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Effect of adding ceramic thermal barrier coating on the turbocharger efficiency, external and internal heat transfer

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Abstract

Turbocharger is a device installed on an internal combustion engine to boost its thermal efficiency. A turbocharger consists of three main components, namely the turbine, central housing, and compressor. The common material for commercial turbine housing is cast iron, for its lower cost yet resilience at elevated temperature. Given the high exhaust temperature a turbocharger is exposed to, energy loss in the form of heat transfer is inevitable. It is known to H turbine efficiency by up to 30%. This research aims to determine the turbocharger efficiency in the presence of thermal barrier coating (TBC) on the inner surface of turbine volute. Particularly, this work will focus on the internal and external heat transfer of the turbine and its impact on efficiency. The subject turbocharger is a commercial single-scroll vaneless unit commonly used in gasoline passenger vehicle. Yttria-Stabilized Zirconia (YSZ) is chosen as the TBC material, due to high melting point (around 2700°C), good thermal insulation property and very low thermal expansion compared to other ceramic materials. The YSZ was applied to the inner surface of turbine volute via plasma coating technique. However, due to the large disparity in thermal expansion between YSZ and cast iron, the TBC is prone to cracking at elevated exhaust temperature. Thus, an Inconel 718 turbine housing, with closer thermal expansion to YSZ, was refabricated for the use of this study The turbocharger performance was experimentally measured on the LoCARtic turbocharger gas stand. The turbine inlet temperature (TIT) was varied at 150, 350, 550, 650 and 750°C, while the compressor operating condition was maintained throughout the testing for equivalent comparison. From the result, the turbocharger efficiency drops when TIT is increased, and the turbine pressure ratio becomes lower. Overall, the external heat transfer loss is found to reduce 7% to 40% and no significant difference noticed on the internal heat transfer.

1. Introduction

Currently, many car manufacturers are using turbocharged engines to boost its thermal efficiency. However, these engines are subject to very high temperatures and corrosive exhaust gases. Heat transfer becomes a major issue due to this extreme temperature, so it is important to select a suitable coating material that is highly durable, as it will affect the engine-turbo matching [1]. Engine downsizing is the current activity of the car engine designer in improving power weight ratio of the internal combustion engine. There are several ways in which this can be done, one of the ways is by improving on material of the components. In addition to this, the designer also working on the environmental friendly performance. One component that can be studied as a prototype research is on the 'Turbocharger Turbine Volute Casing' which to be designed and fabricated using ceramic, many have been done on the ceramic turbine rotor blade but not on the turbine volute.

Turbocharger nowadays has been remarkably proven as the engine downsizing method with increases the power density [2], reduces the engine displacement [3], consequently decreases the cost and weight as well as the mechanical friction losses [4,5]. Despite of the benefit, turbocharger suffer a heat loss during the operation. Hence to solve this, a low thermal conductivity and applying thermal barrier coating (TBC) was proposed to solve this problem.

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Materials can be categorize as thermal barrier coating (TBC) when it possess high melting point, high thermal durability, low thermal conductivity and chemical activity and no phase changes between room and operation temperature [6]. TBC initiate to minimise the thermal stress and reflex much of radiant heat from hot gas [7]. TBC acts as a heat insulation to protect the underlying metal component working in high temperature environment. It has been used for various engineering components such as piston surface, engine head and valves [8] and also turbine blades [9]. By insulating the hot components with TBC, the heat loss are reduced, hence increased the thermal efficiency of the engine [10,11].

In terms of heat transfer effect, the influence of TIT towards compressor efficiency is significant at low turbocharger speed and at high speed, the heat transfer effect can be neglected [12]. The gas flowing heats up the turbines walls and the wall transfer the heats to fresh air in compressor, coolant, lubricant and ambient. However, with the ceramic coating, it reduce the heat through turbine wall [13]. An insulated turbocharger with silicate insulation material can reduced heat flux by 5% to 70% compared to standard turbocharger and improved the fuel consumption by 3% [14]. Ceramic material will also tremendously reduce the surrounding high temperature, thus will protect material of the surrounding components such as cables, rubber materials and piping.

2. Methodology

A turbocharger experiment and heat transfer modelling were conducted to measure and evaluate the performance of a turbocharger and heat transfer effect with uncoated turbine volute and coated turbine volute. The volute was fabricated using rapid prototyping Inconel 718 as main material and Yttria-stabilized zirconia (YSZ) act as the thermal barrier coating layer (TBC).

Experiment was conducted with a constant compressor loading across all conditions. Four different speed at 60 kRPM, 90 kRPM, 105 kRPM and 120 kRPM were tested for the turbocharger. The coolant temperature fixed to 80 °C and 90 °C. The turbine inlet temperature (TIT) were varied from the cold flow testing to maximum at 750 °C.

The test start with cold flow performance evaluation. Assuming the cold flow testing is the zero-heat loss condition. In order to achieve this condition, turbine outlet temperature (TOT) was set to a constant value 30 degC across all speed. This to prevent condensation at the turbine outlet and heat transfer effect from the turbine inlet. Maximum TIT tested is 750 °C because that is the engine operating conditions. In order to see the HT effect of different TIT, the TIT was varied from 150 °C, 350 °C, 550 °C and 650 °C.

2.1. Thermal Barrier Coating

Thermal-barrier coating (TBC) systems are worked at very severe conditions, combining high temperatures, oxidative and corrosive environmental, additional mechanical loading as well as rapid temperature transients. The thermal properties of that coating layer will be changed with different temperature and time because of microstructure evolution and inter diffusion. Inconel 718 were used to fabricate the turbine housing and YSZ were used in this research to make that TBC inside the turbine housing. TBC thickness set to 300 μm based on the previous research[15] and applied in the inner volute because 80.4% of the heat transfer occur in volute and only 17.3% at diffuser [16]. Uncoated turbine casing wall thickness is 4.0 mm and for coated volute wall thickness is 3.7 mm, an offset of 0.3 mm (300 μm) for the coating. Figure 1. shows the final product of the turbine housing with internal coating. The turbine casing made into two pieces for ease of coating process.

