The Analysis of Heat Transfer in Automotive Turbochargers

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Heat transfers in an automotive turbocharger comprise significant energy flows, but are rarely measured or accounted for in any turbocharger performance assessment. Existing measurements suggest that the difference in turbine efficiency calculated in the conventional way, by means of the fluid temperature change, under adiabatic conditions differs considerably from the usual diabatic test conditions, particularly at low turbine pressure ratio. In the work described in this paper, three commercial turbochargers were extensively instrumented with thermocouples on all accessible external and internal surfaces in order to make comprehensive temperature surveys. The turbochargers were run at ranges of turbine inlet temperature and external ventilation. Adiabatic tests were also carried out to serve as a reference condition. Based on the temperature measurements, the internal heat fluxes from the turbine gas to the turbocharger structure and from there to the lubricating oil and the compressor, and the external heat fluxes to the environment were calculated. A one-dimensional heat transfer network model of the turbocharger was demonstrated to be able to simulate the heat fluxes to good accuracy, and the heat transfer coefficients required were ultimately found to be mostly independent of the turbochargers tested. [DOI: 10.1115/1.3204586]

1 Introduction and Objectives

The aims of this project were to understand the effects on performance of automotive turbochargers, which are due to several influences not recognized in current turbocharged engine simulations, and to add new features to those simulations in order to account for them. The effects of concern are heat transfer, internal and external to the turbocharger; friction loss in bearings; and the influence of exhaust pressure pulsations on turbine performance. This paper describes the heat transfer investigation. Another paper [1] covers the pulse flow performance measurement and modeling.

The study comprised the experimental testing of commercial turbochargers to gather a database of information and the use of that database to develop methods by which the effects could be modeled. An important consideration in modeling was that the methods developed should be capable of implementation in industrial circumstances. A typical engine simulation is iterative, and the turbocharger performance may be calculated many times per cycle. Computationally-intensive methods based on computational fluid dynamics (CFD) and finite element analysis (FEA) would not be suitable for the purpose. Simulations are also used to predict the performance of proposed engine systems, so the methods must be capable of being used predictively and should not require large data input of existing turbochargers. Clearly some data input of the turbocharger geometry and flow conditions is necessary, but this must be limited to essential parameters that can be easily estimated for new projects.

By way of introduction, Fig. 1 shows all of the energy transfer processes that occur in a conventional turbocharger. These include the work transfer along the shaft from turbine to compressor, the work converted into heat in the bearings (i.e., the bearing power loss), and the internal and external heat transfers that apply to each major component of the turbocharger: the compressor, bearing housing, and turbine, together with the flows of energy associated with each stream of fluid entering and leaving the turbocharger. Applying energy conservation to each component, in turn, provides a means to define figures of merit (efficiencies) for each component and to determine the heat transfers that must be quantified to allow those figures of merit to be applied in data analysis and turbocharged engine modeling.

1.1 Thermodynamic Analysis. The thermodynamic analyses for the compressor and turbine are based on comparing an isentropic adiabatic process, forming the ideal reference; an adiabatic nonideal process; and a diabatic nonideal process. The last of these represents the actual process that occurs in turbochargers in engine operation and in "hot" tests on the gas stand. The adiabatic nonideal process can be simulated on a gas stand if precautions are taken to minimize the internal and external heat transfers. External heat transfer is readily dealt with by means of thermal insulation. Internal heat transfer can only be minimized by a process of thermal matching, in which the compressor air, turbine gas, and bearing oil temperatures are controlled to minimize the temperature differences that drive internal heat transfers. Adiabatic testing is rarely done in industrial gas stand testing but can be achieved in laboratory tests.

In an adiabatic process or test, the change in total enthalpy across the machine, which can be measured by means of the total temperatures at inlet and exit, is equal to the shaft work transfer. In a diabatic process or test, the change in total enthalpy across the machine is equal to the algebraic sum of the work and heat transfers, and this therefore may not be an accurate method to measure the work transfer, although that will depend on the relative magnitudes of the two effects. The practical purpose of an adiabatic test program is thus to obtain an accurate measurement of the work transfer. In a turbocharger, an adiabatic test invariably means running the turbine "cold," i.e., at some inlet temperature closer to ambient. This influences the match between the turbine and compressor and often restricts the range of operation that can be covered on the gas stand.

Because of the practical difficulties and limitations of adiabatic testing, it is uncommon in the industrial situation. The conventional process is to apply the diabatic gas stand efficiency to the engine condition without any correction for the different heat transfers that are likely to exist in the two situations. The accuracy of work transfer measurement in diabatic testing can be improved if an estimate or calculation of the heat transfer can be made, and the results can be corrected for this. This means that the diabatic efficiency measured on the gas stand can be corrected to an adia-

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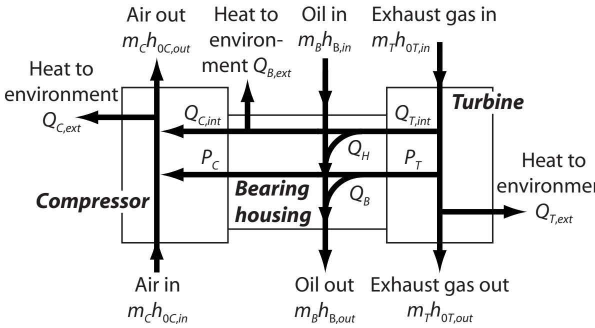

Fig. 1 Energy transfer in a turbocharger

batic efficiency, which, because it is subject only to the internal gas dynamic processes occurring in the machine, will apply equally to operation on-engine. For such operation, the adiabatic efficiency must be readjusted for the heat transfer that occurs in the engine environment, which is not necessarily the same as the gas stand heat transfer, in order to obtain the on-engine diabatic efficiency.

