AD-A073 573 BENNETT AEROTECHNICAL AUBURN AL F/G 21/7

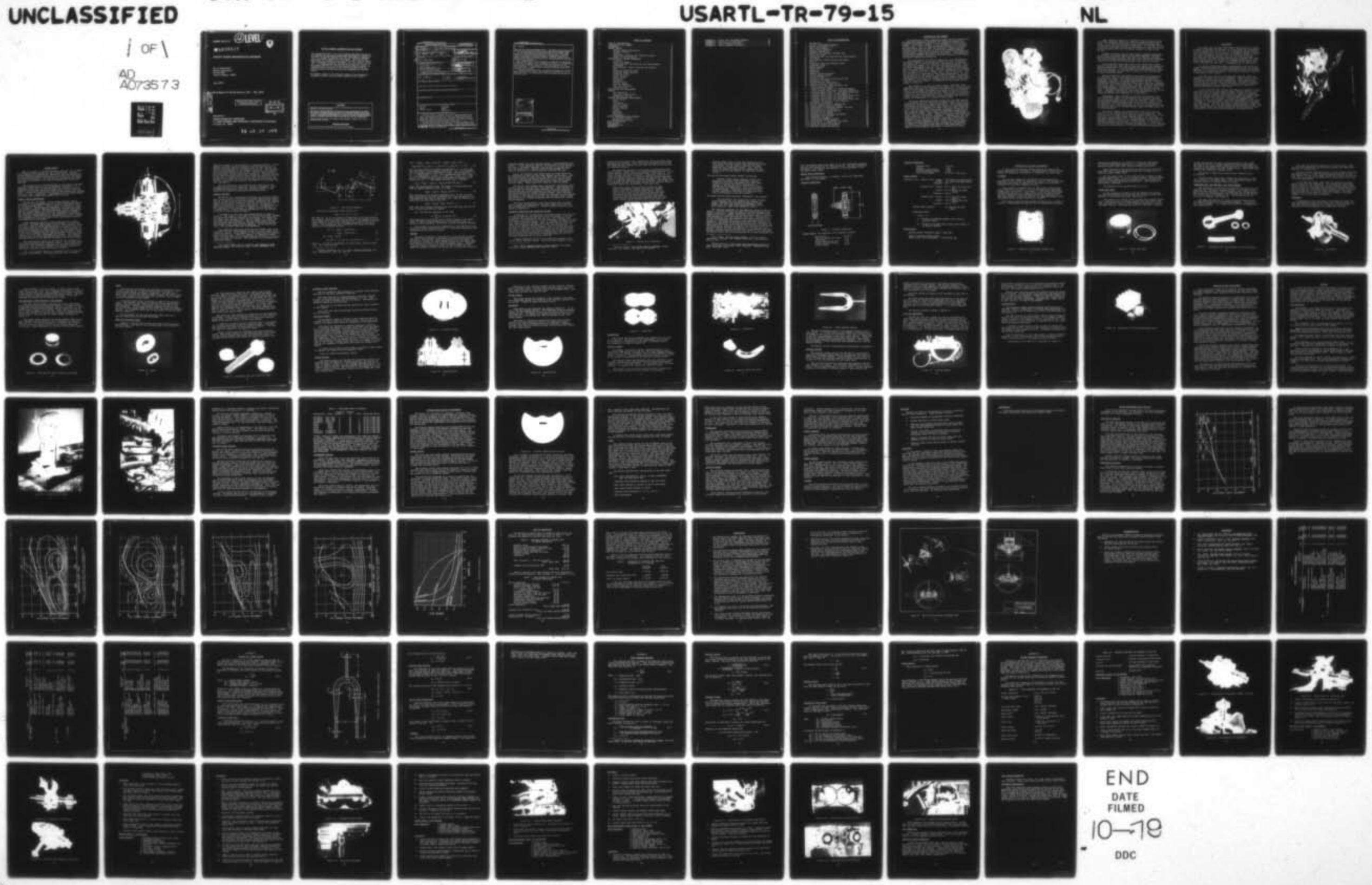
MINI-RPV ENGINE DEMONSTRATOR PROGRAM.(U)

JUN 79 G E ABERCROMBIE DAAJ02-77-C-0014

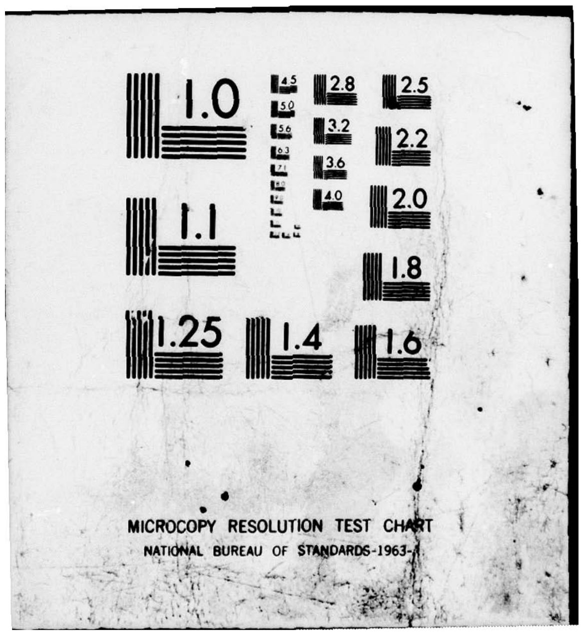
UNCLASSIFIED USARTL-TR-79-15 NL

1 OF 1

AD
A073573



END
DATE
FILMED
10-79
DDC


1.0
1.1
1.25
1.4
1.6
1.8
2.0
2.2
2.5
2.8
3.2
3.6
4.0
4.5
5.0
5.6
6.3
MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

(12) LEVEL II

USARTL-TR-79-15

AD A073573

MINI-RPV ENGINE DEMONSTRATOR PROGRAM

Gary E. Abercrombie
Bennett Aerotechnical
P.O. Box 1032
Auburn, Alabama 36830

June 1979

Final Report for Period February 1977 - May 1979

DDC FILE COPY

Approved for public release;
distribution unlimited.

Prepared for

APPLIED TECHNOLOGY LABORATORY
U. S. ARMY RESEARCH AND TECHNOLOGY LABORATORIES (AVRADCOM)
Fort Eustis, Va. 23604

79 09 07 004

APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

The reported R&D effort is one of two advanced development programs for 20 hp engines to experimentally demonstrate technology for engines for mini-RPV's through the integration of high rate production components into a two-cylinder opposed, two-stroke engine. The 20 hp mini-RPV demonstrator was considered to be successful in that it met the objectives of horsepower, weight per horsepower, life, low vibration, and low cost. It is expected that the data from this program will be used in the upcoming Mini-RPV Full-Scale Engineering Development Program. Appropriate technical personnel of this Laboratory have reviewed this report and concur with the conclusions and recommendations contained herein.

Mr. Edward T. Johnson of the Propulsion Technical Area, Aeronautical Technology Division, served as project engineer for this effort.

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER
18 USARTL-TR-79-15 19 TR-79-15

2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle)
6 MINI-RPV ENGINE DEMONSTRATOR PROGRAM.

5. TYPE OF REPORT & PERIOD COVERED
9 Final Report.
Feb[redacted]77 [redacted] May [redacted]79,

6. PERFORMING ORG. REPORT NUMBER

7. AUTHOR(s)
10 Gary E. Abercrombie

8. CONTRACT OR GRANT NUMBER(s)
15 DAAJ02-77-C-0014

9. PERFORMING ORGANIZATION NAME AND ADDRESS
Bennett Aerotechnical New
P.O. Box 1032
Auburn, Alabama 36830

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
16 [redacted] 1L762732 AH34 [redacted]

11. CONTROLLING OFFICE NAME AND ADDRESS
Applied Technology Laboratory
U.S. Army Research and Technology Laboratories
(AVRADCOM), Fort Eustis, VA 23604

12. REPORT DATE
11 Jun[redacted]79

13. NUMBER OF PAGES
92

14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)
12 93p 17 01

15. SECURITY CLASS. (of this report)
Unclassified

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Mini-RPV	Two-Stroke
Engine	Propulsion
Remotely Piloted	Geared
	Low Vibration

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Increased interest in the past few years in remotely piloted vehicles (RPV's), and especially Mini-RPV's, has led to a similar interest in the power plants required to propel them. This report describes a program to demonstrate a Mini-RPV engine of a unique design. The program consisted of design, fabrication, test and evaluation of the engine.

Important objectives were low production cost, good performance, reliability, low vibration levels and light weight. To meet these objectives a two-stroke cycle, twin-cylinder, twin-crankshaft, geared, simultaneous firing

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

411360

Unclassified
SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

20. Abstract

spark ignition engine was designed and built. The engine utilizes several high production parts including cylinders from the Homelite 270 industrial engine.

The twin-crankshaft, geared configuration allows almost complete engine balance with the resulting exceptionally low vibration levels. The geared output also permits the engine output speed to be matched to airframe and propeller requirements for maximum efficiency. Separated crankcases provide better fuel/air distribution contributing to a low idle speed and giving better operation during maneuvers.

With straight exhausts, over 25 horsepower was obtained on the dynamometer at 7556 rpm with a specific fuel consumption of 0.9 lb/hp-hr. At reduced power, specific fuel consumption as low as 0.68 lb/hp-hr were obtained with the straight exhaust. In this configuration the engine weighs 24.5 lb, excluding the alternator/PCU. Production design of housing and ignition system should reduce this weight to 23 lb.

Employing a tuned exhaust system, over 28.5 hp was measured on the dynamometer at 7556 rpm with a specific fuel consumption of 0.79 lb/hp-hr. At reduced power levels, fuel consumption as low as 0.60 lb/hp-hr was obtained with the tuned system.

Unclassified
SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

INTRODUCTION AND SUMMARY

The emergence of small remotely piloted vehicles (Mini-RPV's) as potentially important defensive and offensive weapons has prompted considerable interest in all elements of the Mini-RPV system. One critical element in the development of all aircraft is the engine. The U.S. Army recognized the lack of a suitable Mini-RPV engine and initiated two programs to demonstrate the technology required for such an engine and identify possible problem areas. This report presents the results of the demonstrator program undertaken by Bennett Aerotechnical. The program included engine design, fabrication of parts peculiar to the design and engine testing and evaluation.

The requirements of critical importance to be considered in the engine design, as stated by the Army, in order of priority, are engine cost in future production, performance, reliability, low vibration, weight, IR signature and maintainability. Also, to reduce development time and cost, available production parts were to be used when practical. Specific targets of the program were an output of 20 hp, a specific fuel consumption of 0.8 lb/hp-hr, a weight of 24 lb (1.2 lb/hp), and a life of 150 hours. The BAT 282 engine was designed to meet these requirements. It is a two-stroke cycle, two-cylinder, simultaneous firing, air-cooled, reciprocating, rotary valve, geared, twin-crankshaft, spark ignition engine. The engine was designed around parts from the Homelite 270 industrial engine. A photograph of the engine is presented in Figure 1.

The simplicity of the two-stroke engine, requiring fewer components than its four-stroke cousin, makes it an obvious choice when cost and weight are important considerations. A simultaneous firing, two-cylinder arrangement was chosen to provide an efficient means of producing the required power as well as presenting a means to reduce the vibration level. The engine can be effectively cooled in flight by the air stream without the additional weight and complexity of a liquid cooling system.

The major difference between the BAT 282 and other engines in its general class is the twin-crankshaft design with the two crankshafts geared to a common output shaft. This geared design adds weight and possibly cost to the engine, but the advantages in performance far outweigh the small cost and weight penalties. First, the twin-cylinder, twin-crank arrangement allows for almost complete engine balance with the subsequent exceptionally low vibration levels. This unique feature will be described in detail later in this report. Second, the design also allows complete separation of the two crankcases. Together with the twin carburetors (one for each cylinder), good fuel distribution is assured, even in maneuvers. Mixture distribution is often a problem in common crankcase twins. Third, the gearing offers an important advantage in the area of propeller performance. The engine can be geared down from the high crank speed to a speed at which propeller efficiency is considerably higher.

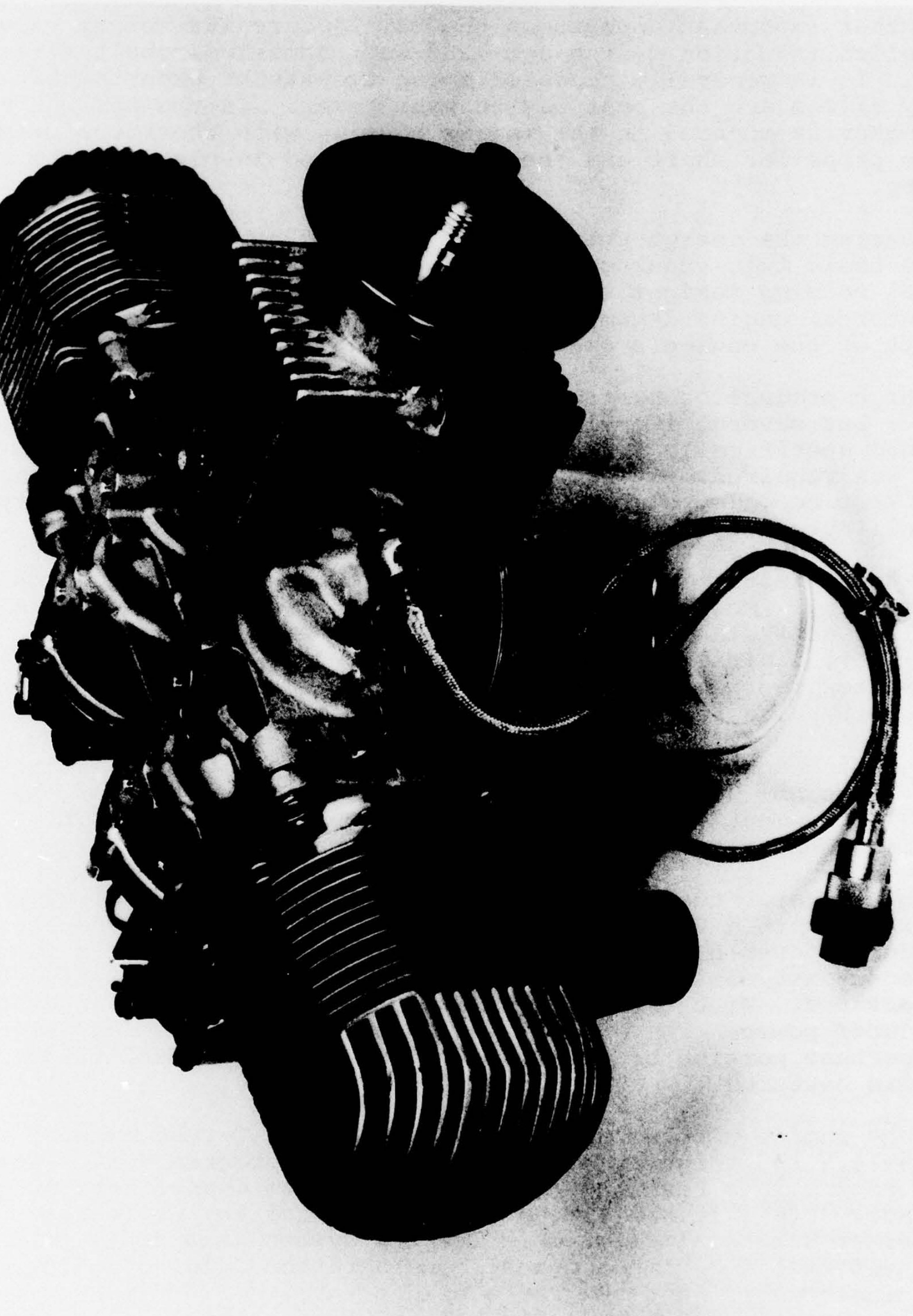


Figure 1. The BAT 282.

Other important aspects of the BAT 282 are its rotary valve controlled induction system and 1250-watt samarium-cobalt alternator. It is generally accepted among two-stroke experts that rotary valves are the best way to make power. In the BAT 282 the alternator is mounted in the engine housing with the rotor pressed on the propeller shaft and the stator clamped in place by the housing.

During the design phase of the program, numerous variations on the basic twin-crank geared design were examined, including several housing designs that were split in different locations. The internal engine dynamics, bearing loads, gear loads, and other aspects of the engine's operation were also studied in detail.

High production parts were used in the engine where appropriate, but several items such as housings, gears, and cranks were designed specifically for this engine, and fabrication of these parts was required. It was decided that these experimental parts could best be made, on the basis of quality, cost, and delivery, by small specialty shops.

Engine testing was performed by both Bennett Aerotechnical and the Army. Most of the in-house testing was performed on a propeller test stand. The prop stand is ideal for endurance, carburetor, ignition, and general performance testing. In-house dynamometer testing was performed where precise data was required and for the fine tuning operations. The Army conducted dynamometer performance tests, a 150-hour endurance test, altitude, noise, icing, electromagnetic interference (EMI), and hot/cold starting tests. The BAT 282 passed all tests with the exception of the EMI test. Government test results will be documented in an ATL Technical Report to be published at a later date.

Dynamometer tests conducted by Bennett Aerotechnical gave over 25 hp with a BSFC of 0.90 lb/hp-hr with straight exhausts. At reduced power BSFC as low as 0.68 was measured. With a tuned U-tube exhaust maximum power of over 28.5 hp with a BSFC of 0.79 was measured. With the U-tube BSFC as low as 0.60 was measured at reduced power. The program goal of 20 hp was exceeded and over an important portion of the performance region the BSFC target of 0.80 was bettered.

The engine without the alternator/PCU and U-tube exhaust weighs 24.5 lb, slightly more than the 24-lb program goal. The extra weight is a result of an ignition system change late in the contract. The original system weighed about 1 lb, while the new system weighs almost 3 lb. The new system is a modified off-the-shelf item and time and cost considerations did not allow the packaging changes required for a flight version. A flight version of the new system is estimated to weigh about 1.5 lb, reducing the overall engine weight to 23 lb and bettering the program goal.

BACKGROUND

In August 1974, Aerotech (now Bennett Aerotechnical) started a study to determine the feasibility of developing an inexpensive mini-RPV system for civilian use. An appropriate engine to power such a vehicle could not be located. A telephone survey of groups concerned with RPV's, including the RPV research centers of the U.S. Army at Redstone Arsenal, the U.S. Air Force at Wright-Patterson AFB, and NASA at Ames Laboratories, revealed that there were no good, low-cost reciprocating engines available for operational Mini-RPV use. Of course, an inexpensive RPV system could not be developed at that time. A powerplant had to come first.

The survey identified several engines being used in RPV programs: McCulloch chain saw engines and Models 91 and 101 go-kart engines; O & R Series 13B and 20A industrial engines; Roper Models 1900 and 3700 industrial engines; Kolbo special engines; Lycoming, a very special small engine.

The chain saw, go-kart, and industrial engines, while not expensive, suffered one or more drawbacks: high vibration levels, fairly high weight, high fuel consumption, limited operational reliability, or inconvenient configuration. The special engines also had special prices. None of the usual aircraft or appliance engine manufacturers would consider developing an appropriate low-cost Mini-RPV engine at that time.

It was concluded that low-cost RPV engines at moderate production rates must be of some special design and must utilize available parts to the largest extent possible. Since this approach does not require extensive manufacturing capability, Bennett Aerotechnical decided to undertake a limited engine development program of its own. This program, initiated in September 1974, resulted in the two-cylinder, twin-crank experimental engine shown in Figure 2. The only special parts in this engine are housings and three gears. Pistons, cylinders, connecting rods, crankshafts, ignition system, etc., are all standard high-production items. The basic engine design was substantiated and the low vibration characteristics of the twin-crank arrangement were demonstrated.

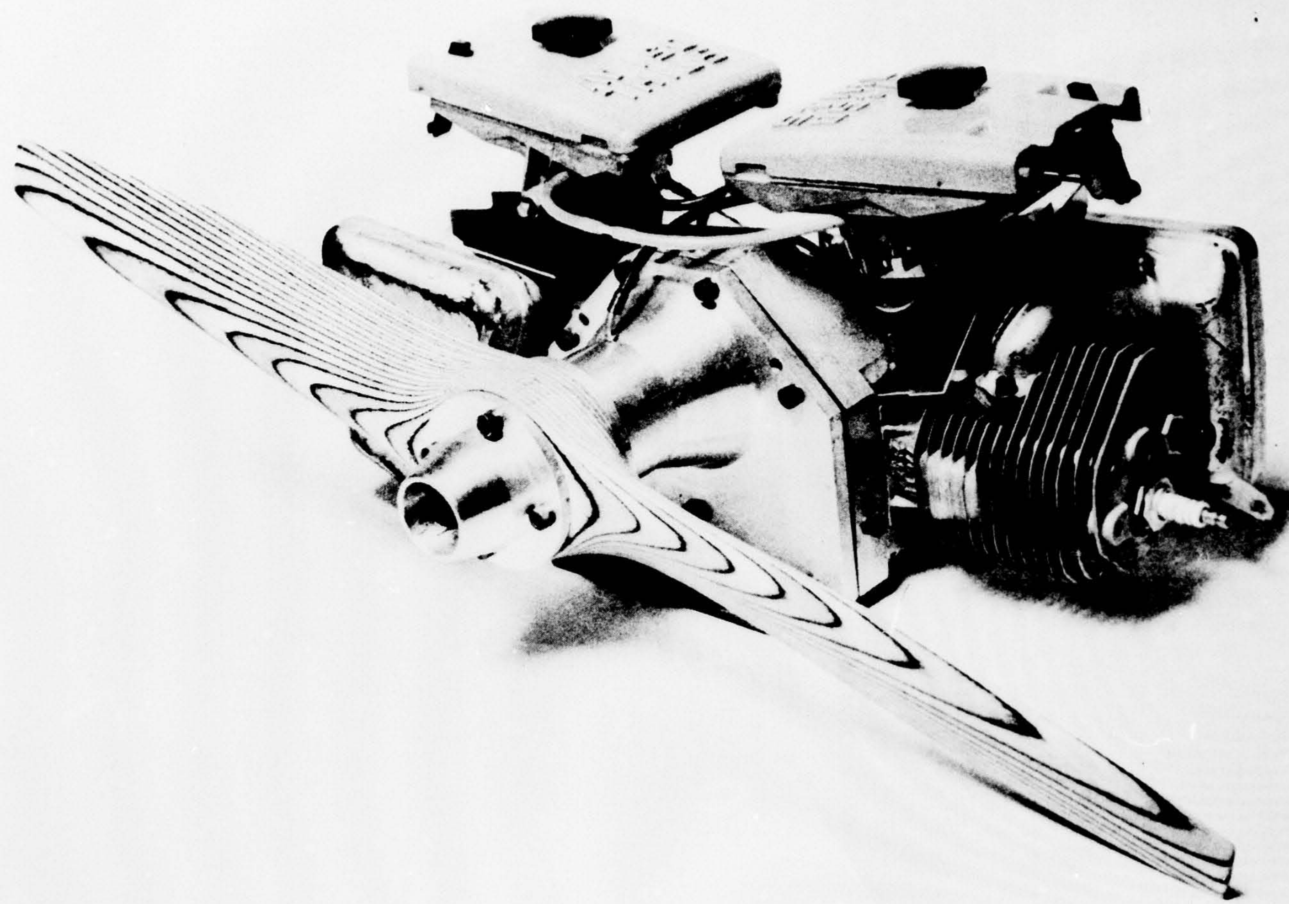


Figure 2. Original Experimental Engine.

ENGINE DESIGN

When the design of the BAT 282 began, the goal was to develop an engine ideally suited for use in a Mini-RPV. This included emphasis on performance, reliability and maintainability, but also included special attention to the effect of the engine on the mission payload. For this reason, the special low vibration twin-crank design of the earlier experimental engine was employed.

The reliability and maintainability of an engine to be operated from remote sites is also of paramount importance. All components of the BAT 282 were designed for a minimum life expectancy of 500 hours, while the target engine life is only 150 hours. Experience gained during tests of the demonstrator engines indicates that maintenance requirements for the engine should be no more than a spark plug change every 50 hours.

GENERAL ENGINE DESCRIPTION

The BAT 282, in effect, is two single cylinder engines connected in an advantageous way by a common housing and geared to a common output shaft. Figure 3 shows a "half-shell" view of the engine. The housing is composed of two identical aluminum castings (slightly different machining) forming the bottom and top of the engine. The twin-crankshaft geared arrangement and the complete isolation of the two crankcases is evident. The induction system consists of twin carburetors, one feeding each cylinder, controlled by the crankshaft-drive rotary valves. The independence of the two sides of the engine adds a measure of reliability to the system.

Other important features of the engine are the integral alternator, the high energy capacitive discharge ignition system, and a special U-tube exhaust system. The alternator rotor is pressed right on the output shaft and the stator is held in place by the housing, as shown in Figure 3. The alternator is capable of generating over 1250 watts at 5000 rpm output speed. It is based on samarium-cobalt technology which enables a high output unit to be both lightweight and compact. The alternator, together with its power conditioning unit (PCU), was designed and manufactured for the BAT 282 by Simmonds Precision Company.

The simultaneous sparks for the BAT 282 are provided by a high energy multiple spark ignition system. The multiple spark feature provides superior combustion and virtually eliminates misfiring. The solid state system is a modified version of a motorcycle ignition system made by Gerex Company.

