Investigations of the Effect of Humid Air on NO_X & PM Emissions of a CNG Engine

FINAL REPORT

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Abstract

Numerical and experimental investigations of the effect of a humid air system on nitrogen oxides (NOx) and Particulate Matter (PM) emissions of a compressed natural gas (CNG) engine have been performed. For the numerical modeling, non-premixed combustion in a single cylinder was simulated using the presumed probability density function combustion model. Simulations were performed for dry as well as humid air intake, ranging from 5% to 30% relative humidity (RH). Numerical results have shown 40% reduction in NO emission at 10% relative humidity, when compared with the corresponding emission with dry intake air. With 15% and 30% relative humidity levels, NO emission were reduced by 65% and 93% respectively.

For experimental investigations, a General Motors inline 4 cylinders, naturally aspirated engine with a maximum rated horsepower (HP) of 50.8 for natural gas fuel was used. The engine was connected to a water-cycled dynamometer. NOx emission was measured by a Horiba portable emission analyzer model 250 and exhaust PM was measured using a dilution tunnel in conjunction with a cyclone with teflo filters. The experiments were carried out at four different horse powers (HP) of approximately 5, 12.5, 25. And 37.5, and three RHs of ambient (30%), 45%, and 60%. Results show for each additional 15% increase in relative humidity, there was approximately 10% reduction in NOx emission, which is nearly consistent with the corresponding drop rate between 15% and 30% relative humidity from the numerical simulations. The PM emission increases with the addition of relative humidity, especially at low HPs. With increased HP, the PM augmentation is reduced significantly and at 37.5 HP , the ratios of PM emitted at 45% and 60% relative humidity, to the corresponding ambient value (30% RH), were near 2.0. When normalized by the corresponding HP, the PM emission decreased with increase in engine loads at both 45% and 60% RHs. For 45% RH, the PM weight (mg) per HP decreased from 0.891 at 5 HP to 0.073 at 37.5 HP and the corresponding values for 60% RH were 1.267 and 0.077 respectively. Our results indicate that humid air system is an effective approach for reducing NOx emission of CNG engines, without significant increase in PM emission which could make them near net zero engines.

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Disclosure

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1.0. BACKGROUND

About 29% of greenhouse gas (GHG) emissions in the U.S. is produced by the transportation sector. The major GHGs are carbon dioxide (CO_2), nitrous oxide (N_2O), methane (CH_4), and Hydrofluorocarbons (HFC). According to the intergovernmental panel on climate change (IPCC) [1], if no additional measures are taken to reduce the GHG emissions, between years 2000 to 2030, the human source GHG emissions will increase 25% to 90% with CO_2 emissions growing between 40% to 110%. The corresponding global temperature rise will be between 2ºF to 11.5ºF by 2100 with 3-4 feet sea level rise. To limit the global warming to a range of 3.6ºF(2ºC) to 4.3ºF(2.4ºC), the GHG emissions must be reduced 50% to 85% below year 2000 by 2050. To meet this target, multi-disciplinary efforts must be undertaken with transportation playing a major role to limit GHG emissions.

Among strategies for reducing transportation GHG emissions are introduction of low carbon fuels, improving vehicle fuel economy and transportation system efficiency, and reducing carbon-intensive travel activities. Natural gas is a low carbon fuel. Assembly Bill 1007 (AB 1007) required the California Energy Commission (CEC) and the California Air Resources Board (CARB) to work together and in consultation with the State Water Resources Control Board, the Department of Food and Agriculture, and other relevant agencies, for development and adaptation of a state plan to increase the use of alternative transportation fuels in California. A recent report prepared for CEC on full fuel cycle assessment, well to wheels energy inputs, emissions and water impacts [2], provides an assessment and impacts of different fuels that include gasoline and diesel, ethanol, biodiesel and renewable diesel, natural gas, electricity, and hydrogen, among others. The report indicates that with adaptations of CNG vehicles, GHG reduction for passenger cars is 20% to 30% and for heavy duty vehicles is 11% to 23%. Liquid natural gas (LNG) is an alternative fuel for heavy duty vehicles. With LNG vehicles, the GHG reduction for heavy duty vehicles would be between 12% to 16% by the year 2022.

The goal of the proposed investigation was to determine the feasibility of using a humid air system for reducing NOx emissions of CNG engines. Humid air system or fumigation has been an effective approach in reducing diesel NOx emissions. In this method, water vapor is injected in the intake air supplied to the engine cylinders. The process reduces the local temperature in the cylinder and raises the specific heat of the air-fuel mixture which also contributes to the elimination of the hot spots in the engine's cylinders. With decreased temperature, NOx reduction is achieved. With an optimized system, fumigation could reduce NOx emission without significant increases in hydrocarbon emissions. Other benefits of this process include longer life of the engine components due to reduced cycle temperature and reductions in carbon deposits. Such a system has been tested on ocean going vessels (OGVs). According to *Richie et al* [3], the humid air motors system has been tested on one ship, the MS Mariella in Europe with positive results. However, since the system required distilled water, and for these large ships, an additional system needs to be developed for converting seawater to distilled water, the initial investment cost for incorporating such a system has prevented its wide application in OGVs.

Takasaki *et al* [4] studied the application of a stratified fuel water injection system (SFWI) and a direct water injection (DWI) system on the NOx and PM emissions of a heavy-duty diesel engine. They found 70% reduction in NOx emission with 60% water injection and 2% improvement in the brake specific fuel consumption(BSFC). For the DWI, they found the NOx emission was reduced to half with 60% water injection, without any improvement in the BSFC.

