Effects of Lubrication System Parameters on Diesel Particulate Emission Characteristics

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ABSTRACT

A substantial part of diesel particulate emissions comes from the engine lubricant. Some results of a program to evaluate a strategy of particulate emission reduction by controlling oil consumption in a diesel engine are presented. This paper focuses on the effects of lubrication system parameters on particulate emission rate and composition. Engine load, viscosity and a piston-ring parameter, specifically the width of the top-piston-ring gap, were chosen as variables. Particulate rate and composition were measured and analyzed for multiple combinations of lubricant and ring-gap configurations at three different engine operating conditions. Particulate samples were collected on 47mm Teflon-coated filters from diluted exhaust through a scaled-down constant volume sample (CVS) dilution system. Each sample underwent a soxlet extraction to determine the soluble organic fraction (SOF) and a unique gas chromatograph method to determine percent lubricant contribution to the SOF. Particulate rates were found to be highest with the lower viscosity oil and the larger top-piston-ring gap combination. At high load, the difference in particulate rate was due to changes in the non-soluble portion, while at medium and low loads, the change in particulate rate was due to differences in the lubricant derived portion of the SOF. In addition, changes in the fuel derived portion of the SOF were discovered and attributed to changes in the amount of fuel absorption in the oil film. These results show that by properly controlling lubricant properties and piston ring designs, there is a definite potential to reduce particulate emissions from a diesel engine via lubrication system control.

INTRODUCTION

DIESEL EMISSIONS AND PARTICULATES: - Emissions from diesel engines play a major role in the deterioration of air quality. Be it from automobiles, trucks or ships, the aggregate effect of even low emission levels can have a marked effect on the health of humans and the environment. Diesel particulates are commonly defined as any exhaust material, except water, that can be collected on a filter in diluted exhaust at a temperature below 53°C, following protocols established by the US Environmental Protection Agency (EPA) [1]. Particulates consist of aggregates of small particles. Predominantly, each particle comprises a carbon core with a complex collection of hydrocarbons and inorganic compounds which have condensed or adsorbed on the surface [2]. It is the collection of adsorbed compounds, particularly the carcinogenic polynucleic aromatic hydrocarbons (PAH), which may create the most significant health risks [1,3]. Using traditional analysis, these compounds can be extracted using organic solvents such as dichloromethane, and the soluble organic fraction (SOF), i.e. the soluble organic percentage of the total particulate mass, can thus be determined [1]. Clearly, it is desirable to reduce not only the total particulate rate, but the harmful carcinogenic compounds found in the SOF. In conjunction with the U. S. Clean Air Act Amendments of 1990, the U. S. Environmental Protection Agency (EPA) has set strict standards for the reduction of particulate rates. Most significantly, they have set the current (1994) heavy-duty vehicle standard of 0.134 g/kwh (0.10 g/bhp-hr) for particulates and 6.702 g/kwh (5.0 g/bhp-hr) for oxides of nitrogen (NO_x). The 1996 urban bus standards are 0.067 g/kwh (0.05 g/bhp-hr) for particulates and 6.702 g/kwh (5.0 g/bhp-hr) for NO_x [1,3]. The 1998 standards are even more stringent at 0.134, and 5.362 g/kwh (0.10, and 4.0 g/bhp-hr) for particulates and NO_x respectively.

PARTICULATES AND OIL CONSUMPTION CONTROL STRATEGY: - Most emission control strategies via combustion control to reduce NO_x often result in higher particulate rates. Therefore, the simultaneous control of oxides of nitrogen and particulate emissions from diesel engines is a complex problem. Important advances are being made via improvements in the combustion system, including changes such as higher fuel-injection pressures, combustion chamber and piston ring-pack designs, exhaust gas recirculation and electronic controls, etc. In addition to the above, engine manufacturers and suppliers are actively investigating control of particulate emissions contributed by the lubricant. To reach extremely low emission levels, reduction of particulates via oil-consumption control is becoming an essential part of the total strategy. In contrast to other major combustion system changes which are being incorporated in new engines, particulate reduction via lubrication system improvement also offers the potential of application to engines in existing trucks, off-highway vehicles and marine vessels. Much work remains to be done before it can be implemented, however.