The main objectives of an adiabatic test program can also be achieved by measuring the shaft power directly, using a dynamometer, torquemeter, or some such device. In turbocharger testing, the turbine power is occasionally measured directly by means of a dynamometer, but practical limitations invariably rule out shaft-mounted devices such as torquemeters. No reference has been found to direct power measurement as a means of assessing the effects of heat transfer in a turbocharger, but this remains an option if it is sufficient to determine only the total heat transfer from the turbine, external and internal, without regard for the path of the heat transfer.

The thermodynamic analysis of adiabatic and diabatic processes forms the starting point of all the research programs investigated. The diabatic process is considered as the sum of the adiabatic process plus heat transfer, and usually the heat transfer is assumed to follow after the work transfer. The physical basis for this is not often considered. In a compressor, this implies that the work transfer begins at the compressor inlet conditions and occurs in the impeller. The heat transfer then follows and physically must take place in the diffuser and volute. This seems reasonable because the surface area exposed to the air flow is larger in the diffuser and volute than in the impeller. Furthermore, heat transfer from the turbocharger to the impeller must occur by conduction along the shaft, which is a small diameter and is exposed to lubricating oil, so that the shaft temperature will be controlled by the oil temperature and will never rise to the turbine temperature. Such heat transfer is therefore likely to be small.

In a turbine, however, the same argument suggests that most of the heat transfer occurs upstream of the turbine rotor. This will affect the work transfer process because work transfer in any turbomachine scales as the inlet total enthalpy, i.e., $\Delta h_0/h_{01}$ =constant, where Δh_0 is the specific work transfer and h_{01} is the inlet total enthalpy. Since the latter quantity here occurs at the inlet to the rotor and is affected by heat transfer that occurs upstream of the rotor, it may be argued that a more accurate thermodynamic model results from applying the heat and work transfers in this order.

In some thermodynamic analyses [2–4], a division is made between heat transfers that occur before and after the work transfer process in the rotor or impeller, rather than concentrating all of the heat transfer in one place, although the benefits in doing so are not clearly explained. The disadvantage of this level of detail is the practical one that the heat transfers before and after the impeller have to be separately quantified (and this, of course, still neglects any heat transfer in the rotor itself). The number of heat transfer paths to be analyzed in either component is increased from two (internal and external) to at least five (internal and external, upstream and downstream of the rotor, and conduction through the casing between the upstream and downstream portions). Because of the experimental and instrumentation difficulties of keeping track of so many heat transfer processes, the present program does not work at this level of detail, and only lumped heat transfers to the compressor and turbine are considered.

1.2 Comparisons of Diabatic and Adiabatic Tests. In early results, the effects of heat transfer were determined from a comparison of hot and cold tests, in which the latter were assumed to be adequately adiabatic [5,6]. Because the direction of heat transfer is away from the hot turbine, the turbine exit temperature is lower than it would be for the work transfer alone and, in consequence, the diabatic efficiency, measured on this basis, is larger than the adiabatic efficiency.

In this work, a very limited number of surface heat transfer measurements were also made on the turbine housing, These results indicate that the majority of the heat transfer does indeed occur upstream of the rotor. This evidence is important in formulating an appropriate thermodynamic model of the turbine expansion process with heat transfer, as discussed above.

The effect of compressor heat transfer on compressor efficiency has been explored in other studies [3–5] using the same process of comparing tests at different turbine inlet temperatures (TITs). The results are not entirely clear but indicate that the compressor diabatic efficiency does not show a simple dependency on turbine inlet temperature. There is the additional influence is the lubricating oil, which removes some of the heat conducted through the bearing housing from the turbine to the compressor.

In the same program, an effort was made to estimate the heat transfer to the oil by measuring the oil flow rate and temperature increase. As Fig. 1 shows, this has two components, being the heat generated in bearing friction and heat transferred by conduction from the turbine through the bearing housing. In the cold test it was assumed that the conducted heat transfer was negligible, and from the results, the bearing friction loss was correlated against a Reynolds number and a Strouhal number, both based on shaft speed. During hot testing, this constituted only about 5% of the total heat to oil at low turbocharger speed, rising to 20% near the design speed, thus, demonstrating the powerful cooling effect of the oil.

By comparing hot and cold test results, it is possible to estimate the net heat transfer from the turbine to the oil and to the compressor. Then in hot testing, assuming that the external heat transfers from the compressor and bearing housing are negligible, the external heat transfer of the turbine is equal to the net change in total enthalpies of the exhaust gas, air, and oil. On this basis it is estimated that the external heat transfer from the turbine accounts for approximately 70% of the total turbine heat transfer. There is a considerable scatter in the data, which is not explained, but presumably is caused by different operating conditions and turbine inlet temperatures. The fraction of the total turbine heat transfer to the oil is roughly 25%, and the remainder (about 5%) is internal heat transfer to the compressor. This breakdown is, of course, specific to the tested turbocharger and relies on some untested assumptions, but is broadly in line with the general understanding that the hot turbine is by far the largest heat source and that the oil has a very significant cooling effect.

