Energy and Exergy Analysis of Automotive Vehicles in Pakistan-Evaluation of Waste Heat Recovery Potential to Drive Parasitic Loads

By

Muhammad Ishaq Khan

Reg# 00000171533

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Supervised by

Dr. Adeel Javed

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National University of Sciences and Technology (NUST)

H-12, Islamabad 44000, Pakistan

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Signature: ________________________________

Name of Supervisor Dr. Adeel Javed

Date: __________________________

Signature (HoD TEE): __________________________

Date: _______________________________________

Signature (Dean/Principal): ___________________

Date: _______________________________________

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This is to certify that work in this thesis has been carried out by **Mr. Muhammad Ishaq Khan** and completed under my supervision in US-Pakistan Center for Advanced Studies in Energy (USPCAS-E), National University of Sciences and Technology, H-12, Islamabad, Pakistan.

Supervisor: ______________

Dr. Adeel Javed
USPCAS-E
NUST, Islamabad

GEC member # 1: ______________

Dr. Majid Ali
USPCAS-E
NUST, Islamabad

GEC member # 2: ______________

Dr. Muhammad Zubair
USPCAS-E
NUST, Islamabad

GEC member # 3: ______________

Dr. Hassan Abdullah
USPCAS-E
NUST, Islamabad

HoD- (TEE) ______________

Principal/ Dean ______________

Dr. Zuhair S. Khan
USPCAS-E
NUST, Islamabad

Dedication

I humbly dedicate this work to my parents whose endless prayers, continuous motivation and infinite sacrifices equipped me with everything it takes to be an accomplished human being.

Acknowledgment

I would like to thank my thesis advisor Dr. Adeel Javed of the USPCAS-E NUST. He is the driver of my research idea, from the formulation of the problem statement to the result of my research. He consistently allowed this study to be my own work but steered me in the right direction whenever he thought I needed it. His directions were succinct and encouraging.

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Abstract

Road transport is considered as the back bone of Pakistan's economy. Road transport vehicles are powered by ICEs. The thermal efficiency of ICE is about 30-35%, rest of the thermal energy is wasted through the cooling water and exhaust gases (contributes to global warming). The fuel consumption is also increased due to the engine parasitic loads. Energy and exergy analysis of the exhaust gases of the selected internal combustion engines have been done in this study. Recoverable waste heat is estimated in the study. This thesis proposes an E-Turbine system to recover waste heat and to replace the parasitic loads (alternator and AC compressor). Engines R06A (0.6L), F8B (0.8), K10B (1L), 2NZ-FE (1.3L) and 2ZR-FE (1.8L), running in Pakistan, are modeled in MATLAB at full load. Brake power, torque, exhaust energy and mass flow rate of exhaust gases are plotted at different rotational speed of internal combustion engines. Temperature of exhaust gas is plotted as a function of brake power and rpm. Exergy analysis of exhaust gases is also done in this study. At 3500 rpm and full load, exhaust gases of engine K10B have the exhaust energy of 25.69 kW and exergy of 13.64 kW. In the latter part of the paper, for the design point 3500 rpm of engine K10B, radial turbine is selected from the NsDs diagram. Diameter of the turbine is 0.061 m at the blade tip speed of 500 m/s. The selection of high-speed generator is done from the literature. The battery sizing and cost analysis of the system has been done.

Keywords: AC compressor, Alternator, MATLAB, Exergy, E-Turbine

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1. **Muhammad Ishaq Khan,** A. Javed "system design of high-speed E-Turbine for waste heat recovery in automotive vehicles**"** International conference on renewable applied and new energy technologies, November 2016, Air University Islamabad

Nomenclature

GDP	Gross domestic product
IEA	International Energy Agency
WHRT	Waste Heat Recovery Technologies
LP	Low Pressure
HVAC	Heating, Ventilation and Air Conditioning
EMF	Electromotive force
FY	fiscal year
x_b	Mass fraction of fuel burned
θ	Crank Angle
θ_s	Crank angle at the start of injection
θ_d	Duration of Combustion
n	Wiebe form factor
a	Wiebe efficiency factor
V_θ	Instantaneous cylinder volume
Vc	Clearance volume
Vmax	maximum volume
B	cylinder diameter
Y	instantaneous distance of piston from top dead center
A	Availability
Q_{exh}	Exhaust energy
m_{exh}	mass flow rate of exhaust gases
T_{amb}	Ambient temperature
Ns	specific speed of turbine
Ds	specific diameter of turbine
Δhos	isentropic enthalpy change
D	diameter of turbine
Po_1	pressure of exhaust gas before turbine
Po_2	pressure of exhaust gas after turbine
P_{amb}	Ambient pressure

Chapter 1

Background and Scope

Contemporary research and efforts for the technological advancement in ICEs and vehicle design are primarily motivated by the dire need to reduce the worldwide fossil fuel consumption and the resultant emissions of greenhouse gases. Oil and natural gas are currently the significant energy carriers in transportation. Across IEA countries, the transport sector accounted for the highest share of energy consumption in 2014 (34%) [1]. In Pakistan, the petroleum consumption in the sector reaches 10 million TOEs (Tons of Oil Equivalent) in 2013, registering a growth rate of 1.8 percent in the last decade [2]. Worldwide oil production is likely to peak before 2030.

Internal combustion engine is the power house for many applications such as automobiles, trucks, light aircrafts, ships, and electric power generators. Due to its high power to weight ratio, simplicity and ruggedness, they are considered as the most favorable option for power producing operations. There are two types of internal combustion engines based on the method of ignition. 1) spark ignition: in this type of engine, air-fuel mixture is compressed in the compression stroke and then spark is generated by the spark plug just before the completion of compression stroke. The compression ratio is limited in these types of engine due to the auto-ignition of the fuel. 2) Compression ignition: air is compressed above the auto-ignition temperature of the fuel such that when the fuel is sprayed (atomized), the combustion starts spontaneously. The compression ratio is higher than the spark ignition engine since only air is compressed in compression ignition engines and there is no possibility of knocking [3].

Based on working cycle, the internal combustion engine is further classified into two groups. 1) 4-Stroke engine: engine in which the thermodynamic cycle completes in 4 strokes of the piston (intake stroke, compression stroke, power stroke and exhaust stroke) or in two mechanical cycles. The power stroke is produced after every two revolutions of crankshaft. 2) 2-Stroke engine: engine in which the thermodynamic completes in two strokes (compression stroke and power stroke) of the piston or in one mechanical cycle. The power stroke is produced after each revolution of the crankshaft [4].

The thermal efficiency of modern internal combustion is around 30-35%. Complex irreversible processes inside the engine cylinder puts an upper limit on engine thermal efficiency. A high temperature variation occurred due to the prompt expansion of combustion gases in the engine's cylinder, due to which turbulence in fluid motion and large amount of heat is transferred between piston and cylinder walls. It has been estimated that around 33% of fuel energy is carried away by the engine cooling system because for better working of IC engine, it must be in the safe temperature range, so it has a good cooling system. Around 30-35% of fuel energy is lost through engine exhaust to the atmosphere, containing greenhouse gases, contributing to the global warming. In the past decade, out of world's total CO_2 emissions from fuel combustion, 23% was emitted by the transportation which includes 17% CO_2 emissions by the road transport [5]. The existing automotive sector in Pakistan is hardly complying with obsolete Euro II emissions standards. According to World Bank report, Pakistan's transport sector is responsible for 362 Mt (million metric tons) CO_2 emissions in 2011.

The fuel consumption of conventional automotive engine is also negatively affected by the two parasitic loads, namely alternator and AC compressor of the car. Both these loads are driven by the engine crankshaft through the belt drive system. These two parasitic loads also affect the pick of the vehicle. Almost all vehicles in Pakistan are powered by the conventional automotive engines (naturally aspirated). On one side, huge of amount of fuel energy is wasted through exhaust and on the other side, the two parasitic loads are increasing the fuel consumption of engine. This should be an alarming issue for a developing country like Pakistan.

This thesis proposes the system that will be used for the waste heat recovery of combustion engine and will replace the two parasitic loads namely 1) alternator 2) AC Compressor. The E-Turbine concept is a novel and innovative idea for the recovery of automotive waste heat. This technology is best suitable for conventional automotive vehicles. The E-Turbine system consist of turbine, spindle and generator. The working principle of E-Turbine system is that exhaust gases pass through the turbine that is placed in the exhaust manifold of engine. Turbine extract power from the exhaust that will otherwise be wasted. Turbine shaft rotates, upon which an electric generator is mounted. Due to change in EMF,

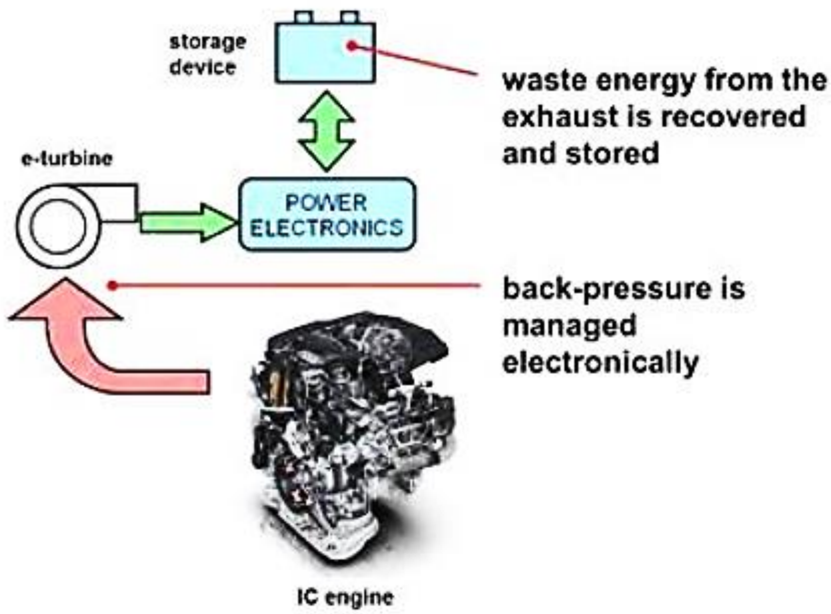

Figure 1: Illustration of E-Turbine concept

generator produced electricity. This electrical power is used for the operation of HVAC compressor and for battery charging which is used to run other electrical accessories of the vehicle. This technology replaces the two parasitic loads that are discussed above and recover the waste heat in a simple way.

The first and the most important step in the designing of the E-Turbine system is the thermodynamic analysis of the automobiles running in Pakistan. The energy and exergy analysis of the exhaust gases of the engines R06A (0.6L), F8B (0.8L), K10B (1L), 2NZ-FE (1.3L) and 2ZR-FE (1.8L) are done in this thesis. This is important because the quantity and quality of the energy of exhaust gases is necessary for the selection and designing of the turbine. Radial turbine selection is done from NsDs diagram based on certain parameters which are discussed in the next chapters. The diameter of the radial turbine is calculated, and its value is 0.061 m. The blade tip speed of the turbine 500 m/s.

Summary

Road transport is considered as the back-bone of Pakistan's economy. Internal combustion engines are used to power this type of raod transport which includes spark ignition engines and compression ignition engines. The thermal efficiency of IC engines is very low and they also great contribution in CO_2 emssions causing global warming. An E-Turbine system is proposed for the waste heat recovery of exhaust gases and to replace the parasitic loads of AC compressor and alternator on the IC engine, thus improving the fuel consumption of automotive vehicles in Pakistan.

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[2] Pakistan Energy Yearbook (2013), Hydrocarbon Development Institute, Ministry of Petroleum and Natural Resources, Government of Pakistan, Islamabad

[3] Heywood. McGraw Hill. internal combustion engine fundamentals.1988.

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[5] https://data.worldbank.org/indicator/EN.CO2.TRAN.ZS?locations=PK

Chapter 2

Introduction to IC Engines

Internal combustion engine is the power house for many applications such as automobiles, trucks, light aircrafts, ships, and electric power generators. Due to its high power to weight ratio, simplicity and ruggedness, they are considered as the most favorable option for power producing operations. The internal combustion engine has several complex processes that take place inside the cylinder. To understand the details of the processes, take place in the cylinder, one must have some knowledge of the basic terminologies. This section of the thesis presents some of the basic geometric and performance parameters.

Geometric Parameters

The geometric parameters are those which are related to the geometry of the engine i.e. cylinder, piston etc. some of the geometrical parameters are discussed below.

I. Cylinder Bore

The diameter of the cylinder is called bore of the cylinder. It is represented by B.

II. Stroke

Stroke is the distance the piston covers when it moves from TDC to BDC or from BDC to TDC. It is represented by S.

III. Top Dead Center

The position of the cylinder when the cylinder has its minimum volume is called Top dead center. It is represented by TDC.

IV. Bottom Dead Center

The position of the cylinder when the cylinder has its maximum volume is called Bottom dead center. It is represented by BDC.

V. Clearance Volume

The volume formed inside the cylinder when the piston is at TDC is called clearance volume. It is represented by Vc.

VI. Swept Volume

The maximum volume the cylinder swept when the piston moves from TDC to BDC is called Swept volume. It is also called displacement volume. It is represented by V_d.

VII. Maximum Volume

The maximum volume is the summation of clearance volume and swept volume. It is represented by Vmax.

VIII. Compression Ratio

The ratio of maximum volume to the minimum volume or clearance volume is called compression ratio. It is represented by r_c.

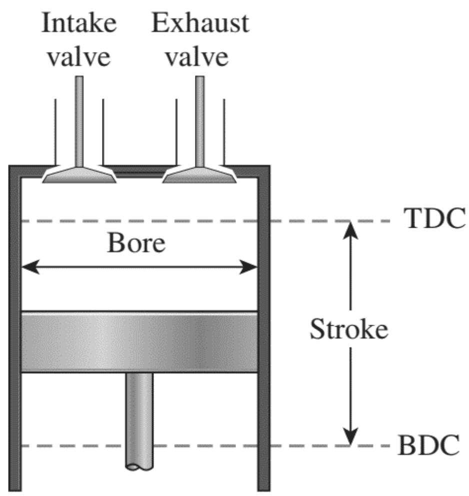


Figure 2: Geometry of engine cylinder

Performance Parameters

The performance of internal combustion engine is calculated in terms of certain parameters which are discussed below.

I. Thermal Efficiency

Thermal efficiency is the ratio of net work produced by the engine to the total heat input. It is represented by η_{th}. The mathematical equation is given as under,

$$\eta_{th}=\frac{W_{net}}{Q_{in}}$$

II. Mean Effective Pressure

Mean effective pressure is the virtual pressure that if it acted upon the piston would produce the same power output as that produced in the actual cycle. It is represented by MEP. Mean effective pressure is used to compare the performance of engines having the same displacement volume. Engines having high MEP would produce more net power than the one having low MEP. The mathematical equation is given as under,

$$\mathrm{MEP} = \frac{\mathrm{W_{net}}}{\mathrm{V_{max}} - \mathrm{V_{min}}}$$

V_{min} is the clearance volume.

III. Brake Power

Brake power is another significant performance parameter of IC engine. It is the power obtained at the shaft of the engine. It is represented by $\dot{W}_b$.

IV. Torque

The performance of internal combustion is also characterized by measuring the torque obtained at the shaft. Torque is the measure of power per unit rotation of crankshaft. It is represented by τ. Maximum torque occur at low engine rpm.

V. Specific Fuel Consumption

Specific fuel consumption is the fuel flow rate per unit power output. It is represented by sfc.

Ideal Thermodynamic Cycles for IC Engine

Actual processes have irreversibilities and do not have enough time to achieve equilibrium conditions. Therefore, the study of actual processes is very difficult and require the detail information of all the process. When all the irreversibilities and heat transfer etc. are neglected in the cycle, the cycle we obtained is called ideal cycle. The ideal cycle gives the general trend of the processes and do not go in the detail of the process.

There are two ideal cycle for the internal combustion engine which are discussed below.

Otto Cycle

Otto cycle is the ideal cycle for the spark-ignition IC engines. It is named after Nikolaus A. Otto. Otto cycle is consisting of four internally reversible processes.

Process 1-2; Isentropic compression: working fluid is compressed isentropically (entropy constant).

Process 2-3; constant volume heat addition: heat is added to the working fluid at constant volume which increase the pressure and temperature of the air.

Process 3-4; isentropic expansion: air is expanded isentropically producing power output.

Process 4-1; constant volume heat rejection: heat is rejected from the air to the environment and the working fluid reaches to its initial state.

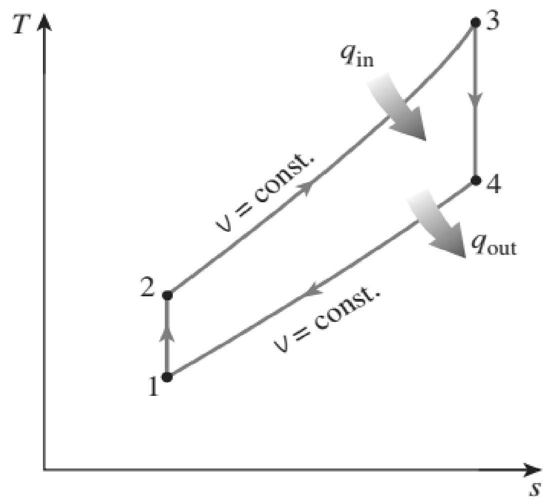


Figure 3: T-s diagram of Otto Cycle

The thermal efficiency for the Otto cycle is given by,

$$\eta_{th,otto} = 1 - \frac{1}{{r_c}^{\Upsilon-1}}$$

Diesel Cycle

Diesel cycle is the ideal cycle for the compression ignition engines. It is named after Rudolph Diesel. It is similar to the Otto cycle; the only difference is in the heat addition process. The heat addition in diesel cycle occurs at constant pressure rather than constant volume as in the Otto cycle. The Diesel cycle consists of four internally reversible processes.

Process 1-2; isentropic compression: working fluid is compressed isentropically (entropy constant).

Process 2-3; constant pressure: heat addition occurs to the cycle take place at constant pressure.

Process 3-4; isentropic expansion: air is expanded isentropically producing power output.

Process 4-1; constant volume heat rejection: heat is rejected from the air to the environment and the working fluid reaches to its initial state.

The thermal efficiency for the Diesel cycle is given by

$$\eta_{th,diesel}=1-\frac{1}{r_c^{\Upsilon-1}\left[r_{cr}^{\Upsilon-1}/\Upsilon(r_{cr}-1)\right]}$$

r_{cr} is the cut off ratio. Cut off ratio is the ratio of volume after heat addition to the volume before heat addition.

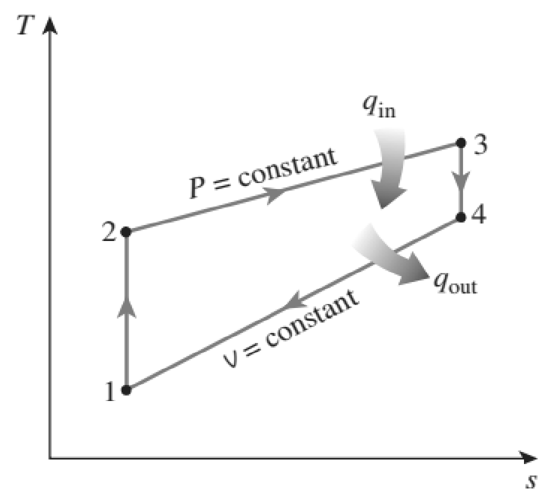


Figure 4: T-s diagram of Diesel cycle

Summary

Internal combustion engine is the power house for many applications due to its high power to weight ratio, simplicity and ruggedness. The important geometrical parameters include cylinder bore, stroke, TDC, BDC, clearance volume, swept volume and compression ratio. Thermal efficiency, MEP, brake power and torque are considered as the performance parameters of the IC engine. Otto Cycle and Diesel Cycle are the thermodynamic ideal cycles for SI ignition and Compression ignition engines respectively. The difference between Otto Cycle and Diesel Cycle is the condition at which heat transfer to the working fluid take place.

