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Improved Heat Transfer Correction Method for Turbocharger Compressor Performance Measurements

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Improved Heat Transfer Correction Method for Turbocharger Compressor Performance Measurements

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Abstract. The compressor adiabatic performance map is important for turbocharger matching with engine. Its accuracy was affected by the heat transfer phenomenon in the turbocharger. The corrected method for compressor performance measurement which affected by heat transfer was widely investigated by some authors. The aim of this work was to measure compressor adiabatic efficiency by simplified method, and estimated the effects of internal heat transfer on compressor efficiency. This approach only added five temperature sensors on the turbocharger based on the standard instruments and doesn't need other special instruments, and the heat transfer properties of turbocharger and compressor adiabatic efficiency could be calculated by lumped model. This method was validated by other experiments. Finally, the heat absorbed by compressor under different operating conditions was analyzed by this method. The results showed water-cooling was main influenced factor, and water-cooling was a barrier for heat transfer from turbine to compressor..

1. Introduction

Turbocharger plays an dominant part in emission and energy reduction of road vehicles. The precise adiabatic efficiency of compressor is very important for engine simulating at the stage of engine design. The efficiency of compressor which was measured at standard test bench is not exacted due to heat transfer in turbocharger [1]. The traditional way to calculate the compressor efficiency, based on a ratio of adiabatic and actual of compressor enthalpy from measurements of a test-bench showed in Eq.1, the detailed about Eq.1 was presented in the reference [1], provides inaccurate results. This inaccuracy comes from the no consideration of heat fluxes effect in the turbocharger [1]. In order to corrected results of measured, a lot of experts and scholars did a great deal of researches for behavior of turbocharger heat transfer in the last few decades [2].

$$\eta = \frac{\triangle h_{is,C}}{\triangle h_{ac,C}} = \frac{C_{P,air}T_1(\Pi^{(k_{air}-1)/k_{air}}-1)}{C_{P,air}(T_{2,adi}-T_1)} \tag{1}$$

One dimensional lumped model of turbocharger heat transfer was studied by some authors. Turbochargers were simplified and divided into different one-dimensional nodes in the lumped model, convective or thermal conductance were used to connected different nodes of turbocharger [3]. The

model of calculation was validated by some experiments. The modeling methodology of one dimensional lumped model was given by Reyes-Belmonte et.al [4] and Shaaban et.al [5] .The heat transfer property of turbocharger was needed to measure by some particular instruments. The special oil as work medium was used in turbine and compressor in order to measure heat transfer properties of turbocharger. Turbocharger was often independent tested on test bench. In Ref. [6], a turbocharger was installed on a diesel engine and experiments performed at different engine speeds and loads. The test results showed that the temperature of the engine exhaust gases has a large impact for turbocharger internal heat transfer.

The lumped model was analyzed and validated by some researchers, and some other authors applied this method to correct the turbocharger performance of measurement[7-14]. The special and expensive instruments were required in the lumped model solving process.

The measured procedure of compressor adiabatic efficiency that applied in the previous studies is complex. Some simple and exact method was needed to investigate. The exacted and simplified methodology to measure adiabatic efficiency for compressor was presented in this paper. Experimental tasks was described in the first part, focusing on the description of sensors and equipment used, the testing methodology and the analysis of testing results were also reported in this part. The second part of this paper is focusing on validation for the testing methodology. The quantities of heat absorbed by compressor was analyzed for different operating conditions. The main conclusions of this work are summarized in the final part of this paper.

