EXPERIMENTAL INVESTIGATION OF WALL WETTING EFFECT ON HYDROCARBON EMISSION IN INTERNAL COMBUSTION ENGINE

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Abstract - In naturally aspirated Spark Ignition engine, about 60-80% of total unburned hydrocarbon emissions are produced during initial stage of vehicle operation under cold start or warm up condition. Wall wetting is predominant effect occurs in idle and part load conditions due to impingement or condensing of un-vaporized fuel droplets around the intake wall, combustion chamber liners and top of the piston. These deposits can cause incomplete combustion which will impact on increase in total hydrocarbon emissions. In this study, wall wetting parameters like fuel density, intake duct geometry, wall film thickness, wall film height, mixture preparation, fuel vaporization has been investigated theoretically by considering droplet evaporation and temperature model for cylinder wall film through mathematical equations. The main objective is to control equivalence ratio and to maintain surface temperature is the most effective way to reduce unburnt hydrocarbon emissions due to cold start wall wetting during steady state and transient conditions. This methodology was carried out on four stroke single cylinder Spark Ignition engine, where additives were used with gasoline fuel of different proportions which could intern reduce the intake and combustion chamber deposits during steady state and transient conditions. This experimental analysis was analyzed at different speed and load conditions. Based on these experimental results, Hydrocarbon emissions were reduced by nearly 40% in steady state and 30% in transient state. It is observed that, bi-fuel injection strategy can be implemented for injecting an ample amount of additive in to cylinder before compression stroke at 110 deg of crack angle during cold start, which it enhances the performance and emission characteristics furthermore.

1. INTRODUCTION

Improving engine efficiency and emission performance are the most important issues in modern automotive engine design. A modern automobile vehicle powered by a naturally aspirated spark ignition engine produces approximately 60%-80% of its tailpipe unburned hydrocarbon (UHC) emissions during the first twelve minutes of engine operation during cold start [4]. Unburned Hydrocarbons are also called volatile organic compounds. The presence of liquid fuel film in the combustion chamber and intake manifold can be a significant source of hydrocarbon (HC) emissions in port fuel injection (PFI) relative to gasoline direct injection (GDI) engines [6]. Under such conditions, there is insufficient time for the fuel injected in the intake port to completely vaporize due to the low surface temperatures of the intake port and intake valves [9]. Possible sources of hydrocarbon emission are wall wetting or flame quenching, decreased in-cylinder and exhaust oxidation which result in lower average temperature due to the overall lean operation. Liquid fuel impingement on in-cylinder surfaces, absorption of fuel vapour into oil layers during intake and compression and desorption of fuel vapour during expansion and exhaust stroke [7]. In PFI engines, some of the fuel that does not vaporize completely in the port is condensed on the walls of the
combustion chamber, where it doesn’t under-go combustion. This behaviour occurs predominantly under cold-start and warm-up condition is called wall wetting. This process of wall-wetting results in slower vaporization of the liquid fuel which contributes to higher HC emissions during early stage development. Thus current fuel control strategies like adding detergents and sending hot EGR were not capable of controlling the equivalence ratio during cold-engine speed and load transients [5]. Therefore, fuel is getting rich at corners of combustion chamber and decreasing its performance and drivability characteristics with increase of emissions. The primary concern is to reduce the wall-wetting phenomenon in the port fuel injection. Most of the HC emissions result from the flame being unable to propagate in fuel-air mixture located in cold regions and narrow passages around the combustion chamber [7]. Combustion Chamber deposits play an important role in contributing HC emissions. If engine operated for a long period of time under cyclic and variable load transient conditions result in increase of deposit thickness in combustion chamber walls.

The primary objective is to investigate and estimate the wall wetting effect in in-cylinder Spark Ignition Engine by numerical prediction on hydrocarbon emissions occurs at initial stage of development. The secondary objective is to reduce the HC emissions in homogenous conditions on port fuel injection relative to direct gasoline injection in steady state and transient conditions by adding necessary additives into gasoline.

1.1 Wall Wetting Phenomenon

In wall wetting, fuel gets atomized into tiny droplets so as to form vaporized fuel and air mixture. Certain portion of the vaporized fuel would condense on the intake manifold, forming the fuel puddles due to low evaporation rate. However, after valve closes during suction due to lack of temperature and oxygen required to burn the fuel at times, un-vaporized fuel impinge on the cylinder liners and piston top. Wall wetting or Flame quenching occurs, as the flame approaches the walls in the combustion chamber heat is lost due to low surface temperature at the walls. This result in temperature drops in that particular reaction zone. This will lead to lower the gas temperature below ignition temperature.

