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THEORETICAL BACKGROUND AND LITERATURE REVIEW ON INTAKE AND EXHAUST SYSTEMS OF DECK'S DIESEL ENGINES

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ABSTRACT: In this review research, the air intake and exhaust systems in internal combustion deck's engine are investigated and studied from the considerations of Introduction; Theoretical Background which includes: Introduction, Gas Exchange Processes, Quasi Static Effects – Volumetric Efficiency of an Ideal Cycle, Effect of Fuel Composition, Phase, and Fuel / Air Ratio, Combined Quasi – Static and Dynamic Effects, Frictional Losses, Ram Effect, Variation with Speed, and Valve Area, Lift and Timing, Exhaust Gas Flow Rate and Temperature Variation, Diesel Engine, Volumetric Efficiency, Air Path. The Literatures Review are discussed from the following viewpoints: Air and Fuel Induction, Exhaust flow, Air System, Turbo Charging System, Turbo Charged Diesel Engine Models, Methods of Power Boosting, Supercharging of Diesel Engine, Diesel-Engine Compression Ratio, Performance of Supercharged Diesel Engines, Compressor and Turbine, Intercooler, Intake Manifold, Combustion and Torque Production, Exhaust Manifold, Effect of Length of IM Runners on the Volumetric Efficiency; and finally comes the important conclusions of this study.

KEYWORDS: Intake; Exhaust; Deck's diesel engines; Theory; Literature review

1 INTRODUCTION

There are many types and arrangements of internal reciprocating deck's engines, and some classification is necessary to describe a particular deck's engine adequately. Two main methods of classification are employed, the first is by the fuel used and the way in which the combustion is initiated, and the second is by the way in which the cycle of processes is arranged. When a deck's engine is selected to suit a particular application, the main consideration being its power/speed characteristics. Important additional factors are initial capital cost and running cost. In order that different types of engines or different engines of the same type may be compared, certain performance criteria must be defined. These are obtained by measurements of the quantities concerned during bench test, and calculation is done by standard procedures. The results are plotted graphically in the form of performance curves.

The testing of a particular deck's engine consists of running it at different loads and speeds and taking sufficient measurements for the performance criteria to be calculated. The intake systems and flow processes are of a major importance in the performance tests. They are influenced by so many factors; e.g., configuration of the intake system, which includes; valve lift, profile, design, and discharge coefficients; manifold dynamics and ram effect; inlet Mach number index; and volumetric efficiency which is governed by the factors; fuel type, phase, and amount; intake manifold temperature; compression ratio; inlet/exhaust pressure ratio; frictional losses. Study proposed will involve the influences of all the above-mentioned factors on the intake systems and flow processes in internal reciprocating deck's engines.

The effect of the volumetric efficiency on the deck's engine effective power is investigated in this work. The effective power is found to be influenced by the volumetric efficiency parameters in different ways, depending on the influencing parameters. Some parameters result in proportional relation while others result in an inverse relation. Refer to [1] – [8].

2 THEORETICAL BACKGROUND

2.1 Introduction

When air enters to the engine cylinder, it must pass through different components like, air cleaner, intake plenum, and intake port and intake valve. Each of these components affects the velocity and pressure of the flow and thus has impact on the volumetric efficiency of the engine.

The intention of the intake system is to minimize pressure losses at wide open pass to the cylinder. Most engine performance is aimed to increasing the airflow rate through the engine. Thereby increasing the amount of fuel that can be burned per unit time by:

* Increased valve area.
* Increased valve lift duration.
* Intake and exhaust tuning.
* Low pressure drop, air clearance and manifolds.
* Free flow exhausts system.
* Increased displaced volume.
* Variable valve timing.
* Increase maximum engine speed.
* Use turbo charging.

2.2 Gas exchange processes

This deals with the fundamentals of the gas exchange processes intake and exhaust in four – stroke cycle engines and scavenging in two – stroke cycle engines. The purpose of the exhaust and inlet processes or of the scavenging process is to remove the burned gases at the end of the power stroke and admit the fresh charge for the next cycle. Equation (1), shows that the indicated power of an internal combustion engine at a given speed is proportional to the mass flow rate of air [9].

$$P = \frac{\eta_f m_a N Q_{HV} \left(\frac{F}{A}\right)}{n_R} \rightarrow \qquad (1)$$

$P = power.$
η_f = fuel conversion efficiency.
n_R = number of crank revolution per stroke.
Q_{HV} = fuel heating value.
$N = crankshaft\ rotational\ speed.$
$m_a = mass\ of\ air\ per\ cylinder\ per\ cycle.$

Thus, inducting the maximum air mass at wide – open throttle or full load and retaining that mass within the cylinder is the primary goal of the gas exchange processes. Engine gas exchange processes are characterized by overall parameters such as volumetric efficiency (for four – stroke cycles), and scavenging efficiency and trapping efficiency (for two – stroke cycles). These overall parameters depend on the design of engine subsystems such as manifolds, valves, and ports, as well as engine operating conditions. Thus, the flow through individual components in the engine intake and exhaust system has been extensively studied. Supercharging and turbo charging are used to increase airflow through engines, and hence power density. Obviously, whether the engine is naturally aspirated or supercharged (or turbocharged) significantly affects the gas exchange processes, [9].

For spark – ignition engines the fresh charge is fuel, air, and (if used for emission control) recycled exhaust, so mixture preparation is also an important goal of the intake process. Mixture preparation includes both achieving the appropriate mixture composition and achieving equal distribution of air, fuel, and recycled exhaust amongst the different cylinders. In diesels, only air (or air plus recycled exhaust) is inducted.

A third goal of the gas exchange processes is to set up the flow field within the engine cylinders that will give a fast-enough combustion process for satisfactory engine operation.

In a diesel engine intake system, the carburetor or electronic fuel injection (EFI) system and the throttle plate are absent. Diesel engines are more frequently turbocharged. Figure 1 below shows the intake and exhaust process for turbocharged four-stroke cycle engine. The turbocharger compressor C raises air-pressured temperature from ambient P_a, T_a to P_i, T_i. Cylinder pressure during intake is less thanP_i. During exhaust, the cylinder gases flow through the exhaust manifold to the turbocharger turbine T. Manifold pressure P_e, may vary during the exhaust process and lies between cylinder pressure and ambient, [9].

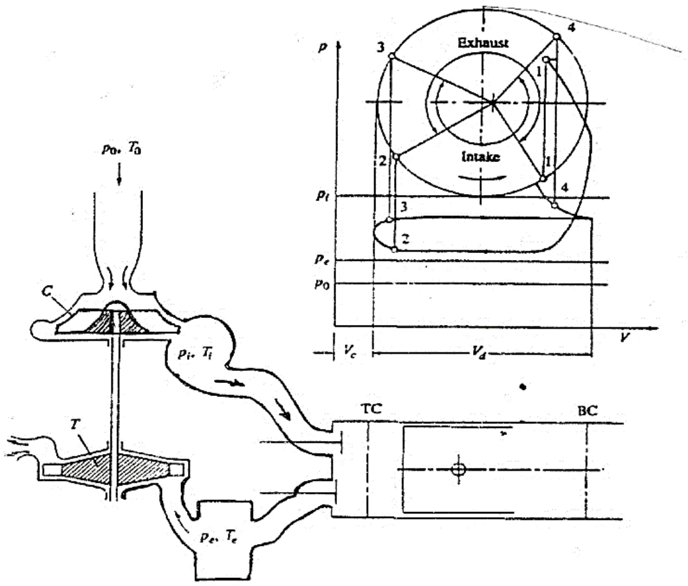

Figure 1. The intake and exhaust process for turbocharged four-stroke cycle engine

A similar set of diagrams illustrating the intake and exhaust processes for a turbocharged four – stroke diesel is shown in Figure 1 when the exhaust valve opens, the burned cylinder gases are fed to a turbine, which drives a compressor that compresses the air prior entering the cylinder. Due to time changes, valve open area and cylinder volume, gas inertial effects, and wave propagation in the intake and exhaust systems, the pressures in the intake, cylinder and exhaust vary during these gas exchange processes. In practice, these processes are often treated empirically using overall parameters such as volumetric efficiency to define intake and exhaust system performance, [9].

