HUMIDITY AND TEMPERATURE CORRECTION FACTORS FOR NO_X EMISSIONS FROM SPARK IGNITED ENGINES

FINAL REPORT

SwRI® Project No. 03.10038

Prepared for

ENVIRON International Corporation
101 Rowland Way, Suite 220
Novato, CA 94945-5010

October 2003

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SAN ANTONIO
HOUSTON
DETROIT
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Prepared by:

Jess W. Gingrich, Timothy J. Callahan, and Lee G. Dodge
Southwest Research Institute
6220 Culebra Road
San Antonio, Texas

October 2003

Approved:

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Bruce B. Bykowski, Director
Department of Engine and Emissions Research
Engine, Emissions, and Vehicle Research Division

TABLE OF CONTENTS

EXECUTIVE SUMMARY

All of the current humidity correction factors for NO_x were found to be based on historical data taken in 1971 and 1972. Some of the engines today are more technically advanced than those engines, incorporating port or throttle-body fuel injection, air-fuel ratio feedback, exhaust aftertreatment, and knock detection. While many off-road vehicles do not have all of these features, this technology is becoming more prevalent in those engines as well. The analysis conducted for this project indicated that the historical correction factors do not adequately account for operating cycles with higher load factors, or advanced technologies such as A/F control and knock detection. No engine test data were found documenting humidity effects for these additional variables. Therefore, the recommendations given here were based on the correlations developed from engine tests conducted in the early 1970's, and the slopes for those correlations were adjusted based on engine modeling results that addressed the effect of higher load factors, A/F controlled to a constant value, and A/F fixed at a different value from the earlier tests.

The model results showed these effects to be significant and the results were used to modify the historical correction procedures. If a more rigorous approach is desired, SwRI would recommend engine testing to quantify the effects for different engine/vehicle classes.

The recommended equation to adjust standardized emissions for **carbureted heavy-duty on-road or off-road (above 19kW) engines** under non-standard inlet air conditions takes the following form:

$$C_{SwRI}(H,T) = 1 + 0.0022\cdot(T - 25) - 0.0280\cdot(H - 10.71) \qquad (13)$$

Where:
T = Temperature of the inlet air [^oC]
H =Absolute humidity of the inlet air [g of H_2O/kg of dry air]

For **heavy-duty on-road or off-road (above 19kW) spark-ignition engines that use a 3-way catalyst** (A/F control, typically with port fuel injectors), the recommended NO_x correction equation is as follows:

$$C_{SwRI}(H) = 1 - 0.0232\cdot(H - 10.71) \qquad (11)$$

with no correction for ambient temperature.

For **light-duty, spark-ignition engines**, the recommended practice is whatever procedure is used in Mobile 6, which can be approximated by Equation 4.

$$C = NOx_{corr\_MOBILE}(H_a) = \begin{cases} 1.2 \ \text{ if } H_a \le 20 \\ (-0.004\cdot H_a + 1.28) \ \text{ if } 20 < H_a < 120 \\ 0.8 \ \text{ if } H_a \ge 120 \end{cases} \qquad (4)$$

Where:
H_a = Absolute humidity of the inlet air [grains/lb]

For **small off-road, spark-ignition engines (< 19kW)**, the recommended practice is,

(14)

$$C = 1 - \frac{546}{AFR} \cdot (\omega - 0.01071)$$

Where:
AFR = Air-fuel ratio of the engine
ω= Absolute humidity of the inlet air [kg/kg]

1.0 BACKGROUND

Emission regulations continue to place additional restrictions on urban areas trying to achieve ambient air quality standards. Although ambient air quality standards are national, achieving the standards is a regional problem delegated to the states. However, the certification procedures for on-road and off-road spark-ignited engines are standardized without regard for regional variation in ambient conditions like temperature and humidity. As early as 1970[(1)], it was recognized that the concentration of oxides of nitrogen (NO_x) in engine exhaust is significantly affected by the thermodynamic conditions of the intake air. Specifically, the intake air temperature and humidity have the dominant effects[(1)(2)(3)]. Because of these sensitivities, it is reasonable to assume regional variations in temperature and humidity can significantly impact engine-out emission levels. Emissions inventory models such as the Environmental Protection Agency's (EPA) MOBILE and NONROAD[(4)(5)(6)] have been developed to account for pollutants attributed to both on-road and off-road mobile sources. These models use local information to adjust the inventory based on average regional temperature and humidity for specific categories of engines.

Historically, the impact of ambient temperature and humidity on emissions was of interest because it was difficult to make comparisons of the NO_x emissions from engines tested at different locations and/or with variations in the ambient conditions. In an effort to allow these day-to-day and location-to-location comparisons, various correction factors have been developed. The goal for all of these correction factors is to standardize the NO_x emissions back to selected standard reference conditions, or to provide an adjustment to the emissions inventory models enabling a more accurate prediction of ambient air quality.

In light of the pressure on states and urban areas for implementing and achieving air quality standards, it seems appropriate to account for regional differences imposed by prevailing ambient conditions. Of particular interest is the impact of ambient conditions on oxides of nitrogen NO_x, a major contributor to ambient air ozone levels.

2.0 OBJECTIVE

The objectives of this project were to review existing data and correction procedures for adjusting spark-ignited Otto-cycle engine NO_x levels for ambient temperature and humidity, and to assess the applicability of these procedures for a number of different mobile sources.

3.0 APPROACH

The existing procedures for correcting NO_x emission levels during standardized tests for ambient temperature and humidity were reviewed, along with the original reference work that developed these procedures. The correction procedures were compared to each other and to accepted engine performance and emission models for quantitative effects. Recommendations were then made on the application of the correction factors to the engine subcategories. For the purpose of this text, the main category should be considered spark-ignited, Otto-cycle engines containing the subcategories: light-duty vehicles and engines, heavy-duty vehicles and engines, and off-road engines.

3.1 Existing NO_x Correction Procedures

A survey of standardized procedures found two methods for correcting ambient humidity and temperature during engine and vehicle tests. The first one is for heavy-duty engines, and the second is another used for light-duty on-road sources as well as off-road mobile sources such as recreational, small off-road, and marine SI engines. Current emissions models such as MOBILE6 and the model developed through the California Air Resource Board (CARB), EMFAC2002, were also explored to identify alternative methods currently in use to correct NOx for ambient temperature and humidity. Other correction algorithms have been developed for specialized cases, though not found in a standardized procedure.

3.1.1 Standardized Corrections

The EPA has promulgated the following correction factor (KH in English units and KH_{SI} in SI units) for NOx based on ambient humidity in multiple sections of CFR Title 40[7]. The correction factor is based on work performed by Manos in 1973[2]:

$$KH = \frac{1}{1 - .0047 \cdot (H - 75)} \qquad KH_{SI} = \frac{1}{1 - .0329 \cdot \left(H_{SI} - 10.71\right)} \qquad (1)$$

Where:
H = Absolute humidity of the inlet air [grains H_2O/pound dry air]
H_{SI} = Absolute humidity of the inlet air [grams H_2O/kg dry air]

The standard absolute humidity for the EPA is 75 grains/lb or 10.71 g/kg. These equations, in some form, are used in:

1. CFR Title 40 §86.144-94 for 1977 and later model year light-duty vehicles
2. CFR Title 40 §86.1342-90 for transient tests on Otto-cycle light-duty engines
3. CFR Title 40 §90.419 for small spark ignited off-road engines below 19kW
4. CFR Title 40 §91.419 for marine spark-ignited engines
5. CFR Title 40 §1051.501 for off-highway vehicles including ATV's and snowmobiles.