Percentage contribution of lubricant to the total particulate matter varies predominantly by engine condition, reaching as high as 70% of total particulates [4,5] and 90% of the soluble organic fraction in the particulates [6-8] under some operating conditions, viz. light to medium loads. The problem is less severe at high loads, as most of the consumed oil is oxidized due to the higher temperatures. For engines of older vintage, the oil contribution to particulates alone far exceeds the current standard for the total particulate matter for new engines. The lubricant contribution will become increasingly important as the total particulate emission level continues to be reduced in the future.

PRIOR INVESTIGATIONS: - Mayer et al [9] of General Motors Research Labs were among the first ones to report the significance of lube-oil contribution to diesel particulates. It was reported that the potential reduction in particulates in a 1978 vintage diesel engine was up to 0.73 g/kwh (0.54 g/hp-hr), which is more than five times the level allowed by current regulations of *total* particulate matter for heavy-duty diesels. Since then, researchers from piston-ring, piston, engine and lubricant manufacturers and others [5,6,10-13] have reported the opportunities for particulate reduction via component and lubricant changes. Dowling [5] found that particulate rate increases with decreasing viscosity and increasing volatility. In addition, he concluded that oil contribution increases with increasing speed and decreasing load. Andrews et al. [8], found that there is also significant influence of oil age which is caused by fuel dilution and carbon thickening, thereby altering the oil viscosity. Zelenka et al. [12] and Essig et al. [10] show that the emission rate of oil derived particulates is directly proportional to the oil consumption rate. They further conclude that reduction of cylinder wall oil consumption, the main source of lubricant derived particulates, should be a primary focus in reducing overall particulate mass. Experimental data show that oil film thickness increases with decreased viscosity [14]. This could indicate that lower viscosity oils leave more residual oil on the liner, therefore explaining an increase in particulate rate with decreased viscosity. Conclusions in recent reports [1,15] indicate that it is necessary to further reduce lube oil consumption to reach mandated diesel particulate emission levels.

Researchers have postulated several sources of oil consumption. These include (a) the release of the accumulated oil on the top piston land into the combustion chamber due to inertial forces, (b) the piston ring-pack dynamics and flow driven release of oil through the top ring [16], and (c) oil volatility effects. For many years, verification of these models has been lacking, due to the difficulty in measuring the oil behavior in the piston-ring pack region. The current study of the effects of ring-gap width and lubricant properties will help understand these oil consumption mechanisms and through which develop additional strategies of particulate reduction.

PROJECT SCOPE AND PRESENTATION

This paper studies the effects of lubrication system parameters on particulate emission rate and composition. Specifically, engine load, lubricant viscosity, and a piston-ring parameter: the width of the top-piston-ring gap, were chosen as variables. Fifteen tests were run for each of three lubricant-and-ring combinations: 10W-30 oil with standard ring configuration, 15W-40 oil with standard ring configuration, and 15W-40 with an enlarged top ring gap configuration. Each fifteen-test group included five tests each at three engine operating conditions. The results should further reinforce prior reports on the relationship between oil consumption and particulate emissions, while giving additional credence to ring-pack theories of oil consumption.

The engine, particulate sampling system, experimental and analytical procedures will first be described, followed by results and discussion. Implication of the results will also be presented.

EXPERIMENTAL APPARATUS

ENGINE CHARACTERISTICS:- The experimental engine used was a single cylinder Ricardo/Cussons standard Hydra engine connected to a Dynamatic Model 20

AC dynamometer with a Digalog controller. Details of the engine are given in Table 1.

