Steam Turbocharging

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Abstract— A new concept of steam turbocharging is proposed in this project which finds a solution for the major disadvantages of conventional turbocharging systems. In conventional turbocharging system the turbocharger starts operating only after certain engine rpm. The back pressure generated at the engine exhaust is also high in conventional systems. These disadvantages are being solved in this proposed project.

The major objectives of this project are to achieve a target boosting pressure of 1.5 bar at the engine inlet, to reduce the backpressure at the engine exhaust and to increase the operating range of turbocharger. The IC engine exhaust energy is used to generate steam and then drive the turbine. Part of steam expansion power is used to drive air compressor. The heat exchanger and turbine are designed and simulated and the performance is analyzed.

The results show that IC engine power can be increased by increasing the inlet boosting pressure. The analysis shows that in steam turbocharging system, the turbocharger starts working at 1000 rpm and has less exhaust back pressure compared to ordinary turbocharger.

Keywords—*Steam Turbocharging, Computational fluid dynamic*

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I. Introduction

The power performance and fuel economy of an Internal Combustion(IC) engine can be increased by boosting the intake pressure. By increasing the intake boosting pressure the BMEP also increases which leads to higher thermal efficiency. By increasing the boosting pressure the engine displacement can also be downsized. Exhaust turbocharging is most widely used method when compared to other boosting pressure technologies. Exhaust turbocharging is a method of Exhaust Energy Recovery (EER) in which the energy of the exhaust gas is used to run a turbine which in turn runs a compressor and boost the intake air to IC engine and thus improves its performance.

In conventional exhaust turbocharging systems used in diesel engines only a part of the exhaust energy can be used efficiently the rest of the exhaust energy is wasted. This is because the exhaust gas contains mainly of thermal energy than pressure energy. In exhaust turbocharging an additional back pressure is also created which leads to less volumetric efficiency of the IC engine and more work during the exhaust stroke. Therefore some of the exhaust energy recovered is used to overcome this pumping loss. At low speeds of IC engine the exhaust energy may be less than the pumping loss, so at lower speeds the target boosting pressure may not be achieved. So exhaust turbocharging is not the best method to recover the exhaust energy or boost the intake pressure. Here we introduce a new concept which utilizes IC engine exhaust thermal energy to boost intake pressure. Since it is based on steam power cycle it is named as Steam Turbocharging, which is more effective when compared to IC engine exhaust turbocharging.

II. RESEARCH METHODOLOGY

The proposed concept of steam turbocharging is based on Rankine steam cycle. As said earlier in steam turbocharging thermal energy of exhaust gas is used instead of pressure energy. The exhaust thermal energy is recovered using heat transfer and this energy is used to generate effective work using a turbine. Finally, the output power of the turbine is used to run a compressor. Figure 1 shows the schematic diagram of steam turbocharging system. As shown in Figure 1, the steam turbocharging system consists of, valve, water tank, pump, heat exchanger, steam turbine, and air compressor, etc. Among these components, valve is used to adjust the mass flow rate of working medium water, while pump is used to control the pressure through the cycle. A motor is coupled to the transmission shaft connecting turbine and compressor to control the energy flow in the cycle. The modes of operation of motor are described as follows. (a) Driving the air compressor: at lower rpm of engine speed, the steam turbine does not work instantly and air compressor is driven by the motor to obtain target boosting pressure; since at low speeds the exhaust energy is less both motor as well as steam turbine is used to run the compressor (b) At higher engine speeds the turbine power generated is greater than required power to run the compressor so this energy is used to generate electricity from the motor.

Both the air compressor and motor is run by steam turbine so the extra energy generated by the turbine is converted to electrical energy by motor. The steam turbocharging system shown in Figure 1 is an open steam power cycle system, which can be also designed as a closed system based on its application. In closed system the water used in the cycle is used again and again whereas in open system it is used only during a single cycle. Since there is no condenser and condensation in open cycle system it is comparatively simple. The cost of open cycle system is also less when compared to closed cycle system. Steam turbocharging based on open cycle has its major application in steam power plants. The working medium that is water must be available in plenty.

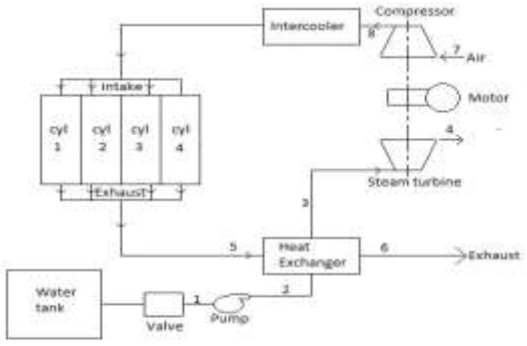

Fig.1.Schematic Diagram of Steam Turbocharging

Steam turbocharging based on closed system has wider applications since the working medium is recycled throughout the cycle. It can me both used in stationary as well as mobile applications such as in automobile and marine engines. This paper mainly deals with open cycle system and the working principle of both systems are moreover same.

