Contract No:

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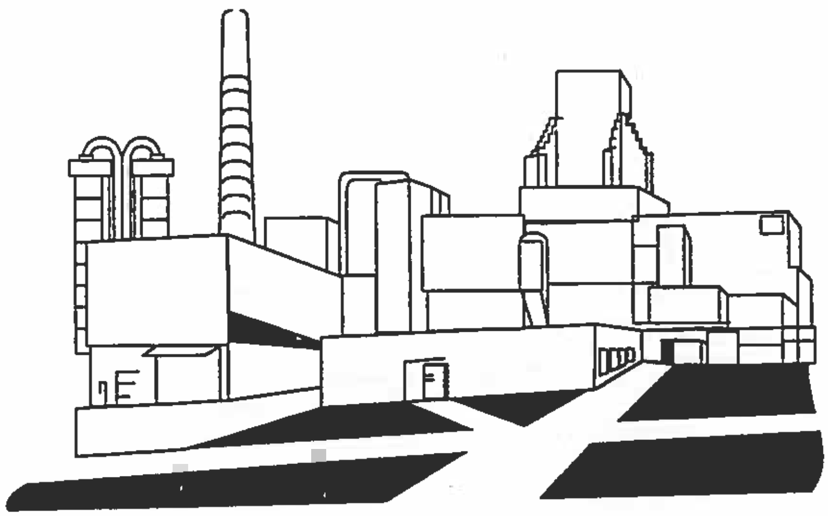
REVISION 0



TECHNICAL REPORT

GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

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GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

1.0 INTRODUCTION

In late 1990, two of K-Reactor's emergency diesels sustained multiple failures of a critical engine part which required an extensive rebuild and re-certification effort prior to restart, reference 1. During subsequent testing following repairs, both analog and digital engine operating data were acquired and recorded in references 2, 3.

In the spring of 1993, an annual preventative maintenance action (PM), and full-load test were completed on both of the Building 108K GM diesels. The following report documents and discusses trends in critical parameters. Also included are an estimate of the condition and health of the machines taking into account known predictive and preventive maintenance information.

Since the GM diesels are not required to remain operable during the current cold standby status of K-reactor, this report is being issued for historical purposes.

2.0 SUMMARY

GM-1 has operated approximately 340 hours since the last 1991 rebuild, and GM-2 has operated approximately 430 hours. During that period, the engines were operated for a period of time at no-load (monthly surveillance), at 40-50% load (hotel service during transformer room outages), and at full 1200 kw load (during 1-2/93 testing).

On engines that accumulate many operating hours, jacket water (JW) and lubricating oil (LO) pressures/temperatures are normally plotted, in a format similar to Appendix A, so that fluctuations in developing or established trends is readily noticeable.

However, for emergency and standby power sources like the GM diesels, meaningful data is obtained only a few times a year, and with so few engine hours (data points), recognizing trends is difficult. Therefore, JW and LO data were not graphed as part of this review. Instead, average temperatures and pressures at comparable load are evaluated and compared. The critical parameters reviewed and trended include Exhaust Temperature, Firing and Compression Pressures, Lubricating Oil Analysis.

A review of all data clearly indicates that the GM diesels are operating satisfactorily. There are no indications of problems at this time.

3.0 DISCUSSION

3.1 Engine Lubricating Oil

3.12 Oil Pressure and Temperature - GM-1K

Comparing data from May 25, 1991 against February 20, 1993 for the same load of 1200 kW (nominal), oil pressure to the engine (header) was 47 psig with 23-24 psig at the blower end for both dates. Engine oil temperature was regulated at 139°F/173°F (in/out) for 1991 and 132/168°F for 1993. Oil temperature rise (delta) across the engine is 34 and 36 degrees respectively, giving indication that heat transfer capabilities have not deteriorated in the past 2 years. Appendix B discusses the General Motors design value of 25-40°F temperature rise. Ambient temperatures, see Table 1, for May 25 and February 20 may account for any slight difference.

Table 1

AMBIENT CONDITIONS*

Date/time	1-15-93/1700 hrs	2-20-93/1400 hrs	5-25-91/0800 hrs	6-26-91/1000 hrs
Outside Air	51°F	45°F	70°F	80°F
Cooling Water	55°F	50°F	67°F	77°F

*source: Bush Field meteorological data. Cooling water temps taken from logsheets.

3.12 Oil Pressure and Temperature - GM-2K

Comparing data from June 26, 1991 against January 15, 1993 for the same load of 1200 kW (nominal), oil pressure to the engine was 46 psig in June, 48 psig in January with 24-25 psig at the blower end for both. Engine oil temperature was regulated at 141°F/169° for 1991 and 128/159°F for 1993. Oil temperature rise across the engine is acceptable at 28 and 31 degrees respectively.

3.2 Jacket Water

3.21 Water Pressure and Temperature - GM-1K

Diesel Jacket Water (DJW) pressure was 29 psig in June and 26 psig in January. The 3 psi reduction may be due to pump wear (increased impeller to wear ring or housing clearance) causing a reduced throughput; it could also be due to a mid-year gage calibration. Historically, from 1988-1990, pump pressure measured at the gauge board has been 27 psig. Measuring jacket water flow through the pump with an ultrasonic flowmeter and comparing to the ~300 gpm figure of 1991 would help evaluate pump condition.

The Amot type regulating valve kept engine jacket water temperature at 155/171°F (in/out) for 1993 and 154/174°F for 1991 at full load for the dates and times given in Table 1. All values are within established parameters. No significant variation is noted.

3.22 Water Pressure and Temperature - GM-2K

Diesel Jacket Water (DJW) pressure was 28 psig this year and was the same 27/28 psig in June 1991. Jacket water was regulated on this engine at 150 / 170 °F this year and 156/168 °F in 1991. There seems to be a bit more temperature rise across the engine this year, i.e., 20° vs. 10-12°. A lower rise across the engine is desirable to inhibit thermal stresses on power packs and reduce challenges to sealing components, see Appendix B. If further testing confirms the "delta T" trending further upward, inspection of cooling water piping and passages in the engine may be warranted. For now, all values are within established parameters.

3.3 Firing and Compression Pressures

3.31 Firing Pressure GM-1K

The maximum pressure developed in each of the engine's 16 cylinders during combustion is measured by connecting a 0-2000 psi capacity Keine pressure gauge to each cylinder head test valve. Firing pressures are balanced by metering fuel injector delivery via micro-adjusting rods.

Figures 5 and 6 show firing pressures and difference-from-average pressures at full load. This engine was balanced more evenly during 1991 testing when the firing pressure average was 958 psig - vice 996 for 1993. From our experience with this engine, we might conclude that the engine was operating above 1200kw (~1720 bhp) load at the time these firing pressures were taken. Latest average firing pressure is in line with that shown on GM's original testing curves, figure 12, for 1750 bhp.

Difference from average firing pressure on all cylinders was 50 psid, and in 1993 it is 80 psig. Cylinders 1, 2, 3, 5, 7 were slightly high in firing pressure but did not exceed the 1050 psig limit. Note that cylinder #5 had high firing pressure and low exhaust temperature in 1993, whereas in 1991 the same cylinder had low pressure and high exhaust temperature. For both years, cylinder #5's firing pressure was higher than average so it appears to be taking its share of load. Cooler exhaust temperature this year may be due to pyrometer inconsistency, slightly different injector spray pattern, or better efficiency caused by tighter ring-to-liner seal and resulting higher compression pressure.

3.31 Firing Pressure GM-2K

Figure 1 shows GM-2K firing pressures at various loads, and figure 2 shows the balanced condition at 1200 kW for 1991 vs. 1993. This engine is capable of being balanced within 50 psig on all cylinders at 1000 kW, within 50 psig sometimes at 1200 kW, but at all times within 80 psig. See Appendix C for a discussion of cylinder pressure and load balancing. Basically, the closer all cylinders are balanced, the more even is the stress distribution in power pack components, on crankshaft journals and bearings, as well as the engine frame itself. Figure 3 shows differences from average firing pressures which should be held below 80 psig as earlier mentioned. Cylinders #5 and #14 on this engine for some reason tend to have the highest and lowest pressures (respectively).

This is a well balanced engine and shows a steady trend from 1991-1993. Firing pressures are well within the 1050-1100 psig allowed by General Motors Cleveland Maintenance Manuals.

3.32 Compression Pressures

Compression pressures normally indicate tightness of .the piston rings, liner, exhaust valves, and head gasket seal. The reading is obtained through the same test valve as firing pressure without the injector pumping fuel into the cylinder. Trends in this parameter tell much about the conditions of the mechanical components in the cylinder. When an engine is first rebuilt, compression pressure will be slightly low, but they will still be high enough to allow compression ignition of the fuel when injected. Once the ring-to-cylinder liner wear-in process is complete, compression pressures trend upward to the desirable or expected value. Figure 8 shows the most recent values on a steady upward trend. The trend for both these engines will continue to increase with operating hours to the upper normal range (600-650 psig) unless the engine runs no-load or lightly loaded. Light loading will cause the piston rings to glaze or scuff, possibly stick with deposits allowing compression gases to escape into the crankcase (blowby). The most common results of light loading or no load operation are low compression, blowby and poor overall engine efficiency and performance. However, a crankcase explosion may result if combustion flames ignite flammable vapors in the crankcase.

Excessive clearance between the piston (wrist) pin and wrist pin bushing will also cause a sharp decrease in compression. This wear phenomena was observed in 1991, and is further discussed below in the conclusions section.

3.4 Exhaust Temperatures

3.41 At 1200 kW - GM-1K

Average exhaust temperatures for the two runs are markedly different. Even though firing pressures were higher in 1993 (996 psig), the average exhaust temperature was ~90°F cooler. Raw (cooling) water and air temperatures in February were lower and may have caused the lower exhaust temps. However, GM-2K water and air were 22 and 29°F lower respectively and exhaust temperatures were not significantly different (732-756°F). Thermocouples may need calibration. In Appendix C, an ex-Cleveland design engineer acting as consultant, discusses pyrometer accuracy and what can cause exhaust temperatures to vary.

3.42 <u>At 1200 kW - GM-2K</u>

Figure 4 illustrates the average exhaust temperatures for GM-2K diesel at full load with lowest and highest exhaust temperature. According to the GM Cleveland design engineer, cylinder #5 (or 6) vice cylinder #14 juxtaposition of high/low firing pressure and low/high exhaust temperature may be due to pressure waves in the exhaust manifold that affect scavenging, see Appendix D.

Exhaust temperature trends throughout testing were satisfactory. At steady loads, temperatures will tend to rise and fall with the day's ambient temperature. A maximum 200°F difference between any two cylinders and a 900°F maximum individual cylinder temperature is allowed for this engine. Average full load temperatures will range from 750-850 °F.

This is close to the values given in GM Cleveland's plot for a 6 cylinder engine, figure 13. In order to compare the GM data to our 16-278A, we use the parameter brake mean effective pressures (BMEP). BMEP is defined as the theoretical constant pressure which can be imagined exerted during each power stroke of the engine to produce power equal to the brake power, reference 4, p-45. Using appropriate equations, it can be shown that a 6-278A engine running at 750 rpm developing 650-670 brake horsepower (bhp) is equivalent to the 16-278A running at 720 rpm developing ~1720 bhp. From the graph, an average exhaust temperature for this load can be estimated at 700-750°F with 1-3 inches Hg backpressure.

3.5 Oil Consumption and Analysis

3.51 <u>GM-1K</u>

Normal oil consumption for this engine, from Appendix B, is one gallon in 1000-3500 bhp-hr.

Using data from February 20 logsheets, the GM-1K diesel used 6 gallons of lube oil in 8 hours at full load for an hourly rate of 0.75 gph, or 4.41EE -4 gal/bhp-hr. Expressed as the inverse, one gallon of oil used for every 2270 bhp-hr (bhp = 1200kw/0.746 x generator efficiency ~.94 = 1700 hp).

Figure 9 shows the wear metal concentration in lube oil for GM-2K. The trend for GM-1K is comparable except that iron levels are lower at 12 ppm by weight. Figure 10 shows an acceptable trend in copper particulate in the lube oil for both engines. Figure 11 summarizes the critical chemical and physical parameters trended and allowable limits for the GM diesel lubricating oil.

3.52 GM-2K

From January 15 logsheet data, it appears this engine used about 9-12 gallons in 17 hours for an average 0.7 gph oil consumption. This is equivalent to using one gallon of lube oil in 2400 bhp-hr again comparing favorable with the original GM data of one gallon/1000-3500 bhp-hr. As the compression rings, oil control rings, and cylinder liners wear, oil consumption will trend toward the lower 1000 bhp-hr/gal limit.

4.0 CONCLUSIONS

All engine trends are satisfactory. One concern is the GM-1K jacket water pump, another is the jacket temperature rise across GM-2K. Although acceptable, they bear watching in the future. Firing pressures on both engines should be tuned to the lowest pressure variation possible so that exhaust temperatures and engine stresses are kept in line. Fuel delivery equipment and governing on this engine is near its limit when running 1250 kW, so the fuel racks should be adjusted carefully and any item with discernible wear replaced immediately.