2.3 Quasi static effects – Volumetric efficiency of an ideal cycle

For the ideal cycles, an expression for volumetric efficiency can be derived which is a function of the following variables: intake mixture pressure P_i, temperatureT_i, and fuel / air ratio (F/A), compression ratior_c, exhaust pressureP_e, molecular weight M and γ for the cycle working fluid. Therefore, the overall volumetric efficiency is:

$$\eta_v = \frac{m_a}{\rho_{a,0} V_d} = \frac{m(1 - x_r)}{\rho_{a,0}[1 + (F/A)]} \frac{r_c}{(r_c - 1)V_1} \rightarrow \tag{2}$$

Where m is the mass in the cylinder at point 1 in the cycle. Since,

$$p_{a,0} = \rho_{a,0} \frac{\tilde{R}}{M_a} T_{a,0}$$

and $$p_i V_1 = m \frac{\tilde{R}}{M} T_1$$

The expression for η_v can be written as follows:

$$\eta_v = \left(\frac{M}{M_a}\right)\left(\frac{p_i}{p_{a,0}}\right)\left(\frac{T_{a,0}}{T_i}\right)\frac{1}{[1+(F/A)]}\left\{\frac{r_c}{r_c-1} - \frac{1}{\gamma(r_c-1)}\left[\left(\frac{p_e}{p_i}\right)+(\gamma-1)\right]\right\} \rightarrow (3)$$

2.4 Effect of fuel composition, phase and fuel / air ratio

In a spark Ignition engine, the presence of gaseous fuel (and water vapor) in the intake system reduces the air partial pressure below the mixture pressure. For mixtures of air, water vapor, and gaseous or evaporated fuel, we can write the intake manifold pressure as the sum of each component's partial pressure:

$$p_i = p_{a,i} + p_{f,i} + p_{w,i}$$

Thus, with ideal gas law gives:

$$\frac{P_{a,i}}{P_i} = \left[1 + \left(\frac{\dot{m}_f}{\dot{m}_a}\right)\left(\frac{M_a}{M_f}\right) + \left(\frac{\dot{m}_w}{\dot{m}_f}\right)\left(\frac{M_a}{M_w}\right)\right]^{-1} \rightarrow \quad (4)$$

The water vapor correction is usually small (≤ 0.03). This ratio, $\frac{P_{a,i}}{P_i}$ for several common fuels as a function of $\left(\frac{\dot{m}_f}{\dot{m}_a}\right)$ is shown in Figure 2. Note that, $\left(\frac{\dot{m}_f}{\dot{m}_a}\right)$ only equals the engine operating fuel / air ratio if the fuel is fully vaporized, [9].

Figure 2 below illustrates the effect of fuel (vapor) on inlet air partial pressure.

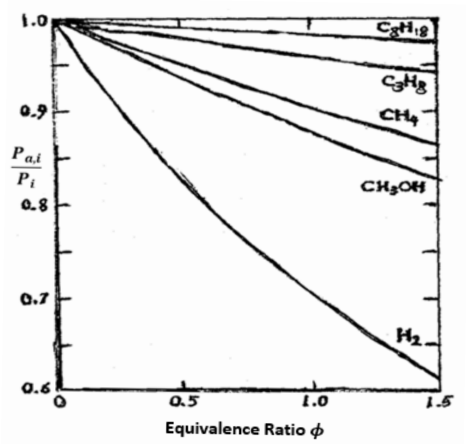

Figure 2. Air fuel pressure ratio versus equivalence ratio ϕ

The ratio of air inlet pressurep_a, to mixture inlet pressure, p_i versus fuel/air equivalence ratio ϕ for iso-octane vapor, propane, methane, methanol vapor, and hydrogen, [9]. For conventional liquid fuels such as gasoline the effect of fuel vapor, and therefore fuel / air ratio, is small. For gaseous fuels and for methanol vapor, the volumetric efficiency is significantly reduced by the fuel vapor in the intake mixture.

2.5 Combined quasi – Static and dynamic effects

When gas flows unsteadily through a system of pipes, chambers, ports, and valves, both frictions, pressure, and inertial forces are present.

The relative importance of these forces depends on gas velocity and the size and shape of these passages and their junctions. Both quasi-steady and dynamic effects are usually significant. While the effects of changes in engine speed, and intake and exhaust manifold, port and valve design are interrelated, several separate phenomena, which affect volumetric efficiency, can be identified, [9].

2.6 Frictional losses

During the intake stroke, due to friction in each port of the intake system, the pressure in the cylinder p_c is less than the atmospheric pressure $p_{atmos.}$by an amount dependent on the square of speed. This total pressure drop is the sum of the pressure loss in each component of the intake system: air filter, carburetor and throttle, manifold. In inlet port and inlet valve, each loss is a few percent, with the port and valve contributing the largest drop.

As a result, the pressure in the cylinder during the period in the intake process when the piston is moving at close to is maximum speed can be 10 to 20 percent lower than atmospheric. For each component in the intake (and the exhaust) system, Bernoulli's equation gives:

$$\Delta p_j = \xi_j \rho v_j^2$$

Where ζ_j, is the resistance coefficient for that component which depends on its geometric details andv_j, "the local velocity". Assuming the flow is quasi-steady, v_j is related to the mean piston speed $\bar{S}_p$ by:

$$v_j A_j = \bar{S} A_p$$

Where A_j and A_p are the component of minimum flow area and of the piston area respectively. Hence, the total quasi – steady loss due to friction is:

$$p_{atm} - p_c = \sum \Delta p_j = \sum \zeta_j \rho v_j^2 = \rho \bar{S}_p^2 \sum \zeta_j \left(\frac{A_2}{A_J}\right)^2 \rightarrow \qquad (5)$$

Equation (5) indicates the importance of large component flow areas for reducing frictional losses, and the dependence of these losses on engine speed, [9].

2.7 Ram effect

The pressure in the inlet manifold varies during each cylinder's intake process due to the piston velocity variation, valve open area variation, and the unsteady gas – flow effects that result from these geometric variations. The mass of air inducted into the cylinder, and hence the volumetric efficiency, is almost entirely determined by the pressure level in the inlet port during the short period before the inlet valve is closed. At higher engine speeds, the Inertia of the gas in the intake system as the intake valve is closing increases the pressure in the port and continues the charging process as the piston slows down around BC and starts the compression stroke. This effect becomes progressively greater as engine speed is increased. The inlet valve is closed some 40º to 60º after BC, in part to take advantage of this ram phenomenon, [9].

2.8 Variation with speed, and valve area, lift and timing

Flow effects on volumetric efficiency depend on the velocity of the fresh mixture in the intake manifold, port, and valve. Local velocities for quasi – steady flow equal the volume flow rate divided by the local cross – sectional area. Since, the intake system and valve dimensions scale are proportional to the cylinder bore, mixture velocities in the intake will scale with piston speed. Hence, volumetric efficiency as a function of speed, for different engines should be compared at the same mean piston. Figure 3 shows typical curves of volumetric efficiency versus mean piston speed for a four–cylinder automobile indirect injection diesel engine and a six – cylinder spark – ignition engine.

