“MODIFICATION AND EVALUATION OF FUEL INJECTION SYSTEM FOR IDI DIESEL ENGINE FUELED WITH JATROPHA STRAIGHT VEGETABLE OIL (JSVO)”

BY

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R K Tripathi

Declaration

I hereby declare that this submission is my own work and that, to the best of my knowledge and belief, it contains no material previously published or written by another person nor material which has been accepted for the award of any other degree or diploma of the university or other institute of higher learning, except where due acknowledgment has been made in the text.

Date: Mar 11 (R K Tripathi)

Certificate

This is to certify that the thesis entitled " Modification and Evaluation of Fuel Injection System for IDI Diesel Engine Fueled with Jatropha Straight Vegetable oil (JSVO)" submitted by Raj Kishor Tripathi to University of Petroleum & Energy Studies, Dehradun for the award of the degree of Doctor of Philosophy is a bona fide record of the research work carried out by him under my supervision and guidance. The content of the thesis, in full or parts have not been submitted to any other Institute or University for the award of any other degree or diploma.

Date: Mar 11 (Dr. P K Sahoo)

Place: UPES, Dehradun

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Executive Summary

Combustion of fossil fuels is the major factor for environmental degradation and limited reserve of fossil fuel generates the necessity of search for alternate fuel. There are enormous efforts made in this direction by the researchers. Most of researchers are working on production of new source of energy in the form of alternative fuel based upon the physico-chemical properties similar to diesel. The existing engines are normally designed and manufactured in accordance to commercial diesel as fuel. The suitability of alternate fuels is yet to be studied whether these fuels give desired performance and emission characteristics. The major reasons for using small diesel engine for the study are decentralized energy generation in rural areas; on site oil production and improvement in socio-economic conditions of the farmers.

It is reported that straight vegetable oil gives somewhat acceptable results when used in the existing engines having indirect fuel injection. The direct injection engine does not perform efficiently with straight vegetable oil because of poor atomization characteristics. The available recent literatures stated the limitations of straight vegetable oil as alternate fuel in the existing engines. The performance and emission parameters of existing IDI engine operated at recommended design operating parameters are better than DI engine because of presence of pre combustion chamber. Pre combustion chamber enables the engine to handle even low grade fuel. The other technical problems reported are lower BTE and emissions. Injector chocking, high deposits, lubricating oil dilution and piston ring sticking under long term use were also reported. In order to overcome the technical problems, various engine modifications were recommended by the researchers to use SVO as alternate fuel. Some of the recommended modifications are change in combustion chamber design, change in design of piston crown, design change for the location of inlet and exhaust port, incorporating the fuel preheating system, changes in fuel injection point and change in fuel injection pressure etc. The comprehensive literature survey related to SVO and engine modifications were analyzed and a low speed indirect injection (IDI) diesel

engine was selected for the present investigation. The engine was tested with Jatropha Straight Vegetable Oil (JSVO) as fuel. The emphasis was made to study the technical feasibility of JSVO for existing diesel engine after engine hardware modification such as fuel injection timing, fuel injection pressure and fuel preheating etc.

The objectives of investigation were elevating the temperature of fuel before the fuel is injected into combustion chamber, advancing the fuel injection point and increasing the fuel injection pressure.

Initially the physico-chemical properties such as viscosity, density, calorific value, surface tension of diesel and JSVO were found out by the prescribed methods with the help of well calibrated precision equipments. The comparison of physico-chemical properties of JSVO with diesel was carried out. The viscosity which is an important property, was measured by Alpha series rotational viscometer manufactured by Fungi Lab. This instrument is capable of finding out absolute as well as relative viscosity. The rotational viscometer has accuracy $\pm$ 0.01 cSt. The densities of diesel and JSVO were measured by hydrometer. The hydrometer has an accuracy of $\pm$ 0.001. The surface tension of diesel and JSVO were also measured by multi frequency ultrasonic interferometer. The principle involved in measurement of surface tension is based upon the accurate determination of wave length in the medium. The surface tension was calculated from wavelength by using empirical formula. The cetane index of diesel and JSVO were also found out in the laboratory with the help of aniline apparatus. The major objective of this work for assessing the effects of these properties on the engine efficiency. The effect of these properties on the performance and emission were analyzed.

The engine selected for the experiment was indirect injection (IDI), low speed diesel engine having rated output 7.35 kW with diesel fuel. This engine is single cylinder, four stroke, vertical, water cooled system having a bore of 120 mm and stroke of 139.7 mm. The test engine was directly coupled to a 230 V single phase AC generator of 7.5 kVA capacity to absorb maximum power produced by the engine. The hydraulic loading arrangement was used for applying load on the engine. Brake power on the engine was monitored with the help of voltmeter and ammeter. Thermocouples were installed at heat exchanger inlet and out let for taking the temperature of exhaust gas and cooling water temperature at the engine for recording the cold and hot water temperature. Lubricating oil temperature and air inlet temperature were also recorded for future analysis. Two types of thermocouples were used; one was J-type having range of 500^0 C for exhaust gas temperatures and other was PT- type having range 100^0 C for the temperature at remaining locations from where the temperatures were observed. The digital tachometer was used for measuring the engine RPM. The burette installed in fuel supply system was used for finding out the volumetric rate of fuel consumption. The emission parameters of exhaust gas were found out with the help of AVL make smoke meter and exhaust gas analyzer.

The repeatability and reproducibility of data was ascertained by operating the engine five times at fixed operating parameter and monitoring the observations. All the sets of operating conditions as decided for experimentation were grouped in a sequence so that the observations could be made at all the operating points in a gradual ascending or descending order. This helped in the evaluation process of engine performance and it can be viewed easily by seeing the trends of data variation in charts.

The modifications in two important technical design parameters considered, were fuel injection pressure and the advance angle of fuel injection. The engine was designed to operate at 175 kgf/cm^2 fuel injection pressure and fuel injection point 20^0 before top dead centre (BTDC). In this investigation, the engine was operated at fuel injection pressures 175 kgf/cm^2, 185 kgf/cm^2, 195 kgf/cm^2 , 205 kgf/cm^2 and fuel injection angles 20^0 BTDC,22^0 BTDC,24^0 BTDC and 26^0 BTDC. The above sets of operating parameters were used in all the three phases of experiment are existing engine testing with commercial diesel, JSVO and modified (engine installed with heat exchanger to pre heat the JSVO) engine with JSVO.

Initially the existing engine was operated with diesel as test fuel and afterwards the same engine was operated by using the JSVO fuel. In the third phase of experiment the heat exchanger was designed, fabricated by taking the technical parameters such as exhaust gas temperature, volume flow rate, heat transfer rate required by the fuel, specific heat of JSVO and exhaust gas etc. The heat exchanger was used as a modification for pre heating the JSVO to 90^0 C. The heat exchanger was installed between fuel pump and the fuel injector for preheating the fuel after considering the feasibility of tapping the exhaust gas and the quantity of heat available in it. The engine was run with JSVO for entire test cycle. During this experiment the temperature of exhaust gas was observed at downstream of heat exchanger.

Subsequently, the IDI diesel engine was operated with diesel fuel at different load conditions such as no load (0% load), 1.84 kW (25% load), 3.68kW (50% load), 5.52 kW (75% load) and 7.35 kW (100% load). The fuel injection angles were varied from 20^0 BTDC to 26^0 BTDC in the steps of 2^0 while keeping the fuel injection pressure constant at 175 kgf/cm^2. The technical parameters such as the brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), emission parameters (CO, CO_2, HC, NO_x and smoke) were recorded. The above process was also repeated for injection pressures 185kgf/cm^2, 195 kgf /cm^2 and 205 kgf/cm^2 subsequently. After completion of the test cycle with diesel fuel, the engine was evaluated with the JSVO and operated under similar test environment for next cycle.

The kinematic viscosity of diesel was obtained to be 4.3 cSt whereas the kinematic viscosity of JSVO was found 48.7 cSt at 30^0. The density of diesel and JSVO were found to be 817 kg/m^3 and 910 kg/m^3 respectively. The calorific values of these fuels were found to be 39, 000 kJ/kg for JSVO and 42,000 kJ/kg for diesel. Since the density of JSVO is higher, therefore even though the calorific value of JSVO in kJ/kg is lesser than the diesel yet energy contained in 1 liter JSVO which is 35490 kJ/liter is higher in comparison with energy contained in one liter which is 34314 kJ/liter. The high density of JSVO has another advantage that it assists in quick scavenging process of burnt gases from combustion chamber.

The surface tension of diesel was found 0.028 N/m and for JSVO it was 0.042 N/m. The surface tension of heated JSVO at 90^0 C was found to be 0.32 N/m. The effect of these properties on performance and emission was analyzed. The cetane index of JSVO was 38 in comparison to cetane index 46 for diesel. The possibilities of improvement in fuel properties which can improve the performance were blending the JSVO with low viscous fuel, preheating the JSVO, using the additives etc.

It was observed that the brake thermal efficiency, brake specific fuel consumption and also the emission parameters were not favorable in the case of JSVO. The combustion quality of JSVO was the major factor for poor performance. The higher viscosity of JSVO (about 10-12 times the viscosity of diesel at same temperature) was identified to be the prime factor for poor combustion quality.

Preheating the fuel would reduce the viscosity and thereby improve the fuel atomization quality. This is because of the fuel particle size in combustion chamber would be smaller at low viscosity. There are many methods of preheating the fuel but in the present work, the exhaust gas was used as heat source. The literature review supports the argument of viscosity effects on combustion quality, these effects can also be minimized by providing more time for fuel atomization process before the combustion is established. This can be achieved by advancing the fuel injection angle. As recommended by the engine manufacturer, the fuel injection angle was 20^0 BTDC and was increased by adjustment on fuel pump. As mentioned above the objectives of establishing efficient fuel atomization phenomena in combustion chamber the fuel injection pressure can also be elevated. The engine was tested for injection pressure 175 kgf/cm^2 (rated injection pressure for this engine with diesel fuel), 185kgf/cm^2, 195kgf/cm^2 and 205 kgf/cm^2. The adjustment of fuel injection pressure was done on the fuel injector with the help of injector tester.

The effects of incorporating the heat exchanger on engine performance were analyzed. The logical conclusions based upon analysis of observed data were compared with the information available from existing literatures reported by the researchers. Some other important factors governing the fuel performance, efficiency and engine output were also been considered. In the present study, efforts were made to reduce the viscosity by pre-heating the vegetable oils, prior to injection in pre-combustion chamber of the engine.

To achieve the objective of preheating the JSVO to about 90^0C a counter flow heat exchanger was designed to use the exhaust gas as heat source. The heat exchanger was designed for installation immediately after the fuel pump and before the injector. The temperature of JSVO was observed 90^0C successfully when the engine was running at 7.35 kW load. However the exhaust gas temperature at inlet to heat exchanger was decreased at part load engine running. Therefore, a provision was made to alter the mass flow rate in heat exchanger by providing the flow regulator at downstream to heat exchanger.

After the fuel was charged in pre combustion chamber at appropriate injection pressure, the fuel particles were atomized and initial combustion process took place in pre combustion chamber. The entire process involves ignition delay period, rapid or uncontrolled combustion (pre mixed flame) and controlled combustion. During combustion phenomena the pressure is so high that the fuel droplets injected during the last stage burns almost as they enter. Any further pressure rise can be controlled by only mechanical means such as by the injection rate and after burning stage. Theoretically it is expected that the combustion process ends at third stage however due to poor distribution of fuel particle, combustion continues during the part of remainder of the expansion stroke.

In this investigation the fuel injection pressure which was second most important parameter was selected for the study. The theoretical analysis of empirical relation for Sauter Mean Diameter (SMD) was carried out to study the effect of increase in fuel injection pressure on fuel particle size. The empirical relation for SMD was programmed in MAT LAB version February 2011 and the results were analyzed.

It was observed that the diameter decreased when the fuel injection pressure increased. The fuel particle size of diesel fuel at175 kgf/cm^2 was compared with fuel particle size of JSVO at195 kgf/cm^2. It was found that the SMD of JSVO was reduced and remained higher than SMD of diesel.

The BTE of existing engine at rated engine load with diesel fuel was obtained 32.96% at fuel injection point 20^0 BTDC and fuel injection pressure 175 kgf/ cm^2. The efficiency was decreased while the injection pressure was increased from175 kgf/cm^2 or fuel injection angle was advanced from 20^0 BTDC. The BTE of existing engine with JSVO fuel at 175 kgf/cm^2 injection pressure and 20^0 BTDC was 30.62%. The BTE was improved from 30.62 % to 31.04% after preheating. By increasing the injection pressure from175 kgf/cm^2 to195 kgf/cm^2 , BTE was improved from 30.62% to 31.45%. Advancing the angle from 20^0 BTDC to 24^0 BTDC, the BTE was improved from 30.62% to 31.04%. The BTE was reduced when the fuel injection point was advanced beyond 24^0 BTDC and the injection pressure was increased beyond195 kgf/cm^2. Therefore, the fuel was required to be preheated while keeping the injection pressure at 195kgf/cm^2 and the fuel injection point at be 24^0 BTDC for maximum BTE..

The BSFC of unmodified engine with JSVO was 0.301 kg/kWh and was reduced to 0.266 kg/kWh after preheating the engine, advancing the fuel injection point to 24^0 BTDC and increasing the fuel injection pressure to 195 kgf/cm^2. The BSFC of unmodified engine with diesel fuel was 0.260 kg/kWh.

It was observed that the emission (CO, CO2, NO_x, HC and smoke) parameters improved as the fuel injection pressure was increased from 175kgf/cm^2 to 195kgf/cm^2 but it started deteriorating on further increase to205 kgf/cm^2. This trend was observed by unmodified and modified engine at all the operating conditions. The CO emission of existing engine with diesel and JSVO operated at 20^0 BTDC fuel injection point, 175 kgf/cm^2 injection pressure and 7.35 kW engine load was 0.05% Vol. and 0.10 % Vol. respectively. The CO emission of engine was reduced from 0.10 % Vol. to 0.045% Vol. with JSVO fuel by implementing the fuel preheating using exhaust gas, increasing the advance angle of fuel injection to 24^0 BTDC and fuel injection pressure to195 kgf/cm^2. The CO emission was reduced to level even below the CO emission with diesel fuel. The CO_2 emission of unmodified engine operated with diesel and JSVO at 20^0 BTDC and 175kgf/cm^2 was observed 1.24 % Vol. and 0.75 % Vol. respectively. The CO_2 emission of modified engine at 24^0 BTDC and 195 kgf/cm^2 was found 1.24 % Vol. Therefore, the CO_2 emission after implementing the modifications was increased substantially and it was improved to level of CO_2 emission for diesel fuel. The NO_x emission of unmodified engine with diesel and JSVO at 20^0 BTDC and 175 kgf/cm^{2}was found 360 ppm and 430 ppm respectively. The NOx emission of modified engine with JSVO was reduced by implementing modification to 330 ppm. The NOx emission of modified engine with JSVO was found better than the diesel fuel. The HC and smoke emission were also found to improve by 23.25 % and 41.30 % respectively.

It was concluded that the engine performance with JSVO fuel was improved after implementing the fuel preheating using exhaust gas, advancing the fuel injection point and increasing the fuel injection pressure. The efficiency of engine with JSVO was improved from 31.04% to 34.76% , which was 1.8% higher than the efficiency of unmodified engine with diesel fuel. The BSFC was reduced after incorporating the engine modifications from 0.301 kg/kWh to o.266 kg/kWh which was almost equal to the BSFC (0.260 kg/kWh) of unmodified engine with diesel fuel. The emission characteristics were appreciably improved for all the exhaust gas constituents.

List of Symbols

%	per cent
α	thermal diffusivity of fuel
β	Volumetric thermal expansion
v	Kinematic viscosity
μ	Dynamic Viscosity
ρ	Density
σ	surface tension
°C	degree Celsius
Cal	calorie
cc	cubic centimeter
cm	centimeter
cSt	centistokes
f	frequency
g	grams
h	hour
j	joule
kCal	kilocalorie
kg	kilogram
kW	kilowatt
L	Stroke length
m	meter
min	minutes
MJ	mega joule
ml	milliliters

N	Newton
Nu	Nusselt Number
rpm	revolutions per minute
Ra	Rayleigh number
Re	Raynold’s Number
s	seconds
U	overall heat transfer coefficient
V	volume of cylinder
W	watt

List of Abbreviations

A/F	air fuel ratio
BTDC	before top dead centre
BDC	bottom dead centre
B T E	brake thermal efficiency
BSFC	brake specific fuel consumption
BHP	brake horse power
CI	compression ignition
CN	cetane number
CO	carbon monoxide
EGR	exhaust gas recirculation
EFI	electronic fuel injection
FID	flame ionization detector
HP	horse power
HC	hydrocarbon; various hydrocarbons
IDI	in- direct injection
IVOP	inlet valve opening pressure
JSVO	Jatropha straight vegetable oil
LMTD	log mean temperature difference
LPG	liquid petroleum gas
MSME	micro small & medium enterprise
NO_x	nitrogen oxides
OAE	own account enterprise
ON	octane number
PAN	proxy acetyle nitrate
ppm	parts per millions
RPM	revolution per minute

SMD	sauter mean diameter
TDC	top dead centre
TSP	total suspended particle
WVO	waste vegetable oil

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CHAPTER 1 INTRODUCTION

The diminishing reserves of fossil fuels coupled with spiraling prices and higher environmental degradation has led to the interest in development of environment friendly alternate fuel and alternate sources of energy. One of the sources is straight vegetable oil. The idea of using straight vegetable oil as alternate fuel is as old as 1912. Mr Rudolph Diesel in his 1912 patent mentioned that "use of vegetable oil for engine fuel may seem insignificant today but such oil may become as important as petroleum in the course of time" [1]. The search of alternate energy sources become more important for the countries whose energy demand is met by foreign sources. The global concern about environmental pollution resulting from exhaust of internal combustion engines also attracted the attention of researchers to work in this area. The replacement of petroleum diesel by vegetable oil is one important potential initiative.

1.1 Prospects of Alternate Energy Sources

The world is confronted with the twin crises of fossil fuel depletion and environmental degradation. The indiscriminate extraction and consumption of fossil fuels have led to reduction in petroleum reserves. Alternative fuels, energy conservation and management, energy efficiency and environmental protection have become important in recent years. The increasing import bill has necessitated the search for liquid fuels as an alternative to diesel, which is being used in large quantities in transport, agriculture, industrial, commercial and domestic sectors.

There are so many sources of vegetable oil e.g. forest, vegetable oil crop and oil bearing biomass material. An important concern about it is that many of the straight vegetable oils are edible oils, if these oils are used for this purpose then the cost of edible oil and its direct effect on human food materials will be very adverse and not acceptable. Thus there is need to focus research and development on utilizing non-edible seeds from oil bearing trees. Some of the sources of non edible straight vegetable oils are Linseed oil, Mahua, Rice bran oil, Jatropha oil, Karanja oil etc. The research work carried out at various research institutions over the last decade have resulted in standardization of production technologies for production of bio-diesel from oil seeds from Jatropha, Karanja etc. The research in the area of using straight vegetable oil directly in the IC engine is also a prominent field of research. The properties of straight vegetable oil carry an important role for engine performance and emission characteristics.

In present scenario the straight vegetable oil is processed to produce biodiesel and then it is used in engines. There are different methods of generating the biodiesel a replacement of petro diesel by chemical processes at industry. Once the Straight vegetable oil is processed at the industry level for making biodiesels, it will be sold in the market at commercial prices. The farmers, who are the potential target in this study are using the IDI engine for their agricultural applications, power generations, pumping the water etc. will have to depend upon market once again and pay the cost of biodiesel fixed by industries. Apart from the cost and affordability, the accessibility is another problem especially for remotely established places. The utilization of straight vegetable oils has not been found favorable due to various technical limitations imposed primarily because of its high viscosity. However, straight vegetable oils could be advantageous in certain specific application areas such as remote village electrification and energizing pump-set for rural irrigation needs due to lower cost and simpler production technology.

In present era, the global objectives are to develop in the areas of resources, technology, automation and social values. These objectives can be fulfilled by ensuring energy availability not only from the available petroleum reserves but also by considering new sources of alternative fuels. These new sources of energy will give us liberty to use them as per the need of development for long term.

1.2 Energy requirement in remote areas and agriculture

As the economy of the countries are growing at rapid pace and also the technological developments have taken place very rapidly during near past, the energy requirements of human beings not only in cities but also in rural areas have hiked. In rural areas the energy requirements to meet the livelihood and agricultural needs have become more critical.

One needs to differentiate between the energy security of rural and urban areas because energy dynamics of both the areas are quite different. Energy security perhaps is more important for the rural people because they are very vulnerable, marginalized and lack access to most of the basic resources. Majority of rural households in developing countries like India depend on traditional fuels like firewood to meet most of their energy requirements, supplemented by small amounts of kerosene, coal and electricity for lighting. As per the statistics available in literatures:

I. About 70% of the Indians live in rural areas and use animal dung, agricultural waste and fire wood as fuel for cooking,

II. Particulate matter in the Indian rural households is 2000 $\mu g/\ m^3$ which is much higher than the permissible 150 $\mu g/\ m^3$,

III. Use of traditional fuel is estimated to cause around 400,000 premature annual deaths due to various respiratory problems.

IV. 75% of rural households depend on firewood for cooking, 9% each on dung-cake and on LPG as against 22% of urban households using firewood for cooking, another 10% on kerosene and about 57% on LPG.

V. 44% of rural households depend on kerosene and another 55% on electricity, while in urban areas dependency is 89% on electricity and 10% on Kerosene for domestic lighting.

'Rural' is usually equated with agriculture and rural energy with cooking and lighting; which certainly misses out the energy requirements of various other rural facets like rural schools and rural enterprises etc. Most neglected policy dimension has been energy for manufacturing in rural areas. Some of the statistical data about rural areas are listed below. These data explains the energy requirements in rural areas for rural development.

a. About 87% of the schools in the country are located in the rural areas.
b. About 25.81 million or (61.3%) of the enterprises in India are located in the rural areas (Economic Census 2005).
c. Among Micro, Small and Medium Enterprises (MSME) alone, around 44.52% of the registered units and around 54.68% unregistered units are in rural areas.
d. Besides, there are thousands of rural artisans who operate as Own Account Enterprises (OAEs) e.g. weavers whose productivity could be enhanced by supply of clean energy on a regular basis.

'Agriculture' is always considered to be synonymous with 'rural', especially in countries like India. Agriculture consumes power both in the animate and non-animate forms. During the first five year plan power consumption in Indian agriculture was 316 GWh (3.97% of total power consumption in India). In 2005 total power consumption in agriculture stood at 88,555 GWh, which is approximately 22.93% of total energy consumption in India. Gujarat (46%), Andhra Pradesh (43%) and Haryana (45%) are the leading states regarding electricity power consumption in Agriculture (TERI energy data directory & yearbook, 2005, P: 240).

Biomass based energy generation has a good potential for our country because of the rich biodiversity and a huge population. The potential sources of biomass energy are:

a. Non-edible oil producing plants/trees-biofuel/bioenergy
b. Wood biomass-lignocellulosic
c. Forest litter biomass
d. Bamboo energy
e. Household/domestic waste
f. Daily/weekly market waste
g. Wastes from rural based industries
h. Algal Energy

1.3 Vegetable oil as a potential substitute for diesel

Considering the above aspect, an initiative is taken through this study to make the farmers self dependent. If the petro diesel is replaced by vegetable oil directly which farmers can cultivate in their field. The Straight vegetable oil as alternative fuel is not enough but machines which can use those alternative fuels with optimum efficiency and economy are also required.

The replacement of petro diesel by vegetable oil will depend upon various parameters. The major parameters which make the straight vegetable oil as substitute are combustion quality, emission characteristics, and its physical availability, impact on engine parts i.e. short t and long term effects.

If the entire world production of 115 billion liters of vegetable oil had been used for fuel in 2007, neglecting the conversion losses as well as the debate on the use of food materials for fuel, this would satisfy about 75% of US diesel demand. The use of locally grown non edible plant oil as fuel in slow speed diesel engine has potential to provide a low cost sustainable energy solution.

1.4 Use of Straight vegetable oil in Diesel Engine

There are many applications of diesel engine in the rural areas such as electricity generation, running the machines like flour mills, oil expellers, rice mills, tube wells, pumping water etc. The farmers depend on petro-diesel available in the market at high prices. Sometimes, the farmers travel long distances for procuring the commercial diesel to meet the agricultural and domestic needs at typically established remote areas.

Exploring the possibilities of utilizing the vegetable oil as alternate fuel will make the farmers self-dependent. They can cultivate energy crops in their fields which may improve the socio-economic conditions. The literatures available in these areas suggest that the already available IDI engines with farmers or manufactures are not compatible with straight vegetable oil. Therefore, there is need to modify the existing engines which can perform efficiently with SVO.

CHAPTER 2 SURVEY OF LITERATURES

2.1 Introduction

Many plant oils have a similar energy density to fossil diesel but several properties of plant oils are considerably different from those of diesel. In Rudolph Diesel's preface to his 1912 patent he wrote that, "use of vegetable oil for engine fuel may seem insignificant today but such oil may become in the course of time, as important as petroleum"

The use of Straight vegetable oils as a fuel for compression ignition engines is restricted by some unfavorable properties e.g. high viscosity, low cetane rating, low calorific value and higher emission characteristics. The high viscosity of SVOs causes poor fuel atomization which leads to incomplete fuel combustion and carbon deposition on the injector and valve seat resulting in serious engine fouling. When the direct injection engines were run with neat vegetable oils, injectors got choked after a few hours. This choking also leads to poor fuel atomization and incomplete combustion. Due to incomplete combustion, partially burnt vegetable oil runs down the cylinder walls and dilutes the lubricating oil.

Despite the above mentioned limitation of SVOs, it could be possible to use them for certain low end applications, such as energizing single cylinder diesel engine which are widely used in rural areas. However, this would call for an additional fuel supply setup for starting and stopping of the engine has to be done on mineral diesel only to avoid deposition of neat oil on various engine parts, which would affect cold starting performance of the engine. The heat available in exhaust gas of the engine may be utilized to reduce the viscosity of intake oil through appropriate heat exchange device. The experiments conducted at various institutes have concluded that engines running on neat SVOs with the integration of above mentioned additional sub-systems could perform effectively for around 250 hours.

An exhaustive analysis of references in terms of technical parameters of results obtained focus on the need and scope of modifying the point of fuel injection, fuel injection timing, compression ratio, fuel injection pressure and also some of the mechanical design of engine. These modifications on diesel engines can overcome the above listed limitations and make the SVO suitable fuel while giving good performance. If the entire world production of 115 billion liters of vegetable oil had been used for fuel in 2007, neglecting the conversion losses as well as the debate on the use of food materials for fuel, this would satisfy about ¾ of US diesel demand. The use of locally grown non edible plant oil as fuel in slow speed diesel engine has potential to provide a low cost sustainable energy solution. The vegetable oil combustion studies have found that the SVO to be used as fuel for diesel engine is not suitable for DI engines whereas it gives good results in IDI Engine.

It has been reported that compared to commercial diesel fuel, all the vegetable oils are much more viscous, much more reactive to more oxygen and have higher cloud and pour points. The viscosities of vegetable oils were found to range from 10 to 20 times greater than diesel fuel. Increased carbon chain length and reduced number of double bonds were associated with increased oil viscosity, cetane rating and gross heat content. It was found that except for castor oil, there was little difference between gross heat content of any of the vegetable oils. Heat contents of SVOs were approximately 88% of that of diesel. The high viscosity of SVO is basic cause of poor combustion and atomization characteristics. The viscosity of Jatropha straight vegetable oil is about 10-12 times the viscosity of diesel. The viscosities of some of the important SVOs and diesel are shown in table 2.1 [1]. It has been reported that high viscosity and low volatility of vegetable oils are generally considered to be the major drawbacks for their utilization as a fuel in diesel engines. The high viscosity of vegetable oil cause problems in the injection process, leading to an increase in smoke levels, and the low volatility of the vegetable oils result in oil sticking of the injector or cylinder walls, causing the deposit formation which interferes with the combustion process.

Table 2.1: Kinematic viscosity of vegetable oils at different temperature [1]

SVO Type	Temperature (^{0} C)	Kinematic Viscosity (c St)
Soybean	25-160	33
Rapeseed	25-200	37-42
Peanut	25-200	40
Palm	25-145	39
Jatropha	15-145	34-37
Diesel	25-125	2.6-3.6

The performance of a direct injection, 3-cylinder, 2600 series Ford tractor engine with 1:3 (v/v) blends of soybean oil and sunflower oil with diesel fuel for 200 hour has been evaluated. After the 200 hours test it was concluded that as far as power output, thermal efficiency and lubricating oil data were concerned, the 1:3(v/v) blends of soybean oil and sunflower oil with diesel fuel performed satisfactory. However, when the general condition of the combustion chamber and fuel injectors after 200 hour of operation were considered, the performance was not satisfactory. All combustion chamber parts and injector tips were coated with carbon deposits. This suggested that different operating conditions or modification of vegetable oils could help in improving the conditions of the engine. There are different methods to reduce the viscosity of fuel such as preheating, blending it with some other low viscous fuel, using additives. In this work the heating approach is utilized to reduce the viscosity. The curve shown in figure 2.1 indicates that the viscosity of JSVO matches with base line fuel as diesel when the JSVO is heated to 90^0 C. Because of high viscosity the injector choking is one problem along with some others e.g. ceaseing of fuel injector, gum formation and piston sticking on long term. [2]

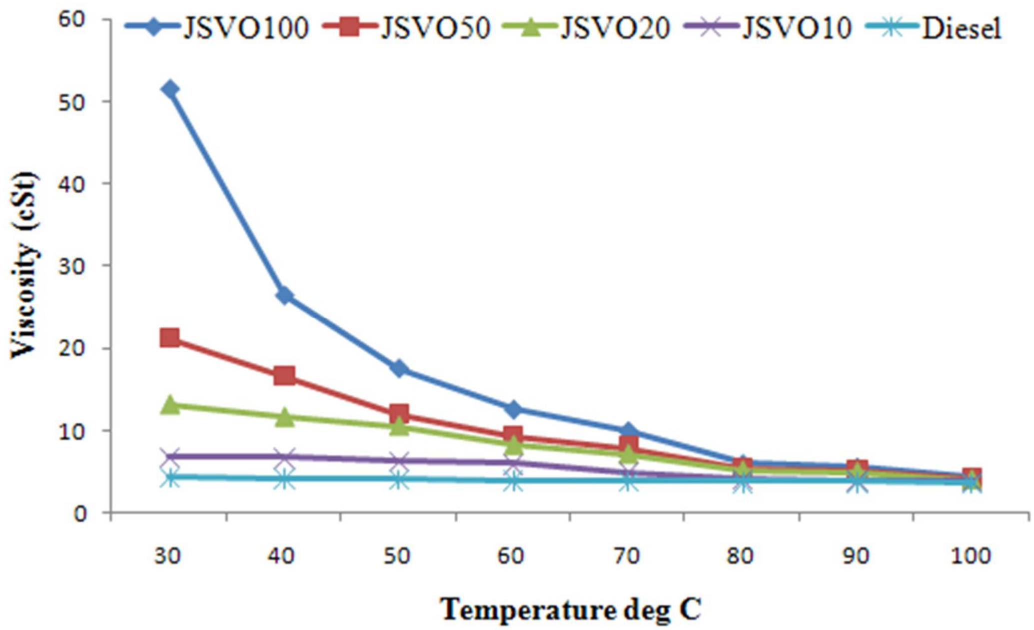


Figure 2.1: Viscosity variation of different fuels with temperature

Mustafa Canacsi [3] has tested the SVO for combustion analysis in IDI diesel engine and reported that high viscosity and density of vegetable oil led to problem in the injection system and combustion chamber of the engine for a long term usage. He also commented that the high viscosity problem has been solved in several ways such as preheating the oil or blending it with petroleum base diesel fuel. He experimented that excessive carbon deposit takes place in combustion chamber when Soybean (SVO) is used as fuel. This happened because of high viscosity. The Jatropha has very good corrosion resistant quality since it has almost nil sulphur content. Deepak Agrawal [4] has tested the SVO in CI engine and reported that the long term use of SVO in engine show high carbon deposit and lubricating oil dilution. He also observed that the SVO has poor atomization characteristics because of low cetane number. Panwar N L [5] investigated the performance of caster seed oil in diesel engine and reported that the specific fuel consumption was higher for caster seed oil fuel and high NO_x emission was also observed.

One hundred percent vegetable oil can be used safely in an indirect injection engine but not in direct engine due to high degree of atomization required for this type. He also commented that, the use of SVO in diesel engine present problems primarily due to their high viscosity and low volatility. The long term tests exhibits symptoms of injector cocking, cylinder deposits and ring sticking. Chocking of injector leads to poor fuel atomization and dilution of lubricating oil by unburnt vegetable oil which runs down the cylinder wall. The testing of SVO in IDI engine showed that the fuel heating increased peak cylinder pressure and has no significant effect on the performance criteria at high engine speed of 3000 rpm. It is also reported that the unheated vegetable oil fuel produced the highest mechanical efficiency throughout the load range. This fact contradicts the results reported by majority of researchers. This may be due to none efficient and calibrated instruments used for data collection or improper result compilation. The friction power was noted to increase by the same amount as the viscosity of the fuel decreases. The effect of decrease in viscosity on hydrocarbon emission was reported to increase [6].

Machacon [7] investigated that besides the viscosity effect the surface tension also affects the spray particle size. Higher the surface tension, higher the droplet size. The droplet size of fuel in combustion chamber directly influences the fuel atomization efficiency.

Habbal OD [8] reported that the viscosity of Deccan Hemp oil (non edible SVO) on heating to 95^0 C became almost equal to the viscosity of petro diesel at 30^0 C. The Habbal OD also reviewed the comparison of Deccan Hemp oil performance with Jatropha. He states that that the brake thermal efficiency of engine fueled with Deccan Hemp oil is higher than the Jatropha for entire range of operation. BSFC of Deccan Hemp oil is lower than the Jatropha.

2.2 Cetane Rating

The cetane rating of a diesel fuel is a measure of its ability to auto ignite quickly when it is injected into the compressed and heated air in the engine. It is related to volatility characteristics of fuel. Cetane number is a parameter by which the ignition delay is determined. Cetane number of a fuel is percentage by volume of cetane ($C_{16}H_{34}$) in a mixture of cetane and α-methyl naphthalene ($C_{10}H_7CH_3$) that has same performance in the standard test engine as that of the fuel. Octane and cetane are inverse measurement of the same property. A simple, though not rigorous, relationship between cetane number and octane number is:

$$CN = (104 - O.N.) / 2.75 \qquad \ldots (2.1)$$

A good diesel engine fuel is a bad gasoline engine fuel. Diesel fuel is having cetane 40-60 while high octane fuel like gasoline is having cetane number 10-20. Cetane number is the most important single fuel property which affects the exhaust emission, noise and start ability of diesel engine. In general lower the cetane number, higher is the hydrocarbon emission and noise level,

Basinger M [1] states that CN is a measure of a fuels ignition delay quantity. A higher cetane number corresponds to shorter ignition delay. Long ignition delay is undesirable due to consequence of engine knock. He suggested that some negative aspect of SVO's ignition quality can be lessened or avoided by tuning the injection timing. The cetane number of Jatropha oil is 38-45. The cetane number of petro diesel and Jatropha is comparable. However the increase in cetane number will subsequently enhance the combustion quality. A cetane number 40-60 is desirable. The cetane number of diesel is 45-56 where as the cetane number of Jatropha is 38-45.

Haldar SK [2] reported that the cetane number of Jatropha is 33.7 where as cetane number of diesel is 46.3. Because of the low cetane number and higher viscosity of the non edible vegetable oil in comparison to diesel, several difficulties in IC engine such as engine choking, cease of fuel injector, gum formation and piston sticking under long term use are noted.

Mustafa canakci [3] compared the cetane number 36.7, of preheated crude sunflower oil with cetane number of petro diesel as 46. He pointed out that low cetane number of crude sunflower oil is reluctant in fuel atomization and therefore require more time for atomization process which can be fulfilled by advancing the fuel injection point.

Deepak Agrawal [4] commented on cetane number that it is comparable to mineral diesel. He also reported that free fatty acids are also found in vegetable oils. The large molecular sizes of the triglycerides results in the oils having higher viscosity and low volatility compared to mineral diesel oil. The propane and location of double bond in the structure of typical SVO as mentioned in figure 2.2 affects the cetane number of vegetable oil.

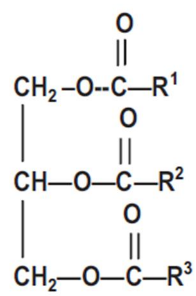


Fig. 2.2 Structure of SVO

Deepak Agrawal also stated that the vegetable oils have comparable heat content, cetane number, heat of vaporization, stoichiometric air / fuel ratio with mineral diesel. He comments that the proportion and location of double bonds affects cetane number of vegetable oil.

2.3 Fuel Injection Advance Angle

The first stage of combustion in the CI engine, i.e. the delay period, exerts a very great influence on both engine design and performance and therefore needs a detailed study. The ignition delay can be roughly divided into two parts (1) physical delay, which is time between the beginning of injection and the attainment of chemical reaction condition and (2) chemical delay, in which pre flame reaction starts slowly and then accelerate until local inflammation or ignition takes place. Generally chemical delay is longer than physical delay. However it depends upon temperature, at higher temperature chemical reaction is quicker and physical delay is longer than chemical delay. The delay period refers to the sum of physical delay and chemical delay. In most of the CI engine the ignition delay is shorter than the duration of injection. The longer the delay period the more rapid and higher is the pressure rise, since more fuel will be present in the cylinder before rate of burning comes into the control.

Ignition delay depends upon fuel quality, injection pressure, injection advance angle, compression ratio, intake temperature, jacket water temperature, supercharging, engine speed, air fuel ratio, engine size and type of combustion chamber. Out of these many parameters fuel injection pressure, size of droplets and fuel injection advance angle is taken for investigation and modification. Basinger M. [1] reported that the increase in fuel injection advance angle increases the smoke opacity, but the increase in opacity is almost negligible up to about 25^0 BTDC fuel injection advance angles where as it becomes significantly high beyond 25^0 BTDC. He also reviewed the observation of Haldar S K [2] that the engine performance gets enhanced by advancing the fuel injection timing; they attribute this to lower cetane number of SVO. He also commented on the timing effect on WVO in a DI engine, stating that the advanced timing improved the efficiency and reduced CO emission, though it elevated NO_x emission. Nwafor also found benefits by advancing the timing of a rapeseed SVO fueled IDI engine. He reported that the engine ran smoother and both CO and CO2 emission improved. This was an attribute to longer ignition delay and slower burning rate of plant oil. However the delay period was also found to be influenced by engine load, speed and temperature. Similar to IVOP findings, advancing the timing too far can have negative consequences resulting in erratic engine behavior.

Haldar SK [2] has observed that the performance data at various timings 45^0,40^0 and 35^0 BTDC in fig 2.3 , it is concluded that the oil of Karanja, Jatropha and Putranjiva gives more yield at 45^0 BTDC timing than diesel and at 40^0 BTDC timing diesel gives more. It may be due to the different cetane number and ignition temperature of the non edible oils compared with diesel. Cetane number of straight vegetable oil is less than diesel which demands ignition timing to be more for better combustion and hence improves the engine performance and emission.

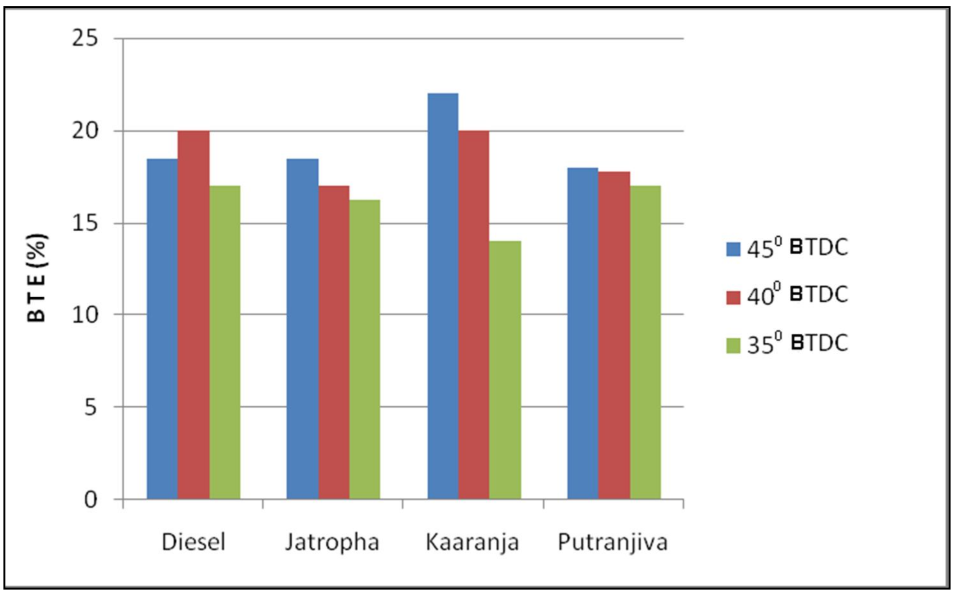


Figure 2.3 Performance data at various timings 45^0,40^0 and 35^0 BTDC

Mustafa Canakci [3] investigated that the cylinder gas pressure depends upon the combustion rate in the premixed combustion phase. This phase is controlled by the ignition delay period and spray envelop of the injected fuel. Therefore, the viscosity and volatility of the fuel have very important role to increase atomization rate and to improve air fuel mixing. Because of high viscosity and low volatility of PCSO (Preheated Crude Sunflower Oil), a slight decrease in the maximum cylinder gas pressure was obtained at all engine speeds [6].

Herchel T C [7] studied the performance and emission characteristics of a diesel engine fueled with coconut oil-diesel fuel blend and reported that there is slight shortening of the ignition delay for the case of pure coconut oil as compared with pure diesel fuel, which may indicate the change in the combustion pattern with the use of coconut oil. The ignition delay is, of course, composed of the physical and chemical delay the physical delay depends on atomization, vaporization and fuel vapor mixing with air, while the chemical delay depends upon the pre combustion reactions of the fuel. Since the atomization of coconut fuel due to its higher viscosity and surface tension is poor, the physical delay would have been longer. Since the surface ignition temperature of coconut oil is higher than that of pure diesel fuel, therefore longer ignition delay was expected. In view of these, the trend of ignition & the trend of ignition delay is somewhat surprising and quite complex. This need to be further investigated. The brake thermal efficiency of engine fueled with deccan hemp oil is reported to be low and BSFC was found to be lower throughout the range of operation [8]. M basinger, Vijay Kumar [9] undertaken the endurance test of CI engine fueled with SVO. He reported that injector choking and piston sticking was observed under endurance test. This may be because of high viscosity leading to poor fuel combustion.

2.4 Fuel Injection Pressure

In CI engine the atomization of fuel within the very small time available has great impact on the quality of combustion process taking place in the combustion chamber. For efficient atomization process the size of fuel particle released by the injector at the time of injection process depends upon injection pressure. Smaller fuel particle is desirable for improved atomization quality and subsequently good combustion process taking place. Abolle [10] reported the results on the study of fuel particle size (Sauter Mean Diameter) and suggested following empirical equation. The equation shows that the fuel particle size will get reduced as the injection pressure is increased.

$$\text{SMD} = 3.08\, \nu^{0.335} (\sigma \rho_{\text{fuel}})^{0.737} \rho_{\text{air}}^{\ 0.06} (\Delta P)^{-0.54} \qquad \text{.....(2.2)}$$

Where v is the kinematic viscosity (m^2/s); σ is the surface tension (kg/ s); ρ_{fuel} is the fuel density (kg/m^3); ρ_{air} is the ambient air density (kg/m^3); ΔP is the injector opening pressure (Pa). In addition to fuel injection pressure other parameters such as kinematic viscosity, density of fuel; density of air; surface tension of fuel also affects the SMD [10].

Avinash Agarwal [11] studied the performance of CI engine with Karanja oil. He reported that loss of efficiency has occurred and rate of fuel combustion was increased. This may be due to inefficient combustion of Karanja oil due to high viscosity. The injector released bigger fuel particle size due to high viscosity. The atomization of fuel spray having bigger fuel particle is poor. Giannelos P N [12] tested tomato seed oil by using it directly in engine. He reported that the chemical composition of SVO is such that it contain more number of carbon atom and double bonds therefore it exhibit reluctance towards efficient combustion. The Winfried Russ [13] tested the food industry waste oil in diesel engine and mentioned that the oil has high cetane number and low calorific value and therefore its usability in IC engine is lean. The smaller fuel particle can be achieved by injecting the fuel at elevated pressure. The amount of increase in pressure is to be decided very carefully because very small fuel particle at very high injection pressure is also undesirable because very small fuel particle will have very small momentum so the fuel particle will not be in position to travel in combustion chamber for long distances creating partial suffocation by its own product of combustion. Secondly, as the pressure after ignition depends on the area of inflammation, smaller the size and greater the number of droplets the larger will be the aggregate area of inflammation and therefore the greater the uncontrolled pressure rises. The disadvantage of larger droplet is of course that subsequent rate of burning is too slow [14]. The surface tension is the fuel property that affects spray atomization, droplet size and other important properties of the diesel spray. The high viscosity leads to poorer atomization of fuel spray and less accurate operation of the fuel injectors [15]. Bhupinder Singh Chouhan

[16] determined the important physico-chemical properties of Jatropha oil. . He reported that diesel fuel can contain both saturated and unsaturated hydrocarbons, but latter is not present in large amounts, making oxidation a problem. Vegetable oil has different chemical structure, where R^1, R^2 and R^3 are alkyle groups of different carbon chain length (varying from12 to 18), and-COO- is a carboxyl group (figure 2.2). Large size of vegetable oil molecule and the presence of oxygen in the molecule suggest that some fuel properties of vegetable oil are different from diesel fuel. Forson FK [17] did experiment on direct injection single cylinder diesel engine with Jatropha oil and its fuel blends with diesel. He reported that the introduction of Jatropha oil into diesel fuel appears to be effective in reducing the exhaust gas temperatures since the Jatropha oil could be considered to be emulsified as water was introduced into the milled Jatropha seed during the extraction process. He also reported that the Jatropha oil can be used as an ignition accelerator additive for poor diesel.

Reddy J N [18] in his parametric study for improving the performance of a Jatropha oil fueled compression ignition engine reported that the injector timing has to be advanced as compared to base fuel diesel operation in order to compensate for the higher ignition delay. He reported that by advancing the injection timing by 3^0 crank angle (i) BTE increased from 25.7 % to 27.3% (ii) HC level was reduced from 2350 ppm to 2068 ppm (iii) smoke level was reduced from 3.9 BSU to 3.3 BSU. He also reported that increase in injection pressure from rated value of 205 bar to 220 bar resulted in significant improvement in performance and emissions. Purushothaman K [19] tested a single cylinder CI engine with orange oil and reported that orange opil exhibits a longer ignition delay and higher combustion duration compared to diesel. The heat release rate was also reported to be higher for orange oil than the diesel. The BTE at full load was found to be 31.7% with orange oil in comparison with BTE 29.3% with diesel fuel. Yusuf Ali [20] in his search for the alternative fuel to diesel reported that the SVO can not be used in CI engine directly. However the technology is available to permit these of modified vegetable oils as diesel fuel substitute.

Dorado MP [21] studied the emission characteristics of diesel engine using the waste olive oil and mentioned that significant reduction in CO (up to 58.9%) , CO2 (up to 8.6%), NO (up to 37.5%) and NO_x (up to 32%) in comparison with diesel fuel was observed. Herchel Thaddeeus C [22] investigated the effect of blend of coconut oil and diesel oil on diesel engine performance. He observed that the brake mean effective pressure was lower in the case of coconut oil in comparison with diesel fuel and there by BSFC was reduced. It was also observed that the NO_x and CO emission was also less with coconut fuel. Volkhard Scholz [23] did the experiment on castor oil fuelling in CI engine and reported unfavorable engine related technical properties of straight castor oil. He explained the possibility to produce a methyl or ethyle ester by transesterification, which can be possibly added to fossil diesel fuel in small proportions. However, castor oil will only play a minor role in future oil and fuel market.

The smoke opacity of SVO is higher than that of diesel fuel due to high viscosity of vegetable oil which results in incomplete combustion [26, 27]. The NO_x emissions of JSVO at all operating point were found lower than the diesel engine. This characteristic is very important as NO_x emission is most harmful gaseous emission [28, 29]. All the vehicles and combustion devices using hydrocarbons and their derivatives as fuels contribute to air pollution; the amount of emission largely depends upon their design, operating conditions and the characteristics of the fuel. The global vehicle pollution excluding motorcycles increased from 250 million in 1970 to 671 millions in the year 2000. The vehicle population is projected to grow closed to 1300 million by the year 2030. The largest growth is expected in South and East Asia where the vehicle numbering around 61 millions in the year 2000 would reach close to 350 million by 2030 [30].

