MACHINERY'S DATA SHEETS

REVISED AND RE-ARRANGED IN LIBRARY FORM

No. 7

Shafting, Keys and Keyways

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CONTENTS

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MACHINERY'S DATA SHEET SERIES

COMPILED FROM MACHINERY'S MONTHLY DATA SHEETS AND ARRANGED WITH EXPLANATORY MATTER

No. 7

Shafting, Keys and Keyways

CONTENTS

In the following pages are compiled a number of diagrams and concise tables relating to shafting, keys and keyways, carefully selected from MACHINERY'S monthly Data Sheets, issued as supplements to the Engineering and Railway editions of MACHINERY since September, 1898. A number of additional tables also are included which are published here for the first time.

In order to enhance the value of the tables and diagrams, brief explanatory notes have been provided wherever necessary. In these notes references are made to articles which have appeared in MACHINERY, and to matter published in MACHINERY'S Reference Series, giving additional information on the subject. These references will be of considerable value to readers who wish to make a more thorough study of the subject. In a note at the foot of the tables reference is made to the page on which the explanatory note relating to the table appears.

SHAFTING, KEYS AND KEYWAYS

Horsepower Transmitted by Shafting

A question which often meets the machine designer is that of determining the horsepower that may be safely transmitted by a shaft of a given diameter at a given number of revolutions per minute. Quite as frequently the horsepower and the speed are known, and it is required to find the diameter of the shaft which will safely transmit the given power. On page 4 a table is presented giving working proportions for shafting of medium steel; this table will be found useful whenever either of the above problems are met with.

Assume, for example, that it is required to find the diameter of a shaft for transmitting 40 horsepower at a speed of 250 revolutions per minute. The shaft is not subjected to any bending action except its own weight. Consulting the table to the left on page 4 and locating 40 in the body of the table, in the column under 250 revolutions per minute, we find in the extreme left-hand column that the diameter of the required shaft should be two inches. The table also gives the maximum permissible distance between the shaft bearings, which in this case is slightly more than 14 feet.

When the exact horsepower given cannot be found in the table, it is advisable to take the nearest larger value listed in the table, and find the diameter of shaft required to transmit this horsepower.

On page 5 a table is given for finding the horsepower which can be safely transmitted by cold rolled steel line shafting. The body of the table gives the horsepower. For example, assume that a 3-inch shaft revolves at a speed of 400 revolutions per minute. What power can this shaft safely transmit? By locating 3 inches in the left-hand column, and 400 at the top of the vertical columns at the head of the page, and following the vertical column downward until opposite 3 inches, we find that under the given conditions 154 horsepower may be safely transmitted.

On pages 6 and 7 are given the horsepower which may safely be transmitted by turned steel line shafting. In this case the diameters are carried up to 12 inches. The tables on pages 5, 6 and 7 are used by the transmission department of the Jones & Laughlin Steel Co. These tables are based on the assumption that bearings are placed at intervals of from 8 to 10 feet, and that all pulleys are located near the bearings. The reason why the table for cold-rolled steel shafting is carried up only to 5 inches diameter, is that 5 inches is the largest diameter cold rolled at the present time.

Diagrams for Strength of Round Shafts

On pages 8 to 11, inclusive, are given diagrams for determining the dimensions of round shafts under different conditions. The diagram on page 8 is intended for finding the diameter of the shaft when the twisting moment and the fiber stress are known. Assume as an example that a shaft is subjected to a twisting moment of 100,000 inch-pounds, and that the allowable fiber stress is 8000 pounds per square inch. The twisting moments, in thousands of inch-pounds, are given on the scale at the bottom of the table, and the fiber stresses are represented by the diagonal

(Continued on page 16.)

HORSEPOWER TRANSMITTED BY SHAFTING

Working Proportions for Shafting of Medium Steel.												
	Transmitting power, but subject to no bending action except its own weight.						Transmitting power, and subject to bending action of pulleys, belting, etc.					
Diameter of Shaft in Inches	Revolutions per Minute.					Maximum Distance in Feet Between Bearings	Revolutions per Minute.					Maximum Distance in Feet Between Bearings
	100 H.P.	150 H.P.	200 H.P.	250 H.P.	300 H.P.		100 H.P.	150 H.P.	200 H.P.	250 H.P.	300 H.P.	
1 1/2	7	10	14	17	20	11.7	5	7	10	12	14	6.8
1 5/8	9	13	17	21	26	12.4	6	9	12	15	18	7.2
1 3/4	11	16	21	26	32	13.0	8	11	15	18	22	7.5
1 7/8	13	20	26	33	40	13.6	9	14	19	23	28	7.9
2	16	24	32	40	48	14.2	11	17	23	28	34	8.2
2 1/8	19	29	38	48	58	14.8	14	21	27	34	42	8.6
2 1/4	23	34	46	57	68	15.4	16	24	33	41	48	8.9
2 3/8	27	40	54	67	80	16.0	19	29	38	48	58	9.2
2 1/2	31	47	63	78	94	16.5	22	33	45	55	66	9.6
2 3/4	42	62	83	102	124	17.6	30	44	59	74	89	10.2
3	54	81	108	134	162	18.6	39	58	77	96	116	10.8
3 1/4	69	103	137	172	206	19.7	49	74	98	123	148	11.4
3 1/2	86	129	172	215	258	20.7	61	92	123	153	184	12.0
3 3/4	105	158	211	264	316	21.6	75	113	151	188	226	12.5
4	128	192	256	320	384	22.6	91	137	183	228	274	13.1

HORSEPOWER TRANSMITTED BY COLD ROLLED STEEL LINE SHAFTING

Diameter of Shaft	Number of Revolutions Per Minute															
	100	125	150	175	200	225	250	275	300	325	350	400	450	500	550	600
1 1/2	4.8	6.0	7.2	8.4	9.6	10.8	12.0	13.2	14.4	15.6	16.9	19.2	22	24	26	29
1 9/16	5.5	6.8	8.2	9.5	10.9	12.2	13.6	15.0	16.4	17.7	19.0	22	25	27	30	33
1 5/8	6.1	7.6	9.2	10.7	12.2	13.8	15.3	16.8	18.4	19.9	21	24	28	31	34	37
1 11/16	6.9	8.6	10.3	12.0	13.7	15.4	17.1	18.8	21	22	24	27	31	34	38	41
1 3/4	7.7	9.6	11.5	13.4	15.3	17.2	19.1	21	23	25	27	31	34	38	42	46
1 13/16	8.5	10.6	12.7	14.8	16.9	19.0	21	23	25	28	30	34	38	42	46	51
1 7/8	9.4	11.7	14.1	16.4	18.8	21	23	26	28	31	33	38	42	47	52	57
1 15/16	10.4	13.0	15.6	18.2	21	23	26	29	31	34	36	42	47	52	57	62
2	11.4	14.3	17.2	20	23	26	29	32	34	37	40	46	51	57	63	69
2 1/16	12.6	15.7	18.9	22	25	28	31	35	38	41	44	50	56	63	69	76
2 1/8	13.7	17.1	21	24	27	31	34	38	41	45	48	55	61	68	75	82
2 3/16	15.0	18.7	22	26	30	34	37	41	45	49	52	60	67	75	82	90
2 1/4	16.3	20	24	29	33	37	41	45	49	53	57	65	73	81	89	98
2 5/16	17.7	22	27	31	35	40	44	49	53	57	62	71	80	88	97	106
2 3/8	19.2	24	29	34	38	43	48	53	57	62	67	76	86	96	105	115
2 7/16	20	25	30	36	41	46	51	56	61	66	72	81	91	102	112	122
2 1/2	22	28	33	39	45	50	56	61	67	72	78	89	100	112	123	133
2 9/16	24	30	36	42	48	54	60	66	72	78	84	96	108	120	132	144
2 5/8	26	32	39	45	52	58	64	71	77	84	90	104	116	129	142	155
2 11/16	28	35	42	48	55	62	69	76	83	90	97	111	124	138	152	166
2 3/4	30	37	44	52	59	67	74	81	89	96	104	119	133	148	163	178
2 13/16	32	40	47	55	63	71	79	87	95	103	111	127	143	159	174	190
2 7/8	34	42	51	59	68	76	85	93	101	110	119	135	152	169	186	203
2 15/16	36	45	54	63	72	81	90	99	108	117	127	144	162	181	199	217
3	39	48	58	67	77	87	96	106	116	125	135	154	173	192	212	231
3 1/8	44	54	65	76	87	98	109	120	131	142	152	174	196	218	240	261
3 1/4	49	61	73	86	98	110	122	135	147	159	172	196	221	245	270	294
3 3/8	55	69	83	96	110	124	137	151	165	179	192	220	247	275	302	330
3 1/2	61	77	92	107	123	138	153	169	184	199	214	245	276	307	337	367
3 5/8	68	85	102	119	136	153	170	187	204	221	238	272	306	340	374	408
3 3/4	75	94	113	132	151	170	189	207	226	245	264	301	340	377	415	452
3 7/8	83	104	125	145	166	187	207	228	249	270	291	332	379	415	456	498
4	92	114	137	160	183	206	229	252	274	297	320	366	411	457	501	549
4 1/8	101	125	150	175	201	226	251	276	300	325	351	401	451	501	551	601
4 1/4	110	137	164	192	219	246	273	301	328	355	383	438	492	547	601	657
4 3/8	120	150	180	210	239	268	298	328	358	388	418	478	538	597	657	717
4 1/2	130	163	195	228	261	293	326	358	391	423	455	521	586	651	716	781
4 5/8	141	177	212	247	283	318	354	389	425	460	495	566	636	707	777	848
4 3/4	153	191	230	268	307	344	382	421	459	497	537	613	688	765	840	919
4 7/8	166	207	249	290	331	372	413	455	496	538	580	662	745	827	909	994
5	179	224	268	313	358	402	447	492	537	581	625	715	805	895	984	1074

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HORSEPOWER TRANSMITTED BY TURNED STEEL LINE SHAFTING

Diameter of Shaft	Number of Revolutions Per Minute															
	100	125	150	175	200	225	250	275	300	325	350	400	450	500	550	600
1 1/2	3.7	4.7	5.6	6.6	7.5	8.4	9.4	10.3	11.2	12.2	13.1	15.0	16.9	18.8	21	22
1 9/16	4.2	5.3	6.4	7.4	8.5	9.5	10.6	11.6	12.7	13.8	14.8	17.0	19.0	21	23	25
1 5/8	4.8	5.9	7.1	8.3	9.5	10.7	11.9	13.1	14.3	15.5	16.6	19.0	21	24	26	28
1 11/16	5.3	6.7	8.0	9.3	10.7	12.0	13.4	14.6	16.0	17.4	18.7	21	24	27	29	32
1 3/4	5.9	7.4	8.9	10.4	11.9	13.4	14.9	16.4	17.9	19.3	21	24	27	30	33	36
1 13/16	6.6	8.2	9.9	11.5	13.2	14.8	16.5	18.1	19.8	21	23	26	30	33	36	40
1 7/8	7.3	9.1	11.0	12.8	14.7	16.5	18.3	20	22	24	26	29	33	37	40	44
1 15/16	8.1	10.0	12.1	14.1	16.1	18.2	20	22	24	26	28	32	36	40	44	48
2	8.9	11.1	13.3	15.6	17.8	20	22	24	27	29	31	35	40	44	49	53
2 1/16	9.8	12.3	14.7	17.2	19.6	22	24	27	29	32	34	39	44	49	54	59
2 1/8	10.6	13.3	16.0	18.6	21	24	27	29	32	35	37	43	48	53	58	64
2 3/16	11.6	14.6	17.5	20.0	23	26	29	32	35	38	41	47	52	58	64	70
2 1/4	12.6	15.8	19.0	22.0	25	28	32	35	38	41	44	51	57	63	70	76
2 5/16	13.7	17.2	21	24	27	31	34	38	41	44	48	55	62	69	76	82
2 3/8	14.9	18.6	22	26	30	33	37	41	45	48	52	60	67	74	82	89
2 7/16	16.0	20	24	28	32	36	40	44	48	52	56	64	72	80	88	96
2 1/2	17.4	22	26	30	35	39	43	48	52	56	61	69	78	87	95	104
2 9/16	18.7	23	28	33	37	42	47	51	56	61	66	75	84	94	103	112
2 5/8	20	25	30	35	40	45	50	55	60	65	71	80	90	100	110	120
2 11/16	21	27	32	38	43	48	54	59	65	70	76	86	97	108	118	129
2 3/4	23	29	35	40	46	52	58	63	69	75	81	92	104	115	127	138
2 13/16	25	31	37	43	49	56	62	68	74	80	87	99	111	124	136	148
2 7/8	26	33	40	46	53	59	66	73	79	86	92	105	119	132	145	158
2 15/16	28	35	42	49	56	63	70	77	84	91	99	113	127	141	155	169
3	30	37	45	52	60	67	75	82	90	97	105	120	135	150	165	180
3 1/8	34	42	51	59	68	76	85	93	102	111	119	136	152	170	186	203
3 1/4	38	48	57	67	76	86	95	105	114	124	134	153	172	191	210	229
3 3/8	43	53	64	75	85	96	107	118	128	139	150	171	192	213	235	256
3 1/2	48	60	72	83	95	107	119	131	143	155	167	190	214	238	262	286
3 5/8	53	66	79	93	106	119	132	145	159	172	185	211	238	265	291	317
3 3/4	59	73	88	103	117	132	146	161	176	190	205	234	264	293	322	351
3 7/8	65	81	97	113	129	145	161	177	194	210	226	258	291	322	355	387
4	71	89	107	125	142	160	178	195	213	231	249	284	320	356	391	427
4 1/8	78	98	117	136	156	176	195	215	235	254	273	312	351	390	429	468
4 1/4	85	107	128	149	170	192	213	234	256	277	298	341	385	426	469	511
4 3/8	93	116	139	163	186	210	233	256	279	303	326	372	419	466	512	559
4 1/2	102	127	152	178	203	228	253	279	305	330	356	405	456	507	558	610
4 5/8	110	138	165	193	220	247	275	302	330	358	385	440	495	550	605	660
4 3/4	119	149	179	209	238	268	298	327	357	396	416	476	537	595	654	714
4 7/8	129	161	193	226	258	290	322	355	387	420	452	516	581	646	710	775
5	139	174	208	244	278	313	347	382	417	452	486	557	625	695	765	835

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HORSEPOWER TRANSMITTED BY TURNED STEEL LINE SHAFTING

Diameter of Shaft	Number of Revolutions Per Minute															
	100	125	150	175	200	225	250	275	300	325	350	400	450	500	550	600
5 1/8	150	187	225	262	300	337	375	412	450	487	525	600	675	750	825	900
5 1/4	161	201	242	281	322	362	403	443	483	523	564	644	725	805	886	966
5 3/8	172	215	259	301	344	387	430	473	516	559	602	689	775	861	947	1033
5 1/2	184	230	277	322	369	415	461	507	553	599	645	738	830	922	1014	1106
5 5/8	197	247	297	345	395	445	495	545	593	642	692	791	890	989	1088	1186
5 3/4	211	264	317	369	422	475	528	581	633	686	739	844	950	1055	1161	1260
5 7/8	225	282	339	394	451	507	564	621	677	733	789	902	1015	1128	1240	1353
6	240	300	360	419	480	540	600	661	720	780	840	960	1080	1200	1320	1440
6 1/8	255	320	384	446	511	575	639	704	766	830	894	1022	1150	1278	1405	1533
6 1/4	271	339	407	473	542	610	678	747	813	881	949	1084	1220	1355	1491	1626
6 3/8	287	360	432	502	575	647	720	792	863	935	1007	1151	1295	1439	1582	1726
6 1/2	305	382	459	535	611	687	764	841	917	993	1069	1222	1375	1528	1680	1833
6 5/8	322	403	484	564	644	725	806	887	966	1047	1127	1289	1450	1611	1772	1933
6 3/4	341	427	513	598	682	767	853	939	1023	1108	1194	1364	1535	1705	1876	2046
6 7/8	361	452	543	632	722	813	903	994	1083	1173	1264	1444	1625	1805	1986	2160
7	381	476	573	667	762	857	953	1049	1143	1238	1333	1524	1715	1905	2096	2286
7 1/8	401	502	603	702	802	902	1003	1104	1203	1303	1404	1604	1805	2005	2206	2406
7 1/4	423	529	636	742	847	952	1059	1166	1270	1375	1481	1693	1905	2116	2328	2540
7 3/8	445	557	670	782	891	1003	1115	1227	1336	1448	1559	1782	2005	2228	2450	2673
7 1/2	468	586	704	822	938	1055	1173	1291	1406	1523	1641	1875	2110	2344	2578	2814
7 5/8	492	616	740	864	987	1108	1231	1355	1477	1600	1722	1969	2215	2461	2707	2953
7 3/4	516	646	776	904	1033	1162	1293	1422	1550	1679	1808	2066	2325	2583	2841	3100
7 7/8	545	682	820	957	1091	1227	1365	1502	1637	1772	1909	2182	2455	2728	3000	3273
8	568	712	855	998	1138	1280	1423	1567	1707	1848	1991	2275	2560	2844	3128	3413
8 1/8	593	742	892	1041	1187	1335	1484	1634	1780	1928	2076	2373	2670	2966	3263	3560
8 1/4	623	780	937	1094	1247	1402	1559	1717	1870	2025	2181	2493	2805	3116	3428	
8 3/8	651	816	980	1145	1305	1467	1632	1796	1957	2119	2282	2608	2935	3261	3586	
8 1/2	681	853	1025	1197	1364	1533	1707	1879	2047	2216	2387	2728	3070	3411	3751	
8 5/8	713	892	1072	1252	1427	1605	1785	1964	2140	2318	2496	2855	3210	3566		
8 3/4	744	931	1119	1306	1489	1675	1863	2050	2233	2419	2605	2977	3350	3722		
8 7/8	766	972	1167	1363	1553	1747	1943	2139	2330	2523	2718	3106	3495	3883		
9	809	1013	1217	1421	1620	1822	2027	2231	2430	2632	2834	3240	3645			
9 1/8	844	1056	1269	1482	1689	1900	2113	2325	2533	2744	2955	3377	3800			
9 1/4	878	1099	1321	1542	1758	1977	2198	2420	2637	2855	3075	3515	3955			
9 3/8	915	1145	1376	1606	1831	2060	2291	2521	2747	2975	3204	3662				
9 1/2	951	1191	1431	1671	1904	2142	2382	2622	2858	3094	3334	3808				
9 5/8	989	1238	1488	1737	1980	2227	2477	2726	2972	3218	3464	3960				
9 3/4	1029	1288	1548	1808	2060	2317	2577	2837	3090	3346	3604					
9 7/8	1069	1338	1608	1878	2140	2407	2677	2947	3210	3476	3744					
10	1111	1388	1666	1944	2222	2500	2778	3055	3333	3611	3888					
10 1/4	1195	1497	1798	2100	2393	2692	2994	3295	3590	3888						
10 1/2	1285	1608	1934	2258	2573	2895	3219	3543	3860	4180						
10 3/4	1379	1726	2074	2422	2760	3105	3453	3800	4140	4484						
11	1477	1850	2223	2595	2958	3327	3700	4073	4437							
11 1/2	1688	2114	2540	2966	3380	3802	4247	4654	5070							
12	1918	2402	2886	3369	3840	4320	4804	5288	5760							

