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Characterization of the Lean Misfire Limit

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and B. D. Kelly**

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Characterization of the Lean Misfire Limit

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AUTOMOTIVE ENGINES ARE WELL KNOWN as major contributors to the air pollution problem. In an effort to reduce the exhaust emission of the three major pollutants, hydrocarbons, carbon monoxide, and oxides of nitrogen, much research in the field of lean burning engines has been conducted. Roughness or uneven combustion is a general characteristic of an internal combustion engine running at lean air-fuel mixtures. Roughness is said to produce poor driveability of an automotive engine; therefore, its reduction is highly desirable.

Uneven combustion, either among cylinders, or for consecutive cycles for a single cylinder, is the source of roughness. Cycle to cycle variations in combustion have been of particular interest to engine researchers. The change in combustion on a cycle to cycle basis includes flame speed, shape, and growth as well as total misfire. In many of the previous investigations as well as in this study, the combustion varia-

tions are measured using the combustion chamber pressure as an indicator.

RELATED RESEARCH

The bulk of research in the field has been concerned with finding the effect of various engine variables upon cycle to cycle variations in combustion, or "cyclic dispersion." In general, it has been shown that cyclic dispersion is dependent upon the homogeneity of the air-fuel mixture, turbulence at the time and in the vicinity of the ignition process, residual exhaust gas mixing, and various ignition parameters.

Soltau (1)* and Tanuma, et al. (2) have shown that the manner in which the air-fuel

*Numbers in parentheses designate References at end of paper.

ABSTRACT

Cycle by cycle variations in combustion are investigated as a means to characterize the lean misfire limit of a spark ignited, internal combustion engine. The cyclic variations are represented as parameters of the cylinder pressure versus crankangle curves. The specific parameters investigated are peak pressure, maximum rate of pressure rise, crankangle of maximum pressure, crankangle of maximum rate of rise, indicated mean effective pressure, and area of the pressure curve. The variations are statistically analyzed on a digital computer. Exhaust gas analysis is

performed to determine the correlation between the misfire limit and emissions of hydrocarbons, carbon dioxide, carbon monoxide, nitric oxide, and formaldehyde.

Imep and area variations are found to be the most promising as an indicator of the lean misfire limit. As the equivalence ratio decreases from stoichiometric, the variations stay constant, until a sharp linear increase is achieved. Correlation with hydrocarbon emissions shows a strong combustion change occurs at this point of rapid variation increase. This point is adopted as the definition of the lean misfire limit.

mixture is ignited has a significant effect upon cyclic dispersion. Reductions of combustion variations have been made through the use of multiple plug ignition, as well as increased spark plug gaps. Hotter or multiple sparks would have a greater chance of properly igniting a less than ideal air-fuel mixture.

The air-fuel homogeneity is an obvious source of cyclic dispersion. If the vaporization or concentration of the fuel in the mixture is erratic, excessively fuel lean zones may occur within the combustion chamber, which would prevent or impede combustion. Patterson (3), Hansel (4), and Quader (5) have shown significant reductions in variations when changing from liquid to gaseous fuels, as well as due to improvements in manifolding. In particular, Hansel reduced the pressure variations for a single cylinder by $1/3$, by changing to a gaseous fuel in a V-8 engine. The mixture homogeneity becomes of increasing importance as the air-fuel ratio is raised. This effect is also important with regard to the cylinder to cylinder variations in multi-cylinder engines.

Residual exhaust gas mixing has been shown to be a source of dispersion in the works of Soltau and Hansel. To test for the effect of exhaust dilution, zero residual exhaust conditions have been attained by not firing the spark plug for several cycles. This has the effect of fully scavenging the exhaust gas in the cylinder before the data cycle is ignited. Hansel reported a one third reduction in cyclic variation, when the exhaust gas residual was eliminated.

It should be noted that Patterson (3) found no significant effect in eliminating exhaust gas residual. Part of this discrepancy is due to a difference in the definition of cyclic dispersion.

