University of Bath

DYNAMIC IDENTIFICATION OF THERMODYNAMIC PARAMETERS

FOR TURBOCHARGER COMPRESSOR MODELS

R.D.Burke[1], P. Olmeda[2], J.R. Serrano[2]

1. Dept. of Mechanical Engineering, University of Bath, Bath, UK, Tel: +441225383481, email:

R.D.Burke@bath.ac.uk

2. CMT-Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n., 46022 València,

Spain

ABSTRACT

A novel experimental procedure is presented which allows simultaneous identification of heat and

work transfer parameters for turbocharger compressor models. The method introduces a thermally

transient condition and uses temperature measurements to extract the adiabatic efficiency and

internal convective heat transfer coefficient simultaneously, thus capturing the aerodynamic and

thermal performance. The procedure has been implemented both in simulation and experimentally

on a typical turbocharger gas stand facility.

Under ideal conditions, the new identification predicted adiabatic efficiency to within 1%point[1] and

heat transfer coefficient to within 1%. A sensitivity study subsequently showed that the method is

particularly sensitive to the assumptions of heat transfer distribution pre and post compression. If

20% of the internal area of the compressor housing is exposed to the low pressure intake gas, and

---

[1] This paper deals with both absolute and relative errors. To avoid confusion, absolute errors in

compressor efficiency will be expressed in %points whereas relative errors in other quantities will be

expressed in %.

this is not correctly assumed in the identification process, errors of 7-15%points were observed for compressor efficiency. This distribution in heat transfer also affected the accuracy of heat transfer coefficient which increased to 20%. Thermocouple sensors affect the transient temperature measurements and in order to maintain efficiency errors below 1%, probes with diameter of less than 1.5mm should be used.

Experimentally, the method was shown to reduce the adiabatic efficiency error at 90krpm and 110krpm compared to industry standard approach from 6% to 3%. However at low speeds, where temperature differences during the identification are small, the method showed much larger errors.

Keywords: Heat transfer, Turbocharger Mapping, Transient, Model, 1D modelling, Test methods

NOMENCLATURE

A	*Area*	m^2
c_p	*Specific heat capacity*	J/kgK
d	*diameter*	m
h	*Convective heat transfer coefficient*	W/m^2K
k	*Conductivity*	
$\dot{m}$	*mass flow*	kg/s
N	*rotational speed*	rpm
Nu	*Nusselt number*	
Pr	*Prandtl number*	
PR	*Pressure ratio*	
Re	*Reynolds number*	
T	*Temperature*	K
v	*velocity*	m/s
α_A	*Heat transfer area coefficient*	

η	*efficiency*	
μ	*dynamic viscosity*	
ρ	*density*	kg/m^3

Subscripts

1	*Pre compressor*
1'	*Pre Compression*
2'	*Post Compression*
2	*Post compressor*
3	*Pre Turbine*
3'	*Pre expansion*
4'	*Post expansion*
4	*Post turbine*
C	*Compressor*
s	*isentropic*
tip	*thermocouple tip*

1   INTRODUCTION

Turbochargers are becoming an important component for the downsizing of internal combustion engines which improve vehicle fuel efficiency by increasing the specific power and reducing weight and friction. However the turbocharger and internal combustion engine have very different flow characteristics and require careful matching in order to optimize their operation when working together in a system. This is achieved in a first instance through zero and one dimensional modeling of the turbocharged engine. The current modelling approach for turbochargers is to use performance maps derived from experiments on a steady flow gas stand facility that describe the flow and isentropic efficiency characteristics whilst assuming that the compressor and turbine operate under adiabatic conditions. This assumption is valid at high power conditions where the work transfer in the device is much greater than the heat transfers, but is inaccurate at lower power conditions. Consequently the simulation tools are compromised when simulating engine part load

conditions and their use is limited to high power operating scenarios. In order to increase the

usefulness of the simulation tools, new models have been proposed that include heat transfer

effects, however their parameterization can be time consuming making them impractical for

industrial applications.

In this paper, a new dynamic mapping approach is proposed for use with advanced compressor

models with heat transfer features. The method is investigated both in a virtual and real laboratory

and is capable of identifying the parameters aerodynamic and thermal model parameters through a

single experiment, suitable for use in a production environment.

2 Background

2.1 Heat Transfer in Turbochargers

A number of studies have been conducted over the past 10 years to quantify heat transfer in

turbochargers [1-6]. The majority of these compare the apparent behavior of the compressor (based

observed gas temperature rise) for different conditions of turbine inlet temperature (TIT) and

external heat transfer. Whilst the absolute values of heat transfer vary substantially depending on

operating and environmental conditions, when operating under conditions representative of those

seen on engine, around 70% of heat transfer from exhaust gases to the turbine housing are

transferred to ambient. Of the remaining 30% conducted along the bearing housing, a substantial

proportion is transferred to the lubricating oil and water cooling if present, meaning only a fraction

of the heat lost from exhaust gases is transferred to the compressor housing. Such is the

arrangement, heat transfer only becomes a significant factor at lower turbocharger speeds where

the mechanical work transfer is also small. As this corresponds to lower engine powers, this

compromises the turbocharged engine simulations for part load. As a result, simulations are

primarily used for full load matching of engine and turbomachinery whilst part load control is

addressed once experimental hardware is available. This has led to the motivation to model heat

transfer in turbochargers to allow better prediction of real driving scenarios which can then better advise the selection of hardware at the matching stage.

2.2 Heat Transfer models

A number of heat transfer models have been proposed by various authors that simplify the heat transfer into a 1-D problem [3, 5, 7-10]. The main differences between these models lie in their treatment of the central bearing housing which can be solved analytically or treated as one of more thermal nodes.

A common point for all models is the treatment of heat and work transfers in the compressor and turbine. It is assumed that heat and work transfer occur independently. These processes are described in the enthalpy-entropy diagrams in figure 1. The isentropic efficiency is used to calculate the temperature rise due to compression (processes 1' to 2' and 3' to 4') whilst heat transfer is modelled using convective heat transfer correlations adopted from flows in pipes [7] or measured specifically for this application [10] (processes (1 to 1', 2' to 2, 3 to 3' and 4' to 4).

2.3 Model Parameterization

Each of the 1D turbocharger heat transfer models are fundamentally based on compressor characteristic maps for efficiency and flow rate. Whilst the flow map is assumed to be independent of the thermal state of the machine, the efficiency map must be representative of true adiabatic conditions. Measurements from conventional gas stand setups are therefore not suitable for this model. Special adiabatic conditions can be created in laboratory setups where the turbine inlet and lubricant temperatures are controlled to the same as the compressor outlet temperature. In this case, heat transfer is suppressed by the removal of the driving temperature gradients across the device. Serrano et al [11] propose to undertake a full thermal characterization of the turbocharger housing such as to be able to use it as a heat flux probe. This then allows the measurement of heat flows during conventional mapping.

Each of these approaches requires a significant amount of experimental effort that is not practical for an industrial environment. This paper proposes a new dynamic mapping method that can capture both efficiency (work performance) and heat transfer characteristic in a single test.

