THESIS

A STUDY OF THE DESIGN OF AN AUTOMOBILE MOTOR.

A Report Submitted to the
Faculty of the
Michigan Agricultural College

By

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June, 1921.

THES.

GENERAL CONSTRUCTION.

The motor designed is intended to be used in a car of approximately one hundred sixteen inch wheel base and weighing about twenty four hundred pounds.

In general the motor is of the four cylinder type of three and one half inch bore by five inch stroke. The main ideas to be worked out in the design of the motor are, a crankshaft of the three bearing counterweight type, an overhead camshaft operating two inlet and two exhaust valves to a cylinder through rocker arms, connection between the crankshaft and camshaft by helical gears on a vertical shaft, generator and water pumps to be driven by a horizontal shaft at right angles to the crankshaft and connected with it through helical gears, and pressure lubrication.

The design of the parts will be given in the order it was taken up. It is believed this method will present the computations most clearly.

CRANKSHAFT AND BEARINGS.

Crankshaft.

The crankshaft is to be of the three bearing counterweight type, dropforged to rough dimensions and then machined all over. Extra material is left in the counterweights so as to bring an unbalanced couple at these points to be balanced out by a (Carwin Static Dynamic) balancing machine. In case a Carwin machine is not available or it would not be good policy to buy one, the weights should be then computed so that the couple formed by the counterweights just balances the couple formed by the crank pins and the lower half of the connecting rods.

Let p = piston displacement,

d = Diam. of crank pin and main crank journal.

L_1 L_2 and L_3 = Length of main crank journals front to rear.

w = width of crank arm.

t_s = thickness of short crank arm,

t_l = thickness of long crank arms.

$$p = \frac{\pi}{4} 3.5^2 (5) = 48.1 \text{ cu. in.}$$

From Heldt Vol. 1 - 189, 190, 191.

$$d = \sqrt{\frac{p}{16}} = \sqrt{\frac{48.1}{16}} = 1.733.$$

It is desired to drill a 3/4" hole through the pins so let us equate the section modulus of the solid shaft to that of a hollow shaft 3/4" inside diam. and 2" outside diameter.

$$\frac{\pi d^3}{16} = \frac{\pi p^3}{16} - \frac{\pi d_1^3}{16}$$

$$d^3 = p^3 - d_1^3$$

$$1.733^3 = 2^3 - .75^3$$

$$5.18 = 8 - .421 = 7.579$$

We are safe in the dimensions assumed.

From Heldt.

$$L = 1.25\ d = 1.25\ (1.733) = 2.168 = 2\ 3/16''.$$

$$L_1 = L_2 = 1.25\ L = 1.25\ (2.168) = 2.71''$$

$$L_2 = 2.75''$$

It is required that bearing one be as short as possible so let us compute the area required and find the length required in this manner.

Area required = 1.733 (2.71) = 4.70 sq. in.

Length = $\frac{4.70 \text{ sq. in.}}{2''} = 2.35''.$

Allowing 1/4" for lubrication grooves gives us 2 5/8".

$$L_3 = 1.75\ L = 1.75\ (2.168) = 3.79''$$

Length required = $\frac{3.79\ (1.733'')}{2} = 3.28''.$

Use 3 5/8" to allow for 1/4" oil groove,

$$w = 1.4\,d = 1.4\,(1.733) = 2.43" \quad \text{use } 2.5"$$

$$t_s = .6\sqrt{\frac{d^3}{w}} = .6\sqrt{\frac{1.75^3}{2.5}} = .876" \quad \text{use } 1"$$

$$t_1 = .8\sqrt{\frac{d^3}{w}} = .8\sqrt{\frac{1.75^3}{2.5}} = 1.17" \quad \text{use } 1\ 1/2"$$

t_s and t_1 are made extra large to reduce bending of the shaft under load to a minimum.

Steel for Crankshaft recommended by Heldt.

Chemical Composition.

		%
Carbon		.45
Manganese		.60
Silicon	not over	.18
Sulphur	not over	.04
Phosporus	not over	.04

Physical Properties

Tensile strength .... 90,000 lbs. per sq. in.

Elastic limit .... 70,000 lbs. per sq. in.

Elongation in 2 inches 15%

Contraction of area 38%

Crankshaft Bearings.

Use die cast bushings.