A great variety of work has been accomplished in the investigations of turbulence in the combustion chamber and its association with cyclic dispersion. For any combustion process, a turbulent flow field will increase the rate of combustion and the flame speed. Turbulence is particularly important in the development of the flame front during and shortly after ignition. A flame kernel is the immediate product of ignition. Small scale turbulence can easily affect the kernel and its growth. This in turn is said to have significant effect upon the establishment of the flame front. Turbulence and its relation to cyclic dispersion are difficult to investigate due to both lack of control over the flow field and difficulties in measurement.

Tanuma, et al. (2) recorded the reduction in cyclic variation through the use of shrouded valve seats in an engine. The air-fuel mixture was swirled and turbulated as it entered the combustion chamber through the special valve sets. It was hypothesized that the improve-

ment was due to the increase in flame speed, as well as a more homogeneous mixture of air, fuel, and residual exhaust gases.

Hansel (4) deduced the amount of cyclic dispersion due to turbulence. He had reduced variation by one third due to air-fuel mixture homogeneity and found a further reduction of one third due to exhaust gas mixing. Therefore, he assumed the last one third was due to turbulence effects. Hansel (6) also hypothesized the source of cyclic dispersion as being due to interactions between the flow field, the air-fuel mixture, and the formation of the flame front. As the flame kernel emerges from the spark plug gap, it is affected by the turbulence and the local air-fuel ratio. The transition from the kernel to the established flame front will be fast or slow depending upon these conditions of ignition. This in turn will vary the fully established flame front, in terms of speed and possibly shape. Because the local air-fuel ratio and turbulence in the vicinity of the spark plug can vary from cycle to cycle, cyclic dispersion is observed.

Patterson (3) also investigated the effect of turbulence through the use of shrouded valves. His tests indicated that the actual variation (in terms of rate of pressure rise) did not change, but the variation based upon a percentage basis showed a reduction. The increased turbulence generated by the shrouded valves causes an increase in the burning rate and a corresponding increase in the rate of pressure rise. It was hypothesized that the turbulence and swirl in the combustion chamber distorted and wrinkled the flame front. This would increase the burning rate according to the degree of turbulence. He also stated that the effect would be largest or most effective immediately after ignition.

Due to difficulties in the control and measurement of turbulence in the combustion chamber, experiments using combustion bombs have been run. Advantages of bomb experiments are that the amount of turbulence can be controlled and hot wire anemometry methods can be calibrated and used for these systems.

Bolt and Harrington (7) used a rotating vane to generate swirl in a constant volume bomb, a conventional spark plug to ignite the mixture, and hot wire anemometry to measure turbulence and flow. Their tests show the rate of pressure rise to be greatly influenced by the swirl velocity due to the increase in the flame propagation rate. These effects become more pronounced as the fuel mixtures are leaned out.

Similar bomb experiments conducted by Cole and Mirsky (8) used a jet of air-fuel mixture to generate turbulence at the point of spark ignition. The experiment was directed at finding the effect of large scale turbulence upon the developing flame kernel. Like the

Fig. 1 - Typical pressure versus crankangle curve

previous experiment, they found that an increase of turbulence caused an increase in the burning rate and a corresponding increase in the rate of pressure rise. The increase was directly related to the distortion of the flame front area. They concluded that cycle by cycle variations could be reduced by reducing the mixture motion variation in the vicinity of the spark plug.

Later research has attempted to model the combustion process through the measurement of the turbulence which occurs in an engine. The experiments of Barton, et al. (9) as well as Winsor and Patterson (10) have reinforced the conclusions of earlier studies and have extended results through the reporting of mean burning times and velocities.

Lancaster, et al. (11) used a two-zone model in the analysis of the combustion in a cylinder. The experimental apparatus used shrouded and non-shrouded valves in a CFR engine. The ignition process and flame development were found to be one third of the combustion duration. The fully developed flame speed ratio, defined as the turbulent flame speed divided by the laminar flame speed, was found to be proportional to the turbulent intensity.