In this research, during combustion by adding additives at early stage of flame development that looks brief over the pre-oxidation stage. Wall wetting occurs due to parameters such as thickness and structure wall geometry, decrease in fuel consumption, irregular mixture of air fuel, flame speed propagation, which lead to sudden drop in temperature and turbulence created inside combustion chamber [4].

1.2 Literature Survey on Cold Start Emission

It is known that, when engine is operated in cold start conditions it decreases the fuel economy and increase in tailpipe UHC emissions. The main problem is influence of controlling equivalence ratio and temperature of exhaust during early flame development stage are starting an engine with optimum fuel control achievement is extremely a difficult task because of the variation in engine speed airflow manifold pressure at which it creates residual mass fraction in cylinder in fraction of time.

By literature survey, it is observed that,

- Lack of air inside combustion chamber causes increase of residual gas mixture and drop in pressure in combustion chamber causing back flow through intake [3].

- Preheating of air intern increase in fuel atomization rate but it will decrease in air density ultimately effecting volumetric efficiency [8].
• Sending hot EGR will increase fuel vaporization but it will increase No emission due to high temperature [5].

• Stoichiometric combustible range needs to be maintained at lean mixture due that flame temperature is stabilized to maintain good thermal stability [13-14].

2. NUMERICAL ESTIMATION OF WALL WETTING

Investigation on wall wetting effect is particularly done by considering parameters which is shown in introduction according to engine operation at cold start. Identifying the fuel vaporisation and finding out the consumption of fuel at running condition is more difficult because usually the parameters vary over the whole operating range. The on-going adaptation of these parameters is difficult because the continuous variation of the operating conditions which will not allow to identify a particular algorithms. This can be obtained by developing models and which will be derived by using physics of problem. This model is taken from a base model of extended Kalman filtering method in which this helps to find the unburnt residual mass fraction in cylinder through off-line strategy [4]. Evaporation from wall film represents that the amount of fuel vaporize from the wall at certain speed and load operating conditions. The rate of evaporating fuel is strictly dependent on the size of the surface that is exposed to the air flow. To estimate, fuel mass entering in to cylinder and fuel mass evaporating from wall surface need to be calculated. [4]

Flame propagation and heat transfer normal to single wall done by conduction

\[
\delta q = \frac{a_u}{s_l} \times \frac{c_{pu}}{c_{pb}} \times \frac{\Delta T_c}{\Delta T_f}
\]

Where, \( \delta q = \text{quench distance} \)
\( \Delta T_c = \text{temperature difference for heat transfer} \)
\( \Delta T_f = \text{flame temperature} \)
\( s_l = \text{laminar flame speed} \)
\( m_{wv} = \text{fuel mass injected - evaporated mass} \)
\( m_{evf} = \text{fuel evaporated from film} \)

Quench distance is directly proportional to Peclet number and laminar flame speed. Peclet number for flame quenching between two parallel plates is 5 times of single wall quench distance.

Quench thickness ranges from 0.04 to 0.2 mm for SI engines. In this particular study, pressure is at 10 bar and \( s_l = 0.2 \text{ m/s} \) then wall quench estimated to be 0.05 to 0.1 mm.

Effects of wall film is determined by A/F sensor only after opening of exhaust valves and delay of gases in exhaust is determined by considering a homogeneous mixture by once in every cycle on the basis of speed and load characteristics. Small error in estimation of air and residual gas mixture leads to problem in identification. Air flow speed is calculated by measurement of air mass flow through intake and across duct surface.
3. EXPERIMENTAL SETUP

The experiment was conducted on a single cylinder, 4 stroke naturally aspirated Spark Ignition engine varying with different speed and load conditions. The specification details of the engine are listed in Table 1. The schematic diagram of the experimental setup is shown in Figure 1. The engine was coupled with eddy current dynamometer (50kW) as shown in pictorial view in Figure 2. Time taken for 10cc of fuel consumption was taken. The inlet and exhaust outlet gas temperatures are measured using k-type thermocouples. The Horiba gas analyzer was used to measure the concentration of exhaust emissions (CO, CO2, HC, No and Lambda). Dynamometer load is controlled by adjusting pressure on water inlet and outlet pipes. Since the experiment was conducted on cold start or engine warm-up condition, we have done the analysis at idle speed with no load and part load conditions. Several iterations has been done by varying its speed conditions at 2000 rpm, 2400 rpm, 2800 rpm and 3200rpm, where loads at 0%, 25%, 50% and 75% in both steady and transient state conditions.

