EXPERIMENTAL MEASUREMENT AND CHARACTERISATION OF THE EFFECT OF OFFTAKE FLOW ON THE TIME CONSTANTS OF A COMPRESSOR CASING

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ABSTRACT
This paper describes measurements of the thermal response times of a compressor casing from a large civil engine as a function of offtake flows. The experiments used a full-size engine compressor casing and full-sized offtake mass flow rates. The experiments were performed using the Oxford Transient Heat Transfer facility (THTF).

The temperature response rates of the casing to a fast ramp change in input air temperature were measured at several metal and air locations. It is shown that the transient temperature behaviour at each location can be accurately characterised by a single time constant. The variation of time constant with offtake mass flow is also shown.

Such characterised experimental data can provide valuable validated information on casing thermal response much earlier than the approach of waiting for an engine development programme thermocouple test, and thus bring forward the achievement of optimum transient casing behaviour.

KEYWORDS
COMPRESSOR CASING, THERMAL RESPONSE, THERMAL GROWTH, RIG TEST

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Area for convective heat transfer</td>
</tr>
<tr>
<td>a</td>
<td>Proportional immediate response of air temperature</td>
</tr>
<tr>
<td>b</td>
<td>Intrinsic time constant of solid</td>
</tr>
<tr>
<td>c_p</td>
<td>Specific heat capacity of air</td>
</tr>
<tr>
<td>C</td>
<td>Specific heat capacity of solid</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
</tr>
<tr>
<td>M</td>
<td>Mass of solid</td>
</tr>
<tr>
<td>Q</td>
<td>Heat flow</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate</td>
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<tr>
<td>t</td>
<td>Time</td>
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<td>T_A</td>
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<tr>
<td>T_M</td>
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<td>ρ</td>
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<td>HTC</td>
<td>Heat Transfer Coefficient</td>
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<td>THTF</td>
<td>Transient Heat Transfer Facility</td>
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INTRODUCTION

The compressor casing of a large civil aero engine has several functions, including forming part of the engine structure and serving as a mount for stator vanes. The casing also forms the outer wall of the main gas path annulus, and the gap between the casing and the rotor blades accounts for the rotor tip clearance. The thermal growth of the compressor casing has a significant effect on the tip clearance, particularly during transient thermal operations, it is therefore desirable to understand the thermal growth rates of the different parts of the casing.

The thermal response rates of the different parts of the compressor casing are influenced by the heat capacity of the different parts and the convective heat transfer coefficient (HTC) on the surface of the parts. In turn the HTCs are set by the flowfield around and within the casing. This flowfield comes from compressor bleed air, extracted through annular slots in the compressor casing before exiting the casing through several pipes.

The complex flowfield in the cavities around the casing is difficult to simulate, making predictions of HTC and hence growth rate of the casing difficult. It is also difficult to predict the HTC in this region using simple correlations. The thermal response rates of the compressor can be measured on a real engine test, but these tests occur late in the engine development stage, making this method unsuitable for optimisation of the compressor growth rates before completion of engine development.

The aim of this paper is to investigate the option of rig testing the heat transfer in the compressor casing offtake cavities, at engine size and levels of offtake mass flow, with a view to being able to inform decisions during engine design phase and thereby avoid the potential for costly re-work following first engine thermocouple test.

EXPERIMENTAL FACILITY

The Oxford Transient Heat Transfer facility (THTF) was used in this investigation. The facility comprises a 2 m diameter spherical pressure vessel, connected to a heated high-pressure air supply. The pressure vessel is large enough to accommodate full-sized components from a large civil aero-engine. The development and initial application of the facility are described by Van Paridon et al. (2015) and Dann et al. (2017).

Compressed air for the rig is sourced from a tank, initially at 28 bara, which is regulated down to the experimental test pressure (typically 12 bara). The setup can maintain a flow of 2.4 kg/s for around 8 minutes. The air is heated by an Osram 600 kW electric heater up to a set temperature (typically 200 °C) before entering the facility and the experimental test section.

