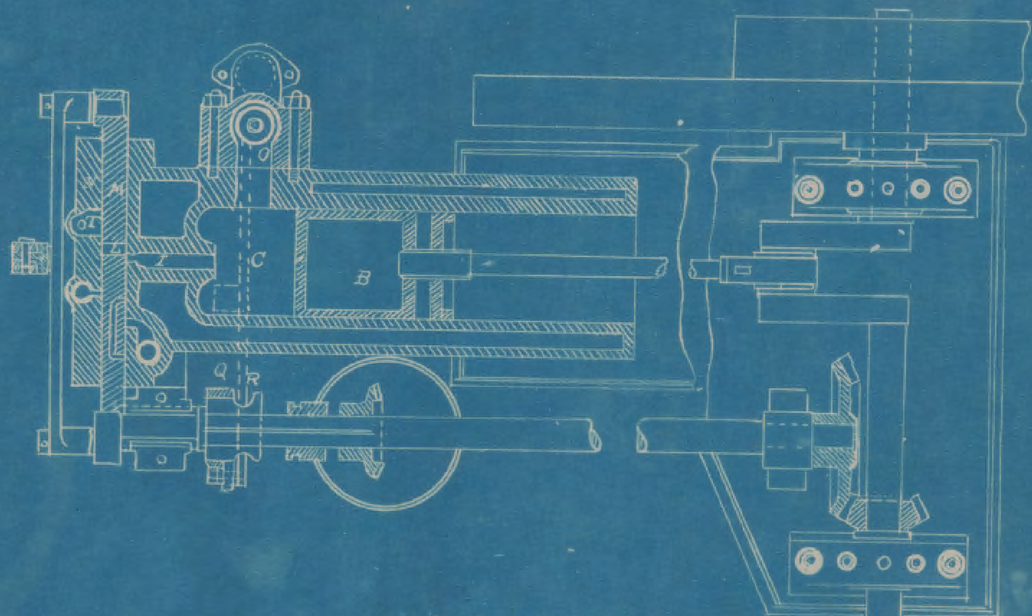
14837

Annie J. Grimm

OTTO



Otto Engine (Sectional plan).

Efficiency Test of the "Otto" Gas Engine.

A Thesis

by

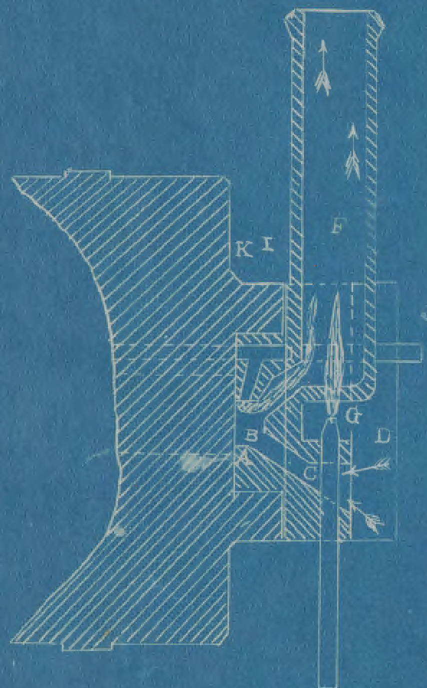
Ralph George Taylor

Presented for the Degree of Mechanical Engineer.

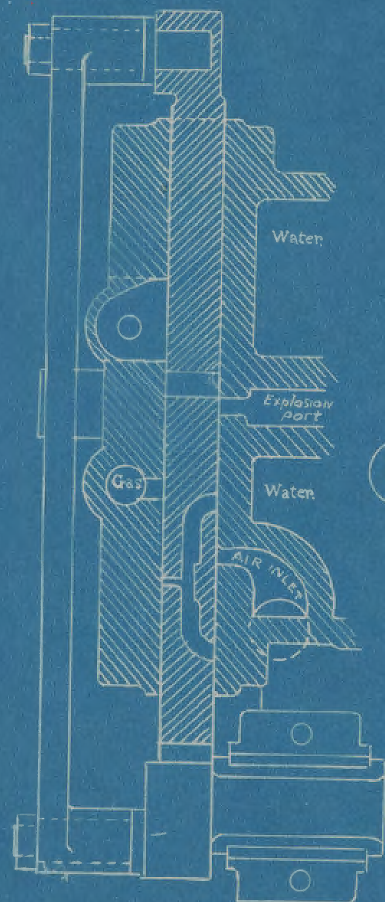
at

University of Minnesota

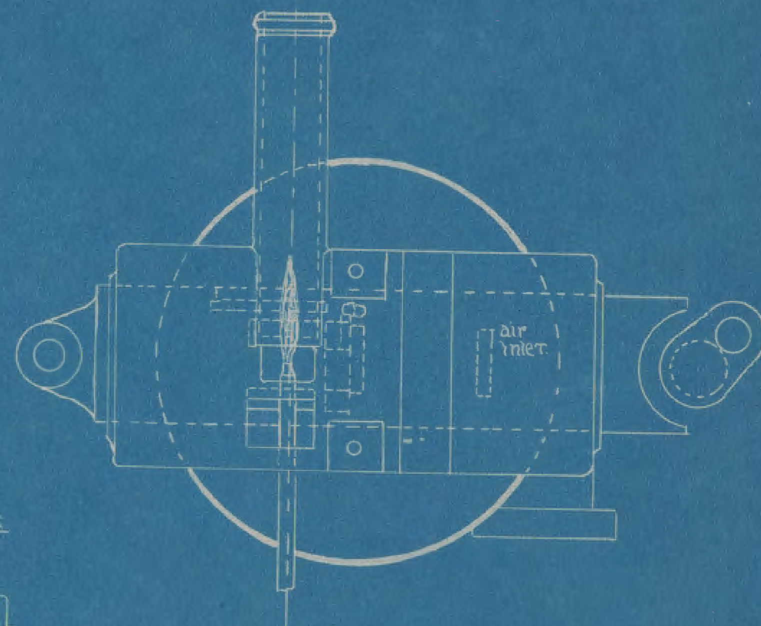
June 1902.



Section, Otto Igniting Valve.



Sectional Plan, Otto Igniting Valve.



End Elevation, Otto Igniting Valve.

Index.

Part 1

Consisting of five tests in the "Otto" gas engine for the purpose of determining the increase or decrease of efficiency of the engine occasioned by additional compression.

Introduction.

In the last en years gas and petroleum motors have made such great strides that at present for small power they are superseding the steam engine, and in some localities they are even going so far as to compete with high power steam engines. In England gas engines have been constructed which develope twenty-five hundred horse power.

The reasons for this unexpected progress are quite evident when looked at from the view of first cost as well as cost of operation and maintenance.

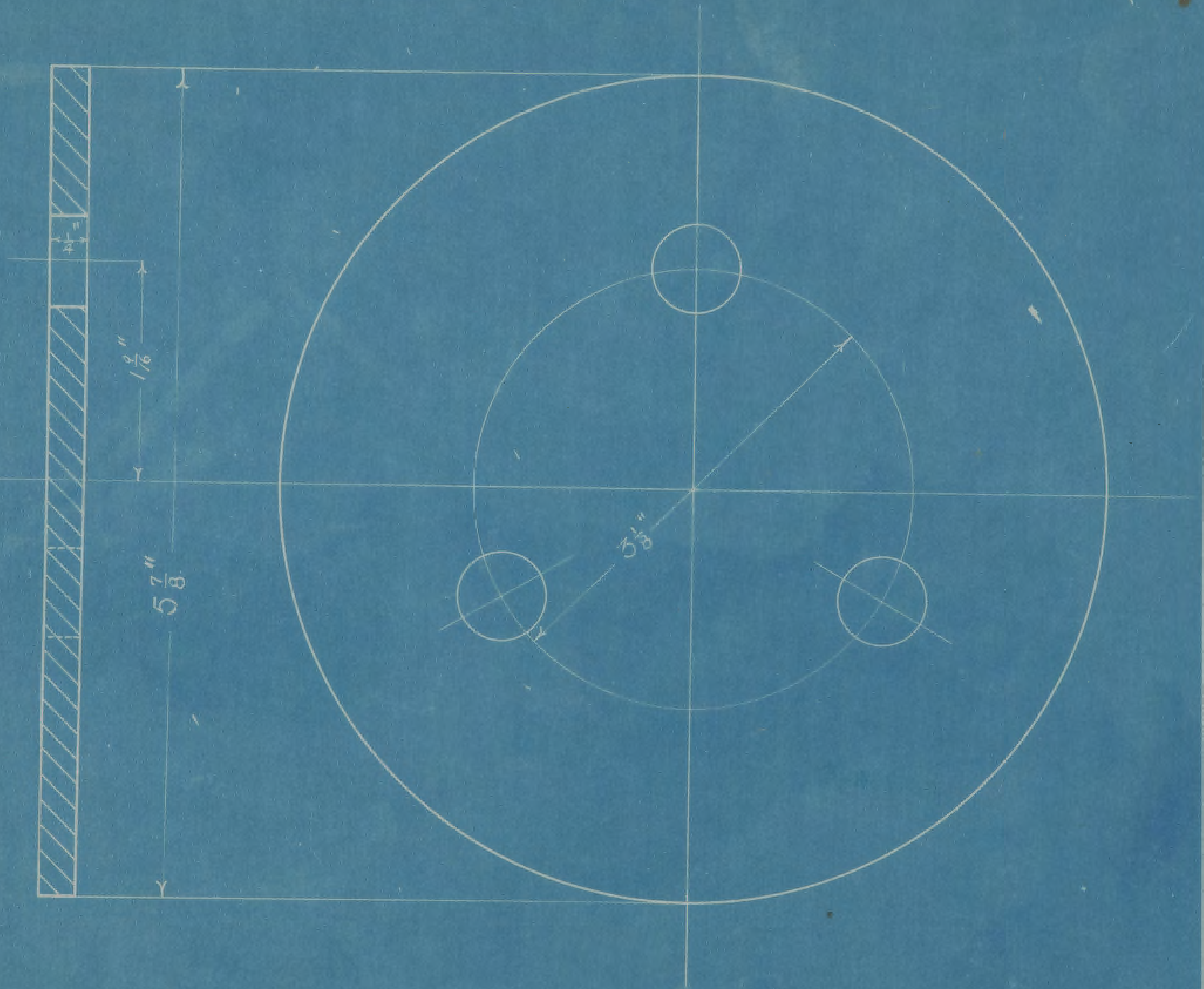
The steam engine although on an average is less costly to operate, requires high-priced labor as well as stokers for boilers, more floor space, accomodations for fuel, boiler cleaning and inspection; all of which necessitate expense.

The gas engine on the other hand is self-contained and once in operation needs little or no attention.

Starting apparatus once so meager and hard to operate, are now on a fair way to perfection, and it is now possible to start most any size motor without difficulty.

The only obstacle now to be removed by the experimentor and inventor is to make gas and petroleum engines more economical than the steam engine when the cost of operation is taken into account.

DISK NO. 1.



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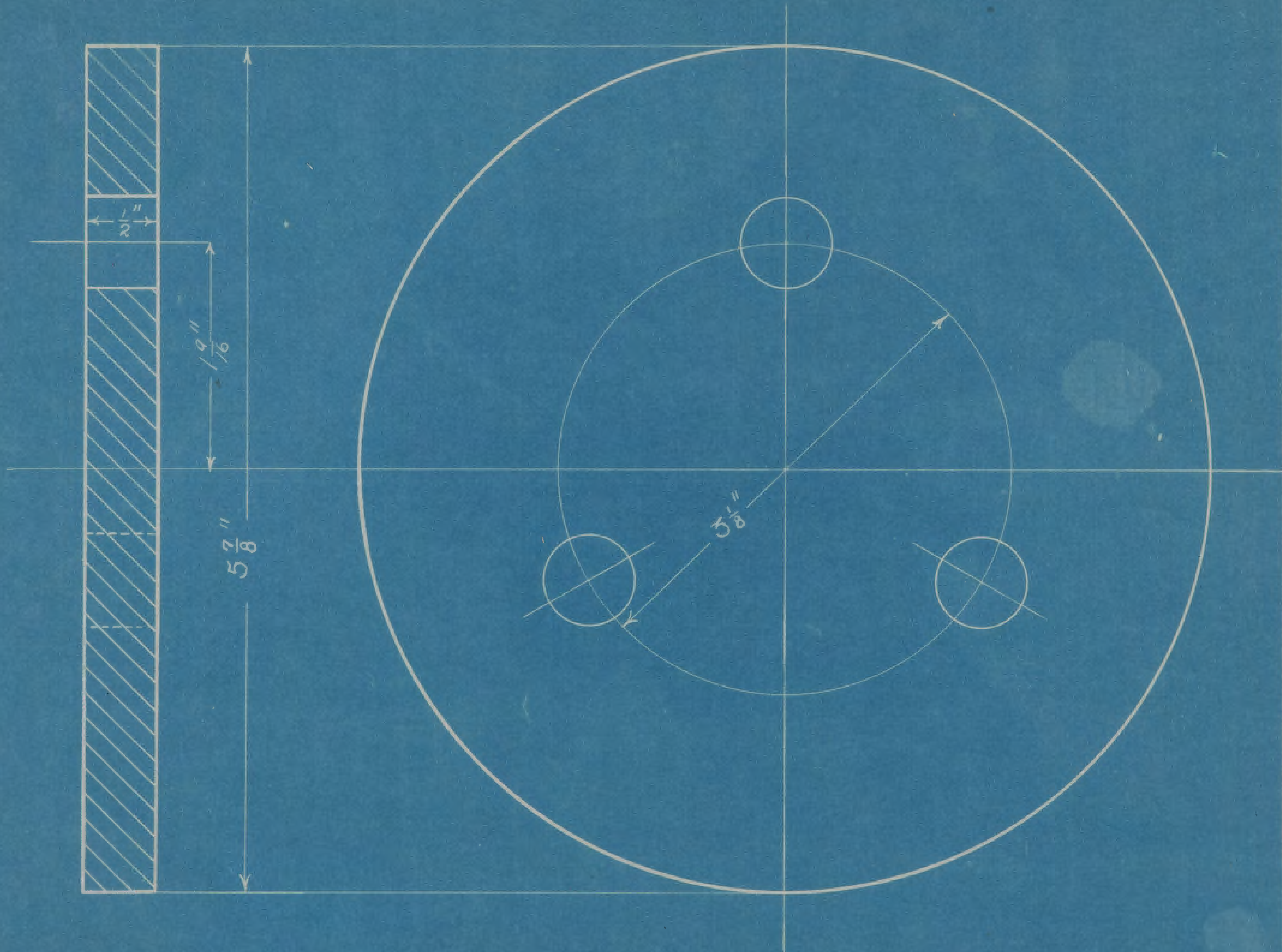

