COMPARISON OF ENGINE PERFORMANCE WITH GASOLINE, A 10 PERCENT ETHANOL/GASOLINE BLEND AND 180 PROOF ETHANOL

by

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A Thesis Submitted to the Faculty of the

DEPARTMENT OF SOILS, WATER
AND ENGINEERING

In Partial Fulfillment of the Requirements
For the Degree of

MASTER OF SCIENCE
WITH A MAJOR IN AGRICULTURAL ENGINEERING

In the Graduate College

THE UNIVERSITY OF ARIZONA

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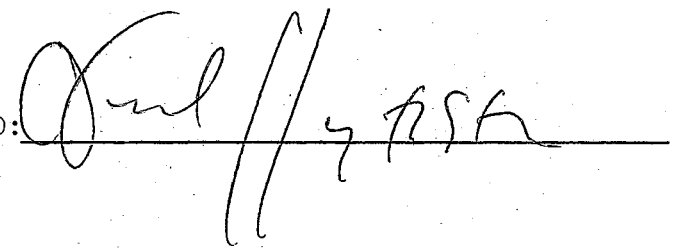
SIGNED: 

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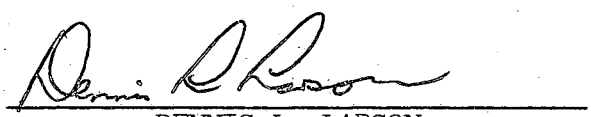

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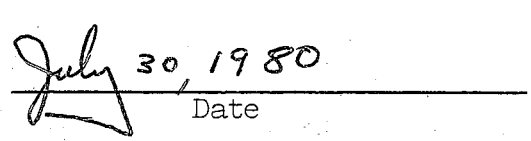

Date

Dedicated to

my wife, Julie

with love

ACKNOWLEDGMENTS

I would like to extend thanks and appreciation to Mr. Duane Magni of BASEF, Tempe, Arizona for financial support of this research, Dr. Dennis Larson for review and criticism of the manuscript, Mr. Barry Miller for his help in operation of the dynamometer and data collection, Dr. S. Chester of the Center for Quantitative Analysis, for technical advice on the statistical analysis, and Mrs. Evelyn Jorgensen for the many pages of typing.

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ABSTRACT

Using a spark ignition internal combustion engine, dynamometer tests were conducted to determine the comparative engine performance with (1) baseline unleaded gasoline, (2) a 10% by volume 200 proof anhydrous ethanol/90% baseline unleaded gasoline blend and (3) straight 180 proof ethanol. Power output, torque and fuel consumption were measured for each fuel and statistically analyzed. The ethanol blend and baseline gasoline were compared at identical carburetor and spark ignition timing settings. The straight ethanol fuel required an increase in main metering jet diameter and spark timing changes. The effect on research octane number by the addition of ethanol to the baseline gasoline also was determined.

Engine exhaust was analyzed using an infrared analyzer. Steady state exhaust levels of carbon monoxide and hydrocarbons were measured for each fuel.

When compared to the baseline gasoline, the ethanol/gasoline blend showed no statistically significant difference between power output, torque and fuel consumption. However, an increase in thermal efficiency was noted. The addition of ethanol to gasoline increased the research octane number.

Steady state emission results showed a reduction of carbon monoxide levels for the blend fuel. There was no significant change in hydrocarbon exhaust emissions.

The power output and torque using straight ethanol were similar to values obtained with the gasoline and ethanol/gasoline blend fuels. The fuel consumption with straight ethanol was considerably greater than with either gasoline or blend fuels. Ethanol emission analysis showed increased levels of hydrocarbon and reduced carbon monoxide emissions.

CHAPTER 1

INTRODUCTION

Alcohols have been considered as an internal combustion engine fuel since the early 1900's (Lucke and Woodward 1907). Researchers in the 1930's studied the effect of ethanol and ethanol/gasoline blends on engine performance (Miller 1934, Teodoro 1935). Agricultural products were used as feedstocks for industrial alcohol production of fuels and synthetic rubber during World War II. Post war alcohol distillation from farm crops decreased as a less expensive process using natural gas was developed (U.S. Dept. of Energy 1980). The use of alcohol as an engine fuel was almost non-existent during the 1950's and 1960's. Recent rising gasoline prices, together with the need for new markets for agricultural products, is renewing interest in ethanol as an engine fuel.

This study evaluated engine performance using unleaded gasoline, a 10% anhydrous ethanol/unleaded gasoline blend and 180 proof ethanol as fuels. Power output, torque and fuel consumption for each fuel were determined and statistically compared. Steady state engine exhaust emissions of carbon monoxide and hydrocarbons also were measured and statistically compared.

CHAPTER 2

LITERATURE REVIEW

Before making comparisons of unleaded gasoline and gasohol (10% 200 proof ethanol, 90% unleaded gasoline by volume) a review of engine behavior and exhaust emissions is necessary.

The three regulated exhaust emissions are:

1. Carbon Monoxide (CO)
2. Hydrocarbons (HC)
3. Oxides of Nitrogen (NO_x)

The relative mixture of fuel and air metered to the engine is a crucial factor in the formation of these regulated exhaust emissions. Depending upon the chemical composition of the fuel, there is a theoretically chemically correct or stoichiometric ratio of the fuel and air metered to provide enough oxygen (O_2) to convert all the carbon in the fuel to carbon dioxide (CO_2) and all of the hydrogen to water vapor (H_2O) without residual fuel or oxygen (Obert 1973, Taylor and Taylor 1961). Figure 1 shows the relative amounts of exhaust constituents, thermal efficiency and power as a function of fuel/air ratio (Ecklund 1978b, McCallum and Timbario 1978).

The amount of carbon monoxide is highly dependent upon fuel/air ratio. Mixtures richer than stoichiometric lead directly to higher carbon monoxide emissions. With a fuel rich (or oxygen deficient)

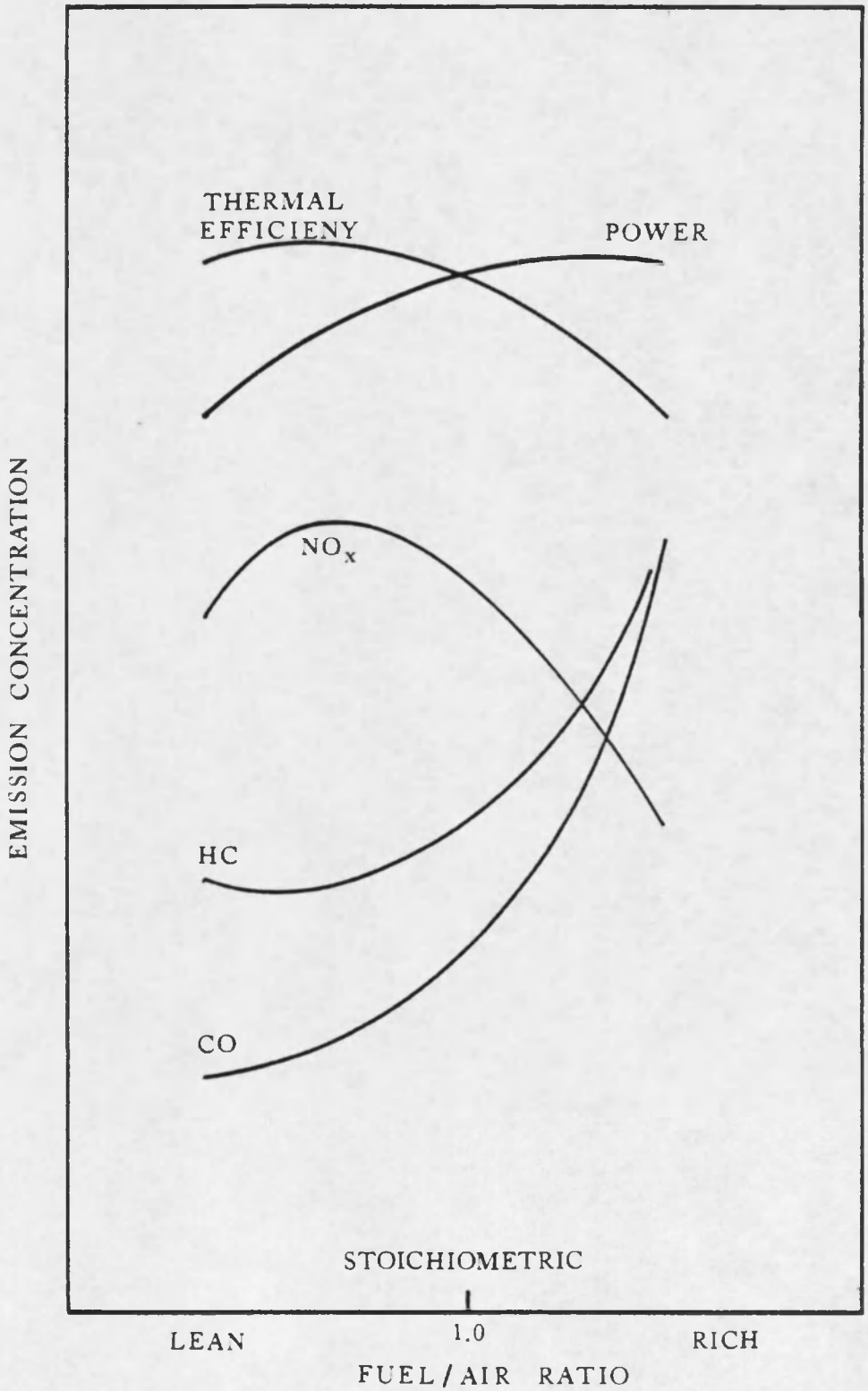

Figure 1. Efficiency, Power, and Emissions as a Function of Fuel/Air Ratio

mixture, some carbon atoms partially oxidize to carbon monoxide (CO). This partial oxidation process increases as the mixture enriches. There is always some carbon monoxide (CO) present in the exhaust gas because of fuel/air mixing problems and cylinder to cylinder charge distribution (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961). However, carbon monoxide emissions can be held at a minimum with engine operation at leaner than stoichiometric conditions.