The test section, shown in Figure 1, was constructed from a HP compressor inner casing from a current large aero-engine. The section consists of the rearmost four stages of the compressor. The stator vanes were replaced with machined segmented rings of the same cross-section as the stator outer platforms, the outer annulus line of an engine was therefore replicated. An annular slot was created by adding a central cylindrical body inside of the casing, giving a 1.8 mm radial gap and a nominal annulus outer radius of 289.5 mm. The casing featured an offtake ramp and casing slots, this allowed for extraction of bleed air through the casing.

The casing assembly features front and rear support cones. These serve a structural assembly role, and also create the cavity which gathers the air that flows through the offtake ramp and slots. It is the heat transfer in this cavity, and how it is affected by the level of offtake mass flow, which is the subject of this investigation.

When testing, the air enters the offtake cavity via the offtake ramp and the adjacent slots in the casing. The flow exits the cavity through a set of eight equi-spaced 32 mm diameter
transfer holes in the cavity front support cone, after which the air flows towards the four smaller diameter offtake pipes set in the main casing.

The design of compressor bleed offtake slots has been and continues to be the subject of patent and research activity. Patents can be found in Klasing et al (2018) and King et al (2015). The offtake slots used in this experiment are broadly of the type in King et al. Research on offtakes has been conducted by Leishman et al. (2003, 2007a, 2007b), and by Grimshaw et al. (2015a, 2015b). The focus of this research is the minimisation of pressure losses in the offtake, or its effects on the aerodynamics in the annulus. There is very little research into the effects of the offtake flow on the heat transfer in the offtake cavity.

The flow through the offtake exits the test section via four pipes, spaced around the test section. Valves can switch the flow in these four pipes independently, allowing for the offtake mass flow to be varied. The flow can also be directed through the annulus to the rear of the offtake, rather than though the offtake itself, however for this investigation this rearwards flow path was closed off.

INSTRUMENTATION

Thermocouples were located in the test section as shown in Figure 2. Metal temperature thermocouples have been labelled as locations 1 to 12. Four thermocouples were placed at each location, circumferentially 90° apart. Similarly air temperatures were also measured at several locations, labelled as A1 to A4. Not shown in Figure 2 is location A1 at the heater exit. The thermocouples were type K and mineral insulated, with the thermocouple data being recorded at a rate of 10 Hz. It was found that the air thermocouple time constant at location A3 was 6s; this delay was corrected out before the characterisation and calculation of the time constants at the metal locations.

TEST CONDITIONS

The test facility was originally designed to test full-sized engine hardware at engine conditions, therefore requiring minimal scaling. The nominal datum case test conditions and a
Figure 2: Air and metal thermocouple locations in test section

Table 1: Scaling of experiment from real engine conditions, indirect parameters in italics

<table>
<thead>
<tr>
<th></th>
<th>Scaling from engine</th>
<th>Datum case test conditions</th>
</tr>
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<tbody>
<tr>
<td>Geometric dimensions</td>
<td>1.0</td>
<td>289.5 mm casing radius</td>
</tr>
<tr>
<td>Offtake mass flow</td>
<td>1.0</td>
<td>2.4 kg/s</td>
</tr>
<tr>
<td>Pressure</td>
<td>1.2</td>
<td>12 bara</td>
</tr>
<tr>
<td>Temperature</td>
<td>0.71</td>
<td>473 K (200 °C)</td>
</tr>
<tr>
<td>Air density</td>
<td>1.68</td>
<td>8.84 kg/m³</td>
</tr>
<tr>
<td>Annulus axial Mach Number</td>
<td>0.48</td>
<td>0.19</td>
</tr>
<tr>
<td>Reynolds Number</td>
<td>1.25</td>
<td>$8.1 \times 10^6$</td>
</tr>
<tr>
<td>Nusselt Number</td>
<td>1.20</td>
<td>-</td>
</tr>
</tbody>
</table>

DATUM TEST RESULTS

The test sequence was started by setting up the required flow, then switching on the heater to give a fast ramp change in air temperature. The metal temperature responses to this step change were then recorded. Only one test was run in a day, to give time for the rig to fully cool, ensuring the metal had a uniform starting temperature.

