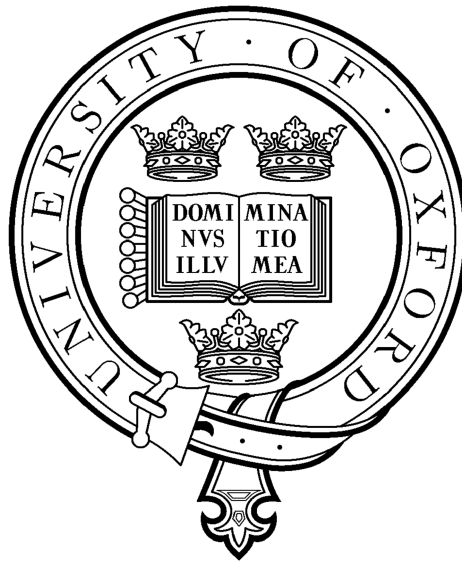


On The Thermal Behaviour Of Gas Turbine Filament Seals

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A thesis submitted in partial fulfilment of the
requirements for the degree of *Doctor of Philosophy*

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Abstract

ON THE THERMAL BEHAVIOUR OF SOME TURBINE SHAFT FILAMENT SEALS

Doctor of Philosophy in Engineering Science

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Advanced rotating shaft seals have the potential to significantly increase the efficiency and performance of steam and gas turbines. Two such seals, brush and leaf seals, rely on the use of thousands of flexible filaments to close clearances between rotating components and their static casings. The current life of the components is poor compared to the rest of the gas turbine, limiting the seals' deployment, particularly in the jet engine at high temperature and pressure. Poor understanding of the seal installation response to frictional heat generated at the point of filament-rotor contact during operation has limited the ability to predict engine closures and hence seal behaviour and life. The resulting temperature rises may compromise the mechanical integrity of the engine rotor in extremis leading to a shaft failure.

This thesis considers the heat transfer mechanisms that govern frictional heating, of both the fluid and solid components in the vicinity of such seals, characterising the process both experimentally and using numerical models. Through the identification of key features of the heat transfer a simple numerical methodology is shown to predict the thermal behaviour of the seal installation sufficiently accurately for engine design purposes.

A low order heat transfer model, using a simple electrical analogy for heat transfer is used to investigate frictional heat generation. When contact occurs between the rotor surface and the seal filaments, mechanical energy is dissipated as heat at the interface. This is conducted into the rotor and the seal filaments in proportions that depend on

the heat transfer characteristics of both contacting bodies (thermal resistances). To calculate the heat partition ratio and the resulting contact temperature, the thermal resistances of both rotor and seal need to be known.

To that end, a new test facility, the Seal Static Thermal Test Facility (SSTTF), is developed. This is first used to study the convective heat transfer occurring in the vicinity of the seal; heat transfer coefficients based on appropriate, scalable, gas reference temperatures are reported. Importantly the results show a larger area on the rotor surface affected by the presence of the seal than was assumed by previous workers. The test rig is further modified to generate heating in a static test rig equivalent to the frictional heating at the filament tips. The test rig allows the contact temperature between rotor and seal, a critical previously unknown parameter to be measured in a well-conditioned environment.

The presence of many thousands of vanishingly small flow passages in filament seals makes their explicit modelling unfeasible for engine design purposes. Thus the results from the experimental campaign are used to develop a simple computational fluid dynamic model of the seal, including empirically derived frictional heating, and seal porosity models, to achieve similar leakage and surface heat transfer to the rotor as was seen in the static experiments.

The low order CFD methodology presented in the thesis is finally employed to model the transient operation of a brush seal under engine representative rotor surface speeds and differential pressures. Experimental data were generated in the Oxford Engine Seal Test Facility for a typical brush seal rubbing against a high growth rotor. These experiments were modelled using CFD and finite element analysis using parameters derived from static tests for the porous modelling of the seal leakage. Comparison of results shows that, without further tuning, the thermal behaviour is captured well with a moderate conservative overestimation of rotor heating with increased differential pressure across the seal allowing the strategy to be used as an engine design tool.

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Nomenclature

Roman Symbols

A	Area [m^2]
b	Backing ring/downstream coverplate clearance [m]
Bi	Biot number, $Bi = (htc L) / k$ [-]
C_p	Specific heat capacity [$\text{J kg}^{-1} K^{-1}$]
D	Seal inner diameter (filament tips) [m]
e	Effective clearance [m]
F	Dimensionless flow function, refer to Eq 5.17 [-]
f	Friction factor [-]
htc	Convective heat transfer coefficient [$\text{W m}^{-2} K^{-1}$]
I_r	Rotor inertia [kg m^2]
k	Thermal conductivity [$\text{W m}^{-1} K^{-1}$]
K_{seal}	Seal thermal constant, refer to Eqs 5.2-5.3 [$\text{kg}^{0.5} \text{m}^2 K^{-1} s^{-2.5}$]
L, l	Length [m]
L_w	Axial width of filament pack [m]

$\dot{m}$	Mass flow rate [kg/s]
N	Number of filaments [-]
P	Perimeter [m], power [W]
Pe	Peclet number, $Pe = (v L) / (4 \alpha)$ [-]
Pr	Prandtl number, $Pr = (C_p \mu) / k$ [-]
Q	Heat, frictional power [W]
q	Heat flux [$W m^{-2}$]
R	Thermal resistance [$K W^{-1}$]
r	Recovery factor [-]
Re	Reynolds number, $Re = (\rho v L) / \mu$ [-]
T	(Static) Temperature [K], torque [Nm]
t	Time [s]
V	Voltage [V]
v	Velocity [$m s^{-1}$]

Greek Symbols

α	Thermal diffusivity [$m^2 s^{-1}$]
δ	Leaf thickness [m]
ρ	Density [$kg m^{-3}$]
ρ_{Ω}	Resistivity [Ωm]
ϵ	Efficiency [-]

η	Heat partition ratio [-]
γ	Ratio of specific heats [-]
μ	Viscosity [$\text{kg s}^{-1}\text{m}^{-1}$]
ω	Angular speed [rad/s]

Subscripts

0	Initial, ambient
<i>avg</i>	Average
<i>aw</i>	Adiabatic wall (temperature)
<i>c</i>	Contact
<i>driv</i>	Driving (temperature)
<i>eq</i>	Equivalent
<i>ext</i>	External
<i>f</i>	Friction, fluid, flash (temperature)
<i>int</i>	Internal
<i>m</i>	Metal (seal material) property
<i>n</i>	Nominal
<i>pack</i>	Related to filament (bristle/leaf) pack
<i>ref</i>	Reference conditions
<i>s</i>	Static (temperature)
<i>tc</i>	Thermocouple

Tot Total

Acronyms / Abbreviations

CFD Computational Fluid Dynamics

ESTF Engine Seal Test Facility, see section 6.1

FEA Finite Element Analysis

FE Finite Element

IR Infra Red

RSS Root-Sum-Square equation

SDP Seal Differential Pressure

SSTTF Seal Static Thermal Test Facility, see section 4.1

Chapter 1

Introduction

The importance of turbomachines in today's world is unquestionable. Compressors and turbines are essential components present in gas and steam turbine engines (Fig 1.1 and 1.2). These power plants, which represent together with piston engines the main industrial prime-movers, could be considered among the most complex devices designed and built by humans.

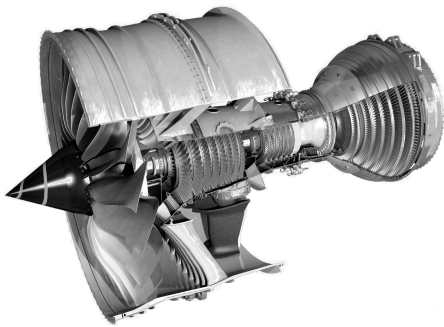


Figure 1.1: Aircraft gas turbine,
Rolls-Royce (2016)

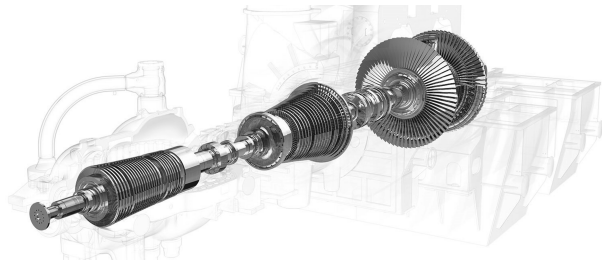


Figure 1.2: Steam turbine power plant, *General Electric (2016)*

Gas turbines, in their simplest form, follow the open Joule-Brayton cycle, whose main mechanical components are depicted in Fig 1.3. The working fluid, generally air, is compressed in the first process of the cycle. Then, part of the chemical energy contained in the fuel is transferred to the gas, raising its temperature in the combustion chamber. Expansion of the hot working fluid in the turbine then produces a power

output greater than that required to drive the compressor. The shaft of the turbine can be coupled to an external load to transmit this useful power. Otherwise, if the working fluid is expanded in the turbine just to produce enough power to drive the compressor, the gas leaving the turbine at high pressure and temperature can be further expanded to atmospheric pressure in a nozzle. The resulting high-velocity jet is commonly used to propel an aircraft forward.

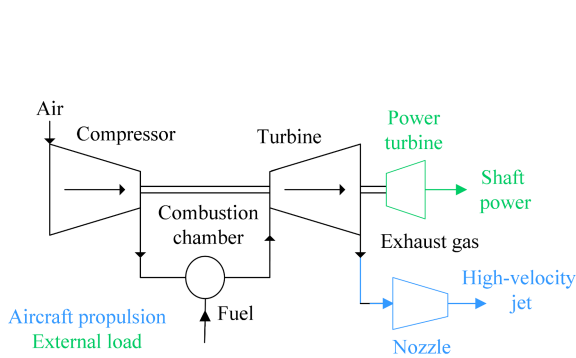


Figure 1.3: Gas turbine engine

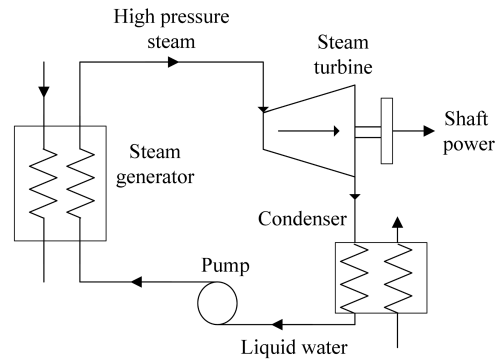


Figure 1.4: Steam turbine engine

Steam power plants operate on the Rankine cycle (Fig 1.4) or on some modification of it, with water being the most common working fluid employed. The main difference between Rankine cycle and other gas cycles, such as Diesel or Joule-Brayton, is the fact that in the former, the working fluid appears in both liquid and gaseous state. The Rankine cycle commences with the compression of the working fluid, in liquid state, in a pump. Then, heat is added at constant pressure in the steam generator. Fossil fuels, nuclear fission or renewable energy are examples of heat sources used in steam power plants. The high pressure steam is finally expanded in a turbine, which is connected to an external load.

The useful output generated in the Joule-Brayton or Rankine cycles is nowadays used in many applications. Gas turbines were initially conceived as military aircraft engines. The advent of the Second World War motivated the need for faster combat planes. The higher power to weight ratio and compactness of gas turbines promoted their substitution for piston engines in aviation. Since then, huge improvements in power and efficiency

have converted gas turbines into a reliable and flexible engine suitable for an increasing number of applications. The great development of global tourism has only been possible due to the progress made in aero engines, gas turbines being the main propulsion system in both military and civil aircraft at present. In those applications where size and weight are not so critical, both steam and gas turbine engines find great usage. They can be employed for electric power generation if the turbine is coupled with an alternator. Also, they are suitable for driving a ship or submarine propeller. Gas turbines are commonly exploited to drive a pump for oil extraction or natural gas transmission along pipelines. Used in combination, gas and steam cycles achieve thermal efficiencies of close to 60 % (Saravanamuttoo et al. (2009)). Steam cycles are used with a wide variety of primary heat sources for the the production of electricity. They are employed in conventional energy generation systems such as nuclear and fossil fuel plants and also in alternative technologies such as solar, bioenergy and geothermal. In terms of the world's energy supply, power plants based on steam and gas turbines play a very important role. In the UK alone, approximately 83% of the electricity production in the first quarter of 2016 was generated using gas and steam turbines (Energy UK, (2016)). While current demand for baseload supply is under threat from renewables, a shift towards electric vehicles may reinvigorate the need for large scale nuclear in the future.

1.1 Sealing in Gas and Steam Turbines

There is a demand for continual improvements in the operating efficiency of steam and gas turbines. Improved performance from the secondary air system in a gas turbine is critical in meeting this goal. The engine secondary air system includes those airflows which do not directly contribute to produce thrust or shaft power but that are vital for appropriate engine operation. In a modern gas turbine, up to one fifth of the total engine core airflow is bled off at various compressor stages according to Rolls-Royce (2005). Very little data exist in the open literature referencing this value, however, *GASTURB* calculations based on the Trent XWB engine used in the A350 suggest cooling flows accounting for 15% of the mainstream flow, making a total secondary air requirement of

20% a conservative estimate [Halliwell (2014)]. This airflow is used to cool critical components, manage thrust loads on bearings, control blade tip clearances, provide bearing chamber sealing and prevent hot gas ingestion into the turbine disc cavities. In order to achieve high thermal efficiency and specific power, gas turbines must be operated at high pressure ratios and temperatures. The high turbine inlet temperature found in a modern engine (1600 - 1900 K) is above the melting point of the most advanced blade materials. Advance cooling methods using secondary air are therefore essential to prevent blades from melting and limit creep and thermal stresses. Since the secondary air system makes no direct contribution to produce useful power, it is important to reduce its mass flow rate and operating pressure by minimizing leakages and pressure loss. Moreover, secondary air system optimization generally produces performance gains at a lower investment cost in comparison with the development of main gas-path technologies (Hendricks et al. (1998), Chupp et al. (2006)). The operation of steam turbines is also improved by increasing steam temperature and pressure. With higher cycle pressures sealing and clearance control between rotating and static components becomes more challenging. Minimizing steam leakage from the rotor ends or through leakage paths between the high and intermediate pressure sections of the turbine strongly contributes to increase the efficiency of the power plant. In addition, an adequate sealing of the rotating blade tip clearances in the high and intermediate pressure turbine greatly improves its performance.

Seals are the critical components employed to manage secondary airflows and control turbomachinery clearances. A wide variety of seal types are installed in steam and gas turbines. The choice of the sealing method is dependent upon several factors, such as the surrounding temperature and pressure, seal degradation and heat generation, weight and space available and clearance variation with operating conditions. Some of the sealing technologies commonly used in turbomachinery are labyrinth, ring, carbon, brush, leaf and finger seals. For instance, Fig 1.5 illustrates the airflow path in a hypothetical secondary air system of a gas turbine with its sealing arrangement. It can be observed how secondary air is delivered to the turbine blades and discs to control their temper-

atures. These airflows are channelled by means of various seals, such as labyrinth and brush seals in rotating positions, but cloth, strip, bellows and piston ring seals in other static locations. Fig 1.6 shows the use of leaf seals in a Mitsubishi high efficiency steam turbine (Watanabe et al. (2003)). These are installed at the rotating blades for sealing the tip clearance and at the sealing section between the high pressure exhaust and the gland seal to reduce steam leakages.

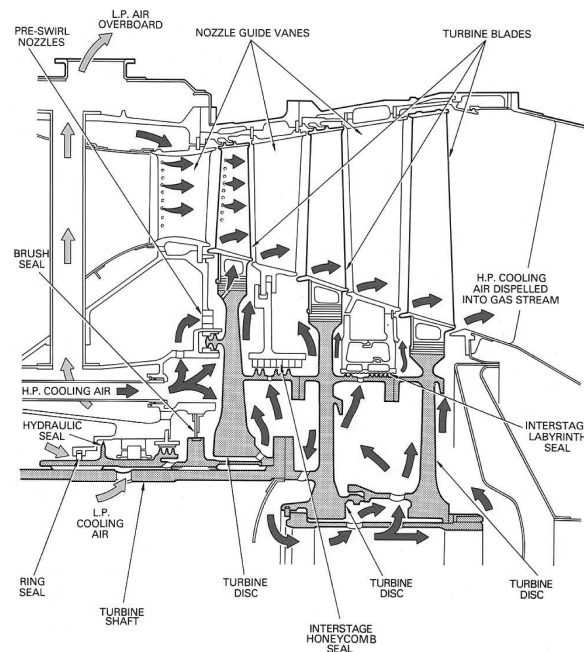


Figure 1.5: Example of a gas turbine secondary air system, Rolls-Royce (2005)

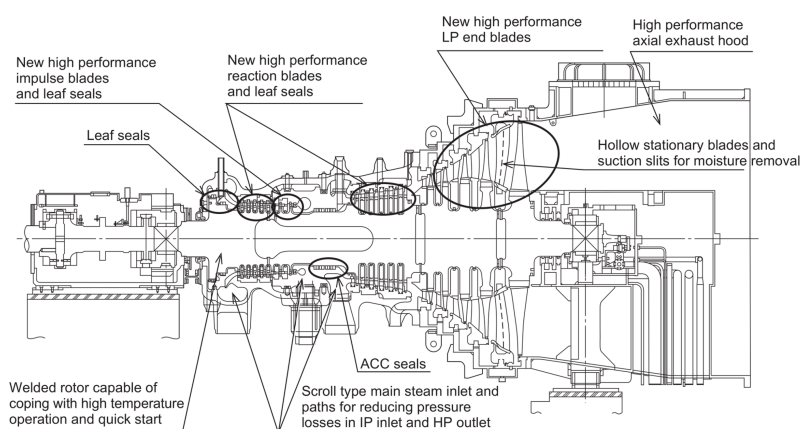


Figure 1.6: Application of the leaf seal to a steam turbine, Watanabe et al. (2003)

1.2 Shaft Seals

Since the early 1990's, research into turbomachinery sealing has been carried out at the University of Oxford. This has focused on shaft seals, which are employed to close clearances at static to rotating interfaces. These sealing locations consist of a stationary land, to which the seal is attached, and a rotating component. Rotor surface speeds can be in excess of 400 m/s, and seal differential pressures reach 10 bar (up to 25 bar for a steam turbine). Moreover, the axial and radial clearances between the rotating and static parts are subjected to variation at different engine operating conditions. Some arise due to the thermal and centrifugal growth of the rotor and casing occurring throughout the engine cycle. In aircraft gas turbines, flight manoeuvres and transient accelerations of the plane can result in rotor eccentric misalignment and excursions. In steam turbines, whirl and ovality occur. In addition, the uncertainty in the size of the rotor-land gap increases when the manufacturing tolerances of engine components are considered. Therefore, shaft seals must be designed to maintain low leakages rates in this changing and adverse environment without incurring in the risk of an excessive contact with the rotor that may result in a failure of the installation. An effective seal is one which keeps leakage to a minimum over the entire engine cycle with the least possible performance degradation over time.

Labyrinth seals are extensively used for rotary sealing in steam and gas turbine since their origins in the 1870s. An engine schematic showing different labyrinth seal configurations is shown in Fig 1.7. This type of shaft seal consists of an alternating set of teeth or fins located in close proximity to the rotating surface. These create a tortuous path along which some of the kinetic energy of the leakage flow is dissipated causing a pressure drop. The distance between the fins and the rotating surface must be large enough to avoid contact even in the event of a rotor excursion. Different versions of labyrinth seals have been designed; some run on abradable materials that are shaped on the first engine run (allowing the gaps to be as small as possible) and reduce the risk of failure if contact occurs. With increasing seal differential pressure, a larger number

of fins are required resulting in an increment in the axial length and weight of the installation. The major limitation of labyrinth seals is the inherent necessity of allowing a certain clearance between fins and rotor to avoid contact and damage at all the possible running conditions of the engine cycle. This results in relative high leakage rates and a poor off-design performance of the seal. On the plus side, labyrinth seals offer minimum heat dissipation and torque losses from the shaft. Despite the drawbacks of labyrinth seals, they serve as a robust and reliable seal that have been installed and proven in numerous commercial engines.

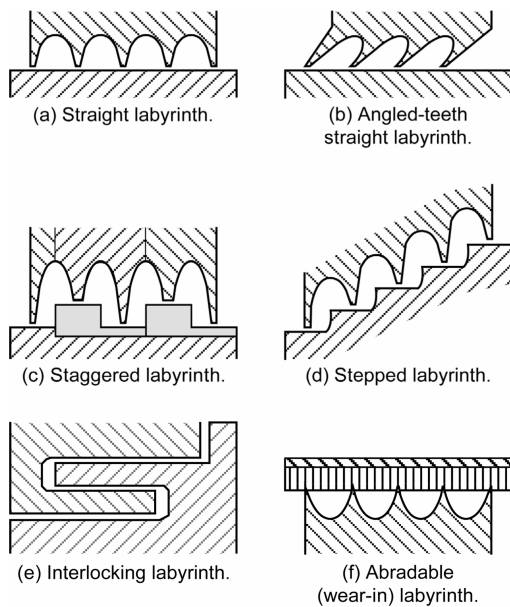


Figure 1.7: Typical labyrinth seal configurations, Chupp et al. (2006)

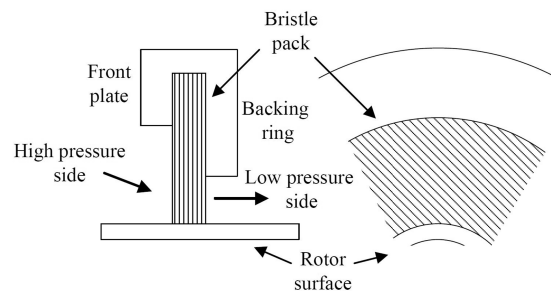


Figure 1.8: Brush seal schematic

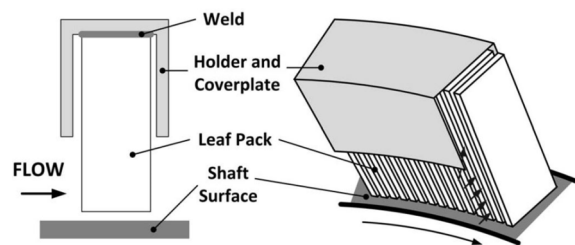


Figure 1.9: Leaf seal schematic, Jahn et al. (2013)

Brush seals emerged as an alternative to labyrinth seals in the 1970s, initially conceived for aircraft gas turbines applications (Ferguson and Waters (1975)). The typical arrangement of a brush seal, shown in Fig 1.8, consists in a pack of fine metallic flexible bristles that are joined to an outer ring at their roots. The outer ring is housed into the static land, while the tips of the bristles remain in close proximity (or even contacting) the rotating surface. The bristle pack is contained between a front plate and a backing

ring, whose configuration largely governs the dynamics of the seal filaments. In order to leak, the fluid has to move within the tightly packed bristles, experiencing a large pressure drop. The backing ring provides structural support to the flexible bristles and also acts as an additional flow obstacle. This advanced shaft seal is intended to provide a better flow restriction and a more rotor-compliant behaviour than labyrinth seals. The bristles are inclined to the radial direction so that they can deflect to accommodate variations in rotor radius due to thermal and centrifugal growths. In addition, brush seals are not damaged by a rotor excursion, such as would typically result with labyrinth fins. Brush seal testing has shown that they present leakage rates of about one third of those using a conventional labyrinth seal (Ferguson (1988)). In addition, for a given pressure drop, the brush seal requires less axial space and weight than the labyrinth seal. An example of the application of brush seals to a range of General Electric gas and steam turbines is provided in Dinc et al. (2002).

A relatively new shaft seal design is the leaf seal, which is disclosed in several patents (Flower (1992), Wright (2001), Shinohara et al. (2002)). Similarly to brush seal, it is made up of thousands of flexible filaments, in this case, thin plates. These are arranged around the circumference of the rotor as shown in Fig 1.9. The leaves are usually welded at the root and inclined to the radial direction forming a compliant seal. The leaf pack is inserted between the upstream and downstream coverplates, which strongly determine the aerodynamic behaviour of the leakage flow. Leaf seals present some advantages in comparison with brush seals. Although they offer similar leakage levels, leaf seals are more rotor-compliant due to their lower radial stiffness, produce less wear and performance degradation with time, and present a higher axial rigidity that allows them to cope with higher differential pressures. The radial stiffness of the leaf seal is almost an order of magnitude lower than that of the brush seal and therefore they can accommodate larger rotor movements. Contrarily to brush seals, leaf seals do not suffer from pressure stiffening, i.e. a reduction in compliance as the pressure differential across the seal is increased (Ingo et al. (2008)).

Although leaf seal technology is largely at a development stage, some seals have

successfully been tested in real engines. Nakane et al. (2004), from Mitsubishi Heavy Industries (MHI), reported results from 1000 h of successful operation of a prototype leaf seal in an M501G gas turbine. In addition, MHI currently has in service leaf seals in some of their commercial steam turbines (Mitsubishi Hitachi Power Systems, (2015)), though they are no longer offered in current products. Rolls-Royce plc has been researching leaf seals in the last decade in close collaboration with the University of Oxford and Alstom Power Ltd. The main technical challenge for the development of leaf seals arise from their unstable mechanical behaviour. The aerodynamic forces exerted by the leakage flow result in vibration and flutter of the seal filaments, causing a deterioration of the seal or even the loss of leaves (*private communications with Rolls Royce Plc, February 2016*). In addition, there is still little information about the leaf seal characteristics in terms of variation in wear rate with operating pressure and temperature.

1.3 Research Problem

Brush and leaf seals employ flexible filaments (bristles and thin plates respectively) to seal clearances between a rotating shaft and a stationary land. Although their sealing performance is superior to that of labyrinth seals, contact between the filament tips and the rotor surface occurs even under normal operation conditions. When a brush or leaf seal is pressurized, the leakage flow generally exerts an aerodynamic force that pushes the bristles or thin plates radially inwards. This phenomenon, known as blow-down, improves the compliance of the seal at the expense of increasing friction contact. This implicit operating mode of the seal involves two main disadvantages. First, the seal suffers from wear and performance degradation. While the initial leakage allowed by these seals can be low, continuous contact with the rotating surface means that, after a certain number of hours of operation, this increases markedly. Performance engineers need to know the evolution of seal behaviour with operating time to ensure a correct supply of secondary air in the case of a gas turbine. Moreover, seal wear significantly increases the maintenance cost of the engine, and also reduces the interval between overhauls.

The second issue arising from seal-rotor contact is loss in shaft power. There is an evident drag force acting on the engine spools, which becomes significant when a large number of these seals are installed. If friction occurs, part of the mechanical energy of the rotor is converted into heat at the contact interface. The order of magnitude of this power loss can approach 20 kW in an extreme transient with an unfavourable seal geometry (*private communications with Rolls Royce Plc, October 2015*). This results in a temperature rise that can compromise the mechanical integrity of the seal installation. The frictional heat generated is dissipated through conduction into the rotor and into the filament pack, and thus, to the leaking flow. The resulting contact temperature is dependent on the ability of both seal and rotor to dissipate the heat released. The heat transferred to the seal is rapidly convected to the airflow that leaks through the filaments. The remaining heat is conducted along the rotor and progressively convected back to the surroundings. The heating of the rotor leads to the possibility of a runaway rotor growth event. This occurs when the rotor grows radially into the seal filaments due to its thermal expansion. The resulting increment in friction produces additional heat generation and rotor growth. This unstable operation eventually causes the failure of the seal installation. Moreover, the temperature rise may also impact the operational life of the seal if local temperatures reach its melting range due to the formation of hot spots.

One of the potential advantages of leaf seals is the possibility of inducing air-riding effects. Leaf seals can exhibit hydrodynamic air-riding under certain operating conditions (Jahn et al. (2013)). When this phenomenon occurs, a tiny air gap forms between the leaf tips and the rotor surface. This behaviour is desirable since it prevents seal degradation and contributes to retain seal performance, without increasing leakage dramatically. Therefore, air-riding represents the ideal mode of operation of the seal. The coverplate configuration of the leaf seal can also be adjusted to balance the forces acting on the leaves in a way to minimise wear and heat generation. However, the mechanisms that govern air-riding are not fully established and its presence cannot always be guaranteed. Leaf seals may experience conditions of heavy contact and severe heat generation

during certain parts of the engine cycle, such as aircraft take-off or at speed ranges where air-riding does not occur. Also, large rotor growths or eccentricities cannot be accommodated with air-riding alone.

