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Proceedings of the Fourth International Symposium on Fluid Machinery and Fluid Engineering

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FOREWORD

Following the experience gained in organizing the International Symposium on Fluid Machinery and Fluid Engineering in 1996, 2000 and 2004, it was decided to hold the Fourth International Symposium on Fluid Machinery and Fluid Engineering. This fourth symposium is now to convene on November 25-27 in Beijing.

The Chinese Society of Engineering Thermophysics (CSET) is a well-established engineering society devoted to theoretical and applied research in the thermal and fluid sciences. It was first founded by the late Dr. C.H. Wu, well-known leader in the field of turbomachinery. The Chinese Society of Engineering Thermophysics (CSET) organized the First, the Second and the Third International Symposium on Fluid Machinery and Fluid Engineering, in 1996, in 2000, and in 2004 successfully. Fluid machinery is a kind of widely used machines and has a great action to all fields of the national economy. The purpose of the Fourth Symposium is the same as before, to provide a common forum for exchange of scientific and technical information worldwide on fluid machinery and fluid engineering for scientists and engineers. The main subject of this symposium is "Fluid Machinery for Energy saving". There is the "Mei lecture" in the symposium to make reports on the development and the new research area of fluid machinery in order to commemorate the late professor Mei Zuyan in the field of fluid machinery in China.

This volume of proceedings contains 69 highly informative technical papers that have been selected by peer review and are to be presented at the Mei lecture session and the technical sessions of the symposium. They cover very well the latest practice and findings in the fields of fluid machinary and fluid engineering.

Jianzhong XU, Professor
Chairman of the Organizing Committee
September 2008

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Heat Transfer in an Automotive Turbocharger Under Constant Load Points: an Experimental and Computational Investigation

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Abstract Nowadays the turbocharger is one of the most commonly used devices to supercharge an engine. Heat fluxes in the turbocharger are not negligible and affect the performance prediction of the turbine. From different experimental investigations it has become clear that heat fluxes from the turbine to the compressor have a great influence on the compressor performance and therefore on the overall turbocharger performance. For this reason understanding the heat transfer within the turbocharger components and from the turbocharger to the ambient environment is essential important to determine the critical heat paths to be considered in the design tools.

In order to investigate the behavior of the heat fluxes occurring in the turbocharger an experimental and computational study has been carried out at Imperial College on a Ford 2.0 liter diesel engine. Beyond the standard measurements necessary to determine the operating points of the compressor and turbine, a novel set of seventeen thermocouples was installed on the turbocharger measuring the inner and outer wall temperature of the turbine and compressor casing, the bearing housing and exhaust manifold temperatures. In addition to these, the air and oil flow rate, temperature and pressure were also measured.

A one-dimensional model was also developed. The developed algorithms are merged using a MATLAB programme that calculates the compressor non-adiabatic efficiencies and exit temperatures based on the turbocharger geometry, the turbine inlet temperature and the maps of the turbine and the compressor in adiabatic conditions. The simplification of the turbocharger was kept as low as possible and, unlike the other models, no heat transfer coefficients were used.

The test results provided profound insight into the temperature distributions occurring within the turbochargers and in particular the role played by the exhaust manifold. Furthermore, the data generated with the test enabled us to quantify the heat fluxes and to validate the one-dimensional model. The model prediction of the temperature and non-adiabatic efficiencies is a significant improvement on previous models. The main outcomes of the research carried out at Imperial are reported in this paper.

Keywords heat transfer, turbocharger, turbine, compressor, engine, model, experimental, performance

Nomenclature

Eng	*Engine side*	
Top	*Top side*	
Ext	*External side*	
I/O	*Ratio between Inner and Outer wall*	
O/I	*Ratio between Outer and Inner wall*	
η	*Efficiency*	
h	*Enthalpy*	[W/mK]
T	*Temperature*	[K]

Subscripts

adi	*Adiabatic*
dia	*Non-adiabatic*
C	*Compressor*
is	*Isentropic*
1	*Inlet to the compressor*
2	*Exit to the compressor*

1 Introduction

In general, most processes occurring in turbo machinery applications are treated as adiabatic since the influence of heat transfer on the calculation is often negligible. However in some cases, heat transfer can have an influence on performance thus making a non-adiabatic treatment more appropriate. Since the hot turbine of the turbocharger is located in close proximity to the relatively cold compressor, it is obvious that there will be heat exchange between turbine and compressor. In general, the non-adiabatic process can be separated into three stages of heat transfer. The first stage involves heat that is transferred before the compression or expansion process starts. In the second stage, a fraction of heat is introduced during the process and the third stage accounts for the heat that is added after the process is completed.

