GENERAL MOTORS SINGLE CYLINDER DIESEL ENGINE MODEL 1-53X3 INSTALLATION AND TESTS

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THESIS

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MODEL 1-53X3 INSTALLATION

AND TESTS

LCDR Stanley R. McCord

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INSTALLATION AND TESTS

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INSTALLATION AND TESTS

by

Stanley R. McCord

Lieutenant Commander, United States Navy

Submitted in partial fulfillment of
the requirements for the degree of

MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School

Monterey, California

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ABSTRACT

The GM 1-53X3 diesel engine was purchased by the Postgraduate School to be installed as a two cycle diesel engine, laboratory instruction unit. The objectives of this thesis were to design or select and integrate the necessary components to provide a laboratory test engine, to determine and document an initial set of performance data and to record a complete description of the entire installation.

The engine is described. The parameters demonstrating the engine performance are determined, presented and discussed. In the appendices a detailed description of the component systems is presented. Illustrations and graphs have been utilized to make system descriptions and performance characteristics clear. Operating instructions, safety precautions and sample calculations are discussed.

The engine is ready for student laboratory exercises. The instruments will permit exercises to determine output power, fuel consumption, air consumption and power distribution among the various energy loss factors. The engine is adaptable to other exercises with additional instrumentation. These modifications are discussed.

ACKNOWLEDGEMENTS

The writer wishes to express his appreciation for the assistance and encouragement given him by Professors Paul F. Pucci and Charles P. Howard during the pursuit of this project. The writer wishes to thank Mr. Joseph Beck for his assistance in assembling the equipment, and Mr. J. J. May of General Motors Corporation, for his advice and assistance.

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1. Engine description

The GM 1-53X3 is a single cylinder two stroke cycle diesel test unit built by General Motors Corporation, Detroit Diesel Engine Division, preliminary to the building of their series 53 production engines. It has a design power of 30HP at 3000 RPM. The production engines have subsequently been placed on the market in two, three or four cylinder units and do not in any exterior way resemble this experimental unit.

From personal correspondence with the project engineer at GM Corporation certain historical information has been determined. During the development stage two different type cylinder heads were created. One has four exhaust valves while the other has two exhaust valves. The four valve head has the larger exhaust area. Both type heads have been furnished with this engine. The four valve head has been used for conducting tests during this project because there are extensive differences in the connections and control arrangements to the two different heads, and the four valve head gives the better performance at the higher speeds. Previous tests at General Motors Corp. showed that the two valve head gives poorer performance by zero percent at low speed to 10 percent at top speed. As a result GM has installed the two valve head on the slower speed industrial models which operate at 1800 to 2200 RPM while the high speed engines up to 2800 RPM use the four valve head.

The two valve head had enough space to cast in a boss sufficiently large enough to bore out to a size that would permit using the larger size indicator pickups, such as the Photocan, Li or balanced pressure type. There was not enough space in the four valve head to cast in a

boss; instead, a stainless steel sleeve of smaller diameter was pressed in. The smaller diameter indicators, such as the S.L.M. can be used in the four valve head. Adapters can be made to permit the use of the S.L. M. indicators in the two valve head if desired.

The drilling done to install the pressure pickup sleeve, mentioned previously, necessitated reversing the injector rack and throttle linkage to such an extent, that in order to use the spare injectors with the 4 valve head, it will require that their racks be extended and reversed. It is recommended that one spare be so modified and the second spare be kept for the two valve head. Each head will then have one installed injector and one spare.

To use the two valve head, it will still be necessary to repipe the cooling water discharge connection and to install an opposite side throttle control system. The 4 valve head was used during all tests because of these differences.

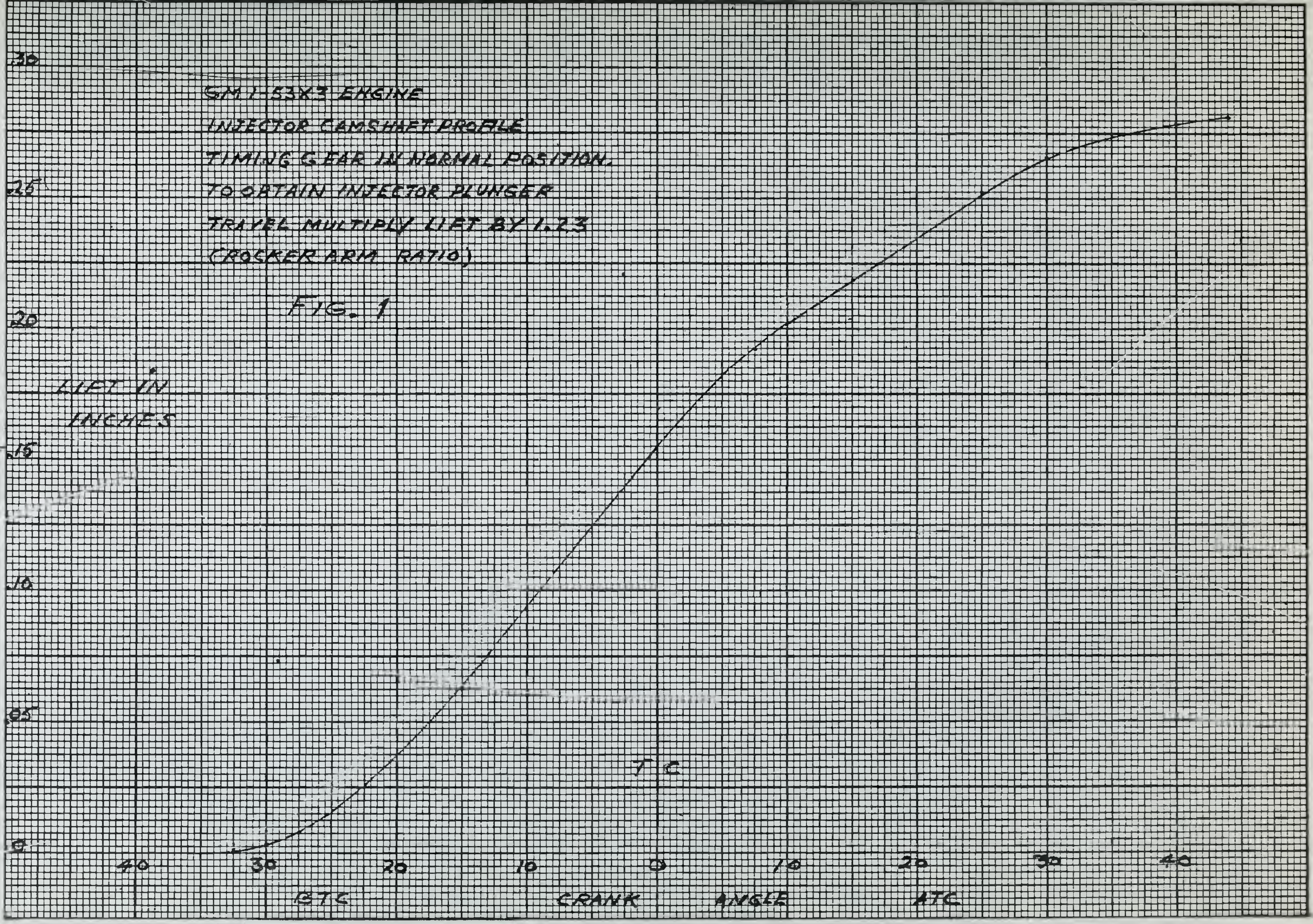
The engine frame is fabricated of steel weldments and has a wet type cast iron cylinder liner gasketed into bushings in the welded frame. The liner then forms the innermost wall of the cooling water jacket and the air box. It uses a forced feed lubricating oil system that is supplied by a pump and reservoir in the accessory drive box. The box is powered by 3 V-belts from the gear drive assembly.

The primary and secondary components of the reciprocating inertial forces are compensated for by the use of eccentric shafts for each component. The primary shafts rotate at engine speed and, of course, the secondary shafts rotate at twice engine speed.

All of the moving parts derive their timing from a gear train

located on the front panel of the engine frame in a gear train case. The fuel injection is accomplished by a unit injector which meters and times injection after positioning of the throttle which controls the injector rack and plunger. The exhaust valves and injector are operated by rocker arms and pushrods from their respective camshafts. The camshafts are located symmetrically on each side of the engine. Cam profiles have been taken of each camshaft for reference use in making timing changes. Fig. 1 is the displacement diagram for the injector cam and Fig. 2 is the displacement diagram for the exhaust valve camshaft.

Up to about 6 degrees of injector timing change can be obtained by using injector timing gauges (adjusting the plunger position). Fig 3 shows the timing possible by this means. Fig. 4 shows the theoretical injector performance for the normal injector timing position which is 1.484 inches.


GM 1-53X3 ENGINE
INJECTOR CAMSHAFT PROFILE
TIMING GEAR IN NORMAL POSITION.
TO OBTAIN INJECTOR PLUNGER
TRAVEL MULTIPLY LIFT BY 1.23
(ROCKER ARM RATIO)
LIFT IN
INCHES
.30
.25
.20
.15
.10
.05
0
40
30
20
10
0
10
20
30
40
TC
BTC
CRANK ANGLE
ATC
FIG. 1

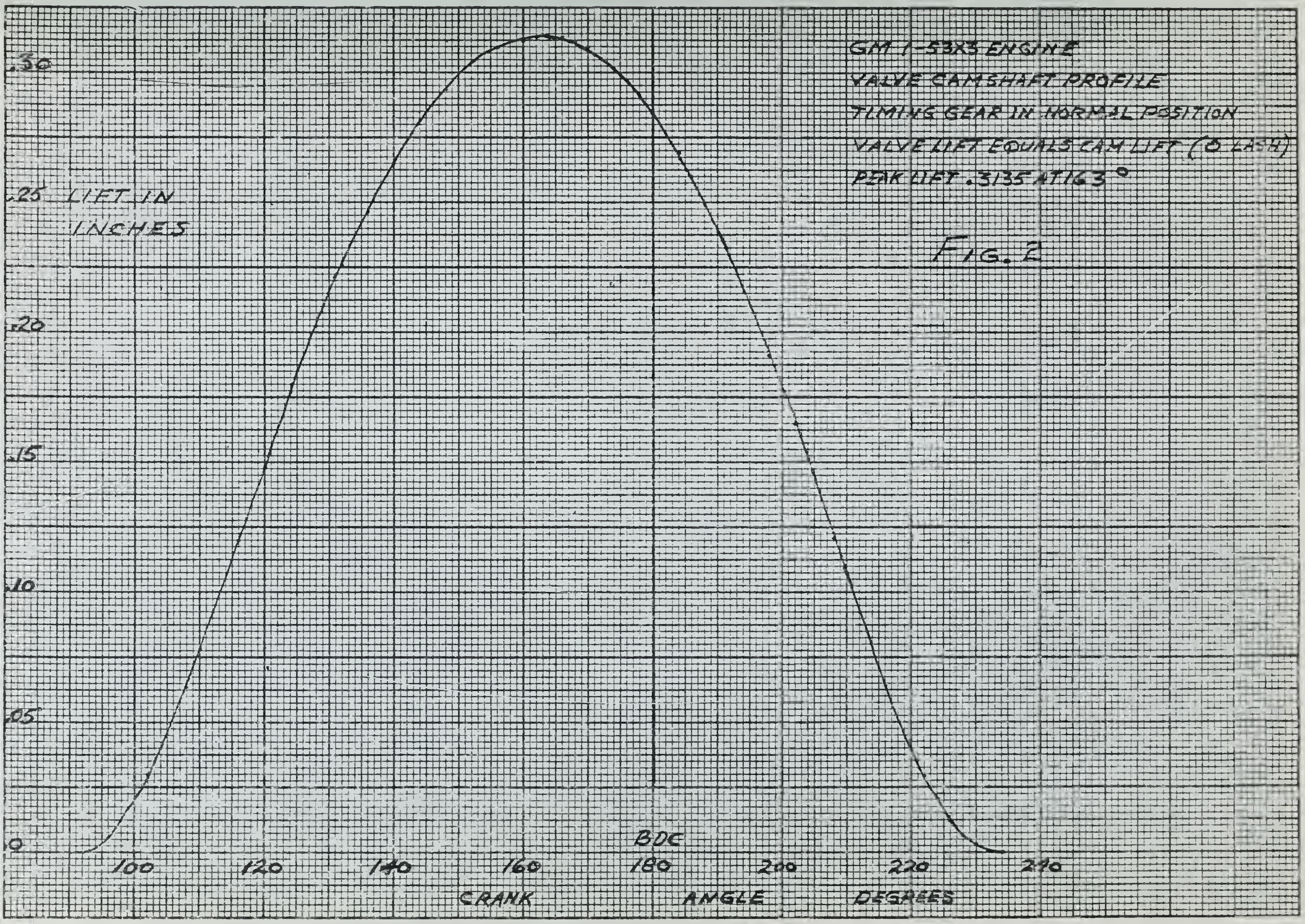
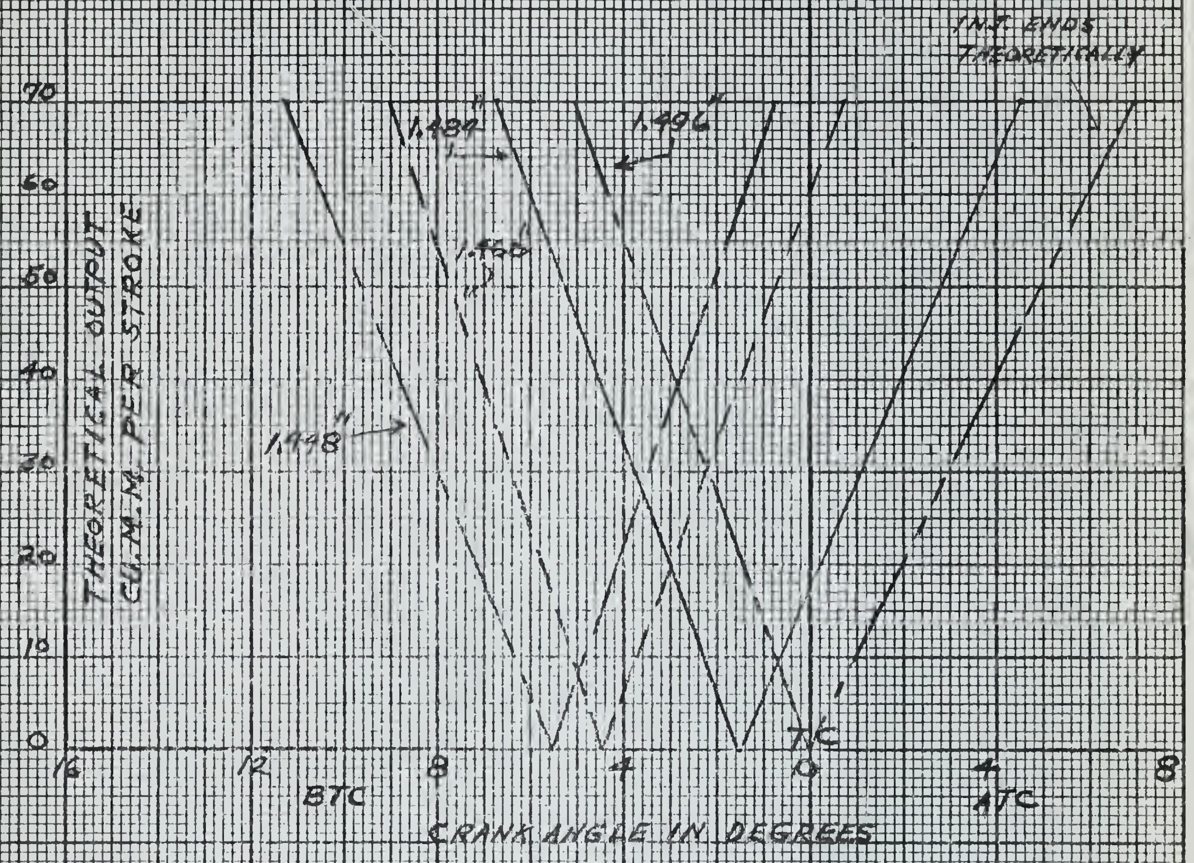
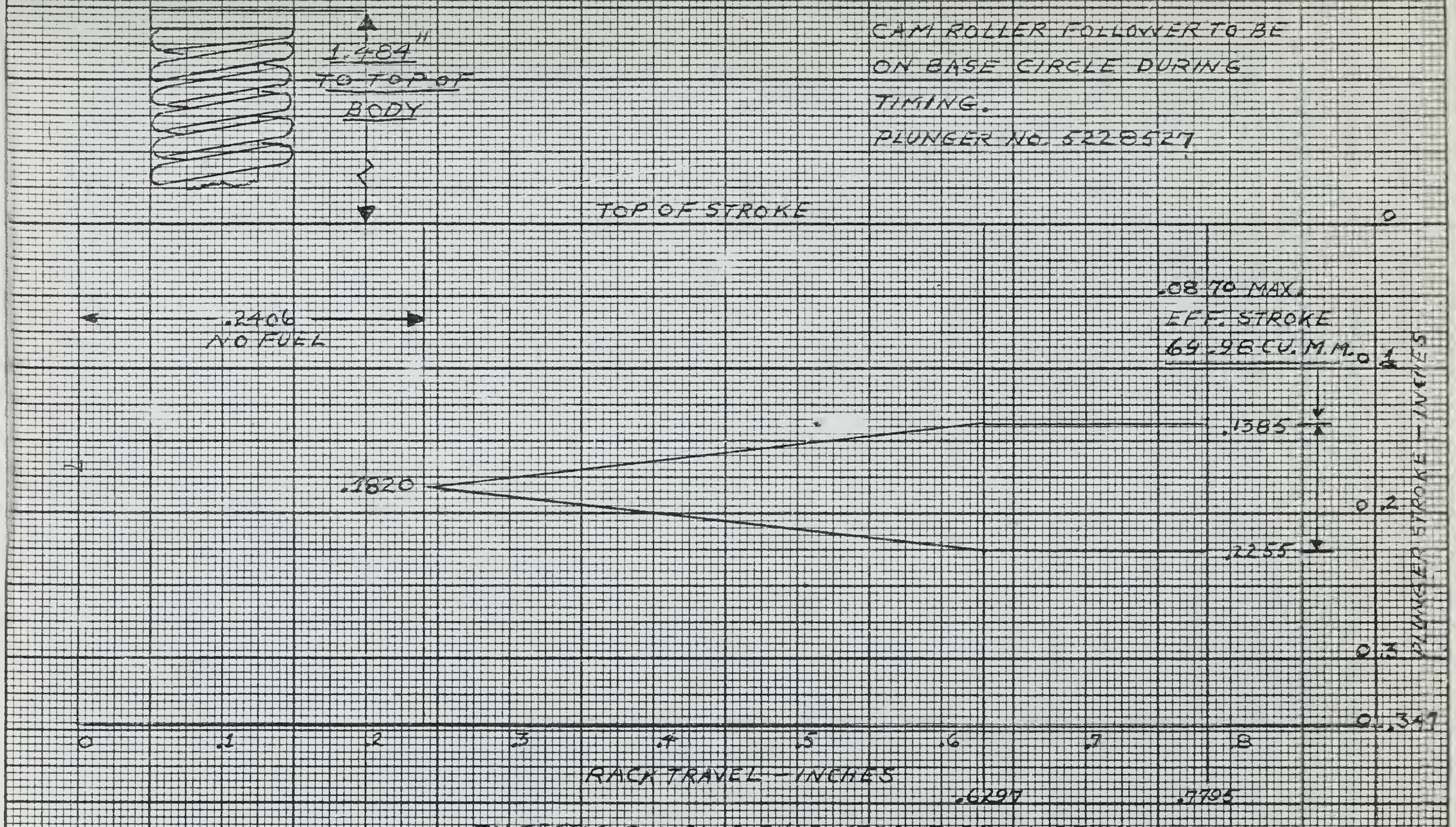

GM 1-53X3 ENGINE
VALVE CAMSHAFT PROFILE
TIMING GEAR IN NORMAL POSITION
VALVE LIFT EQUALS CAM LIFT (0 LASH)
PEAK LIFT .3135 AT 163°
LIFT IN INCHES
.30
.25
.20
.15
.10
.05
0
100
120
140
160
BDC
180
200
220
240
CRANK ANGLE DEGREES

FIG. 2



RANGE OF INJECTOR TIMING POSSIBLE ON 1-55X3 ENGINE BY TIMING GAUGE DIMENSION CHANGE. CRANK ANGLES BASED ON NORMAL INJ. CAMSHAFT POSITION (TIMING MARKS ALIGNED)

Fig. 3



INJECTOR PUMPING CHARACTERISTICS MODEL 71

FIG. 4

Any further and larger changes in timing must be made by shifting the timing gears in the gear train box. In order to make this a reasonably simple mechanical shift, special gears have been designed to replace the present cast steel camshaft gears. They will consist of a set of helical gears identical to the present gears except for the provision of loosening the hub from the rim and shifting the camshaft up to 15 degrees on either side of the normal position.

Air is supplied to the engine by a Roots three lobe helical blower. It is driven by V-belts from the front of the engine. This supplys scavenging air and a short period of supercharging, if the exhaust valve timing is so adjusted to allow time for supercharging.

The detailed engine specifications are contained in Appendix A.

Figures 5-10 show photographic views of the installation and the pertinent components.