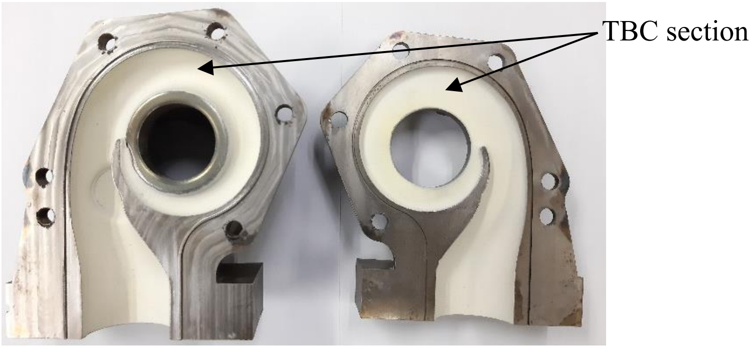


Figure 1. Final product of TBC on turbine casing.

2.2. Turbocharger Experiment

Figure 2. shows the schematic diagram of the turbocharger test rig. Two orifice plate were used to measure the turbine and compressor mass flow rate. Turbine air flow delivered from the screw type air compressor machine with maximum flowrate 0.1 kg/s. Turbine air flow regulated using one way control valve. A combustor was used for hot test and a heater used for cold flow testing. Nine pressure transducers with accuracy ± 0.04% were used to measure the pressure on specific location as shown in figure 2. Three K-type thermocouple were used in turbine side and seven T-type thermocouples used on compressor side, coolant, and lubricant, respectively.

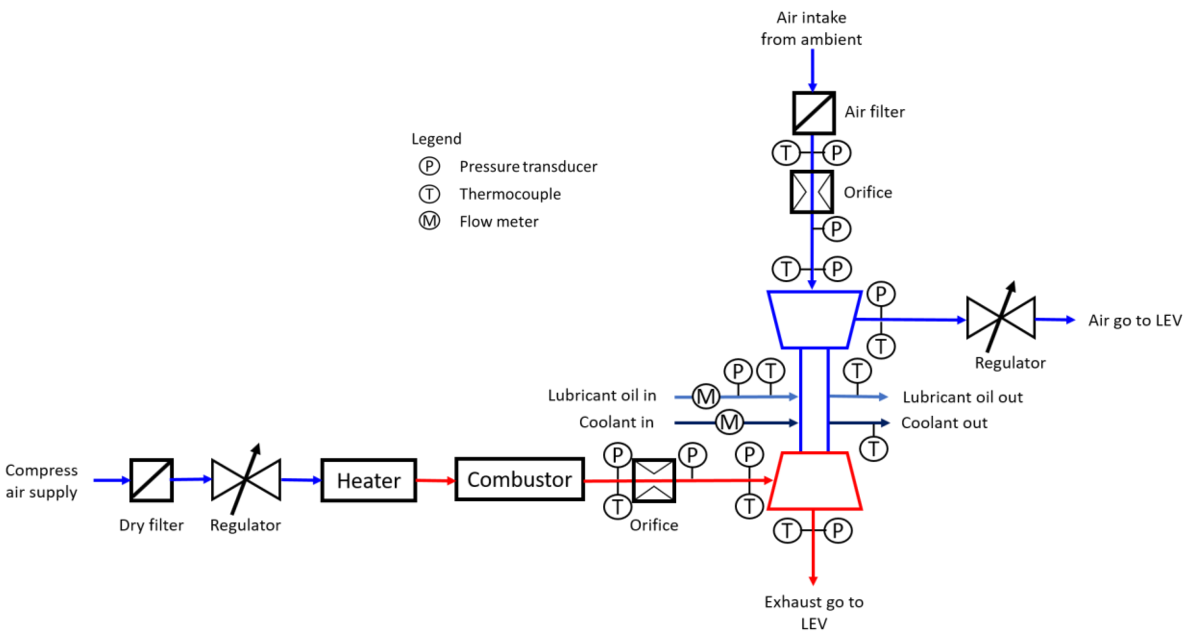
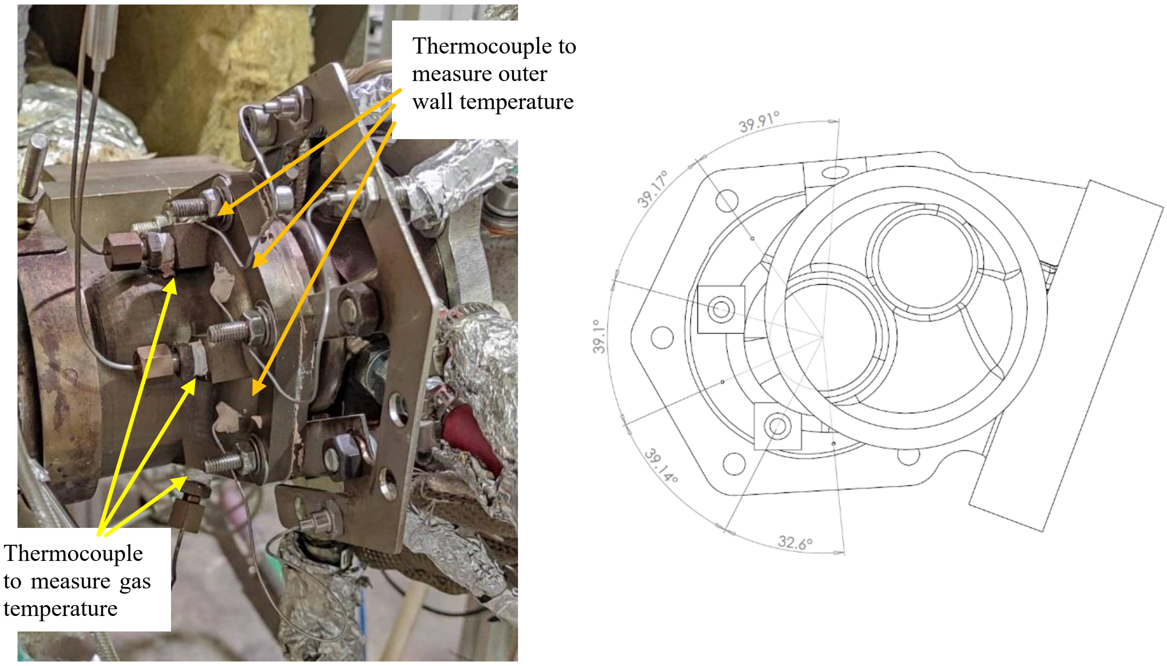


