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ENGINE PERFORMANCE IMPROVEMENT FOR THE 378-FOOT HIGH ENDURANCE CUTTER

E. A. Kasel, et al

June 1978

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ENGINE PERFORMANCE IMPROVEMENT FOR THE 378-FOOT HIGH ENDURANCE CUTTER

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JUNE 1978

FINAL REPORT

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16. Abstract

Methods for improving the performance of the main diesel engines of the 378-foot Coast Guard High Endurance Cutter have been investigated. These engines are models FM38TD8-1/8 rated for 3600hp at 900rpm. Present engine performance was evaluated through questionnaires and interviews with cognizant Coast Guard personnel and visits to selected cutters. In addition, operating records of the Coast Guard and Colt Industries were reviewed. Limited engine testing was performed to verify data trends. We concluded that improvements in engine performance can be made by converting these engines to a new style turbocharger system, pistons, and injectors. Recommendations are also made on methods to improve data taking in full-power trials and engine trend analysis.

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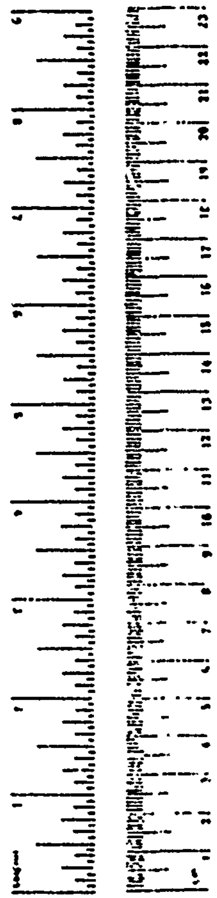
PREFACE

The reported work was performed under contract to the Department of Transportation, Transportation Systems Center, for the U.S. Coast Guard, Office of Research and Development. The Coast Guard project officers were Lt. Thomas J. Marhevko and Fred Weidner for R&D and CWO Robert P. Simpson for the Office of Engineering. The Transportation Systems Center Technical Monitor was Robert Walter. The efforts of the 378-foot High Endurance Cutters commanding and engineering officers are gratefully acknowledged as are the special efforts of Lt. F.L. Ames of the U.S. Coast Guard Third District in obtaining test data on the USCGC GALLATIN (WHEC 721) diesel engine conversion to turbocharger series scavenging-air system.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

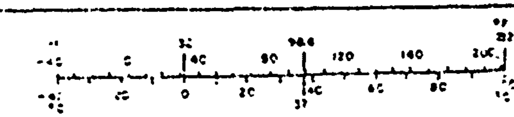


TABLE OF CONTENTS

TABLE OF CONTENTS (CONTINUED)

1. SUMMARY

The program was initiated to identify, by study and test, the most cost-effective methods of improving performance and economy while reducing smoke emissions and eliminating the operational problems of the Fairbanks Morse (FM) 38TD8-1/8 turbocharged diesel engines which power the 378-foot High Endurance Cutters (WHEC). These cutters are powered by a CODOG propulsion plant consisting of two 18,000 shp aircraft-type turbines and two 3,500 shp turbocharged opposed piston diesel engines. Both propulsion plants are connected through reduction gears to variable pitch propellers.

A study of major operating conditions and identification of major operating problems associated with the diesel engines on the 378-foot WHEC cutters were required as well as recommendations of methods to minimize any operational deficiencies. Data for this report were obtained from:

1) a reveiw of original engine test records,
2) Fairbanks-Morse Customer Service Department engine test reports,
3) a pointed questionnaire of all 378-foot WHEC cutters,
4) special in-house laboratory engine tests,
5) tests on the USCGC GALLATIN (WHEC 721),
6) personal visits to selected cutters for discussion with engineering officers and cognizant propulsion plant personnel.

2. BACKGROUND

This effort is a follow-on to work initiated in FY'74 by the U.S. Coast Guard through TSC to minimize the smoke emissions from CG icebreakers (Contract DOT-TSC-905). That work was to verify, through actual field tests, the effectiveness of proposed engine modifications to improve engine efficiency and reduce exhaust emissions of the 38D8-1/8 Fairbanks Morse blower-scavenged, opposed-piston (O.P.) engine. Those modifications, which had been tested in the laboratory, consisted of newer style pintle-type nozzles, shimmed injection pumps, and advanced injection timing.[1] The field test on the #3 Main Diesel Engine of the USCGC MACKINAW (WAGB-83) compared baseline (unmodified) and modified engine test results while the shop performed the routine maneuvers of engine start-up, idle, undocking, docking, and steady steaming. Gaseous and smoke emissions, fuel consumption, and other pertinent engine parameters were measured as a function of engine speed and load.[2]

3. CONCLUSIONS AND RECOMMENDATIONS

We have identified methods of improving engine performance through engine, operational, and maintenance modifications to the FM 38TD8-1/8 engine when used as main propulsion power in the 378-foot WHEC. In general, these improvements are associated with problem areas being encountered by these engines on the cutters. The identified problem areas and recommended methods for improvement are listed below. The conversion from the existing turbocharger scavenging-air-system to a series-type system offers the most promise for improved engine performance (Section 5.2).

The following 10 problem areas have been identified for remedial action:

Conclusions:	Recommendations:
1. Some Engines are not capable of meeting full-power trial requirements without excessive exhaust temperatures or reduced power settings.	1. Modify the full-power trial directive to affect procedural changes in these trials.
2. The existing turbocharger scavenging air-system gives excess air at low engine torque and only adequate air at rated torque.	2. Convert all FM38TD8-1/8 engines to series turbocharging. This air system is better suited for variable speed, variable torque operation, and will improve fuel economy 2 to 10 percent. Conduct field tests on an engine, prior to and after conversion, to fully document the improved performance.
3. Some engines experience excessive fuel, lube, water, and manifold leaks which waste resources and reduce engine efficiency.	3. Convert all engines to gasketless fuel injection nozzles to eliminate high pressure fuel leaks. Specify "O" rings of viton in areas of water leakage problems. Up-date manifold joints to present production engines with new or machined parts. Reduce leakage and fire hazards by converting to latest production water-cooled decks above the exhaust manifold (Section 5.3).

4. Three different piston styles are in use in the 378 fleet-an unnecessary complication that was aggravated by poor liason between FM and CG elements.

4. Convert all engines on an as-needed basis to the latest production-style piston, which will offer economy of operation and reduced smoke. Improve liason between FM and CG to assure the CG is being kept aware of the latest production components (Section 5.4).

5. Trend analysis is not always properly conducted to maximize its usefulness in determining engine condition and performance deterioration.

5. Obtain trend analysis data (more important than full power trials) every 200 hours at a set engine speed and fuel rack setting with compensation for ambient conditions (Section 5.6). It is also recommended that engine diagnostics be developed to assist in engine performance evaluation (Section 5.9).

6. Engine overhaul schedules and parts replacement procedures require review.

6. Base maintenance on a compromise approach with some components on a scheduled basis and other components on an as-needed basis. Schedule maintenance for engine components such as damper assemblies, flexible drives, and engine bearings which have a known useful life and experience similar wear-rates regardless of engine speed and load.

7. Long periods of engine idle for warm-up and engine stand-by are being practiced, a procedure considered to be counter-productive and degrading to engine performance.

7. Reduce engine idle periods to an absolute minimum--any warm-ups in excess of 5 minutes can only result in reducing the viscosity of the oil (Section 5.7).

8. One-engine operation with proper care exercised to eliminate overtorquing is not being practiced.

8. Practice one-engine operation when possible since it reduces total fuel consumption and improves engine performance. The engine torque is nearer the optimum fuel economy point with better brake specific fuel consumption. This also keeps engine temperatures higher, reducing the possibility of oil build-up in the manifolds. However, it is important that the governor linkage be properly set to assure the engine cannot be over-torqued (Section 5.10).

9. A method of measuring engine load that would be more useful in engine condition monitoring is required.

9. Utilize a calibrated fuel-rack position indicator for a reliable measure of engine torque in light of the past CG experience and the complication of shaft Hp meter (Section 5.8).

10. Some cutters have experienced bearing failures.

10. Minimize catastrophic bearing failures by increased use of engine starting and shut-down interlocking in the lube oil circuit and strainers to eliminate human error. A good spectrographic lube oil analysis program should indicate normal bearing wear. The importance of crew training must be emphasized (Section 5.11).

4. APPROACH

4.1 QUESTIONNAIRE SURVEY

A questionnaire survey, relating to all main diesel engines in the 378-foot WHEC fleet, was answered by the ships' engineering officers. The responses showed a similarity in problems and potential solutions from ship to ship. Some of the problems were isolated cases but the major problems occurred throughout the fleet. The most serious problems identified were: (1) engine not capable of producing full power, (2) exhaust manifold leakage with resultant engine room exhaust gas, (3) fuel and lube oil leakage, (4) liner seal leaks, (5) oil leakage at the exhaust-deck seals, and (6) high exhaust temperatures encountered during efforts to obtain full power. Detailed discussion of problems mentioned above are included in Appendix A.

4.2 SHIPS VISITS AND INTERVIEWS

The visits to various 378-foot WHEC cutters and interviews with the responsible engineering personnel generally confirmed the results of the questionnaire survey. Cutters visited included the USCGC's DALLAS (716), GALLATIN (721), HAMILTON (715), SHERMAN (720), RUSH (723), MIDGETT (726), and MUNRO (724). Discussions were also conducted with various district type-desks. The results of these interviews are included as part of the major problem areas discussed in Section 4 and detailed in Appendix A.

The following topics were covered:

1. Cylinder Liner Seals
2. Exhaust Manifold Gaskets and Leaks
3. Exhaust Manifold Flexible Connectors
4. Exhaust Manifold Fires and Gasket Groove Design
5. Fuel Injection Compartment Fuel Leaks
6. Scavenging Air-Piping Leaks
7. Oil Leakage into Engine Cylinders
8. External Oil Leakage at Exhaust Belts
9. Oil Leakage at Front Cover of the Engine
10. Water Leakage at "O" Rings and Gaskets
11. Front Cover Removal Difficulty
12. Cylinder Air Start - 12 versus 6 Air Start Valves

13. Crankcase Vacuum Problems
14. Lube Oil Pump Level
15. Exhaust Temperatures Higher than Recommended
16. Piston Designs Available.

4.3 ORIGINAL ENGINE ACCEPTANCE TEST DATA

The original shop test logs of the engines built for the Coast Guard 378-foot WHEC cutters were reviewed (Appendix B) and the average performance data of every engine at each 25 percent increment of load was tabulated on a different log sheet for ease of comparison (Figures B-1 to B-10). Careful examination will show the effect of engine air inlet temperature on overall engine performance. For instance, note the wide variation of compressor air inlet temperature at 100 percent load (Figure B-7). With the existing style scavenging air system, the turbo air inlet temperature is not only dependent on ambient temperature but also on work done by the engine driven blower.

The 100 percent load air receiver pressure (Figure B-7) shows how similar all engines performed at the shop test trials. They all ran between 32 and 34.8 inches of mercury air receiver pressure at rated load and speed. The average data of all the engines is plotted on Figures B-13 and B-14 of Appendix B. The rated speed and load fuel consumption averaged .376 lbs/bhp/hr with a new engine. This consumption can improve up to 2.5 percent as the engine becomes well run-in.

A variable load and speed test was performed on one of the subject engines, S/N 38D867070-TDG12. Data is plotted on Figures B-15, B-16, and B-17, and B-18 of Appendix B. Figure B-18 is quite useful in determining engine load-speed characteristics for optimum fuel consumption of the main propulsion diesel engine. If the engine torque were held at around 75 percent to 80 percent of rated torque with the scavenging air system when operating at reduced speeds, the fuel rack reading would be about 5.5 and the brake specific fuel consumption (BSFC) would be .360, .364, .368 and .370 lbs/bhp/hr respectively at 500, 600, 700 and 800 engine rpm while the horsepowers would be 1500, 1850, 2150 and 2900. The 800 rpm point is at 90 percent torque and 6.3 rack. At rated speed and 100 percent torque, the rack would read 7.3 with a BSFC of about .380. RPM, horsepower, and BSFC values are 500 - 1200 - .368, 600 - 1700 - .3675, 700 - 2256 - .368, 800 - 2900 - .370, and 900 - 3600 - .380 if one follows the torque limiting curve per the engine instruction book (Figure B-19). By comparing these numbers to those in the above paragraph, we can see a potential reduction in fuel consumption (of 2.2 percent at 500 rpm and 1 percent at 600 rpm) as a result of increasing the engine load above the design torque curve.

4.4 LABORATORY TEST DATA

Laboratory tests were run on a 6-cylinder engine with turboblower, series-scavenging system operating along the contract load-speed specifications of Curve 11188CH (Appendic C). In support of the GALLATIN conversion to a series scavenging system, data was obtained with various size nozzle rings and is shown in Appendix D. These results will be discussed in more detail under Section 5.2 on the turbocharger-series-conversion.

5. 378-FOOT WHEC MAIN DIESEL ENGINES - PROBLEMS AND CORRECTIVE ACTIONS

5.1 FULL-POWER TRIALS

An engine in normal operating condition should have no problems performing a full power trial. The full-power trials, as presently required by the Coast Guard on these engines, will always show up poorly for the engine when power demand is more than engine rating. Engine operating parameters such as rack readings are better indicators of the horsepower required by the ship's propeller than the ship's speed, horsepower meter or prop-pitch setting. With the engine fuel-racks limited to full power setting and balanced per FM instructions, the propeller pitch/speed should be changed to produce engine rated speed and load. Engine and loading data should then be obtained to help determine changes in engine and propeller efficiency with time.

If the engine performance data is adequate, the percentage change in propeller horsepower indication is more than likely in the horsepower meter and/or propeller pitch and ship efficiency.

The horsepower output of the engine is only proportional to the engine speed while the propeller requirement is the cube of the speed (Figure 1). In meaningful terms, because of hull fouling or improper pitch setting, the ship's propeller requires engine rated power at a 1 percent reduction in propeller speed (148.5 rpm versus 150 rpm), and the engine would have to be overfueled more than 3 percent to return to rated speed of 150 rpm. The actual engine full torque will then occur at an engine speed of 887 rpm and at 98.5 percent rated power. To obtain rated speed would require the engine to put out 3607 shp instead of 3500 shp. It would be better to load the engine at rated speed until the full power fuel-rack setting is reached and then obtain engine performance data for comparison with prior data.

An alternative to a full power trial would be to run at full torque at rated speed or rated torque at some lower speed to determine the condition of the engine and ship propulsion system. This

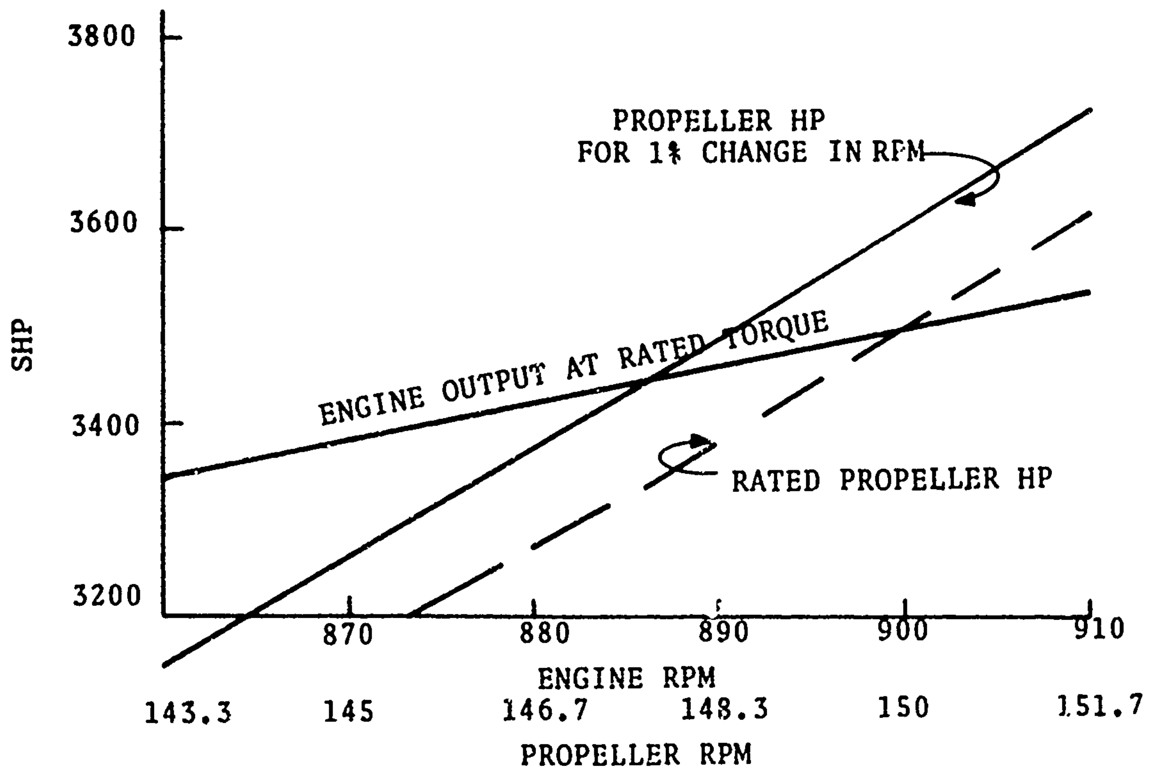


FIGURE 1. ENGINE POWER VS SPEED CURVE FOR PROPELLER LOADING

could be accomplished by:

1. Rated torque and speed power trials. Reduce propeller pitch to allow engine to come to rated rpm. Increase pitch until full power rack-reading is obtained ($\sim$7.5). Proceed with 2 hour test run and log all propulsion data as well as engine temperatures, pressures, etc. Compare data with previous test runs to determine if any deficiencies exist in the engine or loading system.

2. Rated torque or fuel input at something less than rated speed. The pitch could always be maintained at a constant setting (say 1.0) and engine speed increased until the average fuel-rack reading remained at 7.0. Proceed with 2 hour test run and log all engine and propulsion data, provided the manufacturer's limits on engine operating parameters are not exceeded.

5.2 SCAVENGING AIR SYSTEM

Turbocharging increases the efficiency of all types of diesel engines but makes them more prone to inefficient performance unless they are properly operated and maintained. Many of the present operating complaints are directly or indirectly related to the existing scavenging air system which gives a fixed mass of air as a function of engine speed up to approximately 85 percent of rated torque.

This system (GALLATIN excluded) gives more air than needed at low engine torques and only adequate air at rated torque even if the total engine system is in good operating condition. This air system is optimum at 75 to 85 percent of rated torque. The air is supplied by a roots-type, positive-displacement blower connected in series with, and up stream of, the turbochargers. Above 85 percent torque the engine operates as a self-sustained, turbocharged, two-cycle engine. However, the roots-blower continues to take some power and heats the inlet air to the turbocharger. This heated air reduces the efficiency of the turbochargers.

Many of the problems associated with the air scavenging system can be alleviated by conversion to the turbocharger series blower.

In this arrangement the engine scavenging air system is converted to place the scavenging-air, roots-blower after the turbocharger. Minimum equipment changes required included: (1) new air cooler supports and piping from supports to blower, (2) modification of the blower inner bearing plate to allow air discharge directly into the engine air-receiver, (3) blower-drive gears, (4) crankcase ejector system, and (5) possible turbocharger nozzle ring modification. This air system, which is standard on all new turbocharged engines, is much better suited for variable-speed, variable-torque type engine operation. Air flow can be tailored to the engine application by varying the turbocharger nozzle-ring size and blower-drive gear ratio.

The engine brake specific fuel consumption can be reduced from 2 to 10 percent through the engine speed range of 500 to 900 rpm by changing to the turbo-blower series scavenging air system. The pre-turbine exhaust will generally reach temperatures between 500 and 900°F in the range of 500 to 900 rpm engine speed with a fixed-pitch propeller loading.

The extra scavenging air available from the conversion to the turbocharger series blower also reduces the thermal stresses at the exhaust m[illegible] joints and increases the over-fueled smoke limiting load of the engine.

5.2.1 Turbo-Blower Series Conversion - parts and costs

The conversion list price is $15,896.75 per engine less Coast Guard discount in effect at time of placing the order. Labor for the conversion is not included in this estimate. In addition to changing out the parts listed below, work necessary during the conversion includes cutting two openings in the blower inner bearing plate and inspecting and rebuilding the blower if deemed necessary.

			FM part #
1.	Blower Drive Gears	-	16 401 281
2.	Piping Turbo Inlet	-	16 608 398
3.	Piping Turbo Discharge	-	16 608 489
4.	Gasket Kit	-	16 608 405
5.	Flange	-	16 608 224
6.	Piping Spacer Kit	-	16 608 498
7.	Crankcase Ejector	-	16 608 423

The ratio of specified blower gears are 1.385 and should not require a turbocharger nozzle ring change.

In the above conversion, if the present air check-valve housing were reused as part of the air piping and the opening blocked off by making flanges and gaskets, the price would be reduced by $798.19. If the air coolers have excess cooling capacity, the air header above the first cooler and the connector between the two headers would not require replacement, and another $2,913.79 can be saved for a final cost of $12,381.79. A 5 percent improvement in fuel consumption could amount to $9000 in a period of 5000 hours operation at an average load of 2000 horsepower with 35 cents per gallon fuel cost. The savings would be about .1 cents per horsepower - hour with a 5 percent improvement in fuel consumption.

If the average 378-foot WHEC uses 1,000,000 gal of fuel per year and the assumption is made that 70 percent of this fuel is burned by the main diesel engines, then a 5 percent savings at $.35 per gallon would be $12,250. It would appear that the payback time for a two-engine conversion would be two to four years, not including, the operational and safety benefits accrued from such a conversion.

5.2.2 <u>Cutter Gallatin Conversion - Experience and Test Results</u>

In 1977 the USCGC GALLATIN (WHEC-721) engines were converted to turbo-blower series scavenging system via Government Register Number 92-930466. The system consisted of new air piping, slower blower drive gears (1.51:1 ratio), crankcase ejector kit, gasket kit, and spacer kit. The engine had 10° gasketless nozzles and the rotating piston combination (See Section 7.4.) After the turbo-charger nozzle rings were changed from 19 to 21 square inches, operational engine data were obtained. These data are shown in Appendix D as Curves 63 and 64.

Curve 63 shows engine performance between 500 and 900 rpm with both engines operating. Note the combined exhaust temperatures to the turbocharger remained between 600°F. and 875°F. throughout the engine operating speed range. These temperatures are high enough to prevent accumulation of oil in the exhaust system and low enough to prevent premature failure of the exhaust manifold gaskets and flexible connectors.

The initial excess air problems associated with USCGC GALLATIN conversion have been resolved by going to a 21 square inch nozzle ring in the turbochargers. However, the real cause seems to be the blower drive gear ratio used in the conversion kit (same as used for stationary 720 rpm conversions) and the 10 degree injection nozzles. The nozzles are being changes to 15°, but test data are not yet available. However, the conversion has performed faultlessly to data, other than the high scavenging and firing pressures observed at rated speed and load. While it is believed that the engine performance could be slightly improved by going to slower blower-drive gears and a smaller turbocharger nozzle-ring, the data indicate the engine is performing satisfactorily.

Curve number 64 shows operation of the GALLATIN with one engine in the engine speed range of 500 to 720 rpm. The fuel-rack readings indicate that the same engine torque occurs at 700 to 720 engine rpm with one engine operation as occurs at 850 rpm with two-engine operation. Here again, it is believed that the performance could be improved with a slightly slower blower and smaller turbo-charger nozzle-ring area.

5.2.3 Laboratory Test Data

As previously mentioned, some laboratory tests were run at the FM plant on a six cylinder engine with the series conversion and 19 and 21 in. nozzle rings. The results are detailed in Appendix C in curves 60 through 62.

The fuel consumption numbers from a 6-cylinder engine cannot be used as shown for comparison to the 12 cylinder engine because the engine loading is not corrected for the water-pump which requires 43 horsepower at 900 rpm. The correction required changes the .371 BSFC to .362 with the Elliott turbocharger at 900 rpm engine speed. The interesting aspects of these data are the changes in exhaust temperature with speed and the change of air-manifold pressure with engine torque and speed. The old-style scavenging system gave a range of exhaust temperatures from 650 to 960°F. versus 740 to 915°F. for the turbo-blower series arrangement at contract torque speed-loading. The comparison would be even more startling at torque per the propeller curve loading.

Propeller curve data run on a lab engine in 1971 with the turbo-blower series scavenging system and rotating turbocharged engine pistons is attached in Appendix C as Curve number 66. The fuel consumption was decreased 2 percent to 8 percent when compared with the present engine scavenging system with typical propeller-engine loading. This data is very similar to what should be expected of the GALLATIN engine conversion with a slightly smaller scavenging air-blower.

Further laboratory tests to determine the operating parameters of a turbo-blower series engine at various speeds and torques are required to develop good performance maps of the engine in marine service.

5.3 FUEL, LUBE OIL, AND WATER AND AIR LEAKS

Leaks waste vital resources and, in the case of air and exhaust leaks, can affect engine efficiency. The engine must be kept free of external leaks to prevent loss of fuel and lubricating oil. Air and exhaust gas leakage is a form of wastegating energy in the scavenging system of an engine. This reduces the overall efficiency of the engine turbocharger and is directly reflected in the engine fuel consumption. The propulsion plant personnel should always check for leaks and make necessary repairs at the first opportunity. The engine should not be operated with large scavenging air or exhaust gas leaks except in cases of emergency.

The engines have been or are being converted to the newer style gasketless injection nozzles. This conversion will eliminate nearly all sources of high pressure fuel oil leakage.

Engine availability can be increased by specifying O-rings of viton material for areas where water leaks have developed prior to replacement at overhaul time. However, these rings are about 2-1/2 times more costly than the standard O-ring material.

Engine efficiency can also be improved by reducing manifold gas leakage. Manifold joints can be updated to current production standards by machining or applying new parts. One of the ships was using two flexitallic exhaust manifold gaskets in each joint to correct for exhaust manifold flange warpage. This makes it very difficult to maintain a tight joint due to the gasket creepage and loss of joint bolting torque. The loose bolts then allow the gaskets to leak and burn out. Engine fire hazard will be reduced by adopting these changes as well as new pistons (Section 5.4) and can be further improved by incorporating the latest production water-cooled deck above the exhaust manifolds.

5.4 PISTONS AND INJECTORS-EXISTING AND RECOMMENDED STYLES

Engine power pistons are presently of the rotating style or the Mexican hat fixed design. The 378-foot WHEC's have been changing over to the Mexican hat fixed design as the rotating pistons require replacement. The MUNRO AND MIDGETT engines were supplied with Mexican hat fixed pistons while all other cutters originally had the rotating piston. Generally, the engines with fixed pistons have had no piston failures, except due to external causes, while several engines with rotating pistons had one or more failures. The fixed Mexican hat piston generally produced more smoke, more port carboning, and more cracked cylinder liners but experienced less mechanical problems and liner seal leakage problems.

When changing to the fixed style piston, the latest production version should be adopted in lieu of the Mexican-hat style. This new piston is more durable and will produce less smoke and provide better fuel consumption than the Mexican hat piston.

All FM opposed piston turbocharged diesel engines use the same cylinder liner. There are three different pistons available for the 38TD8-1/8 engine:

1. Rotating piston. The upper and lower pistons have different cup volumes and must be used in combination of one upper and one lower piston per cylinder. The piston assembly has a bearing between the piston and piston carrier to allow the piston to rotate freely and is retained to the carrier with lock-plates and capscrews.

2. Mexican hat fixed piston. The upper and lower pistons have the same cup volumes but are different due to fuel slots for the injection nozzle fuel spray located at the edge of the crown. They must be used in combination of one upper and one lower piston per cylinder. The piston is fastened to the piston carrier with four capscrews and is not free to rotate.

3. Turbo-fixed piston (present production piston). The upper and lower pistons are identical and two of these pistons are used per cylinder. The piston has a combustion chamber or crown shape similar to the rotating style pistons. The piston is fastened to the piston carrier with four capscrews and is not free to rotate.

For this same engine there are two different injection nozzles available:

a. 15° Injector. This injector has a holder with the tip pointing 15° off the centerline of the injector body. It is intended for use with all engines not having Mexican hat pistons and is the only injector recommended for use with the turbo-fixed piston. This injector can, however, be used successfully with the other type pistons available for the engine.

b. 10° Injector. This injector has a holder with the tip pointing 10° off the centerline of the injector body. It is intended for use with the Mexican hat piston and can be used successfully with the rotating piston.

The advantages and disadvantages of these various power combinations are discussed in Appendix E.

It became apparent that some elements of the Coast Guard were not cognizant of the latest production components. In order to alleviate this situation, direct liaison between the WHEC type desk of the Coast Guard's Naval Engineering Division (G-ENE) and FM engineering should be established on a semi-annual basis. This will keep ENE aware of the latest design production changes available. However, some of the latest changes are listed below, and it is recommended that they be incorporated into CG engines as early possible:

1) exhaust manifold gasket grooves,

2) air piping packing assembly,

3) exhaust belt seals and gasket grooves,

4) gasketless fuel injectors,

5) exhaust pipe screens to protect turbos,

6) water cooled injection compartment decks.

5.5 MAINTENANCE AND OVERHAUL

The maintenance of the 378-foot WHEC engines is fairly good considering the high turnover rate of shipboard mechanics. It was noted that a considerable variation existed in engine room morale from ship to ship. Recurring engine problem areas can contribute to poor morale. If the basic causes of engine problems are not resolved, an engine that requires nearly continuous attention for barely acceptable engine performance may result. Also, if the maintenance requirements become overbearing for the crew, general engine neglect becomes the rule.

The Coast Guard sends nearly all their engine-men through a one-week school held by FM in Beloit, Wisconsin. This school is brief but quite informative on the unique features of the opposed piston engine, both naturally aspirated and turbocharged.

The larger engine maintenance jobs, including overhauls on these ships, are usually undertaken with the direction of an FM factory representative and a Coast Guard maintenance team.

<u>Scheduled versus As-Needed Maintenance and Overhaul</u>

Service parts replacement and overhaul should be carried out as necessary and only on an as-needed basis.

To overhaul an engine on an as-needed basis requires excellent records of power parts replacements and visual and dimensional inspections. It does not mean an engine should be run until a breakdown occurs. A complete overhaul should be scheduled when enough of the engine power parts are approaching their wear limit since it is more economical to replace all parts rather than only those in need of replacement. This will usually occur when the upper piston compression rings are worn out. However, depending on the engine service, the O.P. engine will require two lower piston ring changes per upper piston ring change. Lower pistons can be removed and rerung easily with about 3 man-hours per cylinder labor and requires only new rings and new cotter keys for the connecting rod bolts. The current Overhaul Directive is attached as Appendix F.

Scheduled maintenance should only be applied to engine components that generally have a known useful life and see a similar wear-rate regardless of engine operating conditions. Some of these components are damper assemblies, flexible drives, engine bearings, etc. Scheduled maintenance on power components will generally only work well and be economical if the engine operating conditions remain the same day after day. Scheduled maintenance is costly and may cause unexpected maintenance requirements due to poor repairs. On the other hand, properly performed scheduled maintenance may alleviate unexpected engine failures.

The best approach would be scheduled power component inspections in conjunction with trend analysis (Section 5.6). The upper compression rings can easily be inspected for wear and breakage through the air-ports at intervals of 2500 hours or less. The lower piston compression rings can be inspected through the exhaust ports any time the exhaust manifolds are removed for re-gasketing or by dropping one or two piston assemblies. Any time a piston assembly is removed, the compression rings should be replaced. Always note the hours on rings and their condition as an indicator for when the other lower rings should be replaced.

Liner bore conditions can be determined by visual and feel inspection through the exhaust ports or when the lower piston is removed. If an individual upper piston requires replacement of broken rings prior to other rings wearing out, it can be removed down through the liner with available tools and rigging without removal of the upper crankshaft. Generally, inspections should be conducted as scheduled and as-needed plus whenever other engine work would make inspection convient. Engine components such as air start valves and injectors should have a scheduled inspection and be repaired as indicated by these inspections. Injection pump condition can be determined by visual inspection of one or two assemblies. Condition of plunger helix barrel port holes, rack and pinion teeth, and delivery valves should be noted and recorded. In all cases, different assemblies can be inspected at each scheduled inspection interval. This inspection should include a dimensional check of all wearing type components and a record of operating hours and the dimensions.

As previously stated, good general engine maintenance, such as repairing all air, exhaust, and fuel and water leaks as they develop, will make it much easier to spot internal engine deficiencies. The engine instrumentation and connection lines must be kept clean and in good repair for proper monitoring of engine performance.

5.6 ENGINE TREND ANALYSIS

Engine trend analysis data can be more important than full power trials as an indicator of engine condition. It reflects changes in engine performance with time whereas the full power trial indicates the system capability as only one time annually.

The trend analysis is useful for determining engine deterioration and pin-pointing operational problems. Without good trend records it becomes very difficult to continue performing good engine preventive-type maintenance. A well conducted program can keep engines in top operating condition.

We recommended obtaining operating data every 200 hours or whenever engine difficulties are suspected. It is very important that data for trend plotting be obtained at comparable engine operating conditions. Good trend data is obtained each time at the same engine rpm and fuel rack reading. In a turbocharged engine, scavenging pressures and the other engine parameters affected by scavenging pressure will show a change with speed and torque. Engine ambient conditions should be taken into consideration and plotted along with propulsion loading data. The engine rpm and fuel rack setting are the most important baseline references when obtaining engine data; not ship speed, propeller pitch, water depth, hull cleanliness, etc.

The following should be considered in trend analysis:

1. Cylinder Compression and Firing Pressures. These pressures are most helpful in pinpointing a malfunctioning cylinder. Generally, if other engine operating pressure and temperature indications are normal, it is not worth the effort to check the firing and compression pressures of every cylinder. Checking the firing pressure of one cylinder every 200 hours would be worthwhile to assure that the engine and fuel settings have not changed the overall engine peak combustion pressures. If the check indicates a change, then other cylinder readings should be taken to verify the change and appropriate action taken to correct the cause.

2. Cylinder Exhaust Temperatures. The exhaust temperatures are a good guide to the efficiency of the cylinder combustion. (The two-cycle O.P. turbocharged engine exhaust temperatures should never be balanced by adjusting the fuel racks). The indicated temperature is a time-average of the real temperature to which the

measuring thermocouple is exposed, and in the case of FM engines, the cylinder causing the higher temperature may be actually upstream from the one indicating the largest temperature increase. It is best to plot the average temperature of each three-cylinder manifold grouping for analysis and log the individual temperatures on a separate sheet. Individual or group changes in the exhaust temperatures without a change in other engine parameters point to faulty fuel injection components. A temperature change in all cylinders indicates a timing or fuel change to all cylinders. Always study the temperatures and try to logically explain changes as indicative of changes in air manifold pressure, load, ambient temperature, air manifold temperature, or fuel-rack reading. If only one temperature is greatly different, it may be instrumentation. A faulty reference temperature for the pyrometer can also lead to a change in all the temperatures. Also, a missing number one compression ring on the lower piston will give an increase in cylinder-exhaust temperature. It will show up as in increase in temperature of the cylinder affected and other cylinders in the same manifold grouping. The average temperature of this group of cylinders will increase while other groups show little or no change in average temperature.

3. Crankcase Vacuum. The crankcase vacuum normally remains stable if the engine is in good mechanical condition. A significant change in crankcase vacuum should be cause for immediate concern and all possible effort made to determine the reason for the change. A rapid change in crankcase vacuum is usually caused by a severe engine malfunction, such as a failed or cracked piston, failed turbocharger bearing and seal, failed blower bearing and seal, or some reason for improper operation of the ejector system. Crankcase explosions are nearly always caused by overheated power parts, such as failed bearings or a severely burned piston.

4. Engine Scavenging Pressures. The engine scavenging pressures should be plotted against operating hours to help monitor engine condition. In a two-cycle engine, pressures should be recorded before and after each function that caused a change in

scavenging system pressures. In the turbo-blower series engine configuration, log and plot: (a) pressure from the turbocharger, (b) the pressure from the blower in the air manifold or pressure from the cylinders in the exhaust manifold, and (c) the pressure in the exhaust stack from the turbochargers. The changes in air manifold pressure should be studied in conjunction with any pressure rise across the turbocharger and scavenging air blower as well as the pressure drop across the engine cylinders and turbocharger. If the scavenging air manifold pressure increases, study the pressures to determine the cause (turbo blower series only). Some of the causes are:

1) a load increase on the engine (check racks);

2) lower air temperature to the blower which increases the blower air mass capacity (check temperature to blower);

3) colder ambient temperature which increases the pressure rise across the turbo compressor because of increased efficiency and blower air mass capacity due to the higher pressure to the blower unit (check ambient temperature);

4) carboned cylinder ports which increase the pressure drop between the air manifold and exhaust manifold (check for carbon);

5) blockage at the turbocharger nozzle ring and turbine which increases the exhaust manifold pressure to the turbocharger, usually associated with a larger temperature drop of the exhaust gases across the turbocharger (check for blockage);

6) increased turbocharger speed which is usually caused by a blockage as in 5 above (This shows up as an increase in air temperature across the compressor.);

7) extra exhaust gas energy to the turbocharger in the

form of increased exhaust gas temperatures which can be caused by a change in cranklead, late injection timing, or faulty injection nozzles;

8) blockage in the exhaust piping f m the turbocharger (check for blockage).

5. Lube Oil Consumption. Lube oil consumption records provide data on the engine for comparative purposes with similar engines and help estimate future lube oil requirements. In an O.P. engine, oil consumption will increase as the engine power-parts wear to their condemnable limits. Increased oil consumption can also be caused by losses through external leakage, the turbocharger, the scavenging air-blower seals, and filter or strainer cleaning.

6. Fuel Rack and Governor Indication. Generally, fuel pump deterioration is so slow that any significant increase in fuel-rack reading should be considered as a change in the engine load or in engine efficiency. Other indicators will help determine where the change occurred. If the engine is performing well, it is reasonable to suspect an overload condition.

7. Pressure Differential Across Lube Oil Strainers and Filters. If the pressure differential is plotted versus time, it will normally show a slow, steady pressure drop. A rapid increase indicates a failed internal engine component. A bearing or piston failure will show up in one to three minutes by nearly plugging the strainer screen creating a subsequent large pressure drop across the strainer. The engine should be stopped immediately for internal inspection of the engine. As the lube-oil strainer screen begins to clog from normal crankcase lube-oil dirt particles, the rate of pressure drop increases rapidly over several hours, due to plugging of the smaller screen openings with large particles and subsequent finer filtration of the lube oil. Smaller particles that would normally pass through the strainer are suddenly trapped. At each cleaning of the oil strainer and each filter change, the elements should be examined for metal particles which may be indicative of metal from various engine internal components such as a vertical-drive bearing cage or gear-teeth.

8. Engine Water and Lube Oil Temperature and Temperature Differential. These should remain within published specifications. An increase in differential temperature indicates a decrease in flow or an increase in heat load. A decrease in flow of the lube oil can be caused by an open relief valve or a blockage in the circulating system. A decrease in water flow will occur due to an increase in the system pressure drop.

9. Exhaust Appearance. Exhaust color is a good indicator of engine combustion efficiency. Black smoke can be caused by a sudden speed-load increase, or overloading, or injection system deterioration. Nozzle deterioration will show up as an increase in steady-state black smoke.

10. Instrumentation. Instrumentation must be kept in good condition and calibrated. Before trend data is obtained, accumulated liquids in air and exhaust pressure gauge-lines must be blown clean. Test and calibrate instrumentation if any doubt exists as to its accuracy.

5.7 ENGINE IDLING

Long periods of engine idle for warm-up or engine stand-by are not desirable for any engine. Extended idling contributes to increased engine wear, degrades lube oil by fuel dilution and, in the case of the O.P. engine, contributes to lube oil build-up in the exhaust manifolds. This subsequently leads to manifold fires and related safety problems. Any warm-up time beyond five minutes only results in reducing the viscosity of the lube oil and, therefore, oil pressure at rated engine speed. The engine normal operating temperatures can generally best be obtained with the engine operating under loaded conditions. Low speed engine loading can commence within five minutes after engine light-off time.

5.8 ENGINE LOAD METERING

A meaningful indication of engine load will allow operating the engine to obtain best performance. In light of the CG's previous experience and the complication of a sophisticated horsepower meter, the 378-foot WHEC fleet should adopt a simple calibrated fuel-rack position indicator. This indicator would give a very reliable indication of engine torque. These devices can operate directly off the engine governor or from a potentiometer at the fuel-pump control rack.

5.9 ENGINE DIAGNOSTICS

Available engine diagnostics presently require highly technical personnel to obtain useful information. Such instrumentation must be simplified into useful hardware for the average engineman. Good diagnostics equipment, if and when available, would help in determining engine performance deterioration. A competent engine-man with knowledge of the particular engine type can sense and anticipate engine service requirements. However, a simple and easy way to interpret a diagnostic instrument that keeps the engineman involved and assists him in detecting engine malfunctions would be most useful.

5.10 ONE ENGINE-ONE SHAFT OPERATION AND PART LOAD-SPEED OPERATION

The use of one engine in place of two, when possible, offers very important economies of operation, safety, and improved engine performance. Two engines should be run only when needed for speed and safety. Engine fuel consumption and performance can be improved at reduced engine speeds by keeping the engine torque near the optimum fuel consumption point. Fuel rack position versus engine speed information can readily be compared to the engine performance map to obtain maximum economy with the variable pitch propeller.