1.3 Heat Transfer Modeling. In addition to the purely experimental testing described above, attempts have also been made to model the heat transfer processes occurring in a turbocharger. The solution method described by Chapman et al. [7] uses a finite element analysis of the turbocharger structure to determine the distributions of temperature and heat flux. The fluid surface boundary conditions required for this are the fluid state and the convective heat transfer coefficients. On the internal surfaces, these were calculated using a CFD computation. Unfortunately, it is not clear how the external surfaces were handled in this model.

It is uncertain whether the method is completely satisfactory as a predictive tool, since there appear to be large differences in the measured and predicted compressor exit temperature ranging from 10 K at about 60% speed up to 40 K at 100% speed. These differences are equivalent to about 25–30% of the total temperature increase in the compressor.

The conjugate heat transfer (CHT) method [2,8,9] is a mathematically more sophisticated approach, which involves the direct coupling of the fluid flow and solid body computations using the same discretization and numerical principles. This is computationally intensive and run times are lengthy. In the fluid flow, the Navier–Stokes equations are solved for the fluid state and velocity, and in the solid elements, the Fourier equation is solved for the temperature. The fluid boundary conditions are the fluid conditions at inlet and exit of the turbocharger determined by conventional measurement, and the external surface temperature was obtained by a thermographic camera [8]. This avoids the need for explicit modeling of the external heat transfer, which can vary from gas stand to engine operation, and is likely to be a source of considerable uncertainty. A small number of surface-mounted resistance temperature devices (RTDs) were also used for point measurements to check the surface temperatures from the thermographic images.

One interesting aspect of the thermographic measurements was the determination of the emissivity coefficients for each housing component of the turbocharger. This was necessary because the external heat transfer is a combination of radiation and convection, whereas the CHT model includes the effects of convection only. The thermographic results therefore had to be adjusted for the effects of radiation. Although the emissivity coefficients so determined were specific to the turbocharger tested, all turbocharger manufacturers tend to use similar materials and manufacturing processes for these components, and so the emissivity results are likely to be valid for most commercial turbochargers.

The published results [2] are largely concentrated on the compressor heat transfer. Predictions of the heat flux to the inner surface of the compressor housing, as a function of distance from the inlet, show negligible heat flux until the leading edge of the impeller. Thereafter the heat flux increases as a result of the compression process, with the largest heat flux occurring at the lowest mass flow and, therefore, the highest pressure rise. This result argues in favor of the thermodynamic model proposed in this project, in which the heat transfer occurs after the work transfer. However, no comparative test data are shown, and it is not possible to draw any conclusions about the accuracy of prediction.

1.4 Summary. The thermodynamic analysis can be used to identify the heat transfer processes that must be quantified in order to determine and ultimately to create a predictive model of the actual diabatic performance of the turbocharger. In turn, this defines the instrumentation and test requirements. Comparing these with test information obtained from the literature has proved difficult because full information is rarely given, but it is quite apparent that this present project goes beyond what has been published elsewhere. Even though only limited information is available, it is possible to draw some conclusions that were used to inform the present test program.

- The heat transfers of greatest magnitude and significance to turbocharger performance on-engine are, perhaps not surprisingly, external from the turbine to the environment and internal from the turbine to the bearing housing.
- Radiation makes an appreciable contribution to the external heat transfer.
- The bearing oil has an important function in cooling the turbocharger and acts as a heat sink.
- Internal heat transfer from the bearing housing to the compressor makes a considerable difference to the measured compressor performance, but the magnitude of this heat transfer is largely unaffected by the turbine gas temperature.
- Confirmation of the importance of the bearing oil in the internal heat transfer would come from tests with varying oil temperature, but this does not appear to have been done.
- The external heat transfer from the compressor to the environment appears to be less influential than the internal heat transfer from the bearing housing.

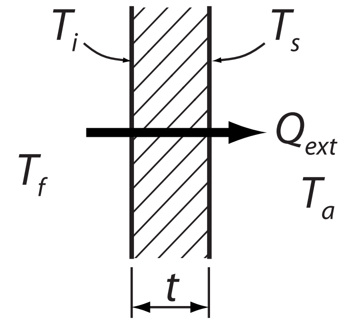

Fig. 2 Heat transfer through housing wall

2 Heat Transfer Methodology

In this section, the heat transfers that occur in a turbocharger, based on Fig. 1, and the methods by which they can be modeled are considered.

2.1 External Heat Transfer. External heat transfers from all parts of the turbocharger are combinations of convection to the air surrounding the turbocharger, radiation to surrounding parts and equipment, and conduction through the connecting pipes and mountings of the turbocharger. In this analysis, the effects of conduction are treated simply. In practice it is likely that some components immediately adjacent to the turbocharger will be at temperatures very similar to the relevant parts of the turbocharger itself (e.g., the exhaust manifold will be at about the same temperature as the inlet of the turbine housing). Differences are most likely to occur in medium and large turbochargers that require mounting provisions separate from the flanges. These may be connected to the engine block or head or to a structural part separate from the engine itself. The complete provision for such effects in the model would require a large amount of physical detail of the vehicle system and is likely to make the model quite unwieldy. It was therefore assumed that conduction effects can be modeled by means of a simple heat conduction coefficient. For similar reasons, in treating radiation and convection, it was assumed that there is an ambient temperature, which is the "sink" temperature for all of the external heat transfer terms. This implies that there are no significant temperature gradients in the environment surrounding the turbocharger. Given these provisos, the external heat transfer can be written in general terms as

$$\begin{aligned} Q_{\text{ext}} &= Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{cond}} \\ &= \bar{h}_s A_s (T_s - T_a) + \varepsilon \sigma (T_s^4 - T_a^4) + \kappa A_c (T_s - T_a)/x \end{aligned} \tag{1}$$