Figure 2. Turbocharger test rig schematic diagram.

To compute the heat transfer across turbine casing, six K- type thermocouples were mounted on the turbine casing, three measure gas (fluid) temperature and three measure the outer wall temperature as shown in figure 3a. and figure 3b. Thermocouple location was determine from the centre of the turbine rotor. All thermocouple locations are equal spacing, except one point because space constraint. Another two surface thermocouples use to measure temperature of bearing housing and compressor housing, respectively. All thermocouples are class 1 accuracy. Since the volute is refabricated, a special thermocouple mounting was designed together with the turbine housing to measure the gas (fluid) temperature along the volute. The thermocouple was positioned.



a) Experimental setup b) Thermocouple location

Figure 3. Thermocouple setup to measure gas temperature and outer wall temperature.

Compressor mass flow parameter (MFP) and turbine MFP was calculated using the equation (2.1) and (2.2) while the compressor pressure ratio (PR) and turbine PR was calculated using equation (2.3) and (2.4). Turbocharger performances were calculated using equation (2.5), ratio of compressor isentropic power to turbine isentropic power.

$$MFP_c = \dot{m}_c \frac{\sqrt{T_{0,1}}}{P_{0,1}} \tag{2.1}$$

$$MFP_t = \dot{m}_t \frac{\sqrt{T_{0,3}}}{P_{0,3}} \tag{2.2}$$

$$PR_c = \frac{P_{0,2}}{P_{0,1}} \tag{2.3}$$

$$PR_t = \frac{P_{0,3}}{P_4} \tag{2.4}$$

$$\eta_{turbocharger} = \frac{\dot{m}_c \cdot c_{p,c} \cdot T_{0,1} \left[PR_c^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\dot{m}_t \cdot c_{p,t} \cdot T_{0,3} \left[1 - \left(\frac{1}{PR_t} \right)^{\frac{\gamma-1}{\gamma}} \right]} \tag{2.5}$$

2.3. *Heat Transfer Modelling*

External and internal heat transfer modelling was based on the proposed method by Baines et al. 2010 [17]. Figure 4. shows the external heat transfer diagram. Three K-type thermocouple were used to measure fluid temperature and another three K-type thermocouples to measure outer wall temperature.

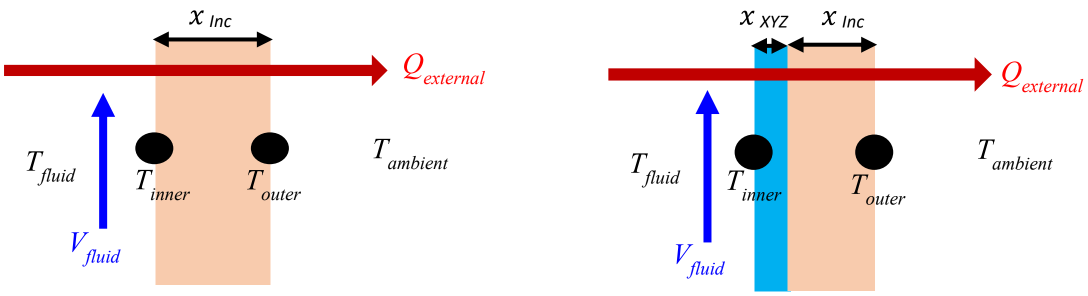


Figure 4. External heat transfer diagram.

Fluid temperature was measured because it drives the external heat transfer and internal heat transfer. Fluid temperature referring to the gas temperature inside the turbine volute. Considering a 1D heat transfer through housing, hence equation (2.6) used to determine external heat transfer for uncoated and equation (2.7) used to determine external heat transfer for coated. Additional variable was added because present of TBC. The fluid temperature was determined from average of three thermocouple measure the fluid temperature and outer temperature an average from three thermocouple measure the outer surface temperature. The thermal properties for Inconel 718 get from the research paper [18] and YSZ thermal conductivity is 1.0 W/mK [19].

$$Q_{ext\ uncoated} = \frac{A \cdot \left(T_{fluid} - T_{outer}\right)}{\frac{1}{h_{fluid}} + \frac{x_{\mathrm{Inc}}}{\kappa_{\mathrm{Inc}}}} \tag{2.6}$$

$$Q_{ext\ coated} = \frac{A \cdot \left(T_{fluid} - T_{outer}\right)}{\frac{1}{h_{fluid}} + \frac{x_{\mathrm{Inc}}}{\kappa_{\mathrm{Inc}}} + \frac{x_{\mathrm{YSZ}}}{\kappa_{\mathrm{YSZ}}}} \tag{2.7}$$

