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EVALUATION OF HEAT TRANSFER EFFECTS ON TURBOCHARGER PERFORMANCE

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ABSTRACT

Test data on several small turbochargers with different levels of heat transfer from the turbine to the compressor have been obtained through cooling of the turbocharger center housing and by testing in hot and cold test stands. This data identifies the strong effect of the heat transfer on the apparent efficiency of the compressor and turbine, particularly at low speeds and low mass flows.

A simplified theory is used to explain the apparent effect of the heat transfer on the work input and efficiency. The results confirm that conventional performance maps underestimate the efficiency of the compressor stage and overestimate the efficiency of the turbine by as much as 20% points at low speeds. A correction procedure for this effect is defined which converts performance maps obtained with heat transfer to performance maps for adiabatic conditions (for both compressor and turbine) without any prior knowledge or measurement of the heat transfer. The practical significance of the results with regard to turbocharger performance and the relevance to a broader class of turbomachines is discussed.

NOMENCLATURE

a	=	speed of sound (m/s)
c	=	absolute flow velocity (m/s)
c_p	=	specific heat at constant pressure (J/kg K)
c_s	=	slip velocity (m/s)
D_2	=	impeller tip diameter (m)
h	=	specific enthalpy (J/Kg)
j	=	specific dissipation work (J/Kg)
k_c	=	non-dimensional heat transfer coefficient (-)
$\dot{m}$	=	mass flow rate (Kg/s)
M_{u2}	=	tip speed Mach number, $M_{u2}=u_2/\sqrt{(\gamma R T_{t1})}$ (-)
n	=	polytropic exponent (-)
p	=	static pressure (N/m^2)
P_c	=	Compressor shaft power (W)
q	=	specific heat transfer (J/Kg))
$\dot{Q}$	=	heat transfer rate (J/s)
R	=	gas constant (J/KgK)
s	=	specific entropy (J/kgK)
T	=	temperature (K)
u_2	=	impeller blade tip speed (m/s)
v	=	specific volume (m^3/Kg)
w_t	=	specific shaft work (technical work) (J/Kg)
y	=	specific pressure change work (J/Kg)

Greek Symbols

β	=	relative flow angle (°)
γ	=	isentropic exponent (-)
η	=	efficiency (-)
λ	=	work input or enthalpy rise coefficient $\lambda = \Delta h / u_2^2$ (-)
ρ	=	density (kg/m^3)
σ	=	non-dimensional slip velocity (c_s/u_2) (-)
ϕ	=	flow coefficient, $\phi = \dot{m}/(\rho_{t1} u_2 D_2^2)$ (-)
ψ	=	pressure rise coefficient $\psi = \eta\lambda$ (-)

Subscripts

1	=	inlet conditions
2	=	outlet conditions
a	=	apparent
c	=	compressor
p	=	polytropic
rev	=	reversible
s	=	isentropic
t	=	total and turbine

INTRODUCTION

To obtain a better understanding of the energy balance of an engine with turbochargers whole-engine simulation systems are used. To analyze a power-train with a turbocharger these require accurate performance curves for the turbo-components as input to the simulations. The necessary performance maps of the turbine and compressor for such simulations are generally measured on hot turbocharger gas test stands in which, as in the real engine environment, there is a heat flow from the turbine to the compressor. Considerable interest has been shown in recent years on understanding the effect of this heat transfer on performance, both in connection with turbochargers and in micro-turbomachinery applications; see Van den Braembusche [1], Gong et al. [2], Sirakov et al. [3], Casey and Fesich [4], and Baines et al. [5] and many references given in these publications.

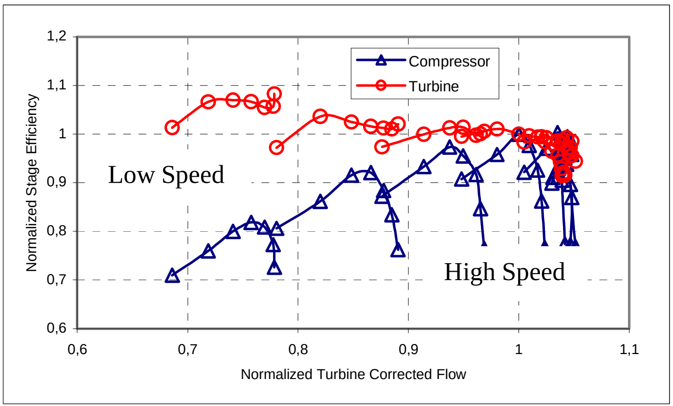

Figure 1: Typical efficiency variation at low speed from performance maps for a turbocharger compressor and a turbine

It is known that in turbochargers the amount of heat transfer is small relative to the work transfer from the turbine to the compressor and has no really noticeable thermodynamic or aerodynamic consequences on the pressure ratios and true performance of the turbocharger (except possibly during extreme operating conditions such as cold start, which concerns primarily the turbine.) The test data to be presented later in this paper also confirm this view. The possible aerodynamic effects associated with heat transfer are related to changes in density due to the change in temperature, and the associated change in velocity triangles due to this. The thermodynamic effects are related to the fact that more work is required to compress a gas as its temperature is increased due to the diverging constant pressure lines on a T-s diagram. Nevertheless, the small amount of heat transfer causes a serious problem with the apparent compressor and turbine efficiencies in the performance maps. The compressor efficiency is generally evaluated on the assumption of an adiabatic flow and the temperature rise caused by the heat transfer is then interpreted as additional work transfer. The heat flow causes an apparent increase in the power consumption and an apparent drop in efficiency of the compressor. As the turbine power and efficiency is then derived from the apparent power of the compressor the turbine appears better than it really is and a shift of efficiency from compressor to turbine results, especially at low speeds, see figure 1.

The erroneous performance maps cause several problems. Firstly, they may cause significant performance matching issues in whole engine powertrain simulations. It is often necessary in matching calculations with an engine to make use of a compressor and a turbine from different turbocharger tests to develop an optimum solution for a particular engine. If the maps include errors due to heat transfer, then the switching of components from one turbocharger to another is inexact as the amount of heat transfer may be different in the different turbochargers. Secondly, the drop in performance of the compressor and rise in efficiency of the turbine at low speeds is not related to any real increase in the losses and is therefore not predicted by any conventional preliminary design methods or computational fluid dynamics tools (CFD). This causes considerable uncertainty when trying to understand the performance and loss mechanisms of the turbomachinery and when trying to develop improved correlations for the losses. The second author has seen examples where the apparent additional compressor work at low speeds has been modeled by a decrease in the slip factor and as additional parasitic work input, both of which are clearly incorrect! In addition, cases are known where the apparent improvement in turbine efficiency at low speeds leads to isentropic efficiencies that are greater than unity, which would actually violate the second law of thermodynamics. Thirdly, and more importantly, an increasing number of engine manufacturers rely entirely on performance maps to make turbocharger selection and erroneous maps may ultimately lead to suboptimal engine system performance. In addition, there is an image issue with customers using these maps due to the apparent poor low speed performance of the compressor, which will vary for different designs of turbocharger and for different suppliers. Finally the trend towards increased use of turbocharging in small petrol engines with higher turbine inlet temperatures and smaller sizes will lead to a larger heat transfer than in diesel engines. This requires that this heat transfer effect is properly modeled in the maps to accurately calculate the matching of the components. This paper provides a technique for more accurate but relatively simple modeling of these effects in the performance maps.