- In a recent paper, Quader (12) has focused on two different physical mechanisms leading to lean misfire, failure to ignite and poor flame propagation. He points out the importance of the ignition variables on these two operational limits. At the lean misfire limit both phenomena appear to be important.

LEAN MISFIRE LIMIT

As the air-fuel mixture becomes leaner, eventually some of the combustion variations will appear as total misfire. For these cases, the local air-fuel mixture is too low or the turbulent intensity too high for combustion

to be initiated and/or sustained. In effect, the lean flammability limit is surpassed in these cycles. For the engine as a whole, a lean limit cannot be defined in a classical combustion sense. Under stable engine operation with occasional misfire, there is only one air-fuel ratio of the engine. Yet some cycles have reached their flammability limit while others can easily sustain combustion. In view of this, there is a clear problem in the definition of a lean misfire limit in an engine.

A precise definition and determination of a lean misfire limit would be useful in the evaluation of lean burning combustion engines and research to reduce roughness. As variations are reduced, the misfire limit should be extended. Only a few studies have used the concept of lean misfire limit and these were based upon arbitrary definitions. Tanuma, et al. (2) used the air-fuel ratio at which the first misfires started as a definition for an engine misfire limit. Quader (5) used the point at which 0.5-0.8 percent of the cycles were total misfires. He describes this percentage of misfire as being unacceptable for normal driving. In the discussion of Quader's paper by Ryan and Lestz (5) the validity of this definition with regard to emissions evaluation is raised. Ryan, and Lestz suggest a better definition may be based upon the peak pressure of a typical cycle at the air-fuel ratio corresponding to the minimum specific fuel consumption. The reasoning behind this definition is that it would not consider any cycles whose combustion was poor enough to increase hydrocarbon emissions as well as specific fuel consumption. Quader notes that a lean misfire limit so defined would be difficult to measure. He also points to the need for a better definition of lean misfire limit. Ryan, et al. (13) used a misfire frequency of 5% to define the lean misfire limit. Harrington (14) used a misfire frequency of 0.5 per minute for the engine running at 1500 rpm (0.067%) as the lean misfire limit in his work.

The lean misfire limit should reflect the abilities of the air-fuel mixture to be burned. Therefore, it would seem reasonable to base its definition upon a parameter which closely follows the combustion process. One parameter which is relatively easy to monitor is the pressure versus time curve. The previous definitions of lean misfire limit indirectly use the pressure trace as an indicator, except they look for instances where the pressure trace is exactly that of a motored (no combustion) engine. It would be reasonable that the lean misfire limit, or some characteristic which could define a lean misfire limit, could be found in the form of cycle by cycle variation in combustion.

A typical pressure versus crankangle curve for a single cycle is shown in Figure 1. There are a number of different parameters

Fig. 2 - Variations in consecutive pressure traces

which could be used as a characteristic for the curve. The most obvious is the maximum pressure and, associated with it, the crank-angle at which maximum pressure occurs. Another is the maximum rate of pressure rise and its crankangle of occurrence. The area under the pressure versus crankangle curve can also be used as a characteristic. Typical variations for consecutive cycles are shown on Figure 2. The variation of all five parameters are noticeable. Lastly, the indicated mean effective pressure can be derived from the crankangle and corresponding pressures and is closely related to the area of the pressure-crankangle curve.

There has been little common use of parameters to describe cyclic dispersion. Soltau (1) and Hansel (4) used the difference of the highest and lowest maximum pressures, divided by the average maximum pressure, for a series of cycles. Karim (15) apparently was the first to report a "coefficient of variation," which is the standard deviation divided by the average maximum pressure. Other studies have simply used the difference of highest and lowest maximum pressures. Research has shown (Allwood (16), Chen and Krieger (17)) that the maximum pressure distributions are essentially normal, thereby validating the use of standard deviation in the measurement of cyclic variations. Most of the work in cyclic dispersion has been concerned with variations associated with the peak pressure. The general reason for its use is simply that it is the most obvious and easiest to measure.