While the equation is consistent throughout Title 40, it should be noted that the use of the correction factor is not defined uniformly. In some instances KH is defined as a multiplicative correction factor to the NO concentration, while other sections define KH as the correction for the humidity effects on NO_2 formation. However, in practice, the applications of these correction factors have all been applied to the total NO_x emission numbers. Equation 1 has also been incorporated into SAE J1088, a test procedure for measuring gaseous emissions from small utility engines[(8)], and in the Texas Natural Resource Conservation Commission Technical Analysis Division specifications for vehicle exhaust gas analyzer systems[(9)]. CARB has also uses this equation in their exhaust emissions standards and test procedures for 2001 model year and later spark-ignited marine engines[(10)] and small off-road engines[(11)]. All of the previous procedures define KH be set to 1 for two-stroke-engines. This definition is not explained in the CFR, but Brereton and Bertrand explain that carbureted two-stroke handheld engines are particularly hard to characterize from a regulator perspective because of the erratic dependence exhaust emissions have on ambient temperature and humidity[(12)].

The original work performed by Manos tested eight vehicles based on the Federal Register Volume 30, Number 108. These vehicles were selected to represent the various engine configurations and carburetion systems found in the United States at that time. During the tests, humidity was varied from 20 to 180 grains of H_2O per pound of dry air (~2.85 to 25.2 g/kg); however, the regression analysis excluded the data above 120 grains per pound (~17.2 g/kg). The temperature range was determined in accordance with the Federal Register to be between 68 and 86 °F.

For gasoline-fueled heavy-duty engines, the EPA presents a correction factor for NO_x based on the humidity of the inlet air. This correction factor was established based on the work of Krause[(3)] in 1971. The vehicles were tested according to the Federal Heavy-Duty Test cycle and the resulting humidity correction can be calculated with the following equation:

$$KH_{HDV}(G) = 0.6272 + .00629 \cdot G - .0000176 \cdot G^2 \tag{2}$$

Where:
G = Absolute humidity of the inlet air [grains H_2O/pound dry air]

The promulgated correction is solely a function of the inlet air humidity. The original correction equation was a regression of the observed dependence of NO concentration in ppm on the inlet air humidity. The range of absolute humidities tested was from 20 to 110 grains/pound. Krause also established a correction equation for the mass emissions of NO_2 g/bhp-hr as seen in the following equation:

$$KH_{HDV\_NO2}(G) = 0.634 + .00654 \cdot G - .0000222 \cdot G^2 \tag{3}$$

Equations 2 and 3, in conjunction with NO_x emissions modeled in Southwest Research Institute's ALAMO_ENGINE cycle simulation computer model, have shown that correction factors established with concentration data and the corresponding mass-based emissions are similar. Therefore, a correction factor developed from concentration data imparts little error when applied to the mass-based emissions of an engine. Krause developed equations to correct carbon monoxide and unburned hydrocarbons emissions for ambient conditions, but the statistical correlation's were not as strong as those from which equations 2 and 3 were derived.

The correction factors shown in equations 1-3 are used to adjust measurements of NO_x to a reference humidity. The emission inventory models use correction factors in an inverse manner, to convert a standardized emissions rate to an actual rate. The equation and figures in the following sections are defined to adjust a standard emission rate to an actual rate based on ambient conditions.

3.1.2 EPA Emission Inventory Models

The emissions inventory models used by EPA and CARB correct NOx emissions based on the average ambient conditions during engine operation. Emission models have been developed for both on-road and off-road mobile sources. The NOx correction factor used for a light-duty gasoline vehicle in EPA's MOBILE6 is shown in the following equation[13] and Figure 1 (equation is estimated from figure provided in EPA documents):

$$NOx_{corr\_MOBILE}(H_a) := \begin{cases} 1.2 \text{ if } H_a \le 20 \\ (-.004 \cdot H_a + 1.28) \text{ if } 20 < H_a < 120 \\ .8 \text{ if } H_a \ge 120 \end{cases} \quad (4)$$

Where:
H_a = Absolute humidity of the inlet air [grains/lb]

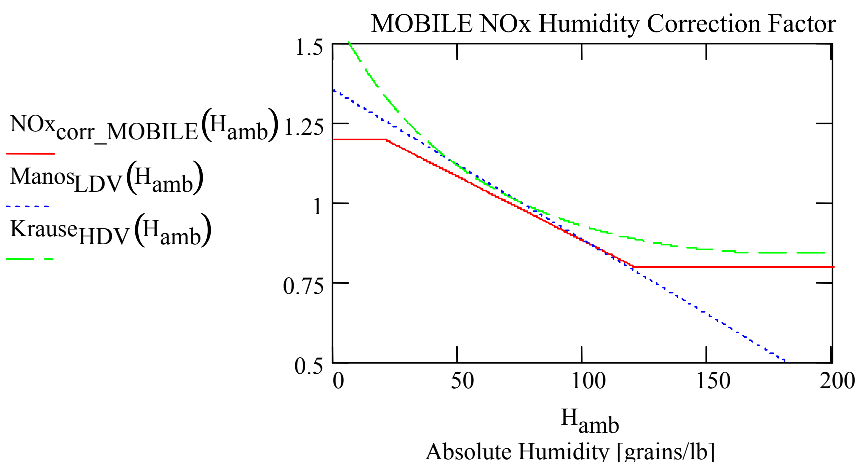

Figure 1. Humidity Correction Factors as a function of Intake Air Absolute Humidity for Three Cases: MOBILE6 (equation 6), Reciprocal of Manos Light-duty Vehicle (Equation 1), and Reciprocal of Krause Heavy-duty Vehicle (Equation 2).

Figure 1 shows three correction equations as a function of the inlet air humidity. While the empirical basis of the MOBILE6 function was not found, it appears to resemble the equation developed by Manos in the humidity ranges where the original regression analysis was performed. Continuing this speculation, it is probable that MOBILE6 model developers were not willing to extrapolate the Manos equation beyond the bounds of the regression analysis, and therefore, may not be accounting for the actual humidity effects on NO_x formation at high humidity.

Temperature correction factors in MOBILE6 are determined separately for each of the three segments of the FTP for light-duty gasoline fueled vehicles. For ambient temperatures below 75-°F the temperature correction factor (TCF) for NO_x emissions is as follows[4]:

$$\mathrm{TCF(b)} = e^{[\mathrm{TC(b)} \cdot (\mathrm{T}-75)]} \tag{5}$$

Where:
TC(b) = Coefficient for the particular test segment
T = Ambient temperature [°F]

TC(b) is dependent on the test segment, the ambient temperature, and the model year of the vehicle. At certain temperatures the TCF factor is combined with effects of fuel volatility as measured by the Reid vapor pressure. EPA's NONROAD2002, the off-road emissions model, calculates the correction factors with the same algorithm as MOBILE6, but with a matrix of TC(b) specific to off-road, four-stroke engines. NONROAD2002 does not apply a correction factor to two-stroke engine emissions due to a lack of data for these engine types.