References

[1] Heywood, Internal combustion engine fundamentals

[2] Younus Cengel, Fundamentals of Engineering Thermodynamics

Chapter 3

Literature Review

Socio-economic development of a country greatly depends on an effective transport and communication sector. An efficient transport system with modern infrastructure is indicator of the economic growth of a country. The transport sector contributes around 10% to the Pakistan's GDP and 6% employment to the country [1]. Roads, railways, air transport, ports and shipping services are the most common modes of transportation. Road transport is the most used mode of transport in any developing country. In Pakistan, 96% of inland freight service is done by the road transport and covers 92% of passenger traffic, therefore is the backbone of Pakistan's economy [2]. All road transport vehicles are powered by the internal combustion engine. ICEs are the main consumers of fossil fuel. Contemporary research and efforts for the technological advancement in ICEs and vehicle design are primarily motivated by the dire need to reduce the worldwide fossil fuel consumption and the resultant emissions of greenhouse gases.

The finite reserves of fossil fuel are one of the utmost significant reasons of these efforts. Oil and natural gas are currently the significant energy carriers in transportation [3]. Across IEA countries, the transport sector accounted for the highest share of energy consumption in 2014 (34%) [4]. In Pakistan, the petroleum consumption in the sector reaches 10 million TOEs (Tons of Oil Equivalent) in 2013, registering a growth rate of 1.8 percent in the last decade [5]. The number of vehicles, mostly using petroleum products and natural gas, has increased rapidly from 2.7 million vehicles in 1990 to 9.8 million in 2010[6]. Worldwide oil production is likely to peak before 2030 [7].

A second, even more significant, reason behind the need to the technological advancement is the emissions of greenhouse gases. CO_2 emissions are generated by the combustion of fossil fuels which has the major contribution in global warming [8]. In the past decade, out of world's total CO_2 emissions from fuel combustion, 23% was emitted by the transportation which includes 17% CO_2 emissions by the road transport [9]. Since 1970, there is increase in CO_2 emissions by 80%, whereas more than 100% increase in CO_2 emissions is recorded from the transportation sector [10]. The existing automotive sector in Pakistan is hardly complying with obsolete Euro II emissions standards. According to World Bank report, Pakistan's transport sector is responsible for 362 Mt (million metric tons) CO_2 emissions in 2011. In 1971, CO_2 emissions from the transport sector were 20.038% of the total fuel combustion increasing to 28.9% in 2014 [11].

On the other side, at the technology level, ICEs have very low thermal efficiency. The ICEs are capable to convert only 30-35% of chemical energy of fuel to useful mechanical work and 60-70% of energy is remaining unused [12]. Complex irreversible processes inside the engine cylinder puts an upper limit on engine thermal efficiency. A high temperature variation occurred due to the prompt expansion of combustion gases in the engine's cylinder, due to which turbulence in fluid motion and large amount of heat is transferred between piston and cylinder walls. It has been estimated that around 33% of fuel energy is carried away by the engine cooling system because for better working of IC engine, it must be in the safe temperature range, so it has a good cooling system. Around 30-35% of fuel energy is lost through engine exhaust to the atmosphere, containing greenhouse gases, contributing to the global warming. The temperature of exhaust gases in the exhaust manifold is around 650K-1150K having high energy content [14].

In addition to the wastage of exhaust energy, conventional automotive engine has two parasitic loads (coupled with crankshaft pulley through belt drive system) that are also

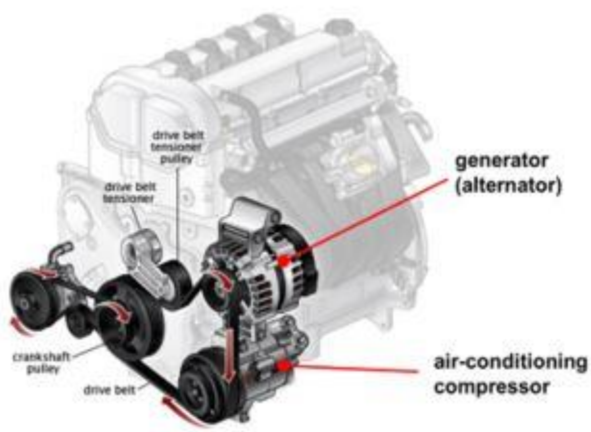


Figure 5: Conventional IC Engine setup

responsible for its low efficiency. These two parasitic loads are 1) battery charging system. Vehicle battery is charged by the electrical current produced by the alternator (generator). The alternator is powered by the crankshaft pulley through belt-drive system. 2) Vehicle HVAC system. HVAC system is an integral part of automotive vehicles, which ensures the thermal comfort of the passengers and responsible for the fresh air in the passenger cabin. The HVAC compressor is coupled with the crankshaft pulley through belt and draws several kilowatts from IC engine. On average the air conditioning system can increase the fuel consumption of a vehicle by 20% [19]. Setup of conventional automotive engine is shown in figure 1. In Pakistan, almost all the vehicles are equipped with conventional automotive engine (naturally aspirated and having these parasitic loads). The HVAC load is also substantial for the vehicles in Pakistan due to the long duration of summer season (7-9 months). The use of ACs in the car cabin is very necessary for the thermal comfort of passengers. Sanden AC compressors are usually installed in the vehicle used in Pakistan. It draws 5-7 kW power from the engine at its maximum refrigerating capacity.

Waste Heat Recovery Technologies

Major portion of waste heat from the engine is in the form of exhaust gases. The engineers and researchers tried to recover this exhaust energy. Various waste heat recovery technologies are developed. These technologies are discussed below.

Turbocharger

The turbocharger is a technologically mature system for waste heat recovery application. Turbocharger is primarily a turbine-driven compressor and remains largely associated with internal combustion (IC) engines. The introduction of turbocharger technology had improved the combustion inside the engine and its emissions. This technology also reduced the size of engine compared to the non-turbocharged engine for the same power output, due to which engine become light weight, small size and more efficient. Turbocharger is turbine-driven compressor which is used to compress the intake air [15].

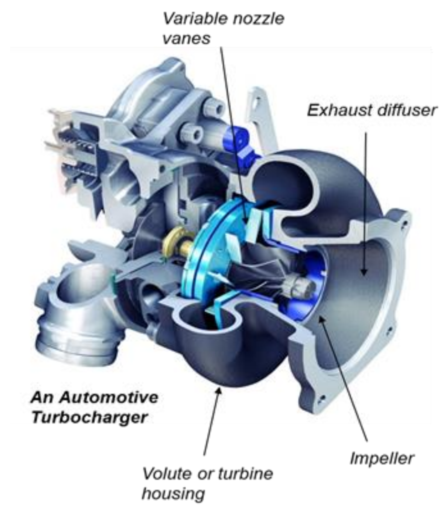


Figure 6: Turbocharger technology

The exhaust gas energy that would have otherwise been wasted is used to drive the turbine, which then transmits the shaft power to the compressor. Conventional turbochargers, however, face many aero-thermal and structural issues which influence their aerodynamic coupling or matching with the IC engine.

Electric Turbocharger

Electric turbocharging is another technique to recover automotive waste heat. The working principle of electric turbocharger is s that the exhaust gas coming out of the engine is directed to pass through the turbine of the turbocharger. The part of shaft work produced by the turbine is used to compress the air and supply it to the cylinder, and part of the shaft work is used to produce electricity via electric generator that is mounted on the common shaft of the turbocharger. This electrical power is used to run the electrical

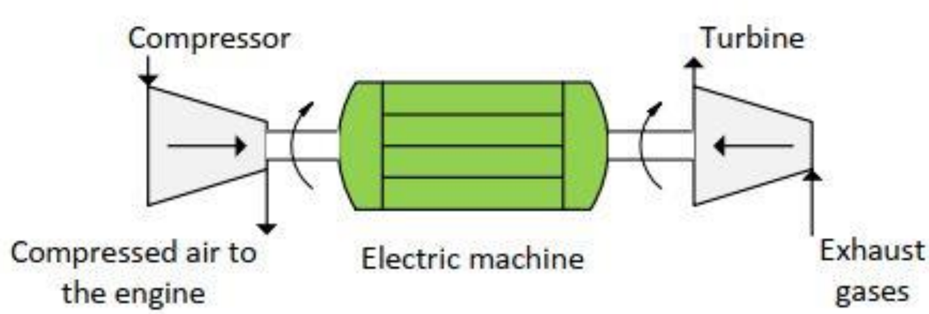


Figure 7: Electrical Turbocharger

accessories of the vehicle and can also be used to accelerate the turbocharger and reduce turbo lag at low speed engine operation [15].

Turbocompound

The turbocompound is another promising technology for the waste heat recovery of the internal combustion engines. An additional turbine, typically downstream of the turbocharger turbine, is placed in the exhaust system. The exhaust gases after the turbocharger turbine is expanded in the LP turbine. The expansion of the exhaust gases in the LP turbine produces the shaft work. This shaft work is either directly used to power the vehicle's wheel or used to drive the generator and produce electricity [16].

There are two types turbocompounding which are discussed below.

Mechanical Turbocompound

In mechanical turbocompounding, the shaft power is directly transferred to the wheels. The shaft power is reduced to the crankshaft's speed via gear trains [16].

Electric Turbocompound

In electric turbocompounding, an electric machine is mounted on the turbine shaft which produces the electricity which is used to run the electrical accessories of the vehicle [17].

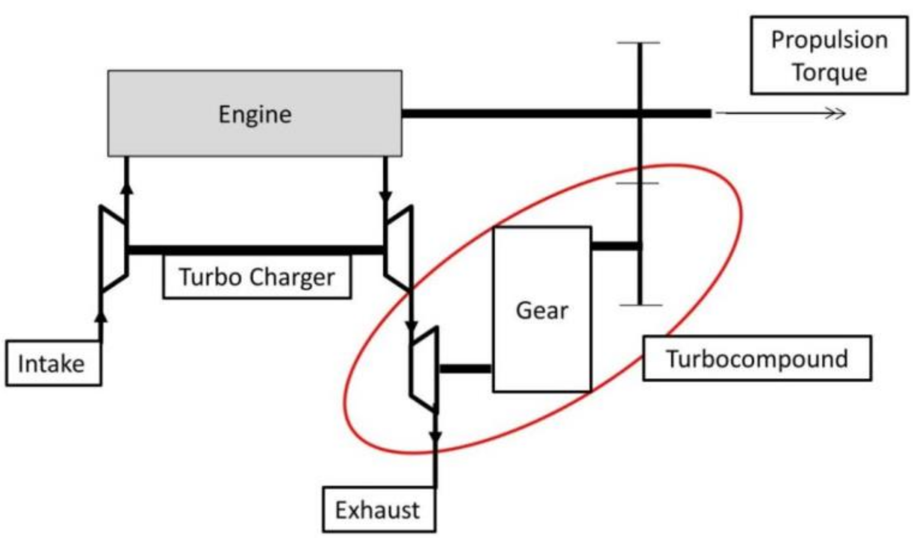


Figure 8: setup of mechanical turbocompound

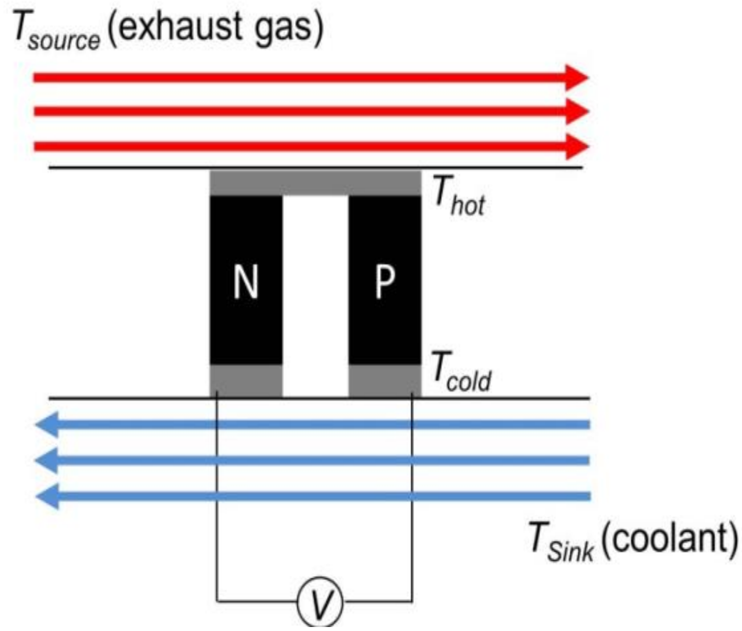


Figure 9: Seebeck effect

Thermoelectric Materials

Thermoelectric materials work on the principle of seebeck effect. The seebeck effect was discovered by Thomas Johann Seebeck in 1823. According to this effect, an electric potential is generated when there is a temperature gradient between two different conductors.

Thermoelectric materials are used for the waste heat recovery for automotive vehicles. The thermoelectric material is placed in the exhaust manifold, the hot exhaust gases and cooling fluid is passed which produces the temperature gradient across the junction of the device [13]. This produce the electric potential which can be used to run the electrical accessories. The schematic diagram of the thermoelectric generator installed for the waste heat recovery in the automotive vehicle.

Thermodynamic Cycles

Various thermodynamic cycles are used for the exhaust heat recovery. Sterling cycle, Kalina cycle, Brayton cycle and Rankine cycle are used for this purpose. The exhaust gases are used as a heat source for these cycles. Among them, Organic Rankine cycle

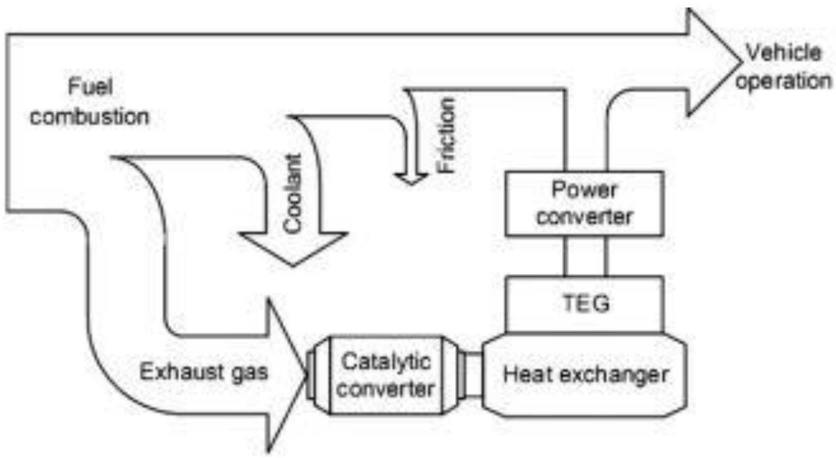


Figure 10: layout of thermoelectric generator

(ORC) is the most promising technology for the waste heat recovery of combustion engine.

Organic Rankine Cycle

Organic Rankine Cycle is the most suitable technology used for low and moderate heat energy i-e engine exhaust or coolant. The working fluid in the ORC has a low boiling point in addition with the other favorable properties. The pros of the Organic Rankine Cycle include simple structure, easy maintenance and high reliability [18].

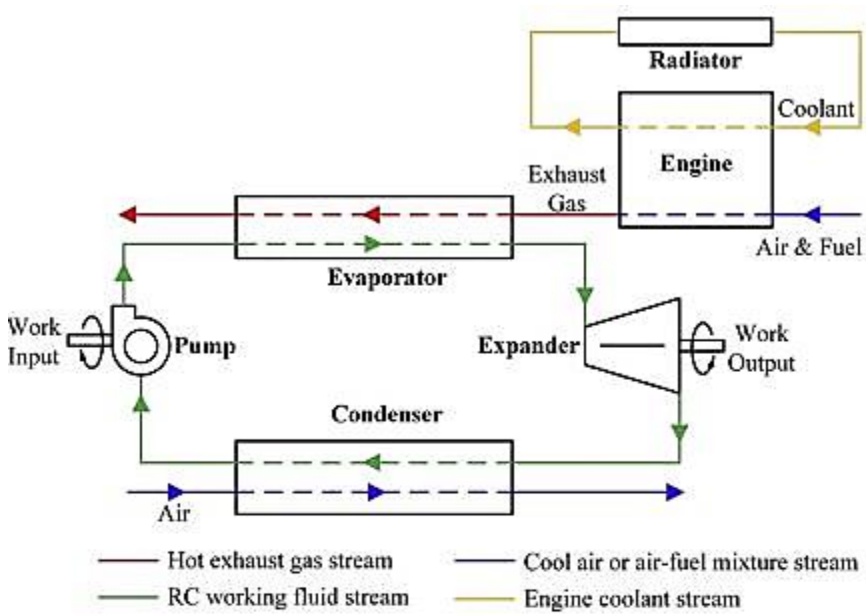


Figure 11: Setup of ORC as WHRT

Automobile sector of Pakistan

The automobile industry is the fastest growing industry in Pakistan. The automobile industry in Pakistan includes companies involved the production/assembling of passenger cars, light commercial vehicles, trucks, buses, tractors and motorcycles. The auto spare parts are also an integrated industry with it. They both form the major sector in Pakistan. Pakistan's auto industry is enjoying a boom as all leading automobile brands such as Suzuki, Toyota and Honda have reported high profit. The Automobile sector recorded a growth of 23.43 percent during July-March FY2016 compared to 17.06% in same period last year [20]. The production capacity of cars including jeeps, LCV, Pickup and Vans is 285500 units, against which 185000 units were produced in 2014-2015. The automobile industry is employing around 2.5 million direct and indirect labors and has a potential to employ much more than this but needs some innovations. The trend is shown in the figure.

