Advantages of internal combustion engines engineering essay

The most widely used heat engine is the internal combustion engine. The advantages that it has over gas turbines have seen its widespread usage in passenger car applications. [1]

All the components of internal combustion engines work at an average temperature which is below the maximum temperature of the working fluid in the working cycle.

This is because the high temperature of the working fluid in the cycle persists only for a very small fraction of the cycle time.

As a result, fluids with high working temperatures can be used to increase thermal efficiency at moderate maximum working pressures.

Weight to power ratio is less than that of steam turbine and gas turbines.

It is therefore possible to develop reciprocating IC engines of very small power output with reasonable thermal efficiency and cost.

Higher brake thermal efficiency can be obtained as only a small fraction of heat energy of the fuel is dissipated to the cooling system.

Initial cost is low.

Materials used in the manufacture of gas turbines must be strong and heat resistant in order to sustain the heat generated. Machining operations required for gas turbines construction are also more complex.

Reciprocating IC engines are more efficient at idle speeds than gas turbines in terms of fuel consumption at idling.

Gas turbines have delayed responses to different power requirements changes.

Gas turbines must be removed for overhaul and servicing, which is usually not the case in internal combustion engines.

Gas turbines require more air than IC engines for its normal operation. It also consumes more fuel whenever the load fluctuates, which is common in the domestic usage.

All these explain why passenger cars do not use gas turbine engines, but use internal combustion engines instead.

Question 2

Define the following parameters and give typical values for spark-ignition and compression ignition IC engines:

Specific fuel consumption,

Specific fuel consumption (SFC) is the fuel flow rate per unit power output []. It measures how efficiency of an engine in using the fuel to produce useful work.

The equation for the specific fuel consumption is:

Where:

Ke= specific fuel consumption

K: Fuel Consumption, kg/s

Pe= Useful work per cycle, i = 0. 5 for 4Î, 1 for 2Î

ne= real efficiency

HÎ± = Heat of Combustion = 42. 000 KJ/Kg

Low values of SFC are obviously desirable.

For SI engines typical values of brake specific fuel consumption are about 270 g/kWh. Range (345 – 285 g/kWh)

For CI engines, values are lower and in large engines can go below 200 g/kWh.

Range (285 – 190 g/kWh) [2]

Mean effective pressure,

Relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the mean effective pressure (mep).

Where:

W= Indicated Work:

Vh= Piston Displacement (cylinder) Volume (cc, cm3, lt)

H= Length TDC – Length BDC

For,

Naturally aspirated spark ignition engines, maximum values are in the range 850 to 1050 kPa at the engine speed where maximum torque is obtained (about 3000 rev/min).

Turbocharged automotive spark-ignition engines the maximum bmep is in the 1250 to 1700 kPa range.

Naturally aspirated four-stroke diesels, the maximum bmep is in the 700 to 900 kPa range

Turbocharged four-stroke diesel maximum bmep values are typically in the range 1000 to 1200 kPa

Turbocharged aftercooled engines this can rise to 1400 kPa

Two-stroke cycle diesels have comparable performance to four-stroke cycle engines. Large low-speed two-stroke cycle engines can achieve bmep values of about 1600 kPa.

[2]

Power-torque relation as function of engine rpm,

Engine torque is measured using a dynamometer. The engine is clamped and the output shaft is connected to the dynamometer rotor. The rotor is coupled electromagnetically, hydraulically, or by mechanical friction to a stator, which is supported in low friction bearings. The stator is balanced keeping the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs, or pneumatic means.

Fig. 1 Brake dynamometer- engine torque test [2]

Torque is a measure of an engine’s ability to do work; and power is the rate at which work is done. The value of engine power measured as described above is called brake power Pb. This power is the usable power delivered by the engine to the load-in this case, a “ brake”.

Fig. 2 Engine power, torque vs. speed plot [3]

Correlation between measured force and engine torque:

Measured power: (1 PS = 0. 736 kW)

Conversion between different units may be necessary for power, torque, or angular speed. For example, if rotational speed (revolutions per time) is used in place of angular speed (radians per time), a factor of 2Ï€ radians per revolution have to be multiplied.