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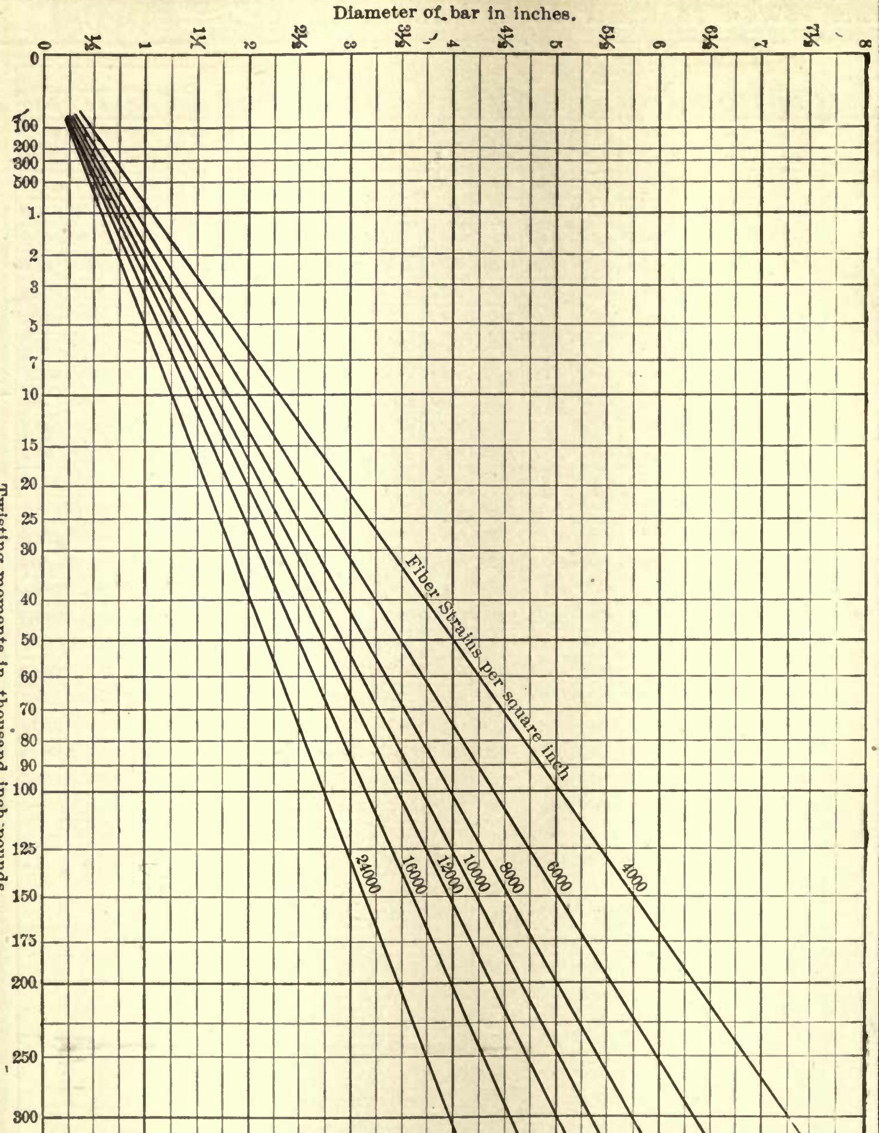
TABLE I. DIAGRAM OF TORSIONAL SHEARING STRENGTH OF ROUND SHAFTS FOR DIFFERENT FIBER STRAINS PER SQUARE INCH



Compiled by the Garry Iron & Steel Co., Cleveland, Ohio. MACHINERY'S Data Sheet No. 26. Explanatory note: Page 3.

TABLE 2. DIAGRAM OF TRANSVERSE STRENGTH OF ROUND SHAFTS FOR DIFFERENT FIBER STRAINS PER SQUARE INCH

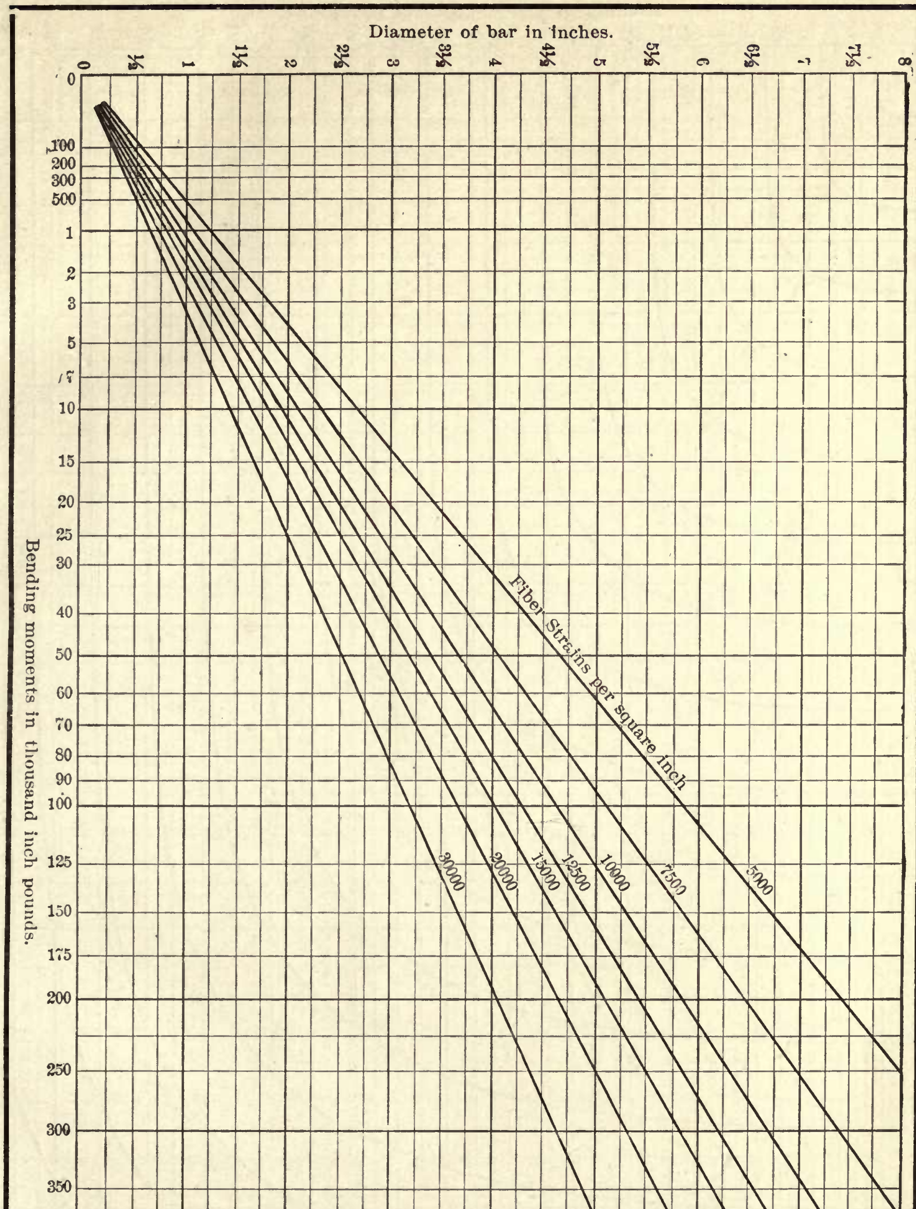


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TABLE 3. DIAGRAM OF STRENGTH OF ROUND SHAFTS FOR COMBINED TWISTING AND BENDING MOMENTS

Calculated on tensile strength of 7500 and shearing strength of 6000 pounds per square inch. For other strengths, after finding diameter by this table, get new diameter from Table No. 4.

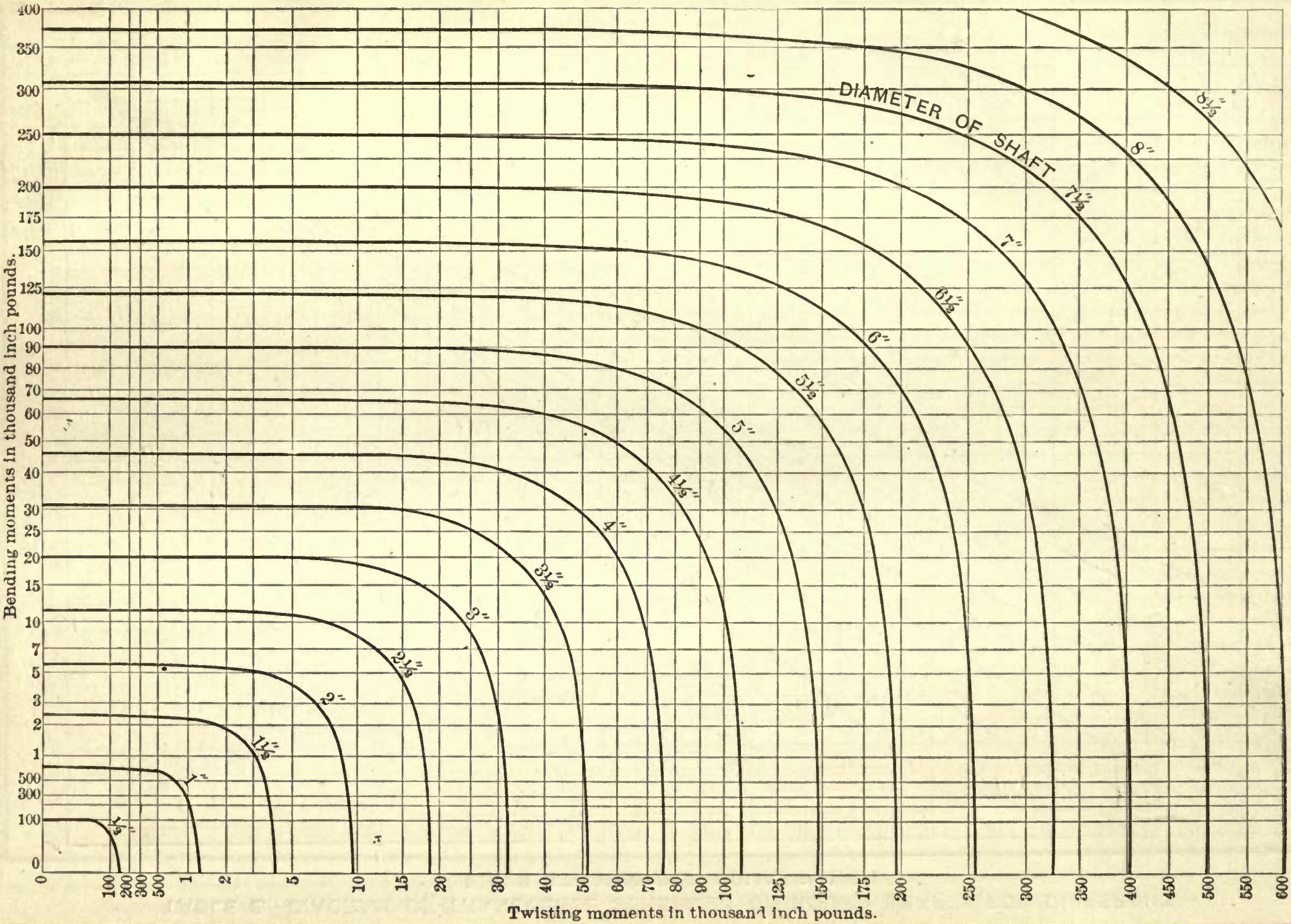
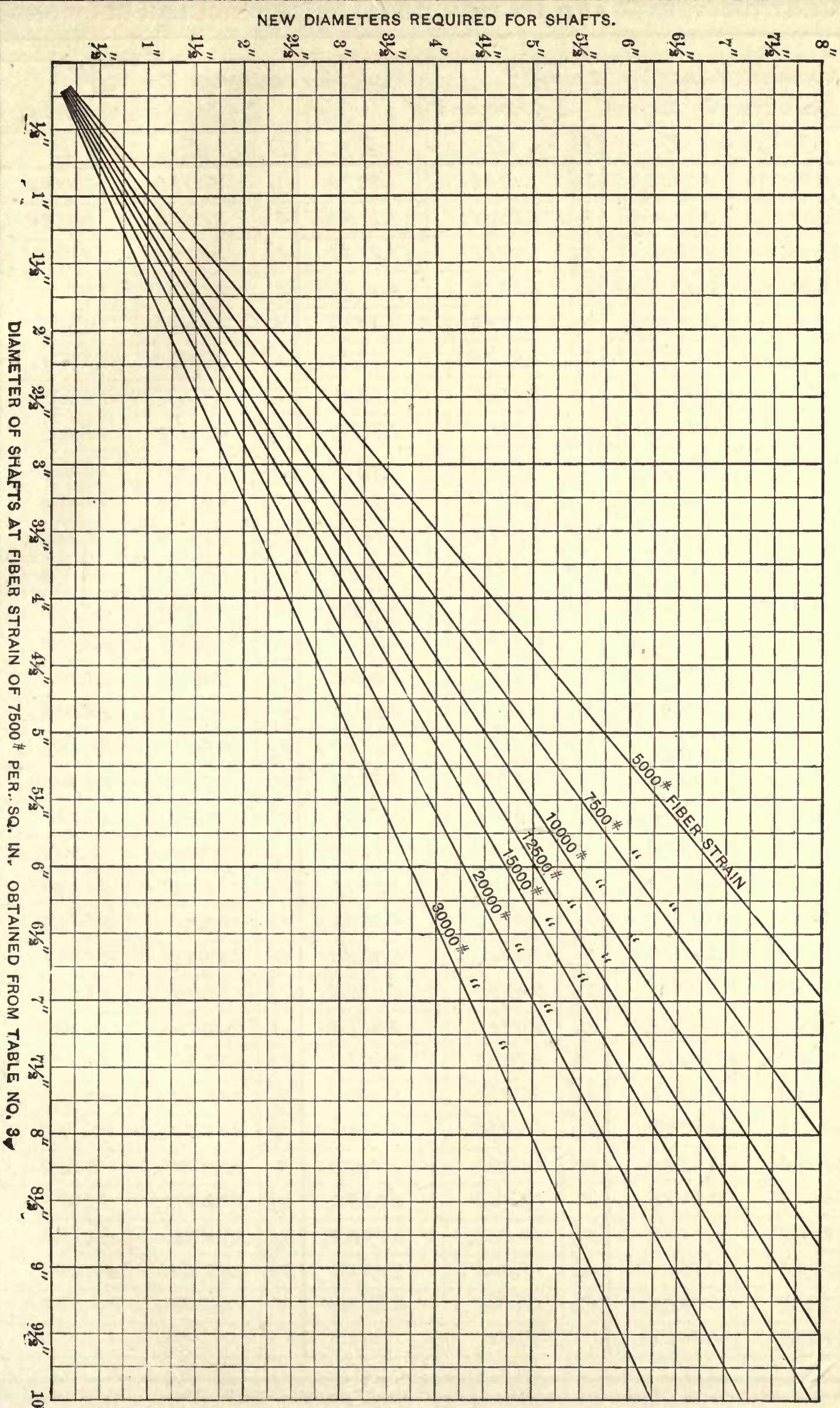


TABLE 4. DIAGRAM OF ROUND SHAFTS

For Finding Diameters at Different Fiber Strains when Diameter for Fiber Strain of 7500 pounds has been found from Table No. 3.



Compiled by the Garry Iron & Steel Co., Cleveland, Ohio. MACHINERY'S Data Sheet No. 26. Explanatory note: Page 3.

MOMENT OF INERTIA AND SECTION MODULUS OF CIRCULAR SECTIONS

Moment of Inertia $I = \frac{\pi D^4}{64}$ Section Modulus $Z = \frac{\pi D^3}{32}$

D = Diameter in Inches. I = Moment of Inertia. Z = Section Modulus.