Heldt vol. 1 - 347.

A = Bearing diameter = 2"

B = Outside diam. bushing = 1.2A =

1.2 (2) = 2.4" use 2 1/2"

Thickness of flange = .1A = .1(2) = .2 use 1/4".

Diameter of flange = B + .2A = 2.4 + .2(2) =

2.8" use 2 7/8"

Use 4 1/2" bolts for each bearing.

FLYWHEEL

Let flywheel diameter equal 14" and thickness $1\frac{1}{4}$"

Heldt Vol 1 - 302.

$$\text{Weight flywheel} = 8.4\left(\frac{\text{Bore (stroke)}}{\text{radius wheel}}\right)^2 = 8.4\left(\frac{3.5\,(5)}{7}\right)^2 = 52.5\text{ lb.}$$

$$\text{Width (approx)} = \frac{Wt}{.28}\cdot\frac{1}{\pi\,\text{diam. (thickness)}} =$$

$$\frac{52.5}{.28}\cdot\frac{1}{\pi\,(4)\,1.25} = 3.41\text{" say } 3.5\text{"}$$

Let thickness of web equal .5 inches which is generally used.

Diameter of Flange = Length of stroke = 5"

Heldt 1 - 305

Width of Flange = 1/16 (inch of bore)(square root of number of cylinders) =

1/16 (3.5) (2) = .437" say .5"

Diameter of bolts = Thickness of flange = .5"

COMBUSTION CHAMBER.

Assume height of chamber = 1 inch.

Clearance volume = Depth (Area rectangle - area rounded corner) - volume of piston above cylinder proper - $1 \left[4\,(4.625) - \left[2.25^2 - \frac{\pi}{4}\,(2.25)^2 \right] - \left[\frac{\pi}{4}\,(3.5^2)\ 3/8 \right] \right]$ = 12.81 cu. in.

Total vol. = 12.81 cu. in. + 48.1 cu. in = 60.91 cu. in.

Compression ratio = $\frac{60.9}{12.81}$ = 4.75

Assume an inlet pressure of 13 pounds per square inch.

Compression pressure = inlet pressure $(\text{ratio})^{1.3}$ = $13(4.75)^{1.3}$ = 98.3#/sq. in. absolute = 83.6 lbs. per sq. in. gauge.

This pressure is satisfactory for a high speed engine so it will not be necessary to carry the work further.

CYLINDER AND CRANKCASE.

DIMENSIONS.

Heldt Vol. 1 - 83.

Thickness = $\frac{\text{bore}}{30}$ + 1/8 in. = $\frac{3.5}{30}$ + 1/8 = 2416 in.

use 3/8 in. cylinder wall.

Water space = 1/2"

Jacket wall = 3/16" on the sides,

3/4 to 1 1/2 in. on the top of Cylinder.

Crank case thickness = 1/4 in.

Flanges = 1.5 (thickness) 1.5(1/4) = 3/8 in.

Use 1/4 in. bolts with a 4 in. span.

CONNECTING ROD AND PINS.

Let ratio connecting rod to stroke equal 2.2

Length connecting rod 2.2(5) = 11 in.

Use nickel steel s = 22000 lb. per sq. in.

Heldt 1 - 143.

Explosion force = compression pressure (4) =

98.3 (4) = 393.2

Total force on piston = $\frac{\pi}{4}\ 3.5^2\ 393.2$ = 3780 lbs.

Heldt 1 - 220 chart III.

T = .159 use 5/32".

Height of section = .159(5.7) = .906" use 15/16".

Width of section = .159(3.8) = .604" use 5/8"

The rod is of this cross section for entire length. Enlarging the rod at the crank pin end has no advantage as the greatest stress comes at the middle of the rod due to whipping. If any cross section should be made larger this should be.

Heldt 44 Equa. 24.

F_a = force due to reciprocating parts,

w = wt. of reciprocating parts,

assume 4.5 lb.

L = Length of stroke in inches,

N = R.P.M.

n =ratio connecting rod "center to center" to stroke.

$$F_a = .0000142\ WLN^2(\cos\theta + 1/2\ n \cos 2\ \theta)\ \text{lb.} =$$

$$.0000142\ (4.5)\ 5\ (3000)^2(1 + \frac{1}{2(\frac{11}{5})} =$$

3540 lbs.