Convective heat transfer coefficient was determine using relationship between equation (2.8) and (2.9). The L in equation (2.8) and (2.10) refer to the streamwise distance used to measure the growth of the boundary layer through convection occurs. The arbitrary constant value of a, b and c was referring to table 1. The dynamic viscosity determined using the Sutherland Viscosity Law as shown in equation (2.12)

$$h = \frac{Nu \cdot k}{L} \tag{2.8}$$

$$Nu = a\, Re^{b} Pr^{c} \tag{2.9}$$

$$Re = \frac{\rho \cdot V \cdot L}{\mu} \tag{2.10}$$

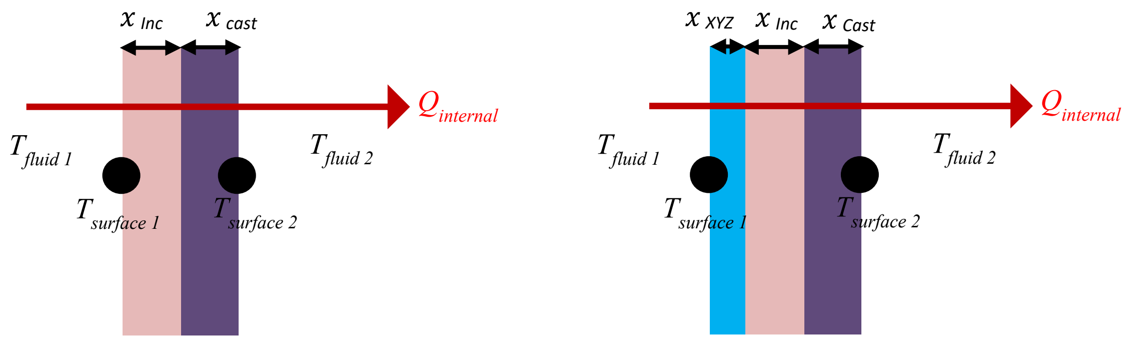
$$Pr = \frac{c_p \cdot \mu}{\kappa} \tag{2.11}$$

$$\mu = \frac{C_1 T^{\frac{3}{2}}}{T + C_2} \tag{2.12}$$

Where $C_1 = 1.458 \text{ x } 10^{-6}$ kg/msK$^{\frac{1}{2}}$ and $C_2 = 110.4$ K.

Table 1. Constant for convective heat transfer coefficient [17].

Forced convection constants	*a*	*b*	*c*
Turbine housing inner surface	0.032	0.7	0.43
Oil	0.040	0.8	0.43



a) Uncoated b) Coated

Figure 5. Internal heat transfer schematic diagram.

Figure 5. shows the internal heat transfer schematic diagram. The T_{fluid1} refer to the hot fluid gas temperature inside turbine volute (source temperature) and the T_{fluid2} refer to lubricant inside the central housing (sink temperature). T_{fluid2} is the average of the lubricant temperature between inlet and outlet oil stream. The turbine housing and central housing is two different materials, therefore distance was divided into two. Thermal conductivity for cast iron is 50 W/mK [20] and assume to be constant across different TIT. Internal heat transfer was calculated using equation (2.13) for uncoated and equation (2.14) for coated. Convective heat transfer coefficient method determines same with the equation (2.9). For the coolant effect, T_{fluid2} in equation (2.13) and (2.14) replace with the T_{fluid3}. Hence for internal heat transfers, the effect of lubricant and coolant were total together using equation (2.15).

$$Q_{int\ uncoated} = \frac{A \cdot (T_{fluid1} - T_{fluid2})}{\left(\frac{1}{h_{fluid1}} + \frac{x_{\text{Inc}}}{\kappa_{\text{Inc}}} + \frac{1}{h_{fluid2}} + \frac{x_{\text{cast}}}{\kappa_{\text{cast}}}\right)} \tag{2.13}$$

$$Q_{int\ coated} = \frac{A \cdot (T_{fluid1} - T_{fluid2})}{\left(\frac{1}{h_{fluid1}} + \frac{x_{\text{Inc}}}{\kappa_{\text{Inc}}} + \frac{x_{\text{YSZ}}}{\kappa_{\text{YSZ}}} + \frac{1}{h_{fluid2}} + \frac{x_{\text{cast}}}{\kappa_{\text{cast}}}\right)} \tag{2.14}$$

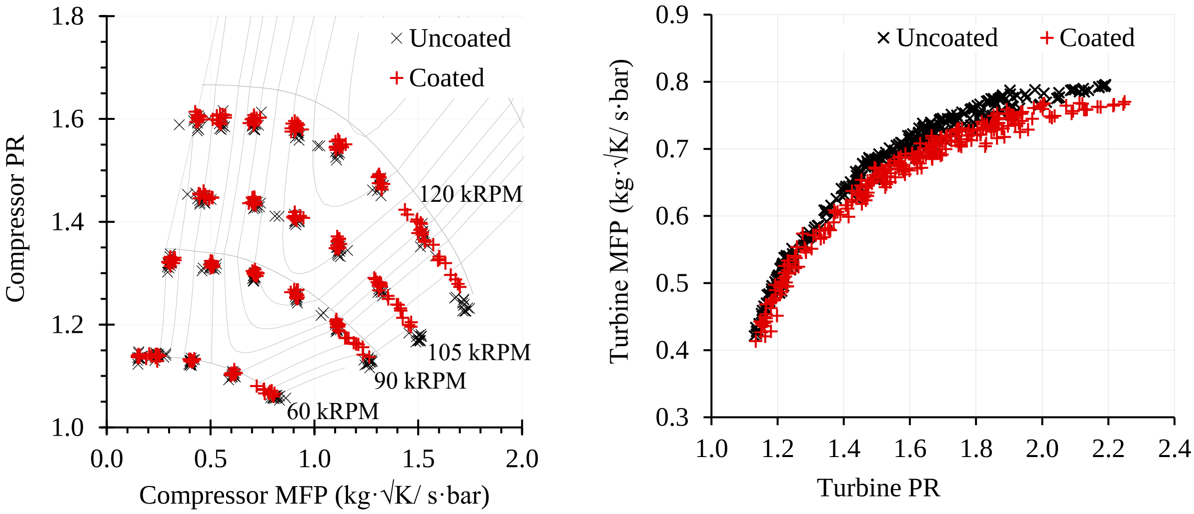
$$Q_{int\ total} = Q_{int\ lubricant} + Q_{int\ coolant} \tag{2.15}$$