2. Experiment investigation

2.1. Test rig

The turbocharger used in this study was for gasoline engine, with turbine and compressor wheel diameters of 37 mm and 44 mm respectively. The experimental activity was developed at Shouguang Kangyue Turbocharger company. A schematic of the test rig is presented in Figure 1. This test bench is able to simulate the conditions of engine intake and exhaust system. The mass flow rate of 0.02-1.0 kg/s fresh air supplied by compression engine to turbine. The compressor sucks fresh air from environments and the operating states were set by valve (up and down stream of compressor) controlling. The hot air of turbine was heated by neutral gas in the combustion chamber. An combustor allows performing experimental investigations with turbine temperature up to 1100℃. An independent lubrication and water supply system is installed. System inlet pressure and temperature could be independent modified. The efficiency of compressor is calculated from the temperature of inlet and outlet, and the turbine efficiency is calculated by compressor power consumption. Data acquisition and actor controlling of this test rig was performed by NI (National instruments) hardware. The software of this test rig is developed by the company of KRATZER. Efficiency and other main operating parameters of turbocharger was recorded by automatic measured system of test rig.

The static air pressure was estimated by the piezo-resistive transducers with an accuracy of ±0.3% of full scale. The air temperature was measured by K type thermocouples (accuracy ± 0.4% of full scale). The thermal mass flow meter with an accuracy of ±1% at full scale was used to measure fresh air mass flow rate of compressor and turbine. The turbocharger rational speed was measured by an sensor (ACAM PICOTURN-BM) installed on the compressor volute and was able to detect the compressor wheel revolution with an accuracy of ±0.2% of full scale.
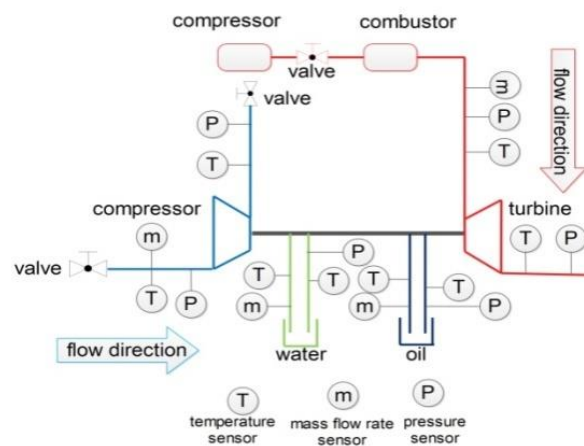
Figure. 1 Layout of turbocharger test bench

Additional temperature sensors were installed on components of turbocharger due to the process of turbocharger heat transfer is complex. The static temperature of nodes were measured by these sensors. Values of those sensors measured were used to reflect temperature distribution of turbocharger under different conditions, the heat transfer of model was validated by those values to a certain extent. In order to measure temperature of nodes C, three K type thermocouples (accuracy ± 0.4 % of full scale) were installed at the volute compressor. Two metal temperatures were also measured on the bearing casing (nodes of H1 and H3), one nearing the turbine casing and another nearing the compressor casing. Details of the thermocouple installation are given in Figure 2 and Figure 3. The temperature of turbine surface was measured by infrared thermometer (accuracy ± 1.5% of full scale) due to K type thermocouple was difficult to fix on the high temperature surface.
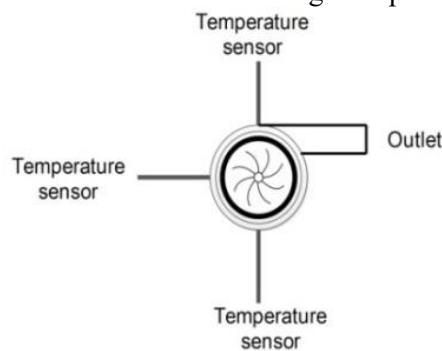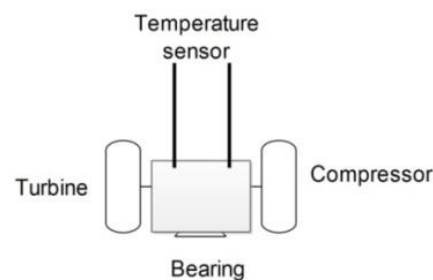