Heat transfer analysis usually involves quantifying the heat transfer rate for some known temperature difference. It is recognised that heat can be transferred by one or a combination of three separate modes known as conduction, convection and radiation. Conduction occurs in a stationary medium; convection requires a moving fluid while radiation occurs in the absence of any medium distinguishing it as a part of the electromagnetic spectrum. Although it is useful to look at each one of these processes in a distinct way, they often occur together. In particular, on a turbocharger all of these three processes occur at the same time and are closely interrelated. The complexity of turbocharger geometry introduces many possible heat transfer mechanisms inside the turbocharger as well as from the turbocharger to the environment, as illustrated in Fig. 1.

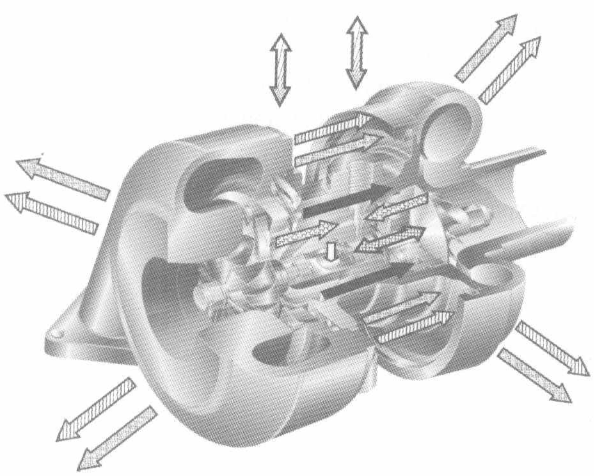

Fig. 1 Heat fluxes in a turbocharger (Shaaban 2006)

Heat transfer from the turbine to the compressor through the bearing housing must be considered even though the cooling oil reduces the amount of heat transfer that is transferred by conduction from the turbine to the compressor; in addition heat transfer from the turbine to the environment takes place by means of radiation and free convection and heat transfer from the compressor to the environment takes place likewise by means of radiation and free convection, even though radiation heat transfer from the compressor is very small because of the low emissivity.

Such a complex pattern of heat fluxes makes experimental investigation of the heat transfer very difficult. As a consequence of this, not many experiments on heat transfer have been carried out in the past. Rautenberg et al. (1983 and 1984) first analysed the influence of heat transfer from the hot turbine to the compressor by testing two different turbochargers with different axial distance. The results showed that the axial length plays an important role in the deterioration of the mechanical power. Beyond the standard measurements necessary to determine the operating point of a compressor and turbine, Bohn et al. (2003) also measured the surface temperature of the turbine casing, showing that the temperature of the turbine casing varies linearly with the inlet temperature.

All of the experiments carried out so far were performed in a purpose-built test facility that enabled a wide range of test conditions to be covered. However the facility did not permit the analysis of the real conditions occurring when the turbocharger is installed on the engine. These are extremely important since the close proximity to the engine and in particular to the exhaust manifolds make the effects of heat transfer on the turbocharger performance even more relevant. Therefore the aim of the current research is to estimate the role played by the engine in the overall heat transfer occurring within the turbocharger from both a qualitative and quantitative point of view. A commercial turbocharger was installed on a 2.0 litre diesel engine and an experimental and computational investigation was carried out and is reported here.

2 Experimental Investigation

A schematic diagram of the engine test rig at Imperial College is shown in Fig. 2. A 25kW DC electric motor/generator supplies air to the inlet manifold via the intercooler either by the compressor of the turbocharger or by a roots blower supercharger externally driven through a multiplication gearbox. An eddy current dynamometer was used to keep the engine load constant at a desired value. The engine was operated via an instrumentation rack consisting of controls to operate the dynamometer, to crank and run the engine and to stop operation in case of an emergency.

