DESIGN OF FLEXIBLE TEST ENGINE

THESIS FOR THE DEGREE OF M. E.

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DESIGN OF FLEXIBLE TEST ENGINE

By

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INTRODUCTION

Instruction in the internal combustion engine field is complicated by the number and idiosyncrosies of the numerous component parts. In contrast to electrical engineering where the units follow a given pattern accurately and can therefore be predicted by mathematical expressions including the calculus, the effect of changing a single adjustment of an engine is difficult enough to forecase accurately and may be entirely different when made in conjunction with other improvements in design. The one real way to find out whether a unit will perform as desired or not is to try it. As a matter of fact, the early "automotive engineers" were simply individuals who took the time and effort to build a number of engines and automobiles. After several failures, it was quite evident that certain features were good and others bad. Mr. Kettering's advice often is to ask the question "What does the engine think about it?"

At times, accidents resulting in the destruction of valuable experimental equipment serve a useful purpose in subjecting our formulas to the ultimate test. As an example of this, water got into the combustion space of an experimental engine without the operator knowing anything about it. This particular engine had individual cylinder heads which usually are torn loose from the cylinder block when an incompressible fluid such as oil or water fills up the combustion space. In this case, however, the crankshaft was broken into many pieces when the engine was turned over to start in the usual manner, indicating to the engineer in charge that his analysis of the crankshaft did not agree with complete factual data.

Granting that the experience gained by building, testing and discarding engines has constituted the training of many engineers in positions of importance in the industry, limitations of time and expense rule out this elaborate method for the large numbers of beginning engineers needed to keep increasing the comfort, economy and safety of automobile transportation. We need a faster and cheaper yet highly efficient system suitable for "mass production" at least in comparison with the method outlined above.

The curriculum for such a program should include technical engineering subjects to create a background of knowledge by which the size and arrangement of various units may be determined; but even more important, practical problems dealing with the actual operation of engines and automobiles. Technical data on a printed page is useful only to the extent that it is applied in such a way as to become an integral part of the reasoning process.

Two types of instruction are very important in assisting the student to create vivid sensory impressions associated with textbook information:

1. Establishing engine performance under stated conditions on the dynamometer.
2. Studying the effect of operating variables on engine performance by using special test engine equipment, and an electronic indicator.

The latter method has not been utilized sufficiently in many programs and deserves serious consideration in course planning.

This enables the student to set up a development program for changing the performance of the engine, estimate the result on the basis of technical information available, and then find out for himself by personal observation just what actually does happen.

Such instruction has been a part of the curriculum in the design of engine and chassis units at General Motors Institute for a number of years but has been limited by the available equipment to the following types of tests:

Variable speed.

Variable load.

Variable compression ratio.

Variable spark advance.

Suggested test demonstrations possible with suitable equipment are:

Effect of spark plug location and multiple plugs firing simultaneously on rate of pressure rise.

Effect of valve timing and duration upon the power peak speed.

Investigation of possible air/fuel ratios especially with lean mixtures.

Performance possible with new "super fuels."

Possible saving in fuel consumption by varying valve timing with load.

The purpose of this paper is to set forth the specifications and design suggestions for a single-cylinder test engine with which this instruction may be accomplished.

SPECIFICATIONS

Specifications for possible changes in the engine operation to be made while it is running.

Compression ratio from $3\frac{1}{2}$ to 15 to 1.

At least 2 indicator holes (18 mm.) and spark plug hole.

Opening for 4 or more spark plugs and 2 indicators.

Variable valve lift.

Variable duration of valve opening in camshaft degrees.

Variable timing for inlet and exhaust valves.

Adjust either valve by itself.

Because of the vibration inherent in a single-cylinder engine, balancing of reciprocating forces is recommended.

CYLINDER HEAD DESIGN

Overhead

High compression work is almost necessarily done on a valve-in-head type of cylinder arrangement since the valve space in an ell head limits its maximum ratio to about 10. Most laboratory engines of this type have a flat, cylindrical space as shown in Fig. 1.

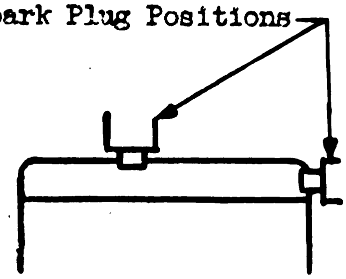


Fig. 1

Even with a central spark plug location the area to volume relationship is higher than normal, and when using the side plug location (which is often done) the flame pattern is distorted so as to be virtually useless for investigating the performance of conventional engines.

Following this line of reasoning, the first trial is illustrated in Fig. 2. The domed head permits a normal flame propagation with centrally located spark plug. Four other openings are provided - exhaust valve, inlet valve and two 18 mm. indicator holes. Because

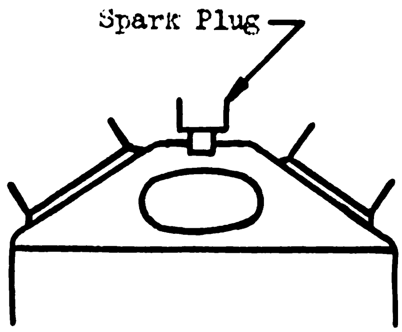


Fig. 2

of the space occupied by the spark plug, the valves were small by comparison with similar engines but since the maximum ratio was 12 to 1, this design was discarded. (See appendix for layout of this chamber space.)

The design finally adopted is a compromise. The head is slightly domed as illustrated in Fig. 3 with valves of reasonable size. With this arrangement it is not possible to put the spark plug in the center of the space, therefore it is located as nearly there as possible. This gives a 15 to 1 maximum ratio easily, but there is still the problem of 2 indicator openings. The solution arrived at is illustrated in the appendix and consists of a small connecting passage from the combustion chamber to the two 18 mm. openings.

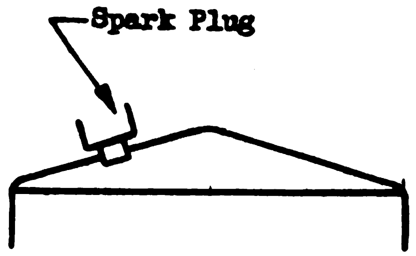


Fig. 3

This chamber permits reasonable flame propagation patterns with varying ratios and is superior to any now in use. The maximum value (15 to 1) is just barely sufficient to test the latest fuels under best conditions. Pure triptane has a critical compression ratio of 15 to 1, but to test this fuel with added tetra ethyl lead, it is necessary to supercharge the cylinder.

<u>Ell Head</u>

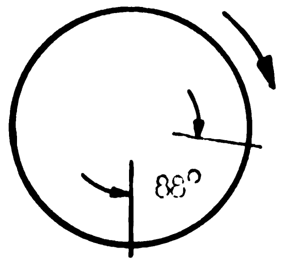
While limited insofar as maximum compression ratio is concerned, this arrangement lends itself very easily to investigation in the area of flame propagation. The proposed design allows for four 10 mm. spark plug openings and two 18 mm. indicator openings. Maximum ratio from the preliminary layout was 10 to 1, but an increase in the height of the valve chamber is suggested so that the cylinder will breathe better with maximum ratio about 9 to 1. (See the general design layout for further information.)

These two designs will satisfy all of the requirements and are easily interchanged. The valve mechanism is designed purposely to allow a small variation in assembled dimensions by using the hydraulic valve lash adjuster.

BREATHING SYSTEM -- GENERAL

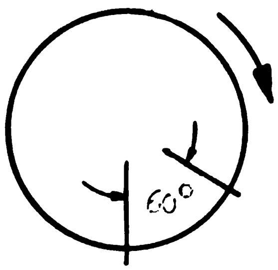
The breathing system of an internal combustion engine is vital in assuring good power output and economy as installed. The usual development program in this area is to make several experimental camshafts and try each under varying conditions until the best apparent combination is selected. Adjustments in timing are very coarse since the gears or chain sprockets must change to the next tooth unless s special driving hub is provided with three keyways indexed to split the tooth space into equal parts. This cut-and-try process is exceedingly time-consuming and expensive. An experimental camshaft, for example, costs several hundred dollars.

This means that only a few in the engineering department even have the opportunity of working out a problem of this sort and then only seldom since the expense is so great. For students in school a series of camshafts would be expensive and their use inconclusive since the timing change affects both valves at the same time and one is never <u>sure</u> whether the effect noted is due to one, the other, or the combination. The proposed specifications create an opportunity of changing any part of the valve movement for either, separately, while the engine is running, so that the actual effect may be noted. As Mr. Kettering so often says, we <u>think</u> that we know why a given mechanism performs as it does, but the only way of finding out for sure is to test it. As mentioned earlier, a typical assignment would be to set specifications similar to a commercial engine designed to operate economically at low speeds and establish the characteristics including power peak. Then by changing specifications to those for a high speed automotive-type engine the effect is shown clearly. Carrying this farther, we know that the mechanism adopted about fourteen years ago for adjusting the spark advance according to load increased the fuel ecnomy about 10% at small initial cost. We also know that present engines are designed to give maximum output at high speeds with reasonable economy at ordinary speeds. This means that the exhaust valve opens considerably before bottom dead center so that the products will have a chance of escaping in the short time available. The two-cycle diesel is an extreme example of this with the exhaust valve opening 88° early which literally wastes the energy still in the fuel at that point. Four-cycle engines do not need quite so much time, and the usual exhaust valve opening is 50-70° before bottom dead center in the high-speed automotive type. Slower speed engines open their valves later and are therefore able to utilize more of the energy stored up in the gases after combustion.