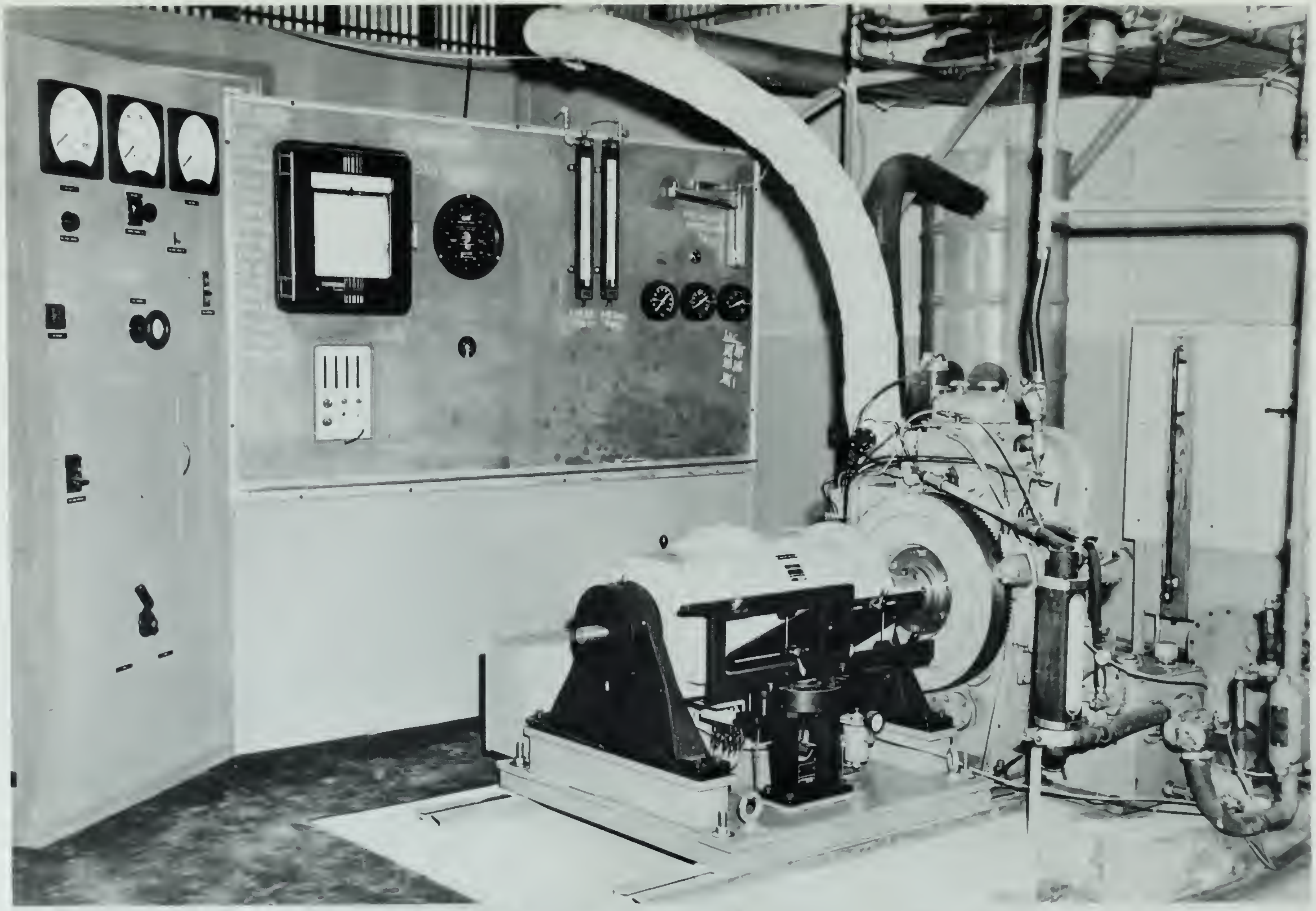


Fig. 5. General view of the GM 1-53X3 test installation.

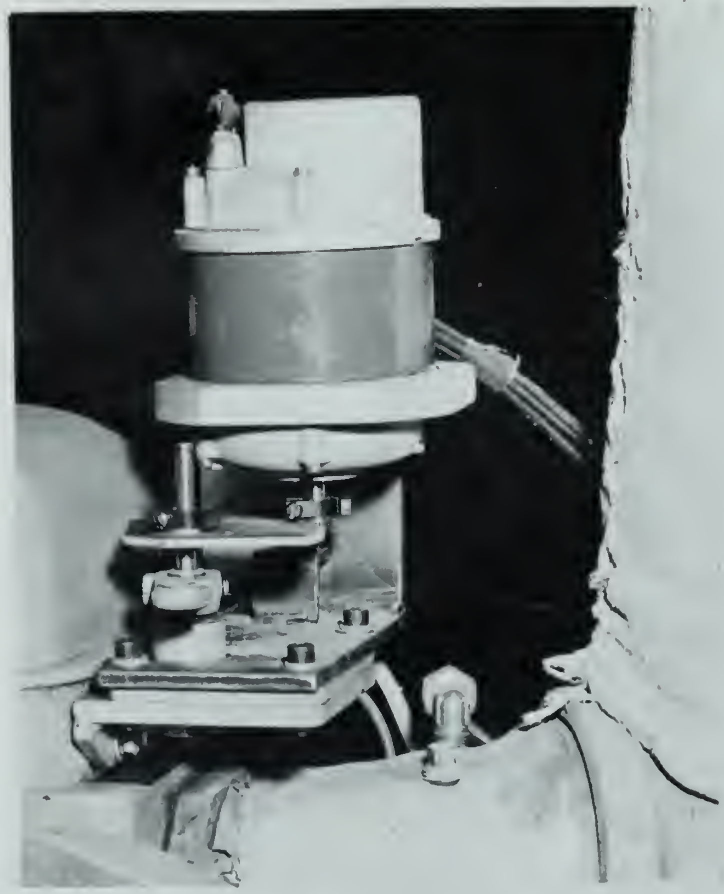


Fig. 6. Throttle control arrangement (close up view).

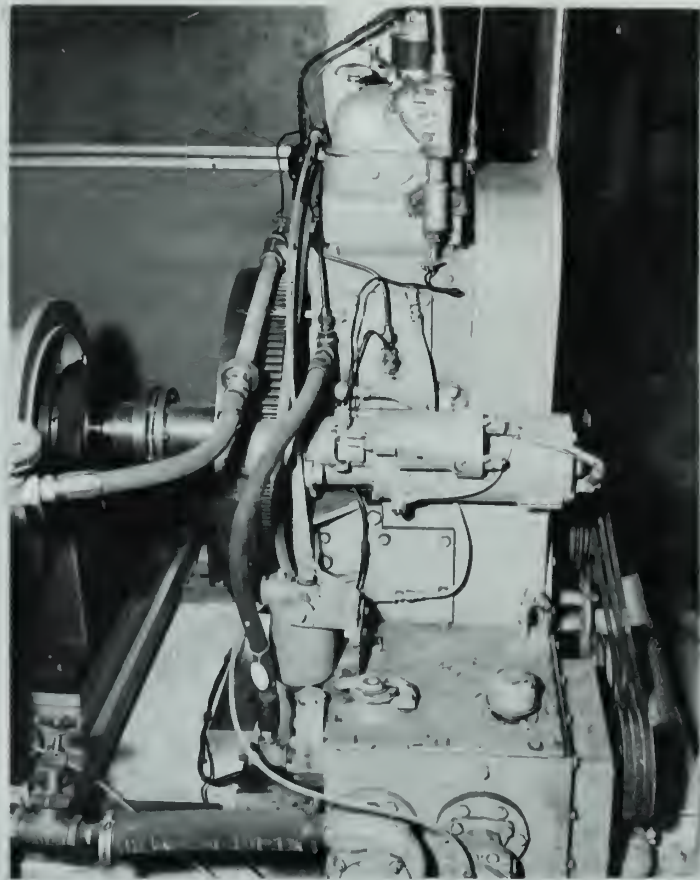


Fig. 7. GM 1-53X3 engine and accessory drive box.

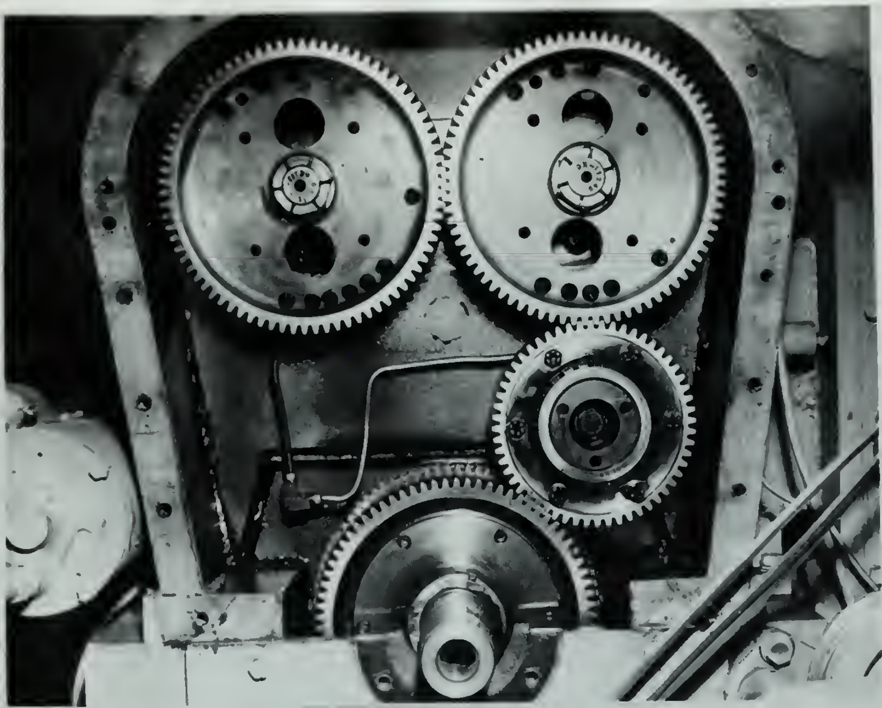


Fig. 8. Timing Gears.

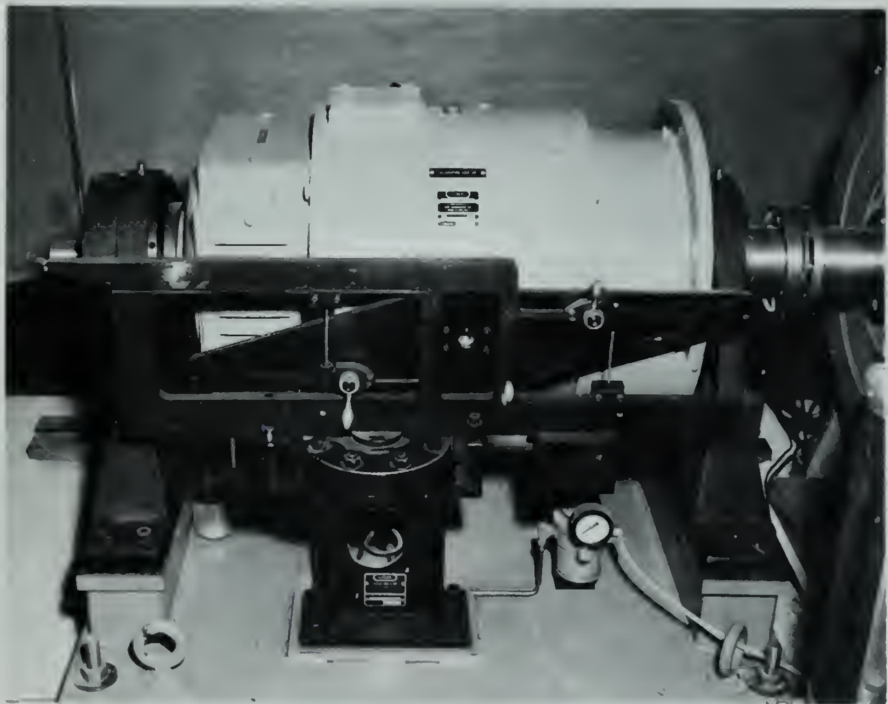


Fig. 9. Generator and link "UNIBEAM", model 302-35 dynamometer arrangement.

Fig. 10. Air measuring chamber and air flow nozzles.

2. Performance tests.

The following material covers running tests conducted on the GM 1-53X3 engine upon completion of the installation and instrumentation work.

In order to have a reference from which to judge the performance of the engine, the normal position of the timing gears was elected to be with the marks on the gears aligned according to the blueprint furnished with the engine. This was not the manner of alignment when the engine was received from the GM laboratory. As received the injector timing gear was advanced two teeth and the injector plunger timing was set at 1.500 inches. The exhaust valve timing gear was retarded one gear tooth, that is, it occurred later. Since there are 78 teeth per 360 degrees on each of these gears, one tooth represents 4.6 degrees of cyclic timing change.

The gear train was disassembled and shifted to bring the marks back into alignment. A dial micrometer was mounted on the top of the rocker arm push rods for each camshaft and the cam profiles were determined as shown in the first section of this report. By using this cam lift diagram, Fig. 1, and the injector timing diagram, Fig. 3, the theoretical injection time can be determined for any subsequent setting of the camshafts and the injector.

As an additional feature to be added to the engine, a new set of adjustable timing gears were designed as part of this project. When they are completed and installed, it will become easier to make finite timing changes in both the injection and in the exhaust valve popping time.

In the meantime, tests were conducted by pulling gears and carefully shifting them to obtain the different timing conditions desired. The

injector plunger timing measurement that is used on the production version of this engine is 1.484 inches. It is recommended for use on the 1-53X3 engine as well. The injector characteristics are known for this setting as shown on Fig. 4.

Three different timing configurations were elected for investigation. They were assigned code designations of A-1, A-2, and A-3 respectively. Test series A-1 was elected to be that configuration which provided alignment of the gear train in the normal manner as shown on the blueprint. Test series A-2 was selected as that arrangement which afforded a definite advance in fuel injection time over the normal configuration. Test series A-3 was conducted with no further change in fuel injection time, but a change was made that delayed the exhaust valve popping time. The latter timing configuration is that arrangement that existed when the engine was received from the GM laboratory.

The sequence of events, as they occur in one cycle of operation, are visually depicted in the timing diagrams - Figures 11, 12, and 13.

The following relationships were determined from the recorded data of tests: A-1 and A-2:

a) Engine characteristic curve: BHP vs. RPM at full load.

b) Torque vs. engine RPM, at full load.

c) Specific fuel consumption vs. engine RPM, at full load.

d) Brake thermal efficiency vs. engine RPM, at full load, and also at varying loads while at constant speed.

e) Air-fuel ratio vs. BHP, at different engine constant speeds.

f) Power distribution vs. speed at full load (Test A-3 only).

g) Scavenging ratio vs. engine RPM, at full load.

For test A-3, only the full load characteristics were determined for comparison with those of test A-2.

In addition to the above investigations a test was conducted to determine the friction and pumping losses of the engine. This was accomplished by bringing the engine to operating temperature (primarily the lube oil temperature) and then driving the engine with the dynamometer while the fuel was not being injected into the engine.

Procedure: To take data for tests A-1 and A-2, the engine was started and brought to operating temperature. A warm up temperature of 180 degrees for water and 150 degrees for lube oil was used as a starting point for the tests. The engine throttle and torque loading was adjusted to give maximum possible torque loading at approximately 500 or 600 RPM (depending on how the engine tended to labor at this slow speed). The throttle was wide open and the loading was at the maximum. In test A-2 the minimum speed possible was 600 RPM because the engine could not be stabilized at 500 RPM with full throttle and the laboring load. When the load and speed stabilized the water cooling flow was adjusted to bring temperature equilibrium to the cooling system. Readings of all instruments and a fuel metering run were obtained while another operator maintained the speed as constant as possible by adjusting the load. All speeds were maintained within $\pm$ 0.5% while taking the data. Upon completion of obtaining the full load data, the throttle and load were adjusted to 3/4 of the torque loading and the same speed. After stabilization a complete set of data was again recorded. This was repeated for 1/2 load, 1/4 load and on test A-2, it was done for minimum load with the generator open circuited.

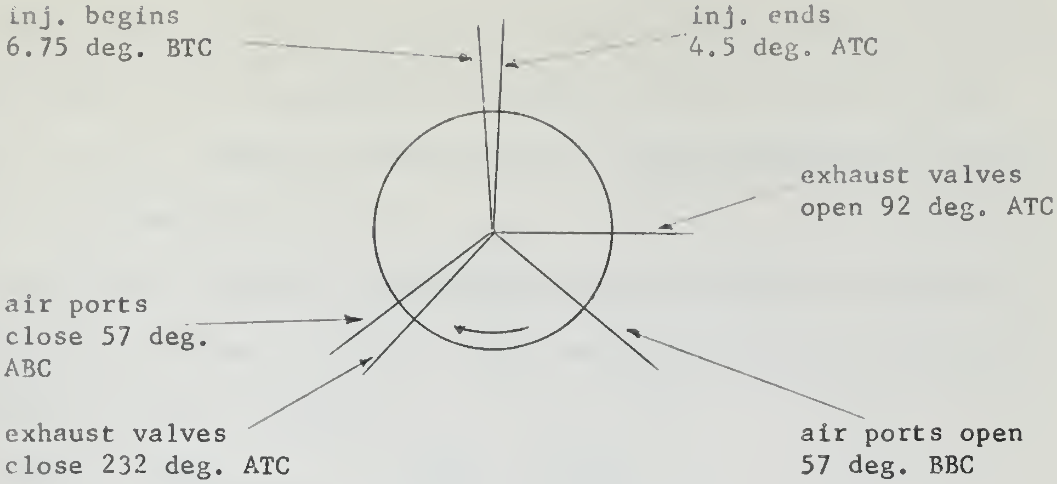


Fig. 11. Timing diagram for test A-1

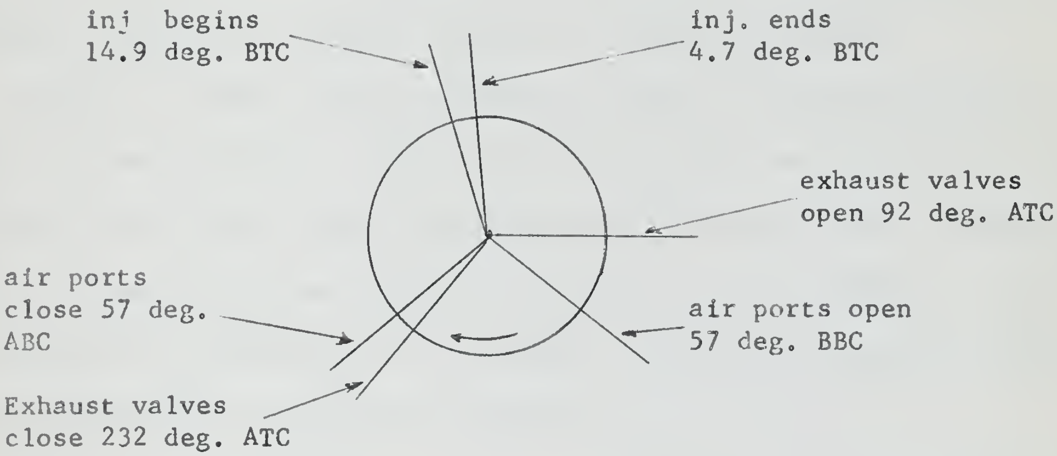


Fig. 12. Timing diagram for test A-2

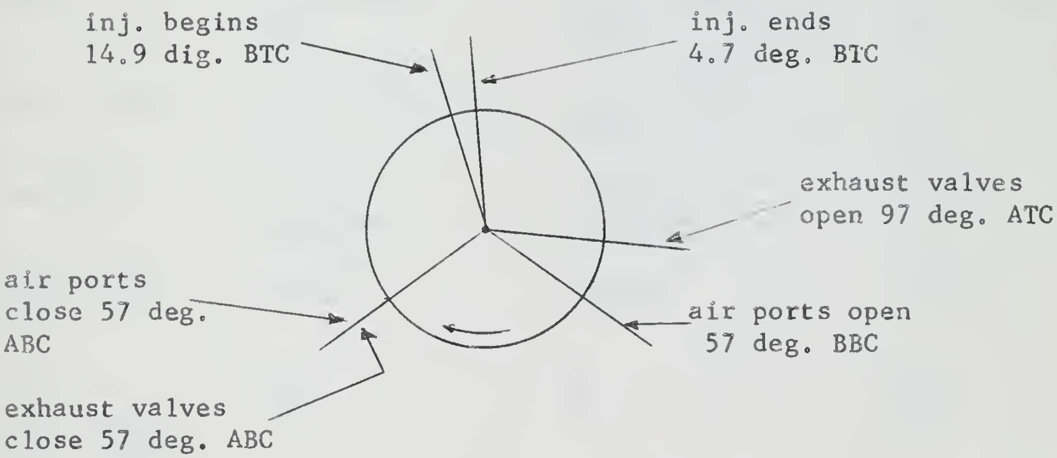


Fig. 13. Timing diagram for test A-3

For test A-3 the same procedure was used except that only the full load characteristic was desired for comparison with test A-2, since there was only a slight difference in the timing configuration during test A-3 compared to test A-2.

All brake horsepower calculations were standardized using a correction factor taken from [5]. It is determined from the following formula, the S.A.E. standard:

$$BHP_{std} = BHP_{observed} \times \frac{29.92}{p_o} \sqrt{\frac{T_o}{520}}$$

where T_o is the laboratory temperature in degrees Rankine, and p_o is the laboratory barometric pressure reading in inches of mercury.

Tables 1, 2 and 3 show the tabulated summary of calculated data reduced from the test data. Fig. 14 shows the engine characteristic curves of BHP vs. RPM at full load, and torque vs. RPM at full load. Reference lines of constant BMEP are drawn in to show clearly the BMEP that exists at any point on the BHP curve. From Fig. 14, the following observations can be noted and are tabulated as follows:

ITEM	Test A-1	Test A-2	Test A-3
a) Max. BHP	22.6 @ 2750 RPM	25.3 @ 2955 RPM	25.5 @ 2700 RPM
b) Max. Torque	63.0 @ 1200 RPM	62.5 @ 1500 RPM	63.6 @ 1500 RPM
c) Max BMEP	87 psi @ 1100 RPM	87 psi @ 1250 RPM	89 psi @ 1250 RPM
d) BMEP @ max. power	63 psi	66 psi	70 psi
e) speed @ $\frac{\text{max. power}}{\text{max. torque}}$	$\frac{2.29}{1}$	$\frac{2}{1}$	$\frac{1.8}{1}$

Note how the shift of timing has increased the power output in the range from 1500 to 3000 RPM.

