T H E S I S

THE THERMODYNAMICS AND DESIGN OF SOLID INJECTION INTERNAL COMBUSTION ENGINES

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For the Degree of Master of Science

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Fort Collins, Colorado

January 16, 1926.

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January I6, I926

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I N D E X

TABLES

THE THERMODYNAMICS AND DESIGN OF SOLID INJECTION INTERNAL COMBUSTION ENGINES

INTRODUCTION

All modern gas engines are operated on the internal combustion method of adding heat to a mass of working gas both explosively and non-explosively, a fact that is responsible for one group of limitations in the results obtainable. There are two direct consequences of importance - first, the charge must be renewed each cycle, fresh air and fuel coming into the working chamber and hot products of combustion leaving alternately, with the other phases between, and second, the obtainable temperature rise with its corresponding volume increase, is fixed by fuel-air-mixture properties. Furthermore, as all these operations take place in cylinders having pistons that require lubricant and which must, therefore, be kept cool enough to prevent destruction of the lubricant, a heat exchange takes place between gases and walls alternately in each direction which causes corresponding volume and temperature changes in the gas not contemplated by the cycle nor subject to computation or formulation. Besides these influences, there are others chargeable to mechanical construction or adjustments that are responsible for further departures in the pressure, volume, and temperature changes from the ideal cyclic ones, as for example, too early an exhaust curtails expansion; too late a closure of admission valve similarly delays the beginning of compression and reduces the total amount. In Diesel engines where the fuel burns as fast as it enters, inaccurately graduated injection may cause the combustion to depart from the truly constant pressure sort, too rapid a feed causing pressure to rise; too slow a feed permitting a fall in pressure.

The mean effective pressure, thermal efficiency, and other characteristic performances differ in real gas engines from their cyclic equivalents, but this does not in any way reduce the value of cyclic analysis. There are two sorts of contributions of practical value in the cyclic analysis taken in conjunction with the performance of real engines. The first is of the fundamental scientific order, giving numerical value to the extent of the possible improvement of real engines of any one class and, pointing out just where the losses occur with the amount of each indicates where any improvement must be made. The second is of more direct practical value to builders and users of engines as it is concerned with the prediction of results, for a given cylinder and fuel just what horse power and fuel comsumption may be expected, or what cylinder size is required for a given output. In these latter cases the cyclic analysis is responsible for the form of equations for mean effective pressure and thermal efficiency, actual tests furnishing numerical values for constants of proportionality.

As cyclic performance is to be the basis of all computations on the approach to perfection of performance in real engines and of their probable power and efficiency, it is necessary to select the standard cyclic equations as a first step. It has been shown that comparatively simple performance equations are derivable for Diesel engine cycles if the specific heats of gases are assumed to be constant, whereas it is known that they are not constant. But with any proposed law of variation in specific heats, it has also been shown that the equations for cyclic performance are very difficult of any solution and impossible to exactly solve. This makes it difficult to decide on a course of procedure for practical computations in the first group of comparisons that are concerned with the approach to perfection of real engines. Strictly speaking, they should be compared with cyclic performance as computed on the basis of variability of specific heats, but in view of first, the uncertainty of the law of variation, and second, of the complexity of the cyclic equations, this is not yet a feasible thing as a matter of regular engineering routine.

The best method is the cyclic results based on constancy of specific heats as the basis of real engine.

performance, and in the simple every-day predictions of probable power and efficiency this is quite as good as any other, because a single constant factor can include all cyclic departures as well as the losses in the engine itself. The next step in establishing the cyclic standard, is to fix the physical properties of the working gases, for these gases include some fuel, some air, and some products of combustion left in the cylinder from the previous explosion. Every change in fuel, or in proportions of fuel to air, to burnt gases, involves a different specific heat for the working mixture and even for a constant mixture, the expansion stroke, being made only with burnt gases, will have different specific heats than the compression stroke where the mixture is yet unburnt.

TYPES OF DIESEL ENGINES

The Diesel Engine is usually built 4 stroke cycle and single-acting in sizes up to 1000 horse power per cylinder. The 2 stroke cycle single-acting type is made by a number of European builders in sizes up to 1250 horse power per cylinder; this type is more easily made reversible and consequently has been adopted for marine use. A 2 stroke cycle cylinder developes from 170 to

180 per cent of the work of a 4 cycle cylinder of the same piston displacement, but its fuel consumption per brake horse power is about 10 per cent greater. The mechanical efficiency of the 4 stroke cycle (2 stroke cycle) type is about 75 (70) per cent.

The air for spraying the oil is supplied to Diesel engines by 2 or 3 stage compressors at a pressure of from 800 to 1100 pounds per square inch. The air required varies from 16 to 34 cubic feet of free air per brake horse power; the power required by the compressor is from 4 to 7 per cent of the rated power of the engine. Diesel engines are usually rated with an overload capacity of 10 to 15 per cent. The heat carried away by the jackets is 2500 to 3000 B.T.U. per brake horse power per hour, corresponding to a heat transfer of about 4000 (9000) B.T.U. per square foot of cylinder surface per hour in 4 stroke cycle (2 stroke cycle) engines. Regulation usually within 3 per cent. Fuel consumption as low as 0.4 pounds per brake horse power per hour, with good fuel (19500 B.T.U. per pound). Lubricating oil 0.01 pounds per brake horse power per hour. Cost of engine per brake horse power $63 to $48 respectively for 200 (1000) brake horse power units. Weight per brake horse power 250 (500) pounds.

LIQUID FUELS

Any combustible liquid can be used as a fuel in an engine designed and adopted to it. Liquid fuels are vaporized either before or during their mixture with the air for combustion. Gasoline and similar light hydrocarbons can be vaporized either by passing air over the surface or bubbling it through the liquid (carburetting by evaporation), or by atomizing them in the air current (carburetting by injection). Petroleum, crude oil, coal-tar oil, alcohol, and other liquid fuels whose boiling points are high, are vaporized by heating the liquid. The lighter of these liquids - petroleum, light crude oil, and alcohol - are used in engines of the explosive type, in which the liquid may be vaporized (a), in a vaporizer outside of the engine which is kept hot by the exhaust gases; (b), in a vaporizer-- hot bulb--connected to the combustion chamber of the engine and kept hot by the explosions; or (c), against a hot part of the engine - a plate projecting from the cylinder head. In the last case the engine must be run on gasoline until the plate becomes sufficiently hot to vaporize the fuel - kerosene. The heavier liquids can be used only in engines of the constant-pressure-combustion type. The heavier the fuel used, the more finely must it be divided when injected into the cylinder in order to insure complete vaporization and ignition. If the heavier oils are injected in large particles only the surface of the particles will be burnt while the center will be converted by the heat into a pitchy substance, which will be deposited on the cylinder walls and valves.

RELATIVE INCREASE OF FUEL CONSUMPTION PER BRAKE HORSE POWER AT PARTIAL LOADS

Full-load	3/4 load	1/2 load	1/4 load
1.00	1.02-1.15	1.07-1.25	1.40-1.90

In explosion engines the total on load consumption goes as high as 30-45 per cent. of the total consumption at normal load; in constant-pressure-combustion engines this figure is only 20-25 per cent.

While different types of engines have been mentioned this treatise will deal only with one type with a specific brake horse power, and designed to operate at its highest efficiency at sea level where the volumetric efficiency is (one) with properly designed valves.

SYMBOLS

t and T - Temperature, degrees, fahr. and abs. respectively.

p = Absolute pressure lb. per sq. in.

v = Volume of gas or of mixture of gases cu. ft.

c_p and c_v = Specific heats at constant (pressure and volume).

$x = c_p / c_v$ = Ratio of specific heats.

J = I/A = 778 = Mechanical equivalent of heat.

$R = 778 (c_p - c_v)$ = Gas constant. For air R = 53.2

M = 144 pv/RT - Weight of volume v of gas.

Q = B.T.U. added or substracted during a given operation.

W = Work done in ft. lbs. during this operation.

n = Exponent of the polytropic curve $pv^n = K$

$r_c = v_1/v_2$ = Volumetric compression ratio

$r_d = v_3/v_2$ = Constant pressure ratio, ratio of cut off volume to clearance volume.

$r_e = v_1/v_3$ = Ratio of expansion

B.H.P. = Brake horse power

C = Fuel consumption in lbs. per B.H.P. per hour

C_h = Total fuel consumption per hour at normal load in lbs.

L = Volume of air actually in the mixture per cu. ft. per lb. of fuel, in cu. ft.

H = Lower heat value of 1 lb. of the fuel under standard conditions. (29.9 in. mercury and 62 deg. fahr.).

d = Piston diameter ft.

L = Stroke ft.

v_d = Piston displacement per stroke, cu. ft.

N = Revolutions of crank shaft per min.

p_m = Indicated mean effective pressure, lb. per sq. in.

E_t = Theoretical indicated thermal efficiency

E_i = Actual indicated thermal efficiency.

E_m = B.H.P./I.HP. = Mechanical efficiency.

E_s = Volumetric efficiency of the suction stroke.

$E_c = E_i/E_t$ = Ratio between the indicated and theoretical thermal efficiencies = Diagram factor.

$E_e = E_c E_t E_m$ = Economic efficiency = Thermal efficiency at the brake

THE WORKING PROCESSES IN FOUR-STROKE CYCLE DIESEL ENGINE

THE SUCTION STROKE

The compression line follows the equation $pv^n =$ constant. At the innermost position of the piston the compression space is filled with v_c cu. ft. of unexpelled exhaust gases whose weight in pounds is $M_r - 144\ p_r v_c/R_r T_r$. If the temperature of the charge before admission is T_m, the weight of the mixture drawn in is $M_m = (E_s v_d/T_m)$ $(144 \times 14.7/R_m)$ lb. at the end of the suction stroke, and the weight of the total charge in pounds is:

$$M_s = M_r\ M_m = 144\ p_s\ vt/T_s R_s,$$

Then - - $p_s vt/T_s = p_r v_c/T_r + 14.7 E_s v_d/T_m$.

(The subscript t with v equals the temperature T) provided it is assumed that $R_s = R_m = R_r$, which can be done without sensible error. In order to obtain a maximum amount of work per stroke E_s must be made as large as possible, by diminishing the suction and exhaust resistances and the suction temperature. The influence of atmospheric pressure on E_s and consequently on the I.H.P. must be especially reconed with plants located at high altitudes.

THE COMPRESSION STROKE

Since the compression line follows the curve $pv^n =$ constant and we assume the value of 12.5 for p_s the value of having been determined by experiment to be 1.41, we get:

$$n = 1.41 \text{ and } r_c = 15 \quad 1 - \frac{1}{r_c^{\,n-1}} = \begin{array}{l} p_c = 569.1 \\ E_t = \ \ 64.2 \\ T_c = 2125 \end{array}$$

The clearance volume for a four stroke cycle engine of the constant pressure type should be from 7 to 8 per cent of the piston displacement, and by experiment the following values were found to be more nearly correct for the design of solid injection engines.

p_c = 500 lb. Clearance 7 to 6 per cent t_c = 1070

The volume of the atomizer in oil engines should vary with the compression pressure, but with compression above 400 pounds per sq. in. the atomizer could be dispensed with except to allow for starting and other contingencies. To allow for starting and slow speeds the ratio of $V_v/V_c = .04$ is taken.

THE EXPANSION STROKE

The variation of the explosion from the theoretical constant volume combustion is indicated by the inclination of the actual combustion line and by the varying expansion lines in sketch 3, page 12. When the explosion is retarded so that

$V'_{ex} \quad v_c, \; T'_{ex} = T_c \, p'_{ex} \, v'_{ex}/p_c v_c = p'_{ex} \, v'_{ex}/MR_e.$

If the value of P'_{ex} can be taken directly from an indicator diagram, the above equation will give the actual explosion temperature T'_{ex}; if, however, p_{ex} or T_{ex} is taken from a hypothetical indicator card, the value of T'_{ex} or P'_{ex} can be found only by multiplying by a reduction factor which takes into account the decrease of temperature or pressure due to heat losses, cooling, etc. The value of this factor is not far from that of the card factor E_c of the cycle. Values of explosion and terminal pressures and temperatures from practice are as follows:

t_{ex}= 2240 to 3140 t_e = 1190 to 1940 p_{ex}= 200 to 375 p_e=37 to 75

THE EXHAUST STROKE

The exhaust valve usually begins to open at between 80 per cent and 90 per cent of the expansion stroke. During the establishment of pressure equilibrium, the velocity of the gases is very great, about 2800 ft. per sec.

If the opening of the exhaust valve is retarded, the negative work is increased and the cylinder temperature is increased which in turn decreases the volumetric efficiency, E_s, and the allowable compression pressure. With correctly timed valve opening, sufficient valve lift, few changes of direction of the gases, and sufficiently large exhaust pipe, the resistance p_r will be a minimum, and together with properly designed inlet valves and passages will insure a maximum value of the volumetric efficiency E_s. Under these conditions and with an exhaust pipe fitted with a long vertical riser, to act as a chimney, it is possible through the kinetic energy of the exhaust gas column to bring the resistance p_r down to atmospheric or even a little below.

The pressure along the exhaust line is usually from 15 to 17 lb. abs.; the exhaust temperature (outside the cylinder and close to the exhaust valve) varies from 575 to 1000 deg. fahr.; this may be raised considerably by the use of rich mixtures and by after-burning due to late ignition or improper mixtures. The average percentage composition of the exhaust gases by volume is CO_2, 10; O_2, 5 to 10; Co, o or only a slight trace; H_2 or CH_4 none; N, 80 to 85. For economical running there should be no trace of combustible gases in the exhaust gas.

THE THERMODYNAMICS OF THE DIESEL ENGINE

WORK DEFINED

Work, in the proper sense of performance of any labor is not a sufficiently precise term for use in computations, but the analytical mechanics have given a technical meaning to the word which is definite and which is adopted in all thermodynamic analyses. The mechanical definition of work is mathematical inasmuch as work is always a product of forces opposing motion and distance swept through, the force entering with the product being limited to that acting in the direction of the motion. The unit of distance in the English system is the foot, and of the force the pound, so that the unit of work is the foot pound, Thus the lifting of one pound weight one foot requires the expenditure of one-foot pound of work. It is not only by lifting and falling weights that work is done; for if any piece of mechanism be moved through a distance of one foot, whether

CYCLE DIAGRAMS

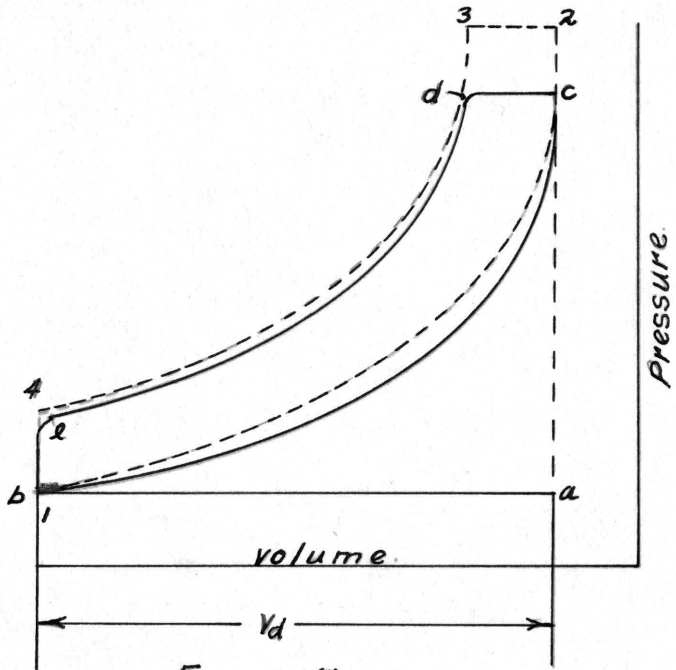


Figure (1)

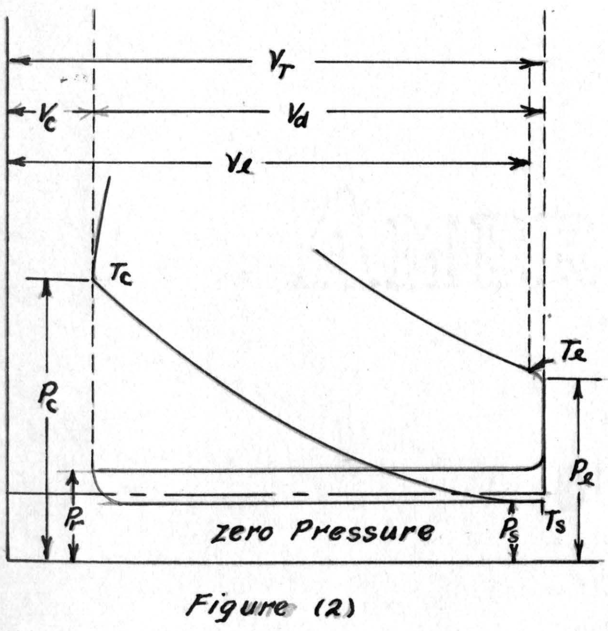


Figure (2)

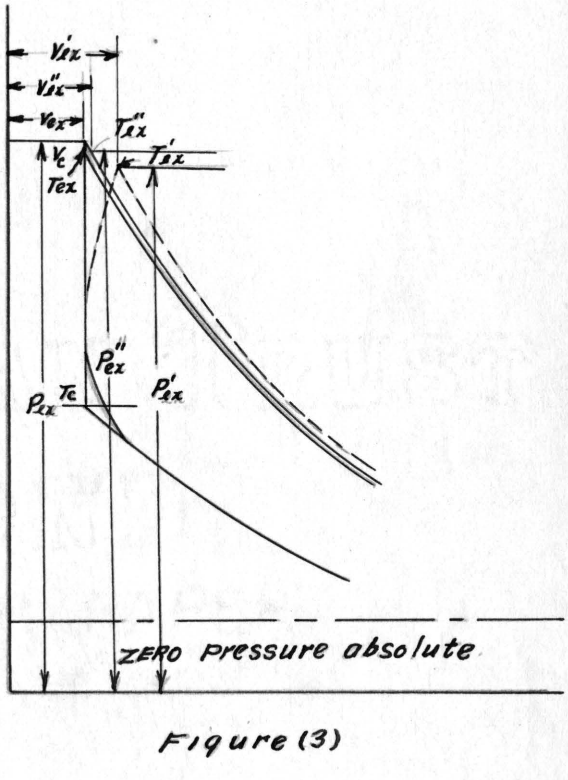


Figure (3)

in a curved or straight path, and its movement be resisted by a force of one pound, there will be performed one foot pound of work against resistance.

Work is used in the negative sense as well as in the positive sense, as the force considered resists or produces the motion, and there may be both positive and negative work done at the same time; similar distinctions may be drawn with reference to the place or location of the point of application of the force. Consider, for example, the piston rod of a direct acting pump in which a certain force acting on the steam end causes motion against some less or equal force acting on the water end. Then the work at the steam end of the pump may be considered to be positive and at the water end negative, so far as the movement of the rod is concerned; when, however, this same movement causes a movement of the water, work done at the water end (although negative with reference to the rod motion, since it opposes that motion) is positive with reference to the water, since it causes this motion. It may also be said that the steam does work on the steam end of the rod and the water end of the rod does work on the water, so that one end receives and the other delivers work, the rod acting as a transmitter or that the work performed at the steam end is the input and that at the water end the output work.

POWER DEFINED

Power is defined as the rate of working or the work done in a given interval of time, thus introducing a third unit of mechanics, time, so that power will always be expressed as a quotient, the numerator being a product of force and distance, and the denominator, time. The unit of power in the English system is the horse power, or the performance of 550 foot pounds per second or 33,000 foot pounds per minute.

WORK IN TERMS OF PRESSURE AND VOLUME

Another of the definitions of mechanics fixes pressure as force per unit area so that pressure is always a quotient, the numerator being force and the denominator, area or the length to the second power. If, therefore, the pressure of a fluid be known, and according to hydromechanics it acts equally and normally over all surface in contact with it, then the force acting in a given direction against any surface will be the product of the pressurer and the projected area of the surface, the projection being on a plane at right angles to the direction considered. In the case of pistons and plungers the line of direction is the axis of the cylinder, and the projected area is the area of the piston, less the area of any rod passing completely through the fluid that may be so placed. When this plane area moves in a direction perpendicular to itself, the product of its area and the distance will be the volume swept through and if the piston be involved, the volume is technically the displacement of the piston. Accordingly, work may be expressed in three ways:

Work = Force times Distance
Work = Pressure times Area times Distance
Work = Pressure times Volume

The product should always be in foot pounds, but will be only when appropriate units are chosen for the factors. These necessary factors are given as follows:

WORK IN FOOT POUNDS = Force in lbs. x Distance in ft.

= Pressure in lbs. per sq. ft. x area in sq. ft. x Distance in ft.

= Pressure in lbs. per sq. in. x area in sq.in. x Distance in ft.

= Pressure in lbs. per sq. ft. x Volume in cu. ft.

= Pressure in lbs. per sq. in. x 144 x Volume in cu. ft.

As pressures are in practice expressed in terms not only as above, but also in heights of columns of common fluids and in atmospheres, it is convenient for calculation to set down factors of equivalence.

	lbs. per sq. foot	lbs. per sq. inch	in. of Hg. 32° F.	Atmos. sea level
One lb. per sq. ft.	1	.006944	.014139	.000472
One lb. per sq. in.	144	1	2.03594	.06802
One oz. per sq. in.	9	.0625	.127246	.004252
FLUID PRESSURES				
One ft. H_2O 62° F	62.355	.43302	.88080	.02946
One in. H_2O 62° F	5.196	.03608	.07340	.00245
One in. Hg 32° F	70.729	.49117	1.0000	.03341
One ft. air 32° F 1 atmos. pres.	.0807	.00056	.00114	.000038
One ft. air 62° F.	.0760	.00052	.00107	.000035

In thermodynamic computations the pressure volume product as an expression for work is most useful, as the substances used are always vapors and gases, which have the valuable property of changing volume indefinitely with or without change of pressure, according to the mode of treatment. Every such increase of volume gives, as a consequence, some work since the pressure never reaches zero, so that to derive work from vapors and gases they are treated in such a way as will allow them to change volume considerably with as much pressure acting as possible.