In 1990-91, repeated wrist pin bushing failures were attributed to poor workmanship and non-GM specified materials. In order to monitor continued health of the highly loaded OEM bushing replacements, three critical parameters were monitored during testing: copper concentration in the lube oil, lubricating oil pressure and temperature (main header), firing and compression pressures. Review of the critical parameters concludes that the bushings are performing as designed. No abnormal wear is indicated.

5.0 REFERENCES

1. WSRC-TR-92-42-011, Engineering Evaluation of the General Motors (GM) Diesel Rating and Capabilities (U), RE Gross, April 1992.

2. Temporary Operating Procedure, TOP-K-91-115, Test - GM-#1 Diesel Generator with Load Banks, May 14, 1991, Tom Hipp et al.

3. Temporary Operating Procedure, TOP-K-91-141, Test - GM-#2 Diesel Generator with Load Banks, May 29, 1991, Tom Hipp et al.

4. Internal Combustion Engines and Air Pollution, Edward F. Obert, 1973 Harper and Row.

6. ATTACHMENTS

LIST OF FIGURES

APPENDICES

Appendix A - Diesel Engine Trend Analysis graphs (DETA), courtesy Naval Ships Systems Engineering Station Code 031-B.

Appendix B - Model 16-278A Engine Thermal Problems, Engineered Applications Corporation, PJ Louzecky, 6/22/93.

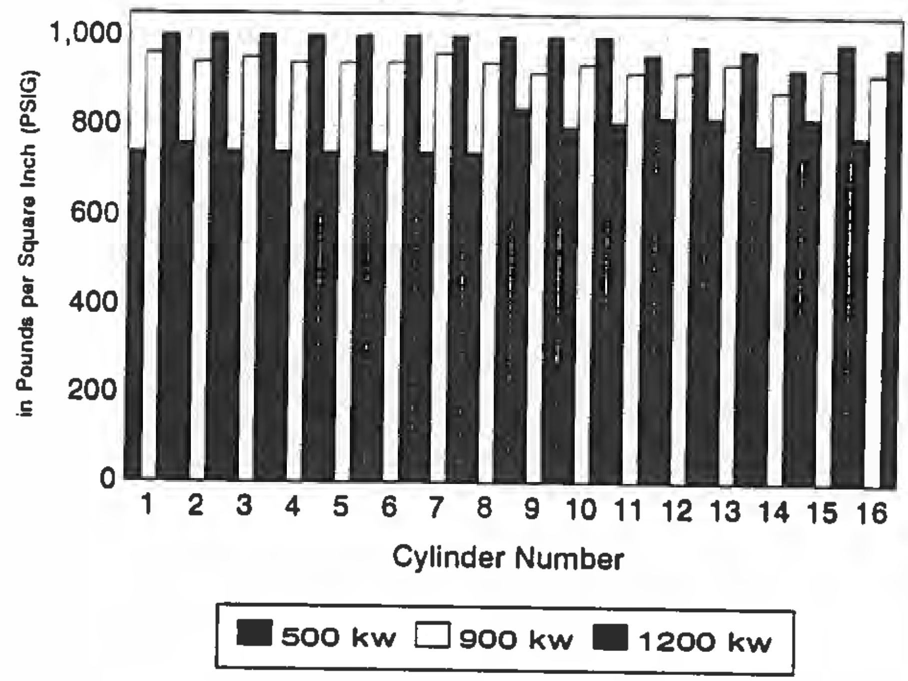
Appendix C - Model 16-278A Cylinder Balancing and Cylinder Exhaust Gas Temperatures, Engineered Applications Corporation, PJ Louzecky, 7/3/91.

Appendix D - Pressure Waves in the Exhaust Manifold of the Model 16-278a Engine and Other Engineering Information", Engineered Applications Corporation, PJ Louzecky, August 5, 1991.

GM-2K Diesel Firing Pressure vs Cylinder

Measured During 1993 Annual Load Testing

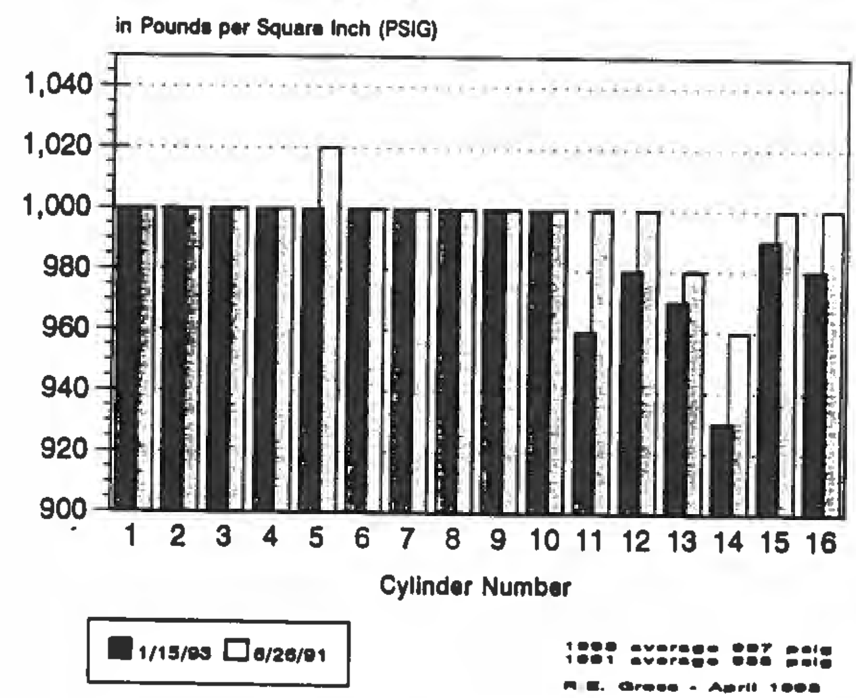
FIGURE 1



GM-2K Diesel Firing Pressure vs Cylinder

at 1200KW - January 15, 1993 vs June 26, 1991

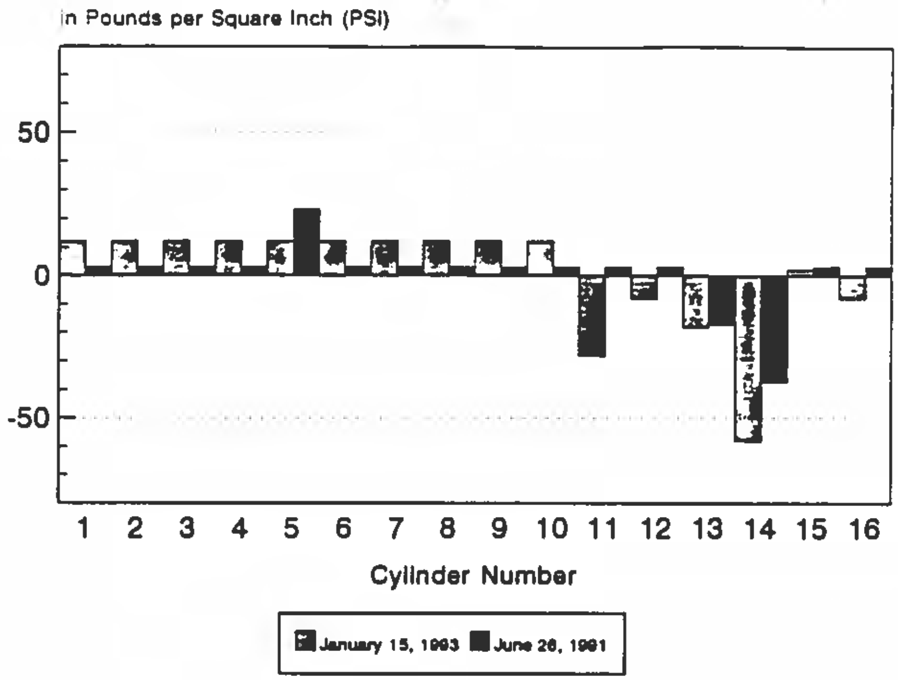
FIGURE 2



GM-2K Diesel Difference from Average Firing Pressure

at 1200KW - January 15, 1993 to June 26, 1991

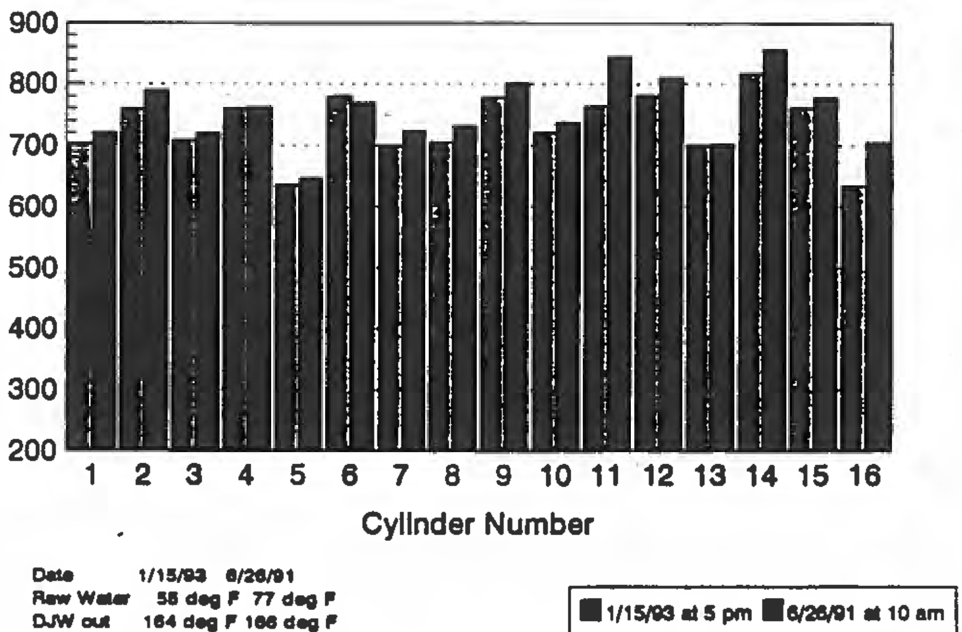
FIGURE 3



GM-2K Diesel Exhaust Temperature vs Cylinder

January 1993 Measurements Compared to June 1991 at 1200kw

FIGURE 4



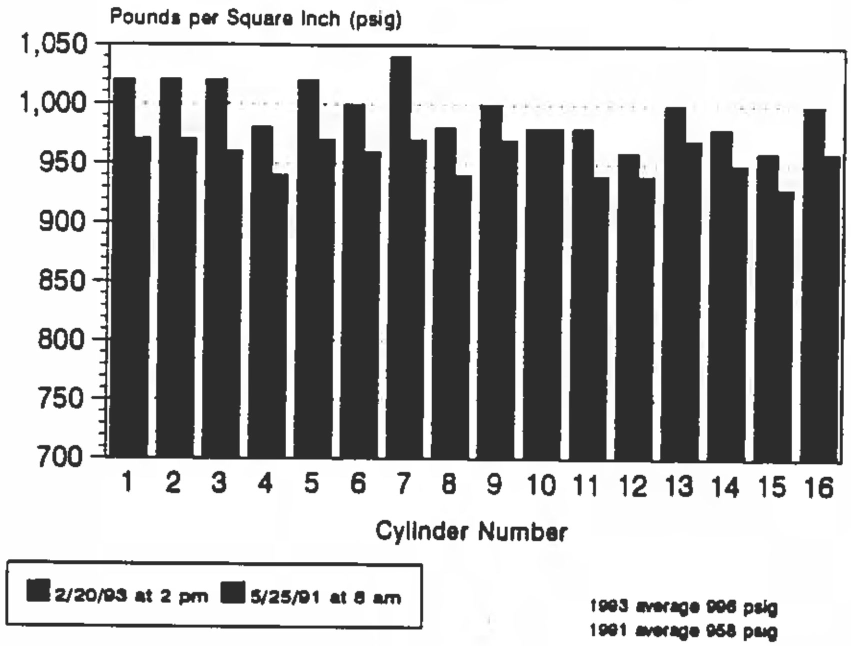
Date	1/15/93	6/26/91
Raw Water	58 deg F	77 deg F
DJW out	164 deg F	166 deg F
DLO out	158 deg F	169 deg F
Exh Ave	732 deg F	758 deg F