3   Dynamic Identification Method

The proposed identification method makes use of similar assumptions to those for the heat transfer models found in the literature. The temperature rise in a turbocharger compressor is due to both work and heat transfers which are assumed to occur independently (represented on the enthalpy-entropy diagram in figure 1a). Heat is transferred to the intake gas at constant pressure between points 1 and 1' and described by equation 1.

$$T_{1'} = T_1 - \frac{h\alpha_A A_c (T_1 - T_c)}{\dot{m}_c c_{p,air}} \qquad\qquad 1$$

Work is then added adiabatically between points 1' and 2', raising the pressure and temperature of the gases. This process is described by an adiabatic but non-reversible process in equation 2, where the efficiency term[2] η_c captures the irreversibility due to friction and flow leakage.

$$T_{2'} = \frac{T_{2's} - T_{1'}}{\eta_c} \qquad\qquad 2$$

Finally further heat transfer occurs between points 2' and 2 (equation 3).

$$T_2 = T_{2'} - \frac{h(1 - \alpha_A)A_c (T_2 - T_c)}{\dot{m}_c c_{p,air}} \qquad\qquad 3$$

---

[2] This efficiency represents the apparent isentropic efficiency that would be observed if the compressor was operated with no heat transfer between the working fluid and the compressor housing (adiabatic conditions). Through this term, the mechanical load on the compressor rotor and the enthalpy rise of the fluid due to work addition can be correctly estimated.

In equations 1 and 3, α_A determines the ratio of internal area of the compressor housing exposed to intake air at the inlet temperature; conversely, *(1-α_A)* determines the proportion of internal area exposed to the intake gas post compression, at the higher temperature.

The goal of the proposed method is to identify the parameters η_c in equation 2 and *h* in equations 1 and 3. Because it is not possible to measure temperatures at the stations 1' and 2', this identification must be performed using only measurements of T_1, T_2 and a measure to the compressor housing temperature T_c. Under thermally stable conditions there exists an infinite number of combinations of efficiency and heat transfer coefficient that will yield these measured temperatures when solving equations 1 to 3 (each describing a different distributions of temperature rise due to compression and heat transfer). The novel method proposed in this paper aims to estimate adiabatic compressor efficiency and heat transfer by fitting equations 1 to 3 to measurements from a thermal transient. During a thermal transient, the thermal inertia of the compressor housing will cause T_c to change relatively slowly. This in turn causes variations in the temperature difference between the gas and housing in equations 1 and 3.

The method makes an additional assumption that for a given compressor operating condition (constant shaft speed and pressure ratio), the heat transfer coefficient *h* and efficiency η_c are constant. For isentropic efficiency this is a current assumption of the map based models as described by equation 4.

$$\eta_c = f(N_c, \dot{m}_c) \qquad\qquad 4$$

For the heat transfer coefficient, this assumption warrants further discussion. The convective heat transfer can be characterized by a correlation between Nusselt, Reynolds and Prandtl number (equation 5), which can be expanded into equation 6.

$$Nu = f(Re, Pr) \qquad\qquad 5$$

$$h = f\left(d, v, k_f, C_p, \rho, \mu_{bulk}, \mu_{skin}\right) \qquad\qquad 6$$

Considering each of the parameters of equation 6: *d* is the characteristic diameter which is purely a function of geometry and is constant for a given device. *v* is the velocity and is related to mass flow, density and the geometry according to equation 7. Hence, if mass flow and fluid density, ρ, remain constant, so will the gas velocity.

$$v = \frac{\dot{m}}{\rho A} = \frac{4\dot{m}}{\rho \pi d^2} \qquad\qquad 7$$

The fluid properties are all dependent on gas temperature. For a particular transient, the gas temperature at the outlet of the compressor, T2, will vary depending on the amount of work transfer. The lower limit is close to the intake temperature and an upper limit will be restricted by the maximum work transfer. For a turbocharger operating on a gas stand facility, these limits are approximately 20-200°C. In addition, the gas temperature will vary by up to 26°C over the thermally transient period required for the identification.

Density of air can be estimated using the perfect gas law (equation 8) and simple evaluation of this function in the region of 20-200°C and 1-2bar pressure shows that a 26°C change in temperature will result in a 4-6% change in density. Changes in velocity as a result in density will be proportional to this change as shown in equation 7.

$$\rho = \frac{P}{RT} \qquad\qquad 8$$

Using tabulated relationships of k_f, C_p and μ for air [12], for a change of 26°C in the relevant temperature range for this work (20-200°C), the respective changes in k_f, C_p and μ are 4-6%, 3-5% and <0.5%.

If the changes in fluid properties are combined into the non-dimensional numbers of equation 5, the resultant change will be 4-6% for Re number and <0.3% for Pr number. If it is assumed that the heat transfer coefficient in the turbocharger compressor can be estimated by analogy to heat transfer in

pipes, then the relationship in equation 5 could be of the form of the Seider-Tate relationship[3] [12] given in equation 9.

$$h = \frac{k_f}{d}\left(0.027Re^{0.8}Pr^{0.33}\left(\frac{\mu_{bulk}}{\mu_{skin}}\right)^{0.14}\right) \qquad 9$$

Combining each of the sensitivities for *Re*, *Pr*, k_f and μ shows that for a 26°C change in air temperature, the change in *h* will be less than 1%. Consequently this transient effect will be ignored and it will be assumed that for a given compressor, equation 6 can be simplified to equation 10, where $\dot{m}$ accounts for large changes in velocity and the pressure ratio accounts for large differences in compressor outlet temperature.

$$h = f(\dot{m}, PR) \qquad 10$$

It is therefore required to induce a state in the compressor whereby speed, pressure ratio and mass flow are constant but temperatures are varying. In practice, this was achieved by inducing a rapid change in compressor operating point. The temperature rise due to compression will change rapidly as the shaft accelerates and back pressure increases. The temperature change due to heat transfer will have a much slower response because of the housing inertia. This operation is illustrated in figure 2: the transient is induced by a rapid change in turbine inlet pressure whilst maintaining turbine inlet temperature and compressor backpressure valve position constant. This causes a rise in shaft speed, compressor mass flow and compressor pressure ratio. The additional work causes a rise in compressor outlet temperature T_2. A combination of this temperature rise and increased mass flow through the turbine increase the temperature of the compressor housing T_c.

Figure 2b shows a zoom on the compressor outlet and housing temperature for the time period over which the identification will be applied. This corresponds to the earliest point at which compressor mass flow, pressure ratio and shaft speed have stabilized thus allowing for the largest possible

---

[3] This relationship is commonly used in heat transfer modelling of turbocharger compressors [5,7,8].

temperature changes over this period. Typical variations in these three stabilized variables over the identification period where 0.8%, 2% and 1.5% respectively.

4   SIMULATION AND EXPERIMENTAL SETUP

The proposed identification methods was applied initially to a 1D turbocharger model where the sought parameters of adiabatic efficiency and heat transfer coefficents were known from the model calculations and provided a useful comparison for the new technique. In this way the physics based model was used as a virtual laboratory and different model aspects were varied to assess the sensitivity of the identification method. The method was subsequently implemented on a gas stand facility to demonstrate the technique experimentally, using results from a detailed thermal parameterization to assess the accuracy of the approach.

For both simulations and experimental results, identification was performed for four turbocharger speeds as described in table 1. In total 4 speed jumps were undertaken. For each of the speed lines, three to five compressor backpressure valve openings were considered providing a number of points along each constant speed line between surge and choke.

