EPA-AA-TSS-83-9

Technical Report

Methanol Vaporization: Effects on Volumetric Efficiency and on Determination of Optimum Fuel Delivery System

by

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August 1984

NOTICE

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I. Introduction

The Office of Mobile Sources within the Environmental Protection Agency has studied and evaluated alternative transportation fuels since its formation in 1970. EPA's responsibilities under the Clean Air Act also have necessitated a significant regulatory role dealing with transportation fuels. In particular, Section 211 of the Clean Air Act requires EPA to play a key role in the introduction of new fuels and fuel additives. Perhaps most visible was EPA's role in the introduction of unleaded gasoline to permit the use of catalytic converters on 1975 and later model year automobiles. More recently EPA has responded to a growing interest in the use of oxygenates (in particular methanol) for use in motor vehicles and for blending with gasoline.

As part of this response to the interest in methanol, a program was undertaken to grade several fuel utilization concepts. One of these concepts that has been evaluated has been loosely labeled fumigation. Normally, the term fumigation is used in the context of diesel engines when all or a majority of the fuel is introduced into the intake track, as opposed to injecting the fuel into the cylinder (or pre-chamber). Generally, fumigated diesel engines are unthrottled engines and rely on the fuel delivery system to regulate the load. In our testing with a throttled spark ignition (SI) engine converted to methanol use, only a portion (20-33%) of the fuel was injected or "fumigated" into the intake track. Two fumigation locations (see figure 1) were used - one significantly upstream from the throttle valve, and the other slightly downstream of the throttle. Both locations were upstream from the normal port injectors which injected the majority of the fuel (for more details see reference 1).

Testing has shown this upstream fumigation of methanol to cause a decrease in volumetric efficiency(1)*. Volumetric efficiency is usually defined as a measure of the mass of air/fuel charge which is successfully taken into the cylinder on each intake stroke. It is an "efficiency" with respect to utilization of the cylinder space, but only indirectly with

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*Numbers in parentheses designate References at end of paper. Reference (1) is reprinted in Appendix VII.

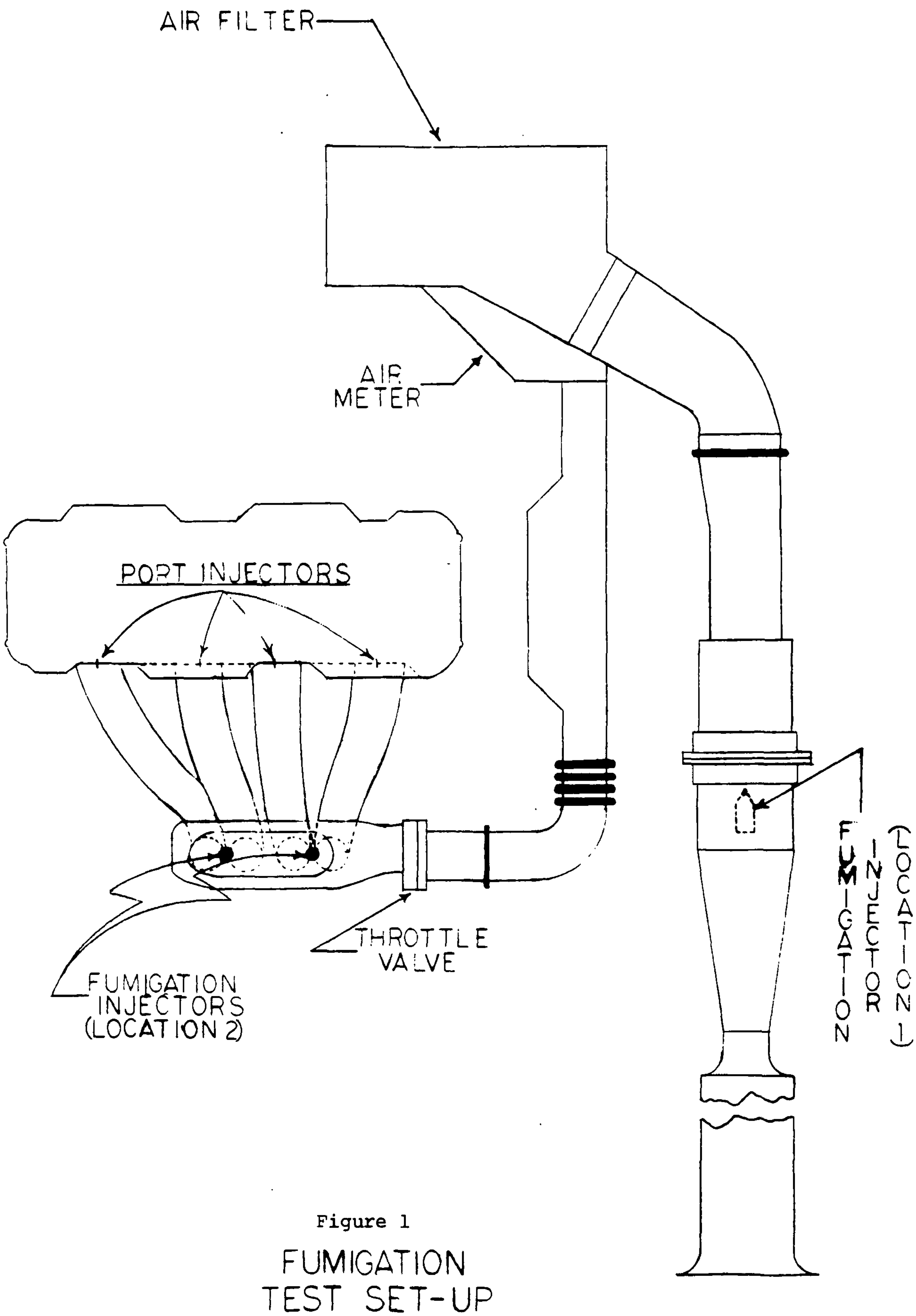


Figure 1

FUMIGATION TEST SET-UP

respect to energy or fuel utilization. A loss in volumetric efficiency would have as its most direct consequence a reduction in power for a given throttle position and engine speed, including a reduction in maximum power. Pumping losses would increase, causing a small reduction in overall engine efficiency. The observed loss in volumetric efficiency was opposite of the commonly held assumption that vaporizing the fuel would cool the intake air and increase the volumetric efficiency. Thus, a small mathematical exercise was undertaken to attempt to quantify the physical processes causing the real effects to be opposite of the anticipated ones. Although the results of this exercise did not uncover new phenomena governing engine operation (as a review of Reference 2 will indicate), it did reinforce the importance of existing but sometimes over-looked phenomena on engine operation. Consideration of these phenomena may be more important for a methanol engine than for a gasoline engine.

II. Summary

EPA tests utilizing fumigation of methanol caused a decrease in volumetric efficiency. The apparent cause of this reversal in the actual versus the expected result was primarily due to heat transfer effects from the walls of the intake passage. A secondary issue deals with the necessity for the combination of the local partial pressure of the fuel and the stagnation temperature to be on the vapor side of the vapor pressure curve for the fuel in question in order to allow vaporization to take place. The partial pressure is the key to this secondary effect and is influenced directly by the local equivalence ratio.* In the cases discussed, this secondary effect had only a minor influence on the results, primarily because the effective equivalence ratio was very lean at the point of fumigation (i.e., the fumigation injectors received only part of the total engine fuel flow resulting in a much lower partial pressure of the fuel in the local area). If, however, the local equivalence ratio at the fumigation injector had approached a stoichiometric mixture or richer, the partial pressure/ temperature effect would prevent a significant portion of the fuel from vaporizing. In this case (rich condition), the intake air temperature depression due to vaporization would be limited to the level allowed by the vapor pressure curve of the substance.

As stated, our experiments were not substantially effected by the secondary effect because our local air/fuel ratios were

* Note: the Greek word "phi" is used to designate equivalence ratio throughout the report.

very lean. Under these conditions, all or nearly all of the fuel was evaporated. On the other hand, even though the lean mixture allowed vaporization, the temperature depression available from this vaporization was limited because of the small mass of fuel involved. Under these conditions, normal component temperatures of the intake air duct-work appeared to be sufficient to cause the initially cooled mixture of fuel and air to heat back up to the original temperature (due to the temperature differential between the wall and the newly cooled mixture). Since the air returned to its original specific volume (at its original temperature), and the vaporized fuel had a much larger specific volume than the liquid fuel, the volume of the mixture (air plus gaseous fuel) was greater than the volume of intake air with no fuel. Because the volume flow rate (Q) of the engine would remain the same for the same manifold vacuum,* an intake charge of this gaseous fuel-air mixture would contain less air than an intake charge of air without the fuel.

This decrease in air flow when using a fuel-air mixture will manifest itself in a low volumetric efficiency (VE). In our case, if heat transfer is sufficient to reheat the mixture to its original temperature, then the theoretical loss in volumetric efficiency (for phi = 0.8) is around 2.3 percent for location 1 and around 3.8 percent for location 2 (see Table 1). The average loss obtained from an analysis of the test results is 2.8 percent for location 1 and 4.1 percent for location 2.

Table 1

Volumetric Efficiency

	Location 1	Location 2
Calculated loss (Including Heat Transfer)	2.3%	3.8%
Measured loss (average)	2.8%	4.1%

*For a complete derivation, see Appendix III. It works out that the total mass flow and total gaseous volume flow vary only slightly with changes in density for a constant pressure drop. What changes is the portion of the total that is fuel, the remainder is the change in inlet air flow. This can readily be seen by comparing the component volumes of a stoichiometric mixture per pound of fuel in Tables AIID-1 to 3 in Appendix II (air component =Mv1, fuel component = Mxv2).

Recognizing that the average test results mask some data scatter, we are left with the fact that fumigating a fraction of the total fuel flow caused a loss in volumetric efficiency. The theory suggests that heat transfer which reheats the charge to the original temperature and expands the fuel volume can explain a majority, if not all, of the difference between the expected results and the actual results. Any differences not explained by heat transfer are probably due to experimental error, or some other phenomonea not yet fully understood.

In principle, it should be possible to reduce the heat transfer down stream of the fumigation point. This might be accomplished, for example, by insulating the inner surface of the intake track. However, the temperature drop would be limited by the dew point of the mixture strength. For example a carbureted system at stoichiometric conditions and 15 in. Hg MAP would require a mixture temperature above 50°F to maintain 100% vapor. (Note: At this temperature, the effect of fuel expansion would cause a theoretical loss in volumetric efficiency of slightly greater than 6%.) On the other hand, if one were willing to tolerate a portion of the fuel in liquid form, there could be an equilibrium point where the mix temperature is depressed sufficiently to allow an improvement in volumetric efficiency. However, distribution of the liquid fuel could become a problem. If partial fumigation with an insulated manifold was used in combination with port injection for cylinder distribution, a small increase in volumetric efficiency might be possible (our testing suggests 2 to 4 percent - see table 3). This increase might only be available under certain operating conditions due to changes in the equilibrium condition of the mix during different operating regimes. Therefore, while it might be possible to obtain some benefit with partial fumigation, it would require careful design to insure that such potential improvements would occur at the design operating point, since it seems possible that the off-design points would incur losses in volumetric efficiency.

The results of this experimental work and theoretical analysis allowed us to make the following generalizations and conclusions.

II. Conclusions

A. Volumetric Efficiency Losses

A loss in volumetric efficiency when mixing methanol with air in the inlet system (compared to air only) is primarily a result of heat transfer effects warming the mixture.

B. Prevention of Volumetric Efficiency Losses

By awareness and proper manipulation of the following five factors a potential loss in volumetric efficiency can most likely be avoided.

1. The partial pressure of the fuel is governed by the mixture strength (f/a), the local manifold air pressure at the time of vaporization, and the portion of the fuel permitted to be vaporized by the partial pressure characteristics at the local temperature.

2. The temperature of the mixture after the heat of vaporization is included is relative to the degree of vaporization, the partial pressure of the fuel, and the amount of the fuel. The temperature of the mixture is also affected to some degree by the initial stagnation air temperature.

3. The heat transfer to the mixture is governed by the difference between the wall temperature and the mixture temperature, but is also influenced by the amount of liquid fuel coming in contact with the wall. The effects of the liquid fuel contact may be substantial.

4. The length of the intake passage from the point of fuel introduction to the cylinder over which the heat transfer can take place is of concern.

5. The time during which heat transfer can take place also affects the results. The time can be influenced by engine speed, load (i.e., throttle position), geometry of the intake tract (i.e., velocity in the system), or, distance (i.e., length of tract).

C. Optimum Fuel Delivery

From these factors, it appears that the best location to add fuel when considering volumetric efficiency (VE) with methanol is directly into the cylinder when the intake valve is open. Direct injection would probably be the best(2), but port injection near the intake valve would be expected to be a better choice on a cost/benefit basis than direct injection. Port injection, of course, would be expected to be much better than carburetion for VE. Because of potential side benefits of improved driveability (less manifold wetting, etc.) and possibly better emission control, port injection should also be better than carburetion on a cost/benefit basis. If carburetion is used, however, more effort should probably be directed at developing a more uniform wet-flow cylinder distribution for the methanol intake manifold than has been found to be necessary for a gasoline manifold. This analysis of methanol evaporation versus vapor pressure and temperature suggests that sufficient manifold temperature will not be available (and may never be available) under enough conditions to avoid wet flow mal-distribution problems (e.g. cold start, warm-up, etc) if a normal gasoline manifold/carburetor combination is used when using methanol.

Considering the qualitative aspects of this analysis, one might predict that phased port injection would perform better than normal port injection. Such a prediction would be based on the assumption that a majority of the fuel would be expected to be evaporated within the cylinder when phased port injection was used. Such evaporation when the intake valve is open would be expected to improve volumetric efficiency because the greater volume to surface area within the cylinder (compared to the intake tract) would tend to limit the heat transfer to the evaporated mixture. Secondly, any evaporation after the intake valve is closed would reduce the work of compression by cooling the mixture. Finally, any heat transfer that did occur would cool the cylinder wall, which would tend to reduce the load on the cooling system. These assumed physical reactions could allow the potentially cheaper phased port injection system to duplicate the performance of a more expensive direct cylinder injection system.

IV. Discussion

The original focus of the fumigation testing was to ascertain any potential benefits that might be available due to the high latent heat of methanol. An analysis in Appendix VI suggests that if we were to cool dry air (i.e. no fuel) down to the lowest fuel dew-point temperature experienced in testing, we could reduce the pumping work by roughly 2 percent for a constant volume flow rate or in the case of constant mass flow rate, 15 percent. We hypothesized that by vaporizing only a small fraction of the fuel upstream of the port injectors, the high latent heat in methanol would allow some improvement in volumetric efficiency, and possibly some reduction in the pumping work. The test results contradicted this hypothesis.

In order to investigate this problem, the basic precepts behind volumetric efficiency were examined. Volumetric efficiency can be defined as the amount of intake air ingested (at STP) expressed as a percentage of the swept volume of the engine. Several factors can influence the volumetric efficiency at wide open throttle; examples are camshaft design, valve size, exhaust restricition, intake path size and length, and resonant effects due to intake geometry. At part throttle all of the above factors are modified by the throttle valve. In our experiments all of the basic factors were held constant since the same engine was used. Also, all tests were conducted at the same RPM and the same low level of torque (which required part throttle operation). The comparisons of volumetric efficiency were all made at the same manifold vacuum level (i.e., same throttle position).

Under these conditions, only two basic factors are readily apparent that could cause the volumetric efficency to decrease when fumigation was added. One is that the basic air ingestion characteristics of the engine were changed with the addition of the fumigated fuel. The other is that the fumigated fuel displaced some of the air that would normally be ingested if the fuel were not there. In the first case, it is hard to imagine how adding fuel to the air stream could change the basic air ingestion characteristics of the engine under the moderately throttled condition as occured in our tests. At wide open throttle, factors such as poor cylinder-to-cylinder distribution, or a change in the resonant effects due to the additional fuel mass might affect the volumetric efficiency, but the power would also be expected to vary, hence dissolving the comparison at constant power. The second consideration, the one of fuel volume replacing air volume is more palatable, but only if the fuel is vaporized. In liquid form, the volume displacement of the fuel is so small that the effect on volumetric efficeincy would be expected to be pratically negligable.

Therefore, if only the vaporized fuel has the potential to cause the majority of observed effects, then the factors influencing vaporization must be evaluated to determine if these factors could have been present in sufficient magnitude to create the observed effects. The factors affecting vaporization are temperature and pressure. As will be shown in the following sections, these factors have a dramatic influence on the amount of vaporization that takes place. The first step taken to evaluate the potential volume increase of the fuel was to identify the expected temperature of the intake air after a given quantity of methanol was evaporated (regardless of other influencing factors). The following sections address the determination of this temperature drop, and how the vapor pressure of the fuel, in concert with heat transfer effects, influence the final fuel volume.