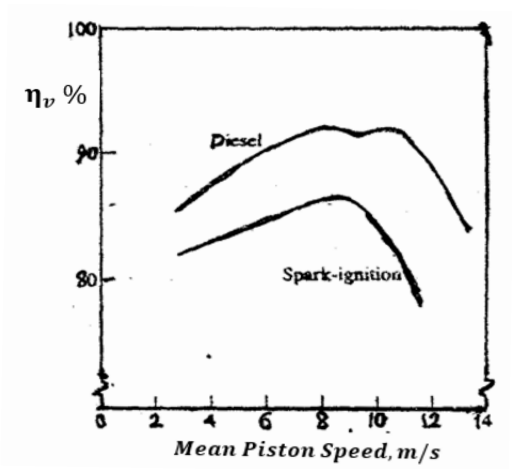

Figure 3. Volumetric efficiency versus mean piston speed for a four-cylinder automobile indirect-injection diesel and a six-cylinder spark-ignition engine

The volumetric efficiencies of spark–ignition engines are usually lower than diesel values due to flow losses in the carburetor and throttle, intake manifold heating, the presence of fuel vapor, and a higher residual gas fraction. The diesel curve with its double peak shows the effect of intake system tuning, [9] and [10].

The shape of this volumetric efficiency versus mean piston speed curves can be explained with the aid of Figure 4 which shows, in schematic form, how the effect on volumetric efficiency of the different phenomena described in this section varies with speed.

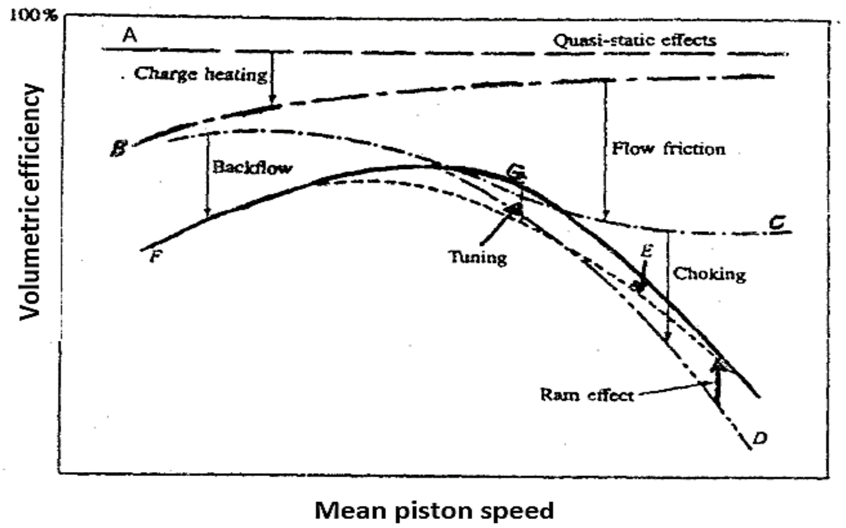

Figure 4. Effect on volumetric efficiency of different phenomena, which affect the airflow rate as a function of speed

Non – speed dependent effects (such as fuel vapor pressure) drop η_{vol} below 100 percent (curve A). Charge heating in the manifold and cylinder drops curve A to curve B. It has a greater effect at lower engine speeds due to longer gas residence times. Frictional flow losses increase as the square of engine, and drop curve B to curve C. At higher engine speeds becomes chocked. Once this occurs, further increases in speed do not increase the flow rate significantly so volumetric efficiency decreases sharply (curve C to D).

The induction ram effect at higher engine speed raises curve D to curve E. Late inlet valve closing, which allows advantage to be taken of increased charging at higher speeds, results in decreasing η_{vol} at low engine speeds due to backflow (curves C and D to F). Finally, intake and /or exhaust tuning can increase the volumetric efficiency (often by a substantial amount) over part of the engine speed range. Refer to curve F to G.

An example of the effect on volumetric efficiency of tuning the intake manifold runner is shown in Figure 5. In an unsteady flow calculation of the gas exchange processes of a four – cylinder spark – ignition engine, the length of the intake manifold runners was increased successively by factors of 2. The 34 cm length produces a desirable "tuned" volumetric efficiency curve with increased low – speed airflow and flat mid – speed characteristics. While the longest runner further increases low – speed airflow, the loss in volumetric efficiency at high speed would be unacceptable, [9].

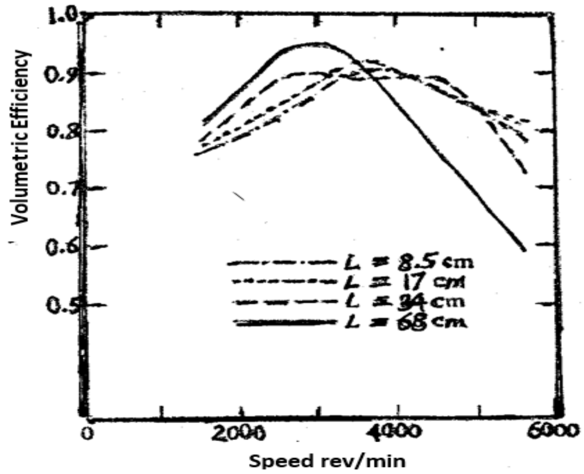

Figure 5. Effect of intake runner length on volumetric efficiency versus speed for 23-dm^3 four-cylinder spark-ignition engine

2.9 Exhaust gas flow rate and temperature variation

The exhaust gas mass flow rate and the properties of the exhaust gas vary significantly during the exhaust process. The thermodynamic state (pressure, temperature, etc.) of the gas in the cylinder varies continually during the exhaust blow down phase, until the cylinder pressure closely approaches the exhaust manifold pressure. In the real exhaust process, the exhaust valve restricts the flow out of the cylinder, the valve lift varies with time, and the cylinder volume changes during the blow down process, but the principles remain the same, [9].

2.10 Diesel engine

According to H.B. Servati and R.G. (1986), the fuel used in almost all heavy-duty trucks today is diesel. The principle after which the diesel engine works is called compression ignition (CI). It means that the diesel is injected into the compressed air and ignited [11] and [12]. Hakan Bengtsson (2002) say that, there are two important differences between the diesel and the gasoline engine when discussing volumetric efficiency. Diesel engine generally have higher compression ratio than gasoline engines. The other difference is that diesel engines have no throttle inlet manifold that gasoline engines have. Generally, diesel engines have higher volumetric efficiency than gasoline engines [13].

2.11 Volumetric efficiency

The intake system restricts the amount of air that can be inducted into the cylinder during one cycle. The volumetric efficiency is the parameter describing the effectiveness of the induction process. The induction process is defined as all events taking place between the inlet valve opening (IVO) and the inlet valve closing (IVC).

Definitions:

There are some different definitions of the volumetric efficiency, in the literature. One definition often used is described as:

$$\eta_{vol} = \frac{120 w_{ic}}{\rho_{im} v_d n_{engine}} \rightarrow \tag{6}$$

Where w_{ic} is the mass flow through the intercooler, ρ_{im} is the density in the inlet manifold. The displacement volume for the engine is v_d and the engine speed is n_{engine}.

An equivalent definition is:

$$\eta_{vol=} \frac{m_c Mix}{\rho_{im} v_d} \rightarrow \tag{7}$$

The total mass of fresh mixture in the combustion champers is denoted by$m_c\, Mix$. The inlet density ρ_{im} is taken in the inlet manifold, which means that only the performance of the inlet valve and the inlet port are considered. Another possibility is to use the atmosphere density. In that case, the performance of the total air path would be considered. However, this often not used for turbo charged engine where the inlet pressure often is higher than the surrounding atmosphere pressure.

Another definition that slightly differ from definition above in characteristics and application is proposed in the following formula:

$$\eta_{vol} = \frac{m_c Mix + m_c R_{es}}{\rho_{im} v_d} \longrightarrow \tag{8}$$

The mass is divided into two parts, fresh mixture, $m_c\, Mix$ and residual gas, $m_c\, R_{ES}$. In this definition, the total mass of residual gas trapped in the cylinder is included in the volumetric efficiency. This definition is practical in some measurement techniques, where the cylinder pressure is used to determine the total mass of gas trapped in the cylinder. In Equations (6) and (7) only the mass of fresh mixture is considered, not the mass of residual gas. Fresh mixture is the new gas that is introduced to the cylinder through the inlet value, i.e., not including (EGR).