Figure 2. Compressor volute   **Figure 3.** Bearing casing

2.2. Experimental study
All performed measurements were processed in order to obtain compressor adiabatic-efficiency (AE). The definition of AE as seen in Ref [12]. To ensure the accuracy of AE, the heat transfer was considered in the $T_{2,adi}$ of measured, $T_{2,adi}$ is an important parameter for AE, these shown in Eq.(2)-(3):

$$T'_{2,adi} = T_{2,adi} - \triangle T \tag{2}$$

$$m * C_{P,air} * \triangle T = Q_L \tag{3}$$

where: $T'_{2,adi}$: corrected temperature of compressor outlet;
$\triangle T$: temperature difference between measured and corrected;

m: mass flow rates of compressor;

Q_L: heat of compressor obtained;

In order to set and validate the model of proposed, main operating parameters of turbocharger were measured under different conditions. The compressor obtained heat was affected by compressor and lubricating system operating states. To evaluate heat transfer affected by compressor operating states, the compressor work at different rotational speed and compression ratio was performed in the experiments. The lubricating states is also related to heat transfer from turbine to compressor, so measurements were performed for oil at different inlet temperature and pressure. The turbocharger was run with and without water in the experiments due to water is important factor for compressor obtained heat from turbine. In order to woken the heat transfer in the turbocharger, the water inlet temperature maintain the same as oil inlet temperature. The turbine inlet temperature plays an important role in the turbocharger internal heat transfer, three different inlet temperature (500℃, 600℃, 700℃) of turbine were given under the other conditions maintained constant.

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2.3. Experimental results

In order to evaluated the effects of heat transfer in the turbocharger, the section of experimental results will be make comparing for results with or without water cooling.
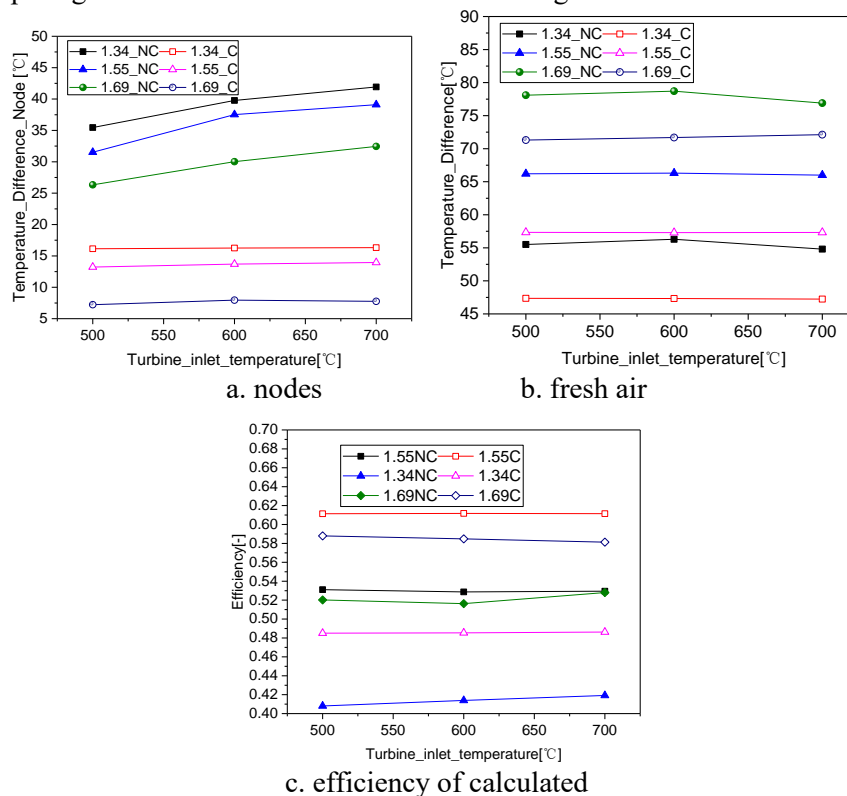