1.4 Thesis Scope

This thesis focuses on the study of the heat transfer mechanisms occurring in brush and leaf seals in the fully contacting regime. The term filament seal will be used throughout this thesis to refer to brush and leaf seals. As will be discussed in the literature review, many previous researchers have attempted to model filament seal frictional heat generation. Nevertheless, most of these studies are based on mathematical models that lack of experimental observations that support the conclusions derived. The convective heat transfer characteristics of the flow upstream and downstream of a filament seal have not been investigated in the past, and results from generic internal flow have been extrapolated. Moreover, the outcome of previous models has not been compared with test data from a running seal, and therefore, their validity is uncertain. The present work aims to fill these gaps in knowledge so that more accurate heat transfer models can be built. Ultimately, the main goal is to develop a methodology that can be employed to predict the temperature rise attained when frictional heating occurs, ensuring the safe operation of the seal installation. The findings on seal heat transfer resulting from this work will be used during engine design. To that aim, the actions listed below have been carried out:

Chapter 2 provides a review on the operating principles and main features of brush and leaf seals, together with a discussion on previous work on their thermal modelling.

Chapter 3 reports and synthesizes pertinent previous work in heat transfer modelling of frictional heat generation. The main existing modelling strategies are examined in detail. From the discussion, an approach based on a simple electrical analogy is chosen as the cornerstone of all the subsequent work, and this identifies where there are current gaps in the knowledge.

Chapter 4 focuses on the study of the convective heat transfer characteristics of the

flow over a rotor where a leaf seal is installed. A new test facility is developed to that end. The intrinsic difficulties found for the application of the conventional heat transfer coefficient approach are set forth: namely the lack of a suitable reference gas temperature. Consequently, a methodology is proposed to alleviate this issue, which is assessed using a computational fluid dynamics (CFD) model. Its application to the experimental data obtained shows for the first time the convective heat transfer coefficient distribution in the close vicinity of a leaf seal.

Chapter 5 addresses the modelling of the seal filament pack heat transfer. In view of the inherent difficulties that a detailed mathematical description of the actual flow path and pack geometry presents, an empirical method is employed. Based on previous work at Oxford, a form of correlation is proposed to estimate the amount of heat that a brush or leaf seal can absorb for a given contact temperature rise and leakage mass flow rate. An adaptation of the experimental facility described in Chapter 4 is used to generate experimental heat transfer data to populate the semi-empirical model. Frictional heat is simulated using an electrical heat source, which produces a known amount of heat that is subsequently transferred to the seal. Brush and a leaf seal geometries are tested in the experimental facility. Analysis of the data obtained confirms the validity of the form of the correlation considered, and allows its calibration. The thermal performance of the two seal geometries investigated are compared, which reveals the higher convective heat transfer occurring within the brush seal.

Finally, in Chapter 6, all the previous research effort is brought together to model the thermal behaviour of a brush seal installation. The existing Engine Seal Test Facility is used to generate heat transfer data of a pair of brush seals running at engine representative surface speeds and differential pressures. Infrared sensors allow the characterization of the rotor temperature rise caused by seal-induced friction. A finite element model of the test rotor, combined with CFD simulations and the empirical correlation previously derived in Chapter 5, is used to predict the transient evolution of the rotor temperature during the tests. The results obtained with the model are analysed and compared with the experimental measurements.

With Chapter 7, this thesis concludes summarizing the main achievements and findings derived from the research and recommending further work.

1.5 Publications

To date, one publication has arisen from this research:

J. D. Pelegrin-Garcia, D. R. H. Gillespie, M. J. Pekris, G. Franceschini, L. Ganin, Experimental Characterization of Rotor Convective Heat Transfer Coefficients in the Vicinity of a Leaf Seal, first presented at *ASME Turbo Expo 2016 (GT2016-57603)*, Seoul (South Korea), June 2016 and published in the *Journal of Engineering for Gas Turbines and Power*, GTP-16-1285, Vol. 139, Issue 3, March, 2017.

Chapter 2

Literature Review

In this chapter, previous research on filament seals relevant to the present work is discussed, and the state of the art described. Detailed reviews of literature on brush and leaf seals were carried out by Franceschini (2005), Jahn (2008) and Pekris (2010). In these works, the potential benefits arising from the use of these technologies, such as improved engine performance, fuel saving and reduction in pollutant emissions, are explained. The historical development of brush and leaf seals from their original patents along with the different improvements and design modifications proposed in the following years are also related. In addition, alternative competing sealing technologies are examined. The main strategies used for modelling and predicting the behaviour of brush and leaf seals are thoroughly analysed by these authors and key experimental observations acquired in the past reported. As explained in the introductory chapter of this thesis, the focus of the present research problem is the investigation of the heat transfer characteristics of brush and leaf seals in the contacting regime. Therefore, here only the literature that is directly related to the thermal analysis of the seal is reviewed. First, a brief discussion on the main design features, operating conditions and fundamental principles under which these seals operate is provided. Second, previous thermal models and observed phenomena are analysed in detail. Finally, a section is included that summarizes the lessons learnt, being these the basis of the subsequent work.

2.1 General Considerations on Brush and Leaf Seals

A deep knowledge of the operation of brush and leaf seals is essential to fully understand the heat transfer mechanisms occurring and tackle the research problem properly. The ideas presented in this section are the result of the study and synthesis of a large amount of work carried out at Oxford during the last twenty years. Technical discussions with Rolls-Royce seal specialists have also been crucial for the author to gain insight into brush and leaf seal behaviour.

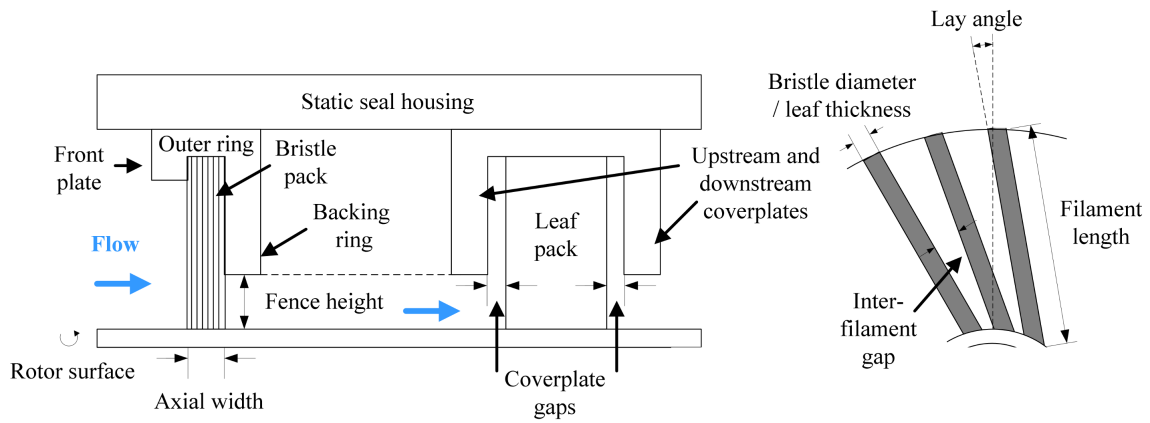


Figure 2.1: Schematic diagram of brush and leaf seals

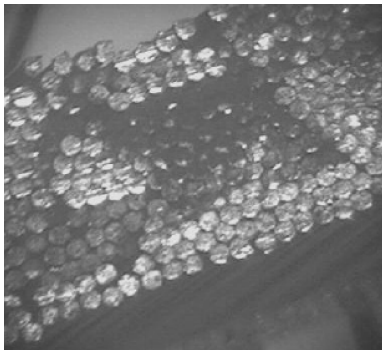


Figure 2.2: Brush seal packing, Franceschini (2005)

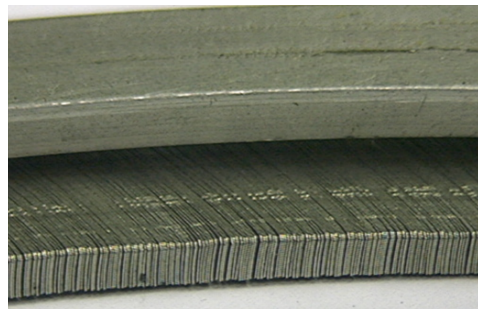
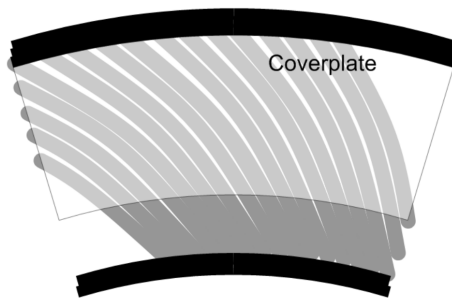


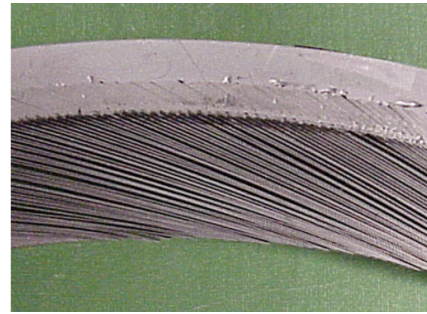
Figure 2.3: Leaf seal packing, Franceschini (2005)

Fig 2.1, 2.2 and 2.3 comprise a schematic diagram and pictures of brush and leaf seals, showing the main geometric variables that are available in the design of a typical

seal. Leaves and bristles, here termed as filaments, are generally welded to the outer ring. The number of compliant elements is in the order of 10^5 (brush seal) and $4 \cdot 10^3$ (leaf seal). Common values of the axial width of the bristle and leaf packs are 1-2 mm and 4-5 mm respectively. In brush seals, the bristles are mostly distributed into a hexagonal close-packed arrangement, as shown in Fig 2.2. For bristles made of a metallic material, wire diameters are around 100 μm , while bristle-bristle gaps are of the order of 1 μm .



(a) Sketch



(b) Photograph of involute prototype

Figure 2.4: *Involute leaf design, Jahn (2008)*

Besides the simplest straight leaf design, different configurations have been proposed to arrange the plates (leaves) that form a leaf seal. In the involute leaf design, the leaves are curved radially inwards in such a way that contact only exists at the leaf tips, while close to the roots, gaps become larger improving the compliance of the seal (see Fig 2.4). Contact between leaves reduces the area available for flow passage and creates considerable damping, thereby limiting vibrations. Leaves with variable thickness have also been employed, where the effect previously described is achieved by thickening the leaves in the section near the tip. The choice of the leaf arrangement is strongly influenced by the radius of the sealing location, and affects the performance of the seal in terms of leakage, differential pressure capability, and leaf vibrations and buckling. Normally, the average gap between leaves is close to 10 μm , and the thickness of the leaves is 100 μm (see Fig 2.3). The distance from the backing ring or downstream coverplate to the rotating surface (“fence-height”) is given by the maximum expected radial growth or eccentric movement of the rotor. A value between 1-3 mm is frequently

employed for this clearance. To increase the compliance of the seal to rotor radial movements, leaves and bristles are inclined to the radial direction forming a typical lay angle between $30\text{-}60^\circ$. The length of the filaments is of the order of 10 mm. Super alloys such as Haynes 25 are commonly used to manufacture bristles and leaves, although non-metallic materials such as aramid fibres have also been employed in brush seals. Wear and frictional heat generation are strongly dependent on the friction coefficient of the tribological pair formed by the rotor and filament materials.

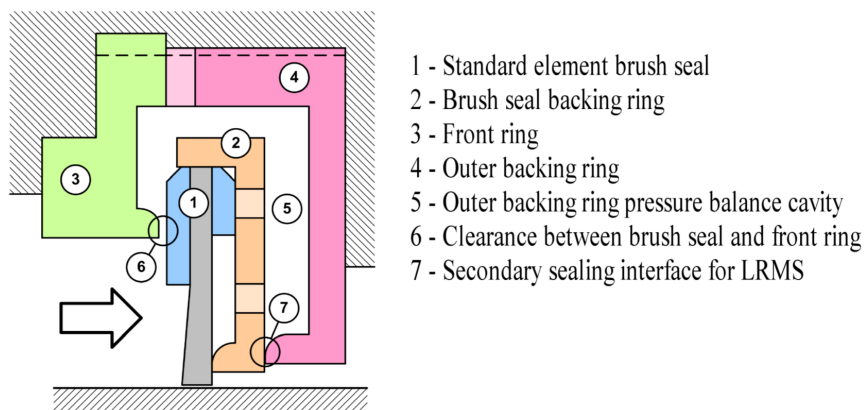


Figure 2.5: Schematic representation of a LRM brush seal with a pressure-balance pocket , Franceschini (2005)

The number of filaments, packing density, and bristle/leaf cross-sectional area influence the overall stiffness of the seal. Seal stiffness is defined as the force exerted by the bristles or leaves when compressed by a radial displacement of the rotor into the filament pack. A key feature of brush and leaf seals is their ability to exhibit a compliant behaviour with the rotor. If the rotor surface moves towards the seal, the filaments must deflect to accommodate the change, but also recover to the initial position when this disappears. A large value of seal stiffness limits the likelihood of seal filaments bending under coverplates, however, the resulting high contact forces increase wear and heat generation. Contrarily, the filaments of a soft seal with a low value of stiffness are easily blown down to the rotor, but also suffer from bristle lock up if pressed away from the rotor, leaving a gap that increases leakage drastically. One of the advantages of leaf seals over brush seals is their similar leakage performance at a lower stiffness (Jahn (2008)),

demonstrated experimentally in Fig 2.6. The large radial movement brush seal (LRMS, see Fig 2.5) was also introduced to reduce the stiffness of conventional brush seals (see Fig 2.7). In such a seal, the bristles are mounted in an outer ring that is capable of moving relative to the static seal housing, which contributes together with the bristle pack in the absorption of large rotor eccentricities (Franceschini (2005)).

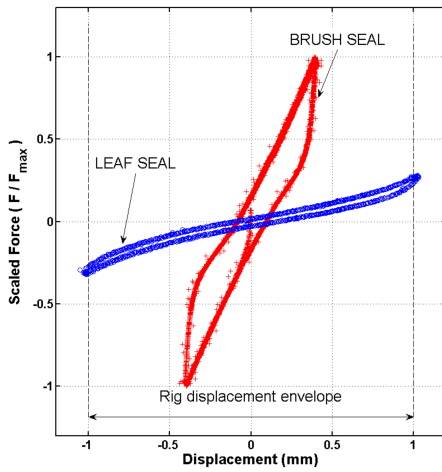


Figure 2.6: Comparison in stiffness between a leaf seal (blue) and a brush seal (red), Ingo et al. (2008)

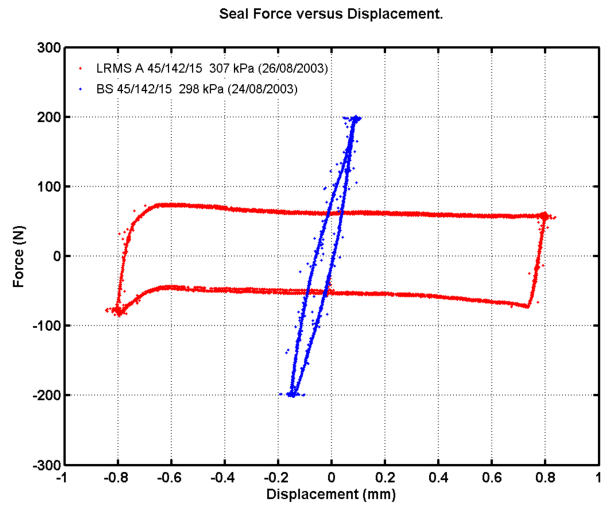


Figure 2.7: Comparison in stiffness between large radial movement brush seal (LRMS, red) and conventional brush seal (BS, blue), Franceschini (2005)

The dynamics of brush and leaf seals under pressurization must be carefully considered in any attempt to model their thermal behaviour. When these seals are subjected to a differential pressure, the resulting pressure field and leakage flow act on the flexible filaments changing their nominal design positions. In brush seals, the rise in differential pressure reduces the inter-bristle spacing and bristle pack porosity. This effect, known as pressure stiffening, greatly increases the stiffness of the seal and heat generation, and impairs its compliance. In addition, the axial forces exerted on the bristle pack once a pressure drop is established, bend the bristles downwards underneath the backing ring. As a result, contact between the last row of bristles and the backing ring occurs. The frictional force between bristles and backing ring can prevent bristles from moving radially inwards after a transient radial movement of the rotor, so that these remain

engaged to the backing ring (bristle lock-up), allowing a clearance to open (Franceschini (2005)). As discussed, large differential pressures and displacements of the rotor surface can introduce a significant deformation on the bristle pack. If the bristles are not able to recover and return to their original positions, then the behaviour of the seal will depend on past history, which is known as brush seal hysteresis. To overcome these issues, pressure balanced brush seals were developed (see Fig 2.5). By introducing a pressure cavity into the backing ring, the net axial force pushing the bristles downwards is reduced and the seal becomes more compliant (a detailed explanation of the operation of such seals can be found in Franceschini (2005)). A close rotor follow-up is achieved in this way without having a too soft seal. The inherent higher axial rigidity of the plates of a leaf seal makes them less prone to pressure stiffening, lock-up and hysteresis. Leaf seals can cope with much higher pressure drops without sustaining permanent deformation. However, leaf seals may suffer from serious problems of vibrations and flutter induced by the aerodynamics forces exerted by the leakage flow on their more extensive surfaces, and require excellent knowledge of the life cycle growth of the seal environment to allow their design (*private communications with Rolls Royce Plc, February 2016*).

In addition to the mechanical filament stiffness forces, there are two other important contributions to consider: the aerodynamic forces generated by the leakage flow and the hydrodynamic lift forces that can exist at the filament tips. The changes in momentum and the radial pressure gradients exert considerable load on the filaments. Generally, brush and leaf seals are designed in such a way that, when pressurized, the net effect is a displacement of the filaments towards the rotor surface, enhancing the compliance of the seal and its sealing performance at the expense of higher contact forces and heat generation. This phenomenon is known as blow-down.

The front and backing ring configuration of the brush seal generates a pressure gradient that leads to a convergence in the streamlines of the leakage flow, as shown in Fig 2.8a. Since the bristles are inclined to the radial direction and have a finite thickness, the turning of the flow radially inwards and the moment created due to a mismatch in pressure at points located equidistant from the filament root (see Fig 2.8b) produce the

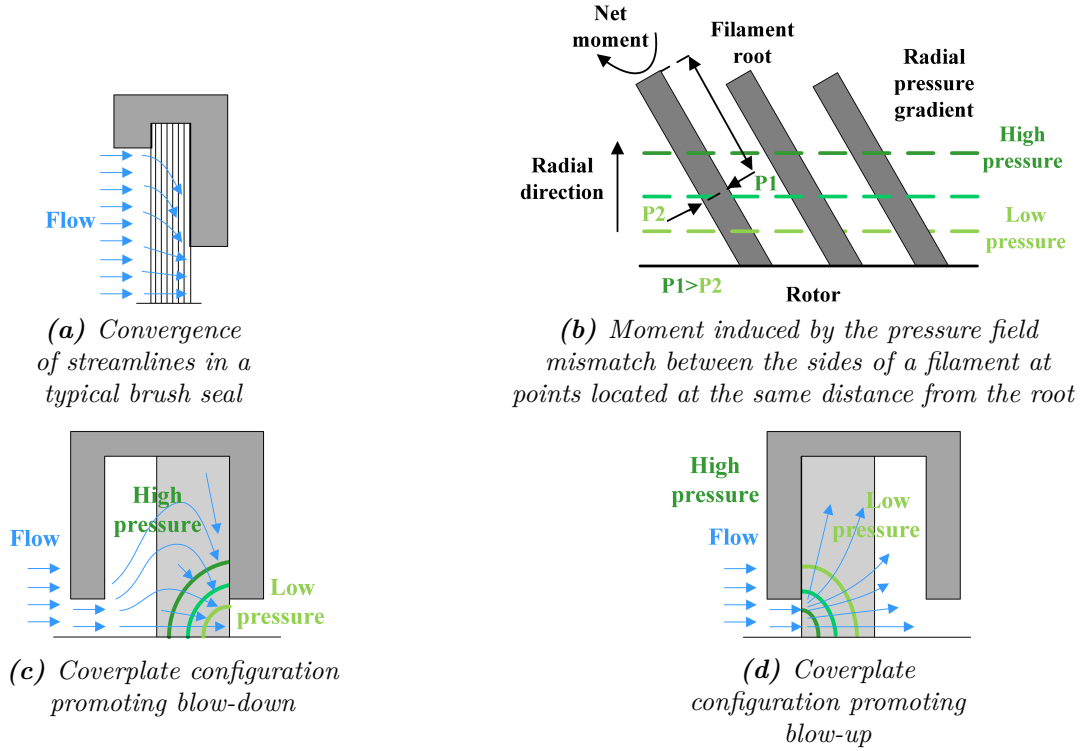


Figure 2.8: Schematic representation of blow-down phenomenon in brush and leaf seals

deflection of the bristles downwards onto the rotating surface.

Similarly, the use of coverplates upstream and downstream of the leaf pack not only increases the pressure drop across the seal, but also allows radially inwards (blow-down) and radially outwards (blow-up) forces to be induced on the leaves (Franceschini et al. (2010)). The coverplate to leaf pack gaps largely govern the aerodynamics of the leakage flow. As a rule of thumb, an upstream coverplate gap larger than the downstream one (Fig 2.8c) results in an aerodynamic force that pushes the leaves towards the rotor (blow-down). Contrarily, if the downstream coverplate gap is larger than the upstream one (Fig 2.8d), the leaves tend to lift (blow-up).

The third type of forces acting on brush and leaf seals are hydrodynamic forces that may develop at the interface between the filament tips and the rotating surface. Under certain conditions, small amounts of gas can be confined between the filament tips and the rotating surface, as show in Fig 2.9a. The resulting pressure field can then generate a net lift force that pushes the filaments away from the rotating surface (Jahn et al.

(2013)). In order to create a substantial hydrodynamic lift force, the height of the tip-rotor gap must be below $2\text{ }\mu\text{m}$; larger values result in an abrupt drop in lift force. In addition, this phenomenon only takes place at very high rotor surface speeds (see Fig 2.9b).

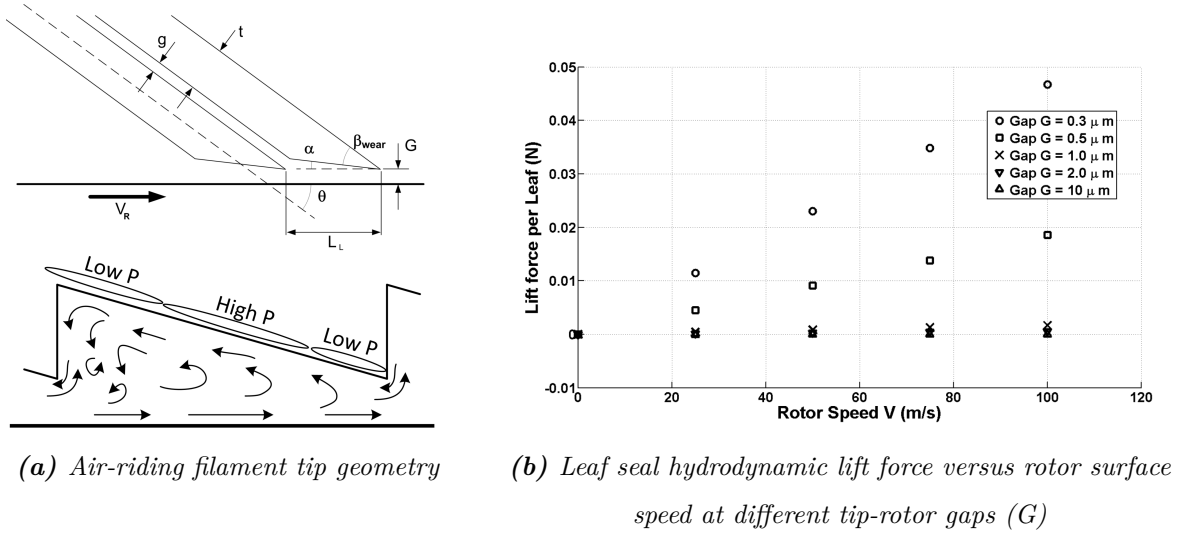


Figure 2.9: Hydrodynamic air-riding in leaf seals, Jahn et al. (2013)

If the hydrodynamic lift forces exceed the stiffness and aerodynamic blow-down forces, then a tiny gap opens and a phenomenon known as air-riding occurs. Further increments in the tip-rotor gap lead to a quickly decay on hydrodynamic lift, so this mechanism is self-regulating. This mode of operation is optimum since prevents seal wear and frictional heat generation without incurring in an excessive increment in leakage. Both brush and leaf seals can potentially exhibit hydrodynamic air-riding, as previous experimental experiences have suggested. Although leaf seals offer a higher capability to air-ride, the addition of tilted pads attached to the bristle tips have been attempted in the past to increase hydrodynamic lift in brush seals (see Fig 2.10, Justak (2002)). These elements however are subjected to considerable axial drag. Nevertheless, this phenomenon is currently not fully established and air-riding cannot be ensured for all the operating conditions experienced by the seal throughout the engine cycle. The study of brush and leaf seals operating in air-riding mode falls outside the scope of the thesis.

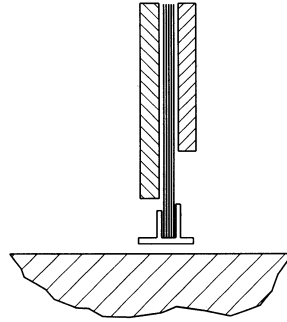


Figure 2.10: Shoed brush seal , Justak (2002)

The highly dynamic behaviour of brush and leaf seals previously described and the strong interaction between seal aerodynamics and solid mechanics enormously complicates the modelling of the filament pack geometry and flow path. Furthermore, the different length scales involved in the problem vary by many orders of magnitude; the length of the filaments may be 10 mm, the bristle diameter or leaf thickness around 100 μm and the inter-filament spacing of the order of 1 μm . This greatly increases the complexity of the grids required in numerical models and the time for computing the solution. As explained, the microscopic spacing between filaments is altered when pressure is applied, resulting in continuous inter-filament contact and complex flow path geometries. Therefore, the validity of conventional heat transfer correlations for internal flows is questionable, given the difficulty in finding an analogous cross-sectional shape or flow channel. Also, the definition of the characteristic length scale (or hydraulic diameter), and reference flow velocity and density required for the definition of the Reynolds and Nusselt numbers is not evident, especially in presence of strong pressure gradients and flow acceleration. Similarly, the choice of an appropriate gas reference temperature for the use of convective heat transfer coefficients is problematic. Whether the laws that govern the flow dynamics and heat transfer in channels of macroscopic dimensions still hold at the microscale of the inter-filament flow passage remains as a fundamental question. For instance, previous experimental works on microtubes (~ 10 μm in diameter) have shown larger heat transfer rates than those expected from clas-

sical theory (Garimella and Sobhan (2003)). Finally, although brush and leaf seals are nominally axisymmetric, their operation is very sensitive to certain geometric variables so that deviations from nominal values due to manufacturing tolerances can cause that, in practice, seal behaviour varies circumferentially.

