Experiential and Theoretical analysis of 4 stroke VCR engine and calculation of Waste heat recovery

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ABSTRACT

A complete experimental analysis of a four-stroke IC engine test rig has been done. A parametric study of the engine's performance by changing fuel injection pressure, the compression ratio has been done. Furthermore, Theoretical analysis on heat loss from the machine can be utilized either in running a thermal system or preheating the metallic component whose feasibility has to be determined. Theoretical analysis of mass loss, work loss, heat loss, and other performance parameters has been made, which gave a complete understanding of its theory. Also, the feasibility of heat which can be utilized in running an engineering device has been done in which analysis of simple absorption system has been done.

KEYWORDS

Diesel engine – Variable Compression ratio - Fuel injection pressure – load - Theoretical analysis – Heat Calculation – Feasibilities

1. Introduction

Variable compression ratio is a technology to vary the compression ratio of an internal combustion engine while the engine is working. This has been done to increase the performance while under varying loads. In diesel engines, variable compression ratio provides control of peak cylinder pressure, improves cold-start ability and low load operation, enabling the multi-fuel capability, increased fuel economy, and reduced emissions. So far, variable compression ratio engines have not reached the market, despite patents and experiments dating over decades. VCR technology can provide the key to enable exceptional efficiency at light loads without loss of full-load performance. The increasingly global problem regarding rapid

Economic development and a relative energy shortage, internal combustion engine exhaust waste heat, and environmental pollution have been more recently emphasized. Out of the total heat supplied to the engine in the form of fuel, approximately 30 to 40% is transferred into valuable mechanical work; the remaining heat is expelled to the environment through exhaust gases and engine cooling systems, resulting in entropy rise and severe environmental pollution, so it is required to utilized waste heat into valuable work. The recovery and utilization of waste heat conserve fuel (fossil fuel) and reduce the amount of waste heat and greenhouse gases damped to the environment. This paper will review VCR engine performance, waste heat calculations, and feasible solutions for waste heat recovery.

2. Literature Review

A better engine design can significantly increase the combustion quality and, in turn, will lead to better brake thermal efficiencies and hence savings in fuel(M.K.G. Babu et al., 2008). The potential of Diethyl ether (DEE) as extra oxygenated fuel in a compression ignition engine has been identified through an experimental investigation. In this study, the tests were conducted on a single-cylinder DI diesel engine fueled with neat diesel fuel and addition of 2, 5, and 10% DEE in diesel fuel to find out the optimal blend based on performance and emission characteristics India though rich in coal abundantly and endowed with renewable energy in the form of solar, wind, hydro and bio-energy has a minimal hydrocarbon reserve (0.4%of the world's reserve) [2].

India is a net importer of energy. Nearly 25% of its energy needs are met through imports, mainly in crude oil and natural gas (Kapilan N et al. 2008). The rising oil bill has been the focus of severe concerns due to its pressure on scarce foreign exchange resources and is primarily responsible for energy supply shortages. The sub-optimal consumption of commercial energy adversely affects the productive sectors, which in turn hampers economic growth. [4].

The present work deals with finding a better compression ratio for the Diesel fuelled C. I. engine at variable load and constant speed operation. The compression ratio of an internal combustion engine or external combustion engine is a value that represents the ratio of the volume of its combustion chamber from its largest capacity to its smallest capacity. It is a fundamental specification for many standard combustion engines.

Experimental results showed a slight increase in brake-specific fuel consumption, brake power, and brake thermal efficiency compared to diesel fuel. In addition, it was found that there is a decrease in smoke, oxides of nitrogen, unburned hydrocarbon, carbon monoxide, and ignition delay, along with an increase in carbon dioxide. (Ashok MP et al. 2007). The diesel engine's performance is increased with the addition of oxygenates to the fuel before the combustion. This paper presents the effect of blending Diethyl ether (DEE) with Diesel at various proportions (5%,7.5%, and 10%)on the performance of diesel engines. The experimental results indicated that DEE concentration to Diesel increases the thermal brake efficiency and mechanical efficiency and decreases the specific fuel consumption. The performance of diesel engines at different compression ratios (18, 16, and 14) for Diesel with a 5% DEE blend was also evaluated in this work. (Subramanian K.A. et.al 2002).The data obtained from the International Journal of Engineering Trends and Technology (IJETT) – Volume 28 Number 1 - October 2015 ISSN: 2231-5381 http://www.ijettjournal.org Page 7 experimentation is presented analyzed in this paper. To find out the Optimum Compression Ratio of the Computerized Variable Compression Ratio (VCR) Single Cylinder Four Stroke Diesel Engine using Experimentation analysis. Various parameters defining the performance of the VCR diesel engine are calculated. They are used to obtain the optimum compression ratio by plotting performance graphs of different loads and different compression ratios from that optimum compression ratio obtained.

3. Experimental Setup

The engine understudy is Diesel injection, a water-cooled one-cylinder diesel engine in which the fuel line pressure, compression ratio, and fuel injection pressure are variable. The machine is coupled with AG 10/ED 1 Eddy current dynamometer. The layout of the experimental setup is shown in fig 3.1. The main components of the system are given below. The engine, Fuel injection Pump, Dynamometer, Device for changing starting of Fuel, Supercharging system, Dynamic injection indicator, Data acquisition system, Smoke meter, Exhaust gas analyzer, Pressure transducer.

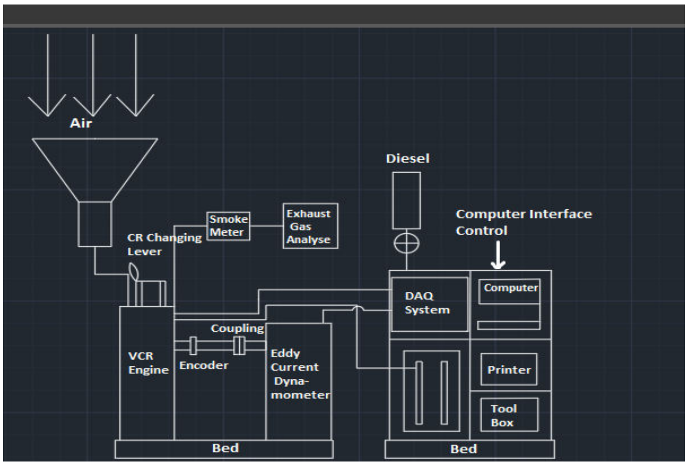


Figure 3.1: Line diagram of VCR Engine

Fig. 3.2Experimental setup of VCR Diesel Engine

Table 3.1 Specification of VCR Engine

No. of cylinders	**1**
No. of Strokes	4
Cylinder Diameter	87.5 mm
Stroke Length	110 mm
Connecting Rod Length	234 mm
Orifice Diameter	20 mm
Dynamometer Arm Length	185 mm

Fuel	**Diesel**
Power	3.5 kW
Speed	1500 RPM
CR Range	12:1 to 18:1
Injection Point Variation(Degree)	0 to 25 ˚BTDC

Fuel	**Petrol**
Power	4.5 kW
Speed Range	1200 to 1800 RPM
CR Range	6:1 to 10:1

THEORETICAL CALCULATIONS

PERFORMANCE

Experimental Procedure:-

Here, we are changing Fuel injection pressure, compression ratio, and load. The load can be varying by using nobe on the setup, which is called Dynamometer Loading Unit. It can vary between 0-20 kg. To change the compression ratio, it is necessary to change the volume of the cylinder. This can be done by loosening the nut on the top of the head. To avoid the leakage gasses outside the cylinder, a gasket is expandable that means if we increase the volume of the cylinder, the gasket will expand and vice versa. There is a long hexagonal nut provided; the Revolution of that nut changes the compression ratio. On the side of that, there are readings on the wire which indicate the compression ratio. Fuel injection pressure can be varying by loosening and tightening the nut. The Revolution of the nut anti-clockwise will increase the fuel injection pressure vice versa.

The test which we are performing is measuring the brake power, indicated power, specific fuel consumption along with thermal brake efficiency, showed thermal efficiency, mechanical efficiency.

