ANALYSIS OF EMULSION FUEL SPRAY CHARACTERISTICS USING SAC AND VCO NOZZLE

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I certify that the thesis entitled " Analysis of Emulsion Fuel Spray Characteristics Using SAC and VCO Nozzle" is written by Che Hanisah Binti Yaacob. I have examined the final copy of this thesis and in my opinion, it is fully adequate in terms of scope and qualify for the award of the Degree of Bachelor Engineering. I hereby recommend that it be accepted in fulfillment of the requirements for the Degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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ABSTRACT

The presence of water within diesel fuel in the form of water-in-diesel (W/D) emulsion lowers the pollution level of nitrogen oxides and particulate matter. The spray penetration and spray angle is the basic phenomenon that can show the combustion inside the chamber where it can determine the time taken for complete combustion. The influences of injector nozzle geometry, injection pressure and ambient air conditions on emulsion fuel spray were examined using simulation ANSYS CFD Fluent 12.1. The emulsion fuel is carried on of 5%, 10% and 15% of water being analyzed at single hole nozzle, 0.2mm on different injection pressure, 0.4MPa and 1.3MPa. This simulation also had been analyzed on different nozzle, SAC and VCO nozzle. The spray penetration showed the differences for both injection pressure, where the highest injection pressure produced the furthest spray penetration compared to the lowest of injection pressure. 5% of water gave the furthest spray penetration due to the emulsion properties of viscosity, where it has highest viscosity compared to the 10% and 15% of water. Comparison were made between different nozzle geometries while the SAC nozzle resulted in furthest spray penetration due to the design and geometry compared to the VCO nozzle under the same conditions

ABSTRAK

Kehadiran air di dalam bahan api diesel dalam bentuk air dalam diesel (A/D) merendahkan tahap pencemaran oksida nitrogen dan habuk terhampai. Panjang penembusan semburan dan sudut semburan memainkan peranan penting dalam konsep pembakaran lengkap di dalam ruang kebuk dimana ia menentukan kepantasan masa yang diambil untuk sesuatu bahan api bertindakbalas dengan udara di dalam ruang kebuk tersebut. Semakin pendek masa yang diambil untuk menghasilkan sebuah pembakaran lengkap akan memberikan daya tujahan yang terbaik dan mengurangkan pengeluaran gas- gas yang tidak diperlukan seperti oksida nitrogen dan habuk terhampai. Panjang semburan dan sudut semburan amat dipengaruhi oleh sifat geometri nozel, tekana suntikan, keadaan udara di dalam ruang kebuk dianalisa menggunakan simulasi ANSYS CFD 12.1. Simulasi ini dijalankan pada kuantiti air yang berbeza iaitu sebanyak 5%, 10% dan 15% yang diuji pada satu lubang yang memilik panjang sebanyak 0.2mm, dan pada tekanan suntikan yang berbeza iaitu 0.4MPa dan 1.3MPa. Selain itu, projek ini turut dijalankan terhadap nozel yang berbeza, SAC dan VCO nozel. Tekanan suntikan yang tinggi menghasilkan panjang semburan yang jauh berbanding tekanan suntikan yang rendah. Kuantiti air sebanyak 5% menunjukkan semburan yang paling panjang dibandingkan dengan 10% dan 15% air kerana ia mempunyai tahap kelikatan yang paling tinggi. Perbandingan juga turut melibatkan jenis nozel yang digunakan dimana SAC nozel menghasilkan panjang semburan yang lebih tinggi berbanding VCO nozel kerana reka bentuk dan sifatnya dalam keadaan yang sama.

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S	Spray penetration length
P	Injection pressure
Δ	Differential

LIST OF ABBREVIATIONS

CFD	Computational dynamic fluid
NOx	Nitrogen oxides
CO	Carbon oxides
VCO	Valve covered orifice
C	Carbon
H	Hydrogen
O	Oxygen
N	Nitrogen

CHAPTER 1

INTRODUCTION

In this chapter, the discussion involved the general information and knowledge about the spray characteristics, emulsion fuel, SAC nozzle and VCO nozzle. There are also problem statements that bring out why this study is carrying and what the benefit the whole world. Here, are also stated the objectives and scopes during the process of this project.

1.1 BACKGROUND OF STUDY

This study contribute to the development of emulsion spray formation and combustion models. The motivation is based on the public need for maintaining or even improving, current prosperity, while preserving the environment and health mankind. In daily practice this means, amongst others, that one has to comply with stringent regulations concerning internal combustion engine emissions. These emissions include pollutants like nitrogen oxides (NOx) and soot. More and more also emission of carbon oxide (CO) is restricted due to its involvement with the reinforcement greenhouse effect. Another implication of this public need together with an increase of the global energy demand is the approaching depletion of fossils fuels, which makes the efficient use of organic fuel necessary.

The presence of the dispersed water droplet phase within a continuous diesel fuel phase leads to the formation of water-in-diesel emulsion (W/D emulsion) or in general, water –in-oil diesel emulsion (W/O emulsion). Numerous industrial and environmental applications involve W/O emulsion. Some examples of these applications are crude oil spillage (Mingyuan, Christy, and Sjoblom, 1992), pipeline transportation of water in heavy crude oil (Pilehvari, Saadevandi, Halvaci, and Clark, 1988), and crude oil-polymer emulsion production during the enhanced oil recovery stage (Ghannam, 2003). There are several other important industries that involve the production of stable emulsion such as the food industries (e. g, mayonnaise), detergency (e. g, removal of oil deposits), pharmacy (e. g, drug emulsion), and cosmetics (e. g, skin lotion). Other potential benefits of emulsified fuel are:

i. Elimination of high cost fireside additives
ii. Reduction in nitrogen oxide due to reduced excess air and lower peak flame temperature
iii. Increase in thermal efficiency and heat rate due to reduced fireside deposits and excess air.
iv. Improved opacity
v. An increase in the range of fuel options

Measured spray characteristics are classified into two basic categories; the first one is macroscopic characteristics, which involve spray tip penetration, spray cone angle, and the derivates of them. The second one is microscopic characteristics which involve droplet velocity, droplet distribution, droplet diameter distribution, air-fuel ratio distribution , and so forth. Macroscopic properties of diesel spray can be recorded and analyzed with lesser and cheaper laboratory equipments than the ones that microscopic properties require. In addition to this, macroscopic characterization more reliable, since they are in bigger dimension and easily detectable. Spray tip penetration is most fundamental characterization among the others. Figure1.1 illustrates the spray images. Below the definitions of the terms of spray images:

i. Tip penetration – the maximum distance between the tip and the root of spray
ii. Spray angle – the angle between the tangents to the spray envelope

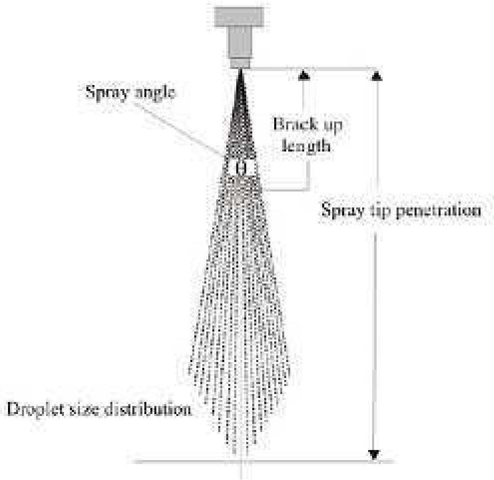


Figure 1.1: Physical parameters on diesel spray

Source: Hiroyasu and Arai (1990)

This study investigates the spray characteristics use emulsion fuel on different nozzle, SAC and VCO nozzle at same diameter of hole.

1.2 PROBLEM STATEMENT

The presence of the emission from diesel fuel within the atmosphere will cause serious damages to the environment such as the green house effect, acid rain, and destruction of the ozone layer. Due to enforced environmental regulations, reducing exhaust gas emissions from diesel engine is necessary. NOx and particulate matter emissions, for example, cause serious problems in urban environment where traffic congestion is very heavy. Also, due to the limited stock of fossil fuel, the search for alternative fuel must be accelerated. Among the potential alternative fuel is emulsion fuel.

1.3 OBJECTIVES

i. To investigate the spray characteristics use the emulsion fuel on different nozzle
ii. To investigate the relationship between the spray characteristics and the corresponding injector parameters.

1.4 SCOPES OF EXPERIMENT

For this project, more focus on the spray tip penetration and spray cone angles and also the spray images. The emulsion fuels consists three different percentage of water at 5%, 10%, and 15%. This project is carried on by using different nozzles, SAC and VCO nozzle in getting the comparison between these two nozzles. ANSYS CFD Fluent 12.1 is use in this project to visualize the spray images.

First of all, a literature study on spray fundamental is reviewed in Chapter 2, with special attention to fuel spray in engine conditions. Then, numerical methods are classified into Chapter 3. For the Chapter 4, the result that obtained from the simulation will be showed and discuss in chapter 5. Lastly, the conclusion will be more discussion and recommendations on chapter 6.

The Figure 1.2 show the flow chart of the project. The project started by finding the related journal and thesis that can be helped to understanding more about the general project. The next step is collecting the literature review by the previous research to identify the suitable method that can be done for this project. After that, study the ANSYS CFD Fluent flow by referring to the related tutorial. Then, create the suitable geometry for the chamber and nozzle. This thing cover the parameters like chamber, diameters of nozzle, types of emulsion and injection pressure. After that, key in the boundary condition and run the simulation. The visualize spray images will be analysis by looking to the images. If it is satisfied, then do the data analysis and discussion. Lastly, the conclusion will be made according to the flow of the project and data analysis.

CHAPTER 2

LITERATURE REVIEW

This chapter discuss about the reading that had been done before planning the methodology. Most of these reading help a lot to obtain more information, knowledge and be a guideline. As a result, chapter 2 will explain details about the literature review that had been done by previous study about the emulsion fuel spray characteristics. This chapter cover the historical of ANSYS CFD FLUENT, the others method that had been done by previous researchers for obtain the accurate result.

2.1 HISTORICAL BACKGROUND

The study of fluids fascinated mankind since the dawn of civilization. The initiation of hydrostatics, static mechanics and measure of objects density and volume is accountable to Archimedes. These ideas were abandoned until the spark of Renaissance in Southern Europe, where great artists with engineering attitudes started again to observe and study natural flows. Lewis Fry Richardson in England (1881-1953) is supposed to have developed the first numerical weather prediction system. This kind of idea could be considered as a very rudimentary CFD (Computational Fluid Dynamic) calculation. Thom in 1933 presented the earliest numerical solution for the flow past a cylinder, while Kawaguti in 1953 achieved similar results for flow around a cylinder by using mechanical desk calculator. A big contribution in the development of CFD numerical methods was done during 1960s by the theoretical division of NASA at Los Alamos and the ubiquitous k-turbulence model are still use nowadays. Hence, CFD is part of the computer-aided engineering (CAE) spectrum of tools, which are constantly used in industry and research institutions.