4.1 Turbocharger Model

The physics based turbocharger model is based on the same assumptions of separate heat and work transfer used by previous authors and described in previous sections. The model structure is illustrated in figure 3. The model inputs are temperatures and pressures at stations 1 and 3 (T_1, P_1, T_3 and P_3) and pressure at turbine outlet (P_4). The heat transfer model then works in tandem with the adiabatic compressor and turbine maps to calculate temperatures and mass flows through the device. The compressor outlet pressure P_2 is calculated using a backpressure valve model and simulating the filling and emptying of the volume between the compressor outlet and this back pressure valve. Further details of the heat transfer model can be found in [8].

In this work the model is primarily used as a virtual laboratory to compute the compressor housing (T_C) and compressor outlet gas temperature (T_2) required for the identification process, although the model can also provide an estimate of turbine outlet temperature T_4. Other intermediate variables are available through the model and most importantly for this work the modelled compressor internal convective heat transfer coefficient and adiabatic efficiency were recorded to validate the proposed identification method.

The thermal transients were induced as described in section 3 using a step change in turbine inlet pressure (P_3) whilst maintaining all other inputs constant. The rise in pressure causes an increase in turbine pressure ratio, resulting in increased mass flow and turbine power. This rapidly accelerates the turbocharger shaft and increases the compressor mass flow, which increases the compressor outlet pressure as it flows against the backpressure valve.

Initially the new method is compared to conventional steady state mapping and an analysis of the influence of time step size and total identification time was conducted. Secondly a number of simulation cases were conducted to vary the following aspects of the turbocharger models to assess the sensitivity of the method to various experimental uncertainties. Each case was simulated for all steps listed in table 1.

1. The effect of internal compressor area heat transfer distribution by varying α_A in equations 1 and 3 between 0 and 0.5.

2. The effect of thermocouple probe size on the compressor outlet gas temperature measurement. Thermocouple diameters of 0mm, 0.25mm, 0.5mm, 1.5mm and 3mm were investigated.

3. The effect of the magnitude of heat transfer from the turbine to the compressor. This was implemented by varying the effectiveness of oil cooling in the heat transfer model.

For point 2, a simplified model was implemented to simulate the thermocouple heating and cooling in the outlet duct of the compressor. The model assumed that the tip can be considered as a small

sphere situated in the flow. Only convective heat transfer was considered and conduction up the thermocouple sheath and radiation to the surrounding pipe was ignored. The convective heat transfer to the thermocouple was modelled using equation 11 [12].

$$Nu_{tip} = \frac{h_{tip}d_{tip}}{k_{gas}} =$$

$$2 + \left(0.4\sqrt{Re_{tip}} + 0.06Re_{tip}^{2/3}\right)Pr^{0.4}\left(\frac{\mu_{bulk}}{\mu_{skin}}\right)^{1/4}$$

11

The temperature of the thermocouple is then determined using equation 12.

$$\frac{dT_{tip}}{dt} = \frac{h_{tip}A_{tip}\left(T_{gas} - T_{tip}\right)}{m_{tip}c_{p,tip}}$$

12

The approximation in equation 12 is similar to a first order lag, however the time constant is dependent on the Reynolds number, and hence the flow rate in the duct where the measurement is being taken. Over the operating regions in question, the time constants can be as large as 10s for a 3mm thermocouple [8]. Temperature measurements of the housing were also performed using thermocouples, however as the heat transfer processes is through conduction and the housing temperature varies much slower than gas temperature, the dynamics of these measurements were ignored.

4.2 Experimental setup

The new identification procedure was also applied to measured data taken from a steady flow gas stand facility: a schematic layout of the facility is shown in figure 4. The turbine is fed with compressed air from an electrically driven screw compressor which can be electrically heated to 500ºC. The compressor takes air from ambient and flows against a back pressure valve such that both this backpressure valve and the screw compressor together control the operating point of the turbocharger. Pressures are measured at the four gas ports using both two independent piezo-resistive sensors and mass flows are measured both using vortex and hot wire sensors. At each of the gas ports, temperatures were measured using 4 k-type thermocouples, with diameter 1.5mm.

Lubricating oil and cooling water flows were provided by a two dedicated hydraulic circuits including electrical pumps and full temperature control.

Although a steady state rig by design, thermal transient were induced in a similar way to the model by rapidly increasing the delivery pressure of the screw compressor by increasing the speed of the motor driving that compressor. The various dynamics in the system mean that the change turbine inlet pressure is somewhat slower than in the model, however the change was acceptably fast to cause the desired thermal transients. The data recorded from one such transient has been presented in figure 2.

Before undertaking the dynamic identification described in this paper, both an adiabatic mapping and a full thermal characterization of the turbocharger used in this work was performed according to Serrano et al. [11]. This allows firstly the determining of the adiabatic efficiency of the compressor (as opposed to apparent efficiency) and secondly to estimate the convective heat transfer coefficient between air flow and internal housing surface. Both these quantities are necessary to provide validation to the output from the new dynamic approach.

5   RESULTS

This section is split into three parts: first, a general analysis of the method will be given in simulation; second, results from a sensitivity study are presented and finally some experimental results will be shown.

5.1 Analysis of dynamic identification method

Before assessing the performance of the new mapping approach, the results from a simulated steady state mapping are presented in figure 5. The apparent efficiencies, as would be measured from the temperature rise over the compressor, are shown for different turbine inlet conditions. These are compared to the sought efficiency that can be directly taken from the turbocharger model in this case, but could be measured experimentally by undertaking the mapping under adiabatic

conditions[4]. This summarizes the accuracy of conventional mapping and provides a basis to compare the new dynamic approach. Figure 5 shows that as the turbine inlet temperature is increased, the apparent efficiency of the compressor is reduced. This is because the temperature rise over the compressor due to heat transfer increases. The effect is barely visible at 150krpm because the heat transfer is negligible compared to the work transfer, but the effect becomes more apparent at lower shaft speeds and powers. At 50krpm, the efficiency is underestimated by around 5%points with the turbine inlet temperature of 500oC compared to adiabatic conditions. This behavior is well documented but provides a comparison for the dynamic approach.

Figure 6 shows the estimated compressor efficiency and heat transfer coefficient obtained by applying the dynamic identification method to the temperature profiles predicted by the physics based turbocharger model for each operating speed and pressure ratio. To obtain these results, the parameters of the physics based model were set such that:

- The turbine inlet temperature remained constant at 500°C

- The split of heat transfer in the compressor is 0% area before compression and 100% after compression (α_A=0)

It should be noted that whilst the first point is easy to control, the latter would not normally be known for a particular turbocharger and would be difficult to determine. In this section, it is assumed that these two assumptions are known to be correct and have therefore been accounted

---

[4] Adiabatic mapping is undertaken when the turbine inlet temperature, oil and water temperatures are controlled to match the compressor outlet temperature. These are referred to as adiabatic conditions as heat transfers are minimised by removing the temperature gradients between working fluids. The turbocharger would also be insulated to avoid heat losses to ambient.

for in the identification procedure. It will be shown in the next section the effect of an incorrect assumption at this stage.

The results in figure 6 show that the dynamic method is capable of capturing both the adiabatic efficiency and the heat transfer coefficient with a high level of accuracy. Compressor efficiency is identified to within 1%point and convective heat transfer with a nRMSE of 3.2%. These results were obtained by applying the identification process to data recorded over a period of 100s, starting as soon as the variation in turbocharger shaft speed, mass flow and pressure ratio were sufficiently low (This typically required a stabilization period of 10s after the turbine pressure rise). The technique is based on fitting the measured data to physical equations and therefore requires a level of transient behavior and sufficient data to capture the efficiency and heat transfer parameters. If the stabilization period becomes too long, the transient effects will have ended and the method could struggle to identify the parameters. Similarly, if the identification period is too long, it may become too heavily weighted towards steady state effect and equally cause problems in the fitting routine. The identification was repeated for all points with a range of stabilization and identification times to assess the robustness of the technique; these results are presented in figure 7.