A. Adiabatic Temperature Drop vs. Vapor Pressure

If heat input is assumed to be equal to zero for the time being, equation AII-1 (ref. 2) in Appendix II provides an estimation of the temperature drop of the intake air. The temperature drop predicted by the equation is governed by the portion of the fuel evaporated and the local fuel-air ratio. Since the first evaluation looked at temperature drop with no heat transfer, the term Q was set at zero for the first analysis.

$$\text{(AII-1)} \qquad \Delta T = [(x)\ (F)\ (HLG) + (Q)]\ /\ (1-F + xF)\ (Cp)^{*}$$

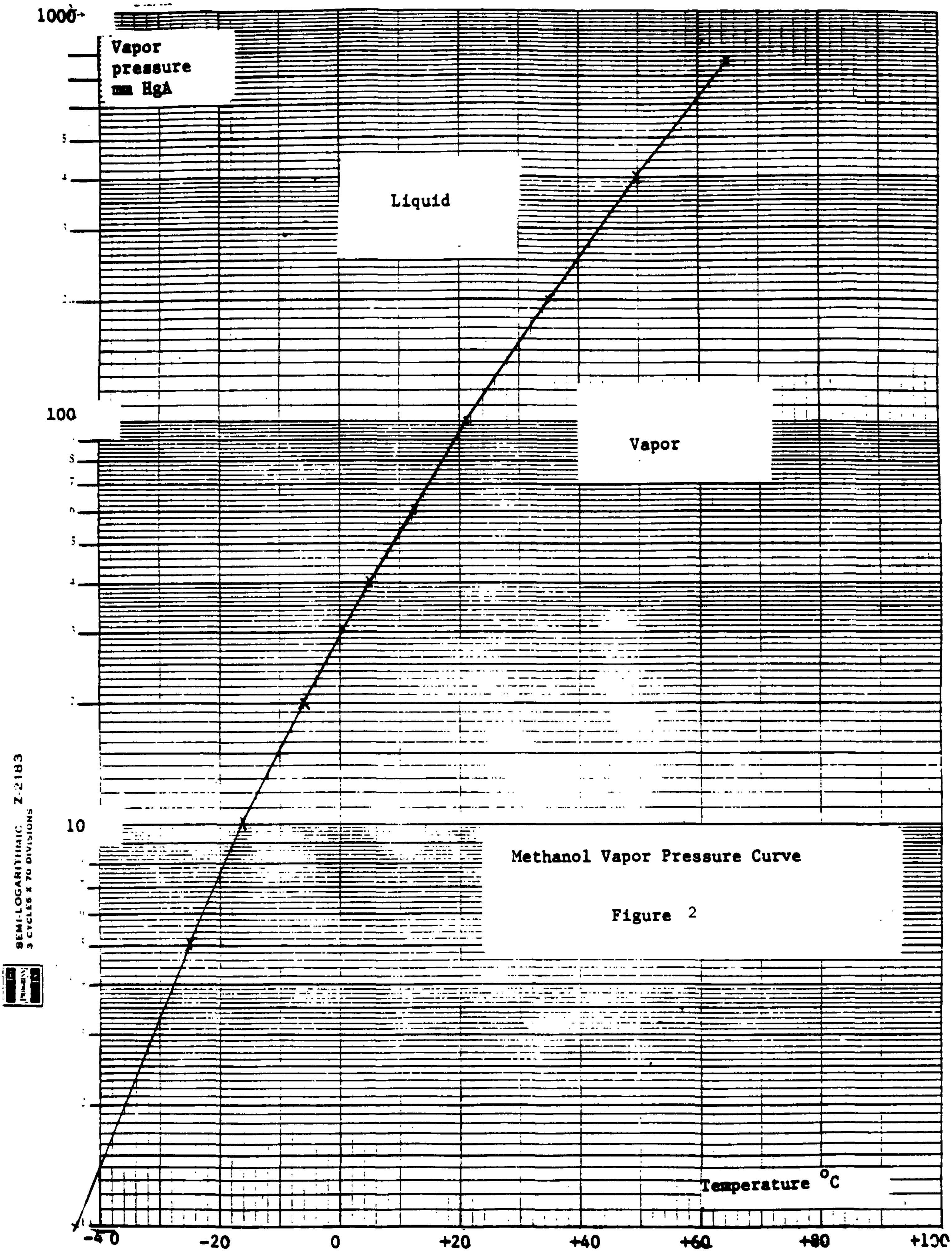
In order to determine the amount of fuel that can be evaporated under any given condition, the vapor pressure versus temperature characteristics of the compound in question must be known. Data on these characteristics was obtained from reference 3 for methanol (see Table 2), and a curve of that data was constructed (see Figure 2). The next step in order to use this data would be to determine the partial pressure of the methanol under the local conditions which are controlled by the local fuel-air ratio and the portion of the fuel evaporated (e.g., 25%, 50%, 75%, 100%). A range of partial pressures can be calculated for each fuel-air ratio (see Appendix I Section IIIC). Using these partial pressures, and the curve in Figure 2, a dew point temperature can be estimated such that if the mixture temperature were below this temperature (and at the same

Table 2

Vapor Pressure Curve for Methanol(3)

mm HgA	°C
1	-44.0
5	-25.3
10	-16.2
20	- 6.0
40	+ 5.0
60	+12.1
100	+21.2
200	+34.8
400	+49.9
760	+64.7

*F = f/a, HLG = heat of vaporization, x = portion evaporated



Methanol Vapor Pressure Curve

Figure 2

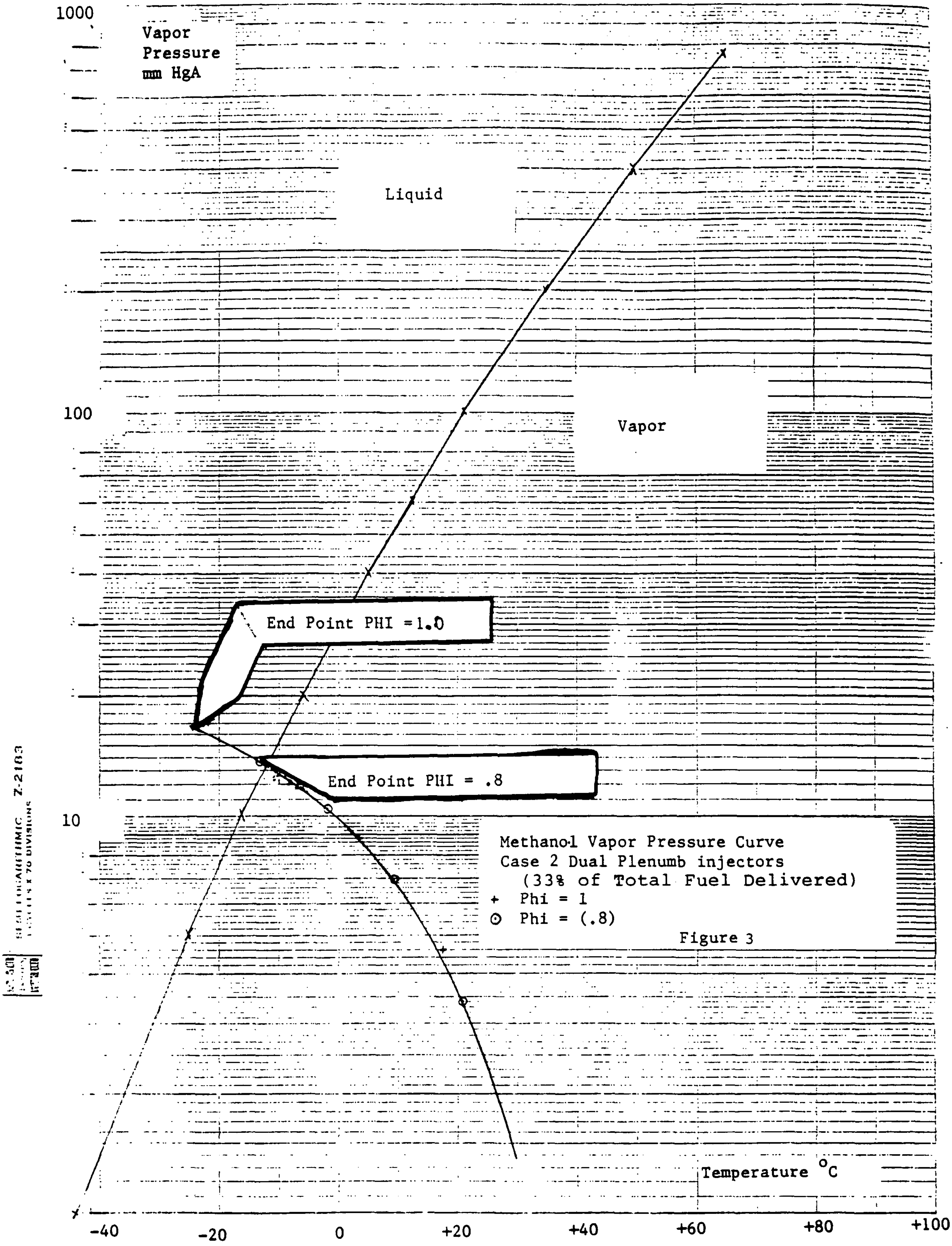

1000
Vapor
Pressure
mm HgA
Liquid
100
Vapor
End Point PHI = 1.0
End Point PHI = .8
10
Methanol Vapor Pressure Curve
Case 2 Dual Plenumb injectors
(33% of Total Fuel Delivered)
+ Phi = 1
⊙ Phi = (.8)
Temperature °C
-40
-20
0
+20
+40
+60
+80
+100
Z-2183

Figure 3

partial pressure), it would be expected that any methanol vapor would condense back into liquid droplets until new equilibrium conditions were satisfied. The same assumed evaporative percentages used to calculate the range of partial pressures can also be substituted into the intake air temperature drop equation (e.g., AII-1), to obtain estimated adiabetic intake air temperatures (T1- ΔT1 in Appendix II Section IIB).

If the estimated intake air temperature from equation AII-1 is warmer than the dew point temperatures (i.e. T1-ΔT1 greater than TDEW, Appendix II, Section IIB) then vaporization could be assumed to take place.

In our case, where we introduced only 20 percent of the fuel at location 1 (Figure 1) upstream of the normal injection system (the remainder from the port injectors), all of the fuel can be evaporated at this upstream location due to partial pressure effects (Location 1, Appendix II, Section IIB). However, the richer of the two equivalence ratios indicates that total vaporization is marginal without additional heat since the depressed intake air temperature is effectively the same as the dew point temperature. If we look at Location 2 (Figure 1) where 33% of the fuel is introduced upstream of the port injectors (Appendix II, Section IIB), the leaner of the two equivalence ratios would have marginal evaporation, while the richer ratio cannot vaporize all of the fuel from partial pressure effects alone (only about 78% will evaporate). The visualization of this effect of incomplete evaporation can be seen by plotting the calculated intake temperature versus the vapor pressure as in Figure 3.

The result of this dew-point limiting temperature can also be seen in the volumetric efficiency changes (CVE) in Table 3 (positive is gain, negative is loss). At location 2, we see a 4.1 percent improvement in volumetric efficiency with 75 percent of the fuel evaporated. Whereas at 100 percent evaporated, the volumetric efficiency improvement drops to only 2.4 percent. This is because the temperature of the mix must be raised from the 75 percent evaporation dew point to the dew point temperature necessary to achieve evaporation of 100% of the fuel.

Comparing the calculated volumetric efficiency improvements in Table 3 to the actual results in Table 4, we see there is a sizeable difference. At an equivalence ratio of 0.8, the calculated results for location 1 predict a 2.5 percent gain, while the actual results show a 2.8 percent loss. The numbers are a 3.8 percent gain, and a 4.1 percent loss for location 2. Obviously the results do not match, and further investigation was in order.

Table 3

Volumetric Efficiency Change "Without"
Heat Transfer from the Engine Structure

Phi**	X	CVE* (%)-Location 1	CVE* (%)-Location 2
1.0	.25	0.7	1.2
	.50	1.5	2.6
	.75	2.4	4.1
	1.00	2.9	2.4
0.8	.25	0.6	0.9
	.50	1.2	2.0
	.75	1.8	3.1
	1.00	2.5	3.8

*CVE is calculated by Equation AIII - 22 in Appendix III
**Represents overall equivalence ratio, the local equivalence ratio at location 1 was 20% of th overall ratio, and 33.3% of the overall ratio at location 2.

Table 4

Average* Volumetric Efficiency
Change (Test Results)

Phi**	CVE (%)-Location 1	CVE (%)-Location
0.8	-2.8	-4.1

*See Appendix IV.

**Several different equivalence ratios were tested. Generally they ranged between 0.7 and 0.9.

B. Heat Transfer

By noting the probability that only marginal evaporation was likely to take place under some conditions (from the 20% and 33% fuel split at the fumigation injectors), the physics seemed to imply that as more fuel was added to the fumigation injectors, the amount of the fuel that could be vaporized was reduced. The apparent reason for this was that the additional fuel flow (either from a larger flow split or a richer fuel air ratio) caused the partial pressure of the fuel to increase which in turn raised the dew point temperature necessary for evaporation. In the extreme, if all of the fuel were to be introduced at the front end of the intake manifold (as in a carburetor) the partial pressure of the fuel would be considerably greater (from 2.8 to 4.5 times greater*) than the cases analyzed. In order for total evaporation to occur under these conditions, the final intake air temperature cquld not be below 45° to 50°F (assuming 15 in. HgA MAP). However, since vaporizing the total fuel flow would depress the inlet air temperature approximately 300°F, and assuming an initial inlet air temperature of around 80°F, a discrepancy of some 265°F exists that must be made up by heat transfer from some source before the fuel can fully vaporize. Equation AII-1 indicated that the addition of heat during evaporation would cause the final intake air temperature depression to be less than in the absence of heat addition. Therefore, heat transfer becomes a logical parameter to be investigated in resolving the difference between the expected results and the measured results.

Since heat transfer would raise the temperature of the mix, it would be useful to assume several recovered charge-air temperatures in order to identify if further investigaton would be fruitful. Then we can perform the heat transfer calculations to identify if there could be a sufficient temperature differential to create the assumed recovery temperatures. The results in Table 5 speak for themselves. The volumetric efficiency losses from heat transfer resemble the actual losses observed (Table 4) much closer than the calculated gains in Table 3. Also, if we were to carburet the entire amount of fuel, as opposed to the fractional fumigation used in these tests, a sizable loss in volumetric efficiency would occur (assuming sufficient heat transfer to vaporize all of the fuel).

*Compare the mole fraction (mf 100) for 4 injectors (4 total) which would be all of the fuel to the mole fraction for 1 injector (5 total) and 2 injectors (6 total) for the two equivalence ratios in Appendix I Section C.

It is interesting to note that the calculated percent loss for volumetric efficiency remains constant for apparently any temperature above the original air temperature (assumed to be 80°F for location 1, and 90°F for location 2). The mechanics of this anomoly (constant VE loss with increasing temperature) are that the comparisons are made with all components at the same recovery temperature. Therefore, the

Table 5

Voluemtric Efficiency Change "with"
Heat Transfer from the Engine Structure
Resulting in an Assumed Charge-Air Temperature

Assumed Charge-Air Temperature	Phi*	CVE (%) Location 1	CVE (%) Location 2
-Partial Fumigation*			
80°F	1.0	-2.9	-3.9
100°F	1.0	-2.9	-4.7
120°F	1.0	-2.9	-4.7
80°F	0.8	-2.3	-3.0
100°F	0.8	-2.3	-3.8
120°F	0.8	-2.3	-3.8
-Carbureted*			
100°F	1.0	-12.8	-13.7
	0.8	-10.5	-11.4

*The equivalence ratio for partial fumigation represents the overall equivalence ratio, the local equivalence ratio at location 1 is 20% of the overall ratio, and 33.3% of the overall ratio for location 2. For the carbureted condition the local and the overall equivalence ratios are the same. For both the fuel delivery senerios, the pressure at location 2 is approximately 15 in. Hg of manifold vacuum.

total volume of fuel and air at 120°F is compared to the volume of air with no fuel at 120°F, not at 80°F (the original inlet temperature). The comparison at the elevated temperatures is made because it is assumed that any heat transfer that would heat the mixture to some temperature above the original air temperature would also heat the intake air in the baseline case (which did not have fuel in the mixture) to the same temperature. The reason the base case temperature would not be higher, is that in the base case the temperature gradient between the duct wall and the intake air is very small. Hence little heat is transfered. However, when methanol is injected, the intake air is cooled creating a reasonably large temperature gradient which allows much more rapid heat transfer. Once the cooled fuel-air mixture has returned to the base case, then any temperature gradient that would have affected the base case would also affect the fuel-air mixture in a similar fashion because the specific heats of air and methanol vapor are very similar. Hence, the conclusion that the volumetric efficiency change between air alone and the fuel-air mixture must be based on the same recovery temperature.

