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Technical Report

Cold Starting Spark Ignition Engines with Methanol: An Analysisof Current Options, and Their Impact on Air Quality

By

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I. Introduction

II. Executive Summary

III. Unassisted Vaporization

A. Introduction

In order to start a Spark Ignited (SI) engine, it is generally accepted that a sufficient quantity of fuel must vaporize to provide a combustible mixture of fuel vapor and air in the vicinity of the spark plug during the ignition event (1). Under cold starting conditions, the latent heat of vaporization and the curve of vapor pressure versus saturation temperature play important roles in determining the quantity of vapor available at the spark plug. Gasoline is a multi-component mixture, and certain components in the fuel can provide sufficient vapor phase fuel under cold starting conditions, provided enough gasoline is added to the air charge. Neat methanol, of course, has no such volatility additives, and has a fixed volatility curve. Although volatility components can be added to methanol, in this section we want to evaluate the requirements for cold starting neat methanol without additives or other starting devices.

In this situation, essentially all the heat used to vaporize the neat methanol must come from the latent heat of the air and liquid fuel charge, from heat transfer from warmer engine surfaces (if any), and from the work of the compression stroke; most will come from the work of compression. The process can be envisioned as consisting of five steps: 1) Liquid fuel and air at the prevailing ambient temperature are mixed. 2) The fuel begins to vaporize and the temperature of the air and remaining liquid fuel is depressed until an equilibrium is reached. The amount that can be vaporized depends on the initial ambient temperature. 3) The depressed mixture temperature leads to heat transfer from engine surfaces which have remained at the ambient temperature. 4) The mixture is then compressed and heated, with some heat loss to the cylinder walls. 5) The heat of compression is used to vaporize more of the methanol until equilibrium is reached. In actuality, the second, third, fourth, and fifth steps overlap in time.

B. Required Properties

Several investigators (2)(3) have used various approaches to model the vaporization or starting capabilities Of neat methanol. Some are more sophisticated than others. Our approach while somewhat simpler than others provides a different perspective to the issue of cold starting with methanol.

Our approach was based on the premise that a certain quantity of vapor is required for the starting of SI engines, and that the macroscopic equilibrium conditions to maintain this vapor must at least be met. Otherwise, some the vapor would condense and reduce the vapor fuel-air ratio to a non-combustible mixture. The temperature of the mixture after adiabatic ideal gas compression and evaporation serves as an indication of equilibrium reserve, and as such must be at or above the saturation or dew temperature for the vapor phase fuel concentration. Any temperature above the dew temperature would be a temperature reserve (TR). This reserve would be utilized for such things as heat transfer to the cylinder walls, non-ideal process losses, rate effects (i.e., insufficient time to reach equilibrium), etc. Because these losses are always present and must be compensated, the temperature reserve would have to be a positive number in order to maintain the fuel in a vapor state under equilibrium conditions. In our evaluation, we compared the calculated temperature reserve (TR) at known start capability conditions to the reserve at unknown cold starting conditions. As long as the reserve at the lower temperature was equal to or greater than the reserve at the higher temperature (with the known start capability), we assumed the potential for a successful cold start existed.

This approach however assumes all of the evaporation necessary to achieve a combustible mixture occurs during the compression stroke. Before accepting the reasonableness of this approach (i.e., no fuel evaporation in the intake manifold or in the cylinder during the intake stroke when cranking), we studied the relationship between the vapor equivalence ratio and the ambient temperature which would be required to support a given vapor ratio after evaporation. Because methanol is a single component mixture, a well defined relationship exists between the partial pressure of methanol and the saturation or dew temperature for methanol. The partial pressure of a given fuel mixture is governed by the concentration of the vaporized methanol (i.e. the vapor f/a ratio), and the local pressure. The following equation from Appendix I in reference (5) was used to determine the partial pressure of methanol for a range of pre-selected vapor equivalence ratios.

(III-1) $$PV = (MAP)\ (1/[1 + (7.155/VPHI)])$$

where:

PV = partial pressure
MAP = manifold air pressure
VPHI = vapor equivalence ratio

The saturation temperature is, of course, the temperature that separates the liquid phase from the gas phase, and in our case it is the temperature which must be maintained in order to prevent our pre-selected vapor equivalence ratio from condensing. Reference (6) describes the relationship between the partial pressure and dew temperature as:

(III-2) $$TDEW = (-1961.8678)/[\log_{10}(PV) - 8.639821)]$$

where:

TDEW =(°K) = saturation temperature
PV = (mm Hg) = partial pressure from Eq. III-1

Equation III-2, does not consider the fact that in the case of an intake manifold, in order to achieve a given vapor equivalence ratio, an equal amount of liquid methanol must be evaporated. The process of evaporating the liquid fuel would depress the surrounding ambient temperature. Therefore, in order to maintain our computed TDEW value for our pre-selected vapor PHI in equation III-1, our initial ambient temperature before evaporation must be above the TDEW value at least by the amount of temperature depression caused by the evaporation of the fuel. The following equation from reference (4) was selected to determine the temperature drop from the evaporating fuel.

(III-3) $$TDROP = [(x)(F)(HLG) + (Q)]/[(1-F+xF)(C_p)](5/9)$$

where:

TDROP = (°C) = temperature drop
x = portion evaporated, in our case = 1.0
F = fuel-air ratio
HLG = heat of vaporization = 474 BTU/lb (ref. 4)
C_p = is for a mixture of fuel-air vapor, reference (4) lists C_p as 0.245 for a stoichiometric mixture of methanol vapor and air, and a C_p of 0.240 for air alone.
Q = heat addition, in our case = 0.0
- Heating or cooling of excess liquid fuel was neglected
- Pressure changes (if any) from cooling were ignored

By substituting the same pre-selected equivalence ratio that was used in equation III-1 into equation III-3, we can identify the temperature drop that would be associated with vaporizing a given amount of fuel corresponding to the dew temperature calculated in equation III-2. Combining the results of III-2 and III-3, we have the initial ambient temperature before vaporization (which is assumed to be the same as the initial intake manifold air temperature) that would support the pre-selected vapor equivalence ratio.

(III-4) TAMB = TDEW + TDROP

Where:

TAMB = ambient temperature.
TDEW = saturation temperature, equation III-2.
TDROP = temperature drop, equation III-3.

Results of equation III-4 for a range of vapor equivalence ratios and manifold air pressures (MAP) are plotted in Figure III-1 and listed in Table III-1 through III-4. The range of manifold pressures used to derive these tables covered the range (suitable for modeling purposes) of observed manifold pressure values for a typical four cylinder engine under cranking conditions at 99kPa(wet) barometric pressure and a range of cranking speeds from 200 RPM to 600 RPM. From Figure III-1 and the Tables, the influence of varying the manifold air pressure slightly from the normal cranking condition (85 kPa) appears to have little effect on the amount of fuel that can be vaporized in the intake manifold at temperatures below 0°C. Large changes in manifold pressure may have an observable affect on the amount of fuel that can be vaporized, but we will investigate that prospect later.

For the time being, these equilibrium data suggest that if we are to consider starting a neat methanol engine at ambient temperatures of -18°C (0° F) or below (TAMB in Tables III-1 to -4), it seems safe to say that there will be very little evaporation occurring within the inlet manifold during cranking, at most maybe a 0.05 to 0.06 vapor equivalence ratio, and at -35°C, maybe a 0.03 vapor equivalence ratio. (Lack of time may prevent even these points from being reached). For a carbureted engine, the lack of fuel vaporization in the inlet manifold could cause serious distribution problems, but more important for our study, the lack of evaporation in the inlet manifold means that the majority of the evaporation to produce an ignitable vapor at cold conditions must occur during the compression stroke. This evaluation, therefore, suggests that our initial approach (i.e., no fuel evaporation in the intake manifold or cylinder during the intake stroke when cranking) is a reasonable assumption to begin our analysis.

We have so far neglected the fact that the temperature drop caused by evaporation will create a temperature gradient and encourage heat transfer from the manifold or cylinder walls, leading in turn to a higher vapor equivalence ratio. Evaporation of enough fuel to cause a vapor equivalence ratio

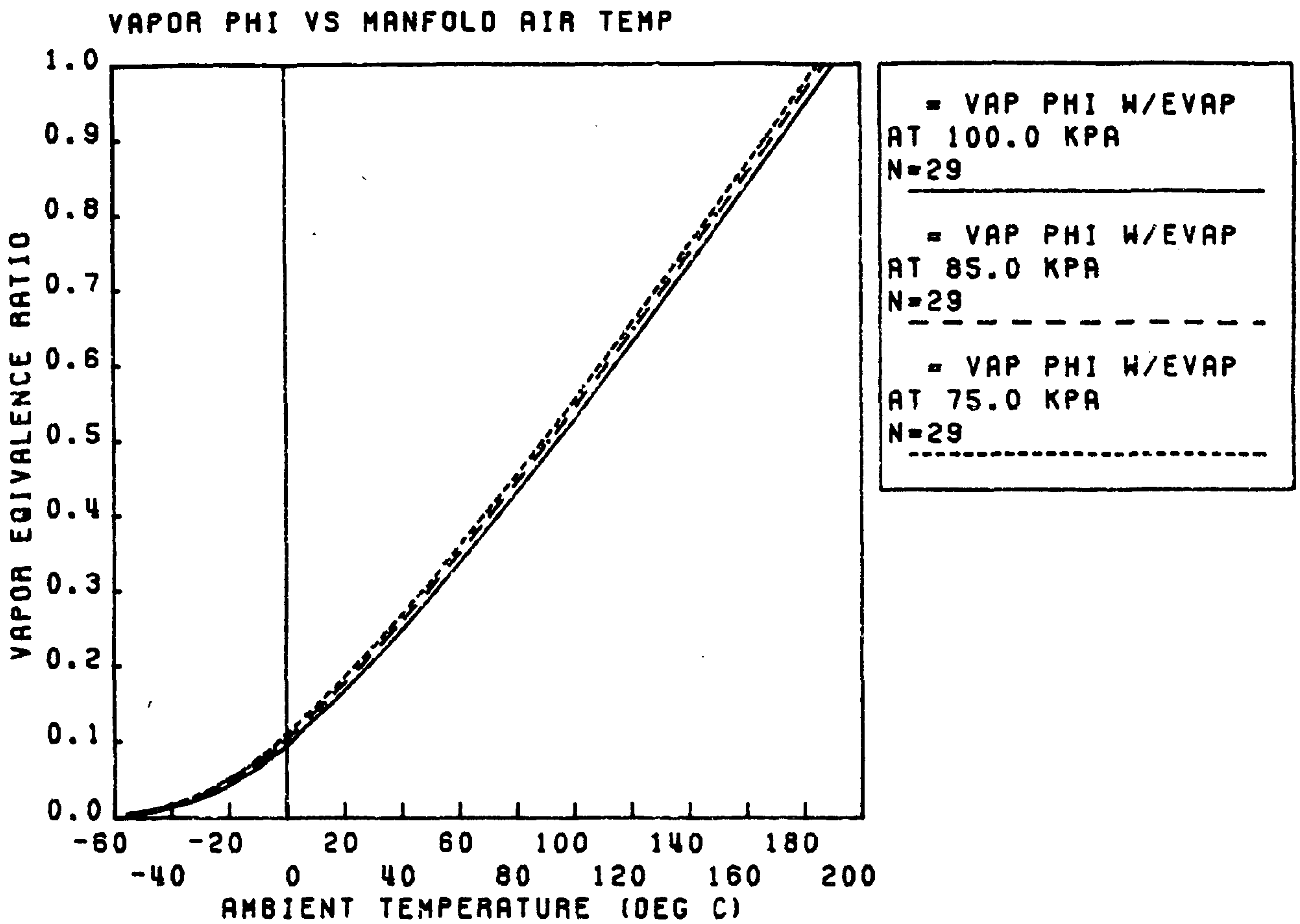
VAPOR PHI VS MANFOLD AIR TEMP
VAPOR EQIVALENCE RATIO
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0.0
-60
-40
-20
0
20
40
60
80
100
120
140
160
180
200
AMBIENT TEMPERATURE (DEG C)
= VAP PHI W/EVAP
AT 100.0 KPA
N=29
= VAP PHI W/EVAP
AT 85.0 KPA
N=29
= VAP PHI W/EVAP
AT 75.0 KPA
N=29