D	I	Z	D	I	Z	D	I	Z
1/16	0.000001	0.000024	2 3/16	1.1240	1.0276	4 1/2	20.129	8.9462
1/8	0.000012	0.000192	2 1/4	1.2581	1.1183	4 5/8	22.460	9.7126
3/16	0.000061	0.000647	2 5/16	1.4038	1.2141	4 3/4	24.989	10.522
1/4	0.000192	0.001534	2 3/8	1.5618	1.3152	4 7/8	27.725	11.374
5/16	0.000468	0.002996	2 7/16	1.7328	1.4218	5	30.680	12.272
3/8	0.000971	0.005177	2 1/2	1.9175	1.5340	5 1/8	33.865	13.215
7/16	0.001798	0.008221	2 9/16	2.1166	1.6520	5 1/4	37.291	14.206
1/2	0.003068	0.012272	2 5/8	2.3307	1.7758	5 3/8	40.972	15.245
9/16	0.004914	0.017473	2 11/16	2.5607	1.9057	5 1/2	44.918	16.334
5/8	0.007490	0.023968	2 3/4	2.8074	2.0417	5 5/8	49.143	17.473
11/16	0.010967	0.031902	2 13/16	3.0714	2.1841	5 3/4	53.659	18.664
3/4	0.015532	0.041418	2 7/8	3.3537	2.3330	5 7/8	58.479	19.908
13/16	0.021393	0.052659	2 15/16	3.6550	2.4885	6	63.618	21.206
7/8	0.028774	0.065769	3	3.9761	2.6507	6 1/8	69.087	22.559
15/16	0.037919	0.080894	3 1/16	4.3179	2.8199	6 1/4	74.902	23.968
1	0.049087	0.098175	3 1/8	4.6814	2.9961	6 3/8	81.076	25.436
1 1/16	0.0626	0.1178	3 3/16	5.0673	3.1794	6 1/2	87.624	26.961
1 1/8	0.0786	0.1395	3 1/4	5.4765	3.3701	6 5/8	94.562	28.547
1 3/16	0.0976	0.1644	3 5/16	5.9101	3.5684	6 3/4	101.90	30.193
1 1/4	0.1198	0.1918	3 3/8	6.3689	3.7742	6 7/8	109.66	31.902
1 5/16	0.1457	0.2220	3 7/16	6.8540	3.9878	7	117.86	33.674
1 3/8	0.1755	0.2552	3 1/2	7.3662	4.2092	7 1/4	135.62	37.412
1 7/16	0.2096	0.2916	3 9/16	7.9066	4.4388	7 1/2	155.32	41.418
1 1/2	0.2485	0.3313	3 5/8	8.4762	4.6765	7 3/4	177.08	45.699
1 9/16	0.2926	0.3745	3 11/16	9.0761	4.9226	8	201.06	50.265
1 5/8	0.3423	0.4213	3 3/4	9.7073	5.1772	8 1/4	227.35	55.127
1 11/16	0.3980	0.4717	3 13/16	10.371	5.4404	8 1/2	256.24	60.292
1 3/4	0.4604	0.5262	3 7/8	11.068	5.7124	8 3/4	287.74	65.769
1 13/16	0.5298	0.5846	3 15/16	11.799	5.9932	9	322.06	71.569
1 7/8	0.6067	0.6472	4	12.566	6.2832	9 1/4	359.37	77.701
1 15/16	0.6918	0.7140	4 1/8	14.212	6.8908	9 1/2	399.82	84.173
2	0.7854	0.7854	4 1/4	16.015	7.5364	9 3/4	443.60	90.994
2 1/16	0.8883	0.8614	4 3/8	17.984	8.2212	10	490.87	98.175
2 1/8	1.0010	0.9421						

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SHEAR STRESSES COMBINED WITH TENSION OR COMPRESSION STRESSES

Let S = Unit Shear, t = Unit Tension or Compression,

S_m = Maximum Combined Unit Shear;

t_m = Maximum Combined Unit Tension or Compression.

Then $S_m = \sqrt{S^2 + \frac{t^2}{4}} = S\sqrt{1 + \left(\frac{1}{2}\frac{t}{S}\right)^2} = Sy.$

And $t_m = \frac{t}{2} + \sqrt{S^2 + \frac{t^2}{4}} = t\left(\frac{1}{2} + \frac{1}{2}\sqrt{\left(2\frac{S}{t}\right)^2 + 1}\right) = tx.$

$\frac{S}{t}$	Tension Factor x	Shear Factor y	$\frac{t}{S}$	Shear Factor y	Tension Factor x
0.05	1.0025	10.0499	0.05	1.0003	20.5060
0.10	1.0099	5.0990	0.10	1.0012	10.5130
0.15	1.0220	3.4801	0.15	1.0028	7.1854
0.20	1.0385	2.6926	0.20	1.0050	5.5250
0.25	1.0590	2.2361	0.25	1.0078	4.5312
0.30	1.0831	1.9437	0.30	1.0112	3.8706
0.35	1.1103	1.7438	0.35	1.0152	3.4006
0.40	1.1403	1.6008	0.40	1.0198	3.0495
0.45	1.1727	1.4948	0.45	1.0250	2.7778
0.50	1.2071	1.4142	0.50	1.0308	2.5616
0.55	1.2433	1.3515	0.55	1.0371	2.3857
0.60	1.2810	1.3017	0.60	1.0440	2.2401
0.65	1.3201	1.2616	0.65	1.0515	2.1177
0.70	1.3602	1.2289	0.70	1.0595	2.0135
0.75	1.4014	1.2019	0.75	1.0680	1.9240
0.80	1.4434	1.1793	0.80	1.0770	1.8463
0.85	1.4862	1.1602	0.85	1.0866	1.7783
0.90	1.5296	1.1440	0.90	1.0966	1.7184
0.95	1.5735	1.1300	0.95	1.1071	1.6653
1.00	1.6180	1.1180	1.00	1.1180	1.6180
1.05	1.6630	1.1076	1.05	1.1294	1.5757
1.10	1.7083	1.0985	1.10	1.1413	1.5375
1.15	1.7540	1.0904	1.15	1.1535	1.5031
1.20	1.8000	1.0833	1.20	1.1662	1.4718
1.25	1.8463	1.0770	1.25	1.1793	1.4434
1.30	1.8928	1.0714	1.30	1.1927	1.4175
1.35	1.9396	1.0664	1.35	1.2065	1.3937
1.40	1.9866	1.0619	1.40	1.2207	1.3719
1.45	2.0338	1.0578	1.45	1.2352	1.3518
1.50	2.0811	1.0541	1.50	1.2500	1.3333

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COMBINED BENDING AND TORSIONAL MOMENTS—I

All values may be read in hundreds or thousands of inch pounds.

Bending Moments in Inch Pounds = M_b		Torsional Moments in Inch Pounds = M_t: 50	100	150	200	250	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	Bending Moment
50	T	70.71	111.8	158.1	206.2	255.0	304.1	403.1	502.5	602.1	2119	2154	2193	2236	2332	2441	2561	2691	2000
	B	120.7	161.8	208.1	256.2	305.0	354.1	453.1	552.5	652.1	4119	4154	4193	4236	4332	4441	4561	4691	
75	T	90.14	125.0	167.7	213.6	261.0	309.2	407.0	505.6	604.7	704.0	803.5	2377	2417	2506	2608	2720	2843	2200
	B	165.1	200.0	242.7	288.6	336.0	384.2	482.0	580.6	679.7	779.0	878.5	4577	4617	4706	4808	4920	5043	
100	T	111.8	141.4	180.3	223.6	269.3	316.2	412.3	509.9	608.3	707.1	806.2	2563	2600	2683	2778	2884	3000	2400
	B	211.8	241.4	280.3	323.6	369.3	416.2	512.3	609.9	708.3	807.1	906.2	4963	5000	5083	5178	5284	5400	
125	T	134.6	160.1	195.3	235.9	279.5	325.0	419.1	515.4	612.9	711.1	809.7	908.6	2786	2864	2953	3053	3162	2600
	B	259.6	285.1	320.3	360.9	404.5	450.0	544.1	640.4	737.9	836.1	934.7	1034	5386	5464	5553	5653	5762	
150	T	158.1	180.3	212.1	250.0	291.5	335.4	427.2	522.0	618.5	715.9	813.9	912.4	2973	3046	3131	3225	3329	2800
	B	308.1	330.3	362.1	400.0	441.5	485.4	577.2	672.0	768.5	865.9	963.9	1062	5773	5846	5931	6025	6129	
200	T	206.2	223.6	250.0	282.8	320.2	360.6	447.2	538.5	632.5	728.0	824.6	922.0	1020	3231	3311	3400	3499	3000
	B	406.2	423.6	450.0	482.8	520.2	560.6	647.2	738.5	832.5	928.0	1025	1122	1220	6231	6311	6400	6499	
250	T	255.0	269.3	291.5	320.2	353.6	390.5	471.7	559.0	650.0	743.3	838.2	934.1	1031	1226	3770	3848	3936	3500
	B	505.0	519.3	541.5	570.2	603.6	640.5	721.7	809.0	900.0	993.3	1088	1184	1281	1476	7270	7348	7436	
300	T	304.2	316.2	335.4	360.6	390.5	424.3	500.0	583.1	670.8	761.6	854.4	948.7	1044	1237	1432	4308	4386	4000
	B	604.2	616.2	635.4	660.6	690.5	724.3	800.0	883.1	970.8	1062	1154	1249	1344	1537	1732	8308	8386	
350	T	353.6	364.0	380.8	403.1	430.1	461.0	531.5	610.3	694.6	782.6	873.2	965.7	1059	1250	1443	1638	4847	4500
	B	703.6	714.0	730.8	753.1	780.1	811.0	881.5	960.3	1045	1133	1223	1316	1409	1600	1793	1988	9347	
400	T	403.1	412.3	427.2	447.2	471.7	500.0	565.7	640.3	721.1	806.2	894.4	984.9	1077	1265	1456	1649	1844	400
	B	803.1	812.3	827.2	847.2	871.7	900.0	965.7	1040	1121	1206	1294	1385	1477	1665	1856	2049	2244	
450	T	452.8	461.0	474.3	492.4	514.8	540.8	602.1	672.7	750.0	832.2	917.9	1006	1097	1282	1471	1662	1855	450
	B	902.8	911.0	924.3	942.4	964.8	990.8	1052	1123	1200	1282	1368	1456	1547	1732	1921	2112	2305	
500	T			522.0	538.5	559.0	583.1	640.3	707.1	781.0	860.2	943.4	1030	1118	1300	1487	1676	1868	500
	B			1022	1039	1059	1083	1140	1207	1281	1360	1443	1530	1618	1800	1987	2176	2368	
600	T			618.5	632.5	650.0	670.8	721.1	781.0	848.5	922.0	1000	1082	1166	1342	1523	1709	1897	600
	B			1219	1233	1250	1271	1321	1381	1449	1522	1600	1682	1766	1942	2123	2309	2497	
700	T				728.0	743.3	761.6	806.2	860.2	922.0	990.0	1063	1140	1221	1389	1565	1746	1931	700
	B				1428	1443	1462	1506	1560	1622	1690	1763	1840	1921	2089	2265	2446	2631	
800	T				824.6	838.2	854.4	894.4	943.3	1000	1063	1131	1204	1281	1442	1612	1789	1970	800
	B				1625	1638	1654	1694	1743	1800	1863	1931	2004	2081	2242	2412	2589	2770	
900	T					934.1	948.7	984.9	1030	1082	1140	1204	1273	1345	1500	1664	1836	2012	900
	B					1834	1849	1885	1930	1982	2040	2104	2173	2245	2400	2564	2736	2912	
1000	T					1031	1044	1077	1118	1166	1221	1281	1345	1414	1562	1720	1887	2059	1000
	B					2031	2044	2077	2118	2166	2221	2281	2345	2414	2562	2720	2887	3059	
1100	T						1140	1170	1208	1253	1304	1360	1421	1487	1628	1780	1942	2110	1100
	B						2240	2270	2308	2353	2404	2460	2521	2587	2728	2880	3042	3210	
1200	T						1237	1265	1300	1342	1389	1442	1500	1562	1697	1844	2000	2163	1200
	B						2437	2465	2500	2542	2589	2642	2700	2762	2897	3044	3200	3363	
1400	T							1456	1487	1523	1565	1612	1664	1720	1844	1980	2126	2280	1400
	B							2856	2887	2923	2965	3012	3064	3120	3244	3380	3526	3680	
1600	T							1649	1676	1709	1746	1789	1836	1887	2000	2126	2263	2608	1600
	B							3249	3276	3309	3346	3389	3436	3487	3600	3726	3863	4208	
1800	T								1868	1897	1931	1970	2012	2059	2163	2280	2408	2546	1800
	B								3668	3697	3731	3770	3812	3859	3963	4080	4208	4346	

M_t = Torsional Moment
M_b = Bending Moment
T = Ideal Torsional Moment
B = Ideal Bending Moment

$T = \sqrt{M_t^2 + M_b^2}$ and $B = M_b + \sqrt{M_t^2 + M_b^2}$

These ideal moments being for use with the Polar Section Modulus.

The values at right of heavy zig-zag line across upper right-hand corner are for the bending moments 2,000 to 4,500, inclusive, in right-hand column.

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COMBINED BENDING AND TORSIONAL MOMENTS—II

Bending Moments in Inch Pounds = M_b		Torsional Moments in Inch Pounds = M_t: 2000	2200	2400	2800	3200	3600	4000	4400	4800	5200	5600	6000	7000	8000	9000	10000	11000	12000
600	T	2088	2280	2474	2864	3256	3650	4045	4441	4837	5235	5632	6030	7026	8023	9020	10018	11016	12015
	B	2688	2880	3074	3464	3856	4250	4645	5041	5437	5835	6232	6630	7626	8623	9620	10618	11616	12615
700	T	2119	2301	2500	2886	3276	3667	4061	4455	4851	5247	5644	6041	7035	8031	9027	10025	11022	12020
	B	2819	3001	3200	3586	3976	4367	4761	5155	5551	5947	6344	6741	7735	8731	9727	10725	11722	12720
800	T	2154	2341	2530	2912	3298	3688	4076	4472	4866	5261	5657	6053	7046	8040	9035	10032	11029	12027
	B	2954	3141	3330	3712	4098	4488	4876	5272	5666	6061	6457	6853	7846	8840	9835	10832	11829	12827
900	T	2193	2377	2563	2941	3324	3711	4100	4491	4884	5277	5672	6067	7058	8051	9045	10040	11037	12034
	B	3093	3277	3463	3841	4224	4611	5000	5391	5784	6177	6572	6967	7958	8951	9945	10940	11937	12934
1000	T	2236	2419	2600	2973	3353	3736	4123	4512	4903	5295	5689	6083	7071	8062	9055	10050	11045	12042
	B	3236	3419	3600	3973	4353	4736	5123	5512	5903	6295	6689	7083	8071	9062	10055	11050	12045	13042
1100	T	2283	2460	2640	3008	3384	3764	4149	4535	4924	5315	5707	6100	7086	8075	9067	10060	11055	12050
	B	3383	3560	3740	4108	4484	4864	5249	5635	6024	5415	6807	7100	8186	9175	10167	11160	12155	13150
1200	T	2332	2506	2683	3046	3418	3795	4176	4561	4948	5337	5764	6119	7102	8090	9080	10072	11065	12060
	B	3532	3706	3883	4246	4618	4995	5376	5761	6148	6537	6964	7319	8302	9290	10280	11272	12265	13260
1400	T	2441	2608	2778	3131	3493	3863	4238	4617	5000	5385	5772	6161	7139	8122	9108	10097	11089	12081
	B	3841	4008	4178	4531	4893	5263	5638	6017	6400	6785	7172	7561	8539	9522	10580	11497	12489	13481
1600	T	2561	2720	2884	3225	3578	3940	4308	4682	5060	5441	5824	6210	7181	8158	9141	10127	11116	12106
	B	4161	4320	4484	4825	5178	5540	5908	6282	6660	7041	7424	7810	8781	9758	10741	11727	12716	13706
1800	T	2691	2843	3000	3329	3672	4025	4386	4754	5126	5503	5882	6264	7228	8200	9178	10161	11146	12134
	B	4491	4643	4800	5129	5472	5825	6186	6554	6926	7303	7682	8064	9028	10000	10978	11961	12946	13934
2000	T	2828	2973	3124	3441	3774	4119	4472	4837	5200	5571	5946	6325	7141	8246	9220	10198	11180	12166
	B	4828	4973	5124	5441	5774	6119	6472	6837	7200	7571	7946	8325	9141	10246	11220	12198	13180	14166
2200	T	2973	3111	3256	3561	3883	4219	4565	4919	5280	5646	6019	6391	7338	8297	9265	10239	11218	12200
	B	5173	5311	5456	5761	6083	6419	6765	7119	7480	7846	8219	8591	9538	10497	11465	12439	13418	14400
2400	T	3124	3256	3394	3688	4000	4327	4665	5012	5367	5727	6093	6462	7400	8352	9315	10284	11259	12238
	B	5524	5656	5794	6088	6400	6727	7065	7412	7767	8127	8493	8862	9800	10752	11715	12684	13659	14638
2600	T	3280	3406	3538	3821	4123	4441	4771	5111	5459	5814	6174	6532	7467	8412	9368	10333	11303	12278
	B	5880	6006	6138	6421	6723	7041	7371	7711	8059	8414	8774	9132	10067	11012	11968	12933	13903	14878
2800	T	3441	3561	3688	3960	4252	4561	4883	5216	5557	5906	6261	6621	7533	8476	9426	10385	11351	12322
	B	6241	6361	6488	6760	7052	7361	7683	8016	8357	8706	9061	9421	10333	11276	12226	13185	14151	15122
3000	T	3601	3720	3842	4104	4386	4868	5000	5325	5660	6003	6353	6708	7616	8544	9487	10440	11402	12364
	B	6601	6720	6842	7104	7386	7868	8000	8325	8660	9003	9353	9708	10616	11544	12487	13440	14402	15364
3500	T	4031	4134	4244	4482	4742	5021	5315	5622	5941	6268	6604	6946	7826	8732	9657	10609	11543	12500
	B	7531	7634	7744	7982	8242	8521	8815	9122	9441	9768	10104	10446	11326	12232	13157	14109	15043	16000
4000	T	4472	4565	4665	4883	5123	5381	5657	5946	6248	6561	6882	7211	8062	8944	9849	10770	11705	12649
	B	8472	8565	8665	8883	9123	9381	9657	9946	10248	10561	10882	11211	12062	12944	13849	14770	15705	16649
4500	T	4924	5009	5100	5300	5522	5763	6021	6294	6580	6877	7184	7500	8322	9179	10062	10966	11885	12816
	B	9424	9509	9600	9800	10022	10263	10521	10794	11080	11377	11684	12000	12822	13679	14562	15466	16385	17316
5000	T	5385	5463	5546	5731	5936	6161	6403	6660	6931	7214	7507	7810	8602	9434	10296	11180	12083	13000
	B	10385	10463	10546	10731	10936	11161	11403	11660	11931	12214	12507	12810	13602	14434	15296	16180	17083	18000
5500	T	5852	5924	6001	6172	6363	6573	6801	7043	7300	7569	7849	8139	8902	9708	10548	11413	12298	13200
	B	11352	11424	11501	11672	11863	12073	12301	12543	12800	13069	13349	13639	14402	15208	16048	16913	17798	18700
6000	T	6325	6391	6462	6621	6800	6987	7211	7440	7684	7940	8207	8485	9220	10000	10817	11662	12530	13416
	B	12325	12391	12462	12621	12800	12987	13211	13440	13684	13940	14207	14485	15220	16000	16817	17662	18530	19416

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lines. Locate, therefore, 100 on the lower scale, and follow the line from the point so located upward until intersecting the diagonal line marked 8000. From the point of intersection follow the horizontal line to the scale at the left-hand side marked "Diameter of bar in inches." It will be seen that a shaft 4 inches in diameter is required.