However, it is also necessary to relate T_s, which is the outer surface temperature, to the local fluid temperature on the inner wall surface and thence to the fluid temperature, since this is the temperature that actually drives the external heat transfer. Consider a simple one-dimensional heat transfer through a housing wall, as shown in Fig. 2

$$\frac{Q_{\text{ext}}}{A} = \bar{h}_i (T_f - T_i) = \frac{\kappa}{t}(T_i - T_s)$$

$$(T_f - T_i) = \frac{Q_{\text{ext}}}{A\bar{h}_i}$$

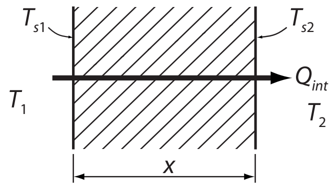

Fig. 3 Turbocharger internal heat transfer

$$(T_i - T_s) = \frac{Q_{\text{ext}}}{A\kappa/t}$$

$$\text{hence} \quad T_f - T_s = \frac{Q_{\text{ext}}}{A}\left(\frac{1}{\bar{h}_i} + \frac{t}{\kappa}\right) \tag{2}$$

For modeling external heat transfer, Eqs. (1) and (2) can be solved simultaneously for the unknowns Q_{ext} and T_s, provided the internal and external convective heat transfer coefficients, the thermal conductivity of the wall material, and the other terms in Eq. (2) are known.

2.2 Internal Heat Transfer. The internal heat transfer is a combination of convection and conduction, i.e., it is similar to the external heat transfer problem except that radiation effects are assumed to be negligible. This can also be addressed by considering the one-dimensional heat transfer from a source fluid temperature T_1 to a sink fluid temperature T_2 via a solid body of length x in the direction of the heat transfer and thermal conductivity κ. The wetted surface temperatures are denoted by T_{s1} and T_{s2} (Fig. 3). In practice, T_1 and T_2 can be any two fluids in the turbocharger: compressor air, turbine exhaust gas, or lubricating oil

$$\frac{Q_{\text{int}}}{A} = \bar{h}_1(T_1 - T_{s1}) = \frac{\kappa}{x}(T_{s1} - T_{s2}) = \bar{h}_2(T_{s2} - T_2) \tag{3}$$

$$T_1 - T_2 = \frac{Q_{\text{int}}}{A}\left(\frac{1}{\bar{h}_1} + \frac{x}{\kappa} + \frac{1}{\bar{h}_2}\right) \tag{4}$$

Providing the fluid temperatures T_1 and T_2 are known, together with the various heat transfer parameters, Eq. (4) models the internal heat transfer Q_{int}.

2.3 Practical Requirements. The equations outlined above provide the basis for predicting the heat transfers in a turbocharger. In order to be able to use such models, it is necessary to know the following:

- fluid and ambient temperatures, measured on test
- surface emissivity of casings, based on thermograph measurements [8]
- thermal conductivity of all casing materials, using published data
- convective heat transfer coefficients at all fluid/structure interfaces

Forced convection can be expressed in a general form as

$$\text{Nu} = a \ \text{Re}^b \ \text{Pr}^c \tag{5}$$

where $\text{Nu}=\bar{h}L/\kappa$ is the Nusselt number, $\text{Re}=\rho CL/\mu$ is the Reynolds number, $\text{Pr}=C_p\mu/\kappa$ is the Prandtl number, and a, b, and c are arbitrary constants. The length scale L is the streamwise distance used as a measure of the growth of the boundary layer through which convection occurs. In free convection, the Reynolds number tends to zero, and buoyancy forces dominate over viscous forces. In this case, which may pertain to the external heat transfer on the outer surface of the turbocharger in circumstances where there is little or no ventilation, Eq. (5) is replaced with

$$\text{Nu} = d \ \text{Gr}^e \ \text{Pr}^f \tag{6}$$

where $\text{Gr}=\beta g \rho^2 L^3 \Theta/\mu^2$ is the Grashof number, and d, e, and f are arbitrary constants.

3 Testing and Instrumentation

Three turbochargers were tested in this project. All were of similar size and flow capacity and were commercial units for automotive truck use. They are designated turbochargers A, B, and C. Turbochargers B and C had identical compressors and turbines and differed in that B used fluid film bearings, and C used rolling element bearings.

Each turbocharger was set up in a conventional cold gas stand, which had the facility to heat the turbine inlet air electrically, but only to about 500–550 K and not as far as typical operating temperatures. The test schedule comprised operation at a series of set speed and turbine inlet temperatures, in each case varying the turbine flow rate within the range of stable compressor operation.

The external heat transfer from the turbocharger could be controlled by changing the environmental conditions. An enclosure could be placed around the turbocharger. This enclosure was ventilated by a fan, and by controlling the fan speed, it was possible to vary the velocity of air across the turbocharger. In still air conditions, the turbocharger was run in the test cell without the enclosure, so that only free convection occurred on the external surfaces.