It is reported that the effect of increase in fuel injection pressure results in improvement in smoke opacity. However he also reported that the increase in fuel injection advance angle increases the smoke opacity, but the increase in opacity is almost negligible at up to about 25^0 BTDC fuel injection advance angles where as it becomes significantly high beyond 25^0 BTDC. The smoke opacity of JSVO is higher than that from diesel fuel due to the high viscosity of vegetable oil which results in incomplete combustion. The smoke opacity increases with increase in engine loads. Higher smoke opacity may be due to poor atomization of SVO. Bulky fuel molecules, higher viscosity of vegetable oil and low volatility results in poor atomization of fuel. Heating vegetable oils result in lower opacity compared to unheated oil but it is still higher than mineral diesel. The smoke opacity percentage for vegetable oil is greater than the diesel fuel because of heavier molecule of hydrocarbons in JSVO which do not burn completely and there is larger formation of particulates. The smoke generation also increases with increase in engine load due to overall richer combustion , increase in overall fuel – air equivalence ratio, longer duration of diffusion combustion phase and reduced oxygen concentration. The smoke can be reduced by reducing period of diffusion combustion by improving fuel atomization. Advancing the injection timing increases combustion temperature and allows more time for oxidation of soot in the expansion stroke thereby reducing the smoke emission. The smoke meter is used for measuring the smoke, the smoke can also be measured by filtering a fixed volume of exhaust gases through filter paper and the stain thus formed is evaluated on a grayness scale by a reflection meter (Bosch Smokemeter). The particulate matter is mainly composed of soot and unburned hydrocarbon adsorbed on soot.

There are two types of pollutants, one is primary pollutants and the other is secondary pollutant. In first category the pollutants are CO, CO_2, HC, NO_x, SO_2, SO_3 where as in second category the pollutants are oxidants like ozone, nitrogen oxides, total suspended particles (TSP) including most of other organic compounds like proxy acetyle nitrate (PAN). The primary pollutants are also termed as vehicle emissions. The most of the vehicle emissions are taking place due to incomplete combustion inside the engine cylinder except for the nitrogen oxide and sulphur di- and tri- oxides which are the product of complete combustion. The amount of sulphur in the current engine fuels gasoline or diesel, is quite small (<500 ppm by mass) and would be lowered further below 50 ppm. Therefore the emission of sulphur di- and tri oxides from engines is not significant from pollution point of view. CO_2 is not pollutant for local environment but is being green house gas; its contribution to global warming is causing an increasing concern. It is estimated that CO2 is responsible for about 50% of the global green house effect. The carbon monoxide (CO) is primarily the result of deficiency of oxygen in the fuel air mixture that leads to incomplete combustion of the fuel. It is an odorless gas but is highly toxic. The CO has 200-240 times greater affinity than oxygen to combine hemoglobin. Thus exposure to CO reduces oxygen carrying capacity of the blood to body tissue. With decrease in air fuel ratio below the stoichiometric, formation of CO increases sharply. During the expansion stroke, piston travels downwards rapidly cooling the combustion products by expansion freezing the reactions that involves NO and CO formations. When combustion temperature is decreased the emission of nitrogen oxides are reduced but the smoke and particulate emissions are increased. If the fuel injection pressure is increased beyond a point, no further improvement in air –fuel mixing results. The retarded fuel injection improves the NO_x emission characteristics. It is also suggested that the significant reduction in NO_x and particulate takes place when 75% fuel is injected in the first pulse and the 25% in the second pulse after a dwell period of 10^0 crank angles [31].

Continuous exhaust sampling and a hot flame ionization detector (FID) with a heated line system were used to measure HC emissions. The HC emissions of the diesel fuel operation were significantly high compared to the test results on plant oil fuels. The unheated vegetable oil offered a net reduction in HC emission over the heated plant fuel. The trend was an increase in HC emission as the viscosity of the fuel decreases. These results indicate that the HC emission is influenced by the fuel viscosity.

The variation of un-burnt HC emissions for diesel and JSVO oil (unheated and preheated) is lower at partial engine load but increased at higher engine load. This is due to relatively less oxygen available for reaction when more fuel is injected into the engine cylinder at higher engine load. Due to large particle size, injector times and nozzle chocking also increased combustion timing. The effect of fuel adsorption in the oil is not significant for small air cooled utility-type engines. HC emissions of unheated JSVO are higher than that of diesel fuel but with heated JSVO this value decreases and at a temperature of about 100^0C, it comes lower than that of diesel. Value of un-burned HC emission from the diesel engine running at constant speed from no load to full load is higher in case of JSVO and lower for mineral diesel oil. HC emissions are lower at part load, but tend to increase at higher load for both the fuels. This is due to lack of oxygen resulting from engine operation at higher equivalence ratio.

Basinger M has opinioned that common modification in CI engine for this purpose is to increase injector valve opening pressure (IVOP). Increasing diesel engines IVOP has shown decrease in fuel spray droplets diameters and increase in velocity and penetration distance resulting in a boost of engine performance and emission improvements. The initial average fuel spray droplet diameter has been shown to be inversely related to its velocity squared. This means that as the velocity increases (from increased IVOP) droplet size rapidly decreases. The droplet evaporation can be described by the D^2 law, which relates the evaporation rate to the droplet diameter squared. Decreasing the droplet diameter can then significantly increase evaporation rate, thus enhancing the combustion efficiency. There is a point where the increasing the IVOP becomes counterproductive. This is due to increased spray penetration resulting in wall impingement. For this reason it can be valuable to tune a particular engine's IVOP for the specific SVO. It is reported that the sauter mean diameter (SMD) which is the ratio of mean volume to the mean surface area of the fuel droplet has important role in defining fuel atomization characteristics. Smaller SMD results better fuel atomization.

$$SMD = \Sigma n_i d_i^3 / \Sigma n_i d_i^2 \quad \dots (2.3)$$

Where n_i is the number of droplets with mean diameter d_i .

Table 2.2: Physico-chemical properties of diesel and Jatropha oil

Properties	**Diesel**	**JSVO**
Density (gm/cc at 30^0 C)	0.817	0.910
Kinematic Viscosity (c St at 30^0 C)	4.3	48.7
Calorific Value (kJ/kg)	42000	39000
Cetane Index	46	38
API gravity	31.7	22.7
Carbon residue (%,w/w)	0.1	0.64
Observation	Stable	Stable

2.5 Effect of Fuel Properties on Engine Performance

In the preparatory phase of experiment the important physico-chemical properties of Jatropha oil were determined by standard methods and compared with diesel. The observed values are shown in Table 2.2. The importance and effect of these properties was found out from literatures available from the present trend of investigation. The results show that the heating value of the jatropha vegetable oil (39,000 kJ /kg) is comparable with heating value of the diesel (42,000 kJ/kg). Since the heating value of JSVO is quite close to heating value of diesel, the engine output also can be expected almost same under similar environment of engine operation [32, 33].

Cetane index, 38 of JSVO is slightly lower than the cetane index 48 of diesel fuel. Cetane index is the indicative parameter of time delay required for fuel combustion. The fuel having smaller cetane index needs larger time delay, so the time delay as designed for this engine may not be sufficient. The increase in delay period will improve the combustion efficiency and ultimately engine output [34, 35].

The kinematic viscosity of Jatropha oil which is 10-12 times of kinematic viscosity of diesel carries prime consideration of the researchers for discussion as it affects the engine performance greatly because of poor fuel atomization quality due to high viscosity. The fuel atomization in combustion chamber is an essential phenomenon taking place before the fully established combustion process. The viscosity of diesel and the viscosity of JSVO were determined in the lab with the help of rotary digital viscometer to have preliminary investigation of JSVO combustion quality. The viscometer was capable of showing the kinematic viscosity of test fuel at constant as well varying temperatures. The result obtained indicated that the viscosity of diesel is almost constant on increasing temperature whereas the viscosity of JSVO was continuously increasing as the temperature was decreased. The variation of kinematic viscosity of Jatropha and diesel on varying temperature has been shown in figure 2.4.

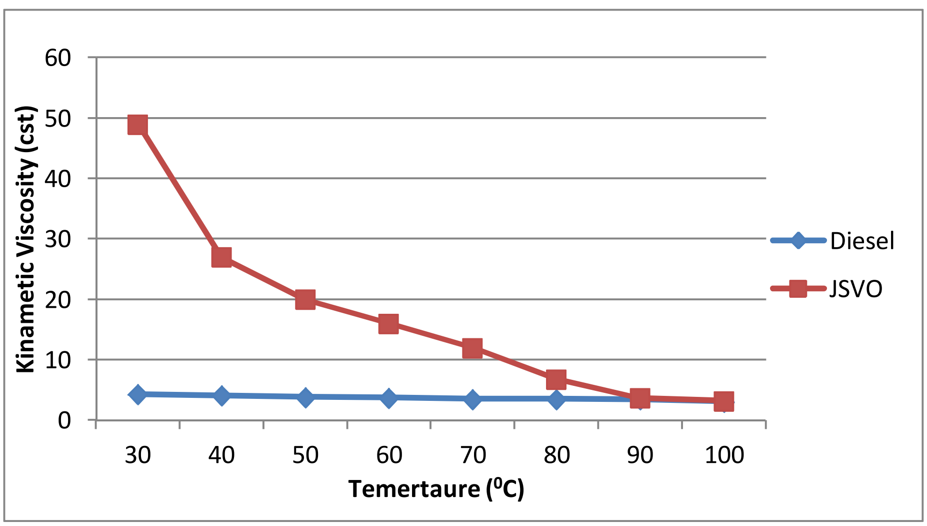


Figure 2.4: Variation of kinematic viscosity of diesel and JSVO with temperature

From the data obtained it is observed that the viscosity variation of diesel is almost negligible at all the temperatures, where as the viscosity of JSVO is varying largely from 48.7 cSt at 30^0 C to 3.68 cSt at 90^0 C. The value of kinematic viscosity of JSVO at 90^0 C is almost same that of diesel viscosity. The vegetable oil combustion studies have found that the SVO to be used as fuel for diesel engine has not been found suitable for DI engines whereas it gives comparatively good results in IDI Engine.

Wagner [36] investigated the engine performance by fuelling it with soybean oil and reported that compared to commercial diesel fuel, soybean oil is much more viscous, much more reactive to oxygen and had higher cloud and pour points. The viscosities of vegetable oils were found to range from 10 to 12 times greater than diesel fuel. Increased carbon chain length and reduced number of double bonds were associated with increased oil viscosity, cetane rating and gross heat content [37]. The high viscosity of JSVO is basic cause of poor combustion and atomization characteristics. The viscosity of JSVO is about 10-12 times the viscosity of diesel. Because of high viscosity the injector choking is one problem along with some others like case of fuel injector, gum formation and piston sticking under long term use. [38]

The study on JSVO for combustion analysis in IDI diesel engine has reported that high viscosity and density of vegetable oil led to problem in the injection system and combustion chamber of the engine for a long term usage. It was also commented that the high viscosity problem has been solved in several ways such as preheating the oil or blending it with petroleum base diesel fuel. The experimental results show excessive carbon deposit taking place in combustion chamber when Soybean (SVO) was used as fuel. This happened because of high viscosity effects on fuel atomization phenomena [39, 40].

The experiment was carried out by preheating the fuel and it was observed that BTE at all the operating points was improved by preheating. This shows that the engine performance improve by preheating the JSVO there by reducing the viscosity [41].

The surface tension is the fuel property that affects spray atomization, droplet size and other important properties of the diesel spray. It has been found that the surface tension of JSVO is 0.04 N/m in comparison with 0.02 N/m for diesel fuel. The higher value of surface tension of JSVO results in poor combustion efficiency due to poor atomization characteristics. Reducing the surface tension will enhance the fuel atomization quality and release of finer fuel particle in combustion chamber from fuel injector. There are various methods to reduce the surface tension of fuel for example by using the additives, heating the fuel or by blending with low surface tension fuel etc. The efforts made for reducing the viscosity by preheating the fuel using of heat contained in exhaust gases will simultaneously reduce the surface tension also. Therefore preheating the fuel will enhance the fuel combustion quality due to improved fuel atomization phenomenon.

The densities of the diesel and JSVO used for the experiment were found out experimentally in the laboratory. The density of JSVO was found 910 kg/m^3 whereas the density of diesel was 817 kg/m^3. Therefore though the volumetric fuel injection by the fuel injector may remain same in both the cases but the delivery of slightly greater mass of JSVO will take place. As the calorific value of JSVO is 39,000 kJ/kg and the calorific value of diesel fuel is 42,000 kJ/kg therefore the net energy supplied to the engine will be almost same in both the cases. Higher density of JSVO will result higher mass of fuel particles injected in the combustion chamber. Because of higher mass the fuel particle will have higher momentum so the distance travelled by the fuel particle in the combustion chamber will also be higher. As we know the rate of fuel burning depends upon the rate at which the product of combustion be removed from the combustion chamber and replaced by the fresh oxygen, i.e. it depends upon the rate at which the burning droplet can move relative to the surrounding air. A heavier droplet will have higher momentum and hence higher relative velocity thereby establishing efficient scavenging process [42, 43]. Therefore higher fuel density is advantageous in both the ways, one in supplying comparatively higher fuel energy in combustion chamber and second in improved combustion process due to efficient scavenging.

About 87% of the schools in the country are located in the rural areas. About 25.81 million or (61.3%) of the enterprises in India are located in the rural areas (Economic Census 2005). The energy need in the rural areas is growing very fast. The search for alternate energy sources are going on globally. This research work will benefit huge population and make them self dependent [44, 45].

The agricultural engineering department at the University of Missouri-Columbia has fueled a 1991 and a 1992 Dodge pickup 5.9 L (360 in^3) Cummins engine with methyl-ester soybean oil (soydiesel) for more than 80,467 km (50,000 miles). Fueling the engines with 100% soydiesel increased engine power by 3% (1991 engine) and reduced power by 7% (1992 engine). Analysis of engine lubrication oil suggested that the engine were wearing at a normal rate. The black exhaust smoke normally observed when a diesel engine accelerates was reduced by 86% when the diesel engine was fueled with 100% soydiesel [46]. The competitive potential of biodiesel is limited by the price of vegetable oil, which strongly influences the final price of biofuels. An appropriate planning and design of the whole production process, from the seed to the biodiesel end product, is essential in order to contain the fallout of energy inefficiencies in the high price of the end product [47]. Vegetable oils and animal fats can be transesterified to biodiesel for use as an alternative diesel fuel. Conversion of low cost feed stocks such as used frying oils is complicated if the oils contain large amounts of free fatty acids that will form soaps with alkaline catalyst. The soaps can prevent separation of the biodiesel from the glycerin fraction. Alternative processes are available that use an acid catalyst [48].

The calorific value of JSVO is 39, 000 kJ/kg and the calorific value of diesel fuel is 42,000 kJ/kg. The calorific value of JSVO is comparable with calorific value of diesel fuel. Since the calorific value of JSVO is marginally less than the diesel fuel but the density is high therefore the energy supplied by the fuel will remain almost same so the performance of the engine can be expected similar as if diesel fuel. The calorific value of diesel fuel (42,000 kJ / kg) in higher than the calorific value of JSVO (39,000 k J/kg), where as the density of JSVO (910 kg/m^3) is higher than the density of diesel (817 kg /m^3). The fuel is sold in the market in the denomination of liter. So the energy contained in one liter of diesel and JSVO is also compared. It is calculated that one liter diesel contains 34314 kJ where as the one liter JSVO contains 35490 kJ. The comparison of engine efficiency at constant energy input can be studied by calculating the brake specific energy consumption (BSEC). The BSEC is obtained by multiplying the BSFC by 42000 in case of diesel fuel and by 39000 in case of JSVO.

There is a huge potential in energy contained with JSVO for the utilization as alternate energy which will reduce the burden of energy demand be met by petro-diesel. As the energy prices are growing high and limited resource of petro-diesel the feasibility of JSVO utilization in diesel engine as replacement of diesel particularly in remotely established rural areas will make the farmers self dependent and also improve their socio-economic conditions.

The Physico-Chemical properties of JSVO are far different than that of diesel. The viscosity of JSVO is about 10 to 12 time than that of diesel whereas the density of JSVO (910 kg/m^3) is also higher than that of diesel (817 kg/m^3). The calorific value of JSVO is about 39000 kJ/kg in comparison with calorific value of diesel which is about 42000 kJ/kg. Therefore energy available with JSVO is comparable with diesel and also the potential of vegetable oil as alternative fuel is much higher. As the Jatropha plant is cultivated primarily in barren land, the use of unproductive land will increase. The research carried out and reports indicate that the existing diesel engines or not compatible with JSVO as fuel. The efforts are required to explore the possibility of utilizing JSVO by either improving the properties of JSVO to suit the requirement of diesel engine or modifications in diesel engine.

Experimental studies on effects of oxygenated fuels in conjunction with single and split fuel injections were conducted at high and low loads on a Caterpillar SCOTE DI diesel engine. At high loads, a significant beneficial effect of oxygenated fuels was seen to reduce soot emissions with little or no penalty on NO_x emissions. Also, at high loads, split injection had an additional favorable effect on soot emissions as compared to single injections, but the soot reducing influence of the oxygenates was not as marked as that seen with the single injection cases. This result indicates that the soot reduction due to the addition of oxygenate to the fuel is most effective in rich combustion as split injections are known to be effective at leaning-out the charge. In fact, at low engine loads when the overall mixture is further leaned-out, the oxygenated fuels had only a slight effect on particulate emissions. Split injections were effective in reducing particulate emissions at low loads particularly at advanced fuel injection timings when overall temperatures would be expected to be higher [49].

Non-edible vegetable oils such as Pongamia oil and Jatropha curcas oil are found to be effective substitute fuels in the low heat rejection diesel engine. Esterification, preheating and increase in injection pressures have been tried for effective utilization of the vegetable oils. Performance parameters such as the brake specific energy consumption (BSEC) and exhaust gas temperature (EGT) have been reported for varying magnitudes of brake mean effective pressure (BMEP.) with different non-edible vegetable oils as substitute fuels. The pollution levels of black smoke and NO_x have been recorded. Combustion diagnosis is also carried out with the aid of a miniature piezoelectric pressure transducer and TDC (top dead centre) encoder [50].

Dynamometer tests have been carried out to evaluate the performance, emissions and wear characteristics of an indirect injection diesel engine when fueled by 10, 20, 30, 40 and 50 per cent blends of ordinary coconut oil with ordinary diesel fuel.

The performance and emissions characteristics results showed that 10-30 per cent coconut oil blends produced slightly higher performance in terms of brake power than ordinary diesel fuel. All the coconut oil blends produced lower exhaust emissions including polycyclic aromatic hydrocarbons and particulate matter. The wear and lubrication oil characteristics results showed that coconut oil blends up to 30 per cent produced similar results to ordinary diesel fuel. This programme will give useful information for further research and development in the future if coconut oil is used as an alternative to ordinary diesel [51].

Neat vegetable oils pose some problems when subjected to prolonged usage in CI engine. These problems are attributed to high viscosity, low volatility and polyunsaturated character of the neat vegetable oils. These problems are reduced to minimum by subjecting the vegetable oils to the process of transesterification. The prepared biodiesel was then subjected to performance and emission tests in order to evaluate its actual performance, when used as a diesel engine fuel. It was found that 20 percent blend of biodiesel gave the best performance amongst all blends. It gave net advantage of 2.5 percent in peak thermal efficiency and there was substantial reduction in smoke opacity values [52].

A kinetic study in free catalyst transesterification of rapeseed oil was made in subcritical and supercritical methanol under different reaction conditions of temperatures and reaction times. Runs were made in a bath-type reaction vessel ranging from 200°C in subcritical temperature to 500°C at supercritical state with different molar ratios of methanol to rapeseed oil to determine rate constants by employing a simple method [53].

The reduction of emissions from engines has become a major factor in the development of new engines and manufacturers are trying to meet the requirements specified by EPA. As a result the use of alternative fuels as a means of meeting these requirements has generated much attention. Today the economics are much more favorable in the production of ethanol and it is able to compete fairly well with standard diesel. Hence there has been renewed interest in the ethanol-diesel blends with particular emphasis on emissions reductions. When considering an alternative fuel for use in diesel engines, a number of issues are important. This purpose of this paper is to review these issues with particular reference to safety and distribution, integrity of the fuel being delivered to the engine, emissions, engine performance and durability [54].

Blends of ethanol and diesel fuel were investigated and found to be technically feasible; however, the high costs of ethanol production meant that the fuel could only be considered in cases of fuel shortages. In the last two decades of the 20thcentury, major advances in engine technology have occurred, leading to greater fuel economy in vehicles. When considering an alternative fuel for use in diesel engines, a number of issues are important. The reduction of emissions from engines has become a major factor in the development of new engines and manufacturers are trying to meet the requirements specified by EPA [55].

A Cummins diesel engine was operated on an 80:13:7 % (V/V) blend of diesel fuel: methyl tallowate: ethanol. Engine exhaust emissions analyses were performed, and the engine oil was analyzed for accumulation of heavy metals at 45 h intervals. It was observed that engine performance was satisfactory for 148 h at which time the injector in cylinder number 2 failed. The injector was changed, and after an additional 11 h (159 h total) of operation the injector in cylinder number 5 failed. That injector also was replaced, and the 200 h procedure was continued. The test was discontinued after 197 h when the supply of the fuel blend was exhausted. The injectors were removed and the injector in cylinder number 1 was observed to be coked. This injector was sent to the Cummins Engine Co. for analysis. If was found that failure was not because of the fuel used, but because a crack had developed across the tip due to an excessively tight overhead adjustment. The power output, torque produced and brake specific fuel consumption of the engine was more or less constant throughout the test. Emissions analyses after every 45 h did not show any increase in exhaust gases or deterioration of engine performance. Engine oil analyses performed for accumulation of heavy metals did not show any excessive wear on the engine parts [56].

The stability and homogeneity of 32 micro emulsions prepared by mixing 150.DEG, 160.DEG, 170.DEG. and 180.DEG. proof ethanol-1-butanol-diesel in different proportions were studied. 13 micro emulsions were stable. However, micro emulsions with 170.DEG. and 180.DEG. proof ethanol in 1:2.5:5.5 and micro emulsion with 180.DEG. proof ethanol-1-butanol-diesel in 1:2:3 ratio were more suitable for fuel use in C I engines based on the comparison of their characteristic fuel properties with diesel [57, 58].

Biodiesel has become more attractive to replace petroleum fuel. As per the reported literature, most of the transesterification studies have been done on edible oils like rapeseed, soyabean, sunflower & canola by using methanol and NaOH/KOH as catalyst. There are very few studies reported on production of biodiesel by utilizing non-edible oils, among which, Karanja is one of the most potential species to produce biodiesel in India, which could offer opportunities for generation of rural employment, increasing income and improving environment [59].

Biofuels derived from renewable plant sources (tree borne vegetable oil) hold immense potential for meeting India's future energy needs. Biodiesel used for these studies was derived from Jatropha curcus. Blends of Biodiesel up to 15% did not affect the engine power while blends with higher proportion of Biodiesel showed tendency to decrease the engine power. Best fuel economy was observed with 10% biodiesel blended fuel. Oxides of Nitrogen (NO_x) emissions, increased under different operating conditions while smoke was reduced at all speed ranges in Road Load Simulation and Wide Open Throttle test modes. Other test data generated revealed that engine oil temperature tend to be less with biodiesel operation and exhaust gas contains more O_2 which is beneficial for the vehicles equipped with diesel oxidation catalyst [60].

A technique to produce biodiesel from mahua oil (Madhuca indica) having high free fatty acids (19% FFA) has been developed. The high FFA level of mahua oil was reduced to less than 1% by a two-step pretreatment process. Each step was carried out with 0.30–0.35 v/v methanol-to-oil ratio in the presence of 1% v/v H_2SO_4 as an acid catalyst in 1-hour reaction at 60°C. After the reaction, the mixture was allowed to settle for an hour and methanol–water mixture that separated at the top was removed. The second step product at the bottom was transesterified using 0.25 v/v methanol and 0.7% w/v KOH as alkaline catalyst to produce biodiesel. The fuel properties of mahua biodiesel were found to be comparable to those of diesel and conforming to both the American and European standards [61].

Large amount of non-edible type oils and fats are available. The difficulty with alkaline-esterification of these oils is that they often contain large amounts of free fatty acids (FFA). These free fatty acids quickly react with the alkaline catalyst to produce soaps that inhibit the separation of the ester and glycerin. A two-step transesterification process is developed to convert the high FFA oils to its mono-esters. The first step, acid catalyzed esterification reduces the FFA content of the oil to less than 2%. The second step, alkaline catalyzed transesterification process converts the products of the first step to its mono-esters and glycerol. The major factors affect the conversion efficiency of the process such as molar ratio, amount of catalyst, reaction temperature and reaction duration is analyzed. The two-step esterification procedure converts rubber seed oil to its methyl esters. The viscosity of biodiesel oil is nearer to that of diesel and the calorific value is about 14% less than that of diesel. The important properties of biodiesel such as specific gravity, flash point, cloud point and pour point are found out and compared with that of diesel. This study supports the production of biodiesel from unrefined rubber seed oil as a viable alternative to the diesel fuel [62].

Pure rubber seed oil, diesel and biodiesel are used as fuels in the compression ignition engine and the performance and emission characteristics of the engine are analyzed. The lower blends of biodiesel increase the brake thermal efficiency and reduce the fuel consumption. The exhaust gas emissions are reduced with increase in biodiesel concentration. The experimental results proved that the use of biodiesel (produced from unrefined rubber seed oil) in compression ignition engines is a viable alternative to diesel [63].

In the wake of the present fuel crisis, it has become essential to identify some renewable and environmentally compatible substitutes to diesel fuel. Three test fuels of neat diesel; blends of 20% volume (B20) of proportions of Jatropha oil methyl ester (JOME) with diesel; and neat JOME (B100) were prepared, analyzed and compared with diesel fuel. The performance and exhaust emission of the engine using blend of 20% JOME With diesel and neat JOME was evaluated in a single cylinder C.I. engine and compared with the performance obtained with diesel. Significant improvement in engine performance was observed compared to biodiesel alone. The specific fuel consumption and the exhaust gas temperature were reduced due to decrease in viscosity of the biodiesel. Acceptable thermal efficiencies of the engine were obtained with neat JOME. Exhaust emission behavior was environment friendly. From the properties and engine test results it has been established that 100% of Jatropha oil methyl ester can be substituted for diesel without any engine modification [64, 65].

HSD and polanga oil methyl ester (POME) fuel blends (20%, 40%, 60%, 80%, and 100%) were used for conducting the short-term engine performance tests at varying loads (0%, 20%, 40%, 60%, 80%, and 100%). Tests were carried out over entire range of engine operation at varying conditions of speed and load. The brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) were calculated from the recorded data. The engine performance parameters such as fuel consumption, thermal efficiency, exhaust gas temperature and exhaust emissions (CO, CO_2, HC, NO_x, and O_2) were recorded. The optimum engine operating condition based on lower brake specific fuel consumption and higher brake thermal efficiency was observed at 100% load for neat biodiesel. From emission point of view the neat POME was found to be the best fuel as it showed lesser exhaust emission as compared to HSD [66, 67].

2.6 Motivation

Substantially large population of India lives in rural areas and use animal dung, agricultural waste and fuel wood as fuel for cooking. About 87% of the schools in the country are located in the rural areas. About 25.81 million or (61.3%) of the enterprises in India are located in the rural areas (Economic Census 2005). The energy need in the rural areas is growing very fast. The search for alternate energy sources are going on globally. This research work will benefit huge population and make them self dependent [44,45].

As there are two ways to meet the requirements, one by processing the JSVO to make the biodiesel and feed in the existing engine and second by modifying the existing engine suitable or using filtered JSVO directly. First method is not convenient for the population which has been targeted in this work because of the complicated technology involved and costly equipments required. If the processing is done on industry level then the basic idea of facilitating the typically established people will get defeated because they are to depend on market and also pay even more cost of bio diesel cost in comparison with diesel in present scenario.

The above requirement motivates the work to follow the second method i.e. making the existing engine compatible with JSVO. The literature survey and the technology involved in IDI engine suggest many types of modifications in the engine i.e. designing the combustion chamber suitable for JSVO, pre heating the JSVO, using multi point fuel injection system, altering the advance angle of fuel injection point, injecting the fuel at elevated injection pressure. The feasibility of modifications on the basis of complication and economy is studied.

2.7 Statement of problem:

The present work would align with the global objective of searching the alternate fuel. The use of Straight vegetable oil in engines will reduce the ever growing load on petro diesel. The cultivation of Jatropha plants at large sale will not only increase the reduce the problem of environmental degradation but also the utilization of non productive land will increase

The non availability of petro diesel to the remote areas is big problem for the farmers and other users in those areas. They have to travel for long distances and also pay ever increasing high cost of diesel. The literature review has convinced that the JSVO can be used in IDI diesel engines easily in comparison with DI diesel engines because of high degree of atomization required in this type. The researchers have reported various problems which were observed on using SVO in IDI diesel engine. The majority of problems were felt because of high viscosity of SVO and poor atomization characteristics.

The improvement in fuel atomization characteristics can be achieved by many methods. . In this work the combination of fuel preheating, altering the fuel injection pressure and the advancing the fuel injection point were utilized. These modifications not only make the engine compatible with JSVO but also enhance the engine performance.

2.8 Need for present work:

The search for alternate fuel under present day today increasing energy demand has become global objective. The JSVO has good potential to meet the objective. There are the research and technology utilization going on to process the JSVO to make the biodiesel. The bio diesel can become a substitute for diesel which can be used in existing engine without any modification. Another possibility of utilizing the JSVO in light of alternate fuel is to fuel the diesel engine with neat JSVO. The literature survey has shown that the neat JSVO has not given good results particularly in DI engine but it gives somewhat better results in IDI engine. The existing IDI engine is required to be modified so that the engine performance with JSVO fuel can be improved.

As an objective of modification in IC engine which can be fueled with JSVO while giving better efficiency, the IDI diesel engine was run on JSVO for its duration test to identify the problems by checking the physical conditions, conducting the performance test, emission test and calculate the test parameters to identify the areas of problems among which some of them can be considered for modification. Based upon the analysis of above stated test results following objectives are identified for this work. As the higher viscosity of JSVO is a major factor which leads to poor combustion, there is need to develop a heat exchanger which can heat up the JSVO and lower the viscosity. This would also improve the fuel atomization quality in combustion chamber. The poor atomization of JSVO is another factor which results in poor combustion and more deposits, there is need to identify the methods for improving the atomization phenomena. Among many of the factors governing the fuel atomization characteristics, one factor is fuel injection pressure. Higher injection pressure will release smaller fuel droplets in combustion chamber. The injection pressure is to be elevated to meet the requirement.

The fuel particle size released in combustion chamber because of high viscosity is larger. The time lag available in existing engine does not meet the requirement of preparing the fuel air mixture for efficient combustion. The time duration for physical delay and chemical delay is required to be increased to make sufficient time available for proper atomization and pre ignition process. The cam follower position which governs the fuel injection point is to be modified so that the fuel injection takes place early. The advancement of fuel injection is to be done by which amount is to be identified and recommend so that presently available IDI engine can use JSVO as fuel directly.

2.9 Objectives of present investigation

Based upon the literature survey and the present trend of research work in this area, the following objectives are identified.

1. Optimization of fuel injection pressure to enhance engine performance and emission characteristics
2. Modification and evaluation of fuel injection time lag on engine performance and emission characteristics
3. Design and fabrication of heat exchanger for fuel preheating in fuel injection system of IDI engine
4. Comparison of modified engine performance with unmodified engine performance fueled with JSVO

CHAPTER 3 TEST RIG DEVELOPMENT AND METHODOLOGY

3.1 Introduction

To study the performance and emission characteristics of the engine with diesel as reference fuel and the JSVO, it was very important to characterize the properties of fuels. The fuel properties such as kinematic viscosity, density, surface tension and calorific value were determined in laboratory with the help of new/calibrated instruments. The engine was integrated with engine loading system, fuel consumption measurement unit and the thermocouples were inserted and attached with indicators to measure the temperatures. The TDC and BDC points were marked with help of magnetic dial indicator. The engine was tested with diesel and JSVO. The heat exchanger was designed, fabricated and installed on the engine. The modified engine was tested with JSVO AND the results were analyzed. The test equipment was checked for data repeatability and reproducibility. The necessary precautions were exercised during the experiments and data recording.

3.2 Measurement of physico-chemical properties of JSVO

The physico-chemical fuel properties of JSVO and diesel has very important role in defining the combustion quality of the fuel. The combustion quality directly relates to engine performance and emission characteristics. Therefore the kinematic viscosity, surface tension, density and calorific value of JSVO as well as diesel were determined in the laboratory. The procedures and the capabilities of the instruments have been discussed in following sub sections.

3.2.1 Measurement of density

The densities of JSVO and diesel used for the experimentation were determined in the laboratory with the help of hydrometer. The hydrometer was capable of finding out the specific gravity of the test fuel with an accuracy of 0.001. Initially 100 ml JSVO was taken in a beaker and a suitable hydrometer was selected to suit the requirement of specific gravity range. The hydrometer was floated in the beaker and was allowed to get stabilized. The reading was directly read on the graduated scale of hydrometer. After finding out the specific gravity of JSVO, the beaker was flushed with diesel. The same procedure was repeated for diesel fuel also. The same experiment was repeated 5 times to ascertain the accuracy of data observed. Finally the mean value was taken for calculation of density. The following expression was used to find the density:

Mass density of test fuel = specific gravity of test fuel x mass density of water

(A) Specific gravity of diesel as observed during the experiment = 0.817

$\therefore$ The density of diesel = 0.817 x 1000 = 817 kg/ m^3

(B) Specific gravity of JSVO as observed during the observation = 0.910

$\therefore$ The density of JSVO = 0.910 x 1000= 910 kg/ m^3

3.2.2 Measurement of kinematic viscosity

The kinematic viscosity of the test fuels was determined with the help of digital rotational viscometer as shown in figure 3.1. A rotor is rotated in test fuel and the resistance offered by the fuel to the rotor due to viscosity was used for measuring the kinematic viscosity. The accuracy of equipment was ± 0.1 cSt and the repeatability of data was checked by repeating the experiment. Initially the diesel fuel was put into the chamber and the viscosity was found.

Figure 3.1: Rotational Viscometer

Table 3.1: Kinematic viscosity of JSVO and diesel at different temperatures

Temperature (^{0} C)	Kinematic Viscosity (cSt)	
	Diesel	JSVO
30	4.3	48.7
40	4.1	27
50	3.8	20
60	3.7	16
70	3.5	12
80	3.49	6.8
90	3.45	3.68
100	3.1	3.16

The viscosity of JSVO was determined at different temperatures to find out the temperature at which the viscosity of JSVO matches with viscosity of diesel at room temperature (Table 3.1).

3.2.3 Measurement of surface tension:

The surface tension of JSVO and diesel was found out in laboratory with the help of Multi Frequency Ultrasonic Interferometer (figure 3.2). The Ultrasonic Interferometer is a simple and direct device to determine the ultrasonic velocity in liquids. The principle used in measurement of velocity (v) is based on the accurate determination of the wavelength (λ) in the medium. Ultrasonic waves of known frequency (f) are produced by a quartz crystal fixed at the bottom of the cell. These waves are reflected by a movable metallic plate kept parallel to quartz crystal. This acoustic resonance gives rise to an electrical reaction on the generator driving the quartz crystal and the anode current of the generator becomes a maximum.

If the distance is increased or decreased and the variation is exactly one half wavelengths or multiple of it, anode current becomes maximum. From the knowledge of wavelength the velocity can be obtained by the relation:

$$\text{Velocity} = \text{Wavelength} \times \text{Frequency}$$

Figure 3.2: Multi Frequency Ultrasonic Interferometer

Table 3.2: Data obtained from interferometer for calculation of surface tension of JSVO

Calculation for surface tension of Jatropha Oil at 27^{0}C							
S. No.	**Micrometer reading corresponding to maxima (mm)**	**λ/2 (mm)**	**S. No.**	**Observation Data**	**λ/2 (mm)**	**Average (λ/2) mm**	**Surface tension (S) = ($2^{3/2}$x 6.3x 10^{-4} x 910)$\lambda^{3/2}$ N/m**
1	0.366		14	4.98	0.356		
2	0.731	0.365	15	5.339	0.359		
3	1.096	0.365	16	5.695	0.356		
4	1.413	0.317	17	6.057	0.362		
5	1.779	0.366	18	6.424	0.367		
6	2.147	0.368	19	6.775	0.351	0.35696	
7	2.481	0.334	20	7.133	0.358		
8	2.844	0.363	21	7.49	0.357		
9	3.187	0.343	22	7.85	0.36		**0.042**
10	3.546	0.359	23	8.211	0.361		
11	3.906	0.36	24	8.573	0.362		
12	4.269	0.363	25	8.93	0.357		
13	4.624	0.355	26	9.29	0.36		

Table 3.3: Data obtained from interferometer
for calculation of surface tension of diesel

Calculation of Surface tension of Diesel at 27^0C							
S. No.	**Micrometer reading corresponding to maxima (mm)**	**$\lambda/2$ (mm)**	**S. No.**	**Observation Data**	**$\lambda/2$ (mm)**	**Average ($\lambda/2$) mm**	**Surface tension (S) = $(2^{3/2}*6.3*10^{-4}*817)\lambda^{3/2}$ N/m**
1	0.336		14	4.662	0.336		
2	0.668	0.332	15	4.992	0.33		
3	1	0.332	16	5.322	0.33		
4	1.33	0.33	17	5.653	0.331		
5	1.662	0.332	18	5.989	0.336		
6	1.995	0.333	19	6.324	0.335	0.33296	
7	2.321	0.326	20	6.66	0.336		
8	2.662	0.341	21	6.993	0.333		
9	2.994	0.332	22	7.327	0.334		**0.028**
10	3.327	0.333	23	7.661	0.334		
11	3.66	0.333	24	7.997	0.336		
12	3.991	0.331	25	8.327	0.33		
13	4.326	0.335	26	8.66	0.333		

3.2.4 Measurement of calorific value

A bomb calorimeter was used to measure the amount of heat generated when a known amount of matter was burnt in a sealed chamber in an atmosphere of pure oxygen gas. It is a simple inexpensive, accurate method for determination of calorific value. Essentially the apparatus consist of following part Bomb, Water jacket, offset stirrer, Calorimeter vessel, bomb firing unit, vibrator, timer, illuminator with magnifier, pressure gauge on stand, gas release valve, pellet press, crucible, ignition wire. The chemicals used for determination of calorific value were benzoic acid, naphthalene, sucrose or cane sugar, standard alkali solution, methyl orange or methyl red indicator.

A known amount of sample fuel was burnt in a sealed chamber (bomb). The air was replaced by pure oxygen. The sample was ignited electrically. As the sample burnt, heat was produced. The rise in temperature was determined. Since barring loss of heat the amount of heat produced by burning the sample must be equal to the amount of heat absorbed by the calorimeter assembly, knowledge of the water equivalent of the calorimeter assembly and of the rise in temperature enables one to calculate the heat of combustion of the sample.

If, W= water equivalent of calorimeter assembly in cal / ^{o} C;

T= Rise in temperature registered by a sensitive thermometer in ^{o} C;

H= heat of combustion of material in cal / gm and

M= Mass of sample burnt in gram.

$$WT=HM$$

H was calculated easily since W, T and M were known.

One gram of air dried sample was compressed into a pellet using the pellet press. The pellet was accurately weighed in the stainless steel crucible and the crucible was placed inside the bomb of 300 ml capacity. A piece of the firing wire (nichrome wire) of known length was stretched across the electrode within the bomb. A known length (15 cm) of cotton thread was tied around the wire. The crucible was placed in position and the loose ends of the thread were arranged so that they are in contact with the material. The same length of thread was used in each determination. 5 ml of distilled water was introduced into the bomb to absorb vapour of sulphuric acid and nitric acid formed during the combustion. The bomb was reassembled avoiding excess pressure. The bomb was charged slowly with oxygen from a cylinder to a pressure of 25 atmospheres without displacing its original air content. The valve was effectively closed using as little pressure as possible and the bomb was detached from the oxygen supply.

A quantity of water was weighed into the calorimeter vessel which should be sufficient to submerge the nut of the bomb to a depth of at least two millimeter leaving the terminals projecting. The same weight of water was used in all the tests. The calorimeter vessel water jacket was transferred and the bomb was lowered carefully into the calorimeter vessel. The bomb was connected to the ignition circuit through a switch for subsequent firing of the charge. The stirrer was adjusted and the thermometer was placed in its position. The stirring mechanism was started which must be kept in continuous operation at a constant speed during the experiment.

After an interval of not less than ten minutes, the temperature was read to 0.001^0C and continued the readings for five minutes, at equal intervals of not more than one minutes, tapping the thermometer lightly during 10 seconds prior to each reading. If, over a period of five minutes, the average deviation of the individual values of the rate of change of temperature is less than 0.00072^0C per minute, the circuit was closed momentarily to fire the charge and continued the observations of the temperature at intervals of similar duration to those of the preliminary period. If the rate of change of temperature is not constant within this limit, extend the preliminary period until it is constant. In the chief period which extended from the instant of firing until the time after which the rate of change of temperature again becomes constant, the earlier reading was taken to the nearest 0.01°C since it would not be possible to take the earlier reading to 0.001°C. The readings were resumed to this - precision as soon as possible. The rate of change of temperature was determined in the after period (which follows the chief period by taking readings at 1 minute intervals for at least five preferably ten minutes). It should be noted that, it was desirable to keep the jacket temperature and the room temperature as close to the calorimeter temperature as possible. The jacket and room temperature should therefore was recorded.

The bomb was removed from the calorimeter and after a lapse of about half an hour from the time of firing allowing the acid mist to settle, the pressure was released by opening the valve. It was verified that the combustion had been completed by noting the absence of any sooty deposit within the bomb. The presence of any trace of sooty deposit indicates incomplete combustion and invalidates the test.

The effective heat capacity of the system was determined by burning pure and dry benzoic acid weighing not less than 0.9 and more than 1.1 gram. The corrected temperature rise was determined (T), from the observed test data, and the un burnt fuse wire was measured, the energy equivalent was computed by substituting in the following equation.

$$W = \frac{HM + E_1 + E_2}{T}$$

W = Energy equivalent of calorimeter in calories per degree centigrade;

H = Heat of combustion of standard benzoic acid in calories per gram;

M = Mass of standard benzoic acid sample in grams;

T = Corrected temperature rise in degrees centigrade;

E_1 = Correction for heat of formation of nitric acid in calories; and

E_2 = Correction for heat of combustion of firing wire, in calories

3.3 Test Rig Development

3.3.1 Engine

The engine selected for this study is widely used mostly for agricultural irrigation purposes and also in many other small and medium scale commercial applications like producing electricity, running flour mills, oil expellers, rice mills etc. This is a single cylinder, four stroke, vertical, water cooled system having a bore of 120 mm and stroke of 139.7 mm. The engine can be started by hand cranking, using decompression lever. The engine was provided with a centrifugal speed governor. The engine was mounted on vibration isolators to avoid the excessive vibrations. The test engines was directly coupled to a 230 V single phase AC generator of 7.5 kVA capacity to absorb maximum power produced by the engine. The coupling of engine and generator was done by V-belt drive power transmission system. The diameters of the pulley mounted on engine and generator was so selected that the engine at designed rpm of 1000 at full load runs the generator at 1500 rpm. The rpm of generator to provide rated output at 235 volt was 1000. The integrated system of experimental set up is shown in figure 3.3. The technical specifications of the engine and generator are listed in table 3.4 and table 3.5 respectively.

Figure 3.3: Field marshal make single cylinder IDI diesel engine

Table 3.4: Specification of Field Marshal make single cylinder IDI diesel Engine

Sl. No.	Particulars	Specifications
1	Make	Field Marshal diesel engines
2	Model	FM-4
3	Rated Brake Power (BHP/kW)	10/7.35110
4	Rated speed (rpm)	1000
5	Number of cylinder	One
6	Bore x Stroke (mm)	120x139.7
7	Compression ratio	17:1
8	Coling System	Water Cooled
9	Lubrication System	Forced Feed
10	Cubic Capacity	1580 cc
11	Nozzle	DL30S1202MICO
12	Nozzle Holder	9430031264 MICO
13	Fuel Pump	9410032034
14	Fuel Pump Plunger	9x03/323 MICO
15	Injection Pressure	145 kgf/cm^2
16	Specific Fuel Consumption	265 gm /kWh OR 195 gm / bhp /h
17	Sump Capacity	4.5 Liter

Table 3.5: Specifications of the alternator coupled with single cylinder diesel engine

Sl No	Particulars	Specifications
1	Make	Field Marshal
2	Alternator Type	Single Phase, 50 Hz, AC
3	Rated Out put	7.5 KVA at 1500 rpm
4	Rated Speed (rpm)	1500
5	Rated Voltage (volts)	230
6	Rated Currents (amp)	32

3.3.2 Engine loading system

The hydraulic loading arrangement was used to load the engine (Figure 3.4). The hydraulic loading arrangement had capability of gradual regulation of load as per the requirements. The study was carried out by loading the engine with 0 kW, 1.84 kW, 3.68 kW, 5.52 kW and 7.35 kW. It had a container of about 40 liters capacity to store water. Two parallel steel plates of defined dimensions were dipped in the water. The plates were regulated to move up or down by hand wheel. The plates were isolated from hand wheel with the help of asbestos link provided in between to avoid the current to pass to hand wheel. The two plates were taken as the electrodes to make the current pass through water. The generator was connected with the loading arrangement with the help of cable. The electricity generated by the generator was made to pass through these plates. The depth of immersion of the plates was regulated with the help of a hand wheel to control the current flowing through it. A digital ammeter having range of 50 Amp and accuracy $\pm$ 0.01 Amp was used to measure the current passing through the loading circuit. The digital voltmeter having range of 50 volt and accuracy 0.01 volt was used to measure the voltage across the terminals. The repeatability and reproducibility of the instrument was ascertained by repeating the observations. The standard deviation of 0.02 Amp was obtained in ammeter and 0.01 volt was obtained in the case of voltmeter. The load on the engine was varied by varying the current at constant voltage difference. The current rating corresponding to the load on engine was calibrated and shown in table 3.6 from fundamental relation of $P = VI \cos \Phi$.

The loading arrangement was capable of offering the load to the engine in the order of above requirements with accurately monitored on the indicator.

$$\text{The load on the engine} = \frac{\text{Voltage x Current}}{\text{1000x Generator efficiency}} \text{ kW}$$

Table 3.6: Current rating for part load estimation

Engine Load (kW)	% of Engine Rated Load	Current (ampere)
0	0	0
1.84	25	6.25
3.68	50	12.5
5.52	75	18.75
7.35	100	25

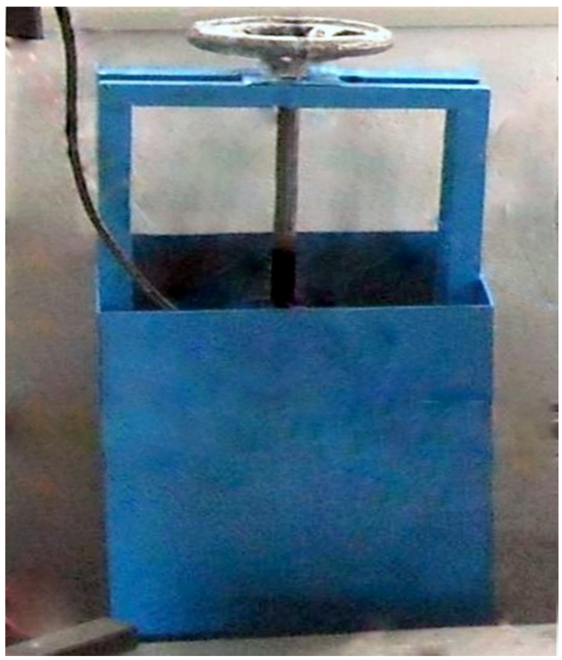


Figure 3.4 : Engine loading system

3.3.3 Engine control panel

The data display board was equipped with indicators connected to different locations of the test engine. Voltmeter, ammeter, temperature display etc. were used for data recording. The standard digital indicators having range as mentioned below were used to display the temperatures, voltage and current. Followings are the indicators installed on display board

I. The voltage indicator: Range 500 V

II. The current indicator: range 50 amp

III. Temperature indicators to indicate the water temperature of at inlet to engine cooling system.

IV. Temperature indicators to indicate the water temperature of at outlet from engine cooling system

V. Temperature indicator to indicate the exhaust temperature at inlet to heat exchanger

VI. Temperature indicator to indicate the exhaust temperature at outlet from heat exchanger

VII. Lubricating oil temperature

Figure 3.5: Data display board

3.3.4 Fuel supply system:

A burette was placed on the control panel to measure the volumetric fuel consumption of the engine at different loads as shown in figure 3.6. The fuel flow was measured by noting the time taken for 50 cc of fuel consumed by the engine. The same fuel supply system was used during all the three test phases i.e. existing engine running with diesel, JSVO and modified engine running with JSVO. Fuel was fed to the injector pump under gravity. The volumetric fuel consumption rate was measured by isolating the burette from main fuel supply line with the help of two flow regulators installed in main supply line and fuel supply from burette. The time for 50 cc fuel consumed was recorded with the help of stopwatch.