FIG. III -1

TABLE III-1

VAPOR EQUIVALENCE RATIO

MAP (KPA)	VAPOR PHI	PART PRESS	TDEW (DEG C)	TDROP (DEG C)	TAMB (DEG C)
100.000	0.005	0.070	-53.221	0.850	-52.371
100.000	0.010	0.140	-45.547	1.701	-43.847
100.000	0.020	0.279	-37.328	3.401	-33.926
100.000	0.030	0.418	-32.246	5.102	-27.143
100.000	0.040	0.556	-28.510	6.803	-21.707
100.000	0.050	0.694	-25.536	8.503	-17.033
100.000	0.060	0.832	-23.056	10.204	-12.852
100.000	0.070	0.969	-20.922	11.905	-9.018
100.000	0.080	1.106	-19.047	13.606	-5.442
100.000	0.090	1.242	-17.372	15.306	-2.066
100.000	0.100	1.378	-15.858	17.007	1.149
100.000	0.150	2.053	-9.880	25.510	15.630
100.000	0.200	2.719	-5.499	34.014	28.515
100.000	0.250	3.376	-2.023	42.517	40.494
100.000	0.300	4.024	0.865	51.021	51.886
100.000	0.350	4.664	3.339	59.524	62.863
100.000	0.400	5.295	5.503	68.028	73.531
100.000	0.450	5.917	7.427	76.531	83.958
100.000	0.500	6.532	9.160	85.035	94.194
100.000	0.550	7.138	10.735	93.538	104.273
100.000	0.600	7.737	12.180	102.042	114.221
100.000	0.650	8.328	13.513	110.545	124.058
100.000	0.700	8.912	14.750	119.048	133.799
100.000	0.750	9.488	15.904	127.552	143.456
100.000	0.800	10.057	16.986	136.055	153.041
100.000	0.850	10.618	18.002	144.559	162.561
100.000	0.900	11.173	18.961	153.062	172.023
100.000	0.950	11.721	19.868	161.566	181.434
100.000	1.000	12.262	20.729	170.069	190.798

TABLE III-2

VAPOR EQUIVALENCE RATIO

MAP (KPA)	VAPOR PHI	PART PRESS	TDEW (DEG C)	TDROP (DEG C)	TAMB (DEG C)
85.000	0.005	0.059	-54.948	0.850	-54.097
85.000	0.010	0.119	-47.396	1.701	-45.695
85.000	0.020	0.237	-39.312	3.401	-35.910
85.000	0.030	0.355	-34.316	5.102	-29.214
85.000	0.040	0.473	-30.645	6.803	-23.842
85.000	0.050	0.590	-27.723	8.503	-19.219
85.000	0.060	0.707	-25.286	10.204	-15.082
85.000	0.070	0.824	-23.191	11.905	-11.286
85.000	0.080	0.940	-21.349	13.606	-7.744
85.000	0.090	1.056	-19.705	15.306	-4.399
85.000	0.100	1.172	-18.217	17.007	-1.211
85.000	0.150	1.745	-12.351	25.510	13.160
85.000	0.200	2.311	-8.052	34.014	25.962
85.000	0.250	2.870	-4.642	42.517	37.875
85.000	0.300	3.421	-1.810	51.021	49.211
85.000	0.350	3.964	0.615	59.524	60.139
85.000	0.400	4.500	2.737	68.028	70.764
85.000	0.450	5.030	4.623	76.531	81.154
85.000	0.500	5.552	6.321	85.035	91.356
85.000	0.550	6.067	7.865	93.538	101.403
85.000	0.600	6.576	9.280	102.042	111.322
85.000	0.650	7.079	10.586	110.545	121.131
85.000	0.700	7.575	11.799	119.048	130.847
85.000	0.750	8.065	12.929	127.552	140.481
85.000	0.800	8.548	13.988	136.055	150.043
85.000	0.850	9.026	14.984	144.559	159.542
85.000	0.900	9.497	15.923	153.062	168.985
85.000	0.950	9.963	16.811	161.566	178.377
85.000	1.000	10.423	17.654	170.069	187.723

TABLE III-3

VAPOR EQUIVALENCE RATIO

MAP (KPA)	VAPOR PHI	PART PRESS	TDEW (DEG C)	TDROP (DEG C)	TAMB (DEG C)
75.000	0.005	0.052	-56.259	0.850	-55.409
75.000	0.010	0.105	-48.800	1.701	-47.099
75.000	0.020	0.209	-40.817	3.401	-37.416
75.000	0.030	0.313	-35.886	5.102	-30.784
75.000	0.040	0.417	-32.263	6.803	-25.461
75.000	0.050	0.520	-29.381	8.503	-20.877
75.000	0.060	0.624	-26.977	10.204	-16.773
75.000	0.070	0.727	-24.910	11.905	-13.005
75.000	0.080	0.829	-23.094	13.606	-9.489
75.000	0.090	0.932	-21.472	15.306	-6.166
75.000	0.100	1.034	-20.006	17.007	-2.999
75.000	0.150	1.540	-14.222	25.510	11.288
75.000	0.200	2.039	-9.985	34.014	24.029
75.000	0.250	2.532	-6.625	42.517	35.892
75.000	0.300	3.018	-3.835	51.021	47.186
75.000	0.350	3.498	-1.446	59.524	58.078
75.000	0.400	3.971	0.644	68.028	68.671
75.000	0.450	4.438	2.501	76.531	79.033
75.000	0.500	4.899	4.174	85.035	89.208
75.000	0.550	5.354	5.694	93.538	99.232
75.000	0.600	5.803	7.087	102.042	109.129
75.000	0.650	6.246	8.373	110.545	118.918
75.000	0.700	6.684	9.566	119.048	128.615
75.000	0.750	7.116	10.679	127.552	138.231
75.000	0.800	7.542	11.722	136.055	147.777
75.000	0.850	7.964	12.702	144.559	157.260
75.000	0.900	8.380	13.626	153.062	166.688
75.000	0.950	8.791	14.500	161.566	176.066
75.000	1.000	9.197	15.330	170.069	185.399

TABLE III-4

VAPOR EQUIVALENCE RATIO

MAP (KPA)	VAPOR PHI	PART PRESS	TDEW (DEG C)	TDROP (DEG C)	TAMB (DEG C)
65.000	0.005	0.045	-57.739	0.850	-56.889
65.000	0.010	0.091	-50.383	1.701	-48.682
65.000	0.020	0.181	-42.515	3.401	-39.113
65.000	0.030	0.271	-37.656	5.102	-32.554
65.000	0.040	0.361	-34.088	6.803	-27.285
65.000	0.050	0.451	-31.249	8.503	-22.745
65.000	0.060	0.541	-28.882	10.204	-18.678
65.000	0.070	0.630	-26.847	11.905	-14.942
65.000	0.080	0.719	-25.059	13.606	-11.454
65.000	0.090	0.807	-23.463	15.306	-8.157
65.000	0.100	0.896	-22.020	17.007	-5.013
65.000	0.150	1.335	-16.328	25.510	9.182
65.000	0.200	1.768	-12.161	34.014	21.853
65.000	0.250	2.194	-8.857	42.517	33.661
65.000	0.300	2.616	-6.113	51.021	44.907
65.000	0.350	3.031	-3.765	59.524	55.760
65.000	0.400	3.441	-1.711	68.028	66.317
65.000	0.450	3.846	0.115	76.531	76.646
65.000	0.500	4.246	1.759	85.035	86.793
65.000	0.550	4.640	3.252	93.538	96.790
65.000	0.600	5.029	4.621	102.042	106.663
65.000	0.650	5.413	5.885	110.545	116.429
65.000	0.700	5.792	7.057	119.048	126.105
65.000	0.750	6.167	8.150	127.552	135.702
65.000	0.800	6.537	9.174	136.055	145.229
65.000	0.850	6.902	10.136	144.559	154.695
65.000	0.900	7.263	11.044	153.062	164.106
65.000	0.950	7.619	11.903	161.566	173.468
65.000	1.000	7.971	12.717	170.069	182.786

of 0.05 at -19°C and 85 kpa MAP implies a mixture temperature drop of about 8.5°C which seems small enough that the resulting heat transfer can initially be ignored.

If there is essentially no temperture drop and no heat transfer, then from an equilibrium point of view, evaporation before compression does not affect the energy balance which determines the conditions at the end of compression, so such evaporation can be ignored in the equilibrium calculations. With these considerations in mind, in order to evaluate the effect on equilibrium conditions of the incrementally increasing pressure that occurs as the piston travels upwards, we began by reevaluating equation III-2 and equation III-1. Equation III-2 can be written as

(III-5) $TDEW = A/[\log_{10}(PV)-B]$

where:

$A = -1961.8678$

$B = +8.6339821$

Performing some algebra on III-5, we can write the solution for PV as:

(III-6) $PV = 10^{J}$

where:

$J = (A/TDEW) + B$

If we assume for this analysis that the initial pressure levels in the cylinder under cranking conditions are the same as the average intake manifold conditions, then we can substitute for PV in the partial pressure equation (equation III-1) with the relationship identified in III-6.

(III-7) $10^{J} = (MAP)\ (1/[1 + (7.155/VPHI)])$

Solving for VPHI, we have

(III-8) $VPHI = 7.155/[(MAP/10^{J}) - 1]$

Solving III-8 by substituting selected values of TDEW in the "J" identity results in approximately the same values as in Tables III-1 through III-3. (round-off errors are assumed to be the cause of the small differences occuring in the third decimal place). However, equation III-8, is now a general equation relating pressure (MAP) and saturation temperature in the "J" term to the vapor equivalence ratio (VPHI). Further, if we solve equation III-4 for TDEW, and substitute these results in the expression for "J", we have a relationship that now includes the effect of the temperature drop from the vaporizing fuel, and the ambient temperature necessary to maintain the vapor equivalence ratio.