In addition, adiabatic tests were performed on each turbocharger. The intention of this was to eliminate all internal and external heat transfers, so that the heat transfer to the lubricating oil would be solely due to bearing friction. These test results were used to provide estimates of the oil temperature increase due to bearing friction alone, which in nonadiabatic tests could be subtracted from the total oil temperature increase, the result being a measure of the internal heat transfer to the oil.

The adiabatic tests were performed by eliminating, as far as possible, all heat transfer by running the complete turbocharger at constant temperature. This involved insulating the outside surfaces of the turbocharger and the supply lines, at least as far as the inlet and outlet thermocouple locations, to reduce the external heat transfer to a minimum. The internal heat transfer was minimized by controlling the turbine air inlet temperature to be equal to the compressor air delivery temperature at each operating point. In all cases, this was sufficiently far above ambient to prevent any water vapor condensing in the turbine expansion. The adiabatic test requirements, in combination with the surge and choke limits of the compressor, unfortunately meant that only a small range of conditions could be measured adiabatically.

In addition to the conventional instrumentation used for compressor and turbine performance, the turbochargers were instrumented for a full assessment of energy flows. The additional instrumentation comprised the following.

- oil inlet and exit temperatures
- oil flow rate
- thermocouples mounted on the internal and external surfaces of the compressor and turbine housings
- compressor back face and turbine heat shield thermocouples
- bearing housing thermocouples. Access to the bearing housing was limited, so that the coverage was incomplete.
- ambient temperature and air flow rate through the enclosure when fitted

4 Turbocharger Surface Heat Temperature Results

It was hoped that the thermocouples installed on the accessible surfaces of the turbochargers would provide some insight into the spatial distributions of temperature, as well as providing average temperatures that could be used for determining the heat transfers in the various parts of the turbocharger. This was done for all the test conditions because the ultimate objective of this work was to

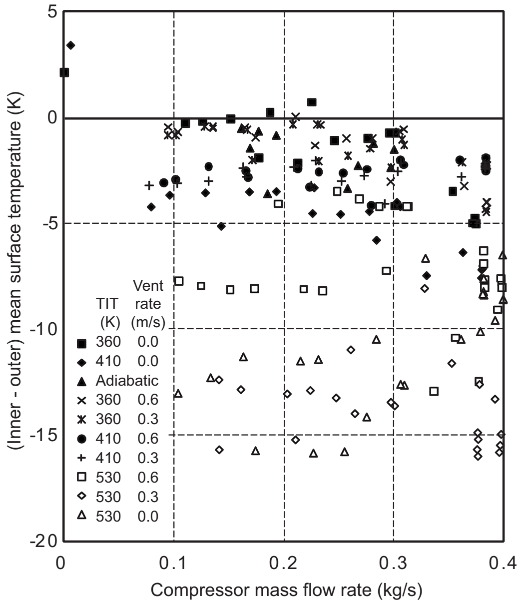

Fig. 4 Turbocharger A compressor, showing average temperature difference between inner and outer housing surfaces as a function of air flow rate, turbine inlet temperature, and external ventilation

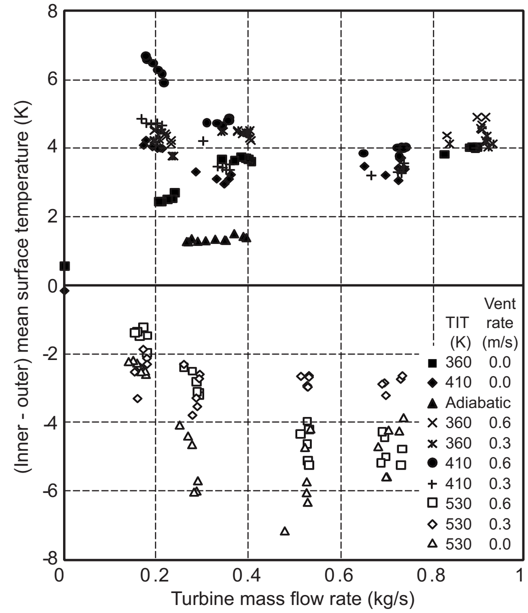

Fig. 5 Turbocharger A turbine, showing average temperature difference between inner and outer housing surfaces as a function of air flow rate, turbine inlet temperature, and external ventilation

arrive at heat transfer models that were either independent of operating condition, or could be correlated against operating condition.

For the compressor housing, the differences between the inner and outer surface temperatures lay within the scatter of the data, as did the variation around the circumference of the housing. The only firm conclusions that can be drawn are that the compressor housing is very nearly isothermal and any heat transfer from the compressed air to the environment through the housing must be very small. However, when all of the temperature measurements on the inner and outer surfaces are averaged, a clearer picture emerges (Fig. 4). For the adiabatic test, and at the lowest TIT, the compressor housing temperature difference is only a very few degrees and within the experimental uncertainty. At a higher TIT, the difference increases slightly but only at the highest TIT does the temperature difference become significant. In this last case, the temperature difference varies only slightly with compressor air mass flow rate, but is reduced by several degrees at the highest ventilation rate.