Hydrocarbon (HC) exhaust emissions are mostly unburned fuel. An engine operating at a richer than stoichiometric fuel/air mixture cannot oxidize all the fuel particles completely, resulting in unburned hydrocarbons. The most important mechanism for hydrocarbon production is flame quenching (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961). As the high temperature combustion flame front approaches the cooler combustion chamber walls, heat flows from the flame to the wall. At some given point near the wall, enough heat is removed to put out or quench the flame. This quench layer covers the entire combustion chamber area and contains a substantial volume of hydrocarbons that never burn before the exhaust valve opens.

Hydrocarbon emissions decrease as the mixture becomes leaner (Fig. 1) until the lean misfire limit is reached where it then increases. If the fuel/air ratios are so lean that the combustion flame cannot be sustained or propagated, hydrocarbons as unburned fuel will escape through the exhaust valve (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961).

Figure 1 shows that hydrocarbon and carbon monoxide emissions reach minimum values at leaner than stoichiometric fuel/air ratios.

Fuel economy and thermal efficiency are also near maximum at leaner mixtures. This is because the engine is burning the fuel as completely as possible and extracting the maximum amount of chemical energy from the fuel (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961).

Oxide of nitrogen (NO_x) emissions result when nitrogen (N_2) reacts with oxygen (O_2) at temperatures above 1093°C (2000°F). The high temperatures necessary for oxide of nitrogen formation are easily reached during combustion. While high combustion temperatures help in reducing carbon monoxide and hydrocarbon emissions, the amount of nitrogen oxide increases (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961).

Power output is maximized at a slightly richer than stoichiometric fuel/air ratios. This results from high flame speeds and favorable combustion chamber environment. Further enrichment drops power due to incomplete combustion of the fuel (Ecklund 1978b, Obert 1973, Taylor and Taylor 1961).

Since different fuels have different stoichiometric fuel/air ratios, the raw value of fuel/air ratio does not give enough information for fuel comparisons. The parameter, equivalence ratio Φ is defined as the actual fuel/air ratio to the stoichiometric fuel/air ratio (Brinkman et al. 1975, Ecklund 1978b, Obert 1973, Taylor and Taylor 1961). For a stoichiometric mixture the equivalence ratio equals 1.0; for a fuel rich mixture the equivalence ratio is greater than 1.0; for a fuel lean mixture the equivalence ratio is less than 1.0.

Characteristics of Ethanol-Gasoline Blends

Because alcohols contain oxygen they have very different stoichiometric fuel/air ratios. When ethanol is added to gasoline, the physical qualities of the blend do not significantly differ from the physical qualities of gasoline. A carburetor would meter the blend fuel as it would meter gasoline (Brinkman et al. 1975, Ecklund 1978b, McCallum and Timbario 1978). However, a 10% by volume mixture of ethanol and gasoline does raise the stoichiometric fuel/air ratio (see Table 1). If the carburetor is calibrated for gasoline at an equivalence ratio of 1.0 and the ethanol-gasoline blend is substituted, the equivalence ratio is reduced (Ecklund 1978b).

$$\Phi = \frac{0.0677 \text{ (Actual F/A Ratio for Gasoline)}}{0.0707 \text{ (Stoichiometric F/A Ratio for Ethanol Blend}} = 0.958$$

This is called the blend leaning effect and causes the engine to run at slightly leaner mixtures. The exhaust emission response to the blend leaning effect depends upon how rich or lean the engine runs on straight gasoline.

The addition of 10% ethanol by volume to gasoline has the effect of increasing the octane number of the blend over that of the base gasoline (Allsup and Eccleston 1979, Cox 1979, Heitland et al. 1977, U.S.D.A. 1980, University of Nebraska Extension Service 1980). However, the percent increase in octane number is reduced as the quality of the base gasoline is improved.

Research on ethanol-gasoline blend water tolerance has shown that a 10% ethanol-gasoline blend will tolerate approximately 0.47%

Table 1. Fuel Properties

	Fuel		
	Unleaded Gasoline	10% Ethanol Blend	180 Proof Ethanol
Chemical Formula	CH_2	C_2H_5OH	$C_{6.8}H_{14.1}O_{0.26}$
Molecular Weight	14.03	46.07	99.8
Specific Gravity	0.7467	0.7515	0.7939
Stoichiometric F/A Ratio	0.0677	0.0707[1]	0.111
Heating Value			
Higher	47070 $\frac{kJ}{kg}$ (20240 $\frac{BTU}{lb_m}$)	45020 $\frac{kJ}{kg}$ (19360 $\frac{BTU}{lb_m}$)	29700 $\frac{kJ}{kg}$ (12770 $\frac{BTU}{lb}$)
Lower	44110 $\frac{kJ}{kg}$ (18970 $\frac{BTU}{lb_m}$)	41720 $\frac{kJ}{kg}$ (17940 $\frac{BTU}{lb_m}$)	27000 $\frac{kJ}{kg}$ (11610 $\frac{BTU}{lb_m}$)
Heat of Vaporization	349 $\frac{kJ}{kg}$ (150 $\frac{BTU}{lb_m}$)	437 $\frac{kJ}{kg}$ (188 $\frac{BTU}{lb_m}$)	921 $\frac{kJ}{kg}$ (396 $\frac{BTU}{lb_m}$)
Research Octane No.	92.2	96.1	106

1. See Appendix A for Calculation of Stoichiometric Fuel/Air Ratio.

water before separation occurs. A 25% ethanol-gasoline blend will tolerate about 1.0% (Cox 1979, McCallum and Timbario 1978). The use of cosolvents for ethanol-gasoline blends is being investigated (Cox 1979).

The effect of ethanol-gasoline blends under high altitude conditions was studied using Federal Test Procedures by Richardson 1978. The results indicate decreases in carbon monoxide emissions with slight increases in oxides of nitrogen. Hydrocarbon emissions, although erratic, decrease slightly. A slight reduction in fuel economy is noted.

Based on 1.9 million miles of driving experiences, the State of Nebraska with the University of Nebraska report reductions of 30% in carbon monoxide emissions with no significant change in hydrocarbon and nitrogen oxide emissions (Scheller 1977).

The University of Miami (1978) compared steady state exhaust emissions using indolene clear reference fuel and 20 volume percent ethanol-indolene blend. The following results were obtained:

A. The ethanol blend reduces oxides of nitrogen at near stoichiometric equivalence ratios.

B. Hydrocarbon levels with the ethanol blend are reduced from baseline indolene at any equivalence ratio.

C. Carbon monoxide emission levels show no significant difference between baseline and ethanol blend.

D. The ethanol blend shows a slight increase in torque over the baseline fuel.

Pure Ethanol as an Automotive Engine Fuel

Pure ethanol as an internal combustion engine fuel has characteristics that make it an attractive candidate as an automotive fuel. Pure ethanol has a high octane rating without requiring the addition of octane boosters. This feature allows the engine to be operated at higher compression ratios and therefore, higher thermal efficiencies (Obert 1973, Taylor and Taylor 1961). Volkswagen do Brasil's research and development program of pure ethanol engines has led to increasing the compression ratio from 7.2:1 to 11.1:1 with resulting increases in maximum power output and maximum torque of 18.5% and 20.0%, respectively (Heitland et al. 1977). These increases in power and torque are a result of better internal cooling, increased volumetric efficiency and improved combustion at higher compression ratios (Ecklund 1978b).