(III-9) $J = [A/(TAMB-TDROP)] + B$

where:

TAMB = (°K)

(III-10) $TDROP = (5/9)(1/C_p)[(f/a)_s(VPHI)(HLG)] = (°K)$

$(f/a)_s = 0.155$ = stoichiometric fuel air ratio

VPHI = vapor equivalence ratio

HLG = 474 BTU/lb = heat of vaporization (ref. 4)

The relationship for "J" can further be modified to include the adiabatic temperature which would reflect a new saturation temperature as the cylinder pressure is increased.

(III-11) $J = [A/(TAD - TDROP)] + B$

where:

TAD = adiabatic temperature (°K) $= TAMB\ (p_2/MAP)^R$

TAMB = (°K) = ambient temperature

p_2 = new cylinder pressure

$R = (k-1)/k$

k = 1.38 = ratio of specific heats for a methanol and air mixture (4)

MAP = manifold air pressure which is assumed to be the same as the initial conditions in the cylinder during cranking conditions.

Putting everything together, we have

(III-12) $VPHI = 7.155/[(p_2/10^J) - 1]$

where:

p_2 = (mm Hg) = cylinder pressure (divide kPa by 0.133224 to obtain mm Hg)

$J = (A/[TAD - (5/9)(1/C_p)(f/a)_s\ (VPHI)(HLG)]) + B$

A = -1961.8678

B = +8.6339821

$C_p = .240$

$(f/a)_s = 0.155$

(HLG) = 474 BTU/lb

(III-13) TAD = (°K) = $TAMB\ (p_2/MAP)^R$ = adiabatic temperature resulting from increasing the cylinder pressure from MAP to p_2. For our analysis, p_2 was incremented from the MAP value to determine TAD values.

TAMB = (°K) = ambient temperature

$R = (k-1)/k$

k = 1.38

With equation III-12, we now have an equation* that allows us to increase the cylinder pressure from the manifold conditions for a given ambient temperature, and observe the resultant vapor equivalence ratio that would be allowed to occur under equilibrium conditions. If the vapor ratio goes down as the pressure is increased, then we can assumed that some of any pre-vaporized fuel will condense, if it goes up we can assume that any fuel vaporized in the intake tract (in this analysis) will remain vaporized.

Figure III-2 shows the results of solving equation III-12 for four ambient temperature levels. From 85 kPa (the approximate MAP observed during cranking) to over 3500 kPa (absolute pressure), we see a steadily increasing vapor equivalence ratio that is supportable under the equilibrium conditions given. Equation III-12 also allows us to explore the concept of temperature reserve (TR) as it relates to the potential for predicting the upper limits of cold start performance with neat methanol. The assumption is that equilibrium values will define the maximum permissable performance, and other factors such as heat transfer rates, and allowable fuel evaporation rates will only serve to reduce the maximum perfomance. The magnitude of these compromising factors and their influence is for the time being ignored in order to determine the upper limit of performance.

Since it is generally accepted that a certain amount of fuel vapor is necessary to initiate combustion, we can use equation III-12 to determine an equilibrium temperature reserve in the following manner. First, we initially define the temperature reserve (TRX) as the difference between the temperature that theoretically exists in the cylinder after adiabatic compression (TAD) minus fuel evaporation temperature losses (TDROP) and minus the saturation temperature (TDEW) of the vapor mixture, without accounting for heat losses to the cylinder walls.

(III-14) $$TRX = (TAD - TDROP) - TDEW$$

Note, that the adiabatic temperature and the temperature drop are related (in the "J" term) to the vapor equivalence ratio (VPHI) for a given cylinder pressure by equation III-12. The calculated vapor equivalence ratio also defines the saturation temperature (TDEW). If we were to use the

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* Note that equation III-12 has VPHI in the "J" term as well. In order to solve III-12, we seeded a computer program with VPHI and iterated III-12 until the seeded value and the calculated value were essentially equal.

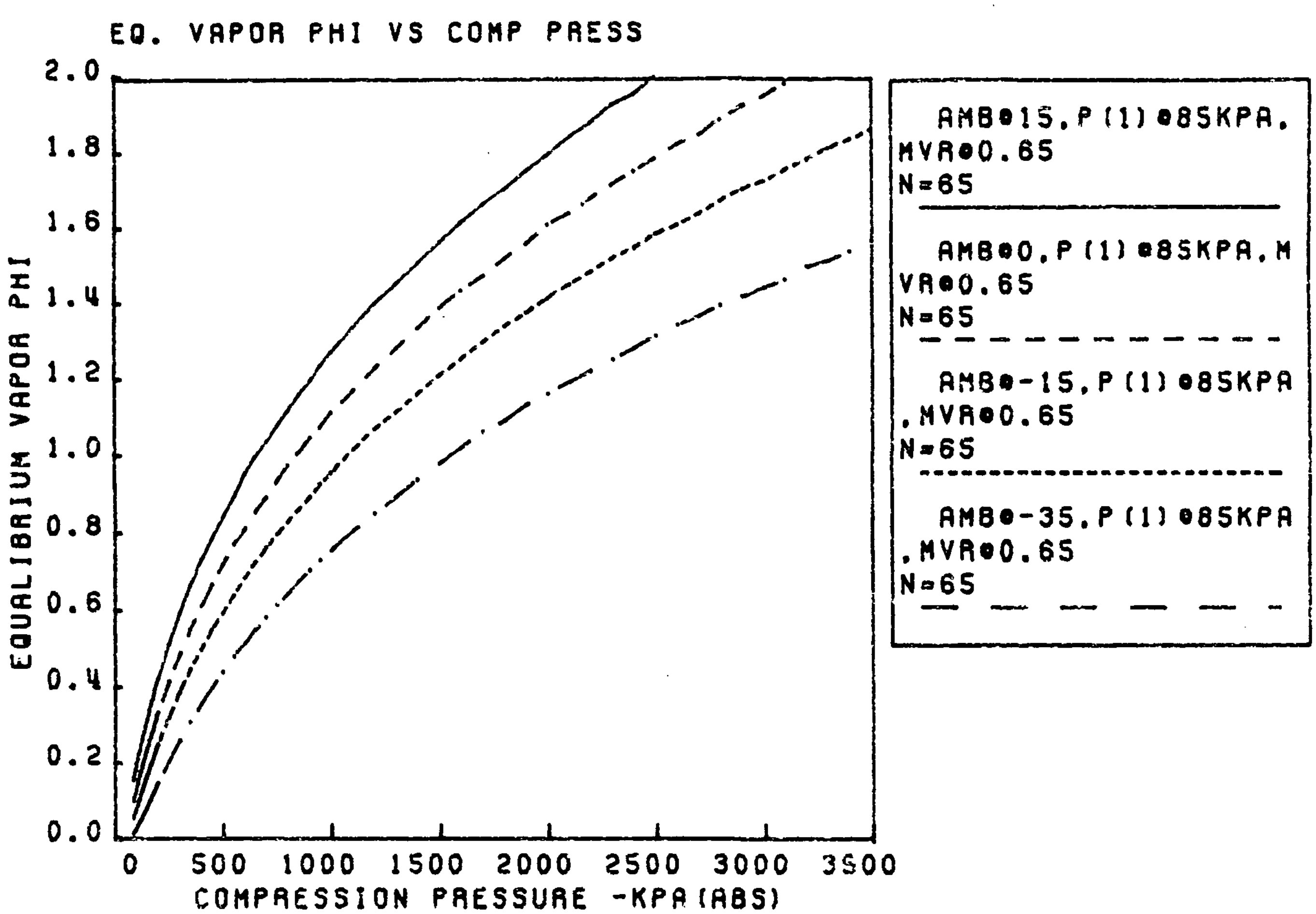
EQ. VAPOR PHI VS COMP PRESS
EQUALIBRIUM VAPOR PHI
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0.0
0
500
1000
1500
2000
2500
3000
3500
COMPRESSION PRESSURE -KPA(ABS)
AMB@15,P(1)@85KPA,
MVR@0.65
N=65
AMB@0,P(1)@85KPA,M
VR@0.65
N=65
AMB@-15,P(1)@85KPA
,MVR@0.65
N=65
AMB@-35,P(1)@85KPA
,MVR@0.65
N=65

FIG. III - 2

related values for TAD, TDROP, and TDEW corresponding to a given VPHI, we would find that the temperature reserve would be essentially zero. But, if we limit the saturation temperature (TDEW) to the value that occurs at the vapor equivalence ratio corresponding to minimum vapor ratio (MVR) required to initiate combustion, we would find that the temperature reserve would begin to increase once the value TAD minus TDROP exceeded our artificially limited TDEW. This TRX value is essentially a measure of the equivalence ratio reserve between the maximum vapor equivalence ratio that can be sustained under the particular pressure and temperature conditions after evaporation, and the MVR value. If we truely wish to represent the temperature reserve (TR) above the MVR, then we must also limit the temperature drop (TDROP) in equation III-14 to that which would occur when evaporating only the amount of fuel necessary to achieve the MVR selected. Thus equation III-14 becomes:

(III-15) TR* = TAD - (TDROP @ MVR + TDEW @ MVR)

Figure III-3 is graphic depiction of this phenonon for several ambient starting temperatures and an assumed minimum vapor ratio (MVR) of 0.65.

Before we pursue the temperature reserve concept further, it would be useful to investigate the effect that the assumed minimum vapor ratio (MVR) has on the conclusions that may be drawn from the model. For instance the lower flamability limit (LFL) for methanol at atmospheric pressure corresponds to a vapor equivalence ratio of 0.455(19). However, Browning (3) suggests that an MVR of 0.63 is necessary for initiation of methanol combustion. Bardon (2) on the other hand, suggests a ratio 0.61 is necessary.