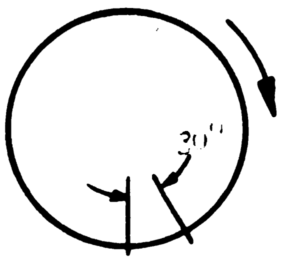
High Speed Diesel
Exhaust Valve Timing

Fig. 4



Automotive Type Engine
Exhaust Valve Timing

Fig. 5



Industrial Engine
Exhaust Valve Timing

Fig. 6

In addition to this point, while the engines are designed to operate on full load at high speed, in an automobile the load factor is usually one half or less which imposes a completely different set of conditions on the engine from those on which the design was based.

Assuming that the economy of engine operation in an automobile at constant speed on a level load, especially at low speeds, would be increased by allowing the products of combustion to expand as far as possible by opening the exhaust valve later, the following questions are important:

1. What increase in economy will result from the proposed installation?
2. How much will it cost per car?

The proposed design makes possible a laboratory demonstration of this problem to check the hypothesis and answer the first question.

It should be noted, however, that the inlet valve operation must be unchanged and the exhaust valve closing remain constant. This is done by altering the timing and the duration until the opening and closing events occur at the desired interval. A later valve closing changes the scavenging of the cylinder which may or may not affect the engine output.

Following the same line of reasoning, the power output is limited mostly by the inlet valve timing and duration. While we want the breathing capacity to be maximum at all times, it may be that optimum operations at half speed and load actually dictate completely different specifications. These we do not know at present except by opinion, since no one has tried it so far as the writer is aware.

VALVE OPERATING MECHANISMS

Production valves were selected for both arrangements for ease of procurement, and cam dimensions are established by the ell head design since the follower operates directly upon the can with no rocker arm as is the case with the overhead design. The valve sizes are 1 7/8 inlet and 1 5/8 outlet with 5/16 lift.

This is a modification of the 3-curve cam as usually designed consisting of a nose radius, with an involute and a radial flank blending to the base circle. (See Fig. 7.)

This cam is characterized by rather high initial acceleration on the radial flank, decreasing to zero on the involute flank then decelerating at nearly constant rate. (See Fig. 8 and the Appendix for a more complete analysis of inlet and exhaust cams.) It is noted that there is a short ramp on this cam, much less than is common with fixed lash adjustment. When using the hydraulic lash adjusters, it is usually recommended that the ramp be very short or eliminated altogether.

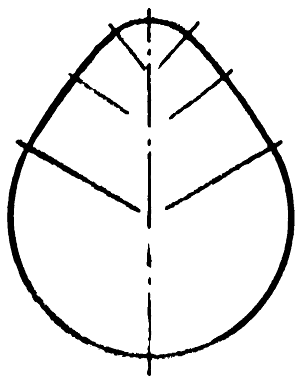


Fig. 7

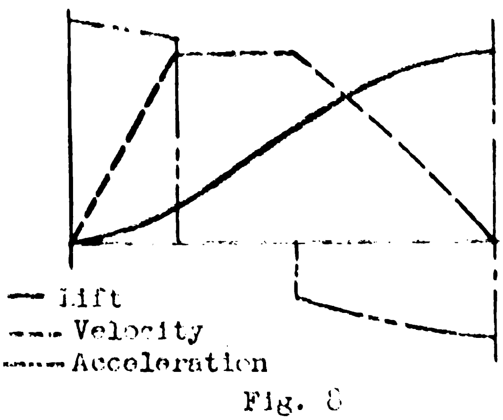


Fig. 8

Variable Timing

The proposed mechanism

for altering the angular relationship between crankshaft and camshaft utilizes one of the most common gear forms. With the worm in a given position, lengthwise, the angular velocity of the camshaft is exactly

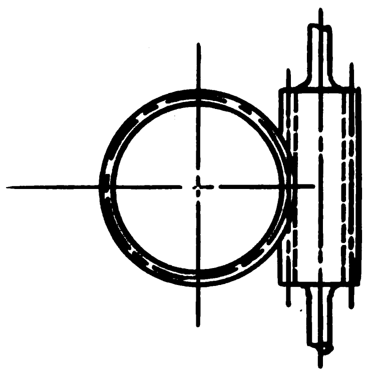


Fig. 9

proportional to that of the crankshaft. To change the angular relationship simply move the worm lengthwise in its bearings. This advances or retards the camshaft as desired. With a 4-cycle engine having a ratio of crankshaft to camshaft of 2 to 1 and a quadruple worm, it is necessary to turn the worm at twice engine speed. The gears will probably be noisy but in combination with the geared balance weights should not be objectionable.

<u>Variable Duration</u>

Any one of several quick-return mechanisms is suitable for

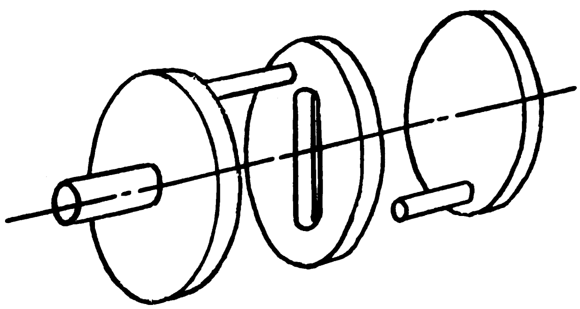


Fig. 10

increasing or decreasing the duration of valve opening while the engine is running. The one selected consists of a driving flange and pin, driven flange with pin 180° from the former, and a link which rotates about its own axis and may be offset from the axis of driving and driven flanges. With the link "on center," motion is transmitted uniformly from driver to driven, but when off-set up the driven member has a non-uniform motion, faster in the upper position (opposite from that shown) and slower in the one shown. Graphical layout and maximum non-uniformity are illustrated in the appendix.

Variable Lift

The conventional pivoted lever is used to vary the valve lift from 25% over that of the cam (to 3/8 total) and decrease to about 30% of the actual maximum lift or one eighth inch approximately. This enables the operator to vary the breathing capacity simply while the engine is running by increasing or decreasing the valve lift.

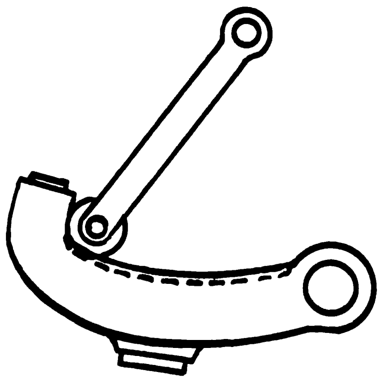


Fig. 11

Cylinder Elevating Mechanism

Engines of this type usually use a worm or thread arrangement for raising and lowering the cylinder with respect to the piston; and differences are in the locking device used. A positive lock as shown in the illustration is preferred. Using four sets of right and left hand screws, each one opposite, and driving from a pair of gears, worm "A" turns opposite from worm "B" but since threads A, B are themselves opposite, the upper casting moves up or down with respect to the lower. By turning A alone, the two screws are opposed and the set locks in a given position.

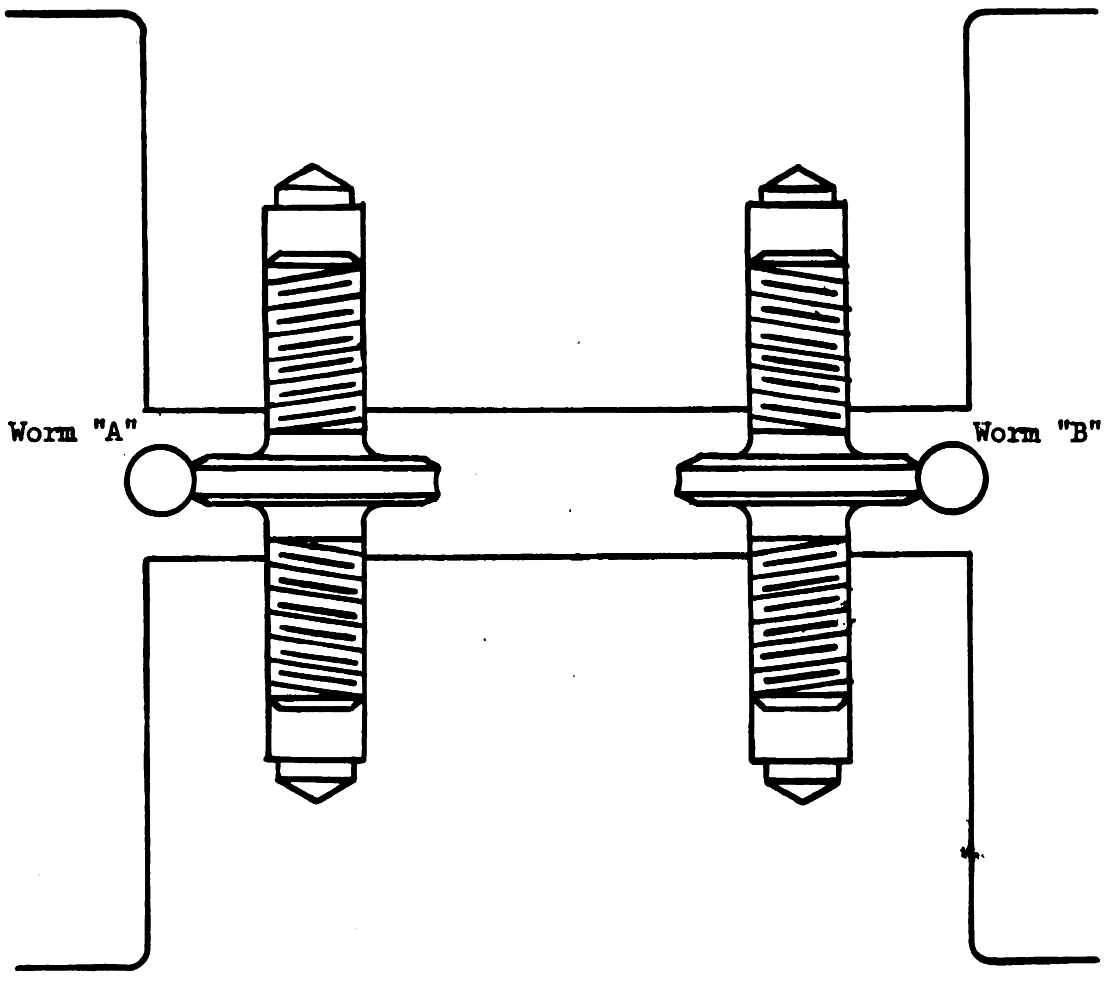


Fig. 12

INERTIA FORCES OF RECIPROCATING PARTS

Movement of the piston by a crankshaft and connecting rod

$$S = L + R - l \cos 0 - R \cos \theta \qquad (1)$$

$$L^2 = R^2 \sin^2 \theta + L^2 \cos^2$$

$$L \cos 0 = \sqrt{L^2 - R^2 \sin^2 \theta}$$

substituting back into equation (1)