The results for identification duration show that the accuracy of the efficiency prediction improves as the duration is increased from 5 to 100s after which there is no noticeable improvement or deterioration is observed. Heat transfer coefficient identification accuracy is stable provided the identification period is longer than 25s. The effect of stabilization time in figure 7 (b) shows that a longer wait before identification results in reduced accuracy of both efficiency and heat transfer coefficient as the resulting identification time comprises less transient components. It should be noted that in all cases the error in efficiency prediction was less than 1%point proving that in an ideal environment without random measurement errors, the method is robust.

5.2 Sensitivity Study

The results in this section refer to three important parameters that characterize the heat transfer and transient behavior of the compressor.

Firstly, the distribution of heat transfer before and after the compressor were investigated. This should be viewed as investigating the influence of assuming an incorrect value for α_A in the identification process. In order to achieve this, the turbocharger heat transfer model was used to simulate transient experiments with different values of α_A ranging from 0 to 0.5.

Figure 8 shows the results for simulations with α_A=0.2. If the identification process is performed by incorrectly assuming a value α_A=0, then the error in predicted efficiency can be large (up to 10%points at 150krpm). However, if the correct value of α_A is assumed in the fitting algorithm, then an almost perfect estimate of the efficiency is obtained. The effect on heat transfer coefficient estimation is shown in figure 8 (b). Here the assumption of an incorrect value of α_A has little effect on the accuracy of the parameter estimate and in fact gave slightly larger errors. This is a limitation of the identification routine which is apparent by comparing figure 8(b) with figure 6(b): the assumption a single heat transfer coefficient for both pre and post compression heat transfer is not-correct, and therefore an average coefficient is identified which will not be equal to the model heat transfer coefficient.

This analysis shows the importance of understanding the distribution of heat transfer pre- and post-compression and presents a limitation for the new method at present. A detailed investigation into the distribution of heat transfer is beyond the scope of this paper, however it is discussed in more detail in other publications [13, 14].

A second factor affecting the accuracy of the dynamic identification is the measurement of gas temperature at the compressor outlet. All previous results have been obtained using ideal temperatures obtained from the simulation model. However, in practice these temperatures will be

measured using temperature probes such as thermocouples that rely on heat transfer from the gas to the tip in order to give the temperature reading. Using the simplified thermocouple model, simulations were conducted to predict the effect of different diameter thermocouples and the identification results are presented in figure 9. Application of the identification method to each of the temperature profiles calculated for the different thermocouples give a variation in the predicted compressor efficiency and heat transfer coefficient. For thermocouples of diameter 0.5mm and 1.5mm, the estimated efficiency is within 1%point of the sought adiabatic efficiency. However, much larger errors are apparent for the 3mm thermocouple, with a failure to identify the efficiency at 50krpm with this size probe. Similar observations are made for the heat transfer coefficient as shown in figure 9 (b).

It is self-evident that the smallest possible thermocouples should be used for measuring the compressor outlet temperature T_2 to ensure best transient accuracy of the measurement, however this will always be a tradeoff between response time and sensor durability. This analysis has demonstrated that probes larger than 1.5mm must be avoided and subsequent experimental work was undertaken using k-type thermocouples of diameter 1.5mm (*supplier TC Direct*).

The magnitude of heat transfer along the bearing housing from the turbine to the compressor will affect the temperature of the compressor housing. The magnitude of this heat is determined by the effectiveness of the lubricating and cooling systems to evacuate the excess heat. A number of simulations were performed whilst varying this level of heat evacuation and in each case the compressor efficiency and heat transfer coefficient were identified using the dynamic approach. Figure 10 shows identified efficiency vs. model efficiency for five different levels of heat transfer. In each case the accuracy of the identified efficiency is similar showing that the single measurement of compressor housing temperature is sufficient to capture a range of different heat flows from the turbine.

5.3 Experimental Validation

Transient experiments were subsequently carried out experimentally on a turbocharger gas stand facility. The dynamic identification results from this are shown in figure 11. These are compared to the adiabatic efficiency, measured under adiabatic test conditions and the apparent efficiency measured with turbine inlet temperature of 500°C. These are presented individually for clarity and efficiency axes vary between graphs. It should be noted that although similar in size, the compressor in this section is different to that in the simulation section which explains the differences in efficiency between the two sets of results.

As in simulation, the apparent efficiency for steady state hot measurements is lower than that under adiabatic conditions except at 150krpm. For the dynamic identification, the results at 90krpm and 110krpm are more accurate than the hot steady state results: the error is less than 3%points whereas the non-adiabatic steady state tests give up to 6%points error. At the low speed case, the dynamic identification gives a poor over prediction of the efficiency of more than 10% points.

Before analyzing the heat transfer coefficient identification it should be noted that the thermal identification showed different correlations depending on the direction of heat transfer between the housing and the gas. This explains the two clusters of "measured" points in figure 11 (b). The dynamic identification gives reasonable estimates of the coefficients for the heating of the compressor housing; however it fails to capture the correlation for cooling of the housing.

Figure 12 plots the error in identified efficiency using the dynamic method against the magnitude of the temperature change at the outlet of the compressor (dT_2) during the identification period. This shows that the highest errors in efficiency occur during the smallest temperature rises. As this temperature change decreases, any measurement errors will become a larger proportion of the desired transient signal that is being captured. Therefore to promote higher accuracy from the approach larger temperature gradients over the identification period should be encouraged.

6   DISCUSSION

The results from the simulation aspect of this study showed that the methodology has the potential to allow a simultaneous identification of adiabatic efficiency and heat transfer coefficient for the turbocharger compressors. This offers the possibility for turbocharger manufacturers to undertake this identification in house for a large number of devices as it is considerably less time consuming than a full thermal identification. It may also offer the opportunity to reduce testing time compared to conventional mapping as it removes the need to wait for thermal stabilization before measurements are recorded.

The sensitivity study has highlighted that thermocouple inertia is an important factor in the accuracy of the method as it affects the apparent gas temperature measurements. If these measurements are to be used directly without any dynamic compensation in the form of a thermocouple model, then the results suggest that the use of thermocouples smaller than 1.5mm will provide an accuracy of around 1%point of compressor efficiency. This may be smaller than is used in robust industrial configurations, but should be sufficiently large to avoid excessive sensor failures.

The sensitivity study has also showed that by including a measurement of compressor housing temperature, the heat transfer from the turbine does not affect the accuracy of the method. However, the details of heat transfer within the compressor housing do have a significant influence. In this work the compressor housing has been described as a single mass that is exposed to gas both before and after the compression. This simple approach has been sufficient to show large effects on the accuracy of the identification procedure. Relatively little has been published on this topic and further investigations are required to better understand the behavior in this area of the device and subsequently to inform the method presented here.