The U-tube exhaust system is the design of one of our consultants, John Brooks. The system operates like a standard

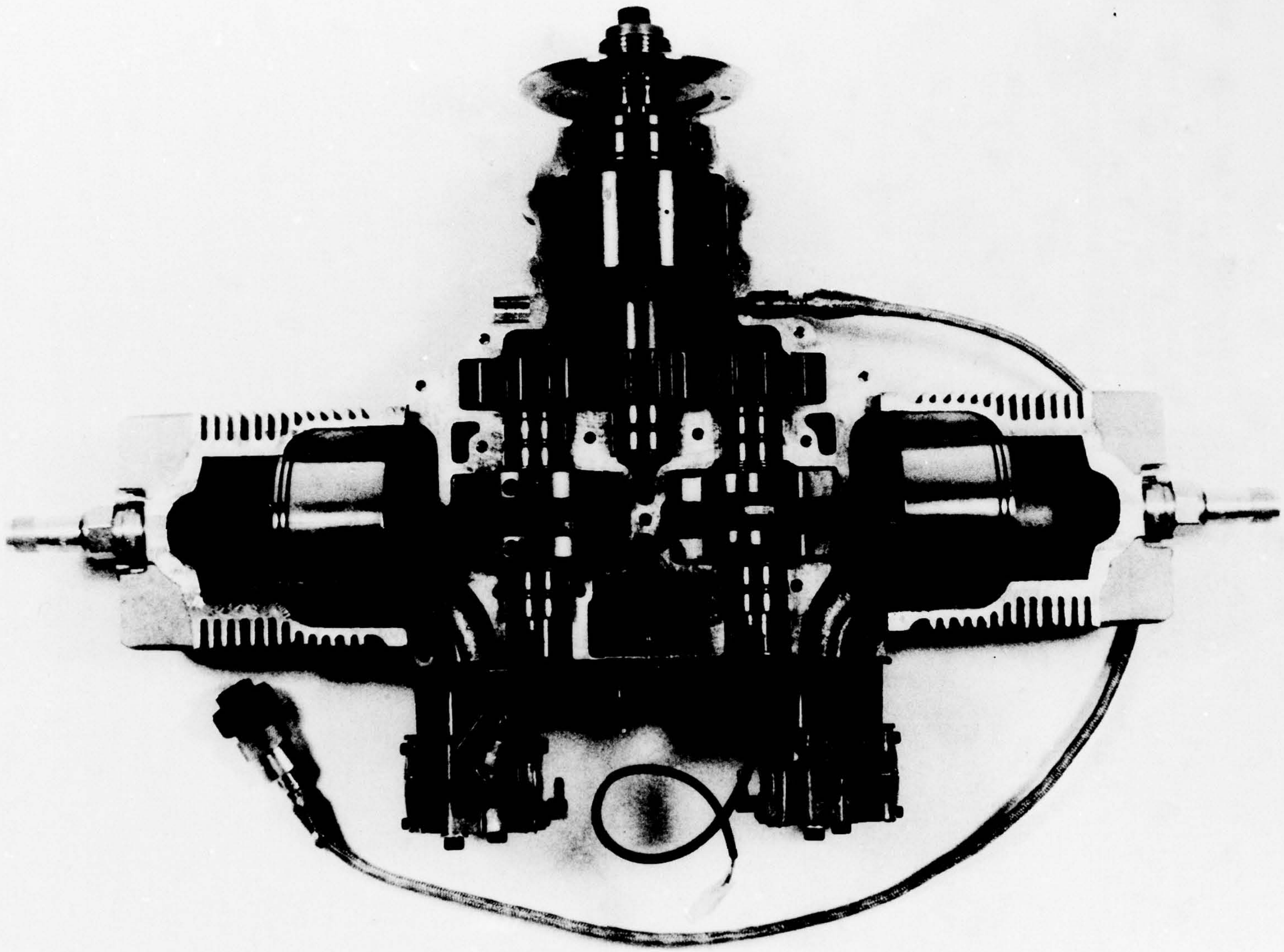


Figure 3. Engine "Half-Shell" View.

expansion chamber on the evacuation of exhaust products. As the exhaust from each cylinder expands in the diffuser section, a negative pressure pulse is reflected back to that cylinder and aids in the evacuation of the combustion products. The remainder of the process differs in the U-tube. The two exhaust pulses meet and travel through each other unaffected. The positive exhaust pulse from each cylinder then reaches the opposite cylinder at the time of transfer port closing, with the result that unburned mix is forced back into the cylinder before exhaust port closing. The U-tube has provided improvements in both power and fuel consumption.

The big difference in the BAT 282 and conventional two-cylinder engines is the twin-crank, geared arrangement. This arrangement provides the basis for achieving several of the engine's most important performance characteristics.

BALANCE CRITERIA

The BAT 282 has demonstrated extremely low vibration levels. This is due in part to the basic twin-crank design which allows the two cylinders to be directly opposed, cancelling the external effects of the piston forces and totally eliminating the rocking moment characteristic of single-crank twins. In addition, a dynamical analysis of the engine resulted in a technique for almost complete dynamic balance. A summary of this analysis is presented below. A more complete presentation of the analysis is given in an SAE Technical Paper.[1]

A schematic of the twin crank configuration is shown in Figure 4. The symbols to be used for the various forces, lengths, and angles involved are defined in the figure. The cranks rotate about axes through A and B and are geared to a common output shaft with axis through 0. The two cranks rotate in the same direction and have relative angular orientation such that the pistons travel symmetrically opposite. In the two-stroke cycle, the cylinders then fire simultaneously.

The separated crank axes allow the two cylinder axes to be coincident and the rocking moment present in usual single crank designs is absent. From symmetry it is evident that all external force sums are zero. There is, however, an unbalanced moment about 0 arising from the transverse bearing forces at A and B and from the piston side forces. The objective is to minimize this unbalanced moment.

---

[1]G.E. Abercrombie and A.G. Bennett, Low Vibration 20 HP Mini-RPV Engine, SAE Technical Paper 780764, September 1978.

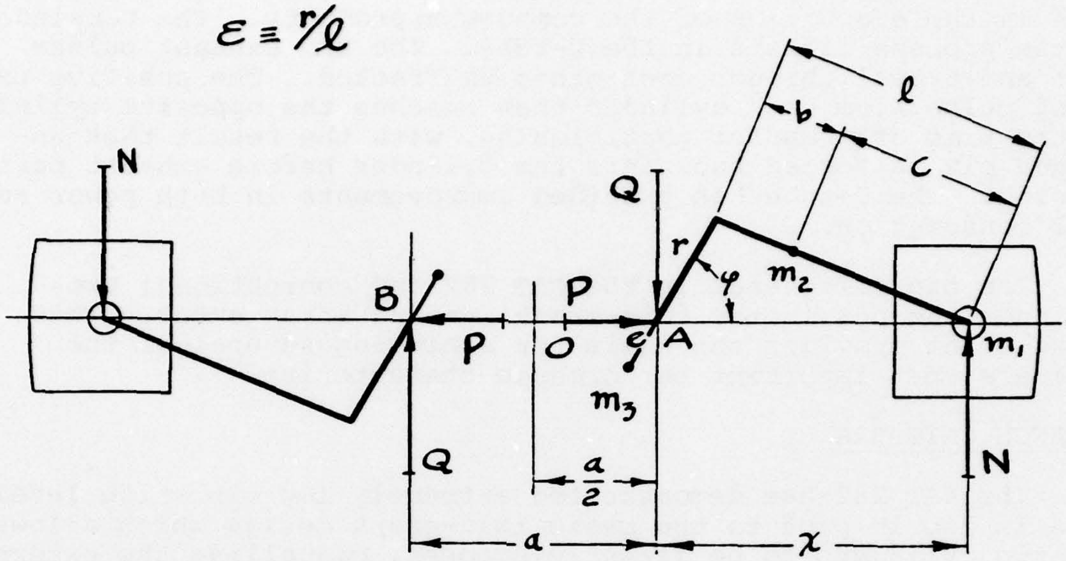


Figure 4. Twin-Crank Schematic.

The sum of the moments about point 0 is

$$\sum M_o = (N - Q)\ a + 2\ Nx \qquad (1)$$

The term (N - Q) is the classical frame force on a single cylinder machine, and Nx is equal to the time rate of change of the angular momentum of crank and connecting rod. Expressions for both these terms are given in any thorough discussion of engine dynamics. The expressions from Timoshenko and Young[2] are translated into the present notation.

$$N - Q = (m_2 b\varepsilon - m_3 e)\ddot{\phi}\cos\phi$$
$$- (m_2 b\varepsilon - m_3 e)\dot{\phi}^2 \text{ Sin } \phi \qquad (2)$$

$$Nx = \dot{\phi}^2 m_2 \varepsilon\ (bc-k^2)$$
$$(c_1 \text{ Sin } \phi - c_3 \text{ Sin } 3\phi + c_5 \text{ Sin } 5\phi + \cdots) \qquad (3)$$

where $\varepsilon = r/\ell$ and k = connecting rod mass center radius of gyration. Eq. 1 then becomes

[2]Stephen Timoshenko and D.H. Young, Advanced Dynamics, New York: McGraw-Hill, 1948, pp. 136-143.

$$\sum M_o = [(m_2 b\varepsilon - m_3 e)\, a \cos\phi]\ddot{\phi} - [(m_2 b\varepsilon - m_3 e)\, a \sin\phi$$

$$- 2m_2\varepsilon(bc-k^2)(c_1 \sin\phi - c_3 \sin 3\phi + c_5 \sin 5\phi + \cdots)]\dot{\phi}^2 \qquad (4)$$

For any usual engine the angular acceleration, $\ddot{\phi}$, is much less than the square of the angular velocity, $\dot{\phi}^2$. So, it is appropriate to drop the angular acceleration term. The constants c_1, c_3, c_5, $\cdots$ depend upon $\varepsilon = r/\ell$. For a reasonable and usual value $\varepsilon = 0.25$, Reference 1 gives $c_1 = 1.008$, $c_3 = 0.024$, $c_5 = 0.001$. And to good approximation we may drop terms multiplied by constants c_3, c_5, $\cdots$. The moment relation, Eq. 1, becomes

$$\sum M_o = [(m_3 e - m_2 b\varepsilon)a - 2m_2\varepsilon c_1(bc-k^2)]\dot{\phi}^2 \sin\phi \qquad (5)$$

Under the approximations made, the moment is minimum when the term in square brackets in Eq. 5 is zero.

$$(m_2 b\varepsilon - m_3 e)a + 2m_2\varepsilon c_1\ (bc-k^2) = 0 \qquad (6)$$

This relation can be further simplified if c_1 is set to unity. This approximation is most reasonable since the variation of c_1 from unity is less than 1% for usual r/ℓ. Then the balance criterion for the twin-crank engine is

$$(m_2 b\varepsilon - m_3 e)a + 2m_2\varepsilon\ (bc-k^2) = 0 \qquad (7)$$

Note that this relation contains only the crank mass and the rod mass. Piston mass does not appear.

Eq. 7 can now be rewritten in the form

$$e = \frac{m_2}{m_3}\,\varepsilon[b + (2/a)\,(bc-k^2)] \qquad (8)$$

which specifies the location of the center of mass of the crankshaft in terms of the parameters of the connecting rod, the weight of the crankshaft, the stroke, and the crankshaft spacing.

The balance criterion was implemented in the BAT 282 by balancing the crank as specified in Eq. 8. The other parameters were determined by the choice of parts.

GEARING

The major advantage of the gearing is the ability to match the engine output speed to propeller and vehicle requirements. Gearing for the BAT 282, when equipped with 12 pitch gears, may be chosen to produce an output speed as high as 1.53 times crankshaft speed or as low as .65 times crankshaft speed. Assuming a typical maximum crankshaft speed of 7500 to 8000 rpm, this makes the geared up engine well suited for a ducted fan requiring propeller speeds of 10,000 rpm and higher to reach maximum efficiency.[3] A more important benefit for Mini-RPV's which typically use open or shrouded propellers (not ducted) is the ability to gear the engine down. Maximum propeller efficiency for this class of propellers is usually between 4000 and 6000 rpm.

First, the original gear design using 16 pitch gears did not adequately account for the loads imposed on the gears. Gear failures occurred at times after less than 1 hour of run time. The current design uses Metric Module 2.5 (10.16 pitch) gears which are approximately 50 percent stronger than the original 16 pitch gears. These gears have shown no signs of damage.

The second, and probably most significant, deficiency found in the original gears was poor heat treating. This resulted in part because of incorrect heat treatment and in part because of insufficient gear tooth thickness to allow surface hardening to the depth required to prevent surface fatigue. More thorough testing and a new heat treat solved the first part of the problem. The change to the coarser pitch gear with thicker teeth allowed specification of a thicker hardened case and solved the surface fatigue problem.

The gears manufactured to the final design have run flawlessly through performance and endurance tests and no further gear problems are anticipated. A gear strength analysis is presented in Appendix C.

SEPARATED CRANKCASES AND INDUCTION SYSTEM

In addition to the advantages in engine balance and gearing, the twin-crankshaft configuration offers advantages in the way it handles the induction of fuel and air. The isolated twin crankcases, with each fed by its own carburetor, minimize any tendency toward mixture problems. This is a problem in common crankcase twins where one cylinder often runs 50°F hotter than the other due to unequal distribution of the fuel/air mixture. In the BAT 282 there is little tendency, even in violent maneuvers, for most of the fuel/air mix to be forced to one cylinder while the other is starved. To further prevent fuel problems in maneuvers and during acceleration and deceleration, the carburetors are mounted so that the internal diaphragm pumps and the fuel metering diaphragm will be least affected.

Another important feature of the induction system is the clean, direct induction ports. The ports were designed to have

---

[3] R.W. Hovey, Ducted Fans for Light Aircraft, R.W. Hovey, Publisher, Second Edition, 1973, pp. 1-3, 19.

smooth turns and gentle area changes and create as little flow resistance as possible. Also, the ports are positioned so that the fresh mix will impinge on the crank pin bearing and provide the required cooling.

The induction process is controlled by crankshaft-driven rotary valves. These valves are located between the rear of the engine housing and the bolt-on back plate. See Figure 3. The rotary valves, as shown in Figure 5, repeatedly cover and uncover the induction port, alternately sealing against the back-plate to maintain crankcase pressure and opening the induction port to allow the fresh mix into the crankcase. The rotary valve is considered by most two-stroke experts to be the best means of induction control for two-strokes. Jennings[4] says:

> "The rotary valve still is best in terms of sheer engine performance, whether arranged for maximum power or for an ultra-broad power range. The rotary valve is free of the really serious blow-back problem afflicting piston-controlled valving, and it offers much less resistance to flow than reeds. People who are currently so infatuated with the reed-valve concept should consider that in the world of

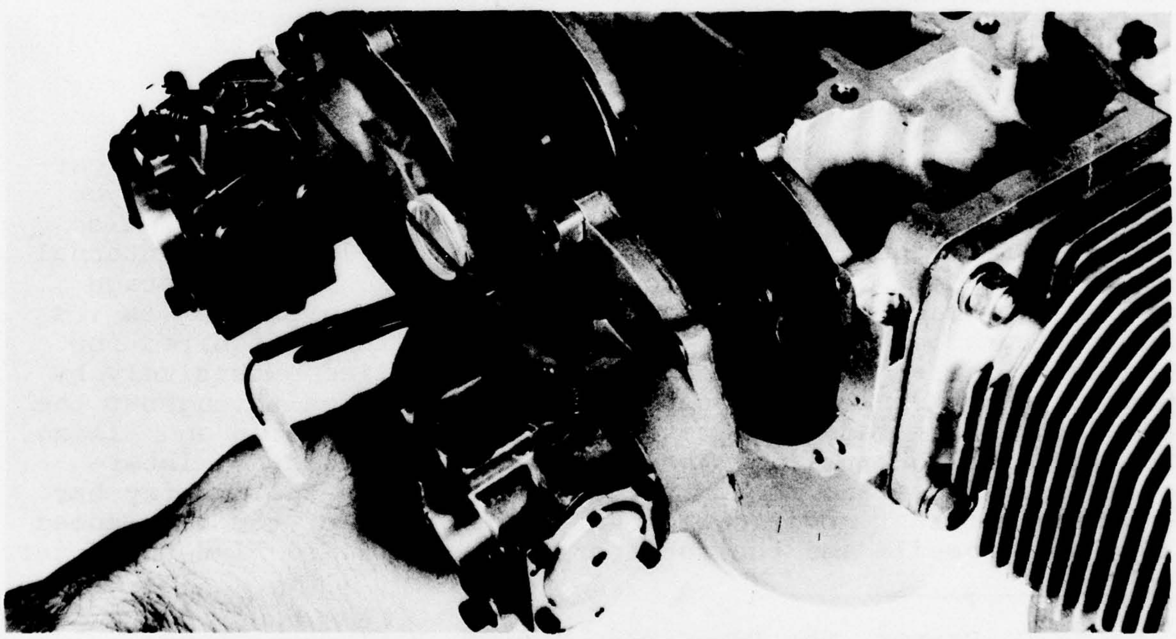


Figure 5. Rotary Valve Induction.

---

[4] Gordon Jennings, Two-Stroke Tuner's Handbook, Tucson, Arizona: H.P. Books, 1973, pp. 51-73, 88-90.

karting, where there is much more experience with both reeds and rotary valves than motorcyclists have accumulated, the two types of engines have been separated into different classes. Why? Because while the reed-valve engines are inexpensive, they cannot match the performance of those with rotary-valves."

The British two-stroke expert Draper[5] reiterates:

"Whilst rotary valves increase slightly the manufacturing costs, their employment improves volumetric efficiency throughout the speed range, starting is easier and petrol consumption is lowered by the elimination of loss of fuel blown back through the carburetter, thus also making for a cleaner engine. Volumetric efficiency is still further improved by the removal of the inlet port from its proximity to the hot cylinder."

The noted Italian engine designer, Bossaglia[6], also states:

"The preference for rotary disc valves in competition engines, and also recently for touring purposes, indicates the undoubted advantages of this design over the others previously described [third port induction, reed valve induction, etc.]."

The remaining component of the induction system is the carburetor. The BAT 282 uses two Walbro WB series diaphragm-type pumper carburetors. Pumper carburetors use the pressure pulse supplied from the crankcase through an external hose or internal pulse passage to draw fuel from the fuel tank. The advantage of the diaphragm over the standard float-type carburetor is its ability to accurately meter fuel in all attitudes required for maneuvers. The Walbro carburetors were modified extensively by Bennett Aerotechnical to provide smooth operation throughout the speed and power range and to improve fuel atomization and mixing. The modification included the addition of off-idle and intermediate orifices and the introduction of a high-speed spray bar. Another important modification was the removal of the high-speed adjustment needle and the incorporation of a fixed high-speed jet.

---

[5]K.G. Draper, The Two-Stroke Engine, Henley-on-Thames, Oxfordshire, England: G.T. Foulis & Co. Ltd., Fifth Edition, 1973, p. 46.

[6]Cesare Bossaglia, Two-Stroke High Performance Engine Design and Tuning, Chislehurst, Kent, England: Lodgemark Press Ltd., 1972, pp. 76-84.

This improvement means that there is one less adjustment required and one less possibility for operator error. The only carburetor adjustment to be made by the operator is the adjustment of the low-speed needle valve.

ENGINE VITAL STATISTICS

This section presents, in summary, some of the important parameters of the engine.

External Dimensions

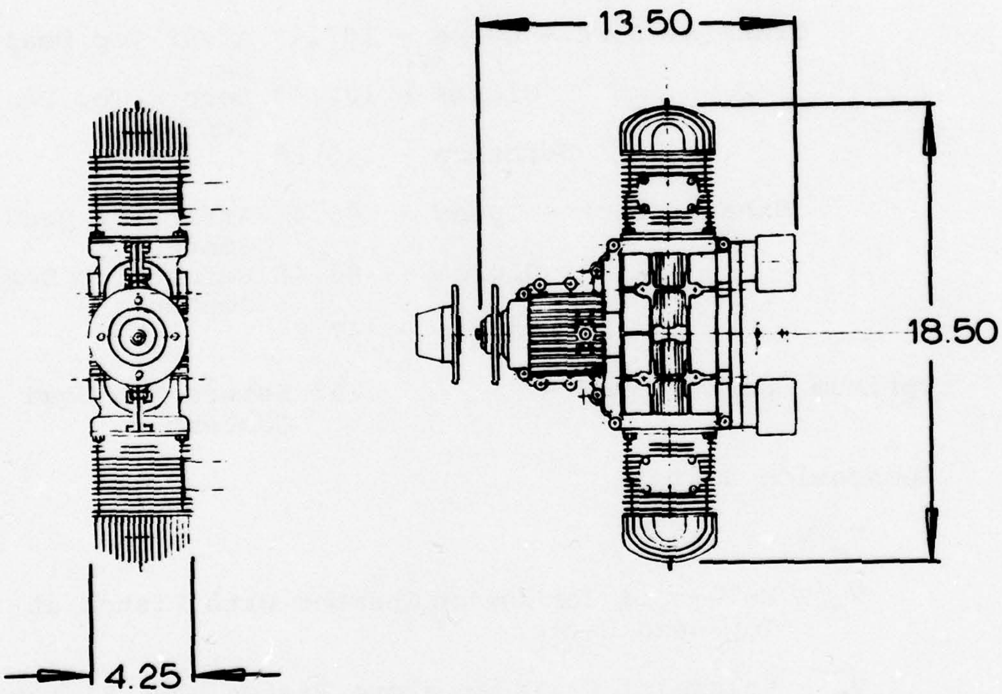


Figure 6. External Dimensions.

Engine Weight (see Appendix A for component weights)

Bare Engine	21.82 lb
Ignition System and Coils	2.72
Alternator	2.57
Power Conditioning Unit	5.67
Exhaust System (U-Tube)	3.41
All-up weight	36.19

Internal Dimensions

Cylinder Bore	2.50 in.
Stroke	1.75
Connecting Rod Length	2.375
Distance between Cranks	3.571
Total Displacement	17.18 in.^{3} (281.6 cc)

Engine Timing

Induction Port - Rotary Valve	Opens -	132° Before Top Dead Center
	Closes -	52° After Top Dead Center
	Duration -	184°
Transfer Port	Opens -	107.4° After Top Dead Center
	Closes -	107.4° Before Top Dead Center
	Duration -	145.2°
Exhaust Port	Opens -	96.4° After Top Dead Center
	Closes -	96.4° Before Top Dead Center
	Duration -	167.2°

Optimum Spark Timing - 26° Before Top Dead Center

Compresion Ratio

$V_C/V_E = 7.3$

V_C - Volume of Combustion Chamber with Piston at Top Dead Center

V_E - Volume of Cylinder above Piston when Piston is at Exhaust Opening.

Engine Speed

Maximum Design Crankshaft Speed - 8000 rpm

Range of Maximum Design Output Speeds (Gear ratio dependent) - 5200-12,240 rpm

DESCRIPTION OF ENGINE COMPONENTS

In this section the basic characteristics of each of the engine components are described. The manufacturer should be consulted for additional information on components not given here.

CYLINDER

The cylinder chosen for the BAT 282 is manufactured by Homelite for their Model 270 industrial engine. The cylinder has previously been used by Homelite as the basis for a high performance twin-cylinder snowmobile engine.

The cylinder is designed for use with rotary-valve or possibly reed-valve induction. It has a hemispherical combustion chamber with squish band. See Figure 7. The squish band design concentrates the mix in a tight pocket under the spark plug and prevents detonation which normally occurs away from the plug. This type chamber is the most popular for high-performance, high-compression-ratio (greater than 6.5:1) engines.

The Homelite cylinder has splined transfer ports as shown in Figure 7. This design is based on ease of manufacture and until

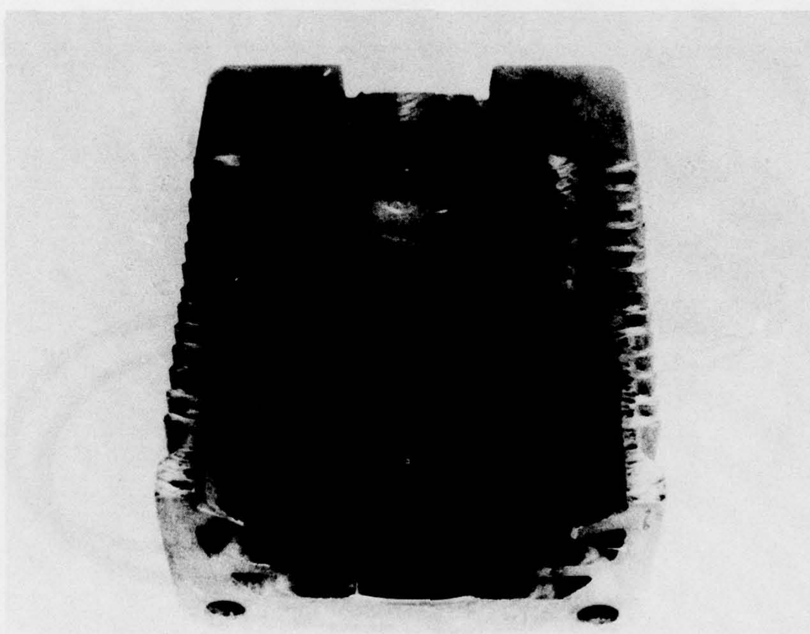


Figure 7. Homelite 270 Cylinder Cutaway View.

recently was thought to be inferior in terms of performance. But recent unpublished results by B. L. Shaeffer show that splined transfer ports offer some performance advantages.

Other features of the cylinder are generous use of cooling fins which will insure adequate cooling and a chromed bore instead of a heavy steel liner.

The only major fault to be found with the cylinder is the bridged exhaust port. It is of an older design. Casting changes could be made to replace the two bridges by a single bridge or substitute a modern bridgeless exhaust port. The exhaust port has caused no problems.

The diameter of the cylinder bore is 2.5 in.

PISTON AND RINGS

It was originally planned to use the Homelite 270 piston, but some shortcomings in cooling and pin location dictated the design of a new piston and rings (see Figure 8).