Radloff and Gautier [5] have provided results of a field testing of a charged air water injection system on diesel NOx emission of an OGV. The emission measurements were performed according to ISO 8178-4-E3 test protocol with marine diesel oil and the intermediate fuel oil. Their results showed NOx reduction between 10% and 35% with the upper end reduction corresponding to engine loads higher than 50%. For the intermediate fuel oil, the NOx reduction was accompanied with increases in the particulate matter and carbon monoxide.

Park *et al* [6] studied the effect of steam and hot water injected simultaneously into the combustion chamber of a low speed diesel engine. The role of the steam was to increase the internal energy of the heated water, to form vapor. Their results indicate 10%-38% reduction in NOx for humidity ratios of 27-59 grams water, per kilogram of dry air, with a slight increase in fuel consumption at less than 50% of the engine load. There were significant increases in carbon monoxide for 25% and 50% of the loads for up to 30 g/Kg humidity ratio. The carbon monoxide emission was reduced with higher humidity ratios.

Zhao *et al* [7] performed numerical simulations of a coaxial jet diffusion flame and a counter-flow diffusion flame, formed using methane gas and air. The effectiveness of the addition of the steam to either air or the fuel injection side was investigated. Their results showed for both cases, adding steam to the air side was more effective in reducing the flame temperature than the fuel side.

Rahai *et al* [8,9] and Farahani et al [10] have investigated the effects of humid air on the performance of a naturally-aspired three cylinder diesel engine with a low sulfur diesel fuel. The addition of the humidity to the intake air was performed with a variable steam generator using distilled water, where the relative humidity levels of the intake air were changed from the ambient conditions of 65% to 75% and 95% levels. The tests were performed at two approximate engine output break horse powers (BHP) of 5.9, and 8.9. Results showed approximately 3.7% and 22.5% reduction in NOx emissions when the relative humidity of the air was increased from 65% (the ambient relative humidity) to 75% and 95% respectively. The addition of the humidity results in increases in the CO, CO2,and particulate matter (PM), by approximately 3.7,3.55, and 14.9 percent at 5.9 BHP and 22, 2.8, and 9.3 percent at 8.9 BHP. There was no change in the brake specific fuel consumption (BSFC) at 5.9 BHP and about 2.7 percent increase in the BSFC at 8.9 BHP. Their results indicate that for both mobile and stationary diesel engines, the humid air system is a viable option for attaining significant reduction in NOx emissions.

2.0. METHODOLOGY

The study was divided into two parts. In part one, numerical investigations of the effect of humid air at different levels of relative humidity on NO, CO, and CO_2 emissions of a non-premixed combustion of air and methane were performed. The study was performed using the existing combustion model of the Star CCM+ software by CD Adapco. The model solves the transport equation for the concentration of NO_X and is available for non- premixed and partially pre-mixed combustion. The outputs are fuel NO_X and thermal NO_X. In this study, the focus was on the thermal NO_X, since it consists a significant portion of the overall NO_X produced in CNG engines.

Four basic combustion models are currently available in STAR-CCM+ as follow:

Presumed Probability Density Function (PPDF) models
Eddy break-up (EDU) models
The Homogeneous Reactor model
The Coherent Flame model (CFM)

The choice of combustion model is decided by knowing the Damkohler number defined as:

$$D_a = \frac{t_{mix}}{t_{rxn}}$$

Where t_{mix} is the mixing time scale and t_{rxn} is the reaction time scale. When the Damkohler number is large, the turbulent mixing that brings reactants together at the molecular scale controls the reaction rate. In this limit, the standard EBU and the equilibrium PPDF models are fairly accurate because they assume the reaction occurs instantaneously upon micro mixing. In order to solve a non-premixed problem we can make the assumption that Damkohler number is fairly large because more time is needed for the fuel and air to mix than them to react.

The PPDF model parameterization depends on whether the flow is assumed to be locally adiabatic or non-adiabatic. Under the adiabatic assumption, only the scalar mixing process and the initial temperature of the control volume affect chemical equilibrium within a control volume. Since heat losses or gains are not desired as controlling factors in this simulation, The PPDF adiabatic model was used.

This simulation involves a turbulent, compressible and multi-component gas flow. Combustion modeling was done based on the CO and NO_X emissions. This model requires each gas component to be added by the user. CNG mostly consists of methane. Hence, other components were neglected for the purpose of avoiding more complexity and the fuel was considered to be 100% methane. The global reactions of methane are:

$$CH_4 + {}^1/_2\, O_2 \rightarrow CO + 2H_2$$
$$CO + {}^1/_2\, O_2 \rightarrow CO_2$$
$$H_2 + {}^1/_2\, O_2 \rightarrow H_2O$$

with the extended Zeldovich mechanism defined as:

$$O + N_2 \rightarrow N + NO$$
$$N + O_2 \rightarrow O + NO$$
$$N + OH \rightarrow H + NO$$

CH_4, CO, CO_2, H_2O, O_2, N_2, and NO were chosen to be the multi gas component for the calculation of CO and NO.

After selecting the gas components of the fuel and oxidizer stream, their temperatures and mass fractions were defined. For all runs the fuel stream was defined as Methane gas (CH4) with temperature set at 330K and mass fraction of 1. In the first run (with no humid added) the oxidizer stream consisted of O2 and N2 with temperature of 330K and 0.233 and 0.767 mass fractions respectively. For the second run, 5% RH, $H2O$ was added to the oxidizer stream with 0.22135 and 0.72865 mass fractions of O2 and N2 respectively. For the 3rd run, 10 % RH, water vapor was added with 0.2097 and 0.6903 mass fraction of O2 and N2. Similar process was followed as the RH was increased.

