Brayton Cycle Based Turbocharging

Cyril Jose Palathinkal[1] Hayns Andrew Kurian[2] Jibin Jojo[3] Jojin N J[4] Nandhu S[5]
[1,2,3,4]UGStudent [5]Assistant Professor
[1,2,3,4,5]Department of Mechanical Engineering
[1,2,3,4,5]Saintgits Collage of Engineering, Kottayam

Abstract— Traditional turbocharging systems make use of the energy carried by exhaust gases by placing a turbine-compressor arrangement in the exhaust line of the automobile. The major problem with this system is the high back-pressure developed in the engine cylinders due to the obstruction provided to the exhaust flow by the turbine blades. The back-pressure results in higher operating temperatures, higher fuel consumption and overall lower engine performance The proposed system replaces the turbine-compressor arrangement from the exhaust line and places a heat exchanger which extracts heat from the exhaust and transfers this heat to pressurized air on the other side of the heat exchanger; it is this hot pressurized air that is used to drive the turbine-compressor arrangement. Since, the arrangement does not impede the flow of the exhaust it is expected that the back pressure will decrease resulting in the overall improvement of engine efficiency.
Key words: Turbocharging, turbine-compressor

I. Introduction

An emerging topic of research in the field of Internal Combustion engines is waste heat recovery techniques. This field has been gaining popularity rapidly owing to the immense demand of energy efficiency in engines and the fact that most I.C engines are able to convert only 30% of the actual chemical energy in the fuel to useful work. It is estimated that a major chunk of the energy obtained through combustion is lost by means of heat carried by the coolant or the exhaust gas. Out of all the waste heat recovery systems one of the most common waste heat recovery systems seen nowadays is the conventional turbocharging technique. This technology works on the same basic principle of harvesting the thermal waste energy contained in the exhaust [1].

The conventional system of turbocharging suffers from two fundamental defects; (i) Backpressure and (ii)Turbo-lag. Engine exhaust back pressure is defined as the exhaust gas pressure that is produced by the engine to overcome the hydraulic-resistance of the exhaust system in order to discharge the gases into the atmosphere. The major problems with increased back-pressure are: Increased pumping work, reduced intake boost pressure, cylinder scavenging problems, higher peak temperature-resulting in higher NOx emissions, higher specific fuel consumption. And Turbo-lag basically refers to the time required for the turbo to spool up i.e. the time required for the required boost pressure to be achieved, this occurs mainly because of the finite time required for the pulse of exhaust gas to reach the turbine, hence making the turbocharger very unresponsive and sluggish at low speeds.

II. Literature review

There have been several studies on the topic of waste heat recovery from exhaust and also regarding the different methods of turbocharging. Obviously these studies are of most importance as they focus on improving the overall performance of automobiles-which is crucial given the fact that the world's reserves of fossil fuels is rapidly depleting and pollution control and exhaust emission levels is of prime importance. The studies conducted in the relevant fields of turbocharging and waste heat recovery are as follows:

1) 'A comparative study on various turbocharging approaches based on I.C Engine exhaust gas energy recovery' [2].
2) 'Experimental investigation on heat recovery from diesel engine exhaust using finned shell and tube heat exchanger and thermal energy storage' [3].

III. Design of the system

The proposed system of turbocharging which is to be designed for demonstration and experimental purposes is shown below in Fig 1. In the proposed system the back pressure on the engine is minimized by removing the exhaust line from the turbocharger turbine end and passing it through a heat exchanger. Compressed air is passed through the other side of the HPHE. The heat exchanger transfers heat from the hot exhaust gases to the compressed air stream on the cold side of the heat exchanger. The high pressure hot air from the heat exchanger then passes to the turbine side of the turbocharger where it expands isentropically and transmits the work to the compressor side which is used to compress the inlet air to the engine.

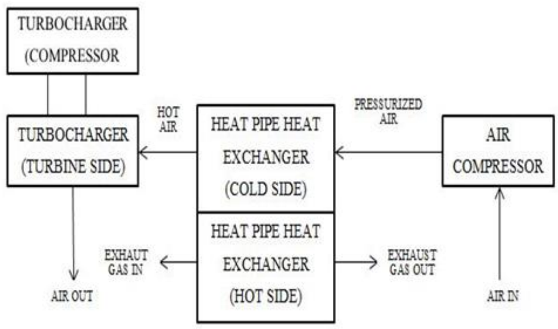

Fig. 1: Proposed System

Since the system is designed such that there is no direct interaction between the turbine side of the turbocharger and the exhaust gases, the hydraulic resistance the exhaust gases have to overcome while travelling through the exhaust system is reduced substantially thereby decreasing the backpressure against which the engine has to operate.