The lean flamability limit of pure ethanol allows internal combustion engines to operate at leaner equivalence ratios than gasoline without misfire (Allsup 1979, Ecklund 1978b). Because ethanol can be burned at lower equivalence ratios than gasoline, the effect upon regulated exhaust emissions is significant. Reduction of carbon monoxide and oxides of nitrogen will occur (U.S. Department of Energy 1978). Engines operated on straight ethanol have shown reduced oxides of nitrogen levels of 50% (Bechtold and Pullman 1979).

There are, however, inherent problems associated with the use of straight alcohols, including ethanol. Pure ethanol-fueled engines are difficult to start cold. The required rich fuel/air ratio needed for cold starting is difficult to attain. This is due to the low

vapor-liquid ratio of ethanol at low temperatures. Attempts to solve this problem have shown the addition of C_4 to C_7 hydrocarbons will lower the flamability temperature (Ecklund 1978a), allowing for easier starting. The addition of butane is effective but has adverse affects on volatility (Ecklund 1978a). Adequate cold starting can be provided by the addition of from 5% to 12% gasoline (Ecklund 1978a, Heitland et al. 1977). Researchers at Volkswagen do Brasil have developed a separate, 2 liter gasoline tank, electric fuel pump, solenoid valve, an injection nozzle and necessary hardware to inject straight gasoline into the induction system (Heitland et al. 1977).

The single boiling point characteristic of ethanol, which makes it difficult to evaporate at low temperatures, causes the additional problem of vapor lock at high temperatures by producing too much vapor in the liquid fuel system. This problem can be resolved by design changes such as insulation of fuel lines and larger fuel vapor return lines.

Due to the low calorific value of ethanol compared to gasoline (Bandel 1977, Ecklund 1978b, Obert 1973, Taylor and Taylor 1961), ethanol fuel flow-through must be increased by approximately 65% over gasoline to maintain torque and power levels. This requires carburetor modification which gives higher volumetric fuel consumption in terms of liters of fuel per kilowatt output, as compared with gasoline.

Experimental work performed at the University of Santa Clara analyzed steady state and Federal Test Procedure simulation exhaust emissions using straight ethanol compared to baseline indolene values

(Ecklund 1978a). The following trends were suggested:

A. Oxides of nitrogen were reduced from the baseline exhaust emission values, except at very rich or very lean equivalence ratios.

B. Unburned hydrocarbons with ethanol were very close to baseline values.

C. Carbon monoxide emissions with straight ethanol were reduced by 60% when compared to indolene at stoichiometric fuel/air ratios.

The results cited in the Literature Review demand careful interpretation, especially those results concerning values determined by different types of emission analyzer instrumentation and sampling procedures. For example, hydrocarbons analyzed with nondispersive infrared analysis techniques tend to undermeasure hydrocarbon content. Flame ionization offers greater sensitivity to hydrocarbons regardless of hydrocarbon size or structure (Obert 1973). Steady state, part throttle emission numbers are not comparable to the Federal Test Procedure (FTP) constant volume sampling (CVS) values, as the sampling techniques are entirely different.

CHAPTER 3

TESTING PROCEDURE AND ANALYSIS

Experimental Procedures

The Society of Automotive Engineers' Engine Test Code — Spark Ignition and Diesel — SAE J816b was followed for the engine dynamometer testing used in fuel comparisons. The engine dynamometer testing was done at wide-open throttle with the carburetor mixture needle adjusted for maximum power at 1500 RPM and the spark timing set for the highest torque at 900 RPM. The test engine was allowed to warm up until the oil and coolant temperatures stabilized. Fuel flow, duration of test, number of revolutions, beam load, coolant inlet and outlet temperatures, oil pressure and temperature, inlet air manometer level, barometric pressure, wet and dry bulb temperatures were recorded for each individual run. Sample data and calculations are shown in Appendix B.

The 10% ethanol blend and the baseline unleaded gasoline were compared at identical engine carburetor and spark ignition settings. The baseline unleaded gasoline and 200 proof anhydrous ethanol was used for all fuel blending.

To compensate for ethanol's low volumetric energy content, the carburetor main metering jet flow area had to be increased before any dynamometer runs were made with 180 proof ethanol. The carburetor mixture needle and spark ignition timing were adjusted to give maximum power at 1500 RPM and highest torque at 900 RPM respectively before tests were conducted.

The research octane number for the baseline unleaded gasoline and ethanol blend was determined by ASTM D 2699-79. The research octane number for 180 proof ethanol was cited from references.

The procedure used to measure exhaust emissions was not the same as the procedure used to determine power output. Steady state exhaust emissions were measured at a fixed engine speed of 1200 RPM. Carbon monoxide and hydrocarbons were each recorded at different carburetor fuel/air mixture settings and plotted versus equivalence ratio. Carbon monoxide was measured as a percent of total exhaust and hydrocarbons in parts per million.

Equipment

A four cylinder Case industrial engine connected to a General Electric direct current dynamometer was used for all fuel comparisons. All exhaust emissions were measured by a Sun Equipment infrared exhaust analyzer (see Appendix C for complete equipment description).

Statistical Analysis

Different regression models were fitted to the observed data. Linear, quadratic and logarithmic transformation models were all tested. The model that fit the observed data and had the simplest regression equation was chosen as the regression model. Common model type was selected for comparison of each characteristic, i.e. power, torque, etc. To test the fit of the regression model to the data, the sum of squares error was broken into two components (1) pure error component, and (2) lack of fit component. The pure error component was determined by using replications of observations and degrees of freedom to calculate

the sum of squares pure error and pure error mean square. The lack of fit component is the sum of squares error minus the sum of squares pure error. The F test statistic was used to determine the fit of the regression function. This test procedure is applicable no matter what form of regression function is being tested.

The method used for testing the equality of two regression functions consisted of these basic steps:

1. Fit the full, unrestricted model to observed data and obtain the sum of squares error SSE(F).

2. Fit the reduced model and determine the sum of squares error SSE(R).

3. Calculate the F statistic and test whether the two regression functions are equal.

The full model provides separate regressions for each data group to be compared. The error sum of squares for fitting the full model for the entire set of data is:

$$SSE(F) = SSE(1) + SSE(2)$$ where $SSE(1)$ = Sum of squares error for data set 1

$SSE(2)$ = Sum of squares error for data set 2

Fitting the reduced model implies fitting a single regression function to the combined data and determining the sum of squares error SSE(R). The value of SSE(R) will be greater than SSE(F).

The test for equality of regression functions compares SSE(R) and SSE(F). If the difference in SSE(R) and SSE(F) is small, the variation of the observations around the regression line for the full model is almost as great as the variation around the regression line for the reduced model. In this case the full model does not significantly reduce the variation in the dependent variable. If SSE(R) - SSE(F) is small, the evidence suggests that the intercepts and slopes of the separate regression functions are equal. If SSE(R) - SSE(F) is large, the full model helps to decrease the variation indicating that the regression functions are not the same (Neter and Wasserman 1974). Once the level of significance has been decided upon, the F statistic is used to test for equality.

The statistical procedure used to determine the fit of the regression model was first completed before any other analysis was done. After fitting the appropriate model, the regression functions were formally tested for equality. These procedures were followed for all observed data before any conclusions were drawn.

Regression equations and ANOVA tables for the performance data and exhaust emissions are shown in Tables 2-16. The regression equations for power, torque and fuel consumption are valid for the 800-1600 RPM range. The regression equations for exhaust emissions are valid for equivalence ratios from 1.0 to 1.7. Table 17 presents a sample lack of fit test of a regression model. Table 18 compares regression models for power output obtained with unleaded gasoline with the model for power obtained using ethanol blend. This test determines whether the models are significantly different.

Table 2. Regression Analysis of Power Output Using Unleaded Gasoline.

Regression Equation : HP = 4.65 + 0.014 RPM
Coefficient of Determination: 0.979

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	437.311	1	437.311
Error	9.277	33	0.281
TOTAL	446.588	34	437.592

Table 3. Regression Analysis of Power Output Using 10% by Volume Ethanol Blend.

Regression Equation : HP = 4.56 + 0.014 RPM
Coefficient of Determination: 0.986

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	490.252	1	490.252
Error	6.936	33	0.210
TOTAL	497.118	34	490.452

Table 4. Regression Analysis of Power Output Using 180 Proof Ethanol

Regression Equation : $HP = 9.96 + 0.0399\ RPM - 0.000011\ RPM^2$
Coefficient of Determination: 0.984

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	535.276	2	267.638
Error	8.505	43	0.198
TOTAL	543.781	45	267.836

Table 5. Regression Analysis of Engine Torque Using Unleaded Gasoline.

Regression Equation : TQ = 114.25 - 0.016 RPM
Coefficient of Determination: 0.862

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	571.436	1	571.436
Error	91.501	33	2.773
TOTAL	662.937	34	574.209

Table 6. Regression Analysis of Engine Torque Using 10% by Volume Ethanol Blend.