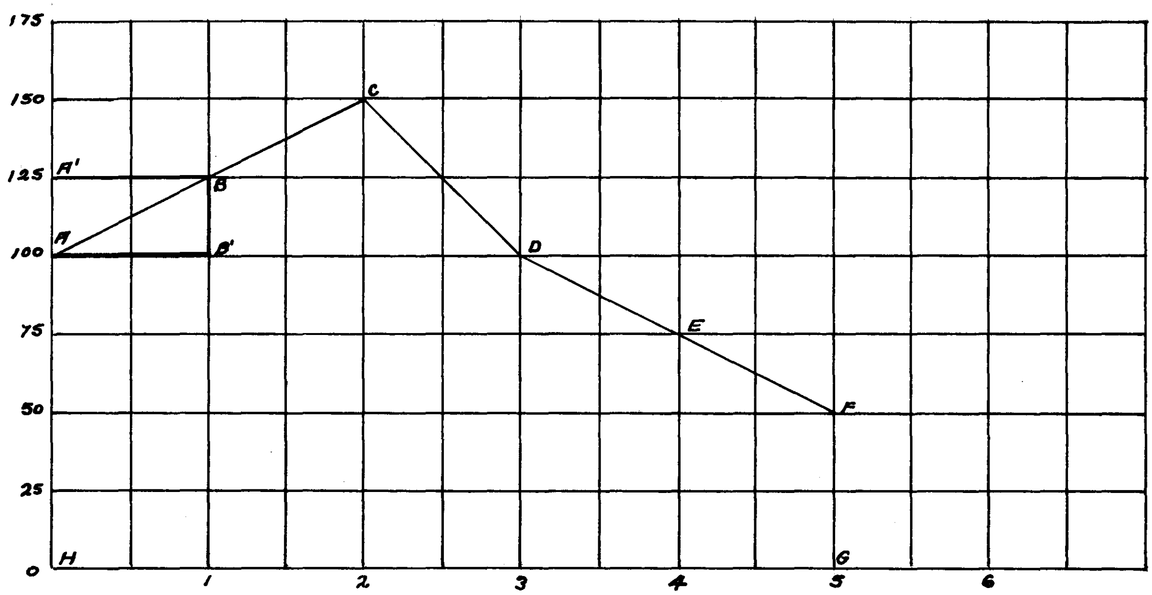
It should be noted that all true pressures are always absolute, that is, measured above a perfect vacuum or counted from zero, while most pressure gages and other devices for measuring pressure, such as indicator, gives the result measured either above or below atmospheric pressure. In all problems involving work of gases or vapors, the absolute values of pressures must be used; hence, if a gage or indicator measurement is being considered, the pressure of the atmosphere found by means of the barometer must be added to the pressure above atmosphere in order to obtain the true pressure.

WORK BY PRESSURE VOLUME CHANGE

Suppose that instead of being constant the pressure were irregular and being measured at intervals of 1 cu. ft. displacement, as follows:

Pressure Lbs. per sq. ft.	Displacement volume cu.ft.
100	0
125	1
150	2
100	3
75	4
50	5

This condition might be plotted in the following manner, using the letters A, B, C, D, E, F, G, H, as the observation points. The work done will be the area under the line joining the observation points. In the absence of exact data on the nature of the pressure variations between the two observation points, A and B, a variety of assumptions might be made as to the precise evaluation of this area, as follows:



(a) The pressure may have remained constant at its original value for the first cubic foot of displacement as shown dotted A-B', and then suddenly have risen to B. In this case the work done for this step would be 100 ft. lbs.

(b) Immediately after the measurement at A, the pressure may have risen to A' and remained constant during displacement A' B, in which case the work done would be 125 ft. lbs.

(c) The pressure may have risen regularly along the solid line A B, in which case the work area is a trapezoid and has the value $\frac{100 + 125}{2} \times 1 = 112.5$ ft. lbs.

It thus appears that for the exact evaluation of work done by volume change, continuous data are necessary on the value of pressure with respect to volume. If such continuous data, obtained by measurement or otherwise be plotted, there will result a continuous line technically termed the pressure volume curve for the process.

The work done during a displacement of this nature under continuely varying pressure is likewise the area between the curve and the horizontal axis when pressures are laid off vertically, and will be in foot pounds if the scale of pressures is pounds per square foot, and volumes, cubic feet. Such an irregular area can be divided into small vertical rectangular strips, each so narrow that the pressure is sensibly constant, however, much it may differ in different strips. The area of the rectangle is P V, each having the width V and the height P, and the work area will be exactly evaluated if the strips are narrow enough to fulfill the conditions of sensibly constant pressure in any one. This condition is true only for infinitely narrow strips having the width dV and height P, so that each has the area PdV and the whole area or work done is $W = \int PdV$.

$$W = \int P \, dV$$

This is the general algebraic expression for work done by any sort of continuous pressure volume change. It thus appears that whenever there are available sufficient data to plot a continuous curve representing a pressure volume change, the work can be found by evaluating the area lying under the curve and bounded by the curve coordinates and the axis of volumes. The work done may be found by actual measurement of the area or by the algebraic solution of equation $W = \quad P\,dV$ which can be integrated only when there is a known algebraic relation between the pressure and the corresponding volume of the expansive fluid, gas or vapor.

WORK OF EXPANSION AND COMPRESSION

Any given quantity of gas or vapor confined and not subject to extraordinary thermal changes, such as explosion, will suffer regular pressure changes for each unit of volume change, or conversely, suffer regular volume change for each unit of pressure change, so that pressure change is dependent on volume change and vice versa. When the volume of a mass of gas or vapor, V_1, is allowed to increase to V_2 by the movement of a piston in a cylinder, the pressure will regularly increase or decrease from P_1 to P_2, and experience has shown that no matter what the gas or vapor or the thermal conditions, if steady, the volumes and pressures will have the relation for the same mass,

$$P_1V_1^n = P_2V_2^n = K$$

or the product of the pressure and n power of the volume of a given mass will always be the same. The exponent n may have any value, but usually lies between 1 and 1.5 for conditions met in practice.

The precise value of n for any given case depends on:

(a) The substance

(b) The thermal conditions surrounding expansion or compression, n being different if the substance receives heat from, or loses heat to, external surroundings, or neither receives nor loses.

Not only does the equation $P_1 V_1{}^n = P_2 V_2{}^n = K$ express the general law of expansion, but it likewise expresses the law of compression for decreasing volumes in the cylinder with corresponding rise in pressure.

From the equation -

1 . . . . $P_2 = P_1 \dfrac{V_1^n}{V_2^n}$ = pressure after expansion

or raising both sides to the $\dfrac{1}{n}$ power, we get

2 . . . . $V_2 = V_1 \dfrac{P_1^{1/n}}{P_2^{1/n}}$ = the volume after expansion so that the final volume depends on the original volume, on the ratio of the two pressures and on the exponent.

From 1 it can be seen that the pressure after expansion depends on the original pressure, on the ratio of the two volumes and on the exponent.

The general equation for work of expansion or compression can now be integrated by means of the equation $P_1 V_1 = P_2 V_2^n = K$ which fixes the relation between pressures and volumes.

Disregarding subscripts and solving for P, we get

$$P = \frac{K}{V^n}$$

which, substituted in the equation $W = \int P\, dV$, we get

$$W = \int \frac{K\, dV}{V^n}$$

but as K is constant,

$$3 \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; W = K \int \frac{dV}{V^n}$$

The integral of equation 3 will have two forms -

(1) When n = I, in which case $P_1 V_1 = P_2 V_2 = K^1$;

(2) When n is not equal to I.

Taking first the case when n = I

Whence $$W = K_1 \int_{V_1}^{V_2} dV/V$$

When n - I
$$\left\{ \begin{aligned} W &= K^1 \, \text{Log}_e \, V_2/V_1 \; . \; . \; . \; . \; . \; . \; a \\ &= P_1 V_1 \, \text{Log}_e \, V_2/V_1 \; . \; . \; . \; . \; . \; b \\ &= P_2 V_2 \, \text{Log}_e \, V_2/V_1 \; . \; . \; . \; . \; . \; c \\ &= K^1 \, \text{Log}_e \, P_1/P_2 \; . \; . \; . \; . \; . \; d \\ &= P_1 V_1 \, \text{Log}_e \, P_1/P_2 \; . \; . \; . \; . \; . \; e \\ &= P_2 V_2 \, \text{Log}_e \, P_1/P_2 \; . \; . \; . \; . \; . \; f \end{aligned} \right.$$

All of the above equations are equal but are set down in different forms for convenience in computation; in them -

V_2 equals largest volume equals initial volume for compression equals final volume for expansion.

P_2 equals smallest pressure equals initial pressure for compression equals final pressure for expansion.

V_1 equals smallest volume equals final volume for compression equals initial volume for expansion.

P_1 equals largest pressure equals final pressure for compression equals initial pressure for expansion.

These equations all indicate that the work of expansion and compression of this class is dependent only on the ratio of pressure or volumes at the beginning and end of the process, and the PV product at either beginning or end, this product being of constant value.

When the exponent n is not equal to one, the equation takes the form -

$$W = K\int_{V_1}^{V_2} dV/V^n = K\int_{V_1}^{V_2} V^{-n}\, dV$$

$$= \frac{K}{I - n}\left(V_2^{\,I-n} - V_1^{\,I-n} \right)$$

As n is greater than one, the denominator and exponent will be negative, so changing the form to secure positive values,

$$W = \frac{K}{n-1}\left(I/V_1^{\,n-1} - I/V_2^{\,n-1}\right)$$

This can be put in a still more convenient form. Multiplying and dividing by $I/V_2^{\,n-I}$ or $I/V_1^{\,n-I}$ we get,

$$W = \frac{K}{n-I}\,\frac{I}{V_2^{\,n-I}}\left[(V_2/V_1)^{n-1} - 1\right] = \frac{K}{n-I}\,\frac{I}{V_1^{n-I}}\left[I - (V_1/V_2)^{n-I}\right]$$

Substituting the value $K = P_2V_2^{n} = P_1V_1^{n}$

$$W = \frac{I}{n-1}\,\frac{P_2V_2^{n}}{V_2^{n-1}}\left[(V_2/V_1)^{n-1} - 1\right] = \frac{I}{n-I}\,\frac{P_1V_1^{n}}{V_1^{n-1}}\left[(V_1/V_2)^{n-1} - I\right]$$

Whence

When n = I . .

$$\begin{cases} W = \dfrac{P_2V_2}{n-1}\left[(V_2/V_1^{\,n-1} - 1\right] & \ldots\ a \\[2ex] = \dfrac{P_2V_2}{n-1}\left[(P_1/P_2)^{\frac{n-1}{n}} - I\right] & \ldots\ b \\[2ex] = \dfrac{P_1V_1}{n-1}\left[I - (V_1/V_2)^{n-1}\right] & \ldots\ c \\[2ex] = \dfrac{P_1V_1}{n-1}\left[I - (P_2/P_1)^{\frac{n-1}{n}}\right] & \ldots\ d \end{cases}$$

The above equations give the work done for this class of expansion and compression in terms of pressure ratios and volumes ratios and in them -

V_2 equals largest volume equals initial volume for compression equals final volume for expansion.

P_2 equals smallest pressure equals initial pressure for compression equals final pressure for expansion.

V_1 equals smallest volume equals final volume for compression equals initial volume for expansion.

P_1 equals largest pressure equals final pressure for compression equals initial pressure for expansion.

The work of expansion or compression is dependent upon the ratio of pressures or volumes at the beginning and end of the process, the exponent, and on the pressure volume product appropriately taken. The work done by expansion or compression of both classes, shows that it is dependent on the initial and final values of pressures and volumes and on the exponent n, which defines the law of variation of pressure with volume between the initial and final states.

VALUES OF EXPONENT n DEFINING SPECIAL CASES OF EXPANSION OR COMPRESSION

The method for determining n as applied to Diesel engines is termed the experimental method. If by measurement, the pressures and volumes of a series of points on an expansion or compression curve, obtained by test with appropriate instruments, for example the indicator - be set down in a table and they be compared in pairs, values of n can be found as follows:

Calling the points a, b, c, etc.,

$$P_a V_a^n = P_b V_b^n \ ,$$

And $\log P_a + n \log V_a = \log P_b + n \log V_b$

or $n (\log V_b - \log V_a) = \log P_a - \log P_b \ ,$

hence $\quad n = \dfrac{\log P_a - \log P_b}{\log V_b - \log V_a}$ . . . . a

or $\quad n = \dfrac{\log (P_a/P_b)}{\log (V_b/V_a)}$ . . . . b

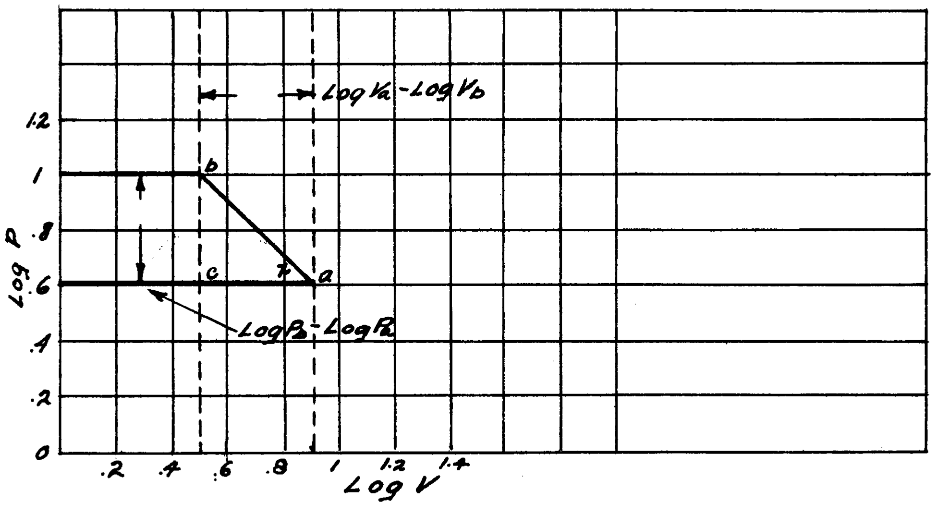
According to equation a, if the difference between the logarithms of the pressures at b and a be divided by the differences between the logarithms of the volumes at a and b respectively, the quotient will be n. According to equation b, the logarithm of the ratio of the pressures, b to a, divided by the logarithm of the ratio of the volumes, a to b respectively, will also give n. It is interesting to note that if the logarithms of the pressures be plotted vertically and logarithms of volumes horizontally, then the line ac equal to the intercept on the horizontal axis represents the difference between the logarithms of volumes, or

$$ca = \log V_a - \log V_b$$

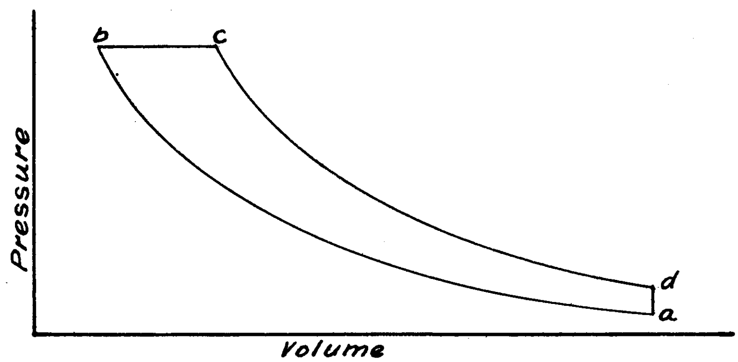
and similarly $\quad cb = \log P_b - \log P_a$

Hence

$$n = \frac{cb}{ca} = \tan x$$



Let C_p and C_v be the specific heats at constant pressure and constant volume, and n their ratio; also call Q_a and Q_b respectively the heat received during combustion and the heat rejected during exhaust, and finally call r the ratio of total volume to clearance volume and e the ratio of cut off volume to clearance volume.



Let a, b, c, in the above figure be a pressure volume diagram of the Diesel cycle with constant pressure combustion with adiabatic compression and expansion.

The efficiency will be -

$$E = I - \frac{Q_a}{Q_b} = I - \frac{C_v(T_d - T_a)}{C_p(T_c - T_b)}$$ substitute $n = \frac{C_p}{C_v}$ and

factor out the compression-line temperatures. This gives

$$E = I - \frac{T_a(T_b/T_a - I)}{nT_b\ (T_c/T_b - I)}$$ but by Charles' law, $\frac{T_c}{T_b} = \frac{V_c}{V_b} = e$

also, $\frac{T_b}{T_a} = (V_a/V_b)^{n-I} = r^{n-I}$ for adiabatic compression

also, $\frac{T_d}{T_c} = (V_c/V_d)^{n-I} = \overline{(V_c/V_a)\ (V_b/V_d)}^{\,n-I} = (e/r)^{n-I}$

And $\frac{T_d}{T_a} = (T_d/T_c)\ (T_c/T_b)\ (T_d/T_a) = (e/r)^{n-I}e.r^{n-I} = e^n$

Substituting in equation 2, we get

$E = I - I/nr^{n-I}\ (e^n - I)/(e - I)$

$n = I.4$ $r = I.2786/.0836 = 15.25$ $e = .215/.0836 = 2.57$

Admission during 27 degrees of working stroke.

$V_c = 18\%\ V_d = .215$ cu. ft.

$E = I - I/I.4 \times 15.25^{.4}\ (2.57^{I.4} - I)/2.57 - I$

$E = I - .625 = 37.5\%$

$\frac{P_b}{P_a} = \frac{500}{14.7} = 34$ Atmos. Volume ratio $= \left(\frac{P_b}{P_a}\right)^{\frac{I}{n}} = 34^{.714} = 12.4$

Temperature ratio $= \left(\frac{P_b}{P_a}\right)^{\frac{n-I}{n}} = 34^{.28} = 2.684$

Initial temperature $= 120^{\circ}$

Compression temperature $= 2.684\ (460 + 120) = 1552^{\circ}$ Abs.

$$\text{Combustion temperature} = \frac{1923 + 1092 + 460}{1092 + 460} = 2.28 = e$$

The temperature rise during combustion assuming 32 pounds of airper pound of oil burned, and 3 pounds injection air with a mean specific heat of .26

$$\frac{18{,}000}{(32 + 3 + I)\ .26} = 1923^{\circ}$$

Combustion temperature $= 3015^{\circ}$ F. or 3465° Abs.

In terms of atmospheres of pressure

$P_a = I \quad P_b = P_c = 34 \quad P_d = P_c\ (V_c/V_d)^{1.4} = 34\ (.18)^{1.4} = 3.08$

Volumes in cubic feet

$V_a = V_d = 1.195 + .0836 = 1.2786 \quad V_c = .215 \quad V_b = .0836$

Absolute temperatures F.

$T_a = 460 + 120 = 580^{\circ} \quad T_d = T_a\ (P_d/P_a) = 580\ (3.08) = 1785^{\circ}$

$T_b = 1552^{\circ} \quad T_c = 3465^{\circ}$

$Q_2 = C_v\ (T_d - T_a) = .186\ (1785 - 580) = 224$ B.T.U.

$$Q_1 = \frac{C_h\ H}{30\ N} = \frac{236\ (18000)}{30\ (250)} = 567 \text{ B.T.U.}$$

$$E = \frac{Q_1 - Q_2}{Q_1} = \frac{567 - 224}{567} = \frac{343}{567} = 60\%$$

$$W = J\,(Q_1 - Q_2) = 778\,(567-224) = 778\,(343) = 266854 \text{ ft. lb.}$$

$$\text{B.T.U. per hr. per I.H.P.} = \frac{2545}{E} = \frac{2545}{.6} = 4241.6$$

$$\text{I.H.P.} = \frac{\text{P.L.A.N.}}{33000} = \frac{100 \times 1.41 \times 124.4 \times 125}{33000} = 66.4$$

$$66.4 \times 8 = 531.2 \text{ I.H.P.}$$

$$\text{Mechanical Efficiency} = \frac{500}{531.2} = 94\%$$

$$\text{Theoretical M.E.P.} = \frac{J\,(Q_1 - Q_2)}{V_d \;\; 1728} = 109 \text{ lb. per sq. in.}$$

THE DESIGN OF THE DIESEL ENGINE

SPECIFICATIONS

Brake Horse Power - - - - - - - -	500	
Revolutions per minute - - - -	250	
Number of Cylinders - - - - - - -	8	
Diameter of Cylinder - - - - - -	1.045	ft.
Length of Stroke - - - - - - -	1.41	ft.
Ratio of Diameter to Stroke - -	1.35	
B.H.P./I.H.P. - - - - - - - - - -	.80	
Specific Capacity - - - - - - - -	2880	ft./lb./cu.ft./sec.
Thermal Eff. at Brake - - - - - -	.30	
Volumetric Eff. - - - - - - - -	1.00	
Piston Speed - - - - - - - - - - -	700	ft./min.
Piston Displacement - - - - - - -	1.195	cu. ft.
Clearance Volume - - - - - - - - -	.0836	cu. ft.
M. E.P. - - - - - - - - - - - - - - -	100	lb./sq.in.
Fuel - Crude Oil - - - - - - - - -	-18,000	B.T.U./lb.
Fuel Consumed per hour - - - -	236	lbs.
Fuel Consumed per stroke - - -	.0314	lbs.
Method of cooling - - - - - - -	Water	
Method of starting - - - - - - -	Air	

DETERMINING THE PRINCIPAL DIMENSIONS OF AN ENGINE

A simple basis for determining the principal dimensions is the amount of air necessary for combustion. This method permits of taking into consideration the properties of the fuel, and requires but few assumptions and established empirical factors.