MANUFACTURER:	G. CUSSONS LTD.
MODEL:	HYDRA RESEARCH ENGINE
NUMBER OF CYLINDERS:	1
DISPLACEMENT:	0.45 L
BORE:	80.26 MM
STROKE:	88.9 MM
MAXIMUM SPEED:	4500 RPM
MAXIMUM POWER:	8 KW
MAXIMUM CYLINDER PRESSURE:	120 BAR
COMPRESSION RATIO:	20:1
INJECTION:	DIRECT

Table 1 Engine Characteristics

Injection timing settings were manually controlled by the operator. For this study, the timing was changed with engine speed to the value recommended by the manufacturer to be "optimum" [17]. The piston has a three-ring configuration, i.e., two compression rings and an oil control ring. It was the top compression ring that was altered for this study. The engine is fully instrumented. Oil, coolant, air, and exhaust temperatures were monitored at various locations. This monitoring allows the operator to insure constant conditions between tests. Engine speed is measured by a magnetic pick-up mounted from a gear on the drive shaft. Engine load is measured by a load cell mounted at a fixed distance from the drive shaft axis between the dynamometer and test bed. Both the oil and coolant are both pumped and can be heated electrically, allowing great flexibility of temperature and ease of flushing.

PARTICULATE SAMPLING SYSTEM:- The particulates were collected using a scaled-down version of a constant volume sample (CVS) dilution tunnel and system similar to that described by Wong et al. [18]. A diagram of the system is shown in Figure 1. Compressed air is supplied through a 5.08 cm (2") line from large shop compressors. A 5.08 cm (2") Balston A15/80-DX filter is installed upstream of the tunnel to remove oil and moisture from the air. The stated removal effectiveness of this filter is 93%. As a background test, a sample of air was drawn through the particulate system with the engine off to determine if this filter sufficiently cleaned the air. No measurable material collected on the particulate filter after a one hour test.

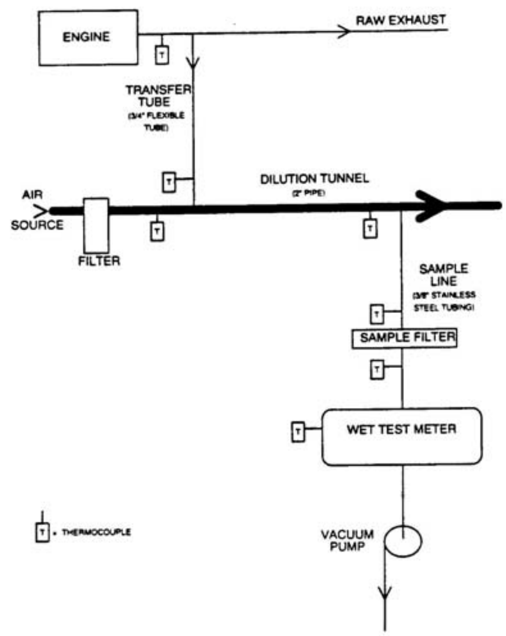

Figure 1 Particulate Sampling System

After filtration, the air enters the 5.08 cm (2") pipe dilution tunnel. An exhaust sample is drawn into the tunnel through a 1.91 cm (3/4") flexible transfer tube using a venturi powered by the dilution air. The transfer tube is connected to the main 5.08 cm (2") raw engine exhaust line. The air and exhaust are mixed over an approximately 91 cm (3') tunnel length. The diluted sample is then drawn through the filter and wet test meter. The sample line is 0.95 cm (3/8") 316 stainless steel tubing. Pallflex Teflon coated 47 mm fiber filters, P/N TX40H I20WW, are used to collect the sample, and are mounted in a Graseby Anderson 316 Stainless Steel filter holder, P/N SE273. The sample line can be isolated from the dilution tunnel by an installed high temperature stainless steel ball valve. After filtration, the sample is drawn through a wet test meter manufactured by Precision Scientific Petroleum Instruments Company. This meter measures total volumetric flow of the sample. A positive displacement vacuum pump draws the sample through the sample line.