For the turbine housing, the scatter in the data again makes interpretation difficult and is further complicated by the large thermal masses of the waste gate and the actuator attachments that are cast integral with the housing. The overall averages of the inner and outer surface temperatures are shown as a function of turbine air flow rate in Fig. 5. The adiabatic test shows a mean difference of about 1 K within the accuracy of the thermocouples. It is very striking that the mean difference of the tests at TIT=530 K is consistently negative, implying that the outer surface of the housing is hotter than the inner surface, whereas for lower TIT values the difference is positive. The effect of external ventilation, at constant TIT, is much smaller and a clear trend is not evident. This does suggest that the external heat transfer has less influence on the housing temperature than the other heat transfer effects, which are strongly affected by the turbine inlet temperature.

Many of the internal surfaces of the bearing housing could not be accessed for temperature measurement, and the coverage was not sufficient to give a spatial survey of the surface temperatures. Figure 6 therefore shows the oil inlet and exit temperatures, together with the average of the bearing and outer surface temperatures, as a function of compressor air flow rate for one test. The difference between the oil inlet and exit temperatures is a combination of the bearing friction and the heat transfer from the turbine to the oil. The housing surface temperature follows closely the oil exit temperature, and the bearing temperature is partway between the oil inlet and exit temperatures. By comparison, in adiabatic testing, the oil inlet to exit temperature difference is much smaller, typically only about 10 K. This is a measure of the relative effects of bearing friction and heat transfer to the oil.

5 Heat Transfer Analysis

The analysis was performed in two phases. First, the test data (primarily, the fluid and surface temperature measurements) were

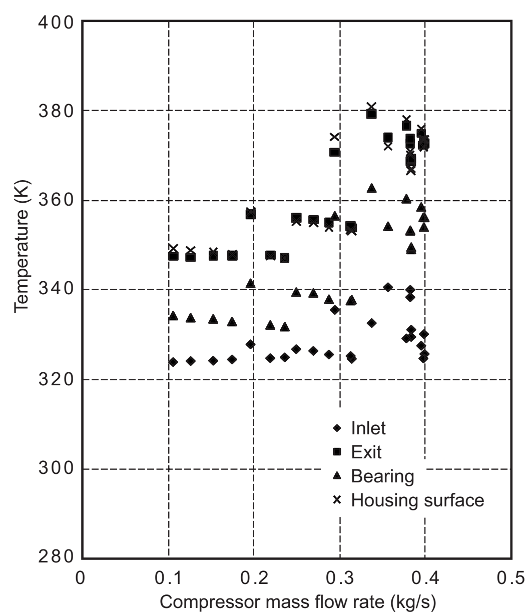

Fig. 6 Turbocharger A bearing housing temperatures, TIT =530 K, external vent rate=0.6 m/s

Table 1 Summary of convective heat transfer coefficients

	Turbocharger A			Turbocharger B			Turbocharger C		
Forced convection constants	a	b	c	a	b	c	a	b	c
Turbine housing inner surface	0.032	0.7	0.43	0.032	0.7	0.43	0.032	0.7	0.43
Turbine housing outer surface	0.6	0.4	0.33	0.6	0.4	0.33	0.6	0.4	0.33
Compressor housing inner surface	0.032	0.62	0.43	0.032	0.62	0.43	0.032	0.62	0.43
Compressor housing outer surface	0.032	0.8	0.43	0.032	0.8	0.43	0.032	0.8	0.43
Turbine back face	0.6	0.4	0.33	0.6	0.4	0.33	0.6	0.4	0.33
Compressor back face	0.032	0.8	0.43	0.032	0.8	0.43	0.032	0.8	0.43
Oil	0.04	0.8	0.43	0.04	0.8	0.43	0.08	0.8	0.43
Free convection constants	d	e	f	d	e	f	d	e	f
Turbine housing outer surface	0.2	0.25	0.25	0.1	0.25	0.25	0.1	0.25	0.25
Compressor housing outer surface	0.678	0.25	0.25	0.2	0.25	0.25	0.2	0.25	0.25

analyzed in order to estimate the magnitudes of the heat transfers occurring in the turbocharger. Then, correlations of the heat transfers from different operating conditions and turbochargers were achieved and checked for consistency in the overall energy balance of the turbocharger.

A certain selectivity was required in correlating the heat transfer data because when all of the conduction, convection, and radiation processes are considered, the number of heat transfer coefficients (thermal conductivity, convective heat transfer coefficient, and emissivity) exceeds the number of separate measurements. It was decided that the greatest uncertainty would lie in the convective heat transfer coefficients and that suitable values for the thermal conductivities and emissivities could be obtained from the literature. It is, therefore, inevitable that any errors or uncertainties in these parameters will be subsumed into the convective heat transfer coefficients and correlations. The results obtained here are conditional on the assumed values for these parameters, and any deviations from these values will be reflected in a greater uncertainty in the heat transfer estimates.

For the analysis of test data that preceded the correlation phase of data reduction, the situation was different in that there were more temperature measurements available than are strictly necessary to deduce all of the heat fluxes, particularly because conservation of energy can be applied to the complete turbocharger and to each major component separately. There is thus a certain redundancy in the information available, which was used for checking and estimating uncertainties.

5.1 Energy Network Model. The energy network model in Fig. 1 shows the power and heat fluxes that formed the basis of the heat transfer calculation. In this one-dimensional model, the compressor, turbine, and bearing system were each treated as isothermal nodes of a temperature determined by the relevant working fluid. Temperature change in the working fluid was considered when determining the enthalpy fluxes required to complete the energy balance of each component.