Indicated power: $p_{im}LANxc$, Watt

Brake Power: $2\pi NT$ or $\frac{WN}{C}$; C=Constant, W=Load

Friction Power: $ip - bp$

Mechanical efficiency (η_m): $\frac{bp}{/ip}$

Indicated Thermal Efficiency (η_{ith}): $\frac{ip}{mfCV}$

Brake Thermal Efficiency (η_{bth}): $\frac{bp}{mfCV}$

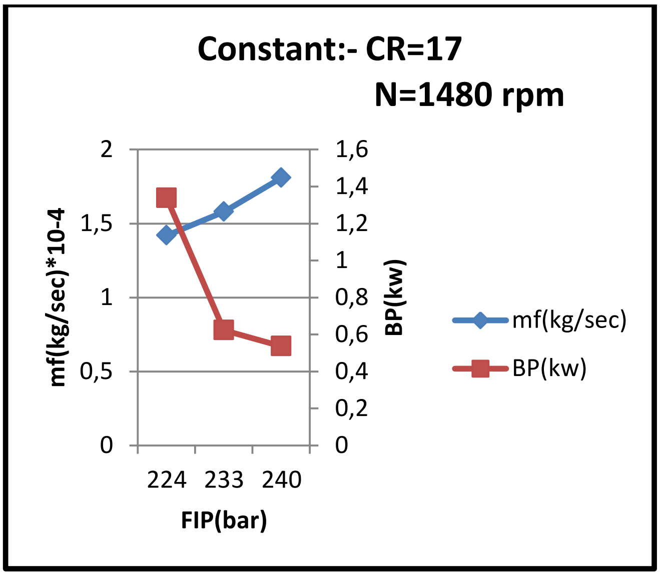
Indicated Specific Fuel Consumption isfc: $\frac{mf}{ip}, {kg}/{Ws}$

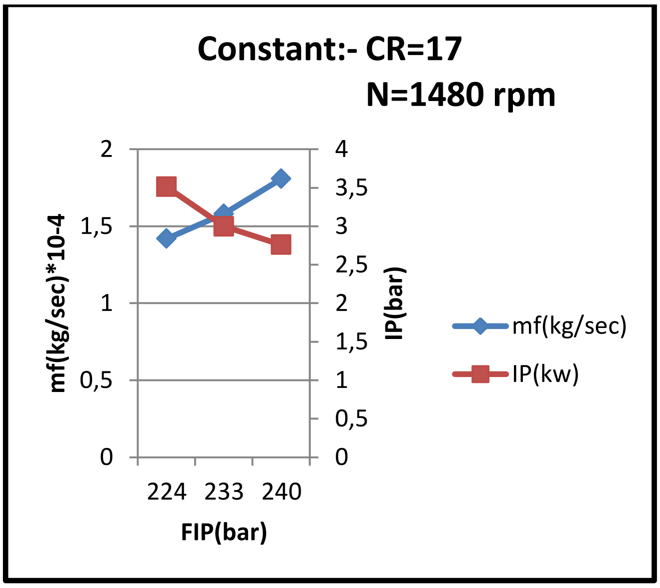
Brake Specific Fuel Consumption bsfc: $\frac{mf}{bp}, {kg}/{Ws}$

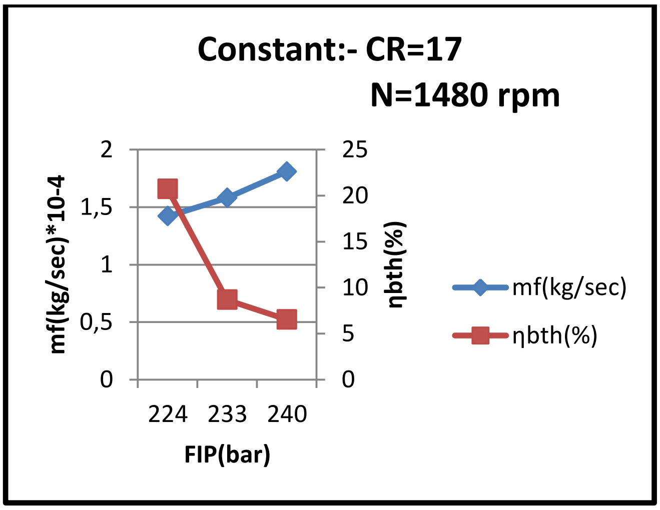
All these parameters were measured at the same compression ratio and inline fuel pressure but different loads. After that, inline fuel pressure was varied at the same compression ratio, and measurements were taken at other loads. Further, the compression ratio was varied, and measures were taken at different loads but the same fuel pressure. Based on additional readings, we get the conclusion.

Results and Discussion

1. At the constant CR=17 and N=1480 rpm, we got the value of fuel consumption(m_f), brake power, indicated power, friction power, mechanical efficiency, indicated specific fuel consumption (isfc), Brake specific fuel consumption (bsfc) along with thermal brake efficiency, indicated thermal efficiency.

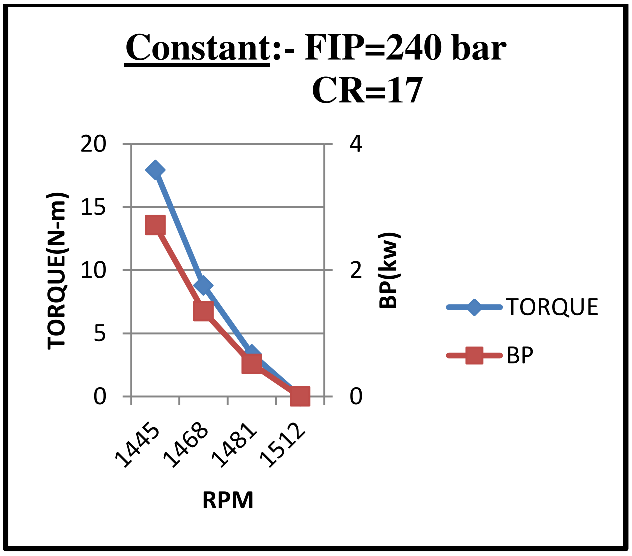


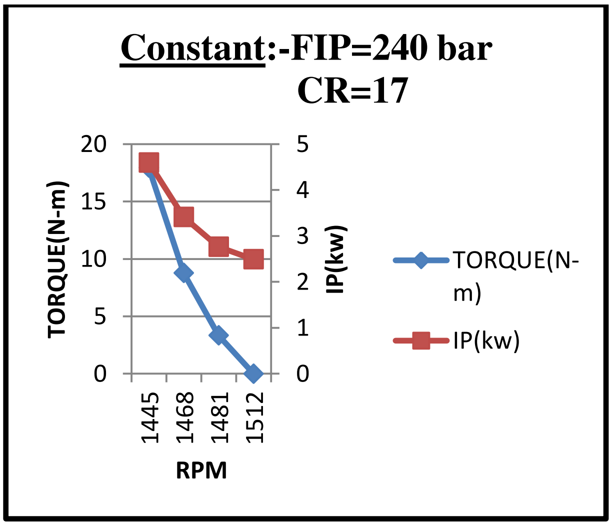


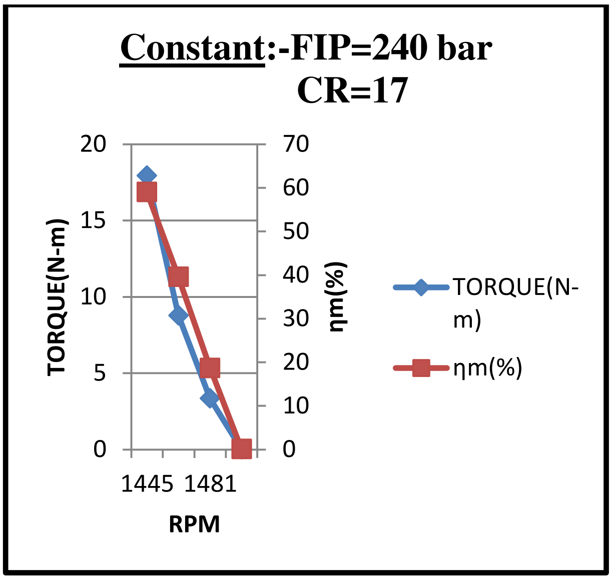


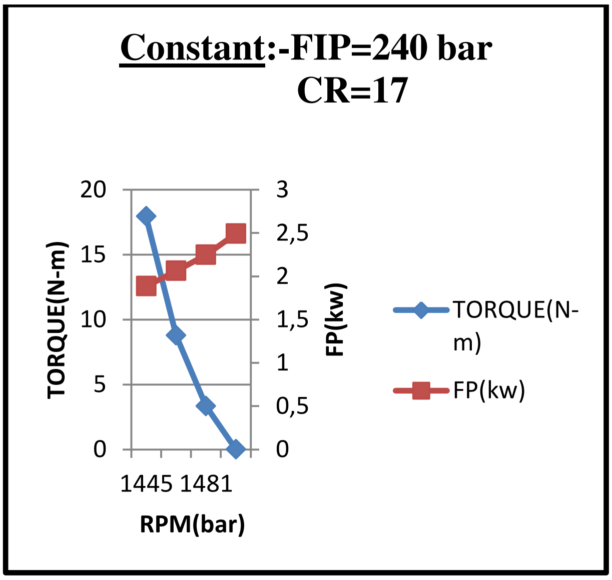
At constant CR = 17 and R.P.M=1480 rpm, they increase the value of fuel injection pressure from 224,233,240 bars, respectively. Values of indicated power, brake power, mechanical efficiency, and brake thermal efficiency **decrease**, brake specific fuel consumption **increases**. Values of indicated specific fuel consumption **increases** throughout. The value of indicated thermal efficiency **decreases** throughout.