$$S = R - R \cos \theta + \frac{L}{R} \left(1 - \sqrt{1 - \frac{R^2}{L^2} \sin^2 \theta}\ \right)$$

or approximately $S = R\,(1 - \cos \theta) + \frac{R}{4L}\,(1 - \cos 2\,\theta)$ (1a)

$$\text{Velocity} = \frac{d\theta}{dt} \cdot \frac{dt}{d\theta} = Rw\,(\sin \theta + \frac{R}{L} \sin 2\,\theta) \qquad (2)$$

$$\text{Acceleration} = \frac{dv}{d\theta} \cdot \frac{d\theta}{dt} = Rw^2\,(\cos \theta + \frac{R}{L} \cos 2\,\theta) \qquad (3)$$

$$\text{Inertia Force} = \frac{W}{g}\,Rw^2\,(\cos \theta + \frac{R}{L} \cos 2\,\theta) \qquad (4)$$

$$= .0000\ 284 N^2 WR\,(\cos \theta + \frac{R}{L} \cos 2\,\theta)$$

See - INTERNAL COMBUSTION ENGINES by Lichty, page 481.

PRIMARY INERTIA FORCES

The $\frac{R}{L}$ ratio for this engine is $\frac{2.25}{11} = .2045$. Working the problem in pound units at 2500 rpm the primary inertia force is $F = .0000284\ N^2WR \cos\theta$ with maximum value at 0°.

Maximum value $F = .0000284 \times 2500^2 \times 3.75 \times 2.25 = 1500$ lbs.

Equal and opposite forces are introduced by attaching reciprocating weights to the crankshaft by means of short connecting rods and an eccentric, or pairs of counter-rotating unbalanced weights which create a simple harmonic force in a vertical direction equal and opposite to the disturbing force.

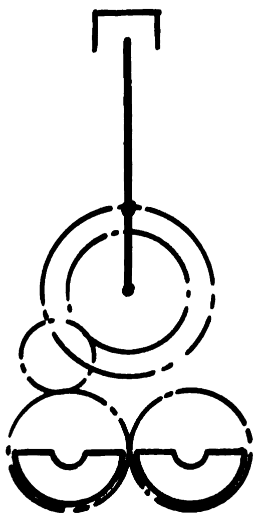


Fig. 13

Horizontal forces are equal and opposite therefore cancel each other.

Since the primary is by definition the component part of the total piston motion which is a simple harmonic motion at crankshaft speed, the force exerted by the two weights will always be equal to the disturbing primary and opposite in direction. These weights must be located immediately beneath the piston or, using two sets or two gears each, so that the resultant force will be. (See appendix for further analysis and sample computation of weights.)

SECONDARY UNBALANCED FORCES

From inertia force equation (4) the expression for secondary inertia force is

$$F = .0000284 N W \times R \left(\frac{R}{L} \cos 2\theta\right)$$

In this instance the secondary force amounts to

$F = .0000284 \times 2500^2 \times 3.75 \times 2.25 \times .2045 \cos 2\theta$ with a maximum value when $\theta = 0$ or 180 $F_o{}^o = 306\#$.

These forces are opposed by two sets of counter-rotating weights driven at twice crankshaft speed and disposed so that their resultant force is equal in amount to the disturbing secondary but opposite in direction.

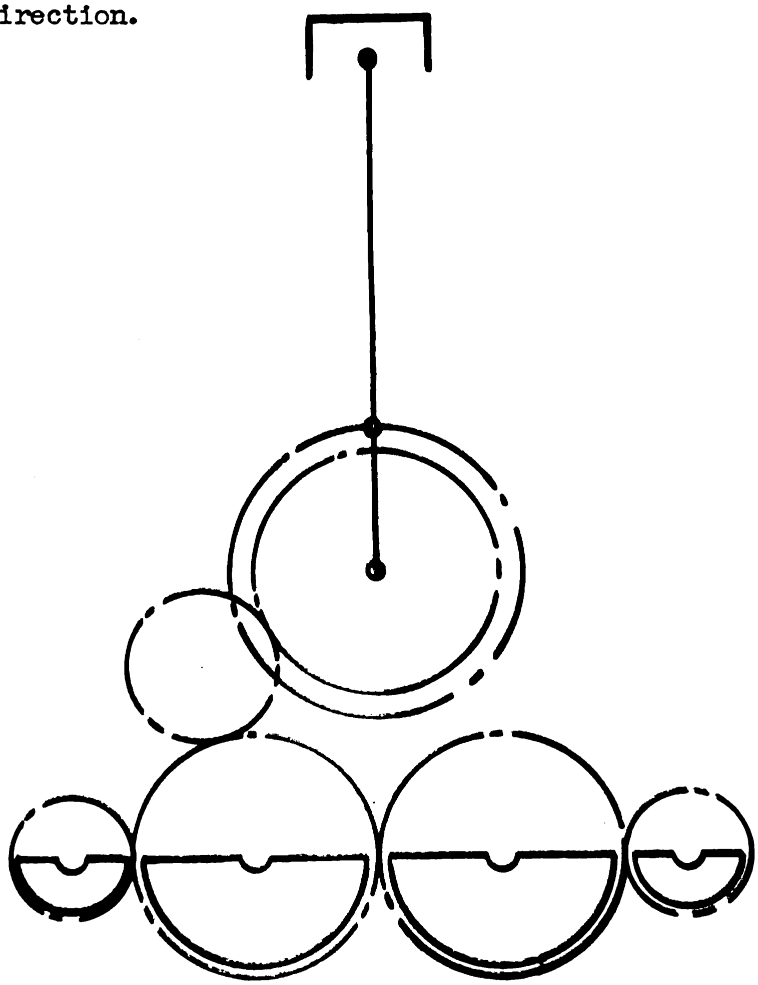


Fig. 14

Diagram for primary and secondary balance weights.

APPENDIX

C. Ratio	8.29 in^2 Clearance in^2	Clearance @ $3\frac{1}{4}$ Dia.	Less .521
15	2.66	.321	0
14	2.87	.346	.025
13	3.11	.775	.054
12	3.39	.409	.088
11	3.73	.450	.129
10	4.15	.5	.179
9	4.67	.554	.233
8	5.33	.641	.320
7	6.22	.75	.429
6	7.46	.9	.579
5	9.33	1.125	.804
4	12.42	1.5	1.179
3.4	14.9	1.8	1.479

Clearance = 1.83 (irregular volume)
.0858 ($\frac{1}{4}$ dia. hole)
.0247 (clearance in individual holes)
1.9405

for 15/1 clearance = 2.66 in^3 $\left(\frac{37.3}{14}\right)$

subtract 1.94 leaving .72 in^3

@ 3.3125 dia. .72 in^3 corresponds with

$\frac{.72}{8.6}$ area = .084

$S = R(1 - \cos\theta + .5K \sin 2\theta)$ $K = \frac{2.25}{11} = .2045$ $\frac{L}{K} = 4.89$

	V_x(in.)	$\frac{5.14286}{V_x}$	$(\ )^{1.33}$	X 12	V_x/V_3	$750/(\ )^{1.38}$	$570/(\ )^{1.28}$
0	.64286	8	15.9	191	1	1	570
2	.64451	7.97	15.8	189.8	1.0025	1.003	569
4	.64946	7.91	15.6	187.2	1.0104	1.0134	563
6	.65766	7.81	15.45	185.5	1.022	1.0283	555
8	.66926	7.69	15.1	181.2	1.04	1.0515	542
10	.68396	7.51	14.6	175.2	1.064	1.0826	526
12½	.70696	7.26	14	168	1.1	1.1282	506
15	.73486	7	13.3	159.8	1.143	1.1867	481
17½	.76786	6.69	12.5	150	1.195	1.254	455
20	.80566	6.36	11.7	140.5	1.257	1.34	425
25	.89436	5.75	10.25	123	1.39	1.525	374
30	1.00186	5.14	8.82	106	1.56	1.766	323
35	1.12586	4.56	7.77	93.4	1.752	2.05	278
40	1.26486	4.06	6.45	77.4	1.97	2.382	239
45	1.41786	3.63	5.55	66.5	2.2	2.75	207
50	1.58186	3.24	4.78	57.4	2.46	3.16	180.3
60	1.94286	2.64	3.635	43.6	3.024	4.13	138
70	2.32286	2.21	2.87	34.4	3.62	5.2	110
80	2.72486	1.881	2.32	27.8	4.25	6.39	89.4
90	3.12480	1.64	1.93	23.2	4.865	7.6	75
100	3.50886	1.465	1.663	19.9	5.46	8.8	64.8
120	4.18686	1.228	1.314	15.8	6.5	11	51.8
140	4.70286	1.09	1.123	13.5	7.31	12.8	44.6

	V_x(in.)	$\frac{5.14286}{V_x}$	$(\ )^{1.33}$	x 12	V_x/V_3	$750/(\ )^{1.38}$	$570/(\ )^{1.28}$
160	4.94286	1.04	1.0535	12.7	7.68	13.6	42
180	5.14286	1	1	12	8	14.3	40

Assuming ratio of 8/1 decreases, inches $= \frac{4.5}{7} = .64285$

Assume $P_L = 12$ psi abs.