Dilution ratio is measured by sampling the CO_2 concentration of the exhaust before and after dilution. Temperatures can be measured at various locations in the sampling system. Most importantly, thermocouples are located before and after the filter enclosure to insure that the sample is below 51.7°C [19], thereby allowing sufficient hydrocarbon condensation. Temperatures of the transferred exhaust, incoming air, and diluted air can also be measured. The transfer tube is insulated with a high temperature fiber "exhaust blanket." Otherwise, the system was not insulated. As a result, there are significant heat losses to the environment. In some cooler conditions, dilution is not necessary as sufficient cooling occurs from the losses alone. To reduce the possible catalytic effects, the dilution tunnel and sampling lines are mild or stainless steel. All components of the system were cleaned with a de-greasing agent and dried before construction. Care was taken to minimize or eliminate any thread compounds from residing on the inside surface of the tunnel. Before testing, the engine was run at its hottest condition without dilution air to oxidize any possible remaining contaminants.

EXPERIMENTATION

TEST PROCEDURE:- The particulate filters were conditioned and weighed to 10 µg by a commercial test laboratory and shipped for testing [20]. In addition, each filter was weighed in the engine laboratory to 100 µg. The filter was then placed in the filter holder and wrench-tightened to avoid O-ring bleed. Starting sample volume was recorded. After the engine was settled at the proper operating condition, the test began by opening the sample line isolation valve, starting the vacuum pump, and recording the time. The dilution air remained on whenever the engine was running to keep the temperatures stabilized. Other pertinent data were recorded while the sample was being collected. At the end of the sampling interval, the isolation valve was closed, the vacuum pump secured, and the final readings taken. The filter was immediately weighed, packaged in double sealed plastic bags and cooled to refrigerator temperature (4°C). The samples were then shipped to the analysis laboratory. While weights were recorded in the engine laboratory, only values obtained by the analysis laboratory were used, as the samples were post-conditioned prior to final weighing. Immediately after packing the sample, the engine load was changed in preparation for the next test. Between each round of fifteen tests, the lubricant or top ring was changed, as these were the two primary variables in the experiment. Oil changes were completed using a fill-flush-fill technique. Old oil was heated to 80°C and drained from the sump. The discharge line from the oil pump was then disconnected and re-routed into a collection vessel. Per manufacturer's recommendation, the pump was then turned on and allowed to suck all remaining oil out of the sump [17]. The oil system was then reassembled. The engine was then filled with new oil of the same viscosity as the replacement oil. It was heated to 80°C and pumped through the system as the engine was motored at 1200 rev/min for 15 minutes. The same technique was then used to drain and pump out this flushing oil. With the sump dry, the filter was changed and the engine re-filled with the new oil. After each change, the engine was run at 3000 rev/min and high load for 5 hours per the findings of an earlier study [21] that most volatility reduction in new oil occurs during the first five hours of high temperature operation. The top ring was changed by removing the piston through the top of the liner. The three ring gaps were oriented at 120° from each other to minimize the possibility that the gaps would align, thereby increasing blow-by.

TEST MATRIX:- Three variables were used for this study. Each test condition was run five times for repeatability. Two production oils of differing viscosity (10W-30 & 15W-40) with the standard manufacturer's ring configuration were used for the first comparison. Then, using the 15W-40 oil, the top ring gap was enlarged for a second comparison. For each of these three lubricant parameter conditions, the engine was tested at three loads. 45 total tests were conducted for final analysis. Table 2 summarizes the test matrix.

LUBRICANT	RING CONFIGURATION	ENGINE LOAD	# OF TESTS
10W-30	STANDARD	HIGH	5
		MEDIUM	5
		LOW	5
15W-40	STANDARD	HIGH	5
		MEDIUM	5
		LOW	5
15W-40	ENLARGED TOP GAP	HIGH	5
		MEDIUM	5
		LOW	5

Table 2 Test Matrix

Engine Load Conditions:- All tests were conducted at 2400 rev/min. Table 3 defines the target values for the engine loading conditions.

Variable Specifics:- The lubricants selected were production oils designed for diesel applications. While original intentions were to use specially formulated oils with known and controlled volatilities, logistical constraints dictated the use of production lubricants. Table 4 details the lubricant properties. The rings used were those provided by the engine manufacturer. For the altered condition, the top ring cold gap was enlarged to approximately six times the standard value. This alteration was purposely extreme to insure that the results show a significant change in emissions. The specifics of the alteration are listed in Table 5.

