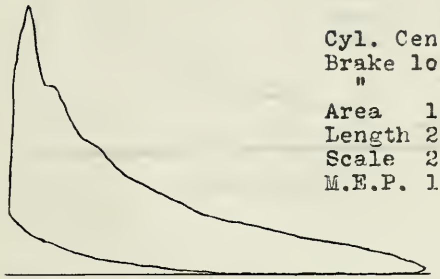


Cyl. left
Card no.1
Load 202 lb. gr.,
172 lb net.
A - 1.34
L - 2.95
Spring 240 lb.
M.E.P. 109 lb.



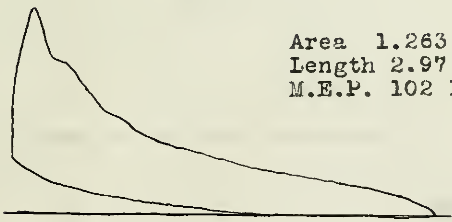
L - 2.995
Spring 20 lb.

Figure 30



Cyl. Center
Brake load 202 lb. gr.
" " 170 lb. net.
Area 1.44 sq. in.
Length 2.93 in.
Scale 240
M.E.P. 118 lb.

Figure 31



Area 1.263 sq. in.
Length 2.97 in.
M.E.P. 102 lb.

Figure 32

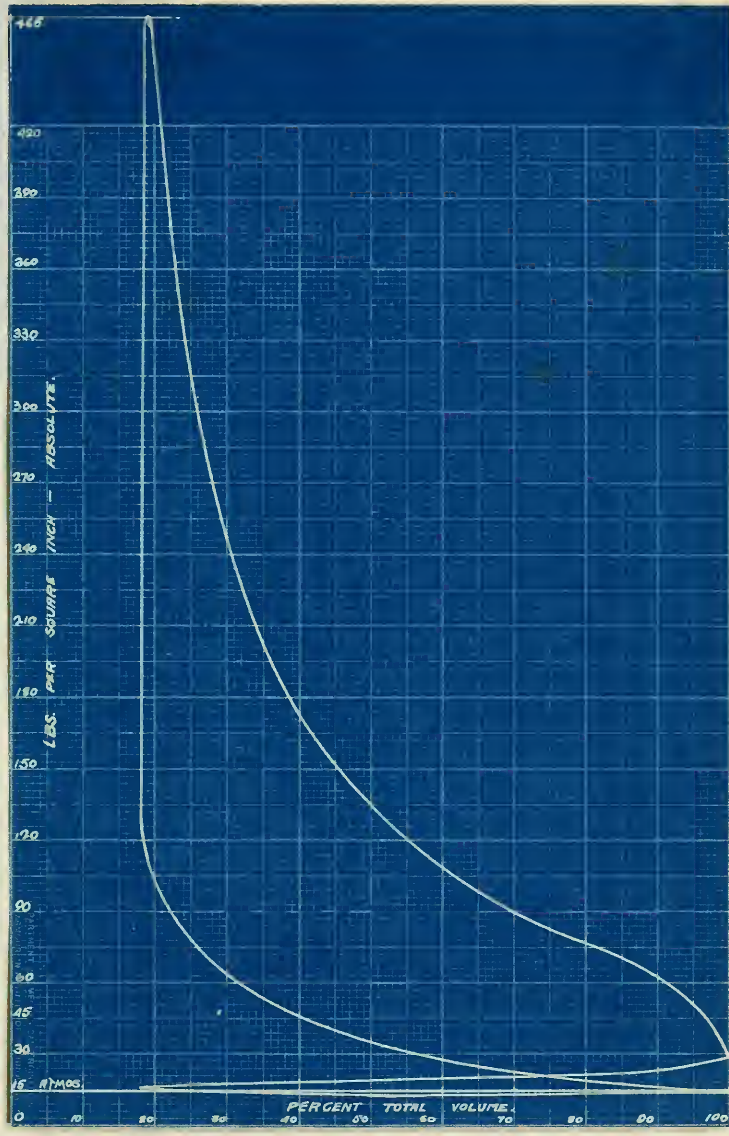

420
390
360
330
300
270
240
210
180
150
120
90
60
45
30
15 ATMOS.
0
10
20
30
40
50
60
70
80
90
100
LBS. PER SQUARE INCH — ABSOLUTE.
PERCENT TOTAL VOLUME.

Figure 36

FIGURE 25.

FIGURE 24.

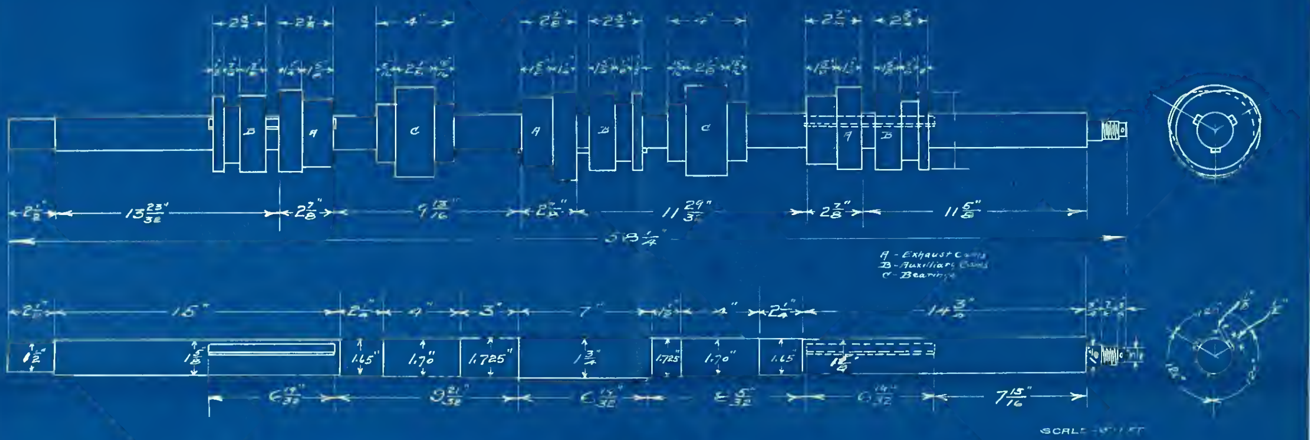


FIGURE 27.