Figure III-4 provides an insight on the effect that the MVR has on the temperature reserve. Trends are plotted for MVRs of 0.6 and 0.7, and for two ambient starting temperatures. Note that the leaner the assumed MVR, the greater the temperature reserve. Expressed otherwise, a high

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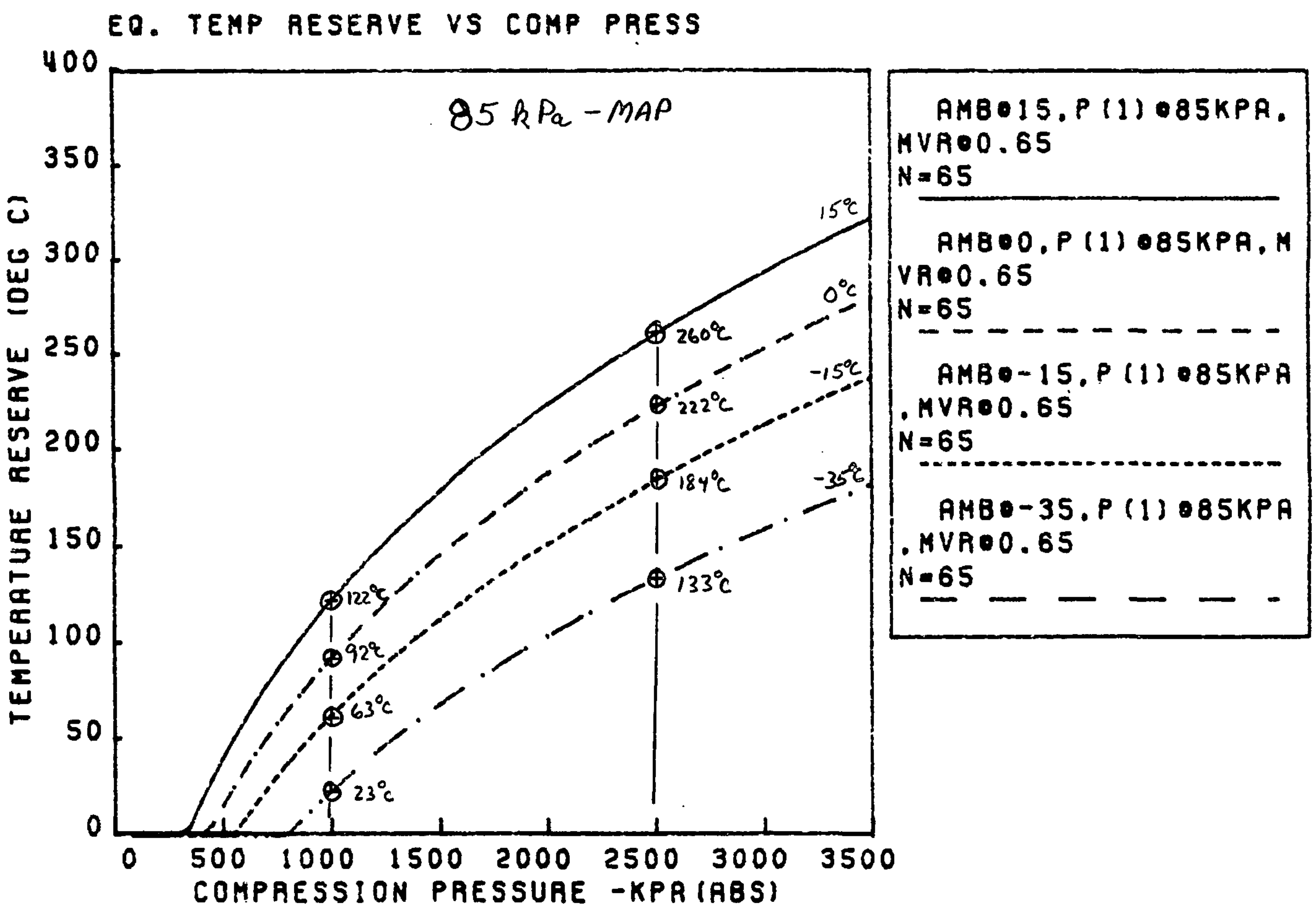
*Note, whenever TR was less than or equal to zero, our computer program automatically set TR equal to zero. The following figures include this assumption. Future work may find the amount of negative TR which is not shown in our figures useful in determining the amount of energy required to be added to the system to achieve a startable temperature reserve. However, for our initial understanding, we choose to set negative TR values equal to zero.



EQ. TEMP RESERVE VS COMP PRESS
85 kPa - MAP
TEMPERATURE RESERVE (DEG C)
COMPRESSION PRESSURE -KPA (ABS)
0
50
100
150
200
250
300
350
400
0
500
1000
1500
2000
2500
3000
3500
15°C
0°C
-15°C
-35°C
260°C
222°C
184°C
133°C
122°C
92°C
63°C
23°C
AMB=15, P(1)=85KPA, MVR=0.65 N=65
AMB=0, P(1)=85KPA, MVR=0.65 N=65
AMB=-15, P(1)=85KPA, MVR=0.65 N=65
AMB=-35, P(1)=85KPA, MVR=0.65 N=65

FIG. III -3

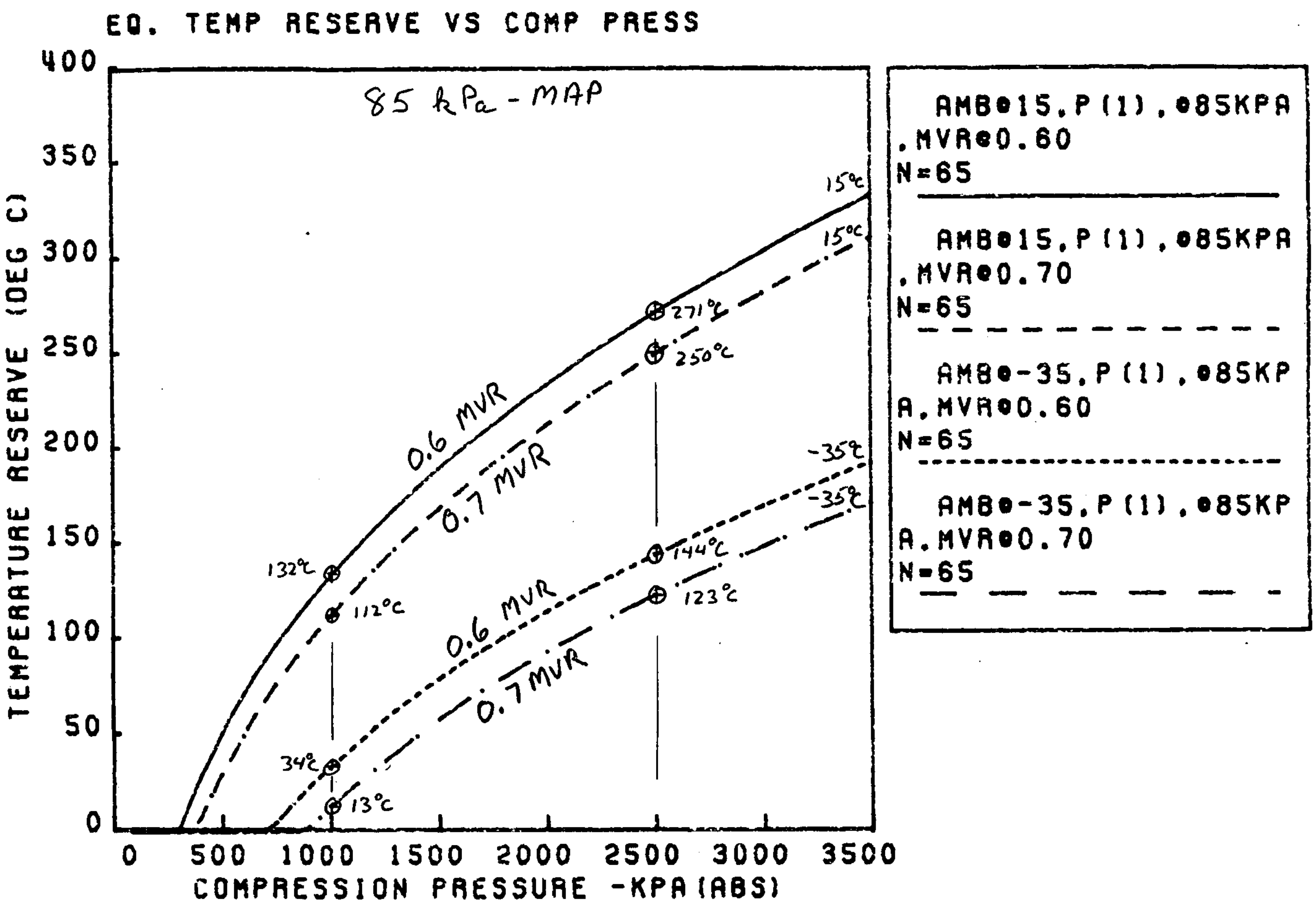

EQ. TEMP RESERVE VS COMP PRESS
85 kPa - MAP
TEMPERATURE RESERVE (DEG C)
400
350
300
250
200
150
100
50
0
0
500
1000
1500
2000
2500
3000
3500
COMPRESSION PRESSURE -KPA (ABS)
0.6 MVR
0.7 MVR
0.6 MVR
0.7 MVR
15°C
15°C
-35°C
-35°C
271°C
250°C
144°C
123°C
132°C
112°C
34°C
13°C
AMB@15, P(1), @85KPA, MVR@0.60
N=65
AMB@15, P(1), @85KPA, MVR@0.70
N=65
AMB@-35, P(1), @85KPA, MVR@0.60
N=65
AMB@-35, P(1), @85KPA, MVR@0.70
N=65

FIG. III -4

vapor equivalence ratio requirement consumes temperature reserve. This of course is due to the leaner mixture having a lower saturation temperature, and to less sensible heat being removed to vaporize the smaller amount of fuel. Also note that for a given compression pressure, there is approximately a 21°C difference in temperature reserve between an MVR of 0.6 and an MVR of 0.7 at either ambient temperature. However, if we review figure III-3, we observe approximately a 30°C to 40°C difference in temperature reserve between different ambient temperatures (+15, 0, -15*) at a constant MVR (0.65). Therefore, the selection of MVR can have an appreciable effect on the results, since the effect of the value selected can be a sizable portion of the effect due to a change in temperature with a constant MVR.

The MVR selected for modeling cold start performance may be a function of engine combustion chamber design or it may be a combination of factors such as the dryness of the methanol, the chamber design, or the ignition energy. It appears that one should be careful in selecting an MVR for modeling purposes, and at least attempt to be consistent when possible in comparing data from other investigations. For our analysis we choose an MVR of 0.65 for the reason that at least two methanol engines with different combustion chamber designs (Nissan NAPS-Z and Ricardo HRCC) have demonstrated (19)(22) that they can be operated with reasonable performance at this overall equivalence ratio in a warmed-up condition. Also, an MVR of 0.65 is slightly conservative from the values used by Browning and Bardon.

Now that the concept of what we call "temperature reserve" has been explained, we can proceed to use this concept to evaluate the potential limits for cold starting with neat methanol. Noticing that the temperature reserve in Figure III-3 is a function of compression pressure, we can compare the temperature reserve at the maximum compression pressure with one engine at a given compression ratio and ambient temperature to the reserve for another engine with a different compression ratio or at a different ambient temperature. For example, in Figure III-5 an engine with a

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* The difference in temperature reserve between -15°C and -35°C is approximately 40°C to 50°C.

peak compression pressure* of 1000 kPa-abs** (130 psi gage) would have a temperature reserve of 122°C at an ambient temperature of 15°C. That same engine at an ambient temperature of -35°C would only have a 23°C temperature reserve. If it were assumed that a 122°C temperature reserve were necessary to start a methanol engine, from Figure III-5 the maximum compression pressure would need to be increased to around 2300 kPa-abs (320 psi gage)*** to obtain a vapor equivalence ratio of 0.65 under equilibrium conditions and an ambient temperature of -35°C.

Obviously increased compression is one means to increase the temperature reserve and improve the fuel evaporation during the compression stroke. Another means is by increasing the throttling during cranking. Because the volume ratio for compression would remain the same when comparing wide open throttle (WOT) to closed throttle (CT), and if dynamic effects were ignored, the adiabatic pressure ratio would remain the same for either WOT or CT. Because the pressure ratio would be the same, the adiabatic temperature (TAD, Equation III-12) would remain the same for either WOT or CT (assuming that T1 is essentially the same for both WOT and CT cranking conditions (ref. 18)). However, under CT conditions, the absolute values of p_2 and p_1 would be lower than in the WOT case. But, because the adiabatic compression temperature (TAD) remains the same, a re-computation of equation III-12 and III-14 with the lower p_2 pressure would result in a higher temperature reserve than would occur under the original WOT conditions because the partial pressure of the methanol vapor would be less.