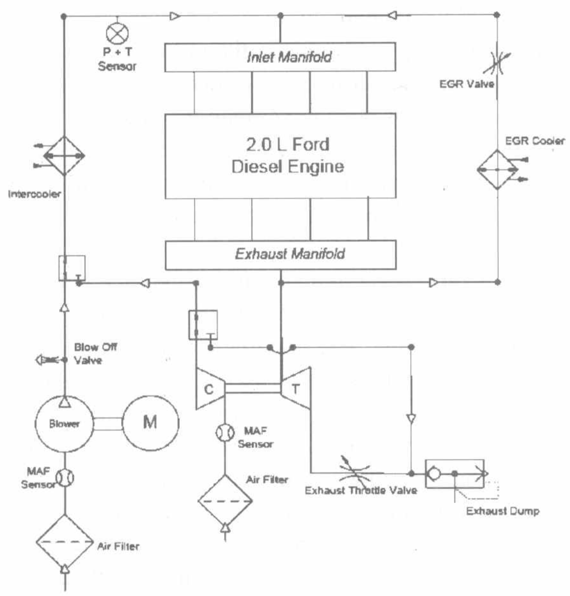

Fig. 2 Test rig layout (Kyartos 2006)

Beyond the standard measurements necessary to determine the operating points of the compressor and turbine, the turbocharger was set up in order to enable the monitoring of the temperatures at seventeen stations, as shown in Fig. 3.

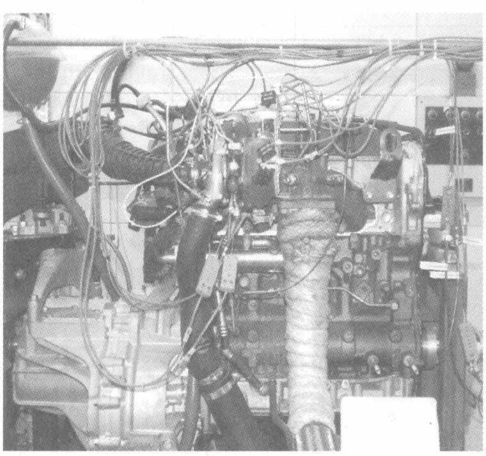

Fig. 3 Test rig

In detail the following measurements have been carried out:

- Inlet and exit air temperature to the compressor
- Total pressure at the inlet and exit to the compressor
- Inlet and exit air temperature to the turbine
- Total pressure at the inlet and exit to the turbine
- Inner and outer temperatures along the compressor casing in three different locations
- Inner and outer temperatures along the turbine volute in three different locations
- Surface temperature of the intake manifolds
- Surface temperature of the bearing housing
- Inlet and exit temperature of the oil flow
- Air and oil flow rate
- Shaft speed

3 Test Results

The turbocharger under study was tested under constant load points for a range of engine speeds. Measurements were obtained for engine speeds of between 1000 and 3000 rpm at steps of 500 rpm. For each engine speed the load applied was varied from 16 to 250 Nm, as reported in Table 1.

Table 1 Test conditions

Speed [RPM] / Torque [Nm]	16	50	100	125	150	200	250
1000	√	√	√				
1500	√	√	√	√	√	√	√
2000	√	√	√		√	√	√
2500	√	√	√		√	√	
3000	√	√	√		√	√	

(1) Performance comparison

In order to evaluate the effects of heat transfer on the deterioration of the mechanical power, the non-adiabatic efficiency was used as dimensionless parameter. The non-adiabatic efficiency represents the apparent compressor efficiency measured under non-adiabatic operating conditions and it is defined as the ratio between the isentropic and the actual enthalpy rise.

$$\eta_{dia,c} = \frac{\Delta h_{adi,is}}{\Delta h_{dia}} = \frac{T_{2,adi,is} - T_1}{T_2 - T_1} \tag{1}$$

$\eta_{dia,c}$ differs from the adiabatic efficiency (below) in which the adiabatic enthalpy rise is taken into account:

$$\eta_{adi,c} = \frac{\Delta h_{adi,is}}{\Delta h_{adi}} = \frac{T_{2,adi,is} - T_1}{T_{2,adi} - T_1} \tag{2}$$

A comparison between the non-adiabatic efficiencies and the correspondent adiabatic efficiencies extrapolated by the cold maps provided by the turbocharger manufacturer is reported in Fig. 4 in terms of *relative efficiency*. This parameter has been introduced for reasons of confidentiality and is defined as the ratio between the compressor peak efficiency and the measured efficiencies. It can be seen that the difference existing between the adiabatic and the non-adiabatic efficiencies tends to increase as the turbine inlet temperature increases. The deviation of the non-adiabatic efficiency from the adiabatic efficiency goes from a maximum of 30% (at low speeds) to a minimum of 15% (at high speeds). This can be explained if we consider that at high rotational speeds, the turbocharger works in conditions similar to those at design point. At this point all of the losses occurring within the turbocharger are assumed to be at a minimum and the efficiency drop is less significant.