Figure 3.6 : Apparatus to measure rate of fuel and air consumption

3.3.5 Heat exchanger

Preheating the JSVO was one of the objectives of this study. The heat exchanger was designed to transfer the heat from exhaust gas to the fuel before injected into the combustion chamber (figure 3.7).

Fuel Injection Pipe to pass through proposed heat exchanger

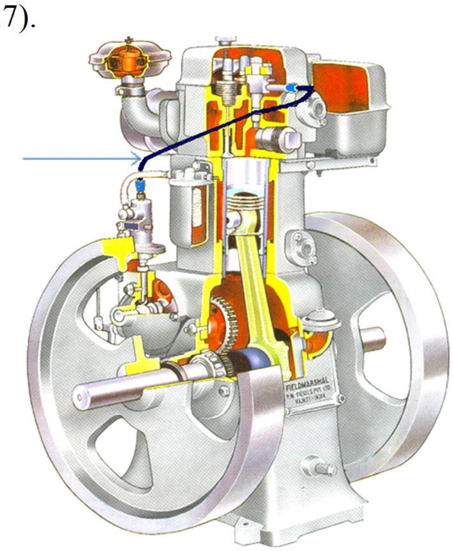


Figure 3.7 : Fuel injection pipe of IDI diesel engine considered for modification

The engine was considered to be running at 100% load for designing the heat exchanger. This is because of the linear relationship of heat transfer and the fuel consumption of the engine. The temperature of the exhaust gas was controlled by reducing the load on the engine. The variations of fuel supply and the temperature variation of heat exchanger necessitated the provision of regulating the mass flow of the exhaust gas by the flow regulator installation at the downstream of the heat exchanger.

The fuel preheating was done by cross flow heat exchanger. The heat exchanger was designed (figure 3.8 and figure 3.9).Since the exhaust gas temperature at part load was less so the heat available to get transferred to oil was also less. The requirement of heat availability was met by regulating the mass flow rate of the exhaust gas through the exchanger.

Figure 3.8: Heat exchanger with thermal insulation

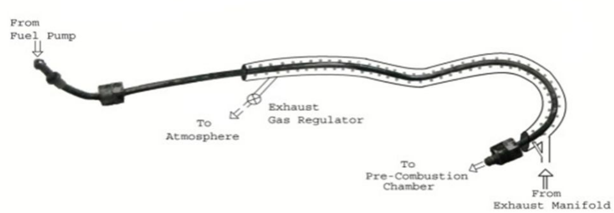


Figure 3.9: Sketch of heat exchanger

3.3.6 Heat exchanger design

The following parameters and assumptions were made for designing the cross flow heat exchanger

Fuel consumption = 0.265 kg / kWh

Engine rating = 7.35 kW

Rate of Fuel consumption (m) = $0.265 \times 7.35 = 1.2127$ kg/hr = 3.3686×10^{-4} kg/s

Properties of JSVO taken on tube side:

Density of SVO = 910 kg/m^3

t_i = Ambient Temp. 30^0 C (assumed)

t_o = 90^0 C (t_o = Desired outlet Temperature)

Assuming the specific heat of SVO = specific heat of diesel

C_p for diesel = 1800 J/kg K

The heat transfer rate, $Q = mC_p\Delta T = 3.3686 \times 10^{-4}. 1800. (90\text{-}30)$

$$Q = 36.38088 \text{ W}$$

Table: 3.7 Fuel properties data sheet used for heat exchanger design

Tube (Fuel)	**Shell (Exhaust)**
ρ_t =910 kg/m^3	T_i = 265^0 C at 100% load
ν_t= 48.7 cSt = 4.432x10^{-2} kg/m-s	T_o =100^0 C
k_t= 0.18W/m-K	ρ_s =1.12 kg/m^3
d_{ti}=2x10^{-3} m	μ_s=15.8x10^{-6} Pa-s
d_{to}=6x10^{-3} m	k_s= 0.02825W/m-K
	D_s=3.81x10^{-2} m
	Cp_s = 1068 J/Kg K

Determination of heat transfer coefficient of tube side:

Thermal diffusivity, $\alpha_t = \dfrac{k}{\rho Cp} = \dfrac{0.18}{910 \times 1800} = 1.0989 \times 10^{-7}$ m^2/s

Coefficient of thermal expansion, $\beta_t = \dfrac{1}{90 + \frac{30}{2} + 273} = 3.003 \times 10^{-3}$ K^{-1}

Rayleigh's Number, $Ra_t = \dfrac{\beta_t g(t_o - t_i) d_{ti}{}^3}{\nu_t\ \alpha}$

$$Ra_t = \frac{3.003 \times 10^{-3} \times 9.81(90-30)\left[2\times 10^{-3}\right]^3}{48.7 \times 10^{-6} \times 1.0989 \times 10^{-7}}$$

$Ra_t = 264.23 \times 10 = 2642.3$

For above calculated values of Ra_t, the Nusselt Number (Nu) is given by

$$Nu = 0.61(ReRa)^{\frac{1}{5}} \left[1 + \frac{1.8}{(ReRa)^{\frac{1}{5}}}\right]$$

$$Nu = 0.61(6.628)\left[1 + \frac{1.8}{6.628}\right]$$

Nu = 5.14

Also, $\dfrac{h_{ti} d_{ti}}{k_t} = Nu$

$$\therefore \frac{h_{ti} d_{ti}}{k_t} = 5.14$$

Or , $h_{ti} = \dfrac{5.14 \times 0.18}{2 \times 10^{-3}} = 462.7$ W/m^2 K

From the above value of h_{ti} and tube diameters h_{to} is given by,

$$h_{tio} = h_{ti} \times \frac{ID}{OD} = 462.7 \times \frac{2}{6} = 154.2$$

Determination of heat transfer coefficient of shell side:

Reynold's Number , $Re = \dfrac{4m_s}{\pi \mu_s d_{eq}}$

Where, $m_s = \dfrac{Q}{Cp_s(T_i - T_o)} = \dfrac{36.38}{1068(265-100)} = 2.4 \times 10^{-4}$

$$\therefore\ Re = \frac{4 \times 2.4 \times 10^{-4}}{\pi \times 15.8 \times 10^{-6} \times 44.1 \times 10^{-3}} = 438.8$$

Coefficient of thermal expansion on shell side, $\beta_s = \dfrac{1}{t_{fs}}$

t_{f_s} = mean film temperature

$$\beta_s = \frac{1}{\frac{265+100}{2} + 273} = 2.195 \times 10^{-3}\ K^{-1}$$

Thermal diffusivity on shell side, $\alpha_s = \dfrac{k}{\rho C_p}$

$$\alpha_s = \frac{0.02825}{1.12 \times 1068} = 2.36 \times 10^{-5}\ m^2/s$$

Rayleigh's Number, $Ra_s = \dfrac{\beta_s g\,(T_i - T_o)\,(D_s - d_{to})^3 \rho}{\alpha_s\,\mu}$

$$Ra_s = \frac{2.195 \times 10^{-3} \times 9.81(265-100)\left(3.81\times10^{-2} - 6\times10^{-3}\right)^3 \times 1.12}{2.36\times10^{-5} \times 15.8\times10^{-6}}$$

$Ra_s = 3.5298 \times 10^5$

For above calculated values of $Ra_{S,}$ the Nusselt Number is given by

$$Nu_s = 0.61(Re\ Ra)^{\frac{1}{5}}\left[1 + \frac{1.8}{(Re\ Ra)^{\frac{1}{5}}}\right]$$

$Nu_s = 0.61(68.86)\left[1 + \frac{1.8}{68.86}\right] = 43.1$

Also, $Nu_s = \frac{h_s(D_s - d_{to})}{k_s} = 43.1$

$\therefore h_s = \frac{43.1 \times 0.02825}{(3.8\times10^{-2} - 6\times10^{-3})}$

$h_s = 37.93$

Log mean temperature difference, $LMTD = \frac{105}{ln\frac{175}{70}}$

Or, $LMTD = 114.6$

The overall heat transfer coefficient for a thin walled tube is given by,

$$\frac{1}{u} = \frac{1}{h_s} + \frac{1}{h_{t_{io}}} = \frac{1}{37.93} + \frac{1}{154.23}$$

$\therefore u = 30.44$

$Q = U.\ A.\ LMTD$ $(A = \pi DL)$

$36.38088 = 30.44 \times \pi \times 6 \times 10^{-3} \times L \times 114.6$

$\therefore\ L = 0.55m$

The heat exchanger was fabricated as per the designed parameters and was installed before the fuel injector immediately after the fuel pump as shown in figure 3.10.

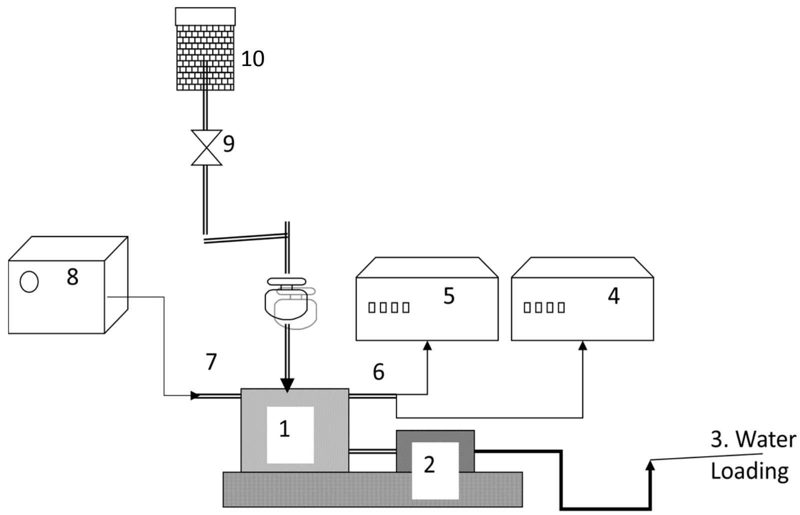
Figure 3.10: Heat Exchanger installed on the engine fuel supply system

3.3.7 Modified engine

Arrangements were made to find the temperature at downstream of heat exchanger. The experiment was carried out by fueling the engine with JSVO. The same test matrix and data sheets as were used to record the observations. The observation sheets are attached as annexure IX-XII.

3.3.8 Development of experimental test set up

The sub systems of the test rig are integrated and the schematic diagram of experimental test set up is shown below in figure 3.11.



1. Single cylinder 4-stroke diesel engine, 6 kW	6. Exhaust manifold
2. Alternator	7. Intake manifold
3. Hydraulic load	8. Air drum
4. Gas Analyzer	9. Control valve
5. Smokemeter	10. Fuel Tank

Figure 3.11:Schematic diagram of test set up for single cylinder IDI diesel engine

3.4 Measurement of Power

The engine was started and the outlet cooling water temperature was monitored. At the stage when the temperature became constant, the engine reached to the steady state condition. After attaining the steady state condition the voltage and current corresponding to all the operating points were recorded from the voltmeter & ammeter mounted on the control panel. The voltage (V) & current (I) was read by the voltmeter and ammeter connected with the hydraulic load. The product of voltage and current gave the actual load on engine alternator system.

Brake power output of the engine, P = $\frac{\textbf{voltage (V)x Current(I)}}{\textbf{1000x efficiency of generater}}$ kW

3.5 Measurement of temperature

The thermocouples of appropriate range were installed to capture the temperatures at cooling water inlet point, cooling water outlet point, exhaust manifold, lubricating oil sump, air inlet duct and the exhaust temperature at outlet from heat exchanger. The thermocouples were inserted into the flow line to a sufficient extent to capture the temperature correctly. The suitable fabrication work was done to accommodate the thermocouples at desired locations. The wires communicating the signals from thermocouples to the digital indicators were connected to the appropriate terminals rigidly. The digital indicators of appropriate range were used with these thermocouples to indicate the temperatures on data display board. (Figure 3.6)

The range and the type of temperature indicators installed on data display board are as follows:

I. Exhaust gas temperature indicator: Range 500 ^{0}C (J 100 type)
II. Exhaust gas temperature indicator at heat exchanger outlet: Range 500 ^{0}C (J 100 type)
III. Cooling water inlet temperature indicator: Range 100^0 C (Type PT 100)
IV. Cooling water outlet temperature indicator: Range 100^0 C (Type PT 100)
V. Lubricating oil temperature indicator: Range 100^0 C (Type PT 100)
VI. Inlet air temperature at inlet to Air tank indicator: Range 100^0 C (Type PT 100)

The display board was mounted on steel stand at appropriate height so that the readings can be taken comfortably.

3.6 Measurements of fuel injection angle

The measurement of crank advance angle at the moment fuel injection starts was done by making a 360^0 dial on white paper pasted on flywheel (See figure 3.12 &

figure 3.13). On the dial TDC & BDC points was marked by use of arrow pointer and magnetic dial test indicator on piston crown. After the points of TDC and BDC were identified the cylinder head was mounted. To pin point the fuel injection advance

angle the drop test was carried out. In this test the spring of the fuel pump was removed so that the drops starts coming out from fuel pump without any load. This corresponds to the start of the fuel injection. The crank advance angle corresponding to this point was reproduced on dial with the help of arrow pointer. The fuel injection advance angle could be altered by changing the length of cam follower. The change in length of follower was done by either screwing out or screwing in the nut after unlocking the lock nut. The hit and trial method was used to adjust the length of follower and after wards measuring the advance angle of fuel injection. After getting the required fuel injection angle the lock nut was tightened and the spring was put back at the position. This method was repeated for adjusting the fuel injection advance angle at 20^0 BTDC, 22^0 BTDC, 24^0 BTDC &26^0 BTDC. These were the fuel advance angles at which the engine was tested with all the combination of types of fuel and also the fuel injection pressures. By advancing the fuel injection point we increased the time delay, so the time available for fuel atomization was increased. The excessive advance angle increases the time delay on one side which is favorable for fuel atomization but also increases the tendency of knocking. Therefore a compromise between these effects was considered to decide the fuel injection advance angle.

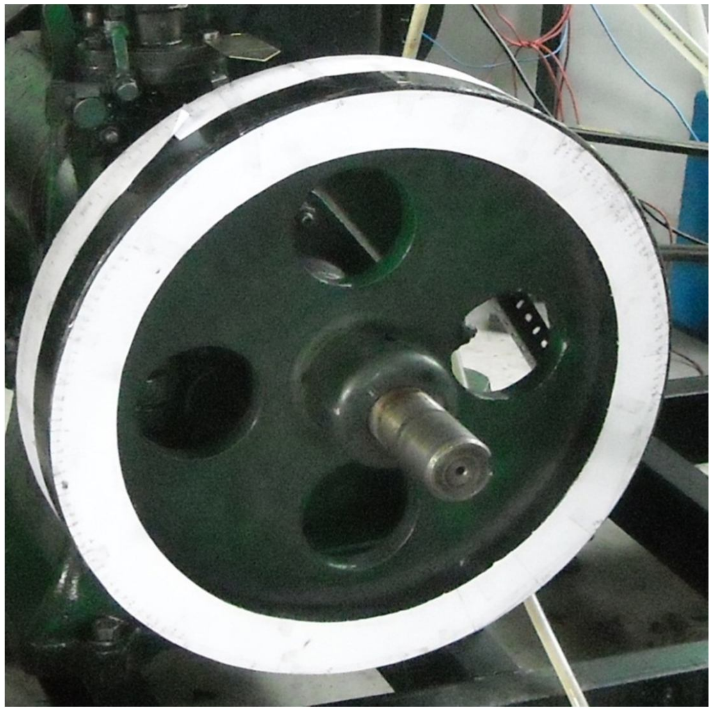


Figure 3.12: Arrangement for locating fuel injection point

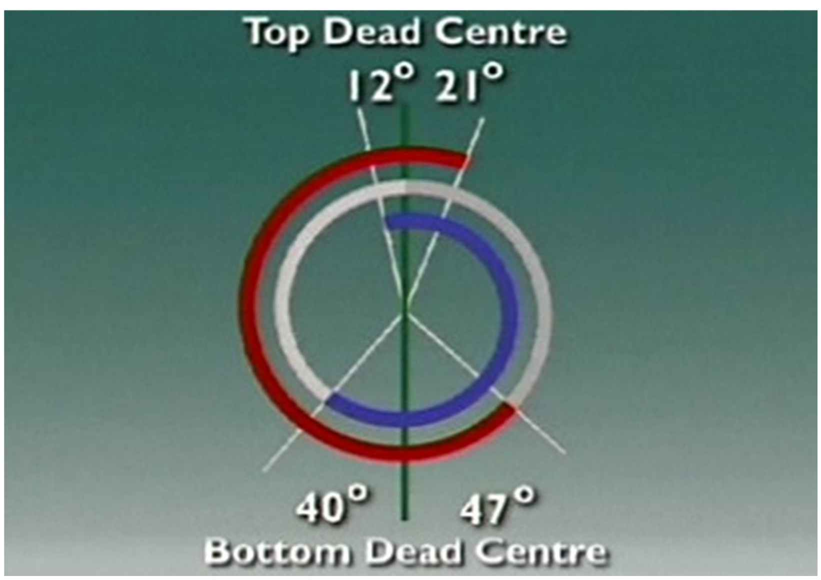


Figure 3.13: General fuel timing diagram

3.7 Measurement of fuel injection pressure

Fuel atomization characteristics largely depend upon the fuel injection pressure. The particle size of fuel injected into the combustion chamber at higher injection pressure becomes smaller. This fact was also validated from the results obtained from program of SMD equation. The calculation of the fuel particle size at different fuel injection pressures keeping all other parameters constants was done with the help of following empirical relation. By using this relation we found out the fuel injection pressure of JSVO at which the fuel particle size was reduced to reasonable extent.

$$\mathrm{SMD} = 3.08\nu^{0.335}(\sigma\rho_{\mathrm{fuel}})^{0.737}\rho_{\mathrm{air}}{}^{0.06}(\Delta P)^{-0.54}$$

Where SMD is sauter mean diameter which is the ratio of mean volume to the mean surface area of the fuel droplet. The empirical relation stated that the fuel particle size depends upon the viscosity of fuel, density of fuel, density of air, fuel injection pressure. The viscosity of JSVO was varied by changing the temperature. The viscosity of JSVO at 90^0 C was equal to the viscosity of diesel fuel. The injector pressure was changed with the help of injector tester (See figure 3.14 and figure 3.15). The spring tension of the fuel injector was altered by either screwing in or screwing out the nut supporting the spring. For adjustment the locking nut was opened and then the main nut was screwed in or out to alter the spring tension. By increasing the spring tension the injection pressure was increased and vice versa. This engine was designed to operate at fuel injection pressure of 175 kg f/cm^2 when petro diesel was used as test fuel. We operated the engine at elevated fuel injection pressure of 185 kgf/cm^2, 195 kgf/cm^2 and 205 kgf/cm^2 subsequently. The experiments were carried out at all the combinations of fuel injection angles i.e. 20^0 BTDC, 22^0 BTDC , 24^0 BTDC, 26^0 BTDC and fuel injection pressures i.e. 175 kgf/cm^2 , 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kg f/cm^2. The test was carried out with diesel fuel, JSVO fueled in existing and the JSVO fueled in modified engine at injector pressure.

Figure 3.14: Fuel injection pressure tester

Figure 3.15: Fuel injection pressure test in progress

3.8 Measurement of speed and air flow rate

The engine speed depends upon the load applied on the engine. The engine speed was measured with digital tachometer. The tachometer used was a digital tachometer capable of measuring the speed in RPM while engine is running. The tachometer's adapter was engaged with the groove of engine crank shaft firmly and the speed was shown on indicator in digits. For the air flow measurement the following arrangement was used.(figure 3.16)

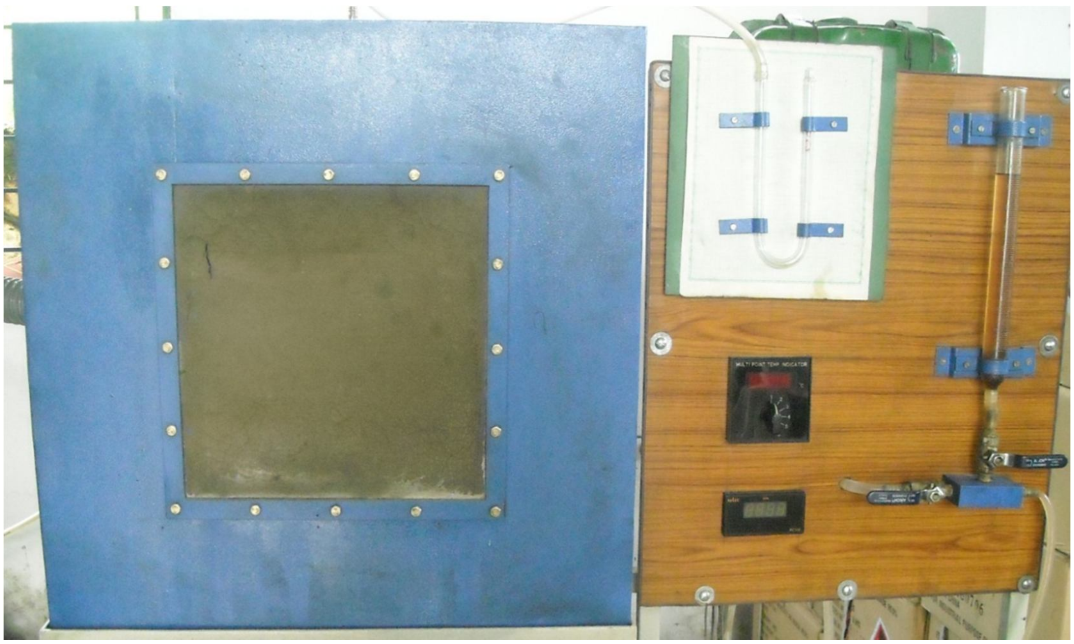


Figure 3.16: Apparatus to measure the rate of fuel and air consumption

It has an air chamber. The air chamber was equipped with an inlet orifice having coefficient of discharge 0.6. The pressure difference in the inlet manifold was measured by a normal U-tube manometer. Airflow was measured by taking the difference in heights of the water column in the two legs of the manometer and area of orifice of the surge tank. The following numerical formula was used for air flow calculation.

$$\text{Air Consumption } (m_a) = 3600\, C_d A (2g\rho_w h_w/\rho_a)^{1/2}\, \rho_a \text{ in kg/hr}$$

Where, C_d= coefficient of discharge=0.6

A = area of orifice in m^2

ρ_w = Density of water = 1000 kg/m^3

ρ_a = Density of air = 1.129 kg/m^3

h_w = Difference in heights of water columns in the two legs of the manometer in m

3.9 Engine Emission tests

The exhaust gas composition was measured using exhaust gas analyzer (AVL DIGAS-4000 model) shown in figure 3.17.

Figure 3.17: AVL smoke meter

It measures NO_x, CO_2, CO, HC and O_2 in the exhaust gases. The basic principle for Measurement of NO_x, CO_2, CO, and HC emissions is non-diffractive infrared radiation (NDIR) and an electrochemical methods for oxygen measurement. Measurement range and resolution for different gases by the exhaust gas analyzer used are given in table 4.5.

The opacity of the exhaust gas was measured by smoke meter (AVL-AUSTRIA,437) shown in figure. The exhaust opacity is defined as the extinction of light between the light source and receiver in a pipe filled with exhaust gases. The smoke opacity is usually measured to quantify the amount of particulate matter present in the engine exhaust. In the smoke meter, exhaust gases flow through a chamber having non-reflective inner surfaces. The light is passed through this chamber. The light source is an incandescent bulb with a temperature between 2800K and 3250K. The light travels through the chamber and falls on a photocell placed at the other end of chamber. The current delivered by the photocell is the linear function of the intensity received by it. When the light is passed through the chamber with exhaust smoke, the particulate matters present in the smoke, hinder the path of light. Thus only a fraction of light reaches the photoelectric cell and generates a voltage signal. The voltage signal is reciprocal to the opacity of the exhaust gases.

AVL-437 smoke meter measures the opacity of the polluted exhaust, in a particular diesel exhaust gases (in a measurement chamber of a defined measurement length). The effective length of the measurement chamber was 0.430 ± 0.005 m.

Table 3.8: capabilities of measurement AVL-437 smoke meter

Exhaust Gas	**Measurement Range**	**Resolution**
NO_x	0 – 4000 Vol. ppm	1 vol. ppm
CO	0 – 10 vol. %	0.01 vol. %
CO_2	0 -20 vol. %	0.1 vol. %
HC	0 – 20,000 ppm	1ppm
O_2	0 – 22 vol %	0.01 vol %

3.10 Engine test process

The sub systems required for measuring the different technical parameters of the engine performance and emission characteristics were integrated with the engine. The necessary attachments to measure the test parameters were organized. The fuel supply lines were primed and the engine was made ready for starting and data collection. The entire test process was carried out in three phases as described below:

3.10.1 Testing of existing engine with diesel fuel

The existing engine was run with diesel fuel under following test matrix. Initially the fuel injection advance angle was set at 20^0 BTDC using the drop test and the fuel injector pressure was adjusted at 175 kg f/cm^2 with the help of injector tester. The fuel tank was flushed with diesel and then the tank was filled with diesel fuel. The engine was started and the outlet cooling water temperature was monitored. As the temperature became stable the steady state of engine was achieved. The applied load on the engine was made zero by withdrawing the plates of loading system fully out and making the current flowing in load circuit to zero. Then the parameters like voltage, current, temperatures of all the 6 points, engine RPM, manometer reading and time for 50 cc fuel consumption were recorded. After recording these parameters the exhaust gas emission parameters like HC, NO_x, CO_2, CO were recorded by putting the probe of AVL gas analyzer into the exhaust pipe. For noting down the smoke (opacity) the exhaust gas was directed to AVL smoke meter and the opacity was recorded.

After completing the above data observation work, the load on the engine was increased to 1.84 kW by regulating the plates of loading system into the water up to the extent so that the current flowing through the load circuit is 6.25 ampere. The 6.25 amp current corresponds to 1.84 kW loads getting applied on the engine. The engine was allowed again to run for about 10-15 minutes to get stabilized. After this time duration all the process described in above paragraph were repeated and careful recording of technical parameters were done at new engine operating point. The load on the engine was increased to 3.62 kW, 5.62 kW and 7.35 by adjusting the current flowing through the load circuit to 12.5 ampere , 18.75 ampere and 25 ampere respectively and keeping the fuel injector triggering pressure and fuel injection advance angle unchanged from 175 kgf/cm^2 and 20^0 BTDC.

The engine was run to attain the steady state condition. After attaining the steady

state condition the observations were taken at all above mentioned operating points. The engine was stopped, the fuel injection angle was altered to 22^0 BTDC and the engine was run once again as mentioned in previous case and the data was recorded on observation table. This process was repeated for fuel advance angles of 24^0 and 26^0 BTDC.The entire process as mentioned above was repeated for fuel injection pressure 185 kgf/cm^2 ,195 kgf/cm^2 and 205 kgf/cm^2 . The data obtained in all the above combinations of engine operating parameters are shown in annexure-I to IV.

3.10.2 Testing of existing engine with JSVO fuel

The diesel from fuel tank, fuel supply pipelines and fuel filters was drained. The fuel supply system was flushed with fresh Jatropha oil and then the fuel tank was filled with JSVO. The air trapped in fuel pipe lines and the fuel filter was primed. The engine was run for sufficient time duration to ensure that the petro diesel fuel phase is over and the engine has started running with Jatropha as fuel. The entire process followed in phase I was repeated while engine running with JSVO as fuel. The observation data sheets is attached as annexure-V to VIII.

3.10.3 Testing of modified engine with JSVO fuel

In third phase of the experiment the modified engine was tested with modification for preheating the fuel. The entire process followed in phase I was repeated while modified engine running with JSVO as fuel. The observation data sheets are attached as annexure-IX to XII.

Table 3.9: Observation Table at different combinations of operating parameter for existing diesel engine fueled with Diesel

Test fuel	**Fuel injection angle**	**Engine Load (kW)**	**Fuel injection pressure (kgf/cm^2)**			
			175	**185**	**195**	**205**
Diesel Fueled in Existing Engine	20^0 BTDC	0	Annex. I	Annex. I	Annex. I	Annex. I
		1.84	Annex. I	Annex. I	Annex. I	Annex. I
		3.68	Annex. I	Annex. I	Annex. I	Annex. I
		5.52	Annex. I	Annex. I	Annex. I	Annex. I
		7.35	Annex. I	Annex. I	Annex. I	Annex. I
	22^0 BTDC	0	Annex. II	Annex. II	Annex. II	Annex. II
		1.84	Annex. II	Annex. II	Annex. II	Annex. II
		3.68	Annex. II	Annex. II	Annex. II	Annex. II
		5.52	Annex. II	Annex. II	Annex. II	Annex. II
		7.35	Annex. II	Annex. II	Annex. II	Annex. II
	24^0 BTDC	0	Annex. III	Annex. III	Annex. III	Annex. III
		1.84	Annex. III	Annex. III	Annex. III	Annex. III
		3.68	Annex. III	Annex. III	Annex. III	Annex. III
		5.52	Annex. III	Annex. III	Annex. III	Annex. III
		7.35	Annex. III	Annex. III	Annex. III	Annex. III
	26^0 BTDC	0	Annex. IV	Annex. IV	Annex. IV	Annex. IV
		1.84	Annex. IV	Annex. IV	Annex. IV	Annex. IV
		3.68	Annex. IV	Annex. IV	Annex. IV	Annex. IV
		5.52	Annex. IV	Annex. IV	Annex. IV	Annex. IV
		7.35	Annex. IV	Annex. IV	Annex. IV	Annex. IV

Table 3.10: Observation Table at different combinations of operating parameter for existing diesel engine fueled with JSVO

Test fuel	**Fuel injection angle**	**Engine Load (kW)**	**Fuel injection pressure (kgf/cm^2)**			
			175	**185**	**195**	**205**
JSVO in Existing Engine	20^0 BTDC	0	Annex. V	Annex. V	Annex. V	Annex. V
		1.84	Annex. V	Annex. V	Annex. V	Annex. V
		3.68	Annex. V	Annex. V	Annex. V	Annex. V
		5.52	Annex. V	Annex. V	Annex. V	Annex. V
		7.35	Annex. V	Annex. V	Annex. V	Annex. V
	22^0 BTDC	0	Annex. VI	Annex. VI	Annex. VI	Annex. VI
		1.84	Annex. VI	Annex. VI	Annex. VI	Annex. VI
		3.68	Annex. VI	Annex. VI	Annex. VI	Annex. VI
		5.52	Annex. VI	Annex. VI	Annex. VI	Annex. VI
		7.35	Annex. VI	Annex. VI	Annex. VI	Annex. VI
	24^0 BTDC	0	Annex. VII	Annex. VII	Annex. VII	Annex. VII
		1.84	Annex. VII	Annex. VII	Annex. VII	Annex. VII
		3.68	Annex. VII	Annex. VII	Annex. VII	Annex. VII
		5.52	Annex. VII	Annex. VII	Annex. VII	Annex. VII
		7.35	Annex. VII	Annex. VII	Annex. VII	Annex. VII
	26^0 BTDC	0	Annex. VIII	Annex. VIII	Annex. VIII	Annex. VIII
		1.84	Annex. VIII	Annex. VIII	Annex. VIII	Annex. VIII
		3.68	Annex. VIII	Annex. VIII	Annex. VIII	Annex. VIII
		5.52	Annex. VIII	Annex. VIII	Annex. VIII	Annex. VIII
		7.35	Annex. VIII	Annex. VIII	Annex. VIII	Annex. VIII

Table 3.11: Observation table at different combinations of operating parameter for modified diesel engine fueled with preheated JSVO

Test fuel	Fuel injection angle	Engine Load (kW)	Fuel injection pressure (kgf/cm^2)			
			175	**185**	**195**	**205**
JSVO Fueled in Modified Engine	20^0 BTDC	0	Annex. XIX	Annex.XIX	Annex. XIX	Annex. XIX
		1.84	Annex. XIX	Annex.XIX	Annex. XIX	Annex. XIX
		3.68	Annex. XIX	Annex.XIX	Annex. XIX	Annex. XIX
		5.52	Annex. XIX	Annex.XIX	Annex. XIX	Annex. XIX
		7.35	Annex. XIX	Annex.XIX	Annex. XIX	Annex. XIX
	22^0 BTDC	0	Annex. X	Annex. X	Annex. X	Annex. X
		1.84	Annex. X	Annex. X	Annex. X	Annex. X
		3.68	Annex. X	Annex. X	Annex. X	Annex. X
		5.52	Annex. X	Annex. X	Annex. X	Annex. X
		7.35	Annex. X	Annex. X	Annex. X	Annex. X
	24^0 BTDC	0	Annex. XI	Annex. XI	Annex. XI	Annex. XI
		1.84	Annex. XI	Annex. XI	Annex. XI	Annex. XI
		3.68	Annex. XI	Annex. XI	Annex. XI	Annex. XI
		5.52	Annex. XI	Annex. XI	Annex. XI	Annex. XI
		7.35	Annex. XI	Annex. XI	Annex. XI	Annex. XI
	26^0 BTDC	0	Annex. XII	Annex. XII	Annex. XII	Annex. XII
		1.84	Annex. XII	Annex. XII	Annex. XII	Annex. XII
		3.68	Annex. XII	Annex. XII	Annex. XII	Annex. XII
		5.52	Annex. XII	Annex. XII	Annex. XII	Annex. XII
		7.35	Annex. XII	Annex. XII	Annex. XII	Annex. XII

3.11 Theoretical Considerations

After recording the test parameters following calculations are done:

Measure of rate of fuel consumption

Let time taken in consuming 50 cm^3 fuel= t second

Fuel consumed in one second= 50 cm^3/ t

= 50 $x10^{-6}$ x density of fuel / t

(a) For diesel fuel: = 50$x10^{-6}$ x 817/t kg/s

(b) For JSVO: = 50$x10^{-6}$ x 910/t kg/s

Measurement of brake power output

$$\text{Power (P)} = \frac{\text{Voltage x Current}}{\text{1000 x efficiency of generater}}\ kW$$

Measurement of break specific fuel consumption (BSFC)

$$\text{BSFC} = \frac{\text{rate of fuel consumption}}{\text{Brake power output}}$$

$$= \frac{50\text{x}10^{-6}\text{ x density of fuel / time}}{\frac{\text{voltage x current}}{\text{1000 x generater efficiency}}}$$

(a) For diesel fuel:

$$\text{BSFC} = \frac{50\text{x}10^{-6}\text{ x 817 / time}}{\frac{\text{voltage x current}}{\text{1000 x generater efficiency}}}$$

$$= \frac{\text{40.85 x 3600 x generater efficiency}}{\text{voltage(V)x current(I)x time(t)}}\ \text{kg/kWh}$$

$$= \frac{1.47060\text{x}10^{5}\text{x generater efficiency}}{\text{voltage(V) xcurrent(I)Xtime(t)}}\ \text{kg/kWh}$$

(a) For JSVO:

$$\text{BSFC} = \frac{50\text{x}10^{-6}\text{ x }910\text{ / time}}{\frac{\text{voltage x current}}{1000\text{x generater efficiency}}}$$

$$= \frac{45.5\text{x}3600\text{xgenerater efficiency}}{\text{voltage(V)x current(I)x time(t)}}\ \text{kg/kWh}$$

$$= \frac{1.638\text{x}10^{5}\text{x generater efficiency}}{\text{voltage(V) x current(I)x time(t)}}\ \text{kg/kWh}$$

Measurement of brake thermal efficiency (BTE)

$$\text{BTE} = \frac{\text{brake power output x 100}}{\text{rate of fuel consumption x calorific Value}}\ \%$$

(a) For diesel fuel:

$$\text{BTE} = \frac{\text{voltage x current x time}}{1.7157\text{ x }10^{4}\text{xgenerater efficiency}}\ \%$$

(Calorific Value of diesel= 42000kJ/kg)

(b) For JSVO:

$$\text{BTE} = \frac{\text{voltage x current x time}}{1.7745\text{ x }10^{4}\text{x generater efficiency}}\ \%$$

(Calorific Value of JSVO= 39000kJ/kg)

The bar chart of BTE against 0 kW, 1.84kW, 3.68kW, 5.52kW, and 7.35kW load for Diesel fuel and JSVO fueled in existing engine and also for modified engine fueled with JSVO was drawn. The experiment was carried out at advance angles of fuel injection 20^0BTDC, 22^0 BTDC, 24^0 BTDC and 26^0 BTDC. The above stated charts were plotted for the fuel injection pressures 175 kgf/ cm^2 , 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kgf/cm^2.

CHAPTER 4 RESULTS AND DISCUSSION

4.1 Introduction

The observations of experiments to find the fuel properties and engine performance were processed for analysis and results. The importance of some of the important fuel properties on engine performance and emission characteristics is discussed in this chapter. The engine performance parameters BTE, BSFC and also emission parameters CO, CO_2, NO_x, HC and smoke were discussed. These parameters were based upon the experimental results obtained with diesel and JSVO test fuel in existing engine as well as modified engine.

The high viscosity of fuel which was one of the important fuel propertiesaffecting the engine performance was reduced by fuel preheating using the exhaust gas as heat source. The effect of preheating the fuel to 90^0 C resulted in engine performance improvement. The engine output at these suggested operating parameters for JSVO fuel improved the BTE of engine from 30.62% to 34.76% in comparison with 32.96% of diesel test fuel at designed operating parameter as recommended by manufacturer for this engine. The appreciable improvements in emission characteristics were shown. Some of the problems like injector choking, deposits build up, lubricating oil dilution and piston ring sticking reported by researchers as a result of endurance test was not investigated in this work. The investigation of engine performance after incorporating these modifications on these reported problems can be considered as future scope of investigation.

4.2 Effect of temperature on viscosity of Jatropha oil and diesel

The high viscosity was an important factor resulting low brake thermal efficiency of engine due to poor fuel atomization. To lower the viscosity of JSVO to an extent of the viscosity of diesel at room temperature, the JSVO was to be heated to 90^0 C. The exhaust gas was utilized as a heat source. The counter flow heat exchanger was designed to transfer the heat from exhaust gas to JSVO fuel. After the fabrication of heat exchanger, the heat exchanger was installed with necessary modification to direct the exhaust gas through it. The heated fuel was supplied to the fuel pump and from there it was supplied to combustion chamber. Fuel while traveling from fuel pump to combustion chamber looses the heat and temperature gets reduced. This difficulty was overcome by installing the heat exchanger after fuel pump and utilizing the heat of exhaust gas.

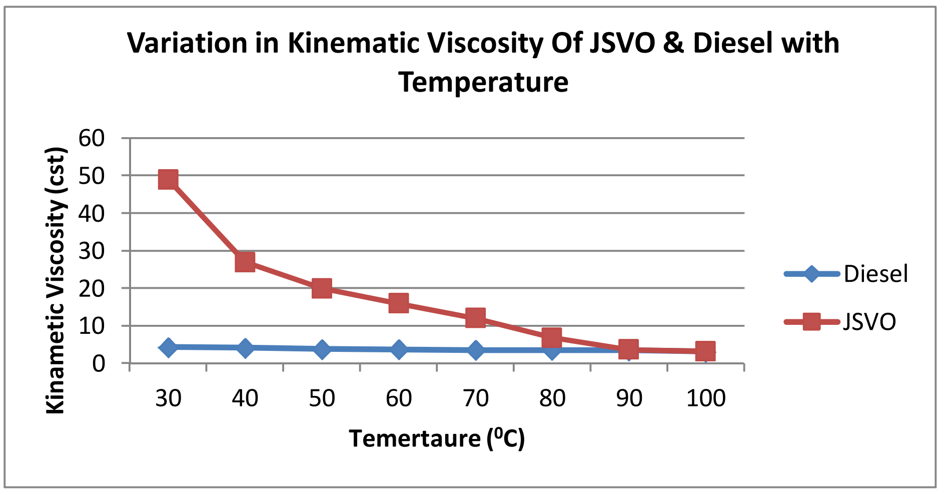


Figure 4.1: Variation of viscosity with temperature

4.3 Engine Performance

4.3.1 Brake Thermal Efficiency

The brake thermal efficiency indicates the net output of the engine available on engine shaft for getting utilized to operate machines. The study of brake thermal efficiency of existing engine fueled with diesel, existing engine fueled with JSVO and modified engine fueled with JSVO was carried out while engine operating under following variable parameters and fuel injection conditions, (1) when the engine was operated with diesel fuel at different fuel injection points, (2) when the engine was operated on different fuel injection pressures, and (3) when the heat exchanger was installed on the engine for preheating the fuel.

It was observed that the brake thermal efficiency (BTE) of engine with diesel fuel was higher than the BTE of engine with JSVO fuel at manufacturer's recommended operating parameter. The calorific value of JSVO as found in laboratory was close to the calorific value of diesel. The experiments were carried out under same environment, so the lower BTE was an indication that the fuel combustion efficiency of JSVO was inferior to the diesel. The combustion efficiency largely depends upon the fuel atomization quality. The fuel atomization efficiency depends upon the viscosity of fuel, fuel injection pressure and the time available for atomization process. The viscosity of JSVO was about 10-12 times the viscosity of diesel. Therefore the JSVO was required to be preheated before injecting into the combustion chamber. The heat exchanger was used for preheating the fuel. The exhaust gas was used as heat source. Apart from preheating two more factors i.e. fuel injection advance angle and the fuel injection pressure governing combustion efficiency were also required to be tested to find the optimum angle and injection pressure which can improve the BTE.

4.3.1.1 Effect of varying fuel injection point at constant fuel injection pressure

The fuel injection point was taken for modification because the time duration between the point of fuel injection and the start of combustion process was felt be one factor which was not sufficient to allow the fuel particles to get atomized properly [1]. This time duration which is known as time delay is in two parts, one is physical delay where the atomization process takes place and the other is the chemical delay where the chemical reaction and temperature rise takes place to bring the air fuel mixture at the point of established combustion process. The increase in time delay period provides more time available for this pre-combustion process. Haldar S K [2] stated that the JSVO demands ignition timing to be more for better combustion and hence improves the engine performance and emission characteristics. The advancing the fuel injection point by 1^0 results in increase in delay period of approximately 0.08 second because this engine is self governed operating at 1000 rpm irrespective of engine load. This resulted in better fuel atomization characteristics and enhanced the combustion efficiency. The increase in delay period was achieved by advancing the fuel injection point. In present engine the design point of fuel injection was 20^0 BTDC for diesel fuel. The fuel injection point used for the experiment was 20^0 BTDC, 22^0 BTDC, 24^0 BTDC and 26^0 BTDC. Advancing the fuel injection point was limited because increased delay period increases the possibility of engine knocking and also the ldecreased efficiency [2]. The possible reason for knocking and reduction in engine performance is due to the fact that advanced ignition timing results in an increased peak cylinder pressure and temperature as the combustion occurs earlier in the cycle and more heat is released before and around the top dead centre.

It was also found that the BTE of existing and modified engine fueled with diesel and JSVO increased as the engine load was increased for the entire range of operation. The comparative BTE at different load conditions as obtained from experiments in the cases of diesel, JSVO fueled in existing engine and also in modified engine have been shown in figures from 4.2 to figure 4.5.

The analysis of the results obtained from the experiments revealed that the BTE at a constant fuel injection pressure decreased continuously from 20^0BTDC to 26^0 BTDC in case of diesel fuel (figure 4.6). The same experiment was carried out at injection pressures 175 kgf/cm^{2,} 185 kgf/cm^2, 195 kgf/cm^2 and 205 kgf/cm^2. The same trend of performance deterioration was observed at all the fuel injection pressures in the case of diesel engine. Also it was observed from the graph that the angle of fuel injection to give optimum output with diesel fuel was 20^0 BTDC, this is quite in accordance with the manufacturer recommendation for this engine.

When the existing engine was fueled with JSVO the BTE of engine at 20^0 BTDC fuel injection point and 175 kgf/ cm^2 fuel injection pressure was observed 30.5% which was much less than the efficiency with diesel (32.96%). This indicates that the existing engine when fueled with JSVO did not perform as efficiently as with diesel fuel. In contrary to diesel fuel, it was observed that the BTE of JSVO gradually increased as the fuel injection was advanced to 22^0 BTDC & 24^0 BTDC where as the BTE started decreasing as the fuel injection point was advanced further to 26^0 BTDC. The above findings indicates that the desirable fuel injection advance angle at a constant fuel injection pressure to give better efficiency for JSVO was 24^0 BTDC (figure 4.6).

The BTE was compared when diesel fuel and JSVO was used in existing engine and in modified engine at 7.35 kW engine load by varying the fuel injection point to 20^0 , 22^0, 24^0 and 26^0 at constant fuel injection pressure of 175 kgf/cm^{2,} 185 kgf/cm^2, 195 kgf/cm^2 and 205 kgf/cm^2 . It was found that the BTE of diesel decreased as the advance angle was increased. The BTE of existing as well as modified engine fueled with JSVO was increased up to 24^0 BTDC. It was observed that the BTE started decreasing when advance angle was increased beyond 24^0 BTDC to 26^0 BTDC. The efficiency of modified engine at all the points was improved in comparison with existing engine fueled with JSVO. This may be because the JSVO was injected in modified engine at elevated temperature which reduced the viscosity of JSVO and therefore improved the combustion characteristics (figure 4.6).

4.3.1.2 Effect of varying fuel injection pressure at constant fuel injection point:

The effect of varying fuel injection pressure at a constant fuel injection point was investigated in the experiment at different pressures 175 kgf/cm^2, 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kgf/cm^2. The investigation was carried out with objective that the fuel atomization characteristics which indirectly depend upon fuel droplets size injected into the combustion chamber gets improved as the fuel droplets size reduces at high fuel injection pressures.

The following expression shows the variation of fuel particle size in terms of SMD at elevated pressure. The sauter mean diameter (SMD) which is the ratio of mean volume to the mean surface area of the fuel droplet and has an important role in defining the fuel atomization characteristics. Smaller SMD results better fuel atomization and ultimately the fuel combustion efficiency. [10]

$$\mathrm{SMD} = 3.08\nu^{0.335}(\sigma\rho_{\mathrm{fuel}})^{0.737}\rho_{\mathrm{air}}^{0.06}(\Delta P)^{-0.54}$$

Where , ν is kinematic viscosity in m^2 /s , σ surface tension of fuel in kg/cm^2 , ρ is density in kg/m^3 and Δ P is pressure across the injector In Pa.

Following values of fuel properties were taken for SMD analysis.

Table 4.1: Test fuel properties

Properties	Diesel at room temp	JSVO at room temp	JSVO at 90^0 C
Kinematic viscosity (m^2/s) x10^{-6}	4.3	48.7	3.68
Surface tension (kg/s^2)	0.28	0.42	0.32
Density of fuel (kg/m^3)	817	910	910

The expression for SMD was programmed in MATLAB version February 2011, to find the effect of injection pressure, viscosity and surface tension on sauter mean diameter of fuel particle. The three sets of data i.e. first for diesel fuel at room temperature, second for JSVO at room temperature and third for JSVO at 90^0 C was used in program and the results have been calculated. The density of air is taken as 1.22kg/m^3. The graph was plotted for SMD against fuel injection pressure. The numerical values of results obtained have been shown in table 4.2. The SMD of diesel at room temperature and injection pressure 175 kgf/cm^2 was found 3.82 x 10^{-5} m. The SMD of diesel was reduced continuously from 3.82 x 10^{-5} m to 3.50 x 10^{-5} m as the pressure was increased from175 kgf/cm^2 to 205 kgf/cm^2.

Table 4.2: Estimated values obtained from SMD program

Fuel injection pressure (kgf/cm^2)	SMD (m) $x10^{-5}$ Diesel at room temperature	SMD (m) $x10^{-5}$ JSVO at room temperature	SMD (m) $x10^{-5}$ JSVO at 90^0 C temperature
175	3.82	12.56	4.33
185	3.68	12.19	4.20
195	3.59	11.84	4.08
205	3.50	11.53	3.97

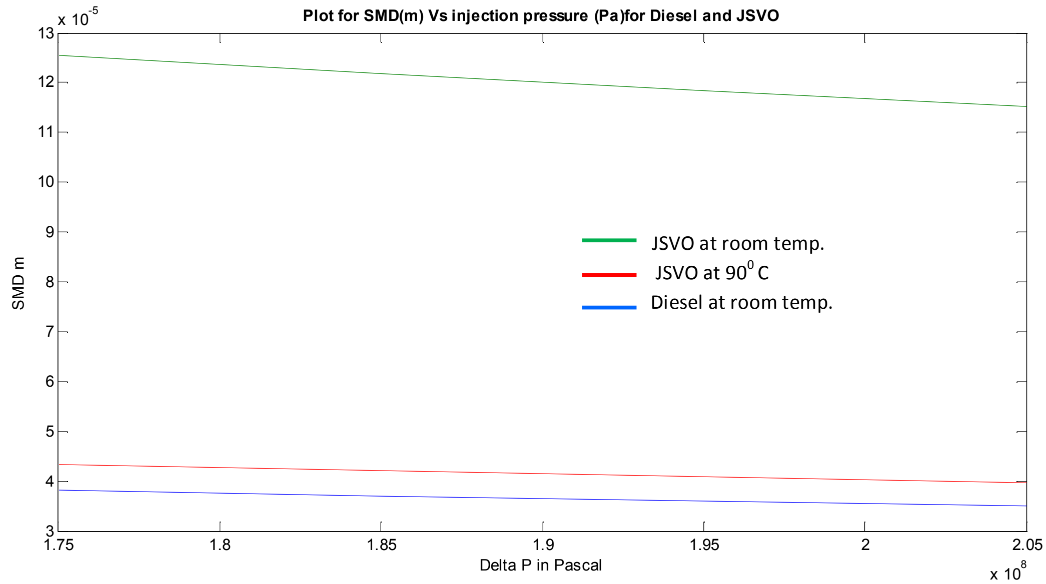
The SMD of JSVO at room temperature and 175 kgf/cm^2 was obtained 12.56 x 10^{-5} m. The SMD of JSVO was reduced linearly from 12.56 x 10^{-5} m to 11.53 x 10^{-5} m as the fuel injection pressure was increased from 175 kgf/cm^2 to 205 kgf/cm^2 . The SMD of JSVO at 90^0 C and 175 kgf/cm^2 was obtained to be 4.33 x 10^{-5} m , this value was reduced to 3.97 x 10^{-5} m as the pressure was increased from 175 kgf/cm^2 to 205 kgf/cm^2 .