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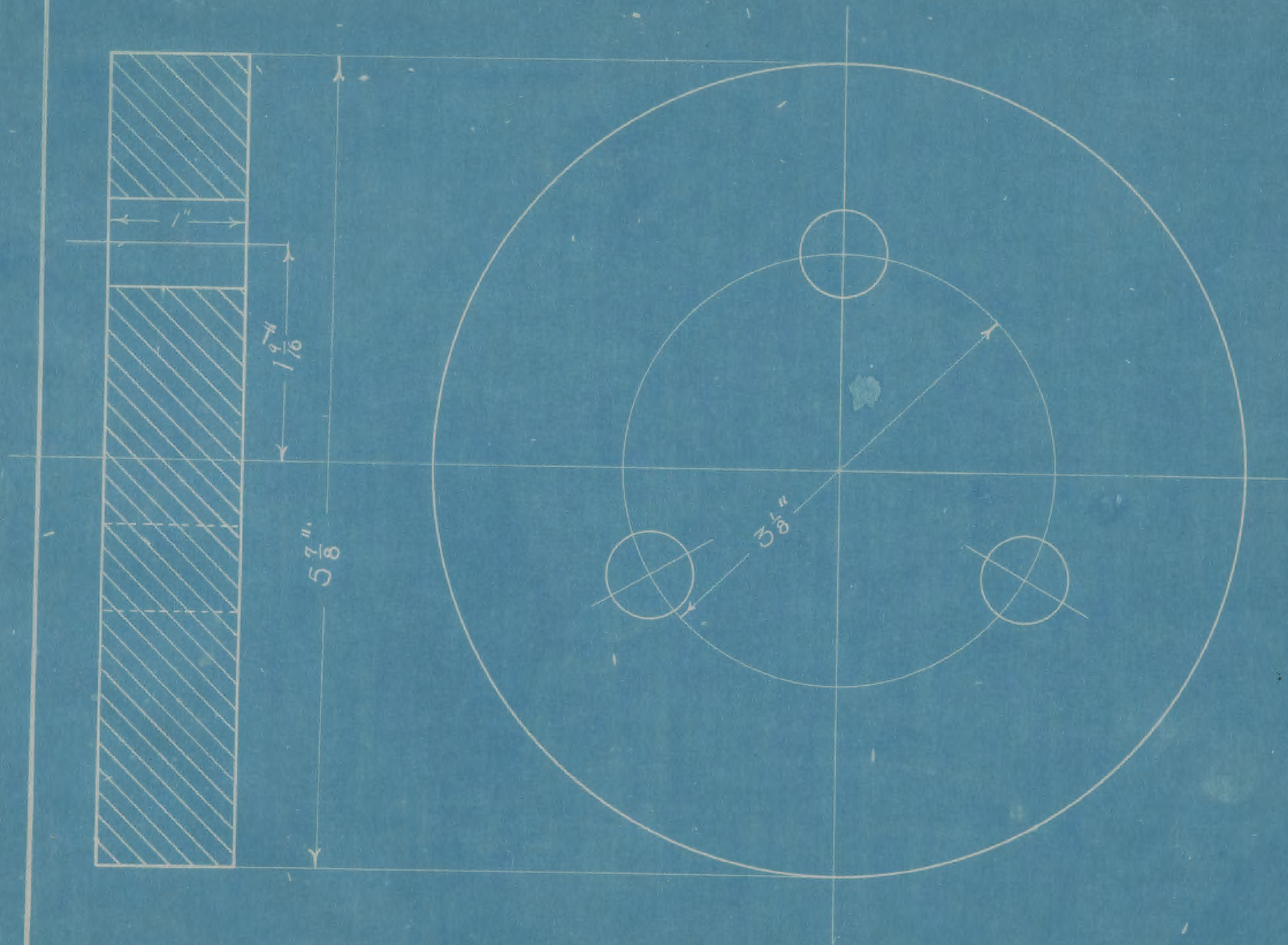

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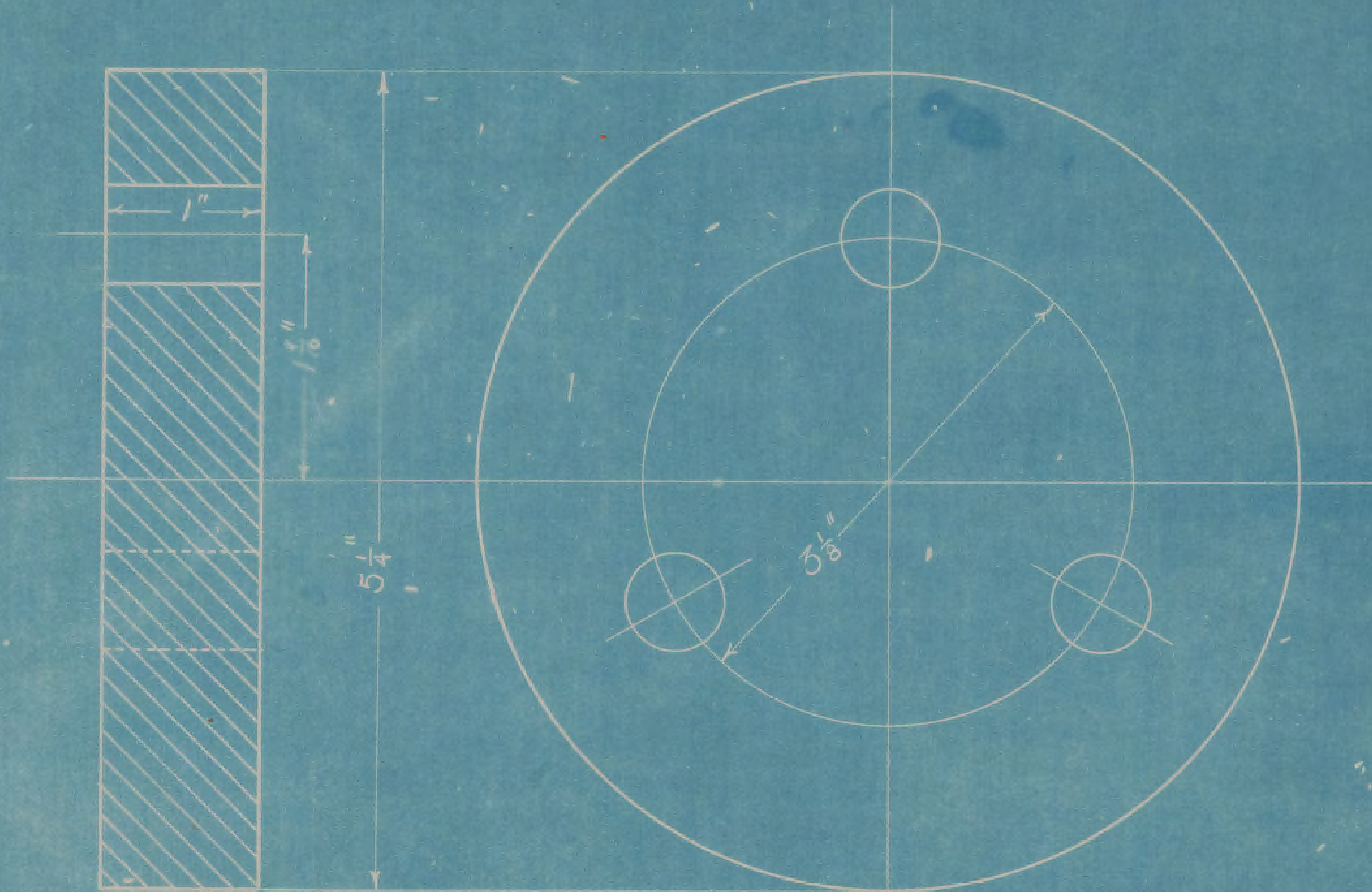

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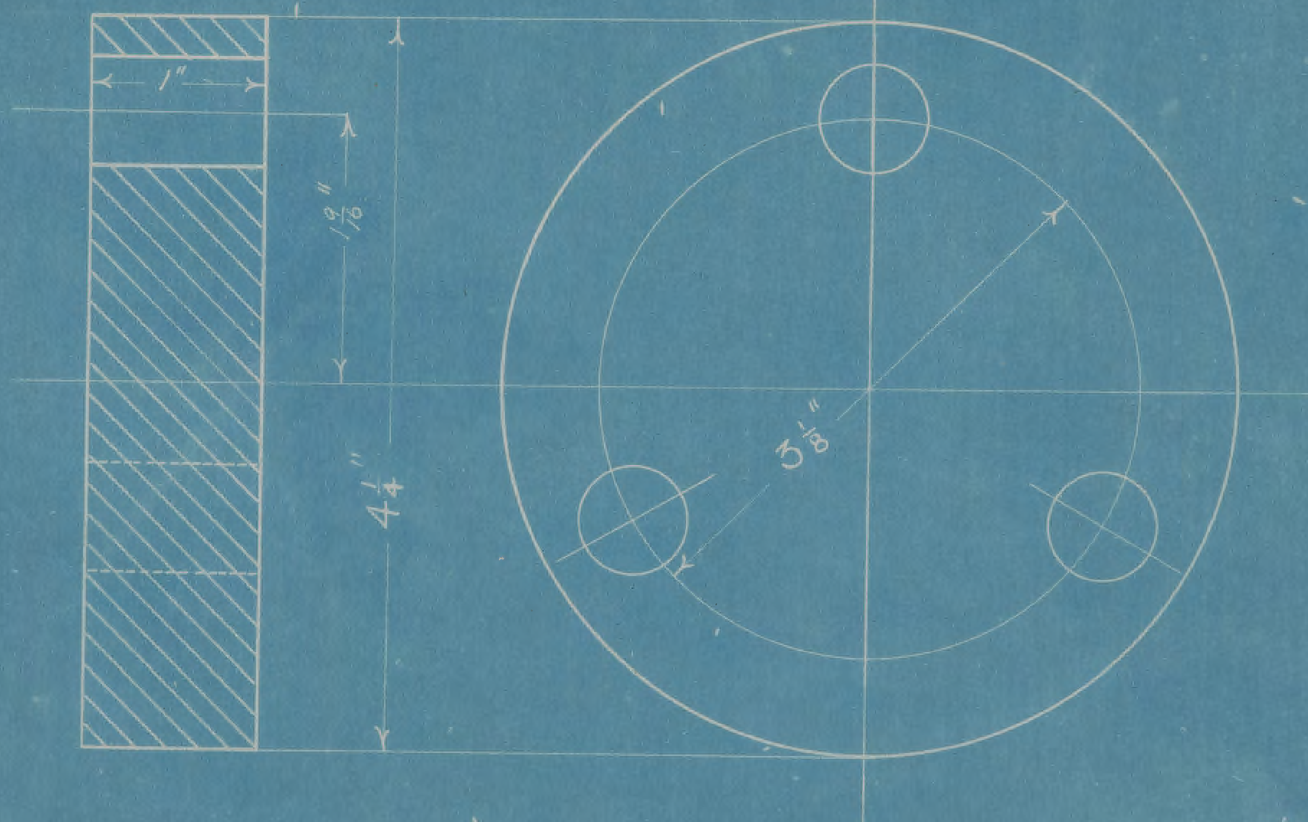


Object.

Numerous experiments have been made with engines using high initial pressure of air and gas before ignition, which show that high compressions are more economical of oil and than the low initial compression.

The best known experiments along this line were made by Prof. James E. Denton of Steven's Institute, New York.

The object of this Thesis will be therefore to determine the amount of increase or decrease of efficiency and gas comsumption brought about by each small additional compressien until a maximum is reached.

The increased compressions were obtained by means of additianal disks shown in drawings 1, 2, 3, 4, 5.

These disks were attached to the inner end of the plunger and held in place by three 5/8" bolts.

It was found necessary to make fourth and fifth disks smaller in diameter than the diameter of the cylinder proper because of the fact that the cylinder and the clearance space is reduced in diameter by the addition ofa flange and also by an exhaust port.

Apparatus.

In all tests of this kind where differential changes occur as in the addition of thin disks producing slight changes in the initial pressure it is necessary that all of the apparatus used should be of the best possible design.