3. Result and Discussion

3.1. Turbine and Compressor Map

Figure 6. shows the tested results plotted on the compressor and turbine maps. For both coated and uncoated testing sets, the testing points are done in the mean of maintaining similar compressor operating condition in both cases. The compressor condition was kept constant by maintaining the compressor PR, MFP, reduce speed and compressor speed. Compressor must be kept constant, to ensure the variable created by the coated and uncoated turbine can be identify and differentiated.

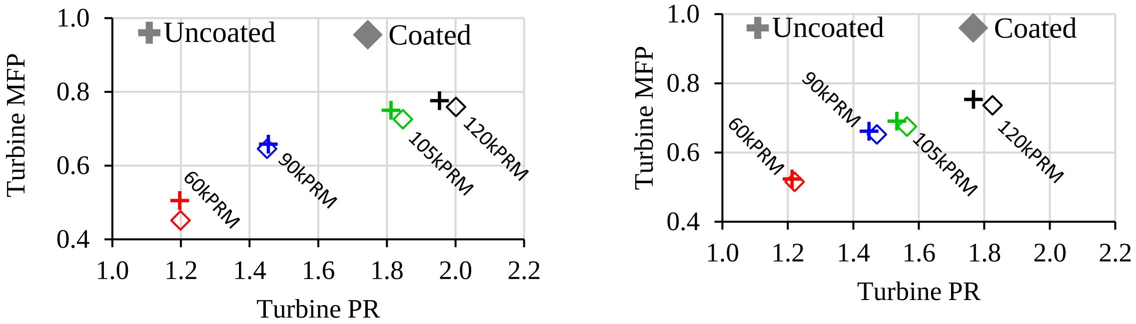
It can be seen in figure 6a. where the tested points of both cases well overlapping each other across 4 constant speed lines. Therefore, the loadings of the turbocharger of both coated and uncoated cases at the similar operating conditions are maintained. Under the similar turbocharger loading, it is found that there is a slight shift of turbine coated case's operating points in comparison with the uncoated case's as indicated in figure 6b. Under the same turbocharger loading conditions, the turbine operating points of the coated case have lower MFP and higher PR. This trend has been observed across the entire tested range.



a) Compressor map b) Turbine map

Figure 6. Compressor and turbine map for both cases (uncoated and coated volute).

To get a better insight on the shift of turbines operating conditions, a few points with identical turbocharger loadings at two different TIT (150 degC and 550 degC) are selected and are plotted in figure 7. At both TIT, the shift is observed, but is more obvious in the case of higher TIT. The shift of turbine operating conditions' effect on the turbocharger performance can be evaluated via the investigation into the effective turbocharger efficiency.

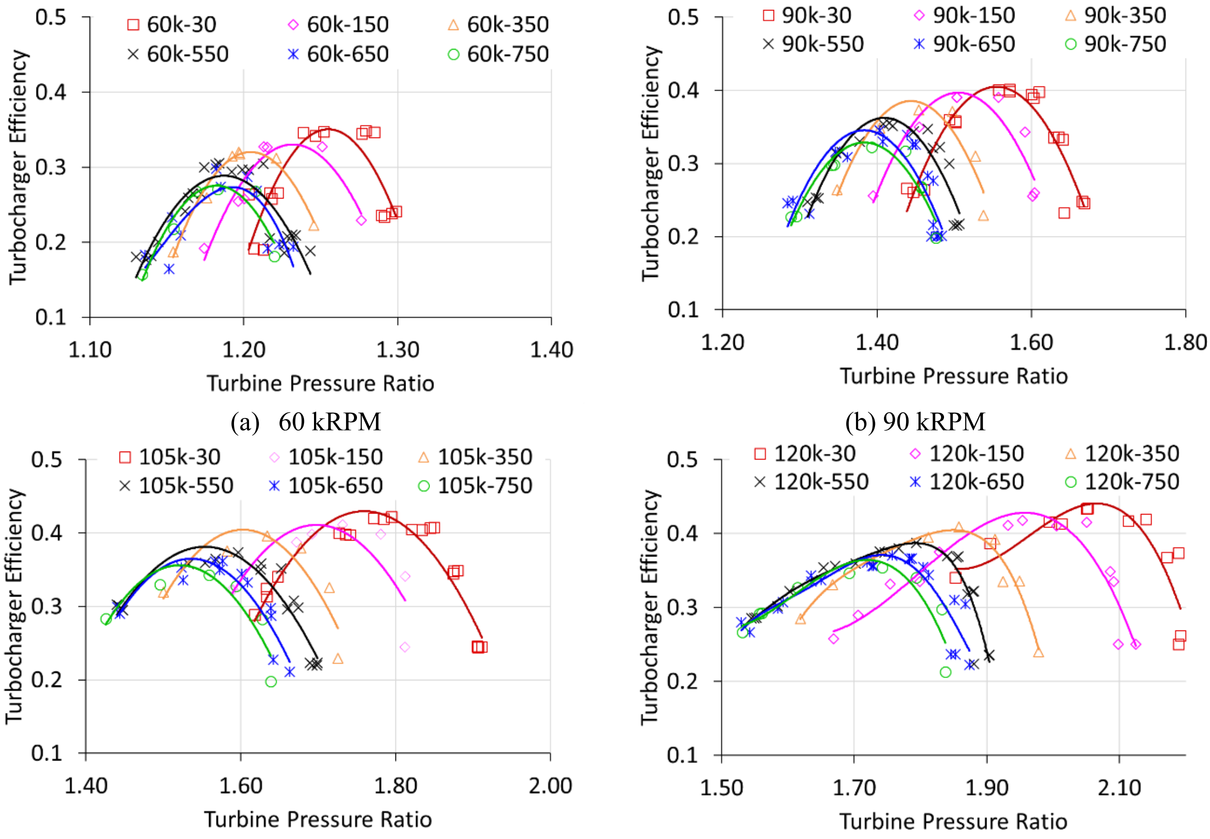


a) TIT 150 degC b) TIT 550 degC

Figure 7. Turbine map point to point comparison on selected cases.