The segregated numerical model along with the standard K-ε turbulence model was used in all simulations. The segregated flow model solves the fluid flow equations for each component of velocity and for pressure in a segregated uncoupled manner for attaining higher computational efficiency.

The initial pressure and temperature of 1 bar and 330 K were assigned for all cases. The boundary type for exhaust was considered to be atmospheric pressure. Turbulence intensity and viscosity ratio values were set to 1% and 10 respectively.

The computational domain was tubular with five holes at the inlet. Figure 1. Shows the tubular combustor. The central hole was the fuel line with 4 holes of oxidant (air) equally spaced circumferentially around the central hole, at 38 mm from the center hole. The center hole diameter was 25 mm and the surrounding holes were 12.5 mm diameter. The surrounding holes were at 60 degree angle with respect to the streamline, impinging air on the fuel line, to enhance mixing and thus combustion. For the fuel inlet, velocity was set to 0.2 m/s which was calculated based on the geometry and desired mass flow rate. Based on the intake geometry and 14.7:1 air fuel ratio the velocity for each of air inlets was set to 3 m/s.

Figure 1. The numerical combustor.

The 288 core high performance computing (HPC) facility of the CSULB College of Engineering (COE) along with the implicit unsteady CFD solver were used for all numerical investigations. For this solver, smaller time steps lead to more continuous results over time and a more accurate solution. For this purpose the time step was set at 0.01 second. Number of inner iterations for each time step was selected to be 50 which was enough to observe the convergence at each time step. Figure 2 shows the fluid temperature during the iterations 200 to 1000, which illustrates convergence for each inner iteration.

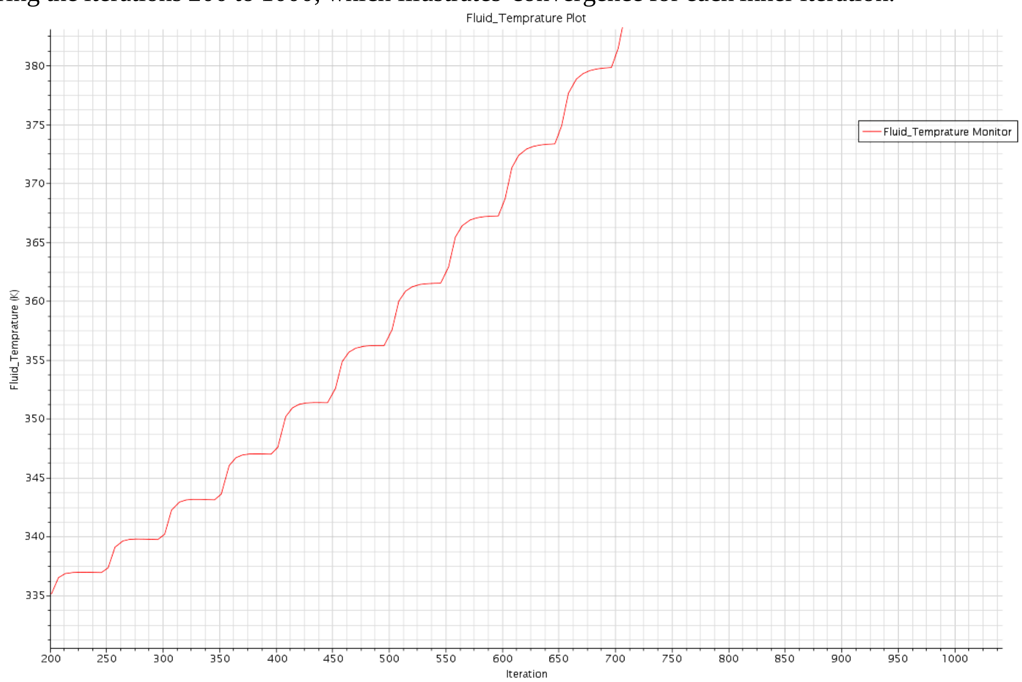


Figure 2. Time-step convergence in numerical simulations.

For each case, the simulation was carried out for 6 seconds for a total of 30,000 iterations. Grid dependency was performed with the case of no humidity added, starting at over 2.6 million grid nodes and then doubling the number of grid points each time, and monitor NO and temperature variation. Less than

1% variation was found when the number of grids exceeded the 2.6 million grid nodes, and thus for all simulations, this number of grid nodes was used.

For experimental investigations, a General Motors inline 4 cylinders, naturally aspirated engine with a maximum rated horsepower (HP) of 50.8 for natural gas fuel was used. The engine was connected to a water-cycled dynamometer from Land & Sea which is equipped with automated data acquisition for engine performance tests. Figure 3 shows the experimental set-up. A special mixing tube was designed to add humidity to the intake air. A Rasco Vapour machine with distilled water was used to generate the added fog to the intake air. The humidity level of the intake air before and after adding humidity was measured with two TSI VelocCalc model 9565-P anemometers.

The experiments were performed at three levels of humidities and four engine rated horse powers (HP) of 5, 12.5, 25, and 37.5. The first level of humidity was the ambient humidity and the subsequent higher level of humidity was obtaind by increasing the percent humidity by 15% each time.

NOx emission was measured by a Horiba portable emission analyzer model 250. It uses non-dispersive IR detection for CO, SO_2, and CO_2; chemiluminescence (cross-flow modulation) for NO_x; and a galvanic cell or an optional zirconium oxide sensor for O_2 measurements.

The exhaust PM measurements were involved using a dilution tunnel connected to a cyclone with Teflo filter. The raw exhaust gas was transferred via the sampling Pitot tube to the dilution tunnel via a heated transfer tube. The dilution tunnel was also supplied with filtered dry air equipped with temperature, pressure, and flow control sensors. Two stainless steel tubes of 0.635 cm ID were used for sampling the diluted flow downstream of the venturi. One tube is connected to the Horiba PG-250, and another to the cyclone.