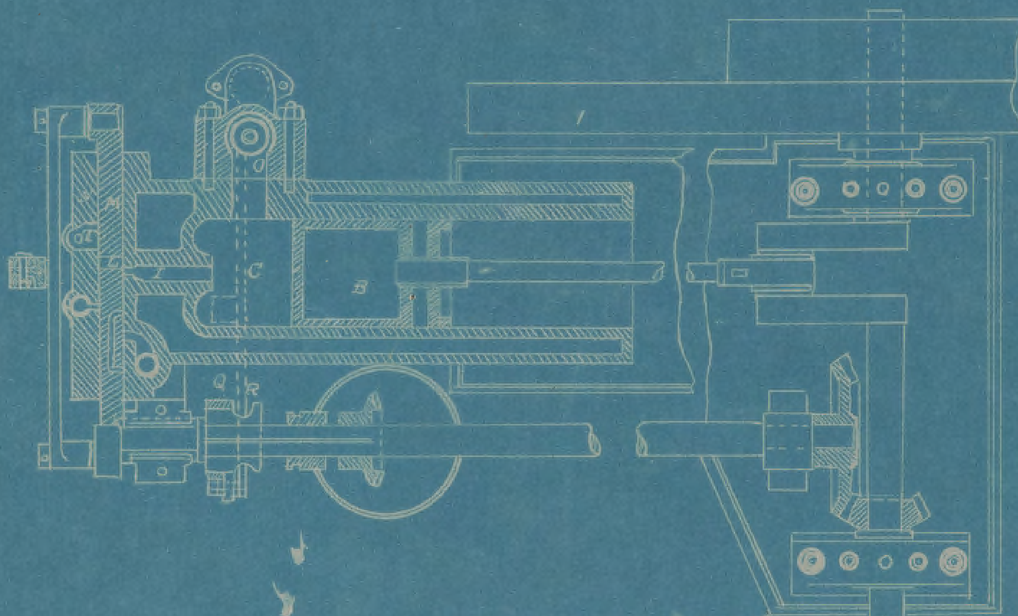
For obtaining the indicated work in the cylinder a Bachelder Indicator was used because of its adaptability for this kind of work where high temperatures are to be encountered.By suitable arrangement the spring is made in the form of a cantilever which projects out to one side of the Indicator pressure cylinder and has an adjustable fulcrum which may be changed in a minute by unscrewing a thumb-screw; thus giving any strength of spring necessary. This arrangement has many advantages over the spiral form of spring in that it may be changed at will with little or no trouble as well as being removed from the heat caused by the hot cylinder gases.

The number of revolutions which the engine made was registered by an ordinary floor stroke counter which was operated by means of a string attached to a small crank pin set in the end of the shaft.

The number of explosions was recorded by means of a small Veeder stroke counter which was fastened to the end of the cylinder near the igniting apparatus in such a way that when the pendulum governor moved ove r and opened the valve of gas supply the counter was moved with the valve plunger; the connection being made by a short piece of copper wire.

Readings of temperature were made on Fahrenheit thermometors. Four different readings being taken: the temperature of the engine room, of the exhaust gases, and the temperatures of the injected and discharged cooling water. The thermometors were placed in oil cups on the exhaust muffler or drum and also on the cylinder.

For obtaining the brake horse power a prony brake was used with an arm sixty-three and oneeighth inches in length. The brake wheel being furnished with a deep flange so that the wheel might be water cooled.



Otto Engine (Sectional plan).

A.Fairbank's scale reading to ounces was used to measure the load at the end of the brake-arm, the dead load being adjusted so as to be an even number of pounds; five pounds in number.

The City pressure was measured by means of a monometer which was connected directly to the gas supply pipe by means of a rubber hose. The reading gauge was graduated to read in inches of water and in ounces direct.

The amount of gas used by the engine for a run was registered by a forty-light gas meter furnished by the City Gas Co.

The disks used in the test are shown in drawings 1, 2, 3, 4, and five. They were made of cast iron cast in the University of Minnesota foundry. The machine work was also done in the University shops. The only real difficulty encountered being to drill the holes for the 5/8" bolts so that ~~they~~ all of the disks might be inter-changeable and used as a unit.

The cooling water was kept, as the tables show, at as high a temperature as possible so that the efficiency might also be high. The temperatures of the discharged cooling water at times reached as high a point as 205 degr Fahrenheit.

As the thermo-dynamic efficiency is increased with an increase of temperature it is advisable to maintain the temperature as high as possible, on the other hand if the increase of temperature be too great the internal friction of the engine becomes excessive: it is necessary therefore to obtain the happy medium in order to derive the best results.

Gas Engine in General.

The "Otto" gas engine on which the tests were made is one used at University of Minnesota in the mechanical laboratory for experimental purposes. The cylinder dimensions are 5-7/8 x 12-5/8. The engine is governed to run at 180 revolutions per minute on maximum load and at other loads the variation was found to be not more than ten per cent which is very good considering the date the engine was constructed.

The "Otto" Company claim that their product would develope four actual horse power, bu t practical tests show that teir claim was much over-estimated, or else the quality of gas furnished is not as good as that which on which they based their estimate.

The governor is of novel construction and has long since become obsolete, but it works very well. It is operated on the pendulum plan. When the engine runs too slow the weight at the bottom of the oendulum has a tendency to retard the motion of the pendulum and therefore the arm at right angles to the pendulum is lowered and fails tp open the gas supply valve: on the other hand if the engine is running under spped the pendulum has a tendency to hang in a vertical position at all times and when in this position the arm at right angles to the pendulum which is of the same casting is in the same plane as the valve operating the admission of gas and therefore gas is supplied more frequently until the speed again becomes normal.

Heat in a Cubic Ft. of Gas.

The number of British thermal units in one cubic foot of coal gas as furnished the mechanical laboratory by the City Gas Co. was calculated from an analysis made by Prof. George B. Frankforter professor of Chemistry in the University of Minnesota. as follows:

CO_2 =	5.6%
O =	1.7 %
C_2H_4 =	6 %
Heavy Carbon =	9 %
CO =	20.4 %
H =	29.2 %
CH_4 =	18 %
N =	16.1 %
	100 %

Now considering the heavy carbon to be (1/3 C_2H_4) and (2/3) of it to be (C_2H_2) we find that the heat given off in burning one cubic foot of gas to be 503.6 British Thermal Units, as computed from a table of Heat of Combustion for the different gases found in Robinson's book on Gas and Petroleum Engines, page 355.

C_2H_2 = 6% of 868.02 = 52.08
C_2H_4 = 3% of 932.05 = 27.96
CO = 20.4 % of 190.02 = 38.76
H = 29.2% of 191.13 = 55.80
CH_4 = 18% of 584.38 = 105.18
Total in centigrade units = 279.78

" " B. T. U. = 279.78 x 9/5 = 503.6 B. T. U. in one cubic foot of gas.

Result of Test.

In order to make the tables more clear it might be well to explain how some of the results were obtained.

The clearance of the gas engine was determined by the water method, that is by pouring water into the clearance space and noting the number of pounds of water which it holds the result of this method being verified by moving the piston forward after the clearance space is filled, until the same number of pounds have again been poured into the cylinder and clearance space combined, and then the plunger travels is noted. The volume of the clearance space and per cent may be calculated from these results.

The initial compression was measured directly from the Indicator cards.

The number of cubic feet of gas per indicated horse power per hour was derived by dividing the total gas consumption for two hours by two and then dividing the result by the number of indicated horse power developed during that hour.

The same may apply to the number of cubic feet of gas per brake horse power per hour except ~~at~~ that the quotient is divided by the actual horse power developed instead of the indicated horse power.

The heat supplied to the gas engine per hour is found by multiplying the number of cubic feet of gas used per hour by the number of B. T. U. in one cubic foot of the gas.

The heat supplied to the jacket water was found by subtracting the temperature of the cooling water injected from the temperature of the cooling water discharged and multiplying the result by the number of pounds of cooling water used in that hour.

The thermal efficiency is found by dividing the amount of indicated heat given out in actual work, by the total amount of heat which has been supplied to the engine. Both expressed in B. T. U.

Results.

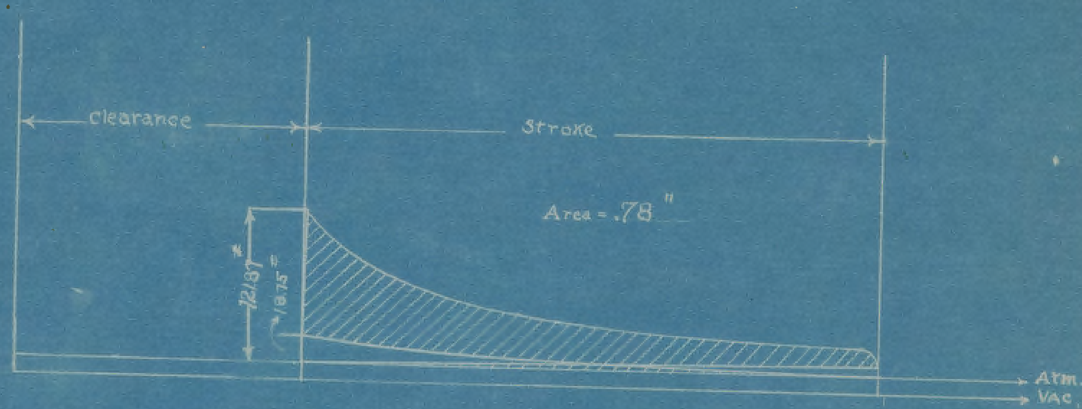
The results of this experiment indicate that the efficiency has been increased by decreasing the clearance volume and hence increasing the initial compression. By b observing the Table 1 it may be seen that in the first run where the initial compression was only 18-3/4 lbs. the thermal efficiency was relatively low; being 12 per cent.

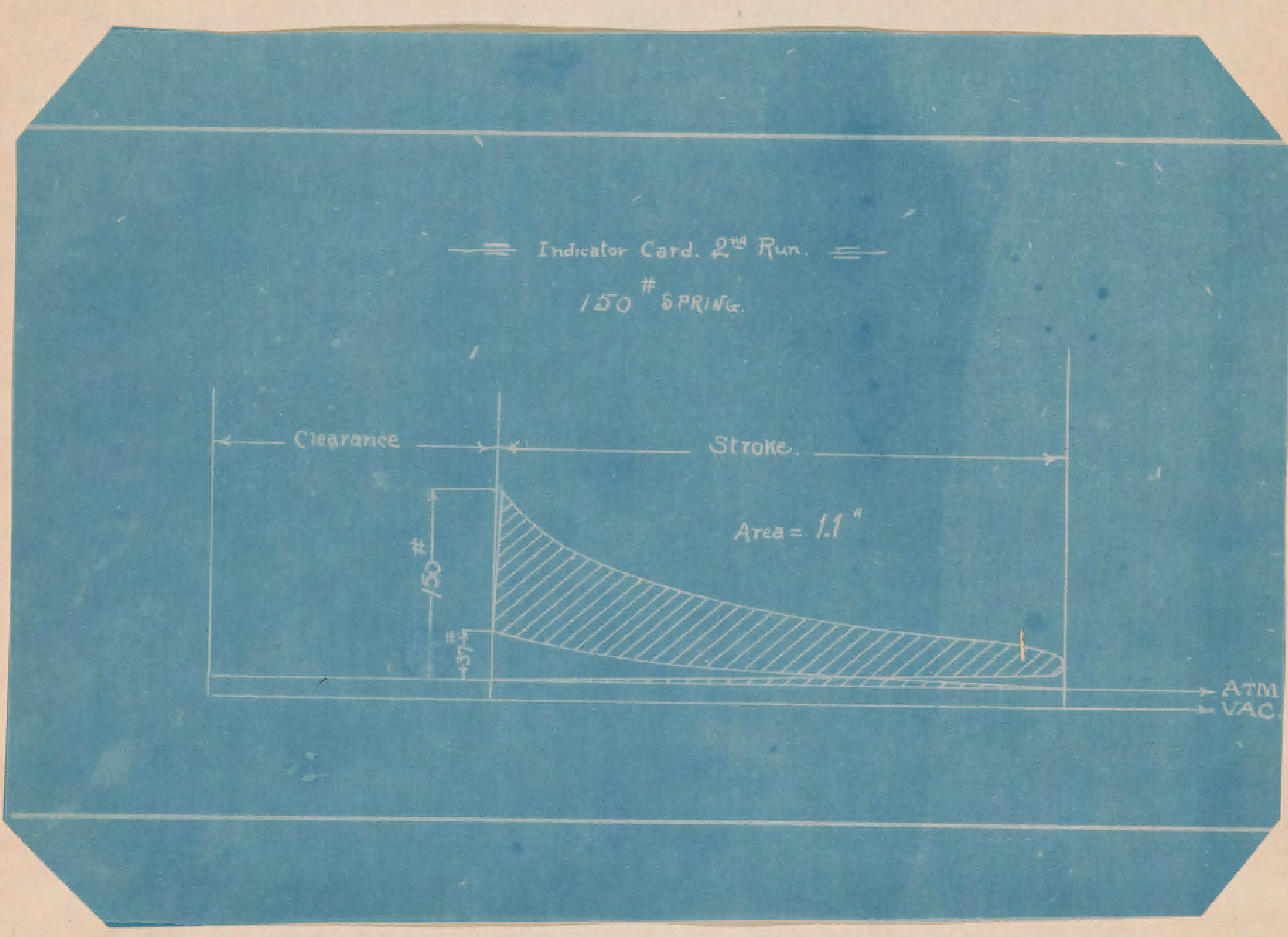
The second run was made with an initial compression of 37 lbs. and the thermal efficiency was found to be fifty per cent greater than the preceding one. The third run used an initial of 37-1/2 lbs. and the thermal efficiency is still on the increase, being 18-1/2 per cent.