In diesel engine, the fresh mixture contains of air, which e.g., contains of water vapor, oxygen and nitrogen. The gas left in the charge from previous cycle is called residual gas. The charge is the content in the cylinder when all valves are closed. Another slightly modified version is used:

$$\eta_{vol} = \frac{m_c Mix + m_c Egr}{\rho_{im} v_d} \rightarrow \tag{9}$$

The different is a wider interpretation of which mass trapped in the cylinder to be considered. Besides the fresh mixture also the mass from EGR,$m_c\,Egr$ is counted here. If the engine is not equipped with an EGR system, the two definitions in Equations (8) and (7) are equal. However, the mass of residual gas and the mass of back flow are not counted.

Back flow occurs when the intake and the exhaust valve are open at the same time and exhaust gas can flow from the exhaust port to the inlet port.

There are two reasons why Equation (8) is preferred. The main reason is that it fits better into the MVEM. This is because the mass flow from the fresh mixture and the EGR are merged in the inlet manifold. The second reason is that it gives a simpler model. If the mass of EGR must be separated from the mass of fresh mixture, one more variable must be used in the model [14].

2.12 Air path

In the combustion process, fuel and oxygen are needed to produce torque. Besides that, heat and exhaust gas are produced. There are legal restrictions on how much emission that is admitted. The emissions are e.g., nitric oxides and smoke. The emissions depend among other on how the air to fuel ratio was in the combustion. Therefore, it is important to know how the amount of air that can be inducted depends on different factors. Often the volumetric efficiency is used to calculate how much air that is inducted in the cylinder. The volumetric efficiency is important both in the design of the engine and in the electronic control system, e.g., to know how much fuel to inject. The air path has in this study been divided into three parts: Inlet, volumetric and Exhaust [13].

2.12.1 Inlet part

The inlet part of the air path describes how the air is affected in the path from the atmosphere to the inlet manifold. Effect of the EGR system is included in both the inlet and the exhaust parts. The effects in inlet part will affect the amount of air that can be inducted into the cylinder. However, according to the definition of volumetric efficiency in Equation (8), only the performance of the inlet part and valve are included. Therefore, the inlet part of the air path does not directly affect the volumetric efficiency. A schematic schedule, which shows the inlet part of the air path, is presented in Figure 6, [13].

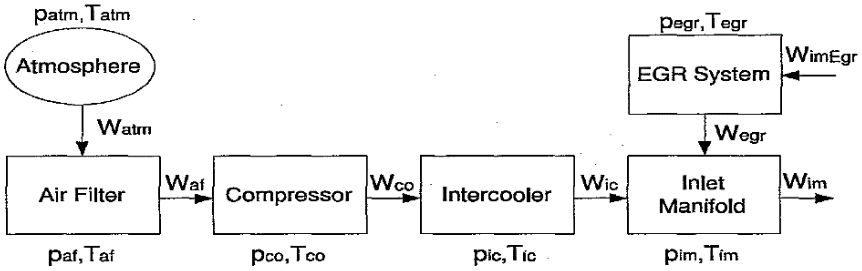

Figure 6. A schedule of the flow through the inlet part (Arrows shows the positive flow direction)

2.12.1.1 Air filter

The first component in the air path is the air filter, which cleans the air from pollutants. Over the air filter, there will be a small pressure drop [13].

2.12.1.2 Compressor

A turbo charger has two separate components that affect the mass flow through the engine: the compressor and the turbine. The compressor pumps the fresh mixture into the inlet manifold and thus increases the density of the air. The gas temperature after the compressor will increase according to thermodynamics laws. The energy needed to drive the compressor comes from the turbine via a connecting turbine shaft.

If the engine is equipped with a VGT the geometry of the turbine can be controlled, which gives an extra degree of freedom to the system. In Figure 7, the difference between an ordinary turbine and a VGT can be seen. With an ordinary turbine, only one position is possible and thus only one line. In VGT the pressure in the inlet manifold and the exhaust manifold are plotted with three different VGT control signals, [5].

As the figure shows, the pressure quotient over the engine$\left\{\frac{P_{em}}{P_{im}}\right\}$ is approximately constant for a fix VGT position. However, the constant differs with the control signal.

The pressure quotient over the engine has been assumed constant in some models of volumetric efficiency. The figure clearly shows that the quotient significantly varies with the VGT control signal.

$$\frac{P_{em}}{P_{im}} = f(u_{vgt}) \longrightarrow \tag{10}$$

Where $u_{vgt}$is the control signal to the "VGT". Therefore, it is important to have a model of volumetric efficiency that works with different pressure quotient.

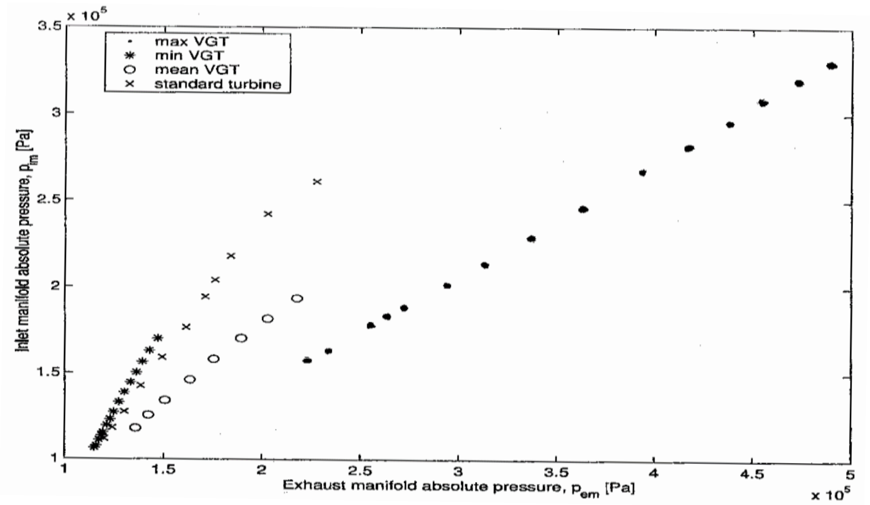

Figure 7. Measurements with three different control to the VGT and one with a standard turbine

2.12.1.3 Intercooler

The intercooler cools the air heated by the compressor and therefore increases the density. The air flows through many thin pipes. Around these pipes, air with lower temperature is streaming and cooling the air inside the pipes. Thus, a pressure drops also over the intercooler.

2.12.1.4 Inlet manifold

The inlet manifold is the volume placed just before the inlet part. It only acts as a volume and no pressure drop occurs over it. The function is to divide the airflow from the intercooler to the different inlet parts leading to each combustion chamber. A principal sketch over the inlet manifold is seen in Figure 8. In the definition of volumetric efficiency in Equation (8) the pressure and temperature of the air in the inlet manifold are used, [13].

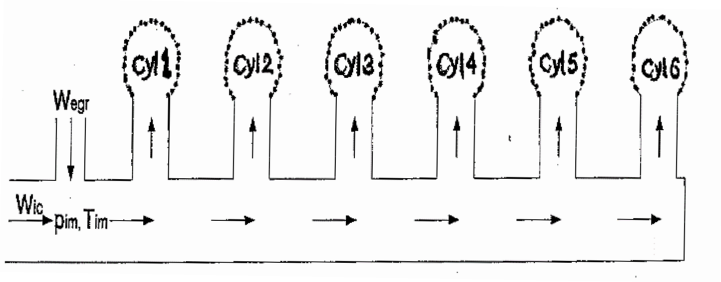

Figure 8. Sketch over the inlet manifold

2.12.1.5 EGR system

If the engine is equipped with an EGR system, exhaust gas is added to the fresh mixture and mixed. The purpose is to affect the combustion process so that the temperature at combustion is lower which will give less nitric oxide in the exhaust gas [13].