However, in one engine operation, care must be taken not to over-torque the engine. Proper adjustment of the torque limiting aspect of the engine governor is required when operating with one engine or continuous overtorqueing of the engine could result. One-engine operation is usually in the range of 460 to 770 engine rpm. On one ship visited, fuel injection pump rack readings were allowed to go to 7 and higher anywhere in this speed range. These rack readings certainly indicated that the torque-limiter portion of the governor was not working or that the fuel linkage was improperly adjusted. If the governor and linkage is per FM instruction book, Section L, the fuel rack will be limited to 55 percent torque at 450 rpm and allowed to increase 1 percent each 10 rpm up to 900 rpm. This set-up gives power and torque curves per Curve 8893CH on page B8 of the instruction book. Such a correct set-up will give maximum engine break-mean-effective pressures of 70 at 450 rpm, 89 at 600 rpm, 108 at 750 rmp and 127.3 and 900 rpm, or fuel rack readings of 5.1, 5.7, 6.3 and 6.9 respectively. It is important that the governor linkage be set-up to assure proper action of the torque limiting feature built into the governor. If this is not the case, the engine can be severely overloaded at low engine rpm. The engine turbocharger is not matched to the engine for high torques at low speeds. The torque curve of the engine allows adequate margin for good acceleration when the propeller is matched to the engine at 900 rpm, 100 percent torque. With the torque limiter working properly, there should be no problems with black smoke due to overload. However, there may be many cases where the engine will not pick-up speed when operating as a single unit, plus it will take longer to go from 450 to 900 rpm engine speed. This overtorquing at low speed cannot be tolerated by a turbo-charged engine in the manner which a naturally-aspirated unit can. The naturally aspirated unit, theoretically, has the same air available per cycle at all engine speeds. Whereas, in a turbo-charged engine, the cylinder air capacity will double as the air pressure approaches 15 psig due to increases in air manifold pressure, thus allowing the engine to produce twice as much torque by burning more fuel per cycle. It can be shown that, even though the engine torque curve is as explained above, the particular engine, with a full size scavenging air blower, should suffer no operational problems, including smoke, up to about 85 psi bmep or 67 percent torque at any engine speed. And, in fact, since some charging is obtained from the turbocharger at all speeds, 78.5 percent torque or 100 bmep may be a reasonable limit. However, the engine should never be allowed to operate above 80 percent torque below 700 rpm. For an engine with the series blower, a review of data from the USCGC GALLATIN engine test in Appendix D shows a torque increase throughout the speed range with one shaft operation. It is estimated that the full-load fuel rack-setting would have been reached at 700 to 750 engine rpm. If the governor torque limiting is set up properly the fuel racks will reach rack 6 at about 725 engine rpm. If the governor torque limiting is inoperative, the engine can be overtorqued at 725 rpm. Also, every time a speed increase is called for, the engine will be overloaded until the new speed setting stabilizes. The overfueling will also produce a dark exhaust color until adequate air flow is developed by the turbocharger or the fuel racks return to their normal steady-state setting.

Since the turbo-blower series engine requires less fuel for full load, it is advisable to reduce the setting of the torque limiting governor to limit maximum fuel to no more that rack 7 as shown in the instruction booklet. With the turbo-blower series engine, the blower size is reduced by about 30 percent in capacity and acts as the second stage in the scavenging air supply. At low speeds and loads, the air flow will be approximately 25 to 30 percent less than with the older style scavenging system. This helps accomplish the desired results at lower engine loads. The USCGC GALLATIN test results give a minimum pre-turbine temperature of 590°F at 500 rpm with a fuel rack reading of 2.0 (less than 25 percent torque). At 50 percent torque, around 650 rpm, the exhaust temperatures were up to 750°F.

When operating under part load and speed conditions, the propeller law requires that the engine power requirement decrease rapidly as engine speed is reduced. With the present engines, the air flow remains proportional to engine speed up to about 85 percent torque. This type operation leads to lube oil fouling of the exhaust system at a low engine torque. Low engine torques allow the exhaust gas temperature to drop below the flash temperature of lube oil, and the lube oil builds up in the exhaust system. The Coast Guard issued a letter in 1969 which directs periodic increases in engine load for a period of about 30 minutes per each twelve hours of operation. The letter recommends full pitch at low engine speeds to accomplish the desired results. This statement is correct since the engine air is a function of engine speed at these speeds and loads.

5.11 BEARING FAILURES

Since premature bearing failures were reported by some of the cutters interviewed, all action possible should be taken to minimize this occurrence. As some of the failures were attributed to human error, training of personnel is most important. Again the use of the FM school is encouraged. Also, additional valve interlocking and warning lights could be provided in the oil flow circuit. Further, gradual degradation of the aluminum bearings should be apparent in a well run lube oil spectrographic analysis program.

5.12 OTHER PROBLEM AREAS

Other problem areas such as leakage, serviceability, exhaust manifold gaskets and seals are covered in more detail in Appendix A.

At least one cutter reported problems with injection pump tappet rollers. Some of these parts were made of defective material. This problem has been resolved by improving the material quality and should correct itself as these parts are replaced. However, this is a highly stressed component in nearly all diesel engines and failure can also be caused by dirt or high injection pressures.

REFERENCES

1) E.A. Kasel and C.L. Newton, "U.S. Coast Guard Pollution Abatement Program: Icebreaker Smoke Reduction," Report No. CG-D-179-5, November 1975.

2) E.A. Kasel et al, "Field Tests of In-Service Modifications to Improve Performance of an Icebreaker Main Diesel Engine," Report No. CG-D-8-77, August 1977.

APPENDIX A

REPORT AND DISCUSSION OF

INTERVIEWS AND VISITS TO SHIPS

CONTRACT NO. DOT-TSC-905

Colt Industries Customer Order No. 11-938642

FM 38TD8-1/8 Performance and Smoke Improvement

378' High Endurance Cutters (WHEC's)

MONTHLY PROGRESS REPORT - FEBRUARY & MARCH 1977

Prepared by: E. A. Kasel

E. A. Kasel

U. S. DEPARTMENT OF TRANSPORTATION CONTRACT NO. DOT-TSC-905

Modification No. 5

FM 38TD8-1/8 Performance and Smoke Improvement

378' High Endurance Cutters (WHEC's)

Monthly Progress Report - February & March 1977

Visited four U.S. Coast Guard High Endurance Cutters (378 class) as part of Contract DOT-TSC-905, Modification number 5.

USCGC Dallas (WHEC 716) Mr. Dennis Purvis
USCGC Gallatin (WHEC 721) Mr. Mike Goodwin
USCGC Hamilton (WHEC 715) Mr. Frank Tintera
USCGC Sherman (WHEC 720) Mr. Paul Hagstrom

The USCG Cutters Dallas and Gallatin were visited with the District Engineering Officier, Lt. F.L. Ames, while the cutters Hamilton and Sherman were visited with Mr. Bob Walter of the Department of Transportation.

The main propulsion engines in the 378' WHEC are Fairbanks Morse turbocharged twelve cylinder opposed piston Model 38TD8-1/8 units rated 3627 BHP at 900 RPM. The engines are connected by clutch to a variable pitch propeller thru a 6:1 reduction gear.

The engines were manufactured by Fairbanks Morse between the time frame of 1964 and 1970. The engines scavenging system is blower-turbo-intercooled for part load operation and turbo-intercooler for full load operation. An air inlet check valve arrangement located at the turbocharger air inlet allows some air to flow directly from the engine air supply to the turbocharger air inlet at full load. The air inlet check valve around the blower opens when the exhaust energy to the turbocharger is sufficient for the turbocharger air compressor to demand more air flow than the engine driven blower capacity. The engine blower capacity is sufficient for 75 to 90% rated torque. The air inlet check valve is designed to open in this torque range and allow more air to flow directly to the turbocharger air inlet.

These engines as built by Fiarbanks Morse included the following major power components.

-12 cylinder blower - 34 inches long.
-blower drive ratio - 2:013:1
-turbochargers - Elliott H-56, 18 sq. in. nozzle ring.
-cylinder liners - turbo-diesel.
-piston - rotating design WHEC 715-723
- fixed design (Mexican Hat) WHEC 724-726

-Compression rings - barrel faced - 16704845
-oil rings - standard production (2 per piston)
-injection cams - .468 inches lift
-injection pumps - 5/8 inch., V.E. helix
-injection nozzles - 15^{o} pintle - WHEC 715-723
- 10^{o} pintle - WHEC 724-726
-exhaust gas screens - changed to production cone design
-water keep warm - 110^{o} to 140^{o}F.

Some of the problem areas reviewed during these visits with personnel of the Cutters are listed and elaborated on below. Also included in this text is input from the Main Diesel Engine Questionnaire Survey of all 378' Cutters and a review of FM files on Service Repair work and Service Parts records of the subject engines.

1. Cylinder liner to jacket seal leakage at the exhaust port end. Nearly all engines have had liner seal leaks, one engine has had 31 liners replaced due to seal leakage. Several cutters now demand liners be resealed with high temperature O-rings.

2. Exhaust manifold gaskets leak liquids and exhaust gases. Exhaust manifold joints warp and come loose along with deterioration of the flexitallic gaskets. The copper gasket installed at the turbocharger four pipe flange last about one year. The initial proposal by FM for using a metal spacer at this joint was to allow the four exhaust passages to be sealed by flexitallic gaskets 16702533 in conjunction with copper flange spacer 16107551. The capscrews for the joint were changed from grade 2 to grade 5 and torqued 50 to 55 ft-lbs. If the copper spacer is oxidizing due to temperature a mild steel spacer per drawing 16107551 could be used in place of the copper material. The main function of the spacer is the carry the joints mechanical loading and give a definite installed compression to the flexitallic gaskets.

   The exhaust gas leaks cannot be tolerated by the engine since this energy is then lost to the turbocharger. It is felt that the majority of the engine room smoke is coming from vapor fumes off the external hot surfaces. If the manifold joints loosen they will leak at start-up and continue to leak during operation. The leakage is generally most visible during start-up since the exhaust may have a white color following engine start-up. This exhaust leakage must not be allowed to continue except under emergency conditions, since this is a method of waste gating turbocharger energy. The more the leakage the higher the exhaust temperatures will become at loads above 75% torque and the more difficult will be the exhaust manifold sealing problem. The resultant higher exhaust manifold temperatures will lead to further warpage of the manifold joints and relaxing of the joint fasteners.

3. Failure of exhaust manifold flexible connectors is generally associated with misalignment or extremely high exhaust manifold temperatures. The flexible bellows connectors all have an internal bellows guide which limits the axial movement within an acceptable range. Manifold extension pipes must fit at assembly without being forced to bring on centerline or rotated to line bolt holes. The pipes are assembled with a tension in the bellows to give optimum flexible connector load at operating temperature. High exhaust temperatures will lead to premature failure of the bellows due to the hotter environment and excessive thermal expansion of the exhaust manifold components.

4. Exhaust manifold associated fires are caused by liquid leakage from the exhaust piping joints and engine leakage from areas above the exhaust manifold. Most fires occur at the front of the engine where leakage from the front cover drips onto the hot heat shields and manifold. It seems that leakage from the front of the engine warrants further investigation by FM. If leakage cannot be eliminated in this area it appears that a low cost leakage collector system could be applied to the front of the engine.

As mentioned in Item 3, exhaust manifold leakage must be kept to a minimum for good engine performance. Engine operational procedures help control the excessive oil in the system at engine start-ups but oil continues to accumulate in the exhaust system during engine warm-up and light load operation. This oil then leaks thru the manifold joints causing engine room smoke and a fire hazard when the engine load is increased. Use of the latest design exhaust manifold gasket groove dimensions and surface finishes will reduce leakage from these joints provided joint distoration does not take place due to operation with high exhaust gas temperatures. The latest production groove design allows the joints to pull up metal to metal, the flexitallic gasket compression is then 15 to 25% of its thickness. Excess joint loading is taken by the metal to metal contact of the flanges. With the old designed groove the gasket takes the entire joint loading which eventually leads to gasket collapse and complete loss of gasket flexibility. These gaskets have a maximum safe operating temperature. The production gasket drawing specifies 1100°F operating temperature and the type 347 stainless steel used in the gasket manufacture is rated at 1600°F maximum.

5. Injection compartment fuel leakage is reported as a fire hazard when fuel sprays from the injection compartment, leaks thru the water jumper opening in the cylinder block or if the injection compartment lower deck becomes extremely hot from the high exhaust manifold temperatures and exhaust manifold leakage. Injection compartment leakage is generally from:

   a. Injection pump
   b. Injection nozzle
   c. Drain line breakage and injection tube

Injection pump leakage along the fuel racks is generally an indication of failure of the lapped surface between the pump body and barrel or excessive plunger to barrel clearance. Other possible leakage areas would be the fuel header to pump body gasket and discharge valve cage gasket. Generally it requires a new gasket to repair leakage at these areas.

The injection nozzle leakage can occur at the fuel inlet fitting gasket or needle stop gasket and generally requires a new gasket to repair leakage in these areas. Best results at the needle stop gasket is obtained by assembling with freshly annealed gasket. After dipping all internal parts in clean fuel oil snug assemble by hand, then torque in one sweeping motion to 45 to 50 ft-lbs. After this initial torquing, if the sleeve is disturbed it will generally result in gasket failure. Also, if either of these copper gaskets leak during engine operation, leakage will generally worsen if you try to tighten it.

Injection tube leakage is either due to insufficient tightness of the fittings or at the nozzle end a crack in the ferrule sleeve and requires replacing the sleeve or installing a new injection tube to repair same. Two wrenches should be used, one at the nozzle fitting and one on the tube nut, when tightening the fuel line or the fuel inlet fitting gasket may be damaged.

Leakage due to failed fuel drain lines is usually caused by overtightening the drain line fittings. At least one vessel reported stripping the threads on the brass banjo fitting for the drain line. The fitting should only require good snugging for proper seal.

Another possible area of fuel leakage is the thread on the nozzle opening pressure adjustment screw. This can be eliminated by using a thread sealant between the adjustment screw and spring sleeve when servicing the nozzle assembly.

At least one of the cutters has converted to the latest production gasketless nozzle assembly. Service kit P/N 16609071 includes nozzles, collars, injection tubing and drain lines for one cylinder, figure 1. This assembly has no gaskets exposed to high pressure fuel no external pressure adjustment screw and virtually eliminates high pressure fuel leakage.

The latest production engine incorporates a water cooled barrier below the injection compartment deck. This barrier is integral with the water passages from the exhaust belts into the cylinder liner. This design also seals the cylinder block opening where the water jumper passes thru the injection compartment deck. Laboratory test recorded up to a maximum of 200°F lower injection compartment deck temperatures with the water cooled heat shielding. Catalog number 2115, figure 2. This feature is available for field retrofit and offers reduction in fire hazards.

6. Scavenging air leakage occurs at the two rectangular slip joints in the air ducting between the turbocharger and air coolers. This joint has been redesigned for production engines and is depicted in figure 3. Part numbers of new parts required are shown. This new design is being used successfully in engines operating with air box pressures up to 25 PSIG, and can be easily field retrofitted.

7. Oil leakage into the cylinders is reported occurring on some engines during shut down and idle operation. It is suggested that oil is still leaking past the upper pistons due to failure to bar the engine over one half hour after shut down, or excessive oil is entering this cylinder during the initial thirty minutes after shut down. Other areas where oil can enter the engine during shut down and slow idle include liner to cylinder block seals due to deterioration or damage during installation and the injection pump tappet housing to upper crankcase fit, just below the camshaft. Earlier engines relied on a metal to metal fit to seal lube oil into the crankcase and air into the air receiver at the point where the tappet housing passes thru the crankcase.

   A production change was made in May of 1968 to include an o-ring seal at this interface. In fact the WHEC 724-726 may have these particular seals. Oil leakage at this point would show up as an oil washing of the tappet housing inside the engine air receiver. This seal can be added to earlier engines by machining the groove in the tappet housing and breaking the sharp edge of the cylinder block to allow assembly without shearing the o-ring. The seal application is depicted on figure 4 and the groove details are shown on figure 5.

   It was noted that at least one of the engines visited had only two oil rings assembled to the upper pistons. To slow the accumulation of oil in the exhaust system during idle warm-up and light load operation it is recommended that the third oil ring groove be fitted with a 16101191 oil drain ring.

8. Lube oil leakage at the exhaust belts to cylinder block fit is becoming a problem again. The accepted solution for earlier engines was to add two drilled hold down capscrews with oil shields to allow this leakage to be pulled back into the engine crankcase before it reached a depth where it would overflow its built in dam. It shows up as oil coming from off the block deck below the exhaust manifold and running down between the lower crankcase doors and accumulating on the block mounting rails. Figures 6, 7, and 8 show the hardware and modification. Laboratory experience with this modification shows that eventually these capscrews become clogged with dirt and must be removed to clean the oil drain hole and the dirt accumulated in the cylinder block deck capscrew hole.

   The leakage is due to a humping tendency of the exhaust belt with temperature causing a gap to develop between the block and the exhaust belt on the centerline of the engine. This gap is then impinged with high velocity lube being thrown from the crankshaft bearings. The result is sufficient oil pressure to cause oil to leak outward across the horizontal gap.

Production engines have an o-ring installed at this joint. Again the WHEC 724-726 more than likely had this ring installed during engine manufacturing. This o-ring can be added to earlier engines by machining the o-ring groove in the lower face of the exhaust belt. The o-ring is the same as used on the lower liner to belt seal which is supplied with all new cylinder liner kits.

The exhaust belt to cylinder block o-ring groove dimensions are .102 to .110 inches groove depth with 9.51 I.D. and 9.86 O.D. This seal application is depicted on figure 9, catalog 4018.8.

This would be the proper time to also modify the exhaust belt to manifold gasket groove to the depth of the present production engines. Groove depth .100 to .105 inches with 4.32 I.D. and 4.78 O.D.

The liner to exhaust belt seal furnished with new liner kits should be used in all turbocharged engines. The only precaution needed is to make certain the leading edge of the exhaust belt fit is adequately broken to prevent shearing of the o-ring at assembly. Later exhaust belts incorporate a 30 degree lead in at this fit. This chamfer could be added when making above modifications to the exhaust belts.

9. Leakage at the front of the engine is from various capscrews, covers and areas where shafts pass thru the cylinder block and front cover. One consistent leak is the seal at the end of the fuel control shaft. A new seal (P/N 16106089) will fix this leak. To prevent further deterioration of this seal a metal heat shield should be attached to the cover capscrews to prevent direct manifold heat radiation to the fuel control shaft and seal. Most of the front end oil leaks can be repaired with sealant and careful application of covers.

10. Water leaks besides the liner seals mentioned earlier include adapter copper gaskets, adapter o-rings and exhaust belt triangular gaskets.

Generally adapter copper gaskets leaks will be eliminated if the instruction book is followed. It is important that the threads and gasket surfaces are properly cleaned and free of nicks and pits. The copper gasket must be annealed, properly centered on the adapter and held in place with grease, beveled side towards the adapter. Invert the adapter to be sure the gasket stays in place. Use adapter installation tools and snug with a hand wrench. Again as with the injection nozzle try to torque the adapter with one sweeping motion. The copper actually flows and work hardens during the torqueing action.

Premature failure of the adapter o-rings generally indicate excessive cooling water temperatures. In certain engine applications there has been some deterioration problems with the inner o-rings used on the indicater valve adapter. A high temperature o-ring is available for this seal at a cost premium of about 100%. The ring is made of viton material, P/N 16704654 and replaces o-ring 16701279. Viton for the other adapters o-ring would be P/N 16704650 replacing P/N 16701278 and P/N 16704651 replacing P/N 16701280.

Since these engines have been built the cylinder liner adapter material has been changed from cold rolled steel to a stainless steel. This new adapter is much more corrosion-resistant and has better manufacturing control on the copper gasket seat machining. The stainless steel parts are automatically furnished by the service department when new adapters are ordered.

The triangular gasket used to seal the water jumpers on the exhaust belts has caused many leaks in the past. A new gasket design released in 1974 has eliminated this problem. The new gasket is directly interchangeable with the older part and carried the same part number 16102122. Figures 10 and 11 show the new gasket and instructions for proper application of this gasket.

The engine front cover and exhaust piping design is a very common engine servicing complaint. The front cover cannot be removed from the engine without removing the turbocharger exhaust gas outlet piping. The only other possible way to remove this cover is with the upper crankshaft. It is also difficult to remove the upper crankshaft without taking the front cover with it. In either case the front cover must be loosened from the engine block to remove the upper crankshaft.

Because of the design it is difficult to change the engine overspeed setting, replace or tighten the timing chain, balance the timing between sides of the engine, or clean the crankcase breather screens. It seems that with the design of the turbo blower series air piping minor design changes could relocate the exhaust piping to give adequate work clearance.

12. At least one ship wanted to know why these engines had 12 cylinder air start valves instead of 6 used on stationary applications. The engine was reported to start much easier with only 6 valves plus require less starting air. One ship has converted to six air start valves and some are adding extra air compressor capacity. The twelve valves are used on all Marine engines as a standard production item by Fairbanks Morse. The extra valves assure that even with one valve mulfunctioning there wuuld be no position at which the engine could stop where the air would not bar it over for starting. This extra starting reliability is quite important for marine engines. The six valves are adequate to start the engine provided the Coast Guard Cutters can tolerate a possible situation where the engine may not bar over on command. Since these engines are not reversible it may seem to be desirable to block off six of these valves. Every other valve in the firing sequence of the engine would be made inoperative by blocking the air start distributor openings and disconnecting the air start valve pilot line, cylinder liner dummy plugs are available from FM. However, loss of starting reliability under all maneuvering conditions would result and only the Coast Guard can evaluate the risks versus the return.

13. Loss of crankcase vacuum is generally an indication of a mal function in the engine. The malfunction should be located and corrected. Other than a cracked piston the loss of vacuum will generally be associated with engine scavenging air pressure leaking into the engine crankcase. These leaks can occur at the turbocharger and blower seals or between the air receiver and upper crankcase deck. The other possible source would be down thru the exhaust belt along the cylinder liner fit.

The leaks at the air receiver or exhaust belt would be due to liner to block seal deterioration or damage at assembly. Again some of the earlier engines did not have a seal between the injection pump tappet housing and the cylinder block or the cylinders liner and the exhaust belt. These parts can be modified to allow application of the production seals.

With the advent of higher air supercharging pressures in the O.P. engine it became necessary to rework the crankcase ejector system to assure adequate crankcase ejector capacity. The change includes removing all restrictions from the in flow passages to the ejector and reoptimizing the ejector nozzle configuration. The larger flow passages included opening the 1-1/4 inch cored hole in the ejector body to 1-3/4 inches and using gaskets with holes of at least 1-1/2 inch diameter. The new ejector nozzle has a smaller hole and extends further into the ejector body. Figures 12 and 13 show the ejector body modification and the new nozzle. These parts are automatically furnished by the service department if new parts are ordered. Part numbers remain the same.

The new ejector does not eliminate the need for preventive maintenance towards keeping the crankcase leakage at normal levels for a given engine. Anytime there is a reduction in crankcase vacuum an investigation to locate the cause must be made and corrective action taken.

14. The Coast Guard lowered the oil sump levels per FM recommendation. This change reduced the oil consumption to about one half of that experienced prior to the change. The change was made by marking the level stick as a funtion of normal ship rake and a certain level of oil at the center of the oil pan.

There was some concern as to whether the lower lube oil level could in any way be related to the bearing failures due to loss of oil to the pump suction at engine start-up or in rough sea operation. It does not seem that this would be the case, however, a recording of the engine lube oil pump discharge pressure along with engine rpm during typical rough sea operation may be desireable.

Bearing failures on the Hamilton and Sherman were due to running the engine with a ruptured strainer on the Hamilton and with no lube oil on the Sherman.

15. A common complaint was that the engines would not come up to speed or if they did, only with excessively high exhaust temperatures. Temperatures in the 1100 to 1200°F were mentioned as extremes and 1050°F as common.

These high exhaust temperatures are due to either engine overload or engine performance deterioration. The overload will always be indicated by the fuel rack position which should follow the curve of Figure 14 for an engine in good operating condition. Since these are turbocharged engines and the turbochargers are matched for the maximum power and speed point (3627 HP @ 900 RPM) full load rack reading should only occur at 900 RPM. If the propeller pitch has been increased at low engine speeds to keep the exhaust temperatures above 600°F the pitch must be decreased to normal setting before running higher engine speeds. As stated earlier, generally the engine blower will put out sufficient air for 75 to 90% torque at any engine speed. The highest exhaust temperatures will occur at the speed and load where the air inlet check valve is just ready to open.

Engine performance deterioration can include many items, such as, engine settings, fuel injection pumps, injection nozzles, turbochargers, air coolers, air leaks, exhaust leaks, pistons, piston rings as well as characteristics of the fuel used in the engine.

To provide greater reliability, cooler operation, longer life and reduced fire hazard the turbo blower series scavenging system may be desirable. This scavenging system increases the amount of excess air available to the engine at the higher loads. Besides making the deterioration margin larger the rate of deterioration is reduced because of the cooler normal operating temperatures. The exhaust system remains tight much longer, and the power parts generally have a lower rate of normal wear.

16. These engines have pistons of both the rotating type and Mexican Hat type. Vessel WHEC 715 thru 723 were built with rotating pistons while WHEC 724 thru 726 were furnished with the Mexican Hat fixed type pistons. It seems that most of the cutters are changing from the rotating piston to the Mexican hat piston. The change is being made piecemeal (one cylinder at a time) and by sets.

The latest production piston is of fixed version having a combustion chamber configuration similar to the rotating pistons. The upper and lower piston and insert are identical. The latest production piston has proven to be much more durable than either of the earlier pistons. The newer piston has excellent internal oil cooling surfaces, giving low ring temperature and longer ring life.

It is recommended that all new pistons be of the new style when changing pistons either by cylinder or set change out. Figure 15, catalog no. 5.6, shows the parts needed to install the latest style production pistons.

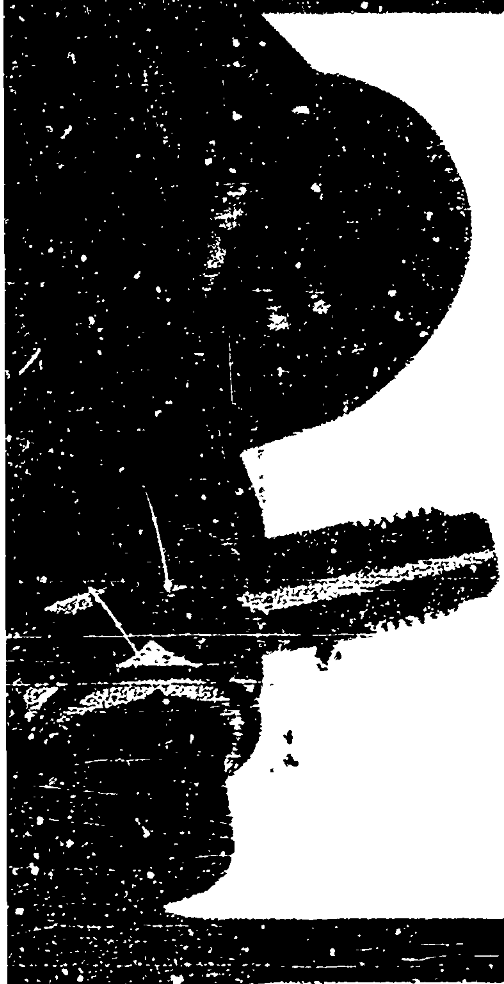
The new piston requires that the injection nozzle assemblies have 15 degrees holders. This is the angle of the spray tip relative to the centerline of the holder. The latest engines in the fleet were built with nozzles having 10 degree holders. The original nozzle assembly internal parts were the same for the 10 and 15 nozzles. The 10 degree holder is P/N 16201441 while the 15 degree holder is P/N 16200805. Since some of the cutters are changing to the gasketless nozzle the fleet should have available extra 15 degree nozzle holders as well as assembly components. The 10 degree holders should not be used with the new style fixed piston assemblies. These new style pistons are the correct parts for engine conversion to the turbo blower series scavenging systems.

GASKETLESS FUEL INJECTION NOZZLE ASSEMBLY

FOR FAIRBANKS MORSE
OPPOSED PISTON ENGINES

NEW...

for high
fuel pressure
operation!



- Nozzle assemblies available for diesel, turbo diesel, dual fuel and turbo dual fuel.
- Elimination of 3 gaskets as high pressure seals.
- Replaceable on most engines in the field.
- Factory warranty on every replacement part.
- Precision manufactured parts insure engine operating economy.
- Designed and built by skilled experts who built your engine originally.

FIGURE 1.

A-13

Colt Industries

Fairbanks Morse
Engine Division
BELOIT, WISCONSIN 53511

File 3[illegible]

Continuing engine developments keep your Fairbanks Morse Opposed Piston Engine operating efficiently and economically.

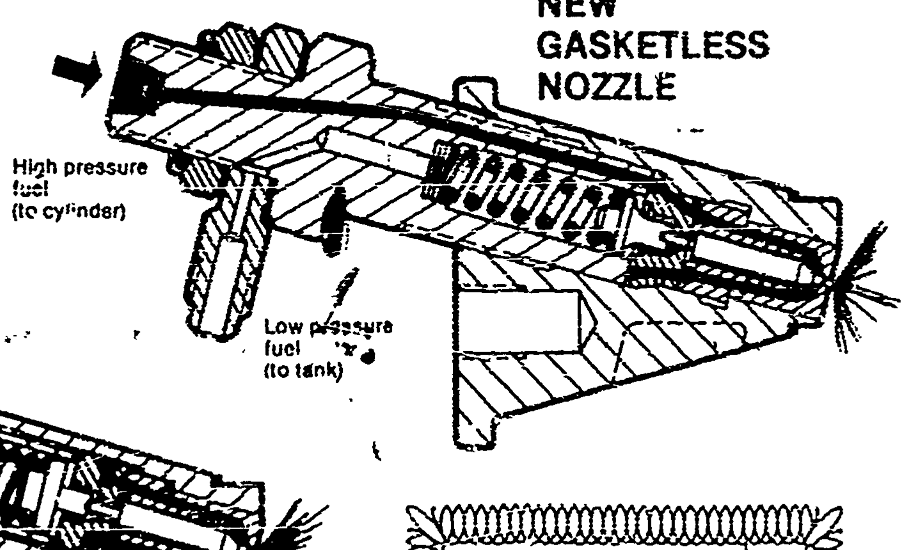
Metal to metal sealing surfaces withstand high pressure!

The new gasketless nozzle uses high pressure metal to metal sealing surfaces with matching faces, precision ground and lapped. This method offers the ultimate in a leak proof seal and withstands extremely high pressure when properly torqued.

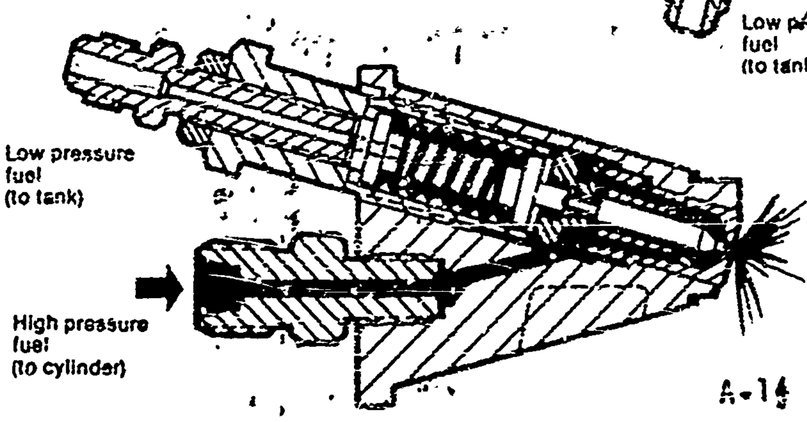
Three copper gaskets which were used for high pressure seals, have been eliminated in this new design. This practically eliminates any possibility of leakage of fuel. This new nozzle unit is completely replaceable on most engines in the field and offers fewer parts, direct fuel flow, less maintenance and increased tip life can be expected. When replacing nozzle tip in the field no special tools are required.

Your periodic nozzle maintenance schedule can also be lengthened because these highly machined surfaces offer fewer maintenance problems in sealing high pressure fuel than copper gaskets. By replacing your present nozzles you are assured of a longer period of consistent fuel delivery to all cylinders. You get improved engine operation through correct fuel quantity to each cylinder on every pumping stroke.

NEW GASKETLESS NOZZLE



OLD DESIGN

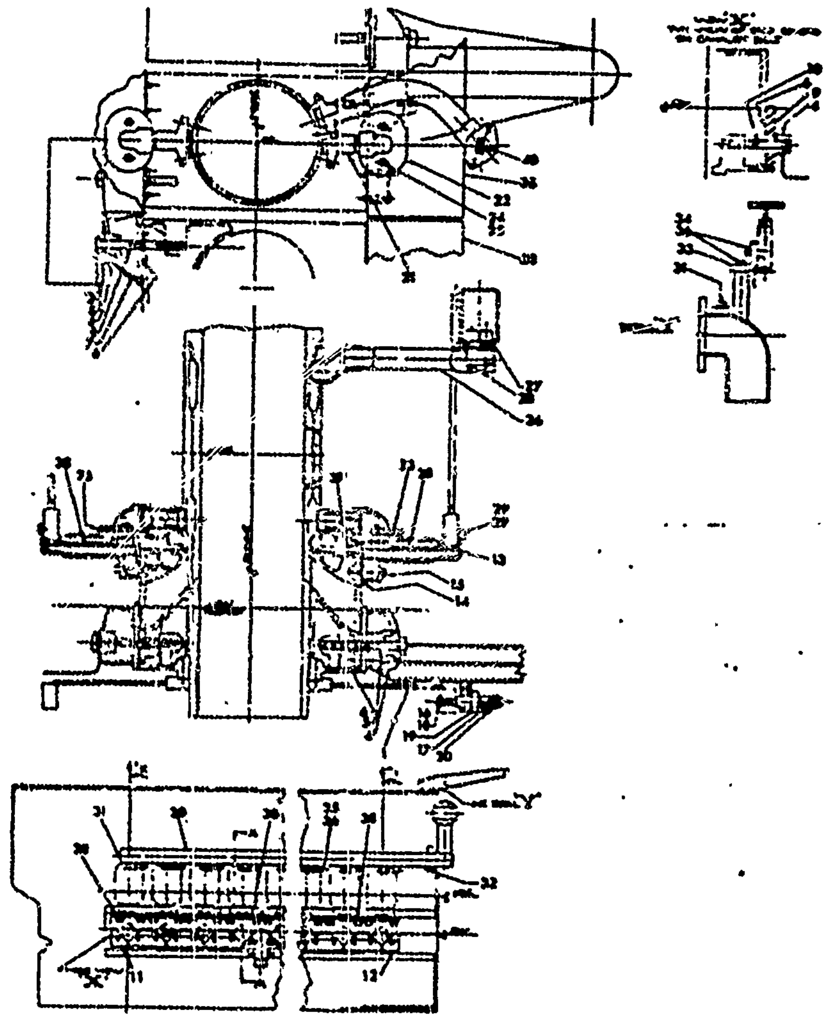


FULL WARRANTY

A full factory warranty on every replacement part · for your protection

PIECE NO.	NAME OF PART	PART NUMBER	NUMBER USED
1	FITTING, Inlet water	16 102 230	2
2	WASHER, Seal	16 107 849	40
3	CAPSCREW 3/8-16x1	81 340 310	40
4	GASKET, water by-pass fitting	16 102 [illegible]	40
7	FITTING, water by-pass	16 207 093	20
8	HEAT SHIELD, lower tower	16 107 777	20
9	STRIP, exhaust belt	16 103 319	8
10	CAPSCREW 3/8-16x1	81 343 336	4
11	HEAT SHIELD, end	16 104 223	2
12	HEAT SHIELD, end	16 110 232	2
*13	TUBE, Cooling Deck	16 1[illegible] 867	24
*14	GASKET	16 559 [illegible]	24
*15	CAPSCREW 3/8-16NF-7/8	81 345 879	96
16	GASKET, Header Flange	16 701 734	4
17	OUTLET, water Inlet Header	16 187 244	4
18	CAPSCREW 1/2-13x1	81 387 021	4
19	NUT C-16	81 376 974	4
20	NUT 1/2-13	81 184 [illegible]	4
*21	FITTING, Inlet	16 201 206	24
*22	ELBOW, Inner water Inlet	16 704 973	24
*23	NIPPLE	16 103 [illegible]	24
*24	CAPSCREW 3/8-16x1	81 301 [illegible]	48
*25	WASHER 3/8	81 365 870	48
26	PIPE, Inner Outlet to Header	16 404 501	12
27	GASKET	16 702 287	12
28	CAPSCREW 3/8-16x7	81 387 019	24
*29	WASHER 3/8	81 389 793	24
30	HEADER, water	16 403 375	1
31	PLUG, Pipe 1	17 837 [illegible]	2
32	PLUG, Pipe 1/2	17 031 [illegible]	1
33	BRACKET, Water Header	16 102 287	1
34	CAPSCREW 3/8-20x1-1/2	81 387 017	2
35	CAPSCREW 1/2-20x1-3/4	81 302 [illegible]	2
36	WASHER 1/2	81 301 336	11
*37	"O" Ring	16 708 [illegible]	12
*38	DECK, Cooling	16 [illegible]	48
*39	CAPSCREW 3/8-24x7/8	81 302 [illegible]	4
*40	PLUG, Pipe 3/8	92 608 [illegible]	24

*Conversion Items



WATER HEADER WITH COOLING DECK

FIGURE 2

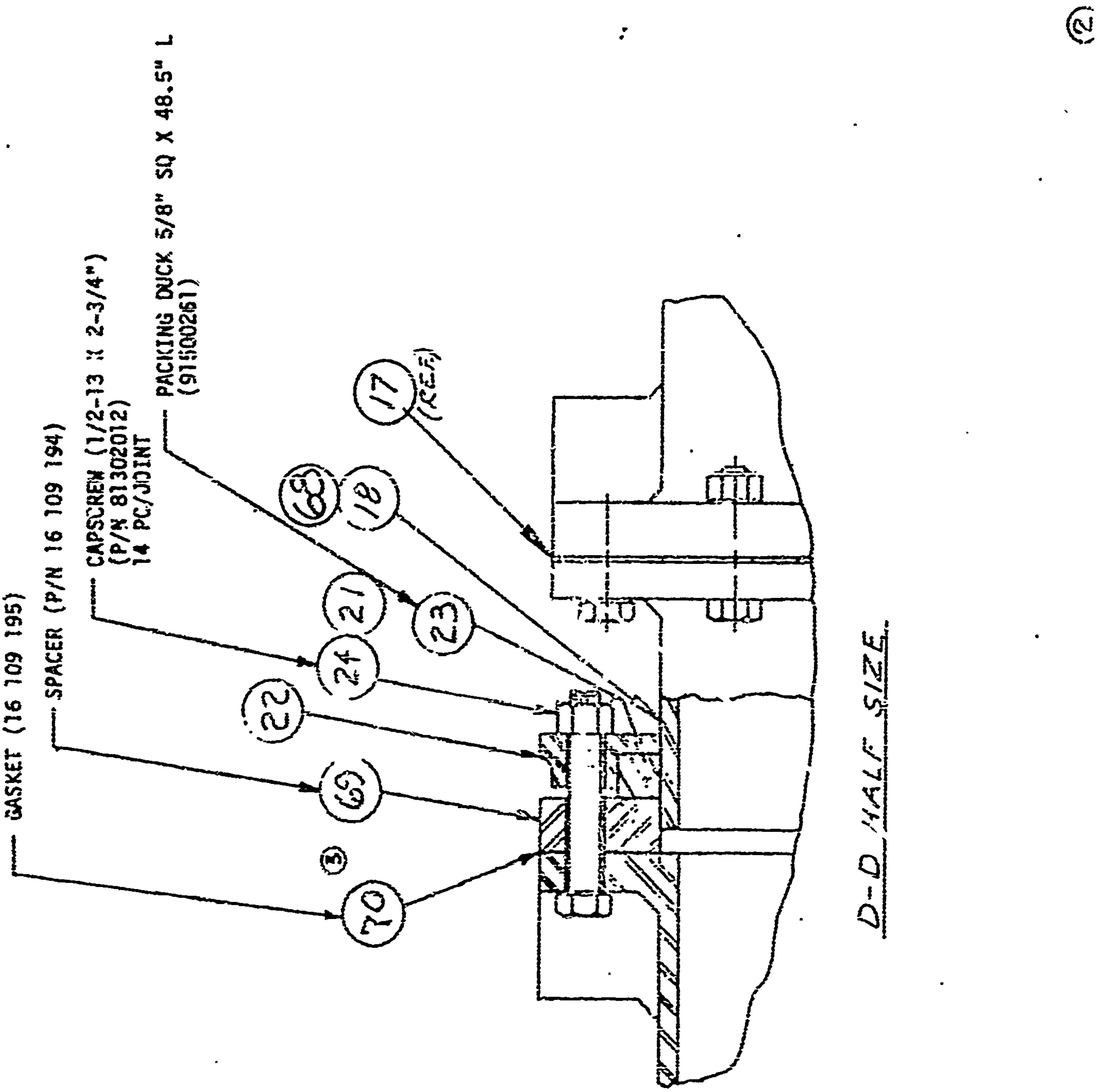
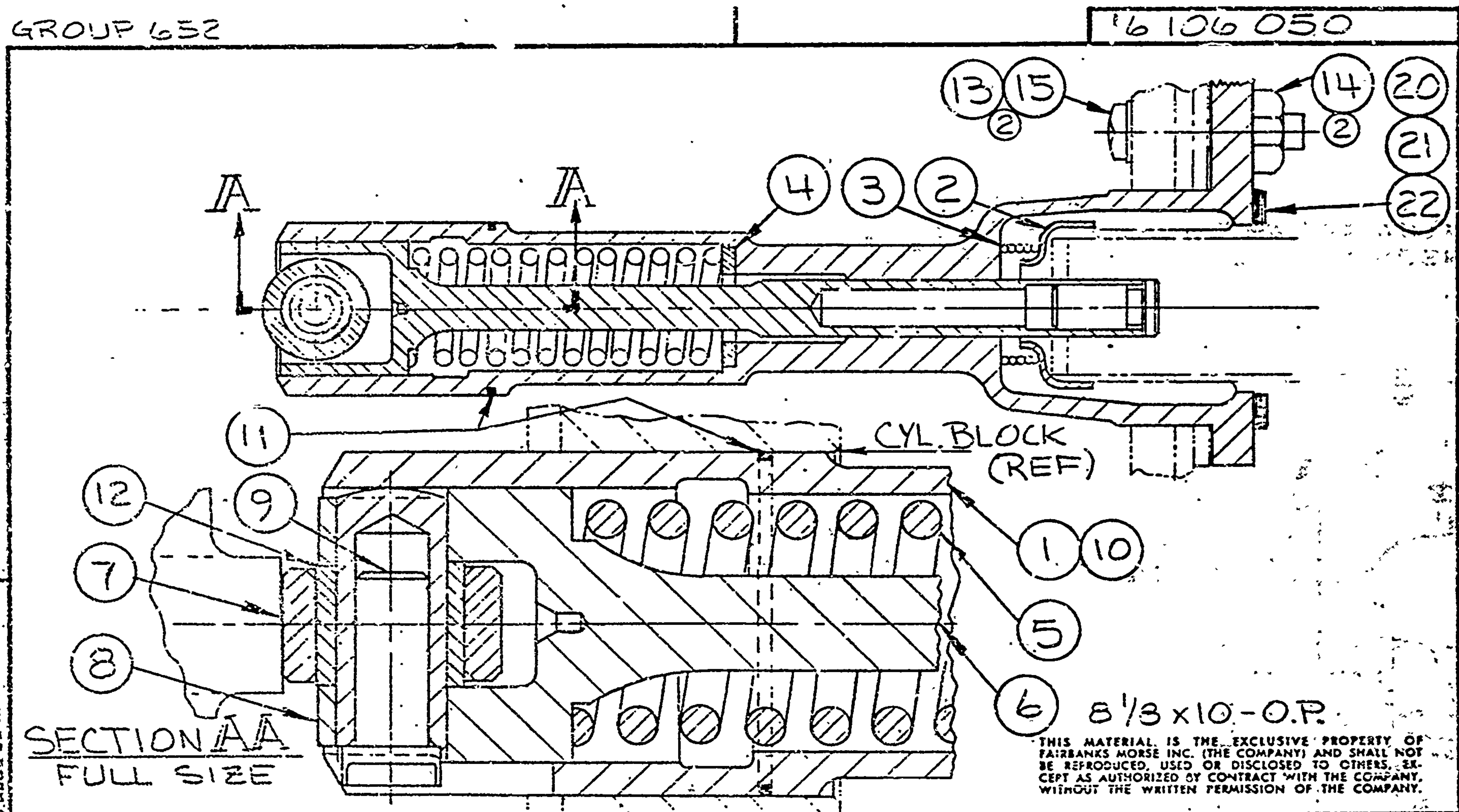