The piston has a conical crown to match the cylinder squish band. The piston pin is located further from the piston crown than the Homelite piston to offer adequate cooling for the pin and upper bearing. The Homelite piston was designed to operate

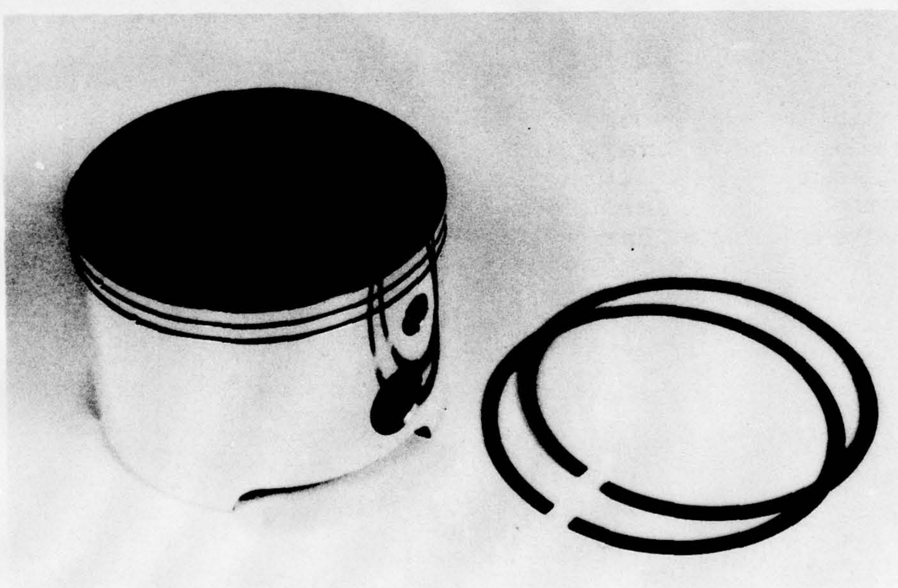


Figure 8. Piston and Rings.

at 3600 rpm and did not have a cooling problem at that speed. Cooling of the pin is further enhanced by the cooling holes in the side of the piston which allow fresh mix to flow through the interior of the piston into the transfer ports. These holes also provide some additional transfer area.

The piston is made from a high silicon aluminum alloy, A-132 T-65.

The new piston design also offered the opportunity to use thin (.024 in.) tool steel rings. Thin rings offer better performance because of reduced friction losses.

CONNECTING ROD, ROD BEARINGS, AND GUIDE WASHERS

The connecting rod and both rod bearings also come from the Homelite 270. The rod is of split design. It is made in one piece and then fractured. This technique gives a better rod fit. The rod was initially thought to be the weak link in the engine, but it has given no problems at all. The connecting rod is 3.375 in. in length. See Figure 9.

The upper end bearing is a drawn cup needle bearing manufactured for Homelite by the Torrington Company. It has a .5-in. inside diameter. It is supplied by Homelite preassembled in the connecting rod.

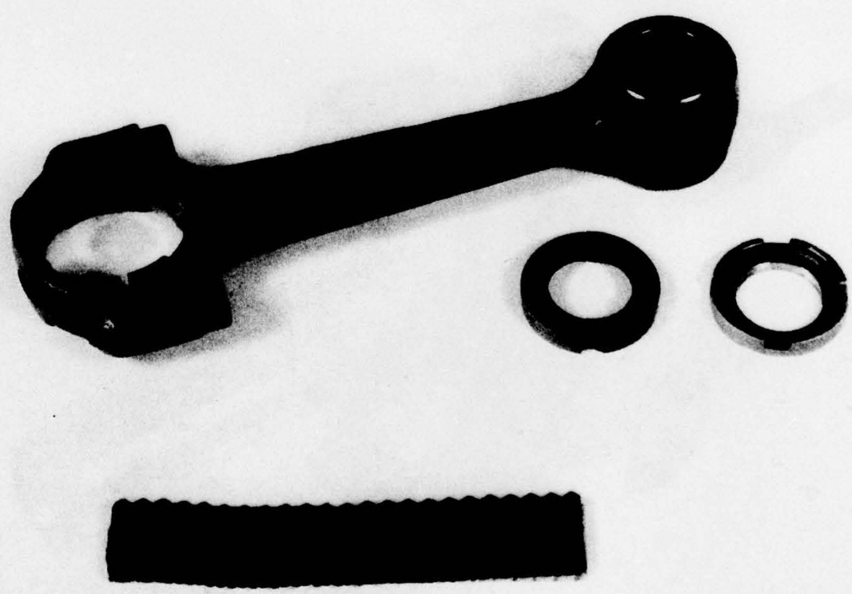


Figure 9. Connecting Rod, Rod Bearings, and Guide Washers.

The lower end bearing consists of 23 loose rollers. They are also manufactured by the Torrington Company and supplied to Homelite in a waxed strip.

Neither of these bearings employs a cage, as some two-stroke designers suggest. A caged bearing becomes important at very high speeds. At the lower speeds (8000 rpm) at which the BAT 282 runs, the cage would be the weakest link, especially in the case of a split cage design.

In the Homelite design (3600 rpm) the connecting rod was guided axially on the crankshaft. To increase the operational speed capability, the connecting rod was narrowed on the bottom and two piston guide washers were installed on the piston pin to top guide the rod. The piston guide washers are made from steel and are shaped to fit over the upper end bearing. The guide washers have slots as shown in Figure 9 to allow a sufficient supply of oil to the upper end bearing.

CRANKSHAFT

A photograph of the crankshaft is presented in Figure 10. This crankshaft was made from a steel bar. Earlier shafts were made from Homelite forgings, but insufficient material was available in the forgings and the cranks were too weak.

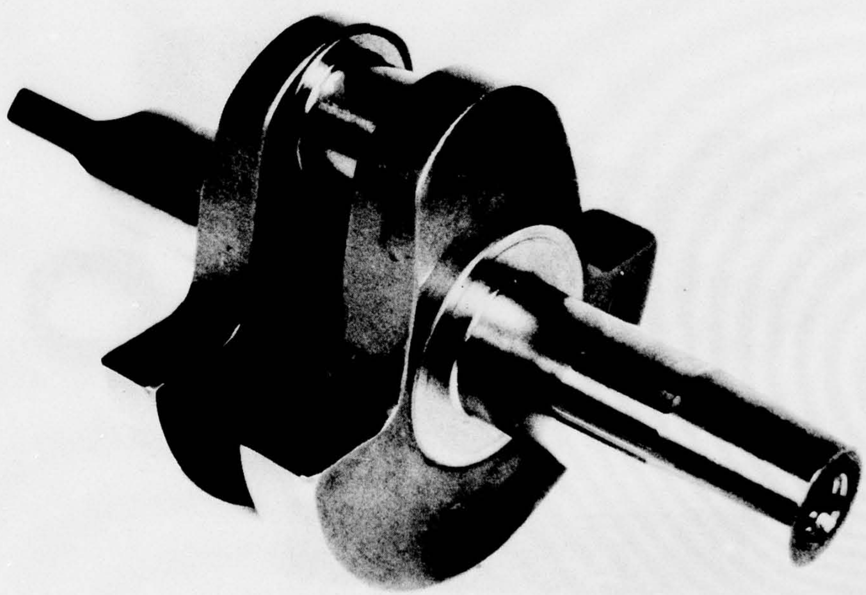


Figure 10. Crankshaft.

The crank has a 1.75-in. stroke. It has a taper on the front end to accept a gear and a flatted round tang to drive the rotary valve on the rear. The crankshaft is designed according to the balance criteria presented under Engine Design. SAE 8620 steel is used in the manufacture of the shaft. It is case hardened by carburizing to 60 Rockwell C scale.

MAIN BEARINGS, THRUST BEARINGS, AND WASHERS

The four main crankshaft bearings are heavy cased needle roller bearings made by INA Bearing Company. The bearing has a 19 mm bore, 27 mm outside diameter, and is 20 mm long. It has a dynamic load rating of 2080 lb and a static rating of 2765 lb. It is rated at speeds to 21,000 rpm. Under the conditions present in the BAT 282, the bearing has a minimum expected lifetime of 584 hours. The bearings and thrust washer are shown in Figure 11.

The needle thrust bearings and thrust washers serve to accept the thrust loads imposed on the crankshafts by imperfect gear loading. The thrust bearing has a dynamic load rating of 1770 lb, a static rating of 2690 lb, and a speed rating of 14,300 rpm. These components are made by the Torrington Company.

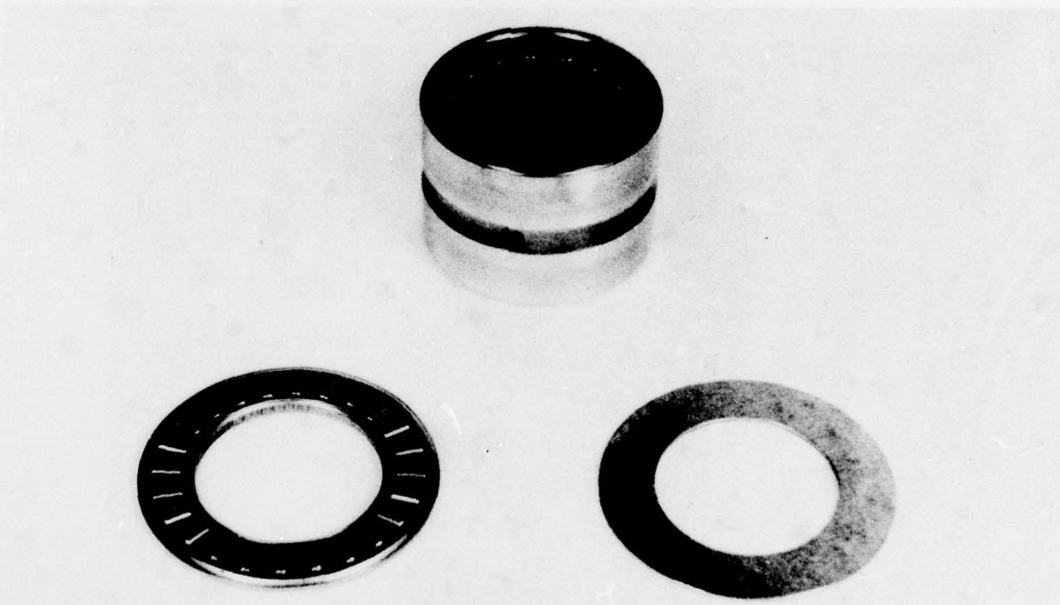


Figure 11. Main Bearing, Thrust Bearing and Washer.

SEALS

The pressure seal between the crankcase and gearbox is maintained by the smaller of the two seals shown in Figure 12. A spring-loaded single lip design is employed. The seal has a steel outer ring and the sealing member is made from Viton, which is the best commercially available seal material. Two of these seals are required in each engine.

A spring-loaded double lip design is used on the propeller shaft. The external lip serves to prevent foreign matter from working into the gearbox. Polyacrylic nitrile is used in the sealing member. A thin steel housing supports the sealing member on both the outside diameter and on the face near the sealing surface. The propeller shaft seal is the larger of the two seals shown in Figure 12.

The crankshaft seal and the propeller shaft seal are a special design made for the BAT 282 by NOK.

GEARS AND PROPELLER SHAFT

Gearing in the BAT 282 has been discussed previously under Engine Design. This section will present some of the details of the final design.

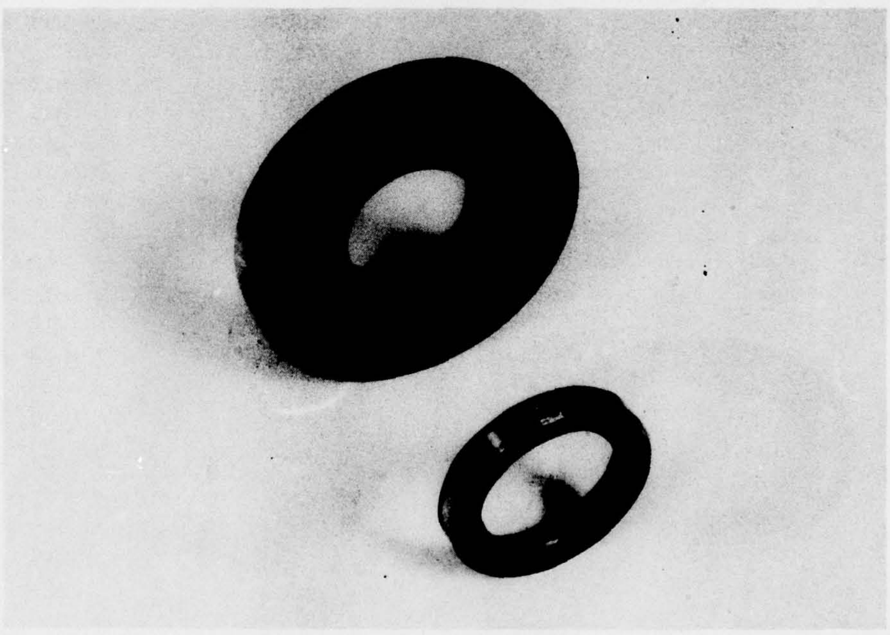


Figure 12. Seals.

All three gears are made from the highest quality gear material, SAE 9310 nickel alloy steel. The gears are Module 2.5, full depth, 20° pressure angle. The Metric Module gear was chosen over the standard 10 pitch for its increased contact area. Module 2.5 is equivalent to a diametral pitch of 10.16. The center distance between crankshaft gear and propeller shaft is 1.785 in., which gives in one configuration a 17-tooth crankshaft gear and an 18-tooth propeller shaft gear. Of course, other gearing configurations may be and have been used. The face width of the crankshaft gear is .71 in. and the propeller shaft gear is wider at .77 in. This overlap in the mating gears prevents high stress edge loading.

The gears are case hardened by carburizing to approximately 55 Rockwell C scale and an effective depth of .030 in. The propeller shaft is double heat treated to increase the effective depth of the case to .060 in. on the rear journal.

A taper fit is used to attach crankshaft gear to crankshaft and to attach propeller flange to propeller shaft. (The gear and crankshaft are keyed to insure proper alignment.) The central gear is integral to the propeller shaft.

Figure 13 shows both the crankshaft gears and propeller shaft. Additional features of the propeller shaft to be noted are the ground diameter onto which the alternator rotor is pressed and the groove which holds the bearing retaining ring.

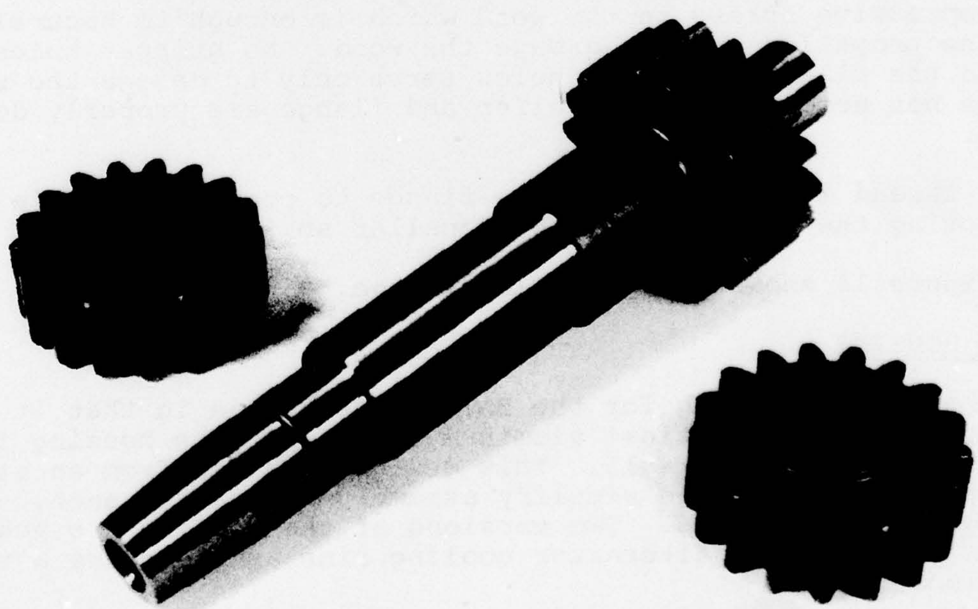


Figure 13. Crankshaft Gear and Propeller Shaft.

PROPELLER SHAFT BEARINGS

The rear propeller shaft bearing is a needle roller bearing identical to the main crankshaft bearings.

The front bearing is a high production 6204 ball bearing. It has a 20 mm bore and an outside diameter of 47 mm. Load ratings are 2200 lb dynamic and 1370 lb static. The bearing is rated for speeds up to 15,000 rpm.

A snap ring is included with the bearing for axial location in the housing.

This 6204 ball bearing performed without problems throughout all testing.

PROPELLER FLANGE

The propeller flange is secured to the propeller shaft by a taper fit held in place by a 3/8 in screw. The flange is not keyed to the shaft since relative orientation is not important.

The flange was designed to optimize propeller retention. A 4 in diameter was chosen to provide sufficient area and moment arm to grip the propeller. The bolt circle for the propeller retaining screws was sized to give equal area inside and outside the bolt circle to provide uniform compressive stresses in the wooden propellers. Six No. 10-24 screws are used to retain the propeller. When torqued to 50 in.-lb they provide about 600 psi compressive stress in the wood which is enough to securely hold the propeller yet not damage the wood. No gripper holes are used in the flange. Gripper holes serve only to damage the wood and are not necessary if propeller and flange are properly designed.

A thread is cut right on the flange to provide a simple means of removing the flange from the propeller shaft.

Figure 14 shows the propeller flange.

ENGINE HOUSING

The engine housing for the BAT 282 is unique in that it is machined from two identical aluminum castings. The housing is split as shown in Figure 15. This design resulted from an attempt to reduce cost and simplify assembly and maintenance. The casting alloy is 356-T6. Two versions of the housing are available, a version with alternator cooling fins as shown and a version without fins.

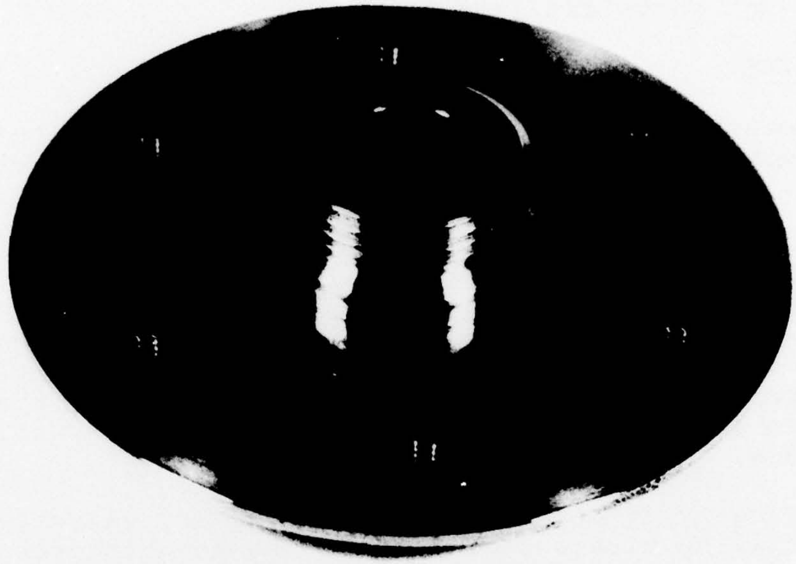


Figure 14. Propeller Flange.

Figure 15. Engine Housing.

Machining of the castings includes mating surfaces, bearing bores, alternator bore, cylinder pads, mounting pads, rear surface, snap ring groove, clearance holes, and threaded holes. The crankcases and ports are left as cast.

ROTARY VALVES

The rotary valves are formed by a thin flapper valve steel disc sandwiched between two hardened steel hubs and retained with six screws. Figure 16 shows the rotary valve.

BACKPLATE

Both sides of the backplate are shown in Figure 17. It is also a 356-T6 aluminum casting. The backplate contains mounting stacks for the carburetors as well as the initial portion of the induction port. It serves as a sealing surface for the rotary valve and in it are mounted the ignition pickup as well as a timing window.

Machining is limited to facing the rotary valve sealing surface, drilling clearance holes for the mounting screws, drilling and tapping threaded holes for carburetor retention, milling a slot for the ignition pickup, and drilling and tapping for the timing window.

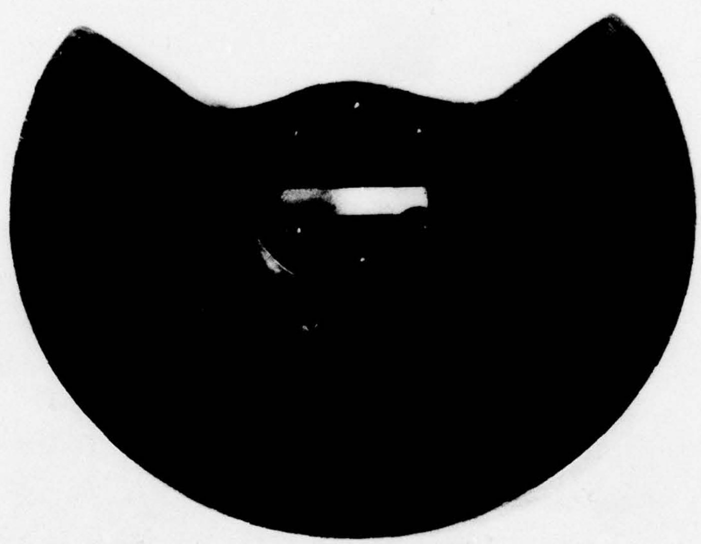


Figure 16. Rotary Valve.

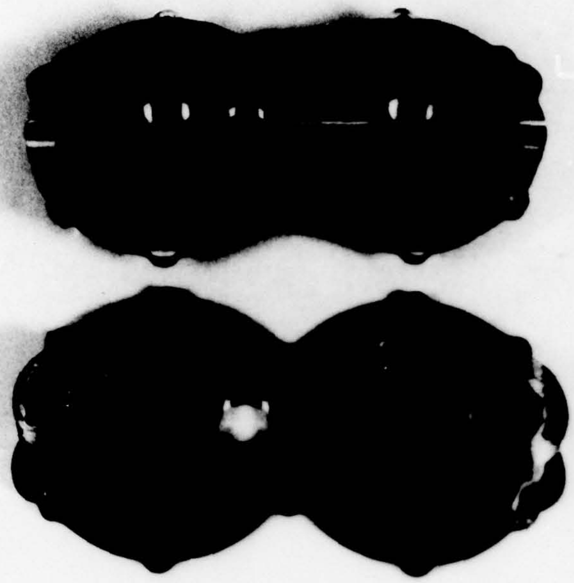


Figure 17. Backplate.

CARBURETORS

Twin Walbro WB series diaphragm-type carburetors are used on the BAT 282. Modifications and improvements to the basic WB carburetor are discussed in the section on Engine Design. The carburetor is shown in Figure 18.

EXHAUST SYSTEM

The simple, or straight exhaust system consists of the flange and elbow shown in Figure 19. The tuned exhaust system consists of the flange, the elbow, and a U-tube expansion chamber, shown in Figure 20. The physical principles behind the operation of the system are presented in the section on Engine Design.

The exhaust flange was designed during the contract to give a smooth, unrestricted transition from the roughly rectangular exhaust port to the round outlet. It is another 356-T6 aluminum casting. It replaced the restrictive Homelite flange.

The exhaust elbow is made by bending common aluminum tubing. The elbow is joined to the transition by welding.

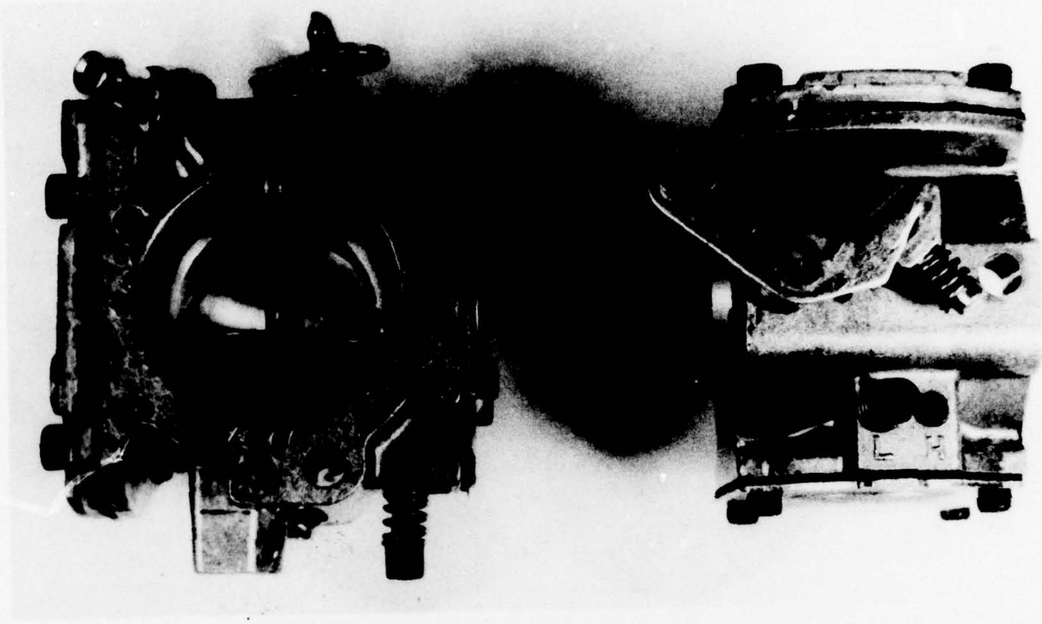


Figure 18. Carburetor.