The manufacturer lists the following rated horsepowers for the production 3 cylinder version of the 1-53X3 engine [6]. These ratings are compared on a unit cylinder basis with the best performance of the 1-53X3 and are presented in Fig. 15.

RPM	Torque lb ft	STD BHP	Fuel H.P.	B.th.eff. percent	Specific fuel Consumption	Scavenging ratio	Air-fuel Ratio
526	50.7	4.94	27.6	17.8	.738		25/1
524	37.1	3.72	13.75	27.0	.505	1.3	50.4/1
526	25.1	2.46	10.00	24.6	.540		69/1
526	12.5	1.22	6.84	17.9	.745		101/1
987	62.4	11.41	47.1	24.4	.547		28.2/1
987	46.6	8.59	30.8	27.9	.478	1.395	43.2/1
987	31.4	5.77	22.95	25.2	.529		58/1
987	15.6	2.86	15.00	19.1	.695		89/1
1495	61.5	17.1	67.6	25.2	.526		34/1
1495	46.1	12.8	45.1	28.4	.467	1.63	51.2/1
1495	31.0	8.60	33.9	25.4	.523		68.4/1
1495	15.6	4.34	25.1	17.3	.770		92/1
1998	53.4	20.6	86.0	22.8	.576		31.8/1
1998	40.0	15.41	61.6	25.0	.553		44.3/1
1998	26.4	10.2	44.6	22.8	.605	1.47	61.4/1
1998	13.6	5.25	33.8	15.5	.890		81/1
2496	47.1	22.8	105.4	21.6	.640		34.5/1
2496	36.7	17.7	77.0	23.0	.600	1.56	47.4/1
2496	23.8	11.5	60.4	19.06	.726		60.2/1
2496	12.1	5.84	44.4	13.0	1.051		82/1
2955	39.8	22.8	124.8	18.22	.759		34/1
2955	30.1	17.2	94.5	18.15	.761	1.56	44.8/1
2955	19.8	11.3	78.5	14.36	.960		54/1
2955	10.0	5.71	59.7	9.55	1.441		71/1

Table 1. Tabulated results of performance test A-1.

RPM	Torque lb-ft.	Std. BHP	Fuel HP	B.th.eff. Percent	Specific fuel Consump.	Scav. Ratio	Air-Fuel Ratio
608	52.0	5.95	29.4	20.6	.664	1.24	25.1/1
605	38.9	4.50	17.9	25.1	.545		41.1/1
607	26.0	3.01	13.0	23.2	.590		56.6/1
604	13.0	1.50	9.45	15.9	.862		77.9/1
608	1.40	0.163	5.90	2.75	4.96		124.9/1
1015	55.6	10.8	48.4	22.4	.614	1.37	26.7/1
1005	41.5	8.00	29.7	26.9	.510		43.5/1
1004	27.4	5.27	22.6	23.4	.586		51.1/1
1002	14.1	2.69	15.85	17.0	.806		81.5/1
1003	1.60	0.308	9.40	3.34	4.19		137.8/1
1506	62.5	18.0	63.0	28.8	.478	1.46	33.0/1
1505	46.3	13.3	47.2	28.2	.485		44.3/1
1504	30.8	8.85	36.2	24.4	.558		57.5/1
1502	15.5	4.46	25.0	17.88	.771		82.5/1
1500	1.8	0.516	17.0	3.04	4.51		119.0/1
1990	55.2	21.0	85.9	24.4	.560	1.47	32.9/1
1985	41.3	15.7	63.1	24.9	.550		44.8/1
1989	26.7	10.25	47.0	21.8	.638		60.2/1
1992	13.4	5.11	34.8	14.71	.936		81.1/1
1991	2.0	0.761	26.3	2.89	4.76		107.5/1
2493	51.0	24.3	102.9	23.9	.578	1.56	35.6/1
2503	37.8	18.05	77.0	23.5	.583		47.5/1
2503	25.4	12.17	60.4	20.1	.680		60.5/1
2506	12.8	6.15	47.1	13.05	1.05		77.5/1
2500	2.1	1.005	35.6	2.82	4.86		102.5/1
2952	45.0	25.4	121.9	20.9	.656	1.55	34.8/1
2959	33.9	19.2	95.4	20.1	.681		44.4/1
2954	22.3	12.6	77.0	16.4	.837		55.0/1
2957	11.4	6.45	60.8	10.6	1.29		69.6/1
2958	2.4	1.351	49.8	2.71	5.02		85.0/1

Table 2. Tabulated results of performance test A-2

RPM	Torque lb-ft	Std BHP	Fuel HP	Bth. eff. Percent	S.F.C	Scav Ratio	Air-Fuel Ratio
608	54.8	6.35	28.8	22.0	.617	1.271	26.3/1
1014	57.5	11.1	48.0	23.1	.588	1.385	28.2/1
1502	63.6	18.22	64.0	28.5	.478	1.465	33.4/1
2010	58.0	22.2	86.0	25.8	.527	1.490	33.9/1
2514	53.2	25.2	101.2	25.0	.544	1.520	36.2/1
2905	45.1	25.0	120.5	20.8	.656	1.55	35.8/1

Table 3. Tabulated results of performance test A-3.

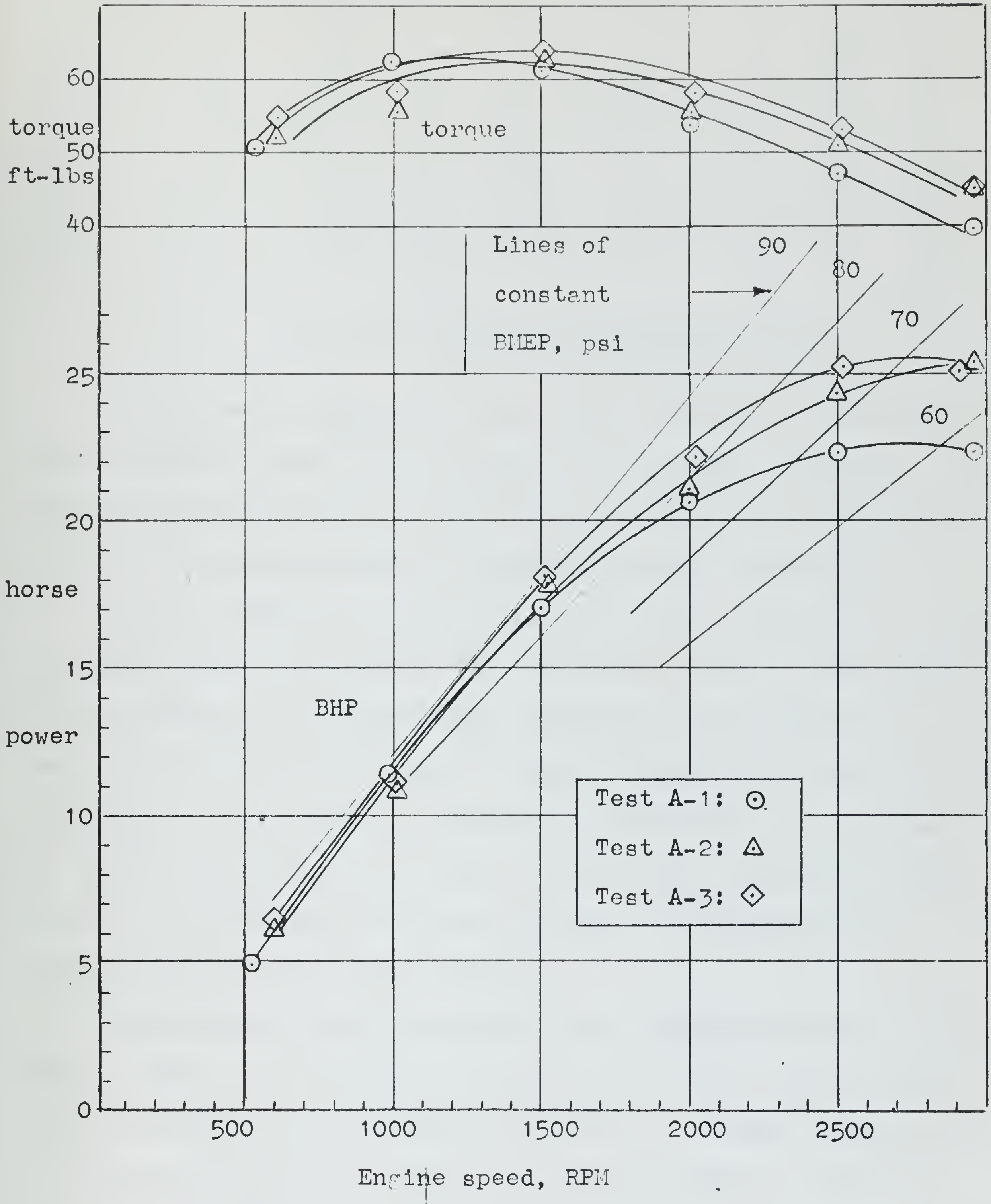


Fig. 14. Engine full power characteristics.

RPM	BHP		TORQUE	
	3-53	1-53	3-53	1-53
1000	12	12	62	62
1500	18	18	62	63
2000	21.3	22.5	61	61
2500	26.6	25.25	56	53

Fig. 15. Comparison of manufacturer's performance claims with results of tests.

From the above comparison it would appear that there is close agreement between test results and the manufacturer's claimed performance. The data on the GM 3-53 with the 4 valve head was obtained from curves presented in an advertising brochure for the series 53 diesel engines [6].

Fig. 16, shows the full load brake specific fuel consumption vs. RPM and brake thermal efficiency vs. RPM. From these curves the improvement in both efficiency and specific fuel consumption can be readily noted after advancing the fuel injection. There was insignificant difference during test A-3 as a result of delaying the exhaust valve popping time. It should be noted that data points on these curves at 1000 and 2000 RPM appear to be inaccurate, but the data for test A-3 substantiated the curves that were faired in for test A-2.

The manufacturer lists the specific fuel consumption of the production engines based on the continuous ratings which range from 83% of the rated BHP at 1000 RPM to 75% of rated BHP at 2500 RPM. An example specific fuel consumption listed by the manufacturer is 0.442 lbs/BHP-hr at 1500 RPM when delivering 83% of rated load. This compares with 0.467 lbs/BHP-hr obtained under similar power conditions. Otherwise, specific fuel

consumptions determined during these tests were somewhat higher by 0.1 lbs/BHP-hr.

Fig. 17A and 17B show the brake thermal efficiency vs power output as calculated from test data. Examination of the figures shows that the best operating range of the engine is 1000 to 2200 RPM. The manufacturer lists the maximum continuous engine speed under load as 2200 RPM. This is for all versions of the 53 production series except the 2-53 with the two valve head. The results of this testing will support the manufacturer's recommendation.

Fig. 18A and 18B show the range of air-fuel ratios. In the two cycle diesel engine using a scavenging blower the air-fuel ratio has no real significance since it does not truly state the mixture of air and fuel. The scavenging ratio (discussed in Appendix E) does give a better index of the air-fuel mixture. The scavenging efficiency is to the two cycle engine what the volumetric efficiency is to the four cycle engine.

Fig. 19 shows a percentage breakdown of the distribution of the fuel horsepower supplied to the engine while operating at full load and variable speed. Data is from test A-3.

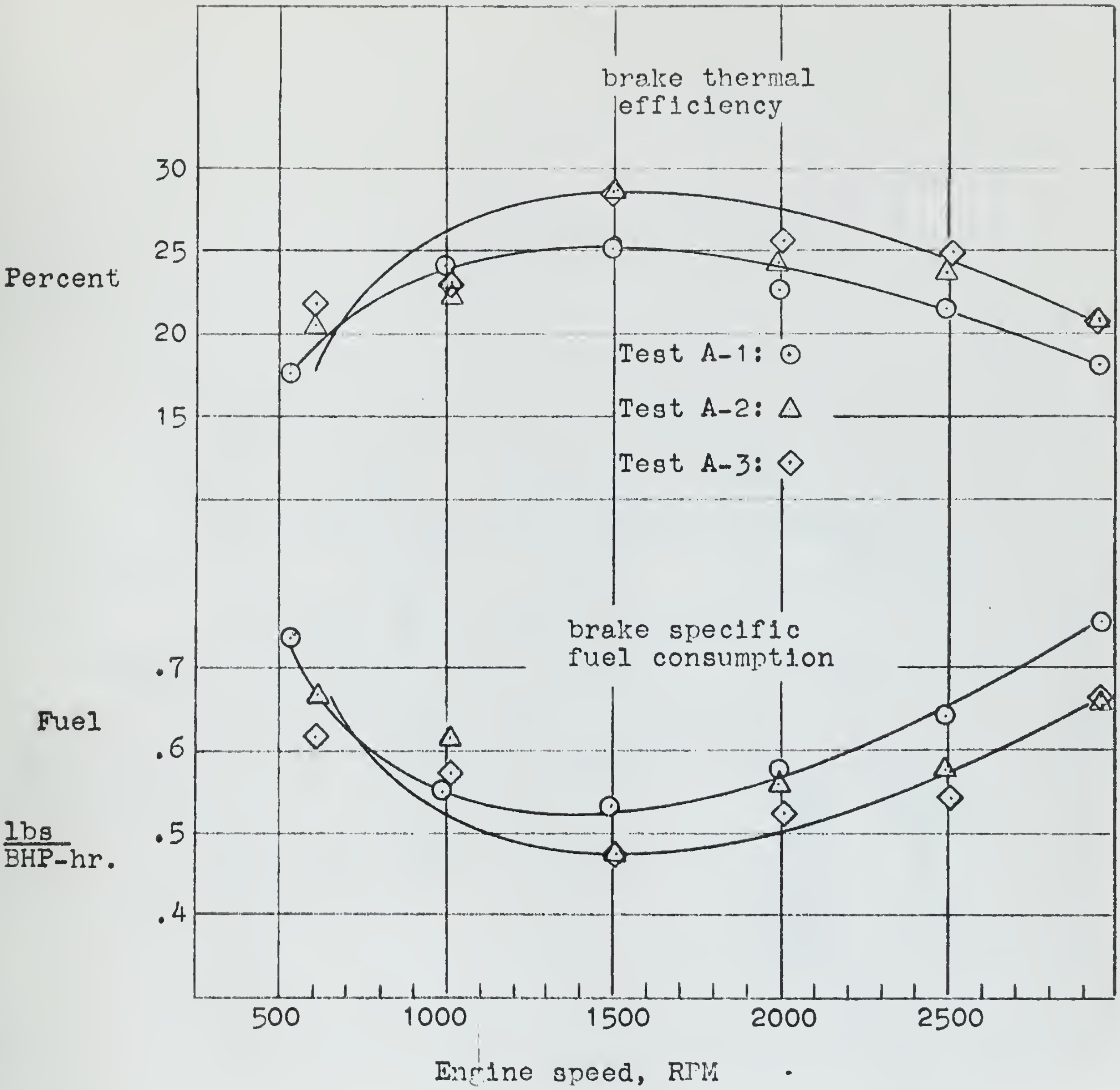


Fig. 16. Full load fuel consumption and brake thermal efficiency vs. engine speed.

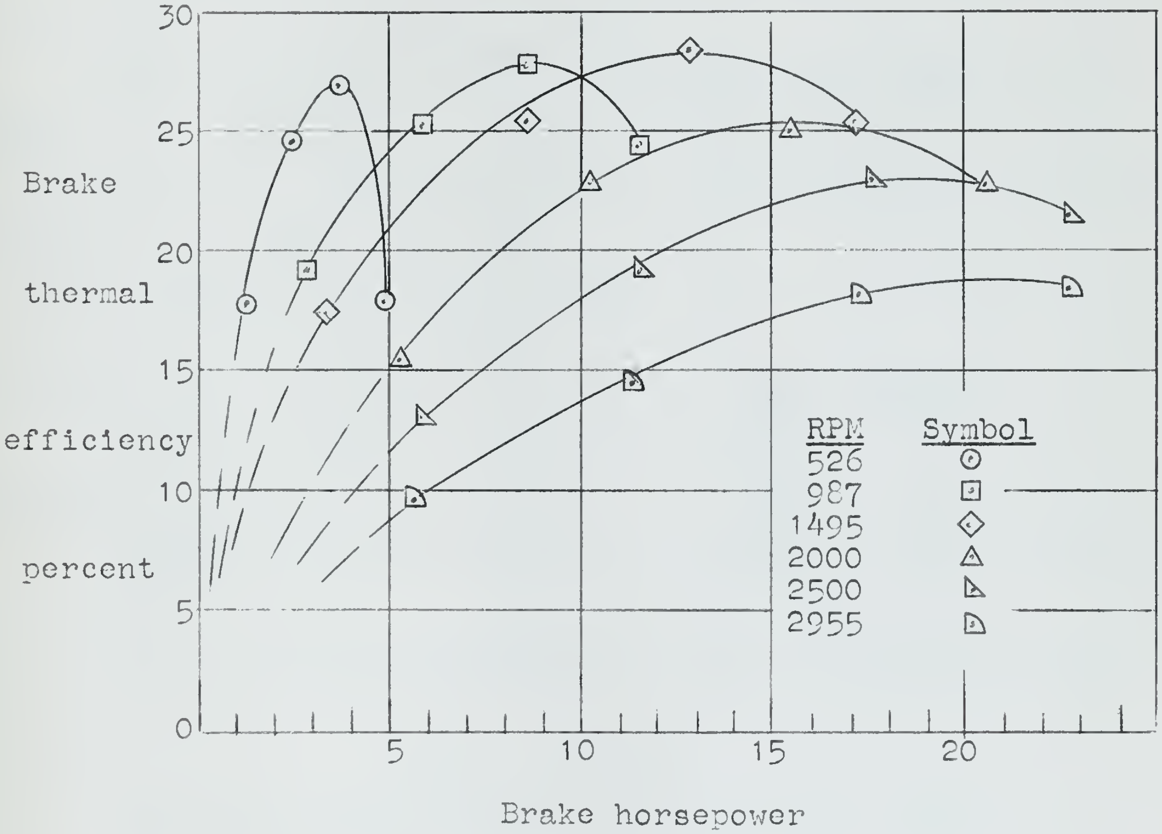


Fig. 17A. Brake thermal efficiency vs. brake horsepower; from test A-1

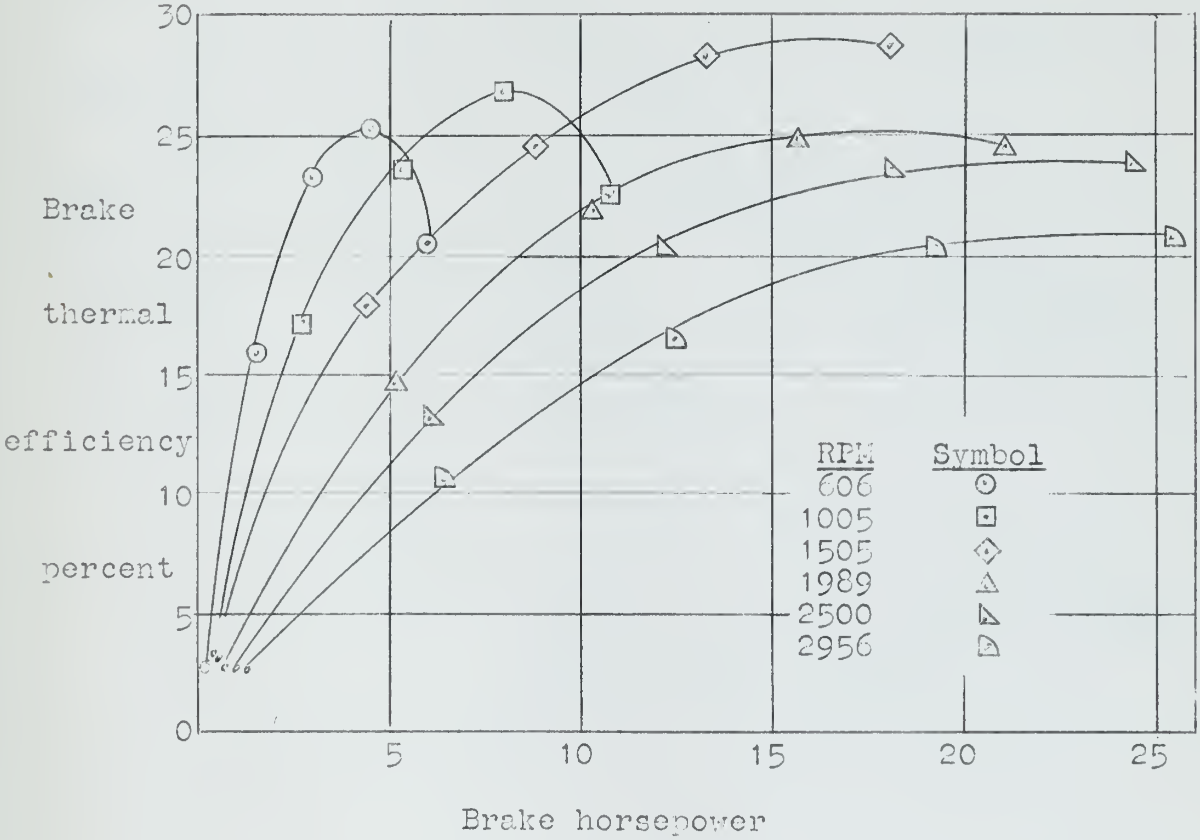


Fig. 17B. Brake thermal efficiency vs. brake horsepower; from test A-2.