Hence, it can be theoretically reasoned that this new system of turbocharging improves overall engine performance parameters such as specific fuel consumption, engine operating temperatures and inhibits NOx formation in Diesel Engines [4].

The subsequent sections of the paper mainly focus on a brief treatment of the methodology adopted during the design stage of the experimental setup and the various important considerations and formulae used.

A. Determining The Ratings For The Compressor And The Heat Pipe Heat Exchanger:

Since air compressors are very commonly available in the market, the design procedure involves selecting an air compressor that will be able to match the flow requirements of the turbocharger selected and then designing the heat pipe heat exchanger so as to ensure a particular level of performance is obtained from the turbocharger.

B. Turbocharger Selection:

Turbochargers are generally selected on the basis of the compressor maps available from the manufacturer which will indicate whether the specified turbocharger is capable of providing the required boost pressure and mass flow rate to the engine in question at the operating speed without passing the surge limit of the compressor as can be seen from the compressor map of Garrett GT1544 shown in Fig. 2.

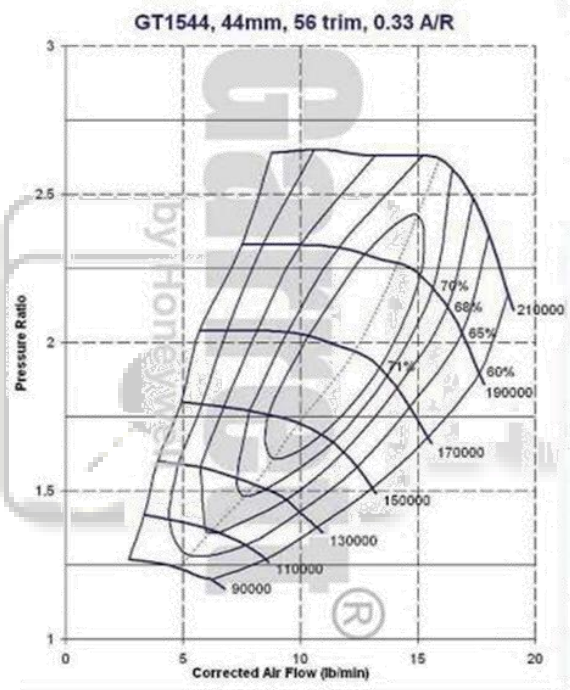

Fig. 2: Compressor map of Garrett GT1544

In this specific demonstration a Garrett GT1544 or equivalent turbocharger has been used (Selected on the basis of engine air requirements and the compressor maps available in the site for turbocharging manufactures such as Borg Warner/Garrett). The turbine map for the Garrett GT1544 is shown in Fig. 3 and has been used to determine the mass flow rate required to pass through the turbine and the corresponding pressure ratio.

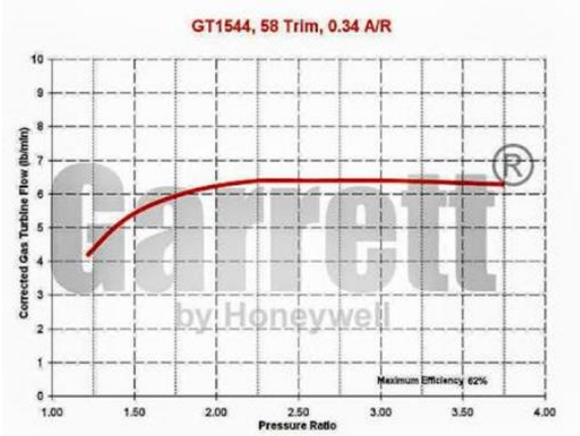

Fig. 3: Garrett GT1544 turbine map.

The turbocharger selected has following key characteristics as obtained from the manufacturer of the turbocharger-Garrett:[5]

Isentropic efficiency, $\eta_{max} = 62\%$
Mass flow rate of air, $m_a = 6$ lb/min
Mass flow rate of exhaust, $m_e = 1.8$ kg/min
Pressure drop $= P_{1t} / P_{2t} = 1.75$

C. Compressor Selection:

The compressor was fixed depending on the requirement of flow of the turbocharger. Hence, the compressor was selected with the following specifications:

Mass flow rate, $M_e = 0.05$ kg/s (As obtained from turbine map)

Pressure boost in the compressor = 35 psi ≈ 2.5 bar (fixed)

Based on the data obtained above, the compressor is selected such that it provides a 35 psi absolute pressure boost and has a 0.05 kg/s mass flow rate at 1500 rpm.