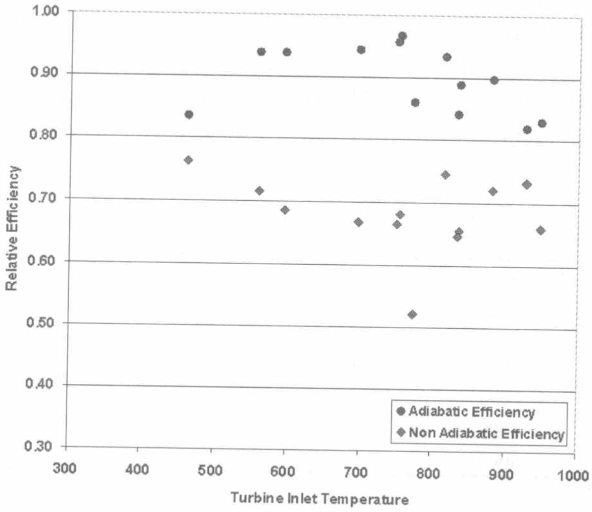

Fig. 4 Comparison between adiabatic and non-adiabatic compressor performance

(2) Temperatures on the Turbine and Compressor casing

For a fixed engine speed, the inner and outer wall temperature of the turbine and compressor casings was measured in three different locations (*Engine, Top, and External*), as shown in Fig. 5.

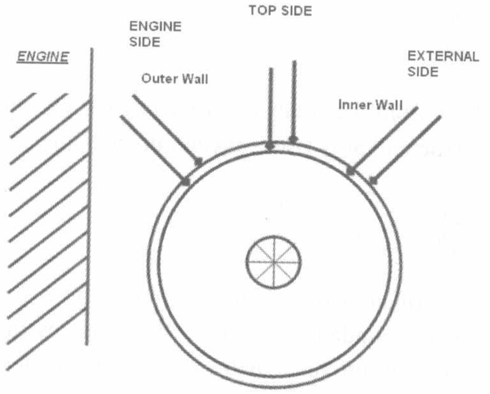

Fig. 5 Thermocouple positions

The heat transfer mechanism occurring in the turbine can be described as follows: the hot gases flowing into the turbine heat up the inner surface of the casing by forced convection. A temperature gradient between the inner and the outer surface of the casing is therefore created, leading to the generation of a heat flux towards the external wall where, by means of radiation and natural convection to the environment, the turbine is cooled down. In contrast to the turbine, the inner wall temperature of the compressor casing is lower than the outer one. This can be attributed to the air flowing in the compressor which tends to cool down the casing subjected to radiation and conduction due to the turbine, the bearing housing and the exhaust manifold.

Table 2 reports the temperature difference between the inner and the outer wall for both the turbine and the compressor casings. The maximum temperature observed on the compressor casing at maximum engine speed and load is almost five times lower than the maximum temperature of the turbine. Furthermore, a gradient of up to 4.23% has been measured between the outer and inner wall of the compressor casing at 3000 rpm on the external side. This is more than 50% less than the temperature gradient seen in the turbine and demonstrates that, although the effects of heat transfer on the compressor side is less significant than those occurring in the turbine; they are nevertheless not negligible since they significantly affect the compressor performance.

Table 2 Temperature difference between the inner and the outer wall for both the turbine and the compressor

	Turbine			Compressor		
	Eng	Top	Ext	Eng	Top	Ext
1050 rpm	I/O	I/O	I/O	O/I	O/I	O/I
8 Nm	2.99%	2.63%	7.34%	0.16%	0.54%	0.32%
50Nm	3.22%	3.53%	10.63%	0.38%	0.93%	1.16%
1500 rpm						
8 Nm	3.12%	2.73%	9.91%	0.03%	0.45%	1.47%
250 Nm	3.40%	5.51%	11.01%	7.22%	4.12%	2.15%
2000 rpm						
8 Nm	3.42%	4.33%	10.13%	2.35%	1.70%	1.91%
250 Nm	4.98%	5.56%	10.64%	12.2%	3.57%	2.31%
2500 rpm						
8 Nm	2.95%	2.57%	10.38%	6.16%	2.35%	2.40%
200 Nm	5.52%	4.90%	10.61%	8.46%	3.84%	4.09%
3000 rpm						
16 Nm	3.81%	3.64%	9.04%	1.36%	1.96%	2.79%
200 Nm	4.54%	5.52%	11.09%	2.38%	4.23%	4.17%

(3) Bearing housing and Exhaust manifold surface temperature

The surface temperatures of the bearing housing and of the exhaust manifold were also measured. The results are reported in Table 3.