The paper first describes a theoretical analysis of the heat transfer which leads to equations for the correction of the work input and efficiency of the compressor and turbine due to the heat transfer, following the approach of Casey and Fesich [4]. Their analysis for compressors is simplified here, and is extended to include the performance of the turbine and the overall turbocharger. The equations include a single non-dimensional empirical coefficient for the strength of the heat transfer. This can be determined from conventional turbocharger hot rig test data over a range of speeds and allows the apparent diabatic test performance (with heat transfer) to be corrected to the true adiabatic conditions (without heat transfer). It is important to note that this correction can be performed without any prior knowledge or measurements of the heat transfer levels in the turbocharger.

This is followed by the description of a systematic series of tests on turbocharger test rigs under the influence of heat transfer from the turbine to the compressor in which the amount of heat transfer is varied. The experimental test data includes measured characteristic curves of high accuracy at different rotational speeds with and without cooling of the bearing housing and with a hot (620°C) and a cold turbine (turbine inlet temperature set to the compressor exit temperature). The compressor impellers used are small (D_2 ~ 40mm) and these have been selected as the heat transfer effects should be larger on a smaller machine (with a higher ratio of surface area to volume). These test data identify the strong effect of the heat transfer on the apparent efficiency of the compressor and turbine, particularly at low speeds and low mass flows.

The practical significance of the results with regard to turbocharger performance is discussed and some considerations of possible further refinements of the procedure are given. A discussion of the applicability of the method to a broader class of turbomachines operating under diabatic conditions is also provided.

EFFICIENCY ANALYSIS WITH HEAT TRANSFER

Compressor efficiency

We consider the compression process between the inlet state (1) and outlet state (2) of a steady flow in a compressor. As turbocharger suppliers generally base their analysis on isentropic or adiabatic efficiencies, this analysis is described first. It needs, however, to be firmly stated that the use of an isentropic or adiabatic efficiency to describe a diabatic flow is thermodynamically flawed, see Casey and Fesich [4].

For the isentropic analysis we define a virtual or hypothetical state 2s, which has the same outlet pressure as the real process but the same entropy as at the inlet state. From the second law formulated in terms of the Gibbs equation:

$$dh = vdp + Tds \qquad 1$$

we see that the enthalpy change in an adiabatic compression process, with no entropy change due to heat transfer, is the sum of the pressure change work (*vdp*) and the dissipation (*Tds*). The isentropic process then describes the ideal work of a perfect adiabatic machine with no dissipation losses and no change of entropy. For a constant entropy process we integrate equation 1 to obtain

$$h_{2s} - h_1 = \int_1^{2s} v\,dp \qquad 2$$

The combined work and heat transfer of the process is defined by the first law for steady flow through an open system as

$$w_{t,12} + q_{12} = h_2 - h_1 + \frac{1}{2}(c_2^2 - c_1^2) \qquad 3$$

If we have an adiabatic flow then the specific work input is the same as the change in enthalpy (for clarity we neglect the changes in kinetic energy and consider total conditions at states 1 and 2). In this way we determine the usual expression for the isentropic efficiency in an adiabatic flow as

$$\eta_s = \frac{\Delta h_s}{\Delta h} = \frac{\int_1^{2s} vdp}{h_2 - h_1} = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \qquad 4$$

The main problem with this equation for a diabatic flow is that the shaft power is no longer equal to the change in enthalpy, so the denominator of equation 4 is actually wrong. The measured enthalpy and temperature rise overestimates the actual work input due to the warming caused by the heat addition and this needs to be subtracted to get a proper measure of the real diabatic isentropic efficiency. Several authors have tried to make a correction for this effect. Firstly, we assume that the heat is added after the compression process. Then, if we further neglect the kinetic energy terms we obtain from equations 1, 2 and 4 that for a flow with heat transfer the “diabatic isentropic efficiency”, that is the true isentropic efficiency η_s with no heat transfer, can be derived from the apparent efficiency with heat transfer η_{sa} as

$$\eta_s = \frac{h_{2s} - h_1}{(h_2 - h_1) - q_{12}} = \frac{T_{2s} - T_1}{(T_2 - T_1) - q_{12}/c_p}$$

$$\eta_s = \frac{1}{\frac{1}{\eta_{sa}} - \frac{q_{12}}{c_p(T_{2s} - T_1)}} = \frac{1}{\frac{1}{\eta_{sa}} - \frac{q_{12}}{h_{2s} - h_1}} \qquad 5$$

This equation indicates that if heat is added to the flow then the apparent efficiency is actually lower than the real efficiency, as already shown in figure 1, and is a result obtained by several previous authors. The problem with the isentropic analysis is that the value of h_2 is actually floating, and depends on the heat addition distribution and thus the isentropic efficiency is changing for the same inlet conditions, pressure ratio and the same heat addition. This is why an assumption had to be made for equation 5 and the simplest one was used for this argument - adding all heat after the compression process.

A similar but thermodynamically correct analysis on the basis of a polytropic efficiency is explained in detail by Casey and Fesich [4]. This does not include any thermodynamic inconsistencies and the heat transfer can take place before, during or after the compression process. This results in a very similar equation for the apparent and real polytropic efficiencies, as follows:

$$\eta_p = \frac{1}{\frac{1}{\eta_{pa}} - \frac{q_{12}}{y_{12}}} \qquad 6$$

where y_{12} is the actual pressure change work of the process, see equation 7 below. For further details and an extensive discussion of the analysis readers are referred to the paper of

Casey and Fesich [4]. It should be noted that for turbochargers on the low speed characteristics with low pressure ratio, where the heat transfer effects are largest, there is no practical difference between the isentropic and polytropic analysis. In this paper, however, the polytropic efficiency is used in the analysis.

Compressor work input

For an adiabatic flow we can express the polytropic pressure rise coefficient in terms of polytropic efficiency and the enthalpy rise coefficient, as follows

$$\psi_p = \frac{\int_1^2 vdp}{u_2^2} = \frac{y_{12}}{u_2^2} = \frac{y_{12}}{w_{t12}} \frac{w_{t12}}{u_2^2} = \eta_p \lambda \qquad 7$$

In the diabatic flow we similarly obtain for the apparent performance

$$\psi_{pa} = \eta_{pa} \lambda_a \qquad 8$$

The measurements below show that the heat transfer has no effect on the pressure ratio, so the pressure rise coefficient with and without heat transfer is effectively unchanged. This identifies that the aerodynamic and thermodynamic effects of the heat transfer are small for the studied turbochargers.

