An Experimental Evaluation of NO_x Reductions from Hydrogen Enhanced Diesel Combustion

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1 Abstract

Hydrogen (H_2) has long been recognized as a carbon-free energy carrier having excellent combustion and emissions characteristics suitable for application in internal combustion (IC) engines. Compared to its application in spark ignition (SI) engines, the burning of H_2 in compression ignition (CI) engines using a dual fuel combustion mode can obtain higher brake thermal efficiencies with a more flexible requirement of the quality and quantity of H_2 containing gaseous fuels, provided that knock is avoided. The addition of H_2 into the intake mixture of small diesel engines has been shown to improve the brake thermal efficiency with reduced emissions of particulate matter (PM), carbon monoxide (CO) and unburned hydrocarbon (HC). However, the addition of H_2 to diesel engines was reported to increase the emissions of oxides of nitrogen (NO_x). Typically, past research in this area was performed using small diesel engines with PM data measured using a smoke meter and NO_x emissions reported in ppm. There is recent interest in examining the effects of H_2 addition on the performance, combustion and emissions characteristics of heavy-duty diesel engines with PM and NO_x emissions measured using well accepted emission measurement approaches and reported in g/bhp-hr.

This research investigated the effects of H_2 addition on performance, combustion, and emission characteristics of two turbocharged heavy-duty diesel engines. The effects of H_2 addition, the engine load, speed, and diesel fuel flow rate on the brake thermal efficiency, cylinder pressure, combustion process, and exhaust emissions of NO_x, PM, HC, HC, CO_2 and unburned H_2 were explored. The engine load was varied from 10% to 100% with H_2 concentration in the fresh intake mixture (H_2/(H_2+Air), vol. %) varied from 0 to 7.5%. Engine speed was varied from 1200 rpm to 1800 rpm.

The addition of H_2 to the 1999 Cummins ISM370 engine (without exhaust gas recirculation (EGR)) and the Mack 2004 MD11 engine (with EGR) was shown to significantly reduce the emissions of PM. However, its addition to the 1999 Cummins ISM370 diesel engine at medium to high load was shown to substantially increase the NO_x emissions especially when a relatively large amount of H_2 was added. In comparison, the addition of H_2 into the Mack 2004 MD11 diesel engine at medium to high load was found to have less effect on NO_x emissions. For both engines, the addition of a relatively small amount of H_2 at low load was shown to have negligible effect on NO_x emissions. In comparison, its addition at a relatively large amount under very narrow low load operation range reduced the emissions of NO_x. Based on the experimental data obtained in this research, the significant reduction in NO_x through H_2 enrichment of heavy-duty diesel engines was found to be infeasible.

The improvement in brake thermal efficiency has been one of the main objectives of engine research. Based on the heating value of diesel fuel and H_2 without considering the extra energy cost for the production of H_2, the addition of relatively large amount of H_2 at medium and high load was shown to improve the brake thermal efficiency benefiting from the enhancement to the combustion process, and improvement to the indicated thermal efficiency and mechanical efficiency. In comparison, the addition of a small amount of H_2 at low load was shown to lower the brake thermal efficiency. The different effect on brake thermal efficiency of the addition of H_2 at a relatively small and a relatively large amount demonstrated the presence of a minimum H_2 supplementation limit. The addition of H_2 less than this limit resulted in a penalty in the brake

thermal efficiency. The minimum H_2 flow rates needed for a positive effect on the brake thermal efficiency were 272.7-90.3 liter/minute (l/m) and 284.3-152.0 l/m (standard conditions) for 15% to 70% load operation of the 1999 Cummins ISM370 and 2004 Mack MD11 engine, respectively, when operated at 1200 RPM.

The effect of H_2 addition on the cylinder pressure and combustion process was experimentally investigated. The addition of a relatively large amount of H_2 at medium to high load was shown to increase the cylinder pressure and advance the peak cylinder pressure phasing. This was due to the development of an enhanced heat release peak observed at the diffusion combustion stage. In comparison, the addition of H_2 at very low load was shown to reduce the cylinder pressure and retard the peak pressure phasing due mainly to the retarded injection timing and the inhibited premixed combustion. Based on the heat release process information obtained through processing the cylinder pressure data, the addition of H_2 to a diesel engine was shown to retard and inhibit the premixed combustion. The featured three-stage heat release process of diesel-H_2 dual fuel combustion was only observed at medium to high load operation when a healthy H_2-air flame can be initiated. In comparison, the addition of H_2 to these two engines at low load was shown to significantly inhibit the premixed combustion without initiating a strong H_2 flame in the diffusion combustion stage, which resulted in deteriorated brake thermal efficiency.

Dual fuel engine operation suffers from the slip of the gaseous fuels supplemented into the intake mixture especially at low load operation. The emissions of the unburned H_2 not only compromise its potential in producing power and improving the thermal efficiency of diesel fuel, but also raise severe safety concerns. The experimental data obtained in this research demonstrated the preference of H_2 addition in relatively large amount under medium to high load operation to obtain high H_2 combustion efficiency and improved brake thermal efficiency. In comparison, the addition of H_2 at low load operation should be avoided due to the unacceptably high H_2 emissions and dramatic deterioration in the brake thermal efficiency.

The feasibility of improving the brake thermal efficiency of heavy-duty diesel engines using on-board H_2 production devices was examined and discussed. In principle, H_2 can be produced by H_2O electrolysis utilizing the electricity produced on-board with the consumption of mechanical work of the diesel engines. It can also be produced on-board by reforming diesel fuel to produce H_2 rich gaseous fuels. Based on a review of literature, the conversion efficiency of diesel fuel to H_2 utilizing H_2O electrolysis was 26.6% when evaluated using the lower heating value of diesel fuel and H_2. In comparison, the conversion efficiency of H_2 production through diesel fuel reforming was about 75%. When evaluated on the basis of diesel fuel consumption, the integration with heavy-duty diesel engines of the on-board H_2 production devices using H_2O electrolysis will lower substantially the overall brake thermal efficiency. In comparison, the application of diesel reforming technology will also reduce but at less extent the overall brake thermal efficiency. Based on the data measured in the research and H_2 production efficiency assumed in this research, it seems infeasible to improve the brake thermal efficiency of the heavy-duty diesel engines using the on-board H_2 production technology.

2 Introduction

Diesel engines are widely used in on-road and off-road vehicles due to their high power density and desirable fuel conversion efficiency. However, further development of advanced diesel engine technologies is facing significant challenges. These include the requirements of simultaneously reducing exhaust emissions of both NO_x and PM, the two problematic pollutants associated with diesel engines. In principle, NO_x of diesel engines is formed under high combustion temperature and oxygen (O_2)-rich operating conditions. The approaches that tend to reduce the combustion temperature will reduce the engine-out NO_x emissions usually accompanied with reduced brake thermal efficiency. In comparison, PM is formed in diesel engines due mainly to the heterogeneous combustion characteristics of diesel engines. The presence of fuel rich mixture under high temperature and pressure enhances the formation of PM due to the lack of sufficient O_2. The approaches that help to improve the fuel vaporization and its sufficient mixing with air will reduce the formation of PM but enhance the formation of NO_x due to the increased combustion temperature. The application of cooled EGR has been demonstrated as an effective approach that can significantly reduce NO_x emissions. However, the application of EGR increases the emissions of PM especially under high load operation.

Hydrogen has long been recognized as a carbon-free fuel having excellent combustion and desirable emissions characteristics for applications in IC engines. The burning of H_2 in air produces mainly H_2O, a small amount of NO_x but without fuel resourced PM emissions. These features make H_2 an excellent fuel for both traditional power production devices including IC engines and the latest innovative power devices such as fuel cells. The application of H_2 in these devices makes it possible to potentially meet the increasingly stringent environmental regulations of exhaust emissions, including the possible elimination of greenhouse gas emissions and significant reduction in exhaust emissions. The concise statements and detailed discussions of the positive features of H_2 as a fuel and the associated limitations to its wide application have been the subject of much research and many publications [Furuhama, 1983; Das, 1990; Karim, 2003; White, et al. 2006].

As a promising fuel, H_2 can be used in SI and CI engines, gas turbines, and fuel cells. Operation of most fuel cells requires pure H_2. The presence of other components such as CO and sulfur dioxide (SO_2) could significantly deactivate the catalyst of fuel cells and substantially reduce the service life. Such an excessively demanding requirement for high purity H_2 makes the operation of H_2 fuel cells economically uncompetitive though high thermal efficiency could be achieved without formation of pollutants. In comparison, IC engines can burn almost any low purity H_2 even with the presence of quite a large amount of diluents. For example, the reformed gas containing mainly H_2 with the presence of CO, CO_2, H_2O, and N_2 has been demonstrated as a fuel showing desirable H_2-like combustion properties. The application of H_2 as sole fuel or mixtures with traditional slow burning fuels also offers the opportunity of optimizing engine performance and reducing exhaust emissions. Considering the significant difference in the combustion system of gasoline engines and diesel engines, the application of H_2 in these engines is to be reviewed, respectively.

2.1 Hydrogen Application in SI Engines

Hydrogen has long been recognized as a fuel having some unique and highly desirable properties, such as low ignition energy, very fast flame propagation speed, and wide lean operational range. H_2 has also been demonstrated as a fuel being able to support a propagating flame with extremely lean mixture, which is a very attractive property of the H_2 as a SI engine fuel. The extensive research of pure H_2 as fuel has led to the development of the SI H_2 engine. For example, Ford developed the P2000 lean burn SI H_2 engine, which was used to power Ford's E-450 Shuttle Bus [Tang, et al., 2002; Stockhausen, et al., 2002; Gopalakrishnan, et al., 2007]. BMW developed a 6 liter, V-12 engine using liquid H_2 as fuel. With an external mixture formation system, this engine has a rated power of 170 kW and rated torque of 340 N.m [Kiesgen, et al., 2006].

Most of the research associated with H_2 application in the SI engines focused on its partial substitution for gasoline, known as H_2 enrichment. Extensive research has demonstrated that H_2 enrichment help to improve the performance of the SI engine for the following reasons: (1) Enhancing the flame propagation rate: the propagation rate of H_2 flame is about 4 times that of traditional fuels. The addition of H_2 to SI gasoline engines was shown to enhance the flame propagation rate; (2) Expanded lean operational region due to the extended lean operational limit benefiting from the super capability of H_2 in supporting flame propagation in a very lean mixture. The addition of H_2 to SI gasoline and natural gas engines has been shown to expand the operational region toward the leaner mixture, which is very important to obtain extremely low NO_x emissions; (3) Improving combustion stability (less cycle to cycle variation) and enhancing combustion efficiency with reduced emissions of CO and HC. Detailed information can be found in the literature [Das, 1990; Karim, 2003; Li and Karim, 2005; Munshi, et al., 2004; Topinka, et al., 2004; and White, et al., 2006]

2.2 Hydrogen Application in Diesel Engines

Diesel engines are widely used in on-road and off-road vehicles due to their high power output and attractive higher thermal efficiency compared to SI engines. Although a smaller population compared to SI engines, diesel engines consume a large amount of fossil fuels and contribute to a large portion of PM and NO_x emissions. The development and application of innovative combustion control and after-treatment systems has significantly reduced the emissions of PM and NO_x. However, their potential in reducing fuel consumption and GHG emissions are very limited. In fact, the application of EGR and after-treatment devices was demonstrated to increases the fuel consumption due to either the deteriorated combustion or the specific operating conditions that have to be satisfied for the after-treatment device to function properly. There is increasing recent interest in burning H_2 in diesel engines aiming to reduce both NO_x and PM emissions as well as CO_2 emissions.

2.2.1 Application of H_2 in Diesel Engines as Sole Fuel

Although the energy needed to ignite an H_2-air mixture is much less than that for most hydrocarbon fuels, the ignition temperature of H_2 is much higher than diesel fuel. Without the help of an ignition assistant device such as a spark or glow plug, it is very difficult to ignite H_2 through compression only. For example, the research of Homan, et al. [1979] demonstrated that it was very difficult to operate an H_2 compression ignition engine due to the high self-ignition temperature. Wong [1990] investigated the feasibility of using H_2 as the sole fuel in a direct injection diesel engine. It was concluded that the application of H_2 as sole fuel in direct injection diesel engine without an external ignition source was neither practical nor feasible. A number of researchers demonstrated that it was necessary to employ a suitable assistant device to ignite an H_2-air mixture. The approaches tested included glow plug, spark assistance, and pilot injection of diesel [Karim and Klat, 1982; Karim, 1976; Gopal, et al., 1982]. For example, Furuhama and Fukuma [1986] developed a turbo-charged, two-stroke diesel engine fuelled with H_2. A hot surface igniter was developed and demonstrated to be necessary to ignite H_2 supplied by direct in-cylinder injection.

2.2.2 Application of H_2 in Diesel Engines as Supplemental Fuel

Considering the high temperature needed to ignite an H_2-air mixture, external ignition assistance is needed for the reliable, stable, and repeatable ignition of H_2-air mixtures. It is well accepted that H_2 can be burned in diesel engines as a supplemental fuel either mixed with air in the intake manifold or injected directly into the combustion chamber prior to the injection of diesel. Prior to the injection of pilot diesel fuel, H_2 and air have been mixed well and formed to some extent a homogeneous fuel-air mixture. The mixture is then compressed to a high temperature, but not high enough to initiate the auto-ignition process of H_2 in air. After being injected into the cylinder, the pilot diesel fuel is atomized, vaporized, mixed with H_2-air mixture and ignited through auto-ignition. Under suitable conditions, the burning of diesel fuel serves as an energy resource to ignite the H_2-air mixture at multi-points. The H_2-air mixture will either be burned through flame propagation similar to that of an SI engine if the H_2-air mixture is rich enough to support the propagation of the flame, or the H_2 may be oxidized in air if the H_2-air mixture is leaner than the flammability limit. Numerous relevant experiments were reported in the literature. For example, Gopal, et al. [1982] tried to operate a diesel-H_2 dual fuel engine in which the compressed hot H_2-air mixture was ignited by means of a pilot diesel injection process. It was shown that the supplementation of H_2 to the diesel engine would increase the energy release rate of the premixed combustion stage and thereby increase the brake thermal efficiency. The detailed information of dual fuel engine combustion processes can be found in the literature [Boehman and Corre, 2008 and Karim, 2003].

2.3 Review of H_2-Diesel Dual Fuel Engines

The research of H_2-diesel dual fuel engines was initiated by the anticipated positive effects in enhancing the engine performance such as improved power production efficiency and its substitution to traditional fuel. The concern for CO_2 emissions and potential of H_2 in reducing PM and possible NO_x emissions are the recent triggers for its application in diesel engines. Similar to diesel engines, the exhaust emissions of an H_2-diesel dual fuel engine includes PM, NO_x, CO, unburned HC, and also unburned H_2. The past research has demonstrated that the addition of H_2 into diesel engines would lengthen the ignition delay, retard the combustion phasing and reduce the emissions of PM, CO, HC, and CO_2. However, research conducted by Varde and Frame [1983] reported that both CO and HC emissions increased with H_2 addition, which was not consistent with most of the past research. Considering the relatively small emissions values of diesel engines, the emissions of CO and HC will not be reviewed in this research. Following are brief reviews to the emissions of PM and NO_x, the two problematic pollutants of diesel engines, and the effect of H_2 addition on the brake thermal efficiency. The emissions of the unburned H_2 raise both fuel economy and safety concerns will also be reviewed.

2.3.1 NO_x Emissions

NO_x is formed under high combustion temperature with excess O_2 present with N_2. In principle, NO_x of combustion devices burning N-free fuels can be formed through 1) Thermal NO_x mechanism; 2) Prompt NO_x mechanism; and 3) N_2O intermediate mechanism. When operated under medium to high load, the combustion temperature of diesel engines is usually higher than the threshold value for NO_x to be formed through thermal NO_x mechanism. Correspondingly, NO_x emissions of diesel engines are dominated by the combustion temperature when operated at medium to high load. In comparison, the N_2O intermediate mechanism makes significant contribution to NO_x emissions under low load operation when the combustion temperature is low accompanied by the presence of large amounts of O_2 and unburned HC. However, prompt NO_x is usually formed in the unburned mixture prior to the flame front of a fuel-rich mixture. Its contribution to NO_x emissions of a diesel engine is usually very small. It is well accepted that combustion temperature is one of the main parameters that dominates the formation of NO_x in diesel engines. In general, the approaches that tend to decrease the temperature of the combustion products usually reduce the formation of NO_x. For example, the application of cooled EGR helps to reduce NO_x emissions benefiting from the deteriorated combustion process and higher heating capacity of the CO_2 contained in the exhaust gases recycled, both decrease the temperature of the combustion products.

The addition of H_2 to diesel engines was shown to lengthen the ignition delay and retard the combustion phasing beyond top dead center, which tended to decrease the combustion temperature. Many of the H_2 research projects expressed the wishes that the addition of H_2 to diesel engines should reduce NO_x emissions. However, the addition of H_2 also enhance the combustion process and reduce the combustion duration once the H_2 combustion is initiated, which has the potential to increase the temperature of the combustion products and increase NO_x emissions. For example, Varde and Frame [1983] reported some of the earlier work in examining the effect of H_2 addition on exhaust emissions using a single cylinder diesel engine with a rated power of 5 kW. A chemiluminescent NO-NO_x analyzer was used to measure the emissions of NO_x. The addition of H_2 to this single cylinder diesel engine was reported to increase substantially the emissions of NO_x. When operated under full load, the emissions of NO_x increased almost linearly from 900 ppm for pure diesel operation to 1150 ppm when 15% of energy was provided by H_2. For 82% load operation, the NO_x emissions increased from 750 ppm linearly to 950 ppm when 17% of diesel fuel was substituted by H_2. The increase in NO_x emissions could be due to the following reasons: (1) The increased combustion temperature due to the burning of H_2 with a higher adiabatic temperature; (2) Increased premixed combustion ratio resulted from longer ignition delay; (3) Reduced combustion period due to the significant increase in premixed combustion ratio; (4) Increased energy release rate due to the spontaneous combustion of H_2 and diesel fuel; (5) Increased water emissions that increases the NO_x level measured on dry basis, which may affect the NO_x emission values reported in ppm (dry).

Kumar, et al. [2003] investigated the effect of H_2 addition on NO_x emissions of a single cylinder, four stroke, water-cooled diesel engine with a rated power of 3.7 kW at 1500 rpm. For both standard diesel and Jatropha oil as the main fuel, the addition of H_2 increased the NO_x emissions when operated under high load. In comparison, the addition of H_2 showed negligible effects on NO_x emissions when operated at low to medium load (<60%). This was consistent with the NO_x emissions obtained in a 0.64 liter, single cylinder, direct-injection, diesel engine [Tomita et al., 2001]. In the experiment, the overall equivalence ratio was kept constant while adding H_2 to the intake manifold to substitute up to 80% of the diesel fuel. When operated low to medium load operation with overall equivalence ratio value of 0.3 and 0.4, respectively, substitution of up to 70% diesel with H_2 was found to have negligible effect on the emissions of NO_x. When operated at overall equivalence ratio of 0.5 (about 60% load), the substitution of diesel fuels with H_2 increased the NO_x emissions from 200 ppm for pure diesel operation to about 450 ppm. It seems that engine operating load plays an important role in determining the effect of H_2 addition on NO_x emissions. Saravanan and Nagarajan [2008] reported a different trend in examining the effect of engine load and H_2 addition on NO_x emissions in a single cylinder, four-stroke, diesel engine with a rated power of 3.7 kW. The addition of 10% and 20% H_2 into a diesel engine was found to increase NO_x emissions corresponding to the overall load range examined. However, the NO_x emissions were found to decrease with a further increase in the amount of diesel fuel substituted by H_2 (>30% H_2 substitution). A minimum NO_x level of 575 ppm was achieved with 90% H_2 addition at 60% engine load.

Recently, numerous research projects were conducted using light duty multi-cylinder diesel engines with modern engine technologies. For example, Bika, et al. [2008] examined the effect of H_2 addition on NO_x emissions of a 1999 Volkswagen 4 cylinder, 1.9 liter, turbo-charged, direct injection diesel engine. The addition of H_2 for the substitution of up to 40% diesel fuel was reported to have negligible effect on NO_x emissions. Shirk, et al. [2008] examined the effect of H_2 addition on exhaust emissions of a 1.3 L, turbo-charged, light duty diesel engine with a common rail fuel system. The mixture of 20% bio-diesel with 80% petroleum derived diesel fuel was used as the main fuel. When installed in an engine test cell, eight steady state operating points were tested with H_2 flow rate equivalent to 0%, 5% and 10% of the total fuel energy. The emissions of NO_x decreased slightly with the addition of H_2 while its effect on efficiency was small. This engine was also installed in a 2005 Chevrolet Equinox and tested on a chassis dynamometer following the UDDS and 505 urban driving cycles. The addition of 5% H_2 was shown to have a negligible effect on NO_x emission rates. However, the substitution of 10% diesel by H_2 reduced the NO_x emission rate from 0.75 g/mile to 0.6 g/mile, a 20% reduction compared to pure diesel operation.

EGR is widely used in modern diesel engines to reduce the engine-out NO_x emissions by reducing the temperature of combustion. However, its application usually increase the emission of PM. Benefiting from carbon-free fuel properties of H_2, the addition of H_2 into diesel engine with EGR may extend the application range of EGR system without raising the issues of PM emissions. For example, McWilliam, et al. [2007] investigated the effects of H_2 addition and EGR application on the combustion and emissions of a modern 2 liter, turbo-charged, 4 cylinder, 4 stroke diesel engine equipped with high pressure common rail fuel injection system and cooled EGR. The effects of engine load, EGR rate and H_2 addition on exhaust emissions were experimentally investigated. The addition of H_2 to this modern diesel engine was found to increase NO_x emissions especially for low load operation without EGR. As shown in Fig. 1, the diesel engine operation at 2.7 bar without EGR produced 6.2 g/kW-hr of NO_x. With the addition of H_2 without EGR, the emissions of NO_x increased almost linearly to 19.2 g/kW-hr with the addition of up to 6% H_2 into the intake air. The author explained this NO_x increase by the following reasons: (1) the more pronounced premixed combustion due to the increased ignition delay; (2) the reduced combustion periods due to the presence of H_2 which enhanced the combustion process once ignited; (3) the combustion temperature of H_2 is higher than that of diesel fuels. In comparison, the application of EGR helped to weaken the hydrogen's enhancing effect on NO_x emissions. When 10% EGR was applied, the addition of 6% H_2 produced NO_x emissions at a rate of 12 g/kW-hr, which is about 2.2 times of that for pure diesel engine operation. As shown in Fig. 1, the extent of increase in NO_x emissions was weakened when operated at higher load with the application of EGR. When operated under 5.4 bar BMEP with 10% EGR, the NO_x emissions increased almost linearly from 2.0 g/kW-hr with pure diesel operation to about 4.8 g/kW-hr when 6% H_2 was mixed into the intake air, which is only slightly higher than that of pure diesel operation. With the help of EGR, it was possible to obtain lower NO_x emissions even with the addition of H_2. With the help of H_2 as supplemental fuel, it may be possible to develop suitable EGR strategies that can reduce both NO_x and PM emissions while overcoming the penalty in fuel economies associated with EGR.

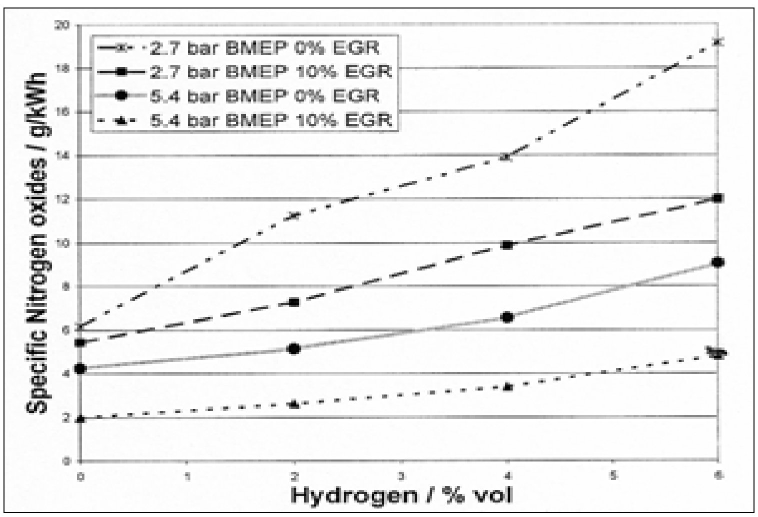

Figure **1** Effect of H_2 Addition, Engine Load and EGR on NO_x Emissions [McWIlliam, et al., 2008]

2.3.2 PM Emissions

The emissions of PM have continued to be one of the main problematic pollutants associated with diesel engines since the stringent PM emission regulations were implemented. PM is formed in diesel engines under fuel-rich conditions presented under high temperature and pressure. In diesel engines, PM is usually formed during the diffusion combustion stage especially resourced from the portion of diesel fuel injected at the end of injection process when the combustion has been initiated. The approaches that help to enhance premixed combustion and reduce diffusion combustion usually reduce the emissions of PM. The addition of H_2 into diesel engines has been demonstrated to lengthen the ignition delay, which may increase the amount of heat released during the premixed combustion stage. Plus, the burning of less diesel fuel also helps to reduce the opportunity for PM to be formed during diesel combustion.

In the early decades of H_2-diesel dual fuel engine research, the level of PM emissions were qualitatively measured using simple smoke meters such as the devices developed by AVL and Bosch, which reported the Filter Smoke Number (FSN) or Bosch Smoke Number (BSN). For example, Varde and Frame [1983] examined the effect of H_2 substitution to diesel fuels on PM emissions using a Bosch Smoke Meter. The addition of a relatively small amount of H_2 was shown to reduce the PM emissions. Further increasing the H_2 flow rate beyond that to obtain minimum BSN, was shown to produce more PM as represented by the increased BSN number. The author believed that this was due to the displacement of air by H_2, which reduced the overall air/fuel ratio of the diesel fuel and increased formation of PM during combustion. Tomita, et al. [2001] examined the PM emissions of a H_2-diesel dual fuel engine using a Bosch type smoke meter (Zexel, DSM10). The addition of H_2 to this small diesel engine reduced the emissions of PM especially when operating at medium to high load (about 60% load). Recently, some research has been conducted using multi-cylinder diesel engines. For example, McWilliam, et al. [2008] examined the effect of H_2 addition on the PM emissions through measuring variation of FSN of a turbo-charged light duty diesel engine. The addition of H_2 to this diesel engine was found to significantly reduce PM emissions especially when operated at high load without EGR.

The current emissions regulation requires reporting the quantitative data of PM emissions. Recently, there is increasing interest to quantitatively measure the PM emissions of dual fuel diesel engine with H_2 supplemented into intake mixture. For example, Saravanan and Nagarajan [2008] characterized the PM emissions of a single cylinder H_2-diesel dual fuel engine. The PM emissions were reported both qualitatively in BSN and quantitatively in g/kW-hr without describing the measuring approach. The addition of H_2 to the diesel engine was shown to substantially reduce the emissions of PM. For example, the substitution of 90% diesel fuel with H_2 was found to reduce the emissions of PM by 70%. Masood, et al. [2007] examined the effect of compression ratio and H_2 substitution ratio on PM emissions at full load operation using a single cylinder diesel engine with a rated power of 3.7 kW. The PM emissions rated as g/m^3 were reported without describing the measurement method. When operated a compression ratio of 18.35, the substitution of 90% diesel by H_2 reduced PM emissions by 92.7%.

Some research scale PM characterization devices were also used to explore the effect of H_2 addition on PM emissions. For example, Bika, et al. [2008] examined the effect of H_2 addition on PM emissions of a 1.9 Liter Volkswagen TDI engine while using ultra low sulfur diesel and soy methyl ether (SME) bio-diesel. The PM emissions were measured using a Scanning Mobility Particle Sizer (SMPS). The amount of H_2 added varied from 10%-40% evaluated on an energy basis. For all load conditions examined, the addition of H_2 into this light-duty diesel engine was shown to reduce the PM emissions ranging from 10% to 50% in both total PM concentration and total PM number. The PM concentration and soot diameter distribution were used to calculate the PM mass flow rate using the assumed density of 1 g/cm^3 of the spherical soot particles. The addition of H_2 to the diesel engine reduced the PM mass emissions beyond the expected reduction resulting from the reduced diesel flow.

2.3.3 Brake Thermal Efficiency

Thermal efficiency is one of the main parameters used to evaluate the performance of I.C. engines. Most of the research on H_2-diesel dual fuel engines in the earlier stage focused on the improvement of the brake thermal efficiency. For example, Varde and Frame [1983] examined the effect of H_2 addition on the brake thermal efficiency of a small single cylinder diesel engine. The addition of H_2 improved substantially the brake thermal efficiency. When operated at full load, the brake thermal efficiency increased from 30.53% for pure diesel operation to 33.68% for dual fuel operation when 12.5% of the energy was supplied by H_2. However, the addition of a small portion of H_2 (<5% of total energy) was found to reduce the brake thermal efficiency. It was believed that this was due to the extremely lean H_2-air mixture, which could not support the flame propagation and resulted in low combustion efficiency of H_2. As an example, the equivalence ratios of the premixed H_2-air mixture at these fueling rates were extremely low, typically having an equivalence ratio of approximately 0.03, which caused H_2 to burn in a very erratic manner. This was further confirmed by the experimental results obtained when H_2 was added at a rate of 0.65 kJ/s. The engine efficiency was consistently lower than pure diesel operation. With an increased H_2 flow rate to 1.65 kJ/s, the addition of H_2 was found to improve the brake thermal efficiency.

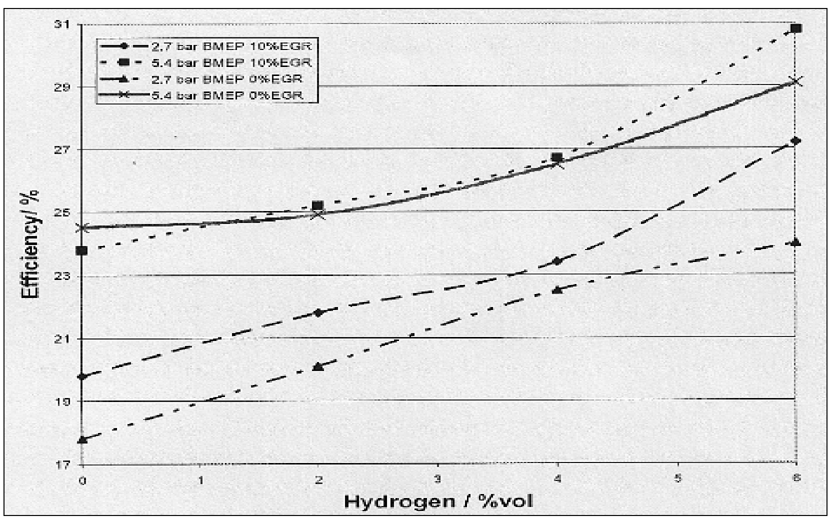

Figure 2 Effect of H_2 Addition, Engine Load and EGR Rate on Fuel Conversion Efficiency [McWilliam, et al., 2008]

Recently, McWilliam, et al. [2008] investigated the effect of H_2 addition on the thermal efficiency of a light duty diesel engine modified for H_2-diesel dual fuel operation. As shown in Fig. 2, the addition of H_2 to this multi-cylinder diesel engine improved substantially the thermal efficiency. As shown in Table 1 for operation at 5.4 bar BMEP with 10% EGR, the addition of 6% H_2 into the intake mixture improved the thermal efficiency from 23.81% for pure diesel operation to 30.76%, an improvement of 29.2%. When operated at 2.7 bar BMEP, the brake thermal efficiency was increased by 37.39% when the intake mixture contained 6% H_2. Due to the lack of experimental data of H_2 supplementation less than 2%, the possible reduction in brake thermal efficiency with the addition of a small amount of H_2 was not observed.

Table **1** Effect of H_2 Addition on Brake Thermal Efficiency [McWilliam, et al., 2008]

5.4 bar BMEP			2.7 bar BMEP		
[H_2] in intake, vol.%	Thermal Efficiency	Thermal efficiency improvement compared to that of pure diesel, %	[H_2] in intake, vol.%	Thermal Efficiency	Thermal efficiency improvement compared to that of pure diesel, %
0	23.81%	0%	0	19.80%	0%
2	25.19%	5.78%	2	21.79%	10.07%
4	26.66%	11.96%	4	23.41%	18.22%
6	30.76%	29.20%	6	27.20%	37.39%

2.3.4 Hydrogen Emissions

When supplied to diesel engines, H_2 is usually mixed with air in the intake manifold to form a homogeneous mixture prior the injection of the pilot diesel. When supplied at a small flow rate, the H_2-air mixture is usually too lean to support a healthy propagating flame. Without the presence of diesel fuel, the H_2 can only be burned through auto-ignition or left as unburned H_2 due to the high temperature needed for H_2 to auto-ignite. After pilot diesel fuel is injected into the combustion chamber, the H_2 present in the plume of the diesel spray may be burned completely if mixed well with diesel fuel. However, the H_2 present outside of the diesel spray plume may survive the main combustion process due to the lack of a propagating flame and also the low bulk mixture temperature. The H_2 exiting from crevices after the main combustion process may also survive without releasing any chemical energy and making contribution to the production of power. The survived H_2 then exits the diesel engines as unburned gaseous fuel without participating in the combustion process.

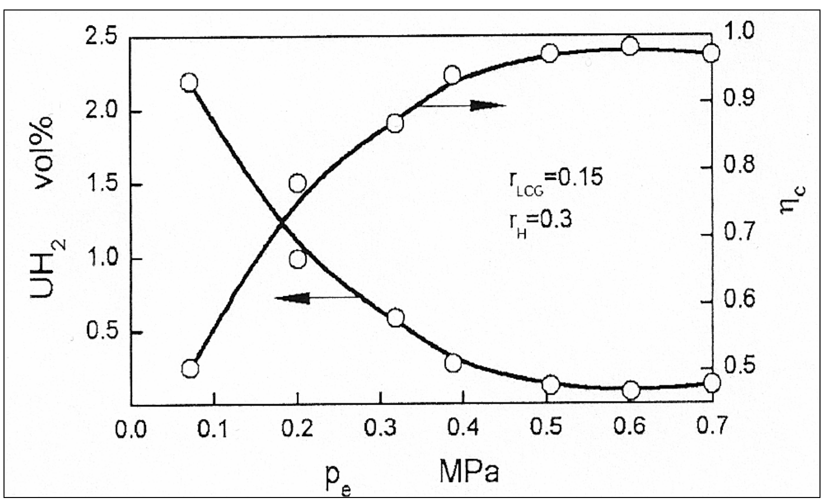

Figure 3 Effect of Engine Load on the Emissions of H_2 and Its Combustion Efficiency [Mohammadia, et al., 2007]

A number of researchers examined the emissions of unburned H_2 of dual fuel engines supplemented by H_2 or H_2-rich gaseous fuel mixtures. For example, Mohammadia, et al. [2007] investigated the combustion and emissions characteristics of a dual fuel diesel engine supplemented with low calorific gaseous fuels consisting of H_2 and N_2. The emissions of unburned H_2 (UH_2) were measured and used to calculate the combustion efficiency of H_2. As shown in Fig. 3, the reduction of engine load below 0.4 MPa BMEP reduced substantially the combustion efficiency of H_2. When operated at 0.2 MPa, about 22% of the H_2 exited the engine without participating in the combustion process. When operated at 0.08 MPa, about 50% of H_2 exited the engine as unburned fuel without attending the combustion process. It was evident that the substitution of H_2 to diesel fuel was not preferred at low load operation.

Abu-Jrai, et al. [2007] examined the emissions of unburned H_2 of a diesel engine supplemented with reformed exhaust gas recirculation (REGR). Table 2 summarizes the emissions of unburned H_2 and CO. With the addition of 7.5% H_2 into the intake mixture, the dry exhaust gas contained 1.53% and 0.73% H_2 when operated at 2.8 bar and 4.0 bar imep, respectively. When both H_2 (4.5%) and CO (3.0%) were supplemented into the intake mixture, the emissions of H_2 was about 1.21% and 0.6% for 2.8 and 4.0 bar, respectively. In comparison, the emissions of CO were about 0.6% and 0.47%, respectively. The emissions of unburned H_2 and CO reflect the low combustion efficiency of gaseous fuels supplemented, which reduced the overall thermal efficiency of the dual fuel engine. At low load operation, the substitution of diesel with H_2 or CO resulted in their insufficient combustion.

Table **2** Exhaust Emissions of Unburned H_2 and CO of Dual Fuel Diesel Engine, [Abu-Jrai, et al. 2007]

Load, imep	[H_2] in intake mixture	[CO] in intake mixture	[H_2] in Exhaust	[CO] in Exhaust	[H_2]/[CO] (intake)	[H_2]/[CO] (Exhaust)
2.8 bar	7.5%	0%	1.53%	NA	NA	NA
4.0 bar	7.5%	0%	0.73%	NA	NA	NA
2.8 bar	4.5%	3.0%	1.21%	1.05%	1.5	1.15
4.0 bar	4.5%	3.0%	0.6%	0.47%	1.5	1.28

Tsolakis, et al. [2005] investigated the emissions of unburned H_2 and CO of a single cylinder dual fuel diesel engine supplemented with reformed EGR, a simulated syngas containing H_2 and CO. For the cases of engine operation on diesel only and dual operation with reformed exhaust gas recirculation (REGR) that contained H_2 only, the CO concentration in the exhaust gas was not affected by the engine load and the addition of REGR. However, the addition of the REGR containing 10% CO increased substantially the emissions of CO. It was believed that the extra CO emissions were due to the insufficient combustion of gaseous CO fuel supplemented into intake mixture. The CO concentration in the bulk mixture was too low to reach the lean flammability limit. The CO away from the diesel spray plume cannot be burned due to the lack of a propagating flame. This was further confirmed by examining the variation of CO emissions with engine load. The increase in engine load reduced the emissions of CO benefiting from the increased diesel flow rate. The emissions of unburned H_2 were also examined. As shown in Table 3, H_2 emissions follow a similar trend to that of CO emissions. It should be noted that CO emissions originate from both CO fuel in the intake mixture and that formed during oxidation of the diesel fuel. It was also noted that the ratio of H_2 over CO in the exhaust gas was lower than that in intake mixture showing that the combustion efficiency of H_2 was higher than that of CO.

Table 3 Exhaust Emissions of Unburned H_2 and CO of Dual Fuel Diesel Engine, [Tsolakis, et al. 2005]

Reformed EGR	Load	[H_2] in intake mixture	[CO] in intake mixture	H_2 emissions (dry)	CO emissions (dry)	[H_2]/[CO] (intake)	[H_2]/[CO] (Exh.)
REGR Rate 20%	25%	3.0%	2.0%	1.22%	1.01%	1.5	1.21
	50%	3.0%	2.0%	0.67%	0.57%	1.5	1.17
	75%	3.0%	2.0%	0.25%	0.24%	1.5	1.05
REGR Rate 10%	25%	1.5%	1.0%	0.57%	0.51%	1.5	1.12
	50%	1.5%	1.0%	0.29%	0.27%	1.5	1.07
	75%	1.5%	1.0%	0.19%	0.21%	1.5	0.90

2.4 Summary

Based on the review of the literature, hydrogen is usually burned in a diesel engine by dual fuel mode using pilot diesel to ignite the H_2-air mixture. Extensive research has been conducted to investigate the effect of H_2 supplementation on the engine performance, combustion process, and exhaust emissions of H_2-diesel dual fuel engines. The earlier research was conducted using single cylinder diesel engines employing old engine technologies. More recently, a number of projects were conducted to examine the emission characteristics of a light-duty multi-cylinder diesel engine. The corresponding research on heavy-duty H_2-diesel dual fuel engines has not been reported.

The addition of H_2 to a diesel engine can significantly reduce PM emissions when operated at low and medium load. When operated at full load, the PM emissions may increase due to the reduction in O_2 concentration resulted from H_2 induction into air. When operated under medium to high load, the addition of H_2 into the diesel engine was shown to increase NO_x emissions. However, some literature reported that NO_x emissions can be reduced by H_2 addition at some conditions associated with low load operation.

The PM emissions were qualitatively measured using simple smoke meters such as the devices developed by AVL and Bosch, which reported the Filter Smoke Number (FSN) or Bosch Smoke Number (BSN). The emissions of NO_x were usually reported in ppm. There is a need to examine the effect of H_2 addition on PM and NO_x emissions using well accepted measuring approaches and quantitatively report the specific emissions data in g/bhp-hr.

The exhaust emissions of unburned H_2 could be a potential issue for safety and economic reasons. Limited experimental data have been reported in the literature and demonstrated the low combustion efficiency of H_2 at low load operation. Detailed research of H_2 emissions and also its combustion efficiency needs to be examined.

3 Experimental Setup and H_2 Fuel System

This research investigated the performance, combustion and emission characteristics of two turbocharged heavy-duty H_2-diesel dual fuel engines. The effects of the addition of H_2, engine load, engine speed, and diesel fuel flow rate on the brake thermal efficiency, cylinder pressure, combustion process, and exhaust emissions of NO_x, PM, CO, HC, and unburned H_2 were investigated. The engine load was varied from 10% to 100% with the addition of 0~7.5% H_2 into the intake mixture. The emissions of baseline engines and also that with the addition of 2% and 4% H_2 were measured using the steady state 13-Mode ESC cycle.

3.1 Engines and Dynamometer

The diesel engines used in this research included a 1999 Cummins ISM370 engine and a 2004 Mack MD11 engine. The 1999 Cummins ISM370 was a turbocharged, 6 cylinder heavy-duty diesel engine without an EGR system. It was certified by the EPA as having emissions at or below 4.0 g/bhp-hr NO_x and 0.10 g/bhp-hr PM. The 2004 Mack MD11 was a turbocharged, 6 cylinder heavy-duty diesel engine with cooled EGR system. It was certified by the EPA as having emissions at or below 2.5 g/bhp-hr NO_x and 0.10 g/bhp-hr PM. As shown in Fig. 4, these two engines had similar power out. Both were typical representatives of heavy-duty diesel engines for a class-8 heavy-duty truck. Detailed specifications of both engines can be found in Table 4.

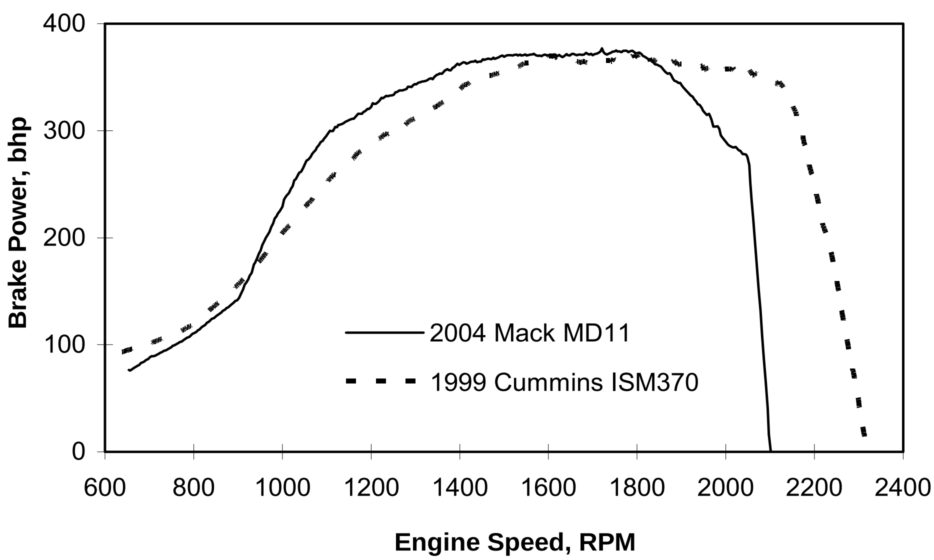

Figure 4 Comparison of Engine Map of the 2004 Mack MD11 and 1999 Cummins ISM370

The test engine was mounted to a 550 hp General Electric DC dynamometer, which was used to absorb engine load and control engine speed. Additionally, the engine was instrumented for the measurement of manifold air pressure, air intake restriction, total exhaust backpressure, manifold intake temperature, coolant temperature, oil temperature, and exhaust temperature according to CFR 40 Part 86 requirements. Figure 5 shows the engine and dynamometer facility used in this research.

Table 4 The Specifications of the Test Engines.

Engine Manufacturer	Mack	Cummins
Engine Model	MD11	ISM370
Model Year	2004	1999
Displacement	10.8 Liter	10.8 Liter
Power Rating	355 bhp @ 1800 rpm	370 bhp @ 2100 rpm
Torque Rating	1360 ft-lbf @ 1200 rpm	1350 ft-lbf @ 1200 rpm
Configuration	Inline 6-cylinder	Inline 6-cylinder
Bore x Stroke	4.84 in x 5.98 in	4.92 in x 5.79 in
Induction	Turbocharged with in-house Aftercooler	Turbocharged with in-house Aftercooler
Fuel Type	Diesel	Diesel
Engine Strokes per Cycle	4	4
Injection	Direct Injection, Electronic	Direct Injection, Electronic
Cooled EGR	Yes	No
VGT	Yes	No

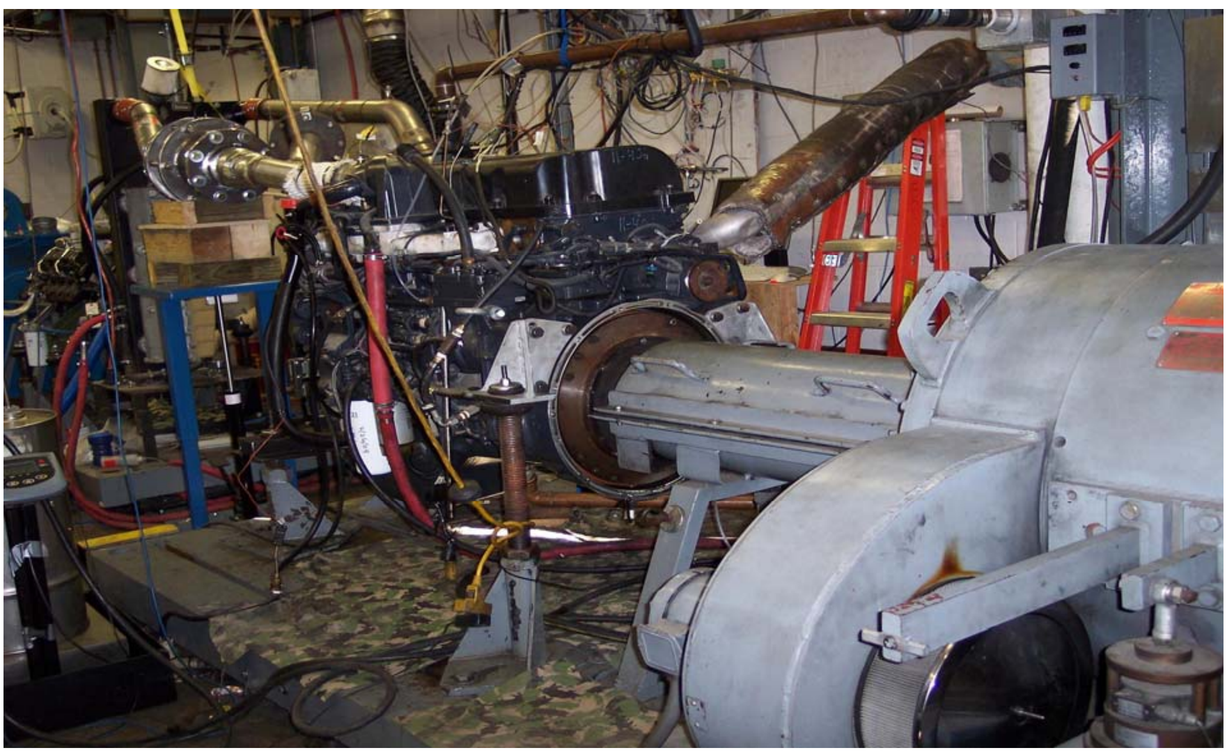

Figure 5 2004 Mack MD11 Diesel Engine Mounted to 550 hp DC Dynamometer

3.2 Emission Measurement

Engine exhaust was ducted to a full-scale dilution tunnel (18 inches in diameter, 20 feet in length) with flow control accomplished using a critical flow venturi-constant volume sampler (CFV-CVS) system. A 10-inch diameter orifice, located 3 feet from the tunnel entrance, ensured that exhaust was thoroughly mixed with dilution air before it reached the emissions sampling zone, located ten tunnel diameters downstream. The CFV-CVS controlled the dilute exhaust flow at a nominal 2400 scfm throughout the test program. Instantaneous measurement of CFV-CVS flow was accomplished using a fast-response thermocouple and pressure transducer at the venturi entrance. Dilute exhaust samples were drawn from the tunnel using heated sample probes and lines to individual emissions analyzers. Dilute exhaust was also drawn from the dilution tunnel sample plane through a series of two 70mm filters to obtain particulate matter samples. Temperature and humidity of the engine and dilution tunnel intake air was controlled to 68-86°F and 50+/-5% for the duration of the testing. To minimize the contribution of background particulate matter on dilute exhaust PM measurements, dilution air was drawn through HEPA filters located in the intake conditioning system. These HEPA filters, each at a 2400 scfm capacity, were installed in parallel to provide up to 4800 scfm capacity.

Table 5 Exhaust Gas Analyzer Specifications

Exhaust Gas Constituent	**Measurement Technology**	**Manufacturer**	**Model**
Oxide of Nitrogen (NO_x)	Chemiluminescence	Eco Physics	CLD822CMH
Nitric Oxide (NO)	Chemiluminescence	Eco Physics	CLD822CMH
Total Hydrocarbon (THC)	Flame Ionization Detection	California Analytical Instruments	600M HFID
Carbon Monoxide (CO)	Non-dispersive infrared	Horiba	AIA-210
Carbon Dioxide (CO_2)	Non-dispersive infrared	Horiba	AIA-210
Particulate Matter (PM)	Dilution Tunnel	PM sampling according to CFR 40 Part 86	N.A.
Hydrogen (H_2)	Electron Pulse Ionization (EPI) Mass Spectrometer	V & F LCC	H-Sense

The gas analysis bench was equipped with exhaust sample conditioning and analysis systems following CFR 40 Part 86 requirements. Regulated emissions of HC, CO, NO_x, NO, PM, and CO_2 were measured and reported in brake-specific mass units along with brake-specific fuel consumption. For the 13 mode tests, an integrated PM sample was obtained. Measurement of CO and CO_2 were performed using non-dispersive infrared (NDIR) analyzers, NO_x and NO measured using a wet chemiluminescence analyzer, and HC was measured using a heated flame ionization detector (HFID). H_2 was measured using an H-sense Electron Pulse Ionization (EPI) Mass Spectrometer (MS). Data from the exhaust analyzers, sampling trains, double dilution tunnel, and the engine were acquired and archived at a rate of 1 Hz. Calibration procedures and intervals were followed according to CFR 40 Part 86 requirements. A laboratory checkout following the procedures listed in CFR 40 Part 86 was performed prior to the collection of the data.