Based on the test data, the internal and external heat transfers were calculated using the appropriate combinations of convection, conduction, and radiation processes, as described by Eqs. (1)–(4). The energy conservation equations were then applied to the compressor and turbine in order to estimate the mechanical power transfer of each (see Fig. 1)

$$P_C = m_C(h_{0C,\mathrm{out}} - h_{0C,\mathrm{in}}) + Q_{C,\mathrm{ext}} - Q_{C,\mathrm{int}} \tag{7}$$

$$P_T = m_T(h_{0T,\mathrm{in}} - h_{0T,\mathrm{out}}) - Q_{T,\mathrm{ext}} - Q_{T,\mathrm{int}} \tag{8}$$

Another redundancy check is that the difference between these two powers is equal to the power absorbed in bearing friction, which was separately measured in the adiabatic tests.

The conservation of energy equation can also be applied to the bearing housing as follows (see Fig. 1):

$$m_{\mathrm{oil}}(h_{0\mathrm{oil,out}} - h_{0\mathrm{oil,in}}) = P_T - P_C + Q_{T,\mathrm{int}} - Q_{C,\mathrm{int}} - Q_{B,\mathrm{ext}} \tag{9}$$

Since all of these quantities have been measured or obtained earlier in the analysis, this equation is used as a check on the accuracy of the results.

5.2 Convective Heat Transfer Correlations. The coefficients in the convective heat transfer equations (5) and (6) that satisfy the requirements of the energy network model above, for the three turbochargers, are listed in Table 1. These were obtained by fitting the measured temperatures to the model. The forced convection equation (5) was used in all analyses except for external heat transfer to the environment in the cases of zero ventilation, for which the free convection equation (6) was used.

The fact that the coefficients listed here are, with few exceptions, common to all of the turbochargers tested, demonstrates that the heat transfer network model that is largely independent of the turbocharger model. The only exceptions are the heat transfer to the oil, which is significantly different between the turbochargers with fluid film bearings (A and B) and with rolling element bearings (C) and free convection, which presumably depends more than forced convection on the details of the housing geometry, but in reality free convection is only relevant when there is no flow of air around the turbocharger. It is noteworthy that studies of heat transfer to engine exhaust manifolds [10,11] also show that the heat transfer can be correlated against Reynolds number with an exponent that is typically about 0.75, which is quite similar to the value determined here for the similar situation of internal heat transfer to the turbine housing.

An example of the heat fluxes predicted by the model for Turbocharger A is illustrated in Fig. 7, which shows the external (to the environment) and internal (to/from the bearing housing) heat transfers for the compressor and turbine, across the range of testing. The turbine heat transfers are strong functions of the turbine inlet temperature, and at the highest value of TIT, the variation with Reynolds number is also strong (the Reynolds numbers here are local Reynolds numbers and have different length scales for the internal and external heat transfers). Comparing magnitudes, the internal heat transfer is much greater than the external heat transfer, illustrating the cooling effect of the lubricating oil and also demonstrating how lagging a turbocharger externally, but without paying attention to the internal thermal matching will not provide anything approaching adiabatic test results.

For the compressor, the variation in heat flux with turbine inlet temperature is much less strong because the lubricating oil acts as a heat sink and is more a function of the compressor operating condition. The external heat transfer is consistently low, but clearly not a negligible fraction of the overall heat transfer. The