Among the first to use the rate of pressure rise as an indicator of cyclic dispersion was Patterson (3). Barton, et al. (18) attempted a cross correlation between the peak pressure, maximum rate of pressure rise, and their respective positions of occurrence. They concluded that there was a strong relation between the variations of peak pressure and maximum rate of rise.

DATA REDUCTION

Fig. 3 - Engine test facility

Winsor and Patterson (10) state that peak pressures reflect piston motion and the total mixture burned, and is a complex indication of the combustion process. A better indicator would be the rate of pressure rise, which directly reflects the combustion rate and piston velocity. The maximum rate of rise is generally located near top dead center.

Indicated mean effective pressure (imep) was investigated by Borman and Peters (19) as a measure of cycle by cycle variations. They found no correlation between imep and peak pressures and concluded the two were not simply related.

In the attempt to find a good definition of the lean misfire limit through the use of combustion variations, all six mentioned parameters were investigated. The method of measurement for variations used the standard deviation of these parameters, often normalized by the average value.

EXPERIMENTAL APPARATUS

The engine experimental apparatus was composed of two basic systems, data collection and data reduction. Figure 3 shows the engine test facility. A single cylinder, variable compression ratio CFR engine was used for all tests. A water cooled Kistler 601A piezoelectric pressure transducer was mounted in the engine combustion chamber through an unused spark plug opening. A Kistler charge amplifier converted the transducer output into a voltage signal which was then fed into a Phillips Analog 7 frequency modulated data tape recorder. Two magnetic timing pickups were used. One was mounted adjacent to the camshaft and indicated bottom dead center of the intake stroke. The other was mounted adjacent to the engine flywheel. Three hundred and sixty steel strips

Fig. 4 - Spark timing

were glued to the flywheel, each corresponding to one degree of crankangle. Using these two timing signals, the crankangle at any time could be determined. The magnetic timing signals were amplified and fed into the data tape recorder. The output of the recorder was connected to an oscilloscope to monitor the taping process.

For the recorded data reduction, amplification and filtering were performed. The data were then digitized and analyzed by a PDP computer.

A homogeneous air-fuel mixture was promoted through the use of a gaseous fuel and the insertion of a coarse metal wool packing to provide mixing. Mixture homogeneity was not checked during the course of testing. The natural gas flow rate was measured with a positive displacement meter and its composition determined with a Hewlett Packard 5750 gas chromatograph. The air supply was the high pressure house air and its flow rate was monitored using a standard ASME sharp edged orifice and manometer. All tests were conducted at wide open throttle. A pressure regulator adjusted the intake manifold to the pressure desired.

The engine speed was held constant by an electronic dynamometer control. Speed adjustments were made through adjusting the engine load to attain the desired speed. Brake torque was measured by the resisting force applied to a moment arm on the housing of the dynamometer. Ignition timing was adjusted by hand and checked with a timing light.

Exhaust gas analysis was performed for some engine tests. The gases detected were total hydrocarbons (FID), carbon monoxide (NDIR), carbon dioxide (NDIR), nitric oxide (NDIR), and aldehydes (chromotropic acid).

Table 1 - Engine Operating Conditions

Case	Timing	Data	ϕ
A	constant at MBT for $\phi=1$	pressure	.59 - 1.1
B	MBT	pressure	.56 - 1.1
C	MBT	pressure, aldehydes	.56 - 1.0
D	MBT	pressure, HC, CO, CO ₂ , NO	.51 - 1.0

For all cases: compression ratio = 8.0; engine speed = 900 rpm; fuel = natural gas; intake manifold pressure = 98 kPa.

EXPERIMENTAL PROCEDURE

To calculate equivalence ratios, the stoichiometric air fuel ratio of the fuel must be known. The molecular weight of the fuel is also required in the flow rate calculations. The possibility of varying compositions can arise by using natural gas as fuel. On a day of engine tests, a sample of natural gas was injected into the chromatograph and its composition determined. Although there were detectable changes in composition on a day to day basis, the molecular weight as well as the stoichiometric air-fuel ratio were found to be very constant, due to the large amount of methane in the natural gas. The mean stoichiometric air-fuel ratio was a constant 17.0 to one.