Regression Equation : TQ = 114.17 - 0.015 RPM
Coefficient of Determination: 0.866

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	577.924	1	577.924
Error	89.602	33	2.715
TOTAL	667.526	34	580.639

Table 7. Regression Analysis of Engine Torque Using 180 Proof Ethanol.

Regression Equation : $TQ = 69.67 + 0.067\ RPM - 0.000035\ RPM^2$
Coefficient of Determination: 0.851

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	879.930	2	439.965
Error	154.615	43	3.596
TOTAL	1034.545	45	443.561

Table 8. Regression Analysis of Fuel Consumption Using Unleaded Gasoline.

Regression Equation : $FC = 1.94 - 0.0018\ RPM + 0.00000061\ RPM^2$
Coefficient of Determination: 0.951

ANOVA TABLE

Souuce of Variation	SS	df	MS
Regression	0.19624	2	0.09812
Error	0.01010	32	0.00032
TOTAL	0.20634	34	0.09844

Table 9. Regression Analysis of Fuel Consumption Using 10% by Volume Ethanol Blend.

Regression Equation : $FC = 1.97 - 0.0018\ RPM + 0.00000063\ RPM^2$
Coefficient of Determination: 0.964

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	0.19803	2	0.09901
Error	0.00736	32	0.00023
TOTAL	0.20539	34	0.09924

Table 10. Regression Analysis of Fuel Consumption Using 180 Proof Ethanol.

Regression Equation : $FC = 3.40 - 0.0036\ RPM + 0.0000014\ RPM^2$
Coefficient of Determination: 0.851

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	0.30476	2	0.15238
Error	0.07228	43	0.00168
TOTAL	0.37704	45	0.15406

Table 11. Regression Analysis of Steady State Carbon Monoxide Emissions Using Unleaded Gasoline.

Regression Equation : $CO = 5.87 - 14.66\,\Phi + 8.76\,\Phi^2$
Coefficient of Determination: 0.998

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	14.841	2	7.421
Error	0.021	12	0.0017
TOTAL	14.862	14	7.4227

Table 12. Regression Analysis of Steady State Carbon Monoxide Emissions Using 10% by Volume Ethanol Blend.

Regression Equation : $\log_{10} CO = 0.88 + 8.38 \log_{10} \Phi$
Coefficient of Determination: 0.98

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	5.040	1	5.040
Error	0.099	16	0.006
TOTAL	5.139	17	5.046

Table 13. Regression Analysis of Steady State Carbon Monoxide Emissions Using 180 Proof Ethanol.

Regression Equation : $CO = 10.01 - 18.38\,\Phi + 8.43\,\Phi^2$
Coefficient of Determination: 0.986

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	17.173	2	8.587
Error	0.243	15	0.0162
TOTAL	1 .416	17	8.6032

Table 14. Regression Analysis of Steady State Hydrocarbon Emissions Using Unleaded Gasoline.

Regression Equation : $HC = 200.88 - 430.3\ \Phi + 269.40\ \Phi^2$
Coefficient of Determination: 0.982

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	44160.912	2	22080.456
Error	789.088	15	52.606
TOTAL	44950.000	17	22133.062

Table 15. Regression Analysis of Steady State Hydrocarbon Emissions Using 10% by Volume Ethanol Blend.

Regression Equation : $HC = 238.88 + 294.4\ \Phi - 27.83\ \Phi^2$
Coefficient of Determination: 0.986

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	14688.804	2	7344.402
Error	204.529	12	1 .044
TOTAL	14893.333	14	7361.446

Table 16. Regression Analysis of Steady State Hydrocarbon Emissions Using 180 Proof Ethanol.

Regression Equation : $HC = 1102.0 - 1\ 17.47\ \Phi + 588.46\ \Phi^2$
Coefficient of Determination: 0.919

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	20355.931	2	10177.965
Error	1794.069	15	119.605
TOTAL	22150.000	17	1029 .570

Table 17. Decomposition of Error Component for Power Output Using Unleaded Gasoline.[1]

Source of variation	SS	df	MS
Pure Error	0.789	5	0.158
Lack of Fit	8.488	28	0.303
TOTAL	9.277	33	

To test the fit of the regression model, the appropriate test statistic is:

$$F^* = \frac{MSLF}{MSPE} \qquad \text{where} \quad MSLF = \text{Lack of fit mean square}$$

$$MSPE = \text{Pure error mean square}$$

The hypothesis tested is whether the regression model fits the observed data. The comparative hypotheses are:

$$H_o: \; Y = B_0 + B_1 x$$

$$H_a: \; Y \neq B_0 + B_1 x$$

With the level of significance $\alpha = 0.05$, the decision rule is:

If $F^* \leq F(0.95, 5, 28)$, conclude H_o

If $F^* > F(0.95, 5, 28)$, conclude H_a

Calculations:

$$F^* = \frac{0.303}{0.158} = 1.92$$

$$F(0.95, 5, 28) = 2.53$$

Since $F^* \leq F$, conclude H_o, that the linear regression model fits the observed data. Further regression analysis is shown in Table 18.

---

1. See Table 2 for complete Regression Analysis and Anova Table.

Table 18. Regression Analysis of Combined Power Outputs for Baseline Unleaded Gasoline and Ethanol Blend Fuels - Reduced Model.

Regression Equation : Hp = 4.60 + 0.014 RPM
Coefficient of Determination: 0.982

ANOVA TABLE

Source of Variation	SS	df	MS
Regression	928.16	1	928.16
Error	16.50	68	0.24
TOTAL	944.66	69	928.40

To test the equality of regression models, the appropriate test statistic is:

$$F^* = \frac{\frac{SSE(R) - SSE(F)}{2}}{\frac{SSE(F)}{n_1 + n_2 - 4}}$$

where SSE(R) = Error sum of squares for reduced model

SSE(F) = Summation of error sum of squares for the two fitted regression functions [2]

n_1 = Number of observations for Regression Model 1

n_2 = Number of observations for Regression Model 2

2. See Tables 2 and 3 for complete Regression Analysis and Anova Table.

The hypothesis tested is whether or not the regression models are the same. The comparative hypotheses are:

H_o: $B_1 = B_2$ and $B_{11} = B_{12}$ (checking equality of intercepts and slopes respectively of each model)

H_a: $B_1 \neq B_2$ or $B_{11} \neq B_{12}$ or both

With the level of significance $\alpha = 0.05$, the decision rule is:

If $F^* \leq F(0.95, 66)$, conclude H_o

If $F^* > F(0.95, 66)$, conclude H_a

Calculations:

$$F^* = \frac{\frac{16.50 - 16.21}{2}}{\frac{16.21}{66}} = 0.590$$

$F(0.95, 2, 66) = 3.15$

Since $F^* \leq F$, conclude H_o, that the regression models are not significantly different.

CHAPTER 4

RESULTS AND CONCLUSIONS

Using a spark ignition internal combustion engine, dynamometer tests were conducted to determine the comparative performance with unleaded gasoline and a 10% by volume ethanol/90% unleaded gasoline and 180 proof ethanol. The baseline unleaded gasoline and 200 proof anhydrous ethanol were used for all fuel blending.

Engine power output, torque and fuel consumption were plotted versus engine RPM for each fuel tested. Figures 2 and 3 show data scatter and fitted regression curves for the baseline unleaded gasoline and the ethanol/gasoline blend fuel, respectively. Figure 4 shows data scatter and regression lines for 180 proof ethanol.

The fuel consumption and lower heating value of the fuel are used to calculate the enthalpy thermal efficiency (see sample calculations for example). Figure 5 shows thermal efficiency versus engine RPM for each fuel.

Exhaust emission levels are plotted against equivalence ratio for each fuel. Carbon monoxide emissions are measured as a percent of total exhaust. Hydrocarbon levels are measured in parts per million of exhaust gas. Figure 6 shows data scatter and fitted regression curves for carbon monoxide emission for each fuel. Figure 7 shows data scatter and fitted regression curves for hydrocarbon levels for each fuel tested.