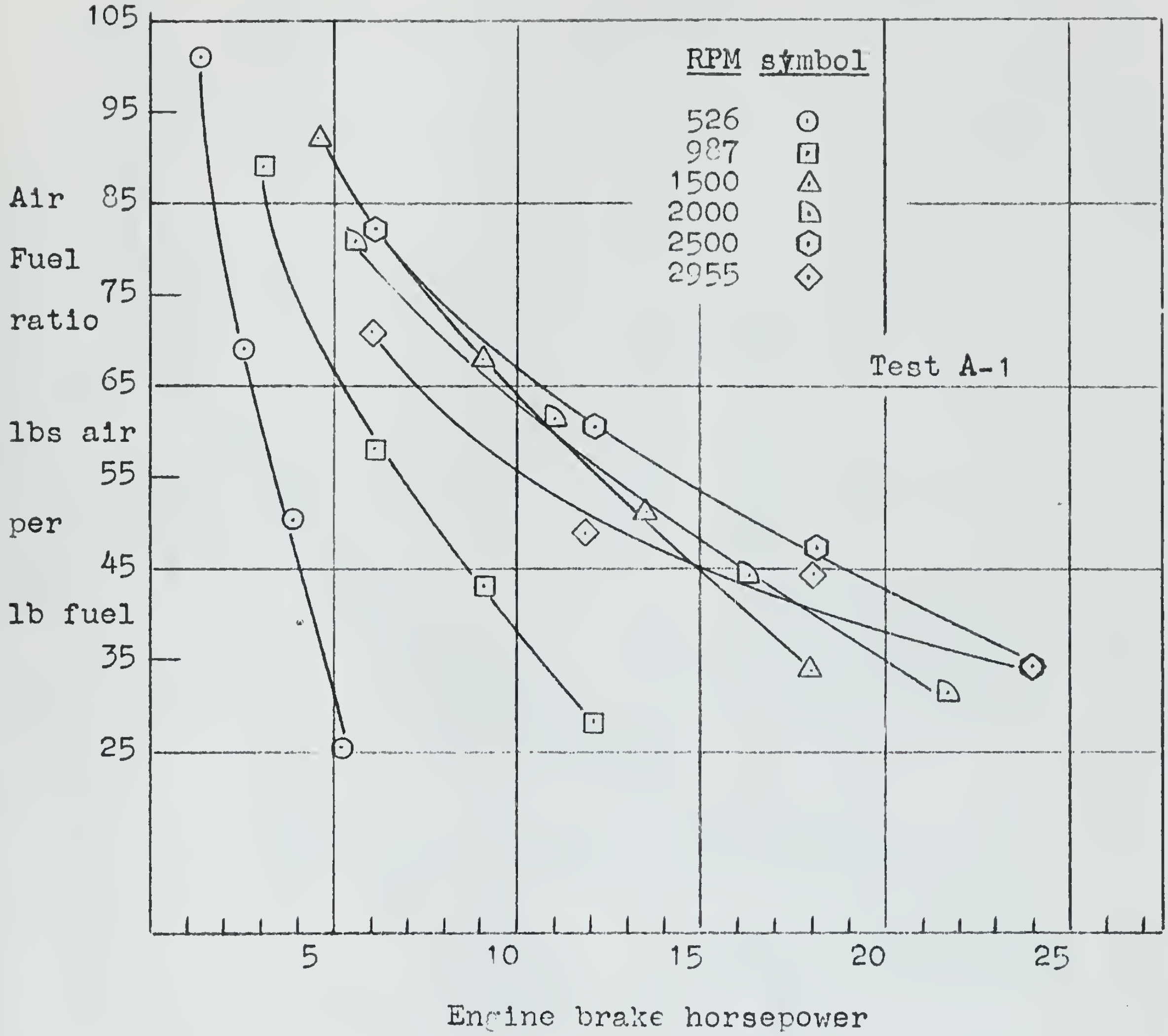


Fig. 18A. Air/fuel ratio vs. brake horsepower.

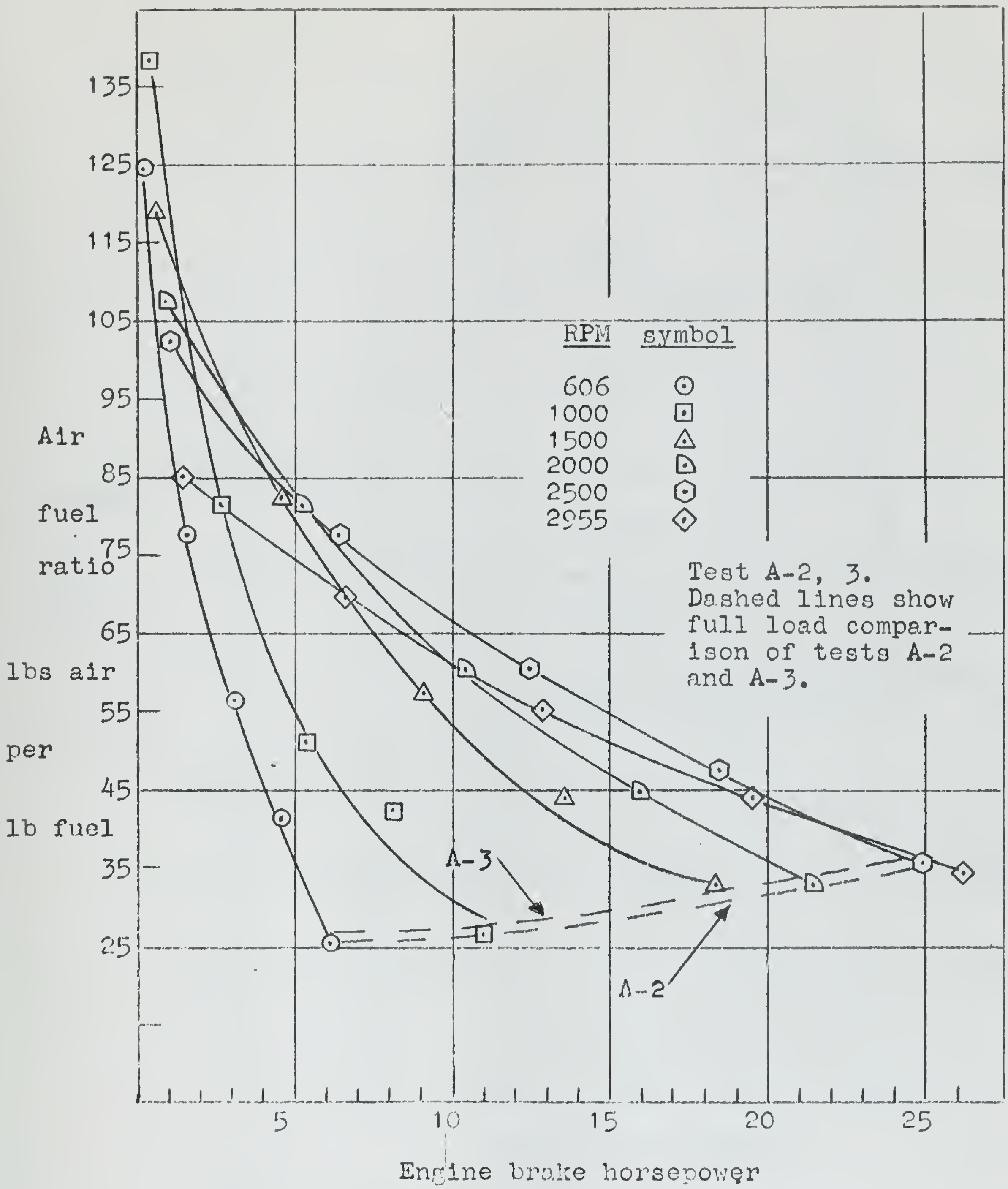


Fig. 18B. Air/fuel ratio vs. brake horsepower.

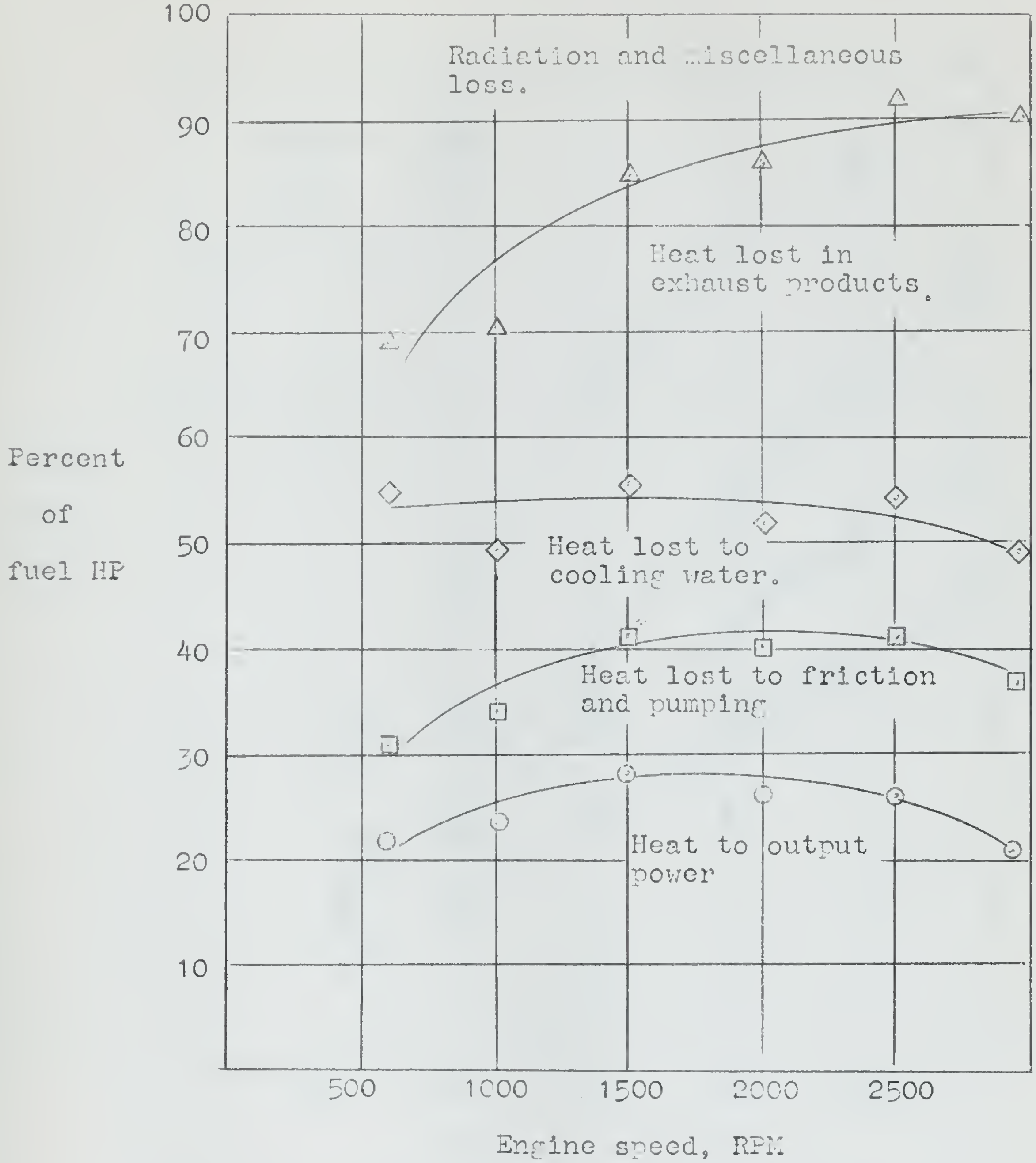


Fig. 19. Power distribution, variable engine speed at full load; from test A-3.

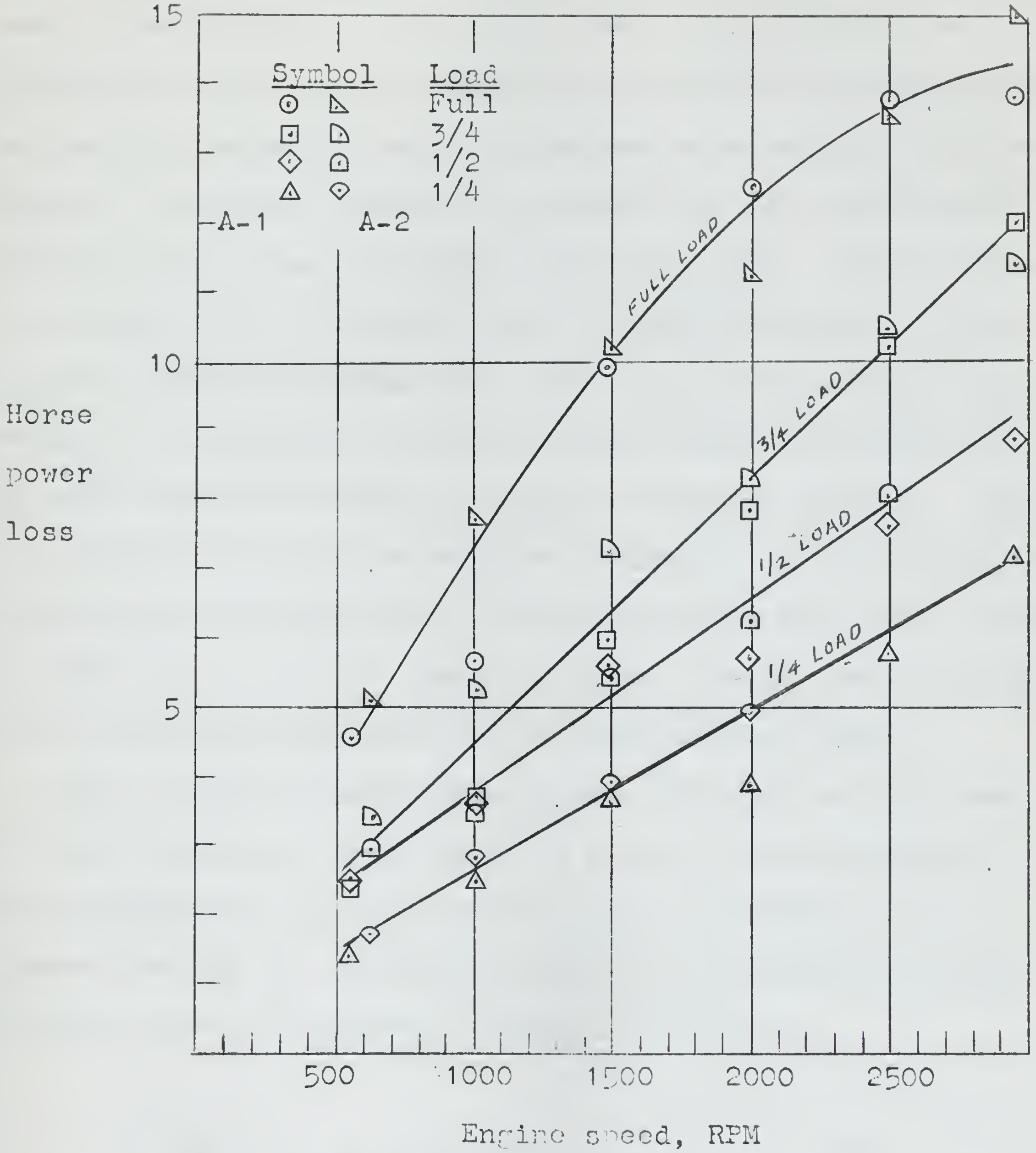


Fig. 20. Curves of cooling water power loss vs. engine speed; from data collected during tests A-1, A-2.

Determination of mechanical losses:

In order to determine the friction and pumping losses on a single cylinder engine it is necessary to drive the engine by motoring the dynamometer from the 115 volt D. C. supply. Due to the voltage and motoring limitations, 1500 RPM was the highest speed attained in this manner without greatly exceeding the current limitations on the switchboard and the generator. Horsepower readings were obtained from the torque meter and tachometer data in the usual manner as for output data. From this data a logarithmic plot of horsepower loss vs. engine RPM was made. The result was a reasonably straight line. This, then affords a certain reassurance in extrapolating the data to obtain values of the losses at the higher speeds unattainable. The plot is presented as Fig. 21. From Fig. 21 the slope of the line determines the exponential relationship between power loss and engine RPM's. It is of the form Power Loss = constant X $(RPM)^n$; where n is the slope of the line. In this case n is 1.36. Data for determining mechanical loss is taken from this figure.

Fig. 22 shows the engine characteristic full power curve for test A-3 and the mechanical losses plotted on power vs. RPM coordinates. The mechanical loss curve was taken from Fig. 21; the output curve is repeated from Fig. 14. From these curves, the corresponding mechanical efficiency curve is determined by using the relationship:

$$\eta_m = \frac{BHP}{IHP} = \frac{BHP}{BHP + \text{mech. losses}} \times 100$$

where η_m = mechanical efficiency.

BHP = brake horsepower.

IHP = indicated horsepower.

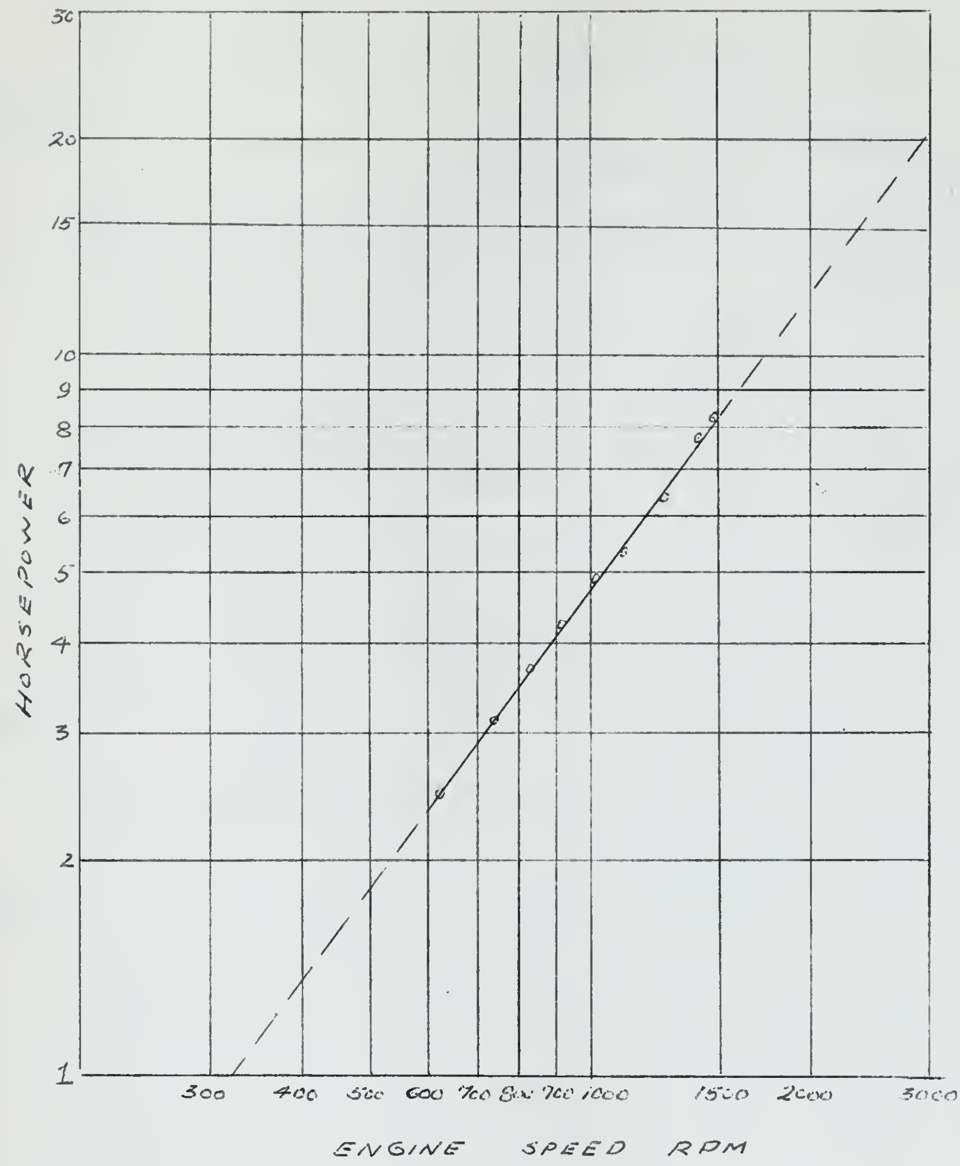


FIG. 21. LOGARITHMIC PLOT OF MECHANICAL LOSS IN HORSEPOWER VS. ENGINE RPM. ENGINE OIL TEMPERATURE 165° F.

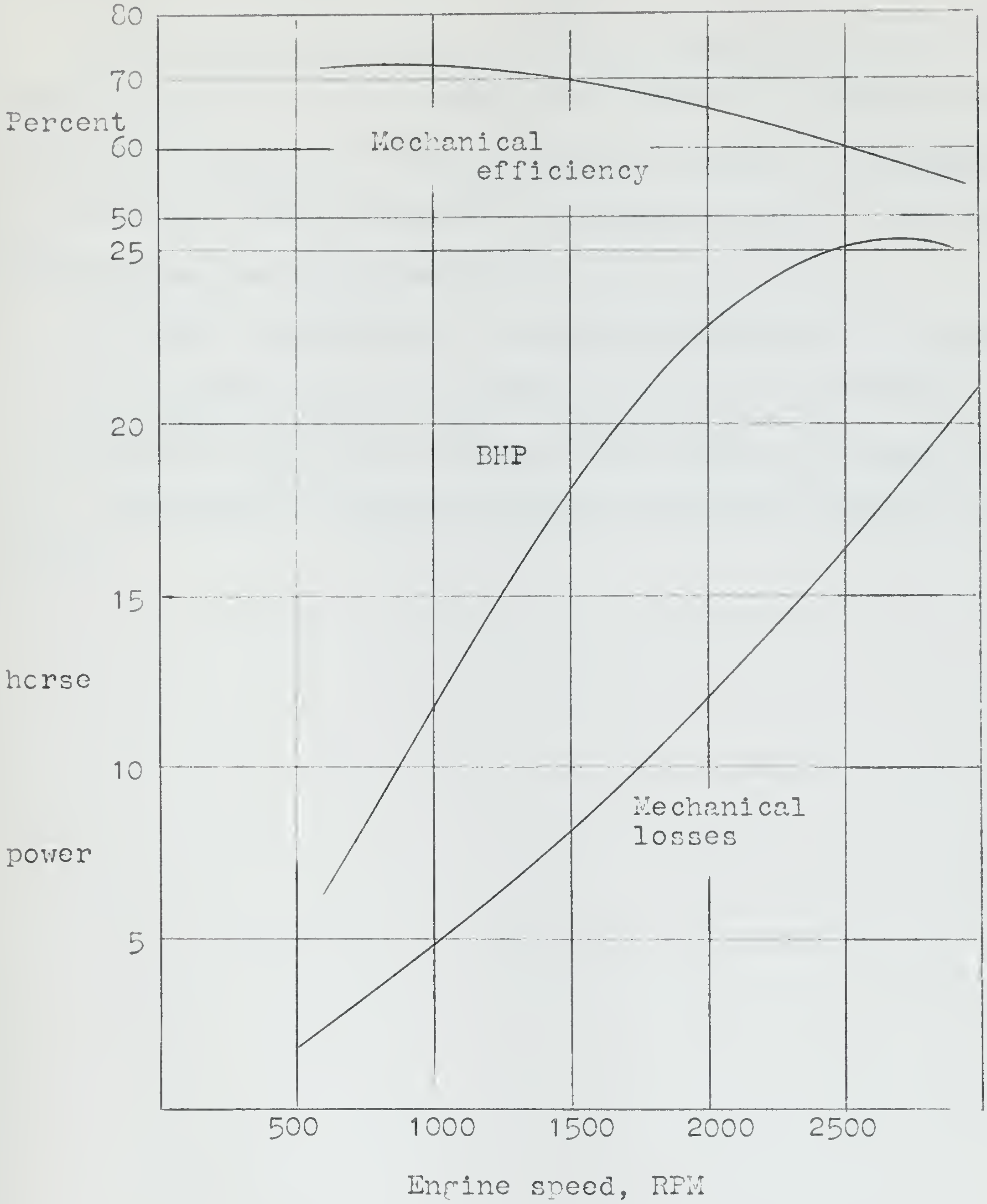


Fig. 22. Mechanical efficiency vs. engine speed at full load; from test A-3

Observe that the best mechanical efficiency occurs at 800 RPM, delivering 8.75 horsepower, and is 72 percent.

