M.E.
Thesis rack

THE POSSIBILITY OF CHANGING A
1916 HENDERSON MOTORCYCLE ENGINE
FOR
LIGHT PLANE WORK.

Thesis M.E.Dept., Mass. Inst. Tech. Sept. 3, 1924

Van J. Weaver.

Cambridge, Mass.,

September 3, 1924.

Professor A. L. Merrill,
Secretary of the Faculty,
Massachusetts Institute of Technology.

Dear Sir:

In accordance with the requirements for the Degree of Bachelor of Science in Mechanical Engineering I submit herewith thesis entitled, "The Possibility of Changing 1916 Henderson Motor Cycle Engine for Light Plane Work."

Respectfully,

V. J. Weaver.

142932

ACKNOWLEDGEMENT.

I wish to acknowledge hereby my deep appreciation of the assistance and suggestions given by Prof. Fales, Prof. Smith, Mr. Chayne, Mr. English and Mr. Hardy.

V. J. Weaver.

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ILLUSTRATIONS.

OBJECT.

The purpose of this experiment was to determine whether or not the engine from a 1916 Henderson motorcycle could be adapted to use in Light Aeroplane work. The features desired in this motor were -- Reliability -- Light weight per horsepower -- A maximum horsepower of from 15 to 20 at a speed which would make possible the use of a direct driven propeller of the Curtiss-Reed forged duraluminum type.

To attain these qualities the expense and time required to make the alterations were to be kept as low as possible and were to be of such a nature that the ordinary mechanic could readily carry them out with the tools usually found in machine shops.

DESCRIPTION OF THE ENGINE.

Name	1916 Henderson Motorcycle.	
Type	Vertical Air-Cooled 4-Cycle.	
Cylinders		
Bore	2.5 in.	
Number	4	
Type	L-head Flanged	
Material	Cast Iron.	
Crankcase	Aluminum Flywheel housing and transmission case and integral part of same.	
Pistons	Drilled for lightness. Wrist pin set screwed to piston. Oil groove at bottom.	
Rings	Two 1/8 in.	
Material	Cast Iron.	
Crankshaft		
Type	Three bearing. Note : A fourth bearing in the transmission housing served to support the flywheel and transmission bevel driving pinion on end of crankshaft.	
Bearing sizes		
All 1 in. diameter	Front	1 1/4 in. long.
	Center	1 1/2 in. ''
	Rear	1 1/2 in. ''
	Pinion	1 in. ''
Material	Steel	
Connecting Rods.		
Length	5.5 in. center to center.	
Piston End.	Bushed for 1/2 in. wrist pin.	
Crank End.	Babbit lined, bronze backed bearing 7/8 in. diameter 1 in. long.	
Material	Steel Drop forged.	
Flywheel		
Diameter	10 in.	
Thickness	15/16 in. Bolted to plate flange on crankshaft.	
Material	Cast Iron.	

Camshaft	
Type	Two bearing.
Length	15 in.
Diameter	9/16 in.
Material	Steel.
Valves	
Intake	Removable cage on top of cylinders.
Operation	Rocker arm thru push rod.
Diameter	1 1/4 in.
Stem	1/4 in.
Lift (Approx)	5/32 in.
Exhaust	
Operation	Tappet directly
Diameter	1 1/4 in.
Stem	1/4 in.
Lift (Approx)	7/32 in.
Timing	Head gear train.
Magneto	Berling High Tension Model N-44
Spark Plugs	Mosler Aviation 1/2 in. Met.
Carburetor	Schebler Model HX -176
Intake Manifold	Cast Iron Baffled
Flange openings	7/8 in.
Exhaust Manifold	Pressed Steel.
Flange openings	1 in.

Transmission.

Two bearing propeller shaft at right angles to crankshaft in horizontal plane. Propeller shaft carries clutch and driving sprocket on one end and starting ratchet on other. Propeller shaft driven thru 17 to 28 bevel gears.

Note : Poorly designed as the thrust acts down on this shaft and is taken by studs screwed into the top of the aluminum transmission case. It is understood that this caused a great deal of trouble as these studs would pull out. Later models put driven bevel gear on propeller shaft on the opposite side of driving pinion on crankshaft. The thrust is then upward against the transmission case.

Chain drive from clutch sprocket to rear wheel. Ratio of sprockets 17 to 41

Compression Ratio (After Regrinding) 2.941

Piston Displacement ' ' ' 61.525 cu.in.

Approximate speed of engine based on motorcycle speed of

50 miles per hour with wheels 28 in. in diameter = 2400 r.p.m.

INSPECTION, ASSEMBLY and TEST of ORIGINAL ENGINE.

The engine in its original condition was disassembled and given a careful inspection for possible weaknesses and breaks which might have occurred during its previous service. Beyond the fact that one of the transmission propeller shaft bearing studs was pulled loose and that there was a slight crack in the upper half of the crankcase, apparently having been caused by the letting go of the connecting rod of No.1 cylinder, there seemed to be nothing seriously wrong. The loose stud mentioned above was tightened up, a nut put on its upper end, and then the engine was reassembled.

A few preliminary runs indicated the necessity of procuring the following new parts ----

New Intake valve stem guides.

New Magneto drive shaft

New Carburetor.

It was necessary to make the new guides and magneto drive shaft because the Henderson agency did not carry parts for a model so old as this one.

NEW INTAKE VALVE STEM GUIDES.

The intake valve stem guides screw into the top of the cast iron intake valve cages. There was therefore no difficulty in removing the worn guides. The new guides were made on a lathe, using 7/16 in hexagonal cold rolled steel and the old guides for patterns.

Since the valve stems themselves were considerably worn it was first necessary to center them carefully, turn the stems down from 1/4 in. to 7/32 in., and then to square up the valve itself, making it true with the stem. The small diameter of the valve stem made it impossible to turn the guides themselves down by use of a mandrel and therefore, the hole in the new guide was not exactly true with the threads on the outside by which the guide was positioned in the cage. This necessitated truing up the seats in the cage with the new guides. A Ford valve reseating tool was used for this purpose, it being only necessary to turn down the stem on the reseating tool to 7/32 in.

NEW MAGNETO DRIVE SHAFT.

The magneto drive shaft carries on its front end a gear, which is one of the timing gear train. The shaft itself is carried in a bushing of bronze, which is fastened to the back of the timing gear housing with four screws. The rear end of the shaft fits into a sleeve fastened to the magneto armature shaft. The sleeve and the drive shaft are pinned to gether by a taper pin running thru both of them. There is therfore no adjustment of the magneto timing once the gear is keyed to the driving shaft and the shaft pinned to the magneto sleeve.

This drive shaft and the bronze bushing were very badly worn. The bushing was first turned out as much as necessary and the new driving shaft made to fit the new hole. In other respects the new shaft was just the same as the old one. A great deal of care is to be exercised in putting the magneto in position. It is necessary that perfect alignment be secured since there is no flexible joint in this driving system. If this alignment is off the bronze bearing, which is not amply provided with lubrication, will heat up very rapidly.

NEW CARBURETOR.

The original carburetor was very badly worn and in need of complete renovation. Investigation, however, disclosed the fact that a new carburetor could be obtained from the manufacturers' agent for a trifle more than it would cost to recondition the old one. It was desired to use a Zenith carburetor but the cost of this type and the delay in procuring the same decided in favor of procuring a new Schebler. (Plate A #1.)

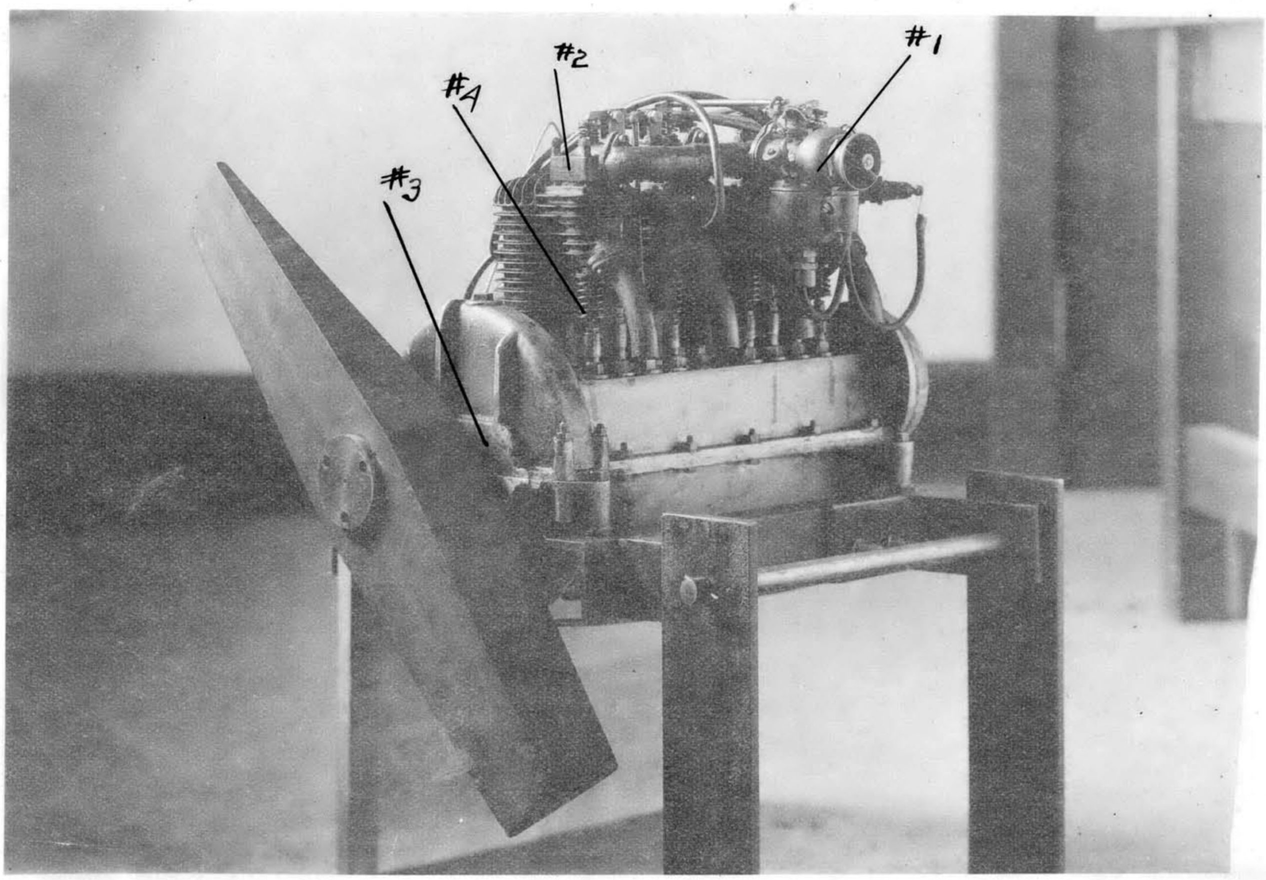


Plate A

TRIAL and CHANGES.