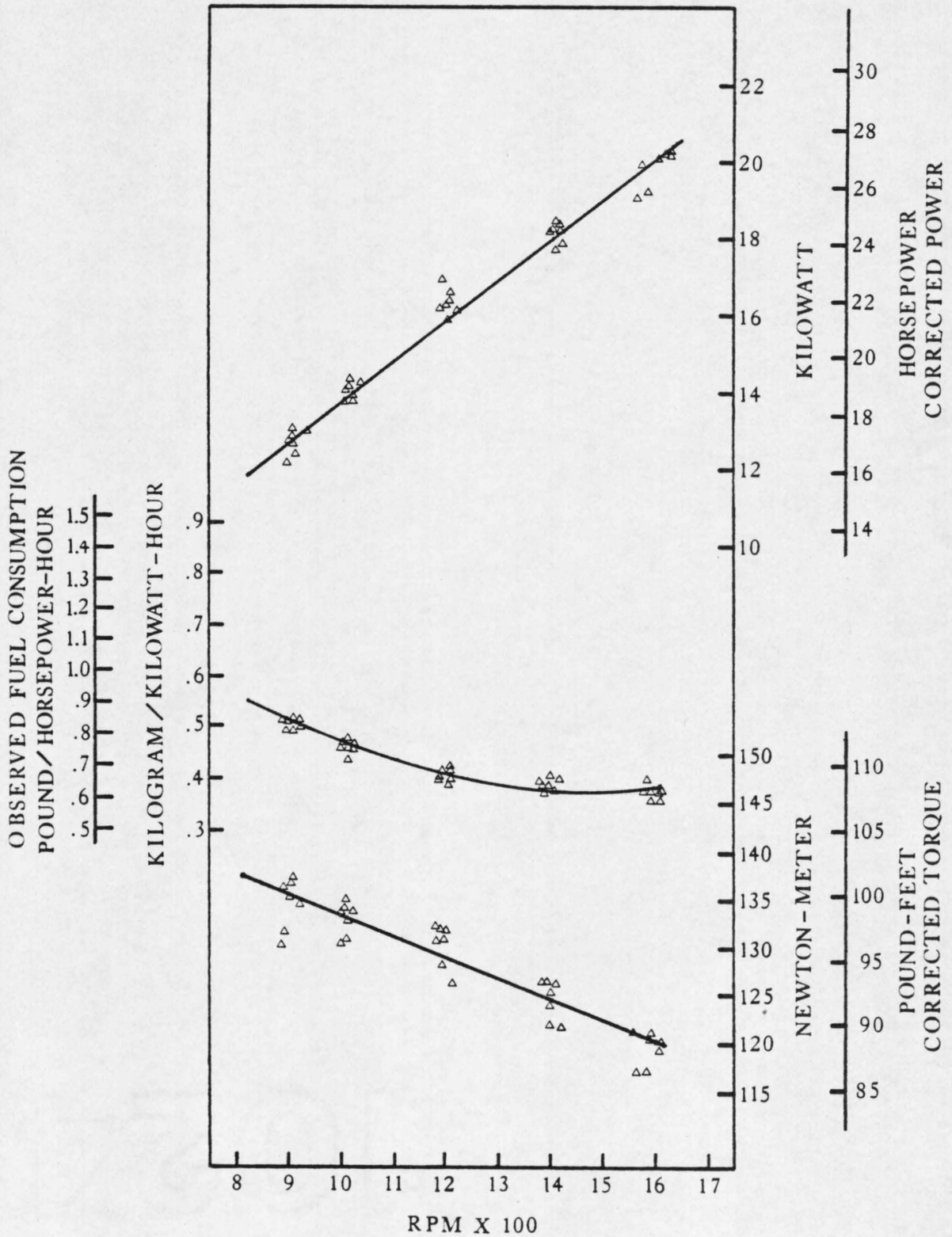

Figure 2. Engine Performance Curves for Unleaded Gasoline

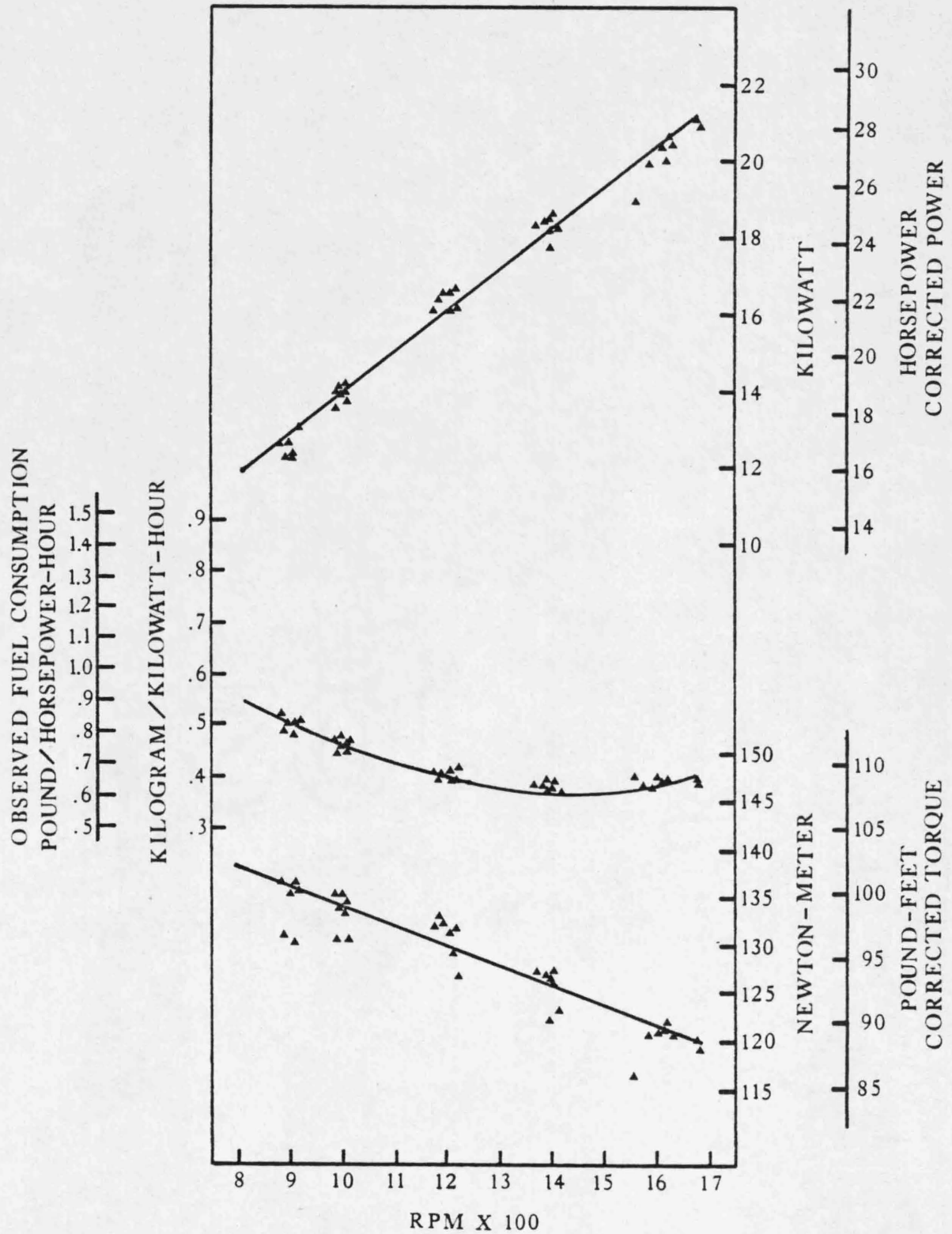

Figure 3. Engine Performance Curves for 10 Volume % Anhydrous Ethanol/ Unleaded Gasoline Blend.