ENGINE LOAD	TORQUE (FT-LB)	FUEL FLOW RATE (CC/MIN)	EXHAUST TEMP. (°C)
HIGH(100%)	15.0	25.8	460
MEDIUM(67%)	10.0	19.8	350
LOW(33%)	5.0	15.4	270

Table 3 Engine Load Conditions

LUBRICANT	MANUFACTURER/ BRAND	VISCOSITY (cST @ 100°C)	FLASH POINT °F	POUR POINT °F
10W-30	SHELL/ROTELLA T®	10.9	405	-35
15W-40	SHELL/ROTELLA T®	14.0	415	-35

Table 4 Lubricant Properties [18]

INITIAL COLD RING GAP (IN)	.020
ALTERED COLD RING GAP (IN)	.119

Table 5 Ring Dimensions

SAMPLE ANALYSIS:- Subsequent to testing, each filter was packaged and sent to the analytical laboratory for analysis. Samples were handled per U. S. EPA guidelines as stated in 40 CFR 86. The soluble organic portion of the sample was extracted using the soxhlet extraction method [20]. SOF was then calculated by analytical laboratory.

The soluble organic samples were then analyzed for fuel-versus-lubricant contribution. The fuel/lubricant contribution analysis was completed using a unique method developed by the Cummins Engine Co. The method is a modification of ASTM D2887 SIMDIS of petroleum products. The fractions of fuel and lubricant in the particulate sample are determined using a ratio of integrated times obtained from chromatograms of three samples: the extracted SOF, "topped" fuel, and new lube oil. (The "topped fuel" is the remainder of the fuel after 30% by volume is distilled.) The precision of lubricant derived portion of the SOF result is on the order of 0.001 g/bhp-hr.

RESULTS

TOTAL PARTICULATE RATE:- As predicted, mean particulate rate was higher using the 10W-30 oil, the lower viscosity oil, in all three loading conditions. Figure 2 displays this trend. The high and medium load conditions showed modest differences in particulate rate, while the low load condition had a more pronounced change.

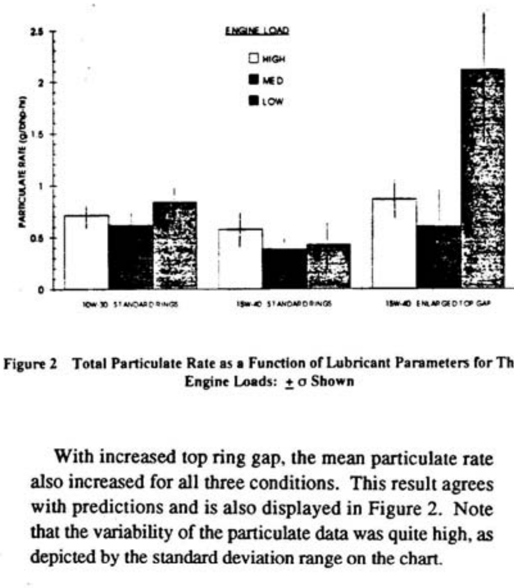

Figure 2 Total Particulate Rate as a Function of Lubricant Parameters for Three Engine Loads: $\pm\sigma$ Shown

With increased top ring gap, the mean particulate rate also increased for all three conditions. This result agrees with predictions and is also displayed in Figure 2. Note that the variability of the particulate data was quite high, as depicted by the standard deviation range on the chart.

Variability of engine load also created a consistent difference in particulate rate. Figure 2 shows that the rate was highest at low load and lowest at medium load for all lubrication conditions.

PARTICULATE COMPOSITION:- Consistent particulate composition trends were quite intricate. Each condition had clearly different results. Figure 3 shows the mean compositions for the three lubrication conditions in the high load condition. Both the fuel and lubricant derived portion of the SOF were almost constant, while the non-soluble content increased with decreasing viscosity and increased ring gap. This implies more than the traditional belief that the change in particulate rate as a function of lubricant parameter is created by a change in lubricant-derived soluble organic compounds. The results indicate that the lubricant also affected the non-solubles, i.e. carbon, etc. The implications are discussed in a later section.