2.12.2 Volumetric part

The effects on the air between the inlet manifold and the exhaust manifold has been defined as the volumetric part of the air path.

Instead, the volumetric efficiency takes care of these effects. This is where the model of volumetric efficiency comes in. A schematic schedule, which shows the volumetric part of the air path, is presented in Figure 9, [13].

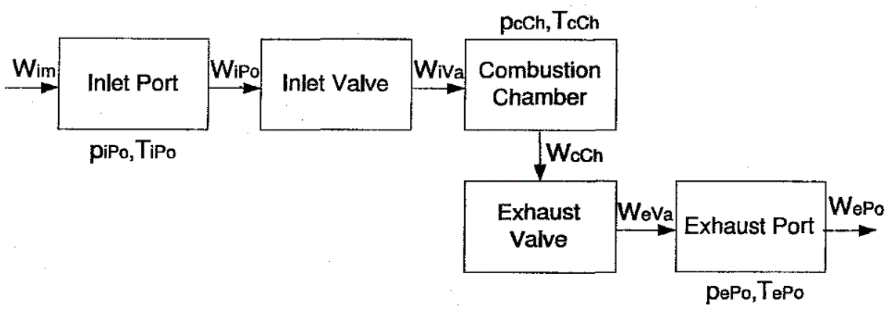

Figure 9. A schedule of the flow through the volumetric part (Arrows shows the positive flow direction)

2.12.2.1 Inlet part

When the air flows through the inlet part, heat is exchanged between the metal and the gas. This will rise the temperature on the gas while passing through the inlet port. This phenomenon is called heat transfer and will affect the temperature in the gas when it flows into the combustion chamber.

2.12.2.2 Inlet valve

The inlet valve controls the gas flow from the inlet manifold into the combustion chamber. The valve is controlled via the camshaft, which lifts the valve from its closed position. The camshaft has a theoretical curve profile, which described how the valve lift depends on the crank angle. For the engine used in the measurements. A simple sketch over the volumetric part is shown in Figure 10, [13].

The valve head is placed in the combustion chamber and therefore the valve will be very hot and will expand. Thus, there must be a clearance. This means that the actual clearance depending on the temperature must be subtracted from the theoretical curve to get the actual lift.

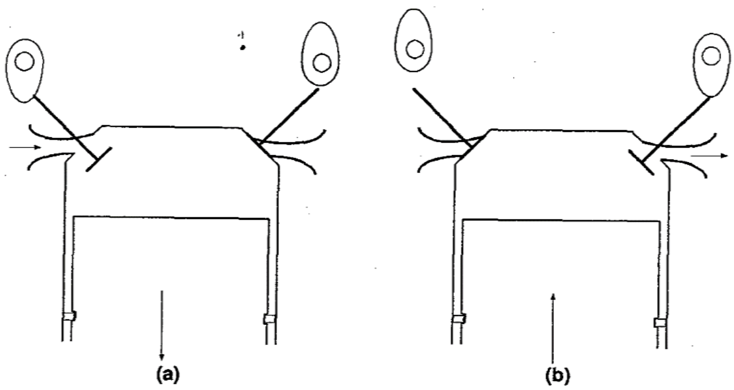

Figure 10. A sketch of the volumetric part of the air path. Arrows shows the positive flow direction and the piston movement. (a) Intake stroke, (b) Exhaust stroke

The flow through a valve is often expressed by the equation for compressible flow through a flow restriction, which is derived from the one – dimension isentropic flow, [13].

$$\dot{m} = \frac{C_D A_R p_0}{\sqrt{RT_0}} \psi \left(\frac{p_T}{p_o}\right) \rightarrow \tag{11}$$

Where $\dot{m}$ denotes mass flow through the valve and

$$\frac{p_T}{p_0} \leq \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}$$

Here C_D is an experimentally determined discharge coefficient, P_o andT_o the upstream pressure and temperature. The constant A_R is the reference area, which varies with the actual lift. The variable P_T is the pressure at the throat, i.e., where the flow area is smallest. The constant γ is the specific heat ratio taken for the gas flows through the valve.

When the condition $\frac{P_T}{P_O} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma+1}}$is valid, the flow is chocked. This means that the maximum flow rate is reached and the flow has become saturated.

2.12.2.3 Combustion chamber

In the combustion chamber, the temperature will be very high and the cylinder walls will be much hotter than the gas during the intake. Thus, if heat transfer from the metal to the gas is considered significant, this will be much larger than the energy that is transferred in the inlet port. The volume of the combustion chamber varies with the crank angle.

2.12.2.4 Exhaust valve

The exhaust valve is similar to the inlet valve. One difference is that it is harder to cool the exhaust valve and thus, more clearance normally is needed. For the exhaust valve, Equation (11) is also valid. The dominating flow direction is from the intake port via the combustion chamber to the exhaust port. However, two phenomena do not follow this and are very important in the study of volumetric efficiency that are back flow and residual gas.

As can be seen in Figure 11 the inlet valve opens before the exhaust valve has closed. If the pressure in the inlet manifold is below the pressure in the exhaust manifold, the flow will go in the negative direction.

The residual gas is the exhaust gas that remains in the cylinder after the blow out. If the pressure in the cylinder is higher than the inlet manifold pressure, gas will flow out in the inlet port when the inlet valve opens.

The mass of back flow and residual gas takes room from the fresh mixture and thus they will decrease the volumetric efficiency. The back flow and the residual gas depend on the properties of the gas directly after the exhaust valve. Figure 11 below shows the overlap between the inlet and the exhaust valves curves [5].

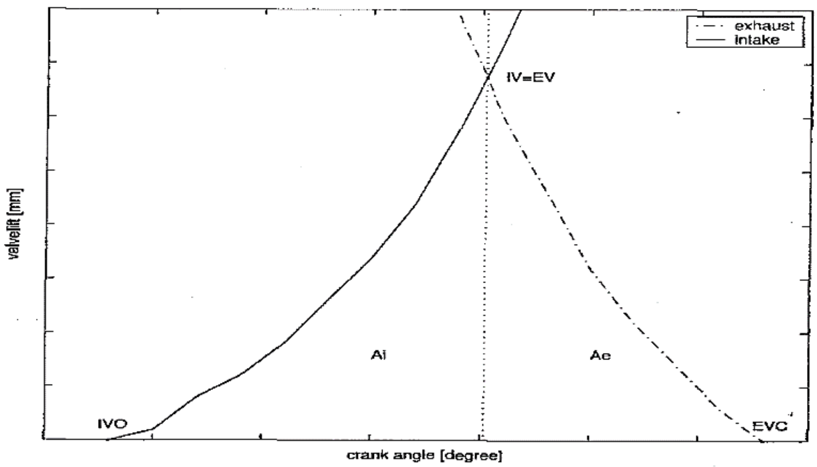

Figure 11. Overlap between the inlet and the exhaust valves curves

2.12.3 Exhaust part

All effects that take places from the exhaust manifold to the atmosphere are called the exhaust part of the air path. In addition, the EGR system is included in the exhaust part. A schematic schedule, which shows the exhaust part of the air path, is presented in Figure 12, [13].