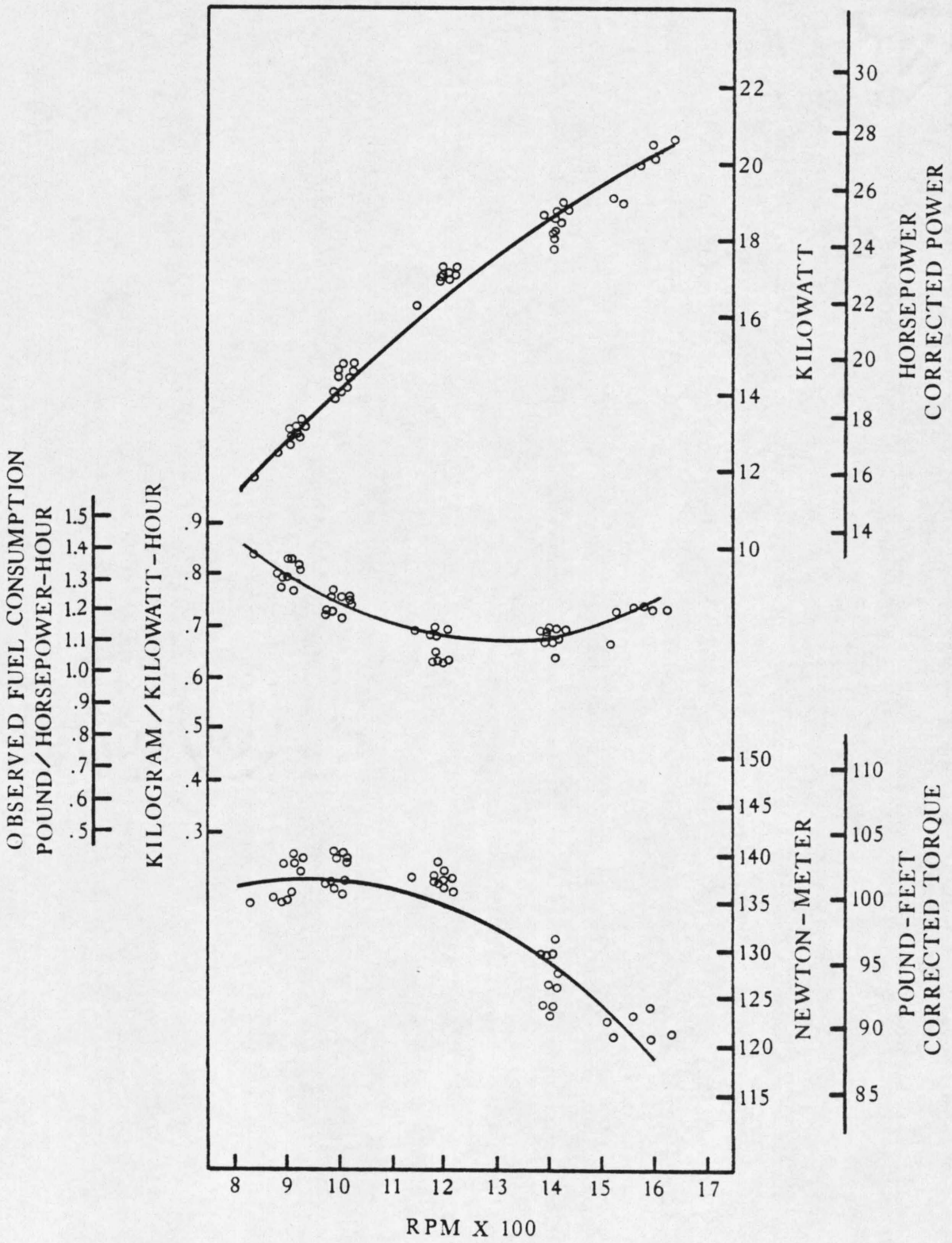

Figure 4. Engine Performance Curves for 180 Proof Ethanol

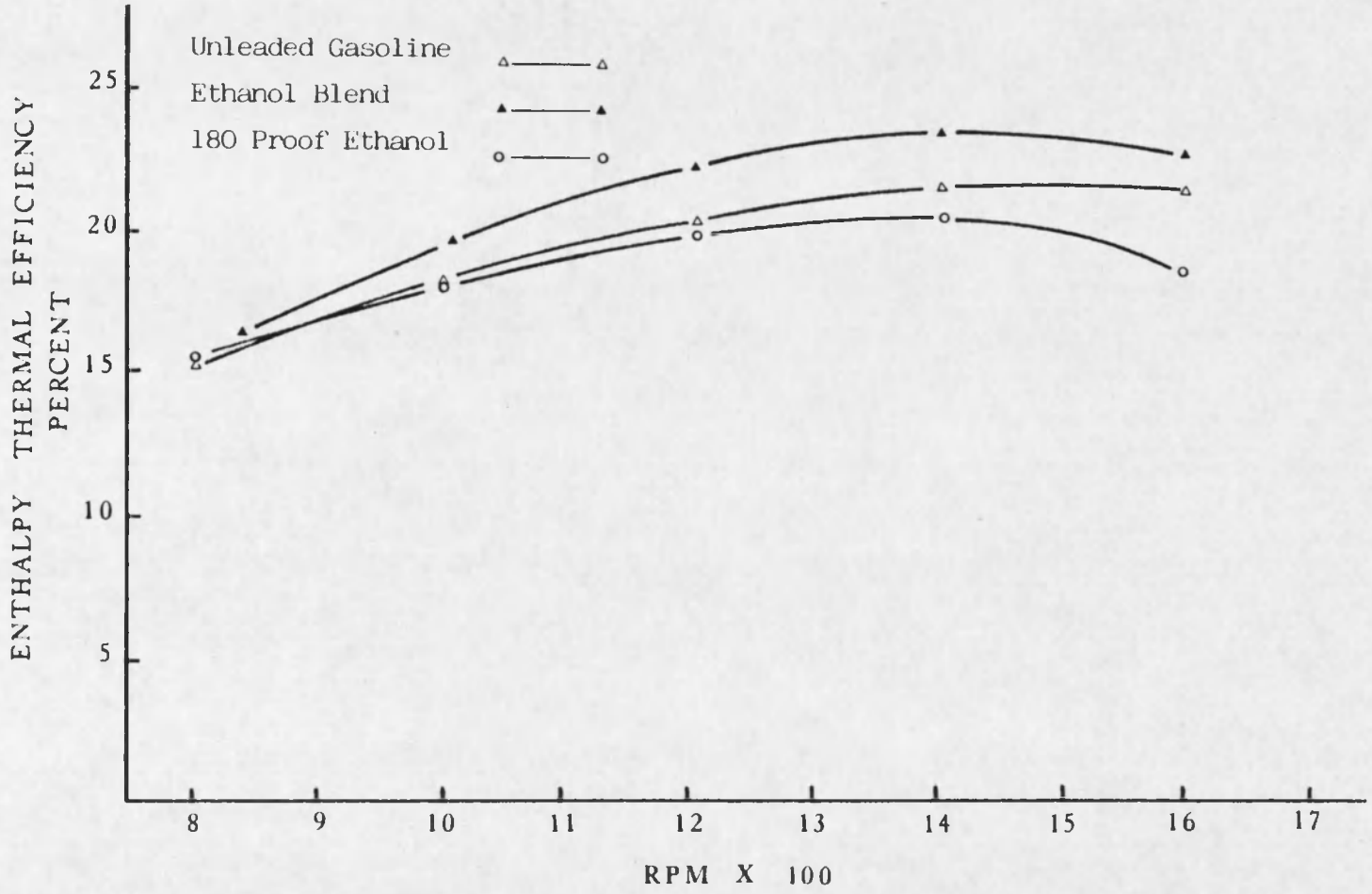

Figure 5. Enthalpy Thermal Efficiency vs RPM for Unleaded Gasoline, 10 Volume % Anhydrous Ethanol/Unleaded Gasoline Blend and 180 Proof Ethanol.

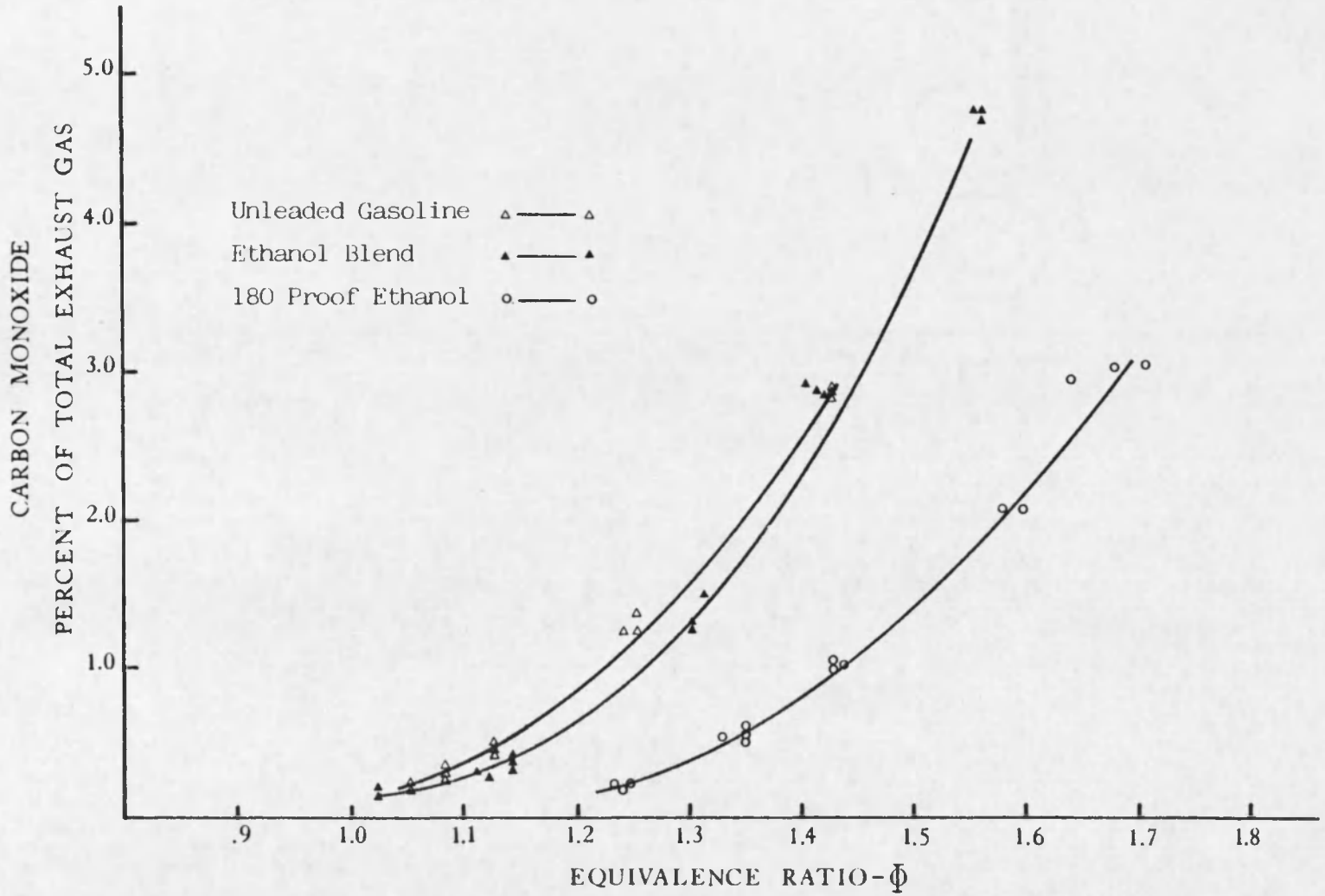

Figure 6. Carbon Monoxide Emissions vs. Equivalence Ratio for Unleaded Gasoline, 10 Volume % Anhydrous Ethanol/Unleaded Gasoline and 180 Proof Ethanol.