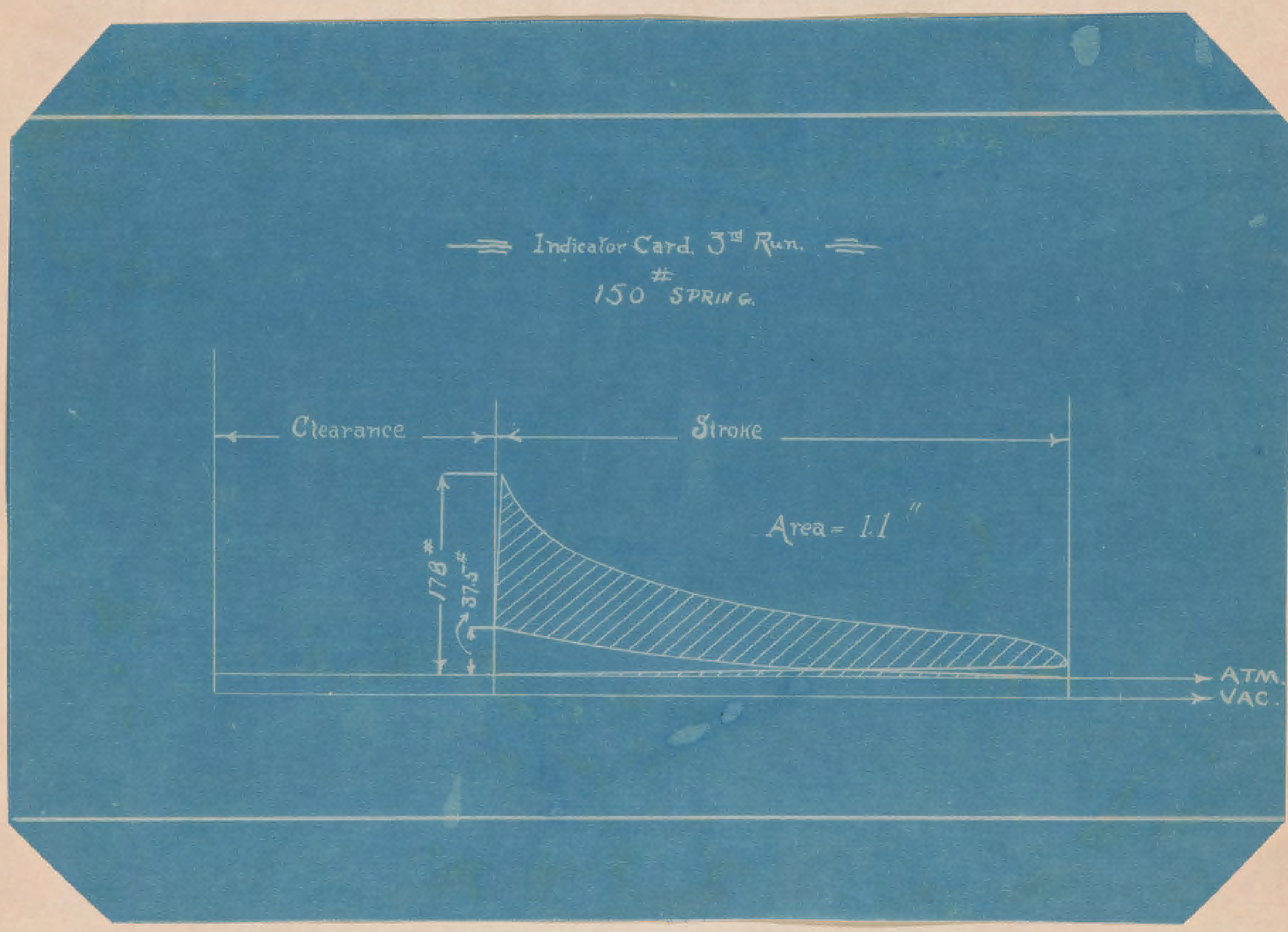
The fourth run was made with an initial compression of 38 lbs. giving a thermal efficiency of 18.8 per cent. While the last run used an initial compression of 38-1/2 lbs. and the thermal efficiency took a drop to 18.1 per cent probably because of the insufficient air supply preceding the explosion.

Indicator Card. 1st Run.

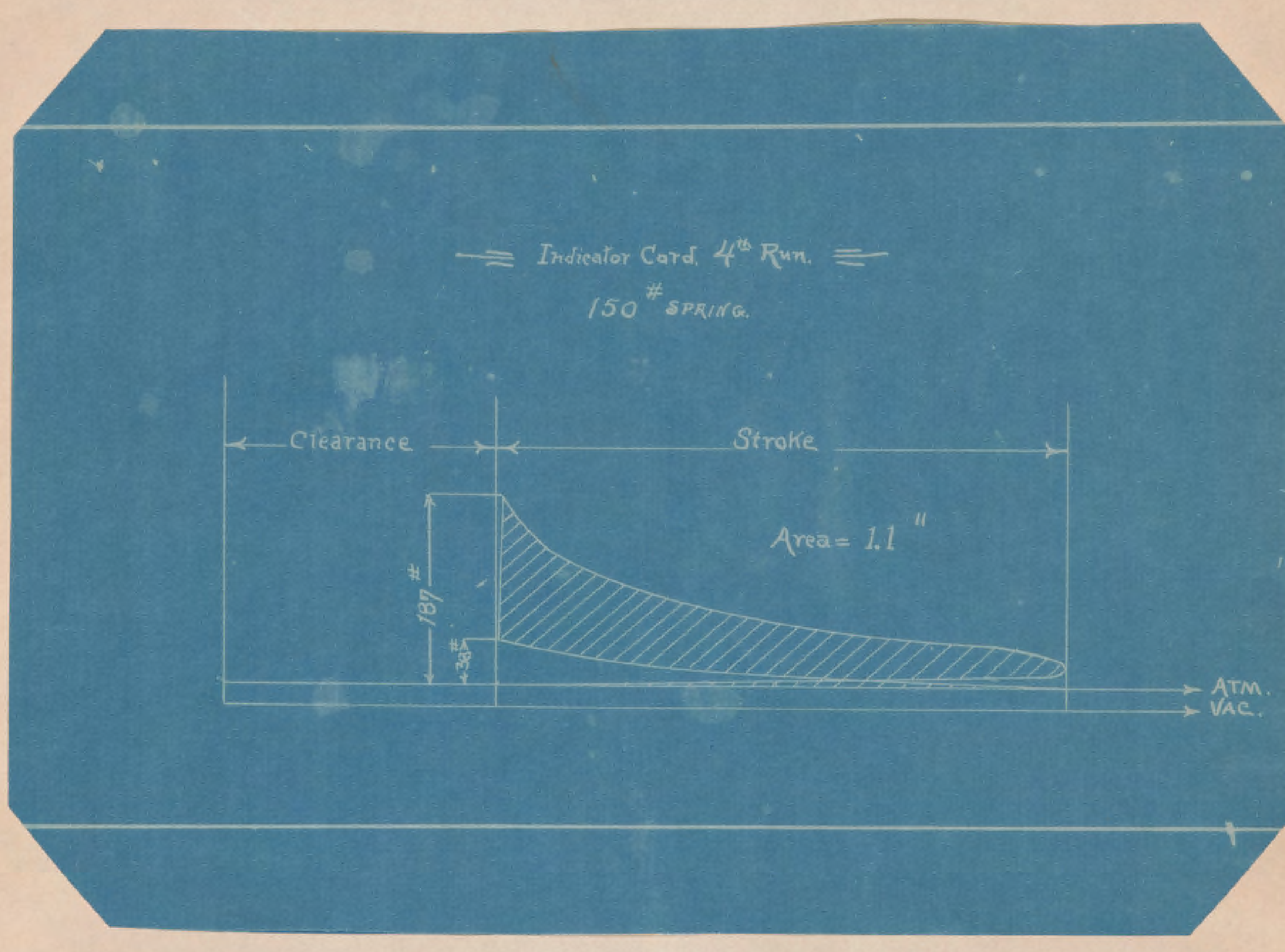
150# SPRING.



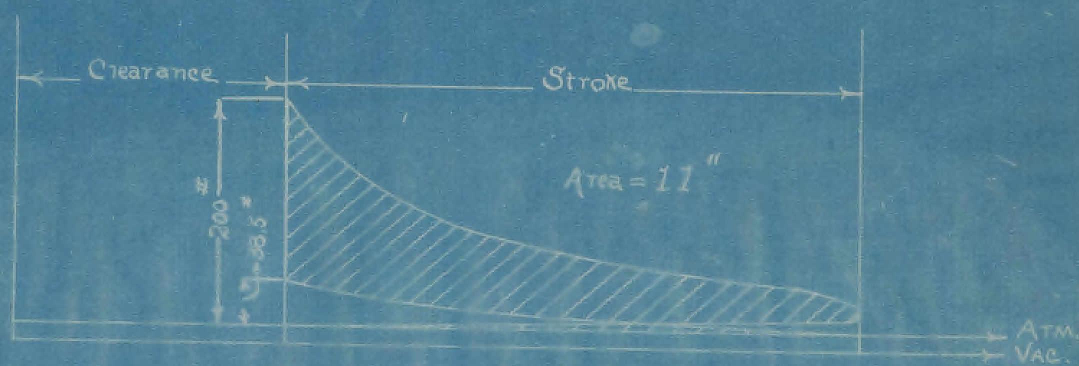

Indicator Card. 2nd Run.
150# SPRING.
Clearance
Stroke.
Area = 1.1"
150#
37#
ATM
VAC.



Indicator Card. 3rd Run.
150# SPRING.
Clearance
Stroke
Area = 1.1"
178#
37.5#
ATM.
VAC.


Indicator Card. 4th Run.
150 # SPRING.
Clearance
Stroke
Area = 1.1 "
187 #
38 #
ATM.
VAC.

Indicator Card, 5th Run.



Duration trial, hrs	2.	Time for 100 Revs. Sec.	34.6
— Dimensions of Engine —		Max. Pr. per Sq. in from card.	121.87#
Diameter of piston In.	$5\frac{7}{8}$	— Results —	
Area of piston Sq. in.	27.15	Indicated M. E. P. M. E. P.	$33.\frac{1}{4}$#
Length of stroke. In.	$12\frac{5}{8}$	Indicated H. P. I. H. P.	2.35
Piston displacement Cu. ft.	2.	— Dynamometer work —	
Clearance. Cu. ft.	1.153	Lbs on scales	10.
Clearance. Per ct.	$33\frac{1}{3}$%	Dynamometer D. H. P. D. H. P.	1.71
Compression Lbs	18.75#	Mechanical Efficiency D.H.P. ÷ I.H.P.	73%
Revs. per Minute.	173.5	Friction Horse Power I.H.P.–D.H.P.	.64
Explosions per Minute.	83.2	— Heat per Hour —	
— Temperatures —		Supplied B.T.U.	52,878.
Room F°	85°	Absorbed by jacket water B.T.U.	22,250.
Exhaust F°	374°	— Efficiency —	
— Jacket-Water. —		Thermal	12%
Inj. F°	64°		
Disch. F°	173.3°	— Remark —	
Wt. Pounds.	372.	1st Run with no compression as for starting.	
— Gas —			
Cubic Feet Gas	210.		
Pressure Water-In.	$2\frac{3}{4}$"		
Cu. ft. gas per I. H. P.	44.75		
" " " D. H. P.	61.		
Thermal equiv., cu. ft. gas B.T.U.	503.6		

Duration trial. hrs.		2.
Dimensions of Engine.		
Diameter of piston	In.	$5\frac{7}{8}$
Area of piston.	Sq. in.	27.15
Length of stroke.	In.	$12\frac{5}{8}$
Piston displacement	Cu. ft.	2.
Clearance	Cu. ft.	1.153
Clearance	Per. ct.	$33\frac{1}{3}$%
Compression	Lbs	37.#
Revs. per Minute.		175.
Explosions per Minute.		78.6
Temperatures.		
Room. F°		83.3°
Exhaust, F°		490.2°
Jacket-Water.		
Inj. F°		56.44°
Disch. F°		191.°00.
Wt. Pounds.		367.#
Gas.		
Cubic Feet Gas.		190.
Pressure	Water. In.	$2\frac{3}{4}$"
Cu. ft. gas per I. H. P.		28.8
" " " D. H. P.		36.75
Thermal equiv. cu. ft. gas.	B. T. U.	503.6

Time for 100 Revs. Sec.		34.3
Max. Pr. per Sq. in. from card		150#
Results.		
Indicated. M. E. P.	M. E. P.	49.5#
Indicated. H. P.	I. H. P.	3.3
Dynamometer work		
Lbs on scale		15#
Dynamometer	D. H. P.	2.58
Mechanical Efficiency	D. H. P. ÷ I. H. P.	79.%
Friction Horse-Power	I. H. P. – D. H. P.	.72
Heat per Hour.		
Supplied.	B. T. U.	47900
Absorbed by jacket water	B. T. U.	24,750
Efficiency		
Thermal		18%

— Remark. —

2nd Run with only ordinary compression.