Heldt 60 Equa. 31

F = Centrifugal force,

W = Weight of revolving parts,

N = Rev. per sec. = 3000/60

r = radius in feet,

$$F = 1.226\ WN^2\ r\ \text{lb.}$$

Assume lower half of connecting weighs 2 lbs. and r = 3".=

$$F = 1.226(2)\ \frac{3000}{60}^2\ \frac{3}{12} = 1530\ \text{lbs.}$$

Total force = 3540 lb. + 1530 lb. = 5070 lb.

Using two bolts stress per bolt = 2035 lb.

Assume a stress of 22,000 lb. per square inch.

Area each bolt $= \frac{2538}{22000} = .1154$ sq. in.

Use a 7/16 in. S.A.E. bolt,

a = .12.7 sq. in.

Length bolts = 1 1/2".

PISTON AND PISTON PIN.

Heldt 144 - 146.

Average piston pressure = 2500 lb. per sq. in.

$\therefore$ Area of bearing $= \frac{3780}{2500} = 1.51$ square inch.

Assume 7/8" outside diameter.

Length $= \frac{1.51}{7/8} = 1.73$ say 1.75".

Let d_1 = inside diameter,

d = outside diameter,

b = piston bore,

p = Explosion pressure,

s = Stress = 20,000 lb. per sq. in.

(3.5 nickel steel)

$$d_1 = \sqrt[4]{d^4 - \frac{db^3p}{2s}} = \sqrt[4]{.875^4 - \frac{.875(3.5)^3\ 3780}{2\ (20,000)}} =$$

.681 use 5/8"

PISTON DIMENSION.

Average practice as given by Heldt at 1800 R.P.M. an average side thrust of 110 lbs. was found.

Allowing 9 lb. per sq. in. bearing pressure on the piston we have

Length piston = $\frac{110}{9(3.5)}$ = 3.5" bearing length.

Diameter piston at top = .998 (3.5) = 3.493 in.

Diameter piston at bottom = .9995 (3.5) = 3.498 in.

Thickness of head = .032 (3.5) + .060 = .182 use $\frac{3}{16}$"

Depth of ring grooves = .04(d) = .04(3.5) =
.140 say 9/64"

Thickness of piston wall upper rod =
.062(3.5) + .10 = .317 use 5/16.

Thickness wall lower end = .02(3.5) + .05 = 12
use 1/8 in.

Use 2 1/4 in. set screws to hold the pin in position.

A good grade of cast iron to be used.

VALVES AND ROCKER ARMS

An effort has been made to keep the weight of valves and rocker arm to as low a figure as possible to insure quiet running.

Maximum speed = 3000 R.P.M.

Maximum piston speed = 3000. 10/12 = 2500 Feet per minute.

Favary 64 - 65.

Let p = piston diam. = 3.5"

Let s = piston speed = 2500 ft. per min.

Let G = gas velocity = 8180 ft. per min.

Let A = Area of one valve (two intake and two exhaust per cylinder)

$$A = \frac{1}{2} \quad \frac{.7854\ (p)^2\ s}{G} = \frac{1}{2} \quad \frac{(.7854)\ 3.5^2 (2500)}{8180} =$$

. 147 sq. in.

Diam. of valve $= \sqrt{\frac{1.47}{.7854}} = 1.37"$ use 1 3/8 in.

Lift of valve $= \frac{9}{32} + (d - 1\ 1/4)\ \frac{4}{32} =$

$$\frac{9}{32} (1\ 3/8 - 1\ 1/4)\ 1/8 = \frac{19}{64} \text{ say } \frac{20}{64} \text{ or } \frac{5}{32}"$$

Valve Dimension:

Outside diam. = 1.15 (1.375) = 1.58 say 1 9/16"

Thickness of head = .15 (1.375) = .2060 say 3/16"

Valve stem diameter = .15 (1.375) + .15 = .3560 say 5/18"

Radius of valve = .25 (1.375) = .344 say 5/16"

VALVE SPRINGS

Weight of valve, disc, spring and rocker arm.

Wt. valve = $.28\left(4.5\left(\frac{\pi}{4}(.3125)^2 + .1875\ \frac{\pi}{4}(1.375)^2\right)\right) =$

$28(.620) = .174$ lb.