R.E. Gross April 26, 1993

GM-1K Diesel Firing Pressure vs Cylinder

at 1200KW February 20, 1993 vs May 25, 1991

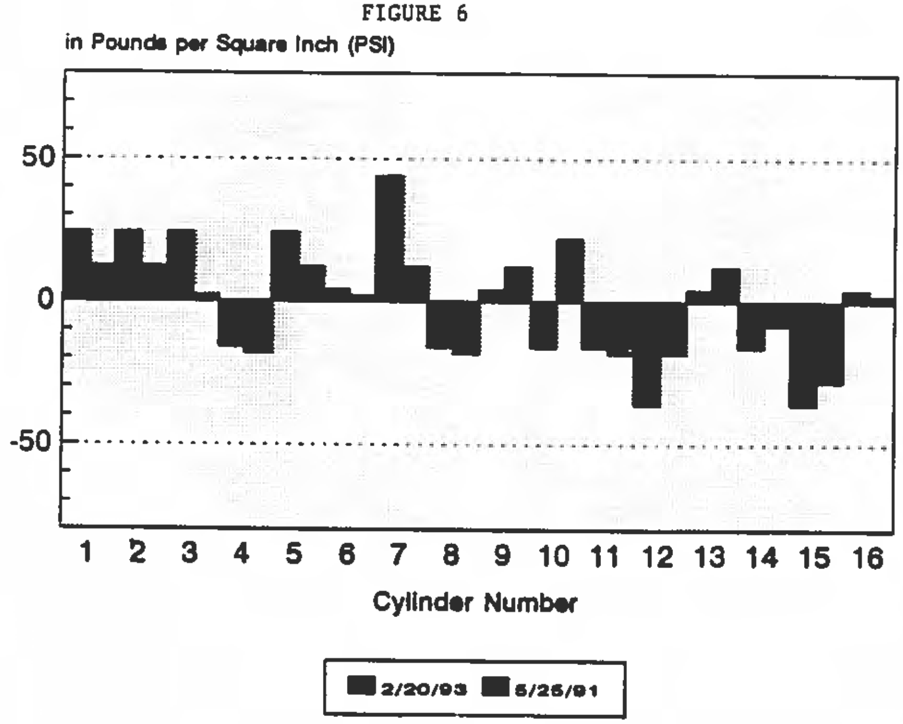
FIGURE 5



GM-1K Diesel Difference from Average Firing Pressure

at 1200 KW - February 20, 1993 to May 25. 1991

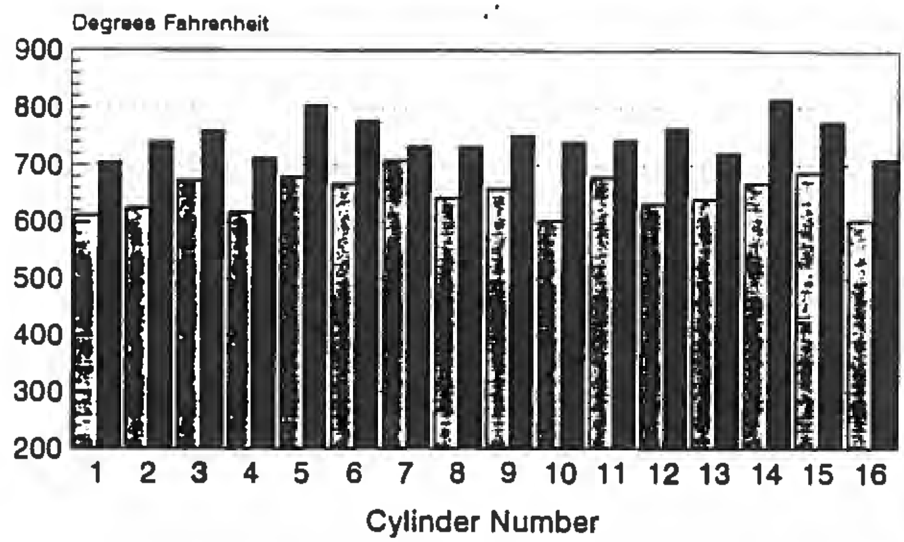
FIGURE 6



GM-1K Diesel Exhaust Temperature vs Cylinder

February 20, 1993 compared to May 25, 1991 at 1200kw

FIGURE 7



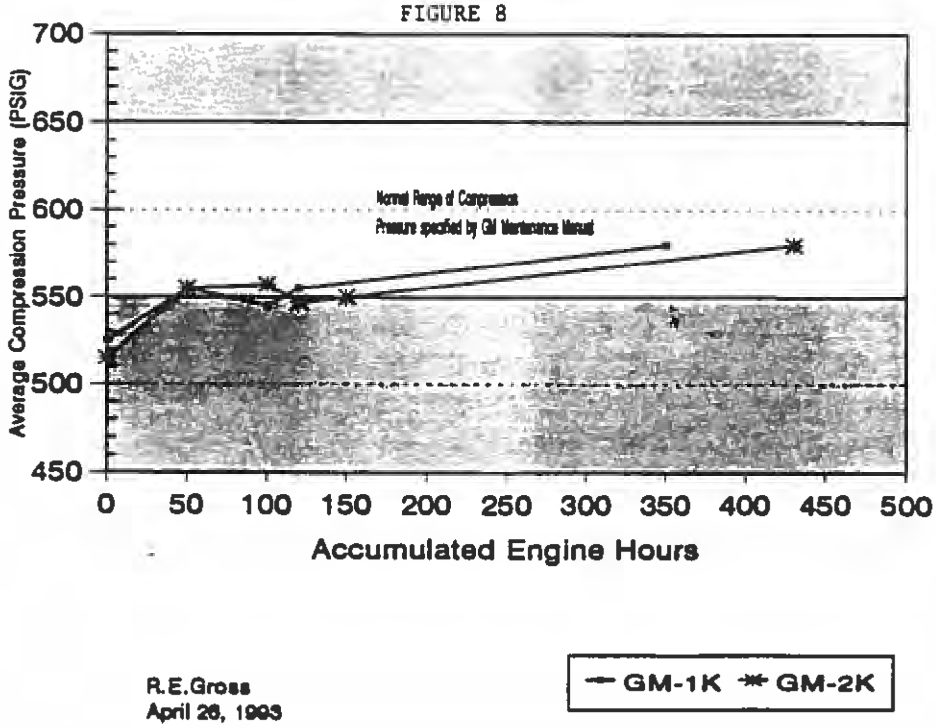
Date	2/20/93	5/25/91
Raw Water	50 deg F	67 deg F
DJW out	170 deg F	175 deg F
DLO out	170 deg F	172 deg F
Exh Ave	640 deg F	750 deg F

R.E. Gross April 26, 1993

GM Diesel Compression Pressure Trends

May 1991 to April 1993

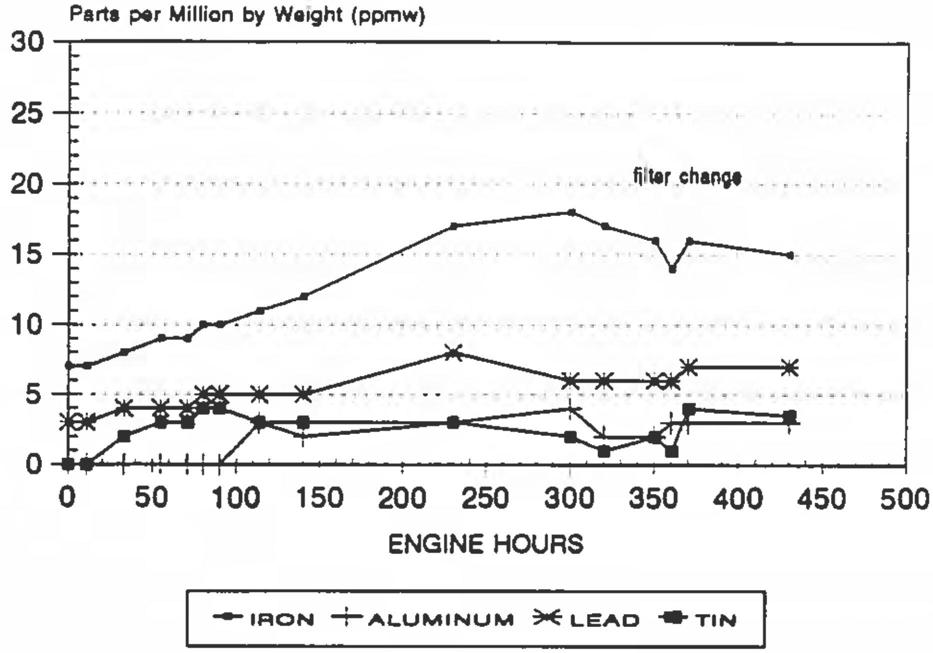
FIGURE 8



R.E.Gross
April 26, 1993

WEAR METAL CONCENTRATIONS IN LUBRICATING OIL vs Engine Hours - GM-2K Diesel

FIGURE 9



COPPER CONCENTRATION IN LUBRICATING OIL vs Engine Hours - GM DIESELS

FIGURE 10

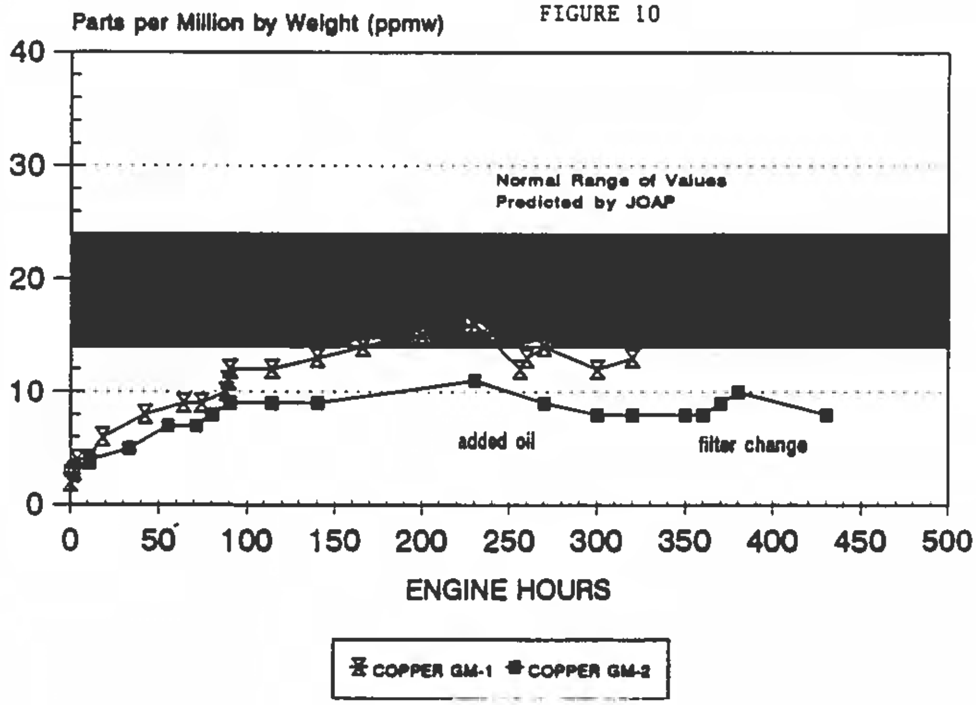


FIGURE 11

GM LUBE OIL ANALYSIS AT 400-500 HOURS

ATTRIBUTE	200-HOUR ANALYSIS		500-HOUR ANALYSIS		NEW OIL	TYPICAL LIMIT	SOURCE
	GM-1	GM-2	GM-1	GM-2			
WEAR METAL (ppm)							
Iron (Fe)	16	18	12	15	4	0-24	Joint Oil Analysis Program manual Navy/Army/Air Force (JOAP) for GM 12-278 engine
Chromium (Cr)	5	3	3	3	1	5-9	JOAP for 16-248 engine
Aluminum (Al)	5	4	3	3	0	0-5	JOAP; 12-278
Copper (Cu)	16	9	13	8	2	15-30	DuPont Standard (1987) PL24 Table 6B
Tin (Sn)	4	3	2	3	0	0-5	PL-24
Zinc (Zn)	1260	1310	1184	1550	1400	800-1400	PL-24
WATER % VOLUME	<0.05	<0.05	<0.5	<0.05	0	(0.5% max)	PL-24
FUEL DILUTION	<0.5	<0.5	<0.5	<0.5	0	(5% max)	Naval Ships Technical Manual (NSTM) Chapter 233, "Diesel Engines"
TOTAL SOLIDS % Volume	0.3	0.7	0.2	0.1	0-0.2	(3% max)	PJ Louzecky (Cleveland Diesel) and PL-24
TOTAL BASE NUMBER (TBN)	8.33	7.91	8.46	7.08	7.89	(4.0 min.) (<2.0)	PJ Louzecky (Cleveland Diesel) ASTM D-2896
VISCOSITY (cst @ 100°C)	15.8	15.3	15.0	14.8	15.6	12.9-16.8 (40% inc. - 15% dec.)	General Motors Detroit Diesel Allison Technical Guide (DDA)