Table 1. Engine Specifications

| Engine Name          | Briggs and Statorn 21RB |
|----------------------|-------------------------|
| Engine Configuration | Vertical Shaft          |
| Engine Displacement (cc) | 392                    |
| No. Of cylinders/cycle | One/4-Stroke           |
| Compression Ratio    | 8.1:1                   |
| Rated Power (KW)     | 9                       |
| Ignition System      | Naturally Aspirated Spark Ignition |
| Bore (mm)            | 87.3                    |
| Stroke (mm)          | 62.4                    |
| Injection Type       | Port Fuel Injection     |
| Cooling Medium       | Air Cooled              |
| Lubrication System   | Pressure Lubrication     |

3.1 Methodology

Before starting the experiment, basic elements required such as Eddy current Dynamometer, Load controller, Gas analyser and data acquisition from k-type thermocouple. Initially, Engine and Dynamometer is coupled by cardan shaft with universal joints at both ends along with flexible couplings. Secondly, by neutralizing the water flow in the pipe at particular pressure level. At engine exhaust, gas analyser and probe was connected. Thirdly, filling fuel tank with gasoline which is located at higher end and run the engine in idle speed with no load condition pure gasoline fuel. Fuel consumption at 10cc is measured along with exhaust temperature and base line data is generated by gas analyser. Test is subsequently operated at suitable speed and load conditions which are given on load controller. Thereafter, fuel tank is filled and run the test with 90vol% of gasoline along with additives of acetone 5vol% and N-butanol 5vol%. The same process is repeated by changing proportions of acetone 10vol% and N-butanol 10vol% along with gasoline 80vol%. The noted values
to be marked as base reference. By taking obtained parameters, formulated data can be used to solve wall wetting film condensation on cylinder wall by numerical modelling. The performance, combustion and emission characteristics are investigated, which includes brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), Thermal stability and gas emissions and temperature from the exhaust manifold are calculated. while, in this study HC emissions are concentrated at early stage of flame development. Furthermore, emissions like NO, CO and Co2 is also observed.

**Figure 1. Schematic Diagram of Experimental Setup**

![Figure 1. Schematic Diagram of Experimental Setup](image1)

**Figure 2. Pictorial View of Experimental Setup**

![Figure 2. Pictorial View of Experimental Setup](image2)
Table 2. Properties of Additives and Gasoline

| Parameter                                      | Gasoline | Acetone | N-Butanol |
|------------------------------------------------|----------|---------|-----------|
| Chemical Formula                               | C_{4}-C_{12} | C_{3}H_{6}O | C_{4}H_{9}OH |
| Composition (Mass %)                           | 86, 14, 0 | 65, 13.5, 21.5 | 62, 10.5, 27.5 |
| Lower Heating Value (MJ/kg)                    | 43.4     | 33.1    | 29.6      |
| Density (kg/m³)                                | 715-765  | 810     | 790       |
| Energy Density (MJ/l)                          | 32.2     | 26.81   | 23.38     |
| Octane Number                                  | 90       | 87      | -         |
| Boiling Temperature (°C)                       | 25-215   | 118     | 56.2      |
| Latent Heat of Vaporization (kJ/kg)            | 380-500  | 716     | 518       |
| Self-Ignition Temperature (°C)                 | ~300     | 343     | 465       |
| Stoichiometric Air/Fuel Ratio                  | 14.7     | 11.2    | 9.5       |
| Laminar Flame Speed LFS (cm/sec)               | ~33      | ~48     | ~34       |
| Mixture Calorific Value (MJ/m³)                | 3.72     | 3.82    | 4.04      |
| Ignition Limits in Air (vol.%)                 | 0.6-8    | 1.4-11.2| 2.6-12.8  |
| Solubility in water at 20°C                    | <0.1     | 7.7     | Fully Miscible |

4. RESULTS AND DISCUSSION

4.1 Hydrocarbon Emission

As discussed earlier, it was observed that performance and emission characteristics are observed by conducting on test bed. Additives added into gasoline at different proportions for overall run of engine at different speed and load conditions to verify it in steady state and transient conditions. Due to equal density ratio, additives can able to mix with gasoline directly without any transification process. Insertion of additives intern provides oxidation and increase in temperature inside combustion Load

![Figure 3. Variation of HC Emission at No Load](image1)

![Figure 4. Variation HC Emission at 2.5Nm](image2)
chamber to avoid presence to quench liquid film during flame development stage.

From the above Figures 3,4,5,6; it is shown that, at low speed 2400rev/min condition Acetone 5vol.% and N-butanol 5vol.% along with gasoline 90vol.% [A5+B5+G90] gives less HC emission compared to pure base gasoline and Acetone 10vol.% and N-butanol 10vol.% with gasoline 80vol. %[A10+B10+G80] in all load conditions. As HC emission trend keeps increasing at 2800rev/min in all three compositions. Still [A5+B5+G90] gives very less emission during that particular speed at idle load and part load conditions due to oxidation from additives. Whereas, at full load condition pure gasoline gives better efficient on HC compared to both. At 3200rev/min high speed condition, A5+B5+G90 shows drastic increase of HC emission due to increase in temperature by oxidation into combustion chamber. Moreover, A10+B10+G80 gives better than other one at full speed and high load condition. It is inferred that A5+B5+G80 gives impact during idle and part load low speed condition during cold start or engine warm up.