It follows directly from equations 7 and 8 that the apparent increase in work input and enthalpy rise due to heat transfer effects is then related to an apparent decrease in the efficiency. We define the apparent enthalpy rise coefficient as

$$\lambda_a = \frac{\Delta h_{t12}}{u_2^2} = \frac{w_{t12} + q_{12}}{u_2^2} = \lambda + \frac{q_{12}}{u_2^2} \qquad 9$$

which includes the enthalpy rise due to work and heat transfer. This corresponds to the apparent measured enthalpy rise coefficient and the difference to the adiabatic coefficient can be given as

$$\lambda_a - \lambda = \frac{q_{12}}{u_2^2} \qquad 10$$

The true polytropic efficiency of the compressor stage can then be determined from the apparent efficiency with heat transfer as follows

$$\eta_p = \frac{1}{\frac{1}{\eta_{pa}} - \frac{(\lambda_a - \lambda)}{\psi_p}} \qquad 11$$

Note that under the assumption that the pressure coefficient does not change under the influence of heat transfer then this also leads to the equation

$$\eta_p = \eta_{pa} \frac{\lambda_a}{\lambda} \qquad 12$$

Turbine efficiency

The turbine enthalpy change is determined from the power consumption of the compressor and not from the turbine temperature measurements, so that we do not have an effect of the turbine heat transfer on the efficiency as in the compressor. If the actual enthalpy drop across the turbine were measured accurately there would however be a discrepancy between this and that derived from the compressor power. The heat loss from the turbine causes a larger temperature drop across the turbine than that which is due to the power production alone. The isentropic efficiency of the turbine (including bearing losses) can then be estimated from the compressor power as

$$\eta_t = \frac{P_c}{\dot{m}_t (h_3 - h_{4s})} = \frac{\lambda u_2^2 \dot{m}_c}{\dot{m}_t c_p (T_3 - T_{4s})} \qquad 13$$

The misinterpretation of the heat transfer to the compressor as compressor work input leads to an apparent error in the work coefficient so that the apparent turbine efficiency with heat transfer to the compressor is given by

$$\eta_{ta} = \frac{\lambda_a u_2^2 \dot{m}_c}{\dot{m}_t c_p (T_3 - T_{4s})} \qquad 14$$

So the true turbine efficiency can be calculated from the apparent turbine efficiency as follows

$$\eta_t = \eta_{ta} \frac{\lambda}{\lambda_a} \qquad 15$$

The misinterpretation of the heat transfer as work input leads to a higher apparent turbine efficiency and this effect increases on the low-speed characteristics, as shown in figure 1.

Overall turbocharger efficiency

It should be noted here that the form of equations 12 and 15 leads to the interesting result that the product of the compressor and turbine efficiencies in the case of the presence of heat transfer is not affected by the heat transfer. Because of this the overall turbocharger efficiency determined from the measurements is not changed by the presence of the heat transfer. It is simply the shift of efficiency from compressor to turbine that results from the effect of the heat transfer, and this happens in such a way that the improvement in turbine efficiency exactly compensates the reduction in compressor efficiency. Again this only holds true if the thermodynamic and aerodynamic effects of heat transfer are small as is the case for most turbochargers.

Heat transfer model

We now consider the effect of the heat transfer on the temperature change in the compressor. Clearly for a given amount of heat transfer the temperature rise will be higher for a lower compressor mass flow, so we can expect the effects of the heat transfer to decrease as the compressor speed and mass flow is increased. The specific heat flow then depends on the mass flow through the compressor and using conventional non-dimensional parameters we can derive that

$$\dot{m} = \rho_{t1} \phi_{t1} u_2 D_2^2 \qquad 16$$

and that the power input is

$$P_c = \dot{m} w_{t,12} = \dot{m}\lambda u_2^2 = \rho_{t1}\phi_{t1}\lambda u_2^3 D_2^2 \quad 17$$

The specific heat flow per unit mass flow can be expressed as

$$q_{12} = \dot{Q} / \rho_{t1}\phi_{t1} u_2 D_2^2 \quad 18$$

And we expect the difference between the apparent and real work coefficients to be given by

$$\lambda_a - \lambda = \frac{q_{12}}{u_2^2} = \frac{\dot{Q}}{\rho_{t1}\phi_{t1} u_2^3 D_2^2} \quad 19$$

where the denominator is proportional to the power of the compressor. We can express this non-dimensionally as

$$\lambda_a - \lambda = \frac{\dot{Q}}{\rho_{t1} a_{t1}^3 D_2^2} \frac{1}{\phi_{t1} M_{u2}^3} = k_c \frac{1}{\phi_{t1} M_{u2}^3} \quad 20$$

where k_c is a dimensionless coefficient that depends on the rate of heat transfer per unit area into the compressor.

It should be noted that this coefficient differs slightly from that used by Casey and Fesich [4] in that it is based on the global flow coefficient rather than the impeller outlet flow coefficient and so different numerical values result from this. If the heat transfer were to be constant across the whole performance map the analysis suggests that the difference between the apparent and true work coefficient is inversely proportional to the product of the cube of the impeller tip speed Mach number and the inlet flow coefficient, so that the largest effects can be expected close to surge on the low speed characteristics. This assumption of a constant heat transfer rate is not a necessary condition of the method, but it is shown below that this is a reasonable approximation in most cases. Discrepancies from the assumption of constant heat transfer are discussed later.

EXPERIMENTAL STUDIES OF HEAT TRANSFER EFFECTS IN TURBOCHARGER TEST RIGS

The objective of the tests described here are to demonstrate the effect of heat transfer on the turbocharger performance by directly changing the amount of heat transfer so that the non-dimensional heat transfer coefficient, as defined in equation 20, is changed. In the paper of Casey and Fesich [4] performance curves for two different turbochargers on several speed-lines were analysed to determine the value of the coefficient k_c for these compressors. This confirmed the use of an equation similar to equation 20 for these two turbochargers but no change in heat transfer rates was made. In the work presented here tests on 4 different turbocharger configurations are analysed, whereby the tests include cases in which the amount of heat transfer is deliberately modified. The modifications are made by cooling the center housing between the compressor and the turbine stages, by changing the turbine inlet temperature and by changing the turbocharger configuration retaining the same compressor geometry. This form of validation was not made by Casey and Fesich [4] who simply used the different speed lines to evaluate a change in the amount of heat transfer relative to the mass flow.

Description of test rigs and test procedures

The measurements are taken on a typical turbocharger test rig as used in industrial component testing, see the description by Baines [6]. In most tests presented here high pressure hot gas (constant T = 620°C) is driving the turbine but in some tests air at compressor exit temperature is used. The flow in the turbine is separate from the flow in the compressor. The turbine is directly coupled to the compressor with a shaft (Figure 2). The compressor speed is controlled by varying the turbine inlet pressure. At a constant speed, the compressor mass flow is controlled by a throttle valve at the compressor discharge. The compressor mass flow is measured with a long radius nozzle (bell mouth device ISO 5167-3) at the compressor inlet while the turbine and fuel flows are measured with orifice plates (5167-2) upstream of the burner. The compressor inlet and outlet static pressures and the turbine inlet pressure are measured with Druck 4000 sensors in piezometric chambers. The turbine exit pressure is assumed atmospheric. Compressor inlet and exit total temperatures are measured using platinum resistance thermometers (RTD PT100, IEC 751, Class A). Turbine inlet temperature is measured with Thermocouples (K-type, IEC 584-2, Class 1). The fuel flow is controlled to obtain the desired air to fuel flow ratio. The gas stand cell installation and measurements follows ASME PTC22 and PTC10 code. Corrected mass flow is obtained directly from mass flow measurements, ambient conditions, and turbine inlet conditions. Pressure and temperature ratios can be obtained for the compressor and the adiabatic isentropic efficiency can be computed. The experimental errors of the test rig instrumentation are considered to be ±1.3°C in temperature measurement, up to ± 200 N/m2 in pressure measurement and ±1.0% in mass flow or volume flow. On the low speed characteristics the observed repeatability in efficiency is ±1.5% points.