Figure 3. The experimental set-up.

The raw exhaust gas flowed due to the negative pressure created by the venture (a component of the dilution tunnel). Tracer of the CO_2 gas was measured in the raw exhaust gas, incoming dilution air, and diluted exhaust gas for calculation of the dilution ratio. For PM measurements, 47 mm Teflo filters were used. The Teflo filters were conditioned (dried) in a uniform temperature at 72F inside a class 10,000 clean room for at least 24 hours both prior to and after the experiments, before weight measurements. Weight measurements were performed, using a Mettler-Toledo MT5 analytical microbalance.

4.0. RESULTS AND DISCUSSIONS

Part I: Numerical Results

Figures 4-6 show axial variation of mean temperature along the tube mid-section for three cases of dry air (no humidity), 5% and 10% relative humidity (RH) at 0.5 sec., 1.5 sec., and 2.5 sec. Figures 7 and 8 show time variations of the exhaust mean temperature and percent mass fraction of NO, CO, and CO_2 gases. At 0.5 sec., the contours of the mean temperature indicate reduced temperature gradient at 10% RH and the maximum temperature has been reduced with the addition of the humidity. With dry air, the maximum temperature is at 2253 K and with the addition of 5% RH and 10% RH, the maximum temperatures are 2139K and 2024 K respectively. At 1.5 sec., with the addition of humid air, regions of high temperature are expanded, but the temperature gradient remains relatively the same. As in 0.5 sec., the maximum temperature is reduced with the addition of humid air. At 2.5 sec., with the addition of humidity, there are significant reductions in the mean temperature and temperature gradient, which indicate the effectiveness of the humidity in reducing the combustion temperature and hot spots and more uniform combustion.

Exhaust temperature for all three cases approach constant values, beyond 2.5 sec. with the maximum temperature is reduced by more than 9% and 17% with 5% RH and 10% RH respectively.

Variation of percent mass fraction for NO and CO_2 exhaust gases show significant reductions with the addition of the humid air. However, the CO emission shows significant increases with increased humidity. Table 1 shows percent change in exhaust gases and the total exhaust emissions. Here, the total emissions were calculated from the area under the curve. With the addition of 5% RH, NO and CO_2 are decreased by more than 53% and 28% respectively, while CO emission is increased by more than 30%. The total emission which is the summation of these numbers indicates a reduction of more than 18%.

With 10% RH, the reductions in NO and CO_2 emissions are at more than 72% and 56% respectively and the increase in CO emission is at nearly 49%. The total emission reduction is more than 38%. The results indicate that the addition of the humidity results in significantly more reduction in total exhaust emission than increases in particular gas emission and thus in total, not only there is a significant reduction in NO emission, but also in total gas emission.

Table 2 shows the rate of change in emissions for all three cases investigated. It shows with the addition of humidity, the rate of reduction in NO is higher than the rate of increase in CO emission with the rate of change in CO_2 remains negative, but less at 10% RH as compared with 5% RH. The rate of decrease in total emission is slightly higher at 10% RH.

One of the major parameter in deciding the optimized factor for maximum NOx and in this case also total emission reduction is the humidity to fuel mass ratio. Previous investigations on the impact of fumigation or humid air system on diesel NOx emission has shown an optimum ratio of 3:1. For 5% RH, the ratio is 0.735 which increases to 1.47 for 10%RH. We also have investigated the effect of 15% and 30% RH on the exhaust emissions and results have indicated higher than 85% in NO reduction. For 15% RH, the humidity/fuel ratio was 2.20 and for 30%RH it was 4.4, much higher than 3:1 ratio.

Based on past experiences, the uncoupled numerical investigations with standard K-ε turbulence model significantly over estimated the output and it was expected that with the engine experiment, a more realistic results could be obtained. Here we just use the numerical investigation to understand the relationship between increased %RH and reduction in NO emission. Also, while the numerical investigations indicate significant increase in CO emission with increased humidity, however the over-estimation could also applies here and we did not expect in a realistic setting, for the CO emission to increase significantly.

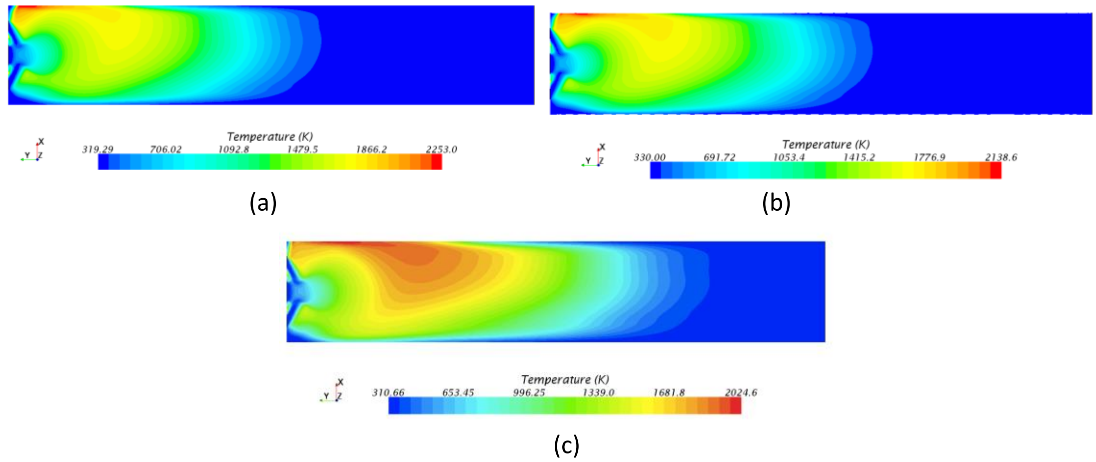


Figure 4. Axial mean temperature distribution along the tube mid-section at 0.5 sec. (a) dry air, (b) humid air with 5% RH, and (c) humid air with 10% RH.