Wt. spring cap = $.28\left(\frac{\pi}{4}(2^2)\ .0625\right) = .28\ (.196) = .055$ lb.

Effective weight of spring = 1/2 total weight.

Assume a spring of 10 coils, 2 in. diam. and .134 in. wire diam.

Wt. spring = $.28\ \ 10\ (2\pi)\ \frac{\pi}{4}\ (.134^2) = .248$ lb.

Effective weight = 1/2 (.248) lb. =.124 lb.

Consider rocker arm as two bars 1/2 in wide tapering to 1/4 inch, 5/16 in. thick, 1 1/4 in. long and revolving through an arc of 5/16 in with one end fixed.

Wt. rocker arms = .28(2) (1.25) (.375) (.3125) = .082 lb.

Their weight with the motion spoken of it approximately equivalent to half their weight reciprocating through a distance of 5/16 in. Therefore, the approximate weight = .041 lbs.

Total weight = .174 + .055 + .124 + .041 = .394 lb.

Let us use a mushroom cam with a top circle of 1/10 in. Distance center top circle to center of camshaft =

D = .875 which will be arrived at in the camshaft computation.

Heldt 1 - 265.

Let F. = spring force,

W = wt. reciprocating parts = .394 lb.

N = Revolution of camshaft = 1500

D = Value given above = 22.08 lb.

$$F = \frac{WN^2\ D}{35,200} = \frac{.394\ (1500)^2\ .8750}{35,200} = 22.08 \text{ lb.}$$

allowing 20% for friction of guides, etc., we have

F = 22.08 (1.20) = 26.496 lb.

. = 26.5 lb.

Heldt 582.

Let us use No. 11 wire diam. .12 in. and 1 1/4" diam. spring. Taking E = 10,000,000 the allowable load = 27.1 lb. and the deflection per coil per 100# = .7538".

$$\text{Deflection for } 27.1\# = .7538\ \frac{27.1}{100} = .204 \text{ in.}$$

When the spring is fully deflected the load should not increase by more than 20%. Since the load in proportional to the deflection the allowable deflection per coil = .20 (.204) = .0408 in.

No coils = total deflection ÷ allowable deflection per coil. $= \frac{.3125}{.0408} = 7.64$

Use 10 coils, 8 active.

Height of spring when valve is closed, allowing 3/8" between the coils when valve is opened wide =

$$10(.12) + 3/8 + .3125 = 1.8875$$

say 1 7/8 ".

Height of spring when free should be 1 7/8" + .211 (8) =

3.56".

CAMSHAFT, ROCKER ARM SHAFT AND BEARINGS.

Heldt 1 - 278.

Use 3 bearings,

Length between bearings = L = 11.0"

X = ratio of distance of cam from bearing to total distance = 8/11 = .727

a = .069

Exhaust valves.

Load on camshaft = Pressure of valve spring + Pressure due to inertia of reciprocating parts + pressure on exhaust valve.

Pressure of valve spring = 27.1 lb.

Pressure due to inertia of reciprocating parts = 26.5 lb.

Pressure on exhaust valve = $50 \frac{\pi}{4} (1.375)^2$ = 73.5 lb.

Total pressure = 127.1 lb.

Total pressure two valves = 2 (127.1) = 254.2 lb.

Consider that the two valves are both located at the point 8" from the farther bearing. Their condition may be safely assumed as it will cause a greater deflection than will the two followers where they actually are.

$$d = \sqrt[4]{\frac{P L^3 a}{8,800,000(.002)}}$$

where d = diam. camshaft required

P = load exerted by follower,

a = factor depending on ratio given above

.002 = allowable deflection.

$$d = \sqrt[4]{\frac{254\ (11^3)\ .069}{8,800,000\ (.002)}} = 1.073 \text{ say } 1\ 1/16 \text{ in.}$$

TUBE ON WHICH ROCKER ARM TURNS.

Boyd's Strength of Materials, page 176, equation 6 solved, gives for a beam fixed at the ends with concentrated load at the middle.

$$E\ I\ y = Mo\ \frac{x^2}{2} + \frac{Vo\ x^3}{6} \quad \text{where } Mo = \frac{Pab^2}{L^2}.$$

Solving this equa. for the case where $a = b = x = \frac{L}{2}$ and substituting the moment of inertia $\frac{d^4}{64}$ of a solid circular section for I and solving for d we get

$$d = \sqrt[4]{\frac{64\ PL^3}{192\ (\pi\ E\ y)}}$$

where d = diam. shaft

P = load

L = length between bearings = 4.375"

E = Modulus of elasticity = 30,000,000

y = Allos deflection = .002 in.