2.2 SPRAY BREAKUP MODEL

Fluent software provides the option to use two different models for the breakup of the fuel spray called TAB (Taylor Analogy Breakup) model and Wave model. TAB model is reported to be widely applicable model for many engineering situations. The analogy is created between an oscillating and distorting droplet and a spring mass system. The breakup model assumes the division of a large particle into small particles. Wave model is an alternative to the TAB model for high- Weber-number flows, based on the work Of Reitz (Reitz, 1987). Wave model considers the breakup of the droplets to be induced by the relative velocity between the gas and liquid phase. The models also predicts the parameters like particle diameter and particle velocity after breakup. The Wave model is used in the present work (fluent). Advanced version of TAB model, called E-TAB is not available in Fluent software.

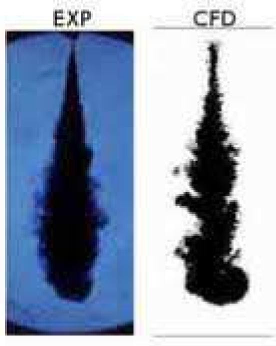


Figure 2.1: Spray images of experimental and simulation

Source: Reitz (1987)

Figure 2.1 show the spray images that obtained from the experimental and simulation. From the observation, spray images for experiment has a better penetration form compared to the simulation due to the atomization of the droplet molecules to the air inside the chamber.

2.3 DRAG MODEL

The Dynamic drag model calculates and updates the droplet drag coefficient, accounting for variations in the droplet shape in high Weber number sprays. The drag acting on a particle depends on the shape of particle. The model interpolates the drag of a particle by calculating the distorted shape between the shapes of a sphere and that of a flat disk, by making use of the drag co-efficient distribution, which is assumed linear (Heywood, 1988).

2.4 INJECTION MODELLING

The simulation of spray requires specification of the injector model. The compression ignition engine use multi-hole injector. The group type single-hole injector is chosen for the simulation purposes, which use information including nozzle diameter, mass flow rate, initial droplet size, droplet size distribution, droplet velocity, and position of injector. All such information completely defines the fuel injection model.

A Fiat single-hole 0.25mm diameter orifice nozzle, used by Mirza (Mirza, 1991), is use made of his published experimental results on fuel injection characteristics of the pump-line-injector combination using distribution type commercial fuel pump. The simulation results on spray penetration rates are compared with the empirical correlations.

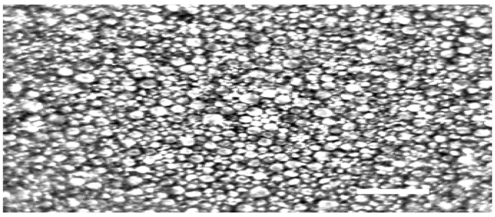
2.5 DROPLET COLLISION MODEL

Injection of fuel is assumed to consist of N number of particles. The droplet collision model handles the effective computation of the possibility that any two particles out of N particles will collide by introducing the concept of parcel. This reduces the computational cost several thousand times. Parcel is defined as a group of particles, which behave in a similar fashion collectively. Fluent uses O'Rourke's algorithm (O'Rourke, 1981) to estimate probability of collisions and its outcome in the form of coalescence or bouncing

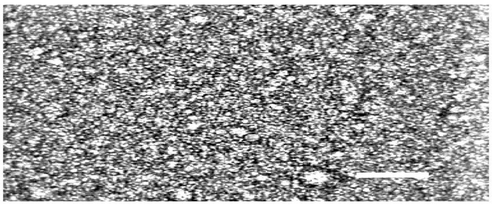
The model assumes that collision frequency is very small as compared to the time step. To adjust this, particle length scale is to be adjusted according to the distance traveled by the particle in the present time step. Because of the assumptions of collision in the same cell, grid dependant artifacts like stratification of particles can been seen. To avoid such situation a more refined grid should be adopted.

2.6 EMULSION STABILITY BEHAVIOUR

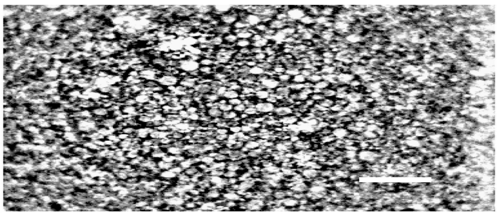
Muzio and Quartucy (1997), attributed the formation of NOx to the presence of nitrogen within the diesel fuel, excess oxygen and high gas combustion temperature. When water is added into diesel fuel in the form of W/D emulsion, however, the W/D emulsion fuel produces less NOx emission than the no emulsion diesel fuel. The emulsion with 20% water content reduced NOx emission by 56.8% (Lin and Wang, 2004). According to Lin and Wang, during the combustion process of W/D emulsion, it is atomized into numerous liquid droplets through a nozzle. Because the water's boiling point is less than that diesel fuel, the water enveloped layer explodes through the outer oil layer. As a result of micro explosion behavior, the atomized emulsion drops are further atomized into much finer droplets. This mechanism of micro explosion leads to a stronger mixing and a faster reaction rate between the atomized fuel droplets and the surrounding air. Therefore, a higher extent of combustion is completed. In addition, the emission of NOx, particulate matter and smoke is significantly reduced.



(a) 10/90 W/D emulsion (100x) Bar = 100 μm



(b) 30/70 W/D emulsion (100x) Bar = 100 μm



(c) 50/50 W/D emulsion (100x) Bar = 100 μm

Figure 2.2: Photomicrographs of 10/90, 30/70, and 50/50 emulsions

Source: Ghannam (2009)

Figure 2.2 show the photomicrographs of emulsion on different percentage of water. From the observation, 30/70 emulsion has the smallest micro molecules compared to the 10/90 emulsion and 50/50.

Based on experimental studies and dimensional analysis, (Pal, 1998) proposed an empirical viscosity model for mono-dispersed emulsion, with similar phase densities and with low interfacial tension. The flow is considered to be steady and the Brownian movement of droplets is neglected. (Paul, 2001) has recently reviewed and evaluated several theoretical viscosity models for several dilute and concentrated emulsions. All these models only describe the variation of viscosity as a function of dispersed phase volume fraction or the ratio between the viscosity of dispersed phase and that of the continuous phase. The significant effect of temperature on the viscosity is not considered in any of models.

Table 2.3: Chemical characteristics and properties of commercial diesel and emulsion fuel

Properties	Diesel	Emulsified fuel-1[a]	Emulsified fuel-2[b]
Density (kg m^{-3})	839.5	852.0	845.0
Cetane no.	52.6	45.5	45.9
Viscosity (cSt)	2.973	3.16	3.05
Lower heating value (MJ K g^{-1})	42.89	39.51	40.69
Sulfur content (ppm w)	410.0	345.0	321.0
C (% w/w)	85.24	72.67	73.56
H (% w/w)	13.652	11.86	11.97
N (% w/w)	0.063	0.050	0.054
O (% w/w)	–	12.689	13.782
Aromatic content (% w/w)	29.73	26.03	26.38
Poly aromatic content (% w/w)	0.51	0.45	0.48
Molecular weight (g/mol^{-1})	211.9	189.0	189.67
Adiabatic flame temperature (K)	2730.6	2704.6	2714.8

[a] Diesel containing 15% water contents stabilized by conventional surfactant.
[b] Diesel containing 15% water contents stabilized by Gemini surfactant.

Source: Nadeem et al (2006)

The Table 2.3 show the chemical characteristics and properties of commercial diesel and emulsion fuel. This property is resulted from the experiment by testing two different additives that mixed at the same percentage water, 15% of water. The additives that has been used was Gemini and conventional surfactant where the Gemini surfactant showed the greater properties compared to the conventional surfactant.

Mattiello et al (1992) provided evidence of the modification of a heavy oil spray due to the secondary atomization in emulsion spray flames. Their light scattering measurements in the near burner region indicated a significant increase in particle number density and a reduction of ~50% in the droplet mean diameter. An important decrease in soot formation was also reported in the emulsion flame compared with the neat oil flame. The combustion of fuel in oil in the form of oil-water emulsions is also found to affect the production. The work of (Kozinski, 1994), on heavy oil combustion demonstrated that the formation of polycyclic aromatics in the emulsion flame was significant reduced. Data on the emission of NOx are scarce. No differences in NOx emission between neat oil and oil-water emulsions flames have been encountered in some studies.

Reported by Ahmad and Gollahali (1984), some variations but they were unrelated to the presence of the water and were probably connected with the nature of the surfactant. Cunningham et al, 1983, measured a decrease of ~10% in the emission of NOx when burning heavy oil-water emulsions. This change was attributed to the lower flame temperature due to additional water. Some results suggest a possible influence of the added water on the NOx formation through the increase in the OH radical pool, leading to a reduction in oxygen atom concentration and hence in NO formation. Several basic studies on the evaporation and combustion of isolated drops and burning sprays of emulsions have appeared by Gerhard et al (2010). Also, many researches on the use of emulsion in conventional combustors have been reported. These studies have shown that the effect of using emulsified fuel with the heavier fuels such as residual oils is more obvious where particulate matter emission and flame radiation are generally reduced by emulsification (Allouis et al, 2005). However, the CO, NO and hydrocarbon emissions and the thermal efficiency of combustion devices don not always improve when fuel are emulsified with water.

The effect of single and multi-point water addition system on the NOx and ssot emissions of a vehicular heavy-duty diesel engine have been investigated by Samec et al, 2000 and coworkers Cernej, 1993). Their result confess that both system (single and multi-point) demonstrate practically the same propitious influence on NOx emission reduction, but rather a poor effect on soot emissions. However, the result of several other investigations performed recently using water in fuel emulsion (Radloffand Mello, 1999) have concluded that more promising results on NOx and soot reduction may be expected.

Several experimental investigation were carried out on industrial furnaces and external combustion system Gunnerman (1997), diesel engines Storment (1978), Vchniesky (1975), and gas turbines Arias (2003) and Nageli (1978) to discuss some of the benefits of using water in fuel as an approach to improve the engine emission criteria, reduce the specific fuel consumptions, control the engine thermal loading, and maximize the combustion pressure Yoshimoto (1996) and Tsukahara (1989), however the variations in results from one set of experiments to another encouraged researchers to model the combustion of emulsified fuels and compare the modeling result with the actual combustion system results to predict the possible improvement that can be suggested.

Schlitt and Exner (1991) have compared water-in-diesel emulsions with humidified intake air; i. e, water in the form of aerosol. It was found that both systems reduced the NOx level compared to traditional diesel fuel. The percentage of water in fuel in the studies of diesel emulsions varies. Most of the water investigators used water contents 5-10%, however the use of higher percentage needs to be thoroughly investigated. It has been claimed that the optimum water content for NOx reduction is between 10 to 20 % (Bartok, 1991).