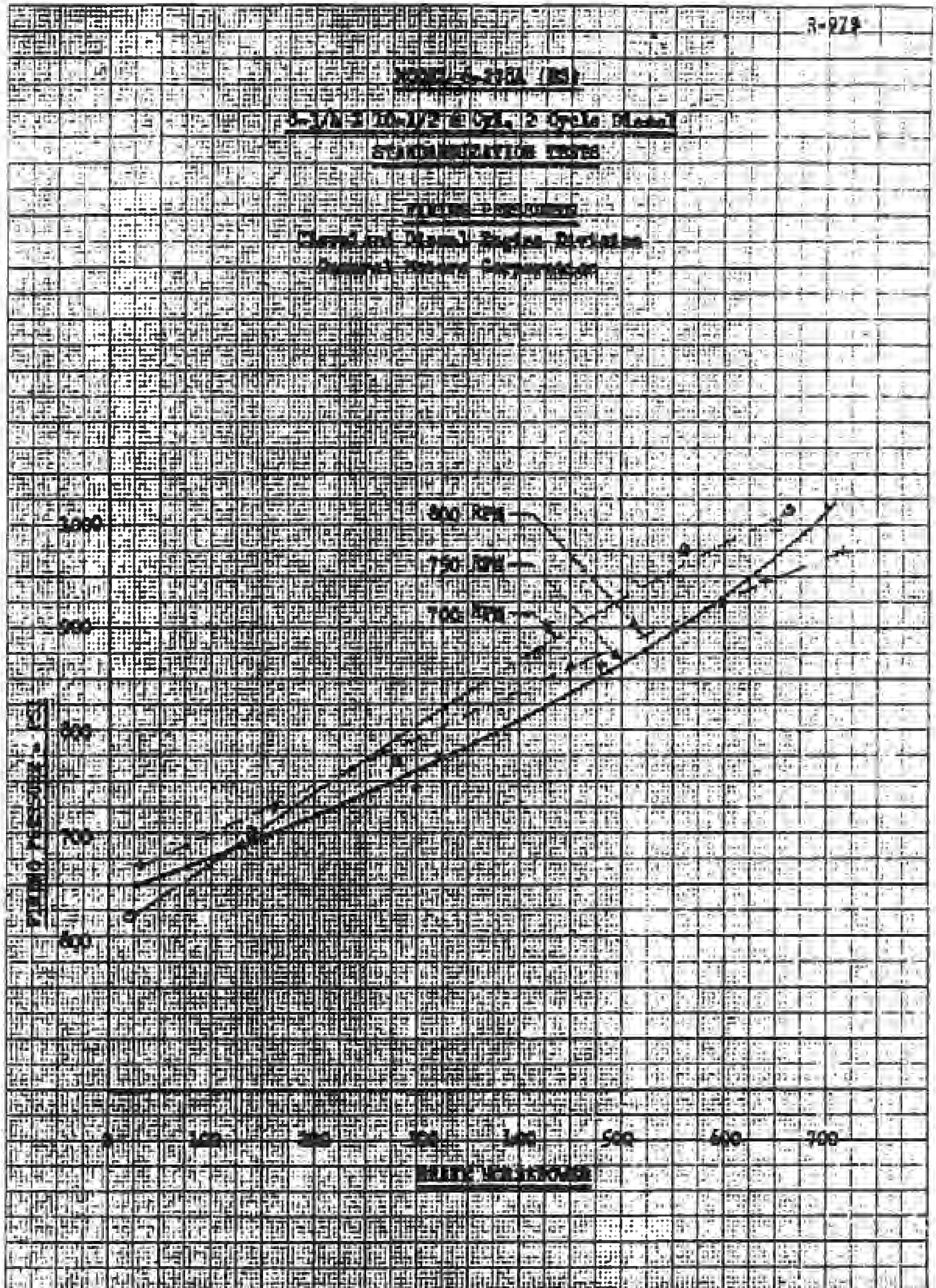


FIGURE 12

RPS
6-30-54

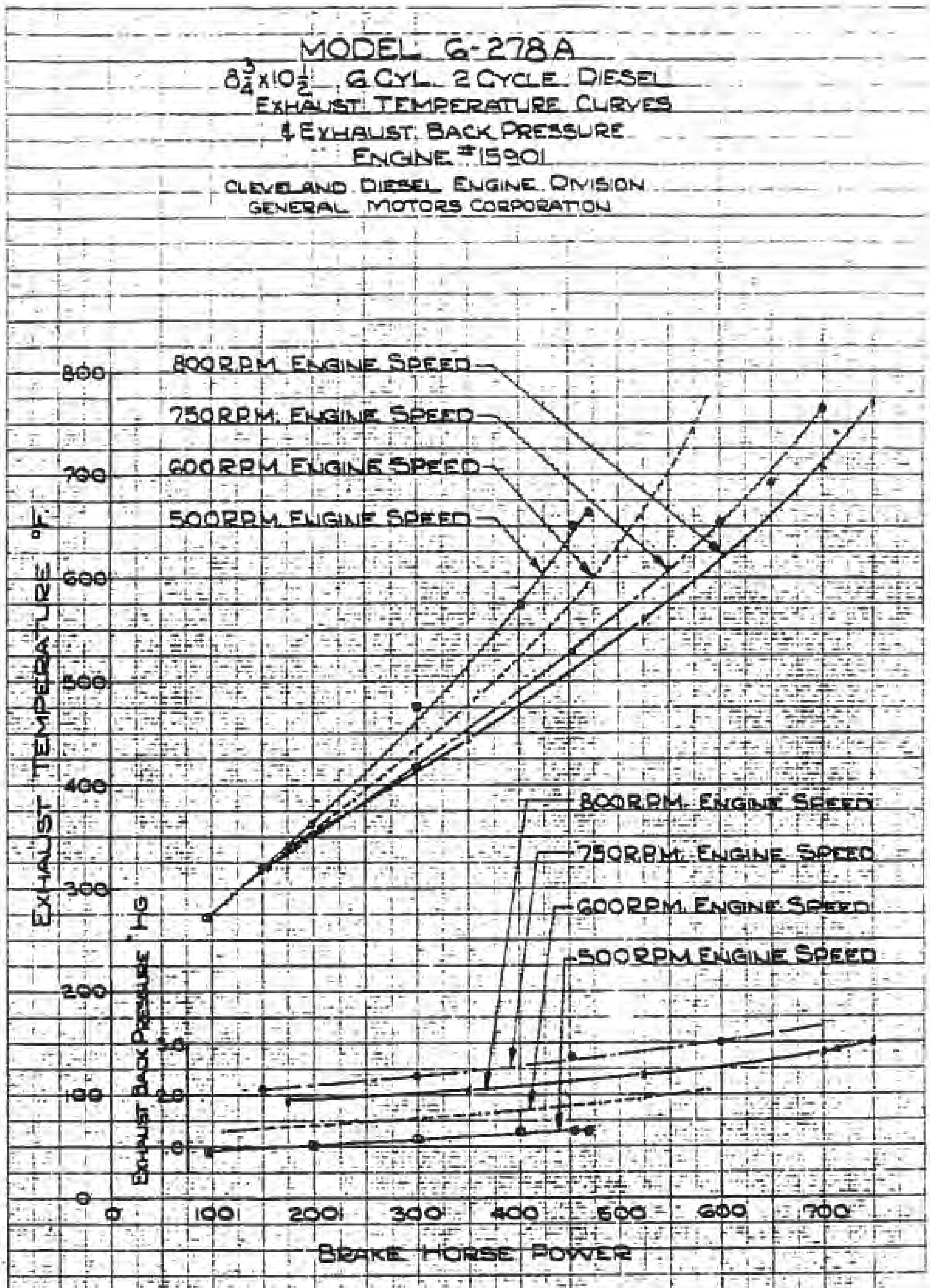

MODEL 6-278A
8 3/4 x 10 1/2 6 CYL. 2 CYCLE DIESEL
EXHAUST TEMPERATURE CURVES
& EXHAUST BACK PRESSURE
ENGINE #15901
CLEVELAND DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION
800 R.P.M. ENGINE SPEED
750 R.P.M. ENGINE SPEED
600 R.P.M. ENGINE SPEED
500 R.P.M. ENGINE SPEED
EXHAUST TEMPERATURE °F
EXHAUST BACK PRESSURE " HG
BRAKE HORSE POWER
0
100
200
300
400
500
600
700
800