2.1 MODELLING

2.1.1 DESIGN PROCEDURE OF COMPRESSOR

A compressor is designed to produce 1.5 bar boosting pressure at the engine intake. To achieve that boosting pressure we have designed a compressor using the following formula. The analysis is carried out at the engine speed of 1000 rpm.

$$\text{Air flow} = \frac{\text{Displacement of engine X Engine rpm X Volumetric efficiency}}{3456}$$

$$\text{Pressure ratio} = \frac{\text{Target boosting pressure}}{\text{Inlet pressure}}$$

$$\text{Corrected air flow rate} = \text{Pressure ratio} \times \text{airflow}$$

$$\text{Mass flow rate} = \text{Density} \times \text{Corrected air flow rate}$$

$$P_{com} = m_{in} \cdot c_p \cdot T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \frac{1}{\eta_{com}} \quad (5)$$

Where, m in is the mass flow rate of intake gas; C_P is the constant pressure specific heat of intake gas; T_1 and T_2 are the intake gas temperature at the inlet and outlet of compressor, respectively. P_1 and P_2 are the intake gas pressure at the inlet and outlet of compressor, respectively; r is the specific heat ratio of intake gas. η_{com} is the isentropic efficiency of compressor.

The power required by the compressor is thus calculated from the above equation. By analyzing the results we came to a conclusion that garret 1544 turbocharger compressor will produce the desired pressure boosting.

Table 1 Basic Parameters of Test Engine

Item	Content
Engine type	Inline four cylinder
Displacement (L)	1.573
Compression Ratio	18
Rated Power (kW/rpm)	80/4000

Table 2 Theoretically Calculated Compressor Specification

Engine RPM	1000
Mass flow rate	0.0158757
Power Required (kW)	0.82

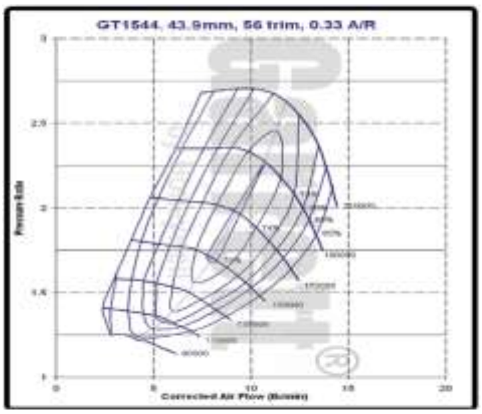

Fig 2 Compressor Chart of Selected Compressor

Fig.3. Proposed Turbocharger Compressor

Fig 3 is the picture of the turbocharger compressor we selected after the calculations. From the compressor chart in Fig 2 we can see that this compressor meets our requirements.

2.1.2 DESIGN PROCEDURE OF HEAT EXCHANGER

The exhaust gas from the engine is supplied to a heat exchanger in this proposed project. The energy of the exhaust gas is used to convert pressurized water to steam.

The steps involved in designing a heat exchanger are:

Step 1. The thermal and physical properties of hot and cold fluid is obtained and these properties are calculated at mean temperature.

Step 2. From energy balance equation obtain the energy transferred to heat exchanger.

Step 3. The value of overall heat transfer coefficient is assumed a approximate value (Uo, assm).

The assumed value of Uo can be obtained from heat and mass transfer data book.

Step 4. Determine required number of shell and tube passes (p, n).

Step 5. Calculate area of heat transfer (A) required.

Step 6. Select material of tube, select the tube diameter, its wall thickness and length of tube (L). Also calculate the number of tubes.

Step 7. Determine type of shell and tube exchanger. Select the tube pitch (Pt), decide inside shell diameter (Ds) that can accommodate the calculated number of tubes (n).

Step 8. Assign fluid to shell or tube side.

Table 3 Theoretical Design Values of Heat Exchanger

Item	Content
Area of Heat Exchanger	0.2802
No of Tubes	30
Outside Diameter of Tube	31mm
Inside Diameter of Tube	25.4mm
No of Passes	2
Shell Diameter	28.62 cm
Length of Heat Exchanger	118cm
Exhaust gas temperature at 1000 rpm	801.1 K
Exhaust mass flow rate at 1000 rpm	0.087 kg/s

In the designed heat exchanger the value of heat transfer coefficient is assumed to be 150 W/m^2. The exhaust gas is passed through the tube side and the pressurized water is passed through the shell side.