a. nodes   b. fresh air

c. efficiency of calculated

Figure 4. Temperature difference cooling or no cooling

Difference value of temperature between nodes of C and H3 was presented in Figure 4 a (NC is no cooling, C is cooling). It's shown that temperature difference between nodes of C and H3 was affected by the compression ratio, air temperature of turbine inlet and water cooling states. It was direct proportion to air temperature of turbine inlet, inverse ratio to compression ratio. This phenomenon was caused by internal heat transfer of turbocharger. The quantity of heat obtained by compressor can be directed reflected by the temperature difference between nodes of C and H3, so the heat transfer from turbine to compressor is higher when compression ratio decrease and turbine inlet temperature increase. That heat flux will be partly removed by the coolant liquid in case it exists, so the difference is higher at the no water cooling. b and c of Figure 4 show difference value of fresh air temperature between compressor inlet and outlet and efficiency of calculated at different conditions. Compressor adiabatic efficiency was determined by rational speed and compression ratio of compressor. The traditional way to calculate the compressor efficiency, based on compression ratio and outlet temperature of compressor, therefore, the same trend was observed in these two pictures. More heat was absorbed by compressor under no water cooling due to water is the barrier of heat from turbine to compressor. The value of no-cooling is larger than cooling. The compressor outlet temperature and efficiency increased when compressor compression ratio increases and it will cause the temperature difference between C and H3 decreasing, and the heat transfer from turbine to compressor was woken by this phenomenon, therefore, the heat of compressor absorbed is lower when compressor work at higher compression ratio operating points. From above analysis, the heat transfer can be separate reflected by two different parameters of difference temperature of nodes C and H3 and difference value of fresh air temperature between compressor inlet and outlet. The error of calculated efficiency was lead by heat transfer in the turbocharger. The heat absorbed by compressor can be calculated by solved equation about k and temperature difference of nodes C and H3, difference value of fresh air temperature between compressor inlet and outlet. The more detailed solution process was described in the section 3.

3. Turbocharger thermal model

In order to simplified the complex heat transfer phenomena in the turbocharger, heat has been assumed extracted from the exhaust gas of turbine to the fresh air in the compressor before it is expanded in the turbine stator (see Figure 5). Water and oil is the barrier to prevent heat transfer from turbine to compressor. Turbocharger was simplified into different nodes. The bearing casing was divided into three part in order to make sure accuracy of model. The heat transfer in the turbocharger is very complex and its affected by the water and oil operating states, the energy obtained from surroundings was omitted due to the surface of compressor is small in area and coefficient of heat transfer by free convection is little [13], therefore, the heat obtained by fresh air in compressor equalled to volute of compressor transfer from bearing casing due to the energy balance. The heat of compressor obtained is sampled to nodes H3 transfer to C and heat absorbed by compressor was calculated according to Eq.(4.) The temperature of nodes were measured by thermocouples. Details of the thermocouple installation are given in Figure 1. In order to solve the value of heat transfer coefficients (K) , the different heat transfer conditions were given under compressor operating at the same rotational speed and compression ratio (see Figure 5). The form of equation set was presented as Eq. (5)-(6). Because compressor adiabatic efficiency only determined by rotational speed and compression ratio, heat transfer coefficients (K) of bear casing to compressor volute can solved by Eq. (5)-(6) and not need some specific test bench, the value of K is 0.03467818 (unit was omitted and the unit of other variable is ISO). Q is the heat of turbine released or compressor obtained.
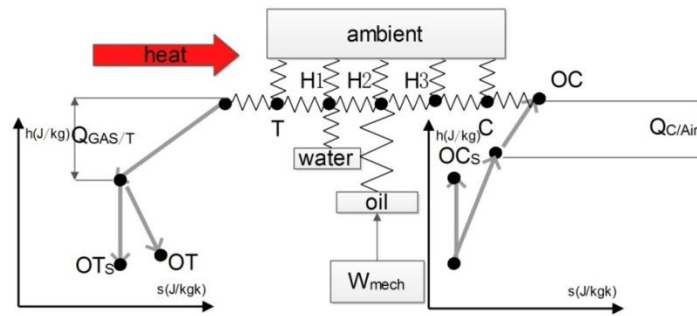
Figure 5. Schematic of turbocharger heat transfer