Results from the experimental study show some promising behavior in the mid speed range but also the limitations of the method when applied in a non-ideal case, using data subject to measurement errors. As a result, the method is not yet capable of reproducing results taken under rigorous

adiabatic conditions but did offer an improvement over industry-standard non-adiabatic mapping approach in the region of 90-110krpm. Here the error of the dynamic method compared to the adiabatic case was less than 3% points compared to 6%points for the conventional approach. As the inaccuracies are primarily at low speed conditions, the sensitivity study would suggest this is a result more of thermocouple inertia rather than heat transfer distribution. However, dominant heat flows at low speeds may be different to those at high speeds and this again emphasizes the need for a detailed investigation into compressor heat flows.

The potential outcome of this new method could be the implementation of a new style of compressor mapping as illustrated by figure 13. The conventional approach measures points along constant speed lines as shown in figure 13 (a). The small changes in operating points between each case have the advantage of minimizing the thermal stabilization time between each case. If the method in this paper is used, then there is a requirement to maximize the thermal gradient between point, consequently the mapping will jump between speed lines as illustrated in figure 13 (b). Such an approach would have an added benefit of allowing for better detection of measurement system drift during the mapping process.

7   CONCLUSIONS

A novel method for compressor efficiency and heat transfer identification has been presented based on the analysis of the thermally transient behavior of the device. The method has been implemented both in simulation and on an experimental turbocharger gas stand and the following conclusions have been drawn:

1. Under idealized conditions, the identification method performed well giving an accuracy of less than 1%point for compressor adiabatic efficiency. The method was found to be robust to identification times without the need for thermal stabilization. This is particularly interesting at lower speeds where conventional mapping approach was seen to result in 5%point error at 50krpm.

2. The new method offers the ability to predict internal convective heat transfer coefficient with an accuracy of 3.2%. An estimation for this parameter is currently only achievable using and elaborate identification process.

3. A sensitivity study showed that assumptions regarding the heat transfer process in the compressor housing were crucially important to allow high accuracy of the method. If 20% of the internal area of the compressor is exposed to pre-compression gas, and this is not reflected in the identification method, errors of 7-15%points in predicted efficiency will be made. Further research is required into the thermal behavior of turbocharger compressor housings.

4. The sensitivity study also showed the importance of thermocouple response time: using a 3mm thermocouple could results in errors in compressor efficiency of over 10%points, especially at low speed conditions. It is recommended that probes no larger than 1.5mm be used to maintain an accuracy of less than 1%point.

5. When implemented experimentally, the method is subjected to measurement errors and random perturbations that are not present in the simulation case. Under these conditions the method performed well in the mid-speed region, providing efficiency estimates within 3%points of the adiabatic case compared to 6%points for the industry-standard mapping approach. This shows that the method offers improvements over current industry practice, but is not yet able to replace adiabatic mapping undertaken in research environments.

ACKNOWLEDGMENTS

REFERENCES

1.   Chesse, P., Chalet, D. and Tauzia, X., Impact of the heat transfer on the performance calculations of automotive turbocharger compressor (Influence des transferts thermiques sur le calcul des performances des compresseurs de suralimentation). Oil and Gas Science and Technology, 2011. 66(5): p. 791-800, DOI: 10.2516/ogst/2011129

2.   Cormerais, M., Hetet, J-F, Chesse, P. and Maiboom, A., Heat transfers characterisations in a turbocharger: Experiments and correlations, in 2006 Spring Technical Conference of the ASME Internal Combustion Engine Division, May 7-10 2006, Aachen, Germany: American Society of Mechanical Engineers.

3.   Shaaban, S., Experimental investigation and extended simulation of turbocharger non-adiabatic performance, PhD Thesis, Fachbereich Maschinenbau, Universität Hannover, Germany, 2006.

4.   Serrano, J.R., Guardiola, C., Dolz, V., Tiseira, A. and Cervello, C., Experimental Study of the Turbine Inlet Gas Temperature Influence on Turbocharger Performance, SAE Paper Number 2007-01-1559. 2007, DOI: 10.4271/2007-01-1559

5.   Baines, N., Wygant, K.D., and Dris, A., The Analysis of Heat Transfer in Automotive Turbochargers, Journal of Engineering for Gas Turbines and Power - Transactions of the ASME, 2010. 132(4), DOI: 10.1115/1.3204586

6.   Serrano, J.R., Olmeda, P., Arnau, F., Rayes-Belmonte, M. and Lefrbvre, A., Importance of Heat Transfer Phenomena in Small Turbochargers for Passenger Car Applications. SAE Int. J. Engines, 2013. 6(2): p. 716-728, SAE Paper Number 2013-01-0576, DOI: 10.4271/2013-01-0576

7.   Romagnoli, A. and Martinez-Botas, R., Heat transfer analysis in a turbocharger turbine: An experimental and computational evaluation, Applied Thermal Engineering, 2012. 38(3): p. 58-77, DOI: 10.1016/j.applthermaleng. 2011

.12.022

8.   Burke, R.D., Analysis and Modelling of the Dynamic Behavior of Automotive Turbochargers, in 2013 Fall Technical Conference of the ASME Internal Combustion Engine Division, Dearborn, USA, ASME Paper Number ICEF2013-19120.

9.   Cormerais, M., Chesse, P. and Hetet, J-F., Turbocharger heat transfer modeling under steady and transient conditions, International Journal of Thermodynamics, 2009. 12(4): p. 193-202.

10.   Olmeda, P., Dolz, V., Arnau, F. and Reyes-Belmonte, M., Determination of heat flows inside turbochargers by means of a one dimensional lumped model. Mathematical and Computer Modelling, 2013. 57(7-8): p. 1847-1852, DOI: 10.1016/j.mcm.2011.11.078

11.   Serrano, J.R., Olmeda, P., Paez, A. and Vidal, F., An experimental procedure to determine heat transfer properties of turbochargers. Measurement Science & Technology, 2010. 21(3), DOI: 10.1088/0957-0233/21/3/035109

12.   Incropera, F.P. and De Witt, D.P., Introduction to Heat transfer. 2nd ed. 1985, New York: John Wiley and sons Inc, ISBN: 0-471-51728-3

13.   Bohn, D., Heuer, T. and Kusterer, K., *Conjugate flow and heat transfer investigation of a turbo charger,* Journal of Engineering for Gas Turbines and Power, 2005. 127(3)(3): p. 663-669, DOI: 10.1115/1.1839919

14.   Burke, R.D., Copeland, C.D. and Duda, T., *Investigation into the Assumptions for Lumped Capacitance Modelling of Turbocharger Heat Transfer*, in 6th International Conference on Simulation and Testing, Berlin, May 15-16 2014, Expert-Verlag: Berlin, Germany.