PM was measured using a proportional sampling system. Dilute exhaust was drawn from the full flow dilution tunnel into a stainless steel 4-inch diameter by 30-inch long secondary dilution tunnel and, in turn, was drawn through a stainless steel filter holder that contained two Pallflex 70mm diameter Model T60A20 fluorocarbon-coated glass microfiber filters in series. The sample stream was maintained at temperatures below 125 ºF as measured at the inlet of the TPM filter holder. Secondary dilution air, which can be used to lower the temperature of the dilute sample upstream of the sample filters, was not required during this evaluation. Prior to weighing (both before and after PM measurement), sample filters were conditioned in an environmentally controlled room to a nominal 22 ºC dry bulb, 9.5 ºC dew point, and 45% relative humidity, in compliance with requirements presented in CFR 40 Part 86. Filters were weighed using a Sartorius microbalance in the same controlled environment.

3.3 Cylinder Pressure Measurement and Combustion Process Analysis

In-cylinder pressure was acquired from the 2004 Mack MD11 engine and the 1999 Cummins ISM370 engine using a Kistler model 6125B and a Kistler Model 6125C pressure transducer, respectively, with a Kistler model 5010B charge amplifier. The in-cylinder pressure was acquired from one cylinder (cylinder 1 for the Mack engine and cylinder 6 for the Cummins engine) at 0.25 degree crank angle (CA) resolution. At the end of each emission sampling period, 200 consecutive pressure curves were collected and averaged. A low-pass filter was applied to the measured in-cylinder pressure to minimize fluctuations caused by the pressure wave and reduce the high frequency combustion noise. The cut-off frequencies applied to the in-cylinder pressure of the 2004 Mack MD11 engine and 1999 Cummins ISM 370 were 2000Hz and 2500Hz, respectively. The dynamic pressure of the in-cylinder pressure transducer was referenced using a constant polytropic coefficient since the measurement of manifold air pressure (MAP) was not required [Brunt and Pond, 1997]. The combustion characteristics (gross indicated mean effective pressure, mean cylinder gas temperature, etc.) were solved using methods reported in the literature [Brunt and Pond, 1997; Brunt and Platts, 1999 and Brunt and Emtage, 1997]. The mean cylinder gas temperature was solved using the ideal gas law ($T = PV/mR$) using the in-cylinder pressure and cylinder volume at the intake valve closing as a reference condition ($T = PV*(T_{IVC}/P_{IVC}*V_{IVC})$).

The approach used in this paper to extract heat release information from the engines involved the use of the well-known traditional single zone zero-dimensional heat release model [Heywood, 1988]. While spatial characteristics internal to the cylinder were not studied, conclusions on engine performance and emissions may still be drawn on the spatially averaged burn rate. Utilizing the first law of thermodynamics and assuming a uniform pressure, uniform temperature, and ideal gas with the substitution of the specific heat ratio (γ) and the substitution of crank angle for time reduces to an expression for the gross heat release rate. The well-known Woshni [1967] equation was used to calculate the heat loss from the bulk mixture temperature to coolant.

3.4 *Hydrogen Fuel System and Safety Measures*

3.4.1 Hydrogen Fuel System

Figure 6 shows the schematic diagram of the H_2 fuel system, which can be divided into two sub-systems: (a) up-stream H_2 fuel system from fuel tank to emergency valve A as shown also in Fig. 7; and (b) down-steam H_2 fuel system from emergency valve B to intake manifold, as shown in Fig. 8. As shown in Figs. 6 and 7, up-stream H_2 fuel system consisted of fuel tanks, H_2 fuel module, which consisted of two set of pressure regulators each connected with 6 H_2 tanks with maximum pressure of 2000 Psi, emergency valve (A) for the shutdown of H_2 fuel in case of emergency. The H_2 module was also equipped with safety valves stopping the back flow of H_2 from H_2 module to the H_2 tank. The H_2 fuel cannot flow from high pressure H_2 tank to low pressure one. The pressure regulator was used to reduce and regulate H_2 fuel to a constant pressure. The main function of the up-stream H_2 system was to continuously provide H_2 flow under constant pressure and shut down the H_2 supply in case of an emergency.

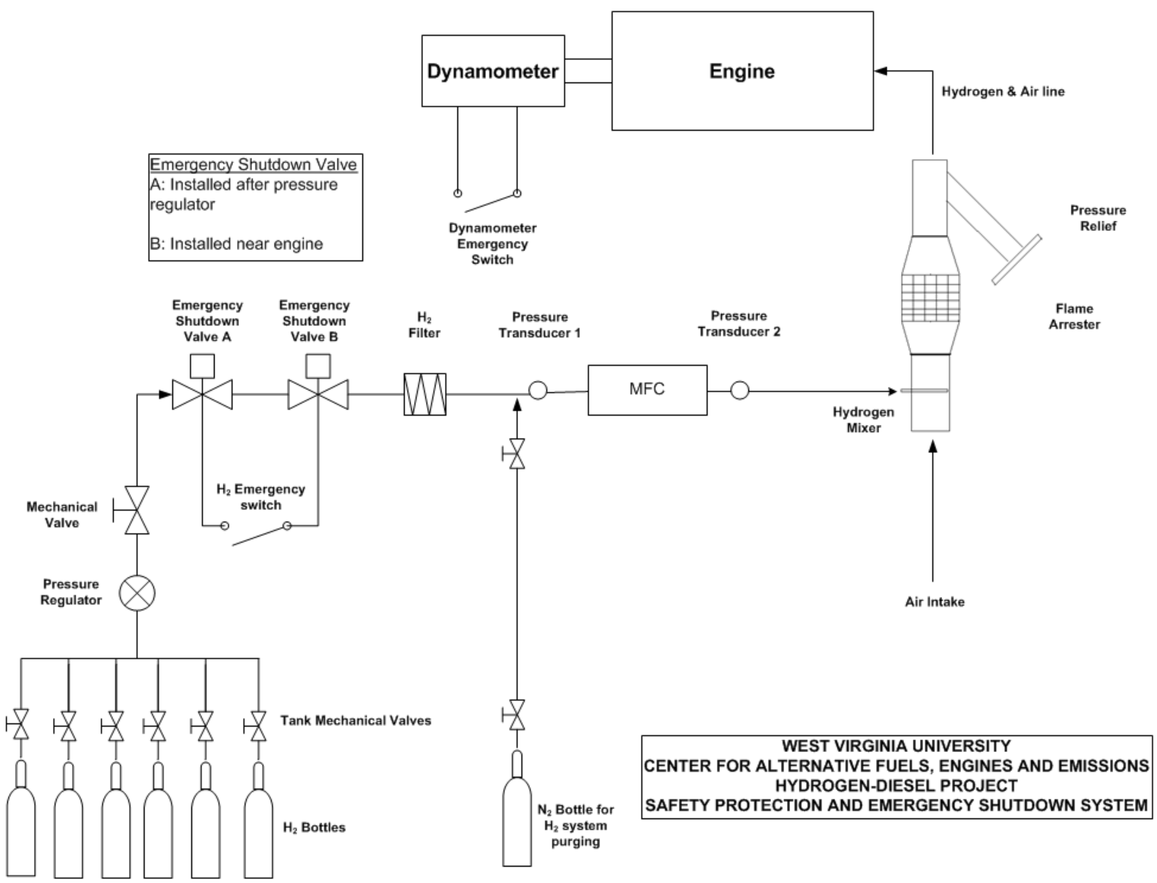

Figure 6 Schematic Diagram of H_2 Fuel System

Figure 7 Hydrogen Station with Pressure Regulation Module and Emergency Valve A

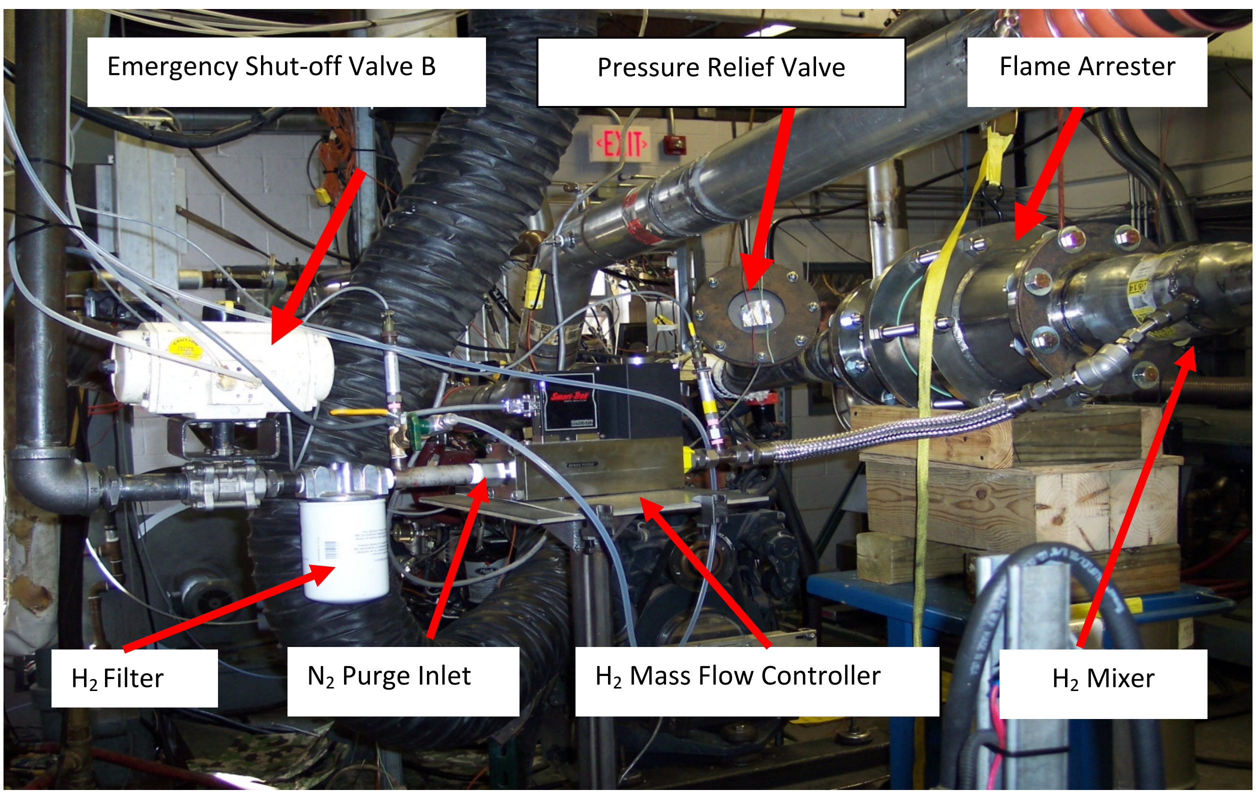

Figure 8 Hydrogen Fuel System Safety Approaches

As shown in Figs. 6 and 8, the down-stream H_2 fuel system consisted of a emergency shutdown valve (B), filter, purging valve, mass flow controller, and H_2 mixer. The main function of the down-stream H_2 system was to deliver the accurate amount of H_2 into engine intake system with proper approach to eliminate back-fire hazards. A fuel mixer, manufactured in-house as shown in Fig. 9, was installed about 0.5 feet prior to the flame arrester. The large volume of the flame arrester and intake pipe line from the flame arrester to the intake manifold allow for the good mixing of H_2 with the fresh intake air. A flame arrestor was installed in the intake system to quench the propagation of the H_2 flame initiated by backfire. A pressure relief valve was installed to release the pressure established in the intake system in case a backfire occurred.

Figure 9 In-House H_2 Mixer

3.4.2 Safety Features

Safety is one the major concerns when a large amount of H_2 is used in the research laboratory. The research team identified the following safety concerns associated with H_2 application in the engine research laboratory: (1) the leakage of H_2 into engine laboratory; (2) better ventilation to eliminate the accumulation of H_2 in the laboratory especially the ceiling area; (3) the occurrence of backfire and relief of pressure; (4) shut down of H_2 fuel system in case of emergency, and (5) the avoidance of over-dosing of H_2 flow, which may result in abnormal combustion such as back fire and the onset of knock. In this research, the following safety approaches were developed and implemented during the test.

Leakage Test: (a) Detailed leakage test was conducted using soap bubbles approach after the system was built, first a few day of testing and at least once a week during the testing period; (b) Leakage test after system modification, change of mass flow controller and suspected leakage after leakage test using pressure drop approach; (c) leakage test for connector and high pressure H_2 line after the switching of H_2 tanks; (d) the leakage check using simple pressure drop methods prior to the purging of the H_2 fuel system, which is filled with pure N_2 after the finishing of the previous test.

Better Ventilation: The H_2 fuel module from the H_2 tank to emergency valve A was installed in a shed as shown in Fig. 7. This shed was located outside the laboratory and open to atmosphere. Any H_2 leaked from this system would be ventilated into air without accumulation around the H_2 fuel system. The ventilation of the laboratory was maintained by turning on the ceiling ventilation fan in the lab prior to the start of testing. The ceiling fan was kept running for at least 20 minutes after the finishing of the test.

Purging of the H_2 Fuel System with Pure N_2 after Each Test: After the experiment was finished, the H_2 system from the mass flow controller to the H_2 regulator was purged with pure N_2. After purging, the H_2 fuel system from pressure regulator to mass flow controller was filled with pressurized N_2. It was believed that such purging was necessary to eliminate the potential H_2 leaking source when the engine was not running. The pressurization of the fuel system with N_2 was also considered as a safety procedure to eliminate the possibility for air entering the H_2 fuel system. Prior to the second day of testing, the H_2 system was purged with H_2 with the engine operated at 30% load with 4% H_2 addition for at least 10 minutes.

Elimination of Backfire Damage: The damage of backfire was eliminated through the implementation of the following approaches:

(1) Design of the test matrix to avoid the addition of H_2 at high concentration which may lead to the occurrence of backfire. In this research, the addition of H_2 was limited to 7.5% for low load operation, 6% for 70% load and 5% for 100% load.

(2) The relief of the high pressure established in the intake system once backfire occurs: a pressure relief valve was installed between the intake manifold and flame arrestor. The relief valve would blow off once the intake manifold pressure reached 50 psi for any reason. This value was demonstrated to be effective in eliminating the backfire hazards. During this research, the backfire occurred three times without causing any damage to the intake system other than blowing off the pressure relief valve. The installation of the pressure relief valve is shown in Fig. 6 and 8.

(3) To minimize the amount of H_2-air mixture burned by backfire: A flame arrestor was installed in the intake system and used to quench the flame initiated by backfire.

(4) To shut down of the H_2 system and minimize the amount of H_2 leaked into the engine laboratory once backfire occurred and the pressure relief valve was blown off. This was activated by shutting down the two emergency valves and also the mass flow controller. Once activated, there was no further H_2 flow into the intake system of the engine.

An emergency system was established to shut down the H_2 fuel system in case of emergency and also force the engine back to idle operation. As described before, there were two emergency valves activated by compressed air via solenoid valves, which were connected to an emergency button. The emergency button was designed in such a way that it would shut down the two emergency valves and the mass flow controller once activated.

Safety approaches were also implemented into the throttle control system and the operation of the mass flow controller. The control system would force the engine back to idle and shut down the mass flow controller once the following cases were detected: (1) the engine load was much higher than the set value; (2) the flow of H_2 was much higher than set value.

(5) The avoidance of the activation of the mass flow controller without turning on the emergency valve. A pressure transducer was installed prior to the mass flow controller and used as a reference signal for the opening and closing of the mass flow controller. The test cell control system was designed in such a way that it would not turn on the mass flow controller unless the pre-set H_2 pressure was established and would shutdown the mass flow controller once the H_2 pressure dropped below the set value. This prevented the over-dosing of the mass flow controller resulted from the malfunction of the emergency shutdown valves. In case of failure of on-time opening of the emergency valve, the mass flow controller could not deliver the required H_2 flow, which would force the mass flow controller to 100% open. With a fully open mass flow controller, the sudden opening of the emergency valve would make the fully open mass flow controller to deliver much more H_2 than required and raise safety issues associated with the over dosing of H_2 fuel such as backfire and the onset of knock. In case of late or failed opening of the emergency value, the mass flow controller would not be open until the transducer installed prior to the mass flow controller detected the preset pressure value, an indicator of the effective opening of the emergency shutdown valve.

4 Safety Issues Encountered and Operation Procedure Revised

Although the project was well prepared, there were still three incidents associated with the H_2 fuel system including two backfires and one failure of a high pressure pigtail tube. On the first day of H_2 operation, the engine was shutdown immediately after a preliminary test with the addition of H_2. After the diesel fuel tank was refilled, the start of the engine initiated an immediate backfire. As shown in Fig. 10, the pressure relief valve burst and released the pressure of the intake system without causing any damage to the engine intake system. It was believed that the occurrence of this unexpected backfire was due to the accumulation of H_2 in the intake system. After the H_2 system was shutdown, the H_2 contained in the pipeline between the mass flow controller and mixer entered the engine intake system. The H_2 contained in the pipeline between the second emergency shut-down valve and mass flow controller may also have leak into the intake system. After this incident, the engine procedure was revised. After the H_2 system was shut down, it was required that the diesel engine must continue to operate with pure diesel for at least 10 minutes to burn any residue H_2 that may have entered the intake system.

Figure 10 Pressure Relief Valve and Its Burst due to Backfire

The second incident occurred when operated at 100% load. During the transition from 0% to 3% H_2, the mass flow controller was commanded to provide 3% H_2 into the intake air without turning on the emergency shutdown valves. This forced the mass flow controller to open fully. After the emergency valve was opened, the wide open mass flow controller delivered much more H_2 than required and initiated backfire. After this incident, the operation procedure of H_2 fuel system and mass flow controller was revised: 1) the H_2 fuel system could only be turned on with pure diesel operation without H_2 flow expected; 2) The mass flow controller was not allowed to turn on until the pressure prior to the mass flow controller reached the predetermined value; 3) The mass flow controller was programmed to open up slowly to avoid large amounts of H_2 flowing into intake manifold. After the operation procedure was revised, no further backfire was observed during transition of engine operation conditions with different H_2 flow rate.

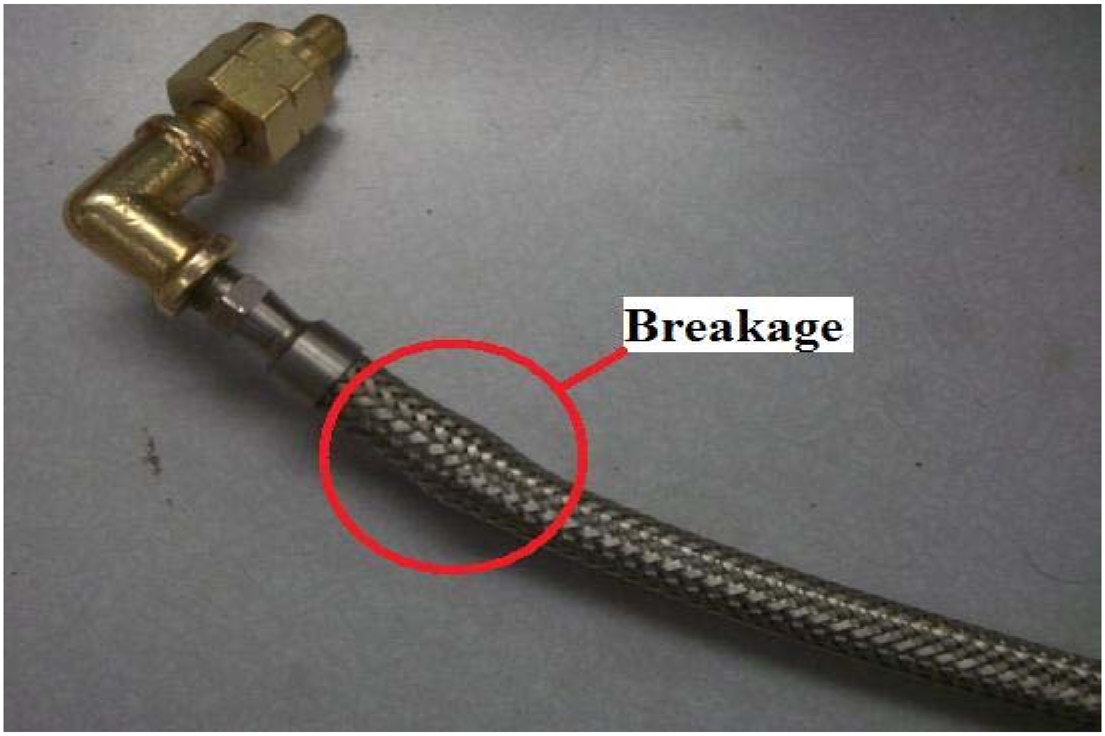

Figure 11 Failure of High Pressure Pigtail Pipe

The last incident was the failure of a high pressure flexible pigtail tube used to connect the high pressure H_2 tank to the H_2 module. After the full H_2 tanks were switched and turned on to provide H_2, the second pigtail tube broke and leaked compressed H_2 into air. As shown in Fig. 11, the breakage was about 0.5 inch from the connecter to the H_2 tank. It was suspected that the pigtail pipe was weakened by excess bending during installation.

5 Experimental Results and Analysis: 1999 Cummins ISM370 Diesel Engine

5.1 Test Matrix

The test matrix was designed to investigate the effect of H_2 addition, the engine load, engine speed and diesel flow rate on the engine performance, combustion and exhaust emissions of H_2-diesel dual engine operation. As shown in Table 6, the addition of 2% and 4% H_2 on the exhaust emissions was measured using the steady-state 13-mode ESC cycle. The modes were determined according to the engine map shown in Fig. 12. The detailed 13-mode settings can be found in Fig. 13 and Table 7. The effects of H_2 addition on the cylinder pressure, combustion, and exhaust emissions were examined under the conditions described in Table 8. The flow rate of H_2 at 6% was also listed in Table 8 to demonstrate the amount of H_2 added into this heavy-duty diesel engine. As shown in Table 9, the effect of engine speed on the operation of H_2-diesel duel fuel engine was examined under constant torque of 700 ft-lbf with the addition of 4% H_2 into the intake mixture. Table 10 shows the operating conditions under constant diesel flow rate operation at 1200 RPM. The engine load was varied by adjusting the amount of H_2 supplied into the diesel engine. In this test, the flow rate of diesel fuel was kept approximately at a constant 22.35 kg/hr, corresponding to 50% load for pure diesel operation. The maximum H_2 supplemented was 5.5% in the intake mixture.

Table 6 Test Matrix of Hot Start 13-Mode ESC Cycle Emission Test

13-Mode Emissions Test	Test 1	Test 2	Test 3
$H_2/(H_2+Air)$, vol. %	0	2	4

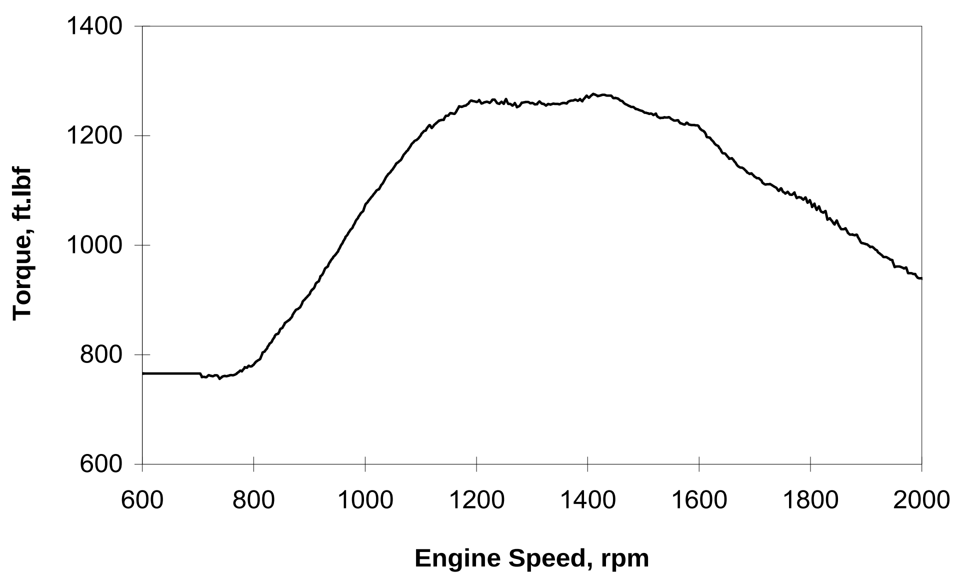

Figure 12 Load Map of 1999 Cummins ISM370 Diesel Engine

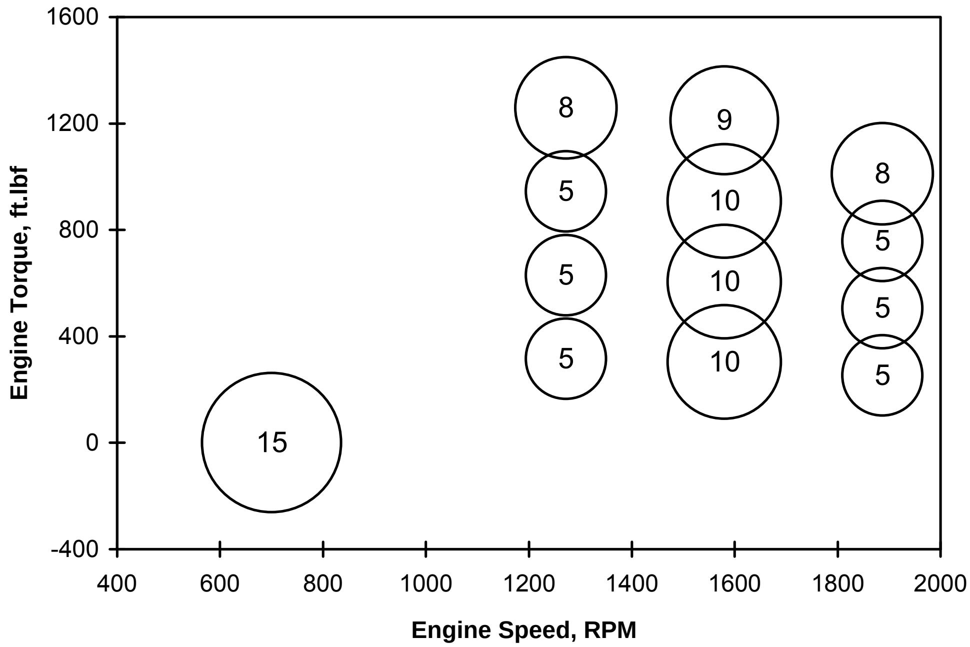

Figure 13 Set Points for 13-Mode ESC Cycle Emission Test

Table 7 Set Points for 13-Mode ESC Cycle Emission Test

Mode	Speed	Torque, ft-lbf	Weighting Factors, %
1	700	0	15
2	1272	1258	8
3	1580	606	10
4	1580	908	10
5	1272	629	5
6	1272	944	5
7	1272	315	5
8	1580	1211	9
9	1580	303	10
10	1887	1011	8
11	1887	253	5
12	1887	758	5
13	1887	506	5

Table 8 Constant Load Test Matrix, N=1200 RPM

Torque	$H_2/(H_2+Air)$, vol. %	H_2 Flow at $H_2/(H_2+Air)$=vol. 6%	
		kg/hr	l/min.
10%	0%, 1%, 2%, 3%, 3.5 4%, 4.5%, 5%, 5.5%, and 6%	2.10	391.1
15%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 3%, 3.5% ,4%, 4.5%, 5%, 5.5%, 6%, 6.5%	2.08	387.5
20%	0%, 1%, 2%, 3%, 3.5%, 4%, 4.5%, 5%, 5.5%, 6%, 6.5%	2.10	391.4
30%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 3%, 3.5%, 4%, 4.5%, 5%, 5.5%, 6%, and 6.5%	2.30	430.1
50%	0%, 1%, 2%, 3%, 3.5, 4%, 4.5%, 5%, 5.5%, 6%, and 6.5%	2.78	518.4
70%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 2.5%, 3%, 3.5%, 4%, 4.5%, 5%, 6%	3.30	615.3

Table 9 Test Matrix for Constant Torque with Variable Engine Speed

Engine Torque	$H_2/(H_2+Air)$, vol. %	Engine Speed, RPM
700 ft-lbf	4%	1200, 1300, 1400, 1500, 1600, 1700, and 1800

Table 10 Test Matrix for Constant Diesel Fuel Flow Rate

Diesel Flow Rate	$H_2/(H_2+Air)$, vol. %
22.35 kg/hr	0%, 1%, 2%, 3%, 4%, 4.5%, 5%, 5.5%

5.2 13-Mode Exhaust Emissions

Tabl3 11 shows the effect of the addition of H_2 on the exhaust emissions measured using the steady-state 13-mode ESC cycle. The addition of 2% H_2 into this 1999 Cummins ISM370 diesel engine was found to increase PM emissions from 0.032 g/bhp-hr to 0.037 g/bhp-hr (+15%). The addition of 4% H_2 reduced PM emissions from 0.032 g/bhp-hr to 0.029 g/bhp-hr (-10%).

Table 11 Effect of H_2 Addition on Exhaust Emissions, 13-Mode ESC Cycle, g/bhp-hr

$H_2/(H_2+Air)$, vol. %	PM	NO_x	CO	HC	CO_2
0% H_2	0.032	3.41	0.25	1.97	474.72
2% H_2	0.037	3.59	0.26	2.00	416.74
4% H_2	0.029	3.94	0.21	2.08	347.98

As shown in Table 11, the addition of 2% and 4% H_2 increased the emissions of NO_x from 3.41g/bhp-hr to 3.59 (+5.3%) and 3.94 g/bhp-hr (+15.5 %), respectively. Fig.14 shows the NO_x emissions measured using 13 mode ESC cycle. The addition of 2% and 4% H_2 was found to increase the NO_x emissions in all modes with the exception of mode 9 (1580 rpm, 303 ft-lbf) and 11 (1887 rpm and 253 ft-lbf), representing the low load operation. This may imply the different response of diesel engines to the addition of H_2 under high and low load operation. For example, the addition of H_2 at mode 10 (1887 rpm, 1011 ft-lbf), 12 (1887 rpm, 758 ft/lbf) and 13 (1877 rpm 506 ft-lbf) was shown to increase the emissions of NO_x.

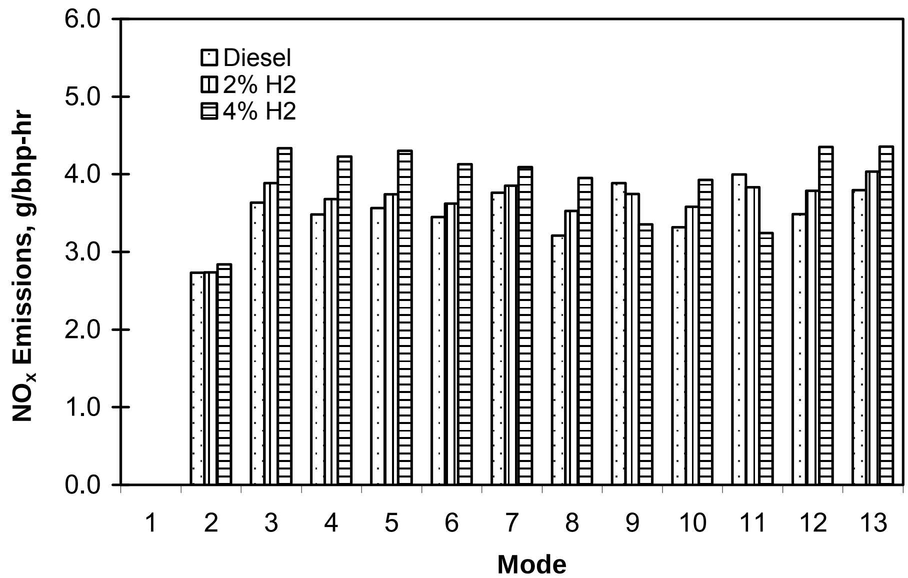

Figure 14 Effect of H_2 Addition on NO_x Emissions Measured Using 13-Mode ESC Cycle

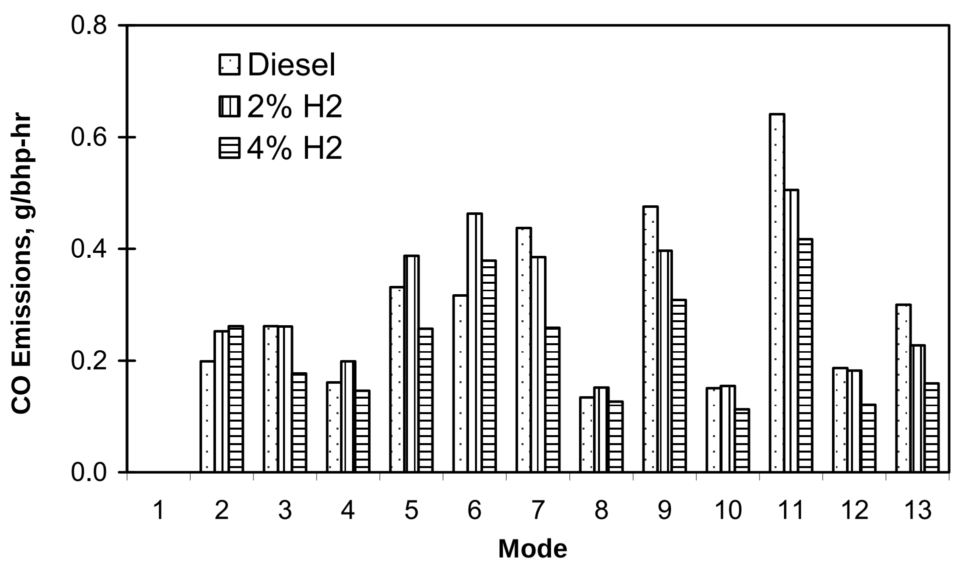

Figure 15 Effect of H_2 Addition on CO Emissions Measured Using 13-Mode ESC Cycle

The effect of H_2 addition on CO emissions can be found in Table 11 for integrated emissions and Fig. 15 for those measured in each mode. As shown in Table 11, the addition of 2% H_2 had a negligible effect on CO emissions measured using 13-mode emission cycle. However, the addition of 4% H_2 was found to reduce CO emissions from 0.25g/bhp-hr to 0.21 g/bhp-hr. As shown in Figure 15, the addition of 2% H_2 increased the emissions of CO with exception of mode 7, 9, 11, 12 and 13. In comparison, the addition of 4% H_2 reduced the emissions of CO with the exception of mode 2, a full load operation at 1272 rpm.

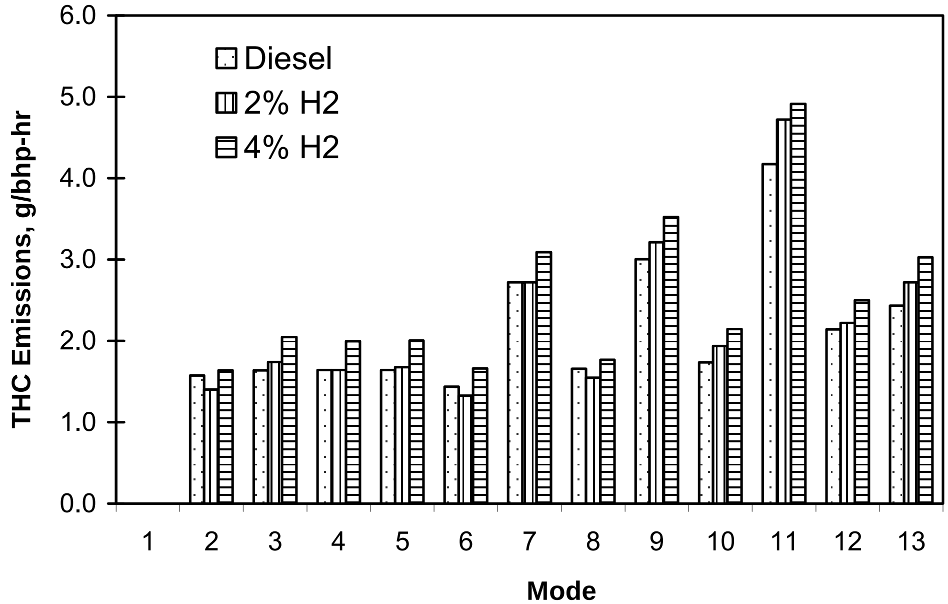

Figure 16 Effect of H_2 Addition on THC Emissions Measured Using 13-Mode ESC Cycle

As shown in Table 11, the addition of 2% H_2 and 4% H_2 was found to slightly increase THC emissions from 1.97 g/bhp-hr to 2.00 g/bhp-hr (+1.5%) and 2.08 g/bhp-hr (+5.58%), respectively. As shown in Fig. 16, the addition of 4% H_2 was found to increase THC emissions for all of the emissions modes. However, the significant effect of adding 2% H_2 on THC emissions was only observed for modes 9, 10, 11, and 13.

5.3 Effect of H_2 Addition and Engine Load on Exhaust Emissions

5.3.1 NO_x Emissions

The effect of H_2 addition on NO_x emissions was examined for 10% - 70% load. As shown in Fig. 17 for medium to high load operation, the addition of H_2 into the intake mixture of this 1999 Cummins ISM370 diesel engine gradually increased the emissions of NO_x, which was consistent with data reported in literature [Varde and Frame, 1983 and McWilliam et al. 2008]. This was due to the increased maximum average bulk mixture combustion temperature with H_2 addition as shown in Fig. 18 for 70% load operation. As shown in Fig. 18, the addition of a small amount of H_2 into the diesel engine slightly reduced the maximum averaged bulk mixture temperature without causing a notable reduction in NO_x emissions. As shown in Fig. 19, the slight increase in NO_x emissions may have been due to the increased conversion of NO to NO_2, which has a larger molar weight. It should be noted that the maximum averaged bulk mixture temperature calculated from the pressure curve represents the average temperature of the bulk mixture, which is much lower than the combustion temperature. The latter dominates the formation of the NO_x in diesel engines. Further increasing the amount of H_2 beyond 2% was shown to increase the maximum averaged bulk mixture temperature accompanied with increased NO_x emissions.

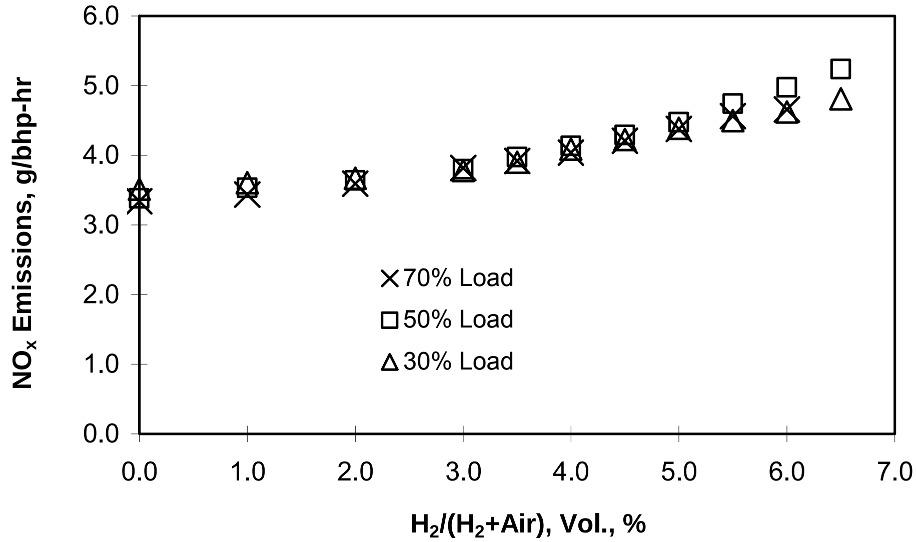

Figure 17 Effect of H_2 Addition and Engine Load on NO_x Emissions, N=1200 RPM, 30-70% Load

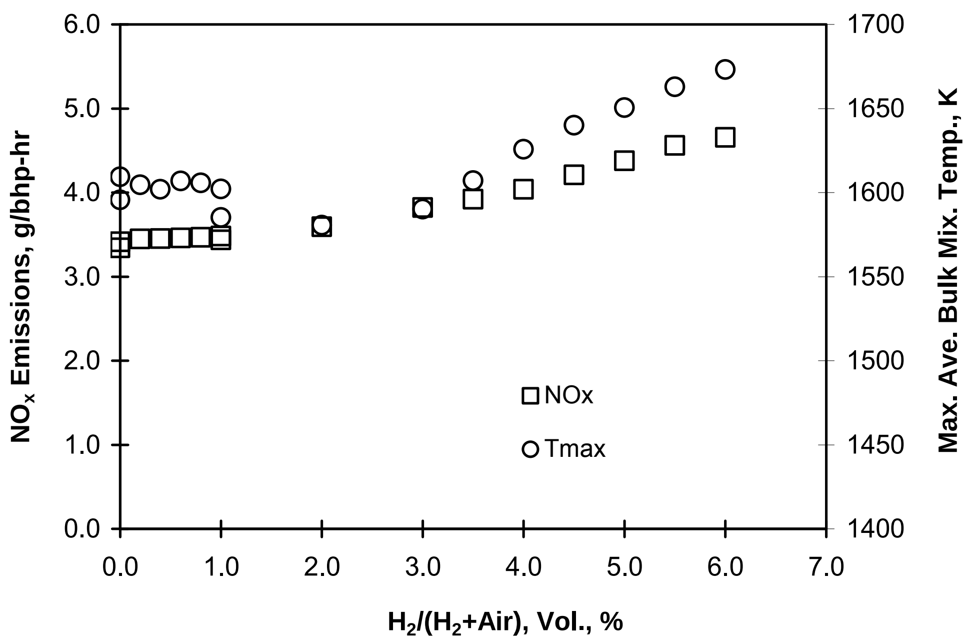

Figure 18 Effect of H_2 Addition on NO_x Emissions and Maximum Average Bulk Mixture Temperature Calculated Using Cylinder Pressure, N=1200 RPM, 70% Load

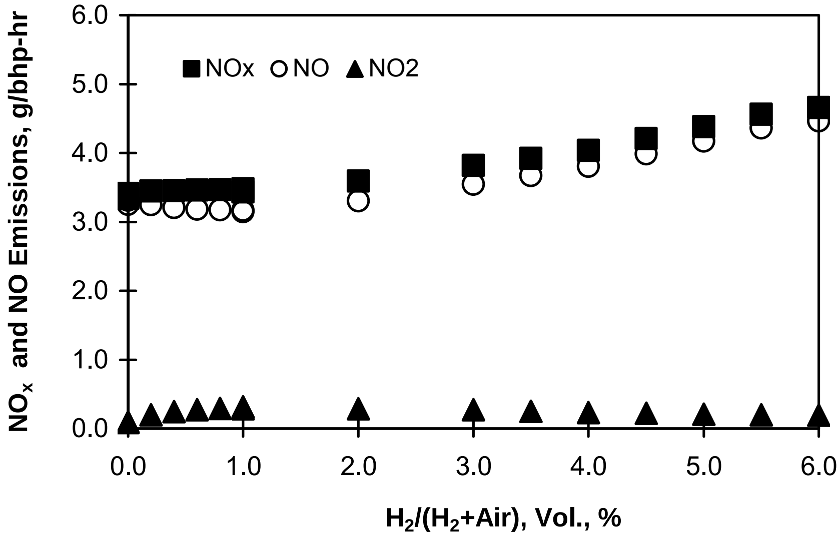

Figure 19 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 70% Load

The effect of H_2 addition on NO_x emissions was also examined under low load operation. As shown in Fig. 20, the addition of a small amount of H_2 (<3%) at 10% load operation had a negligible effect on NO_x emissions. Increasing the amount of H_2 supplemented beyond 3% reduced the NO_x emissions. In comparison, the addition of H_2 at 15% and 20% load slightly increased the emissions of NO_x until a maximum value was observed. Further increasing the addition of H_2 reduced substantially the emissions of NO_x. Compared to that of pure diesel operation, the reduction of NO_x emissions through the addition of H_2 was only obtained at a narrow operational range (10 to 15% load with the addition of H_2 over 3%)

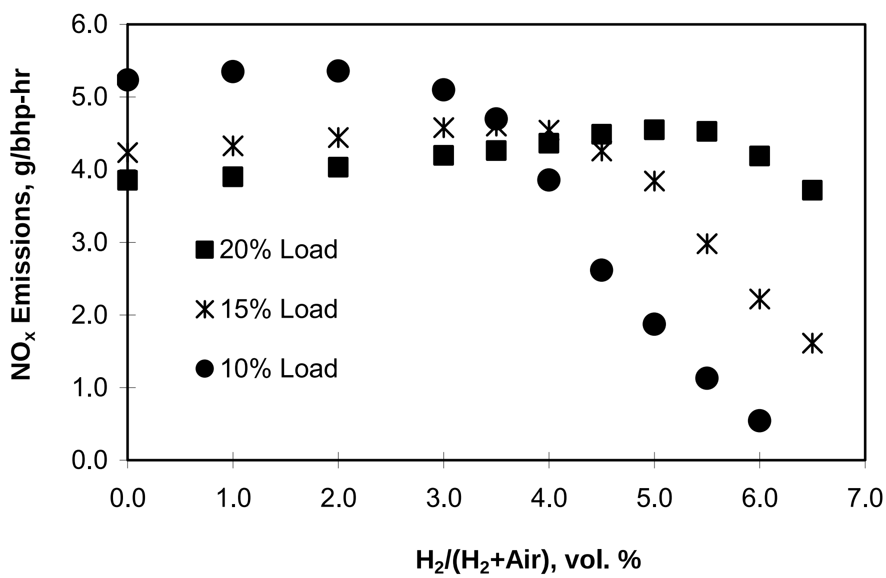

Figure 20 Effect of H_2 Addition and Engine Load on NO_x Emissions, N=1200 RPM, 10%-20% Load

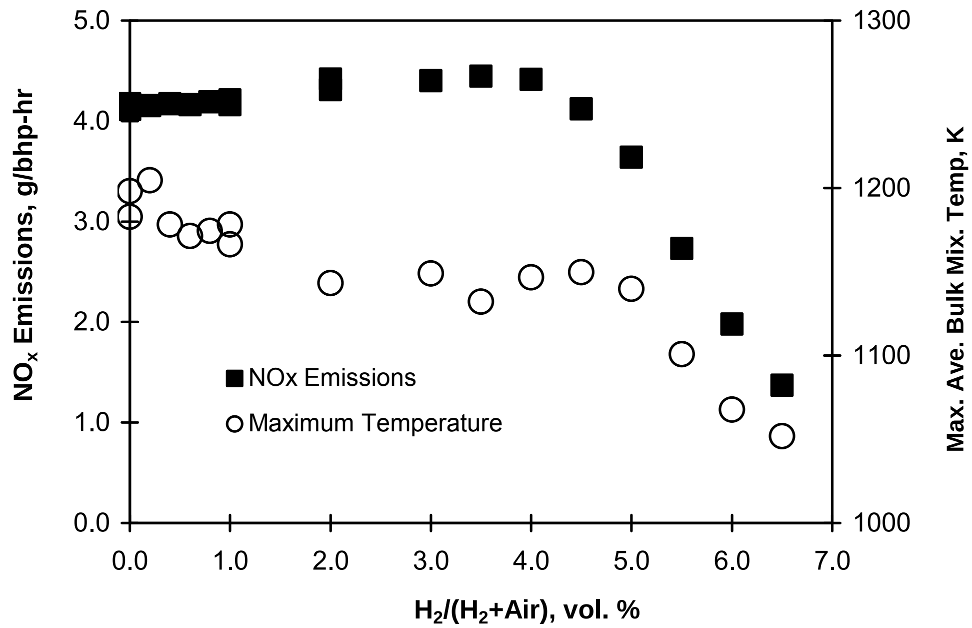

Figure 21 Effect of H_2 Addition on NO_x Emissions (mass) and Maximum Average Bulk Mixture Temperature, N=1200 RPM, 15% Load

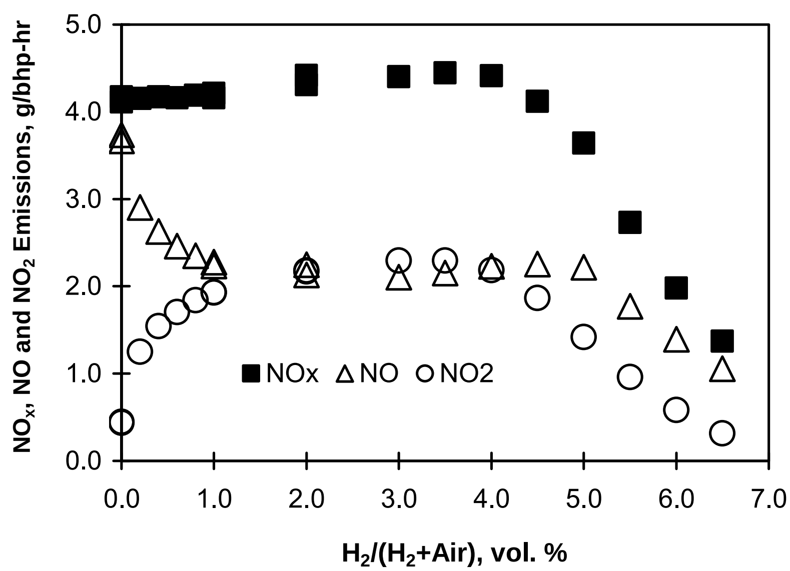

Figure 22 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 15% Load

As shown in Fig. 21, the addition of a small amount of H_2 at 15% load operation was found to gradually reduce the averaged bulk mixture temperature accompanied with increasing NO_x emissions. This is not consistent with the traditional NO_x formation mechanism. As shown in Fig. 22, the slight increase in NO_x emissions was due to the conversion of NO to NO_2, a compound having a higher molar weight than NO. This was further confirmed by examining the emissions rate of NO_x on a molar basis. As shown in Fig. 23, the addition of H_2 at 15% load was found to slightly reduce the NO_x emissions examined on a molar basis. As shown in Figs. 21 and 23, the dramatic reduction in NO_x emissions associated with the addition of a relatively large amount of H_2 was due to the reduced bulk mixture temperature.