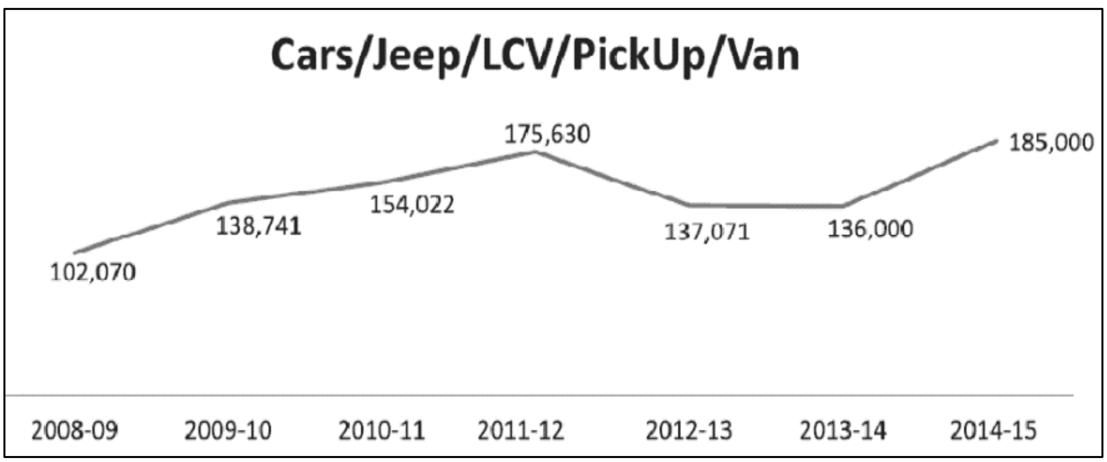


Figure 12: Automobile production in Pakistan

Most of the people in Pakistan use passenger cars and light commercial vehicles. These vehicles are equipped with the internal combustion engines namely R06A, F8B, K10B, 2NZ-FE and 2ZR-FE having the displacement volumes of 0.6L, 0.8L, 1L, and 1.3L and 1.8L respectively. These engines are naturally aspirated (no turbocharger is installed). The R06A internal combustion engine is installed in Suzuki WagonR, Alto and Carry having displacement volume of 0.6 L. Suzuki WagonR is the one of the most running cars in Pakistan. The R06A engine is inline-three engine that is produced by Suzuki since 2011. The F8B engine is inline-three 6-valves installed in Suzuki Mehran. The K10B internal combustion engine is installed in Suzuki Cultus, Suzuki Alto and Maruti WagonR. It is DOHC 12-valve having displacement volume of 1 L. Toyota Corolla, Toyota Belta, and Toyota Vitz are equipped with the Toyota 2NZ-FE engine. It is naturally aspirated, inline-four, DOHC 16-engine having displacement volume of 1.3 L. 2ZR-FE engine is installed in Toyota Grande etc. These engines are naturally aspirated (no turbocharger is installed). The exhaust energy is wasted in much more amount than the turbocharged engines. In the HVAC system of the automobiles in Pakistan, Sanden AC compressor is installed. It draws around 5 kW from the engine at its maximum refrigerating capacity. Therefore, in this thesis, engines R06A (0.6L), F8B (0.8L), K10B (1L), 2NZ-FE (1.3L) and 2ZR-FE (1.8L) displacement volumes are studied. Detail Engine specifications are given in the table.

(a)

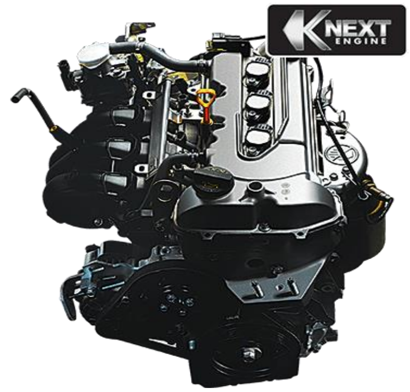
(b)

(c)

(d)

Figure 13: (a) Suzuki WagonR (b) Suzuki Cultus (c) Suzuki Corolla (d) Suzuki Mehran

(a)



(b)

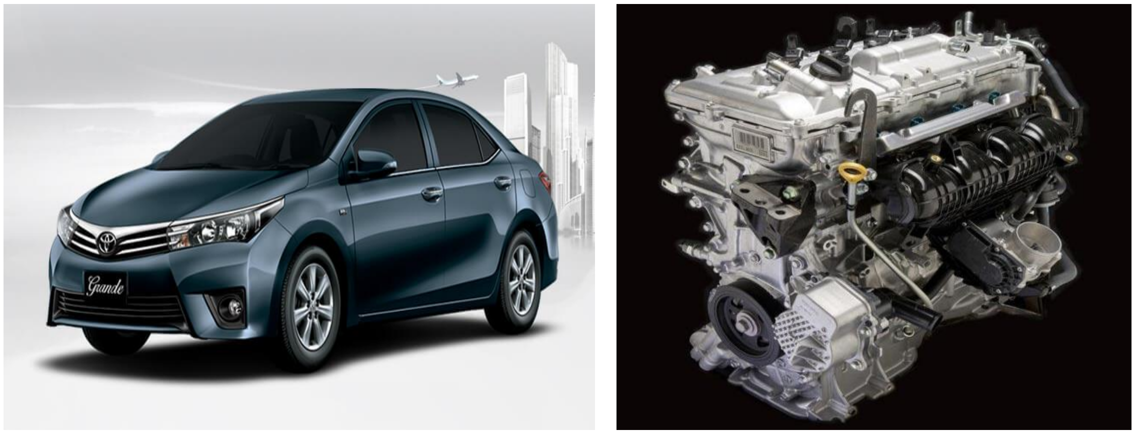
(c)

(d)

Figure 14: (a) Engine R06A (b) Engine K10B (c) Engine 2NZ-FE (d) Engine F8B

Table 1: Engine Data Specifications

Engines	R06A	F8B	K10B	2NZ-FE	2ZR-FE
No. of cylinders	3	3	4	4	4
Bore×Stroke(mm)	64×68.5	68.5×72	67×70	75×73.5	80.5×88.3
Displacement Vol. (L)	0.6	0.8	1	1.3	1.8
Compression ratio	11.5	11	9	10.5	10
No. of valves	12	6	16	16	16
Power (kW/rpm)	40/6000	29/5500	50/6200	63/6000	103/6400
Torque (Nm/rpm)		59/3000	90/3200	121/4000	173/4000
Fuel	Gasoline	Gasoline	Gasoline	Gasoline	Gasoline
Working Cycle	Natural	Natural	Natural	Natural	Natural



(a) (b)

Figure 15: (a) Toyota Grande (b) Engine 2ZR-FE

Summary

Internal combustion engine converts only 25-35% of fuel energy to the useful mechanical power while the rest of the fuel energy is wasted through the cooling water and exhaust gases to the environment. Several waste heat recovery technologies were introduced for the automotive vehicles. Turbocharger, turbocompounding, thermoelectric generator and Organic Rankine cycle are specifically used for the waste heat recovery in automotive vehicles to improve its fuel consumption. In Pakistan, automobile industry is fastest growing sector. Road transport covers almost 96% inland freight service while 92% of passenger traffic. Suzuki, Toyota and Honda are the major stakeholders in Pakistan's automobile industry. Suzuki WagonR, Suzuki Mehran, Suzuki Cultus and Toyota Corolla are the most running cars in Pakistan.

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Chapter 4

Engines Analysis and Methodology

This chapter covers the models and methodology followed for the energy and exergy analysis of engine and designing of the E-Turbine system. First the mathematical models will be discussed and then the MATLAB modelling will be studied in detail.

Heat Release Model

In the ideal cycles of internal combustion engines, it is assumed that the combustion takes place at constant volume (Otto cycle) or at constant pressure (Diesel cycle). But these idealizations do not match the actual engine pressure and temperature. The finite energy release model also called Wiebe Function is more realistic heat release model and address this issue. This is zero-dimensional model because it is a function of crank angle only. Spatial coordinates do not have any effect on it. This model study the effect of spark timing, duration of combustion on engine work and efficiency. The equation is given by

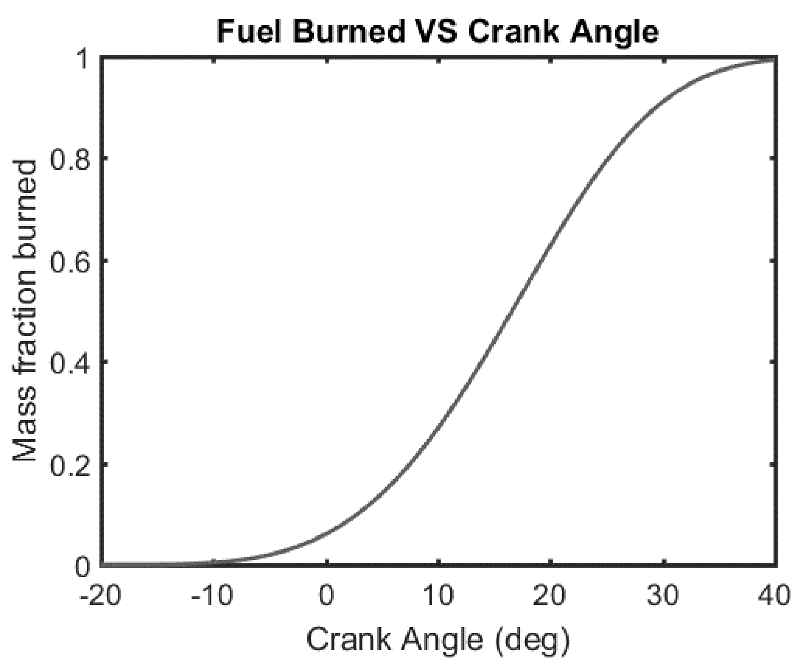

Figure 16: Mass fraction burned

$$x_b(\theta)=1-\exp\left[-a\left(\frac{\theta-\theta_s}{\theta_d}\right)^n\right]$$

x_b is the mass fraction of fuel burned, θ is the crank angle that varies between -180 to 180 degrees,$\theta_s$is the crank angle at which the start of ignition take place and it is usually -20 to -10 degrees for modern engines, $\theta_d$is the duration of combustion process varying between 60 to 70 degrees, a is the Wiebe efficiency factor, n is the Wiebe form factor. The S-shaped curve is obtained for the mass fraction of fuel burned.

Cylinder Volume Model

The volume of the cylinder is continuously changing due to oscillatory motion of the piston. When the piston is at top dead center, it forms the minimum cylinder volume. The cylinder volume goes on increasing when the piston moves from the TDC towards the BDC and reaches maximum when the piston reached to BDC. The instantaneous cylinder volume is measured by the equation given below

$$V_\theta=V_c+\frac{\pi}{4}B^2y$$

$$y=l+a-\left[(l^2-a^2\sin^2\theta)^{\frac{1}{2}}+a\cos\theta\right]$$

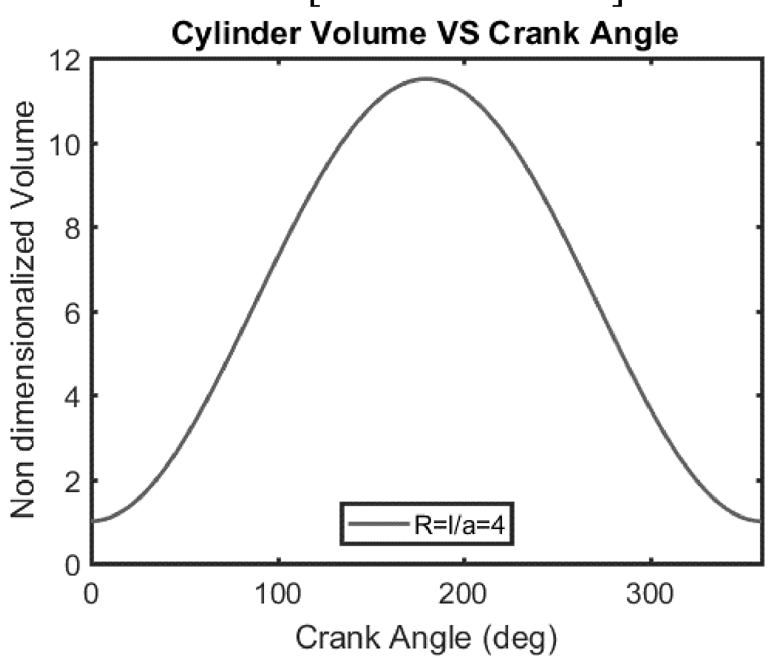

Figure 17: Cylinder volume variation w.r.t Crank angle

y is the instantaneous in-cylinder volume at any crank angle, V_c is the clearance volume, B is the bore of the cylinder, y is the instantaneous distance of the piston from the top dead center, l is the connecting rod length, a is the crank radius. The profile followed by the cylinder volume is plotted against crank angle. The cylinder volume first increases when the piston moves from TDC, reaches to maximum at BDC, then gradually decreases when the piston moves towards TDC.

In-Cylinder Pressure Model

The pressure inside the cylinder varies w.r.t to the crank angle due to certain processes. The pressure of the air-fuel mixture (SI Engine) or air (CI Engine) is gradually increases due to the compression stroke. The piston compresses the working fluid. The spontaneous increase in the cylinder pressure occur due to the combustion process. The pressure inside the cylinder then gradually decreases during the power stroke imparting power to crank shaft. The in-cylinder pressure variation against crank angle is given by the equation,

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

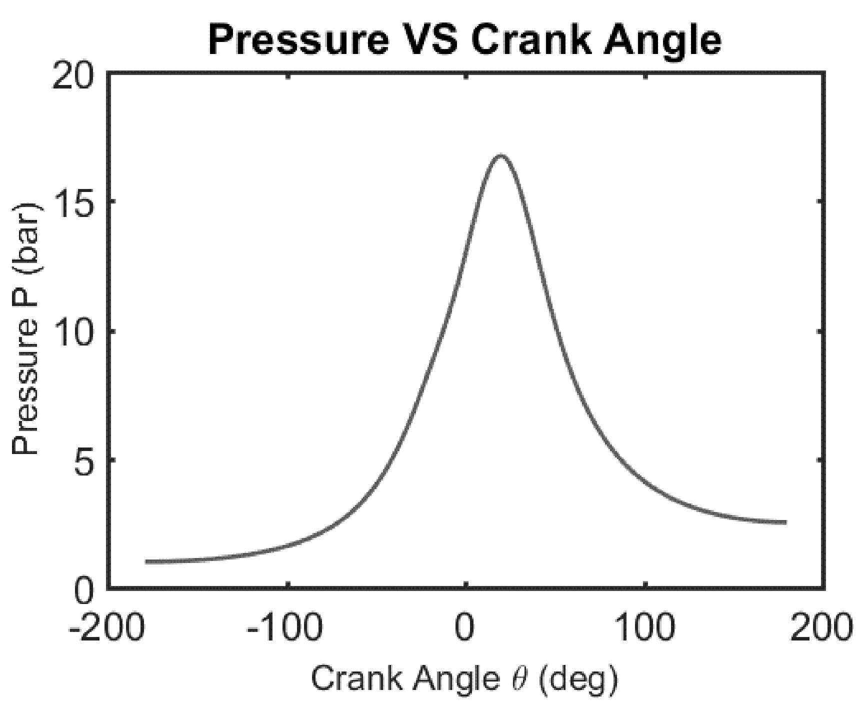

Figure 18: Pressure variation w.r.t Crank angle

$$\frac{dQ}{d\theta}=Q_{in}\frac{dx_b}{d\theta}$$

$$C=\frac{\dot{m}_1}{m}$$

P is the in-cylinder pressure, Q_{in} is the heat energy input, γ is the ratio of specific heats, *V* is the in-cylinder volume, ω is the rotational speed in radians, *C* is the mass blow by coefficient which is mass leaked by the piston rings etc. C value is 0.5% for new engine. The profile of in-cylinder pressure is plotted against the crank angle.

In-Cylinder Temperature Model

The in-cylinder temperature also varies w.r.t the crank angle due to certain processes. The temperature of the working fluid is slightly increasing with the compression. The combustion process abruptly increases the temperature of the working fluid. The working fluid is now the combustion gases and consists of various gases. The temperature of the combustion gases slightly decreases during the power stroke. The in-cylinder temperature model is given below,

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

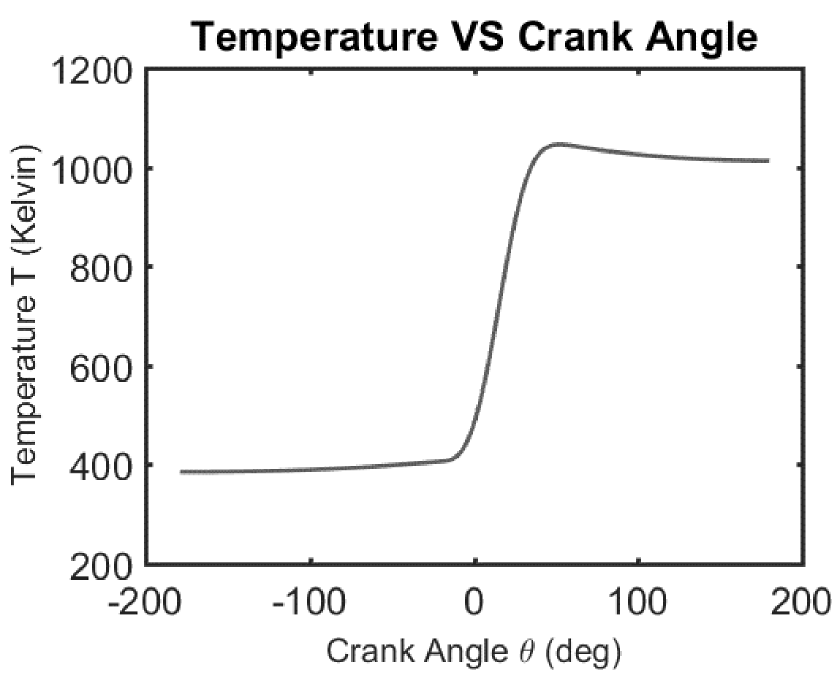

Figure 19: Temperature variation w.r.t Crank angle

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

T is the in-cylinder temperature of the combustion products. P is the in-cylinder pressure, V is the cylinder volume, θ is the crank angle. The in-cylinder temperature of the combustion products is plotted against the crank angle. The profile followed by the in-cylinder temperature is shown in the figure.

Indicated Work

The work produced in the cylinder of the engine after each thermodynamic cycle is called indicated work. The indicated work is then transfer to the shaft after pumping losses etc. It is given by the mathematical equation,

$$\frac{dW}{d\theta}=P\frac{dV}{d\theta}$$

W is the indicated work, P is the pressure inside the cylinder, V is the cylinder volume, and θ is the crank angle.

Brake Power

The power obtained at the crankshaft is called brake power. The brake power is the difference of indicated power and frictional losses. The brake power is less than the indicated power due to the pumping losses, mechanical friction etc.

$$\dot{W}_b= \dot{W}_i-\dot{W}_f$$

The indicated power is multiplied by the mechanical efficiency to obtain the brake power. Mechanical efficiency is the ratio of brake power to the indicated power.

$$\eta_{mech}=\frac{\dot{W}_b}{\dot{W}_i}$$

The brake increases with the rotational speed of the engine up to certain point then with the increase in rpm, the brake power decreases. It is due to the fact that the frictional losses exponentially increase at the higher rpm.

$$W_b=\frac{n_c W_i rps}{2}\eta_m$$

n_c is the number of cylinders in the engine, rps is the rotational speed of the engine per second. The equation is divided by two for the 4-stroke engine because in 4-stroke engine, the power stroke is delivered after 2 revolutions of the crank shaft. With greater number of cylinders, the brake power will be more.

Torque of the Engine

Torque is the measure of work done per unit rotation of the crank shaft. The maximum torque occurs at low rotational speed of the engine than the maximum brake power. The equation to calculate the torque is given by,

$$\dot{W}_b=2\pi\tau \text{ rps}$$

τ is the torque produced by the engine.

MATLAB Model

Engines R06A, K10B and 2NZ-FE having displacement volume of 0.6L, 1L and 1.3L, are modelled in MATLAB. Zero dimensional model is generated for the study. It is called zero-dimensional model because it is the function of only crank angle of the crank shaft. No spatial coordinates are considered. ODE23 function is used for solving the multiple differential equations. It is MATLAB built-in function for solving multiple differential equations simultaneously. Displacement volume V_d, clearance volume V_c, maximum volume V_{max}, compression ratio r_c are the required geometrical inputs on which the performance of an engine depends. Initial conditions required for the MATLAB model to solve the differential equations are gamma Υ, omega ω, rotational speed of engine (rpm), spark ignition timing θ_s, duration of combustion θ_d. The MATLAB model calculates the required result at each and every crank angle i.e. -180 degree to 180 degree crank angle. To consider the effect of the combustion process i.e. spark timing, duration of combustion, "for loop" is used for the Wiebe function for $-20\leq\theta\leq60$. The model first calculate the in-cylinder pressure and temperature w.r.t crank angle. Then indicated work, brake power, torque are calculated. Flow chart of the MATLAB model is given.