Dividing on the left by 60 seconds per minute and by 1000 watts per kilowatt gives us the following.

mbox{power (kW)} = frac{ mbox{torque (N}cdotmbox{m)} times 2 pi times mbox{rotational speed (rpm)}} {60, 000}

Volumetric efficiency

Volumetric efficiency is the ratio of the mass inside the engine cylinder to the mass of air of the displacement volume at atmospheric conditions. It measures the effectiveness of an engine’s induction process. Volumetric efficiency is used for four-stroke cycle engines which have a distinct induction process and not for two stroke engines.

Where pai is the inlet air density

Alternatively volumetric efficiency can also be defined as,

Indicative values: 4Î-Otto: 0. 7 – 0. 9

2Î-Otto: 0. 5 – 0. 7

Typical maximum values of Î·v for naturally aspirated engines are in the range 80 to 90 percent. The volumetric efficiency for diesels is somewhat higher than for SI engines.

[2]

Engine real efficiency as function of engine power, fuel consumption and fuel calorific value

The real engine efficiency of an engine can be found out using the relation

Where,

ne= real efficiency

nth= theoretical thermodynamic efficiency

ng= quality coefficient (0. 4-0. 7 Otto; 0. 6-0. 8 Diesel)

nm= mechanical efficiency (0. 8)

ni= actual efficiency (nth. ng= Pi/Q)

K= fuel consumption Kg/s

HÎ± = Heat of Combustion = 42. 000 KJ/Kg

Question 3

Describe with simple terms the main air flow path developing inside the cylinder of IC engines relative to the piston motion; make a simple schematic to indicate them.

Laser Doppler Velocimetry (LDV) helps us to visualise the charge motion within the cylinder with the help of optically transparent research engines. Computational Fluid Dynamics (CFD) can help in validating the average flow field in the cylinder but the process is expensive. One such CFD software is KIVA-4v, which helps to predict the air charge motion.

Swirl flow

Swirl is defined as the micro mass rotational motion of charge within the cylinder. It is generated by constructing the intake system to give a tangential component to the intake flow as it enters the cylinder. This is done by shaping and contouring the intake manifold, valve ports, and sometimes even the piston face. Swirl enhances the mixing of air and fuel to give a homogeneous mixture in a short time in modern high-speed engines. It is also responsible for very rapid spreading of flame front during the combustion.

Fig. 3 Swirl flow in the engine cylinder [3]

Swirl flow can be generated by changing the geometry of the inlet port

Fig. 4 Geometry of inlet port affecting swirl flow [3]

(a) Deflector wall (b) directed (c) shallow ramp helical (d) steep ramp helical

Similarly inlet valve approach geometry can also generate swirl flow by producing net in-cylinder angular momentum of the charge.

Fig. 5 Inlet valve geometry affecting swirl flow [2]

Squish flow

When the piston approaches TDC at the end of the compression stroke, the volume around the outer edges of the combustion chamber reduces drastically. New combustion chamber designs have the clearance volume near the centerline of the cylinder. As the piston approaches TDC, the gas mixture occupying the volume at the outer radius of the cylinder is forced radially inward as this outer volume is reduced to near zero. This radial inward motion of the gas mixture is called squish. It adds to other mass motions within the cylinder to mix the air and fuel, and quickly spreads the flame front. Maximum squish velocity usually occurs at about 10°bTDC.

During combustion, the expansion stroke begins and the volume of the combustion chamber increases. As the piston moves away from TDC, the burning gases are propelled radially outward to fill the now-increasing outer volume along the cylinder walls. This reverse squish helps to spread the flame front during the latter part of combustion

Piston motion influences squish as in the case of wedge shaped and bowl-in combustion chambers.