Duration trial, hrs.		2.
Dimensions of Engine		
Diameter of piston	In.	$5\frac{7}{8}$"
Area of piston	Sq. in.	27.15
Length of stroke	In.	$12\frac{5}{8}$
Piston displacement	Cu. ft.	2.
Clearance	Cu. ft.	1.101
Clearance	Perct.	$32\frac{1}{4}$%
Compression	Lbs	37.5#
Revs. per Minute		168.6
Explosions per Minute		83.36
Temperatures		
Room F°		73.5°
Exhaust F°		450°
Jacket-Water		
Inj. F°		64.5°
Disch. F°		178.4°
Wt. Pounds		420#
Gas		
Cubic Feet Gas		197.5
Pressure	Water-In.	$2\frac{3}{4}$"
Cu. ft. gas per I.H.P.		28.25
" " " D.H.P.		33.75
Thermal equiv., cu. ft. gas	B.T.U.	503.6

Time for 100 Revs. Sec.		35.6
Max. Pr. per Sq. In. from card		178.125#
Results		
Indicated M. E. P.	M. E. P.	49.5#
Indicated H. P.	I. H. P.	3.5
Dynamometer work		
lbs on scales		17.5#
Dynamometer D. H. P.	D. H. P.	2.925
Mechanical Efficiency	D.H.P. ÷ I.H.P.	84%
Friction Horse Power	I.H.P. – D.H.P.	.575
Heat per Hour		
Supplied	B.T.U.	49,750
Absorbed by jacket water	B.T.U.	24,000
Efficiency		
Thermal		18.5%

– Remark. –

3rd Run with Additional compression, caused by the Addition of the $\frac{1}{4}$" disk.

Duration trial, hrs		2.
— Dimensions of Engine —		
Diameter of piston	In.	$5\frac{7}{8}$"
Area of piston	Sq. in.	27.15
Length of stroke	In.	$12\frac{5}{8}$
Piston displacement	Cu. ft.	2.
Clearance	Cu. ft.	1.06
Clearance	Perct.	$31\frac{1}{4}$%
Compression	Lbs	38.#
Revs. per Minute		172.4
Explosions per Minute		84.075
— Temperatures —		
Room F°		73.16°
Exhaust F°		460°
— Jacket-Water —		
Inj. F°		63.08°
Disch. F°		182.84°
Wt. Pounds		430.#
— Gas —		
Cubic Feet Gas		194.5
Pressure	Water In.	$2\frac{3}{4}$"
Cu. ft. gas per I. H. P.		27.4
" " " D. H. P.		30.
Thermal equiv., cu. ft. gas	B. T. U.	503.6

Time for 100 Revs. Sec.		34.8
Max. Pr. per Sq. in from card		187.#
— Results —		
Indicated, M. E. P.	M. E. P.	49.75#
Indicated, H. P.	I. H. P.	3.55
— Dynamometer work —		
lbs on Scale		19.#
Dynamometer D. H. P.	D. H. P.	3.24
Mechanical efficiency	D. H. P. ÷ I. H. P.	91.5%
Friction Horse Power	I. H. P. – D. H. P.	.31
— Heat per Hour —		
Supplied	B. T. U.	48,975.
Absorbed by jacket water	B. T. U.	25,750
— Efficiency —		
Thermal		18.8%

– Remark. –

4th Run with additional compression, caused by the addition of the $\frac{1}{2}$" disk.

Duration trial, hrs.		2
Dimensions of Engine		
Diameter of piston	In.	$5\frac{7}{8}$"
Area of piston	Sq. in.	27.15
Length of stroke	In.	$12\frac{5}{8}$
Piston displacement	Cu. ft.	2.
Clearance	Cu. ft.	1.01
Clearance	Per. ct.	$30\frac{1}{4}$%
Compression	Lbs.	38.5#
Revs. per Minute		160.
Explosions per Minute		77.55
Temperatures		
Room. F°		83.72°
Exhaust, F°		441°
Jacket-Water		
Inj. F°		60.2°
Disch. F°		180°
Wt. Pounds		433.5
Gas		
Cubic Feet Gas		204.5
Pressure	Water-In.	$2\frac{3}{4}$"
Cu. ft. gas per I. H. P.		27.9
" " " D. H. P.		32.4
Thermal equiv. cu. ft. gas	B. T. U.	503.6

Time for 100 Revs. Sec.		37.5
Max. Pr. per Sq. in. from card		200#
Results		
Indicated M. E. P.	M. E. P.	54.9#
Indicated H. P.	I. H. P.	3.6
Dynamometer work		
Lbs on scales		19.5#
Dynamometer D. H. P.	D. H. P.	3.15
Mechanical Efficiency	D. H. P. ÷ I. H. P.	87.5%
Friction Horse Power	I. H. P. − D. H. P.	.45
Heat per Hour		
Supplied	B. T. U.	51,493.
Absorbed by jacket water	B. T. U.	25,900
Efficiency		
Thermal		18.1%

— Remark —

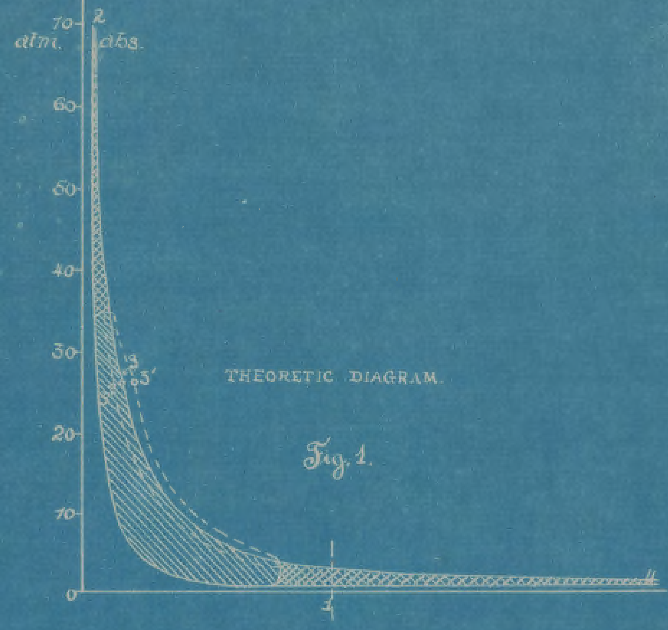
5th Run with Additional compression, caused by the addition of the $\frac{1}{4}$" & $\frac{1}{2}$" Disks.

Part 2.

Including a Description and test

of

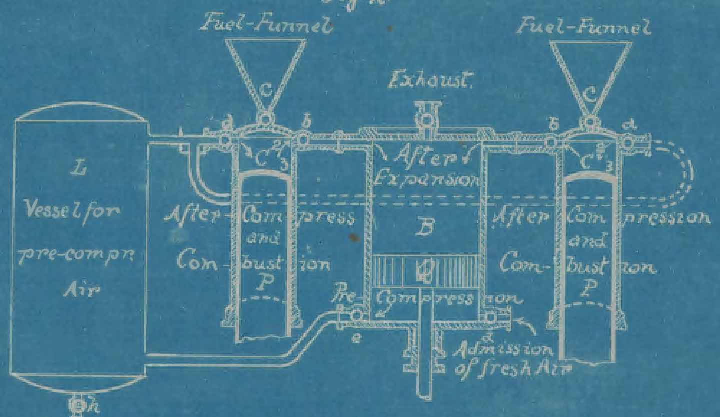
Diesel's Rational Heat Motor.



THEORETIC DIAGRAM.

Fig. 1.

Fig. 2.



SCHEMATIC REPRESENTATION OF THE DIESEL MOTOR.

As many tests have been made by other experimentors along the line of determining the increase of efficiency by increased compression. The writer takes the liberty to use the data derived by one of the most prominent experimentors as well as to describe somewhat in detail the engine used by him.

Figure 1 represents the theroretical card assumed to have been taken from the Diesel motor. In reference to it pure air is compressed in the cylinder according to the

curve 1 to 2, so that by it alone(before combustion and independent of it)a combustion temperature is affected whereupon the fuel, according to the curves 2 to 3, is so gradually introduced that the combustion is going on isothermally, on account of the motion of the piston, and the expansion of the air is thereby affected without materially effecting the pressure. After the supply of fuel has been shut off the gas in the cylinder will further expand from 3 to 4 as is shown by the curve.

In practice the narrow part (2) of the diagram will not come out so pointed as obtained on the ideal card, but will assume a more rounded form; neither will the

different curves run exactly according to the theory and the curve of combustion may more or less deviate from the isothermal curve without materially changing the nature of the process.

There is a decided advantage in omitting long stretched extremities at (2) and (4), since by doing away with them we succeed in obtaining at (2) far less and consequently more practical heights of compression and at (4) smaller cylinder dimensions.

Therefore it is advisable to cut off these attenuated dimensions, as is shown by the dotted lines and thus obtain the real diagram indicated by the lighter hatched portion, such as will be actuallly obtained from the experimental engine.

From figure (1) the efficiency of the engine may be regulated. The shape of the combustion curve is changed 3' to 3" 3# as shown by dotted lines, by changing the duration of the fuel supply; thereby also changing the total height of the diagram therefore, also its area by introducing the fuel at various points on the compression line as designated by dotted lines.

Figure (2)

Figure 2. shows a motor design for the purpose of realizing in practice that which is set forth by theory. The values therin are but crudely indicated and foundation, connecting rod, fly wheel and etc. are omitted.
There are also two combustion cylinders (C) with plunger (P) details of which are adapted to stand high pressure, especially in regard to piston packing. By means of geared valves small (b) both these cylinders (C) are connected to the two sides of this larger intermediate cylinder (B); the combustion cylinders, communicate by means of the geared valve (a) with the air vessel (L). The cranks of both cylinders have the same position and are diametrically opposite to the intermediate cylinder crank.

When piston (Q) moves up it draws air through the valve (d) which is compressed during the downward motion to several atmospheres, whereupon it is forced through the valve (e) into the air vessel (L). The lower part of the intermediate cylinder only, serves as an air pump and brings about the complete compression of the combustion air. This precompression however is to be effected only so far that the heating of the air thereby caused is kept withing a moderate limit.

During the downward stroke the piston performs the second period of compression to the required height. The extreme lower and upper positions of the piston are indicated by dotted lines.

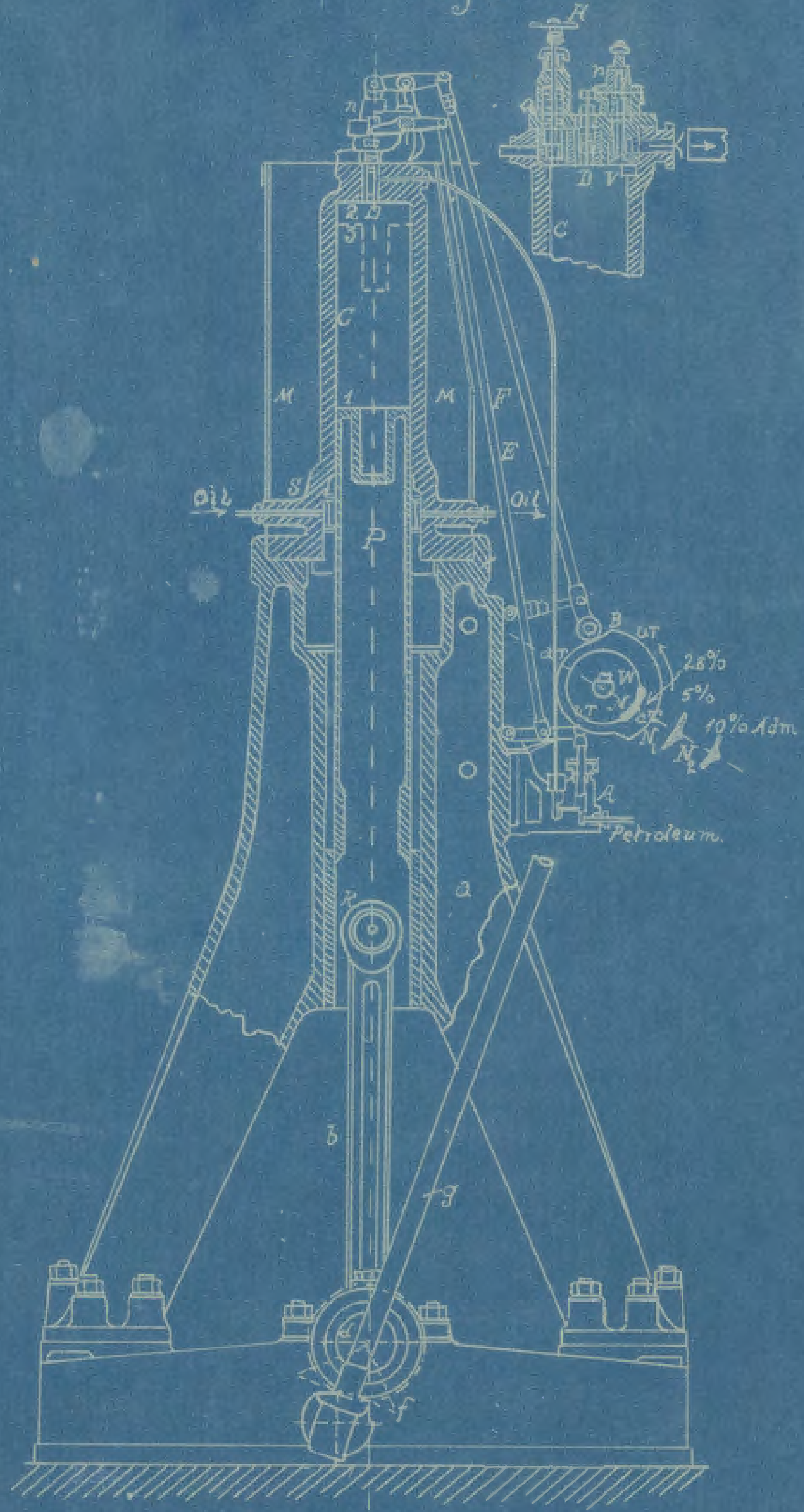
Hereupon the piston moves downward to the position (3) during which period fuel is gradually being introduced and consumed as was previously explained.