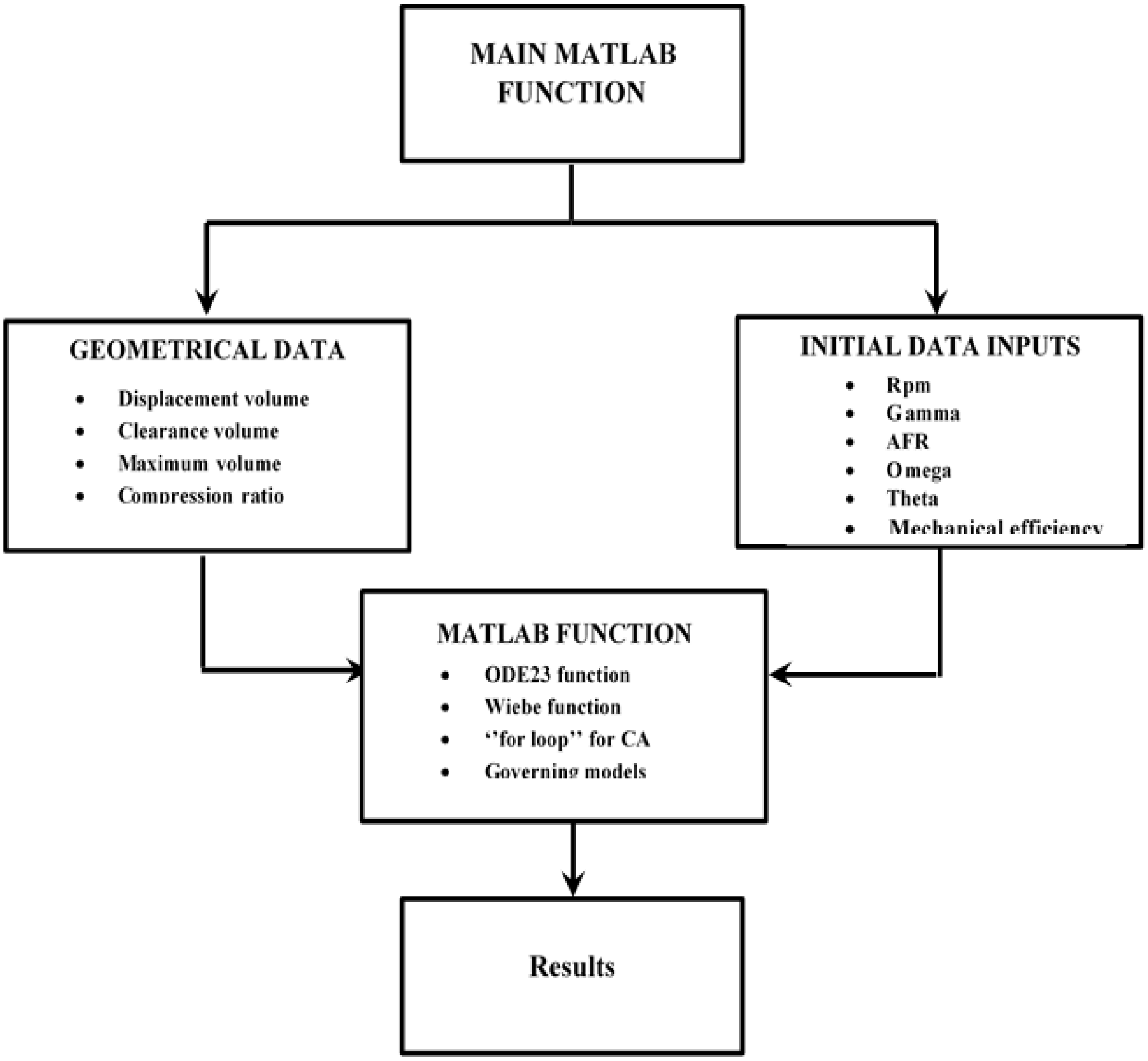

Figure 20: Flowchart of MATLAB Model

Validation of MATLAB Model

The MATLAB model used for results given in this thesis is validated by the published data. Zhijun Peng et al studied 1.4L turbocharged engine and plotted the distribution of exhaust gases temperature as a function of torque and rotational speed [1]. The figure shows that at full load operation, the temperature of exhaust gases increases with increasing torque. The exhaust temperature reaches to above 1000K at full load. The exhaust temperature for naturally aspirated engine is higher than the turbocharged engine.

This engine is then modelled in MATLAB model used for the results in this paper. The torque is plotted against the rpm at full load. The MATLAB model gives the result in which the maximum torque occurs at 4000 rpm while in the published data, the maximum torque occurs at 3500 rpm which are quite close to each other. The dotted blue line shows the experimental result while the dotted black line shows the result of MATLAB model.

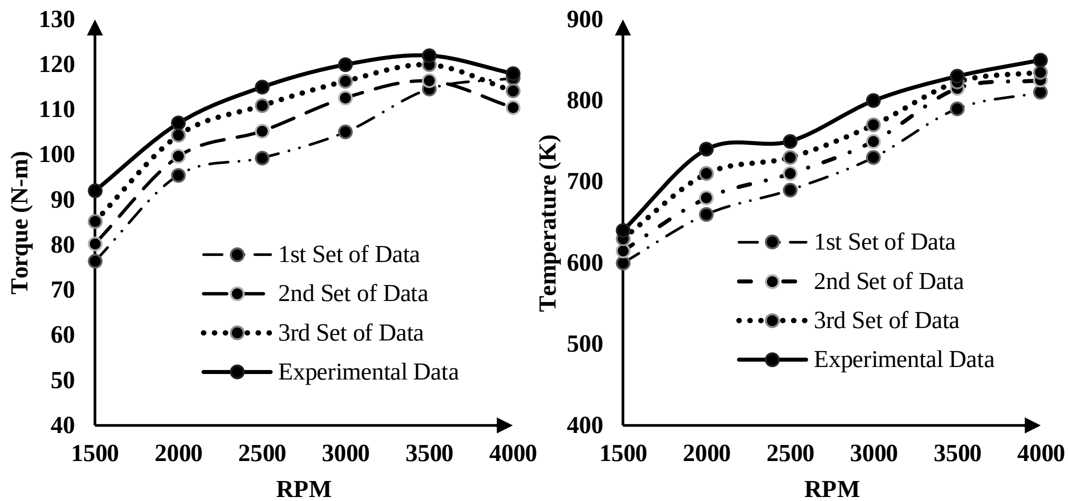

Figure 21: Validation of MATLAB Model

Therefore, the MATLAB model used for the study is validated. Distribution of exhaust gases temperature followed the same pattern as in the published data. The engine torque increases with increasing rpm. The maximum torque occur at around 3500 rpm after this the torque then decreases.

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Chapter 5
Results and Discussion

The results obtained from the MATLAB model are presented in this chapter. Brake power, torque, mas flow rate of exhaust gases and exhaust energy is obtained from the model. Exergy analysis of engine RO6A, K10B and 2NZ-FE is done by using the mathematical model. After that turbine selection is done from the NsDs diagram. The diameter and rotational speed is calculated from the model. High speed generator is selected.

Engine R06A

Engine R06A having the displacement volume of 0.6 L has been modelled in MATLAB. Brake power, mass flow rate of exhaust gases, and exhaust energy are plotted against the rotational speed of engine. All the results are plotted when the engine operating at full load.

With the increase in rotational speed per minute, brake power increases reaching its maximum value of 50 kW at 6000 rpm. The brake power starts declining with further increase in rpm after 6000 rpm. This is due to the fact that frictional losses increase exponentially with the rotational speed. The temperature of exhaust gases increases with the brake power and rpm. The temperature of exhaust gases is 860 K at 6000 rpm.

Mass flow rate of the exhaust gases are calculated by using law of conservation of mass. According to the law, mass flow in is equal to mass flow out.

$$\dot{m}_{in} = \dot{m}_{out}$$

The power is calculated by the MATLAB model. Mass flow rate of fuel is calculated by the equations,

$$\eta_{th} = \frac{\dot{W}_b}{\dot{m}_{fuel} \times Q_{in}}$$

$$\dot{m}_{fuel} = \frac{\dot{W}_b}{Q_{in} \times \eta_{th}}$$

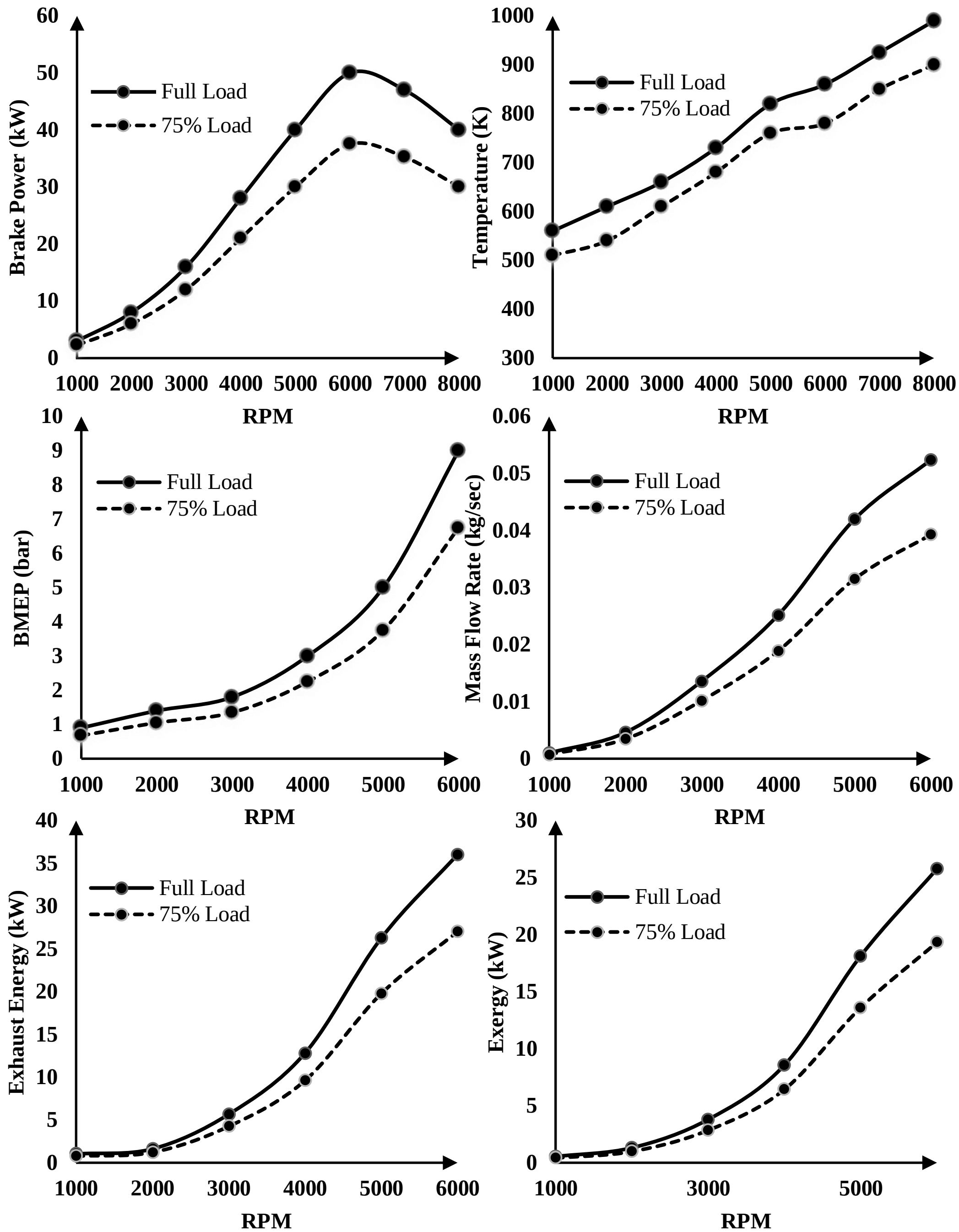

Figure 22: Performance Results of Engine R06A

Air flow is calculated from air to fuel ratio by the following equation,

$$AFR=\frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

Total flow is the summation of air flow and fuel flow.

$$\dot{m}_{exh} = \dot{m}_{air} + \dot{m}_{fuel}$$

The pattern followed by mass flow rate and exhaust energy are shown in the figure. Mass flow rate and exhaust energy at 6000 rpm are 0.0524 kg/s and 36.02 kW respectively. A huge amount of energy is wasted through the exhaust gases to the environment.

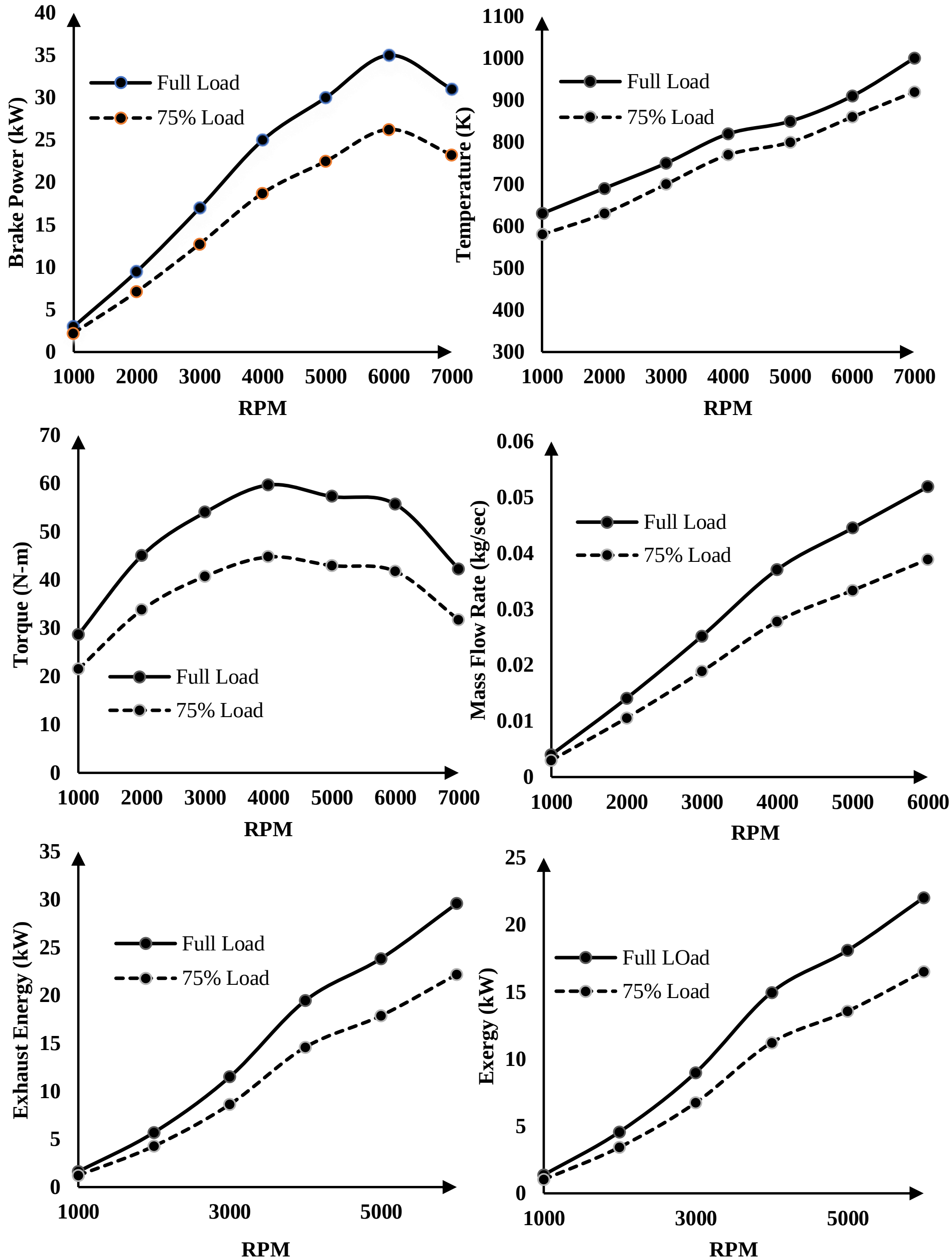

Figure 23: Performance Results of Engine F8B

Engine F8B

Engine F8B is an inline-three OHC-6 valves internal combustion engine having displacement volume of 0.8L. These engines are installed in Suzuki Mehran and Maruti

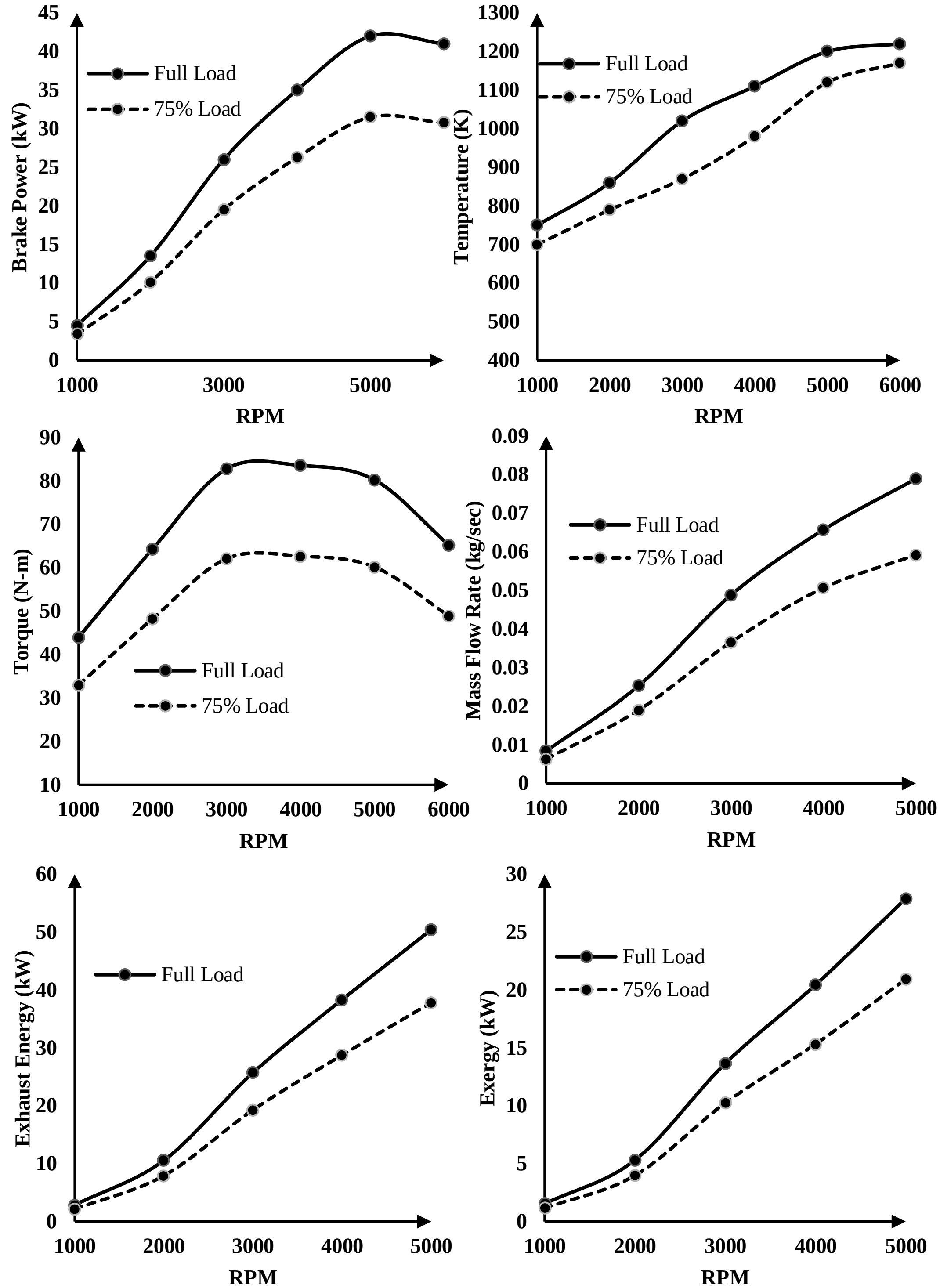

Figure 24: Performance Results of Engine K10B

vehicles. This engine has been modelled in MATLAB. Brake power, torque, mass flow rate and exhaust energy are plotted, and the results are shown in the figures.