On page 9 is given a diagram of transverse strength of round shafts for different fiber stresses. Assume in this case that a shaft is subjected to a bending moment of 80,000 inch-pounds and that a fiber stress of 12,500 pounds per square inch is allowable. The bending moments in thousands of inch-pounds are given on the scale at the bottom of the diagram, and the fiber stresses are represented by the diagonal lines, the same as in the previous diagram; hence by locating 80 on the lower scale and following the vertical line from the point so located until it intersects the diagonal line marked 12,500, and from the point of intersection following the horizontal line to the left, we find that the diameter of the required shaft is 4 inches.

On page 10 is given a diagram for the strength of round shafts subjected to a combined twisting and bending moment. This diagram is calculated for a tensile strength of 7500, and a torsional shearing strength of 6000 pounds per square inch. The twisting moment in thousands of inch-pounds is located on the scale at the bottom of the diagram, the bending moments are located on the scale at the left-hand side, and the diameter of the required shaft is determined by the curve which comes nearest to the intersection between the vertical line from the twisting moment and the horizontal line from the bending moment. Assume, as an example, that a shaft is subjected to a twisting moment of 175,000 inch-pounds and a bending moment of 90,000 inch-pounds. The two lines corresponding to these values are found to intersect very nearly on the 6-inch curve. A shaft 6 inches in diameter is thus required.

On page 11 an auxiliary diagram to that on page 10 is given, from which the required diameter of round shafts may be found for other fiber stresses than 7500 pounds per square inch, for which the diagram on page 10 is made up. When using this table, the diameter for a fiber stress is 7500 pounds per square inch is first found from page 10. This diameter is then located on the lower scale in the diagram on page 11. The vertical line from the point so located is followed until it intersects the diagonal line representing the allowable fiber stress; from the point of intersection a horizontal line is then followed to the left-hand scale, where the corrected diameter for the permissible fiber stress is read off. For example, if we have found from the diagram on page 10 that for given conditions a six-inch shaft is required at a fiber stress of 7500 pounds per square inch, we find from this diagram that if we increase the stress to 12,500 pounds, a shaft 5 inches in diameter would be sufficient. [MACHINERY, September, 1905, Computing Hollow and Solid Shafting.]

Moment of Inertia and Section Modulus of Circular Sections

When calculating the strength of shafting, tables of the moment of inertia and section modulus of circular sections, for diameters varying by small fractions of an inch, are very convenient. On page 12 such a table is given. The values in this table are used when the shaft is subjected to bending moments only. For torsional moments the polar moment of inertia and section modulus should be used; but since these quantities are, in this specific case, exactly double those given in the table, the tabulated values may simply be multiplied by 2 in cases where torsional moments are dealt with.

The use of the table can be best illustrated by an actual problem. Assume that the maximum combined bending moments on a shaft are 52,900 inch-pounds. Using a fiber stress not exceeding 10,000 pounds per square inch, what size shaft would be required? The section modulus in this case is

$$Z = \frac{52,900}{10,000} = 5.29.$$

Referring now to the table on page 12 we find that the diameter corresponding to this section modulus is 3 13/16 inches, approximately. This diameter corresponds to a value of $Z = 5.44$, and is thus on the side of safety. [MACHINERY, May, 1908, Maximum Stresses.]

Shear Stresses Combined with Tension or Compression Stresses

The question of shearing stresses combined with tension or compression stresses is one which always causes considerable difficulty. On page 13 a table of factors is given by means of which the maximum combined unit shear and the maximum combined unit tension or compression may be determined when the forces causing shear and tension or compression are known. For example, assume that S (see table on page 13) = 9000, and t = 12,000 pounds per square inch; then $\frac{S}{t} = 0.75$, and from the table we find that the tension (or compression) factor x then equals 1.40. This means that if the shear is 75 per cent of the tension, the maximum combined tension will be 1.40 times what it would have been if there had been no shear. This table makes it possible to quickly determine the maximum stresses in shafts subjected to combined tension and compression stresses, provided the separate unit stresses are known. [MACHINERY, March and April, 1904, Notes on Design; May, 1908, Maximum Stresses; MACHINERY'S Reference Series No. 12, Mathematics of Machine Design, Chapter I: Machinery Shafting.]

Table of Combined Bending and Torsional Moments

One of the most familiar examples of combined stresses in shafting is that of torsion and bending, the torsional stresses being shearing stresses, and the bending stresses being tension and compression stresses. The maximum stress may be found by calculating each separately, and combining them by the aid of the table on page 13, as already mentioned. The tables on pages 14 and 15 also may be used for more directly combining these stresses. If the bending and the torsional moments, both in inch-pounds, are known, they are located at the left-hand side and at the top of the tables, as indicated. The body of the tables then gives the maximum or "ideal" torsional moment in the line marked T to the left, and the maximum or "ideal" bending moment in the lines marked B. For example, a shaft 3½ inches in diameter is subjected to a torsional moment of 36,000 inch-pounds and a bending moment of 35,000 inch-pounds. What is the combined shearing stress and the combined tension and compression stress?

Referring to the table on page 15 and remembering that all values may be multiplied by 10, we find, by locating the torsional moment 3600 (instead of 36,000) at the top of the column, and the bending moment 3500 (instead of 35,000) in the left-hand column, that the maximum twisting moment, in this case, is 50,210, and the maximum bending moment 85,210 inch-pounds. Having now found the maximum moments, we can find the maximum combined unit shear and unit tension or compression. From page 12 we find that the section modulus Z for a 3½-inch shaft is 4.209. The polar section modulus being twice this, we have $Z_p = 8.418$.

Using the notation,

Maximum combined unit shear $= S_m$,

Maximum combined unit tension or compression $= t_m$,

(Continued on page 24.)

DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—I

Fiber Stress in Pounds per square inch			Torsional Moments in Thousands of Inch Pounds.																		Fiber Stress in Pounds per square inch		
7500			0	3.75	7.5	11.25	15	18.75	22.5	30	37.5	45	52.5	60	67.5	75	90	105	120	135			7500
	10000		0	5	10	15	20	25	30	40	50	60	70	80	90	100	120	140	160	180		10000	
		12500	0	6.25	12.5	18.75	25	31.25	37.5	50	62.5	75	87.5	100	112.5	125	150	175	200	225	12500		
Bending Moments in Thousands of Inch Pounds.																					Bending Moments in Thousands of Inch Pounds.		
0	0	0	0	1 3/8	1 23/32	1 31/32	2 3/16	2 11/32	2 1/2	2 3/4	2 15/16	3 1/8	3 9/16	3 7/16	3 19/32	3 23/32	3 15/16	4 5/32	4 11/32	4 1/2	0	0	0
3.75	5	6.25	1 23/32	1 27/32	2 1/32	2 7/32	2 3/8	2 1/2	2 5/8	2 27/32	3 1/16	3 7/32	5 15/16	5 31/32	5 31/32	6	6 1/16	6 1/16	6 1/8	6 3/16	250	200	150
5.625	7.5	9.375	1 31/32	2 1/32	2 3/16	2 5/16	2 15/32	2 19/32	2 11/16	2 29/32	3 3/32	3 1/4	3 13/32	3 9/16	6 3/16	6 3/16	6 1/4	6 1/4	6 5/16	6 3/8	275	220	165
7.5	10	12.5	2 3/16	2 7/32	2 5/16	2 7/16	2 9/16	2 21/32	2 25/32	2 31/32	3 5/32	3 5/16	3 7/16	3 19/32	6 5/16	6 3/8	6 3/8	6 7/16	6 7/16	6 1/2	300	240	180
9.375	12.5	15.63	2 11/32	2 3/8	2 7/16	2 17/32	2 21/32	2 3/4	2 27/32	3 1/32	3 3/16	3 11/16	3 1/2	3 5/8	3 3/4	6 1/2	6 9/16	6 9/16	6 5/8	6 5/8	325	260	195
11.25	15	18.75	2 1/2	2 1/2	2 9/16	2 31/32	2 3/4	2 13/16	2 29/32	3 3/32	3 1/4	3 13/32	3 17/32	3 21/32	3 25/32	6 5/8	6 11/16	6 11/16	6 3/4	6 13/16	350	280	210
15	20	25	2 3/4	2 3/4	2 25/32	2 27/32	2 29/32	3	3 1/16	3 7/32	3 11/32	3 1/2	3 5/8	3 3/4	3 27/32	3 31/32	6 13/16	6 7/8	6 7/8	6 15/16	375	300	225
18.75	25	31.25	2 15/16	2 15/16	2 31/32	3 1/32	3 3/32	3 1/8	3 3/16	3 11/32	3 15/32	3 19/32	3 23/32	3 13/16	3 15/16	4 1/32	4 7/32	7 3/16	7 3/16	7 1/4	437.5	350	262.5
22.5	30	37.5	3 1/8	3 5/32	3 5/32	3 3/16	3 1/4	3 9/32	3 11/32	3 7/16	3 9/16	3 11/16	3 25/32	3 7/8	4	4 3/32	4 9/32	4 15/32	7 1/2	7 1/2	500	400	300
26.25	35	43.75	3 9/32	3 9/32	3 5/16	3 11/32	3 3/8	3 13/32	3 15/32	3 9/16	3 21/32	3 3/4	3 7/8	3 31/32	4 1/16	4 5/32	4 11/32	4 1/2	4 21/32	7 13/16	562.5	450	337.5
30	40	50	3 7/16			3 15/32	3 1/2	3 17/32	3 19/32	3 21/32	3 3/4	3 27/32	3 15/16	4 1/16	4 1/8	4 7/32	4 13/32	4 9/16	4 23/32	4 27/32	50	40	30
33.75	45	56.25	3 19/32				3 5/8	3 21/32	3 11/16	3 25/32	3 27/32	3 15/16	4 1/32	4 1/8	4 7/32	4 9/32	4 7/16	4 5/8	4 3/4	4 29/32	56.25	45	33.75
37.5	50	62.5	3 23/32				3 3/4	3 25/32	3 13/16	3 7/8	3 15/16	4 1/32	4 1/8	4 3/16	4 9/32	4 11/32	4 1/2	4 21/32	4 13/16	4 15/16	62.5	50	37.5
45	60	75	3 15/16				3 31/32	4	4 1/32	4 1/16	4 1/8	4 3/16	4 9/32	4 11/32	4 13/32	4 1/2	4 5/8	4 25/32	4 29/32	5 1/32	75	60	45
52.5	70	87.5	4 5/32				4 3/16	4 7/16	4 7/32	4 1/4	4 5/16	4 3/8	4 13/32	4 15/32	4 17/32	4 5/8	4 3/4	4 7/8	5	5 1/8	87.5	70	52.5
60	80	100	4 11/32					4 3/8	4 3/8	4 13/32	4 15/32	4 1/2	4 17/32	4 19/32	4 21/32	4 3/4	4 27/32	4 31/32	5 3/32	5 7/32	100	80	60
67.5	90	112.5	4 1/2						4 17/32	4 19/32	4 5/8	4 21/32	4 23/32	4 3/4	4 13/16	4 7/8	4 31/32	5 1/16	5 3/16	5 5/16	112.5	90	67.5
75	100	125	4 11/16						4 23/32	4 23/32	4 3/4	4 25/32	4 13/16	4 7/8	4 15/16	4 31/32	5 3/32	5 3/16	5 9/16	5 3/8	125	100	75
82 5	110	137.5	4 13/16						4 27/32	4 7/8	4 29/32	4 15/16	4 31/32	5	5 1/32	5 3/32	5 3/16	5 9/32	5 3/8	5 15/32	137.5	110	82.5
90	120	150	4 31/32							5	5 1/32	5 1/16	5 3/32	5 1/8	5 5/32	5 7/32	5 9/32	5 3/8	5 15/32	5 9/16	150	120	90
105	140	175	5 7/32							5 1/4	5 9/32	5 5/16	5 11/32	5 11/32	5 3/8	5 7/16	5 1/2	5 9/16	5 21/32	5 23/32	175	140	105
120	160	200	5 15/32								5 1/2	5 17/32	5 17/32	5 9/16	5 19/32	5 5/8	5 11/16	5 3/4	5 13/16	6	200	160	120
135	180	225	5 11/16								5 23/32	5 3/4	5 3/4	5 25/32	5 25/32	5 13/16	5 7/8	5 15/16	6	6 1/16	225	180	135

Values at right of heavy zig-zag line across upper right-hand corner are for the bending moments in upper part of three right-hand columns.

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DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—II

Fiber Stress in Pounds per square Inch: 7500, 10000, 12500. Torsional Moments in Thousands of Inch Pounds (columns). Bending Moments in Thousands of Inch Pounds (rows).

7500	10000	12500																		
		7500	0	150	165	180	210	240	270	300	330	360	390	420	450	525	600	675	750	825	900
		10000	0	200	220	240	280	320	360	400	440	480	520	560	600	700	800	900	1000	1100	1200
		12500	0	250	275	300	350	400	450	500	550	600	650	700	750	875	1000	1125	1250	1375	1500
0	0	0	0	4 11/16	4 13/16	4 31/32	5 7/32	5 15/32	5 11/16	5 7/8	6 1/16	6 1/4	6 7/16	6 9/16	6 3/4	7 1/16	7 7/16	7 11/16	8	8 1/4	8 1/2
45	60	75	3 15/16	5 5/32	5 9/32	5 3/8	5 19/32	5 13/16	6	6 3/16	6 3/8	6 1/2	6 11/16	6 13/16	6 15/16	7 5/16	7 5/8	7 7/8	8 1/8	8 3/8	8 5/8
52.5	70	87.5	4 5/32	5 1/4	5 11/32	5 15/32	5 11/16	5 7/8	6 1/16	6 1/4	6 7/16	6 9/16	6 3/4	6 7/8	7	7 5/16	7 5/8	7 15/16	8 3/16	8 7/16	8 5/8
60	80	100	4 11/32	5 5/16	5 7/16	5 17/32	5 3/4	5 15/16	6 1/8	6 5/16	6 7/16	6 5/8	6 3/4	6 7/8	7 1/16	7 3/8	7 11/16	7 15/16	8 3/16	8 7/16	8 11/16
67.5	90	112.5	4 1/2	5 13/32	5 1/2	5 5/8	5 13/16	6	6 3/16	6 5/16	6 1/2	6 5/8	6 13/16	6 15/16	7 1/16	7 3/8	7 11/16	8	8 1/4	8 1/2	8 11/16
75	100	125	4 11/16	5 15/32	5 19/32	5 11/16	5 7/8	6 1/16	6 1/4	6 3/8	6 9/16	6 11/16	6 7/8	7	7 1/8	7 7/16	7 3/4	8	8 1/4	8 1/2	8 3/4
82.5	110	137.5	4 13/16	5 9/16	5 21/32	5 3/4	5 15/16	6 1/8	6 5/16	6 7/16	6 5/8	6 3/4	6 7/8	7	7 1/8	7 1/2	7 3/4	8 1/16	8 5/16	8 1/2	8 3/4
90	120	150	4 31/32	5 21/32	5 3/4	5 27/32	6	6 3/16	6 3/8	6 1/2	6 5/8	6 3/4	6 15/16	7 1/16	7 3/16	7 1/2	7 13/16	8 1/16	8 5/16	8 9/16	8 3/4
105	140	175	5 7/32	5 13/16	5 7/8	5 31/32	6 1/8	6 5/16	6 7/16	6 5/8	6 3/4	6 7/8	7	7 1/8	7 1/4	7 9/16	7 7/8	8 1/8	8 3/8	8 5/8	8 13/16
120	160	200	5 15/32	5 31/32	6 1/16	6 1/8	6 5/16	6 7/16	6 9/16	6 11/16	6 7/8	7	7 1/8	7 1/4	7 3/8	7 5/8	7 15/16	8 3/16	8 7/16	8 11/16	8 7/8
135	180	225	5 11/16	6 1/8	6 3/16	6 1/4	6 3/8	6 9/16	6 11/16	6 13/16	6 15/16	7 1/16	7 3/16	7 5/16	7 7/16	7 11/16	8	8 1/4	8 1/2	8 11/16	8 15/16
150	200	250	5 7/8	6 1/4	6 5/16	6 3/8	6 1/2	6 5/8	6 13/16	6 15/16	7 1/16	7 3/16	7 1/4	7 3/8	7 1/2	7 3/4	8 1/16	8 5/16	8 9/16	8 3/4	8 15/16
165	220	275	6 1/16	6 7/16	6 1/2	6 9/16	6 5/8	6 3/4	6 15/16	7	7 1/8	7 1/4	7 3/8	7 1/2	7 9/16	7 7/8	8 1/8	8 3/8	8 9/16	8 13/16	9
180	240	300	6 1/4	6 9/16	6 5/8	6 11/16	6 3/4	6 7/8	7	7 1/8	7 1/4	7 5/16	7 7/16	7 9/16	7 11/16	7 15/16	8 3/16	8 7/16	8 5/8	8 7/8	9 1/16
195	260	325	6 7/16	6 11/16	6 3/4	6 13/16	6 7/8	7	7 1/8	7 3/16	7 5/16	7 7/16	7 9/16	7 5/8	7 3/4	8	8 1/4	8 1/2	8 11/16	8 15/16	9 1/8
210	280	350	6 9/16	6 13/16	6 7/8	6 15/16	7	7 1/8	7 3/16	7 5/16	7 7/16	7 1/2	7 5/8	7 3/4	7 13/16	8 1/16	8 5/16	8 9/16	8 3/4	8 15/16	9 3/16
225	300	375	6 3/4	6 15/16	7	7 1/16	7 1/8	7 1/4	7 3/8	7 7/16	7 1/2	7 5/8	7 11/16	7 13/16	7 15/16	8 1/8	8 3/8	8 5/8	8 13/16	9	9 1/4
262.5	350	437.5	7 1/16	7 1/4	7 5/16	7 3/8	7 7/16	7 1/2	7 9/16	7 5/8	7 3/4	7 13/16	7 15/16	8	8 1/8	8 5/16	8 9/16	8 3/4	8 15/16	9 1/8	9 3/8
300	400	500	7 7/16	7 9/16	7 9/16	7 5/8	7 11/16	7 3/4	7 13/16	7 7/8	8	8 1/16	8 1/8	8 3/16	8 5/16	8 1/2	8 11/16	8 7/8	9 1/8	9 5/16	9 7/16
337.5	450	562.5	7 11/16	7 13/16	7 7/8	7 7/8	7 15/16	8	8 1/16	8 1/8	8 3/16	8 1/4	8 5/16	8 7/16	8 1/2	8 11/16	8 7/8	9 1/16	9 1/4	9 7/16	9 5/8
375	500	625	8	8 1/16	8 1/8	8 1/8	8 3/16	8 1/4	8 5/16	8 3/8	8 7/16	8 1/2	8 9/16	8 5/8	8 11/16	8 7/8	9	9 3/16	9 3/8	9 9/16	9 3/4
412.5	550	687.5	8 1/4	8 5/16	8 3/8	8 3/8	8 7/16	8 7/16	8 1/2	8 9/16	8 5/8	8 11/16	8 3/4	8 13/16	8 7/8	9	9 3/16	9 3/8	9 1/2	9 11/16	9 7/8
450	600	750	8 1/2	8 9/16	8 9/16	8 5/8	8 5/8	8 11/16	8 3/4	8 3/4	8 13/16	8 7/8	8 15/16	9	9 1/16	9 3/16	9 3/8	9 1/2	9 5/8	9 13/16	10