$$Q_{C/Air} = (T_{H3} - T_C) * K_{H1/C} \qquad (4)$$

$$Q = (T_{H3} - T_C) * K_{H1/C} \qquad (5)$$

$$Q' = (T'_{H3} - T'_C) * K_{H1/C} \qquad (6)$$
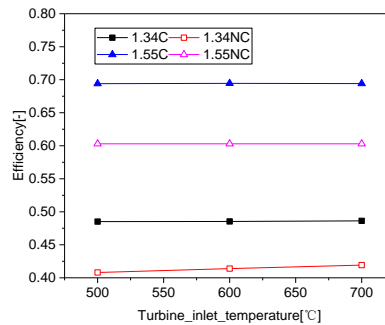
4. Model validation

Figure 6. Efficiencies of no correction

Table1. Efficiency of correction

Turbine_inlet_T[°C]	1.34NC [-]	1.34C [-]	1.55NC [-]	1.55C [-]
500	0.556487974	0.556488	0.699807	0.699807443
600	0.556487974	0.556488	0.699807	0.699807443
700	0.556487974	0.556488	0.699807	0.699807443

The validation for corrected method was reported in the below. The comparison of efficiency at with or without cooling, different compression ratio and turbine inlet temperature is given in Figure 6. The compressor adiabatic efficiency was only determined by the rotational speed and compression ratio of compressor under ideal condition. Compressor adiabatic efficiency measured in hot condition is less than the cooling condition due to heat transfer in the turbocharger. The value of measured compressor adiabatic efficiency also was affected by other operating (e.g. oil operating states, turbine inlet temperature, rotational speeds of turbocharger) and geometry parameters. The heat transfer of turbocharger also was affected by these parameters. The discrepancy of corrected results between with and without cooling is less than 1e-4 as shown in Table 1. The discrepancy was shown in the Table 1 was caused by four main factors. First of all, some heat dissipated by bearing casing was omitted in corrected process. Second factor contributing to the overall discrepancy is that, some compressor absorbed heat of radiated by turbine was also neglected in corrected results. Third discrepancy influenced factor is that, compressor adiabatic efficiency has tiny difference when fresh air of compressor has different temperature state. Another overall discrepancy influenced factor is that, some measured error due to the temperature sensor was connected with turbocharger by high temperature

glue and some other uncertainty factors in the measured process. This method can be improved in four sides of expressed above.

5. Conclusion

Traditionally, heat losses have been neglected in the compressor map measured process and the behaviour of the compressor has been predicted by direct use of manufacturer maps to interpolate without any corrected, therefore, the accuracy of map can't reach up to the requirement of engine design.

This paper reported the exacted methodology to measure efficiency and heat absorbed for compressor. This method isn't complex and can be performed at standard test bench of turbocharger .The following main conclusions can be drawn:

(1) Water cooling significantly impacts heat transfer between the compressor and turbine for gasoline turbocharger. Temperature of water is more close to temperature of compressor than turbine operating temperature. Qc of without water-cooling is three to four times for with water-cooling.

(2) The improved method gives exact measured results of adiabatic efficiency and Qc of compressor that only adding five temperature sensors based on standard test bench of turbocharger. Those sensors were used to measure temperature of nodes C and H3.

(3) Less quantities heat transfer from turbine to compressor at higher rational speeds due to the work range of compressor move to high compression ratio direction. The average temperature of fresh air in compressor was increasing when compression ratio increase. The temperature difference between nodes C and H3 was woken. Less quantities of heat was absorbed by compressor at higher compression ratio corresponding the same reason.

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