$P_1V_1^{\ n} = P_x V_x^{\ n}$ $\quad P_x = P_1\left(\frac{V_1}{V_3}\right)^n$ $\qquad P_3V_3^{\ n} = P_xV_x^{\ n}$

Use $n = 1.33$ compression $\qquad P_x = P_3\left(\frac{V_x}{V_3}\right)^n$

1.29 expansion

2500 rpm $\quad 208.73 \times 2.500 \times 2.25 = 49.1$

S   R(1 - cos θ + .5K sin 2 θ)   K $\frac{2.25}{11}$ .2045 $\frac{L}{K}$ 4.89

	1 cos θ	Sin θ	Sin 2 θ	X $\frac{K}{L}$	S Factor	RX Factor = Stroke
0	0	0	0	0	0	0
2	.00061	.0349	.00122	.0001248	.000735	.00165
4	.00244	.06976	.00486	.000497	.002937	.0066
6	.00548	.10453	.0109	.001115	.006595	.0148
8	.00973	.13917	.0194	.001985	.011715	.0264
10	.01519	.17365	.0301	.00308	.01827	.0411
12½	.02370	.21644	.0469	.0048	.0285	.0641
15	.03407	.25882	.067	.00685	.04092	.092
17½	.04628	.30071	.0902	.00923	.05551	.125
20	.06031	.34202	.117	.01197	.27228	.1628
25	.09369	.42262	.178	.0182	.11189	.2515
30	.13397	.5	.25	.0256	.15957	.359
35	.18085	.57358	.328	.0335	.21435	.483
40	.23396	.64229	.413	.0422	.27616	.622
45	.29289	.70711	.5	.0511	.34399	.775
50	.35721	.76604	.589	.0602	.41741	.939
60	.5	.86603	.75	.0766	.5766	1.3
70	.65798	.93969	.88	.090	.74798	1.68
80	.82635	.98481	.969	.099	.92535	2.082
90	1	1	1	.1022	1.1022	2.482
100	1.17365	..98481	.969	.099	1.27268	2.866
120	1.5	.86603	.75	.0766	1.5766	3.544

	1 cos θ	Sin θ	Sin 2 θ	x $\frac{K}{L}$	S Factor	RX Factor = Stroke
140	1.76604	.64279	.413	.0422	1.80824	4.06
160	1.93969	.34202	.117	.01197	1.95166	4.3
180	2	0	0	0	2	4.5

Assuming ratio of 8/1 decreases, inches = $\frac{4.5}{7}$ = .64285

Assume P_L = 12 psi abs.

$P_1V_1^n = P_xV_x^n$ $\quad$ $P_x = P_1\left(\frac{V_1}{V_x}\right)^n$ $\quad$ $P_3V_3^n - P_xV_x^n$

$P_x = P_3 \div \left(\frac{V_x}{V_3}\right)^n$

Use n = 1.33 compression

1.29 Expansion

Velocity = .00873 NR$(\sin\theta + \frac{K}{2}\sin 2\theta)$

Acceleration = .000914 $N^2R(\cos\theta + K\cos 2\theta)$ = 12,880

In Force = .0000284 N WR$(\cos\theta + K\cos 2\theta)$ =

	Sin 2 θ	X · 1022	Sin θ	Vel Factor		Cos 2 θ	X K	Cos θ
0	0	0	0	0	0	1	.2045	1
2	.06976	.00714	.03490	.04204	2.06	.99756	.102	.99939
4	.13917	.01425	.06796	.08221	4.04	.99027	.1012	.99756
6	.20791	.02125	.10453	.12578	6.26	.97815	.1	.99452
8	.27564	:02817	.13917	.16734	8.21	.96126	.0984	.99027
10	.34202	.035	.17365	.20865	10.24	.93969	.096	.98481
12½	.42262	.0436	.21644	.26004	12.78	.09631	.0928	.97630
15	.5	.0511	.25882	.30992	15.2	.86603	.0886	.96593
17½	.57358	.0586	.30071	.35931	17.62	.81915	.0837	.95372
20	.64279	.06565	.34202	.40767	20	.76604	.0785	.93969
25	.76604	.0785	.42262	.50112	24.6	.64279	.0657	.90631
30	.86603	.0886	.5	.5886	28.85	.5	.0511	.86603
35	.93969	.096	.57358	.66958	32.8	.34202	.035	.81915
40	.98481	.1008	.64279	.74359	36.45	.17365	.01775	.76604

	Sin 2 θ	X 1022	Sin θ	Vel Factor		Cos 2 θ	X K	Cos θ
45	1	.1022	.70711	.80931	39.7	0	0	.70711
50	.98481	.1008	.76604	.86684	42.6	-.19081	-.0195	.64279
60	.86603	.0886	.86603	.95463	46.8	-.5	-.0511	.5
70	.64279	.06565	.93969	1.00534	49.4	-.76604	-.0785	.34202
75	.5	.0511	.96593	1.01703	49.8			
80	.34202	.035	.98481	1.0198	50.	-.93969	-.096	.17365
85	.17365	.01776	.99619	1.01395	49.7			
90	0	0	1	1.	49.1	-1	-.1022	0
100	-.34202	-.035	.98481	.94981	46.5	-.93969	-.096	-.17365
120	-.866	-.0886	.86603	.7743	38	-.5	-.0511	-.5
130	-.98481	-.1008	.76604	.66684	32.8	-.17365	-.01775	-.64279
140	-.98481	-.1008	.64279	.54199	26.6	+.17365	+.01775	-.76604
150	-.866	-.0886	.5	.4114	20.2	+.5	+.0511	-.86603
160	-.64279	-.06565	.34202	.28637	14.08	+.76604	+.0785	-.93969
170	-.34202	-.035	.17365	.13865	6.8	+.93969	+.096	-.98481
172	-.27564	-.02817	.13917	.111	5.45	+.96126	.0985	-.99027

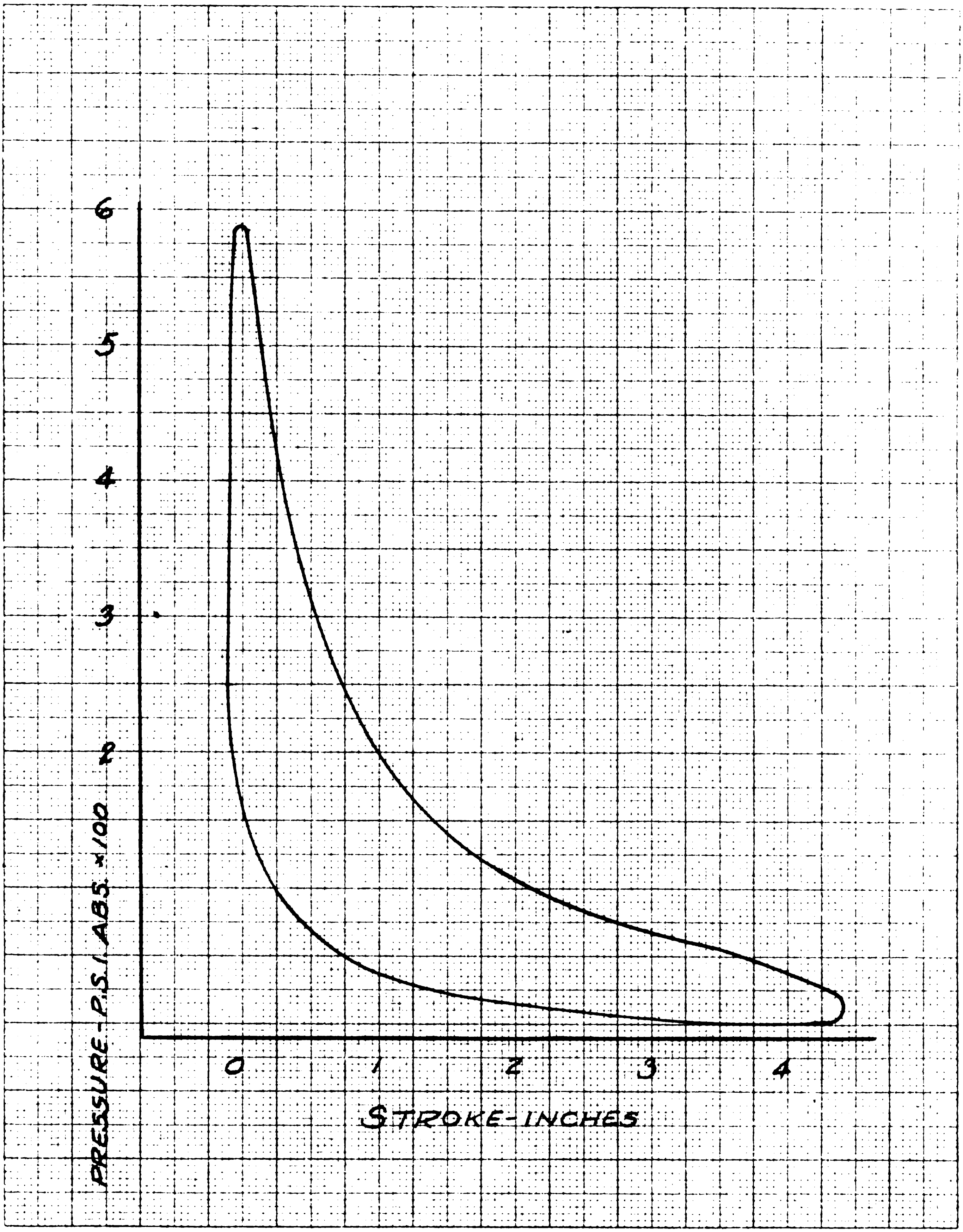
	Sin 2 θ	X · 1022	Sin θ	Vel Factor		Cos 2 θ	X K	Cos θ
174	-.20791	-.02125	.10453	.08328	4.09	+.97815	.1	-.99452
176	-.13917	-.01425	.06976	.05551	2.72	+.99027	+.1012	-.99756
178	-.06796	-.00714	.03490	.02776	1.36	+.99756	+.102	-.99939
180	0	0	0	0	0	1	+.1022	-1