C_s = Fuel consumption per suction stroke at the rated load.

L_s = Actual amount of air required per suction stroke at the normal load, determined from C_s and L under standard conditions (29.9 in. Hg and 62 deg. fahr.) in cu. ft.

Then per cylinder end,

$$C_h = 2545 \text{ B.H.P.}/HE_e \text{ (cu.ft. or lb. per hour)}$$

$$C_s = 2C_h/60N = 84.8 \text{ B.H.P.}/E_e\, HN \text{ (cu.ft. or lb.)}$$

$$L_s = C_sL = 84.6 \text{ B.H.P.}/E_e\, HN \text{ (cu. ft.)}$$

$$C_h - 2545 \times 500/18000 \times .30 = 236 \text{ lb. per hr.}$$

$$C_s = 2 \times 236/60 \times 250 = .0314 \text{ lb. per stroke.}$$

$$L_s = .0314 \times 305 = .972 \text{ cu. ft.}$$

$$V_d = 84.8 \text{ B.H.P. } L/E_eE_s\, HN$$

$$V_d = 84.8 \times 62.5 \times 305/.30 \times 1 \times 18000 \times 250 = 1.195$$

$$d = 216 \text{ B.H.P. } L/E_sE_eHS \qquad \text{in. ft.}$$

$$d = 216 \times 62.5 \times 305/1 \times .3 \times 18000 \times 700 - 1.045 \text{ ft.}$$

$$l = 108 \text{ B.H.P. } L/E_s E_e H N d^2 \text{ in ft.}$$

$$l - 108 \times 62.5 \times 305/\ I \times .3 \times 18000 \times 250 \times \overline{1.045}^2 = 1.41$$

$$N = 108 \text{ B.H.P.}/E_e E_s H I d^2 \text{ ------ R.P.M.}$$

$$N = 108\ (62.5)(305)/I\ (.30)\ (18000)(I.41)(1.09) = 250$$

VALVES

Air inlet valves should have a mean velocity of 120 to 140 ft. per sec. $a = V/v$ sq. ft., where V equals displacement of piston, cu. ft. per sec. v equals mean velocity, ft. per sec. a equals area of valve in sq. ft., or a equals Fc/v, where c equals piston speed, ft. per sec., and F equals area of piston in sq. ft.

$$a = I.195\ (250)/60(145)(2) = .017 \text{ sq. ft.}$$

$$d_v = 5.6 \text{ in.}$$

The exhaust valves of Diesel engines are constructed with the same dimensions as the inlet valves.

The safety valve area will be 2 sq. in. and set toopen at 600 lb. per sq. in.

CYLINDERS AND LINERS

In practically all Diesel engines, the cylinder and liner are made separately. The two parts are bolted rigidly at the back,and front end having only a rubber ring joint. Thus, the liner, heated by combustion of the fuel and the cylinder casing, cooled by water circulation between them can expand and contract freely. Strains due to the impulses behind the piston are transmitted through the casing, and the liner withstands only radial stresses. The liner is made from hard, close grained cast iron, of a tensil strength of approximately 35000 lb. per sq. in. Assuming a pressure of 510 lb. per sq. in. of piston area, the pressure P on the piston would be $P = 510\ (.7854\ d^2) = 400\ d^2$, where d = diameter of cylinder in. The thickness S of the cylinder liner may be taken as $S = .07d$ in. To this should be added $\frac{1}{4}$ in. thickness of metal for reboring.

$$S = .07\ (12.55) + .25 = 1.13 \text{ in.}$$

If D = mean diameter of the cylinder jacket wall, in.; s = thickness of cylinder jacket wall in.; d = diameter of piston in., the cylinder casing or jacket wall must withstand in the direction of its axis, a pulling force of $F = 400d^2$. The cross-sectional area, $A = 3.1416\ Ds$. The stress per sq. in., $F = 400d^2/Ds$ or $s = 400d^2/3.1416DF$. Where F = 1800 lb. per sq. in. which is the desirable stress to assume $s = 0.071\ d^2/D$

$$s = 156.5\ (.071)/17 = .65 \text{ in.}$$

CYLINDER HEAD BOLTS

Constructed from wrought iron or soft steel having a tensile strength of 60000 lb. per sq. in. The size of the bolts is determined by the maximum pressure. $F = 400d^2$ plus 25% for tightening.

$F = 400\ (156.5) = 62600$ lb.

$62600/6000 = 10.45$ sq. in.

15 1 in. bolts required per cyl.

THE CYLINDER COVER

The cover is made from close grained, soft charcoal iron; it contains air-inlet, exhaust, fuel-inlet and starting valves, which are so arranged that the cooling water circulates freely around each valve housing. The thickness of the metal S of the cover is computed from the formula $S = r^2/6$, where S is the thickness of metal of the inner walls of the cover, in inches; r = radius, in., of the largest circle that can be described on the plain surface of metal existing between the different supports for valves.

$$S = (2.19)^2/6 = .8 \text{ in.}$$

THE CONNECTING ROD

A drop forging made from a very good grade of steel. Let d = mean diameter of rod, in.; D = diameter of cylinder, in.; l = distance between centers of rod, in.; m = maximum pressure, lb. per sq. in. S = stroke in. When l = 2.58 S to 3 S, $d = 0.028 \quad Dl \quad m$

$$d = 0.028 \quad 12.55\ (50)(22.6) \qquad = 2.34 \text{ in.}$$

CONNECTING ROD BOLTS

Made from tough wrought iron of such size as to have the stress not to exceed 6000 lb. per sq. in. at the beginning of the suction stroke.

Two bolts on each side 1 in. diameter giving a factor of safety of 50. This insures any possibility of the bolts failing and causing great damage to the engine.

CRANK SHAFT

The crank shaft is forged from material with an ultimate tensile strength of 80000 lb. per sq. in. Minimum; elastic limit not less than 48000 lb. per sq. in. with an elongation of 25% in 2 in., and a reduction of area not less than 45%. Where the maximum pressure does not exceed 500 lb. per sq. in; Diameter of crank shaft

$C = D^2 (AS + BL)$, where D - diameter of cylinder, in.; S - length of stroke, in. L - span of bearings adjacent to a crank, in., measured from inner edge to inner edge.

4-Cycle single-acting engines	AS + BL
8 cylinders - - - - - - - - - - - - - - - - -	.099S + .054L

The following proportions of Diesel crank shafts, built of steel of 75000 lb. per sq. in., tensile strength are recommended by the Diesel Engine Users Association (England).

Diam. of crank pin and journal	.524D	to	.54D
Length of journal	.750D	"	.80D
Length of crank pin	.524D	"	.54D
Thickness of web	.320D		

Proportions of Diesel engine crank shafts

H.P. in one cylinder	62.5
Type of shaft	Forged
Diam. of crank-pin	7.5 in.
Length of crank-pin	7.5 "
Diameter of main bearing	7.5 "
Length of main bearing	10.5 "
Diameter of fly-wheel section	8.0 "
Diameter of outboard bearing	6.5 "
Length of out-board bearing	9.0 "
Thickness of crank web	4.3 "
Width of crank web	10 "
Diameter of coupling end	7 "

BALANCING

Balancing of the reciprocating and rotating parts may be computed by use of the following formula: $W_1 = W_2 r/R$, where W_1 = weight of the balance-weight, Lb.; R = radius of the center of gravity of the balance weight, ft.; W_2 = (weight of the crank pin and the big end of the connecting rod), lb.; r = throw or radius of the crank, ft.

PISTONS

The Trunk Type

The trunk type piston is used almost exclusively in engines where the cylinder diameter is 21 in. or less. In cylinders of this size the volume of gases is such that the cooling by the surrounding water jackets will maintain proper temperatures without internal piston cooling, the pistons being made in one piece. The length of the piston should not be less than 1.6 diameter.

DIMENSIONS OF PISTONS

D - Diameter of piston

Diameter of wrist pin - - - - - - - - - - - - - - -	.3D to .4D
Diameter of wrist pin boss - - - - - - - - - - - -	.2D
Length of wrist pin bearing - - - - - - - - - - - -	.5D
Length of wrist pin bearing in piston - - - - -	.25D
Thickness of piston head - - - - - - - - - - - - -	.08D to .125D
Thickness of piston barrel (Head end) - - - - -	.09D to .ID
Thickness of piston barrel (Crank end) - - - -	.06D to .075D
Number of piston rings - - - - - - - - - - - - - -	6 to 7
Width of piston rings - - - - - - - - - - - - - -	$\frac{1}{4}$ to $\frac{1}{2}$ in.
Depth of piston ring grooves - - - - - - - - - -	$\frac{1}{2}$ to 3/4 "

Clearance, head-end tapering to rear of last piston ring (for expansion of head end) .0025 in. per in. diam, tapering to .0015 in. per in. diam. of body.

Vertical engines carry a wiper ring located below the wrist pin and also an oil catcher near the bottom of the piston. The inside of the piston head should be ribbed to allow for radiation.

Diameter of piston = 1.045 ft.

Diameter of wrist pin = .35 (12.55) = 3.83 in. D_1

Diameter of wrist pin boss = .2 (12.55) = 2.5 in. D_2

Length of wrist pin bearing = .5 (12.55) = 6.27 in. L_1

Bearing in piston (length) = .25 (12.55) = 3.18 in. L_2

Thickness of piston head = .1 (12.55) = 1.2 in. T

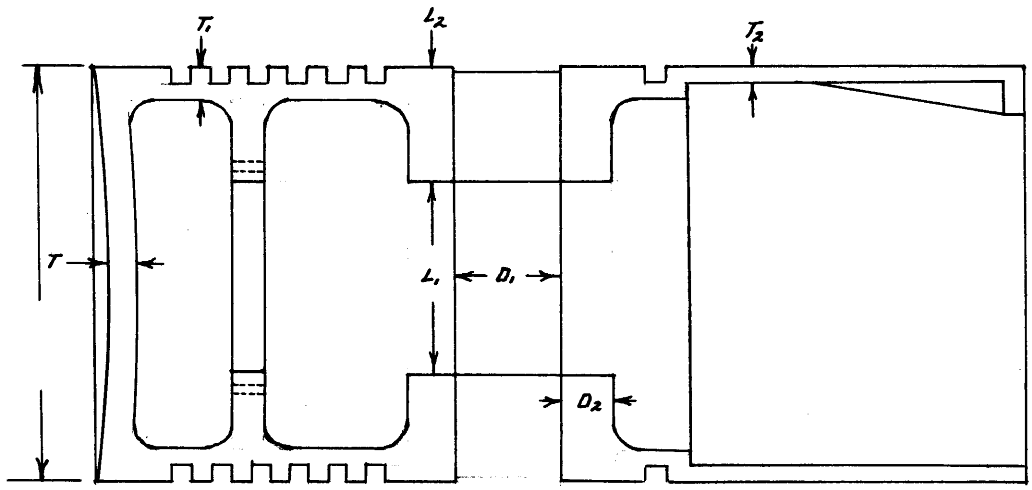
Thickness of piston barrel = .09 (12.55) = 1.13 in.(H.E.) T_1

Thickness of piston barrel = .06 (12.55) = .754 in.(C.E.) T_2

Width of rings - - - - - - - - - - $\frac{1}{2}$ in.

Depth of groove - - - - - - - - - $\frac{1}{2}$ "

Number of rings - - - - - - - - - 6



AIR COMPRESSOR

Three stage type driven from an extension on the crank shaft. On the downward stroke the air is admitted into the low pressure cylinder, and then near the end of the up stroke is discharged at a pressure of about 60 pounds through an inclined plate valve into a cooling coil, and on the next stroke, enters through a plate valve to the space swept by the difference in diameters between largest and intermediate pistons. On the return stroke this air is discharged through the plate valve in the same cage and passes through another cooling coil. On the next stroke the air is drawn in through a poppet valve and enters the space above the smallest piston. On the return stroke the highly compressed air is discharged through another poppet valve and passes through an after cooler composed of a coil of pipe surrounding the upper part of the compressor cylinder.

By dividing the compression into three stages with inter-coolers between stages, the work of compression is reduced, very small clearances are avoided, and the temperature rise in each stage is kept down to 250 or 260 degrees F.

Capacity of air compressor to be such that it will handle 24 cu. ft. of air per H.P. to be compressed to 1200 pounds per sq. in. as a maximum load.

The power required to drive the compressor will be about 4% of the rated H.P. of the engine or 20 H.P. Since the engine has an overload capacity of at least 10%, this does not interfere with its original rating.

THE FUEL PUMP

The fuel pump is built of massive proportions due to the fact that it must handle the fuel under pressures of 600 to 1000 pound pressure. Its displacement must be at least twice that required by actual consumption in order to care for leaks, overloads, and volumetric efficiency.

The best plan is to have individual pumps for each cylinder as the pressure is more easily maintained and finer adjustments can be obtained, without relying on the vaporizers to care for fuel adjustment.

SPEED REGULATION

With all stationary engines the speed variation should not exceed $1\frac{1}{4}$% on either side of the uniform speed of rotation, nor exceed 4% when the load is varied from full to $\frac{1}{4}$ load, nor 5% when varied from full to no load. In Diesel Engine the governing system most generally used alters the point in the stroke of the pump at which the suction valve contacts with its seat. At full load, the valve seats earlier and a full amount of oil is delivered to the sprayer; at lighter loads, the valve seats later and less fuel is delivered.

THE EXHAUST PIPE

The exhaust pipe between the exhaust valve and the silencer should have an area of 1.2 that of the exhaust valve, and an area equal to the exhaust valve beyond the silencer.

COOLING

Cooling arrangements necessary to carry off 33% of the total B.T.U. supplied to the engine are important. The amount of cooling varies considerable with the design of the engine but in 2-cycle Diesels, the standard practice is to assume from 8 to 10 gallons per H.P., and in 4-cycle engines from 4 to 5 gallons per H.P. With an initial temperature of 50° F. the outlet temperature will be from 140° to 160° F. If plenty of free water is at hand the ideal way to supply cooling water is by the use of a pump driven from the engine with controlling arrangements for increasing or decreasing the supply.

The centrifugal type pump is best adapted for this class of work as it will furnish water at more constant pressure with less expenditure of power.

THE EXHAUST SILENCER

Volume should be 6 to 8 times the volume of the piston displacement.

THE VAPORIZER

For high compression engines, the required volume of the vaporizer bears a certain relation to the clearance volume of the engine, expressed as:

$$V_v/V_o = (I-(P_2/400)^{I/n}) \;/\; (P_3/400)^{I/n} - (P_2/400)^{I/n}$$

in which V_o = total clearance volume of the engine; V_v = required volume of vaporizer; P_3 = maximum explosion pressure, absolute; P_2 = compression pressure, absolute; n = exponent in the equation $PV^n = K$.

Since it has been determined that engines with compression above 400 lb. per sq. in. require no method of vaporizing the fuel except as a means of decreasing the speed where the compression will fall below 400 lb. per sq. in., the vaporizers will be constructed on a ratio of

$V_v/V_o - 0.06$

$$.0836 = 0.06\,V_o$$

$$V_o = 1.39$$

VALVE GEAR

The cam shaft is constructed of machine steel and the cams from hardened steel; cam levers from soft or vast steel; driving gears from steel upon cast iron or bronze, or if well lubricated, cast iron on cast iron for screw gears; and cast iron on cast iron for spur or bevel gears. The tooth angle for the screw gears is 63 deg.

25 min. for the driving gear and 26 deg. 35 min. for the driven gear. Large cam diameters permit accurate adjustment of the valve motion but the peripheral velocity of the cam should not exceed 3 ft. per sec., as knocking against the cam rollers is likely to result and the torque of the shaft is increased. The width of the cam rollers for inlet valves should be .3 of the diameter of the roller, and for the exhaust valve .4. Strength of the valve levers should be based on a pressure at the moment of opening of 30 lb. per sq. in for the inlet and 75 lb. per sq. in. for the exhaust valve.

FIRING ORDER

1, 5, 7, 3, 6, 8, 4, 2, 6

VALVE SETTING

Inlet valve starts to open when crank is 15 degrees from head-end-dead-center (approaching).

Exhaust valve closes when crank is 12 degrees from hed-end-dead-center (leaving).

Exhaust valve starts to open when crank is 140 degrees from head-end-dead-center (leaving).

Inlet valve closes when crank is 200 degrees from head-end-dead-center (leaving).

Spray valve opens when crank is 7 degrees from head-end-dead-center)approaching).

Spray valve closes when crank is 20 degrees from head-end-dead-center (leaving).

POSITION OF CAMS

Number I piston top dead center

Firing center No. 1 Spray 360 degrees; exhaust 102 degrees; inlet 158 degrees.

No. 2 Spray 270 degrees; exhaust 12 degrees; inlet 72 degrees.

No. 3 Spray 135 degrees; exhaust 237 degrees; inlet 297 degrees; air 135 degrees.

No. 4 Spray 225 degrees; exhaust 327 degrees; inlet 27 degrees; air 225 degrees.

No. 5 Spray 45 degrees; exhaust 147 degrees; inlet 207 degrees; air 45 degrees.

No. 6 Spray 315 degrees; exhaust 57 degrees; inlet 117 degrees; air 315 degrees.

No. 7 Spray 90 degrees; exhaust 192 degrees; inlet 252 degrees.

No. 8 Spray 180 degrees; exhaust 282 degrees; inlet 342 degrees.

Cams to be of free type which enables interchanging when necessary. Slip gear attachment on driven gear of cam shaft to allow for an adjustment of 5 degrees either way from setting.

LUBRICATION

The hourly bearing oil circulation should be about .05 gallons per B.H.P. The amount of oil actually used up and not recoverable will vary between .00035 and

.00075 gallons per B.H.P. per hour with well designed oiling systems. The consumption of cylinder oil, none of which can be recovered varies between .00024and .00042 gallons per B. H.P. per hour, depending upon the care given the engine.

The type of pump used to furnish oil is the impeller type which will furnish oil at constant pressure and volume by regulating the suction capacity. The pump is driven from an extension on the crank shaft directly forward of the air compressors.

FRAME

Determinations of the thickness of walls or webs in frames on the basis of strength, give webs so thin as to give rise to molding and casting difficulties and to bad casting strains. The walls are, therefore, made thicker than necessary for the required strength. The box frame is most commonly used for vertical engines because it is especially suitable for dusty locations, and the oiling is simplified by the splash system.

The usual thickness, s, for various cylinder diameters, d, are as follows:

d (inches) =	5	7	9	11	13	15
s (inches) =	½	9/16	5/8	11/16	3/4	13/16

FOUNDATION

The reciprocating parts of combustion engines are greater and the speeds of rotation higher than in the case of steam engines, necessitating more mass to the foundation, which should always be carried down to a firm footing.

VOLUME OF MATERIAL IN FOUNDATIONS

For vertical engines with outboard bearings- 9.8 to 10.5 B.H.P. results in cu. ft.

FLY WHEELS

In four-cycle engines the inertia effects of the reciprocating weights may be neglected in fly wheel computations. The mean pressure of the suction and exhaust is only about I per cent of the mean pressure of the positive work in modern large engines with mechanically operated valves of large area, and therefore, this negative work has practically no effect on the turning moment; it has less effect upon the coefficient of fluctuation than has the frictional resistance of the reciprocating parts of whose magnitude and distribution over the four strokes of the cycle nothing is known. Sufficient accuracy is attained by considering only the negative work of the compression stroke and the positive work of the combustion and expansion stroke, neglecting the suction and exhaust strokes.

The following method of determining the fly wheel weight gives results more quickly than the graphical method and is of sufficient accuracy.

Let M = necessary fly wheel weigh for single cylinder

I.H.P. = Maximum indicated H.P.

$P_m c$ = mean indicated pressure of compression stroke lb. per sq. in.

P_m = mean effective pressure of the cycle lb. per sq. in.

k = coefficient of fluctuation

V = velocity of the center of gravity of the rim cross section, ft. per sec.