1. We got the torque value at constant compression ratio=17 and FIP=240, indicated mean effective pressure, indicated power, brake power, friction power, and mechanical efficiency.



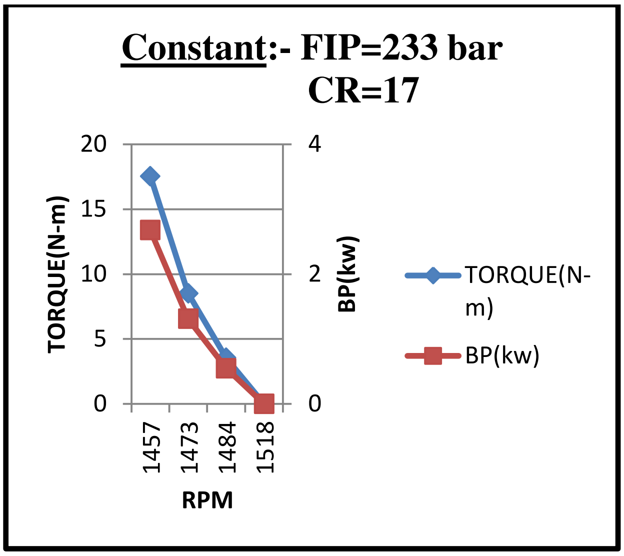


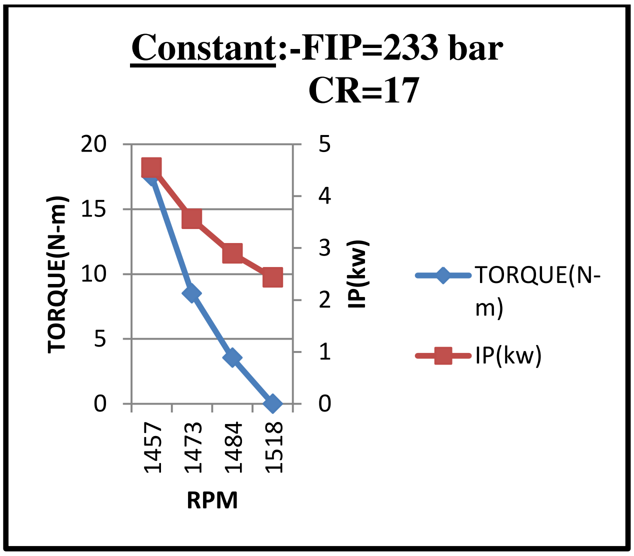


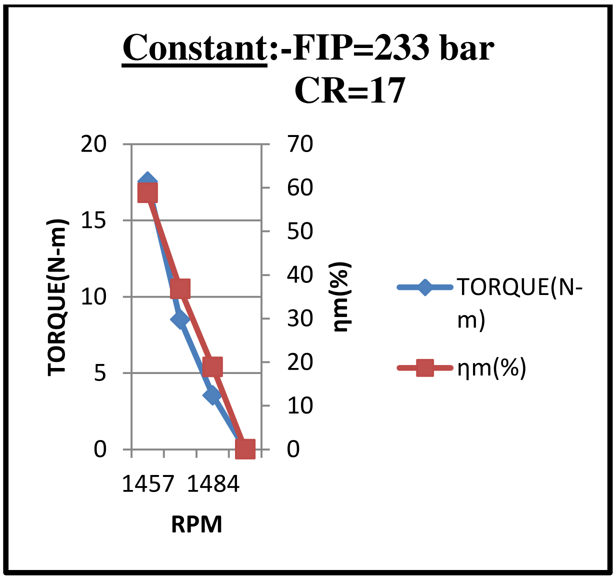


- At constant CR=17 and FIP=240 bar, increasing RPM value from 1445,1468,1481,1512 respectively, the values of indicated power, brake power, mechanical efficiency, torque indicated mean effective pressure **decreases** and Friction power **rises.**

2. We got the torque value at constant compression ratio=17 and FIP=233, indicating mean effective pressure, indicated power, brake power, friction power, and mechanical efficiency.







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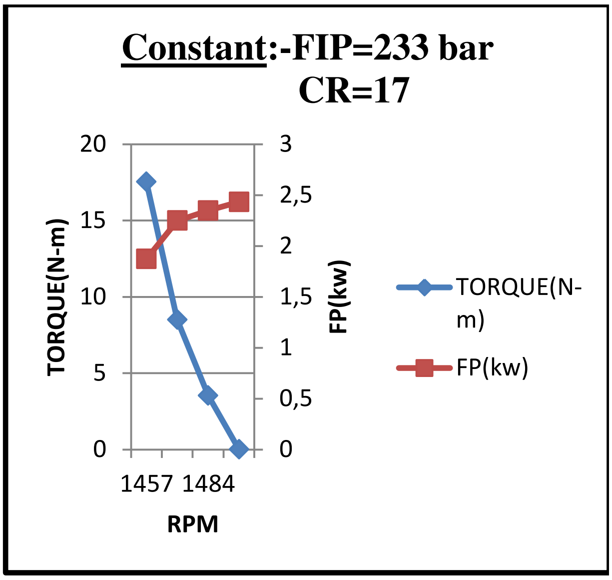
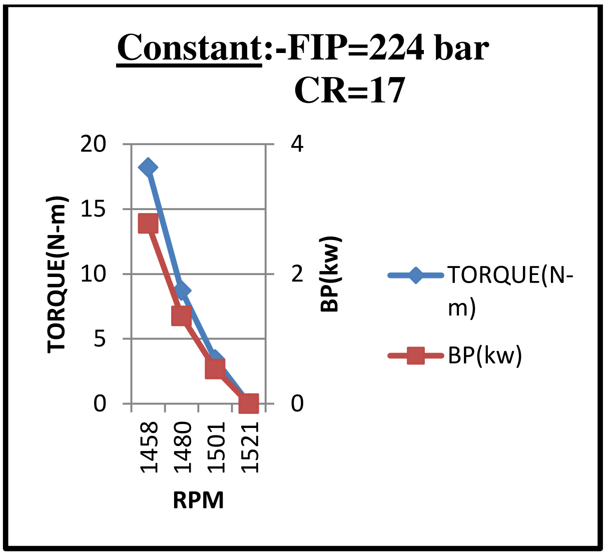
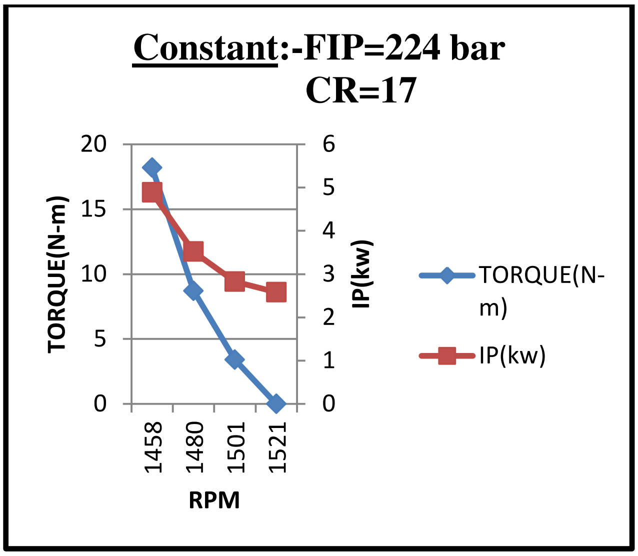
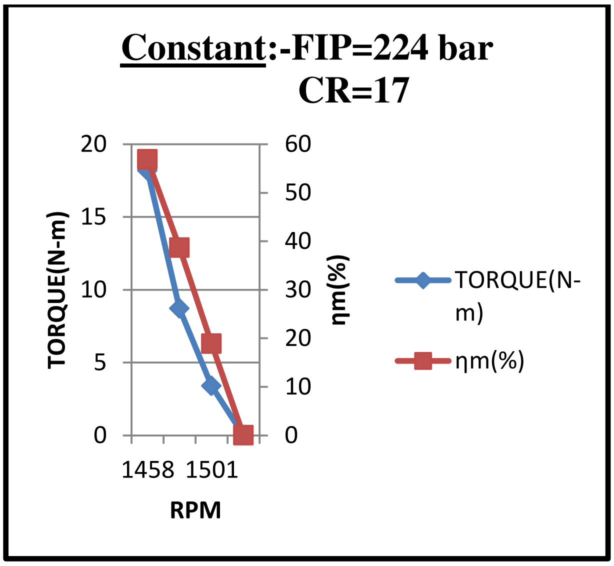