Our greatest load on the rocker arm tube will be when the cam starts to raise the valve.

From scaling off the drawing we find we have a load of 254 lb. at 28° to horizontal (cam on follower), and a load of practically 254 lbs. acting vertically upward. Solving to get the resultant load on the rocker arm shaft we have $R = \sqrt{(254 - 254 \sin 28^\circ)^2 + (254 \cos 28^\circ)^2}$

$= 262$ lbs.

$$\text{so } d = \sqrt[4]{\frac{64\ (262)\ 4.375^3}{192\ (\pi)\ 30{,}000{,}000\ (.002)}} = .444 \text{ in.}$$

However it is desired to use a hollow tube. Let us see if a 1/2" tube with a 1/4" hole has a section modulus equal to that of the .444 in. solid tube.

$$\frac{\pi d^3}{32} = \frac{\pi D^3}{32} - \frac{\pi d_1^3}{32}$$

$$d^3 = D^3 - d_1^3$$

$$.444^3 = .5^3 - .25^3$$

$$.0875 = .125 - .25 = .1094$$

The hollow tube is the stronger and will give even less deflection than the solid tube. The best timing of the valves for a motor of this speed (up to 5000 R.P.M.) to compromise between high efficiency and power at high speeds and smooth operation at low speeds seemsto be -

Let inlet cam open 15° past top dead center,

Let inlet cam close 40° past lower dead center,

Let exhaust cam open 48° before lower dead center,

Let exhaust cam close 10° past top dead center.

Inlet cam:-

Heldt 1 - 264.

R_1 = radius of cam at base = .625"

r = radius of top circle = .0625

L = left = .312

D = Center top circle to center of shaft =

$R_1 + L - r = .6250 + .3125 - .0625 =$

.8750.

ϕ = Angle of opening on camshaft =

$$\frac{180 + 40° - 15°}{2} = 102.50°$$

$$\frac{\phi}{2} = 41.25°$$

R = Radius of flank circle =

$$\frac{D^2 - r^2 + R_1^2 - 2R_1 D \cos\frac{\phi}{2}}{2(R_1 - r - D\cos\frac{\phi}{2})} =$$

$$\frac{(.8750)^2 - .0625^2 + .625^2 - 2(.625)\ .875\ (.\cancel{124})}{2(.625 - .0625 - (.8750)..626)} = 16.14 \text{ in.}$$

Exhaust cam:-

$$\phi = \frac{180° + 48° - 10° = 218°}{2} = 109° \text{ on camshaft.}$$

$$\frac{\phi}{2} = 54.5° \quad \cos\frac{\phi}{2} = .58$$

$R_1 = .625$

$r = .0625$

$P = .8750$

$$R = \frac{.8750^2 - .0625^2 + .625^2 - 2(.625)\ .875\ (.580)}{2\ (.625 - .0625 - (.875).58)} = 4.66''$$

Heldt 1 - 275.

Exhaust cam width = 1/3 equivalent diameter of valve equal in area to the two valves. = 1/3 (1.94) = .647 in.

Allowing 1/16" for oil hole in follower gives .647 + .0625 = .7095 say 3/4".

Inlet cam width = 1/3 (1.375) + 1/16" for oil hole

= .458 + .0625

= .5205 say 9/16"

CAMSHAFT BEARINGS.

Practice is to make the length of the cam shaft bearings from 2 to 3 times camshaft diameter or from 1 1/8 (2 to 3) or 2 1/4" to 3 3/8".

Let us make front bearing 3 in. long

Let us make middle bearing 3 in. long

Let us make rear bearing 2 1/2 in. long

GEARS DRIVING CAMSHAFT, WATER PUMP, GENERATOR, and BEARINGS.

Horse Power to Drive Camshaft.

Consider the work done in one revolution.

Load to open one exhaust valve = 127.1

Load to open eight exhaust valves = 8(127.1) = 1016.8

Load to open one inlet valve = 50.45

Load to open eight inlet valves = 8(50.45) = 403.60

Total load = 1465.40.