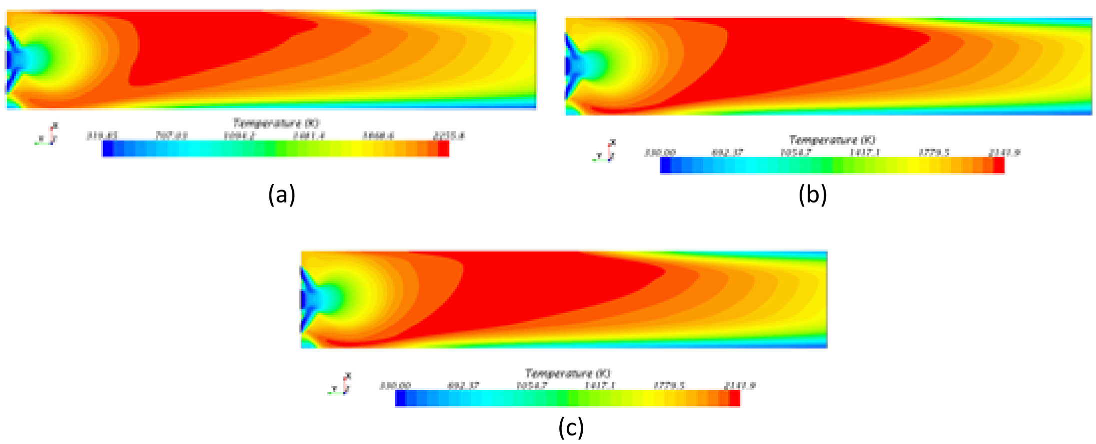
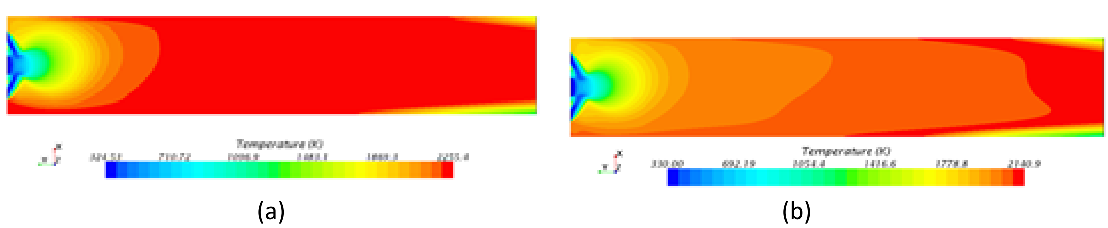
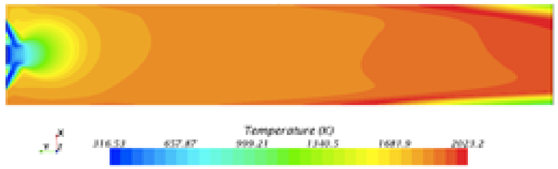


Figure 5. Axial mean temperature distribution along the tube mid-section at 1.5 sec. (a) dry air, (b) humid air with 5% RH, and (c) humid air with 10% RH.





(c)

Figure 6. Axial mean temperature distribution along the tube mid-section at 2.5 sec. (a) dry air, (b) humid air with 5% RH, and (c) humid air with 10% RH.