After these new parts were installed, a great deal of trouble was experienced in eliminating air leaks in the intake manifold connections and under the intake valve cages, which are fastened to the cylinders by four studs. (Plate A #2) Even when these leaks were entirely eliminated, Cylinder No. 3 refused to function properly, apparently not getting the proper or sufficient mixture. (Note : Cylinder No. 1 nearest timing gears.)

OLD and NEW INTAKE MANIFOLDS.

The original manifold was long and of poor external design, so it was decided to cut it in half and to see what the interior design was like. When this was done (Plate B #1), it was found that the baffles were so arranged that the mixture could hardly be expected to have the proper distribution. The interior, too, was very rough and offered considerable resistance to the flow of the mixture from the carburetor to the cylinders.

Keeping in mind the fact that it would not be expected of the ordinary mechanic to design and construct by the electrical deposition of copper on a wax and graphite form a scientifically designed manifold, it was decided to try a manifold made up of copper tubing, brass plates for flanges and brass pipe fittings of the type shown in Plate C #2. This manifold required two elbows, three tees, enough plass plate to make the flanges for the intake valve cages and the carburetor flange and enough copper tubing to join the pipe fittings together. All of the

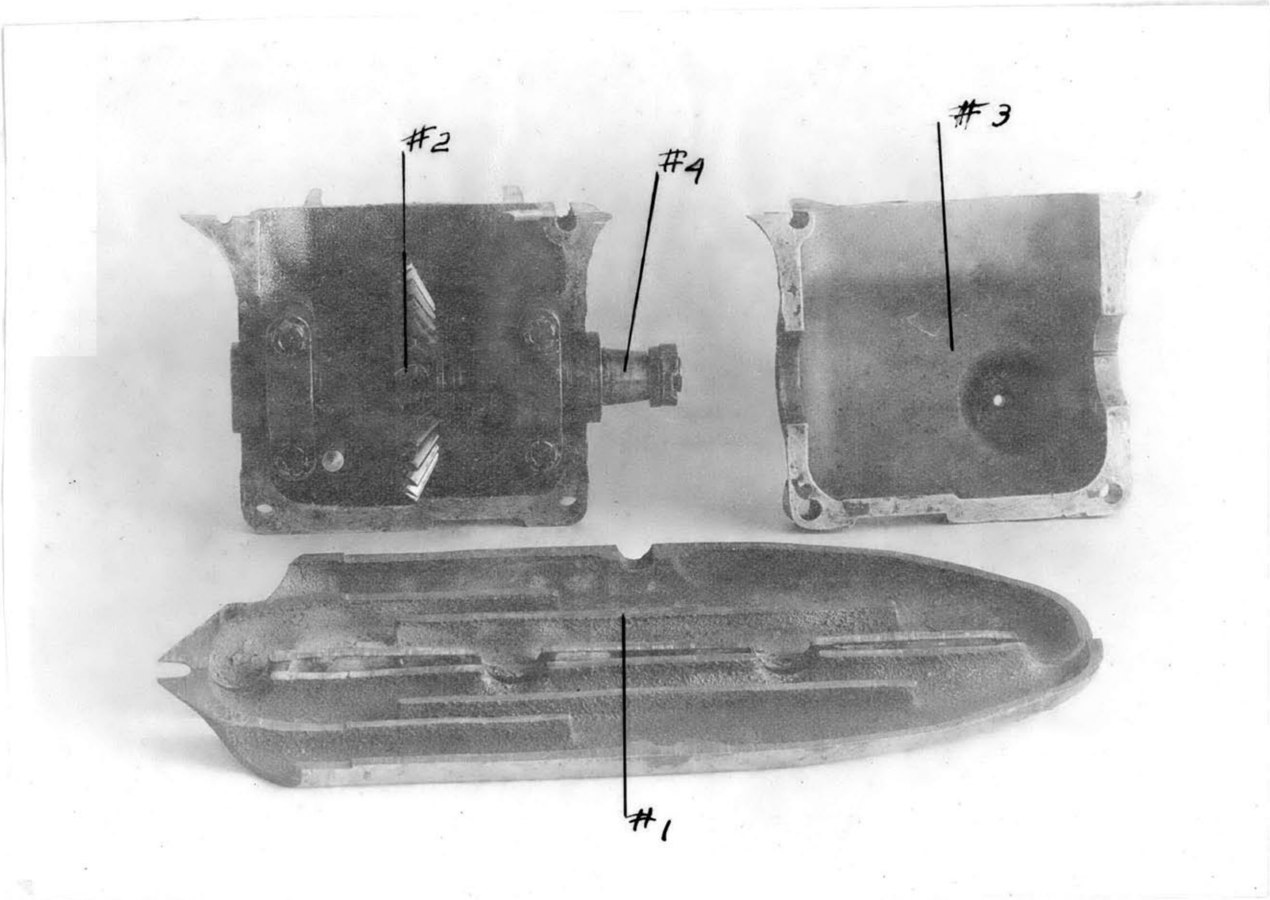


Plate B.

joints -- eleven in number -- were carefully fitted by lathe turning and then brazed together, making a tight unit. (Plate C #1 and Plate D #3).

No attempt was made to finish off the exterior after the brazing was completed. However, the interior was smoothed up by placing a lot of steel slugs of various shapes within the m manifold and then covering the openings with two boards. This assembly was then fastened to a slowly revolving pulley and after several hours of ''tumbling'' in this manner, the interior was smoothed up very nicely. Although the turns in this manifold are of a radius much shorter than that desired -- 2 1/2 times the diameter of the inside of the pipe being the radius usyally employed -- it was the best that could be done with the materials at hand and the conditions imposed. This manifold represents a lot of careful lathe work due to the length of time required to chuck up the awkward elbows and tees in the ordinary lathe chuck.

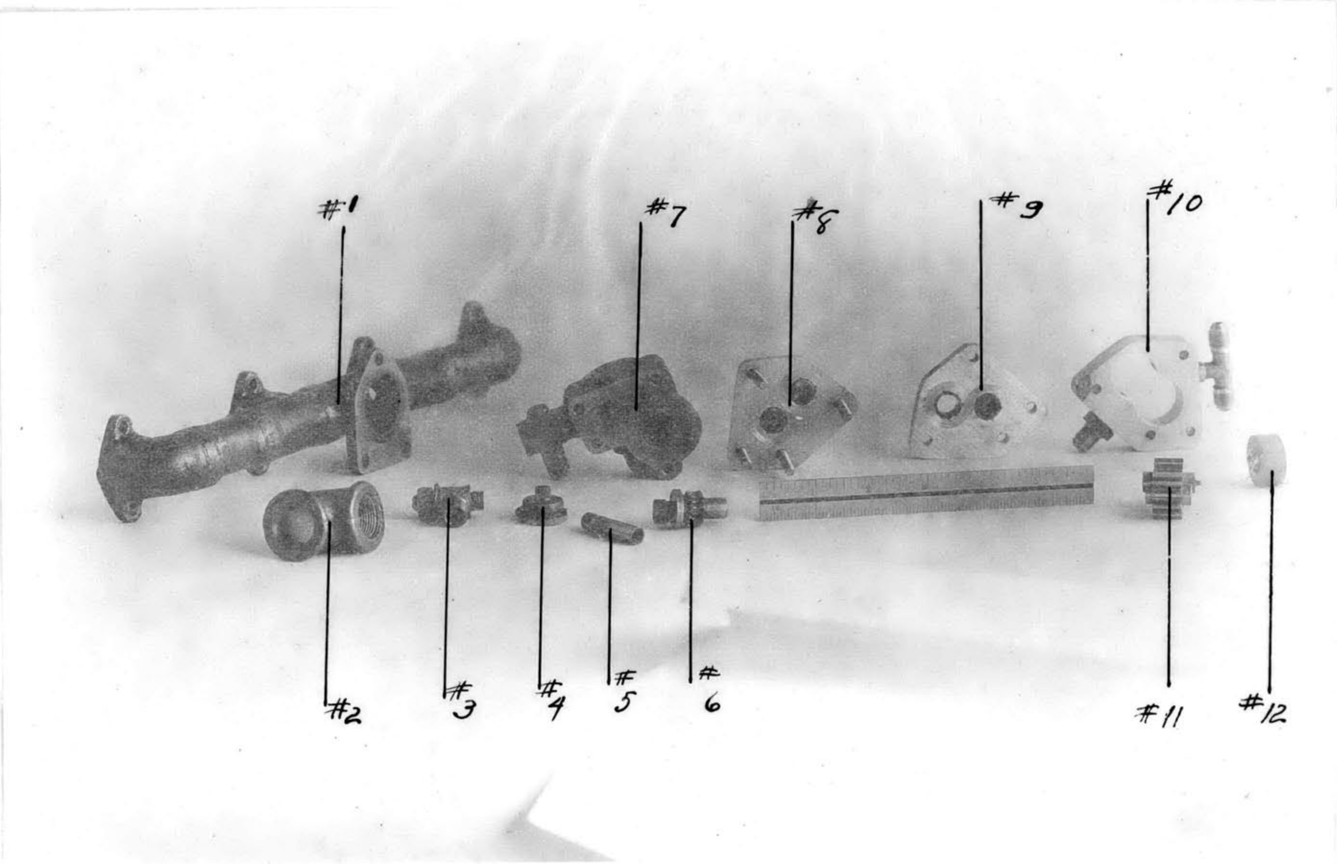


Plate C.

RUN WITH NEW MANIFOLD.