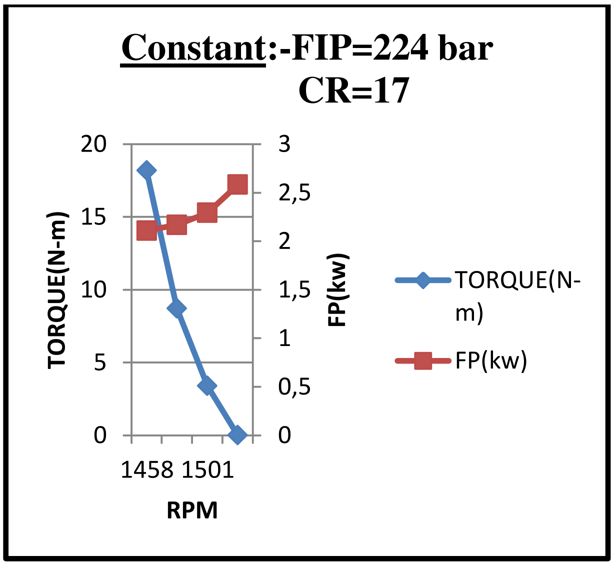
- At constant CR=17 and FIP=233 bar, increasing RPM value from 1457,1473,1484,1518 respectively, the indicated power, brake power, mechanical efficiency, torque indicated mean effective pressure **decreases** and Friction power **rises**.

3. We got the torque value at constant compression ratio=17 and FIP=224, indicating mean effective pressure, indicated power, brake power, friction power, and mechanical efficiency.



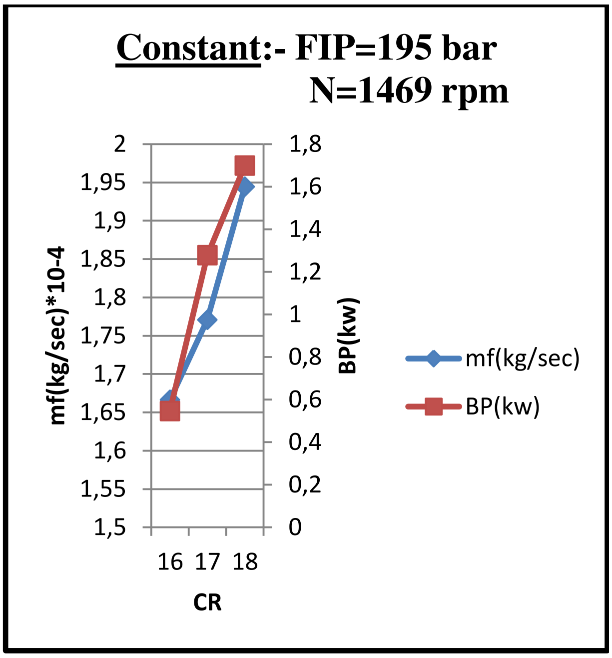


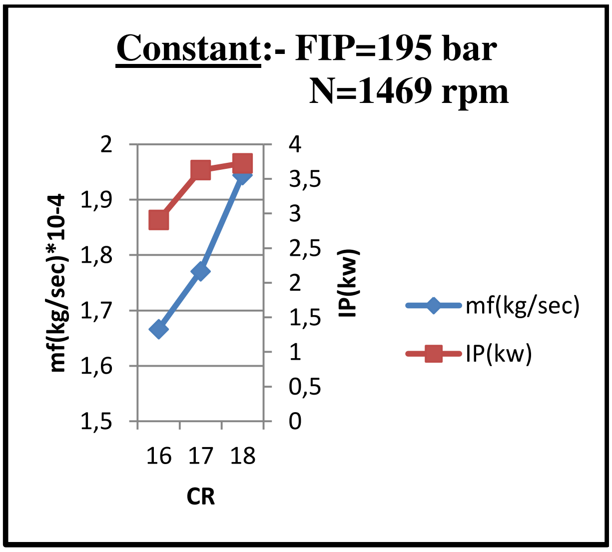


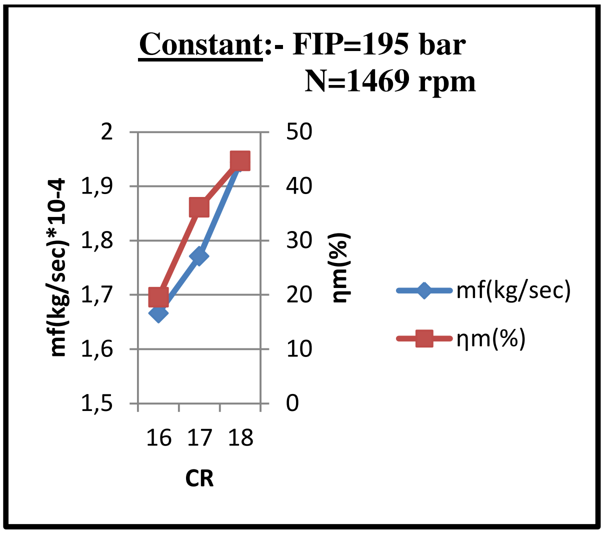


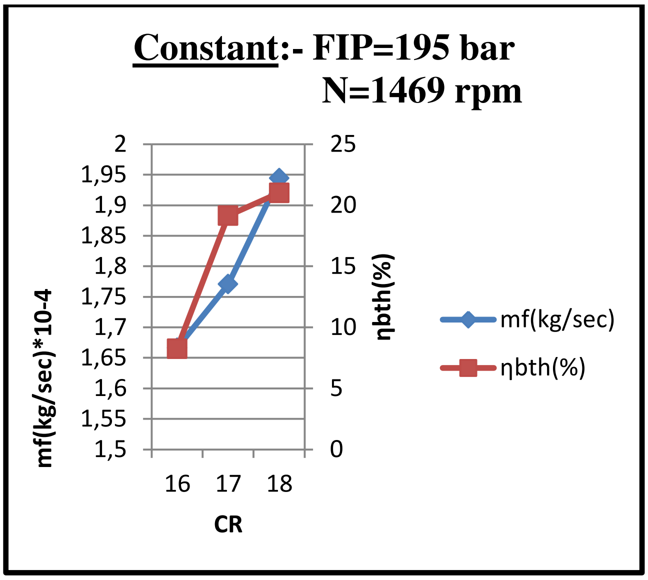
- At constant CR=17 and FIP=224 bar, increasing RPM value from 1458,1480,1501,1521 respectively, the indicated power, brake power, mechanical efficiency, torque indicated mean effective pressure **decreases** and Friction power **rises**.

4. At constant N=1469 rpm and FIP=195, we got the value of fuel consumption (m_f), brake power, indicated power, friction power, mechanical efficiency, indicated specific fuel consumption (isfc), Brake specific fuel consumption (bsfc) along with thermal brake efficiency, indicated thermal efficiency.









- At constant N=1469 rpm and FIP=195 bar, increasing the compression ratio value from 16,17, 18 respectively, the importance of indicated power, brake power, mechanical efficiency, brake thermal efficiency increase, and brake specific fuel consumption **decreases**. Values of indicated thermal efficiency **increase** when CR is from 16 to 17 because the rise in the rate of indicated power is high and **falls** when CR is from 17 to 18 because of an increase in change of fuel consumption. Values of Indicated specific fuel consumption **decreases** when CR is from 16 to 17 because of less fuel consumption and high brake power and **increases** when CR is from 17 to 18 because of high fuel consumption and low brake power.

RESULTS

Following conclusions can be drawn from the experimentations carried out on the C. I. engine with Diesel at various compression ratios. The optimum compression ratio is 18 as operation for the given engine. Better fuel economy is obtained at the compression ratio of 18. Fuel consumption is higher at a compression ratio of 16.

4. Validation of theoretical Analysis

Waste heat losses arise both from equipment inefficiencies and thermodynamic limitations on equipment and processes. For example, consider internal combustion engine approximately 30 to 40% is converted into practical mechanical work. The remaining heat is expelled to the environment through exhaust gases and engine cooling systems. It means approximately 60 to 70% energy losses as waste heat through exhaust (30% as engine cooling system and 30 to 40% as the environment through exhaust gas). At higher pressure, there is a leakage of an air-fuel mixture or combustion gases between a piston and the cylinder wall into the crankcase of an automobile technology, which is the mass blow by process and which is accompanied by the heat loss from the engine. For the recovery of heat from the sides of the cylinder wall, we developed a Matlab code. The equations used are as follows:

Energy is released by heating the fuel; out of this total energy, some proportion of energy is lost; hence energy utilized is,

$dQ = Q_{in}dx – dQ_1$

Heat loss is dQ1, and it is given as follows, using Newton's law of cooling process,

$$\frac{dQ}{d\theta} = hA(T\text{-}T)$$

Differential form of the first law of thermodynamics applied to energy equation for an open system is given as follows:

$$\frac{dQ}{d\theta} - P\frac{dV}{d\theta} = mc_v\frac{dT}{d\theta} + c_vT\frac{dm}{d\theta} + \frac{\dot{m}_1 h_1}{\omega} \qquad (1)$$

The logarithmic derivative of equation of state is given as follows:

$$\frac{1}{P}\frac{dP}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} = \frac{1}{m}\frac{dm}{d\theta} + \frac{1}{T}\frac{dT}{d\theta} \qquad (2)$$

Where, m_l is instantaneous mass loss in the cylinder or blowby from the cylinder.