Work per rev. = total load (lift)

= 1420.40 .5/16 .1/12

= 370 ft. lb.

$$H_1P_1 = \frac{\text{work per rev. (R.P.M.)}}{33{,}000} = \frac{370(1500)}{33{,}000} = 1.682$$

Allowing 20% for friction loss

$$H_1P_1 = 1.682\ (1.20) = 2.01840$$

$$\text{Torque} = \frac{63{,}000\ (H_1P_1)}{\text{R.P.M.}} = \frac{63{,}000\ (2)}{15{,}000} = 84 \text{ in. lb.}$$

Front End Gears.

Assume diam. of 3" for gear on crankshaft

Assume diam. of 2" for lower gear on vertical shaft

Assume diam. 1 1/4" for upper gear on vertical shaft

Assume diam. 3 3/4" for gear on camshaft.

Marks 729 - 30

Consider gear on crankshaft and the one on vertical shaft meshing with it.

Let N_b = R.P.M. of crankshaft = 3000

N_c = R.P.M. of vertical shaft = 4500

R_b = Radius of crankshaft gear = 1.5 in.

R_c = Radius of vertical shaft gear = 1 in.

X_c = ∠ of true pitch line to axis of gear D

X_b = ∠ of true pitch line to axis of gear D

$$\frac{N_c}{N_b} \quad \frac{1.5}{1} \quad \frac{\cos X_h}{\cos X_c}$$

$\cos X_c = \cos X_d$

but $\cos X_c = \sin. (90 - X_c) = \sin. X_d$

$\sin X_c = \cos X_c$

$X_c = 45^\circ$

$X_b = 90^\circ - 45^\circ = 45^\circ$

Let us now consider the camshaft gear B and the vertical shaft gear A. Notation as before

$$\frac{N_A}{N_B} = \frac{R_B \ \cos X_B}{R_A \ \cos X_A}$$

$$\frac{4500}{1500} = \frac{1.875}{.625} \quad \frac{\cos X_B}{\cos X_A}$$

$$1 = \frac{\cos X_B}{\cos X_A}$$

$X_B = X_A = 45^\circ$

Let us check the gears for strength.

Marks 732.

Load on gear B = $\frac{84 \text{ in. lb.}}{1.875 \text{ in.}}$ = 44.8 lb.

Speed of pitch line (apparent) = $\frac{1500\ (1.875)}{12}$ =

234 ft. per min.

f = 12,000 (table 21 cast steel gears)

X = pitch line angle = 45°

b = width of face = $\frac{5}{16}$ in.

P = circular pitch

Let us try diametral pitch of 8

$p = \frac{\pi}{8}$

Unwin's formula for helical gears,

Load = w = .0833 b p f cos. ^2X

$= .0833 \left(\frac{5}{16}\right) \frac{\pi}{8} (12000)\ .707^2$

= .614#

Oversize gears are to be used to make the radial loads and thrust loads smaller and to obtain a fairly large number of teeth per gear in order that the gears may run silently.

Load on teeth of upper gear on the vertical shaft = 44.8 lb. Barring the thrust load due to friction which would be small, 44.8 lb. is the thrust to be taken care of in the vertical shaft and camshaft.

This thrust load is greater than the thrust loads on the lower gear of the vertical shaft or the thrust load on the pump and generator shaft. This shaft runs at 1 1/2 crankshaft speed and meshes with the crankshaft by a gear the same size as that on the lower end of the vertical shaft, and will ordinarily take less than 2 horse power.

Because of this a combined radial the thrust ball bearing suitable to take the loads here can also be used in the other places.

Let us take a new departure combined radial and thrust bearing 20 m.m. bore, (.787"), A = 1.027" in. inside shoulder, B. = 1.595 in. outside shoulder, diam. ; 2.05" width = .591". This bearing will take care of 125 lb. at 3000 R.P.M. so is of ample size.

On the vertical shaft use a modified square block type of universal joint, couple it with the shafts carrying the gears.

INTAKE AND EXHAUST MANIFOLDS.

Intake Manifold.

The design of the intake manifold is of considerable importance in the performance in an engine. The one factor is the smoothness in the operation of an engine, is that each cylinder receive a charge identical with that received by each of the other cylinders.