In this case the fuel is used in the form of coal dust which is admitted to the cylinder during the prescribed admission period by slowly turning a rotating cock provided with a small lateral groove.

At (3) the fuel supply is shut off and the expanding of the air is continued. The piston having reached the lower position(1), valve (b) is opened; at this moment piston (Q) is in its upper position; by continuing the action (P) and (Q) down, the ga ses of combustion expanding further to the volume of the cylinder (B). Hereupon valve (b) is closed and (f) opens; the combustion gases are then expelled into the open air at the next upward stroke of the piston (Q).

As the cylinders (C) have only one combustion period every other revolution, as by the arrangement of the two cylinders , a combustion for each revolution is effected, that is; one working period is taking place alternating at right and left.

Fig. 3.



RATIONAL HEAT MOTOR

Fig. 4.

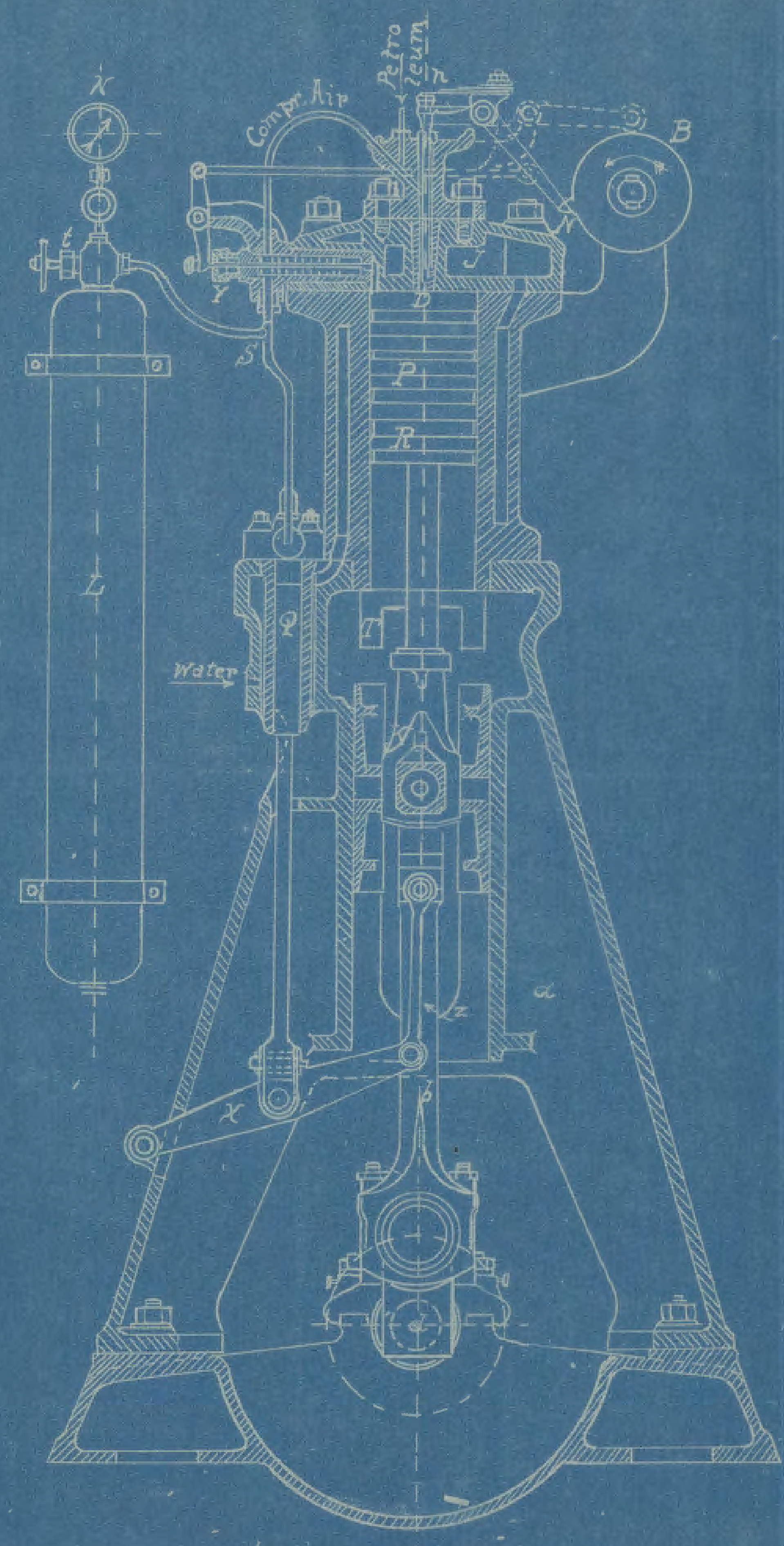
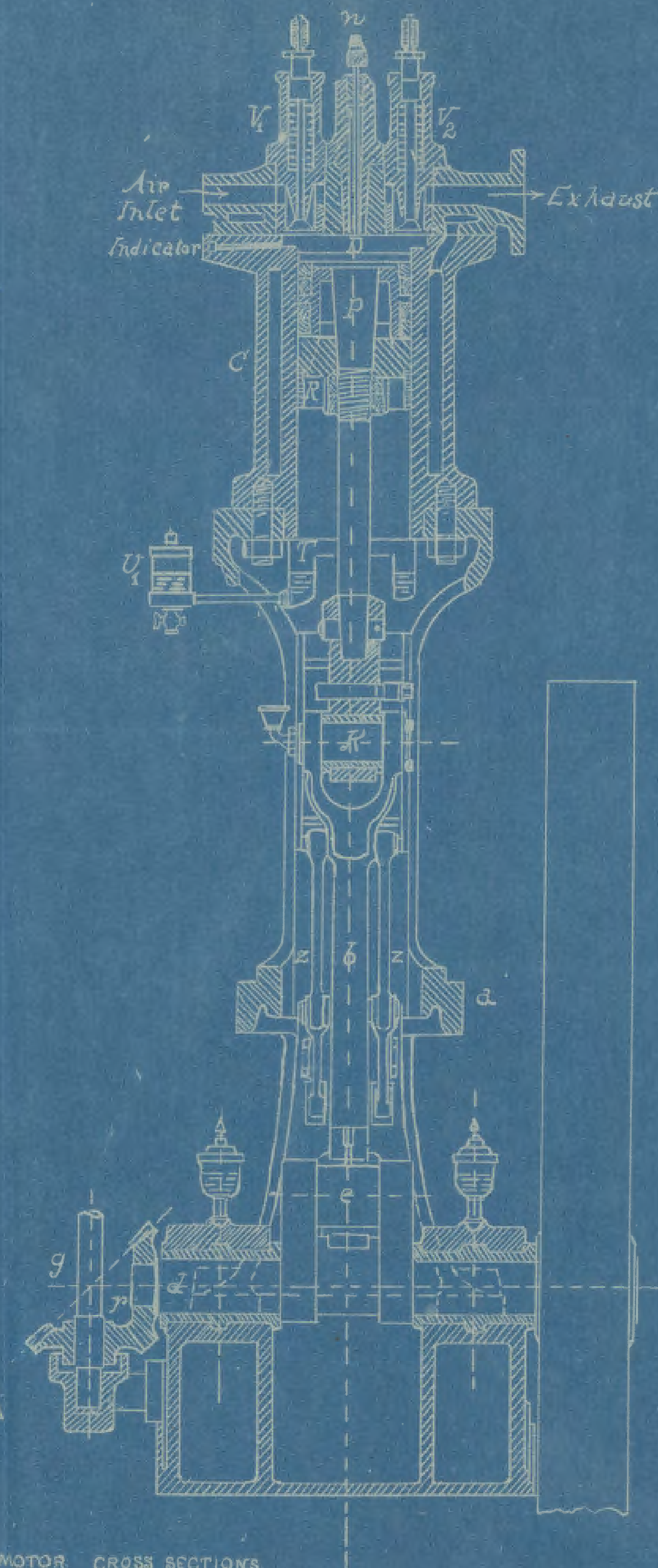


Fig. 5.



RATIONAL HEAT MOTOR CROSS SECTIONS.

By comparing this motor with the steam engine it will be seen at a glance that the motor must be far superior.

First of all, owing to the intermediate combustion of the fuel in the cylinder, no steam boiler is necessary $(n)_{I}$ becoming equal to 1. Furthermore, the ~~th~~ greatest theroretical efficiency $(n)_{II}$ is for the new motor as has been proved equal to from 50 to 70 per cent, that is, an average of more than twice as much as that of the best and largest steam engines.

The indicated efficiency $(n)_{III}$ likewise will become considerably greater as we have not to deal with condensable gases and many other sources of loss has been abolished.

It was to be anticipated however that the mechanical efficiency would result in a low figure, on account of the high pressure that is involved in negative work. Henceforth this was the point that had to stand the energetically attacks of criticism, which endeavored to demonstrate by calculation that this mechanical efficiency would be found such a low one that the fully recognized theroretical advantages might not be realized, and the motor was expected to give lower practical results than any other motor previously designed.

After much theroretical research and practical trial it was decided to build an engine which would represent a commercial value for small and medium sizes. The compound motor which was the inventor's first idea was preceded by a single cylinder engine. It was furthermore determined that the experiments were to be conducted first with liquid, then with gases and afterwards with pulverized fuel. It may be mentioned here that the idea prevailed at the beginning that the generation of gas from coal would be simpler and cheaper than its crushing and sifting, and that the application of coal dust, although it seemed to be very tempting from the first, would offer disadvantages rather than advantages as compared with the use of gas.

Based on this theory the engine which is represented by figures (3)-(4)-(5)-(6)-(7) was built. They show motor with a single acting cylinder (C) and plunger (P), whose details are constructed to stand high pressure. Piston(P) is connected to the fly-wheel axis (d) by means of cross-guide (a) connecting rod (b) and crank in the usual manner By means of bevel gears the fly-wheel shaft turns the inclined shaft (g) which in turn revolves the gear shaft(W). On this shaft two cams are fastened, opening respectively the air valve (V) and the petroleum valve (D) at a

Fig. 6.

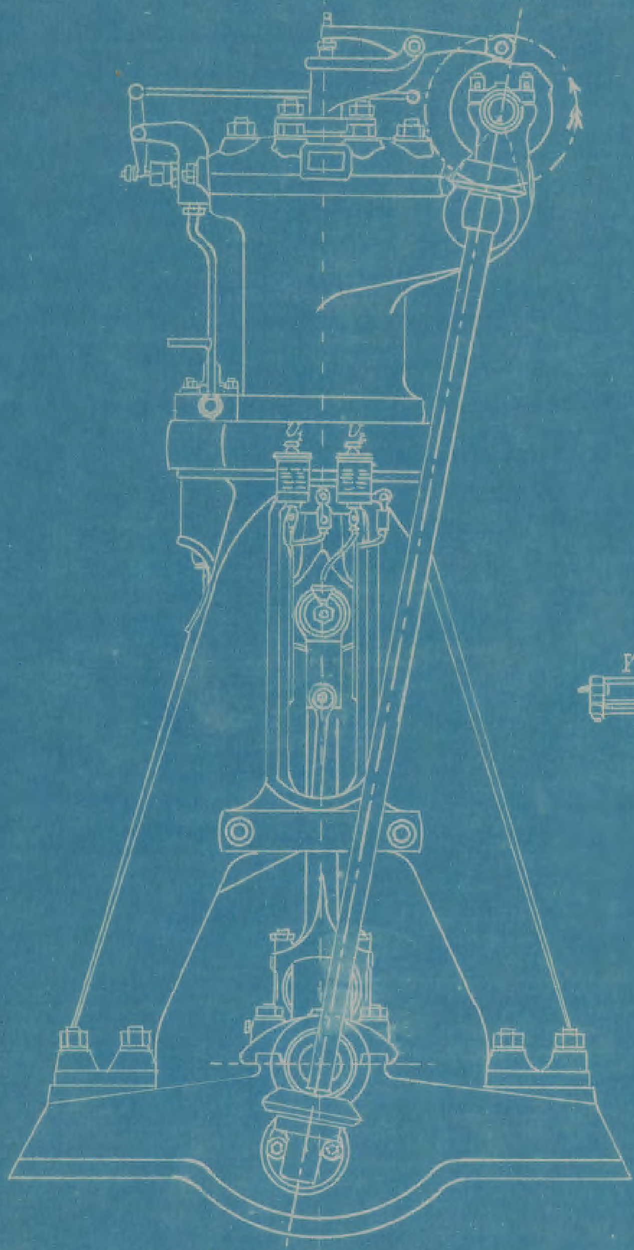
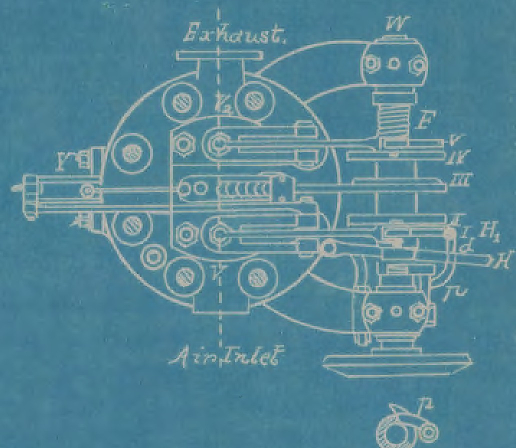


Fig. 7.