GASKET (16 109 195)
SPACER (P/N 16 109 194)
CAPSCREW (1/2-13 X 2-3/4")
(P/N 81302012)
14 PC/JOINT
PACKING DUCK 5/8" SQ X 48.5" L
(91500261)
17
(REF)
68
18
21
23
24
22
69
70
3
D-D HALF SIZE
2

FIGURE 3



REV NO	CHANGE ORDER NO	DATE	DESCRIPTION	CHK'D. BY
			REDRAWN I.D.I. NO CHANGE (REF)	BJC
2	8334	4-19-3	ADDED ITEMS 14 & 15 & CAPSCREW & NUT IN VIEW	JH 4-17-73
1	8106	5-4-72	ADDED NEW ITEMS 20, 21 & 22	MW

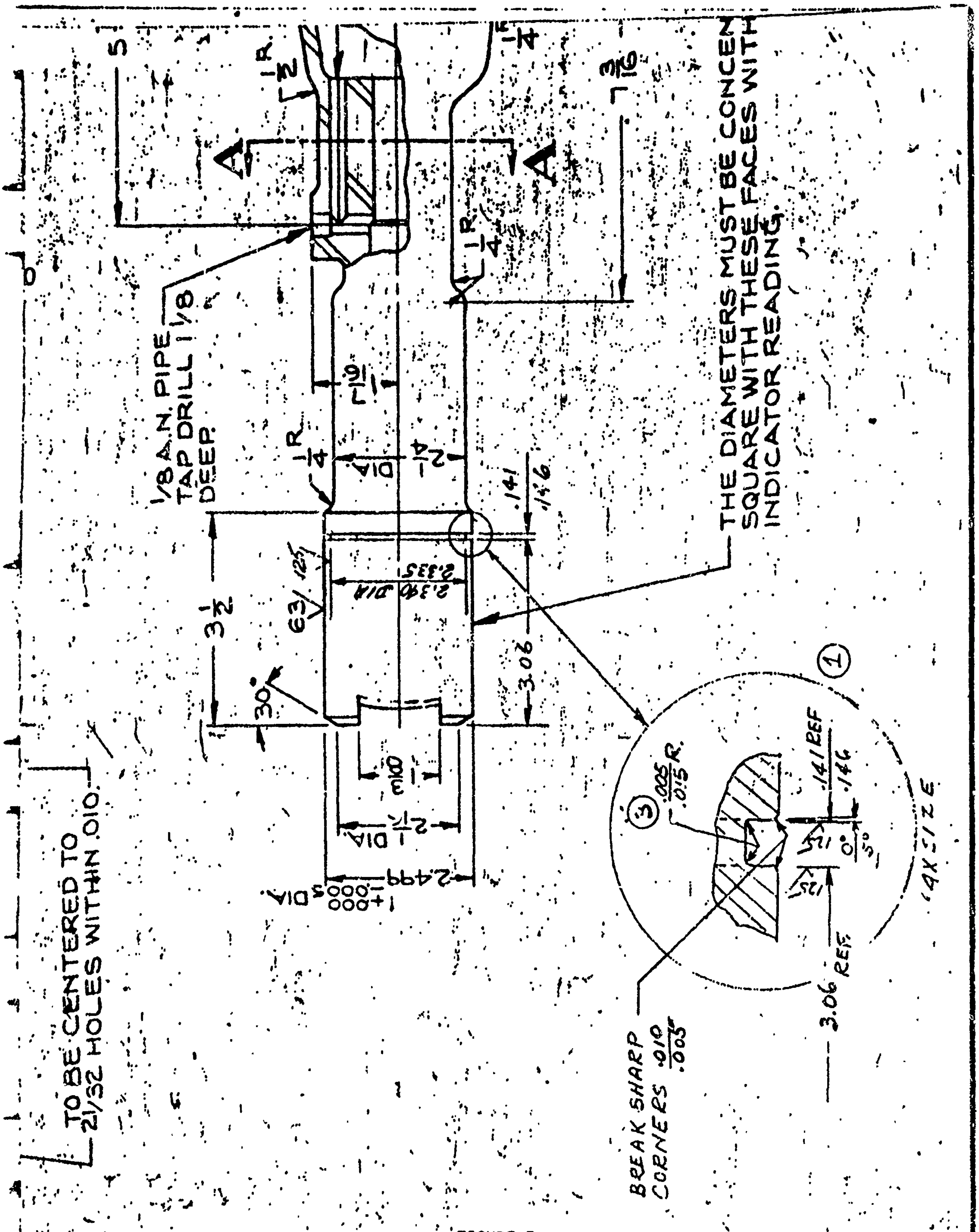
TITLE
TAPPET ASSEMBLY

TOLERANCES
(Unless Otherwise Specified)
FRACTIONAL MACHINING ± 1/64
TOOL DESIGN ± .002
FLAME CUTTING, SHEARING, NIBBLING, FORMING AND WELDING ± .060
DECIMAL MACHINING TOLERANCES
0. ± .060 0.0 ± .030 0.00 ± .010

Colt Industries — Fairbanks Morse Engine Division

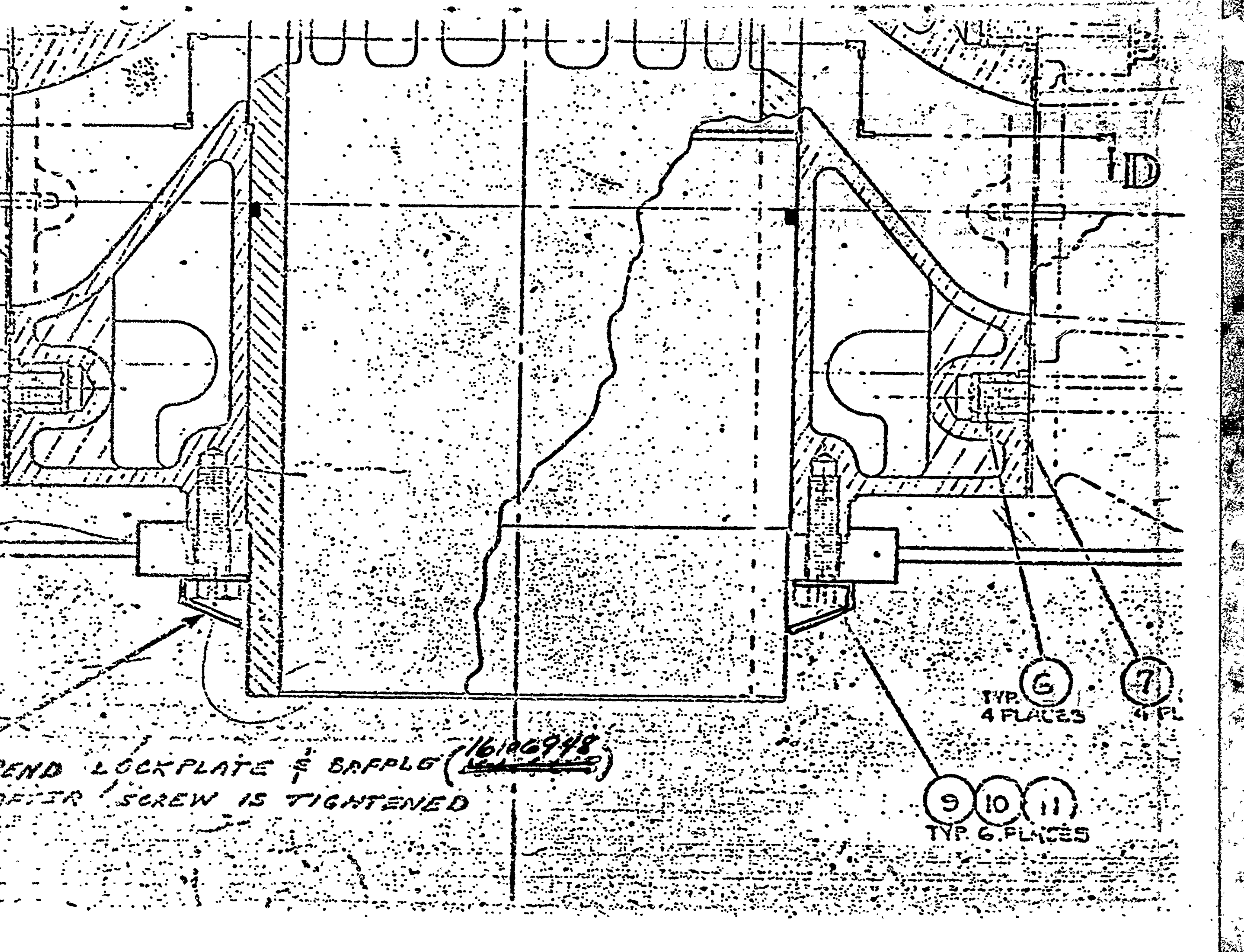
REL. NO. 0392	12/16/64	ASSY. OR LO.	
DR. BY KWS	9/22/66	SCALE 1/2 & 1/1	
CHK'D. BY R.M.	9/27/66	HEAT TR.	
MAT'L SEE DETAILS			
APPV.	CAST.	MAT'L.	
	MACH.	ENGR.	
	WELD.		
SHEET SIZE A	DWG. NO. 16 106 050	SHEET OF	

FIGURE 4
A-17



TO BE CENTERED TO
21/32 HOLES WITHIN .010.
1/8 A.N. PIPE
TAP DRILL 1 1/8
DEEP
THE DIAMETERS MUST BE CONCEN
SQUARE WITH THESE FACES WITH
INDICATOR READING.
A
A
3 1/2
3.06
30°
2.340 DIA
2.335
63/125
1 R/4
2 1/4 DIA
2 1/2 DIA
2.499
1.000 DIA
.141
.146
3/16
5
BREAK SHARP
CORNERS .010/.005
3.06 REF.
.141 REF
.146
.005/.015 R.
10X SIZE

FIGURE 5



6
TYP. 4 PLACES
7
4 PL
9 10 11
TYP. 6 PLACES
BEND LOCKPLATE & BAFFLE (16106948)
AFTER SCREW IS TIGHTENED

FIGURE 6

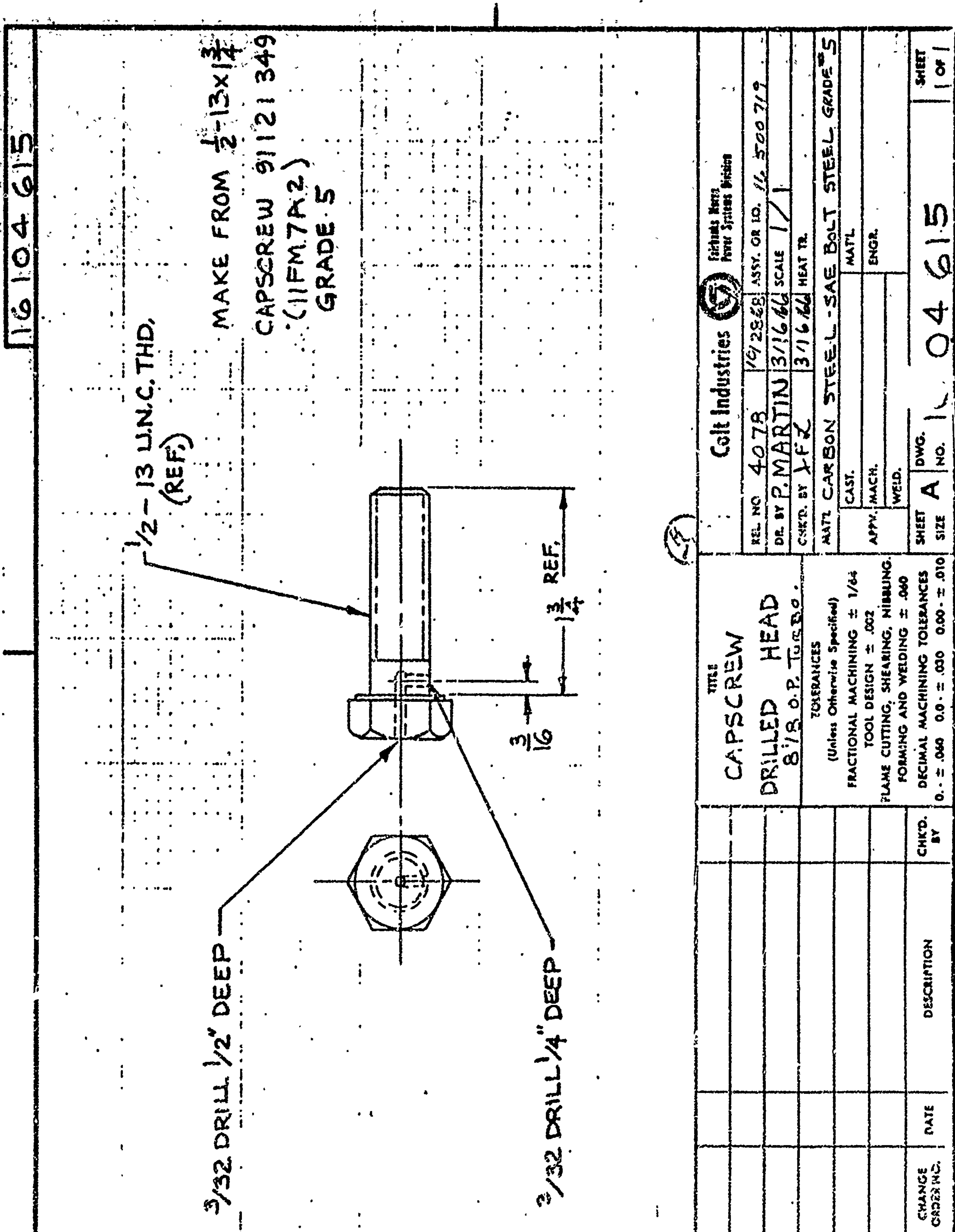
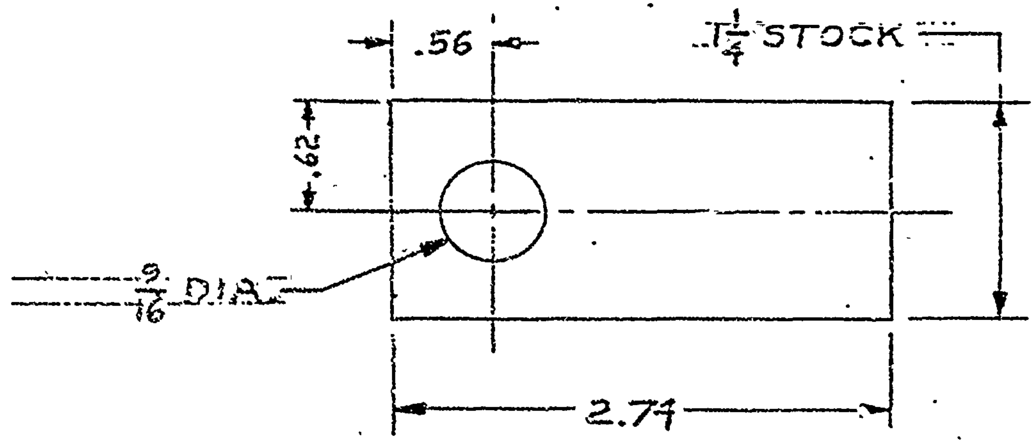

16 104 615
MAKE FROM 1/2-13x1 3/4
CAPSCREW 911 21 349
(11FM7A.2)
GRADE 5
1/2-13 U.N.C. THD.
(REF.)
1 3/4 REF.
3/16
3/32 DRILL 1/2" DEEP
3/32 DRILL 1/4" DEEP
Colt Industries
Fairbanks Morse Power Systems Division
REL. NO 4078
10/29/68
ASSY. OR I.D. 16 500 719
DR. BY P. MARTIN
3/16/66
SCALE 1/1
CHK'D. BY
3/16/66
HEAT TR.
MAT'L CARBON STEEL - SAE BOLT STEEL GRADE #5
CAST.
MACH.
WELD.
APPV.
MAT'L
ENGR.
TITLE
CAPSCREW
DRILLED HEAD
8 1/8 O.P. TURBO
TOLERANCES
(Unless Otherwise Specified)
FRACTIONAL MACHINING ± 1/64
TOOL DESIGN ± .002
FLAME CUTTING, SHEARING, NIBBLING, FORMING AND WELDING ± .060
DECIMAL MACHINING TOLERANCES
0. ± .060 0.0 ± .030 0.00 ± .010
SHEET SIZE A
DWG. NO. 16 04 615
SHEET 1 OF 1
REV. NO.
CHANGE ORDER NO.
DATE
DESCRIPTION
CHK'D. BY

FIGURE 7

16106943



#20 GA. (.0359) STOCK THICKNESS

REV NO	CHANGE ORDER NO.	DATE	DESCRIPTION	CHK'D. BY

TITLE
WASHER-LOCK
CYL. BLOCK TO
EXHAUST BELT

3½ Tuac? TOLERANCES
(Unless Otherwise Specified)
FRACTIONAL MACHINING ± 1/64
TOOL DESIGN ± .002
FLAME CUTTING, SHEARING, NIBBLING, FORMING AND WELDING ± .060
DECIMAL MACHINING TOLERANCES
0. ± .060 0.0 ± .030 0.00 ± .010

Colt Industries — Fairbanks Morse, Power Systems Division

REL. NO. 4078 | 10/28/69 | ASSY. OR LO. 16500719
DR. BY DEK | 10/11/69 | SCALE FULL
CHK'D. BY | / / | HEAT TR.
MAT'L STRIP ST. 1010-T4
APPV. CAST. | MAT'L
MACH. | ENGR.
WELD.
SHEET SIZE A | DWG. NO. 16106948 | SHEET OF

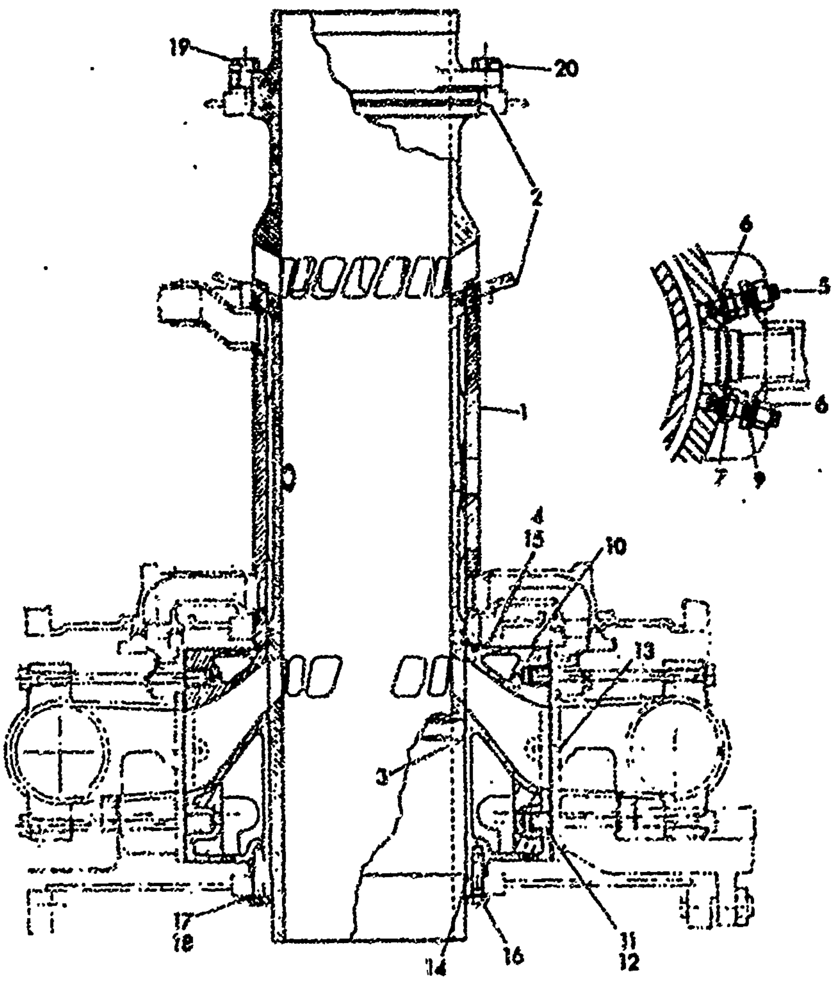
FIGURE 8

4018.10 - Rev.

TD Marine

PIECE NO.	NAME OF PART	PART NUMBER		* NUMBER USED
1	Cylinder Liner - Service Kit (Pc. 1-9)	16 607 672		1
2	RING, Liner to Block	16 102 902		2
3	RING, Oil Seal	16 106 469		1
4	RING, Exhaust Belt Seal	16 101 262		1
5	STUD, Assembly, Water Connection (Pc. 5-7) . . .	16 104 860		6
6	THREADSEAL	16 106 518		6
7	NUT, Water Connection Stud	81 328 591		6
8	NUT, Water Connection Stud	81 344 094	3/8-16	6
9	WASHER, Water Connection Stud	16 101 325		6
10	Exhaust Belt - Assembly (Pc. 10-15)	16 605 431		1
	BELT - Not furn. sep.	16 702 306		1
11	INSERT, Manifold Capscrew	16 701 263		4
12	LOCKRING, Insert	16 701 266		4
13	DOWEL, Exhaust Belt	16 101 320		4
14	RING, Oil Seal	16 106 469		1
15	RING, Exhaust Belt Seal Ring Support	16 102 412		1
16	CAPSCREW, Exhaust Belt	91 121 349	1/2-13x1-3/4	6
17	LOCKWASHER, Exhaust Belt Capscrew	16 101 324		6
18	WASHER, Exhaust Belt Capscrew	81 309 830		6
19	CAPSCREW, Cylinder Liner to Block	91 125 658		4
20	LOCKPLATE, Cylinder Liner to Block Capscrew . .	16 107 826		4

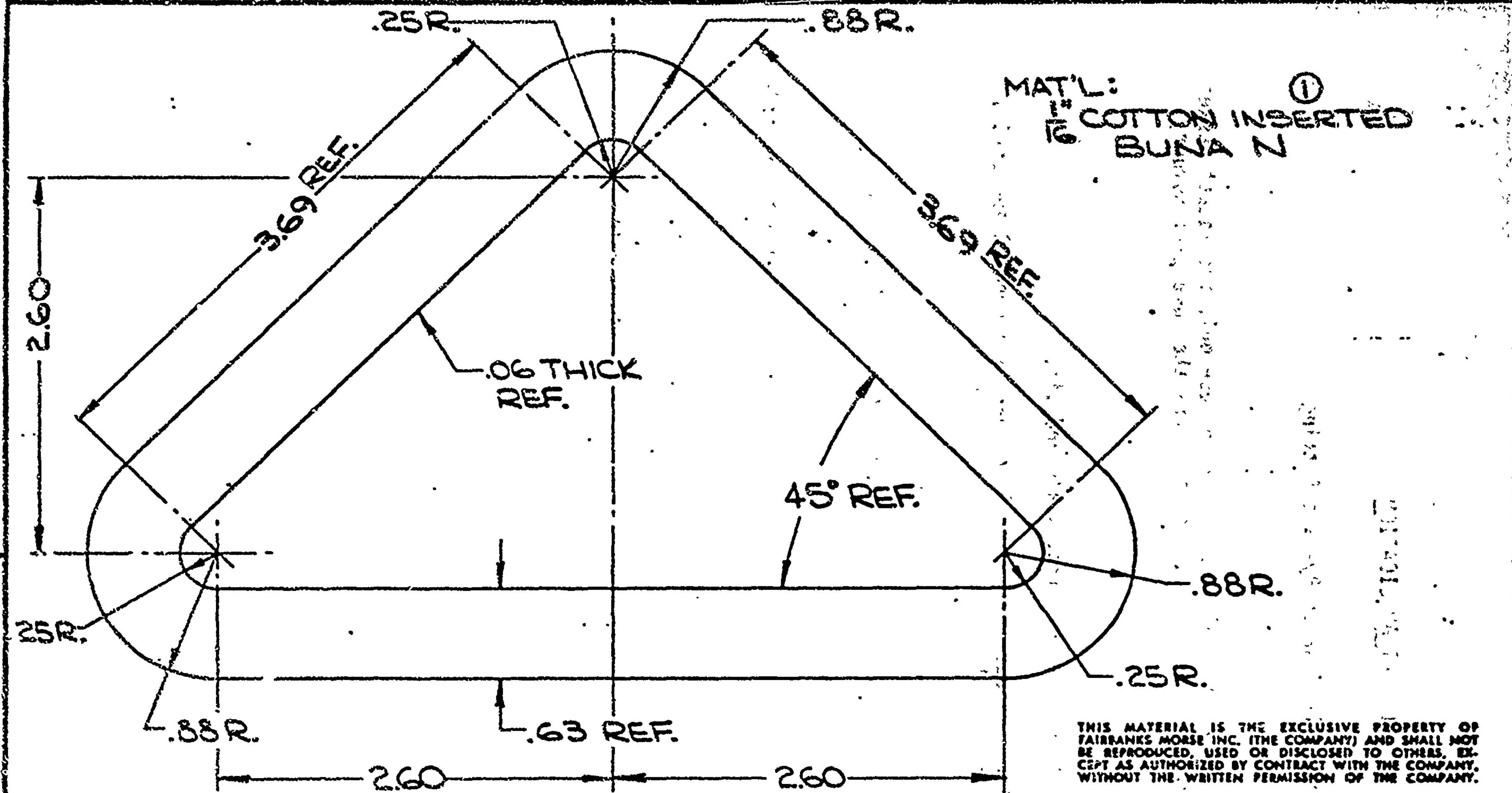
* Per Cylinder



CYLINDER LINER

FIGURE 9

16 102 122



REV. NO.	CHANGE ORDER NO.	DATE	DESCRIPTION	CHK'D. BY
2	9175	9-24-74	TITLE WAS "GASKET WATER JUMPER ELBOW	JJB
2	"	"	① WAS 1/16" COTTON IMPREGNATED.	
1	9143	9-6-74	REVISED AND REDRAWN	JJB

TITLE
GASKET
WATER BY-PASS
FITTING

TOLERANCES
(Unless Otherwise Specified)
FRACTIONAL MACHINING ± 1/64
TOOL DESIGN ± .002
FLAME CUTTING, SHEARING, NIBBLING, FORMING AND WELDING ± .060
DECIMAL MACHINING TOLERANCES
0.- ± .060 0.0- ± .030 0.00- ± .010

Colt Industries — Fairbanks Morse Power Systems Division

REL. NO. ———	/ /	ASSY. OR LO.
DR. BY G.U. - IDI	8/27/74	SCALE FULL
CHK'D. BY JJB	9/4/74	HEAT TR.
MAT'L SEE NOTE - PURCHASE		
APPV. CAST. / MACH. / WELD.		MAT'L / ENGR.
SHEET SIZE A	DWG. NO. 16 102 122	SHEET 1 OF 1

872009-2

FIGURE 10
A-23

GASKET APPLICATION

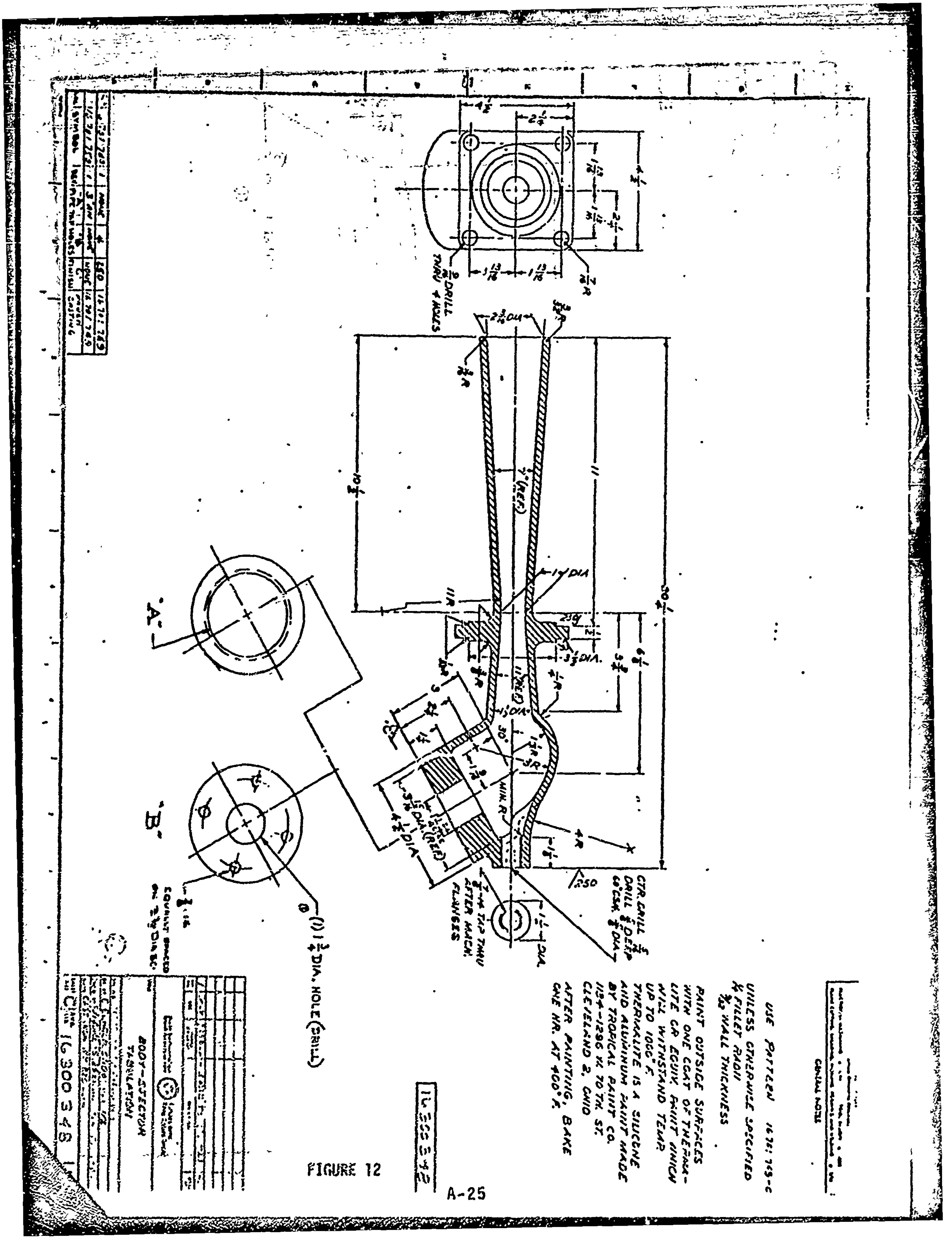
Exhaust Belt Water By-Pass Fitting

Do not use the gasket with a 90° angle cross section, use only the latest gasket with flat cross section. The new and old gaskets carry the same part number, namely 16102122.

Apply the new gasket as follows:

1. Clean all gasket surfaces free of dirt, scale and oil.
2. Coat gasket area of the exhaust belt with "Goodyear" Pliobond 20 industrial adhesive or equivalent. Apply light coat with brush.
3. Firmly apply gasket to adhesive before it dries.
4. Clean all threads. Chase threads on belt studs with a 5/8-11 die nut.
5. Apply new "O"-ring to gland nut.
6. Lubricate the "O"-ring seat by applying a light coating of grease to the hole in the water by-pass fitting.
7. Lubricate the stud to nut threads and nut bearing area with molycote or equivalent.
8. Torque nuts with heat shields under the nut head, 30 to 35 ft.-lbs.

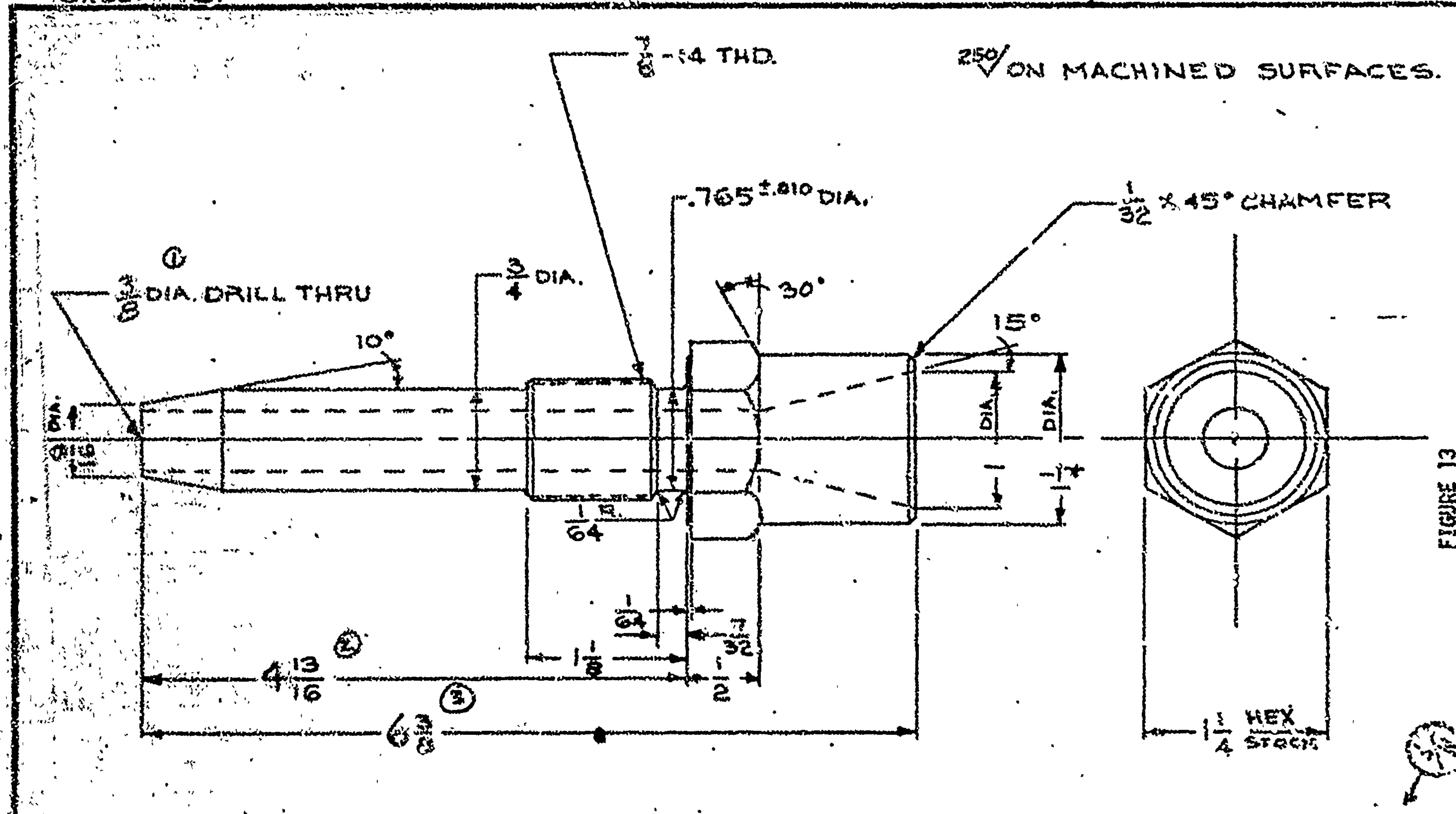
FIGURE 11



USE PATTERN
UNLESS OTHERWISE SPECIFIED
FILLET RADII
WALL THICKNESS
PAINT OUTSIDE SURFACES WITH ONE COAT OF THERMA-LITE OR EQUIV. PAINT WHICH WILL WITHSTAND TEMP UP TO 1000°F.
THERMALITE IS A SILICONE AND ALUMINUM PAINT MADE BY TROPICAL PAINT CO. 1134-1296 W. 70 TH. ST. CLEVELAND 2, OHIO
AFTER PAINTING, BAKE ONE HR. AT 400°F.
BODY-SECTION
TABULATION
1C 300 348

FIGURE 12

16 102 006



					TITLE NOZZLE-EJECTOR	Cox Industries		
1	9388	3-24-75	(3) WAS 5 3/8	WWK		REL. NO.	11/3/61	ASST. OR LG.
1	"	"	(2) WAS 3 13/16	WWK		DR. BY CONOVER	3/9/61	SCALE FULL
1	"	"	(1) WAS 7/16 DIA. HOLE	WWK	TOLERANCES (Unless Otherwise Specified) FRACTIONAL MACHINING ± 1/64 TOOL DESIGN ± .032 FLAME CUTTING, SHEARING, NIBBLING, FORMING AND WELDING ± 1/16	CHK'D. BY	3/13/61	APPV.
						MAT'L C.R. STL. 4FM1320		HEAT TR.
REV. NO.	CHANGE ORDER NO.	DATE	DESCRIPTION	CHK'D. BY		SHEET SIZE A	DWG. NO. 16102006	SHEET 01

FIGURE 13

Ref.: [illegible]
T.d.A.: 10-10-67

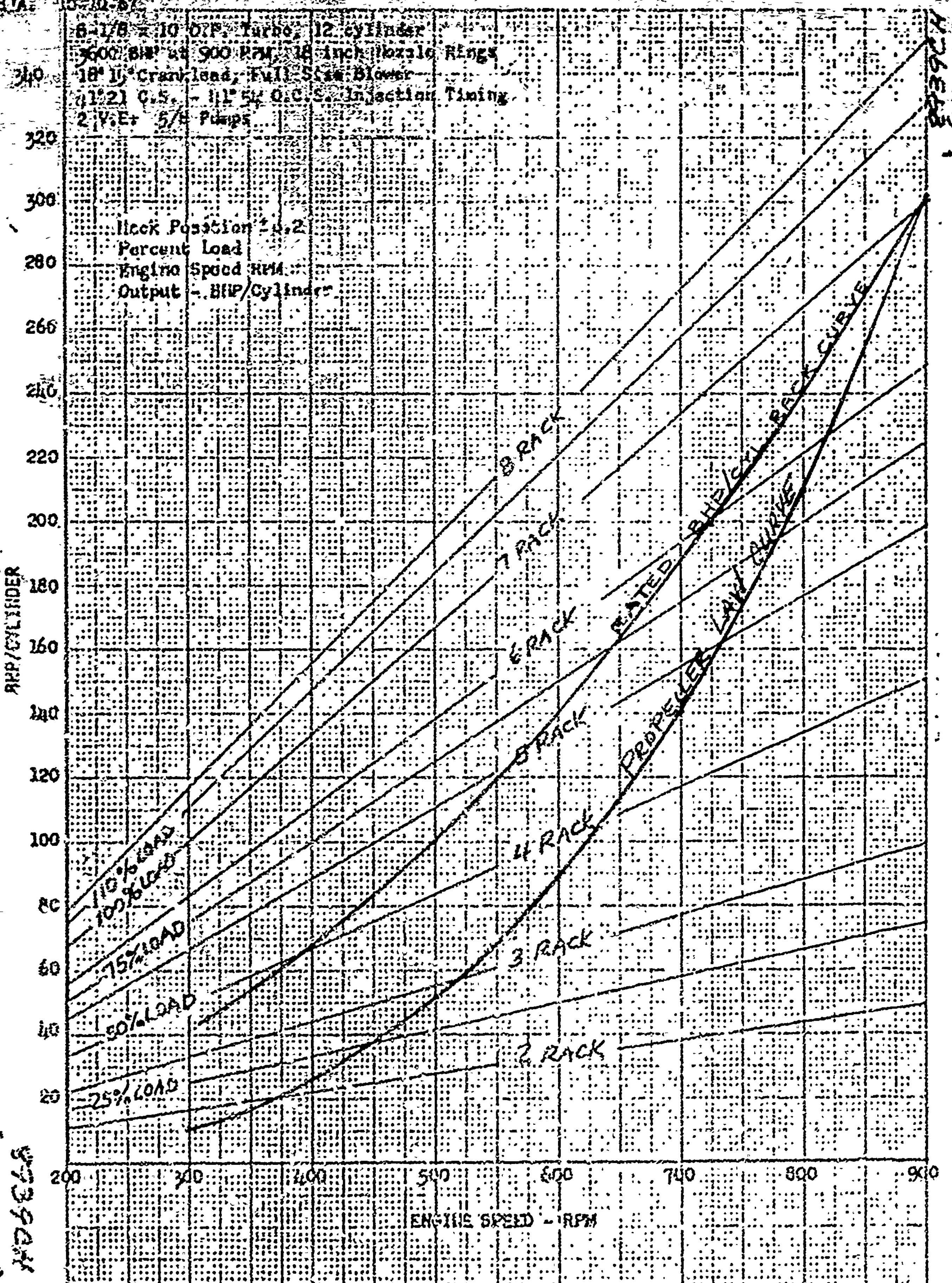


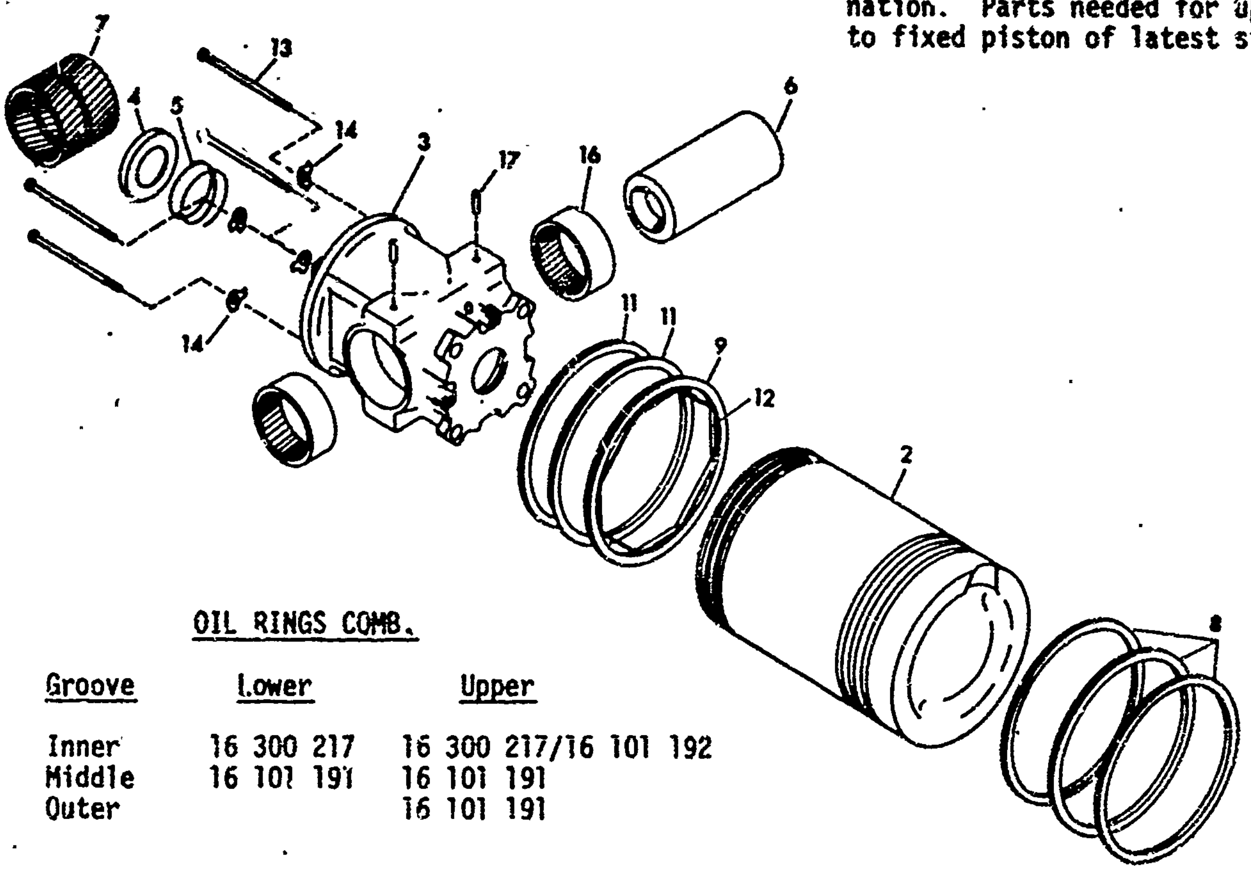
FIGURE - 14

(11206EH)
8739EH

PIECE NO.	NAME OF PART	PART NUMBER	* NUMBER USED
2	Upper Piston - Complete	16 608 301	1
2	PISTON, Upper	16 401 902 *	1
3	Piston Insert Assembly - (Pc. 3, 16 & 17)	16 608 129 *	1
16	BUSHING, Piston Insert	16 300 358	2
17	PIN, Bushing Lock	16 101 188	2
4	RETAINER, Oil	16 101 189	1
5	SPRING, Oil Retainer	16 101 190	1
6	PIN, Piston	16 200 274	1
7	BUSHING, Piston Pin	16 701 767	1
8	RING, Compression	16 704 845	3
9	RING, Oil Scraper	16 300 217	1
11	RING, Oil Drain	16 101 191	2
12	RING, Oil Scraper Expander	16 101 192	1
13	BOLT, Hex	16 704 740 *	4
14	LOCKPLATE	0014012.02.10*	6

* No. Used per Piston

*Upper & lower piston assy. the same except for oil ring combination. Parts needed for up-date to fixed piston of latest style.