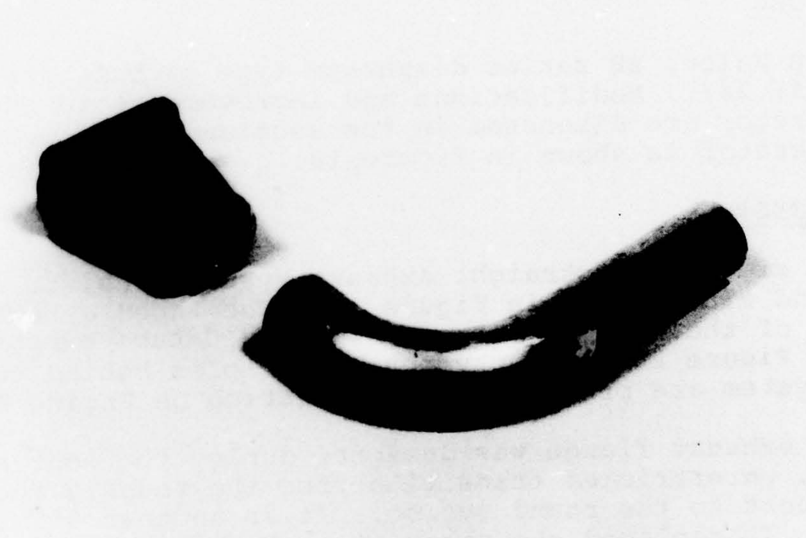


Figure 19. Exhaust Flange and Elbow.

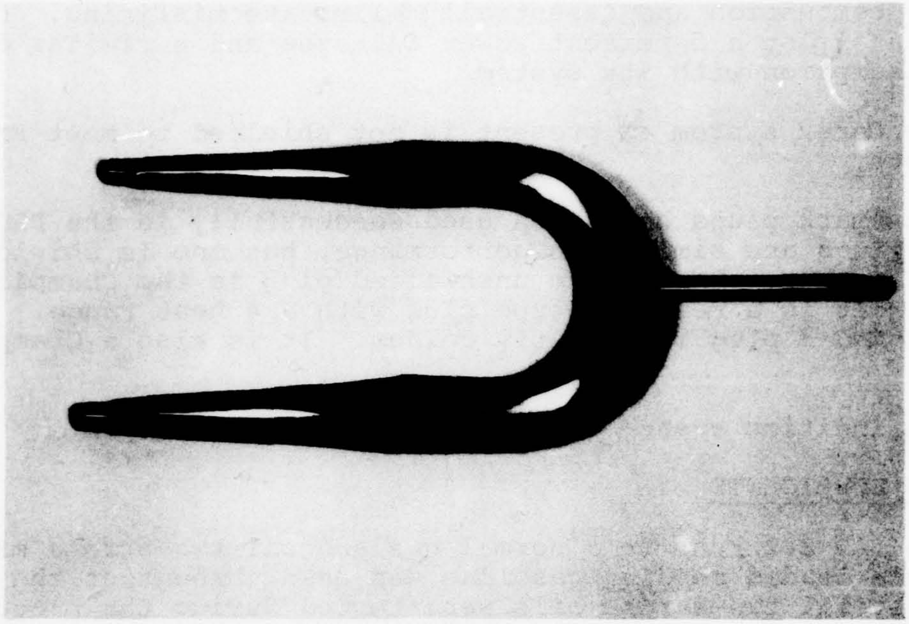


Figure 20. U-Tube Exhaust System.

The most interesting piece of the exhaust system is the U-tube itself. It is made from two flat sheets of aluminum cut in the shape of the U and with the width of the sheet corresponding to the half circumference of the final round section. These two pieces are welded together along their edges and then inflated to the desired shape by some proprietary process. The U-tubes were the design of John Brooks and were manufactured by Mueller-Kautzer Welding.

The details of the U-tube design are presented in Appendix B.

IGNITION SYSTEM

The ignition system used on the BAT 282 is a solid state capacitive discharge system based on a production unit made for motorcycles by Gerex Company. Gerex modified the stock unit to run on 28 volts and reworked their Hall effect pickup system to trigger off the trailing edge of the rotary valve.

The unit has high quality components and supplies extremely high energy to the plugs. Yet the power consumption of the system at 7500 rpm was measured at only 21 watts. Another important feature of the system is multiple sparks. About half of the energy is supplied to the plug at the normal firing point, followed by 10-13 weaker sparks. The multiple sparks insure complete combustion and essentially eliminate misfiring. Gerex expects at least a 5-percent power increase and a similar drop in fuel consumption with its system.

The Gerex system at present is not shielded to meet EMI requirements.

Two spark plugs have been used successfully in the BAT 282. The two plugs are similar in performance, but one is shielded while the other is not. The unshielded plug is the Champion RCJ-585. It is a resistor type plug with a 4 heat range. The shielded RMJ-3 plug is slightly colder. It is also a Champion plug.

The ignition system is shown in Figure 21.

FUEL AND LUBRICANTS

The BAT 282 runs on a normal gas and oil two-stroke mix. Pump grade leaded regular gasoline was used throughout the program. Several two-stroke oils were tested during the program with varying degrees of success. Finally a two-stroke oil called MC-1 made by Belray Company was tested. This oil was found to be superior to all the others tested in every category. The oil is a synthetic. It reduces wear in the engine to the point that it is undetectable. It runs very clean. An engine that was run

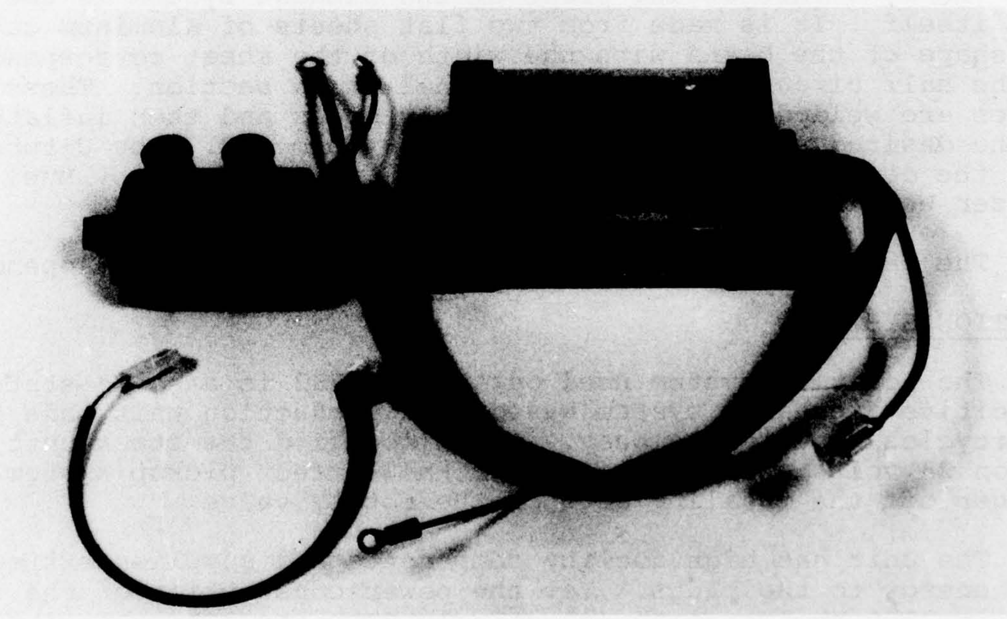


Figure 21. Ignition System.

with another two-stroke oil and accumulated considerable carbon deposits was cleaned in about an hour of running on MC-1. Being a synthetic, very high gasoline-to-oil mixture ratios can be used. The normal mixture ratio is 50:1.

Several gear lubricants were also tested during the course of the program. The one proven to be the best was Almasol #607 made by Lubrication-Engineers. Almasol is a paraffin-based oil with extreme pressure additives. Its high film strength helps to keep the gear teeth separated, reducing wear. About 100 ml of gear lubricant is used in the gearbox.

ALTERNATOR/PCU

The permanent magnet alternator/power conditioning unit (PMA/PCU) system is comprised of a single power generation circuit with a rectified DC output and a fully regulated power conditioning unit.

The PMA consists of a rare earth samarium-cobalt permanent magnet rotor and a three-phase stator. It is capable of delivering in excess of 40 amperes at 28 VDC through the PCU. Only minimal cooling of the alternator is required due to its inherently low winding resistance.

A shielded interconnecting cable is used to connect the PMA output to the PCU. Wire size was selected to minimize voltage drops and limit temperature rise. Shielding was selected to effectively reduce EMI.

The PCU consists of a full wave bridge, a series switching regulator, signal conditioning, and circuit protection sufficient to meet criteria of the designated specification.

A photograph of the PMA/PCU is presented in Figure 22.

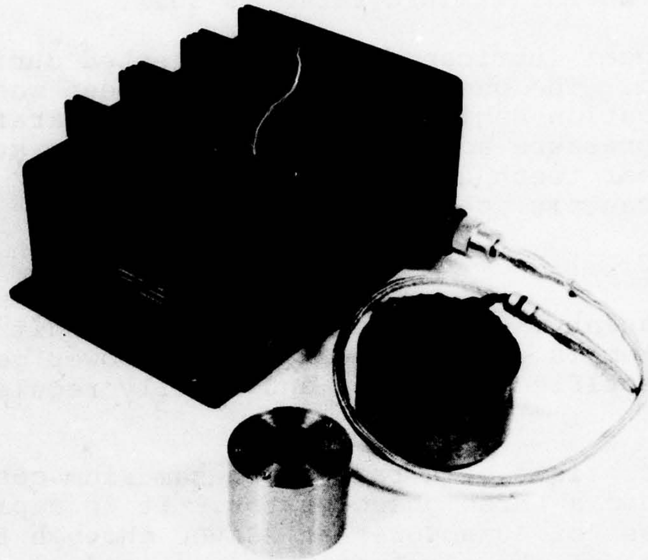


Figure 22. Alternator and Power Conditioning Unit.

FABRICATION AND PROCUREMENT

One of the major tasks of the contract was the procurement of engine components. Both high production off-the-shelf items and newly designed items requiring from-scratch fabrication were required.

The standard components employed in unmodified form from the Homelite Model 270 were obtained in a timely fashion and no procurement problems were encountered for these parts. Similarly, the procurement of other high production items such as bearings, fasteners, etc., presented no problems. The manufacturers and distributors showed interest in the engine and supplied the technical information desired and insured that parts were available.

The newly designed components presented a different question. What is the best approach to take in the fabrication of low production engine components? This question was not only important during the demonstrator phase but will also be important during the production phase where production levels will still be quite low. Three possible approaches were considered: (1) in-house fabrication, (2) subcontracts to large manufacturers of engine components, and (3) subcontracts to specialty job shops.

Total in-house fabrication was only briefly considered. In a low production program, where there is sufficient work to keep man and machine busy, in-house machining of several of the key components such as housings might be reasonable. Regardless of production levels, some in-house fabrication capabilities are required for the fabrication of numerous jigs and fixtures, modification to engine parts, and fabrication of several components. Most of the propeller flanges, backplates, and exhaust flanges were machined in-house.

During the program, several large engine manufacturers were contacted about possible manufacture of parts. These companies included manufacturers of small engines and aircraft engines. They offered high quality components, modern equipment, production experience and a capability for high production. They also offered high tooling costs, high unit production costs, and generally unacceptable deliveries. The problem was extremely high labor rates and overhead and their unwillingness to put aside manufacture of high production items to do a short run of parts for the demonstrator engines or for low production rates.

The specialty job shop approach was determined to be the best, both for this demonstrator program and for production. High quality, low cost, and good deliveries are available from these specialty job shops.

TESTING

A large portion of the program consisted of testing and validating the engine design. Early in the program an Engine Test Plan was prepared which included inspection of parts, assembly checking, engine break-in procedures, selection of spark and induction timing, endurance test procedures, component test procedures, and criteria for the evaluation of component performance. The plan was followed to the extent possible. Early problems with rotary valves and bearings required special testing and deviations from the test plan.

TEST APPARATUS

The propeller test stand used for most of the testing was designed in-house and is shown in Figure 23. The engine is mounted in a three-point c.g. mount. The propeller is totally enclosed by rolled aluminum plate and wire screen. Engine starting is accomplished through a rubber-lined friction cone powered by an automotive starter motor. Two 12-volt batteries wired in series supply the 24 vdc power for both starter and ignition system.

The tachometer used for measuring engine speed is an Accutac electro optical device made by BWT Systems.

Temperature measurements were made on the prop stand and dynamometer using a Bailey Instruments Model TZF-HR pyrothermometer and thermocouple probes. Worst-case accuracy is $\pm$ 2°F.

For checking spark timing a Sears Model 244.21682 inductive pickup timing light was used on both the prop stand and dynamometer.

The dynamometer is an Eaton Dynamatic Model 758DG. It is an eddy current inductor dynamometer rated at 50 horsepower and 8000 rpm. The dyno and auxiliaries are shown in Figure 24.

Engine speed is measured on the dynamometer by a tach-generator and an electromagnetic sensor which senses a 60-tooth gear mounted on the dynamometer shaft. RPM is output in both analog and digital form.

The load absorbed by the dynamometer is measured by a Toledo hanging scale Model 2110. The load is applied at a moment arm length of 1 ft and gives a torque reading directly in foot-pounds.

Fuel flow is measured by an electro-optical device. The device senses the displacement of an opaque marble which is proportional to the flow rate. The flow meter is a modification of a unit distributed by Heathkit; Model Cl-1078. The guaranteed

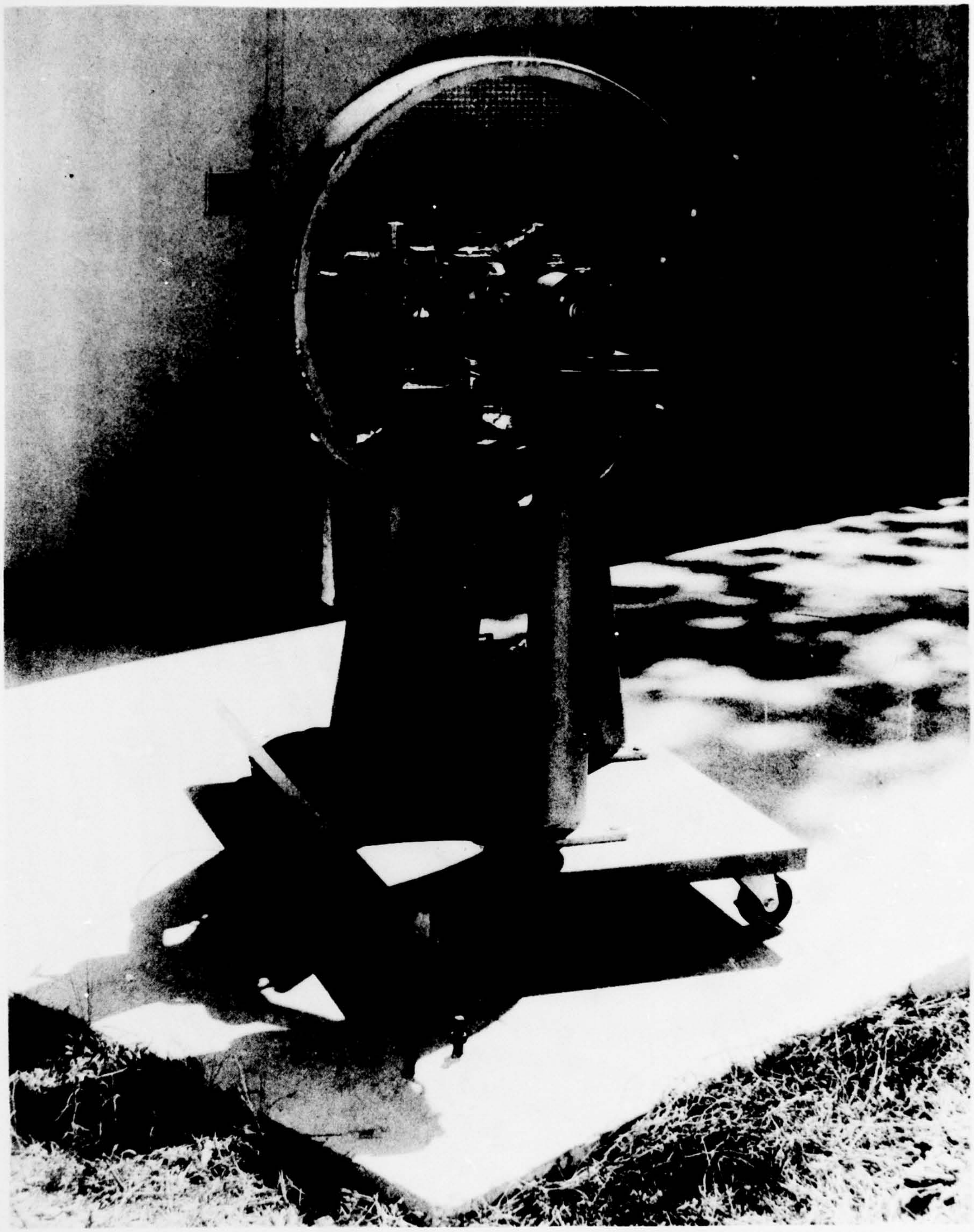


Figure 23. Propeller Test Stand.

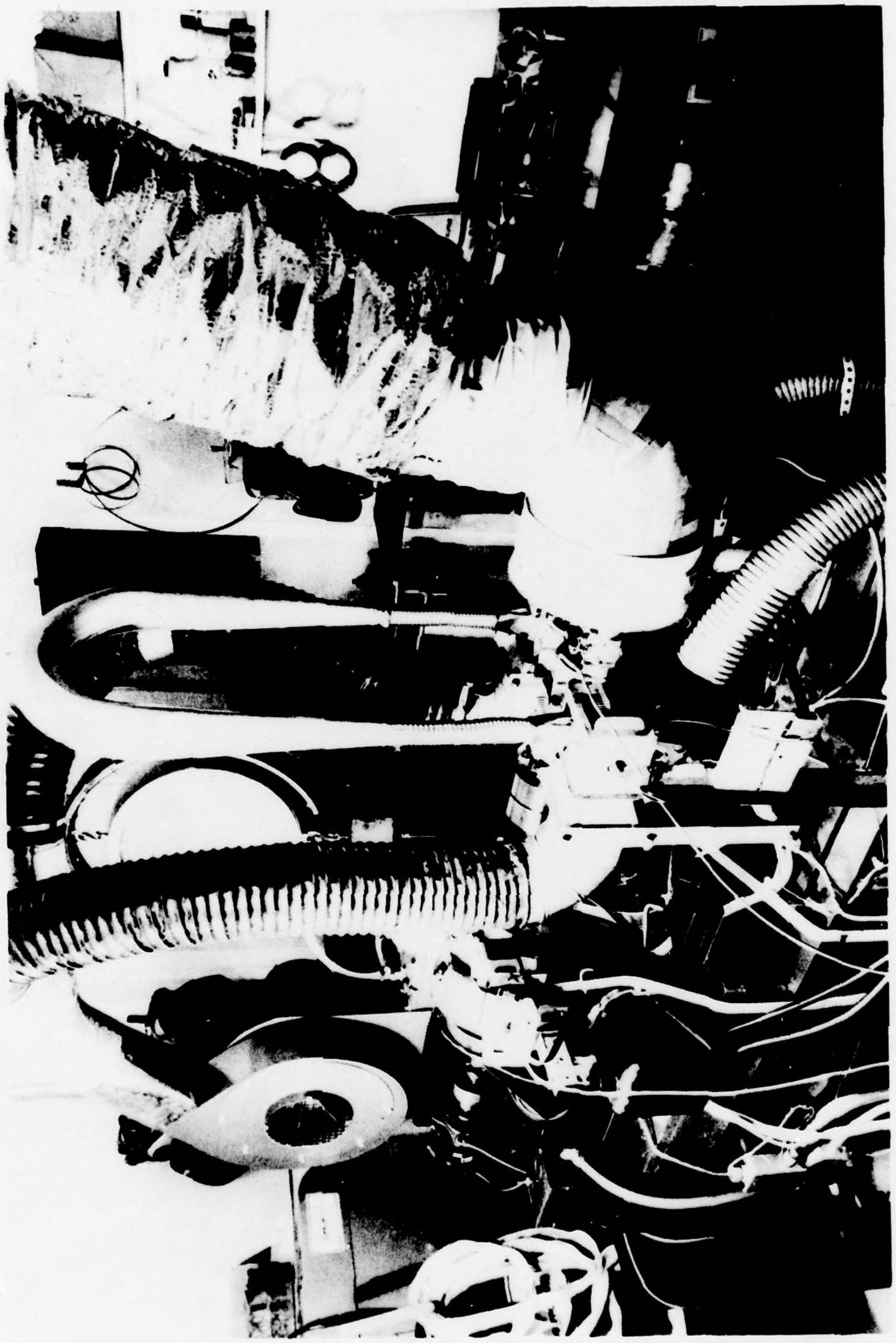


Figure 24. Dynamometer and Auxiliaries.

accuracy is ± 5 percent; however, frequent and careful calibration should improve the accuracy to ± 2 percent.

The airflow measurement equipment consists of a Meriam laminar flow element, Model 50MCZ-2, a Meriam Model 40 HE35 WM inclined manometer, 10 inch diameter insulated ducting, and a large plenum chamber. The laminar flow element is rated at 100 cubic feet of air per minute and calibrated to ± 5 percent. The inclined manometer has a 30-inch scale and a pressure range of 8 inches of water.

Engine cooling on the dynamometer is provided by a 1200 CFM High Pressure Dayton Blower, Model 4C329. Air from the single blower is ducted to the two cylinders and directed around the cylinder by cooling shrouds.

Alternator performance was measured with a resistive load bank equipped with a voltmeter and ammeter. Switching is arranged so that three different loads are available: about 400, 800, and 1200 watts. This unit was used on both the prop stand and dynamometer.

PROPELLER STAND TESTING

All of the early engine testing was performed on the propeller test stand. Almost all of the important parameters of an engine can be studied on the propeller stand with test equipment limited to a tachometer, a timing light, and a temperature measuring device. If calibrated propellers are used, power can be calculated with reasonable accuracy by the cube law.

Several different propellers were used during testing to provide different loads at different speeds and to develop the retention technique. The first propellers used were stub test clubs. These were abandoned due to their high moment of inertia which resulted in some of the early gear problems. The most satisfactory propellers used were made by Sensenich and Kolbo. Table 1 describes the propellers used. Most of the testing was done with tractor propellers, although brief testing was performed with pusher propellers.

One advantage of prop stand testing is that troubleshooting is facilitated since auxiliary equipment such as blowers does not hide the sound of the engine and modifications can often be made in place. Another important advantage is that it more nearly simulates the real operational environment. Also, the soft engine mounts used on the prop stand are much easier on the engine than the hard mounting required on a dynamometer.

Initial engine testing could be categorized as development testing. The engine was set up for low performance to evaluate its basic running characteristics. Parameters such as spark

TABLE 1. PROPELLERS USED IN TESTING

Manufacturer	Model	Pusher-P Tractor-T	Diameter in.	Pitch	Estimated Rating
Arrowprop	Test Club	T	20	--	25 hp @ 6000 RPM
Arrowprop	Test Club	T	20	--	20 hp @ 6000 RPM
Arrowprop	Test Club	T	20	--	28 hp @ 6000 RPM
Sensenich	W26GX-13	T	26	13	14 hp @ 7000 RPM
Kolbo	2811T	T	28	11	25 hp @ 7500 RPM
Kolbo	2811P	P	28	11	25 hp @ 7500 RPM
Kolbo	2724T	T	27	24	26 hp @ 6000 RPM
Kolbo	2724P	P	27	24	26 hp @ 6000 RPM
Sensenich	W-28GX-15	T	28	15	25 hp @ 7500 RPM
Sensenich	W-28GXL-15	P	28	15	25 hp @ 7500 RPM

timing, induction timing, and compression ratio were the subject of tests. Engine components were evaluated and clearances between mating parts examined. A great deal of carburetor testing was performed, resulting in a significantly improved carburetor. Endurance tests were performed to determine the fatigue characteristics of some of the key components. Crankshafts, gears, and rotary valves were the subject of much of this testing. Tests of the alternator and exhaust systems were run. Performance tests including power, fuel consumption, throttle response, and idle were conducted.

DYNAMOMETER TESTING

After the bugs were out of the engine and good estimates of carburetion, timing, etc., were obtained, dynamometer testing began. To get the dynamometer to function properly required almost as much development as the engine. The problems encountered with the dynamometer were the usual problems with couplings, cooling blowers, and torque measuring equipment.

After the problems were solved, the dynamometer proved to be a valuable tool. One of its functions was to provide more accurate data than that obtainable from propeller tests for the fine tuning of carburetors, the exhaust system, and induction and spark timing. A laminar flow element and inclined manometer were installed as part of the dynamometer setup and used for airflow measurement. This data was extremely valuable in the analysis of the engine's "breathing".

After the tuning stage of testing, the dynamometer facility was used to generate a complete map of the engine, including speed, rpm, torque, fuel flow, airflow, carburetor inlet temperature, and cylinder head and exhaust temperatures. Performance results are presented in the section on Engine Performance/Test Results.

PROBLEM AREAS/DESIGN IMPROVEMENTS

The design of a new piece of machinery invariably goes through a series of changes and improvements as testing reveals problem areas and attempts are made to upgrade performance. This section describes the design evolution of the BAT 282, tracing some of the troublesome components from original to final design and noting some of the improvements and simplifications made during the program.

Three basic design iterations were accomplished during the program, although several more design iterations on specific components were required to arrive at the final design. Two iterations were performed before the first five engines were delivered to the Army. These iterations including initial design were accomplished in 16 months. A few problems, including a fatigue problem with the crankshaft and an ignition problem, still existed at this time. These problems had been recognized, but insufficient time for manufacture of new parts prevented their correction in the first delivery engines. The third iteration was completed by the 22nd month of the contract. This iteration resulted in what is referred to as the "final design" in this report.

ROTARY VALVES

The first and most severe problem encountered during early testing was breakage of rotary valves. The loads on the rotary valves were just too high for the material and design chosen. Modification to the valves was complicated by the requirement that it perform the additional role of magnet carrier. A 3/32-in.-thick magnet rotated with the rotary valve and was used to trigger the ignition system.

The original rotary valve design consisted of a 1/8-in.-thick phenolic disc with a drive hole in the center as shown in Figure 25. A valve of this design normally lasted about 5 minutes when the engine was run at moderate speeds.