Samec et al (2002) studied the effect of 10 and 20% water-in-diesel on emulsion level of NOx, hydrocarbons and soot, as well as on the specific fuel consumption. The values obtained, compared to those of the neat diesel show considerable reduction in both hydrocarbons and soot at 10% water, however in their work the NOx reduction seems to be more water sensitive than the hydrocarbon and soot, therefore the 20% water in fuel level is needed to be investigated and reviewed thoroughly to clarify the discrepancies in result between the earlier work of Lawson, for Heavy-Duty Diesel Emission Control in 1986 (Lawson, 1986).

Many researchers concentrated on the secondary atomization phenomena and emulsified fuel penetration concept. Zhou and Thorp (Yoshimoto, 1989 and Lasheras, 1980) have presented both theoretical and experimental studies on the differences between pure fuel and emulsified fuel atomization and discussed the effects of emulsified fuel atomization on fuel combustion. They measured the spray tip penetration and spray angle in the combustion chamber of a marine diesel engine (Ruston 6APC) by using a high speed camera with a micro-lens. In their study they found that the pure fuel spray compared to that of the emulsified fuel has longer spray tip penetration and wider spray angle. Also they proved that the tip penetration increase as the water percentage increase within the range 5-20% water in fuel. The number of countable droplets of emulsion fuel was much greater than that for pure fuel, indicating that the emulsion fuel spray processes a larger total surface area. In addition, mathematical models for the prediction of spray tip penetration and spray angle were proposed.

2.7 MECHANISMS OF ATOMIZATION

Soteriou et al (1995) reported that cavitation in the nozzle hole is the predominant mechanism causing atomization in the spray. The cavitation in beneficial to spray atomization and causes atomization of the jet immediately on nozzle hole exit. There are two different mechanisms that cause cavitation in diesel fuel injection equipment. The cavitation that result from these mechanisms could be referred to as dynamically induced and geometry induced cavitation. The cavitation occurred in the holes of standard direct injection nozzles is categorized as geometry induced cavitation which could occur in steady state as well as in transient flow. It is initiated by local high velocities within separated boundary. The high velocities could result in sufficiently large reduction in local pressure to cause the formation of vapor bubbles. This cavitation process produces a homogenous opaque foam, rather than large voids. The intensity of geometry induced cavitation of an orifice could be indicated by a cavitation number which is defines as the ratio of a factor tending to create cavitation, such as average flow velocity or pressure drop across the orifice, to a factor tending to suppress it, such as downstream pressure. The intensity of cavitation increase with cavitation number. The jet from each nozzle hole diverges and atomizes when cavitation first occurs within the hole. The spray angle increase significantly once the cavitation extends across and down to the bottom of the hole, and the flow consists of an opaque white form.

As Arcoumanis et al (1997) reported, three different atomization models had been used in their study; aerodynamic-induced atomization, jet turbulence-induced atomization, and cavitation-induces atomization. The models are briefed as follows. In the aerodynamic-induced atomization model, it was proposed that wave are developing on the surface of the liquid jet, caused by relative motion between the injected fuel and the gas. In terms of dimensionless parameters, the Weber number determines the grow rate of these waves and the disintegration of the jet into smaller droplet. In the jet turbulence-induced atomization model, it was proposed that for fully turbulence flow condition in the injection nozzle holes, the radial velocity component in the jet soon leads to disruption of the surface film, followed by general disintegration of the jet. Even when injected the vacuum, the jet will disintegrate solely under the influence of its own turbulence. In the cavitation-induced atomization model, it was proposed that the liquid jet emerging from the injection hole disintegrates due to the collapsing of the cavitation bubbles present at the exit of the holes. Since the pressure around the emerging jet is much higher than the pressure inside the cavitating bubbles, these bubbles gradually collapse while they are convected by the internal jet turbulence. This process causes perturbation to be formed on the surface of the liquid jet. The perturbations lead to jet disintegration and formation of smaller droplets at the time of total bubble collapse or at the time the bubble reach the jet surface. Based on the results of this study, it showed that the hole cavitation strongly affects the injection velocity and droplets sizes and the cavitation-induced atomization model predicted the droplet sizes more accurately as compared to the other two models.

By understanding the literature review of the past researcher, a lot of information can be used to continue this project and get the result.

CHAPTER 3

METHODOLOGY

This chapter will closely discuss details more about the procedures for the simulation analysis. Before that, some properties and model of drawing is needed to proceed the procedures. The properties that concern is the emulsion fuel that contain different presence of water, the various injection pressure and different nozzle, SAC and VCO.

3.1 EMULSION FUEL PROPERTIES

Table 3.1: The emulsion fuel properties table

% of water / Properties	5%	10%	15%
Density (kg/ m^3)	850.0	861.0	890.2
Dynamic viscosity (kg/ms)	0.006	0.00532	0.00472
Boiling point (K)	443	508	573

Source: Roberto et al (2005)

Figure 3.1 show the emulsion fuel properties for different percentage water added into the diesel. 5% of water has the lowest density which is only 850 kg/ m^3 compared to the 10% of water which is 861 kg/ m^3 and 15% of water has 890.2 kg /m^3. Whereas the 5% of water has the highest dynamic viscosity, 0.006 kg/ms, while 10% of water has 0.00532 kg/ms and 15% of water has 0.00472 kg/ms. 15% of water has the highest boiling point , 573 K while 10% of water has 508 K and 5% of water has 443 K.

3.2 PREPARING ANSYS SIMULATION

3.2.1 Creating Geometry

i. Construct the 3D model in Solid Work for the SAC and VCO nozzle. The dimension is 60 mm x 20 mm. Save the file in specialize folder and save as again with IGES format to make easier to import in ANSYS.

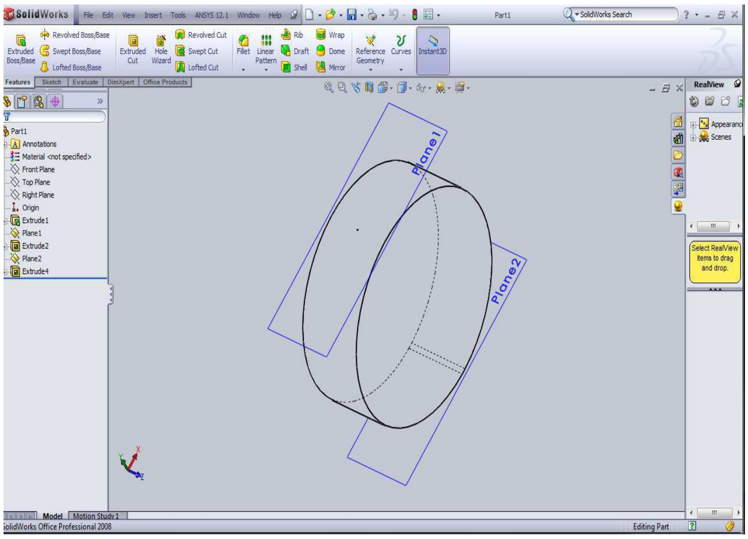


Figure 3.2: 3D model construction of VCO nozzle

ii. Open the ANSYS 12.1/Workbench. Import the 3D model of geometry with IGES file. Edit the geometry to the FLUID and generate. Closed the Design Modeler box and save the progress to Workbench.

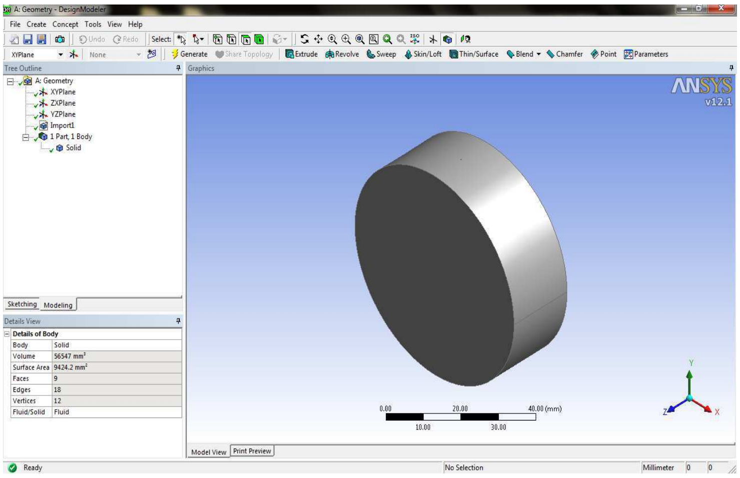


Figure 3.3: Design modeler box

iii. Right click Fluid Flow (Fluent) under the Project Schematic and drag the geometry. Right click Geometry and select Properties. Under Advanced Geometry Options, change the Analysis Type from 2-D to the 3-D and close the properties box.

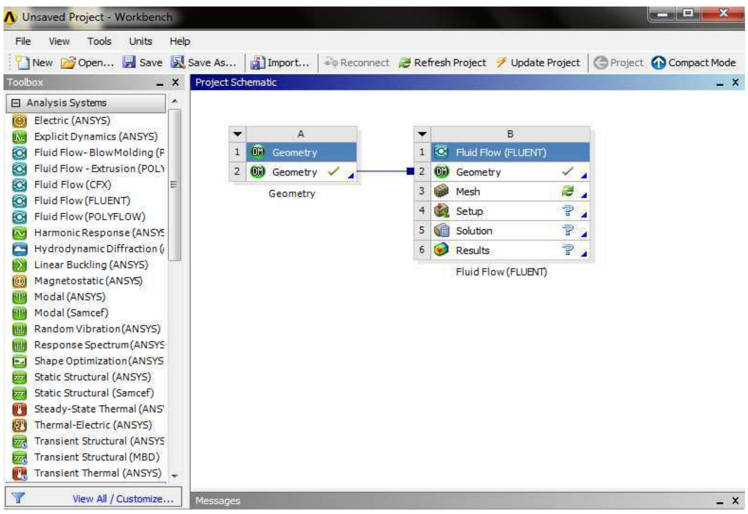


Figure 3.4: Workbench box

3.2.2 Mesh Generation

In the Project Schematic, right click the Mesh. It may take a minute or two to load. Select Mesh from the outline window. Select the face and named the selection to the pressure inlet and pressure outlet. In the Details View, click the plus sign to expand the sizing option. Change the Relevance Center to the Fine, Transition Fast, Span Angle Center Fine. Click the Inflation, change the Use Automatic Tet Inflation Program Controlled, Smooth Iteration 20. Right click Mesh and select Update. The mesh should now look similar to the following. Closed the Meshing window and save the project.