Removing dT/dΘ from equations (1) and (2), and using

$$\frac{dm}{d\theta} = -\frac{\dot{m}_1}{\omega} \quad ,$$

$$\frac{dP}{d\theta} = -\gamma\frac{P}{V}\frac{dV}{d\theta} + \frac{\gamma-1}{V}\frac{dQ}{d\theta} - \frac{\gamma}{\omega}\frac{\dot{m}_1}{m}P$$

Hence,

$$\frac{d\tilde{P}}{d\theta} = -\gamma\frac{\tilde{P}}{\tilde{V}}\frac{d\tilde{V}}{d\theta} + \frac{\gamma-1}{\tilde{V}}\left[\tilde{Q}\frac{dx}{d\theta} - \tilde{h}(1+\beta\tilde{V})(\tilde{P}\tilde{V}/\tilde{m} - \tilde{T})\right] - \frac{\gamma C\tilde{P}}{\omega}$$

$$\frac{d\tilde{W}}{d\theta} = \tilde{P}\frac{d\tilde{V}}{d\theta}$$

$$\frac{d\tilde{Q}_1}{d\theta} = \tilde{h}(1+\beta\tilde{V})(\tilde{P}\tilde{V}/\tilde{m} - \tilde{T})$$

$$\frac{d\dot{m}}{d\theta} = -C\frac{\tilde{m}}{\omega}$$

Here, $C = \frac{\dot{m}_1}{m}$ is called the blowby coefficient.

Data for ideal cycle parametric analysis:

thetas = -20; % start of heat release (deg)
thetad = 40; % duration of heat release (deg)
r =18; % compression ratio
gamma = 1.4; % gas const
Q = 20.; % dimensionless total heat release
h = 0.2; % dimensionless ht coefficient
tw = 1.2; % dimensionless cylinder wall temp
beta = 1.5; % dimensionless volume
a = 5; % weibe parameter a

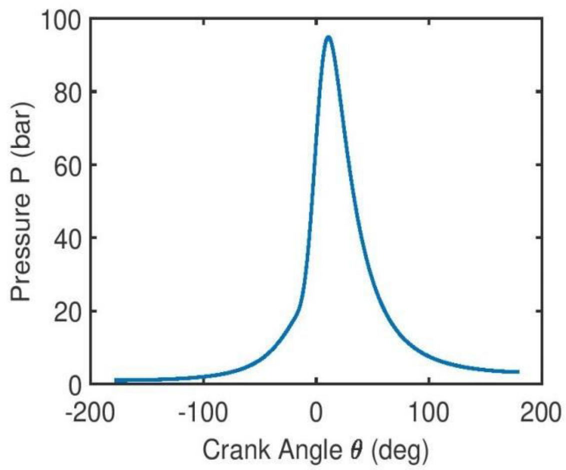
n = 3; % weibe exponent n
omega =200.; % engine speed rad/s
c = 0.8; % mass loss coeff
step=1; % crankangle interval for calculation/plot
NN=360/step; % number of data points
theta = -180; % initial crankangle

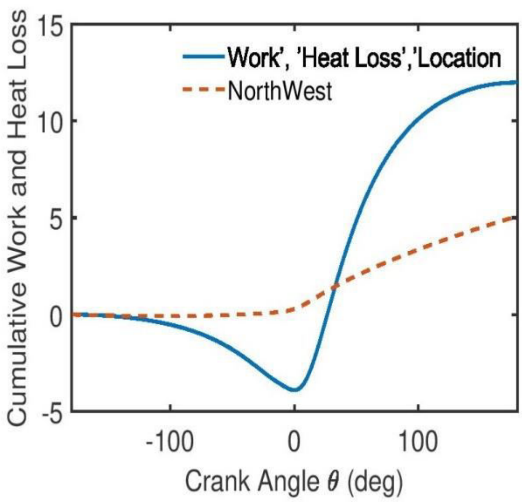
PARAMETRIC ANALYSIS

We developed a Matlab code and varied these parameters as follows:

1) VARIATION IN DIMENSIONLESS TOTAL HEAT RELEASE (Q):-

PARA METER	(Mass loss)/m	(Heat loss) /P_1V_1	(Net work)/ P_1V_1	Efficiency	Imep/ P_1
17	0.025	3.82	8.79	0.517	9.77
19	0.025	4.24	9.86	0.519	10.95
20	0.025	4.45	10.39	0.520	11.55
21	0.025	4.66	10.92	0.520	12.14
23	0.025	5.08	11.99	0.521	13.32





- As dimensions total heat release increases from 17 to 23, the mass loss remains constant, heat loss, network, efficiency, and Imep gains throughout.

2) VARIATION IN COMPRESSION RATIOS (r):-

PARA METE R	(Mas loss)/ m	(Heat loss)/P_1V_1	(Network) / P_1V_1	Efficienc y	Imep / P_1
16	0.025	3.88	11.67	0.584	12.45
17	0.025	3.93	11.82	0.591	12.56
18	0.025	3.98	11.96	0.598	12.66

- As the Compression ratio increases from 8 to 12, the mass loss remains constant, heat loss decreases, network, efficiency, and Imep increases throughout.

6. Heat loss from Cylinder Wall

Total heat loss from engine cylinder wall =Heat loss*p_1v_1

Where,

$p_1 = 0.98*10^5$ N/m^2

$v_1 = 661$ cc

Table 6.1 heat loss calculation at cylinder wall for different CR

Sr. No	r	Q_i(kW)
1	16	0.251
2	17	0.254
3	18	0.258

- At optimum CR=18, we got 0.258 kW as internal heat loss from the engine.

7. Calculation of Heat Loss through the Exhaust in Internal Combustion Engine

- The quantity of waste heat contained in the exhaust gas is a function of both the temperature and the mass flow rate of the exhaust gas:

$$Q_e = m_E \, c_P \, \Delta T$$

Where Qe is the heat loss (kJ/min); m_E is the exhaust gas mass flow rate (kg/min); c_P is the specific heat of exhaust gas (kJ/kg°K), and ΔT is a temperature gradient in °K.

To enable heat transfer and recovery, the waste heat source temperature must be higher than the heat sink temperature. Moreover, the magnitude of the temperature difference between the heat source and sink is an essential determinant of waste heat's utility or "quality." The source and sink temperature difference influence the rate at which heat is transferred per unit surface area of the recovery system and the maximum theoretical efficiency of converting thermal from the heat source to another form of energy (i.e., mechanical or electrical). Finally, the temperature range has an essential function for the selection of waste heat recovery system designs.

Table 7.1 Temperature Range from Diesel Engine

Sr. No	Engine	Temperatures in °C
1	Single Cylinder Four Stroke Diesel Engine	456
2	Four Cylinder Four Stroke Diesel Engine (Tata Indica)	448
3	Six Cylinder Four Stroke Diesel Engine (TATA Truck)	336
4	Four Cylinder Four Stroke Diesel Engine (Mahindra Arjun 605 DI)	310
5	Genset (Kirloskar) at power 198hp	383
6	Genset (Cummims) at power 200hp	396

We did an exhaustive survey to measure exhaust temperature from automotive vehicles and stationary engines' internal combustion engines. It is shown in Table.