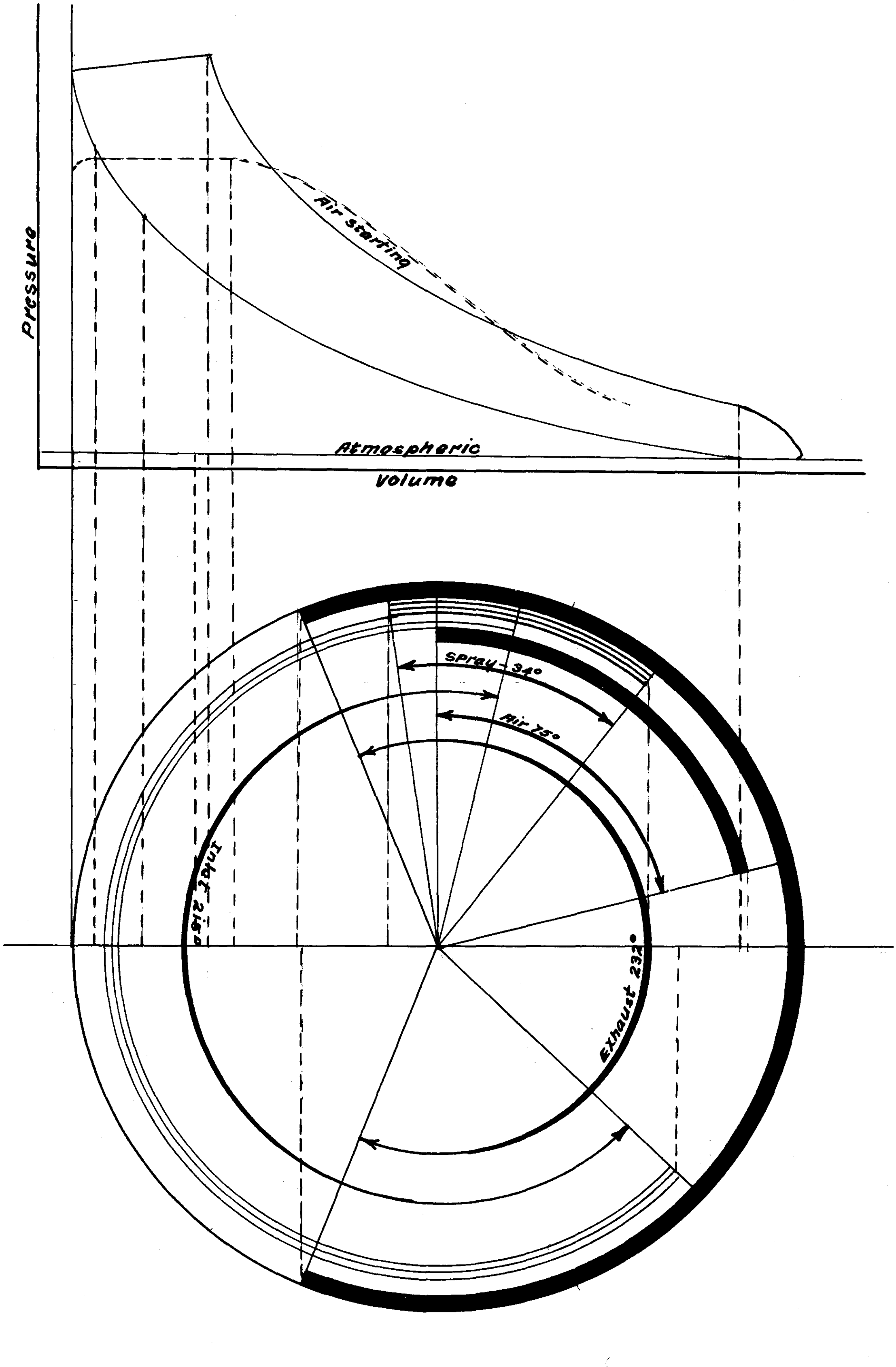
$R_W = \frac{1}{2} D_W$ = radius of the equivalent wheel = distance from center of gravity of rim cross-section to center of shaft, ft.

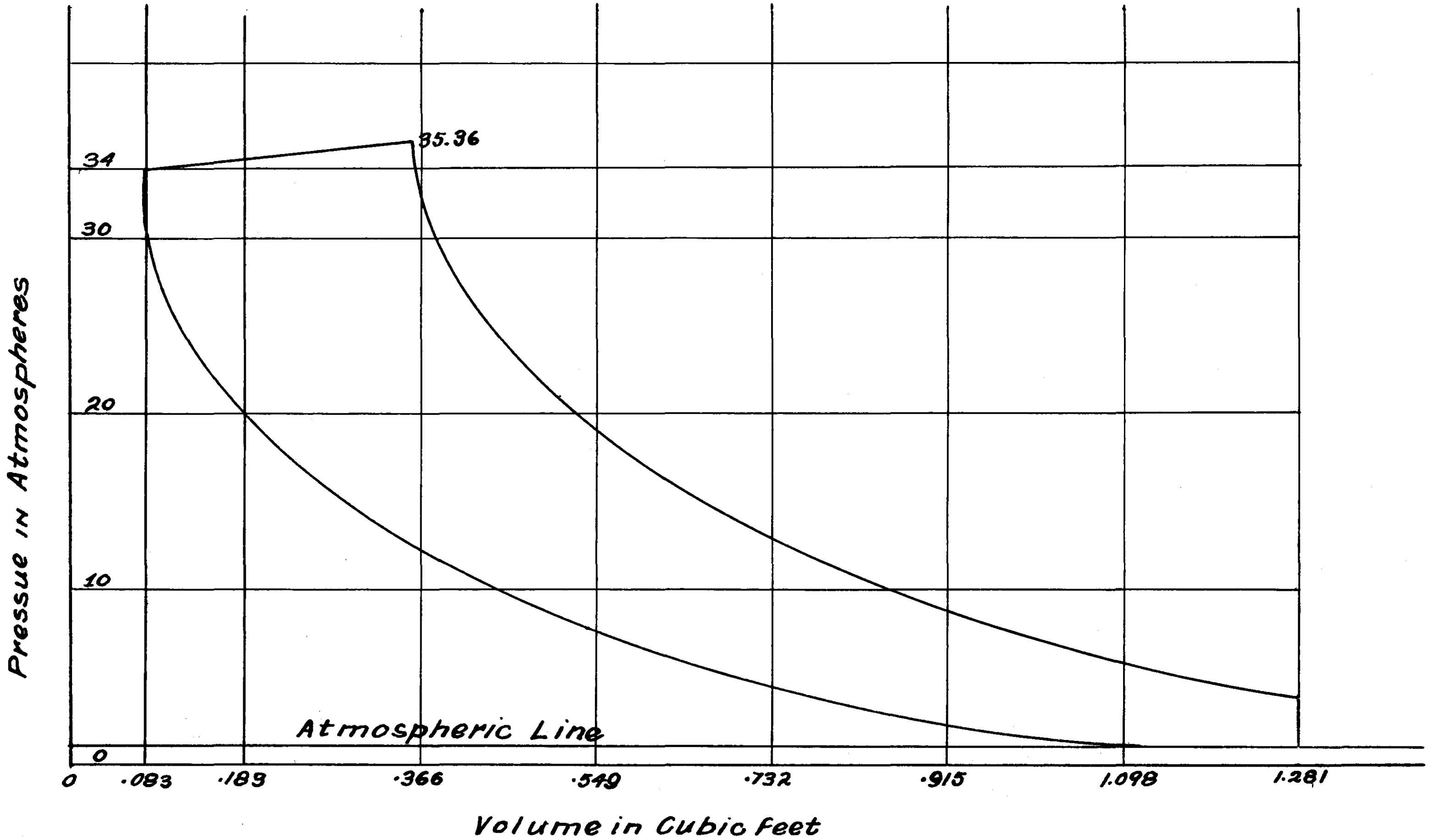
$$M - \frac{2,125,0009\ (62.5)\ (.75\ p_{mc}/p_m)}{k\ V^2\ N}$$

Value of p_{mc}/p_m =0.5

Value of k = 0.025 for ordinary power purposes

Value of V = 100 ft. per sec.


Pressure
Air starting
Atmospheric
Volume
Spray 34°
Air 75°
Inlet 215°
Exhaust 232°



This card drawn from data used in the design of a 500 H.P. Diesel

The rise in pressure during admission is due to air injection

GENERAL DESIGN

Notation

Let t and T = temperature, deg. fahr. and absolute, respectively

p = absolute pressure, lb. per sq. in.

v = volume of gas or of mixture of gases, cu. ft.

c_p and c_v = specific heats at constant pressure and constant volume

$x = c_p/c_v$ = ratio of specific heats

$J = 1/A = 778$ = mechanical equivalent of heat

$R = 778\ (c_p - c_v)$ = gas constant. For air, $R = 53.2$

$M = 144\ pv/RT$ = weight of volume v of gas

Q = B.t.u. added or abstracted during a given operation

W = work in ft.-lb. done during this operation

n = exponent of the polytropic curve pv^n = const.

$r_c = v_1/v_2$ = volumetric compression ratio (see Figs. 1 and 2)

$r_d = v_3/v_2$ = constant pressure ratio, ratio of cut-off volume to clearance volume, in the case of constant-pressure combustion

$r_e = v_1/v_3$ = ratio of expansion

b.h.p.$_n$ = rated brake horse power of an engine, usually 0.85 of the maximum power

C = fuel consumption in cu. ft. or lb. per b.h.p.$_n$ per hour

C_h = total fuel consumption per hour at normal load in cu. ft. or lb.

L = volume of air actually in the mixture per cu. ft. of gas or per lb. of fuel, in cu. ft.

H = lower heat value of 1 cu. ft. or 1 lb. of the fuel under standard conditions. (29.9 in. of mercury and 62 deg. fahr.)

d = piston diam., ft.

l = stroke, ft.

v_d = piston displacement per stroke, cu. ft.

N = revolutions of the crank shaft per minute

p_m = indicated mean effective pressure, lb. per sq. in.

E_t = theoretical indicated thermal efficiency

E_i = actual indicated thermal efficiency

E_m = b.h.p./i.h.p. = mechanical efficiency

GENERAL DESIGN (Continued)

Notation

E_s = volumetric efficiency of the suction stroke (taken at 29.9 in. of mercury and 62 deg. fahr.)

$E_c = E_i/E_t$ = ratio between the indicated and theoretical thermal efficiencies = diagram factor

$E_e = E_c E_t E_m$ = economic efficiency = thermal efficiency at the brake

Unexplained suffixes refer to points on Figs. 4 and 5.

MECHANICAL EFFICIENCIES

(Lucke, "Thermodynamics")

Type of engine	Mech. efficiency	
	4-cycle	4-cycle
Small high-speed auto, multi-cylinder, single-acting	0.75	......
Small single-cylinder, boat engine, single-acting	0.85	0.68
Small or medium single-cylinder, stationary, single-acting	0.87	0.70
Small or medium 2-cylinder, stationary, single-acting	0.84	......
Small or medium 3-cylinder, stationary, single-acting	0.82	......
Small or medium 4-cylinder, stationary, single-acting	0.80	......
Large single-cylinder, stationary, single-acting	0.90	0.70
Large 2-cylinder, stationary, single-acting	0.86	to
Large 4-cylinder, stationary, single-acting	0.84	0.80
Large single-cylinder, stationary, double-acting	0.83	0.75
Tandem 2-cylinder, stationary, double-acting	0.81	0.73
Tandem twin 4-cylinder, stationary, double-acting	0.77	0.69

INFLUENCE OF THE HEIGHT ABOVE SEA LEVEL ON THE VOLUMETRIC SUCTION EFFICIENCY

(Atmospheric pressure as given by average barometer reading b)

Height above sea level, ft.	Atmos. pressure, in. of mercury (b)	Relative volumetric efficiency (E_s), b : 29.92	Height above sea level, ft.	Atmos-pressure, in. of mercury (b)	Relative volumetric efficiency (E_s), b : 29.92
0	29.92	1.000	4,000	25.85	0.865
500	29.41	0.984	4,500	25.37	0.848
1,000	28.85	0.965	5,000	24.92	0.833
1,500	28.33	0.948	5,500	24.46	0.818
2,000	27.82	0.931	6,000	24.00	0.803
2,500	27.31	0.914	8,000	22.17	0.742
3,000	26.82	0.892	10,000	20.34	0.681
3,500	26.35	0.882			

VALUES OF COMPRESSION PRESSURES, TEMPERATURES AND THEORETICAL THERMAL EFFICIENCIES

(p_s taken at 12.5 lb. per sq. in. abs., and T_s at 700 deg. fahr. (t_s = 240 deg. fahr.)

	r_c =	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0	12.0	15.0
n = 1.30	pc=	63.7	75.8	88.3	101.3	128.4	156.9	186.6	217.5	249.4	316.1	422.5
$1-\frac{1}{r_c^{n-1}}$	=Et=	0.313	0.340	0.363	0.383	0.416	0.442	0.464	0.483	0.499	.....	.....
	Tc=	1019	1061	1099	1134	1198	1255	1306	1353	1397	1475	1577
n = 1.35	pc=	67.8	81.2	95.2	109.8	140.4	172.9	207.1	242.7	279.8	357.9	483.8
	Et=	0.355	0.384	0.409	0.431	0.466	0.494	0.517	0. 57	0.553	.....	.....
	Tc=	1085	1137	1185	1230	1311	1383	1449	1510	1567	1670	1806
n = 1.41	pc=	73.1	88.3	104.2	120.9	156.4	194.3	234.6	276.9	321.3	415.5	569.1
	Et=	0.402	0.434	0.460	0.483	0.520	0.550	0.574	0.594	0.611	.....	.....
	Tc=	1170	1236	1297	1354	1459	1554	1642	1723	1799	1939	2125

VALUES OF K IN FORMULA FOR CYLINDER DIAMETER

(Table based on a value of $E_s = 0.85$)

Fuel	Lower heat value (H) B.t.u. per		Air required-cu. ft. Theoretically (L_o) per		Air required-cu. ft. Actually, to give best results (L) per		K
	cu.ft.	lb.	cu.ft.	lb.	cu.ft.	lb.	
Natural gas	950	....	9.4	....	14	....	4.01
Illuminating gas	600	....	5.5	....	8	....	3.81
Coke-oven gas	580	....	5.4	....	7.8	....	3.86
Producer gas	135	....	0.99	....	1.3	....	4.33
Blast-furnace gas	100	....	0.73	....	1.1	....	5.34
Gasoline	.....	20,500	.....	189	......	300	3.72
Kerosene	.....	20,300	.....	187	......	300	3.76
Alcohol, 90% by vol.	.....	10,900	.....	101	......	165	3.85
Crude oil	.....	18,000	.....	176	......	305	4.31

THERMAL BRAKE EFFICIENCIES OF DIFFERENT TYPES OF ENGINES

Type of Engine	Ee, per cent.
Small high-speed auto, multi-cylinder, single-acting	15.0 to 27.5
Small or medium, single cylinder, stationary, single-acting	10.0 to 17.5
Small or medium, two-cylinder, stationary, single-acting	15.0 to 20.0
Small or medium three-cylinder, stationary, single-acting	20.0 to 27.5
Large single-cylinder, stationary, single-acting	17.5 to 25.0
Large two-cylinder, stationary, single-acting	17.5 to 25.0
Large single-cylinder, stationary, double acting	20.0 to 25.0
Double-acting tandem, two-cylinder	20.0 to 30.0
Double-acting twin tandem, four-cylinder	20.0 to 27.5

COEFFICIENTS OF FLUCTUATION

For ordinary power purposes	1/30 to 1/40
For direct-current generators, direct-connected	1/100 to 1/120
For direct-current generators, belt-driven	1/70 to 1/80
For alternating-current generators, direct-connected	1/175 to 1/200
For alternating-current generators, belt-driven	1/125 to 1/150

CAPACITY COEFFICIENTS FOR SINGLE-ACTING FOUR CYCLE ENGINES

Fuel, assuming $E_m = 80$ per cent	M.e.p. p_m lb. per sq. in.	Specific capacity $W_o = \frac{144 p_m E_m}{4}$ ft.-lb. per cu. ft. per sec.	Specific displacement $V_o = \frac{550}{W_o}$ cu.ft. per cu.ft. per sec.	Capacity constant $K_c = \frac{p_m E_m}{168,000}$	Relative capacity referred to illuminating gas engine = 1.00
Natural gas	85.0	2450	0.224	0.000405	1.10
Illuminating gas	77.5	2230	0.247	0.000369	1.00
Coke-oven gas	77.5	2230	0.247	0.000369	1.00
Producer gas	62.5	1800	0.306	0.000298	0.81
Blast-furnace gas	57.5	1660	0.331	0.000274	0.74
Gasoline	75.0	2160	0.255	0.000357	0.97
Kerosene	55.0	1580	0.348	0.000262	0.71
Alcohol	55.0	1580	0.348	0.000262	0.71
Crude oil (Diesel)	100.0	2880	0.191	0.000477	1.29

USUAL COMPRESSION PRESSURES FOR INTERNAL COMBUSTION ENGINES

Fuel	Types of engines	Range of compression, lb. per sq. in. gage	Average compression in lb. per sq. in. gage
Gasoline	Automobile	45-95	65
Gasoline	Stationary	60-105	70
Kerosene	Hot bulb, 250-500 r.p.m.	30-75	60
Kerosene	Vaporized before entering cylinder	45*-85†	65
Alcohol	Vaporized before entering cylinder	120-210	150
Fuel oil	Injected into hot bulb before compression - Hornsby-Akroyd	45	45
Fuel oil	Injected after compression	255	255
Fuel oil	Diesel cycle	510	510
Natural gas . . .	Medium and large engines ‡	75-160	120
Coke-oven gas . .	Large engines (in Germany)	105-135	120
Coal gas	Mostly small, very few large engines	75-120	100
Carburetted water gas	Mostly small, very few large engines	75-105	90
Producer gas	Both large and small engines	100-160	130
Blast-furnace gas	Largest engines built	120-190	155

* With hot mixture without water injection. † With water injection

‡ Wide variation due to variation in composition of natural gas from various localities.

HYPERBOLIC LOGARITHMS

These pages give the natural (hyperbolic, or Napierian) logarithms ($\log_e$) of numbers between 1 and 10, correct to four places. Moving the decimal point n places to the right (or left) in the number is equivalent to adding n times 2.3026 (or n times 3.6974 to the logarithm. Base e = 2.71828 +

n	n(2.3026)	n(0.6974-3)
1	2.3026	0.6974-3
2	4.6052	0.3948-5
3	6.9078	0.0922-7
4	9.2103	0.7897-10
5	11.5129	0.4871-12
6	13.8155	0.1845-14
7	16.1181	0.8819-17
8	18.4207	0.5793-19
9	20.7233	0.2767-21

Number	0	1	2	3	4	5	6	7	8	9	Ave. diff.
1.0	0.0000	0100	0198	0296	0392	0488	0583	0677	0770	0862	95
1.1	0953	1044	1133	1222	1310	1398	1484	1570	1655	1740	87
1.2	1823	1906	1989	2070	2151	2231	2311	2390	2469	2546	80
1.3	2624	2700	2776	2852	2927	3001	3075	3148	3221	3293	74
1.4	3365	3436	3507	3577	3646	3716	3784	3853	3920	3988	69
1.5	0.4055	4121	4187	4253	4318	4383	4447	4511	4574	4637	65
1.6	4700	4762	4824	4886	4947	5008	5068	5128	5188	5247	61
1.7	5306	5365	5423	5481	5539	5596	5653	5710	5766	5822	57
1.8	5878	5988	6043	6043	6098	6152	6206	6259	6313	6366	54
1.9	6419	6471	6523	6575	6627	6678	6729	6780	6831	6881	51
2.0	0.6931	6981	7031	7080	7129	7178	7227	7275	7324	7372	49
2.1	7419	7467	7514	7561	7608	7655	7701	7747	7793	7839	47
2.2	7885	7930	7975	8020	8065	8109	8154	8198	8242	8286	44
2.3	8329	8372	8316	8459	8502	8544	8587	8629	8671	8713	43
2.4	8755	8796	8838	8879	8920	8961	9002	9042	9083	9123	41

HYPERBOLIC LOGARITHMS (Continued)

Num-ber	0	1	2	3	4	5	6	7	8	9	Ave. diff.
2.5	0.9163	9203	9243	9282	9322	9361	9400	9439	9478	9517	39
2.6	9555	9594	9632	9670	9708	9746	9783	9821	9858	9895	38
2.7	0.9933	9969	*0006	*0043	*0080	*0116	*0152	*0188	*0225	*0260	36
2.8	1.0296	0332	0367	0403	0438	0473	0508	0543	0578	0613	35
2.9	0647	0682	0716	0750	0784	0818	0852	0886	0919	0953	34
3.0	1.0986	1019	1053	1086	1119	1151	1184	1217	1249	1282	33
3.1	1314	1346	1378	1410	1442	1474	1506	1537	1569	1600	32
3.2	1632	1663	1694	1725	1756	1787	1817	1848	1878	1909	31
3.3	1939	1969	2000	2030	2060	2090	2119	2149	2179	2208	30
3.4	2238	2267	2296	2326	2355	2384	2413	2442	2470	2499	29
3.5	1.2528	2556	2585	2613	2641	2669	2698	2726	2754	2782	28
3.6	2809	2837	2865	2892	2920	2947	975	3002	3029	3056	27
3.7	3083	3110	3137	3164	3191	3218	3244	3271	3297	3324	27
3.8	3350	3376	3403	3429	3455	3481	3507	3533	3558	3584	26
3.9	3610	3635	3661	3686	3712	3737	3762	3788	3813	3838	25
4.0	1.3863	3888	3913	3938	3962	3987	4012	4036	4061	4085	25
4.1	4110	4134	4159	4183	4207	4231	4255	4279	4303	4327	24
4.2	4351	4375	4398	4422	4446	4469	4493	4516	4540	4563	23
4.3	4586	4609	4633	4656	4679	4702	4725	4748	4770	4793	23
4.4	4816	4839	4861	4884	4907	4929	4951	4974	4996	5019	22
4.5	1.5041	5063	5085	5107	5129	5151	5173	5195	5217	5239	22
4.6	5261	5282	5304	5326	5347	5369	5390	5412	5433	5454	21
4.7	5476	5497	5518	5539	5560	5581	5602	5623	5644	5665	21
4.8	5686	5707	5728	5748	5769	5790	5810	5831	5851	5872	20
4.9	5892	5913	5933	5953	5974	5994	6014	6034	6054	6074	20

HYPERBOLIC LOGARITHMS (Continued)