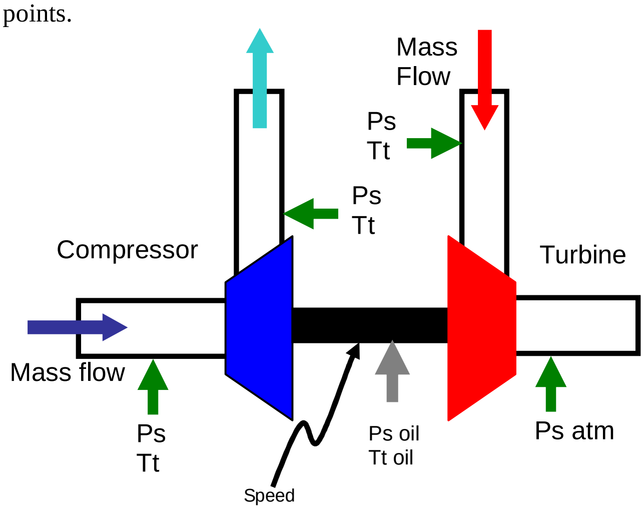

Figure 2: Schematic for the measurements made on the turbocharger gas stand used for this investigation

To obtain the combined efficiency for the turbine and the bearings a power balance equation is used. The total-to-static pressure ratio measured across the turbine stage and the turbine mass flow are used along with the information obtained for the compressor. Note that in this process the estimated power of the compressor is used so that any error in this repeats in the determination of the turbine efficiency. The compressor choke flow is defined at certain efficiency. The compressor surge line is determined by detecting fluctuations of compressor inlet and outlet pressures. Measurements are taken at a number of speed lines and for a number of points at each speed line.

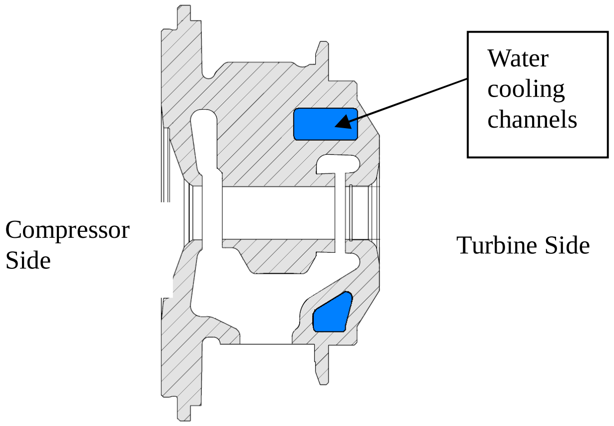

Figure 3: Sketch of water cooled center housing

In the studies of the effect of the heat transfer described here two different strategies have been used. In one series of tests on a specific stage (described here as compressor 0) a water cooling jacket (see Figure 3) was designed in the center housing to keep the oil temperature in the bearings within the acceptable limits and to reduce the temperature in the structure and protect it from the hot turbine stage. From the sketch it is clear that the heat transfer to the compressor can only be reduced with the selected water cooling system but not eliminated. In a second series of tests (on compressors A, B/C and D below) the compressor characteristics were measured both on a hot turbine test rig and a different cold turbine test rig (with turbine inlet and oil flow temperatures set equal to the compressor exit temperature to minimize the heat transfer). In the second series of tests there will clearly be some difference in performance due to the differences in the tests stands, but it is shown below that this effect is small compared to the heat transfer effect. The basic geometrical parameters of all the stages considered are typical of backward-swept turbocharger impellers with vaneless diffusers and no further details are given here.

Experimental results with and without water cooling of the center housing

The tests carried out with water cooling of the center bearing housing on compressor 0 are described in some detail in this section. This explains many aspects of the results which are valid for all turbochargers tested so that the discussion of the subsequent tests on stages A, B/C and D can be reduced to a discussion of the interesting differences in these cases.

An important result using compressor 0 is that the tests with and without water cooling do not produce any change in the compressor pressure rise characteristics, see figure 4. The solid lines indicate results with cooling and dotted lines without cooling for all figures. Each speed line of the compressor has the same value of the non-dimensional pressure rise coefficient with and without cooling, within the typical experimental error, and the operating range of the compressor is not substantially modified by the cooling. A similar result is obtained with the other compressors when tested with a hot and a cold turbine inlet temperature. This demonstrates that the heat transfer has no large thermodynamic or aerodynamic influence in turbochargers, as has already been demonstrated in earlier publications. Although there is no change in the pressure rise, there is a large change in the apparent compressor efficiency at low speeds, which is not a thermodynamic effect, but is simply wrong book-keeping in that the heat transfer is falsely attributed as an additional work input leading to an accounting error in the efficiency. This is shown in figure 5 where the efficiency ratios of the two cases are plotted in a non-dimensional form, as a normalized efficiency relative to a reference efficiency in the map.

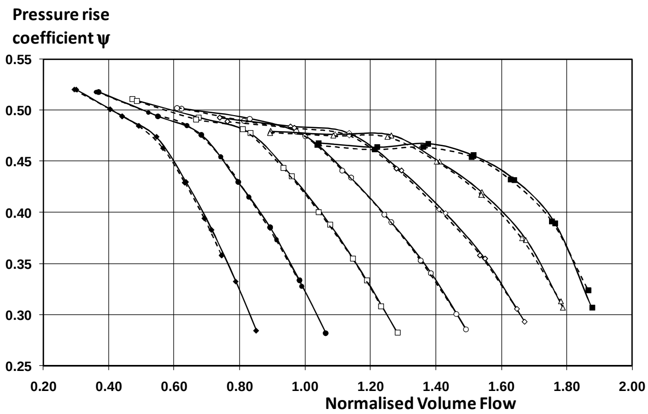

Figure 4: Pressure rise characteristics (pressure rise coefficient versus normalized volume flow) of compressor 0 with and without cooling over a range of speeds. Solid lines indicate with cooling and dashed lines without cooling.

The effect of the incorrect accounting of the compressor work leads to an apparent increase in the turbine efficiency at low speeds. This is shown in figure 6, where the turbine efficiency ratio is plotted, also made non-dimensional with a reference efficiency in the map. This result has been found to be similar for all other tests mentioned here and so this form of data presentation will not be repeated for all cases.