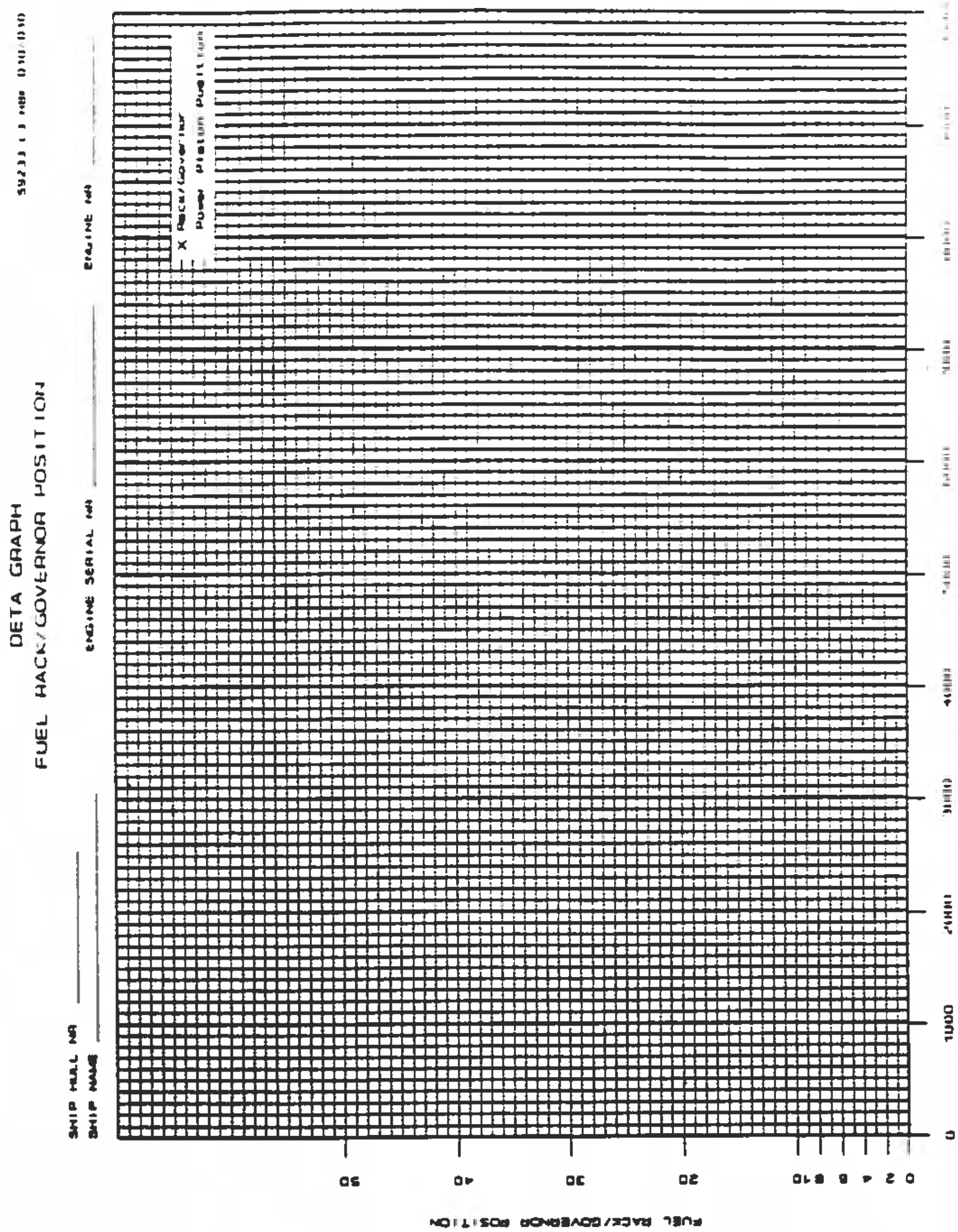
FIGURE 13

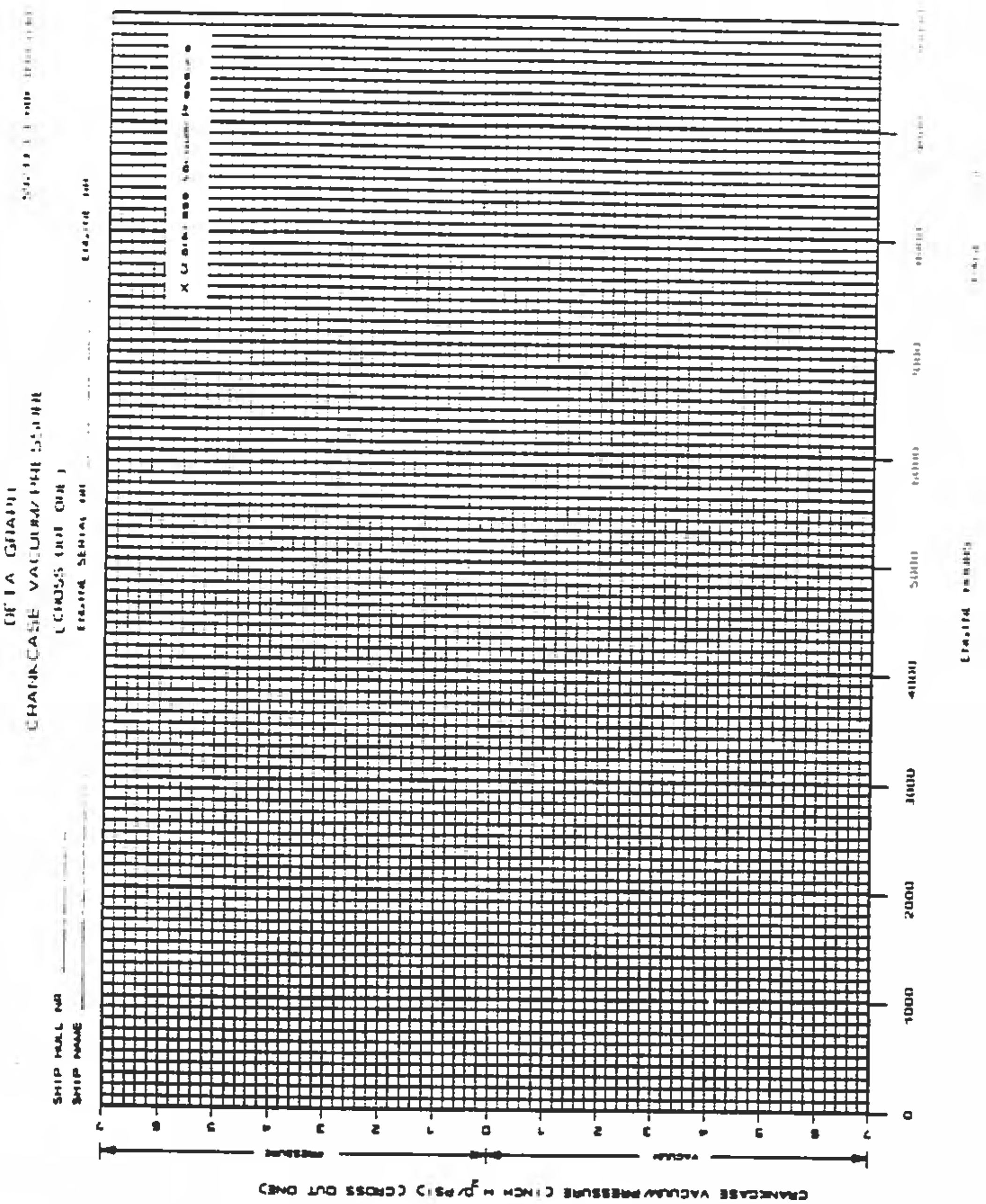
WSRC-TR-93-42-056

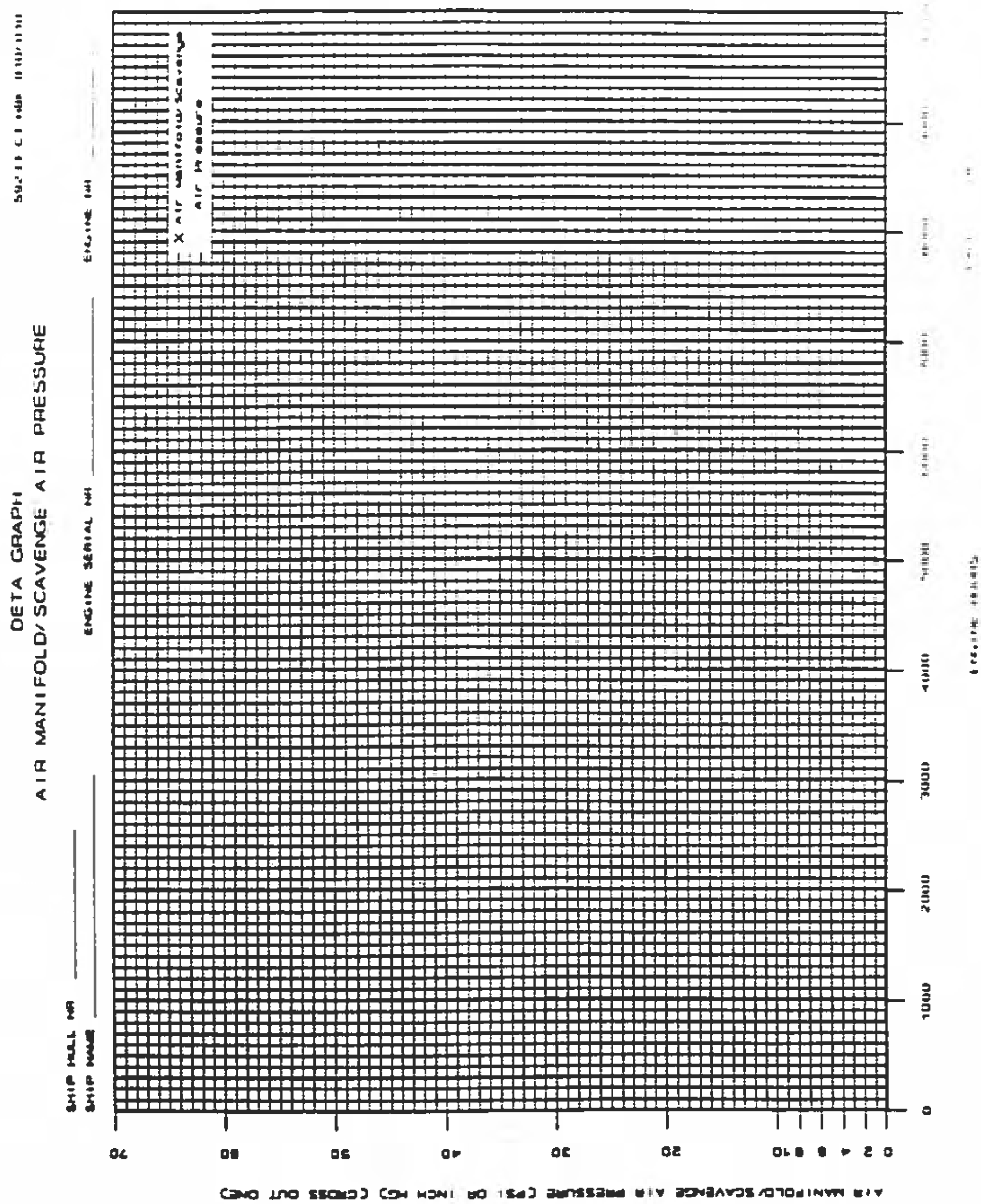
GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

APPENDIX A

DETA GRAPH FORMS


DETA GRAPH
FUEL RACK/GOVERNOR POSITION
SHIP HULL NR
SHIP NAME
ENGINE SERIAL NR
ENGINE NR
X Rack/Governor
Power Piston Position
FUEL RACK/GOVERNOR POSITION
0 2 4 6 8 10
20
30
40
50
0
1000


SHIP HULL NR
SHIP NAME
0
1000
2000
3000
4000
5000


DETA GRAPH
AIR MANIFOLD/ SCAVENGE AIR PRESSURE
SHIP HULL NR
SHIP NAME
ENGINE SERIAL NR
ENGINE NR
X Air Manifold/Scavenge Air Pressure
0
1000
2000
3000
4000
ENGINE HOURS
0 2 4 6 8 10
20
30
40
50
60
70
AIR MANIFOLD/SCAVENGE AIR PRESSURE (PSI OR INCH HG) (CROSS OUT ONE)

WSRC-TR-93-42-056

GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

APPENDIX B

MODEL 16-278A ENGINE THERMAL PROBLEMS

ENGINEERED APPLICATIONS CORPORATION
ENGINEERING AND DESIGN CONSULTANTS
476 Sarsfield Drive, Rochester Hills, Michigan 48307
Phone: 313 - 853-3992

ENGINE ... TESTS AND ... PROBLEM DETERMINATION

ENGINE DESIGN
SPECIAL TESTS AND
ENGINE SERVICES

June 22, 1993

Rec'd 7-1-93
R92

Mr. Robert E. Gross P.E.
Reactor Engineering
Westinghouse Savannah River Company
P.O. Box 616
Aiken, SC 29802

Subject: Model 16-278A Engine Thermal Problems

Dear Bob:

Enclosed is my report on the heat rejection and design temperature rise of the engine water and lubricating oil. Also I discuss the engine oil consumption using different cylinder liners.

Because both the 278A and 567 engines are 2 cycle units having their air intake ports at the bottom of the cylinder liner there will a tendency of the oil sweeping into the air port area, which later could be blown into the engine combustion air in the cylinder combustion space. This is one drawback of this 2 cycle engine design.

I hope this work covers your request. If not give me a call.

2.

Enclosed, also is my statement for my work on your problems, (copy).

It has been a real joy working with you on your problems.

Best wishes,

Paul

MODEL 16-278A HEAT REJECTION, TEMPERATURE RISE, AND OIL CONSUMPTION

ENGINE COOLING WATER

The heat rejection to a two cycle engine cooling system is as shown by a typical engine heat rejection curve, Figure 1. The curve shows that for a large 2 cycle non turbocharged engine the heat rejection, at full load, is about 30 BTU per BHP per minute.

Then knowing the engine cooling water flow, the water temperature rise through the engine can be calculated and measured.

Usually, this water temperture rise is about 15 F. The reason for this low temperature rise is to minimize the thermal stresses and stress distortion in the engine.

The thermal expansion between engine parts for this temperature rise is about .0001 inches per inch and the stress is about 700 -1000 PSI per inch of length, but for large engine parts this thermal stress is much smaller because the temperature difference between parts is much less than 15 F.

The oil and water top operating temperature range is generally between 160-180 F. By keeping both temperatures about the same it helps minimize the engine thermal stresses.

Because of the flow characteristics of the typical centrifugal engine cooling water pump, see Figure-2, the engine cooling water temperature rise increases slightly as the cooling system fouls.

The engine, raw water cooling system, is generally designed with a 10°F temperature rise because it has a greater tendency to foul the cooler and a greater temperature rise can be tolerated before cleaning is required.

ENGINE OIL SYSTEM

The temperature rise of the engine oil is generally higher than for the engine cooling water even though the heat rejection is less,see Figure-3. Generally the engine oil temperature rise on this model engine is about 25-40 F.

In both cases the top oil and water temperatures are kept nearly the same, so as to minimize thermal distortion.

In designing the engine oil system the temperature rise of the oil in the engine crankshaft bearings is a determining factor. Calculating the temperature rise of the oil in the bearings the oil flow is adjusted to give a rise of about 30 F. In this way the bearing thermal distortion is minimized. Then knowing the engine oil requirements, the oil pump is sized. With this information the oil temperature rise through the engine is determined.

Generally the top oil and water temperatures are between 160-180 F, and it is desirable to keep both the water and the oil temperatures about the same. The lower the oil temperature the greater is the oil film thickness.

ENGINE LUBRICATING OIL CONSUMPTION

The engine lubricating oil consumption for the typical 2-cycle engine varies with the engine speed, the type of cylinder liners used, their roughness, the piston ring combination, the oil used, the overall condition of the engine and the way it was run-in and operated.

The typical 16-278A and 16-567 engines oil consumption with porous chrome plated cylinder liners is about 1000-3500 BHP HRS. per Gallon of oil.

If the engine uses channel chrome plated liners the oil consumption is about 700-1500 BHP HRS per Gallon. Engines with these liners tend to push the oil up through the channels into the combustion chamber where it is burned.

If the engine uses cast iron cylinder liners with a pearlitic microstructure, the oil consumption is about 2000-5500 BHP HRS. per Gallon.

The emergency diesel engines do so little running that often the piston rings do not become properly seated and the result is high oil consumption, oil contamination and sludge. Therefore any extra running during run-in is helpful and improves seating of the rings. Also the oil must be filtered properly or changed frequently to prevent contamination and to keep the rings free.

The Model 278A and 567 Engines are two cycle units with the air intake ports at the bottom of the cylinder liner. As the piston rings move past the ports some oil is swept into the port intake area. Some of this oil contaminates the air box and some is blown into the combustion space along with the air.

If the air intake port area is not properly relieved (mae-wested) the oil pumping problem can be aggravated.

This possible combustion air contamination can account for a higher oil consumption in some cases. For this reason the piston rings must be kept in good operating condition.

Paul J. Louzecky

June 22, 1993

Per phonecon PJL/REG on 7/9/93, The following 567B curves can be used for 278A.