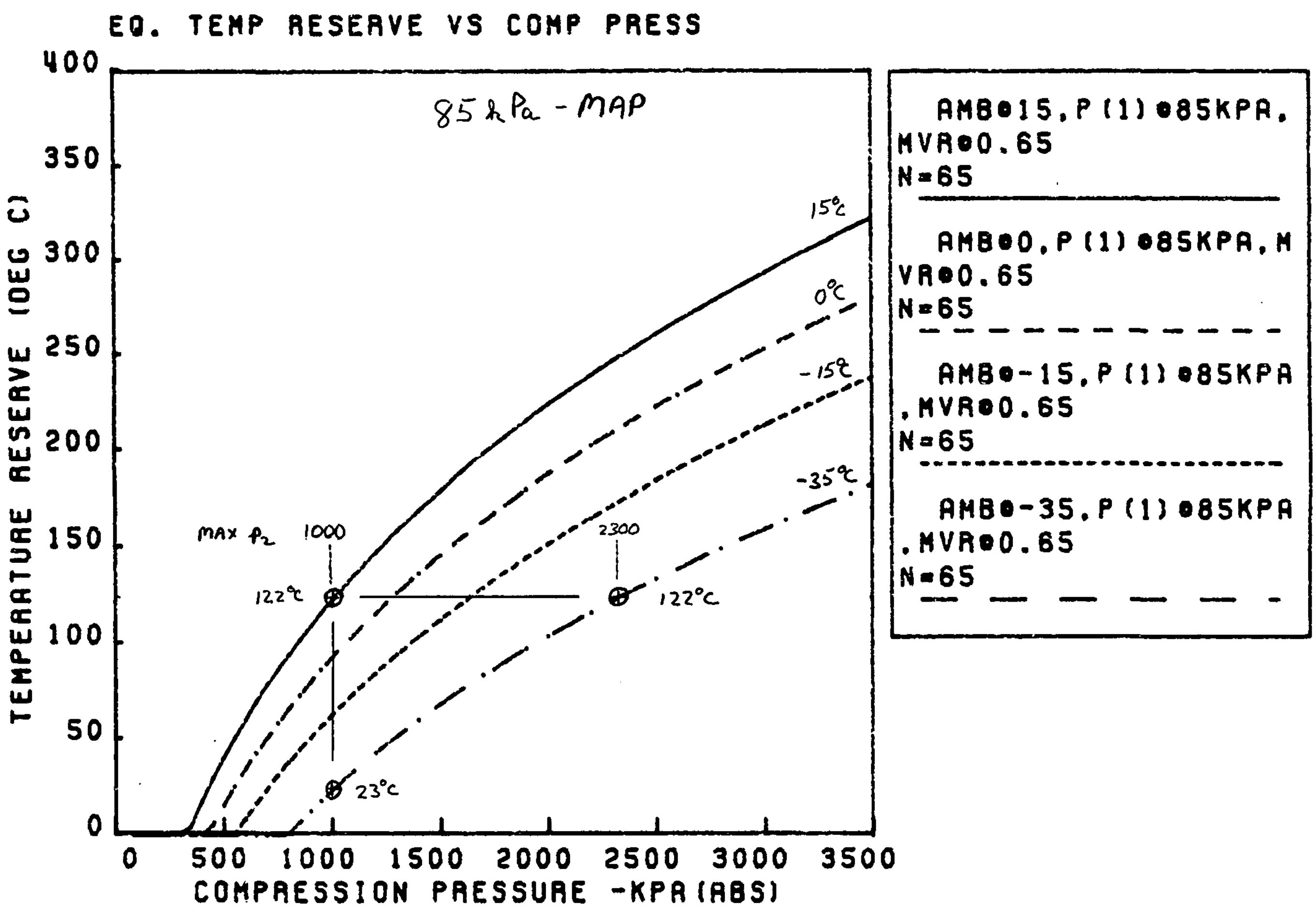
A typical 4 cylinder engine was motored at speeds from 200 RPM to 600 RPM. An analysis of the vapor equivalence ratio versus ambient temperature (at the rounded MAP values from Table III-4) was shown in figure III-1, and indicated relatively little improvement in the ability to pre-vaporized the fuel prior to the compression stroke. However, if we were to enter these rounded-off results into equation III-13 for the "MAP" term, and recompute the temperature reserve by

---

* Assumes that measured cranking compression adequately represents the actual "p_2" in the P-V cycle.

** This pressure is assumed to be rather typical of low compression SI engines converted to run on methanol.

*** Our current test engine, a 2 liter Nissan NAPS-Z engine with 12.8:1 compression ratio has a cranking compression of around 1960 kPa-abs (270 psi gage) when motored on a dynamometer at 200 RPM, note the normal in-vehicle cranking speed for this engine is specified as 300 RPM.



EQ. TEMP RESERVE VS COMP PRESS
85 kPa - MAP
TEMPERATURE RESERVE (DEG C)
COMPRESSION PRESSURE -KPA (ABS)
0
50
100
150
200
250
300
350
400
0
500
1000
1500
2000
2500
3000
3500
15°C
0°C
-15°C
-35°C
MAX P_2
1000
2300
122°C
122°C
23°C
AMB@15, P(1)@85KPA, MVR@0.65
N=65
AMB@0, P(1)@85KPA, MVR@0.65
N=65
AMB@-15, P(1)@85KPA, MVR@0.65
N=65
AMB@-35, P(1)@85KPA, MVR@0.65
N=65

FIG. III -5

the method in equation III-15, we would find that the temperature reserve would increase more markedly at the low temperatures. Figure III-6, suggests that for an engine with 1000 kPa maximium cranking pressure and 85 kPa MAP , the temperature reserve at 15°C ambient temperature would be increased by about 35 percent by lowering the MAP from 85 kPa to 65 kPa, but at -35°C ambient, the reserve would be increased by almost 160 percent over the 85 kPa value. In fact the temperature reserve of 60°C at -35°C ambient and 65 kPa is almost 50 percent of the reserve at 15°C ambient and 85 kPa, a condition at which most neat methanol engines should start.

Table III-4

Cranking MAP+

RPM	MAP*	Round-off**
200	91.72	90
400	75.53	75
600	62.86	65

* curve fit (R^2 = .995), MAP = 105.5567 - (.072125)(RPM) 300 RPM = 84 kPa rounded to 85 kPA.

** Round-off represents arbitrarily picked numbers close to the observed values for analysis purposes.

\+ Barometric Pressure = 99.17 kPa (wet).

It should be noted, however, that this analysis assumes that some means is used to maintain the maximum cranking pressure at a constant level when lowering the MAP. If such means were not employed, then the maximum compression pressure would be lowered by the ratio of the new MAP to the old MAP, unless other factors such as leakage during the compression stroke would limit the maximum compression pressure at the higher MAP condition to a value that is below that which could be achieved at the lower MAP condition with no leakage. In comparing the measured p_2 in our engine at the different cranking speeds to that at 200 RPM, the actual p_2 is about 7.5 percent higher at 400 RPM and about 12 percent higher at 600 RPM. For a large V-8 gasoline truck engine these values are about 10 percent at 400 RPM and 18 percent at 600 RPM. Because these values are relatively small, in this analysis we will simply assume a straight ratio of the new MAP value divided by the MAP at the normal cranking speed to be multiplied by the p_2 at the normal cranking speed. Those values will then be rounded to the nearest 50 kPa which will be designated as "MAP corrected p_2" or "corrected p_2".

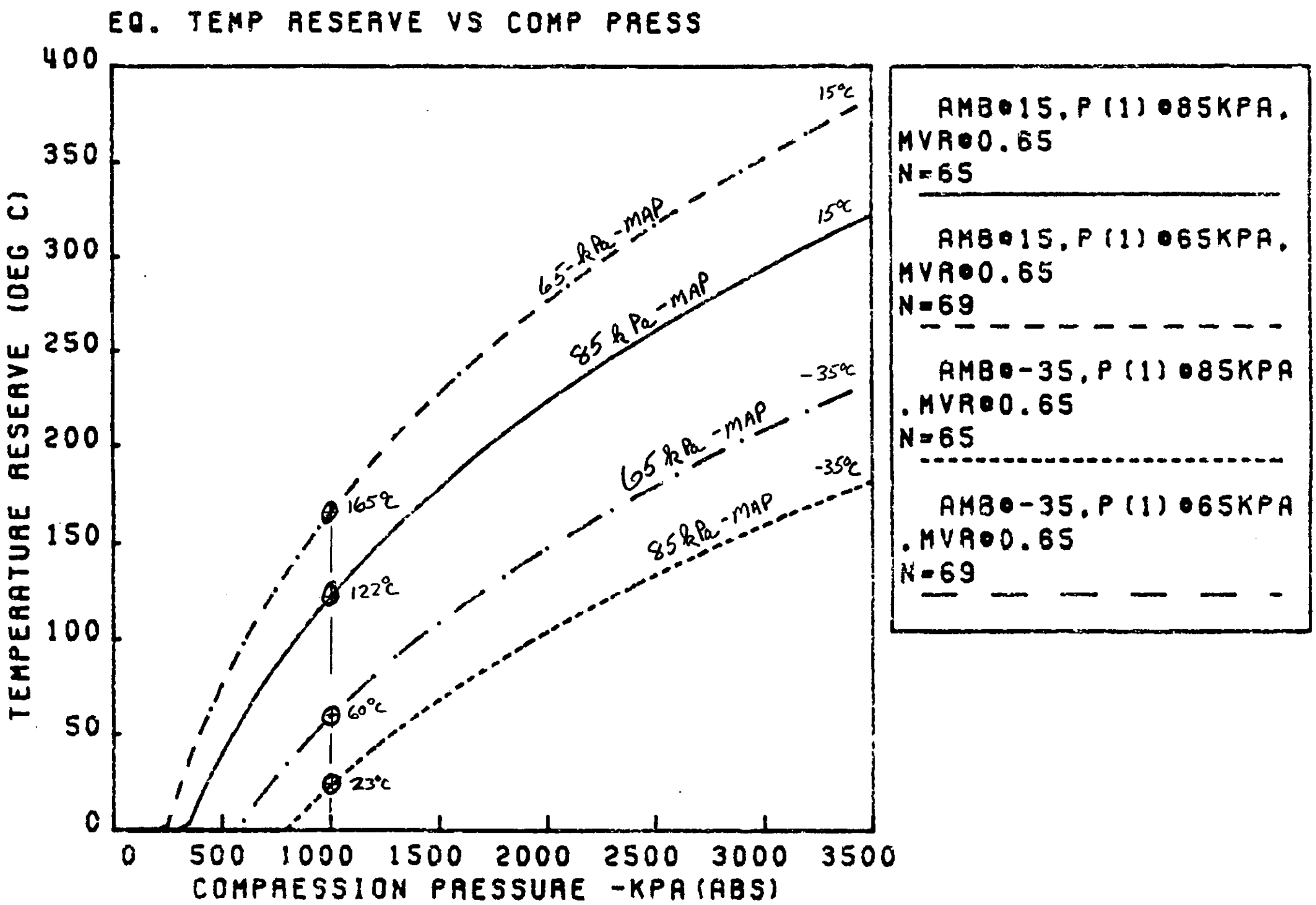

EQ. TEMP RESERVE VS COMP PRESS
TEMPERATURE RESERVE (DEG C)
0
50
100
150
200
250
300
350
400
0
500
1000
1500
2000
2500
3000
3500
COMPRESSION PRESSURE -KPA (ABS)
15°C
65 kPa-MAP
15°C
85 kPa-MAP
-35°C
65 kPa-MAP
-35°C
85 kPa-MAP
165°C
122°C
60°C
23°C
AMB●15, P(1)●85KPA, MVR●0.65
N=65
AMB●15, P(1)●65KPA, MVR●0.65
N=69
AMB●-35, P(1)●85KPA, MVR●0.65
N=65
AMB●-35, P(1)●65KPA, MVR●0.65
N=69