4.2 Influence on Exhaust Temperature

Maintaining exhaust and in cylinder temperature is very crucial parameter for depletion of impingement film layer over the cylinder wall. When engine is running on dynamometer operated at high speeds, maximum temperature which can obtained is about 700°C. In this particular wall wetting phenomenon exhaust temperature is one of the main constrained to show that wetting or quenching doesn’t occur. As we know that, gasoline produce an exhaust temperature due to flame development inside combustion chamber while we can see by adding additives to gasoline the exhaust temperature keeps increasing and increasing tremendously where it maintains constant throughout the operation.

Figure 7. Variation of Exhaust Gas Temperature at No Load  Figure 8. Variation of Exhaust Gas Temperature at 2.5 Nm Load
From above Figures 7, 8, 9, 10 it is examined that A5+B5+G80 and A10+B10+G90 attains higher exhaust temperature than pure gasoline at low speed to high speed with variations of loads. Whereas, pure gasoline step by step increased drastically through the operation of vehicle in steady state and transient conditions. Additives give oxidation equally thought out the operation and maintain constant temperature at low and high speed and idle part load. However, in cold start during low speed no load condition A5+B5+G80 attains more temperature than other due to that complete combustion takes place and decrease in presence of quench film can also be seen. There after it should maintain constant throughout out to increase its thermal stability. During cold start flame doesn’t propagate properly to the combustion wall so that it requires more oxidation which it can be provided from additives which is been added.

4.2.1 Overall Variation In Between HC and NO Formation

This variation gives more precise formation at times with HC and NO. Because, when vol% of ppm decrease in HC may increase of NO due to increase in temperature inside combustion chamber. Here it is shown that at idle and part load conditions variation can be seen in between HC and NO.

From the Figure 11 & 12, it is shown that, in both graphs similarity is shown at different speeds and load conditions among HC and NO formation. According to trend line, At low speed part load condition, A5+B5+G90 and A10+B10+G80 gives HC and NO low constant emission at 2400rev/min.
compared to pure gasoline due to increase in oxygen content in combustion chamber. Whereas, in high speed at 3200rev/min HC and NO drastically increased to 310ppm and 265ppm due increase in temperatures at exhaust. It is seen that high speed conditions A10B10G80 gives less NO emissions compared to HC emission for gasoline and A5B5G90.

**Figure 13.** Overall Variation of HC vs. Speed at 5 Nm Load

**Figure 14.** Overall Variation of NO vs. Speed at 5Nm Load

From the Figure 13 & 14, it is shown that, in both the graphs similarity is shown with respect to HC and NO formation. According to trend line, at low speed medium load condition, A5+B5G90 and A10B10G80 gives HC and NO gives linearly constant increased emission compared to pure gasoline due to increase in oxygen content in combustion chamber. HC and NO for pure gasoline are indirectly proportional from low speed to high speed at same load condition. Whereas, A10B10G80, HC drastically increased to 184ppm and 117ppm during 2800 rev/min and 3200rev/min and NO is maintained same compared to HC. It noticed that at high speed conditions NO A10B10G80 and A5B5G90 gives less NO emissions compared to gasoline emission.

**5. CONCLUSION**

The following conclusion is made from the present study:

- Additives like Acetone and N-Butanol used to increase oxygen content and temperature rise in-cylinder which helps to reduce HC emission and segregation of fuel puddles due to wall wetting during cold start at steady state condition as well transient condition.

- From experimental results, HC emission is minimized to a certain extent at low speed low load conditions by slight increase of NO due to increase in temperature compared to pure gasoline.

- Compared to the pure gasoline fuel, at 2400rev/min, Acetone 5vol% and N-Butanol 5vol% mixture with gasoline 90vol.% gave maximum reduction in unburnt hydrocarbons, CO and HC by 14.80%, 18.59%, 32.53% respectively than at 2800rev/min and 3200rev/min.

Furthermore, Bi-fuel injection strategy can be implemented before compression stroke directly into combustion chamber by adding additives of acetone and butanol mixture in gasoline during cold start condition with respected to speed with for a little span of engine operation and shutoff.

**ACKNOWLEDGEMENT**

The author would like to thank Mr. M.Sivanesan for supporting this research with valid suggestions and I would also like to thank specially Amrita Automotive Research and Technology Centre for arranging required experimental apparatus for completion of the project and also like to thank Head of the Department Dr. S.Thriumalini and Prof. S.Srihari for their assistance.
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