3.1.3 CARB Emission Inventory Model

The CARB motor vehicle emission inventory model, EMFAC2002, corrects NO_x emissions for the ambient conditions where the vehicle/engine operates. CARB based their humidity correction methodology on that published by Manos. CARB expands on the original methodology by adding a factor that is determined by the technology class for a given vehicle. The technology classes are differentiated by the method of fueling, such as multi-point fuel injection or carburetion, and the exhaust aftertreatment. The factors were created through a linear regression analysis of data recorded between 1989 to 1995 including 885 light-duty trucks, 116 medium-duty vehicles and 3447 passenger vehicles ranging in model 1962 to 1995[6]. The equation developed to estimate NO_x emissions for a vehicle operating at a humidity other than the standard 75 grains/pound takes the form:

$$E_{amb} = E_{Standard} \cdot \frac{\left(1 + m_{manos}\left(H_T - H_S\right)\right) \cdot \left(1 + m_{class}\left(H_{amb} - H_S\right)\right)}{1 + m_{class}\left(H_T - H_S\right)} \tag{6}$$

Where:
E_{amb} = Corrected NOx mass emission
$E_{Standard}$ = NOx mass emissions at standard conditions
m_{manos} = -0.0047
m_{class} = ARB developed technology factors
H_T = Absolute humidity during the vehicle/ test [grains/lb]
H_S = Standard absolute humidity [grains/lb]
H_{amb} = Ambient absolute humidity during vehicle operation [grains/lb]

The m_{class} factors were obtained through linear regressions. None of the published statistical R^2 values were larger than 0.033. Therefore, the adjusted correction has no statistical benefit over the original equation published by Manos.

EMFAC2002 corrects for the ambient temperature with technology specific correction factors for on-highway vehicles. The base equation takes the following form:

$$TCF = A\cdot(T-75) + B\cdot(T-75)^2 + C\cdot(T-75)^3 \qquad (7)$$

Where:
A, B and C = Technology specific coefficients
T = Ambient temperature [°F]

Technology specific coefficients apply to the individual FTP segments and will adjust for engine differences such as fueling methods, exhaust aftertreatment, and air conditioning.

3.1.4 Specialized Corrections

The effects of ambient conditions on the performance and emissions of two-stroke and four-stroke hand-held engines were explored by Brereton and Bertrand. They established a correction that takes the form:

$$K_H = \frac{1}{1 - \frac{546}{AFR}\cdot(\omega - .01071)} \qquad (8)$$

Where:
AFR = Air-fuel ratio of the engine
ω = Absolute humidity of the inlet air [kg/kg]

This equation accommodates the range of AFR (A/F) that a hand-held carbureted engine may experience in actual use. Equation 8, with an operating A/F of 16, will reduce to the form defined by the EPA for use with small off-road engines.

3.1.5 Comments on Current Correction Practices

Both the light-duty and heavy-duty correction equations, equations 1-3, were developed to correct for the minor variations in ambient humidity during a standardized test. The promulgated corrections are not suited for the wide range of humidity seen throughout the country. Use of any correction at conditions outside of the bound of the original regression should be considered an extrapolation. Equation 1, adopted by the EPA, did not include the effects of temperature because of the limited temperature range, and subsequent minimal effect on NO_x concentration. If the temperature adjustment is not removed from Manos' original equation to adjust an observed NO_x emissions to a standard humidity, it takes the following form:

$$Kh(T_{amb}, H_{amb}) := \frac{7.165}{\left[7.165 + 0.0290\cdot(T_{amb} - 78) - 0.0337\,(H_{amb} - 75)\right]} \qquad (9)$$

Where:
T_{amb} = Ambient temperature [°F]
H_{amb} = Absolute humidity of the inlet air [grains/lb]

Figure 2 graphically represents Equation 9 for a range of possible ambient temperatures and humidities. The patch-marked and non-patch-marked sections of the figure delineate the regions of interpolation and extrapolation, respectively, when using equation 9. The area outside the bounds of the statistical analysis were explored with the cycle simulation code ALAMO_ENGINE to help characterized the validity of such extrapolations.

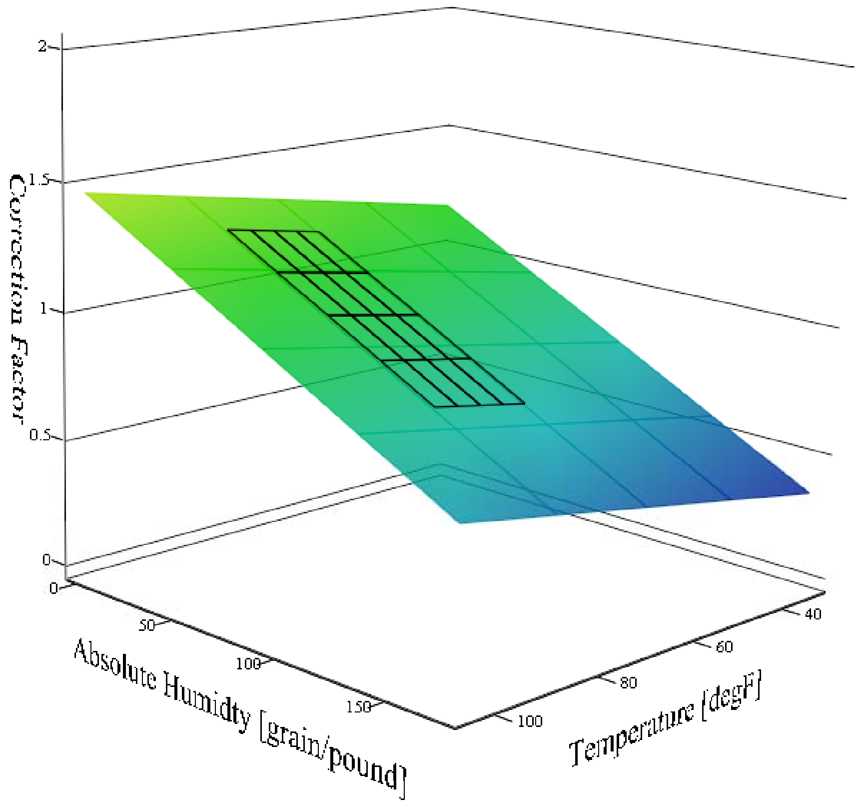

Figure 2. NO_x Correction Factor for Ambient Temperature and Humidity including the Bound of the Statistical Regression

The corrections presented in equations 1-3 were also developed on carbureted engines. The NO_x emissions from a carbureted engine will be affected by changes in humidity through three independent mechanisms. First, the water vapor in the intake will act as a diluent for the combustion charge, reducing the flame temperature, and therefore, the reaction rates for forming NO. Second, the diluent will slow the fuel burning rate, moving the average combustion later in the cycle where temperatures are lower due to the expansion cooling. (However, an engine with a knock sensor will not be affected in the same way.) Third, for a carbureted engine, an approximately fixed volume of air including humidity flows into the engine at a given throttle position. Water vapor displaces dry air, and therefore, increased humidity enrichens the (dry) A/F of the engine, which effects all engine emissions. Krause reported that a 100 [grain/lb] increase in humidity generally decreased the engine A/F by 0.4. The shift in A/F will have the most pronounced influence on NO_x formation if the engine is tuned to operate just lean of stoichiometric. Many light and heavy-duty vehicles operating today have tight A/F control to maintain the exhaust composition required for the aftertreatment process. Therefore, the NO_x emissions from these types of engines will not respond as strongly to ambient humidity changes as the carbureted engines did during the original testing of Manos and Krause. Figure 3 shows NO_x corrections based on Manos (Equation 1), Krause (Equation 2), and modeled results for both floating and fixed equivalence ratio.