To conclude this section, a brief discussion on the nature of the flow through brush and leaf seals is provided. The characteristics of the flow upstream of the seal will be determined by the engine hardware existing in this region. Assuming that it consists of a plain annular passage comprised between a rotating shaft and a static casing (see Fig 2.11), pressure gradients and losses may be expected to be negligible. Therefore, the flow is unlikely to present a high axial or radial velocity components. However, due to the rotation of the shaft, a significant swirl in the circumferential direction can be induced into the flow. This swirling motion is thus the main mechanism that governs convective heat transfer away from the seal inlet. As the flow approaches the filament pack, the reduction in cross-sectional area introduced by the front plate or upstream coverplate generates a local acceleration and a radial turning of the flow. These plates represent an additional flow restriction and cause a small pressure drop in the flow. As discussed before, the plate configuration can be manipulated to influence the flow passing through the seal, improving the blow-up or blow-down of the filaments. The pressure drop exerted by the seal mostly takes place within the filament pack, especially in the last rows of bristles or rear of the leaves. Therefore, strong gradients in pressure exists across the filament pack, which results in a great acceleration of the flow. However, despite the high flow velocity and inertia, if the Reynolds number is calculated based on the inter-filament spacing ($\sim 10\text{ }\mu\text{m}$) and average velocity, values around 100 are obtained. The term average velocity corresponds to the value obtained considering the actual leakage mass flow rate and the total cross-sectional area available. This value is far below conventional Reynolds numbers at which transition from laminar to turbulent conditions occurs (~ 2000 for pipe flow). In consequence, laminar flow theory has commonly been employed in the past to study the inter-filament flow. In leaf seals, the presence of leaves prevents any flow in the circumferential direction, so the swirl

acquired upstream of the seal is removed. Similarly, the absence of significant pressure gradients in the circumferential direction, and the blockage introduced by the bristles, means that brush seals also straighten the flow.

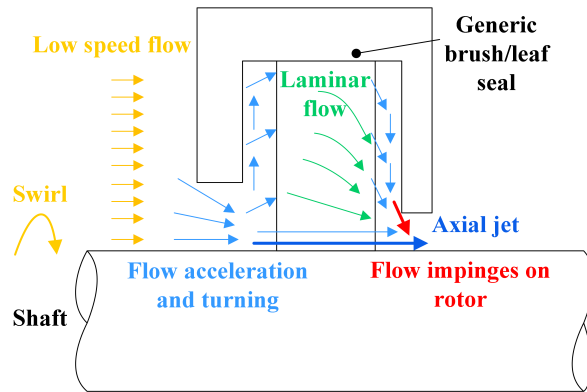


Figure 2.11: Schematic representation of the flow through a generic brush/leaf seal

As shown schematically in Fig 2.11, the flow leaving the filament pack presents two important features. First, there is a high velocity axial jet that flows parallel to the rotor surface. Second, a stream moves radially inwards impinging on the rotor surface in the first millimetres downstream of the filament pack. Due to the high velocity of the flow, a high degree of turbulence is expected. The large variations in velocity can also lead to local drops in the flow static temperature. The fast axial and radial motion of the flow largely determine convective heat transfer in this region, while the effect of the swirl induced by the shaft rotation is less important. The characteristics of the exit flow need to be considered when integrating the seal into the engine system, since they may affect downstream components.

2.2 Previous Investigations on Brush and Leaf Seal Heat Transfer

2.2.1 Overview

As explained in Chapter 1, the objective of the present research programme is the study of heat transfer in brush and leaf seal applications. This is mainly governed by the heat generation at the seal-rotor contact interface caused by high speed rotary friction. The blow-down effect, where the seal filaments deflect radially inwards due to the aerodynamic forces caused by the leakage flow, enables brush and leaf seals to sustain contact with the rotor and hence achieve exceptionally low leakage at the different operating conditions of the engine cycle. However, large filament-tip forces result in high contact pressures, and thus, increased frictional heating. An excessive temperature rise can lead to a rapid degradation in sealing performance, higher wear rates, or even a catastrophic failure of the installation. In addition, when these seals are installed in the secondary air system of a gas turbine, a rise in the airflow temperature reduces its downstream cooling capacity. An accurate prediction of the seal operating temperature is essential to ensure the correct functioning of the engine system.

The heat generated by friction is transferred to both rubbing bodies in a proportion that depends on their relative capacities at removing heat from the contact interface. Bristles and leaves act as a very efficient heat exchanger fin, dissipating heat rapidly to the airflow as the heat is conducted into the seal. The tortuous inter-filament gas-path together with the large area exposed to the flow are expected to result in large heat transfer rates between fluid and solid. Within the filament pack, high convective heat transfer is likely to occur given the very small inter-filament passages and relatively large local velocity of the leakage flow. This partly alleviates the problem of heat generation, since at relatively high differential pressures, brush and leaf seals tend to dissipate efficiently the heat that they generate. Understanding which seal design features enhance heat transfer and contribute to maintaining low contact temperatures is therefore of

paramount importance.

In the next section, the main research work carried out in the past on the thermal aspects of leaf and brush seals is examined in detail, following a chronological and technical development order. The lessons learnt will be the basis of the work developed in this thesis. Although a large amount of work has been done on the mechanics of brush and leaf seals, there are fewer studies on the heat transfer characterisation in spite of the big impact that frictional heating has on seal life and performance.

2.2.2 Brush and leaf seal heat transfer

One of the earliest studies on brush seals relevant to their heat transfer modelling is due to Hendriks et al. (1991). In this paper, the authors develop a bulk flow model of a brush seal based on a porous medium. The model was used to predict the variation in leakage with clearance and pressure drop, and results were compared with data provided by the manufacturer. In this work frictional heat (Q_f) is calculated as:

$$Q_f = \mu_c F_c v_s \quad (2.1)$$

where μ_c is the predicted friction coefficient, F_c the contact force acting on the bristle tips and v_s the rotor surface speed. This formula was later used by many researchers to calculate the amount of heat generated by friction, assuming that all frictional work is transformed into heat.

Owen et al. (2003) at Oxford University proposed an analytical model for the heat transfer occurring in a brush seal under contact conditions. All the frictional heat was assumed to be transferred to the surrounding flow, either by convection at the rotor walls or conduction along the bristles and subsequent convection to the leakage flow. Later, Pekris (2010) extended this approach to leaf seals. Their models are illustrated in Fig 2.12. The bristle/leaf pack acts as a heat exchanger, absorbing heat from the contact interface and dissipating this energy to the leakage flow. Some heat conducts laterally along the rotor to be convectively transferred to the airflow. The contact region, where friction generates heat, was assumed to be at a particular temperature common to the

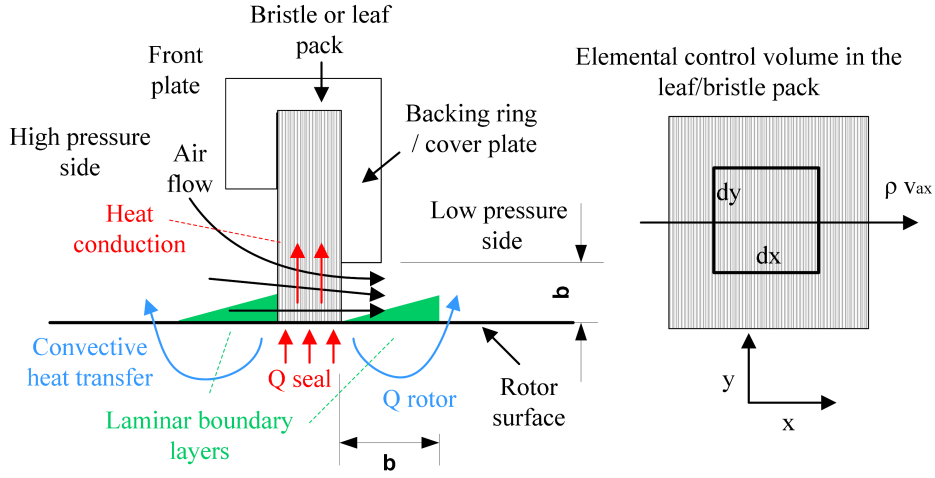


Figure 2.12: Schematic diagram of brush/leaf seal heat transfer mechanism (left), elemental control volume considered for an energy balance in the bristle/leaf pack (right), adapted from Owen et al. (2003)

rotor surface and filament tips. This contact temperature was considered to be uniform given the relatively high thermal conductivity of the rotor, which keeps lateral gradients to a minimum. Both models are directly relevant to the work presented later in this thesis and are thus described in detail below.

Given the nature of the flow through brush and leaf seals, the convective heat transfer coefficient on the filament surfaces was expected to be large. This was estimated by the authors using available correlations for the flow around an isolated cylinder (brush seal) and within a rectangular duct with an infinite aspect ratio (leaf seal). Bristle diameter and the analogous hydraulic diameter of the inter-leaf flow passage were used for the calculation of the Reynolds and Nusselt numbers. The typical values of the heat transfer coefficients thus obtained were $2500\text{--}7500\text{ W/m}^2\text{K}$ for the brush seal and $10000\text{ W/m}^2\text{K}$ for the leaf seal. In view of this highly convective environment, the authors assumed that the frictional heat absorbed by the seal is rapidly transferred to the leakage flow; with the majority of heat transfer occurring in the tip region. In the case of a brush seal, conduction along the bristle was estimated using:

$$Q_{conduction} = \frac{\pi d^2}{4 l} k \Delta T_l \quad (2.2)$$

where d , k and l are the bristle diameter, thermal conductivity and length respectively, and ΔT_l the temperature drop along the wire. Similarly, convective heat transfer was calculated by:

$$Q_{convection} = \pi d l htc \Delta T_a \quad (2.3)$$

where htc is the inter-bristle convective heat transfer coefficient and ΔT_a the difference between bristle and air temperatures. Since the heat flows from conduction and convection must be of the same order, the ratio of ΔT_l to ΔT_a was computed by the authors as:

$$\frac{\Delta T_l}{\Delta T_a} = \frac{\pi d l htc}{\frac{\pi d^2}{4} k} \approx 1400 \quad (2.4)$$

In Eq 2.4, typical values of l , d and k for a brush seal were employed. The authors argue that the temperature difference between air and bristle is much smaller than the drop along the wire. In consequence, they assume that air and metal present locally the same value of temperature. These considerations are also valid for leaf seals. The high heat transfer coefficient within the leaves justifies the assumption that local metal and air temperatures are equal, simplifying the subsequent analytical model.

An energy balance was performed in an elemental control volume of the filament pack (see Fig 2.12). The flow was considered to be solely in the axial direction and heat conduction in the radial direction. Although the flow within the bristle or leaf pack presents an important radial motion, the assumption of one dimensional axial flow is adequate in the tip region, where most of the heat transfer takes place. Applying the assumption of equal air and metal temperatures, for a brush seal, Owen et al. (2003) related the increment in air temperature as it flows through the bristle pack to the radial solid conduction along the wires by:

$$\frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) \delta y = \frac{\partial}{\partial x} (\rho v_{ax} c_p T \delta y) \delta x \quad (2.5)$$

where ρ , c_p , T and v_{ax} are the density, heat capacity, temperature and axial velocity of the leakage flow respectively. Pekris (2010) obtained a similar expression for the case of a leaf seal. These equations were solved using a Laplace transform. A heat input

was imposed as boundary condition at the filament tips, while a zero temperature rise referred to the inlet gas temperature was applied to the roots. As explained before, all the frictional heat conducted into the bristle/leaf pack was assumed to be transferred to the leakage flow before reaching the outer radius of the filaments. The solution was integrated along the axial length of the bristle and leaf packs. The resulting total heat absorbed by the seal, Q , is given by:

$$Q_{brush} = 2 \sqrt{\frac{k c_p L D}{b}} \dot{m}^{1/2} (T_c - T_{gas}) = K_{seal} \dot{m}^{1/2} (T_c - T_{gas}) \quad (2.6)$$

$$Q_{leaf} = \frac{3}{4} \sqrt{\frac{k c_p N L \pi \delta}{b}} \dot{m}^{1/2} (T_c - T_{gas}) = K_{seal} \dot{m}^{1/2} (T_c - T_{gas}) \quad (2.7)$$

where N is the number of leaves, L the axial width of the filament pack, δ the leaf thickness, b the backing ring clearance and D the diameter of the seal at the filament tips. All these parameters are lumped into a constant (K_{seal}) that is an intrinsic characteristic of the seal geometry and does not depend on the operating conditions. In the above equations, the leakage mass flow rate is represented by $\dot{m}$, while T_c and T_{gas} stand for the seal-rotor contact temperature and upstream gas temperature respectively. The functional dependency suggested in Eqs 2.6 and 2.7 ($Q \propto \dot{m}^{1/2} \Delta T$) was confirmed by Pekris (2010) using dimensional analysis.

Eq 2.6 and 2.7 correspond exclusively to the amount of frictional heat that is transferred to the seal. In addition, part of the frictional heat is conducted into the rotor. The authors assumed that this portion is entirely convected to the flow in the close vicinity of the seal. The heat transferred to the airflow from the rotor walls was estimated using a standard correlation for a laminar boundary layer on a flat plate. This convective heat transfer was assumed to occur over a distance equal to the backing ring or coverplate height, as shown in Fig 2.12. Including both upstream and downstream regions, the heat convected to the airflow from the rotor was calculated as:

$$Q_{rotor} = 1.66 (\mu D \pi)^{1/2} c_p \dot{m}^{1/2} (T_c - T_{gas}) \quad (2.8)$$

where μ is the gas viscosity. The fractions of heat conducted into the seal and into the rotor were added to compute the total frictional heat generated. For the brush seal geometry studied by Owen et al. (2003), and according to the analytical model, the frictional power was found to be related with leakage mass flow rate and contact temperature rise by:

$$Q_{total} = Q_{brush} + Q_{rotor} = (161.7 + 7.6) \dot{m}^{1/2} (T_c - T_{gas}) = 169.3 \dot{m}^{1/2} (T_c - T_{gas}) \quad (2.9)$$

It can be seen that the amount of heat absorbed by the seal is much larger than the rotor term. Eq 2.9 was compared with experimental data obtained using the Oxford Engine Seal Test Facility. This comprises a rotor running in contact with a pair of identical brush seals immersed in a pressurized cavity. The rig was operated at representative engine differential pressures and surface speeds, which resulted in typical frictional powers of up to 3 kW. Infrared detectors were used to measure the temperature of the rotor close to the contact interface with the seal. A more detailed description of the test facility is provided in Chapter 6, since this rig is also used in the present programme.

Three brush seal geometries were tested by Owen et al. (2003) at different rotor eccentricities and backing ring configurations. In these experiments, the rotor was accelerated to approximately 7000 RPM; then a seal differential pressure was applied. The rotor was decelerated due to the resulting friction torque with the seals and the windage and bearing torques. The seal-induced friction torque was calculated from the measured total torque discounting any parasitic losses. Frictional heat generation at the seal-rotor contact interface was derived from the product of rotational speed and seal torque. During the deceleration of the rotor, time instants were identified where the system was in thermal equilibrium and the rotor metal temperature gradients were small. Under these conditions, all the heat generated was convected into the airstream, being the analytical model applicable. The heat transfer data measured at these time instants for the different brush seal geometries and configurations tested are shown in Fig 2.13. The authors performed linear regression through the experimental data to find

the value of the gradient of the best linear fit. Although the experimental constant that relates total frictional power with $\dot{m}^{1/2}$ and $(T_c - T_{gas})$ was found to be different from the theoretical value (85 versus 169.3 respectively), the distribution of the data agreed with the linear dependency predicted with the analytical model. The discrepancy between these two values was explained by the authors due to the uncertainty in the values of the seal parameters and the simplifications adopted in the model. Moreover, Pekris (2010) carried out further testing of brush seals using the same test facility. The results obtained also showed a strong linear relationship between total frictional heat and the term $\dot{m}^{1/2}(T_c - T_{gas})$.

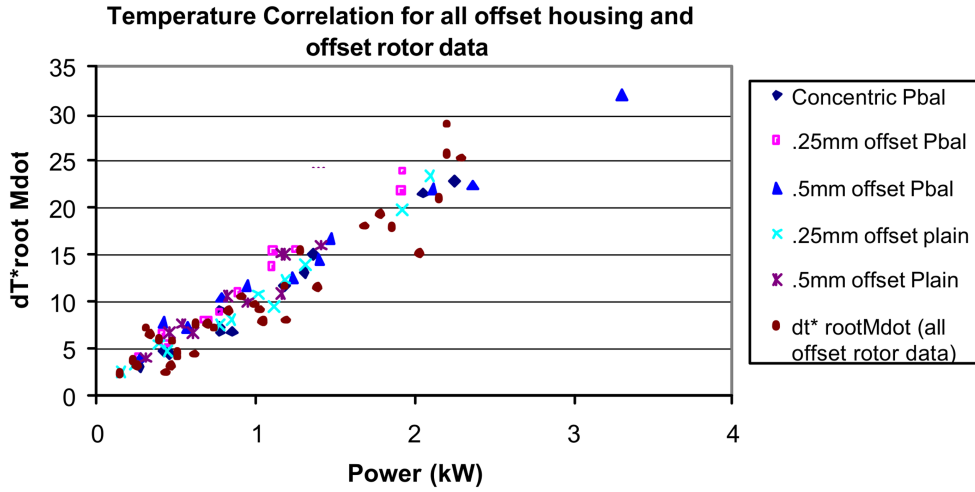


Figure 2.13: Experimental data for brush seal thermal correlation, Owen et al. (2003)

Fig 2.14 shows the experimental data obtained by Pekris et al. (2016) for a leaf seal tested in the Engine Seal Test Facility using the same methodology. In this case, the data also collapse very well to give a positive correlation between the derived model variables. The application of the analytical model to the leaf seal geometry investigated gave:

$$Q_{total} = Q_{leaf} + Q_{rotor} = (119 + 16) \dot{m}^{1/2} (T_c - T_{gas}) = 135 \dot{m}^{1/2} (T_c - T_{gas}) \quad (2.10)$$

As in Owen et al. (2003), the leaf seal contribution is larger than that of the rotor. The data shown in Fig 2.14 correspond to the seal pair tested; the value of the constant

for a single seal is calculated dividing by $\sqrt{2}$, which gives $158 \text{ m}^2 \text{kg}^{0.5} \text{s}^{-2.5} \text{K}^{-1}$. This is very close to the theoretical value of $135 \text{ m}^2 \text{kg}^{0.5} \text{s}^{-2.5} \text{K}^{-1}$ predicted by the authors.

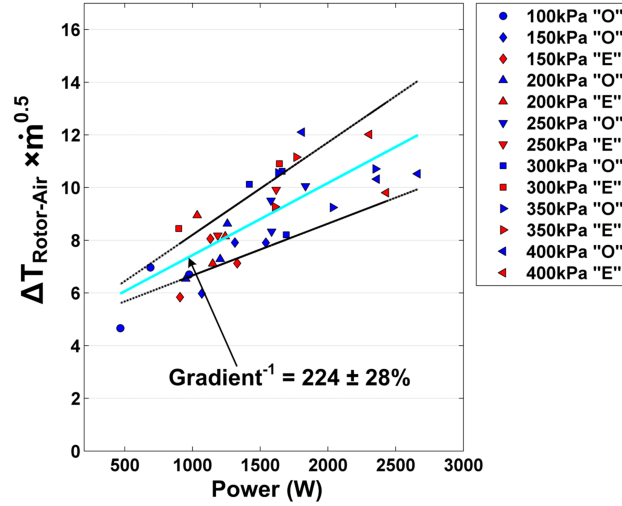


Figure 2.14: Experimental data for leaf seal thermal correlation, Pekris et al. (2016)

As a summary of these works, it can be concluded that the functional dependency expressed in Eq 2.6 and 2.7 proves to be a robust method for correlating heat transfer data of brush and leaf seals in the contacting regime. In addition, Eq 2.6 and 2.7 allow the identification of those seal parameters than mainly affect heat transfer, which provides a design criterion to minimize the resulting temperature rise. However, the major deficiency of these models is the calculation of the rotor heat transfer. According to the description of the flow through the seal presented in Section 2.1, the assumption of laminar heat transfer on the downstream side may not be entirely appropriate, resulting in an under-prediction of the rotor term (Q_{rotor}). In these analyses, an arbitrary boundary layer height and extent has been assumed. In both cases, the boundary layer height is taken as the backing ring height, although there is not a real physical basis behind this assumption. In addition, if not all the frictional heat transferred to the rotor is convected back to the airflow, then the material and geometric properties of the rotor and its surrounding boundary conditions must be accounted for.

Chew and Guardino (2004) studied the flow between the bristle tips and the rotor of an idealized brush seal using CFD simulations. The bristle tip surfaces were inclined

relative to the rotor to investigate the resulting lift forces and aerodynamic air-riding effect. Fig 2.15 shows the solution domain investigated by the authors, which is confined to the region of the bristle tips. Each bristle was modelled as a circular straight tube inclined to the radial direction. Four hexagonal or rectangular packed bristles were considered. An angle between the bristle tip and rotor was introduced to study its effect on the lift exerted on the filaments. Fully compressible flow solutions with solid heat conduction modelling in the bristles were obtained.

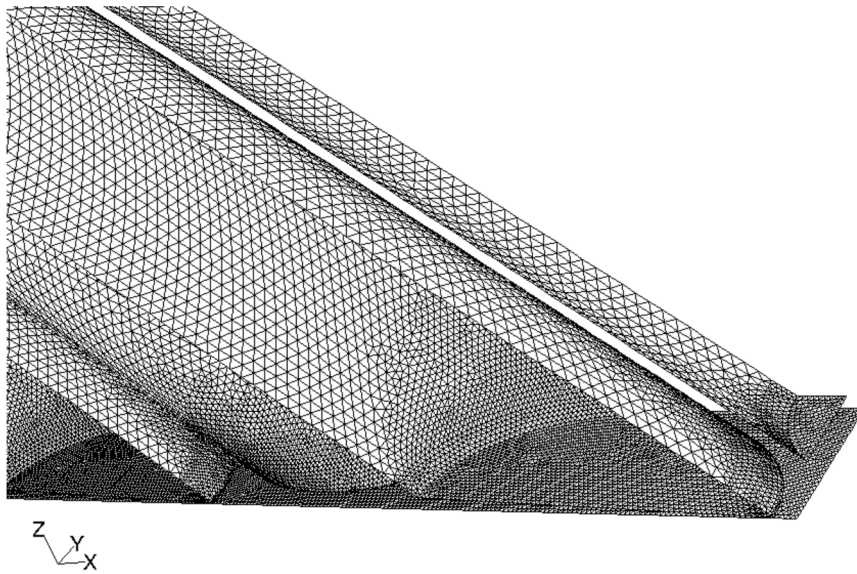


Figure 2.15: *CFD domain considered in Chew and Guardino (2004), comprising the bristle tip region and the rotor surface*

The authors included the effect of heat generation due to contact friction and heat conduction through the bristles. In the CFD model, frictional heat was modelled as a uniformly distributed heat source in the bristle tip. Heat generation was calculated as the product of the normal contact force, the coefficient of friction (assumed equal to 0.3) and the rotor surface speed. At the bristle top surfaces, two different boundary conditions were employed: adiabatic wall and a constant temperature equal to the inlet flow temperature. Results showed the importance of heat conduction in the bristles as an effective mechanism to dissipate the frictional heat. The authors recognise that in practice, the problem is further complicated due to the presence of heat transfer to the

rotor and conduction between contacting bristles. Lelli et al. (2006), extended modelling capabilities to include coupling between the fluid flow and mechanical movement of bristles, allowing bristle clumping to be predicted for the first time.

Another attempt for an analytical model of heat transfer in leaf seals was presented in Franceschini (2005). This author developed a two-dimensional flow model to compute leakage and aerodynamic loading on the leaves. The flow upstream and downstream of the pack was approximated from considerations of inviscid flow while under and between the leaves a simplification of the Reynolds' equation was employed. To extend the use of the mechanical model to heat transfer studies, the author initially considers modelling the leaf seal as a group of fins. The development of a two-dimensional (axial and radial) leaf conduction model running in parallel with the flow model is discussed by the author. The main limitation of this approach is the requirement of a convective heat transfer coefficient distribution over the leaf surface to couple the flow and solid conduction models. The work presented in Franceschini (2005) is merely a proposal, no further implementation or validation was carried out with the thermal model envisaged.

Dogu and Aksit (2006) investigated the temperature distribution in a brush seal as a function of frictional heat generation at the bristle tip. Two approaches were used to that end. First, the seal was modelled using a two-dimensional axisymmetric CFD model which treats the bristle pack as a porous medium. Second, an analytical analysis was performed where a single bristle is modelled as a fin. For a brush seal operating in contact with the rotor, the authors describe a heat transfer mechanism (shown in Fig 2.16) similar to that discussed in Owen et al. (2003). Frictional heat was calculated as the product of the friction coefficient, the bristle contact load and the rotor surface speed, as in Hendriks et al. (1991). A value of 0.3 was chosen by the authors as a typical friction coefficient of the Haynes 25 alloy. Overall seal stiffness from published experimental data was employed to estimate typical values of contact pressure. Limit values for the seal operating conditions were employed to identify the minimum and maximum heat generation levels that may exist in common applications. This resulted in lower and upper bounds of $q_{min} = 20 \text{ kW/m}^2$ and $q_{max} = 75,000 \text{ kW/m}^2$. The authors

admit the difficulty in determining how the frictional heat is shared by rotor and bristles. They assumed that half of the heat generated is transferred to the bristles, while the remaining is conducted into the rotor, which is clearly problematic.

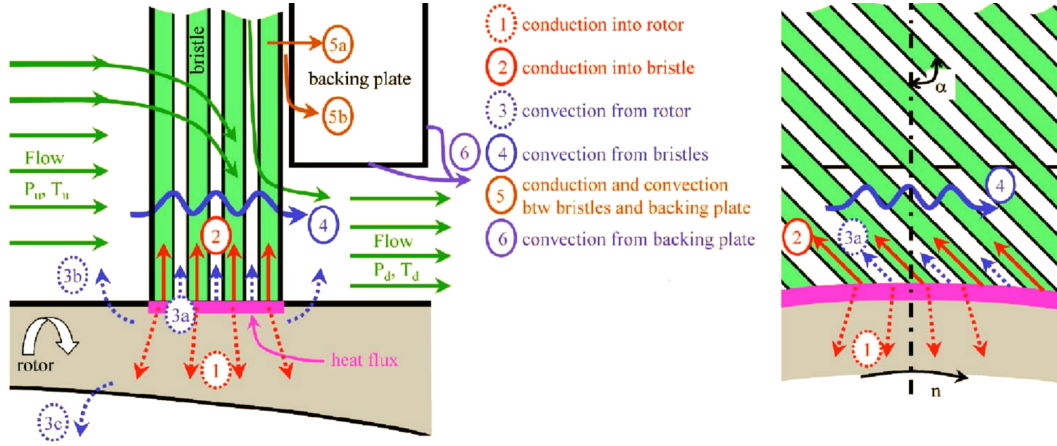


Figure 2.16: Frictional heat generation and heat flow distribution, Dogu and Aksit (2006)

The CFD domain comprises the upstream and downstream cavities, the bristle pack, and the front and backing plates. Fluid-solid interfaces were defined for the front and backing plate surfaces to account for solid conduction. The rotor geometry was not modelled, being its surface the lower boundary of the domain. The fluid zone corresponding to the bristle pack was modelled as a porous medium, which implies solving the set of Navier-Stokes equations with some modifications. Additional resistance terms were added to the momentum equations to capture the pressure drop exerted by the seal. Experimental data on seal leakage was used to derive the drag coefficients required. In the energy equation, the thermal conductivity of the bristle material (Haynes 25) was used instead of that of the gas. A heat source boundary condition was imposed at the bristle-rotor interface to simulate frictional heat. On the upper plane of the domain (the casing or stator), the wall was assumed to be adiabatic.