APPENDIX A. Model Parameters for virtual laboratory

NAME	DESCRIPTION	VALUE	UNIT
Compressor back pressure valve			
$V_{c,out}$	Volume between compressor outlet and compressor back pressure valve	0.78×10^{-3}	m^3
$D_{bp\ valve}$	Orifice area	$0\text{-}4 \times 10^{-3}$	m^2

Cd_{bp_valve}	Discharge Coefficient	*0.6*	
Turbocharger base model			
I_{turbo}	Shaft inertia	*16×10⁻⁶*	kgm²
$\dot{m}_{turb}$	Corrected turbine flow map (from tabulated performance data, *f(N, PR_{turb})*)	*0-0.12*	kg/s
η_{turb}	Turbine efficiency map (tabulated data, *f(ṁ_{turb}, PR_{turb})*)	*0-75*	%
$\dot{m}_{comp}$	Corrected compressor flow map (tabulated data, *f(N, PR_{comp})*)	*0-0.18*	kg/s
η_{comp}	Normalized compressor efficiency map (tabulated data, *f(ṁ_{comp}, PR_{comp})*)	*0-74*	%
VGT	VGT Position (fixed in all simulations)	*30*	%open
Turbocharger Heat transfer model			
$A_{c,int}$	Total internal surface area of compressor	*29×10⁻³*	m²
$A_{c,ext}$	Total external surface area of compressor housing	*79×10⁻³*	m²
$A_{t,int}$	Total internal surface area of turbine	*24×10⁻³*	m²
$A_{t,ext}$	Total external surface area of turbine housing	*80×10⁻³*	m²
$d_{c,inlet}$	Compressor intake pipe diameter	*40×10⁻³*	m
$d_{c,outlet}$	Compressor outlet pipe diameter	*35×10⁻³*	m
$d_{t,inlet}$	Turbine intake pipe diameter	*40×10⁻³*	m
$d_{t,outlet}$	Turbine outlet pipe diameter	*45×10⁻³*	m
Nu_c	Internal Convective correlation compressor	$Nu{=}0.004Re^{1.13}Pr^{1/3}(\mu/\mu_{skin})^{0.14}$	
Nu_t	Internal Convective correlation Turbine	$Nu{=}0.038Re^{0.662}Pr^{1/3}(\mu/\mu_{skin})^{0.14}$	
k, μ, c_p	Air Data (tabulated data from [12])		
L_{bh}	Bearing housing length	*45×10⁻³*	m
A_{bh}	Bearing housing section area	*7×10⁻⁴*	m²
h_{ext}	External free convection correlation	$h_{ext} = 1.25(T_{wall} - T_{amb})^{1/3}$	W/m²K
ε_c	External emissivity of compressor housing	0.1	
ε_t	External emissivity of turbine housing	0.6	
T_{amb}	Ambient temperature	25	ºC
$P_{c,inlet}$	Ambient Pressure at compressor inlet	1	bar
$P_{t,outlet}$	Ambient pressure at turbine discharge	1.10	bar
m_c	Mass of compressor housing	0.93	kg
m_t	Mass of turbine housing	3.15	
Cp_c	Thermal capacitance of compressor housing (*f(T_{wall})*)	930-990	J/kg
Cp_t	Thermal capacitance of turbine housing (*f(T_{wall})*)	500-700	J/kg
k_{bh}	Thermal conductivity of central bearing housing (*f(T_{wall})*)	45-50	W/m

Test #	Target Speed (rpm)	Step
1	50000	110000 to 50000
2	110000	50000 to 110000
3	90000	150000 to 90000
4	150000	90000 to 150000

Table 1: Target speeds and magnitudes of speed jumps to induce thermal transients
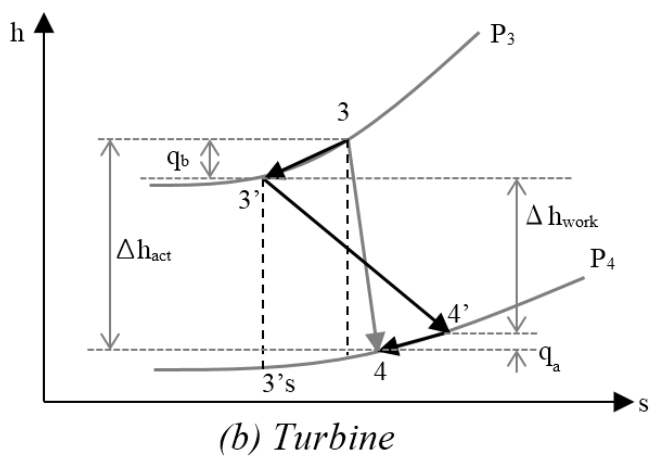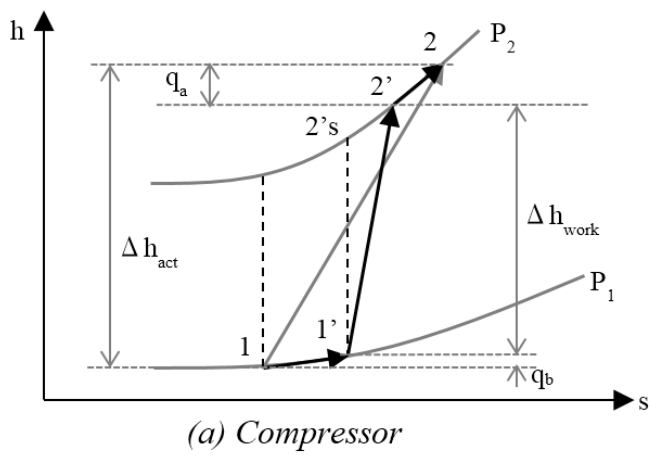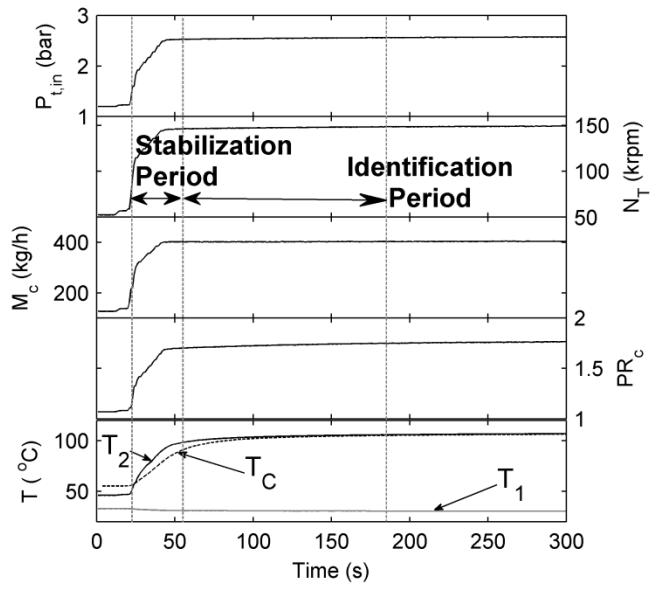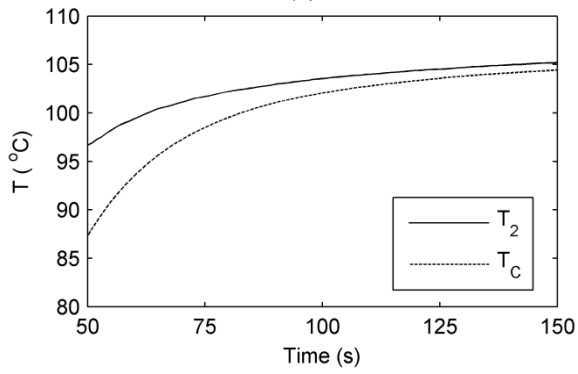
Figure 1: Idealized compression and heat transfer processes in the turbine and compressor

(a)

(b)