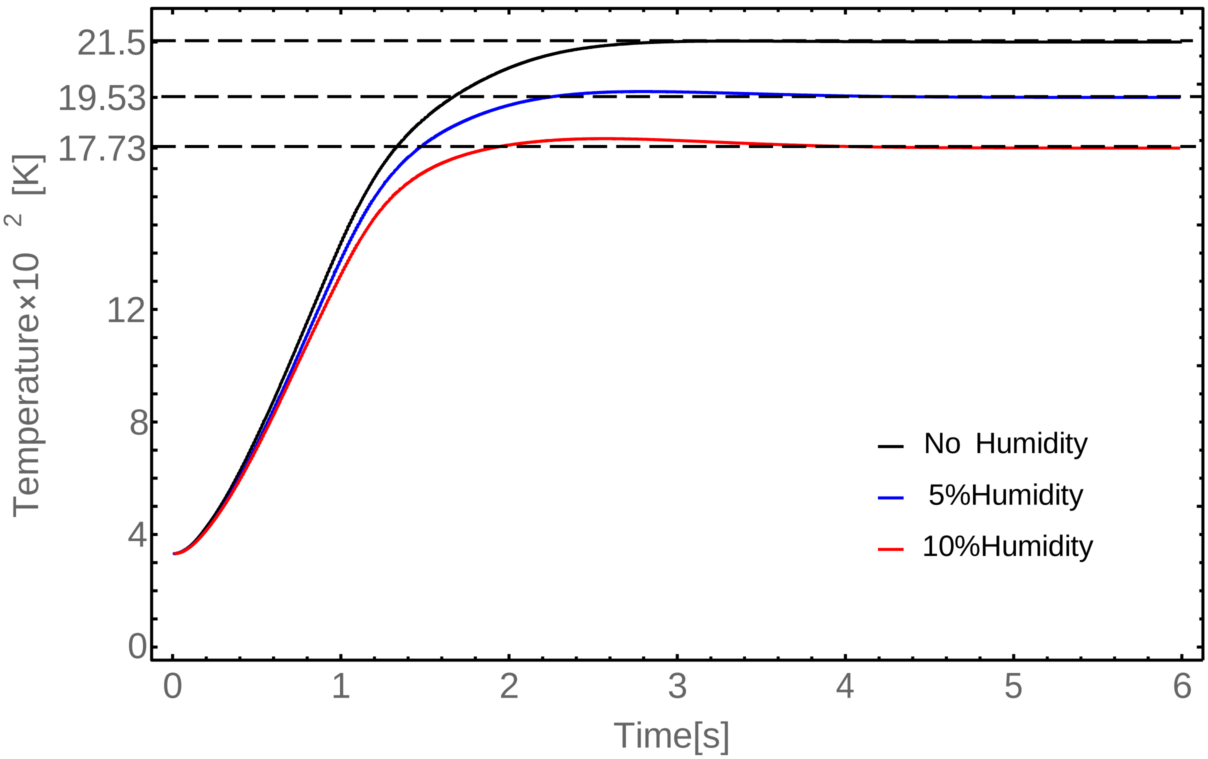
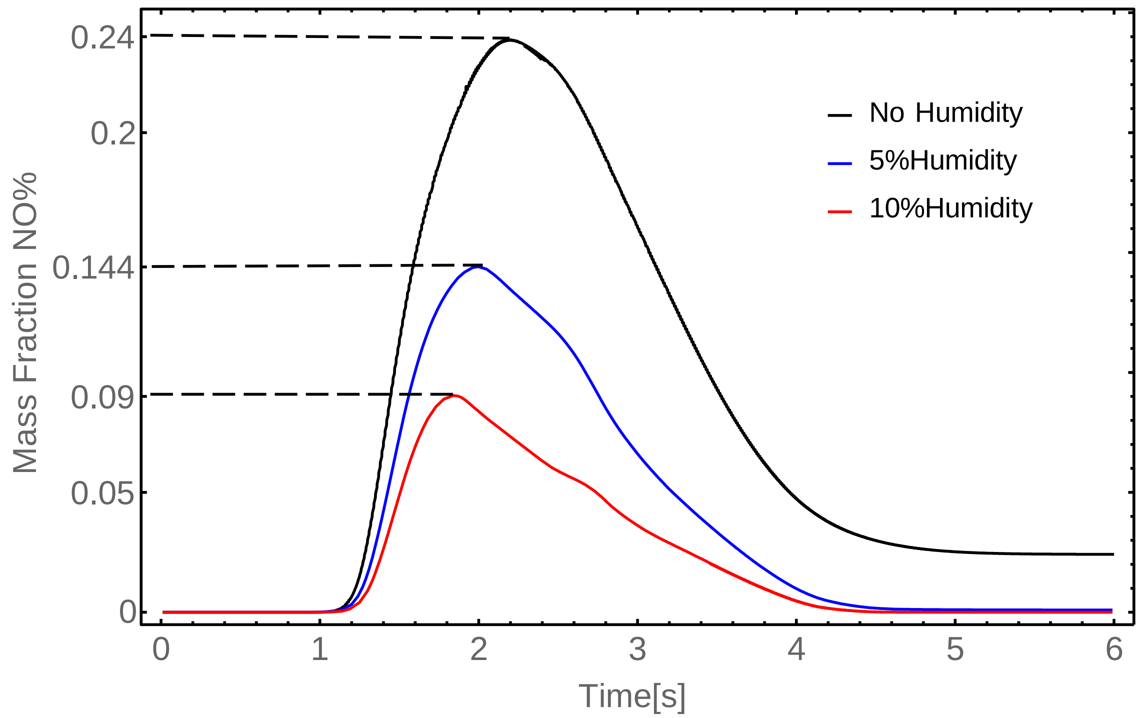
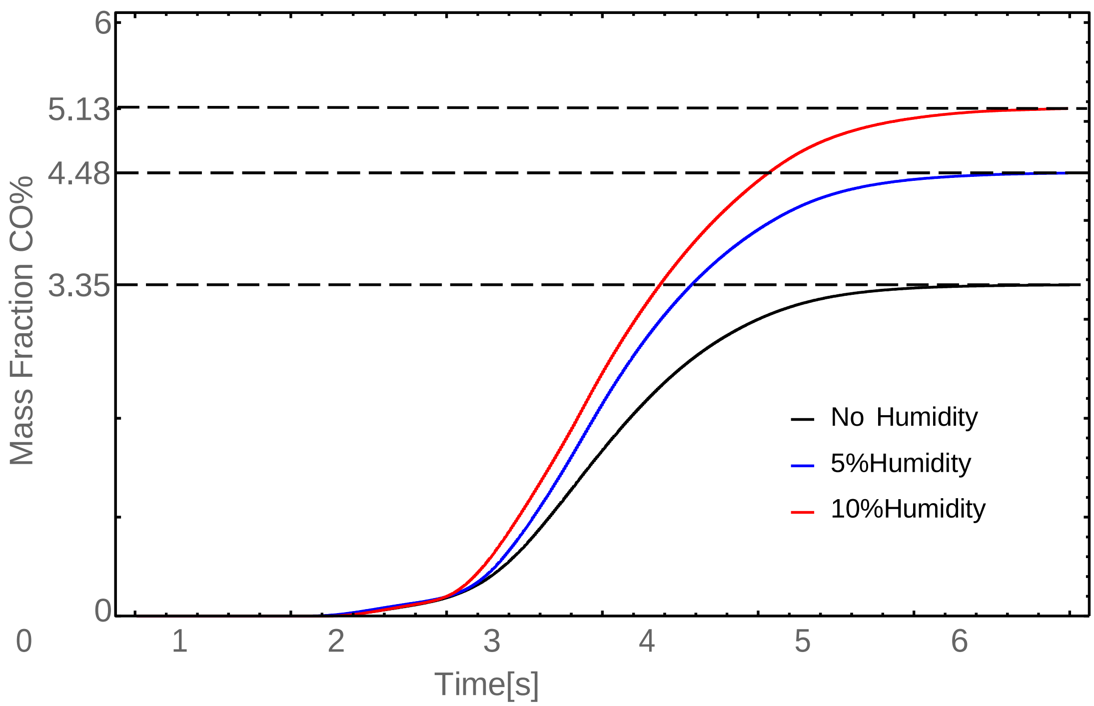