In the design, at attempt has been made to give each cylinder a charge of the same quality and the same volume at the same temperature or pressure, as that of each of the other cylinders. Sharp corners have been avoided in order that the friction loss be as small as possible.

Exhaust Manifold.

As much radiation surface as possible has been given the exhaust manifold so that the volume of the gas will be reduced, thereby reducing the pressure at the exhaust valves.

WATER PUMP AND COOLING SYSTEM.

A single bearing centrifugal pump is to be used.

Heldt 1 - 512 - 513.

D_1 = Impeller diameter,

V = Cu. in. piston displacement,

N = Ratio of pump speed to crankshaft speed,

C = Vane width at impeller circumference,

C_1 = Vane width at inlet circumference,

$$D_1 = 1.1 \sqrt[4]{\frac{V}{N}} = 1.1 \sqrt[4]{\frac{192.4}{1.5}} = 3.9 \text{ say } 4 \text{ in.}$$

$C = .12\,D = .12\,(4) = .48 = .48$ say 1/2"

$C_1 = .18\,D = .18\,(4) = .72$ say 3/4"

Water jacket space = 1/2".

An attempt has been made to provide for equal cooling for the cylinders by making the jackets as symmetrical as possible.

FAN CALCULATIONS.

Size and Capacity.

Heldt 499

The B.T.U. to be disposed of per minute approximately equals 68 (maximum Brake Horse Power). This figure is only a fairly close approximation as it is ordinarily used with an L or T head motor.

Assuming the maximum brake horse power at 48, the B.T.U. per minute to be disposed of = 68(48) =

3260.

Assuming maximum speed of 70 miles per hour, general practice has shown that .6 cu. ft. of air is required per B.T.U. entering radiator.

∴ cu. ft. of air per minute maximum speed = 3260 (.6) =

1956.0

Let us design a four blade face of the helical blade type.

Assume outside diameter = 16"

" inside " = 4"

" width of 2"

" pitch of 20"

" maximum speed of 4000 R.P.M.

" a 3" diam. for fan pulley,

" a 4" diam. for driving pulley,

Heldt 1 - 492.

$$V = N W \pi N \sum (R^2 - R_1^2) \quad \frac{2 \quad P\left(\frac{R + R_1}{2}\right)}{4 \quad {}^2\frac{R + R_1}{2}^2 + P^2}$$

$R^2 - R_1^2$	$\frac{2 \; x \; P}{4 \; 2 \, x \, 2 + P^2}$		$(R^2 - R_1^2) \frac{2 \; x \; P}{4 \; 2 \, x \, 2 + P^2}$
$\frac{8^2 - 7^2}{144} = .104$	.600		.0623
$\frac{7^2 - 6^2}{144} = .0902$	.628		.0566
$\frac{6^2 - 5^2}{144} = .0764$	.660		.0505
$\frac{5^2 - 4^2}{144} = .0625$	.686		.0429
$\frac{4^2 - 3^2}{144} = .0486$	.706		.0343
$\frac{3^2 - 2^2}{144} = .0348$	.713		.0248
		Total =	.2714

$V = 4 \frac{2}{12}$ 3.142 (4000) 2770'/ min.

The capacity is enough greater to take care of slip in the belt.

POWER TO DRIVE FAN.

Heldt 1 - 446.

$$H_1P_1 = \frac{Nn^3 \pi W P^2}{49{,}660{,}000{,}000} \left(R^2 - r^2 - \frac{P^2}{4 \quad 2} \log_n \frac{R^2 + \frac{P^2}{4 \quad 2}}{r^2 + \frac{P^2}{4 \quad 2}} \right)$$

Where N = no blades,

n = R.P.M.

w = width of blades in feet,

P = pitch in feet,

R = Radius to large circumference,

r = Radius to small circumference,

$N = 4 \quad n = 4000 \quad w = \frac{2}{12}$ ft. $\quad P = \frac{20}{12}$ ft.

$R = \frac{8}{12}$ ft. $\quad r = \frac{2}{12}$ ft.

The H. P. solves out to 6.57.

$$\text{Torque} = \frac{63{,}000\ (H.P.)}{N} = \frac{63{,}000\ (6.57)}{4{,}000} = 10.36 \text{ in. lb.}$$

$$\text{Net driving tension} = \frac{10.36}{1.5} = 6.9 \text{ lb.}$$

DESIGN OF FAN BELT.