- Calculation:-

Assuming, Volumetric efficiency (η_v) is 0.8 to 0.9

Density diesel fuel is 0.84 to 0.85 gm/cc

Calorific value of diesel is 42 to 45 MJ/kg

Density air fuel is 1.167 kg/m^3

Specific heat of exhaust gas is 1.1-1.25 KJ/kg°K

- Exhaust heat loss through diesel engine compression ratio(V_r):

$$V_r = 1 + V_S/V_C$$

$$17.5V_C = V_C + 6.61*10^{-4}$$
$$V_C = 4 * 10^{-5} \ m^3$$

- TOTAL VOLUME = $V_C + V_S$

$$= 4*10^{-5} + 6.61*10^{-4}$$

$$= 7.01*10^{-4} \ m^3$$

- Mass flow rate at CR 18 $m_f = 1.944* 10^{-4}$ kg/sec

- Volume Rate = swept volume * speed

$$= V_s * n$$

$$= 6.61 * 10^{-4} * \frac{1500}{2}$$

$$= 0.4957 \ m^3/min$$

$$= 8.262 * 10^{-3} \ m^3/sec$$

- **Volumetric efficiency(η_v)**

$$\eta_v = \frac{volume\ of\ air}{swept\ volume}$$

$$\eta_v = m_a / \varrho_a * n * V_S$$

$m_a = \eta_v * \varrho_a * n * V_S$

$=0.9 * 1.160 * 1500/2 * 6.61 * 10^{-4}$

$=0.5175$ gm/min

$=8.625$ gm/sec

- **Mass flow rate of exhaust gas (m_E)**

$m_E = m_f + m_a$

$=0.1944 + 8.625$

$=8.8194$ gm/hr

$=8.819 * 10^{-3}$ gm/sec

- **Heat loss in exhaust gas(Q_E)**

$=m_E\ c_P\ \Delta T$

$= 8.819 * 10^{-3} * 1.1 * (450-30)$

$=4.074$ KJ/sec

Overall Heat Loss

- Total Heat Loss = $Q_{in} + Q_{ex}$

$= 0.258 + 4.074$ kW

$= 4.332$ kW

Therefore the total heat loss from the engine is 4.332kW.

8. Feasible solutions using IC Engine waste heat

A huge quantity of hot flue gases is generated from internal combustion engines Etc. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. It depends upon the mass flow rate of exhaust gas and the temperature of exhaust gas. The internal combustion engine energy lost in waste gases cannot be fully recovered. However, much of the heat could be recovered, and losses are minimized by adopting specific measures. The heat recovery from exhaust gas and conversion into mechanical power is possible with the help of Rankine, Stirling, and Brayton thermodynamic cycles, vapor absorption cycle. These cycles are proved for low-temperature heat conversion into usable power.

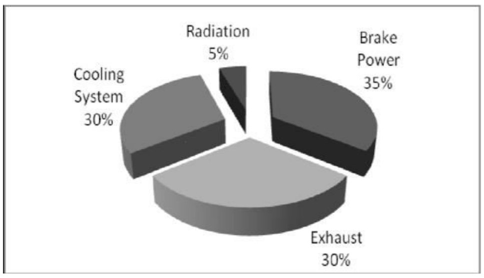


Fig 8.1 Total Fuel Energy Content in I. C. Engine

Waste heat is heat generated in a process by way of fuel combustion or chemical reaction and then "dumped" into the environment even though it could still be reused for some valuable and economic purpose. This heat depends partly on the temperature of the waste heat gases and the mass flow rate of exhaust gas. Waste heat losses arise both from equipment inefficiencies and thermodynamic limitations on equipment and processes. For example, consider internal combustion engine approximately 30 to 40% is converted into valuable mechanical work. The remaining heat is expelled to the environment through exhaust gases and engine cooling systems. It means approximately 60 to 70% energy losses Review on Exhaust Gas Heat Recovery for IC Engine J. S. Jadhao, D. G. Thombare PG Student, Automobile Engineering Department, R.I.T., Sakharale, Dist. Sangali, (MS) Professor, Automobile Engineering Department, R.I.T., Sakharale, Dist. Sangali, (MS) ISSN: 2277-3754 ISO 9001:2008 Certified International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 12, June 2013 94 as waste heat through exhaust (30% as engine cooling system and 30 to 40% as the environment through exhaust gas). Exhaust gases immediately leaving the engine can have temperatures as high as 842-1112°F [450-600°C].

Consequently, these gases have high heat content, carrying away as exhaust emission. Efforts have done to design a more energy-efficient reverberatory engine with better heat transfer and lower exhaust temperatures; however, the laws of thermodynamics place a lower limit on the temperature of exhaust gases. Fig. show total energy distributions from the internal combustion engine.

Possibility of Waste Heat from Internal Combustion Engine

Today's modern life dramatically depends on automobile engines, i.e., Internal Combustion Engines. Most vehicles are still powered by either spark ignition (SI) or compression ignition (CI) engines. CI engines, also known as diesel engines, have a wide field of applications, and as energy converters, they are characterized by their high efficiency. Small air-cooled diesel engines of up to 35 kW output are used for irrigation purposes, small agricultural tractors, and construction machines, whereas large farms employ tractors of up to 150 kW output. Water or air-cooled engines are used for a range of 35-150 kW, and unless a strictly air-cooled engine is required, water-cooled engines are preferred for higher power ranges. Earthmoving machinery uses engines with an output of up to 520 kW or even higher, up to 740 kW. Marine and locomotive applications usually employ machines with an output range of 150 kW or more. Trucks and road engines usually use high-speed diesel engines with 220 kW output or more. Diesel engines are used in small electrical power generating units or standby units for medium capacity power stations.

Table 8.1 Various Engine and There Output

Engine type	Power output kW	Waste heat
Small air cooled diesel engine	35	30-40% of Energy Waste loss From I.C. Engine
Small agriculture tractors and construction machines	150	
Water air cooled engine	35-150	
Earth moving machineries	520-720	
Marine applications	150-220	
Trucks and road engines	220	

The Table shows the various engine and their power ranges. In general, diesel engines have an efficiency of about 35%, and thus, the rest of the input energy is wasted. Despite recent improvements in diesel engine efficiency, a considerable amount of energy is still expelled to the ambient with the exhaust gas. In a water-cooled engine, 35 kW and 30-40% of the input energy are wasted in coolant and exhaust gases, respectively. The amount of such loss, recoverable at least partly, greatly depends on the engine load. Mr. Johnson found that for a typical 3.0 l engine with a maximum output power of 115 kW, the total waste heat dissipated can vary from 20 kW to 400 kW across the range of usual engine operation. It is suggested that for a typical and representative driving cycle, the average heating power available from waste heat is about 23 kW, compared to 0.8–3.9 kW of cooling capacity provided by typical passenger car VCR systems. Since the wasted energy represents about two-thirds of the input energy and for the sake of a better fuel economy, exhaust gas from Internal Combustion engines can provide a vital heat source that may be used in several ways to provide additional power and improve overall engine efficiency. These technical possibilities are currently under investigation by research institutes and engine manufacturers. For the heavy-duty ISSN: 2277-3754 ISO 9001:2008 Certified International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 12, June 2013 95 automotive diesel engines, one of the most promising technical solutions for exhaust gas waste heat utilization appears to be the use of helpful work.

POSSIBLE WAY OF USING HEAT RECOVERY SYSTEM

The increasing fuel costs and diminishing petroleum supplies are forcing governments and industries to increase the power efficiency of engines. A cursory look at the internal combustion engine heat balance indicates that the input energy is divided into roughly three equal parts: energy converted to practical work, the energy transferred to coolant, and energy lost with the exhaust gases. There are several technologies for recovering this energy on an internal combustion engine, whereas the dominating ones are: Waste heat can be utilized for heating purposes, power generation purpose, refrigeration purpose, Etc.

A Heating Purpose

Waste heat can be utilized for heating purposes like space heating, Preheating intake air and fuel, dryer, Etc. Typical examples of use would be preheating combustion air, space heating, preheating boiler feed water or process water, Etc. A waste heat recovery system can be utilized for preheating intake air and intake fuel. Mhia Md et al. investigate the effect of preheating intake air on Nox emission on a diesel engine. They have designed waste heat recovery for preheating intake air and fabricated, and its impact has been tested on diesel combustion and exhaust emissions. The result shows that NOx emission is reduced with the new air preheating waste heat recovery setup. Higher inlet air temperature is caused by the lower ignition delay, which is responsible for lower NOx formation with air preheating. Uniform or better combustion is occurred due to preheating of inlet air, which also causes lower engine noise. They have represented easy vaporization, and better mixing of air and fuel occurs due to warm-up of inlet air, which causes lower CO emission. Heat energy is recovered from the exhaust gases, which causes lower heat addition, thus improving engine thermal efficiency. Low-grade fuel, such as kerosene, can blend with conventional diesel fuel in diesel engines. Using the air preheating system and 10% kerosene blend as fuel, the thermal efficiency is improved, and exhaust emissions (NOx and CO) are reduced compared to neat diesel fuel without using an air preheating system. F. Karaosmanoglu studied the use of alternative fuel in internal combustion engines leads to problems such as poor fuel atomization and low volatility, mainly originating from their high viscosity, high molecular weight, and density. It is reported that these problems may cause critical engine failures such as piston ring sticking injector coking, formation of carbon deposits, and rapid deterioration of lubricating oil after the use of alternative fuel for an extended period. Waste heat recovery helps preheat alternative fuel to reduce the viscosity of fuel, better fuel atomization, and low volatility of fuel.