RATIONAL HEAT MOTOR. SIDE VIEW AND PLAN OF VALVES.

specified moment. The gearing is fully shown in figure ((3); that for the valve (V) performs its work in a similar manner. Both valves being forced on their seats by means of springs as soon as the cam is released.

The process in the cylinder (C) is carried on in accordance with the four-cycle period, as follows:

First: Downward motion of the piston (P) caused by the visviva of the fly wheel which was received from the preceding work-stroke. Atmospheric air is drawn through the open valve (B) into the cylinder (C); the lowest position of the piston being designated by (s).

Second: Upward motion of the piston (P) still caused by the vis viva of the fly-wheel, valve(V) being closed. The air drawn into the cylinder is compressed to such a high pressure that the temperature at which the combustion following the same is going on is generated by this compression alone. This compression pressure can only be caused by the piston (P), after having, in its end position compressed the air drawn in it, to the required temperature.

Third: The second downward motion of the piston (P), or proper working stroke, follows. The introduction of the fuel takes place during the motion from position (2) to (3) of the position, by the little pump (A) operated by several projecting cams (N N N) regulating the reqired periods of fuel supply.

In position (3) the admission of the fuel is shut off and the air expands until the piston reaches the lower dead center.

Fourth: The second upward motion of the Piston (P) due to the vis viva of the fly-wheel; the gases are expelled through the valve (V) to the outer atmosphere. After this second upward motion the cycle is again repeated.

The motion is started by admitting air through a proper valve from the storage tank, the latter being constantly filled with compressed air while the engine is working. It may be noted here that the engine was run without a cooling jacket, thereby proving the possibility of working without cooling water, as was theroretically anticipated. In later constructions the cooling jacket for practical reasons was added which permits of greater effects to be obtained from the same cylinder dimensions.

From the numerous results obtained based on experiments it may be said that a water jacket is the only obstacle in the way of realizing a greater efficiency for the Diesel motor.

With all caloric motors, the carrying away of the certain amount of heat is necessary from a theroretical standpoint. In the steam engine a special apparatus, the condensor is devised for this purpose. Incombustion motors the condensor is represented on the one hand, by the cooling jacket, and on the other by the atmosphere, which receives the heat of the exp elled gases. Both quantities of heat carried off provide certain losses, gen erally lower than suggested, are not taken into consider- ation but represent those quantities of heat to be taken away in order to perform process in the right way. Theres fore, the water jacket is no irredeemable fault, but a theroretical necessity as is the condensor to the steam engine.

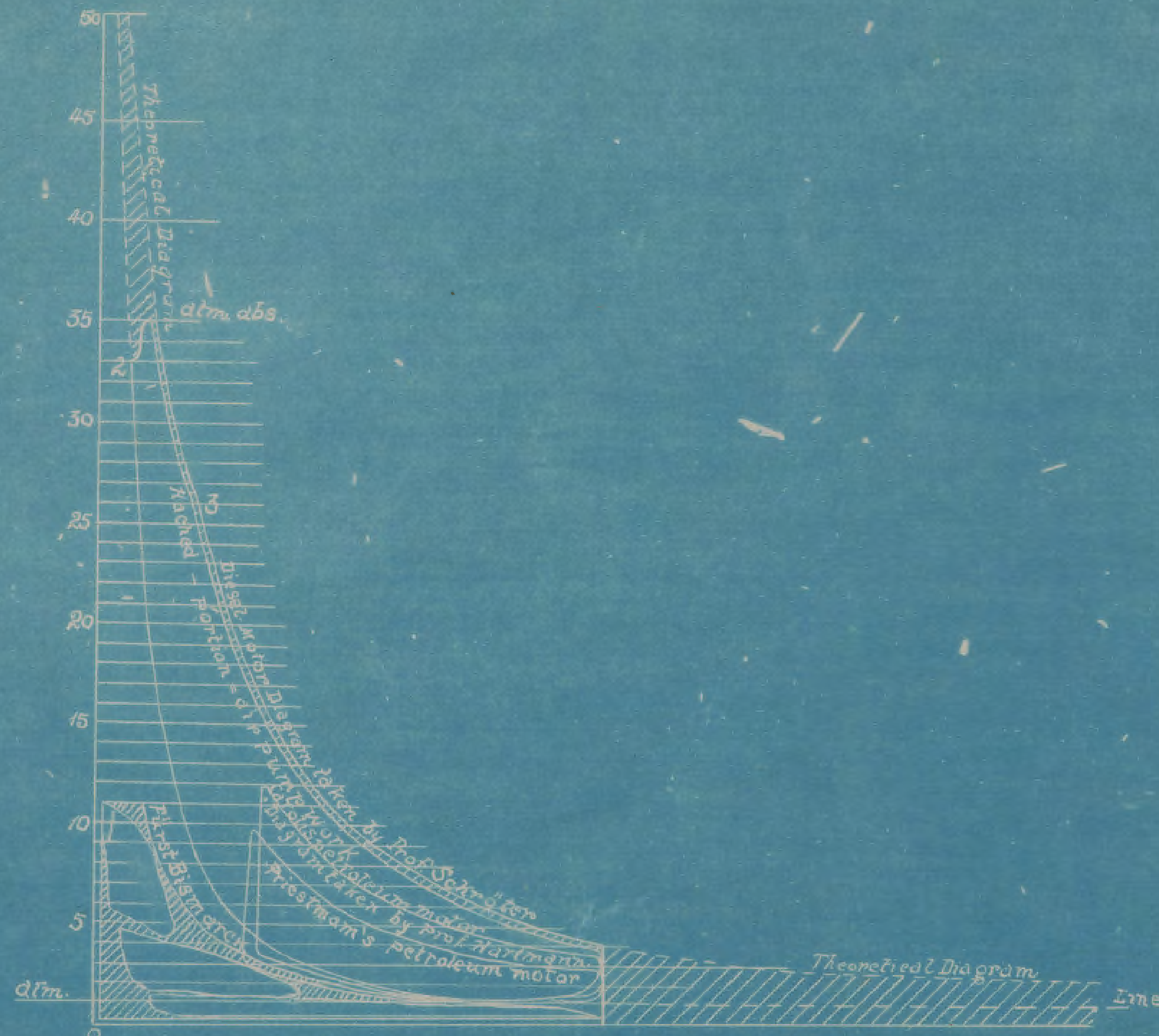
The numerous efforts to save as far as possible the heat that is carried away are incomprehensible, and their failure to do so is evident. This effort resembles the tendency to save heat in steam engines by introducing as little cooling water into the condensor as possible. If one is desirous of saving the heat carried away, there is but one way and that is, to choose such a process of com- bustion at the start that from a merely theroretical standpoint a smaller loss of heat is necessary. It would take too long to prove this opinion, but it is based upon accumulated experimental facts.

As the new process necessitates high compression pressures with simultaneous high temperatures, so many new and high claims were attributed to it that only re- mained of former experiences to depend upon, nearly every single detail had to be adapted to its special purpose by tedious study and continuos building and re-building.

Great difficulties were to be overcome in constructing the numerous valve, piston and gearing apparatus as well as to determine the fuel supply necessary: because the fuel was introduced in small quantities; in an exactly regulated manner, by strong and durable apparatus. Two years were occupied in investigating and testing the

Diesel motor before it was possible to undertake to re- building of a new motor which was to include all former experience and which would really be fit for practical use.

Fig. 8.



COMPARATIVE DIAGRAMS OF A STEAM ENGINE, PETROLEUM MOTOR AND THE RATIONAL HEAT MOTOR FOR EQUAL CYLINDER VOLUMES AND DRAWN TO THE SAME SCALE.

From figures (3) to (7) inclusive may be seen that the Deisel motor for reasons previously explained is provided with a water jacket. The small air pump (Q) driven by connecting rod (Z) and lever (X), keeps the vessel (L) filled with compressed air under a higher pressure than the highest compression attained in the cylinder. By means of the pipe connection (S) the same excess of pressure is connected with the interior of the

injector valve (D). In it the petroleum is collected during the intervals in the four-cycle period betwen the combustion periods. This petroleum is introduced by a little pump which is not shown in the figure. By opening the valve (n) the fuel is by its high pressure caused to flow through the nozzle opening large (D) to the combustion chamber of the engine, thus creating the combustion period in accordance with Diagram (1)

Figure (8) shows in detail the entire gearing and especially the starting apparatus of a motor by the compressed air vessel (L). (W) represents the gear shaft having a number of cams (I to V). Cams (I) operate the valve (V), ~~1-to-3~~ (III) the fuel valve in the nozzle (D) and (V) the exhaust valve (V) of the engine. The whole of the gearing serves to start the engi ne by passing compressed air from (L) through the valve (Y) (figure 4) to the cylinder, pushing the piston ahead and escaping through the main valve (V). During this very short starting period the lever (H) (figure 7) is in the position (H), so

that the valve (Y) is geared by cam (II), valve (V) by cam (IV) instead of (V) while fuel plate (III) and cam (I) of the admission valve are disconnected. After a few revolutions the engine attains its normal spped. At this moment the pin (d) (figure 7) which has kept the lever (H) in position is removed and the latter is automatically, by means of the spring (F) snapped into its normal position (H) and with it the five cams, and thus the normal running is effected without interrupting the working of the engine. The engaging of the cams at the right moment can only take place if a notch, especially provided for that purpose in the cam system passes before the lever (p)

Practical Test.

After having described the Diesel motor in detail it is now fitting to extend the discussion by giving the result of a practical test made on the motor by Prof. Denton.

Tests were made with ordinary domestic kerosene and with various grades of petroleum which are used as fuel under stationary boilers, the operation of the engine covered some two hundred and fifty hours: in observing the behavior of the oil and the condition of the interior of the engine after using each different oil during all this time efficiency tests were made of from one to ten hours duration each as is shown in the Table.

The principal commercial results taken from this Table are as follows: First: With ordinary colorless domestic kerosene of 120° Fahrenheit flash, 150° Fahr. fire test and of.784 sp. gravity, running at 187 R.P. M. and giving an average B. H. P. of 20-8/10 with an oil consumption of .534 lbs. or .82 U. S. Gls. per hour per B. H. P. ; 13.05 B. H. P. with an oil consumption of .58 6 lbs. or .0896 U. S. Gals. per hour per B. H. P. ; and 8.78 B.H. P. with an oil consumption of .72 lbs. or .11 U. S. gals. per hour per B. H. P These results show an efficiency on B. H. P. of 25.8 per cent, 23.5 per cent add 19 per cent respectively. The corresponding indicated horse power efficiencies were 37.7 - 41.8- 41.2 respectively. These results practically confirm the tests made on an engine at a 19 brake horse power at Augsburg, Germany by Prof. Schroeder and other experimentors, which for a 17.33 English horse power showed an efficiency in oil consumption of .54 lbs. or .0189 U. S. gals of American kerosene per ~~houn~~ per B. H. P; and for a 9.315 B. H. P. an oil consumption of .63 lbs. or.0956 U. S. gals. per hour per brake horse power, efficiencies being as follows:

	17.33 H. P.	9.315 H. P.
Thermal Effic. on Ind. H. P.	34.15 %	38.18 %
" " " B. H. P.	25.81 %	22.12 %

The same increase in indicated efficiency with reduction of load to one-half shown by the by the Germmn test is found in the New York test as per line 33 in Table. Tee mechanical efficiency or the per cent of power utilized at the brake of the New York motor is not as great as that of the motor tested in Ger many-a result which is possibly attributable to the insufficiency of the foundation of the New York motor, which by permitting excessiwe vibratinn may have caused a waste of power between the cylinder and the brake. The New York motor showedgreater indicated efficiency however, which is probably due partly to the slighlty increased temperature of the cooling water which reduced the jacket waste of the engine. The heat balance of the two cases is as follows:

Heat Balance.

	New York		Augsburg.	
Heat of combust. accounted	B. H. P.			
for by ind.	21.51--	8.78--	17.33--	9.315
power---	37.2%	41.2%	34.7%	38.9%
removed by jacket--	35.4%	37.4%	40.3%	45.1%
remainder--	27.4%	21.4%	25. %	16 %
Total Heat of Combustion	100.%-	100 %-	100%-	100%

An examination of the interior of the engine during the use of the kerosene showed that there was no deposit on the surfaces, other than an infinitesimal thin film of lamp-black on the cylinder and piston-head, which did not increase with time. Second: With Pratt's fuel oil which is a dark straw colored clear petroleum of 200° Fahren. flash, 268° fire test/ sp. gravity of .852, being considerably used as a steam boiler fuel, at practicably the same number of revolùtions per minute, 20.07 B. H. P. were developed with an oil consumption of .613 lbs. or .0862 U. S. gals. per hour per B. H. P. ; and 12.56 B. H. P. with an oil consumption pf.681 lbs. or .0958 U.S. gals. per hour per B. H. P. The operation of this oil was attended the same insignificant lamp-black product.