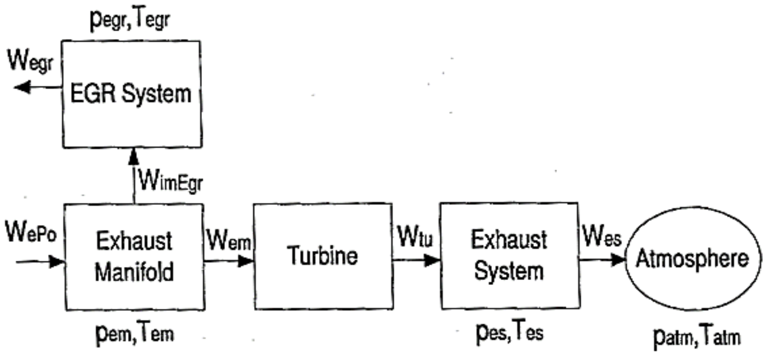

Figure 12. A schedule of the flow through the exhaust part. Arrows shows the positive flow direction

2.12.3.1 EGR system

The EGR system involves pipes, EGR coolers and a variable valve used to control the amount of EGR flow. When the EGR is added to the air in the inlet manifold, it affects the temperature and the pressure there.

The definition of volumetric efficiency in Equation (9) includes both the mass of fresh mixture and EGR. The physical constants used in Equation (11) are approximately equal for exhaust gas and fresh mixture. Thus, not all the components in the volumetric part will behave different with EGR from with fresh mixture [15].

2.12.3.2 Exhaust manifold

After the exhaust port there is an exhaust manifold. The function is opposite to the inlet manifold: To bring the flow from each exhaust port into one pipe. The exhaust manifold is modeled as a volume with no pressure drop.

The pressure that is important in the study of volumetric efficiency is the one just before the exhaust valve closes. Another measurement related problem occurs when the exhaust valve opens. Inside the combustion chamber, gas at very high pressure is confined when the exhaust valve opens the gas expands quickly out from the combustion chamber and causes a pressure peak in the exhaust manifold.

The same phenomenon is present in the inlet manifold. However, the amplitude is larger in the exhaust manifold and thus, more important to have in mind. In fact, this phenomenon is present in all measurements of volumetric efficiency. In Figure 13, a simulated exhaust manifold pressure over one cycle is shown, [13] and [16].

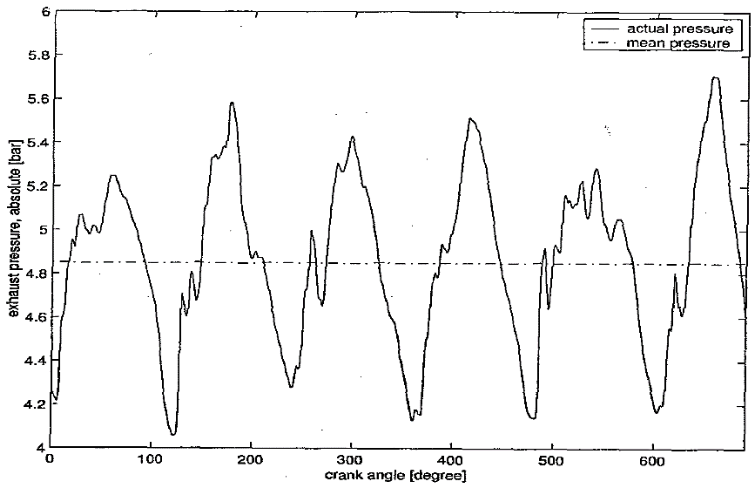

Figure 13. Simulated exhaust manifold pressures over one cycle

2.12.3.3 Turbine

The turbine is the second part of the turbo. The purpose is to utilize some of the energy wasted with the exhaust gas and to transform it into torque, used to drive the compressor. In a function, describing the mass flow through the turbine is used. The function depends on the quotient $\frac{P_{em}}{P_{im}}$ the turbine speed and the control signal,u_{vgt}, that controls the geometry of the VGT.

2.12.3.4 Exhaust system

Between the atmosphere and the turbine there is a pipe leading the exhaust gas and a muffler that should reduce the noise. Over the muffler, there is a pressure droop. In future engines there also can be a catalyst used for after treatment of pollutants [13].

3 LITERATURES REVIEW

3.1 Air and fuel induction

This research study describes intake system of deck's engine and how air and fuel are delivered into the cylinder. J. Bitesus (2002) says that the object of the intake system is to deliver the proper amount of air and fuel accurately and equally to all cylinder at the proper time in the engine cycle. Flow into an engine is pulsed as the intake valves open and close but can generally be modeled as quasi- steady state flow. Correct, consistent induction of air and fuel into an engine is one of the most important and difficult processes to obtain in engine design. High volumetric efficiency of intake system, giving a maximum flow of air, is important to supply the oxygen needed to react with the fuel [9].
Ideally, the engine should receive a constant amount of air cylinder to cylinder

and cycle to cycle. This does not happen due to turbulence and other flow inconsistencies, and engine operation must be limited by statistical averages [9]. M. Nilsson (2000) says that air is supplied through an intake manifold, with flow rate controlled on SI engine by throttle butterfly valve and uncontrolled on CI engines. Inlet air pressure is ambient or is increased with super charger, a turbo charger, or crank case compression CI engines inject fuel directly into the combustion chamber and control engine speed by injection amount, [17].

3.2 Exhaust flow

After combustion is completed and the resulting high-pressure gases have been used to transfer work to the crankshaft during the expansion stroke, those gases must be removed from the cylinder to make room for the air-fuel charge of the next cycle [18].

Diana Fan Kiev. (1995) said that the exhaust process of a four-stroke cycle CI engine is two-step process: blow down and exhaust stroke. Blow down occurs when the exhaust valve opens late in the expansion stroke and the remaining high pressure in the cylinder force the exhaust gases through the open valve, sonic velocity occurs and flow is choked, As the exhaust gas experiences blow down, the temperature decreases due to expansion cooling. The high kinetic energy of the gas during blow down is dissipated quickly in the exhaust manifold. In addition, there is a momentary rise in the temperature again from the resulting increase in specific enthalpy. The exhaust valve must open soon enough so that blow down is complete when the piston reaches BDC. At this point, the cylinder is still filled with exhaust gas at about atmospheric pressure, and most of this is now expelled during the exhaust stroke, [10].

J. Bitesus (2002) declares that two stroke cycle engines experience exhaust blow down but have no exhaust stroke. Most of the gas that fills the cylinder after blow down is expelled by as scavenging process when inlet air enters at elevated pressure.

To reduce the generation of nitrogen oxides many deck's engines have exhaust gas recycling, with some of the exhaust flow ducted back into the intake system. That engine equipped with turbochargers use the exhaust flow to drive the turbine, which in turn drives the inlet compressor [9].

3.3 Air system

Merrion, D.F (1994) said: The air system is composed of the following components: turbo charger, charge cooler (or inter cooler), intake manifold and exhaust manifold. The purpose of the engine's intake air system is to supply fresh air to the engine power cylinders in the proper quantity and under the proper condition of pressure and temperature to meet the needs for cylinder scavenging and combustion. Similarly, the purpose of the engine exhaust system is to collect, transport, and discharge the exhaust gases from the engine cylinder to the vehicle exhaust system [19].

The engine designer should define the maximum intake airflow, the maximum exhaust temperature, and the maximum exhaust volume flow, for the restrictions that can occur when the engine is installed in the vehicle. Typically, the maximum air intake restriction set by the individual OEM is 12 in H_2O (three K Pa) with new air cleaners and limited to 20 in H_2O (five K Pa) with dirty air cleaners. Where the total intake restriction is determined by the summation of the individual engine components and the rain shield, air cleaner, ducting and bending restrictions, typically the maximum exhaust pressure at rated speed and load set by the individual OEM is 3 in Hg (10.1 K Pa). This requirement is a fine balance between noise control using high backpressure mufflers, the effect backpressure has on the turbo charger performance, and the resulting engine efficiency.

Although naturally aspirated engine has the advantage of simplicity of air system design, their specific power output is limited by the quantity of air available for combustion. Increased power is typically accomplished by increasing the charge air density, with a pressure boost device. Although turbo charging is the primary mean used to increase charge air density, it has the undesirable side effects of raising the charge air temperature. This temperature rise is offset, by installing a heat exchanger between the turbo charger and the intake manifold [20].