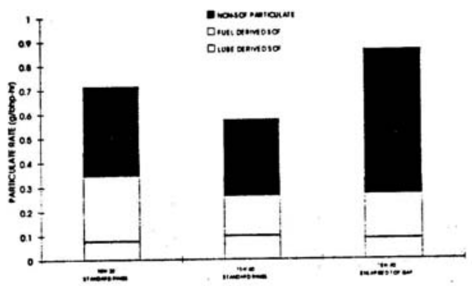

Figure 3 Particulate Composition as a Function of Lubricant Parameters - High Load Condition

Figure 4 shows the composition of the particulates from the medium load condition. In this case, the non-soluble content remains almost constant with changing lubricant parameters, while the lubricant derived portion of the SOF changes as predicted, i.e. it increases with decreasing viscosity and increasing ring gap. In addition, the fuel derived portion of the SOF increased with decreasing viscosity and enlarged ring gap.

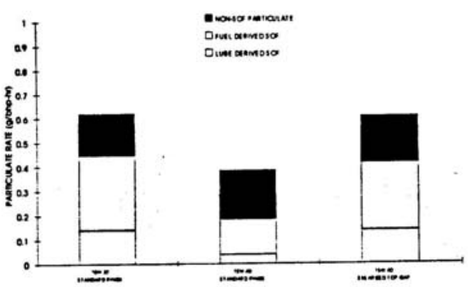

Figure 4 Particulate Composition as a Function of Lubricant Parameters - Medium Load Condition

At low load condition, the variability of results is larger, but also show increased lubricant-derived SOF with increasing ring gap. Figure 5 illustrates the data. In addition, the fuel-derived SOF also varies significantly with lubricant parameter. The data show a fuel trend similar to that of the oil. At low load, it is clear that the lubricant parameters affect the fuel composition of the particulate.

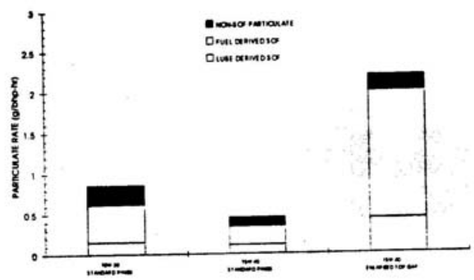

Figure 5 Particulate Composition as a Function of Lubricant Parameters - Low Load Condition

DISCUSSION

As predicted, total particulate rate increased with decreasing viscosity. This result is not only consistent with previous studies, but also is physically reasonable. As mentioned earlier and shown in another concurrent study, experimental data indicate that oil film thickness increases with decreasing viscosity [14]. Therefore, more lower viscosity oil will remain on the liner during the engine expansion stroke, exposing more oil to the flow-entrainment and combustion process. This additional exposure is then manifested as a higher particulate rate, as more oil participated but was not totally oxidized in the combustion process.

A second argument also supports the data. According to the a ring-pack theory of oil consumption, called the Puddle Theory, oil consumption depends on the size of a puddle of oil near the top ring-gap. This puddle of oil is blown through the top ring gap during a brief period of reverse gas flow through the gap when the land pressure exceeds the cylinder pressure. The theory indicates that as viscosity decreases, Taylor Number (Ta = U/s) increases, causing an increase in the puddle area. This larger puddle then increases oil consumption and particulate rate as a larger puddle enters the combustion chamber through the ring gap [16].

Similarly, the total particulate rate increased with the enlarged ring gap. Again, this data agree with the predicted result such as the ring-gap reverse flow theory. An enlarged ring gap allows a bigger puddle of oil to pass into the combustion chamber when the cylinder pressure drops below the inter-ring pressure during expansion. This increased puddle size is also manifested as an increase in particulate rate, as more oil enters the combustion chamber.

At the high load condition, the predominant change in particulate composition is the non-soluble portion. Due to both viscosity and ring configuration changes, the traditional belief is that the change in particulates would be due to a change in oil-derived SOF, as it is the combustion of additional oil that causes the change. In the high load condition, however, this was not the case. With the hotter in-cylinder temperatures, it is quite plausible that the additional oil was partially oxidized and formed carbon. Such a process would cause an increase in carbonaceous, or non-soluble, particulate, exactly as the data indicate.