The bristles were also modelled by the authors as fins attached to the rotor surface. At the fin-rotor contact interface, frictional heat generation was introduced as a heat flux boundary condition. The bristle temperature rise far from the contact interface was assumed to be negligible due to the highly convective surroundings. The flow tem-

perature within the bristle pack was considered to be constant and equal to the inlet temperature, which implicitly involves the assumption of infinite heat capacity of the gas. The convective heat transfer coefficient over the fin surface was taken as constant and calculated using correlations from cross flow around cylinders and through a bank of staggered tubes. The diameter of the wire was chosen as the characteristic length for the calculation of the Reynolds and Nusselt numbers. For the range of leakage mass flow considered, values between 1400 and 2200 W/m^2K were obtained. The one-dimensional heat conduction equation of an infinitely long circular fin was solved to yield a closed-form solution for the heat flux transferred to a single bristle:

$$q_{brush} = 2 \sqrt{\frac{k}{d}} htc_{fin}^{1/2} (T_c - T_{gas}) = K'_{seal} htc_{fin}^{1/2} (T_c - T_{gas}) \quad (2.11)$$

where K'_{seal} is a constant dependent on bristle diameter (d) and thermal conductivity (k), htc_{fin} the convective heat transfer coefficient over the fin surface, and T_c and T_{gas} the temperatures of the bristle tip (contact) and of the gas upstream of the seal. This expression is analogous to the equations derived in Owen et al. (2003) and Pekris (2010) (Eq 2.6 and 2.7). The heat absorbed by the bristle pack is proportional to a constant term that is a function of some seal parameters, and the product of the temperature rise at the filament tips and the square root of a term dependent on the leakage flow characteristics. While the flow term is the mass flow rate in the case of Owen and Pekris' models, the fin model employs the convective heat transfer coefficient, which must be related to mass flow rate via Nusselt and Reynolds numbers. Typically, for flow past a cylinder, $Nu \propto Re^{0.5-0.7}$, thus the exponent on mass flow rate predicted using equation 2.11 is likely to be very much lower ($Re^{0.25-0.35}$) than for the predictions of either Owen et al. (2003) or Pekris (2010). The key difference in approach lies in the assumption of infinite flow heat capacity in the case of the fin based model.

The authors calculated the temperature distribution within the bristle pack for the range of heat flux inputs considered. Fig 2.17 shows the temperature solution obtained from four CFD cases. The authors highlight the exponential decay down to the inlet gas temperature that exists in the radial direction. Most of the temperature drop oc-

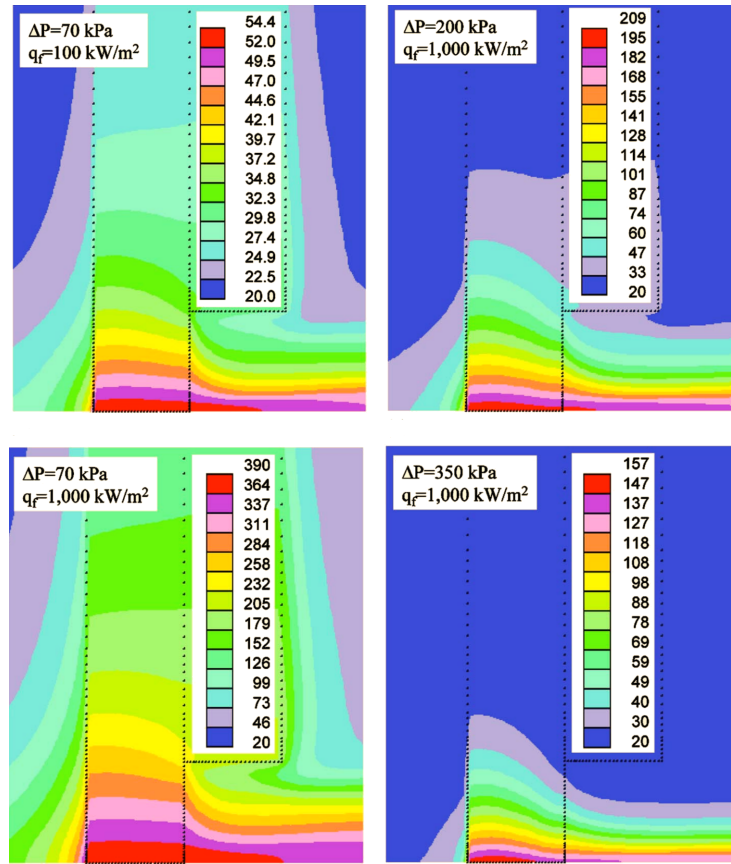


Figure 2.17: Temperature contours (in degrees Celsius), bulk porous medium approach, Dogu and Aksit (2006)

curs in the backing ring-height region for combinations of high mass flow leakages and low-moderate frictional heats. In the axial direction, the temperature gradients are negligible. For the largest frictional heats imposed, the numerical model predicted contact temperatures above 1000 °C. The results obtained with the fin analysis were compared with the CFD model. Although the analytical model predicted a faster cooling mechanism, both models presented comparable results and a similar trend.

Another study on frictional heat generation in brush seals is due to Demiroglu and Tichy (2007). As other researches in the past, frictional heat generation was calculated as the product of the normal contact force, the friction coefficient and the rotor surface speed. The authors carried out a series of friction coefficient measurement tests (not described in the paper) to derive an average value of 0.24 for the tribopair considered. They also presented a close-form equation to calculate the contact load exerted on the

rotor by the seal. Bristle tip forces were calculated using as input variables the number of bristles, rotor outer diameter, and bristle modulus of elasticity, lay angle, diameter, length and deflection in the radial direction. Notably, the formula proposed by the authors is only applicable when the seal is not pressurized. These findings were brought together resulting in an empirical equation to calculate frictional heat generation. To validate the expression derived, the authors measured the temperature rise at the seal-rotor interface in a test facility. The experimental data obtained was subsequently compared with the predictions of a FE model of the test rotor, where the frictional heat was calculated with the empirical equation that the authors suggest. The test rig consisted in a pressure vessel with a pair of brush seals housed inside and running in contact with a rotor. An electric motor coupled to the rotor allowed surface speeds up to 68 m/s. The authors employed infrared thermography to measure rotor and bristle temperatures. Fig 2.18 shows the setup of the experimental facility and the measurement target area. The test duration was 3 minutes, which was found to be sufficient to reach steady-state conditions.

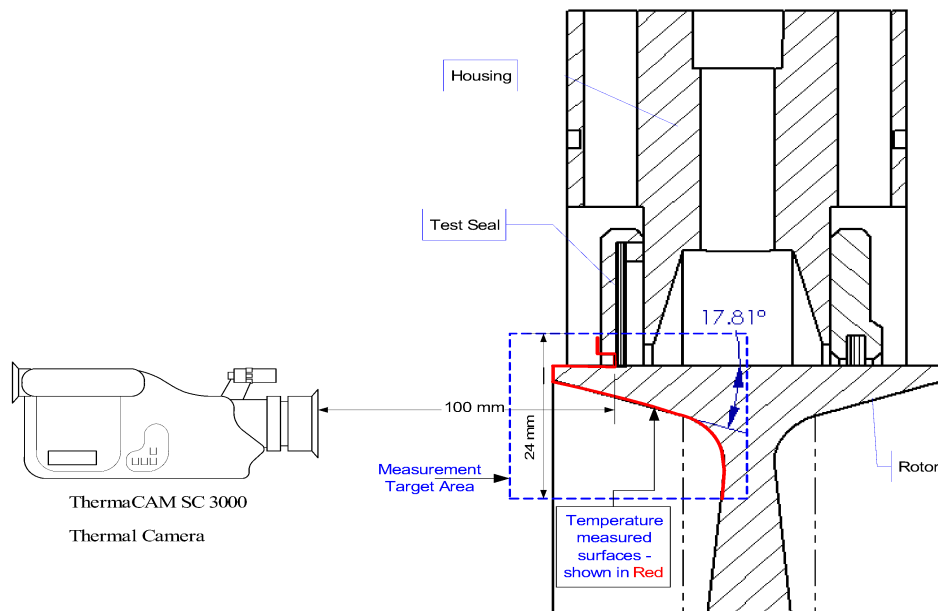


Figure 2.18: Test rig set up and measuring location, Demiroglu and Tichy (2007)

Experimental observations showed the large rotor temperature rise occurring in ab-

sence of leakage flow through the seal. Maximum temperatures of 336 °C were attained at the maximum surface speed (68 m/s) tested. This demonstrates the risk of a failure of the seal installation at those engine operating conditions where the seal differential pressure is close to zero. However, when pressure was applied to the seal, the authors found a rapid cooling of the seal-rotor interface by the axial flow. Fig 2.19 shows the temperature distribution measured with and without leakage flow for a fixed rotor speed. It can be observed the circumferential variation in temperature that exist in spite of the nominal axial symmetry of the system. Although the expected pressure stiffening and blow-down effects increase contact forces and heat generation, cool down caused by the leakage flow was observed in all tests. The authors point out the dominance of the improved cooling effect over the increment in frictional heating. Fig 2.20 illustrates the peak in rotor temperature that immediately occurred when pressure was supplied to the seals and the bristle blew-down, followed by a strong cooling event induced by the leakage flow.

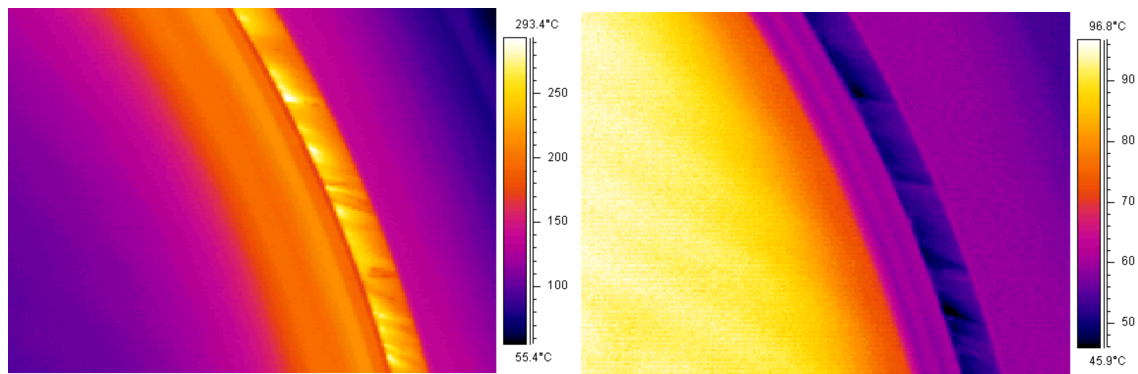


Figure 2.19: Rotor and brush seal temperature distribution measured with infrared thermography by Demiroglu and Tichy (2007). The left image corresponds with a case where no pressure was applied to the seal, while the right image shows the large cooling effect when leakage flow is introduced. In both cases, the rotor speed was the same

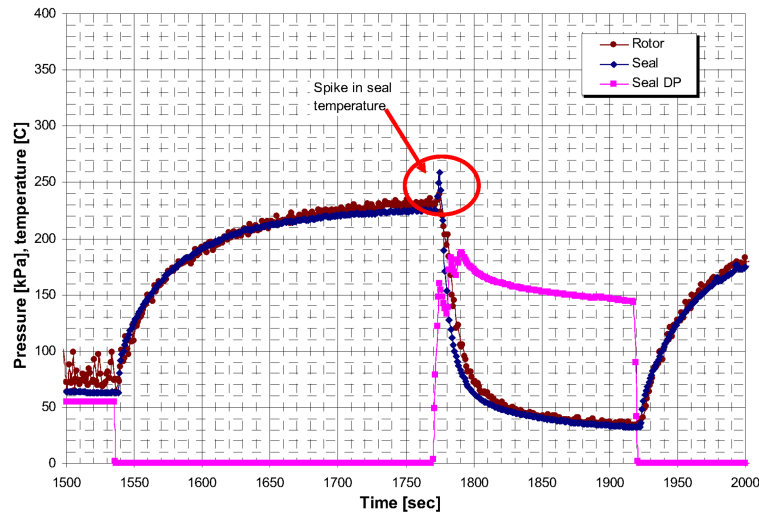


Figure 2.20: Variation of seal-rotor interface temperature with time in a typical experimental test, Demiroglu and Tichy (2007)

The experimental temperature field was compared with results from thermal FE analysis of the test rotor for the cases with no leakage flow. The expression developed by the authors was used to calculate total frictional heat generation, which was imposed in the model as a constant heat flux at the seal-rotor interface. For the cases where the seals were not pressurized, the authors argued that the partition of heat was proportional to the thermal conductivities of the material pair, assuming similar surrounding temperatures for both bodies and same area for heat conduction. Given the similarity in thermal conductivity between the bristle and rotor materials, it was considered that half of the frictional heat was transferred to the rotor. The calculation of the heat partition in presence of leakage flow was not attempted by the authors. However, they comment that it changed drastically in favour of the bristles when high pressure was delivered to the seals and the leakage flow enhanced convective heat transfer. According to Fig 2.19, the cooling of the bristle pack when the seal is pressurized is larger than in the rotor, which indicates that the seal acts as a stronger heat sink absorbing a bigger proportion of the frictional heat. The authors also assumed that the temperature at the contact interface is common to each contacting body, which was later supported by the numerical solutions. Convection boundary conditions were prescribed on the remaining rotor surfaces. The value of the convective heat transfer coefficient was estimated using cor-

relations from the literature. Results from the numerical model matched reasonable well the measured temperature distribution. Discrepancies are explained by the authors due to errors in the convection coefficients, rotor thermal conductivity and variation with operating conditions of the friction coefficient (assumed constant). The major limitation of the work presented by these authors is that its applicability only extends to operating conditions at zero seal differential pressure.

Ruggiero et al. (2009) also employed infrared thermography to investigate frictional heat generation in Kevlar (aramid) fibre brush seals. Compared with traditional metallic brush seals, bristles made of aramid fibres present a smaller wire diameter offering tighter sealing capability. They also present higher flexibility, which results in a more compliant behaviour that encourages the blow-down of the bristle pack and a rapid recovery in the event of a transient radial displacement of the rotor. The authors highlight the importance of the choice of an adequate value for the bristle pack radial stiffness. This parameter must be large enough so that the bristles remain compliant with the rotor and can recover from variations in rotor radius without incurring in high levels of heat generation. In the paper, the authors describe a testing technique for characterizing the frictional heat generation rates produced by a Kevlar fibre brush seal.

The setup of the experimental facility was similar to that used by Demiroglu and Tichy (2007) (see Fig 2.18). A single seal was installed in contact with a rotating disc inside a pressurized cavity. The seal-rotor interface temperature was measured using an infrared camera. Tests were carried out in the absence of leakage flow, i.e. the seal differential pressure was always zero. Rotational speeds of up to 20000 RPM were attained during testing. Tests were run long enough to reach steady-state conditions. To infer the frictional heat generation characteristics of the aramid seal tested, the authors built a FE model of the test rotor. On the seal-rotor contact interface, a heat source boundary condition was imposed to simulate the frictional heat generation. The value of this heat source was iterated to match the rotor temperature field measured at steady-state. At the inner radius of the rotor, the temperature measured experimentally was applied. Convective heat transfer coefficients from existing correlations were imposed

on the rotor surfaces exposed to the surrounding air. The good agreement between the numerical temperature solution and the experimental measurements gives confidence to the accuracy of the value derived for the rotor heat input. As expected, it was found that increasing bristle pack stiffness leads to greater heat generation rates. However, this resulted in a rapid bristle tip degradation and thus, a reduction in contact forces and less heat generation, being this mechanism self-regulating. Although the methodology described in this paper can be used to compare the level of friction that different seal designs present, it does not fully establish the total frictional heat generated. The fraction of the frictional heat transferred to the bristle pack was not accounted in the model. Therefore, the authors only estimated the heat conducted into the rotor, not the overall heat generated by the seal. In addition, heat generation and contact temperature rise were not considered for a pressurized seal.

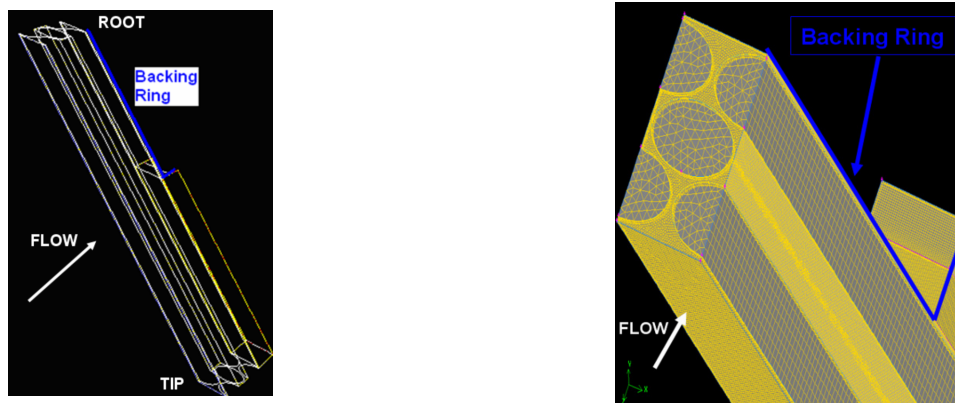


Figure 2.21: Idealised brush seal domain and mesh of CFD model, Pekris et al. (2011)

Pekris et al. (2011) described a methodology to investigate the effect of the main seal geometric variables on key performance parameters based on computational fluid dynamics (CFD) simulations using ANSYS Fluent. The inter-bristle flow was modelled using the domain shown in Fig 2.21. This comprises a single circumferential bristle row including 5 bristles in a hexagonal close-packed arrangement and an exit region under a backing ring. Total pressure was prescribed at the inlet and outlet, while periodic boundaries were imposed on the left and right sides of the domain. Heat flux boundary conditions were applied at the bristle tips and at the rotor-air interface. The value of

this heat source was obtained from experimental measurements of typical seal-induced frictional powers carried out at Oxford. It was assumed that half of the total frictional heat measured was conducted up the solid bristles and convected to the air in-between the bristles. Heat conduction through the bristles was modelled, and convection to the leakage flow enabled. At the fluid-solid interfaces, the temperatures of the air and bristles were assumed to be equal. The temperature at the bristle roots and backing ring were set equal to the inlet flow temperature (300 K). The rotor geometry was not included in the domain and a static wall boundary condition was applied to the plane of the bristle tips.

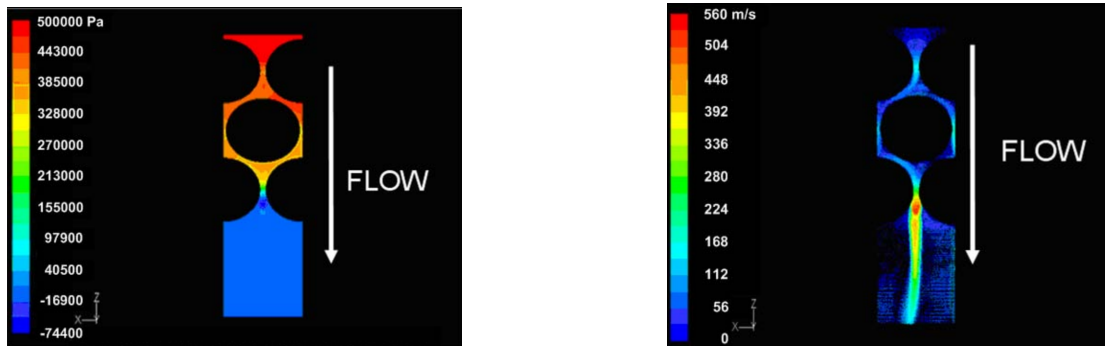


Figure 2.22: Pressure and velocity fields, Pekris et al. (2011)

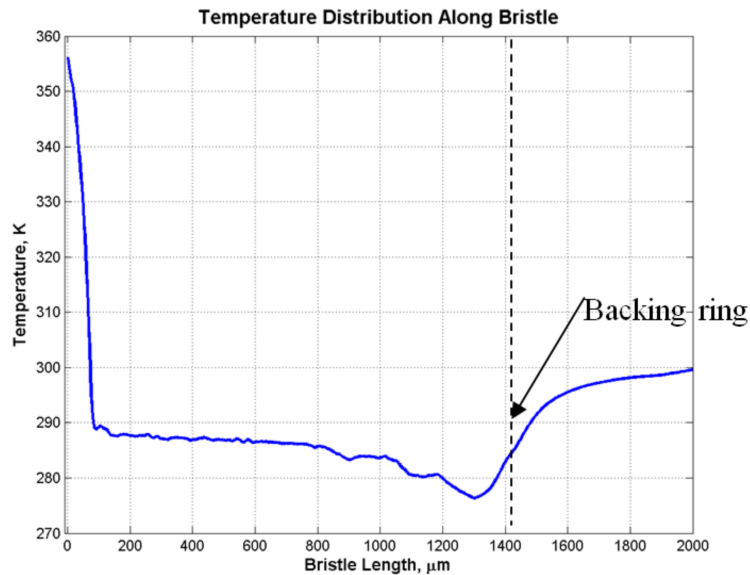


Figure 2.23: Average bristle temperature variation with radial position (last row of bristles), Pekris et al. (2011)

Given the expected low inter-bristle Reynolds number (~ 300), a laminar solver was employed by the authors to compute the flow solution. Fig 2.22 shows the typical pressure and velocity fields obtained. The largest pressure drop occurred in the last bristle, resulting in a high velocity jet downstream of the bristle pack. Flow separation from the backing-ring was also observed and a strong radial velocity component was found in the downstream region. The flow solution showed that the highest temperatures were located at the bristle tips, as expected. The variation of bristle surface temperature circumferentially indicated that the lowest values appear on the upstream side, where the flow is brought to rest allowing very effective heat transfer to occur. In the radial direction, the temperature falls rapidly with distance away from the tip. This is illustrated in Fig 2.23, which shows the variation of the circumferentially averaged bristle temperature with radial direction. In addition to provide a comprehensive discussion on the inter-bristle flow features, the authors carried out a parametric study using the CFD model. The main geometric variables of a brush seal (backing ring clearance, inter-bristle spacing, number of rows, and bristle diameter, lay angle and length) were ranged to investigate their effect on performance parameters such as bristle forces, leakage and tip temperature. They found that maximum bristle temperature presents a strong dependence on all six input variables, with bristle spacing and diameter being of highest importance. These two parameters determine the size of the inter-bristle flow passage, which governs the velocity and heat transfer levels developed.

More recently, a number of studies have been carried out that tackle the problem at a higher resolution using bigger computational domains. The next three papers model heat transfer and flow together with the mechanics of the seal filaments, with the last paper including a comparison between numerical predictions and experimental results.

Qiu and Li (2013) developed a brush seal heat transfer model combining CFD and FEA (ANSYS). The authors argued that the mechanical and aerodynamic aspects of the problem are strongly coupled and must be studied simultaneously. The leakage flow largely influences the bristle tip contact forces, which determine the amount of frictional heat. The flow through the bristle pack was simulated with a CFD model based on a

porous medium approach, in a similar manner to Dogu and Aksit (2006). The CFD domain included the geometry of the rotor, so that the frictional heat partition between seal and rotor was implicitly calculated by the solver. Conjugate heat transfer was defined at the rotor and front and backing plate walls and heat conduction within these solids modelled. Additional resistance terms, calculated from experimental leakage data, were added to the flow momentum equations to represent the pressure drop experienced by the leakage flow. The flow energy equation was supplied with an effective thermal conductivity defined as the volume average of the fluid and solid conductivities. The resulting resistance forces acting on the leakage flow were considered to be equal to the aerodynamic loads on the bristles. Consequently, these forces were applied as boundary conditions in the FE mechanical model of the bristles. The FE model was then used to calculate bristle deflections and contact forces, which were multiplied by the friction coefficient and rotor surface speed to calculate frictional heat generation. The resulting frictional heat was later applied to the CFD model as a circumferential heat source at the bristle-rotor interface. In addition, a new bristle pack geometry and CFD mesh were generated according to the predicted bristle deflections. The authors repeated this iterative procedure until the residual of bristle deflections and aerodynamic forces was less than 0.01%.

Results showed that bristle-rotor normal contact force and frictional heat generation almost linearly increased with seal differential pressure due to the pressure stiffening and blow-down mechanisms. The pressure drop was found to mostly take place at the rear of the bristle pack around the fence height region, as expected. Fig 2.24 shows a contour plot of the typical temperature field predicted by the authors and the radial temperature distributions obtained at several differential pressures. The maximum temperature obtained from the different simulations reached 377 °C at the bristle tips. The authors remark the exponential drop in bristle temperature that exists in the radial direction with increasing distance from the interface with the rotor. The radial temperature gradients were higher at the fence height region, where the pressure drop mostly occurs and the flow is strongly accelerated, resulting in a high convective cooling capability. In Fig

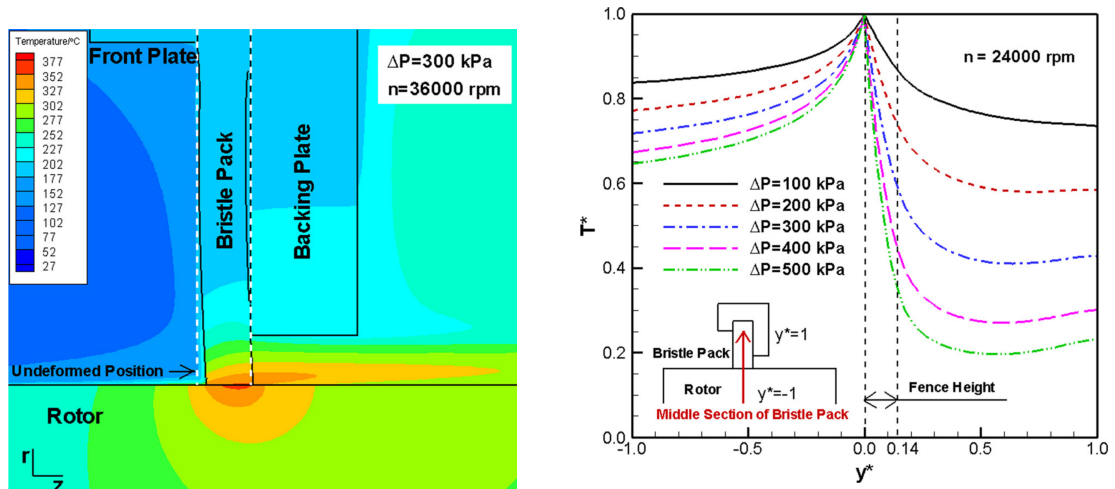


Figure 2.24: Typical temperature contours (left) and radial temperature distribution at the middle of the bristle pack versus seal differential pressure (right), Qiu and Li (2013)

2.24, it can also be observed the continuity assumed by the authors in contact temperature at the seal-rotor interface. Unfortunately, no considerations on the frictional heat partition between seal and rotor was provided by the authors; neither on the convective heat transfer characteristics of the flow upstream and downstream of the bristle pack.

As shown in Fig 2.24 (right), for the case at a rotor speed of 24000 RPM, increasing the seal differential pressure results in higher leakage flow rates and an enhancement in the cooling of the rotor and bristle pack. However, this also involves higher contact forces and further frictional heat generation. The authors found that a critical value of seal differential pressure exists below which the additional heat generation dominates the increment in heat removal capacity of the leakage flow and therefore, maximum temperatures rise. The value of the seal differential pressure at this critical point increases with rotational speed, as shown in Fig 2.25.

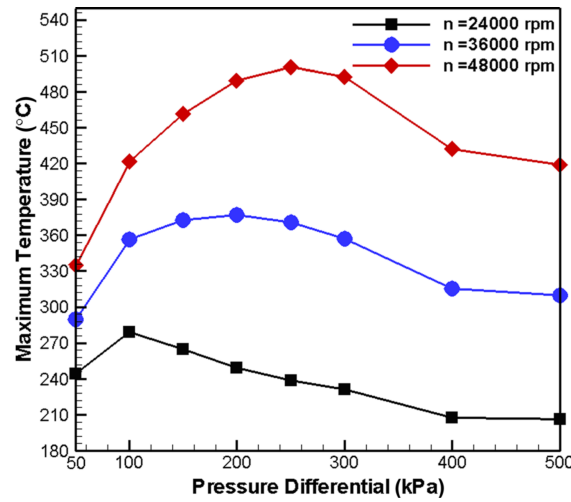


Figure 2.25: Bristle maximum temperature at various seal differential pressures and rotational speeds, Qiu and Li (2013)

Later, these authors further improved their flow model (Qiu et al. (2015)) implementing a dual energy equation to account separately for solid conduction through the bristles and heat transfer within the leakage flow. The inter-bristle flow was still modelled using the porous medium approach; however in the new model, the flow energy equation uses the thermal conductivity of the air instead of an effective bristle-air thermal conductivity. An additional energy equation for solid conduction along the bristles was added to the model. The flow and bristle energy equations were linked by a convective heat transfer term. This required as input the convective heat transfer coefficient, which was estimated by the authors using a standard experimental correlation for the flow through banks of staggered tubes. The convective heat transfer coefficient was assumed to be constant throughout the bristle pack, and for its derivation, bristle diameter and average flow velocity based on the leakage mass flow rate in the fence-height region were considered. For the brush seal geometry modelled, the authors obtained a value of heat transfer coefficient varying between 1200-1900 W/m^2K in the range of 2-4 seal pressure ratio.