Because the investigation of heat transfer effects began after the tests were run, many of the temperature measurements that would be useful for the heat transfer calculations were not available. Therefore, many of these temperatures were assumed based on other factors. The issue here is to identify if rough heat transfer approximations with reasonable temperature assumptions can account for any of the discrepancy between the expected volumetric efficiency change and the measured change. As it turned out, even with extremely unsophisticated heat transfer approaches, heat transfer does account for a majority if not all of the difference between the expected and the observed.

The first assumption was that the only heat transfer that occurred, occurred between the intake tract wall and the vapor phase methanol-air mix. By using this assumption, we were able to apply equation (13.2 - 25) from reference 4 for convection in tubular heaters (this equation is repeated as AV-1. The various boundary assumptions used for the analysis are listed in Appendix VI.

The first case analyzed was for the single injector upstream of the inlet to the EFI system (ie. upstream of the air filter). The total distance from single injector to the throttle body mounted on the intake air collector was approximately 80 inches.

Since nearly all of this distance was surrounded by atmospheric temperature, it was assumed that at least the initial portion of the inlet system was at room temperature (approximately 80°F). It was further assumed that the temperature gradually increased along the length of the duct to a final temperature of 160°F. This temperature rise is considered to be an upper limit for estimation purposes for the following reasons. At about the 40 percent point in the intake tract, the intake air passes through a large box-like structure that houses an air filter. In leaving the air filter box, the intake air passes through a spring loaded air meter device. The box and air meter are metal and are in the vicinity of the exhaust manifold (within 10-12 inches). It is assumed some radiative heat transfer occurrs to the air filter/metering system. However, no attempt was made to correct for any of the potential heat transfer improvements due to the proximity of the exhaust manifold to the filter/metering box, to the increased surface area in the air filter, or to the increased residence time in the air box other than to assume the above temperature gradient along the duct. Following the air box was a long section of stamped steel duct work in close proximity to the cylinder head. And finally, a short section in front of and including the throttle valve was heated by engine coolant. The coolant was maintained at about 180°F on the engine dynamometer test stand.

By applying these boundary conditions to the tubular heater equation, the length of the intake tract necessary to recover the original air temperature (after the temperature depression due to evaporation) varies between a low estimate of 85 inches (phi = 0.8) to a high value of 137 inches (phi = 1.0). The variation in these calculated estimates is due to a range of assumed hydraulic diameters of the irregularly shaped intake tract. The measured distance on the hardware was approximately 80 inches.

The discrepancy between the measured tract length and the calculated range can be attributed to several reasons. First, mentioned before, is that more heat transfer could have occurred in the air filter box than was accounted for in the calculations. Second, and most likely, is the estimation procedure itself. All of the temperatures were estimated and not measured (note: if 100°F had been used for the final wall temperature (To2) instead of 160°F, the calculation would predict slightly less than a doubling in length). Probably more important, however, is the procedure only considered convective heat transfer to the gaseous methanol-air mixture. Heat transfer with a change in phase (fuel evaporation) can be very complex. The effects on the system due to convective heat transfer on any liquid fuel droplets that might have impinged on the wall could have had a noticeable influence on the total heat transfer rates. This effect, however, was not considered in the rough approximation. Finally, there is the additional possibility that some other unknown parameter influenced the volumetric efficiency results.

Even though some people might consider the discrepancy between the calculated and measured results to be minor (i.e., substantially less than one magnitude of difference), investigation into the results at Location 2 might support one of the suggested reasons for the discrepancy -- more specifically, the issue of neglecting the influence of heat transfer to liquid fuel droplets. At Location 1, a small amount of fuel was sprayed into the center of a large diameter duct (4.5 inches) approximately 6 to 8 inches upstream from the entrance to the normal inlet duct-work. At Location 2, 65 percent more fuel was sprayed into an air collector about 2 to 3 inches from the entrance of four 1.35 inch intake runners. Because of the physical set-up, more liquid droplets would be expected to impinge on the duct walls at Location 2 than at Location 1. If the heat transfer effects from the droplets were significant, one would expect the simple calculation procedure (i.e., neglecting droplet effects) to show less agreement at Location 2 than at Location 1.

The simple calculation predicts a path length for Location 2 of between 31.9 and 32.2 inches (phi = 0.8 and 1.0 respectively). The approximate measured distance is 14 inches. The over estimation by the simple approach is about 128% in this case (note: the over estimation value would increase if lower wall temperatures were used). At Location 1, the over estimation for the smallest assumed duct diameter is only about 6%, whereas the length of the largest assumed diameter is over estimated by around 58%. From this comparison, it is difficult to scientifcally say that the simple approach is more accurate at Location 1 than at Location 2. Hence, the hypothesis that liquid fuel droplets impinging on the intake walls increased the heat transfer can not be completely confirmed. However, since the convective heat transfer coefficient (h) for boiling liquids is approximately 100 times that for forced convection of gases, and about 5 times that of gases for viscous fluids(4)(5), there certainly is room for engineering judgement to expect that any fuel droplets on the walls would substantially enhance the heat transfer. Enhanced heat transfer would shorten the effective length of the intake track necessary to recover the original intake air temperature, thus bringing the calculated lengths more in line with the actual hardware.

In conclusion, the effects of heat transfer appear to play an important part in explaining the difference between the expected VE results and the actual results. One may argue that the estimated component temperatures are too high, or that the study should have investigated the effects of droplets on the walls more thoroughly, but in the final analysis, the aggregate evidence says heat transfer plays a role in the volumetric efficiency of the engine when methanol is introduced into the inlet track. Furthermore, because of the difference in the heat of vaporization*, the heat transfer role appears to be more dramatic for methanol than gasoline, and therefore this role should be given more consideration in the design of methanol fuel delivery systems.

*Obert(6) provides a procedure to calculate the dew point of gasoline (a multi-component mixture). Between 10% and 90% evaporated, the dew point temperature is very similar between gasoline and methanol. However, due to differences in the heat of vaporization and the specific heats, methanol will have an adiabatic temperature drop eight times (on the average) greater than for gasoline. Therefore, to maintain a similar dew point mixture temperature in the intake manifold, the heat transfer required for the methanol engine would be substantially greater than for the gasoline engine.

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Appendix I

Partial Pressure Calculation

I. Basic Equation
Phi = 1

(AI-1) $CH_3OH + 1.5(O_2 + 3.73\ N_2 + .04\ Ar) \rightarrow CO_2 + 2H_2O + 1.5\ (3.73N_2 + .04Ar)$

II. Molecular Weights

$CH_3OH = 32.043$ $\quad$ $N_2 = 28.014$

$O_2 = 32$ $\quad$ $Ar = 39.948$

III. Partial pressure (Pv) of Methanol

A. Equation for mole fraction with 100% vaporization

(AI-2) $$(mf100) = \frac{(z)}{(z)+(1.5)+(1.5)(3.73)+(1.5)(.04)}$$

where z = % of the total fuel used at specific location times 10^{-2}

mf = mole fraction

or

(AI-3) $$(mf100) = \frac{(z)}{(z) + (7.155/PHI)}$$

where Phi = (f/a) actual/(f/a) stoichiometric = equivalence ratio

B. Equation for mole fraction wih x% vaporized

(AI-4) $$mfx = \frac{(z)(x)}{(z)(x) + (7.155/PHI)}$$

where x = % of z amount of fuel that is vaporized times 10^{-2}

(e.g. 25, 50, 75, 100)

C. Values from equation AI.4.

	Configuration	Z	Phi	mf 25	mf 50	mf 75	mf 100
C1.	1 injector* (5 total)	.2	1	.007	.014	.021	.027
	1 injector* (5 total)	.2	.8	.006	.011	.015	.022
	4 injectors+(5 total)	.8	1	.027	.053	.077	.101
	4 injectors+(5 total)	.8	.8	.022	.043	.063	.082
C2.	2 injectors*(6 total)	.333	1	.012	.023	.034	.044
	2 injectors*(6 total)	.333	.8	.009	.018	.027	.036
	4 injectors+(6 total)	.667	1	.023	.045	.065	.085
	4 injectors+(6 total)	.667	.8	.018	.036	.053	.069
C3.	4 injectors+(4 total)	1.0	1	.034	.065	.095	.123
	4 injectors+(4 total)	1.0	.8	.027	.053	.077	.101

* Fumigation Injector
\+ Port Injector

D. Partial Pressure (pv)

(AI-5) (pv) = (mfx)(local absolute pressure)

Appendix II

Temperature Drop with Vaporization

I. Injector Location

A. Location 1

1. Upstream single injector with 4 port injectors.

2. Fuel flow split: 20% at Location 1, 80% at the normal port injectors

3. P1 = 29 in. HgA (736.6 mm HgA), T1 = 80°F (26.7°C)

4. Overall phi ratios examined = 1.0, 0.8

B. Location 2

1. Two air collector injectors with 4 port injectors.

2. Fuel flow split: 33.3% at Location 2, 66.7% at the normal port injectors

3. P2 = 15 in. HgA(381.0 mm HgA), T2 = 90°F (32.2°C)

4. Overall phi ratios examined = 1.0, 0.8

II. Calculations

A. Temperature drop (ΔT) Equation for x% evaporated(2)

(AII-1) $\Delta T = [(x)(F)(HLG) + (Q)]/(1-F + xF)(Cp)$

where

x = % evaporated = 0.25, 0.5, 0.75, 1.0

F = (f/a) = (flow split)(.155)(phi)

Cp = .240 for F less than 0.5; = .245 for F greater than 0.5 (assumption for approximate calculations)

HLG = heat of vaporization = 474 BTU/lb.
ΔT = °F
Q = zero for the first stage of analysis

1. Location 1 $f/a = (.2)(.155)(phi)$

phi	x	ΔT1(°C)	T1-ΔT1(°C)	Pv(mm)	TDEW(°C)*
1.0	.25	8.7	18.0	5.2	-25.6
1.0	.5	17.3	9.4	10.3	-15.9
1.0	.75	25.7	1.0	15.5	- 9.8
1.0	1.0	34.0	-7.3	19.9	- 5.9
.8	.25	6.9	19.8	4.4	-27.8
.8	.5	13.8	12.9	8.1	-19.4
.8	.75	20.5	6.2	11.8	-13.9
.8	1.0	27.2	- .5	16.2	- 9.1

2. Location 2 $f/a = (.333)(.155)(phi)$

phi	x	ΔT2(°C)	T2-ΔT2(°C)	Pv(mm)	TDEW(°C)*
1.0	.25	14.7	17.5	4.6	-27.2
1.0	.5	29.1	3.1	8.8	-18.2
1.0	.75	43.0	-10.8	13.0	-12.5
1.0	.78	44.7	-12.5	13.3	-12.1
1.0	1.0	56.6	-24.4	16.8	- 8.6
.8	.25	11.7	20.5	3.4	-31.2
.8	.5	23.1	9.1	6.9	-21.7
.8	.75	34.3	-2.1	10.3	-15.9
.8	1.0	45.3	-13.1	13.7	-11.7

* Calculated from: $\text{Log}_{10}\ Pv = [-0.2185(A)/K]+B$
$A = 8978.8$ Range: -44°C to 224°C
$B = 8.639821$
$K = °K$

Source: CRC Handbook of Chemistry and Physics, 53rd edition, 1972-1973.

B. Specific Volume of Air

1. Location 1

$v = RT/p$

where

$p = (29.0)$ in hg $(70.73) = 2051.2$ psf

$T = (459.7 + 80) - \Delta T1 = (539.7° - \Delta T1)°R$

$R = 53.345$ ft-lb /°R

v = specific volume = cu ft/lb air

phi	x	T (air, °R)	v(cu ft/lb air)
1.0	0	539.7	14.04
1.0	25	524.0	13.63
1.0	50	508.6	13.23
1.0	75	493.5	12.83
1.0	100	481.1	12.51
with heat transfer to 80°,100°, and 120°F			
1.0	100	539.7	14.04
1.0	100	559.7	14.56
1.0	100	579.7	15.08
.8	0	539.7	14.04
.8	25	527.3	13.71
.8	50	514.9	13.39
.8	75	502.9	13.08
.8	100	490.8	12.76
with heat transfer to 80°, 100°,and 120°F			
.8	100	539.7	14.04
.8	100	559.7	14.56
.8	100	579.7	15.08

2. Location 2

$v = RT/p$

where

$p = 15$ in HgA $(70.73) = 1061.0$ psf

$T = (459.7 + 90°) - \Delta T2 = (549.7° - \Delta T2)°R$

$R = 53.345$ ft-lb/°R

phi	x	Tair	v
1.0	0	549.7	27.6
1.0	25	523.2	26.3
1.0	50	497.3	25.0
1.0	75	472.3	23.7
1.0	100	476.2	23.9

with heat transfer to 80°, 100°,and 120°F

1.0	100	539.7	27.14
1.0	100	559.7	28.14
1.0	100	579.5	29.15
0.8	0	549.7	27.6
0.8	25	528.6	26.6
0.8	50	508.1	25.5
0.8	75	487.9	24.5
0.8	100	470.6	23.7

with heat transfer to 80°,100°, and 120°F

0.8	100	539.7	27.14
0.8	100	559.7	28.14
0.8	100	579.7	29.15

C. Specific Volume of Fuel

1. Location 1

v = (RT/p) vapor

where

p = (29.0) in hg (70.73) = 2051.2 prf

T = (459.7 + 80) $-\Delta T1$ = (539.7° $-\Delta T1$)°R

m = 32.043

mR = 1544 ft-lb/°R
v = specific volume = cu ft/lb of vapor

1. Case 1

phi	x	T	v
1.0	0	539.7	12.68
1.0	25	524.0	12.31
1.0	50	508.6	11.95
1.0	75	493.5	11.59
1.0	100	481.1	11.30
	with heat transfer		
1.0	100	539.7	12.68
1.0	100	559.7	13.15
1.0	100	579.7	13.62
0.8	0	539.7	12.68
0.8	25	527.3	12.39
0.8	50	514.9	12.10
0.8	75	502.9	11.81
0.8	100	490.8	11.53
	with heat transfer		
0.8	100	539.7	12.68
0.8	100	559.7	13.15
0.8	100	579.7	13.62

2. Location 2

$$v = RT/p$$

where

$p = 15$ in HgA(70.73) = 1061 psf

$T = (459.7 + 90) - \Delta T2 = (549.7° - \Delta T2)°R$

$m = 32.043$

$mR = 1544$

phi	x	T	v
1.0	0	549.7	25.0
1.0	25	523.2	23.8
1.0	50	497.3	22.6
1.0	75	472.3	21.4
1.0	100	476.2	21.6
	with heat transfer		
1.0	100	539.7	24.51
1.0	100	559.7	25.42
1.0	100	579.7	26.33
0.8	0	549.7	25.0
0.8	25	528.6	24.0
0.8	50	508.1	23.1
0.8	75	487.9	22.2
0.8	100	470.6	21.4
	with heat transfer		
0.8	100	539.7	24.51
0.8	100	539.7	25.42
0.8	100	579.7	26.33

D. <u>Compute mass Volume</u>

V = (mass air)(v1)+(mass fuel)(x)(v2)+(Mass fuel)(1-x)(v3)

where

mass air = 6.452 lb (phi = 1)
mass fuel = Location 1: 20% (1 lb) = .2 lb for phi of 1.0; =.16 lb for phi of 0.8

= Location 2: 33.3%(1 lb) = .333 lb for phi of 1.0; = .266 lb for phi of 0.8

= Carbureted : 1 lb for phi = 1.0, 0.8 lb for phi = 0.8

x = % of fuel evaporated = 0, .25, .50, .75, and 1.0
v_1= Specific volume air (cu ft/lb)
v_2= Specific volume evaporated fuel (cu ft/lb)
v_3= Specific volume liquid fuel = .0201 cu ft/lb
V = Cubic feet of mixture per pound of fuel consumed
v_c= Specific Volume of gaseous portion of mixture
v_c= V/[(mass air) + (x) (mass fuel)]

Table AII-D-1

Specific Volume of Partially Fumigated Mix at Location 1

phi	x	Mv1	Mxv2	M(x-1)v3	V	Specific Volume of Mix (vc)
1.0	0	90.59	0	.004	90.59	14.04
1.0	.25	87.94	.62	.003	88.56	13.62
1.0	.50	85.36	1.20	80.33	96.56	13.21
1.0	.75	82.78	1.74	.001	84.52	12.80
1.0	1.00	80.71	2.26	0	82.97	12.47
	with heat transfer 80, 100, 120°F					
1.0	1.00	90.59	2.54	0	93.13	14.00
1.0	1.00	93.94	2.63	0	96.57	14.52
1.0	1.00	97.30	2.72	0	100.02	15.04
0.8	0	90.59	0	.0032	90.59	14.04
0.8	.25	88.46	.50	.0024	88.95	13.06
0.8	.50	86.39	.97	.0016	87.36	13.37
0.8	.75	84.39	1.42	.0008	85.81	13.06
0.8	1.00	82.33	1.84	0	84.17	12.73
	with heat transfer 80, 100, 120°F					
0.8	1.00	90.59	2.03	0	92.61	14.01
0.8	1.00	93.94	2.10	0	96.04	14.53
0.8	1.00	97.30	2.18	0	99.48	15.05