Number	0	1	2	3	4	5	6	7	8	9	Ave. diff.
5.0	1.6094	6114	6134	6154	6174	6194	6214	6233	6253	6273	20
5.1	6292	6312	6332	6351	6371	6390	6409	6429	6448	6467	19
5.2	6487	6506	6525	6544	6563	6582	6601	6620	6639	6658	19
5.3	6677	6696	6715	6734	6752	6771	6790	6808	6827	6845	18
5.4	6864	6882	6901	6919	6938	6956	6974	6993	7011	7029	18
5.5	1.7074	7066	7084	7102	7120	7138	7156	7174	7192	7210	18
5.6	7228	7246	7263	7281	7299	7317	7334	7352	7370	7387	18
5.7	7405	7422	7440	7457	7475	7492	7509	7527	7544	7561	17
5.8	7579	7596	7613	7630	7647	7664	7681	7699	7716	7733	17
5.9	7750	7766	7783	7800	7817	7834	7851	7867	7884	7901	17
6.0	1.7918	7934	7951	7967	7984	8001	8017	8034	8050	8066	16
6.1	8083	8099	8116	8132	8148	8165	8181	8197	8213	8229	16
6.2	8245	8262	8278	8294	8310	8326	8342	8358	8374	8390	16
6.3	8405	8421	8437	8453	8469	8485	8500	8516	8532	8547	16
6.4	8563	8579	8594	8610	8625	8641	8656	8672	8687	8703	15
6.5	1.8718	8733	8749	8764	8779	8795	8810	8825	8840	8856	15
6.6	8871	8886	8901	8916	8931	8946	8961	8976	8991	9006	15
6.7	9021	9036	9051	9066	9081	9095	9110	9125	9140	9155	15
6.8	9169	9184	9199	9213	9228	9242	9257	9272	9286	9301	15
6.9	9315	9330	9344	9359	9373	9387	9402	9416	9430	9445	14
7.0	1.9459	9473	9488	9502	9516	9530	9544	9559	9573	9587	14
7.1	9601	9615	9629	9643	9657	9671	9685	9699	9713	9727	14
7.2	9741	9755	9769	9782	9796	9810	9824	9838	9851	9865	14
7.3	1.9879	9892	9906	9920	9933	9947	9961	9974	9988	*0001	13
7.4	2.0015	0028	0042	0055	0069	0082	0096	0109	0122	0136	13

HYPERBOLIC LOGARITHMS (Continued

Number	0	1	2	3	4	5	6	7	8	9	Ave. diff
7.5	2.0149	0162	0176	0189	0202	0215	0229	0242	0255	0268	13
7.6	0281	0295	0308	0321	0334	0347	0360	0373	0386	0399	13
7.7	0412	0245	0438	0451	0464	0477	0490	0503	0516	0528	13
7.8	0541	0554	0567	0580	0592	0605	0618	0631	0643	0656	13
7.9	0669	0681	0694	0707	0719	0732	0744	0757	0769	0782	12
8.0	2.0794	0807	0819	0832	0844	0857	0869	0882	0894	0906	12
8.1	0919	0931	0943	0956	0968	0980	0992	1005	1017	1029	12
8.2	1041	1054	1066	1078	1090	1102	1114	1126	1138	1150	12
8.3	1163	1175	1187	1199	1211	1223	1235	1247	1258	1270	12
8.4	1282	1294	1306	1318	1330	1342	1353	1365	1377	1389	12
8.5	2.1401	1412	1424	1436	1448	1459	1471	1483	1494	1506	12
8.6	1518	1529	1541	1552	1564	1576	1587	1599	1610	1622	12
8.7	1633	1645	1656	1668	1679	1691	1702	1713	1725	1736	11
8.8	1748	1759	1770	1782	1793	1804	1815	1827	1838	1849	11
8.9	1872	1872	1883	1894	1905	1917	1928	1939	1950	1961	11
9.0	2.1972	1983	1994	2006	2017	2028	2039	2050	2061	2072	11
9.1	2083	2094	2105	2116	2127	2138	2148	2159	2170	2181	11
9.2	2192	2203	2214	2225	2235	2246	2257	2268	2279	2289	11
9.3	2300	2311	2322	2332	2343	2354	2364	2375	2386	2396	11
9.4	2407	2418	2428	2439	2450	2460	2471	2481	2492	2502	11
9.5	2.2513	2523	2534	2544	2555	2565	2576	2586	2597	2607	10
9,6	2618	2628	2638	2649	2659	2670	2680	2690	2701	2711	10
9.7	2721	2732	2742	2752	2762	2773	2783	2793	2803	2814	10
9.8	2824	2834	2844	2854	2865	2875	2885	2895	2905	2915	10
9.9	2925	2935	2946	2956	2966	2976	2986	2996	3006	3016	10
10.0	2.3026										

REQUIRED MATERIAL

II5 Feet I/8" high pressure copper tubing

20 Feet 3/4" high pressure copper tubing

38 Feet I I/2" Double strength copper tubing

I35 Feet I/2" high pressure copper tubing

I68 FeetI I/4" standard wrought iron pipe

I4 Unions I I/4"

38 Elbows I I/4"

22 Nipples I I/4"

9 Nipples I/2" high pressure copper

9 Packing glands I/2" high pressure copper

9 Unions I/2" high pressure copper

I6 Nipples I/8" high pressure copper

I6 Connectors I/8" high pressure copper

208- II/I6"x 6" Bolts and nuts (standard)

36-I I/4"x 6 I/2" Studs and nuts (steel)

II2- I"x I4" Studs and nuts (steel)

96- I"x 6 I/2" Bolts and nuts (standard)

24- 7/8"x 6" Bolts and nuts (standard)

84- 7/I6"x 4" Cap screw (standard)

68- I/2"x 2" Bolts and nuts (standard)

I2- I/2"x 9" Studs and nuts (steel)

208- 3/8"x I" Cap screws (standard)

64- 3/4"x 2 I/2" Studs and nuts (steel)

30- I/2"x 2" Bolts and nuts (standard)

28- 3/8"x I" Set screws (standard)

OIL and WATER PUMPS

While the design of the crank case is such as to permit the oil and water pumps to be driven by the engine this feature is not at all desirable due to the fact that it is not possible to circulate water or oil through the engine when not in operation. Due to the rather massive construction of the Diesel type engine it is almost necessary to circulate water through the engine for a few minutes after shuting down, therefore the drive on the water pump should be from some other source. It is well after the engine has been idle for some time to circulate oil through it for a few minutes before starting up, which makes it desirable to equip the oil pump with some motive power other than that furnished by the engine.

Expansion tanks are sometimes used to cool the engine down after it is stopped. This tank is placed in a position above the engine and acts as a radiator in cooling the engine there being two pipes leading from this tank to the engine, the hot water pipe leading from the upper most part of the engine to some position well toward the top of the tank, the cold water pipe leading from some low point on the engine to the bottom of the tank.

A hand pump is sometimes installed in the oil line to supply oil to the bearings and working parts of the engine when an engine driven oil pump is used.

INDEX OF PLATES

B I B L I O G R A P H Y

Elements of Engineering Thermodynamics

James A. Moyer	Director of the Massachusetts Department of University Extension
James P. Calderwood	Professor of Mechanical Engineering, Kansas State Agricultural College.
Andrey A. Potter	Dean of Engineering, Purdue University.

Engineering Thermodynamics

Charles Edward Lucke	Professor of Mechanical Engineering, Columbia University.

Internal Combustion Engines

Robert L. Streeter	Formerly Professor of Steam and Gas Engine Design, Russell Sage Foundation, Rensselaer Polytechnic Institute, Mechanical Engineer, Aluminum C o. of America.

American Machinists Hand Book

Fred H. Colvin	Associate Editor of the American Machinist
Frank A. Stanley,	Associate Editor of the American Machinist.

Vega Logarithms

Baron Von Vega.

Translated from the Fortieth or Dr. Bremiker's, by W.L.F. Fischer, M.A., F.R.S. Fellow of Clare College.

Marks Mechanical Engineering Hand Book

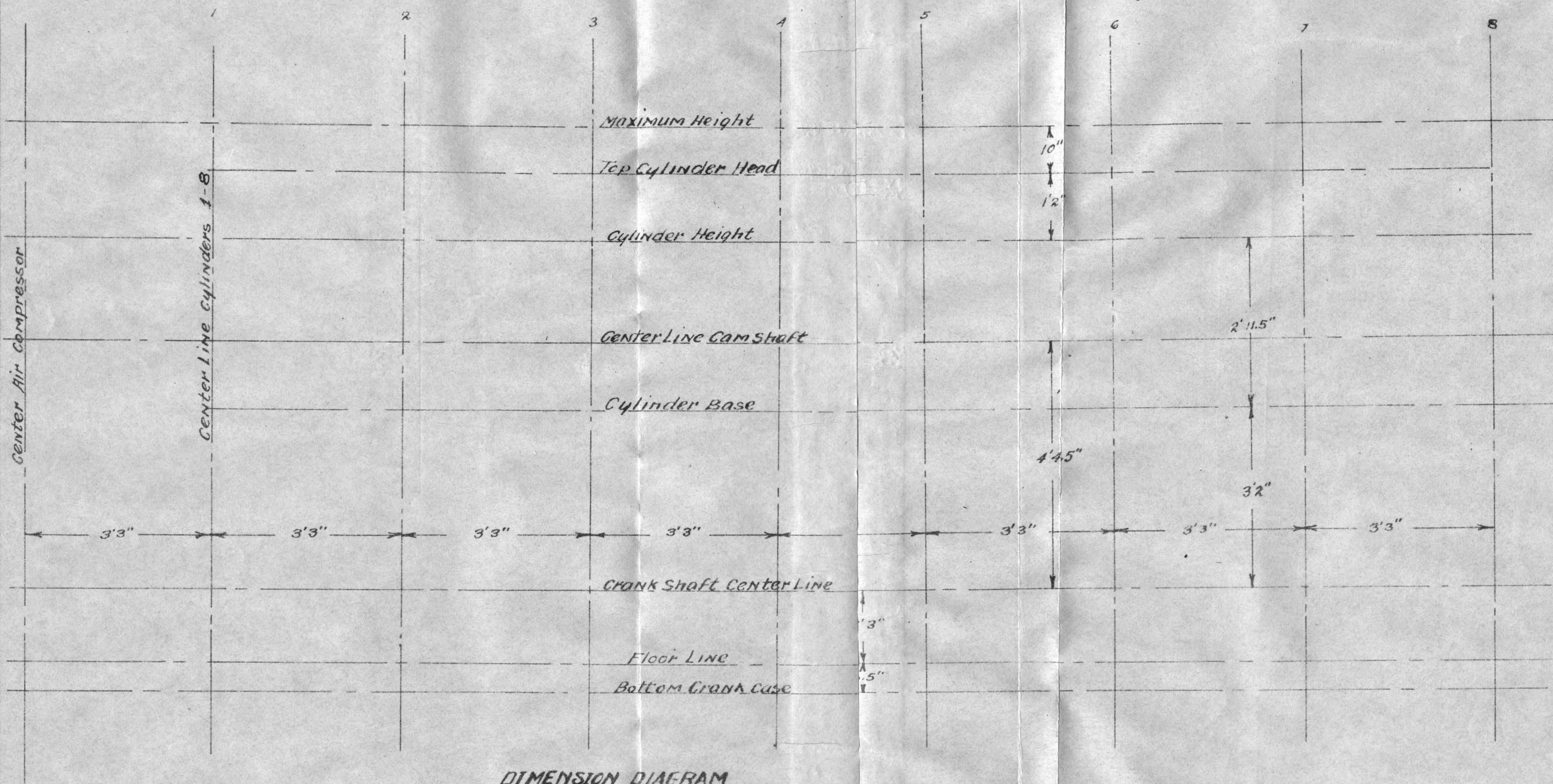
Professor of Mechanical Engineering, Harvard University, and Massachusetts Institute of Technology.

Kents Mechanical Engineers Hand Book

William Kent, M.E., Sc. D.

Mechanical Engineering

Published by the American Society of Mechanical Engineers.



DIMENSION DIAGRAM

Total height necessary to permit removal of cylinder liners 10 ft. 4 in.

Total Length of engine 33 ft. 9 in.

Engine should be bolted to steel channels set in concrete with center seal to allow for expansion.

Maximum width not over 6 ft.

PLATE - 1

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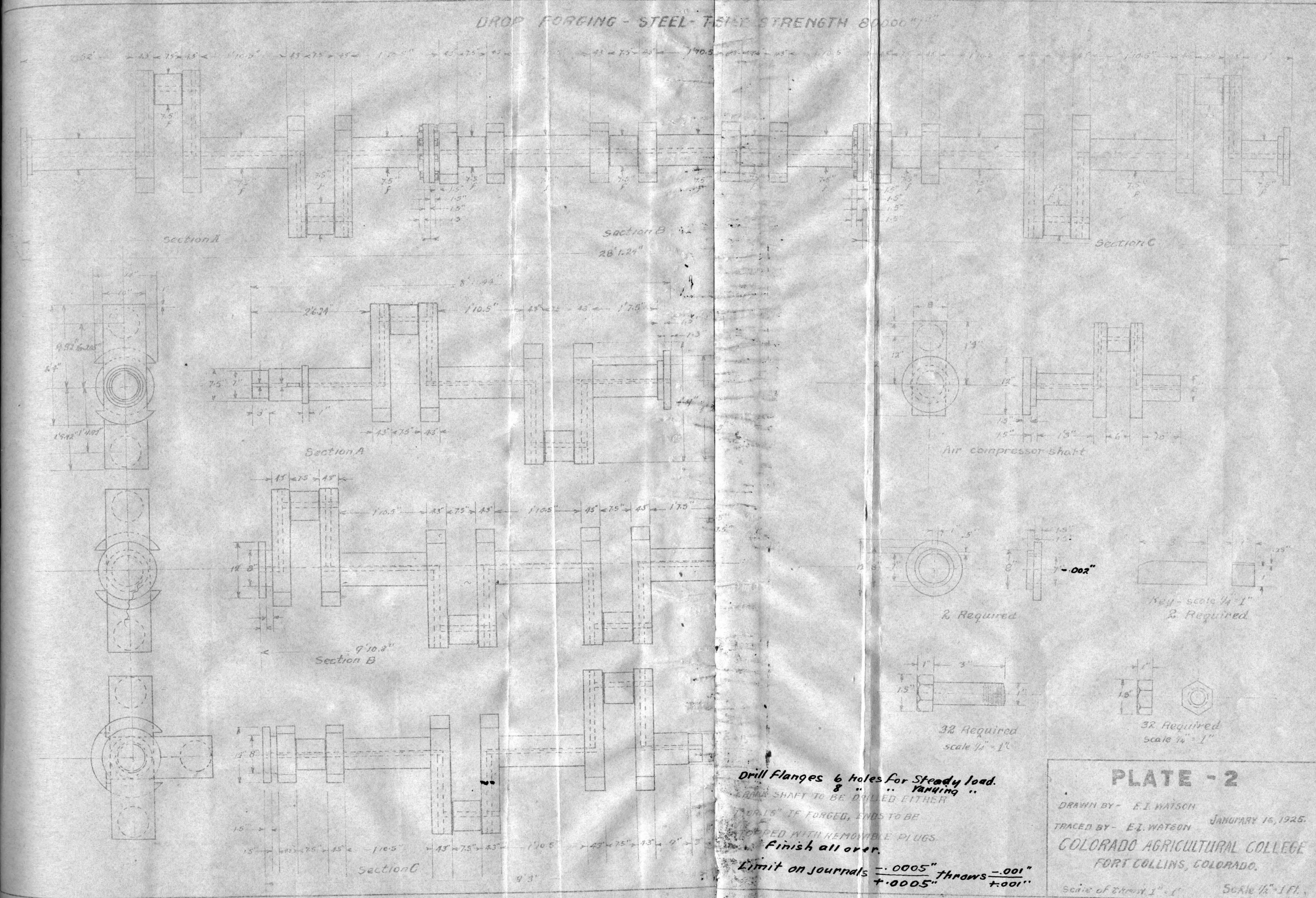
TRACED BY E.I. WATSON

COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO
SCALE 1/2" = 1'


DROP FORGING - STEEL - T. S. STRENGTH 80,000 #/□"
Section A
Section B
28' 1.24"
Section C
Section A
Air compressor shaft
Section B
2 Required
Key - scale 1/4" - 1"
2 Required
Section C
32 Required
scale 1/4" = 1"
32 Required
Scale 1/4" = 1"
Drill flanges 6 holes for Steady load.
8 " " Varying "
CRANK SHAFT TO BE DRILLED EITHER
1" OR 1.5" IF FORGED, ENDS TO BE
STOPPED WITH REMOVABLE PLUGS.
Finish all over.
Limit on journals -.0005" / +.0005" throws -.001" / +.001"
PLATE - 2
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TRACED BY - E.I. WATSON
JANUARY 16, 1925.
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
Scale of Throw 1" - 1'
Scale 1/2" - 1 Ft.

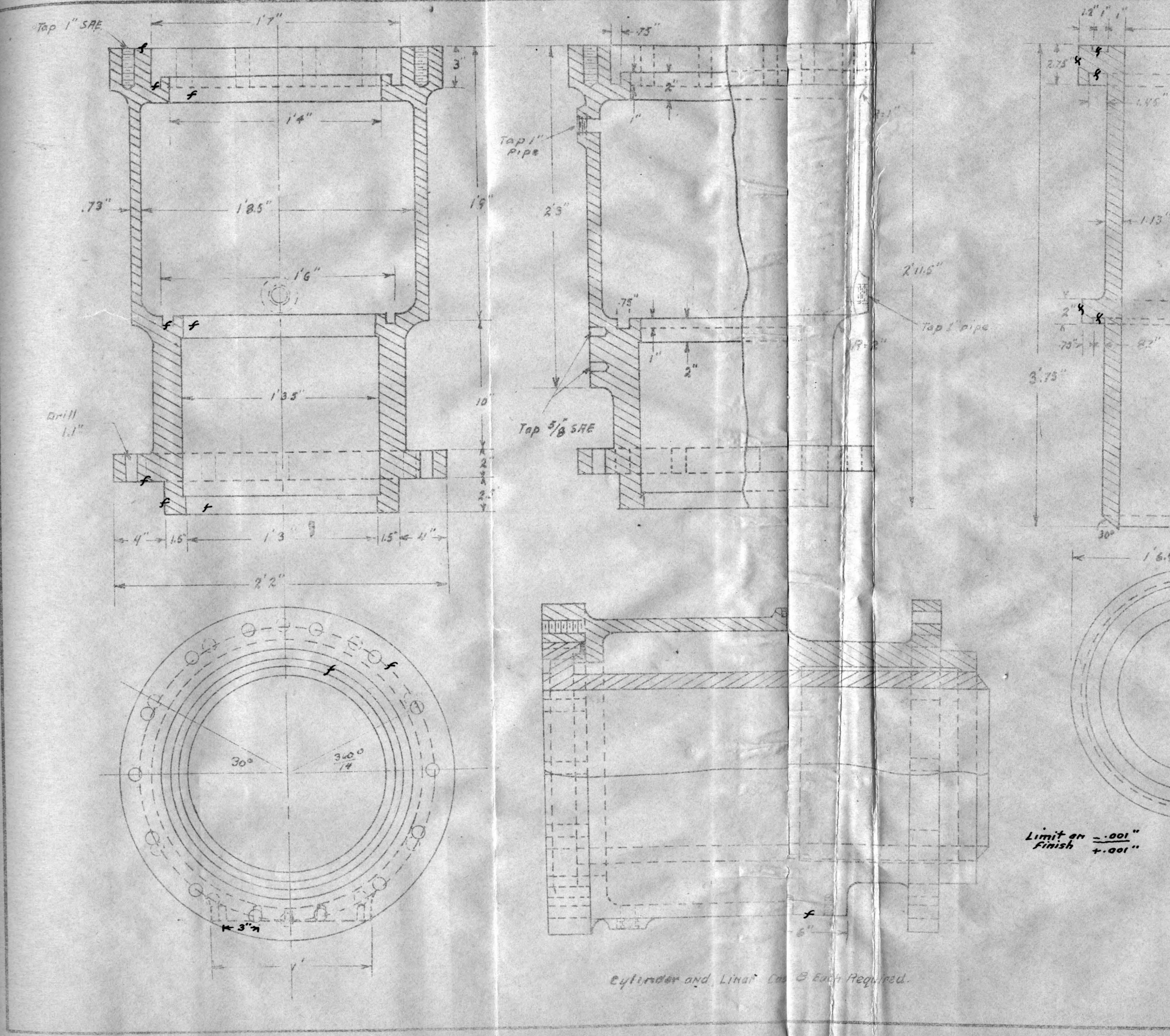

PLATE - 3

DRAWN BY E.I.WATSON
TRACED BY E.I.WATSON
JANUARY 16, 26

COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE 1.5" = 1 FT.