B Vapour Absorption Refrigeration System

Heat recovery from automotive engines has been predominantly for turbocharging or cabin heating with absorption chillier. The experiments conducted on the system prove that the concept is feasible and could significantly enhance system performance depending on the part-load of the engine. Also, the idea could be used for refrigeration and air conditioning of vehicles. Heat recovery from automotive engines has been predominantly for turbocharging or cabin heating with absorption chillier. The experiments conducted on the system prove that the concept is feasible and could significantly enhance system performance depending on the part-load of the engine. Also, the idea could be used for refrigeration and air conditioning of vehicles.

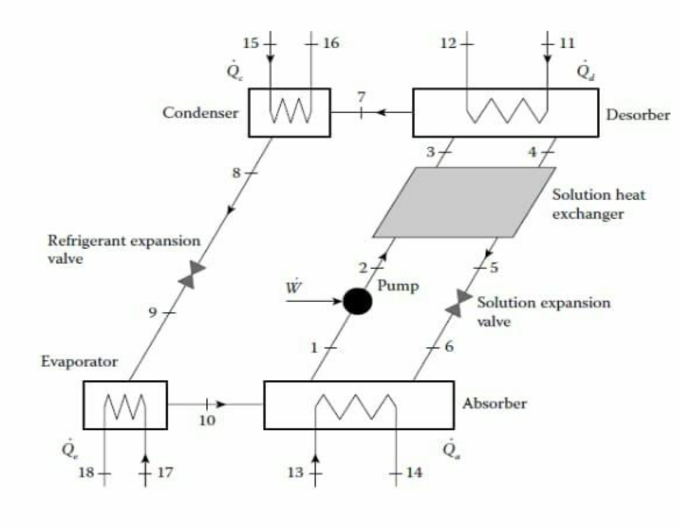


Fig 8.2 Vapour Absorption Cycle

Where,

T_{11}=Inlet temperature of the generator, T_{13}=Inlet temperature of the absorber, T_{15}=Inlet temperature of condenser, T_{17}=Inlet temperature of the evaporator, m_{11}=Mass flow rate of the generator, m13=Mass flow rate of an absorber, m_{15}=Mass flow rate of condenser, m_{17}=Mass flow rate of an evaporator, U_{Aa}=Internal energy of Absorber, U_{Ac}=Internal energy of condenser, U_{Ae}=Internal energy of evaporator, U_{ag}=Internal energy of generator.

In the vapor absorption refrigeration cycle, heat is provided at the generator, which generates the refrigerant vapors. These vapors are then condensed in a condenser by losing heat. The high-pressure liquid refrigerant is then throttled through an expansion valve to lower pressure at the evaporator. The refrigerant at much lower pressure and temperature evaporates and produces a cooling effect. The refrigerant vapors then pass to the absorber. The weak solution in the absorber absorbs the refrigerant vapors, and the answer is pumped to a higher pressure to the generator by a pump. The diluted solution from the generator is fed back to the absorber, where it absorbs the refrigerant vapors coming from the evaporator.

VARS in EES Software

Engineering Equation Solver (EES) is a commercial software package used to solve systems of simultaneous non-linear equations. EES stores thermodynamic properties, which eliminates iterative problem solving by hand through code that calls properties at the specified thermodynamic properties. In the EES software, we are changing temperatures, internal energy, and mass flow rate to find out the maximum heat required in the generator.

Pre-Determine value in EES Code:

```
SI=2

{Input data}
Eff_Hx=.64
{UAs=.132}
m[1]=.05

UAa=1.8            {kW/K}
UAc=1.2            {kW/K}
UAg=1
UAe=2.25

T[13]=25           {C}
m[13]=.28          {kg/sec}
T[15]=25
m[15]=.28
T[11]=100
m[11]=1.0
T[17]=10
m[17]=.4

Q[8]=0
Q[10]=1.0
```

Results

From the EES observations, we can say that heat required in the generator is a minimum of 12.581 kW to 16.128 kW.

The total % of the heat used in the VARS system is:-

$$\frac{\textit{Heat Generated from Engine}}{\textit{Heat Required in VARS System.}}$$

So, $Q_{min} = \frac{4.332}{12.581}$ kw $Q_{max} = \frac{4.332}{16.128}$ kw

Q_{min} = 34.43% Q_{max} = 26.86%

So, we can give 27% to 35% of heat from the exhaust heat to the VARS system.
Now, we know that,

1 Refrigeration Ton = 3.51 kW

So, by changing the temperature and internal energy of the evaporator, we can get the Q_{evp} and Q_{gen} for different tons and find the percentage of heat that can be used in the VARS system by dividing Q_{evp} by 3.51 kW.

Table 8.2 Percentage of heat use in VARS for different Ton

Ton	Q_{evp}(kW)	Q_{gen}(kW)	% heat used In VARS
1.5	5.366	8.376	51.71
2	7.146	10.376	41.75
3	10.702	14.651	29.56
4	14.266	17.284	25.06
5	17.943	20.445	21.18

C. Mechanical Turbo-compounding

A compressor and turbine on a single shaft boost the inlet air (or mixture) density. The energy available in the engine's exhaust gas is used to drive the turbocharger turbine, which causes the turbocharger compressor which raises the inlet fluid density before entry to each engine cylinder. The fig shows a turbocharged and turbo-compounded internal combustion engine is shown in fig. The turbo demonstrates a method that is presently utilized widely to convert waste energy to improve the efficiency and power output of the internal combustion engine. The problem with current turbochargers is that they do not extract all the potential energy available. The concept of using a turbine to recover energy comes from the turbocharger. The turbocharger is a mechanism that increases the power output of the engine using a turbine. Rather than using the turbine to power a compressor, the turbine could be connected to a generator. Alternatively, a series of turbines could be connected to a series of generators.

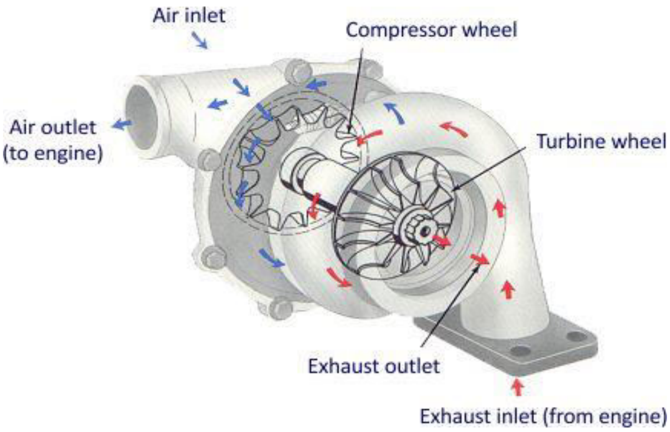


Fig 8.3 Turbocharger

If an efficient design was implemented, the alternator could be removed from the car to improve the engine's efficiency by lowering the load on it and by decreasing the weight of the vehicle itself. A turbine of this nature would have to be situated after the catalytic converter.

D. Rankine Cycle

The system is based on the steam generation in a secondary circuit using the exhaust gas thermal energy to produce additional power utilizing a steam expander. A particular case of low-temperature energy generation systems uses certain organic fluids instead of water in the so-called Organic Rankine Cycle (ORC). This technique has the advantage compared with turbo-compounding that does not have such a significant impact on the engine pumping losses and concerning thermoelectric materials that provide higher efficiency in using the residual thermal energy sources. The figure shows the waste heat recovery from the Rankine cycle operated at the low-temperature difference using unconventional fluids (refrigerants, CO2, binary mixtures). At a shallow heat source temperature, the trans-critical CO2 process produces the highest net power output. Rankine bottoming ISSN: 2277-3754 ISO 9001:2008 Certified International Journal of Engineering and Innovative Technology (IJEIT) Volume 2, Issue 12, June 2013 98 cycle techniques maximize energy efficiency; reduce fuel consumption and greenhouse gas emissions.