Table AII-D-2

Specific Volume of Partially Fumigated Mix at Location 2

phi	x	Mv1	Mxv2	M(x-1)v3	V	Specific Volume of Mix (vc)
1.0	0	178.08	0	.0067	178.08	27.60
1.0	.25	169.69	1.98	.0050	171.67	26.27
1.0	.50	161.30	3.76	.0033	165.06	24.94
1.0	.75	152.91	5.34	.0017	158.25	23.61
1.0	1.00	154.20	7.19	0	161.40	23.79
	with heat transfer to 80°, 100°, and 120°F					
1.0	1.00	175.08	8.16	0	183.24	27.01
1.0	1.00	181.56	8.46	0	190.03	28.01
1.0	1.00	188.05	8.77	0	196.82	29.01
0.8	0	178.08	0	.0054	178.08	27.60
0.8	.25	171.62	1.60	.0040	173.22	26.57
0.8	.50	164.53	3.08	.0027	167.61	24.45
0.8	.75	158.07	4.44	.0013	162.51	24.43
0.8	1.00	152.91	5.69	0	158.60	23.61
	with heat transfer to 80°, 100°, and 120°F					
0.8	1.00	175.08	6.52	0	181.60	27.03
0.8	1.00	181.56	6.76	0	188.32	28.03
0.8	1.00	188.05	7.00	0	195.05	29.03

Table AII-D-3

Specific Volume of Carbureted Mix
with Complete Fraction of Fuel and
a Charge-Air Temperature of 100°F

phi	v	Mv1	Mxv2	M(x-1)v3	V	Vc	VB
				Location 1			
1.0	1.00	93.92	13.15	0	107.06	14.37	14.56
0.8	1.00	93.92	10.52	0	104.43	14.40	"
				Location 2			
1.0	1.00	181.56	25.42	0	206.98	27.78	27.60
0.8	1.00	181.56	20.33	0	201.90	27.84	"

Table AII-D-4

Specific Volume of Partially Fumigated
Mix (all locations) vs. Fraction Evaporated

Phi	x	Location 1 VB	Location 1 Vc	Location 2 VB	Location 2 Vc
1.0	0	14.04	14.04	27.60	27.60
"	.25	"	13.62	"	26.27
"	.50	"	13.21	"	24.94
"	.75	"	12.80	"	23.61
"	1.00	"	12.47	"	23.79
.8	0	14.04	14.04	27.60	27.60
"	.25	"	13.70	"	26.57
"	.50	"	13.37	"	25.45
"	.75	"	13.06	"	24.43
"	1.00	"	12.73	"	23.61
with heat transfer (assumed charge temperature)					
1.0/80°F	100	14.04	14.00	27.60	27.01
1.0/100°F	"	14.56	14.52	28.14	28.01
1.0/120°F	"	15.08	15.04	29.15	29.01
0.8/80°F	100	14.04	14.01	27.60	27.03
0.8/100°F	"	14.56	14.53	28.14	28.03
0.8/120°F	"	15.08	15.05	29.15	29.03

Appendix III

Volumetric Efficiency Change Due to Adiabatic Temperature Drop

In order to compare the effects of the adiabatically cooled charge on volumetric efficiency, it would be useful to perform this comparison at a constant manifold pressure (which is also consistent with the analysis in Reference 1). If we consider the Bernoulli equation of the form;

(AIII-1) $0.5d_1(V_1)^2 + P_1 = 0.5d_2(V_2)^2 + P_2$*

we can rearrange it to,

(AIII-2) $P_1 - P_2 = 0.5d_2(V_2)^2 - 0.5d_1(V_1)^2$

If we consider condition (1) to be at ambient conditions, then the term $(P_1 - P_2)$ would be our constant manifold vacuum (MV), and the V_1 term would be zero. Therefore Equation AIII-2 would become

(AIII-3) $MV = 0.5d_2(V_2)^2$

Now we can evaluate the effects of charge-air cooling on the velocity (V) term by forming a ratio of the baseline case (B) to the cooled case (C).

(AIII-4) $MV_C/MV_B = [0.5\ w(V_2)^2]_C/[0.5\ w(V_2)^2]_B$

Since $MV_B = MV_C$ we can change (AIII-4) to,

(AIII-5) $[(V_2)^2]_C/[(V_2)^2]_B = w_B/w_C$

Also recognizing that the weight density (w) can be expressed as the inverse of the specific volume, we have Equation AIII-6 which says the velocity change in the system is inversely proportional to the square root of the ratio of the specific volumes.

(AIII-6) $[V_2]_C = [V_2]_B\ (v_C/v_B)^{0.5}$

The mass flow change due to the charge cooling can now be calculated from the information on the velocity change and the specific volume change. This mass change can then be related to a change in volume flow at the inlet to the induction system which is at constant temperature and pressure. First, the mass rate (M) can be described as,

*d = mass density = weight density/gravitational constant
= w/g.
v = velocity.

(AIII-7) $M = (d)(V)(A)$

Next, we can look at the change in mass from the baseline (M_B) due to charge cooling (M_C) by forming a ratio,

(AIII-8) $M_C/M_B = [dVA]_C/[dVA]_B$

Cancelling the areas (A) in AIII-8, and substituting for V_C from AIII-6, we have

(AIII-9) $M_C/M_B = (d_C V_B)(v_C/v_B)^{0.5}/(d_B V_B)$

Substituting for d ($d = w/g$) and cancelling V_B, we have

(AIII-10) $M_C/M_B = w_C\ (v_C/v^B)^{0.5}/w_B$

Substituting the specific volume (v) for the density (w) ($w = 1/v$),

(AIII-11) $M_C/M_B = (v_B/v_C)(v_C/v_B)^{0.5}$

Equation AIII-11 describes the change in mass flow at constant manifold vacuum due to charge cooling in terms that we have calculated in Section D of Appendix II (specific volume). Note that the specific volume of the cooled charge (v_C) is the specific volume for a mix of vaporized fuel plus air while the specific volume of the base case (v_B) is for air alone. Therefore, the mass flow change includes the change in the mass of air, and the change in the mass of fuel.* In order to identify the change in volumetric efficiency, we must separate out the change in the mass of the intake air from the total mass change. This can be determined essentially on a unit ratio basis where the original air mass is unity, and the portion of the change in air mass (M_{AC}) is the ratio of the mass of air (a) to the sum of the mass of air (a) plus the mass of fuel that has evaporated (xf).

(AIII-12) $M_{AC} = (M_C/M_B)(a/a + xf)$

or,

(AIII-13) $M_{AC} = (M_C/M_B)[1/(1 + xf/a)]$

Since we considered the old air mass equal to unity, we can now compute the change in air mass (CAM) between the two conditions.

*The change in mass of fuel refers to the gaseous portion and hence is essentially governed by the fraction evaporated. Liquid fuel is neglected.

(AIII-14) $CAM = M_{AC} - 1$

or,

(AIII-15) $CAM = (v_B/v_C)(v_C/v_B)^{0.5}[a/(a + xf)] - 1$

We can use this change in mass flow to compute the change in volume flow at the inlet to the system (i.e., at the air flow meter). Recognizing that the change in mass flow is uniform across the system, the change at the inlet can be written as a ratio of the difference between the change in air flow due to cooling (M_{AC}), and the baseline case, or

(AIII-16) $CAM = (M_{AC} - M_B)/M_B$

Using the continuity Equation (AIII-7) and inlet conditions, we can rewrite AIII-16 as,

(AIII-17) $CAM = [(dVA)_{ACi} - (dVA)_{Bi}]/[dVA]_{Bi}$

At the inlet to the system, the temperature and pressure for all practical purposes is the same between the charge cooled case, and the baseline case because all of the potential charge cooling would occur farther downstream. Hence the inlet density for the charged cooled case is essentially the same, and cancels in AIII-17 leaving only the velocity term (V) and the area term (A). The product of the velocity (V) and area (A) happens to be the volume flow rate (Q). The result of substituting this identity into AIII-17 provide us with a change in volume flow rate

(AIII-18) $CAM = (Q_{ACi} - Q_{Bi})/Q_{Bi} = CQ_i$

If we define volumetric efficiency (VE) as inlet volume flow rate (Qi) divided by engine displacement per unit time, the change in volumetric efficiency (CVE) would be equivalent to the change in volume flow rate (CQi).

(AIII-19) VE = Qi/(Displ./Time)

(AIII-20) CVE = CQi

(AIII-21) CVE = CAM

Substituting the conditions resulting from any charge-air cooling for CAM (AIII-15) into equation AIII - 21, we now have a function for the change in volumetric efficiency (CVE) as a function of the specific volume resulting from the temperature change due to fuel evaporation, and the amount of fuel that has evaporated.

(AIII-22) $CVE = (v_B/v_C)(v_C/v_B)^{0.5}[a/(a + xf)] - 1$

Appendix IV

Observed Volumetric Efficiency Loss

The volumetric efficiency (VE) change is based on a constant pressure drop across the throttle (i.e., constant manifold vacuum, MV). Analysis of the test results from Figure 3 in Appendix VII indicate the following relationships.

Baseline:	VE = -2.43 MV + 70.50	(RSQ = .96)
Location 1:	VE = -2.57 MV + 71.17	(RSQ = 1.0)
Location 2:	VE = -1.99 MV + 63.44	(RSQ = .77)

Substituting a range of manifold vacuums from 10 to 15 inches Hg, we arrived at the following tables:

Table AIV-1

Volumetric Efficiency vs. Manifold Vacuum

MV	Baseline	Location 1	Location 2
10	34.05	32.62	33.15
12.5	40.13	39.05	38.13
15	46.2	45.47	43.10

Table AIV-2

Volumetric Efficiency Changes (from Base)

MV	CVE (%)-Location 1	CVE (%)-Location 2
10	-1.6	-6.7
12.5	-2.7	-5.0
15	-4.2	-2.6
Average Loss	2.8	4.1

Appendix V

Heat Transfer Calculation

I. Equation for correlation coefficient (C) by Sieder and Tate (4)

(AV-1) $(Tb2 - Tb1)(D)(Pr)^{.667}(u)^{-0.14}/4(L)[(To - Tb)ln] = C$

where

$[(To - Tb)\ ln] = (A - B)/[ln(A/B)]$

A = To1 - Tb1
B = To2 - Tb2
Pr = (Cpu/k) @ Tb
u = (u @ Tb/u @ To)

II. Boundary Conditions

A. Boundary Conditions for All Locations

1. Air flow = 19.6 CFM = 1176.4 CFH
2. k = .014 (approx. = air) (conduction k)
3. Cp = .240 for air; = .245 for phi of 1 @ 60°F (ref. 2)
   a. Cp = (R/J)[k/k-1]
   b. R = mR/m = 1544/m
   c. m = (% fuel)(phi)(32.043) + [1-(% fuel)(phi)](28.85)
   d. k = 1.4 for air (ratio of specific heats)
   e. k = 1.38 for menthanol/air at phi = 1 (ref. 2)
4. Geometry similar to tubular heater
5. Re, Pr, Nu numbers computed at bulk temperature (Tb) = (.5) (Tb1 + Tb2)

B. Boundary Conditions for Location 1

1. For phi = 1

a) Tb1 = 19°F
b) Tb = 50°F
c) ub = 3.7 (E-7) (32.17) (3600) lbm/ft.hr @ 50°F
d) uo = 3.9 (E-7) (32.17) (3600) lbm/ft.hr @ 80°F
e) Cp = .240
f) To1 = 80°F
g) To2 = 160°F
h) Tb2 = 80°F
i) Re neglects fuel mass flow.

2. For phi = .8

a) Tb1 = 31
b) Tb = 56
c) ub = 3.7 (E-7) (32.17) (3600) lbm/ft.hr @ 50°F

d) uo = 3.9 (E-7) (32.17) (3600) lbm/ft.hr @ 80°F
e) Cp = .240
f) To1 = 80°F
g) To2 = 160°F
h) Tb2 = 80°F
i) Re neglects fuel mass flow

C. <u>Boundary Conditions for Location 2</u>

1. <u>For phi = 1</u>

a) Tb1 = 7.5°F
b) Tb2 = 90°F
c) Tb = 48.7°F approx = 50°F
d) ub = 3.7 (E-7) (32.17) (3600) lbm/ft.hr @ 50°F
e) uo = 3.95 (E-7) (32.17) (3600) lbm/ft.hr @ 90°F
f) To1 = 180°F
g) To2 = 240°F
h) Re neglects fuel mass flow
i) Cp = .241 for k est. @ 1.395

2. <u>For phi = .8</u>

a) Tb1 = 8.4°F
b) Tb2 = 90°F
c) Tb = 49.2°F approx = 50°F
d) ub = 3.7 (E-7) (32.17) (3600) lbm/ft.hr @ 50°F
e) uo = 3.95 (E-7) (32.17) (3600) lbm/ft.hr @ 90°F
f) To1 = 180°F
g) To2 = 240°F
h) Re neglects fuel mass flow
i) Cp = .241 for k est. @ 1.395

III. Calculations

A. Location 1, phi = 1

° $L/D = [(Tb2 - Tb)(Pr)^{.667}(u)^{-0.14}]/4(C)[(To - Tb)ln]$

$-(Pr)^{.667} = (Cp\ u/k)^{.667} = [(.240)(3.7)(E-7)(32.17)(3600)/(0.14)]^{.667}$

$-\ (u)^{-0.14} = (ub/uo)^{-0.14} = (3.7/3.9)^{-0.14} = 1.0074$

° $[(To - Tb)ln)] = (A - B)/[ln(A/B)] = 70.1$

$-\ A = To1 - Tb1 = 80-19 = 61$

$-\ B = To2 - Tb2 = 160-80 = 80$

° $Re = Dvd/u$

Where

d = slugs/cu. ft. = 2.3(E-3) slug/cu. ft. @ 59°F

v = ft./sec.

u = lb.-sec./sq. ft.

D = duct diameter.

° For various estimates of intake duct hydraulic diameter

D (in.)	D ft.)	A (sq.ft.)	v (ft.sec.)	Re	C
2	.167	.022	14.989	1.56(E+4)	.00375
2.5	.208	.034	9.586	1.28(E+4)	.00380
3.0	.25	.049	6.662	1.04(E+4)	.00395

° $L = (D/C)(K)(Tb2 - Tb1)/4[(To - Tb)ln]$

$-\ K = .8021$

° For various estimates of intake duct hydraulic diameter

D (in.)	Estimate of L Required to Achieve Tb2	L (hardware measurement)
2.0"	95.2"	80"
2.5"	117.4"	80"
3.0"	137.2	80"

B. Location 1, phi .8

- Pr, Re, C, and u are the same as in Case 1, phi = 1.0
- [(To - Tb)ln] = (A - B)/[ln(A/B)] = 63.2

  - A = To1 - Tb1 = 80-31 = 49

  - B = To2 - Tb2 = 160-80 = 80
- For various estimates of intake duct hydraulic diameter

D (in.)	Estimate of L Required to Achieve Tb2	L (hardware measurement)
2.0"	84.8"	80"
2.5"	104.6"	80"
3.0"	122.3	80"

C. Location 2, phi = 1

- D of intake track runner = 1.35 in. = .1125 ft.
- Q per runner = Q/4 = (.327 ft.^{3}/sec.)/4 = .0818 ft.^{3}/sec.
- v = 8.22 ft./sec.
- Re = dvD/u = 5.748 E+3
- C = .004
- $Pr^{.667}$ = .7376
- $(u)^{-.14}$ = 1.009
- K = $(Pr)^{.667}$ $(u)^{-.14}$ = .7442
- [(To - Tb)ln] = (A - B)/[ln(A/B)] = 161

  - A = To1 - Tb1 = 172.5

  - B = To2 - Tb2 = 150
- L = (D/C)(K)(Tb2 - Tb1)/4[To - Tb)ln]
- L (calculations) = 32.2 in.
- L (hardware measurement) = 14 in. (approx.)

D. Location 2, phi .8

- Pr, Re, C, D, and u are the same as in Case 2, phi .8
- [(To - Tb)ln] = (A - B)/[ln(A/B)] = 160.5

  - A = To1 - Tb1 = 171.6

  - B = To2 - Tb2 = 150
- L (calculated) = 31.9 in.
- L (hardware measurement) = 14 in. (approx.)