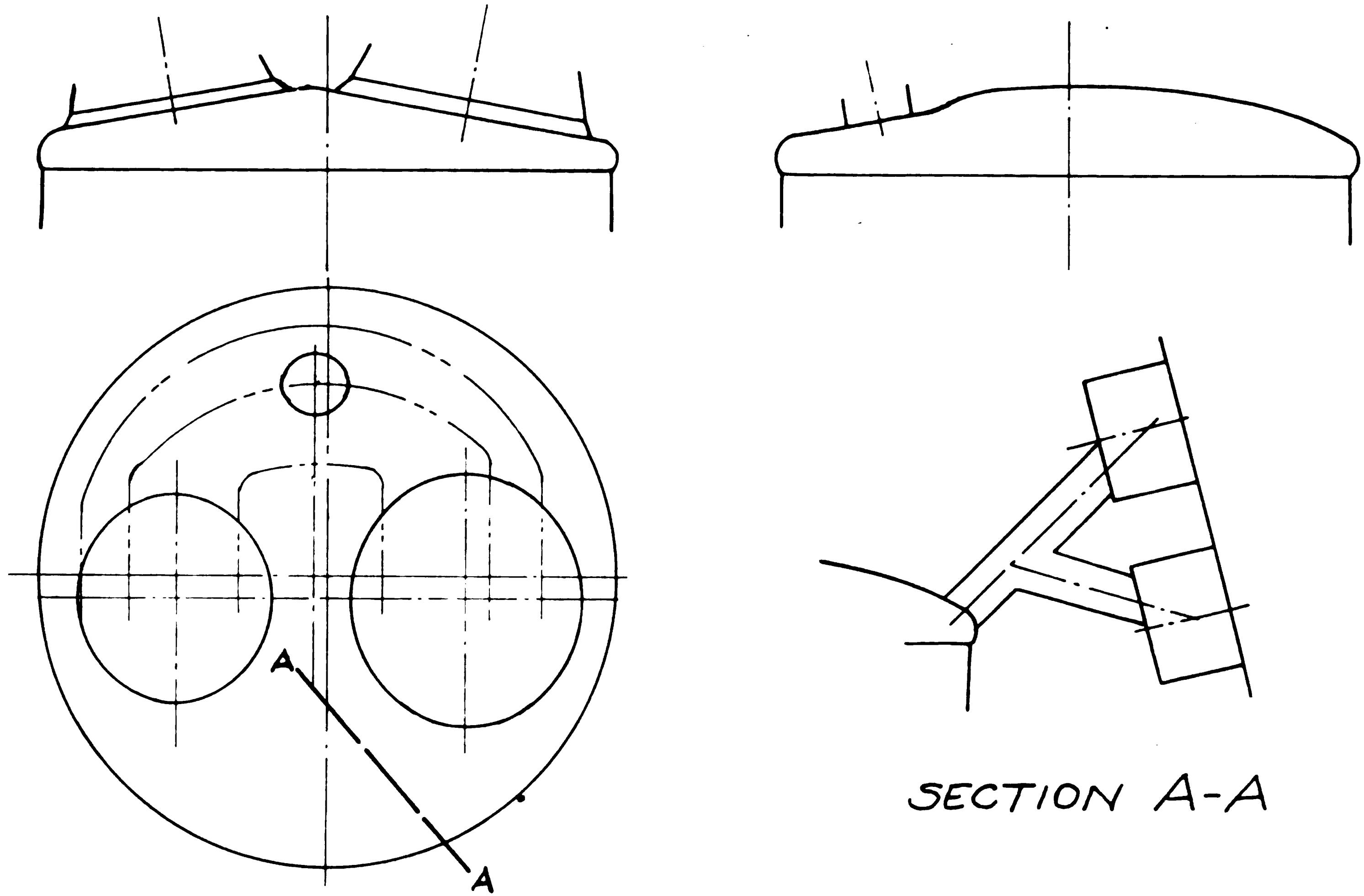
Acceleration $= .000914\, N^2R(\cos\theta + K\cos 2\theta) = 12{,}850\ (\qquad)$

Inertia Force $= .0000284\, N^2WR(\cos\theta + K\cos 2\theta) = 1500\ (\qquad)$

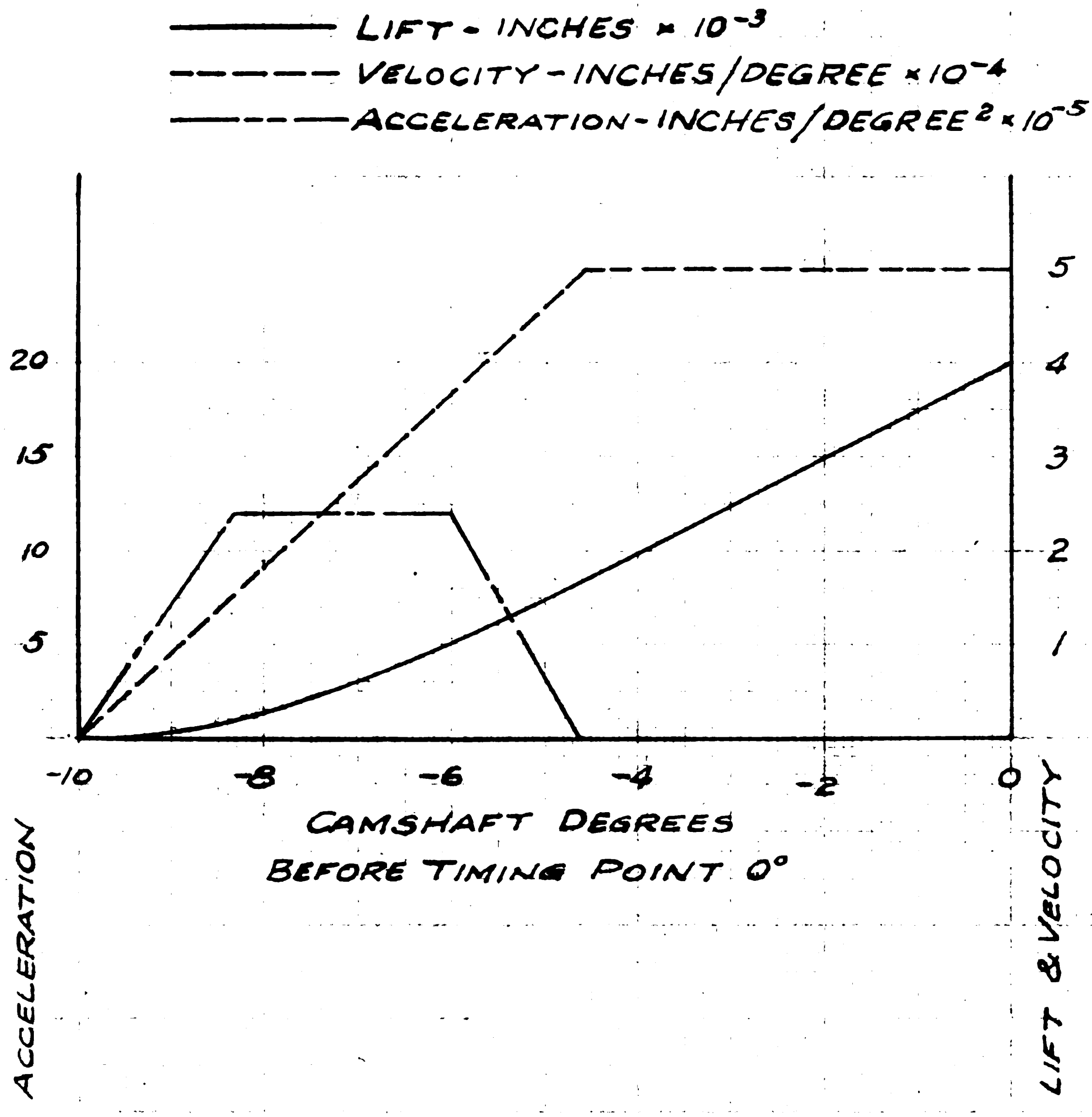
	Cos 2 θ	X K	Cos θ	Factor	Accel.	In Force
0	1	.2045	1	1.2045	15,480	1800#
2	.00756	.204	.99939	1.2034	15,470	1800
4	.99027	.2025	.99756	1.20	15,410	1796
6	.97815	.2	.99452	1.1945	15,340	1785
8	.96126	.1968	.99027	1.1871	15,220	1775
10	.93969	.192	.98481	1.1768	15,100	1759
12½	.90631	.1853	.97630	1.1616	14,910	1740
15	.86603	.1771	.96593	1.1438	14,700	1711
17½	.81915	.1671	.95372	1.1208	14,400	1679
20	.76604	.1569	.93969	1.09659	14,100	1641
25	.64279	.1313	.90631	1.03761	13,300	1550
30	.5	.1022	.86603	.96823	12,420	1450
35	.34202	.0699	.81915	.88905	11,410	1330
40	.17365	.0357	.76604	.80174	10,300	1200
45	0	0	.70711	.70711	9,090	1058
50	-.19081	-.039	.64279	.6058	7,800	909
60	-.5	-.1022	.5	.3928	5,110	595
70	-.76604	-.157	.34202	.1850	2,376	276
80	-.93969	-.192	.17365	-.018	-231	26.9
90	-1	-.2045	0	-.2045	2,625	306
100	-97969	-.192	-.17365	-3.6565	4,695	545.5
120	-.5	-.1022	-.5	-6022	7,740	900