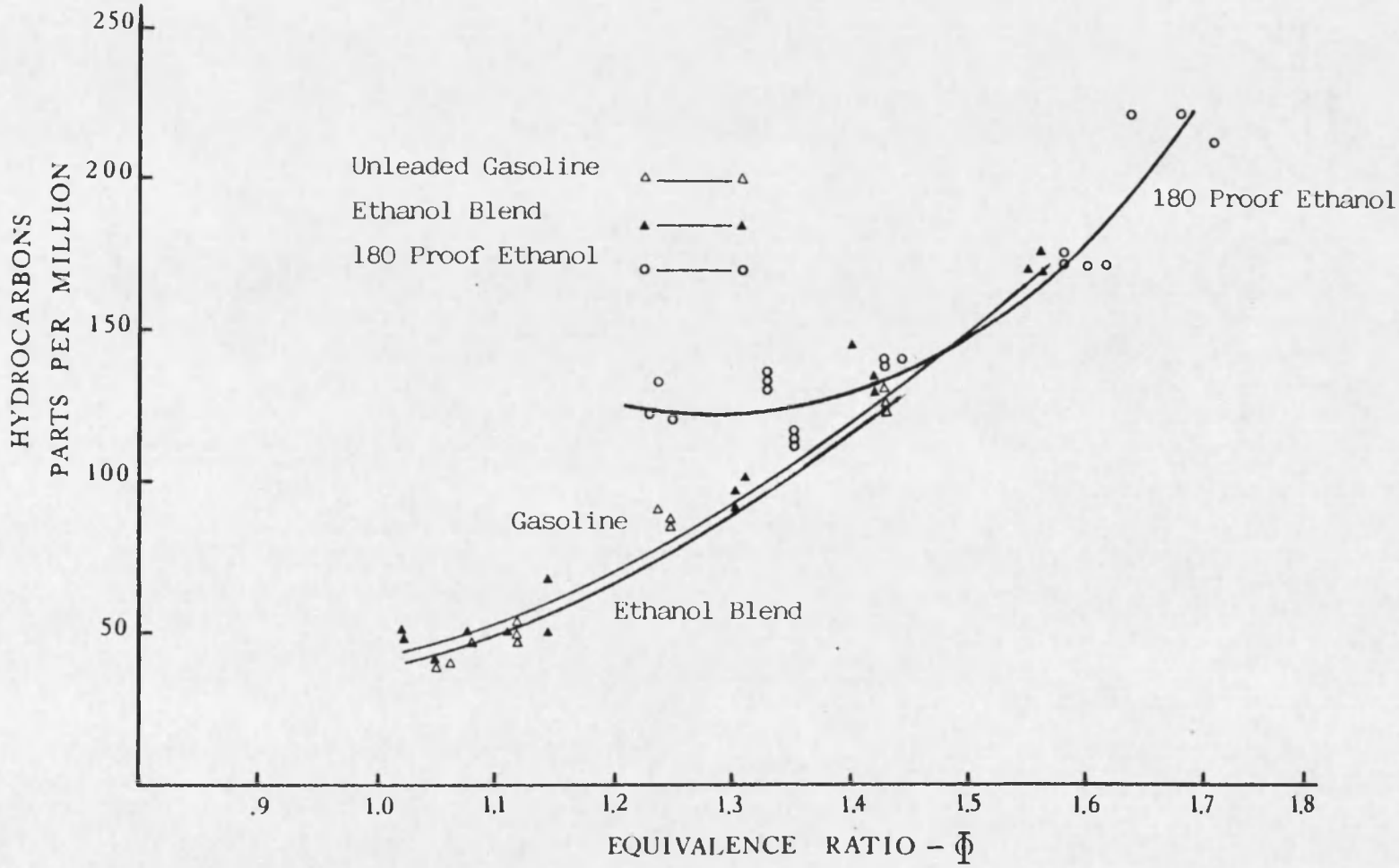

Figure 7. Hydrocarbon Emissions vs Equivalence Ratio for Unleaded Gasoline, 10 Volume % Anhydrous Ethanol/Unleaded Gasoline and 180 Proof Ethanol.

Statistical comparisons of the regression functions at the 0.05 level of significance show:

1. No statistical difference among fuels when comparing power output, torque and fuel consumptions.

2. The enthalpy thermal efficiency using the fuel's lower heating value increased an average of 6.4% for the ethanol/gasoline blend fuel compared with gasoline (see Fig. 5).

3. The ethanol/gasoline blend fuel emitted less carbon monoxide than the baseline gasoline fuel. The blend fuel showed a reduction of up to 30% in carbon monoxide emissions during steady state emission runs. However, the blend fuel demonstrated no statistical difference in hydrocarbon exhaust emissions.

4. The addition of 200 proof ethanol to unleaded gasoline increased the research octane number by 4 points, from 92.2 to 96.1.

5. The ethanol/unleaded gasoline blend fuel was closely observed over the testing period for water tolerance. No phase separation occurred during any of the fuel tests.

Carburetor modifications and spark ignition timing changes were necessary when running the engine on ethanol. Performance data gathered on the use of straight 180 proof ethanol as fuel showed:

1. Engine torque with ethanol was 3.0% greater and peaked at a higher RPM when compared to unleaded gasoline. There was no significant difference in power output with ethanol and unleaded gasoline.

2. Because of ethanol's lower energy content per unit volume, fuel consumption was 66% greater than the baseline unleaded gasoline.

3. Sharply reduced levels of carbon monoxide were obtained for ethanol when compared to baseline unleaded gasoline. Steady state carbon monoxide levels were reduced up to 60% when compared to baseline unleaded gasoline.

4. Increased levels of hydrocarbons were obtained for ethanol when compared to baseline unleaded gasoline.

CHAPTER 5

RECOMMENDATIONS

The engine's compression ratio has to be raised to approximately 11:1 to 1 to take advantage of ethanol's high octane number. This would increase power output and thermal efficiency.

More research is needed before a complete evaluation of straight 180 proof ethanol as an engine fuel can be made. Problems were encountered with fuel vaporization and inlet manifold distribution. Because of the inlet manifold configuration, it was not possible to have additional mixture heating to vaporize the ethanol. Some of the ethanol passed into the combustion chamber unvaporized and exited into the exhaust as unburned fuel raising the hydrocarbon levels.

APPENDIX A

CALCULATION OF STOICHIOMETRIC FUEL/AIR RATIO FOR BLEND FUEL

To determine the stoichiometric fuel/air ratio for the blend fuel, the number of gram/moles of ethanol and gasoline per liter of mixed fuel must be known. The blend fuel is a 10% by volume mixture of ethanol and gasoline.