Appendix VI

Changes in Pumping Work Due to Temperature Changes

It is generally accepted that if the air-charge of an engine is cooled, the volumetric efficiency is increased. The question is then what is the general magnitude of the differences between the work to pump cold air versus hot air.

As indicated in other appendices, if fuel is used to cool the charge, the volumetric efficiency may or may not improve. For the purposes of this appendix, we will assume that there is no fuel in the air charge, and that the air charge is cooled by some means other than fuel evaporation.

The simple model we will use is a long tube of constant cross section connected to a pump which exits to the atmosphere. In order to consider only the work necessary to pump a given quantity of fluid, and not the work to speed up the air, we will assume the pump exit area to be the same as the long inlet tube. The following conditions will be assumed:

Position 1 (inlet)

$P_1 = Pa$
$V_1 = 0$

Position 2 (inside tube)

$V_2 = 9.586$ ft/sec (from Section III, Appendix V)
$D_2 = 2.5$ in $= .208$ ft (from Section III, Appendix V)

Position 3 (Exit)

$P_3 = Pa = P_1$
$V_3 = V_2$
$D_3 = D_2$

Equation AVI-1 (pgs. 213, 216, Reference 4) is a restatement of the Bernoulli equation, and in effect says that the work needed to pump a given amount of fluid across our simple system from position 1 to position 3 is equal to the work needed to change the velocity, plus the work needed to change the hydraulic head, plus the work needed to change the pressure, plus the work needed to overcome friction.

$$\text{(AVI-1)} \quad W = (.5)V^2 - g\ h-(RT/M)\ln(P_3/P_1)-(.5V^2Lf)/R$$

The change in velocity is simply the outlet velocity minus the inlet velocity.

$$(.5)V^2 = .5(V_3)^2 - .5(V_1)^2$$

$$V_1 = 0$$

$$V_3 = V_2$$

(AVI-2) $$(.5)V^2 = .5(V_2)^2$$

The change in hydraulic head is zero.

(AVI-3) $$h = 0$$

The change in pressure is zero

$$P_1 = P_2$$

(AVI-4) $$\ln (P_1/P_2) = \ln 1 = 0$$

If we look at the change in work per unit length at position 2, the unit work due to friction is

$$= (.5)(V_2)^2(1)(f)/(.5)(D_2)$$

(AVI-5) $$= (V_2)^2(f)/D_2$$

Substituting AVI-2 to 5 into AVI-1, provides for the work per unit length to pump fluid through our simple system,

(AVI-6) $$W = -(.5)(V_2)^2 - (V_2)^2(f)/D_2$$

The friction factor (f) is related to the Reynold's No.

$$Re = DVd/u$$
$$d = P/RT^*$$
(AVI-7) $$u = (.3170)(E-10)(T)^{1.5}[734.7/(T+216)]^{**}$$

From Appendix II, we can assume a dry air temperature of 80°F at condition 2. The lowest dew point of the fuel air mixture was around -12°C (10.4°F). Therefore, we will assume that we can cool dry air to that temperature for this comparison. Using these assumptions results in the following:

Temperature	u	d	Re
80°F	3.86(E-7)	2.214(E-3)	1.42(E+3)
10.4°F	3.46(E-7)	2.542(E-3)	1.82(E+3)

* d = mass density = w/g.
**T = °R, from Reference (7).

Since the Reynold's No.s calculated are in the laminar range, the friction factor (f) becomes,

(AVI-8) $f = 16/Re$

Therefore, the work Equation (AVI-6) becomes,

(AVI-9) $W = -(.5)(V_2)^2 - (V_2)^2(16)/(D)(R_e)$

To determine the magnitude of the difference in work per unit length to pump hot air (W_H) versus cold air (W_C), a simple ratio can be formed with Equation AVI-9, and the appropriate values for position 2 and the Reynold's No. can be substituted.

(AVI-10) $W_C/W_H = Z$

or,

(AVI-11) $W_C = .9785\ W_H$

Because the velocity (V_2) and area (A_2) remained the same between the comparison of the required pumping work with hot air to the required work with cold air, the results of Equation (AVI-11) are applicable to constant volume flow-rate. In this case, artifically reducing the temperature of dry air by approximately 70°F, reduced the estimated work per unit length by slightly more than 2 percent.

However, power is a function of mass of air flow, not volume. If we want to hold mass flow (M) constant for this exercise, then Equation AVI-13 tells us that mass density (d) and velocity must vary with temperature.

(AVI-13) $M = dVA$

Since we have previously calculated the change in mass density (d) with temperature, we can determine the subsequent change in velocity by forming a ratio of the cold mass flow rate to the hot mass flow rate. Cancelling terms, we have

(AVI-14) $V_C/V_H = d_H/d_C = .87097$

Substituting AVI-14 into AVI-9 and 10 results in the ratio of work per unit length to pump the same mass flow at two different temperatures.

(AVI-15) $W_C/W_H = .87097\ (Z)$

or,

(AVI-16) $W_C = .8522\ W_H$

The results of AVI-16 indicate that the work to pump the same mass of cold fluid is approximately 15 percent less than that required for a hot fluid. But remember that this is for air that was cooled by some means other than fuel evaporation. Also this was a very simple example that did not consider the intricacies in the geometry of a real engine. The potential effects of the practical engine cycle (intake, compression, power, exhaust, and valve overlap) were similarly not considered. And finally the myriad of effects involved in pulsating flow were ignored. Nonetheless, the general trend is that if you can cool the air by some means other than adding fuel, it will require less power during the intake stroke.

The amount of work reduction is, however, obviously not only dependent on the percentage reduction, but also on the base level of intake work. Furthermore, the relation of the intake work to the other losses in the engine cycle must be of sufficient magnitude, in order for a small percentage improvement in intake work due to a cooler charge to be measureable in the overall engine performance.*

*Note: engine power increases due to a cooler intake charge are essentially due to an increase in charge mass density which increases the mass throughput of the engine.

Appendix VII

EPA MEMO: Results of Methanol
Fumigation Investigations

UNITED STATES ENVIRONMENTAL PROTECTION AGENCY

ANN ARBOR, MICHIGAN 48105

DATE: APR 22 1983

OFFICE OF
AIR, NOISE AND RADIATION

SUBJECT: Results of Methanol Fumigation Investigations

FROM: R. Bruce Michael Bruce Michael
Technical Support Staff

William Clemmens, Project Manager WB Clemmens
Technical Support Staff

TO: Charles L. Gray, Jr., Director
Emission Control Technology Division

THRU: Phil Lorang, Chief Phil Lorang
Technical Support Staff

The hypothesis of this study was: "Can methanol injected into the upstream air passages (additionally to the normal port injection) cool the inlet air sufficiently through vaporization to increase the volumetric efficiency, and will the expected increase in volumetric efficiency translate into improved thermal efficiency?" This hypothesis is derived from the mathematical equation of volumetric efficiency in which the efficiency is proportional to the mass of air per unit of time flowing through the engine relative to the swept volume of the engine during that time. Vaporization (through fumigation) in the inlet air should cause a temperature drop of the air, which would increase density and mass flow. To test this hypothesis, two methods of fumigation were tested, both occurring along with the normal port injection of the Nissan 2.0 litre NAPS-Z engine. Neither method improved volumetric or thermal efficiency. In fact, the normal port injection without fumigation gave slightly better results. Because none of the fumigation systems tested demonstrated any increase in efficiency at the lower power levels tested, we recommend that we abandon any plans for fumigation testing on this engine in the future.

Test Configuration

Fumigation of methanol was obtained by adding EFI style fuel injectors upstream of the standard port injectors. The fumigation injectors were controlled by the EFI computer in the same manner and simultaneously with the standard port injectors. The standard port injectors are located approximately 3 to 5 inches from the intake valve seat.

Three locations for the fumigation injectors were tested. The first location was in the intake manifold collector. The collector is a log style manifold downstream of the single throttle valve. Four intake runners enter the bottom of the collector. They are ranged in pairs - cylinder 1 and 2, and cylinders 3 and 4. The intake runners are on the order of 12 inches from the intake valve seat to the collector. The single fumigation injector was mounted on the top side of the collector, and roughly equally spaced between the intake runner pairs of cylinders 1 and 2 and cylinders 3 and 4 (i.e., between cylinders 2 and 3).

The second location was also in the collector, but two injectors were used instead of a single injector. The two injectors were located over the intake runner pairs - one injector over the pair of runners for cylinders 1 and 2, and the other injector over the pair of runners for cylinders 3 and 4.

The third location utilized a single injector placed upstream of the entrance to the EFI vehicle system. This injector was centrally located in a short section of 4 inch O.D. tubing connected to the downstream end of the intake air flow meter. This configuration is identified as "central injection" or "central fumigation".

Engine Operation

Engine operation with fumigation eminating from the first location was not satisfactory. Rough engine running, extreme sensitivity to ignition timing and ever present detonation were characteristics of this location. We speculate that the central location of the injector created a situation in which cylinders 2 and 3 robbed fuel from cylinders 1 and 4 creating a very lean condition in cylinders 1 and 4. Because of these problems, all fumigation testing at this location was terminated.

The second location for the fumigation injectors seemed (to a degree) to solve the assumed distribution problems, but on a qualitative basis the engine still did not operate quite as well as in the standard configuration (i.e., no fumigation). For instance, some of the tests at 90 ft.lb. and 2000 RPM were scrubbed from the test program because of an instability in the MBT point for ignition timing, which led to creeping detonation. Another handicap with the two injectors was that even with the adjustable air/fuel control box for the EFI, the sum of the fuel flow from the two fumigation injectors plus the four standard injectors made it difficult to achieve equivalence ratios leaner than 0.8. Operation at equivalence ratios richer than 0.8 would normally depress the brake thermal efficiency somewhat, but Table 1 and Figure 1 suggest that peak efficiency with the two injector set-up is slightly richer than 0.8. Directionally the shift to a richer point from the baseline (0.7 to 0.8) for peak efficiency is in the wrong direction.

Only the central injector seemed to operate as well as the standard configurations. No difference in driveability was observed. The only handicap encountered was that the central fumigation system was potentially capable of running leaner than our adjustable EFI control box would allow.

Results and Conclusions

The most interesting result of this testing was that upstream injection (fumigation) of methanol did not improve volumetric efficiency. In fact, as shown in Table 1 (and Figures 1 and 2), fumigation of methanol on this port injected engine actually decreased volumetric efficiency by 4 to 10 percent. In three out of four cases, the decrease in volumetric efficiency resulted in a noticeable loss in thermal efficiency (5-9%). The only case that showed an increase in thermal efficiency (0.5 percentage point), with a decrease in volumetric efficiency was the test point at high power using the central upstream injector (Inlet to EFI, Table 1). The trend of this increase in efficiency as shown in Figure 2 (central fumigation) is plateauing around an equivalence ratio of 0.8, the limit of our EFI control with the central injector. Potentially, improvement could be obtained at leaner equivalence ratios, but returning to Figure 1, we see that we were able to sufficiently enlean the system to get a peak in the efficiency trend with the central injector. At the lower power in Figure 1, the peak efficiency for the central injector was below the peak efficiency for the baseline.

It is not known exactly why fumigation did not increase volumetric efficiency, but information from the recent Nissan work may shed some light. Nissan was surprised to find that efficiency went down with methanol, as compared to gasoline, and theorized that fuel vaporization was caused mainly by absorbing heat from the cylinder walls rather than the air. This would, of course, result in little or no gain in charging efficiency since the incoming air would not change density. The lack of a density change results in a negative effect on volumetric efficiency for the following reason. When the fuel vaporizes, it increases the gas volume in the air stream and has the effect of restricting or slowing down the air flow. This lowers charging efficiency, more than offsetting any gain. While this is not exactly analogous to our fumigation work, it may be that the restriction of air flow (due to fumigation) was the dominant effect, which lowered the volumetric efficiency.

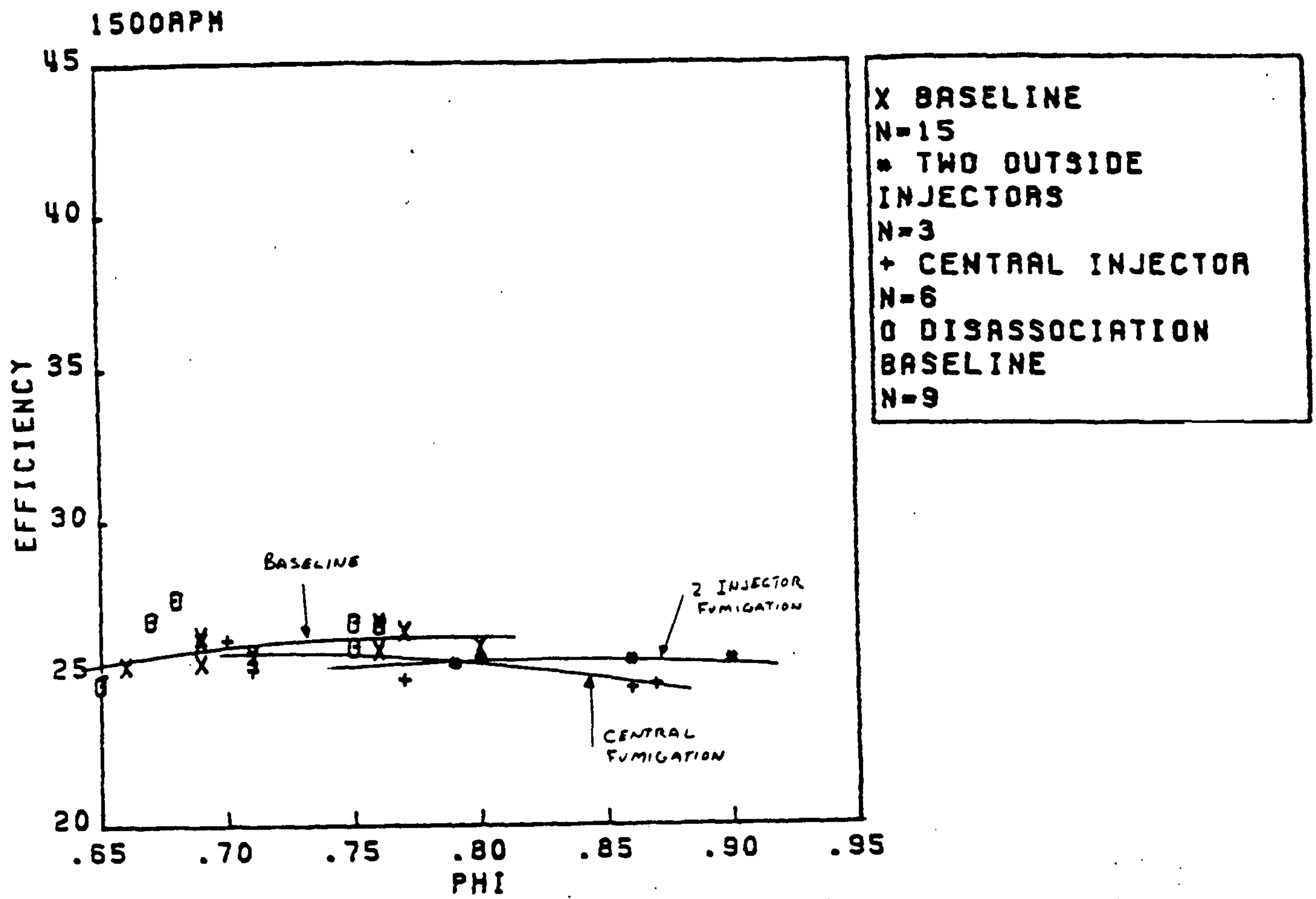
Table 1

Methanol Fumigation
Test Results

		Injector Position		
	Baseline*	Intake Collector 1 Injector	Intake Collector 2 Injectors	Inlet to EFI 1 Injector
90 ft-lbs @ 2000 RPM				
Best Average Efficiency	38.9	-**	35.5	39.4
Equivalence Ratio	.79	-	.93	.83
Volumetric Efficiency	69.3	-	65.5	65.1
Number of Tests	9		3	3
29.5 ft-lbs @ 1500 RPM				
Best Average Efficiency	26.4	-	24.4	25.2
Equivalence Ratio	.70	-	.86	.78
Volumetric Efficiency	38.0	-	33.8	35.5
Number of Tests	12		3	3