	Cos 2 θ	X K	Cos θ	Factor	Accel.	In Force
130	-.17365	-.0354	-.64279	-.67819	8,700	1012#
140	+.17365	+.0354	-.76604	-.73064	9,390	1091
150	+.5	+.1022	-.86603	-.76383	9,800	1140
160	+.76604	+.157	-.93969	-.78269	10,050	1168
170	+.93969	.192	-.98481	-.79281	10,180	1182
172	.96126	.1968	-.99027	-.79347	10,195	1185
174	.97815	.2	-.99452	-.79452	10,200	1189
176	.99027	.2025	-.99756	-.79506	10,210	1190
178	.99756	.204	-.99939	-.79539	10,220	1191
180	1	.2045	-1	-.7955	10,230	1193


PRESSURE-P.S.I. ABS. ×100
6
5
4
3
2
0
1
2
3
4
STROKE-INCHES

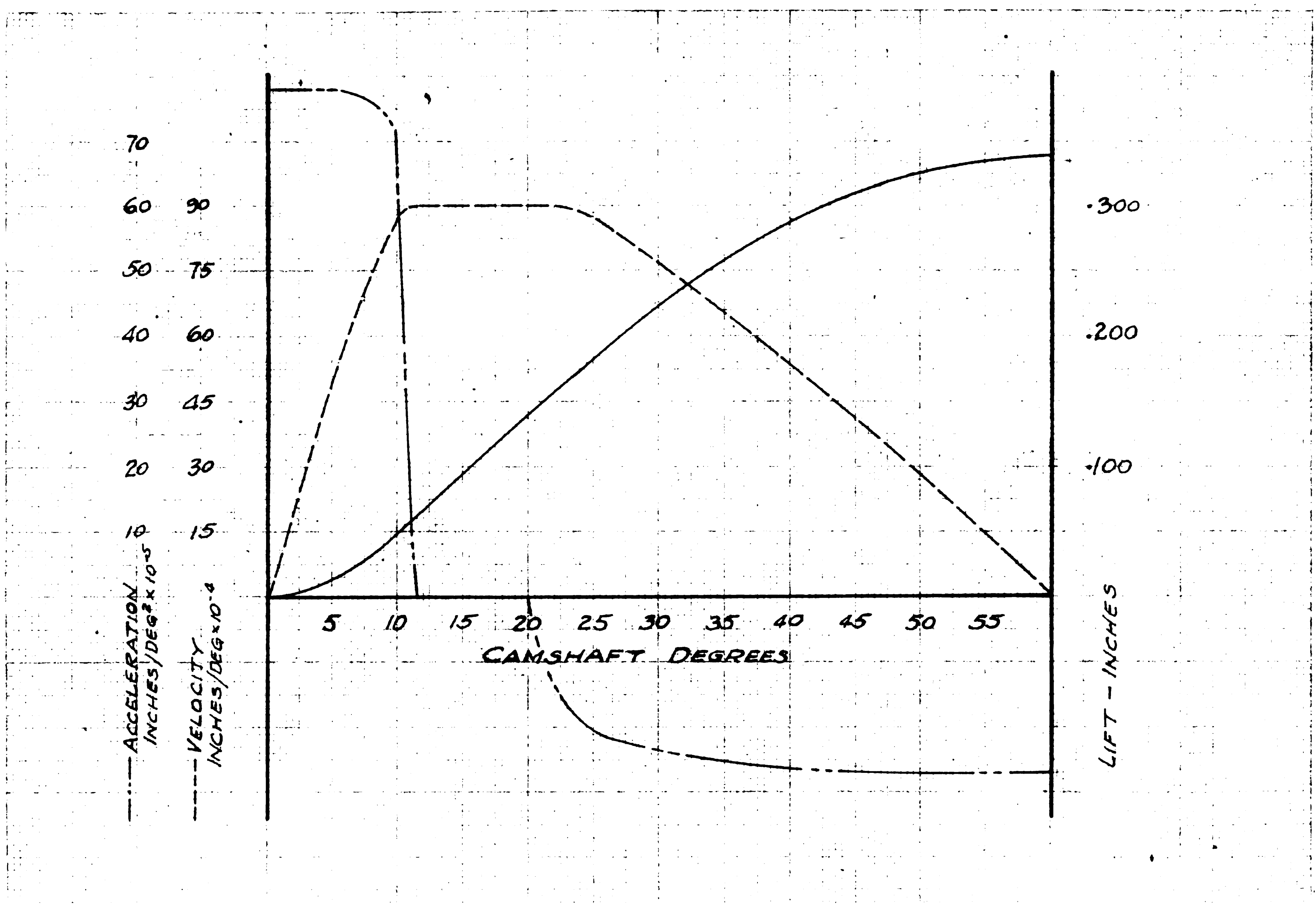


SECTION A-A

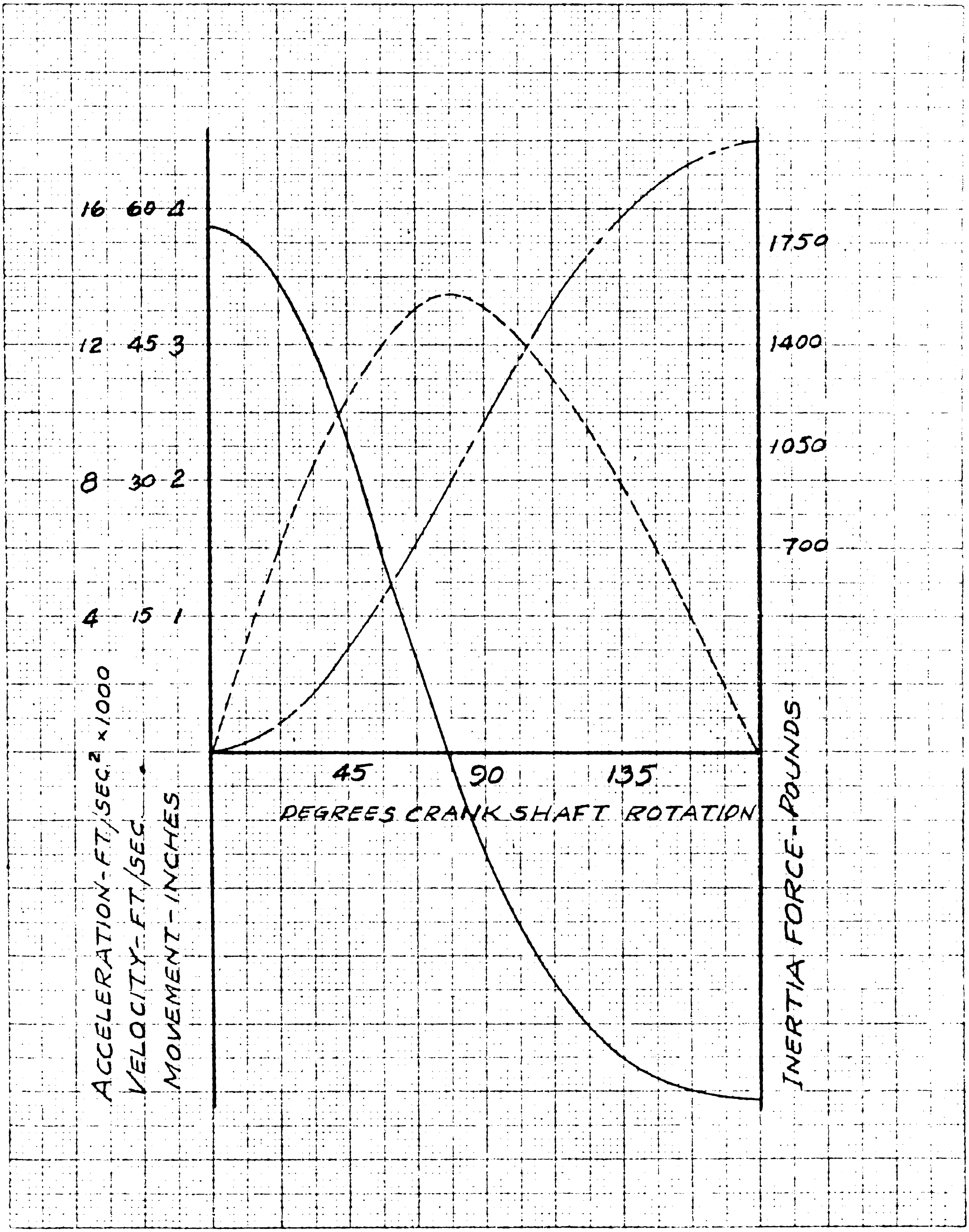


CHARACTERISTIC CAMSHAFT CURVES

RAMP

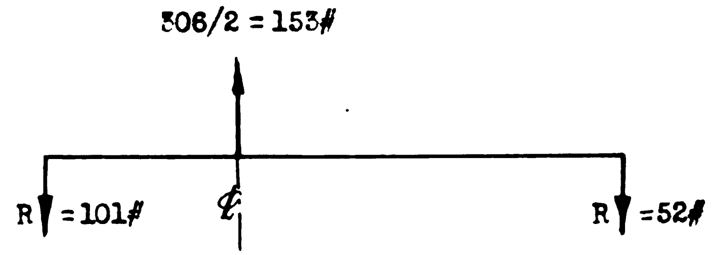


70
60
50
40
30
20
10
90
75
60
45
30
15
ACCELERATION
INCHES/DEG² x 10⁻⁵
VELOCITY
INCHES/DEG x 10⁻⁴
5
10
15
20
25
30
35
40
45
50
55
CAMSHAFT DEGREES
.300
.200
.100
LIFT - INCHES