Figure 7. Exhaust temperature for three cases studied.



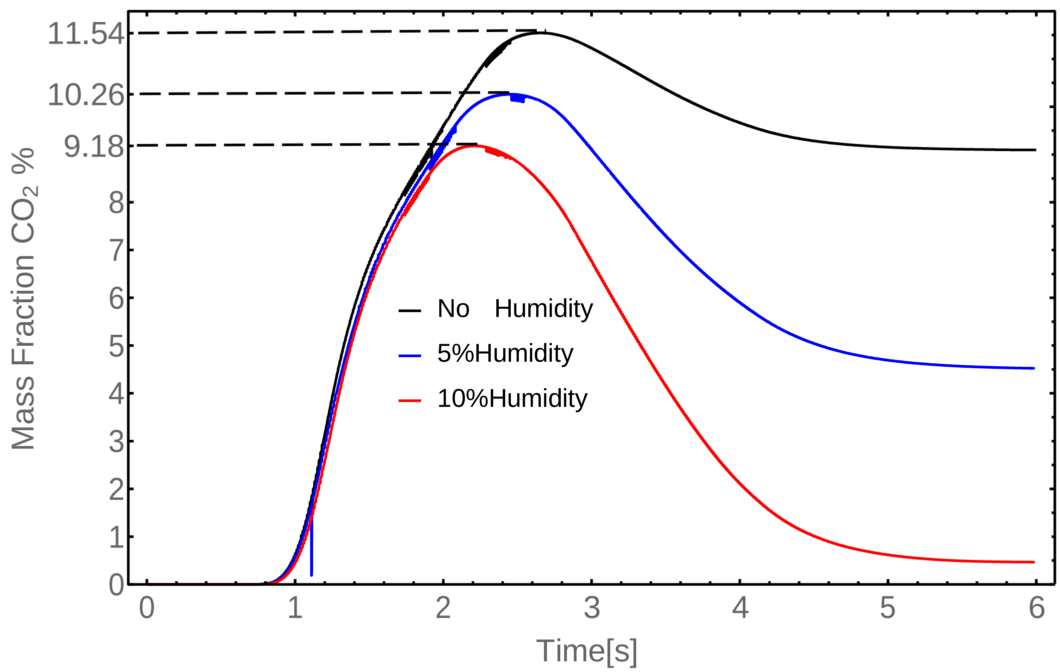
0.24
0.2
0.144
0.09
0.05
0
Mass Fraction NO%
0
1
2
3
4
5
6
Time[s]
No Humidity
5%Humidity
10%Humidity

(a)



6
5.13
4.48
3.35
0
Mass Fraction CO%
0
1
2
3
4
5
6
Time[s]
No Humidity
5%Humidity
10%Humidity

(b)



(c)

Figure 8. Percent variation of (a) NO, (b) CO, and (c) CO_2 for all three cases studied.

	NO	CO2	CO	Total Emission
Base Line (No Humidity)	NA	NA	NA	NA
5% Humidity	-53.19	-28.91	30.61	-18.65
10% Humidity	-72.34	-56.52	48.98	-38.37

Table 1. Percent change in emitted gases and total emission.

	NO	CO2	CO	Total Emission
Base Line (No Humidity)	NA	NA	NA	NA
5% Humidity	-10.64	-5.78	6.12	-3.73
10% Humidity	-7.23	-5.65	4.90	-3.84

Table 2. Rate of change in emitted gases and total emission.

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Part II:

Tables 3 and 4 show results of both gaseous and PM emissions of the engine tests at various loads. Cases 1-4 corresponds to the engine loadings of 5 HP, 12.5 HP, 25 HP, and 37.5 HP. The tests were performed at three different RHs of 30%, 45%, and 60%. The 30% RH corresponds to the ambient condition for the day that the test was performed. With addition of humidity, the NOx emission is reduced significantly for all cases studies. For case 1, at 60% RH, more than 60% reduction in NOx emission has been achieved and for case 4, which corresponds to the maximum loading condition, the reduction in NOx emission is at nearly 80%. The corresponding humidity-fuel mass ratio ranged from 0.64 to 0.82. Results indicate strong correlation between the humidity-fuel mass ratio and the % NOx reduction. It is expected that with a humidity-fuel mass ratio near 0.9, more than 90% NOx reduction could be achieved, especially at high loading conditions.

At low loads, the addition of the humidity results in significant increases in exhaust PM. For case 1, for 45% and 60% RH, the PM ratios, when compared to corresponding ambient PM (30% RH), are 5.6 and 8.3. These ratios for case 2 are 31.49 and 39.19 and for case 3 are 8.39 and 6.49 respectively. However at the highest loading condition (case 4), the ratios drop to around 2. Taking into account the loading HP, it indicated the PM weight per HP decreases significantly with increased HP.