Fig. 6 Piston motion generating squish [2]

(a) Wedge shaped SI combustion chamber (b) bowl-in-piston DI Diesel combustion chamber

Tumble

As the piston nears TDC, squish motion generates a secondary rotational flow called tumble. This rotation occurs about a circumferential axis near the outer edge of the piston bowl

Fig. 7Tumble- result of piston motion and squish [3]

Turbulence

Due to the high velocities involved, all flows into, out of, and within engine cylinders are turbulent flows. The exception to this is those flows in the corners and small crevices of the combustion chamber where the close proximity of the walls dampens out turbulence. As a result of turbulence, thermodynamic transfer rates within an engine are increased by an order of magnitude. Heat transfer, evaporation, mixing, and combustion rates all increase. As engine speed increases, flow rates increase, with a corresponding increase in swirl, squish, and turbulence. This increases the real-time rate of fuel evaporation, mixing of the fuel vapor and air, and combustion.

Intake turbulent mixture flow Turbulence superimposed on mixture swirl

Fig. 8 Turbulence of the charge within cylinder [4]

Question 4

The Figure below shows a conceptual model of a quasi-steady Diesel combustion plume, as presented by Dec et al in 1997. Indicate the following areas shown on this schematic:

liquid fuel ,

rich vapour fuel-air mixture ,

fuel-rich premixed flame,

initial soot formation ,

diffusion flame boundary ,

thermal NO production zone ,

soot oxidation zone ,

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Fig. 9 Quasi-steady Diesel combustion plume [5]

The above figure describes the formation and features of a quasi-steady diesel fuel jet. This model is applicable to large bore, quiescent chamber combustion or a free fuel jet without wall interactions. At the point of fuel injection, fuel penetrates into the combustion chamber and air which is at a high temperature due to end of compression stroke begins to mix with the spray. Fuel absorbs energy from the hot air and evaporates. This process continues until a point where no liquid fuel is present. The point at which this occurs is called the liquid length. This liquid length reduces after the start of combustion but thereafter remains constant until the end of injection. Beyond the liquid length, the rich premixed fuel and air are still heated by the surroundings until they start to react in the rich premixed zone. The products of rich combustion continue downstream and diffuse and mix radially outward until reaching the surrounding cylinder gases. At a location where the rich products and cylinder gases mix to produce a stoichiometric mixture, a diffusion flame is produced. The diffusion flame surrounds the jet in a thin turbulent sheet, which extends upstream towards the nozzle. The axial distance from the nozzle exit to the diffusion flame is the lift-off length. The lift-off length controls the amount of oxygen mixed into the fuel jet and therefore the stoichiometry. Soot is burned out and NOX is produced on the outside of the diffusion flame, where temperatures are high and oxygen and nitrogen are abundant.

Question 5

What are the main requirements of the fuel injection system for a direct injection engine?

In recent years, significant progress has been made in the development of advanced computer-controlled fuel injection systems, which has had much to do with the research and development activities related to Direct Injected engines being expanded.

[6]The main requirements of the fuel injection system for a direct injection engine are:

Well atomised fuel spray independent of chamber pressure conditions

Injection during the compression stroke against pressures up to 20bar

Injection during the intake stroke against atmospheric pressures with stoichiometric homogeneous mixture

To have uniform distribution of fuel in a multi cylinder engine

To improve breathing capacity of an engine i. e. volumetric efficiency

To reduce or eliminate detonation

To prevent fuel loss in the form of scavenging in the case of two stroke engines.

For an efficient combustion of a stratified mixture, a stable and compact spray geometry is necessary

Injection pressure has been determined to be very important for obtaining both effective spray atomization and the required level of spray penetration.

Accurate fuel metering (generally a +2% band over the linear flow range);

Desirable fuel mass distribution pattern for the application;

Minimal spray skew for both sac and main sprays;

Good spray axisymmetry over the operating range;

Minimal drippage and zero fuel leakage, particularly for cold operation;

Small sac volume;

Good low-end linearity between the dynamic flow and the fuel pulse width;

Small pulse-to-pulse variation in fuel quantity and spray characteristics;

Minimal variation in the above parameters from unit to unit.

Question 6

Describe the injection process requirements for direct injection Diesel engines and the evolution of the fuel injection equipment over the last few decades.