The response of the air thermocouples at the different locations are shown in Figure 3. It can be seen that the response rate of the air slows down at the downstream locations, this is due to the air losing heat to the metal as it passes through the rig.

The change in air temperature drives a response in the metal temperatures, Figure 4 shows the temperature at a fast and slow responding metal locations (locations 1 and 5 respectively). The offtake ramp air temperature, $T_{A3}$, has been added as reference, as this is the input air temperature which drives the metal temperature responses.
Figure 3: Response of air temperature thermocouples

Figure 4: Response of metal temperature thermocouples to driving air temperature

Figure 5: Schematic showing characterisation of temperatures
CHARACTERISATION EQUATIONS

Consider the scenario shown in Figure 5. A mass flow of air $\dot{m}$ with an inlet total temperature $T_{in}$, passes heat into a solid block of mass $M$ and specific heat capacity $C$. The conductivity of the solid is sufficiently large that its temperature $T_M$, can be considered uniform at any point in time. The block receives heat input $\dot{Q}$ over a surface area $A$, with an effective average heat transfer coefficient $h$. This heat flow reduces the temperature of the air stream from its inlet value of $T_{in}$ to an exit value $T_A$.

Firstly, it is useful to derive a general equation for how $T_A$ relates to the inlet air temperature and block temperature. Equating the convective heat flow from the air into the metal with the enthalpy drop in the air flow gives:

$$\dot{Q} = \dot{m}c_p(T_{in} - T_A) = hA(T_A - T_M) \quad (1)$$

Note that $h$ is the effective averaged value. Equation 1 can be rearranged as:

$$T_A = a(T_{in} - T_M) + T_M \quad (2)$$

where:

$$a = \frac{\dot{m}c_p}{\dot{m}c_p + hA} \quad (3)$$

The parameter $a$ is labelled here as the source strength. It can be seen to tend towards 0 at small mass flows (in proportion to the product $hA$); conversely to tend towards 1 at large mass flows or low $hA$.

Equation 2 shows that the exit air temperature sits at a constant proportion of the gap between the inlet temperature $T_{in}$ and the block temperature $T_M$, irrespective of the time history of either.

Specific solutions are now derived for the case with a step change in inlet air temperature $T_{in}$ at time $t = 0$.

**Infinite source**

If the source strength $a = 1$, equation 2 becomes $T_A = T_{in}$. The heat flow into the block, and the rate of its temperature rise are:

$$\dot{Q} = hA(T_A - T_M) \quad (4)$$

$$MC\frac{dT_M}{dt} = hA(T_A - T_M) \quad (5)$$

Eliminating $T_A$ gives the equation for $T_M$ as:

$$b\frac{dT_M}{dt} + T_M = T_{in} \quad (6)$$

where:

$$b = \frac{MC}{hA} \quad (7)$$

The parameter $b$ is labelled here as the intrinsic time constant of the block. For the case of a step input of the inlet temperature to $T_{in}$ at $t=0$, the block’s temperature response becomes the exponential solution i.e.:
\[ T_M = T_0 + (T_{in} - T_0) \left(1 - e^{-t/b}\right) \] 

This means the block temperature responds with a single time constant, equal to \(b\).

**Finite source**

If the source strength \(0 < a < 1\), then the above derivation changes as follows: equation 5 still holds, but now \(T_A \neq T_{in}\) but is given by equation 2. The equation for the metal temperature then becomes:

\[ b \frac{dT_M}{dt} + a T_M = a T_{in} \] 

And the solution for \(T_M\) becomes:

\[ T_M = T_0 + (T_{in} - T_0)(1 - e^{-ta/b}) \] 

This means the block temperature responds with a single time constant \(\tau = b/a\).