OIL RINGS COMB.

Groove	Lower	Upper
Inner	16 300 217	16 300 217/16 101 192
Middle	16 101 191	16 101 191
Outer		16 101 191

UPPER PISTON
or Lower

FIGURE 15

APPENDIX B

ORIGINAL ENGINE ACCEPTANCE TEST DATA

U.S. Department of Transportation Contract No. DOT-TSC-905

Modification No. 5

FM 38TD8-1/8 Performance and Smoke Improvement

378' High Endurance Cutters (WHEC's)

Monthly Progress Report - January 7, 1977

The shop test logs from fourteen of the subject engines were reviewed and the average performance data of each engine was tabulated on shop test logs. All data is at 900 rpm and was tabulated for each 25% increment of rating, including 110% load. The 100% load log sheet includes engine settings, such as turbocharger nozzle ring area, cranklead and injection timing.

The average data of these engines is plotted versus brake mean effective pressure and attached as Curve No. 56, sheets 1 and 2. The break noted in the data lines of the exhaust temperatures and scavenging air pressures is typical of the 38TD8-1/8 having the scavenging air blower working ahead of the turbocharger compressor. If you examine the scavenging air temperature to the turbocharger inlet, you will note the decrease in temperature with increased bmep. This temperature is a function of the ambient air temperature and the work of the scavenging air blower. The air temperature from the compressor has a break in its curve at the point where the scavenging air check valve opens. This valve opens when the displacement of the scavenging air blower is less than the engine requires, dictated by the energy to the turbocharger and the overall efficiency of the turbocharger. Above this point the engine scavenges without the aid of the scavenging air blower.

Engine serial no. 38D867070 was run through the speed range of 300 to 900 rpm at each 100 rpm increment during the shop test. These data are plotted as Curve No. 57, sheets 1 through 3. The data is plotted versus rpm and as constant torque lines.

Air manifold pressure in inches of mercury as a function of load and speed are shown on Curve No. 11205CH.

The average preturbo exhaust temperature in degrees Fahrenheit as a function of engine load and speed is shown on Curve No. 11207CH.

The engine brake specific fuel consumption obtained during shop test as a function of total engine horsepower and speed is shown on Curve No. 11204CH.

The engine fuel rack position as a function of engine total horsepower output at various engine speeds is shown on Figure 1 dated January 6, 1977. Lines of constant torque and fuel consumption are also plotted on Figure 1.

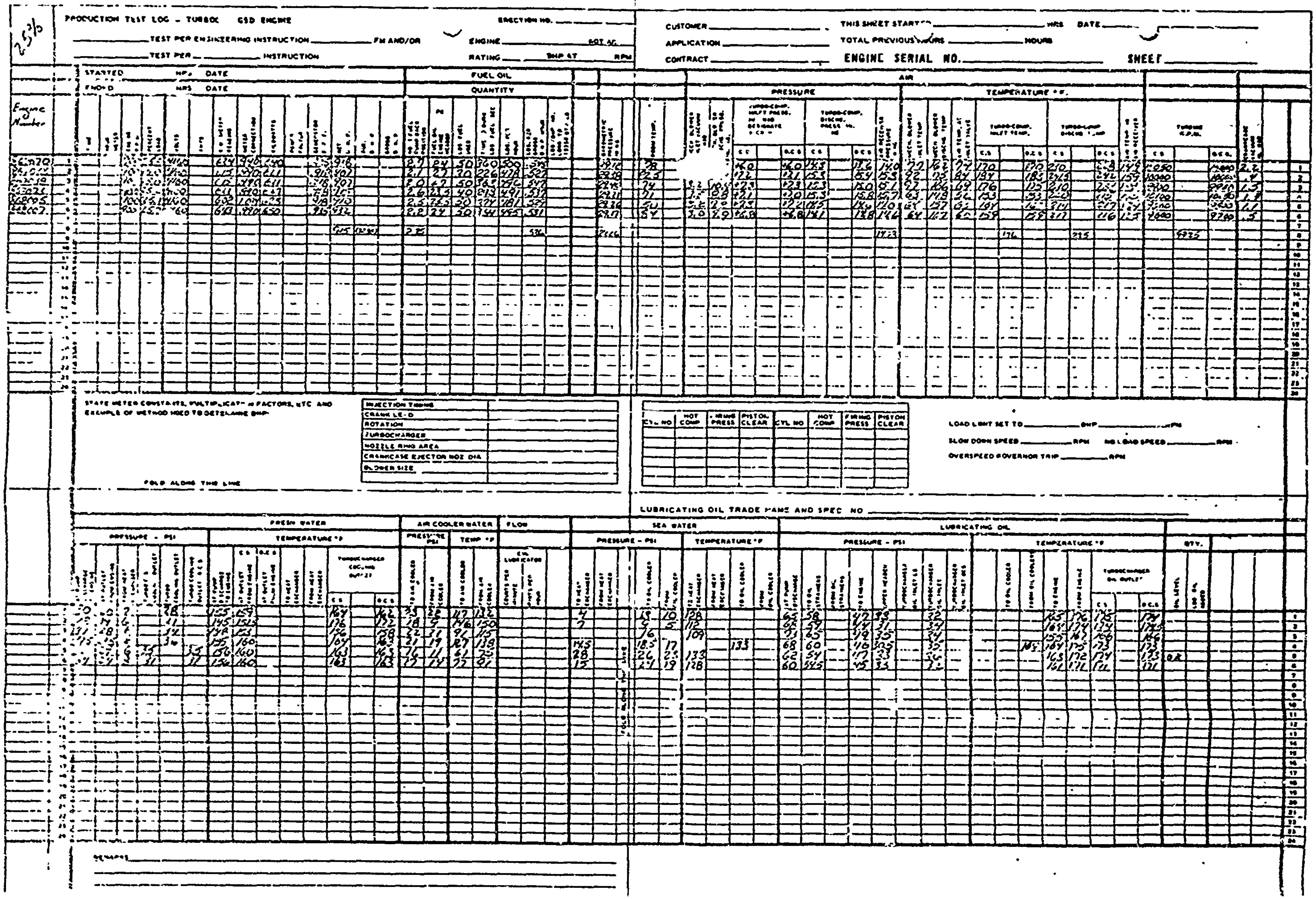


FIGURE B-1.

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	FRESH WATER											
	PRESSURE - PSI						TEMPERATURE °F					
								C.S.	TURBOCHARGER COOLING OUTLET C.S.	O.C.S.		
1	70	7	2		28		155	159	167	162	[illegible]	[illegible]
2	22	7	6		31		145	152	176	172	18	
3	37	28	6		34		155	151	176	158	32	
4	12	25	6		36		155	160	164	163	21	17
5	[illegible]	37	6	35		15	156	160	161	163	76	11
6	374	24	3	31		32	156	160	163	163	72	14

	SEA WATER								
	PRESSURE - PSI					TEMPERATURE °F			
1	4			18	10	128			
2	7			5	5	112			
3				76		109			
4	14.5			10.5	17			135	
5	28		3	26	2.5	135			
6	15		3	24	18	138			

	LUBRICATING OIL																
	PRESSURE - PSI								TEMPERATURE °F								QTY
													TURBOCHARGER OIL OUTLET C.S.		O.C.S.	OIL LEVEL	
1	65	58		47	35		32				185	176	178		174		
2	58	59		44	31		34				164	172	174		174		
3	73	65		49	35		34				155	167	166		166		
4	68	60		46	32.5		35			147	164	175	173		173		
5	62	54		47	33		36				165	172	174		172	10.4	
6	60	54.5		45	33		32				166	171	171		126		

REMARKS

FOLD ALONG THIS LINE

NATURAL GAS FUEL

	EXHAUST TEMP FROM CYLINDERS °F																	
	CYL. NO. 1	CYL. NO. 2	CYL. NO. 3	CYL. NO. 4	CYL. NO. 5	CYL. NO. 6	CYL. NO. 7	CYL. NO. 8	CYL. NO. 9	CYL. NO. 10	CYL. NO. 11	CYL. NO. 12	CYL. NO. 13	CYL. NO. 14	CYL. NO. 15	CYL. NO. 16	CYL. NO. 17	CYL. NO. 18
1	520	480	450	520	505	480	480	475	485.5	485.5	475	495						
2	420	426	452	454	470	458	440	412	446	533	552	484						
3	450	475	500	485	462	462	480	472	410	452	462	475						
4	498	470	471	476	458	476	522	512	470	506	507	465						
5	472	470	480	460	480	470	460	500	470	510	502	452						
6	413	495	485	460	480	498	495	475	475	505	505	455						

	COMBINED EXHAUST TO TURBO		TEMPERATURE °F TEMP FROM TURBOCHARGER			PRESSURE TO TURBOCHARGER IN. HG.			PRESSURE	
	CS °F	OCS °F	C.S.		O.C.S.	C.S.		O.C.S.		
1	535	530		450		10.7		10.1	1.8	
2	437	440		446		10.6		10.0	1.2	[illegible]
3	505	500		472		10.0		9.5	1.2	N/A
4	525	475		462		11.5		10.7	1.3	N/A
5	480	500		420		11.3		10.8	1.5	N/A
6	495	475		472		11.5		10.0	3.0	N/A
8	495			[illegible]			10.6			

FIGURE B-2

INSPECTION	TESTER
FIRST SHIFT	FIRST SHIFT
SECOND SHIFT	SECOND SHIFT
THIRD SHIFT	THIRD SHIFT

50%

PRODUCTION TEST LOG - TURBOCHARGED ENGINE

ERECTION NO ______

______ TEST PER ENGINEERING INSTRUCTION ______ PM AND OR

______ TEST PER ______ INSTRUCTION

ENGINE ______ RATING ______ BHP AT ______ RPM

CUSTOMER ______

APPLICATION ______

CONTRACT ______

THIS SHEET STARTED ______ HRS. DATE ______

TOTAL PREVIOUS HOURS ______ HOURS

ENGINE SERIAL NO. ______ SHEET ______

STARTED HRS. DATE

ENDED HRS. DATE

FUEL OIL

QUANTITY

AIR

PRESSURE

TEMPERATURE °F.

STATE METER CONSTANTS MULTIPLICATION FACTORS, ETC. AND EXAMPLE OF METHOD USED TO DETERMINE BHP

INJECTION TIMING	
CRANK LEAD	
ROTATION	
TURBOCHARGER	
NOZZLE RING AREA	
CRANKCASE EJECTOR NOZ. DIA.	
BLOWER SIZE	

FOLD ALONG THIS LINE

CYL. NO.	HOT COMP.	FIRING PRESS.	PISTON CLEAR.	CYL. NO.	HOT COMP.	FIRING PRESS.	PISTON CLEAR.

LOAD LIMIT SET TO ______ BHP ______ RPM

SLOW DOWN SPEED ______ RPM   NO LOAD SPEED ______ RPM

OVERSPEED GOVERNOR TRIP ______ RPM

LUBRICATING OIL TRADE NAME AND SPEC. NO. ______

FRESH WATER

PRESSURE - PSI

TEMPERATURE °F

AIR COOLER WATER

FLOW

SEA WATER

LUBRICATING OIL

REMARKS ______

FIGURE B-3.

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LUBRICATING OIL TRADE NAME AND SPEC. NO. ____________

Line	Fresh water pressure – psi						Fresh water temp. °F (in)	Fresh water temp. °F C.S. outlet	Turbocharger cooling outlet C.S.	Turbocharger cooling outlet S.C.S.	Air cooler water pressure psi (to cooler)	Air cooler water pressure psi (from cooler)	Air cooler water temp. °F (to cooler)	Air cooler water temp. °F (from cooler)
1	97	20	7		23		121	122	167	162	21	20	93	111
2	[illegible]	20	8		22		144	139	169	163	12	8	116	112.5
3	[illegible]	28	8		[illegible]		141	151	151	158	51	20	129	118
4	[illegible]	28	8		18		136	164	170	168	51	19	129	140
5	[illegible]	29	[illegible]	25		23	130	136	165	165	20	16	105	113
6	[illegible]	29	[illegible]	24		22	136	161	163	164	27	14	86	98
7	[illegible]	31	5	21		21	139	[illegible]	173	156	23	19	95	112
8	[illegible]	35	6	143	—	143	134	140	161	155	14	11	133	142
9	[illegible]	[illegible]	8	[illegible]		[illegible]	135	139	153	126	65.5	[illegible]	85	114
10	[illegible]	[illegible]	7	18		26	135	[illegible]	119	135	21	19	128	117
11	[illegible]	10	5.5	26		17	127	[illegible]	123	135	21	23	30	129
12	55	28	5	31		32	[illegible]	129	124	172	12.5	13	31	91

Line	Sea water pressure – psi (to heat exchanger)	Sea water pressure – psi	Sea water pressure – psi	Sea water pressure – psi (to oil cooler)	Sea water pressure – psi (from oil cooler)	Sea water temp. °F (to heat exchanger)	Sea water temp. °F (to oil cooler)	Sea water temp. °F (from oil cooler)	Lub. oil pressure – psi (at pump)	Lub. oil pressure – psi	Lub. oil pressure – psi (to engine)	Lub. oil pressure – psi	Lub. oil pressure – psi (turbocharger oil inlet)	Lub. oil temp. °F (from oil cooler)	Lub. oil temp. °F (to engine)	Lub. oil temp. °F (from engine)	Turbocharger oil outlet C.S.	Turbocharger oil outlet S.C.S.	Oil level
1	5			15	11	111			69	67	50	15	34		157	171	174	123	
2	5.5			9	8	102			70	61	45	32	35		128	172	125	125	
3				17		100			71	63	48	15	33		156	172	173	171	
4	11.5			17.5	16		125		68	60	46	10.4	35	163	163	176	178	172	
5	26		3	30	29	114			63	56	48	14	37		159	171	176	125	
6	8		3	25	12	117			56	52.5	43	14	31		164	177	171	180	
7	21		3	22	22		112		71	62	54	14	34		150	171	168	171	OK
8	35		2	26.5			94		80	69	52	13	35		138	173	177	112	[illegible]
9	13		3	20	16			128	76	65	56	12	41		161	176	177	172	[illegible]
10	29		3	20	26		125		78	65	38	40	41		137	173	177	175	OK
11	26		3	25	32			88	72	64	55	163	38		155	170	172	135	[illegible]
12	36	36	3	25			92		75	65	53	59	38		[illegible]	180	[illegible]	180	[illegible]

REMARKS

FOLD ALONG THIS LINE

FOLD ALONG THIS LINE

NATURAL GAS FUEL

EXHAUST

Line	Cyl. No. 1	Cyl. No. 2	Cyl. No. 3	Cyl. No. 4	Cyl. No. 5	Cyl. No. 6	Cyl. No. 7	Cyl. No. 8	Cyl. No. 9	Cyl. No. 10	Cyl. No. 11	Cyl. No. 12	Combined exhaust to turbo CS °F	Combined exhaust to turbo OCS °F	°F Temp from turbocharger	Pressure to turbocharger C.S.	Pressure to turbocharger	Pressure to turbocharger S.C.S.	Pressure	
1	695	610	610	715	720	625	610	685	610	675	675	510	715	710	615	12.8		12.6	2.2	
2	640	600	610	716	722	590	771	720	560	700	615	605	710	716	600	12.4		11.4	[illegible]	Chec
3	690	[illegible]	[illegible]	650	615	610	[illegible]	640	520	610	610	570	705	640	560	12.3		12.0	1.5	[illegible]
4	651	650	[illegible]	553	660	550	702	57	528	615	[illegible]	572	704	690	570	13.2		13.1	1.6	Nite
5	621	610	550	685	[illegible]	[illegible]	610	[illegible]	[illegible]	[illegible]	651	540	705	700	580	13.2		12.2	1.7	Nite
6	680	645	615	645	620	630	645	[illegible]	530	705	675	535	640	645	530	—		11.4	1.6	[illegible]
7	690	675	[illegible]	650	675	640	675	645	600	705	600	610	715	710	610	14.3		11.1	2.6	[illegible]
8	[illegible]	[illegible]	[illegible]	[illegible]	600	615	710	710	605	715	715	670	715	710	605	13.2		11.6	2.6	Nite
9	600	675	[illegible]	610	715	[illegible]	700	710	640	770	745	650	715	710	610	12.3		12.9	2.4	Chec
10	700	690	690	715	740	650	750	645	650	745	640	600	720	710	645	12.5		12.6	2.2	Nite
11	690	580	520	705	695	620	735	700	600	725	625	600	725	720	610	12.1		12.6	2.5	[illegible]
12	650	650	605	715	650	580	690	715	560	725	745	625	620	720	535	12.6		12.2	2.3	Nite
13													723		607		12.6			

FIGURE B-4.

INSPECTION
FIRST SHIFT ____________
SECOND SHIFT ____________
THIRD SHIFT ____________

TESTER
FIRST SHIFT ____________
SECOND SHIFT ____________
THIRD SHIFT ____________

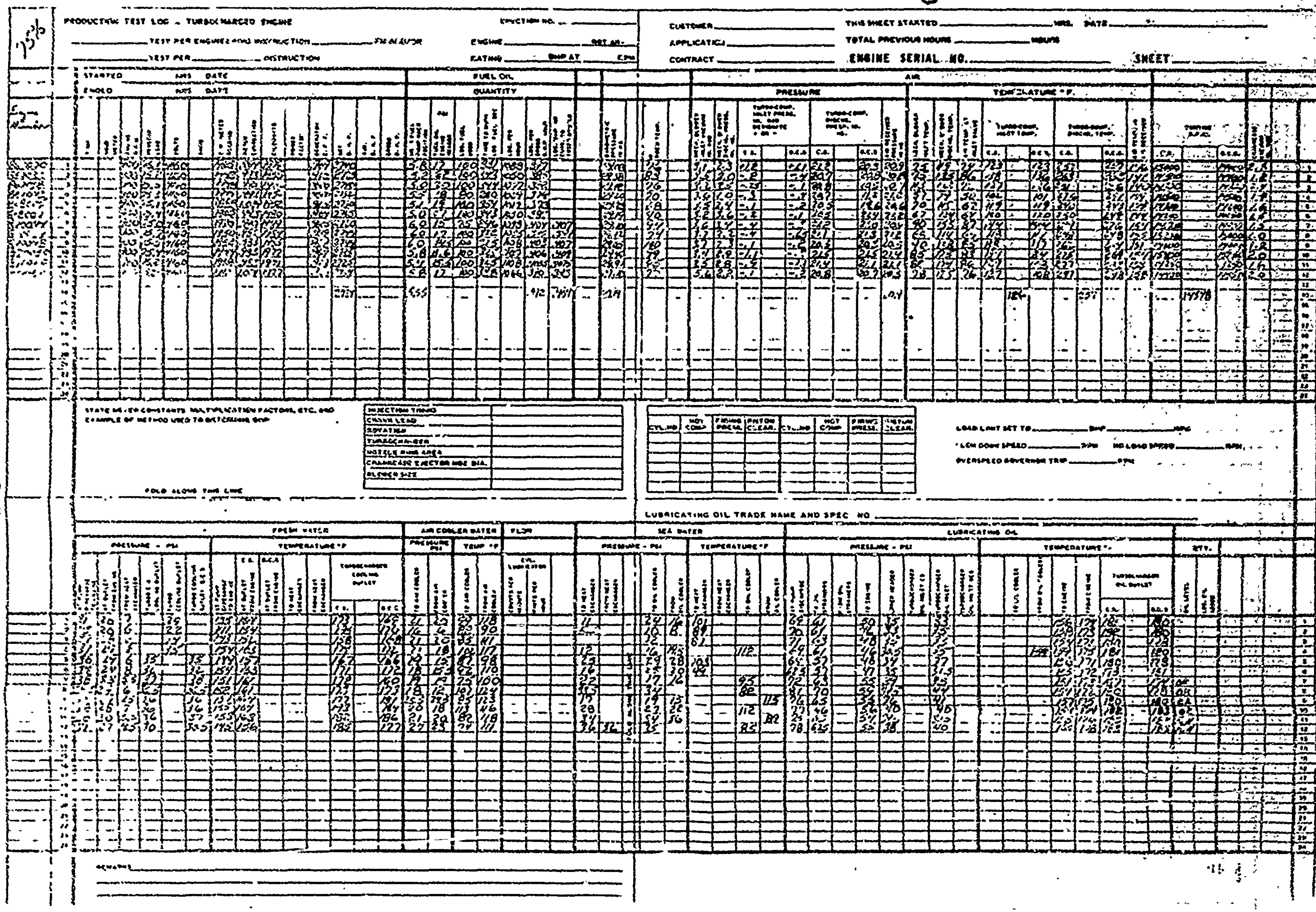

FIGURE B-5. THIS PAGE IS BEST QUALITY PRACTICABLE FROM COPY FURNISHED TO DDQ

FRESH WATER	AIR COOLER WATER	FLOW	SEA WATER	LUBRICATING OIL	MTT.
PRESSURE - PSI / TEMPERATURE °F	PRESSURE PSI / TEMP. °F	CYL. LUBRICATION	PRESSURE - PSI / TEMPERATURE °F	PRESSURE - PSI / TEMPERATURE °F	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

REMARKS

FOLD ALONG THIS LINE

NATURAL GAS FUEL	EXHAUST			
PRESSURE / TEMPERATURE / PRESSURE	EXHAUST TEMP. FROM CYLINDERS °F (CYL. NO. 1–18)	COMBINED EXHAUST TO TURBO	TEMPERATURE	PRESSURE
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

INSPECTION	TESTER
FIRST SHIFT	FIRST SHIFT
SECOND SHIFT	SECOND SHIFT
THIRD SHIFT	THIRD SHIFT

FIGURE B-6.

100%

PRODUCTION TEST LOG - TURBOCHARGED ENGINE

ERECTION NO. ______

______ TEST PER ENGINEERING INSTRUCTION ______ FM AND/OR

______ TEST PER ______ INSTRUCTION

ENGINE ______ SET NO. ______

RATING ______ BHP AT ______ RPM

CUSTOMER ______

APPLICATION ______

CONTRACT ______

THIS SHEET STARTED ______ HRS. DATE ______

TOTAL PREVIOUS HOURS ______ HOURS

ENGINE SERIAL NO. ______ SHEET ______

STARTED ______ HRS. DATE ______

ENDED ______ HRS. DATE ______

Engine Number

FUEL OIL QUANTITY

AIR PRESSURE

AIR TEMPERATURE °F.

[illegible]

STATE METER CONSTANTS, MULTIPLICATION FACTORS, ETC. AND EXAMPLE OF METHOD USED TO DETERMINE BHP

INJECTION TIMING	
CRANK LEAD	
ROTATION	
TURBOCHARGER	
NOZZLE RING AREA	
CRANKCASE EJECTOR NOZ. DIA.	
BLOWER SIZE	

FOLD ALONG THIS LINE

CYL. NO.	HOT COMP.	FIRING PRESS.	PISTON CLEAR.	CYL. NO.	HOT COMP.	FIRING PRESS.	PISTON CLEAR.

LOAD LIMIT SET TO ______ BHP ______ RPM

SLOW DOWN SPEED ______ RPM NO LOAD SPEED ______ RPM

OVERSPEED GOVERNOR TRIP ______ RPM

LUBRICATING OIL TRADE NAME AND SPEC. NO. ______

FRESH WATER: PRESSURE - PSI; TEMPERATURE °F

AIR COOLER WATER: PRESSURE PSI; TEMP °F

FLOW

SEA WATER: PRESSURE - PSI; TEMPERATURE °F

LUBRICATING OIL: PRESSURE - PSI; TEMPERATURE °F

QTY.

[illegible]

REMARKS ______

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FIGURE B-7.

PRESSURE – PSI | TEMPERATURE °F | PRESSURE – PSI | TEMPERATURE °F | PRESSURE – PSI | TEMPERATURE °F

REMARKS

FOLD ALONG THIS LINE

FOLD ALONG THIS LINE

NATURAL GAS FUEL | EXHAUST

EXHAUST TEMP. FROM CYLINDERS °F | COMBINED EXHAUST TO TURBO | TEMPERATURE | PRESSURE

INSPECTION
FIRST SHIFT
SECOND SHIFT
THIRD SHIFT

TESTER
FIRST SHIFT
SECOND SHIFT
THIRD SHIFT

FIGURE B-8.

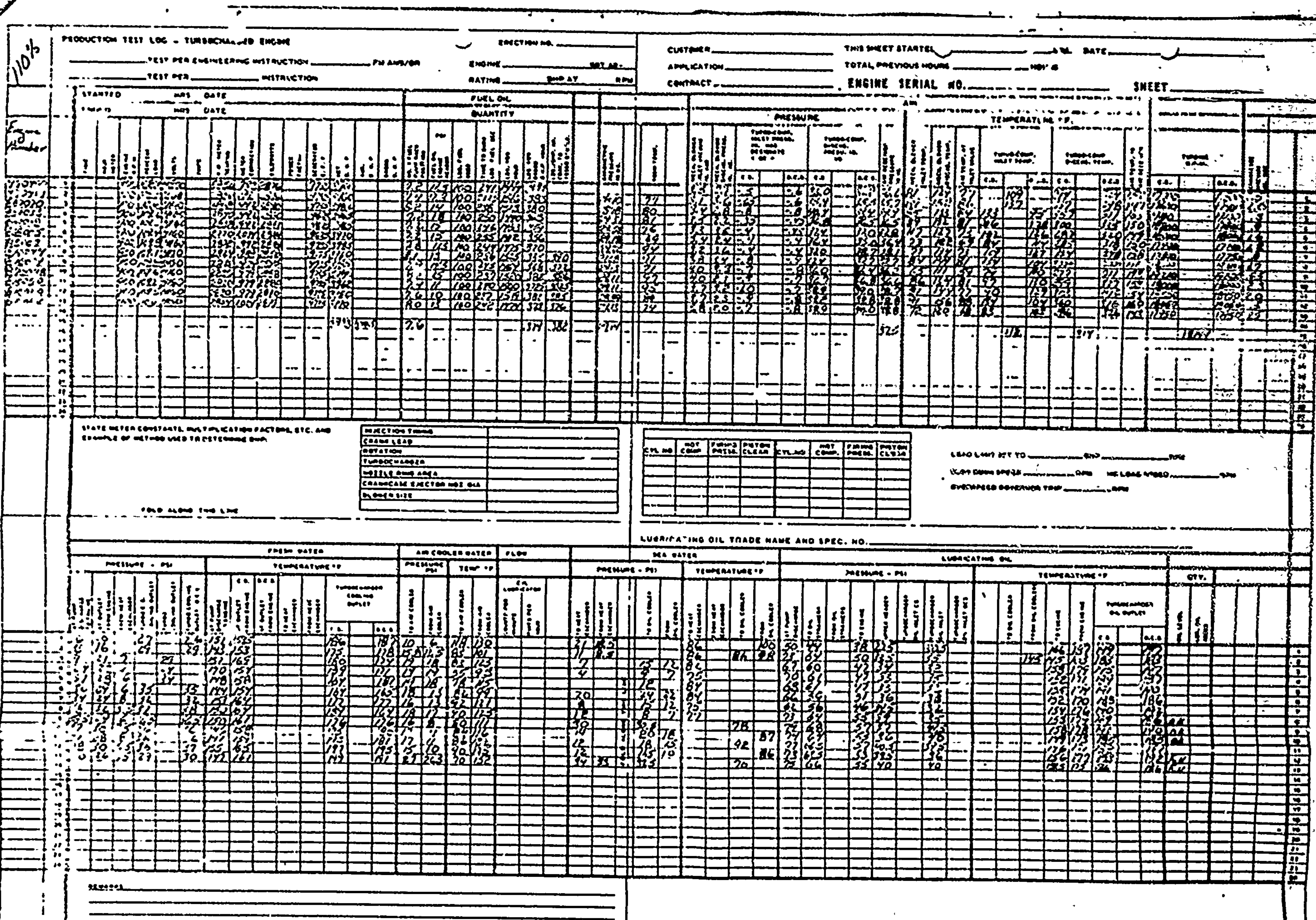

FIGURE B-9.

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FRESH WATER		AIR COOLER WATER		FLOW	SEA WATER		LUBRICATING OIL		OTY.
PRESSURE - PSI	TEMPERATURE °F	PRESSURE PSI	TEMP. °F	CYL. LUBRICATOR	PRESSURE - PSI	TEMPERATURE °F	PRESSURE - PSI	TEMPERATURE °F	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

REMARKS

FOLD ALONG THIS LINE

FOLD ALONG THIS LINE

NATURAL GAS FUEL		EXHAUST				
PRESSURE	TEMPERATURE	PRESSURE	EXHAUST TEMP. FROM CYLINDERS °F	COMBINED EXHAUST TO TURBO	TEMPERATURE	PRESSURE
			CYL. 1–18	CS / OCS	°F TEMP. FROM TURBOCHARGER	PRESSURE TO TURBOCHARGER IN. HG.
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

INSPECTOR	TESTER
FIRST SHIFT	FIRST SHIFT
SECOND SHIFT	SECOND SHIFT
THIRD SHIFT	THIRD SHIFT

FIGURE B-10.

T d'C. 10-10-67

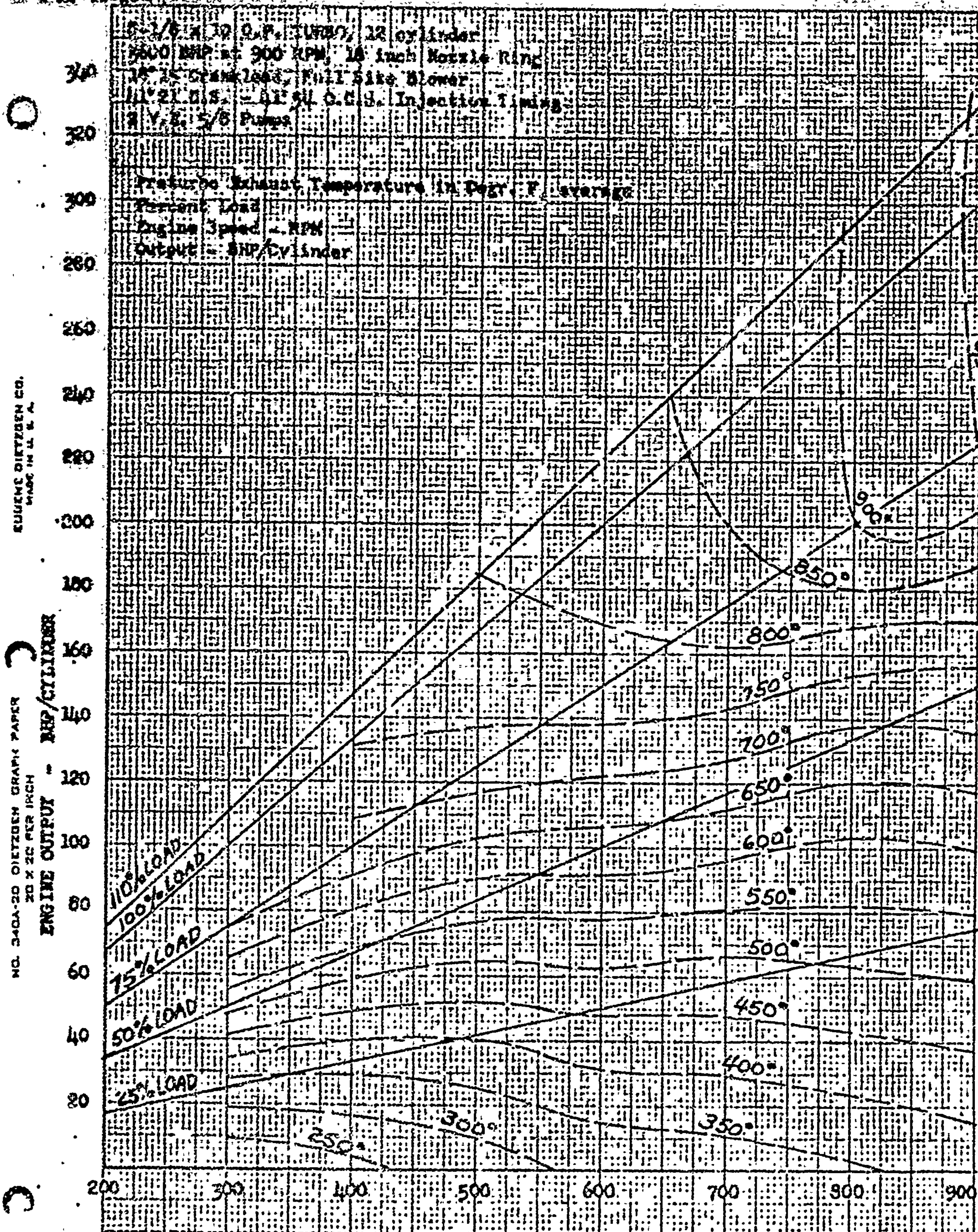


FIGURE B-11.

11207 CH

ORIGINALS AT DIESEL ENGINEERING

11207CH

Ref.: 38D867070TDG12
T.d'A. 10-9-67

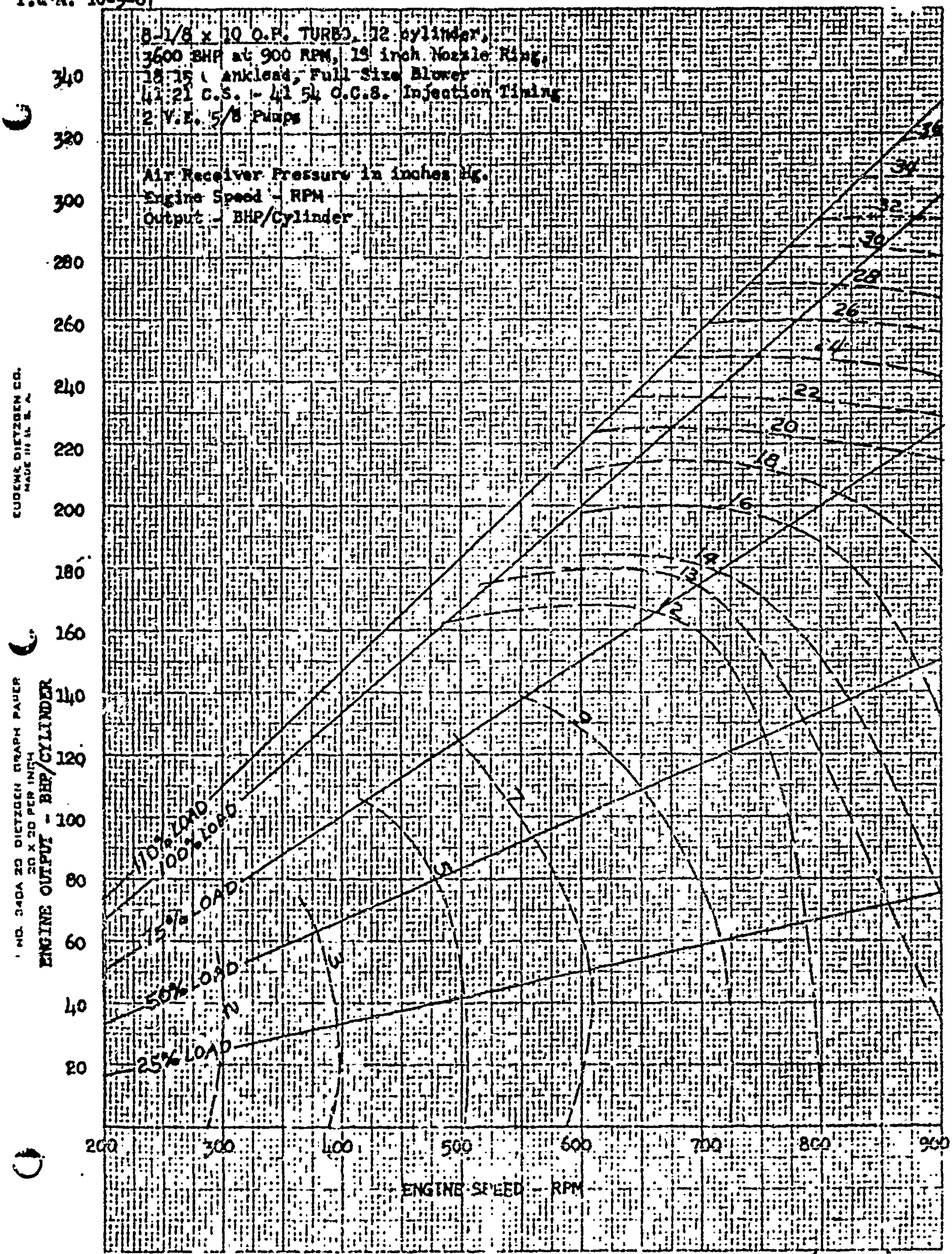


FIGURE B-12.