With this manifold and with all joints carefully shellaced to prevent air leaks, the motor ran very well up to moderate speeds byt very poorly at higher speeds. Further investigation seemed to indicate the desirability of making the following changes before attempting any further tests ----

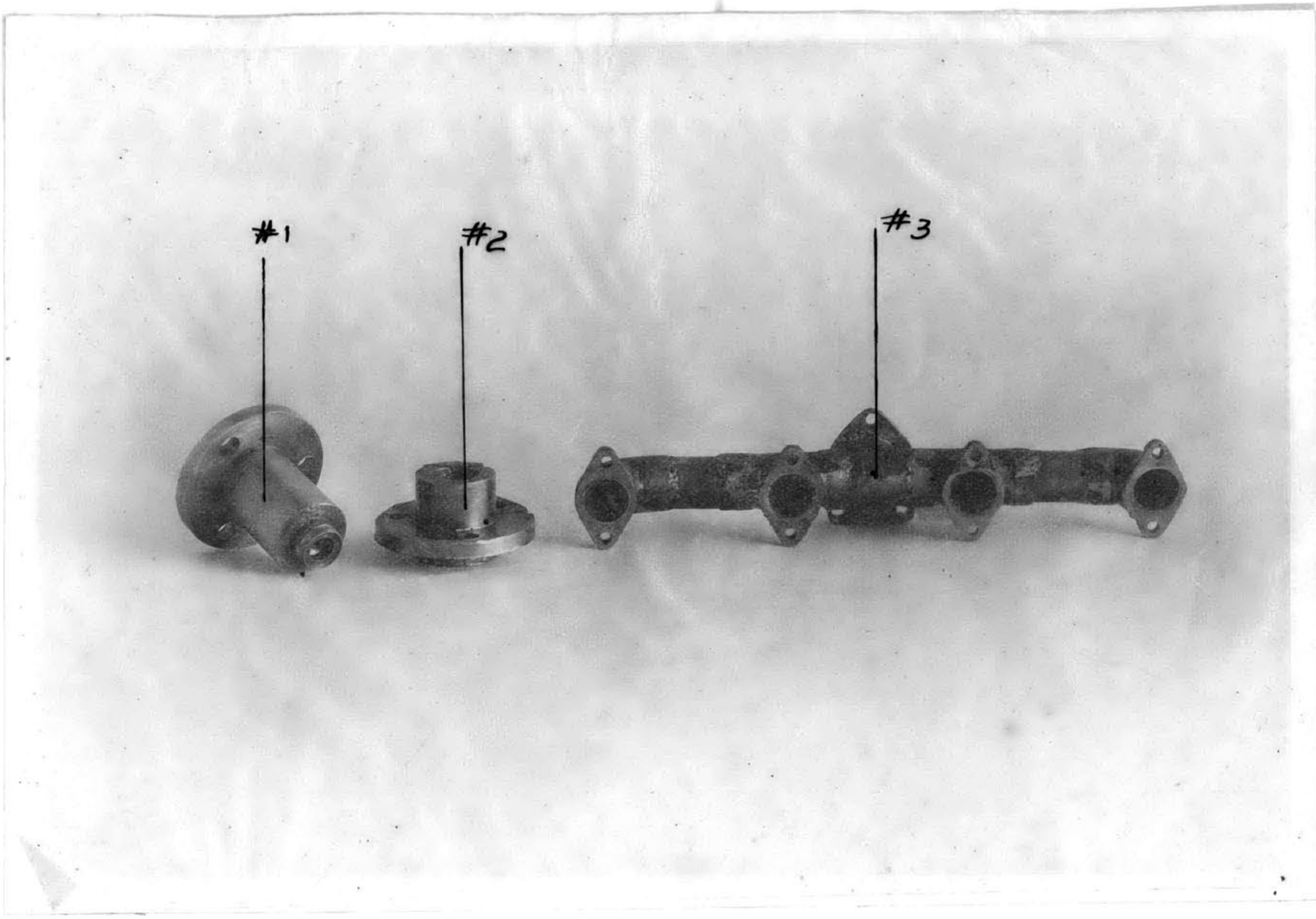


Plate D

1. New oiling system.
2 New exhaust valve springs.
3. Regrind cylinders - new pistons, rings and pins.
4. Different type dynamometer.
5. Provide club brake.
6. Eliminate flywheel and transmission.
7. Provide thrust bearing.
8. Provide hub for club.

OILING SYSTEM. OLD and NEW.

The lubrication of the engine is by the splash system small projections on the connecting rod bearing caps dipping into the oil in the pockets in the crank case (Plate F) and splashing it all over the parts. This system works out very well and is used in many racing automobiles staisfactorily. Of course the proposition of using a wet sump motor in an aeroplane is not thought highly of to-day, but, it must be remembered that this motor is not intended for use in a machine where acrobatic work -- upside-down flying and the like -- will be done. A light plane is not designed for that purpose. Its maneuvers on the whole are gentle -- no steep banks, dives, glides, etc., so that this wet sump engine should function nicely in this respect.

Teh old oil pump (Plate E #1) was of the piston and cylinder type. The Piston (Plate C #5) was driven up and down in the cylinder by an eccentric mounted on a shaft which was in turn driven by a small worm and wheel system. (Plate C #6) The oil was sucked from the crankcase into the cylinder (Plate C #7), check-valved and forced out thru a pipe delivering it to the top of the timing gear housing. (Plate G #1) Here it was allowed to flow down over the timing gears and thence into the crankcase. The rapidity with which this pump circulated the oil was very poor and after a few minutes of fast running the oil became very hot and thin.

The new pump (Plate E #3) was designed to keep the oil in rapid circulation and to thereby attain some degree of cooling from it. The new system consisted of a pipe (Plate F #1) running from the bottom of the flywheel housing to the suction side of the pump (Plate E #4) and thence from the delivery side of the pump (Plate E #5) to a line running to the top of the timing gear housing as before and another line (Plate F #2)

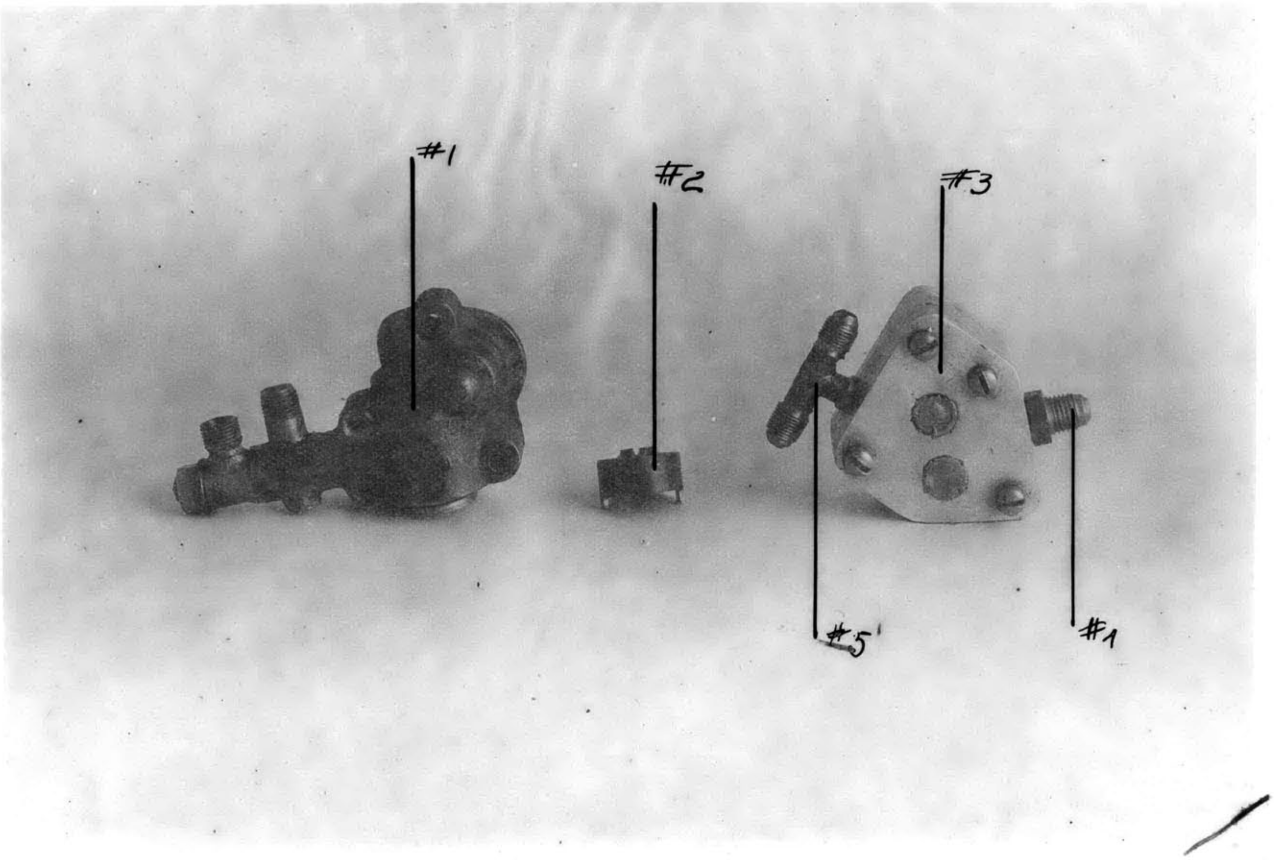


Plate E.

running to the bottom of the crankcase and connecting each of the four oil pockets in the latter. (Plate F)

The new oil pump was of the gear type. The two gears were 12-tooth, having 3/4 in. pitch diameter, 1/2 in. faces and a 3/8 in. hole in the center thru which axises were run. (Plate C #11). The housing housing consisted of three pieces of aluminum -- a front and back plate, each 1/4 in. thick and a center plate 1/2 in. thick. The front plate (Plate C #8) and the back plate (Plate C #9) each have two brass bushings which carry the gear axises. The center plate (Plate C #10) has two overlapping holes of such size that the two gears when in mesh, fitted them tightly. Holes were drilled in the edges of this piece and tapped to take the suction (Plate E #4) and the delivery (Plate E #5) pipe fittings. The shape of the pump housing was made to correspond with the holes in the timing gear housing cover to which the pump was attached by four screws. (Plate G #2) The part of the old pump drive, which originally carried the worm for the worm and wheel drive, was altered to drive the new pump gears thru a tongue and groove connection (Plate E #2) -- the tongue being made on the end of the pump driving gear shaft (Plate C #11).