All tests were run at 900 rpm, with the valve timing as well as compression ratio held constant. The intake manifold pressure was also held at 98 kPa for all tests. A lower than atmospheric pressure was chosen because of the low pressure natural gas fuel system and to insure that a flow reversal was never allowed.

The engine was run for at least one hour before any data were taken. For one set of tests, Case A, the ignition timing was held constant at the point of the maximum brake torque (MBT) for the engine operating at the stoichiometric air-fuel ratio, Figure 4. For all other sets of tests, Cases B, C, and D, the timing was adjusted to MBT timing for each air-fuel ratio run. The difference between Cases B, C, and D may be attributed to the replacement of the ignition power supply for Case D. Engine operating conditions are summarized in Table 1. After the engine was stabilized at the desired operating point, the pressure signals and the two timing signals were recorded. The recorder was run at a tape speed of 76.2 cm/sec in an

FM mode so there was no low frequency limitation upon the input signals. Sufficient data were recorded to insure that 100 consecutive cycles were available for processing. A correlation between exhaust emissions and cycle by cycle variations was also attempted.

DATA ANALYSIS

After the engine testing was complete, the recorded data were analyzed with a PDP-7 computer. The data were played back at a tape speed of 2.4 cm/sec, which gives a 32 fold time expansion. This tape speed allowed an acceptable data rate for the computer. The data signals were first amplified and the pressure signal filtered to reduce noise. The filtering was tailored to insure that most of the high frequency noise was smoothed out and that no attenuation or phase shifting occurred. The data were then digitized and 100 consecutive cycles stored on the computer magnetic tape.

An assembly language program controlled the data digitization and storage. The pressures corresponding to each crankangle during the compression and power strokes (180° BTDC - 180° ATDC) were stored on the computer magnetic tape after each cycle was digitized. After 100 cycles were stored, the computer program would start the analysis of each individual cycle.

The program would retrieve one cycle and find or calculate the maximum pressure and its angle of occurrence, the maximum rate of pressure rise and its angle of occurrence, imep, and area under the curve. The rate of pressure rise was calculated with a Lagrangian differentiation formula. The indicated mean effective pressure and the area were summed over the power stroke (0° TDC - 180° ATDC). Imep can be calculated through the knowledge of the crankangle and its corresponding piston displacement. For convenience the area was summed over the same interval as imep, using a simple integration formula.

After these six characteristics of the curve were found for the cycle, the values as well as their squares were added to running totals. These are needed to calculate the standard deviation of each characteristic. This sequence was repeated for 100 cycles, then the standard deviations and averages were computed.

The pressure data position in the array gives the corresponding crankangles of occurrence, therefore checks in the program tested for poorly synchronized data. Incorrect synchronization could occur through errors in timing signals during the digitization process. These checks prevented errors in the two positional data as well as integrations over incorrect intervals for imep and area.

In early runs, the computer calculations gave some negative values for imep or area, and upon consideration of their definitions,

Fig. 5 - Effect of reference pressure correction upon indicated mean effective pressure

these are impossible cases. The problem was traced to a shifting reference point for the charge amplifier and piezoelectric pressure transducer unit.

The piezoelectric pressure transducer is a quartz crystal which produces a charge proportional to the pressure upon it. The charge amplifier converts the charge to a low impedance voltage. The reference point for the output voltage is set by the amplifier, by pressing the zero button when the transducer is subjected to the reference pressure. In this manner, "static" measurements may be made with the system, if the time between referencing and measurement is small enough. After longer periods of time the signal will begin to drift and seek its own reference.

Due to long warmups and long times to set up a desired operating point, the amplifier will have found an arbitrary equilibrium zero point. The data were corrected by viewing the pressure traces on an oscilloscope. The pressure at the time of bottom dead center during the intake strokes would be very close to the manifold pressure, because the intake process has just finished, and the piston has not yet begun to compress the fuel mixture. The pressure was taken to be equal to the manifold pressure (98 kPa). The difference between this and the arbitrary reference of the amplifier could be calculated and the data could be corrected. Justification for this correction can be seen in Figure 5. The indicated power when uncorrected falls below the brake power, which is impossible. The corrected indicated power is at a constant value above the brake power, which is the actual case for constant speed runs.