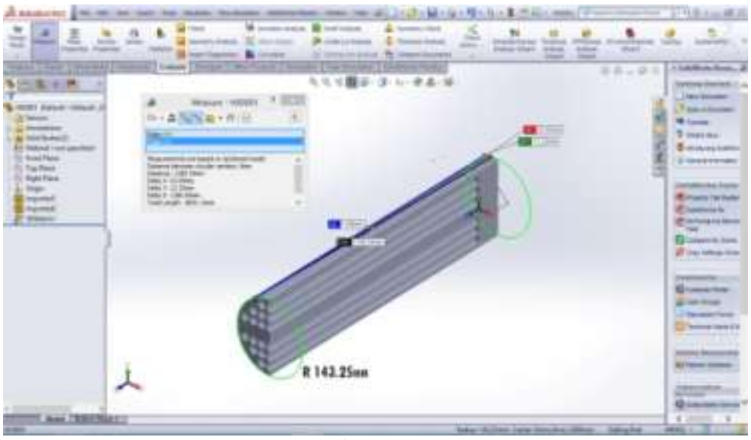

Fig.4. Solidworks Model of Designed Heat Exchanger

2.1.3 DESIGN OF REACTION TURBINE FOR POWER GENERATION

The pressurised water which is fed into the heat exchanger is turned to steam by using the exhaust gas energy from engine. The steam at 5 bar pressure from the outlet of heat exchanger is directly fed to a reaction turbine to generate power required to run the compressor.

The design of reaction turbine is done using ANSYS Vista RTD software. The geometry of turbine blade is generated using vista RTD and it is developed using BladeGEN of ANSYS. The analysis of the turbine blade is done using ANSYS Fluent 15. The model of the turbine blade developed through Vista RTD is shown in the Figure 5.

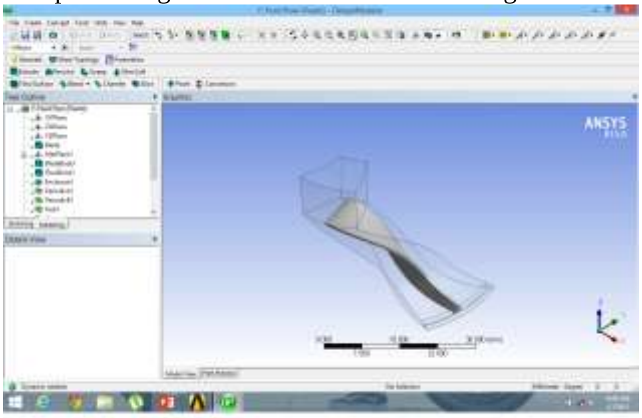

Fig.5. Turbine Blade Geometry

Figure 5 shows the geometry of turbine blade which is designed using Vista RTD. The envelop over the blade geometry shows the flow path or control volume. The blade is so designed that the steam gets expanded to a pressure of 1 bar at the outlet of the turbine.

III. RESULT AND DISCUSSIONS

The heat exchanger and turbine analysis were done on ANSYS Fluent 15.0. The results are shown below:

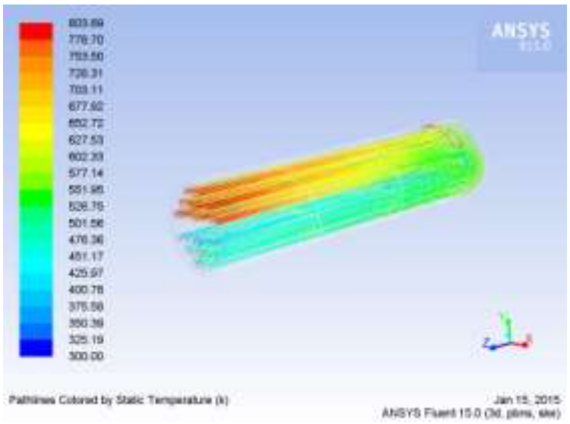

Fig 6 Tube side Temperature Distribution

The Figure 6 shows the temperature distribution of tube side fluid. The temperature of exhaust gas at tube inlet is around 801K and when it comes out of heat exchanger its temperature decreases to 450K. Figure 7 shows the shell side temperature distribution of same heat exchanger. In the shell side water enters at around 303K and leaves at around 423K as steam. The water is supplied to the heat exchanger at a pressure of 5bar using a pump.

Figure 8 shows the contours of static pressure in a turbine blade. The steam enters at a pressure of 4.7 bar and expands to around 1 bar pressure. There are a total of 9 blades in the runner designed to generate required power.