As the JSVO was heated to 90^0 C the SMD at 175 kgf/cm^2 was reduced from 12.56 x 10^{-5} m to 4.33 x 10^{-5} m. This is substantial reduction in SMD by preheating. The preheating the fuel has lowered the viscosity from 48.7 x 10^{-6} m^2 /s to 3.68 x 10^{-6} m^2 /s as obtained in laboratory , also the surface tension was reduced from 0.42 kg /s^2 to 0.32 kg /s^2. The comparison of SMD for JSVO and diesel at room temperature showed that the difference between them was very high (8.74 x 10^{-5} m). This difference was reduced to a great extent (0.51 x 10^{-5} m) as the JSVO was preheated. The appreciable improvement is due to the fact that the viscosity and surface tension were reduced to very high extent. By injecting the JSVO at recommended injection pressure (195 kgf/cm^2), the difference was further reduced to 0.26 x 10^{-5} m. That above results revealed that SMD of JSVO at

90 ^{0} C and 195 kgf/cm^{2} injection pressure was almost equal to the SMD of diesel fuel.

The almost equal values of SMDs for preheated JSVO and diesel must have brought the atomization characteristics of these test fuels at same level. This fact is quite in accordance to the BTE of engine obtained from preheated JSVO operating at recommended parameters. The BTE (34.76%) of engine with preheated JSVO was found higher than the BTE of engine with diesel fuel (32.96%), this difference may be due to the fact that the fuel injection timing was also advanced by 4^{0} degrees of crank rotation.. Advancing the injection point by 4^{0} results in an additional 0.32 second time available for fuel atomization process; also the increase in efficiency can be attributed to the higher energy supplied to the engine by an equal volume of JSVO than the diesel, this is due to the fact that the density of JSVO is higher than the density of diesel which results higher energy supply even though the calorific value (kJ/kg) of JSVO is slightly lower than the calorific value of diesel fuel.

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%Program for SMD (metre) Vs fuel injection pressure (Pa)        %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

clear all;

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                    For JSVO                                   %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

nue_Jsvo1 = 48.7*10e-6; % m2/s
sigma_Jsvo1 = 0.042; % kg/s2
nue_Jsvo2 = 3.68*10e-6
sigma_Jsvo2 = 0.032
rho_Jsvo = 910; % Unit Kg/m3
rho_air = 1.22; % kg/m3
delta_P = input('Value of delta_P in pa is:');
%delta_P= 175*10e5:10*10e5:205*10e5; % Pa
SMD_Jsvo1   =   3.08   *   nue_Jsvo1^0.335   .*   (sigma_Jsvo1   *
rho_Jsvo)^0.737 * rho_air^0.06 .* (delta_P).^(-0.54);
SMD_Jsvo2   =   3.08   *   nue_Jsvo2^0.335   .*   (sigma_Jsvo2   *
rho_Jsvo)^0.737 * rho_air^0.06 .* (delta_P).^(-0.54);
disp('Value  of  SMD_Jsvo1  in  metre  is:'),format  long,SMD_Jsvo1,
single(SMD_Jsvo1)
disp('Value  of  SMD_Jsvo2  in  metre  is:'),format  long,SMD_Jsvo2,
single(SMD_Jsvo2)

%plot (delta_P,SMD_Jsvo)

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%                          For Diesel Fuel                      %
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