The tests carried out with and without water cooling on compressor 0 have also been analyzed using the approach given by Casey and Fesich [4], by comparing the apparent work input with the theoretical work input derived from the Euler equation. This relies on an approximate analysis to determine the outlet velocity triangle, given in detail by Casey and Schlegel [7]. In this procedure an approximate determination is made of the impeller outlet velocity triangle on the basis of two assumptions. Firstly the disc friction work input is estimated and subtracted from the actual measured work input, to obtain the Euler work input. Secondly an assumption is made for the pressure recovery in the vaneless diffuser and volute. This allows the static pressure at impeller outlet to be determined from the total pressure at stage outlet. In the calculations made here it is assumed that the diffuser pressure recovery is constant across the characteristic but this assumption could be replaced with a more appropriate variation of the pressure recovery with flow. In fact the results of this process for the determination of the work coefficient are relatively insensitive to the value of the pressure recovery coefficient chosen. The simple procedure makes use of the Euler equation, the continuity equation and the test performance to determine the outlet velocity triangle by iteration.

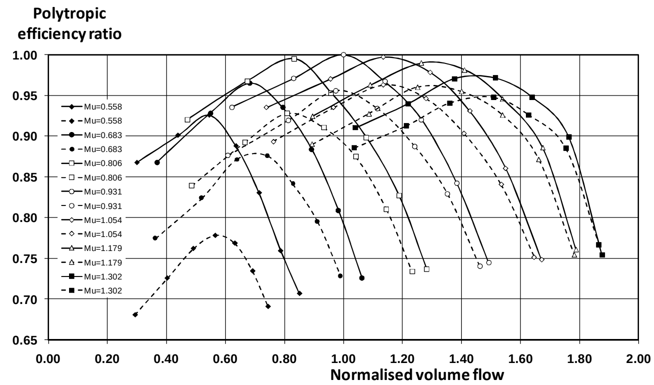

Figure 5: Compressor efficiency ratio characteristics of compressor 0 with and without cooling of the bearing housing. Solid lines indicate with cooling and dashed lines without cooling.

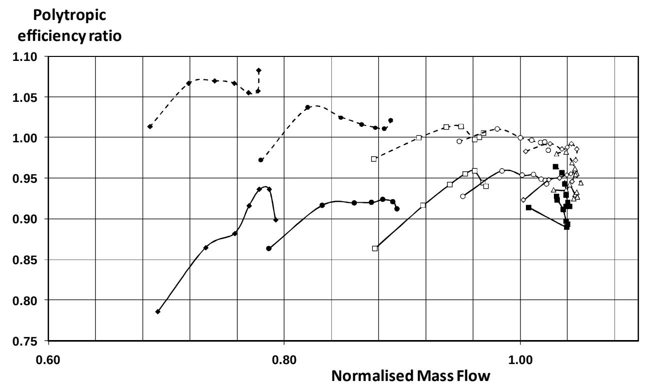

Figure 6: Turbine efficiency ratio characteristics of compressor 0 with and without cooling. Solid lines indicate with cooling and dashed lines without cooling.

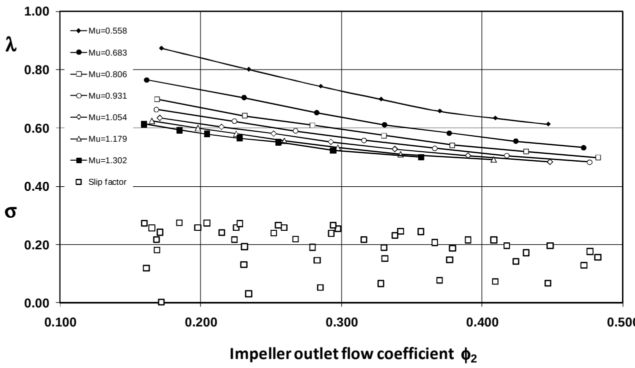

Figure 7: Work coefficient and slip factor versus impeller outlet flow coefficient at different speeds for compressor 0 without cooling and with no correction for heat transfer effects.

The result of such an analysis for compressor 0 without cooling is shown in figure 7. The apparent work input coefficient versus the impeller outlet flow coefficient is shown for a number of different speed lines. The Euler equation suggests that there should be a single unique and approximately linear variation of work with flow in such presentations, and measurements in large-scale industrial compressors without heat transfer confirm this, see Dalbert et al. [8]. As there is a difference between the individual speed lines in figure 7, this can only be interpreted as the effect of the heat input. The difference between these work input characteristics also increases at lower flow coefficients and lower tip-speed Mach number, as suggested by equation 20.

An additional justification that this is an effect of heat transfer can be determined from the apparent work input. The calculated values of the non-dimensional slip velocity, that is the slip velocity divided by the impeller tip-speed ($\sigma = c_s/u_2$), based on the apparent work input are also shown as open symbols lower down in figure 7. Note that the term c_s/u_2 is used to denote the slip in this paper, and is also called the slip factor, rather than the term $1- c_s/u_2$, which is sometimes used and also sometimes denoted as the slip factor. The data in the figure shows no sensible trends across the different speed lines. Figure 8 shows the same experimental data but in this case each experimental point has also been corrected for an estimate of the heat transfer using equation 20, using a constant value of the non-dimensional heat transfer coefficient of $k_c = 0.0031$ for the whole map. Note that the value of the heat transfer coefficient has been estimated on the basis that the work coefficient and the slip factor should be a unique line as a function of flow coefficient. The effect of correcting the work input curves is to produce a nearly single band of points representing the corrected work input for all speed-lines (black diamonds in figure 8) as a function of flow coefficient. No curve fit has been made to these points but it can be seen that these points at all speeds now correspond closely to the work input curves for the highest speed lines (which are closest to being adiabatic). The data also shows that the estimated slip factor now also produces a narrow band of points, which in fact is nearly constant across the whole range. The largest discrepancies occur on the low speed characteristics, where the actual measured temperature rise is only 15 to 20°C leading to higher experimental error in measuring the temperature rise. A similar diagram with corrected values is provided in figure 9, for compressor 0 when tested with water cooling. Of great interest is the fact that the agreement shown is obtained with a heat transfer coefficient of k_c = 0.001, indicating that more than two thirds of the heat transfer to the compressor is removed by the housing cooling system, but not all of it.

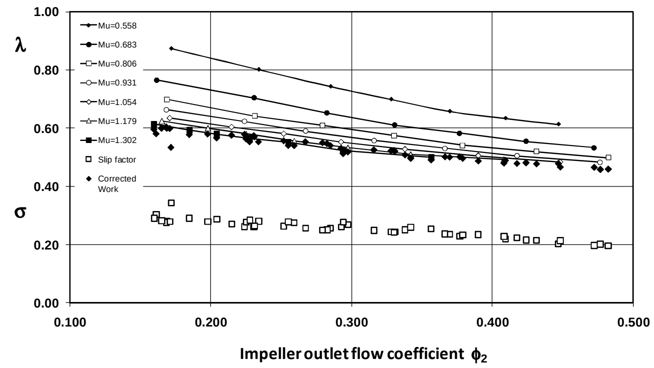

Figure 8: Work coefficient and slip factor versus impeller outlet flow coefficient at different speeds for compressor 0 without cooling but with correction for heat transfer effects.