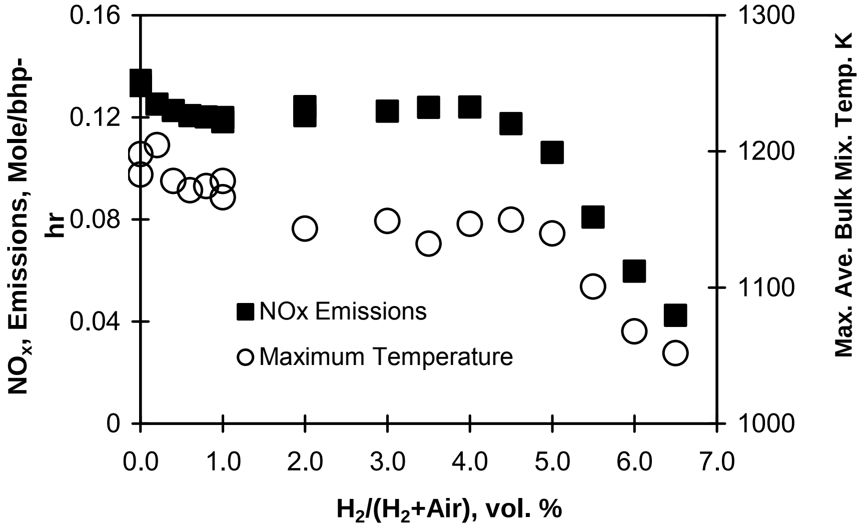

Figure 23 Effect of H_2 Addition on NO_x Emissions (mole) and Maximum Average Bulk Mixture Temperature, N=1200 RPM, 15% Load

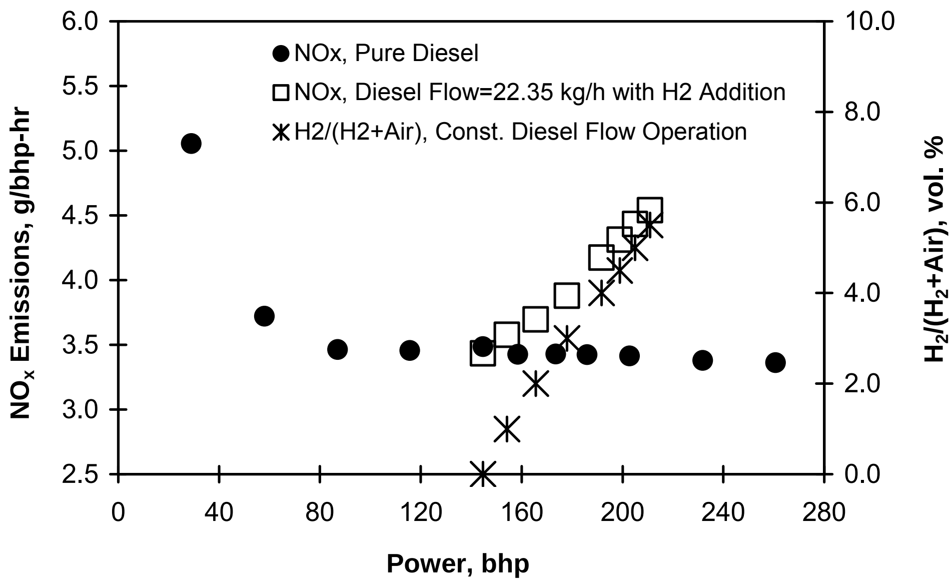

Figure 24 Effect of H_2 Addition on NO_x Emissions Operated with Constant Diesel Flow Rate of 22.45 kg/hr (Corresponding to 50% Load for Pure Diesel Operation). N=1200 RPM. For Constant Diesel Fuel Flow Rate Operation, Engine Load was varied by Adding H_2

The enhancement of H_2 addition on NO_x emissions for operation at medium to high load was also demonstrated the addition of increasingly amount of H_2 under constant diesel flow rate. Compared to that of pure diesel operation, the addition of H_2 at medium to high load increased the emissions of NO_x as shown in Fig. 24.

5.3.2 PM Emissions

The effect of H_2 addition on the emissions of PM is shown in Fig. 25 and 26. As shown in Fig. 25 for medium to high load operation, the emissions of PM were gradually reduced with the addition of H_2 until 4.5% H_2 was supplied into the intake mixture. The addition of H_2 beyond 4.5% was shown to have a negligible effect on the emissions of PM.

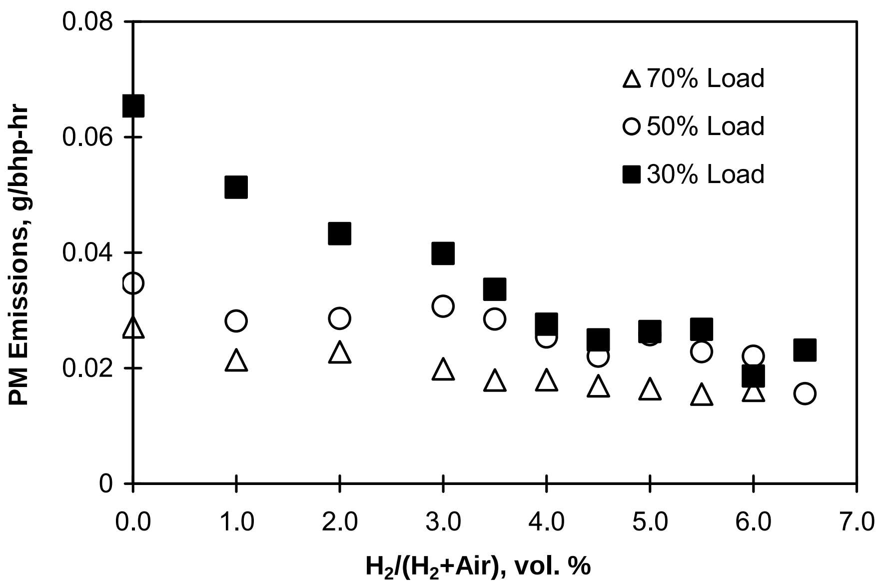

Figure 25 Effect of H_2 Addition and Engine Load on PM Emissions, N=1200 RPM, 30%-70% Load

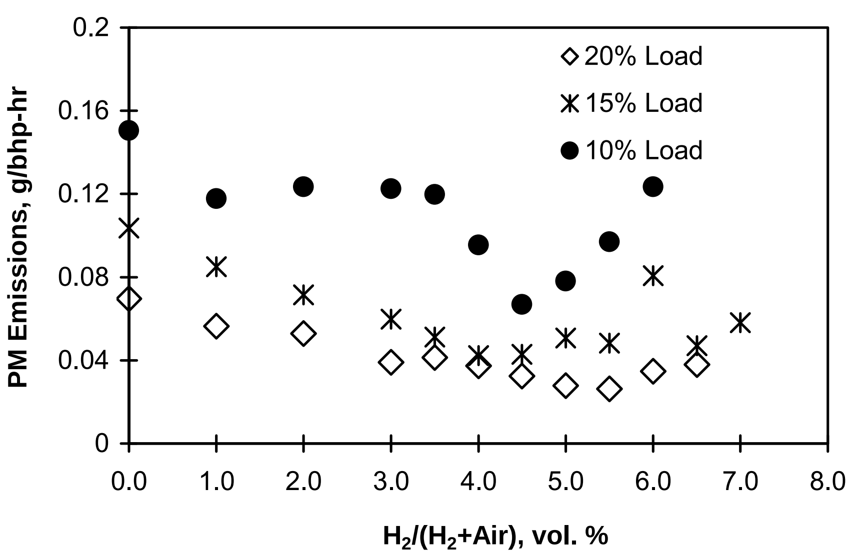

Figure 26 Effect of H_2 Addition and Engine Load on PM Emissions, N=1200 RPM, 10%-20% Load

Fig. 26 shows the effect of H_2 addition on the emissions of PM when operated at low load. The addition of H_2 below 4.5% was found to gradually reduce the emissions of PM. However, the addition of H_2 beyond 4.5% was found to gradually increase the emissions of PM. This may be due to the burning of diesel fuel injected in late phasing in hot bulk gases when the H_2-air mixture was rich enough to support a propagating flame. The detailed explanation of this phenomenon may need further detailed experimental research. It was evident that the addition of H_2 below 4% reduced the PM emissions. Compared to pure diesel operation, the maximum PM reduction obtained was 40% to 65% when operated from 10% to 70% load. In comparison, the PM reduction measured using the 13-mode emissions cycle was much lower. As shown in Table 12 for 2% and 4% H_2 addition, the PM reduction measured using 13-mode ESC emission cycle were much lower than that obtained under constant load operation of 10%-70% load. This may be due to the effect of H_2 addition on PM emissions at full load or at high engine speed, which has not been measured in this research.

Table 12 Effect of H_2 Addition on PM Emissions Measured under Constant Load and the 13-Mode Emission Cycle, g/bhp-hr

Load	Pure Diesel	$H_2/(H_2+Air)$, vol. 2%		$H_2/(H_2+Air)$, vol. 4%	
	PM Emissions g/bhp-hr	PM Emissions g/bhp-hr	PM Reduction	PM Emissions g/bhp-hr	PM Reduction
10%	0.151	0.124	-17.88 %	0.095	-37.09%
15%	0.104	0.0715	-31.25%	0.0423	-59.33%
20%	0.096	0.053	-44.79%	0.0374	-61.04%
30%	0.0654	0.043	-34.25%	0.0276	-57.80%
50%	0.0357	0.0286	-19.89%	0.0254	-28.85%
70%	0.0271	0.022	-18.82%	0.018	-33.58%
13-Mode ESC Cycle	0.032	0.037	+15.63%	0.029	-9.38%

5.3.3 CO Emissions

Carbon monoxide is formed in the diesel engine due to the incomplete combustion of diesel fuels and also the dissociation of CO_2 under high combustion temperature conditions. As shown in Figs. 27 and 28, the emissions of CO were found to decrease almost linearly with the addition of H_2 into this diesel engine with the exception of 70% load operation. This was due mainly to the reduced consumption of diesel fuel. The presence of a large amount of H_2 enhanced the complete combustion CO formed due to the incomplete combustion of diesel fuel. As shown in Fig. 28 for 70% load operation, the addition of a small amount of H_2 (below 3.5%) had a negligible effect on CO emissions. Increasing the amount of H_2 beyond 4% slightly increased the emissions of CO.

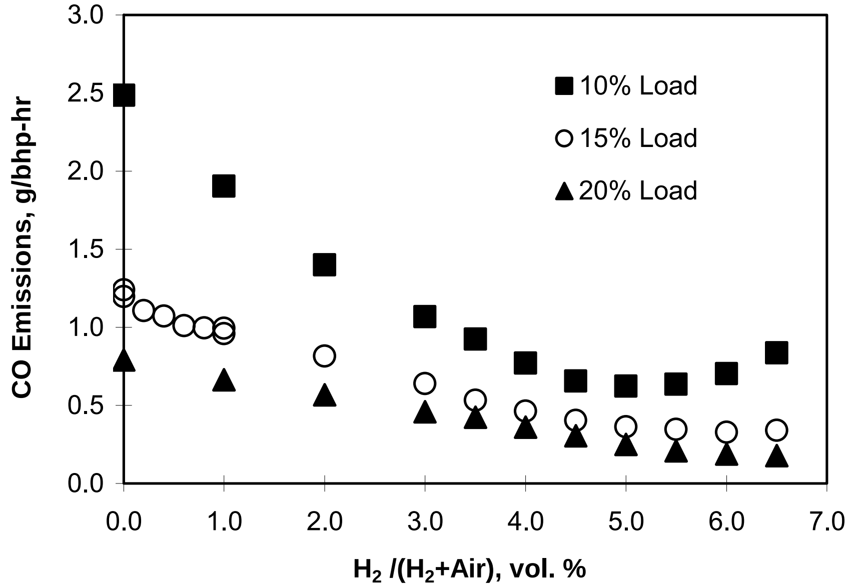

Figure 27 Effect of H_2 Addition and Engine Load on CO Emissions, N=1200 RPM, 10%-20% Load

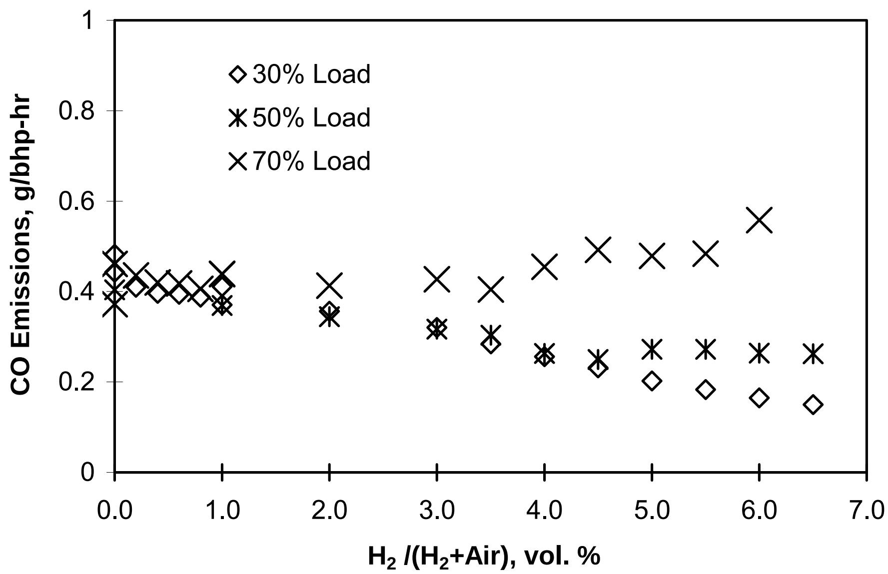

Figure 28 Effect of H_2 Addition and Engine Load on CO Emissions, N=1200 RPM, 30%-70% Load

5.3.4 THC Emissions

The effect of H_2 addition on THC emissions can be found in Figs. 29 and 30. As shown in Fig. 29, the addition of H_2 at low load increased the emissions of THC. When operated at medium to high load, the addition of H_2 had negligible effect on the emissions of THC as shown in Fig. 30. This is consistent with the results reported in the 13-mode test, which showed very small effect of H_2 on THC emissions.

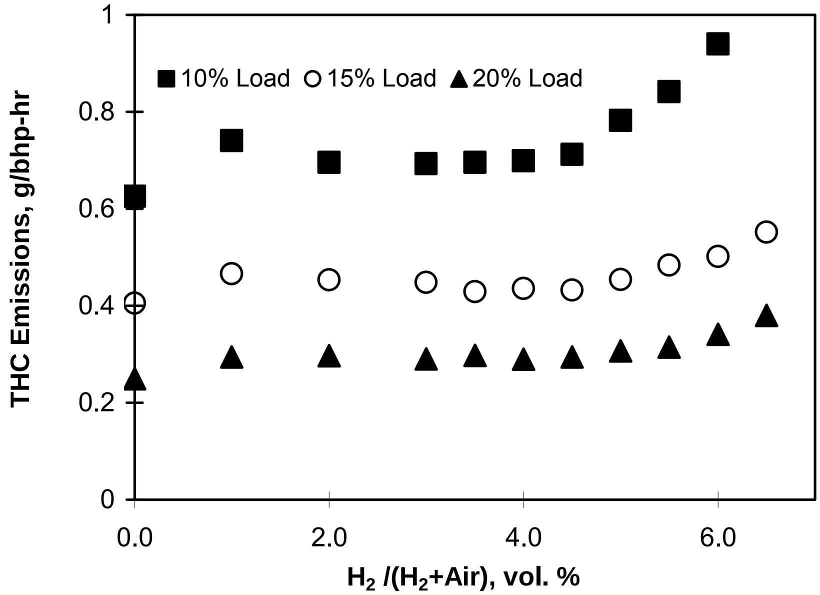

Figure 29 Effect of H_2 Addition and Engine Load on HC Emissions, N=1200 RPM, 10%-20% Load

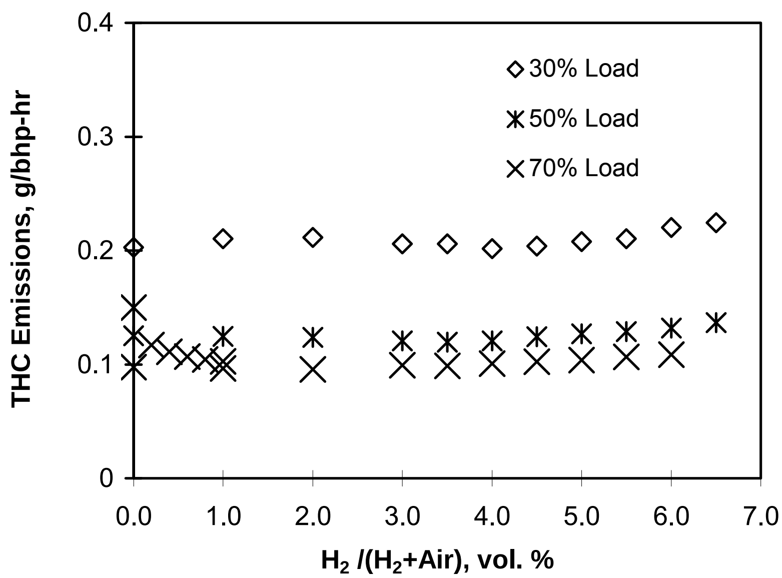

Figure 30 Effect of H_2 Addition and Engine Load on HC Emissions, N=1200 RPM, 30%-70% Load

5.3.5 CO_2 Emissions

The effect of H_2 addition on CO_2 emissions can be found in Fig. 31. As expected, the addition of H_2 into the diesel engine gradually reduced the emissions of CO_2. With the addition of a small amount of H_2, the operation at high load produced less CO_2 than low load operation due to the higher brake thermal efficiency obtained at high load operation. However, with the addition of relatively large amount of H_2, the operation at low load produced less CO_2 than at high load operation. This was due to mainly the substitution of large percentage of diesel fuel by H_2 for low load operation.

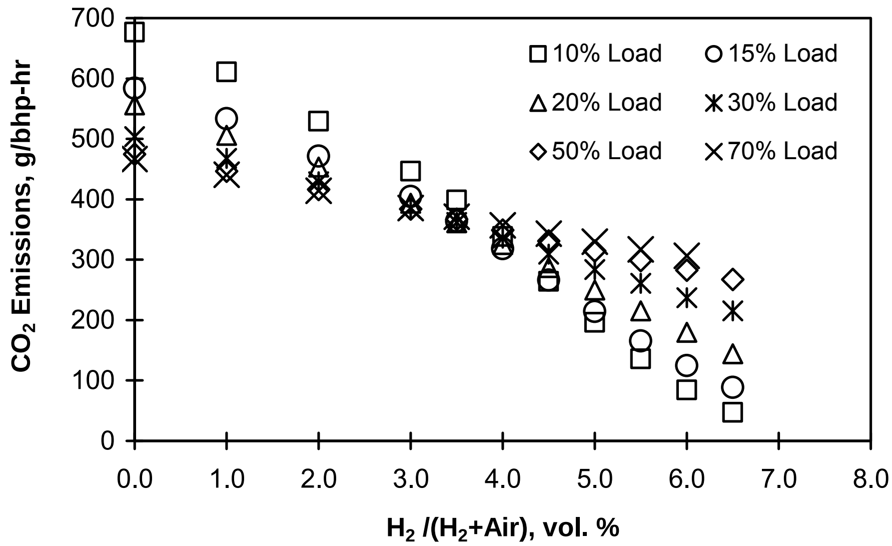

Figure 31 Effect of H_2 Addition and Engine Load on CO_2 Emissions, N=1200 RPM, 10%-70% Load

5.4 The Exhaust Emissions of H_2 and Its Combustion Efficiency

Despite benefits in obtaining higher thermal efficiency with multiple fuel flexibility, there are numerous issues associated with the burning of gaseous fuels in CI engines using dual fuel combustion mode. When H_2 is supplemented into the intake mixture, it forms a homogeneous H_2-air mixture in the combustion chamber. In principle, the supplemented H_2 could be burned through one of the following modes: (1) the spontaneous combustion with diesel fuel when present within the diesel spray plume and mixed with diesel vapor; (2) the consumption of H_2 by a propagating turbulent H_2-air-diluent flame initiated by diesel combustion, which can only be obtained when H_2-air mixtures richer than the flammability limit of H_2; (3) the oxidation (auto-ignition) of H_2 in air under high temperature resulting from compression of piston movement, and combustion of H_2 and diesel. The high oxidation temperature of H_2 in air makes it very difficult for H_2 to be burned through auto-ignition only. The initiation and development of a propagating turbulent flame requires the formation of a combustible H_2-air mixture. The failure to initiate a propagating H_2 flame or its quenching during its propagation through the bulk mixture will make the remaining H_2 pass through the engine without participating in the combustion process. The presence of crevices in a diesel engine combustion chamber can also trap H_2-air mixture during the compression stroke and release it during the expansion stroke. This contributes to the slip of H_2 through the engine without participating in the combustion process.

As shown in Fig. 32 for 15% load operation, the exhaust emissions of H_2 increased almost linearly with the addition of a small amount of H_2. This was due to the lack of a healthy H_2 flame, which made it impossible to burn the lean H_2-air mixture located outside of the diesel spray plume. The maximum H_2 emissions were observed when 3.5% H_2 was supplemented into the intake air, which was leaner than that needed to support a healthy H_2 flame (4% in pure air at ambient temperature). The addition of H_2 beyond 4% gradually reduced the emissions of H_2. This was due to the initiation of a propagating H_2-air flame. The H_2 emissions data measured was further processed to calculate the combustion efficiency of H_2, defined as the molar percentage of H_2 burned in the engine relative to H_2 supplied into the intake mixture. As shown in Fig. 32, the H_2 combustion efficiency of 67% was observed when small amounts of H_2 were supplemented into the intake mixture. The increase in the amount of H_2 supplemented slightly increased the combustion efficiency until H_2 reached 3% in the intake mixture. Increasing the amount of H_2 added beyond 3% improved substantially the combustion efficiency of H_2. The combustion efficiency of 94.5% was obtained when the amount of H_2 added reached 6.5% in the intake mixture.

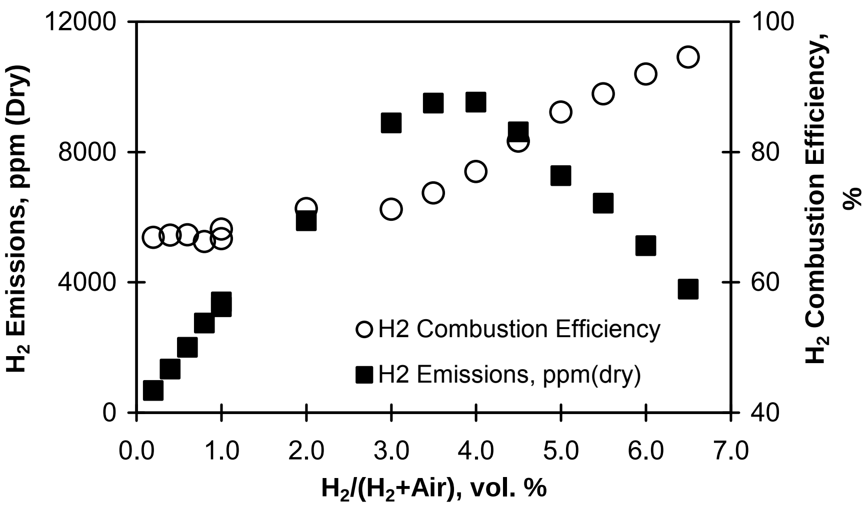

Figure 32 Effect of H_2 Addition on the Emissions of H_2 and Its Combustion Efficiency, N=1200 RPM, 15% Load

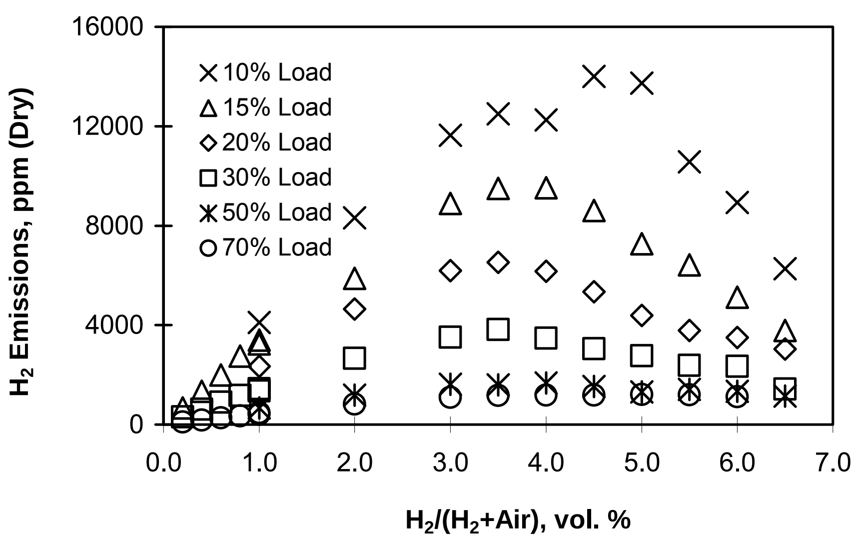

Figure 33 Effect of H_2 Addition and Engine Load on the Emissions of H_2, N=1200 RPM, 10%-70% Load

Figure 33 shows the effect of engine load and the H_2 addition on the emissions of H_2. When operated at higher load, the emissions of H_2 were found to drop drastically. This was due mainly to the increased diesel flow rate, which had a larger diesel spray plume and helps to burn more H_2 mixed with diesel vapor in a wider area. With the increase in engine load, the maximum H_2 emissions were observed in a smaller H_2 addition ratio as shown in Fig. 34. With the increase in engine load, the effect of H_2 addition on H_2 emissions became weaker. As shown in Fig. 33, the effect of H_2 addition on H_2 emissions became negligible for 50% and 70% load operation.

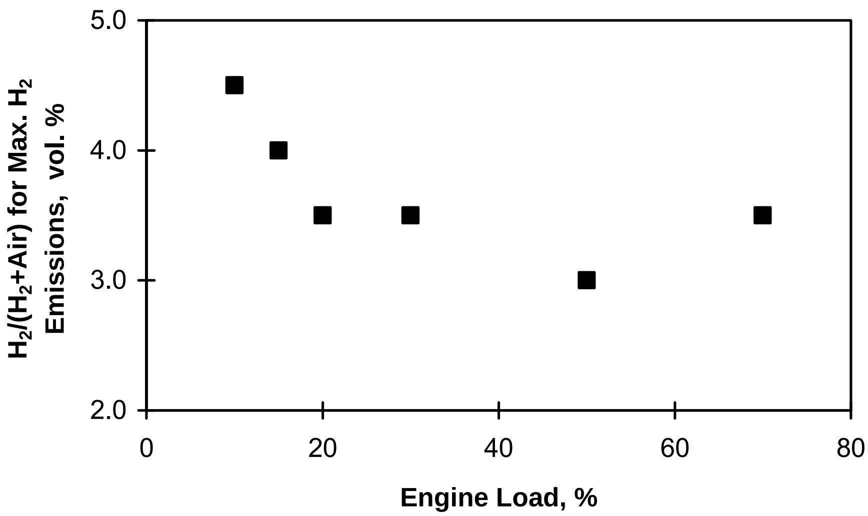

Figure 34 Effect of Engine Load on H_2 Addition Limit to Obtain Maximum H_2 Emissions

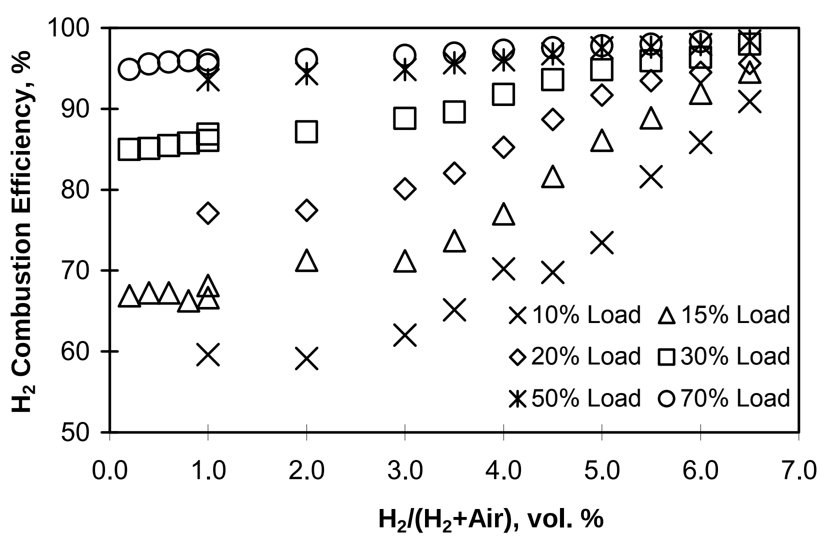

Figure 35 Effect of H_2 Addition and Engine Load on the Combustion Efficiency of H_2, N=1200 RPM, 10%-70% Load

Figure 35 shows the effect of engine load and H_2 addition on the combustion efficiency of H_2. The combustion efficiency was found to be about 60% when a small amount of H_2 (<3%) was added at 10% load operation. It was evident that the addition of a small amount of H_2 at low load operation should be avoided. As shown in Fig. 35, the increase in engine load and the amount of H_2 added beyond 3% improved substantially the combustion efficiency of H_2. It was also noted that the effect of engine load on the combustion efficiency of H_2 became negligible when operated at medium to high load operation (50% and 70% load). The maximum H_2 combustion efficiency of 98.3% was obtained for 70% load operation with the addition of 6% H_2 into the intake mixture. Due to the characteristics of H_2 mixing with air in the intake mixture and the presence of crevices in the engine combustion chamber, it is impossible to get 100% combustion efficiency as the H_2 fuel exiting the crevices late in the expansion stroke cannot be burned completely.

5.5 Brake Thermal Efficiency and Its Improvement

The effect of H_2 addition on the brake thermal efficiency was also examined. As shown in Fig. 36 for 30% load operation, the addition of a small amount of H_2 was found to reduce the brake thermal efficiency with its minimum value observed with the addition of H_2 at about 1-2%. The addition of H_2 beyond 2% was shown to gradually improve the brake thermal efficiency. With the addition of 3% H_2, the brake thermal efficiency obtained was comparable to that of pure diesel operation. With the addition of 6.5% H_2 into the intake mixture, the brake thermal efficiency obtained was 41.56%, a much better value than 37.9% obtained for pure diesel operation. The brake thermal efficiency of H_2-diesel dual fuel operation was compared to that of pure diesel operation to calculate the improvement to the brake thermal efficiency. As shown in Fig. 36, the addition of H_2 over 3% was shown to improve the brake thermal efficiency. With the addition of 6.5% H_2, the improvement to the brake thermal efficiency reached 9.54%.

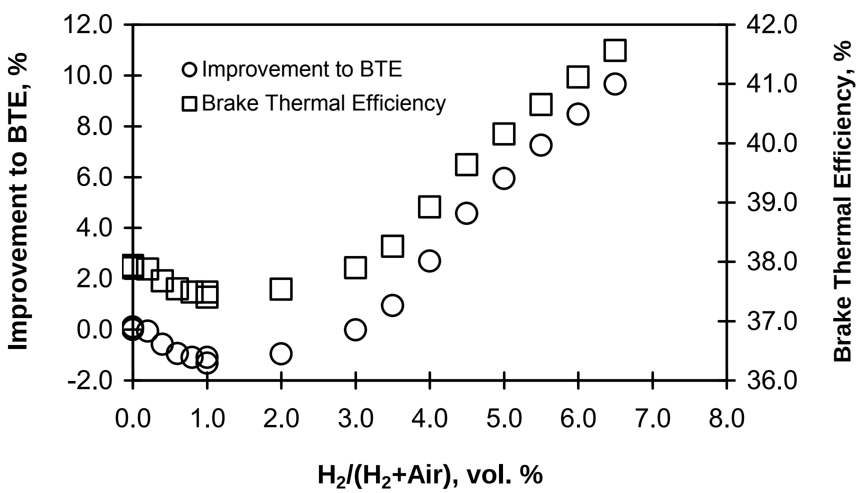

Figure 36 Effect of H_2 Addition on Brake Thermal Efficiency (BTE) and Its Improvements, N=1200 RPM, 30% Load

The improvement to the brake thermal efficiency could be due to improvement to the indicated thermal efficiency resulted from the enhanced combustion and the reduction of mechanical loss through the improvement to mechanical efficiency. As shown in Fig. 37, the addition of H_2 into the diesel engine was found to gradually improve the mechanical efficiency. With the addition of 6.5% H_2 into the intake mixture, the mechanical efficiency was increased from 82% to 85.1%, which contributed to about 1.5% (absolute) improvement to the brake thermal efficiency considering the range of indicated thermal efficiency (45%-50%) obtained at 30% load. As shown in Fig. 38, the addition of a small amount of H_2 was shown to reduce the indicated thermal efficiency, which contributed to the deterioration in the brake thermal efficiency. Similar to the brake thermal efficiency, the minimum indicated thermal efficiency of 45.12% was also observed at around 1% H_2. Increasing the amount of H_2 beyond 1% was shown to improve the indicated thermal efficiency. Compared to pure diesel operation, the improved indicated thermal efficiency was obtained with the addition of H_2 beyond 4%. With the addition of 6.5% H_2, the indicated thermal efficiency of 48.84% was obtained, a much better value compared to 46.21% observed for pure diesel operation.

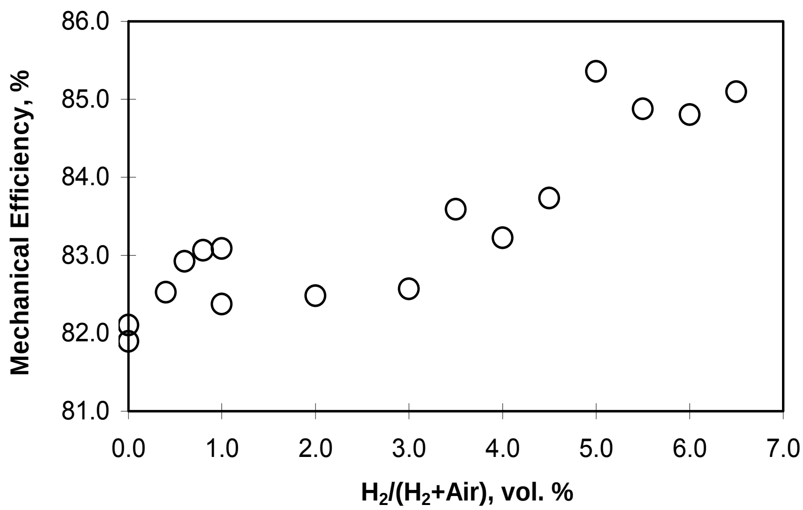

Figure 37 Effect of H_2 Addition on Mechanical Efficiency, N=1200 RPM, 30% Load

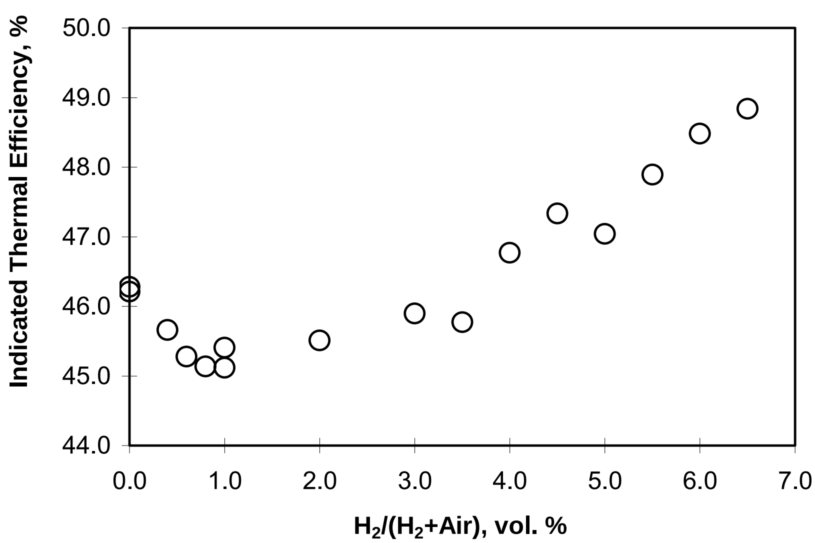

Figure 38 Effect of H_2 Addition on Indicated Thermal Efficiency, N=1200 RPM, 30% Load

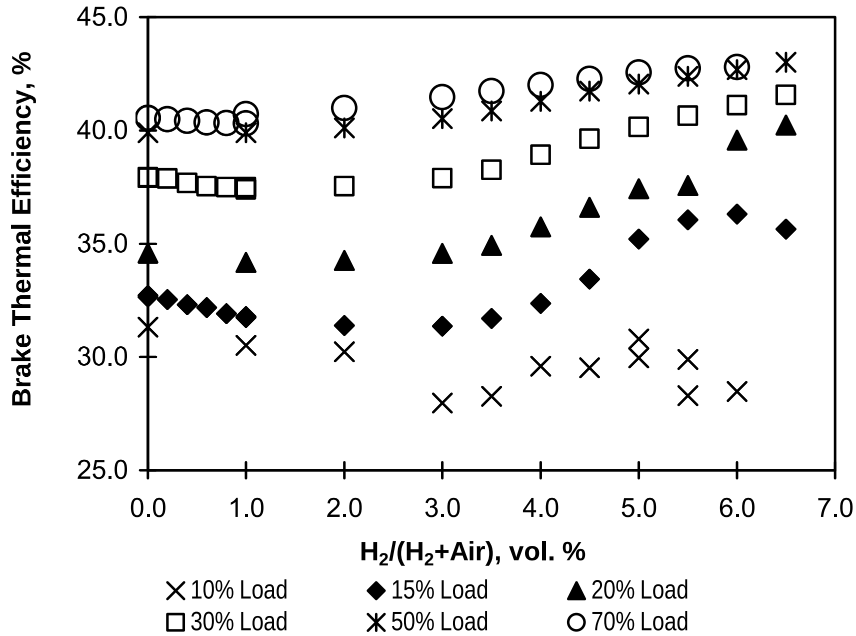

Figure 39 Effect of H_2 Addition and Engine Load on the Improvement to Brake Thermal Efficiency (BTE), N=1200 RPM, 10%-70% Load

Figure 39 compares the effect of engine load and H_2 addition on the brake thermal efficiency. The addition of a small amount of H_2 at low load was found to reduce the brake thermal efficiency. As shown in Fig. 39, the addition of H_2 up to 6% at 10% load was found to reduce the brake thermal efficiency. With the increase in engine load, the addition of a relatively large amount of H_2 was shown to improve the brake thermal efficiency. For example, improved brake thermal efficiency was obtained with the addition of 4.5% H_2 at 15% load operation. It was evident that there was a minimum H_2 supplementation rate, beyond which a positive effect on brake thermal efficiency could be obtained. Such a rate was defined as the minimum H_2 supplementation limit. As shown in Fig. 40, with the increase in engine load, the minimum H_2 supplementation limit to obtain a positive effect on the brake thermal efficiency was found to expand toward lower H_2 concentration. For example, the addition of H_2 beyond 3% improved the brake thermal efficiency when operated at 20% load. When operated at medium (50%) to high load, the addition of H_2 over 1% improved the brake thermal efficiency. It was evident that low load operation required the addition of more H_2 to obtain an improvement to the brake thermal efficiency.

Figure 41 shows the effect of H_2 addition and engine load on the improvement of the brake thermal efficiency. The maximum improvement to the brake thermal efficiency of 16.3% was obtained for 20% load operation with the addition of 6.5% H_2. As shown in Fig. 42 for the addition of 6% H_2, the improvement to the brake thermal efficiency was found to decrease with the increased engine load with the exception of 10% load operation. The addition of H_2 at 10% load operation did not improve the brake thermal efficiency.

The effect of engine speed and H_2 addition on the improvement of the brake thermal efficiency can be found in Fig. 43. The addition of 4% H_2 under constant torque found to improve the brake thermal efficiency. As shown in Fig. 44, the engine speed had negligible effect on the improvement to the brake thermal efficiency resulting from the addition of 4% H_2. The addition of 4% H_2 was shown to improve the brake thermal efficiency by approximately 3.8% for the range of engine speed examined.

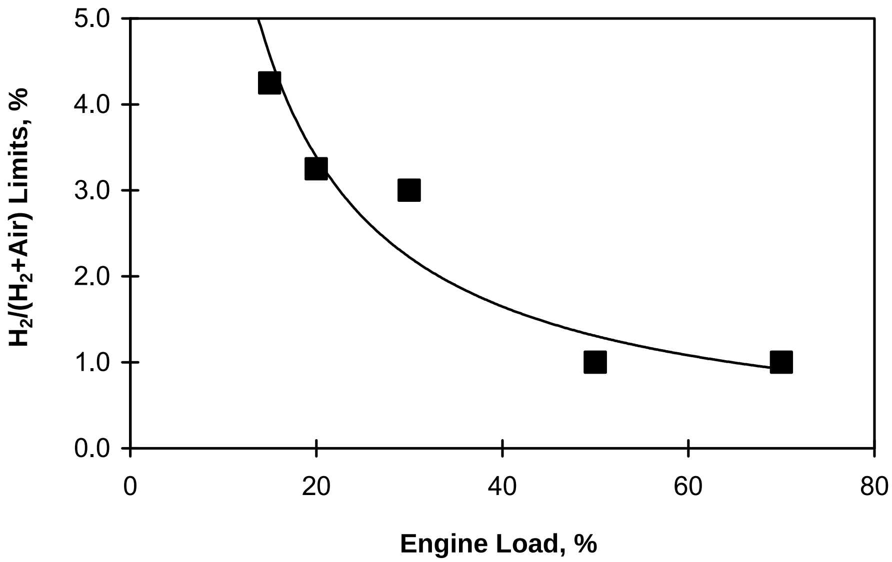

Figure 40 Effect of Engine Load on the Minimum H_2 Supplementation Rate Needed for Positive Effect on Brake Thermal Efficiency

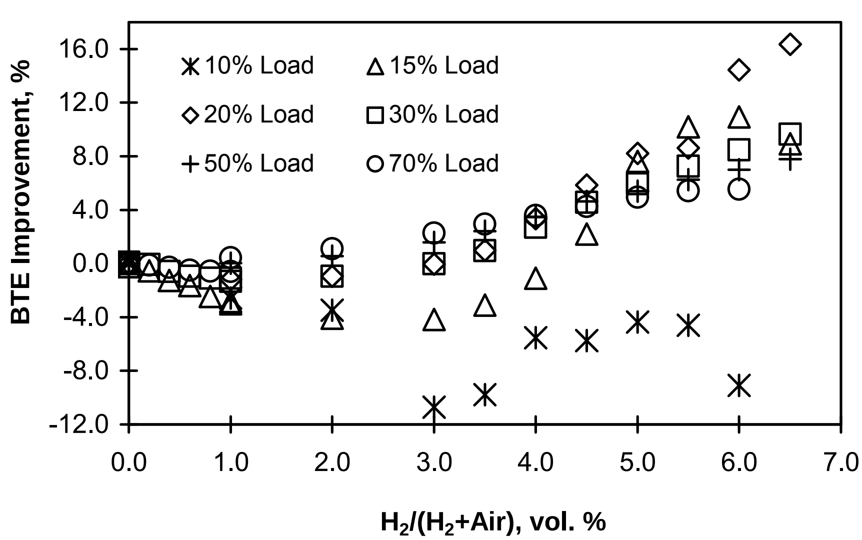

Figure 41 Effect of H_2 Addition in Improving the Brake Thermal Efficiency, N=1200 rpm, 10%-70% Load

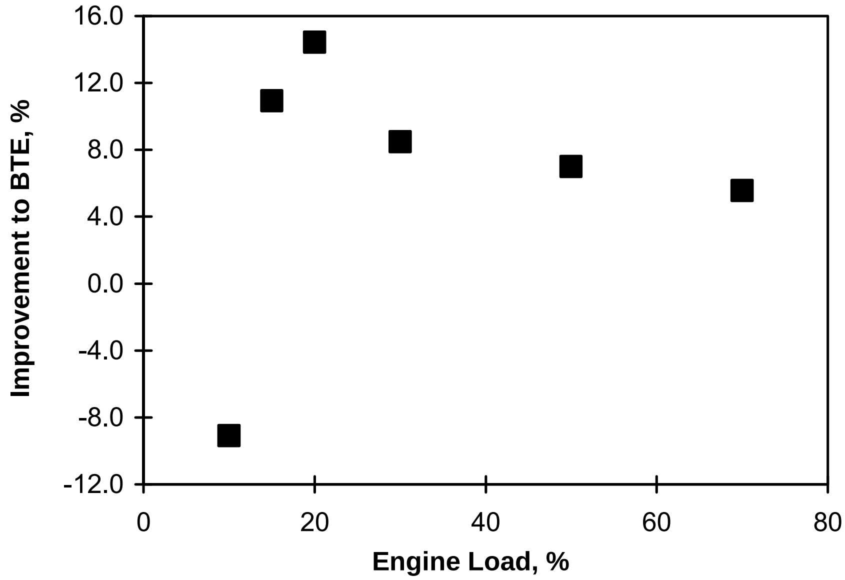

Figure 42 Effect of Engine Load on the Improvement to Brake Thermal Efficiency with the Addition of 6% H_2, N=1200 RPM

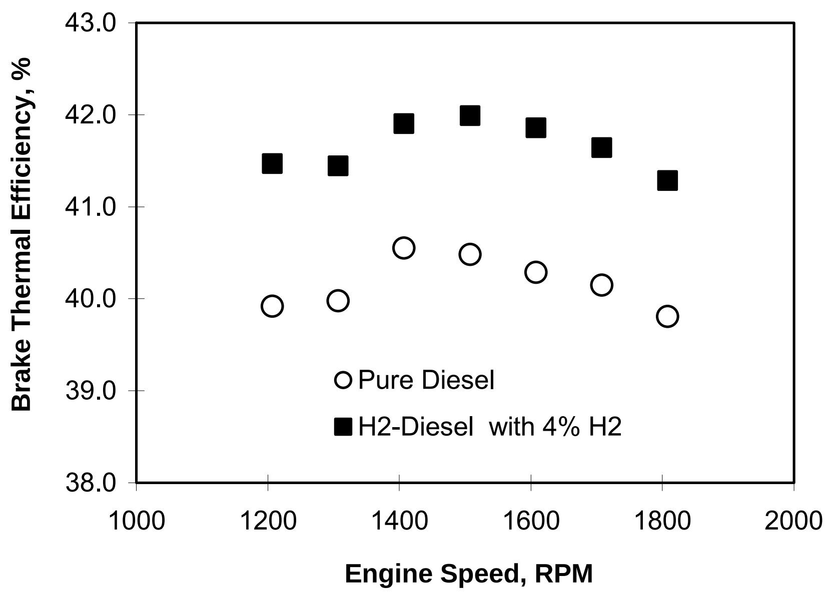

Figure 43 Effect of H_2 Addition and Engine Speed on Brake Thermal Efficiency, Torque=700 ft-lbf, $H_2/(H_2+Air)$=4% vol.

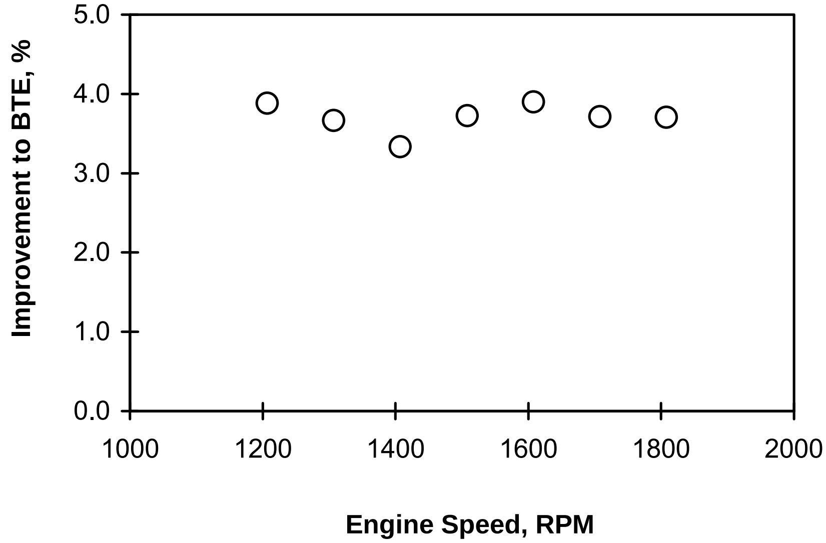

Figure 44 Effect of H_2 Addition and Engine Speed on the Improvement to Brake Thermal Efficiency, Torque=700 ft-lpf.

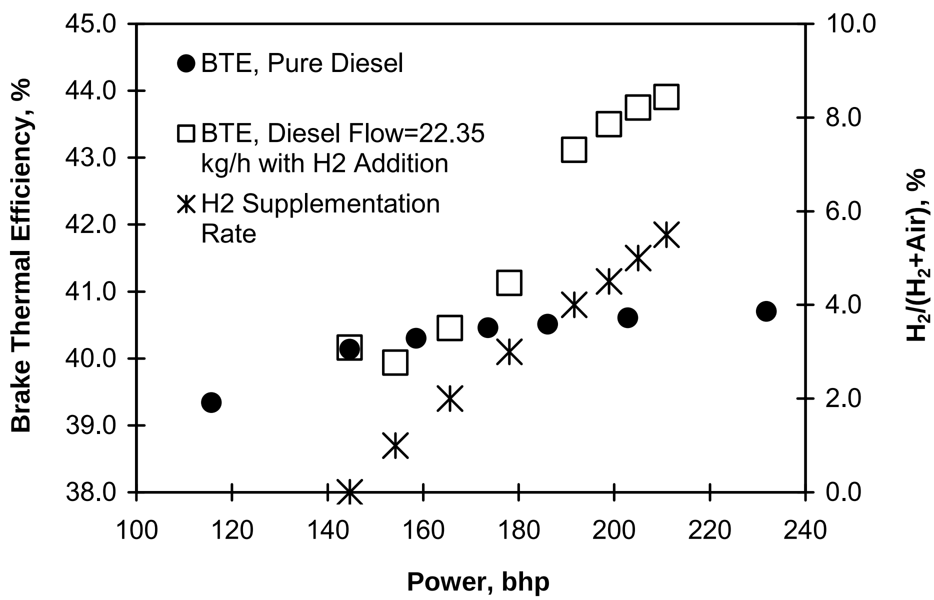

Figure 45 Effect of H_2 Addition on Brake Thermal Efficiency. N=1200 RPM. For Constant Diesel Flow Operation, Engine Load Was Increased by the Addition of H_2

The effect of H_2 addition on the brake thermal efficiency was also investigated while the diesel fuel flow rate was held constant. The engine load was changed by gradually adding more H_2 into the intake mixture. Compared to pure diesel operation, the addition of a small amount of H_2 was found to deteriorate the brake thermal efficiency as shown in Fig. 45. With the addition of 2% H_2, the brake thermal efficiency obtained was comparable to that of pure diesel operation. Further increasing the addition of H_2 beyond 2% was demonstrated improved substantially the brake thermal efficiency. The maximum brake thermal efficiency of 43.9% was obtained with the addition of 5.5% H_2 into the diesel engine.

5.6 Cylinder Pressure and Heat Release Rate

The cylinder pressure was measured and processed to obtain sets of engine cylinder pressure and combustion process. As an example, Fig. 46 shows the effect of H_2 addition on cylinder pressure when operated at 70% load. With the addition of a small amount (<3%) H_2, the cylinder pressure prior to the initiation of combustion was higher than that of pure diesel operation. This was due to the increased intake pressure as shown in Fig. 47. The addition of H_2 beyond 2% reduced gradually the intake pressure and also the cylinder pressure prior to the initiation of combustion. As shown in Fig. 46, the addition of H_2 into the diesel engine was shown to increase the cylinder pressure after the combustion was initiated. This was further demonstrated by examining the variation of the peak cylinder pressure with the addition of H_2. As shown in Fig. 48, the addition of a small amount of H_2 increased the peak cylinder pressure with negligible effect on the phasing when peak cylinder pressure was observed. Further increasing the amount of H_2 beyond 3% continued to increase the peak cylinder pressure and advance phasing when peak cylinder pressure was observed. With the addition of 6% H_2, the peak cylinder pressure was increased from 106.4 bar to 120.5 bar.