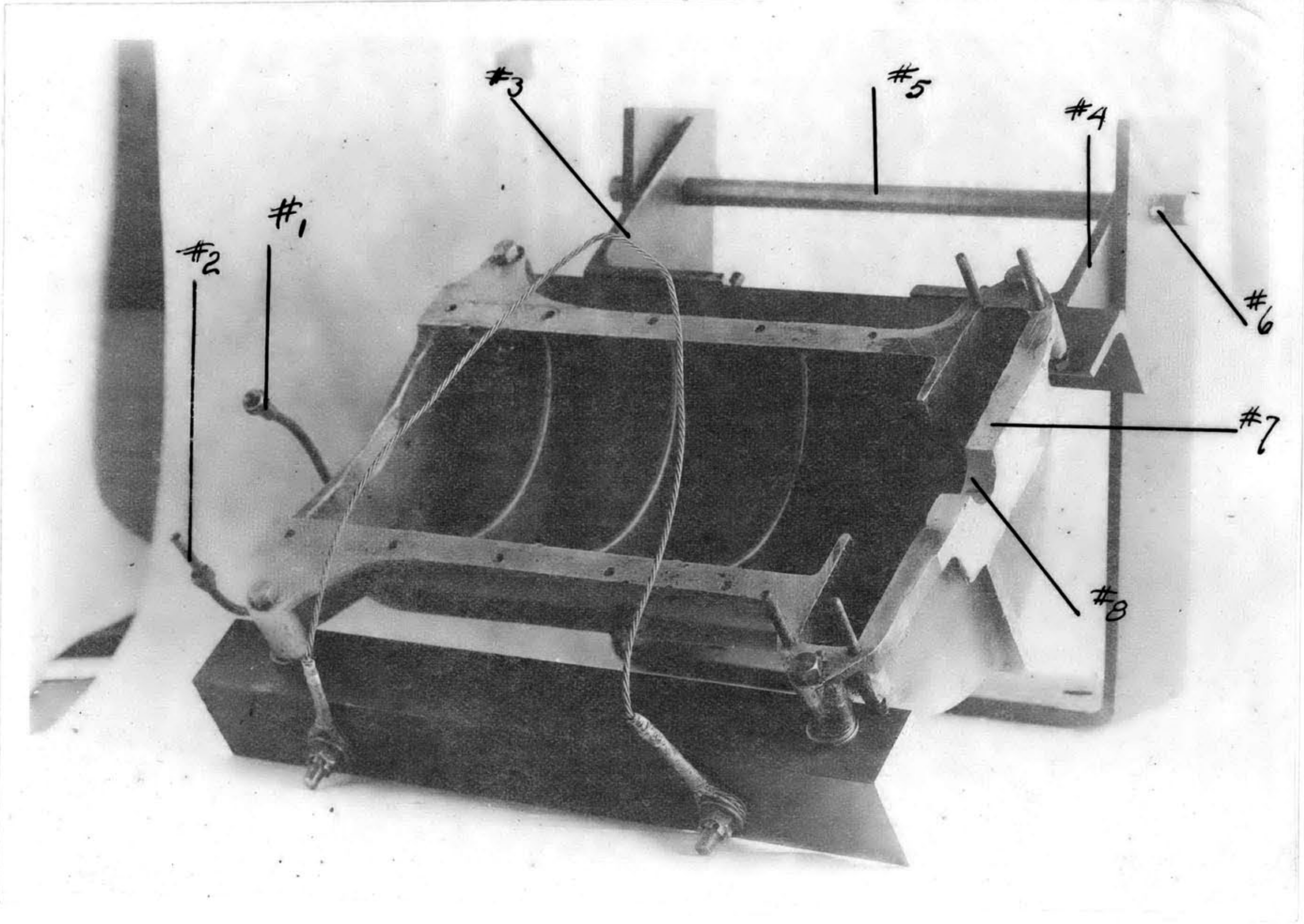


Plate F.

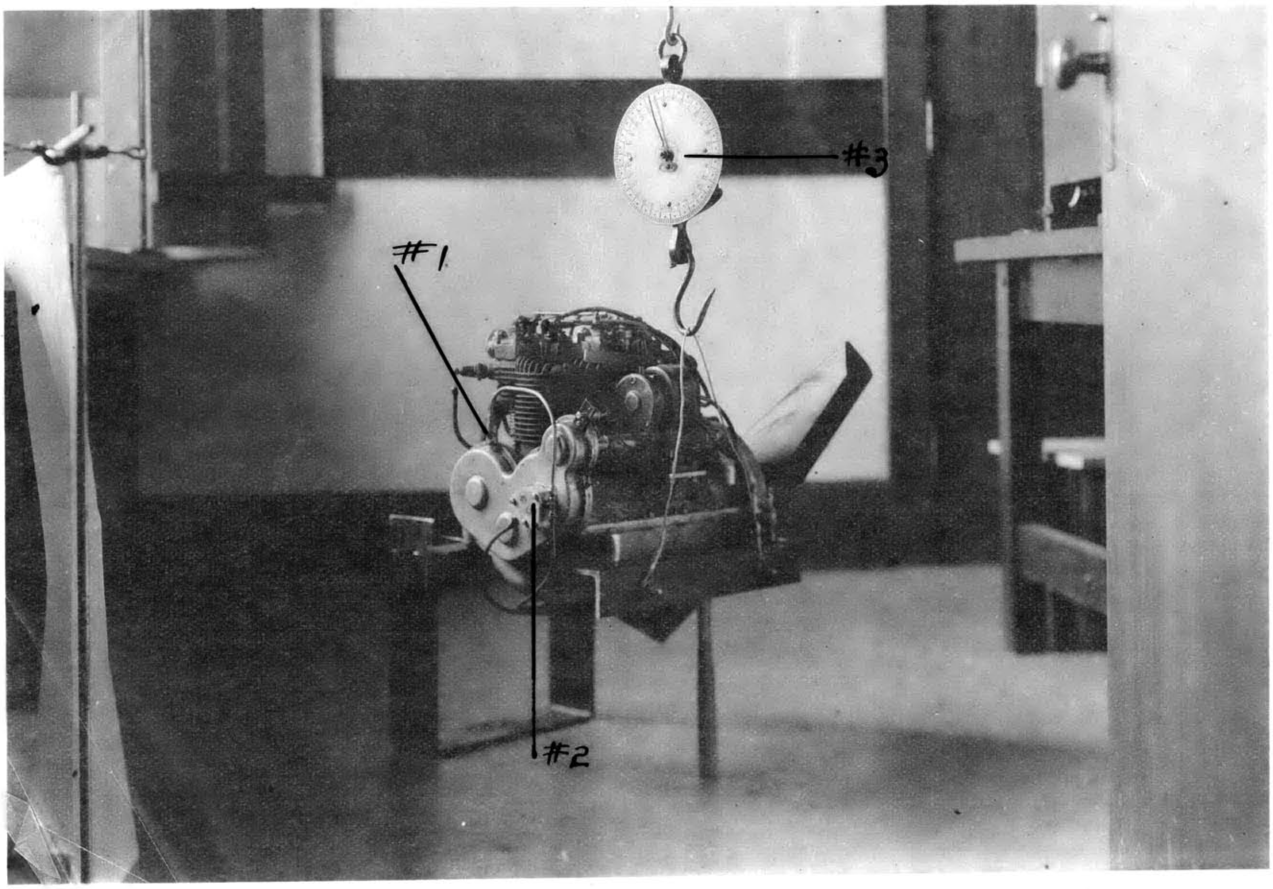


Plate G.

This new oil pump has considerably fewer parts than the original one and circulates the oil much more rapidly, besides being lighter and just as easily removed and examined.

NEW EXHAUST VALVE SPRINGS.

The original exhaust valve springs were of such strength that the valves would open under 15 lbs. pressure. This was insufficient to give the valves quick and positive closing at high speeds and therefore stronger springs were sought. The Ford valve spring was cut in two and used to make two exhaust valve springs for this engine. With these springs the valves opened at 23nlbs. pressure.

A HIGH SPEED DYNAMOMETER.

By G. E. Scholes, M.Sc.

FOR determining the brake horse power of an engine running at high speeds, the types of brake available are somewhat limited. As a rule, manufacturers use either a hydraulic brake of the Heenan & Froude type, or a fan brake. The electrical dynamometer is also sometimes employed. An ordinary rope brake is rarely used, although satisfactory results are obtainable with engines running at lower speeds, and such a device is both cheap to manufacture and easy to apply. Against this type of brake the objection may be raised that the unbalanced net load which is carried by the shaft produces a bending movement.

In fig. 1 the gross load is W, the spring balance load w, and the net load is (W – w) lb. (neglecting the mass of the short lengths of rope). This net load has to be carried by the shaft and thus produces stresses due to bending.

The rope is carried on the rim of the flywheel, and in all engines using cast-iron flywheels the diameter is such that this rim speed is sensibly constant and in the region of 90ft. per second. For instance, an engine running at 200 r.p.m. would have a flywheel of approximately 8ft. diameter, while on an engine running at 1,500 r.p.m. the flywheel would have a diameter of about a foot. In each case the net load required on the brake per brake horse power developed would be the same, and therefore each shaft would be subjected to the same unbalanced force due to the rope brake when developing the same brake horse power. The stresses induced in the crankshaft due to this load would, however, be much greater in the high-speed engine, on account of the much smaller diameter crankshaft used, and it is this effect which makes the use of the ordinary rope brake inadvisable with high-speed engines. On the other hand, the rope brake has much to commend it, satisfactory service being combined with smooth running, and where its use is permissible it is in many respects ideal.

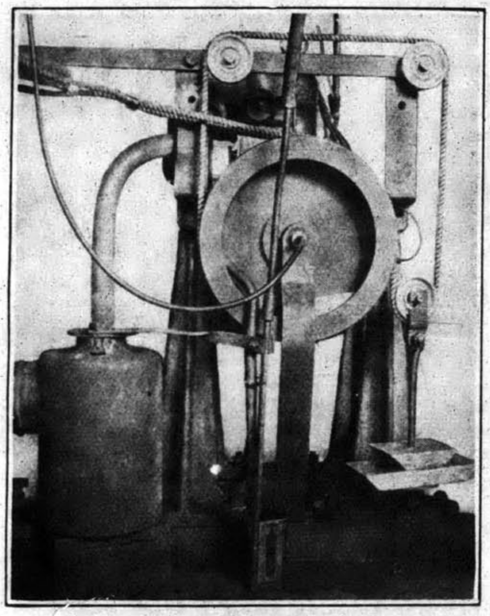


Fig. 4.

In order to eliminate the unbalanced force mentioned, the author constructed a brake in which the rope system was modified as shown in fig. 2. The behaviour of the brake is entirely satisfactory, and it should prove useful in providing a cheap and accurate means for determining the brake horse power of any petrol engine on the shaft of which it is convenient to fit a flywheel having a water-cooled rim.

From the illustration, fig. 2, it will be seen that a rope having one end attached at A is passed round the flywheel from B to C, over pulleys D, E, and F, and then round the flywheel from G to B, both ends of the rope being attached at A. These two ends are held apart by a small clamp to enable the rope CD to pass between them, the spring balance hook being attached to the clamp. The pulleys D, E, and F have roller bearing centres, being of the type used in ships' rigging, and the rope is of cotton ½in. diameter. In this arrangement the unbalanced force is only that due to the spring balance reading, and is so small that its influence may be neglected. By using two spring balances and winding the rope an additional half turn, as in fig. 3, the brake would be perfectly balanced as far as force is concerned, but this refinement is considered unnecessary.