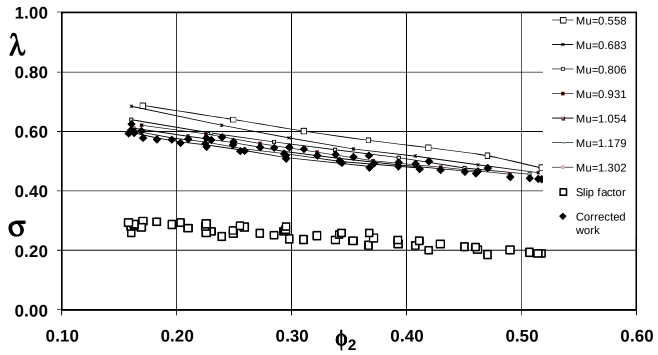

Figure 9: Work coefficient and slip factor versus impeller outlet flow coefficient at different speeds for compressor 0 with cooling and with correction for heat transfer effects

Of importance here is the fact that the analysis produces nearly unique curves for the important impeller parameters (the slip factor and the corrected work coefficient as a function of outlet flow coefficient) in figures 8 and 9, so that the procedure can be considered to have removed the heat transfer effect to produce an adiabatic reference base-line for the compressor. The baseline values for the work coefficient and the slip factor in figures 8 and 9 are nearly identical within the level of the experimental error and this justifies that both curves are now equivalent to the adiabatic case. This is further justified in figure 10 which shows the corrected efficiency ratios taking into account the heat transfer using equation 11. In both cases the correction for the effect of the heat transfer leads to an efficiency which is essentially the same (within 2% over most of the characteristic with a larger error near surge on the two lowest speed characteristics). This can be interpreted as the efficiency that would ensue under adiabatic conditions and it is interesting to note that the peak efficiency remains sensibly constant towards lower speeds, as would be expected if there is only a small effect of the Reynolds number. Note that the correction of the efficiency at low speeds is equivalent to more than a change in 20% points of efficiency.

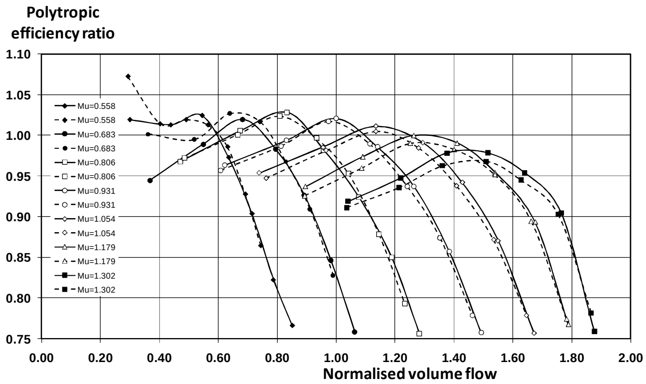

Figure 10: Efficiency ratio for compressor 0 with and without water cooling at different speeds corrected to adiabatic conditions. Solid lines indicate with cooling and dashed lines without cooling.

With this correction to the efficiency and work coefficient of the compressor it is now possible to correct the turbine efficiencies using equation 15 which also leads to an equivalent adiabatic turbine efficiency, as shown in figure 11. Note that the error of around 5% on the lower speed curves only corresponds to about 2% points as the efficiency is quite low here. The trend of the efficiency with speed is now similar for both the case with and without cooling and gives a very different impression to that derived without correction of the heat transfer effect.

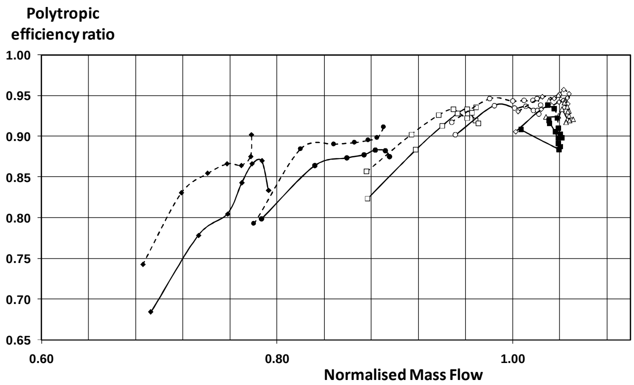

Figure 11: Corrected turbine efficiency ratios at different speeds. Solid lines indicate with cooling and dashed lines without cooling.

Based on these results it can be concluded that the proposed method can be used effectively by the turbocharger and engine manufacturers to correct their compressor and turbine map databases back to adiabatic conditions without changing significantly the accuracy. The method enables the correction to adiabatic maps to within 2% points of efficiency from the actual adiabatic levels. The accuracy of the corrected maps will of course still depend on the accuracy of the maps obtained on the hot gas stand.

Experimental results on hot and cold turbine rigs

Performance tests have been carried out with stages A, B/C and D in turbocharger test rigs with a hot and a cold turbine. An analysis equivalent to that presented for compressor 0 has been carried out and this confirms the findings in the water cooling tests that the pressure coefficient is not affected by the heat transfer, and that the heat transfer leads to a shift in the apparent efficiency from compressor to turbine and so is not repeated here. Some special features of the tests are however useful to examine. Firstly stage B/C has been tested to very high rotational speeds with a pressure ratio of nearly 4 on the surge line. In this case the derivation of the adiabatic curve for the work coefficient shown in figure 12 agrees closely with the measured work coefficient curves at very high speeds. This is simply an effect of the scaling with the cube of the tip-speed Mach number, as shown in equation 20, which indicates that the heat transfer effect becomes very small at high tip-speeds. In this case the measured high speed curve with heat transfer is very close to the adiabatic base-line performance. This compressor has a different design of bearing housing and the value of the heat transfer coefficient for this case is $k_c = 0.0024$. In this case the calculated slip factor is not constant with flow but still has a narrow band of points determining a clear trend with the flow coefficient.

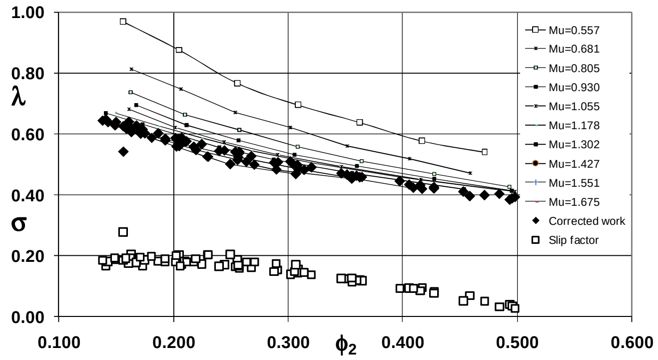

Figure 12: Work coefficient and slip factor versus impeller outlet flow coefficient at different speeds for compressor B/C tested up to high tip speeds.