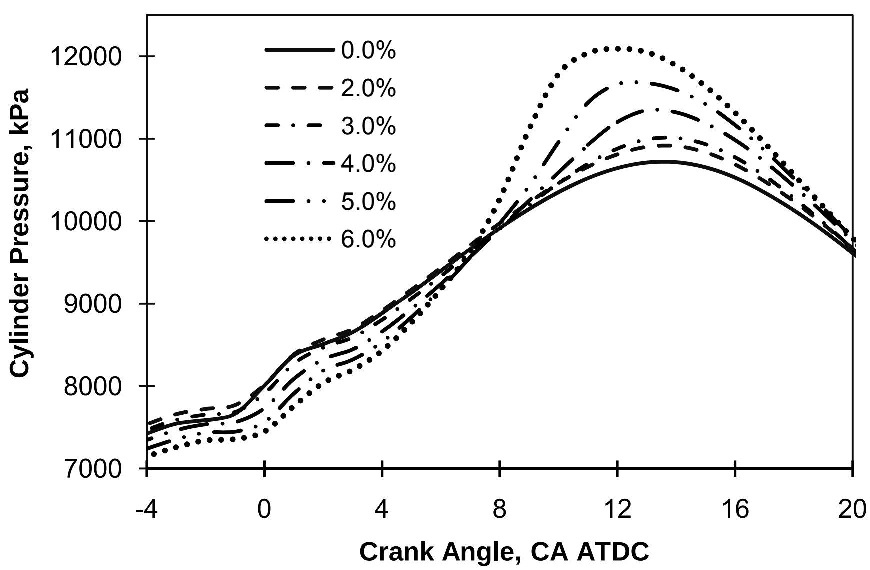

Figure 46 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 70% Load

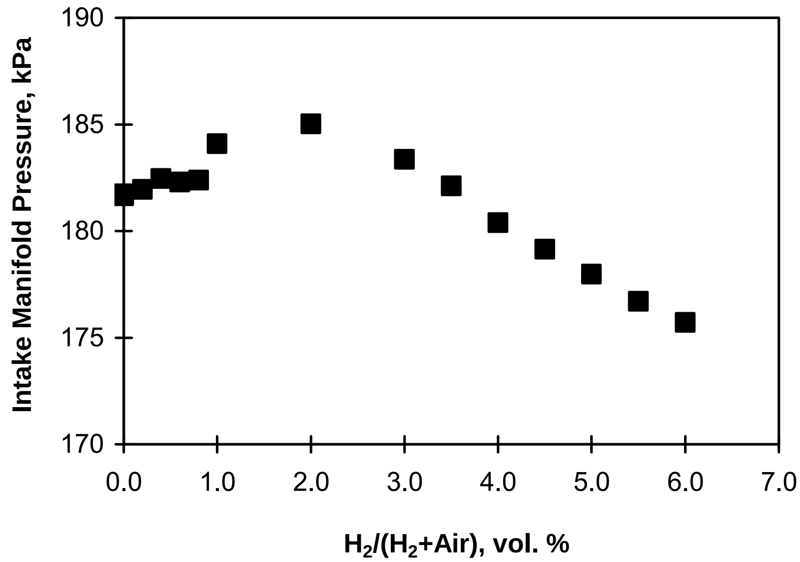

Figure 47 Effect of H_2 Addition on Intake Manifold Pressure, N=1200 RPM, 70% Load

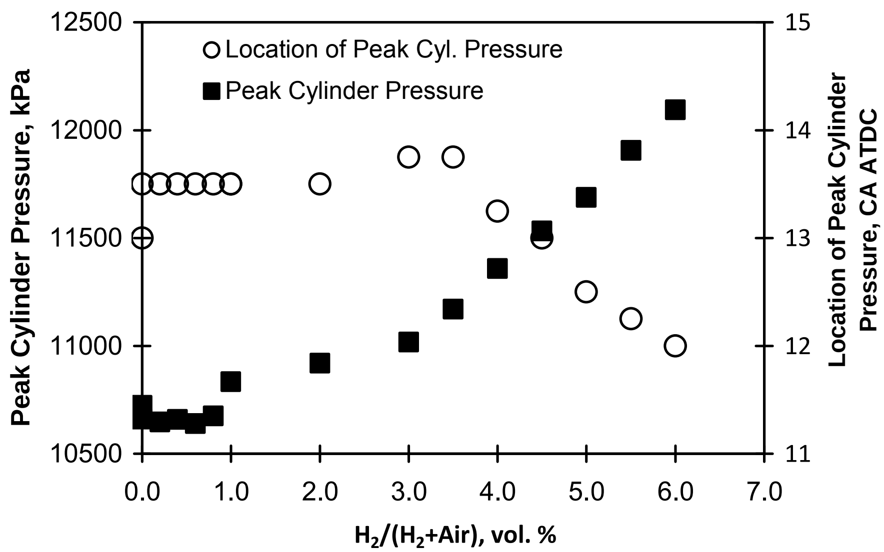

Figure 48 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 70% Load

The cylinder pressure was processed to obtain the heat release rate. As shown in Fig. 49 for premixed combustion, the addition of H_2 into this diesel engine was shown to retard the initiation of premixed combustion and slightly reduce the peak heat release rate obtained in the premixed combustion stage. As shown in Fig. 50, the addition of H_2 into the diesel engine enhanced the mixing controlled diffusion combustion process. When a large amount of H_2 (>4%) was added, the traditional two-stage combustion process of pure diesel operation shown in Fig. 51 became a three-stage combustion process featured with a significantly enhanced heat release process observed at the middle of diffusion combustion. This was further demonstrated in Fig. 52 for the addition of 6% H_2. Similar to pure diesel combustion, the premixed combustion was followed by a diffusion combustion process resulted from the continuous injection, atomization, vaporization and mixing of diesel fuel with air. However, a sudden increase in heat release rate beyond that of traditional diesel diffusion combustion was observed in the middle of diffusion combustion process. Such a “huge” heat release peak was due to the initiation of an H_2-air flame, which burned the H_2 quickly through the propagation of a turbulent flame and released extra energy to that of the diffusion combustion of diesel fuel. The combination of the diesel diffusion combustion and the turbulent flame of H_2-air-diesel fuel mixture created the heat release peak featured with H_2-diesel dual fuel combustion. When the turbulent H_2-air-diesel flame propagated completely through the bulk mixture or quenched due to the slow reaction rate, the diffusion combustion of diesel would continue until completion of diesel combustion, noted as late diesel diffusion combustion. As shown in Fig. 51 and 52, the peak heat release rate of H_2-diesel dual fuel engine was much higher than that of pure diesel combustion. In comparison, the diffusion combustion of pure diesel operation was relatively “flat”. As shown in Fig. 52, the dual fuel diesel engine combustion consisted of premixed diesel combustion, early stage diffusion combustion of diesel, spontaneous combustion of diesel fuel diffusion combustion and turbulent combustion of H_2 and continued late diffusion combustion of diesel fuel after the completion of the H_2 combustion. The featured three-stage combustion process included the premixed combustion of diesel fuel (stage 1), diffusion combustion of diesel fuel (stage 2) and turbulent combustion of H_2-air-diesel mixture with multi ignition points (stage 3).

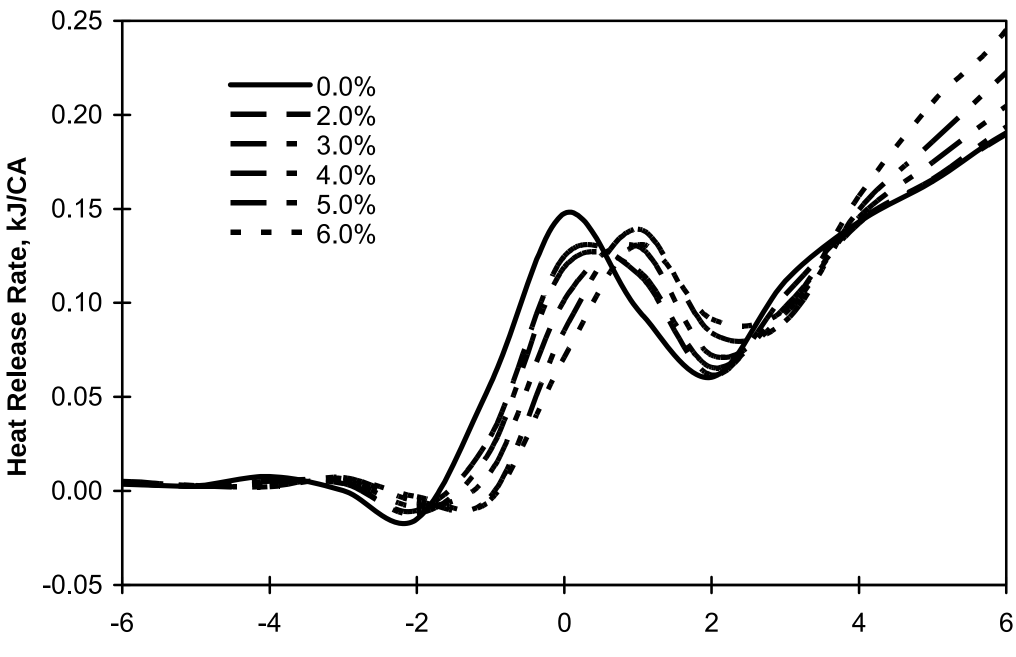

Figure 49 Effect of H_2 Addition on Heat Release Rate of Premixed Combustion, N=1200 RPM, 70% Load

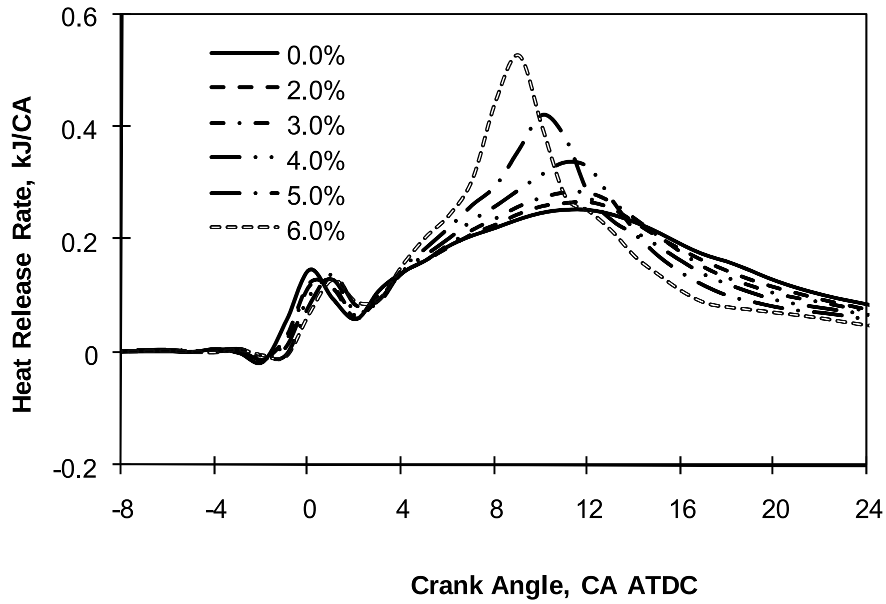

Figure 50 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 70% Load

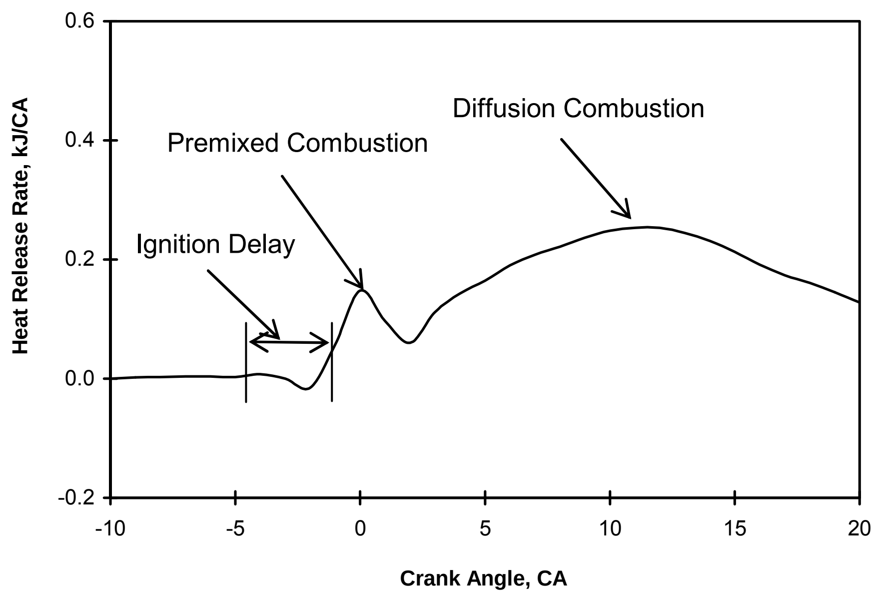

Figure 51 Two-Stage Heat Release Process of Pure Diesel Operation, N=1200 RPM, 70% Load

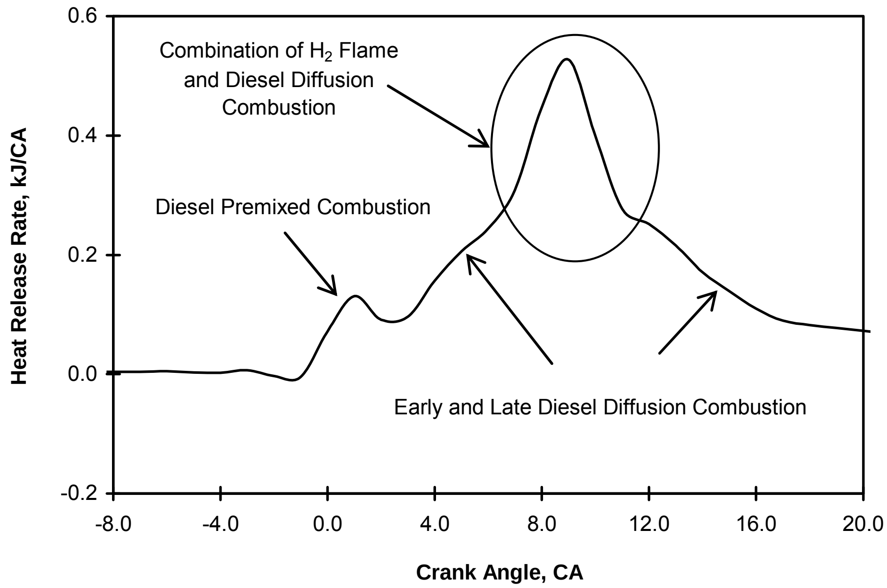

Figure 52 Featured Three-Stage Heat Release Process of H_2-Diesel Dual Fuel Engine, N=1200 RPM, 70% Load, $H_2/(H_2+Air)$=6%

As shown in Fig. 51, the start of injection was defined as crank angle when the first tiny "heat release" bump associated with the injection of diesel was observed. The start of combustion was defined as the crank angle when the heat release rate at premixed stage reached 0.05 kJ/deg.CA. The crank angle period between the start of injection and that of combustion was defined as the ignition delay. As shown in Table 13, the addition of H_2 retarded slightly the injection timing of diesel fuel. Compared to pure diesel operation, the addition of 6% H_2 retarded the start of diesel injection by approximately 1 °CA, which might be due to reduced diesel flow. In comparison, the addition of H_2 was found to have negligible effect on ignition delay as shown in Table 13. When H_2 was added to this 1999 Cummins ISM370 engine at 70% load, the retarding of the premixed combustion was due to the retarded fuel injection.

Table 13 Effect of H_2 Addition on Start of Injection (SOI), Start of Combustion (SOC) and Ignition Delay (ID) when Operated at 1200 rpm, 70% Load

$H_2/(H_2+Air)$, vol. %	SOI, CA ATDC	SOC, CA ATDC	Ignition Delay, CA
0	-5.20	-1.1	4.1
1	-5.10	-0.85	4.25
2	-5.05	-0.75	4.30
3	-5.00	-0.68	4.32
4	-4.90	-0.55	4.35
5	-4.40	-0.37	4.03
6	-4.20	-0.26	3.94

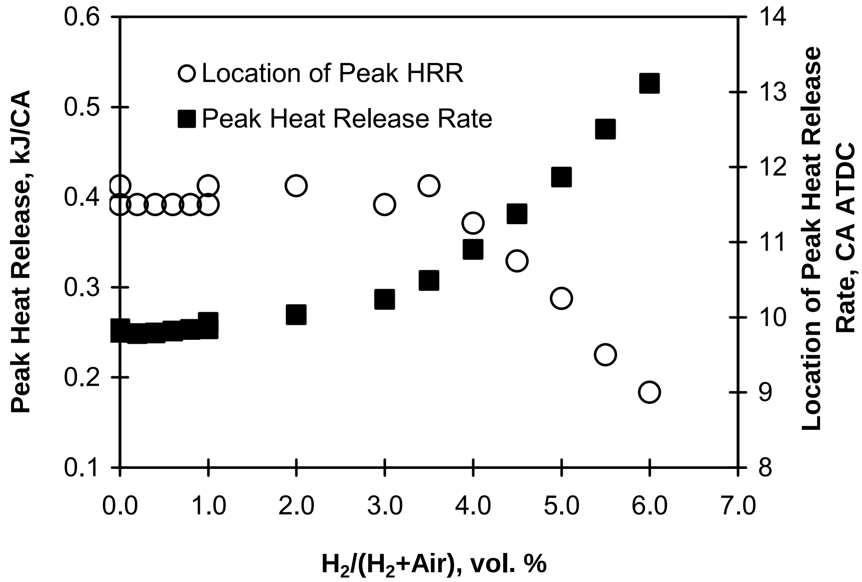

Figure 53 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 70% Load

As shown in Fig. 53, the addition of a small amount of H_2 (<3%) was shown to slightly enhance the peak heat release rate without affecting the phasing when the peak heat release rate was observed. However, further increasing the addition of H_2 beyond 3.5% increased substantially the peak heat release rate and advanced the phasing when the peak heat release rate was observed. This was due to the gradual development of the turbulent flame of H_2-air-diesel at the early stage of diesel diffusion combustion, which released more energy at advanced phasing as shown in Figs. 50 and 52.

Figures 54-58 show the effect of H_2 addition on cylinder pressure, intake pressure, peak cylinder pressure, heat release process, and maximum heat release rate when operated at 30% load. As shown in Fig. 54, the addition of H_2 at 30% load increased slightly cylinder pressure after combustion was initiated. However, its effect on the cylinder pressure prior to the initiation of the combustion was relatively small due to the relatively weak effect of H_2 addition on the intake pressure as shown in Fig. 55. As shown in Fig. 56, the addition of H_2 at 30% load slightly increased the peak cylinder pressure with negligible effect on the phasing when peak heat release rate was observed. As shown in Figs. 57 and 58, the addition of H_2 at 30% load slightly enhanced the premixed combustion and retarded its phasing. In comparison, its addition was shown to have negligible effect on the diffusion combustion process. Compared to high load operation, the peak heat release rate was obtained at the premixed combustion stage when operated at 30% load as shown in Fig. 57.

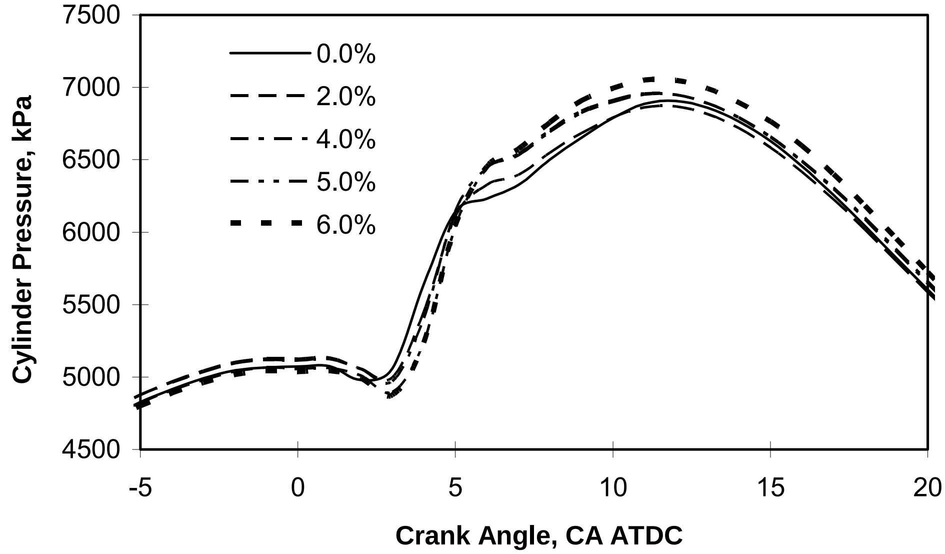

Figure 54 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 30% Load

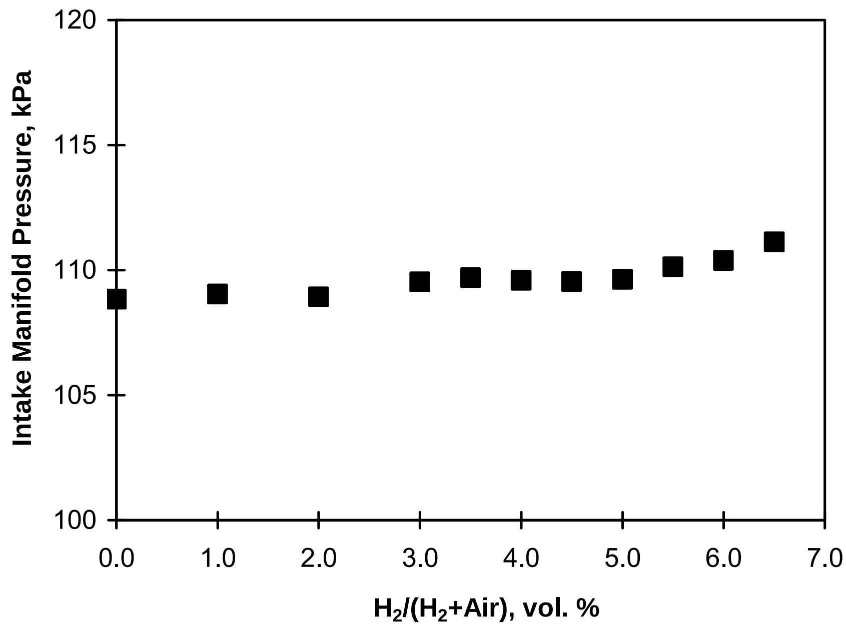

Figure 55 Effect of H_2 Addition on Intake Manifold Pressure, N=1200 RPM, 30% Load

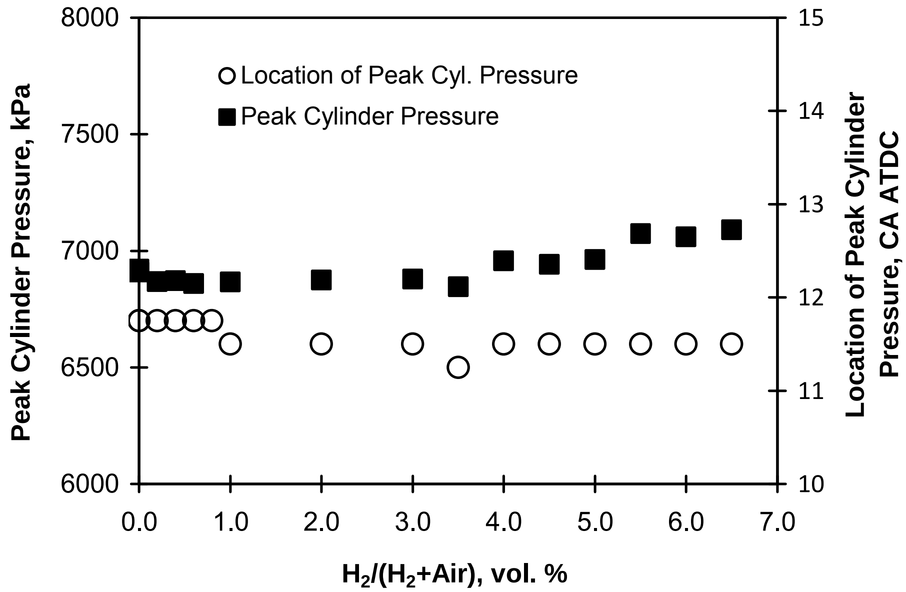

Figure 56 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 30% Load

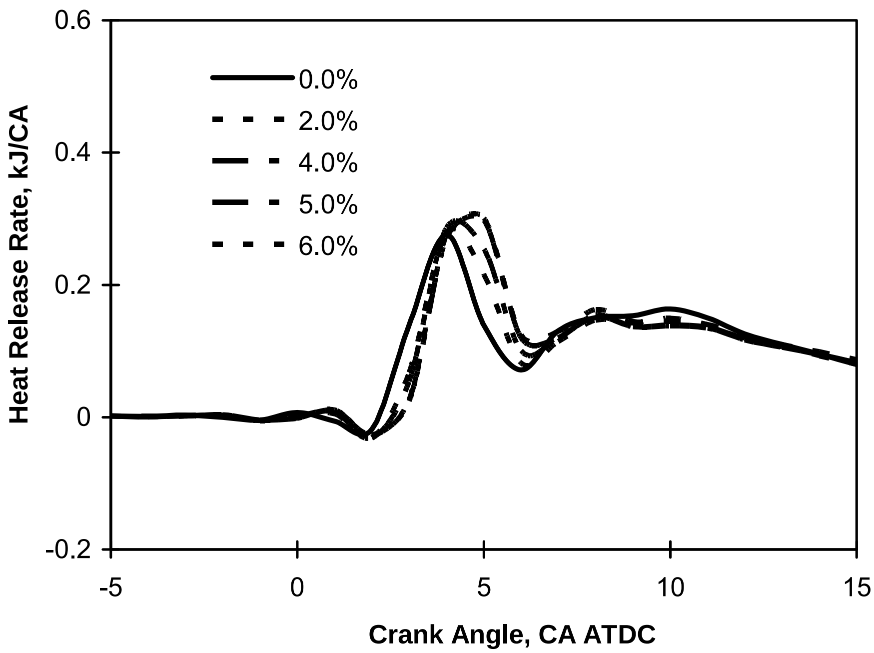

Figure 57 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 30% Load

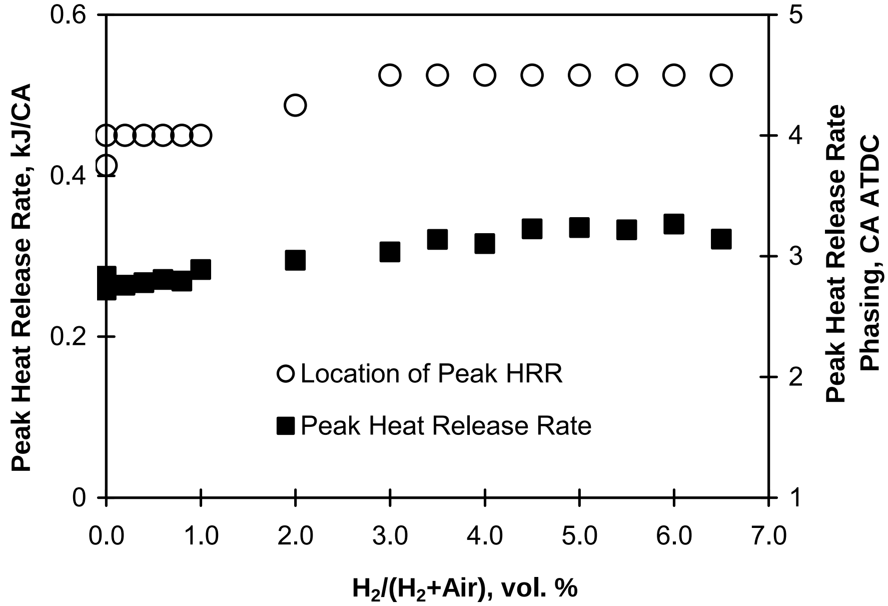

Figure 58 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 30% Load

Figs. 59-63 show the effect of H_2 addition on cylinder pressure, peak cylinder pressure and its phasing, heat release process, and maximum heat release rate and its phasing when operated at 15% load. As shown in Figs. 59 and 60, the addition of H_2 at 15% load was shown to reduce substantially the cylinder pressure after the combustion was initiated. The phasing of the peak cylinder pressure was lightly advanced. This was due mainly to the deteriorated premixed combustion process as shown in Fig. 61. In comparison, the diffusion combustion process was further retarded with the enhanced heat release rate observed at the late stage of diffusion combustion. Similar to 30% load operation, the peak heat release rate was obtained at the premixed combustion stage. As shown in Fig. 62, the addition of a small amount of H_2 (less that 2%) was shown to have negligible effect on the peak heat release rate and its phasing. Increasing the amount of H_2 beyond 3% was shown to reduce substantially the peak heat release rate with negligible effect on the phasing when peak heat release rate was observed. The reduction in the peak heat release rate may be due to the reduction in diesel flow rate. As shown in Fig. 63, the peak heat release rate was found to decrease with reduced diesel flow following the addition of H_2. The reduction in the peak heat release rate at low load operation was due to mainly the reduced diesel flow. Similar to 30% load operation, the peak heat release rate was obtained during the premixed combustion stage at 15% load as shown in Fig. 61.

Table 14 shows the effect of H_2 addition on the injection timing, start of combustion and ignition delay at 15% load operation. The addition of H_2 was shown to have negligible effect on the start of injection and also the ignition delay.

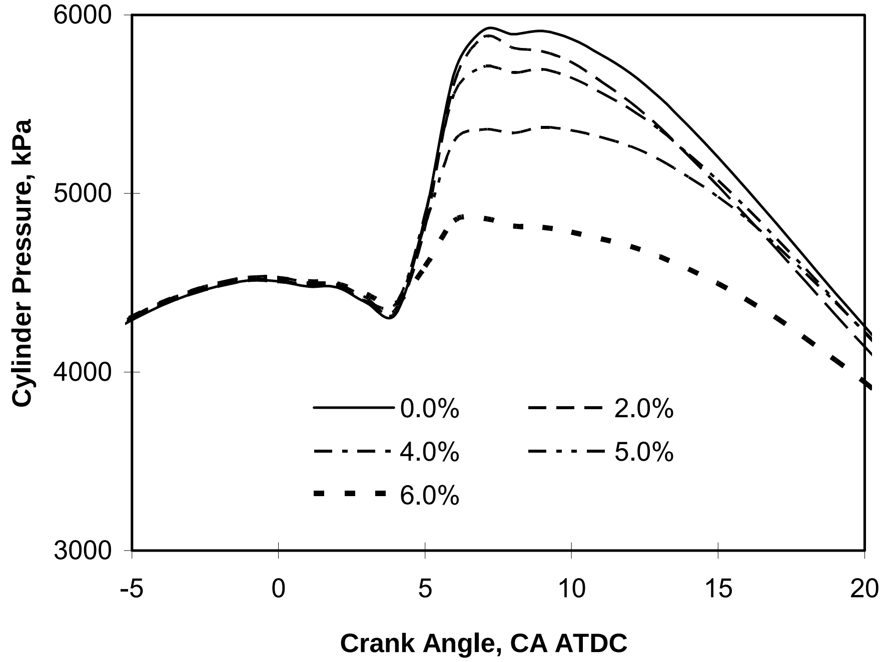

Figure 59 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 15% Load

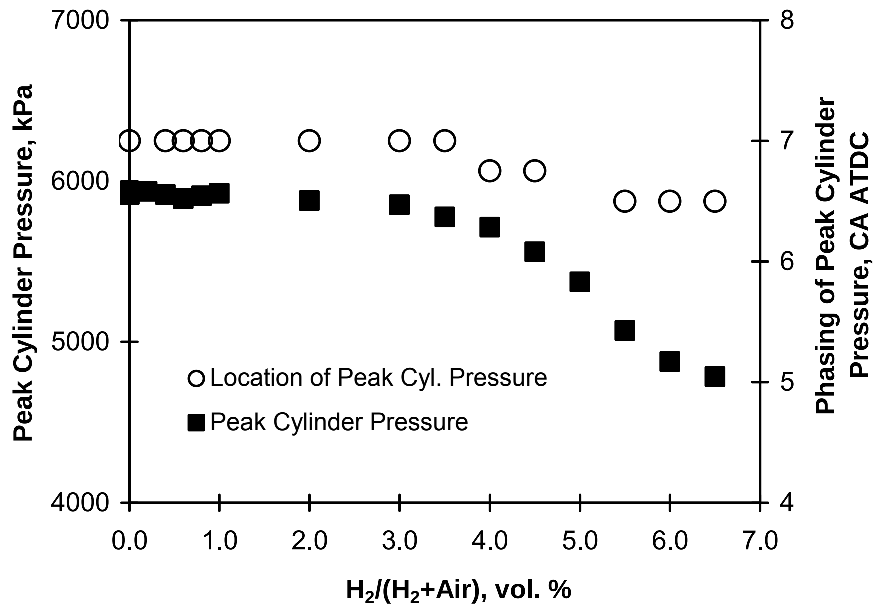

Figure 60 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 15% Load

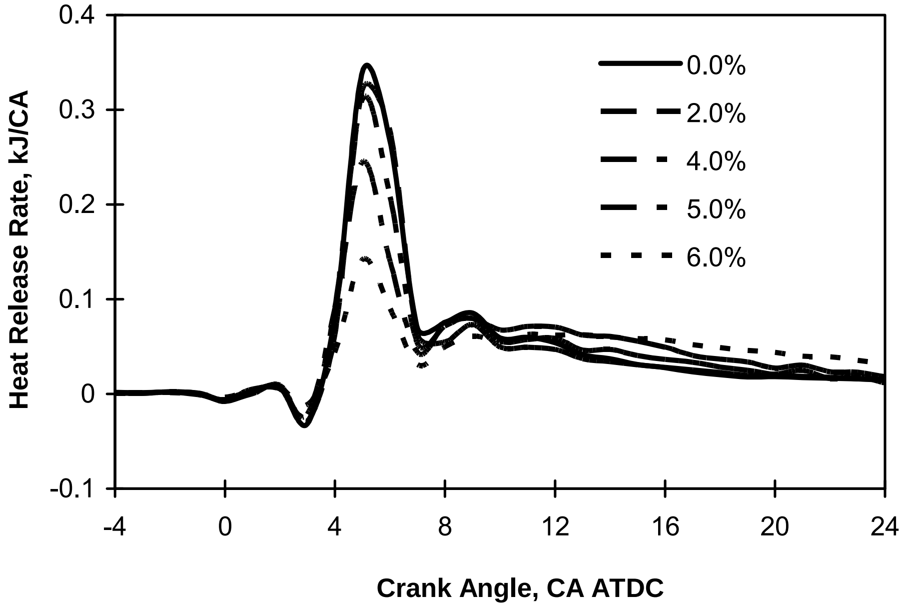

Figure 61 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 15% Load

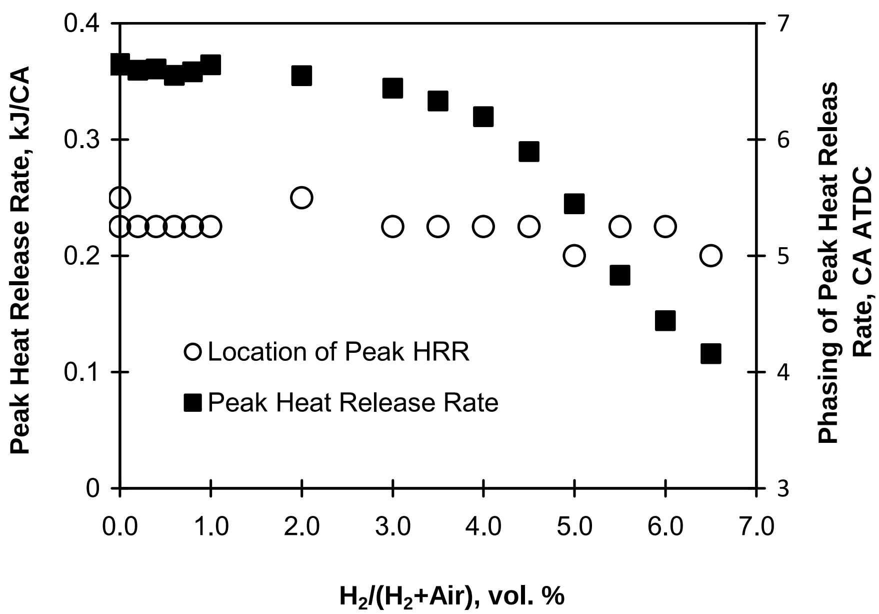

Figure 62 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 15% Load

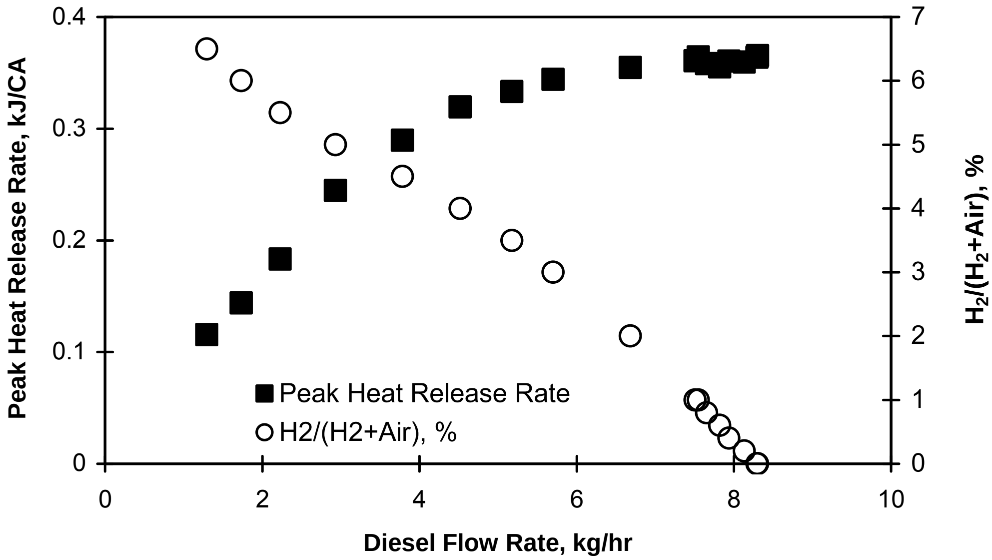

Figure 63 Effect of Diesel Flow Rate on Peak Heat Release Rate of H_2-Diesel Dual Fuel Engine, N=1200 RPM, 15% Load

Table 14 Effect of H_2 Addition on the Start of Injection (SOI), Start of Combustion (SOC) and Ignition Delay when at 15% Load, 1200 RPM

$H_2/(H_2+Air)$, vol.%	SOI, CA ATDC	SOC, CA ATDC	Ignition Delay, CA
0	0.0	3.82	3.82
2	0.1	3.92	3.82
4	0.0	3.75	3.75
5	-0.1	3.72	3.82
6	0.0	4.08	4.08

Figure 64 compares the effect of engine load and the addition of H_2 on the peak heat release rate. The addition of small amount of H_2 (<4%) was shown to have negligible effect on the peak heat release rate. The increased addition of H_2 beyond 4% increased substantially the peak heat release rate at 70% load but reduced substantially the peak heat release rate at 15% load operation. When operated at 30% load, the addition of H_2 was found to slightly increase the peak heat release rate, which was observed at premixed combustion stage as described earlier.

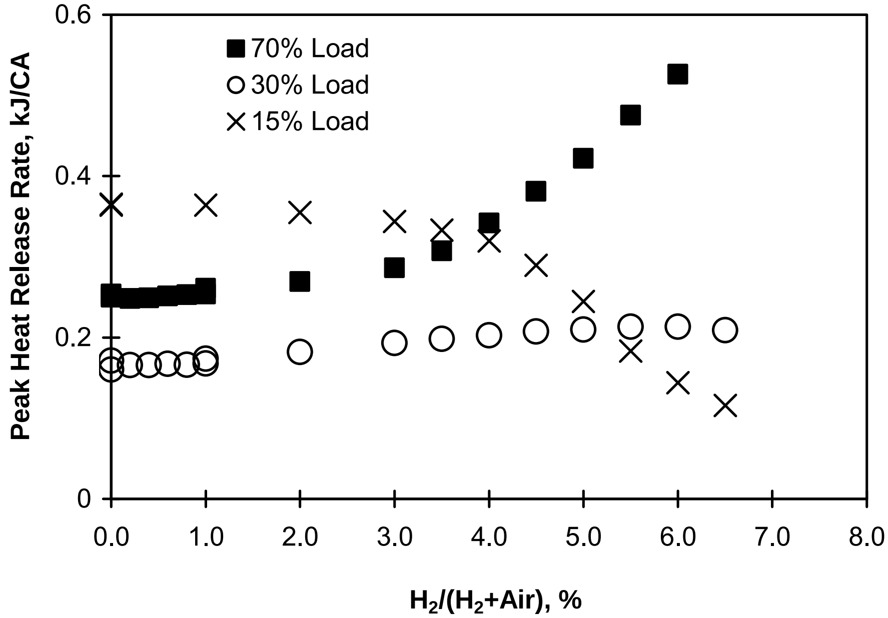

Figure 64 Effect of H_2 Addition and Engine Load on the Peak Heat Release Rate, N=1200 RPM

5.7 Summary

The effect of the addition of H_2, engine load, engine speed and the diesel fuel flow rate on the performance, combustion, and exhaust emissions of a 1999 Cummins ISM370 diesel engine has been experimentally investigated without modifying the engine control and fuel injection strategies. Following is a brief summary based on the limited results obtained in this research:

- The addition of H_2 was found to substantially reduce the PM emissions measured under constant load. When operated at 10% - 70% load, the addition of H_2 can reduce the PM emissions by up to 40% to 65%. When measured using the 13-mode emission cycle, the addition of 2% H_2 increased PM emissions by 15.6%. In comparison, the addition of 4% H_2 reduced the PM emissions by 9.3%.

- The addition of H_2 at medium to high load (>30%) increased substantially the emissions of NO_x. When operated at low load (10%-15%), the addition of a small amount of H_2 had negligible effect on NO_x emissions. However, the addition of a relatively large amount of H_2 (beyond 3%-4% depending on load) at this very narrow operational region reduced substantially the emissions of NO_x. When measured using the 13-mode ESC cycle, the addition of 2% and 4% H2 increased the emissions of NO_x by 5.3% and 15.5%, respectively.

- The addition of H_2 under constant load reduced substantially the emissions of CO with the exception of 70% load. When operated at 70% load, the addition of relatively large amount of H2 increased slightly the emissions of CO. When measured using the 13-mode ESC cycle, the addition of 2% H_2 had a negligible effect on CO emissions. In comparison, the addition of 4% H_2 reduced the emissions of CO by 16%.

- The addition of H_2 at medium to high load (30%-70%) had a negligible effect on the emissions of HC. In comparison, the addition of H_2 at low load (10% -20%) was shown to slightly increase the emissions of HC. When measured using the 13-mode ESC cycle, the addition of 2% and 4% H_2 slightly increased the emissions of HC by 1.5% and 5.6%, respectively.

- The addition of a relatively small amount of H_2 was shown to deteriorate the brake thermal efficiency. However, the addition of relatively large amounts of H_2 at medium to high load improved the brake thermal efficiency. The addition of 6% H_2 was shown to improve the brake thermal efficiency by 14%-4% when operated at 15%-70% load.

- There presents a minimum H_2 supplementation limit for the positive effect on the brake thermal efficiency. The addition of H_2 below this limit reduced the brake thermal efficiency. Such a limit increased with the reduction in engine load. The addition of H_2 at low load (10% load) should not be considered as no improvement to the brake thermal efficiency could be obtained.

- The exhaust emissions of H_2 at low load operation raised a safety concern. The maximum H_2 emission of 1.4% was obtained with the addition of 4.5% H_2 at 10% load operation. The H_2 emissions were substantially reduced with the increase in engine load.

- When operated at high load, the addition of H_2 into this diesel engine affected mainly the diffusion combustion process. In comparison, the addition of H_2 at low load affected mainly the premixed combustion. With the addition of H_2 at large amounts under high load, a featured three-stage combustion process of H_2-diesel dual fuel engine was observed.

- With the addition of H_2, the retarded combustion phasing was mainly due to the retarded fuel injection. In comparison, the effect of H_2 addition had negligible effect on the ignition delay of this heavy-duty diesel engine.

6 Experimental Results and Analysis: 2004 Mack MD11 Diesel Engine

The 2004 Mack MD11 is a turbocharged, 4-stroke, 6 cylinder, heavy-duty diesel engine with cooled EGR system and certified at 2.5 g/bhp-hr NO_x and 0.1 g/bhp-hr PM. As shown in Table 15, the NO_x emissions were 2.02 g/bhp-hr and 2.29 g/bhp-hr when measured using the 13-mode ESC cycle and hot FTP emission cycle, respectively. The engine torque map was measured with/without the installation of the flame arrester. As shown in Fig. 65, the installation of the flame arrester had negligible effect on the engine torque map.

Table 15 2004 Mack MD11 Engine Baseline Emission Test Results (g/bhp-hr)

Cycle	PM	NO_x	NO	NO_2	CO	HC	CO_2
ESC	0.040	2.02	1.95	0.07	0.75	0.073	477.05
FTP (Warm)	0.056	2.29	1.73	0.56	1.11	0.18	521.47

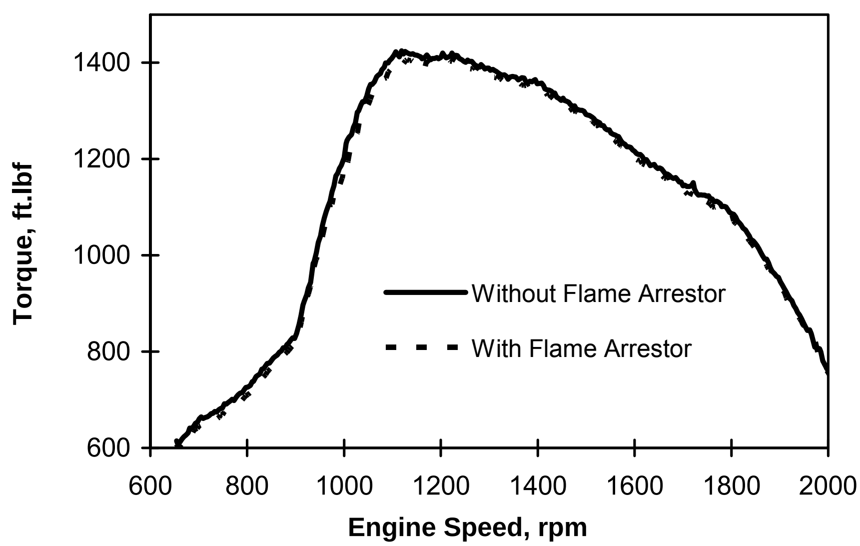

Figure 65 Effect of Flame Arrestor on 2004 Mack MD11 Engine Map

6.1 Test Matrix

The test matrix was designed to investigate the effect of H_2 addition, engine load and diesel flow rate on the engine performance, combustion and exhaust emissions of the 2004 Mack MD11 engine. As shown in Table 16, the effect of the addition of 2% and 4% H_2 into the diesel engine on the exhaust emissions was measured using the steady state 13-mode ESC cycle. Table 17 and Fig. 66 show the set points of 13-mode ESC cycle test determined using the map shown in Fig. 65 with flame arrestor installed. The effects of H_2 addition and engine load on engine performance, cylinder pressure, combustion process, and exhaust emissions were examined at 10% to 70% load and engine speed of 1200 rpm as described in Table 18. The flow rate of H_2 at 6% was also listed in Table 18 to demonstrate the amount of H_2 added to this heavy-duty diesel engine.

Table 16 Test Matrix Hot Start 13 Mode Emission Test

13 Mode Emissions Test	Test 1	Test 2	Test 3
$H_2/(H_2+Air)$, vol.%	0	2	4

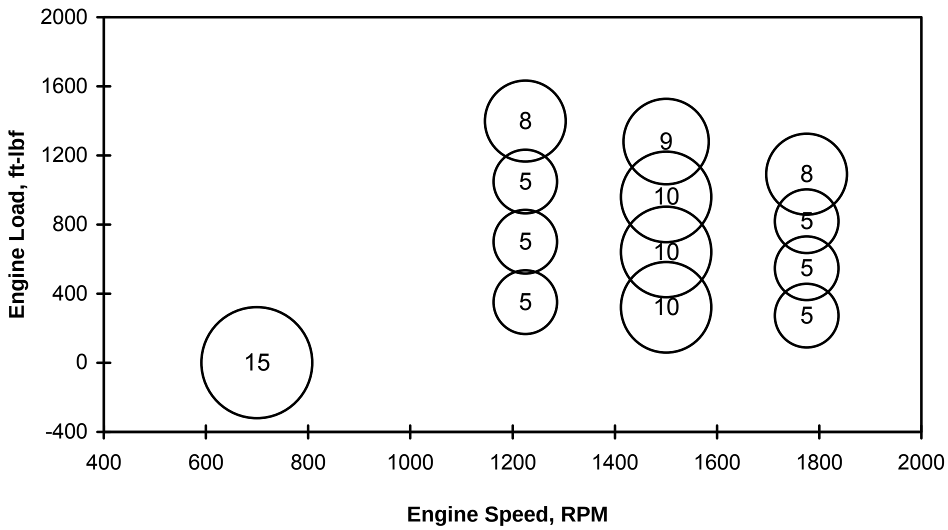

Figure 66 Set Points for 13-Mode ESC Cycle Emission Test for the 2004 Mack MD11 Diesel Engine

Table 17 Set Points for 13-Mode ESC Cycle Emission Test for the 2004 Mack MD11 Diesel Engine

Mode	Speed	Torque, ft-lbf	Weighting Factors, %
1	700	0	15
2	1225	1399	8
3	1501	640	10
4	1501	959	10
5	1225	700	5
6	1225	1049	5
7	1225	350	5
8	1501	1279	9
9	1501	320	10
10	1776	1091	8
11	1776	273	5
12	1776	818	5
13	1776	546	5

Table 18 Test Matrix for Constant Load Operation, Maximum Torque=1399 ft-lbf, N=1200 RPM

Load	$H_2/(H_2+Air)$, vol. %	H_2 Flow at $H_2/(H_2+Air)$=6% vol.	
		kg/hr	L/min.
10%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 3%, 4%, 4.5%, 5%, 5.5%, 6%, 6.5%, 7% and 7.5%	1.42	264.7
15%	0%, 1%, 2%, 3%, 3.5% ,4%, 4.5%, 5%, 5.5%, 6%, 6.5%, 7%	1.52	284.3
20%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 3%, 4%, 4.5%, 5%, 5.5%, 6%, 6.5%, 7% and 7.5%	1.69	315.5
30%	0%, 1%, 2%, 3%, 4%, 4.5%, 5%, 5.5%, 6% and 7%	1.95	363.4
50%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 3%, 3.5, 4%, 4.5%, 5%, 5.5%, 6%, and 6.5%	2.97	553.5
70%	0%, 0.2%, 0.4%, 0.6%, 0.8%, 1%, 2%, 2.5%, 3%, 3.5%, 4%, 4.5%, 5% and 6%	3.84	716.8
100%	0%, 2% (1225 rpm), 3%, 4% (1225 rpm), 5%	N.A.	N.A.