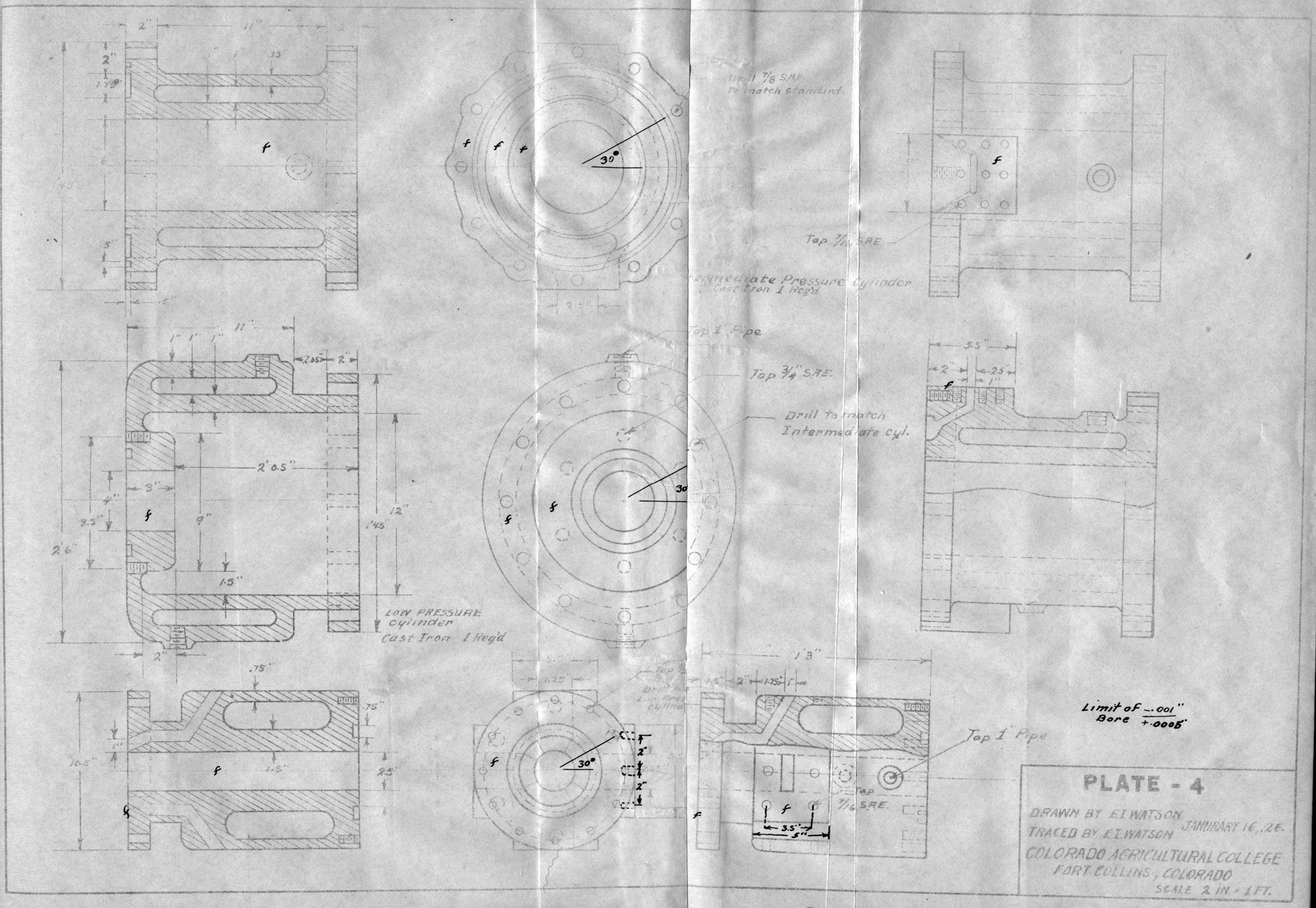

PLATE - 4

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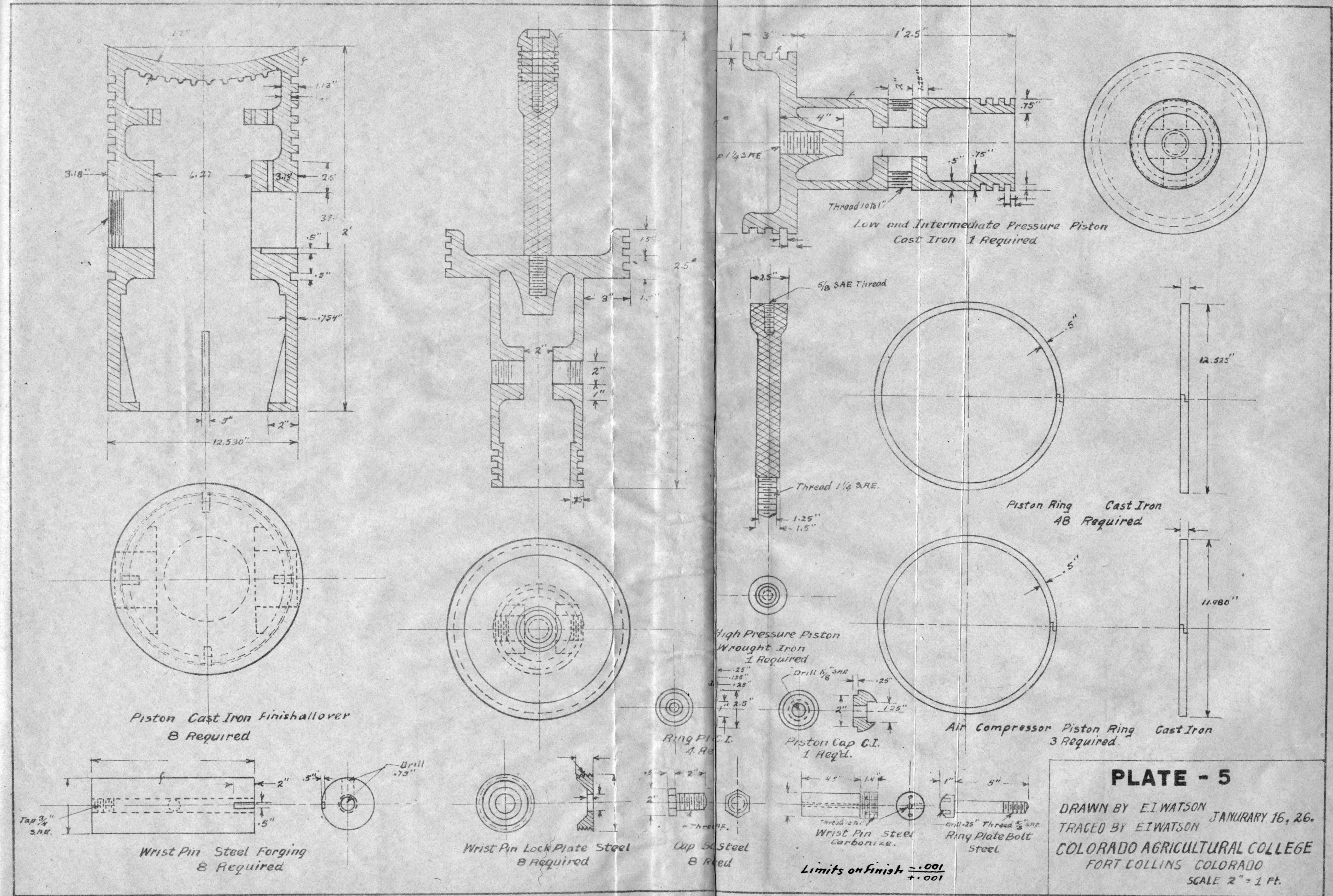
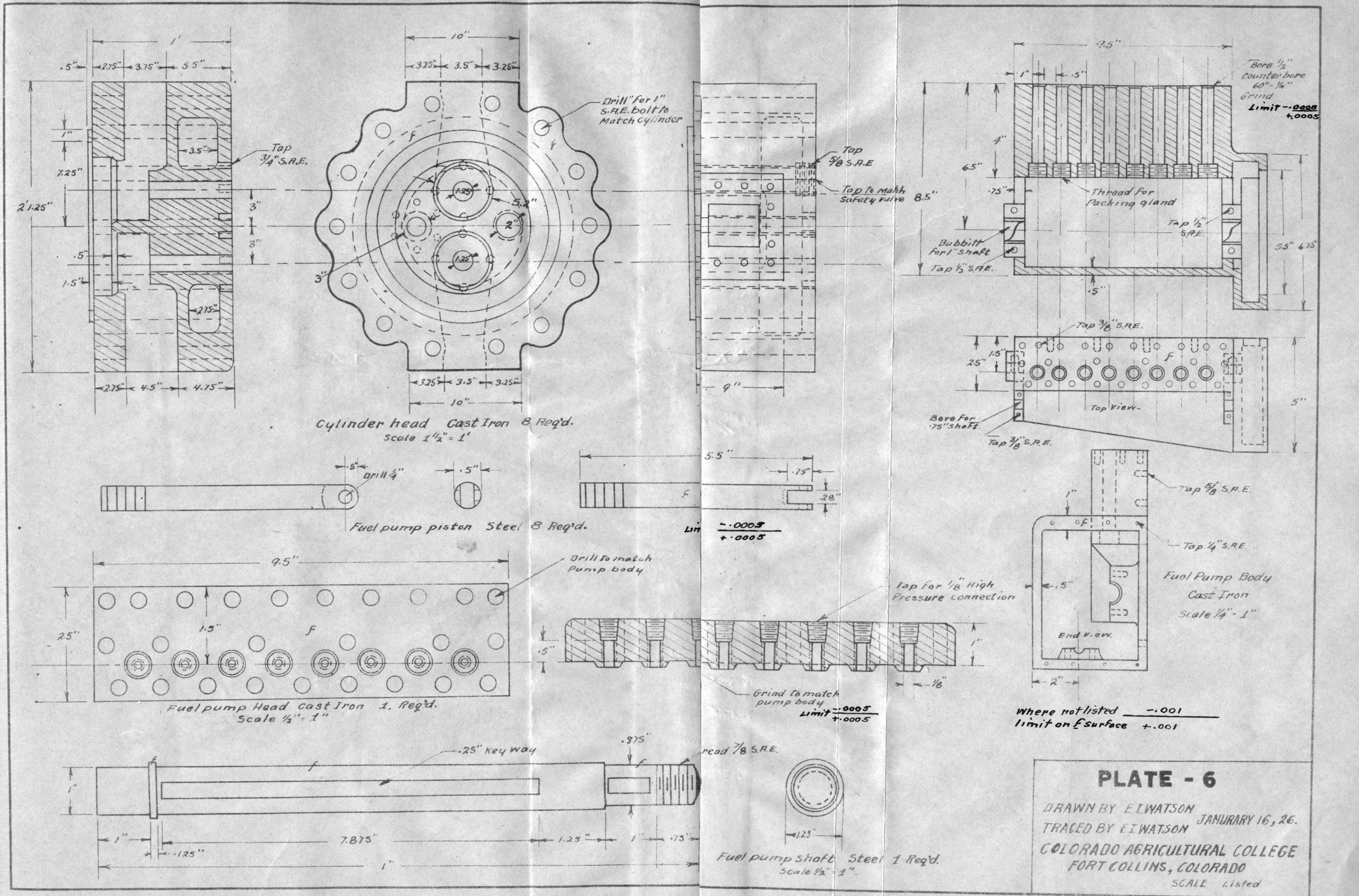
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO
SCALE 2 IN. = 1 FT.



Piston Cast Iron Finishallover
8 Required
Low and Intermediate Pressure Piston
Cast Iron 1 Required
High Pressure Piston
Wrought Iron
1 Required
5/8 SAE Thread
Thread 1 1/4 SAE.
Piston Ring Cast Iron
48 Required
Air Compressor Piston Ring Cast Iron
3 Required.
Piston Cap C.I.
1 Reqd.
Wrist Pin Steel Forging
8 Required
Wrist Pin Lock Plate Steel
8 Required
Wrist Pin Steel
Carbonize.
Ring Plate Bolt
Steel
Limits on Finish $\frac{-.001}{+.001}$
PLATE - 5
DRAWN BY E.I. WATSON
TRACED BY E.I. WATSON
JANURARY 16, 26.
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS COLORADO
SCALE 2" = 1 Ft.


Drill "For 1" S.A.E. bolt to Match cylinder
Tap 3/4" S.A.E.
Cylinder head Cast Iron 8 Req'd.
Scale 1 1/2" = 1'
Tap 5/8 S.A.E
Tap to Match Safety valve
Bore 1/2" Counterbore 60° - 1/4" Grind
Limit -.0005 +.0005
Thread for Packing gland
Babbitt for 1" Shaft
Tap 1/2" S.A.E.
Tap 3/8" S.A.E.
Top View-
Bore for .75" Shaft
Tap 3/8" S.A.E.
Drill 1/4"
Fuel pump piston Steel 8 Req'd.
Limit -.0005 +.0005
Tap 5/8" S.A.E.
Tap 1/4" S.A.E.
End View.
Fuel Pump Body Cast Iron Scale 1/4" - 1"
Drill to match Pump body
Tap for 1/8" High Pressure connection
Grind to match pump body
Limit -.0005 +.0005
Fuel pump Head Cast Iron 1. Req'd.
Scale 1/2" = 1"
Where not listed -.001
limit on f surface +.001
.25" Key way
Thread 7/8 S.A.E.
Fuel pump Shaft Steel 1 Req'd.
Scale 1/2" - 1"
PLATE - 6
DRAWN BY E.I.WATSON
TRACED BY E.I.WATSON
JANURARY 16, 26.
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO
SCALE Listed

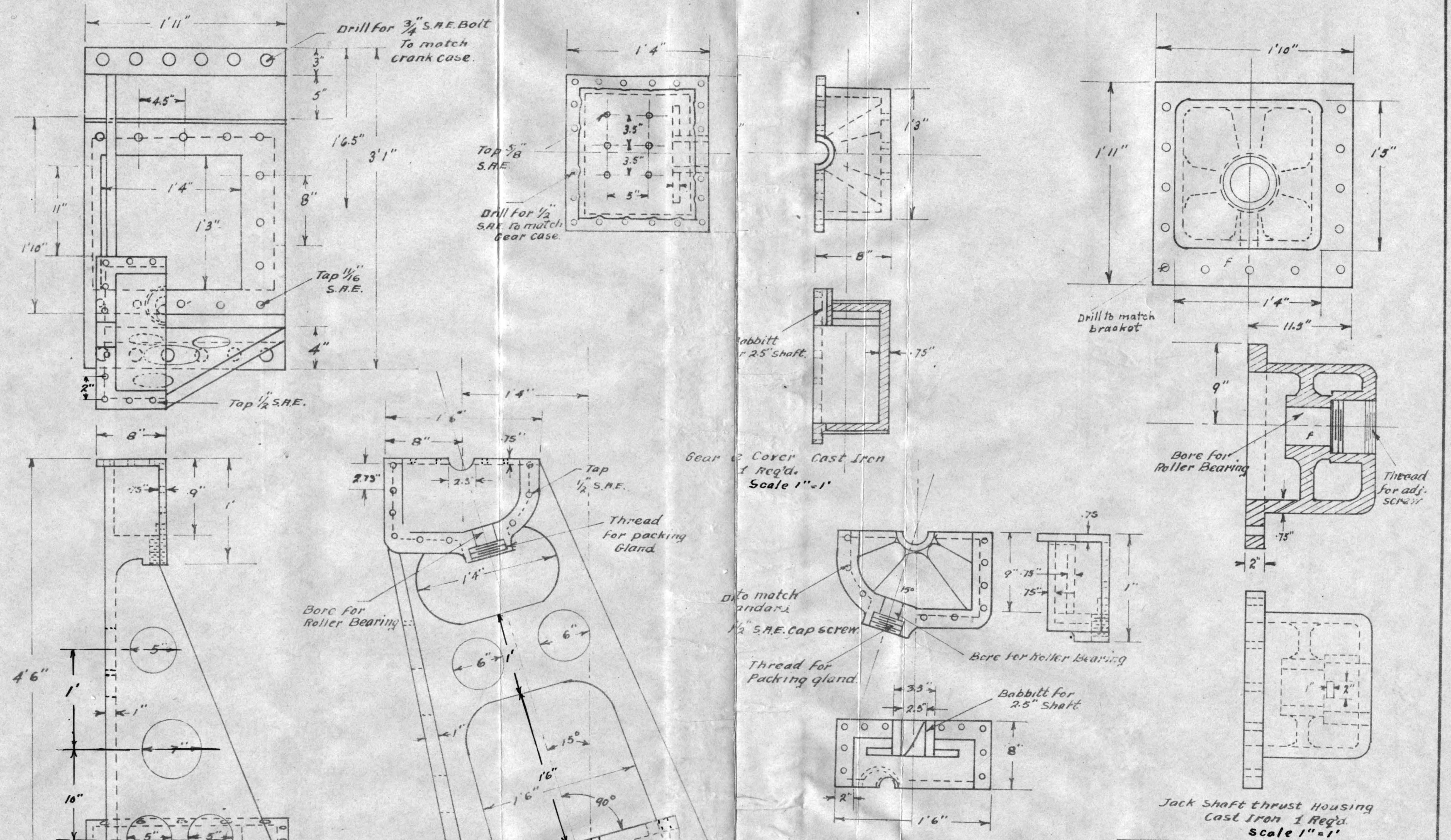


PLATE - 7

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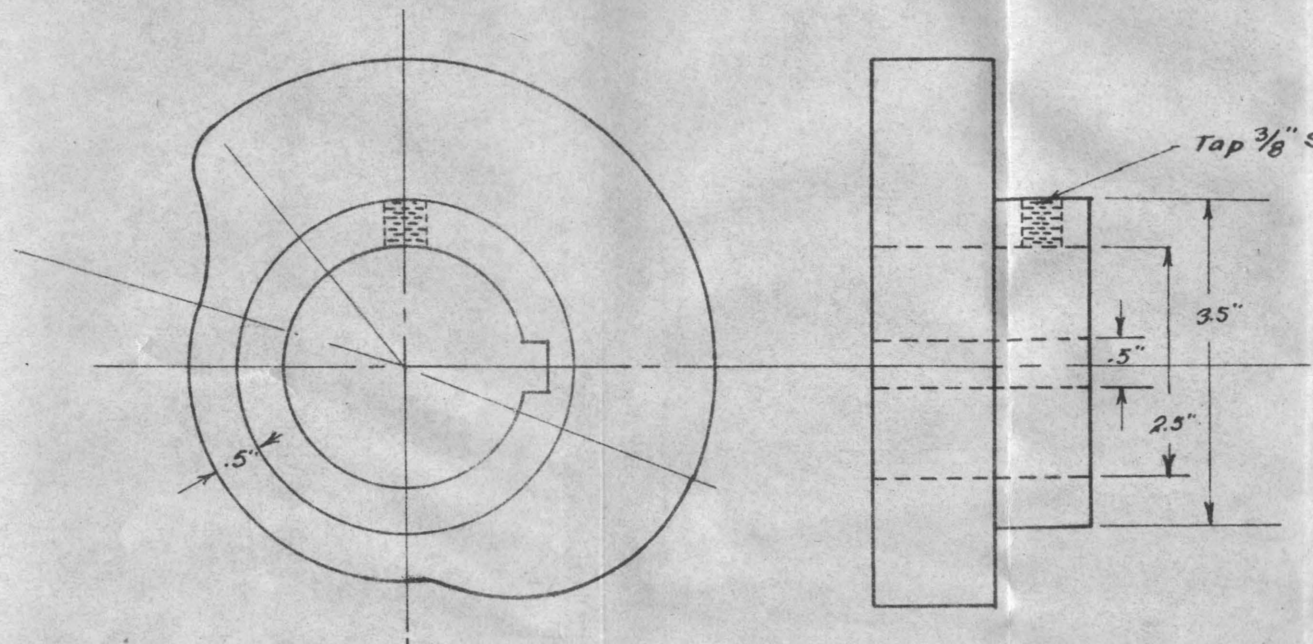
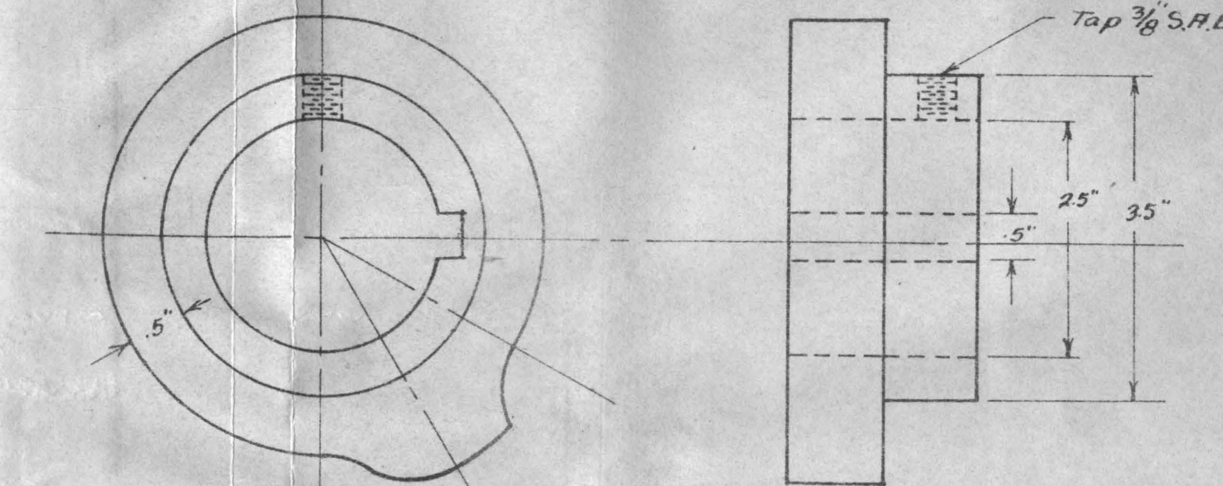
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE - GIVEN

GRIND FACE OF CMS

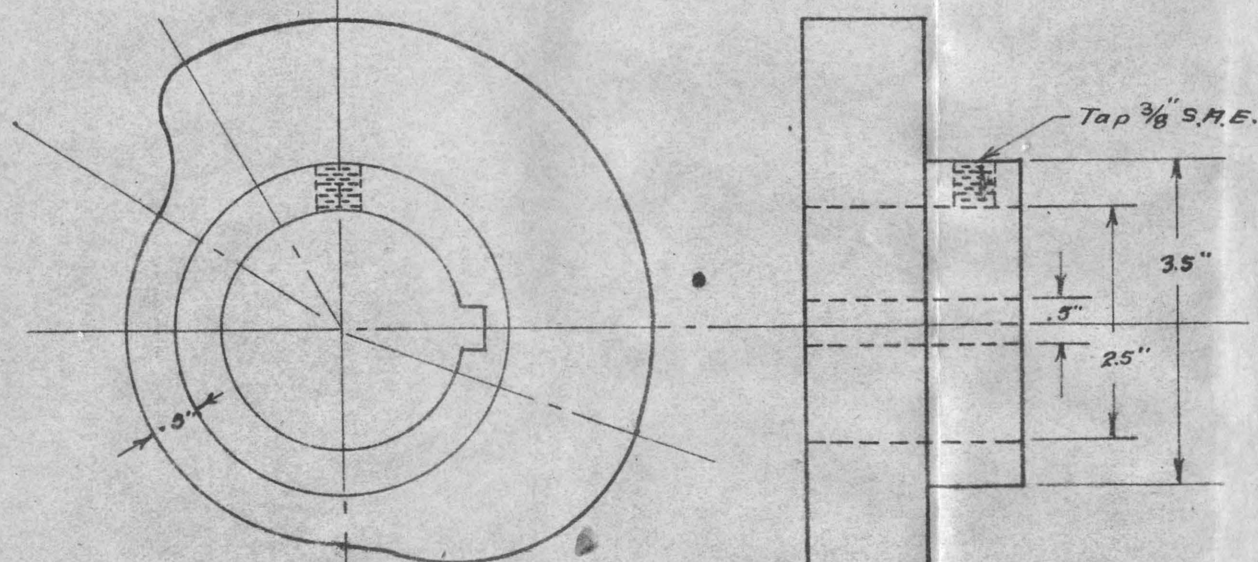


Exhaust Cam Steel Finish all over, "Hard i"
Lift 1" 8 Req'd.