Table 3 Surface temperature of the exhaust manifold and bearing housing

	Exhaust Compressor side	Bearing Housing	Exhaust Turbine side
1050 rpm			
8 Nm	333.5 °K	330.3 °K	343.2 °K
50 Nm	366.6 °K	335.1 °K	385.4 °K
1500 rpm			
8 Nm	336.9 °K	326.5 °K	348.0 °K
250 Nm	593.4 °K	418.8 °K	659.6 °K
2000 rpm			
8 Nm	341.6 °K	345.0 °K	361.6 °K
250 Nm	613.6 °K	422.0 °K	683.4 °K
2500 rpm			
8 Nm	370.9 °K	350.9 °K	382.8 °K
200 Nm	615.0 °K	411.3 °K	654.6 °K
3000 rpm			
16 Nm	414.2 °K	366.9 °K	437.8 °K
200 Nm	614.9 °K	406.1 °K	698.6 °K

The surface temperature of the bearing housing can be considered as the sum of the cooling effects of the oil and of the convective, radiative and conductive heat fluxes due to the proximity of the turbine. However, the test results demonstrate that the surface temperature of the bearing housing is of the same order of magnitude as the oil. This means that the role played by the oil on the overall temperature of the bearing housing is greater than that played by the turbine and, since most of the heat transferred from the turbine to the compressor occurs through the bearing housing, this implies that the heat flux through the bearing housing will have to be predicted very accurately in the model.

In addition to the surface temperature of the bearing housing, the surface temperatures of two pipes of the exhaust manifold were also measured. As can be seen in Table 3 a significant temperature difference exists between the pipes on the compressor and turbine sides. Such a difference ranged from a minimum of 10 °K at 1000 rpm to a maximum of 80 °K at 3000 rpm. Since the temperatures on the turbine casing are of the same order of magnitude as those measured on the exhaust manifold, the close proximity of the exhaust manifold to the turbine casing is unlikely to affect the turbine heat transfer. However, since the surface temperatures measured on the compressor casing are much lower than those on the turbine casing, the close proximity of the exhaust manifold to the compressor will tend to transfer heat by radiation and convection to the compressor casing and this will tend to heat up the air flowing inside. As a consequence, a temperature rise in the air flowing through the compressor is expected to occur, leading to a further decrease in the compressor efficiency. The exhaust manifold has not been considered in the model but there are good reasons however for including it in a future model development.

4 One-Dimensional Model

In this section a description of the one-dimensional model implemented in MATLAB is provided. The model was developed considering the main heat transfer mechanisms: conduction, radiation and convection. The estimation of the heat fluxes made it possible to calculate the following parameters:

- exit temperatures for both the turbine and the compressor
- non-adiabatic efficiency of the compressor

A good prediction of the compressor exit temperature is crucial to the model validation; this is because these temperatures form the boundary conditions for any engine model and they are significantly affected by the heat transfer processes.

In Fig. 5 the physical model of the turbocharger is illustrated. The compressor, the turbine and the bearing housing were modelled as three cylinders and for each of these three bodies the heat fluxes were calculated. Such a simplification was necessary for several reasons: first of all, a full description of the turbocharger geometry would require geometrical data that were not available, as the design of the turbine could not be disclosed for reasons of confidentiality. Secondly an over-detailed geometrical analysis of the turbocharger would make the model too complex and would also constrain the outcomes to the particular case under study.

The heat fluxes that are of direct interest in the estimation of the non-adiabatic efficiencies are the heat fluxes that enter or leave the turbine and the compressor, as illustrated in Fig. 6. Due to the hot exhaust gases entering the turbine volute, a high surface temperature can be expected on the turbine side while the compressor is comparatively cold and hence only a negligible amount of heat is transferred at the compressor. The surface of the bearing housing lies between the turbine and the compressor and is therefore also taken into account. In the oil channel between the shaft and the bearing housing the oil flow causes forced convection on the shaft and on

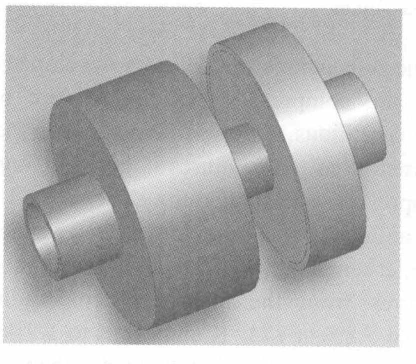

Fig. 6 Turbocharger physical model

the inside of the bearing housing. As the shaft is rotating at high speeds the flow field in the oil channel is very complex and must be carefully treated.