Fig 2.26 shows the typical bristle and flow temperature fields obtained with this approach. It can be observed that fluid and solid temperatures are similar locally. In addition, there was a rapid decrease in flow and bristle temperatures in the radial

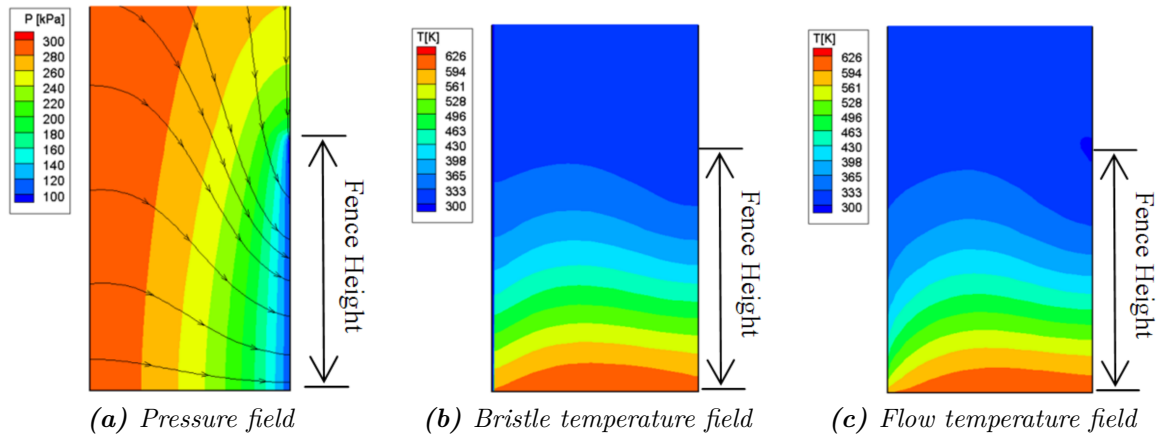


Figure 2.26: Pressure and flow and bristle temperature distributions in the bristle pack region at 9000 RPM and pressure ratio=3, Qiu et al. (2015)

direction, while the gradients were lower axially. An interesting feature found by the authors was the drop in leakage flow that occurred with increasing rotor speed. This is illustrated in Fig 2.27. They argue that increasing the rotational speed involved higher frictional heat generation rates, which resulted in an overall increment in leakage flow temperature. For a fixed seal differential pressure, this effect caused a reduction in fluid density and thus, leakage mass flow rate. This phenomenon implies that frictional heating contributes to improve the sealing performance of the brush seal. Moreover, for a fixed rotational speed, the effect of increasing the seal differential pressure was found to be a drop in the average bristle temperature but an increase in bristle maximum temperature (Fig 2.28). Although the increment in leakage flow with pressure enhanced the overall cooling of the bristle pack, the higher contact forces and frictional heat produced that locally maximum (tip) temperatures rose.

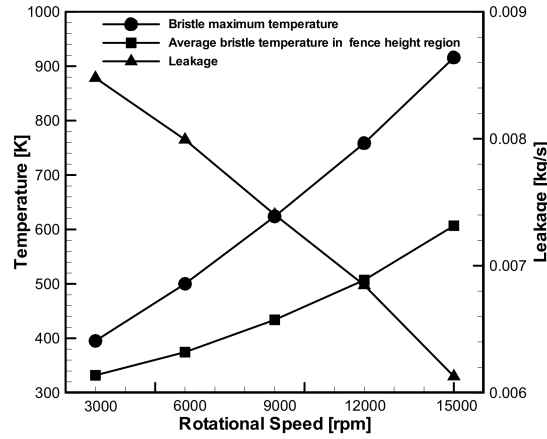


Figure 2.27: Bristle maximum and average temperatures in the fence height region versus rotational speed at pressure ratio=3, Qiu et al. (2015)

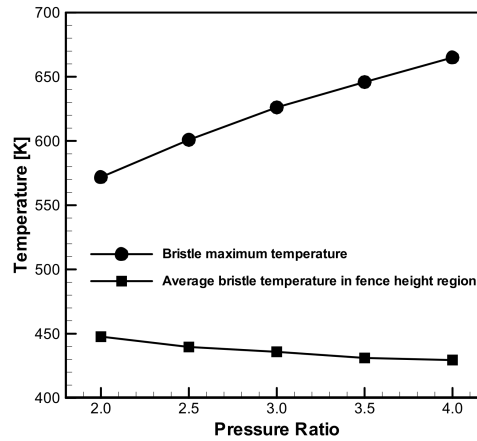


Figure 2.28: Bristle maximum and average temperatures in the fence height region versus pressure ratio at 9000 RPM, Qiu et al. (2015)

Huang et al. (2014) carried out a theoretical and experimental investigation on the mechanical and thermal characteristics of the brush seal. A schematic of the experimental setup employed by the authors is shown in Fig 2.29. The instrumentation of the test facility includes a torque sensor, pressure gauges, a flow meter and an infrared camera for the measurement of the seal-rotor interface temperature.

The operation of the brush seal installation was simulated using combined CFD and FE models (ANSYS). Fig 2.30 shows the 3D domain considered by the authors in the CFD model, which represents a small circular sector of the test brush seal. Fluid-solid interfaces were defined to solve the temperature field within the rotor and backing-ring.

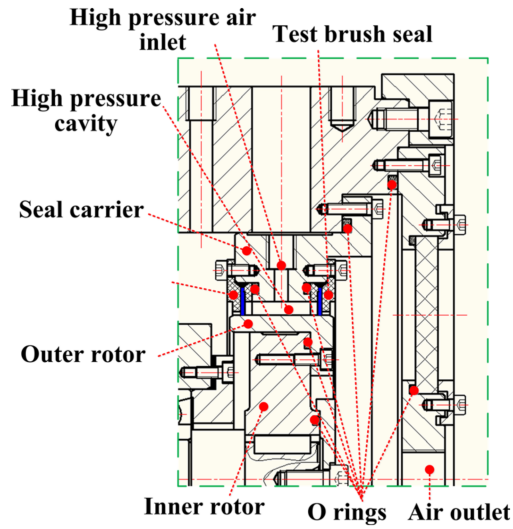


Figure 2.29: Cross-sectional view of the test facility, Huang et al. (2014)

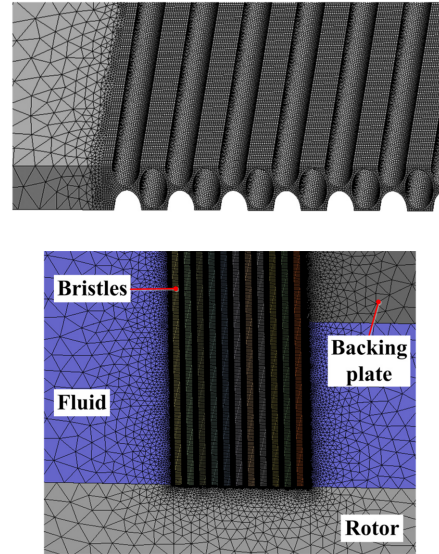


Figure 2.30: CFD domain and mesh, Huang et al. (2014)

The geometry of the bristles was modelled as straight tubes with constant inter-bristle spacing. The size of the inter-bristle spacing was adjusted to match experimental leakage data. To simplify the meshing of the domain and reduce computational time, the authors focused their model on the bristle tip region, arguing that most of the pressure drop and heat transfer occur in this region. The aerodynamic loads on the bristles from the CFD simulations were employed to calculate tip forces assuming that bristles were cantilevers in FEA. These contact forces were used to derive the frictional heat generation rate, using the general formula where frictional heat is the product of the friction coefficient, normal contact force and rotor surface speed. The authors also carried out a set of independent friction tests to find the value of the friction coefficient of the seal-rotor tribopair used in their experiments. The resulting frictional heat was imposed as boundary condition at the bristle-rotor interface in the CFD model. As other authors previously, a temperature equal to the inlet flow temperature (300 K) was imposed at the upper boundary of the bristle pack (outer radius). A laminar compressible flow solver was employed according to the low Reynolds number estimated by the authors.

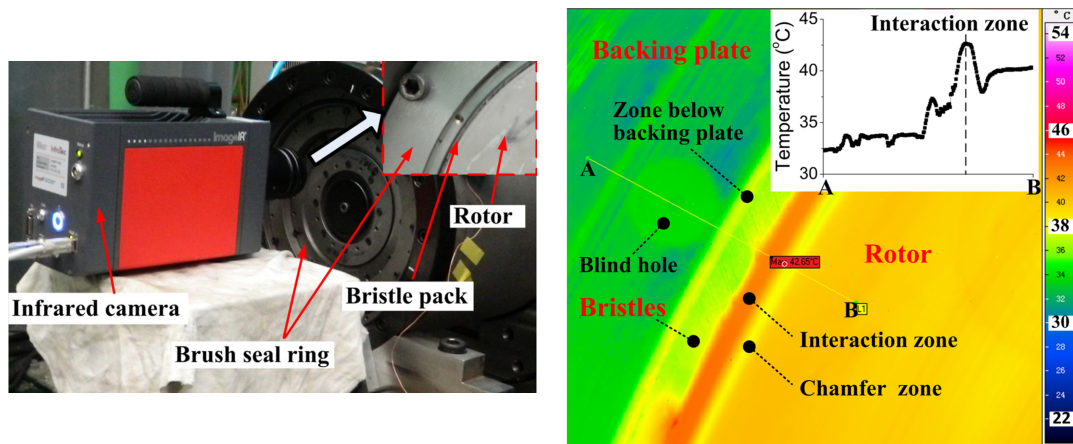


Figure 2.31: Infrared camera set up (left) and typical thermal image of bristle-rotor interface (3000 RPM, 0.2 MPa), Huang et al. (2014)

Experiments showed that in the speed range considered, the leakage flow was almost unaffected by rotational speed and only dependent (linearly) upon the seal differential pressure. Test torque and thus frictional heat generation were found to increase with differential pressure, confirming the presence of bristle blow-down. Numerical solutions showed that, not surprisingly, the largest pressure gradients and aerodynamic forces occurred on the downstream bristles in the region close to the rotor surface, where the blow-down effect is stronger. The authors highlight that this axial variation in bristle tip force results in higher frictional heat generation rates and temperatures at the rear of the bristle pack. Fig 2.31 shows a typical thermal image obtained at the bristle-rotor interface. This image confirms the continuity in contact temperature between seal and rotor that exists on a macroscopic scale. It also illustrates the rapid cooling of the bristle tips by the leakage flow in the radial direction. In addition, significant temperature variations in the circumferential direction can be observed, which suggests an uneven thermal behaviour of the seal circumferentially. Although the authors did not calculate how the frictional heat was shared between rotor and seal, the colder temperature field and stronger radial temperature gradients measured in the bristle pack seem to indicate that the seal was absorbing a larger proportion than the rotor. In general, predicted temperatures were consistent with experimental measurements. For tests at 3000 RPM, maximum bristle temperatures were found to increase with differential pressure, being

the resulting increment in contact forces and heat generation larger than the heat transfer enhancement due to the rise in leakage level. However, the authors point out that other speeds may not present the same trend. In addition, both the numerical model and the experiments indicated a linear relationship between rotational speed and maximum tip temperature up to 5000 RPM.

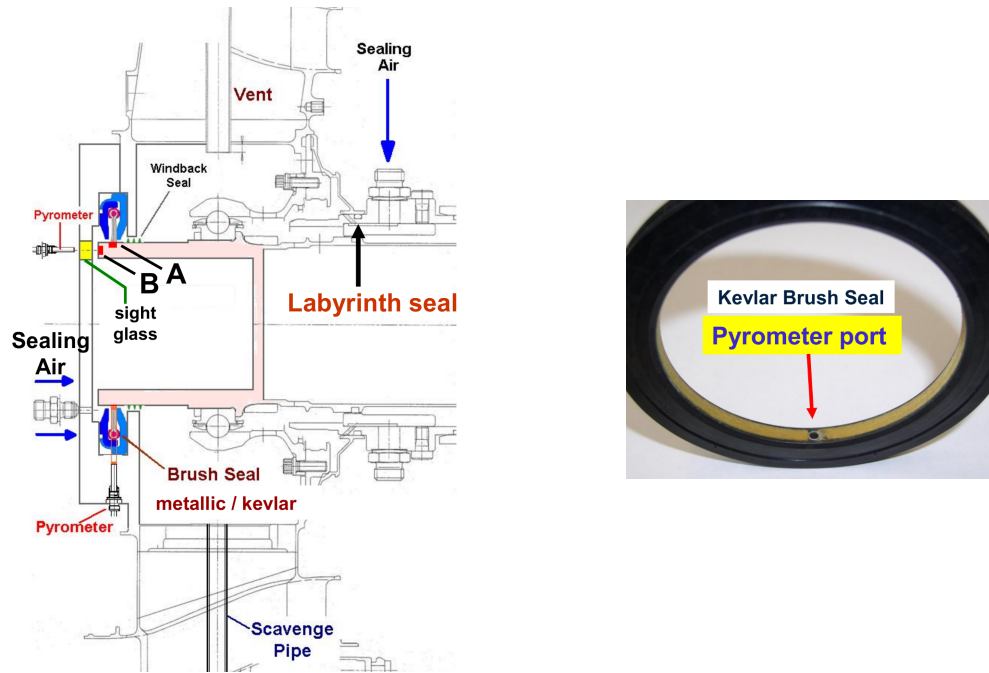


Figure 2.32: Cross-sectional view of the bearing chamber test facility (left) and location within the bristle pack where the pyrometer was installed for the measurement of contact temperature (right), Flouros et al. (2014)

Flouros et al. (2014) tested brush seals with bristles made of Kevlar fibres and conventional steel in a test facility based on an aero engine bearing chamber. The authors used a non-contact infrared pyrometric technique to measure transient temperatures at the bristle-rotor contact zone. A schematic of the experimental setup with the locations of the two pyrometers employed is shown in Fig 2.32. One of the pyrometers was installed within the bristle pack, pointing towards the rotor surface. The authors claim that this pyrometer provided accuracy with an error of ± 7 K at 200°C . Rotational speeds ranging between 6000 to 19500 RPM were tested. Results showed that Kevlar brush seals did not lead to contact temperatures considered elevated by the authors ($>230^{\circ}\text{C}$). Contrarily, metallic brush seals resulted in higher contact temperatures on average, with

the additional presence of transient temperature peaks up to 263 °C. Nevertheless, no deterioration of the rotating components, brush seal, or lubricant was observed after the tests.

In addition, the authors performed a CFD analysis to calculate the frictional heat transferred to the rotor during the experiments. The temperature fields within the rotor and sealing air were solved simultaneously. The modelling of the brush seal is not fully explained in the paper, although the authors comment that it is based on the flow through an equivalent gap causing the same pressure drop. The model boundary conditions were obtained from the experimental measurements of contact temperature, air mass flow rate and temperature, and bearing inner ring temperature. The outcome of these simulations was the heat flux conducted into the rotor at the contact interface with the brush seal. Results of the derived frictional heat are presented normalized by the maximum value obtained; however, it is possible to comment that the metallic brush seal produced higher frictional heating than bristles made of Kevlar fibres.

Fico et al. (2016) used CFD and FE thermal analysis together with experimental data to study the heat transfer characteristics of the leaf seal. They developed a CFD model based on a 3D sector comprising a single inter-leaf gap along with the upstream and downstream coverplate regions. Fig 2.33 illustrates the geometry modelled and the boundary conditions employed. Laminar flow was assumed throughout the domain. In addition to interesting flow features, the author found with this model the convective heat transfer coefficient distribution on the leaf surface using the Reynolds analogy. Both CFD and FE codes were Rolls Royce Plc inhouse codes.

The outcome of the CFD simulations was used to simulate a set of transient tests carried out with the Oxford Engine Seal Test Facility. In these experiments, the rotor was decelerated by the friction with the seals and other resistance torques from a rotational speed of approximately 7000 RPM to rest. During this deceleration, the upstream seal pressure was pulsed several times and the rotor temperature measured with infrared sensors. A separate 2D axisymmetric finite element (FE) model of the test rig was built. Experimental data was used to derive the boundary conditions required for the FE

simulations. Time histories of shaft speed, upstream and downstream air pressure, inlet air temperature, leakage mass flow rate and seal frictional heat were used as input to the model. Seal frictional heat generation was calculated from the torque measurement corrected for rig parasitic (windage and bearing) torques. Fig 2.34 depicts the domain considered in the FE simulations, showing the location where rotor temperature was measured experimentally. Since the airflow supply was alternately switched on and off during the experiments, the FE model boundary conditions were changed between free and forced convection as appropriate. When pressure was delivered to the seals, the heat transfer coefficient distribution obtained from the CFD simulations, multiplied by a matching factor, was applied. The value of this factor was adjusted to find the best fit with the experimental data. Comparison of the model predictions with the rotor temperature measured experimentally was in good agreement during the portions of the tests where the seals were pressurized. However, when there was no flow through the seals and natural convection acts as the main cooling mechanism, the model failed to match experimental values. This demonstrates that the modelling of heat transfer within the leaf pack in the absence of leakage flow is not a trivial problem.

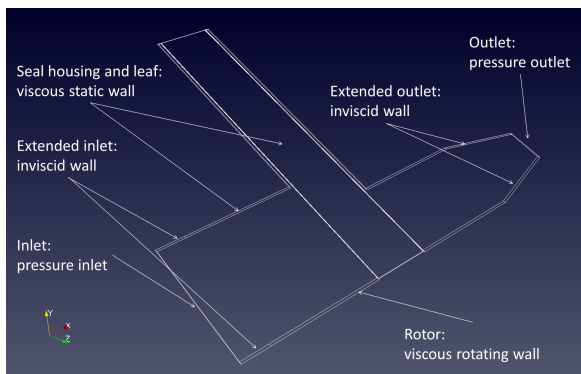


Figure 2.33: Leaf seal CFD domain and boundary conditions, Fico et al. (2016)

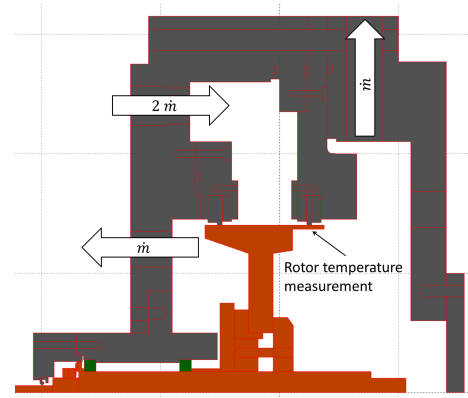


Figure 2.34: Schematic of the FE model domain of the experimental facility, Fico et al. (2016)

2.3 Summary and Concluding Comments

The most meaningful lessons learnt are listed below as a summary of the current state of the art in terms of seal heat transfer modelling capability.

2.3.1 Brush and leaf seal modelling strategies

Two main approaches have been employed in the past for modelling the mechanics of the inter-filament flow: porous medium models and models based on cross flow through an idealised filament pack. The porous medium approach has commonly been implemented in commercial CFD codes (such as ANSYS Fluent). The bristle/leaf pack is treated as a 2D axisymmetric porous medium, within which additional resistance terms are added to the momentum equations. These resistance coefficients are generally obtained by calibration with experimental data on seal leakage variation with pressure difference.

When cross flow through bristles/leaves is considered, the flow within voids between uniformly positioned rigid cylinders/plates is simulated using CFD. Only a limited number of filament rows have been modelled, and solution domains are generally confined to the filament tip region, where most of the pressure drop and convective heat transfer occur. Moreover, the inter-bristle/leaf spacing has been adjusted to match experimental leakage data. This approach allows a more detailed investigation about the effect of seal geometric variables on performance parameters.

In both approaches, the actual seal geometry and flow path are not captured due to bristle/leaf dynamic behaviour during seal operation, and laminar flow solvers have mostly been employed due to the expected low inter-filament Reynolds number.

To study heat transfer within the filament pack, cross flow models impose fluid-solid interfaces, and conjugate heat transfer is enabled. In the case of porous medium models, two strategies exist. First, some models assume local thermal equilibrium, and therefore, negligible temperature difference between the solid (filaments) and the fluid (leakage flow). Then, the flow energy equation is solved with a volume-average

thermal conductivity using the porosity of the medium. The second strategy assumes considerable temperature difference between solid and fluid phases, and local thermal non-equilibrium conditions are considered. A separate solid heat conduction equation is added, so that the flow and bristle energy equations are linked via a convective heat transfer coefficient, derived from conventional empirical correlations for internal flows. Since the porous medium flow model does not account for the actual gas path through the filament pack, local convective heat transfer is not captured properly. From a heat transfer point of view, the physical meaning of the porous medium is not clear.

In terms of analytical modelling, two models stand out. The fin approach models a single bristle or leaf as a fin. The leakage flow is assumed to have infinite heat capacity and a constant temperature equal to that at the inlet of the seal installation. The inter-filament convective heat transfer coefficient, required by the model, is obtained from conventional correlations for internal flow. The outcome of this model shows that the heat absorbed by the fin is proportional to the square root of inter-filament heat transfer coefficient (htc_{fin}) and the temperature rise at the tip (ΔT_c):

$$Q_{seal} \propto htc_{fin}^{0.5} \Delta T_c \quad (2.12)$$

The second remarkable analytical model is that developed at Oxford by Owen and Pekris. Both flow and filaments are assumed to present locally the same temperature. All frictional heat input to the seal is subsequently transferred to the leakage airflow. This formulation shows that the heat absorbed by the seal (Q_{seal} in Eq 2.13) is proportional to square root of the leakage mass flow rate ($\dot{m}$) and the temperature rise at the tip (ΔT_c). This functional dependency was also confirmed by Pekris using dimensional analysis. Although the proportionality of Eq 2.13 was validated with experimental data; the model failed to predict the value of the constant of proportionality, with the experimental value differing in a factor of ~ 2 . These analytical models incur strong simplifications, but still provide primary insight into seal heat transfer.

$$Q_{seal} \propto \dot{m}^{0.5} \Delta T_c \quad (2.13)$$

2.3.2 Boundary conditions commonly employed

The following statements are universally accepted by most previous researchers and form the fundamental basis to set the boundary conditions required by the seal heat transfer models:

- All frictional power is assumed to be dissipated as heat, which is equal to the sum of the seal and rotor heat flows.
- Frictional heat is applied as a uniformly distributed heat source at the filament tips.
- Negligible temperature rise is imposed at the root of the filaments. This assumption will be key in the model presented in Chapter 5.
- Seal and rotor are frequently decoupled and studied separately. Frictional heat partition has been therefore assumed. In Chapter 4 and 5, rotor and seal will be investigated separately in two different test rig configurations. In Chapter 6, an engine representative rotor-seal installation will be modelled, and the frictional heat partition addressed.
- Typical values of the inter-filament heat transfer coefficients, obtained from standard correlations, are of the order of $1000\text{-}7000\text{ W/m}^2\text{K}$. With the experimental results presented in Chapter 5, insight will be gained into the inter-filament flow heat transfer characteristics.
- Total frictional heat is commonly calculated as the product of friction coefficient, rotor surface speed and normal tip contact force. Also, values from experimental measurements of shaft torque and speed have been used.

2.3.3 Key model results and predictions

The phenomena listed below have been predicted by previous models. Some of the assumptions behind the methodology presented in this thesis build on these facts, while some results from the present research are in agreement with them.

- Exponential drop in bristle temperature with increasing distance from tip in the radial direction due to the highly convective environment within the filament pack.
- Convective heat transfer mostly occurs in the tip region.
- Reduction in leakage with heat input to the seal caused by the drop in gas density due to the heating of the flow.
- Increasing seal differential pressure involves larger contact forces and thus heat generation, but also improved seal cooling (conflicting mechanisms).
- No considerations have been provided by previous authors on the convective heat transfer coefficient distribution in the seal vicinity. However, plots showing the flow field predicted by the existing models indicate low speed flow upstream of the seal, and high speed axial jet and flow impingement on rotor surface downstream. Chapter 4 will address this gap in knowledge providing experimental data and numerical predictions.

2.3.4 Outstanding experimental observations

Although there are few studies on seal heat transfer that employ experimental techniques, the following key observations have been found:

- Seal filament tips and rotor surface present a common temperature at the contact interface on a macroscopic scale. This point will be extensively discussed in Chapter 3, and will have a big impact on the model proposed in this thesis.
- As mentioned before, frictional heat partition has not received much attention in the past. Without leakage flow, a 50-50 % split is commonly assumed. However, experimental observations suggest that a larger proportion is transferred to the seal when this is pressurized, due to the great cooling effect induced by the leakage flow ($Q_{seal} > Q_{rotor}$). The seal acts as a strong heat sink. This behaviour will be reproduced in the simulations carried out in Chapter 6.

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- The total heat generated at the seal-rotor interface has been found to be proportional to the square root of leakage mass flow rate and temperature rise at the contact area in both brush and leaf seals. This crucial result is the base of the empirical correlation investigated in Chapter 5.
 - Variation in seal thermal behaviour in the circumferential direction has been identified despite the nominal axial symmetry. In the experiments described in Chapter 5, this lack of circumferential uniformity will be decisive to understand the results obtained.

Chapter 3

Frictional Heat Generation and Sliding Interface Contact Temperatures

As explained in the previous Chapters, the filaments of leaf and brush seals are designed to work in close proximity with the rotor surface in order to maintain low leakage rates. Even under optimal operating conditions, contact between the filament tips and the rotor surface may occur. In addition, off-design transient rotor eccentric movements or radial growths can lead to heavy contact and severe wear during certain parts of the engine cycle. Although contact with the rotor improves the sealing performance of the installation, excessive friction may result, leading to significant heat generation and power loss.

In brush and leaf seal applications, frictional heat is primarily generated at the filament tips-rotor contact interface, and subsequently conducts into the rotor and the filaments. A good prediction of the temperature levels under such conditions is of paramount importance in order to guarantee the proper functioning of the system. To achieve this, the heat partition ratio, i.e. the two fractions of the frictional heat transferred to the rotor and to the seal, need to be known. When there is no pressure difference across the seal, it can be expected that the frictional heat divides among rotor and seal

according to the ratio between the seal contact area and the swept seal contact area on the rotor, provided their thermal conductivities and surrounding sink temperatures are similar. However, as seal differential pressure and leakage increase, the cooling of the bristles/leaves is greatly enhanced, and a larger proportion of the frictional heat is absorbed by the seal.

Frictional heat generation during sliding contact has been a topic of much previous research. The temperature distribution within the rubbing bodies and the heat partition between bodies have been investigated over several decades. The aim of this Chapter is to review some of the previous work on the thermal analysis of sliding systems and discuss those findings that are relevant to the brush and leaf seal application. Besides an improved fundamental insight on frictional heating, the main outcome of this Chapter is the electrical analogy concept, which will be the basis of the modelling strategy developed throughout this thesis.

3.1 The Nature of the Contact Area: Flash and Nominal Surface Temperature Rise

It is generally accepted that the vast majority of the mechanical energy dissipated during rubbing processes converts into heat, which causes an increase in the temperatures of the sliding bodies. In some applications this temperature rise can affect the mechanical and tribological behaviour of the contacting surfaces. Formation of oxide layers, melting, and phase transformation are some of the consequences derived from high temperature exposure at the contact interface.