11205CH

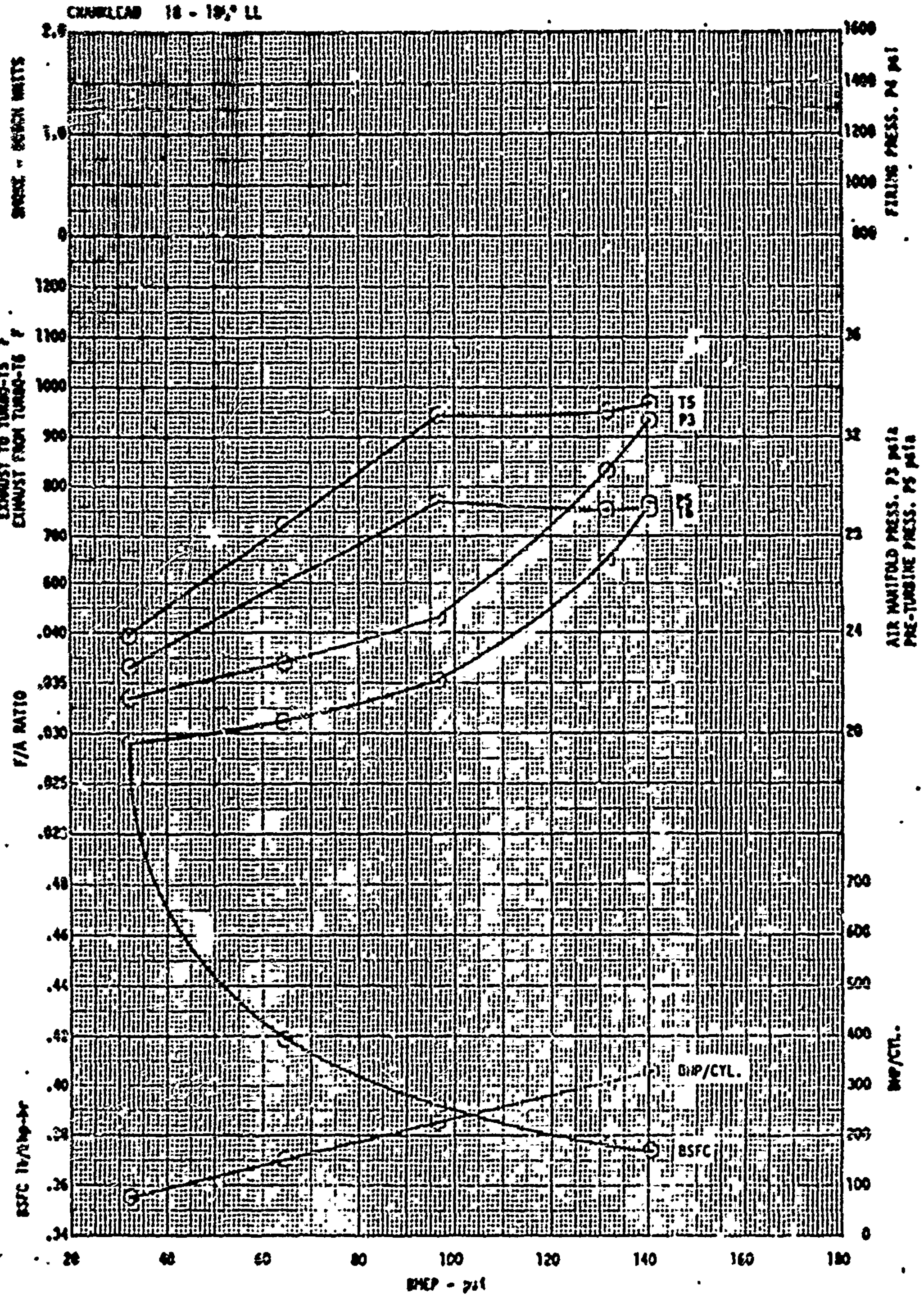
(8978 CH)
11205CH

PERFORMANCE CURVE - VARIABLE LOAD

ENGINE 16 Cyl. Turbo
SERIAL NO.
BORE/STROKE 8-1/6 x 10
COMPRESSION RATIO 11.1:1
TURBOCHARGER Elliot
NOZZLE RING AREA 18 sq.in.
DIFFUSER HEIGHT .740 in.
FUEL NOZZLE 16 704 3/8
CRANKLEAD 18 - 19½° LL

LINER Turbo-Diesel
PISTON 16 40C 332 & 334 (Rot.)
AO 141 AC 255
EO 117 EC 243
FUEL CAM Production
INJECTION P.C. 1.5 - 4.7° BTDCLC
GAS VALVE CAM N/A
GAS VALVE OPEN N/A

FUEL #2 Shop
AMBIENT TEMP.
BAROMETER 29.19 (avg.)
AIR MANIFOLD TEMP.
ENGINE SPEED 900 rpm
LOG NO.
DATE RUN
DIESEL __X__ DUAL FUEL ______

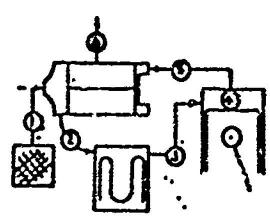


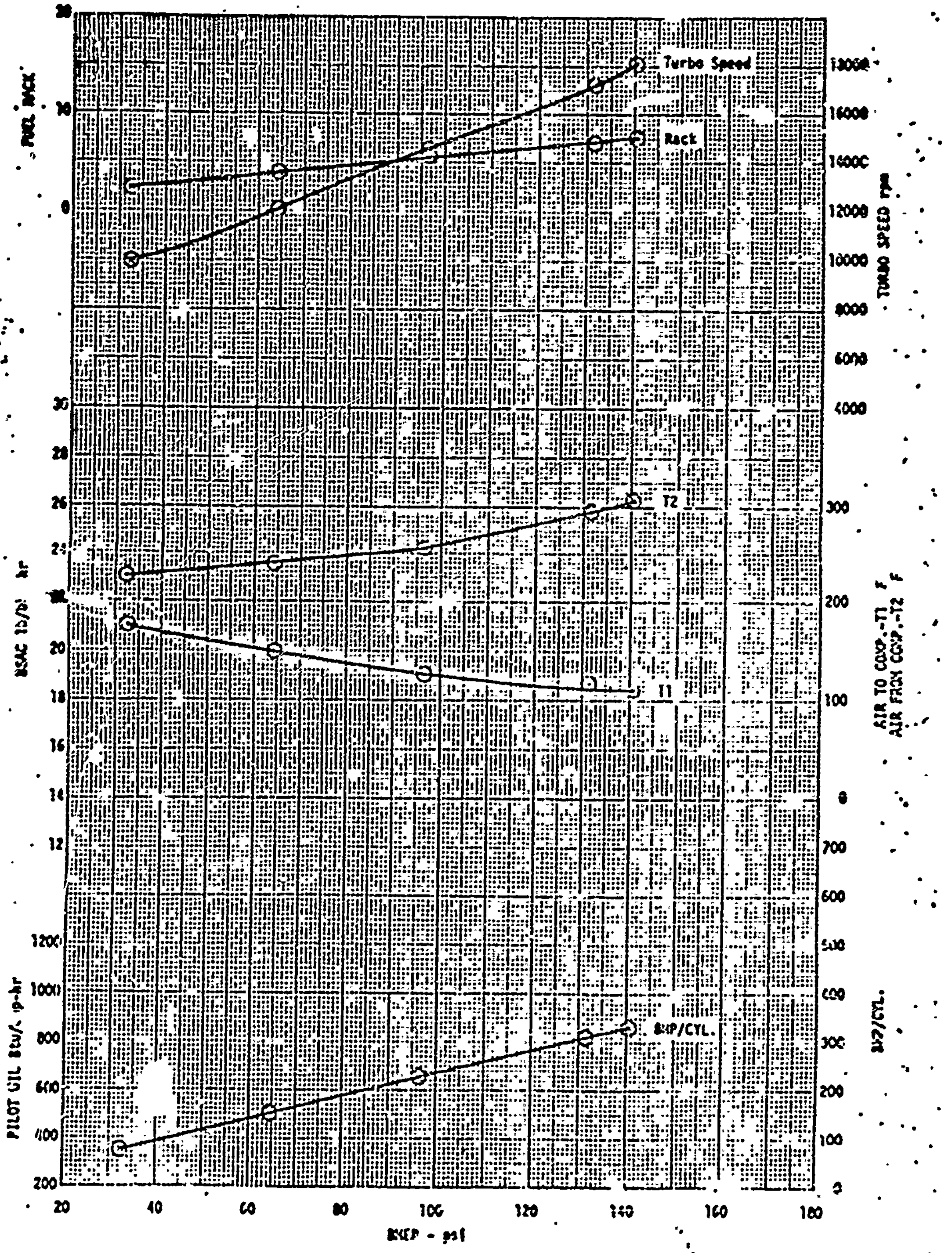
DRAWN BY: R. Pickering
DATE: 12/7/76

Form D

CURVE NO. 56
SHEET 1 OF 2

FIGURE B-13.





DRAWN BY: B. Pickering
DATE: 12/7/76

Form F

CURVE NO. 56
SHEET 2 OF 2

FIGURE B-14.

PERFORMANCE CURVE - VARIABLE LOAD

⬡ = 110%
○ = 100%
△ = 75%
□ = 50%
◇ = 25%

ENGINE 12-Cyl. Turbo
SERIAL NO. 380867070-TDG12
BORE/STROKE 8-1/8 x 10
COMPRESSION RATIO 11.1:1
TURBOCHARGER Elliot
NOZZLE RING AREA 18 sq. inch
DIFFUSER HEIGHT .740 inch
FUEL NOZZLE 16 704 903
CRANKLEAD 18¼° LL

LINER Turbo-Diesel
PISTON 16 400 333 & 334 (Rot.)
AO 141° AC 255°
EO 117° EC 243°
FUEL CAM Production
INJECTION P.C. 1.6° BTDCLC
GAS VALVE CAM N/A
GAS VALVE OPEN N/A

FUEL #2 Shop
AMBIENT TEMP. 75°
BAROMETER 29.00 (avg.)
AIR MANIFOLD TEMP.
ENGINE SPEED Var.rpm
LOG NO. 6-12 (Prod.)
DATE RUN 9-21-67
DIESEL X DUAL FUEL ____

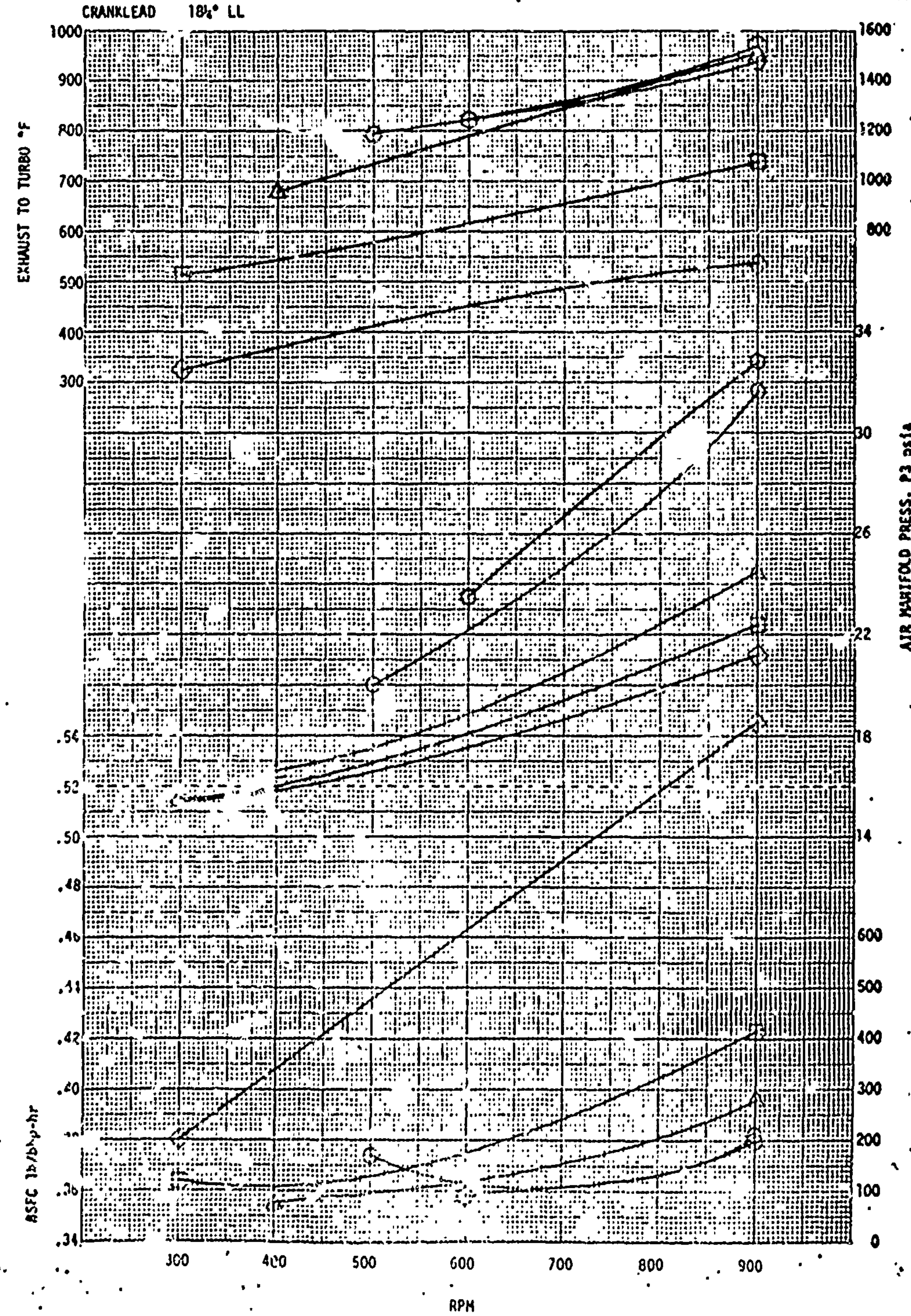


FIGURE B-16.

DRAWN BY: Pickering
DATE: 12/1/76

CURVE NO. 57
SHEET 1 OF 3

PERFORMANCE CURVE - VARIABLE LOAD

ENGINE 12 Cyl. Turbo	LINER Turbo-Diesel	FUEL #2 Shop
SERIAL NO. 380867070-TDG12	PISTON 16 400 333 & 334 (Rot.)	AMBIENT TEMP. 75°
BORE/STROKE 8-1/8 x 10	AO 141° AC 255°	BAROMETER 29.00 (avg.)
COMPRESSION RATIO 11.1:1	EO 117° EC 243°	AIR MANIFOLD TEMP.
TURBOCHARGER Elliot	FUEL CAM Production	ENGINE SPEED Var rpm
NOZZLE RING AREA 18 sq. inch	INJECTION P.C. 1.6° BIDCLC	LOG NO. 6-12 (Prod.)
DIFFUSER HEIGHT .740 inch	GAS VALVE CAM N/A	DATE RUN 9-21-67
FUEL NOZZLE 16 704 908	GAS VALVE OPEN N/A	DIESEL X DUAL FUEL
CRANKLEAD 18¼° LL		

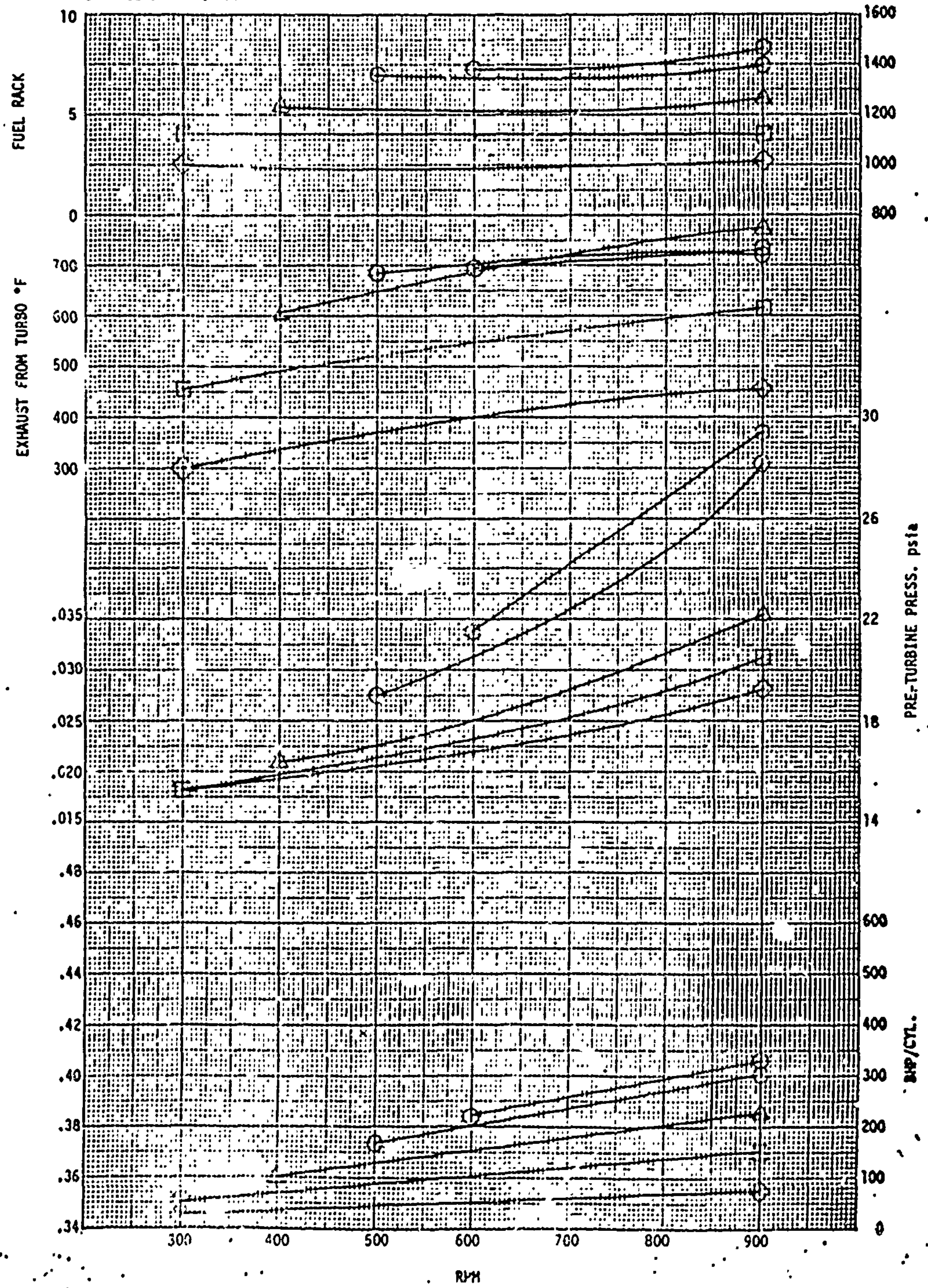


FIGURE B-15.

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DATE: 12/15/76

CURVE NO. 57
SHEET 2 OF 3

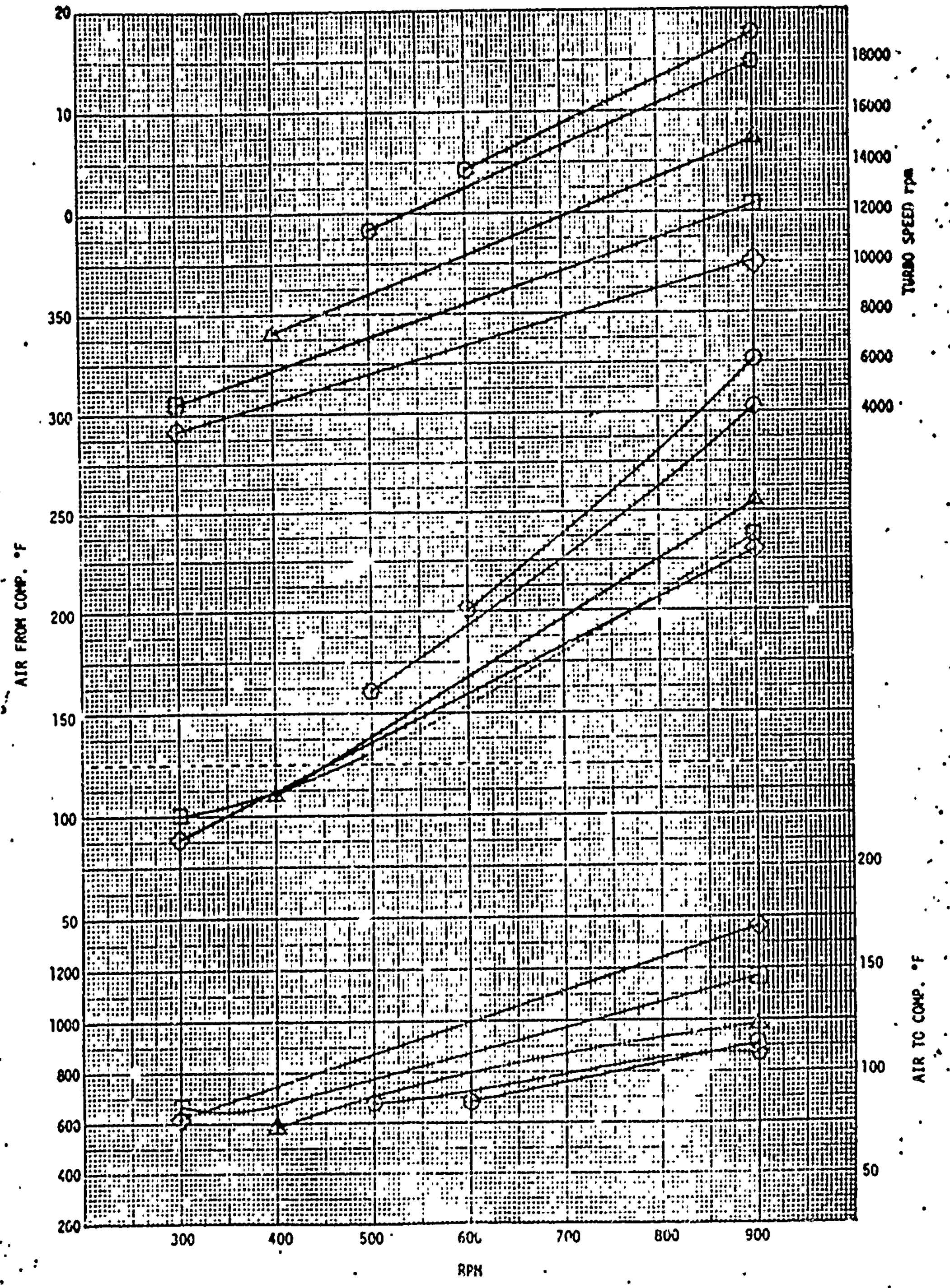

FIGURE B-17.

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DATE: 12/15/76

CURVE NO. 57
SHEET 3 OF 3

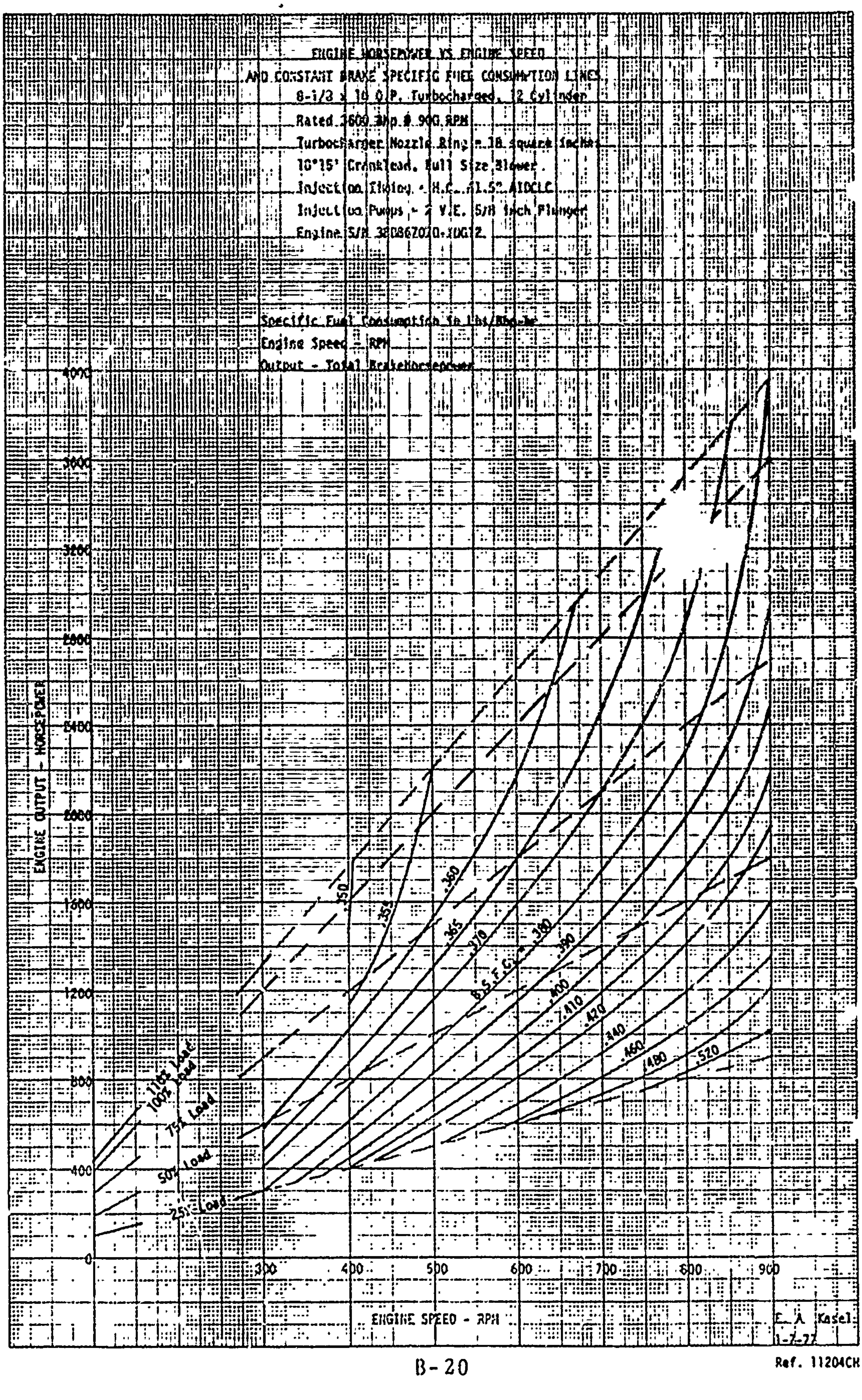


FIGURE B-18.

Ref. 11204CH

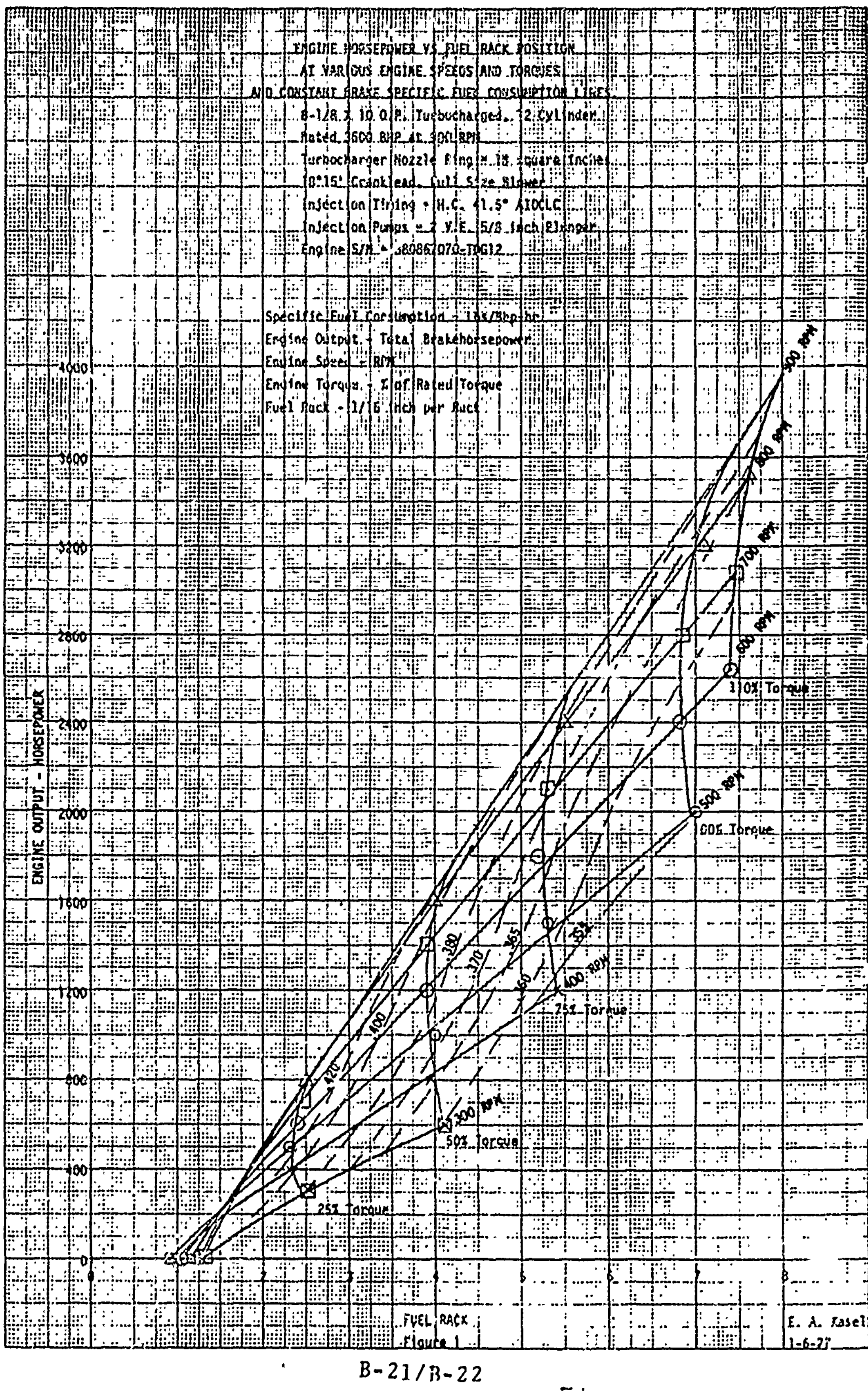

ENGINE HORSEPOWER VS FUEL RACK POSITION
AT VARIOUS ENGINE SPEEDS AND TORQUES
AND CONSTANT BRAKE SPECIFIC FUEL CONSUMPTION LINES
8-1/8 X 10 O.P. Turbocharged, 12 Cylinder
Rated 3600 BHP at 900 RPM
Turbocharger Nozzle Ring = 18 Square Inches
10°15' Cranklead, Full Size Blower
Injection Timing - H.C. 41.5° AIOCLC
Injection Pumps = 2 V.E. 5/8 inch Plunger
Engine S/N = 380867070-TDG12
Specific Fuel Consumption - lbs/Bhp-hr
Engine Output - Total Brakehorsepower
Engine Speed - RPM
Engine Torque - % of Rated Torque
Fuel Rack - 1/16 inch per Rack
ENGINE OUTPUT - HORSEPOWER
4000
3600
3200
2800
2400
2000
1600
1200
800
400
0
0
1
2
3
4
5
6
7
8
900 RPM
800 RPM
700 RPM
600 RPM
110% Torque
500 RPM
100% Torque
400 RPM
75% Torque
300 RPM
50% Torque
25% Torque
380
370
365
360
355
400
420
FUEL RACK
Figure 1
E. A. Kasel
1-6-77

FIGURE B-19.

APPENDIX C

LABORATORY TEST DATA - 6 CYLINDER ENGINE WITH TURBO-BLOWER SERIES SCAVENGING AIR SYSTEM

CONTRACT NO. DOT-TSC-905

Colt Industries Customer Order No. 11-938642

FM 38TD8-1/8 Performance and Smoke Improvement

378' High Endurance Cutters (WHEC's)

MONTHLY PROGRESS REPORT - APRIL 1977

Prepared by: E. A. Kasel
E. A. Kasel

Approved by: C. L. Newton
C. L. Newton

U. S. DEPARTMENT OF TRANSPORTATION CONTRACT NO. DOT-TSC-905

Modification No. 5

FM 38TD8-1/8 Performance and Smoke Improvement

378' High Endurance Cutters (WHEC's)

Monthly Progress Report - April 1977

Tests were run using a 6 cylinder laboratory engine to determine expected performance and engine operating parameters with the latest production engine scavenging air system and power components. Variable speed torque loading was per the initial contract specifications shown on FM curve 11188 CH. Power output was one-half that shown, since tests were run with a 6 cylinder engine.

These data were obtained with 70°F air to the turbocharger compressor inlet and 120 to 125°F air to the engine driven scavenging air blower. The turbocharger was a unit built by Alco instead of Elliott Co., except with the 19 square inch nozzle ring. The data on curve no. 62 with an Alco and Elliott turbocharger shows the difference between the turbochargers is insignificant.

The engine had the latest production turbocharged pistons. The scavenging air blower capacity was 1.23 times the engine displacement volume. The 12 cylinder engine with a 34 inch long blower and 1.51:1 gear ratio, as used for field conversion to the new scavenging air system, gives a ratio of 1.26 times the engine displacement volume. This blower capacity is about 3.6% more than used on production 12 cylinder engines with the 27 inch long blower and 1.835:1 blower drive gear ratio.

The fuel injection pumps for these tests were turbocharged dual fuel type. The pumping characteristic of this pump is the same as the diesel fueled engine injection pump, except for the rack reading. The adjusted rack reading for a diesel pump is shown as a dashed line on the attached curves.

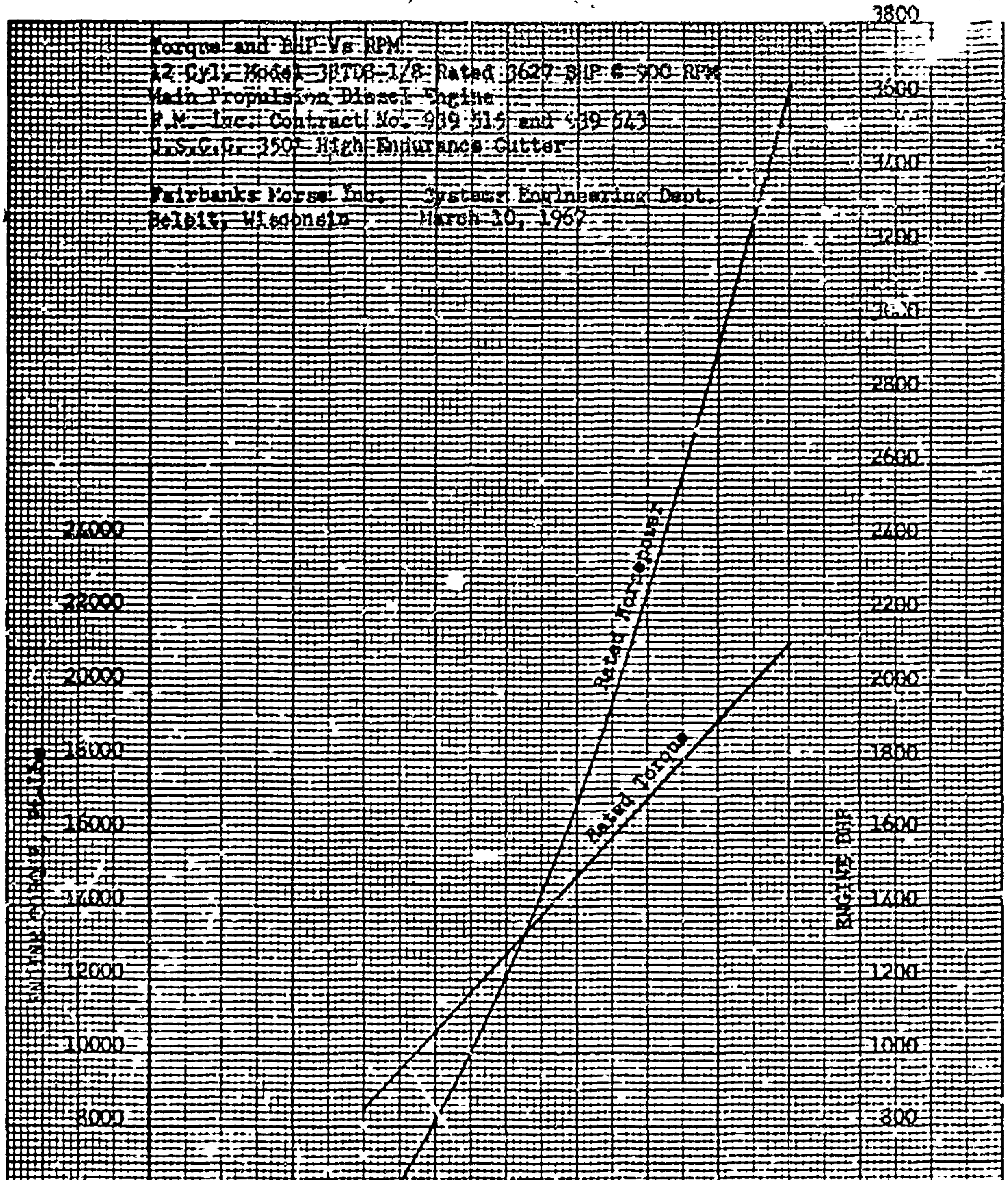
Curve no. 60 shows the above described engine performance with a 17 and 18 square inch nozzle ring with engine loading per curve 11188 CH. Engine performance is very good with the 18 square inch nozzle ring, giving a .368 to .378 brake specific fuel consumption throughout the speed range of 450 to 900 rpm with a maximum preturbine exhaust temperature of 920°F. The F/A ratio is very good and remains in the range of .024 to .026 pounds of fuel per pound of air throughout the operating range.

Curve no. 61 shows the engine performance at maximum rated speed operating between 75% and 110% rated torque. The engine performance is considered very good with the 18 square inch nozzle ring and gives firing pressures of about 1210 psig at rated speed and torque with an injection timing of high cam 36-3/4 degrees after inner dead center of the lower piston.

Curve no. 62 shows the engine performance using a 19 square inch nozzle ring in two different turbochargers. These data show little change from the 18 square inch nozzle ring and are considered accaptable. The fuel consumption is slightly better with a small increase in smoke and exhaust temperatures. The brake specific air consumption remains above 13.9 lbs/bhp-hr. and is considered good.

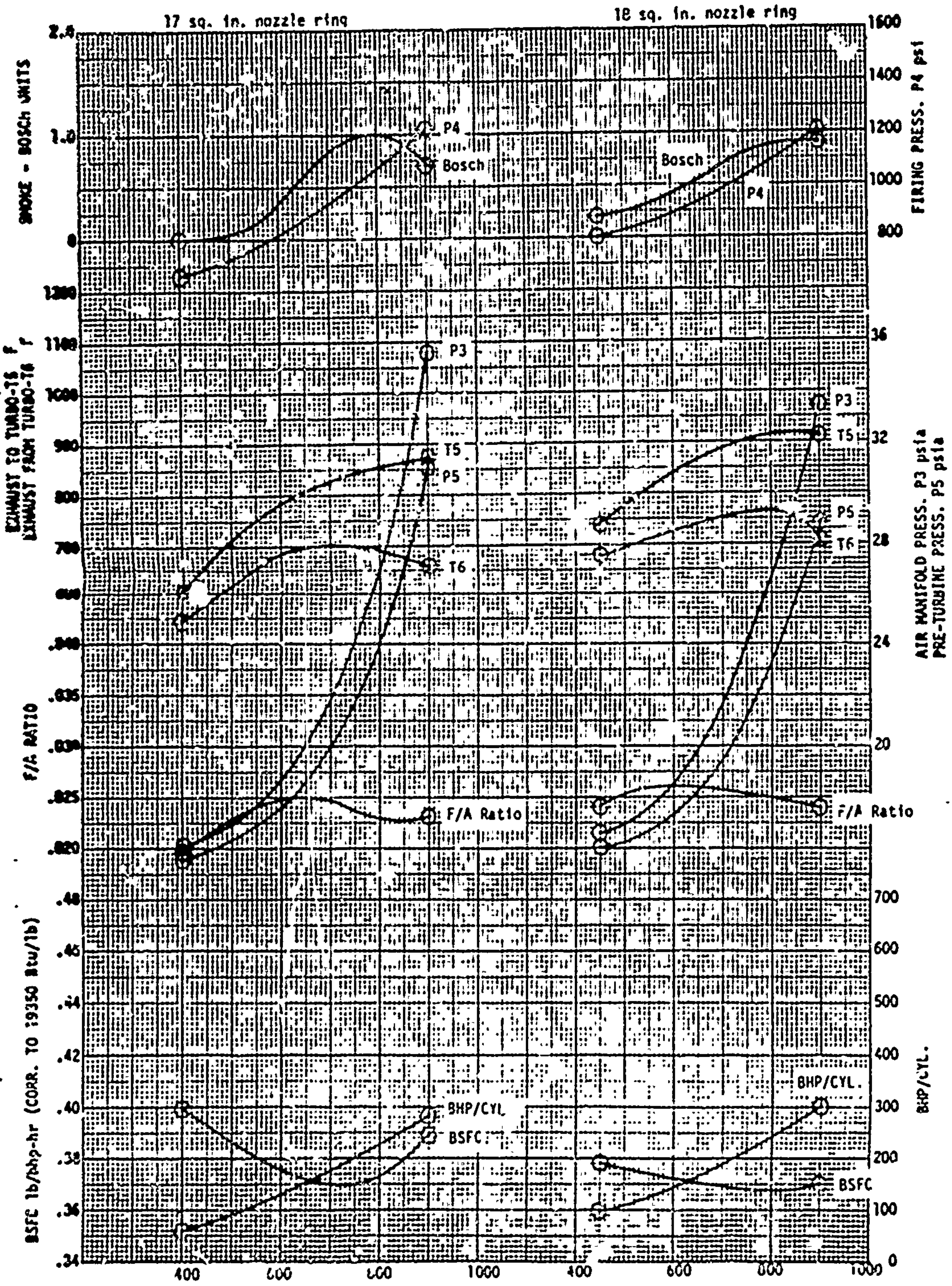
The smoke readings obtained during these laboratory tests are higher than what would be expected. Partly because a 12 cylinder engine generally has a little less smoke and secondly because the injection nozzles were in a deteriorated condition when these data were obtained. Later, engine operation with new injection nozzle tips gave about a 30% reduction in Bosch smoke units.

For 378' Coast Guard Cutter engine conversions to turbo blower se es scavenging air system, the best all around engine performance should be obta d with a 19 square inch nozzle ring. This takes into consideration that the engines are generally not run at 900 rpm and can have air inlet temperatures well below 70°F. The 19 square inch nozzle ring will give adequate air at reduced engine speeds and with turbo air inlet temperatures up to 90°F.