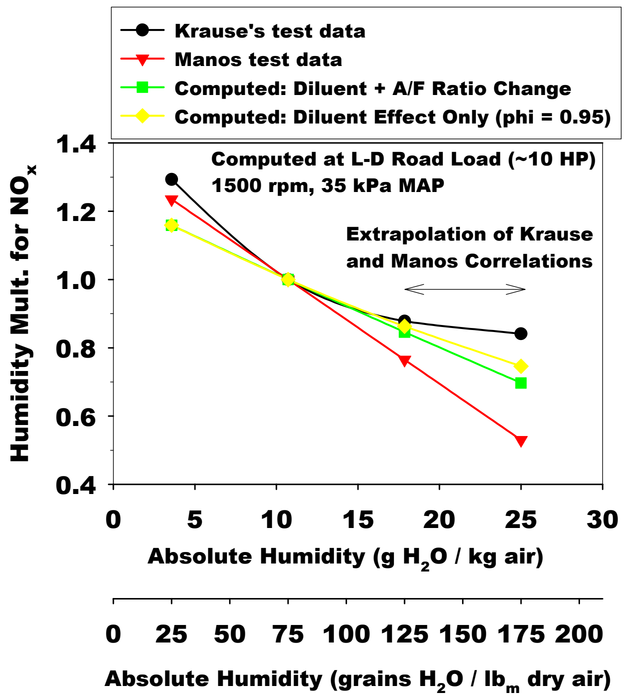

Figure 3. Humidity NO_X correction comparison

The modeled results were from the ALAMO_ENGINE code, which was developed at Southwest Research Institute. The details of how the ALAMO_ENGINE code computes power and NO_x emissions can be found in Appendix A. It should be noted that the humidity effects computed in ALAMO_ENGINE are based on solving chemical equilibria for the burning gases, computing the adiabatic flame temperature, and the kinetics for NO_x formation. Therefore, the computed results are not simply based on some humidity correction scheme, but rather on fundamental principles of the combustion process. The computed results are shown for two different cases. In the case labeled "diluent effect only," the A/F was fixed at 15.3 ($\Phi = 0.95$), and the humidity effect on the flame temperature through both the diluent effect and specific heat effect were accounted for. In the case labeled "diluent effect + A/F change," the A/F ratio was allowed to change from 15.5 at 25 grains/lb (3.6 g H_2O/kg dry air) to 15.3 at 75 grains/lb (10.7 g/kg), to 15.1 at 125 grains/lb (17.6 g/kg) to 14.9 at 175 grains/lb (25 g/kg). These changes in A/F were based on the test results reported by Krause. Therefore, both the change in A/F and the diluent effect were accounted for in this case. In all modern light-duty, on-road, gasoline engines, and most heavy-duty, on-road gasoline engines the A/F ratio remains fixed, independent of the humidity, since a sensor is used to maintain a fixed dry A/F.

As seen in the Figure 3, there is a significant discrepancy between the correction factors for humidity levels greater then 120 grains/lb. The second-order curvature of Krause's regression seems to under predict the slope of the correction factor at high humidity levels while the Manos equation may be over predicting the slope. Both the Manos et al, and Krause studies were conducted with engines that were subject to changes in A/F ratio with changes in humidity.

It should be noted that the ALAMO_ENGINE model results are for a single engine operating point (a light-duty, road-load of ~10 HP), while the data in the literature were for a combination of operating conditions. The sensitivity of the modeled NO_x to humidity was dependent on the engine operating conditions, with higher load conditions showing less sensitivity to humidity as shown in Figure 4. The model results also demonstrate that one would expect air-fuel ratio, or equivalence ratio, to effect the sensitivity of the NO_x emissions to humidity. If the engine is configured to operate rich of stoichiometric, the effect of humidity becomes less significant. Conversely, engines operating lean would be expected to demonstrate a larger sensitivity to humidity, as shown in Figure 5, where the modeling was performed at the Ford World-Wide Mapping Point (WWMP), a light load condition. The commonly used NO_x correction factors for ambient humidity do not account for these effects that could be important, depending on the application of the engine (light-duty or heavy-duty) and the engine fueling strategy (lean, stoichiometric, rich). The slope of the curves presented in Figures 4 and 5 represent the sensitivity of NO_x to ambient humidity. This slope is plotted versus the engine load (BMEP) and equivalence ratio in Figure 6.