nue_diesel = 4.3*10e-6; % m2/s
sigma_diesel = 0.028; % kg/s2
rho_diesel = 817; % Unit Kg/m3
%delta_P = input('Value of delta_P in pa is:');
%delta_P= 175*10e5:10*10e5:205*10e5; % Pa
SMD_diesel   =   3.08   *   nue_diesel^0.335   .*   (sigma_diesel   *
rho_diesel)^0.737 * rho_air^0.06 .* (delta_P).^(-0.54);
disp('Value  of  SMD_diesel  in  metre  is:'),format  long,SMD_diesel,
single(SMD_diesel)
%figure(2)
%plot(delta_P,SMD_diesel,delta_P,SMD_Jsvo1,delta_P,SMD_Jsvo2)
%title('\bfPlot  for  SMD(m)  Vs  injection  pressure  (Pa)for  Diesel
and JSVO')
%xlabel('Delta P in Pascal');
%ylabel('SMD m')
```



Value of delta_P in pa is:175*10e5

Value of SMD_Jsvo1 in metre is:

SMD_Jsvo1 = 1.255920632652550e-004

ans = 1.2559207e-004

Value of SMD_Jsvo2 in metre is:

SMD_Jsvo2 = 4.326730245171972e-005

ans = 4.3267304e-005

Value of SMD_diesel in metre is:

SMD_diesel = 3.815634943854152e-005

ans = 3.8156348e-005

nue_Jsvo2 = 3.680000000000001e-005

sigma_Jsvo2 = 0.03200000000000

Value of delta_P in pa is:185*10e5

Value of SMD_Jsvo1 in metre is:

SMD_Jsvo1 = 1.218793159788331e-004

ans = 1.2187932e-004

Value of SMD_Jsvo2 in metre is:

SMD_Jsvo2 = 4.198823627833311e-005

ans = 4.1988238e-005

Value of SMD_diesel in metre is:

SMD_diesel = 3.702837304294310e-005

ans = 3.7028374e-005

nue_Jsvo2 = 3.680000000000001e-005

sigma_Jsvo2 = 0.03200000000000

Value of delta_P in pa is:195*10e5

Value of SMD_Jsvo1 in metre is:

SMD_Jsvo1 = 1.184633613406750e-004

ans = 1.1846336e-004

Value of SMD_Jsvo2 in metre is:

SMD_Jsvo2 = 4.081141714941745e-005

ans = 4.0811417e-005

Value of SMD_diesel in metre is:

SMD_diesel = 3.599056575280821e-005

ans = 3.5990564e-005

nue_Jsvo2 = 3.680000000000001e-005

sigma_Jsvo2 = 0.03200000000000

Value of delta_P in pa is:205*10e5

Value of SMD_Jsvo1 in metre is:

SMD_Jsvo1 = 1.153069956226953e-004

ans = 1.1530700e-004

Value of SMD_Jsvo2 in metre is:

SMD_Jsvo2 = 3.972402813280712e-005

ans = 3.9724029e-005

Value of SMD_diesel in metre is:

SMD_diesel = 3.503162463694563e-005

ans = 3.5031626e-005

4.3.1.3 Effect of fuel pre heating at constant fuel injection point & constant fuel injection pressure

The BTE of JSVO when fueled in existing engine and in other case when fueled in modified engine were compared at all the operating points. At initial point where optimum BTE of diesel fuel was obtained 32.96%, the efficiency of existing engine fueled with JSVO was observed to be 30.62%. After preheating the JSVO by installing the heat exchanger, the efficiency of modified engine increased to 31.04%. This shows that the fuel preheating improved the efficiency by 0.42%, which is about 1.37% of 30.62%. The similar improvements were obtained at all the operating points. (Figure 4.2 to figure 4.5)

4.3.1.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

It was observed from figure 4.6 that the BTE of existing engine when fueled with JSVO increased as the fuel injection angle increased from 20^0 BTDC to 22^0 BTDC and 24^0 BTDC but it started decreasing when the angle was advanced further to 26^0 BTDC. The same trend was observed with JSVO when fueled in modified engine with fuel preheating with an improvement of BTE at corresponding operating point in modified engine. This improvement can be attributed to the preheating done in modified engine which reduced the viscosity of JSVO and thereby improved the combustion efficiency. It was also seen that the maximum improvement took place at 24^0 BTDC, a downward trend was observed at 26^0 BTDC. Therefore an important inference can be drawn that the maximum BTE in the modified engine fueled with JSVO is at 24^0 BTDC.

The BTE at all pressures but fixed fuel injection points was compared. The increase in fuel injection pressure resulted that the BTE at corresponding points were improved up to 195 kgf/cm^2 which started reducing on increasing the pressure to 205 kgf/cm^2. This ultimately shows that the BTE was improved up to 24^0 BTDC with increasing the injection pressure and also by preheating the fuel.

It can be concluded that the BTE of engine with JSVO fuel at recommended operating point i.e. 24^0 BTDC and the 195kgf/cm^2 with fuel preheating was 34.76% in comparison with 32.96% when diesel fuel was used at 20 ^{0} BTDC fuel injection point and 175 kgf/cm^2 fuel injection pressure. It shows that the modifications not only made the engine compatible with JSVO but also the efficiency was improved from 32.96% to 34.76% which is about 5.5 % of present diesel efficiency.

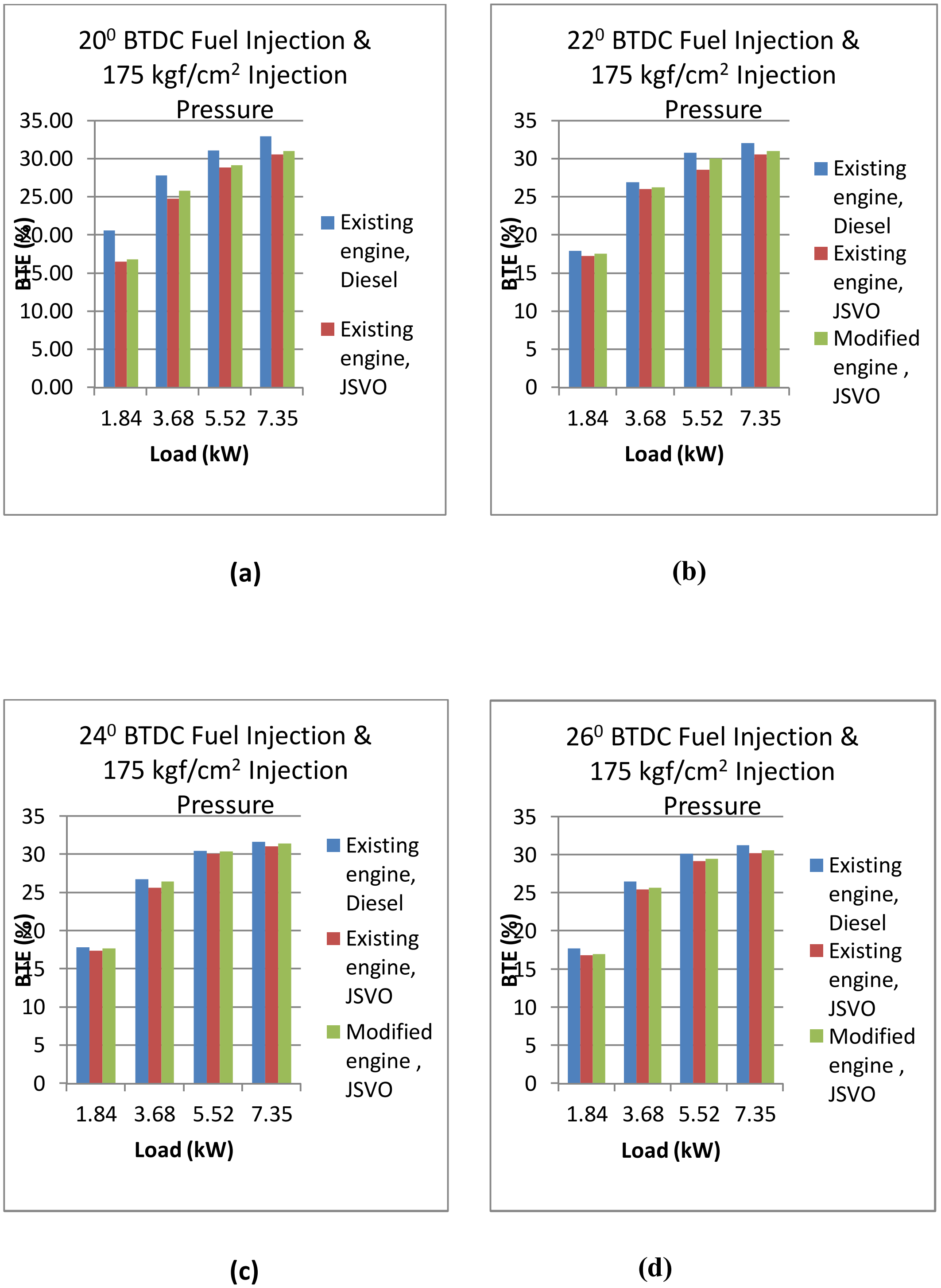
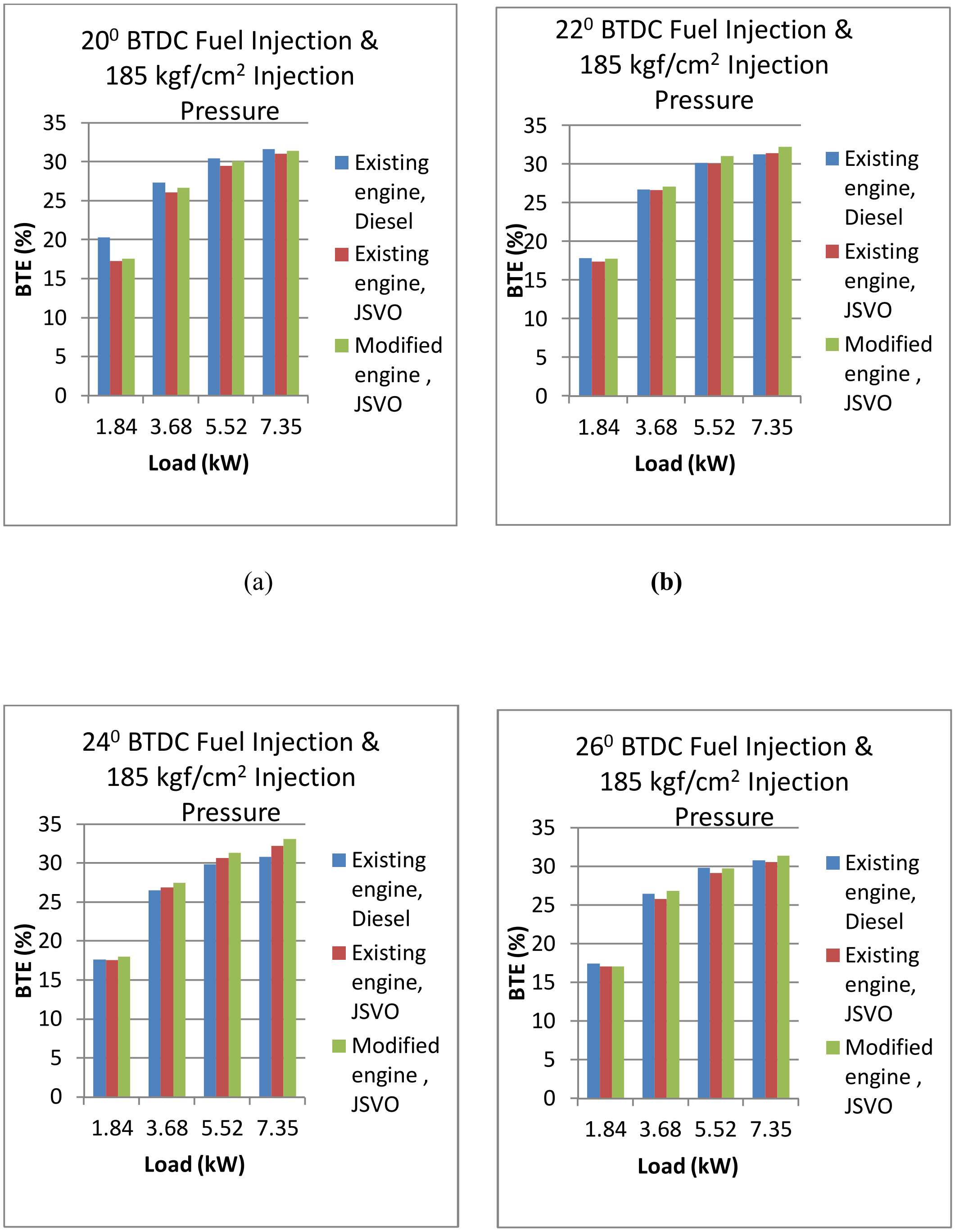


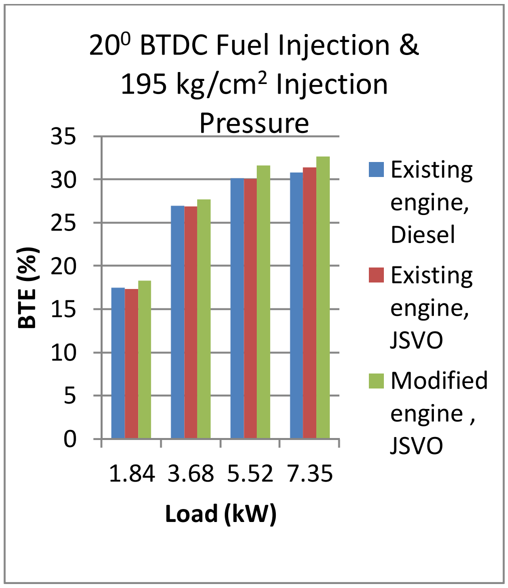
Figure 4.2: BTE vs. Engine load



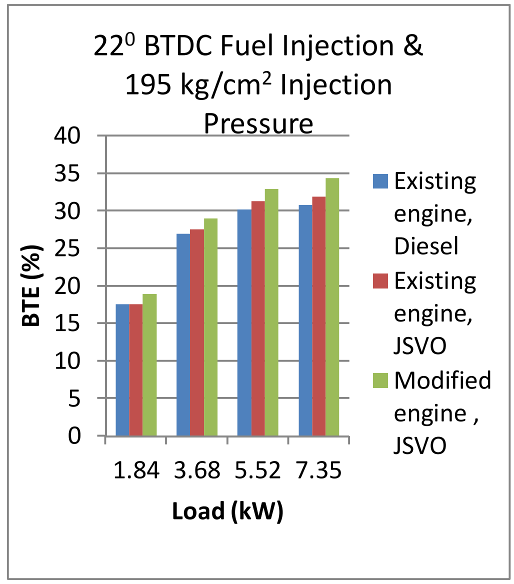
(a) (b)

(c) (d)

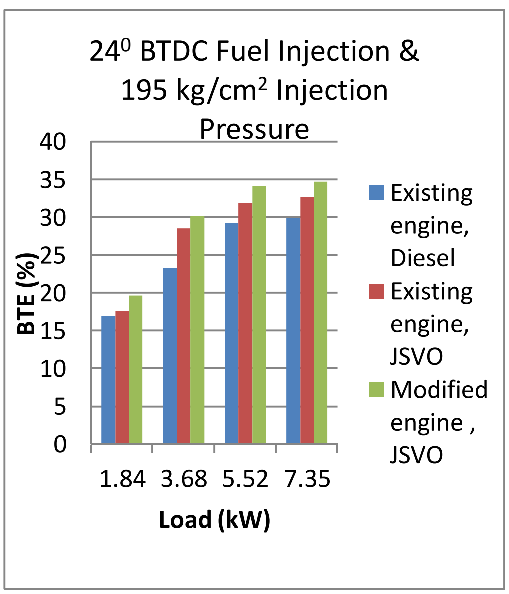
Figure 4.3: BTE vs. Engine Load



(a)



(b)



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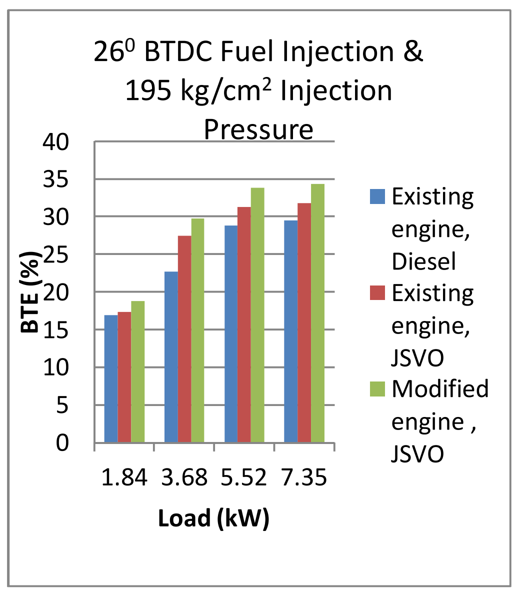


(d)

Figure 4.4: BTE vs. Engine Load

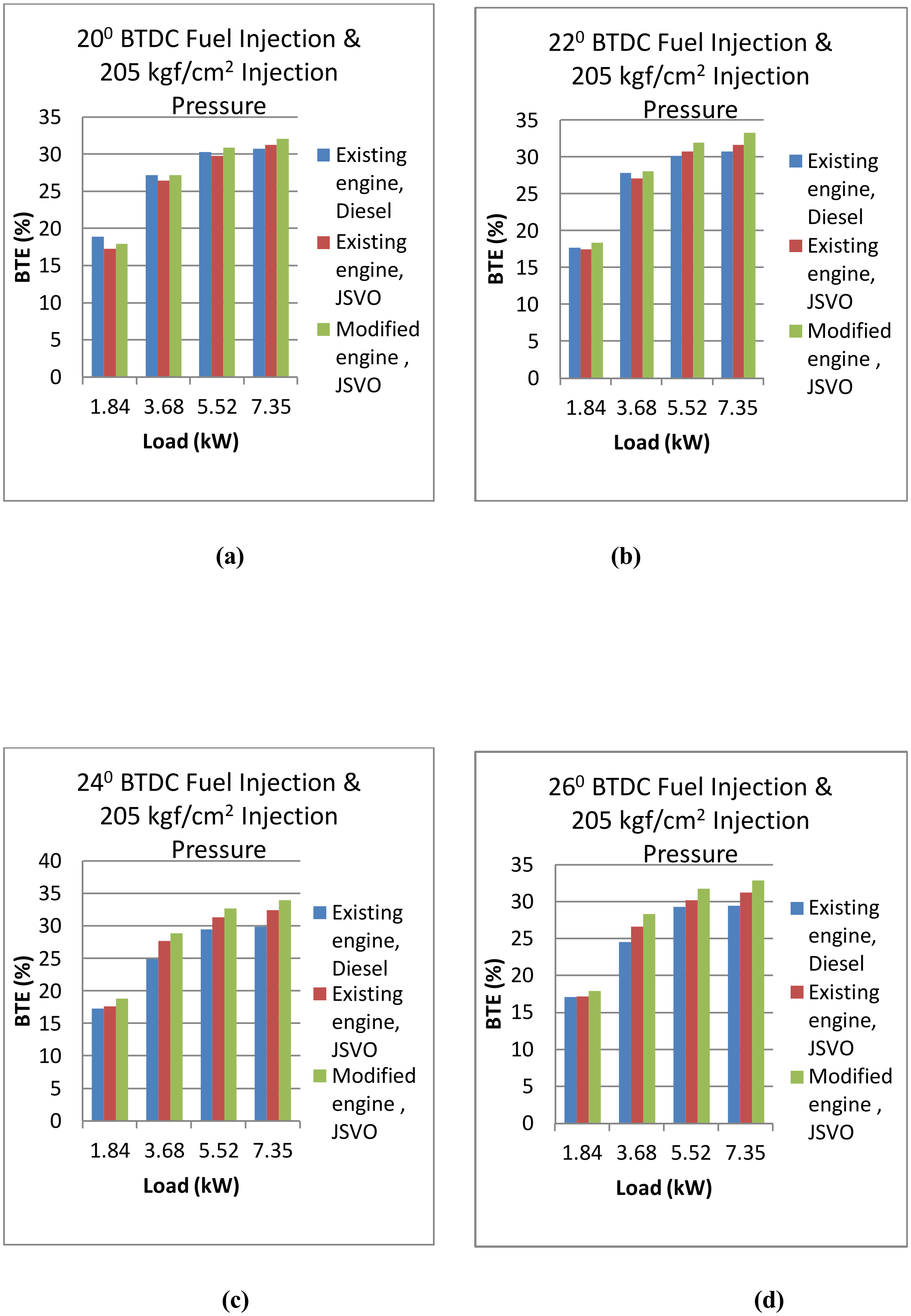
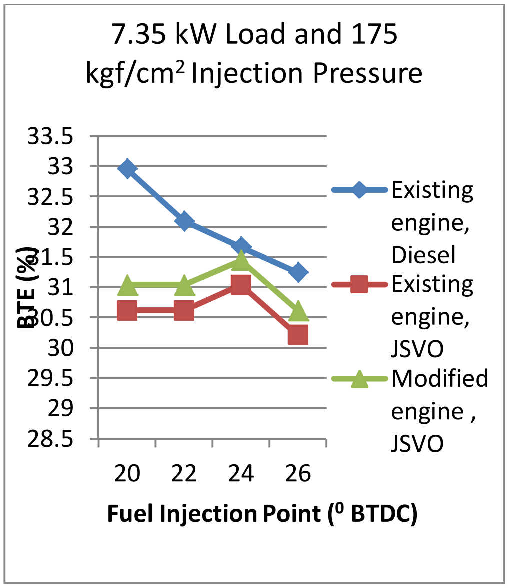
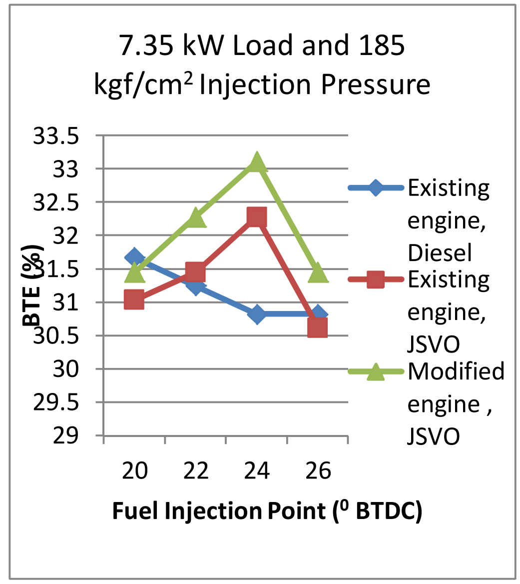


Figure 4.5 : BTE vs. Engine Load



(a)



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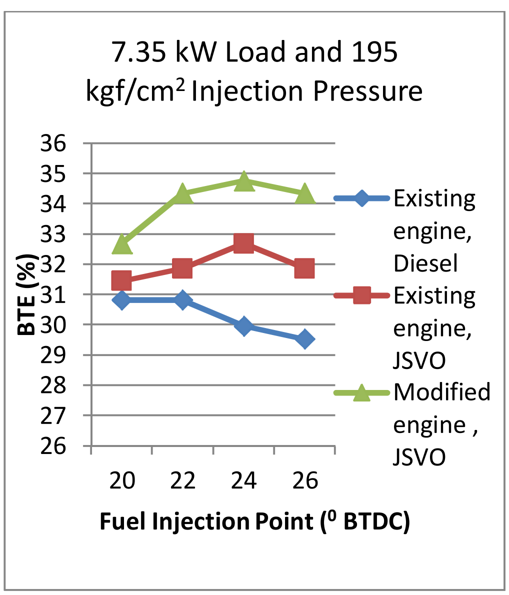

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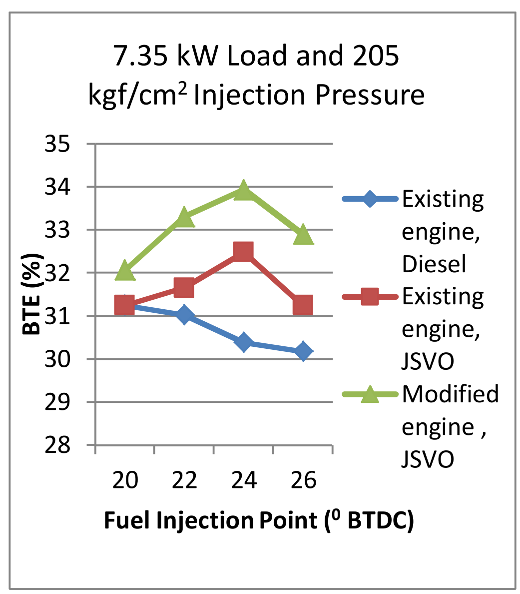
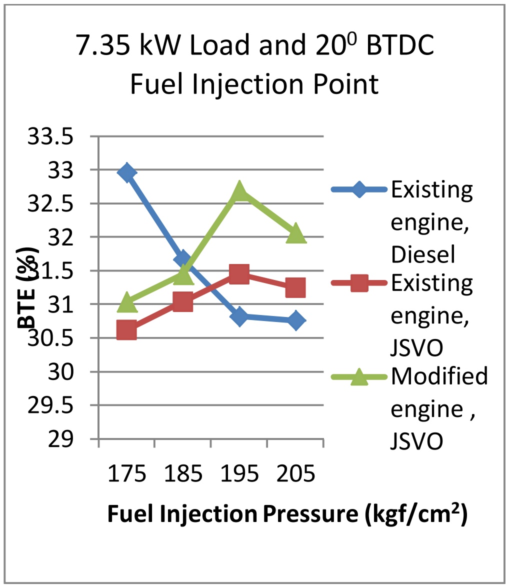
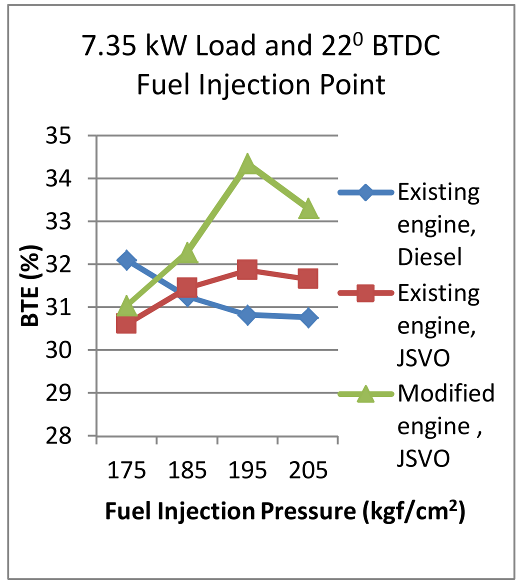


(d)

Figure 4.6: BTE vs. Advance angle of fuel injection



(a)



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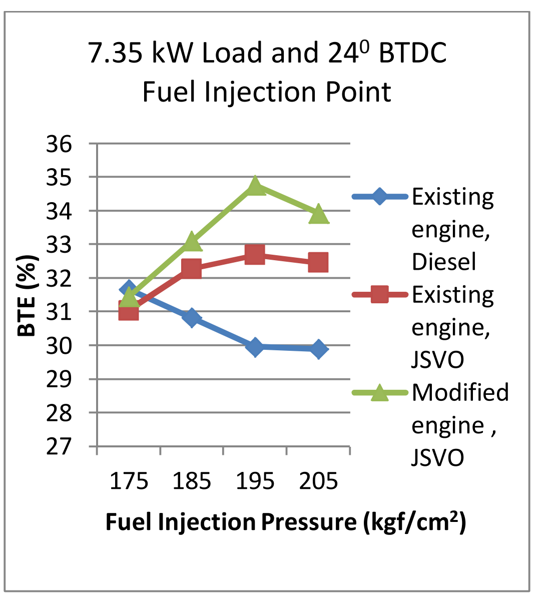

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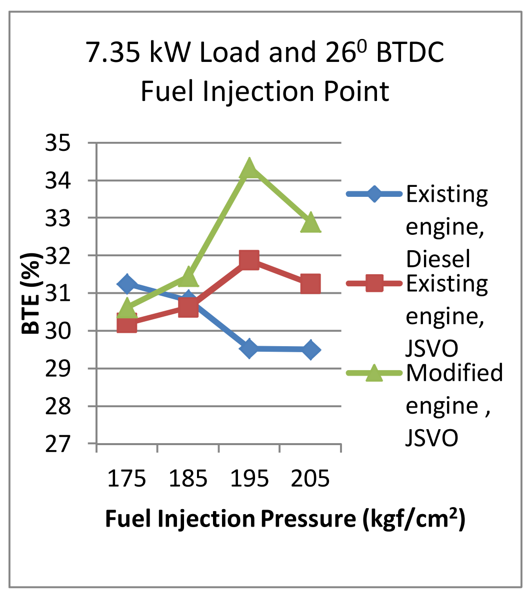


(d)

Figure 4.7: BTE vs. Injection pressure

4.3.2 Brake specific fuel consumption

The amount of fuel consumed for generating unit brake power also called as brake specific fuel consumption is directly connected to the economy of engine utilization, and therefore it was one of the most important parameter to be evaluated under different operating conditions. The observations were made about fuel consumption by monitoring the time duration of 50 cc fuel under previously defined engine operating conditions considered during BTE evaluation. Based upon the results obtained, it was observed that the BSFC of the engine with diesel fuel when used in existing engine at 20^0 BTDC was 0.260 kg/kWh which is optimum at 20^0 BTDC and 175 kgf/cm^2. These values of fuel injection timing and injection pressures were the recommended operating parameter by the manufacturer for his engine for rated power output. The small variation in BSFC was owing to the fact that the engine was run under different geographical location and climatic condition (temperature and pressure) however the engine was tested for repeatability of data. The precision of the instrument has been validated before carrying out any further experiments on the engine. After the engine was validated for the results the BSFC of the engine when fueled with JSVO was found to be 0.301 kg/kWh, which was higher than the BSFC of diesel engine. This was quite obvious due to low calorific value, viscosity, low cetane index, higher surface tension resulting in an all together different combustion characteristics. The trend of variation of BSFC was plotted against the varying (increasing from 0 kW to 7.35 kW, which was rated load for best performance of existing engine for the diesel fuel) load, varying fuel injection point and fuel injection pressure. The trends of variation of BSFC at all possible operating points with increasing load is shown in figure 4.8 to fig. 4.11. The graph shows that the BSFC increased as the engine load was increased from 0 kW i.e. no load condition to maximum rated load of 7.35 kW. These graphs show the comparison of BSFC of engine with diesel fuel, existing engine fueled with JSVO and modified engine (incorporating a heat exchanger) fueled with JSVO.

The specific fuel consumption was observed to be high in case of engine run with JSVO. This indicated that the heat released by fuel combustion was lower in this case which subsequently reduced the brake power output. Because of low power output the BSFC became high. To improve the BSFC, the efficiency of fuel combustion phenomena was required to be enhanced. There are many possible methods of improvement such as design improvement in combustion chamber, design change of piston crown, the design change of inlet and exhaust valve, fuel injection system design improvements etc. Among these possible methods most favorable changes were fuel injection system design changes. The parameters taken in consideration for design change were fuel injection advance angle, fuel injection pressure and method to reduce the viscosity of JSVO to improve the fuel atomization characteristics.

4.3.2.1 Effect of varying fuel injection point at constant fuel injection pressure

As mentioned above, the BSFC of existing engine run with JSVO was higher in comparison with engine fueled with diesel at 20^0 BTDC and 175 kgf /cm^2. From figure 4.12, it was seen that the BSFC of engine with diesel fuel started increasing whenever the operating point was changed from the manufacturer recommended operating point for this engine, the same fact was also reported by Basinger et al in their findings [1]. The BSFC decreased in both modified engine and existing engine when fueled with JSVO as the fuel injection angle was advanced from 20^0 BTDC to 22^0 BTDC and 24^0 BTDC but the BSFC started rising on further increases in advance angle of fuel injection to 26^0 BTDC. This was in correlation with the BTE trends with change in fuel injection point (figure 4.6). It was observed very clearly that the optimum performance of the engine corresponds to 24^0 BTDC with evident decrease in BTE with increasing and decreasing the advance angle of fuel injection point i.e. for 22^0 BTDC and 26^0 BTDC. This was convincing also because the brake thermal efficiency deteriorated at 26^0 BTDC in comparison with 24^0 BTDC which ultimately increased the fuel consumption. These results gave information that the engine efficiency was improved on increasing the advance angle up to 24^0 BTDC. The BSFC of modified engine on comparison with BSFC of existing engine indicated that the trend of change was similar at all the operating points but the BSFC of modified engine was less than the BSFC of existing engine at same combination of advance angle of fuel injection and fuel injection pressure. This shows that the effect of preheating the fuel improved the BSFC substantially.

4.3.2.2 Effect of varying fuel injection pressure at constant fuel injection point

The effect of fuel injection pressure keeping the angle constant has been plotted in figure 4.13.The engine was tested at fuel injection pressures of 175 kgf/cm^2 , 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kgf/cm^2 at all considered fuel injection points (20^0 BTDC, 22^0 BTDC , 24^0 BTDC and 26^0 BTDC) . In the previous section the effect of preheating was discussed and it showed an improvement in BTE and BSFC due to fuel pre heating. This was quite understandable as the preheating the fuel ultimately results in better atomization of fuel in the cylinder when it is injected through the injector due to lower viscosity at higher temperature. The similar atomization characteristics could also be achieved by higher injection pressures and therefore an improvement in BTE and BSFC. The engine BSFC with diesel fuel got increased when the fuel injection pressure was increased from 175 kgf/cm^2 which was the rated fuel injection pressure for this engine. The changes in BSFC at different fuel injection pressure while keeping the advance angle of fuel injection point unchanged indicated that the BSFC reduced at elevated fuel injection pressure for the engine with JSVO. This was also indicated that the trend of changes in BSFC was same both in existing engine as well modified engine when they were fueled with JSVO. However increase in pressure beyond 195kgf/cm^2 was not advisable because the BTE reduced and BSFC increased beyond this pressure (Figure 4.13). This increase in pressure beyond above said pressure was not advisable because of other associated problem of poor combustion efficiency due to low momentum of smaller fuel particles released in combustion chamber as mentioned in the case of BTE discussion.

4.3.2.3 Effect of fuel preheating at constant fuel injection point & fuel injection pressure

The change in BSFC at 7.35 kW engine load at different IVOP (injector valve opening pressure) and fuel injection point was compared for diesel engine, existing engine and modified engine run on JSVO. The BSFC of JSVO without preheating the fuel was higher than the BSFC of JSVO when it was fueled after preheating. It was found with operating point 20^0 BTDC and $175 kgf/cm^2$, the BSFC in both the cases were 0.301 kg/kWh and 0.297 kg/kWh respectively. This shows that the BSFC improved by preheating the fuel which is about 1.33% in this particular case.

The comparison of existing and modified engine at a particular operating point has shown that the BSFC of modified engine was less than the existing engine with JSVO, which was only because of fuel preheating. This indicated that the contribution of heat exchanger to preheat the JSVO improved the efficiency of engine.

4.3.2.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

It was observed from figure 5.13 that the BSFC of existing engine when fueled with JSVO decreased as the fuel injection angle was increased from 20^0 BTDC to 22^0 BTDC and 24^0 BTDC but it started increasing when the angle was advanced further. The same trend was observed with JSVO when fueled in modified engine but there was improvement in BSFC at corresponding operating point in modified and existing engines. It was also seen that the maximum improvement had taken place at 24^0 BTDC, when angle was increased to 26^0 BTDC the values of BSFC at all the corresponding points were increased.

The comparison of the values of BSFC at all but fixed fuel injection point when the fuel injection pressure was increased show that the BSFC at corresponding points was reduced. This ultimately shows that the BSFC was improved up to 24^0 BTDC by increasing the injection pressure and also by preheating of the fuel.

Therefore it can be concluded that the performance of JSVO at recommended operating point i.e. 24^0 BTDC and the 195kgf/cm^2 with preheating the fuel by installing the heat exchanger gave BSFC 0.266 kg/kWh in comparison with 0.260 kg/kWh of diesel at design operating point $20^{\,0}$ BTDC fuel injection point and 175 kgf/cm^2 fuel injection pressure. It shows that the modifications made the engine compatible with JSVO with significant improvement in BSFC of JSVO from 0.301 kg/kWh (at 20^0BTDC , 175 kgf/cm^2 and without preheating) to 0.266 kg/kWh (at 24^0 BTDC, 195 kgf/cm^2 and pre heating the fuel) which was an improvement of about 1.2 % with respect to 0.301 kg/kWh.

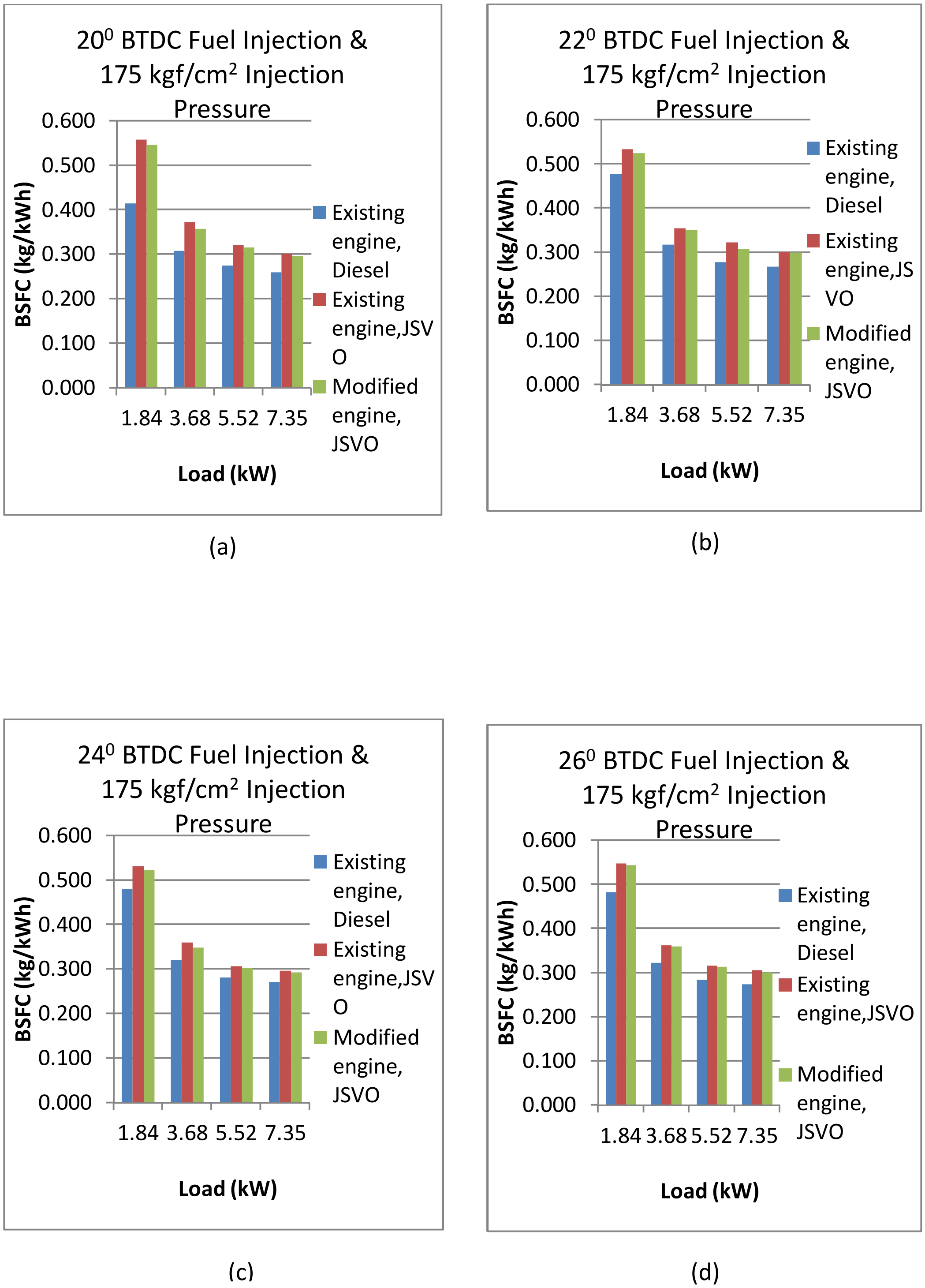


Figure 4.8: BSFC vs. Engine Load

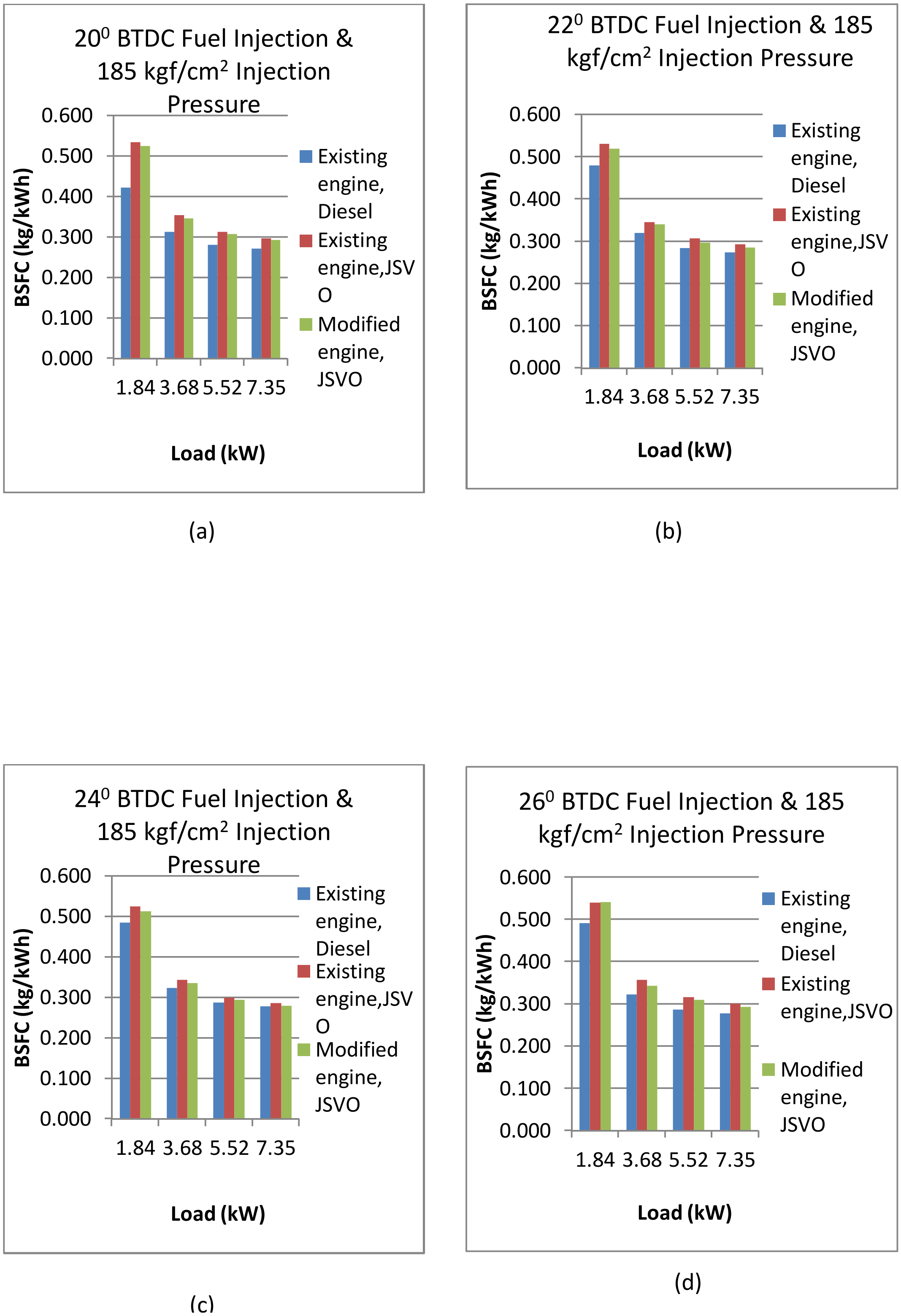


Figure 4.9: BSFC vs. Engine Load

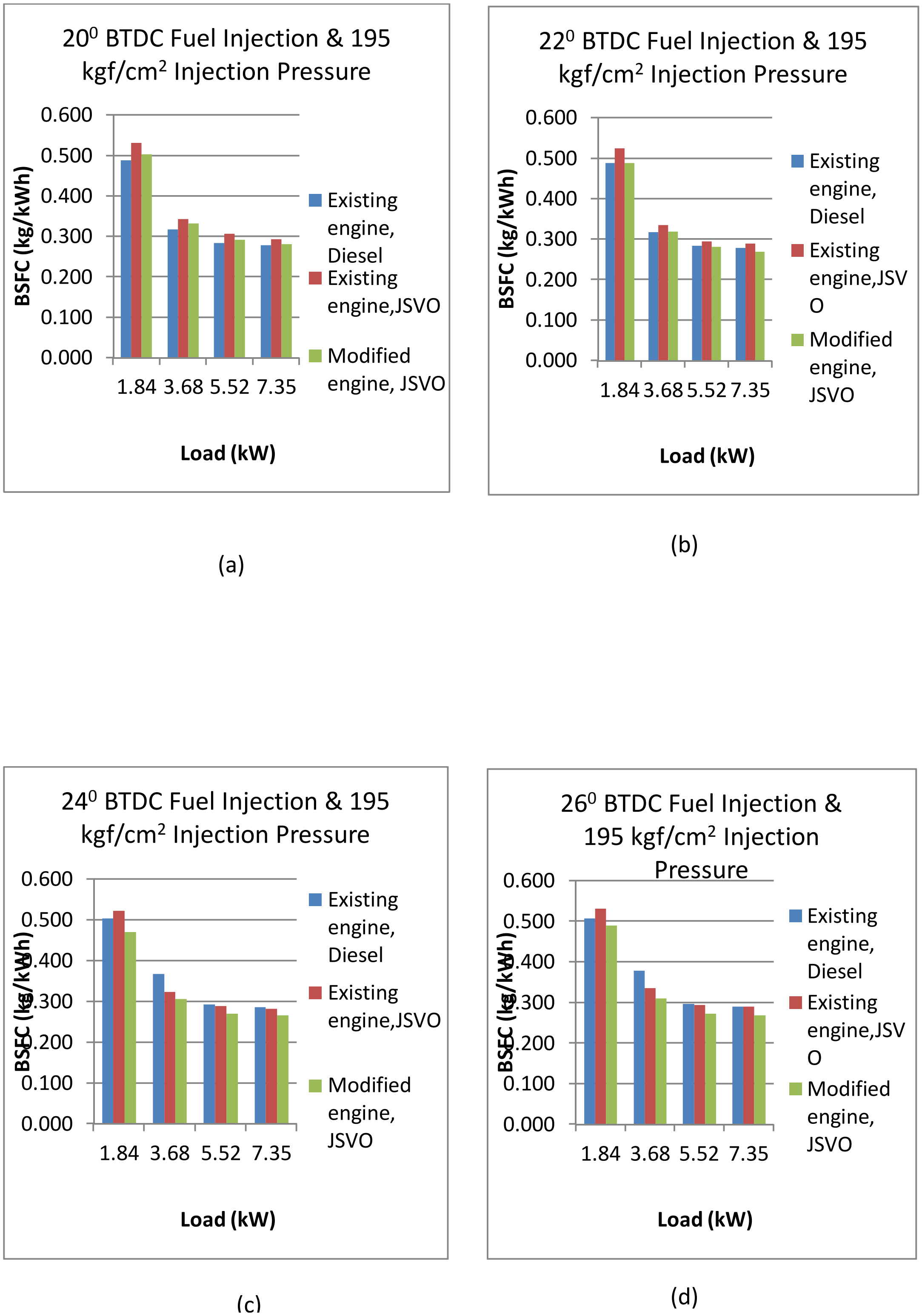


Figure 4.10: BSFC vs. Engine Load

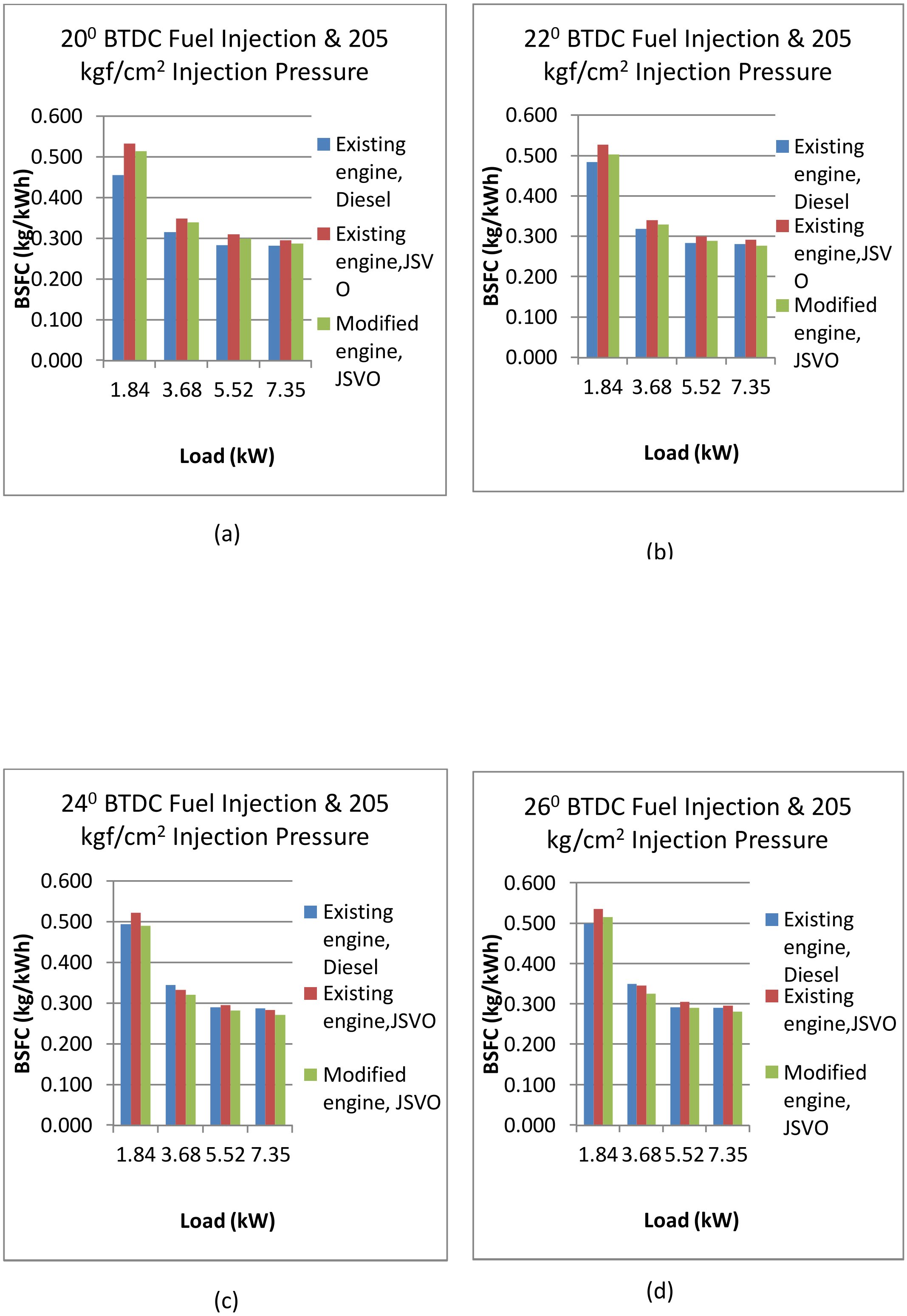


Figure 4.11: BSFC vs. Engine Load

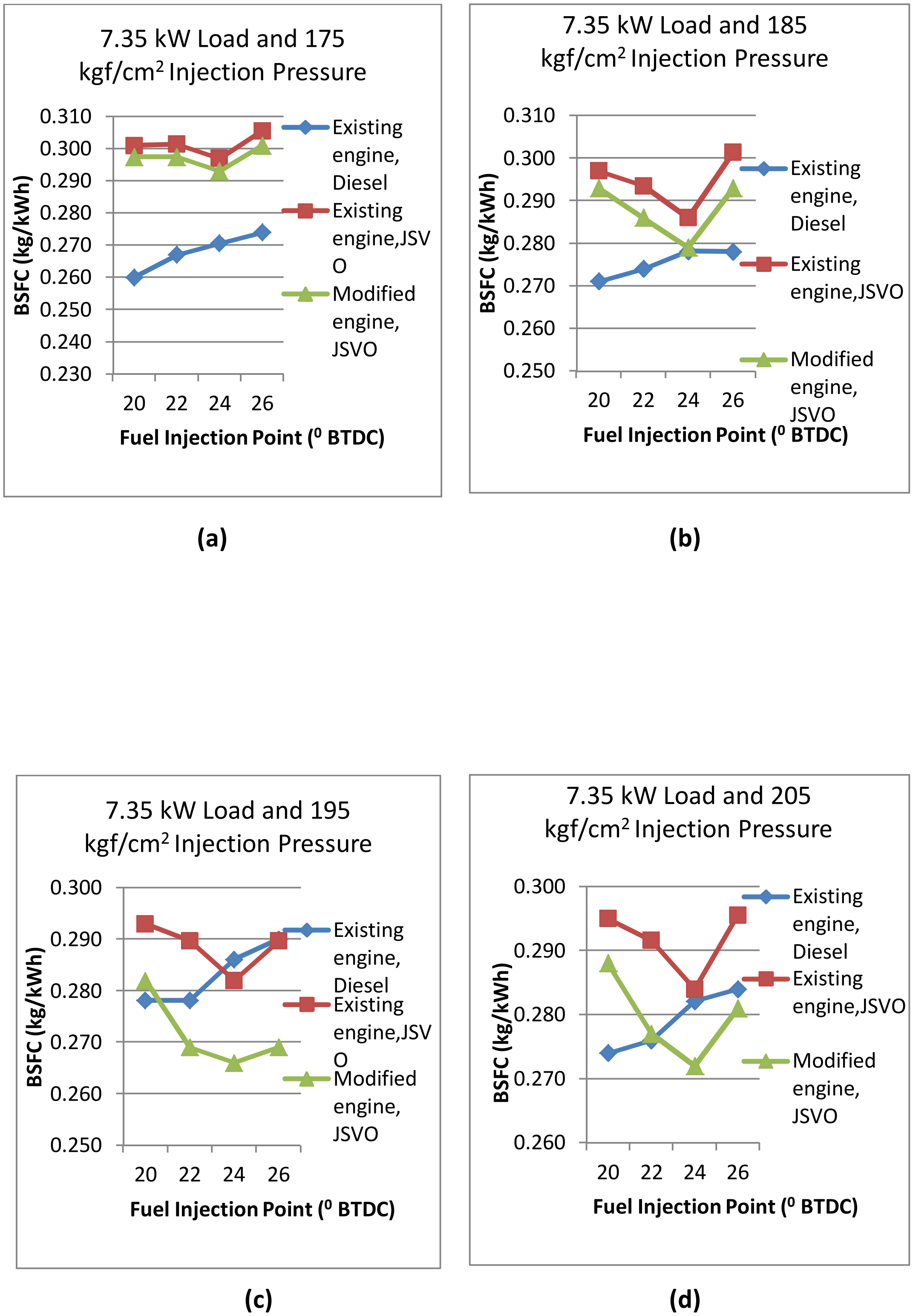


Figure 4.12: BSFC vs. Advance angle of fuel injection

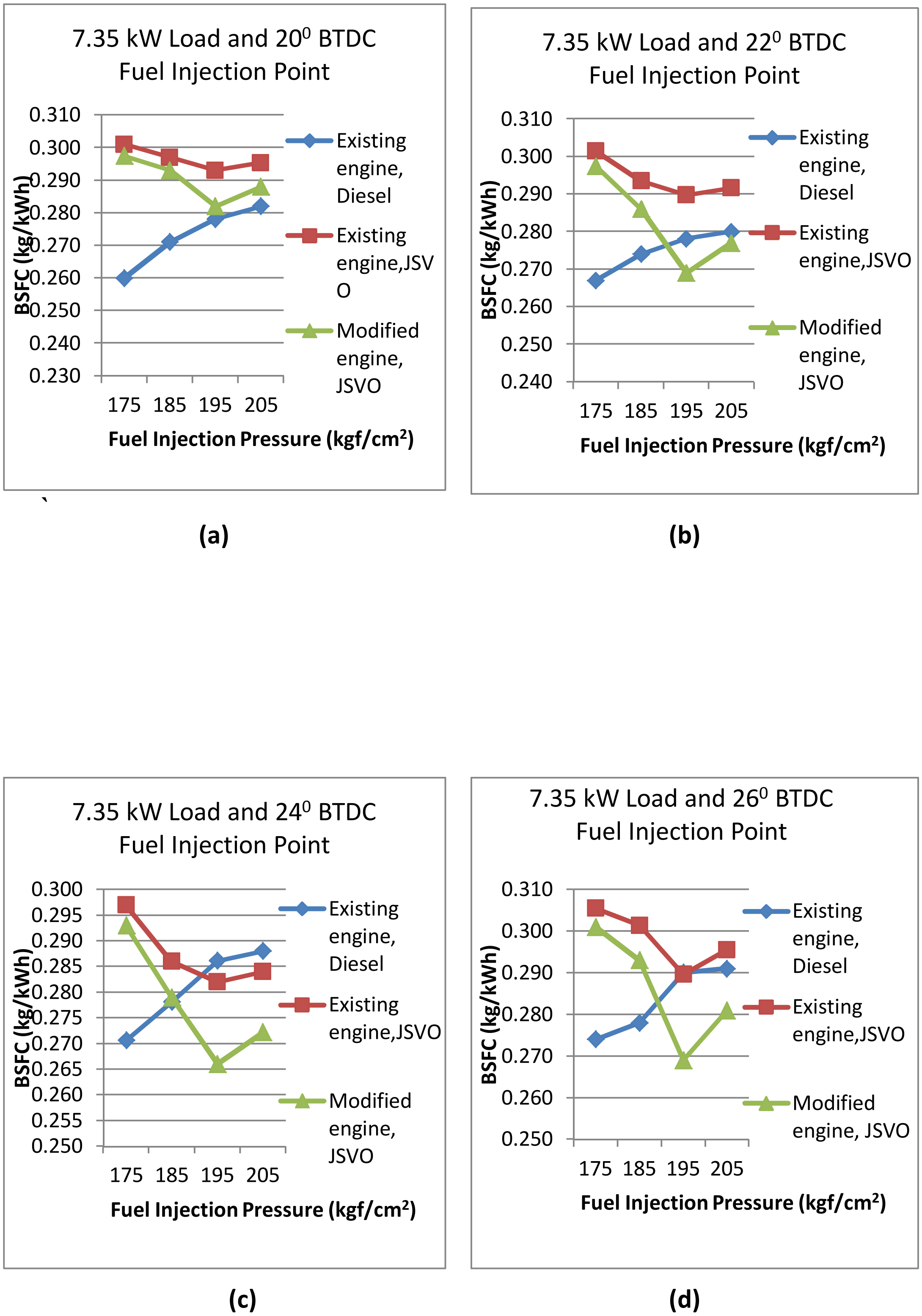


Figure 4.13: BSFC vs. Fuel Injection Pressure

4.4 Emission Characteristics

4.4.1 CO Emission

The engine emissions were measured with AVL-5 gas analyzer (for CO, CO_2, NO_x, HC) and smoke-meter. The measured emissions have been shown graphically. The engine requires rich fuel- air mixture during cold starting engine, warm up and transient like acceleration. The engine operation during those modes contributes significantly to CO emission. The CO emission increases sharply beyond equivalence ratio 1.0 so higher BSFC resulted in higher CO emission.

The CO emission was observed to be high for JSVO fuel which is indication of incomplete combustion. All the measures taken for improving the fuel combustion efficiency ultimately reduced the CO emission. In this work testing the engine at different fuel injection pressure, different fuel injection advance angle and preheating the fuel ascertain the improved in CO emission characteristics.

4.4.1.1 Effect of varying fuel injection point at constant fuel injection pressure

The graph was plotted between CO emission (% Vol) and fuel injection point for the fuel injection pressure of 175 kg f/cm^2, 185 kg f/cm^2, 195 kg f/cm^2 and 205 kgf/cm^2. The CO emission continuously increased for diesel as the angle of fuel injection was increased from 20^0 BTDC to 26 ^{0} BTDC in step of 2^0 for all the load conditions (figure 4.14 to figure 4.17). The CO emission of existing engine with JSVO fuel was observed to be very high at 20^0 BTDC in comparison with CO emission of engine with diesel test fuel at same point. This may be due to high BSFC of JSVO. The high BSFC required more air for complete combustion, so under a constant operating conditions CO formation might have taken place due to incomplete combustion. The CO emission was decreased in the case of JSVO with increase in advance angle of fuel injection up to the angle of 24^0 BTDC at rated (7.35 kW) engine load conditions and at all the fuel injection pressures (175kgf/cm^2, 185kgf/cm^2, 195kgf/cm^2 and 205 kgf/cm^2) but the CO emission started increasing as the angle was further increased to 26^0 BTDC (figure 4.18). This indicated that the combustion efficiency was improved in case of JSVO as the fuel injection angle was increased up to 24^0 BTDC but the combustion efficiency started deteriorating as the angle was further increased. It was observed that the CO emission of engine running with diesel fuel at rated load was least (0.05 % vol) at 20^0 BTDC and 175 kgf/cm^2 which increased on advancing the fuel injection point. The increased CO emission which was an indication of reduced combustion efficiency was in correlation to the change in BTE under similar changes in fuel injection points. The CO emission of the modified engine run with JSVO was also obtained to be decreasing as the angle of fuel injection was increased from 20^0 BTDC to 22^0 BTDC and 24^0 BTDC for all the fuel injection pressures (175kgf/cm^2, 185kgf/cm^2, 195kgf/cm^2 and 205 kgf/cm^2) but the CO emission started increasing as the angle was further increased to 26^0 BTDC (figure 5.19). It was remarkable that the CO emissions of modified engine with JSVO fuel at 24^0 BTDC and 26 ^{0} BTDC measured at all the fuel injection pressures were lower than the CO emission of engine with diesel fuel and also with JSVO.

4.4.1.2 Effect of varying fuel injection pressure at constant fuel injection point