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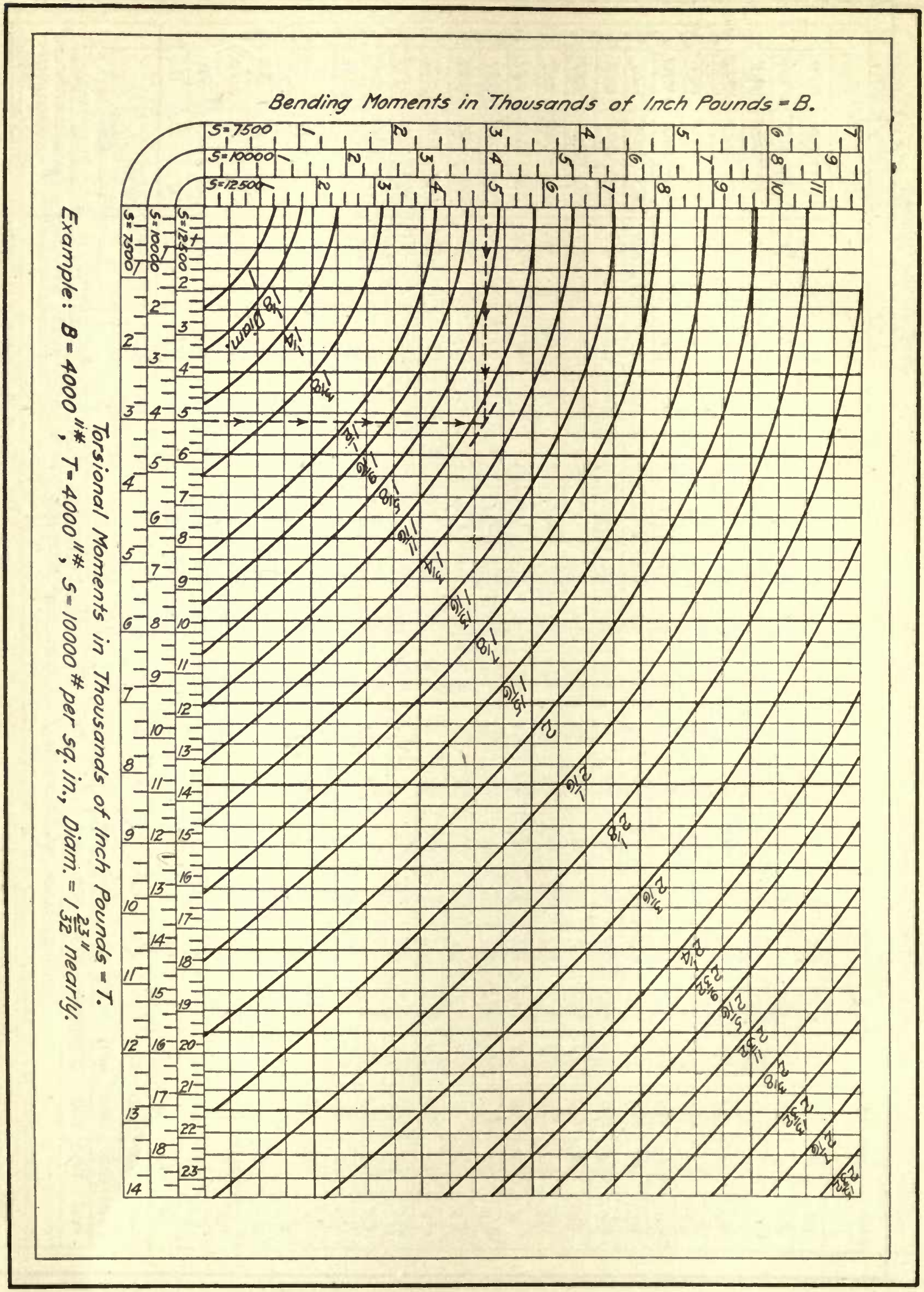
DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—I



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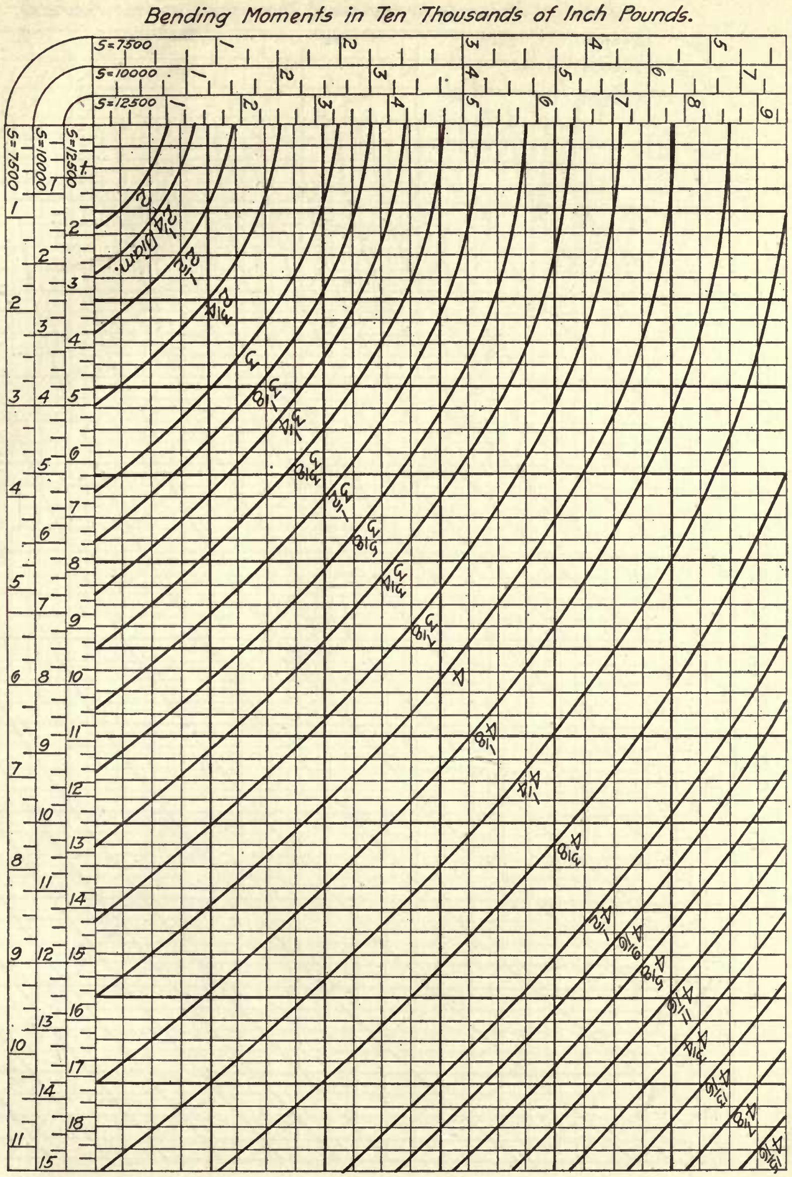
DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—II



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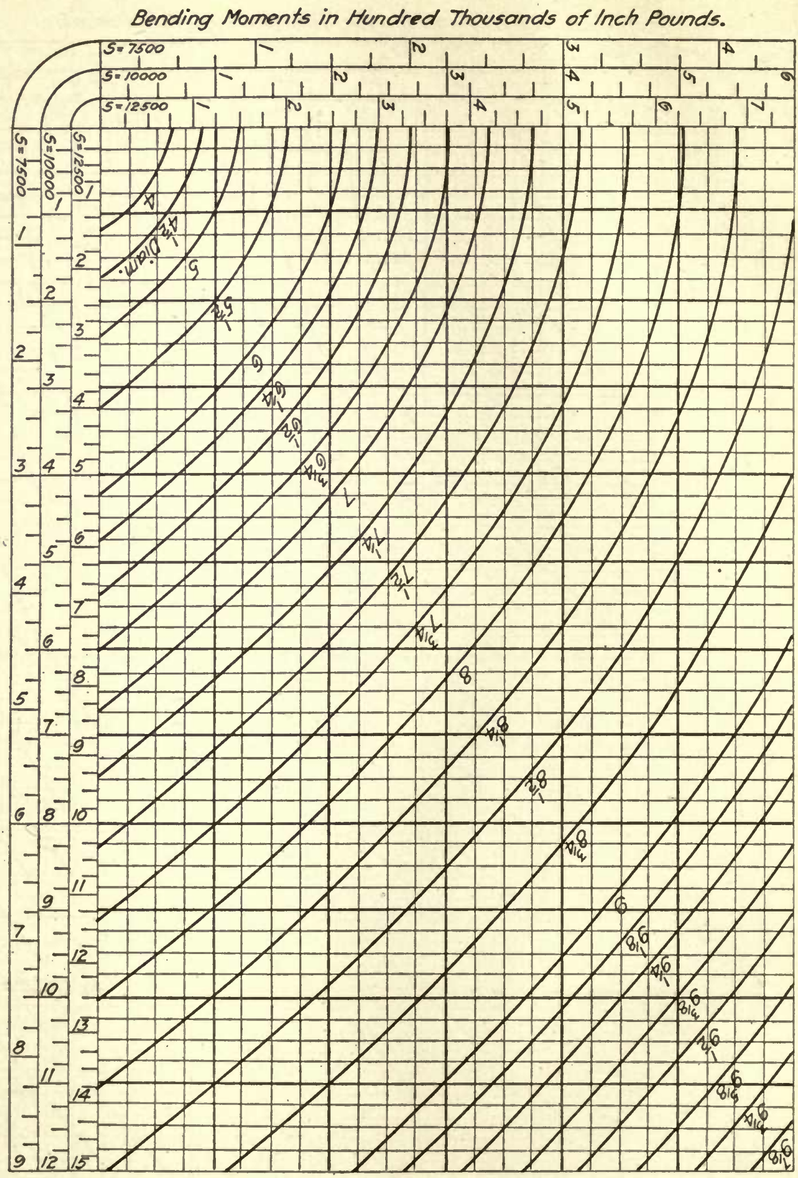
DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—III



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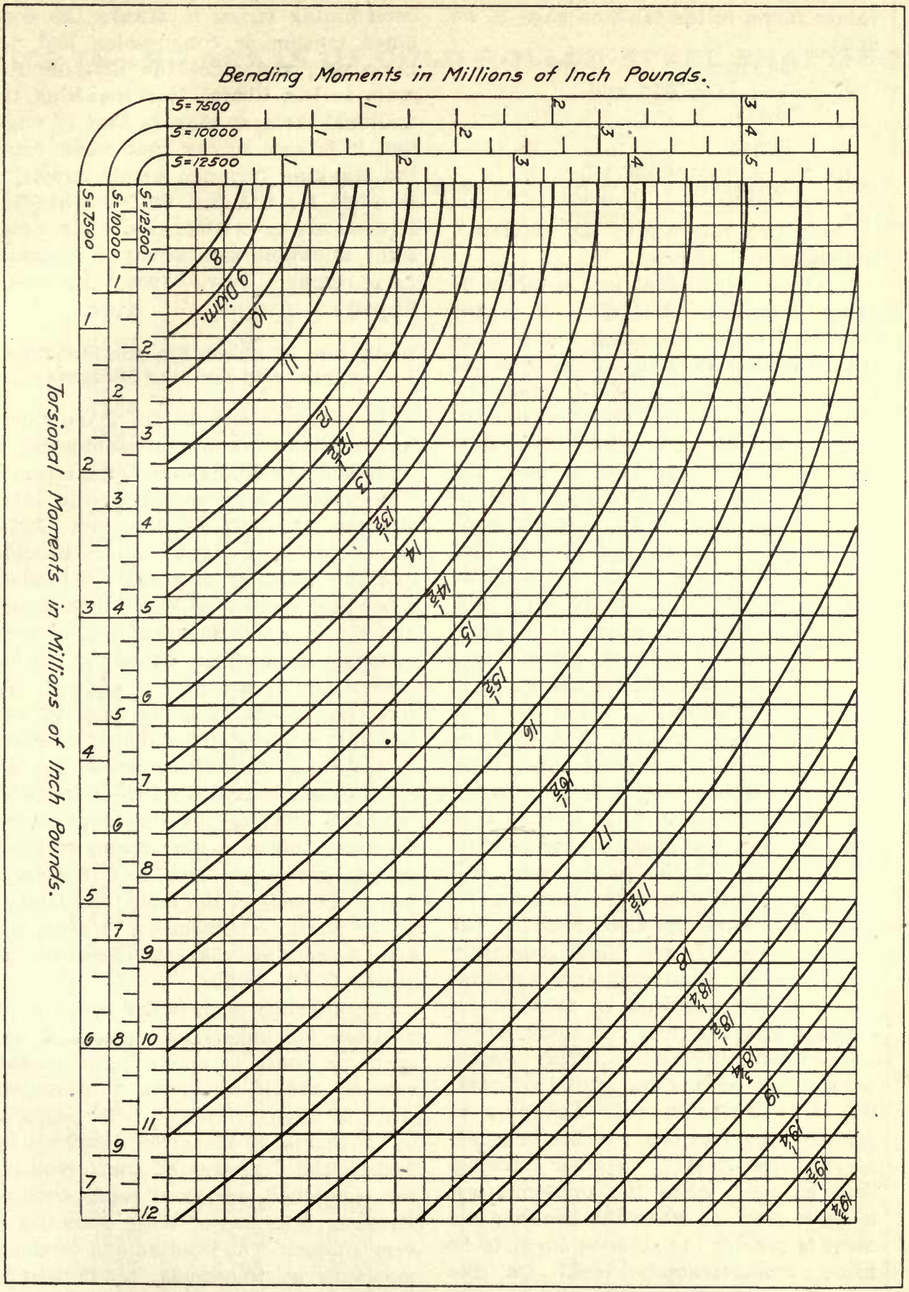
DIAMETERS OF SHAFTS FOR COMBINED TORSION AND BENDING STRESSES—VI



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and proceeding to make use of the values found in the table on page 15 we find:

$$S_m = \frac{50,210}{8.418} = 5970, \text{ and}$$

$$t_m = \frac{85,210}{8.418} = 10,120.$$

These two values give the maximum combined unit stresses.

It will be noted that in the tables on pages 14 and 15 the values B of the maximum or ideal bending moments are always greater than the values T of the maximum or ideal torsional moments. Hence it is the combined tension or compression stresses which determine the size of the section to be used, and the maximum torsional moment may be entirely neglected. All authorities do not agree on the subject of combined torsion and bending. The tables given agree with the formulas given by Rankine. The formula given by Grashof gives a torsional moment which has a greater value than that obtained from the Rankine formula. This latter, however, is commonly used, and shafting designed from calculations based upon this formula has proved satisfactory.