Table 19 Test Matrix for Constant Diesel Fuel Flow Rate Operation

Diesel Flow Rate	$H_2/(H_2+Air)$, vol.%
16.44 kg/hr (30% Load for pure diesel operation)	0%, 1%, 2%, 2.5%, 3%, 4%, 4.5%, 5%, 5.5%, 6%, 6.5%, 7%
25.05 kg/hr (50% Load for pure diesel operation)	0%, 1%, 2%, 2.5%, 3%, 3.5, 4%, 4.5%, 5%, 5.5%

Table 19 shows the operating conditions when diesel flow rate was kept constant at 1200 RPM with the engine load altered by the increasing addition of H_2. In this test, the flow rate of diesel fuel was kept approximately constant at 16.44 kg/hr and 25.05 kg/hr, respectively, which corresponded to 30% and 50% load when operated with pure diesel.

6.2 13-Mode Exhaust Emissions

Table 20 shows the effect of the addition of H_2 addition at 2% and 4% on regulated emissions measured using the steady-state 13-mode ESC cycle. As shown in Table 20, the addition of 2% and 4% H_2 into this 2004 Mack MD11 diesel engine reduced the emission of PM from 0.040 g/bhp-hr to 0.033g/bhp-hr and 0.029 g/bhp-hr, respectively. Compared to pure diesel operation, the PM emissions were reduced by 17.5% and 27.5%, respectively.

As shown in Tables 20 and 21, the addition of 2% H_2 into this turbo-charged diesel engine with cooled EGR had negligible effect on the emissions of NOx. In comparison, the addition of 4% H2 increased the emissions of NOx from 2.02 g/bhp-hr and 2.10 g/bhp-hr. Compared to pure diesel operation, the emissions of NO_x were slightly increased 4.0%. In comparison, the addition of 2% and 4% H_2 into the 1999 Cummins ISM370 engine increased the emissions of NO_x emissions by 5.3% and 15.5%, respectively. It was evident that the addition of H_2 into the 2004 Mack MD11 engine had less effect on NO_x emissions compared to that of the 1999 Cummins ISM370 diesel engine.

Table 20 Effect of H_2 Addition on Exhaust Emissions Measured using 13-Mode ESC Cycle, (g/bhp-hr)

$H_2/(H_2+Air)$, vol. %	PM	NO_x	NO	NO_2	CO	HC	CO_2
0% H_2	0.040	2.02	1.95	0.07	0.75	0.073	477.05
2% H_2	0.033	2.03	1.35	0.68	0.73	0.079	432.2
4% H_2	0.029	2.10	1.46	0.63	0.81	0.072	360.33

Table 21 Effect of H_2 Addition on NO_x Emissions of the 2004 Mack MD11 and 1999 Cummins ISM370 Engine Measured Using 13-Mode ESC Cycle (g/bhp-hr)

$H_2/(H_2+Air)$, vol. %	2004 Mack MD11		1999 Cummins ISM370	
	NO_x Emissions, g/bhp-hr	Changes	NO_x Emissions, g/bhp-hr	Changes
0% H_2	2.02	N.A.	3.41	N.A.
2% H_2	2.03	+ 0.5%	3.59	+5.3%
4% H_2	2.10	+4.0%	3.94	+15.5%

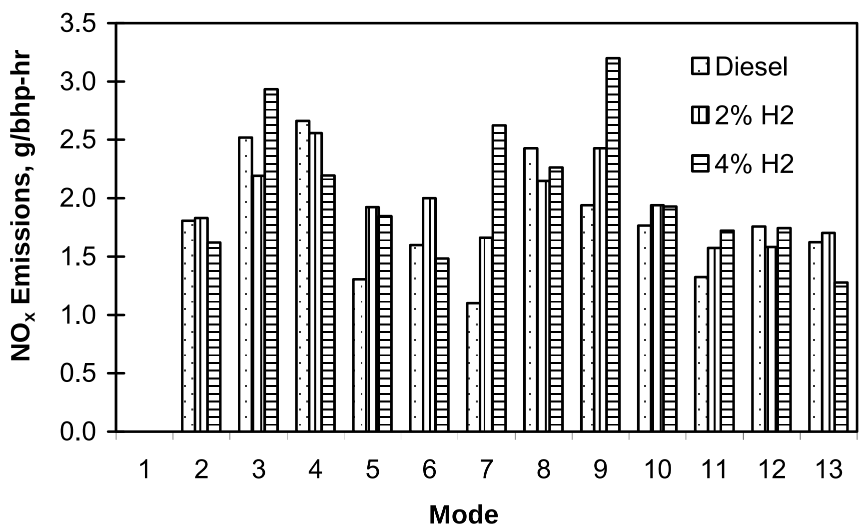

Figure 67 Effect of H_2 Addition on NO_x Emissions, 13-Mode ESC Cycle

The detailed effect of H_2 addition on NO_x emissions of each mode is shown in Fig. 67. The addition of 2% H_2 into the 2004 Mack MD11 diesel engine was shown to increase NO_x emissions at mode 5, 6, 7, 9, 10, 11, and 13. The addition of 4% H_2 increased the emissions of NOx at mode 3, 5, 7, 9, 10 and 11.

As shown in Table 20, the addition of 2% H_2 reduced CO emissions from 0.75 g/bhp-hr to 0.73 g/bhp-hr (-2.7%). However, the addition of 4% H_2 was found to increase the CO emissions from 0.75g/bhp-hr to 0.81 g/bhp-hr (+8%). As shown in Fig. 68, the substantial increase in CO emissions for 4% H_2 operation were observed at modes 2 and 6, the high load operation as shown in Table 17.

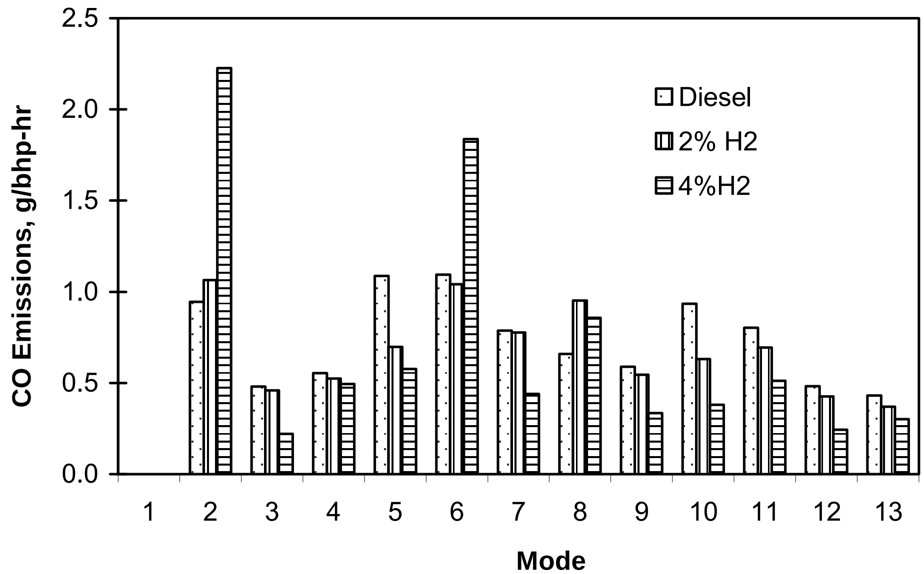

Figure 68 Effect of H_2 Addition on CO Emissions, 13-Mode ESC Cycle

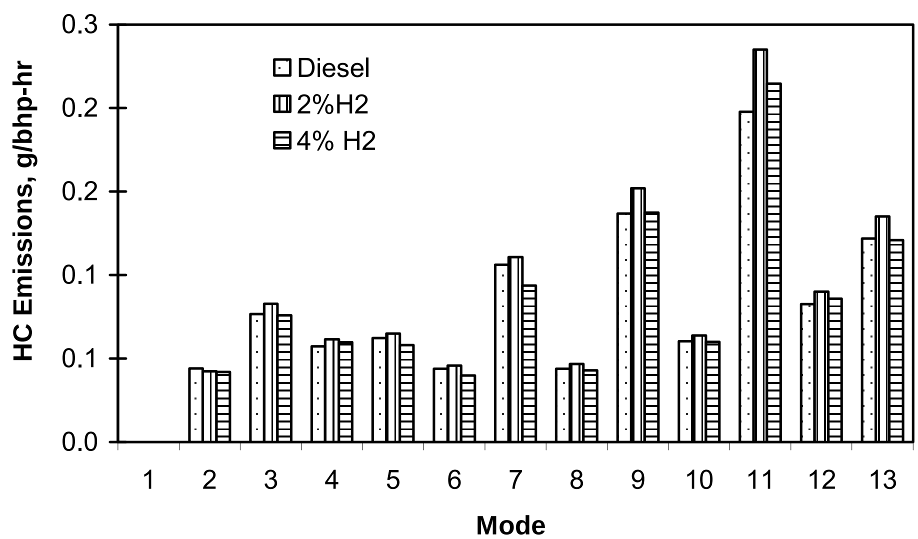

Figure 69 Effect of H_2 Addition on the Emissions of HC, 13-Mode ESC Cycle

The addition of 2% H_2 was found to increase HC emissions from 0.073 g/bhp-hr to 0.079 g/bhp-hr (+8.2%). As shown in Fig. 69, the addition of 2% H_2 was found to significantly increase HC emissions at modes 3, 9, 11, and 13. However, the addition of 4% H_2 had a negligible effect on the emissions of THC as shown in Table 20. As shown in Fig. 69, the addition of 4% H_2 had lower or comparable HC emissions compared to pure diesel operation except mode 11.

6.3 Effect of H_2 Addition and Engine Load on Exhaust Emissions

6.3.1 NO_x Emissions

As shown in Fig. 70 for 10% load operation, the addition of a small amount of H_2 was found to slightly reduce NO_x emissions until the amount of H_2 supplemented reached 4% in the intake mixture. Further increasing the addition of H_2 beyond 4% reduced substantially the emissions of NO_x. This was consistent with the experimental data obtained using the 1999 Cummins ISM370 diesel engine. Similar to that of the Cummins engine operation, the addition of H_2 was found to enhance the conversion of NO to NO_2 as shown in Fig. 70. As shown in Fig. 71, the addition of H_2 at 10% load reduced the NO_x emissions evaluated on molar basis.

Figs. 72 and 73 show the effect of H_2 addition on NO_x emissions when operated at 15% and 20% load, respectively. The addition of H_2 at 15% load was increased slightly the emissions of NO_x. As shown in Fig. 72, further increasing the addition of H_2 beyond 6% reduced the emissions of NO_x. As shown in Fig. 73, the addition of H_2 at 20% load had negligible effect on NO_x emissions with exception of about 4.5% and 5.0%, under which the NO_x emissions was slightly reduced. Further increasing the addition of H_2 beyond 5% was shown to gradually increase the exhaust emissions of NO_x.

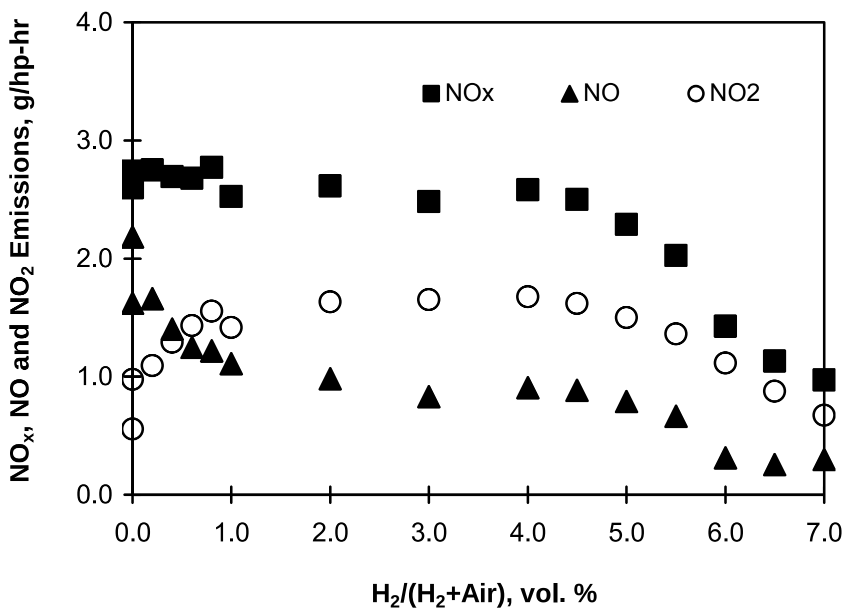

Figure 70 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 10% Load

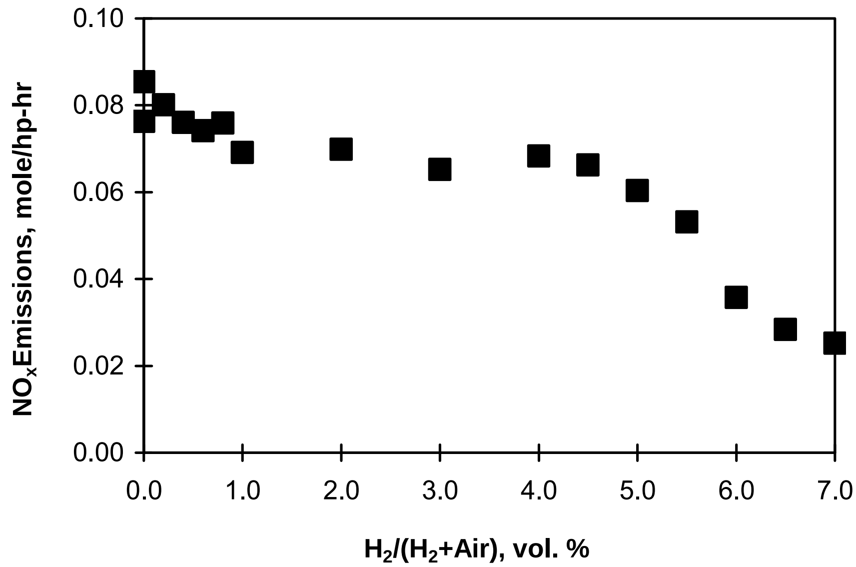

Figure 71 Effect of H_2 Addition on NO_x Emissions (mole/bhp-hr), N=1200 RPM, 10% Load

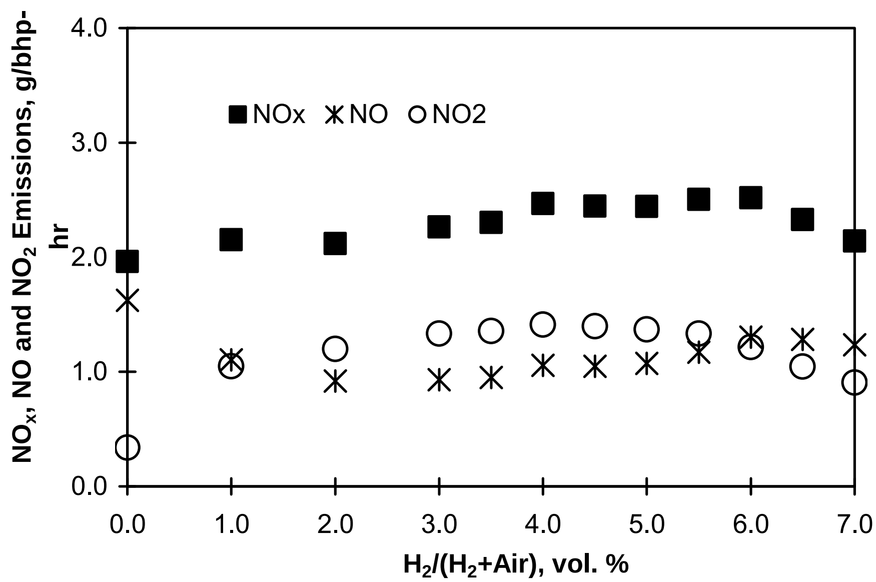

Figure 72 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 15% Load

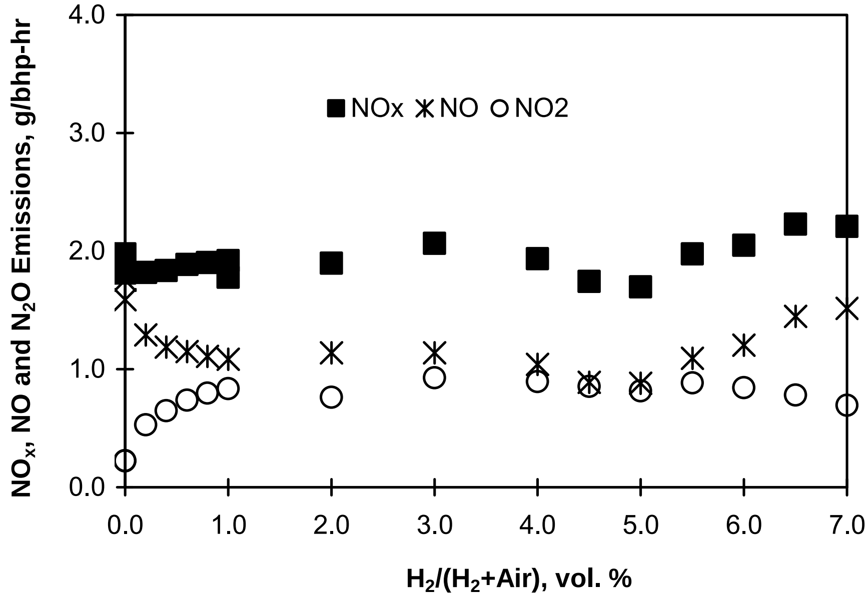

Figure 73 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 20% Load

As shown in Fig. 74 for 30% load operation, the addition of a small amount of H_2 reduced slightly the emissions of NO_x with its minimum value observed with the addition of 4.5% H_2. Further increasing the amount of H_2 added beyond 5% gradually increased the emissions of NO_x. The NO_x emissions obtained at 6% H_2 addition was comparable to that of pure diesel operation.

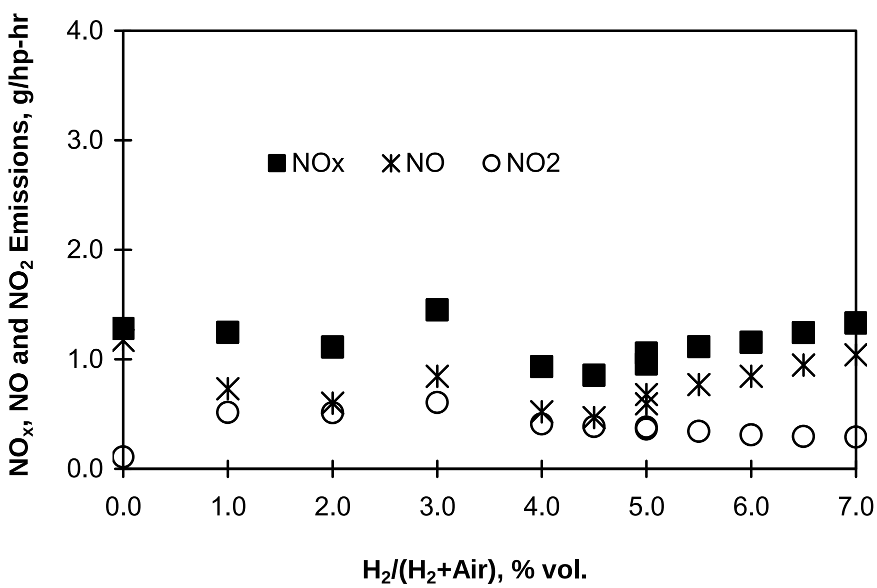

Figure 74 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 30% Load

Fig. 75 shows the effect of H_2 addition on NO_x emissions measured at 50% load operation. The addition of 0.2% and 0.4% H_2 was shown to have negligible effect on NO_x emissions. However, a sudden increase in NO_x emission was observed for the addition of 0.6%, 0.8%, 1% and 2% H_2. It was believed that the sudden increase in NO_x emissions was due to the sudden change in the variable-geometry gas turbine (VGT) and EGR positions as a result of the control strategies, which depended to a large extent on engine speed and the flow rate of diesel fuel as an indication of engine load. As shown in Fig. 76, the increased NO_x emissions were accompanied with increased air flow, indicating the reduced EGR rate. If VGT and EGR system responded properly, the addition of a small amount of H_2 should have negligible effect on NO_x emissions. As shown in Fig. 75, the addition of H_2 beyond 3% gradually increased the emissions of NO_x and NO compared to that of pure diesel operation.

Fig. 77 shows the effect of H_2 addition on the emissions of NO_x, NO and NO_2 at 70% load operation. The addition of H_2 was found to reduce the emissions of NO_x and NO with their minimum values observed with the addition of 4.5% H_2. Further increasing the amount of H_2 beyond 4.5% gradually increased the emissions of NO_x and NO accompanied with further reduction in NO_2 emissions. As shown in Fig.78, the addition of H_2 was found to have negligible effect on NO_x emissions for full load operation at 1200 rpm.

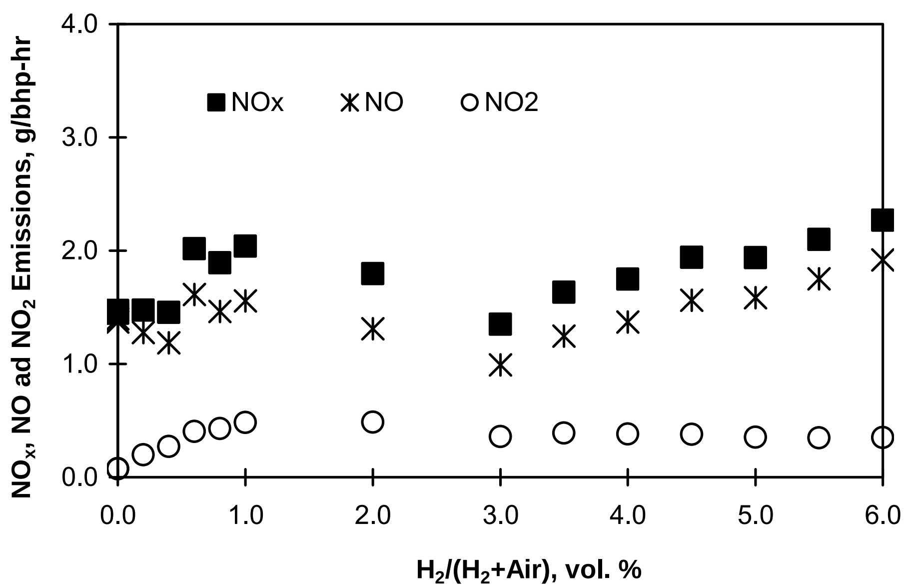

Figure 75 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 50% Load

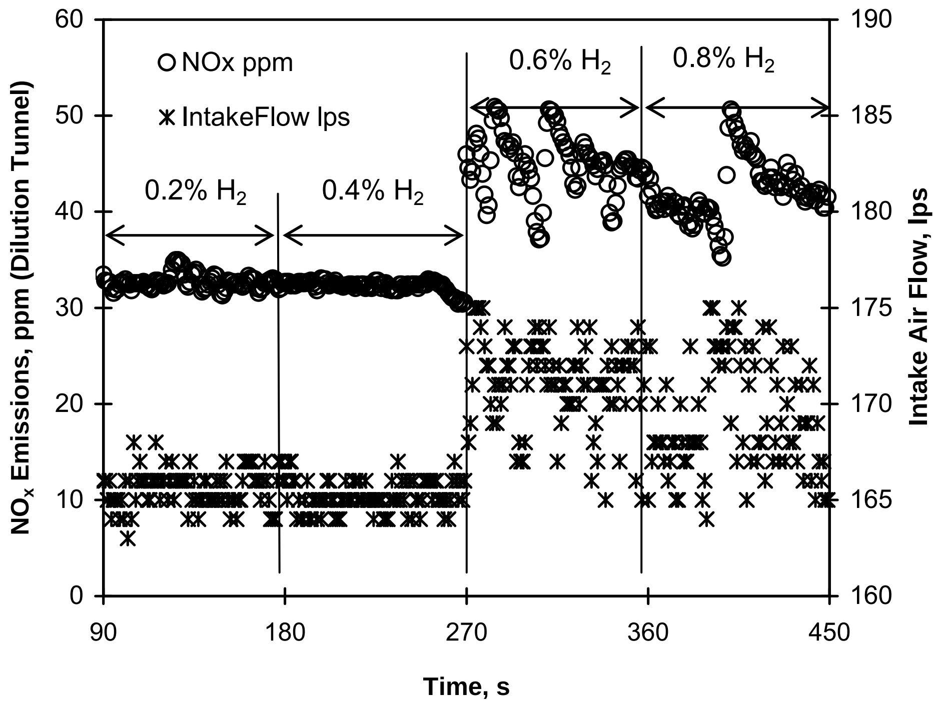

Figure 76 Effect of H_2 Addition on NO_x Emissions and Intake Air Flow (Liter per second (lps)) Measured Continuously, N=1200 RPM, 50% Load

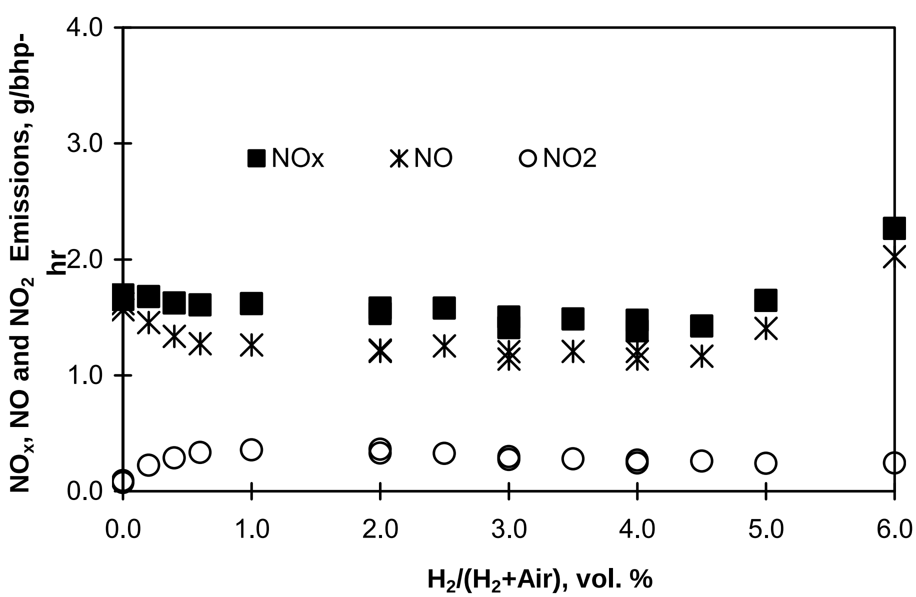

Figure 77 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 70% Load

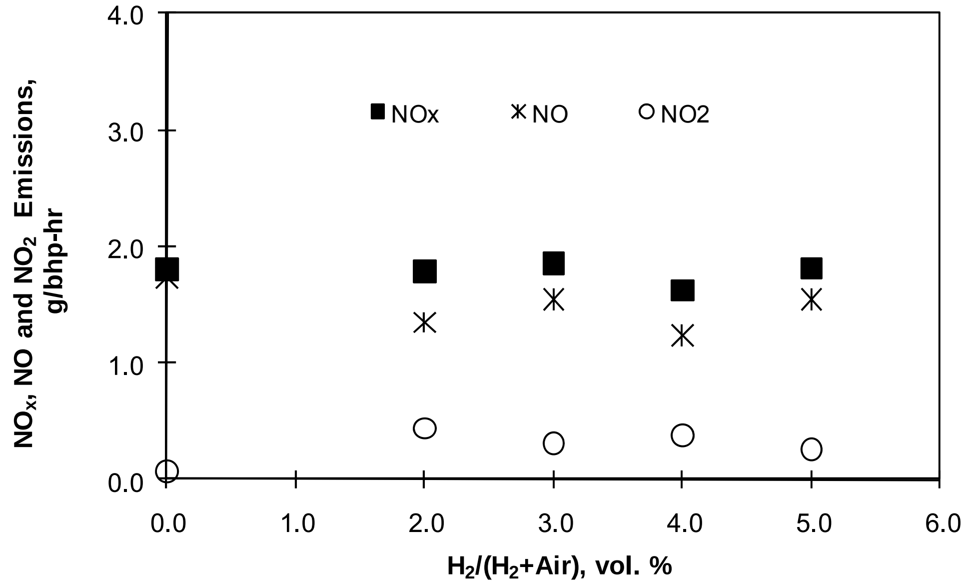

Figure 78 Effect of H_2 Addition on the Emissions of NO_x, NO and NO_2, N=1200 RPM, 100% Load. (2% and 4% Data was Mined from 13-mode ESC Cycle Measured at 1225 rpm.)

6.3.2 PM Emissions

The effect of H_2 additions on PM emissions is shown in Figs. 79-84 for the operation at 10% - 70% load. Similar to that of the 1999 Cummins ISM370 engine, the addition of H_2 to this 2004 Mack MD11 diesel engine reduced substantially the emissions of PM. As shown in Figs. 79, 80 and 81 for low load (10%-20%) operation, the addition of a small amount of H_2 (<3%) reduced substantially the emissions of PM. Further increasing the amount of H_2 added, continued to reduce PM emissions but at a lower rate. Figs. 82, 83 and 84 show the effect of H_2 addition on PM emissions measured at medium to full load. The addition of H_2 gradually reduced the emissions of PM. As shown in Fig. 84, the addition of H_2 was shown to have negligible effect on PM emissions when operated at full load operation.

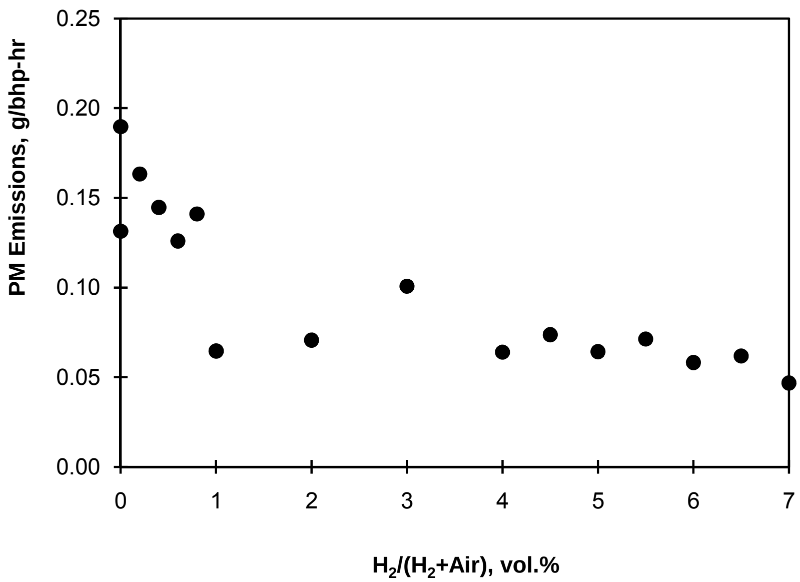

Figure 79 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 10% Load

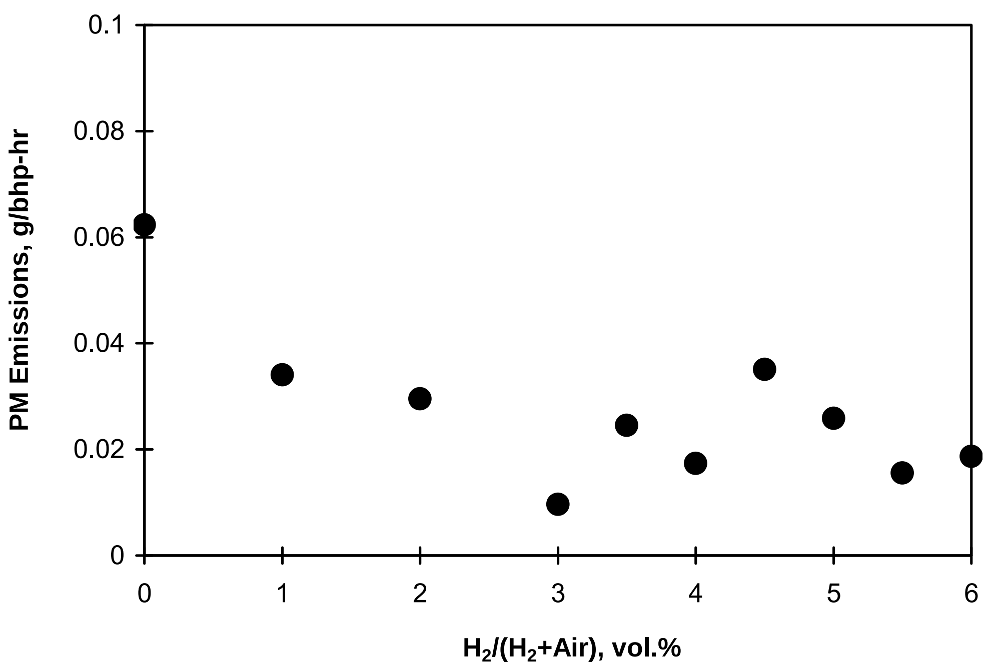

Figure 80 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 15% Load

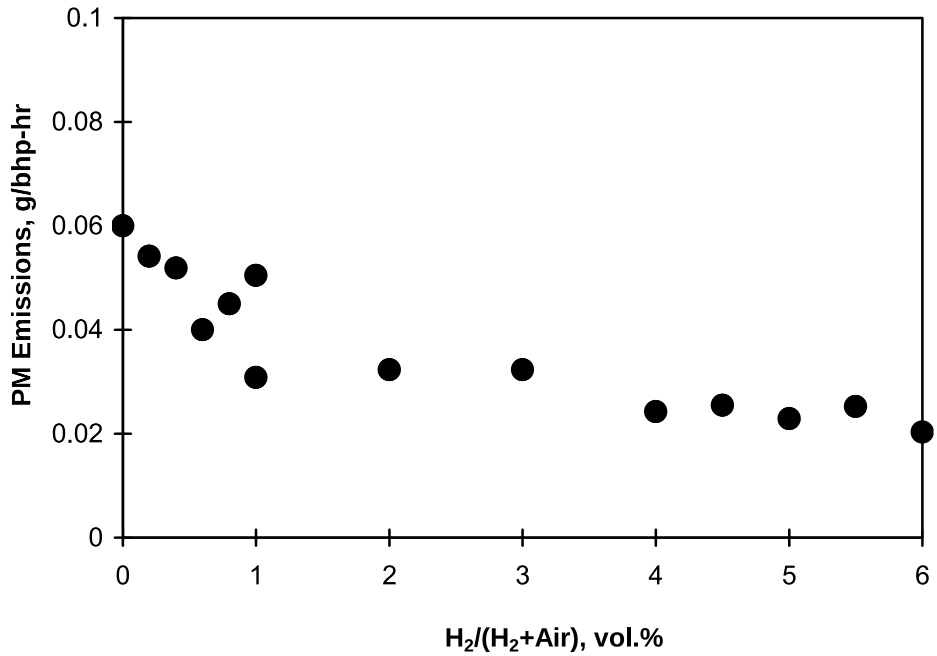

Figure 81 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 20% Load

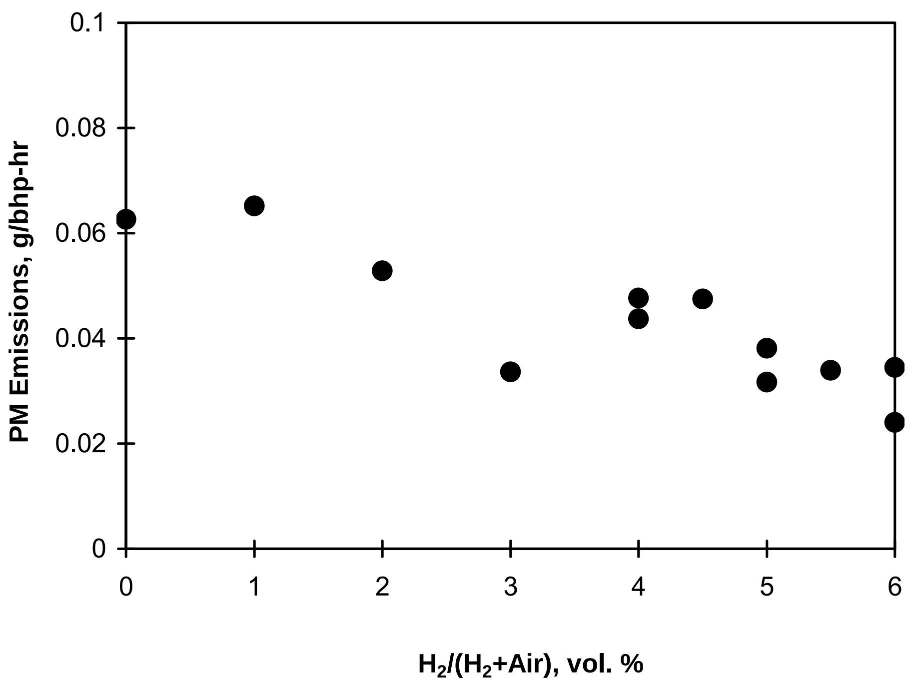

Figure 82 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 30% Load

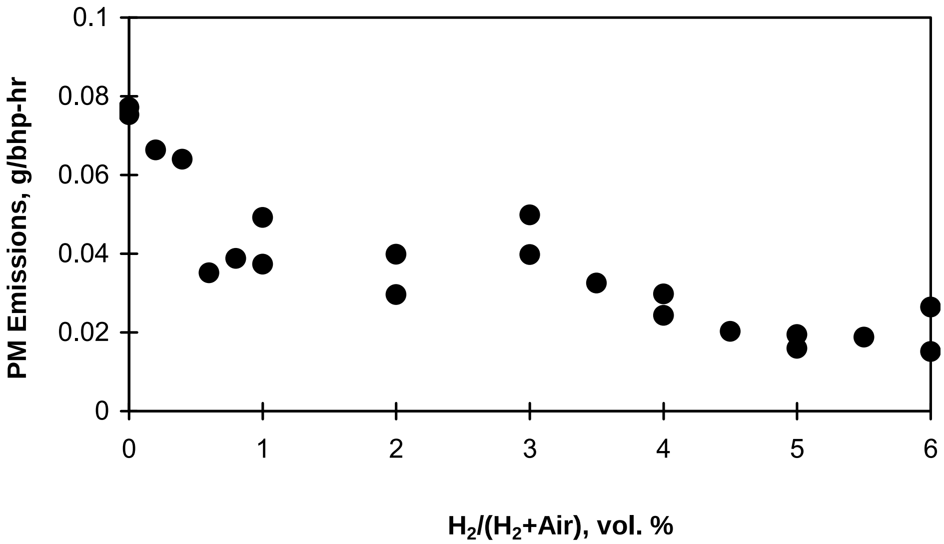

Figure 83 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 50% Load

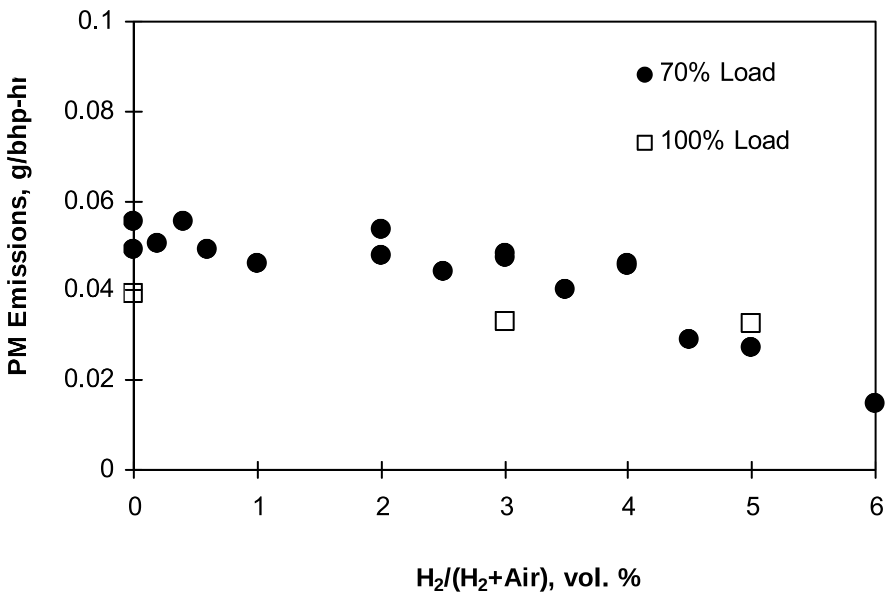

Figure 84 Effect of H_2 Addition on PM Emissions, N=1200 RPM, 70% and 100% Load

6.3.3 CO Emissions

Fig. 85 shows the effect of the addition of H_2 into this diesel engine on the emissions of CO when operated at low to medium load. The addition of H_2 reduced substantially the emissions of CO at low to medium load operation. This was due to the reduction in the amount of diesel fuel burned and the enhancement of H_2 on oxidation of CO to CO_2. However, the addition of a small amount of H_2 at high load operation increased the emissions of CO with its maximum value observed with the addition of H_2 at 4% as shown in Fig. 86. Further increasing the amount of H_2

added reduced substantially the emissions of CO. This could be due to the development of a healthy H_2 flame, which burned the CO survived the main combustion process of diesel fuel.

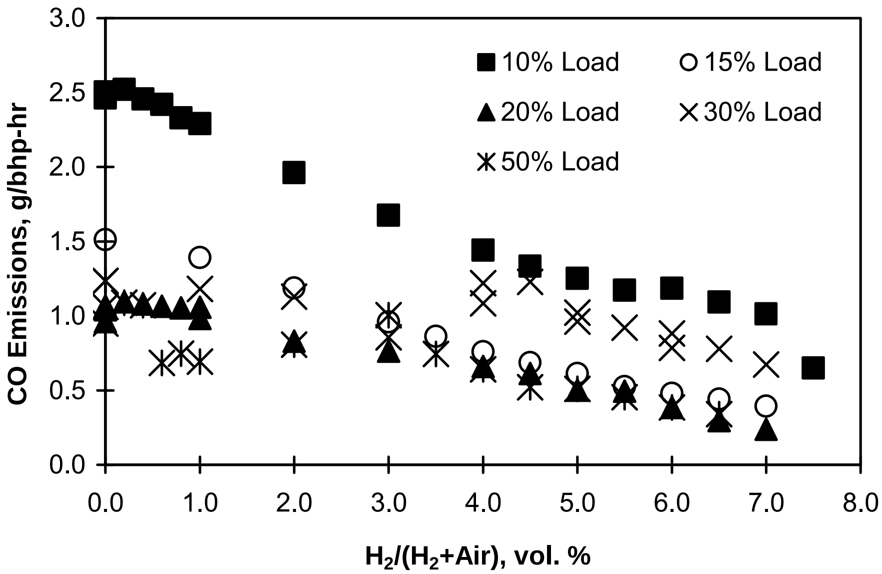

Figure 85 Effect of H_2 Addition and Engine Load on CO Emissions for 10% to 50% Load, N=1200 RPM

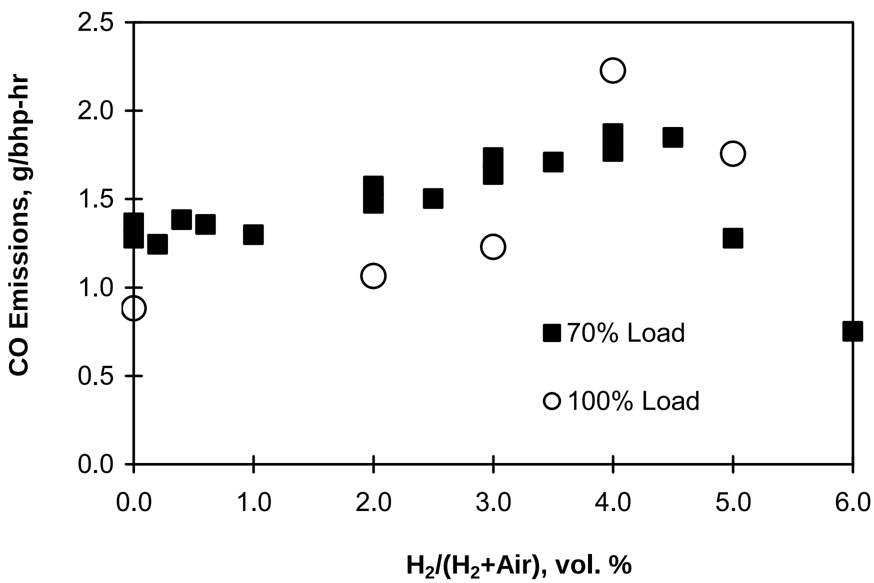

Figure 86 Effect of H_2 Addition and Engine Load on CO Emissions for 70% and 100% Load, N=1200 RPM

6.3.4 HC Emissions

As shown in Fig. 87, the addition of a small amount of H_2 into the intake mixture was found to have negligible effect on the emissions of HC for 10% load operation. Increasing the amount of

H_2 beyond 5% reduced the emissions of HC. In comparison, the addition of H_2 at other loads reduced slightly the emissions of HC.

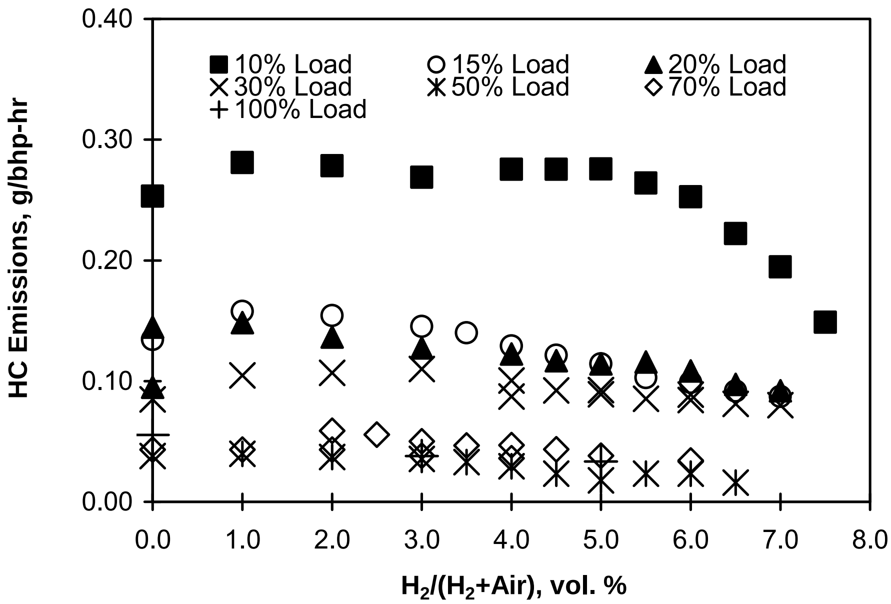

Figure 87 Effect of H_2 Addition and Engine Load on the HC Emissions, N=1200 RPM

6.3.5 CO_2 Emissions

Fig. 88 shows the effect of H_2 addition on the emissions of CO_2. As expected, the addition of H_2 into the diesel engine gradually reduced the emissions of CO_2. This was due to the reduction in diesel fuel flow and also the improvement in brake thermal efficiency when a relatively large amount of H_2 was supplemented.

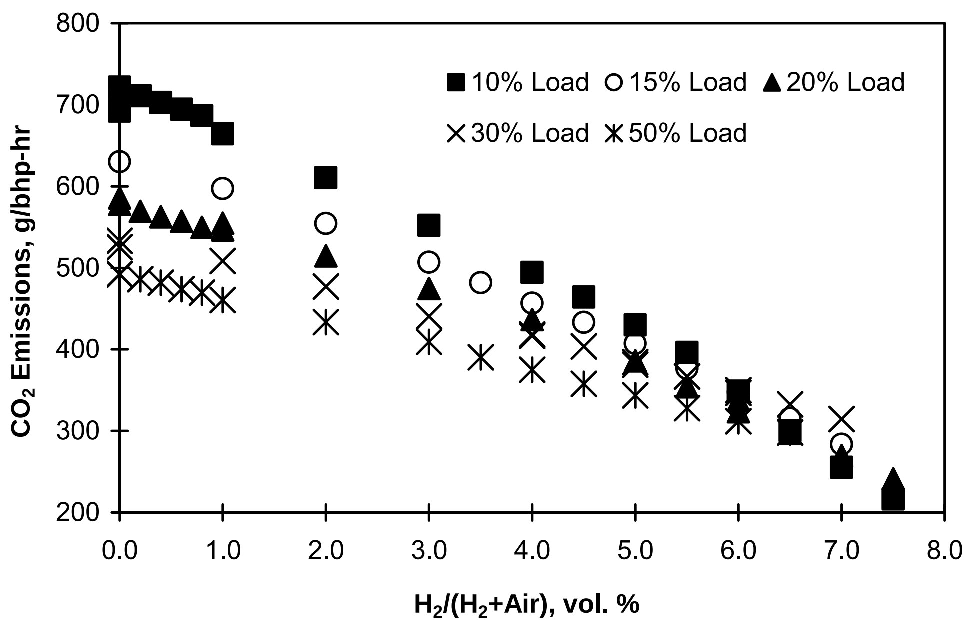

Figure 88 Effect of H_2 Addition and Engine Load on CO_2 Emissions, N=1200 RPM

6.4 *The Exhaust Emissions of H_2 and Its Combustion Efficiency*

As shown in Fig. 89 for 10% load operation, the emissions of the unburned H_2 increased almost linearly with the addition of H_2 into the intake air. The maximum H_2 emission of 1.4% was observed with the addition of 6% H_2. Further increasing the addition of H_2 beyond 6% reduced the emissions of H_2, representing the significant improvement in the combustion efficiency of H_2.