The functional requirements of the fuel injection system are as follows

Accurate fuel metering per engine working cycle

Injection timing to ensure maximum power, good fuel economy and low emissions

Obtain the desirable heat release pattern by control of injection rate

Atomisation of the fuel

Proper spray pattern to ensure better mixing of fuel and air

Uniform distribution of fuel droplets in the combustion chamber

Supply equal quantities of fuel to all cylinders, in the case of multi cylinder engines

Eliminate dripping of fuel droplets into the combustion chamber by eliminating injection lag between start and end of injection

Evolution of fuel injection equipment:

In-line pump

Fig. 10 Layout of In-line fuel injection pump [7]

Though in-line pumps are primitive injection systems, they are still in use among heavy duty marine engines.

Individual fuel pumps fuel each of the injectors

Engine operational speed has a major influence on the fuel injection pressures

As a result, there is a hydraulic delay between the pressure increase and the start of injection

Fuel flows through high pressure connecting pipes

Fuel injection pressures range from 600 – 1200 bar

Injector with discharging in the combustion chamber (the nozzle with one or more holes)

Distributor type pumps

These are still used in a number of engines

Though it started as mechanically operated, now electronic control modifications have been made

It has a mechanism which controls the spill valve responsible for cutting off the high pressure generated inside the pumping chamber, and thus, responsible for the termination of injection

One pumping chamber delivers high pressure to all the injectors of the engine

Pressure depends on engine speed, so a hydraulic delay exists between the pressure generation and start of injection

Relatively low injection pressures (up to 1200bar)

Fig. 11 Distributor type pump (Lucas CAV) [7]

Unit injectors

Consists of the pump and the injector integrated into one body, which does not require a high pressure connection pipe

High fuel pressure is generated close to the nozzle exit, which can be upto 2500 bar.

These gave accurate control over injection

Each cylinder has its own individual system

High pressure developed depends of the engine rpm and the load.

Fig. 12 General Layout of Unit injector [7]

[6]Delphi Diesel Systems electronic unit injectors (EUI fig13.) control the quantity and the timing of injection electronically through a solenoid actuator. The solenoid can respond very quickly (injection periods are of the order 1 ms), to control very high injection pressures (up to 1600 bar or so). The solenoid controls a spill valve, which in turn controls the injection process. The pumping element is operated directly from a camshaft (or indirectly via a rocker), and the whole assembly is contained within the cylinder head.

Fig. 13 Electronic Unit injector (Lucas EUI system) [7]

[6]An alternative approach to the EUI is the Caterpillar Hydraulic Electronic Unit Injector (HEUI, also supplied by other manufacturers). HEUI uses a hydraulic pressure intensifier system with a 7: 1 pressure ratio to generate the injection pressures. The hydraulic pressure is generated by pumping engine lubricant to a controllable high pressure. Similar to CR injection systems, there is control of the injection pressure. The HEUI uses a two-stage valve to control the oil pressure, and this is able to control the rate at which the fuel pressure rises, thereby controlling the rate of injection, because a lower injection rate can help control NOx emissions.

Common Rail fuel injection systems

One of the last improvements to the fuel injection system is the “ Common Rail System” that was implemented first by the Fiat Company.

Fig. 14 Common rail fuel injection system [8]

Common rail (CR) fuel injection systems decouple the pressure generation from the injection process and have become popular because of the possibilities offered by electronic control.

The key elements of a CR fuel injection system are as follows:

A (controllable) high-pressure pump

The fuel rail with a pressure sensor

Electronically controlled injectors

An engine management system (EMS)

The injector is an electro-hydraulic device, in which a control valve determines whether or not the injector needle lifts from its seat. The engine management system can divide the injection process into four phases: two pilot injections, main injection, and post-injection (for supplying a controlled quantity of hydrocarbons as a reducing agent for NOx catalysts). Common rail injection also enables a high output to be achieved at a comparatively low engine speed

Fuel injectors

Fig. 15 Types of nozzles used in Diesel fuel injectors [1]