3.4 Turbo charging system

Diana Fanakiev (1995) reiterated that successful design of turbo charged diesel engine is highly dependent on the choice of system for delivering exhaust gas energy from the exhaust valves or ports, to the turbine. Virtually all the energy of the gas leaving the cylinder arrives at the turbine. Some is lost on the way, due to heat transfer to the surrounding, but this is unlikely to exceed 5% unless water cooled exhaust manifolds are used, and will usually be much less. However, the design of the exhaust manifolds between the exhaust valve and turbine influences the proportion of exhaust gas energy that is available to do useful work in the exhaust system, [10].

3.5 Turbo charged diesel engine models

A normally aspirated gasoline engine model cannot be modified to describe the turbo charged diesel engine. Moreover, drawing a comparison between the SI and CI engines, one can find similarities as well as difference, because as mentioned earlier, there are common principles valid for all internal combustion engines. Several models have been studies, covering the range from quasi steady to filling and emptying methods, and including intermediate levels of complexity.

The model developed by Winter Bone (1977) is a wholly dynamic one, which represents the engine and turbo charger (TC) gas flows by a set of 30 interconnected first order nonlinear differential equations. It employs the filling-and-emptying technique and, being based mostly on physical principles; it gives an accurate description of the engine processes.

Hendricks uses the mean value method to create a series of models for different types of diesel engine in (Hendricks 1986, 1989, Jensen et 1991) [21]. The mean values models are similar to the quasi-steady ones with respect to aspect of simplicity.

However, an important aspect of the mean value method is the use of physically based models. In Hendricks's opinion, the fact that a given system is complex does not necessarily imply that there are not underlying physical principles which can give a simple overall picture of engine operation. The resulting models predict correctly the steady state operating points as well as the most important aspects of dynamic engine response. The model built by Jennies and Blumberg (1980) stands at level between the quasi-steady and the filling-and emptying.

A linearized model by Krutov is discussed by (Kullkamiet et al. 1992). Flower and Gupta (1974), give a state-space from of a simple discrete-time representation of the (TC) diesel engine. Recently, Kao and Moskwa (1993) have tried to summarize the modeling efforts at different level of complexity presenting two models: mean torque production and cylinder by cylinder. The first one uses the quasi-study approach and the latter employs the filling-and-emptying technique. Despite their differences, all models describe the turbo charger diesel engine with charge cooling, which is schematically presented on Figure 14, [10] and [22].

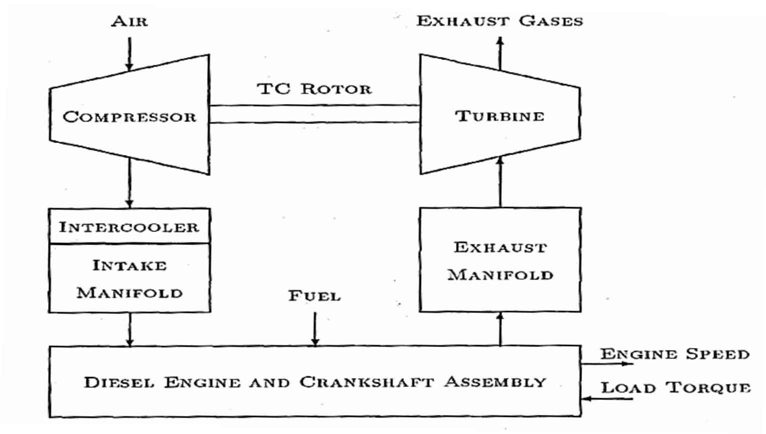

Figure 14. Schematic diagram of turbocharger diesel engine

3.6 Methods of power boosting

Vionov A.N (1965) discusses the maximum power in a given engine. It is limited by the amount of fuel that can be burned efficiently inside the engine cylinder. This is limited by the amount of air that is introduced into each cycle. If the inducted air is compressed to higher density than ambient, prior to entry into the cylinder, the maximum an engine of fixed dimensions can deliver will be increased. This is the primary purpose of supercharging.

The term supercharging refers to increasing the air (or mixture) density by increasing its pressure prior to entering the engine cylinder. Three basic methods are used to accomplish this. The first is mechanical supercharging where a separate pump, blower, or compressor, usually driven by power taken from the engine provides the compressed air. The second method is turbo charging, where a turbocharger, a compressor and a turbine on a single shaft is used to boost the inlet air (or mixture) density. Energy available in the engine's exhaust stream is used to drive the turbo charger turbine, which drives the turbo charger compressor, which raises the inlet fluid density prior to entry to each engine cylinder. The third method is pressure wave supercharging which uses wave action in the intake and exhaust systems to compress the intake mixture. The use of intake and exhaust manifold tuning to increase volumetric efficiency is one example of this method of increasing air density. An example of pressure wave supercharging device is the compress, which uses the pressure available in the exhaust gas stream to compress the inlet mixture stream by direct contact of the fluids in narrow flow channels. The most common arrangement uses a mechanical supercharger or turbo charger. Combinations of an engine driven compressor and a turbo charger are used in large marine engines. Two stages turbo charging is one viable approach for providing very high boost pressure (4 to 7 atmosphere) to obtain higher engine brake mean effective pressures. Turbo compounding, use of a second turbine in the exhaust directly geared to the engine drive shaft is an alternative method of increasing engine power (and efficiency). Charge cooling with a heat exchanger (often called an after cooler or intercooler) after compression, prior to entry to the cylinder, can be used to increase further the air or mixture density [23].

Supercharging is used in form of four stroke cycle engines to boost the power per unit-displaced volume. Some form of supercharging is necessary in two stroke cycle engines to raise the fresh air (or mixture) pressure above the exhaust pressure so that the cylinder can be scavenged effectively. With additional boost in two-stroke cycle engines, the power density is also raised [23].

3.7 Supercharging of diesel engine

Popik K.G. (1965) in Diesel engines supercharging introduces no fuel or combustion difficulties. Actually, the higher compression temperatures and pressures resulting from supercharging tend to reduce the ignition-delay period, and thus to improve the combustion characteristics with a given fuel, or to allow the use of fuel of poorer ignition quality [24]. Thus, the limits on specific output of supercharged Diesel engine are set chiefly by considerations of reliability and durability.

As the specific output increases, due to supercharging, cylinder pressure and heat flow must increase accordingly, and therefore both mechanical and thermal stresses increase unless suitable design changes can be made to control them. However, the possibility of design change is limited, and generally, higher stress must be accepted as an unavoidable accompaniment of an increase in specific output by supercharging. Whether or not such stress increase leads to factory reliability and durability, which depends on the quality of the design, the type of service, and the specific output, expected.

Another consideration that makes supercharging of diesel engines attractive is that, for a given specific output, it is possible to use lower fuel air ratios as the engine designed for un-supercharged operation. It is often found that a modest increase in power, and improvement in reliability and durability, can be obtained by supercharging. This comes about when the increase in airflow achieved by supercharging is accompanied by some whaler smaller increase in fuel flow, thus leading to lower combustion expansion, and exhaust temperatures and reduced smoke and deposits. For these reasons, many diesel engines designed for un-supercharging operation are now provided with superchargers [24].

3.8 Diesel-engine compression ratio

Charles Fayette Taylor (1984) in diesel engines changes in compression ratio have a limited effect on efficiency. In order to avoid excessive peak pressure, there is a tendency to use the lowest compression ratio consistent with easy starting [25]. Engines one fined within heated rooms such as large marine and stationary power plants, have a great advantage here, as is also the case when supercharges are separately driven and the temperature rise thus provided is available for starting [25].

3.9 Performance of supercharged diesel engines

In view of gains in specific output made possible by supercharging with no penalty in fuel economy Popik K. G. (1965), many commercial diesel engines are now supercharged. Most supercharged diesels are designed for un-supercharged operation and are later converted to supercharged operation. Most two-stroke engines use a turbine driven compressor in series with their normal scavenging pump [24].