An additional interesting point is found in comparing the results in the hot and cold rigs for compressor A. The cold rig cannot operate at such high speeds so that only low speed characteristics can be measured. Nevertheless the results shown in figures 13 and 14 seem to indicate that even in the cold gas-stand there is still a small heat transfer effect, as k_c is 0.0036 for the hot rig and 0.0006 for the cold rig. In this case every effort has been made to remove the heat transfer effects by adjusting the oil temperature and the turbine inlet temperature to be equal to the compressor outlet temperature to try to obtain adiabatic conditions. This indicates the great difficulty of comparing efficiencies in turbocharger compressors when tested as a turbocharger unit as the heat transfer effect from the turbine and bearing housing is not known.

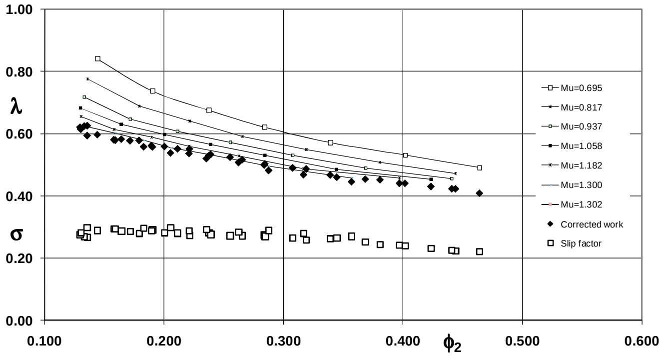

Figure 13: Work coefficient and slip factor versus impeller outlet flow coefficient at different speeds for compressor A tested on a hot gas-stand with correction for heat transfer effects

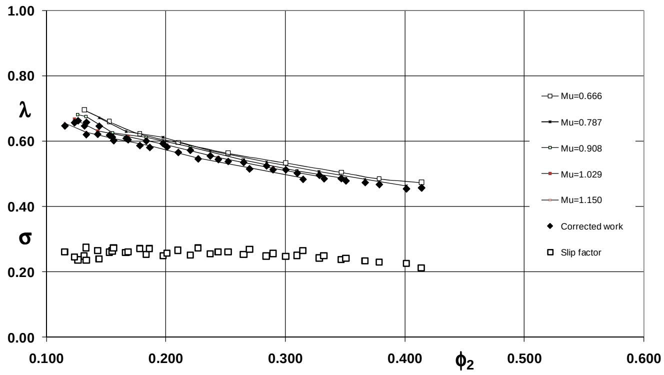

Figure 14: Work coefficient and slip factor versus impeller outlet flow coefficient for compressor A tested on a cold gas-stand at different speeds with correction for heat transfer effects.

The direct comparison of stages D and A is of interest as both of these make use of the same compressor, but in different turbochargers (with different turbocharger housings and different turbines). This comparison is given in figure 13 and 15. Configuration A has a substantially higher heat transfer (k_c is 0.0036) than configuration D (k_c is 0.0018) so that the apparent work coefficient characteristics at different speeds are not the same. When the work input is corrected back to the adiabatic conditions the compressor produces similar performance in both housings, as can be identified from the similar values of the work coefficient and slip factor as a function of flow coefficient.

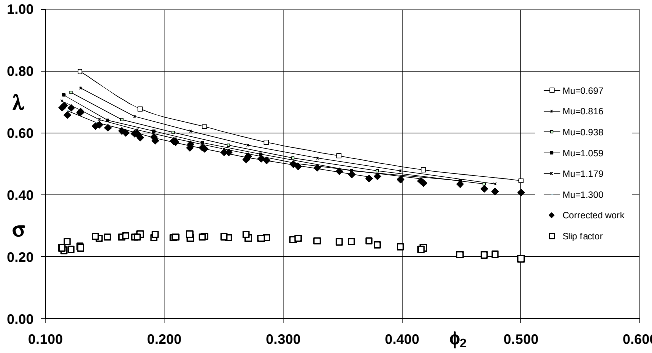

Figure 15: Work coefficient and slip factor versus impeller outlet flow coefficient for compressor D tested on a hot gas-stand at different speeds.

ON THE APPLICATION OF THE METHOD TO A BROADER CLASS OF MACHINES

Both the use of isentropic and polytropic analysis is acceptable for most turbochargers in order to obtain the adiabatic performance due to the fact that the thermodynamic (re-heat) effects and the aerodynamic effect (density, velocity, viscosity, velocity triangles changes) are small. The typical levels of heat entering a turbocharger compressor are not significant to affect the overall performance of the system. In other words, the heat input is small relative to the flow enthalpy at inlet or the heat parameter (Gong et al [2]) is small. In addition the largest effect is on the low-speed characteristics where the flow is effectively incompressible. However, for a broader class of machines including micro-scale compressors that would not be the case in general.

In order to compare the technology of two machines designed for different duties and operating under diabatic conditions both the effect of pressure ratio (reheat) and the effect of heat transfer (reheat and aerodynamic) must be removed. Thermodynamically, both reheat effects have the same consequence since for adiabatic flow reheat is related to the change in the (*Tds*) term in Gibbs equation due to aerodynamic loss, while with heat transfer the effect is related to the change in the (*Tds*) term due to heat addition. Both effects make the subsequent compression steps more difficult (consuming more work for the same rise in pressure) because of the diverging constant pressure lines on a T-s diagram. The aerodynamic effect due to heat transfer will be addressed separately at the end of the section.

Thus the use of the polytropic analysis is necessary first to eliminate the reheat effect due to flow irreversibilities (to compare compressors designed for different duties) and then to eliminate the effect of heat transfer leading to additional reheat between each incremental compression step (to compare compressors based on adiabatic performance).

The elegance of the polytropic efficiency idea is generally appreciated for adiabatic flows as it allows the technology comparison of compressors designed for different duties. This can be understood from the definition of polytropic efficiency, also called the “small-stage” efficiency, Cumpsty [9]. It describes the performance of the compressor during an infinitesimal increase of pressure dp. Thus the complete compression process is the integral of these differential compression steps dp. This idea can be extended to diabatic situations too as described by Casey and Fesich [4].

Unless one is prepared to track every differential step through the compression process some simplifying assumptions are necessary. Generally, the polytropic efficiency is assumed constant for each incremental step. This is more appropriate for some machines like multistage axial compressors with several stages of similar technology levels and less appropriate for others like single stage high speed radial machines. In a radial impeller many loss mechanisms take place near the inlet such as shocks, surface friction loss, re-circulation and mixing loss while most of the pressure rise occurs near the exit with the large change in radius. Therefore, the local polytropic efficiency may be significantly different from one incremental compression step to another. In any case, the beginning and the end states of the actual compression process are kept the same in the polytropic analysis and thus the constant polytropic efficiency can be viewed as a description of the average technology level during the process. It is a much more appropriate metric to compare compressors than using isentropic efficiency.

The situation is more complicated with heat transfer. An assumption to distribute the heat through the process is needed. Casey and Fesich [4] proposed that the incremental heat transfer is proportional to the useful pressure change work at each step, whereas Van den Braembusche [1] suggested that the heat transfer varies with the enthalpy increase in the compressor. Again, these assumptions may be more or less appropriate depending on the details of the machine and operating conditions. Once the total heat is measured or estimated for the actual process it can be introduced in the analysis and distributed between each compression step. Thus, both sources of reheat in the actual process can be accounted for and an average adiabatic polytropic efficiency can be obtained.