Fig. 6 - Variations of MP, MR, area and imep - Case A

Fig. 7 - Variations of PMP and PMR - Case A

EXPERIMENTAL RESULTS

Four complete data runs or cases were made. The equivalence ratio (ϕ) varied from 1.1 or 1.0 to the leanest operation possible, typically $\phi = 0.5 \sim 0.6$. The equivalence ratio is defined as the stoichiometric air-fuel ratio divided by the actual air-fuel ratio.

The first run, or Case A, was conducted to verify a preliminary test, using the same basic system. One modification was the change to a low pressure natural gas fuel as compared to the high pressure bottled methane. Little difference was thought to be encountered because natural gas is approximately 94% methane by volume. As in the preliminary test, the ignition timing was adjusted to MBT at the stoichiometric air-fuel ratio, and held constant for all other points. No exhaust gas analysis was performed on this test.

The cycle to cycle variations in combustion are plotted as normalized standard deviations for the peak pressure, maximum rate of pressure rise, imep, and area in Figure 6. Figure 7 shows the standard deviations of crankangle for the positions of peak pressure and maximum rate of rise. The reason for using only the standard deviation for positional data is obvious when considering the arbitrary nature of an average crankangle. The system used here is referenced to bottom dead center of the intake stroke, therefore the average values of crankangle would be about 180 degrees for both position data. This is an arbitrary reference and any change in the average value will simply distort the true deviation. Although with the system reference to 180° BTDC, the changes in average crankangles are small, the distortion effect would be extreme if the reference were chosen to be 0° TDC.

To condense terminology, the maximum pressure, the maximum rate of pressure rise, the position of maximum pressure, and the position of the maximum rate of pressure rise will be referred to as MP, MR, PMP, and PMR, respectively. Towards the lean equivalence ratios, there is a sharp drop in the variations for MP, MR, and their respective positions. This is thought to be due to an increasing number of total misfires as the air-fuel mixture is leaned out. These misfires would have the same motored pressure trace, and would therefore reduce the total variations. All four of these curves show a maximum variation in the vicinity of $\phi = 0.7-0.8$, with minima at both extreme ends. The minima near stoichiometric equivalence ratio are reasonable, as one would expect optimum conditions for combustion to produce consistent burning.

If the curves were extended into the fuel rich zone ($\phi > 1$), one would expect to get a similar pattern of variation as with the lean case. As the air-fuel mixture became rich, the variations would increase, and finally decline as the misfires began to predominate. This region was not investigated.

There seems to be a correspondence between MP and PMP as well as between MR and PMR. The MR curve reaches a peak at a richer equivalence ratio than the MP curve. The same holds true for PMR and PMP curves. More importantly, the shift between the peaks of the curves is 0.08 equivalence ratio numbers for each pair.

In contrast to these four curves, the imep and area curves have steadily increasing values of variation as the equivalence ratio becomes lean. Despite the similarity of imep and area, the curves are not exactly the same, as the imep curve rises at a steeper rate at a low equi-

Fig. 8 - Variations of MP, MR, area, and imep - Case B

valence ratio. This difference is due to the difference in integration and weighting of the pressure traces. By definition, imep would tend to reduce the effect of variation near top dead center, while the area is an equally time-weighted integration. The imep curve would therefore be less responsive to variations occurring near TDC, which are changes near the peak pressure. Variations in the vicinity of the peak pressure will exist for any air-fuel ratio; however, variations based upon imep would tend to ignore this effect and give an indication of changes affecting the whole pressure trace. In terms of combustion, imep would give a better indication of large scale variations, rather than the small scale. This comparison between imep and MP tends to confirm and possibly explain the observation of Peters and Borman (19) that the two were not simply related.