The problem was diagnosed as a materials problem and the search for a suitable material began. No suitable plastic could be found. The next choice was a thin (.020-in.) stainless steel valve reinforced in the area of the drive hole by welding on a thicker hub. A 300-series stainless steel was chosen because of its strength and the valve's secondary role as magnet carrier which restricted the choice to nonmagnetic materials. The magnet was pressed into a hole swaged in the valve face and further held by epoxy. This valve performed better than the phenolic, but the welds failed after a short period of running and the magnet retention was unsatisfactory. This iteration did prove the value of a hub for the rotary valve.

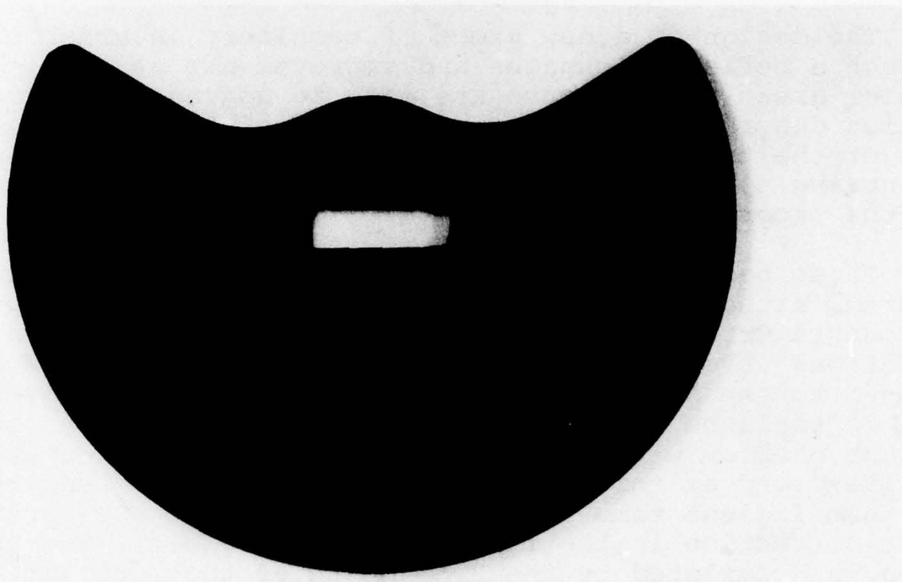


Figure 25. Original Phenolic Rotary Valve.

In an attempt to circumvent the weld and magnet retention problems, a design consisting of a 1/8-in.-thick phenolic valve sandwiched between the halves of a two-piece hardened steel hub and fastened with screws was tried. This valve showed promise. However, the phenolic began to compress after a few hours and the screws loosened. Also, the magnets loosened after a short period and worked themselves into prominent places like between the piston and squish band of the combustion chamber. A valve material that would not compress and a better means of magnet retention were required. This testing did, however, show the merits of a hardened steel hub retained by screws.

In the next design the phenolic was replaced by a valve consisting of a 3/32-in.-thick aluminum core which carried the magnet and two .015-in. stainless steel faces which held the magnet in place. The hardened steel hub and screws were again used. This design obviously solved the magnet retention problem. The all-metal valve was quite heavy, however. This weight served to further increase the load on the hub and crankshaft drive tang. After 4 or 5 hours of run time, fractures began to show up on the hub and after about 35 hours, breaks in the crankshaft drive tang occurred. A fix for the hub problem was shot-peening which greatly increased the fatigue life of the hubs. To stop the crankshaft tang breakage problem, a stronger tang and a lighter rotary valve were required. Strengthening the drive tang was the easier of the two tasks.

The ignition system was giving problems at this time and a change was required. A new ignition system was found that did not require a magnet in the trigger system. This system triggers as the trailing edge of a permeable steel disc passes the sensor. Without the burden of the magnet, the rotary valve was simplified to a single thin sheet of spring steel sandwiched between the two-piece steel hub. This configuration worked well, but a few minor changes were required. In the final design the spring steel was replaced by Swedish flapper valve steel which is more fatigue resistant and the hub diameter was increased for additional strength.

A flapper valve steel rotary valve with a pre-final design hub ran for over 70 hours without showing any sign of damage.

GEARS

The original design called for the crankshaft gears to be made from SAE 9310 steel and the propeller shaft with integral gear to be made from SAE 8620 steel. SAE 9310 steel is the best gear steel made for high shock load application, and the 8620 steel was chosen for its machinability and good shock load properties. The gears were cut to 16 pitch, with 20° pressure angle and full depth teeth, and the teeth were shaved. The face width of the crankshaft gears was .625 in. and prop shaft gears had a face width of .685 in. The different face widths were chosen to eliminate as much edge loading as possible. The gears were specified to be case hardened by carburizing to a depth of .010 to .020 in. and a surface hardness of 60 Rockwell C scale. These gears were of marginal design and, with the further degradation of heat treating which was not to specification, some of them failed in less than 1 hour.

The design modifications incorporated in the next batch were:

Face width increased by .030 in. on both crankshaft gear and prop shaft gear

Propeller shaft material changed to SAE 9310 steel

Gear teeth changed to coarser 14 pitch stub design

Gear teeth ground instead of shaved

Hardened case thickened to .012 to .023 in.

Gears shotpeened.

These gears were far superior to the previous design and the ones which were properly heat treated ran over 50 hours without signs of damage. After that period signs of surface fatigue appeared on the teeth due to the insufficient depth of the case. The ground teeth appeared to show no advantage over shaved teeth.

In the final gear design an even coarser pitch (Metric Module 2.5) was chosen to give a tooth of sufficient thickness to allow a .030- to .040-in. hardened case without becoming too brittle. The coarser pitch gear offers an increase in strength of over 50 percent. The gears of the final design are not ground.

CRANKSHAFTS

The original crankshafts were made from a Homelite Model 270 crankshaft forging. These shafts performed satisfactorily during the initial tests, but began to fail during the longer endurance tests. These fatigue failures normally occurred on the forward crank arm, but failures of the rotary valve drive tang also occurred.

The crankshafts were redesigned to obtain the maximum strength available from the Homelite forging. The gain was minimal, but together with shotpeening the fatigue life of the crank was increased to between 35 and 50 hours. It was clear at this point that the forging would have to be abandoned and the crank made from a billet.

The billet cranks were designed to have a much thicker (30 percent) and wider (32 percent) forward crank arm. Also, a shorter, stronger rotary valve drive tang was designed. These design changes together with the shotpeening have resulted in a crankshaft that should perform reliably at power levels in excess of 30 horsepower for the design life of the engine.

IGNITION SYSTEM

The ignition system, as planned, was a subcontracted item. The initial system consisted of a magnet carried by the rotary valve, a pickup which sensed the magnet as it passed, a capacitive discharge ignition module, and a Homelite coil mounted on each spark plug. In retrospect, it appears that the overall design of the system was adequate. It created problems for the rotary valve, and the original configuration of the coil mounting was inadequate, but it is felt that these problems could have been overcome. At times the system ran flawlessly, but it was extremely unreliable due mainly to insufficient testing. One frequent problem was stray sparks. These seriously aggravated the early gear and rotary valve problems.

After several failures during Government evaluation, the original system was abandoned and a search for a new system initiated. Interim systems built by Kolbo Korp. and DH Enterprises were tested and found to perform adequately, but they were not well suited to the engine.

Finally, a high production motorcycle ignition system built by Gerex Inc. was located. Gerex modified their trigger system to sense the trailing edge of a rotating steel rotary valve and modified their CD ignition module to operate on 28 v. The spark plug mounted coils were replaced by conventional coils and shielded high tension leads. With the advent of the Gerex system, ignition failures and maloperation ceased. Further work to reduce EMI and weight is required.

NEEDLE BEARINGS

Torrington drawn cup needle bearings were initially chosen for the four main crankshaft bearings and the rear propeller shaft bearing. Drawn cup bearings have a thin outer shell and depend in great measure on their housing to retain their shape. These bearings would not work in the split aluminum housing of the BAT 282. The thin shell of the bearings cracked, binding the needles, which ceased to rotate smoothly.

To replace the drawn cup needle bearing, a thick cased needle roller bearing built by INA was found. The INA bearing has worked well. With the exception of the installation of bearing locating pins, no further bearing modifications were made.

THRUST WASHERS

In the original design the crankshaft thrust washers were made from phosphor bronze. This is a common thrust washer material. The material did not, however, meet the requirements of a geared engine. The axial loads imposed on the bronze washers by the imperfect action of the gears resulted in excessive wear and occasional fracture. The bronze washer was replaced by a readily available needle thrust bearing and companion steel thrust washer. No further problems occurred. This change not only improved reliability; it also resulted in a cost savings.

PISTONS

The only problem encountered with the piston was a brief episode of piston seizing. This was remedied by slightly reducing the outside diameter of the piston, thus increasing the piston cylinder diametral clearance from .004 to .006 in.

HOUSINGS

Changes were made in the housing as a result of modifications to other components. These changes included:

1. Crankcase widened to accommodate stronger crankshaft.

2. Gearbox modified to accept wider gears.

3. Induction port widened and shortened, giving larger area for performance and option of using larger diameter rotary valve hub.

4. Cooling fins added to alternator cavity to aid in removal of heat produced by 1250-watt alternator. A nonfinned version is still available.

5. Number of bosses for gear oil fill, drain and vent reduced from four to two to reduce weight.

6. Material added around bearing bores to better support bearings.

BACKPLATE

An all-phenolic backplate made from phenolic sheet and round bar for the carburetor risers was used during early testing. The phenolic was chosen mainly on the basis of weight. Problems with the glue joint between the risers and plate and charring of the surface in contact with the rotary valve necessitated a change.

It was decided at that time that a cast aluminum backplate would be the ultimate solution on the basis of cost, weight and performance. But time did not permit an immediate change. The interim solution was an all-aluminum backplate with welded carburetor riser. The welds failed after about 5 hours due to the low amplitude high frequency vibration of the engine, coupled with the cantilevered load of the carburetors. The fix entailed using a phenolic riser and securing both the riser and carburetor to the backplate with the two carburetor screws threaded into steel thread inserts in the backplate. An O-ring between phenolic and backplate provided the seal. No problems were encountered with this design. However, it was far more complicated than its function warranted.

The cast aluminum backplate seemed to be the best solution. It was designed and built and used on the last batch of engines built under the contract. No problems were experienced.

CARBURETORS

The design modifications and improvements made on the Walbro WB series carburetors were detailed in Engine Design.

ENGINE PERFORMANCE/TEST RESULTS

This section describes in some detail the major performance parameters of the BAT 282. The majority of the data presented is based on in-house dynamometer testing.

QUALITATIVE RESULTS

The most remarkable feature of the BAT 282 to those who are familiar with two-stroke engines is its exceptionally low vibration levels. The BAT 282 is observably smoother than a twin-cylinder engine is expected to be. This program did not include vibration tests, but it is expected that future vibration testing will substantiate the extremely low levels.

The idle characteristics of the engine are also uncommon for a two-stroke engine. The engine will idle smoothly at less than 1000 rpm. Most two-strokes, especially high performance two-strokes, will not idle properly below 3000 rpm. The low idle speed feature results from the improved combustion process provided by the multiple spark ignition system and the good mixture distribution provided by the separated crankcases. An additional benefit of the separated crankcases which will be experienced when flight tests are performed is that the mixture distribution to the cylinders is insensitive to maneuvers. One cylinder will have little tendency to run rich at the expense of the other, as often happens with a common crankcase twin.

Another important feature noted in testing is the crisp throttle response of the BAT. The engine accelerates rapidly and smoothly with no tendency to hesitate.

QUANTITATIVE RESULTS

The results presented in this section are based on dynamometer tests of the "final design" engines.

A comparison of wide open throttle (WOT) power versus crankshaft rpm for three different exhaust schemes is shown in Figure 26. The power has been corrected to standard conditions of 60°F and 29.92 in. Hg. With straight pipe exhausts, a maximum of 25.07 hp was obtained at a crankshaft speed of 8,000 rpm (7,556 rpm output speed). The U-tube tuned exhaust gave a maximum of 28.0 hp at a crank speed of 7,300 rpm. A modified U-tube (stinger shortened 1 in), gave a maximum of 28.56 hp at a crank speed of 8,000 rpm. In all these cases the maximum power greatly exceeds the program target of 20 hp. Only in the case of the modified U-tube was the target Brake Specific Fuel Consumption (BSFC) of 0.80 met at maximum power. But, these results do not give the whole picture on BFSC.

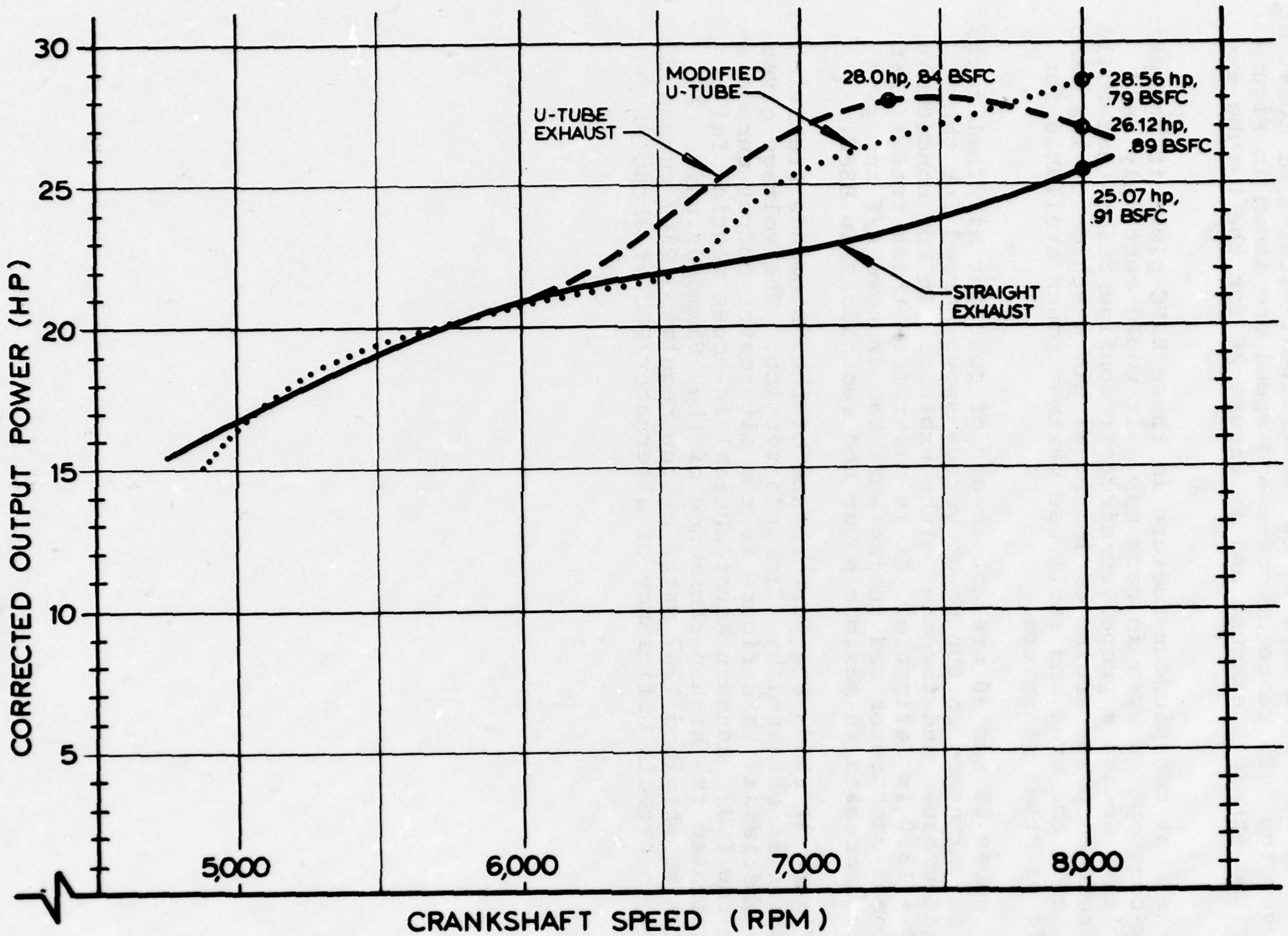


Figure 26. Comparison of WOT Corrected Power for Three Exhaust Schemes.

A large number of dynamometer runs made at partial throttle allowed complete BSFC maps to be constructed. Traces of constant BSFC over the full range of power and speed are shown in Figure 27 for the straight exhaust and in Figure 28 for the U-tube exhaust.

The most conspicuous feature in these BSFC plots is the pair of low-consumption eyes in each map. It would certainly be possible to design a propeller/aircraft configuration that would put a reduced power cruise or loiter within a region of low BSFC near one of the eyes and still use maximum power available for takeoff or other maneuvers.

Figures 29 and 30 present traces of constant air/fuel ratio (A/F) superimposed on the power versus speed curves for the straight exhaust and for the U-tube exhaust. In WOT conditions an A/F $\simeq$ 13.0 is desirable. It is clear from these traces that additional carburetor and tuning work can improve A/F and give further increases in maximum power and reductions in BSFC.

Figure 31 shows the performance of the Simmonds Precision alternator as obtained by Simmonds testing. The voltage given on the abscissa of this figure is the alternator output unregulated by the PCU. Bennett Aerotechnical in-house testing fully substantiated the high performance of the Simmonds unit. At 5000 rpm we obtained 1247 watts at the regulated 28-v output of PCU and an overall efficiency of alternator/PCU of about 80 percent.

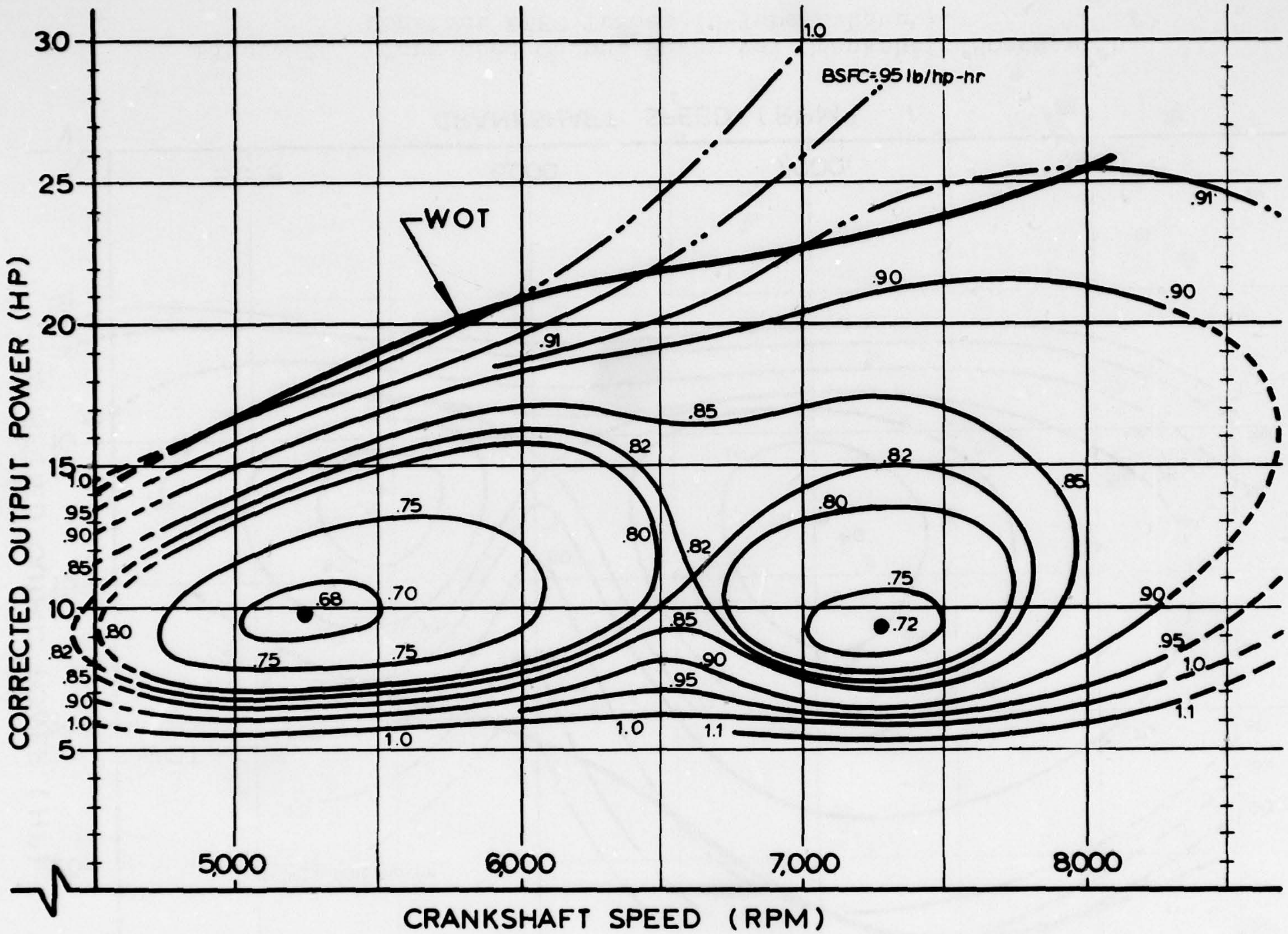


Figure 27. Corrected Output Power vs. Crankshaft Speed with Constant BSFC Traces (Straight Exhausts).

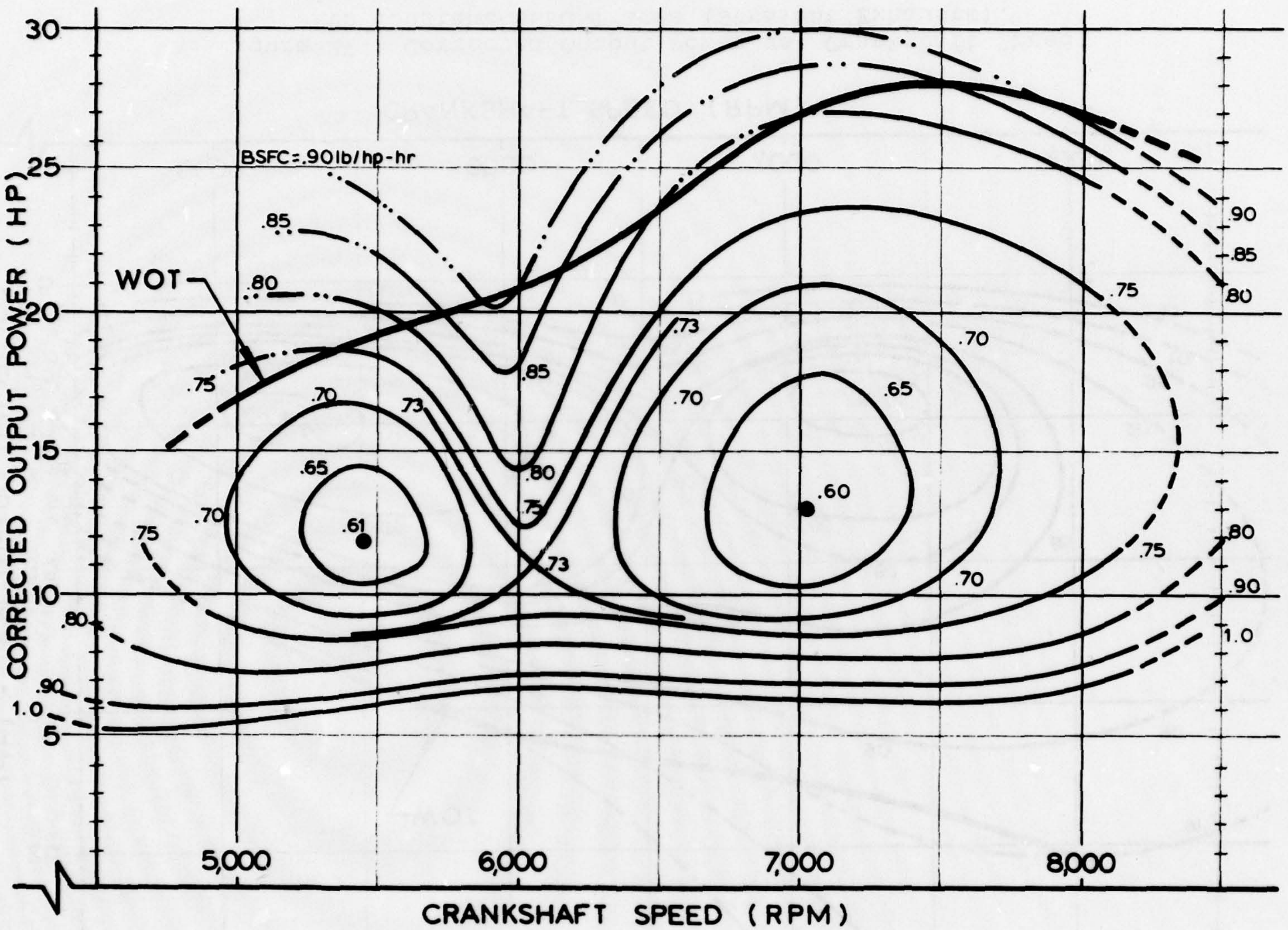


Figure 28. Corrected Output Power vs. Crankshaft Speed with Constant BSFC Traces (U-Tube Exhaust).