Of the timing configurations tested, it appears that the best for operation is A-3. Best performance (most efficient) is obtained in the 1000-2200 RPM range, and this is the range recommended by the manufacturer. The 1-53X3 engine appears to be performing according to the ratings. The advertised data [6] is realistic.

For future investigation of the engines performance, it is suggested that a portable electric or pneumatic motor rig be assembled that will selectively drive the scavenging blower and/or the accessory drive box. In this manner a further breakdown of the power losses may be obtained.

3. Bibliography

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2. C. F. Taylor, The Internal Combustion Engine in Theory and Practice, M.I.T. Technology Press and Wiley and Sons, 1960.

3. S. J. Young and R. W. Pryer, Testing of Internal Combustion Engines, D. Van Nostrand Co., 1937.

4. T. Baumeister, Editor, Mark's Mechanical Engineering Handbook, sixth edition, McGraw-Hill Book Co., Inc., 1958.

5. American Society of Mechanical Engineers, Power Test Codes, Internal Combustion Engines, PTC, 17-1957.

6. Detroit Diesel Engine Division, General Motors Corporation, lithograph number 3 SA 86, "GM Diesel Series 53 Engines".

APPENDIX A

General specifications for single cylinder GM 1-53X3 engine:

ENGINE DESIGN:

Engine type	two stroke cycle
Bore and stroke	3.875 in. X 4.5 in.
Displacement	53.1 cu. in.
Compression ratio	17:1
Engine balance	
Primary balance	2 shafts located in base rotating at engine speed.
Secondary balance	2 shafts located in base rotating at twice engine speed.
Combustion chamber	Open type.
Scavengine system	Uniflow
Fuel injection system	Unit injector
Auxiliary pumps- fuel, water and oil	Accessory box arrangement

BASIC ENGINE PERFORMANCE:

Design horsepower	30 HP at 3000 RPM
Maximum engine speed	3000 RPM
Piston speed, 3000 RPM	2250 ft. per min.
Compression pressure, 3000 RPM	690 psi

MECHANICAL DATA

Piston area	11.8 sq. in.
Max. gas press. used in Calculations	1270 psi
Crank pin bearing press. (max)	2140 psi
Piston bushing press.(max)	3935 psi
Main bearing	
Journal dia.	4.00 in.
Length	1.94 in.
Projected area/bearing	6.16 sq. in.
Material	Steel back - copper lead
Crank pin bearing	
Journal dia.	2.625 in.
Length	1.40 in.
Projected area/bearing	3.675 sq. in.
Material	Steel back - copper lead

APPENDIX A

Camshaft bearing - front	
No. of bushings	2
Diameter, bore	1.49 in.
Length/bushing	1.06 in.
Camshaft bearing - rear	
No. of bushings	2
Diameter, bore	1.49 in.
Length/bushing	0.88 in.
Piston pin bushings	
No. in piston	2
No. in connecting rod	2
Diameter, bore	1.38 in.
Length	0.74 in.
Projected area/bushing	1.02 sq. in.
Piston	
Type	Trunk
Material	Armasteel
Length	5.40 in.
Distance, center of piston pin to top of piston	3.15 in.
Nominal skirt clearance	0.005 in.
Cooling	Oil through rod
Piston ring - fire	
Type	Chrome on steel
No. in piston	1
Width	0.078 in.
Gap	0.020 - 0.040 in.
Distance from top of piston	0.0888 - 0.0988 in.
Piston ring - compression	
Type	Chrome on steel
No. in piston	3
Width	0.093 in.
Gap	0.017 - 0.012 in.
Piston ring oil	
Type	Double scraper
No. in piston	4 (2 per groove)
Location	Bottom of piston
Width	0.093 in.
Gap	0.015 in.
Piston pin	
Type	Full floating
Length	3.240 in.
Diameter	
O.D.	1.375 in.
I.D.	0.88 in.

Connecting rod	
Type	Forged "I" section
Material	Steel
Length, center to center	9.00 in.
Crankshaft	
Material	Steel
No. of counterweights	2
Camshaft - operates 4 exhaust valves	
Material	Steel
Location	Cylinder block
Drive	Gear
Camshaft - operates injector	
Material	Steel
Location	Cylinder block
Drive	Gear
Cylinder block	
Type	Welded
Material	Steel
Liner	
Type	Wet
Material	Cast Iron
Inlet ports	
No.	18
Width	0.50 in.
Height	0.84 in.
Angle	30 degrees
Type	Oval
Timing	
Ports open	57 degrees BBC
Ports close	57 degrees ABC
Exhaust valves	
No.	4 = 1.090 in. dia. 2 = 1.400 in. dia.
Type	Poppet
Operating mechanism	Overhead mechanical, operated in pairs.
Timing	
Valves open	86 degrees BBC
Valves close	52 degrees ABC
Cylinder head	
Material	Cast iron

APPENDIX A

Blower	
Type	Roots (modified 2-71)
Type rotor	3 Lobe helical
Drive	Vee belts
Fuel injector	
Model	71 (HVB)

APPENDIX B

1. Electrical.

The absorption generator of the dynamometer unit is a separately excited shunt generator with a stabilizing series field. It is rated for continuous operation at 240 volts, 105 amps., 1750/3000 RPM with no more than 60 degrees Centigrade rise in temperature. Generator cooling is provided by a separate blower which is driven by a 1/4 H. P. 220/440 volt 3 phase motor attached to the dynamometer foundation. This cooling arrangement exerts no torque on the unit, and is therefore no source of error to the unit. The dynamometer torque sensing element is a Link Engineering Co., "UNIBEAM" pneumatic load cell type, model 302-35.

Electrical connections have been made utilizing existing cables and conduits whereever possible. Fig. 23 is a schematic drawing of the cable runs and their location in the building.

Provision was made for starting the engine by using the generator as a motor and for possible friction motoring tests of the engine. Motoring is accomplished by a simple shift of switches on the control panel. Motoring limitations of the generator restrict the engine mechanical loss tests to those directly determined in the engine speed range of 600 to 1500 RPM.

Fig. 24 is the schematic wiring diagram for the generator-motor system. The variable resistance bank (used alternately as a load grid and starting resistor) is located in the resistance grid house outside the southern wall of building 500. Power for motoring the generator and also for field excitation is taken from the large D. C. generator in the southwest corner of the building. The power is fed to the dynamometer

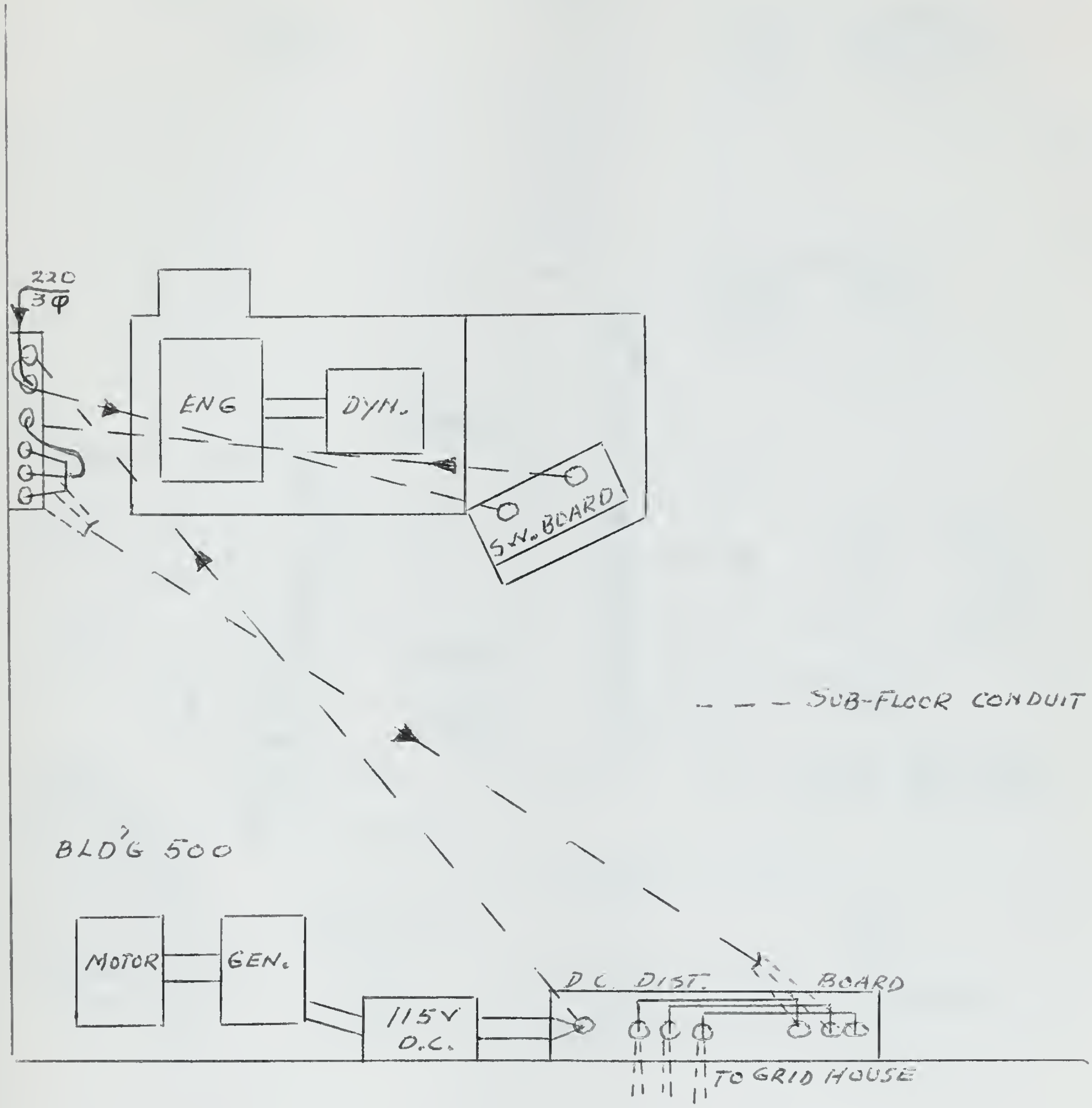
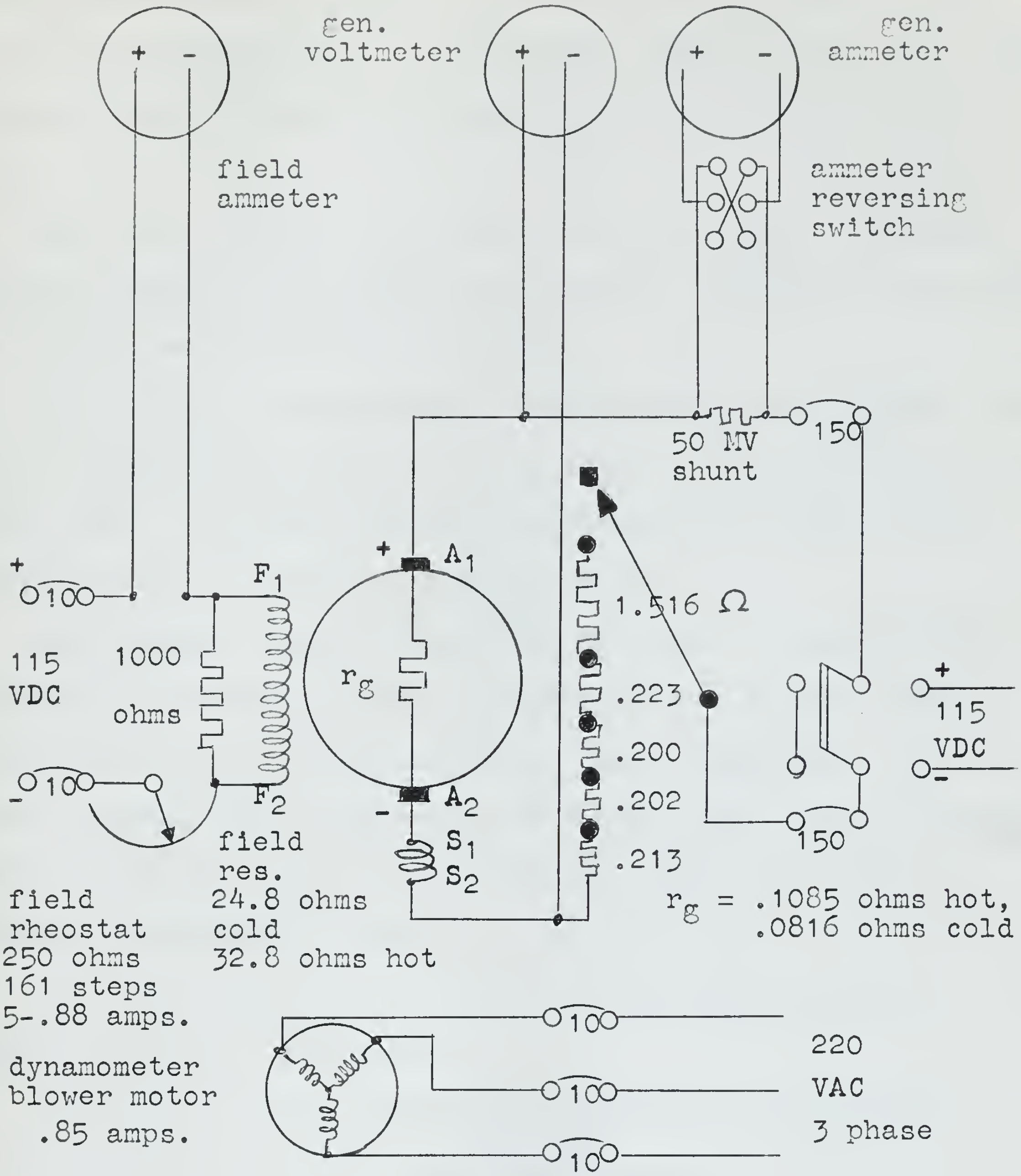


Fig. 23. Electrical power cable runs.



O150O . Circuit breaker/switch, rating indicated.

Fig. 24 . Generator control, schematic diagram.

control panel through a 100 amp. circuit breaker on the main D. C. distribution switchboard. It is circuit breaker number 1. This supplys both starting power, when needed, and field excitation for loading the generator.

Fig. 25 is a sketch of the front of the control panel. All instruments are located for easy reading and operation. The engine can be operated from this control panel by one person.

Fig. 26 is the wiring diagram of the back of the control panel. The rotary resistor selector has been converted from an old starter box contactor plate. It has been modified to withstand the 105 amps of this generator and is completely adequate for the job.

Engine speed control: Provision has been made for controlling the throttle from the control panel by adapting a simple size five synchro transmitter and receiver on the panel and engine respectively. The engine receiver drives the injector rack through a spur pinion and sector gear (see Fig. 6). This can be manually over ridden at the engine by moving the throttle arm by hand.

The various electrical parameters are shown on the electrical figures pertinent to this section.

The engine can also be started by utilizing the 12 volt starter motor attached to the engine frame. This requires a 12V battery or D. C. supply, heavy cabling (No. 2 A.W.G.) and a single pole, single throw light duty switch to operate the starter solenoid.

Fig. 27 is a wiring diagram for this method of starting.

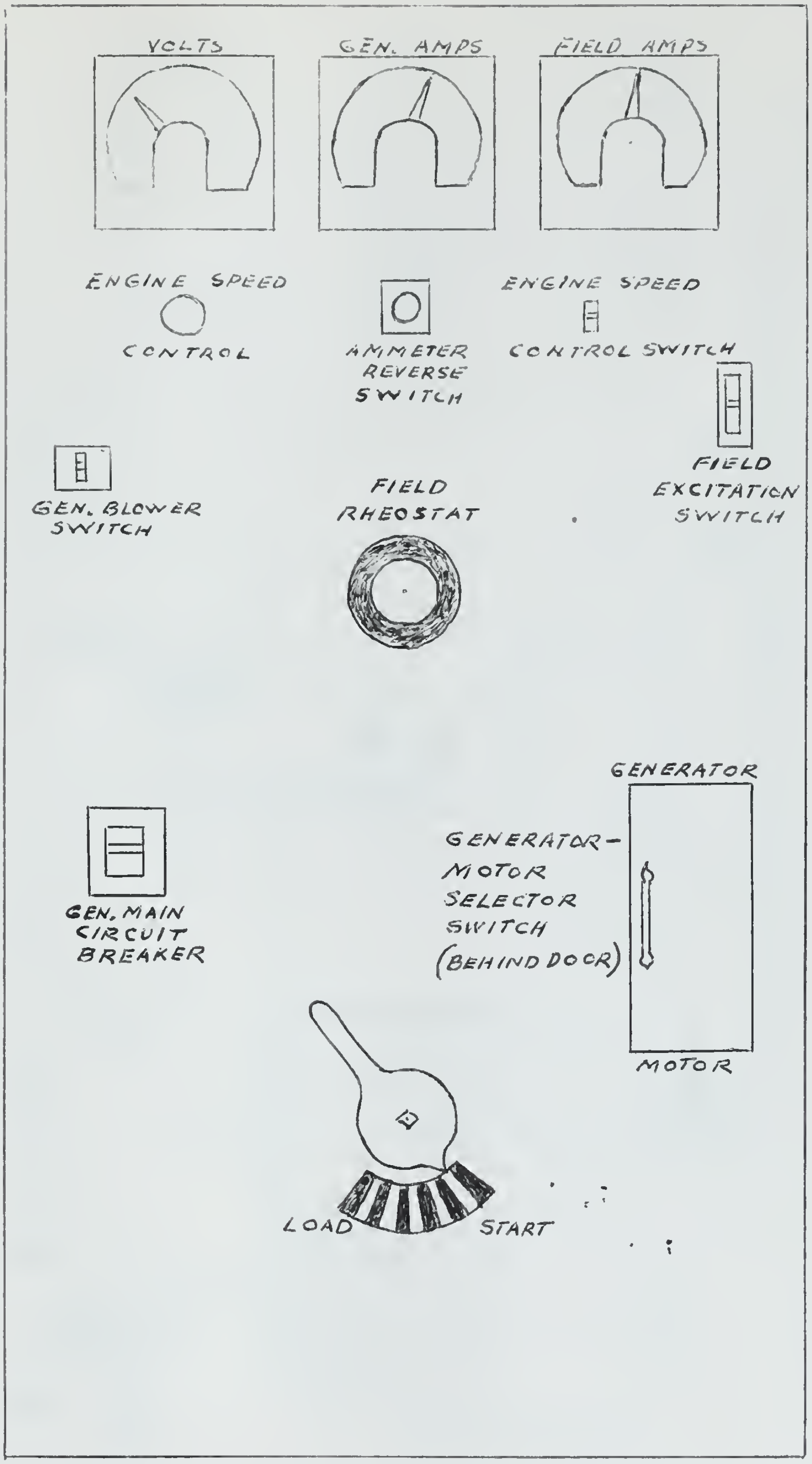


Fig. 25. Electrical switchboard

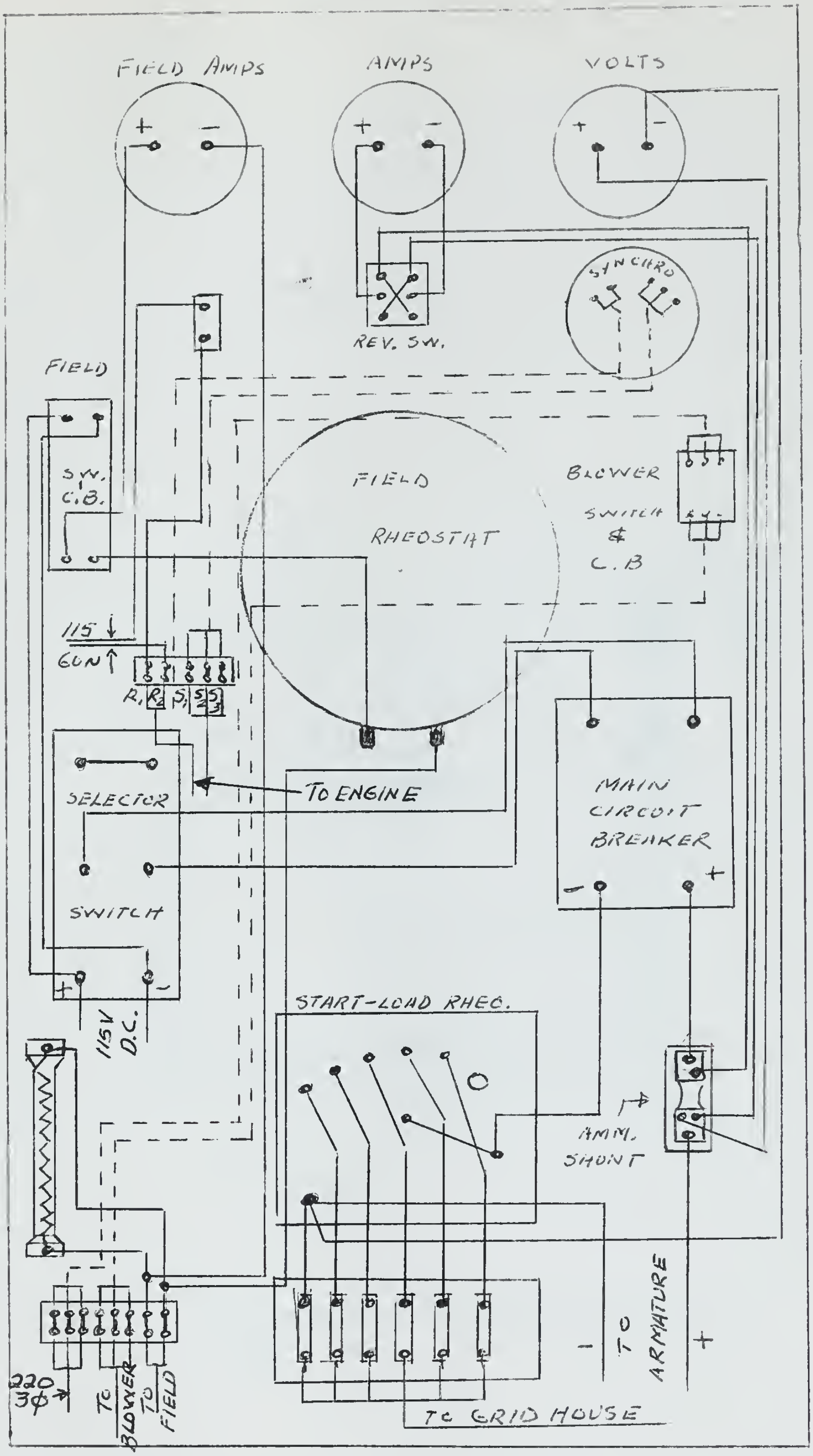


Fig. 26. Electrical switchboard wiring diagram.