In considering some characteristic variations for a lean misfire limit definition, a sharp increase or possibly a narrow peak is desirable. No such characteristic for the Case A tests was found. One factor which could affect the results is the constant ignition timing which is set for stoichiometric MBT. This timing is clearly not optimum for lean air-fuel ratios, where longer times are required to establish and completely burn the mixture. The need to adjust to timing for MBT for each equivalence ratio was recognized and implemented in the next data set. Based on the work of Quader (12), lean misfire in this case would be the result of poor flame propagation.

Case B was run exactly as Case A, but MBT timing employed for each point. The timing was adjusted after the desired equivalence ratio was established. Again, no exhaust gas analysis was attempted.

Fig. 9 - Variations of PMP and PMR - Case B

Figures 8 and 9 show the variations for Case B. Differences between Cases A and B are evident; in general, there is a reduction of variation for Case B for the range of $\phi = 0.7-0.9$. For this region, the maxima of MP, PMP, MR, and PMR in Case A are replaced by linear or constant curves. The maxima are now located at or near the leanest possible running conditions.

The MR, PMR, and PMP variation curves now resemble the imep and area curves. The maximum variations of MR and MP are higher for Case B as compared to Case A, while for PMP and PMR, the maximum variations are the same. The MBT timing has shifted the position of maximum variation to a leaner equivalence ratio in all four characteristics. The leanest possible running conditions (no brake power) have been extended to $\phi = 0.56$.

The imep and area curves are similar for the two cases; however, the second case curves have a sharper increase in variations at lean equivalence ratios. The highest amount of variation has also increased by a third, again at the leanest possible air-fuel ratio. The variations for two curves are also more constant between $\phi = 0.7-1.1$.

General characteristics of all six curves of Case B as compared to Case A, are that the values of variation are uniform near stoichiometric air-fuel ratios and they have a sharper increase in variation as the air-fuel mixture becomes lean. This may be explained by the improved conditions for flame propagation which should exist in Case B. With optimum ignition timing, less variation from poor combustion and misfires will occur at any given equivalence ratio. The effect would be greatest at points far from stoichiometric, because the difference between the optimum and the constant timing used in Case A is large. The sharp

Fig. 10 - Formaldehyde emissions - Case C

increase in variations is assigned to the onset of misfire. The most promising curves to use as indicators of the lean misfire limit are PMP, imep, and area variations. On the basis of these experiments, a lean misfire limit of $\phi \approx .65$ is assigned.

A third set of data, Case C, duplicated the test conditions for Case B, but aldehyde emissions were measured. The curves for the variations of this run are nearly identical to Case B and are not plotted. A graph for the concentration of formaldehyde in the exhaust gas is shown in Figure 10.

The data scatter for the aldehyde concentration is large. The aldehyde data based upon a mass per (brake or indicated) power are also shown. The indicated power is easily calculated from the imep, while the brake power is derived from the dynamometric measurement. The trend of these curves compares favorably to engine aldehyde data taken when the system was built. The increase in aldehyde emissions of $\phi = 0.6$ is not considered to be a significant indicator of the lean misfire limit. The increase is influenced by the decrease in power rather than increase in concentration of aldehydes, especially for the brake power representation.

One reason for the large scatter in the data is that the sampling system was designed for engines whose exhaust aldehyde concentrations were much higher than this engine. This difference in concentration is probably due to the types of fuel used. If one ignores the scatter, the concentration of aldehyde does not seem to have any significant increase at any equivalence ratio. Therefore, even if the sampling system were improved, there would be little correlation with lean misfire limit.

The fourth case was intended to be a repeat of the test procedure as in Cases B and C, but with exhaust sampling for hydrocarbons,

Fig. 11 - Variations of MP, MR, area, and imep - Case D

Fig. 12 - Variations of PMP and PMR - Case D

carbon dioxide, carbon monoxide, and nitric oxide. As testing was started, misfire was audible at air-fuel ratios which were previously suitable for good combustion. The problem was traced to a defective power supply which supplied the ignition system. An alternate battery source was substituted. If the power supply were progressively worsening, the earlier cases may have been affected.