The brake illustrated in fig. 4 has been in use for a considerable time on the flywheel of a four-cylinder petrol engine which has a normal running speed of 1,500 r.p.m. The photograph was taken with the engine under test at 1,100 r.p.m. with an exposure of half a minute. The clearness of the detail illustrates the smoothness with which the engine and brake are operating.

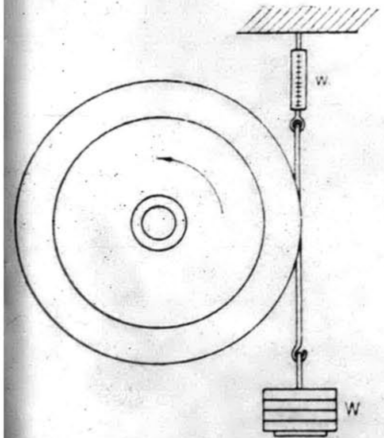


Fig. 1.

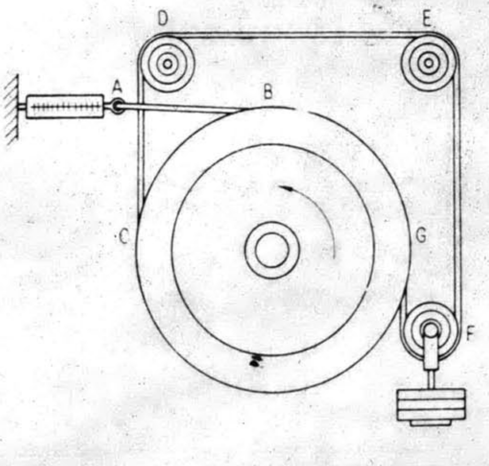


Fig. 2.

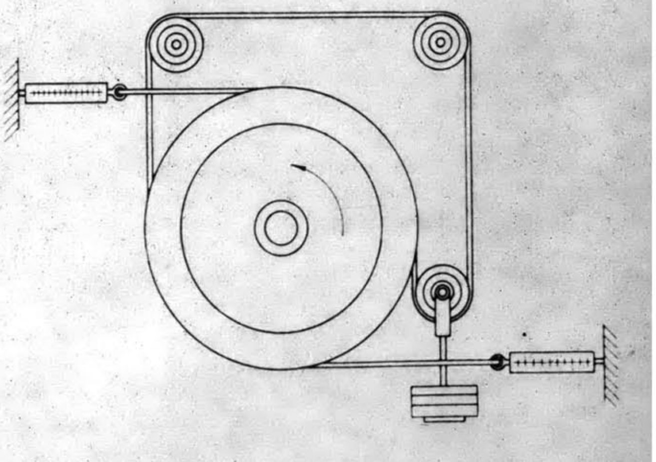


Fig. 3.

REGRINDING.

The job of regrinding the cylinders, fitting new pistons, rings and pins was done by the Dyer Co., Cambridge. It required about five days. The reground bore was 2.555 in.

DYNAMOMETER.

v Up tonthis time the changes made were of such a nature that they would not interfere with the use of the motor in a motorcycle again. It was thought that a very good idea of the suitability of the engine for light plane work could be obtained by running tests for speed and horsepower after making the alterations described. To make such tests it had been proposed to use a high speed rope dynamometer of the type described by G.E.Scholes in the Feb. 1923 issue of the Automobile Engineer. (See accompanying photostat.) It was planned to connect the pulley or drum of this dynamometer to the transmission propeller shaft (Plate B #4) with a special coupling and to provide a special bearing mounted on the engine carrier to support the overhung weight of the drum.

However, with the problem of cooling becoming more and more difficult as higher speeds and longer runs were made and keeping in mind the ultimate desire of using a direct driven propeller, it was decided to abandon this type of dynamometer, to sacrifice the future utility of the engine for motorcycle work - in event it proved unsatisfactory for light plane work - and to make those changes necessary for using a direct driven propeller.

Having reached this descision, it was logical that the use of a club for a brake and a cradle type of dynamometer should be decided upon. There were several advantages to be gained by using this combination, among which were the ease with which both club and cradle could be constructed. Then, too, the club, by giving it pitch, was a fine source of a cooling air stream for keeping the engine temperature down. There were of course disadvantages. The danger of a club revolving at 3000 r.p.m. was not to be overlooked especially with the controls necessarily so close. Also, the club type of brake affords one no method of varying the load it gives at a fixed speed. This was not so bad as might at first appear because if this motor were used for aeroplane work it would be run at or very near to its peak load all the time. It was therefore the horsepower output at peak load which was of paramount importance to determine.

CRADLE DYNAMOMETER.

The cradle dynamometer consisted of a piece of steel 42 in x 4 in. x 5/16 in. bent up into the shape of a U with a flat bottom. (Plates A F G) This U was bolted to the floor. Thru the arms of the U and thru two 6 in. x 4 in. x 5/16 in. pieces of angle iron, 2 1/2 in. long, bolted to the engine bearer on one side (Plate F #3 and #4) was run a 3/4 in. shaft. (Plate F #5) End play was prevented by a pin (Plate F #6) at each end of the shaft. On the opposite side of the engine and supporting the balance of the load was placed at first a Fairbanks platform scale and later a spring balance. (Plate G #3) With the engine running evenly on all four cylinders, the vibration was not enough to justify the expenditure of the time and money necessary to construct a dampening device. The frictional resistance of the bearings was neglected. The difficulty with the Fairbanks scale was that the air stream from the club kept blowing the weights from the weight hanger on the end of the scale beam. The resisting couple, giving a measure of the torque exerted by the engine was the product of the scale reading in pounds and the distance from the scale supporting point to the axis of the cradle, i.e., 18.5 in. or 1.54+ ft.

THE CLUB.

The club was made from a fine, straight-grained piece of White wood, free from knots and blemishes, 36 in.x5in.x3in. It was drilled, tapered and given pitch as shown in Plates A and G. With this pitch and a 12 in. electric fan ample cooling was attained.

ELIMINATION of FLYWHEEL and TRANSMISSION.

The elimination of the flywheel (Plate H #1) was a simple matter, requiring the removal of the transmission driving pinion (Plate H #2) on the end of the crankshaft and the loosening of the three bolts holding the flywheel on to the flange on the crankshaft.

The elimination of the transmission (Plate B #1 and #2 : Plate J #1 : Plate K #1 : Plate L #1) was easily accomplished, also, by sawing the aluminum casing in two just back of the flywheel housing (Plate L #2). This left a large opening (Plate J #2) in the lower half of the crankcase on the rear face of the flywheel housing. It was thru this opening that the oil from the front part of the crank case entered the transmission housing. It was necessary to build this hole up by aluminum welding. (Plate F #7) The upper part of the crankcase at this point, while not having an opening, was

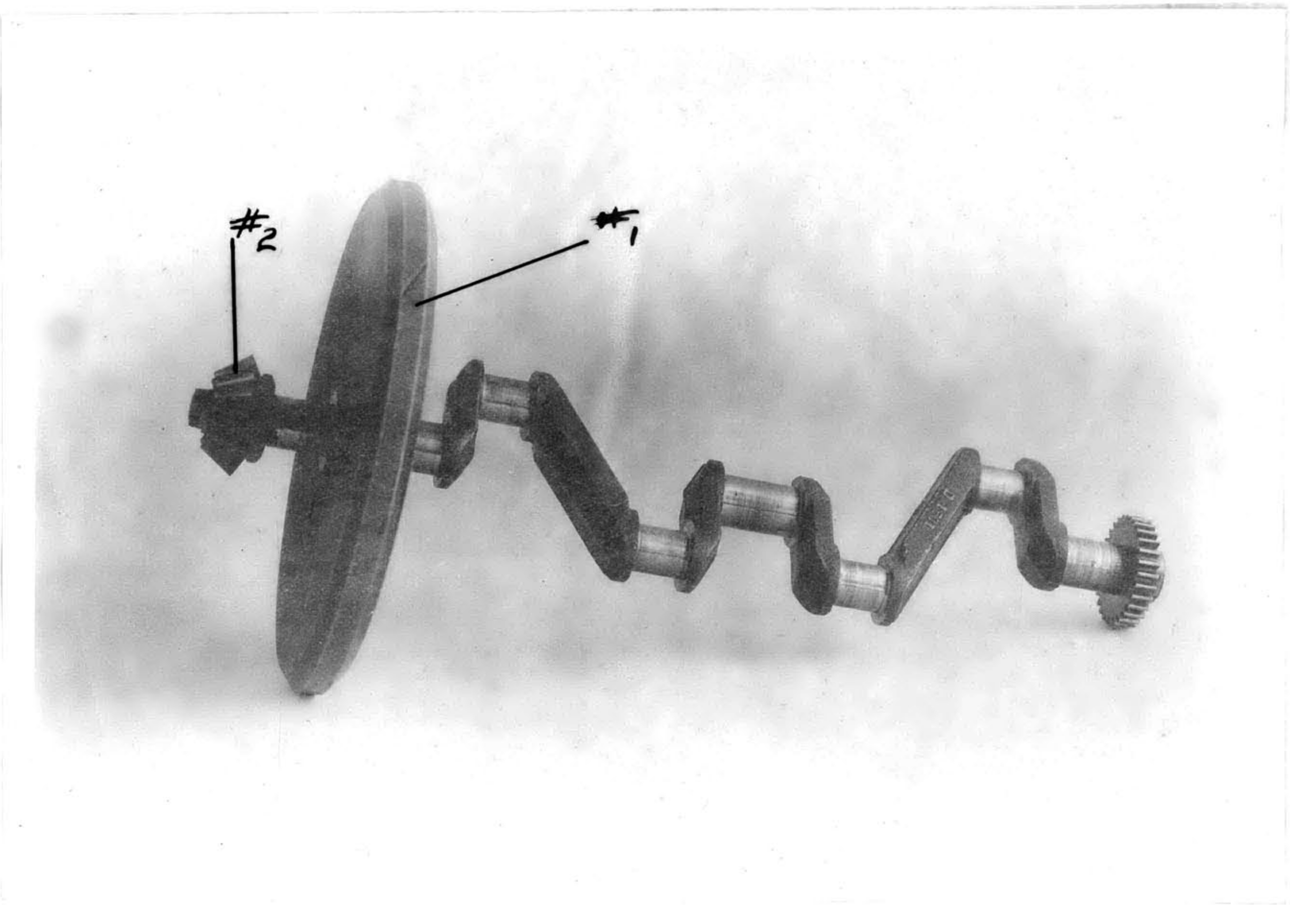


Plate H.

very thin and so it, too, was built up. (Plate A #3: Plate I #1) It was then necessary to fit the two halves of the crankcase tightly together at this welded section and to cut a small hole for the passage of the crankshaft (Plate F #8) This hole was cut a little larger than the crankshaft to permit the use of felt packing between the case and the shaft in keeping the oil from coming out. (Plate I #2) This cutting, of course, eliminated the bearing (Plate K #2) which was between the flywheel and the transmission bevel pinion on the end of the crankshaft (Plate H #2), and in consequence made the club considerably overhung.