Engine K10B

The MATLAB model of K10B having displacement volume of 1L is generated to study its performance at different rotational speed of the engine. Brake power, torque, mass flow rate of exhaust gases and exhaust energy are plotted against rotational speed of engine. The maximum brake power occurs at 5500 rpm having value of 45 kW. With further increase in rotational speed of engine, the brake power starts declining due to frictional losses. The maximum torque occurs at low rotational speed of engine than the maximum brake power. The value of maximum torque is 87.30 N-m at 3500 rpm. The distribution of temperature of exhaust gases as a function of brake power and rpm is shown. Temperature of exhaust gases at 5500 rpm is 1210 K having high energy content.

Mass flow rate and energy of exhaust gases increases with rotational speed of the engine at full load. Mass flow rate of the exhaust gases is calculated from the law of conservation of mass as stated above. Mass flow rate of exhaust gases at 5500 rpm is 0.0845 kg/s. The flow rate of exhaust gases increases with increasing brake power due to the fact that high flow rate of fuel will be required to produce high brake power. By law of conservation of mass, high flow rate into the cylinder will result in high flow rate of exhaust gases. The energy of exhaust gases is more than the energy of exhaust gases of engine having displacement volume of 0.6 L. The exhaust energy is 54.07 kW at 5500 rpm. This is the huge amount of energy that is wasted through the exhaust gases. The exhaust energy is plotted against the rotational speed of engine. The figure shows that exhaust energy increases with increasing the rotational speed and brake power of the engine.

Engine 2NZ-FE

2NZ-FE engine having displacement volume of 1.3 L is modelled in MATLAB. The model predicts the performance of the engine and plotted the results against the rotational speed of the engine. Maximum torque occurs at 4000 rpm having value of 119.3662 N-m. Maximum brake power occur at 6000 rpm having value of 65 kW. At 6000 rpm, 1130 K is the temperature of exhaust gases. Mass flow rate and exhaust energy at 6000 rpm are 0.1156 kg/s and 70.227 kW respectively as shown in figure.

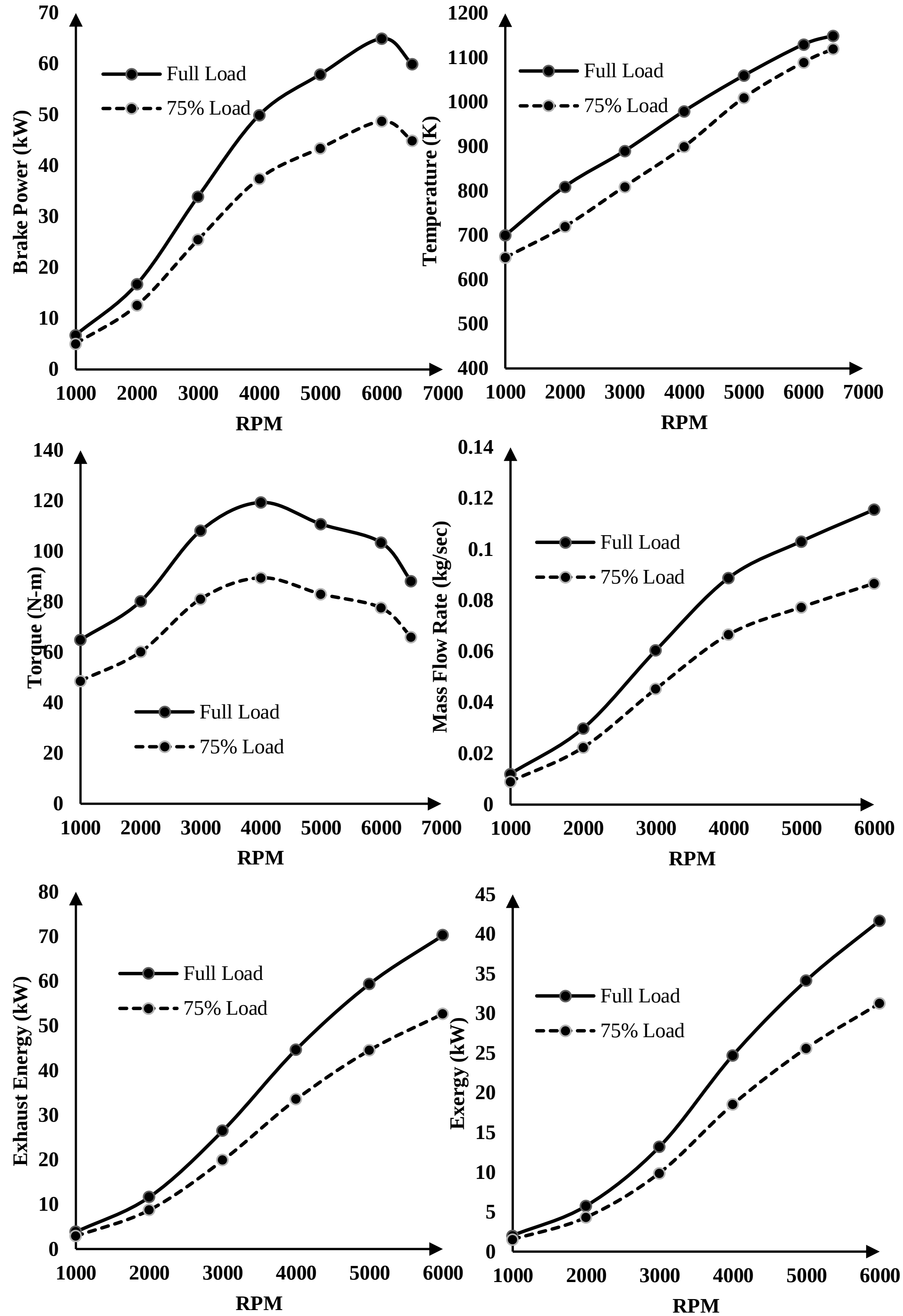

Figure 25: Performance Results of Engine 2NZ-FE

Engine 2ZR-FE

Engine 2ZR-FE is an inline-four DOHC-16 valves internal combustion engine having displacement volume of 1.8L. These engines are installed in Toyota Grande and Premio vehicles. This engine has been modelled in MATLAB. Brake power, torque, mass flow rate and exhaust energy are plotted, and the results are shown in the figures.

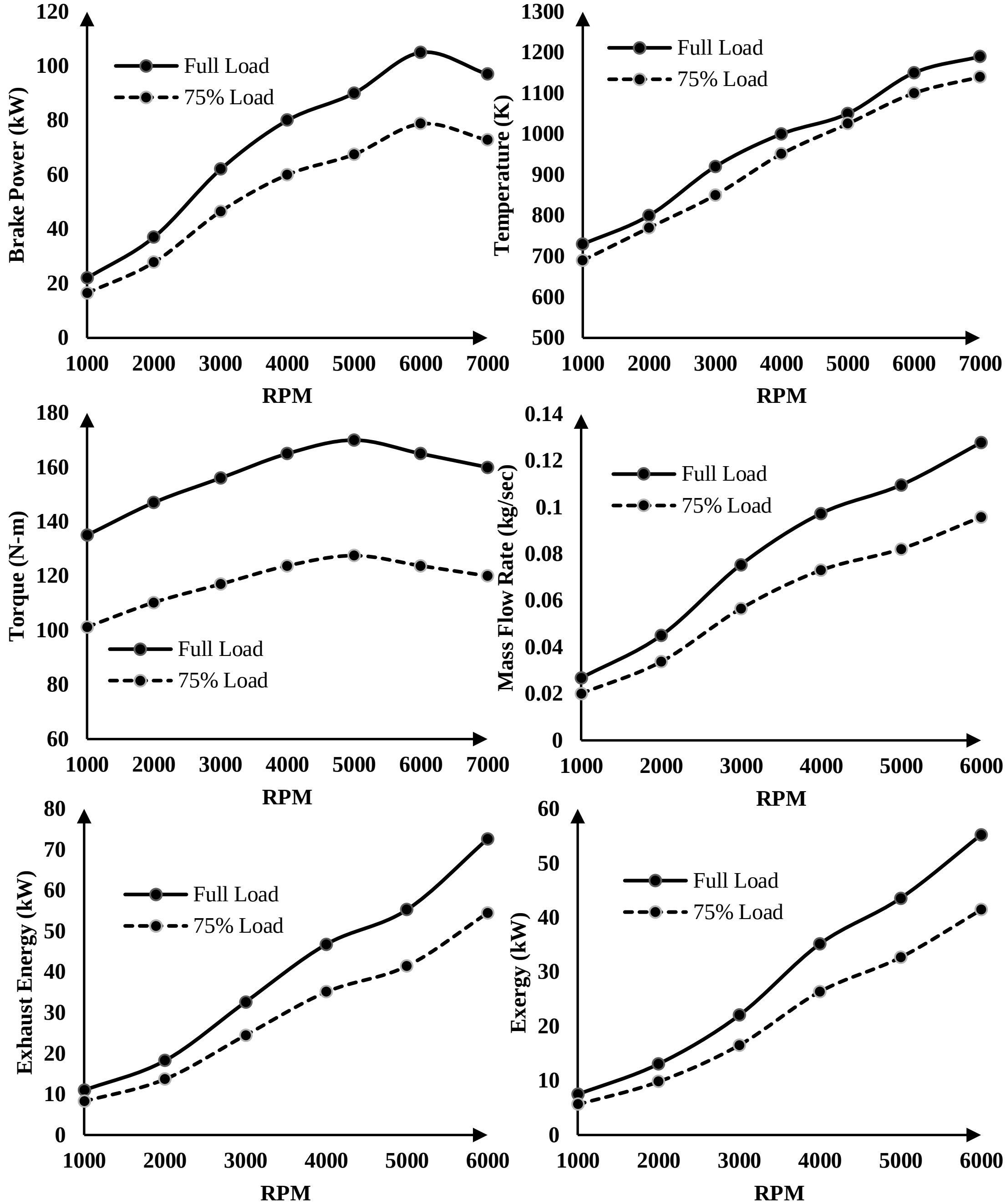

Figure 26:Performance Results of Engine 2ZR-FE

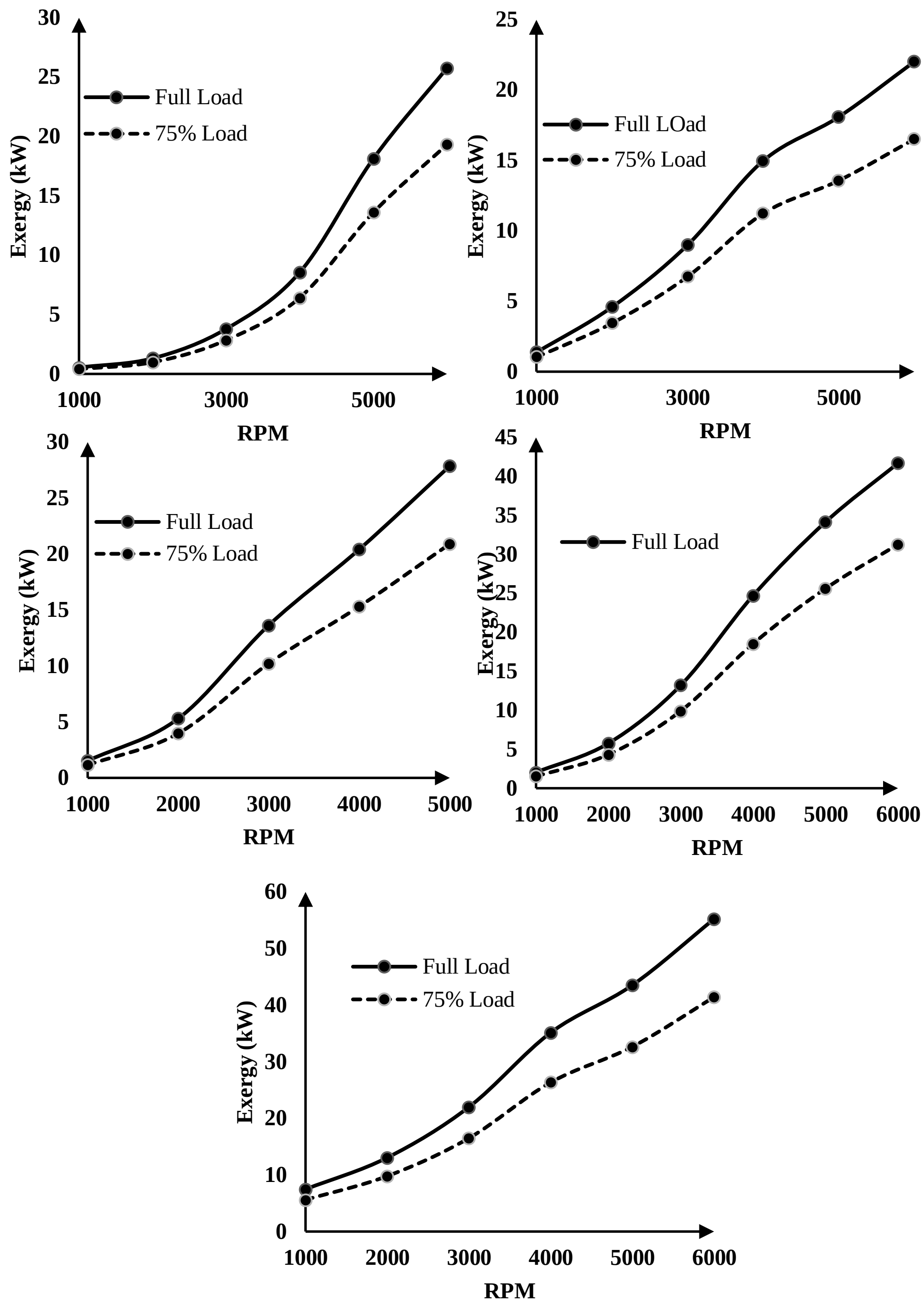

Figure 27: Exergy of Engine (a) R06A (b) F8B (c) K10B (d) 2NZ-FE (e) 2ZR-FE

Exergy Analysis

Exergy analysis of the engines R06A, F8B, K10B and 2NZ-FE is done in the study. The temperature of exhaust gases is high enough having huge amount of energy. This energy is wasted to the environment. All the exhaust energy cannot be recovered to useful work, some energy should be given to the environment. The useful energy that can be recover is called exergy or availability. According to 2nd law of thermodynamics, no system can produce the net work output by transferring the heat with single reservoir. The equation used for the exergy analysis of exhaust gases is given below

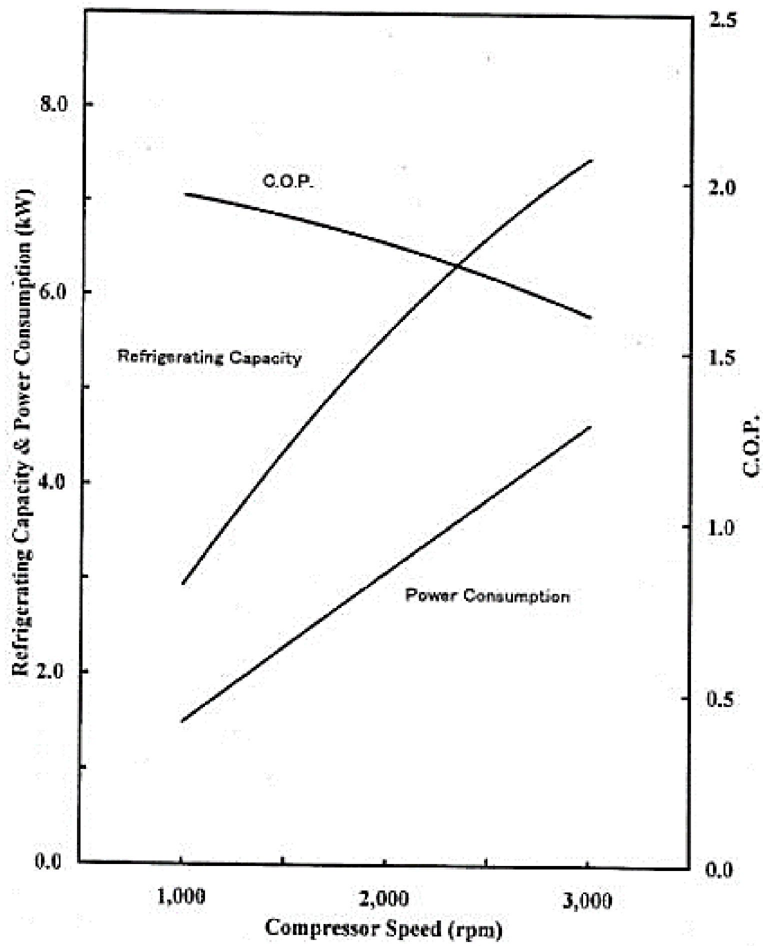

Figure 28: Performance map of SD7H15 Compressor

$$A_{exh}=Q_{exh}+\frac{(m_{exh})T_{amb}\left(C_{pexh}\times Ln\left(\frac{T_{amb}}{T_{exh}}\right)-R_{exh}\times Ln\left(\frac{P_{amb}}{P_{exh}}\right)\right)}{3600}$$

A_{exh} is the available energy, Q_{exh} is the exhaust energy, m_{exh} is the mass flow rate of exhaust gases, T_{amb} is ambient temperature, C_{pexh} is the specific heat of exhaust gases at constant pressure, T_{exh} is the temperature of exhaust gases, R_{exh} is the gas constant of exhaust gases. The exergy of the stated engines is shown in the figure.

HVAC Compressor

Sanden AC compressors are mostly installed in the vehicles of Pakistan. SD7 series and SD5 series are nowadays in use. SD7H15 is installed in the vehicles that are selected for study. The performance map of the compressor is shown in the figure. The map shows that at 3000 rpm, the compressor has its maximum refrigerating capacity having the maximum power consumption of 5 kW. It means that this compressor would draw 6-7 kW power from the vehicle engine when operating at maximum refrigerating capacity.