Fuel : Ethanol - 100% pure
Chemical formula: C_2H_5OH
Density : 0.7939 $\frac{gm}{cc}$
Molecular weight: 46.07
Temperature : 21.1°C (70°F)

The number of gm-moles of C_2H_5OH per 1000cc of blend fuel is:

$$\frac{100cc\ C_2H_5OH \times 0.7939\ \frac{gm}{cc}}{46.07\ \frac{gm}{gm\text{-}mole}} = 1.723\ gm\text{-}moles\ C_2H_5OH$$

Fuel : Unleaded gasoline
Chemical formula: CH_2
Density : 0.7467 $\frac{gm}{cc}$
Molecular weight: 14.03
Temperature : 21.1°C (70°F)

The number of gm-moles of CH_2 per 1000cc of blend fuel is

$$\frac{900cc\ CH_2 \times 0.7467\ \frac{gm}{cc}}{14.03\ \frac{gm}{gm\text{-}mole}} = 47.91\ gm\text{-}moles\ CH_2$$

Total H_2 in gasoline = 47.91 gm-moles H_2
Total C in gasoline = 47.91 gm-atoms C
1000cc blend fuel = 1.723 gm-mole C_2H_5OH + 47.91 gm-atoms C
+ 47.91 gm-mole H_2

The chemical formula for combustion of the blend fuel with dry atmospheric air is as follows:

$$1.723\ C_2H_5OH + 47.91\ H_2 + 47.91\ C + A\ O_2 + 3.76\ A\ N_2 \longrightarrow$$

$$X\ CO_2 + Y\ H_2O + Z\ N_2$$

By balancing the number of carbon, hydrogen, oxygen and nitrogen atoms, the stoichiometric fuel-air ratio can be determined. The chemical formula for stoichiometric combustion of the blend fuel with air is:

$$1.723\ C_2H_5OH + 47.91\ H_2 + 47.91\ C + 77.03\ O_2 + 289.65\ N_2 \longrightarrow$$

$$51.36\ CO_2 + 53.08\ H_2O + 289.65\ N_2$$

For the stoichiometric combustion process for the blend fuel, the fuel/air ratio is:

$$\frac{77.03 + 289.65}{1} = 366.68\ \frac{\text{moles air}}{\text{mole fuel}}$$

$$\text{Molal mass of air} = 28.97\ \frac{\text{gm}}{\text{gm-mole}}$$

$$\text{Molal mass of fuel} = 751.37\ \frac{\text{gm}}{\text{gm-mole}}$$

$$F/A = \frac{751.37\ \frac{\text{gm}}{\text{gm-mole}}}{366.68\ \frac{\text{moles air}}{\text{moles fuel}} \times 28.97\ \frac{\text{gm}}{\text{gm-mole}}} = 0.707\ \frac{\text{gm fuel}}{\text{gm air}}$$

APPENDIX B

SAMPLE CALCULATIONS

Shown below is actual data taken on March 21, 1980.

Date	:	3/21/80
Data Sheet No.	:	23
Time	:	9:40 A.M.
Fuel	:	Gasoline
Fuel Used	:	0.05 lb_m
Duration of Run	:	135.2 sec
Number of Revolutions	:	2263
Beam Loan	:	71.0 lb_f
Coolant Inlet Temperature	:	162°F
Coolant Outlet Temperature	:	170°F
Oil Pressure	:	20 psi
Oil Temperature	:	170°F
ΔP Inlet Air Manometer	:	2.9 in H_2O
Barometric Pressure	:	27.3 in Hg
Wet Bulb Temperature	:	61°F
Dry Bulb Temperature	:	80°F

Observed brake power is calculated by the following equation:

$$bp = \frac{F \times N}{K \times t}$$ where

bp = Observed brake power, Hp

F = Beam load, lb_f

N = Actual revolutions

t = Test time, min

K = Dynamometer constant

Observed torque can be indirectly calculated from the horsepower equation.

$T = 1.313 \times F$ where T = Brake torque, $ft\text{-}lb_f$

F = Beam load, lb_f

Engine performance is affected by barometric pressure, temperature, and humidity of the ambient atmosphere. A correction factor is used to convert the observed data to specified standard atmospheric conditions for comparison purposes.

$$CF = \frac{B_d^*}{B_{dt}} \left[\frac{t_t + A}{t_t^* + A}\right]^{0.5}$$

$$B_{dt} = B - e$$

where CF = Correction factor

B_{dt} = Observed dry barometric pressure, in Hg

B = Observed barometric pressure, in Hg

e = Water vapor pressure in atmosphere, in Hg

B_d^* = Standard dry barometric pressure, 29.00 in Hg

t_t^* = Standard temperature, 85^oF

t_t = Observed air temperature, ^oF

A = Absolute temperature constant, 460^oF

The exact manner in which atmospheric conditions affect fuel consumption is not known, a correction factor is not applied, and specific fuel consumption is reported as observed (SAE Handbook 1979).

Specific fuel consumption is calculated by the following equation:

$$sfc = \frac{M_f}{bp \times t}$$ where

sfc = Specific fuel consumption, $\frac{lb_m}{Hp\text{-}hr}$

M_f = Mass of fuel, lb_m

bp = Observed brake power, Hp

t = Test time, hr

The enthalpy thermal efficiency can be determined from the specific fuel consumption and heating value of the fuel. The lower heating value of the fuel will be used for all calculations.

$$E_t = \frac{K}{sfc \times LHV}$$ where

E_t = Enthalpy thermal efficiency

sfc = Specific fuel consumption, $\frac{lb_m}{Hp\text{-}hr}$

LHV = Lower heating value, $\frac{BTU}{lb_m}$

K = Conversion constant, 2545 $\frac{BTU}{Hp\text{-}hr}$

Engine air flow was measured by a single orifice air box with the pressure differential across the orifice measured by a water

manometer. The equation used to determine air flow is:

$$\dot{m}_{air} = 1.16 \left[\frac{520}{460 + t_t} \times \frac{B}{29.92} \times \Delta P \right]^{0.5}$$

where $\dot{m}$ = Air flow, $\frac{lb_m}{min}$

t_t = Observed air temperature, ^oF

B = Observed barometric pressure, in Hg

ΔP = Inlet air manometer reading, in H_2O

The fuel flow is calculated by the following equation:

$$\dot{m}_{fuel} = \frac{M_f}{t}$$ where M_f = Mass of fuel, lb_m

t = Test time, min

Calculations

$$bp = \frac{71.0\ lb_f \times 2263\ Rev}{4000 \times 2.25\ min} = 17.85\ Hp\ (23.9\ kW)$$

$$T = 1.313\ ft \times 71.0\ lb_f = 93.22\ ft\text{-}lb_f\ (126.4\ N\text{-}m)$$

$$e = 0.383\ in\ Hg\ (from\ Psychometric\ Chart)$$

$$B_{dt} = 27.30\ in\ Hg - 0.383\ in\ Hg = 26.92\ in\ Hg$$

$$CF = \frac{29.00\ in\ Hg}{26.92\ in\ Hg} \left[\frac{(80 + 460)\ ^oR}{(85 + 460)\ ^oR} \right]^{0.5} = 1.077$$

Corrected Power:

$$bp_c = 17.85\ Hp \times 1.077 = 19.22\ Hp\ (25.8\ kW)$$

Corrected Torque:

$$T_c = 93.22 \text{ ft-lb}_f \times 1.077 = 100.4 \text{ ft-lb}_f \quad (136.1 \text{ N-m})$$

Specific Fuel Consumption:

$$sfc = \frac{0.5 \text{ lb}_m}{17.85 \text{ Hp} \times 0.037 \text{ hr}} = 0.745 \frac{\text{lb}_m}{\text{Hp-hr}} \quad \left(0.452 \frac{\text{Kg}}{\text{kW-hr}}\right)$$

Enthalpy Thermal Efficiency:

$$E_t = \frac{2545 \frac{\text{BTU}}{\text{Hp-hr}}}{0.745 \frac{\text{lb}_m}{\text{Hp-hr}} \times 18970 \frac{\text{BTU}}{\text{lb}_m}} = 0.18$$

Engine Air Flow:

$$\dot{m}_{air} = 1.16 \left[\frac{520 \text{ }^{\circ}\text{R}}{460 + 80 \text{ }^{\circ}\text{R}} \times \frac{26.92 \text{ in Hg}}{29.92 \text{ in Hg}} \times 2.6 \text{ in } H_2O\right]^{0.5}$$

$$= 1.74 \frac{\text{lb}_m}{\text{min}} \quad \left(3.84 \frac{\text{Kg}}{\text{min}}\right)$$

Engine Fuel Flow:

$$\dot{m}_{fuel} = \frac{0.05 \text{ lb}_m}{2.25 \text{ min}} = 0.222 \frac{\text{lb}_m}{\text{min}} \quad \left(0.49 \frac{\text{Kg}}{\text{min}}\right)$$

The Resulting Fuel/Air Ratio:

$$F/A = \frac{\dot{m}_{fuel}}{\dot{m}_{air}} = \frac{0.222 \frac{\text{lb}_m}{\text{min}}}{1.74 \frac{\text{lb}_m}{\text{min}}} = 0.127$$

APPENDIX C

TEST EQUIPMENT

Engine

Manufacturer	:	Case SE, No. 5109887 SF
Bore	:	3.5 in
Stroke	:	4.0 in
Compression Ratio	:	6.0:1
Number of Cylinders	:	4
Displacement	:	154 in^3 (2524 cc)

Dynamometer

Manufacturer	:	General Electric
Model	:	Type TLC, Class 1, Direct Current
Voltage	:	250 Volt
Amperage	:	122 Ampere
Capacity	:	45 Hp (60 KW)

Exhaust Analyzer

Manufacturer	:	Sun Equipment
Model	:	APA 75, 47E 14709
Type	:	Infrared
Testing Range	:	0 - 500 ppm Low Scale HC
		0 - 2000 ppm High Scale HC
		0 - 2.5% Low Scale CO
		0 - 10% High Scale CO

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