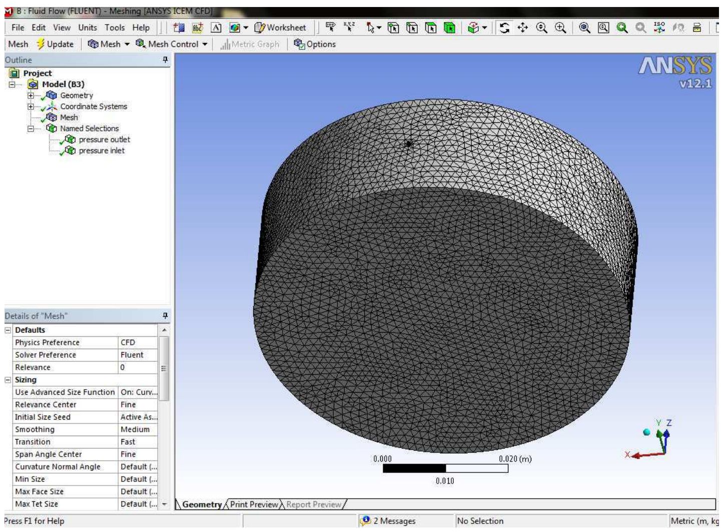


Figure 3.5: Meshing window box

3.2.3 Setup

Right click the Setup icon and when the FLUENT Launcher appears, click ok. Edit the unit to the length and mm. Reorder the Mesh twice until get 1.00 to speed up the solution procedure, which substantially reduce the bandwidth. ANSYS FLUENT was reported the progress in the console.

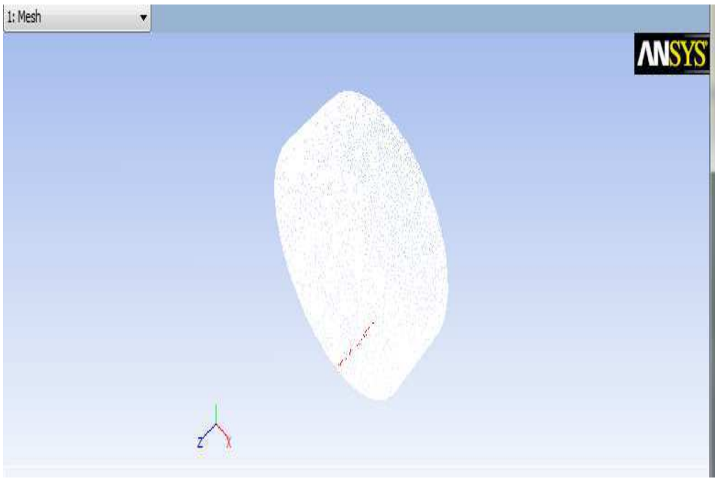


Figure 3.6: The meshing model

```
>> Reordering domain using Reverse Cuthill-McKee method:
     zones, cells, faces, done.
   Bandwidth reduction = 116039/2008 = 57.79
   Done.

>> Reordering domain using Reverse Cuthill-McKee method:
     zones, cells, faces, done.
   Bandwidth reduction = 2008/2001 = 1.00
   Done.
```

Figure 3.7: Reorder the domain

3.2.4 Models

i. Enable the heat transfer by enabling the energy equation.

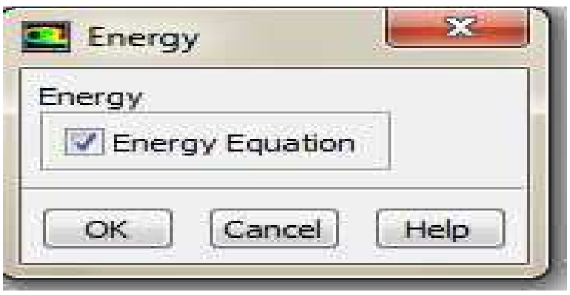


Figure 3.8: The energy box

ii. Enable the reliable k-e turbulence model. Select k-epsilon (2 equation) in the Model list. Select Reliable in the k-epsilon Model list. The reliable k-epsilon model gives a more accurate prediction of the spreading rate of both planar and round jets than the standard k-e model. Retain the default selection of Standard Wall Function in the Near- Wall Treatment list. Click ok to close the Viscous Model

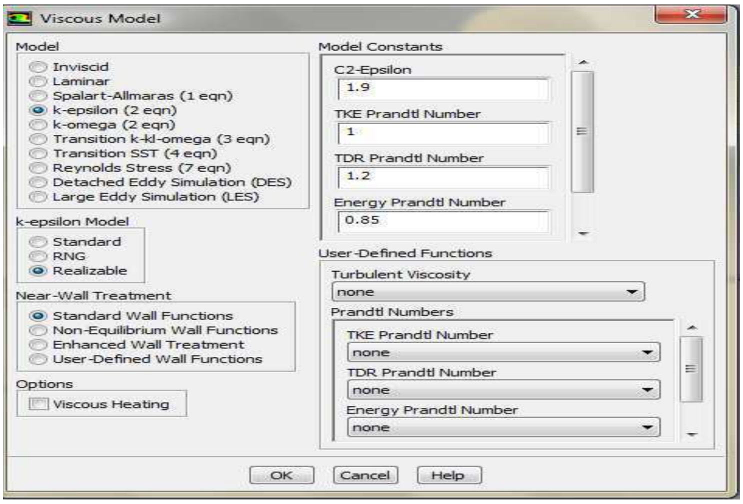


Figure 3.9: The viscous model box

iii. Enable chemical species transport and reaction. Select Species Transport in the Model list. Select diesel- air from the Mixture Material drop- down list. The Mixture Material list contains the set of chemical mixtures that exist in the ANSYS FLUENT database. The chemical species in the system and their physical and thermodynamic properties were defined by the selection of the mixture material. Click ok to close the Species Model dialog box.

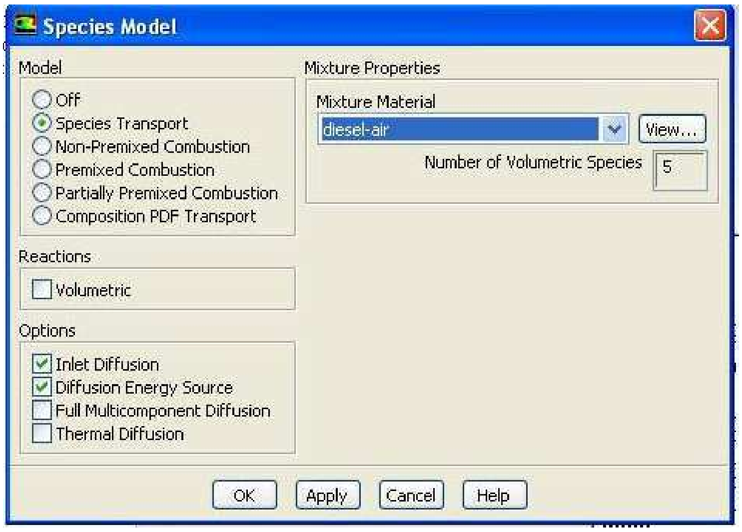


Figure 3.10: The species model box

iv. Define the discrete phase modeling parameters. Enable the Interaction with Continuous Phase in the Interaction group box. This was included the effects of the discrete phase trajectories on the continuous phase. Retain the value of 10 for Number of Continuous Phase Iterations per DPM iteration. Click the Physical Models tab to enable the physical models. Enable Droplet Breakup in the Spray Model group box. Ensure that TAB was enabled in the Breakup Model list. Retain the default value of 0 for y0 and 2 for Breakup Parcels in the Breakup Constants group box.

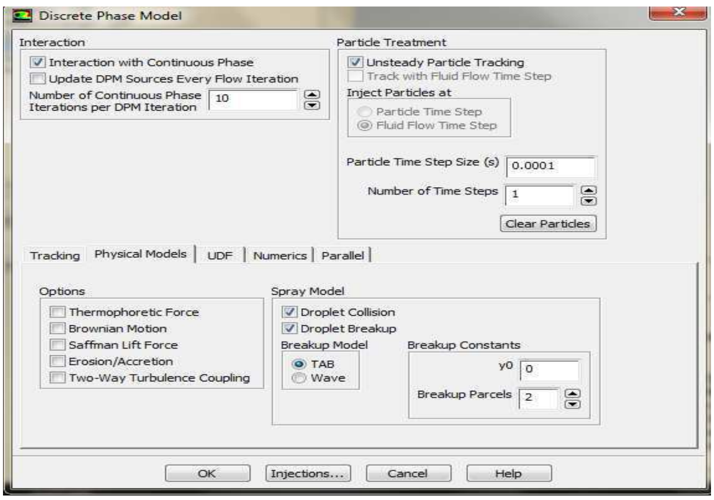


Figure 3.11: The discrete phase model

v. Create the spray injection. This step defined the characteristics of the atomizer. Click the Create button to open the Set Injection Properties dialog box. Select surface the from the Injection Type drop- down list. Select inlet from the release from the surfaces list. Select Droplet in the Particle Type group box. Select fuel-oil-liquid from the Material drop- down list. Enter 0, 0, and 0 for X-Velocity, Y-Velocity, and Z-Velocity, respectively, in the Point Properties tab. Enter 263 K for Temperature. Enter 1.785-3 kg/s for Flow rate. Retain the default Start Time of 0 s and enter 30 s for the Stop Time. For this problem, the injection should begin at t= 0 and not stop until long after the time period of interest. A large value for the stop time (e.g, 100 s) ensures that the injection essentially never stops.

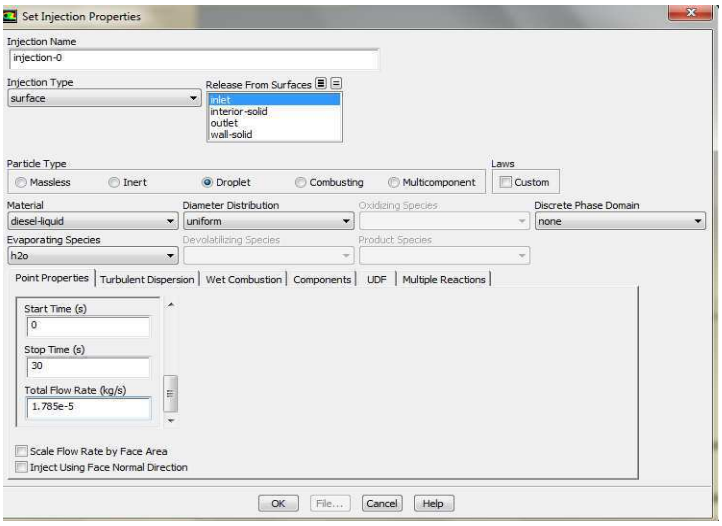


Figure 3.12: The set injection properties

vi. Define the turbulent dispersion. Click the Turbulent Dispersion tab. Enable Discrete Random Walk Model and Random Eddy Lifetime in the Stochastic Tracking group box. These models account for the turbulent dispersion of the droplets. Click OK to close the Set Injection Properties dialog box. Click OK in the Information dialog box to enable droplet coalescence. Close the Injection dialog box.

3.2.5 Materials

Set the droplet material properties. Set the droplet properties because secondary atomization models (breakup and coalescence) were used. Retain the default selection of droplet- particle from the Material Type drop- down list. Enter the 850 for the density and 0.006 for viscosity. Changed the precise linear in the saturation vapor. All these values is only for 5% emulsion fuel. Then, click change/ create and close the materials dialog box.