The solution for this case is illustrated in Figure 5. The block temperature response comes from equation 10, and the air temperature response from equation 2. Three features to note are:

- The block temperature responds with a time constant equal to \(b/a\).
- The air temperature is at all times at the fixed proportion \(a\) between the block temperature and the driving inlet temperature.
- The air temperature at time zero exhibits an immediate response, it being \(a\) times the initial temperature difference \(T_{in} - T_0\).

**CHARACTERISED RESULTS**

The experimentally measured air and metal temperature responses have been characterised using the equations derived previously. A differential form of the temperature response equation was used, given by equation 11 and equation 12 for the metal and air temperatures respectively. These equations were used to time-step through the recorded data, using a time step of 0.1 s.

\[ T_M(i + 1) = T_M(i) + (T_{in}(i + 1) - T_M(i))(1 - e^{-\Delta t/\tau}) \] 

Then

\[ T_A(i + 1) = T_M(i + 1) + a(T_{in}(i + 1) - T_M(i + 1)) \]

Air temperatures were measured at four locations along the flow path. The first, \(T_{A1}\) at the heater exit was taken as measured, and used as the input \(T_{in}\) parameter to the calculation of the characterised \(T_{A2}\) temperature. The two coefficients \(a\) and \(\tau\), were tuned until the best fit was obtained to the full five minute heating curve for \(T_{A2}\). The calculated \(T_{A2}\) value was then used as the input to the calculation of \(T_{A3}\), which was used to calculate \(T_{A4}\).

Figure 6 shows the accuracy of the characterised fits to the air temperatures, and Table 2 the coefficients which generated the curves.

A similar approach was used for the metal temperature locations, however the measured air temperature at the ramp offtake \(T_{A3}\), after correcting out its own time constant delay, was used as the inlet driving temperature for all twelve locations.

Figure 7 shows the accuracy of the fits at an example fast-responding and at a slow-responding metal temperature location.
Figure 6: Air temperature responses showing characterisation

Flow section | Immediate response proportion \((a)\) | Time constant \(\tau = b/a\)
--- | --- | ---
From \(T_{A1}\) to \(T_{A2}\) | 0.60 | 55
From \(T_{A2}\) to \(T_{A3}\) | 0.78 | 50
From \(T_{A3}\) to \(T_{A4}\) | 0.52 | 100

Table 2: Best fit air temperature characterisation coefficients for datum run.

Figure 7: Metal temperature response showing characterisation
Fits to the measured data were of similar quality at all twelve locations, showing that each could be accurately captured by a single time constant. Figure 8 then summarises the result from the datum test, conditions given in table 1, by marking the twelve time constants on to the geometry.

**TESTS AT REDUCED OFFTAKE FLOW**

In addition to the datum test, further experimental runs were conducted varying the offtake mass flow, by varying the number of offtake valves open. Tests were also run with a reduced test pressure, the run list is given in table 3.

It was found that with the tests at a reduced mass flow rate, runs 1 to 3, the time constants increased as the flow rate reduced, by a factor which varied at the different metal locations between $\times 2.2$ and $\times 3.1$. This implies that the heat fluxes, and hence HTCs, show a dependence on mass flow between $\dot{m}_{0.5}$ and $\dot{m}_{0.8}$ at the various locations. Table 4 summarises the time constants across three runs, and gives the best fit power index at each location for the mass flow dependence across the three runs.

**TESTS AT REDUCED PRESSURE**

Tests were also run at lower pressures. This gave various test pairings where the offtake mass flow (in kg/s) was the same but the pressure, and hence air density was different, runs 4 and 2 is one such pairing.

For run 4 the pressure was halved, but as the air temperature was the same as for run 2, the density was doubled and air velocities were halved. The results showing the response at locations 1 and 5 to the driving air temperature $T_{A3}$ for each run, are given in Figure 9.