FIG. III - 6

Reconsidering the lower MAP, if there were a means to substantially reduce the cranking MAP to the vicinity of around 30 kPa - absolute (20.5 inches Hg manifold vacuum), we could essentially raise the temperature reserve from 23°C to 179°C at -35°C ambient if we could maintain a 1000 kPa maximum cranking pressure (i.e., 57°C higher than the value for 85 kPa MAP at 15°C ambient - see figure III-7). However, if we could not maintain the maximum cranking pressure constant, and the maximum p_2 was lowered to 350 kPa (ie. MAP corrected p_2), the temperature reserve would be only 46°C -- 76°C lower than the reserve at the 15°C ambient base condition. Achieving a 50 kPa cranking MAP might be more practical.* A 50 kPa cranking MAP with constant peak pressure would still allow a 97°C temperature reserve at -35°C ambient, nearly equivalent to the temperature reserve of 122°C at a normal cranking speed and 15°C ambient. Reducing the peak pressure by the MAP ratio to 600 kPa, would lower the reserve to just 38°C.

Compared to the 23°C reserve for 85kPa MAP and -35°C ambient (see figure III-7), the 38°C reserve at 50 kPa and the 46°C reserve at 30kPa do show an improvement in the temperature reserve with lower cranking MAP values. However, lowering the ambient temperature causes a decrease in the temperature reserve, and this decrease is not compensated by the MAP effect. Therefore, a substantial amount of the normal temperature reserve at 15°C (e.g., the difference between 122°C and 46°C) would be presumably needed to be made-up by other means if the engine were to start. Further, achieving very low cranking MAPs with practical cranking hardware while maintaining sufficient control of the fuel-air mixture could be difficult. Also, smooth transition from a high vacuum crank condition to a normal idle condition might be tricky. Even so, the effect of lower manifold and cylinder pressures should not be over looked. For a consumer acceptable vehicle, the effect of altitude versus lower cylinder pressure may affect the starting capability, and should also be given consideration (i.e., the 5 inches Hg pressure differential due to an altitude change of 1524 meters -5000 ft.- is approximately 17 kPa).

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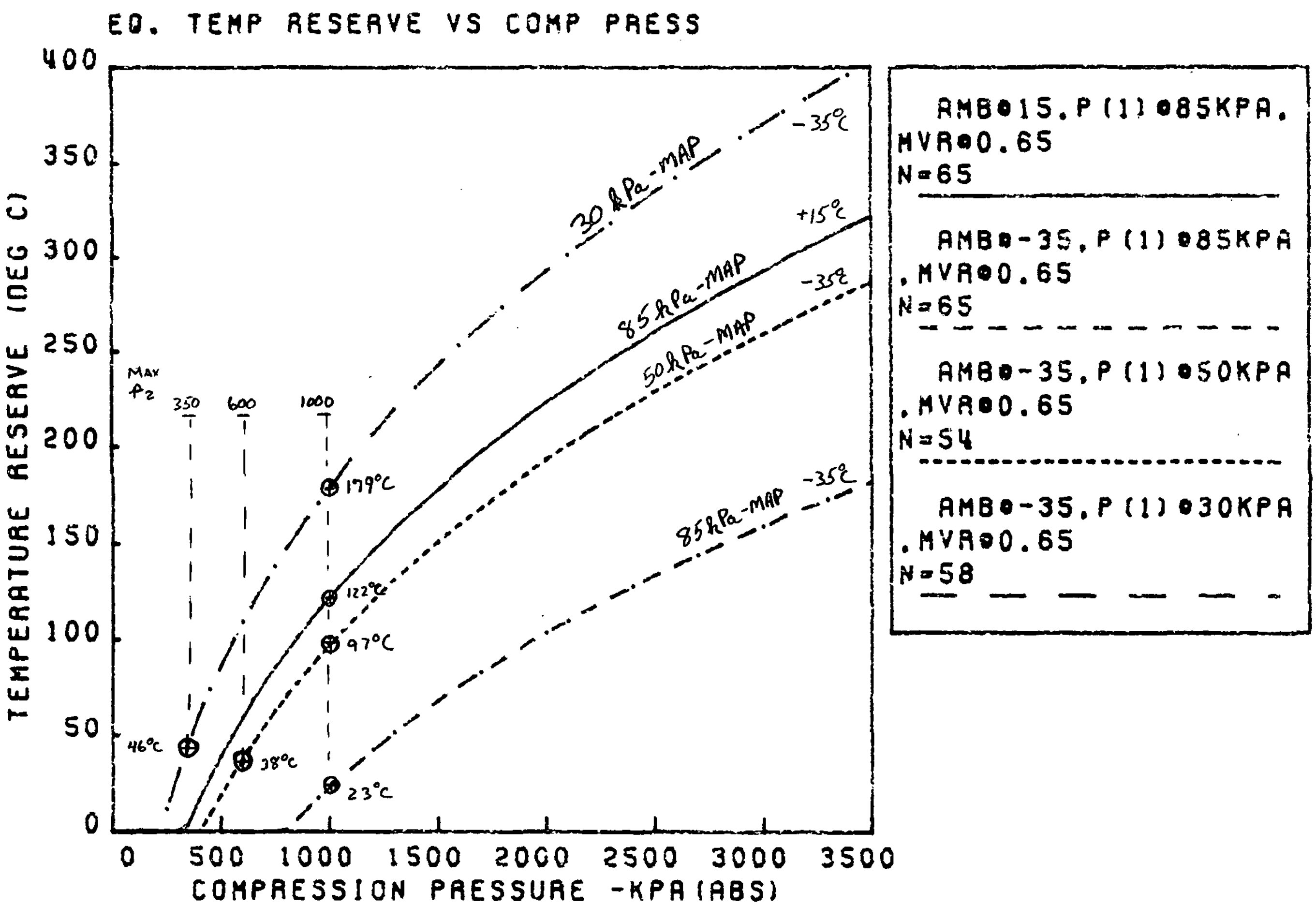
* 50 kPa can be achieved by increasing the cranking speed in our engine to approximately 770 RPM (normal speed is 300 RPM), or presumably by increasing the throttling at the normal cranking speed.



EQ. TEMP RESERVE VS COMP PRESS
TEMPERATURE RESERVE (DEG C)
COMPRESSION PRESSURE -KPA (ABS)
0
50
100
150
200
250
300
350
400
500
1000
1500
2000
2500
3000
3500
MAX p_2
350
600
1000
46°C
38°C
179°C
122°C
97°C
23°C
30 kPa-MAP
-35°C
85 kPa-MAP
+15°C
50 kPa-MAP
-35°C
85 kPa-MAP
-35°C
AMB@15, P(1)@85KPA, MVR@0.65 N=65
AMB@-35, P(1)@85KPA, MVR@0.65 N=65
AMB@-35, P(1)@50KPA, MVR@0.65 N=54
AMB@-35, P(1)@30KPA, MVR@0.65 N=58

FIG. III-7

In discussing means of increasing the temperature reserve, we have digressed somewhat from the original hypothesis of comparing the temperature reserve (TR) at known starting conditions to those at unknown cold conditions. It is generally accepted that the lower temperature limit for unassisted starting of gasoline engines converted to run on methanol is around 10°C (50°F) (10)(11). Without changing pistons such engines usually have rather low compression ratios (in the range of 8:1 to 9:1). A cranking compression of 1000kPa-abs (130 psi gage) was earlier assumed as being reasonably representative of such engines. Under these conditions (10°C ambient and 1000 kPa compression pressure), the temperature reserve would be around 112°C for an MVR of 0.65 and a MAP of 85 kPa (see figure III-8).

Gardiner et. al, (10) extended these limits somewhat by achieving successful cold starts at 0°C. The engine used was a 2 cylinder air cooled 4 cycle SI engine with a compression ratio of 5.84:1. The manufacturer suggests that normal compression pressure on this engine is around 750 kPa-abs (94psi gage). For this pressure and assuming a 65 kPa MAP (since Gardiner's cranking speed was 600 RPM*), the temperature reserve at 0°C ambient with a non-corrected p_2 is 96°C, a value not substantially different from the assumed standard engine at 112°C temperature reserve. The reserve with a corrected p_2 of 550 kPa is 60°C. A visual comparison of Gardiner's engine to the standard engine is shown in figure III-8 as well.

M.A.N., on the other hand, with a modified diesel (CI) engine with spark assist has reported instantaneous starts at -20°C (0°F-ref.12). Informal accounts (15) indicate 3 second starts at -32°C. The exact cranking compression of the M.A.N. engine is unknown to us, but from informal conversations with M.A.N., a range of 25 to 30 bar (2500 to 3000 kPa) was suggested. Typically, CI engines are not throttled, and therefore, the cranking MAP would be closer to atmospheric pressure, or approximately 100 kPa for this analysis. Assuming that the M.A.N. engine has a 100 kPa MAP, the temperature reserve at -35°C ambient would range between 107°C to 131°C (see Figure III-8) for the range of estimated compression pressures, values which are still at or above the value need to start the standard SI engine at +10°C ambient.

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* The normal cranking speed for this engine is between 200 and 400 RPM. A nominal value of 300 RPM and 85 kPA MAP was assumed for this analysis.

If some throttling were used, for instance to achieve 85 kPA MAP during crank, the corrected MAP temperature reserve (approx. 2100 kPa-p_2)at -35°C ambient for the low range (25 bar) would increase from 107°C to 110°C. The reserve would increase approximately 2°C for the higher value (approx. 2500 kPa corrected p_2) as well - see figure III-9. If we were to decrease the MAP even further to 65 kPa, we would increase the temperature reserve another 7°C at the low range and 14°C at the high range (117°C and 147°C respectively). In fact, the temperature reserve for -50° ambient of 109°C at the high compression pressure and 65 kPa MAP is nearly sufficient for starting at -50°C since the reserve of our standard SI engine at 10°C ambient is only 112°C.