In order to evaluate the amount of heat transferred within the turbocharger, the heat transfer coefficients and the temperature distributions occurring on the surfaces constituting the model have been calculated. Since only the steady state is treated in the developed model, no change in internal energy has been taken into account, allowing to simplify the energy balance.

5 Model Results And Validation

(1) Compressor exit temperature

As already shown in the Test Results section, the compressor non-adiabatic efficiency is lower than the compressor adiabatic efficiency and this means that the exit temperature of the compressor in non-adiabatic conditions is higher than in the adiabatic conditions. This is not irrelevant because it leads to a significant inaccuracy in the estimation of the compressor power requirement. The compressor outlet temperature forms one of the boundary conditions used in the engine simulation models and, since the combustion process and the formation of pollutants in the combustion chamber of the engine are very sensitive to the combustion temperature, the boundary conditions have a great influence on the results. Hence the compressor outlet temperature has to be calculated as accurately as possible.

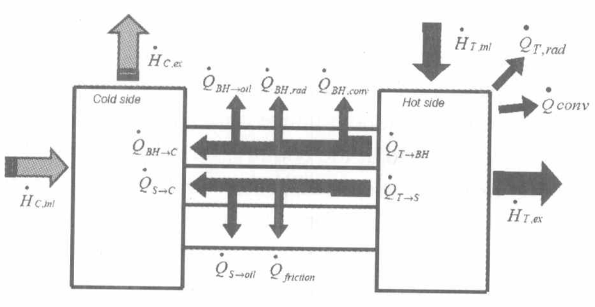

Fig. 7 Heat fluxes

The one-dimensional model here developed enables to calculate the compressor exit temperature under non-adiabatic conditions. A comparison between the predicted and the measured temperatures is given in Fig. 7 against the compressor pressure ratio. The model seems to provide an accurate prediction over a wide range of turbine inlet temperatures. The deviation of the calculated temperatures from the measured temperatures is not greater that 5.8% at 774 °K and drops to 1.9% at 755 °K. On average the deviation is not greater than 4% for both low and high engine speeds and torques and this seems to confirm the effectiveness of the choices made on the heat fluxes occurring within the turbocharger.

Table 4 Compressor exit temperature

Pressure Ratio	1.27	1.57	1.73	1.85	2.08
Turbine Inlet Temperature [°K]	774	755	836	949	928
Deviation: Model – Test [%]	5.8	1.9	2.4	3.8	4.3

(2) Compressor non-adiabatic efficiency

The non-adiabatic efficiencies are calculated as the ratio between the isentropic and the actual enthalpy rise occurring in the compressor (Equation 2). The predicted and the experimental relative compressor non-adiabatic and adiabatic efficiencies are provided in Fig. 8 as a function of the pressure ratio. The model provides a good prediction of the non adiabatic performance over a wide range of pressure ratios from 1.2 to 2.1.