The data observed during the experiment for CO emission was plotted against varying load (0 kW, 1.84 kW, 3.68kW, 5.52kW & 7.35 kW), varying advance angle of fuel injection point (20^0 BTDC, 22^0 BTDC, 24^0 BTDC& 26^0 BTDC) and varying fuel injection pressures (175kgf/cm^2, 185kgf/cm^2, 195kgf/cm^2 & 205 kgf/cm^2). It was observed that the CO emission in case of diesel fuel was increased as the fuel injection pressure was increased from the rated value of 175kgf/cm^2 to 185kgf/cm^2, 195kgf/cm^2 and subsequently 205kgf/cm^2 (figure 4.19). The increase in CO emission on increasing the injection pressure was an indication that the thermal efficiency of engine deteriorated at elevated fuel injection pressure from the manufacturer's recommended value (175kgf/cm^2). So the optimum efficiency was obtained at 175kgf/cm^2 with diesel fuel. From the graph it was seen that the CO emission of the existing as well as modified engine reduced as the fuel injection pressure was increased from 175 kgf/cm^2 to 185 kgf/cm^2 and subsequently to 195 kgf/cm^2 but the CO emission started reducing as the injection pressure was further increased to 205 kg/cm^2. The increase in CO emission at higher pressure may be due to the fact that the spray of smaller fuel particle at higher velocity penetrates more and gets exposed more to air available. These factors lead to better air utilization and combustion characteristics, so the optimum CO emission was obtained at 195kgf/cm^2.The difference of CO emission values between existing and modified engine was due to preheating the fuel. Upon comparison, it was also observed that the CO emission of diesel at all the load conditions was less than the CO emission of JSVO. The CO emission of JSVO, when fueled in modified engine was observed to be less than the CO emission of JSVO when fueled in existing engine.

4.4.1.3 Effect of Fuel preheating at constant fuel injection point and fuel injection pressure

From figure 4.14 to 4.17, it was observed that the CO emission was less at all the operating point in the case of JSVO fuel in modified engine than the CO emission of JSVO fueled in existing engine. The Difference in CO emission of these two engines was due to preheating of fuel because all other operating parameters were the same. It can be concluded that fuel pre-heating reduced the CO emission; the improvement in CO emission at higher temperature of inlet fuel was due to the less heat requirement for getting the fuel atomized. Another effect of fuel preheating was to lower the viscosity which resulted in free flow of fuel and ease of mist formation in the combustion chamber. These phenomena on one side reduce the friction horse power (FHP) and on the other side enhance fuel combustion efficiency. The heat available in the vicinity of fuel particle might be sufficient to vaporize and atomize the fuel; the low emission of CO was obtained which may be due to improved combustion efficiency.

4.4.1.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

The variations in CO emission with change in fuel injection point and fuel injection pressure in both the cases i.e. in modified engine and existing engine exhibited the results that the CO emission characteristics of fuel JSVO was improved by implementing higher degree (24^0) of fuel injection and 195 kgf/cm^2 pressure with a system of fuel pre-heating incorporated in the engine fuel supply system. The present trend of research has accounted the effect of these parameters individually. In this work the effect of these parameters was integrated by testing the engine at all the combinations of operating parameters (fuel injection advance angle, fuel injection pressure and fuel preheating). The result obtained shows that the CO emission at recommended value of injection pressure (195 kgf/cm^2) and fuel injection point (24^0 BTDC) with fuel preheating improved the CO emission of engine with JSVO fuel from 0.10 % vol. to 0.05 % vol. at 7.35 kW engine load. The base line CO emission of engine running at 7.35 kW with diesel fuel is 0.05 % vol. at 20^0 BTDC and 175 kgf/cm^2. This shows that the modifications in fuel injection system improved the CO emission to an extent of base line CO emission (0.05%) at rated engine load condition.

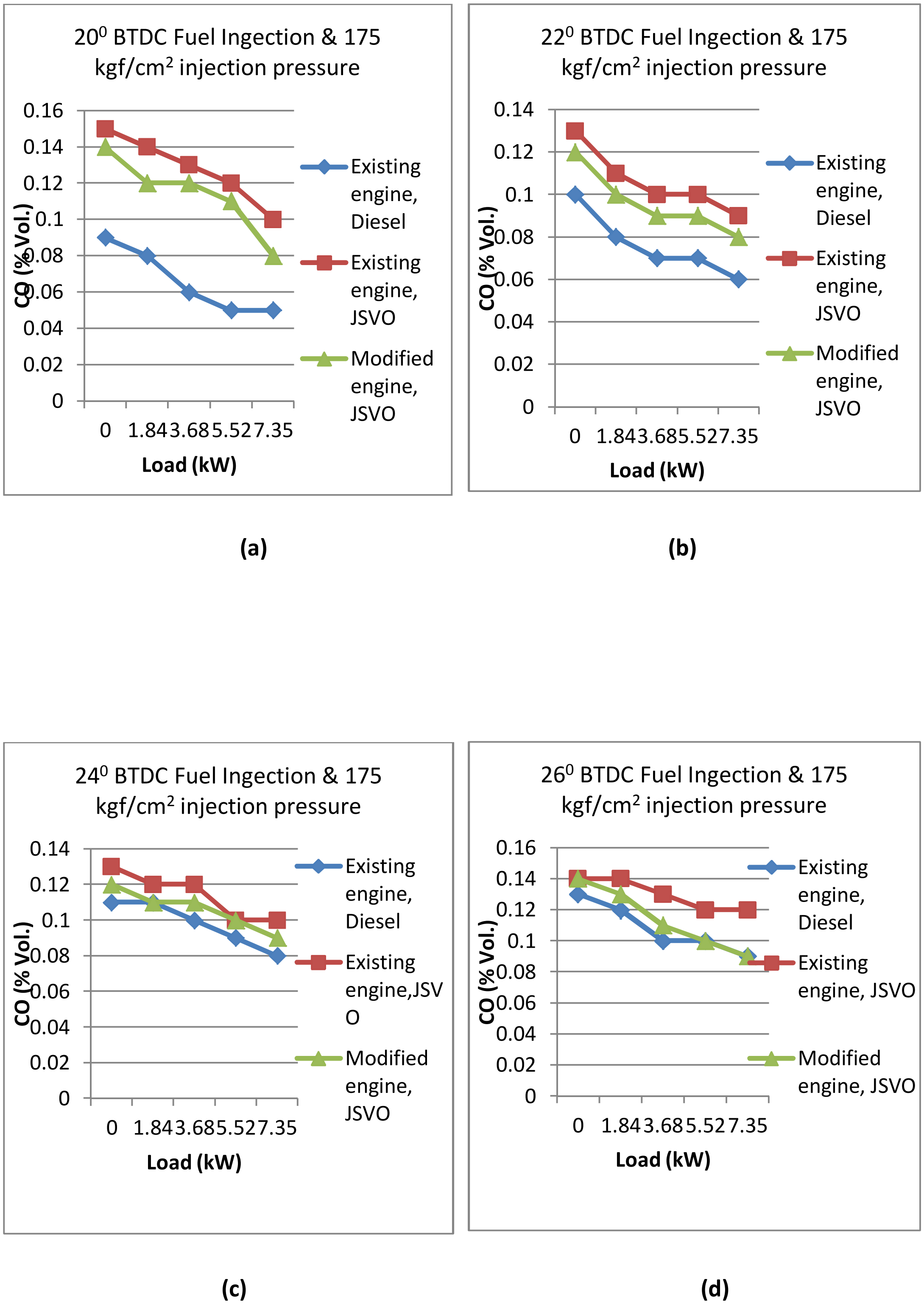


Figure 4.14: CO emission vs. Engine Load

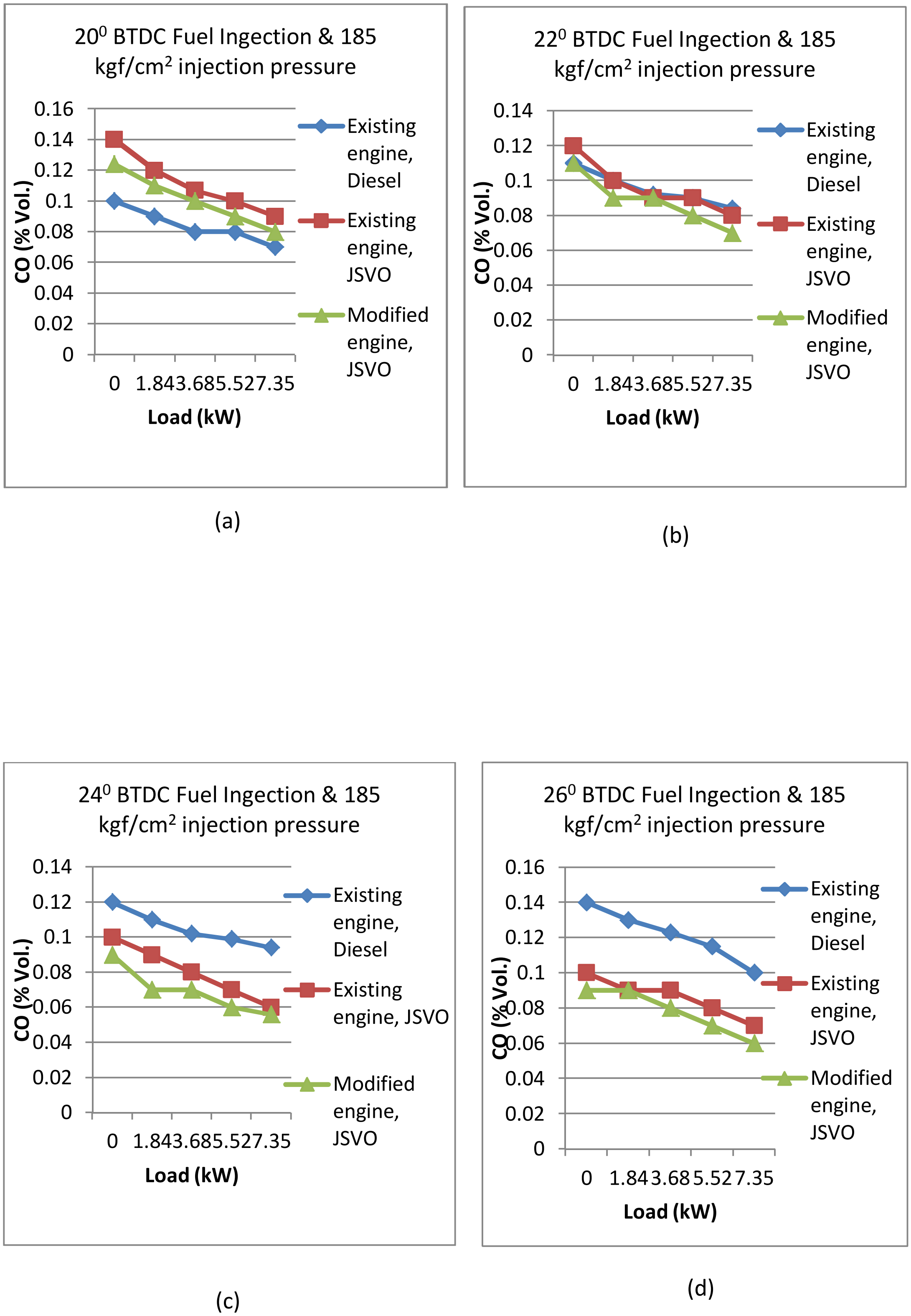
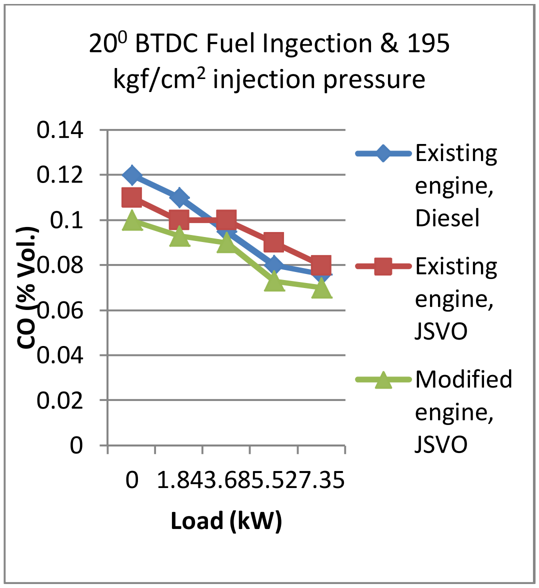


Figure 4.15: CO emission vs. Engine Load



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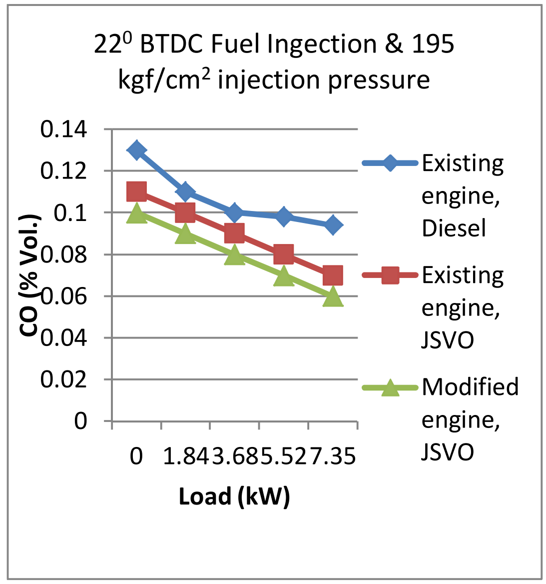

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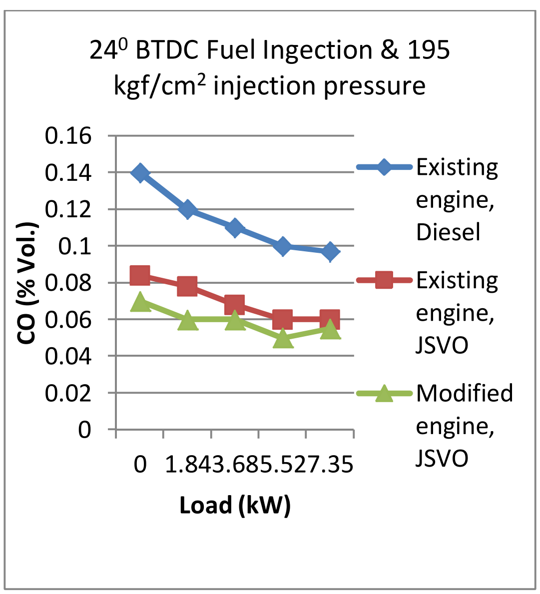

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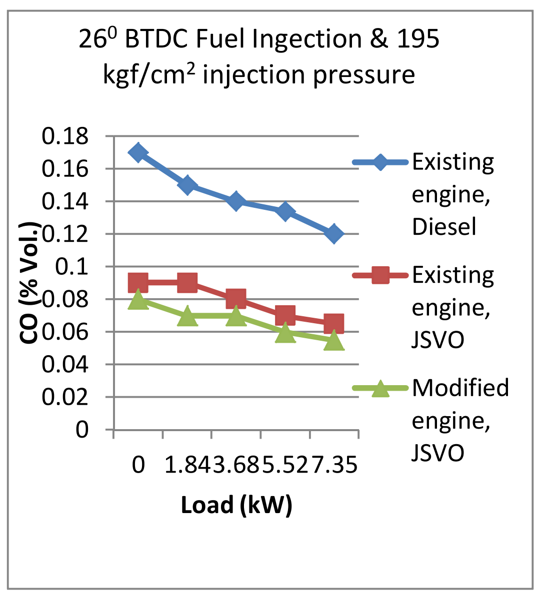


(d)

Figure 4.16: CO emission vs. Engine Load

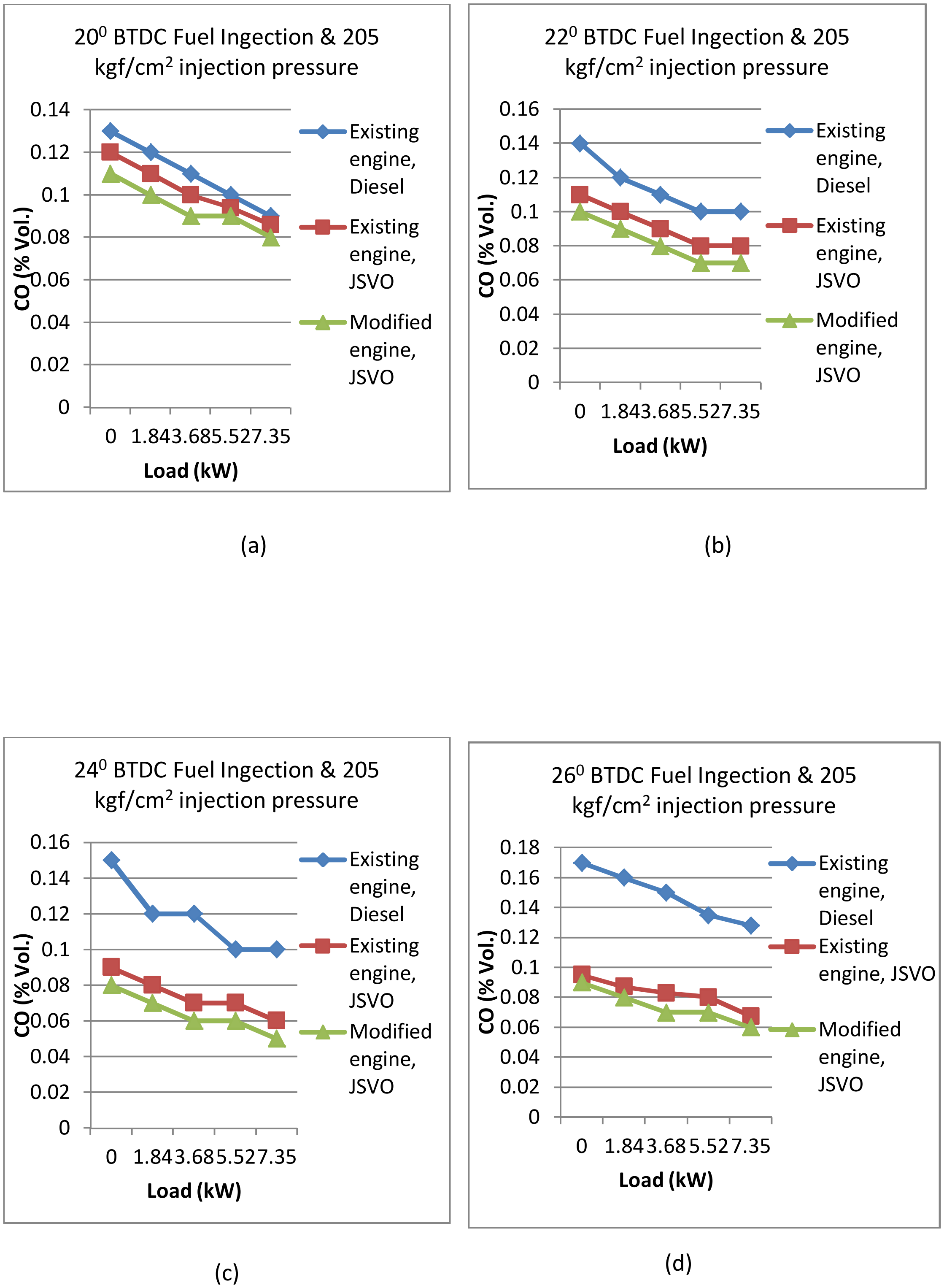


Figure 4.17: CO emission vs. Engine Load

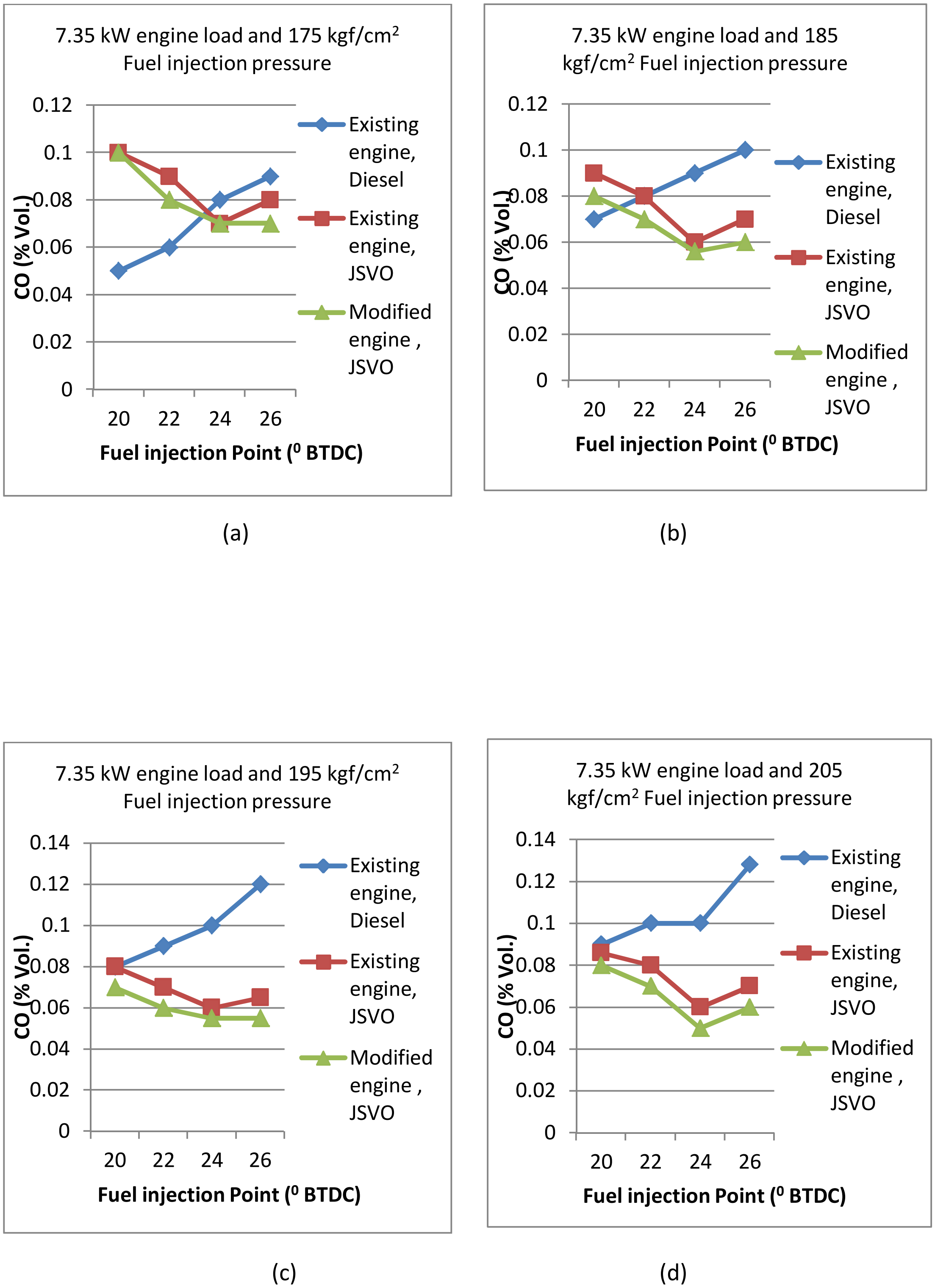


Figure 4.18: CO emission vs. Advance angle of Fuel injection

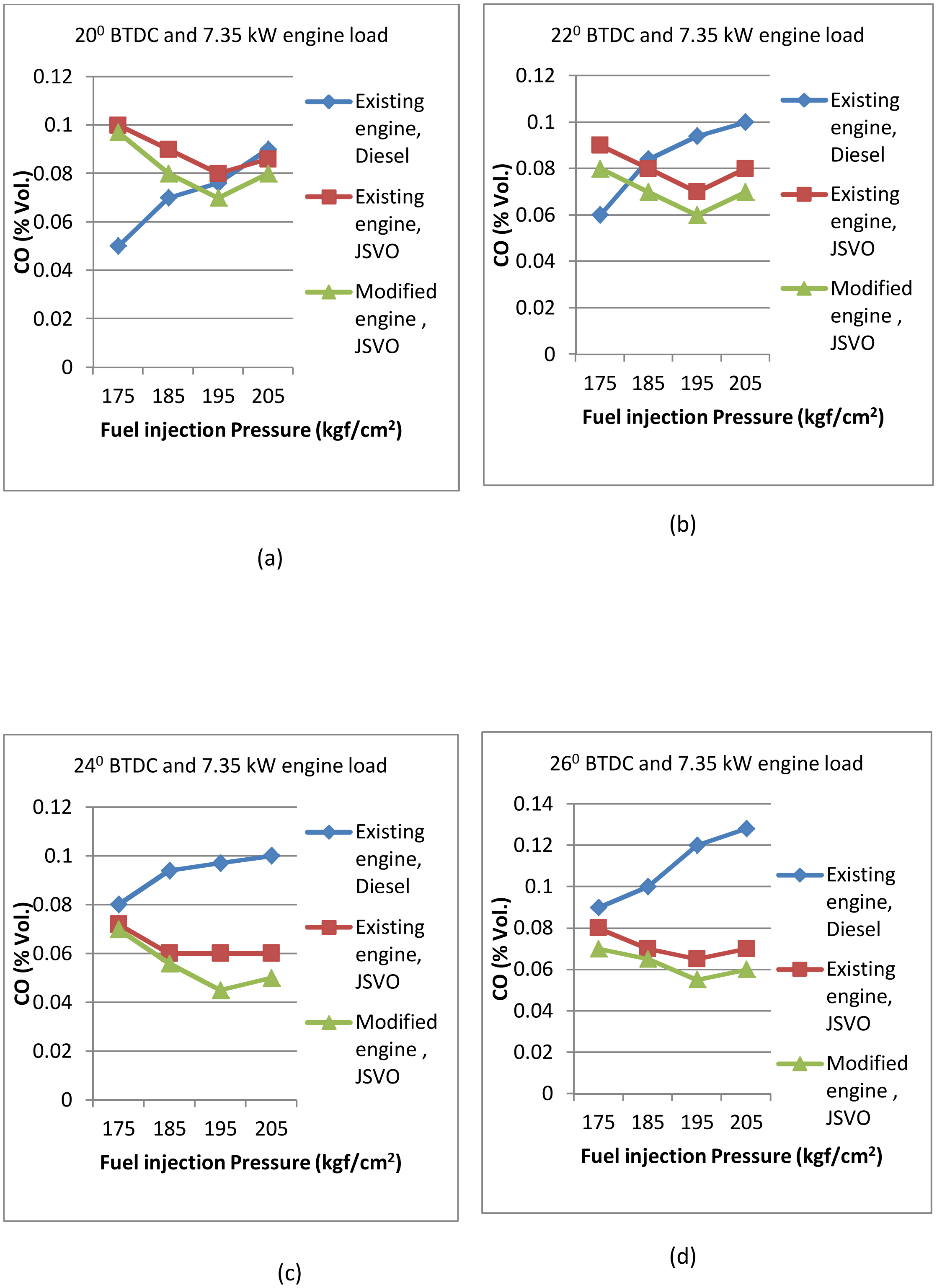


Figure 4.19: CO emission vs. Fuel injection pressure

4.4.2 CO_2 Emission:

The effect of advanced injection timing was evident for the production of carbon dioxide. The highest CO_2 concentrations in the exhaust were recorded when the engine was run on diesel fuel. The rated fuel injection timing (20^0 BTDC) and 175kgf/cm^2 fuel injection pressure offered highest (1.24% Vol.) CO_2 emissions in comparison with CO_2 emission (0.76% Vol.) of engine fueled with JSVO at same injection point and injection pressure. The effect of air fuel mixture on CO_2 emission was that the CO_2 emission increased as the A/F ratio was decreased. The emission of CO_2 was therefore, a measure of combustion efficiency of the system. It is desirable to have high CO_2 and less HC emissions under any operating condition [30].

In the range of the whole engine load, the CO_2 emissions of diesel fuel were higher than that of the JSVO because the vegetable oil contains oxygen element. The carbon content was relatively lower in the same volume of fuel consumed at the same engine load and consequently, the CO_2 emissions from the vegetable oil were higher.

4.4.2.1 Effect of varying fuel injection point at constant fuel injection pressure

The data of CO_2 emission obtained during the experiments were plotted against varying engine load (0 kW, 1.84 kW, 3.68kW,5.52 kW ,7.35kW), varying fuel injection point (20^0 BTDC, 22^0 BTDC, 24^0 BTDC, 26^0 BTDC) and fuel injection pressure (175 kgf/cm^2 , 185 kgf/cm^2, 195 kgf/cm^2, 205 kgf/cm^2) in figure 4.20 to figure 4.24. The increase in CO_2 emission was obtained on increasing the engine load (figure 4.20 to figure 4.23). The increase in CO_2 at increased engine load indicated that the brake engine output in relation to fuel consumption was increased. This may be due to the fact that the engine efficiency increases as the engine approaches to rated engine load due to higher temperature of burnt gases at higher engine load. The higher temperature available in combustion chamber helps in fuel atomization process and also the oxidation of CO available in burnt gases after the combustion process is finished. This results in efficient combustion process and conversion of CO in to CO_2 thereby giving increased CO_2 emission. Decrease in CO_2 emission of engine with diesel fuel was observed at all but fixed fuel injection pressure as the fuel injection advance angle was increased from 20^0 BTDC to 26^0 BTDC in steps of 2^0. This showed that the optimum engine performance was at 20^0 BTDC. The performance deteriorated as the advance angle of fuel injection was increased from its rated fuel injection point. The CO_2 emissions of existing as well as modified engine with JSVO fuel increased gradually as the advance angle of fuel injection was increased up to 24^0 BTDC but the CO_2 emission started decreasing beyond 24^0 of fuel injection which indicated that the fuel combustion efficiency started deteriorating (figure 4.24). The increased CO_2 emission on advancing the fuel injection point may be due to the fact that increased delay period offers more time for pre combustion process. This has also been reported by the researchers that advancing the fuel injection point too far can have negative consequence, resulting in engine knocking and erratic engine behavior [1, 30], these effects reduces the engine performance and hence CO_2 emission.

4.4.2.2 Effect of varying fuel injection pressure at constant fuel injection point

It was observed that the CO_2 emission in case of diesel fuel was decreased as the fuel injection pressure was increased from the rated value of 175kgf/cm^2 to 185kgf/cm^2, 195 kgf/cm^2 and subsequently to 205kgf/cm^2. The graph also indicates that the CO_2 emission was increased as the load was increased from 0 kW to 7.35 kW. In the case of JSVO as test fuel of existing as well as modified engine it was found that the CO_2 emission was increased on increasing the fuel injection pressure up to 195 kgf/cm^2 where as the CO_2 emission started decreasing on further increase in fuel injection pressure to 205 kgf/cm^2. That means the higher fuel injection pressure provides better fuel combustion efficiency due to improved fuel atomization characteristics (figure 4.25). On comparison this was also observed that the CO_2 emission of diesel at all the combinations of operating points were more than the CO_2 emission of JSVO (figure 4.20 to 4.25). However in some of the cases (at 205 kgf/cm^2 fuel injection pressure) this was also observed that CO_2 emission of modified engine was lower than the CO_2 emission of existing engine, it indicated that the effect of reducing the CO_2 emission by increasing the fuel injection pressure from optimum value (195 kgf/cm^2) was more than the effect of preheating the fuel to increase the CO_2 emission. In general the CO_2 emission of JSVO when fueled in modified engine was observed to be more than the CO_2 emission of JSVO when fueled in existing engine.

4.4.2.3 Effect of fuel preheating at constant fuel injection point and fuel injection pressure

It was observed that the CO_2 emission was more at all the operating point in the case of JSVO fuel in modified engine than the CO_2 emission of JSVO fueled in existing engine. The difference in CO_2 emission between these two cases were only because of fuel preheating, as all the other operating parameters were same at any one observation point. This revealed that the fuel pre-heating increases the CO_2 emission. This may be because of improved fuel combustion quality as the viscosity of fuel decreases by pre-heating.

4.4.2.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

The data plotted in figures from figure 4.20 to figure 4.25 for CO_2 emission shows that the change in fuel injection point and fuel injection pressure in both the cases i.e. in modified engine and in existing engine exhibited the result that the CO_2 emission characteristics of fuel JSVO was improved by implementing higher degree (24^0) of fuel injection and 195 kgf/cm^2 fuel injection pressure with a system of fuel pre-heating incorporated in the engine fuel supply system. The CO_2 emission of modified engine at recommended fuel injection point (24^0 BTDC) and fuel injection pressure (195 kgf/cm^2) was observed to be 1.24 % vol. which was an improvement of CO_2 emission of existing engine with JSVO fuel from 0.76% Vol. (at 20^0 BTDC and 175kgf/cm^2) to 1.24 % Vol. (at 24^0 BTDC and 195 kgf/cm^2 with fuel preheating). The existing engine with diesel fuel at rated operating point (at 20^0 BTDC and 175kgf/cm^2) had CO_2 emission 1.24 % vol. This shows that the modified engine emits equal amount of CO_2 with the modifications incorporated to it. The decrease in CO_2 indicated the incomplete combustion, therefore it was concluded that the emission performance of the engine with diesel fuel got deteriorated as the advance angle of fuel injection was increased from the rated design value of 20^0 BTDC and also as the fuel injection pressure was increased from the rated value of 175kgf/cm^2. The improvement in CO_2 emission was observed as the load on engine approached to rated load.

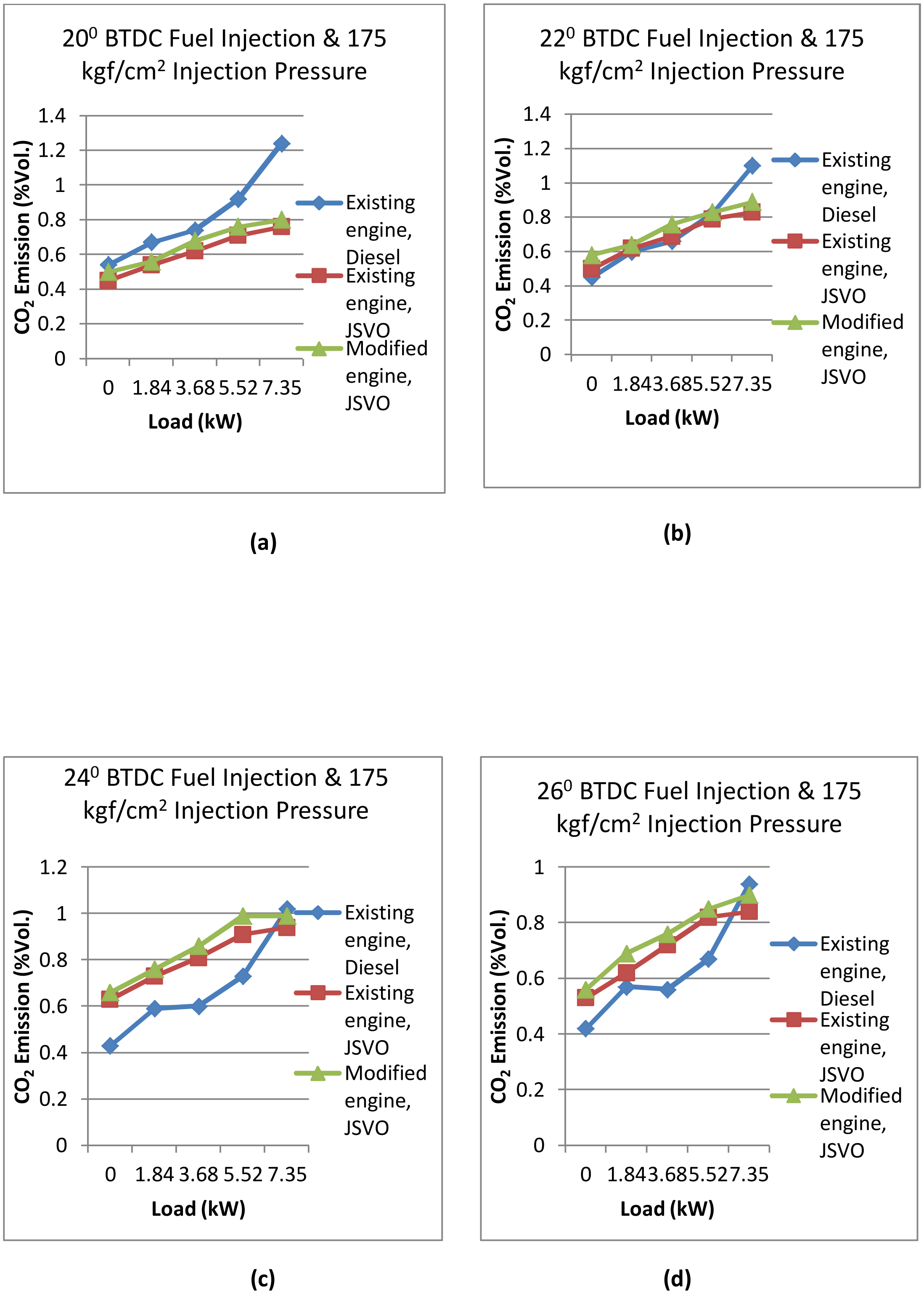


Figure 4.20: CO_2 emission vs. Engine Load

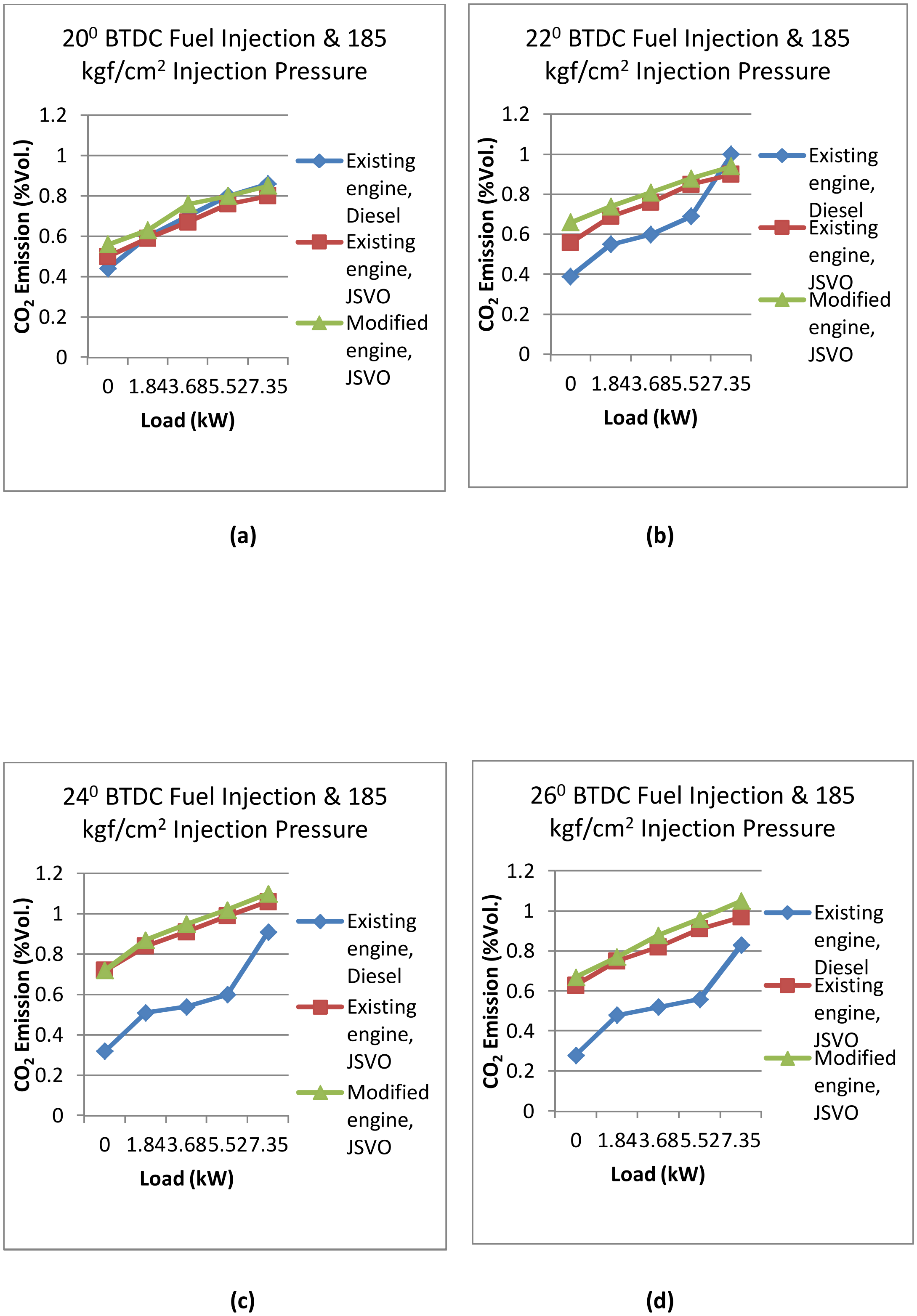
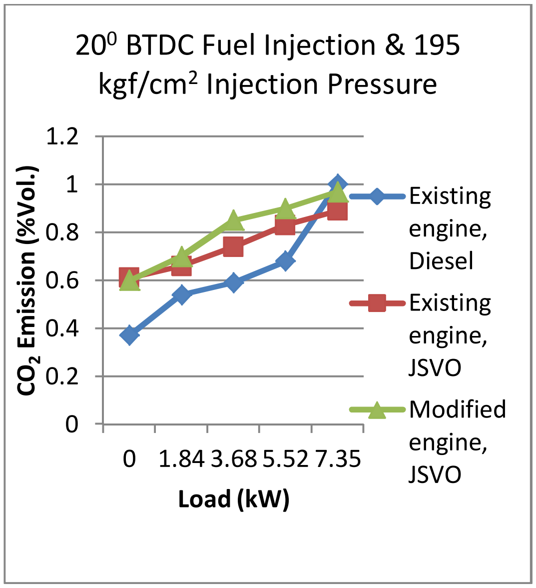


Figure 4.21: CO_2 emission vs. Engine Load



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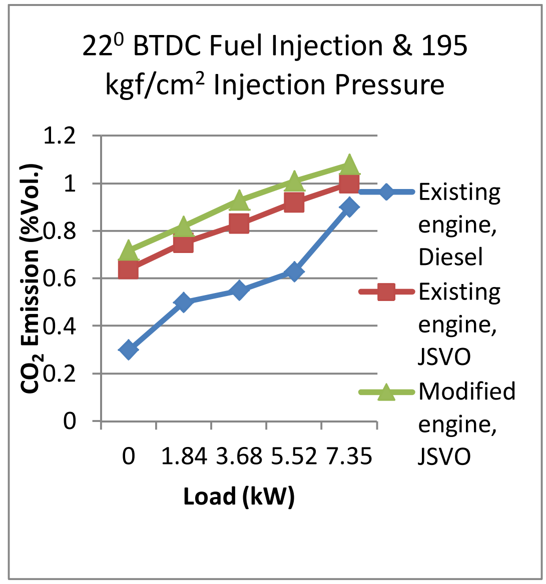

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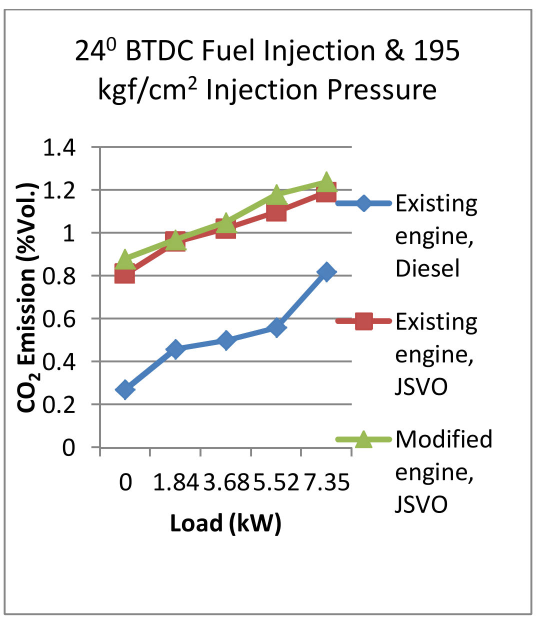

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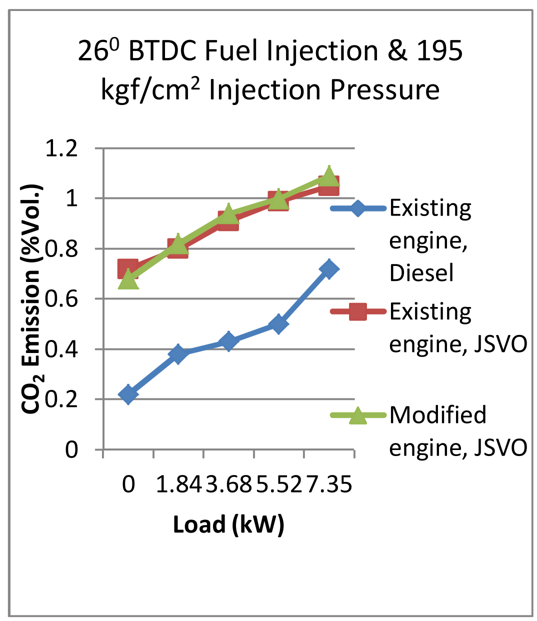

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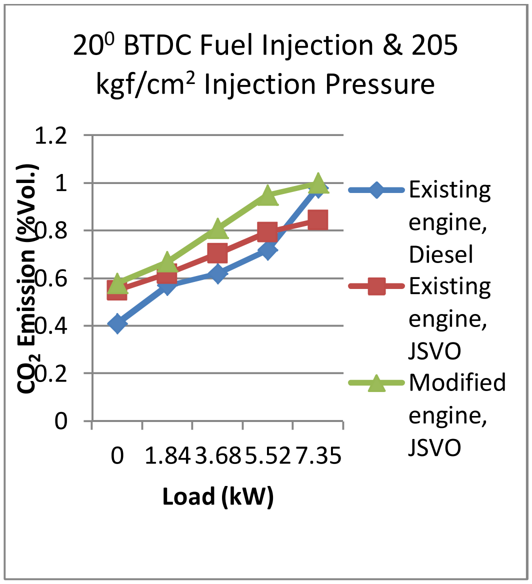
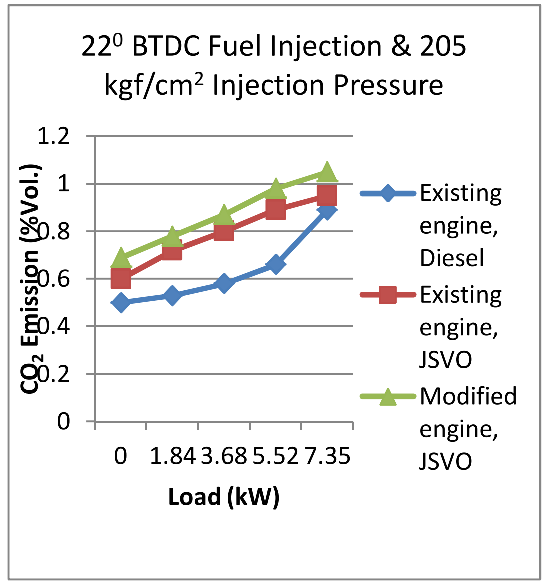
Figure 4.22: CO_2 emission vs. Engine Load



(a)



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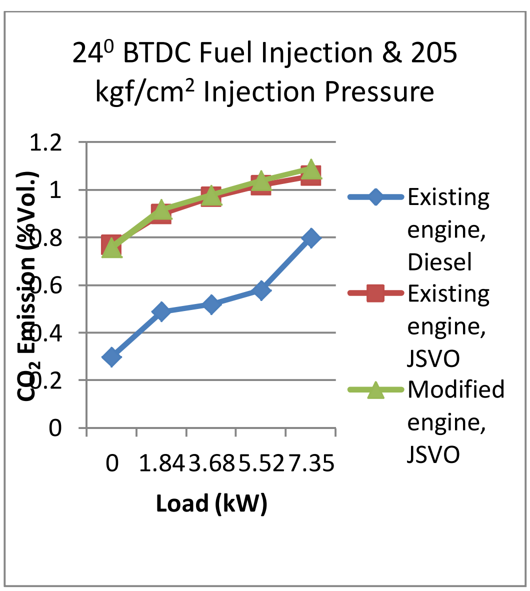

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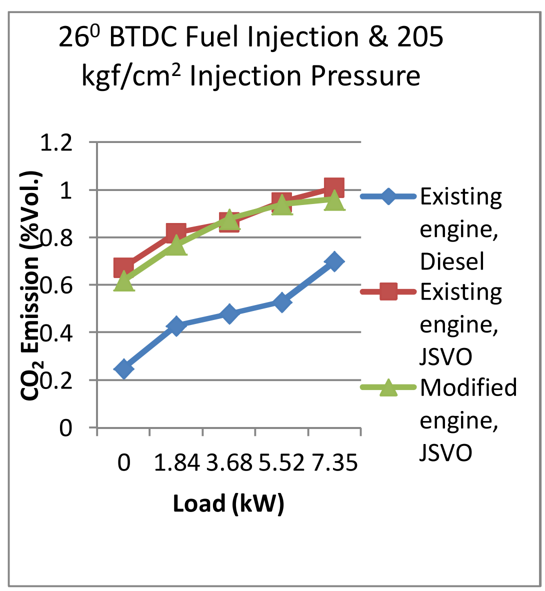

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Figure 4.23: CO_2 emission vs. Engine Load

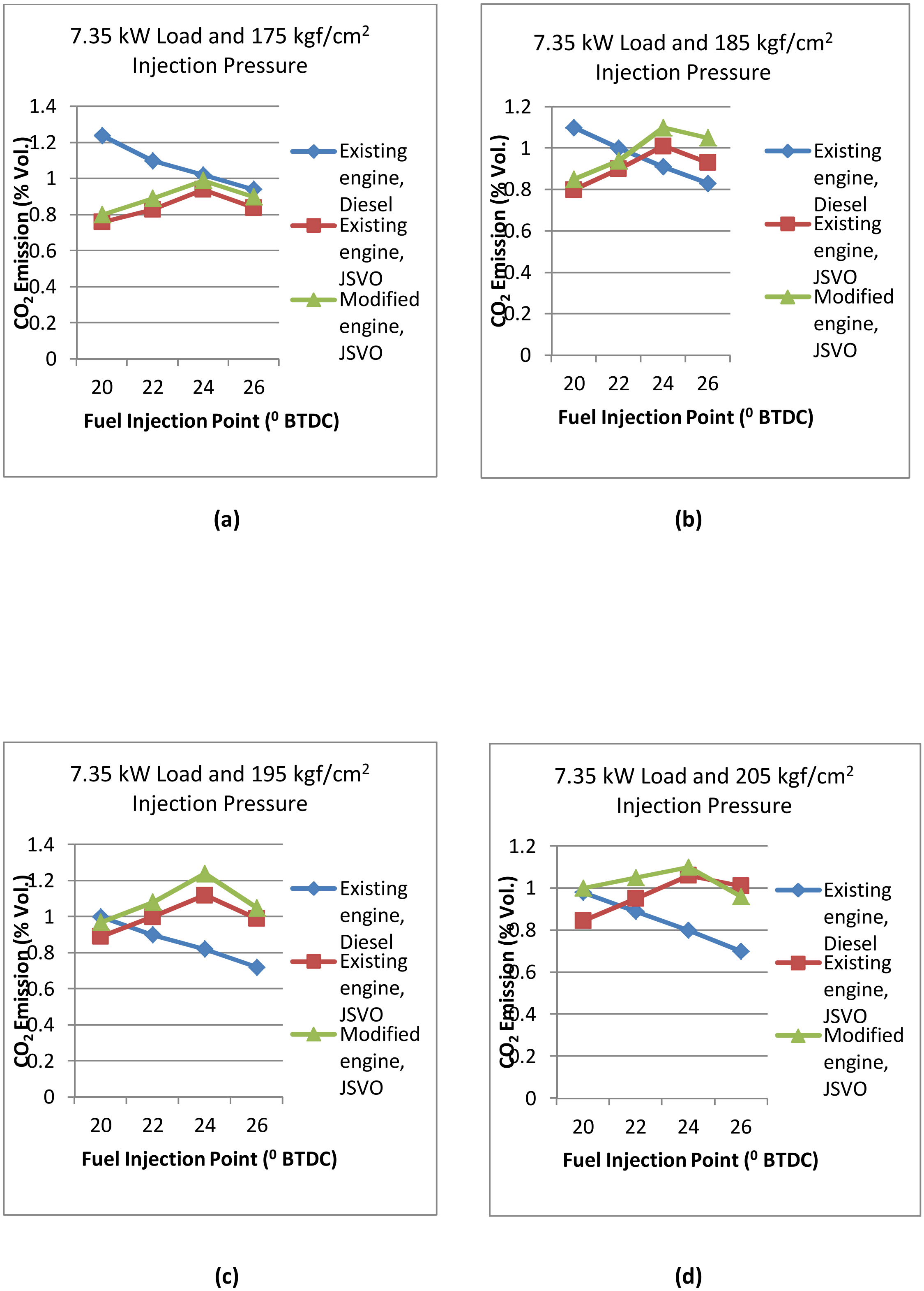


Figure 4.24: CO_2 emission vs. advance angle of fuel injection

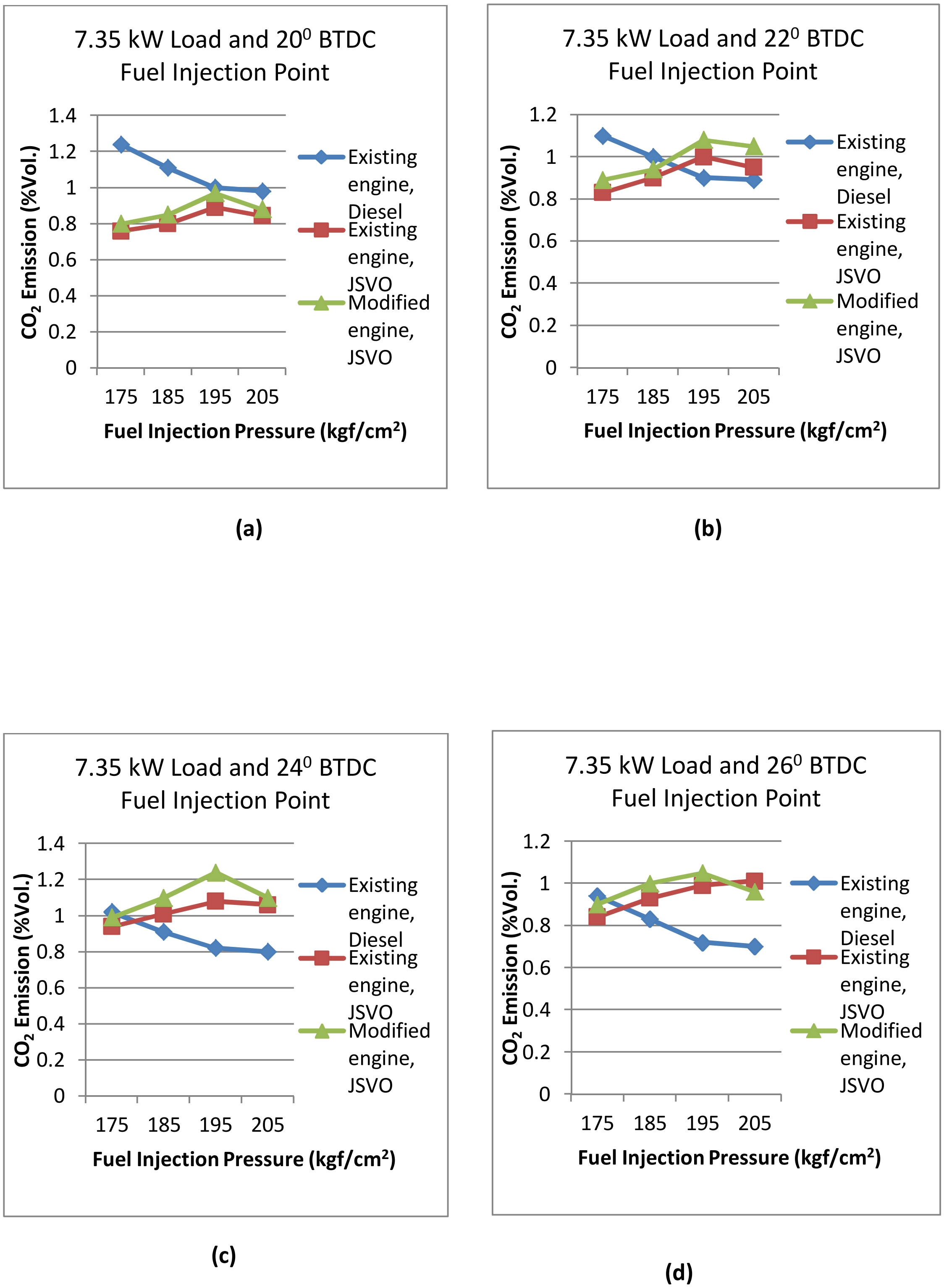


Figure 4.25: CO_2 emission vs. Fuel injection pressure

4.4.3 NO_x Emission:

It is important to consider Nitrogen oxides as it is one of the major pollutant and emission concerns. When combustion temperature is decreased the emission of nitrogen oxides are reduced but the smoke and particulate emission increases [30]. By retarding the injection timing the temperature inside the combustion chamber decreases, and therefore a reduction in the NO_x emission was obtained. To meet the above stated requirements advanced fuel injection technologies that employ very high injection pressure and electronic control of injection timing and rate are being used now a days. The use of electronic fuel injection (EFI) system results into very high injection pressure up to 2000 bar to atomize fuel into very fine droplets for fast vaporization. It has been reported in literatures that increase in fuel injection pressure beyond a point results no further improvement in air – fuel mixing. The retarded fuel injection improves the NO_x emission characteristics. It is also suggested that the significant reduction in NO_x and particulate takes place when 75% fuel is injected in the first pulse and the 25% in the second pulse after a dwell period of 10^0 crank angle [30].

4.4.3.1 Effect of varying fuel injection point at constant fuel injection pressure

The graph was plotted between NO_x emission (ppm) and varying fuel injection point (20^0 BTDC, 22^0BTDC,24^0BTDC, 26^0 BTDC) for the fuel injection pressure of 175 kgf/cm^2, 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kgf/cm^2. The NO_x emission was continuously increasing for existing engine with diesel fuel as the angle of fuel injection was increased from 20^0 BTDC to 26 ^{0} BTDC in step of 2^0 for all the load conditions (figure 4.26 to figure 4.29). The increased NO_x emission at retarded angle of fuel injection point may be due to the increase in peak cylinder pressure and temperature as the combustion occurs earlier in the cycle and more heat is released before and around the top dead centre. The charge elements which burns early in the cycle are subjected to higher temperature and pressure with advance in peak timing and remain at high temperatures for a longer period.

These early burn elements contribute most to NO formation and hence higher NO formation rates result with advanced ignition timing. The NO_x emission for JSVO when fueled in existing engine was observed to be high at 20^0 BTDC in comparison with NO_x emission of diesel at same point. The NO_x emission was decreased in the case of JSVO fuel with increase in advance angle of fuel injection up to the angle of 24^0 BTDC at all the load conditions (except at 7.35kW load and 185 kgf/cm^2 injection pressure) but the NO_x emission started increasing as the angle was further increased to 26^0 BTDC. The reverse behavior of diesel and JSVO in NO_x emission on varying the advance angle of fuel injection angle may be because the established combustion process in JSVO might be still later than the diesel fuel. This consequence of late combustion could have been due to low temperature even little beyond the TDC point. This indicated that the combustion efficiency was improved in case of JSVO as the fuel injection angle was increased up to 24^0 BTDC but the combustion efficiency deteriorated as the angle was further increased (Figure 4.30). The inverse result beyond 24^0 BTDC may be due to possibility of engine knocking and erratic behavior. It has also been observed that the NO_x emission of diesel fuel at all the angles of fuel injection was higher than the NO_x emission of JSVO except at 175 kgf/cm^2 fuel injection pressure. The NO_x emission of the JSVO when fueled in the modified engine was also decreased as the angle of fuel injection was increased from 20^0 BTDC to 24^0 BTDC in step of 2^0 for all the load conditions, but the NO_x emission started increasing as the angle was further increased to 26^0 BTDC (Figure 4.30). It was remarkable that the NO_x emission at recommended operating point (24^0 BTDC, 195 kgf/cm^2) for JSVO (330 ppm) was even less than the NO_x emission of diesel (360 ppm) at rated operating point (20^0 BTDC, 175 kgf/cm^2).

4.4.3.2 Effect of varying fuel injection pressure at constant fuel injection point

From the graph it was observed that the NO_x emission in case of diesel fuel was increased as the fuel injection pressure was increased from the rated value of

175kgf/cm^2 to 185kgf/cm^2, 195kgf/cm^2 and 205kgf/cm^2 (figure 4.31). The graph also indicated that the NO_x emission kept reducing as the load was increased from 0 kW to 7.35kW (figure 4.26 to figure 4.29). The NO_x emission of modified and existing engine was found to be reducing as the fuel injection pressure was increased from 175 kgf/cm^2 to 195 kgf/cm^2 but the trend was reversed when the pressure was further increased (figure 4.32). This may be due to the fact that peak pressure and temperature is very close to TDC or even in some of the cases after the TDC which results in exposing the NO for very small time period to the excess air available for the formation of NO_x. The reverse effect of engine beyond the injection pressure 195 kgf/cm^2 might be due to the possibility of engine knocking and erratic behavior of engine. The NO_x emission of JSVO when fueled in modified engine was observed to be less than the NO_x emission of JSVO when fueled in existing engine. This difference was due to the effect of fuel preheating.