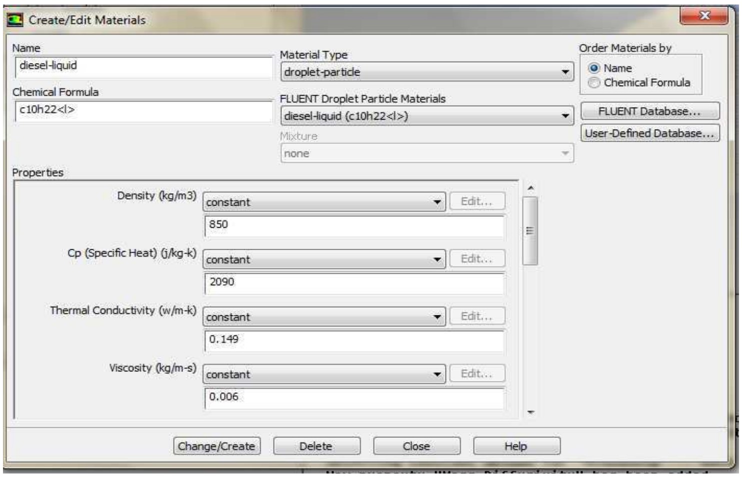


Figure 3.13: The create materials

3.2.6 Boundary Conditions

i. Set the boundary conditions for pressure inlet. Select Pressure Outlet for inlet boundary conditions from the Type drop- down list. Enter 400000 in Gauge Pressure. Select Normal to the Direction Vector. Select Intensity and Viscosity Ratio and change 10 for Backflow Turbulent Intensity and 10 for Backflow Hydraulic Diameter. Click the Species tab and click ok to close the Pressure Outlet dialog box.

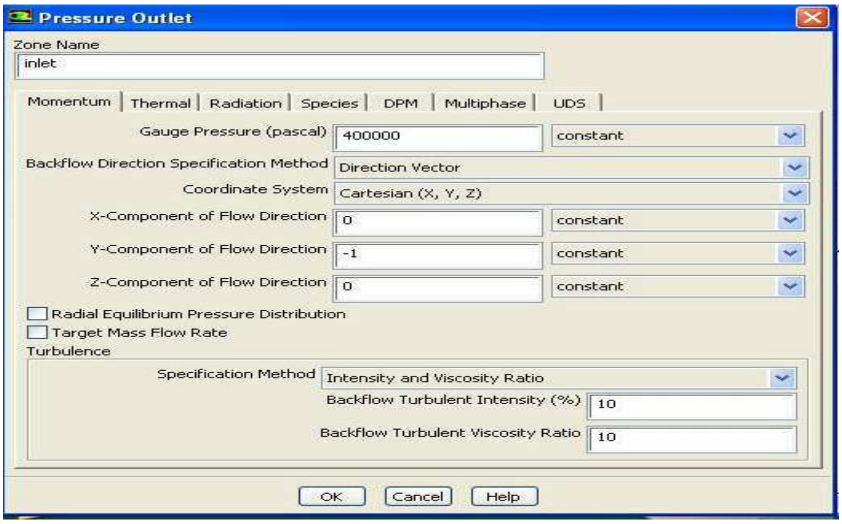


Figure 3.14: The inlet pressure box

ii. Set the Boundary Condition for pressure outlet. Select Pressure Outlet from the Type drop- down list. Enter 0 in Gauge Pressure. Select Normal to the Neighboring Wall Select Intensity and Viscosity Ration and enter 5 for Backflow Turbulent Intensity and 5 for Backflow Click the Species tab and click OK to close the Pressure Outlet dialog box.

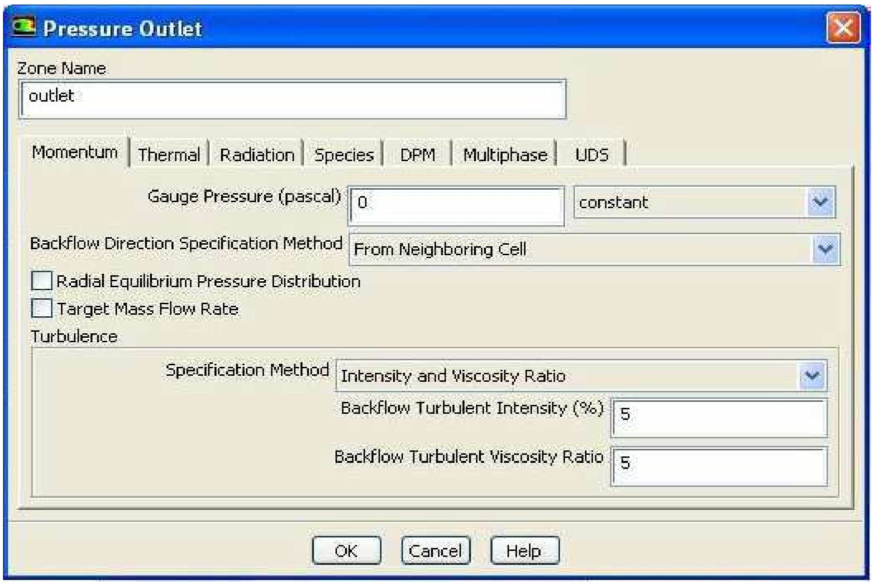


Figure 3.15: The pressure outlet box

iii. Set the boundary conditions for the wall. Select Specified Shear in the Shear Condition list. Retain the default values for the remaining parameters. Close the Wall dialog box.

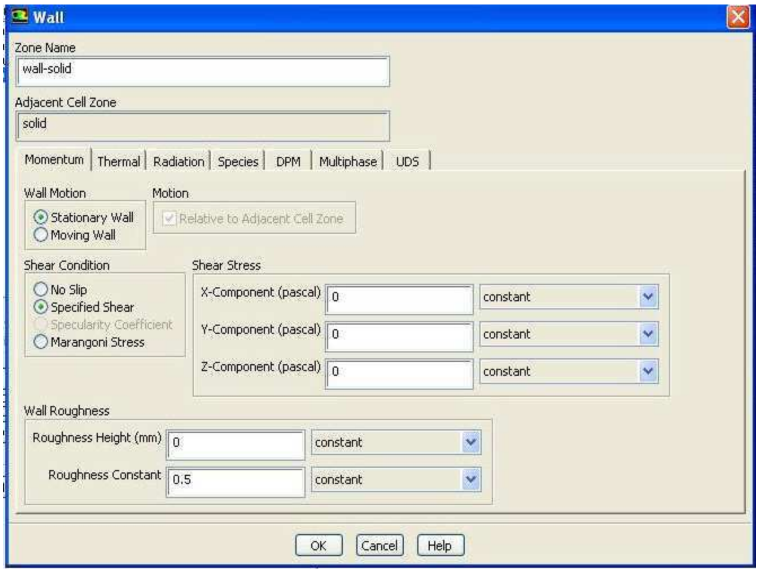


Figure 3.16: The wall dialog box

3.2.7 Solution Methods

Apply First Order Upwind at Solution Method. Select First Order Upwind from drop- down list for all parameter at Spatial Discretization box.

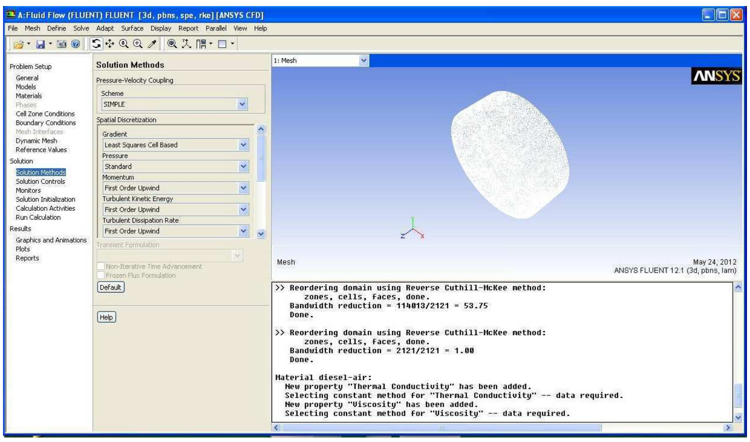


Figure 3.17: The solution method box

3.2.8 Monitors

Enable residual plotting during the calculation. Ensure the Plot is enabled in the Options group box. Disable all the Monitor Check Convergence Absolute Criteria. Click OK to close the Residual Monitors dialog box.

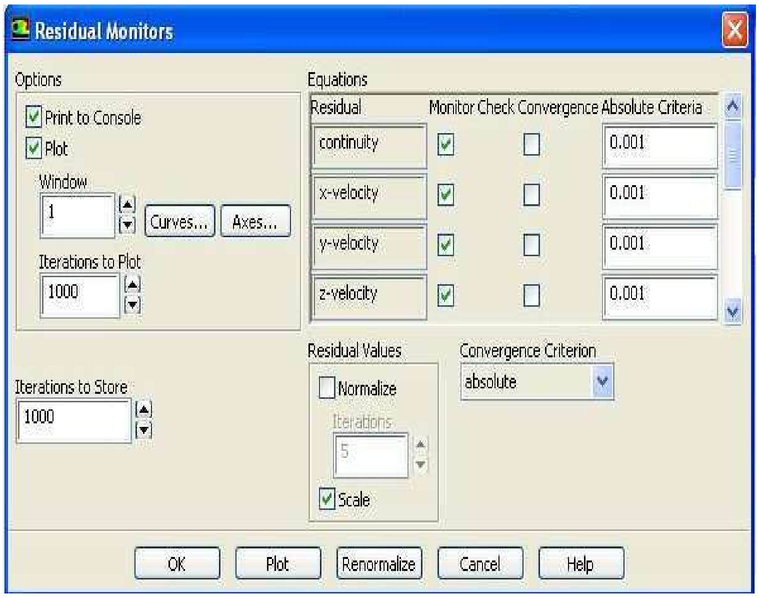


Figure 3.18: The residual monitors

3.2.9 Solution Initialization

Select the pressure inlet from the drop- down Compute and initialize the variables.

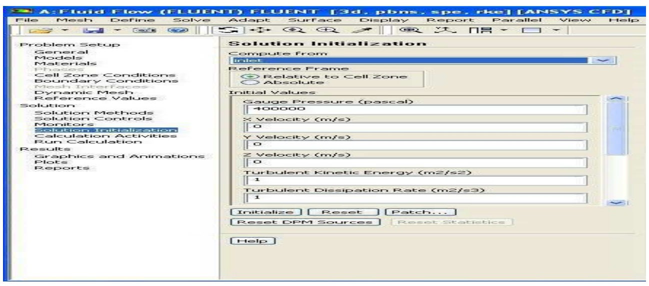


Figure 3.19: The solution initialization box

3.2.10 Run Calculation

i. Enter 43 for Numbers of Iteration and click Calculate. The solution will converge in approximately 43 iterations.