With all these oils, the engine started instantly and p operated very steadily. The action of the valve gear was smooth and practically noiseless. The combustion occurred with perfect regularity and was accompanied with no noise other than that due to the pulsation of the floor under the engine. There was also no noise from the exhaust of the engine audible a few feet distant from the chimney into which the exhaust pipe led. Exhaust gases with all of the oils carried no soot, a pipe of white paper being unsoiled by the impingement of the gases against its surface. No deposit was found in the drum which formed a part of the exhaust pipe.

The cooling water used is shown by line 35 in the Table to be about 60 lbs. per B. H. P. per hour with a discharge temperature of 127° Fahr. The engine operated safely however with the water leaving the jacket at 140° Fahr., which would reduce the water used per hour to 50 lbs. or about 6 gals. per hour per B. H. P. The cost of operation will then be practically confined to the cost of oil and as the motor shows itself to be able to use oil obtainable at about 2 cents per gal., the cost of a brake horse power may be about .2 of one cent per hour which is slightly less than the cost of the same power for the average triple expansion steam engine, with coal at $3.00 per gross ton.

— Diesel Motor Tests —

	With Ordinary Domestic Kerosene of 120° Fahr. Flash and 150° Fahr Fire Test.							
	Approximate Brake Horse Power							
	19 H.P.	21 H.P.	21 H.P.	17 H.P.	15 H.P.	13 H.P.	9 H.P.	9 H.P.
	New York			Schroeter Augsburg	New York.		Schroeter Augsburg	New York
	1	2.	3	4.	5	6	7	8
1 Date	June 30	July 1.	July 9.	-----	July 8.	July 8.	---	July 8
2 Duration hours	2.00	4.25	1.96	-----	2.00	.49	---	1.50
— Pressures. —								
3 Barometer, inches of mercury	30.17	30.19	29.88	----	30.09	29.96	---	29.09
4 Air tank, lbs per sq. in.	636.6	631.7	643.7	----	611.7	621.8	---	618.9
— Temperatures Deg. Fahr. —								
5 Engine room and air supply.	100.5	101.3	88.5	---	93.1	97.3	---	95.1
6 Exhaust gases	740+	740+	740+	712.0	673.	601	500	511.5
7 Inlet cooling water	73.0	74.0	76.	49.32	77.	78	48.38	75
8 Outlet cooling water	109.2	110.6	119.9	68.5	108.6	128.	64.87	120.6
— Power Data: —								
9 Diam of main cylinder. inches.	10.23	10.23	10.23	9.86	10.23	10.23	9.86	10.23
10 Stroke of main cylinder. inches.	16.16	16.16	16.16	15.69	16.16	16.16	15.69	16.16
11 Diam of air cylinder. inches.	3.346	3.35	3.35	2.76	3.35	3.35	2.76	3.35
12 Stroke of air cylinder. inches.	8.05	8.05	8.05	7.87	8.05	8.05	7.87	8.05
13 M.E.P. of main cyl., lbs per sq. in.	99.56	108.9	108.46	104.07	87.612	81.197	75.1	66.76
14 M.E.P. of air cyl., lbs per sq. in.	60.00	62.07	61.7	63.3	64.9	63.5	61.4	63.4
15 Maximum press. in main cyl. from indicator cards lbs per sq. in.	520.63	521.6	521.	512.	517.5	510.	475.	505.
16 Revolutions per minute	187.0	186.6	187	154.	191.5	186.5	154.	188.5
17 Load on brake, lbs.	160.5	172.8	178.4	---	122.7	108.	---	72.
18 Indicated horse power of main cyl.	31.38	32.53	34	24.	28.	25.4	17.2	21.
19 Indicated horse power, air cyl.	2.01	2.07	2.06	1.13	2.2	2.15	1.104	2.17
20 Net indic. horse power (Line 18 – Line 19)	29.32	30.5	31.9	22.9	25.9	23.3	16.12	18.9
21 Brake horse power	19.43	20.8	21.5	17.33	15.16	13.05	9.31	8.78
22 Friction horse power (Line 20 – Line 21)	9.89	9.65	12.42	5.62	10.76	10.205	6.805	10.16
23 Mechanical efficiency (Line 21 ÷ Line 20)	66.27%	68.3%	67.87%	75.5%	58.5%	56 %	57.8%	46.4%
— Oil Consumption —								
24 Total, lbs.	21.19	47.25	23.56	---	18.	3.75	---	9.5
25 Per hour, lbs	10.5	11.12	11.8	9.35	9.00	7.65	5.86	6.33
26 Per hour per brake H.P. — lbs.	.546	.534	.549	.594	.504	.586	.68	.72
27 Per hour per brake H.P. — U.S. Gal.	.0835	.0817	.0839	.0819	.0908	.0898	.0958	.1102
28 Per hour per indicated H.P. of main cylinder — lbs	.338	.342	.347	.388	.32	.301	.34	.30
29 Per hour per indicated H.P. of main cylinder — U.S. Gal.	.0517	.0523	.0531	.0589	.0489	.046	.0516	.0459
30 Per hour per net indicated horse power — lbs.	.36	.385	.37	.408	.347	.328	.365	.334
31 Per hour per net indicated horse power — U.S. Gal.	.055	.0558	.0566	.0619	.0513	.0503	.0554	.0511
32 Efficiency brake H.P. $\frac{2560\times100}{\text{Line 26}\times\text{Line 38}}$	25.2	25.77%	25 %	25.8%	23. %	23.49%	22 %	19.1 %
33 Efficiency on net ind. H.P. $\frac{2560\times100}{\text{Line 30}\times\text{Line 38}}$	38 %	37.7%	37. %	34. %	39.6%	41.8%	38.2%	41.2%
— Cooling Water —								
34 Per hour, lbs.	---	---	1772.	3317.	1923.	---	2948	1034
35 Per hour per brake horse power lbs.	---	---	82.4	101.4	127.	---	316.5	117.8
36 Heat units abs per hour per B.H.P. B.T.U.	---	---	3617	3671	4007	---	5219	5018
— Quality of Oil. —								
37 Calorific val. (Mahler's bomb) B.T.U. per lb.	20158			19783	20166		19783	20156
38 Probable calorific val. (Mahler's bomb) corrected for condensed var. B.T.U. per lb.	18604			18371	186.04		18371	18604
39 Flash test, degrees Fahr.	120				120			120
40 Fire test, degrees Fahr.	150				150			150
41 Specific gravity at 60° Fahr.	.784			.79	.784		.79	.784
42 Wgt. per U.S. gallon, lbs.	.54			6.59	6.54		6.59	6.54

	With Fuel Oil of 218°F Flash and 330°F Fire Test.			With California Crude Oil.			With Eagle Crude Oil of 240°F Flash and 296°F Fire Test.			With Lima Crude Oil.			
	Approximate Brake H.P.			Approximate B.H.P.			Approx. Brake H.P.			Approximate Brake H.P.			
	20 H.P.	13 H.P.	No Load	20 H.P.	20 H.P.	11 H.P.	21 H.P.	11 H.P.	No Load	21 H.P.	15 H.P.	11 H.P.	No Load
	New York.			New York.			New York.			New York.			
	9.	10	11.	12	13	14.	15	16	17	18	19	20	21.
1 Date	July 29	July 29.	July 29.	Aug. 12.	Aug. 12	Aug. 12.	Aug. 27.	Aug. 27.	Aug. 27.	Aug 27.	Aug 29	Aug 27.	Aug 27.
2 Duration hours	1.94	2.07	.81	.99	.99	1.00	.76	1.00	.49	.96	10.	1.01	0.50
— Pressures. —													
3 Barometer, inches of mercury	30.	30.	30.	30.	30.2	30.2	30.1	31.14	30.15	31.14	30.19	30.16	30.16
4 Air tank, lbs per sq. in.	630.	617.4	623.	641.	647.	628.9	651.7	643.	620.4	634.6	653.	650.2	632.5
— Temperatures Deg. Fahr. —													
5 Engine room and air supply.	95.1	99.8	96.2	89.3	92.4	94.	76.	81	81.5	86.7	93.4	89.1	86.25
6 Exhaust gases	---	---	---	---	---	---	740+	516	374.7	740+	628.3	---	---
7 Inlet cooling water	74.	74.6	74.5	76.	76	76	74	74.	74	75	75.6	75.7	75
8 Outlet cooling water	113.5	104.6	98.4	118.5	117.5	110.75	127	123.9	116.5	115.7	126.	125.	119.
— Power Data: —													
9 Diam of main cylinder. inches.	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23	10.23
10 Stroke of main cylinder. inches.	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16	16.16
11 Diam of air cylinder. inches.	3.35	3.35	3.35	3.35	3.74	3.74	3.74	3.74	3.74	3.74	3.74	3.74	3.74
12 Stroke of air cylinder. inches.	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05	8.05
13 M.E.P. of main cyl., lbs per sq. in.	104.6	78.7	39.7	104.8	100.3	72.2	110.5	76.14	---	111.87	87.55	77.95	44.03
14 M.E.P. of air cyl., lbs per sq. in.	68.96	62.86	66.2	70.6	74.5	78.9	76.	79.5	75.6	83.8	79.8	83.8	82.8
15 Maximum press. in main cyl. from indicator cards lbs per sq. in.	522.	516	512	565	557.	542.	549.	529.	523	549.	537	533.	526.
16 Revolutions per minute	184.	185	187.	185	186.	187.	184.	187.	189.	187.	187.	186.	190.
17 Load on brake, lbs.	169	105.	---	169	167.	95.	175	87.8	---	172.	126	89.5	---
18 Indicated horse power of main cyl.	32.34	24.5	18.5	32.1	31.3	22.6	34.	23.9	13.2	35.	27.3	24.4	14.
19 Indicated horse power, air cyl.	2.3	2.25	2.2	2.9	3.1	3.1	3.13	3.3	3.2	3.5	3.3	3.5	3.52
20 Net indic. horse power (Line 18 – Line 19)	30.07	22.3	10.3	29.2	28.2	19.5	31.	20.5	10.	31.6	23.9	20.87	10.5
21 Brake horse power	20.	12.6	---	19.5	20.	11.5	20.8	10.6	---	20.7	15.16	10.77	---
22 Friction horse power (Line 20 – Line 21)	10.	9.68	10.29	9.55	8.15	8.10	10.14	9.95	10.	10.9	8.8	10.10	10.5
23 Mechanical efficiency (Line 21 ÷ Line 20)	66.7%	56.5%	---	67.3 %	71. %	58.6%	67. %	51.5%	---	65.6%	63.8%	51.6 %	---
— Oil Consumption —													
24 Total, lbs.	23.9	17.7	3.57	13.37	12.4	8.	8.75	7.3	2.56	11.75	87.8	7.3	2.56
25 Per hour, lbs	12.31	8.55	4.39	13.51	12.6	8.0	11.5	7.3	5.2	12.4	8.71	7.3	5.12
26 Per hour per brake H.P. — lbs.	.613	.681	---	.686	.625	.699	.556	.641	---	.590	.579	.672	---
27 Per hour per brake H.P. — U.S. Gal.	.0862	.0958	---	.0975	.0838	.099	.078	.097	---	.0826	.0811	.0941	---
28 Per hour per indicated H.P. of main cylinder — lbs	.381	.349	.351	.421	.401	.354	.338	.307	.395	.349	.321	.297	.365
29 Per hour per indicated H.P. of main cylinder — U.S. Gal.	.0536	.0491	.0494	.0596	.0568	.0501	.0477	.0434	.0558	.0439	.045	.0416	.051
30 Per hour per net indicated horse power — lbs.	.409	.384	.427	.463	.445	.409	.373	.356	.520	.387	.365	.347	.487
31 Per hour per net indicated horse power — U.S. Gal.	.0575	.054	.0501	.066	.065	.067	.052	.0503	.0734	.050	.0515	.0485	.0682
32 Efficiency brake H.P. $\frac{2560\times100}{\text{Line 26}\times\text{Line 38}}$													
33 Efficiency on net ind. H.P. $\frac{2560\times100}{\text{Line 30}\times\text{Line 38}}$													
— Cooling Water —													
34 Per hour, lbs.	1989.8	1937.8	1916.5	1997.	1821.	1949.	1242.	951.	963.	1783.	1085	990.	862.5
35 Per hour per brake horse power lbs.	99.	154.	---	101.7	90.8	170.4	59.9	89.8	---	86.	71.6	91.9	---
36 Heat units abs per hour per B.H.P. B.T.U.	3902	4629	---	4322	3768	5921.	3175	4481	---	3580.	3629.	4531.	2161.
— Quality of Oil. —													
37 Calorific val. (Mahler's bomb) B.T.U. per lb.	19980			19376			19742			19548			
38 Probable calorific val. (Mahler's bomb) corrected for condensed var. B.T.U. per lb.													
39 Flash test, degrees Fahr.	200						249.			90			
40 Fire test, degrees Fahr.	268						296.			160			
41 Specific gravity at 60° Fahr.	.852			.846			.849			.856			
42 Wgt. per U.S. gallon, lbs.	7.11			7.06			7.08			7.14			

Conclusion.

From present indications the probable advance in gas and petroleum engines will be along the lines of improvement by means of increasing the temperatures and necessarily higher initial compression of the explosive mixtures.

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