Design Point and Turbine Selection

1000 cm^3 engine is selected for the design point and turbine selection. At 3500 rpm, the available energy in the exhaust gases is 13.64 kW, having a mass flow rate of 0.06kg/s but the turbine is designed for 0.04kg/s to take care of the size of turbine, which is more than enough to drive the compressor after the losses that would be taking place in turbine so 3500 rpm is selected as the design point. Ns Ds diagram is used for the selection of turbine. Radial turbine is selected against Ns=0.42, Ds=2, η=0.77 from the diagram [1]. The diameter of the radial turbine is 0.061m having blade tip speed of 500 m/s after calculation. The equations used for the calculations are given below,

$$N_s=N\left(\frac{\dot{m}}{\rho_e}\right)^{\frac{1}{2}}\times(\Delta h_{0s})^{\frac{-3}{4}}$$

$$\Delta h_{0s}=C_pT_{01}\left[1-\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$D_s=\frac{D\left(\frac{\Delta p}{\rho}\right)^{\frac{1}{4}}}{Q^{\frac{1}{2}}}$$

Δh_{0s} is the isentropic expansion that will take place in the turbine, $\frac{p_{02}}{p_{01}}$ is the pressure ratio against the turbine, ρ_e is the density of exhaust gases, Ns is the specific speed, N is the rotational speed of turbine, Ds is the specific diameter and D is the diameter of turbine. Specific speed, specific diameter and efficiency are chosen for the selection of turbine. Radial turbine is selected against these values. The shaft power produced by the turbine is 6.8 kW, calculated by the equation

$$\mathrm{P} = \dot{\mathrm{m}} \times \Delta \mathrm{h}_{0\mathrm{s}}$$

Selection of High-Speed Generator

Permanent magnet synchronous generator is selected for the study [2]. The PMSG design is simple because it does not have the field excitation circuit which is very difficult to

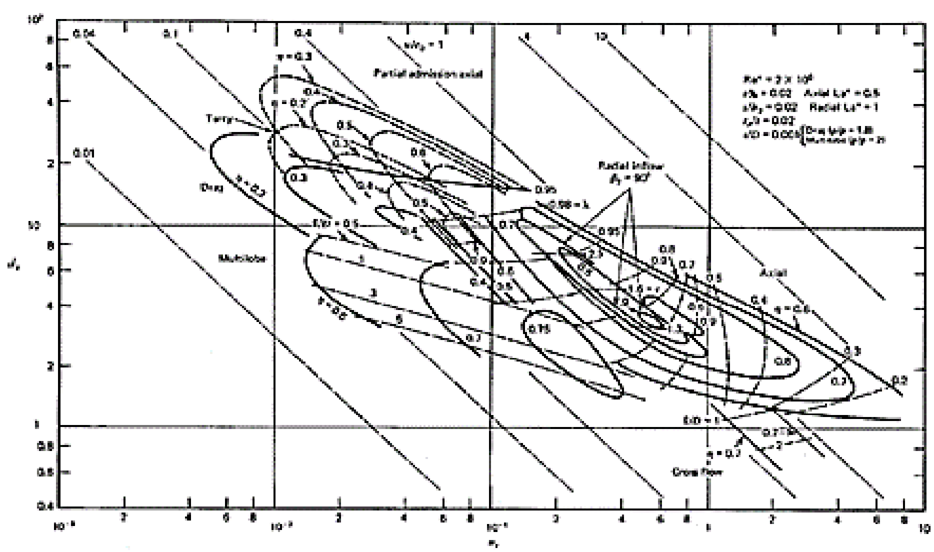

Figure 29: NsDs Diagram of Turbines (Balji diagram)

design. Mechanical design includes the air gap which is 1.5 mm, and the optimum shaft diameter is 15 mm. The stator has 2 poles and 6 slots.

Battery Sizing Model

Battery is an electrical equipment used to store electrical charge and transfer it for further use. Battery sizing has an important role in designing the whole E-Turbine system. The electrical energy produced by the high-speed generator should be stored in the battery to drive the electrical accessories of the vehicle. Therefore, this step is very important for the optimum design of the system.

An electrical AC compressor of automotive vehicles operates on 12-volt and draws 60 amperes current. The energy needs to drive the electrical AC compressor is 60 Ah. Other electrical accessories installed in the vehicle are assuming to draw 30 Ah of electrical energy. Therefore, the need total energy from the battery is 90 Ah. A 12-volt battery having energy storage capacity of 90 Ah is chosen for the study. State of charge (SOC) is selected as 20% which means that when 20% of the battery energy is left in the battery, recharging of the battery will commence.

$$E_{20\%} = E_B \times 20\%$$

Where E_B is the total battery capacity. Energy left in the battery at SOC is calculated by

$$E_{20\%} = 90 \times 20\%$$

$$E_{20\%} = 18\,Ah$$

It means that the battery will not discharge further when 18 Ah energy is left and recharging of the battery will start. Another important term used in the batteries is depth of discharge (DOD) given by the manufacturers to the customer. If the DOD is 80% then the battery will supply 80% of rated battery capacity, more than that is not possible. The total energy required to completely recharge the battery will be

$$E_{100\%} = E_B \times 80\%$$

$$E_{100\%} = 90\,Ah \times 0.80$$

$$E_{100\%} = 72\,Ah$$

To covert the amp-hours to kWh, multiply the rated voltage of the battery to the energy capacity.

$$12\,volts \times 72\,Ah = 864\,Wh$$

$$864\frac{Wh}{1000} = 0.864\,kWh$$

The high-speed generator produced 4 kW of electrical power and this power is transferred to the battery. The time required to completely recharge the battery is given by

$$t = \frac{E_{100\%}}{electric\ power}$$

$$t = \frac{0.864\ kWh}{4\ kW\ \times 0.80}$$

$$t = 17\ mins$$

The unit cell voltage is 3.4 volts then the total number of cells are calculated by

$$N_c = \frac{rated\ voltage}{unit\ cell\ voltage}$$

$$N_c = 4 \text{ cells}$$

Number of series in parallel is given by

$$I_b = I_c \times N_s$$

Where $I_c = Q_c \times c_r$

To find each cell energy capacity, divide the total energy capacity by number of cells.

$$Q_c = \frac{total\ battery\ capacity}{total\ number\ of\ cells}$$

Battery Sizing Calculations	
Rated voltage of battery	12 volts
Storage capacity of battery	90 Ah
Energy of battery	0.864kWh
SOC	20%
DOD	80%
Time for recharge (hrs.)	0.21
Time for recharge (mins)	13
Number of cells in series	4
Number of series in parallel	15
Total number of cells	60
Capacity of cell	1.5
Weight of one cell (g)	54
Weight of battery (kg)	3.24

Stoichiometry

Stoichiometry is the study of combustion process to analyze the reactants and products during the process. Stoichiometric reaction is one in which the fuel is completely burned during combustion resulting the products of H_2O and CO_2. This reaction is used as reference case. The composition of products during combustion of fuel-air mixture depends upon stoichiometry i.e. fuel-rich mixture or fuel-lean mixture. Before going to study these cases in detail, it is necessary to define the important term i.e. fuel-air equivalence ratio.

Fuel- Air Equivalence Ratio

The ratio of actual fuel-air ratio to stoichiometric fuel-air ratio is called fuel-air equivalence ratio. It is used to define that either the mixture is fuel-rich or fuel-lean. It is represented by φ.

$$\boldsymbol{\varphi = \frac{F/A_{actual}}{F/A_{stoic}}}$$

Three cases: If $\varphi < 1$, then the mixture is fuel-lean.

If $\varphi = 1$, then the mixture is stoichiometric.

If $\varphi > 1$, then the mixture is fuel-rich.

Automotive sector of Pakistan has been studied in this thesis, so gasoline (RON-92) is selected as fuel in the following study. All the three cases are discussed in the following section.

Stoichiometric reaction

Stoichiometric reaction is one in which the fuel is completely burned in combustion process and the products are only H_2O and CO_2. This reaction is used as a reference case. The fuel is selected as gasoline (RON-92) C_7H_{17} having a density of 737 kg/m^3. Air act as oxidizing agent. Air is mainly consisting of oxygen and nitrogen. The molar analysis shows that for one mole of O_2 there will be 3.76 moles of N_2 available. The combustion reaction will be

$$\mathbf{C_7H_{17} + v_0\,(O_2 + 3.76\,N_2) \longrightarrow aCO_2 + bH_2O + cN_2 + dO_2 + eCO + fH_2}$$

For the stoichiometric combustion process, the products will only be CO_2, H_2O and N_2 and the coefficients d, e, f will be zero. The new chemical reaction will be

$$\mathbf{C_7H_{17} + v_0\,(O_2 + 3.76\,N_2) \longrightarrow aCO_2 + bH_2O + cN_2}$$

To find out the coefficients v_0, a, b and c, comparing coefficients technique is used.

For **C** $\longrightarrow$ $\mathbf{a=7}$

For **H** $\longrightarrow$ $\mathbf{2b=17}$

$\mathbf{b= 8.5}$

For **O** $\longrightarrow$ $\mathbf{2v_0= 2a + b}$

$\mathbf{v_0 =11.25}$

For **N** $\longrightarrow$ $\mathbf{2(3.76)\ (v_0) = 2c}$

$\mathbf{c= 42.3}$

so, the balanced chemical reaction for the stoichiometric combustion process is given below

$$\mathbf{C_7H_{17} + 11.25\ (O_2 + 3.76\ N_2) \longrightarrow 7CO_2 + 8.5H_2O + 42.3N_2}$$

The stoichiometric air-fuel ratio on molar basis is given by

$$\frac{A}{F}\Big|_{\text{stoich,molar}} = \frac{\text{moles of air}}{\text{moles of fuel}}$$

$$\frac{A}{F}\Big|_{\text{stoich,molar}} = \frac{v_0(1 + 3.76)}{1}$$

$$\frac{A}{F}\Big|_{\text{stoich,molar}} = 53.55$$

To convert the molar basis air-fuel ratio to the mass-based air-fuel ratio, following steps are done.

$$\frac{A}{F}\Big|_{\text{stoich,mass}} = \frac{A}{F}\Big|_{\text{stoich,molar}} \left(\frac{\text{mass of air}}{\text{mass of fuel}}\right)$$

$$\frac{A}{F}\Big|_{\text{stoich,mass}} = 53.55 \left(\frac{28.97}{101.21}\right)$$

$$\frac{A}{F}\Big|_{\text{stoich,mass}} = 15.32$$

The stoichiometric air-fuel ratio for gasoline spark-ignition engine is 15.32. The fuel-air equivalence ratio is 1.

Excess Air Combustion/ Fuel-lean mixture

Fuel- lean mixture means that there is excess air for the given amount of fuel taking part in the combustion process. The fuel is completely burned in the process but still there is some air which did not take part in the process. The products resulting from the fuel-lean mixture combustion are CO_2, H_2O, N_2 and O_2. The fuel-air equivalence ration for this type of process is less than 1. The chemical reaction for this process will be,

$$\mathbf{C_7H_{17} + (PTA)\ v_0\ (O_2 + 3.76\ N_2) = aCO_2 + bH_2O + cN_2 + dO_2}$$

PTA is the percent theoretical air present inside the cylinder at the time of combustion. The analysis has been done for the 150% theoretical air. Then the chemical reaction takes the form,

$$\mathbf{C_7H_{17} + (1.5)\ (11.25)\ (O_2 + 3.76\ N_2) = aCO_2 + bH_2O + cN_2 + dO_2}$$

To find the unknown coefficients a, b, c, d, comparing coefficients technique is used.

For **C** $\longrightarrow$ **a=7**

For **H** $\longrightarrow$ **2b=17**

b= 8.5

For **O** $\longrightarrow$ **2 (1.5) (11.25) = 2a + b + 2d**

d = 5.625

For **N** $\longrightarrow$ **2 (3.76) (1.5) (11.25) = 2c**

c = 63.45

the balanced chemical reaction for 150% theoretical excess air is given,

$$\mathbf{C_7H_{17} + (1.5)\ (11.25)\ (O_2 + 3.76\ N_2) = 7CO_2 + 8.5H_2O + 63.45N_2 + 5.625O_2}$$

Air-fuel ratio on molar basis is find out by,

$$\frac{A}{F}\Big|_{actual,molar} = \frac{\text{moles of air}}{\text{moles of fuel}}$$

$$\frac{A}{F}\Big|_{actual,molar} = \frac{(1.5)\ v_0(1 + 3.76)}{1}$$

$$\frac{A}{F}|_{actual,molar} = 80.325$$

To convert the molar basis air-fuel ratio to the mass-based air-fuel ratio, following steps are done.

$$\frac{A}{F}|_{actual,mass} = \frac{A}{F}|_{actual,molar}\left(\frac{\text{mass of air}}{\text{mass of fuel}}\right)$$

$$\frac{A}{F}|_{actual,mass} = 80.325\left(\frac{28.97}{101.21}\right)$$

$$\frac{A}{F}|_{actual,mass} = 22.9$$

The fuel-air equivalence ratio is calculated by

$$\varphi = \frac{F/A_{actual}}{F/A_{stoic}}$$

$$\varphi = 0.7$$

as the fuel-air equivalence ratio is less than 1, which shows that the mixture is fuel-lean. The oxygen appearing in the products also shows that there is an excess air present during the combustion process.

Fuel-rich Combustion

Fuel-rich mixture means that there is an excess amount of fuel than air during combustion process. In other words, there is not enough oxygen available in the process to oxidize the fuel completely. Therefore, in the combustion products, CO is also present because there is not enough oxygen to completely convert it into the CO_2. The chemical reaction for this process will be,

$$\mathbf{C_7H_{17} + (PTA)\ v_0\ (O_2 + 3.76\ N_2) = aCO_2 + bH_2O + cN_2 + dO_2 + eCO + fH_2}$$

In this study the engines are operated at its full load, therefore the mixture used for the combustion is fuel-rich. The fuel-air equivalence ratio is assumed to be 1.4, then to find the PTA the following steps are carried out,

$$\Phi = 1.4$$

$$\varphi = \frac{F/A_{actual}}{F/A_{stoic}}$$

$$1.4 = \frac{F/A_{actual}}{0.065}$$

$$F/A_{actual} = 0.091$$

$$A/F_{actual} = 10.98$$

$$A/F_{actual,mass} = \frac{A}{F_{actual,molar}} \left(\frac{M_{air}}{M_{fuel}}\right)$$

$$\frac{A}{F_{actual,molar}} = 38.359$$

$$\frac{A}{F_{actual,molar}} = \frac{\text{moles of air}}{\text{moles of fuel}}$$

$$38.35 = (PTA)\ (11.25)(4.76)/1$$

PTA = 0.71 or 71%

The chemical reaction takes the following form,

$$\mathbf{C_7H_{17} + (0.71)\ (11.25)\ (O_2 + 3.76\ N_2) \longrightarrow aCO_2 + bH_2O + cN_2 + dO_2 + eCO + fH_2}$$

The unknown coefficients in the above reaction cannot be find by simple comparing coefficients technique because during the combustion of fuel-rich mixture, carbon monoxide and hydrogen are additional products. The oxygen can be neglected. The product composition for fuel-rich mixture cannot be determined by an element balance alone, an additional assumption should be made for the species of the products.

The combustion equation used to determine the composition of the products is given below,

$$\boldsymbol{\epsilon\ \varphi\ C + 2(1 - \epsilon)\ \varphi\ H_2 + O_2 + \psi\ N_2 \longrightarrow n_i\ CO_2 + n_{ii}\ H_2O + n_{iii}\ CO + n_{iv}\ H_2 + n_v\ N_2}$$

where $\epsilon = \frac{4}{4+y}$

$y = \frac{H}{C}$ molar ratio

Φ = fuel-air equivalence ratio

Ψ = molar N/O ratio= 3.76 for air

The mole fractions for the reaction are determined from the table given below,

Species	**No. of moles/moles of O_2**
CO_2	$\epsilon\varphi - c$
H_2O	$2(1 - \epsilon\varphi) + c$
CO	C
H_2	$2(\varphi - 1) - c$
O_2	0
N_2	Ψ

The unknown c is be determined by solving the quadratic equation given below,

$$\mathbf{(K - 1)\, c^2 - c\, \{K\, [2\, (\varphi - 1) + \epsilon\varphi] + 2\, (1 - \epsilon\varphi)\} + 2\, K\, \epsilon\varphi\, (\varphi - 1) = 0}$$

Where K is called equilibrium constant and can be determined from the equation given by JANAF Table.

$$\mathbf{\ln K(T) = 2.743 - \frac{1.761\times10^3}{T} - \frac{1.611\times10^6}{T^2} + \frac{0.2803\times10^9}{T^3}}$$

where T is in Kelvin. The value of K depends upon temperature of the product. Therefore, the product composition is determined for different engines separately.

Engine R06A

Engine R06A is inline-three 12-valve engine having displacement volume of 0.6L. The product composition has been determined from the above equations.

1. For temperature = 550 K, rpm = 1000 at full load

$$\mathbf{K = 0.0166}$$

$$\mathbf{\epsilon = 0.622}$$

$$\mathbf{\varphi = 1.4}$$

c = 0.0360

yCO_2 = 0.8348

yH_2O = 0.2944

yCO = 0.0360

yN_2 = 3.76

2. For temperature = 610 K, rpm = 2000 at full load

K = 0.0392

ϵ = 0.622

φ = 1.4

c = 0.0698

yCO_2 = 0.8010

yH_2O = 0.3282

yCO = 0.0698

yN_2 = 3.76

3. For temperature = 660 K, rpm= 3000 at full load

K = 0.0708

ϵ = 0.622

φ = 1.4

c = 0.1042

yCO_2 = 0.7666

yH_2O = 0.3626

yCO = 0.1042

yN_2 = 3.76

4. For temperature = 730 K, rpm= 4000 at full load

K = 0.1392

ϵ = 0.622

φ = 1.4

c = 0.1553

yCO_2 = 0.7155

yH_2O = 0.4137

yCO = 0.1553

yN_2 = 3.76

5. For temperature = 820 K, rpm= 5000 at full load

K = 0.2747

ϵ = 0.622

φ = 1.4

c = 0.2185

yCO_2 = 0.6523

yH_2O = 0.4769

yCO = 0.2185

yN_2 = 3.76

6. For temperature = 860 K, rpm= 6000 at full load

K = 0.3527

ϵ = 0.622

φ = 1.4

c = 0.2443

yCO_2 = 0.6265

yH_2O = 0.5027

$$yCO = 0.2443$$

$$yN_2 = 3.76$$

Engine K10B

Engine K10B is inline-four cylinder, 16-valved internal combustion engine having displacement volume of 1L. The product composition is calculated and given below,

1. For temperature = 750 K, rpm= 1000 at full load

$$K = 0.1645$$

$$\epsilon = 0.622$$

$$\varphi = 1.4$$

$$c = 0.1698$$

$$yCO_2 = 0.7010$$

$$yH_2O = 0.4282$$

$$yCO = 0.1698$$

$$yN_2 = 3.76$$

2. For temperature = 860 K, rpm = 2000 at full load

$$K = 0.3527$$

$$\epsilon = 0.622$$

$$\varphi = 1.4$$

$$c = 0.2443$$

$$yCO_2 = 0.6265$$

$$yH_2O = 0.5027$$

$$yCO = 0.2443$$

$$yN_2 = 3.76$$

3. For temperature = 1020 K, rpm = 3000 at full load

$$\mathbf{K = 0.7651}$$

$$\boldsymbol{\epsilon = 0.622}$$

$$\boldsymbol{\varphi = 1.4}$$

$$\mathbf{c = 0.3302}$$

$$\mathbf{yCO_2 = 0.5406}$$

$$\mathbf{yH_2O = 0.5886}$$

$$\mathbf{yCO = 0.3302}$$

$$\mathbf{yN_2 = 3.76}$$

4. For temperature = 1110 K, rpm = 4000 at full load

$$\mathbf{K = 1.0554}$$

$$\boldsymbol{\epsilon = 0.622}$$

$$\boldsymbol{\varphi = 1.4}$$

$$\mathbf{c = 0.3674}$$

$$\mathbf{yCO_2 = 0.5034}$$

$$\mathbf{yH_2O = 0.6258}$$

$$\mathbf{yCO = 0.3674}$$

$$\mathbf{yN_2 = 3.76}$$

5. For temperature = 1200 K, rpm = 5000 at full load

$$\mathbf{K = 1.3757}$$

$$\boldsymbol{\epsilon = 0.622}$$

$$\boldsymbol{\varphi = 1.4}$$

$$\mathbf{c = 0.3981}$$

yCO_2 = 0.4727

yH_2O = 0.6565

yCO = 0.3981

yN_2 = 3.76

Engine 2NZ-FE

Engine 2NZ-FE is inline-four cylinder, 16-valved internal combustion engine having displacement volume 0f 1.3L. The product composition is given below,