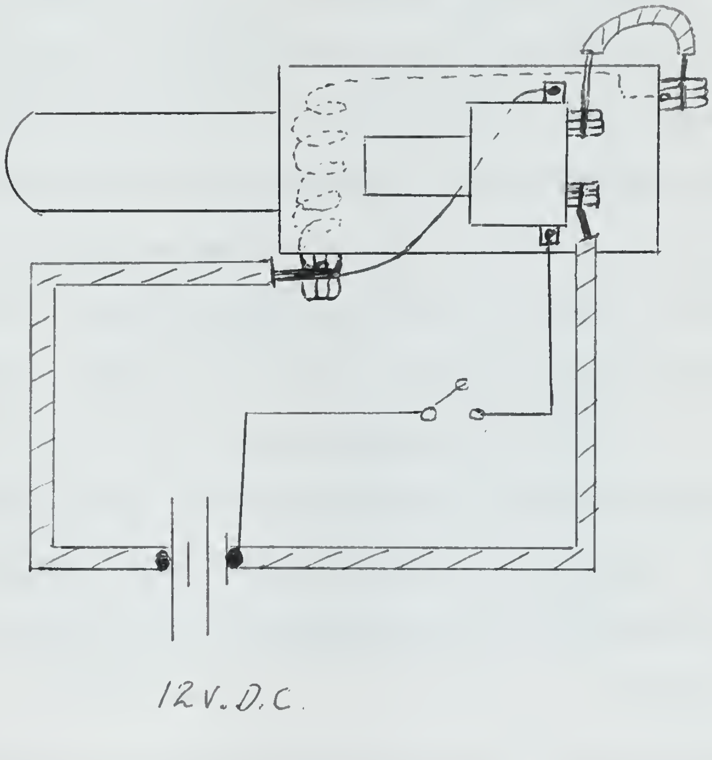


Fig. 27. 12 volt starter, electrical wiring diagram

APPENDIX B

2. Fuel oil system

Fig. 29 is a schematic diagram of the fuel system as installed. The fuel leaves the weigh tank and goes through a strainer to the main stop valve to the G.M. 1-53X3. With the main stop open and fuel manifold supply valve open fuel goes to the fuel pump, then through the filter to the engine unit injector and back to the manifold via the engine spill line. With the burette stop open the fuel will also back up through the burette, out the vent, up to the level of the weigh tank.

To start a metered run it is only necessary to open the burette stop valve and then shut off the fuel tank stop. Both valves are on the manifold. The fuel flow can then be timed from one point in the gauge glass to another. The combined cross-section of the glass and copper pipe is 1.08 sq. in. Upon completion of the timed runs it is only necessary to open the manifold valve from the supply tank. This will refill the burette while still supplying the engine. If the burette is still sufficiently full shut off the burette stop until the next run.

Fuel Burette data: The fuel measuring burette was made after considering a 5 minute student laboratory run at full power. Based on previous data the specific fuel consumption was 0.6 lbs fuel/HP-hr at a sp. gr. of 0.844 or 0.03045 lbs/cu. in. At 30 H.P. for 5/60 hour it will use .6x30x1/12 equal to 1.5 lbs of fuel; therefore, the burette was made of copper pipe 1.06 inches I.D. 80 inches long with a 60 inch sight glass, 1/2 inch I.D. This gives a total cross sectional area of 1.08 sq. in. which will give a drop of 50 inches along the sight glass for a 5 minute full power run.

APPENDIX B

The attached fuel pump is a positive displacement gear type pump which operates at about 30-70 p.s.i. depending on engine RPM. It is important to note that the oil seal rings on the fuel oil pump shaft have deliberately been installed backwards to prevent fuel oil from leaking into the lube oil sump.

Fig. 28 is a photograph of the fuel metering manifold and a portion of the burette. The valve to the left is the fuel tank supply; the valve to the right is the burette supply.

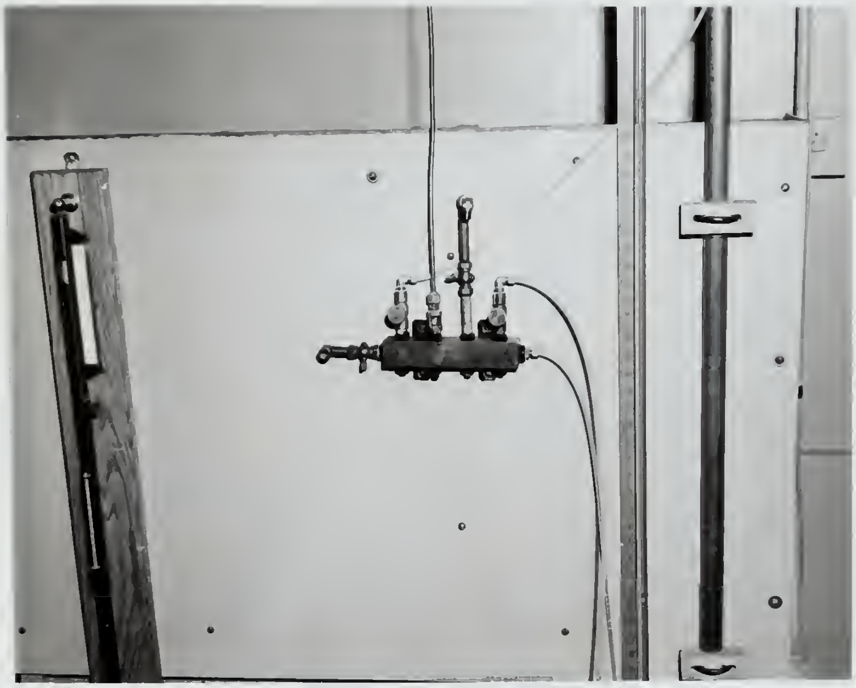


Fig. 28. Fuel metering manifold and burette

APPENDIX B

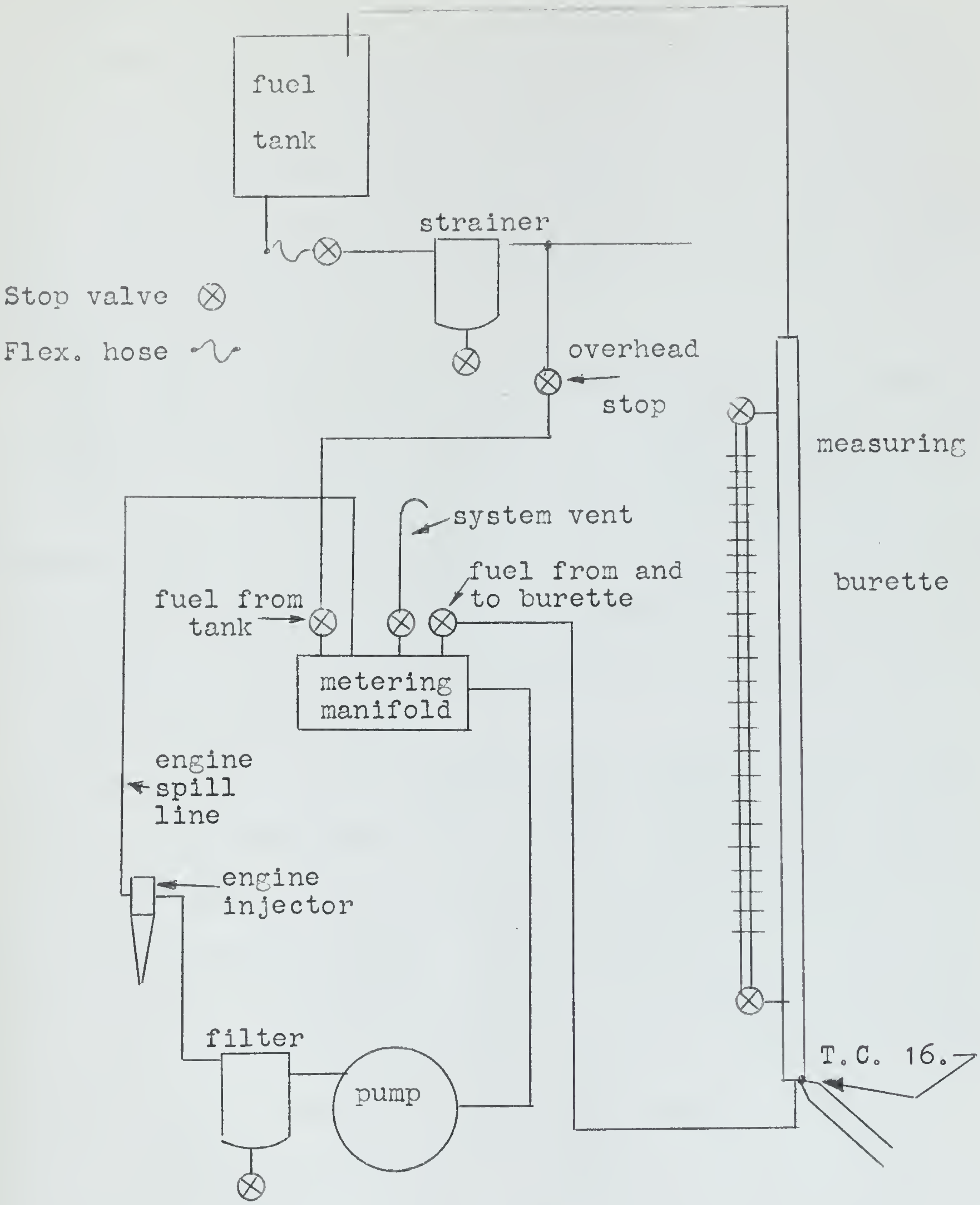


Fig. 29. Schematic diagram of fuel system.

3. Circulating water system.

The engine cooling water system consists of a closed loop engine system with a small vent and expansion tank mounted above the engine, and a cold water cooling system where heat is removed through a small heat exchanger located just before the engine circulating water pump. Provision has been made for using direct cooling by laboratory cold water should the circulating pump fail.

In testing a single cylinder engine of this small size the operating temperatures can have a great influence on the power output, and for this reason, the water temperatures must be regulated carefully to obtain the consistent results desired in a laboratory test. Manually operated valves are installed in both the cold water and hot water systems to obtain the desired operating temperature and temperature rise between inlet and outlet water of the cooling jacket.

The water pump is a small centrifugal pump capable of about 5 gals per min at a top engine speed. It is driven from the accessory drive box through bevel gears and a four pronged fork type flexible coupling. The pump discharges through a flexible vibration absorbing hose to the flow regulating valve and then to the flowmeter. The flowmeter has been calibrated and the calibration curve is shown as Fig. 30. From the flowmeter the water passes to a Weston direct reading thermometer which must be used to correct for temperature when determining cooling water flow. From the thermometer it enters the engine block. Thermocouples record the temperature in and out of the engine. A thermal bulb distant reading temperature gauge has been installed at the engine outlet for quick use

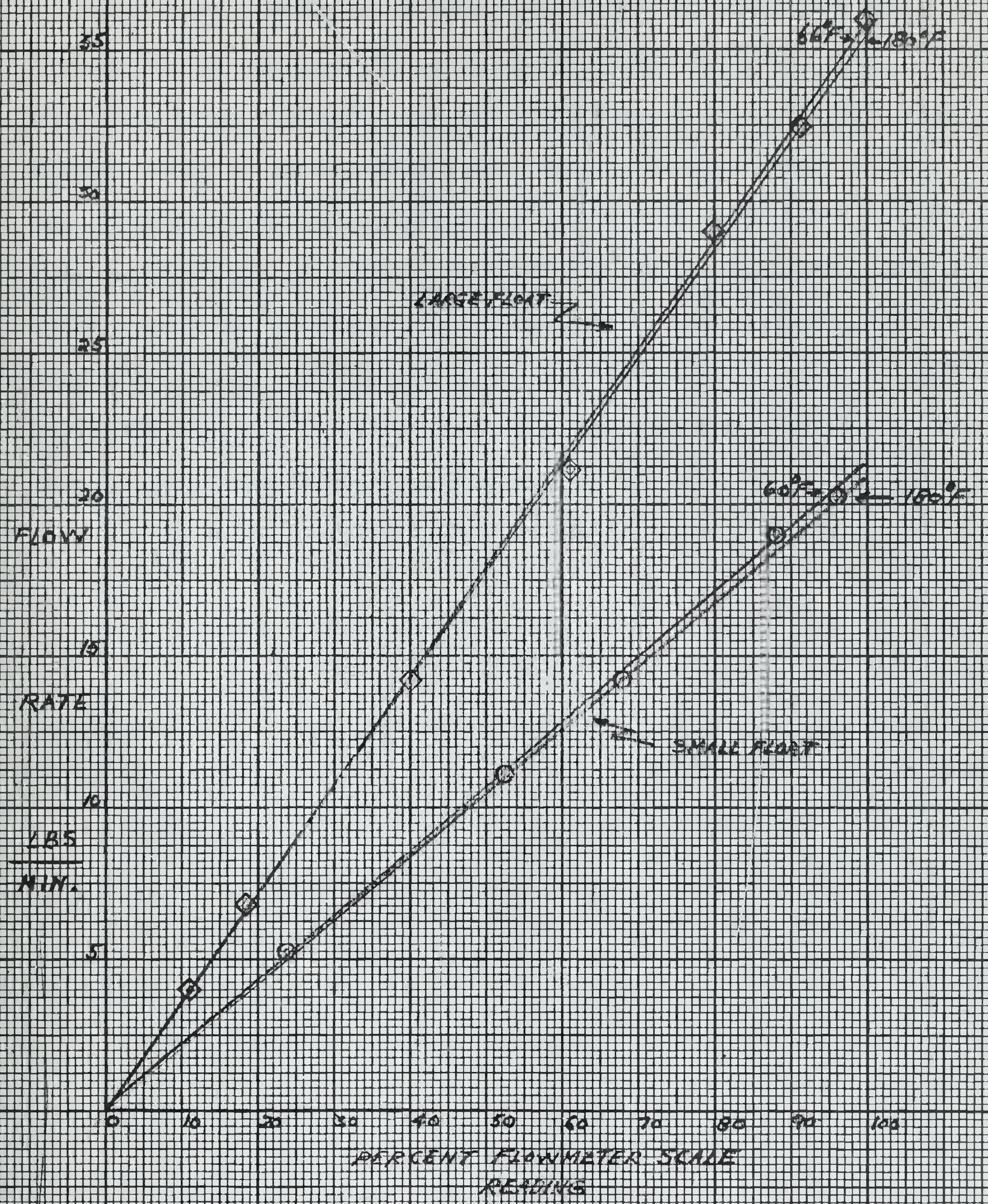


Fig. 30 SHOWING CIRCULATING FRESH WATER MASS FLOW RATE VS. SCALE READING OF FLOWMETER

of operating personnel when the recorder is not being used. The outlet water flows back to the suction side of the pump after passing through the heat exchanger where it can be cooled as needed. The connection to the expansion tank and a discharge to waste are made at the high point in the discharge line. The discharge to waste is made via a stop valve and through a gravity type orifice meter which can measure the cooling water flow should direct cooling be necessary. This direct cooling by laboratory water should not be used because, in addition to providing overcooling with its poor engine performance, it offers the engine too great a thermal shock when the cold water strikes the wet type cylinder liner employed in this engine.

Fig. 31 is a shematic diagram of the water cooling system.

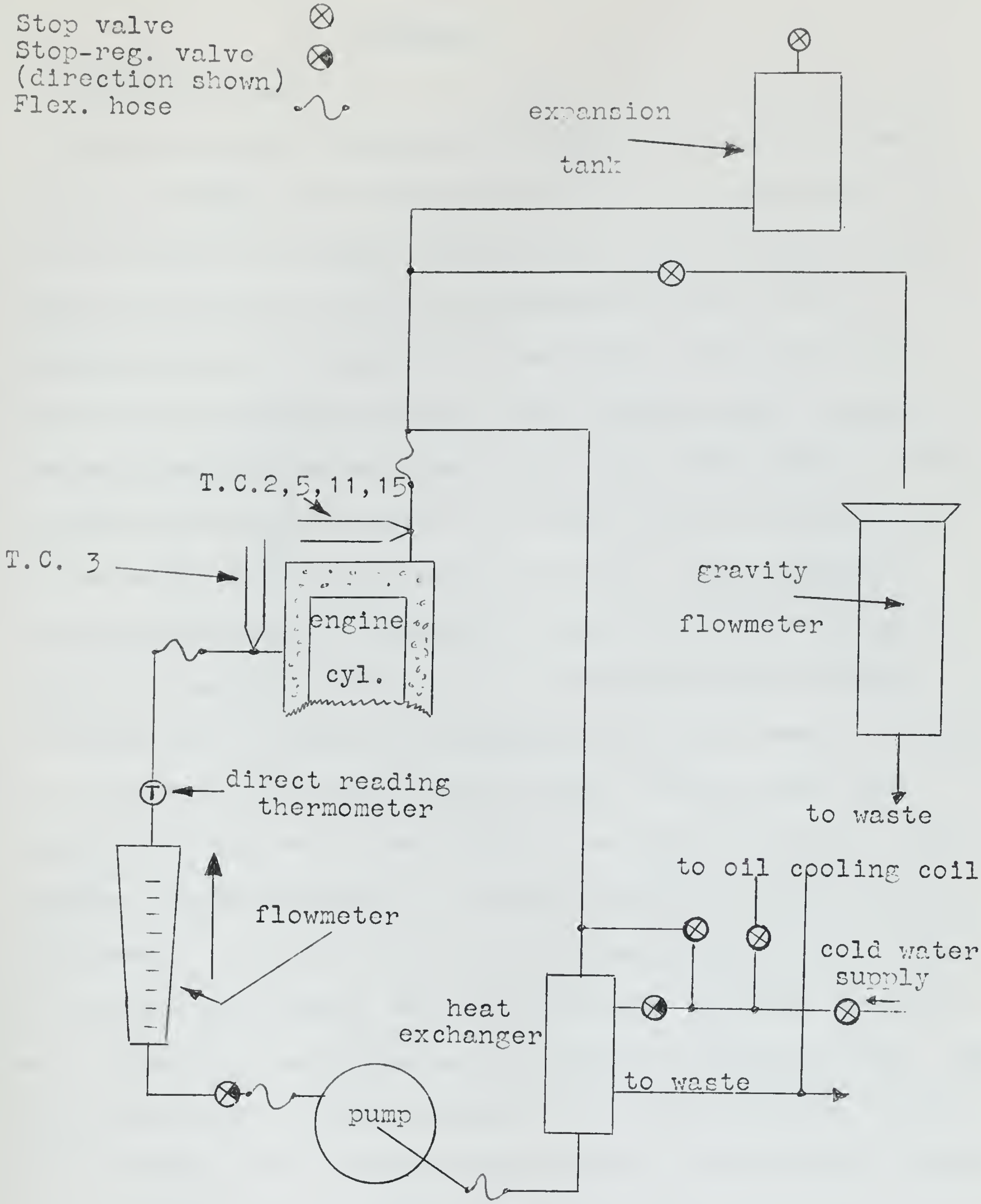


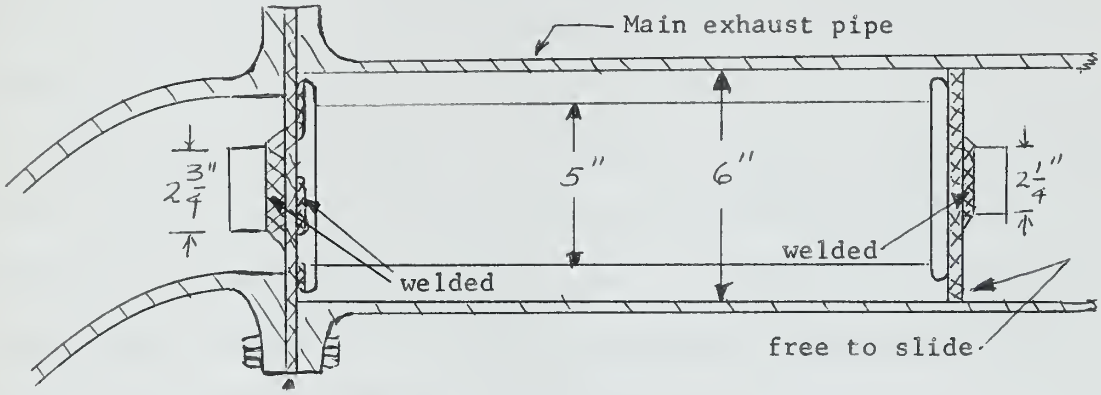
Fig. 31. Schematic diagram of circulating water system

4 Engine air system

Air is supplied to the engine by the positive displacement three lobe helical blower which maintains up to four p.s.i. pressure in the air box section of the engine. Entry is made into the engine cylinder through 18 inlet ports which are oval shaped and direct the air in a peripheral motion at 30 degrees from the normal to the cylinder wall. These ports open 57 degrees BBC and close 57 degrees ABC. The gases are exhausted through 4 overhead exhaust valves each 1.090 inches in diameter. The valves are operated in pairs by two rocker arms and push rods from one cam on the accessory box side of the engine. With timing marks aligned in the engine the exhaust valves open at 88 degrees BBC and close at 52 degrees ABC (See Fig. 11). The exhaust is piped through a flexible steel corrugated and insulated pipe to the atmosphere outside. Air is supplied to the blower intake through a 100 gal surge chamber which has 1/8 inch thick rubber diaphrams installed at each end. These diaphrams have been stretched to a natural frequency of 2700 cycles/min approximately. This will provide good anti-surge characteristics to the air flow measuring nozzles, and there has been no difficulty with resonance at 2700 RPM. Two nozzles are installed for measuring air flow. They are aluminum ASME long radius nozzles of 1.375 inches diameter and 2.00 inches diameter. One is plugged while the other is being used or both are used together. The surge chamber completely removes perceptable oscillation of the vernier manometer. The manometer is a standard Ellison draft vernier gauge. The connection to the engine blower inlet is made via a length of smooth bore flexible rubber suction hose of 4 inch diameter.