For cases with significant heat transfer the proposed correction procedure (Equation 11) needs a minor modification. In general, the achieved pressure ratio, efficiency, and mass flow will not be the same for a case with and without heat transfer. Therefore, only the diabatic compressor performance will be known from the measurements. To obtain the adiabatic efficiency equation 11 can be used with the diabatic pressure rise coefficient replacing the adiabatic one. The diabatic and adiabatic pressure rise coefficients were assumed the same in Casey and Fesich [4] and used interchangeably.

$$\eta_p = \frac{1}{\frac{1}{\eta_{pa}} - \frac{(\lambda_a - \lambda)}{\psi_{pa}}} \qquad 21$$

Equation 21 offers a consistent framework to obtain an average adiabatic polytropic efficiency for compressors operating under diabatic conditions. It distributes the aerodynamic irreversibilities and the heat addition to the flow through the compression process in order to obtain an overall performance metric consistent with the real initial and end states of the fluid. Once the adiabatic efficiency is obtained, the adiabatic pressure ratio can also be computed for the given work input.

One additional assumption is required in order to apply this approach to machines with appreciable heat transfer and reheat effects. The approach is valid only for cases with small aerodynamic effect on the performance of the compressor due to heat addition. Fortunately, most practical situations fall under this category. Clearly, adding heat to the gas will lead to gradual expansion of the fluid decreasing the density and increasing the velocity and viscosity. The velocity triangles may also change. Thus in general Reynolds and Mach numbers will be altered and aerodynamic loss may be affected from one compression step to another. The efficiency for each incremental step will therefore be slightly different and the method will provide some average polytropic efficiency for the process that may be different from the true adiabatic one.

Conceptually the compression process is modelled as the combination of a large number of small steps in the pressure rise. Therefore, each small step can be viewed as adiabatic with the heat transfer and loss added before or after the step. Thus for each step the values of Reynolds, Mach numbers, etc. can be assumed unchanged due to heat transfer. Therefore, the corrected performance of each small stage should be the same despite the changes from step to step of the inlet temperature and thus the polytropic efficiency will remain about the same as long as the Reynolds number does not change significantly between the first and the last steps. Most compressors in practice are characterized under this assumption – Reynolds number effects are taken to be small. Since flows even in micro scale machines remain turbulent the effect of the heat transfer on Reynolds number is expected to remain small (see Casey [10]) even for cases with significant heat addition and changes in density due to it.

Finally it should be noted that in some situations in compressor testing, without adequate insulation, the heat flow due to convection and radiation from the casing is such that the apparent efficiency of the compressor is improved, and not reduced. The equations given here can be applied to this problem if an estimate of the heat loss can be made.

DISCUSSION AND OUTLOOK

The simple corrections to the characteristics described in this work include the assumption that the non-dimensional heat transfer coefficient is constant across the whole performance map leading to a single value for the parameter k_c in equation 20. This is clearly an oversimplification as it assumes that the turbocharger operating point has no effect on the level of heat transfer. There are several possible reasons why this works reasonably well for this simple correction method.

Firstly, the work of Baines et al. [5] has estimated the heat flows into the different components using temperature measurements within the turbocharger. Their overall conclusion is that roughly 70% of the total heat lost from the turbine is lost to the external environment by radiation and convection. The fraction of the total turbine heat transfer to the oil is roughly 25%, and the remainder – only about 5% - is internal heat transfer to the compressor. Clearly the oil has a significant cooling effect such that the heat transfer to the compressor compared to that lost by the turbine is small and the heat transfer to the compressor is then dominated by the temperature of the oil. Any direct heat transfer from the turbine to the impeller must occur by conduction along the shaft or through the bearing housing. The shaft has a small diameter and is exposed to lubricating oil, so the shaft temperature is controlled by the oil temperature and this will naturally dominate this heat flow. Heat flow through the bearing housing is also affected mainly by the temperature of the oil rather than the turbine. The heat flow through the bearing housing will also tend to heat the flow downstream of the impeller in the diffuser and volute, so that it will have little to no effect on the actual work input. This can be implied from the fact that the pressure rise is not affected by the amount of cooling, see figure 4. The experiment with stage 0 described above identifies that in this case more than 2/3 of the heat transfer travels through the bearing housing and appears to be controlled by its temperature, which is close to that of the oil. Because of this the heat transfer to the compressor is mainly influenced by the temperature of the oil in the bearing housing, and not so directly by the temperature of the turbine inlet itself.

Secondly, the form of equation 20 indicates a high sensitivity to speed (through the exponent 3 on the Mach number in equation 20). The largest correction to the efficiency characteristic occurs when the tip speed Mach number is small, and at this condition the temperature rise in the compressor is low. At higher speeds the outlet temperature of the compressor may rise so that less heat flows from the oil to the compressor, but the effect of the heat input relative to the work input drops rapidly with speed so that this heat input has a smaller effect. The correction calculated on the assumption of constant heat transfer may then be slightly overestimated on the high speed characteristics, but as it is in any case very small at these speeds this is not important. Further detailed analysis of the test data for stage 0, not included here, indicates that the value of the heat transfer coefficient is actually not a constant for all operating points but tends to become slightly smaller towards the surge line. This is consistent with lower heat transfer to the compressor as the compressor outlet temperature rises.

Further work is ongoing to analyse these effects of a non-constant heat transfer rate on a wider range of compressors. It is hoped that with this improvement the error from the correction method can be reduced to better than 1% point of efficiency, which is then probably better than the accuracy of the measured performance characteristics themselves.

CONCLUSIONS

The test data and theoretical analysis in this paper show the significance of the heat transfer from the turbine to the compressor on the component performance maps on a small turbocharger. The main conclusions that can be drawn from this work are:

- The thermodynamic and aerodynamic effect of the heat transfer on the overall performance of the turbocharger is small as it leads to no changes in the pressure ratio characteristics of either component.
- The apparent effect on the efficiencies for both the turbine and compressor at low speeds is large, as the heat transfer is interpreted as work transfer in a conventional adiabatic analysis of the performance. This causes a shift in efficiency of up to 20% points from the compressor to the turbine on the low-speed characteristics.
- The fact that the apparent efficiencies on the low-speed characteristics are affected so much by the heat transfer makes comparison of the aerodynamic quality of compressors measured in different turbocharger configurations (housings) and different gas stands very difficult.
- A guideline for the conversion of turbocharger maps measured in situations with heat transfer on hot gas-stands to the equivalent adiabatic performance curves is provided. The approach requires no knowledge of the heat transfer and allows the heat transfer effect to be corrected so that the true efficiencies can be defined to within 2% efficiency points.
- The assumption of a constant heat flow at all operating points appears to be acceptable for all the compressors tested here. This allows a simple correction method with an accuracy of 2% points for the conversion of compressor and turbine performance maps to adiabatic conditions. Work continues on a method based on a variable heat transfer rate, which may be more accurate.
- Experiments on the test rigs in which the amount of heat transfer has been varied can be interpreted and explained in terms of the simple analysis method.
- Justification for the application of the method to a broader class of turbomachines including micro-scale machines is provided.

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