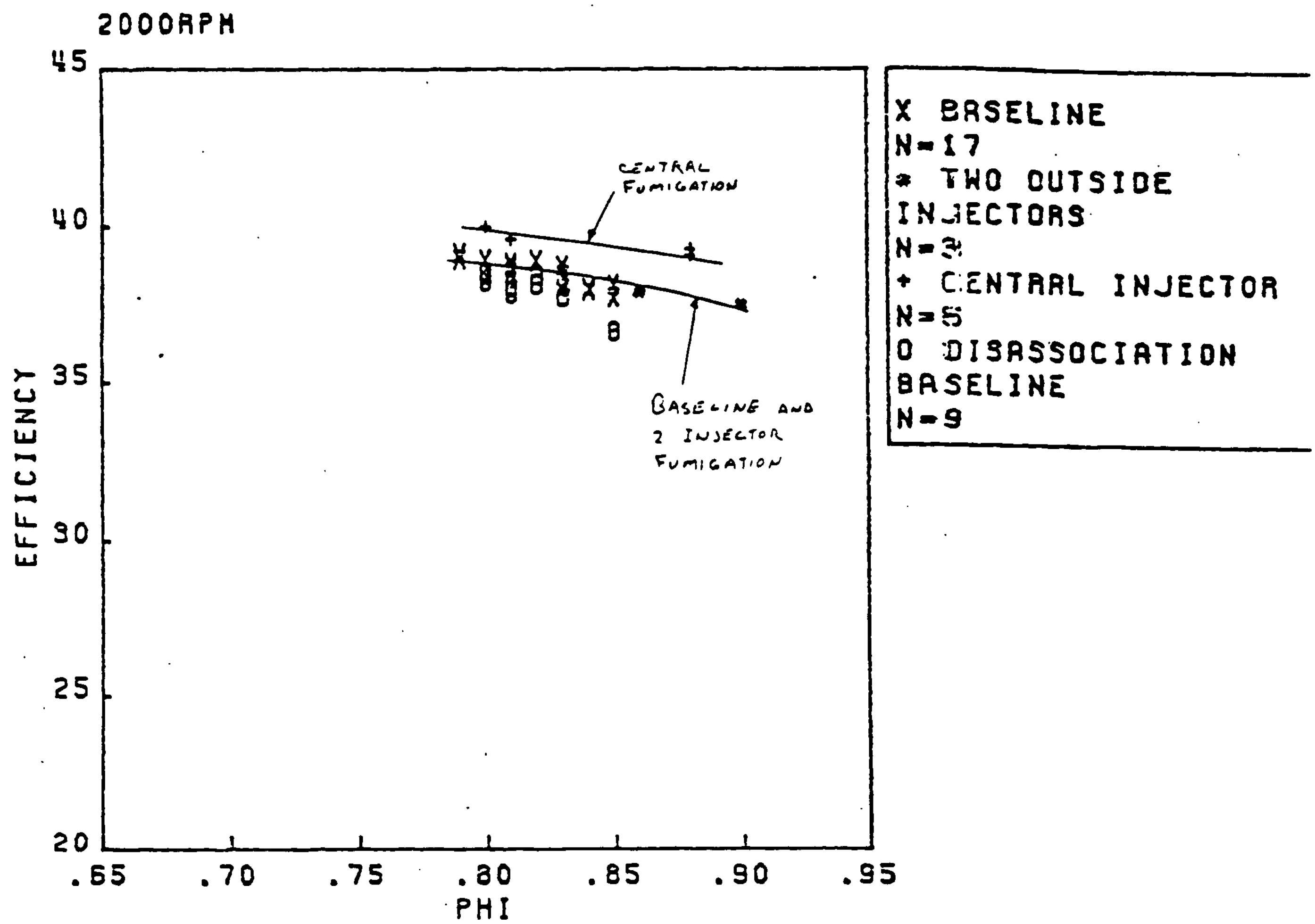
*Based on replicate tests before fumigation testing (V87,V90) and replicates after fumigation (ZM2-V0). The best average (thermal) efficiencies were calculated as follows. First, the data were separated into different equivalence ratio ranges for each version. Second, the average thermal efficiencies of the tests in each range for each version was calculated. Third, for the equivalence ratio range yielding the overall best efficiencies, the average efficiencies for the three versions were averaged. For the higher torque level, the equivalence ratio range which yielded the highest average efficiency was .77-.80. For the lower torque level, the range was .67-.73.

**All testing was aborted in this configuration due to severe detonation problems (no data taken).



Methanol Fumigation

Figure 1



Methanol Fumigation

Figure 2

Further analysis of the data was performed to compare volumetric efficiency to manifold air pressure (manifold vacuum). Theoretically, if the upstream fumigation of methanol cooled the inlet air, the volumetric efficiency should increase for the same manifold air pressure.

Figure 3 shows scatter plots and regression lines of volumetric efficiency versus manifold pressure (MAP) for the normal port injection (baseline) and the two fumigation versions. The regression lines, which are all nearly parallel, again show no advantage to fumigation.*

Figure 4 shows thermal efficiency (BTE) versus manifold pressure for the same tests. In all cases, for a given MAP, efficiency was higher in the baseline configuration, thus suggesting that methanol fumigation does not improve volumetric efficiency.

---

*We should mention that the MAP measurements were assumed to be "dry" measurements (no partial pressure of the fuel in the manifold), which is not really correct. The fumigation would really result in a somewhat wet condition in the manifold, making the measured MAP a wet measurement which is by nature greater than a dry measurement. Therefore, the fumigation points and regression lines really should be slightly to the left of those shown in Figure 3 which translates into worse results. No calculations were made to account for this, because the effect is slight.

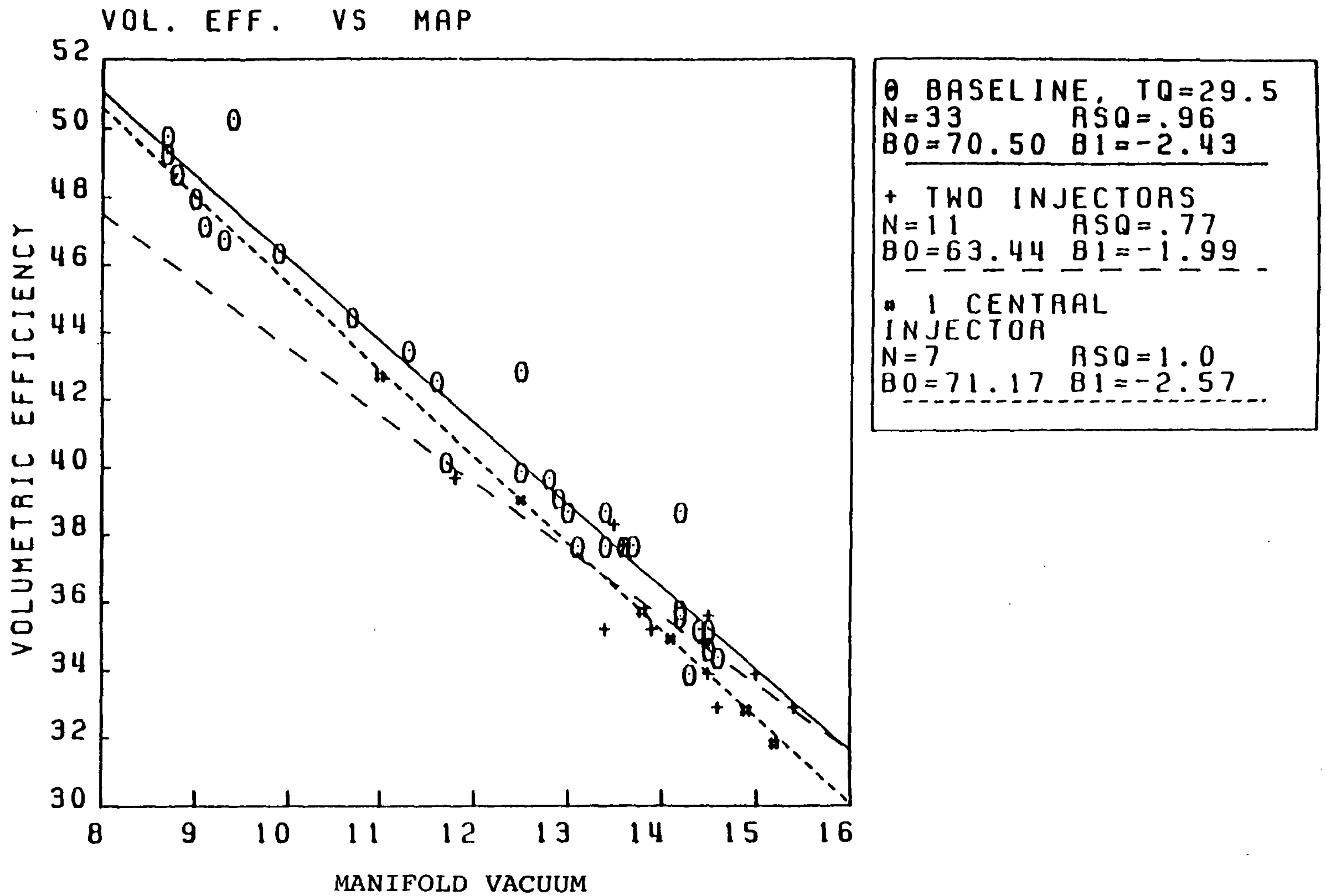

VOL. EFF. VS MAP
VOLUMETRIC EFFICIENCY
52
50
48
46
44
42
40
38
36
34
32
30
8
9
10
11
12
13
14
15
16
MANIFOLD VACUUM
0 BASELINE, TQ=29.5
N=33 RSQ=.96
B0=70.50 B1=-2.43
+ TWO INJECTORS
N=11 RSQ=.77
B0=63.44 B1=-1.99
* 1 CENTRAL INJECTOR
N=7 RSQ=1.0
B0=71.17 B1=-2.57

Figure 3

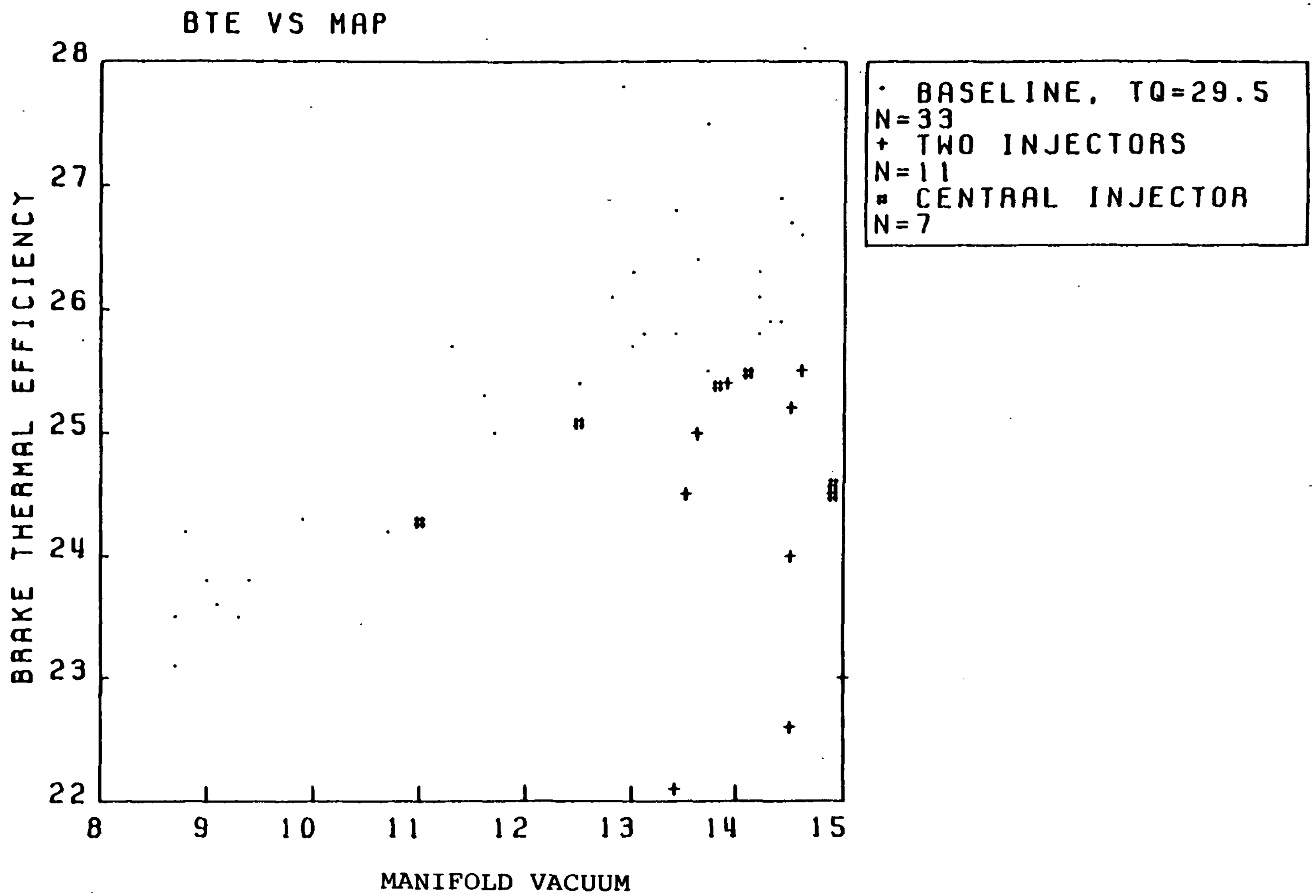
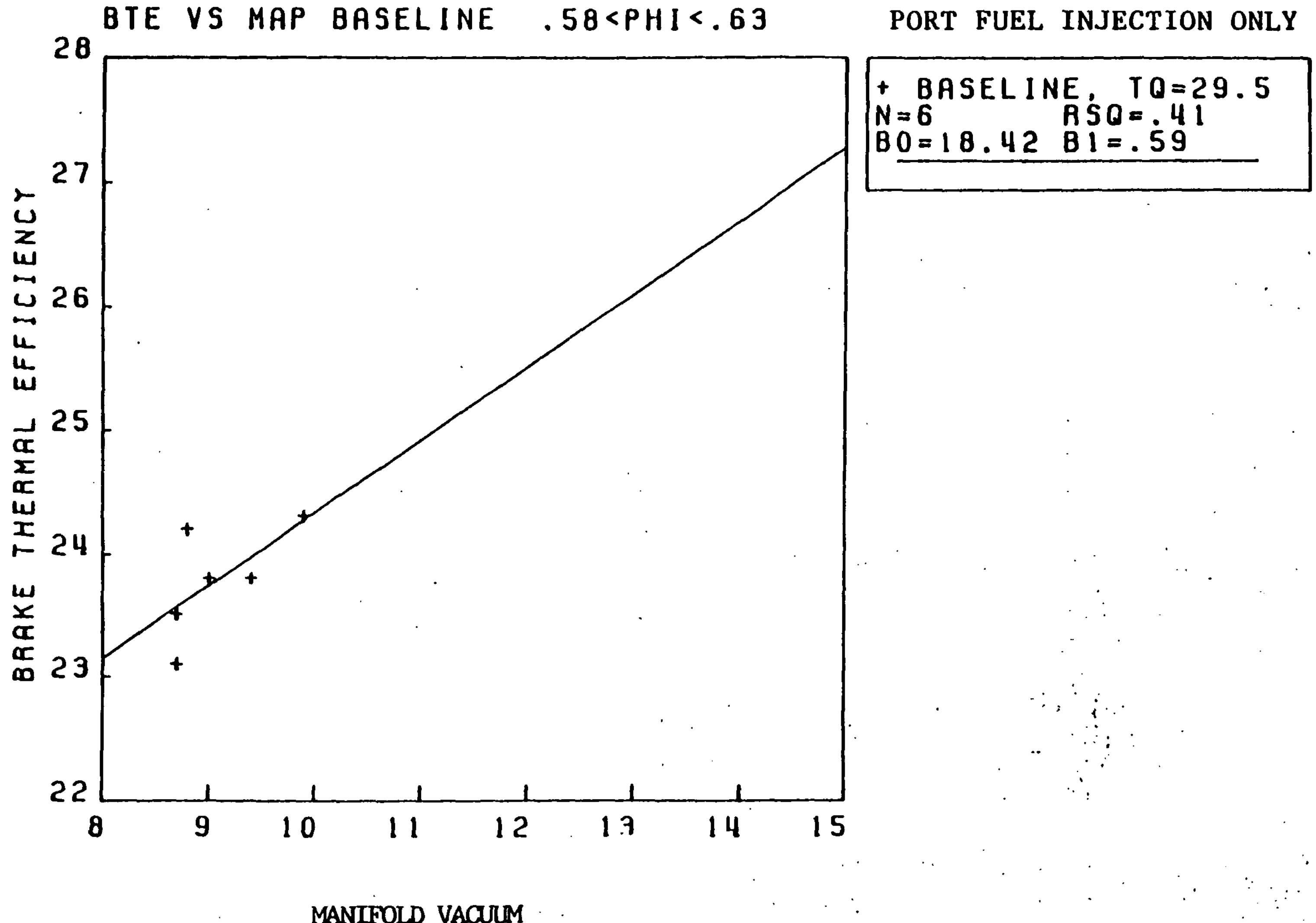

BTE VS MAP
BRAKE THERMAL EFFICIENCY
22
23
24
25
26
27
28
8
9
10
11
12
13
14
15
MANIFOLD VACUUM
· BASELINE, TQ=29.5
N=33
+ TWO INJECTORS
N=11
# CENTRAL INJECTOR
N=7

Figure 4

From another aspect, graphs of BTE vs. MAP can also be valuable. The air-fuel ratio was varied from an equivalence ratio (PHI) of about .6 to .8 for many test points. For the power output and rpm to remain constant, the throttle opening has to be varied somewhat inversely to the different PHI levels - a lower PHI requires a larger throttle opening, for example, which in turn lowers the throttling (pumping) loss because of less restriction of the air flow. Therefore, it can be determined from the results if throttling loss changes have an appreciable effect on thermal efficiency when leaner mixtures (leaner than .8) are used.