At the medium and low load conditions, i.e., when the engine was running cooler, the predicted change in oil-derived particulate did occur, while the non-soluble portion remained somewhat constant. These results support the conclusion that at higher loads, the additional partially burned oil will become part of the carbonaceous particulate. These results also suggest the presence of a threshold temperature, above which the additional oil is oxidized instead of leaving the cylinder as altered hydrocarbon compounds.

Curiously, at all three conditions, the fuel-derived portion of the SOF changed to some degree with the lubricant parameters. The changes were the same as those expected of the oil derived portion, i.e. the fuel derived SOF increased with decreasing viscosity and increased top ring gap. This data suggest that the fuel-derived SOF increases when the lubricant parameters allow a stronger fuel-oil interaction. Preliminary experimental data from other studies show that some fuel is stored in the oil film during the combustion event. The fuel is absorbed during the rapid pressure increase then desorbed when the cylinder pressure decreases, thus escaping the combustion event. Additionally, the studies show more fuel is absorbed with increased film thickness [22]. This phenomenon can explain the changes in fuel-derived SOF as a function of viscosity, as decreases in viscosity create a greater film thickness and therefore more fuel absorption. The changes in oil formulation also may have an effect on the amount of fuel absorbed in the oil film. Increased amounts of oil in the combustion chamber, therefore, will not only create more combusted or consumed oil, but will prevent additional fuel from oxidation.

Regarding variability of the results, ring rotation could be a contributing factor. Schneider et al. [23] found that rings rotate erratically and can reach rates of 0.25 rev/min or faster in a four-cylinder engine. While this rotation rate would indicate that several rotations during the sample duration, the erratic and inconsistent nature of the rotations can still create variability. With the three rings rotating, relative gap orientations are changing, creating occasional situations where two or more of the gaps are aligned. Based on the findings that particulate rate is affected by ring gap parameters, it can be concluded that ring rotation could have a perturbing effect on the particulate rate measurements.

CONCLUSIONS

The results indicate that lubricant viscosity and the gap size of the top piston-ring do have a significant effect on diesel particulate emissions. Various specific conclusions can be drawn from the results and are summarized as follows:

1) Total particulate emission rate increases with decreasing viscosity. This effect is due most likely due to two mechanisms. First, the decreased viscosity leaves a thicker film on the liner during and after the expansion stroke, allowing more oil to participate in the combustion event. Second, decreased viscosity allows a larger puddle of oil to accumulate and subsequently pass through the ring gap when the cylinder pressure falls below the inter-ring pressure, again allowing more oil into the combustion chamber.

2) Total particulate emission rate increases with an increased ring gap width. This result is also derived from the ring-gap reverse flow theory of oil consumption, whereby oil passes through the ring gap and is consumed when the cylinder pressure falls below the inter-ring pressure. The results of this study serve to further confirm the presence of this oil transport mechanism and the significant effect it has on oil consumption and emissions.

3) At hotter conditions, the extra oil present in the combustion chamber due to changes in a lubricant parameter is mostly partially oxidized and emitted as carbonaceous material. At cooler conditions, most of this extra oil remains in some hydrocarbon form and is

eventually adsorbed on the particulate, thereby contributing to the oil derived soluble organic fraction of the particulates.

4) Changes in lubricant parameters affect not only the oil-derived emissions, but fuel-derived emissions as well. Through the process of fuel absorption and desorption, increased oil on the liner prevents additional fuel from participating in the combustion event, thereby increasing the fuel derived portion of the soluble organic fraction of the particulates.

5) Engine load has an effect on particulate emission rate and composition. At higher loads, the soluble organic fraction is reduced. The greatest total particulate rate is at low load, while the medium load condition created the lowest rate. This agrees with traditional findings.