1. For temperature = 700 K, rpm = 1000 at full load

K = 0.1061

ϵ = 0.622

φ = 1.4

c = 0.1332

yCO_2 = 0.7376

yH_2O = 0.3916

yCO = 0.1332

yN_2 = 3.76

2. For temperature = 810 K, rpm = 2000 at full load

K = 0.2569

ϵ = 0.622

φ = 1.4

c = 0.2118

yCO_2 = 0.6590

yH_2O = 0.4702

yCO = 0.2118

$$yN_2 = 3.76$$

3. For temperature = 890 K, rpm = 3000 at full load

$$K = 0.4182$$

$$\epsilon = 0.622$$

$$\varphi = 1.4$$

$$c = 0.2625$$

$$yCO_2 = 0.6083$$

$$yH_2O = 0.5209$$

$$yCO = 0.2625$$

$$yN_2 = 3.76$$

4. For temperature = 980 K, rpm = 4000 at full load

$$K = 0.6482$$

$$\epsilon = 0.622$$

$$\varphi = 1.4$$

$$c = 0.3112$$

$$yCO_2 = 0.5596$$

$$yH_2O = 0.5696$$

$$yCO = 0.3112$$

$$yN_2 = 3.76$$

5. For temperature = 1060 K, rpm = 5000 at full load

$$K = 0.8898$$

$$\epsilon = 0.622$$

$$\varphi = 1.4$$

$c = 0.3476$

$yCO_2 = 0.5233$

$yH_2O = 0.6060$

$yCO = 0.3476$

$yN_2 = 3.76$

6. For temperature = 1130 K, rpm = 6000 at full load

$K = 1.1244$

$\epsilon = 0.622$

$\varphi = 1.4$

$c = 0.3747$

$yCO_2 = 0.4961$

$yH_2O = 0.6331$

$yCO = 0.3747$

During the combustion process the water-gas reaction also takes place which is basically the reaction of CO_2 and H_2 to yield CO and H_2O.

$$\mathbf{CO_2 + H_2 \longrightarrow CO + H_2O}$$

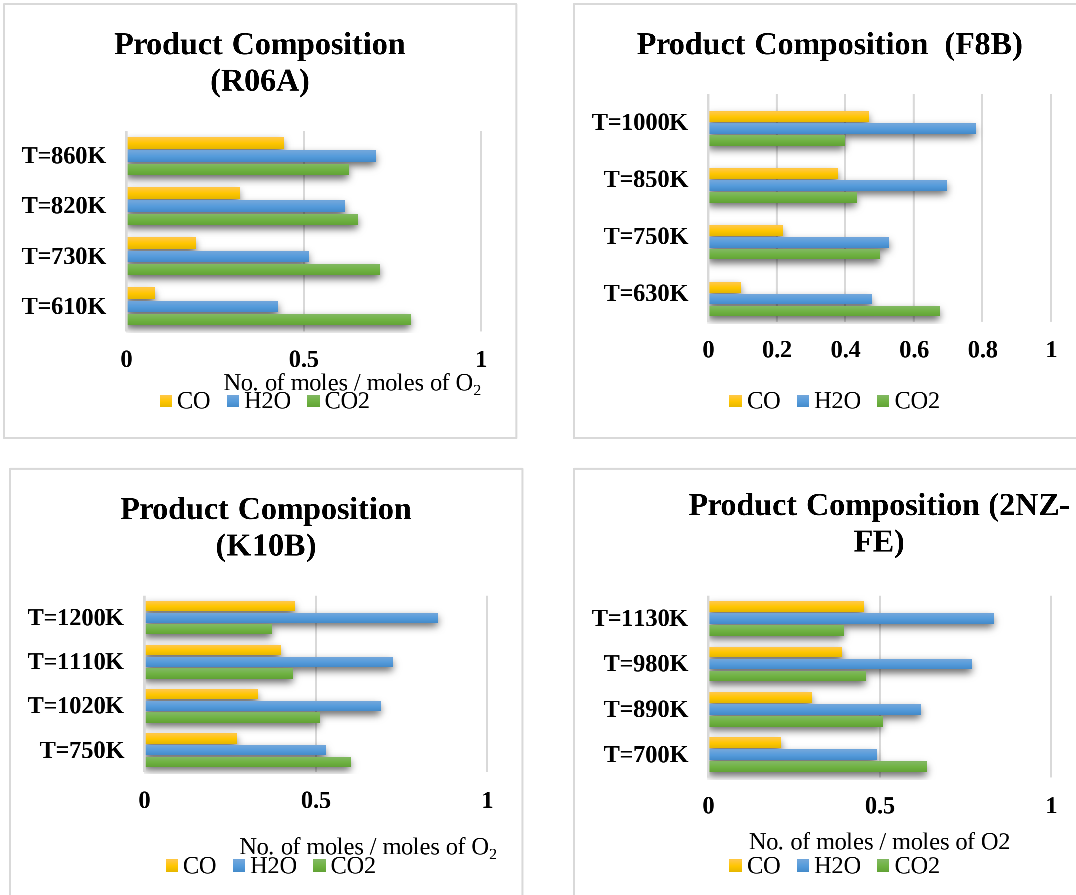

Figure 30: Product Composition for Different Engines

Fuel Saving and Cost Analysis

The ultimate objectives of an E-Turbine system are to improve the performance of the vehicles and to decrease their fuel consumption. Therefore, calculation of fuel saved, and the cost analysis has a key role in the study. The fuel saved by the E-turbine system is calculated by making certain assumptions based on our day-to-day experience.

As we know from the literature that the thermal efficiency is given by,

$$\eta_{th} = \frac{P_{brake}}{\dot{m}_{fuel} \times q_{comb}}$$

The AC compressor of the vehicles draws around 4 kW at its maximum refrigerating capacity, heating value of gasoline is 44000 kJ/kg and by assuming thermal efficiency of 30%. The mass flow rate of the fuel is calculated to be,

$$\dot{m}_{fuel} = 0.00030\ kg/s$$

The summer season in Pakistan is of the duration of almost 7 months, then

Duration of season = 7 months

No. of days = 30 × 7

No. of days = 210 days

On average if a person drives a car for 6 hours daily then the number of driving hours in 7 months will be

No. of hours = 6 × 210

No. of hours = 1260

Time in seconds = 1260 × 3600

Time in seconds = 4536,000

Now to calculate the mass of the fuel consumed by the AC compressor or in other words the mass of the fuel saved by the E-Turbine system in one summer season, multiply the time in seconds with the mass flow rate of the fuel.

$$m_{fuel} = Time\ in\ seconds \times \dot{m}_{fuel}$$

$$m_{fuel} = 1374\ kg$$

$$\mathrm{m_{fuel}} = 1863\ \text{L/season}$$

Pay-Back Period

The main components of E-Turbine system are radial turbine, high-speed generator and Battery Bank. The capital cost of the E-turbine is calculated by

Estimated cost of radial turbine = USD 300

Estimated cost of high-speed 12V PMSG = USD 500

Estimated cost of 12V Lead acid battery = USD 200

Miscellaneous cost = USD 100

Total cost = USD 1100

$$\text{Total cost in PKR} = 152{,}625$$

The total cost of the mass of the fuel consumed by the AC compressor can be calculated by,

$$\text{Price of 1 L gasoline} = 90.38 \text{ PKR}$$

$$\text{Total cost of fuel consumed} = 1863 \times 90.38$$

$$\text{Total cost of fuel consumed} = \text{PKR } 168{,}377/\text{season}$$

To calculate pay-back period,

$$\text{Total cost of fuel consumed/month} = 168{,}377/7$$

$$\text{Total cost of fuel consumed/month} = \text{PKR } 24{,}053$$

$$\text{Pay-back period} = \text{total cost of E-Turbine system/PKR } 24{,}053$$

$$\text{Pay-back period} = 6.34 \text{ or } 6.5 \text{ months}$$

It means that the break-even point will be reach after 6.5 months after which the benefit period will start.

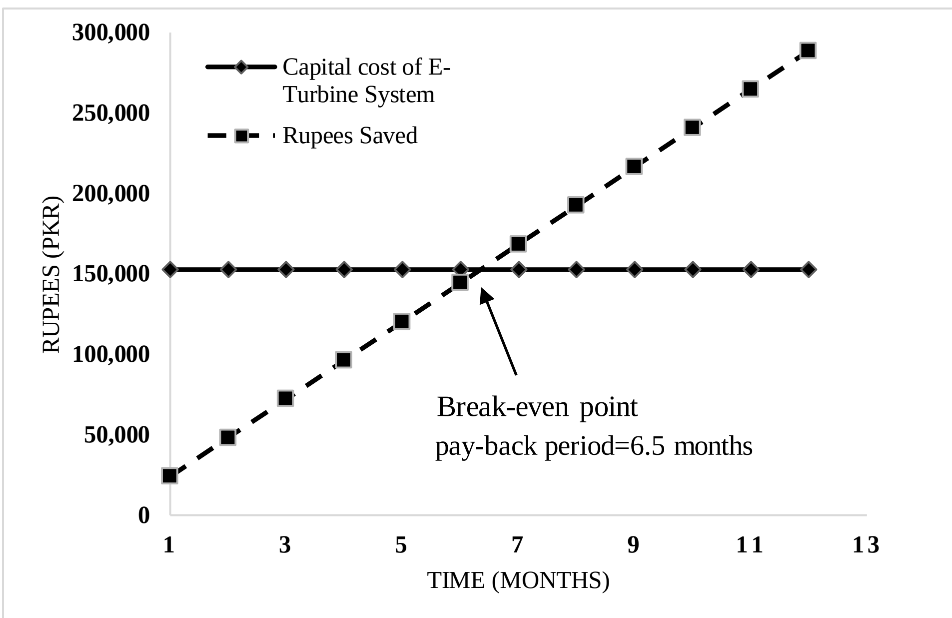

Figure 31: Break-even analysis

References

[1] Dixon, Fluid mechanics, thermodynamics of turbomachinery

[2] "Design and evaluation of a Turbo-Generator as an Auxiliary Power Unit for hybrid vehicles", Antariksh Akhave, Georgios Chrysakis, Atul Gupta, 10.1049/cp.2014.0941

Chapter 6

Conclusions & Recommendations

This chapter presents the conclusions extract from the entire study carried out. The main objective of this thesis was to find out that how much fuel energy is wasted through the exhaust by the vehicles running in Pakistan and to study the parasitic loads on the engine. After taking out the study, we come up with some conclusions that are discussed below. Recommendations for the future work related to this work is also presented in this chapter.

Conclusions

The work done in this project leads us to some conclusions that are presented here,

I. Power and torque curves are plotted for the Engine R06A, Engine F8B, engine K10B, Engine 2NZ-FE and Engine 2ZR-FE. Form the results it can be concluded that the maximum torque and maximum brake power occurred at different engine rpm. The maximum torque occurs at lower rpm than the maximum brake power. It has also been concluded that with the increase in engine rotational speed, torque and brake power start decreasing due to the fact that mechanical efficiency decreases.

II. Engines R06A, F8B, K10B, 2NZ-FE and 2ZR-FE are the most running engines in Pakistan. They all are naturally aspirated engines. A huge amount of fuel energy is wasted through the exhaust gases from all stated engines. Results show that the exhaust gases of R06A, K10B and 2NZ-FE have power of 36.02/6000 kW/rpm, 54.07/5500 kW/rpm and 70.227/6000 kW/rpm respectively. This is a huge amount of fuel energy that is being wasted.

III. Exergy analysis of the exhaust energies of the stated engines has been done in the study. The study shows that the available work in the exhaust energy is more than enough to drive the parasitic loads of the automotive vehicles. Engine R06A has the exergy of 25.76 kW at 6000 rpm. K10B has exergy of 30.31 kW at 5500 rpm while the engine 2NZ-FE has 41.17 kW of available energy at 6000 rpm. Therefore, E-Turbine system has a huge potential to be installed in automotive vehicles as a retrofit technology for waste heat recovery.

IV. Radial turbine is selected from the NsDs diagram for engine K10B and the design point of 3500 rpm. It has been found that the radial turbine produces almost 6.48 kW shaft power which can be used either directly to drive the AC compressor or by coupling it with an electrical generator and the electrical energy produced is then stored in the battery.

V. The high-speed permanent magnet synchronous generator (PMSG) is coupled with the shaft of the radial turbine. The PMSG produces the electrical power of 4 kW which is then used to charge the battery.

VI. The 90 Ah battery is used to drive the AC compressor and alternator. This battery is charge by the electrical current produced by the generator. The battery is recharge in 13 minutes by the generator.

VII. The E-Turbine System would save 1863 L/season when installed in vehicles. This system has a pay-back period of 6 months.

Recommendations for Future Work

The completion of this thesis leads to certain recommendations for the future work.

I. A test-rig should be designed for the experimental study to validate the results of the simulations. A test-rig should include the internal combustion engine which will be used to find the power and torque curves and exhaust energy of the gases. A dynameter should be used for these results. Radial turbine, shafts, generator and battery bank of the known specifications should be including in test-rig.

II. In this thesis, the E-Turbine System is designed only for one internal combustion engine namely engine K10B (1L). For the future, it is recommended that the E-Turbine System shall be designed for different engines studied in this thesis.

III. Comparison f the performance of E-Turbine System for different engines shall be carried out to find out that for which vehicle the E-Turbine System is most feasible.

IV. The idea needs to be commercialized for effective results.

Appendix

```
%RO6A (660cc) Engine Modeling

function[Work_2]=IC()

clear();

P_1=101.325;                              %initial pressure
T_1=298;                                  %initial temperature
Vd=0.0008;                              %Total cylinder volume
r_c=11;                                   %compression ratio
B=0.0685;                                  %Cylinder Bore
S=0.072;                                  %Piston Stroke
V_d=((pi*B^2)/4)*S;                       %per cylinder volume
V_max=(r_c/(r_c-1))*V_d;                  %maximum volume
V_c=V_max/r_c;                            %clearance volume
R_bar=8.314;                              %universal gas constant
M=101.21;                                 %molecular mass of fuel
R=R_bar/M;                                %gas constant
m=(P_1*(V_d+V_c)/(R*T_1));                %initial mass in the cylinder
a=S/2;                                    %Crank Radius
R=4;                                      %%ratio
l=a*R;                                    %connecting rod length
q_c=44000;                                %heat of combustion
Q_in=m*q_c;                               %total heat input
y_c=(4*V_c)/(pi*B^2);                     %clearance
A_o=pi*B*y_c;                             %clearance area
gamma=1.02;                               %specific heat ratio
rpm=1000;
rps=rpm/60;
```

```
omega=(pi/30)*rpm;                    %rotational speed(rad/s)
C=0.05;                               %mass blowby coefficient
thetas=-20;                           %start of combustion
thetad=50;                            %duaration of combustion
n=3;                                  %weibe exponent n
af=5;                                 %weibe parameter a
n_c=3;                                %number of cylinder
eta_i=0.45;
eta_v=0.78;
FA=0.078;
den_i=1.225;
step_1=11.11;                                                   %rotational speed
ZZ=round(4000/step_1);
rpm=2000;
rpme=rpm+step_1;
U_bar=2*rpm*S;                       %mean piston speed
eta_m=0.90;                           %mechanical efficiency
step=1;                               %step
NN=360/step;                          %number of data points
theta=-180;                           %initial crank angle
thetae=theta+step;                    %crank angle
y_theta=l+a-((l^2-(a^2.*sind(theta).^2)).^0.5)+(a.*cosd(theta));              %pi
A_inst=pi*B*y_theta;                  %instantaneous cylinder area
A=A_o+A_inst;                         %total cylinder area
A_p=(pi*B^2)/4;                       %piston area
save.rpm=zeros(ZZ,1);
save.theta=zeros(NN,1);
save.vol=zeros(NN,1);
save.press=zeros(NN,1);
```

```
save.work=zeros(NN,1);
save.temp=zeros(NN,1);
save.W_dot=zeros(ZZ,1);
save.W_brake=zeros(ZZ,1);
save.bmep=zeros(ZZ,1);
fy=zeros(3,1);
fy(1)=1.5;
fy(3)=230;
gy=zeros(3,1);
gy(1)=0;
gy(3)=0;

for i=1:NN,
    [fy, vol]=pressure(theta,thetae,fy);
    theta=thetae;
    thetae=theta+step;
    save.theta(i)=theta;
    save.vol(i)=vol;
    save.press(i)=fy(1);
    save.work(i)=fy(2);
     work_a(i)= fy(2);
     save.temp(i)=fy(3);

end

[pmax,id_max]=max(save.press(:,1))
thmax=save.theta(id_max);
Work_2=save.work(NN,1)
Work_2=work_a;
```

```
Work_2=Work_a;
%W_dot=save.W_dot(NN,1);
W_dot=(n_c*Work_2*rps)/2;
%W_3=save.W_dot(NN,1);
W_brake=(eta_m*n_c*W_dot*rps)/2;
%W_brake=((eta_m*eta_i*q_c*eta_v*den_i*Vd*rps)/2)*(FA/(1+FA));
%W_4=save.W_brake(NN,1);
wb1(i,:)=W_brake;
bmep=(W_brake)/(V_d*rps);
bmep_1=save.bmep(NN,1);
rpm = 2000;
for i=1:1:360

Work_temp = Work_2(i);
w_i(i) = 3*(Work_temp)*(rpm(i)/2);
if (i<360)
rpm(i+1) = rpm(i)+11.11;
end
end
%
% work_brake = eta_m.*w_i;

rpm(1) = 2000;
for i=1:1:360

bmep(i) = (2*(work_brake(i)))/(V_d*rpm(i));
if (i<360)
```

```
rpm(i+1) = rpm(i)+11.11;
end
end

%plot results
plot(save.theta,save.press,'-','linewidth',2)
set(gca,'fontsize',18,'linewidth',2);
title('Pressure VS Crank Angle');
xlabel('Crank Angle \theta (deg)','fontsize', 16)
ylabel('Pressure P (bar)','fontsize', 16)
figure();

plot(save.theta,save.temp,'-','linewidth',2)
set(gca,'fontsize',18,'linewidth',2);
title('Temperature VS Crank Angle');
xlabel('Crank Angle \theta (deg)','fontsize', 16)
ylabel('Temperature T (Kelvin)','fontsize', 16)
figure();

plot(save.theta,save.work,'-','linewidth',2)
set(gca,'fontsize',18,'linewidth',1.5);
xlabel('Crank Angle \theta (deg)','fontsize', 18)
ylabel('Indicated Work W (joule)','fontsize', 18)
figure();

plot(save.theta,W_dot,'-','linewidth',2)
set(gca,'fontsize',18,'linewidth',1.5);
```

```
xlabel('Crank Angle(deg)','fontsize', 18)
ylabel('Brake  Power P (watt)','fontsize', 18)
figure();

plot(save.theta,bmep,'-','linewidth',2)
set(gca,'fontsize',18,'linewidth',2);
xlabel('Crank Angle(deg)','fontsize',16)
ylabel('Brake  Mean Effective Pressure  (Pascal)','fontsize', 18)

function[fy,vol]=pressure(theta,thetae,fy);
[xx,yy]=ode23(@press,[theta thetae],fy);
for j=1:3
    fy(j)=yy(length(xx),j);
end
    function [yprime]=press(theta,fy)
vol=((1.+ (r_c -1)/2.*(1-cosd(theta)))/r_c);
    dvol=(r_c - 1)/2.*sind(theta)/r_c*pi/180;
%vol =(V_c+(pi*B^2/4)*(1+a-(((1^2-(a^2.*(sind(theta))^2)).^0.5)+(a.*cosd(theta)))));
 %dvol   = ((B^2*pi*(a*sind(theta) + (a^2*cosd(theta)*sind(theta))/(- a^2*(sind(theta))^2 + 1^2)
dx=0.;
if(theta>thetas)
    dum1=(theta-thetas)/thetad;
    x=1-exp(-(af*dum1^n));
    dx=(1-x)*af*n*dum1^(n-1)/thetad;
end
term1=-gamma*fy(1)*dvol/vol;
term2=(gamma-1)/vol*(Q_in*dx);
term3=(gamma*C*fy(1))/omega;
yprime(1,1)=(term1+term2-term3);
```

```
 yprime(2,1)=fy(1)*dvol;
 %yprime(3,1)=((n_c*yprime(2,1))*rpm)/2;
 yprime(3,1)=fy(3)*((1/fy(1))*yprime(1,1)+(1/(vol)*dvol)+(C/omega));
 %W_dot=((n_c*yprime(2,1))*rpm)/2;
     end
 end
 function [gy, W]= brake(rpm, rpme, gy);
     function [gprime]=power(rpm,gy)
         W=yprime(2,1);
         gprime(1,1)=((n_c*yprime(2,1))*rpm)/2;
         gprime(2,1)=(eta_m*n_c*save.work(NN,1)*rpm)/2;
         gprime(3,1)=(2*gprime(2,1))/(Vd*rpm);
     end
 end
 disp(wb1)

 end
```