PROVISION of THRUST BEARING.

The thrust required for the propulsion of a light plane is dependeant upon the design of the machine itself. It is assumed that the tractor type is to be used. The thrust is any case would not be large however, and so a plain steel ball thrust bearing was used. (Plate I #3) The size of the bearing - overall - was as follows :- outside diameter 2 1/2 in. inside diameter 1 in. and thickness 13/16 in. The flange (see accompanying photograph of crankshaft) which held the flywheel, was turned down to the outside dimmeter of the thrust bearing, i.e. 2 1/2 in. (Plate H #4). The space between the rear face of the flange and the front side of the rear wall of the flywheel housing was just the right thickness to accomodate this bearing. Any propeller thrust was transmitted to the bearing thru the flange on the crankshaft and then thru the thrust bearing to the flywheel housing and thence to the engine supports.

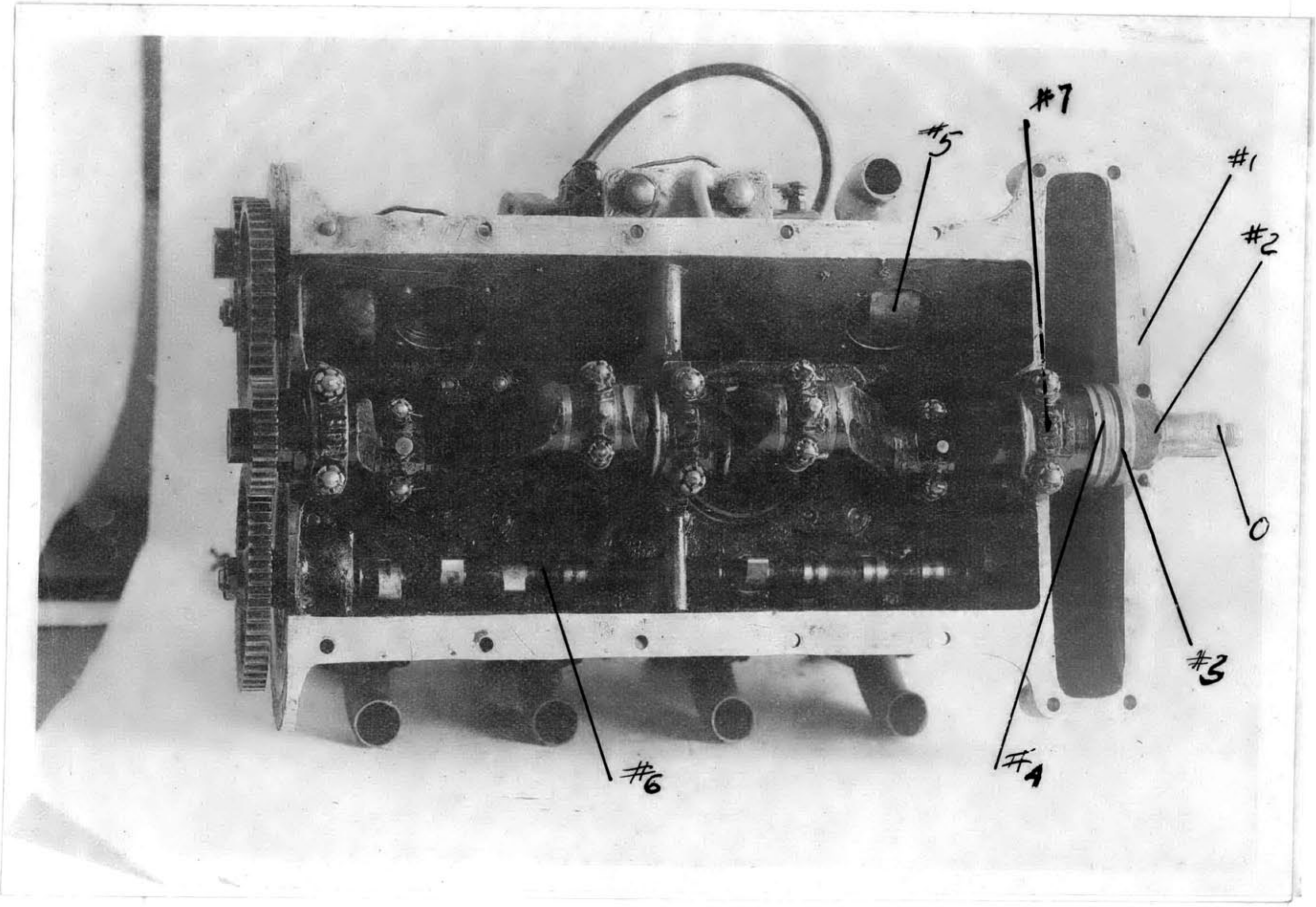


Plate I.

THE HUB.

The hub for fastening the club to the crankshaft is shown in Plate D #1 and #2. The flanges are 3in. in diameter and 5/16 in. thick. The distance between the inner faces of the flanges is 3 in. The holes are for 3/8 in. bolts. The diameter of the section between the flanges is 1 1/2 in. The threaded hole in part #1 is 1 in. deep and tapped 1/2 in. - 20. Part # 2 has a keyway and is placed on the end of the crankshaft first. Next the club is put in place and thru it is put #1, which is then screwed onto the threaded crankshaft until the two parts of the hub are tightly together. In this position the holes in the two flanges line up and bolts are run thru the two flanges and the intervening club. In this way there can be no movement of #1 relative to #2 and therefore the hub cannot come loose. This hub design did very well except that it was difficult to disassemble. In this case it was necessary to put a bar thru the breather tube hole in the side of the crank case (Plate I #5) and to wedge it against the crankshaft so that Part #2 could be prevented from turning while Part #1 was unscrewed from the end of the crankshaft.

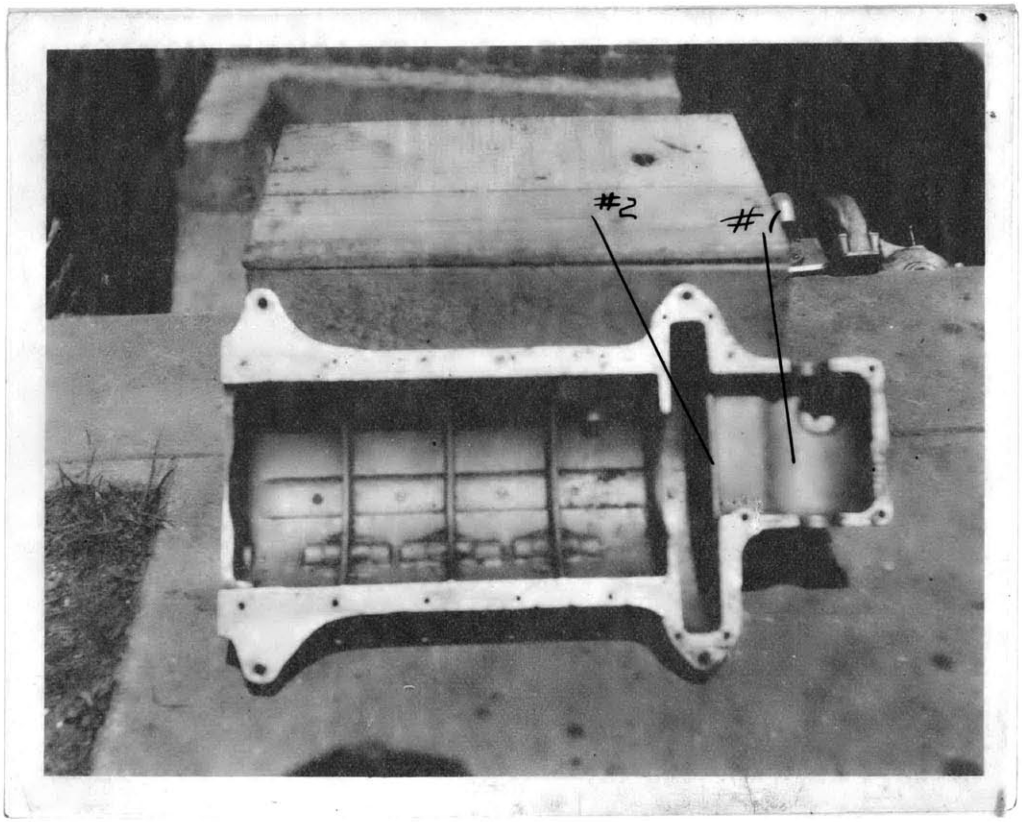


Plate I.