The compressor required should have an Actual Cubic Feet per Minute flow rate, ACFM of 47.325 and a boost pressure of 35 psi (2.5 bar) absolute. Basic thermodynamic analysis of the proposed system of turbocharging conducted revealed that a heat exchanger of approximately 1800W needs to be designed and manufactured to operate the said turbocharger to provide an inlet pressure boost of 29.4 psi absolute(2 bar)to the engine with the compressor of selected ACFM and pressure boost as mentioned in the previous page.

D. Design Of The Heat Pipe Heat Exchanger:

Assumptions made during the design of heat pipe heat exchanger:

- A counter flow type heat exchanger has been designed.
- Since a counter flow type heat exchanger has been designed the temperature profiles along the length are such that there is reasonable constant temperature difference between the hot and cold sides throughout the length of the heat exchanger.
- Hence , the heat transferred (which solely depends on the properties/temperature of air) all other factors remains constant (such as charge fill, evaporating fluid, pipe diameter, diagonal pitch, longitudinal pitch, transverse pitch) can be assumed to be uniformly distributed across the rows of heat pipes of the exchanger.
- Design a heat exchanger for Q = 2000W.
- Consider one row of heat pipe as shown in Fig. 4

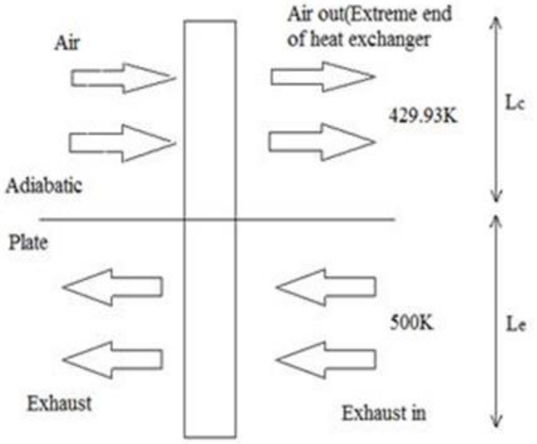

Fig. 4: One row of heat pipe heat exchanger.

The design of a heat pipe heat exchanger is a complicated affair, however for the purpose of this paper, the major formulae for the convective heat transfer coefficients and other important parameters were obtained from previous studies conducted on air- to air heat pipe heat exchangers [6].

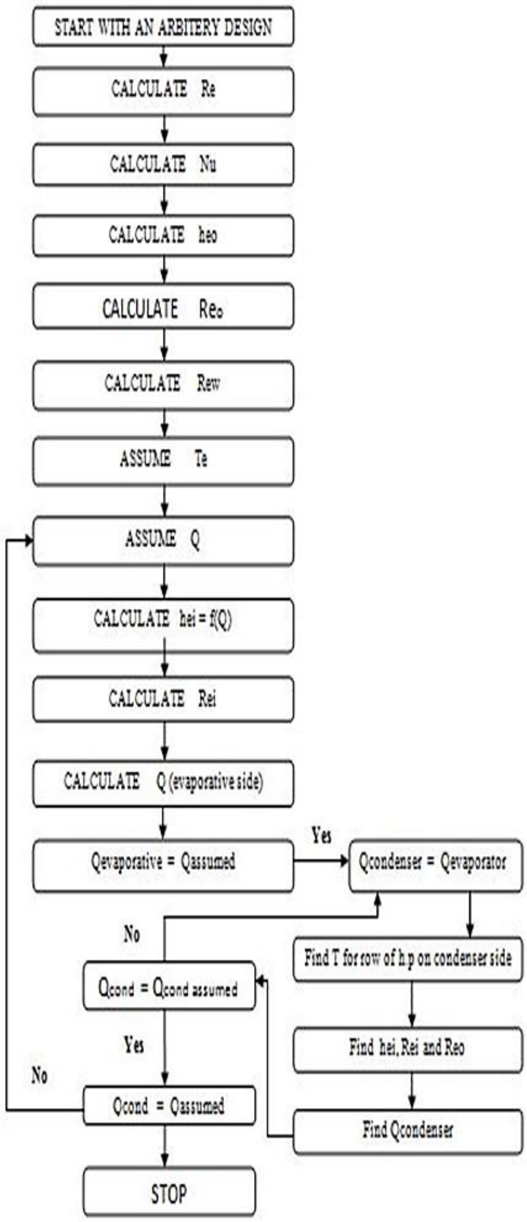

Fig. 5: Design procedure

Since the design involves several complicated interdependent equations, an iterative method was used to arrive at a solution for the design procedure. The methodology is shown in Fig. 5.