The variation data for Case D is shown in Figures 11 and 12. The testing procedure was the same as in Case B, with the exception of the ignition system. Comparisons between Cases B and D show that there were some power supply related differences but that the trends are similar.

One difference from Case B is in the MR curve, which now resembles the MP curve with a maximum which does not occur at the leanest

Fig. 13 - HC, CO, CO₂, and NO emissions - Case D

equivalence ratio. The maximum of these curves seems to occur at the same equivalence ratio. MP and MR show increases near $\phi \approx 0.6$ indicating a slight extension of the lean misfire limit with the second ignition power supply.

There is now a great similarity between PMP, PMR, imep, and area variations. As the equivalence ratio is leaned out, the variations remain virtually constant until $\phi \approx 0.6$ where linear increases in variation occur. Imep has the strongest increase or slope for $\phi < 0.6$, and a very constant variation for $0.6 < \phi < 1.0$. This is followed by area, PMR and PMP respectively. PMR has a stronger slope at $\phi < 0.6$, but its scatter is greater than PMP. The scatter in PMR and MR is greater than any other variation characteristic, due to the pronounced effect of random noise upon the calculation of rates of pressure rise and, subsequently, its position.

The emissions of HC, CO, CO₂, and NO are shown in Figure 13. All are plotted as a volume percent, or mole fraction concentrations, rather than on a mass/power basis, in order to prevent exaggeration of emissions at low power levels. Emission data based upon mass/power units are considered inappropriate for this application of correlation with the lean misfire limit.

The emissions of HC, CO, and NO show a significant change near $\phi = 0.6$. These changes in the emissions reflect a serious change in the combustion process. This is reflected in the increase of variations of combustion as represented by the various pressure data. Of the three, hydrocarbons show the largest change. The hydrocarbon emission appears to be constant for $0.63 < \phi < 1.0$, on the scales shown. At the leanest equivalence ratios, incomplete combustion due to misfire produces large levels of HC. This is reflected, though in a smaller effect, in the CO emissions. At the very lean air-fuel ratios combustion temperature is so

low that very little NO is formed. As the equivalence ratio rises, better combustion is attained and NO levels rise.

None of the emissions data show a dramatic change like that of the variations of imep, but they confirm the location of a misfire limit. It is clear that variations of imep offer the best indication for a characteristic of the lean misfire limit. In reviewing Figure 11, the lean misfire limit would be located between $0.60 < \phi < 0.63$, which are the two closest points tested. A single point can be defined as a lean misfire limit by finding the intersection of the curve fit line for $\phi > 0.63$ and a curve fit line for $\phi < 0.60$. Both regions are quite linear in nature, and will fit with very little scatter. Extrapolating the lines on Figure 11 would give the lean limit equivalence ratio equal to 0.605.

An advantage in the use of this definition for the lean misfire limit is that the engine does not have to be operated at the exact equivalence ratio which defines the lean limit. Previous definitions for the lean misfire limit required the engine to be run until the occurrence of a certain frequency of misfires is observed. Here, the misfire limit can be determined without having to locate the precise limit at the time of experimentation.

CONCLUSIONS

As was found in previous investigations, cycle to cycle variations are easily affected by the ignition process. More importantly, proper timing is essential in the use of the variations to characterize the lean misfire limit. Improper timing tends to make the rapid changes in variations occurring at the lean misfire limit appear smooth or more gradual.

The position of maximum pressure, position of maximum rate of pressure rise, area, and imep all show good characterizations of the

lean misfire limit. Area and imep are the best indicators, with imep showing the sharpest change in variations. Imep shows a sharper change due to its insensitivity to small variations near top dead center and therefore gives a better indication of large scale variations.

The point where the imep variations shows a rapid increase, can be used as a definition for lean misfire limit.

Aldehydes, carbon dioxide, and nitric oxide emissions do not correlate with the lean misfire limit. Increases in hydrocarbons and carbon monoxide emissions indicate the point of the misfire limit. Hydrocarbons show the best correlation. Imep still seems to be the best indicator of the lean misfire limit even when compared to the hydrocarbon emissions data.

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