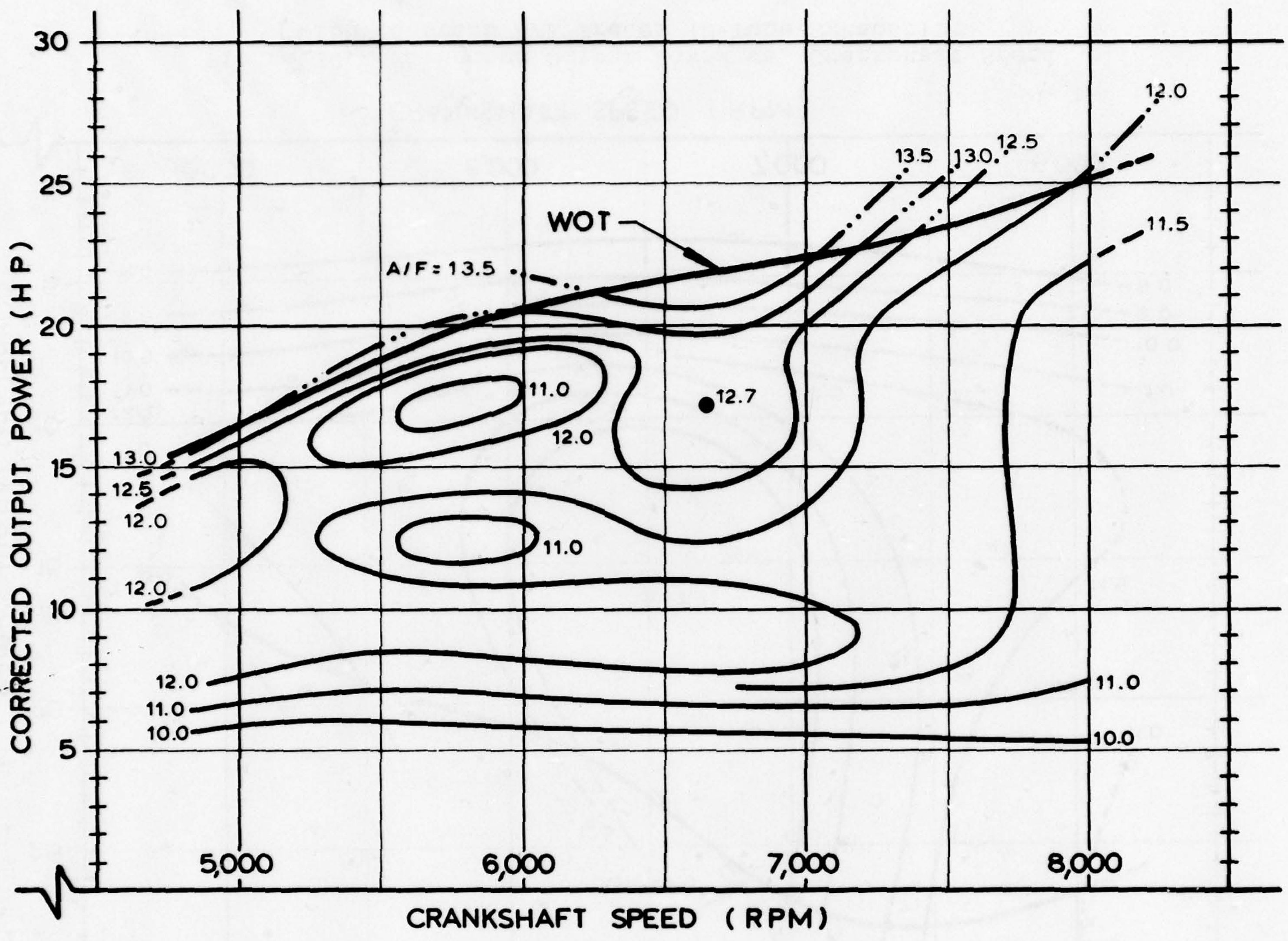


Figure 29. Corrected Output Power vs. Crankshaft Speed with Constant A/F Traces (Straight Exhausts).

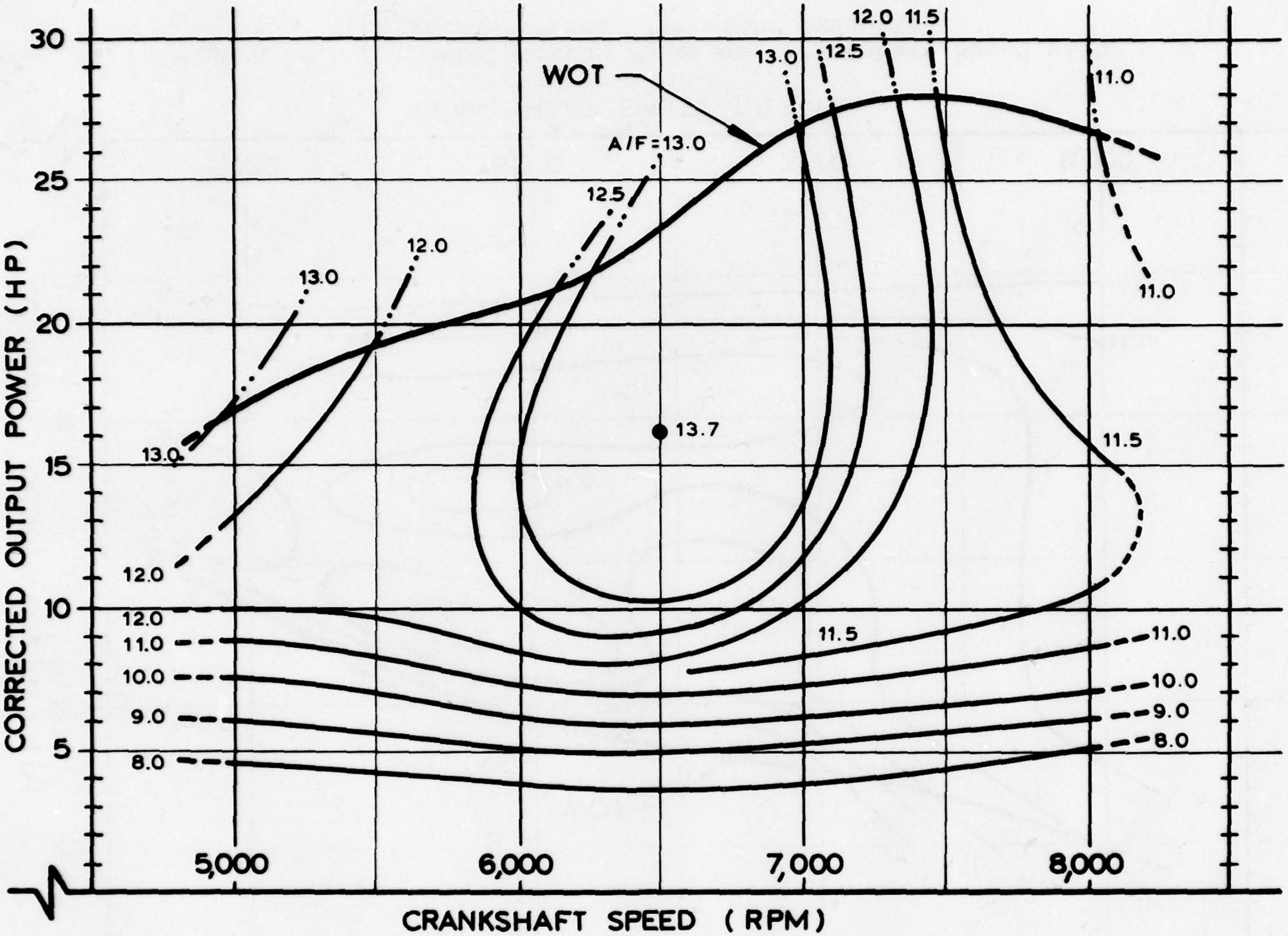


Figure 30. Corrected Output Power vs. Crankshaft Speed with Constant A/F Traces (U-Tube Exhaust).

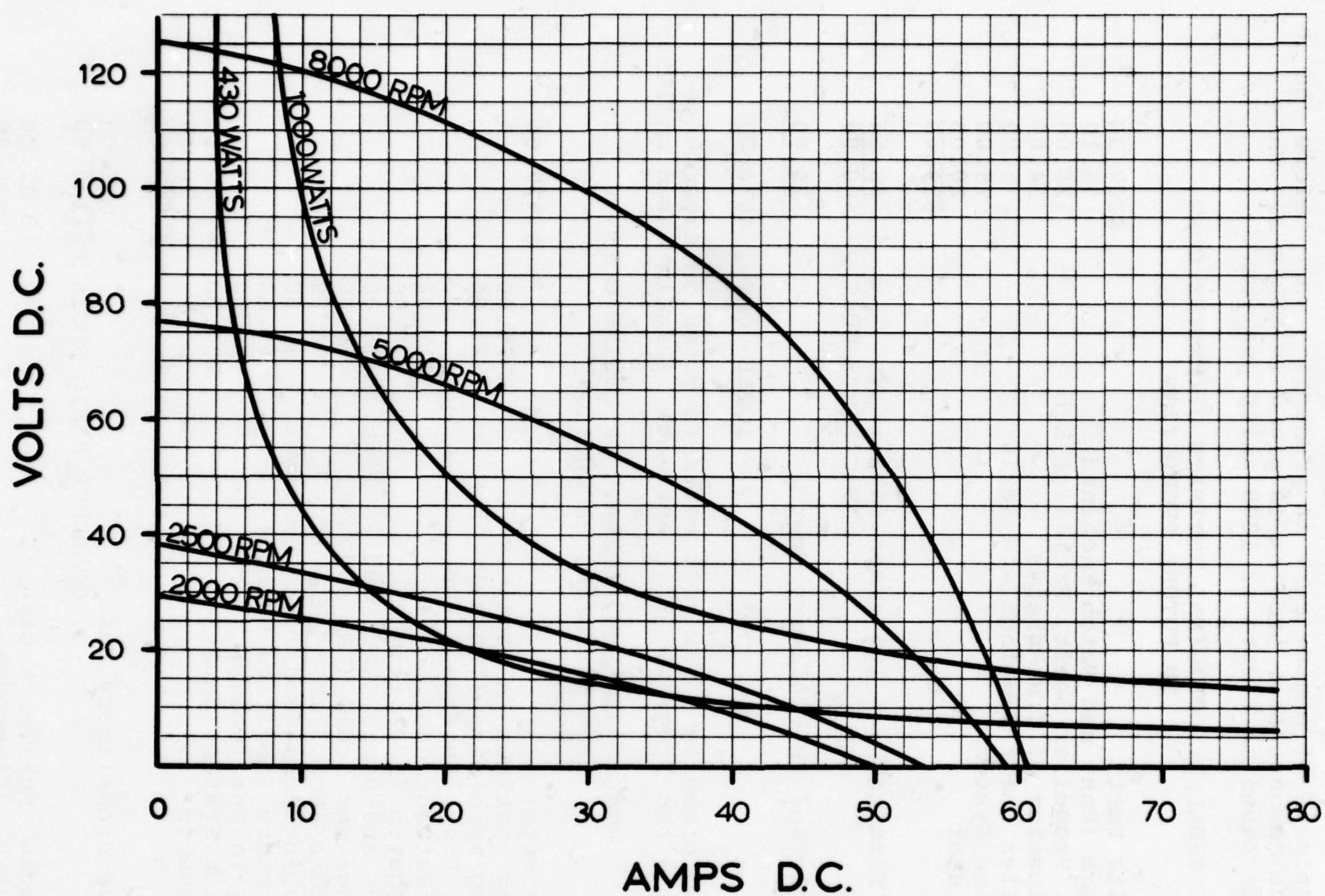


Figure 31. Alternator Performance.

COST IN PRODUCTION

The estimate of engine cost in production made before the start of the demonstrator program is given in Table 2. This estimate was based on a rate of 1000 units per year.

TABLE 2. ORIGINAL ESTIMATE OF ENGINE COST IN FUTURE PRODUCTION

Homelite Parts	$207.36
Housings (Cast and Match Machined)	55.10
Gears, Propeller Shaft, Propeller Hub	120.00
Modification of Crankshafts	54.00
Propeller Shaft Bearings and Seals	8.00
Ignition System	75.00
Carburetor	20.00
	539.46
Misc. Fittings + ≃ 10% error factor	60.54
BASIC PARTS COST	600.00
Assembly and Pre-Delivery Test	70.00
TOTAL COST	$670.00

Table 3 presents a new cost estimate based on experience gained during the program. This estimate is again based on 1000.

TABLE 3. NEW ESTIMATE OF ENGINE COST IN FUTURE PRODUCTION

Major Components:		$ 977.00
Housing castings (Alcast)	$ 75.00	
Housing machining (Intrax)	225.00	
Gears, Prop. Shaft and Hub (Estrada Gear)	115.00	
Crankshafts (Hydroflight)	50.00	
Pistons and Rings (Jade and Pacific)	18.00	
Bearings and Seals (INA and NOK)	26.00	
Cylinders (Homelite)	76.00	
Connecting Rods (modified Homelite)	30.00	
Carburetors (modified Walbro)	45.00	
Rotary Valves (Meigs)	36.00	
Ignition System (Gerex)	280.00	
Other Components		70.00
TOTAL PARTS COST		$1047.00
Assembly and Predelivery Test		125.00
TOTAL FOR BASIC ENGINE		$1172.00
Exhaust System (Mueller-Kautzer)		32.00
Alternator/PCU (Simmonds)		1815.00
TOTAL FOR ENGINE SYSTEM		$3019.00

units and all tooling costs have been absorbed in the costs given. The breakdown into cost elements is different in the two estimates because of changes in design and differences in use of available parts. The increase in basic parts cost estimate from $600 to $1047 is attributable to two items: The housing (+$246) and the ignition system (+$205). The increased housing cost is due simply to poor estimates given to us originally. The increase in ignition system cost is partly caused by application of Military Specification 641 for very low EMI. The increase in assembly and test cost is based on our experience during the demonstrator program. The exhaust system and the alternator/PCU were not included in the original cost estimate.

Table 4 gives a comparison of the original estimate and new estimate, both in 1979 dollars. An inflation factor of 1.207 was used to convert the original 1976 estimates to 1979 dollars.

TABLE 4. COMPARISON OF ORIGINAL AND NEW COST ESTIMATES IN 1979 DOLLARS

	Original Estimate	New Estimate
Total Parts Cost	$ 724.00	$1047.00
Assembly and Predelivery Test	85.00	125.00
Total for Basic Engine	$ 809.00	$1172.00

It is felt that some reductions in housing machining costs can be obtained by closer work with the supplier. It is not felt that a major reduction in ignition system cost is possible in view of the requirements of Military Specification 641.

CONCLUSIONS

1. The demonstrator program has resolved the question of gearing small two-stroke engines within the weight limitations set by RPV use. While the gears employed early in the program were inadequate, the final design resulted in gears that withstood extensive endurance testing and several hours of very hard running on the BAT dynamometer.

2. The theoretical analysis which showed the low vibration possible with a properly configured twin-crank design was substantiated insofar as the engine was observably smooth in operation. Vibration measurements were not included as part of the demonstrator program.

3. The program goal of maximum utilization of off-the-shelf parts is probably not attainable if RPV engine requirements are to be met. While standard cylinders and slightly modified connecting rods were successfully employed, new pistons and crankshafts were necessary. Considerable modification of standard carburetors was required to attain desired performance. These modifications are not complex and can be done in-house or by the carburetor manufacturer if volume warrants.

4. The demonstrator program has shown that the separate crankcase design with individual carburetors gives superior idle performance ($\simeq$ 800 rpm) and more nearly equal distribution to each of the cylinders than is obtainable with a single crankcase design. Adjustment of the individual carburetors to equalize cylinder performance was found to be very easy on the test stand and should present no problem in the field since fixed high-speed carburetor jets are used and only idle speed need be adjusted to match atmospheric conditions.

5. The demonstrator goal of a 24-lb engine without alternator/PCU and exhaust system was slightly exceeded. The final demonstrator engine weighs 24.5 lb but production design of housings and ignition system should reduce weight to 23 lb.

6. The demonstrator goal of 20 hp was greatly bettered. BAT dynamometer tests gave over 25 hp with simple straight exhausts.

7. The unique U-tube exhaust developed during the program gave significant increases in power and reduction in fuel consumption. Power as high as 28.56 hp with a BSFC of 0.79 were obtained. At reduced power levels, BSFC was

as low as 0.60. The exhaust system study was intentionally quite limited and further work is required to develop full potential of the system.

8. Experience during the program substantiated the view that specialty job shops can most economically produce the required parts at the low production rates of 1000 engines per year.

9. The wide, but flat, configuration was thought at the beginning of the program to be good for many RPV designs. Installation studies of the final design made in conjunction with the demonstrator program show that such is the case. An aerodynamically clean installation in a typical modern RPV is shown in Figure 32.

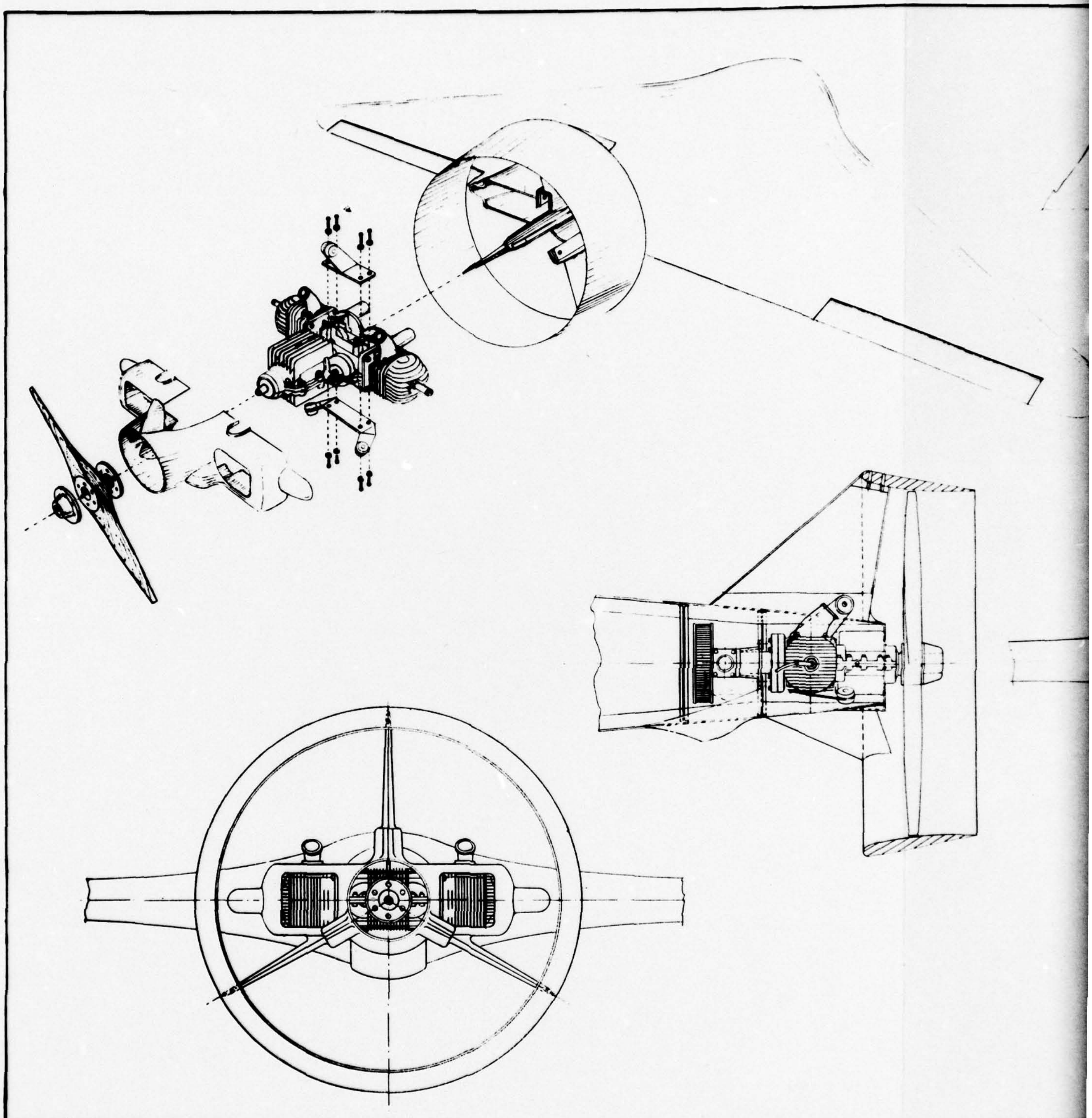
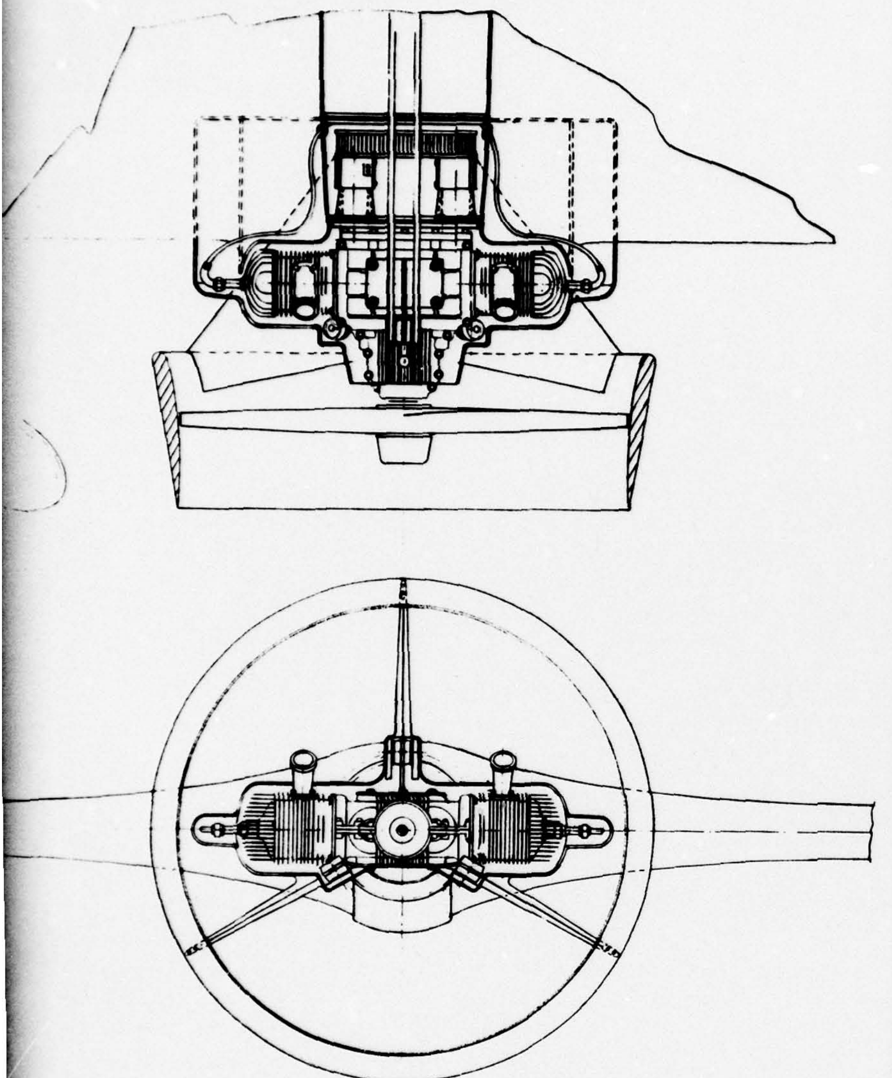


Figure 32. BAT 282 Installation in Typical RPV.



BENNETT AEROTECHNICAL

BAT 282 INSTALLATION
IN MINI-RPV AIRCRAFT

D.G. HOFFMAN 8 AUG 78

SCALE

SHEET 1 OF 1

RECOMMENDATIONS

Should the Government desire to complete engineering development of the BAT 282, particular attention needs be given to the following:

1. Repackage the ignition system to reduce weight and satisfy Military Specification 641 on EMI.

2. Modify design of housing castings and propeller shaft to reduce weight.

3. Carry out complete carburetor/rotary valve/spark timing development to reduce BSFC and increase maximum power. This development can be done with both the straight exhausts and the U-tube exhaust. Experience with other rotary valve engines, when compared with present test results, indicates that over 30 hp can be obtained with reasonable further development.

REFERENCES

1. G.E. Abercrombie and A.G. Bennett, Low Vibration 20 HP Mini-RPV Engine, SAE Technical Paper 780764, September 1978.

2. Stephen Timoshenko and D.H. Young, Advanced Dynamics, New York: McGraw-Hill, 1948, pp. 136-143.

3. R.W. Hovey, Ducted Fans for Light Aircraft, R.W. Hovey, Publisher, Second Edition, 1973, pp. 1-3, 19.

4. Gordon Jennings, Two-Stroke Tuner's Handbook, Tucson, Arizona: H.P. Books, 1973, pp. 51-73, 88-90.

5. K.G. Draper, The Two-Stroke Engine, Henley-on-Thames, Oxfordshire, England: G.T. Foulis & Co. Ltd., Fifth Edition, 1973, p. 46.

6. Cesare Bossaglia, Two-Stroke High Performance Engine Design and Tuning, Chislehurst, Kent, England: Lodgemark Press Ltd., 1972, pp. 76-84.