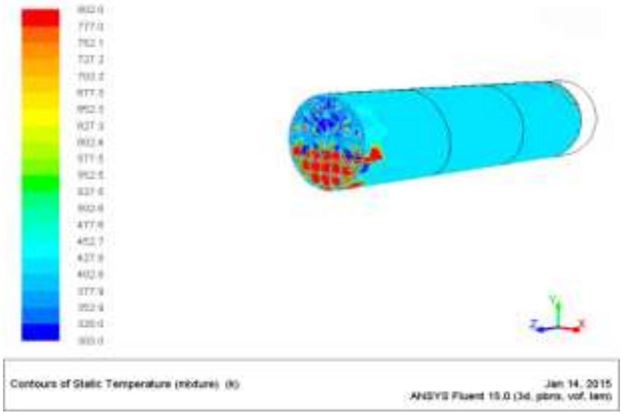

Fig 7 Shellside Temperature Distribution

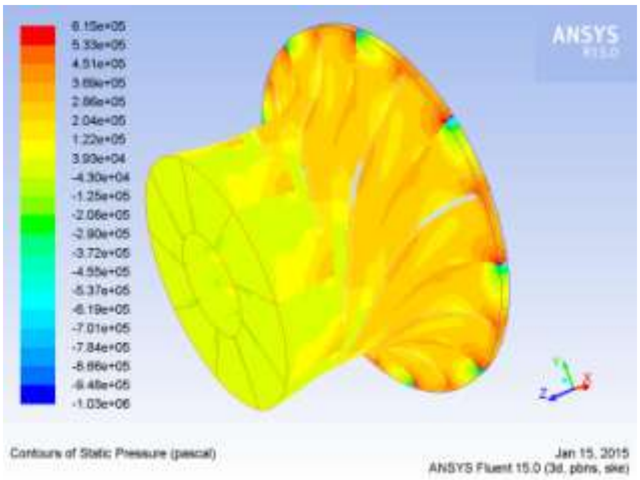

Fig 8 Contours of Static Pressure in Turbine Blades

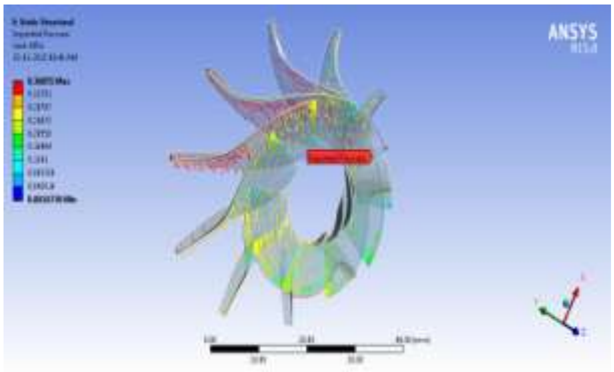

Fig 9 Forces on Turbine Blade

During the analysis of heat exchanger, it was found that there is a pressure drop of 300 Pa and the outlet temperature and pressure of steam from heat exchanger outlet was found to be 423K at 4.7 bar pressure. This steam is expanded through a steam turbine and the analysis result showed that the steam expanded to a pressure of 1 bar, thus a total power of 0.86kW was generated by the turbine which is sufficient to run the compressor and the pump which pressurizes the water at heat exchanger inlet.

From the analysis we found that the temperature at shell inlet is 303 K and shell outlet is 430 K. The maximum force acting on turbine blade is 0.36885 MPa.

IV. CONCLUSIONS

From the analysis it was found that even at 1000 rpm engine speed, the power required to run the compressor is generated by the turbine thus a boosting pressure of 1.5 bar is produced at the engine inlet. By using this proposed system, the turbocharger starts to operate at 1000 rpm, thus the major disadvantage of existing turbocharger systems is solved. Since this system uses exhaust gas temperature, more thermal efficiency is obtained in steam turbocharging.

In steam turbocharging system, since the exhaust gas is directly fed to a heat exchanger and not to the turbine as in conventional systems, the exhaust back pressure is considerably reduced. This increases the volumetric efficiency of the engine. The heat exchanger doesn't hinder the flow of exhaust gas through it, thus the exhaust gas experiences less resistance when flowing through the heat exchanger.

The engine air inlet pressure is maintained at 1.5 bar pressure in this proposed system, which increases the volumetric efficiencies as well as the overall efficiencies of the test engine. By supplying air at this pressure, the combustion takes place completely inside the engine and more power is developed, thus it is seen that by using steam turbocharging system all major disadvantages of conventional turbocharging systems is solved.

Acknowledgment

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