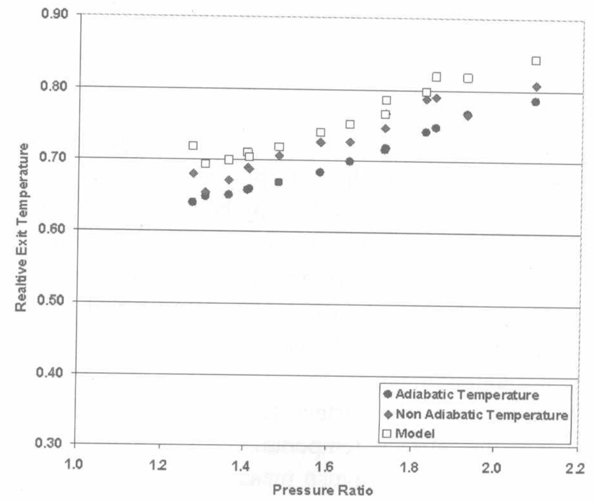

Fig. 8 Relative exit temperature vs. pressure ratio

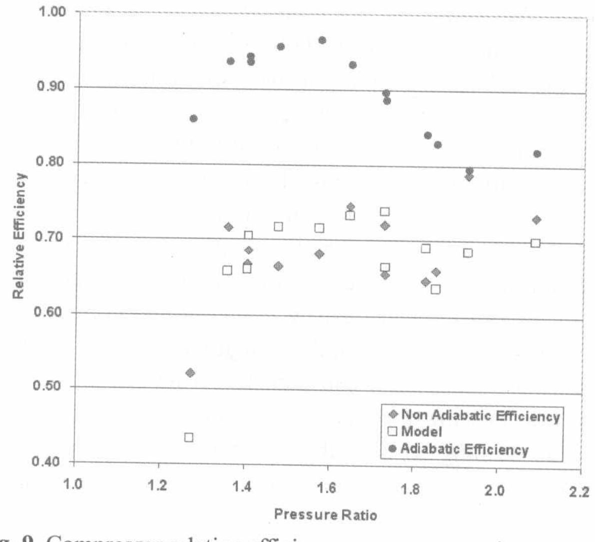

Fig. 9 Compressor relative efficiency vs. pressure ratio

In Table 5 a comparison between the efficiencies is reported. The deviation of the computed data from the test data is not significant for most of the measured points even though a significant scatter can be observed at low pressure ratio. This is because at low rotational speeds (corresponding to a low pressure ratio), the effects of heat transfer on the compressor performance are greater since the compressor works in conditions far removed from the design point. This can lead to instability effects within the flow that are difficult to match in the model as no assumption was made regards the aerodynamic conditions of the flow.

Table 5 Efficiency comparison

Pressure Ratio	1.27	1.57	1.73	1.85	2.08
Turbine Inlet Temperature [°K]	774	755	836	949	928
Deviation Model-Test [%]	17.28	4.73	1.36	3.63	4.30

6 Conclusion

This study has investigated the influence of heat fluxes from the turbine to the compressor in a turbocharger. Firstly a set of tests was carried out on a 2.0 litre diesel engine, monitoring the surface and flow temperatures for both the compressor and the turbine. Secondly, a one-dimensional model was implemented to predict the compressor exit temperature and non-adiabatic efficiencies and the model was validated against experimental data.

The tests were carried out at Imperial College in cooperation with Prof. Alex Taylor and Dr. Yanis Hardalupas. The engine speed was varied from 1000 rpm to 3000 rpm and, for each of these speeds the load was varied from 8 Nm to 250 Nm. The data obtained showed that the non-adiabatic conditions strongly affect the compressor performance. At lower engine speeds and torques, the overall non-adiabatic compressor efficiency deviates by about 30% from the adiabatic efficiencies whilst, at higher engine speeds and loads, the deviation drops to 3%. The test also showed that the bearing housing temperature is strongly affected by the coolant oil and that the close proximity of the exhaust manifold to the compressor casing is an additional source of efficiency loss.

The data generated from the tests were then used to validate a simplified one-dimensional model developed in MATLAB. The turbocharger geometry was simplified and considered as being constituted of an assembly of flat plates with known thermal properties. Heat transfer correlations were used to quantify the heat coefficients for convective heat transfer and were based on the inlet conditions and the geometrical properties. The compressor exit temperature and the compressor non adiabatic efficiencies were calculated.

Comparison between the experimental and computed data showed that the predicted compressor exit temperatures were found to be in very good agreement. An average deviation of 4% was found to occur for the compressor. This result is remarkable since the exit temperatures form the boundary conditions for any engine model available on the market. In addition to the compressor temperatures, the compressor non-adiabatic efficiencies were also computed. An overall averaged deviation from the test values was found to be not greater than 4% at lower engine speeds and around 10% at higher engine speeds. Although these deviations are not negligible, the results obtained here provide a significant improvement over other methods, in which deviation of up to 30% was seen.

In conclusion it can be said that the accuracy of the model can be increased by decreasing the level of simplification on the turbine side. This includes the development of more appropriate heat transfer correlations for the turbine volute and the turbine wheel. Furthermore the level of simplification of the turbine geometry can be reduced to obtain more accurate values for the overall heat loss of the turbine. Besides these basic improvements further experimental work is needed to gain a better insight in the temperature distributions on the surfaces of the turbine and the bearing housing in order to estimate the occurring heat fluxes more accurately.

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