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SYSTEM DESIGN OF A HIGH-SPEED E-TURBINE SYSTEM FOR WASTE HEAT RECOVERY IN AUTOMOTIVE VEHICLES

Muhammad Ishaq Khan
Department of Thermal Energy Engineering
NUST H-12, Islamabad Pakistan
madishaq143@gmail.com

Adeel Javed
Department of Thermal Energy Engineering
NUST H-12, Islamabad Pakistan
adeeljaved@uspcase.nust.edu.pk

Abstract

Due to the rapid industrialization and urbanization of Pakistan, automotive transport sector is one of the fastest growing sectors in Pakistan in terms of output, energy consumption and CO2 emissions. The existing automotive sector in Pakistan, hardly complying to an obsolete Euro II emissions standards, requires a sizable enhancement in energy conversion efficiency. Furthermore, there are two parasitic loads on conventional IC Engine, (1) generate current to recharge the vehicle's battery, The electrical charge is produced by the alternator (generator) that is commonly coupled to the crankshaft pulley of the IC engine through a belt-drive system. (2) The air-conditioning compressor draws several kilowatts of power from the IC engine at peak load, thereby increasing the fuel consumption of the vehicle considerably. On average the air-conditioning system can increase the fuel consumption of a vehicle by 20% . Internal combustion engine wastes a lot of energy through their exhaust. Different IC Engines(660cm^3,1000cm^3,1300cm^3) have been modeled in MATLAB. Results showed that on average the exhaust gas temperature ranges from 850K to 1100K at the engines operating at full load. The mass flow rate of exhaust gases also increases with the engine rpm. The pressure of the exhaust gases is in the range 4-6 bar. This exhaust energy is recovered by an e-turbine system. E-turbine system consists of turbine, generator and spindle. Exhaust energy rotates the turbine placed in the exhaust manifold which in turns turn the shaft. Shaft is placed in the generator and due to EMF , current is produced. The e-turbine system produce power that is almost sufficient to run the vehicle HVAC compressor, replacing the engine parasitic load

Keywords: Electric Turbine, Internal Combustion Engine, HVAC, MATLAB, Alternator

1. INTRODUCTION

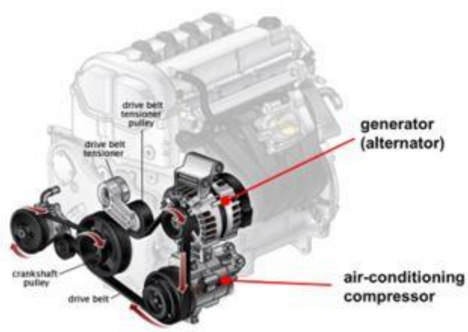
The inadequate accessibility and huge dependency on the fossil fuel due to the recent developments in industrial technologies shifted the attention of research towards the effective and efficient use of energy. Most part of fossil fuels around the world is used by the internal combustion engines. The technological development and a good power to weight ratio made the IC engine to be the best option for transportation, power plants and agriculture etc.[1].

IC engines are most widely used in the field of transportation and mobile power generation purposes [2]. But IC engines waste a reasonable amount of its fuel energy to the environment. Only 30-40% of its fuel energy is converted into useful mechanical work [3]. A huge amount of fuel energy is wasted to the environment in the form of exhaust gases, through the coolant and charge air[2].

In recent years, due to increasing prices and shortage of fossil fuel compel the researchers and engineers to think about an engine design that would use the fuel and produce the work output in an efficient way and with a small and clean waste. They have developed certain methods that reduce emission but still the waste energy is in reasonable amount i-e. supercharge, lean mixture etc. Major portion of waste heat from the engine is in the form of exhaust gases. The engineers and researchers tried to recover this exhaust energy. Various waste heat recovery technologies are developed.

Waste heat recovery technologies have a great potential to increase the overall efficiency of engine and is a hot topic for research [4], [5]. Combustion inside the engine and its emissions have been improved by the turbocharger technology [6], [7]. The turbocharger is now a technologically mature system for waste heat recovery application. Turbocharger is primarily a turbine-driven compressor and remains largely associated with internal combustion (IC) engines. Compared to a non-turbocharged IC engine delivering similar magnitude of power, a turbocharged IC engine will be smaller, lighter, and more efficient [10]. The exhaust gas energy that would have otherwise been wasted is used to drive the turbine, which then transmits the shaft power to the compressor. One of the waste heat recovery technologies is turbocompounding the engine. Low pressure turbine is placed after the high pressure turbocharger compressor. The power produced by the LP turbine is transferred to wheels by coupling it with the engine crankshaft. The main disadvantage of turbocompound engine is its high pumping loses. Engine needs to do more work in the exhaust stroke to throw out exhaust gases from the cylinder [8]. Another method of turbocompounding which is also used for waste heat recovery is electric turbocompounding. ET converts the LP turbine shaft work into electric power by using a high speed generator [9]. Another promising technology for WHR that has been developed nowadays is thermoelectric generator. The working principle behind this technology is seebeck effect. Two different materials are placed in the path of exhaust gases. Electric power is produced due to the flow of energy by virtue of temperature difference of exhaust gases and the external environment. But the low efficiency of this system is the main hurdle in the implementation [9]. Waste heat is also being recovered by using different thermodynamic cycles such as Stirling cycle, Brayton cycle, Kalina cycle, and Rankine cycle [2]. Organic Rankine cycle is the most promising cycle among the above thermodynamic cycles due to its many advantages i-e flexibility, simple design and easy maintenance.

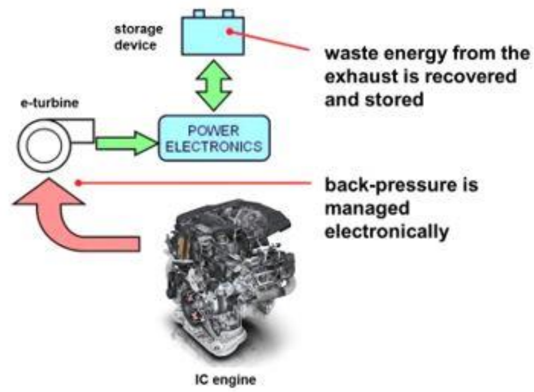
The transport sector, one of the important sectors of Pakistan's economy, contributes around 10 percent to the GDP and 6 percent employment to the country [11]. Oil and Gas is the most consumed source of energy in transport. The petroleum consumption in the sector reaches 10 million TOE (Tons of Oil Equivalent) in 2013, registering a growth rate of 1.8 percent in the last decade [12]. The number of vehicles, mostly using petroleum products and natural gas, have increased rapidly from 2.7 million vehicles in 1990 to 9.8 million in 2010, thereby contributing worryingly to air quality degradation in Pakistan. The existing automotive sector in Pakistan, hardly complying to an obsolete Euro II emissions standards, requires a sizable enhancement in energy conversion efficiency. Also in Pakistan, most of the automotive vehicles are equipped with naturally aspirated engine(non-turbocharged). So the exhaust waste heat is more as compared to the turbocharged engine. In conventional automotive vehicles, there are two parasitic loads on the engine. 1) vehicle's battery charging system. The electrical charge is produced by the alternator (generator) that is commonly coupled to the crankshaft pulley of the IC engine through a belt-drive system. 2) vehicle's HVAC system. . The HVAC system is also an integral automotive system, which ensures the thermal comfort of the passengers and responsible for the circulation of clean air in the passenger cabin. The air-conditioning



compressor draws several kilowatts of power from the IC engine at peak load, thereby increasing the fuel consumption of the vehicle considerably. On average the air-conditioning system can increase the fuel consumption of a vehicle by 20% [13]. The vehicle's performance is also influenced negatively. The HVAC load on the engine is substantial for vehicles operating in the regions with hot climate conditions. In Pakistan, the summer season is of the duration of 7-9 months. So for a thermal comfort, the use of air conditioners in the car cabin is necessary and a lot of engine power is used to run the AC compressor. On one side, a huge of amount energy is wasted through the exhaust and on the other side, these parasitic loads are responsible for the high fuel consumption. This paper proposes a system that is used to recover this waste heat and replaces these parasitic loads improving fuel consumption of the vehicles in Pakistan. The

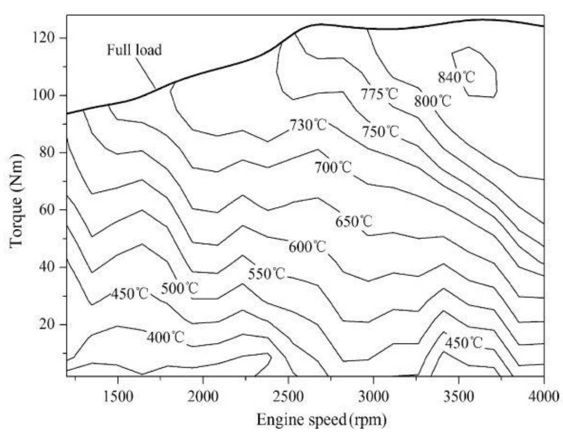
E-Turbine concept is a novel and innovative idea for the recovery of automotive waste heat. This technology is best suitable for conventional automotive vehicles. The E-Turbine system consist of turbine, spindle and generator. the working principle of e-turbine system is that exhaust gases pass through the turbine that is placed in the exhaust manifold of engine. Turbine extract power from the exhaust that will otherwise be wasted. Turbine shaft rotates, upon which an electric generator is mounted. Due to change in

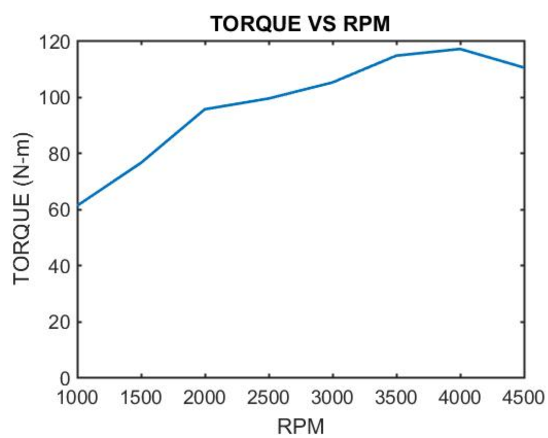
EMF, generator produced electricity. This electrical power is used for the operation of HVAC compressor and for battery charging which is used to run other electrical accessories of the vehicle. this technology replaces the two parasitic loads that are discussed above and also recover the waste heat in a simple way.



1. Methodology

The main step in the designing of E-Turbine system is modeling of different IC Engines that are currently used in Pakistan's transport sector. 660cm^3, 1000cm^3, 1300cm^3 IC engines are modelled in MATLAB. Zero-dimensional model is used for the modelling of these IC engine in MATLAB. Model is verified by





modelling 1.4L engine and its results are compared with the published data[14].

First In-cylinder pressure, temperature and indicated work are calculated by the model.

For pressure calculation, the equation used is

$$\frac{dP}{d\theta} = -\gamma \frac{PdV}{Vd\theta} + \frac{(\gamma-1)}{V}\frac{dQ}{d\theta} - \frac{\gamma \dot{m}_1}{\omega m} P \qquad (1)$$

$$C = \frac{\dot{m}_1}{m}$$

For temperature calculation, the equation used is

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

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

For indicated work, the equation used is

$$\frac{dW}{d\theta}=P\frac{dV}{d\theta} \qquad (3)$$

After calculating this, brake power, torque, mass flow rate of exhaust gases, exhaust energy and availability are calculated at different rotational speeds for the stated engines.

The equations used are,

For brake power,

$$W_b=\frac{n_c W_i rps}{2}\eta_m \qquad (4)$$

For torque,

$$\dot{W}_b=2\pi\tau\ rps \qquad (5)$$

For mass flow rate of exhaust gases,

$$\eta_o=\frac{P}{q_c\dot{m}_f} \qquad (6)$$

$$\dot{m}_a=\dot{m}_f\times AFR$$

$$\dot{m}_{exh}=\dot{m}_a+\dot{m}_f$$

For exhaust energy,

$$\dot{Q}_{exh}=\dot{m}_{exh}\times C_{pexh}\times\Delta T \qquad (7)$$

For availability,

$$A_{exh}=Q_{exh}+\frac{(m_f\times m_a)T_{amb}(C_{pexh}\times Ln\left(\frac{T_{amb}}{T_{exh}}\right)-R_{exh}\times Ln\left(\frac{P_{amb}}{P_{exh}}\right))}{3600}$$

Final step is the selection of turbine from the Ns, Ds diagram. The turbine is selected on the basis of available energy and exhaust mass flow rate from the Ns, Ds diagram.

The equations used are,

$$n_s = N(\frac{\dot{m}}{\rho_e})^{\frac{1}{2}} \times (\Delta h_{0s})^{\frac{-3}{4}} \qquad (9)$$

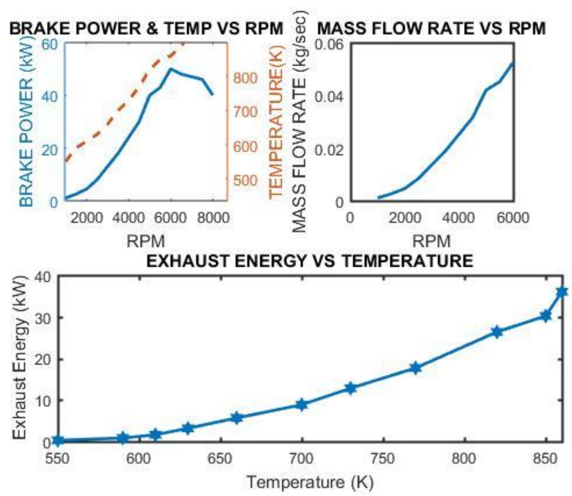
$$\Delta h_{0s} = C_p T_{01}[1 - (p_{02}/p_{01})^{(\gamma-1)/\gamma}]$$

$$D_s = \frac{D(\frac{\Delta p}{\rho})^{\frac{1}{4}}}{Q^{\frac{1}{2}}} \qquad (10)$$

2. Results and Discussion

Engine 660cm^3

Engine R06A having a displacement volume of 660cm^3 has been modelled in MATLAB. Brake power, mass flow rate of exhaust gases and exhaust energy are plotted against different rpm. Brake power increases



with the rpm, reaches its maximum and then decreases. This is due to the fact that frictional loses increases at the higher rpms, so the mechanical efficiency decreases and ultimately power drops. The temperature of the exhaust gases also increases with the rpm at peak load. Exhaust energy and its mass flow rate increase with the rpm. The results are shown in the figure 5 below,

Engine 1000cm^3

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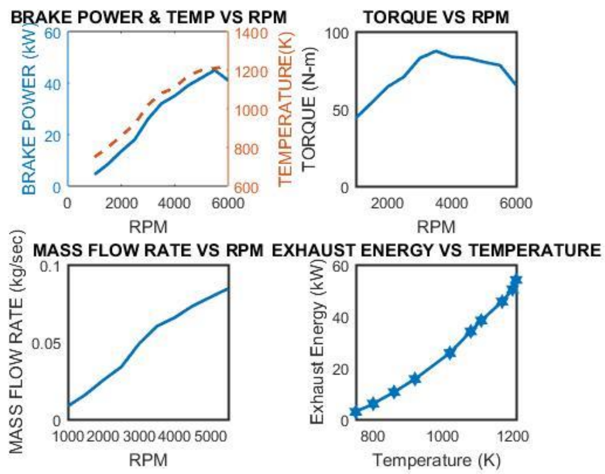
Engine K10B having a displacement volume of 1000cm^3 has been modelled in MATLAB to calculate its brake power, torque, exhaust energy, mass flow rate of exhaust gases at different rpm. The graphs shows that maximum power and maximum torque occur at two different rpms. Maximum torque occurs at low rpm while maximum power occurs at high rpm. The exhaust energy, its mass flow rate, and temperature of



exhaust gases increases with the rpm at peak load.

Engine 1300cm^3

Engine 2NZ-FE having a displacement volume of 1300cm^3 has been modelled in MATLAB to calculate its brake power, torque, exhaust energy, mass flow rate of exhaust energy at different rpm. The exhaust energy, its mass flow rate and temperature increase with rpm at peak load.

Availability Analysis

The exhaust energy that is wasted in the exhaust gases is not completely recoverable. Some portion of exhaust energy should be thrown out to the environment. The available useful energy present in exhaust gases that can be converted into useful work is called exergy. This available energy is less than the total energy present in the exhaust gases. Exergy analysis has been done for the stated engines at different rpm at peak load. The table shows the exhaust energy and exergy for engines at different rpm.

AC Compressor Power

In Pakistan, Sanden AC compressor are widely used in automotive vehicles. SD7H15 has been selected for study to find out that how much power it consume from the engine at its maximum refrigerating capacity. The chart shows that at its maximum refrigerating capacity, it draws almost 5kW from the engine.

Turbine selection

Engine 1000cm3 is selected for which the turbine is designed. The design point is selected at 3000 rpm. Available energy at 3000 rpm is 13.64kW and the power needs to run the compressor is 5 kW. By using the above equations, the selection of radial turbine is made producing 5.5kW power. This power is enough to run the AC compressor of the vehicle at its maximum refrigerating capacity.

CONCLUSIONS

AC compressor load on engine is replaced by the E-Turbine system which improves the fuel consumption of the vehicle. The removal of AC compressor load from the engine also improves the pick of the vehicle. This paper gives the conceptual design of the E-Turbine system. This idea needs to be commercialized for the improved results.

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