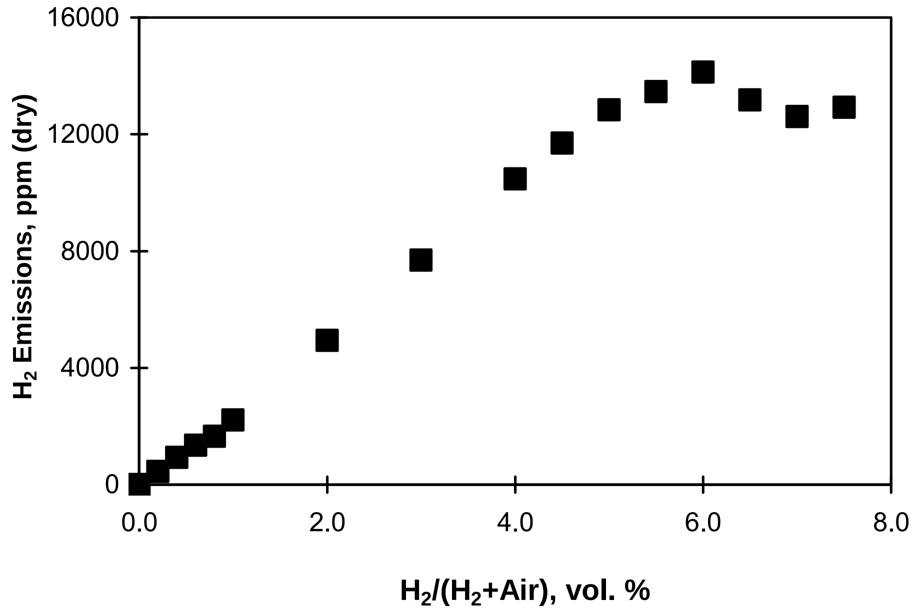

Figure 89 Effect of H_2 Addition on the Emissions of H_2, N=1200 RPM, 10% Load

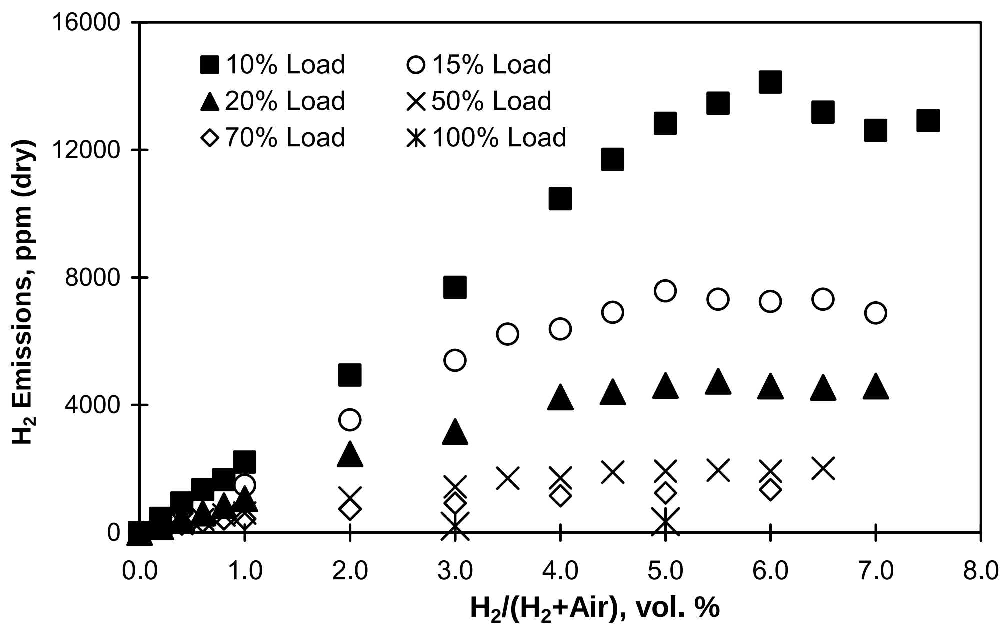

Figure 90 Effect of H_2 Addition and Engine Load on H_2 Emissions, N=1200 RPM

Fig. 90 shows the effect of H_2 addition and engine load on the emissions of H_2. The addition of H_2 at higher load resulted in much lower H_2 emissions. This was further demonstrated with the addition of 1%, 3%, 5% and 6% H_2. As shown in Fig. 91, increasing the engine load reduced substantially the emissions of H_2. It was evident that H_2 should be supplemented into H_2-diesel dual fuel engine under relatively higher load operation.

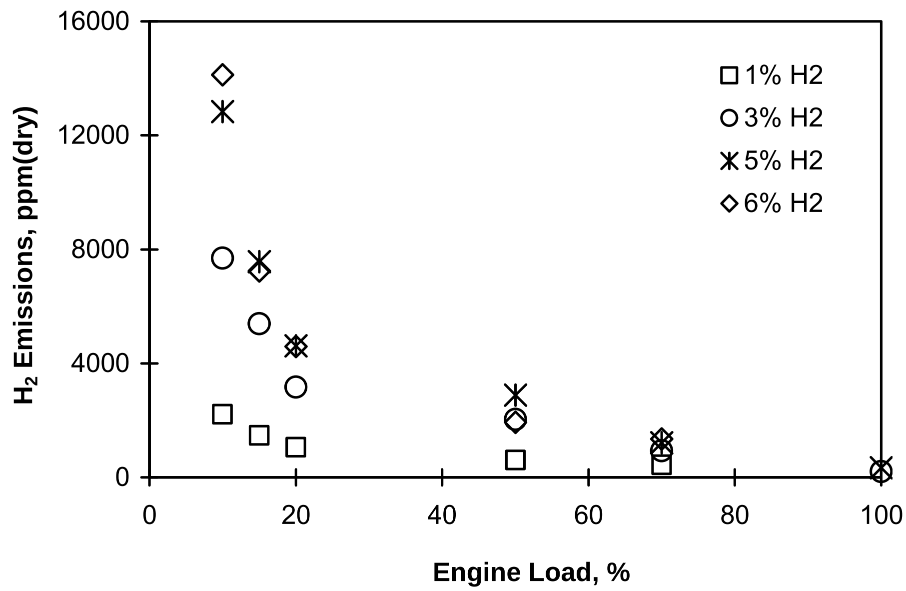

Figure 91 Effect of H_2 Addition and Engine Load on H_2 Emissions, 1200 RPM

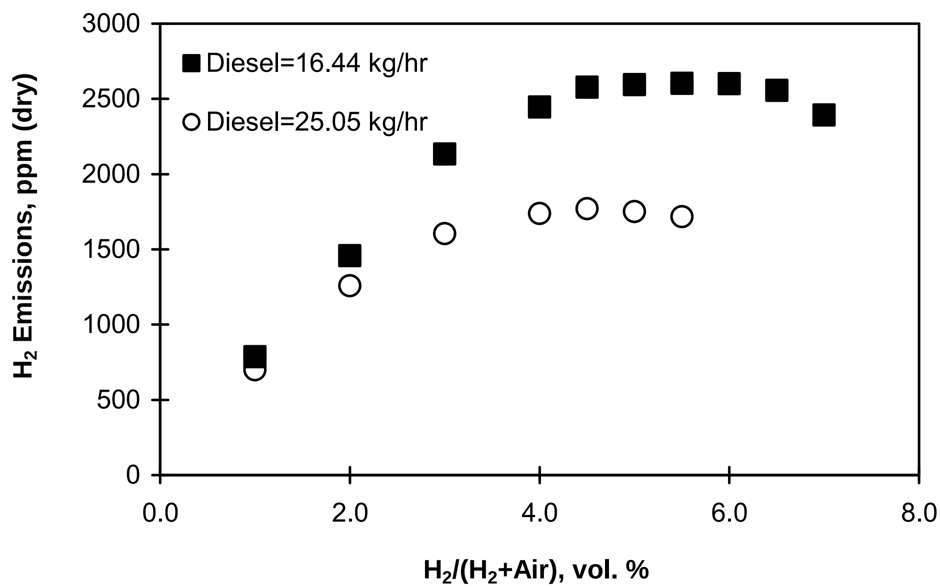

Figure 92 Effect of H_2 Addition and Diesel Flow Rate on the Emissions of H_2 Operated with Constant Diesel Flow Rate, N=1200 RPM.

Fig. 92 compares the effect of diesel fuel flow rate on the emissions of H_2. The emissions of H_2 increased linearly with addition of H_2 until the amount of H_2 supplied reached 4.5% and 4% for diesel fuel flow rate of 16.44 kg/hr and 25.05 kg/hr, respectively. Further increasing the amount of H_2 supplemented did not increase the emissions of H_2. The addition of H_2 at higher diesel fuel flow rate reduced the emissions of H_2, representing the improved H_2 combustion.

The H_2 emissions measured were processed to calculate the combustion efficiency of H_2. As shown in Fig. 93 for 10% load operation, the increasing addition of a small amount of H_2 reduced the combustion efficiency of H_2 with its minimum value of 74.9% obtained with the addition of 4% H_2. This was due to the reduced diesel fuel flow rates, which reduced the volume of the diesel spray plume and correspondingly reduced the amount of H_2 burned. Further increasing the addition of H_2 beyond 4% enhanced the combustion of H_2. This was due to the gradual development of an H_2 flame, which helped to burn more H_2 other than that entrained in diesel spray plume only. As shown in Fig. 93, the combustion efficiency of H_2 reached 84% when its concentration reached 7.5% in the intake mixture, which demonstrated the infeasibility of H_2 addition at 10% load operation with 16%-25% H_2 slip through the engine without participating in the combustion process.

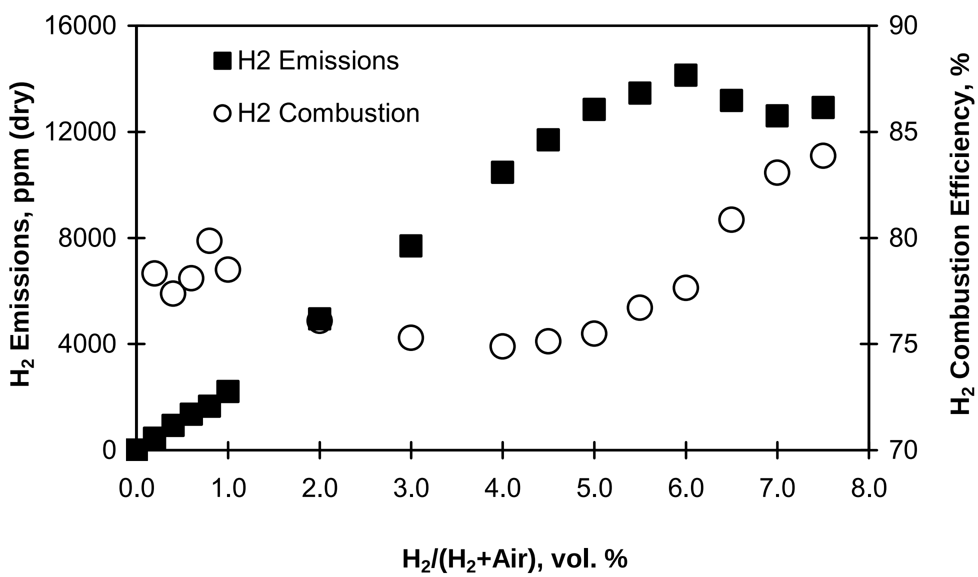

Figure 93 Effect of H_2 Addition on the Emissions of H_2 and Its Combustion Efficiency, N=1200 RPM, 10% Load

The combustion efficiency of H_2 can be improved by adding more H_2 to initiate a healthy H_2 flame or enhancing the entrainment of H_2 into diesel spray plum through the addition of H_2 at higher diesel fuel flow rate. As shown in Fig. 94, the combustion efficiency of H_2 was significantly improved when H_2 was added at higher load. This was due to the increased combustion temperature and enlarged diesel spray plume, which entrained more H_2-air mixture into the spray plume and helped to burn more H_2. This was further demonstrated by examining the effect of engine load on H_2 combustion efficiency while keeping $H_2/(H_2+air)$ constant. As shown in Fig. 95, the combustion efficiency of H_2 was improved substantially with the increasing engine load. It was evident that engine load was one of the main parameters dominating the combustion efficiency of the H_2.

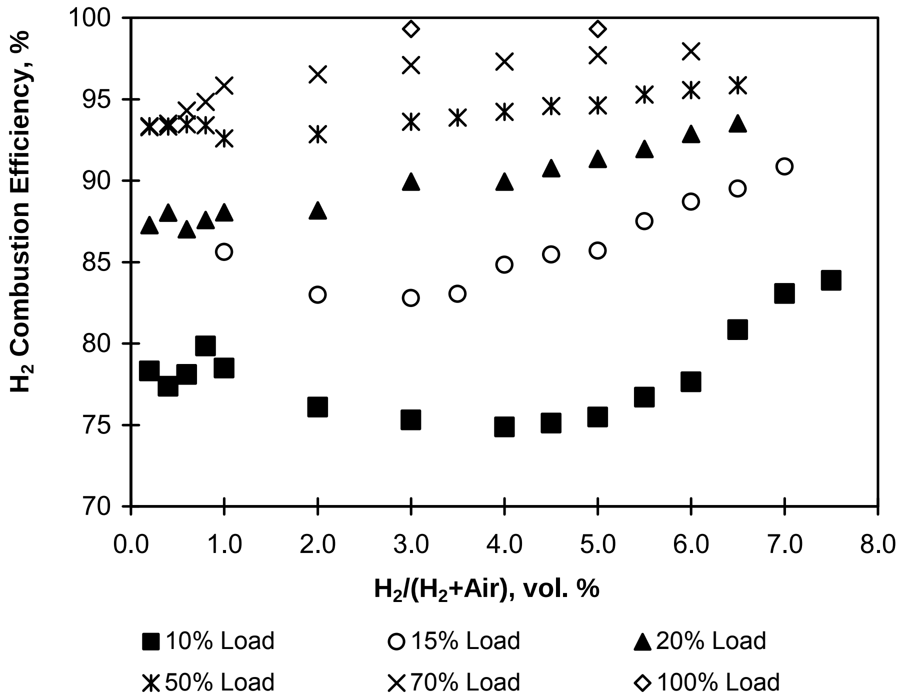

Figure 94 Effect of H_2 Addition and Engine Load on H_2 Combustion Efficiency, N=1200 RPM

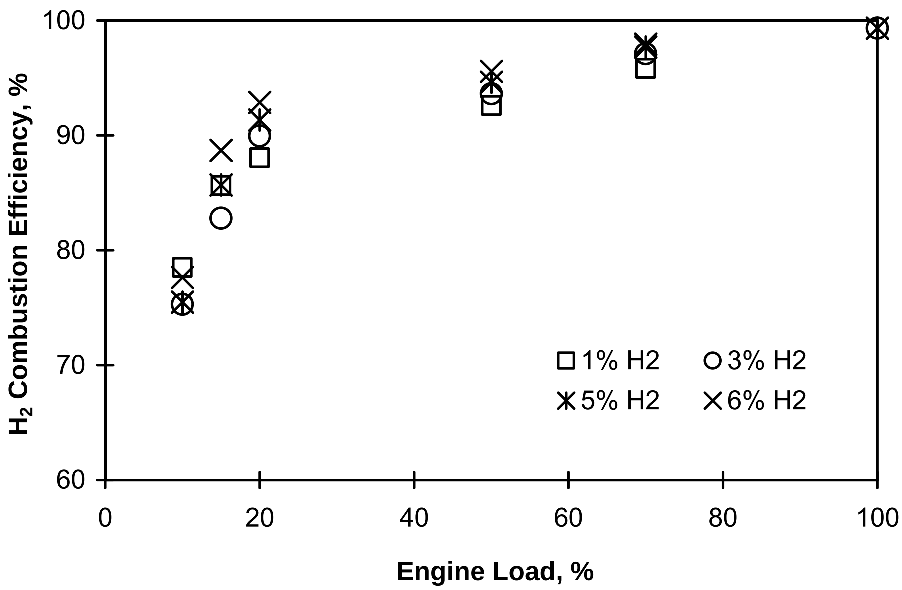

Figure 95 Effect of H_2 Addition and Engine Load on H_2 Combustion Efficiency, N=1200 RPM

The effect of diesel fuel flow rate on the combustion efficiency of H_2 was also examined. As shown in Fig. 96, the combustion efficiency of H_2 increased almost linearly with the addition of H_2. As expected, H_2 tends to burn more completely with high diesel fuel flow rate, though the difference tends to be small at high load operation. As shown in Figs. 94, 95 and 96, it is infeasible for H_2 to burn completely due to the presence of crevice and boundary layer in combustion chamber. The H_2 emitted from crevice at late expansion stroke and those presented in the boundary layer always survive the combustion process and slip through the engine as unburned H_2.

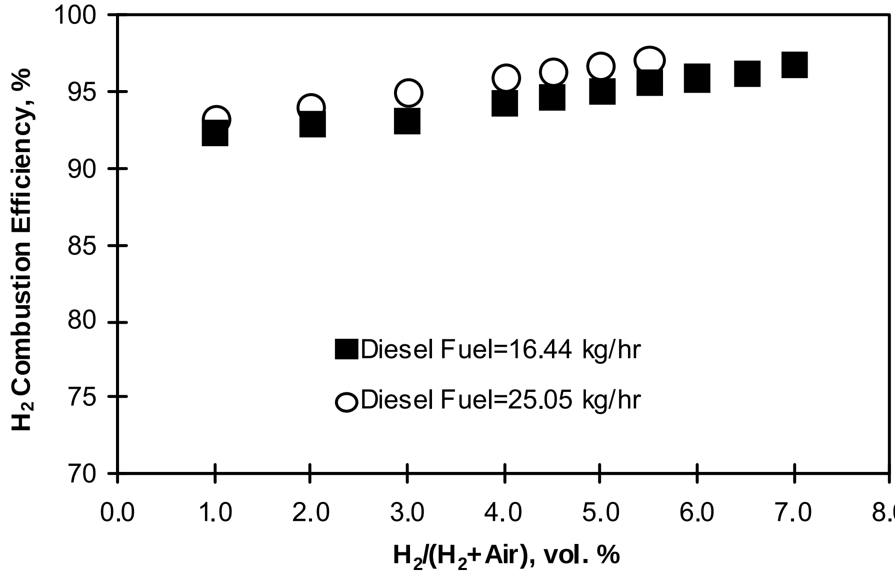

Figure 96 Effect of H_2 Addition and Diesel Fuel Flow Rate on the Combustion Efficiency of H_2, 1200 RPM (Diesel Flow 16.44 kg/hr Correspond to 30% Load for Diesel Only; Diesel Flow 25.05 kg/hr Correspond to 50% Load for Diesel Only)

6.5 Brake Thermal Efficiency and Its Improvement

The brake thermal efficiency is an important parameter describing the engine performance in converting chemical energy contained in fuel to mechanical work. Its value depends on the combustion efficiency of the fuel, combustion phasing, combustion duration (heat release rate), compression ratio and also the heat transfer from the bulk gas mixture to coolant. As shown in Fig. 97 for 10% load operation, the addition of H_2 into this diesel engine reduced substantially the brake thermal efficiency. The minimum brake thermal efficiency of 25.63% was observed when the $H_2/(H_2+Air)$ reached 4.5%, which was comparable to that when the minimum H_2 combustion efficiency was observed. Further increasing the amount of H_2 added beyond 4.5% gradually improved the brake thermal efficiency as the H_2-air mixture was rich enough to support a propagating flame. However, the brake thermal efficiency with the addition of H_2 at 10% load was lower than that for pure diesel operation (27.44%) for the range of tests explored. The further increase in the amount of H_2 added was infeasible as the diesel fuel flow rate became to low to ignite the mixture properly. Based on the data measured in this research, no benefit in brake thermal efficiency was obtained with the addition of H_2 at 10% load.

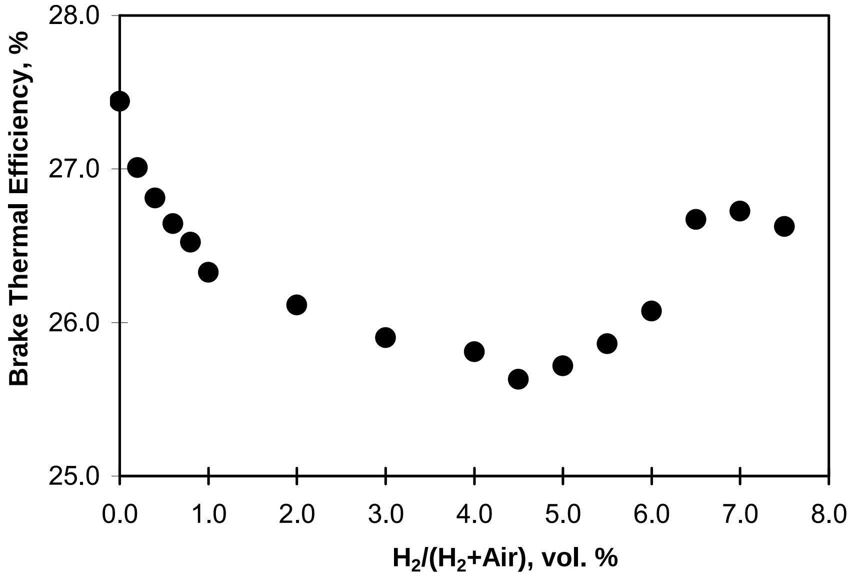

Figure 97 Effect of H_2 Addition on the Brake Thermal Efficiency, N=1200 RPM, Load=10%

Benefiting from the improved H_2 combustion efficiency, the positive effect of the addition of H_2 in improving the brake thermal efficiency was observed at higher load operation. As shown in Fig. 98 for 20% load operation, the brake thermal efficiency was found to be better than pure diesel operation when the $H_2/(H_2$+Air) reached 5.5%. With the addition of 7.5% H_2 into the intake mixture, the brake thermal efficiency of 35.88% was observed, which was better than the brake thermal efficiency of 34.20% obtained with pure diesel operation.

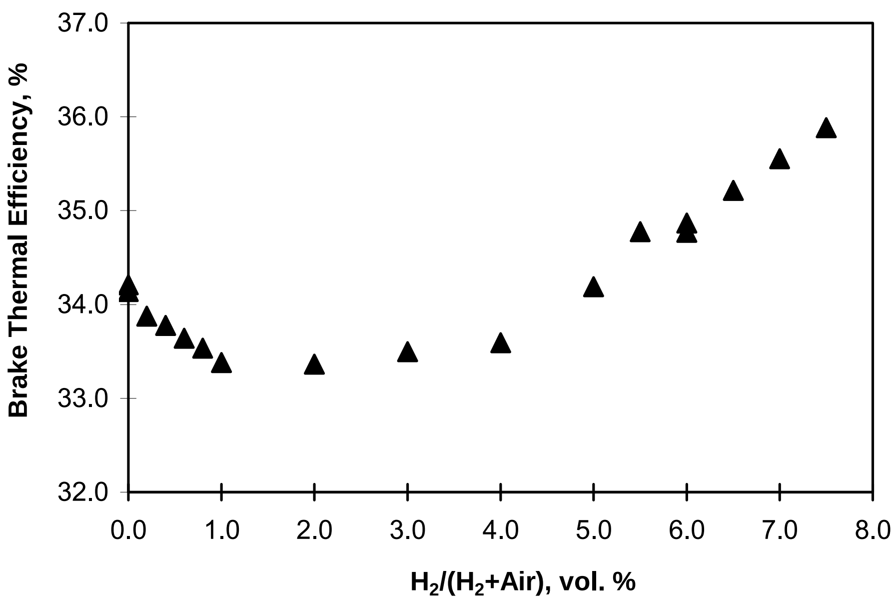

Figure 98 Effect of H_2 Addition on the Brake Thermal Efficiency, N=1200 RPM, Load=20%

Benefiting from the improvement of H_2 combustion efficiency and also an enhanced combustion process with the increasing engine load, the penalty in brake thermal efficiency was gradually limited to the lower H_2 addition range as shown in Fig. 99. When operated at medium to high load, the negative effect of H_2 addition on the brake thermal efficiency was only observed when a very small amount of H_2 (<1%) was added as shown in Fig. 100 for 50%, 70%, and 100% load operation.

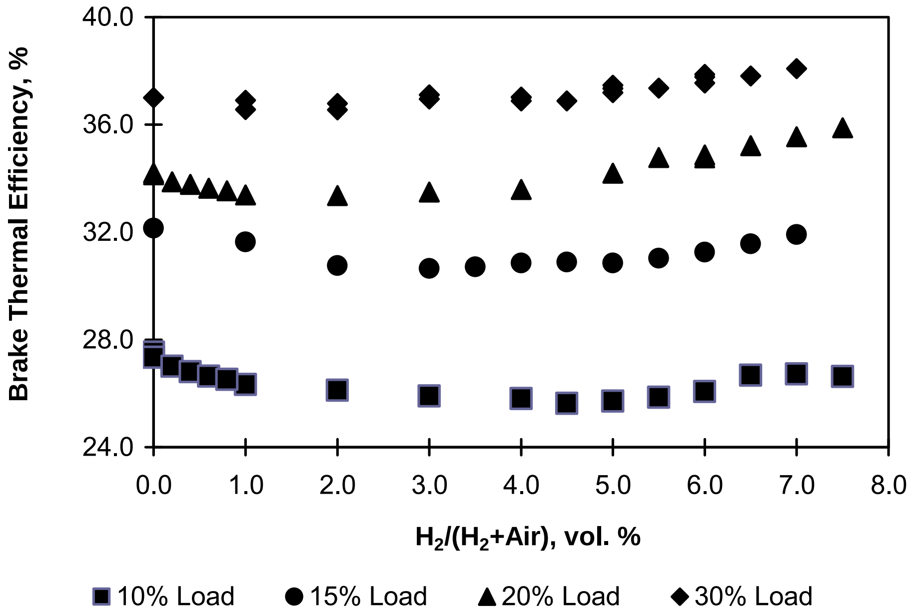

Figure 99 Effect of H_2 Addition and Engine Load on Brake Thermal Efficiency under Low Load Operation, N=1200 RPM

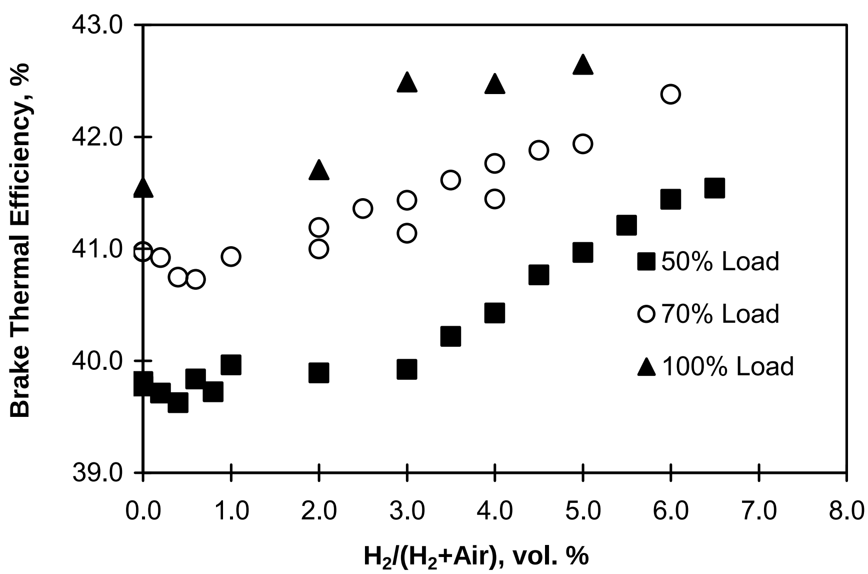

Figure 100 Effect of H_2 Addition and Engine Load on Brake Thermal Efficiency at Medium to High Load Operation, N=1200 RPM (For 100% Operation, 2% and 4% H_2 Data was Measured at 1225 RPM)

The effect of H_2 addition on the brake thermal efficiency was also explored by examining the improvement to the brake thermal efficiency (BTE) compared to that of pure diesel operation. As shown in Fig. 101, the positive effect of H_2 addition on brake thermal efficiency could only be obtained when a relatively large amount of H_2 was supplemented at loads higher than 15%. With the increasing engine load, the positive effect of H_2 addition in improving brake thermal efficiency could be obtained with the addition of less H_2. As shown in Fig. 102 for medium to high load operation, the brake thermal efficiency was improved with as low as 1% H_2 addition

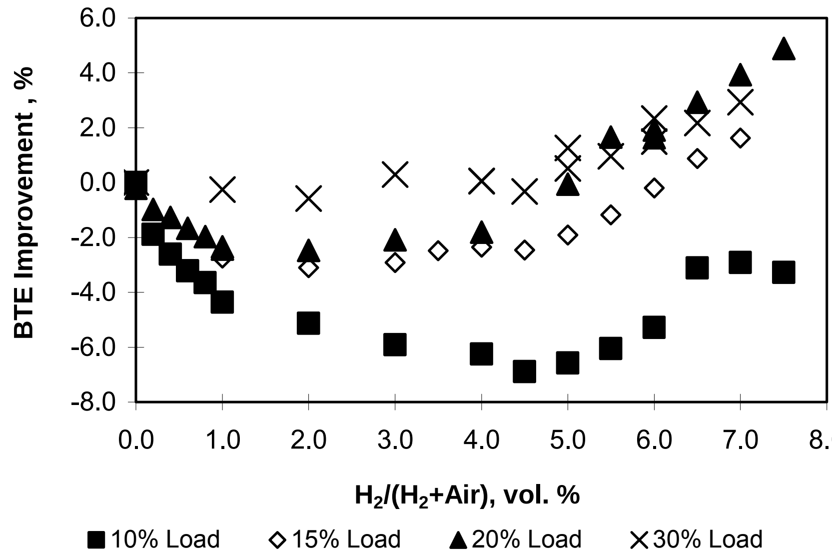

Figure 101 Effect of H_2 Addition and Engine Load in Improving the Brake Thermal Efficiency under Low Load Operation. N=1200 RPM

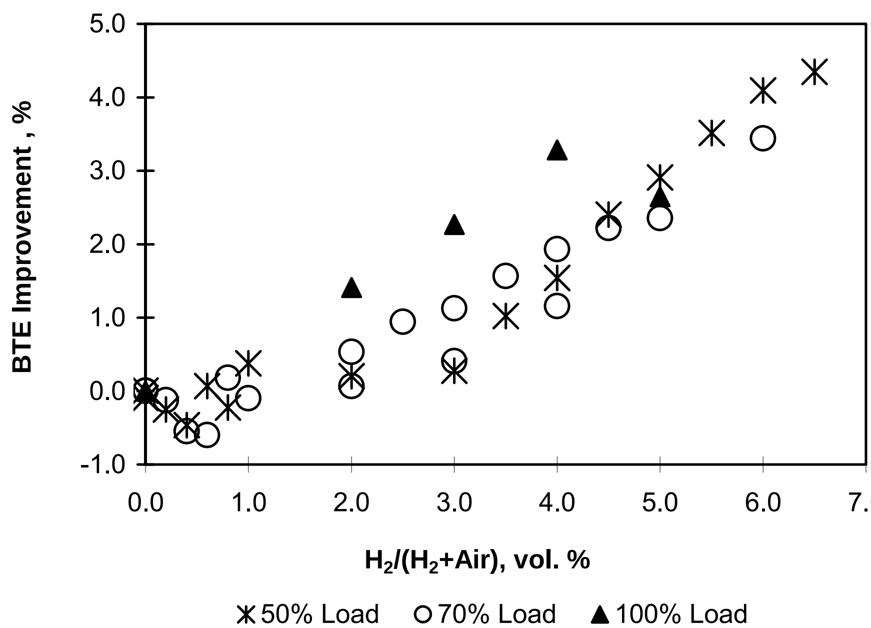

Figure 102 Effect of H_2 Addition and Engine Load in Improving the Brake Thermal Efficiency under Medium to High Load Operation. N=1200 RPM

The effect of engine load on the improvement to the brake thermal efficiency was also examined with the addition of 6% H_2 into the intake mixture. As shown in Fig. 103, the maximum benefit on the brake thermal efficiency was about 4% for the addition of 6% H_2 into this 2004 Mack MD11 diesel engine. Compared to that of the 1999 Cummins ISM370 engine, the addition of H_2 to this 2004 Mack MD11 engine was less effective in enhancing the brake thermal efficiency, especially at low load operation.

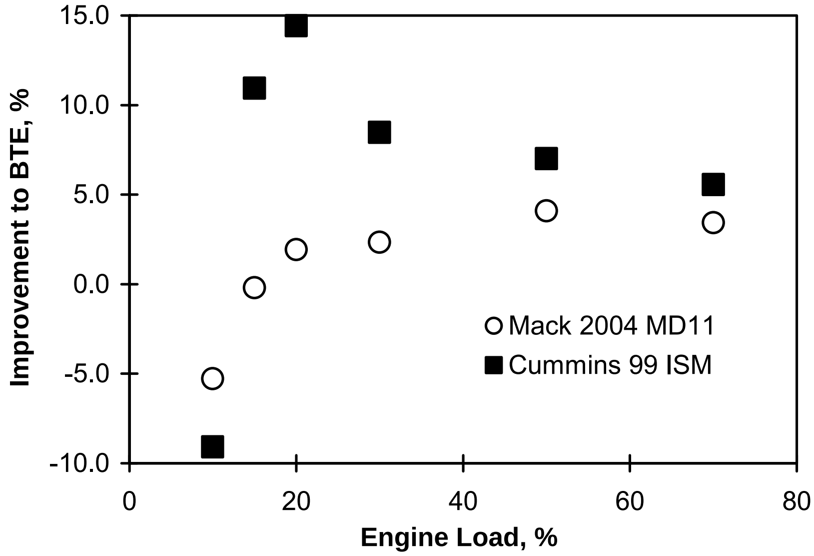

Figure 103 Effect of Engine Load on Brake Thermal Efficiency Improvement with the Addition of 6% H_2, N=1200 RPM.

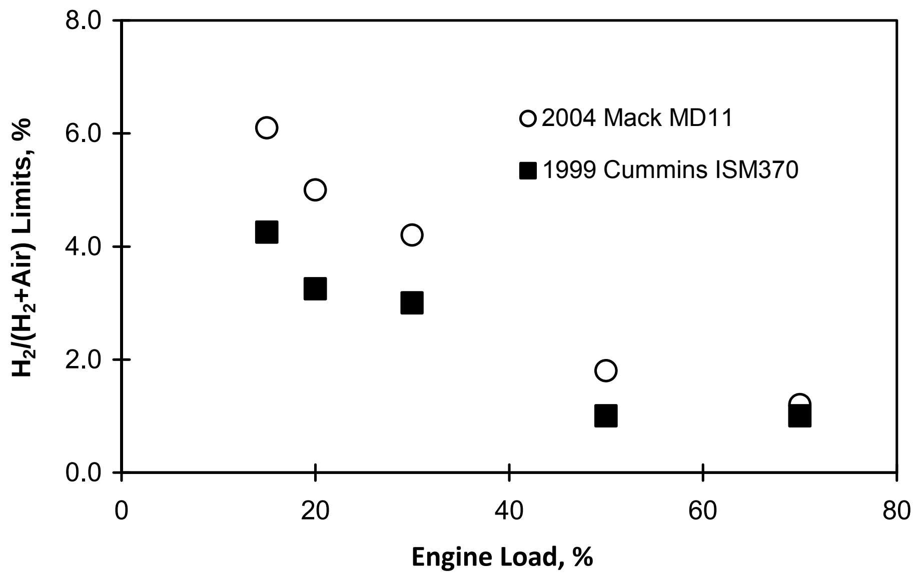

Figure 104 Effect of Engine Load on the Minimum H_2 Supplementation Rate Needed for Positive Effect on Brake Thermal Efficiency, N=1200 RPM

Figure 104 compares the effect of engine load on the minimum H_2 supplementation rate to achieve a positive effect on brake thermal efficiency using both diesel engines. It was shown that more H_2 was needed for the 2004 Mack MD11 to obtain a positive effect in improving the brake thermal efficiency. This may be due to the application of cooled EGR in the 2004 Mack MD11 engine.

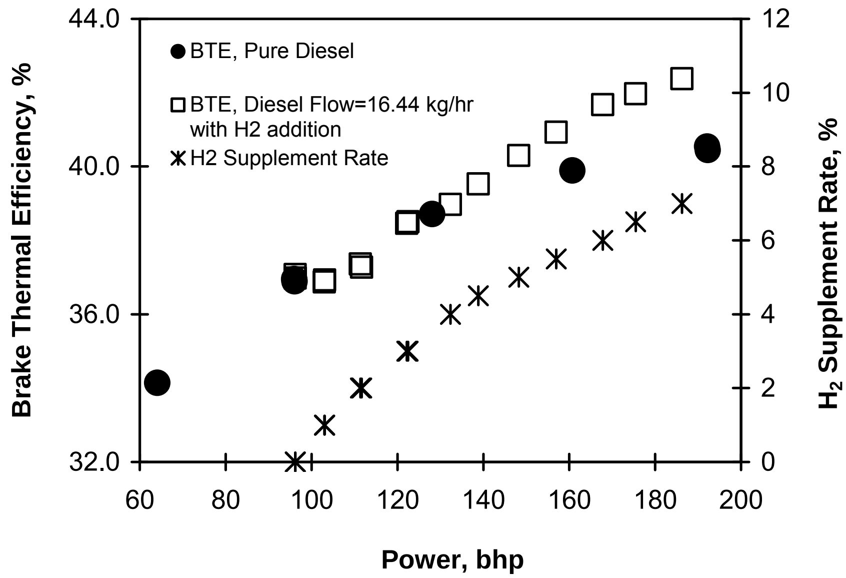

Figure 105 Effect of H_2 Addition on the Brake Thermal Efficiency. N=1200 RPM. For Constant Diesel Fuel Flow Rate (16.44 kg/hr) Operation, Engine Load Was Changed by the Addition of H_2

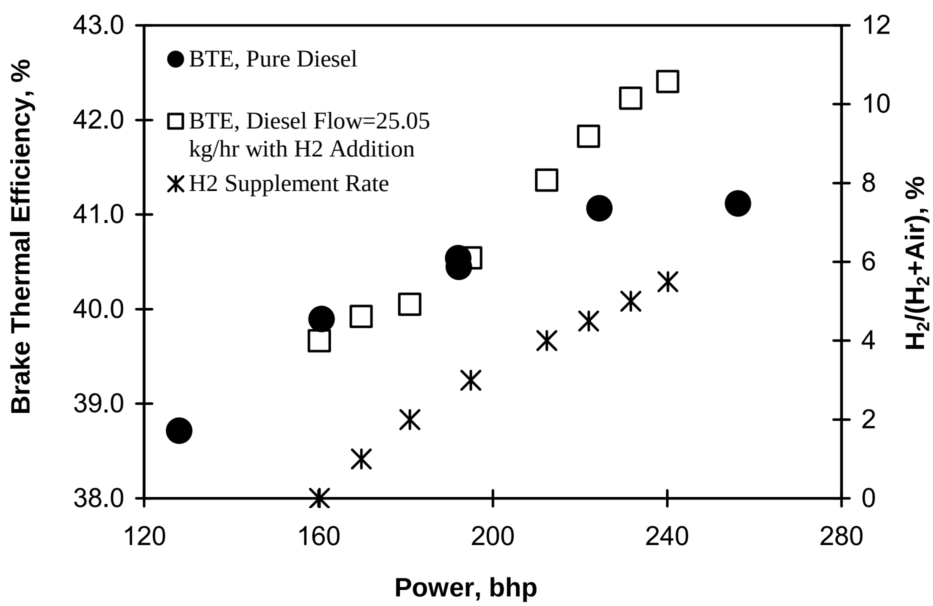

Figure 106 Effect of H_2 Addition on the Brake Thermal Efficiency. N=1200 RPM. For Constant Diesel Fuel Flow Rate (25.05 kg/hr) Operation, Engine Load Was Changed by the Addition of H_2

For constant diesel fuel flow rate operation, engine load was varied by altering the amount of H_2 added. As shown in Fig. 105, the addition of a small amount of H_2 was shown to slightly deteriorate the brake thermal efficiency compared to the burning of pure diesel. The addition of

H_2 beyond 4% was shown to improve the brake thermal efficiency. A similar result was obtained for the addition of H_2 at a higher diesel flow rate. As shown in Fig. 106, the positive effect of the addition of H_2 on brake thermal efficiency was obtained for the addition of H_2 beyond 3%.

6.6 Cylinder Pressure and Heat Release Rate

The addition of H_2 into this diesel engine at high load was shown to significantly enhance the combustion process. As shown in Fig. 107 for 70% load operation, the addition of H_2 was shown to increase substantially the cylinder pressure after combustion was initiated. In fact, its affect on the cylinder pressure prior to the initiation of combustion was small. As shown in Fig. 108, the addition of a small amount of H_2 (<3%) was found to gradually increase the peak cylinder pressure but have negligible effect on phasing when peak cylinder pressure was obtained. However, further increasing the addition of H_2 beyond 3% continued to increase the peak cylinder pressure and substantially advanced the phasing when peak cylinder pressure was observed.

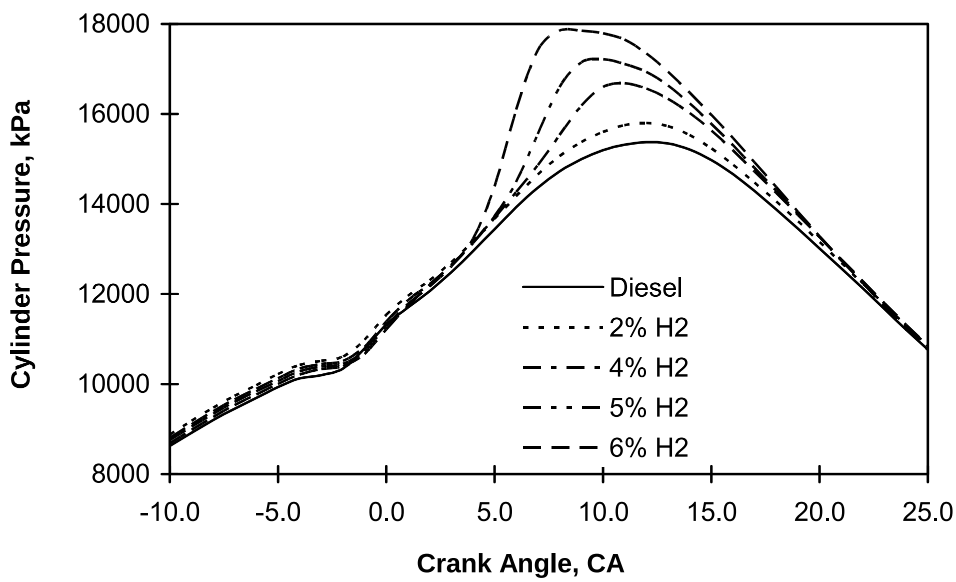

Figure 107 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 70% Load

As shown in Fig. 109, the addition of H_2 was shown to slightly retard the initiation of the premixed combustion and the phasing of the peak premixed combustion heat release rate. As shown in Table 22, the retarding of the remixed combustion was due to slight retard in fuel injection timing and also the slightly longer ignition delay. This made the premixed combustion slightly stronger as more diesel fuel was well mixed with air during the ignition delay period and ready to be burned in the premixed combustion stage. As shown in Fig. 109, the addition of H_2 to the diesel engine was also shown to enhance the heat release rate during the switching process between premixed combustion to diffusion combustion. When $H_2/(H_2+\text{air})$ reached 6%, the induction period from premixed combustion to diffusion combustion became negligible. The combustion of H_2-air mixture may have been ignited by the premixed combustion of diesel fuel prior to the initiation of diffusion combustion. This phenomenon was not observed in the combustion of the 1999 Cummins ISM370 H_2-diesel dual fuel engine.

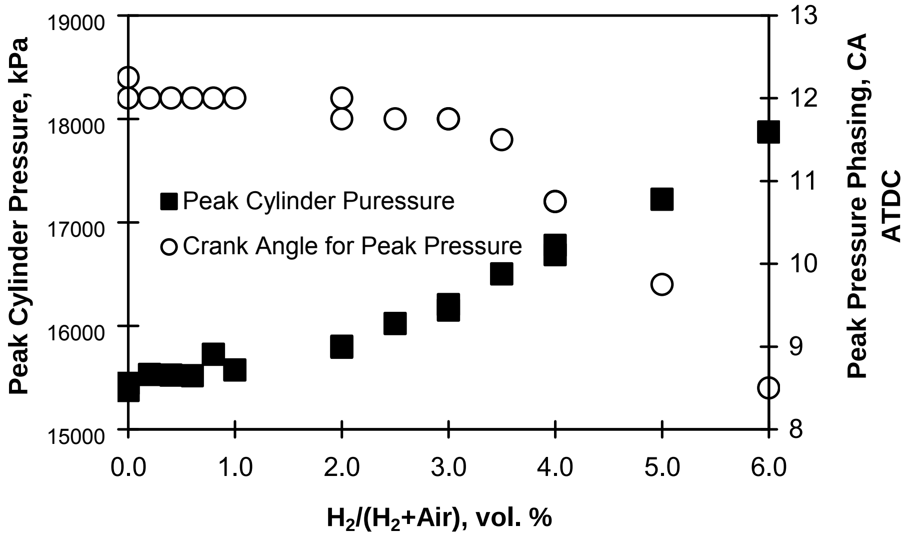

Figure 108 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 70% Load

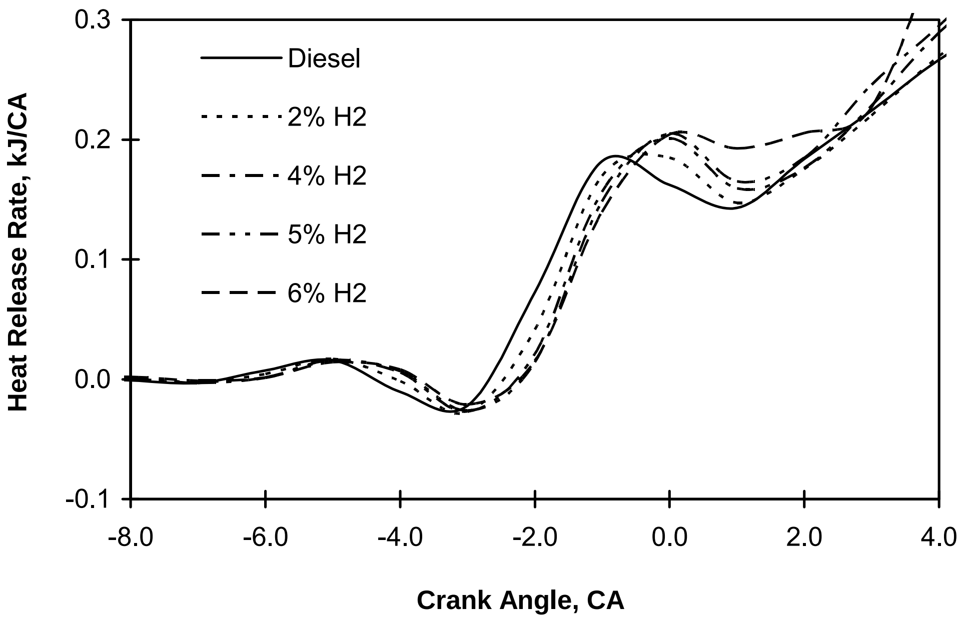

Figure 109 Effect of H_2 Addition on Premixed Combustion, N=1200 RPM, 70% Load

Table 22 Effect of H_2 Addition on Start of Injection (SOI), Start of Combustion (SOI) and Ignition Delay when Operated at 1200 rpm, 70% Load

$H_2/(H_2+Air)$, %	SOI, CA ATDC	SOC, CA ATDC	Ignition Delay, CA
0	-7.15	-2,20	4.95
2	-7.1	-1.95	5.15
4	-6.95	-1.75	5.20
5	-6.95	-1.70	5.25
6	-6.9	-1.68	5.22

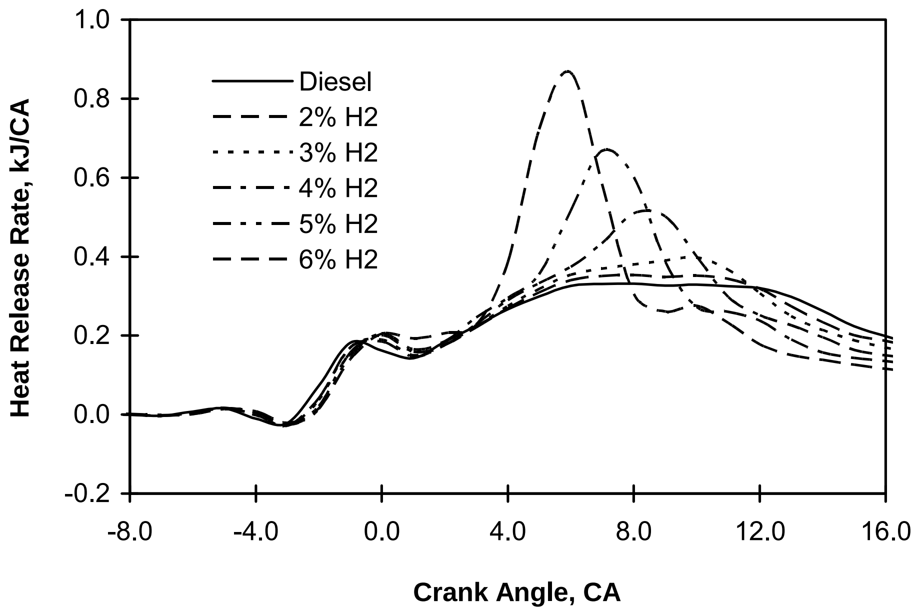

Figure 110 Effect of H_2 Addition on Heat Release Process, N=1200 RPM, 70% Load

Fig. 110 shows the effect of H_2 addition on the heat release process when operated at 70% load. The addition of H_2 was shown to have a more significant effect on diffusion combustion than premixed combustion. The addition of a small amount of H_2 gradually enhanced the heat release rate during the middle stage of diffusion combustion with deteriorated late diffusion combustion. When the amount of $H_2/(H_2+Air)$ reached 3%, the enhanced middle stage diffusion combustion gradually developed to the peak heat release rate at a later phasing compared to diesel combustion, which was associated with the fast burning of H_2. As shown in Fig. 111 for the addition of 6% H_2, the featured two-stage combustion process of the diesel engine developed into a three-stage combustion process of H_2-diesel dual fuel engine including premixed diesel combustion (stage 1), early stage diffusion combustion of diesel combustion (stage 2), combination of diesel diffusion combustion and burning of H_2 by multi-turbulent flame (stage 3) and late diffusion combustion of diesel fuel (continue of stage 2). The enhanced heat release peak in diffusion combustion represented a combination of the diffusion combustion of diesel fuel and fast burning of H_2-air mixture by multi-turbulent flame. Also, the combustion of H_2 finished earlier than slow diffusion combustion of diesel fuel. The fast burning of H_2 fuel contributed to the advanced phasing of peak cylinder pressure obtained in H_2-diesel dual fuel operation mode.