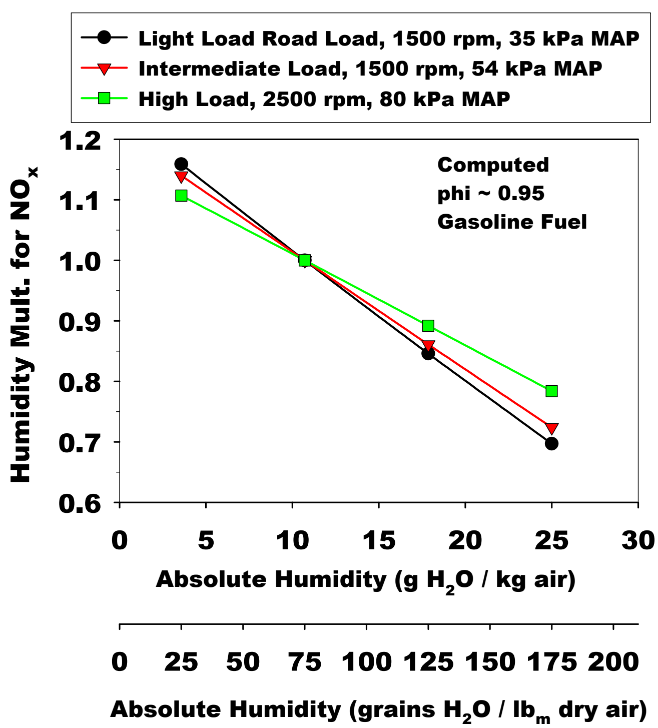

Figure 4. Effect of Load on Humidity Correction Factor

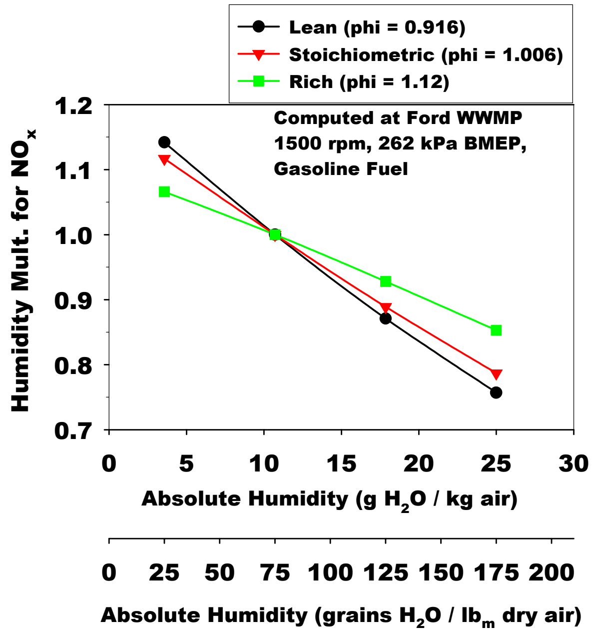

Figure 5. Effect of Equivalence Ratio on Humidity Correction Factor

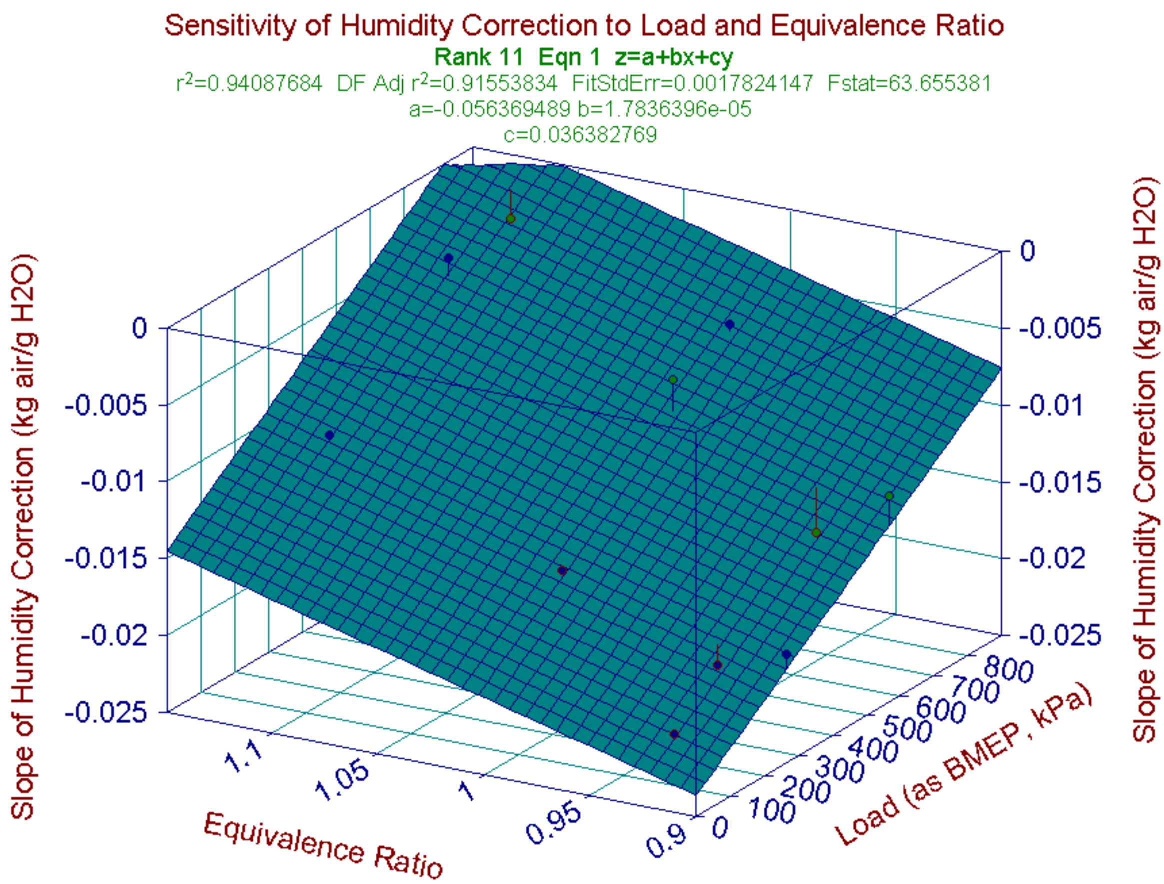

Figure 6. Humidity Correction as Function of Engine Load and Equivalence Ratio

All of the equations presented previously identify ambient conditions as the independent variable. However, it is the conditions present in the intake manifold that actually affect the NO_x formation. While the conditions in the intake manifold are coupled to the inlet conditions, the intake system will have an effect on the thermodynamic conditions presents in the intake manifold. The intake system can be conveniently considered in terms of two sections. The first section is from the ambient to the intake manifold, the second is from the manifold to the cylinder. The first section can be relatively simple, consisting of simply an air filter and a pipe or plenum, typical of a naturally aspirated engine, such as some on-highway and most small off-road engines. Alternatively, the first section can also be very complex, including the use of an exhaust driven turbocharger compressor and heat exchanger. These configurations can be found in large off-road stationary engines, and some on-highway applications. In all of these cases, the first section affects the temperature of the intake air as it passes through the various devices. Additional changes are likely as the air passes through the second section, where it is affected by heat transfer from the hotter surfaces of the intake port and intake valve.

Every different engine design incorporates features that can, and probably do have different effects on the thermodynamic conditions of the intake air as it enters the engine. In this sense, every engine could theoretically have its own correction factors. ALAMO_ENGINE was used to explore the effects of inlet humidity on NO_x formation for circumstances where there are no known standardized correction equations and to model emissions in the regime where other regression equations could only extrapolate. NO_x emissions were modeled with absolute humidity levels ranging from ~2.5 to 25 g/kg at different effective loads, and with multiple fuels. Figure 7 illustrates the humidity correction factor from ALAMO_ENGINE for propane and natural gas compared to gasoline at a light load under stoichiometric conditions. As indicated, under these conditions, propane and natural gas were slightly more sensitive to humidity than gasoline, although, recall that these are model results and not experimental data.

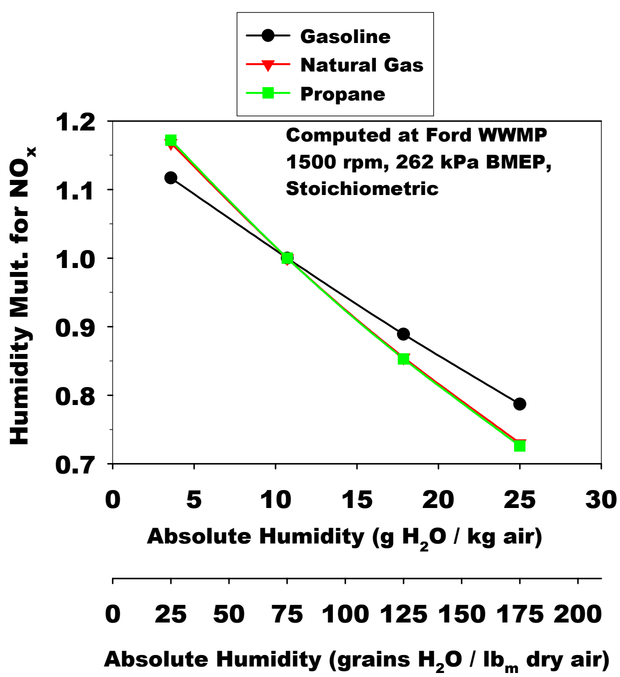

Figure 7. Fuel Effect on Humidity Correction

3.1.6. Effect of Air Conditioning Loads on Humidity and Temperature Correction Factors

Heavy-duty and off-road emissions tests are engine tests, not vehicle tests. Therefore, there is no allowance or concern for air-conditioner loads and their impact on emissions. Air conditioner compressors for passenger cars and trucks require about 1.2 kW (1.6 HP), a value which is likely negligible compared to the engine power required for heavy-duty engines used on a heavy-duty test cycle in vehicles large enough that they would use air conditioners. Air conditioners may have secondary impacts on the underhood temperatures, and this could impact NO_x emissions. These effects are too uncertain and too varied for any recommendations concerning humidity and temperature effects on NO_x emissions.

4.0 RECOMMENDED PRACTICES

4.1 Heavy-Duty On-Road and Off-Road Vehicles/Engines

The majority of gasoline heavy-duty on-highway engines and newer model off-highway heavy-duty engines operate with oxygen sensor feedback to control A/F just rich of stoichiometric to utilize the emission reduction benefits of three-way catalysts. Many also use knock sensors to tune the engine away from the detrimental effects of knock. An engine that is controlled to maintain a specific knock margin through EGR and ignition timing adjustments will also be reducing the humidity effect on burn rate. By reducing the effect of humidity on burn rate, the effect of humidity on NOx emissions will also be reduced. The current practices for adjusting NO_x emissions with intake air humidity do not incorporate any of these modern control practices into their theoretical methodology. Those practices that do correct based on technology classes seem to have extracted statistically irrelevant information from vehicle emission test databases. By modeling NO_x emission under a variety of conditions, SwRI was able to establish a theoretical revision of a standardized correction equation.