Results from baseline tests show that leaner mixtures (with lower throttling losses) did not improve thermal efficiency; in fact the reverse was true. Figures 5-8 show baseline (no fumigation) test data for four categories of PHI. Figure 5 includes test points at the lowest levels of PHI and MAP, which produced the lowest levels of BTE. Figure 6, which includes intermediate low values of PHI and MAP, shows mixed results ranging from nearly the lowest to the highest levels of BTE. PHI values in this category appear to be the transition from low to high BTE. The highest BTE values were from PHI at .66-.67 and the higher MAP values, while the lowest BTE values were from PHI at .63 and the lowest MAP values. Figures 7 and 8 show consistently high levels of BTE for PHI greater than .67 and MAP greater than 13 inches of mercury. Thus it appears that throttling losses are not dominant characteristics on this engine under these loads.

In summary, fumigation did not improve either volumetric or thermal efficiency and we therefore recommend not pursuing it further on this engine.


BTE VS MAP BASELINE .58<PHI<.63
PORT FUEL INJECTION ONLY
+ BASELINE, TQ=29.5
N=6 RSQ=.41
B0=18.42 B1=.59
BRAKE THERMAL EFFICIENCY
22
23
24
25
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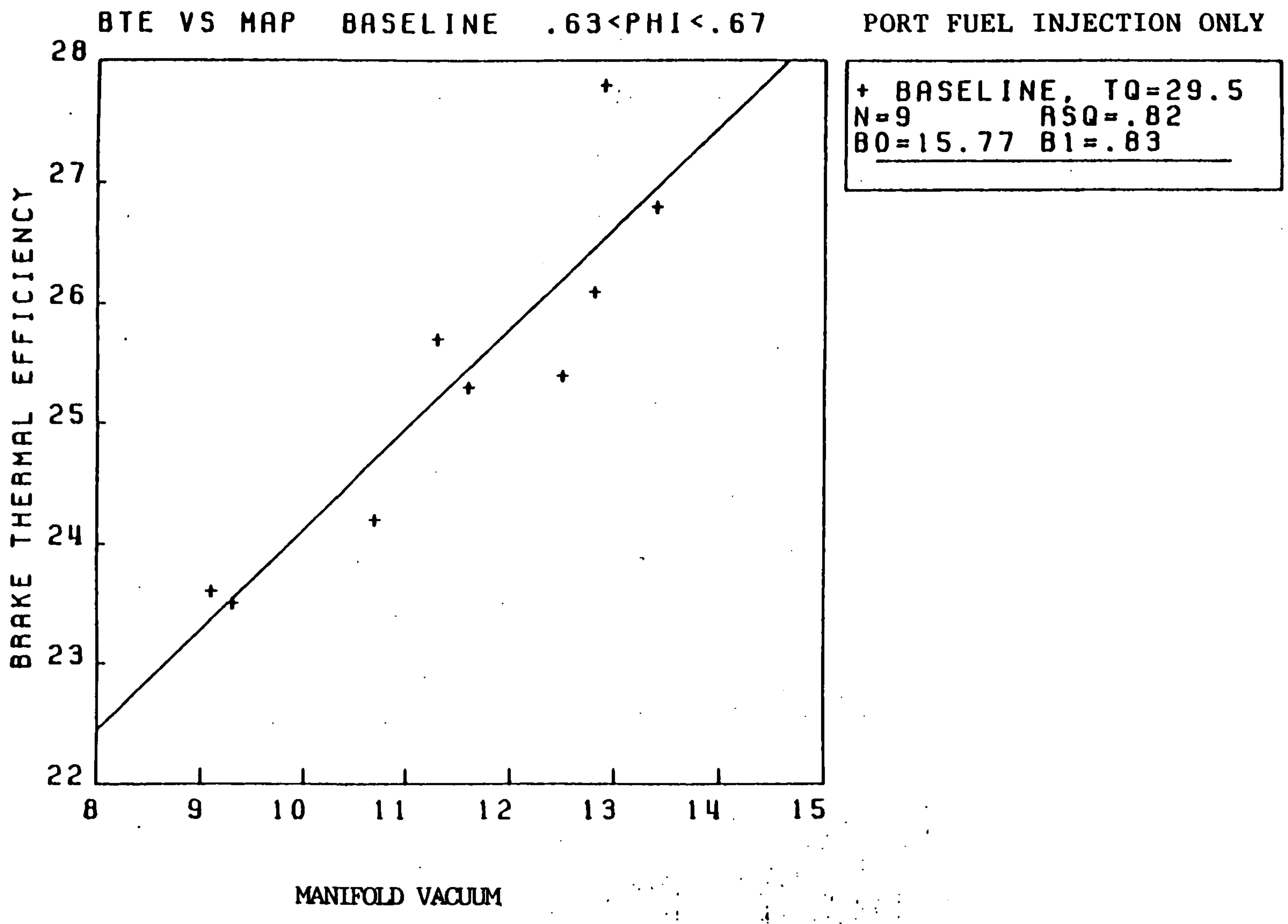
27
28
8
9
10
11
12
13
14
15
MANIFOLD VACUUM

Figure 5


BTE VS MAP BASELINE .63<PHI<.67
PORT FUEL INJECTION ONLY
+ BASELINE, TQ=29.5
N=9 RSQ=.82
B0=15.77 B1=.83
BRAKE THERMAL EFFICIENCY
22
23
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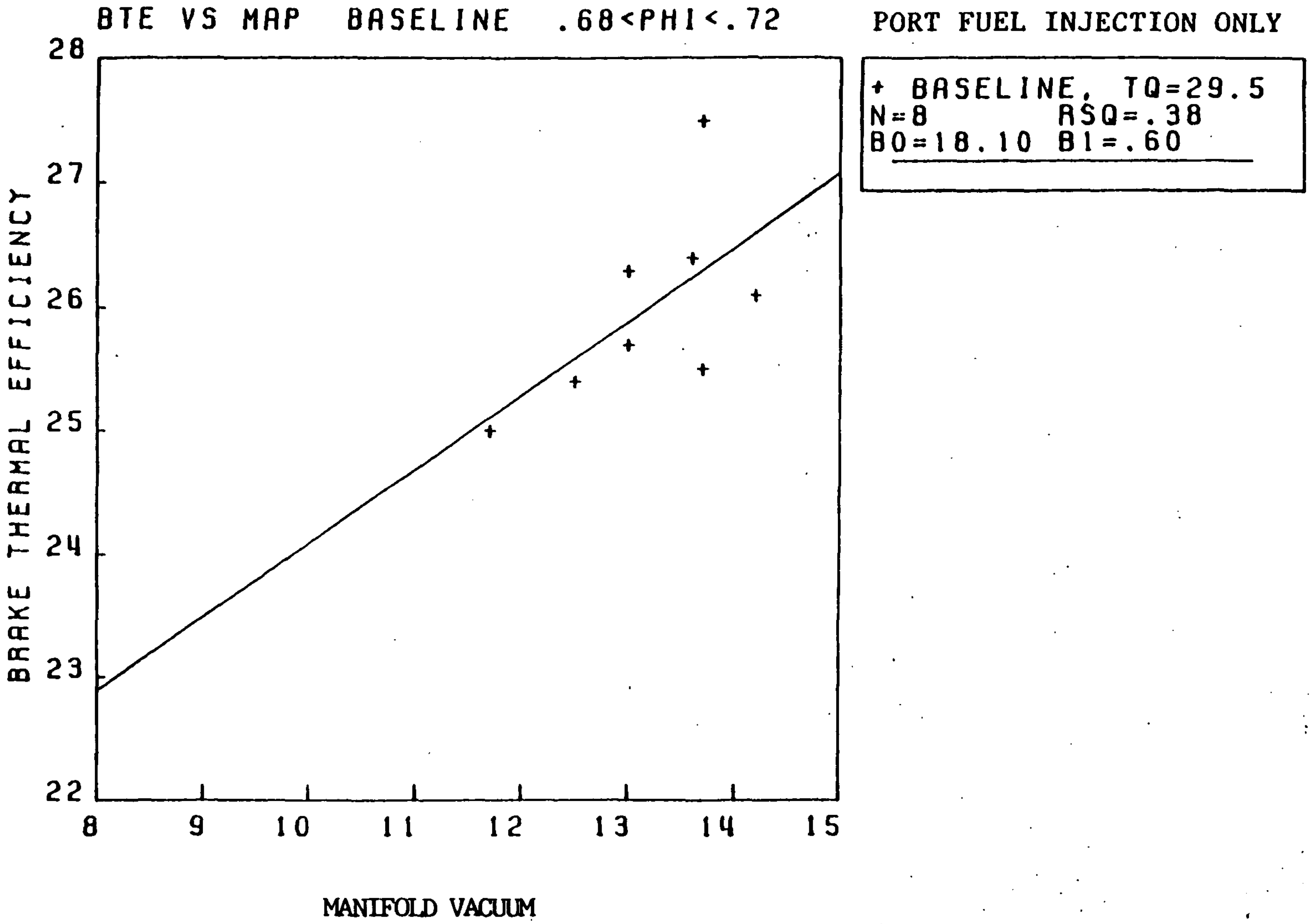
12
13
14
15
MANIFOLD VACUUM

Figure 6


BTE VS MAP BASELINE .68<PHI<.72
PORT FUEL INJECTION ONLY
+ BASELINE, TQ=29.5
N=8 RSQ=.38
B0=18.10 B1=.60
BRAKE THERMAL EFFICIENCY
MANIFOLD VACUUM
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13
14
15

Figure 7

Appendix VIII

Test Data
(Summary Sheets)

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.5	100	210	22.0B	0.	34.91	7.75	0.83	206.7	26.66	0.764	0.359	38.6	6.60	1.620	1.388	408.16	9.555		0.0
2000	91.0	101	110	23.5B	0.	35.12	8.21	0.79	217.9	26.54	0.756	0.355	39.0	6.53	1.878	2.537	398.25	7.505		0.0
2000	90.0	99	080	25.5B	0.	34.74	8.42	0.77	220.2	26.14	0.753	0.354	39.1	6.51	1.895	2.560	398.01	6.318		0.0
1500	39.5	100	1270	32.5B	0.	11.41	8.03	0.81	90.8	11.31	0.991	0.466	29.7	8.57	2.707	3.427	560.25	11.878		0.0
1500	30.0	76	1340	33.5B	0.	8.67	8.87	0.73	87.6	9.88	1.140	0.536	25.8	9.85	6.330	5.272	597.08	4.804		0.0
1500	30.0	76	1070	37.0B	0.	8.67	9.80	0.66	103.3	10.55	1.216	0.571	24.2	10.52	14.888	6.761	473.22	1.673		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	91.0	100	470	22.0B	0.	35.43	7.71	0.84	205.8	26.70	0.754	0.354	39.1	6.52	1.961	2.677	441.49	9.687		0.0
2000	90.0	99	110	25.5B	0.	35.07	8.26	0.78	215.7	26.12	0.745	0.350	39.6	6.44	2.384	2.510	433.92	8.678		0.0
2000	89.7	99	080	25.0B	0.	34.94	8.23	0.79	217.0	26.36	0.754	0.354	39.0	6.52	2.599	2.515	436.46	7.837		0.0
1500	29.5	100	1460	33.8B	0.	8.60	8.38	0.77	80.0	9.55	1.109	0.521	26.6	9.59	8.597	4.495	618.02	8.305		0.0
1500	29.5	100	1310	31.0B	0.	8.60	8.92	0.73	87.6	9.83	1.142	0.537	25.8	9.87	8.588	5.394	614.77	3.911		0.0
1500	28.0	95	1060	34.2B	0.	8.15	10.01	0.65	103.3	10.32	1.267	0.595	23.2	10.95	14.545	7.601	623.48	1.312		0.0

TEST NO.: HD-810858	ENGINE: 380 NAPS-ZM	90	TEST D/T: 9-21-82 9:45	REPORT D/T: 09-27-83 13:56	P062581

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.5	100	110	24.0B	0.	35.10	7.88	0.82	208.9	26.51	0.755	0.355	39.0	6.53	1.608	2.118	439.10	8.058		0.0
1500	29.5	100	1300	38.0B	0.	8.54	9.16	0.71	89.9	9.81	1.148	0.539	25.7	9.93	5.231	4.304	605.85	3.511		0.0

TEST NO.: HD-810859 ENGINE: 380 NAPS-ZM 90 TEST D/T: 9-21-82 10: 0 REPORT D/T: 09-27-83 13:55 P062581

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
															----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ----					
2000	90.0	100	140	24.0B	0.	34.65	7.82	0.83	205.8	26.33	0.760	0.357	38.8	6.57	1.346	2.149	335.40	8.523		0.0
2000	90.5	101	90	24.0B	0.	34.83	7.96	0.81	210.3	26.42	0.759	0.356	38.8	6.56	1.806	2.262	437.19	6.777		0.0
2000	89.5	99	80	27.0B	0.	34.45	8.17	0.79	211.2	25.85	0.750	0.353	39.2	6.49	1.904	2.247	433.69	6.662		0.0
1500	29.5	100	1450	36.0B	0.	8.51	8.56	0.76	80.4	9.39	1.104	0.519	26.7	9.54	4.149	3.259	600.13	4.909		0.0
1500	29.5	100	130	42.0B	0.	8.51	9.42	0.69	89.9	9.54	1.121	0.527	26.3	9.69	5.439	4.057	596.13	1.498		0.0
1500	29.5	100	910	48.0B	0.	8.51	10.30	0.63	109.6	10.64	1.250	0.587	23.6	10.80	17.268	8.327	594.76	0.910		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.0	100	12	22.0B	0.	34.88	7.79	0.83	208.0	26.70	0.765	0.360	38.5	6.62	1.351	1.823	451.18	9.107		0.0
2000	90.0	100	09	22.0B	0.	34.90	7.80	0.83	210.3	26.95	0.772	0.363	38.1	6.68	1.411	1.932	449.84	7.654		0.0
2000	89.3	99	08	24.0B	0.	34.50	7.96	0.81	211.2	26.53	0.769	0.361	38.3	6.65	1.542	1.987	446.63	8.247		0.0
1500	29.6	100	142	44.0B	0.	8.56	8.51	0.76	83.1	9.77	1.142	0.536	25.8	9.87	16.704	3.195	621.66	9.857		0.0
1500	29.5	100	125	47.0B	0.	8.54	9.34	0.69	92.6	9.91	1.161	0.545	25.4	10.03	7.044	4.570	613.92	4.772		0.0
1500	30.8	104	95	47.0B	0.	8.92	10.19	0.64	109.2	10.72	1.202	0.565	24.5	10.39	17.257	6.970	578.51	1.592		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.0	100	120	21.0B	0.	34.81	7.61	0.85	204.4	26.87	0.772	0.363	38.2	6.67	1.462	1.912	455.68	9.348		0.0
2000	89.5	99	100	23.0B	0.	34.62	7.71	0.84	206.7	26.80	0.774	0.364	38.0	6.69	1.583	2.014	450.69	8.974		0.0
2000	89.5	99	090	22.8B	0.	34.61	7.73	0.84	206.7	26.75	0.773	0.363	38.1	6.68	1.623	2.102	451.36	8.194		0.0
2000	89.5	99	080	22.8B	0.	34.61	7.57	0.85	204.4	26.99	0.780	0.366	37.8	6.74	1.637	2.114	455.56	8.200		0.0
1500	29.5	100	1430	38.0B	0.	8.56	8.07	0.80	78.6	9.74	1.139	0.535	25.9	9.84	2.882	3.091	624.25	9.696		0.0
1500	30.5	103	1340	42.5B	0.	8.84	8.42	0.77	83.1	9.87	1.116	0.524	26.4	9.65	3.580	3.433	604.45	7.237		0.0
1501	29.5	100	1170	55.5B	0.	8.56	9.26	0.70	93.5	10.09	1.178	0.553	25.0	10.18	8.550	5.355	609.22	4.131		0.0
1500	29.5	100	940	56.5B	0.	8.56	0.0	0.0	0.0	10.64	1.243	0.584	23.7	10.74	19.079	7.410	593.28	2.066		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) --- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.0	100	11	17.5B	0.	34.87	6.32	1.02	202.2	32.02	0.918	0.431	32.1	7.94	5.207	37.401	471.76	4.904		0.0
2000	91.0	101	08	16.0B	0.	35.20	6.33	1.02	202.2	31.95	0.908	0.426	32.5	7.85	4.103	32.175	474.61	5.049		0.0
2000	89.5	99	08	17.5B	0.	34.65	6.49	1.00	201.3	31.04	0.896	0.421	32.9	7.74	4.796	25.769	475.41	6.097		0.0
1500	29.0	100	145	39.5B	0.	8.42	7.16	0.90	78.6	10.98	1.304	0.613	22.6	11.27	15.957	55.446	625.82	9.828		0.0
1500	29.0	100	134	40.8B	0.	8.41	7.28	0.89	81.8	11.23	1.335	0.627	22.1	11.54	15.030	44.581	632.67	9.250		0.0
1500	30.0	103	1180	42.2B	0.	8.69	7.89	0.82	92.1	11.67	1.342	0.631	21.9	11.60	8.423	5.276	600.16	4.070		0.0