Torque and BHP Vs RPM
12 Cyl. Model 38TD8-1/8 Rated 3627 BHP @ 900 RPM
Main Propulsion Diesel Engine
F.M. Inc. Contract No. 939 515 and 939 643
U.S.C.G. 350' High Endurance Cutter
Fairbanks Morse Inc. Systems Engineering Dept.
Beloit, Wisconsin March 20, 1967
Rated Horsepower
Rated Torque
ENGINE RPM
ENGINE BHP
0
100
200
300
400
500
600
700
800
900
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000
22000
24000
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2600
2800
3200
3400
3600
3800

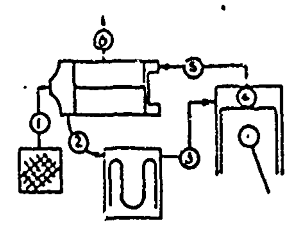
PERFORMANCE CURVE - VARIABLE LOAD
DIESEL CONFIGURATION

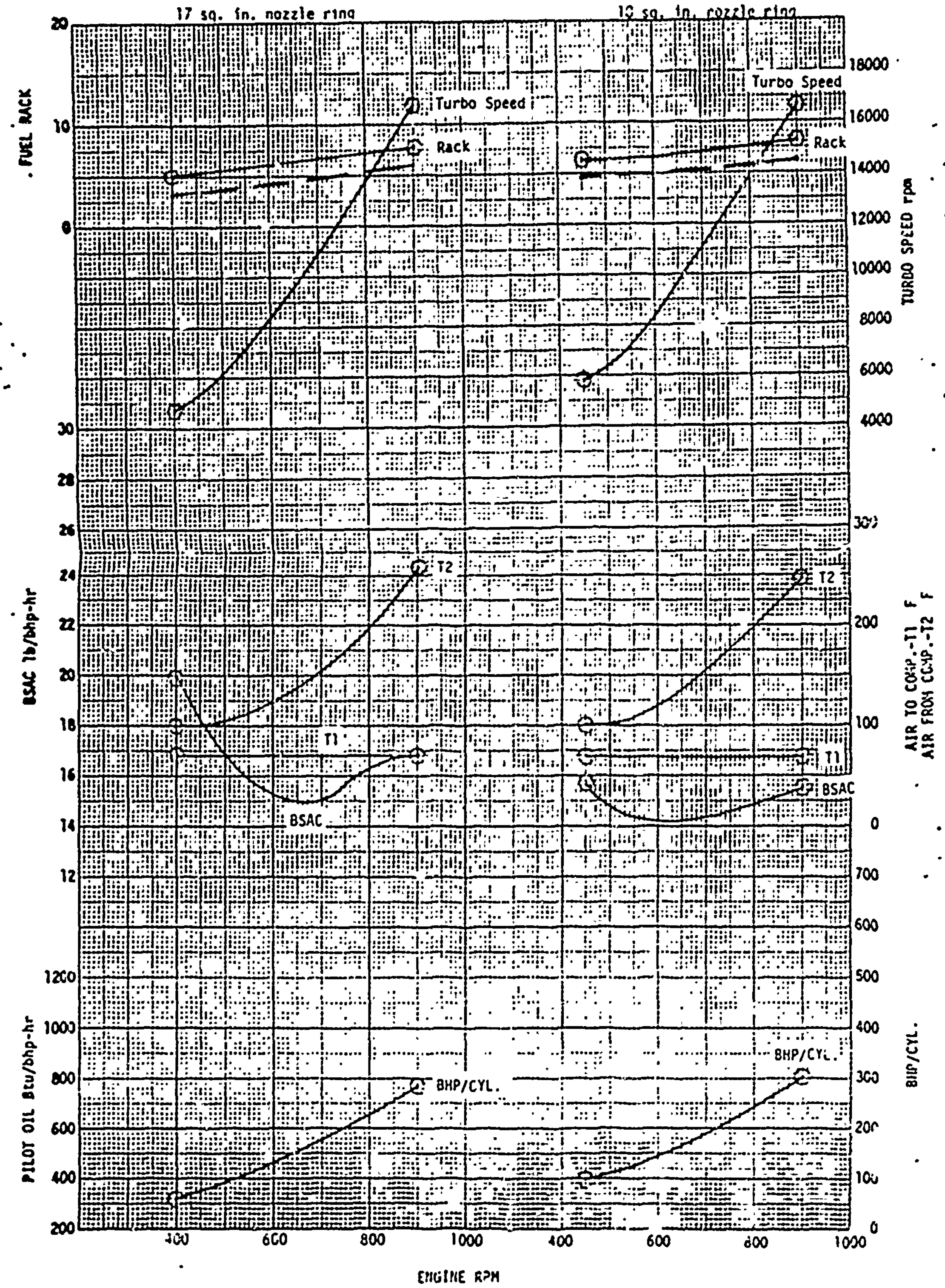
ENGINE West 6	LINER 16 302 239/16 302 233	FUEL #2 Shop
SERIAL NO. 39193	PISTON 16 401 902	AMBIENT TEMP. 63-72°F
BORE/STROKE 8-1/8 X 10	AO 141° AC 255°	BAROMETER 29.42
COMPRESSION RATIO 11:1	EO 117° EC 243°	AIR MANIFOLD TEMP. 130-139°F
TURBOCHARGER Alco 720	FUEL CAM Production	ENGINE SPEED Var. rpm
NOZZLE RING AREA 17 sq.in./18 sq.in.	INJECTION P.C. 6.25° BTDCLC	LOG NO. 2608 & 2610
DIFFUSER HEIGHT .690 inch	CRANKLEAD 18°LL	DATE RUN 2-15-77 & 2-28-77
FUEL NOZZLE Gasketless	BL. ENG. VOL. RATIO 1.23:1	DIESEL X DUAL FUEL ____



C-6

DRAWN BY: B. Pickering
DATE: 1-4-77
Form D
CURVE NO. 60
SHEET 1 OF 2





C-7

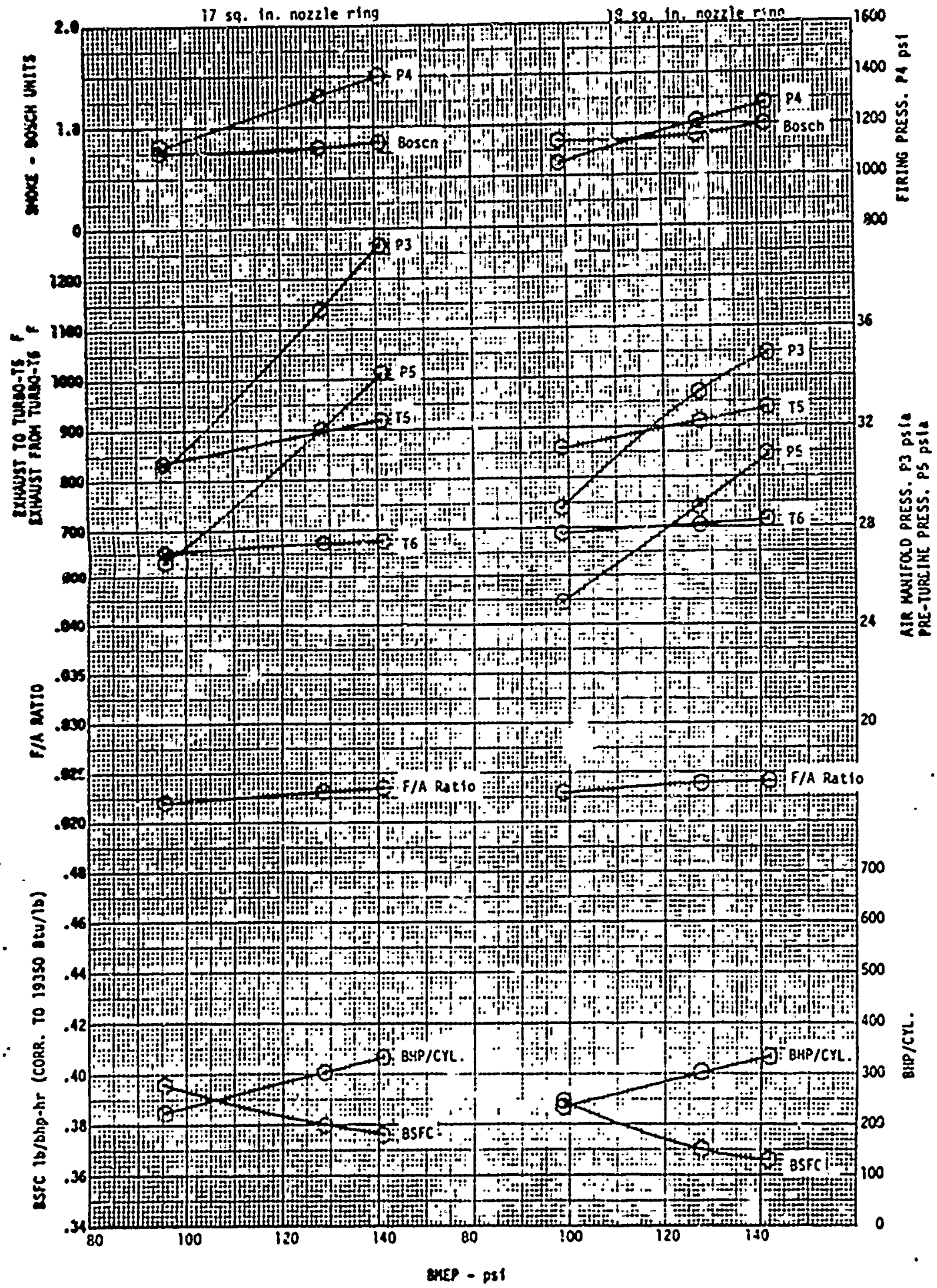
DRAWN BY: B. Pickering
DATE: 3-4-77

Form F

CURVE NO. 60
SHEET 2 OF 2

PERFORMANCE CURVE - VARIABLE LOAD
DIESEL CONFIGURATION

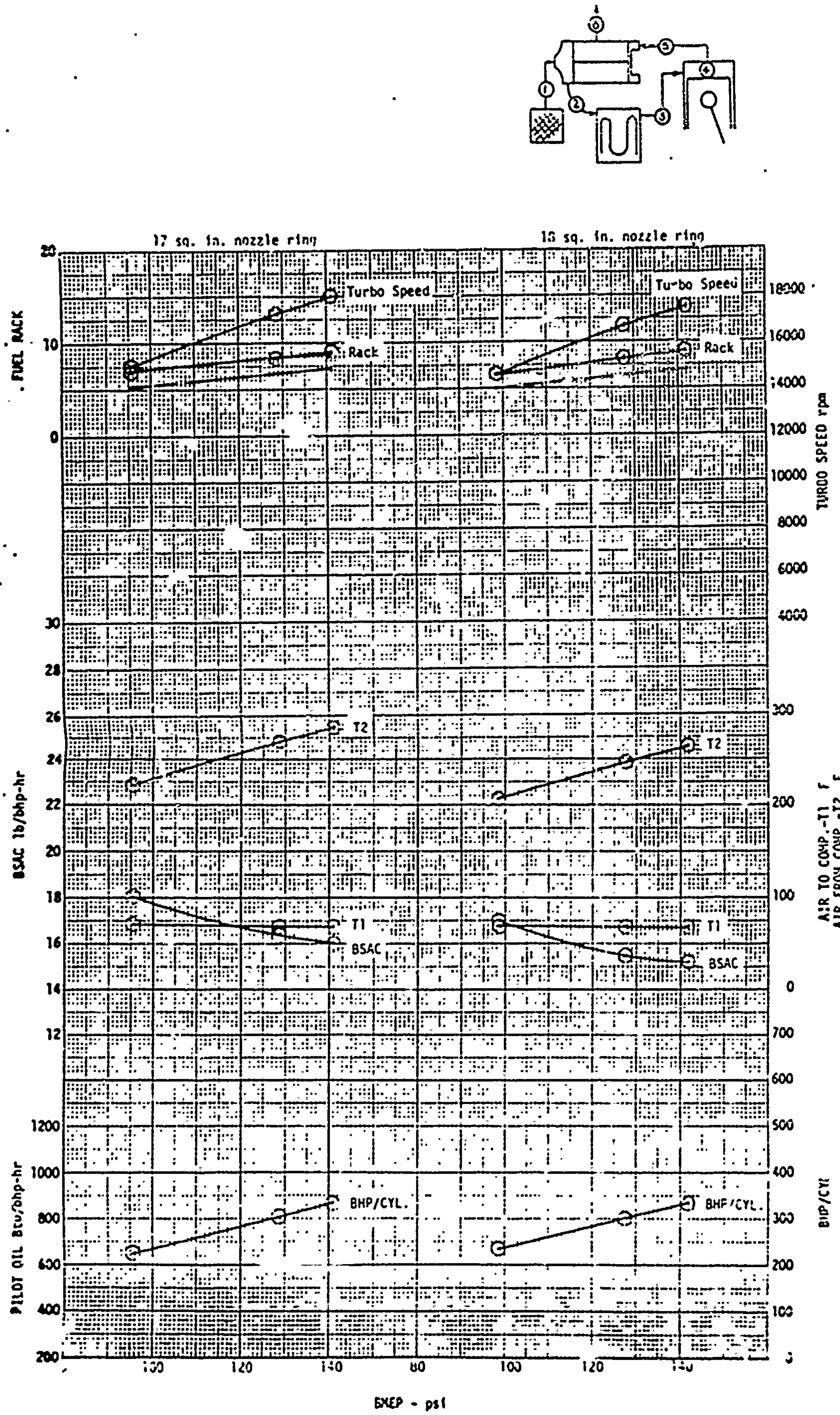
ENGINE Test 6	LINER 16 302 289/16302 283	FUEL #2 Shop
SERIAL NO. 339193	PISTON 16 401 902	AMBIENT TEMP. 68-72°F
BORE/STROKE 8-1/8 x 10	AO 141° AC 255°	BAROMETER 29.21
COMPRESSION RATIO 11:1	EO 117° EC 243°	AIR MANIFOLD TEMP. 163-165°F
TURBOCHARGER Alco 720	FUEL CAM Production	ENGINE SPEED 900 rpm
NOZZLE RING AREA 17 sq.in./18 sq.in.	INJECTION P.C. 6.25° BTDCLC	LOG NO. 2610
DIFFUSER HEIGHT .690 inch	CRANKLEAD 18°LL	DATE RUN 2/28/77
FUEL NOZZLE Gasketless	BL. ENG. VOL. RATIO 1.23:1	DIESEL x DUAL FUEL ______



DRAWN BY: R. Pickering
DATE: 3-4-77

Form D

CURVE NO. 61
SHEET 1 OF 2



C-9

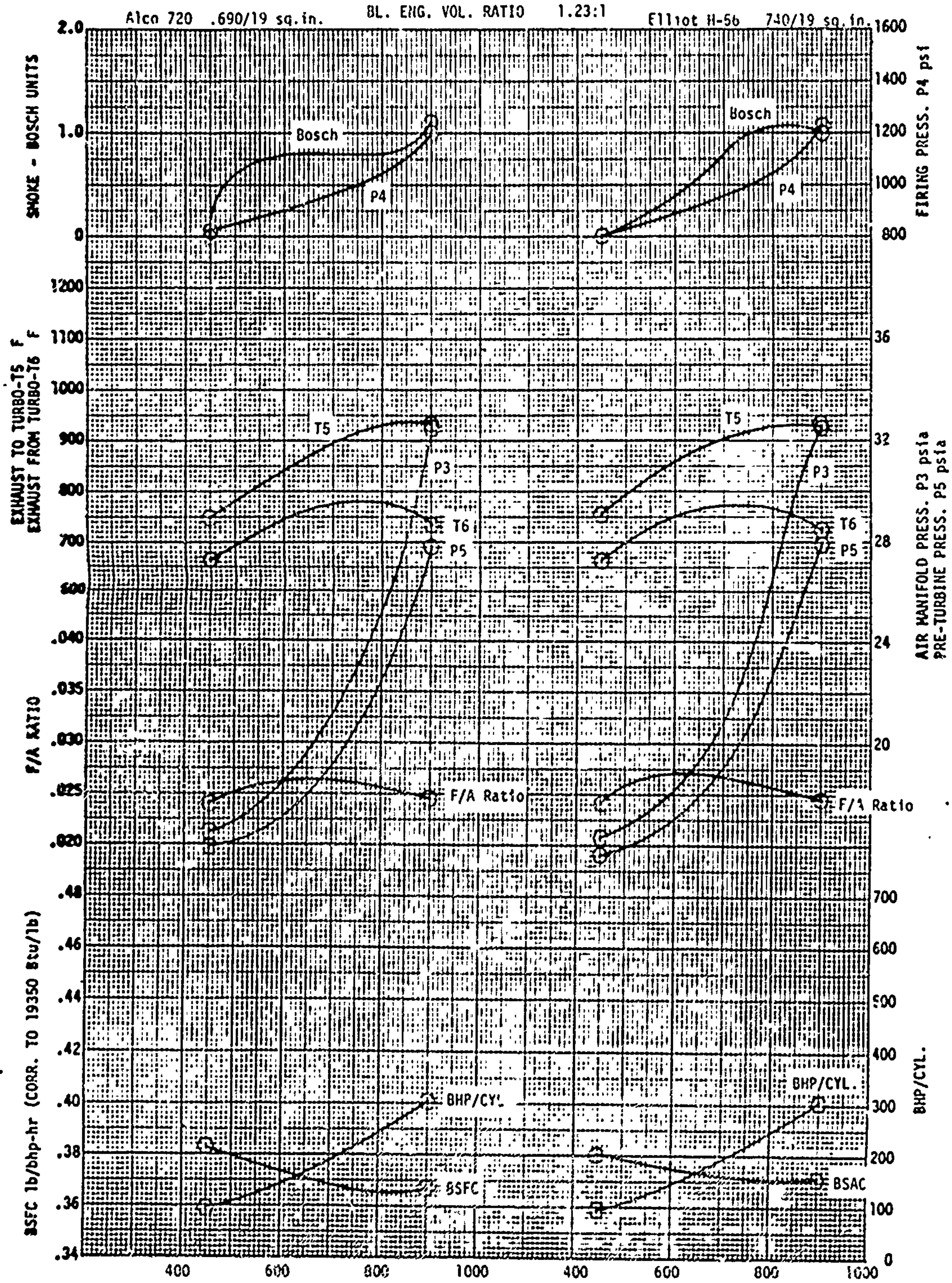
DRAWN BY: ___
DATE: ___

Form F

CURVE NO. ___
SHEET 2 of 2

PERFORMANCE CURVE - VARIABLE LOAD
DIESEL CONFIGURATION

ENGINE Test	LINER 16 392 209/16 392 233	FUEL #2 Shop
SERIAL NO. 9193	PISTON 16 401 902	AMBIENT TEMP. 65-72°F
BORE/STROKE 9-1/8 x 10	AO 141° AC 255°	BAROMETER 29.40/28.90
COMPRESSION RATIO 11:1	EO 117° EC 243°	AIR MANIFOLD TEMP. 126-158°F
TURBOCHARGER Alco 720/Elliot H-56	FUEL CAM Production	ENGINE SPEED Var. rpm
NOZZLE RING AREA 19 sq. inch	INJECTION P.C. 6.25° BTDCLC	LOG NO. 2611
DIFFUSER HEIGHT .690 inch/.740 inch		DATE RUN 3/2/77
FUEL NOZZLE Gasketless	CRANKLEAD 18°LL	DIESEL X DUAL FUEL
	BL. ENG. VOL. RATIO 1.23:1	

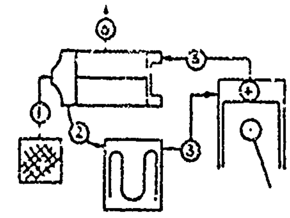


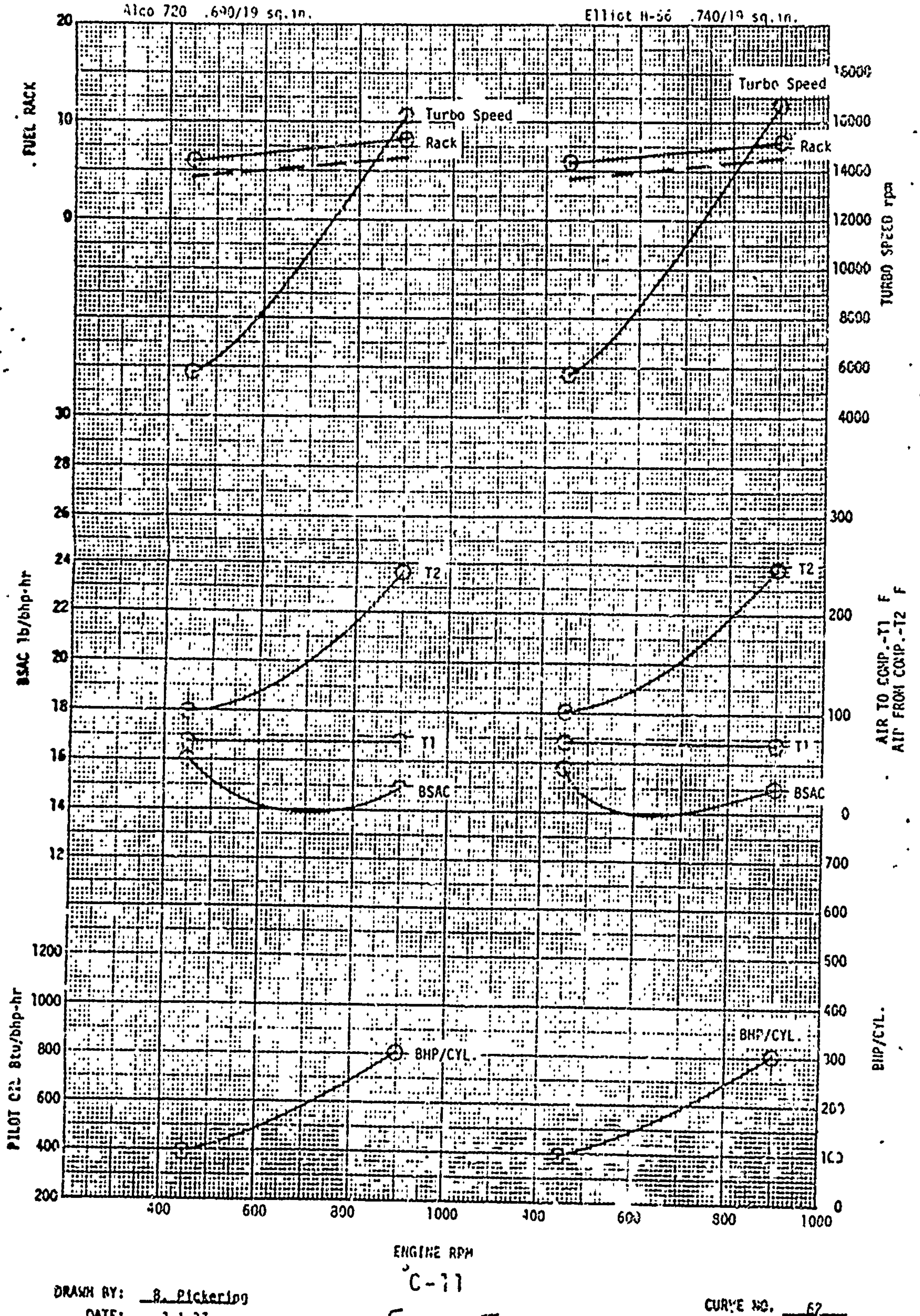
C-10

DRAWN BY: B. Pickering
DATE: 3-4-77

Form D

CURVE NO. 62
SHEET 1 OF 2





C-11

DRAWN BY: B. Pickering
DATE: 1-1-77

Form F

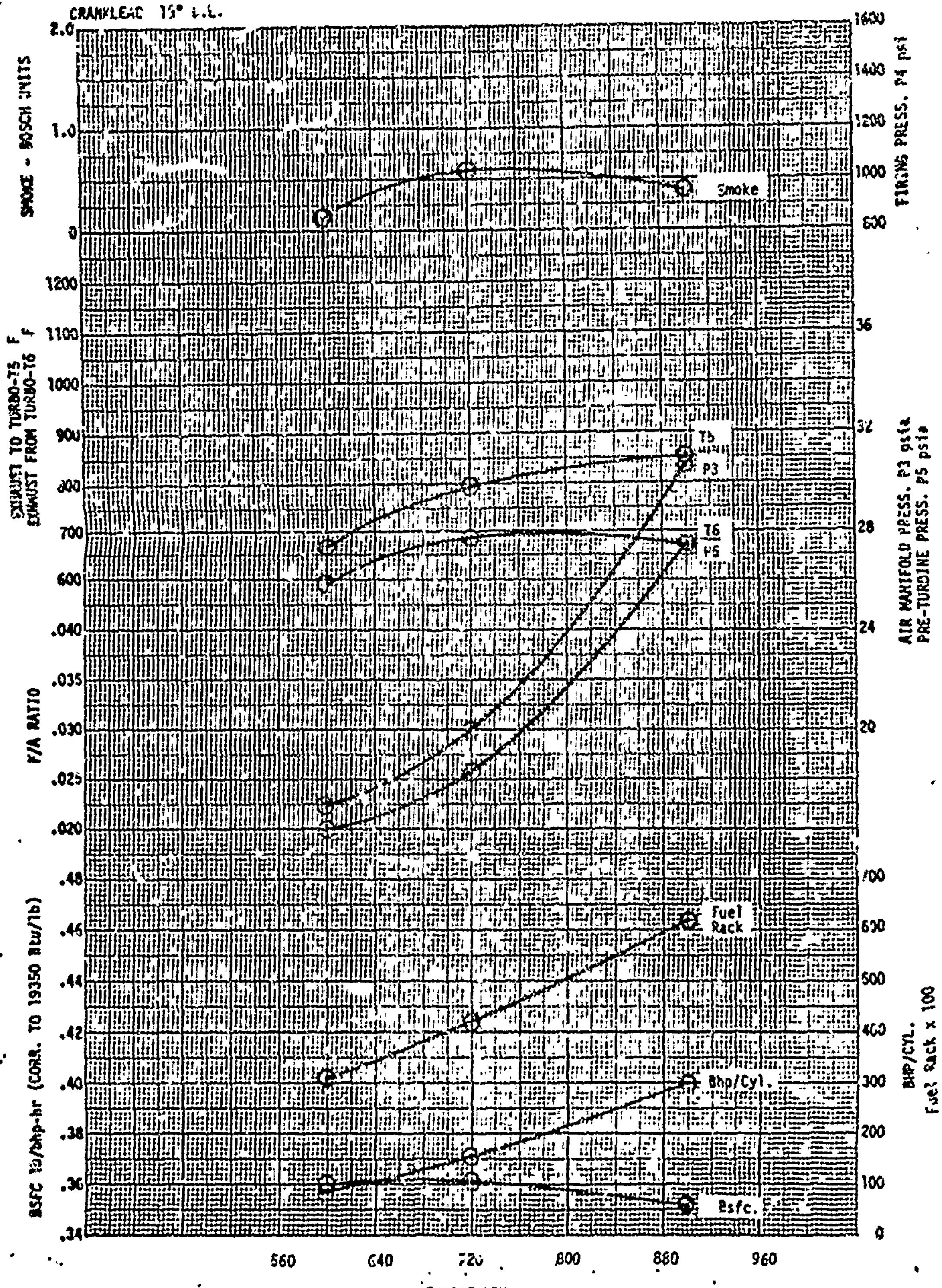
CURVE NO. 62
SHEET 2 OF 2

PERFORMANCE CURVE - VARIABLE LOAD

ENGINE V-6
SERIAL NO. 88419
BORE/STROKE 8-1/8 x 10
COMPRESSION RATIO 17.0:1
TURBOCHARGER Elliott (H-52)
NOZZLE RING AREA 19 in.^{2}
DIFFUSER HEIGHT .74 in.
FUEL NOZZLE Pintle (10° Angle)
CRANKLEAD 15° L.L.

LINER 16401912 (Turbo Diesel)
PISTON 16400233 & 334 (Rotating)
AO 141 AC 255
EO 117 EC 243
FUEL CAM Production
INJECTION P.C. 35.8° ATDCLC
GAS VALVE CVI None
GAS VALVE OPEN None

FUEL #2 Shop
AMBIENT TEMP. 56 to 70°F.
BAROMETER 29.2
AIR MANIFOLD TEMP. 133 to 145°F.
ENGINE SPEED 900 rpm
LOG NO. 1775, 1766 & 1787
DATE RUN 3/23 thru 10/12/71
DIESEL X DUAL FUEL ____



C-12

DRAWN BY: E. Kasel
DATE: 8/29/77

Form D

CURVE NO. 66
SHEET 1 OF 1

BENEFITS OF TURBO-BLOWER SERIES ARRANGEMENT

1. Increase Horsepower 10% with improved piston life and ring wear.
2. Exhaust smoke and exhaust temperature are reduced.
3. On diesel operation the Turbo-Blower Series Arrangement system provides 20% more air at all loads.
4. Cylinder Liner temperature is lowered due to excess air.
5. The air inlet check valve above turbocharger and associated piping has been removed thereby simplifying maintenance.

These benefits are available for existing engines when the conversion kits described in this brochure are applied.

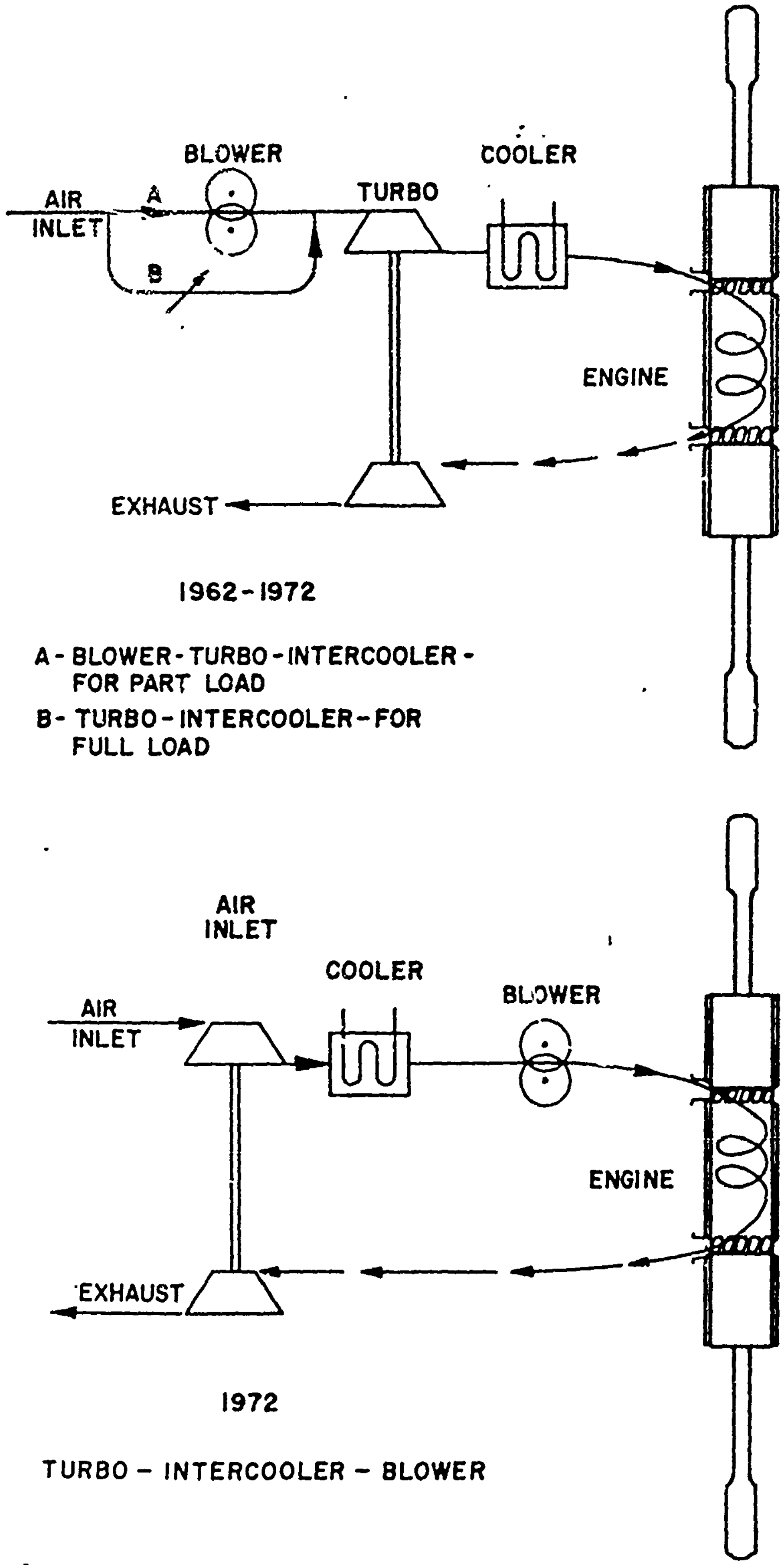

BLOWER
COOLER
AIR INLET
A
TURBO
B
ENGINE
EXHAUST
1962-1972
A-BLOWER-TURBO-INTERCOOLER-FOR PART LOAD
B-TURBO-INTERCOOLER-FOR FULL LOAD
AIR INLET
COOLER
BLOWER
AIR INLET
ENGINE
EXHAUST
1972
TURBO – INTERCOOLER – BLOWER

Figure 1.

DESCRIPTION OF THE TURBO-BLOWER SERIES SCAVENGING SYSTEM

The turbo-blower series arrangement is an improved scavenging system just developed. See Fig. 1. The turbo blower system draws air into the turbo-compressor where it is compressed and discharged through a cooler to the engine driven roots type blower. This second stage blower, operating at a low pressure ratio, discharges the air directly into the engine air box. From here it flows through the cylinder intake ports. After the combustion cycle it passes out the exhaust ports through the exhaust driven turbine and finally out the stack.

The turbo-blower series arrangement substantially improves performance of Model 38TD8-1/8 (Turbocharged Diesel) and the Model 38TDD8-1/8 (Turbocharged Dual Fuel) engines. Power is increased by 10%. Exhaust smoke, exhaust temperature and wear are all reduced. On diesel the turbo-blower series arrangement system provides 20% more air for all operating conditions and lowers the engine exhaust temperature, piston temperature, and liner temperature. This provides longer engine life and decreased wear. The mechanical driven air blower also provides excellent response to sudden load changes.

THE NEW DESIGN OFFERS
SIMPLER PIPING AND LESS MAINTENANCE

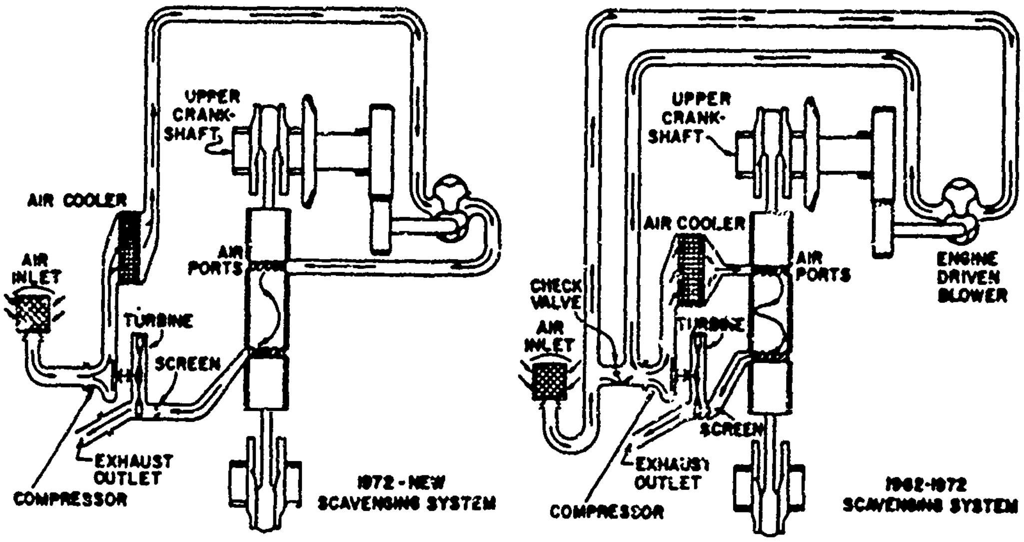


Figure 2.

In the original design the larger engine driven roots type blower was used as a first stage compressor. Air was discharged from it into the turbo-compressor. This roots blower was needed for starting and for part load operation. At full load this blower was by-passed by an automatic valve to save fuel. The original design was basically a blower-turbo series for start and a single stage turbo-system for full load. Because of this parallel system with extra piping and valving involved, the manifold was quite bulky and complicated. The turbo blower system requires less piping and has simplified controls.

A NEW SCAVENGING SYSTEM

The new turbo-blower series arrangement substantially improves performance of the Fairbanks Morse 8-1/8" bore opposed piston engine. Power is now increased by 10% with the potential of another 10% next year. Specific fuel consumption, exhaust smoke, exhaust temperature and wear are all reduced. This is accomplished through a significantly improved air scavenging system.

Figure 1, compares the improved scavenging system to the original system. The new system draws air into the turbo-compressor where it is compressed and discharged through a cooler to the engine driven roots type blower. This second stage blower, operating at a low pressure ratio, discharges the air directly into the engine air box. From here it flows through the cylinder intake ports. After the combustion cycle it passes out the exhaust ports through the exhaust driven turbine and finally out the stack.

In the original design a larger engine driven blower was required as the first stage. The compressed air was then discharged into the turbo-compressor. This blower was needed for starting and for part load operation. At full load this blower was by-passed by an automatic valve to save fuel.

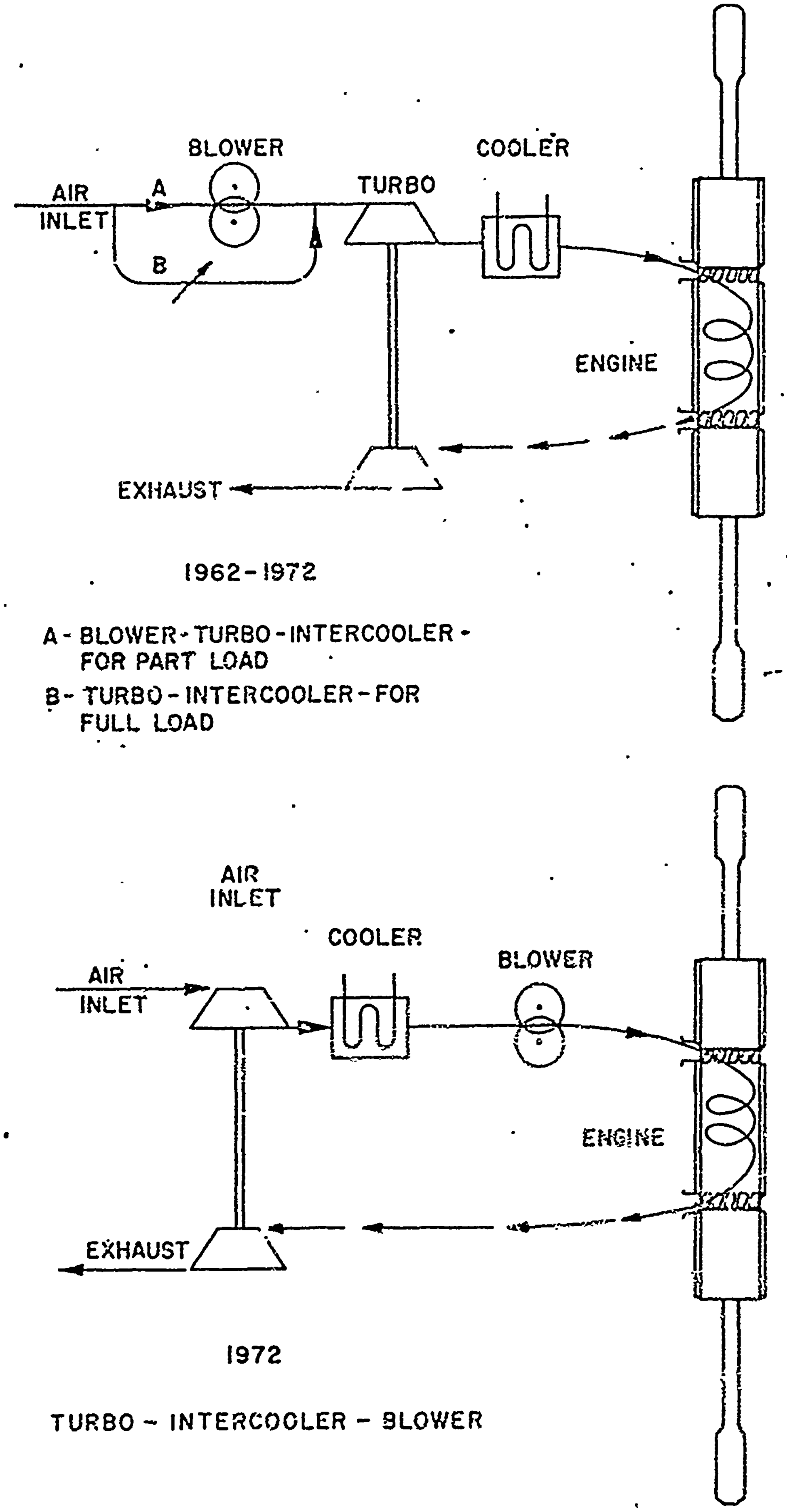
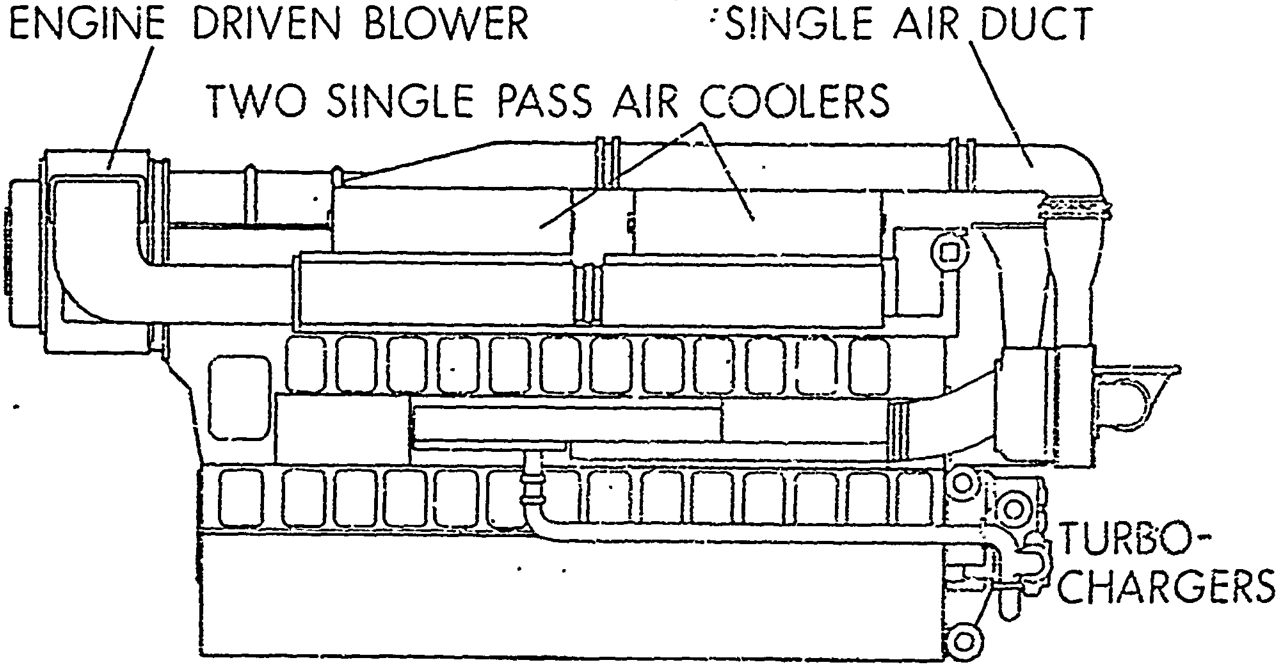

BLOWER
COOLER
AIR
INLET
A
TURBO
B
ENGINE
EXHAUST
1962-1972
A - BLOWER - TURBO - INTERCOOLER - FOR PART LOAD
B - TURBO - INTERCOOLER - FOR FULL LOAD
AIR
INLET
COOLER
BLOWER
AIR
INLET
ENGINE
EXHAUST
1972
TURBO - INTERCOOLER - BLOWER

Figure 1.