In this connection it is well to note that in the case of shafting, the location and direction of the tooth loads, belt pulls, etc., which produce bending, remain fixed while the shaft rotates. The bending stresses are thus constantly varying in direction, and since a greater factor of safety should be used for reversible stresses than for those which are constant in direction, many designers recommend that the allowable working stresses should vary according to whether the torsional or bending moment predominates. Higher stresses may be used when the torsional moment is greater; when the bending moment is greater the stresses ought to be made proportionately less. On the other hand the ultimate tensile stress is approximately 25 per cent greater than the ultimate shearing stress, and as the determining stress is always the combined tension or compression and not the shear, and since the Rankine formula is less liberal in recognizing the torsional moment than is that of Grashof, it is safe to say that when using the Rankine formula, ample provision is made for the fact that the bending stresses are reversible, even when a constant allowable safe stress is assumed. [MACHINERY, July, 1908, Maximum Stresses.]

Diameters of Shafts for Combined Torsional and Bending Stresses

On pages 18 and 19 are given tables for the diameters of shafts subjected to combined torsional and bending stresses. The tables are arranged for fiber stresses of 7500, 10,000 and 12,500 pounds per square inch. As an example, find the diameter of a shaft to sustain a bending moment of 80,000 inch-pounds and a torsional moment of 100,000 inch-pounds, if a fiber stress of 10,000 pounds per square inch is allowed. By referring to the table on page 18, and locating the torsional moment as given in thousands of inch-pounds at the top, and the bending moment as given at the left-hand side, in the line and column corresponding to a fiber stress of 10,000 pounds per square inch, and then locating in the body of the table the diameter of the shaft corresponding to these moments, we find that the diameter required is 4¾ inches.

One difficulty with tables is the interpolation for immediate values. A diagram or chart is much better in this respect, and if drawn to a convenient scale is often preferable. On pages 20 to 23, inclusive, are given diagrams for finding the diameter of shaft required for combined torsional and bending stresses. The use of these diagrams is very simple. The bending and torsional moments in thousands, ten-thousands, hundred-thousands, and millions of

(Continued on page 26.)

WEIGHTS AND AREAS OF COLD ROLLED STEEL SHAFTING.

Diameter, Inches.	Area, Square Inches.	Circumference Inches.	Weight per Foot, Pounds.	Diameter, Inches.	Area, Square Inches.	Circumference, Inches.	Weight per Foot, Pounds.
$\frac{3}{16}$	.0276	.5890	.095	$2\frac{3}{16}$	3.7583	6.8722	12.80
$\frac{1}{4}$	.0491	.7854	.167	$2\frac{1}{4}$	3.9761	7.0686	13.52
$\frac{5}{16}$	.0767	.9817	.260	$2\frac{5}{16}$	4.2000	7.2649	14.35
$\frac{3}{8}$	.1104	1.1781	.375	$2\frac{3}{8}$	4.4301	7.4613	15.07
$\frac{7}{16}$	.1503	1.3744	.511	$2\frac{7}{16}$	4.6664	7.6576	15.89
$\frac{1}{2}$	.1963	1.5708	.667	$2\frac{1}{2}$	4.9087	7.8540	16.70
$\frac{9}{16}$	.2485	1.7671	.845	$2\frac{9}{16}$	5.1572	8.0503	17.55
$\frac{5}{8}$	.3068	1.9635	1.05	$2\frac{5}{8}$	5.4119	8.2467	18.41
$\frac{11}{16}$	.3712	2.1598	1.26	$2\frac{11}{16}$	5.6727	8.4430	19.31
$\frac{3}{4}$	.4418	2.3562	1.50	$2\frac{3}{4}$	5.9396	8.6394	20.21
$\frac{13}{16}$	.5185	2.5525	1.77	$2\frac{13}{16}$	6.2126	8.8357	21.15
$\frac{7}{8}$	.6013	2.7489	2.05	$2\frac{7}{8}$	6.4918	9.0321	22.09
$\frac{15}{16}$	.6903	2.9452	2.35	$2\frac{15}{16}$	6.7771	9.2284	23.06
1	.7854	3.1416	2.68	3	7.0686	9.4248	24.05
$1\frac{1}{16}$	.8866	3.3379	3.02	$3\frac{1}{8}$	7.6699	9.8175	26.09
$1\frac{1}{8}$	.9940	3.5343	3.38	$3\frac{3}{16}$	7.9798	10.014	27.16
$1\frac{3}{16}$	1.1075	3.7306	3.77	$3\frac{1}{4}$	8.2958	10.210	28.22
$1\frac{1}{4}$	1.2272	3.9270	4.17	$3\frac{3}{8}$	8.9462	10.603	30.43
$1\frac{5}{16}$	1.3530	4.1233	4.61	$3\frac{7}{16}$	9.2806	10.799	31.58
$1\frac{3}{8}$	1.4849	4.3197	5.05	$3\frac{1}{2}$	9.6211	10.996	32.73
$1\frac{7}{16}$	1.6230	4.5160	5.52	$3\frac{5}{8}$	10.321	11.388	35.20
$1\frac{1}{2}$	1.7671	4.7124	6.01	$3\frac{11}{16}$	10.680	11.585	36.40
$1\frac{9}{16}$	1.9175	4.9087	6.52	$3\frac{3}{4}$	11.045	11.781	37.57
$1\frac{5}{8}$	2.0739	5.1051	7.06	$3\frac{7}{8}$	11.793	12.174	39.40
$1\frac{11}{16}$	2.2365	5.3014	7.61	$3\frac{15}{16}$	12.177	12.370	41.04
$1\frac{3}{4}$	2.4053	5.4978	8.18	4	12.566	12.566	42.75
$1\frac{13}{16}$	2.5802	5.6941	8.78	$4\frac{1}{4}$	14.186	13.352	48.26
$1\frac{7}{8}$	2.7612	5.8905	9.39	$4\frac{7}{16}$	15.466	13.941	52.62
$1\frac{15}{16}$	2.9483	6.0868	10.03	$4\frac{1}{2}$	15.904	14.137	54.11
2	3.1416	6.2832	10.69	$4\frac{3}{4}$	17.728	14.923	60.88
$2\frac{1}{16}$	3.3410	6.4795	11.35	$4\frac{15}{16}$	19.147	15.512	65.50
$2\frac{1}{8}$	3.5466	6.6759	12.07	5	19.635	15.708	67.45

MACHINERY'S Data Sheet No. 29. Explanatory note: Page 26.

inch-pounds, as the case may be, are located at the left-hand side and at the bottom of the diagram respectively; the horizontal line from the bending moment and the vertical line from the torsional moment are followed until they intersect as shown by the dotted lines on page 20. The curve passing exactly or approximately through the point of intersection then indicates the diameter of shaft required. In the example shown on page 20 it will be seen that the lines intersect between the 1 11/16- and 1 3/4-inch curves. It is always better to make the shaft a trifle stronger than necessary; in this case, then, one of 1¾-inch diameter would be used. This shaft would be of the required size to transmit a torsional moment of 4000 inch-pounds, and could in addition sustain a bending moment of 4000 inch-pounds, at a fiber stress of 10,000 pounds per square inch, these being the known requirements from which the dotted lines in the diagram were traced. [MACHINERY, July, 1908, Maximum Stresses.]

Weights and Areas of Cold-rolled Steel Shafting

When calculating the stresses in shafting, the weight of the shafting itself must be considered whenever the distance between the bearings is considerable. The table on page 25 will be found convenient in such instances, as it gives the weight per foot in pounds of cold rolled steel shafting from 3/16 to 5 inches diameter; besides, the area in square inches and the circumference in inches are given. In calculating the stresses caused by the weight of the shaft itself, the total weight between the bearings is, of course, considered as uniformly distributed along the whole shaft, the shaft being assumed to be supported freely at the bearings. The bending moments caused by pulleys, belting, gears, etc., are then determined and these are added to find the total bending moment.

Allowances and Tolerances for Various Kinds of Fits

Running fits, as implied by the name, are characterized by the condition that of two machine members fitted together, one is free to revolve inside or about the other, the fit, however, being otherwise as close as possible. It is evident that the member that fits inside of the other must be a very small amount less in diameter than the hole into which it fits.

The term "forcing fit" is used when a pin, axle, or other part, which is somewhat larger than the hole into which it is inserted, is pressed into place by a hydraulic press or by other means. The crank-pins and axles for locomotive driving wheels are usually inserted in this way.

The term "shrinking fit" is applied when a part which is to be held in position by being tightly fitted into a hole is first turned a few thousandths of an inch larger than the hole, and then the diameter of the hole increased by heating it, after which the pin is inserted in the heated part. When this part cools down, the consequent contraction of the metal causes it to grip the pin with tremendous pressure. Locomotive tires, for example, are attached to their wheel centers by means of a shrinking fit.

Allowances and tolerances for running fits recommended by the Engineering Standards Committee of Great Britain are given on page 27. The note at the bottom of the page should be carefully read before using the table, in order to avoid misunderstandings.

On page 28 a diagram is given of allowances for forcing, driving and running fits as adopted by the Builders' Iron Foundry, Providence, R. I. In the diagram two heavy lines are drawn for each kind of fit, the upper line indicating the maximum and the lower line the minimum allowance for the respective diameter. For example, assume

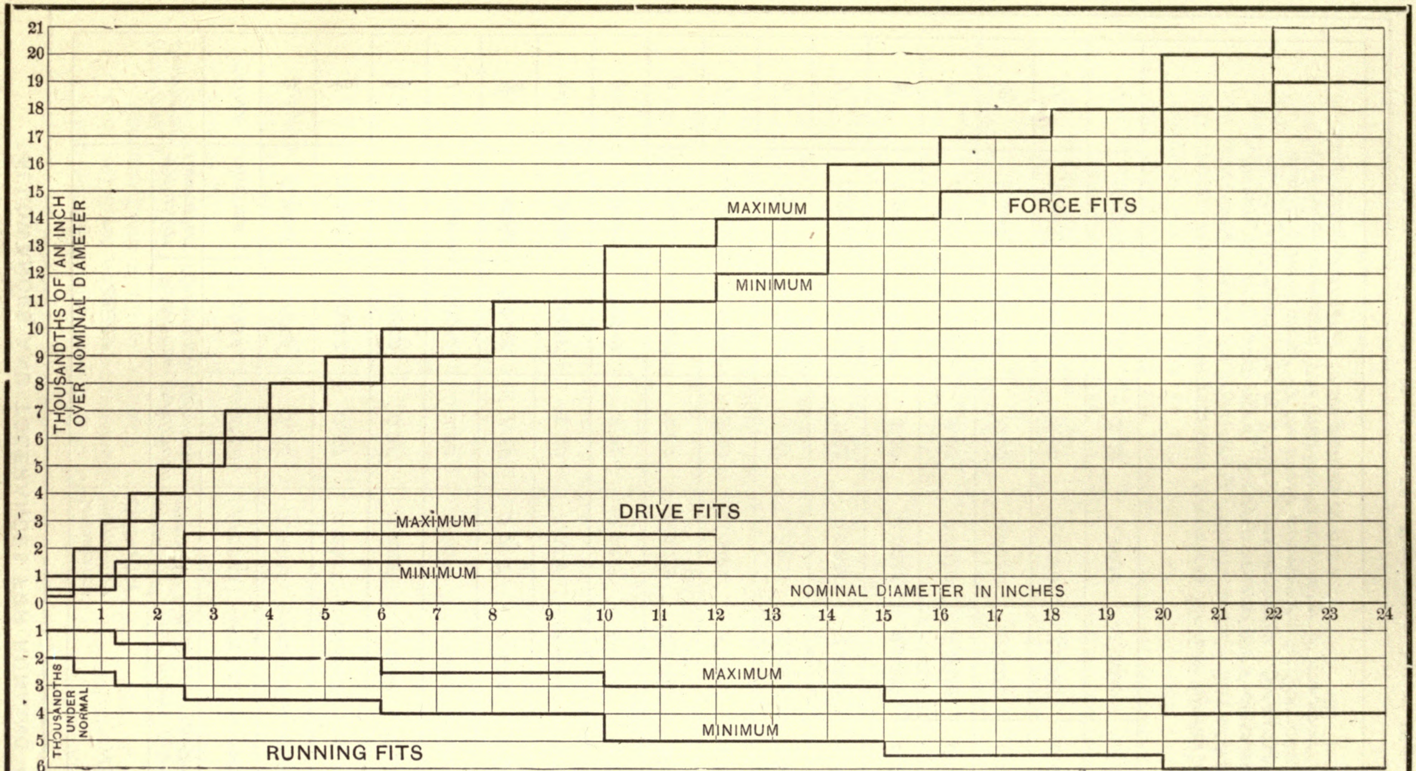
(Continued on page 34.)

ALLOWANCES AND TOLERANCES FOR RUNNING FITS

Nominal Diameter	Shaft			Allowance (Minimum Difference between Shaft and Hole)	Hole		
	Minimum Diameter	Tolerance (Difference)	Maximum Diameter		Minimum Diameter	Tolerance (Difference)	Maximum Diameter
Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
1/4	0.2495	0.0005	0.25	0.0005	0.2505	0.0003	0.2508
1/2	0.4993	0.0007	0.50	0.0007	0.5007	0.0007	0.5014
3/4	0.7491	0.0009	0.75	0.0008	0.7508	0.0009	0.7517
1	0.9990	0.0010	1.00	0.0010	1.0010	0.0010	1.0020
1 1/2	1.4988	0.0012	1.50	0.0012	1.5012	0.0013	1.5025
2	1.9985	0.0015	2.00	0.0015	2.0015	0.0015	2.0030
3	2.9982	0.0018	3.00	0.0018	3.0018	0.0017	3.0035
4	3.9980	0.0020	4.00	0.0020	4.0020	0.0020	4.0040
5	4.9980	0.0020	5.00	0.0020	5.0020	0.0020	5.0040
6	5.9975	0.0025	6.00	0.0025	6.0025	0.0025	6.0050
7	6.9975	0.0025	7.00	0.0025	7.0025	0.0025	7.0050
8	7.9975	0.0025	8.00	0.0025	8.0025	0.0025	8.0050
9	8.9970	0.0030	9.00	0.0030	9.0030	0.0030	9.0060
10	9.9970	0.0030	10.00	0.0030	10.0030	0.0030	10.0060
11	10.9970	0.0030	11.00	0.0030	11.0030	0.0030	11.0060
12	11.9970	0.0030	12.00	0.0030	12.0030	0.0030	12.0060

Note:- The above allowances and tolerances for running fits are recommended by the Engineering Standards Committee of Great Britain, for first-class work. For second- and third-class work, multiply the tolerances by 2 and 3, respectively. For extra fine quality of work, about 2/5 the above allowances for first-class work are recommended. The maximum diameter of the shaft is the nominal diameter in all grades of work.

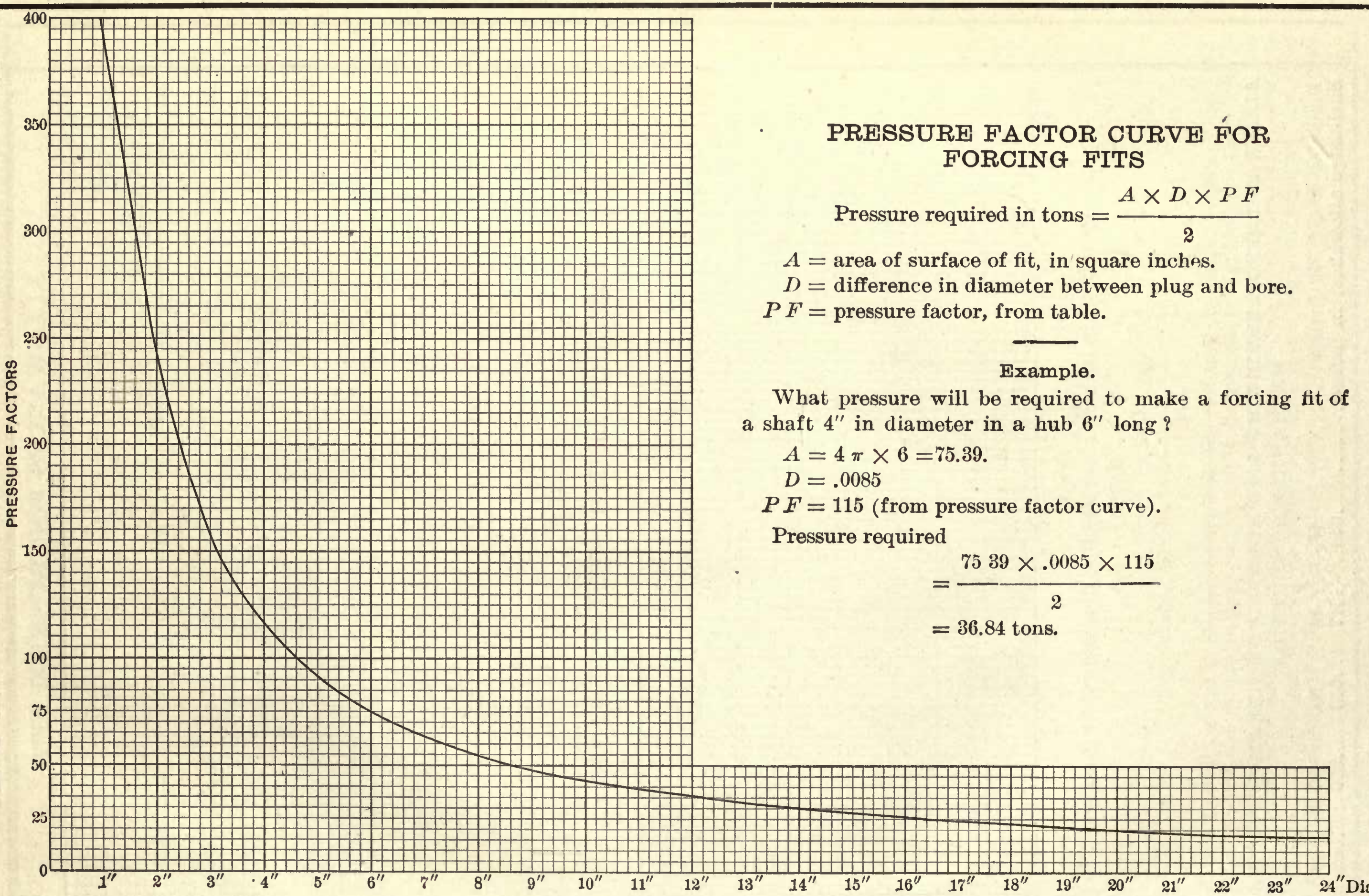
ALLOWANCES FOR FORCE, DRIVE AND RUNNING FITS (Builders Iron Foundry)



LIMITS FOR STANDARD HOLES.