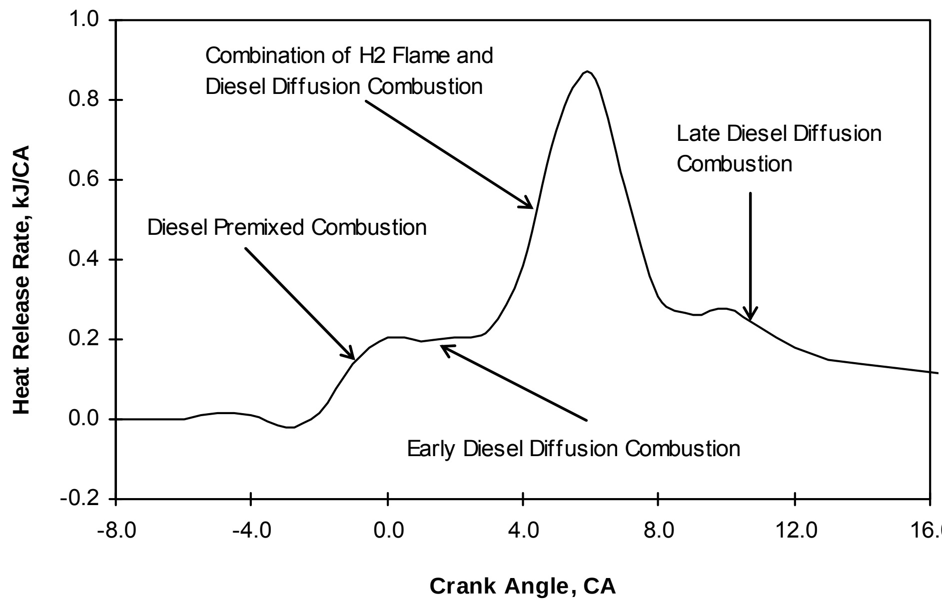

Figure 111 Featured Heat Release Process of H_2-Diesel Dual Fuel Engine Operation, N=1200 RPM, 70% Load, $H_2/(H_2+Air)$=6%

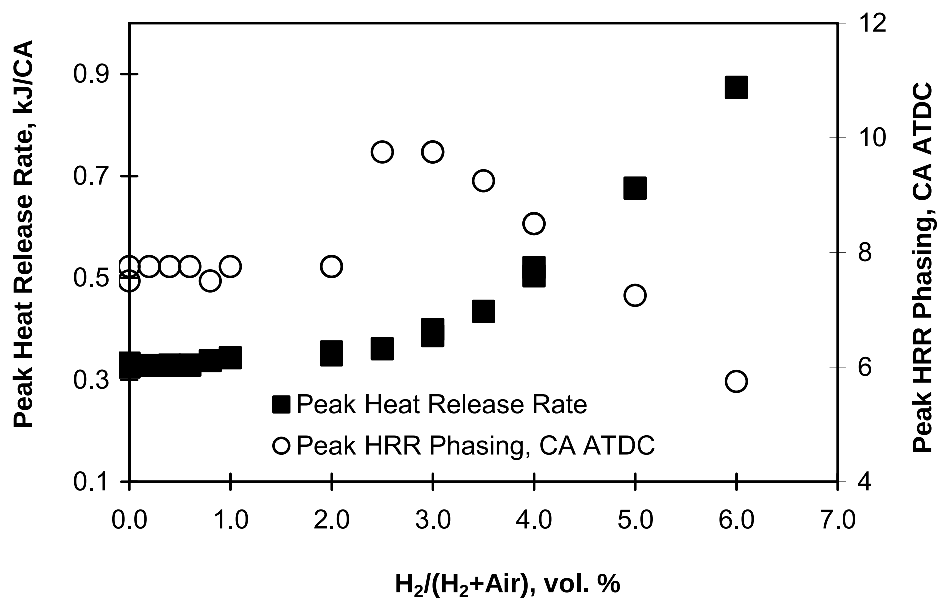

Figure 112 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 70% Load

Figure 112 shows the effect of H_2 addition on peak heat release rate and its phasing. The addition of a small amount of H_2 (<3%) slightly increased the peak heat release rate but had negligible effect on the phasing when peak heat release rate was observed. A sudden retard in phasing of peak heat release rate with the addition of 2.5% and 3% H_2 was due to the development of peak heat release rate obtained at middle stage of diffusion combustion. As described before, this late heat release bump was due to the gradual development of H_2 combustion. As shown in Fig. 112, further increasing the amount of H_2 beyond 3% drastically enhanced the combustion process represented by the increased peak heat release rate observed with substantially advanced phasing.

Figs. 113-117 show the effect of H_2 addition on variation of cylinder pressure and heat release process for 50% load operation. As shown in Figs. 113 and 114, the addition of a small amount of H_2 (<3%) was found to slightly increase the peak cylinder pressure but had negligible effect on its phasing. Similar to 70% load operation, the sudden retard in the phasing of peak cylinder pressure for 3.5% and 4% H_2 addition, as shown in Fig. 114, was due to the development of the second heat release bump in diffusion combustion as shown in Fig. 115. Increasing the amount of H_2 added beyond 4% increased substantially the peak cylinder pressure observed at substantially advanced phasing.

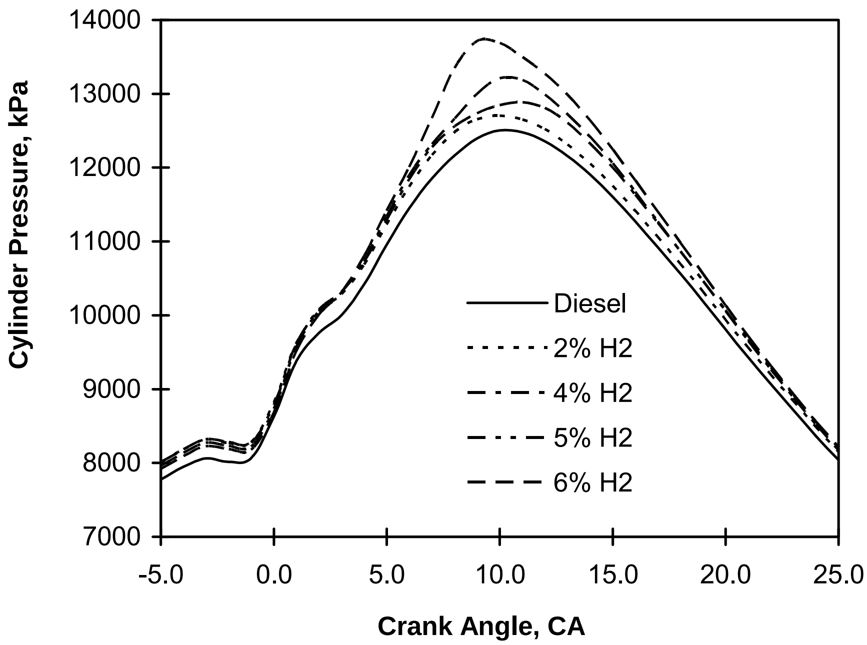

Figure 113 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 50% Load

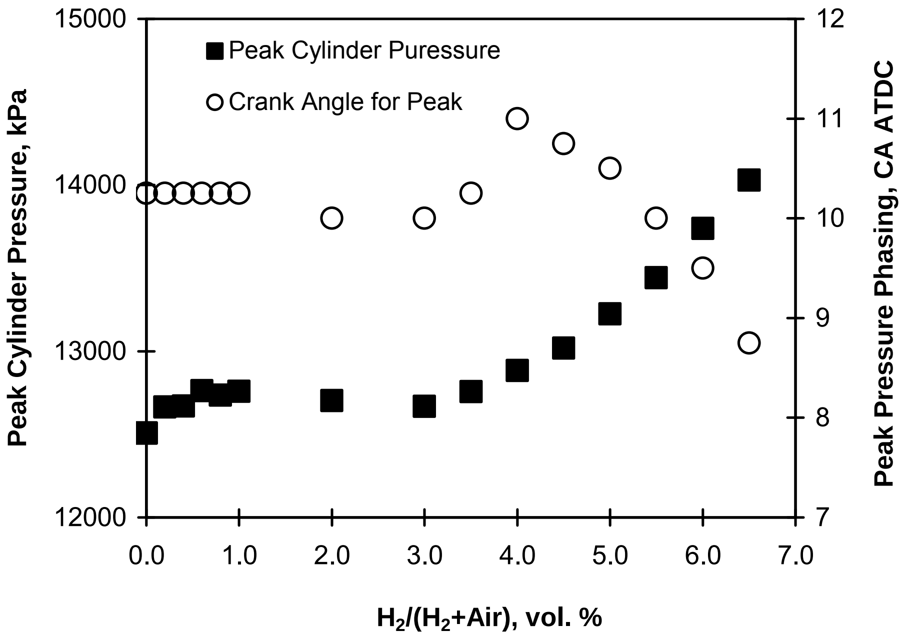

Figure 114 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 50% Load

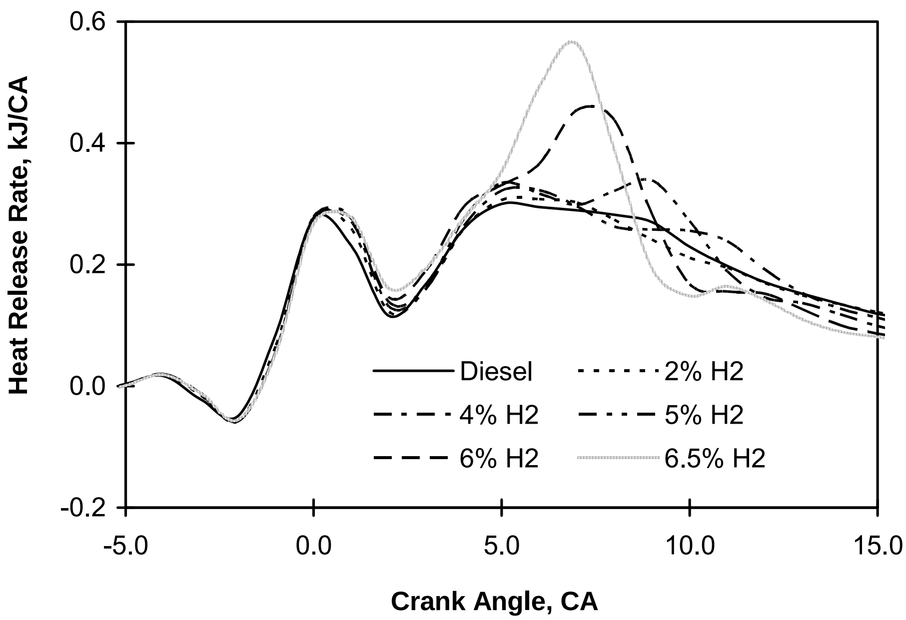

Figure 115 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 50% Load

As shown in Fig. 116, the addition of a small amount of H_2 (<4.5%) slightly increased the peak heat release rate with negligible effect on its phasing. Also, the addition of H_2 was found to enhance the late diffusion combustion as shown by the gradual development of the second heat release peak observed at late diffusion combustion as shown in Fig. 117. Further increasing the amount of H_2 added enhanced and advanced the heat release peak associated with H_2 until it became the dominate one as shown in Fig. 115 for the addition of 6% and 6.5% H_2.

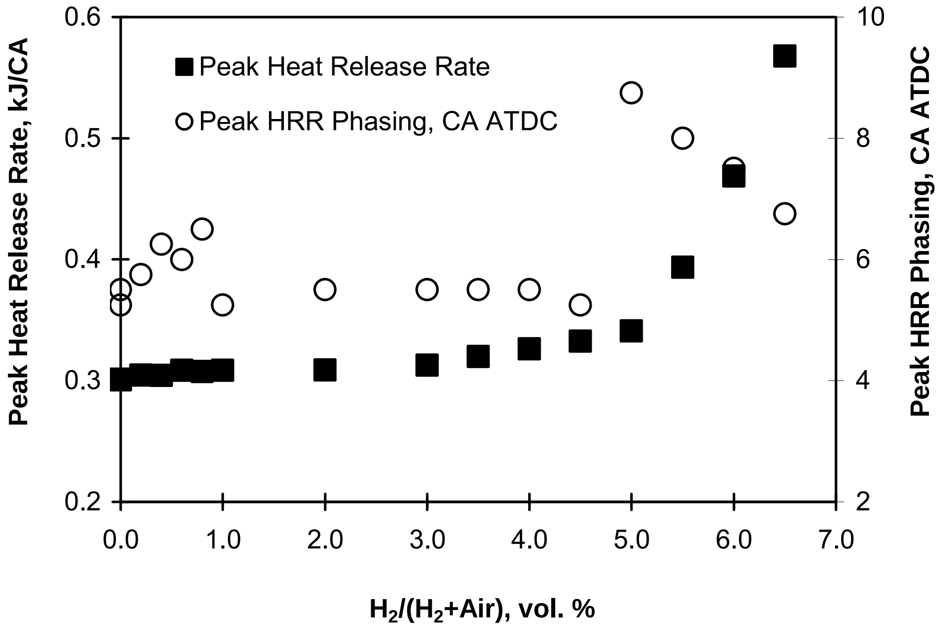

Figure 116 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 50% Load

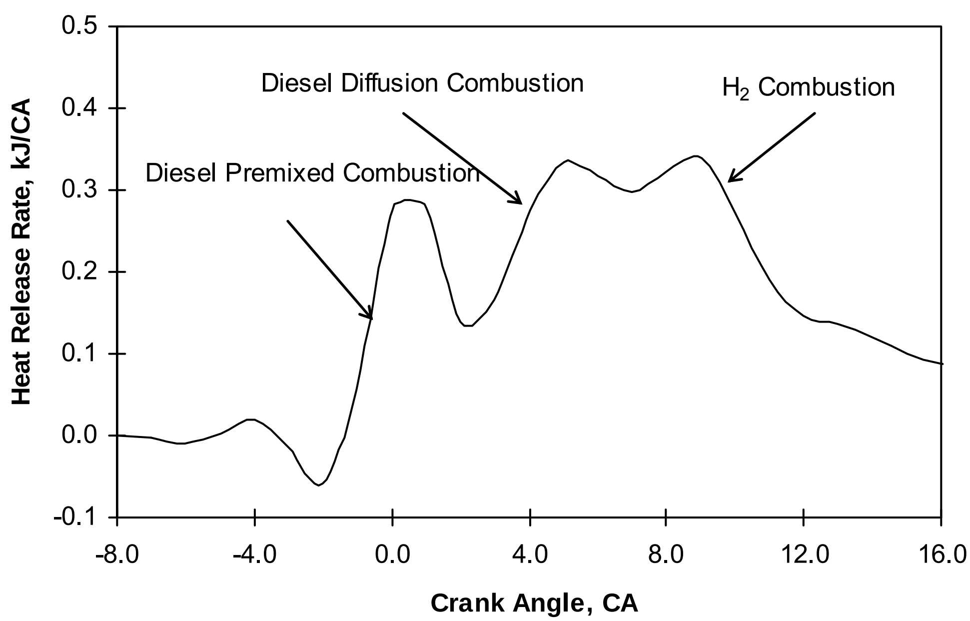

Figure 117 Variation of Heat Release Rate with Changes in Crank Angle, N=1200 RPM, 50% Load, $H_2/(H_2+Air)=5\%$

Figs. 118-121 show the effect of H_2 addition on the cylinder pressure and heat release process for 30% load operation. Compared to 50% and 70% load operation, the addition of H_2 at 30%

load had a much weaker effect on the cylinder pressure. As shown in Fig. 118 and 119, the addition of H_2 below 6% had a negligible effect on the cylinder pressure after combustion was initiated including the peak cylinder pressure but it retarded slightly the peak pressure phasing. Further increasing the amount of H_2 beyond 6% slightly increased the peak cylinder pressure observed at substantially retarded phasing. As shown in Fig. 120, the addition of H_2 at 30% load retarded the initiation of premixed combustion, enhanced premixed combustion, inhibited the early stage diffusion combustion and slightly enhanced the late stage diffusion combustion. With the addition H_2 over 6%, the late stage diffusion combustion developed into a third heat release peak. Further increasing the amount of H_2 added was shown to advance and enhance the third heat release peak until it dominated the heat release process of diffusion combustion. This was due to the gradual development of a healthy H_2 flame. As shown in Fig. 121, the addition of H_2 at 30% load slightly enhanced the peak heat release rate and retarded its phasing. Compared to that of 50% and 70% load with peak heat release rate observed at diffusion combustion stage, the peak heat release rate of 30% load operation was observed in the premixed combustion stage. In comparison, the diffusion combustion was found to be very weak for low load operation.

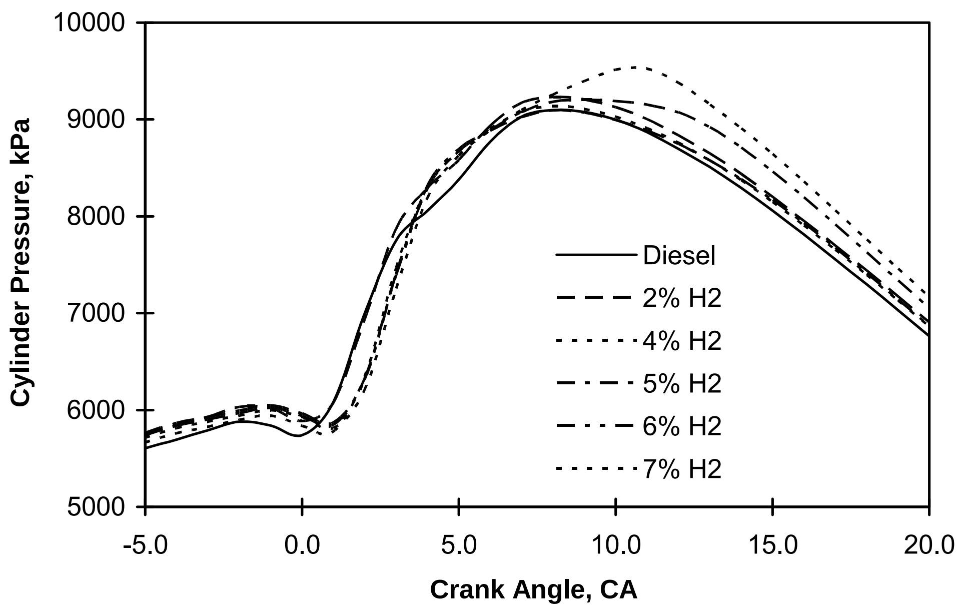

Figure 118 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 30% Load

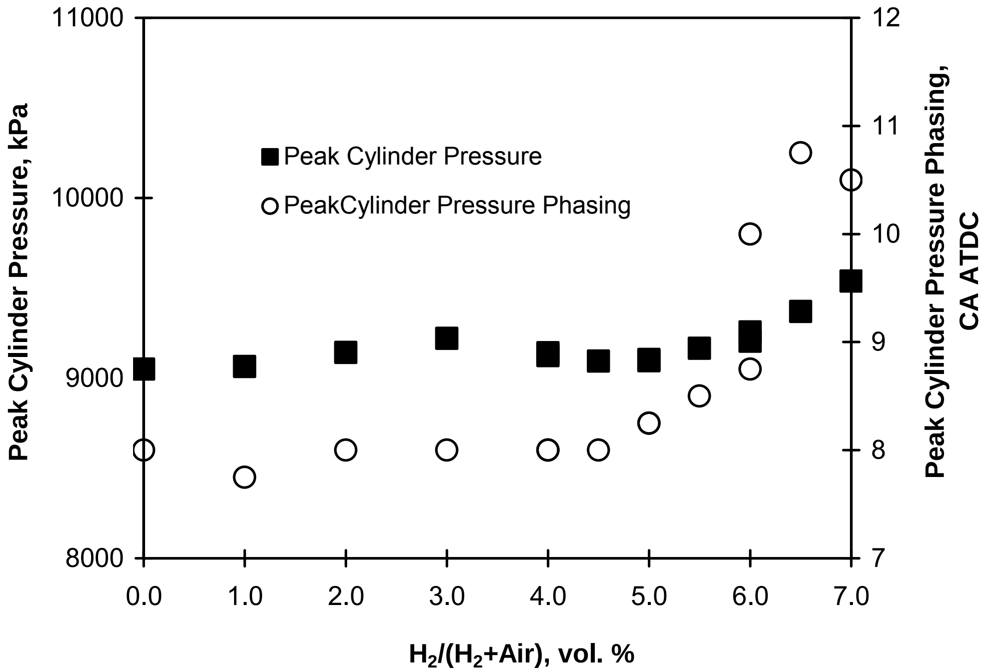

Figure 119 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 30% Load

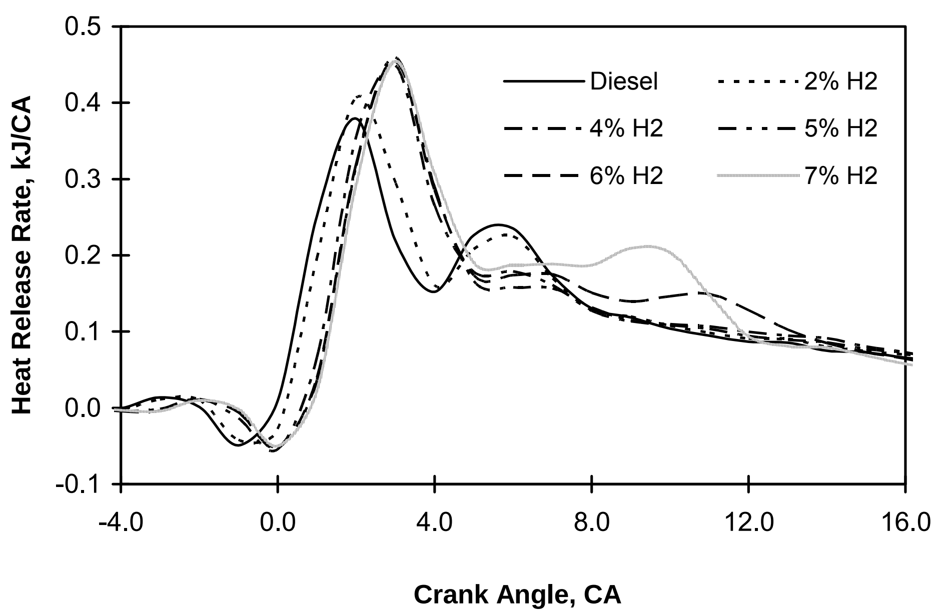

Figure 120 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 30% Load

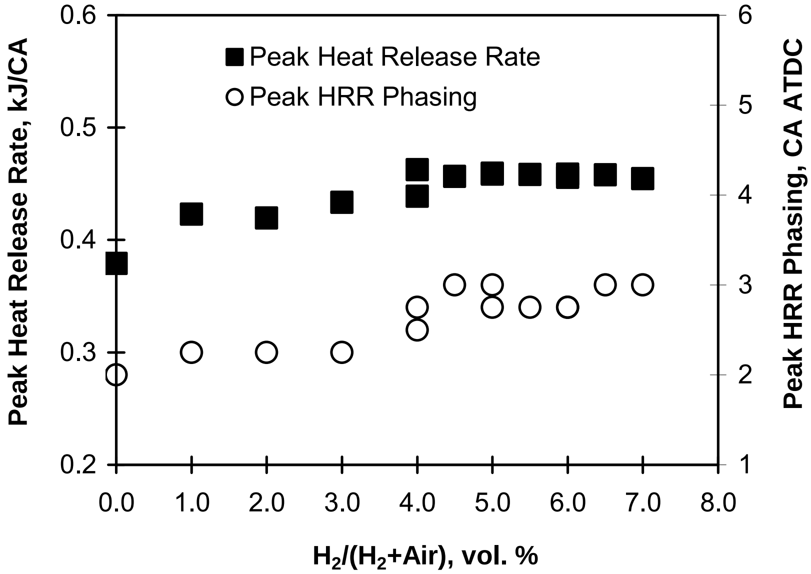

Figure 121 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 30% Load

Figs. 122-125 show the effect of H_2 addition on the cylinder pressure and heat release process when operated at 15% load operation. Compared to medium and high load operation, the addition of H_2 at 15% load educed substantially the cylinder pressure after the combustion was initiated. The reduced peak cylinder pressure was observed at a retarded phasing as shown in Figs. 122 and 123. As shown in Fig. 124, the retarded peak cylinder pressure was due mainly to the later initiation of the combustion process. As shown in Table 23, the retarding combustion phasing was due mainly to the retarded injection timing. In comparison, the effect of H_2 addition on ignition delay was negligible. Similar to 30% load operation, the peak heat release rate was observed in premixed combustion as shown in Fig. 124. In comparison, the diffusion combustion becomes extremely weak. As shown in Fig. 125, the addition of small amount of H_2 enhanced lightly the peak heat release rate. However, the addition of H_2 beyond 4% slightly reduced the peak heat release rate and retarded the phasing when peak heat release rate was observed.

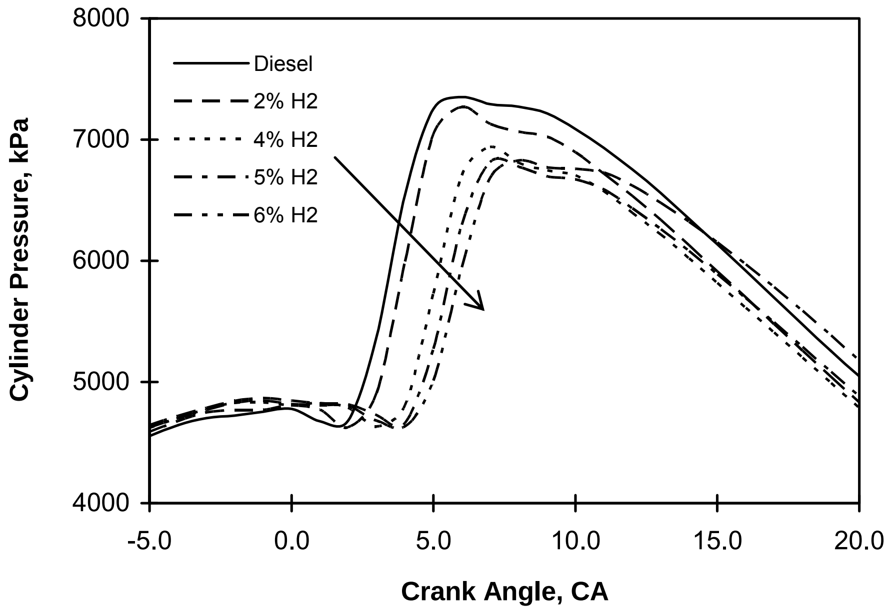

Figure 122 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 15% Load

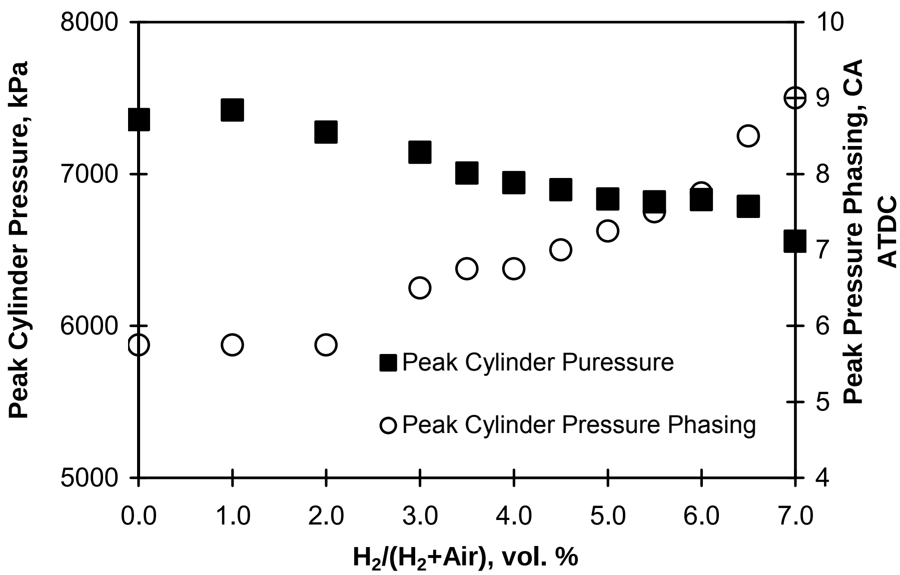

Figure 123 Effect of H_2 Addition on Peak Cylinder Pressure and Its Phasing, N=1200 RPM, 15% Load

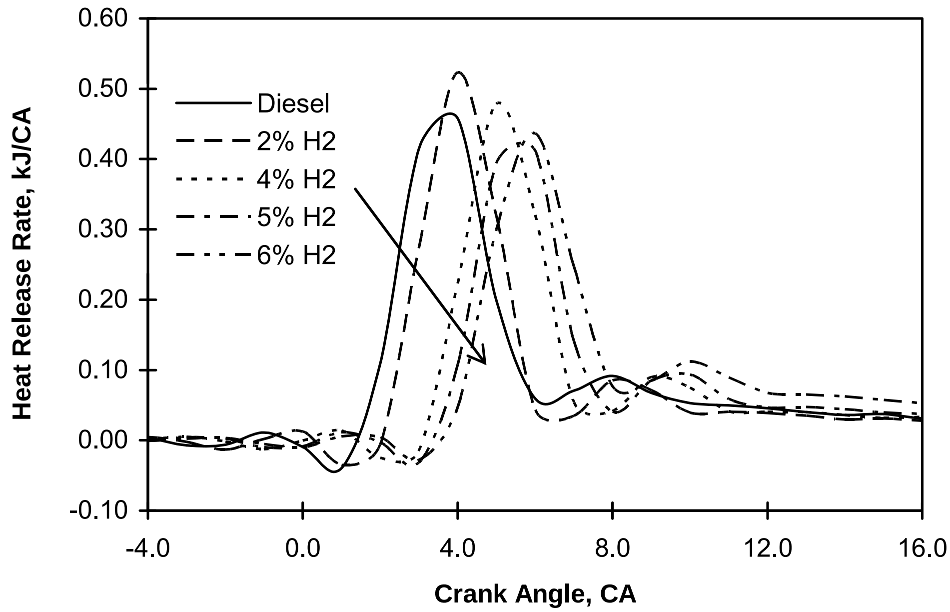

Figure 124 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 15% Load

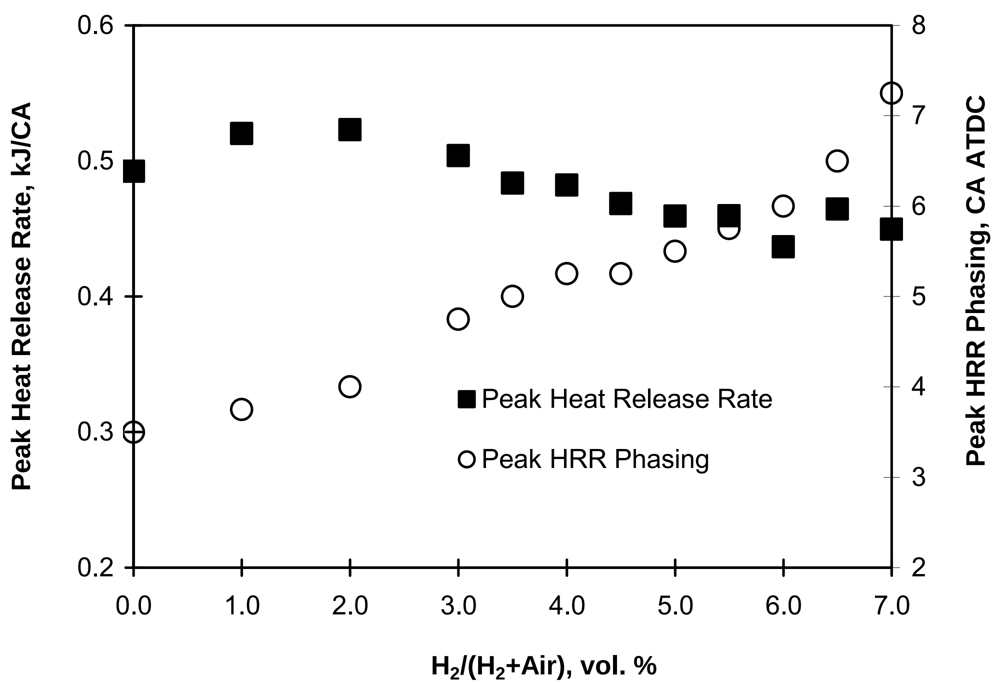

Figure 125 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 15% Load

Table 23 Effect of H_2 Addition on Start of Injection (SOI), Start of Combustion (SOC) and Ignition Delay, 1200 rpm, 15% Load

$H_2/(H_2+Air)$, vol. %	SOI, CA ATDC	SOC, CA ATDC	Ignition Delay, CA
0	-2.50	1.62	4.12
2	-2.05	2.25	4.30
4	-1.0	3.35	4.35
5	-0.5	3.70	4.20
6	-0.15	4.02	4.17

Figs. 126-129 show the effect of H_2 addition on cylinder pressure and heat release process at 10% load operation. As show in Fig. 126, the addition of H_2 at 10% load was shown to retard the initiation of combustion and reduce the cylinder pressure after the combustion was initiated. As shown in Fig. 127, the addition of a small amount of H_2 (<4%) slightly reduced the peak cylinder pressure observed at gradually retarded phasing. Further increasing the addition of H_2 beyond 4.5% was shown to reduce substantially peak cylinder pressure and retard the phasing when peak cylinder pressure was observed. As shown in Fig. 128, with the addition of H_2 less than 5%, the reduced cylinder pressure was due mainly to the retarded combustion phasing without affecting the peak heat release rate. Further increasing the addition of H_2 beyond 5% retarded further the combustion phasing and substantially inhibited the premixed combustion process accompanied with a much enhanced late diffusion combustion. As shown in Table 24, the retarded combustion was also due mainly to the retarded injection timing with notable contribution of lengthened ignition delay for the addition of H_2 over 6%. As shown in Fig. 129, the significant effect of H_2 addition on peak heat release rate was observed when the amount of H_2 added reached 6%.

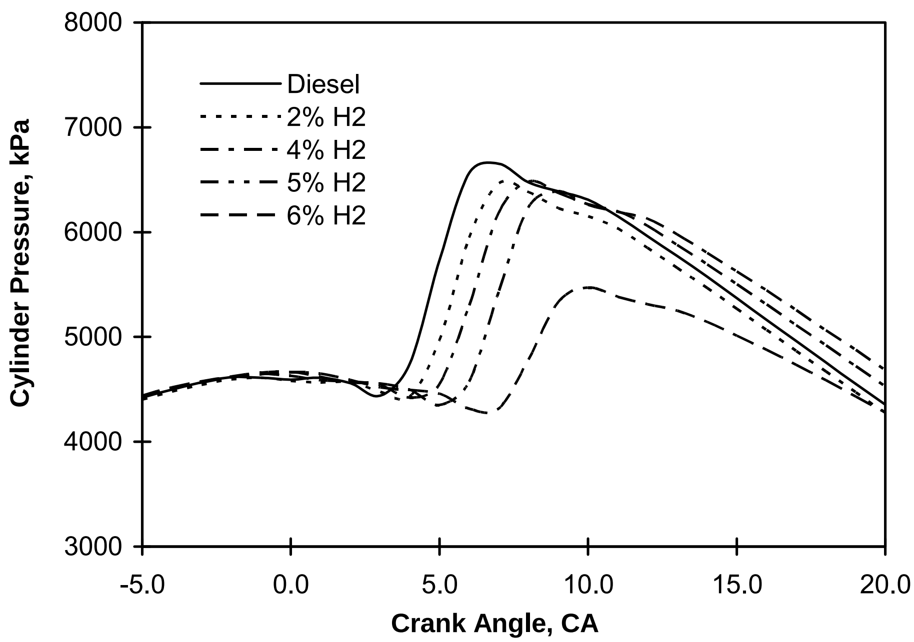

Figure 126 Effect of H_2 Addition on Cylinder Pressure, N=1200 RPM, 10% Load

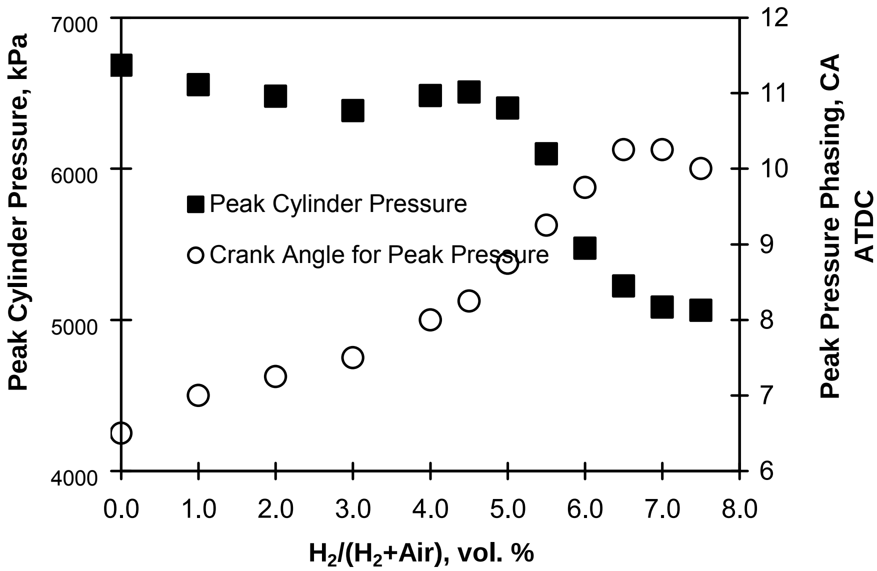

Figure 127 Effect of H_2 Addition on Heat Release Rate and Its Phasing, N=1200 RPM, 10% Load

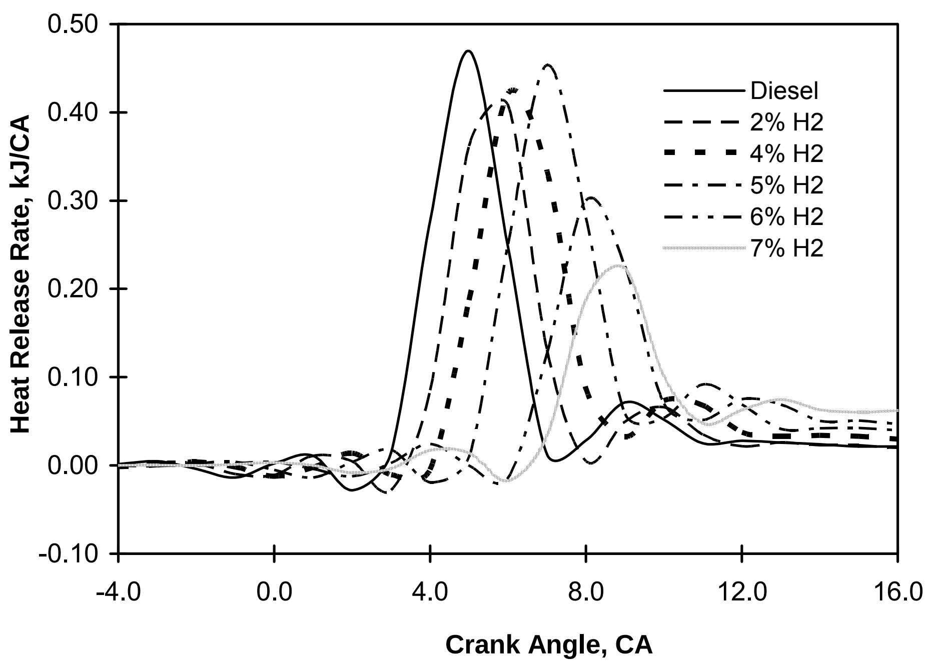

Figure 128 Effect of H_2 Addition on Heat Release Rate, N=1200 RPM, 10% Load

Table 24 Effect of H_2 Addition on Start of Injection (SOI), Start of Combustion (SOC) and Ignition Delay when Operated at 1200 rpm, 10% Load

$H_2/(H_2+Air)$, vol. %	SOI, CA ATDC	SOC, CA ATDC	Ignition Delay, CA
0	-1.1	3.18	4.28
2	-0.35	3.8	4.15
4	0.0	4.36	4.35
5	0.9	5.22	4.32
6	1.9	6.54	4.64
7	2.2	7.15	4.95

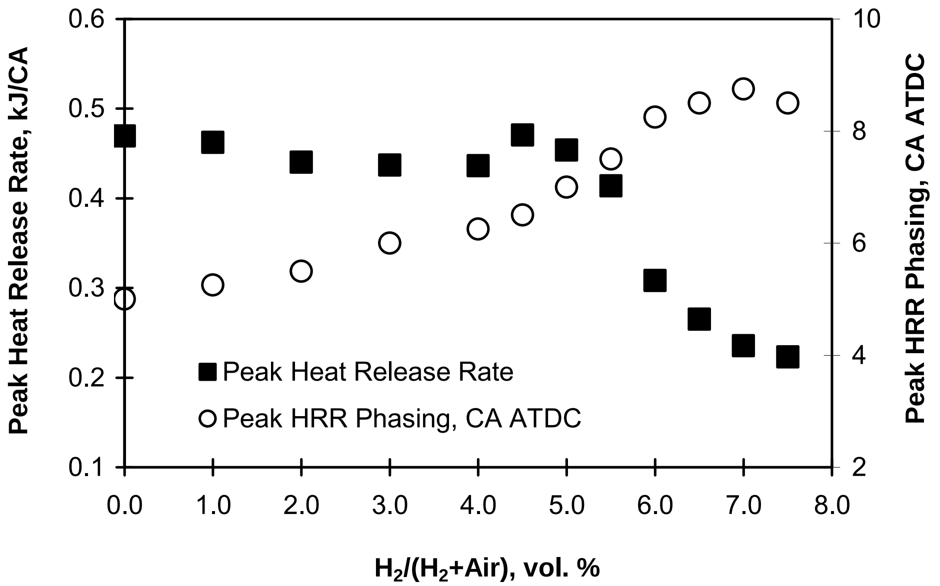

Figure 129 Effect of H_2 Addition on Peak Heat Release Rate and Its Phasing, N=1200 RPM, 10% Load

Figure 130 shows the effect of engine load on the heat release process of pure diesel operation. The increase in engine load was shown to advance the combustion phasing, inhibit the heat release process of premixed combustion and substantially enhance and lengthen the diffusion combustion process. When operated at low load, the peak heat release rate was observed at premixed combustion stage. When operated at high load, the featured early injection of large amount of diesel fuel, and also higher compression temperature and pressure resulted in a weak premixed combustion. In comparison, the mixing controlled diffusion combustion was substantially enhanced and lengthened. The diffusion combustion gradually developed into the main combustion process. When operated at high load, the peak heat release rate was observed at the diffusion combustion stage.

Figure 131 shows the effect of engine load on the heat release process of H_2-diesel dual fuel operation with the addition of 6% H_2. Similar to pure diesel operation, the increase in engine load was shown to advance the phasing of both premixed and diffusion combustion resulting from the advanced fuel injection. The maximum heat release rate of premixed combustion was shown to increase at first and then drop gradually with the development of a strong diffusion combustion. With the increase in engine load, the burning of an increasing amount of H_2 produced a unique and heat release peak as shown for 50% and 70% load operation. The diesel engine featured two-stage (premixed and diffusion diesel combustion) gradually develop into a H_2-diesel dual fuel engine featured three-stage combustion process including premixed and diffusion combustion of diesel fuel, and fast combustion of H_2 burned by numerous turbulent flame as shown in Fig. 131 for 70% load operation.

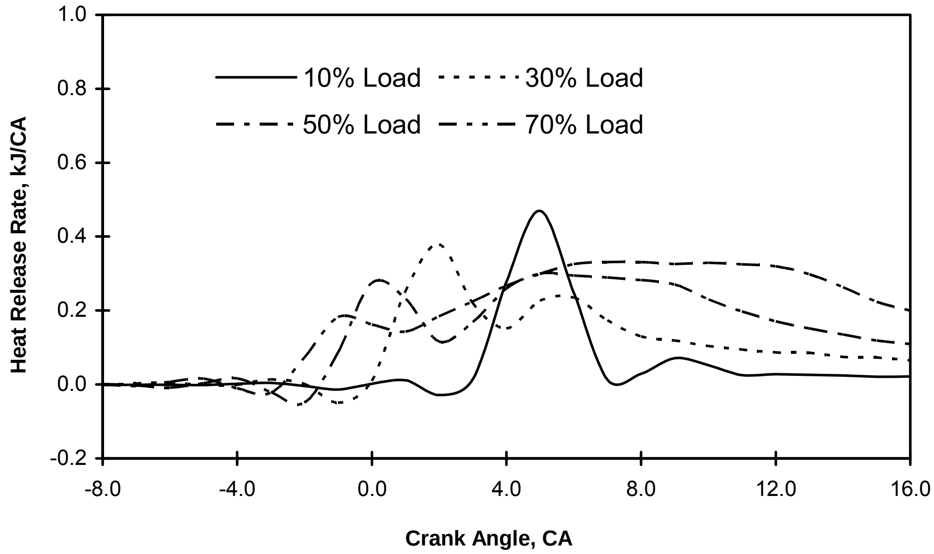

Figure 130 Effect of Engine Load on Heat Release Rate, N=1200 RPM, Diesel Only

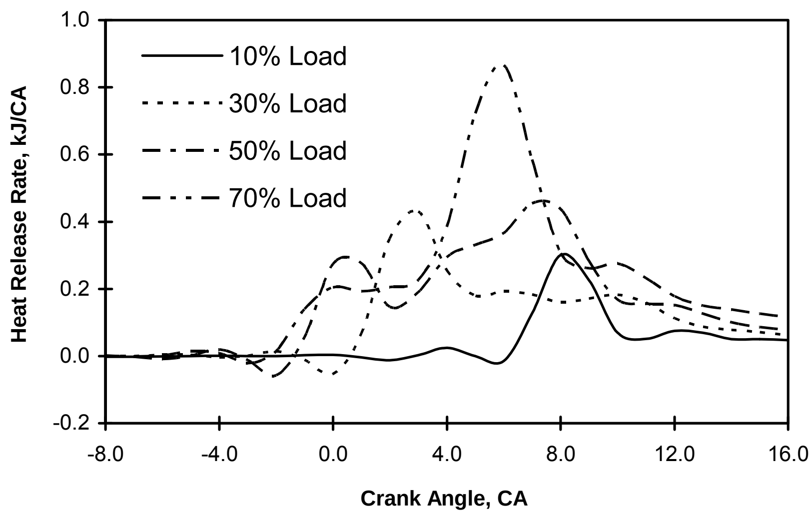

Figure 131 Effect of Engine Load on Heat Release Process, N=1200 RPM, $H_2/(H_2+Air)=6\%$

6.7 Summary

The effect of H_2 addition, engine load, engine speed and diesel fuel flow rate on the performance, combustion, and exhaust emissions of a 2004 Mack MD11 diesel engine has been experimentally investigated without modifying the engine control and fuel injection strategies. Following are a brief summary based on the results obtained in this research:

- The addition of H_2 into this diesel engine reduced the emissions of PM. The extent of reduction in PM emissions depended on the amount of H_2 added and engine load. With the addition of H_2 up to 7.5% at 10%-70% load, the maximum PM reduction of 65% to 80% were obtained. The PM emissions measured using the 13-mode emissions cycle were reduced by 17.5% and 27.5% for the H_2 addition of 2% and 4%, respectively.

- When measured using the 13-mode ESC emission cycle, the addition of 2% H_2 into the intake mixture was found to have negligible effect on NO_x emissions. The addition of 4% H_2 was shown to increase NO_x emissions by 3.96%.

- The examined at low load, addition of H_2 was shown to have small effect on NO_x emissions with the exception of 10% load with a relatively large amount of H_2 addition (>4%), which reduced the NO_x emissions accompanied with a substantial deterioration in the brake thermal efficiency (-6.9%-2.9% with H_2 addition of 4%-7.5%).

- When operated at medium to high load (30%-70%) operation, the addition of a small amount of H_2 (<3-5%) was shown to slightly reduce the emissions of NO_x. The addition of a relatively large amount of H_2 (>3% for 50% load and >5% for 30% and 70% load) increased the emissions of NO_x. When operated at full load, the addition of H_2 was shown to have negligible effect on NO_x emissions with the addition of H_2 up to 5% tested in this research.

- The addition of H_2 reduced substantially the emissions of CO when operated at low to medium load. When operated at high load, the addition of small amount of H_2 increased the emissions of CO with its maximum value observed at 4% H_2. Increasing the amount of H_2 beyond 4% reduced substantially the emissions of CO. When measured using the 13-mode ESC cycle, the addition of 2% H_2 into the diesel engine was shown to reduce CO emissions by 2.7%. The addition of 4% H_2 increased CO emissions by 8%.

- The addition of H_2 reduced the emissions of HC with the exception of 10% load operation. When operated at 10% load, the addition of H_2 less than 5% had a negligible effect on the emissions of HC. The addition H_2 beyond 5% reduced substantially the emissions of HC. When measured using the 13-mode ESC emission cycle, the addition of 2% H_2 increased HC emissions by 8.2%. The addition of 4% H_2 had a negligible effect on HC emissions.

- The addition of a relatively small amount of H_2 lowered the brake thermal efficiency. The desirable positive effect of H_2 in improving the brake thermal efficiency was obtained with the addition of H_2 at relatively large amounts.

- The experimental data demonstrated the presence of the minimum limit to obtain a positive effect on the brake thermal efficiency. Such a limit increased with the reduction in engine load. The addition of a small amount of H_2 into diesel engine was not recommended especially at low load operation.

- With the addition of 6% H_2 in the intake air, the improvement to the brake thermal efficiency was found to be 1%-4% for 20%-70% load operation, which is much lower than the

14-4% improvement in the brake thermal efficiency of the 1999 Cummins ISM370 diesel engine for 15% -70% load operation.

- The exhaust emissions of H_2 at low load operation could be a safety issue. The maximum H_2 emission of 1.4% (dry) was obtained with the addition of 6% H_2 at 10% load operation. The emissions of H_2 can be dramatically reduced with the increase in engine load.

- The addition of H_2 to the diesel engine was found to significantly affect the combustion process including the peak cylinder pressure, combustion phasing, and peak heat release rate. When added at large amounts under high load, a featured three-stage combustion process of H_2-diesel dual fuel engine can be observed.

7 Feasibility of Improving the Brake Thermal Efficiency of Heavy-Duty Diesel Engines Using On-Board H_2-Production Technologies

The experimental results reported in Section 5 and 6 have demonstrated the effect of H_2 addition on the engine performance, combustion and exhaust emissions of heavy-duty diesel engines. The addition of H_2 into two heavy-duty diesel engines was shown to substantially reduce the emissions of PM. However, the addition of H_2 to the 1999 Cummins ISM370 engine was shown to enhance the emissions of NO_x with exception at low load operation with the addition of relatively large amount of H_2. In comparison, the addition of H_2 to the 2004 Mack MD11 diesel engine was shown to have negligibly effect on the emissions of NOx at low load and slightly reduce the NOx emissions at high load operation. The substantial reduction to the emissions of NO_x through the addition of H_2 into a heavy-duty diesel engine seems infeasible. In this section, the feasibility of improving the brake thermal efficiencies of heavy-duty diesel engines through the integration of the on-board H_2-production technologies is to be examined.