7. Joseph E. Shigley, Mechanical Engineering Design, New York: McGraw-Hill, Second Edition, 1972, pp. 495-507.

APPENDIX A

PARTS LIST AND WEIGHT SUMMARY

Group	Part	Source and Part No.	Qty	Unit Wt. (lb)	Total Wt. (lb)
I. Crankshaft	Crankshaft	BAT D-3037	2	1.422	2.844
	Thrust washer	Torrington TRA-1220	4	.006	.026
	Needle thrust bearing	Torrington NTA-1220	4	.009	.035
	Needle roller bearing	INA NK 19/20	4	.073	.291
	Shaft seal	NOK 72AY-188-R	2	.007	.015
	Woodruff key	No. 206	2	.000	.001
	Set screw	5/16-24 x 1/4	2	.003	.006
	Gear	BAT B-3007-A	2	.406	.812
	Flat washer	BAT A-3017	2	.029	.058
	Socket head cap screw	5/16-24 x 3/4	2	.024	.048
	Needle bearing	Homelite A-53197	2	.029	.058
	Connecting rod	Homelite A-53191	2	.200	.401
II. Propeller Shaft	Propeller shaft	BAT B-3006-A	1	1.359	1.359
	Socket head set screw	3/8-24 x 1/4	1	.005	.005
	Needle roller bearing	INA NK 19/20	1	.073	.073
	Alternator rotor	Simmonds Precision 49973	1	1.302	1.302
	Ball bearing	SKF 6204-NRJ	1	.231	.231
	Circular clip	BAT A-3013	1	.002	.002
	Shaft seal	NOK 72AY-189-R	1	.040	.040
	Propeller flange	BAT A-3040	1	.920	.920
	Flat washer	BAT A-3028	1	.036	.036
	Socket head cap screw	3/8-24 x 1	1	.045	.045

Group		Part	Source and Part No.	Qty	Unit Wt. (lb)	Total Wt. (lb)
III.	Housing	Machined engine housing	BAT D-3039	1	5.625	5.625
		Bearing locating pins	INA NR 2.5 x 9.8	5	.003	.015
		Cylinder studs	5/16-18, 5/16-24 x 1 1/4	8	.021	.169
		Heat conducting sheet	Bergquist Sil Pad 400	1	.010	.010
		Alternator stator	Simmonds Precision 49974	1	1.271	1.271
		Oil plug	1/8 NPT	1	.003	.003
		Vent plug	1/8 NPT	1	.003	.003
		Flat washer	McCulloch 105976Z	21	.001	.026
		Socket head cap screw	10-24 x 1	21	.009	.183
IV.	Piston	Piston	BAT D-3014	2	.321	.642
		Piston pin	Homelite A-54727	2	.084	.168
		Piston guide washer	BAT B-3003	4	.008	.032
		Pin retaining rings	Homelite 25502	4	.001	.004
		Piston rings	Pacific Piston Ring 11434	4	.005	.020
V.	Cylinder	Cylinder	Homelite A-54715	2	2.547	5.094
		Cylinder gasket	Homelite 74139	2	.011	.022
		Lock washer	5/16	8	.004	.031
		Hex nut	5/16-24	8	.010	.082

Group	Part	Source and Part No.	Qty	Unit Wt. (lb)	Total Wt. (lb)
VI. Backplate/Rotary Valve	Rotary valve	BAT A-3035	2	.037	.074
	Rotary valve hub	BAT A-3025	2	.049	.099
	Socket head cap screw	6-40 x 3/8	12	.002	.028
	Backplate gasket	BAT A-3051	1	.018	.018
	Backplate	BAT D-3050	1	.733	.733
	Flat washer	#8	6	.001	.004
	Lock washer	#8	3	.001	.002
	Socket head cap screw	8-32 x 3/4	10	.005	.051
	Carburetor gasket	BAT A-3052	2	.002	.004
	Carburetor	Walbro WB series BAT modification	2	.468	.937
	Flat washer	#10	4	.002	.007
	Socket head cap screw	10-24 x 2	4	.016	.065
	Timing window	BAT A-3054	1	.013	.013
	Ignition sensor hold down	BAT A-3053	1	.004	.004
VII. Other	Power conditioning unit	Simmonds Precision 49831	1	5.672	5.672
	Ignition system	Gerex Type 1CB	1	2.906	2.906
	Exhaust gasket	Homelite 50355-A	2	.020	.040
	Exhaust flange	BAT C-3032	2	.249	.499
	Socket head cap screw	1/4-20 x 3/4	8	.013	.106
	Exhaust elbow	BAT A-3036	2	.184	.367
	U-Tube exhaust	BAT B-3032	1	2.438	2.438
	Spark plug	Champion RMJ-3	2	.155	.310

APPENDIX B

DETAILS OF U-TUBE DESIGN

The basic design of the U-tube exhaust system is that of John Brooks, a consultant on the program. The proportions of the system are based on the empirical work of Jennings. The basic principles of the U-tube were presented in Engine Design.

The parameters to be considered are shown in Figure B-1. According to Jennings, the tuned length for the system is given by

$$L_t = \frac{E_o V_s}{N} \qquad \text{(B-1)}$$

where L_t = tuned length (inches)
E_o = exhaust open period = 167.2°
V_s = average wave speed = 1675 ft/sec
N = crankshaft speed = 7500 rpm

Using the values above in Eq. B-1 results in a tuned length of 35.01 in. This length is measured from the piston face in the exhaust port to the point of mean reflection in a standard expansion chamber. Here it is used as the half length of the circuit since one cylinder "stuffs" the other and does not rely on its own reflected wave for "stuffing".

Lead-in Pipe Proportions

The lead-in pipe area should be about 10 to 15 percent greater than the exhaust port area. In the BAT 282 the exhaust port area is .935 in.^{2}. This results in a lead-in pipe area of (using 11 percent) 1.04 in.^{2} and a pipe diameter of D_1 = 1.15 in. The lead-in pipe length should be between 6 to 11 times pipe diameter, the longer lead-in giving a broader power curve. A value of 9 was chosen giving a lead-in length of L_1 = 10.53 in.

Diffuser Proportions

A diffuser divergence angle of α_1 = 3.75° was chosen to give wide range performance. The outlet area should be about 6.25 times greater than the inlet area. Thus

$$D_2 = \sqrt{D_1^2 \, (6.25)} \qquad \text{(B-2)}$$

$$D_2 = 2.88 \text{ in.}$$

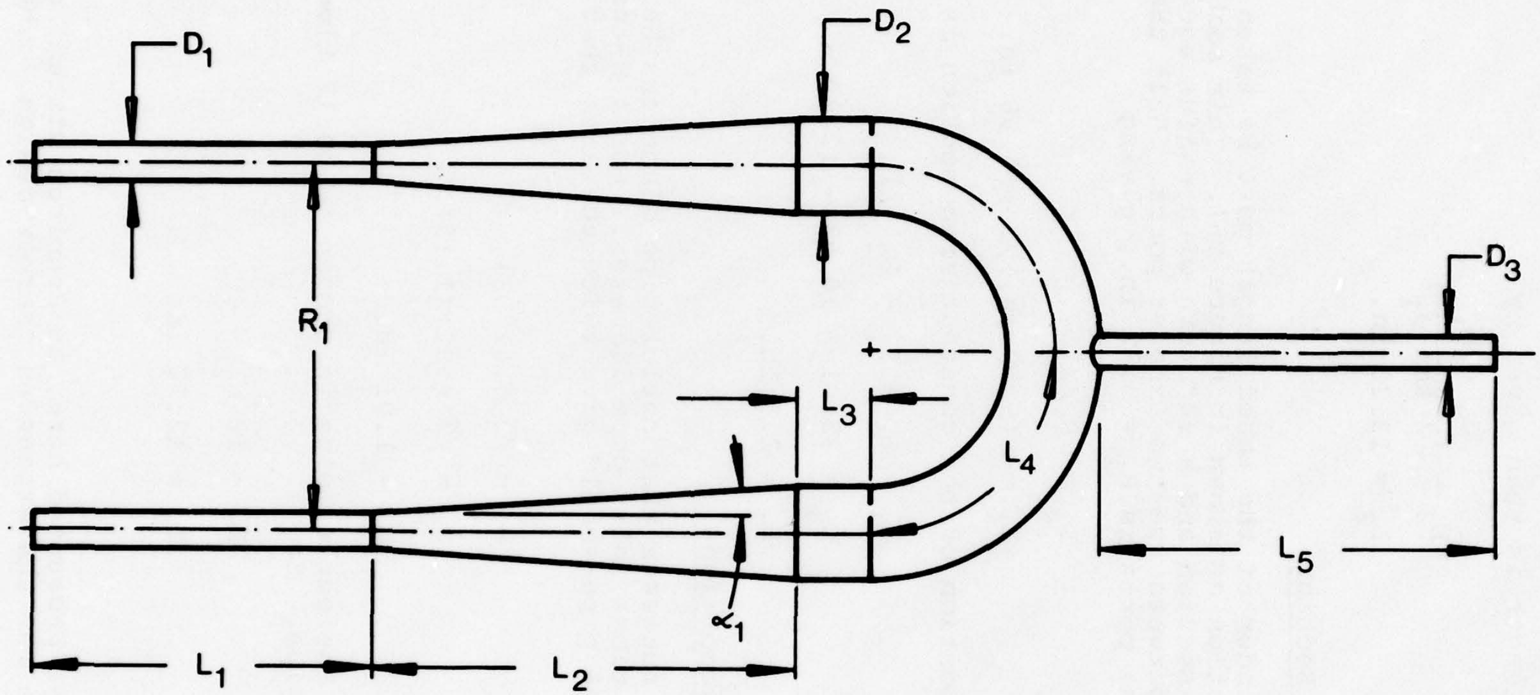


Figure B-1. U-Tube Exhaust System Parameters.

The diffuser length is then given by

$$L_2 = \left(\frac{D_2 - D_1}{2 \tan \alpha_1}\right) \quad \text{(B-3)}$$

$$L_2 = 13.20 \text{ in.}$$

Constant Area Section

The remainder of the tuned length must be taken up by a constant area section as shown in Figure B-1. This section consists of a straight section and a 90° turn with radius equal to half the distance between the two exhaust ports. Half the distance between exhaust ports is $R_1 = 5.72$ in., giving

$$L_4 = \pi R_1/2 \quad \text{(B-4)}$$

$$L_4 = \pi (5.72 \text{ in.})/2 = 8.98 \text{ in.}$$

The straight portion of the constant area section is given by

$$L_3 = L_t - L_1 - L_2 - L_4 \quad \text{(B-5)}$$

$$L_3 = 35.01 - 10.53 - 13.20 - 8.98 \text{ in}$$

$$L_3 = 2.3 \text{ in.}$$

Outlet Pipe Proportions

Jennings suggests that outlet pipe diameters be made equal to .58 to .62 times inlet pipe diameter. For a U-tube with only one outlet, the pipe should give twice the area that Jennings recommends.

$$D_3 = \sqrt{2}\ (.62\ D_1) \quad \text{(B-6)}$$

$$D_3 = \sqrt{2}\ (.62)(1.15)$$

$$D_3 = 1.01 \text{ in.}$$

The length of the outlet pipe is chosen to be 12 times the diameter of the pipe. So

$$L_5 = 12\ D_3 \quad \text{(B-7)}$$

$$L_5 = 12.12 \text{ in.}$$

Comments

The recipe presented here, as Jennings stated, gave a good first estimate for the expansion chamber design. The design was modified and refinements made as a result of testing. Also, the exact values for diameters and lengths which resulted from the recipe were not always used. Standard tubing sizes were always used, usually the next larger size.

APPENDIX C

GEAR STRENGTH ANALYSIS

This gearing analysis is based on the modified Lewis formulation as presented by Shigley.[7] This formulation considers the gear teeth to act much like a cantilever beam in bending. The modified Lewis equation is

$$\sigma = \frac{W_t P}{K_v F J} \tag{C-1}$$

where σ = bending stress (psi)

W_t = transmitted load (lb)

P = diametral pitch (in.$^{-1}$)

K_v = velocity factor

J = geometry factor including stress concentration

F = face width (in.)

This equation will be evaluated for the BAT 282 gears having the following properties and under the following conditions

P = 10.16 in.$^{-1}$
F = .71 in.
d = pitch diameter smallest (weakest) gear = 1.73 in.
r_f = root filet radius = .03 in.
N - number of teeth = 17
t - tooth thickness at base of tooth = .19 in.
ℓ - working depth of tooth = .20 in.
H = engine power per side = 12.5 hp
n - engine speed = 8,000 rpm

Transmitted Load

The mean transmitted load in terms of horsepower speed and pitch diameter is given by

$$W_t = \frac{2(12 \text{ in./ft})(33{,}000 \text{ ft-lb/min/hp})}{2\pi \text{ rad/rev}} \frac{H}{nd} \tag{C-2}$$

$$W_t = \frac{2(12 \text{ in./ft})(33{,}000 \text{ ft-lb/min/hp})\ 12.5 \text{ hp}}{(2\pi \text{ rad/rev})(8{,}000 \text{ rev/min})\ (1.73 \text{ in.})}$$

$$W_t = 113.8 \text{ lb}$$

---

[7]Joseph E. Shigley, Mechanical Engineering Design, New York: McGraw-Hill, Second Edition, 1972, pp. 495-507.

Velocity Factor

The velocity factor accounts for the increase in tooth load due to the rotational velocity of the gear and the extent of dynamic loading. The pitch line velocity is given by

$$V = \frac{2\pi\,(\text{rad/rev})}{12\ \text{in./ft}}\ \frac{d}{2}\ n \qquad \text{(C-3)}$$

$$V = \frac{2\pi\,(\text{rad/rev})}{12\ \text{in./ft}}\ \frac{(1.73\ \text{in.})}{2}\ (8{,}000\ \text{rev/min})$$

$$V = 3623\ \text{ft/min}$$

For precision gears under high dynamic loading, the velocity factor is given by

$$K_v = \sqrt{\frac{78}{78 + \sqrt{V}}} \qquad \text{(C-4)}$$

$$K_v = \sqrt{\frac{78}{78 + \sqrt{3623}}}$$

$$K_v = .751$$

Geometry Factor

The geometry factor accounts for the effects of the shape of the tooth, the position on the tooth at which the critical load is applied and stress concentration. The stress concentration factor for 20° pressure angle spur gears is given by

$$K_t = 0.18 + \left(\frac{t}{r_f}\right)^{0.15} \left(\frac{t}{\ell}\right)^{0.45} \qquad \text{(C-5)}$$

$$K_t = 0.18 + \left(\frac{.19}{.03}\right)^{0.15} \left(\frac{.19}{.20}\right)^{0.45}$$

$$K_t = 1.47$$

The factor is modified to account for notch sensitivity by

$$K_f = 1 + q(K_t - 1) \qquad \text{(C-6)}$$

where K_f is the composite factor and

q is the notch sensitivity factor = .90.

$$K_f = 1 + .90\ (1.47 - 1)$$

$$K_f = 1.42$$

The Lewis form factor, Y, is the other factor which enters into the geometry factor. The value of Y for a 20° full depth gear with 17 teeth is

$$Y = .303$$

The geometry factor may now be written

$$J = \frac{Y}{K_f} \qquad \text{(C-7)}$$

$$J = \frac{.303}{1.42}$$

$$J = .213$$

Bending Stress

The modified Lewis equation (C-1) may now be entered to compute the mean bending stress of the tooth.

$$\sigma = \frac{W_t\ P}{K_v FJ}$$

$$\sigma = \frac{(113.8 \text{ lb})(10.16 \text{ in.}^{-1})}{.751\ (.71 \text{ in.})(.213)}$$

$$\sigma = 10{,}180 \text{ lb/in.}^2$$

Strength of Gear Teeth

To determine the strength of the gear several additional factors must be included. These are surface finish, size, reliability, temperature, and miscellaneous effects. The endurance limit of the gear is then

$$S_e = k_a k_b k_c k_d k_e S_u \qquad \text{(C-8)}$$

where
- k_a = surface finish factor
- k_b = size factor
- k_c = reliability factor
- k_d = temperature factor
- k_e = miscellaneous factor
- S_u = ultimate strength of material (psi)

Following the guidelines of Reference 7,

k_a = .64 for shaved and ground gears,
k_b = 1.0 for diametral pitch greater than 5,
k_c = .737 for a desired reliability of 99.9 percent,
k_d = .838 for an assumed maximum temperature of 280°F,
and k_e = 1.0 for teeth stressed in both directions.

The ultimate strength of SAE 9310 steel is approximately 180,000 psi. The endurance limit for the gear is then

$$S_e = (.64)(1.0)(.737)(.838)(1.0)(180{,}000 \text{ psi})$$

$$S_e = 71{,}148 \text{ psi}$$

Safety Factor

This results in a safety factor

$$k_s = S_e/\sigma$$

$$k_s = 71{,}148 \text{ psi}/10{,}180 \text{ psi}$$

$$k_s = 7.0$$

From Reference 7 the desired safety factor for gearing with high shock loadings is 4.8. This desired safety factor accounts for the fact that the peak loads are several times larger than the main loads, the mounting is not completely rigid (aluminum housing), and uses a desired strength-over-stress ratio of 2.

APPENDIX D

ENGINE ASSEMBLY PROCEDURES

Assembly of the BAT 282 is not a difficult task; however, several precautions must be taken to insure proper operation. All components must be carefully cleaned and the work area must be kept clean and free of foreign matter which might damage the engine. The proper tools should also be used for each assembly procedure. A list of the tools required is given in Table D-1. Additional materials such as assembly lubricant are also needed. They are given in Table D-2.

The engine is most easily assembled if the components are divided into six groups which may be assembled somewhat independently.

As a frame of reference, the propeller is front, the carburetors rear, the screw heads in the engine housing top and the threaded holes bottom.

TABLE D-1. TOOLS REQUIRED FOR ASSEMBLY OF BAT 282

Torque wrench(es)	20 in.-lb to 75 ft-lb
Hex key with adapter to fit torque wrench	9/64 in. 5/32 in. 3/16 in. 1/4 in. 5/16 in.
Vise with soft jaws	4-in. throat (minimum)
Adjustable wrench	8-in. length
Arbor press	8-in. throat (minimum)
Steel sleeve	20 mm I.D., 2 mm minimum wall, 50 mm length
Steel ring	20 mm I.D. x 45 mm O.D. x 2 mm thick
Feeler gauge	.001 in. to .030 in.
Open-end wrench	7/16 in. 1/2 in.
Snap ring pliers	KD #444 or comparable
Socket wrench	13/16 in. spark plug type

TABLE D-2. MATERIAL REQUIRED FOR ASSEMBLY OF BAT 282

Engine pre-lube	Silicone grease or comparable
Cleaning fluid	Alcohol or equivalent de-greaser
Loctite	#35 max strength or equivalent
Silicone RTV Form-A-Gasket	Dow Corning #730 preferred GE MR 10 or comparable acceptable
Gear oil	Almasol #607

CRANKSHAFT GROUP (2 PER ENGINE)

Parts Required:

1 crankshaft, BAT D-3037
2 thrust washers, Torrington TRA 1220
2 needle thrust bearings, Torrington NTA 1220
2 needle roller bearings, INA NK 19/20
1 shaft seal, NOK 72AY-188-R
1 Woodruff key, No. 206
1 crank gear, BAT B-3007-A
1 flat washer, BAT A-3017
1 socket head cap screw, 5/16-24 x 3/4 in.
1 connecting rod, Homelite A-53191
1 needle bearing, Homelite A-53197

Procedure:

1. Thoroughly lubricate thrust washer, thrust bearing, needle roller bearing, and seal with assembly lube and install on tapered end of shaft as shown in Figure D-1.

2. Clean taper with alcohol to remove any lubricant which may have rubbed off.

3. Install Woodruff key in key slot on shaft.

4. Slide gear onto taper making sure that Woodruff key fits into keyway in gear.

5. Secure gear using flat washer and socket head cap screw. Apply small amount of Loctite to end of screw.

6. Mount crankshaft securely in a vise as shown in Figure D-2.

7. Using torque wrench with 1/4-in. hex key, tighten screw to a torque of 45 ft-lb.

8. Wrap waxed needle bearing strip around crank pin. Wax should hold needles in place.

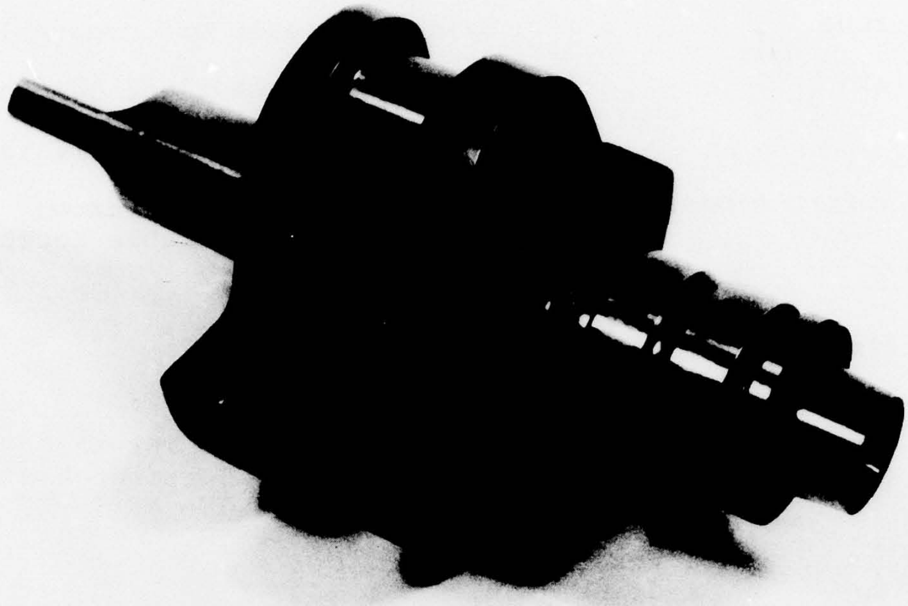


Figure D-1. Installation of Bearings, Washer, and Seal.

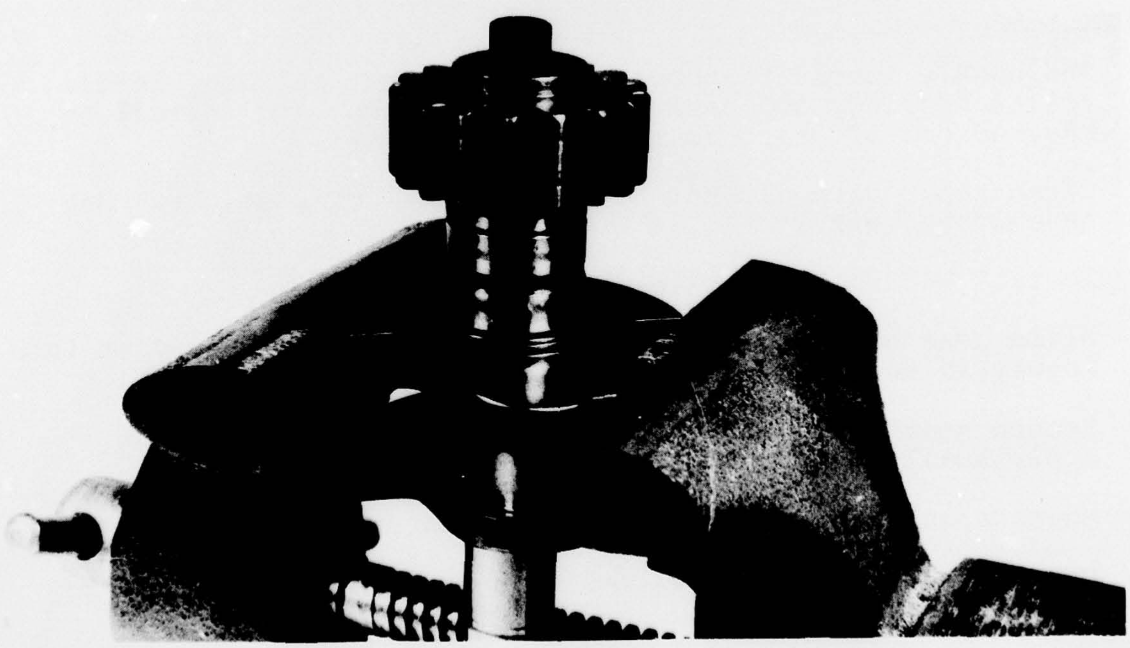


Figure D-2. Vise Mounting of Crankshaft.

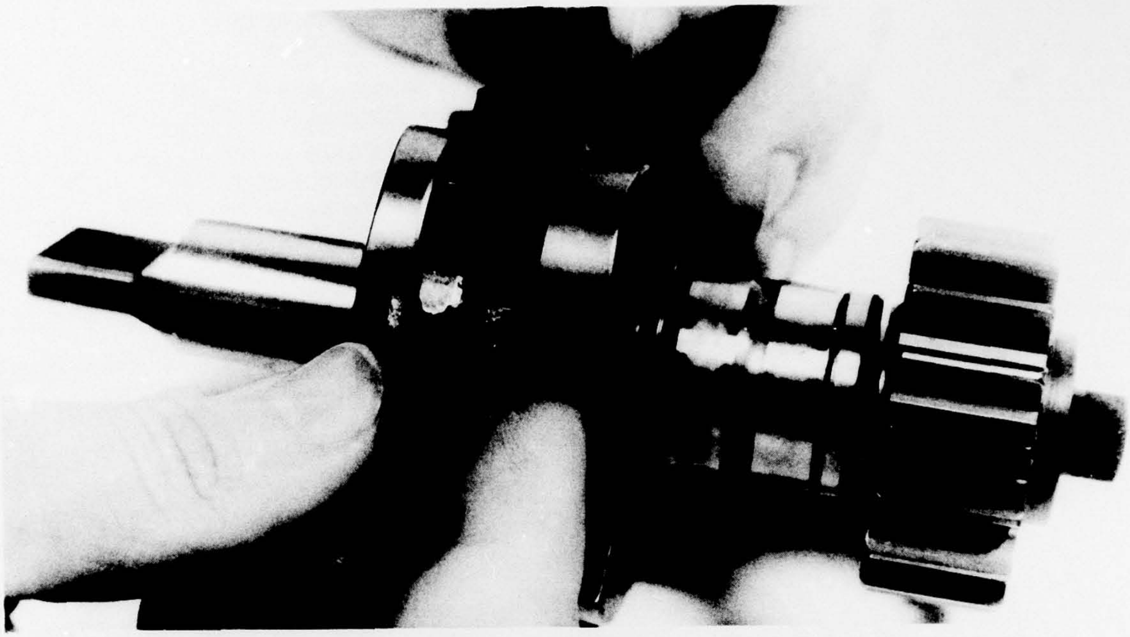


Figure D-3. Installation of Connecting Rod.