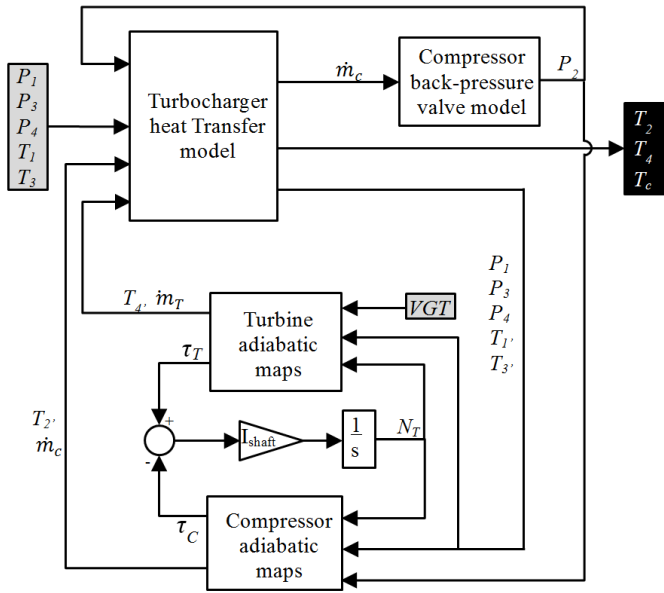**Figure 2: Temperature evolutions over thermal transient following a rapid change in compressor operating point**

Figure 3: Overview of turbocharger model
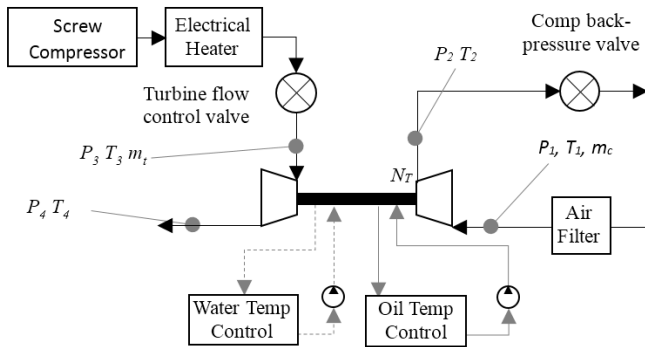
Figure 4: Gas stand configuration
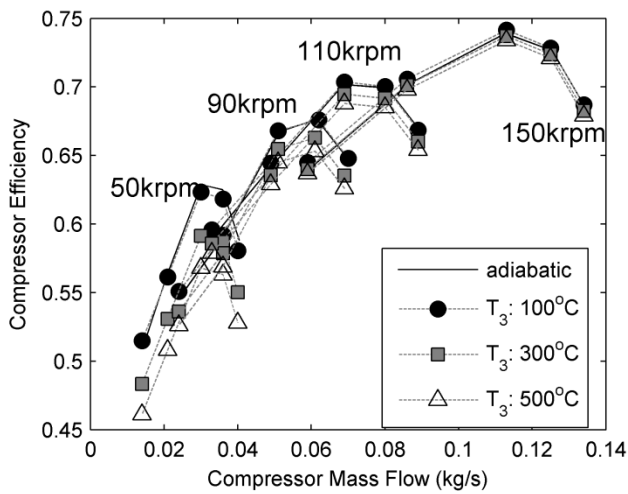
Figure 5: Steady state mapping simulations
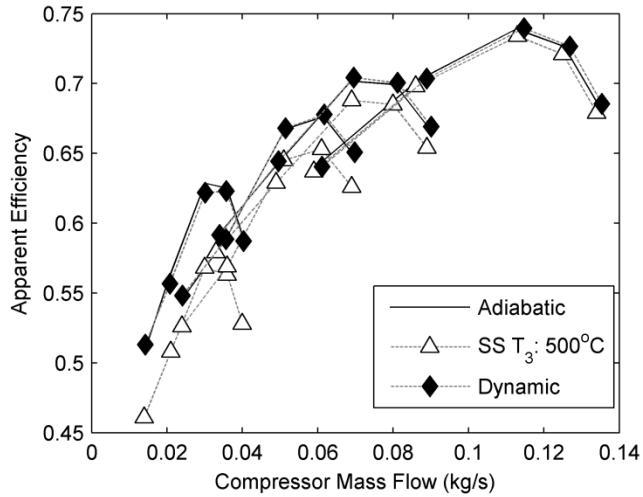
(a)
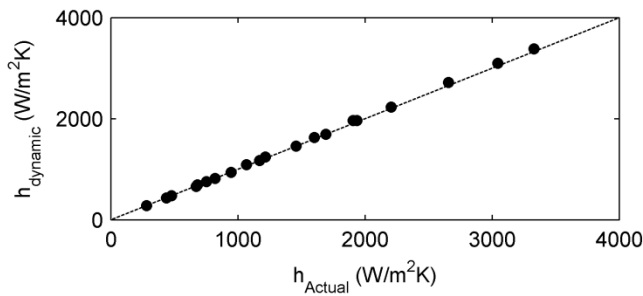
(b)

Figure 6: (a) Efficiency and (b) heat transfer identification using dynamic method vs. steady state approach. All results using $\alpha_A=0$.
(SS: Steady State Mapping)
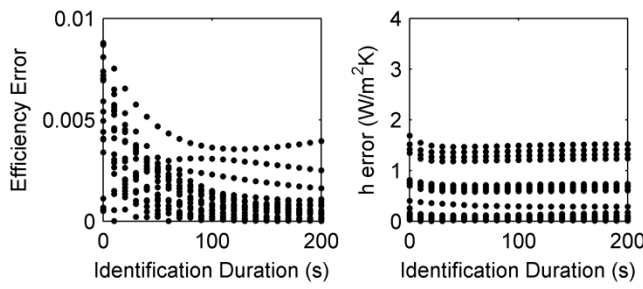
(a)
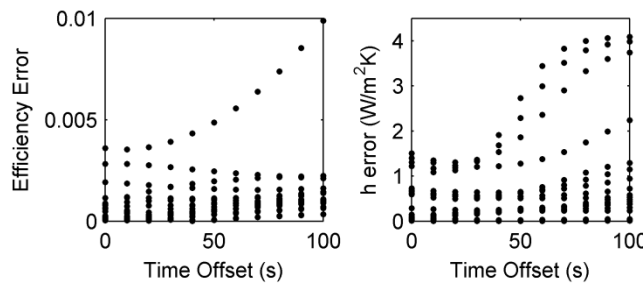
(b)

Figure 7: Accuracy of the identification method with respect to (a) identification period duration and (b) identification period offset with respect to thermal transient initialization.
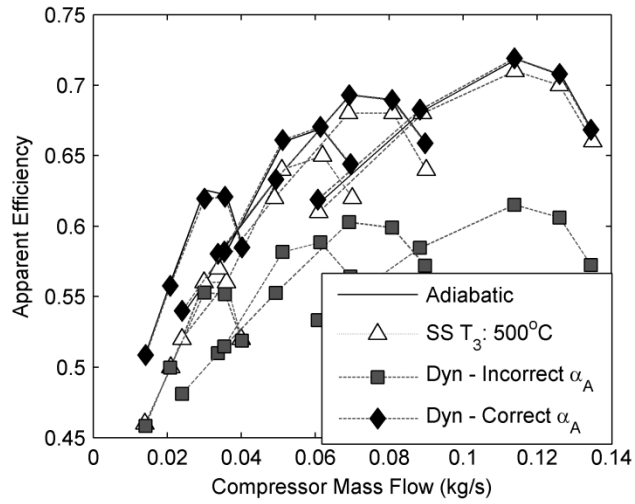
(a)
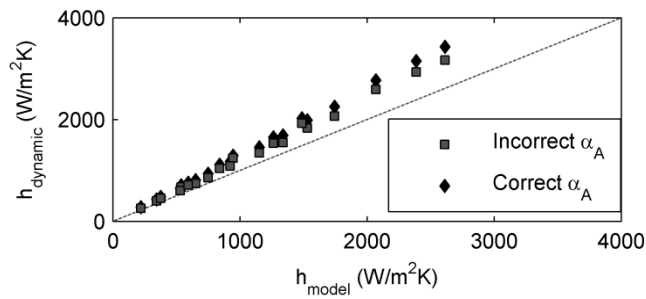
(b)