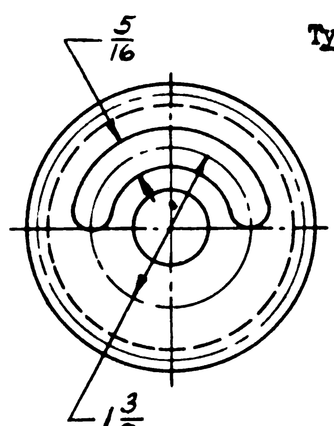


16 60 4
12 45 3
8 30 2
4 15 1
1750
1400
1050
700
45
90
135
DEGREES CRANK SHAFT ROTATION
ACCELERATION-FT./SEC.² ×1,000
VELOCITY-FT./SEC
MOVEMENT-INCHES
INERTIA FORCE-POUNDS

BALANCING WEIGHTS FOR SECONDARY INERTIA FORCES



Limitation of space available immediately beneath the piston necessitates using two sets of 2 weights each. These are not equally situated on each side of the piston center-line, lengthwise, and therefore will not be the same on each end. The diagram above indicates the relative positions and amount of force necessary to oppose the inertia force of the piston.



Steel Gear - 1" Thick

Typical Computation for Weights Shown at Right, In Diagram Above

Area	Lever Arm	Moment
1.12	.359	.401
.44	.225	.099
0.68		0.302

Centroidal distance $\frac{.302}{.68} = .444$

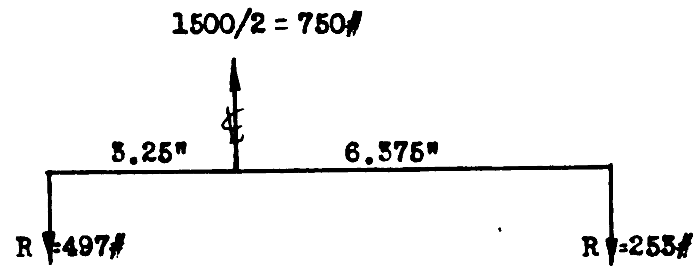
Weight x Radius .194 x .444 = .086

Weight removed .68 x 1" x .284 = 0.194

Centrifugal force corresponding with this weight x radius :

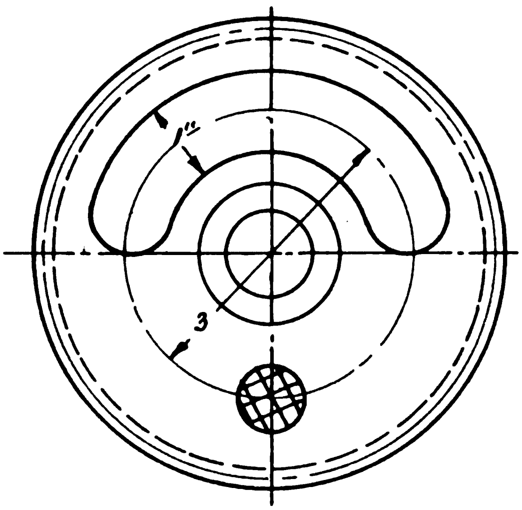
.0000284 x 5000 x 5000 x .086 = 60#

BALANCING WEIGHTS FOR PRIMARY INERTIA FORCES



Limitation of space available immediately beneath the piston necessitates two sets of 2 weights each. These are not equally situated on each side of the piston center-line, lengthwise, and therefore will not be the same on each end. The above diagram indicates relative positions and amount necessary to oppose the inertia force of the piston.

<u>Typical Computation For Weights Shown At Right, Above</u>



Steel Gear - 1" Thick

Force due to weight removed by semi-circular slot :

Area	Lever Arm	Moment
6.28	.850	5.34
1.57	.425	.67
4.71		4.67

Centroidal distance $\frac{4.67}{4.71} = 0.99"$

Weight removed 4.71 x 1" x .284

= 1.34#

Weight x Radius 1.34# x .99 = 1.33

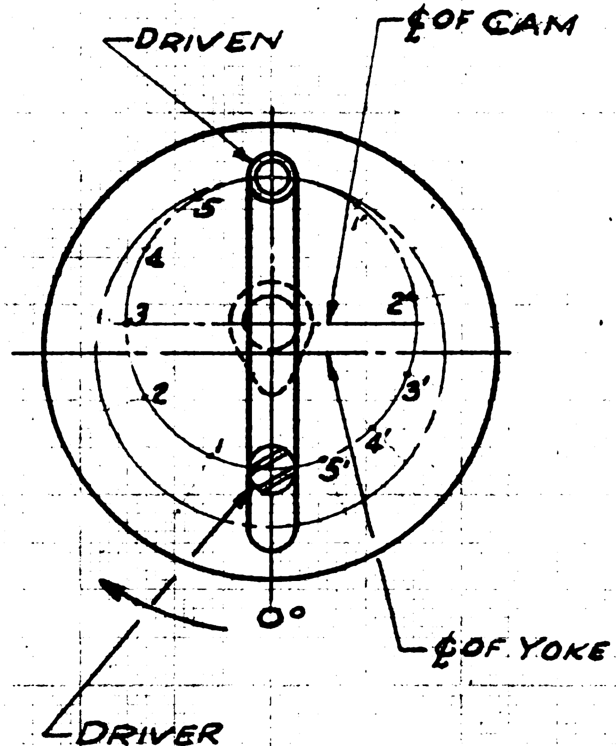
For 3/4 dia. hole filled with lead

Weight x Radius .442 x .1275 x 1.5

= 0.0845

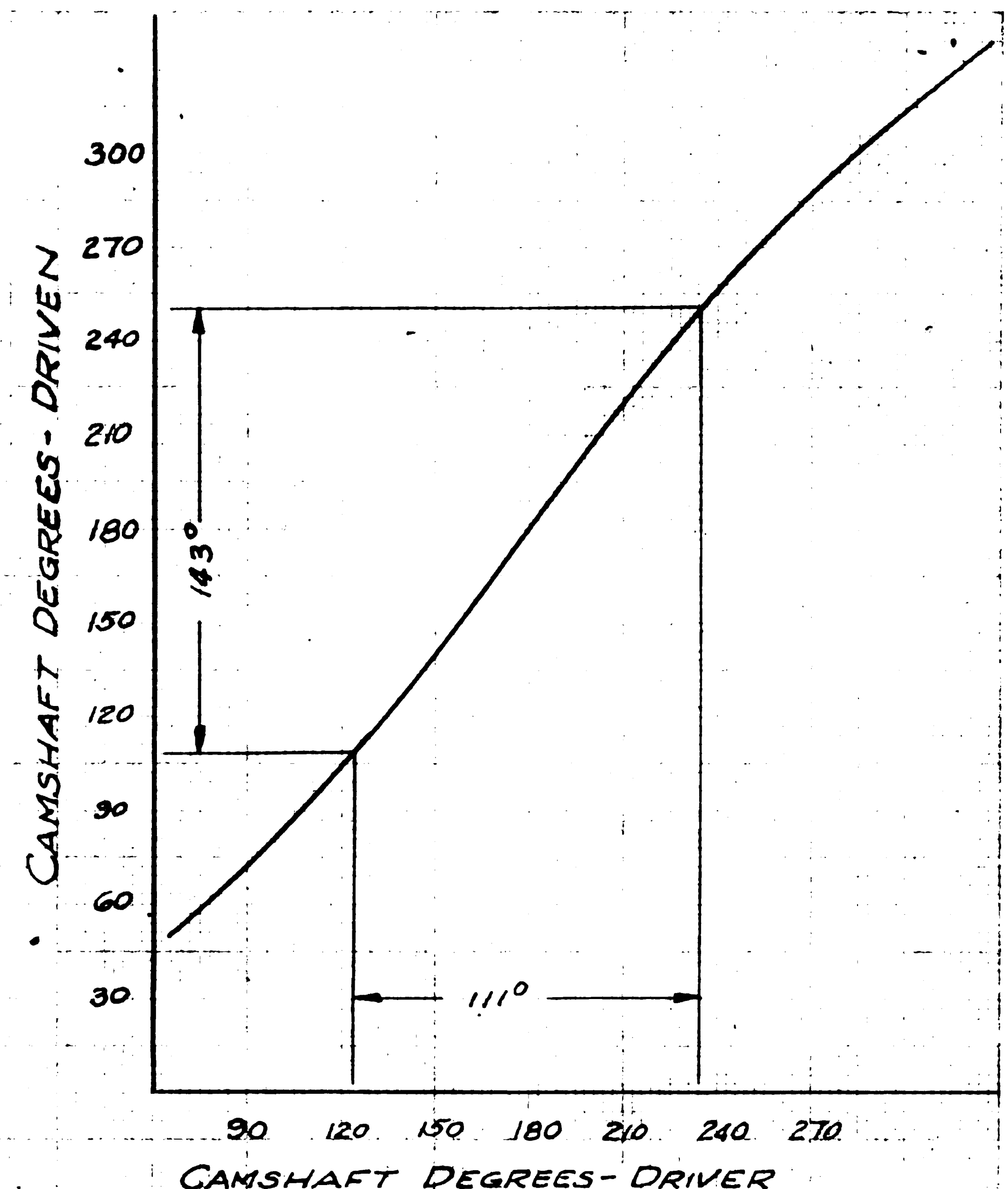
Centrifugal force corresponding with this is :