3.2. Turbocharger Efficiency

The baseline turbocharger efficiency of different speed at varies TIT is shown in figure 8. Curve fitting lines are plotted for each turbocharger loadings to better illustrate the efficiency characteristic. At each speed, the increment of TIT shifting the turbine operating points to lower PR, at the same time reducing the peak efficiency. The cold flow condition (30degC) has the highest peak efficiency in overall across the speeds.



(a) 60 kRPM (b) 90 kRPM

(c) 105 kRPM (d) 120 kRPM

Figure 8. Turbocharger efficiency effect with different TIT (uncoated turbine).

Similar phenomenon is observer across the tested speed range. It is fair to conclude that the shift of PR downwards causing the turbine to operate at lower efficiency region, hence impacting the turbocharger efficiency negatively. However, the heat transfer effect should be taken into account too.

At higher TIT, more heat loss from the turbine to the compressor, the coolant and lubricant, or the environment, is expected due to steeper temperature gradient [21].

In order to further investigate the effect of TIT on turbocharger efficiency, testing points of similar turbine operating condition at turbine PR = 1.55, MFP=0.69 in the range of turbocharger speed from 90 kRPM to 120 kRPM are selected and the corresponding efficiencies are plotted in figure 9. From the plot, it is observed that the turbocharger efficiency drops with the increment of TIT.

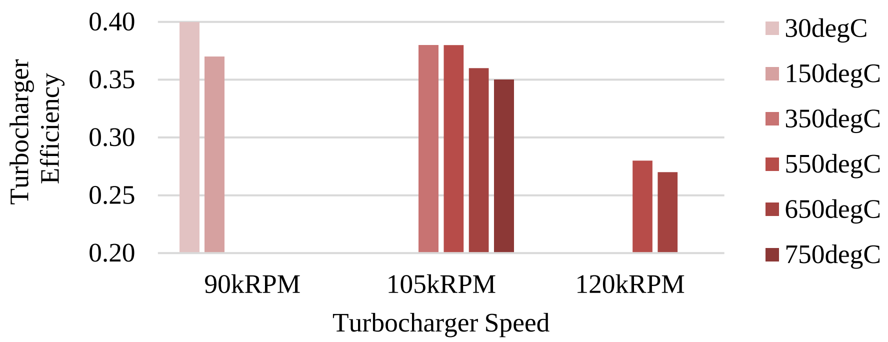
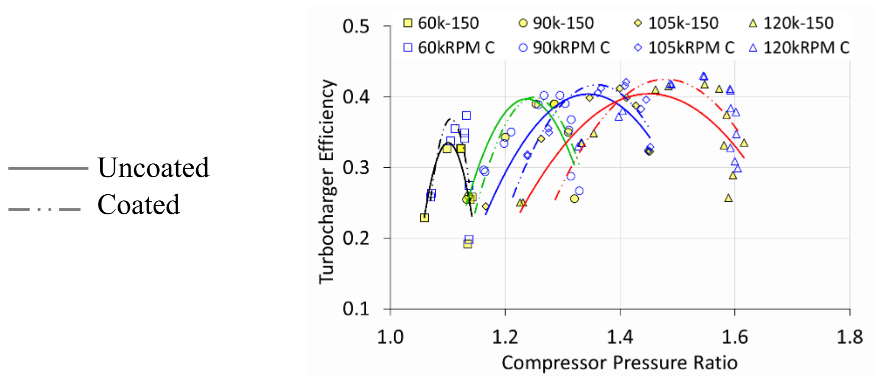


Figure 9. Turbocharger efficiency comparison at same turbine operating conditions.

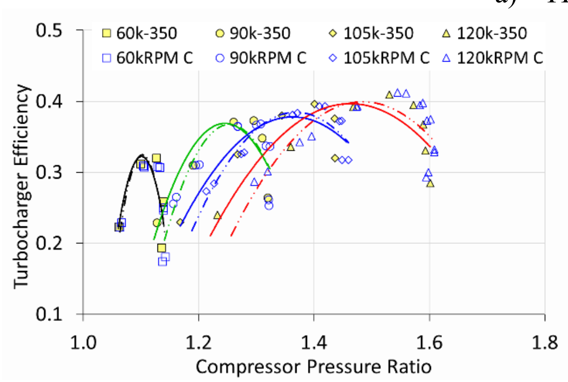
To have a meaningful comparison between the uncoated and coated turbine housing, the turbocharger efficiency is compare based on the turbocharger loading. The turbocharger loading was kept constant for each test point. The performance evaluation focus on the overall turbocharger performance rather than the individual components because the turbocharger tested as one unit. Figure 10. shows the turbocharger efficiency comparison versus the compressor PR for the entire range of TIT. Curves are fitted to the plots to better illustrate the efficiency tabulations. Generally, turbocharger efficiency of the coated turbine housing has higher peak in comparison to the uncoated turbine housings. The coated turbine housing efficiency is also often higher than the uncoated case' at higher turbocharger loading region (after the peak of efficiency). The coating does improve the overall turbocharger performance.