The approach taken here was to start with the test results and correlations developed by Manos et al.[2] and to make certain adjustments to the humidity correction factors suggested by them to account for the following effects:

1. Correct Manos et al.'s data from light-duty test cycles to heavy-duty test cycles.
2. For engines with closed-loop air-fuel ratio control, correct Manos et al.'s data taken on engines with variable air-fuel ratio to a fixed air-fuel ratio.
3. For engines with closed-loop air-fuel ratio control, correct Manos et al.'s data taken on engines with an average air-fuel ratio of 15.3 to a slightly rich of stoichiometric air-fuel ratio of 14.5.

For item 1, to establish a correction procedure for heavy-duty, on-road engines, and for off-road engines over 19 kW, which also operate at higher loads than light-duty applications, the correction procedure of Manos was selected as a baseline procedure. The correction developed by Manos was extrapolated to 175 grains/lb or 25 g/kg, and then the slope of the line was corrected based on theoretical modeling. Because the correction established by Manos was based on light-duty vehicles, the slope of the correction curve should be decreased by 15% due to the decrease in NO_x–humidity sensitivity at higher effective loads indicated by model results. Specifically, the percent change was calculated between the slope of the light-duty, road load in

Figure 4 and the average of the slopes for three loads, the light-duty road load, the Ford world wide mapping point (1500 rpm, 262 kPa BMEP), and the intermediate/high load point of 2500 rpm, 80 kPa intake manifold pressure (rough approximation of a HD cycle). Details are provided in Appendix B.

To account for item 2 above, engines with fixed air-fuel ratio control will not suffer the variable A/F ratio seen by Manos et al., and the resulting humidity effect on air-fuel ratio and on NOx. If the vehicle/engine class in question uses A/F control, the slope should be decreased another 10%. The derivation of this approximate correction factor is given in Appendix B.

To account for item 3 above, consider the following. The engines tested by Manos also operated at an A/F lean of stoichiometric, about 15.3. For engines operating near stoichiometric, 14.6, as typical for on-road, heavy-duty gasoline engines, the modeling study showed an approximately 9% reduction in the slope for the humidity correction factor. This may be seen qualitatively in Figure 5, and the quantitative analysis is given in Appendix B.

Therefore, the recommended equation to adjust standardized emissions for a carbureted heavy-duty on-road or off-road (above 19kW) engine under nonstandard inlet air conditions takes the following form:

$$C_{SwRI}(H) = 1 - 0.0280 \cdot (H - 10.71) \qquad (10)$$

Where:
$C_{SwRI} = NOx_{ambient}/NOx_{standard}$
H = Absolute humidity of the inlet air [g of H_2O/kg of dry air]

For heavy-duty on-road or off-road (above 19kW) engines that use 3-way catalysts (always with A/F control, and typically with port fuel injectors), the recommended NO_x correction equation is as follows:

$$C_{SwRI}(H) = 1 - 0.0232 \cdot (H - 10.71) \qquad (11)$$

Vehicles that utilize knock sensors in their engine control algorithms will further decrease their sensitivity to inlet humidity, though no data were found to quantify the magnitude.

Modeling results for heavy-duty engines running on either natural gas or propane show trends comparable to those seen for engines running on gasoline. Engine tests with natural gas fueled engines at SwRI have documented that humidity has an effect on NO_x emissions.[(14)] However, there is insufficient data on natural gas engines to specify a humidity correction factor for NO_x different from that for gasoline-fueled engines. Therefore, it is recommended that the same correction equations (Eq. 10 and 11) given above for gasoline engines should be used for propane and natural gas engines.

For heavy-duty engines that use carburetors and no aftertreament, Manos et al. measured a temperature effect on NO_x emissions. The ambient temperature affects the density of the air flowing through the carburetor, resulting in a shift in air-fuel ratio. The humidity effect on carbureted engines was quantified by Manos in Equation 9. Recall that Equation 9 would be used to correct to standard conditions and the inverse would be required to predict the effect of non-standard conditions. Using Equation 9 as the basis, the temperature correction term would then become:

$$C_{temp} = 0.0022 \cdot (T - 25) \tag{12}$$

Where:
T = Temperature of the inlet air [°C]

Applying this term to the humidity correction equation for carbureted engines (Equation 10) results in the following equation for NO_x correction that includes both temperature and humidity:

$$C_{SwRI}(H,T) = 1 + 0.0022\,(T - 25) - 0.0280 \cdot (H - 10.71) \tag{13}$$

Where:
T = Temperature of the inlet air [°C]
H =Absolute humidity of the inlet air [g of H_2O/kg of dry air]

For engines that have aftertreatment and accurate control of air-fuel ratio, the ambient temperature effect will be limited to effects on the charge air temperature, which is predominately determined by the manifold wall temperatures. Since this effect is largely unknown, it is recommended that no temperature correction be used for these engines.

To more accurately determine the correction factors for temperature and humidity, further testing is recommended to provide empirical data to clarify the apparent issues that this work could only address through theoretical modeling.

4.2 Light-Duty Vehicles

Unlike heavy-duty vehicles/engines the EPA corrects for the humidity effect on NO_x formation for light-duty, on-highway emissions in MOBILE6. For current modeling purposes, the recommended practice is Equation 4.

$$C = NOx_{corr\_MOBILE}(H_a) = \begin{cases} 1.2 & \text{if } H_a \le 20 \\ (-0.004 \cdot H_a + 1.28) & \text{if } 20 < H_a < 120 \\ 0.8 & \text{if } H_a \ge 120 \end{cases} \tag{4}$$

Where:
H_a = Absolute humidity of the inlet air [grains/lb]

This equation (in the region greater than 120 grain/lb and less than 20 grains/lb) should be modified based on empirical data as soon as valid testing is performed. The testing should also involve the technology classes present in the current inventories, to identify actual technology-based dependencies.

4.3 Small Off-Road Engines

For small off-road engines the recommended practice is Equation 8 for correcting observed NO_x emissions to a standard condition. To approximate a standard engine emission rate at non-standard conditions, the inverse of Equation 8 should be used. While the operating A/F may be difficult to estimate for the lawn and garden style engine inventory, test-cycle averages can be assumed. When the test cycle averages are not known, an estimate for the operating A/F of this class of gasoline engine is 12.0. For two-stroke engines, the NO_x correction factor should be set to 1.0 since no statistically significant dependence has been shown. The NO_x correction factor to estimate NO_x emissions at a non-standard condition from emissions data taken at standard conditions would then be,

$$C = 1 - \frac{546}{AFR} \cdot (\omega - 0.01071) \qquad (14)$$

Where:
AFR = Air-fuel ratio of the engine
ω = Absolute humidity of the inlet air [kg/kg]

5.0 SUMMARY

All of the current humidity correction factors for NO_X were found to be based on historical data taken in 1971 and 1972. Some of the engines today are more technically advanced, incorporating port or throttle-body fuel injection, air-fuel ratio feedback, exhaust aftertreatment, and knock detection. While many off-road vehicles do not have all of these features, this technology is becoming more prevalent. The analysis conducted indicated that the historical correction factors do not adequately account for operating cycles with higher load factors, or advanced technologies such as A/F control and knock detection. No engine test data were found documenting humidity effects for these additional variables.

The model results showed these effects to be significant and the results were used to modify the historical correction procedures. If a more rigorous approach is desired, SwRI would recommend engine testing to quantify the effects for different engine/vehicle classes.