DEVELOPMENT PAYS OFF

Figure 3, shows that the familiar OP engine silhouette remains nearly intact. Clearly the switch to the improved scavenging system has appreciably simplified the front end piping without repositioning any of the major engine components. The number of parts for the air scavenging system has been reduced 50%, which will improve serviceability and reliability. Prior to release for production, the new system was thoroughly evaluated from a durability and performance standpoint. Three test engines, two sixes and a twelve, were used for this program, and many thousands of hours were accumulated.



NEW TURBO-BLOWER SERIES ENGINE

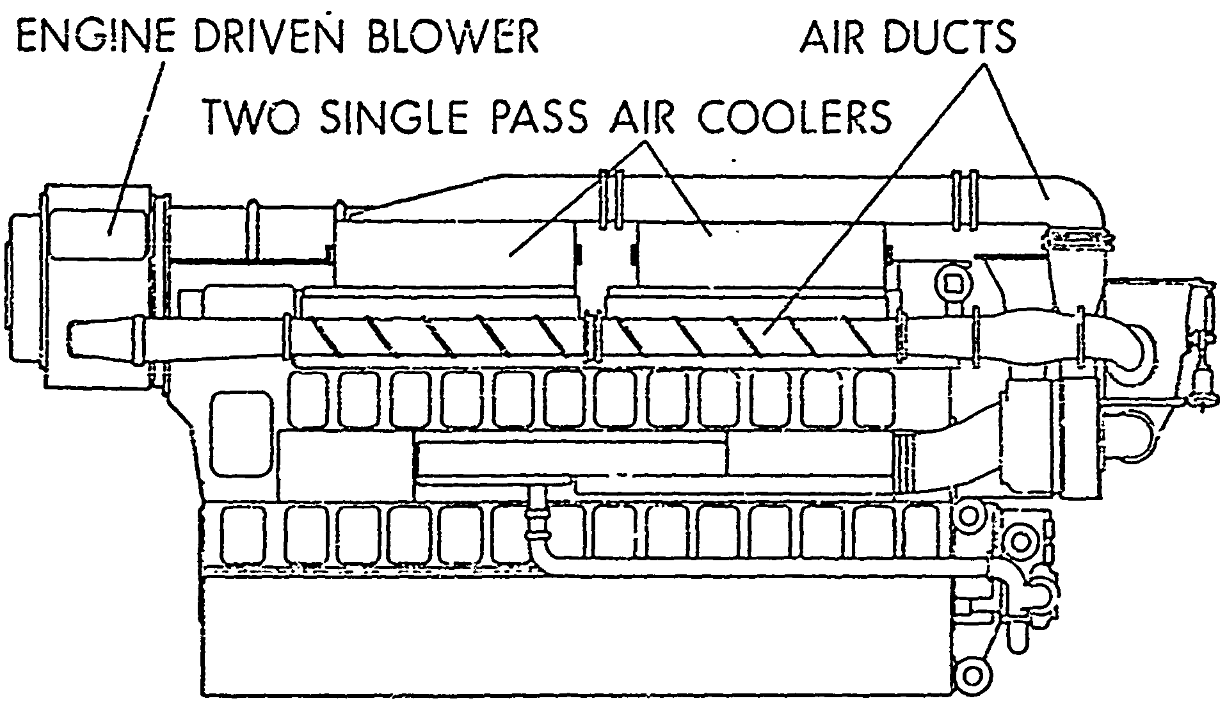


Figure 3.

ORIGINAL MODEL

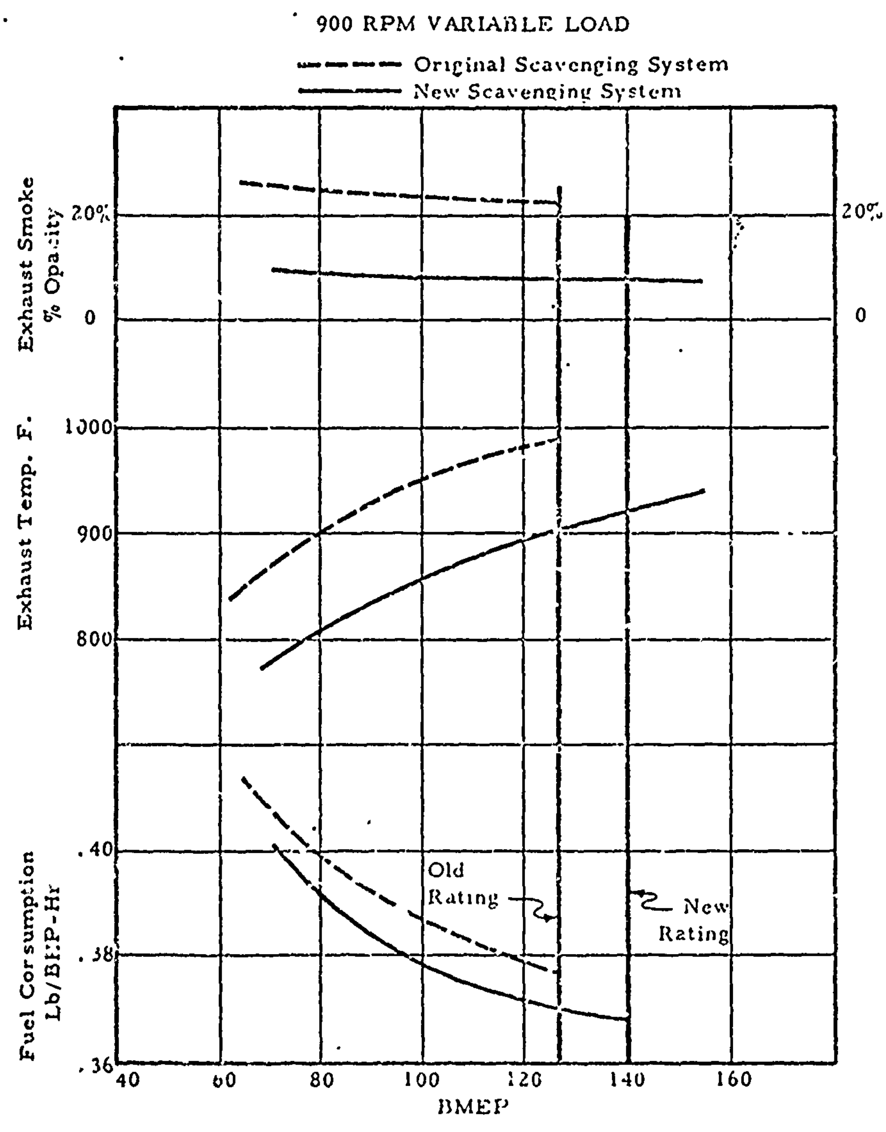
SMOKE REDUCTION

The improved 8-1/8 O-P will be offered in both the diesel and the dual fuel version. The performance is improved by the additional scavenging air. The exhaust temperature is reduced and the cylinder firing pressure has been kept under 1350 psi at the 140 bmep rating. The fuel economy is improved as shown in Figure 6.

Probably the most important improvement is the significant reduction of smoke and emission level. This is important due to increasing awareness and concern of the general public for reducing environmental pollution. Dual fuel and diesel engines, properly applied, are the least polluting and least obnoxious power source.

The smoke for the gas operation is not visible and for the diesel version, it is barely visible and certainly less than 10% opacity. This is well below the limit of many state and federal regulations.

Because of the very clean and lean combustion the exhaust emission level is reduced by 30% over previous designs.



900 RPM VARIABLE LOAD
Original Scavenging System
New Scavenging System
Exhaust Smoke % Opacity
20%
0
Exhaust Temp. F.
1000
900
800
Fuel Consumption Lb/BHP-Hr
.40
.38
.36
Old Rating
New Rating
40
60
80
100
120
140
160
BMEP

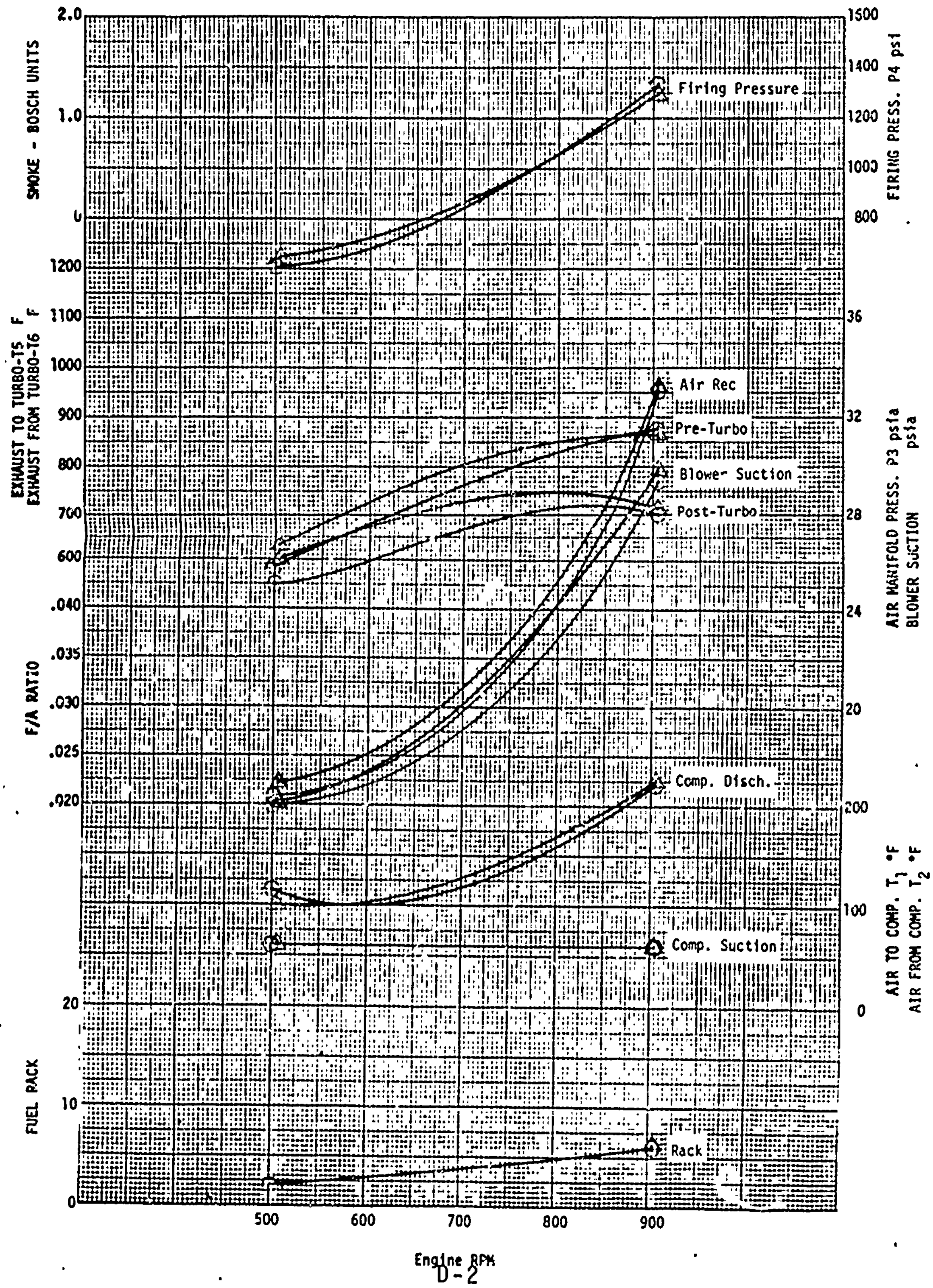
APPENDIX D

RESULTS OF TESTS ON CUTTER GALLATIN AFTER

TURBO-BLOWER SERIES CONVERSION

GALLATIN - WHEC 721
PERFORMANCE CURVE - VARIABLE LOAD
Two Main Diesel Engines - Balanced

ENGINE ○ = #1 MDE △ = #2 MDE	LINER 16 302 289	FUEL #2
SERIAL NO. 380868007/380868005	PISTON 15 400 333/16 400 334	AMBIENT TEMP. 60°F
BORE/STROKE 8-1/8 x 10	AO 141° AC 255°	BAROMETER 30.28
COMPRESSION RATIO 11:1	EO 117° EC 243°	AIR MANIFOLD TEMP. 130-140°F
TURBOCHARGER Elliot H-56	FUEL CAM Production	ENGINE SPEED Var rpm
NOZZLE RING AREA 21 sq. inches	INJECTION P.C. 2° BIDCLC	LOG NO. --
DIFFUSER HEIGHT .740 inch	GAS VALVE CAM None	DATE RUN 5-23-77
FUEL NOZZLE 16 705 664	GAS VALVE OPEN None	DIESEL CONFIGURATION



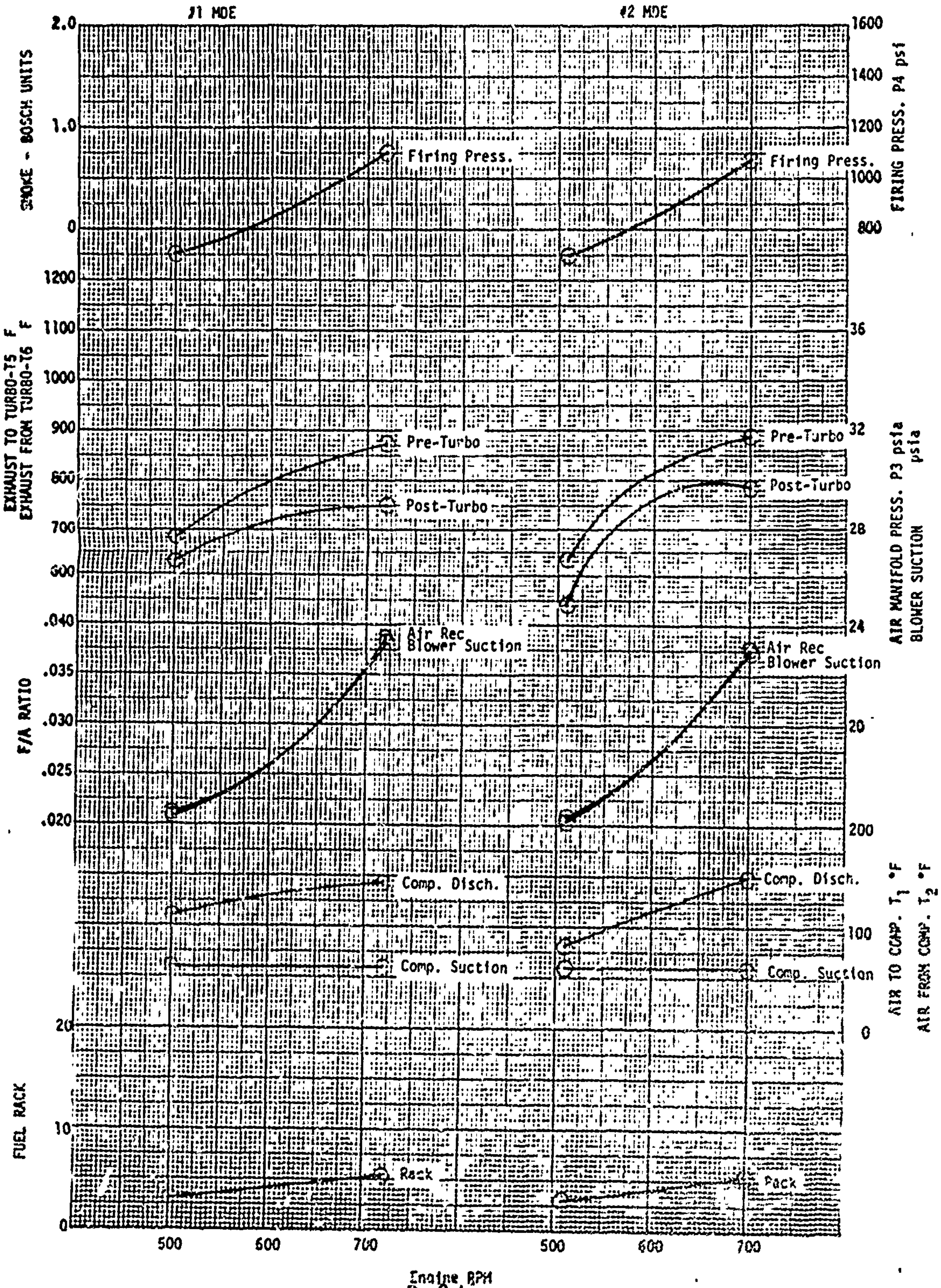
DRAWN BY: D. Pickering
DATE: 6/3/77

Form D

CURVE NO. 63
SHEET 1 OF 1

GALLATIN - WHEC 721
PERFORMANCE CURVE - VARIABLE LOAD
One Engine Operating/One Propeller Free-Wheeling

ENGINE Main Diesels	LINER 16 302 289	FUEL #2
SERIAL NO. 38D868007/38D268005	PISTON 16 400 333 & 334	AMBIENT TEMP. 60°F
BORE/STROKE 8-1/8 x 10	AO 141° AC 255°	BAROMETER 30.23 in.hg.
COMPRESSION RATIO 11.0:1	EO 117° EC 243°	AIR MANIFOLD TEMP. 130-140°F
TURBOCHARGER Elliott H-56	FUEL CAM Production	ENGINE SPEED Var rpm
NOZZLE RING AREA 21 sq. inches	INJECTION P.C. 2° BIDCLC	LOG NO. --
DIFFUSER HEIGHT .74 inch	GAS VALVE CAM None	DATE RUN 5-23-77
FUEL NOZZLE 16 705 664	GAS VALVE OPEN None	DIESEL CONFIGURATION



DRAWN BY: R. Pickering
DATE: 6/3/77

Form D

CURVE NO. 64
SHEET 1 OF 1

APPENDIX E

PISTON, LINER, AND INJECTOR VARIABILITY

U.S.C.G.C. 378' WHEC

MAIN DIESEL ENGINE PISTONS

The Coast Guard is changing from the rotating piston design to fixed pistons on the subject cutters. Fairbanks Morse recommends that the latest production fixed piston be used. (P/N 16401902 Piston, 16508129 Piston Insert, 16704740 Capscrews and 0014012.02.10 Lockplates) All other piston assembly components are the same for all turbocharged engine pistons. A complete list of parts for the piston assemblies is shown on parts list Catalog No. 5.6.

This new piston is used in new turbo blower series scavenged engines and is the piston recommended by Fairbanks Morse for turbo blower series conversion kits. The following precautions should be adhered to when installing the subject piston assembly:

1. Always install as a cylinder set to maintain the correct compression ratio.

2. Only install in a new cylinder liner or a liner that has operated with the rotating style piston. Cylinder liner wear at inner travel of the piston ring with the Mexican Hat fixed piston will lead to ring breakage if the new piston is installed in such a used liner.

3. The injection nozzles should have the 15 degree angle holder. The 15 degree old style nozzle holder would be P/N 16200805, all other parts are the same as the 10 degree old style nozzle.

   Gasketless 15 Degree Nozzle -

Nozzle Kit, Consisting of 4 Items;	16609071
1 - Injection Nozzle Assembly	16705667
1 - Injection Tube Assembly	16204287
1 - Nut - Injection Tube	92004341
1 - Sleeve - Injection Tube	92004353
1 - Collar - Nozzle Hold Down	16108370

   The 10 degree gasketless nozzle can be converted to 15 degree with the following parts:

1 - Injection Nozzle Holder	03730100
1 - Injection Tube Assembly	16204287
1 - Nut - Injection Tube	92004341
1 - Sleeve - Injection Tube	92004353
1 - Injection Nozzle Tip	16102147

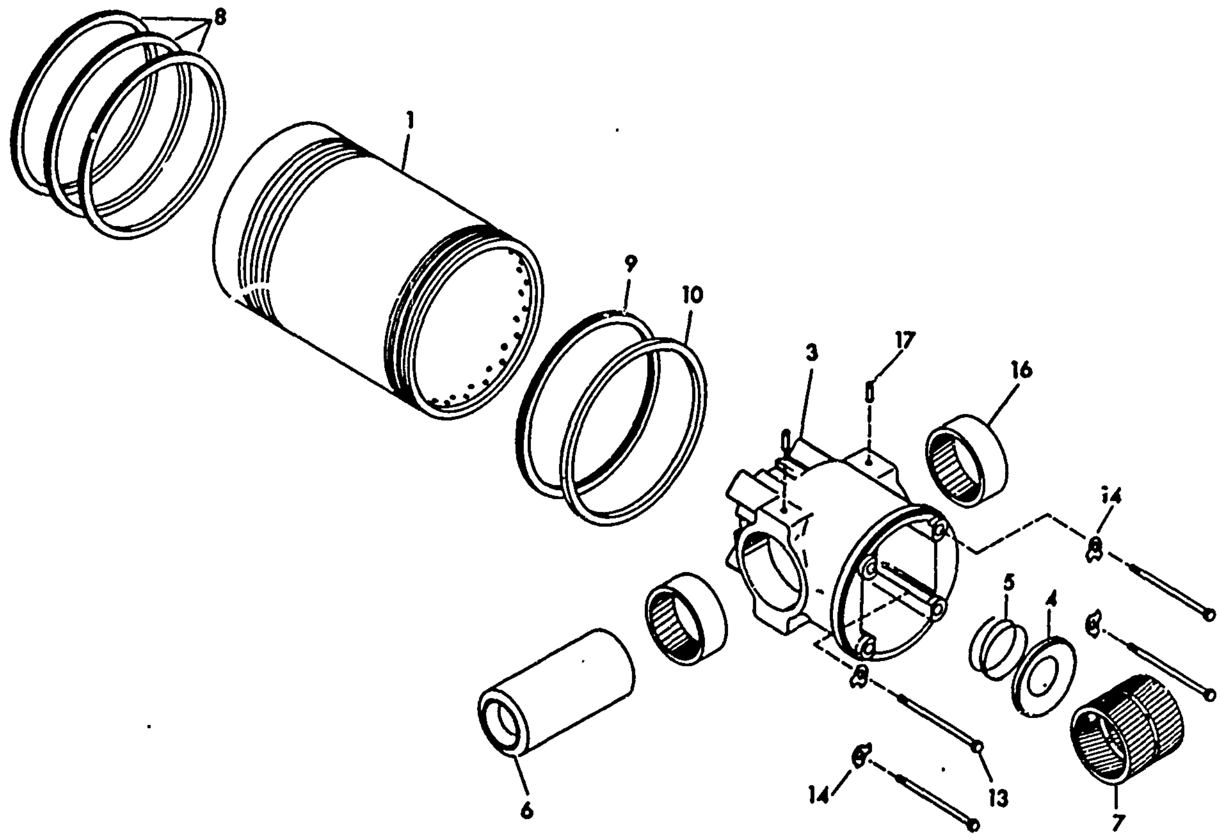
   The 15 degree injection nozzles will work nearly equally well with any of the piston combinations available for the turbocharged 8-1/8 engine.

PIECE NO.	NAME OF PART	PART NUMBER	NUMBER USED *
1	Lower Pist - Complete	16 608 300	1
1	PISTON, Lower	16 401 902	1
3	Piston Insert - Assembly (Pc. 3, 16 & 17)	16 608 129	1
16	BUSHING, Piston Insert	16 300 358	2
17	PIN, Bushing Lock	16 101 188	2
4	RETAINER, Oil	16 101 189	1
5	SPRING, Oil Retainer	16 101 190	1
6	PIN, Piston	16 200 274	1
7	BUSHING, Piston Pin	16 701 767	1
8	RING, Compression	16 704 845	3
9	RING, Oil Scraper	16 300 217	1
10	RING, Oil Drain	16 101 191	1
13	BOLT, Hex	16 704 740	4
14	LOCKPLATE	0014012.02.10	4

* No. Used per Piston

See Page 2 for Upper Piston

NOTE: No Oil Drain Ring used in Piston Skirt End Groove.

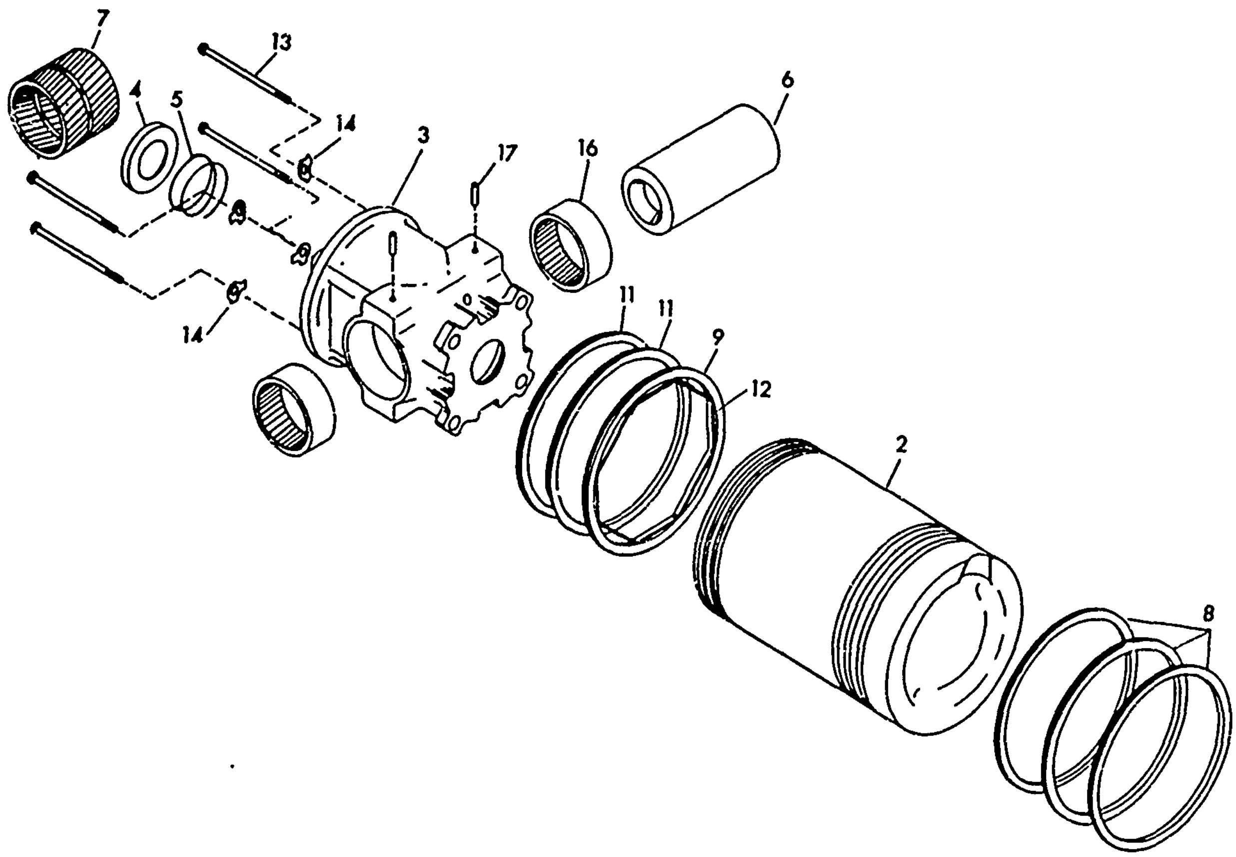


LOWER PISTON

Renewal Parts List

PIECE NO.	NAME OF PART	PART NUMBER	* NUMBER USED
2	Upper Piston - Complete	16 608 301	1
2	PISTON, Upper	16 401 902	1
3	Piston Insert Assembly - (Pc. 3, 16 & 17)	16 608 129	1
16	BUSHING, Piston Insert	16 300 358	2
17	PIN, Bushing Lock	16 101 188	2
4	RETAINER, Oil	16 101 189	1
5	SPRING, Oil Retainer	16 101 190	1
6	PIN, Piston	16 200 274	1
7	BUSHING, Piston Pin	16 701 767	1
8	RING, Compression	16 704 845	3
9	RING, Oil Scraper	16 300 217	1
11	RING, Oil Drain	16 101 191	2
12	RING, Oil Scraper Expander	16 101 192	1
13	BOLT, Hex	16 704 740	4
14	LOCKPLATE	0014012.02.10	4

* No. Used per Piston



UPPER PISTON

U.S.C.G.C. (378') MAIN ENGINE PISTONS

The engine build pistons were of rotating type for the 378' cutters, except for the cutters Munro, Jarvis and Midgett which were built with the Mexican Hat style fixed pistons.

A. Rotating Piston

The rotating piston does not have a heat dam above the compression ring belt and uses a 3/8 inch thick crown with a rotating bearing between the connecting rod insert and the piston. The insert and piston are retained to one another by plates bolted to the insert which run in a radial piston groove. A 1/32 inch clearance in the groove above the lock plates allows for rotating bearing wear and piston length change relative to expansion of the insert and piston crown support struts. If these dimensions get out of tolerance due to wear or temperature expansion, the lock plates start carrying the firing loads. This result causes rapid fretting wear of the rotating bearing and possible complete destruction of the piston.

The rotating piston has a high ring belt. The top ring is 13/16 inch from the crown edge. It allows for minimum mechanical loading of the cylinder liner by exposing a minimum area of the liner to firing pressures. Because of its construction it also runs quite hot and causes the liner to run hotter than necessary, which can lead to thermal loading in the liner.

The ring life is marginal with the rotating piston at 900 rpm engine ratings and yet quiet acceptable at the 720 rpm rating.

B. Mexican Hat Fixed Pistons

The Mexican Hat piston was released for the turbocharged diesel engine in particular because of marine engine problems with the rotating piston. This piston was only used in diesel engines. It greatly improved ring wear with some increase in engine smoke. It also eliminated the rotating piston bearing.

This piston has a heat dam above the top compression ring and a crown thickness of 5/16 inch. It runs cooler both on the piston crown and at the top ring than the rotating piston.

The top ring is located 1-1/2 inches from the crown edge and since this piston has a larger cup volume, it has 0.1 inch less minimum clearance. This allows 1.275 inches more liner bore length to be exposed to combustion gases than with the rotating piston combination.

The upper and lower pistons are not interchangeable for either the rotating combination or the Mexican Hat combination.

The Mexican Hat piston results in extra blow back into the air receiver because of the low ring belt and requires more frequent cleaning of the air ports and air box to prevent possible air box fires.

C. Fixed Piston - Turbo Blower Series

The new piston was released for production along with the turbo blower series scavenging system, diesel and dual fuel version, with a 10% rating increase from 127.3 bmep to 140 psi, bmep. This piston along with the scavenging system gave lower ring wear at the higher rating than obtained with the earlier engine. The top ring is located at 1-3/16 inches from the crown top edge and incorporates a heat dam design. The crown is 5/16 inch thick and has under crown oil cooled support fins. This piston is being successfully laboratory endurance tested at a continuous rating of 148.5 psi, bmep and 350 hp/cylinder at 900 rpm.

The piston rings and crown run very cool, particularly with the turbo blower series scavenging air system. These pistons have a crown shape similar to the rotating piston and were designed to use the same piston for either upper or lower position.

The upper and lower piston assembly components are the same including the piston, insert, capscrews and lock plates.

Piston P/N	16401902
Insert P/N	16608129
Capscrew P/N	16704740 4/Piston
Lock Plates P/N	0014012.02.10 4/Piston

The only precaution one must adhere to when using these new pistons in the injection nozzles must have 15 degree angle holders. That is, either 16200805 holders for the old style nozzle of 16705667 for the gasketless nozzle assemblies. To my knowledge all the cutters, except the three mentioned earlier, have the 15 degree holders. All engines out of Governors Island have been or are being converted to gasketless nozzles.

Fairbanks Morse strongly recommends that all further pistons purchased for turbo engines on the 378s be the new style and engines with a few Mexican Hat pistons have them removed and given to ships with all Mexican Hat pistons.

D. Enclosures on the New Style Piston Relative to the Older Style Pistons

1. Sales pitch on the Improved Piston Cooling and Ring Wear. This data was developed by marketing people in conjunction with management for sales purposes of the newly rated engine.

2. Piston 16401902 (new style) temperatures are shown on curves developed 10/27/73 at 720 and 900 rpm (E-2). Note the low temperatures at the #1 piston ring area; less than 300°F at loads up to 170 psi, bmep. The 720 rpm temperatures generally run higher than the 900 rpm ratings due to the reduced flow of cooling oil to the piston. The air-fuel ratio is nearly constant while the temperature versus time exposure of the piston combustion surface is nearly identical for both speeds. The center of the piston temperature is well below the acceptable limit of 800°F.

These temperatures are with turbo blower series scavenging system.

3. The enclosures E-3 and E-5 show the temperatures of the rotating and Mexican Hat pistons with the older style scavenging system. One can readily see the improvement of the Mexican Hat piston relative to the rotating piston in the ring belt area. The Mexican Hat piston temperatures are about 300°F at a point below the top ring, while the rotating piston temperatures were between 400 and 450°F below the ring and very hot directly above the ring, 750 to 890°F. The heat flow through the top ring to the cylinder liner is considered to be a good share of the piston ring cooling with the rotating design. Note the relationship of the top ring to the top of the crowns for these two piston designs.

4. The enclosure E-4 shows the new piston and cylinder liner temperatures at important locations while operating at 900 rpm with turbo blower series air system. These low temperatures give very good operating results for cast iron rings against chrome.

5. Enclosure E-5 shows piston and liner temperatures of the rotating piston and diesel liner at 720 rpm with the new turbo blower series scavenging air system. Note, while these temperatures are much lower than obtained with the old scavenging air system (piston E-3, January 1967), they are still very high compared to the new piston with turbo blower series scavenging (E-4). The liner temperatures at the exhaust ports are between 300 and 400°F while the piston environment at the top ring is between 400 and 600°F. The ring definitely depends on the liner wall for cooling to an acceptable operating temperature.

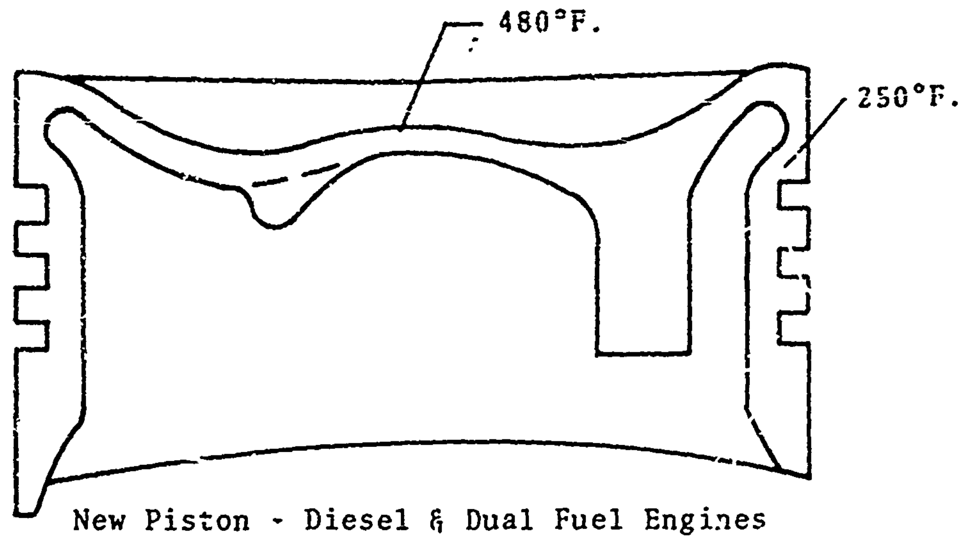
   The high liner temperature will lead directly to liner seal failure with the slightest malfunction as broken top piston ring or momentary loss of jacket water circulation.

IMPROVED PISTON COOLING AND RING WEAR

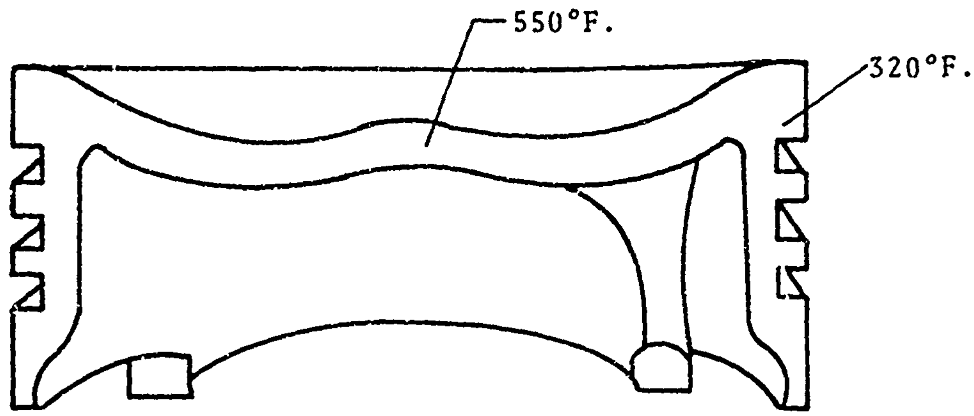
A new and cooler piston design has been developed as shown in Figure 4.

This piston, equally suitable for both diesel and dual fuel engines, has significantly lowered the crown and top ring groove temperatures. This improved cooling contributes to longer piston and piston ring life. This improvement is the result of a modification of the undercrown cocktail shaker design above the top ring, and a new piston crown with proven combustion characteristics.

This new piston is suitable for both upper and lower cylinders, and replaces four older designs in the system.



New Piston - Diesel & Dual Fuel Engines



Old Piston - Dual Fuel Engines

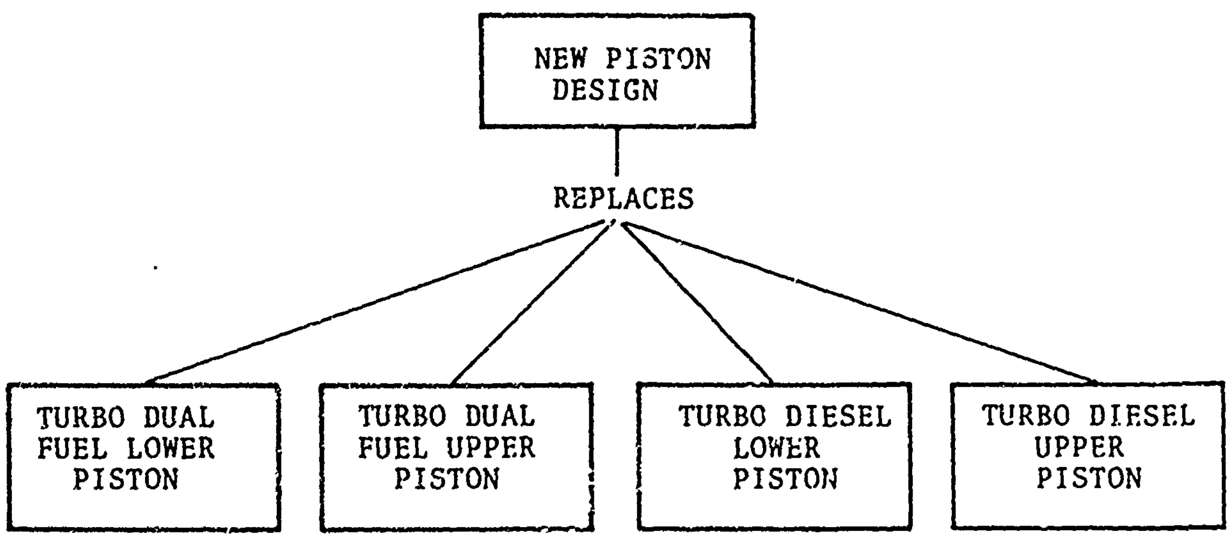


FIGURE 4

ENCLOSURE E-1.

RINGS AND LINERS

The wear of the piston ring and liner has been reduced by a factor of 2 to 3 times, as a result of thousands of hours of laboratory testing to select the best combination. The combination selected for production is shown in Figure 5. The life of the cast iron piston is further improved due to the better cooling, explained earlier. The top compression ring is crowned, high strength cast iron and ferrox filled. Excellent oil control is obtained with an oil ring combination consisting of a conformable two piece scraper seal ring in the inner groove and a single edge ventilated scraper in the second groove. A steel expander is used with the seal ring on the upper pistons only.

A the new rating, the lower piston top ring wear, which is the most servere one due to the exhaust porting, will be less than .004" per 1000 hours. This provides a ring life of over 10,000 hours for normal operation.

The cast iron liner, with the proven strong back design, has a new chrome finish to improve the liner wear. It is a chrome finish with a mechanically produced oil retaining surface pattern supplied by the Chromium Corporation of America. The laboratory testing indicates that the liner life should be increased by a factor of 2 over the old design, and that replating should not be required before 20,000 hours for normal operation.

Enclosure E-1.