3.3. External and Internal Heat Transfer

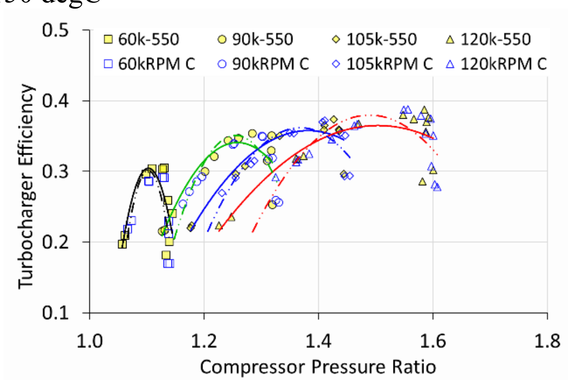
Figure 11. shows the result of external heat transfer measure from the experiment. Higher the temperature, higher the external heat transfer. The dotted line shows the coated turbine volute and solid line is uncoated turbine volute. At low Reynold number or low turbocharger speed, no significant difference between coated and uncoated at TIT 30 degC. As the TIT increase, the difference between uncoated and coated volute become bigger. The reduction of external heat transfer from 7% to maximum of 15% at low turbocharger speed (60kRPM). At high Reynold number or high turbocharger speed, the reduction of external heat transfer become larger, the reduction from 19% to 40%. The reduction is due to increase of thermal resistance when coating was applied as shown in figure 12. With the present of coating in turbine volute, the external heat transfer were reduce and this were supported with finding from other researcher [13].



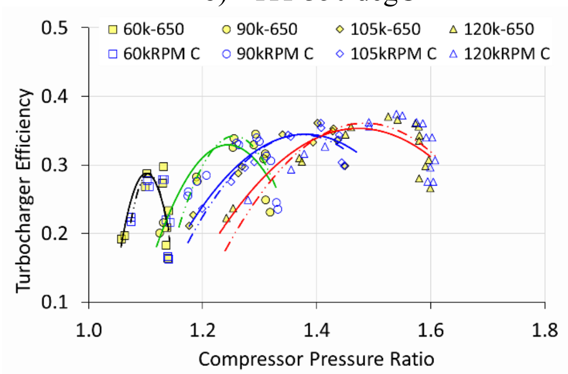
a) TIT 150 degC



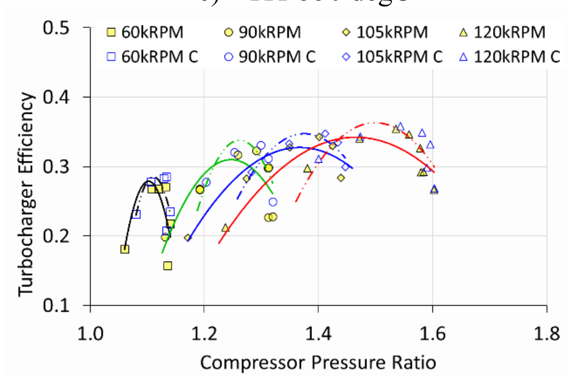
b) TIT 350 degC



c) TIT 550 degC



d) TIT 650 degC



e) TIT 750 degC

Figure 10. Turbocharger efficiency comparison coated and uncoated volute casing for each TIT.

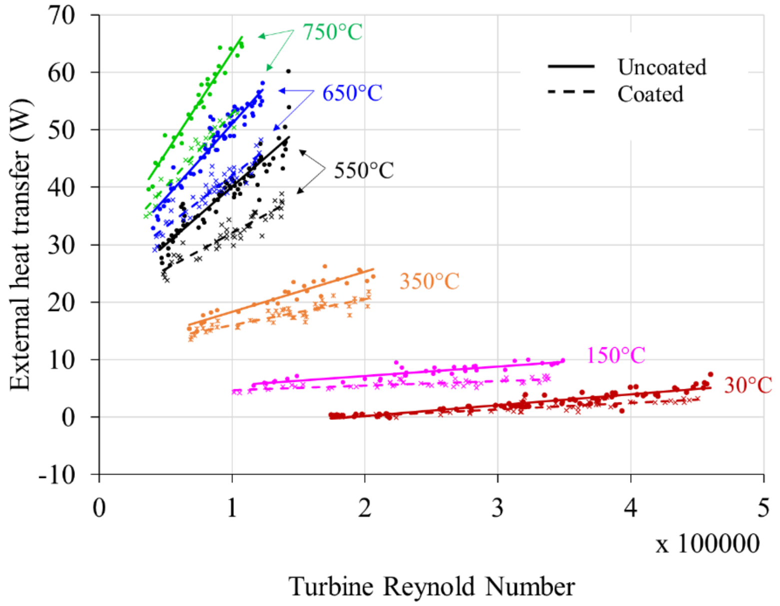


Figure 11. External heat transfer across turbine volute.

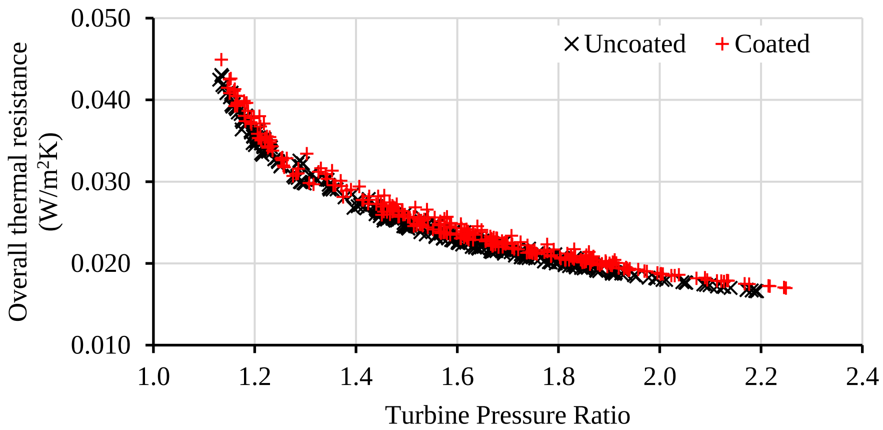


Figure 12. Overall thermal resistance for external heat transfer.