Thakare et al. (2016) carried out an experimental investigation on the wear mechanism occurring in brush seals. Specimens were examined using microscopy and depth-wise surface chemical analysis after being tested in a sliding wear test rig. Fig 3.1 summarizes the main features observed by the authors on worn bristles. As represented in the figure, the analysis of the structure of the worn bristle surface and subsurface revealed the presence of a thin tribo-film (generally tens to hundreds of nanometres thick). Wear

significantly depends on the characteristics of this tribo-film, formed by the compaction of debris particles and oxidation of the substrate material. Low wear is promoted by the existence of a stable film, whose formation is dependent on the availability of wear debris and the occurrence of high temperatures that enhance sintering and oxidation. Accordingly, an increase in sliding speed, and therefore, resulting contact temperatures, was found to lead to a more uniform surface film, and a decrease in wear rates. In addition, the authors observed evidences of plastic flow on the worn surfaces, leading to the formation of “comet tail” type features, and sub-surface recrystallization, both suggesting the occurrence of temperatures probably close to 1000 °C in these regions. Microstructural examination showed that these features are limited to a few microns from the wear interface, which indicates the presence of a steep thermal gradient along the bristle tip.

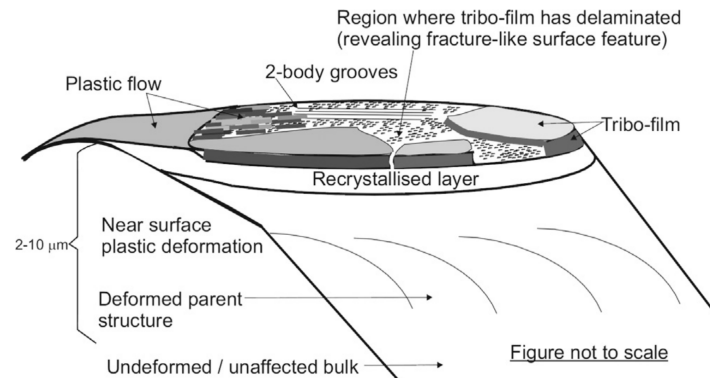


Figure 3.1: Schematic of the main features observed in a worn bristle, Thakare et al. (2016)

The study of frictional heating between two bodies in sliding contact requires the real nature of the contact area to be considered (Greenwood and Williamson (1966)). It is common knowledge that the actual contact area is only a small fraction of the apparent or nominal interfacial area. On a microscopic scale, surfaces are rough and contact between solids is restricted to a discrete number of randomly distributed areas, resulting from the deformation of interacting surface protrusions (asperities). This is schematically illustrated in Fig 3.2.

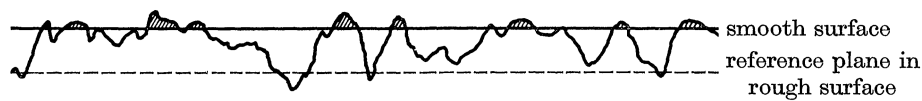


Figure 3.2: Contact of rough surfaces, Greenwood and Williamson (1966)

It is within this real area of contact that friction occurs and heat is generated. When two bodies come into contact, the atoms on their surfaces are brought very close together, stretching their bonds and acquiring potential energy. When they move apart, these bonds recover to their initial state, and the potential energy converts into kinetic energy. The temperature rise found in sliding friction is caused as this kinetic energy is randomly distributed over the adjacent atoms. In some cases, the energy gained by the contacting atoms is high enough to break their bonds, and thus, wear (mass loss) occurs. Since the real contact area is much smaller than the nominal one, the local pressure is higher at a given load. Therefore, the temperatures generated at the real contact area are higher than those obtained if a force was applied over the nominal contact area to obtain the same average heat flow. This phenomenon is known as flash temperatures (Blok (1963)) and explains the occurrence of unexpected high temperatures in many experiments on rubbing systems. It is important as it may determine the nature of the oxide layers formed.

In addition to this flash temperature rise, there is a mean or bulk temperature rise over the nominal contact area. The flash temperature rise is characterized by its short duration, transient nature, high intensity, and changing local position, while the nominal temperature rise is lower and nearly constant at a set engine running condition.

3.2 Frictional Heat Partition Calculation Using Small Scale Heat Transfer Models

One of the first attempts to calculate the temperature rise at the contact interface of two sliding bodies was by Blok (1937). This author focused on surfaces under boundary lubricating conditions, and relatively low contact pressures and temperatures. He studied the temperature distribution within solids subjected to stationary and moving

heat sources. Analytical solutions were obtained based on a semi-infinite body approach (a solid that extends to infinity in all directions except for the direction normal to the contact plane). Apart from the heat source, bodies were assumed to be thermally insulated (adiabatic boundaries). The equations derived were subsequently applied to study the frictional heating occurring in a single surface of actual contact, which represented a microscopic asperity.

Blok introduced the concept of heat partitioning. When heat is generated at the interface of two contacting solids 1 and 2, part of this heat will conduct into body 1 and the remaining into body 2. In order to determine these fractions, temperature continuity between coincident points on both bodies was assumed. Blok pointed out that if this temperature matching is to occur over the entire contact surface, the heat partition ratio cannot be constant with position. To simplify the problem, Blok assumed a constant partition ratio, obtained by matching the maximum surface temperatures of both bodies.

Jaeger (1942) investigated problems involving moving heat sources, which were commonly found in the calculation of temperatures at sliding or cutting contacts. Exact solutions were provided for the temperature field in a semi-infinite medium over which different types of heat sources move with constant velocity. No losses of heat from the remaining surfaces of the body were considered. Expressions for the temperatures attained at the plane of the moving heat source and within the semi-infinite medium were derived. It was shown that in most cases, the heat penetration into the medium was extremely shallow. The temperature rise within a semi-infinite solid due to a stationary heat supply was also calculated. These exact solutions are the basis of most of the subsequent analytical work on heat conduction during sliding contact.

Jaeger combined the aforementioned analytical solutions to address the problem formulated by Blok: one semi-infinite body sliding with a constant velocity over the surface of a second stationary semi-infinite medium (see Fig 3.3). Heat was generated at a uniform rate over the common surface (single microscopic contact), and there was no loss of heat from the remaining surfaces. The analytical solution for a moving heat source was applied to the stationary body, while that for a stationary heat source was

used in the moving body. Jaeger performed a division of the frictional heat similar to that made by Blok. The heat partition ratio was calculated equalling the average temperatures over the interfacial surface obtained for the moving and stationary bodies. Since the temperatures of both media did not match over the entire contact area, Jaeger admitted the approximate nature of his method, but argued that average temperatures are the quantities experimentally observed.

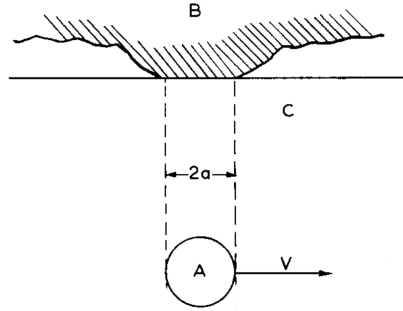


Figure 3.3: Schematic representation of single contact surface between two semi-infinite sliding bodies, Archard (1958)

At this stage of the discussion, it is useful to introduce the dimensionless quantity commonly employed to characterize sliding problems: the Peclet number (P_e). This is defined as:

$$P_e = \frac{V l}{4 \alpha} \quad (3.1)$$

where V is the relative sliding speed, l the characteristic length of the contact area, and α is the thermal diffusivity of the medium. Generally, for large values of the Peclet number, the heat flow in the direction perpendicular to the motion of the heat source can be neglected, and the heat source approximated as an infinitely long band source. For very small Peclet numbers, the temperature distribution is similar to that obtained with a stationary heat source.

Archard (1958) presented an approach similar to those of Jaeger and Blok to estimate contact temperatures when one solid slides over another base solid. Archard made a distinction between two different regimes: one characterized by high sliding speed and short length of heat penetration on the stationary body, and another where the

heat source is applied longer and heat penetrates further. Which of these two models is appropriate for a given problem was determined by the magnitude of the Peclet number. Archard developed simple formulas to relate the heat absorbed by each body to the mean temperature of the contact area. These expressions were based on Jaeger's work, although the author aimed to omit some mathematical complexities. For finding the fractions of the total heat supplied to each of the two contacting bodies, the author proposed the following formula:

$$\frac{1}{T_{avg}} = \frac{1}{T_1} + \frac{1}{T_2} \quad (3.2)$$

where T_1 and T_2 are the average temperature rise at the contact surfaces of bodies 1 and 2 if all the frictional heat was applied to each body independently, and T_{avg} the resulting average temperature rise over the contact area in the actual problem. No explanation or physical fundamentals were given for such a formulation.

After the pioneering works of Blok and Jaeger, researchers mainly focused on developing complex mathematical models that achieve continuity in temperature over the entire interfacial surface and a more realistic representation of the actual contact area.

A number of researchers further developed Blok and Jaeger's methodology during the following decades, see Ling (1959), Cameron et al. (1965), Symm (1967), Bos and Moes (1995), Komanduri and Hou (2001) and Bansal and Streater (2009). All these studies have some important features in common. The interfacial surface, represented as an individual microscopic contact, was modelled as a single square, circle, ellipse or line, over which heat was supplied. Bodies in frictional contact were treated as semi-infinite solids. Analytical solutions for the surface temperature rise were derived for the cases of a moving and a stationary heat source. Both solids were decoupled and modelled separately. The temperature rise at a point far removed from the contact zone was assumed to be zero. With this domain and boundary conditions, the resulting surface temperature rise was always proportional to the heat flux applied on the body surface. Frictional heat was assumed to be generated uniformly, and its value calculated as the

product of the friction coefficient, contact pressure, and sliding speed. Surfaces outside the heat source were considered to be adiabatic (although sometimes this condition was relaxed and convective cooling allowed). The sliding bodies had matching temperatures at every point within the entire contact zone. Although the distribution of total frictional heat was uniform, the fraction of heat absorbed by each body could vary with position. These formulations lead to an integral equation, resulting from equating the surface temperatures of the two rubbing bodies over the contact area. Since total frictional heat was a known quantity, the only unknown was the heat partition distribution. Solving the integral equation involved mathematical complexities, especially at intermediate Peclet numbers. As a result, several numerical schemes were proposed to obtain the heat partitioning function. Bansal and Streater (2009) compared the average interfacial temperature rise obtained with their analysis to that calculated using Jaeger's hypothesis (constant heat partition and average temperature matching). They concluded that Jaeger's approach provides excellent agreement with their more comprehensive analysis over a wide range of Peclet numbers.

All the aforementioned studies assumed a single and continuous contact area. However, more recent work has been done that accounts for real rough contacts and the effect of the surface topography, e.g., Qiu and Cheng (1998), Gao et al. (2000), Chen and Wang (2008), and Kadiric et al. (2008). These authors modelled the contact zone as a grid formed by discrete points representing each micro contact. The interaction between the mechanical and thermal effects was considered in these models. The contact pressure distribution was first calculated according to the surface topography, and then, the heat flux distribution was inferred. The amount of frictional heat generated at each single contact was calculated as the product of the friction coefficient, local contact pressure and sliding speed. Both rubbing bodies were assumed to be very large in relation to the contact area, and therefore, modelled as semi-infinite solids. Surface areas outside the real contact area were assumed to be thermally insulated. The temperature rise at each contact point was calculated separately using analytical solutions. The overall temperature rise was obtained superimposing the individual contributions. Temperatures

were matched at all points in contact, and the heat partition ratio thus obtained varied with position. An illustrative example of the typical contact temperature distribution obtained with this methodology is shown in Fig 3.4.

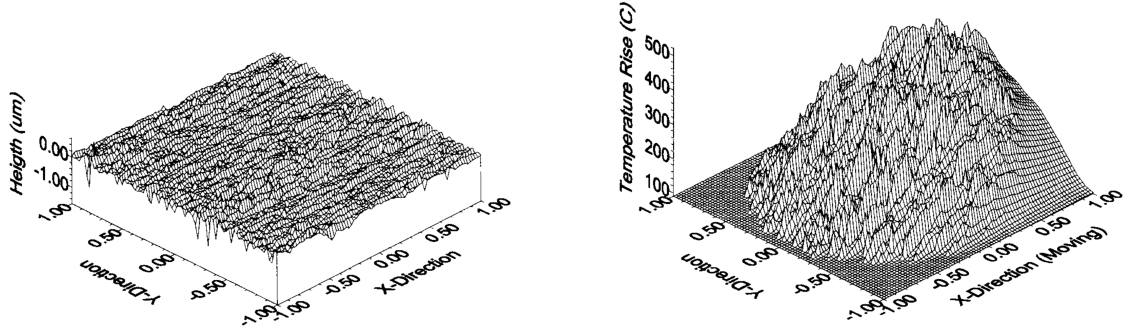


Figure 3.4: Surface roughness profile (left) and resulting typical temperature distribution over the contact area (right), Qiu and Cheng (1998)

So far, frictional heat has been modelled as a surface heat source in all the preceding models. However, the heat energy dissipated during sliding friction has two basic contributors. First, there is the energy associated with the making and breaking of adhesive bonds at the real surface of contact. Second, a volumetric heat generation exists due to plastic deformation within the near surface layers. Only the first component is strictly a surface heat flux, although the subsurface heat generation only extends within the top few microns of the sliding bodies. Nosko (2013) accounted for this volumetric subsurface heat generation in his model. This author considered two semi-infinite solids sliding relatively to each other. Total frictional heat was divided into two fractions, which were supplied to each rubbing body. These fractions were further split using a coefficient that indicated the proportion released due to the adhesive mechanism at the surface. An exponential distribution dependent on the distance from the contact surface was defined to model the remaining volumetric heat dissipation. This methodology is illustrated in Fig 3.5. The governing equations of the temperature field in a semi-infinite body with volumetric heat generation were solved assuming a zero temperature rise at an infinite distance from the interface, and a prescribed heat flux at the contact plane. Continuity in temperature was not imposed at contact. The author referred to other studies to determine a priori the fraction of the total frictional heat that was dissipated

within each body. Notably, the model required as an input the effective thickness of the subsurface layer where heat is released.

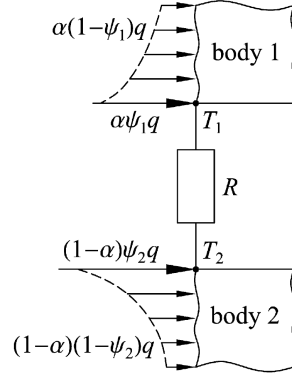


Figure 3.5: Surface and subsurface (volumetric) frictional heat generation, Nosko (2013). α indicates the fraction of the total frictional heat transferred to body 1, while ψ represents the amount of frictional heat dissipated at the contact surface.

As a summary of this section, it can be concluded that all the aforementioned studies have focused on the estimation of the maximum surface temperatures that occur in the real contact area, i.e., the flash temperatures. The difficulties involved in describing the geometry of the actual contact surface have been highlighted along with how the frictional energy is released. The two rubbing bodies have always been decoupled and modelled separately as semi-infinite solids. The link between them has been the condition of some kind of continuity in temperature over the interface. As a consequence, the heat absorbed by each body only depends on a single temperature, the surface temperature, since the temperature rise far from the contact remains unchanged. The frictional heat partitioning is thus determined by the conditions in the microscopic vicinity of the contacting asperities (geometry, thermal properties and sliding speed). For instance, the following formula was proposed by Kennedy and Tian (1994) for the calculation of the heat partition ratio when both bodies are in motion:

$$q_1 = \frac{1}{1 + \frac{k_2}{k_1} \sqrt{\frac{1+Pe_2}{1+Pe_1}}} q_{tot} \quad (3.3)$$

where the subscripts 1 and 2 refer to each of the sliding bodies, k is the thermal con-

ductivity, P_e the Peclet number and q_{tot} the total frictional heat flux. In the case of oil lubrication contact, or when a body moves very fast with respect to the heat source, the assumption of negligible bulk temperature rise could be valid and the semi-infinite solutions hold true. However, in many practical dry sliding applications, a bulk temperature rise does exist over the entire nominal contact region. It is important to point out that flash temperatures and small scale heat flow restrictions have been assumed to be the key mechanism that controls the heat partition so far. In the case of a brush or leaf seal rubbing against a rotor, it seems likely that large scale heat transfer, such as the cooling effect induced by the leakage flow or the bulk heating of the rotor, has a decisive influence on the heat partition. In the next section, the effect on the heat partitioning of large scale heat flow restrictions and nominal contact temperature rise are discussed.

3.3 Frictional Heat Partition Calculation Using Large Scale Heat Transfer Models

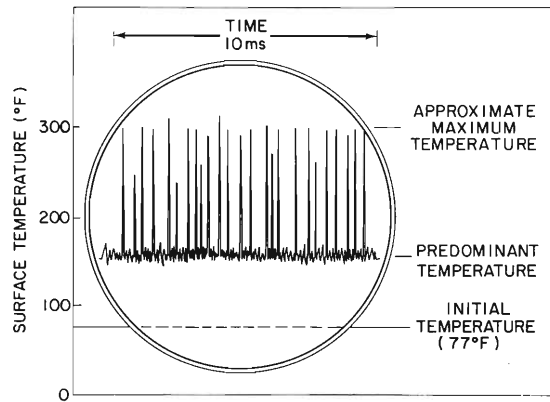


Figure 3.6: Surface temperature pattern, Furey (1964)

As previously discussed, frictional heat generation during sliding contact results in a surface temperature rise formed by two contributors: flash and nominal contact temperatures. In the heat transfer models discussed in Section 3.2, the heat partition ratio was calculated by matching the surface temperatures of the two rubbing bodies on a microscopic level (asperities). According to the definition provided in Section 3.1, these

temperatures correspond to flash temperatures. They are characterised by their high intensity and transient nature, and their value fluctuates after reaching steady-state. Figure 3.6 illustrates a typical surface temperature pattern observed during sliding contact. The sudden peaks (flash temperatures) occurring over the background mean (nominal) temperature rise can be clearly observed.

Moreover, it must be considered that the origin of flash temperatures is the heat flow restriction that the asperities in the rough contact represent. Since surface topography changes with time due to the wear process and thermal deformation, it is not surprising that flash temperatures also do so. Contrarily, the nominal or mean surface temperature rise is nearly constant at steady-state. Whether the flash temperatures are the controlling mechanism that determines the heat partition, or whether it is rather governed by the nominal surface temperature rise, is still a controversial question. In addition, the existence of continuity between the nominal contact temperature rises of the two sliding bodies is not universally accepted.

Ling and Pu (1964) pointed out that in order to obtain accurate solutions for the surface temperature at the interface of bodies in sliding contact, more realistic boundary conditions should be considered. These authors questioned the applicability of Blok and Jaeger formulae, based on semi-infinite solids, to bodies with finite extent such as a rotating shaft. In the following discussion, large scale models that consider the finite nature of the contacting bodies and their surrounding boundary conditions are examined.

Ling and Simkins (1963) carried out an experiment where a rotating disc rubbed against a stationary ring sector. The experimental setup employed is illustrated in Fig 3.7. Temperatures were measured at the locations showed in the diagram. The temperature in the rotating disc was found to be essentially constant at high speed, while the stationary ring temperatures changed with time. Given the difficulties in measuring surface temperatures, they were inferred by extrapolating subsurface temperature measurements assuming linear heat conduction. The surface temperatures thus obtained were termed macroscopic surface temperatures by the authors, and correspond to the temperature rise at the nominal contact area. The authors found a difference between

the contact temperatures of the ring and disc, which denoted a macroscopic jump in interfacial temperature. This discontinuity was explained due to the fact that frictional heat is generated in one or both of the surfaces in sliding contact and not at the interface exactly. According to the authors, a significant amount of heat should be generated within the subsurface layer of the warmer body (in this case the stationary ring) and some heat must be transferred from the ring to the disc by convection and radiation mechanisms. The temperature jump is a consequence of the thermal resistance of the contact. The authors did not claim that there is a jump on the microscopic scale at the instantaneous asperity contacts. To conclude, it was stated that whenever the capacity of one of the bodies to remove heat away is less than the amount of heat generated within its subsurface layer, there will be a macroscopic temperature jump across the interface. No expression was proposed for determining the heat partition ratio.

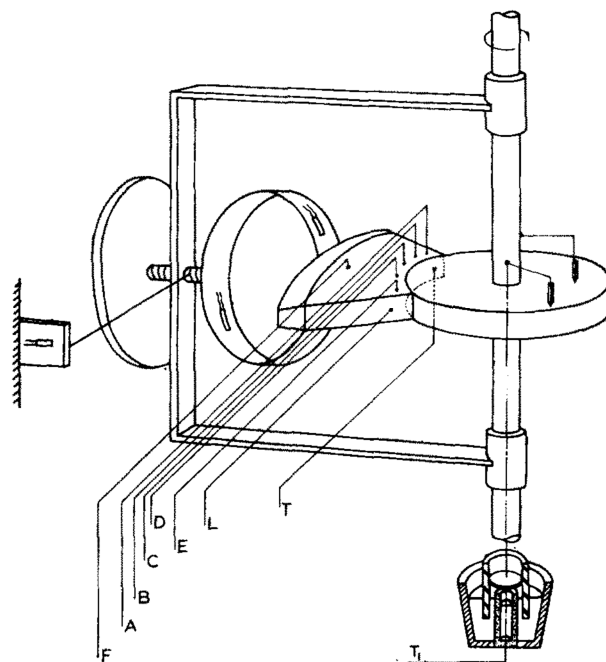


Figure 3.7: Schematic diagram of friction test facility showing the locations where temperature was measured, Ling and Pu (1964)

It is important to take into account that because both macroscopic surface temperatures were extrapolated, their accuracy relies on the correctness of the models employed for their derivation and the precision of the experimental data. In addition, it is some-

what surprising that the authors obtained a nearly constant surface temperature for the rotating disc (heated intermittently) and yet a variation in the surface temperature of the stationary ring (heated continuously). Later, it will be shown (Patula (1981), Gecim and Winer (1984), Clarke et al. (2006)) that the steady temperature field in a rotating disc subjected simultaneously to surface heating and convective cooling is formed by two main components: an interior bulk temperature rise constant with time, and a shallow temperature rise that changes with angular position. This temperature distribution is illustrated in Figs 3.8 and 3.9, which show the typical surface temperature pattern obtained for a rotating disc with its surface exposed to boundary conditions alternating between heat input and convective cooling. Since the thermocouple embedded in the disc (see Fig 3.7, location termed as “T”) was far from the surface, it measured the bulk temperature rise rather than the actual surface temperature peaks. This fact could explain why Ling and Simkins (1963) estimated a lower temperature at the disc contact surface. Nevertheless, since the experience of Ling and Simkins (1963), the existence of continuity in nominal contact temperatures has been a controversial topic. Some authors have referred to Ling and Simkins’ work to justify a macroscopic contact temperature jump (e.g., Barber (1970), Kennedy and Tian (1993), Nosko (2013)).

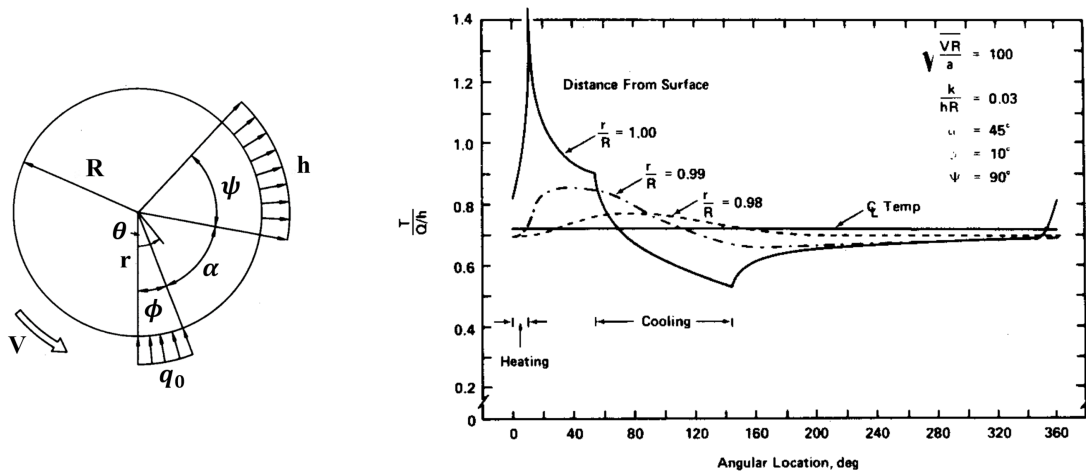


Figure 3.8: Rotating cylinder in sliding friction, geometry and boundary conditions: heat input, thermal insulation and convective cooling (left). Variation of normalized disc temperature with radial and angular position (right), Patula (1981)

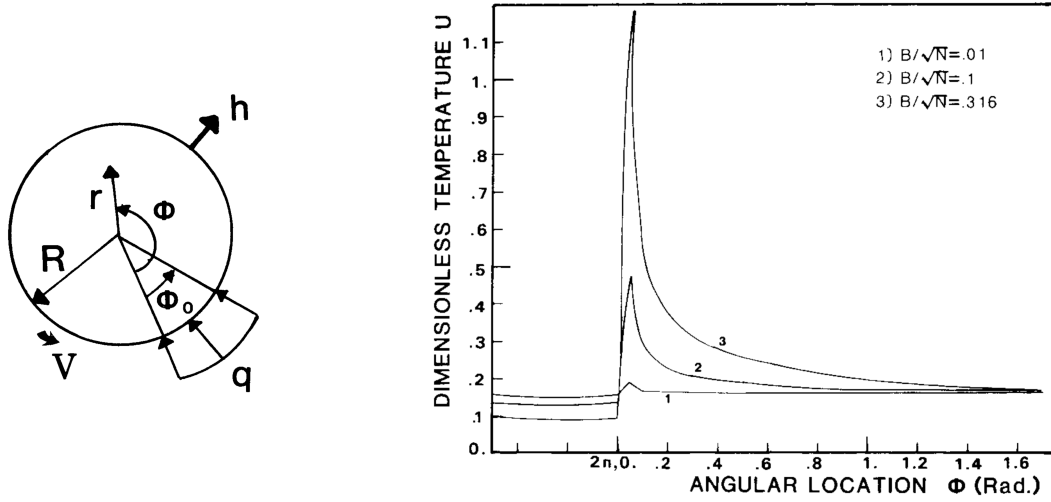


Figure 3.9: Rotating cylinder in sliding friction, geometry and boundary conditions: heat input and convective cooling (left). Variation of dimensionless surface temperature with angular position at different $B/\sqrt{N}$ ratios, B =Biot number, $2N$ =Peclet number (right), Gecim and Winer (1984)

Furey (1964) and Barber (1970) highlighted the importance of considering the bulk temperature field and the large scale heat flow restrictions of each sliding body in the calculation of the heat partition ratio. Furey (1964) observed a lack of agreement between experimental data on surface temperature increments and those obtained using the existing theory (Blok, Jaeger and Archard mainly). Experimental values were found to be considerably lower than the theoretical ones. The author argued that some of the factors that may have caused this discrepancy are the actual finite heat capacity of the contacting bodies and heat losses by convection, which were not accounted in previous analytical models based on semi-infinite and thermally insulated solids. Barber (1970) considered the effects of a difference in the temperature at infinity of the two rubbing bodies, modelled as semi-infinite solids. The introduction of large scale thermal resistances into the system involved understanding that the heat flowing into each body was also related to the temperatures far from the contact. The author found that this modification had a relatively small impact on the temperature at the actual contact area, but the division of heat between both solids was markedly affected.

Francis (1970) studied the interfacial temperature field within a sliding hertzian contact using analytical expressions of a semi-infinite body subjected to an ellipsoidal

heat flux distribution on its surface. The author calculated the heat partition ratio by matching the average flash temperatures of the contacting bodies in a similar manner as Jaeger (1942). However, Francis also considered the case where the temperatures of the two bodies far from the real contact area were not equal. He assumed that, in addition to the frictional heat flow, there was now an additional amount of heat that was transferred from the hotter body to the colder one across the interface. This heat flow was superimposed over the semi-infinite solution and changed significantly the heat partition ratio.