Let us design a V belt of built up construction with sides at an angle of 28°.

Machine Design - Leutwiler - 173.

$$T_1 - T_2 = T_1 - \frac{WV^2}{g} \quad \frac{e^{\frac{U\phi}{\sin.B}} - 1}{e^{\frac{U\theta}{\sin.B}}}$$

$T_1 - T_2$ = difference in tension,

T_1 = Maximum tension,

W = wt. of belt per foot of length,

V = Velocity of belt in ft. per sec.

U = Coefficient of friction,

ϕ = Arc of contact in radians,

B = angle of sides of belt,

$T_1 T_2$ = 6.9

Let mean width of the belt be 1/2 in. and the thickness 1/2 in. Assume blocks to weigh 1/4 of belt proper.

Wt. belting = .035 lb. per cu. in.

$\therefore$ Wt. per ft. of belt = 1.25 (1/2.1/2) 12 (.035) =

= .1313 lb.

$V = 4000\ (\pi \frac{3}{12}) = 3142$ ft. per min. = 52.36 ft. per sec.

U from table = .50

$$6.9 = T_1 - .1313\,\frac{(52.36)^2}{(32.2)}\ \frac{e^{\frac{.5\,(\pi)}{.242}}}{e^{\frac{.5\,(\pi)}{.292}}}$$

$$6.9 = T_1 - 11.1\quad \frac{649}{650}$$

$$6.91 = T_1 - 11.1$$

$$T_1 = 18.01 \text{ lb.}$$

$$f_s = \frac{\text{load}}{\text{area}} = \frac{18.01}{1/2.1/2} = 72.04 \text{ lb. per sq. in.}$$

f_s is well above 400 lb. per sq. in. so the belt is of ample proportions.

LUBRICATION

Quantity of Oil Required.

Allow 25 cu. in. of oil per min. per sq. in. of projected bearing area.

Bearing area crankshaft = 2(4(2 3/16) + 2 5/8 + 2 3/4 + 3 5/8)

= 2 (17.77) = 35.54 sq. in.

Bearing area camshaft = 1.0625 (2.75 + 3 + 2.5) = 8.76 Sq.In.

Bearing area vertical shaft and pump shaft = .75 (3) 1 =

2.25 sq. in.

Total area = 35.54 + 8.76 + 2.25 = 46.55 sq. in.

Cu. In. of oil per min. for bearings = 46.55 (25) =

1163.75 cu. in. per min.

There are 12 - 1/16 in. holes in the rocker arms, lubricating the camshaft.

Assume a 5 lb. per inch head.

$$\emptyset = C_d \; a \; 2 \; gh \; (12)(60)(1728) =$$

$$.61 \left(\frac{\pi}{4}\right)\left(\frac{1}{16^2}\right)\left(\frac{1}{144}\right)\left(\sqrt{64.4 \; (5) \; 2.3}\right)(12) \; 60 \; (1728) =$$

441 cu. in. per min.

This figure is liberal enough to take care of the rocker arm bearings also.

Total quantity of oil per minute = 1163.75 + 441 = 1604.75

cu. in. per minute.

DESIGN OF PUMP.

Let us design a pump of the gear type.

Heldt 1 - 332.

Q = Quantity in cu. in. per minute

D = pitch diameter,

F = Face of gear,

N = R.P.M. of pump,

P = diametral pitch.

$$Q = \frac{2\pi DF.N}{P} \text{ cu. in. per min. or}$$

$$F = \frac{P.Q}{2\pi D.N}$$

Assume a pitch diameter of 1.25" and a diametral pitch of 8

$$F = \frac{8\ (1600)}{2\ (1.25)\ 4500} = .363'' = \text{say } .5''$$

MOUNTING OF ELECTRICAL UNITS.

The motor is designed to take a two unit system with the distributor mounted on the end of the Generator. S.A.E. standard mountings provide for the starter and generator.

A tube is provided for the protection of the ignition cables.

SUSPENSION OF MOTOR FROM FRAME.

The motor is provided with three point suspension, two faces on the flywheel housing and one point in front.

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Oct 24 '50

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Poket empty as of
3/1/11