Figure 13. shows the result of internal heat transfer measure from the experiment. No significant difference for internal heat transfer between coated and uncoated. Furthermore, temperature different between gas temperature and lubricant temperature and water temperature not very significant. Despite of having larger thermal resistance for coated, but the driven fluid temperature difference not very significant, hence the internal heat transfer not very significant. Figure 14. shows the overall thermal resistance and fluid temperature differences for internal heat transfer.

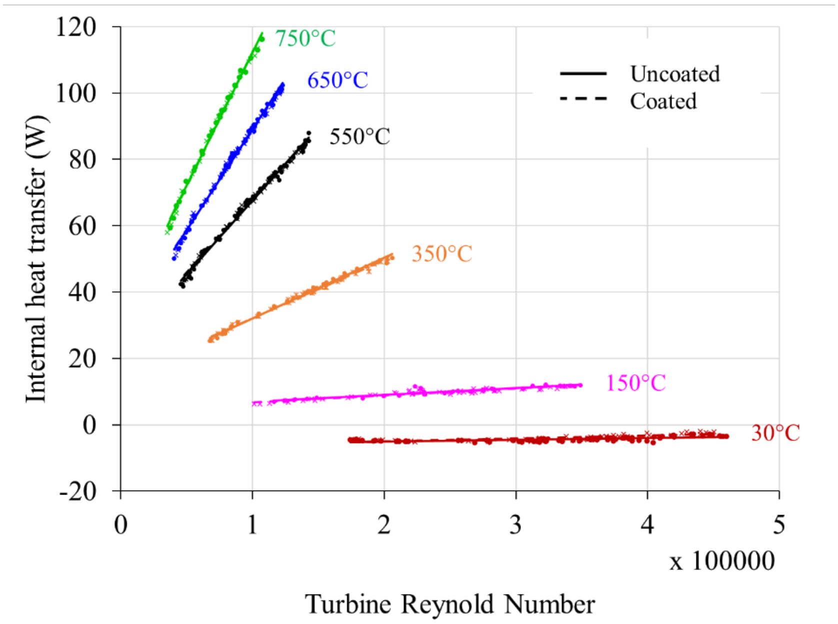


Figure 13. Internal heat transfer from turbine side to central housing.

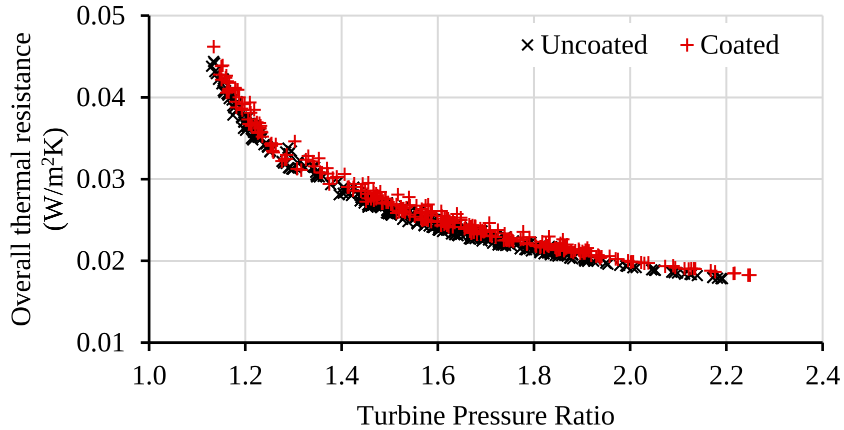


Figure 14. Overall thermal resistance for internal heat transfer.

4. Conclusion

In conclusion, the turbine coating inner volute increase the turbocharger efficiency and very effective to reduce the external heat transfer. However, no significant difference on the internal heat transfer. For the coated turbine inner volute, the mass flow rate lower than the uncoated turbine casing. Since the mass flow is lower, therefore higher PR needed to achieve the target speed. The coated turbine casing increases the turbocharger efficiency significantly across all TIT. Coating reduces the external heat transfer from 7% to 15% at speed 60 kRPM and 19% to 40% at speed 120 kRPM. For internal heat transfer, no significant difference between coated and uncoated turbine due to the temperature difference between two fluid are low. For future work, the coating area can be enlarged to shroud and diffuser area. By enlarging coating area, the external heat transfer can further reduce, and efficiency can increase.

Nomenclature

t	Turbine	0	Total conditions
c	Compressor	1	Compressor inlet
ext	External	2	Compressor outlet
int	Internal	3	Turbine inlet
Inc	Inconel	4	Turbine outlet
YSZ	Yttria Stabilized Zirconia	η	Efficiency
Cast	Cast iron	γ	Gamma
C	Coated turbine	MFP	Mass flow parameter
Q	Heat transfer	PR	Pressure ratio
A	Area (m^2)	$\dot{m}$	Mass flow rate
L	Length (m)	c_p	Specific heat capacity
Nu	Nusselt number	h	Force convective (W/m^2K)
Re	Reynold number	κ	Thermal conductivity (W/mK)
Pr	Prandtl number	x	Thickness material (m)
T	Temperature (K)	ρ	Density (kg/m^3)
P	Pressure (Pa)	μ	Dynamic viscosity (kg/ms)
TIT	Turbine inlet temperature (K)	TOT	Turbine outlet temperature (K)

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