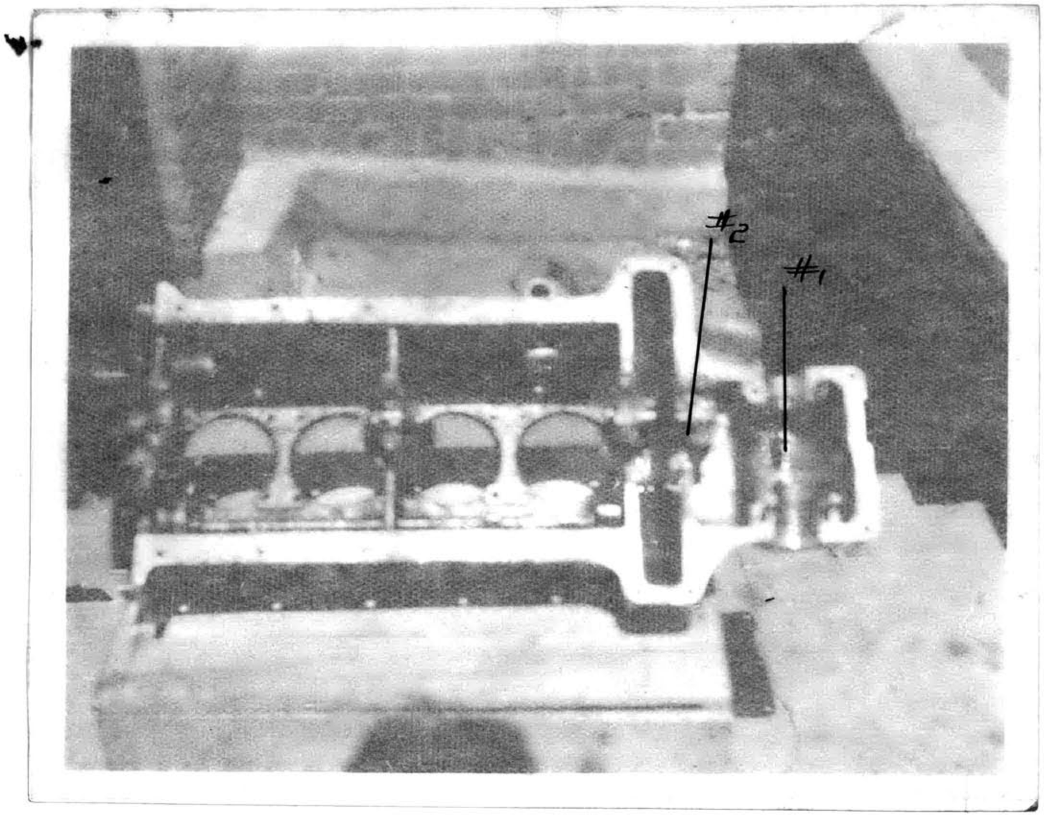


Plate K

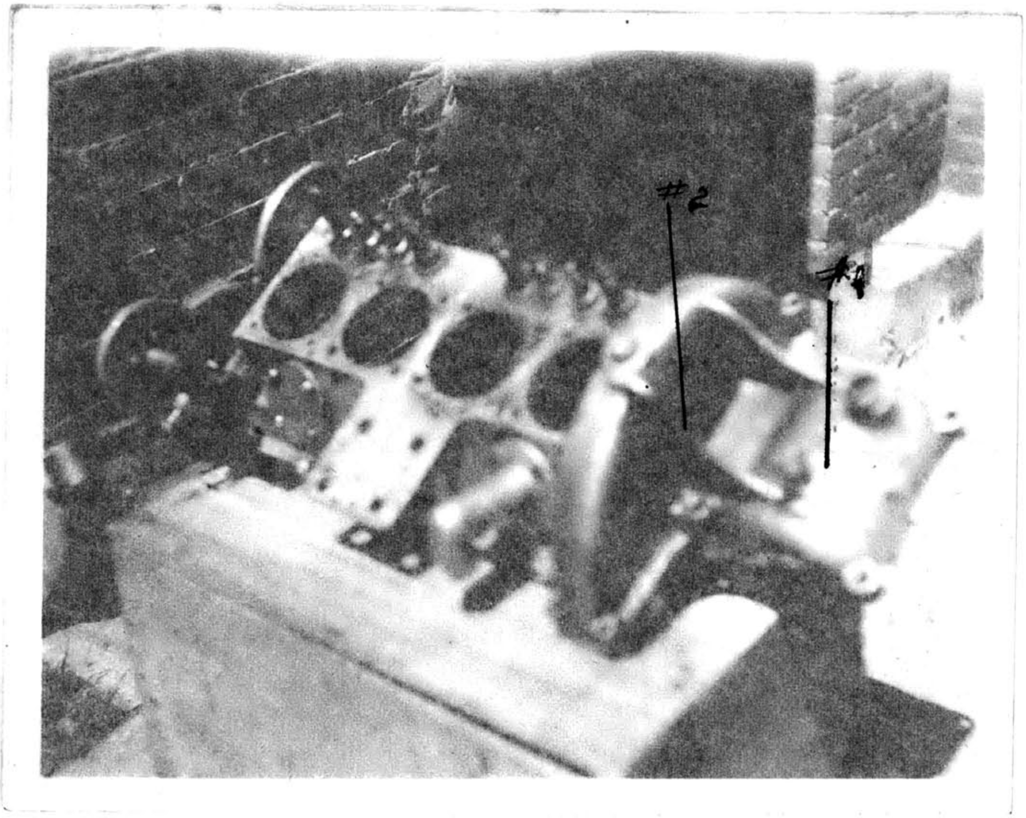


Plate L.

TEST WITH 36-inch CLUB.

When all of the above described changes and parts had been made and the motor reassembled and set up ready to a short run was made to determine whether or not the 36 inch club was too large.

With this club the motor heated up excessively and would not turn up more than 1500 r.p.m. This speed was reached with the throttle about half opened. Further opening of the throttle would not result in any further increase in speed. Due partly to the tightness of the pistons in the reground cylinders but more largely to the fact that the club was undoubtedly too large, the motor heated up very much. The club was therefore removed and four inches cut from each end. The pitch was increased and static balance carefully restored and the club - now 28 in. long was reattached for another trial.

TEST WITH 28-inch CLUB.

With the 28 inch club the motor turned up 2500 r.p.m. It ran very well at low and medium speeds, indicating that this size club was about right in its flywheel action at those speeds, but at speeds in excess of 1800 r.p.m. the engine did not perform so well. The greater pitch which this club possessed gave ample cooling for the cylinders.

To obtain still higher speeds, it was decided to cut the club down to 25 in. and to improve the performance of the engine at high speeds new exhaust valve springs were provided. (Plate A #4) These springs allowed the valves to open at

32 lbs pressure. Greater pressure than this was not deemed advisable because of the light character of the camshaft (Plate I #6)

TABULATED RESULTS OF TEST 28- inch CLUB.

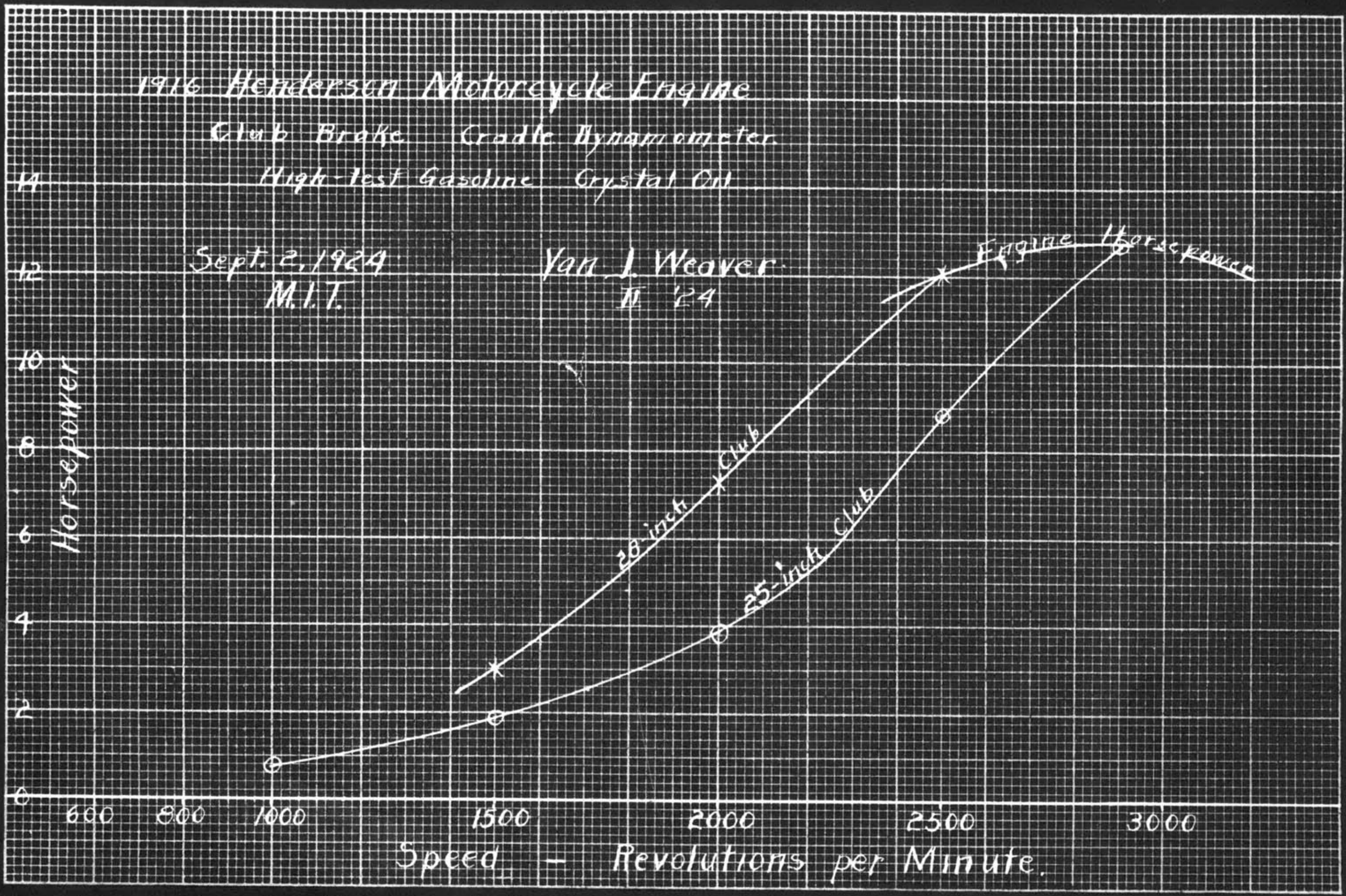
Run	RPM	Pounds	HP
1	1500	64.5	(64.5 - 57.625)(1500)K = 3.026
2	2000	70.0	(70.0 - 57.625)(2000)K = 7.264
3	2500	74.0	(74.0 - 57.625)(2500)K = 12.012

Fairbanks Platform Scale Tare Weight 57.625 lbs.
$K = 0.0002935$

TEST WITH 25-inch CLUB.

In reducing the club from 28 in. to 25 in. it was not removed from the engine. One and a half inches were cut from each end and no attempt was made thereafter to be sure that the static balance was preserved, it being thought that the removal of so small an amount would not sensibly affect the latter.

With this 25 in. club the motor turned over 3030 r.p.m. on a preliminary run and although it did not work well at speeds below 1000 r.p.m., due to the decrease in the flywheel effect of the smaller club, nevertheless, it was decided that higher speeds would not be feasible and that a test of the motors reliability and power output would be made now. The engine accelerated right up to the full open position of the throttle, which seemed to indicate also that speeds in excess of 3000 r.p.m. could not readily be reached. The tabulated results of the test with the 25-inch club, follow :



1916 Henderson Motorcycle Engine

Club Brake Cradle Dynamometer

High-test Gasoline Crystal Oil

Sept. 2, 1924
M.I.T.

Van L Weaver
II '24

Run	R.P.M.	Pounds	HP
1	1000	63.0	(63.0 - 60.5)(1000)K = 0.734
2	1500	65.0	(65.0 - 60.5)(1500)K = 1.981
3	2000	67.0	(67.0 - 60.5)(2000)K = 3.8155
4	2500	72.5	(72.5 - 60.5)(2500)K = 8.805
5	2900	75.5	(75.5 - 60.5)(2900)K = 12.767

Spring Balance Tare Weight 60.5
K = 0.0002935

DISCUSSION OF CURVES.