TEST NO.: HD-810863	ENGINE: 380 NAPS-ZM	91	TEST D/T: 9-23-82 9: 0	REPORT D/T: 09-27-83 13:49	P062581

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	89.5	100	350	20.0B	0.	34.79	6.58	0.98	179.7	27.30	0.785	0.369	37.5	6.78	1.872	4.982	460.89	10.150		0.0
2000	90.0	101	210	17.0B	0.	34.83	7.16	0.90	195.5	27.31	0.784	0.368	37.6	6.78	1.689	1.485	459.36	8.972		0.0
2000	90.2	101	140	18.5B	0.	34.83	7.56	0.86	204.4	27.03	0.776	0.365	38.0	6.71	1.654	1.599	457.28	9.109		0.0
1500	29.5	100	950	32.0B	0.	8.55	7.20	0.90	71.0	9.86	1.154	0.542	25.5	9.97	5.228	3.944	628.36	10.944		0.0
1500	30.2	102	1460	40.5B	0.	8.74	7.56	0.86	76.4	10.11	1.156	0.543	25.5	9.99	4.242	2.912	635.69	14.091		0.0
1500	29.5	100	1390	41.8B	0.	8.55	8.23	0.79	81.8	9.93	1.162	0.546	25.4	10.04	4.099	3.458	636.92	14.477		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	91.0	100	350	20.0B	0.	35.19	6.57	0.98	182.0	27.69	0.787	0.370	37.4	6.80	0.692	1.474	436.91	12.993		0.0
2000	87.0	96	270	21.5B	0.	33.64	7.45	0.87	191.0	25.63	0.762	0.358	38.7	6.59	0.995	1.798	412.89	7.277		0.0
2000	88.0	97	100	24.0B	0.	34.01	7.76	0.83	202.2	26.07	0.767	0.360	38.4	6.63	1.540	2.069	413.55	4.735		0.0
1500	29.0	100	1520	42.0B	0.	8.41	7.47	0.87	74.1	9.92	1.180	0.554	25.0	10.20	2.841	2.971	579.60	12.030		0.0
1500	29.5	102	1380	42.4B	0.	8.56	8.39	0.77	83.1	9.91	1.158	0.544	25.4	10.01	3.354	3.453	565.09	4.162		0.0
1500	30.0	103	1100	46.0B	0.	8.71	9.41	0.69	99.3	10.56	1.212	0.570	24.3	10.48	8.411	5.268	599.31	4.064		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/ BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	90.5	100	390	18.0B	0.	35.03	6.74	0.96	179.7	26.66	0.761	0.358	38.7	6.58	0.809	1.290	428.27	11.570		0.0
2000	88.5	98	290	21.0B	0.	34.25	7.34	0.88	189.6	25.83	0.754	0.354	39.1	6.52	0.977	1.512	417.86	9.264		0.0
2000	90.0	99	120	25.5B	0.	34.84	7.97	0.81	206.7	25.95	0.745	0.350	39.6	6.44	1.283	1.764	409.08	7.549		0.0
1500	29.5	100	1490	40.5B	0.	8.56	7.44	0.87	76.4	10.26	1.198	0.563	24.6	10.36	3.456	2.888	582.72	14.407		0.0
1500	28.8	98	1380	41.5B	0.	8.36	8.37	0.77	83.1	9.93	1.188	0.558	24.8	10.27	3.829	3.483	570.88	4.036		0.0
1500	31.0	105	1200	45.8B	0.	9.00	9.30	0.70	94.4	10.15	1.128	0.530	26.1	9.75	5.100	3.965	502.53	0.798		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	1000* MBTU/BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	91.0	100	420	18.0B	0.	35.19	6.66	0.97	178.8	26.86	0.763	0.359	38.6	6.60	0.851	1.253	430.36	12.169		0.0
2000	89.0	98	300	21.5B	0.	34.41	7.32	0.88	188.7	25.77	0.749	0.352	39.3	6.47	0.901	1.415	414.51	10.553		0.0
2000	90.2	99	130	28.0B	0.	34.87	8.08	0.80	207.6	25.70	0.737	0.346	40.0	6.37	1.115	1.678	406.91	9.544		0.0
1500	29.2	100	1490	41.0B	0.	8.46	7.52	0.86	76.4	10.16	1.201	0.564	24.5	10.38	3.248	2.817	568.25	12.140		0.0
1500	30.0	103	1410	49.8B	0.	8.70	8.11	0.80	81.3	10.03	1.153	0.542	25.5	9.97	3.019	2.780	553.85	7.204		0.0
1500	29.5	101	1250	49.2B	0.	8.56	9.05	0.71	90.8	10.03	1.172	0.551	25.1	10.13	4.050	3.896	512.06	1.124		0.0

TEST NO.: HD-810867 ENGINE: 380 NAPS-ZM 91 TEST D/T: 9-24-82 16: 0 REPORT D/T: 09-27-83 13:49 P062581

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	ENERGY EFFICIENCY 1000* MBTU/BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
1500	30.0	100	1500	40.0B	0.	8.74	7.02	0.92	78.6	11.20	1.281	0.602	23.0	11.07	3.203	13.291	640.18	14.966		0.0
1500	29.5	98	1450	44.5B	0.	8 60	7.68	0.84	80.9	10.53	1.225	0.576	24.0	10.59	3.114	5.575	633.10	12.415		0.0
1500	29.5	98	1350	24.5B	0.	8.59	8.60	0.75	89.0	10.34	1.204	0.566	24.5	10.41	4.228	4.066	633.52	6.504		0.0
1500	29.0	97	1540	21.5B	0.	8.44	7.63	0.85	76 4	10.01	1.185	0.557	24.9	10.25	2.269	11.843	615.45	4.221		0.0
1500	29.5	98	1450	23.0B	0.	8.59	8.23	0.79	82.7	10.04	1.169	0.549	25.2	10.10	2.320	4.813	610.47	6.049		0.0
1500	29.8	99	1360	53.5B	0.	8.68	8.57	0.76	87.6	10.23	1.178	0.553	25.0	10.18	2.966	3.303	636.52	11.002		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	89.0	100	120	20.5B	0.	34 80	7.96	0.81	211.2	26.51	0.762	0.358	38.7	6.59	1.241	2.058	441.85	7.061		0.0
2000	89.5	101	080	22.0B	0.	34.87	7.97	0.81	212.1	26.62	0.764	0.359	38.6	6.60	1.541	2.148	397.81	6.836		0.0
2000	90.0	101	080	22.5B	0.	35.09	7.92	0.82	211.2	26.65	0.760	0.357	38.8	6.57	1.559	2.192	437.53	7.167		0.0
1500	29.5	100	1360	44.5B	0.	8.62	9.10	0.71	87 6	9.63	1.117	0.525	26.4	9.65	0.0	0.0	0.15	0.018		0.0
1500	30.0	102	1130	50.0B	0.	8.77	10.06	0.64	101.1	10.05	1.146	0.538	25.7	9.91	11.336	5.768	566.38	3.759		0.0
1500	30.0	102	880	46.5B	0.	8.77	10.60	0.61	113.2	10.69	1.218	0.572	24.2	10.53	19.589	8.538	559.37	1.332		0.0
2000	89.5	100	120	22.0B	0.	34.89	7.98	0.81	211.2	26.48	0.759	0.357	38.8	6.56	2.108	2.161	445.25	7.759		0.0
2017	89.5	100	080	21.5B	0.	35.17	8.05	0.80	215.7	26.81	0.762	0.358	38.6	6.59	1.983	2.240	442.91	6.404		0.0
2000	90.2	100	080	21.8B	0.	35.14	7.87	0.82	211.2	26.84	0.764	0.359	38.6	6.60	2.063	2.232	444.74	6.408		0.0
1500	29.0	100	1440	28.0B	0.	8.47	8.82	0.73	81.8	9.27	1.094	0.514	26.9	9.45	2.063	2.232	444.74	6.408		0.0
1500	30.2	104	1290	44.5B	0.	8.83	9.70	0.67	90.8	9.36	1.061	0.498	27.8	9.17	2.063	2.232	444.74	6.408		0.0
1516	29.0	100	870	52.0B	0.	8.56	10.87	0.60	116.8	10.75	1.255	0.590	23.5	10.85	2.063	2.232	444.74	6.408		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	1000* MBTU/BHPHR	BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
															----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ----					
2000	89.5	100	140	22.5B	0.	34.47	8.00	0.81	208 9	26.13	0.758	0.356	38.9	6.55	1.776	1.976	453.53	7.787		0.0
2000	90.5	101	080	22.8B	0.	34.85	8.12	0.80	213.4	26.30	0.755	0.355	39.0	6.52	1.621	2.096	446.93	6.372		0.0
2000	89.5	100	080	23.2B	0.	34.44	8.17	0.79	212.5	26.02	0.756	0.355	39.0	6.53	1.679	2.154	445.23	6.136		0.0
1500	29.0	100	1370	44.0B	0.	8.37	9.06	0.71	87.6	9.67	1.156	0.543	25.5	9.99	4.405	4.141	616.63	12.112		0.0
1500	30.0	103	1160	44.0B	0.	8.66	9.79	0.66	98 9	10.10	1.166	0.548	25.3	10.08	10.816	5.993	572.88	4.412		0.0
1500	30.0	103	870	42.2B	0.	8.66	10.38	0.62	114.6	11.04	1.275	0.599	23.1	11.02	22.808	9.047	591.60	2.082		0.0
2000	90.0	100	140	22.5B	0.	34.68	8.03	0.81	211.2	26.31	0.759	0.356	38.8	6.56	1.796	2.212	448.89	7.788		0.0
2000	90.0	100	080	22.5B	0.	34.66	8.08	0.80	213.4	26.43	0.763	0.358	38.6	6.59	1.769	2.138	443.88	6.324		0.0
2000	90.5	101	080	22.0B	0.	34.82	7.95	0.81	211.2	26.55	0.763	0.358	38.6	6.59	1.799	2.100	444.55	6.219		0.0
1500	29.5	100	1420	39.0B	0.	8.52	9.37	0.69	89.9	9.60	1.127	0.529	26.1	9.74	5.229	3.358	626.62	4.457		0.0
1500	29.0	98	1250	45.0B	0.	8.37	10.30	0.63	99.8	9.69	1.157	0.544	25.4	10.00	6.744	5.656	607.30	1.833		0.0
1500	29.8	101	940	40.5B	0.	8.61	10.97	0.59	116.8	10.65	1.237	0.581	23.8	10.69	12.148	8.664	693.74	2.181		0.0

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/ BHPHR	BSFC GAS EQUIV LB/ BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	91.0	100	110	20.0B	0.	35.09	7.86	0.82	211.2	26.87	0.766	0.360	38.5	6.62	2.094	1.883 •	449.19	7.038		0.0
2000	89.8	99	080	21.5B	0.	34.63	8.06	0.80	213.4	26.47	0.764	0.359	38.5	6.61	1.841	2.042	443.94	6.324		0.0
2000	90.2	99	070	21.8B	0.	34.50	7.95	0.81	212.1	26.69	0.774	0.363	38.1	6.69	1.837	2.066	450.21	6.387		0.0
1500	29.0	100	1440	30.0B	0.	8.31	8.66	0.75	81.8	9.44	1.136	0.534	25.9	9.82	4.496	3.665	616.35	6.100		0.0
1500	28.7	99	1200	48.0B	0.	8.23	9.97	0.65	98.0	9.83	1.194	0.561	24.7	10.32	5.956	6.145	590.20	5.151		0.0
1500	29.5	102	930	51.0B	0.	8.46	10.25	0.63	108.7	10.61	1.254	0.589	23.5	10.84	11.200	8.684	582.28	2.521		0.0

TEST NO.: HD-810874 ENGINE: 380 NAPS-ZM2 O TEST D/T: 9-29-82 14: 0 REPORT D/T: 09-27-83 13:57 P06258

SUMMARY REPORT

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	1000* MBTU/BHPHR	BRAKE SPECIFIC EMISSIONS (G/BHPHR) BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
2000	89.5	100	120	21.5B	0.	34.39	7.89	0.82	208.9	26.49	0.770	0.362	38.2	6.66	2.753	2.350	447.81	6.022		0.0
2000	89.0	99	080	23.0B	0.	34.18	8.12	0.80	213.4	26.30	0.769	0.361	38.3	6.65	2.321	2.367	448.51	5.870		0.0
2000	90.0	101	070	22.0B	0.	34.57	7.93	0.82	211.2	26.62	0.770	0.362	38.2	6.66	2.254	2.351	448.39	5.500		0.0
1500	29.8	100	1450	44.5B	0.	8.58	8.65	0.75	81.8	9.46	1.103	0.518	26.7	9.53	8.931	3.042	602.59	10.454		0.0
1500	29.5	99	1340	54.0B	0.	8.48	9.65	0.67	89.9	9.31	1.098	0.516	26.8	9.49	6.532	3.808	578.29	2.580		0.0
1500	29.4	99	990	55.0B	0.	8.45	10.53	0.61	107.8	10.24	1.212	0.569	24.3	10.48	9.161	8.186	577.20	1.109		0.0
2000	90.0	100	090	17.0B	0.	34.56	7.62	0.85	211.2	27.70	0.802	0.377	36.7	6.93	2.827	2.328	437.35	4.783		0.0
2000	90.0	100	080	20.5B	0.	34.57	7.83	0.83	211.2	26.96	0.780	0.366	37.8	6.74	2.300	2.269	425.07	5.204		0.0
2000	89.8	100	080	21.8B	0.	34.49	7.95	0.81	213.0	26.79	0.777	0.365	37.9	6.71	2.300	2.305	423.25	5.671		0.0
1500	30.5	100	1430	52.0B	0.	8.78	8.51	0.76	82.7	9.71	1.107	0.520	26.6	9.57	8.461	2.835	609.72	13.727		0.0
1500	30.0	98	1370	47.0B	0.	8.63	9.47	0.68	87.6	9.25	1.072	0.504	27.5	9.27	6.071	3.234	581.48	2.879		0.0
1500	28.0	92	1060	50.0B	0.	8.05	10.46	0.62	103.3	9.88	1.227	0.576	24.0	10.61	7.955	7.941	604.71	0.946		0.0

TEST NO.: HD-810877 ENGINE: 380 NAPS-ZM2 O TEST D/T: 10-19-82 13: 0 REPORT D/T: 09-27-83 13:59 PO62581

<u>SUMMARY REPORT</u>

RPM	TORQ. FT-LB	% MAX TQ.	THR POSN	IGN. TIMG.	INJ. TIMG.	CORR. BHP	MEAS. A/F	PHI EQUIV RATIO	AIR LB/HR	FUEL LB/HR	BSFC LB/BHPHR	BSFC GAS EQUIV LB/BHPHR	ENERGY EFFICIENCY %	1000* MBTU/ BHPHR	----- BRAKE SPECIFIC EMISSIONS (G/BHPHR) ---- BSHC	BSCO	BSCO2	BSNOX	BSALDYH	BSPART
1500	29.5	100	1420	44.0B	0.	8.51	8.69	0.74	82.7	9.52	1.119	0.525	26.3	9.67	3.299	3.123	599.40	11.074		0.0
1500	29.1	99	1280	48.5B	0.	8.38	9.74	0.66	92.1	9.46	1.128	0.530	26.1	9.75	5.934	4.700	586.13	3.888		0.0
1500	30.0	102	900	45.5B	0.	8.64	10.42	0.62	111.4	10.70	1.237	0.581	23.8	10.69	19.985	8.405	564.77	1.580		0.0