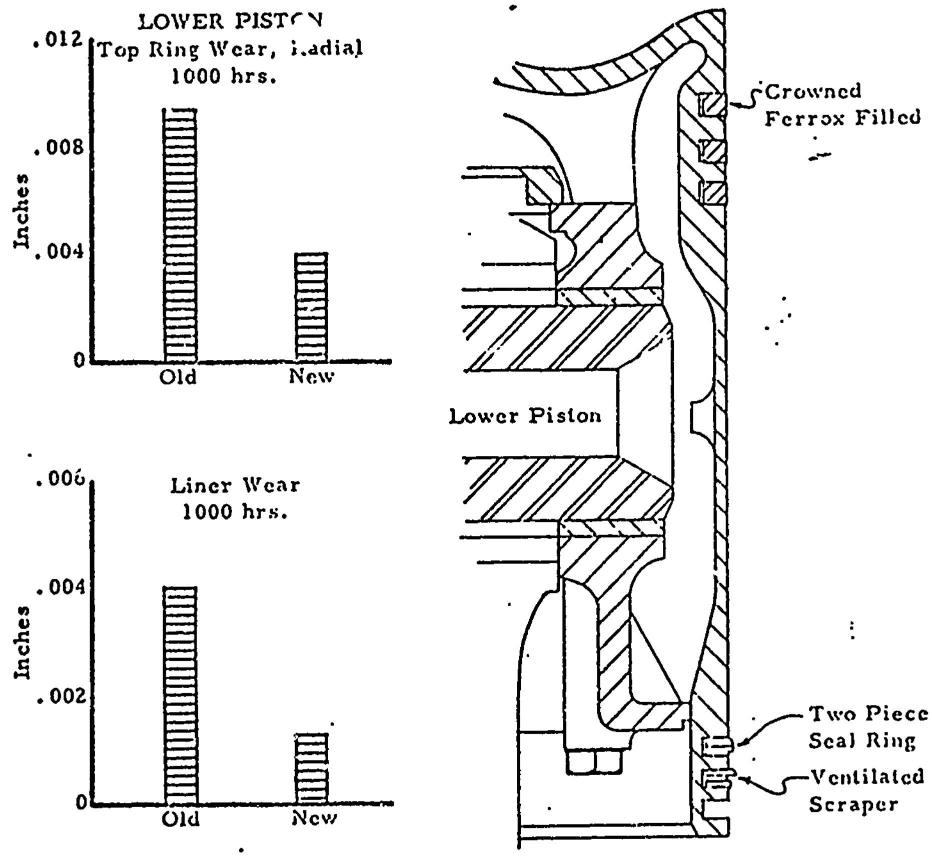
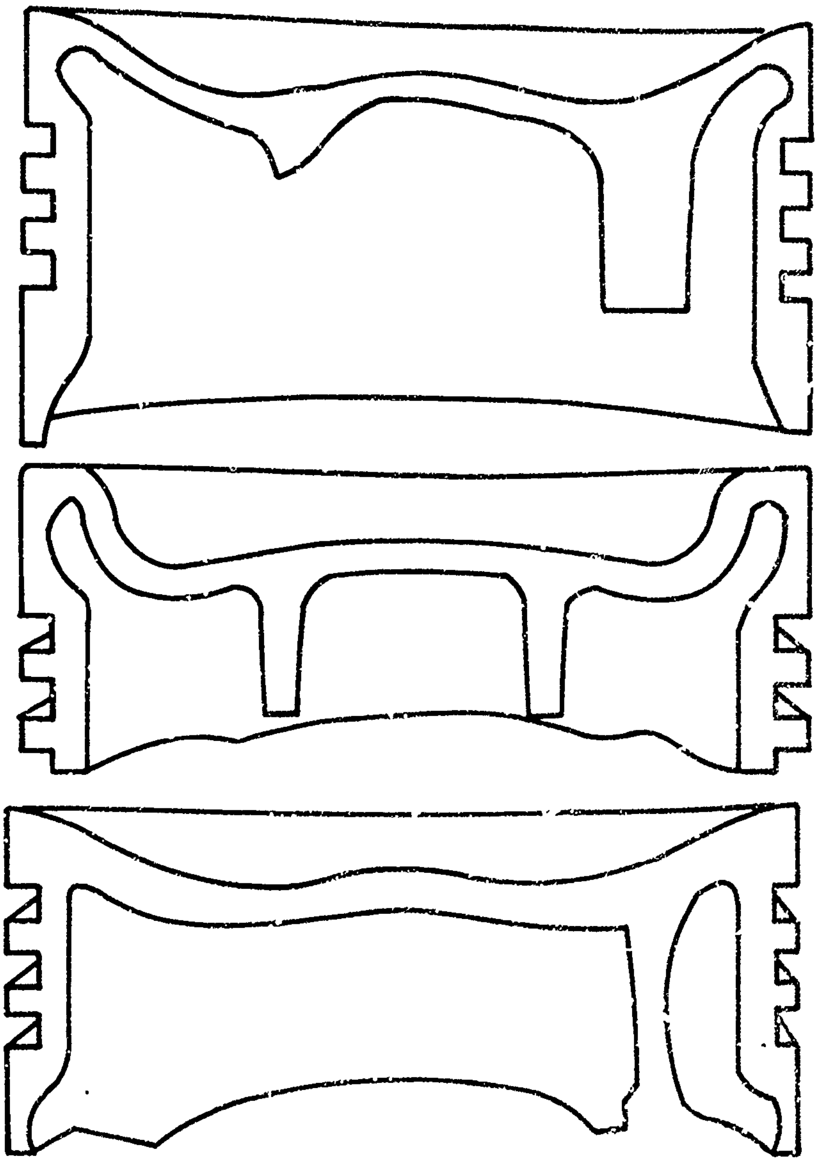


Figure 5.

ENCLOSURE E-1.

38TD8-1/8 ENGINE PISTONS



PRODUCTION PISTON

MEXICAN HAT PISTON

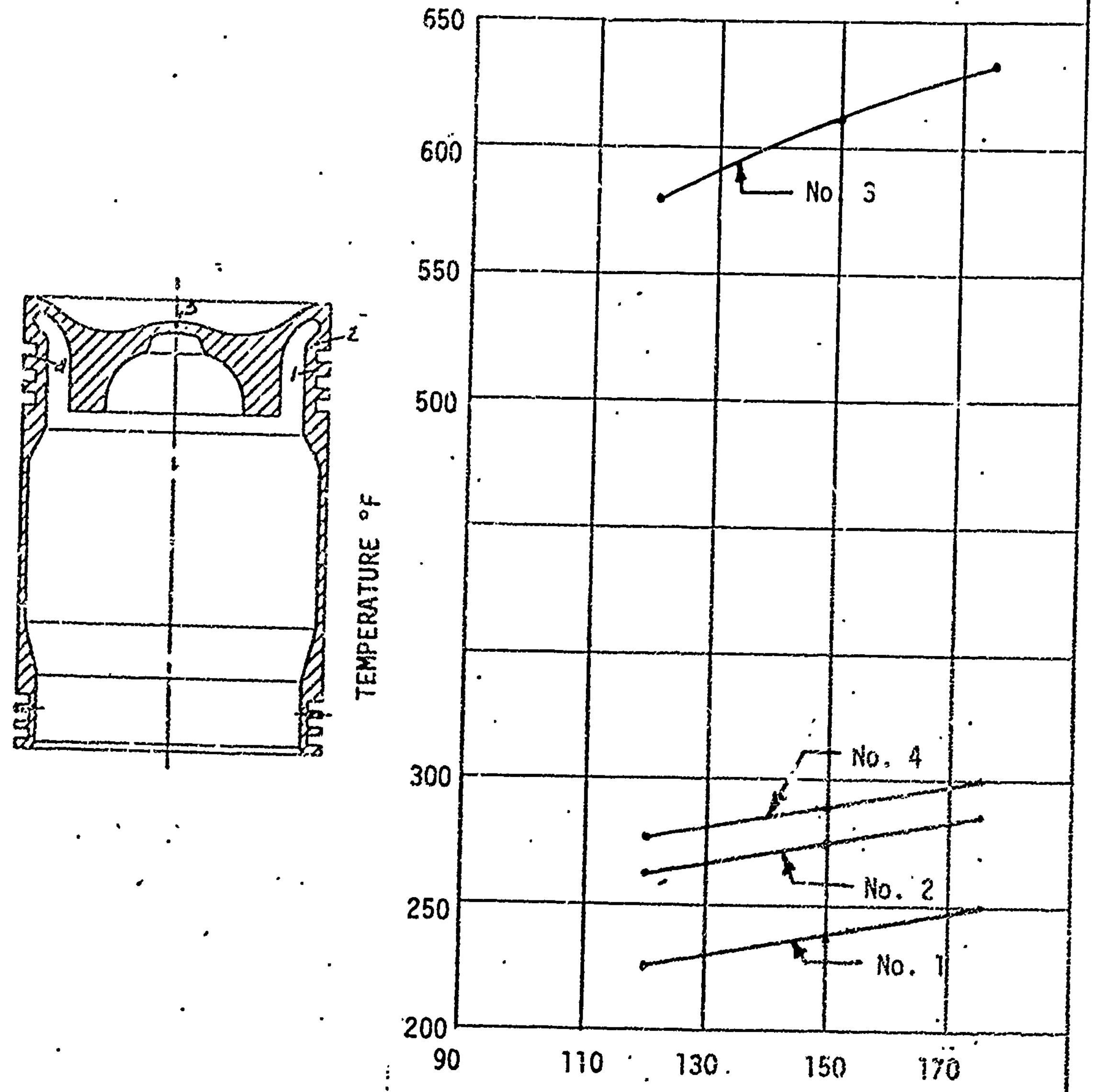
ROTATING PISTON

PISTON CROWN CROSSECTION

ENCLOSURE E-1.

PISTON TEMPERATURES
8-1/8 x 10 TURBOCHARGED (E-6)
BLOWER SERIES SYSTEM
DIESEL @ 720 RPM

TURBO:	H-56
NOZZLE RING:	16 in^2
TIMING:	14.5 BMV
BLOWER:	1.23:1 Vol.
PISTON:	16 401 902
LINER:	16 401 992

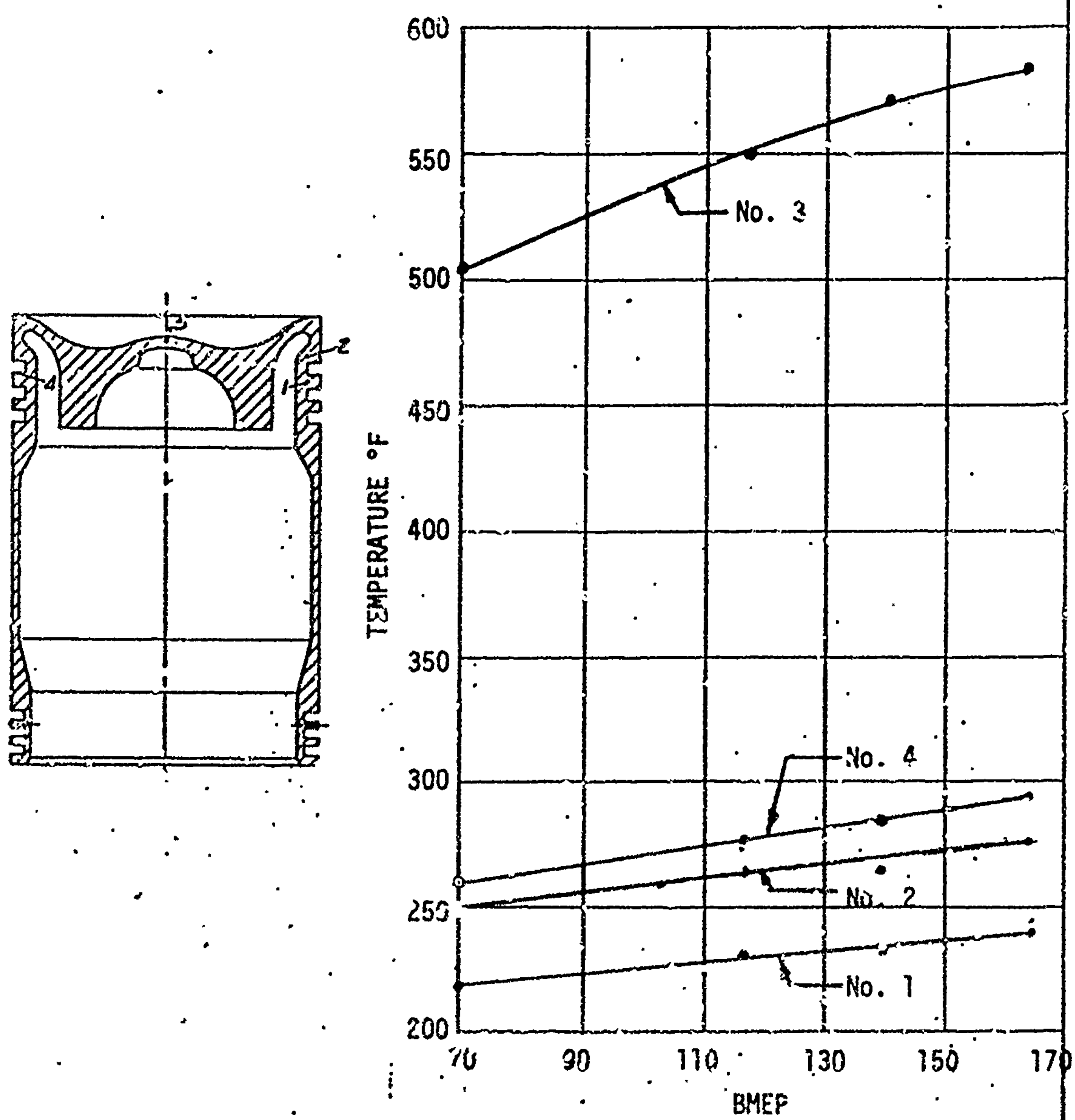


ENCLOSURE E-2

Figure No. ____
By: E. Kasel
Date: 10/27/73

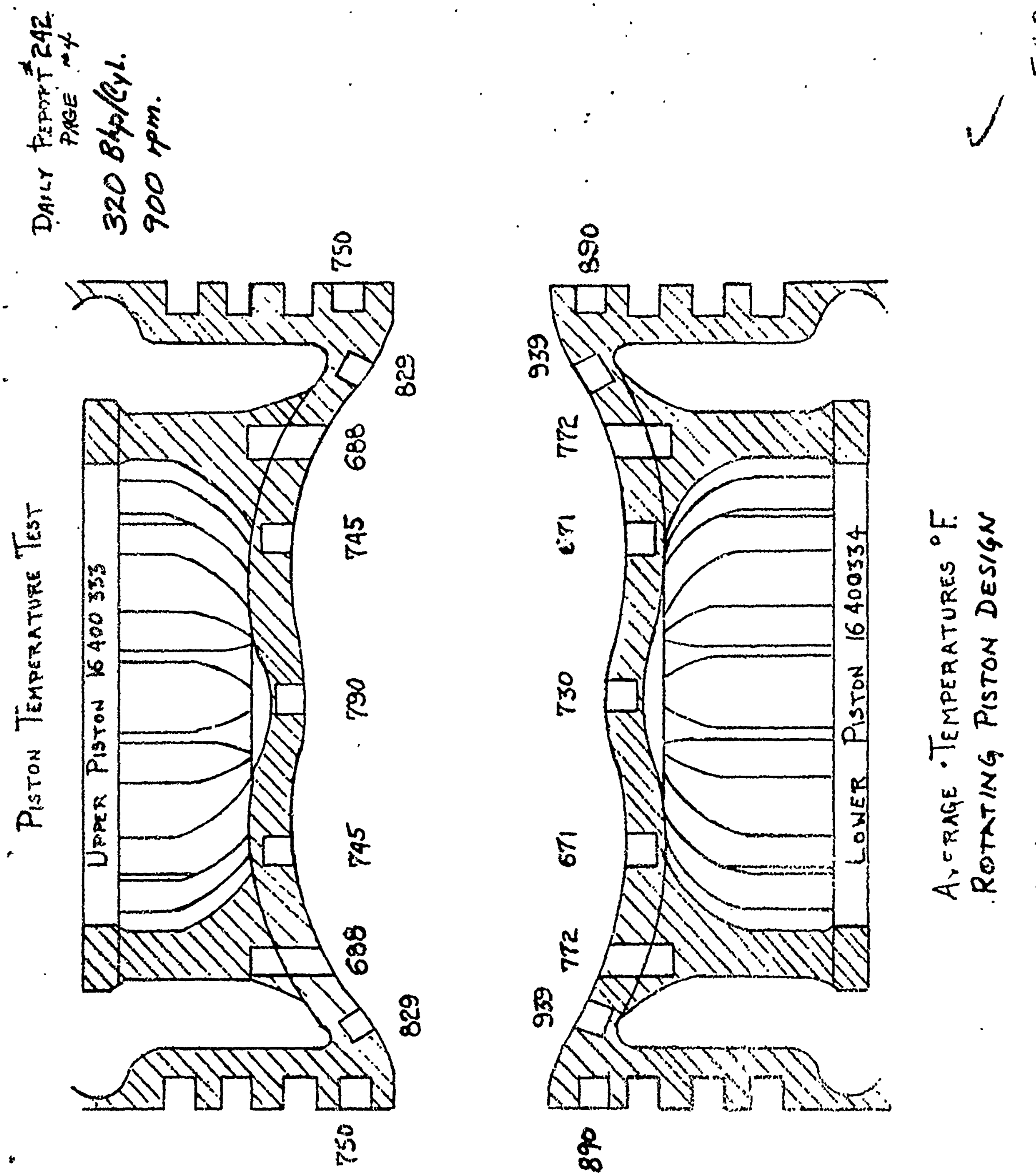
PISTON TEMPERATURES
8-1/8 x 10 TURBOCHARGED (E-6)
BLOWER SERIES SYSTEM
DIESEL @ 900 RPM

TURBO: H-56
NOZZLE RING: 18 in^2
TIMING: 12.5 BMV
BLOWER: 1.23:1 Vol.
PISTON: 16 401 902
LINER: 16 401 992

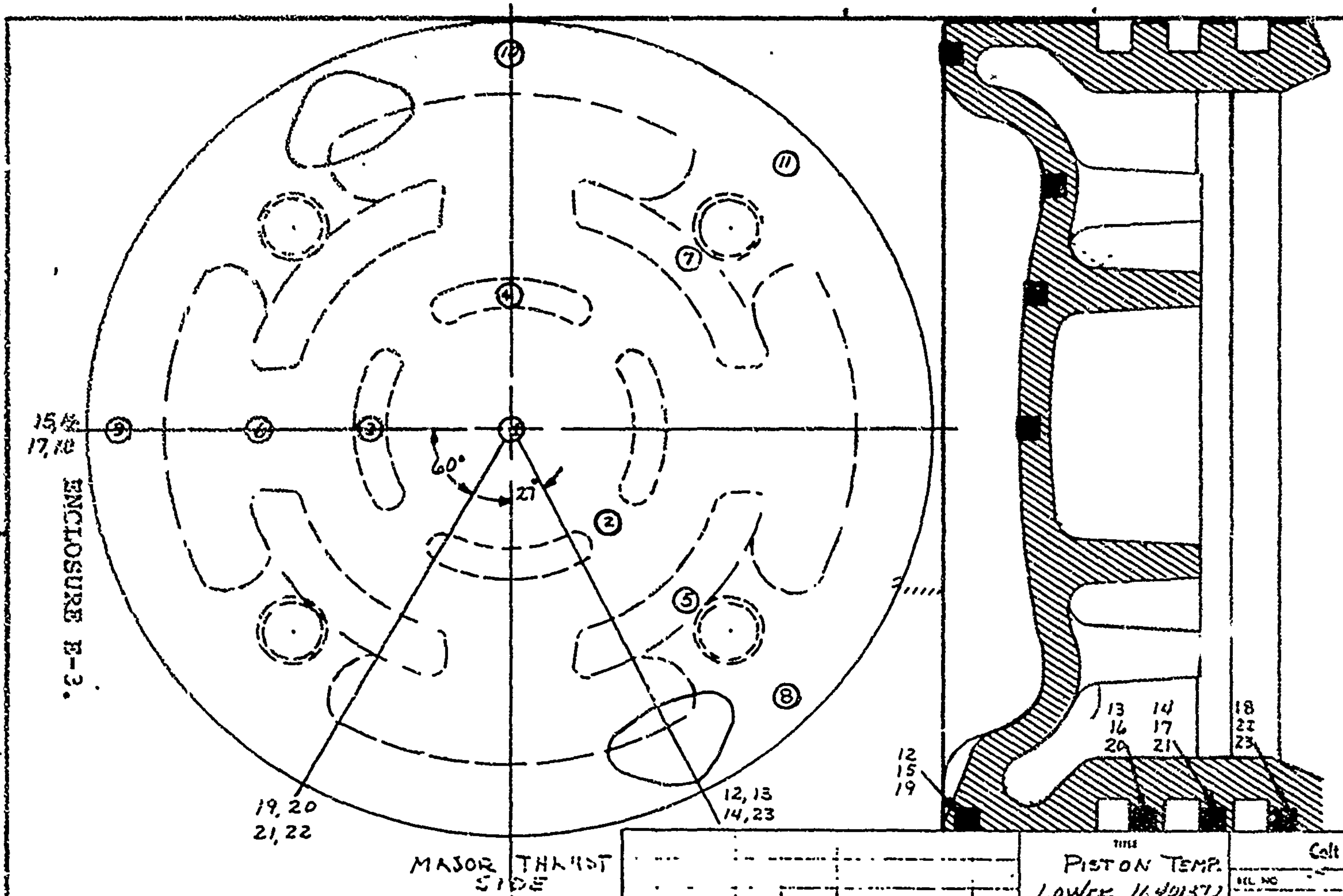


ENCLOSURE E-2.

Figure No. ____
By: E. Kasel
Date: 10/27/73



ENCLOSURE E-3.



ENCLOSURE E-3.

TEMP. PLUG LOCATION	TEMP. COMB. SURFACE	TEMP. 0.2" IN FROM SURFACE
1	790°F	660°F
2	860°	700°
3	790°	690°
4	830°	700°
5	690°	570°
6	640°	550°
7	690°	570°
8	890°	790°
9	750°	660°
10	720°	660°
11	810°	750°
12	750°	700°
13	300°	300°
14	300°	300°
15	620°	620°
16	300°	300°
17	300°	300°
18	300°	300°
19	980°	980°
20	300°	300°
21	300°	300°
22	300°	330°
23	300°	300°

Piston temperatures shown [redacted] were obtained while operating one hour at 320 BHP/Cyl. in cylinder #4 lower position. The plugs were threaded and 0.2 inch long made of temperature sensitive material and were located in the piston as shown. The accuracy of the data is believed to be within 50°F., except temperatures recorded at location 19. This temperature should have been less than that recorded by plugs 11 and 12 and greater than that recorded by plugs 9 and 15. Inspection of the test piston indicated that plug 19 was not properly bottomed at assembly thereby reducing its heat transfer to the piston.

TITLE PISTON TEMP. LOWER 16401372 MEXICAN HAT

Colt Industries

DR. BY KAK

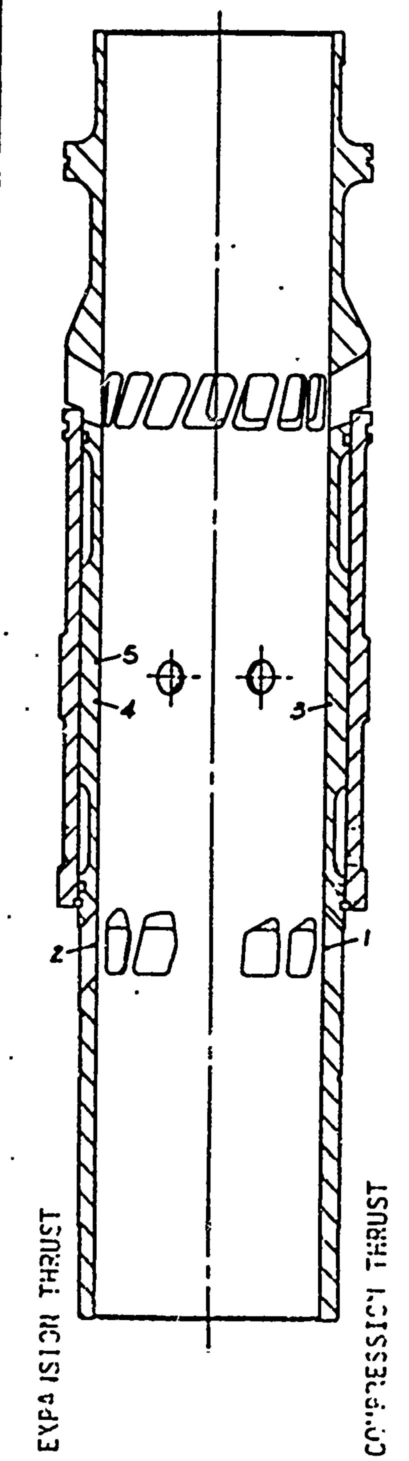
MATL 320 BHP/cyl. - 900 RPM

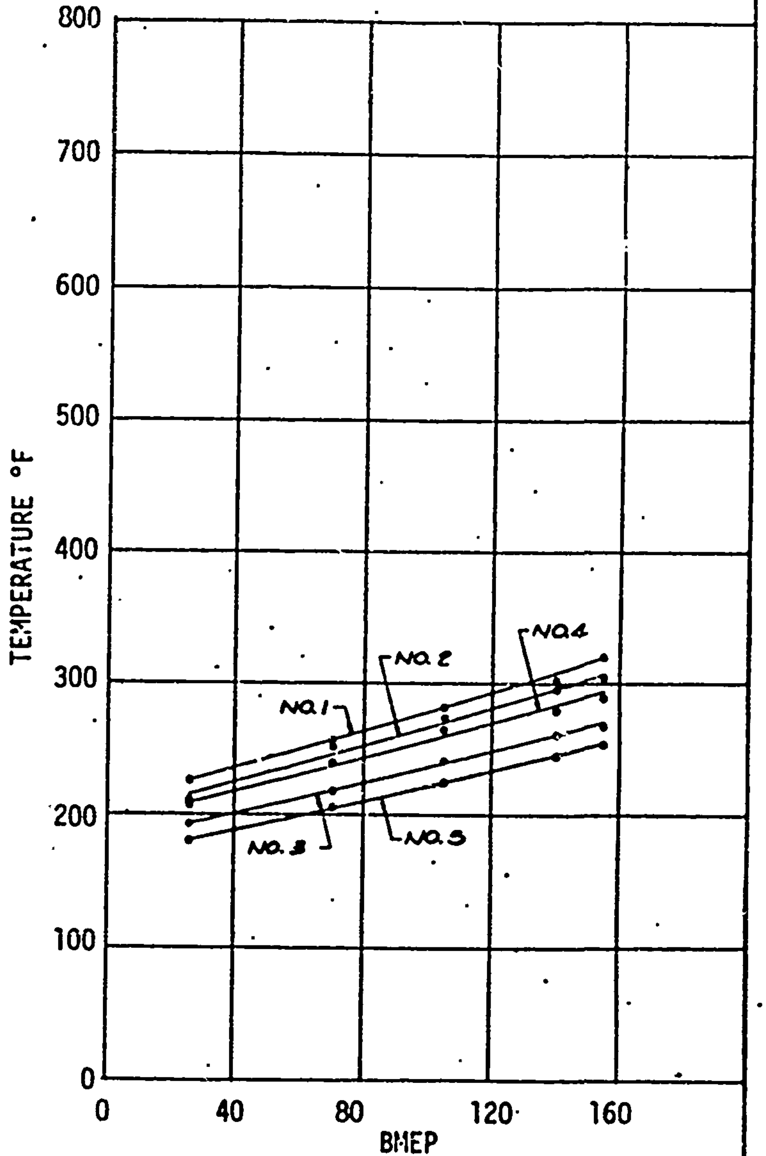
SIZE B DWG. NO. DR. #255

LINER TEMPERATURES
8-1/8 x 10 TURBOCHARGED
SERIES - BLOWER SYSTEM
DIESEL @ 900 rpm

PISTON 16401902

LINER 16 401 992





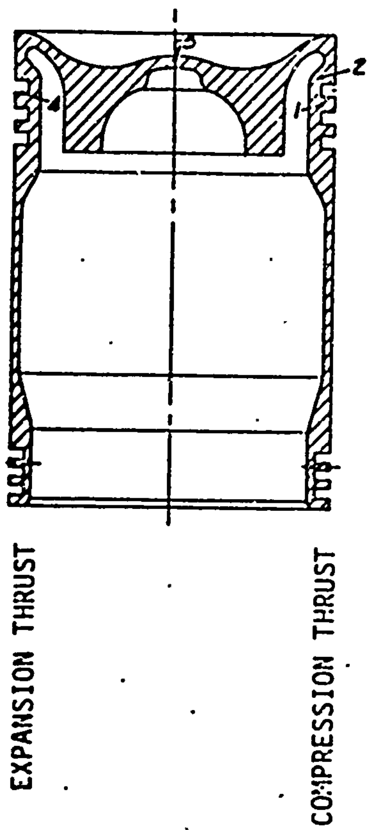
ENCLOSURE E-4.

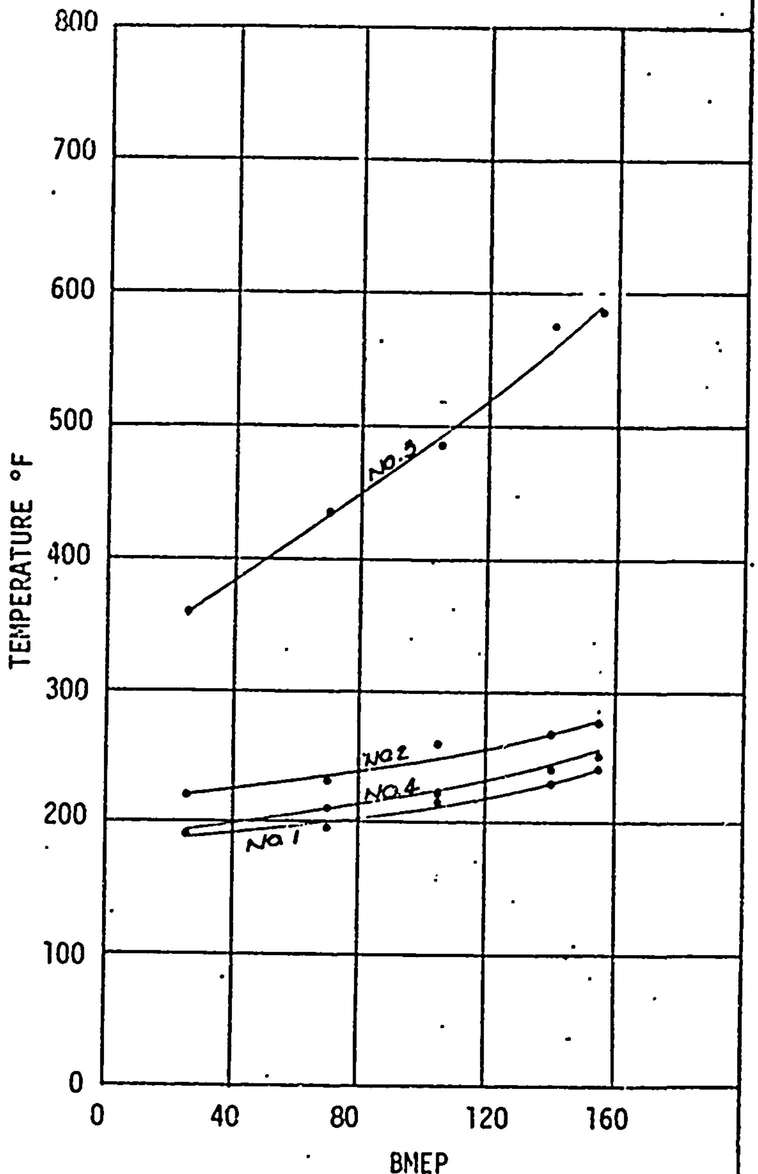
DATE: 4/13/72
BY: J.J.

PISTON TEMPERATURES
8-1/8 x 10 TURBOCHARGED
SERIES - BLOWER SYSTEM
DIESEL @ 900 rpm

DIESEL LINER 16 401 992

PISTON 16 401 902





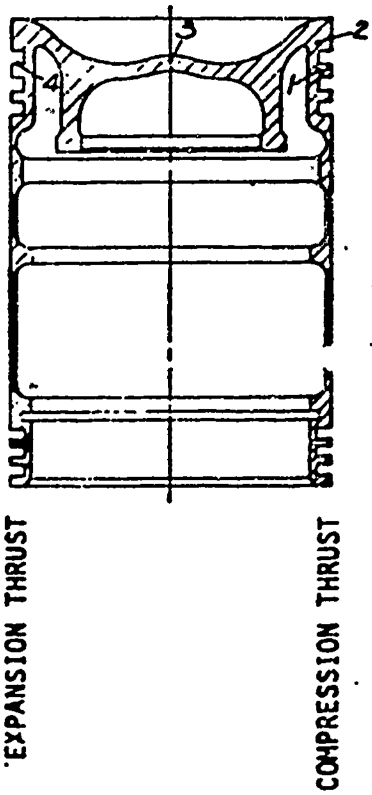
ENCLOSURE E-4.

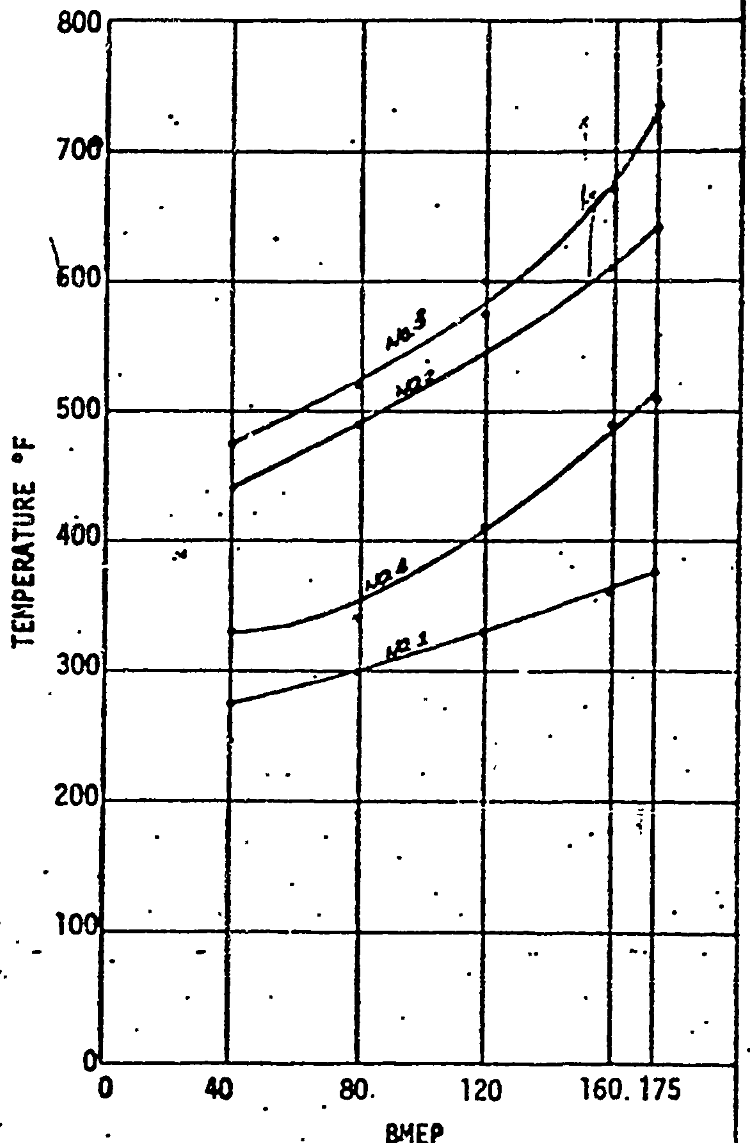
DATE: 4/13/72
BY:

PISTON TEMPERATURES
8-1/3 x 10 TURBOCHARGED
SERIES - BLOWER SYSTEM
DIESEL @ 720 rpm

DIESEL LINER 16 401 992

PISTON 16400334





— ROTATING PISTON

FIG. II

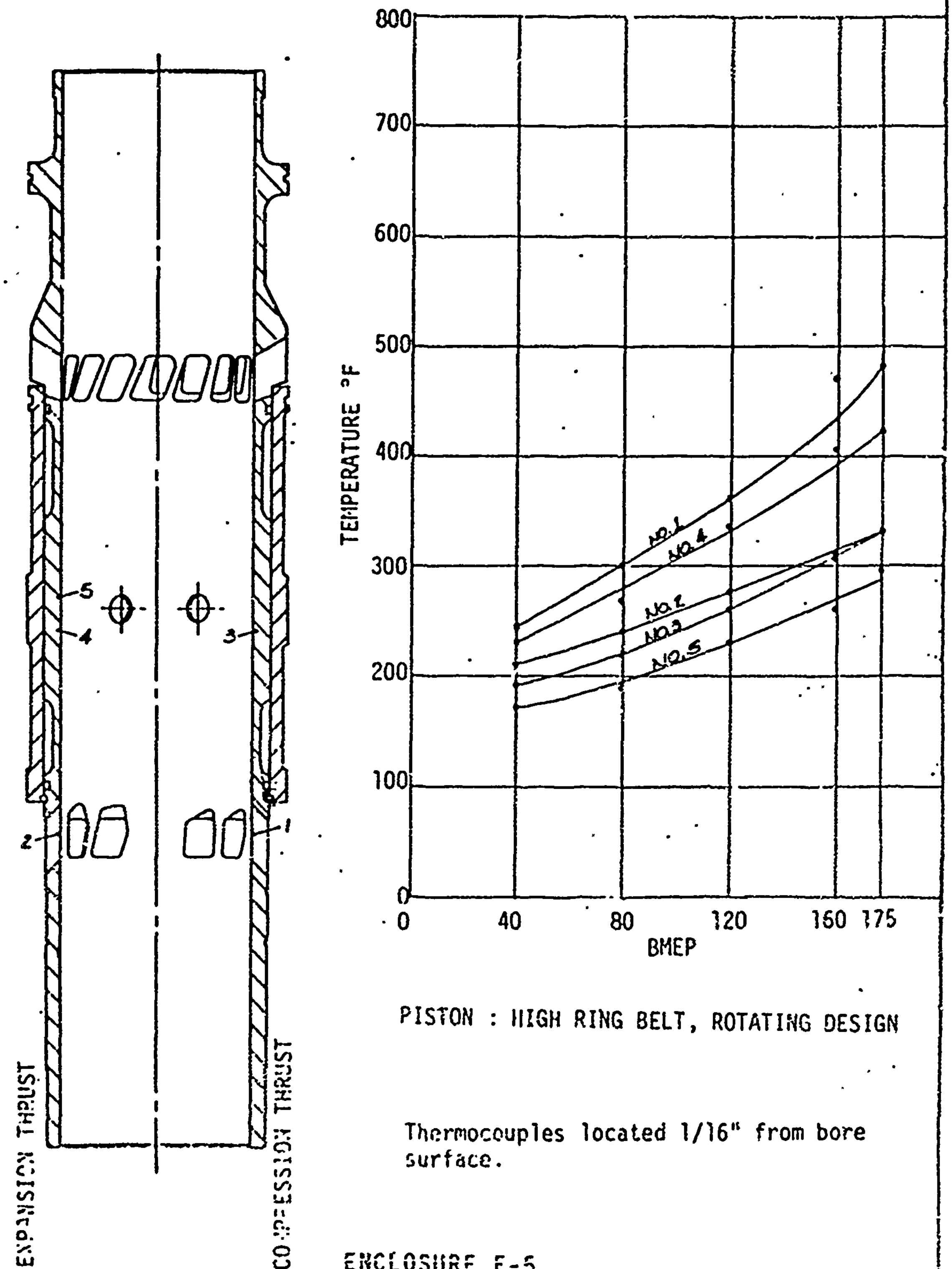
ENCLOSURE E-5.

DATE: 4/13/72
BY: [signature]

LINER TEMPERATURES
8-1/8 x 10 TURBOCHARGED
SERIE' - BLOWER SYSTEM
DIESEL @ 720 rpm

PISTON 16400334

LINER 16401912



PISTON : HIGH RING BELT, ROTATING DESIGN

Thermocouples located 1/16" from bore surface.

ENCLOSURE E-5.

DATE: 4/13/72
BY:

APPENDIX F

ENGINE OVERHAUL DIRECTIVE

COMMANDANT NOTICE 9410

Subj: Marine Diesel Engine Maintenance Program

Ref: (a) Naval Engineering Manual (CG-413)

1. Purpose. The purpose of this Notice is to amend the present Diesel Engine Maintenance Policy set forth in reference (a) in order to increase diesel engine reliability and to provide for more orderly planning, procurement action, and execution of diesel engine center section overhauls.

2. Discussion. A discussion of the past maintenance systems and reasons for moving to the "overhaul as required" policy are contained in reference (a). This "as required" policy has worked to some extent, but in numerous instances, has not provided the desired reliability from our large propulsion engines. The need for unexpected overhauls has also created scheduling and logistic planning problems which have resulted in unavailability of needed parts and "hurry-up" repairs when parts are finally received. Recent experience has also revealed that trend monitoring and lube oil analysis have not, in all cases, indicated when center section components have failed. These failures have been discovered both during inspections and, in some cases, during engine failure that resulted in aborting

operational missions.

This new policy is an attempt to improve this situation by requiring a center section overhaul at an hourly interval. This should result in a more realistic basis for planning maintenance time, a more orderly acquistion of parts, and a better quality of overhaul.

The trend monitoring and lube oil analysis programs are to be continued, since these are tools which, if properly used, provide the operating engineer information concerning the health of his engines. Additionally, there are current Research and Development efforts to develop a simple, state-of-the-art package for diesel engine diagnostics which would be independent of ambient conditions. If these efforts are successful, the center section overhaul policy will be re-evaluated. A secondary benefit of requiring a regular center section overhaul is the "hands on" experienced gained by the Machinery Technicians.

3. Scope. The policy stated herein applies to all cutters having engines listed in enclosure (1).

4. Action.

a. District Commanders and Commanding Officers of cutters having the machinery listed in enclosure (1) shall implement the provisions of this directive.

b. The Lube Oil Analysis program and the Trend Monitoring and Analysis programs required by reference (a) shall be continued. If repair or overhaul is indicated by either of these programs prior to the time between overhaul indicated in enclosure (1), appropriate maintenance action shall be taken.

c. Engine components, such as attached pumps, torsional vibration dampers, blowers, turbochargers, and like parts shall continue to be inspected and overhauled as required by existing Technical Publications or manufacturer's recommendations. The importance of accomplishing these inspections/overhauls cannot be emphasized enough. They are critical!

d. When the hourly interval stated in enclosure (1) has elapsed, the engine shall be opened up, disassembled, and all center section components shall be inspected for distress and wear. Except for compulsory renewal items, which will be listed in forthcoming Technical Publication Amendments, center section components shall be renewed only if the measured wear exceeds one-half the allowable wear. Allowable wear is defined as the new condition dimension minus the condemning limit dimension as stated in the appropriate Technical Publication.

e. A record of measurements taken and components renewed shall be maintained as part of the Machinery History.

f. The Commandant will capitalize and have SICP provide the stock support indicated in enclosure (2). This support is to provide for emergent needs and needs beyond planned available parts. Districts may purchase these items from SICP.

g. The District shall submit, along with letter report required by reference (a) (9004.6.2) a summary report showing engines overhauled, hours between overhaul, and center section parts renewed (other than those required by the T.P. Amendment).

5. <u>Cancellation</u>. This notice is cancelled upon incorporation of the policy in reference (a) or on _______________ which ever comes first.

Enclosure (1) to COMDTNOTE 9410

CENTER SECTION OVERHAUL INTERVALS

ENGINE	* INTERVAL (Hours)
FAIRBANKS	
38TD8 1/8	9000 - 10,000
38D8 1/8	9000 - 10,000
ALCO	
16-251B	10,000 - 12,000
COOPER-BESSEMER	
GN-8	6000 - 7000
GSB-8	6000 - 7000
FVBM	10,000 - 12,000

* Any engine already exceeding the stated interval shall have the center section overhauled within the next 2000 hours.

110 copies