Figure 8: Sensitivity of dynamic identification method to correct selection of compressor housing heat transfer distribution
(SS: Steady State Mapping; Dyn: Dynamic Identification)
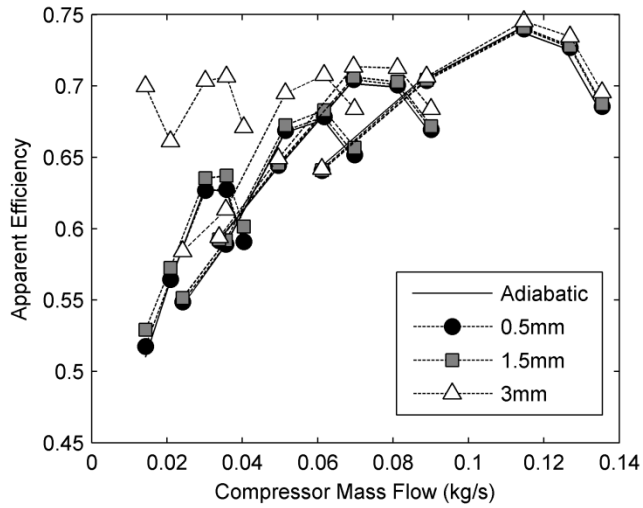
(a)
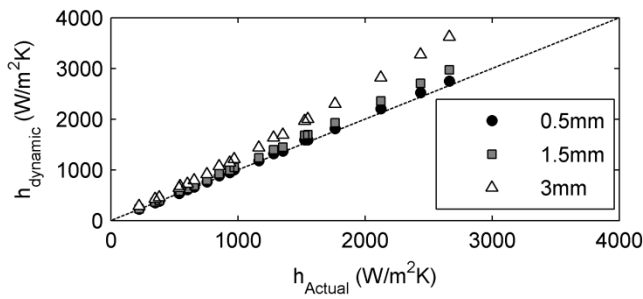
(b)

Figure 9: Effect of thermocouple inertia on estimated compressor efficiency
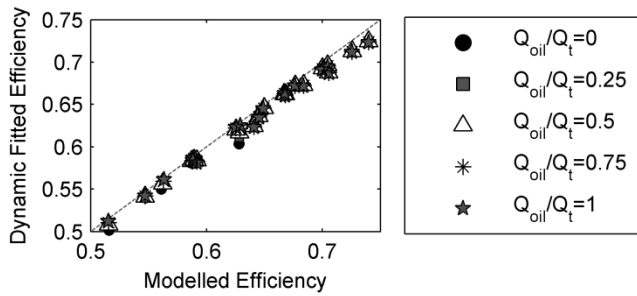
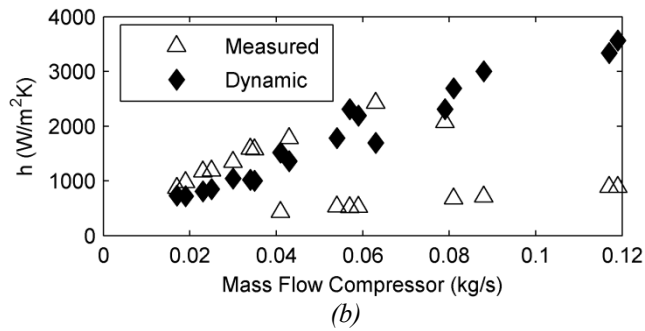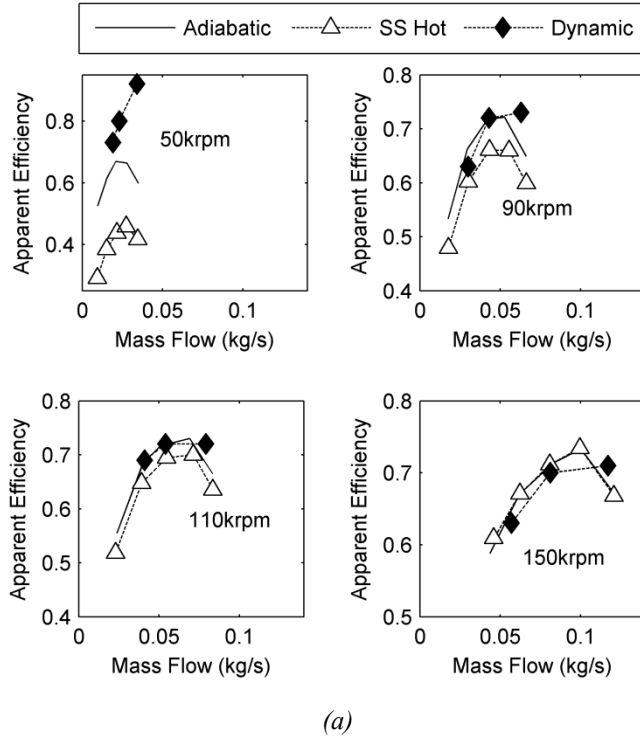**Figure 10: Influence of heat transfer from the turbine to the compressor on identification accuracy**

Figure 11: Experimental Identification results for (a) efficiency and (b) heat transfer coefficient (SS: Steady State Mapping)
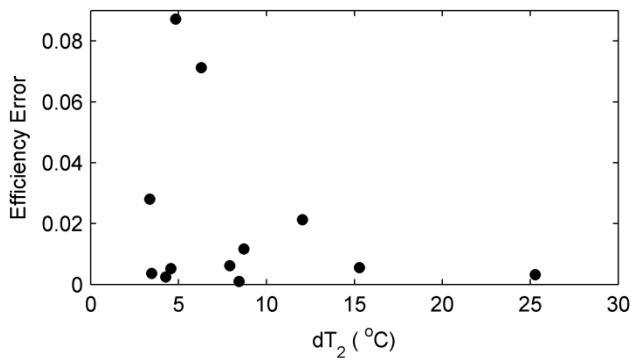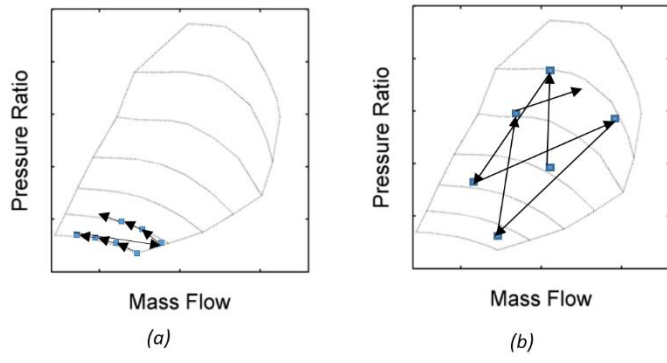

Figure 12: Efficiency error vs. step size showing that higher accuracy is obtained with larger thermal transients

Figure 13: (a) conventional and (b) proposed mapping strategies