The recommended equation to adjust standardized emissions for a **carbureted heavy-duty on-road or off-road (above 19kW) engine** under non-standard inlet air conditions takes the following form:

$$C_{SwRI}(H,T) = 1 + 0.0022\cdot(T - 25) - 0.0280\cdot(H - 10.71) \qquad (13)$$

Where:
T = Temperature of the inlet air [^oC]
H =Absolute humidity of the inlet air [g of H_2O/kg of dry air]

For **heavy-duty on-road or off-road (above 19kW) spark-ignition engines** that use a 3-way catalyst (A/F control, typically with port fuel injectors), the recommended NO_x correction equation is as follows:

$$C_{SwRI}(H) = 1 - 0.0232\cdot(H - 10.71) \qquad (11)$$

with no correction for ambient temperature.

For **light-duty, spark-ignition engines**, the recommended practice is whatever procedure is used in Mobile 6, which can be approximated by Equation 4.

$$C = NOx_{corr\_MOBILE}(H_a) = \begin{vmatrix} 1.2 \ \text{ if } H_a \le 20 \\ (-0.004\cdot H_a + 1.28) \ \text{ if } 20 < H_a < 120 \\ 0.8 \ \text{ if } H_a \ge 120 \end{vmatrix} \qquad (4)$$

Where:
H_a = Absolute humidity of the inlet air [grains/lb]

For **small off-road, spark-ignition engines (< 19kW)**, the recommended practice is,

$$C = 1 - \frac{546}{AFR}\cdot(\omega - 0.01071) \qquad (14)$$

Where:
AFR = Air-fuel ratio of the engine
ω = Absolute humidity of the inlet air [kg/kg]

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APPENDIX A

ALAMO_ENGINE COMPUTER MODEL

ALAMO_ENGINE COMPUTER MODEL

An overall flow chart of the ALAMO_ENGINE computer model is shown in Figure A1. The model consists of three parts, a cycle simulation, a calculation of adiabatic flame temperatures and chemical species, and a submodel for computing NO_x emissions. Each of these submodels is briefly described below. A more complete description of the overall model as applied to diesel engine NO_x predictions was given previously by Dodge, et al. [A1].

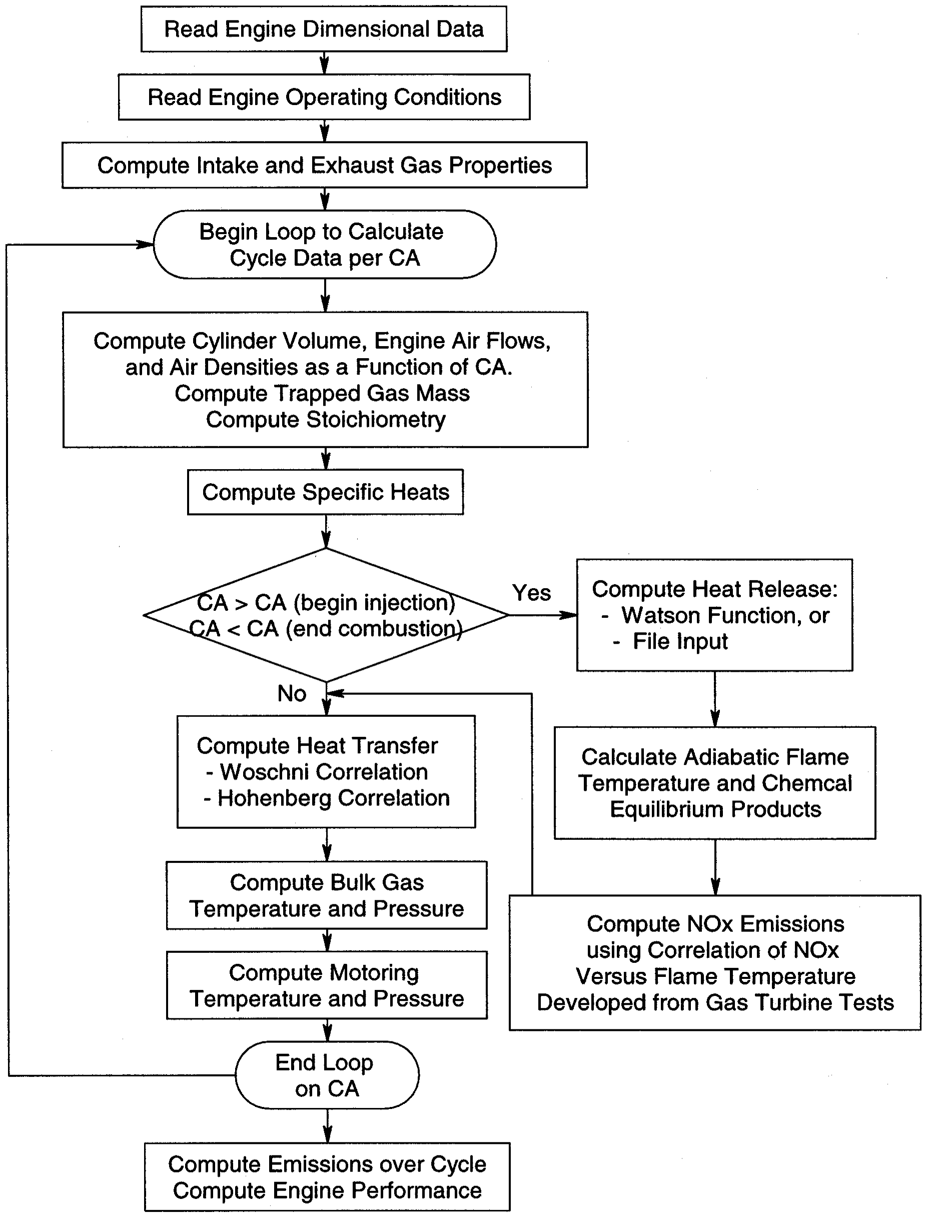

Figure A1. ALAMO_ENGINE Computer Model Flow Chart

CYCLE SIMULATION SUBMODEL

The cycle simulation portion of the model is fairly conventional with several added features for ease of use and to make it particularly suitable for studying combustion effects on NO_x. About 25 engines are stored in a database of engines that can be selected by the user. This database contains all the information required by the program to operate. Provisions are made in the program so that if some information about the engine is unknown, typical values for that size and type of engine are used. Heat release information can be estimated using a Wiebe function for spark-ignited engines or a modified Watson function [A1-A2] for compression ignition engines. Different Wiebe functions are selected for gasoline and natural gas engines to reflect the slower flame speeds of natural gas engines. If the apparent heat release rates have been measured, they may be used rather than the Wiebe or Watson correlations. Detailed gas compositions are computed for the unburned and burned gases based on equations given by Heywood [A3], so that the proper specific heats and other gas properties can be accounted for. The equations by Heywood were expanded to include water vapor from in-cylinder water injection and from emulsified fuels. This allows the program to evaluate the effect of EGR gases, residual gases, humidity, and water injection on NO_x emissions and power. Residual gas concentrations are calculated based on the work of Fox et al. [A4], as modified by Senecal et al. [A5] for turbocharged engines or by direct calculation of residual concentrations from the valve flow model. Residual gas temperatures are computed by iterating through the cycle simulation. The air flow through the engine is computed using a valve flow model. Heat transfer may be estimated from correlations developed by Woschni [A6] or Hohenberg [A7], or it may be turned off using the menu-based input. The Woschni [A6] heat transfer model was used for all calculations shown here, except as noted otherwise. The choice of heat transfer models can have a significant effect on NO_x and power predictions.