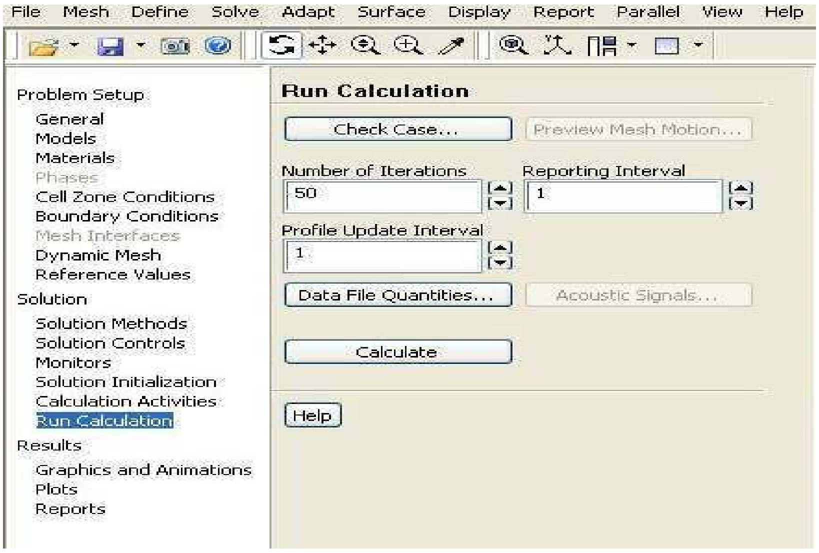


Figure 3.20: The run calculation box

ii. Create a plan to examine the flow field at the midpoint of the surface section. Select Mesh from the Surface of Constant drop- down list. Click Compute to update the minimum and maximum value. Choose Z coordinate and click Create to create the iso-surface and close the dialog box.

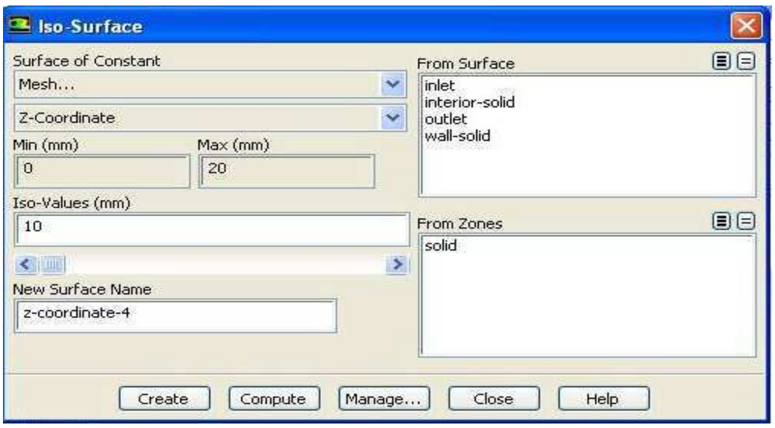


Figure 3.21: The iso-surface box

The steps is repeated for the 1.3MPa at different percentage of water by key in the emulsion properties on droplet properties. Next, the same procedures is done by using SAC nozzle. After that, the result and discussion will be followed up in next chapter.

CHAPTER 4

RESULTS AND DISCUSSIONS

This chapter will focus on the spray images that obtained from the simulation. The details discussion covered about the understanding and information include the spray penetration, spray angle at different properties of emulsion fuel, injected at different injection pressure and based on SAC and VCO nozzle. The factors that contribute the varies penetration length and angle on the spray.

4.1 SPRAY IMAGES

Figure 4.1 show the spray images that obtained from the simulation for 0.4MPa and 1.3MPa of injection pressure at 5%, 10%, and 15% of water used VCO nozzle. As observed from the images, at 5% of water for 0.4MPa is shorter that 1.3MPa. At 10% of water for 0.4MPa also showed the shorter spray penetration length compared to the 1.3MPa and the same goes at 15% of water which is the spray penetration for 0.4MPa is shorter compared to the 1.3MPa. These are due to the injection pressure that influenced the spray penetration length. Higher injection pressure produced the longest spray tip penetration length. The spray penetration length generally defined as the difference pressure on the nozzle to the chamber pressure. In addtion, it has been previously reported that higher injection pressures result in increase in the atomisation rate of the spray and result in decrease in the mean liquid droplet size. It would expected that the smaller droplet woult evaporate at higher rate, possibly resulting in a reduction in the apparent liquid length of the spray. The effect on penetration of a change in injection pressure is reduced at higher injection pressures. In addtion, the increased rate of evaporation at high injection pressures may also influence the penetration rate of the spray.

Discussion based on the spray angle for the spray images on Figure 4.1 also showed that the spray angle at 5% of water for 0.4MPa is wider compared to the 1.3MPa. The same situation goes for the 10% and 15% of water at 0.4MPa is much wider compared to the 1.3MPa. The factors were the injection pressure and viscosity of the emulsion fuel. The spray angle become wider with increasing the injection pressure but reduced by increasing the emulsion fuel viscosity. As shown in Figure 4.1, the spray angle for 5% of water is more narrow compared to the 15% of water at the same injection pressure. The viscosity influenced the spray angle by reducing or increasing the droplet size which is slightly influenced the spray angle. The spray angle generally related to the surface tension and droplet size.

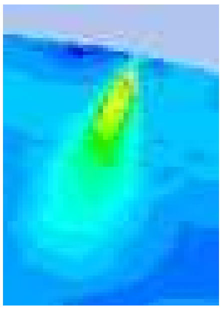
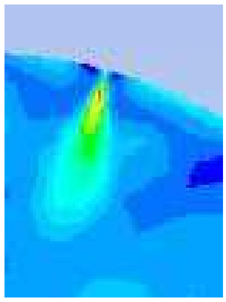
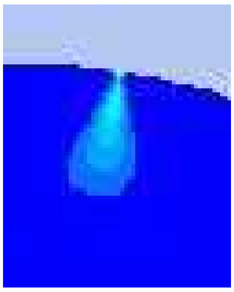
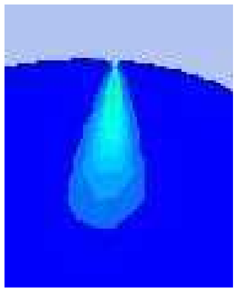
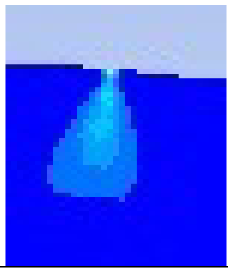
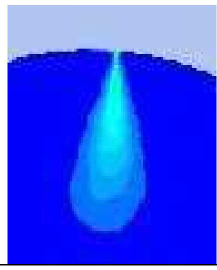
Percentage of water (%)	Injection Pressure (MPa)	
	0.4MPa	**1.3MPa**
5		
10		
15		

Figure 4.1: The spray images for 0.4MPa and 1.3MPa at different percentage of water (VCO nozzle)

Figure 4.2 show the spray images for 5%, 10%, and 15% of water at 0.4MPa and 1.3MPa used SAC nozzle. From the observation, the spray penetration length at 1.3MPa is longer compared to the 0.4MPa at the same percentage of water, 5%. The same phenomena occurred to the 10% and 15% of water, whereby at 1.3MPa resulted in longer spray penetration length compared to the 0.4MPa. The main factor that influenced the spray penetration length was the injection pressure. At higher injection pressure applied, the longer spray penetration length will be obtained. The injection pressure of 1.3MPa can penetrate longer because it has higher pressure compared to the pressure inside the chamber.

From the spray angle prospective, Figure 4.2 show that for 15% of water has wider spray angle than 10% and 5% of water at the same injection pressure, 0.4MPa. This is due to the viscosity of the emulsion fuel itself. 15% of water has the lowest value of viscosity (0.00472 kg/ms) compared to the 5% (0.006 kg/ms) and 10% (0.00532 kg/ms). The same discussion with the spray images on Figure 4.1 for the spray angle.

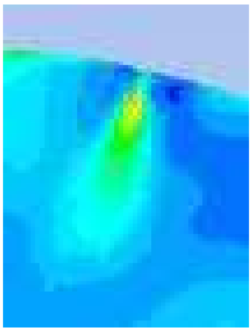
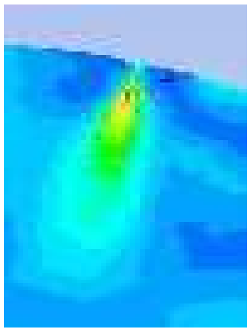
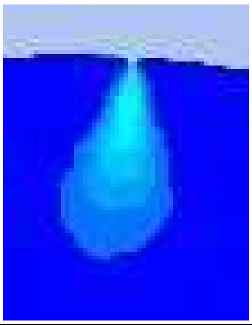
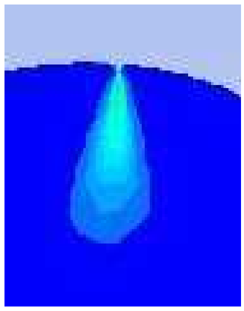
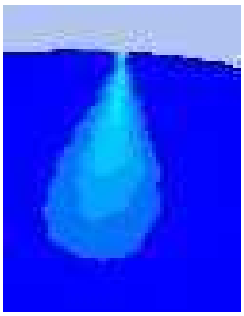
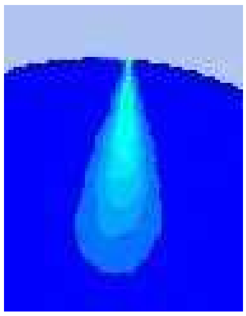
Percentage of water (%)	Injection Pressure (MPa)	
	0.4MPa	1.3MPa
5		
10		
15		

Figure 4.2: The spray images for 0.4MPa and 1.3MPa at different percentage of water (SAC nozzle)

4.2 SPRAY CHARACTERISTICS FACTORS

4.2.1 VISCOSITY AND TEMPERATURE

In Figures 4.3 and 4.4 , the spray penetration length for different presence water added into the diesel is shown grouping according to the VCO nozzle used. In combustion, the shorter time taken react with air inside the chamber is better because the complete combustion can achieved to reduce the emission. When the combustion is rapidly happened, the engine performance gives high efficiency and reduce the emission. The advantages for some fuel that has lowest viscosity, it tend to vaporise easier and react with air in short time. Compared to the high viscosity of fuel, it quiet hard to vaporise and some of the droplet cannot make a complete combuation. As a result, the emission produced in high quantities to the environment.

As can be observed in Figure 4.3 and 4.4, the longest spray penetration at 0.4 MPa on VCO nozzle is 30mm which is contributed by the 5% of water presence while for 1.3 MPa, the longest is 42mm also contributed by the 5% of water. The 10% of water for 0.4 MPa showed 26mm length and for 1.3 MPa is 39mm. The shortest spray penetration was recorded for 0.4 MPa was 23mm while at 1.3 MPa was 35mm.