ENGINE HEAT BALANCE

In making the diesel engine heat rejection study it is often desirable to make a heat balance study. The estimated heat balance for this type of engine is as follows:

Heat to useful work	34%
Heat rejection to engine cooling water	27%
Heat rejection to engine oil	6%
Heat rejection to engine exhaust	28%
Radiation	5%
Total	100%

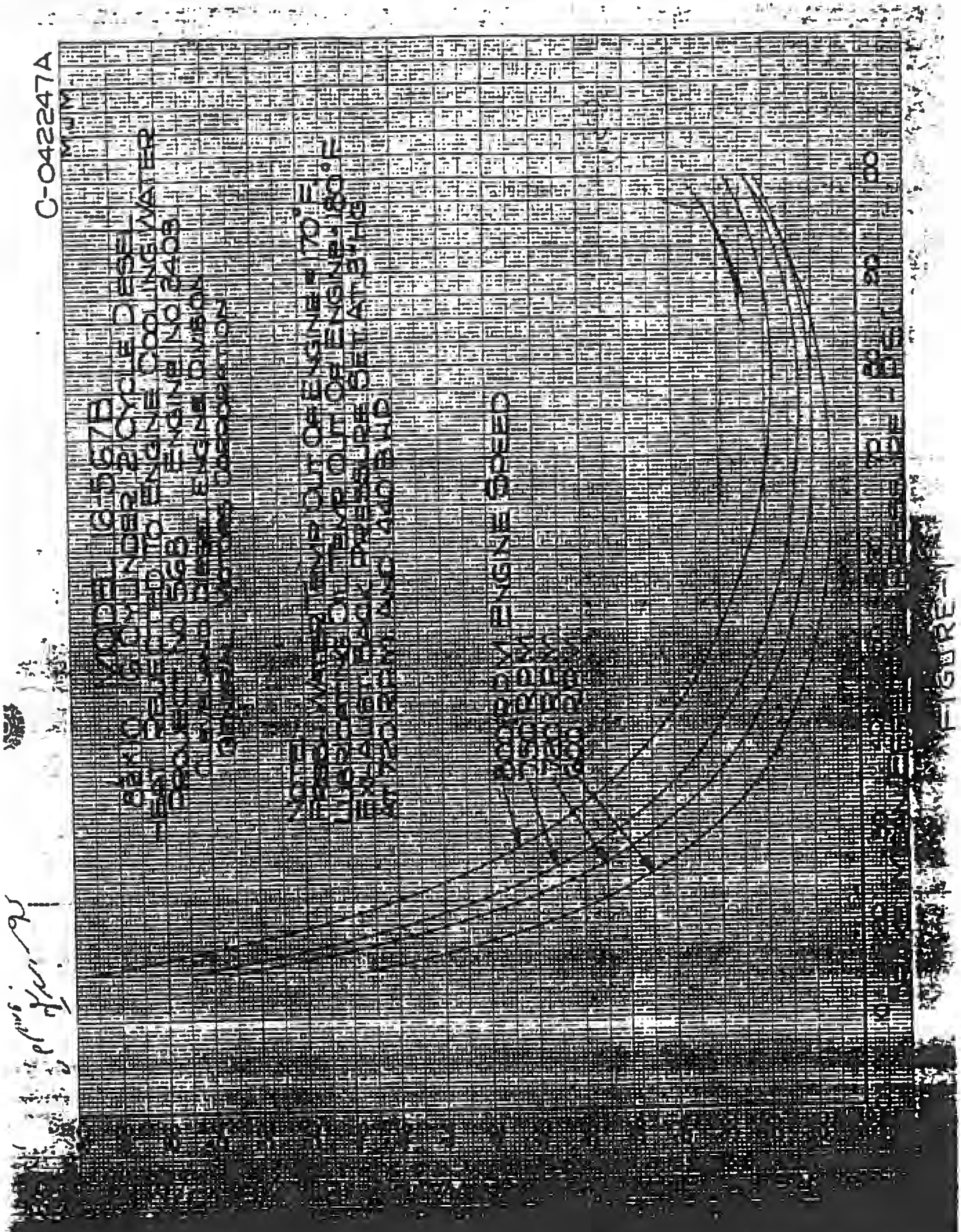
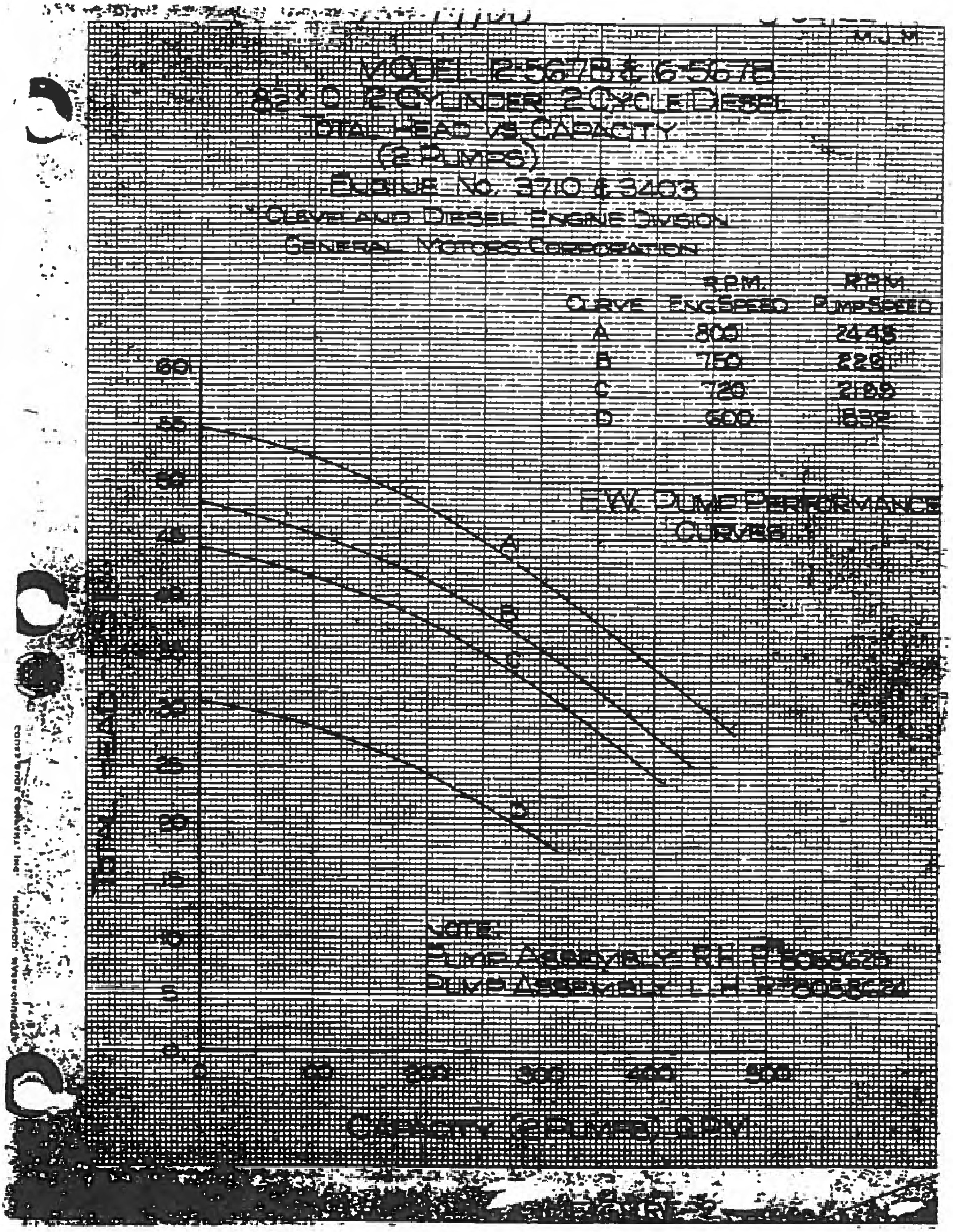

C-042247A
MODEL 16-567B
16 CYLINDER 2 CYCLE DIESEL
HEAT REJECTED TO ENGINE COOLING WATER
ENGINE NO 2403
DIESEL ENGINE DIVISION
NOTE:
FRESH WATER TEMP OUT OF ENGINE = 170°F
EXHAUST BACK PRESSURE SET AT 3" HG
800 RPM ENGINE SPEED
70
80
90
100

FIGURE-1


MODEL 12-567B & 6-567B
8½ X 10 12 CYLINDER 2 CYCLE DIESEL
TOTAL HEAD VS CAPACITY
(2 PUMPS)
CLEVELAND DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION
CURVE
R.P.M. ENG SPEED
R.P.M. PUMP SPEED
A
B
C
D
F.W. PUMP PERFORMANCE CURVES
NOTE:
PUMP ASSEMBLY R.H.
PUMP ASSEMBLY L.H.
TOTAL HEAD
CAPACITY (2 PUMPS) G.P.M.

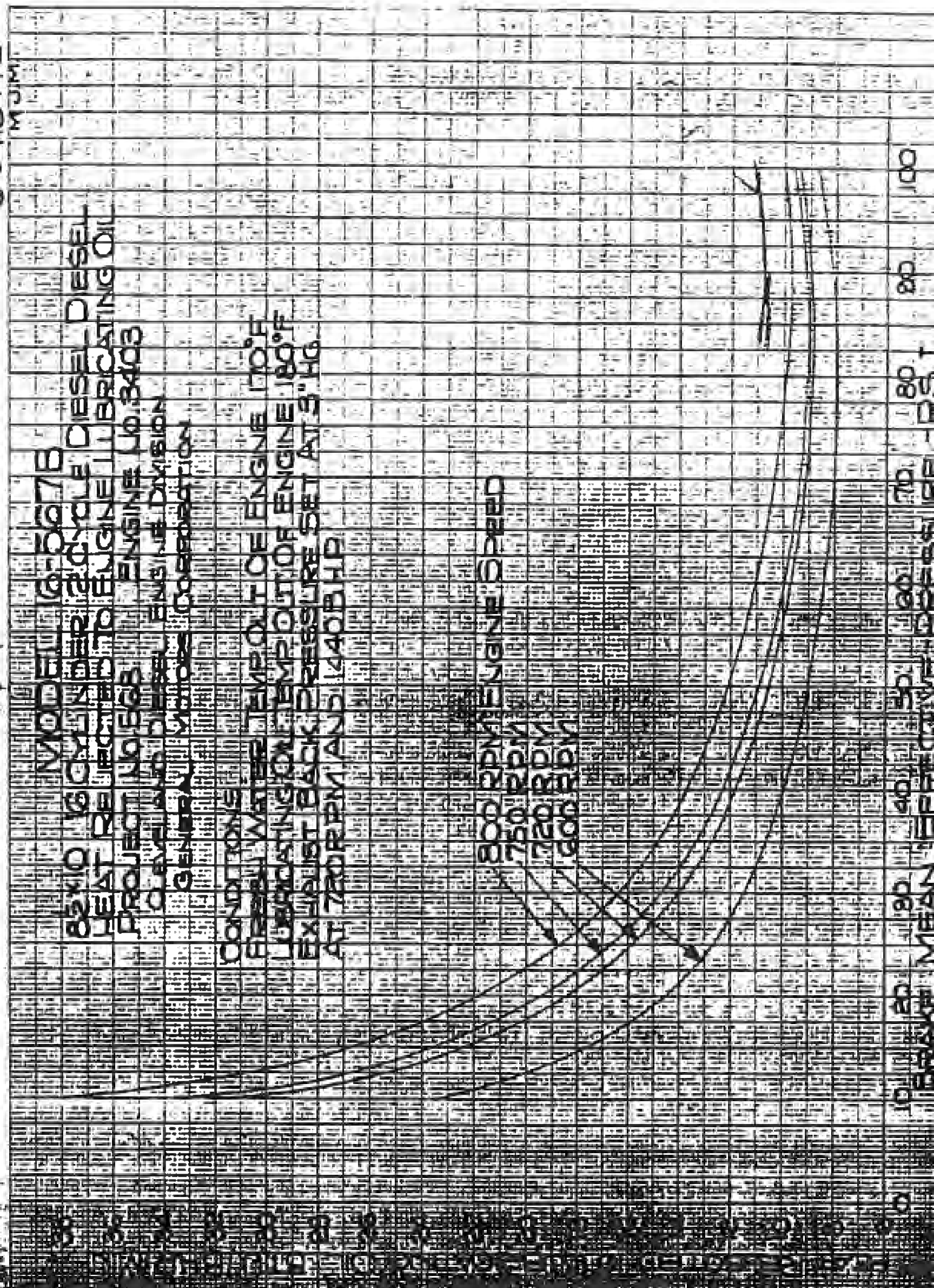

C-041847B
MODEL 16-567B
16 CYLINDER 2 CYCLE DIESEL
HEAT REJECTED TO ENGINE LUBRICATING OIL
PROJECT NO. 508 ENGINE NO. 3403
CLEVELAND DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION
CONDITIONS
FRESH WATER TEMP. OUT OF ENGINE 170°F
LUBRICATING OIL TEMP. OUT OF ENGINE 180°F
EXHAUST BACK PRESSURE SET AT 3" HG.
AT 720 RPM AND 1240 BHP
800 RPM ENGINE SPEED
750 RPM
720 RPM
600 RPM
0
10
20
30
40
50
60
70
80
90
100
BRAKE MEAN EFFECTIVE PRESSURE - PSI

FIGURE - 3

WSRC-TR-93-42-056

GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

APPENDIX C

MODEL 16-278A CYLINDER BALANCING AND EXHAUST GAS TEMPERATURES

ENGINE SYSTEM ANALYSTS

ENGINEERED APPLICATIONS CORPORATION
ENGINEERING AND DESIGN CONSULTANTS
476 Sarsfield Drive, Rochester Hills, Michigan 48307
Phone 313 - 853-3992

ENGINE DESIGN SPECIALISTS

July 3, 1991

Mr. Robert E. Gross P.E.
Westinghouse Savannah River Company
P.O. Box 616
Aiken, SC 29802

Subject: Model 16-278A Cylinder Balancing and Cylinder Exhaust Gas Temperatures

Dear Bob:

Recently your men asked me four questions on the cylinder power balancing and on cylinder exhaust gas temperatures.