UNBURNED AND BURNED GAS TEMPERATURES SUBMODEL AND CHEMICAL EQUILIBRIUM SUBMODEL

The model assumes two in-cylinder gas zones, one for the unburned gas and one for the burned gas. (Shortcomings of this approach for predicting NO_x have been discussed by Raine et al. [A8].). In this approach, the burned gases are assumed to be fully mixed. The approach follows that of Heywood [A9]. The chemical equilibrium submodel was necessary to compute the equilibrium NO level as required by EQ. (6) below. The chemical equilibrium code was described previously by Dodge et al.[A1].

NITRIC OXIDES EMISSIONS MODEL

The NO_x emissions are calculated as NO based on the extended Zeldovich mechanism and a correlation for the Fenimore prompt NO mechanism. The results were summed together without consideration of further interaction. Even though the formation was assumed to be in the form of NO, conversions between NO_x concentrations in ppm and emissions rates expressed as g/HP-hr used the molecular weight for NO_2, in agreement with guidelines given by the U.S. Environmental Protection Agency. The well-known extended Zeldovich mechanism is given by [A10,A11]:

$$O + N_2 <=> NO + N \tag{1}$$

$$N + O_2 <=> NO + O \tag{2}$$

$$N + OH <=> NO + H \tag{3}$$

For the calculations reported here, the rate constants given by Heywood [A12] were used for the extended Zeldovich reactions. A correlation representing the Fenimore prompt NO mechanism is also used. [A13]

$$CH + N_2 <=> HCN + N \tag{4}$$

$$C_2 + N_2 <=> 2\ CN \tag{5}$$

The incorporation of the prompt NO mechanism into the model would have been difficult because of the requirement to add the hydrocarbon chemistry. However, Moore [A14] showed that the prompt NO correlates as a function of the equivalence ratio Φ, and the equilibrium nitric oxide concentration corresponding to the adiabatic flame temperature, $NO_{\text{equil. T adiab}}$

$$NO_{prompt} = f(\Phi)\mathrm{P}^{1/2} NO_{\text{equil, T adiab}} \tag{6}$$

where P is the pressure in atmospheres, and Corr. et al. {A15] curve fit the data of Fenimore to get $f(\Phi)$ as,

$$f(\Phi) = 0.0053 \exp\left(1.004\ \Phi^{4.865}\right) \tag{7}$$

over a range of equivalence ratios from $0.8 < \Phi < 1.2$. Corr et al. [A15] extrapolated this function to Φ=0.6 and got reasonable agreement with their data taken at that equivalence ratio. Both Corr et al. and Fenimore suggested that $f(\Phi)$ be multiplied by 0.75 in the case of methane combustion.

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APPENDIX B

DERIVATION OF CORRECTION EQUATIONS

DERIVATION OF CORRECTION EQUATIONS

This appendix shows the derivation of the amounts of correction required to apply Manos et al.'s experimental data and corrections to heavy-duty spark-ignition engines. Three potential corrections are required:

4. Correct Manos et al.'s data from light-duty test cycles to heavy-duty test cycles.
5. For engines with closed-loop air-fuel ratio control, correct Manos et al.'s data taken on engines with variable air-fuel ratio to a fixed air-fuel ratio.
6. For engines with closed-loop air-fuel ratio control, correct Manos et al.'s data taken on engines with an average air-fuel ratio of 15.3 to a slightly rich of stoichiometric air-fuel ratio of 14.5.

These corrections are approximate, and there are no test data that were found for comparison.

1. Correct Manos et al.'s data from light-duty test cycles to heavy-duty test cycles.
Consider the modeling results from three load conditions: light load, intermediate load, and high load. These data are presented in Figure 4 of the report. The corresponding slopes of the lines are shown in the table below. Application of the light duty correction equations to heavy-duty engines must account for the sensitivity of the humidity effect on load. To accomplish this, the slope of the humidity correction for light load compared to the average for the three cases. That is, the light-load data were assumed to correspond to the test conditions of Manos et al., while the average of the light, intermediate, and high load data were assumed to represent a heavy-duty test cycle. The term that resulted from the ratio was then used to adjust the humidity correction term from Equation 1 of the report.

Load Condition	Slope kg dry air/ g of H_2O
Light Load Road Load	-0.02159
Intermediate Load	-0.0194
High Load	-0.01506
Average	-0.01868
Light Load Road Load/Average	1.15

$$\frac{1}{KH_{SI}} = 1 - 0.0329(H - 10.71) \qquad (1)$$

$$C_{SwRI}(H) = 1 - 0.0329 \cdot \frac{(-0.0187)}{(-0.0216)}(H - 10.71)$$

or

$$C_{SwRI}(H) = 1 - 0.0285(H - 10.71) \qquad (10)$$

2. Correct Manos et al.'s data taken on engines with variable air-fuel ratio to a fixed air-fuel ratio (only applicable to engines with closed-loop air-fuel ratio control).

Equation 1 of the report was also derived from data for engines without air-fuel ratio control. Thus part of the correction accounted for variations in air-fuel ratio as humidity changed. For engines with air-fuel ratio control, the magnitude of this correction would be too large. To adjust for this difference, air-fuel ratio effects were modeled at an intermediate-load condition (the Ford world-wide mapping point of 1500 rpm, 262 kPa BMEP). These calculations were carried out simultaneous changes in humidity and corresponding changes in air-fuel ratio as observed by Krause[3] (0.4 A/F units per 100 grains of humidity change), and compared with changes in humidity with a fixed A/F ratio as observed by Krause of 15.3. For the engine with simultaneous changes in humidity and air-fuel ratio, the slope of the humidity correction was computed to be -0.02159, while for the engine with changes in humidity at constant air/fuel, the slope of the humidity correction was -0.01935. The ratio of these slopes is 0.896, or about a 10% reduction in humidity dependence if the air-fuel ratio is constant as compared to simultaneous changes in humidity and air-fuel ratio.

3. Correct Manos et al.'s data taken on engines with an average air-fuel ratio of 15.3 to a slightly rich of stoichiometric air-fuel ratio of 14.5 (only applicable to engines with closed-loop air-fuel ratio control).

These data are presented in Figure 5 of the report. The slopes of the line represented in Figure 5 for the lean (phi = 0.916) and stoichiometric (phi = 1.006) conditions are –0.0180 and –0.0154, respectively. This represents a 16.7% difference in slopes. The data used in the derivation of Equation 1 from Manos et al. were acquired for an equivalence ratio of approximately 0.95 (air/fuel = 15.3). In order to adjust the humidity coefficient in Equation 10, an interpolation of the humidity effect was used to determine the slope of a line at a 0.95 equivalence ratio. The slope was then used to adjust the humidity coefficient in equation 10 for a stoichiometric equivalence ratio.

$$slope_{0.95} = slope_{0.916} + (slope_{1.006} - slope_{0.916}) * \frac{(0.95 - 0.916)}{(1.006 - 0.916)}$$

$$slope_{0.95} = -0.0180 + (-0.0154 - (-0.0180)) * \frac{(0.95 - 0.916)}{(1.006 - 0.916)}$$

$$slope_{0.95} = -0.0170$$

$$\frac{slope_{1.006}}{slope_{0.95}} = 0.906$$

For engines with three-way catalysts, fixed air-fuel ratio control is used, and the engines are controlled to an equivalence ratio just rich of stoichiometric. Therefore, compared to Manos et al.'s humidity correction factor which was corrected in item 1 above for a heavy-duty cycle to a slope of -0.0285, the slope must be further reduced by the factors in items 2 and 3, or (–0.0285) (0.896) (0.906) = –0.0232. Therefore, for engines with fixed, stoichiometric air-fuel ratios, the humidity correction factor becomes,

$$C_{SwRI}(H) = 1 - 0.0232 \cdot (H - 10.71)$$