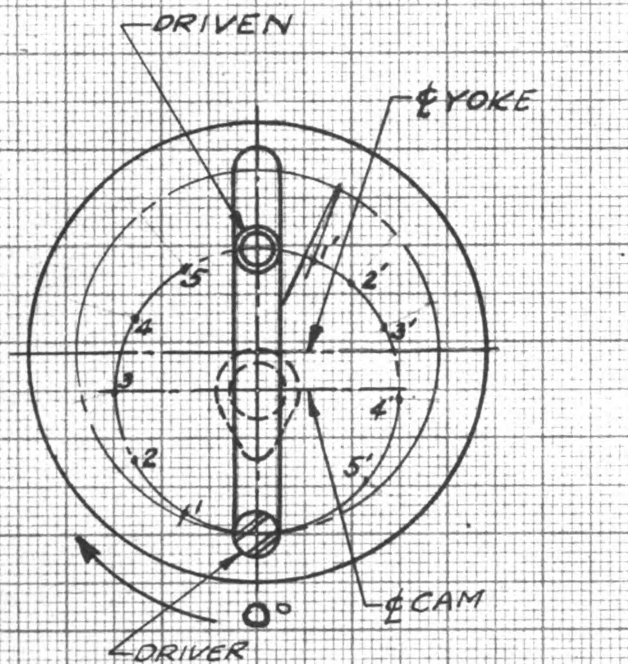
.0000284 x 2500 x 2500 x(1.33 + .0845) = 250#



MAXIMUM OFFSET

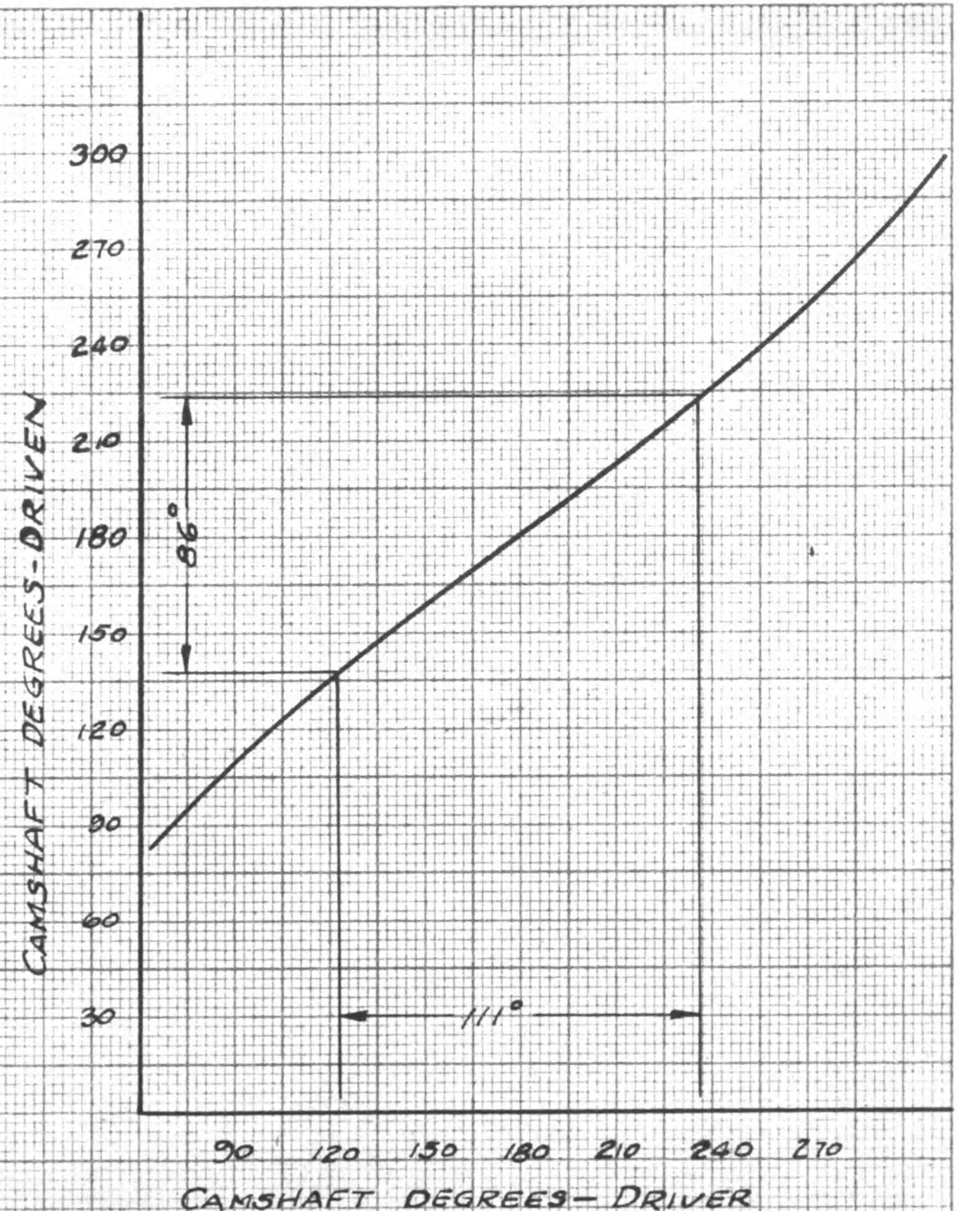
FAST TIMING

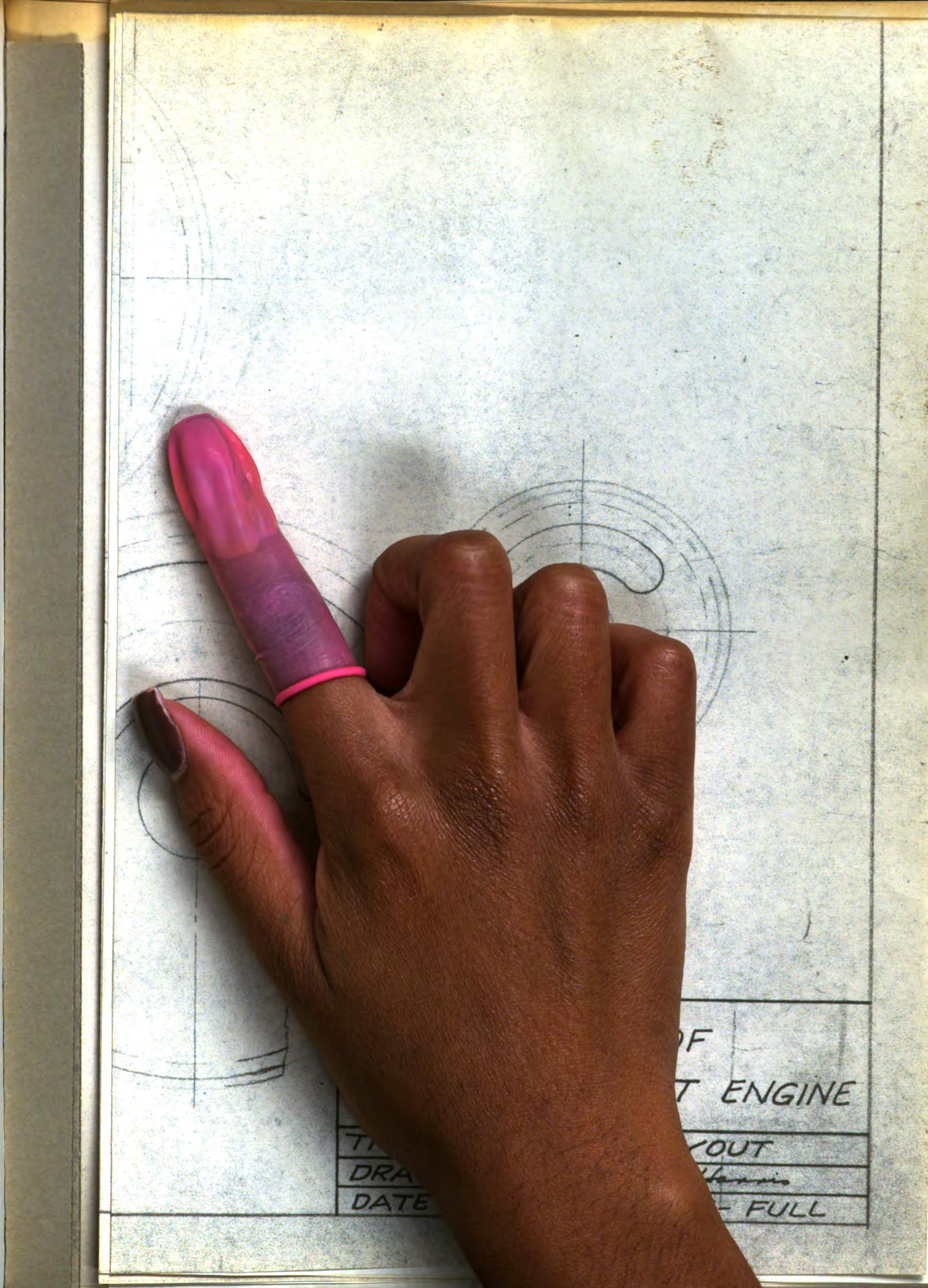




MAXIMUM OFFSET

SLOW TIMING




ENGINE
DATE
FULL