Enclosed are my notes on these questions as follows:

The notes are:

"Balancing the power output between cylinders of the Model 16-278A engine."

"Engine exhaust temperature as a guide to balancing the cylinder to cylinder power output."

"The exhaust gas temperature difference between cylinders also the exhaust gas temperature limit from engine cylinder."

"Why are the exhaust gas temperatures for cylinder no. 5 and 15 different from average."

This last question on cylinders no. 5 and 15 still has me puzzled. I don't have a real good answer for you but if you wish I can make a more detailed study of the problem.

Sincerely,

Paul

Balancing The Power Output Between Cylinders Of The Model 16-278A Engines

The problem of balancing the power output between cylinders of an internal combustion engine is difficult for a number of reasons. The engine cylinders are not exactly the same, the fuel injection systems and spray patterns are different between cylinders and the carbon build up and cylinder breathing and cooling result in power output differences.

This problem of balancing the cylinders has been studied by all the engine builders. The conclusion is that different engine builders use different ways to balance the cylinders or a combination of the different methods.

One way to balance the engine cylinders is by balancing their cylinder firing pressures. This is the way it was done at the Cleveland Diesel Engine Division of GMC.

The reason for balancing the engine cylinders this way is that it improves the engine performance. (the combustion and load between cylinders is more uniform) as each cylinder delivers about the same power. The engine runs smoother and its torsional vibratory amplitudes are more regular and the possibility of exciting minor critical vibrations is reduced.

With this method the loads on the connecting rods, the piston pins and bushings and the loads on the connecting rod bearings are more uniform.

With a uniform load developed by each cylinder, it is easier for the governor to control the engine speed and load, specifically if the engine's cyclic irregularity is close to the systems critical value.

Also, with uniform firing pressure the loading on the cylinder heads and valves is more uniform and the piston side thrust is in better balance.

The piston side thrust on emergency start, keep warm engines has sometimes given problems, specifically with some oils.

Specifically, Cleveland Diesel recommended that the cylinders be balanced by cylinder firing pressure and the pressure variation to be about ± 50 PSI between cylinders.

Again, this method is not perfect but its history has shown it, to be good.

The instrument used for this cylinder balancing is a Kiene pressure gage. It is a pressure gage that is connected to the individual cylinder through a valve. The gage has a needle indicator that tries to follow the cylinder pressure as it changes. To read the gage it is sometimes difficult because the needle and pressure tubing have inertia and sometimes overshoot the cylinder pressure. This combination of conditions help to account for some of the cylinder to cylinder variation. This method, however, seems to be a reasonable practical compromise for measuring the cylinder pressure.

Another method for balancing the load between cylinders is by measuring the exhaust temperature from the cylinder.

This method was tried at Cleveland Diesel, but consistent results could not be obtained. Cleveland Diesel even went to the Bureau of Standards for calibrated cylinder exhaust thermocouples. The thermocouples were accurate to ±1°F. These thermocouples were better, but didn't balance the cylinders much better than the standard thermocouples which were said to be accurate to ± 12.5°F.

The reason for this difficulty may be due to the thermocouple location in the exhaust passage. Also the cylinders are scavenged with air (blower excess air) for cleaning out and cooling the cylinder and valves. The scavenging air may upset the couples reading. The thermocouples tend to average the exhaust temperature.

At Cleveland Diesel the thermocouples were used as indicators of reasonable engine cylinder operation but not for balancing the cylinders.

The third method for balancing the cylinders on some engines is to balance the fuel rack settings. This method also has its drawbacks, as calibrated fuel pumps with rack settings are required, however, the rack settings are not often set for easy reading and for easy adjustment. Also, if the fuel from the pump to spray nozzle are long and not with the same configuration, the expansion and deflection of the fuel lines, under pressure, give erroneous fuel delivery and because the fuel lines bend the pressure waves in the lines influence delivery.

Therefore, it seems that the best method with the least amount of error is to balance out the engine by balancing the cylinder firing pressures.

Paul

June 26, 1991

Engine Exhaust Temperature as a Guide to Balancing The Cylinder To Cylinder Power Output

The use of engine exhaust temperature to balance the cylinder to cylinder power output is only a fair method for balancing the engine performance and the power.

The reason is that there are so many parameters between cylinders that influence the cylinder exhaust temperature that it is difficult to standardize the readings. A few of these items are as follows:

1. The temperature accuracy of the thermocouples is not close enough for good engine control. Experience shows that the thermocouples have an accuracy of about ± 25°F.

2. The location of the thermocouples in the cylinder head is not always the same and they do not always get the same amount of cooling. Depending upon the water flow in the cooling core cavity and how well it is cleaned, will depend on the thermocouple cooling. Then the depth and orientation of the thermocouple will influence its temperature

2.

3. Generally there is an excess amount of air delivered to the engine above that required for combustion. This air is delivered by the supercharger or blower (generally for two cycle engines) and by the turbocharger compressor for both two and four cycle units.

This excess air is used for cooling the cylinder and the exhaust valves. This excess air blowing on the thermocouples influences its reading. This is one of the reasons that the exhaust temperature from the combustion analysis does not agree with the measured temperature.

4. The combustion conditions in the cylinder influence the cylinder exhaust temperature. The spray pattern and burning of the fuel can be different between cylinders.

5. The association reactions in the cylinder and the disassociation reactions of the products of combustion as they leave the cylinder influence the temperature that the cylinder thermocouple sees.

6. The exhaust back pressure of the exhaust and the traveling waves in the

exhaust manifold influence the rate that the exhaust leaves the cylinder. Here we introduce a time function in the exhaust temperature readings.

The length of the exhaust pipe and the location of the muffler add to the traveling wave problem.

Referring to the exhaust pipe pressure drop calculation report for the 16-278A engine it was found that the volume of the exhaust from the engine was 12500 CFM and its weight was 12500 x .0357 = 446.3 pounds per minute. The back pressure on the engine was 32 inches of water. This back pressure was equivalent to

$$\frac{32 \times 62.4 \times 144}{12 \times 144 \times .0357} = 4661 \text{ ft. of exhaust}$$

gas. The engine horsepower to expell the exhaust gas is $\frac{4661 \times 446.3}{33000} = \underline{63\,HP}$ without any efficiency corrections.

This value is about 3.5% of the horsepower developed by the engine.

This horsepower influences not only the engine exhaust temperature, but also some of the pressure waves that help or interfere with individual cylinder breathing and also the roots type scavenging blower requirements.

7. Also there is a possibility of camshaft twist due to cam loads or due to torsional vibrations of the 16-273A engine camshafts. These twists or vibrations allow the exhaust valves to open earlier or later and also to change the valve opening duration.

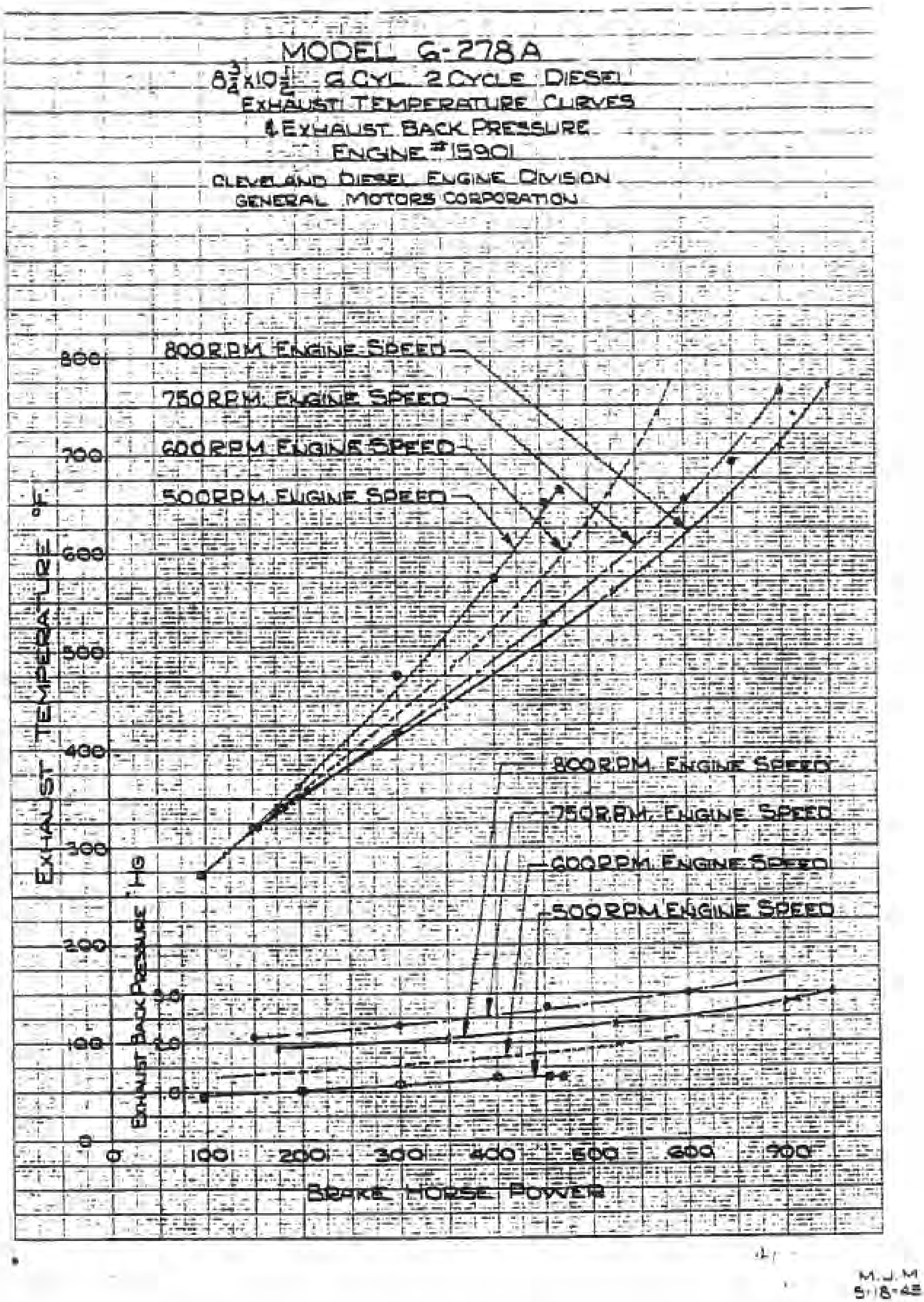
The camshaft is driven from one end so the twist due to cam load changes the valve timing making each cylinder from the drive end open later in the engine expansion cycle, a condition that changes the exhaust gas temperature.

As is noted these different operating conditions cause possible changes in the exhaust gas temperature from the individual cylinders. Therefore to balance out the engine by using the cylinder exhaust gas temperature is difficult at best. Using the firing pressures is a better cylinder balancing criterion.

Attached is an exhaust (cylinder) temperature curve for a 6-278A engine run at different loads and speeds at different exhaust back pressures.

Paul

June 28, 1991


MODEL G-278A
$8\frac{3}{4}$ x $10\frac{1}{2}$ G CYL 2 CYCLE DIESEL
EXHAUST TEMPERATURE CURVES
& EXHAUST BACK PRESSURE
ENGINE #15901
CLEVELAND DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION
800 RPM ENGINE SPEED
750 RPM ENGINE SPEED
600 RPM ENGINE SPEED
500 RPM ENGINE SPEED
EXHAUST TEMPERATURE °F
EXHAUST BACK PRESSURE " HG
BRAKE HORSE POWER
M.J.M
5-18-43

The Exhaust Gas Temperature
Difference Between Cylinders also
The Exhaust Gas Temperature Limits
From Engine Cylinder

As mentioned in a previous writeup, the engine cylinder exhaust temperature is not usually used as a tool for balancing the engine cylinder power output. Exhaust temperature measurements are an indicator that the cylinders are burning the fuel normally.

A 200°F difference in temperature as say ±100°F from the average exhaust of all cylinders is a reason for investigation but considering the many parameters that come into play it is generally no cause for concern unless the cylinder sounds rough or the engine exhaust shows smoke.

However if the deviation is ±200°F from the cylinder average then the problem must be investigated as something needs correcting.

Usually the problem can be a sticky injector or rack, improper injection timing, pump not standard or plugged or cratered spray nozzle. There are other possibilities but generally the mentioned problems are –

the just one to check.

Besides the exhaust temperature problem just mentioned a question was raised about the maximum cylinder exhaust temperature.

The Model 16-278A engine exhaust valves and seats were designed to operate up to 900°F cylinder exhaust outlet temperature. Generally the naturally aspirated engines operate below 800°F but on occasion this temperature is exceeded.