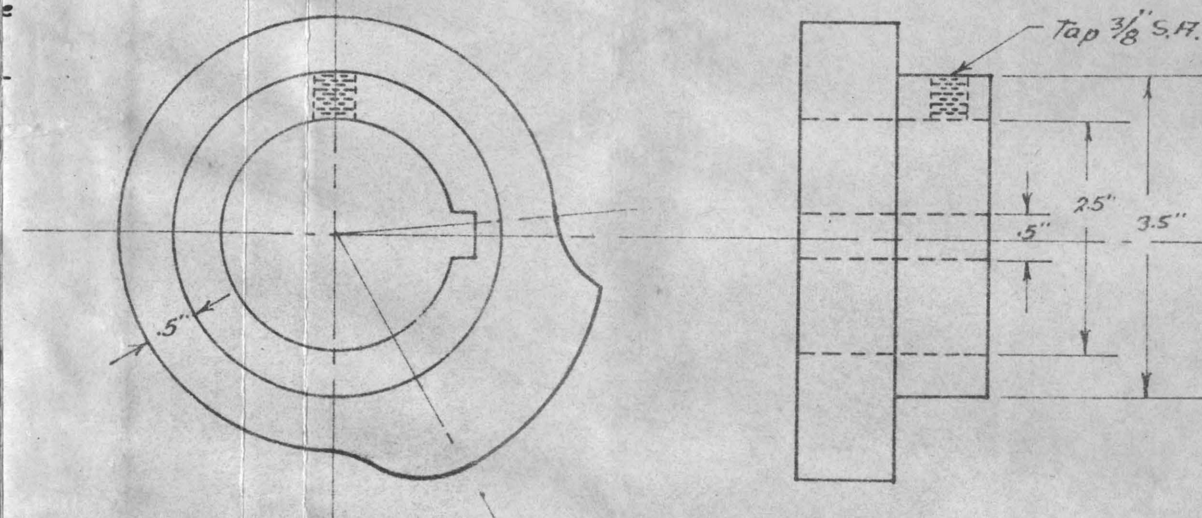


Injection Cam, Steel, Finish all over, "Harden"
Lift .5" 8 Req'd

Ground surf
Limit of err
-.001
+.001



Inlet Cam Steel Finish all over, "Harden"
Lift 1" 8 Req'd.

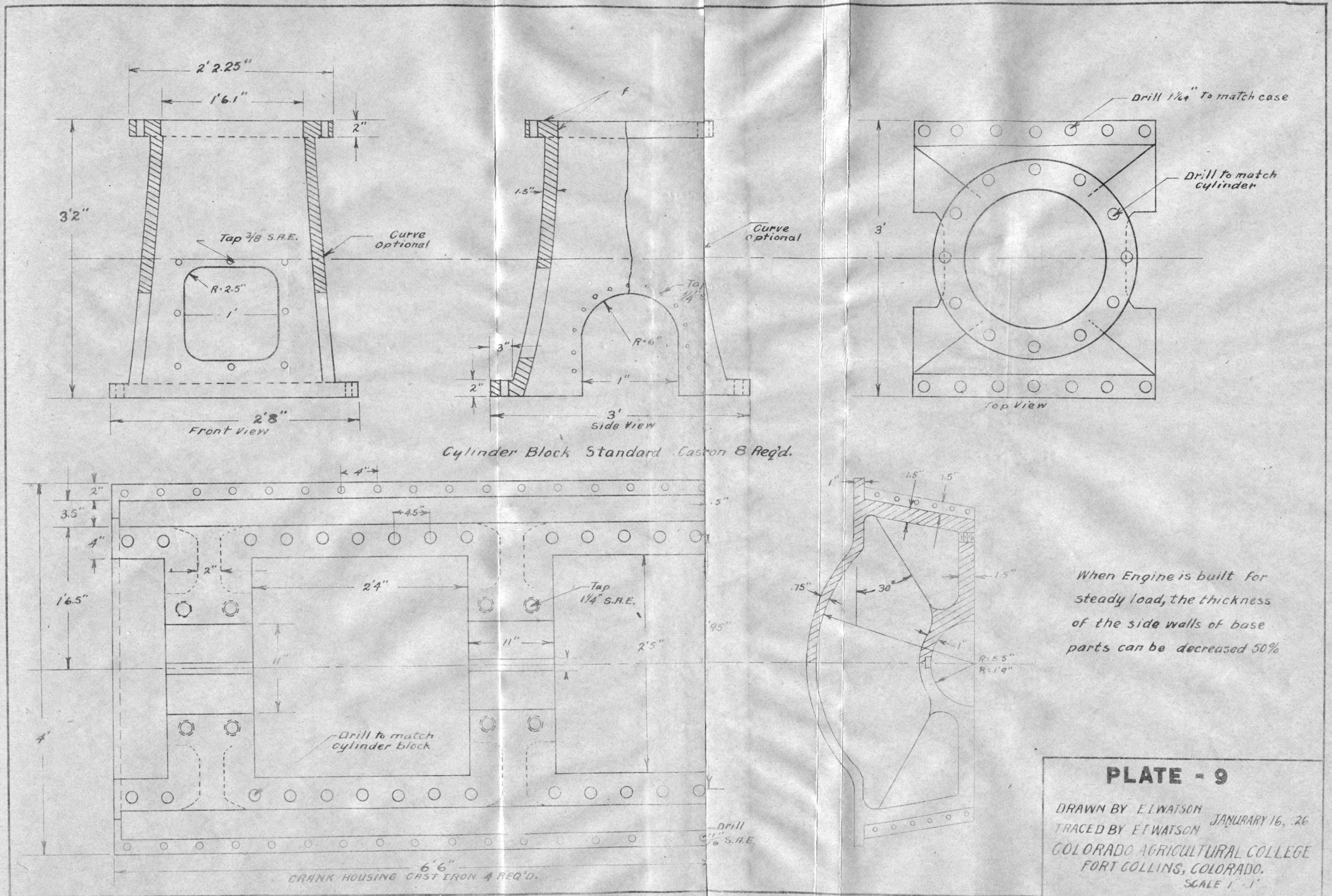


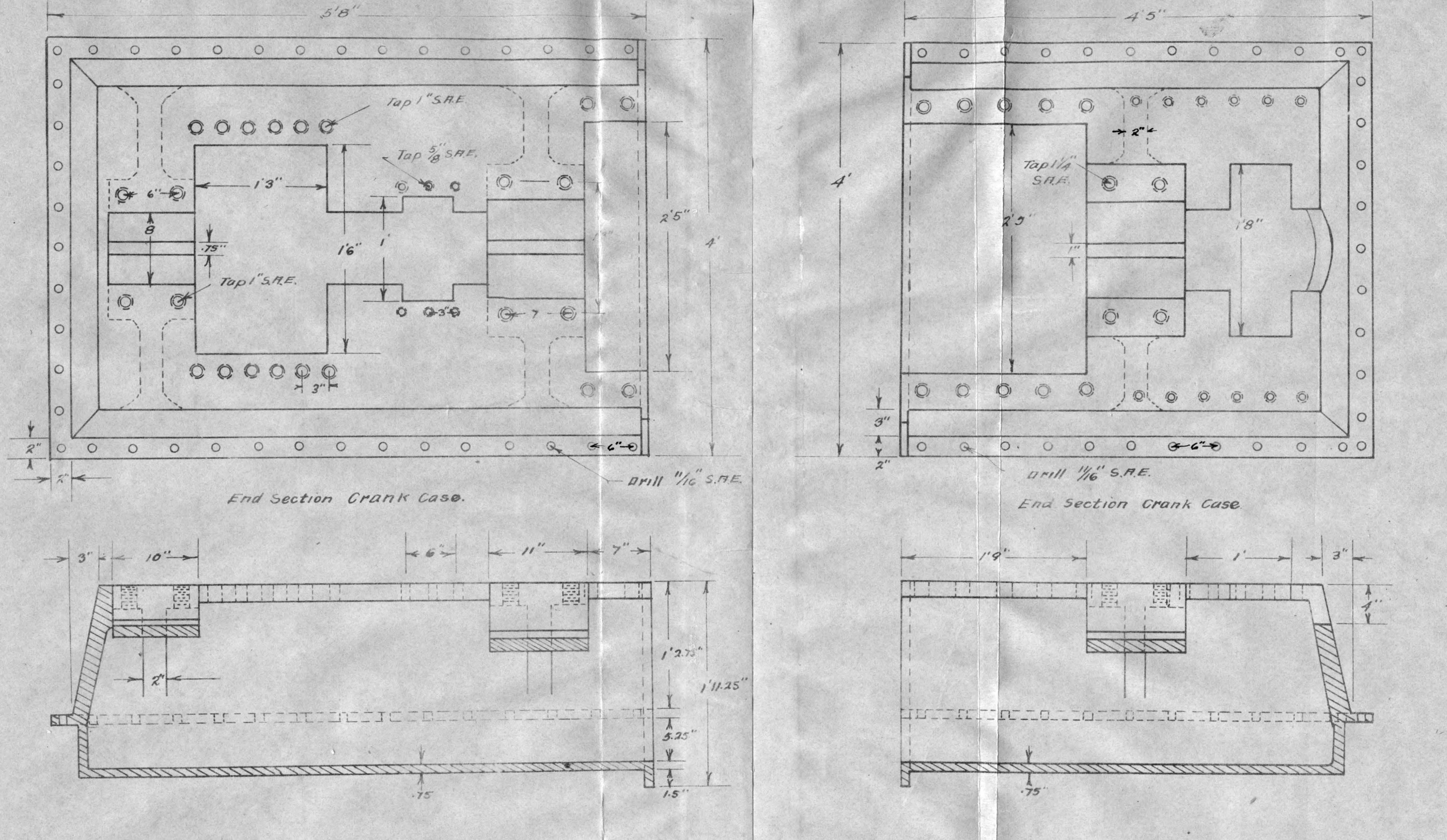
Air Starting Cam, Steel, Finish all over, "Harden"
Lift .5" 4 Req'd.

See plate 12 for lay out cams

PLATE - 8

DRAWN BY E I WATSON
TRACED BY E I WATSON JANURARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE 1/2" = 1"


2' 2.25"
1' 6.1"
2"
3' 2"
Tap 3/8" S.A.E.
Curve Optional
R=2.5"
1'
2' 8"
Front View
f
1.5"
Curve Optional
Tap 1/4" S.A.E.
R=6"
3"
2"
1"
3'
Side View
Drill 1 1/64" To match case
Drill to match cylinder
3'
Top View
Cylinder Block Standard Cast Iron 8 Req'd.
4"
2"
3.5"
4.5"
.5"
4"
1' 6.5"
2"
2' 4"
Tap 1/4" S.A.E.
11"
11"
2' 5"
.95"
4'
Drill to match cylinder block
Drill 11/16" S.A.E.
6' 6"
CRANK HOUSING CAST IRON 4 REQ'D.
1"
1.5"
1.5"
1.5"
.75"
30°
1"
R=5.5"
R=1' 9"
When Engine is built for steady load, the thickness of the side walls of base parts can be decreased 50%
PLATE - 9
DRAWN BY E.I. WATSON
TRACED BY E.I. WATSON
JANUARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE 1" = 1'



See Plate 9 for cross section of se.

PLATE - 10

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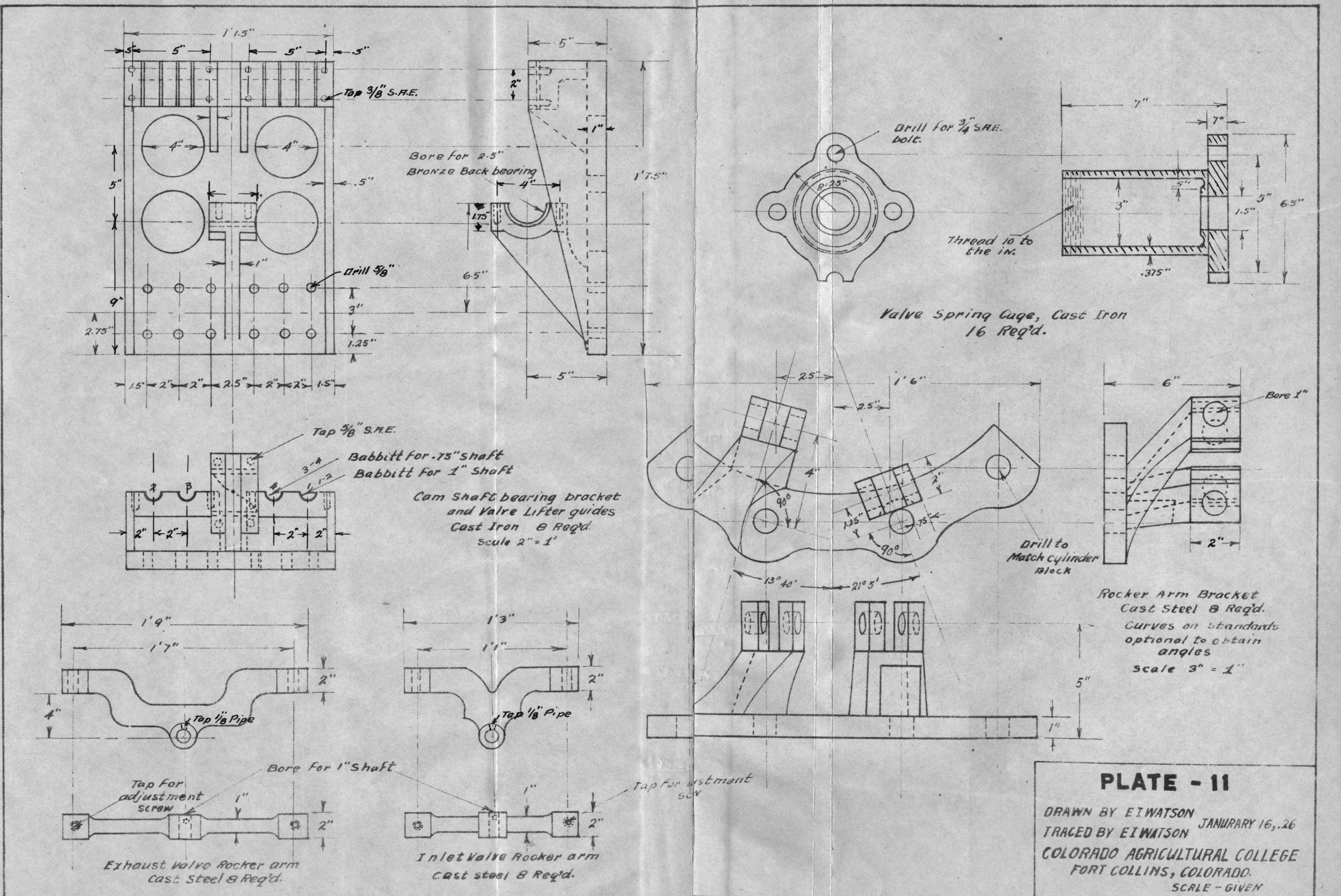
JANURARY 16, .26

COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO

SCALE 1" - 1'


Tap 3/8" S.A.E.
Drill 5/8"
Bore for 2.5" Bronze Back bearing
Tap 5/8" S.A.E.
Babbitt for .75" Shaft
Babbitt for 1" Shaft
Cam Shaft bearing bracket and Valve Lifter guides Cast Iron 8 Req'd.
Scale 2" = 1'
Drill for 3/4" S.A.E. bolt.
Thread 10 to the in.
Valve Spring Cage, Cast Iron 16 Req'd.
Bore 1"
Drill to Match cylinder Block
Rocker Arm Bracket Cast Steel 8 Req'd.
Curves on standards optional to obtain angles
Scale 3" = 1'
Tap 1/8 Pipe
Tap 1/8" Pipe
Tap for adjustment screw
Bore for 1" Shaft
Exhaust Valve Rocker arm Cast Steel 8 Req'd.
Inlet Valve Rocker arm Cast steel 8 Req'd.
PLATE - 11
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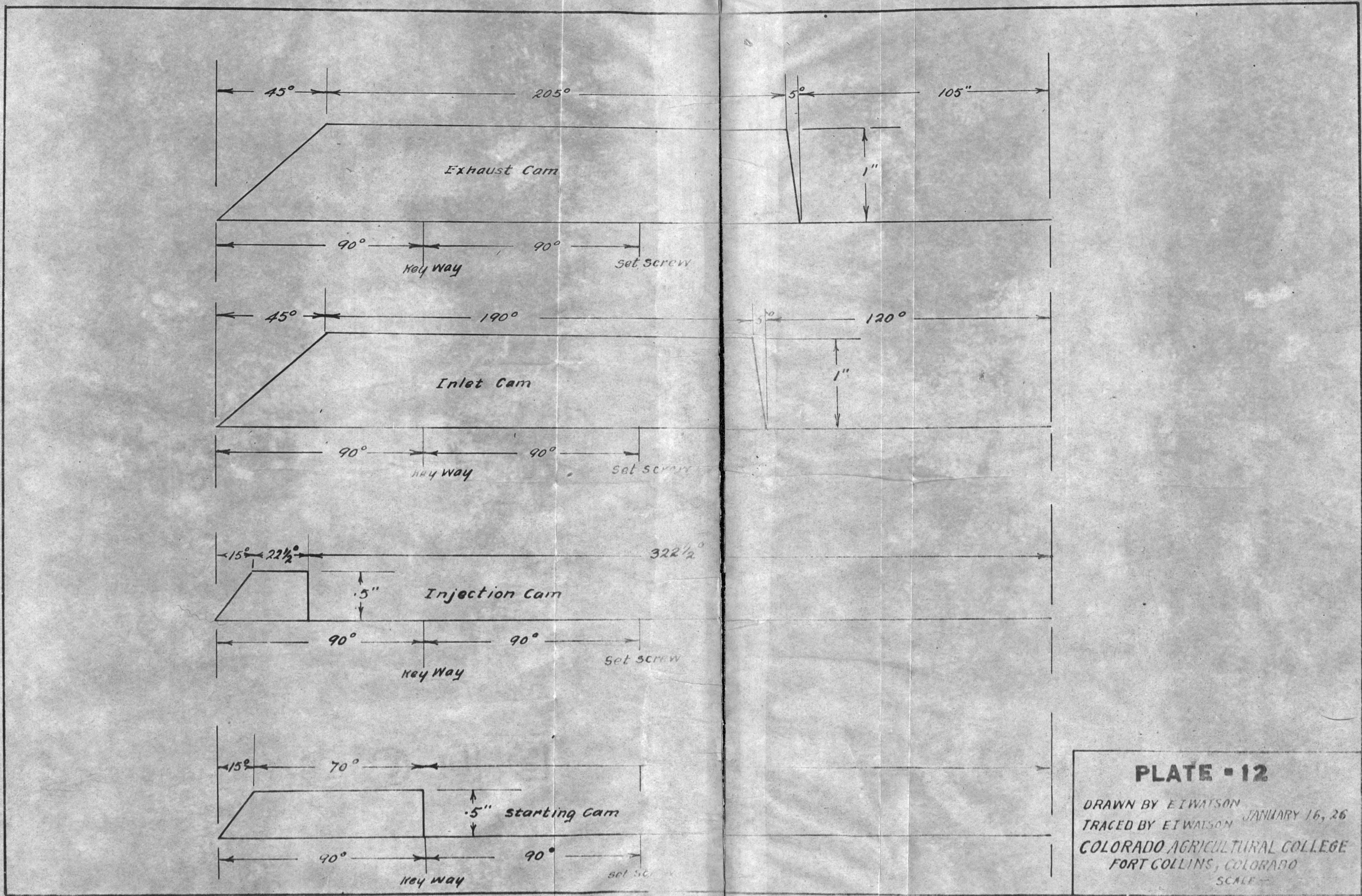
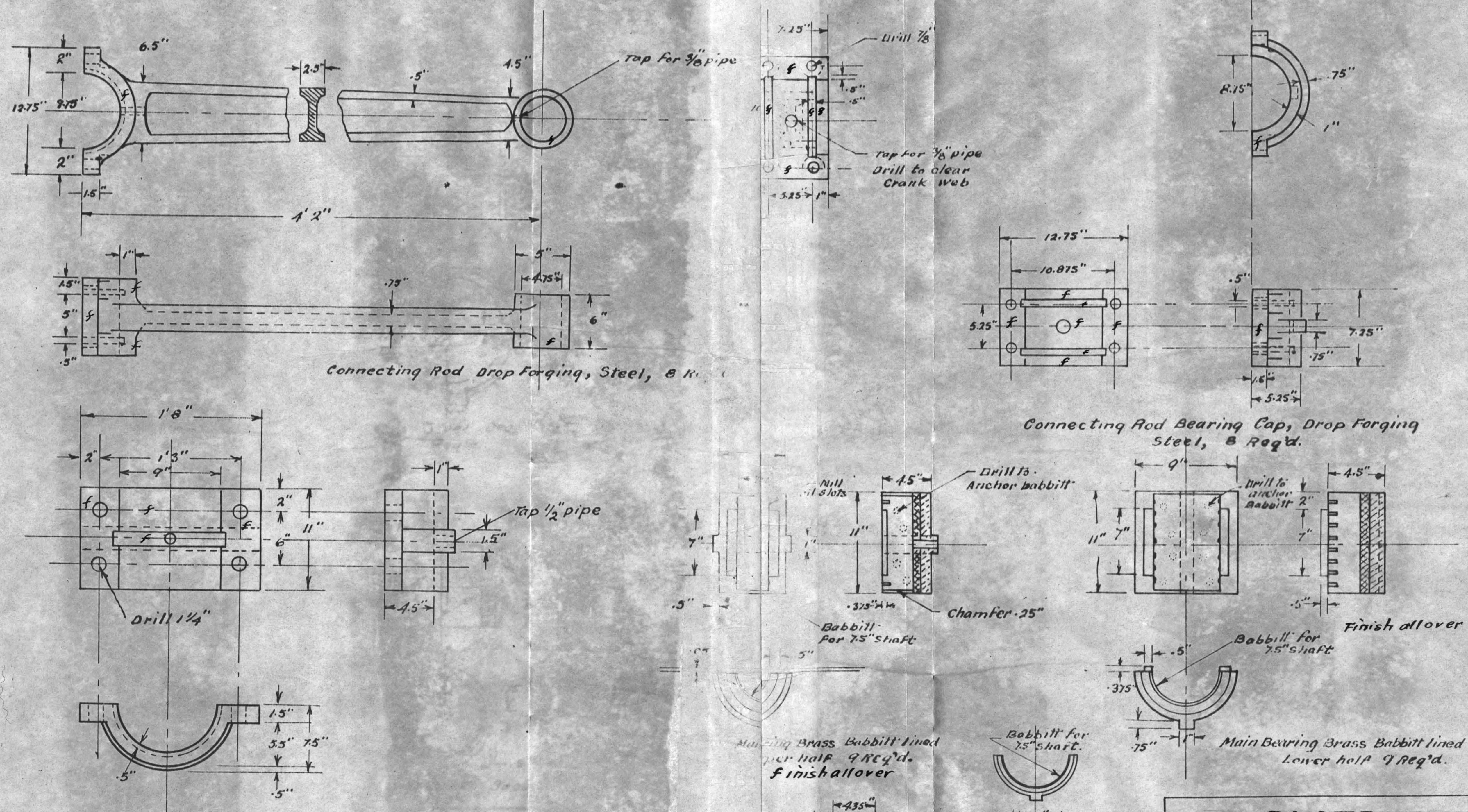
TRACED BY E I WATSON
JANURARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE - GIVEN