Gains in rated output up to 50% have been attained in this way with only minor modifications of the engine. These gains are often accompanied by modest improvements in fuel economy due to the use of leaner mixtures [24].

3.10 Compressor and turbine

Horlak J.H. and WINTERBONE (1986) standard study-state performance maps, which relate the mass flow rate and efficiency of the compressor to its pressure ratio, inlet temperature and the TC rotor speed are used to model this subsystem. The disadvantage of this representation is its complete reliance on empirical data provided by the compressors manufacturer. Unfortunately, it seems to be the only possible approach, since all studied models used it [26].

3.11 Intercooler

The process of compression raises temperature as well as pressure. Since the objective is to increase inlet air density, intercoolers are often used to cool the air between the compressor delivery and the cylinders, so that the pressure increase is achieved with the maximum rise in density. The more involved intercoolers models (Jennings et al. 1986 Winter bone et al.1977) employ energy balance on the gas and coolant flows through the intercooler to determine their respective outlet temperatures. We adopted the simple approach suggested by (Kao and Moskwa 1993) to determine the outlet temperature using the cooling efficiency, [10].

3.12 Intake manifold

Different approaches are used to determine the intake manifold (IM) pressure. The filling- and- emptying method used by (Jennings et al. 1986, Winter bone et al. 1977) provides detailed consideration of all factors associated with the IM. Assuming that the heat transfer is negligible, one can achieve certain simplification as in (Hendricks, 1986, 1989, Jensen et al. 1991, Kao and Maskwa 1993). In addition to that the IM can be viewed as no volume component (i.e., no temperature charge occurs) which leads to an even simpler model. The air mass flow into the cylinder depends on the engine speed and on the volumetric efficiency. Jensen et al. (1991) have stated the fact, that presence of a turbocharger, the volumetric efficiency is a function only of the engine speed as opposed to the case of a normally aspirated engine, where it also depends on the intake manifold pressure [27].

3.13 Combustion and torque production

The major difference in computational complexity of the models is due to their different representation of the torque production process. The combustion can be described based on physical principle as in (Winter bone et al. 1977). The advantages of this approach are its capability to simulate the instantaneous fluctuations of the engine speed. However, they have no effect on the vehicle dynamics; therefore, applying this method will only increase the computational burden without contributing to the accuracy of our model [28].

The other commonly used technique for computation of the indicated torque is based on using steady state data. The engine speed and the air/ fuel mixture are the factors that affect the combustion process. Empirical characteristics, specific to the engine, are used to determine the thermal efficiency and respectively the produced torque, [10].

3.14 Exhaust manifold

The important of this subsystem comes from the presence of the turbocharger. The pressure and temperature in the exhaust manifold are input conditions for the turbine. As in the intake manifold model, several levels of detail are possible. The temperature can be determined either from complete thermodynamic analysis of the combustion process, or using empirical data for the temperature rise through the engine. The pressure calculation depends on the turbocharging method. There are two different ways in which the energy of the exhaust gases can be utilized to drive the turbine with the constant pressure turbo charging, the exhaust port from all cylinders is connected to a single exhaust manifold whose volume sufficiently must be large to ensure that its pressure is virtually constant. The unsteady exhaust flow processes at the cylinder are damped into a steady flow at the turbine.

The objective of pulse turbocharging is to make the maximum use of the high pressure and temperature, which exist in the cylinder when the exhaust valve opens even at the expense of creating highly unsteady flow through the turbine. In most cases, the benefit from increasing the available energy will be more than offset the loss in turbine efficiency due to unsteady flow.

Pulse turbo charging is usually applied to automotive and truck engines. Employing the pressure pulse to drive the turbine implies that an average exhaust manifold pressure model would be inaccurate. Ledger et al (1971) suggest how to avoid the involved thermodynamic calculations and still account for the pulse effect. In addition to the turbine efficiency, they introduce a corrective parameter called apparent efficiency, which reflects the pulsating pressure effect on the turbine, [10], [29], and [30].

3.15 Effect of length of IM runners on the volumetric efficiency

The design of an intake manifold can be accomplished in different ways. Due to advancement of computers and CFD software, using 3-D simulation of the flow within intake manifold is growing fast these days. With using this method, we can predict, observe and analyze the flow within an intake manifold and evaluate how the IM works under steady and unsteady situations.

In 1996, the intake manifold of a 4-cylinder engine was simulated using this method in VW Company. The flow had been simulated in both steady and unsteady states and the results were analyzed to improve the intake manifold performance. In 2001, an intake manifold of a direct injection diesel engine had been studied using 3-D simulation in GM Company to predict and improve cylinder to cylinder to EGR distribution.

In 2003, intake manifold of MRFI was simulated at one speed of engine. In 2004, in Ford Motor company, a model approximation method was used to predict the detailed wave dynamic characteristics and to determine the effects on the volume trice efficiencies. Multi-dimensional computational fluid dynamics simulations were carried out on the intake manifold and cylinder of a four-stroke single cylinder two-wheeler engine in 2005.

Three hypothetical models have been made that all of their runner's length is increased to 110, 120 and 130% of initial value. No sensible change in volumetric efficiency observed in the cases with 110 and 130%-increased length at any speed of engine. For 20% extended runners, the volumetric efficiency increases at 3500 and 4500 rpm. This phenomenon can be explained by tuning. To achieve a favorable volumetric efficiency in a wide range of engine speed, it is suggested to increase the length of IM runners to 120% of initial value [31].

4 CONCLUSIONS

This review research describes intake and exhaust system and how the air and fuel are delivered into the cylinder. The object of the intake system is to deliver the proper amount of air and fuel accurately and equally to all cylinders at the proper time in the deck's engine cycle. After combustion is completed in the exhaust flow a high-pressure gas have been used to transfer work to the crankshaft during the expansion stroke.

The deck's engine designer should define the maximum intake airflow, the maximum exhaust temperature and the maximum exhaust volume.

The first component in the air path is the air filter, which cleans the air from pollutions. Over the air filter, there will be a small pressure drop.

The turbo charging is primarily means increasing charge air density; and it has undesirable side effects or raising the air temperature. The process of compression raises temperature as well as pressure. Since, the objective is to increase inlet air density; intercoolers are often used to cool the air between the compressor delivery and the cylinder. The inlet manifold is the volume placed just before the inlet port. It only acts as a volume and no pressure drop occurs over it. Its function is to divide the airflow from the intercooler to the different inlet ports leading to each combustion chamber.

The Exhaust Gas Recirculation (EGR) system involves pipes, EGR coolers and a variable valve used to control the amount of EGR flow. When the airflow through the inlet port, heat was exchanging between the metal and the gases this will rise temperature on the gas when it flows into the combustion chamber.

The inlet valve controls the gas flow from the inlet manifold into the combustion chamber. The valve is controlled via the camshaft.

In the combustion chamber, the temperature will be very high and the cylinder wall will be much hotter than the gas during the intake. The exhaust valve is similar to the inlet valve.

After the exhaust port there is an exhaust manifold. Its function is opposite to the inlet manifold, which brings the flow from each exhaust port into one pipe. The exhaust manifold is modeled as a volume with no pressure drop. The pressure that is important in the study of volumetric efficiency is the one just before the exhaust valve closes. Another measurement related problem occurs when the exhaust valve opens.

Inside the combustion chamber, gas at very high pressure is confined. When the exhaust valve opens, the gas expands quickly out from the combustion chamber and causes a pressure peak in the exhaust manifold.

The turbine is the second part of the turbo. Its purpose is to utilize some of the energy wasted with the exhaust gas and to transfer it into torque, used to drive the compressor. Between the atmosphere and the turbine there is a pipe leading the exhaust gas and a muffler that should reduce the noise. Over the muffler, there is a pressure drop.

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