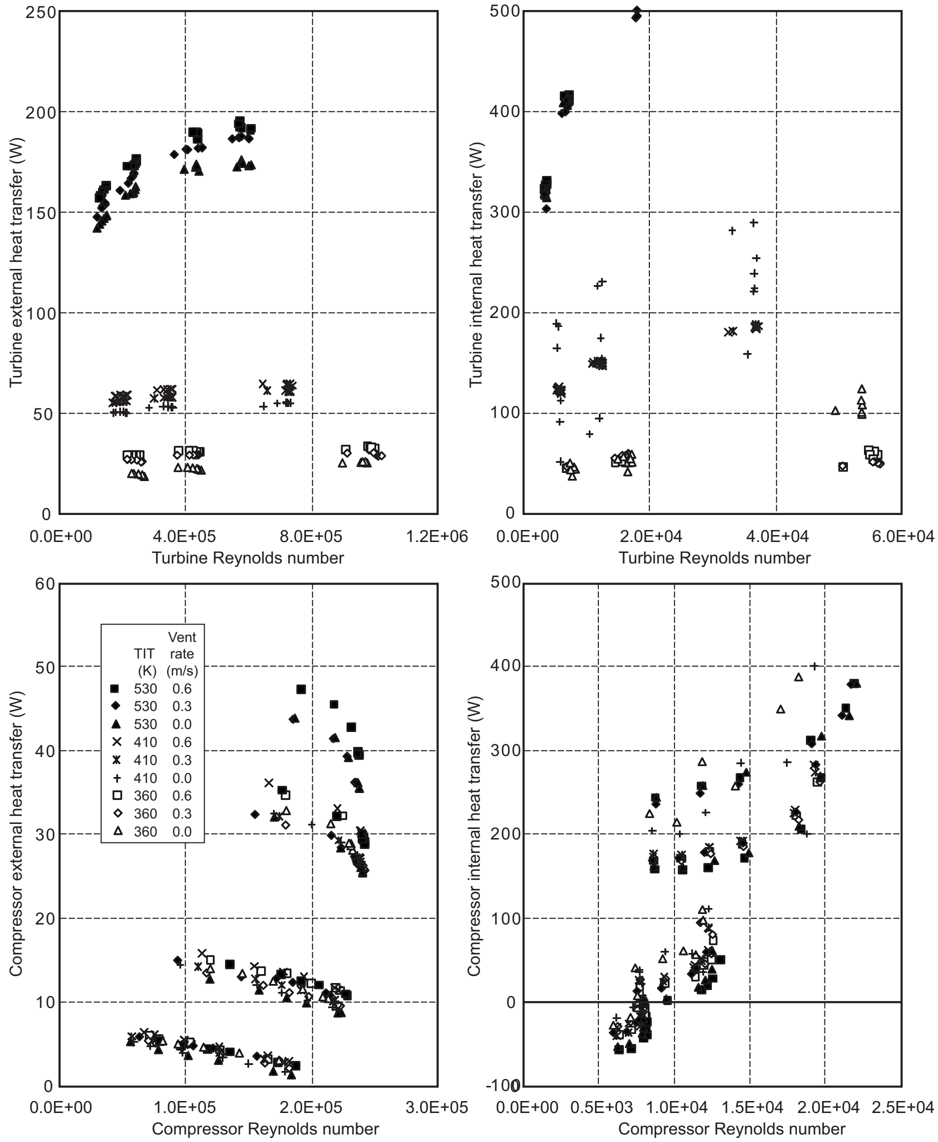

Fig. 7 Heat transfers for Turbocharger A, as functions of turbine inlet temperature and external ventilation

internal heat transfer is mostly positive, i.e., from the compressed air to the bearing housing, but at a very low Reynolds number corresponding to low pressure ratio, the air is actually heated by heat transfer from the bearing housing.

The quality of the model was judged by assessing the overall energy balance of the turbocharger. Figure 8 is an example showing the residual heat flux to the oil, as a fraction of the turbine power output, as predicted by the heat transfer network model for one of the tested turbochargers. A model that satisfied conservation of energy absolutely would give a value of zero residual for all conditions. In practice, all of the forced convection cases are within an error band of $-1/+2\%$. The free convection cases, however, show significantly larger residuals, particularly at low turbine flow (low Reynolds number) conditions where all of the heat fluxes are small and measurement uncertainties are consequently large. The model must therefore be regarded as less reliable for low flows and free convection, but as previously noted, this is not a common condition for the operation of automotive turbochargers.

6 Conclusions

A one-dimensional heat transfer network model of a turbocharger has been developed in order to predict the external heat transfers from the working fluids to the environment and the internal heat transfers within the turbocharger, as part of a project aimed at improving the overall turbocharger simulation and engine matching. Tests were conducted on three turbochargers extensively instrumented with thermocouples, and the models were developed using, and checked against, the database of information so obtained. Conduction and radiation effects can be calculated using existing material thermal properties and data, and so the principal task in developing the model was to simulate the convective heat transfer components. It was found that using conventional convective heat transfer correlations, a set of heat transfer coefficient values could be obtained that were largely independent of the turbocharger model. The only exceptions to this were some external free convection components, which are only required in unventilated turbocharger installations, and the heat transfer to oil, which varies depending on whether the turbocharger bearings are

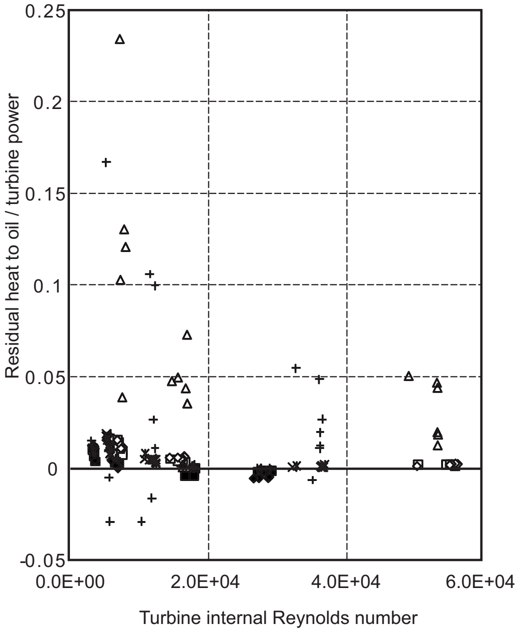

Fig. 8 Overall energy balance, as a check on heat transfer model accuracy

fluid film or rolling element. The model was shown to be able to predict the heat transfers with good confidence, as demonstrated by comparisons with the measured temperatures and by satisfying overall conservation of energy for the tested turbochargers.

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Nomenclature

A = area
C = velocity
C_p = specific heat at constant pressure
Gr = Grashof number
h = enthalpy
$\bar{h}$ = convective heat transfer coefficient
L = length
m = mass flow rate
Nu = Nusselt number
P = power
Pr = Prandtl number
Q = heat transfer
Re = Reynolds number
T = temperature
t = thickness
x = distance
β = coefficient of volume expansion
ε = emissivity of surface
κ = thermal conductivity
μ = dynamic viscosity
ρ = density
σ = Stefan–Boltzmann constant
Θ = temperature difference between surface and fluid

Subscripts

a = ambient
B = bearing
C = compressor
c = cross section
cond = conduction
conv = convection
ext = external
f = fluid
i = inner surface
int = internal
rad = radiation
s = surface
T = turbine

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