E. . Final Design Configuration of Hphe:

After several configurations and iterations, the specifications of heat exchanger designed that would transfer approximately 2000W of energy is shown below

No of heat pipes per row, n = 3
No of rows in HPHE, r = 7
Evaporator section length, Le = 20cm
Condenser section length, L_c = 20 cm
Width of HPHE shell, w = 14 cm
Total length of HPHE, L = 40 cm
Thickness of pipe used, t = 3/4" (STD refrigeration Cu pipe)
Longitudinal pitch, P_l = 40 mm
Transverse pitch, P_t= 40 mm

Thickness of all sheet used for the HPHE shell is ¼" (M.S sheet).

IV. FABRICATION AND SETUP

Copper pipe of $^3/_4$ inch thickness has been used for transferring heat from exhaust gas to the water at low pressure in the evaporator section after which the boiling water condenses and loses its heat to the incoming air via the copper pipe. Since copper has high thermal conductivity it transfers heat at reasonable rate due to inherent low thermal resistance. During the course of fabrication, the 40 cm copper pipe ends were closed by gas welding process. During welding process pipe end is melted and it is closed by means of an anvil and tongs to join the end while applying filler metal rod at the pipe end to make it as a leak proofing pipe.

The heat exchanger shell consists of $^1/_4$ inch thickness mild steel sheet. And the dimensions of mild steel sheet for the various faces of heat exchanger shell are; 14cm×40cm – 3 numbers, 14cm×20cm – 4 numbers, 20cm×40cm-4 numbers. These various sheets are joined by arc welding process. So that it makes a rectangular box, which looks like one box placed over the other box (the lower box is the shell for the evaporator section whereas the upper box is the shell for the condenser section). Compressed air flows through the upper part of the heat exchanger shell which acts as the condenser side whereas the lower part of the heat exchanger is subjected to exhaust gas air flow and acts like an evaporative side. The rectangular shape heat exchanger shell is divided into two portions by an adiabatic partition sheet. And the partition sheet is drilled with 23 holes(staggered) as shown in the Fig 6.

Fig. 6: Heat exchanger shell

The copper pipes are placed inside the holes of the partition sheet of heat exchanger shell. 20 cm of the copper pipe is projected to the evaporative side and remaining 20 cm of the copper pipe is projected to the condenser side. The centre of the copper pipe is joined to the partition wall by brazing process as shown in Fig. 7. The copper pipe is filled with cold water up to the middle of the pipe. Following which the lower portion of copper pipe is heated using welding torch.

Fig. 7: Heat pipe inside the heat exchanger shell.

Due to the higher temperature water boils inside the pipe, after which the upper end of the pipe was closed. So that the copper pipe contains only water and water vapour, as the air is expelled from the tube.

After completing the fabrication of the heat exchanger, the different components were assembled in order to make the test setup which could be used to determine the existence of backpressure in the Brayton Cycle Turbocharger. Overall experimental setup is shown in Fig 8.

Fig. 8: Experimental setup.

In the test setup, an MS pipe with a flange welded at its end was used to guide exhaust from the 4 stroke 4-cylinder diesel engine to the evaporator side of the heat exchanger. Mechanical pressure gauges were used at the exhaust gas entry point and exit point of the heat exchanger to measure the backpressure hydraulic resistance caused due to this system of turbocharging. The compressed air to the inlet of the heat exchanger condenser section was supplied by means of an air compressor that takes drive from the engine setup available at the laboratory.

The compressed air obtained at the outlet of the heat exchanger condenser section is then further guided to the turbine which utilizes the high stagnation enthalpy of the hot compressed air outlet of the heat exchanger to produce work (rotation of turbine). This work is utilized by the turbocharger compressor side to compress the inlet air to the engine cylinders.

V. Conclusion

The experimental setup was fabricated and tested on a 4 stroke 4-cylinder diesel engine with hydraulic dynamometer loading available. The major aim of the experiment was to prove that the system of turbocharging designed would not result in excessive back-pressure that has been proven in the past to have a detrimental effect on the overall engine performance. The experiment was conducted at varying loading conditions for the hydraulic dynamometer keeping the speed of the engine constant and the backpressure was quantified by means of the pressure gauge placed at the inlet of exhaust gases to the heat pipe heat exchanger.

The results obtained during the experiment were promising as it was seen that the designed turbocharging system could function without the formation of excessive backpressure – based on the reading obtained from the pressure gauge fitting mentioned earlier.

The proposed system of turbocharging has several benefits such as:

1) Reduced back pressure – because the exhaust flow is not obstructed by the turbine. This leads to more efficient working of the turbocharger, lesser operating temperatures and less specific fuel consumption.
2) Reduced Turbo-lag - Since the system has the air compressor end (which is responsible for the circulation of working fluid) coupled directly to the engine camshaft, the turbo tends to spool up almost instantly providing required boost pressure even at low speeds of operation.

VI. Acknowledgement

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