This same model engine was later turbocharged and under some conditions the cylinder exhaust did approach about 900°F and no problems developed.

Therefore operating the engine with a cylinder exhaust temperature of 900°F will not cause trouble even when operating in an emergency for two hours or more.

Paul
July 1, 1997

Why Are The Exhaust Gas Temperature For Cylinders No. 5 and 15 Different From Average

It has been reported that the exhaust temperature from cylinders no. 5 and 15 deviate considerably from the cylinder average and the question is why? It is understood that the injection pumps have been changed out but the cylinder exhaust temperature did not change.

The problem appears to be associated in some way with the engine and the way it runs.

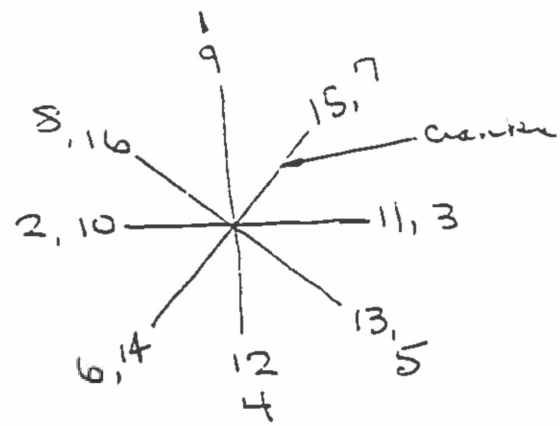
A few possibilities are engine crankcase vibration, camshaft bending or torsional vibrations or the injector control shaft bending or vibrating. It seems that some vibration is causing the fuel injection pump to deliver more or less fuel than it is timed to deliver. Could these vibrations cause the injector control rack to vibrate and move in or out

The engine firing order is:

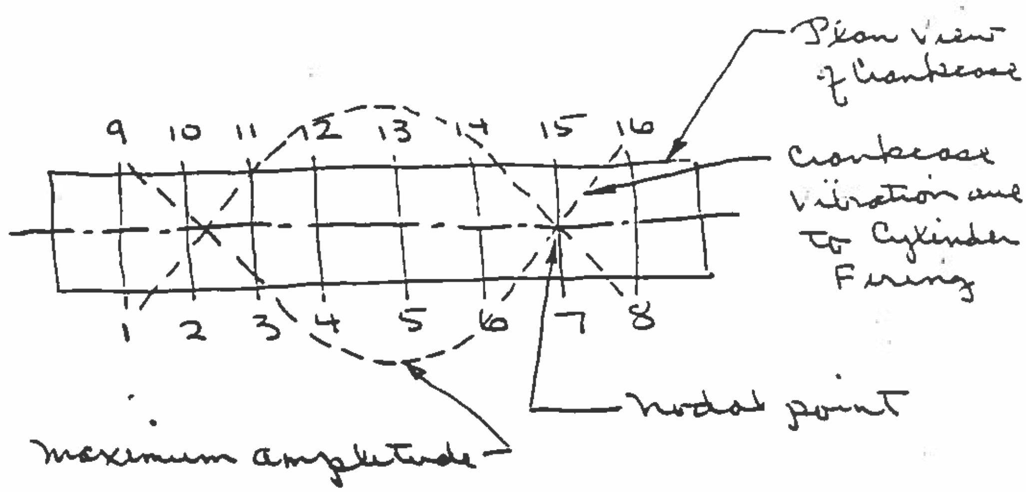
1 9 8 16 2 10 6 11 4 12 5 13 3 14 7 15 1

2.

Looking at the end of the crankshaft the cranks look as follows



Note, that cylinders 4, 5 and 6 and 11, 12 and 13 fire one after the other and they are on one side. This condition causes the crankcase to bend back and forth horizontally about as shown.



3.

With the cylinder firing arrangement shown the engine crankcase will move and vibrate (forced vibration) somewhat as shown. The frequency of this vibration is once per revolution or 12 VPS.

Could this vibration cause the fuel control rod to move the specific injector racks so they deliver more or less fuel. It should be noticed that cylinder no. 5 is at a maximum amplitude point on the crankcase and cylinder no. 15 is at a nodal point.

The injector control shaft is supported in bearings but it still could vibrate and change the injection pumps delivery. The bearings on this shaft should act as dampers.

The crankcase deflection is assumed to be about ±.005 inches. This motion if transmitted directly to the fuel injection pumps would represent about ±1% of the fuel delivery. If 2% of the fuel pump delivery is used this change in fuel would change the exhaust temperature about 15 degrees F. So the injector rack motion must be greater than the total rack motion of .010 inches. In other words there must be some vibration amplification.

Another possibility is the camshaft bending motion which would move the fuel cam and change fuel delivery. Also the camshaft might have a torsional vibration which would rotate the fuel cams and change the injection pump delivery.

The camshaft is driven from the rear of the engine and because of the cam loads the camshaft twists so the injection timing is later for the front cylinders due to this twist. This camshaft twist could also set up a vibration at different cams that would influence individual fuel delivery.

Paul,
July 2, 1991

WSRC-TR-93-42-056

GENERAL MOTORS (GM) DIESEL TREND ANALYSIS RESULTS 1991-1993

APPENDIX D

PRESSURE WAVES IN THE EXHAUST MANIFOLD OF THE MODEL 16-278A AND OTHER ENGINEERING INFORMATION

ENGINE SYSTEM ANALYSTS

ENGINEERED APPLICATIONS CORPORATION
ENGINEERING AND DESIGN CONSULTANTS
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Phone 313 - 853-3992

ENGINE DESIGN SPECIALISTS

August 5, 1991

Mr. Robert E. Gross P.E.
Westinghouse Savannah River Company
P.O. Box 616
Aiken, SC 29802

Subject: Pressure Waves in The Exhaust Manifold of The Model 16-278A Engines and Other Engineering Information

Dear Bob:

As we talked last Friday, the answers to a number of mentioned problems are as follows:

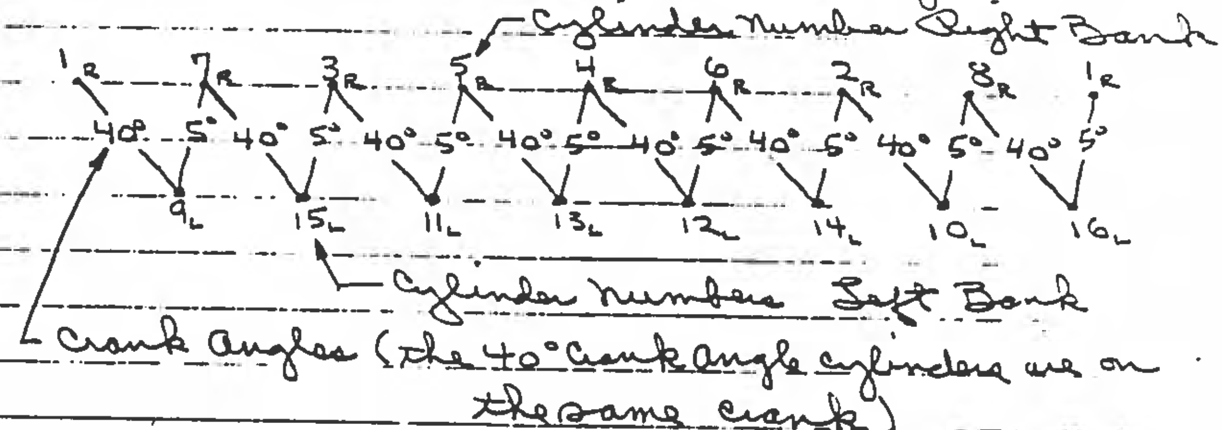
1. Enclosed are the calculations for the possible pressure waves in the engine exhaust manifold. The pressure waves appear to influence the exhaust temperatures of some cylinders.

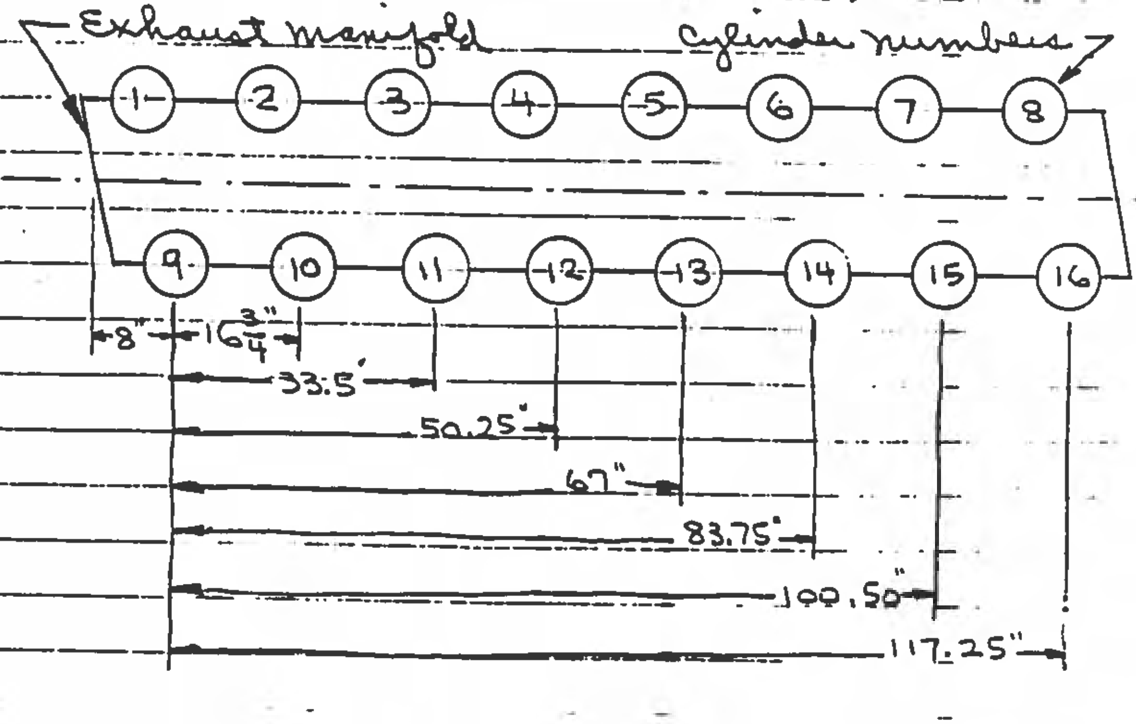
Lowering the exhaust back pressure on the engine would possibly reduce the pressure wave influence. It would also help all the cylinders scavenge better (breathe better).

2. The 100 P.S.I. peak pressure difference between cylinders requires a

Westinghouse Savannah River Company
Model 16-278A Engine
Pressure Waves in Engine Exhaust Manifold

The pressure waves may possibly cause the low cylinder exhaust temperature at cylinder no. 6 and the high temperature at cylinder no. 14 in the manifold.





Assume that the mean exhaust temperature in the exhaust manifold is about 720°F and the temperature of the gas out of the cylinders is about 750°F at both full load and overload.

Also because one bank of cylinders fire 5 crank degrees after the cylinders of the other bank, it is assumed that there are 8 firing impulses per crank rotation.

Also the cylinders in question no. 5, ~~6~~ and 14 are on the same crank. ✓

RSG see my note.

The velocity of a pressure wave (velocity of sound) in the exhaust manifold is $= 48\sqrt{T_e}$ per Taylor in "Internal Combustion in Theory and Practice" Vol. I

Velocity of a pressure wave $= 48\sqrt{720+460} =$ 1649 ft. per sec.

The pressure wave travels from Cyl. no. 6 or 14 and its length of travel is.

$$\frac{5 \times 16.75 + 8}{12} = \underline{7.6} \text{ feet (manifold)}$$

The length of the wave to travel from cyl. no. 6 to the end of the manifold and back to cyl. no. 6 is $\frac{1}{2}$ a wave length, one quarter wave down and one quarter wave back.

one quarter wave — for a total of a half wave

$$\text{the wave length } \lambda = \frac{1649}{\frac{720}{60} \times 8_{\text{cylinders}}} = 17.17 \text{ ft.}$$

$$\text{and a half wave length} = \frac{17.17}{2} = \underline{8.6} \text{ ft.}$$

Looking at the other end of the manifold the length of travel is

$\frac{3 \times 16.75 + 8}{12} = 3.46$ ft. and a quarter wave length is $\frac{17.17}{4} = 4.29$ ft.

So it appears that both the one half and the one quarter waves influence the exhaust flow from cylinders no. 6 and 14. In one case the cylinder breathes better so its temperature is lower because more scavenging air flows out the cylinder and in the other case the pressure waves obstruct the flow from the cylinder so the exhaust temperature is higher.

The engine exhaust back pressure also influences the magnitude of the pressure waves in the exhaust manifold. a decrease in back pressure would let the cylinders scavenge better and would probably reduce the magnitude of the manifold pressure waves.

The exhaust outlet location from the manifold also has some influence on these pressure waves.

Paul

Aug. 1, 1991

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