4.4.3.3 Effect of Fuel preheating at constant fuel injection point and fuel injection pressure

On comparison of figure 4.26 to figure 4.29, it was found that the NO_x emission was less at all the operating point in the case of JSVO fuel in modified engine than the NO_x emission of JSVO fueled in existing engine. This shows that the fuel pre-heating reduces the NO_x emission. This is because of improved fuel combustion quality as the viscosity of fuel was decreased by pre-heating. On comparing the amount of NO_x emission when the JSVO was used on existing engine and in another case when used in modified engine the amount of emission was observed to be reduced which directly relates to better fuel combustion quality. Though due to better combustion quality the temperature gets increased which should increase the NO_x emission but this effect might have been dominated by the decrease in temperature due to fuel injection advance angle. That means the preheating of fuel was an advantageous modification which improved the performance characteristics as discussed earlier and also the emission characteristics.

4.4.3.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

The NO_x emission with change in fuel injection point and fuel injection pressure in both the cases that is in modified engine and in existing engine exhibited that the NO_x emission characteristics of fuel JSVO was improved by implementing higher degree (24^0) of fuel injection and 195 kgf/cm^2 pressure with a system of fuel pre-heating incorporated in the engine fuel supply system.

In association with the effect of fuel injection advance angle and the increase in fuel injection pressure the pattern of decrease in NO_x indicated the improvement in combustion quality of JSVO in combustion chamber. It shows that the emission performance of the JSVO fuel improved as the advance angle of fuel injection was increased from the rated value of 20^0 BTDC upto 24^0 BTDC but it was deteriorated on further increase to 26^0 BTDC. The similar trend was shown on increase in fuel injection pressure from the rated value of 175kgf/cm^2 .

The NO_x emission of existing engine with diesel fuel at rated operating point (20^0 BTDC and 175 kgf/cm^2) was 360 ppm where as the emission of existing engine with JSVO fuel at same operating point was 430 ppm. After the modifications was incorporated, the NO_x emission of modified engine at 24^0 BTDC and 195 kgf/cm^2 was obtained to be 330 ppm. This shows that the modification not only made the engine compatible to the existing engine on the basis of NO_x emission but also the NO_x emission parameter was further improved.

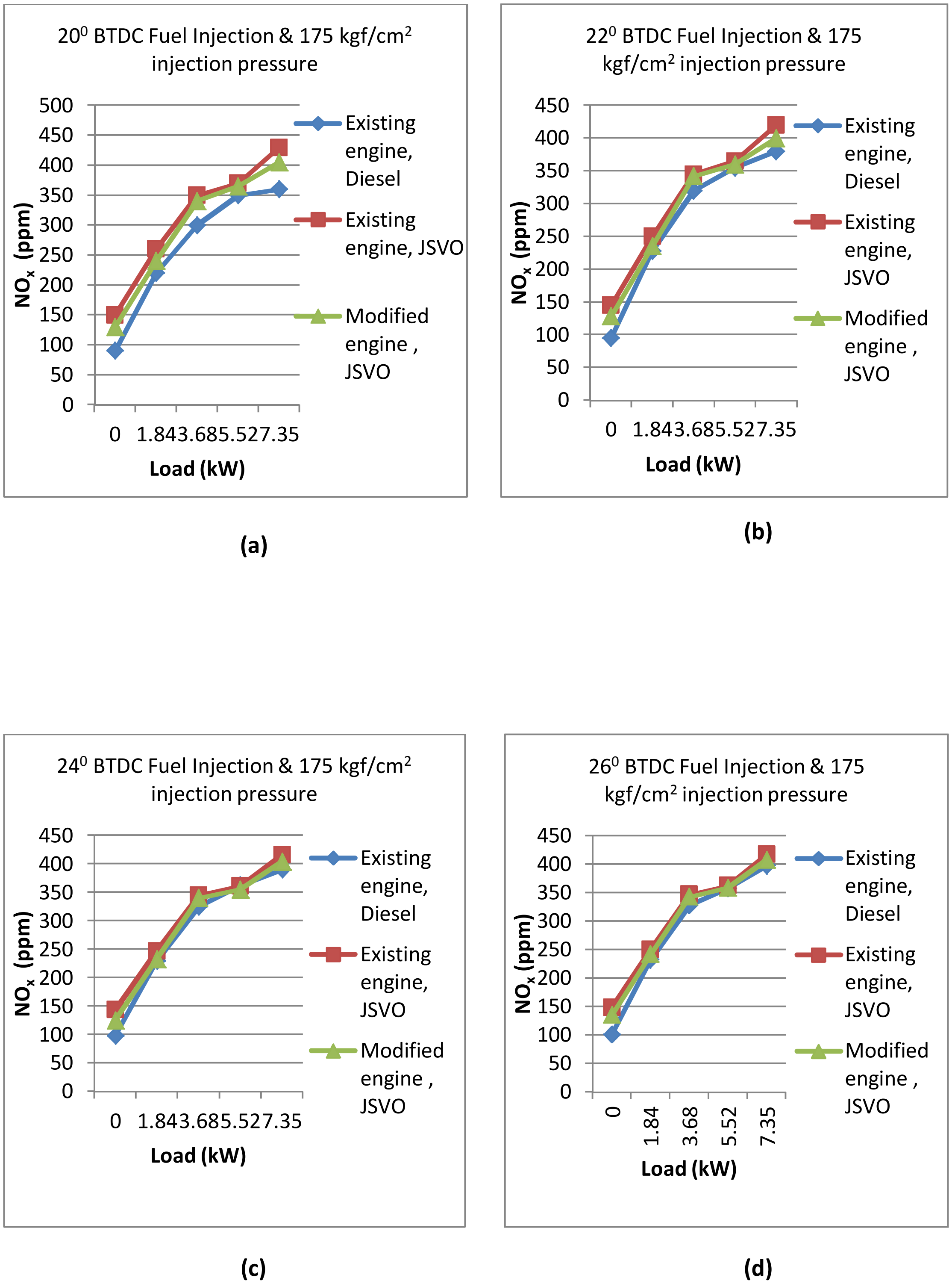
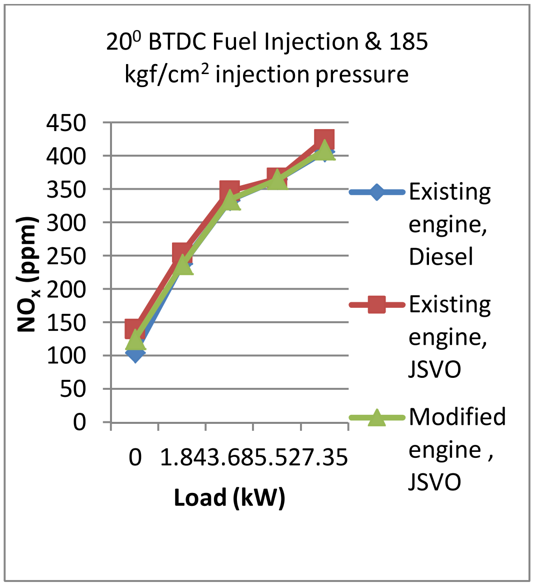


Figure 4.26: NO_x emission vs. Engine Load



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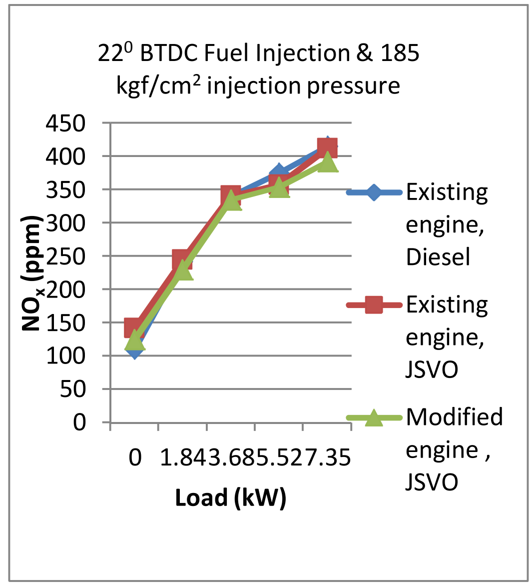

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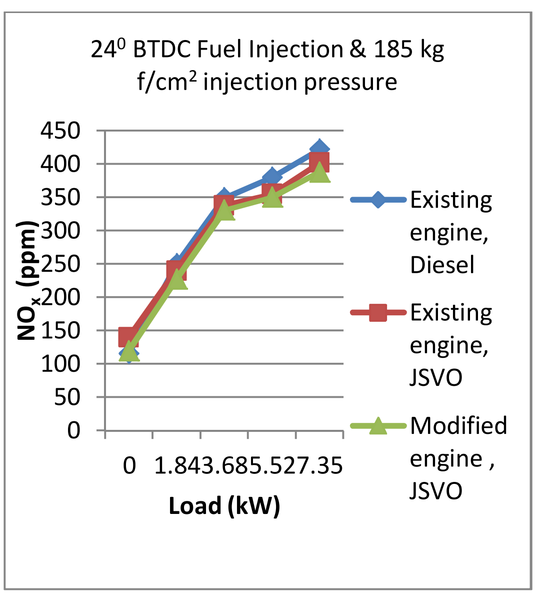

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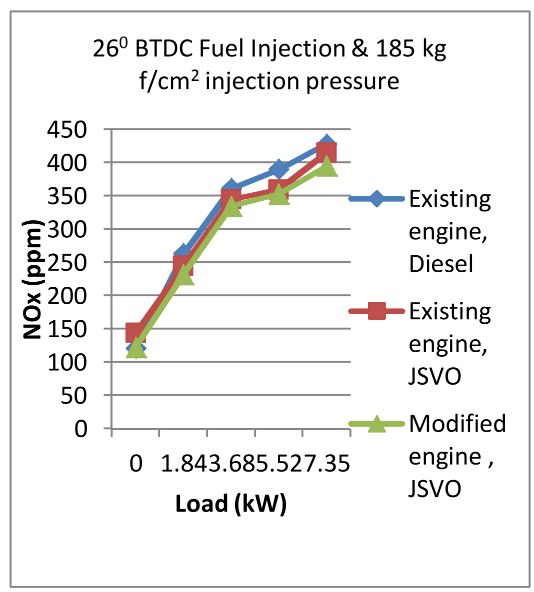

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Figure 4.27: NO_x emission vs. Engine Load

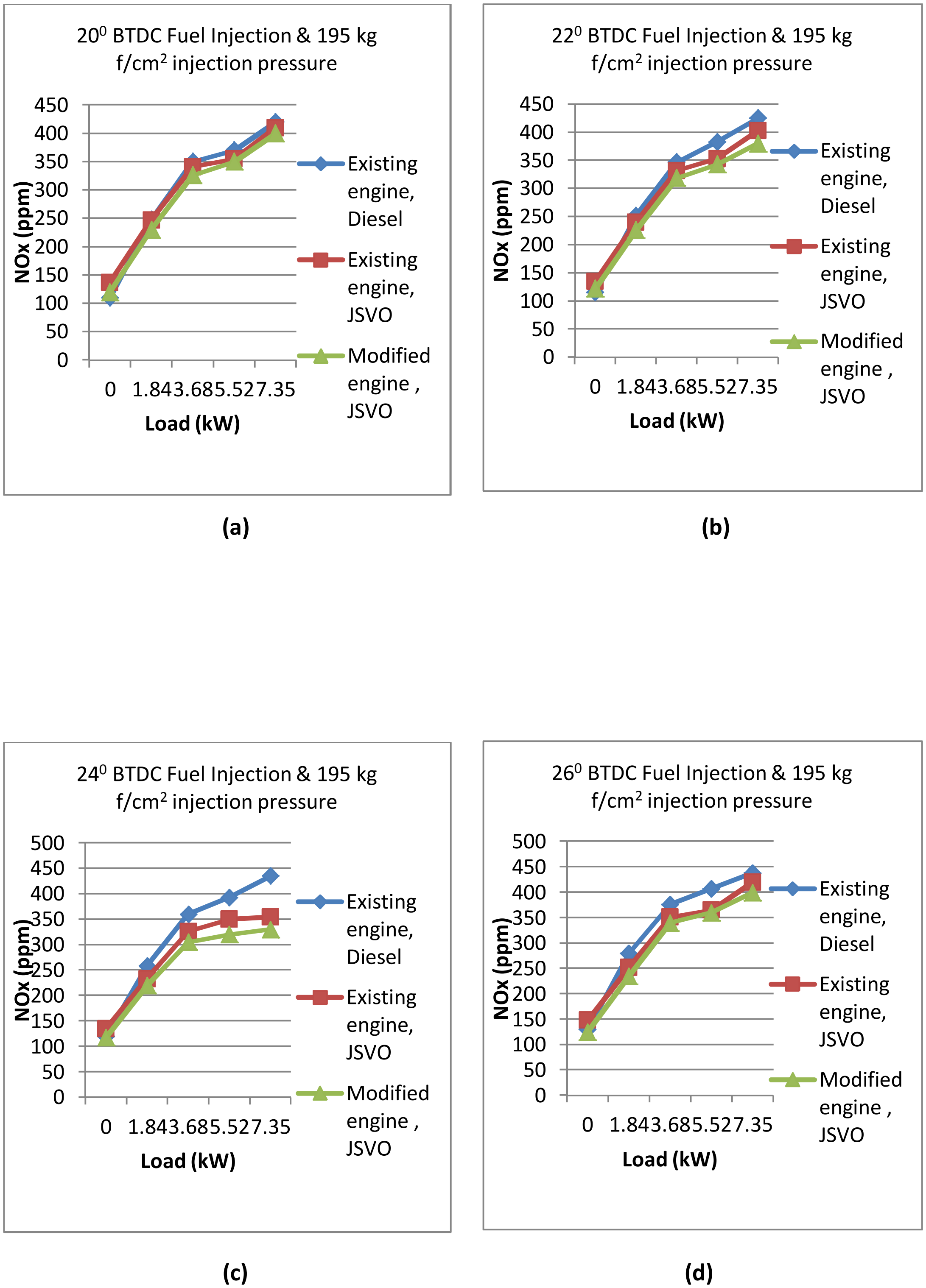


Figure 4.28: NO_x emission vs. Engine Load

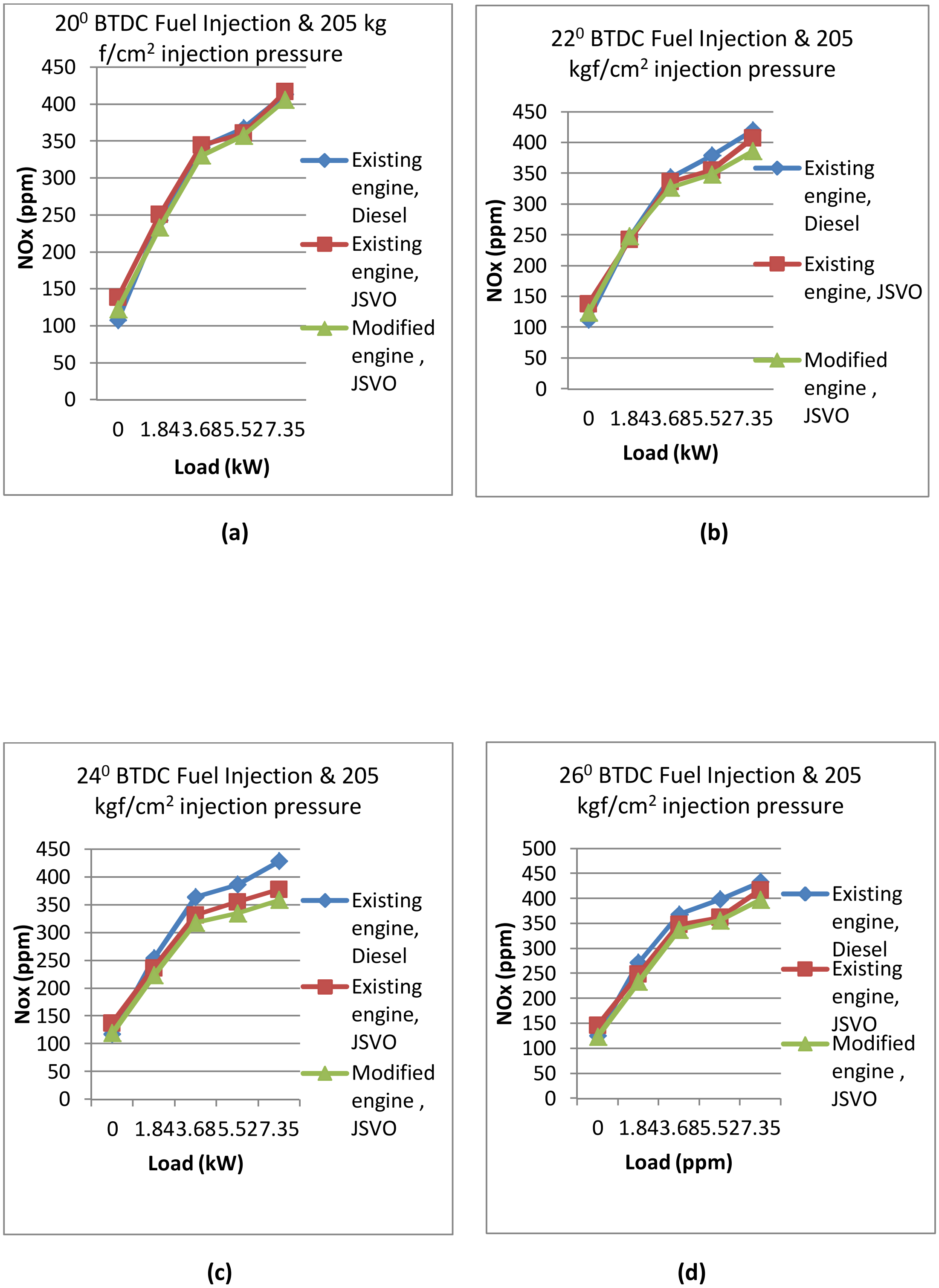
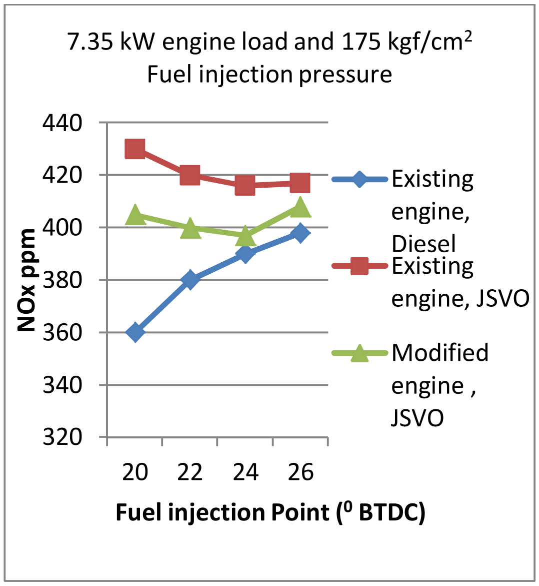


Figure 4.29: NO_x emission vs. Engine Load



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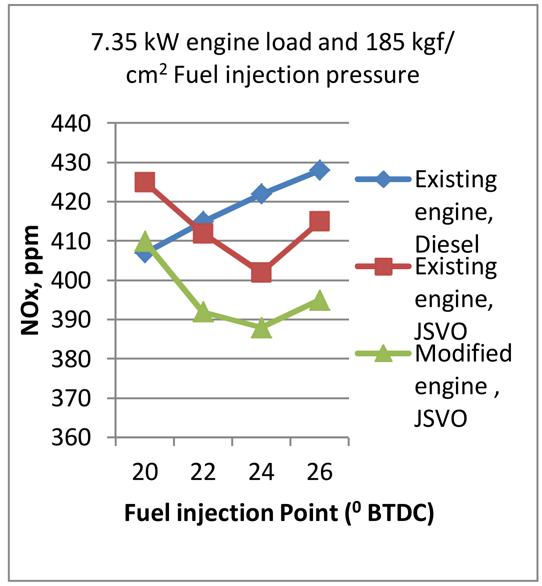

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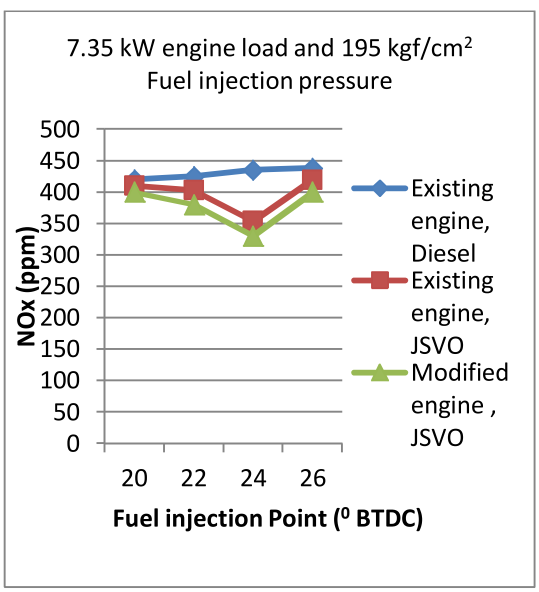

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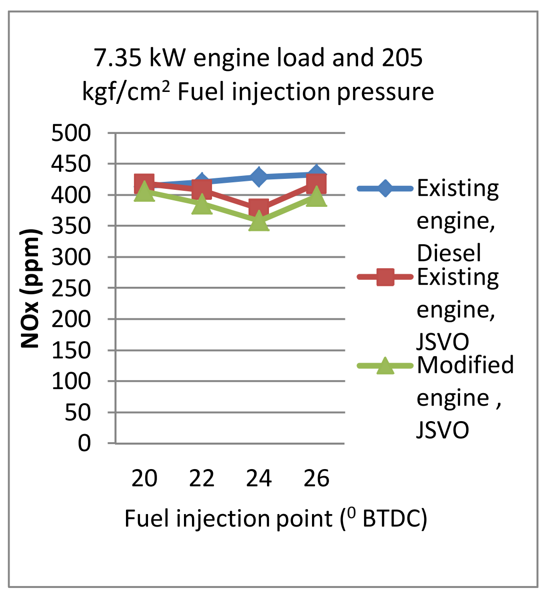

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Figure 4.30: NO_x emission vs. Advance angle of fuel injection

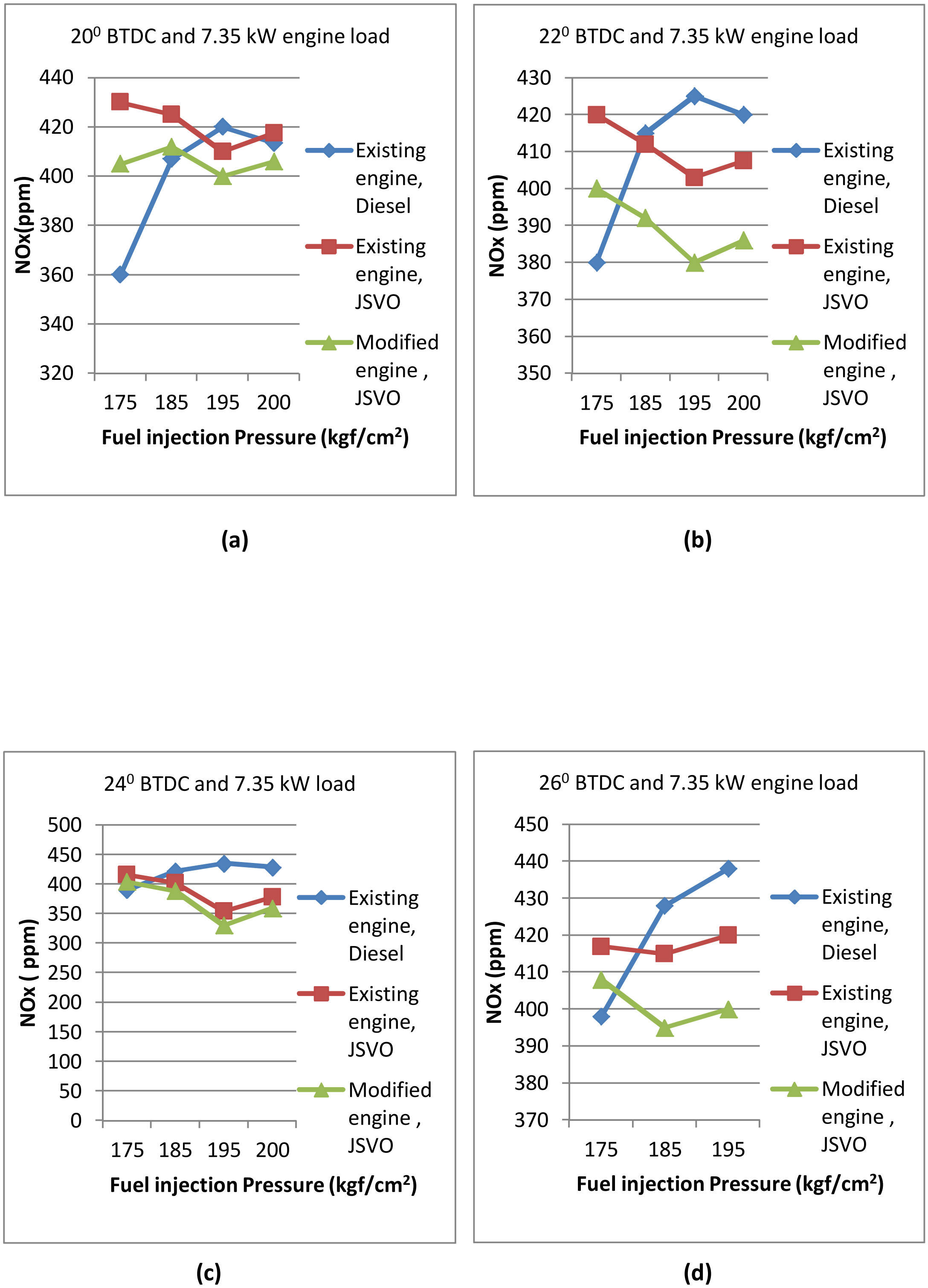


Figure 4.31: NO_x vs. Fuel Injection Pressure

4.4.4 Smoke Emission

4.4.4.1 Effect of varying fuel injection point at constant fuel injection pressure

The graph was plotted between smoke emission (opacity) and varying fuel injection point for the fuel injection pressures 175 kgf/cm^2, 185 kgf/cm^2 , 195 kgf/cm^2 and 205 kgf/cm^2. The smoke emission was found to increase continuously for diesel as the angle of fuel injection were increased from 20^0 BTDC to 26 ^{0} BTDC in step of 2^0 for all the load conditions (figure 4.36). The Smoke emission for JSVO when fueled in existing engine was observed to be high at 20^0 BTDC in comparison with Smoke emission of diesel at same point. The smoke emission was decreased in the case of JSVO with increase in advance angle of fuel injection up to the angle of 24^0 BTDC at all the load conditions but the Smoke emission started increasing as the angle was further increased to 26^0 BTDC. This indicated that the combustion efficiency was improved in case of JSVO as the fuel injection angle was increased up to 24^0 BTDC but the combustion efficiency started deteriorating as the angle was further increased (Figure 4.36). It was also found that the Smoke emission of diesel fuel at all the angles of fuel injection was lower than the smoke emission of JSVO. The smoke emission of the JSVO when fueled in the modified engine was also decreased as the angle of fuel injection was increased from 20^0 BTDC to 24^0 BTDC in step of 2^0 for all the load conditions but the smoke emission started increasing as the angle was further increased to 26^0 BTDC (figure 4.36). It is remarkable that the Smoke emission at recommended operating point for JSVO (24^0 BTDC, 195 kgf/cm^2) was least.

4.4.4.2 Effect of varying fuel injection pressure at constant fuel injection point

It was observed that the Smoke emission in case of diesel fuel was increased as the fuel injection pressure was increased from the rated value of 175kgf/cm^2 to 185kgf/cm^2 ,195kgf/cm^2 and 205 kgf/cm^2 (figure: 4.37). The graph also indicates that the Smoke emission kept reducing as the load was increased from 0 kW to 7.35 kW (figure 4.32 to figure 4.35). The decrease in smoke emission with increase in advance angle of fuel injection was obtained at all load conditions up to 24^0 BTDC. On comparison, this was also observed that the smoke emissions of diesel at all the load conditions were less than the smoke emission of JSVO. The smoke emission of JSVO when fueled in modified engine was observed to be less than the smoke emission of JSVO when fueled in existing engine.

4.4.4.3 Effect of Constant Fuel Injection point and fuel injection pressure

From the graphs it was found that the smoke emission was less at all the operating point in the case of JSVO fueled in modified engine than the smoke emission of JSVO fueled in existing engine. This shows that the fuel pre-heating reduced the Smoke emission. This may be because of improved fuel combustion quality as the viscosity of fuel decreased by pre-heating. On comparison the amount of smoke emission when the JSVO was fueled in existing engine and modified engine the amount of emission was observed to be reduced which directly related to better fuel combustion quality. Though the better combustion quality increases the temperature which should increase the smoke emission but this effect might have been dominated by the decrease in temperature due to fuel injection advance angle. That means the preheating of fuel was an advantageous modification which improved the performance characteristics as discussed earlier and also the emission characteristics.

4.4.4.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point

The variations of smoke emission with change in fuel injection point and fuel injection pressure in both the cases i.e. in modified engine and existing engine was compared. The comparison exhibited the result that the smoke emission characteristics of JSVO fuel was improved by implementing higher degree (24^0)

of fuel injection and 195 kgf/cm^2 pressure with a system of fuel pre-heating incorporated in the engine fuel supply system.

In association with the effect of fuel injection advance angle and the increase in fuel injection pressure the pattern of decrease in smoke indicated improvement in combustion quality of JSVO in combustion chamber. It was found that the emission performance of the JSVO fuel improved as the advance angle of fuel injection was increased from the rated value of 20^0 BTDC upto24^0 BTDC but deteriorated on further increase to 26^0 BTDC. The similar trend was shown on increasing the fuel injection pressure from the rated value of 175kgf/cm^2 .

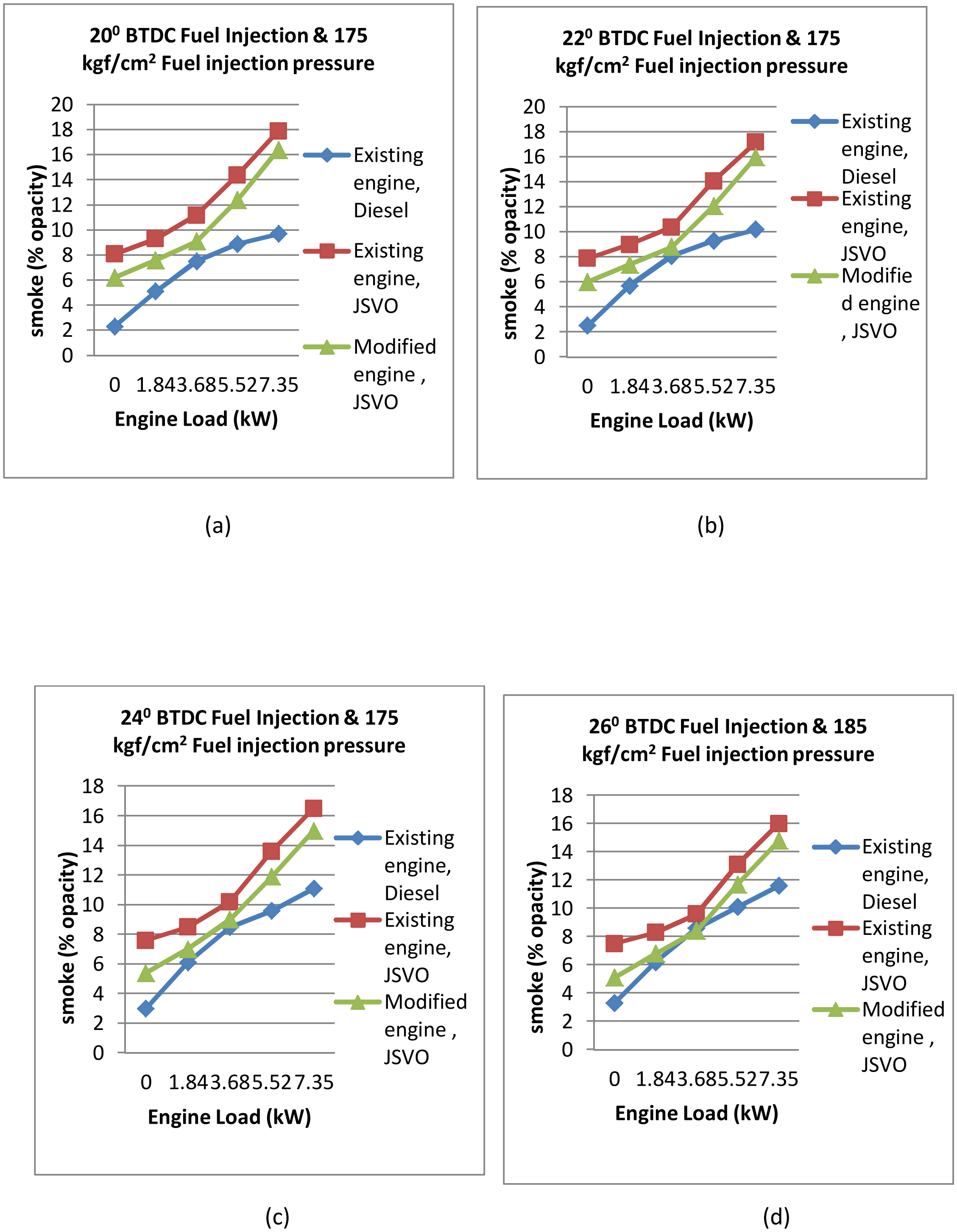
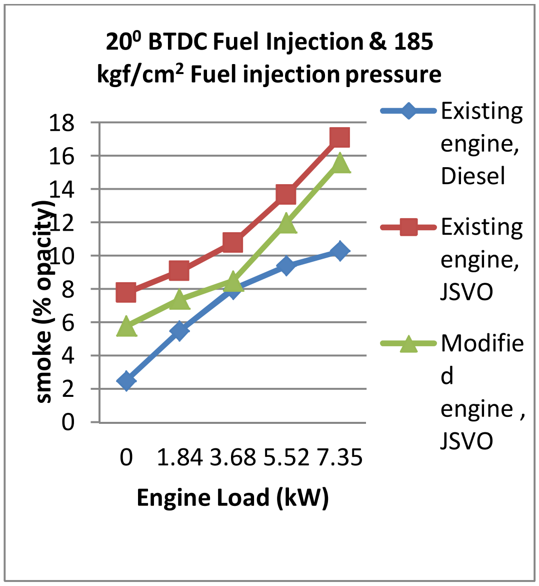


Figure 4.32 : Smoke emission vs. Engine Load



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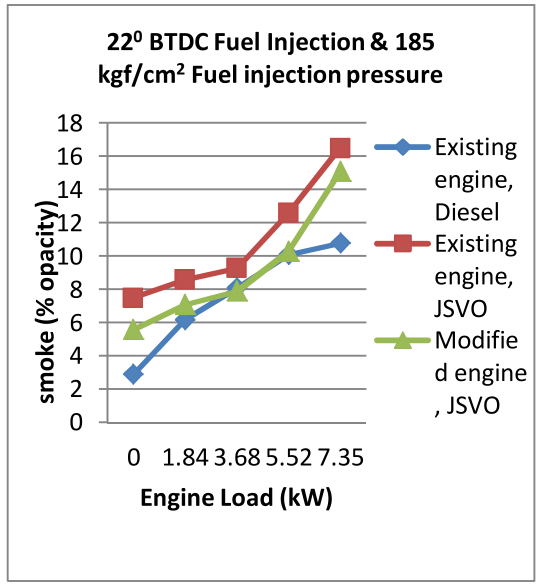

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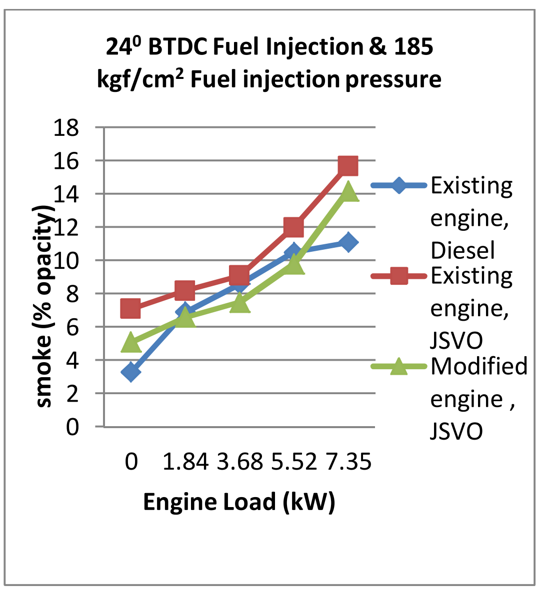

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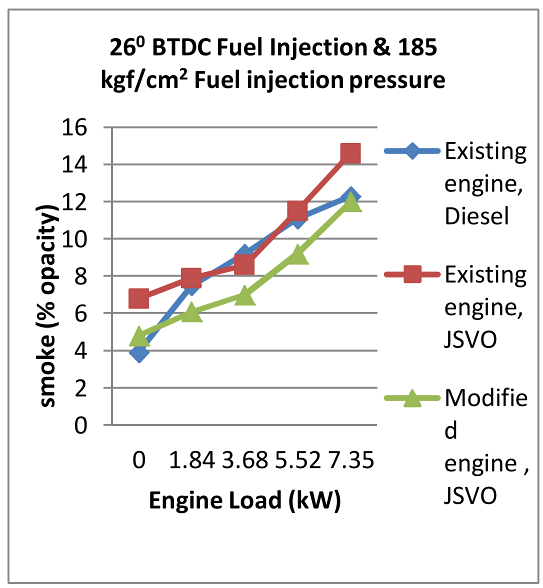

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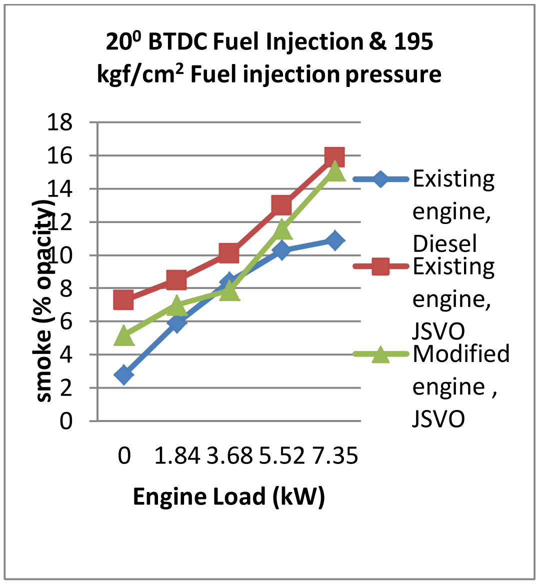
Figure 4.33 : Smoke emission vs. Engine Load



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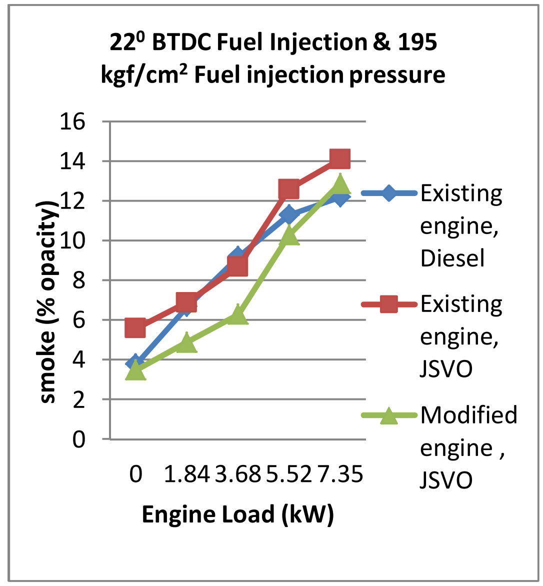

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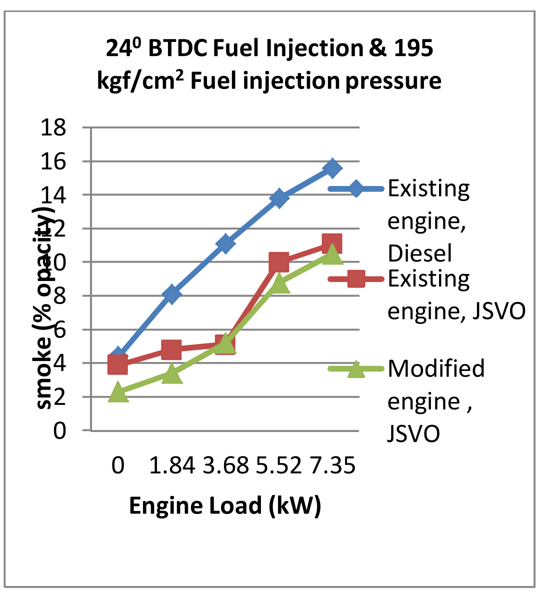

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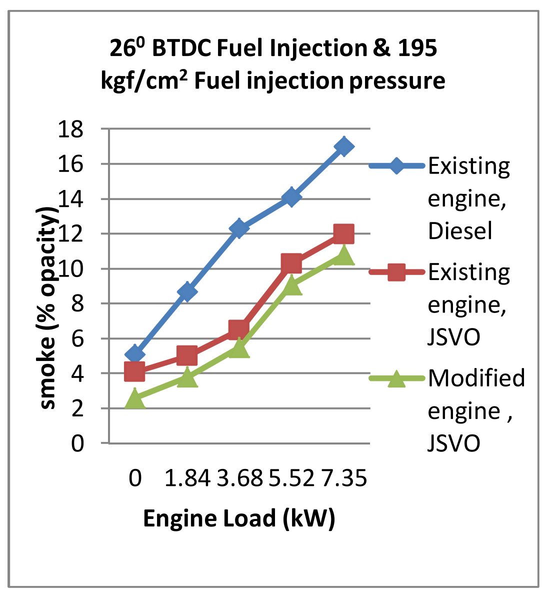

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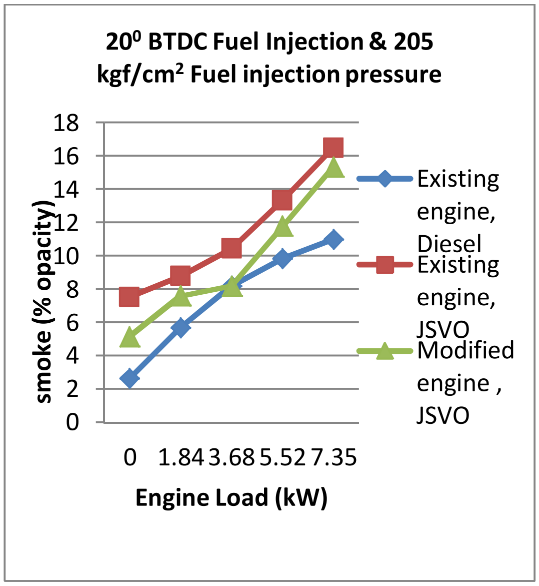
Figure 4.34 : Smoke emission vs. Engine Load



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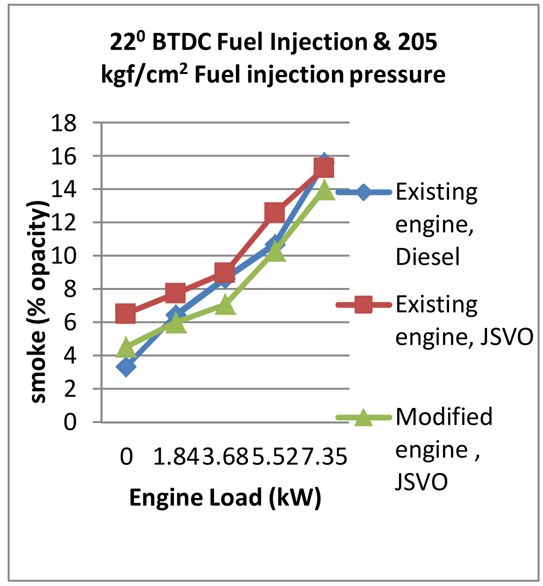

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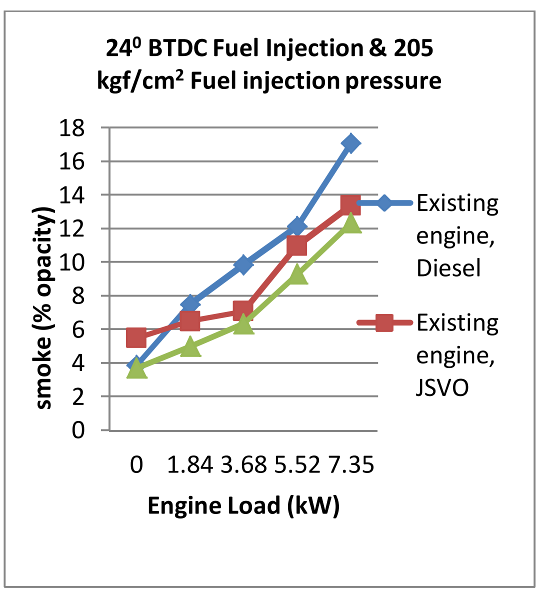

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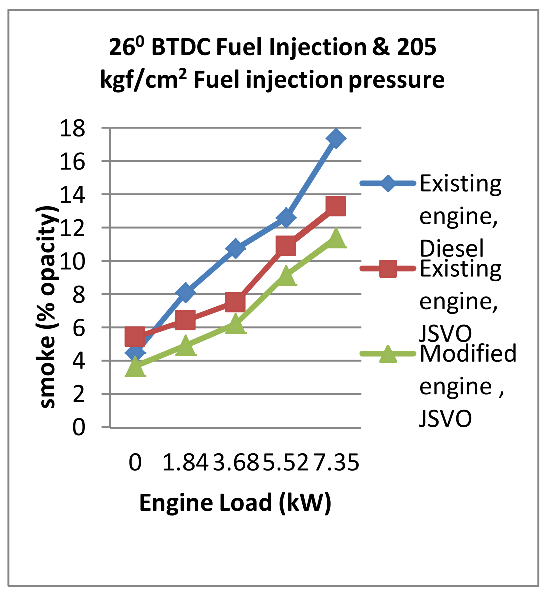


(d)

Figure 4.35 : Smoke Emission vs. Engine Load

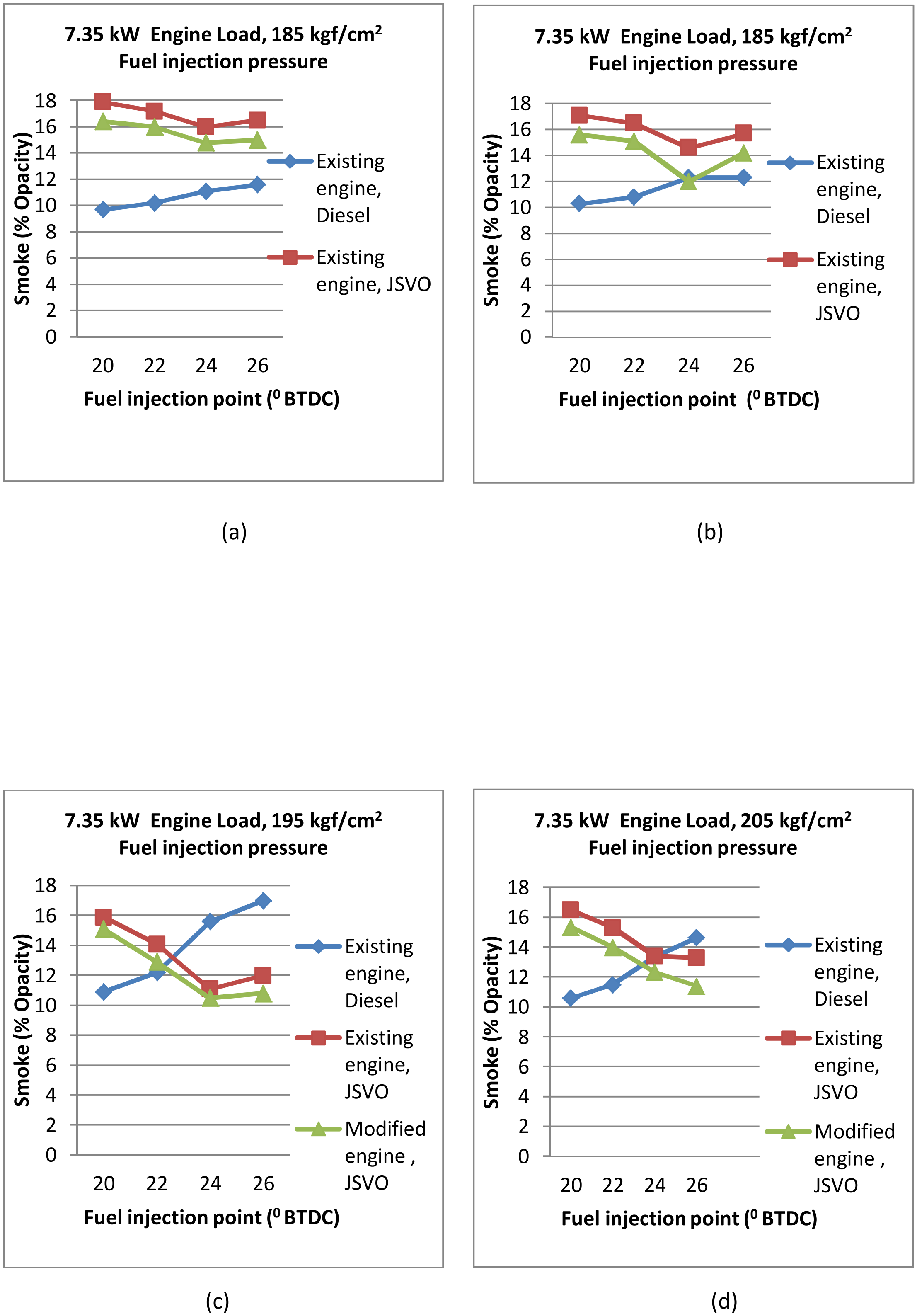


Figure 4.36: Smoke emission vs. Advance angle of fuel injection

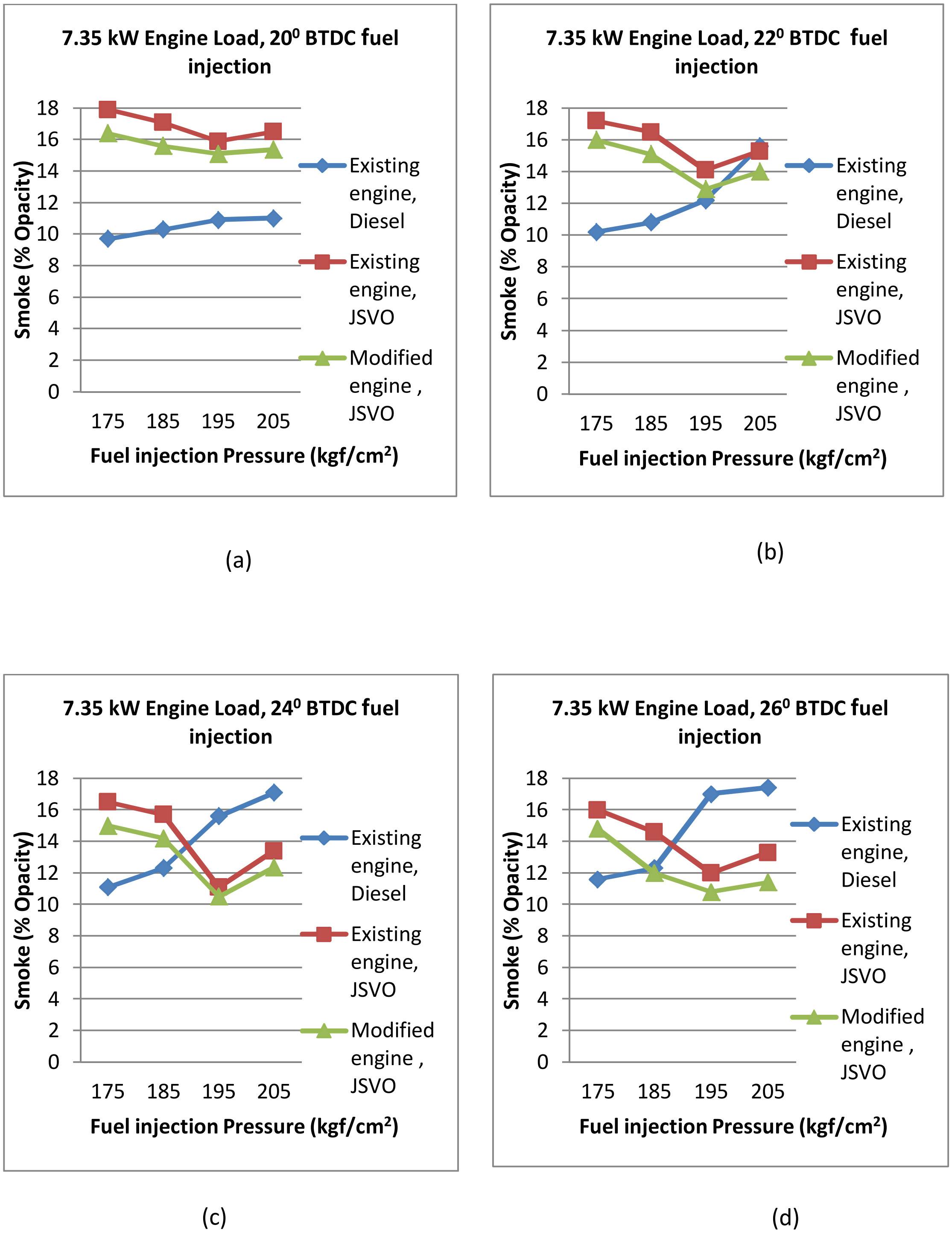


Figure 4.37 : Smoke Emission vs. Fuel injection pressure

4.4.5 HC Emission:

4.4.5.1 Effect of varying fuel injection point on HC emission at constant fuel injection pressure

The viscosity directly affects the HC emission quality of fuel. Because of higher viscosity the injector releases the fuel droplets of larger diameter, which needs more heat as well as more time for proper vaporization. If the advance angle of fuel injection is increased the fuel atomization quality increases and the combustion efficiency increases. The increased combustion efficiency results to low HC emission.

It was observed that at 20^0 BTDC the HC emission was lowest for diesel fuel, where as the modified engine fueled with JSVO was emitting the HC more than the engine fueled with diesel but less than the JSVO fueled in existing engine. The HC emission for modified as well as existing engine fueled with JSVO was reduced up to 24^{0} BTDC but it started increasing when the angle was further increased. The gap between them was continuously reducing as the advance angle of fuel injection was increased because the combustion efficiency of diesel might have started decreasing at higher advance angle but the combustion efficiency of JSVO started increasing resulting low HC emission (figure 4.42).

4.4.5.2 Effect of varying fuel injection pressure on HC emission at constant fuel injection point

The HC emission of diesel increased as the fuel injection pressure was increased from 175 kgf/cm^2 to 185 kgf/cm^2 ,195 kgf/cm^2 and 205 kgf/cm^2 (figure 4.43) . It shows that the combustion quality deteriorated by using the diesel in existing engine at higher pressure than the design value of this engine. The HC emission of JSVO was decreased as the fuel injection pressure was increased this may be due to the fact that the fuel particle size at higher injection pressure gets reduced and the combustion efficiency gets increased. It was found that the HC emission even at higher value of fuel injection pressure of 195 kgf/cm^2 increased at 26^o

BTDC advance angle. It shows that the effect of increasing the HC emission at high advance angle pre dominated the effect of decreases of HC emission at high injection pressure.

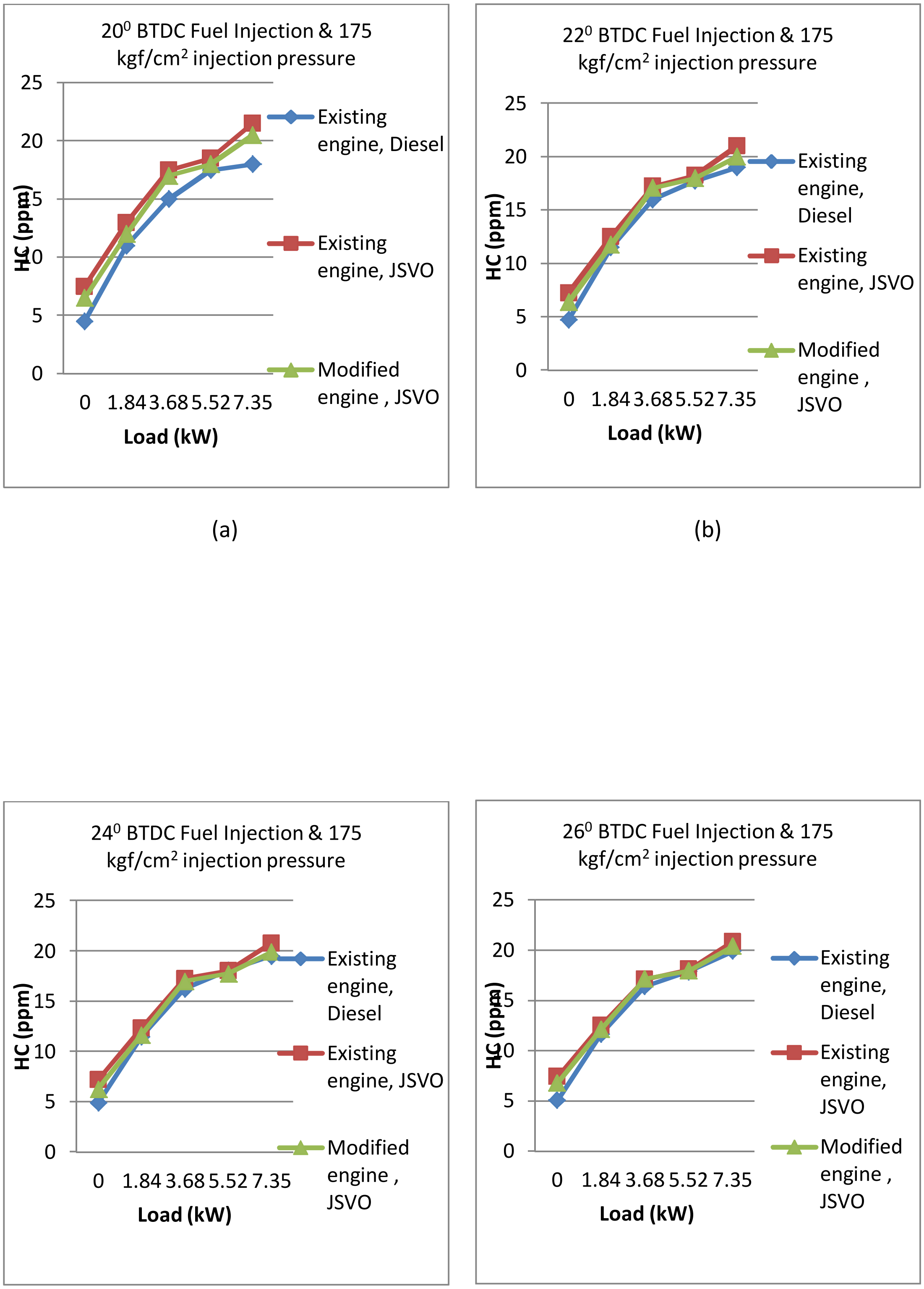
4.4.5.3 Effect of fuel preheating at constant fuel injection point and fuel injection pressure

The HC emission of the JSVO when fueled in existing engine was observed to be higher than the HC emission of JSVO when fueled in modified engine (figure 4.38 to figure 4.43). In modified engine the fuel was preheated so the viscosity was reduced , the reduced viscosity enhanced the fuel injection efficiency so the HC emission was reduced. At all the point of engine running it was found that the HC emission was better in the case of preheating the fuel.

4.4.5.4 Combined effect of fuel preheating, fuel injection pressure and fuel injection point on HC emission

The study of variations in HC emission with change in fuel injection point and fuel injection pressure in both the cases i.e. modified engine and existing engine exhibited the result that the HC emission characteristics of fuel JSVO was improved by implementing higher degree (24^0) of fuel injection and 195 kgf/cm^2 pressure with a system of fuel pre-heating incorporated in the engine fuel supply system. But it was observed that the HC emission increased as the advance angle was increases further.

In association with the effect of fuel injection advance angle and the increase in fuel injection pressure the pattern of decrease in HC indicated the improvement in combustion quality of JSVO in combustion chamber. It was found that the emission performance of the JSVO fuel was improved as the advance angle of fuel injection was increased from the rated value of 20^0 BTDC upto24^0 BTDC but it started deteriorating on further increase to 26^0 BTDC. The similar trend was shown on increase in fuel injection pressure from the rated value of 175kgf/cm^2.



(c) (d)

Figure 4.38: HC emission vs. Engine Load

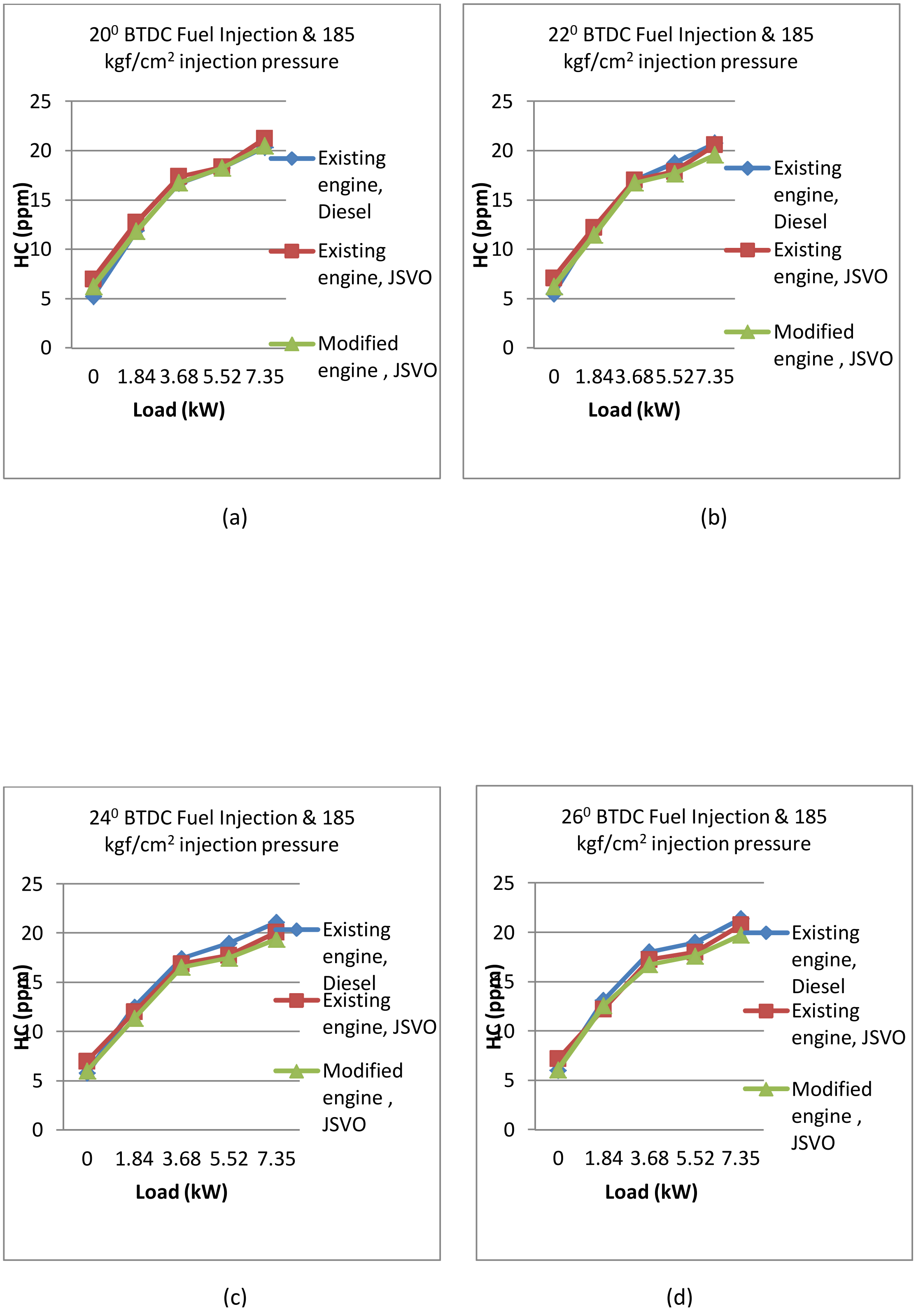
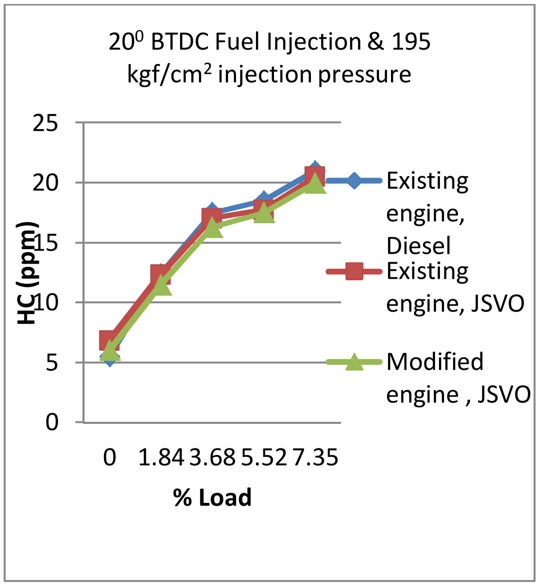


Figure 4.39: HC emission vs. Engine Load



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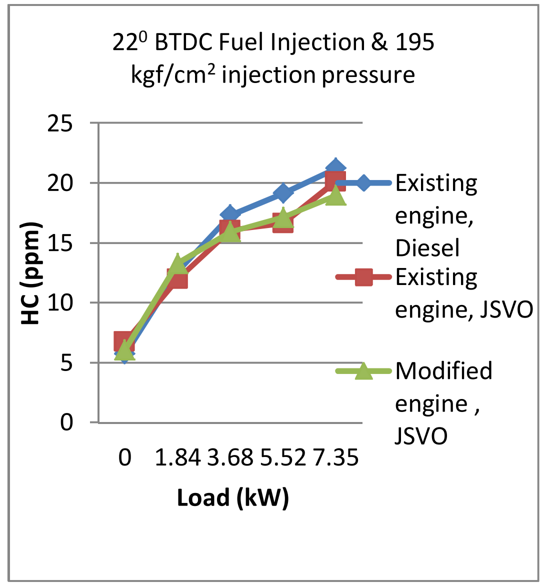

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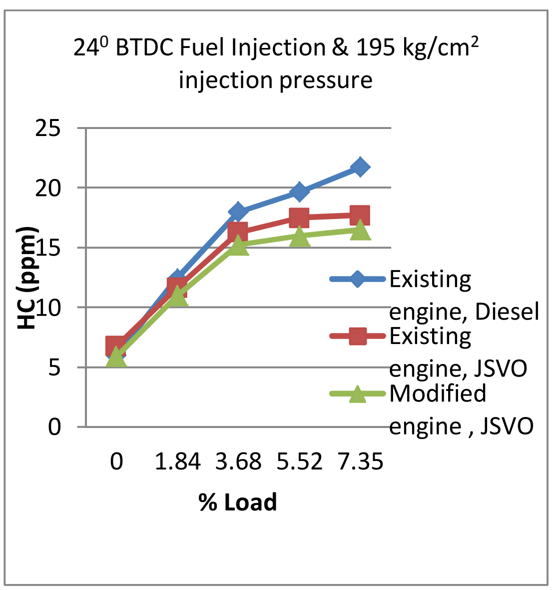

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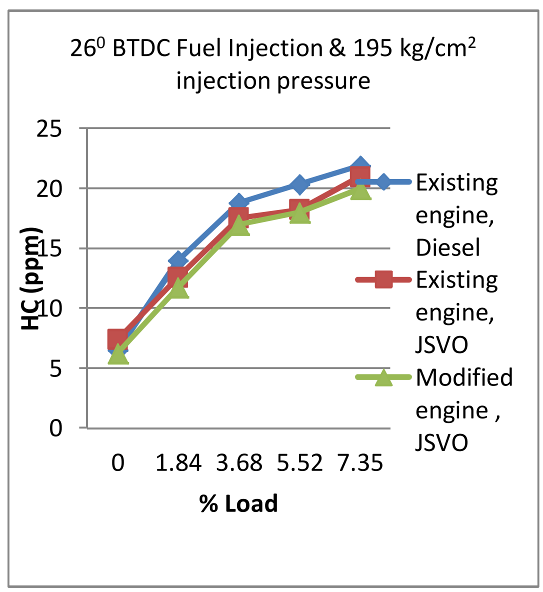

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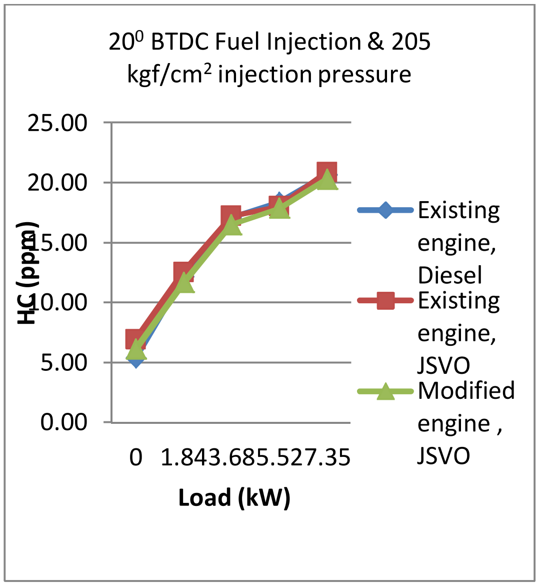
Figure 4.40: HC emission vs. Engine Load



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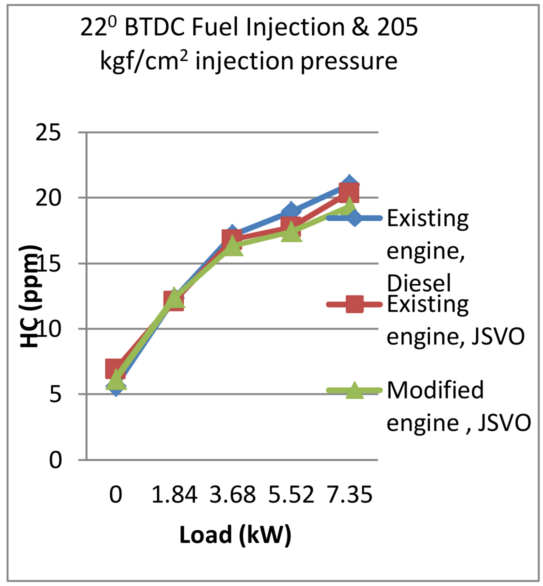

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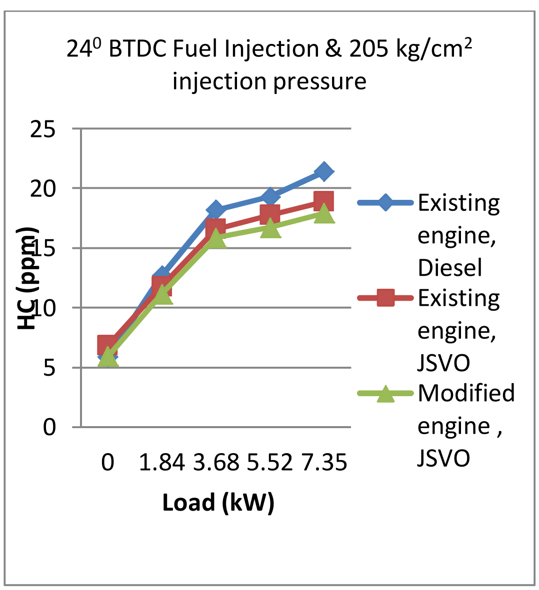

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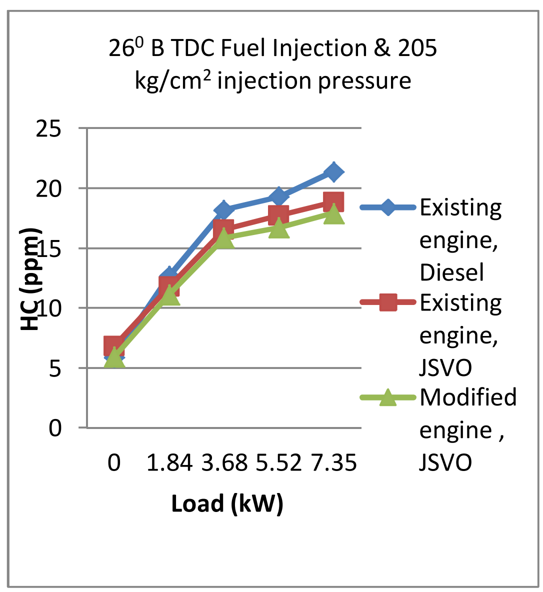

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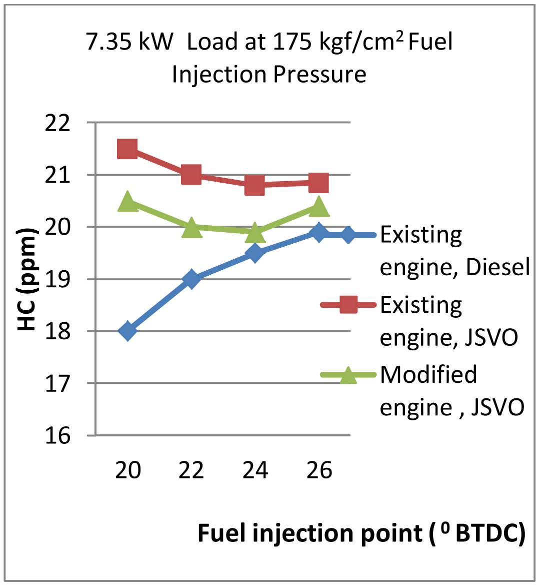
Figure 4.41: HC emission vs.Engine Load:



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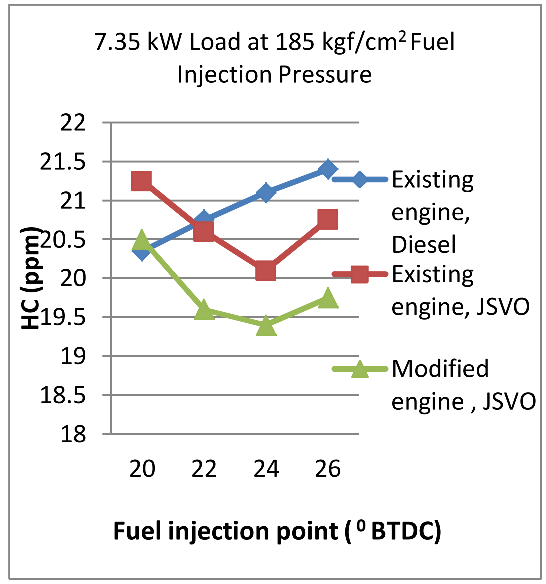

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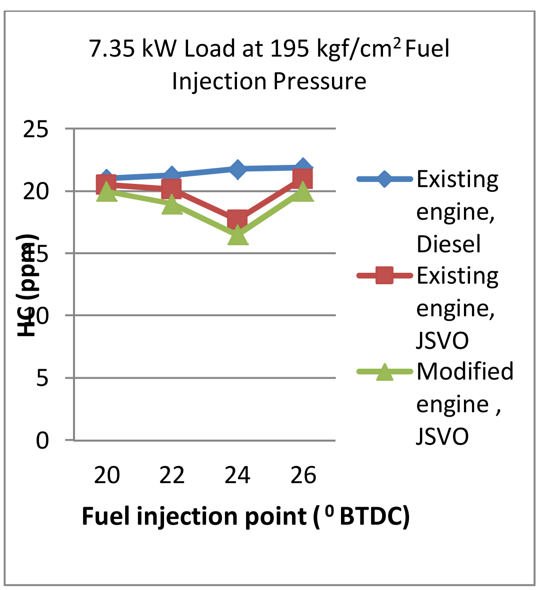

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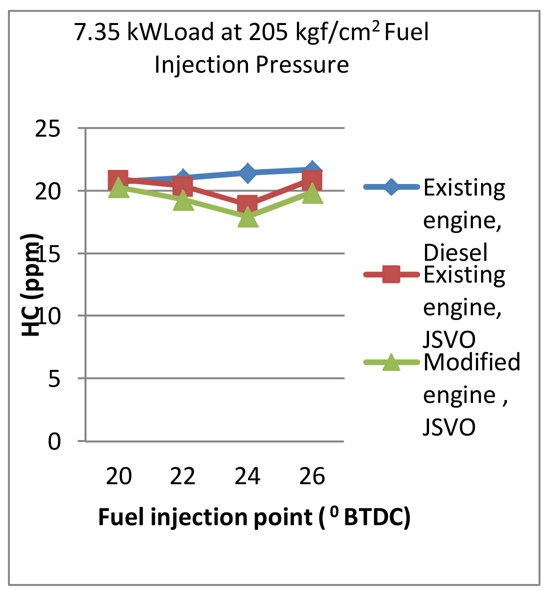

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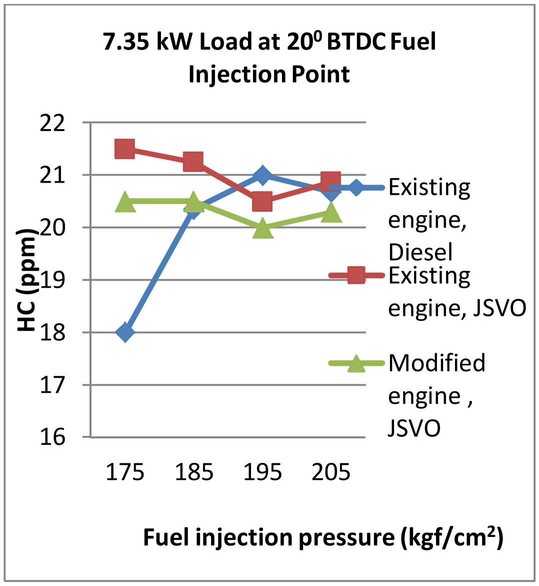
Figure 4.42: HC emission vs. Advance angle of fuel injection



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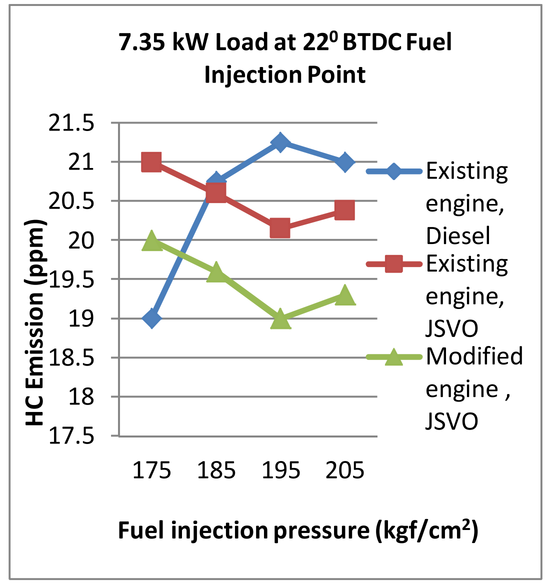

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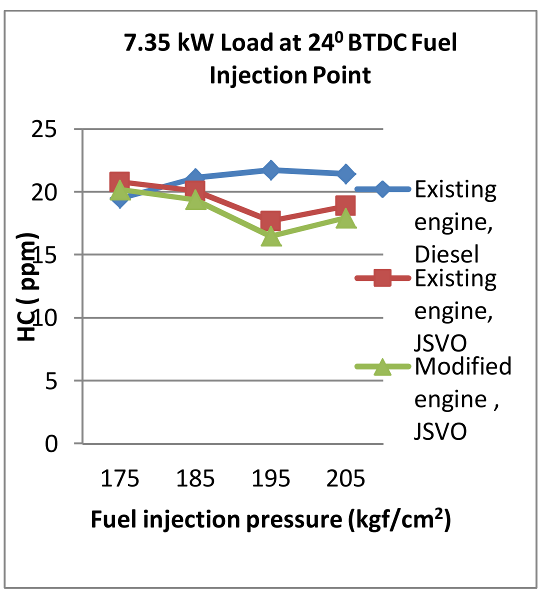

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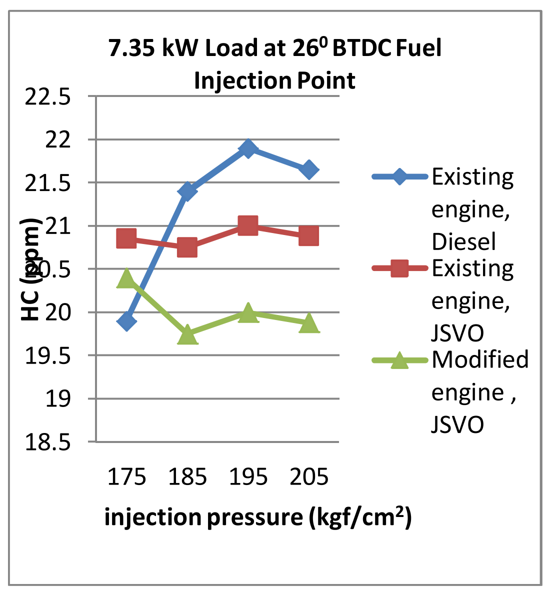


(d)

Figure 4.43: HC emission vs. Fuel injection pressure

4.5 Performance evaluation of existing and modified engine

The existing engine was operated to record the base line data required for comparison with engine performance with JSVO fuel. There were many sets of operating parameter used for testing the engine. Different fuel injection points, fuel injection pressures were used to test the engine with and without pre heating the fuel. After the analysis of data it was found that the engine incorporated with fuel preheating operating at 195 kgf/cm^2 fuel injection pressures and 24^0 BTDC advance angle of fuel injection point gives good results in comparison with existing engine running with diesel fuel. In this chapter the results obtained from the tests of modified engine running with JSVO at recommended operating parameter is compared with base line test results of engine.

4.5.1 Performance of existing engine fueled with JSVO

The existing low speed IDI engine manufactured by field marshal was tested with diesel and JSVO at 20^0 BTDC fuel injection point and 175 kgf/cm^2 fuel injection pressure. These parameters were the manufacturer's recommended operating parameters for this engine with diesel fuel. The engine performance parameters as obtained from the experiments at 7.35 kW load on the engine are listed in table 4.1. The analysis of above listed parameters of the engine shows that the efficiency of the engine 30.62 % with JSVO was quite low in comparison with BTE of engine with diesel fuel which was 32.96% (figure 4.44). The cetane number is the scale for comparing the ignition delay angle of fuel. The higher CN means a lower delay period and smoother engine operation. The cetane number depends upon chemical composition of fuel. If more paraffinic hydrocarbons are contained in the fuel, the cetane number will be higher. The cetane number of JSVO was 38 in comparison with 48 as cetane number of diesel. The fuel with low cetane number requires large time delay and maximum rate of pressure rise. The other important phenomena are fuel atomization which governs the maximum temperature rise in combustion chamber. The larger fuel particle released by the

Table 4.3: Performance parameters of engine at 7.35 kW load

Performance parameters	Diesel fuel	JSVO
BTE (%)	32.96	30.62
BSFC (kg/kWh)	0.260	0.301
CO emission(% vol)	0.05	0.10
CO2 emission(% vol)	1.24	0.76
NO_x emission (ppm)	360	430
HC emission (ppm)	18	21.5
Smoke emission (% opacity)	9.7	17.9

fuel injector due to higher viscosity needs more heat and time to get atomized properly. Since the viscosity of JSVO was 10-12 times the viscosity of diesel so the JSVO also may have required higher heat and delay period. In scarcity of these factors the pre combustion process might be inefficient and would have negative effect on fuel combustion efficiency. The inefficient fuel combustion might have reduced the brake thermal efficiency. The BSFC of engine with JSVO was recorded to be 310 kg/kWh which was much higher in comparison with BSFC (0.260 kg/kWh) for diesel fuel (figure 4.45). The emission parameters of JSVO was poor in comparison with diesel fuel for CO, CO , NO_x , HC and smoke (figure 4.46). These results made the JSVO unfit for use in existing engine. The feasibility of JSVO as efficient alternate fuel was only in modified engine. The existing engine can also be run by fuelling it with processed JSVO. The fuel properties of JSVO will improve by chemical processing. In this work the first method was utilized. The existing engine was modified to have advance angle of fuel injection 24^0 BTDC, fuel injection pressure 195 kgf /cm^2 and also by preheating of fuel. The engine was tested and the results obtained are discussed in following paragraph.