These results indicate that for heavy duty CNG and LNG engines that are used for freight movements, and mostly are operating at high loading conditions, significant NOx emissions could be achieved without comparable increase in PM emissions.

Natural Gas Engine (50HP MAX)	Case 1			Case 2			Case 3			Case 4		
Power (hp)	4.9	4.8	4.9	12.6	12.7	12.8	26.5	25.1	25	37.4	36.5	37.7
Humidity Level	30%	45%	60%	30%	45%	60%	30%	45%	60%	30%	45%	60%
Ambient Humidity (%)	31.5	49.3	65	25	43.5	68.2	27.8	45.2	64.7	24.7	40.3	63.2
Ambient Temperature (°F)	87	91.1	102.4	92.3	96.1	106.6	89.5	98.5	110.4	92.9	97.4	109.1
Air Flow Rate(cfm)	27.02	28.01	29.83	32.51	33.85	36.74	44.1	45.42	47.15	56.32	59.16	62.5
Fuel Consumption Rate (cfm)	2.22	2.30	2.34	2.74	2.82	2.95	3.65	3.61	3.69	4.54	4.47	4.48
<RAW>												
NOx (ppm)	134	108	47	259	115	64	688	203	60	984	238	199
SO2 (ppm)	42.4	34.9	26.8	42.6	22.3	21.1	51.7	18.1	11.6	41	10.2	17.5
CO (ppm)	3780	3055	3109	3690	1965	1880	4550	1745	1103	4467	1245	1785
CO2 (%)	10.02	8.27	5.66	10.23	5.12	4.34	10.03	3.75	2.37	9.07	2.32	3.98
O2 (%)	1.68	5.15	10.08	1.43	11.14	12.54	1.51	13.67	16.22	3.46	16.32	13.01
<Diluted>												
NOx (ppm)	19	19	13	43	38	22	110	94	45	190	142	84
SO2 (ppm)	11.6	11.8	13	12.1	12.4	13	13.2	13.1	12.7	13	12.4	12.7
CO (ppm)	690	673	685	712	712	704	843	817	771	917	803	778
CO2 (%)	1.8	1.81	1.72	1.84	1.81	1.78	1.8	1.81	1.67	1.74	1.59	1.7
O2 (%)	17.2	17.33	17.45	17.19	17.32	17.39	17.23	17.33	17.55	17.33	17.7	17.53
Dilution Ratio	5.70	4.57	3.29	5.56	2.83	2.44	5.57	2.07	1.42	5.21	1.46	2.34
mass_air (g/min)	889.43	915.16	955.03	1059.87	1096.01	1167.53	1445.05	1464.31	1488.35	1834.12	1911.04	1977.40
mass_humidity (g/min)	7.64	14.16	28.10	8.53	17.47	41.17	11.85	26.17	55.80	14.86	29.34	69.50
mass_fuel (g/min)	41.95	43.36	44.12	51.67	53.24	55.68	68.88	68.17	69.68	85.64	84.43	84.55
Humidity-Fuel Mass Ratio	0.18	0.33	0.64	0.17	0.33	0.74	0.17	0.39	0.80	0.17	0.35	0.82
Dilution tunnel flow rate	300	305	305	305	300	305	305	300	295	305	305	300
Dilution tunnel avg temp	93.33	97.00	101.00	95.67	106.67	108.00	99.33	111.33	112.33	70.33	111.67	113.00
Ratio of NOx to baseline	1.000	0.806	0.351	1.000	0.444	0.247	1.000	0.295	0.087	1.000	0.242	0.202
Ratio of CO to baseline	1.000	0.808	0.822	1.000	0.533	0.509	1.000	0.384	0.242	1.000	0.279	0.400

Table 3. Experimental Results for Gaseous Emissions

	PM weight (mg)				PM ratio				PM weight per hp			
Humidity Level (%)	Case 1	Case 2	Case 3	Case 4	Case 1	Case 2	Case 3	Case 4	Case 1	Case 2	Case 3	Case 4
30	0.764	0.194	0.583	1.287	1.00	1.00	1.00	1.00	0.156	0.015	0.022	0.034
45	4.278	6.106	4.890	2.678	5.60	31.49	8.39	2.08	0.891	0.481	0.195	0.073
60	6.210	7.598	3.783	2.907	8.13	39.19	6.49	2.26	1.267	0.594	0.151	0.077

Table 4. Experimental PM Emissions

5.0. CONCLUSIONS

Numerical and experimental investigations of the effects of humid air intake at different humidity ratios, on exhaust NOx and PM emissions of a natural gas combustion model and a compressed natural gas (CNG) engine have been performed. For the numerical investigations, the humidity ratio changed from 5%

to 30% and for the engine experiments, the relative humidity tested were from ambient condition (at 30%) to 45% and 60%. The engine tests were performed at different loading conditions ranging from 5 HP to 37.5 Hp. For the numerical investigations, results indicate more than 90% reduction in NO emission with a humid air input at 30% RH. The experimental results also indicate that with additional 30% RH, nearly 80% reduction in NOx emission could be achieved at high loading conditions. Results of the engine tests have also indicated that there is a strong correlation between the percent reduction in NOx emission and the humidity to fuel mass ratio, especially at the high engine horse power. With a 10% humidity to fuel mass ratio, a 10% reduction in NOx emission is achieved.

With added humidity to the intake air, the PM emission is increased, especially at low horse powers (HPs). For the engine tests, the ratio of PM emission to the baseline emission (at 30% RH) is approaching 2 for the highest engine horse power. When PM emission is normalized by the corresponding HP loading, results indicate a graduate decrease in PM emission with increased loading at 45% RH and 60% RH.

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