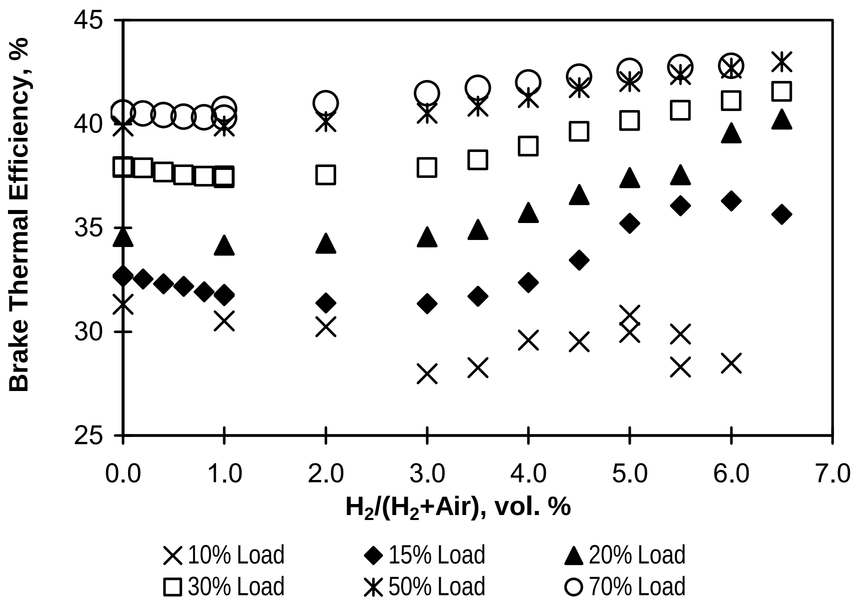

Figure 132 Effect of H_2 Addition and Engine Load on Brake Thermal Efficiency (BTE), N=1200 RPM, 10%-70% Load, 1999 Cummins ISM370 Diesel Engine. BTE was Calculated Using the Lower Heating Values of Diesel and H_2 without Accounting for the Extra Energy Cost for H_2 Production

Figs. 132 and 133 show the effect of H_2 addition on the brake thermal efficiency of the 1999 Cummins ISM370 and 2004 Mack MD11 diesel engine, respectively. It should be noted that the data shown in Figs. 132 and 133 was calculated using the lower heating values of diesel fuel and H_2 without accounting for the extra energy cost for the production and process of H_2 fuel. As shown in Figs. 132 and 133, the addition of a relatively large amount of H_2 at medium to high load was shown to improve the brake thermal efficiency. In comparison, the addition of H_2 at relatively small amounts was shown to deteriorate the brake thermal efficiency. The addition of H_2 at 10% load operation was shown to deteriorate the brake thermal efficiency for the range of operation examined.

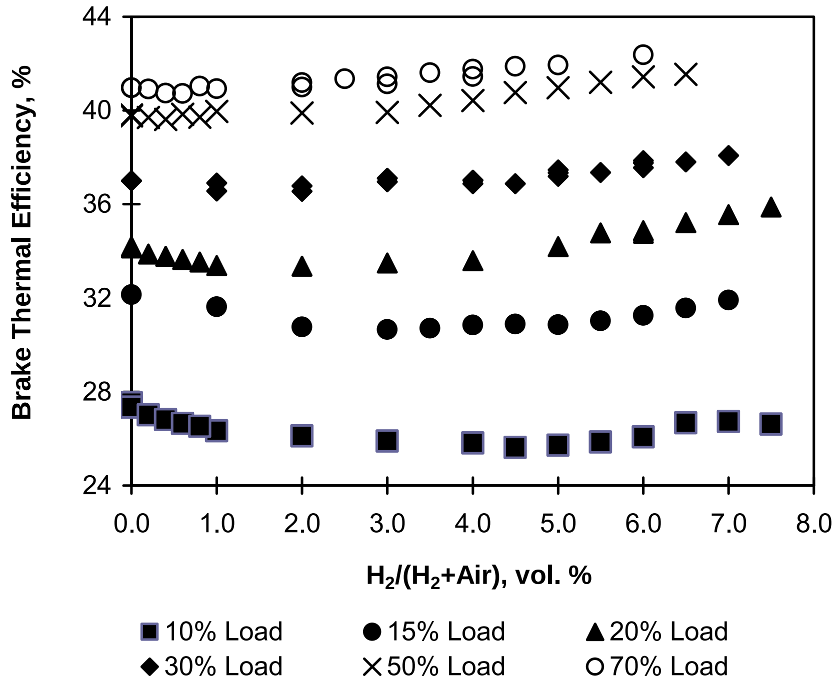

Figure 133 Effect of H_2 Addition and Engine Load on Brake Thermal Efficiency, N=1200 RPM, Load 10%-70%, 2004 Mack MD11 Diesel Engine. BTE was Calculated Using the Lower Heating Values of Diesel and H_2 without Accounting for the Extra Energy Cost for H_2 Production.

There is increasing interest to improve the brake thermal efficiency of diesel engines through on-board production of H_2 with the consumption of the on-board energy resources such as diesel through gas reforming technologies or electricity generated on-board by H_2O electrolysis technologies. In this section, the feasibility of improving the brake thermal efficiency of heavy-duty diesel engines through the integration of the on-board H_2 production devices is examined. The minimum H_2 flow rate needed to achieve a positive effect on the brake thermal efficiency will be presented. The feasibility of using on-board H_2 production devices to improve the brake thermal efficiency will be analyzed and discussed with H_2 production efficiency obtained by literature review and assumption made in this research.

7.1 Minimum H_2 Flow Rates Needed to Improve the Brake Thermal Efficiency without Considering the Extra Energy Cost of H_2 Production

In sections 5 and 6, the effect of H_2 addition in improving the brake thermal efficiency of both engines was examined using the lower heating value of diesel and H_2 without accounting the extra energy cost associated with the production of H_2. As shown in Figs. 134 and 135, the addition of relatively large amounts of H_2 to a diesel engine improved the brake thermal efficiency at medium to high load operation. In comparison, the addition of H_2 at relatively small amounts was shown to deteriorate the brake thermal efficiency. As shown in Figs. 134 and 135, the addition of H_2 into the 2004 Mack MD11 diesel engine was less effective in improving the brake thermal efficiency compared to that of the 1999 Cummins ISM370 diesel engine.

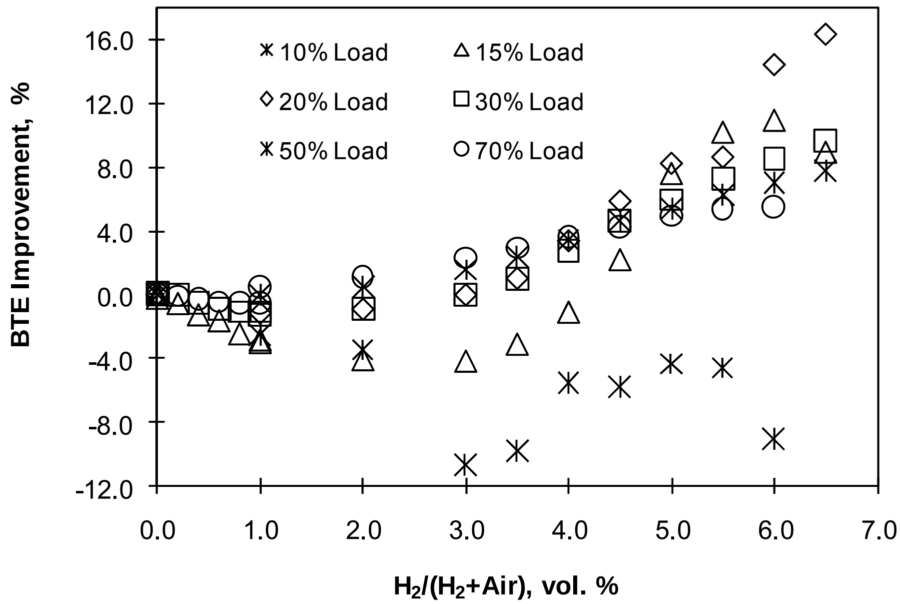

Figure 134 Effect of H_2 Addition in Improving the Brake Thermal Efficiency, N=1200 rpm, 10%-70% Load, 1999 Cummins ISM370. BTE was Calculated Using the Heating Value of H_2 without Accounting for the Energy Cost of H_2 Production

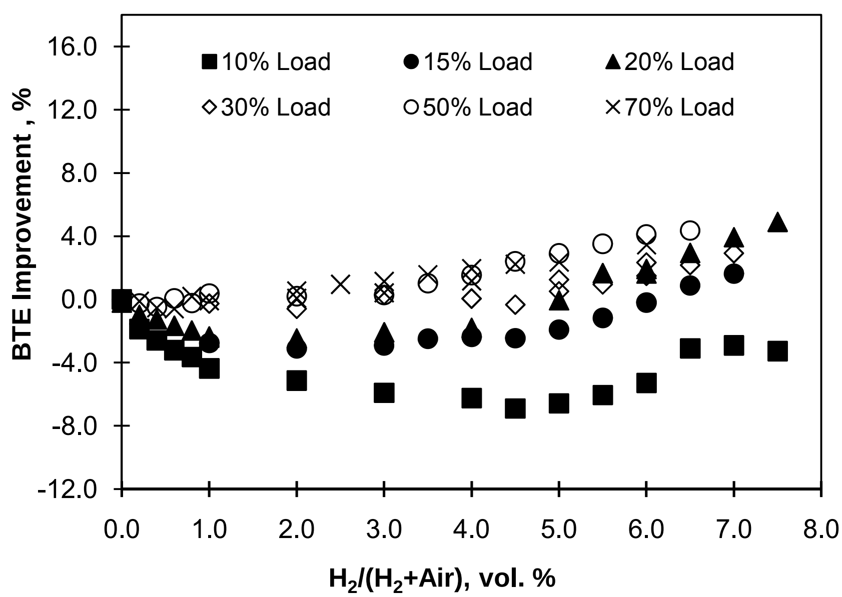

Figure 135 Effect of H_2 Addition in Improving the Brake Thermal Efficiency. N=1200 RPM, 10%-70% Load, 2004 Mack MD11 Diesel Engine. BTE was Calculated Using the Heating Value of H_2 without Accounting for the Energy Cost of H_2 Production

This was further demonstrated by comparing the effect of 6% H_2 addition in improving the brake thermal efficiency as shown in Fig. 136. The addition of 6% H_2 was shown to substantially improve the brake thermal efficiency of the 1999 Cummins ISM370 diesel engine especially at low to medium load. In comparison, its addition was found to have mild effect on the brake thermal efficiency of the 2004 Mack MD11 diesel engine.

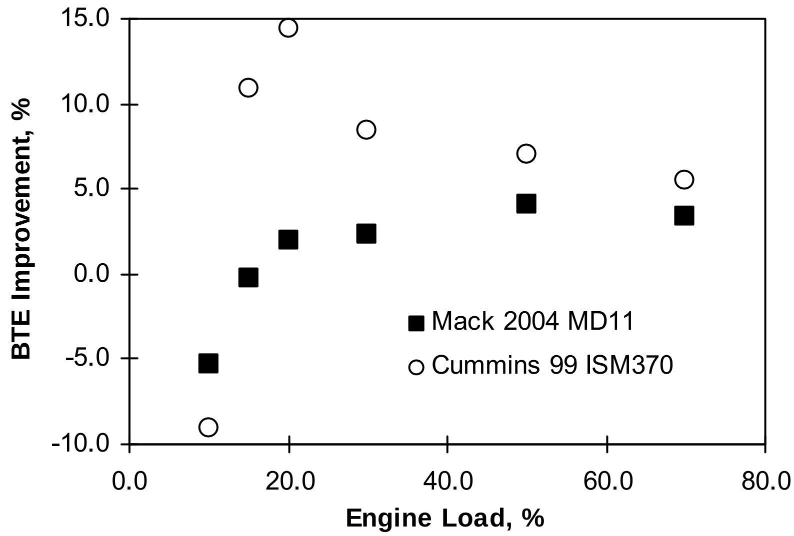

Figure 136 Effect of Engine Load on the Improvement to the Brake Thermal Efficiency, N=1200 RPM, $H_2/(H_2+Air)=6\%$, BTE was Calculated Using the Heating Value of H_2 without Considering the Energy Cost of H_2 Production.

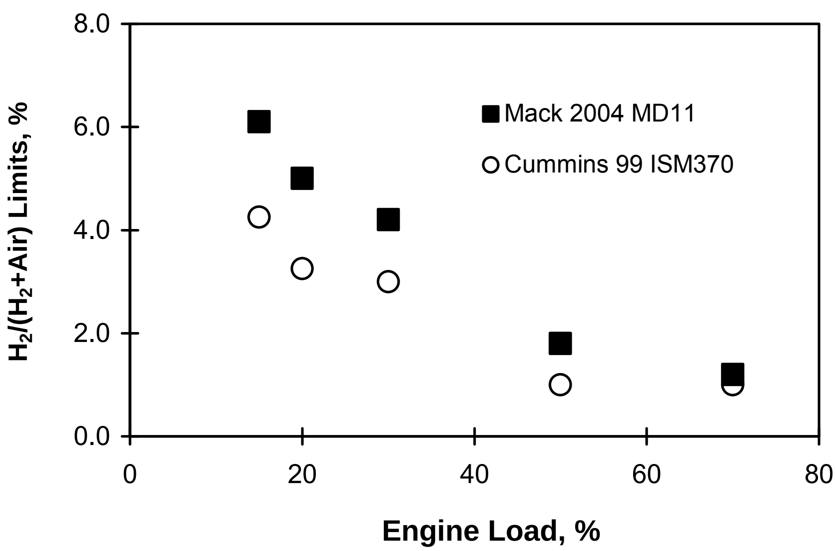

Figure 137 Effect of Engine Load on the Minimum H_2 Supplementation Needed for Positive Effect on the Brake Thermal Efficiency, N=1200 RPM. BTE was Calculated Using the Heating Value of H_2 without Considering the Energy Cost of H_2 Production.

The data showed in Figs. 134 and 135 also demonstrated that there existed a minimum H_2 supplementation limit for a positive effect in improving the brake thermal efficiency. As shown in

Fig. 137, it required more H_2 for the 2004 Mack MD11 engine to obtain improved brake thermal efficiency than the 1999 Cummins ISM370 engine.

The minimum $H_2/(H_2+Air)$ limit measured in each load was processed to obtain the minimum volumetric H_2 flow rate needed for a positive effect on the brake thermal efficiency. As shown in Table 25, the positive effect of H_2 addition on the brake thermal efficiency of the 1999 Cummins ISM370 engine required minimum H_2 flow of 272.7 l/m to 90.3l/m for 15% to 70% load operation at 1200 RPM. As shown in Table 26, the positive effect of H_2 addition on the brake thermal efficiency of the 2004 Mack MD11 engine required the minimum H_2 supplementation rate 284.3 l/m to 152.0 l/m. for 15% to 70% load operation at 1200 RPM. It should be noted that these are the H_2 flow rates needed for H_2-diesel dual fuel engine to produce a brake thermal efficiency comparable to that of pure diesel operation. The addition of H_2 lower than the flow rates described in Table 25 and 26 deteriorated the brake thermal efficiency.

Table 25 Minimum H_2 Flow Rate Needed for a Positive Effect in Improving the Brake Thermal Efficiency (BTE) of the 1999 Cummins ISM370 Diesel Engine, N=1200 RPM

Load	$H_2/(H_2+Air)$ vol. Limit for Positive Effect on BTE	H_2 Flow, kg/hr	H_2 Flow, L/min
10%	NA	N.A.	N.A.
15%	4.25%	1.46	272.7
20%	3.25%	1.15	215.0
30%	3.0%	1.18	220.6
50%	1.0%	0.48	90.3
70%	1.0%	0.59	109.4

Table 26 Minimum H_2 Flow Rate Needed for a Positive Effect in Improving the Brake Thermal Efficiency of the 2004 Mack MD11 Diesel Engine, N=1200 RPM

Load	$H_2/(H_2+Air)$ Vol. Limit for Positive Effect on BTE	H_2 Flow, kg/hr	H_2 Flow, L/min
10%	N.A.	N.A.	N.A.
15%	6.0%	1.52	284.3
20%	5.0%	1.39	259.3
30%	4.2%	1.38	257.8
50%	1.8%	0.93	172.7
70%	1.2%	0.81	152.0

7.2 On-Board H_2 Production Using Water-Electrolysis Technology

Hydrogen can be produced on-board through water-electrolysis technologies using electricity generated on-board by consuming the mechanical work produced by the diesel engine. Table 27 summarizes the H_2 production efficiency (GJ H_2/GJ electricity) of existing and advanced electrolysers reported by International Energy Agency [Mandil, 2005]. As shown in Table 27, the cell efficiency of the traditional electrolyser was lower than 70%. Based on the experience obtained through demonstration projects, the efficiency of onsite H_2 production devices using

H_2O electrolysis was lower than 60% when the energy consumption of H_2O electrolyser and pump were included as shown in Table 28. When performing the Well-to-Wheel analysis of producing H_2 by on-site H_2O electrolysis, Huang and Zhang assumed H_2 production efficiency of 71.5% for 50% probability. Accordingly, it is very aggressive to assume 70% as the overall H_2 production efficiency of the on-board H_2O electrolysis system. It is well accepted that a smaller system usually is less efficient. In this research, it is assumed that the electricity is generated on-board when the engine is operated at medium to high load with a brake thermal efficiency of 40%. The on-board generator will convert the mechanical work of the engine to electricity with an efficiency of 95%. As shown in Table 29, the overall efficiency of on-board production of H_2 through H_2O electrolysis using on-board electricity is 26.6% when considering on-board diesel fuel as the original energy resource.

Table 27 Energy Conversion Efficiency of Existing and Advanced H_2O Electrolysers [Mandil, 2005]

Technologies	Conventional electrolyser	Advanced alkaline electrolysers	Inorganic membrane electrolyser	PEM electrolyser	SOFC High Temp electrolyser
Cell Efficiency	66-69%	69-77%	73-81%	73-84%	81-86%

Table 28 Design and On-site Operation Data of H_2 Production Unit Based on H_2O Electrolysis [Stolzenburg, et al., 2008]

On-site H_2 Production Unit Location	Amsterdam	Hamburg
Production Range (Nm^3/hr)	15-60	30-60
Designed Cell Efficiency (Stack and Pumps)	63%	63%
On-site Operation Efficiency	60%	58%

Table 29 Efficiency of H_2 Production through Water-Electrolysis Using the Electricity Generated on-Board by Consuming the Mechanical Work Produced by the Diesel Engine

Sub-system	Diesel to Mech. Efficiency Using Diesel Engine	Mech. Work to Electricity Conversion Eff.	H_2 Production by H_2O Electrolysis	Overall Eff. from Diesel to H_2 Based on Heating Value
Efficiency	40%	95%	70%	26.6%

Based on the overall efficiency of the on-board H_2 production by H_2O electrolysis using power produced by the diesel engine, the effect of adding the H_2 produced on-board to the diesel engine on the overall brake thermal efficiency was examined using the diesel consumption and lower heating value of diesel fuel. As shown in Figs. 138 and 139, the integration of on-board H_2 production using H_2O electrolysis with a diesel engine resulted in substantially lower overall thermal efficiency of the integrated diesel engine system. The more H_2 added, the lower the overall brake thermal efficiency obtained. This is due mainly to the substantial energy loss during the production of on-board H_2. As shown in Table 29, the energy loss includes 60% from diesel fuel to mechanical work, 5% from mechanical work to electricity, and 30% from H_2O to H_2

fuel. The conversion of diesel fuel to H_2 using diesel-mechanical work-electricity-H_2 loop account for a loss of 73.4% of the chemical energy. Unless substantial improvements to the efficiency of H_2O electrolysis technology can be achieved it is infeasible to improve the brake thermal efficiency of heavy-duty diesel engines by the integration of H_2O electrolysers.

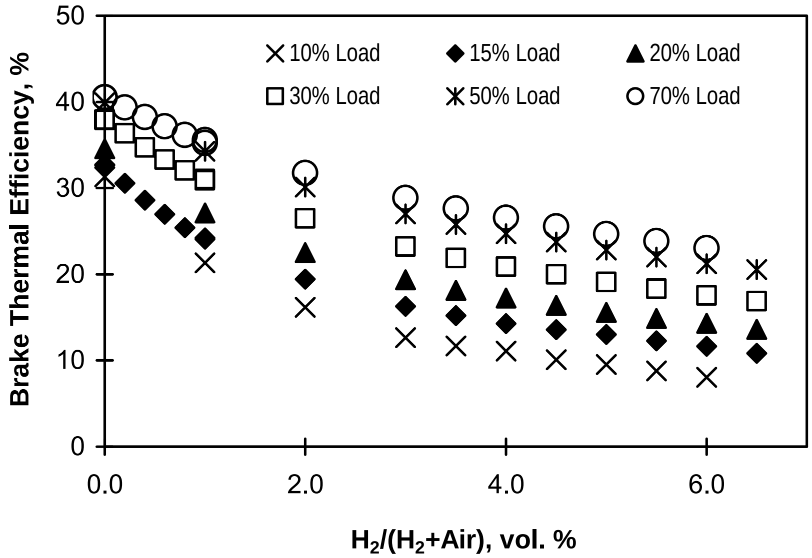

Figure 138 Effect of H_2 Addition and Engine Load on the Overall Brake Thermal Efficiency with the On-board Production of H_2 Using H_2O Electrolysis Technology, N=1200 RPM, 10%-70% Load, 1999 Cummins ISM370 Diesel Engine,

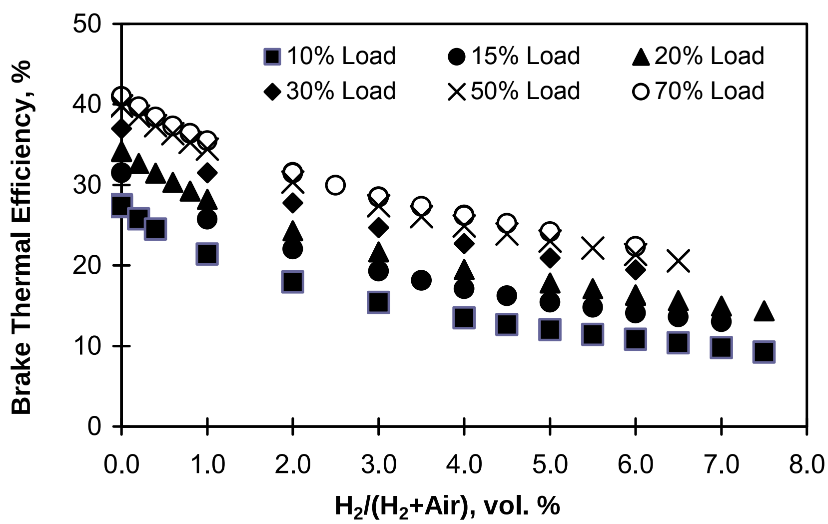

Figure 139 Effect of H_2 Addition and Engine Load on the Overall Brake Thermal Efficiency with the On-board Production of H_2 Using H_2O Electrolysis Technology, N=1200 RPM, 10%-70% Load, 2004 Mack MD11 Diesel Engine

7.3 On-Board H_2 Production through Gas-Reforming of Diesel Fuel

Hydrogen can also be produced on-board from most fossil fuels using steam reforming, partial oxidation, or auto-thermal reforming technologies [Carpenter, et al., 1999]. The raw syngas fuels produced on-board usually contain H_2 in the presence of CO, CO_2, H_2O, N_2 and a small amount of O_2. The syngas can either be further processed to remove N_2, CO_2 and CO for the production of H_2 with high purity or directly burned in diesel engine by dual fuel combustion mode. For example, Brown [2001] investigated the fuel candidates suitable for the on-board H_2 production for fuel cell vehicles. The fuel candidates explored included methanol, ethanol, gasoline, diesel and natural gas. Based on the data reported by Brown, the theoretical efficiency of H_2 production by steam reforming of diesel fuel was about 81%. Bromberg, et al. [2004] developed a Plasmatron Fuel Reformer to produce H_2-rich gases using traditional gasoline, diesel and bio-diesel. For homogeneous reformation, the ratio of the heating value of the H_2, CO and light HC by-products relative to that of liquid fuel was typically 60-70%. Galloni and Minutillo [2007] examined conversion of gasoline fuel to reformate gas produced on-board. The reformer efficiency was calculated by considering chemical energy of H_2 and CO in the reformate gases. The conversion efficiency obtained was about 80%. Minutillo [2005] investigated the reforming efficiency of an exothermal partial oxidation process using ambient air as oxidant for the production of H_2-rich gaseous fuel mixture. As shown in Table 30, the reformer efficiencies measured were 73% and 65% for the reforming of propane and methane, respectively. It is expected that the addition of CO to the diesel engine as supplemental fuel is less effective in improving the brake thermal efficiency compared to that of H_2. The diluents such as CO_2 and N_2 in syngas will also deteriorate the combustion process and reduce the brake thermal efficiency of diesel engines.

Table 30 The Reforming Efficiency of the De-centralized Gas Reforming Technologies Reported by Minutillo [2005]

Fuel	O/C Ratio, vol.	Reformer Efficiency (%, LHV)
Propane	0.35	72
	0.40	73
Methane	0.45	63
	0.46	65

Table 31 The Reforming Efficiency of the De-centralized Reforming Technologies Reported by IEA [Mandil 2005]

Technologies	Conversion Efficiency (%, LHV)
Steam Methane Reforming	71-76
Partial Oxidation	66-76
Auto-thermal Reforming	66-73
Potential of Novel Reforming Technique	>75

The H_2-rich syngas produced on board can be further processed for the production of pure H_2 with the consumption of energy. For example, Mandil [2005] of International Energy Agency (IEA) reviewed the technologies suitable for the production of H_2. As shown in Table 31, the conversion efficiencies of current technologies were lower than 76%. Stolzenburg et al. [2008] reported the achievements of H_2 production by steam methane reforming in the Clean Urban Transport for Europe (CUTE) project. As shown in Table 32, the conversion efficiency obtained for Madrid and Stuttgart station were 62% and 65%, respectively. Huang and Zhang [2006] examined the different pathways of producing pure H_2 using natural gas and reported H_2 production efficiency of 67% for the probability of 50% and increased efficiency of 72% for the probability of 80%. Based on the review of literature, it is very aggressive to assume the overall conversion efficiency of 75% for the on-board H_2 production by gas reforming.

Table 32 Design Data of H_2 Production Units Based on Steam Methane Reforming [Stolzenburg, 2008]

Technologies	Madrid	Stuttgart
H_2 Production Range (Nm^3/hr)	30-50	50-100
Thermal Efficiency (%, LHV)	62%	65%

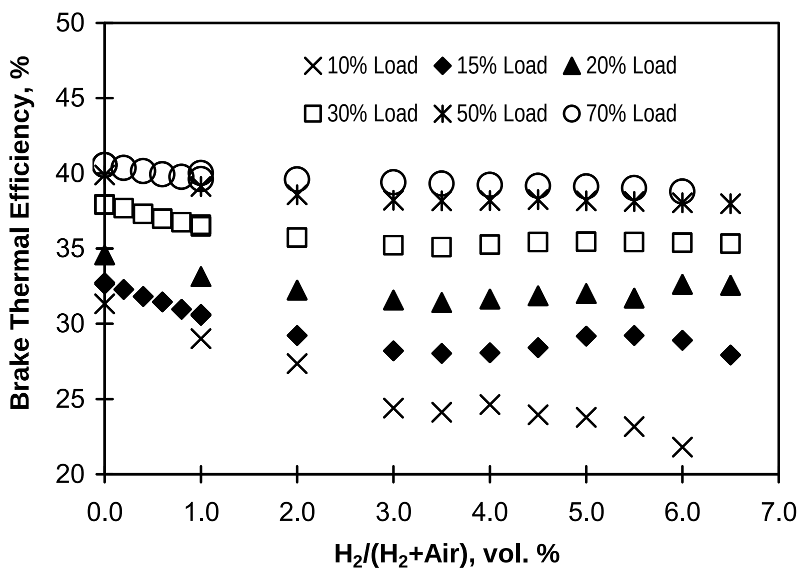

Figure 140 Effect of Engine Load and Adding H_2 Produced on-Board Using Gas Reforming Technologies on the Overall Brake Thermal Efficiency Calculated Using the Overall Diesel Fuel Consumption,, N=1200 RPM, 10%-70% Load, 1999 Cummins ISM370 Diesel Engine

Based on the assumption of 75% diesel-H_2 conversion efficiency using the on-board gas reforming technologies, the effect of the addition of H_2 produced on-board to the engine intake on the brake thermal efficiency of these two heavy-duty diesel engines was examined. As shown in Figs. 140 and 141, the integration of the on-board H_2 production device with heavy-duty diesel engines was shown to reduce the overall brake thermal efficiency but at smaller rate compared to the integration of H2O electrolysers as shown in Figs. 138 and 139. Unless substantial improvements to diesel engine combustion through the addition of H_2 or technique breakthrough in gas reforming technologies can be achieved, the improvement to the brake thermal efficiency of heavy-duty diesel engines by the application of gas reforming of diesel fuel is infeasible.

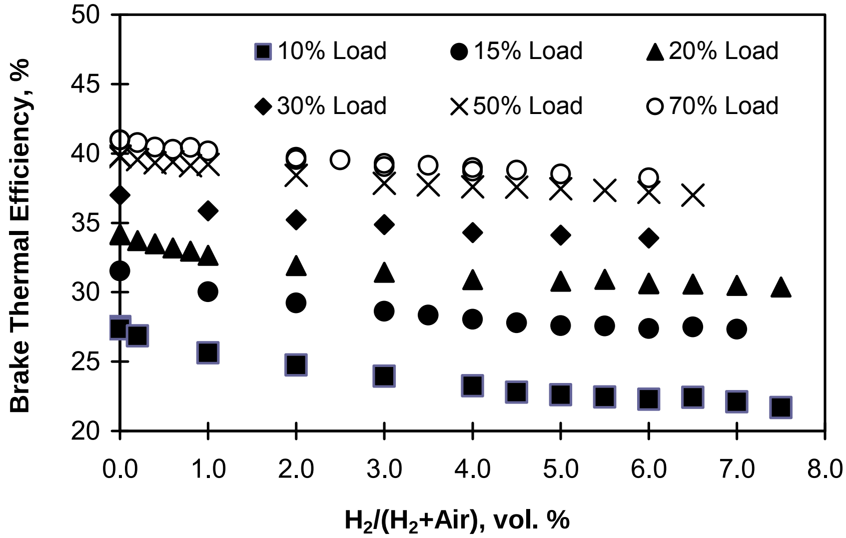

Figure 141 Effect of Engine Load and Adding H_2 Produced on-Board Using Gas Reforming Technologies on the Overall Brake Thermal Efficiency Calculated Using the Overall Diesel Fuel Consumption, N=1200 RPM, 10%-70% Load, 2004 Mack MD11

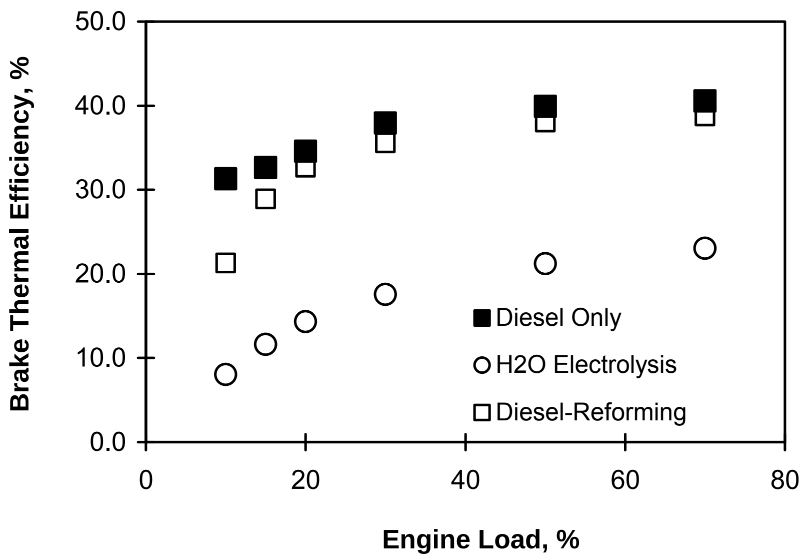

Figure 142 Effect of On-board Production of H_2 on the Overall Brake Thermal Efficiency, N=1200 RPM, $H_2/(H_2+Air)$=6%, 1999 Cummins ISM370 Diesel Engine, Overall Production Efficiency of On-Board H_2 Production Was Assumed as 26.6% and 70%, respectively, for Electrolysis and Gas Reforming Using Diesel as Original Fuel

Some researchers also accounted for the sensible thermal energy contained in the hot syngas produced on board. It should be noted that the operation of heavy-duty diesel engines prefer cool intake temperature for the reduction of PM and NO_x emissions and improvement of fuel consumption. The effective cooling of the hot syngas increases the load of the auxiliary system such as cooling fan and reduces the overall brake thermal efficiency of the engine.

The effect of different on-board H_2 production systems in deteriorating the brake thermal efficiency was also examined for the addition of 6% H_2. Compared to the on-board H_2 production using H_2O electrolysis, the application of on-board production of H_2 through gas reforming of diesel was found to have smaller reductions in the brake thermal efficiency as shown in Figs. 142 and 143, respectively. It is evident that it is very difficult for H_2O electrolyser to obtain positive effect on the brake thermal efficiency of heavy-duty diesel engine.

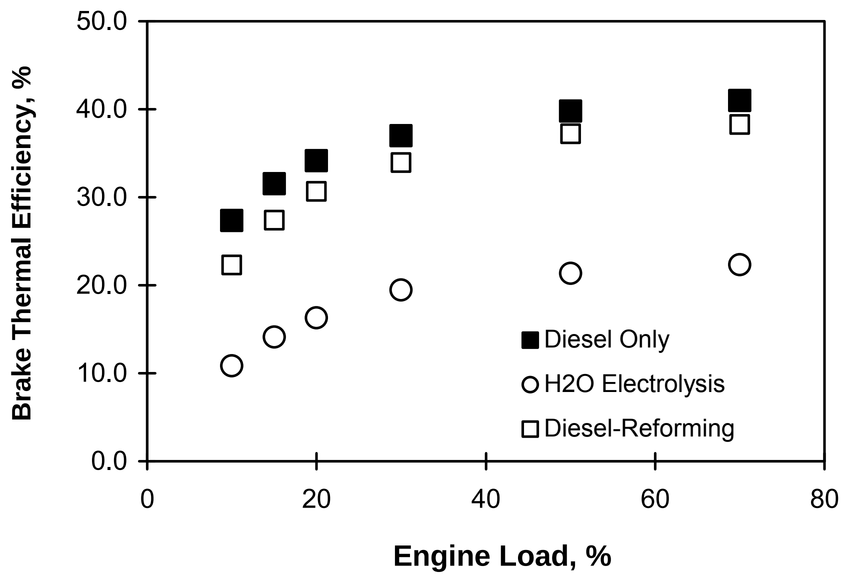

Figure 143 Effect of On-board Production of H_2 on the Overall Brake Thermal Efficiency, N=1200 RPM, $H_2/(H_2+Air)$=6%, 2004 Mack MD11 Diesel Engine, Overall Production Efficiency of On-Board H_2 Production Was Assumed as 26.6% and 70%, respectively, for Electrolysis and Gas Reforming Using Diesel as Original Fuel

7.4 Summary

Without modifying the combustion system and fuel injection strategies, the feasibility of improving the brake thermal efficiency of heavy-duty diesel engines using on-board H_2 production technologies was analyzed and discussed. Based on the review of literature and assumptions made in this research, the overall conversion efficiencies of on-board H_2 production using H_2O electrolysis and reforming technologies were obtained and used to examining their effect on the overall brake thermal efficiency when integrated with heavy-duty diesel engines. Based on the literature review, assumptions and experimental data obtained in this research, the following conclusions can be made:

- When evaluated on the lower heating value of diesel fuel and H_2 without accounting for the extra energy cost of H_2 production, the addition of H_2 to diesel engines could improve the brake thermal efficiency when added in a relatively large amount at medium to high load. When operated at 1200 RPM, the minimum H_2 flow rates needed for a positive effect in improving the brake thermal efficiency are 272.7 l/m -90.3 l/m and 284.3 l/m to 152.0 l/m for 15% to 70% load operation of the 1999 Cummins ISM370 and 2004 Mack MD11 engine, respectively. The addition of H_2 at a flow rate lower than these limiting values reduced the brake thermal efficiency.

- When evaluated on the lower heating value of the H_2 produced on-board and diesel fuel consumed, the overall H_2 production efficiency of the on-board H_2O electrolysis using engine power are about 26.6%. In comparison, the production efficiency of H_2 and syngas gas through gas reforming about 75% could be obtained.

- Based on the brake thermal efficiency of the two H_2-diesel dual fuel engines obtained in this research and assumption of 26.6% conversion efficiency of a H_2O electrolysis device using diesel as fuel, the integration of H_2O electrolysis devices with heavy-duty diesel engines was shown to reduce substantially the overall brake thermal efficiency. It was infeasible to improve the brake thermal efficiency of heavy-duty diesel engines by its integration with the on-board H_2 production devices using H_2O electrolysis technologies.

- Based on the brake thermal efficiency of the two H_2-diesel dual fuel engines measured in this research and assumption of 75% conversion efficiency of diesel to H_2 using gas reforming devices, the integration of the gas reforming devices with heavy-duty diesel engines was found to slightly reduce the overall brake thermal efficiency. Compared to the H_2O electrolysis technology, the integration of the gas reforming technology resulted in less reduction in the overall brake thermal efficiency.

8 Conclusions and Recommendations

This research investigated the performance, combustion, and emission characteristics of H_2-enriched diesel engines using a 1999 Cummins ISM370 (without EGR) and a 2004 Mack MD11 (with cooled EGR). The H_2 was mixed with air into the intake system well before entering the intake manifold with suitable backfire protection devices installed. The effect of H_2 addition, engine load, speed, and diesel fuel flow rate on the brake thermal efficiency, cylinder pressure, combustion process, and exhaust emissions of NO_x, PM, CO, HC, and unburned H_2 were explored. The engine load was varied from 10% to 70% with the addition of H_2 up to 7.5% into the intake mixture. The potential of H_2 enriching in reducing the emissions of PM and enhancing the formation of NO_x was discussed. The feasibility of improving the brake thermal efficiency of heavy-duty diesel engine using on-board H_2 production devices was analyzed based on the experimental data obtained in this research and those published in the literature. Based on the results obtained in this research, the following conclusions can be drawn:

- The addition of H_2 into these two diesel engines reduced substantially the emissions of PM. The extent of PM reduction depended mainly on the amount of H_2 supplemented and engine load.

- The addition of H_2 into the 1999 Cummins SM370 diesel engine was found to enhance the emissions of NO_x when operated at medium to high load. In comparison, the addition of small amount of H_2 at low load was shown to have negligible effect on the NO_x emissions. Its addition at relatively large amounts was shown to reduce the emissions of NO_x when operated at very narrow low load operational range.

- The addition of H_2 to the 2004 Mack MD11 diesel engine had weak effect on the emissions of NO_x. The addition of H_2 at medium to high load reduced slightly the emissions of NO_x. The substantial reduction of NO_x emissions was only observed with the addition of a relatively large amount H_2 (>4%) at very load (10%) operation accompanied with substantial reduction in the brake thermal efficiency.

- When measured using 13-mode ESC cycle, the addition of 2% and 4% H_2 into the 1999 Cummins ISM370 engine increased the emissions of NO_x by 5.3% and 15.5%, respectively. In comparison, the addition of 2% H_2 into the 2004 Mack MD11 diesel engine has negligible effect on the emissions of NO_x. The addition at 4% increased the emissions of NO_x by 4%.

- Based on the experimental data measured under constant load and that using 13-mode ESC cycle, the substantial reduction in NO_x emissions through the addition of H_2 into these heavy-duty diesel engines is infeasible.

- Without considering the extra energy cost associated with the production of H_2, the addition of relatively large amount of H_2 to heavy-duty diesel engines at medium to high load improved the brake thermal efficiency. In comparison, the addition of a relatively small amount of H_2 deteriorated the brake thermal efficiency. With the addition of 6% H_2 to the 2004 Mack MD11 engine, the improvement to the brake thermal efficiency was found to be 1%-4% for 20%-70% load operation. In comparison, the addition of 6% H_2 into the 1999 Cummins ISM370 diesel engine was shown to improve the brake thermal efficiency by 14%-4% for 15%-70% load operation.

- When the extra energy cost of H_2 production was considered, the integration of the on-board H_2 production devices using H_2O electrolysis and gas reforming of diesel fuel was shown to reduce the overall brake thermal efficiency of the two diesel engines. Based on the experimental data obtained in this research and the assumed diesel-to-H_2 conversion efficiency, the improvement to the brake thermal efficiency through the application of on-board H_2 production devices is infeasible.

- When operated at medium to high load, a featured three-stage combustion process of H_2-diesel dual fuel engine was observed. The unique high peak heat release rate observed in diffusion combustion was a combination of mixing controlled diffusion combustion of diesel fuel and the fast burning of H_2 by multi turbulent flame propagating through H_2-diesel-air mixture.

- The emissions of H_2 could be a concern for both safety and economic reasons. The maximum value of H_2 emissions obtained in this research was 1.4%, about 25% of the total amount of H_2 supplied into engine at 10% load. The addition of H_2 into diesel engine at low load should be avoided due to the low combustion efficiency of H_2 fuel and also the deterioration of the brake thermal efficiency.

Based on the results obtained in this research and those reported in the literature, the research team would like to make the following recommendations for future research in this area.

- The experimental data obtained in this research demonstrated the high H_2 emissions at low load operation. The research team recommends conducting a more detailed literature review to explore the potential benefits of engine-out H_2 emissions in improving the operation and performance of diesel after-treatment systems including the regeneration of DPF and operation of SCR. If unburned engine-out H_2 emissions can be effectively utilized to improve the operation of after-treatment system, it is possible to spontaneously improve the brake thermal efficiency and reduce exhaust emissions of NO_x and PM by reducing the fuel economy penalty associated with the application of after-treatment.

9 References

1 Abu-Jrai, A., Tsolakis, A., and Megaritis, A., 2007, "The Influence of H_2 and CO on Diesel Engine Combustion Characteristics, Exhaust Gas Emissions, and After Treatment Selective Catalytic NO_x Reduction", International Journal of Hydrogen Energy 2007; 32; 3565-3571

2 Bika A.S., Franklin L.M., and Kittelson D.B., 2008, "Emission Effects of Hydrogen as a Supplement Fuel with Diesel and Bio-Diesel", SAE Paper 2008-01-0648

3 Boehman, A.L., and Corre, O.L., 2008, "Combustion of Syngas in Internal Combustion Engines", Combustion Science and Technology, Vol. 180, pp1193-1206, 2008

4 Brunt, M. and Emtage, A., 1997, "Evaluation of Burn Rate Routines and Analysis Errors," Warrendale, PA, SAE Paper No. 970037, 1997

5 Brunt, M. and Pond, C., 1997, "Evaluation of Techniques for Absolute Cylinder Pressure Correction," SAE Paper 970036

6 Brunt, M. and Platts, K., 1999, "Calculation of Heat Release in Direct Injection Diesel Engines," SAE Paper No. 1999-01-0187

7 Bromberg, L., Cohn, D.R., Hadidi, K., Heywood, J.B., and Rabinnovich, A., 2004, "Plasmatron Fuel Reformer Development and Internal Combustion Engine Applications", 2004 Diesel Engine Emission Reduction (DEER) Workshop, Coronado, CA, August 29-September 2, 2004

8 Brown, L., 2001, "A Comparative Study of Fuels for On-board Hydrogen Production for Fuel-Cell-Powered Automobiles", International Journal of Hydrogen Energy, Vol. 26, pp.381-397

9 Carpenter, I., Edwards, N., Ellis, S., Frost, J., Golunski, S., Keulen, N.V., Pignon, J., and Reinkingh, J., 1999, "On-Board Hydrogen Generation for PEM Fuel Cells in Automotive Appications", SAE Paper 1999-01-1320

10 Das, L.M., 1990, "Hydrogen Engines: A Review of the Past and A Look into the Future", International Journal of Hydrogen Energy, Vol. 15, No. 6, pp.425-443

11 Furuhama, S., 1983, "State of the Art and Future Trends in Hydrogen-Fuelled Engines," International Journal of Vehicle Design, Vol. 4, pp. 359–385

12 Furuhama, S. and Fukuma, T. 1986, "High Output Power Hydrogen Engine with High Pressure Fuel Injection, Hot Surface Ignition and Turbo-charging," Int. J. Hydrogen Energy 11,399-407

13 Galloni, E., Minutillo, M., 2007, "Performance of a Spark Ignition Engine Fuelled with Reformate Gas Produced On-Board Vehicle", International Journal of Hydrogen Energy , Vol. 32, pp.2532-2538

14 Gopalakrishnan, R., Throop, M. J., Richardson, A., Lapetz, J. M., 2007, "Engineering the Ford H_2 IC Engine Powered E-450 Shuttle Bus", SAE Paper 2007-01-4095

15 Gopal, G., Rao, P. S., Gopalakrishnan, K. V., Murthy, B. S., 1982, "Use of hydrogen in dual-fuel engines," International Journal of Hydrogen Energy. Vol. 7, No. 3, pp. 267-272

16 Heywood, J.B., 1988, Internal Combustion Engine Fundamentals, McGraw-Hill, New York City, NY.

17 Homan, H. S., Reynolds, R. K., Deboer, P. C. T. and Mclean, W. J. 1979, "Hydrogen-Fuelled Diesel Engine without Timed Ignition," International Journal of Hydrogen Energy,

Vol. 4, pp. 315-325

18 Huang, Z. and Zhang, X., 2006, “Well-to-Wheel Analysis of Hydrogen Based Fuel-Cell Vehicle Pathways in Shanghai”, Energy 31, pp.471-489

19 Karim, G.A. and Klat S.R., 1982, “Experimental and Analytical Studies of Hydrogen as a Fuel in Dual Fuel Engine”, ASME Paper No. 75-DGP-9

20 Karim, G. A. 1976. “Hydrogen as a Fuel in Compression Ignition Engines”, Mechanical Engineering. April 1976. pp.34-39

21 Karim, G.A., 2003, “Hydrogen as a Spark Ignition Engine Fuel”, International Journal of Hydrogen Energy, Vol. 28, pp. 569 – 577

22 Kiesgen, G., Kluting, M., Bock, S., and Fisher, H., 2006, “The New 12-cylinder Hydrogen Engine in the 7 Series: The H_2 ICE Age Engine Begun”, SAE Paper 2006-01-0431

23 Kumar, M. S., Ramesh, A., Nagalingam, B., 2003, “Use of Hydrogen to Enhance the Performance of a Vegetable Oil Fuelled Compression Ignition Engine,” International Journal of Hydrogen Energy 28, pp.1143 – 1154

24 Li, H.L. and Karim, G.A. 2005, “An Experimental Investigation of S.I. Engine Operations on Lean Gaseous Fuels Mixtures”, Transaction of SAE, Journal of Engines, Vol. 114-3, pp. 1600-1608

25 Mandil, C., 2005, Prospects for hydrogen and Fuel Cells, International Energy Agency

26 Masood, M., Mehdi, S. N., Reddy, P. R. 2007, “An Experimental Investigations on a Hydrogen-Diesel Dual Fuel Engine at Different Compression Ratios,” Journal of Engineering for Gas Turbines and Power, Vol. 129 / 572-578

27 McWilliam, L. Megaritis, T. and Zhao, H, 2008, “Experimental Investigation of the Effects of Combined Hydrogen and Diesel Combustion on the Emissions of a HSDI Diesel Engine”, SAE 2008-01-0156

28 Minutillo, M., 2005, “On-Board Fuel Processor Modeling for Hydrogen-Enriched Gasoline Fueled Engine”, International Journal of Hydrogen Energy, Vol. 30, pp.1483-1490

29 Mohammadia, A. Shiojib, M., Nakaib, Y., Ishikurab, W., and Tabo, E. 2007, “Performance and Combustion Characteristics of a Direct Injection SI Hydrogen Engine”, International Journal of Hydrogen Energy, vol. 32, pp. 296-304

30 Munshi, S., Nedelcu, C.,Harris, J. Edwards. T, Williams, J.R., Lynch, F., Frailey, M.R., Dixon, G., Wayne, S., and Nine, R., 2004, “Hydrogen Blended Natural Gas Operation of a Heavy Duty Turbo-Charged Lean Burn Spark Ignition Engine”, SAE Paper 2004-01-2956

31 Saravanan, N. and Nagarajan, G., 2008, “An Experimental Investigation of Hydrogen Enriched Air Induction in a Diesel Engine System”, International Journal of Hydrogen Energy, Vol. 33 1769-1775

32 Saravanan, N., and Nagarajan, G., 2008, “An Experimental Investigation of Hydrogen as a Dual Fuel for Diesel Engine System with Exhaust Gas Recirculation Technique”, Int. Journal of Hydrogen Energy, Vol. 33, pp.422-427

33 Shirk, M.G., McGuire T.P., Gary L., Neal, and Haworth, D., 2008, “Investigation of a Hydrogen-Assisted Combustion System for a Light-duty diesel vehicles”, International Journal of Hydrogen Energy, Vol. 33, pp.7237-7244

34 Stockhausen, W.F., Natikin, R.J., Kabat, D.M., Reams, L., and Tang, X., 2002, “Ford P2000 Hydrogen Engine Design and Development Program”, SAE Paper 2002-01-0240

35 Stolzenburg, K., Tsatami, V., and Grubel, H., 2008, "Lessons Learned from Infrastructure Operation in the Cute Project", International Journal of Hydrogen Energy, Paper in Press, Accepted June 2008

36 Tang, X., Kabat, D.M., Natikin, R.J., and Stockhausen, W.F., 2002, "Ford P2000 Hydrogen Engine Dynamometer Development", SAE Paper 2002-01-0242

37 Tomita, E., Kawahara, N., Piao, Z., Fujita, S., and Hamamoto, Y., 2001, "Hydrogen Combustion and Exhaust Emissions Ignited with Diesel Oil in a Dual Fuel Engine", SAE Paper 2001-01-3503

38 Topinka, J., Gerty, M.D., and Heywood, J.B, 2004, "Knock Behavior of a Lean-Burn, H_2 and CO-Enhanced, SI Gasoline Engine Concept", SAE Paper 2004-01-0975

39 Tsolakis, A., Megaritis, A. "Partially Premixed Charge Compression Ignition Engine with On-Board H_2 Production by Exhaust Gas Fuel Reforming of Diesel and Biodiesel," International Journal of Hydrogen Energy 30 (2005) 731 – 745

40 White C.M., Steeper, R.R., and Lutz, A.E., 2006, "The Hydrogen-Fuelled Internal Combustion Engine: a Technical Review". International Journal of Hydrogen Energy 31, 1292-1350

41 Varde, K.S. and Frame G.A., 1983. "Hydrogen Aspiration in Direct Injection Type Diesel Engine-its Effect on Smoke and Other Engine Performance Parameters", International Journal of Hydrogen Energy, Vol. 8, pp.549-555

42 Wong, J.K.S., 1990, "Compression Ignition of Hydrogen in a Direct Injection Diesel Engine Modified to Operate as A Low-Heat-Rejection Engine", International Journal of Hydrogen Energy, Vol.15, No.7, pp.507-514

43 Woshni, G., 1967, "A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine,", SAE Paper No. 670931