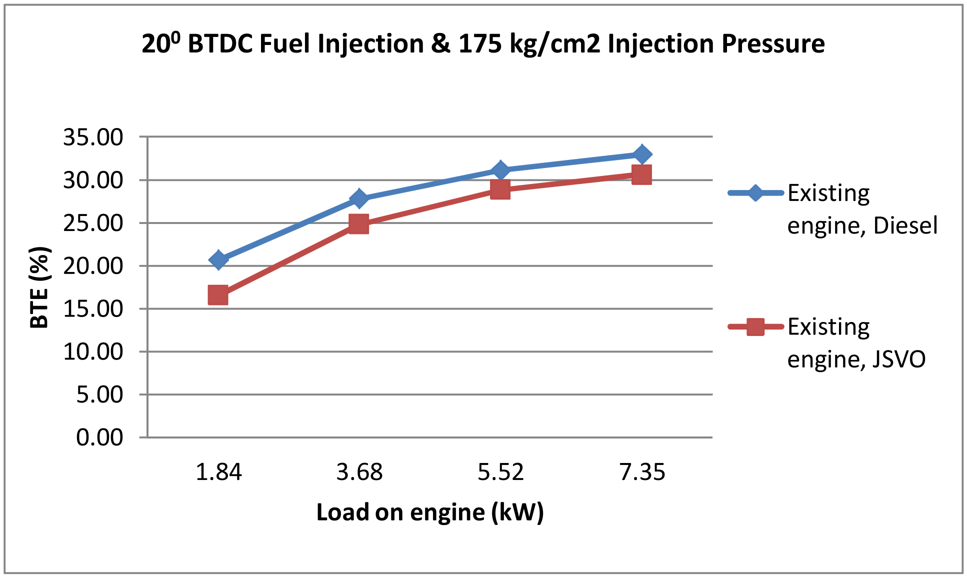


Fig. 4.44: BTE vs Engine Load

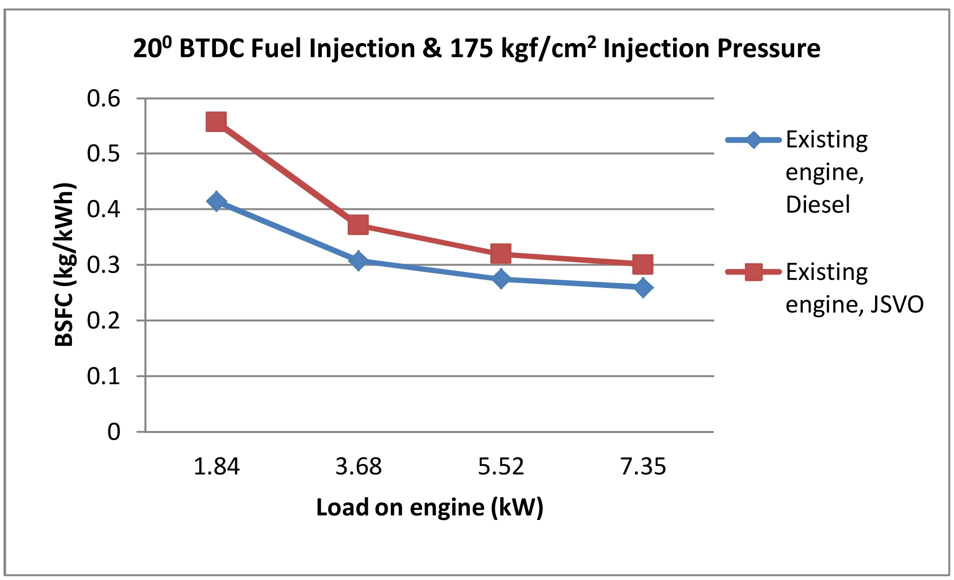
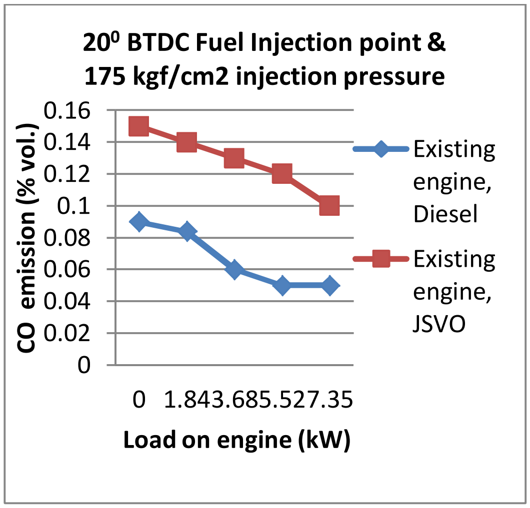


Fig. 4.45: BSFC vs Engine Load



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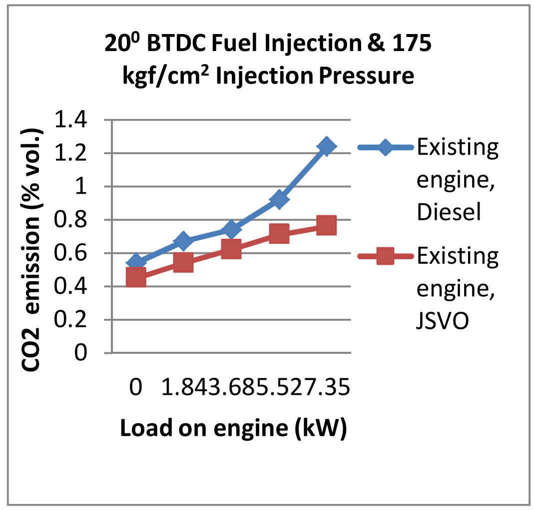

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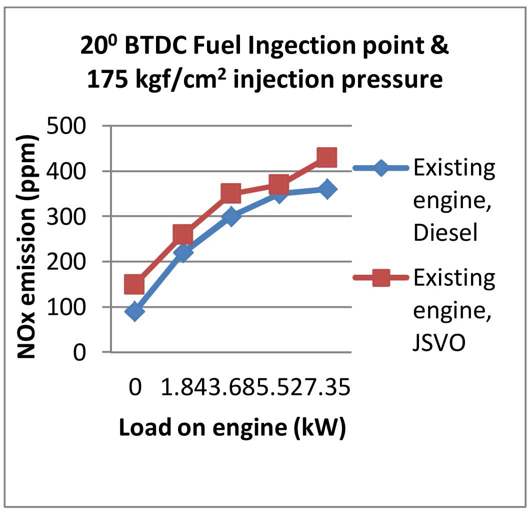

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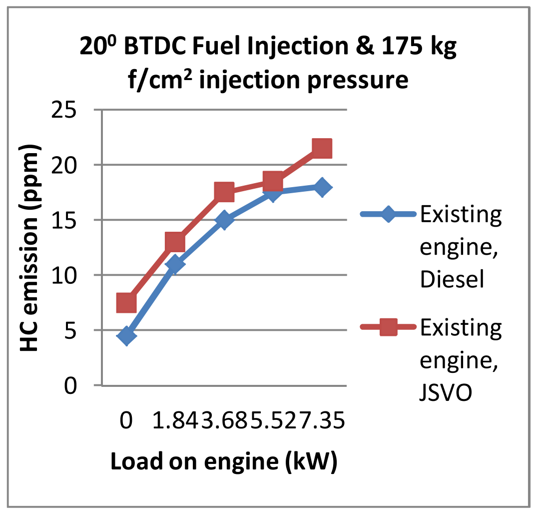

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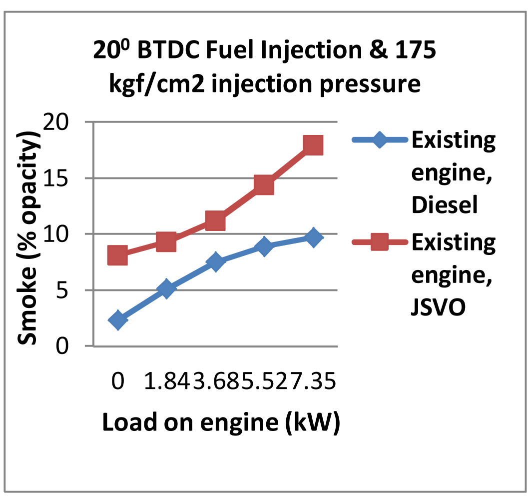


Fig. 4.46: Emission vs Engine Load

4.5.2 Comparison of existing engine performance fueled with diesel and modified engine fueled with JSVO

The performance of existing engine fueled with JSVO at 20^0 BTDC fuel injection point and 175 kgf/cm^2 fuel injection pressure was poor in comparison with engine performance with diesel fuel at same operating parameters. The modifications on advance angle of fuel injection and fuel injection pressure with preheating the fuel to 90^0 C improved the engine performance. The recommended advance angle of fuel injection was 24^0 BTDC and the fuel injection pressure 175 kgf/cm^2 for JSVO to be used as alternate fuel in engine. The comparative engine performance parameters of modified engine to be fueled with JSVO and existing engine fueled with diesel is shown in figure 4.47. The improvements in engine performance after the modification of advance angle of fuel injection point and the fuel injection pressure with heat exchanger to preheat the JSVO to 90^0 C can be seen at a glance from the table 4.2. BTE of modified engine was obtained to be 34.76% which is an improvement of 4.14% on BTE of existing engine with JSVO fuel. It can be seen that the BTE of modified engine with JSVO test fuel was improved to such an extent that it was not only equal but higher than the BTE of existing engine with diesel fuel.

The BSFC was reduced in both the cases as the engine load was increased. The BSFC of existing engine with diesel fuel was found 0.260 kg/kWh. The BSFC of modified engine fueled with JSVO was obtained 0.266 kg/kWh. (figure 4.48). It can be concluded that the modification reduced the BSFC substantially but it was still marginally higher than the BSFC of base line engine.

The emission parameters (CO, CO_2, NO_x, HC and smoke) of modified engine fueled with JSVO and existing engine fueled with diesel were compared (figure 4.49) . It was found that CO emission was improved and it became almost equal to the base line data. The CO_2 emission of existing diesel engine was higher than the modified engine with JSVO fuel but the difference was reduced substantially.

The NO_x emission of modified engine was found to be reduced to an extent even less than emission of existing engine with diesel fuel.. The HC and smoke emission were also improved to very good extent.

Table 4.4: Comparative performance parameters of engine at 7.35 kW load

Performance parameters	Existing engine , Diesel fuel	Existing engine, JSVO fuel	Modified engine, JSVO fuel
BTE (%)	32.96	30.62	34.76
BSFC (kg/kWh)	0.260	0.301	0.266
CO (% vol)	0.05	0.10	0.05
CO2 (% vol)	1.24	0.76	1.08
NO_x (ppm)	360	430	330
HC (ppm)	18	21.5	16.5
Smoke (% opacity)	9.7	17.9	10.5

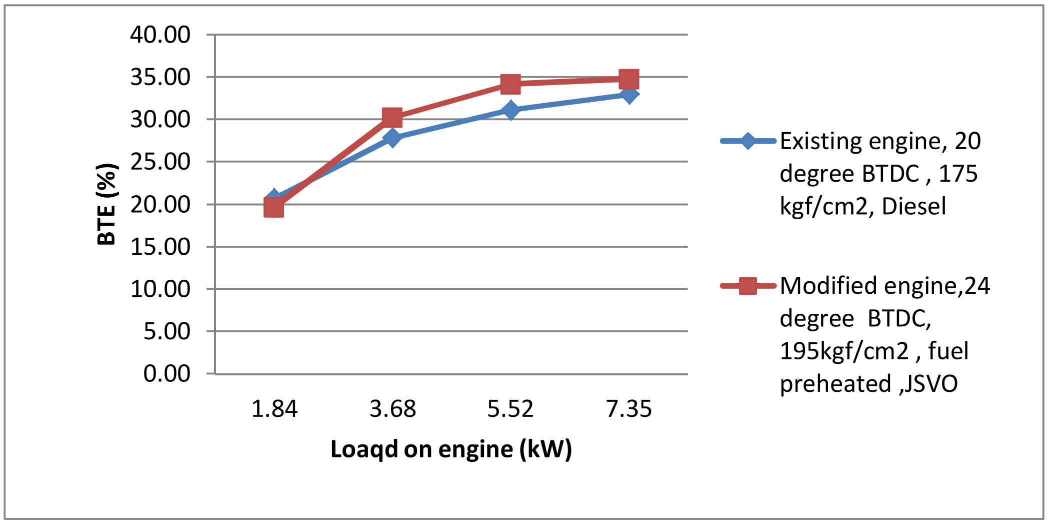


Fig. 4.47: BTE vs Engine Load

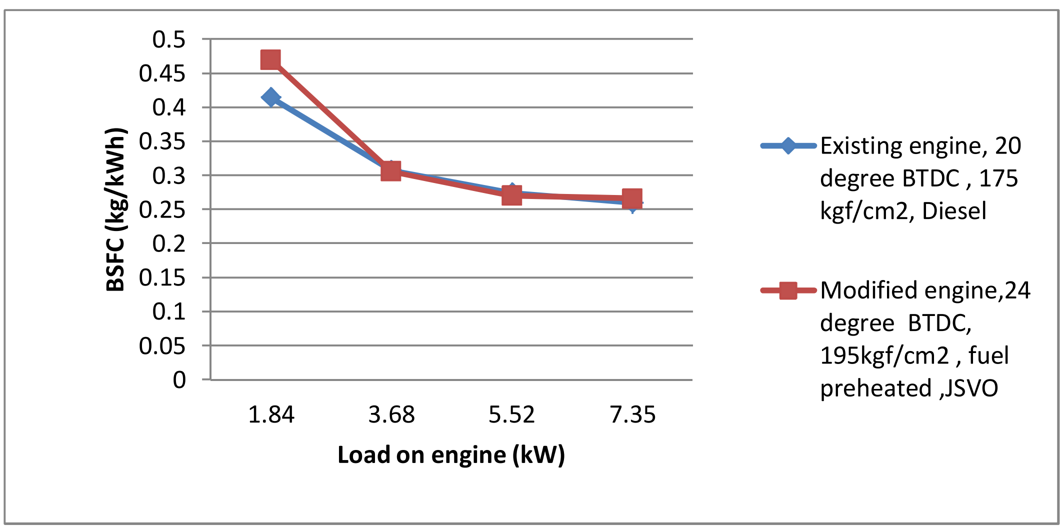
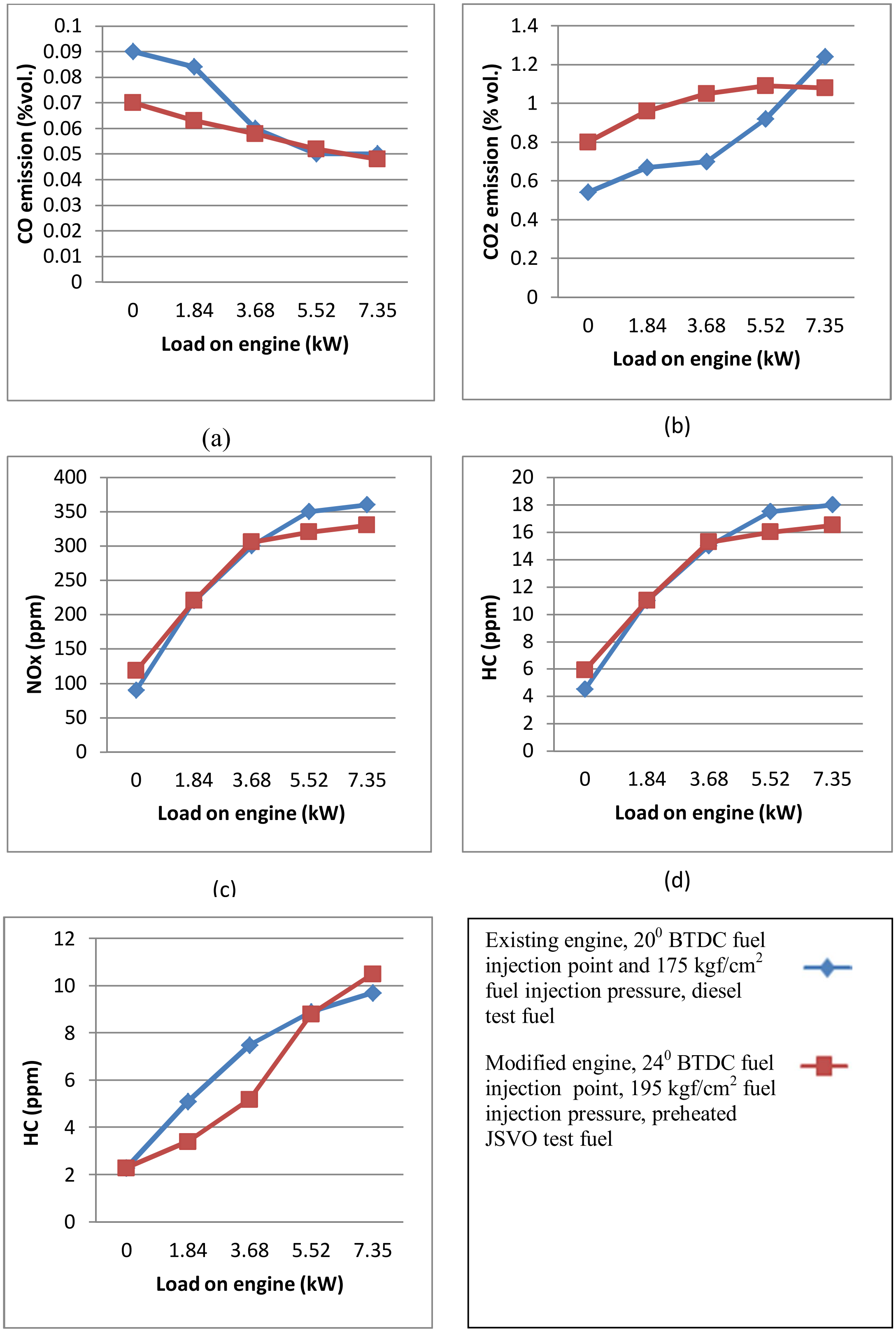


Fig. 4.48: BSFC vs Engine Load



(e) Figure. 4.49: Emission parameters vs Load

4.5.3 Techno economic evaluation of modified engine

The technical aspects of engine modifications are not complicated; it can be done locally for the engines already in use and by the manufacturer at the production level. The modification of engine can be done by the users also after short term training. This will retain the possibility of using the engine with petro diesel also as and when it becomes essential. The financial implications involved in modification are almost negligible.

Based upon the market survey the cost of Jatropha seeds at present is about Rs 20 per kg. The oil yield of the seed which was obtained in our laboratory was 28%. The oil yield of the seeds depend upon factors such as Jatropha species , plantation time, expellers efficiency etc. The production cost of 1 litre JSVO including power used by the expeller and the labor involved is about Rs 75 under present market value. The use of JSVO in scenario of scarce availability of seed and high price projects lean feasibility for industrial and agricultural applications. The cost of one liter JSVO is much higher than the cost of diesel .The cost of JSVO can be expected to reduce as the farmers get encouraged to cultivate jatropha at large scale. The production of oil by the farmers at site will not cost to this extent because there will not be any cost of packing, storage, transportation, traders profit and all other financial liabilities of market.

The engine operating cost and other associated maintenance requirements are not included in this study. The researchers have reported the possibilities of injector choking, gum formation, piston sticking and carbon deposit under the endurance test. These symptoms may increase maintenance and engine operating cost. The test conducted by these researchers have either not utilized the modifications, if utilized then the modifications was carried out partially. The engine performance will improve on these reported symptoms if the engine is run with modification operating at recommended operating parameters. The combustion quality, emission quality and ultimately the behavior of engine has improved to large extent by incorporating the modifications. The improvements in efficiency, BSFC and emission characteristics indicate that the engine will be running trouble free for recommended life span without any change in its maintenance protocol. This shows that there will not be any additional maintenance cost involved by using the JSVO in this engine on regular basis. However the detailed study on these parameters under endurance test is yet needs to be carried out in future research work.

4.5.4 The engine operation on long term basis

The engine after the modification will be functioning efficiently without any starting trouble. Engine starting trouble was not encountered at the time of experiments. Since the experiments were conducted at an ambient temperatures 25^0 C-30^0 C even without any modification of fuel injection system and without preheating the fuel. It can be expected that the staring trouble will not arise even during winter season and also at cold climatic places. However, if it is there then the engine can be started with diesel just for a while and later it can be changed over to neat JSVO.

4.5.5 Socio economic evaluation of modified engine

The use of JSVO as alternate fuel will not only reduce the load on petroleum fuel reserves but also help in global objective of having clean and environmental friendly fuel. The scope of usage of JSVO in engine will encourage the farmers to cultivate the Jatropha. The Jatropha is primarily cultivated at barren land so the land utilization and revenue generation of farmers will increase. The production of JSVO at user end will make them self dependent. The trouble of the farmers to go to market for purchase of diesel will reduces to maximum extent. The self dependence of farmers for the procurement of engine fuel becomes more important for the users residing in remote areas. The barren land utilization by the farmers will enhance their prosperity and generate he employment opportunities to local laborers.

CHAPTER 5 CONCLUSION AND SCOPE FOR FUTURE WORK

The experimental work carried out in this study were analyzed and the results were discussed in previous chapters. The major findings are listed in this chapter:

(i) It is found that viscosity of JSVO at 90^0 C is almost equal to the viscosity of diesel at room temperature. The engine efficiency at maximum engine load and JSVO fuel has improved from 30.62 % to 31.04%.

(ii) Researcher in general installed the heat exchanger before the fuel pump and some external source of heat were used to heat the JSVO for reducing the viscosity. In the present investigation, by installing the heat exchanger between the fuel pump and fuel injector the heat dissipation losses in the section have been avoided. Also by using the heat of the exhaust gases for preheating the JSVO need of some external heat source is completely eliminated.

(iii) The analysis of emission constituent (CO, CO2, NO_x, HC and smoke) on fuel preheating shows that all the emission characteristics of engine improves as the fuel is preheated. Similar trend were observed from both the engines i.e. existing engine as well as modified engine with JSVO fuel and at all the operating points.

(iv) The existing engine at rated engine load with diesel fuel has maximum efficiency 32.96% at 20^0 BTDC and 175 kgf/ cm^2 which deteriorated as the injection angle was advanced and the efficiency was reduced to 31.25% at $26\,^0$ BTDC. This shows that the optimum fuel injection point of existing engine with diesel fuel is 20^0 BTDC.

(v) It was found that the engine efficiency with JSVO fuel at rated fuel injection point (20^0 BTDC) and fuel injection pressure 175kgf/cm^2 is 30.62% which gets increased to 31.04% at 24^0 BTDC at the same fuel injection pressure. The efficiency gets deteriorated to 30.21% by further increase in advance angle of fuel injection to 26^0 BTDC. The decrease in efficiency by advancing the angle beyond 24 ^{0} BTDC may be due to the possibility of knocking on excessive advance angle.

(vi) The emission constituents (CO, CO2, NO_x, HC and smoke) on varying advance angle shows that the emission characteristics improves as the advance angle of fuel injection is increased from 20^0 BTDC to 24^0 BTDC but it starts deteriorating on further advance to 26^0 BTDC. This trend is shown by both the engines i.e. existing engine as well as modified engine. The above results revealed that the optimum advance angle of fuel injection is 24^0 BTDC to provide optimum engine performance and emission characteristics.

(vii) The existing engine at rated engine load with diesel fuel has maximum efficiency 32.96% at 175 kgf/cm^2 which deteriorates as the fuel injection pressure is increased. The efficiency gets reduced to 30.76 % at 205 kgf/cm^2 . This shows that the optimum fuel injection pressure of existing engine with diesel fuel is 175 kgf/cm^2.

(viii) It was found that the engine efficiency with JSVO fuel at rated fuel injection pressure (175kgf/cm^2) and advance angle of fuel injection 20^0 BTDC is 30.62% which gets increased to 31.45% at 195kgf/cm^2 and at the same fuel injection point. The efficiency gets deteriorated to 31.25% by further increase in fuel injection pressure to 205 kgf/cm^2. The decrease in efficiency by increasing the pressure to 205 kgf/cm^2 may be due to the suffocation in combustion chamber. The suffocation in combustion chamber is due to low momentum of fuel particles leading to short distance traversed by the fuel spray.

(ix) The analysis of obtained emission constituent (CO, CO2, NO_x, HC and smoke) on varying fuel injection pressures shows that the emission characteristics improves as the fuel injection pressure is increased from 175kgf/cm^2 to 195kgf/cm^2 but it starts deteriorating on further increase to 205 kgf/cm^2. This trend is shown by both the engines i.e. existing engine as well as modified engine at all the operating points. The above results revealed that the optimum fuel injection pressure for optimum engine performance and emission characteristics is 195kgf/cm^2.

(x) It is concluded that the efficiency of existing engine with diesel fuel at 20^0 BTDC fuel injection point and 175 kgf/cm^2 fuel injection pressure is obtained as 32.96%. The experiment with JSVO as test fuel in same engine at same operating point, the efficiency was obtained as 30.62%. After the modification i.e. installing the heat exchanger, keeping 24 ^{0} BTDC advance angle of fuel injection and 175kgf/cm^2 fuel injection pressure the efficiency was obtained as 34.76%. This shows that by implementing the modifications the engine efficiency with JSVO fuel not only becomes equal to the efficiency of engine with diesel fuel but also the efficiency is enhanced by 1.8%.

(xi) Therefore the existing IDI engine can be run successfully with JSVO and by incorporating the suggested modification in fuel injection systems. The engines may perform equivalent to diesel engine.

(xii) The experiments were conducted at ambient temperatures of 25^{0}C-30^{0}C and no engine starting trouble was experienced during the experiments, even without any modification of fuel injection system and without preheating the fuel, therefore it can be expected that staring trouble will not arise even during winter seasons.

(xiii) The researchers have reported the injector choking, gum formation, oil dilution and piston sticking under endurance test. As the recommended modifications improve not only efficiency and BSFC but also the emission characteristics, therefore it can be expected that the above problems if not eliminated then also will get minimized. However the engine performance on these issues are to be investigated under endurance test in future.

(xiv) It can be recommended that the engine cylinder head should be cleaned at shorter interval till the results are not obtained under endurance test and new maintenance protocol is chalked out.

Scope for future work:

The scope of present investigation is to develop a dedicated JSVO engine. This will improve the prosperity of the remotely situated farmers and small scale industrialist by making them self dependent. The engine users must be benefited as the JSVO can be produced by them locally. This will enhance the utilization of unproductive land, as the Jatropha plant is primarily cultivated in barren land.

The current investigation will also promote the researchers to utilize the finding of this work in their future work. Some of the prominent areas which can be dealt for future investigations are:

(i) Feasibility investigation of using other types of non edible oils in IDI engine and also in DI engine.

(ii) The suitability of use of vegetable oil in tractors which is versatile machine being used for agricultural purposes.

(iii) Utilization of Jatropha de oiled cake for energy generation.

(iv) The search for techniques to improve the Jatropha plant yield.

(v) The search for bio techniques so that the oil content of Jatropha seed can be improved. (Project is going on at FRI Dehradun, where the researchers are developing such species which can give more oil yield)

(vi) The study of other techniques to improve the efficiency further more.

(vii) Endurance test of engine under suggested engine modifications.

Experimental Data Sheet Annexure V

Fuel Injection Angle: 20 degree BTDC Fuel Injection Pressure : 175 kgf/cm^2 Fuel: Jatropha Calorific Value: 39000 kJ/kg Density: 910 kg/m^3

Load (kW)	T_{FUEL} (Sec)(For 50 CC)	H(mm)	RPM	EGT (^0C)	WIT (^0C)	WOT (^0C)	LOT (^0C)	Air Intake Temp. (^0C)	CO (%Vol)	CO_2 (%Vol)	NO_X (ppm)	HC	SMOKE (%opacity)	Voltage (Volt)	Current (Amp.)	Power Output (kW)	BTE %	BSFC (kg/kWh)
0	232	150	1028	130	26	62	38.7	32.5	0.15	0.45	150	7.5	8.1	235	0	0	0.00	#DIV/0!
1.84	160	150	1017	169	26	51.2	48.4	30.7	0.14	0.54	260	13	9.3	235	6.25	1.84	16.55	0.558
3.68	120	150	1006	189	26	57.9	50.1	31.2	0.13	0.62	350	17.5	11.2	235	12.5	3.67	24.83	0.372
5.52	93	150	1003	232	26	68	52.5	32.5	0.12	0.71	370	18.5	14.4	235	18.75	5.51	28.87	0.320
7.35	74	150	1000	260	26	74.2	54.3	32.2	0.10	0.76	430	21.5	17.9	235	25	7.34	30.62	0.301

Fuel Injection Angle: 20 degree BTDC Fuel Injection Pressure : 185 kgf/cm^2 Fuel: Jatropha Calorific Value: 39000 kJ/kg Density: 910 kg/m^3

Load (kW)	T_{FUEL} (Sec)(For 50 CC)	H(mm)	RPM	EGT (^0C)	WIT (^0C)	WOT (^0C)	LOT (^0C)	Air Intake Temp. (^0C)	CO (%Vol)	CO_2 (%Vol)	NO_X (ppm)	HC	SMOKE (%opacity)	Voltage (Volt)	Current (Amp.)	Power Output (kW)	BTE %	BSFC (kg/kWh)
0	243	150	1038	152	26.9	43.8	56.2	30	0.13	0.50	140	7.00	7.80	235	0	0	0.00	#DIV/0!
1.84	167	150	1024	193	26.9	46.6	56.4	30.2	0.12	0.59	255	12.75	9.10	235	6.25	1.84	17.28	0.534
3.68	126	150	1008	232	26.9	50.6	56.4	30.3	0.10	0.67	348	17.40	10.80	235	12.5	3.67	26.07	0.354
5.52	95	150	1004	259	25.9	52.1	56.3	30.8	0.10	0.76	367	18.35	13.70	235	18.75	5.51	29.49	0.313
7.35	75	150	1000	276	26.8	51.6	54.8	30.2	0.09	0.80	425	21.25	17.10	235	25	7.34	31.04	0.297

Fuel Injection Angle: 20 degree BTDC Fuel Injection Pressure : 195 kgf/cm^2 Fuel: Jatropha Calorific Value: 39000 kJ/kg Density: 910 kg/m^3

Load (kW)	T_{FUEL} (Sec)(For 50 CC)	H(mm)	RPM	EGT (^0C)	WIT (^0C)	WOT (^0C)	LOT (^0C)	Air Intake Temp. (^0C)	CO (%Vol)	CO_2 (%Vol)	NO_X (ppm)	HC	SMOKE (%opacity)	Voltage (Volt)	Current (Amp.)	Power Output (kW)	BTE %	BSFC (kg/kWh)
0	236	140	1038	146	26.5	52.6	57.8	33.4	0.11	0.61	137	6.85	7.30	235	0	0	0.00	#DIV/0!
1.84	168	140	1030	181	26.5	56.2	57.7	33.9	0.10	0.66	247	12.35	8.50	235	6.25	1.84	17.38	0.531
3.68	130	140	1020	210	26.4	59.3	57.1	33.8	0.10	0.74	341	17.05	10.10	235	12.5	3.67	26.90	0.343
5.52	97	140	1010	261	26.2	65.3	55.3	34.3	0.09	0.83	355	17.75	13.00	235	18.75	5.51	30.11	0.307
7.35	76	140	1008	303	25.8	68.5	52	33.5	0.08	0.89	410	20.50	15.90	235	25	7.34	31.45	0.293

Fuel Injection Angle: 20 degree BTDC Fuel Injection Pressure : 205 kgf/cm^2 Fuel: Jatropha Calorific Value: 39000 kJ/kg Density: 910 kg/m^3

Load (kW)	T_{FUEL} (Sec)(For 50 CC)	H(mm)	RPM	EGT (^0C)	WIT (^0C)	WOT (^0C)	LOT (^0C)	Air Intake Temp. (^0C)	CO (%Vol)	CO_2 (%Vol)	NO_X (ppm)	HC	SMOKE (%opacity)	Voltage (Volt)	Current (Amp.)	Power Output (kW)	BTE %	BSFC (kg/kWh)
0	240	145	1038	149	26.7	48.2	57	31.7	0.12	0.56	139	6.93	7.55	235	0	0	0	#DIV/0!
1.84	168	145	1027	187	26.7	51.4	57.05	32.05	0.11	0.63	251	12.55	8.80	235	6.25	1.84	17.33	0.533
3.68	128	145	1014	221	26.65	54.95	56.75	32.05	0.10	0.71	345	17.23	10.45	235	12.5	3.67	26.49	0.349
5.52	96	145	1007	260	26.05	58.7	55.8	32.55	0.09	0.80	361	18.05	13.35	235	18.75	5.51	29.80	0.310
7.35	76	145	1004	289.5	26.3	60.05	53.4	31.85	0.09	0.85	418	20.88	16.50	235	25	7.34	31.25	0.295

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