These results of increasing temperature reserve with decreasing MAP and corrected p_2 match our earlier observations with one exception. The exception is, in this case decreasing the MAP causes a more observable change in the projected starting capabilities at the lower ambient temperatures. The key to the more observable change lies simply in the fact that the high compression pressures result in temperatures reserves that are initially nearer to the reserve for our standard engine at 10°C.

Table III-5 provides a handy comparison of our analysis for the three engines that have been discussed, the mythical standard SI engine, Gardiner's engine, and the M.A.N. CI spark assisted engine to projections based on the Temperature Reserve concept. High compression pressures and/or a low MAP during cranking appear to be avenues to achieve potential unassisted cold start capabilities in the -30°C to -40°C range -- the high compression through higher in-cylinder temperatures during the cranking cycle, and the high speed cranking through reduction of the partial pressure of the fuel due to the lower MAP.

The high speed cranking may have an additonal factor that assists in the cold starting phenomonon. With the high cranking speed, there exists the potential for more compression strokes to occur during cranking prior to start. Since the actual engine cranking is not adiabatic, each cranking stroke leaves behind some residual fuel vapor at a temperature higher than the ambient mixture, potentially requiring less heat (or lower temperature reserve) on the next stroke to achieve the MVR (minimum vapor equivalence ratio) necessary for starting. Gardiner's starting procedure of cranking the engine at 600 RPM for 10 seconds (i.e., 50 compression strokes) and then resting for 10 seconds followed by two replications of this cycle, allows up to 150 compression strokes to occur for what Gardiner accepted as a

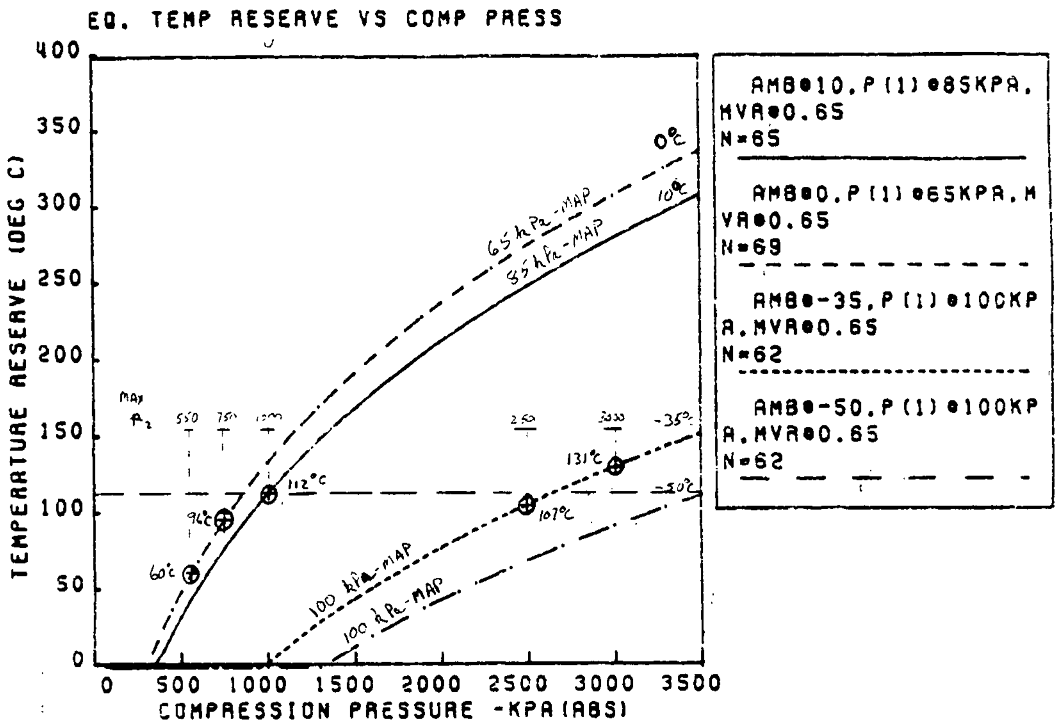

Fig III -8

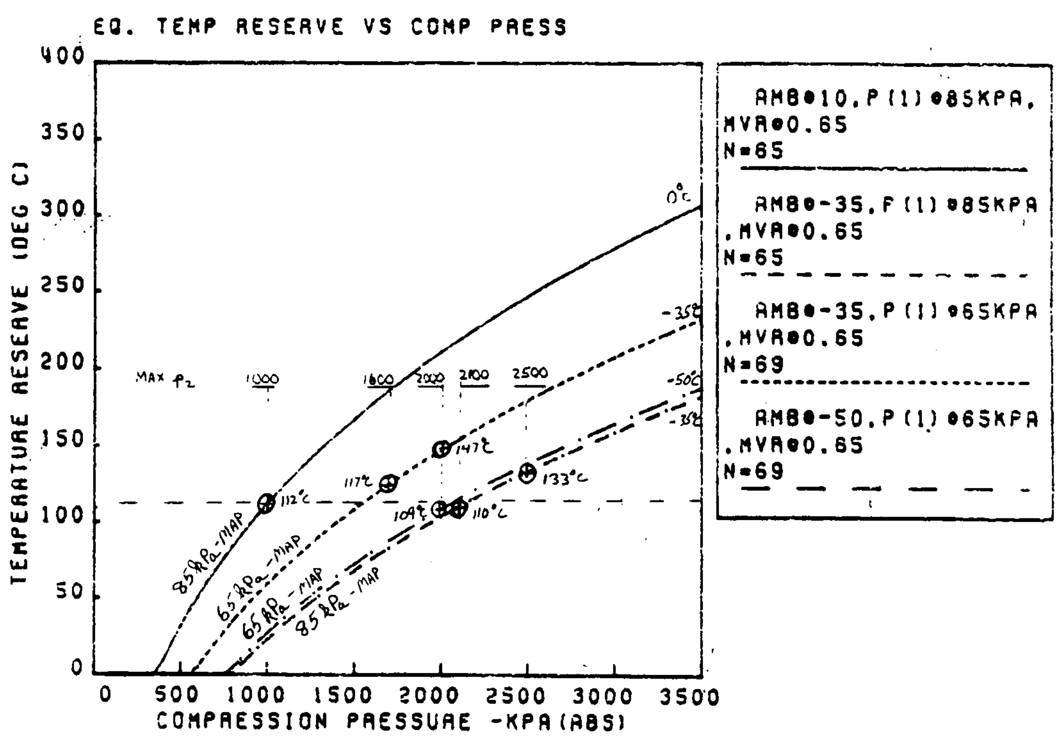

Fig III -9

Table III-5

Actual vs. Projected

Starting Capabilities

Condition	Max Cyl Press(kPa)*	MAP (kPa)	TR (°C)	Ambient Start
Reference				
-Standard SI	1000	85	112	10°C
-Gardiner (600 RPM crank)	550	65	60	0°C
-M.A.N. (12)	2500	100	142	-20°C
	3000	100	173	-20°C
-M.A.N. (15)	2500	100	107	-32°C
	3000	100	131	-32°C
Projected***				
-SI engine with high speed cranking (770 RPM)	600	50	112	- 2°C
-High compression SI engine				
° w/normal cranking (300 RPM)	2100**	85**	112	-34°C
° w/high speed cranking (770 RPM)	1200	50	112	-39°C
-M.A.N. projected capabilities	2500	100	112	-33°C
	3000	100	112	-43°C
-M.A.N. w/intake throttling during crank	1600	65	112	-37°C
	2000	65	112	-49°C

* Corrected P_2

**Max pressure and MAP data from current test engine, a Nissan NAPS-Z with 12.8:1 compression ratio.

***Ambient Start temperatures are linearly interprolated from the computer output matrix based on:

$$AMB_N = AMB_L - ([AMB_L - AMB_H][(TR_S - TR_L)/(TR_H - TR_L)])$$

Where:

AMB_N = new ambient start temperature

AMB_H = ambient temperature with a TR above TR_S

AMB_L = ambient temperature with a TR below TR_S

TR_S = the TR for the standard SI engine at 10°C ambient, 85 kPA-MAP, and 1000 kPa = 111.955°C

TR_H = TR corresponding to AMB_H

TR_L = TR corresponding to AMB_L

successful start. A normal cranking speed (200-300 RPM) would have only 30 to 50 percent as many compression strokes for a given time. Thus if the exhaust residuals which in this case are fuel vapors, build up with successive engine cycles, the high speed cranking could have an advantage in this area as well, and may be partially responsible for Gardiner's ability to start with a lower temperature reserve than the value predicted for other investigators with a so-called standard engine (see figure III-8 and Table III-5).

Detroit Diesel Allison (DDAD) utilized a similar approach with exhaust residuals to develop successful autoignition in its methanol bus engine (13). The autoignition temperature of methanol was identified by DDAD, and the temperature loss due to methanol evaporation was calculated. DDAD then increased boost pressure and temperature, and varied exhaust back pressure to increase the exhaust residuals in the engine in order to recover the original temperature. Quite possibly a butterfly valve in the exhaust pipe similar to those used for early fuel evaporation systems (EFE) on V-8 gasoline engines, when applied to 4 cylinder methanol engines would, when closed during cranking, increase the residual fuel vapor retained in the cylinder from successive cranking cycles. Another possibility would be to delay the fuel (which is possible on electronically fuel injected engines) until after several cranking revolutions in order to have hotter cranking residuals.

Since Table III-5 suggests the potential starting capability in the -30 to -40°C range for high compression ratio engines, and since there is not a plethora of methanol engines with -30 to -40°C starting capability, other factors need to be examined (Note also that most of the cold start studies reported in the literature have generally been with relatively low mechanical compression or low cranking compression engines). First the analysis presented is a bulk equilibrium analysis, and is intended only to show the outer limits of potential start capabilities (i.e., if equilibrium conditions cannot support a combustible vapor mixture, non-equilibrium conditions do not stand much of a chance for starting). In reality, the evaporation process which occurs during the compression cycle is a series of non-equilibrium processes that occur during the finite time of the piston's upward travel. Time is a factor, in heat transfer to both the methanol droplets and to the surrounding structure. Compression heat is continuously transferred to both at a rate depending on the difference in temperature and the heat transfer coefficient for each. The maximum allowable evaporation rate of methanol plays a critical role in determining if a sufficient amount of fuel can be evaporated by the end of the compression stroke.

It is apparent that increasing the compression pressure increases the temperature reserve which should increase the potential for successful starting at cold ambient temperatures. Slightly less apparent is the observation that when the temperature reserve is the greatest (i.e., normal summer temperatures, 25° - 40°C), the portion of the initial liquid fuel that can be vaporized in the inlet manifold (based on equilibrium conditions) is also the most favorable (see figure III-1) thus tending to prevent the evaporation rate of methanol from being a potential limiting factor because the amount that is prevaporized can be a sizable portion of the MVR (20%-30%). Both factors (increased temperature reserve and increased manifold vapor) enhance starting capabilities. Conversely, when the temperature reserve is the smallest (i.e, low ambient temperature), the amount that can be vaporized in the inlet manifold is also minimal, which indicates low ambient temperature conditions have two factors working against successful cold starts - (1) reduced ability to vaporize the fuel during the compression stroke, and (2) reduced capability to pre-vaporize some fuel in the intake manifold prior to the compression stroke.