45°
205°
5°
105"
Exhaust Cam
1"
90°
90°
Key Way
Set Screw
45°
190°
120°
Inlet Cam
1"
90°
90°
Key Way
Set Screw
15°
22½°
322½°
.5"
Injection Cam
90°
90°
Key Way
Set Screw
15°
70°
.5"
Starting Cam
90°
90°
Key Way
PLATE - 12
DRAWN BY E I WATSON
TRACED BY E I WATSON
JANUARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO
SCALE -



Connecting Rod Drop Forging, Steel, 8 Re...

Connecting Rod Bearing Cap, Drop Forging Steel, 8 Req'd.

Main Bearing Cap, Cast Iron, 9 Req'd.

...ing Brass Babbitt lined ...per half 9 Req'd. Finish allover

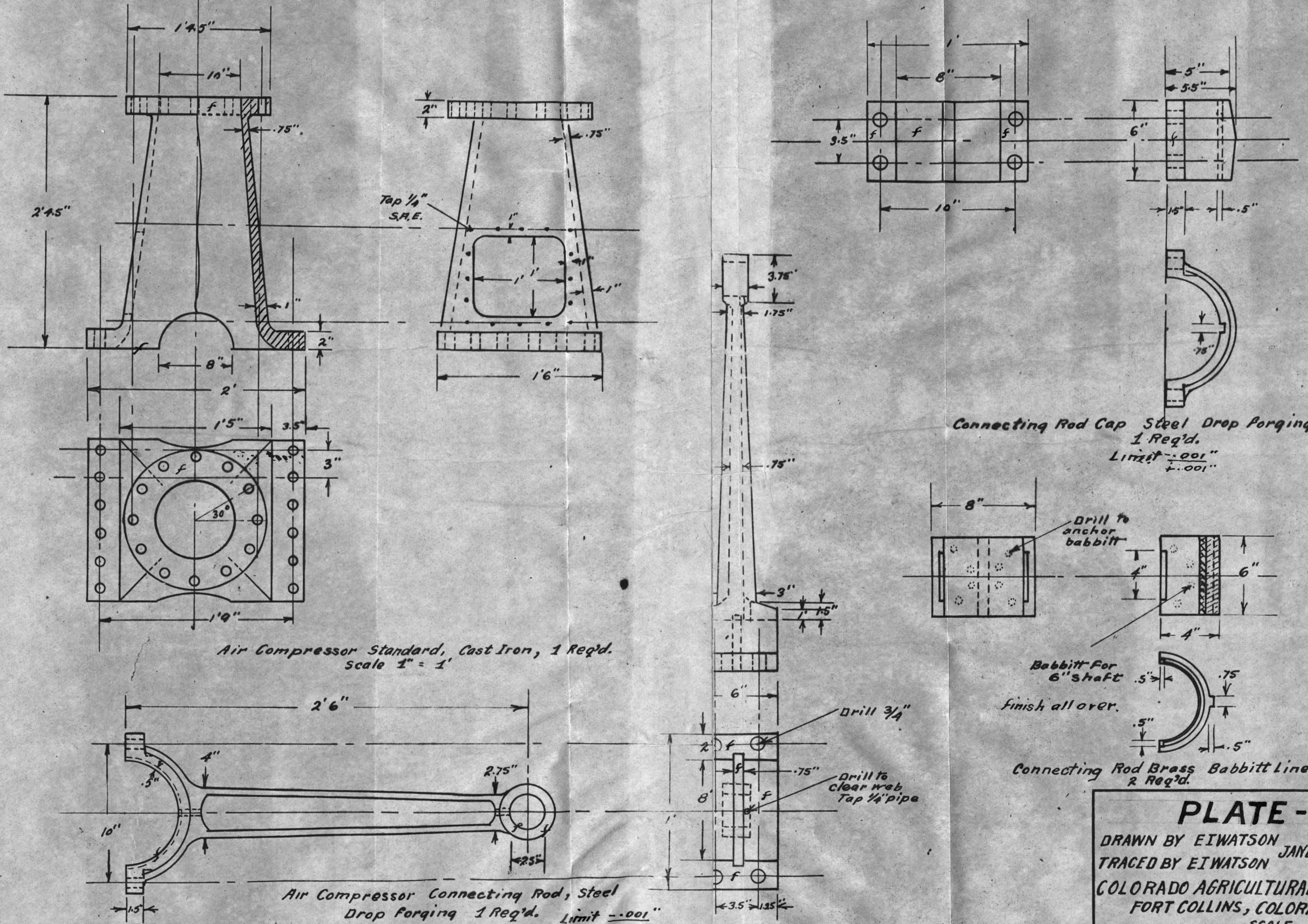
Main Bearing Brass Babbitt lined Lower half 9 Req'd.

Connecting Rod Brass- Babbitt Lined 16 Pieces Req'd. Finish allover.

Limit of all finished parts $\frac{-.001"}{+.001"}$

PLATE - 13

DRAWN BY E I WATSON
TRACED BY E I WATSON
JANUARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE - 1" = 1'



Air Compressor Standard, Cast Iron, 1 Req'd.
Scale 1" = 1'

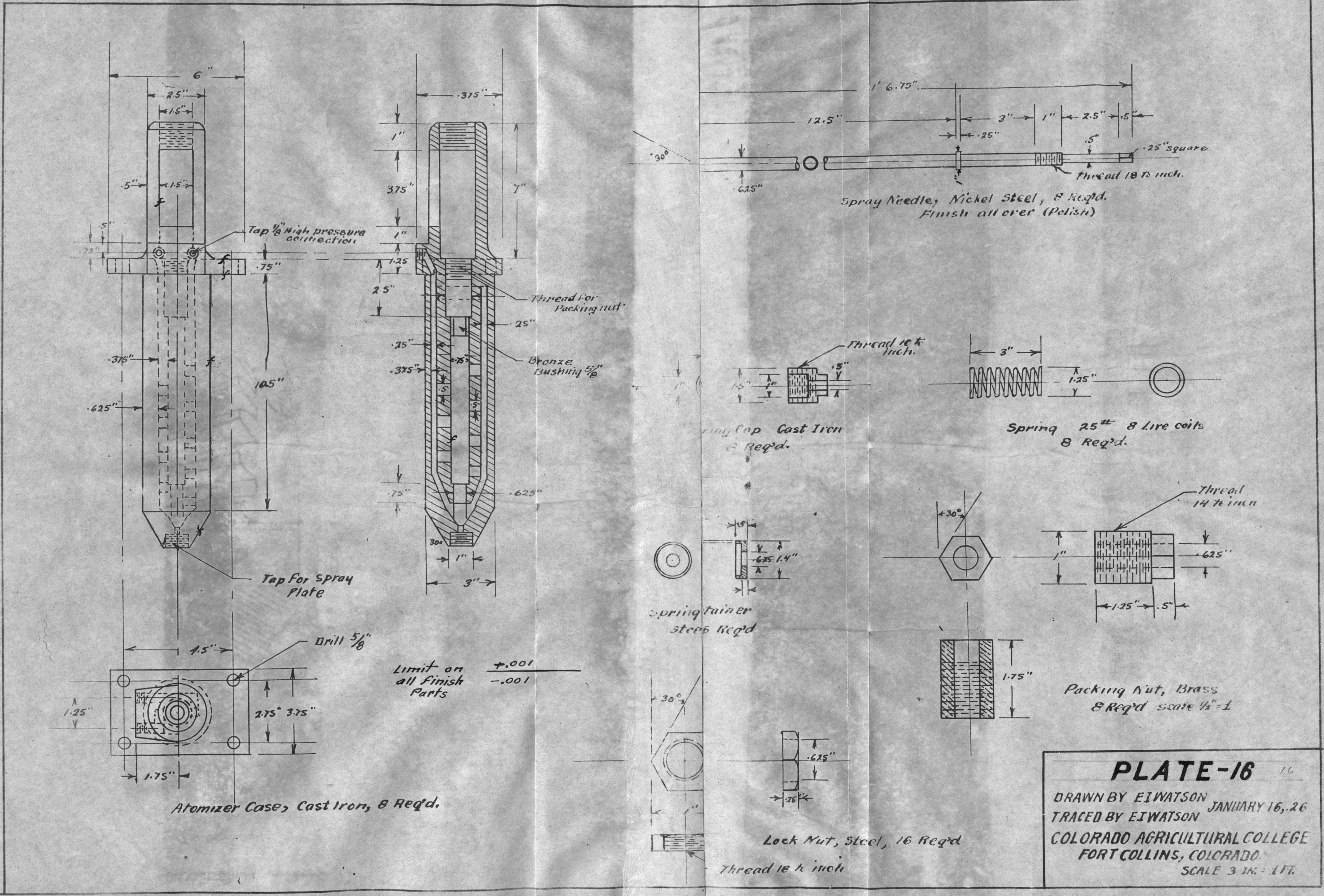
Air Compressor Connecting Rod, Steel Drop Forging 1 Req'd. Limit $\frac{-.001"}{+.001"}$

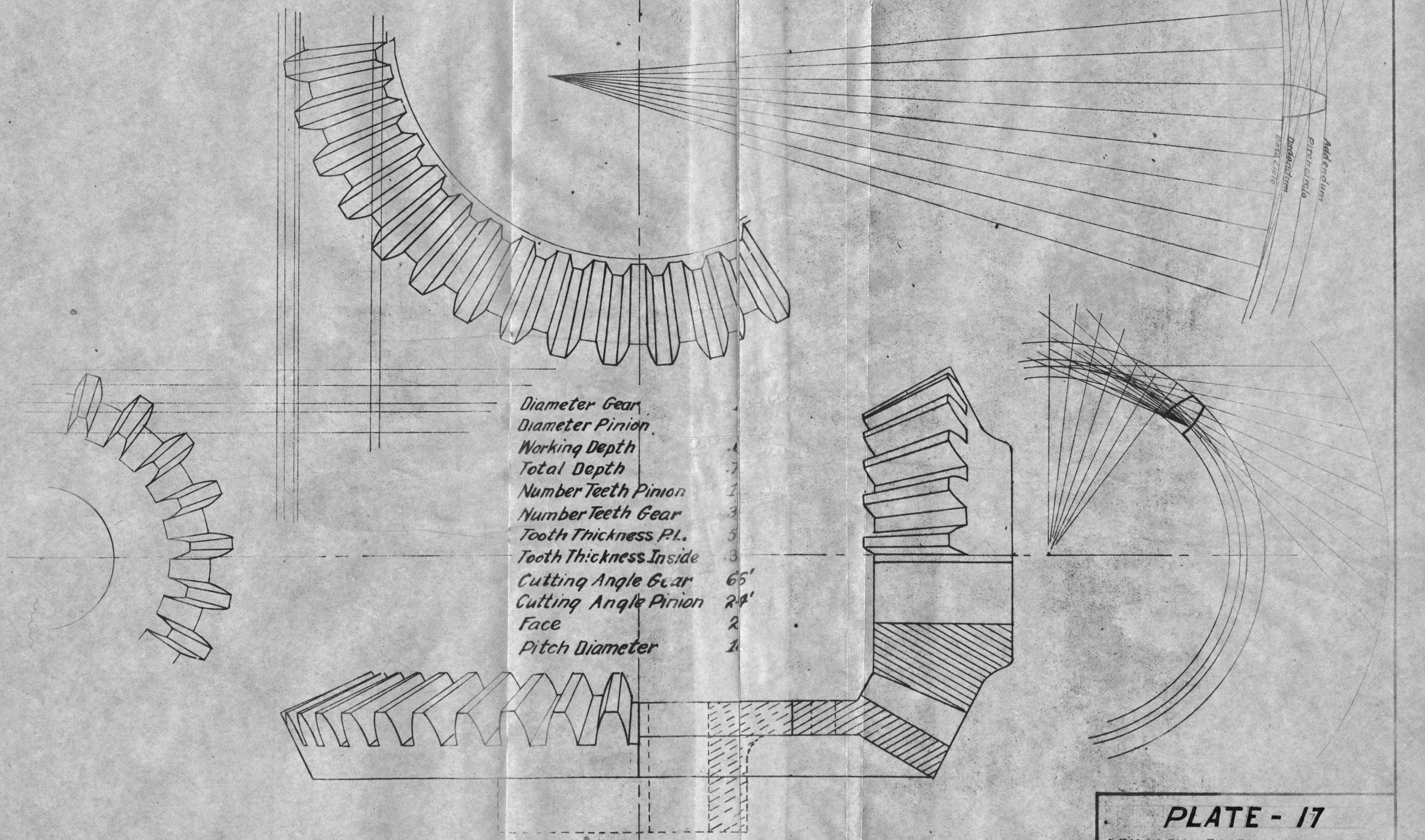
Connecting Rod Cap Steel Drop Forging 1 Req'd.
Limit $\frac{-.001"}{+.001"}$

Connecting Rod Brass Babbitt Lined 2 Req'd.

PLATE - 14

DRAWN BY E I WATSON JANUARY 16, .26
TRACED BY E I WATSON
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE: 1½" = 1'


Tap 1/8" High pressure connection
Thread for Packing nut
Bronze Bushing 5/8"
Tap for spray Plate
Drill 5/8"
Atomizer Case, Cast Iron, 8 Req'd.
Limit on all Finish Parts +.001 -.001
Spray Needle, Nickel Steel, 8 Req'd.
Finish all over (Polish)
Thread 18 to inch.
.25" square.
Thread 16 to inch.
Cap Cast Iron 8 Req'd.
Spring 25# 8 Live coils 8 Req'd.
Spring tainer steel Req'd
Thread 14 to inch
Packing Nut, Brass 8 Req'd scale 1/2"=1
Lock Nut, Steel, 16 Req'd
Thread 16 to inch
PLATE-16
DRAWN BY E I WATSON
TRACED BY E I WATSON
JANUARY 16, 26
COLORADO AGRICULTURAL COLLEGE
FORT COLLINS, COLORADO.
SCALE 3 IN. = 1 FT.



Diameter Gear	
Diameter Pinion	
Working Depth	.6
Total Depth	.7
Number Teeth Pinion	1
Number Teeth Gear	3
Tooth Thickness P.L.	5
Tooth Thickness Inside	.3
Cutting Angle Gear	66'
Cutting Angle Pinion	24'
Face	2
Pitch Diameter	1

2 Gears 12" Diameter Req'd. 1 Ige Connected
1 Hub Connected 1 Pinion 6" Dieter 2.5" Hub
1 Pinion 12" Diameter 2.5" Hu Thickness Hub 1.5"
Thickness Flange 1"

PLATE - 17

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JANUARY 16, '26

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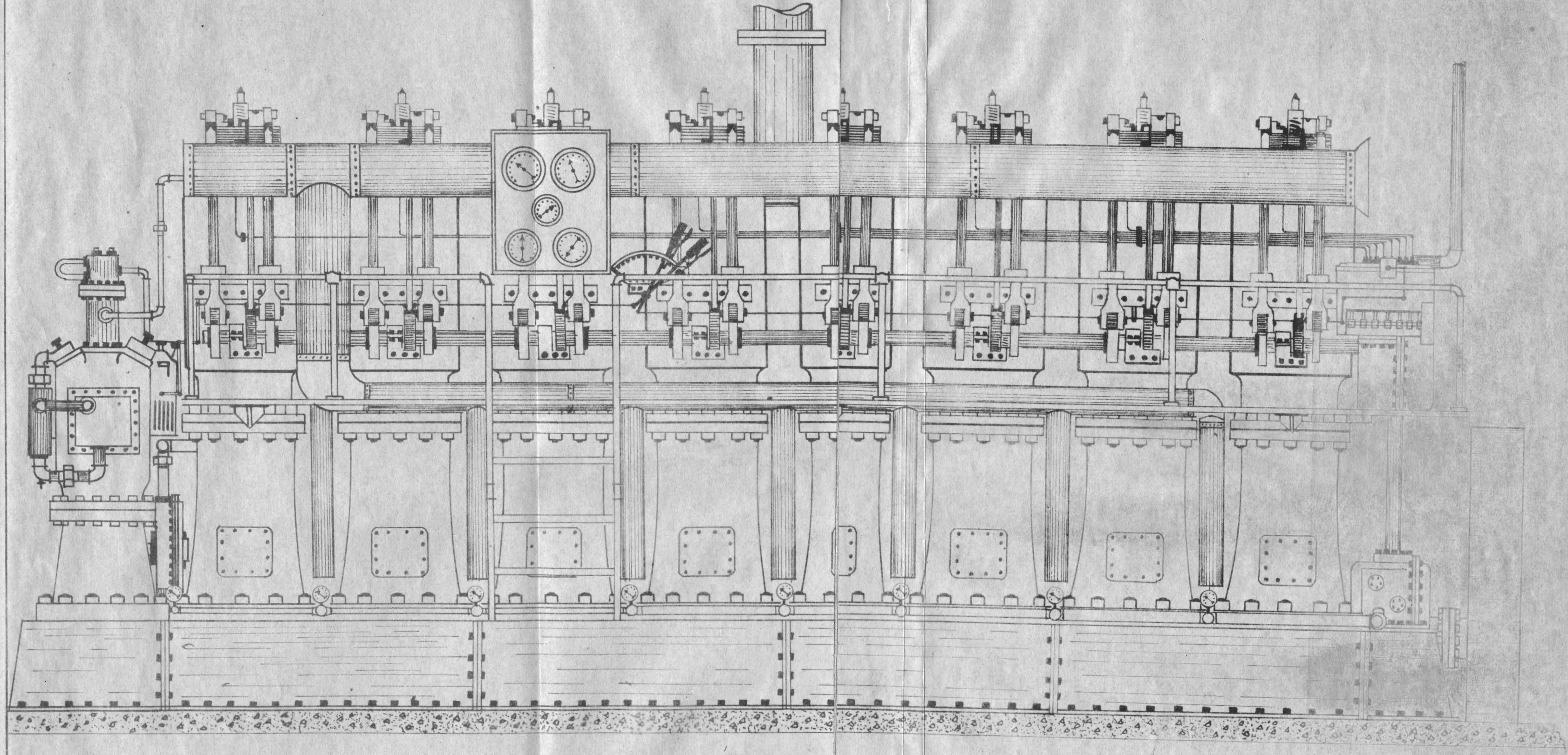
FORT COLLINS, COLORADO.



500 H.P. SOLID INJECTION DIESEL'PE ENGINE 250 R.P.M.

DESIGNED BY E.I. WATSON