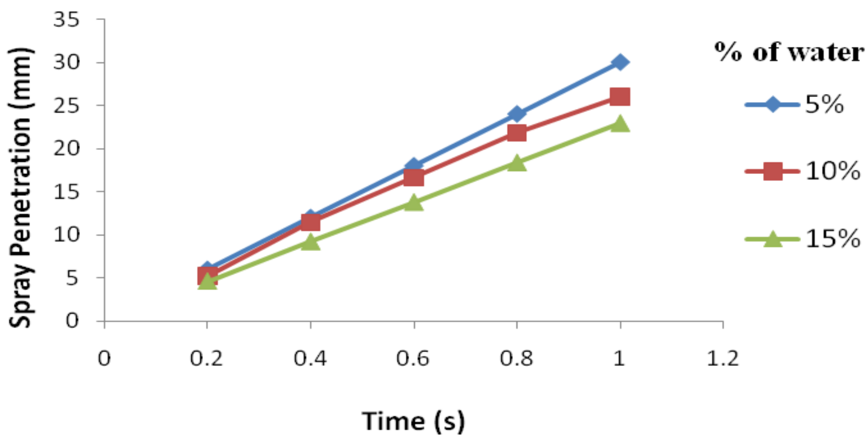


Figure 4.3: Time vs spray penetration for 5%, 10% and 15% of water at 0.4MPa (VCO nozzle)

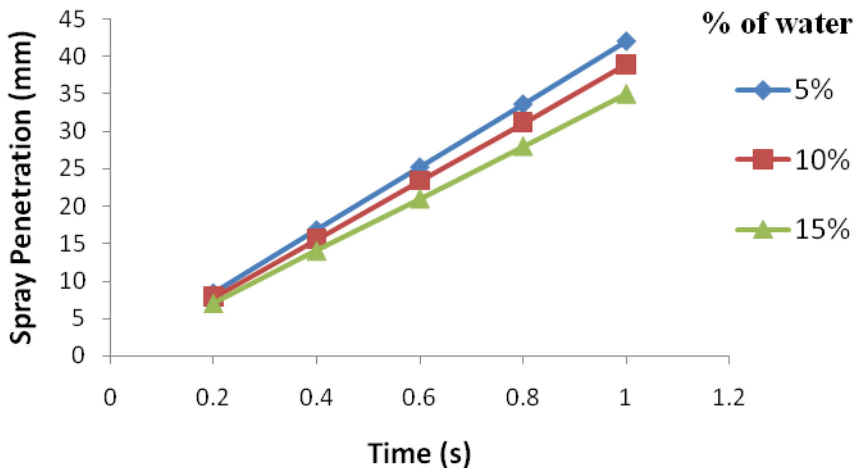


Figure 4.4: Time vs spray penetration for 5%, 10% and 15% of water at 1.3MPa (VCO nozzle)

Figure 4.5 and 4.6 showed the graph for the spray penetration used SAC nozzle at different injection pressure. From the graph, the pattern of the line is similar to the VCO nozzle which is in straight line. 5% of water for 0.4MPa showed 33mm and for 1.3MPa showed 45mm. 10% of water for 0.4MPa contributed 30mm while for 1.3MPa was 41mm while 15% of water, 27mm for 0.4MPa and 38mm for 1.3MPa. The shortest spray penetration was contributed by 15% of water for 0.4MPa which was 27mm while for the 1.3MPa was 38mm also by 15% of water. The longest spray penetration was goes to the 5% of water, 33mm for 0.4MPa and 45mm for 1.3MPa. The reduction of spray penetration was 18% for 0.4MPa while 16% for the 1.3MPa.

The spray penetration influenced by some factors that highly affects the length. These were the viscosity and temperature. Viscosity is defined as the resistance in the flow of a liquid or internal friction present between two layers of a liquid which resists the flow of liquid. Aliquid with high viscosity is thick and flows slowly and vice versa. Different liquid had different value of viscosity. 15% of water showed the shortest spray penetration length compared to the 5% of water due to the viscosity. 15% of water had lowest viscosity, 0.00472 kg/ms compared to the 5% of water, 0.006 kg/ms and 10% of water was 0.00532 kg/ms. The shortest spray penetration is easier to vaporise inside the chamber and react with air. The temperature also one of the factors that contributed to the spray penetration length. Each emulsion fuel that used had different boliling point. The lowest boling point was 443 K for 5% of water while the highest boiling point was 573 K for 15% of water. The viscosity of liquid decrease with increase in temperature. Because an increase in temperature, the forces of attraction between molecules also reduced. Hnece, the spray penetration for 15% of water became the shortest compared to the 5% and 10% of water. There were also others factors that influenced viscosity which were size of molecules, shapes of molecules and intermolecules forces.

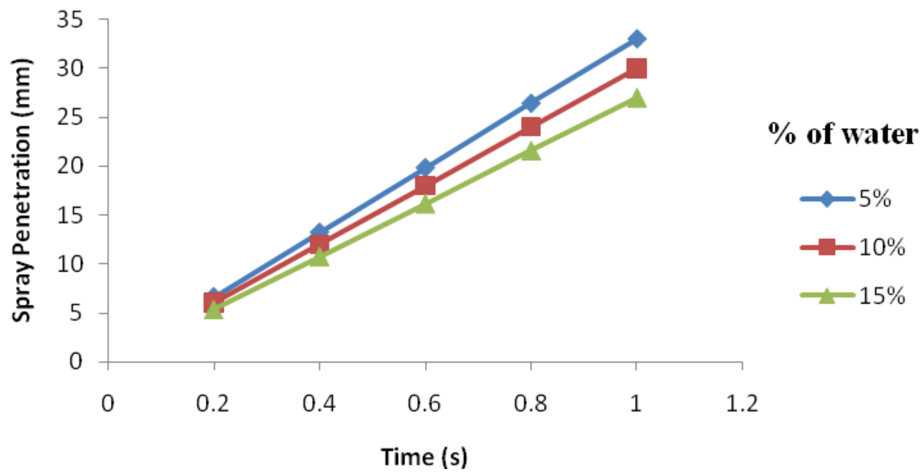


Figure 4.5: Time vs spray penetration on 5%, 10% and 15% of water at 0.4MPa (SAC nozzle)

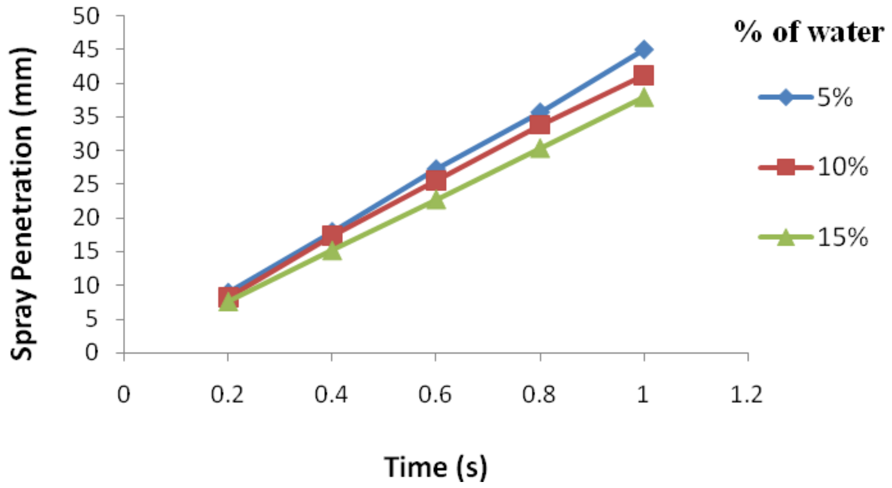


Figure 4.6: Time vs spray penetration at 5%, 10% and 15% of water on 1.3MPa (SAC nozzle)

4.2.2 GEOMETRY OF NOZZLE

Figure 4.7 showed the curves of spray penetration at 10% of water in different nozzle, SAC and VCO. From the graph, the O.4 MPa of SAC nozzle had longer spray penetration, 30mm compared to the 0.4 MPa of VCO nozzle. The same goes to the 1.3 MPa of SAC nozzle that recorded the longest spray penetration, 41mm while 1.3 MPa of VCO nozzle read only 41mm. The differences of the spray penetration at the same pressure is due to the geometry of the nozzle itself. The difference between the two nozzle designs is interesting. None of the empirical models considered account for the geometry upstream of the orifice which is the only difference between the two configurations. Previous work by Heimgartner and Leipertz has reported that the mean droplet size from VCO nozzle can be up to 50% smaller than equivalent mini-sac nozzle. Again, this would result in a higher rate of evaporation and could explain the observed result. It also been shown that VCO nozzle injectors cause a greater pressure loss within the nozzle geometry hence a lower Δp at the nozzle orifice and lower spray penetration. These result emphasises the importance of understanding the influence of nozzle and injector geometry on fuel spray. The SAC nozzle has a room that will maintain the volume and pressure before the injection. The pressure tend to pull the penetration rate volume of emulsion fuel to the furthest length. Generally, the SAC nozzle more power than VCO nozzle because the uniform pressure inside the room. Based on the spray penetration, SAC nozzle tend to produce a more uniform spray than VCO nozzle.

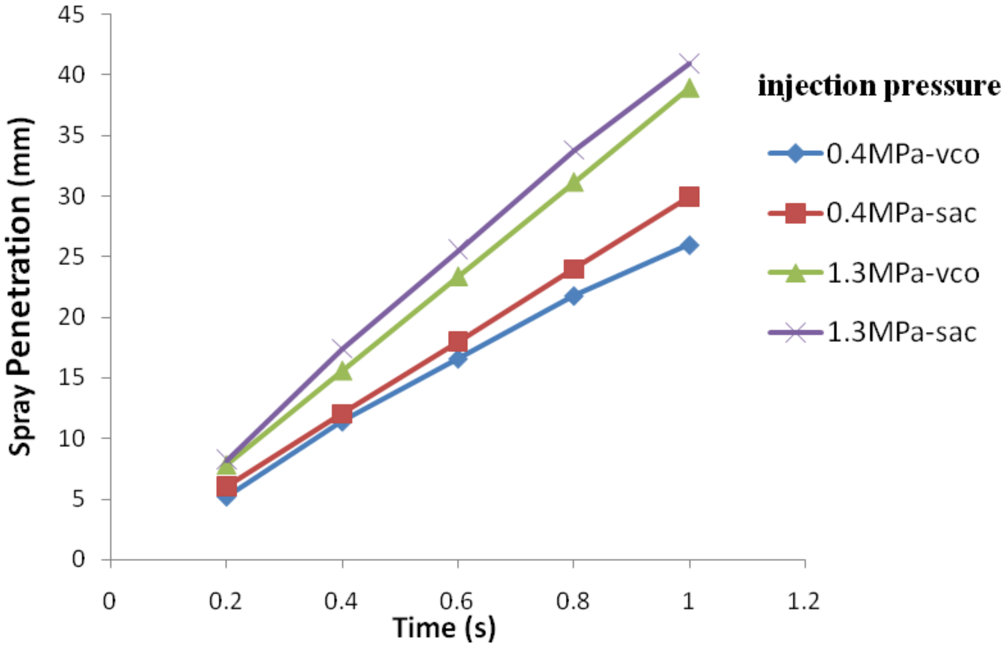


Figure 4.7:Time vs spray penetration at 0.4MPa and 1.3MPa (SAC and VCO nozzle)

4.2.3 INJECTION PRESSURE

Figure 4.8 showed the relationship between injection pressure with spray penetration length for 10% of water presence on VCO nozzle. The curves is slightly propertional of injection pressure to the spray penetration length. The graph approved that the spray penetration length for 1.3 MPa is more longest compared to the 0.4 MPa.The injection pressure is also a main factor that contribute the spray penetration length. As expected, higher injection pressure result in higher spray penetration. There was found that the differences between 0.4 MPa and 1.3MPa is slightly bigger. The spray penetration is generally reported to be a function of the pressure difference across the nozzle and air pressure. The theoritical dependance of the nozzle pressure difference was derived by Naber and Siebers (2000) and reported as below:

$$S \alpha (\Delta P)^{0.25} \quad 4.1$$

According to this relationship, the effect on penetration of a change in injection pressure would be reduced at higher injection pressures. Similarly injector body and nozzle throttling and hence the pressure loss becomes more significant for increasing the injection pressure.In addtion, it has been previously reported that higher injection pressures result in increase in the atomisation rate of the spray and result in decrease in the mean liquid droplet size. It would expected that the smaller droplet woult evaporate at higher rate, possibly resulting in a reduction in the apparent liquid length of the spray. The effect on penetration of a change in injection pressure is reduced at higher injection pressures. In addtion, the increased rate of evaporation at high injection pressures may also influence the penetration rate of the spray.