Diameters.	Minimum.	Maximum.
.00 to 1.24	Diameter − .00025	Diameter + .00025
1.25 to 2.49	Diameter − .00025	Diameter + .00075
2.50 to 5.99	Diameter − .0005	Diameter + .001
6.00 to 11.99	Diameter − .001	Diameter + .001

PRESSURE FACTORS FOR FORCING FITS



PRESSURE FACTOR CURVE FOR FORCING FITS

$$\text{Pressure required in tons} = \frac{A \times D \times PF}{2}$$

A = area of surface of fit, in square inches.
D = difference in diameter between plug and bore.
PF = pressure factor, from table.

Example.

What pressure will be required to make a forcing fit of a shaft 4″ in diameter in a hub 6″ long?

$A = 4\,\pi \times 6 = 75.39.$
$D = .0085$
$PF = 115$ (from pressure factor curve).

Pressure required

$$= \frac{75.39 \times .0085 \times 115}{2}$$

$$= 36.84 \text{ tons.}$$

This curve is an hyperbola whose equation is $PF = \dfrac{500}{D^{1.06}}$

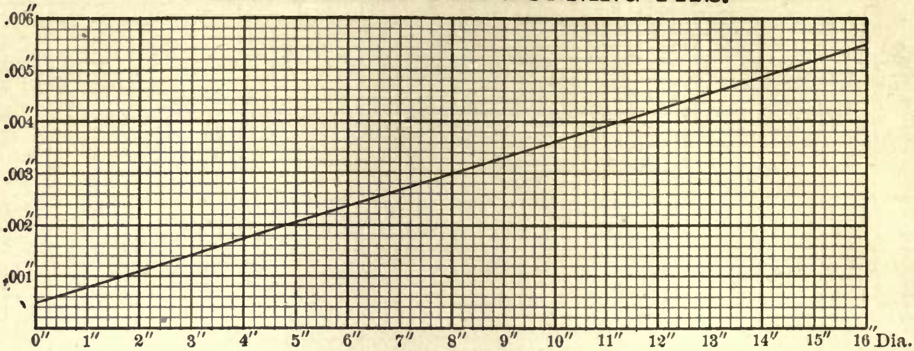
It assumes the hub to be twice the diameter of the plug, the shafts of machinery steel and the hubs of cast iron.

From a paper by Stanley H. Moore read before the American Society of Mechanical Engineers.

MACHINERY's Data Sheet No. 23. Explanatory note: Page 26.

DIAGRAM FOR RUNNING FITS AND LIMIT GAGES

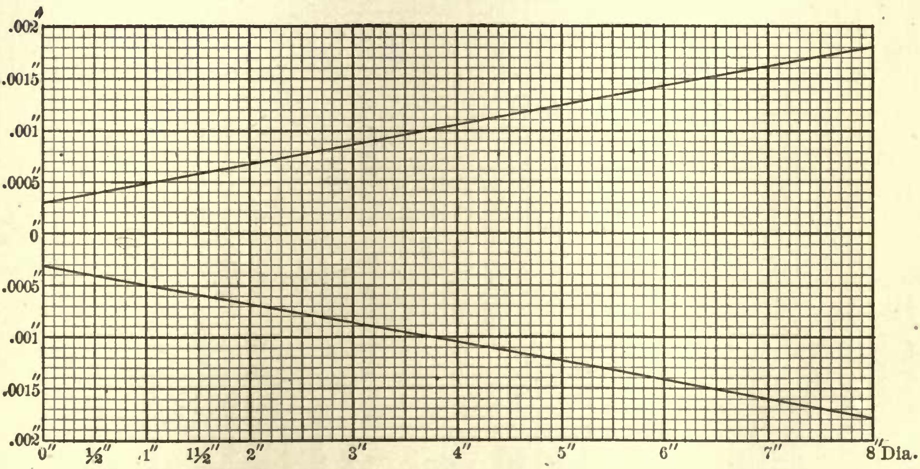
ALLOWANCES FOR RUNNING FITS.



A = allowance in inches. D = nominal diameter of fit in inches.

$$\text{For running fits, } A = \frac{0.31\,D + 0.5}{1000}$$

TABLE OF LIMITS FOR LIMIT GAGES.



L = total limit in inches. D = nominal diameter of fit in inches.

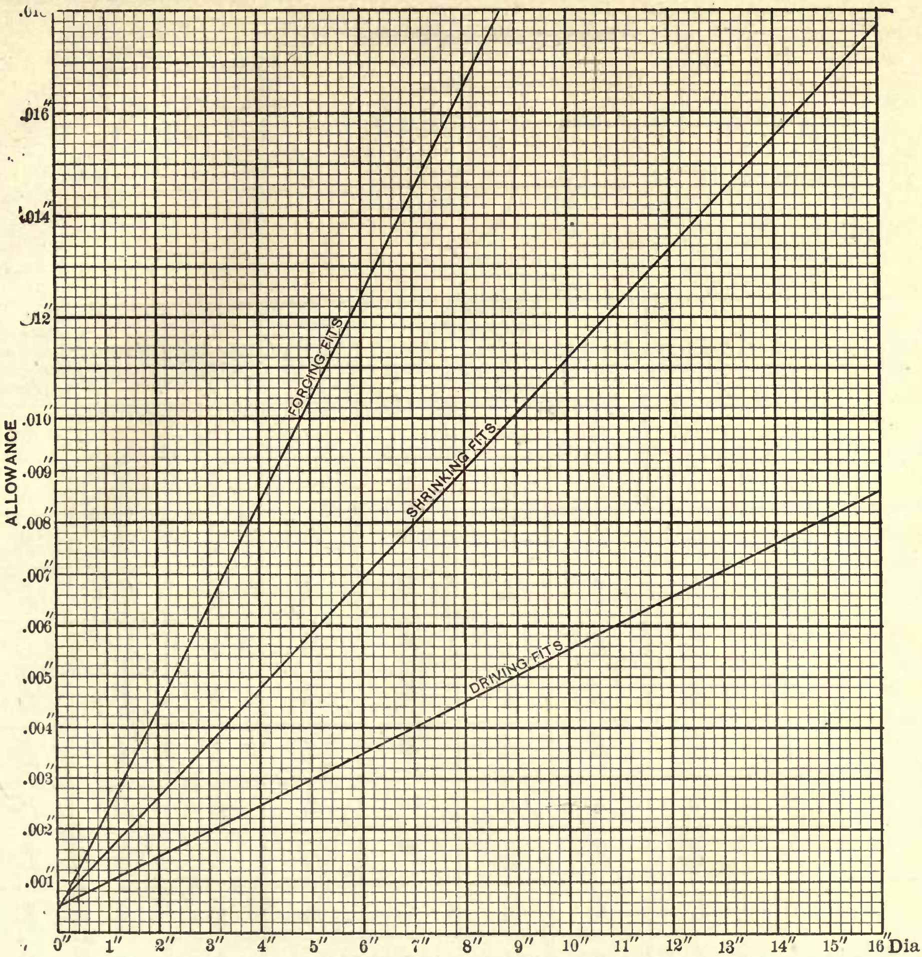
$$L = \frac{0.375\,D + 0.6}{1000}$$

ALLOWANCES FOR SHRINKING FITS ADOPTED BY THE AMERICAN MASTER MECHANICS' ASSOCIATION.

Diameter of tire, in inches	38	44	50	56	62	66
Shrinkage allowance, in inches	.040	.047	.053	.060	.066	.070

From a paper by Stanley H. Moore read before the American Society of Mechanical Engineers.
MACHINERY'S Data Sheet No. 23. Explanatory notes: Pages 26 and 39.

FORCING, SHRINKING AND DRIVING FITS



A = allowance in inches. D = nominal diameter of fit in inches.

$$\text{For forcing fits, } A = \frac{2\,D + 0.5}{1000}$$

$$\text{For shrinking fits, } A = \frac{1.06\,D + 0.5}{1000}$$

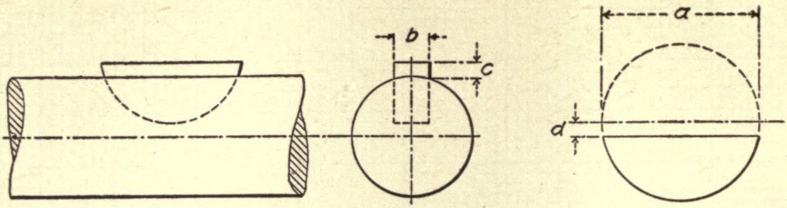
$$\text{For driving fits, } A = \frac{0.5\,D + 0.5}{1000}$$

NOTE.—While the data given in the above table is the result of an investigation of the practice of a large number of shops, the allowances for the large diameters is considered excessive, as they give results which require presses of more than ordinary power to make the fits. It is the practice in a large number of shops to decrease the allowance per inch as the diameter increases. The general rule of .001 inch per inch of diameter has been found very satisfactory for sizes above 6 inches, while the allowances for the smaller sizes correspond more nearly to those given above.

From a paper by Stanley H. Moore read before the American Society of Mechanical Engineers.
MACHINERY'S Data Sheet No. 23. Explanatory note: Page 26.

WOODRUFF KEYS—I

Woodruff Standard Keys

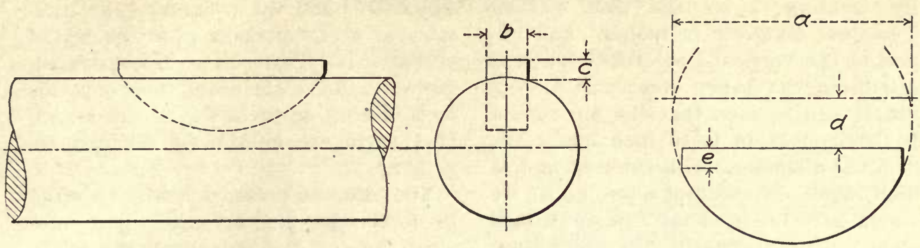


No. of Key	Diam. of Key	Thickness of Key	Depth of Keyway	Center of stock, from which key is made, to top of key	No of Key	Diam. of Key	Thickness of Key	Depth of Keyway	Center of stock, from which key is made, to top of key
	a	b	c	d		a	b	c	d
1	1/2	1/16	1/32	3/64	B	1	5/16	5/32	1/16
2	1/2	3/32	3/64	3/64	16	1 1/8	3/16	3/32	5/64
3	1/2	1/8	1/16	3/64	17	1 1/8	7/32	7/64	5/64
4	5/8	3/32	3/64	1/16	18	1 1/8	1/4	1/8	5/64
5	5/8	1/8	1/16	1/16	C	1 1/8	5/16	5/32	5/64
6	5/8	5/32	5/64	1/16	19	1 1/4	3/16	3/32	5/64
7	3/4	1/8	1/16	1/16	20	1 1/4	7/32	7/64	5/64
8	3/4	5/32	5/64	1/16	21	1 1/4	1/4	1/8	5/64
9	3/4	3/16	3/32	1/16	D	1 1/4	5/16	5/32	5/64
10	7/8	5/32	5/64	1/16	E	1 1/4	3/8	3/16	5/64
11	7/8	3/16	3/32	1/16	22	1 3/8	1/4	1/8	3/32
12	7/8	7/32	7/64	1/16	23	1 3/8	5/16	5/32	3/32
A	7/8	1/4	1/8	1/16	F	1 3/8	3/8	3/16	3/32
13	1	3/16	3/32	1/16	24	1 1/2	1/4	1/8	7/64
14	1	7/32	7/64	1/16	25	1 1/2	5/16	5/32	7/64
15	1	1/4	1/8	1/16	G	1 1/2	3/8	3/16	7/64

MACHINERY'S Data Sheet No. 81. Explanatory note: Page 39.

WOODRUFF KEYS—II

Woodruff Special Keys



No. of Key	Diam. of Key	Thickness of Key	Depth of Keyway	Center of stock, from which key is made, to top of key	Width of Flat	No. of Key	Diam. of Key	Thickness of Key	Depth of Keyway	Center of stock, from which key is made, to top of key	Width of Flat
	a	b	c	d	e		a	b	c	d	e
26	$2\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{17}{32}$	$\frac{3}{32}$	31	$3\frac{1}{2}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{13}{16}$	$\frac{3}{16}$
27	$2\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{17}{32}$	$\frac{3}{32}$	32	$3\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{13}{16}$	$\frac{3}{16}$
28	$2\frac{1}{8}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{17}{32}$	$\frac{3}{32}$	33	$3\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{32}$	$\frac{13}{16}$	$\frac{3}{16}$
29	$2\frac{1}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{17}{32}$	$\frac{3}{32}$	34	$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{13}{16}$	$\frac{3}{16}$
30	$3\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{13}{16}$	$\frac{3}{16}$						

Standard Keys to use with various diameter shafts

Diameter of Shaft	Number of Keys	Diameter of Shaft	Number of Keys	Diameter of Shaft	Number of Keys
$\frac{5}{16}-\frac{3}{8}$	1	$\frac{7}{8}-\frac{15}{16}$	6, 8, 10	$1\frac{3}{8}-1\frac{7}{16}$	14, 17, 20
$\frac{7}{16}-\frac{1}{2}$	2, 4	1	9, 11, 13	$1\frac{1}{2}-1\frac{5}{8}$	15, 18, 21, 24
$\frac{9}{16}-\frac{5}{8}$	3, 5	$1\frac{1}{16}-1\frac{1}{8}$	9, 11, 13, 16	$1\frac{11}{16}-1\frac{3}{4}$	18, 21, 24
$\frac{11}{16}-\frac{3}{4}$	3, 5, 7	$1\frac{3}{16}$	11, 13, 16	$1\frac{13}{16}-2$	23, 25
$\frac{13}{16}$	6, 8	$1\frac{1}{4}-1\frac{5}{16}$	12, 14, 17, 20	$2\frac{1}{16}-2\frac{1}{2}$	25

that it is required to find the diameter to which to turn a pin to be fitted by a forcing fit into a standard 7-inch hole. By locating 7 on the line marked "Nominal diameter in inches," and following the vertical line from 7 until it intersects the heavy lines for forcing fits, it will be seen that the pin should be from 0.009 to 0.010 inch above the nominal diameter. If a running fit had been required instead of a forcing fit, we would have followed the line downward from 7 until intersecting the heavy lines representing the limits for running fits. Assuming the hole to be standard size as mentioned, the pin should thus have been turned from 0.0025 to 0.004 inch below the size of the hole. In the case of running fits, however, it is almost always the practice to make the diameter of the shaft the standard or nominal size, and to provide for the allowance in the hole. In such a case the shaft would have been made 7 inches in diameter while the hole would have been made from 7.0025 to 7.004 inches in diameter.

Whether parts should be assembled by pressing them into place or by the shrinking method depends somewhat upon circumstances. To press a tire, for example, over a wheel center, would be a rather difficult job, owing to the size and shape of the work. On the other hand, a pin is easily forced into place with a hydraulic press if such a tool is available; otherwise the hole can be heated and expanded sufficiently to permit the insertion of the pin by sledging or even by hand. The hydraulic press is more economical for most work, and in addition there is an advantage in its use in that the exact pressure or tonnage required to force the part into place is indicated by a gage, while there is more or less uncertainty connected with a shrinking fit. If the allowance when turning a pin for a shrinking fit were too great, the part into which the pin is fitted might be broken when cooled down, owing to the excessive stresses produced. When using a press this danger is largely eliminated, as the approximate pressure required can be calculated, and the pressure gage indicates at every moment what the actual pressure is. Tests have demonstrated, however, that a shrinking fit is superior to a forcing or press fit, as the assembled parts are held more securely together.

The ultimate pressure finally required to force the pin or other part into place depends not only upon the allowance for the fit, but also upon the length of the bore or the area of the surface of the fit. The pressure required for forcing a pin with a given allowance into a hole may be determined by the formula given with the diagram on page 29, where the pressure factor *PF* is determined from the diagram. This pressure factor varies with the diameter of the pin. For example, if the pin is 6 inches in diameter, then we find from the diagram that the pressure factor is 75. To find this, we locate 6 on the scale at the bottom of the diagram, and follow the vertical line from the point so located until it intersects the curve drawn on the diagram; from the point of intersection, we follow the horizontal line to the scale at the left where the pressure factor 75 is read off. The example given in connection with the diagram and formula indicates clearly their use for practical calculations.

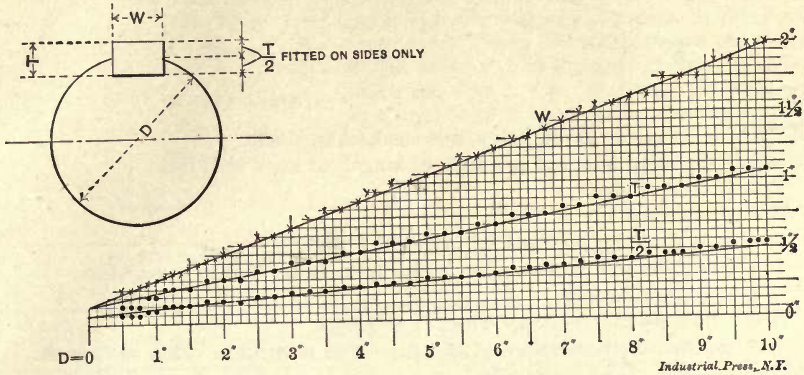
The diagrams for running, forcing, shrinking and driving fits given on pages 30 and 31 are compiled from a paper read by Stanley H. Moore before the American Society of Mechanical Engineers, and are the results of an investigation of the practice in a large number of shops. Before using these diagrams, however, the note at the bottom of page 31 should be read and comparison made with the diagram on page 28. The use of the diagrams on pages 30 and 31 is very simple. On the scale at the bottom of the diagrams are given

(Continued on page 39.)