Ashby et al. (1991) proposed a simple formulation for calculating flash and nominal temperatures during dry sliding. When contacting solids slide, the authors assumed that an amount of heat equal to the normal contact force, friction coefficient and sliding speed is generated at the surface at which they meet. This heat flows into the two solids in a way which depends on their geometrical and thermal properties. They distinguished two temperatures of interest: the mean or nominal surface temperature and the local or flash temperature. The difference of both temperatures is governed by the ratio between the true contact area and the nominal area. The following expression, similar to an electric analogy, was suggested to calculate the nominal contact temperature rise:

$$T_n - T_0 = \frac{Q_{tot}}{A_n} \frac{1}{\frac{k_1}{L_{1n}} + \frac{k_2}{L_{2n}}} \quad (3.4)$$

where T_0 and T_n are the ambient and nominal contact temperatures respectively, Q_{tot} the frictional heat, A_n the nominal contact area, k_1 and k_2 the thermal conductivities of both bodies, and L_{1n} and L_{2n} two lengths. The complexity of the problems is contained in determining these lengths. They are dependent on the large scale cooling characteristics of each body since the heat absorbed must be dissipated by conduction, convection and/or radiation.

An analogous expression was proposed for calculating the flash temperatures (T_f), where the ambient temperature is substituted by the nominal surface temperature, which

acts as the sink temperature:

$$T_f - T_n = \frac{Q_{tot}}{A_r} \frac{1}{\frac{k_1}{L_{1f}} + \frac{k_2}{L_{2f}}} \quad (3.5)$$

where A_r is the real contact area, and L_{1f} and L_{2f} the two effective heat-diffusion distances, which depend on the size of the asperities and whether the heat flow is transient or steady. This methodology implicitly assumes that the nominal interface temperature is equal for both contacting bodies. Some expressions for the lengths required in Eq 3.4 and 3.5 were provided by the authors. The application of the model to data from pin-on-disk experiments showed satisfactory agreement.

Kennedy and Tian (1993) also questioned the validity of the assumption commonly made that the temperature rise in the surrounding vicinity of the real contact area (asperities) is negligible. Besides the localized flash surface temperature rise found at the asperities, there is a nominal contact temperature rise above the initial ambient temperature of the contacting bodies. According to the authors, these ideas have generally been overlooked by surface temperature analysts. They proposed a model based on the concepts of small scale and large scale heat flow restrictions, which is depicted in Fig 3.10. The authors argued that there must exist continuity in temperature wherever real contact occurs, i.e., at the microscopic level where asperity contact happens. Frictional heat is assumed to be released at this true contact points, and subsequently it flows into both rubbing bodies. Due to a small scale heat flow restriction, there is a rapid non-linear temperature drop within the near surface layers of each body. As a result, the nominal contact temperatures of the two bodies are not necessarily equal (see Fig 3.10). The two heat flows further penetrate into each body, causing an additional temperature drop that is governed by the large scale heat flow restrictions.

According to these ideas, the contact temperature is the sum of the local surface temperature rise (flash), the nominal surface temperature rise, and the background ambient temperature. To obtain the local or flash surface temperature rise, the authors proposed the use of Blok's semi-infinite medium approach. Contradictorily, the heat

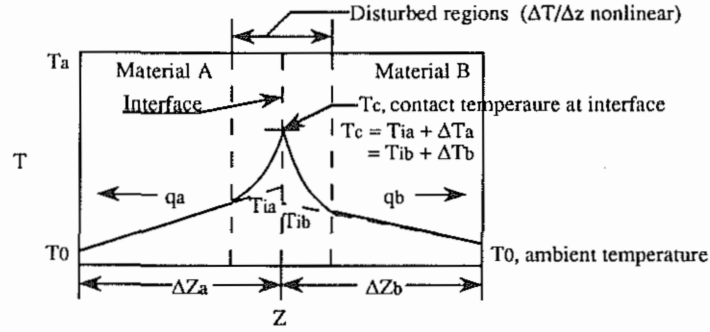


Figure 3.10: Temperature field at contact, Kennedy and Tian (1993). T_c represents the flash temperatures occurring at the asperity level, while T_{ia} and T_{ib} are the nominal contact temperatures at each of the sliding bodies

partition ratio in the authors' model was dictated by analytical solutions that implicitly did not take into account the large scale heat transfer in the contacting bodies. Once the values of the heat fluxes entering each body were known, the nominal surface temperature rise was calculated using a large scale model of the sliding geometries. The fractions of the frictional heat previously derived were applied as an imaginary constant heat flux over the nominal contact area of each body. The resulting nominal contact temperature rise was thus influenced by the large scale cooling of the solids. Finally, a condition of continuity in the overall contact temperature (flash plus nominal) obtained for each body was imposed.

Many researchers have focused in the past on the study of sliding friction in problems of more practical engineering application. In these cases, the rubbing bodies have been modelled on a large scale basis, ignoring the heat transfer mechanisms occurring at the micro contacts. Nominal contact areas and temperatures are thus considered. To calculate the heat partition ratio, surface temperatures are matched at the apparent contact interface. Therefore, the heat partitioning is governed in these models by the large scale cooling capacity of each body. Some of these studies are summarized below as representative examples of this methodology.

Using integral transform techniques, the surface temperature rise in a fast rotating cylinder subjected to constant surface heating and convective cooling was investigated by Gecim and Winer (1984). Fig 3.9 illustrates the domain and boundary conditions

considered. The authors found that the surface temperature rise was largely governed by the ratio of the Biot number to the square root of the Peclet number. The surface temperature obtained reached a peak value over the arc where the heat source was applied, and then, it exponentially decreased to a nearly constant value over the cooling zone. Thermal penetration in the radial direction was shown to be very shallow, leaving the bulk of the body at a uniform temperature. For the case in which the heat source was due to friction, a partition coefficient was derived from the principle that both sliding bodies must present the same temperature at the contact (nominal) area. This partition coefficient included the Biot and Peclet numbers, geometry coefficients, and material properties.

A similar problem was investigated later by Tudor and Khonsari (2006). The steady-state temperature distribution in a wheel exposed to frictional heating due to rail and brake contact was considered. An analytical model of the wheel was employed where the boundary conditions at the surface were convective cooling and three uniform heat fluxes (rail and two brakes). Individual models of the rail, treated as a semi-infinite solid, and the two brake shoes, were developed. The frictional heat flows conducted into the wheel were calculated by assuming continuity between the temperature fields of each body at the three contact zones.

Laraqi et al. (2009) proposed an analytical model to calculate the temperature distribution and the heat partition ratio in a pin-on-disc wear rig (see Fig 3.11). A uniform heat flux was imposed at the contact area, and convective heat transfer allowed on the remaining surfaces. Both disc and pin were considered semi-infinite in the direction perpendicular to the plane of contact.

Examples of the temperature distribution obtained on the plane of contact are shown in Fig 3.11. The formulas derived for the disc and pin surface temperatures indicated that the temperature increments were in both cases directly proportional to the heat flux supplied. In view of this result, all the variables appearing in such expressions were lumped in a single term, designated by the authors as thermal resistance (R). The average contact temperature rise of each body (T_{avg}) was then expressed as the product

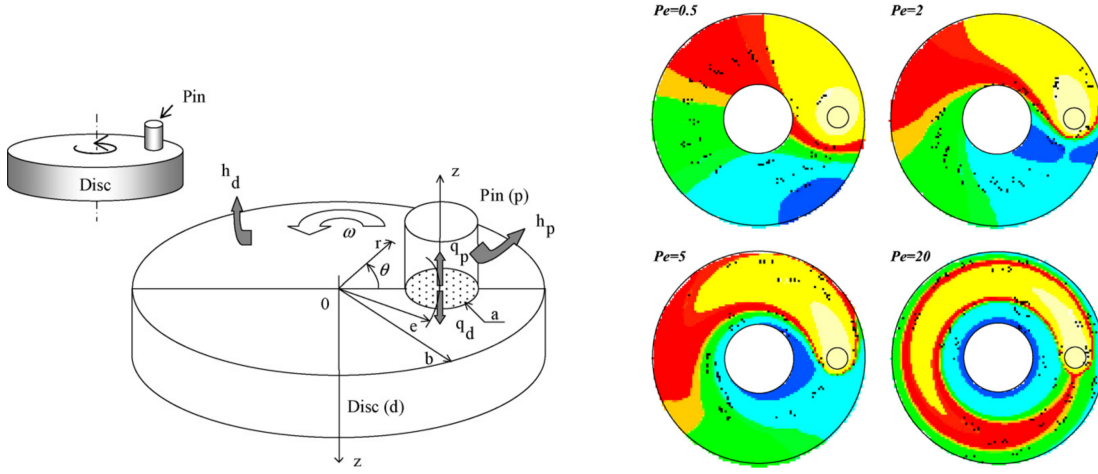


Figure 3.11: Pin-on-disc device (left) and calculated disc surface temperature distributions at various Peclet numbers (right), Laraqi et al. (2009)

of the thermal resistance and the heat input (Q). To determine the heat partition ratio, the average contact temperature rises in the pin and disc were matched:

$$T_{pin\ avg} = R_{pin} Q_{pin} \quad (3.6)$$

$$T_{disc\ avg} = R_{disc} Q_{disc} \quad (3.7)$$

$$T_{pin\ avg} = T_{disc\ avg} \quad (3.8)$$

$$Q_{tot} = Q_{pin} + Q_{disc} \quad (3.9)$$

$$\text{heat partition ratio } \eta = \frac{Q_{disc}}{Q_{tot}} = \frac{R_{pin}}{R_{pin} + R_{disc}} \quad (3.10)$$

The values of the thermal resistances were computed for different operating conditions. The authors found that the heat partition ratio was mainly dependent on the Peclet and Biot numbers of each body. In addition, beyond a certain sliding speed, the heat partition ratio was only affected by the Biot number.

Finite Element Analysis has also been employed to analyse surface temperatures in sliding bodies of complex shape, where integral transform techniques are difficult to apply. Frictional heat can be input as a heat flux boundary condition at the contact surface, and/or as a volumetric heating within the subsurface layers. The temperature field in both contacting bodies can be solved simultaneously, so that no artificial heat

partitioning is required. A condition of continuity in temperature is generally imposed at nodes belonging to both bodies. To compute the temperature distribution in components that are moving relative to the heat source, special formulation is required (Kennedy (1981), Kennedy (1984)). Hybrid methods have thus resulted, which make use of finite element techniques to solve the component subjected to a stationary heat source, and apply transform methods to the component receiving a moving heat supply (Colin and Floquet (1986)).

3.4 Third Body Approach

As discussed in Section 3.3, the existence of macroscopic temperature continuity across the nominal contact area in sliding systems is well known as a problem difficult to accommodate in simplistic analyses. Some researchers have assumed the existence of an intermediate thin layer between the sliding bodies to eliminate the need to analyse heat generation at the molecular level but nonetheless model this temperature jump. This layer has been called the friction layer or “third body”.

Ryhming (1979) introduced this concept to analyse the heat conduction problem found in a tool work-piece interaction. Accordingly, a thin layer was assumed to exist between the tool and the work-piece, as shown in Fig 3.12. Heat was generated in this layer due to friction at a constant rate per unit volume. The temperature distribution in the layer was governed by the heat conduction equation, and it was assumed to be in perfect thermal contact with both rubbing bodies. Therefore, different temperature distributions at the edges of the layer could be accommodated.

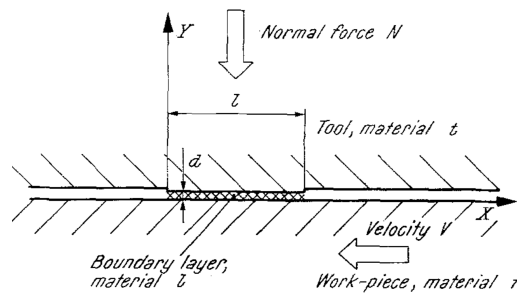


Figure 3.12: Third body representation, Ryhming (1979)

The concept of the contact interface layer has been used in an automotive application in brake friction modelling (see Majcherczak et al. (2005), Osterle and Urban (2006), Loizou et al. (2013)). In these studies, the pad was usually found to be at a higher temperature than the disc at the contact zone. The existence of an intermediate third body would explain this thermal gradient. However, due to the high rotational speeds attained, the disc was studied using two dimensional axisymmetric models that neglect angular variations in temperature. As explained in Section 3.3 (see Fig 3.8 and 3.9), this could lead to an underestimation of the actual disc contact temperature, since the temperature obtained corresponds to a bulk or mean disc surface temperature. The temperature peaks that occur periodically at contact may thus have been overlooked.

The use of a third body approach is not free of problems. Difficulties arise in determining the geometric and thermal characteristics of the interface layer. Thermal conductivity, specific heat, density and especially thickness, are parameters that strongly influence the third body's thermal performance. For instance, if its thickness is assumed to be of the order of microns or below, the contact temperature jump obtained is negligible (Majcherczak et al. (2005)).

3.5 A Brush/Leaf Seal Friction Heat Model

High speed rotary friction between the filament tips of a brush or leaf seal and the rotor surface results in a rise in the interface temperature. As discussed throughout this chapter, contact temperatures (T_c) consist of three terms. First, intense and transient temperature variations occur at the microscopic locations where real contact exists (flash temperature rise, ΔT_f). Second, due to the overall energy input that each sliding body receives, there is a bulk or nominal surface temperature rise over the apparent contact area (ΔT_n). Heat generation causes local increments in the average rise of flash and nominal temperatures. The increment history is added to the initial or ambient temperature of the rubbing bodies (T_0) to give the final contact temperature:

$$T_c = f(\Delta T_f, \Delta T_n, T_0) \quad (3.11)$$

Flash temperatures mainly depend on the microscopic topography of the contacting surfaces, sliding speed and magnitude of the frictional heat generation, and the thermal diffusivities of the sliding solids. While the heat transfer characteristics far from the contact zone have little impact on the flash temperature rise, they strongly influence the nominal surface temperature rise. The estimation of flash temperatures is extremely difficult because of our ignorance of the geometry and thermal properties of the actual contact area and how it evolves as wear progresses. Their measurement experimentally is also challenging. When friction between two sliding systems reaches steady-state conditions, the contact temperature is formed by a high frequency flash temperature rise fluctuation superimposed on a nearly constant nominal surface temperature rise. All the previous works on brush and leaf seal heat transfer (reviewed in Chapter 2) have ignored flash temperatures, and tackled the problem on a large scale. In consequence, the contact temperatures considered by these authors correspond to the nominal surface temperature rise. In Section 3.3, it was shown that, when there is a significant bulk heating of the rubbing bodies and these cannot be considered semi-infinite, then the partition of frictional heat is markedly affected by the surrounding temperature field and boundary conditions. In these cases, large scale models based on macroscopic or nominal contact areas and temperatures are employed. The main limitation of large scale models is the hypothesis implicitly assumed that continuity in nominal contact temperatures exists. The brush and leaf seal heat transfer models described in Chapter 2 presumed that over the nominal contact area, the rotor surface and the tips of the seal filaments present the same temperature. Moreover, experimental observations using infrared thermography (Demiroglu and Tichy (2007), Huang et al. (2014)) have confirmed that rotor and filaments exhibit a common interface temperature, and discard the existence of a significant temperature jump. Whether the short duration flash temperatures are actually the ones which regulate the heat partition ratio, or whether the nominal surface temperature rise is the controlling mechanism is a topic that has been discussed extensively. However, as exposed in this Chapter, no general agreement has been reached. For the brush and leaf seal problem, it seems evident that the airflow around the rotor and through

the seal plays a key role in determining the resulting nominal contact temperature and heat partition ratio. Previous investigations have clearly shown how the large cooling effect induced by the leakage flow enormously increases the amount of frictional heat transferred to the seal. In view of all the aforementioned facts and considerations, an approach based on the large scale heat transfer characteristics of the seal installation is adopted in this thesis to calculate the partitioning of the frictional heat. This is explained next.

As discussed in Section 3.1, under certain operating conditions, a tribo-film of different chemical composition may form at the bristle-rotor interface (Thakare et al. (2016)). This thin layer is essentially the equivalent of the “third body” in some frictional heating models (see Section 3.4), and would explain the potential existence of a discontinuity in nominal contact temperatures between the metal surfaces. Thakare et al. (2016) observed experimentally that oxides and debris particles can get sintered together into a coating over the bristle surface if the temperature at the contact region is high enough. The maximum thickness of this tribo-film found by the authors was ~ 500 nm. For the range of friction powers expected in a typical seal application (<10 kW) and considering the film to be made of pure iron oxide ($k = 0.58$ W/mK), the temperature drop introduced by heat conduction through a tribo-film 500 nm in thickness is expected to be lower than 2 K. This values correspond to a worst case scenario, and therefore, the tribo-film can be considered thermally thin and not requiring modelling.

From the vast amount of work done and the different methodologies proposed in the past, it is generally agreed that two conditions must be satisfied at the contact interface. First, the sum of the two heat fractions transferred to each rubbing body must equal the total heat generated. Second, continuity in temperature must exists wherever the two solids meet. Under these considerations, the heat transfer mechanism that occurs during seal-rotor friction can be studied using an electrical analogy. This concept is illustrated in Fig 3.13.

Frictional heat is assumed to be generated at the surface of the asperities that form the real contact area on a microscopic level. Since this is much smaller than the apparent

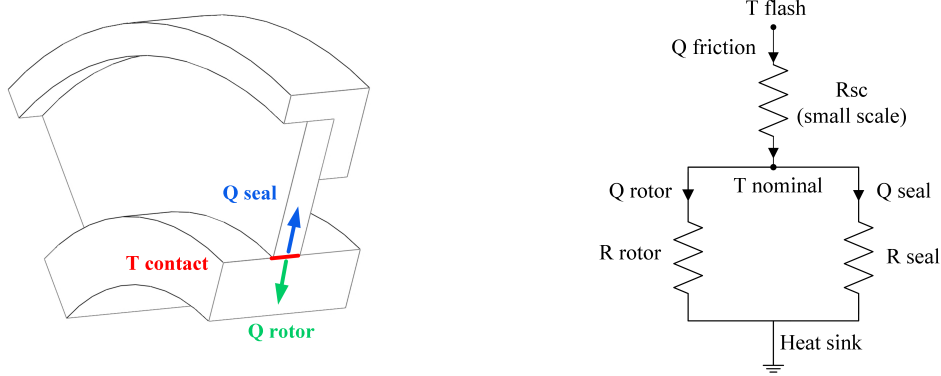


Figure 3.13: Heat transfer mechanism in sliding contact

or nominal contact area, it represents a strong heat flow restriction, which is included in the model as a small scale thermal resistance (R_{sc}). The reduced area available for heat conduction at the actual contact causes the flash temperature rise, which is considered in the model as a temperature drop common to both sliding bodies due to its infinitesimal local extent. As a result, the model assumes that the nominal surface temperature rise is equal for both rotor and seal. Total frictional heat can thus be imposed as a surface heat source over the nominal contact area. This heat source is stationary relative to both rotor and seal, and therefore, the heat partition ratio is not expected to be dependent on the sliding speed. The total frictional heat is divided according to the relative capacities of each contacting body at removing heat from the interface. These are captured in the model as thermal resistances (R_{rotor} , R_{seal}), and are determined by the large scale heat transfer of seal and rotor. The problem of the heat partitioning thus reduces to finding out the values of these thermal resistances, which will be the focus of Chapters 4 and 5. This approach allows us to decouple the seal from the rotor and study them separately, as long as the two linking conditions previously stated hold true (energy conservation and temperature continuity). The investigation of the small scale resistance (R_{sc}) and flash temperatures is out of the scope of this thesis. However, they are a key element in frictional heating, having an important effect on chemical deterioration and oxidation, local melting and softening, and wear rates.

3.6 Summary and Conclusions

The partition of the frictional heat generated at the interface of two rubbing bodies is a difficult task where no general solution has been found yet. The main difficulty lies in the multi-scale dimension of the problem. It is extremely complex to treat the problem with all the detail that it requires, modelling simultaneously the small and large scale heat transfer mechanisms.

Some researchers tackle the problem from a microscope scale (Section 3.2). Frictional heat is assumed to be generated at sliding asperity contacts. These micro contacts act as a bottle neck to the passage of the heat flow causing high flash temperature increases. The real contact area is modelled assuming that the bulk of the sliding bodies are semi-infinite. By applying continuity in temperatures at the asperity tips, a heat partition ratio can be derived which depends solely on the thermal diffusivities, contact geometry and sliding speed.

Other approaches regard the sliding system on a macroscopic scale (Section 3.3). These studies model the entire body geometries and deal with nominal or apparent contact areas and temperatures. The heat partition ratio is thus governed by the large scale heat transfer characteristics and boundary conditions of the system. The main difficulty that these approaches face is finding a correct way of linking the temperature fields of the two contacting bodies, since the assumption of macroscopic contact temperature continuity is controversial.

In addition, where and how frictional energy is released is still one of the most important and fundamental unknowns in the study of sliding contact. Although nearly all analyses apply frictional heat as a surface heat flux, there is experimental evidence of subsurface heat generation.

As a result, engineers have had to make important assumptions and simplifications in order to build practical models that can give us a rough idea of the thermal behaviour of bodies in sliding contact. In this thesis, frictional heat generation is studied from a macroscopic perspective, and temperature continuity is assumed at the seal-rotor

nominal contact area. It should be noted that a parallel investigation focusing on the microscale behaviour of the rotor-filament interface would be of great interest to better understand seal wear and performance degradation.

Chapter 4

Experimental Characterization Of Rotor Convective Heat Transfer Coefficients

When the filaments of a brush or leaf seal rub against the rotor surface, mechanical energy is converted into thermal energy at the interface. The heat generated is shared between the two contacting bodies according to a certain heat partition ratio. As discussed in Chapter 3, this problem can be studied considering an electrical analogy, which is schematically represented in Fig 4.1. Two conditions need to be satisfied at the contact zone. First, the sum of the fractions of heat transferred to the seal and rotor must equal the total frictional power. Second, a bulk or nominal temperature rise occurs at the interface, which is assumed to be common to both rubbing surfaces as contact resistance is low.

The partition of the frictional heat is determined by the heat transfer characteristics of both seal and rotor. These are captured in the model as thermal resistances (R), a concept that allows us to decouple the sliding system and study rotor and seal separately. The geometry and boundary conditions of the rotor depend on its location within the engine and operating conditions. A finite element model can, for instance, be built to study the thermal behaviour of the rotor. The key boundary condition is the

fraction of the total frictional heat that must be supplied at the rotor contact interface (Q_{rotor}). The solution of the rotor temperature field has to meet the bonding condition of continuity in nominal contact temperature rise to be valid.

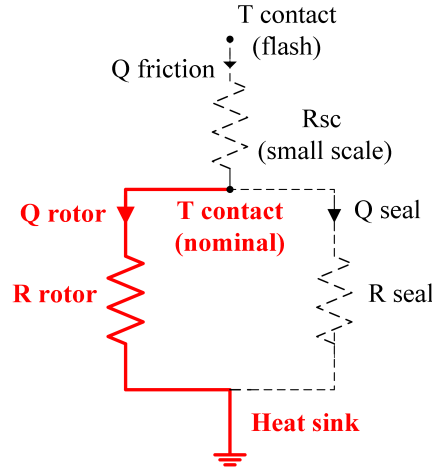


Figure 4.1: Thermal modelling of seal-rotor frictional heat generation using an electrical analogy

The aim of the work presented in this Chapter is to improve our understanding into the convective heat transfer characteristics of the flow upstream and downstream of a filament seal. This will allow us to employ more accurate thermal boundary conditions for a rotor heat transfer model (R_{rotor} , in red in Fig 4.1). As explained in Chapter 2 (see Fig 2.11), relatively far upstream and downstream of the seal, heat transfer is mainly governed by the swirling flow induced by the shaft rotation. At these surfaces, a convective heat transfer coefficient (htc) distribution can be imposed using available correlations from the literature for a rotating cylinder in an annular passage. However, there is a lack of knowledge about the flow characteristics and convective heat transfer in close vicinity to the seal. The leakage flow experiences an intense acceleration as it expands through the seal filaments. Upstream of the seal, the flow is accelerated to enter the inter-filament gaps, which may cause relaminarisation of the flow. Downstream, the high speed jet is vectored towards the rotor surface. In a manner akin to that of an impingement system wall jet, it mixes turbulently with the surrounding flow through a shear layer, until its excess momentum is dissipated.

In order to experimentally investigate the htc distribution upstream and downstream

of the seal, a new experimental facility was developed. In this chapter, the experimental setup and test method are described in detail, and accuracy considerations are included. The methodology employed to derive the htc from the experimental data is explained with reference to an analogous computational fluid dynamics (CFD) model. The work demonstrates the importance of the choice of an appropriate driving gas temperature for the definition of the htc , especially downstream of the seal. Finally, experimental htc maps are presented for a blow-down leaf seal operating over a range of engine representative differential pressures.

4.1 Seal Static Thermal Test Facility (SSTTF): Rotor Thermal Characterization

4.1.1 Description of the experimental facility

The new test rig is designed on the basis that the flow field in the close vicinity of the seal is largely a function of the axial flow, and thus a rotating test section is not required – a comparison is made to other data from the literature when results are presented. Furthermore, the temperature difference in the flow field need not be generated by frictional heating, and thus a heated flow is introduced using a transient electrical heater. A photograph of the rig and cross-sectional schematic of the annular test section is shown in Fig 4.2.

The test section consists of two static concentric cylinders forming a pressure vessel in which a single seal is housed. The outer cylinder (casing), is made from aluminium alloy while mild steel is used for the inner cylinder (which represents the engine rotor). Air at pressures of up to 10 bara is supplied to the vessel through four inlet pipes. Internal buffers eliminate jetting flow and the air passes through the upstream annulus and thence through the seal. An insert made using rapid prototyping can be used to introduce a swirl component in the flow upstream of the seal (however, it should be noted that the swirl generator was not used in the present test campaign). The flow exhausts to ambient pressure through a further perforated flow restriction and four exit

holes. The facility may be used to test labyrinth, brush and leaf seals.

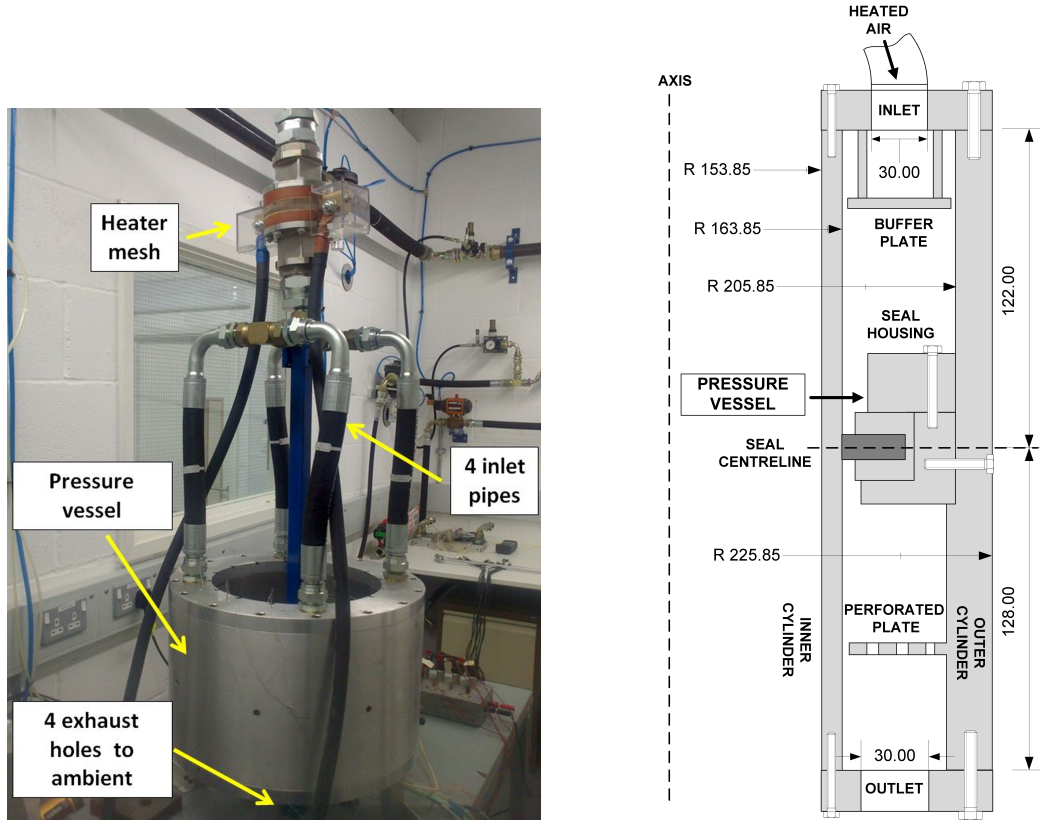


Figure 4.2: Seal Static Thermal Test Facility (SSTTF)

A temperature rise in the incoming flow is required to drive convective heat transfer. This is achieved using a heater mesh upstream of the inlet. The mesh is powered from a high current 0 – 32V DC power supply. For the range of leakage mass flow rates of the leaf seal tested, a typical temperature rise of 20 – 30 K is achieved. To prevent the risk of high temperatures at the brass terminals of the mesh, tightening torques were carefully considered, and the resulting junction temperature assessed using an infrared camera during commissioning. A detailed risk assessment and standard operating procedure of the rig is included in Appendix A.