As noted previously (Pg.16) the club type of brake offers a fixed resistance for each speed at which it revolves. The variation of this resistance with changes in speed is a complicated function depending upon the design of the club. It is therefore impossible to get the maximum power the engine will develop at any given speed except at the maximum speed at which the engine will turn the club in question. At that speed the power of the motor is the maximum and is just equal to the pesistance of the club at that speed. If this were not so the engine would be able to turn the club faster until equilibrium was established. Knowing that the power curve of a gasoline engine increases with an increase in speed until a peak is reached after which the curve decreases with further increase of speed and noting that the increase in horsepower output for this engine for an increase of 400 revoulution per minute was only 0.75 HP, it is safe to assume that the broken curve shown on the accompanying graph is representative of the power curve of this engine between the speeds of 2500 and 3200 revolutions per minute.

A reliability or duration test as planned was cut short by the shearing off of two of the bolts holding the club and hub in place. This happened while the motor was running at full speed, but fortunately the vibration resulting from the loosened condition of the club gave warning that something was wrong in time for the engine to be stopped before any damage of a serious nature occurred. Investigation showed that the hub was so hot th that the wood of the club was smoking considerably. The shearing of the bolts, which were originally under considerable tension, was doubtless caused by the added tension imposed by the expansion of the metal of the hub. Heretofore there had been no excessive heating of the hub.

The extend and amount of heating can better be appreciated when it is understood that in order to remove Part 2 of the hub (Plate D #2) from the crankshaft, it was necessary to saw and chisel it off. A careful inspection of the end of the crankshaft at 0 Plate I will show the roughened condition of the shaft where the heat started to weld the hub to the shaft. The hub bolts when removed looked as though they had been tempered -- the middle was blue colored while the ends were staw.

There are two possible explanations for this heating of the hub, either or both of which seem adequate. It will be remembered that when the club was reduced from 28 in. to 25 in., it was not removed from the hub and the balance was not checked up thereafter. There may have been therefore a slight unbalance which would produce bending of the crankshaft at the rear bearing (Plate I #7). The club being overhung about 4 inches this repeated bending at 3000 r.p.m. would generate a great deal of heat which would be communicated to the hub, while the heat in the crankshaft itself would largely be carried away by the lubricating oil. The other explanation is that the key in the end of the crankshaft did not fit tightly the keyway, which had to be cut by hand, in Part 2 of the hub (Plate D #2) A great deal of heating touble of this nature has been traced to this source in tests on motors with club brakes at McCook Filed, Dayton, Ohio.

At this point it was decided to abandon further tests of this motor, for the following reasons --

The weight of the motor per horsepower was too high. The weight seemed to have been reduced as far as feasible and to increase the horsepower would necessitate the increasing of the compression ratio which would in turn greatly increase the cooling difficulties.

The club type of brake, with the engine controls so close, was a constant and menacing source of danger to the operator and surroundings and to rig up another type of brake would in itself involve considerable expenditure of time and money and would leave the problem of cooling entirely unsolved.

The amount of time and money spend so far was not justified by the results obtained.

CONCLUSIONS.

The reliability of the motor, while not definitely determined by an endurance test, seemed good. An examination of the parts, etc, after the last test showed no reason for condeming the engine on this point.

The horsepower developed was a trifle lower than that expected and as stated originally, to be desired for light plane work.

The speed developed was not too high for the use of a direct driven propeller of the Curtiss Reed forged aluminum type. The Curtiss Co. at Garden City, L.I, N.Y., will build a propeller for this motor for One Hundred Twenty Dollars ($120.00)

The weight per horsepower was too high to warrant the use of the motor, with no further changes, in light plane work. Especially designed motors for this purpose can be purchased which develop from 25 to 35 HP and weigh less than 100 lbs. While there is no imformation readily at hand as to the cost of these motors, it is probable that they are much more expensive than the total cost of rebuilding this Henderson engine.

RECOMMENDATIONS.

Any attempt to carry the development of this work further should be along the following lines and with the following changes ---

A better intake manifold with the use of a Zenith carburetor.

Stronger intake valve springs.

A light roller or ball bearing having its race set into the flywheel housing at #7 Plate F and its cone adjustable on the crankshaft should replace the plain thrust bearing used. Such a bearing would not only take care of the propeller thrust but would also reduce the overhang of the propeller and eliminate any possibility of bending the crankshaft.

The compression ratio should be increased -- possibly to 4 but no more, since the cap screws hold the cylinders to the crank case are screwed into aluminum and are in danger of pulling out, especially when the holes are worn from frequent dismantling. The compression ratio could best be increased by building special pistons with the head higher above the wrist pin than at present. As mentioned before, increasing the compression ratio will also increase the difficulty of cooling the cylinders and it would appear that the best way to overcome this difficulty would be to mount the engine on some movable platform, such as a light automobile chassis or boat -- anything that would enable the propeller to pull it along at a speed giving cooling comparable to that to be expected in a light plane.

The dust caps on the tappets should be removed and flat faces ground on the tappet itself which will enable one to hold them while the adjusting srew and lock nut are set.

Disregarding the cost and time further work with this engine, as outlined above, would be justified, and with out doubt the engine could be made servicable for light plane work.

DATA and CALCULATIONS.

Over-all Dimensions of Altered Engine.

Length		23.5 in.
Width --		16.5 in.
Width	with out carburetor Plate A #1	12.5 in.
Height		17.5 in.

Compression Ratio   Reground.

Piston at bottom dead center	Volume	450.0 cc.
Piston at top dead center	Volume	85.0 cc.

Ratio 450 / 85 = 2.941

Piston Displacement   Reground.

Area of Piston x Stroke x Number of cylinders =

$$\frac{\pi d^2}{4} \times L \times N = \frac{\pi}{4} (2.555)^2 \times 3 \times 4 = 61.525 \text{ cu.in.}$$

Weight of Parts.

Engine complete with oil $\frac{60.5 \times 18.5}{11.5} = 97.3$ lbs.

Note : Distance from cradle axis to scale = 18.5 in.
Distance from cradle axis to C.G. = 11.5 in.
Tare weight = 60.5 lbs.

Flywheel	13 lbs.	25'' Club and Hub	7.5 lbs.
Old Oil Pump	1.0	New Oil Pump	0.5
Transmission	7.5	Welded Aluminum	0.5
Parts Removed	21.5 lbs.	Parts Added	8.5 lbs.

Saving in weight 13.0 lbs. or slightly over 1.0 lb. per Horsepower.

Horsepower Formula

$$HP = \frac{2\pi NM}{33000} = \frac{2\pi\ N\ P\ a}{33000} = \frac{2\ \pi \times 18.5}{33000 \times 12}\ P\ N.$$

$$= 0.0002935\ P\ N$$

$$= K\ P\ N$$

in which N = revolutions per minute
M = P a = turning moment
P = scale reading in pounds
a = moment arm = 18.5 inches
= 18.5 / 12 feet.
33000 = foot-lbs. per minute
K = 0.0002935 = constant.

Mean Effective Pressure.

$$M.E.P. = \frac{HP \times 33000}{(\text{No. Cyls.})(\text{Stroke})(\text{Area Piston})(\text{Working strokes per cyl. per min.})}$$

$$= \frac{12.767 \times 33000}{4 \times \frac{3}{12} \times \frac{\pi}{4}(2.555)^2 \times \frac{2900}{2}}$$

$$= 56.05 \text{ lbs. per sq.in.}$$

ESTIMATE OF TIME.

Disassemble, Inspect, and Reassemble	10 hours.
Make preliminary runs	2
Make new intake valve stem guides	3
Turn intake valve stems and true up valves	1
Reseat valves, including regrinding	2
Make new magneto drive shaft	1
Rebore drive shaft bearing	0.5
Test for air leaks, etc.	3
Cut up old manifold	1
Make new manifold	20
Test run	0.5
Make new oil pump, new pipe lines etc	10
Install new exhaust valve springs	2
Build dynamometer	2
Build club	3
Remove flywheel and transmission	2
Provide thrust bearing -- fitting welded section of crankcase halves, etc.	3
Make hub	5
Test 36 '' club	2
Cut down club	2
Test 28'' club	2
Cut down club	0.25
Test 25'' club	2
Rebuilding motor after regrinding cylinders	4
Making pictures -- Dismantling,etc.	10
Procuring materials and getting outside work done involving innumerable trips all around Boston	20
Writing up report	15
Total	128.25 hrs.

This estimate of the time required is what I should judge would be requred for another to accomplish the same results without having to make all the errors, parts,etc., which I did before arriving at the proper solutions of difficulties. I estimate another 50 hours was lost in the above manner.

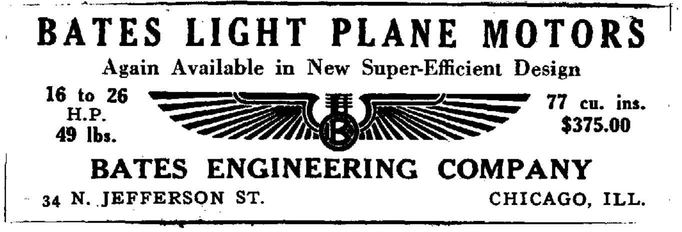
Cost.

The cost of the motor will vary with its condition and wheter or not it can be purchased without the frame, wheels,etc. of the motorcycle. It should not exceed Fifty Dollars ($50.00)

Other items of expense follow --

Starting ratchet	$ 1.20
Connecting rod bearings, bolts, etc.	5.45
Valve grinding compound	.25
Blacksmith work Mr. Lambirth.	3.00
Carburetor	9.00
Steel for hub	2.70
Brass for intake manifold	2.45
Borax	.10
Brazing manifold	5.00
Thrust bearing	1.65
Welding - aluminum	6.00
Valve reseating tool	1.25
Oil pump gears	1.20
Aluminum for oil pimp	3.40
Piston rings	1.95
Regrind cylinders, new pistons and pins	34.80
Bushing	.27
Wood for club	1.50
Valve springs	.80
Iron for cradle dynamometer	3.65
Crystal lubricating oil	2.00
Pictures Prof. Hardy for films	9.00
Miscellaneous and printing	3.00
Nuts and bolts, screws, pipeing etc.	15.00
Approximate total expense of alterations	$ 114.62
Maximum cost of motor	50.00
	$ 164.62

Compare with the accompanying advertisement from the Septermber 1924 issue of Aviation.



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Again Available in New Super-Efficient Design

16 to 26 H.P.
49 lbs.

77 cu. ins.
$375.00

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