6) Simultaneous measurements of the oil distribution in the piston ring-pack, using methods such as the laser-induced-fluorescence, and real-time oil consumption as well as particulate measurements will further elucidate the mechanisms by which lubricant contributes to the total particulate emissions and the opportunities for improvement.

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REFERENCES

1. Johnson, J. H., Bagely, S. T., Gratz, L. D., and Leddy, D. G., "A Review of Diesel Particulate Control Technology and Emissions Effects - 1992 Horning Memorial Award Lecture," SAE Paper 940233, 1994.
2. Heywood, J. B., Internal Combustion Engine Fundamentals, McGraw-Hill Book Company, New York, 1988.
3. Walsh, M. P., "Global Trends in Diesel Particulate Control," SAE Paper 930126, 1993.
4. Cartellieri W.P., et al, "Report on a Preliminary Survey of Strategies to Meet US-1991 HD Diesel Emission Standards Without Exhaust Gas Aftertreatment," SAE Paper 870342.
5. Dowling, M., "The Impact of Oil Formulation on Emissions from Diesel Engines," SAE Paper 922198.
6. Cooke, V.B., "Lubrication of Low Emissions Diesel Engines Part 1 and Part 2," SAE Paper 900814.
7. Cartellieri, W.P., et al. "Particulate Analysis of Light Duty Diesel Engines (IDI & DI) With Particular Reference to the Lube Oil Particulate Fraction," SAE Paper 840418.
8. Andrews, G. E., Abdelhalim, S., and P. T. Williams, "The Influence of Oil Age on Emissions from an IDI Diesel," SAE Paper 931003
9. Mayer, W. J., Lechman, D. C., and Hilden, D. L., "The Contribution of Oil to Diesel Particulate Emissions," SAE Paper 800256, 1980.
10. Essig, G., Kamp, H., Wacker, E., "Diesel Engine Emissions Reduction - The Benefits of Low Oil Consumption Design," SAE Paper 900591, 1990.
11. Jakobs, R. J. and Westbrooke, K., "Aspects of Influencing Oil Consumption in Diesel Engines for Low Emissions," SAE Paper 900587, 1990.
12. Zelenka, P., Kriegler, W., Herzog, P. L., and Cartellieri, W. P., "Ways Toward the Clean Heavy-Duty Diesel," SAE Paper 900602, 1990.
13. Richards, R.R., et al, "Diesel Engine Emissions Control for the 1990's," SAE Paper 880346, 1988.
14. Deutsch, E. J., "Piston Ring Friction Analysis from Oil Film Thickness Measurements," Masters Thesis, Department of Mechanical Engineering, Massachusetts Institute of Technology, 1994.
15. Kawatani, T., et al., "Technology for Meeting the 1994 USA Exhaust Emission Regulations on Heavy-Duty Diesel Engine," SAE Paper 932654, 1993.
16. Hoult, D. P. and Shaw, B. T., "The Puddle Theory of Oil Consumption," Tribology Transactions, Vol 34,1, 1994.
17. The Ricardo/Cussons Standard Hydra Engine and Test Bed - Instruction Manual, G. Cussons, Ltd., Manchester, England.
18. Wong, V. W., Yu, M. L., Mogaka, Z. N., and Shahed, S. M., "Effects of Catalytic Wire-Mesh Traps on the Level and Measurement of Heavy-Duty Diesel Particulate Emissions," SAE Paper 840172, 1984.
19. EPA, Code of Federal Regulations(CFR 40), 1990.
20. Ortech Proposal # 94-E14-7123/FI, Ortech Corporation, Mississauga, Ontario, 1994.
21. Hartman, R. M., "Tritium Method Oil Consumption and Its Relation to Oil Film Thickness in a Production

Diesel Engine," Masters Thesis, Department of Mechanical Engineering, Massachusetts Institute of Technology, 1990.

22. Norris, M., Hochgreb, S., "Novel Experiment on In-Cylinder Desorption of Fuel from Oil Layer," SAE Paper 941962, 1994.
23. Schneider, E. W. and Blossfeld, D. H., "Method for Measurement of Piston Ring Rotation in an Operating Engine," SAE Paper 900224, 1990.