DIMENSIONS OF KEYS—I

PROPORTIONS OF KEYS.

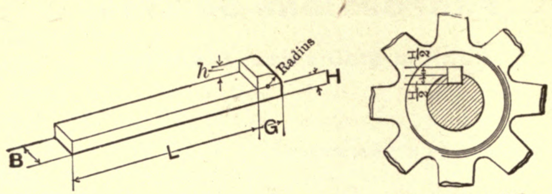
(United States Navy Standard.)



D	W	T	D	W	T	D	W	T
1/2"	7/32"	3/16"	4"	7/8"	1/2"	8"	1 5/8"	7/8"
5/8"	1/4"	3/16"	4 1/4"	15/16"	9/16"	8 1/4"	1 5/8"	15/16"
3/4"	9/32"	3/16"	4 1/2"	1"	9/16"	8 1/2"	1 3/4"	15/16"
7/8"	9/32"	7/32"	4 3/4"	1"	9/16"	8 3/4"	1 3/4"	15/16"
1"	5/16"	7/32"	5"	1 1/16"	5/8"	9"	1 3/4"	1"
1 1/8"	11/32"	1/4"	5 1/4"	1 1/8"	5/8"	9 1/4"	1 7/8"	1"
1 1/4"	3/8"	1/4"	5 1/2"	1 3/16"	5/8"	9 1/2"	1 7/8"	1 1/16"
1 3/8"	3/8"	1/4"	5 3/4"	1 3/16"	11/16"	9 3/4"	2"	1 1/16"
1 1/2"	13/32"	1/4"	6"	1 1/4"	11/16"	10"	2"	1 1/16"
1 3/4"	7/16"	5/16"	6 1/4"	1 5/16"	3/4"	....	....	....
2"	1/2"	5/16"	6 1/2"	1 3/8"	3/4"	....	....	....
2 1/4"	9/16"	5/16"	6 3/4"	1 3/8"	3/4"	....	....	....
2 1/2"	5/8"	3/8"	7"	1 7/16"	13/16"	....	....	....
2 3/4"	5/8"	3/8"	7 1/4"	1 1/2"	13/16"	....	....	....
3"	11/16"	7/16"	7 1/2"	1 9/16"	7/8"	....	....	....
3 1/4"	3/4"	7/16"	7 3/4"	1 9/16"	7/8"	....	....	....
3 1/2'	13/16"	7/16"	....	....	..	....	....	....
3 3/4"	13/16"	1/2"	....	....	..	....	....	....

DIMENSIONS OF KEYS—II

TABLE OF GIB KEYS. Computed by F. D. Buffum, Akron, O



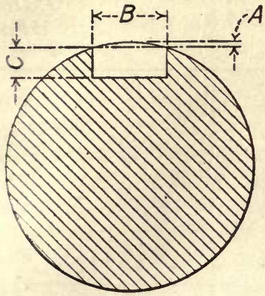
Industrial Press, N.Y.

Keys of proportions given below are weakest in shear.
The safe twisting moment per inch of length of keys = R B S
- R = Radius of shaft.
- B = Breadth of key.
- S = Safe shearing strength of material in key.
- B = $\frac{1}{4}$ bore up to 6 inches. Then B = .211 bore. Taken to eighths.
- G = B approximately.
- H = $\frac{1}{6}$ bore up to 6 inches. Then H = $\frac{1}{8}$ bore.
- h = Radius = $\frac{1}{8}$ bore taken to eighths. But minimum value = $\frac{3}{16}$ inch.
- L = Length of hub + $\frac{1}{2}$ inch.
  Taper $\frac{1}{8}$ inch per foot.

Bore and Shaft Diameter.	Width of Key B.	Height of Key H.	Depth $\frac{H}{2}$	h and Rad.	G	Safe Twisting Moment on Key per inch of Length for S =		
						5000	7500	10000
$\frac{15}{16}$ to $1\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{3}{16}$	$\frac{1}{4}$	630	940	1250
$1\frac{3}{16}$ to $1\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{8}$	1170	1760	2340
$1\frac{7}{16}$ to $1\frac{5}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{8}$	1410	2110	2810
$1\frac{11}{16}$ to $1\frac{7}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{1}{4}$	$\frac{1}{2}$	2190	3280	4380
$1\frac{15}{16}$ to $2\frac{1}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{2}$	2500	3750	5000
$2\frac{3}{16}$ to $2\frac{3}{8}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{8}$	$\frac{5}{8}$	3520	5270	7030
$2\frac{7}{16}$ to $2\frac{5}{8}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{3}{8}$	$\frac{5}{8}$	3910	5860	7810
$2\frac{11}{16}$ to $2\frac{7}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	5160	7730	10313
$2\frac{15}{16}$ to $3\frac{1}{8}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	5620	8420	11250
$3\frac{3}{16}$ to $3\frac{3}{8}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{9}{32}$	$\frac{1}{2}$	$\frac{7}{8}$	7110	10660	14220
$3\frac{7}{16}$ to $3\frac{5}{8}$	$\frac{7}{8}$	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{7}{8}$	7660	11480	15310
$3\frac{11}{16}$ to $3\frac{7}{8}$	1	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{1}{2}$	1	9380	14060	18750
$3\frac{15}{16}$ to $4\frac{1}{8}$	1	$\frac{11}{16}$	$\frac{11}{32}$	$\frac{1}{2}$	1	10000	15000	20000
$4\frac{3}{16}$ to $4\frac{3}{8}$	$1\frac{1}{8}$	$\frac{11}{16}$	$\frac{11}{32}$	$\frac{5}{8}$	1	11950	17930	23910
$4\frac{7}{16}$ to $4\frac{3}{4}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{5}{8}$	1	12660	18980	25310
$4\frac{13}{16}$ to $5\frac{1}{4}$	$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{5}{8}$	$1\frac{1}{4}$	15620	23440	31250
$5\frac{5}{16}$ to $5\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{1}{4}$	18910	28360	37810
$5\frac{13}{16}$ to $6\frac{1}{4}$	$1\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{2}$	22500	33750	45000
$6\frac{5}{16}$ to $6\frac{3}{4}$	$1\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{2}$	24380	36560	48750
$6\frac{13}{16}$ to $7\frac{1}{4}$	$1\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{4}$	26250	39380	52500
$7\frac{5}{16}$ to $7\frac{3}{4}$	$1\frac{5}{8}$	1	$\frac{1}{2}$	1	$1\frac{3}{4}$	30470	45700	60940
$7\frac{13}{16}$ to $8\frac{3}{4}$	$1\frac{3}{4}$	$1\frac{1}{8}$	$\frac{9}{16}$	1	2	36090	54140	72190
$8\frac{13}{16}$ to $9\frac{3}{4}$	2	$1\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{8}$	2	46250	69380	92500
$9\frac{13}{16}$ to $10\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{4}$	2	57660	86480	115320
$10\frac{13}{16}$ to $11\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{4}$	2	70310	105470	140630
$11\frac{13}{16}$ to $12\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{4}$	2	76560	114840	153130

MACHINERY'S Data Sheet No. 14. Explanatory note: Page 39.

TABLE FOR USE WHEN MILLING KEYWAYS

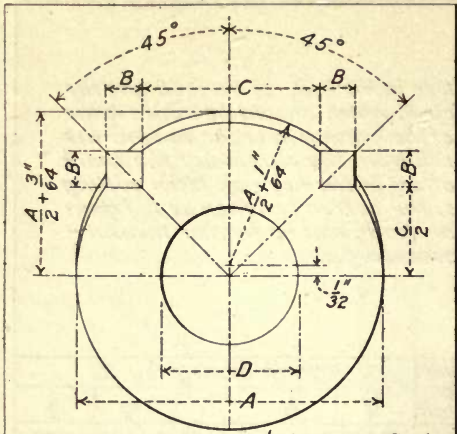


The values in the body of the table give the dimension A, which should be added to the depth C of the keyway in order to find the total depth from the outside of the shaft to the bottom of the keyway. When milling keyways, the cutter can then be fed down this total depth, and no further measuring is necessary.

Size of Shaft	Width of Keyway B					Size of Shaft	Width of Keyway B				
	1/4	5/16	3/8	7/16	1/2		1/4	5/16	3/8	7/16	1/2
1/2	0.0325	—	—	—	—	2 5/16	0.0068	0.0104	0.0155	0.0209	0.0274
9/16	0.0289	—	—	—	—	2 3/8	0.0066	0.0102	0.0152	0.0202	0.0267
5/8	0.0254	0.0413	—	—	—	2 7/16	0.0064	0.0100	0.0149	0.0198	0.0260
11/16	0.0236	0.0379	—	—	—	2 1/2	0.0063	0.0098	0.0146	0.0194	0.0253
3/4	0.0220	0.0346	0.0511	—	—	2 9/16	0.0061	0.0094	0.0142	0.0189	0.0247
13/16	0.0198	0.0314	0.0465	—	—	2 5/8	0.0060	0.0090	0.0139	0.0185	0.0242
7/8	0.0177	0.0283	0.0420	0.0583	—	2 11/16	0.0059	0.0089	0.0136	0.0180	0.0236
15/16	0.0164	0.0264	0.0392	0.0544	—	2 3/4	0.0058	0.0088	0.0133	0.0176	0.0230
1	0.0152	0.0246	0.0365	0.0506	0.0670	2 13/16	0.0057	0.0086	0.0129	0.0172	0.0226
1 1/16	0.0143	0.0228	0.0342	0.0476	0.0625	2 7/8	0.0056	0.0084	0.0126	0.0168	0.0220
1 1/8	0.0136	0.0210	0.0319	0.0446	0.0581	2 15/16	0.0054	0.0083	0.0122	0.0164	0.0216
1 3/16	0.0131	0.0204	0.0304	0.0421	0.0551	3	0.0053	0.0081	0.0119	0.0161	0.0211
1 1/4	0.0127	0.0198	0.0290	0.0397	0.0522	3 1/16	0.0052	0.0080	0.0116	0.0158	0.0207
1 5/16	0.0123	0.0191	0.0279	0.0380	0.0499	3 1/8	0.0051	0.0078	0.0114	0.0155	0.0202
1 3/8	0.0120	0.0185	0.0268	0.0364	0.0477	3 3/16	0.0050	0.0076	0.0112	0.0157	0.0198
1 7/16	0.0114	0.0174	0.0254	0.0346	0.0453	3 1/4	0.0049	0.0075	0.0110	0.0149	0.0194
1 1/2	0.0110	0.0164	0.0240	0.0328	0.0429	3 5/16	0.0048	0.0074	0.0108	0.0146	0.0191
1 9/16	0.0107	0.0158	0.0231	0.0309	0.0412	3 3/8	0.0047	0.0072	0.0106	0.0143	0.0187
1 5/8	0.0105	0.0153	0.0221	0.0291	0.0395	3 7/16	0.0046	0.0071	0.0104	0.0140	0.0184
1 11/16	0.0102	0.0147	0.0214	0.0282	0.0383	3 1/2	0.0045	0.0070	0.0102	0.0138	0.0180
1 3/4	0.0099	0.0142	0.0207	0.0274	0.0371	3 9/16	0.0044	0.0069	0.0101	0.0135	0.0177
1 13/16	0.0095	0.0136	0.0198	0.0265	0.0355	3 5/8	0.0043	0.0067	0.0100	0.0133	0.0174
1 7/8	0.0093	0.0130	0.0190	0.0257	0.0339	3 11/16	0.0042	0.0066	0.0099	0.0131	0.0171
1 15/16	0.0090	0.0127	0.0184	0.0250	0.0328	3 3/4	0.0042	0.0065	0.0098	0.0128	0.0168
2	0.0088	0.0124	0.0179	0.0243	0.0317	3 13/16	0.0041	0.0064	0.0097	0.0126	0.0166
2 1/16	0.0083	0.0117	0.0173	0.0236	0.0308	3 7/8	0.0041	0.0063	0.0096	0.0124	0.0163
2 1/8	0.0078	0.0111	0.0168	0.0229	0.0299	3 15/16	0.0041	0.0062	0.0095	0.0123	0.0161
2 3/16	0.0073	0.0109	0.0163	0.0222	0.0291	4	0.0040	0.0061	0.0094	0.0121	0.0160
2 1/4	0.0070	0.0107	0.0159	0.0216	0.0282						

Contributed by James J. Loftus. Explanatory note: Page 39.

DUPLEX KEYS



Taper, one key only, 1/8 inch per foot.

A	B	C	Bore of Hollow Shaft D
1	3/16	1/2	——
2	3/8	1	——
3	9/16	1 15/16	——
4	3/4	1 31/32	——
5	15/16	2 15/32	——
6	1 1/8	2 31/32	2
7	1 5/16	3 15/32	2 3/8
8	1 1/2	3 31/32	2 3/4
9	1 11/16	4 7/16	3
10	1 7/8	4 15/16	3 3/8
11	2 1/16	5 7/16	3 3/4
12	2 1/4	5 15/16	4
13	2 7/16	6 7/16	4 3/8
14	2 5/8	6 29/32	4 3/4
15	2 13/16	7 13/32	5
16	3	7 29/32	5 3/8
17	3 3/16	8 13/32	5 3/4
18	3 3/8	8 29/32	6
19	3 9/16	9 13/32	6 3/8
20	3 3/4	9 7/8	6 3/4
21	3 15/16	10 3/8	7
22	4 1/8	10 7/8	7 3/8
23	4 5/16	11 3/8	7 3/4
24	4 1/2	11 7/8	8
25	4 11/16	12 11/32	8 3/8
26	4 7/8	12 27/32	8 3/4
27	5 1/16	13 11/32	9
28	5 1/4	13 27/32	9 3/8
29	5 7/16	14 11/32	9 3/4
30	5 5/8	14 27/32	10
31	5 13/16	15 5/16	10 3/8
32	6	15 13/16	10 3/4
33	6 3/16	16 5/16	11
34	6 3/8	16 13/16	11 3/8
35	6 9/16	17 5/16	11 3/4
36	6 3/4	17 25/32	12

the diameters in inches, and on the scale at the left-hand side the allowances. Assume, for example, that we want to find the allowance for a shrinking fit for a 4-inch diameter pin. Referring to page 31, we find by following the vertical line from 4 inches until it intersects the diagonal line for shrinking fits, and from the point of intersection following the horizontal line to the left-hand scale, that an allowance of nearly 0.005 inch is required.

Allowances for shrinking fits adopted by the American Master Mechanics Association are given at the bottom of page 30. These allowances refer directly to tires to be shrunk onto their wheel centers. [MACHINERY, July, 1909, Machine Shop Practice—Shrinking and Forcing Fits.]

Diagram of Limits for Limit Gages

On page 30 a diagram is given showing suitable maximum and minimum limits for limit gages for ordinary work. It will be understood that the upper and lower diagonal lines in this diagram indicate the maximum and minimum limits corresponding to various diameters. To find the limits for any given diameter, say 6 inches, this dimension is first located on the bottom scale, and the vertical line from 6 inches is followed until it intersects the lower diagonal line. From the point of intersection the horizontal line is followed to the left, and the minimum limit read off. This diagram is made up on the principle that the maximum limit is as much above standard size as the minimum limit is below standard size, so that when the minimum limit has been found there is no need of locating the maximum limit. For a 4-inch diameter shaft, for example, the allowable limits would be very slightly more than 0.001 inch above or below the standard size.

Keys

On pages 32 and 33 are given tables of Woodruff standard and special keys. In the lower part of page 33 a table is also given of Woodruff standard keys to be used with various shaft diameters. It will be seen that the designer's judgment must be relied upon to a certain extent, as a number of different sizes of keys may be used for the same diameters. For ordinary practice, when no special considerations have to be taken into account and where more than two keys are given for the same diameters, the medium size key is the most suitable.

On page 35 is given a diagram and table of the United States Navy standard proportions of keys. The diagram is shown only to indicate how the sizes were determined by plotting the dimensions from a curve supposed to give the best theoretical dimensions. When using the information given on page 35, no attention need be paid to the diagram, but the table should be used directly, as all the required information is contained therein.

On page 36 is given a table of gib keys. In addition to the dimensions, it will be seen that the safe twisting moments which the key will sustain for each inch of length, at different shearing stresses, are given. This information will be of considerable value in quickly calculating the strength of keys when the twisting moment is known. [MACHINERY, September, 1901, Notes on Keys and Keyways; March, 1907, Keys and Keyways; MACHINERY'S Reference Series No. 22, Calculation of Elements of Machine Design, Chapter VI, Keys and Keyways.]

Table for Use when Milling Keyways

The table given on page 37 will be found very useful when milling keyways to a given depth. The usual way of measuring the depth of a keyway is to mill off the top of the shaft until the flat on the top is of the same width as the cutter. Then the index is set to zero and the cutter is fed down the required depth C (see page 37). When milling off the top of the shaft, difficulty is experienced in measuring properly the width of the flat. By means of the table a more accurate measurement can quickly be obtained. Bring the cutter down so that it will just touch the work on the top, and set the index to zero. Then add the figures given in the body of the table for the given size of shaft and width of keyway, to the depth C of the keyway. This gives the total depth from the outside of the shaft to the bottom of the keyway. For example: If the size of the shaft is 3 inches and the width of the keyway one-half inch, then, from the table, we find that 0.0211 inch should be added to the given depth C of the keyway—usually made half the width B—in order to find the total depth from the top of the shaft to the bottom of the keyway. In this case, then, this dimension would be $0.250 + 0.0211 = 0.2711$ inch. [MACHINERY, December, 1908, Keyway Gaging in Shafts and Hubs.]