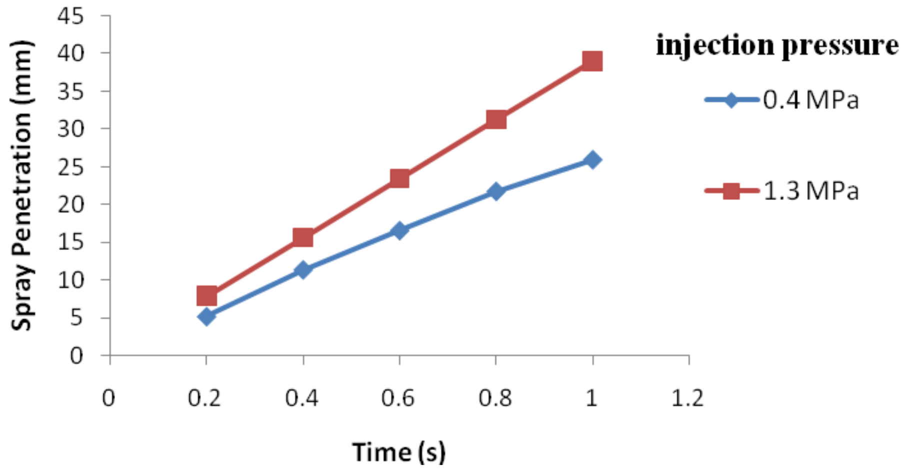


Figure 4.8: Time vs spray penetration at 0.4 MPa and 1.3MPa (VCO nozzle)

4.3 SPRAY ANGLE

As observed from the Figure 4.9, the curves shown based on the spray angle to the injection pressure and percentage of water added to the emulsion fuel for the VCO nozzle.. At 5% of water, at 0.4MPa, the angle that was recorded was 10 degree and at 1.3MPa the degree was 5 degree. For 10% of water, the spray angle at 0.4MPa was 20 degree while at 1.3MPa was 16%. Whereas for the 15% of water added, the spray angle at 0.4MPa was 46 degree and at 1.3MPa. was 20 degree. The widest spray angle contributed by the 15% of water at 0.4MPa which is 46 degree and the narrowest spray angle was recorded by 5% of water at 1.3MPa is 5 degree. The increasing of the spray angle at 0.4MPa was 350% compared to the 300% increasing at the 1.3MPa. The spray angle was influence by the some properties included the injection pressure, the surface tension and the viscosity. Based on the differences injection pressure, the higher the injection pressure, the wider the spray angle and vice versa. The spray angle has an inverse effect on drop size. An increasing in the spray angle will reduce the droplet size, whereas the reduction in spray angle will increase the droplet size. The injection pressure has an inverse effect on droplet size. An increase in pressure will reduce the droplet size due to the decreasing the flow rate of the fuel.

The droplet size has direct effect with the flow rate. For case 0.4MPa, the spray angle is slightly wider than the 1.3MPa where the spray angle be wider with the increasing of the injection pressure but reduced by increasing in liquid density. At other point, the viscosity also influence the spray angle where the liquid with more viscous than water form the smaller spray angle. The emulsion fuel that has lowest viscous tend to reduce the droplet size and the flow rate. Hence, when the droplet size is reduced, the spray angle will be narrow. 15% of water has a lowest viscosity (0.00472) whereas 10% of water has 0.00532 and the 5% of water has 0.006. In additional, the spray angle of 15% of water is wider compared to the 5% of water due to the viscosity factor, hence the result shown is satisfied with the theory. The other factor that contribute to the spray angle development is the surface tension. The surface tension is an important physical property affecting surface formation, and makes the liquid resist breaking into droplets. The main effect of the surface tension is on the spray angle and droplet size of the sprayed fluid as well as the spray distribution. The surface tension has direct effect on the droplet size, where reducing the surface tension also will reduce the droplet size. Hence, the spray angle will be wider.

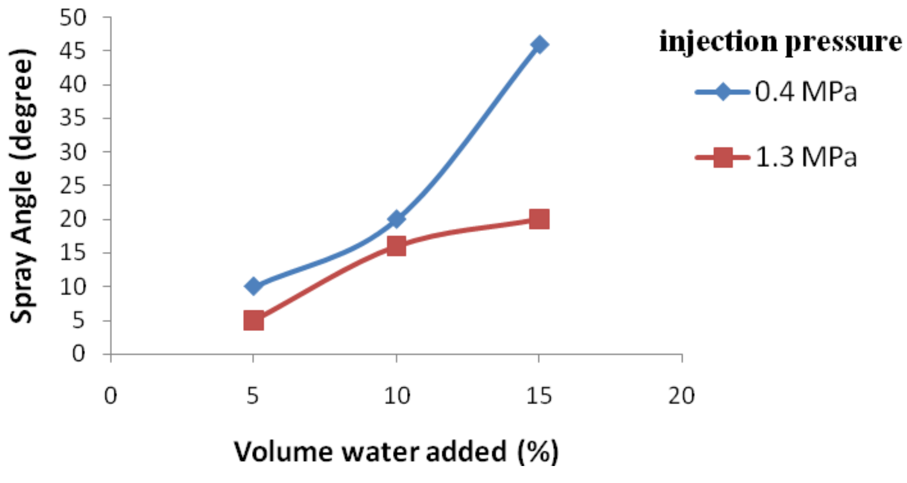


Figure 4.9: Percentage of water vs spray angle at 0.4MPa and 1.3MPa (VCO nozzle)

The spray penetration length influenced by many factors, the viscosity, the temperature, the geometry of nozzle and the injection pressure. All the discussion is strongly refer based on the knowledge and understanding of the spray characteristics and the behavior of the emulsion fuel. The next chapter will conclude and summaries the whole project.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

These chapters covered the overall overview about the whole analysis and introduce some recommendation to improve the more accurate analysis.

5.1 CONCLUSIONS

The influence of nozzle tip geometry, the injection pressure and the presence water added were studied over arrange of conditions representatives of a modern diesel engine. Significant differences in spray structure were observed between SAC and VCO nozzle. These could be due to the differences in internal flow structure within the nozzle at low needle lift. The expected trends with injection pressure were observed, except at high injection pressure where a lower rate of penetration. In addition, the increased rate of evaporation at high injection pressure may also influence the penetration rate of the spray. The injection pressure showed the positive result with the expected one. When the injection pressure increase, the spray penetration will also increase due to the reducing in the flow rate and viscosity.

The VCO nozzles were observed to show a lower rate of penetration than SAC nozzle of the same dimensions. It is proposed that this is due to better atomization of the spray from the VCO nozzle, possibly due to the difference in internal flow of structure, resulting in a higher rate evaporation. Generally in VCO nozzle, the holes lead directly to the needle seat and are therefore shut by the needle itself. Because of this feature, the fuel injection timing and quantities are very accurate but the fuel distribution to different is very sensitive to needle nozzle concentric during needle lift. This could lead to large discrepancies in spray formation between the different holes at small needle lifts. In SAC nozzle, the holes lead to a sac volume below the needle tip and are not directly closed by the needle. The fuel spray obtained with sac nozzle are not sensitive to needle dislocation since the distribution to each holes is balanced by the sac volume (Kennaird et al, 2002). Disadvantages of SAC holes when compared to the VCO nozzle is that the fuel in the SAC can result in fuel droplets leaving the injector after the needle has shut.

The injection pressures also influence the spray penetration length. As we know that, the spray penetration actually is the pressure difference across the nozzle and the air pressure inside the chamber. The greater the injection pressure, the longer the penetration will take place. Regarding the some research and reading, the very high injection pressures produce a small penetration length because the spray penetration will reduce at very high injection pressure. Pressure has an inverse effect on droplet size. An increase in pressure will reduce the droplet size, whereas the reduction in pressure will increase the droplet size.

Viscosity, on the other hand, is related to a liquid's resistance to being deformed or moved. This is caused by the friction between molecules. Compared to viscosity, surface tension is a simpler phenomenon. It is basically stable, changed mostly by temperature and chemical that modifies the bonding characteristics of the molecules. As temperature decrease, surface tension increase. From the simulation, by adding the percentage of water, the viscosity is reduce, result in shortest spray penetration and vice versa. 15% of water has the lowest viscosity compared to the 5% and 10% of water, hence it produced the shortest spray penetration. The temperature also played the important role in judging the spray penetration length. When the boiling point of the emulsion fuel increase , the viscosity is reduce. The same goes to the 15% of water that produced the shortest one. Increasing in temperature will reduce the viscosity by breaking the intermolecular force between the molecules.

Spray angle also has related to the viscosity and injection pressure. The spray angle became wider with increasing the injection pressure but reduced by increasing the emulsion fuel viscosity. Here, the surface tension also took place to the spray angle by reducing the droplet size. When the droplet size is reduced, the spray angle became more wider. The wider the spray angle, the faster the atoms will react with the air inside the chamber and the complete combustion happened in short time.

5.2 RECOMMENDATIONS

For further study, here I would like to make some recommendations regarding to this project. In future, I hope this project will be carried on in experimental concept because it has more clear understanding and experience. By having experimental result, the spray images will be clearly seen compared to the simulation. Some modifications can be done to the injection parameters include the nozzle size that can be varied to several diameter, use the multi-hole injector, increase the injector pressure, prepare the emulsion fuel by own, consider the condition inside the chamber and investigate the emission produce by the engine. For simulation purposes, I recommend to use high power computer to create the smallest meshing to the model. By having the smallest meshing, the spray images can be clearly seen. In addition, the iterations of the simulation can be increased approaching 300 iteration that which give more accurate result and images because it take time during run the project.

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