To further explain the failure of real methanol engines to start at low ambient temperatures for which their theoretical temperature reserve seems adequate based on the testing of low compression engines at higher ambient temperatures (10+°C), we must examine how ambient temperature itself effects the non-equilibrium processes which are at work. One fact which seems important is that as the ambient temperature decreases, the difference between the peak compression temperature and the ambient temperature also decreases. This temperature difference is roughly the driving force for initial heating of fuel droplets which began at ambient temperature. Later, when fuel droplets can be approximated as having been warmed to the boiling point corresponding to the total pressure in the cylinder, the driving temperature difference for heat transfer will also be reduced at lower ambient temperatures since the peak compression temperature is reduced but total cylinder pressure is not. Thus, heat transfer and evaporation will be slower for a given temperature reserve at lower ambient temperatures, impairing starting ability.

In fact, the evaporation rate of methanol may be the only extremely critical factor in starting a methanol engine under cold ambient conditions. When evaluating the M.A.N.

system* (12), it is interesting to note that the fuel is injected late in the compression stroke. This provides a very large temperature difference between the air temperature and the fuel temperature which enhances heat transfer to the fuel. Secondly, and possibly more important is that the engine design does not attempt to evaporate all of the fuel at once. Instead, the fuel is deposited in a bowl in the piston. There, only the exposed surface is open for evaporation, and the evaporation that does occur, cools the piston, not just the air. After ignition by a spark plug, the flame provides for controlled evaporation and combustion of the remaining fuel in a manner that allows an acceptable pressure rise in the cylinder.

Gardiner, on the other hand, indicated that increasing the amount of liquid fuel to the engine from an initial liquid equivalence ratio of about 1.0 to about 3.0 improved the cold starting performance of a carbureted SI engine. However, increasing the initial liquid ratio from 3.0 out to 10.0 made little change in the starting performance. One explanation of this may be that the additional fuel surface area achieved by an equivalence ratio of 3.0 was enough to prevent the evaporation rate from being a limiting factor, such that the work of compression was the limiting factor as equivalence ratio was further increased. A given amount of compression work will support only a certain amount of methanol vapor at equilibrium conditions for a given ambient temperature (see figure III-2). Another explanation is that the evaporation rate was limiting throughout Gardiner's range of equivalence ratios, but because of poor fuel atomimization and distribution, the fuel surface area in the cylinder did not increase past an equivalence ratio of 3.0. Gardiner's own explanation is also plausible, namely that surface area and evaporation rate did steadily increase with increasing equivalence ratio, but (expressed in our terms), the temperature reserve was possibly consumed by the need to heat large amounts of liquid fuel to the boiling point. Gardiner does not indicate whether he attempted operation beyond an initial liquid equivalence ratio of 10.0.

From this information, a hypothesis is put forward that may be difficult to justify, but none the less may stimulate further thought in this area. It is hypothesized that under cold starting conditions that the rate of evaporation of methanol is the limiting factor. During the compression stroke, it is hypothesized that the gas temperature remains

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*A modified diesel engine with spark assist.

well above the saturation temperature for the partial pressure of methanol; but because rate limitations have not allowed sufficient methanol to evaporate, the temperature has not been driven down. Figure 5 in Bardon's paper (2) which shows the measured cylinder temperature after methanol vaporization to be much greater than the theoretical temperature after vaporization tends to support this hypothesis. If sufficient methanol has not evaporated to drive the temperature down, also sufficient methanol vapor is not available to form a combustible mixture.

Increasing the overall liquid mixture in order to potentially increase the surface area of the fuel available for heat transfer and subsequent evaporation (such that a combustible mixture is formed) would require that the remainder of this excess liquid fuel must be consumed by the flame. However, if the excess fuel to be consumed by the flame was delivered to the combustion chamber in a manner such that large droplets remain after the initial vaporization, there is evidence to suggest that the flame will not propagate due to insufficient preliminary volatilization of the large droplets. Hence, the initial flame front could essentially be quenched by the large droplets resulting from the excess fuel causing a no-start condition.

Burgoyne and Cohen (20) conducted an experiment with tetralin and air mixtures which evaluated the flame propagation of various sized fuel droplets surrounded by just-flammable fuel vapor. Their apparatus was configured such that the droplets flowed downward through a burning zone at the end of a long tube. The combustion was supported by the just-flammable fuel vapor, but the vapor flame would not propagate up the tube. When the droplets were added in appropriate amounts the flame then propagated up the tube. They found that for droplets above 300 microns in diameter, the flame would not propagate upwards against the flow which they suggest is due to the velocity of the fall being greater than the burning velocity such that the mass motion of the droplets prevents propagation. Such mass motion is always evident to some degree within the cylinder during the compression stroke. The spark plug could be envisioned as the location of the just-flamable burning zone and the droplets would pass through this region due to the swirl in the cylinder. This might suggest that the more quiescent combustion chambers need less temperature reserve for starting than one with high swirl even though the high swirl may be more desirable once the engine is running. Conversely, the higher swirl engine may need a larger MVR to compensate the initial vapor flame speed for the higher local air velocities.

Another aspect of Burgoyne's and Cohen's work is the concept of two regimes of droplet evaporation prior to combustion. The two methods are evaporation controlled by diffusion, and evaporation controlled by heat transfer. The boundary between the two mechanisms is controlled by what they define as the "stabilized temperature" which is defined as the " 'wet-bulb' temperature which is below the boiling-point by reason of the heat carried away by the vapour". If we compare the heat of vaporization between methanol (474 BTU/lb or 1.102 J/kg - ref. 4) to that of tetralin (0.201 J/kg - ref. 23), we see that the amount of heat that can be absorbed by the heat of vaporization is about 5.5 times greater for methanol than for tetralin. Therefore, one would expect the "stabilized temperature" for methanol to be much higher than that for tetralin because of the additonal heat transferred away from the droplet by the evaporating methanol.

Burgoyne and Cohen also point out that aerosols (droplets under 10 microns) have much higher burning velocities because the time required to evaporate the droplets is much shorter than for larger droplets. If we assume that droplet evaporation will occur much more quickly in the heat transfer controlled regime than in the diffusion controlled regime, then because of the higher stabilized temperature for methanol which keeps the methanol in the diffusion controlled regime longer, one would expect the pre-flame evaporation of methanol droplets to take much longer than tetralin.

Transferring these methanol pre-flame evaporation assumptions back to Burgoyne's and Cohen's propagation tube results, we would predict that the 300 micron upper droplet size limit for propagation with tetralin in air mixtures would be reduced (possibly substantially) for methanol in air mixtures because of the longer time needed for pre-flame volatilization of the methanol droplets. The lack of flame propagation due to the pre-flame evaporation rate and droplet size factors could explain the failure of other experimentors to achieve cold start temperatures as low as predicted here through purely liquid enrichment. To be accurate, however, it must be pointed out that not enough experimentors have reported cold start investigations with high compression ratios or low MAPs to allow a fair appraisal of the limitations of unassisted methanol cold starts. Furthermore, some experimentors have added liquid fuel in a way which may not have increased the surface area available for evaporation (the additional fuel may have merely pooled in the manifold or have merely formed a thicker layer on the cylinder walls), or added fuel in a way that the fuel droplets may have congealed forming droplets too large to allow pre-flame volatilization and subsequent flame propagation.

Based on this hypothesis, CI engines and modified CI engines (M.A.N.) are expected to be easier to start than homogeneous charged SI engines. There are several factors to support this statement. First, CI engines almost always have higher compression ratios which have correspondingly higher compression temperatures. Also, CI engines inject the fuel late in the compression stroke where there is a much larger temperature difference between the air and the fuel (which should increase the vaporization rate). Further, because the fuel is injected by a high pressure injection system, it usually has smaller droplets than a homogeneous charge engine which enhances both diffusion controlled evaporation and pre-flame heat transfer controlled evaporation. Lastly, the initial part of the injection cycle creates a very lean f/a ratio which also improves partial pressure or diffusion controlled evaporation. All four factors, higher compression temperature, larger temperature differential, smaller droplets, and lean f/a ratio enhance the evaporation rate. Because the potential for the evaporation rate during the initial part of the injection cycle is much greater under the CI type conditions than those in an SI engine, less overall liquid fuel should be required to achieve a combustible vapor. Therefore, since less overall liquid may be required, the probability of excess liquid or large droplets quenching the initial flame front would most likely be reduced. Finally, even if excess liquid does occur, certain CI combustion systems (e.g., M.A.N.) allow for controlled evaporation of the excess liquid in order to prevent quenching of the pilot flame.

Since many of the features that make cold starting more probable with a CI or late (near TDC) injecting style of engine may not be easily transferred to an SI engine, the outlook for the cold start performance of homogeneous charged SI engine does not look promising. Items that may be critical to cold start performance of the SI methanol engine without starting aids include devising means to increase the evaporation rates through such mechanisms as higher compression temperatures through increased compression ratios, smaller droplet sizes through improved fuel systems (e.g. EFI, sequential EFI, higher pressure EFI, or other atomization systems), higher cranking speeds or increased throttling, reduced heat transfer to the structure during the compression stroke, increased heat transfer between the air and the fuel during the compression stroke, and possibly devising means to better use the cranking residuals. Flash vaporization might be another possibility as described by Oza (14), however this technique will be discussed more thoroughly in the chapter on physical vaporization.

C. Summary

In summary, we have identified several instances of successful cold starting of unassisted methanol engines (Gardiner (10) at 0°C, and M.A.N. officially at -20°C, unofficially at -32°C). A hypothesis has been developed that suggests that the limiting factor in unassisted methanol cold starting is not the equilibrium vapor pressure curve, but the rate and time sequence of the fuel evaporation. Experimental work by Gardiner (10), and modeling by Bardon (2) and Browning (3) tend to support this hypothesis.

This information then suggests that the potential for cold starting a methanol engine without assist at very low ambient temperatures is still a possibility. High compressions ratios and smaller droplets are two possible factors required to achieve acceptable cold start capabilities. A list of perceived advantages and disadvantages is given below.

1. Advantages

a) Cost of Vehicle Fuel System - Most likely the cost of a single-fuel vehicle fuel system for unassisted methanol cold starting will be less expensive than one with auxiliary heaters and associated power requirements. Certainly the single-fuel system should be less than dual fuel systems.

b) Cost of Fuel - Methanol with no volatility additives should cost less than methanol with volatility additives.

c) Emissions - With a single fuel, it will most likely be easier to measure, characterize, and control both exhaust and evaporative emissions compared to methanol with volatility additives or dual fuel systems.

d) Enforcement - With a single fuel, disputes over using or selling a winter fuel during the summer or in an unacceptable region of the country will not occur.

e) Vehicle Maintenance - A single-fuel vehicle fuel system does not need the range of authority that a system would need for methanol with volatility additives. This could make the single-fuel system less complex and easier to maintain. The single- fuel system would almost certainly be less complex than dual fuel systems.

2. Disadvantages

a) Universal Demonstration - With the exception of a few situations, universal application of successful cold start technology has not been demonstrated. No successful cold start with SI engines in the -35°C range have been demonstrated.