9. Install connecting rod as shown in Figure D-3.

10. Using a torque wrench, with a 5/32-in. hex key, tighten rod screws finger-tight.

11. Check closely to insure that fractured rod mates properly. The fracture should be almost undetectable.

12. Increase torque on rod screws to 110 in.-lb by alternately increasing torque on each screw in 20 in.-lb increments. Hold rod, not crank, with an adjustable wrench as shown in Figure D-4.

13. Lubricate thrust washer, thrust bearing and needle roller bearing and install on shaft as shown in Figure D-5.

PROPELLER SHAFT GROUP (1 PER ENGINE)

Parts Required:
1 propeller shaft, BAT B-3006-A
1 needle roller bearing, INA NK 19/20
1 alternator rotor, Simmonds 49973
1 ball bearing, SKF 6204 NRJ
1 circular clip, BAT A-3013
1 shaft seal, NOK 72AY-189-R

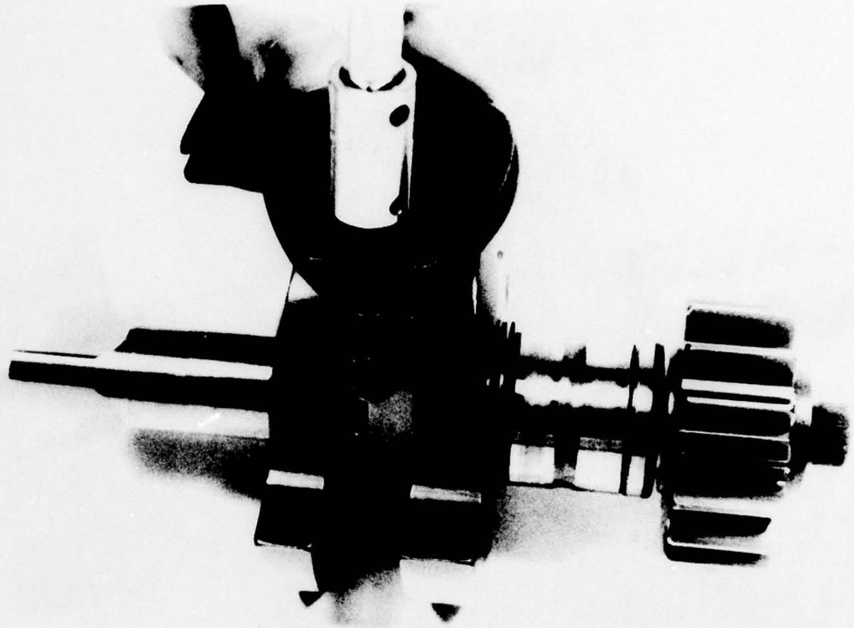


Figure D-4. Torquing of Rod Screws.

Figure D-5. Installation of Bearings and Washer.

1 propeller flange, BAT D-3040
1 flat washer, BAT A-3028
1 socket head cap screw, 3/8-24x1

Procedure:

1. Place alternator rotor on shaft so that chamfered edge of rotor bore faces gear.

2. Use an arbor press to press the shaft into the rotor. Press the shaft until rotor comes into contact with the shoulder. Lubricant should be used.

3. Use the arbor press, the 20 mm ring and sleeve to press the ball bearing into position against the shaft shoulder. The ball bearing should be oriented with the snap ring toward the rear.

4. Using the 20 mm ring, install the circular clip in the ring groove adjacent to the ball bearing. Be sure that circular clip fits securely in the groove. Take care that the seal surface adjacent to the ring groove is not scratched or otherwise damaged.

5. Lubricate the shaft seal and install in contact with the bearing, metal side forward.

6. Clean taper with alcohol to remove any lubricant which may have rubbed off.

7. Install propeller flange, flat washer, and socket head cap screw loosely. Final tightening is performed after propeller is installed.

8. Lubricate and install needle roller bearing on rear of shaft.

HOUSING GROUP (1 PER ENGINE)

Parts Required:
1 machined engine housing, BAT D-3039
2 crankshaft groups
1 propeller shaft group
2 heat conducting sheets, Berqquist Sil Pad 400
1 alternator stator, Simmonds 49974
1 oil plug, 1/8 NPT
1 vent plug, 1/8 NPT
21 flat washers, McCulloch 105976 Z
21 socket head cap screws, 10-24 x 1

Procedure:

1. Orient bottom half of housing, the half with bearing locating pins, so that interior of housing is up.

2. Install the two crankshaft groups and locate the needle roller bearings so that the small oil holes fit over the locating pins in the housings.

3. With a feeler gauge, check the clearance between the forward thrust washer and crankshaft washer face. This clearance should be between .005 and .012 in. If the clearance is not within these limits, remove the rear needle roller bearing and switch it end for end. The oil holes are not perfectly central.

4. When a suitable clearance is achieved, remove the crankshaft groups and apply a fine bead of Loctite in the bearing bores about 1/8 in. on either side of the locating pins. This includes both propeller shaft bearing bores. Apply Loctite forward of snap ring in nose bearing bore and in ring groove.

5. Reinstall the crankshaft groups.

6. Locate heat conducting sheet in alternator cavity to act as heat conductor from stator to housing.

7. Carefully slip alternator stator over gear end of propeller shaft and center stator in rotor. Stator lead should be toward rear.

8. Set propeller shaft in place, making sure that the snap ring fits securely in the snap ring groove.

9. Orient propeller shaft and crankshafts for proper timing. This is tested by holding the shafts securely in the bores and rotating either crankshaft so that the rotary valve drive tang is horizontal. The other tang should then also be horizontal and one crankpin should be up at about 45° and the other down at about 45°. See Figure D-6.

10. Holding the propeller shaft securely, adjust the placement of the stator so that the electrical lead and bushing fit into place without binding as shown in Figure D-7. Care should be taken not to cut the O-ring on the bushing or the spares.

11. Apply a liberal bead of RTV to housing mating surfaces. This material will not harm internal parts.

12. Check to see that bearings are located on pins and that heat conducting pads are in place. Make adjustments as required.

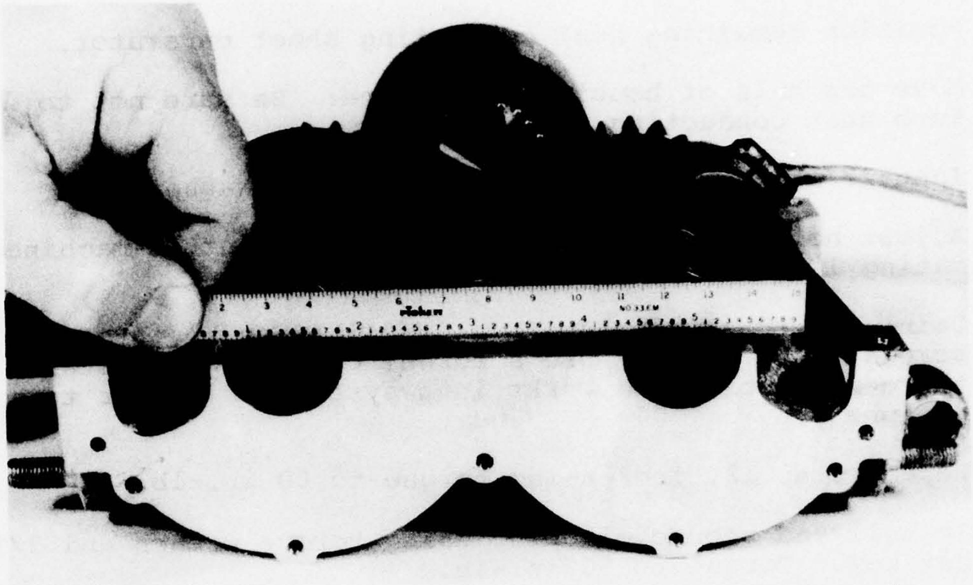


Figure D-6. Crankshaft Timing Check.

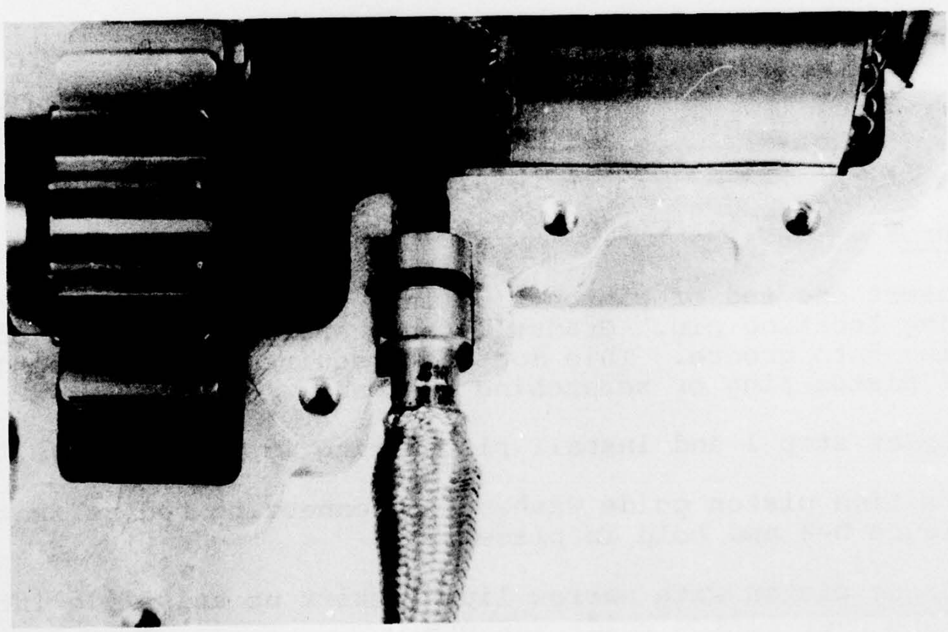


Figure D-7. Alternator Placement.

13. Apply a thin bead of Loctite in the bores of the top housing half as in Step 4.

14. Position remaining heat conducting sheet on stator.

15. Move top half of housing into place. Be sure not to disturb heat conducting sheet.

16. Install socket head cap screws and flat washers.

17. Adjust housing so that no steps are evident at machined mating lines.

18. Using a torque wrench with a 5/32-in. hex key, tighten all screws to 40 in-lb. Use a torque pattern which starts with the center screw and works in a systematic manner toward the periphery.

19. Repeat step 18, increasing torque to 60 in.-lb.

20. Install and secure oil plug using torque wrench and 3/16 in. hex key. Torque to 60 in.-lb.

21. Add 100 ml of Almasol 607 gear oil to gearbox.

22. Install and secure vent plug using 7/16-in. open-end wrench.

PISTON GROUP (2 PER ENGINE)

Parts Required:

1 housing group
1 piston, BAT B-3014
2 piston rings, Pacific Piston Ring 11434
1 piston pin, Homelite A-54727
2 piston guide washers, BAT B-3003
2 pin retaining rings, Homelite 25502

Procedure:

1. Insert one end of piston ring into lower groove adjacent to ring locating pin. Gradually and gently work entire piston ring into groove. This does not require excessive bending of piston ring or scratching piston.

2. Repeat step 1 and install piston ring in upper groove.

3. Position piston guide washers on connecting rod as shown in Figure D-8 and hold in place.

4. Orient piston with narrow lip of skirt up and slide into place over piston guide washers.

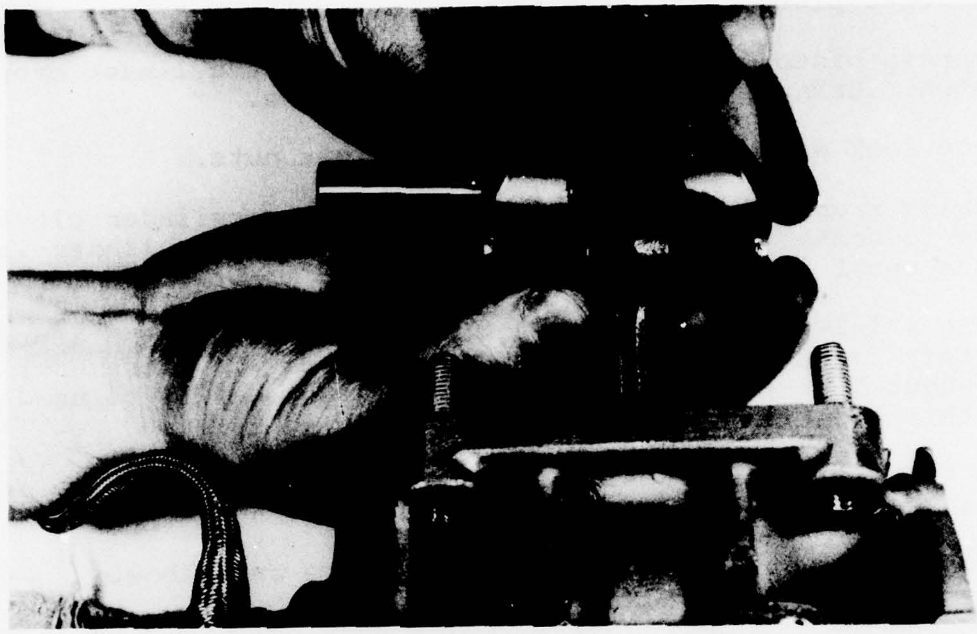


Figure D-8. Piston/Guide Washer Assembly.

5. Lubricate and insert piston pin. Orientation of closed end is not important.

6. Using snap ring pliers, insert pin retaining rings and rotate retaining ring until gap is down, away from piston crown.

7. Lubricate guide washers and upper rod bearing.

CYLINDER/EXHAUST GROUP (2 PER ENGINE)

Parts Required:

1 housing group
1 cylinder, Homelite A-54715
1 cylinder gasket, Homelite 74139
4 lock washers, 5/16
4 hex nuts, 5/16-24
1 exhaust gasket, Homelite 50355-A
1 exhaust flange and elbow, BAT C-3032/A-3036
4 socket head cap screws, 1/4-20 x 3/4
1 spark plug, Champion RMJ-3

Procedure:

1. Install cylinder gasket.
2. Lubricate piston and piston rings thoroughly.
3. Compress piston rings with fingers and slide cylinder over piston. Exhaust port should be facing down.
4. Place lock washers on studs and start hex nuts.
5. Viewing from the spark plug end, rotate the cylinder clockwise to center cylinder on cylinder pad. Hold cylinder in place until screws are snug.
6. Using a 1/2-in. open-end wrench, tighten hex nuts to approximately 15 ft-lb. Torque nuts uniformly in increments of about 5 ft-lb. (A torque wrench cannot easily be used in this location.)
7. Now turn the engine upside down for installation of the exhaust.
8. Install exhaust gasket and exhaust flange and elbow.
9. Insert socket head cap screws and torque uniformly to 10 ft-lb. Apply a small amount of Loctite to end of screws.
10. Set spark plug gap to .060 in.
11. Install spark plug and torque to 150 in.-lb.

BACK PLATE/ROTARY VALVE GROUP (1 PER ENGINE)

Parts Required:

1 housing group
1 backplate, BAT D-3050
1 backplate gasket, BAT A-3051
2 rotary valves, BAT A-3035/A-3025
6 flat washers, #8
3 lock washers, #8
10 socket head cap screws, 8-32 x 3/4
2 carburetors, Walbro WB Series
2 carburetor gaskets, BAT A-3052
4 socket head cap screws, 10-24 x 2
1 ignition sensor, Gerex
1 ignition sensor hold-down, BAT A-3053

Procedure:

1. Position carburetor gaskets and carburetors as shown in Figure D-9. Be sure that carburetor gasket does not cover pulse passage in backplate, and that pulse passages in carburetor and backplate match.

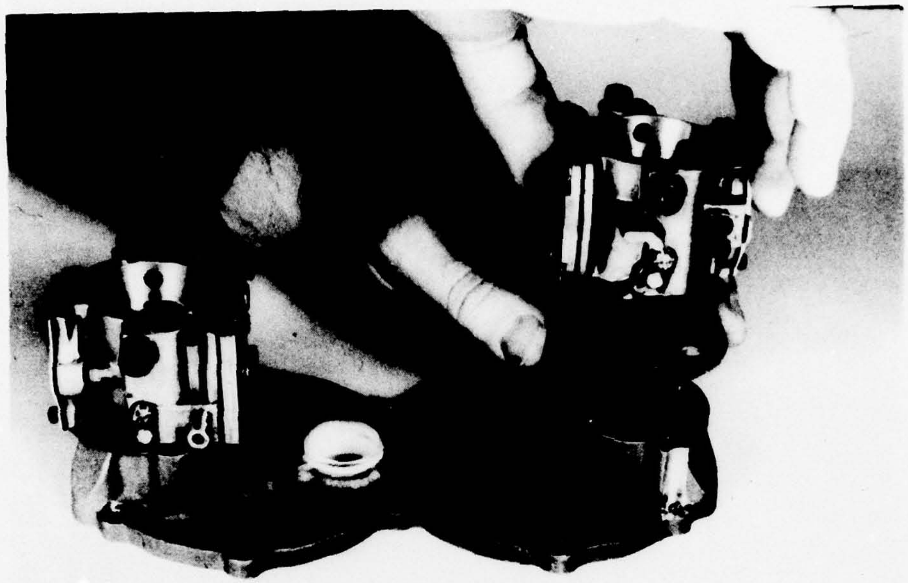


Figure D-9. Installation of Carburetor and Gasket.

2. Install carburetor mounting screws (#10 socket head cap screws) and torque to 40 in.-lb.

3. Rotate crankshaft until piston is at TDC. Lubricate tangs and install rotary valves as shown in Figure D-10. If rotary valves are properly installed, heads of screws will not be visible. Port will be open.

4. Position backplate gasket on housing and set backplate in place.

5. Install flat and lock washers and #8 socket head cap screws as shown in Figure D-11. Note location of three lock washers.

6. Install ignition sensor and secure sensor with hold-down and #8 screw as shown in Figure D-12.

7. Torque #8 socket head cap screws to 35 in.-lb. Use torque wrench and 9/64-in. hex key.

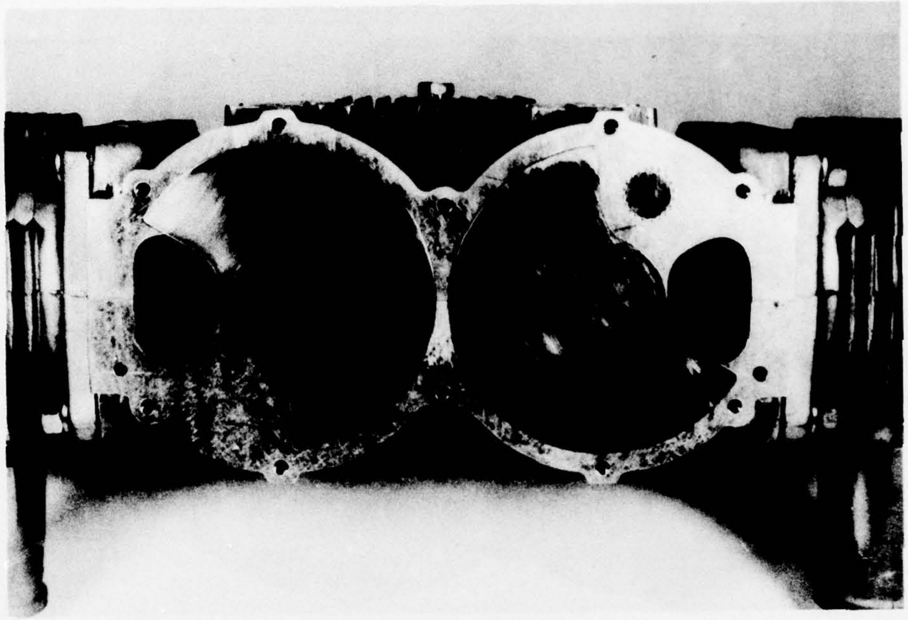


Figure D-10. Installation of Rotary Valves.

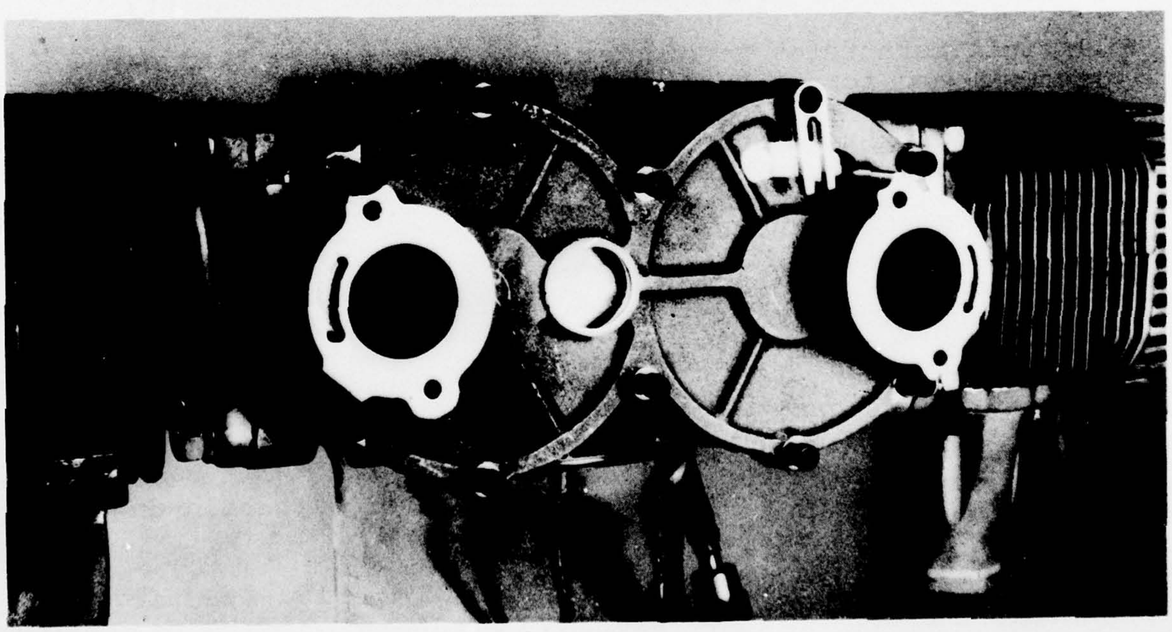


Figure D-11. Backplate Screws and Washers.

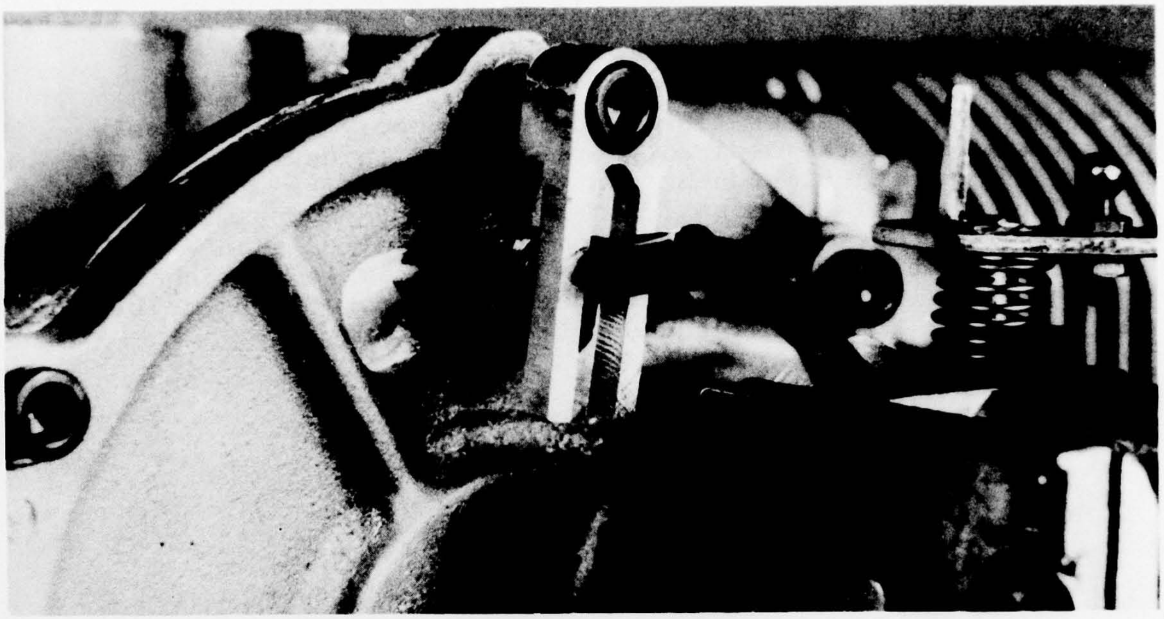


Figure D-12. Ignition Pickup Installation.

This completes the assembly of the engine proper. Connection of alternator, power conditioning unit (PCU), ignition system, and fuel system as well as installation of propeller remains.

PCU CONNECTION

Attach 3-pin connector from alternator into 3-pin receptacle on PCU. Tighten connector finger-tight. The 2-pin output connector is on the opposite side of the PCU.

IGNITION SYSTEM CONNECTION

Attach red and black leads from ignition module to power source; red to +28v, black to ground. Attach remaining two leads from ignition module to coil terminals. Either lead may be attached to either coil terminal. Connect high tension spark plug leads to coil and to plugs. Spark plug leads should be equipped with appropriate terminals to fit in coil securely. Attach the sensor lead from the ignition module to the ignition sensor using the 4-pin plastic connectors.

FUEL SYSTEM CONNECTION

Securely attach two 1/4-in. I.D. fuel lines to carburetor fuel fittings. These lines should contain in-line fuel filters.

PROPELLER INSTALLATION

With the propeller flange mounted loosely as previously described, mount the desired propeller and spinner on the propeller flange and tighten screws to 50 in.-lb for wooden props or as recommended by propeller manufacturer. Now remove the 3/8-in. socket head cap screw from the propeller flange and apply a small amount of Loctite. Replace the screw and, using a torque wrench and 5/16-in. hex key, tighten the screw to 60 ft-lb. Hold the propeller to prevent the shaft from turning.