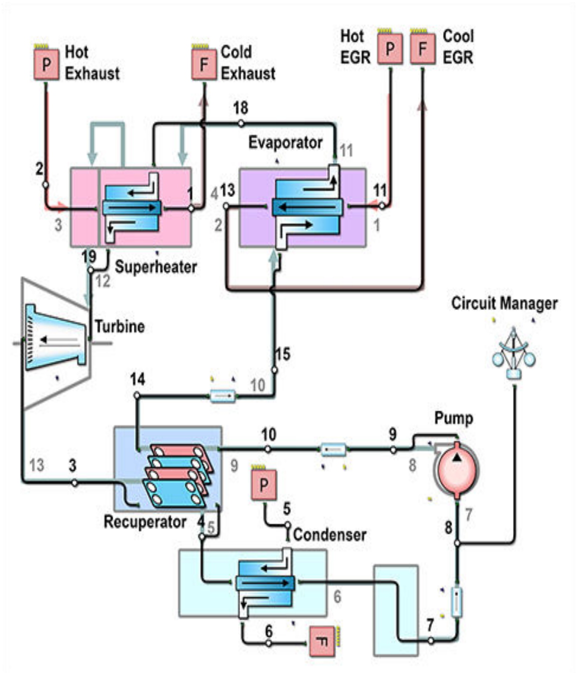


Fig 8.4 Rankine cycle

Recovering engine waste heat can be achieved via numerous methods. The heat can be reused within the same process or transferred to another thermal, electrical, or mechanical operation. Investigation and market evaluation of the Organic Rankine cycle can be applied in several cost-effective areas. Pinch point temperatures, heat exchangers cost, the critical temperature of working fluid would be a restriction for a maximum working pressure of cycle. As in Combined Heat and Power units, Organic Rankine cycles are options to improve total efficiency and reduce the cost. The thermoelectric method of exhaust gas waste heat recovery of a three-cylinder spark ignition engine is carried on the experimental-based test. Waste heat recovery using the Organic Rankine cycle is an efficient method compared with the other techniques, so automobile manufacturers use this method to enhance the efficiency of their products. The economic feasibility of waste heat recovery from diesel engine exhaust gas and analysis of harmfulness of the gases was done using the methods of purification and processing diesel engine exhaust gas. The computational model developed determined diesel exhaust emission rate and diesel exhaust waste heat rate and found beneficial results for diesel engines. The heat recovery can be done and increases with increasing exhaust mass flow rate.

9. CONCLUSION

Following conclusions can be drawn from the experimentations carried out on the CI engine with Diesel at various compression ratios, load and fuel injection pressure with waste heat calculations, and feasibility of waste heat recovery:

- The optimum compression ratio is 18 as operation for the given engine. Better fuel economy is obtained at the compression ratio of 18. Fuel consumption is higher at a compression ratio of 16. Smoke density is less at compression ratio 18. Exhaust gas temperatures are moderate at a compression ratio of 16. For more power, at high loads, the engine should operate at a compression ratio of 18 due to less specific fuel consumption. For lower power output at light loads, the engine should use a compression ratio of 16 due to less fuel consumption.
- We also calculated the Engine cylinder wall and heat loss at the exhaust, equal to 0.256 kW and 4.074 kW, respectively. So the total waste heat recovery from the engine comes out to be 4.332 kW.
- Waste heat recovery defines capturing and reusing the waste heat from the internal combustion engine for heating, generating mechanical or electrical work, and refrigeration system. It would also help to recognize the improvement in performance and emissions of the machine. If the automotive manufacturers adopted these technologies, it would result in efficient engine performance and Low emission.
- The waste heat recovery from exhaust gas and conversion into mechanical power is possible with the help of Rankine, Stirling, and Brayton thermodynamic cycles, vapor absorption. For waste heat recovery thermoelectric generator uses low heat, which has low efficiency.
- By using the EES software, we calculated the heat required in a generator which is necessary to run the VARS system of refrigeration, and at last, concluded the following points:

  If the VARS is 1.5 tons, 51.71% heat can be given from the engine to the generator. If the VARS is 2 tons, 41.75% heat can be provided from the engine to the generator. If the VARS is 3 tons, 29.56% heat can be delivered from the engine to the generator. If the VARS is 5 tons, the machine can provide 21.18% heat to the generator.

REFERENCES

[1] asoudIranmanesh, J.P.Subrahmanyam, M.K.G. Babu, Potential of Diethyl ether as supplementary fuel to improve combustion and emission characteristics of diesel engines, SAE paper number 2008-28-0044 (2008)

[2] M. Senthil Kumar. Experimental investigations on diesel engine with vegetable oils. Ph.D. Thesis. IIT Madras, Chennai, India.

[3] Ganesan V. 1998. Internal Combustion Engines. Tata McGraw-Hill, New Delhi.

[4] BS. Samaga. Vegetable oil as an alternative fuel for CI. engines.8th NCICEC-83. Paper No. AF-2.

[5] J. Ray Smith and Salvador Aceves. 1995. Series Hybrid vehicles and optimized hydrogen engine design. SAE Paper No. 951195.

[6]. Kapilan N., Mohanan P. and Reddy R.P., Performance and Emission Studies of Diesel Engine Using Diethyl Ether as Oxygenated Fuel Additive, SAE Paper number:2008-01- 2466 (2008).

[7]. Ashok MP and Saravanan CG, Effect of Diethyl Ether with Emulsified Fuel in a Direct Injection Diesel Engine, SAE Paper number: 2007-01-2126 (2007)

[8]. Subramanian K.A. and Ramesh A., Use of Diethyl Ether Along with Water-Diesel Emulsion in a Di Diesel Engine, SAE Paper number: 2002-01-2720 (2002).

[9] KS. Narayana, S. Phani Kumar, M.S.S. Srinivasa Rao "Thermo-Structural Finite Element Analysis of IC Engine Pistons "International Colloquium on Materials Manufacturing and Metrology, ICMMM 2014, August 8-9, IIT Madras, Chennai, India. P 882- 84.

[10] Lanka Tata Rao , Katakam Satyanarayana , M.S.S.Srinivasa Rao, T.V. Hanumanta Rao, S.V.UmamaheswaraRao, “stress analysis of 4stroke diesel engine piston "IJCESR ISSN (print): 2393-8374, (online): 2394-0697,volume-2,issue-2,2015

[11] K.Satyanarayana et al., Condition Monitoring And Detection Of Unbalance Using Vibration Signature Analysis. InternationalJournal of Recent Scientific Research Vol. 6, Issue, 8, pp.5796-5800, August 2015.

[12] D. Simhachalam, N. Indraja, M. Raja Roy "Experimental Evaluation of Mechanical Properties of Stainless Steel by TIG Welding at Weld Zone," International Journal of Engineering Trends and Technology (IJETT), V26(3),120-127 August 2015. ISSN:2231-5381

[13] V Ganeshan, “Internal Combustion Engine,” Tata McGraw Hill Publishing Company Limited, Second Edition, pp 35, 606-670.

[14] E.F. Doyle, P.S. Patel, “Compounding the truck diesel engine with an organic Rankine cycle system,” 760343, Society of Automotive Engineers (SAE), 1976.

[15] C. James Conklin, P. James Szybist, “A highly efficient six-stroke internal combustion engine cycle with water injection for in-cylinder exhaust heat recovery,” Energy 35 (4) (2010) 1658-1664.

[16] R. Saidur, M.Rezaei, W.K.Muzammil, M.H.Hassan, S.Paria, M.Hasanuzzaman, "Technologies to recover exhaust heat from internal combustion engines," Renewable and Sustainable Energy Reviews 16 (2012) 5649–5659.

[17] Hou Xuejun and Gao Deli, "Analysis of Exhaust Gas Waste Heat Recovery and Pollution Processing for Z12V190 Diesel Engine," Maxwell Scientific Organization, Res. J. Appl. Sci. Eng. Technol., vol.4, 2012, pp 1604-1611

[18] Mhia Md. Zaglul Shahadat, Md. Nurun Nabi And Md. Shamim Akhter, “Diesel Nox Reduction By Preheating Inlet Air," Proceedings Of The International Conference On Mechanical Engineering 2005. [19] F. Karaosmanoglu, “Vegetable oil fuels: a review,” Energy Sources 21,(1999) 221–231.

[20] Zhang X, Chau KT, “An automotive thermoelectric–photovoltaic hybrid energy system using maximum power point tracking,” Energy Conversion and Management 2011; 52(1):641–7.

[21] Vazquez J, Sanz-Bobi M, Palacios R, "State of the art of thermoelectric generators based on heat recovered from the exhaust gases of automobiles," In Proceedings of the seventh European workshop on thermoelectrics; 2002.

[22] Yuchao Wang, Chuanshan Dai, Shixue Wang, "Theoretical analysis of a thermoelectric generator using exhaust gas of vehicles as the heat source," Appl Energy (2013), http://dx.doi.org/10.1016/j.apenergy.2013.01.018.

[23] Jihad G. Haidar and Jamil I. Ghojel, “Waste Heat Recovery From The Exhaust Of Low-Power Diesel Engine Using Thermoelectric Generators,” 20th International Conference On Thermoelectric (2001), pp 0-7803-5908-9, 02001 IEEE.

[24] k. m. car, m. k. Mansour and m. n. Musa, “Thermal design of automobile exhaust based thermoelectric generators: objectives and challenges,” International Journal of Automotive Technology, (2008) Vol. 9, No. 2, pp. 155-160.