It was found that the time constants in run 4 were unchanged from those of run 2, at all twelve locations. This leads to the conclusions on which parameters control the scaling of the time constants. The time constant at a location scales with offtake mass flow and Reynolds number, but does not scale with air density or annulus Mach number.
<table>
<thead>
<tr>
<th>Location</th>
<th>Run 1</th>
<th>Run 2</th>
<th>Run 3</th>
<th>( \tau \propto \text{Re}^{-\alpha} )</th>
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<tr>
<td>1</td>
<td>31</td>
<td>51</td>
<td>70</td>
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<td>2</td>
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<td>56</td>
<td>88</td>
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<td>24</td>
<td>34</td>
<td>49</td>
<td>0.50</td>
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</table>

Table 4: Time constants with varying offtake flow, runs 1 to 3, pressure = 12 bara

Figure 9: Temperature responses showing effect of testing at 12 bar (run 2) and 6 bar (run 4)
DISCUSSION

Heat Transfer Coefficients

It is interesting to note how widely the heat transfer can vary with location even within an individual compressor casing cavity. The results showed a factor of nearly 6 between fastest and slowest time constants (at locations 2 and 4 respectively). This is despite the fact that the wall section thickness is the same at these two locations. The implication is that the local HTCs vary by a factor of six between the front part of the offtake cavity and the rear part.

The object of the paper is to present the thermal results in terms of time constant, and how they vary spatially and with mass flow. An interpretation in terms of HTCs, as well as time constant, directly from the data is possible at those locations on the cylindrical casing sections where the wall thickness is clear and conduction effects will be small. In these cases the HTC can be inferred from \( h = \frac{(\rho C w)}{\tau} \) where \( \rho \), \( C \) and \( w \) are the material density, specific heat capacity and local wall thickness for the casing respectively. The locations susceptible to this interpretation are locations 1 to 6. These happen to all have sensibly the same wall thickness, from which HTCs for the datum test are inferred as ranging between 350 W/m\(^2\)K at location 2 down to 65 W/m\(^2\)K at location 4. Note that the Biot number is well below 0.1 even for the higher end of this HTC range. At the other locations, locations 7 to 12, the effective mass and corresponding convection surface area are not clear; inference of the HTCs would require matching of the thermocouple data in a combined conduction and convection method such as a finite element model.

Heater Ramp Rate

The heater controlled the ramp up of its power such that it took 40 s to reach 63% of, and 100 s to reach 90% of, the final temperature rise. Moreover, this was compounded by the further thermal slugging introduced between the heater exit and the air getting to the offtake location. The times for the temperature to reach 63% and 90% response were thus 70 s and 160 s.

Despite these time delays being as large as or larger than the casing time constants themselves, it proved possible to establish casing time constants, even down to the fastest one of 18s, with a high degree of resolution. The confidence level on the quoted time constants is better than \( \pm 10\% \), based on the closeness of fit (better than \( \pm 2 \) K) throughout the heating curve of all twelve metal locations and the consistency between tests at different mass flows. This in turn shows that it is unnecessary to have a faster ramp up of the heater input power for testing this type of casing. It also shows the value of measuring the air temperature at the ramp offtake itself rather than referring the metal responses to a more upstream temperature.

CONCLUSIONS

The test facility has been shown to be capable of testing the heat transfer in the offtake cavity regions of engine compressor casing assemblies, at engine conditions up to and including cruise levels of pressure and mass flow.

Metal and air temperatures were measured in a compressor casing in response to a fast ramp change in inlet air temperature. These measurements allowed the time constants in the casing to be characterised.

Time constants can vary considerably between the individual surfaces which bound a specific offtake cavity in a compressor casing assembly. In the datum case reported here the time constants varied in the range between 19 s and 104 s between the various locations. Scaling of
time constants varied slightly between the locations, from a dependence of \( \text{Re}^{0.5} \) up to \( \text{Re}^{0.8} \).

The results showed conclusively that time constants should be scaled between conditions with mass flow, or Reynolds Number. The time constants, at a given mass flow, were found to be independent of air density, axial velocity or Mach Number in the annulus.

The time constant results can be reviewed for which surfaces would benefit from heat transfer reduction measures, such as heat shielding. Moreover a move from engine testing to rig testing means that such studies can occur at an earlier engine design stage.

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