APPENDIX B

During high speed tests of the engine it was determined that a low back pressure muffler would have to be installed because of excessive engine noise. Since time had become of essence in completing the tests a rather rapid solution was adopted to the noise problem. A Maremont IH24 steel muffler was purchased and installed as shown in Fig. 32.



steel annular ring welded to muffler, gasketed and bolted between flanges.

Fig. 32

This installation utilizes the large existing exhaust pipe into which the muffler was inserted and sealed at the elbow flange by welding a 1/16 inch circular steel disk to the front end of the muffler. This steel disk was then clamped between the flange with gaskets and the regular flange bolts thus forcing the exhaust gases to leave through the muffler.

5. Gauge board instrumentation.

The master instrument board contains all the necessary instrumentation to conduct performance tests and operate the engine safely. The board was assembled by utilizing a spare aluminum framed chalk board as the panel into which the instruments are installed. This provides a convenient place to jot down information while providing an easy to see array of instruments. Fig. 33 shows the general arrangement of instruments.

The thermocouple temperature recorder is a Leeds Northrup "Speedomax" unit, type G, model S, 16 channel potentiometer type, calibrated for iron-constantan thermocouples. It has a range of 0-800 degrees Fahrenheit. This range is slightly low to record the exhaust temperature at high power outputs but is adequate for all other temperature ranges.

The tachometer is a Beckman electronic EPUT meter that works in conjunction with a magnetic pickup and a 60 tooth gear on the dynamometer to determine and display at rapid intervals the revolutions per minute with good accuracy. There is a separate instruction book available for describing the modes of operation of this instrument.

The torque meter is calibrated for the Link unibeam dynamometer model 302-35. This meter has been manufactured with a dial zero value of preload equal to 4.25 psi to compensate for the heavier portion of the left side of the unibeam arm which rests on the load cell. Full scale torque reading corresponds to 60 psi pressure in the indicator. This dial type indicator is contained in a pressure tight case and is therefore free from changes of ambient pressure. A separate instruction book has been

provided by the manufacturer for the complete dynamometer set.

Manometers are provided for measuring the exhaust back pressure in inches of water and the air box pressure in inches of mercury.

The air intake depression is measured with an Ellison vernier type manometer calibrated in inches of water and using the special Ellison red oil as a fluid.

A lube oil pressure gauge records the lube oil pressure in psi as it enters the main gallery to the engine block. A fuel oil pressure gage measures the diesel oil pressure as it enters the fuel oil jumper to the injector.

A direct reading water temperature gauge mounted above the pressure gauges is useful for operation without the temperature recorder in use.

Representative ranges of these instruments are shown on Fig. 33.

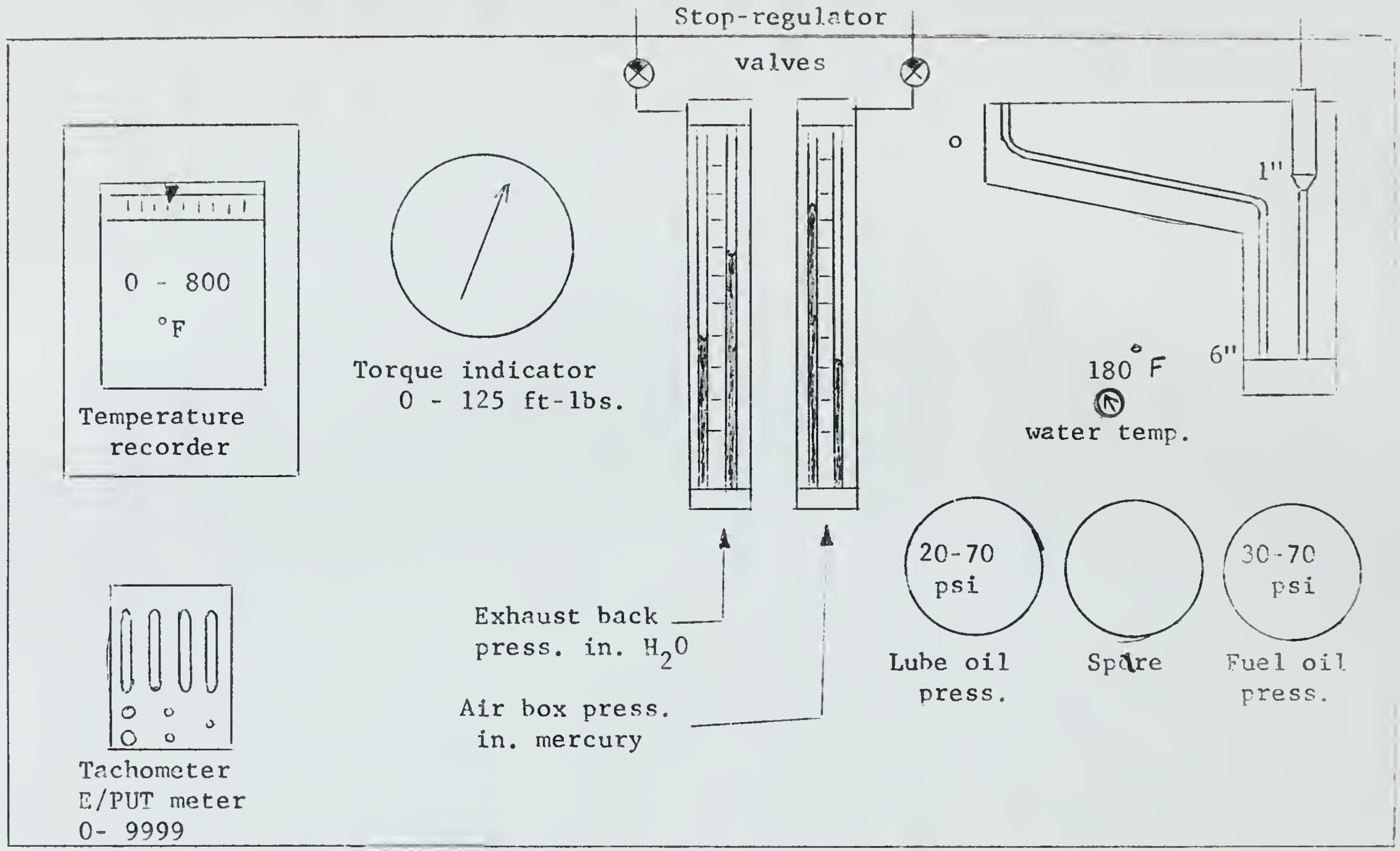


Fig. 33 Showing the instrumentation on the gaugeboard.

6. Foundation.

The foundation is a reinforced concrete block poured into the old pit of an unused engine vibration isolated pad. The new foundation was raised 6 inches above floor level to afford easier accessibility. The engine is secured and leveled directly to the concrete bed where it is held by four 1 inch bolts which were positioned and supplied with long hold down hooks before pouring the concrete. The bolts are located at each corner of the engine and are secured with a two nut locking combination.

To support the dynamometer and to allow for easy shifting, in order to align the coupling, two longitudinal rails were made from four 2x6 standard channel beams. Each rail was formed by welding one quarter inch steel plates across each end to form a box beam with longitudinal slots. The two box rails were spaced by welding 1x1/4 inch steel bars across each end. This will relieve any tendency to spread or close. This assembly was then set on jack screws and leveled. The concrete was then poured to within 1 inch of the top of the rails. The hold down bolts are 3/4 inch bolts with flat large heads welded in place. These are slid under the rail upper flanges. When the nuts are tightened these large heads clamp under the flanges of the rails and prevent movement.

The large slab upon which this foundation was built is surrounded and isolated from the main floor by a 2 inch wall of cork.

Fig. 34 and 35 are photographs taken prior to pouring the concrete.

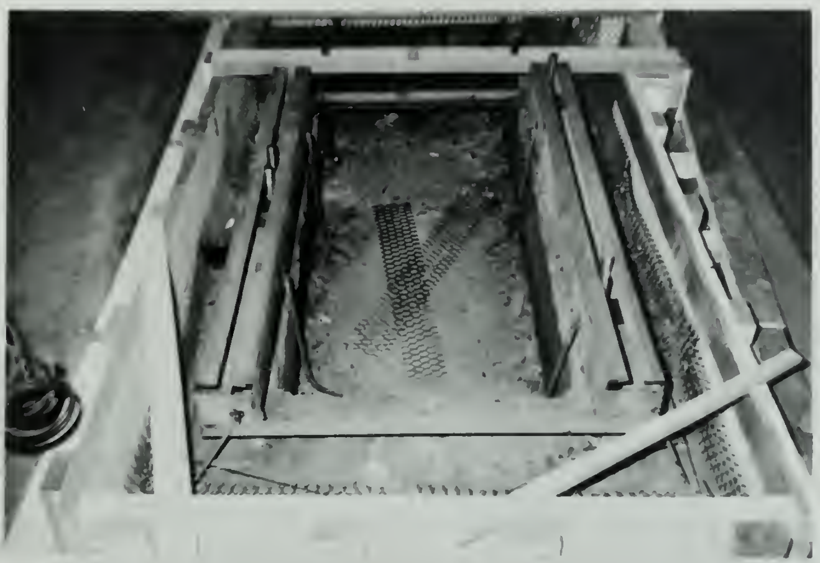


Fig. 34. Looking into pit from East side.

Fig. 35 View from Southeast corner.

7. Engine to dynamometer coupling.

The flexible coupling utilized in this installation is a machined steel type 226 SN coupling manufactured by the Thomas Flexible Coupling Co. of Warren, Pennsylvania, especially for this installation. It consists of a special machined steel flywheel adapter, a machined and hollow floating shaft center section, a keyed flange and two stacks of stainless steel laminated rings. The rings act as the flexing units and through (8) eight specially fitted allen head cap screws the units are connected together to form a double flexing coupling. The flywheel adapter was incorrectly furnished with 4 equally spaced one half inch holes for bolting to the flywheel; therefore, it became necessary to drill holes to fit a six bolt flywheel. Fig. 37 is a detailed sketch of the coupling assembly. The dynamometer flange fits the dynamometer shaft with a medium interference fit.

Alignment; Fig. 36 shows the recommended maximum tolerances in aligning this coupling.

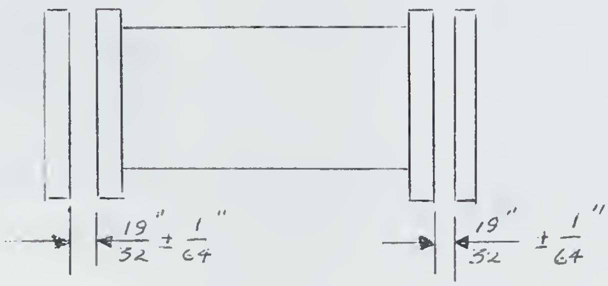


Fig. 36

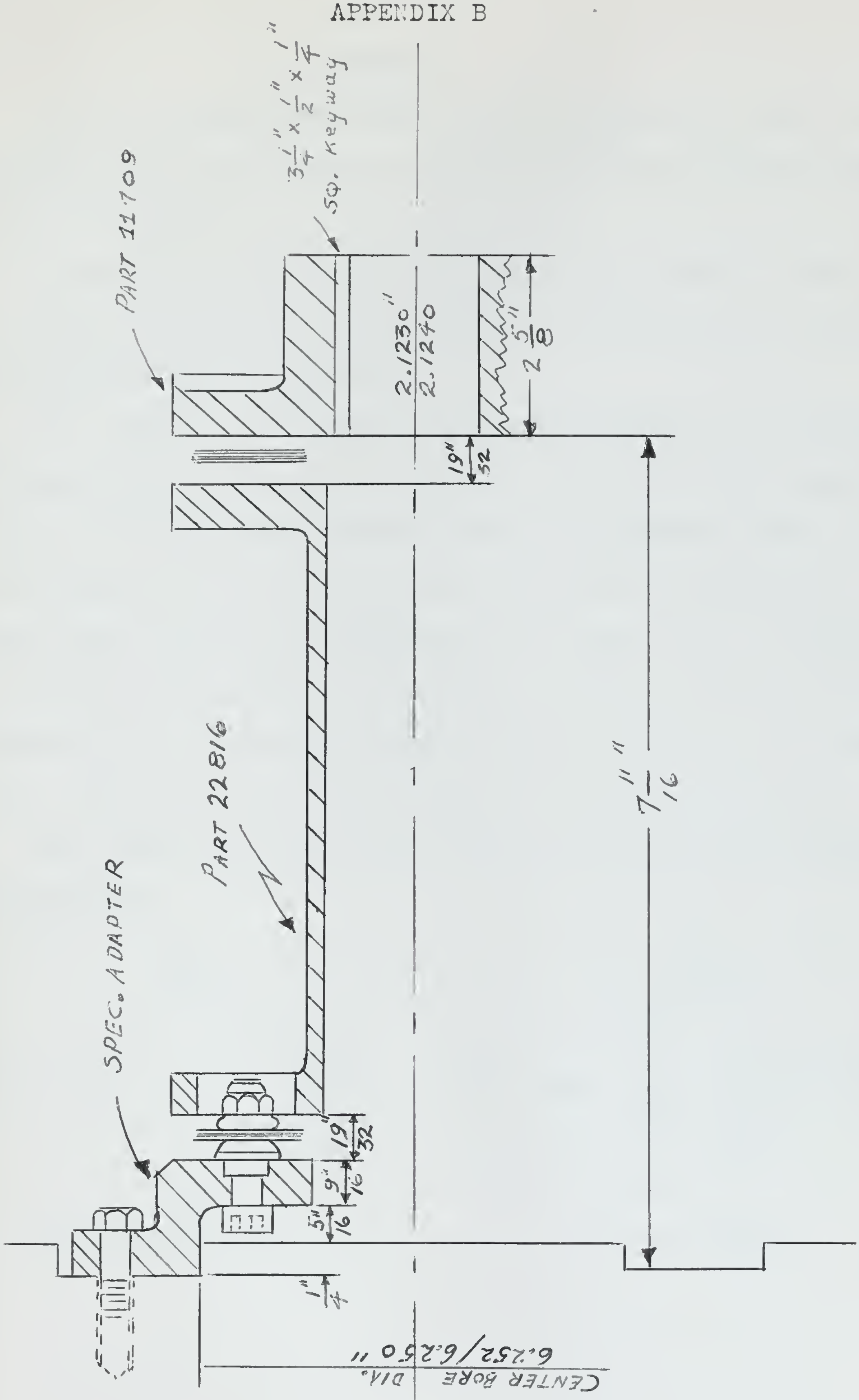


Fig. 37. DETAILS OF FLEXIBLE COUPLING

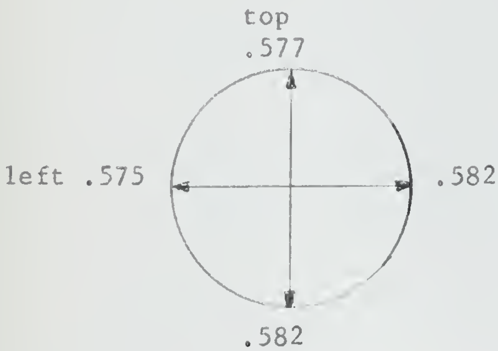
APPENDIX B

If the coupling is installed to meet the requirements of Fig. 36, it should provide good service in absorbing any future slight misalignments and end float.

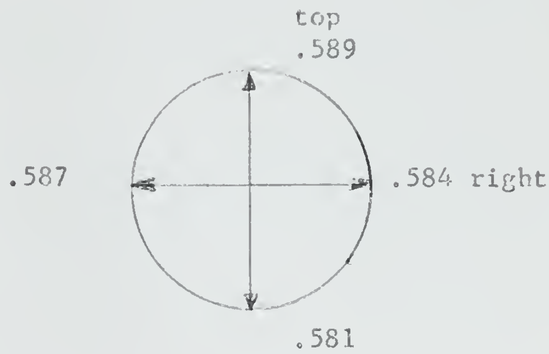
It should be noted at this point that this type coupling, while offering good flexibility, also offers somewhat of a tedious task in getting the flanges within the tolerances.

The alignment is performed by moving the dynamometer into position and tightening all bolts of the coupling. Set the gap between flanges as close to 19/32 inches as possible by moving the dynamometer frame. Tighten the foundation bolts and with a 1/2 inch inside T micrometer take readings between flanges at 4 cardinal points (90 degrees apart-8 readings). The readings should differ by no more than .012 inch. If they do the dynamometer must be loosened, raised and shimmed in the necessary direction to remove the error. It is a tedious, cut and try proposition.

Upon completion of the initial installation the following readings were obtained:



Engine end flanges



Dyn. end flanges

These are considered to indicate good alignment. Subsequent running tests of the unit showed no signs whatever of any heating in the flexible disks, which would have shown unacceptable flexing had it occurred.

APPENDIX C

GM MODEL 1-53X3 DIESEL ENGINE OPERATING INSTRUCTIONS

Preparation:

1. Open fuel stop valve, overhead.
2. Open air to dynamometer stop valve.
3. Turn on tachometer, set to E/UT, 1 second interval; turn on "Speedomax" temperature recorder.
4. Check switches on electrical panel OFF. ENGINE SPEED CONTROL - on.
5. Start D. C. M. G. set (50KW). Energize D. C. distribution board 115 volts. Turn on circuit breaker #1.
6. Set engine throttle about 2/3 open.
7. Lock dynamometer with lever.

Start:

1. Set ammeter reversing switch to IN.
2. Turn all switches on electrical board ON.
3. Set full field on the generator.
4. Place GENERATOR-MOTOR knife switch to MOTOR position.
5. Move START-LOAD RHEOSTAT lever to second position. Watch the ammeter and the oil pressure. Adjust the throttle until engine starts.
6. When ammeter drops to around 20 amps. and engine is running, rapidly push START-LOAD RHEOSTAT lever to the first or OFF position. Set minimum field on the generator; shift GENERATOR-MOTOR knife switch to the GENERATOR position.
7. Place ammeter reversing switch to OUT position.

To load:

1. "UNLOCK" the dynamometer.
2. Put START-LOAD RHEOSTAT lever to the second position. The engine is now lightly loaded.
3. Adjust load with field rheostat and throttle. Use START-LOAD

RHEOSTAT to load only at lower powers.

4. When engine reaches operating temperature of 180 degrees F turn on cooling water to heat exchanger;adjust flow as required.

5. Check lube oil pressure up; at least 20 psig.

APPENDIX D

SAFETY PRECAUTIONS

1. Do not exceed 3000 RPM.

2. Maintain lube oil pressure; minimum: 20 psig.

3. Do not interrupt electrical circuit while under load in excess of 50 amperes current except by main circuit breaker.

4. Do not stand near flywheel and coupling while engine is turning.

5. Belt accessory drive is protected by a screened enclosure, but remember, this belt drive is dangerous.

6. Do not move the START-LOAD RHEOSTAT to the third position except at the proper reduced load. Most of the resistance is in the second position. (See Fig. 24, page 43.)

APPENDIX E

U.S. Naval Postgraduate School
Dept. of Mechanical Engineering
Diesel Engine Test Sheet

1. Data sheet
GM 1-53X3 1 cyl.
bore: 3-7/8 stroke: 4½ fuel: lab diesel Sheet No. 1
comp. ratio: 17:1 sp. gr. .830 @ 60 Test: A-3
lub oil NS 9250 Date: 14 April 1960

Remarks: Fuel injection as in test A-2; exhaust valve popping time delayed 4.6 degrees.

1		Total hrs. 23.5 run	1	2	3	4	5	6
2		Actual RPM	608	1014	1502	2010	2514	2905
3		Dur. of run secs.	29.6	35.5	26.6	19.8	16.8	14.3
4		Corr. torque	54.8	57.5	63.6	58.0	53.2	45.1
5		Corr. BHP	6.35	11.1	18.22	22.2	25.4	25.0
6		Burette temp. °F	71	71	72	74	76	74
7	F	Corr. sp. gr.	.828	----	----	----	----	----
8		Obs. fuel/cycle in	must be calculated					
9	U	Fuel press. in	56	70	68	70	71	74
10		Corr. fuel/cycle in	.0322	.0645	.0645	.0645	.0645	.0645
11	E	Net lbs/cycle	3.92	6.54	8.71	11.7	13.8	16.4
12		Corr. lbs/BHP-hr	.617	.588	.478	.527	.544	.656
13	L							
14								
15	W	Pump press. psig	---	----	----	----	----	----
16	A	Temp. in °F	134	165	155	152	159	150
17	T	Temp. out °F	174	186	176	175	184	180
18	E	Flowmeter %	35	65.5	87	91	95	94
19	R	Flowmeter Temp.	134	164	152	151	158	148
20	O	Pump press. psig.	28	50	57	62	66	65
21	I	Sump temp. °F	150	156	166	174	184	195
22	L							
23								
24		Ch'ging press. in. hg.	1.62	2.85	4.1	6.7	9.5	13.1
25	A	Int'ke dep. in. H_2O	.322	.225	.568	.488	.772	2.20
26		Air box temp. °F	134	138	140	146	158	170
27	I	Intake temp. °F	78	79	80	80	80	80
28		W/B temp. °F	63	----	----	----	----	----
29	R	D/B temp. °F	70	----	----	----	----	----
30		Barometer in. hg.	30.26	----	----	----	----	----
31		Air flow	103	184.5	291	396.9	500	582
32		Man. corr. + 0.01	.332	.235	.578	.498	.782	2.21
33	E	Exhaust temp. °F	570	700	790	800+	800+	800+
34	X	Exh. back press. in. H_2O	2.6	2.5	6.9	10.8	14.2	16
35								
36	H.							
37		Nozzle in use	1	2	2	1&2	1&2	2
38								
39		Start- 1200. Stop- 1410						
40		Hours to date: 25.5						

2. Calculations of brake horsepower and brake mean effective pressure: The mathematical expression connecting engine torque and brake horsepower is

$$BHP = \frac{2\pi NT}{33{,}000}, \text{ where } T = \text{torque in ft. lbs}$$
$$N = RPM$$

The brake mean effective pressure (BMEP) is that part of the mean gas pressure available to do external work at the engine shaft. The indicated mean effective pressure (IMEP) is the constant mean pressure, which when acting through the power stroke, would do the same amount of work as the actual varying pressure; therefore, the BMEP is less than the IMEP by an amount equal to that necessary to overcome the friction and pumping losses. The ratio of BHP to IHP is called mechanical efficiency and is denoted here as η_m. IMEP is denoted by P, then BMEP is $\eta_m P$.

$$BHP = \frac{\eta_m PLAN}{33{,}000}$$

for a single cylinder engine (2 cycle). Where :

$\eta_m P$ is BMEP in lbs/sq. in.

L is length of stroke in feet.

A is piston area in sq. inches.

N is RPM.

For this engine this can be simplified to:

$$BHP = \frac{\eta_m P \times N}{\text{Constant}} \quad \text{or} \quad \eta_m P = \frac{BHP \times \text{Constant}}{N}$$

APPENDIX E

Since $BHP = \frac{TN}{\text{constant}}$; then $\eta_m P = T \times \text{constant}$

For this engine the BMEP (η_m P) constant is calculated to be 1.420 to give η_m P in psi. It is derived from $\frac{2\pi}{LA}$ where L = 0.375 ft. for a 4.5 in. stroke and A = 11.8 sq. in. for a 3.875 bore.

Sample calculation: Data taken from the sample data sheet, test A-3, run 5.

Torque = T = 53.2 ft.-lbs. $BHP = \frac{2\pi NT}{33,000}$

$$BHP = \frac{53.2 \times 2514 \times 2\pi}{33,000} = 25.4 \text{h.p.}$$

$$\text{Correction factor} = \frac{29.92}{30.26}\sqrt{\frac{530}{520}} = 1.00$$

Standard BHP = 25.4 X 1 = 25.4 h.p.

BMEP = T X 1.420 = 53.2 X 1.42 = 75.5 psi

3. Air flow nozzle calculations:

For standard ASME long radius nozzles, as used in this installation, the flow can be calculated and constants determined for the equipment assuming standard conditions that should prevail in this building. The conditions should be about 70 degrees Fahrenheit, 60 percent relative humidity and we will use a standard atmosphere of 29.92 inches of Hg. equal to 14.69 p.s.i.

$$p_{ws} = 0.363 \text{ (saturation)}, \quad p_w = .6 \times .363 \approx .218$$

$$p_t = 14.7, \quad p_{air} = 14.7 - .218 = 14.482 \text{ p.s.i.}$$

$$\text{Density}_{air} = \frac{14.482 \times 144}{53.3 \times 530} = .0756 = \rho_a$$

$$\text{Density}_{water} = \frac{0.218 \times 144}{86 \times 530} = .00069 = \rho_w$$

$$\text{Density}_{mixture} = \rho_m = 0.0763 \text{ lbm/cu. ft.}$$

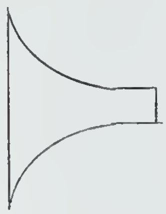
Where p_t and p_w represent pressures, t is for total and w is for water vapor, m is for mixture.

For nozzles used in this manner the upstream velocity is effectively equal to zero.

$V_o = 0$

$p_o = $ atm.

$A_o \approx \infty$



$V_1 = ?$

$p_1 \approx ?$

$A_1 = 0.0103$ sq. ft.

Ideally: $\frac{V_o^2}{2g} + Z_o + \frac{p_o}{\rho_c} = \frac{V_1^2}{2g} + Z_1 + \frac{p_1}{\rho_1}$

and $Z_o = Z_1$, therefore $V_1^2 = 2g\left(\frac{p_c}{\rho_o} - \frac{p_1}{\rho_1}\right)$

Assumption: Incompressible flow applies for range of flows involved.

That is, $\rho_i = \rho_c$

$$V_1^2 = \frac{2g \Delta p}{\rho_c} \; ; \; V_1 = \sqrt{\frac{2g \Delta p}{\rho_c}} \; ; \; \frac{\Delta p}{\rho_c} = \frac{R}{12}\left(\frac{\rho_G - \rho_o}{\rho_c}\right)$$

ρ_G is the density of the gauge fluid = 62.4 lb/cu. ft.

R is the gauge reading in inches.

$$\frac{\Delta p}{\rho_c} = \frac{R}{12}\left(\frac{62.4 - .0763}{.0763}\right) = \frac{R}{12}\left(\frac{62.4}{.0763} - 1\right)$$

$$= \frac{R}{12}(817 - 1) = 68R;$$ therefore without the loss of significant accuracy:

$$\frac{\Delta p}{\rho_c} = \frac{R}{12}\,\frac{62.4}{\rho_c} \quad \text{and :}$$

$$V_1 = \sqrt{64.4 \times 68\ R} = 66.1\sqrt{R} \text{ ft/sec.}$$

The equations for the nozzles are simplified and reduced to a constant X a coefficient X the square root of the gauge reading in inches. The velocity loss coefficient is found in the ASME Flow Measurement standards and is graphed in [1], page 218.

The following formulae can be used:

NOZZLE 1	NOZZLE 2
Throat diameter: 1.375 in.	Throat diameter: 2.000 in.
$V_1 = C_v\ 66.1\sqrt{R}$ ft./sec.	$V_1 = C_v\ 66.1\sqrt{R}$ ft./sec.
$Q = C_v\ 0.68\sqrt{R}$ cu.ft./sec.	$Q = C_v\ 1.44\sqrt{R}$ cu. ft./sec.

NOZZLE 1	NOZZLE 2
$\dot{m}_a = C_v \; 187 \sqrt{R}$ lbs/hr.	$\dot{m}_a = C_v \; 396 \sqrt{R}$ lbs/hr.

These formulae will apply unless the observed laboratory ambient conditions vary from the assumed standards, in which case corrections will have to be made. During these tests the values of the nozzle velocity coefficient have been used from [1]. The correction can be placed in the form of a correction factor to be applied to these tabulated formulae. Thus:

$$V_1 \propto \sqrt{\frac{1}{\rho_o}} \; ; \; \rho_o \propto \frac{p_o}{T_o} \; ; \; V_1 \propto \sqrt{\frac{T_o}{p_o}}$$; therefore a

correction factor can be applied and is equal to:

$$\sqrt{\frac{T_o \, p_{std}}{T_{std} \, p_o}} \; . \quad V_1 = \text{Corr. fac.} \times C_v \; 66.1 \sqrt{R}$$

The same correction factor will apply to Q because $Q = V_1 \times (\text{Area})$; however, $\dot{m}_a = V_1 \, A \, \rho_o$ and the correction factor for $\dot{m}_a$ will be:

$$\frac{T_{std} \, p_o}{T_o p_{std}} \times \sqrt{\frac{T_o \, p_{std}}{p_o T_{std}}} = \sqrt{\frac{T_{std} p_o}{T_o \, p_{std}}}$$

To correct air mass flow rates for laboratory conditions other than the standards used in deriving the formulas, multiply by the correction factor.

Example: data from run 6, test A-3, sample data sheet. Nozzle 2 in use.

$R = 2.20 + 0.01$ corr. $= 2.21$ in. H_2O.

Assuming $C_v = 1$, $V_1 = 66.1 \sqrt{2.21} = 98.5$ ft./sec. uncorrected.

correction factor for V_1 is $= \sqrt{\dfrac{530 \times 29.92}{530 \times 30.26}} = 0.989$

$V_1 + 0.989 \times 98.5 = 97.5$ ft./sec.

$Re = \dfrac{Vd}{\nu}$; where Re is Reynolds No.

V is velocity in ft./sec.

d is throat diameter in ft. $= 0.167$ ft.

ν is the kinematic viscosity of fluid being measured in ft.^{2}/sec.

$Re = \dfrac{97.5 \times 0.167}{1.5 \times 10^{-4}} = 10.82 \times 10^4$

From [1] page 218, for $Re = 10.82 \times 10^4$, $C_v = 0.979$

$\dot{m}_a$ = corr. fac. $\times 0.979 \times 396 \sqrt{2.21} = 582$ lbs./hr.

corr. factor is $\sqrt{\dfrac{530 \times 30.26}{530 \times 29.92}} = 1.01$

This could be repeated starting with the C_v of 0.979 to determine the velocity and a new Re and C_v until precise convergence is reached, but such accuracy cannot be justified in reading the velocity coefficient curve.

4. Determination of scavenging ratio:

Scavenging ratio for a two stroke cycle engine is defined as the ratio of the mass of air supplied to the engine to the mass of air just required to fill the cylinder at bottom dead center with the air ports open and at the exhaust pressure or:

$R_s = \dot{m}_a / N V_c \rho_s$; where:

R_s is the scavenging ratio.

$\dot{m}_a$ is the mass flow rate of inlet air.

V_c is the full cylinder volume $= V_d \dfrac{r_c}{r_c - 1}$

ρ_s is the density of dry air at the inlet temp. and the exhaust pressure.

V_d is the displacement volume.

r_c is the compression ratio.

As a representative calculation, data is taken from the sample data sheet, test A-3, run 1:

$$\dot{m}_a = 103/60 = 1.715 \text{ lbs./min.}$$

$$V_c = \text{constant} = 53.1 \text{ cu. in. } X\ 17/16 = 0.0326 \text{ cu. ft.}$$

$$\rho_s = \frac{P_e}{RT_i} = \frac{14.90 \text{ X } 144}{53.3 \text{ X } 605} = 0.0678 \text{ lbs./cu. ft.}$$

$$R_s = \frac{1.715}{608 \text{ X } 0.0326 \text{ X } 0.0678} = 1.271$$; $P_e \equiv$ exhaust pressure

ΔP =

At 608 RPM the engine is being supplied with 1.271 times the air required to fill the cylinder with clean air. This is assuming the situation of complete displacement by clean air of the exhaust products during the scavenging process. The shortcomings of this process lead to

another definition of use in scavenging parlance. This is given in [2] as the ratio of the mass of fresh air retained in the cylinder for use in combustion to that of the air supplied, and it is called "trapping efficiency". This is a rather difficult item to determine experimentally. Professor Taylor of M.I.T. [2] says that scavenging ratios greater than 1.4 are seldom used.

According to the assumption of complete mixing of air and exhaust gases, mathematical solution of the scavenging efficiency is:

$e_s = 1 - e^{-R_s}$ where: e_s is the scavenging eff.

R_s is the scavenging ratio.

By definition: $e_s = \Gamma R_s$; where Γ is the trapping eff.

Using R_s of 1.271 the scavenging efficiency at 608 RPM was :

$1 - e^{-1.271}$ which is 0.72 or 72 percent.

Table 1, page 19, shows the scavenging ratios for the speed range of the engine. Table 2, page 20, shows the ratios after the muffler was installed. The engine scavenging ratio is essentially independent of load, varying only slightly, at any given speed.

APPENDIX E

5 Determination of power lost to cooling water.

This can be calculated by making use of the installed flowmeter on the engine circulating water system. Determine the mass flow rate of water in percent and convert it to lbs./min. The temperature rise ΔT is determined from the recorded temperatures of the thermocouples installed in the inlet and outlet of the cylinder cooling jacket. A specific heat value for water of 1 BTU/lb - °F is sufficiently accurate to use throughout the range of temperatures and pressures encountered.

Example calculation: Data taken from the sample data sheet, test A-3, run 6.

$\dot{m}_w$ = 19.8 lbs./min. from Fig. 30, for 95% flowmeter reading at 148°F.

$$\Delta T = 180 - 150 = 30\ °F$$

HP (lost to water) $= \dfrac{\dot{m}_w \, C_p \, \Delta T}{42.41}$; where 42.41 is the conversion factor for BTU's/min. to HP.

$$HP(lost) = \frac{19.8 \times 30}{42.41} = 14.02 \text{ h.p. at full load.}$$

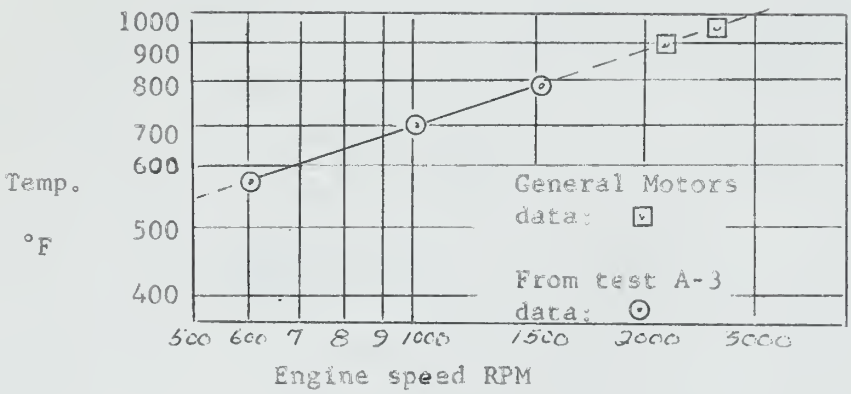
Similarly, the losses at the lesser powers are determined. Mass flow rates are taken from Fig. 30, which is a calibration curve for this flowmeter at 60 degrees and 180 degrees F. Flow rates at intermediate temperatures can be obtained by visual interpolation with little error.

6. Exhaust products heat loss

The following approximations have been made in these calculations due to the limitations of the installed instruments:

1. Combustion products are the result of burning with 200% theoretical air.

2. Exhaust temperatures at the higher powers are estimated from an extrapolated curve (shown below), and verified by previous data obtained at the General Motors test laboratory.



Plot showing how the exhaust temperatures at higher powers were determined where data was not available.

The fuel is a light diesel oil for which the typical formula is $C_{12}H_{26}$. The stoichiometric air-fuel ratio is 15 to 1 [2]. Heat lost to exhaust gases can be approximated from:

$$HP_{lost} = \frac{(\dot{m}_a + \dot{m}_f)\ \bar{C}_P\ (T_e - T_i)}{2544},$$

where $\dot{m}_a$ is the mass flow rate of air in lbs./hr.

$\dot{m}_f$ is the mass flow rate of fuel in lbs./hr.

$\bar{C}_p$ is the average specific heat for 200% theoretical air combustion products in BTU/lb-°F.

T_e is exhaust temperature in °R.

T_i is the air temperature at the cylinder inlet (air box) in °R. $T_e - T_i = \Delta T$.

Example from the sample data sheet, test A-3, run 6:

$\dot{m}_a + \dot{m}_f = (582 + 16.4) = 598.4$

$\bar{C}_p$ from Keenan and Keyes, Gas Tables (Table 8) for $T_{ave.} = \frac{630 + 1435}{2} = 1032$ °R is 0.261.

$T_e = 1435$, $T_i = 630$; $\Delta T = 805$

HP loss $= \frac{598.4 \times 0.261 \times 805}{2544} = 49.5$ h.p.

or $\frac{49.5}{120.5} \times 100 = 41\%$ of the fuel HP

APPENDIX F

Adjustable timing gears:

Two adjustable timing gears are being fabricated from AISI 1046 mild steel. Fig. 38 is a sketch of the gears showing their built up nature. The timing can be shifted by loosening the four lock screws and slipping the hub the desired number of degrees from the zero or reference position which corresponds with the conditions as shown on Fig. 11. The drive of the gear teeth is through a friction clamping between the hub, rim and clamping ring. If this does not maintain the proper grip, then serrating or knurling will have to be utilized.

The steel shows a BHN of 175 and would be better if the teeth were hardened (case hardening); however, considering the expense of obtaining this and the type service to which the gears will be exposed, it is not considered necessary.

Variations up to 15 degrees either side of the normal can be attained with these gears; however, note that any further delay in exhaust valve timing over that in test A-3 would leave the exhaust valves open after the air ports have closed, thus destroying the compression when air is lost.

It may be necessary, when the gears are completed, to take a new reference set of cam profiles for use in making timing changes. This will depend on how accurately the teeth are placed on the new gears with respect to the keyways for the shafts.

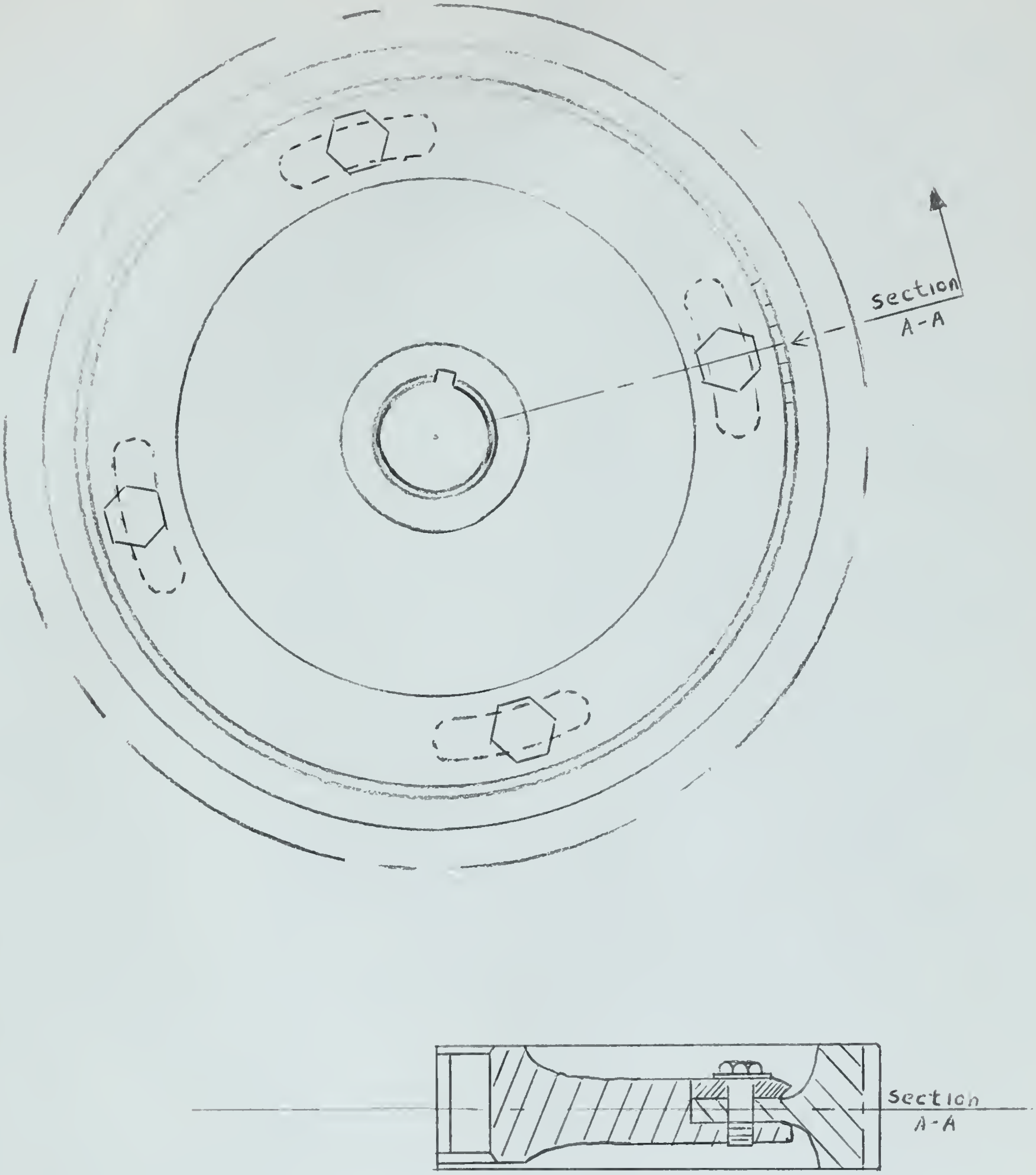


Fig.38. Approximate sketch of adjustable gears for the injector and exhaust valve camshafts.