4.1.2 Instrumentation

Convection can be simply defined using Newton's law of cooling:

$$q = htc (T_{ref} - T_w) \quad (4.1)$$

where q is the heat flux, T_{ref} an appropriate fluid reference temperature, and T_w is the solid wall temperature. This equation introduces the concept of convective heat transfer coefficient, htc , which is a strong function of the hydrodynamic boundary layer, and scales with temperature difference.

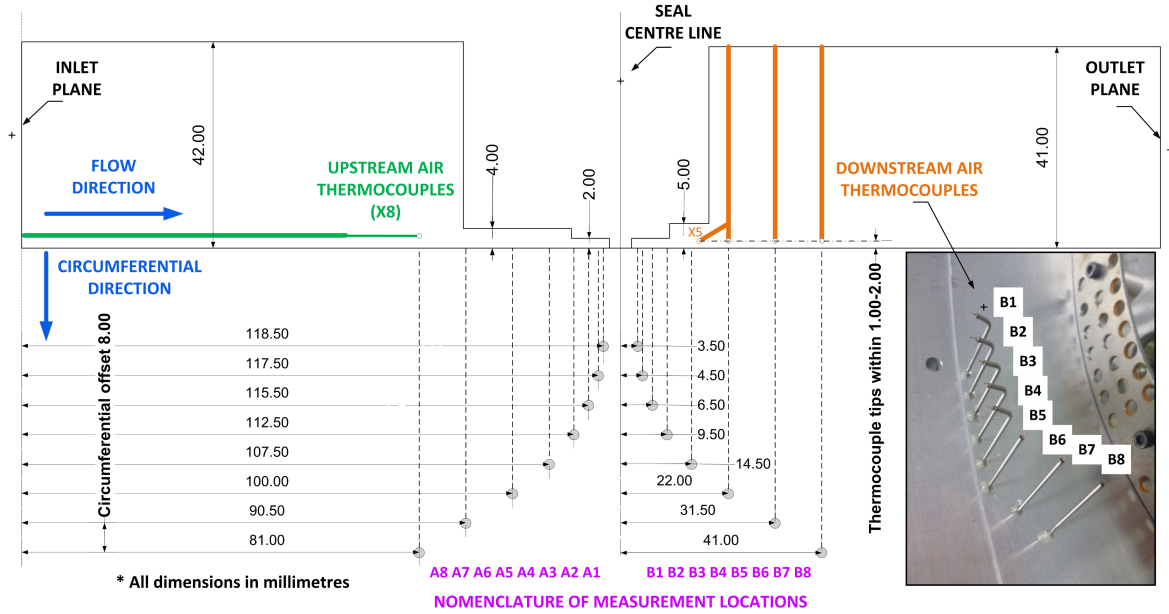


Figure 4.3: Schematic of the test facility thermocouple instrumentation. The gas thermocouples are represented by the green and orange lines on the diagram, and the position of the surface thermocouples and thermocouples embedded 1.5 mm below the surface by the circles in the lower zone of the figure, which shows a part projected surface of the inner cylinder

To derive the htc , three quantities thus have to be determined: fluid to solid heat flux q , the fluid reference temperature T_f (Eq 4.1) and wall temperature T_w . The rig can be considered axisymmetric in the vicinity of the seal, and all the instrumentation is located in one circumferential quadrant of the test rig. Temperatures are measured at a total of sixteen axial locations (eight upstream and downstream). These locations and

the nomenclature employed throughout this Chapter are shown in Fig 4.3. Note that the internal volume is shown in the diagram, and the orientation is rotated 90° with respect to Fig 4.2.

The seal itself can be considered a strong heat sink due to the expected high heat transfer within the filament pack. Therefore, air temperature is measured locally upstream and downstream of the seal to account for any potential heat loss. Eight K-type gas thermocouples ($\sim \varnothing 25 \mu\text{m}$ diameter wire) were mounted upstream of the seal at the same nominal radial distance from the rotor surface, but distributed circumferentially around the annulus such that they are aligned with the surface thermocouples. The eight downstream gas thermocouples are positioned axially $14.5 - 41 \text{ mm}$ from the seal centreline, and as far as possible in close proximity to the equivalent surface thermocouple (see Fig 4.3). In fact, to avoid blockage gas thermocouples B1 - B5 were situated at the same axial location.

OMEGA K-type surface thermocouples ($13 \mu\text{m}$ in thickness) were used to measure surface temperature at the sixteen measurement locations. These are fast response thermocouples which were glued to a substrate made of an *Upilex* polyimide insulating material. A rectangular pocket of 0.66 mm in depth was milled into the inner rotor surface across the measurement sector to accommodate the *Upilex* platform. Underneath this *Upilex* slab and embedded into the metal inner rotor, a second set of K-type thermocouples measure the metal temperature 1.5 mm under the surface thermocouple. This arrangement, depicted in Fig 4.4, forms a sensor that can be used to measure the heat flux exchanged between fluid and solid. According to Eq 4.1, a positive heat flux occurs when heat is transferred from the airflow into the rotor.

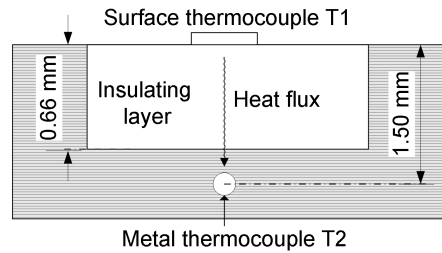


Figure 4.4: Schematic of the heat flux sensor employed (not to scale)

The sensor described is conceptually similar to the direct-heat-flux gauge developed by Epstein et al. (1986) and Piccini et al. (2000). Epstein et al. used thin-film resistance thermometers to measure the temperature on both sides of a thin (25 μm) polyimide sheet, which is adhesively bonded to the test specimen. Piccini et al.'s design employs a thin-film resistance thermometer to measure the top surface temperature of an insulating layer which is bonded to a metal substrate, within which a thermocouple is installed. In the current study the use of thin-film gauges was found to be unreliable, and, as a very high frequency response was not required, a decision was made to use surface and embedded thermocouples.

At low frequencies, these sensors provide the heat transfer rate as the product of the measured temperature drop across the insulating layer per unit thickness and its conductivity (Oldfield (2008)):

$$q = \frac{k}{l} (T_1 - T_2) \quad (4.2)$$

where the subscripts 1 and 2 refer to the local thermocouple positions (see Fig 4.4). The accuracy of direct-heat-flux gauges depends on the existence of a significant temperature difference across the insulating layer. This temperature drop must be large enough to overcome the noise in both thermocouple signals. To achieve this, a sandwich of 6 layers of 50 μm thick *Upilex* and 60 μm thick glue, was used.

Unless the *htcs* are spatially constant, there must be some lateral conduction in the model. Ling et al. (2004) showed that heat conduction can only be assumed to be one-dimensional under the related thermal boundary condition of a step change in heat flux when, $\frac{\partial^2 q}{\partial x^2} \ll \frac{q}{2\alpha t}$ and $\frac{\partial^2 q}{\partial y^2} \ll \frac{q}{2\alpha t}$, where the x direction is parallel to the direction of the freestream flow, the y direction is at right angles to the direction of the freestream flow on the wall plane, and the z direction is into the wall. The term q refers to heat flux, α is the thermal diffusivity of the material and t stands for time. These criteria provide a good test as to whether further post-processing may be necessary. No further processing was required in the current study.

The thermal properties of the combined *Upilex* and glue structure were presented by Piccini et al. [2000]. *Upilex* and glue can be treated as a single homogenous material due to the similarity in their thermal properties (Billiard et al. (2002)). Examples where direct-heat-flux gauges have been employed successfully can be found in Jones et al. (2000) and Guo et al. (2002).

Seal pressure difference is calculated using 0 – 7 barg pressure transducers connected to tappings located in the pressure vessel upstream and downstream of the seal. Leakage mass flow rate is calculated using a BS1042 orifice plate (British Standards Institution, (1981)) located upstream of the heater mesh. This pressure drop is measured by a 0 - 350 mbarg differential pressure transducer to improve measurement accuracy. Since the temperature of the flow at the orifice plate is also required for the mass flow rate calculation, a single K-type thermocouple is installed at that location.

In total, 49 thermocouple and 4 pressure transduce signals are recorded. These data are acquired using a PC equipped with 16-bit data acquisition cards with a data sampling frequency of 16Hz per channel. *LabVIEW* is used to monitor the rig pressures and temperatures, adjust the operating conditions and record data throughout the tests.

4.1.3 Accuracy considerations

The measurands and their precision and bias errors are set out in Table 4.1. The uncertainty in the direct measurements can be obtained from the accuracy specified in the transducer data sheets and the uncertainty in the data acquisition system. The overall uncertainty can be found by combining bias and precision errors using the root-sum-square equation (RSS). To calculate the uncertainty in parameters that are derived from measured quantities, the Jitter technique described by Moffat (1988) was employed. Table 4.2 shows the uncertainty estimated for the mass flow rate, heat flux and heat transfer coefficient (for a given chosen driving gas temperature). The uncertainty in these derived quantities was computed by perturbing the input variables involved in their derivation with their corresponding overall errors. The uncertainty thus obtained is then expressed as a percentage deviation from the value that results if the input variables

are not perturbed. The effect of thermocouple bias error was to some degree mitigated by using surface and gas thermocouples each from common batches of thermocouple wire. In the postprocess of the data, the thermocouple offsets that exist at the start of a test were removed from each data set, resulting in a common starting temperature. Only the absolute orifice temperature is required to determine mass flow rate, all other measurements use temperature difference alone.

	Precision	Bias
Pup orifice plate meter	$\pm 32\text{mbar}$	$\pm 4.62\text{ mbar}$
Delta P orifice plate meter	$\pm 0.66\text{ mbar}$	$\pm 0.22\text{ mbar}$
T orifice plate meter	$\pm 0.3^\circ\text{C}$	$\pm 3.6^\circ\text{C}$
Prig	$\pm 32\text{mbar}$	$\pm 4.62\text{ mbar}$
Atmospheric Pressure	$\pm 0.1\text{mbar}$	$\pm 0.1\text{mbar}$
Surface thermocouple, T1	$\pm 0.3^\circ\text{C}$	$\pm 3.6^\circ\text{C}$
Upilex thickness	$\pm 50\mu\text{m}$	-
Metal thermocouple, T2	$\pm 0.3^\circ\text{C}$	$\pm 3.6^\circ\text{C}$
Initial temperature	$\pm 0.3^\circ\text{C}$	$\pm 3.6^\circ\text{C}$

Table 4.1: Measurands and uncertainties

	RSS error
Mass flow rate	1.23 %
Heat flux	4.26 %
Heat transfer coefficient	7.11 %

Table 4.2: Derived value RSS errors

4.2 Rotor Convective Heat Transfer Coefficient Calculation

4.2.1 The choice of the driving temperature

The primary difficulty in calculating the rotor convective heat transfer coefficients (htc) upstream and downstream of the seal is the choice of the driving gas temperature. Different reference fluid temperatures are used dependent on the application required. In the case of a low speed flow of high heat capacity, a uniform upstream flow temperature is typically employed. The heat transfer coefficient at any point on the surface is then analogous to local heat flux, providing an indication of the heating expected without significant further analysis. Where the mass flow rate is limited and the heat transfer from surface to fluid causes significant change in its temperature, a mixed bulk temperature formed by conducting an energy balance in the streamwise direction is considered more appropriate. This allows scaling to engine conditions provided the non-dimensional thermal boundary layer profile is similar.

In the current research, the flow field is complicated, compressible and there is a very significant heat transfer between the flow, the surroundings, and the seal. The high speed flow jetting from the seal in the downstream region creates a local drop in static temperature, and changes the thermal boundary layer profile. Clearly a simplistic definition of the fluid reference temperature (T_{ref} , Eq 4.1) in this case could lead to physically meaningless htc distributions. Many previous workers (Kays et al. (2004)) have identified the local adiabatic wall temperature as an appropriate reference temperature for convective heat transfer in such situations:

$$T_{aw} = T_s + r \frac{v^2}{2 c_p} \quad (4.3)$$

where r is the local recovery factor in the flow field. This ensures the heat transfer coefficient is scalable to any subsequent engine condition, but is awkward for engine

designers to employ in streamwise energy balance codes, typically used in preliminary engine design.

The problem for the current research is demonstrated in Fig 4.5 by considering local experimental gas and surface temperature histories at two points immediately downstream of the seal. The surface thermocouple corresponds to the first downstream location shown in Fig 4.3 (B1), at 3.5 mm. Here the unheated flow is switched to pass through the test rig at time ~ 5 seconds. After an initial compressive heating event, the surface temperature is seen to drop more quickly than the gas temperature locally measured 1 - 2 mm from the surface. It should be noted that this gas thermocouple is located 14.5 mm downstream of the seal exit because of lack of access nearer to the seal. This would result in a negative h_{tc} , and thus an overestimation of the driving gas temperature. Clearly there must exist strong gradients in the flow static temperature in this region. Unfortunately, the physical space available for the installation of gas thermocouples means that only simple thermocouples can be installed in the flow field, which generate a total temperature with high uncertainty in the recovery factor. Thus, for each location in the high speed flow, the flow reference temperature (the adiabatic wall temperature) must be inferred by other means.

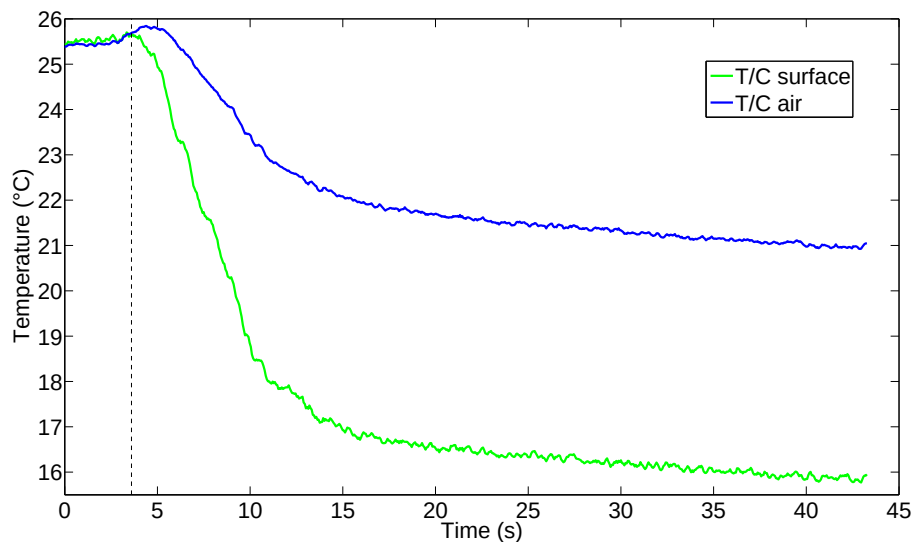


Figure 4.5: Surface and gas temperature histories at the start of an unheated experiment, 2 bar seal differential pressure

4.2.2 Heat transfer coefficient calculation method

In a typical experiment, the pressure difference across the seal is set and kept constant. This results in a near steady velocity field during the entire test. It is noted that the addition of heat to the flow increases the velocity, viscosity, and thermal conductivity, but lowers the density of the air. Analysis of the resulting variation in Reynolds number and conductivity allowed the expected variation in htc to be determined for a 55 °C rise in gas temperature. For a case of laminar heat transfer, where $Nu \propto Re^{0.5}$, a rise of up to 5% in htc is expected. For fully turbulent flow, where it might be expected that $Nu \propto Re^{0.8}$, a drop in htc of 1.5% was found. For the case reported in this experimental campaign, where $Nu \propto Re^{0.67}$, htc increases by 1.3% over the course of the test, which is well within the experimental error. After the heater is switched on, the flow progressively heats up the solid surface. When the heat flux data extracted are plotted against $T_{ref} - T_w$, linear regression can be performed, with the slope of the resulting best fit line corresponding to the htc (Eq 4.1). For a step change in gas temperature, heat flux can be plotted against T_w alone and htc obtained from the gradient of the line. However, in the current experiments, T_{ref} , which changes throughout the test, must be included for the gradient to remain linear. Thus, an available local fluid temperature must be chosen as T_{ref} . Provided that there is a uniform offset (constant throughout the test) between the T_{ref} used, indicated in Fig 4.6 by T_{tc} , and the appropriate local driving temperature, indicated in Fig 4.6 by T_{driv} , regression using an arbitrary T_{ref} yields the same htc as that using an adiabatic wall temperature. When the fluid reference temperature T_{ref} coincides with the local adiabatic wall temperature, the intercept of linear regression with the temperature axis should pass through the origin. By considering when this would occur, a true adiabatic wall temperature can be inferred. This corresponds to the black line regression in Fig 4.6.

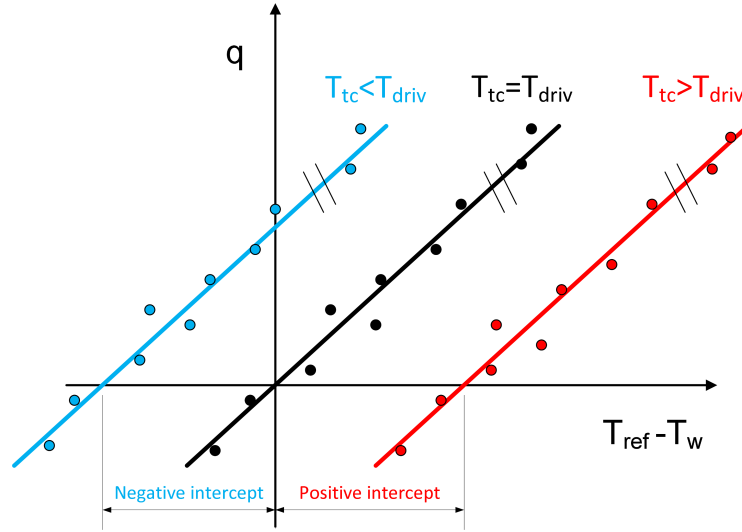


Figure 4.6: Schematic of the data reduction methodology

4.2.3 CFD investigation of heat transfer reduction technique

To help ascertain the variance in the driving gas temperature, a basic CFD study was conducted. A schematic of the CFD domain considered is shown in Fig 4.7. The main geometrical features of the experimental facility are represented in a 2D axisymmetric model. The outlet from the experiment was extended axially to improve convergence of the solution. Pressure boundary condition of 2 bara and 1 bara were imposed at the inlet and outlet of the domain.

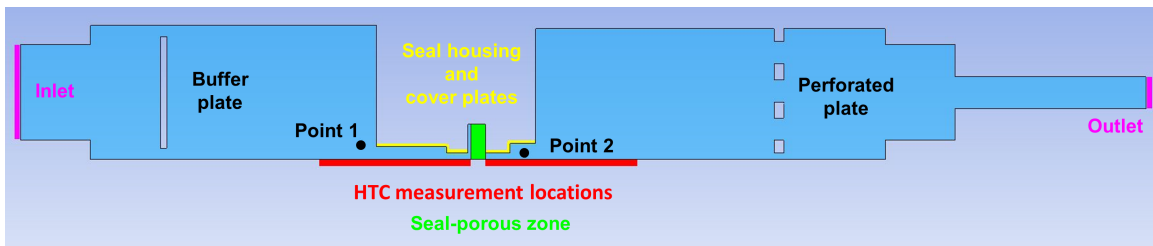


Figure 4.7: CFD domain and boundary conditions. Point 1 and 2 correspond to representative locations for the experimental gas thermocouples in the test rig

The blow-down leaf seal geometry tested experimentally was simulated. The leaf pack was modelled as a porous medium. As shown in Chapter 2, the porous medium approach has been used in brush seal modelling by many authors (see for example Bayley and Long (1992), Chew et al. (1995), Dogu (2005), Qiu and Li (2013), and Flouros et al.

(2014)). Here a separate fluid zone is defined where a resistance source term is added to the momentum equations. This term models the pressure drop experienced by the fluid as it leaks through the seal and is given by:

$$P_i = -(a_i \mu v_i + b_i \frac{1}{2} \rho v_i |v|) \quad (4.4)$$

where P_i is the source term for the i th (x - or y -) momentum equation and $|v|$ the magnitude of the velocity. The first term of Eq 4.4 models viscous losses while the second accounts for inertial losses. In the present 2D model, the two inertial coefficients are set to zero and the viscous terms are chosen to match the mass flow leakage found experimentally for the imposed seal pressure difference. A value of 0.3 was set as the seal porosity.

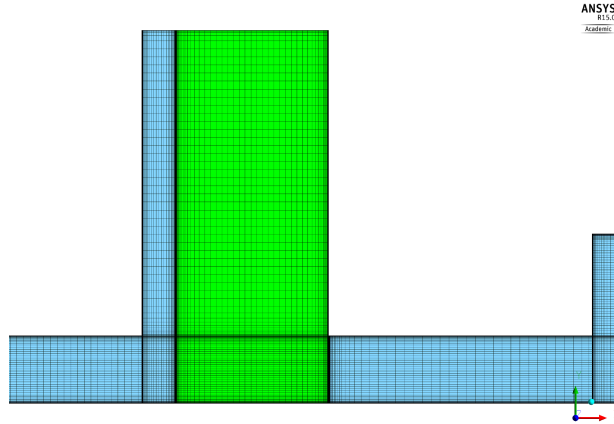


Figure 4.8: Computational mesh. The green region corresponds to the leaf pack porous zone

The computational domain was discretized using a multiblock structured orthogonal mesh built using *ANSYS ICEM*. The wall distance of the first row of nodes in the axial region where heat transfer coefficients need to be determined (red and yellow lines in Fig 4.7) was chosen to achieve a value of $y^+ < 1$. The mesh was also further refined in the transition zones between the porous and fluid zones. Detail of the mesh in the seal pack vicinity is shown in Fig 4.8. The blow-down configuration of the seal modelled involves a large upstream coverplate gap (approximately 1 mm in axial width). Given the small size of the downstream coverplate gap (of the order of microns), it was assumed to be

zero in the model, although this is physically unrealistic. The mesh contained a total of ~120,000 elements.

The steady Reynolds Averaged Navier Stokes Equations (RANS) are solved using *ANSYS Fluent* pressure-based solver with 2nd order discretization and realizable k- ϵ turbulence model with enhanced wall treatment. Other turbulence models were also considered (k- ω and Spalart-Allmaras), but comparison of the resulting flow solutions with experimentally observed flow features proved k- ϵ as the model that better captures the flow behaviour. A mesh dependency study was performed where the mesh was progressively refined in the region where *htcs* are studied until no further refinement resulted in changes in wall temperature.

To obtain the *htc* distribution, constant heat flux boundary conditions were imposed along the walls upstream and downstream of the seal, as shown in Fig 4.7 (red lines). Four cases were run with heat fluxes of 500, 1500, 2000 and 2500 W/m^2 , representative of values during typical experiments. The *htc* distribution along the rotor surface was then obtained through linear regression of the $T_{ref} - T_w$ versus heat flux from all of these tests. An example of the *htc* calculation method is shown in Fig 4.9. In the figure, it is clear that regression of example data in the low speed upstream region passes through the origin, whereas in the downstream region, the temperature at the intercept is negative. This indicates the existence of strong temperature gradients in the flow solution downstream of the seal, while on the upstream side, capturing the driving gas temperature presents less difficulty. The variation in amplitude between the temperature differences required in both cases to attain a given heat flux is also notable. This contrast implies different *htc* levels between the two regions. Several arbitrary fluid temperatures extracted from planes offset from the inner radius and from discrete point locations have been used to investigate the effect of the choice of T_{ref} on the resulting *htcs*. These are compared in Table 4.3.

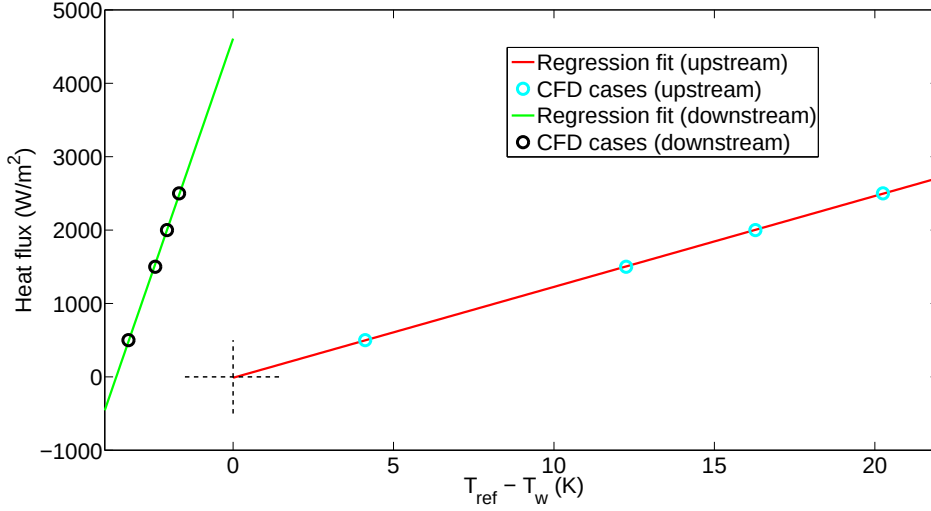


Figure 4.9: Example of linear regression for CFD h_{tc} calculation

Three different sets of boundary conditions sets were used, progressively moving from an ideal case to a situation more representative of an experimental run. Table 4.3 summarizes all the cases run, showing their corresponding boundary conditions and the reference fluid temperature considered. In the first set of simulations (Set 1), the inlet temperature was 325 K and all boundaries were adiabatic except for the two regions where h_{tc} s were to be determined (red lines in Fig 4.7). From these results, four h_{tc} distributions were obtained using alternative definition of fluid reference temperature. At every location where the h_{tc} was calculated, the fluid temperatures 0.5 mm, 1 mm and 1.5 mm perpendicularly from the rotor surface were employed. Additionally, the fluid temperatures at points 1 and 2 on Fig 4.7, which are representative locations for the experimental gas thermocouples in the test rig, were used for the upstream and downstream h_{tc} s respectively.

The second set of simulations (Set 2) differs from the first in that the four heat flux cases were achieved at different inlet temperatures, as would be the case in a real experiment: 303K & 500 W/m², 313 K & 1500 W/m², 318 K & 2000 W/m², 323 K & 2500 W/m². Here, the h_{tc} was calculated using only the fluid temperature at Points 1 and 2. The temperature obtained at Point 1 was employed as T_{ref} in all the linear regressions of the upstream measuring locations. Similarly, the temperature at Point 2

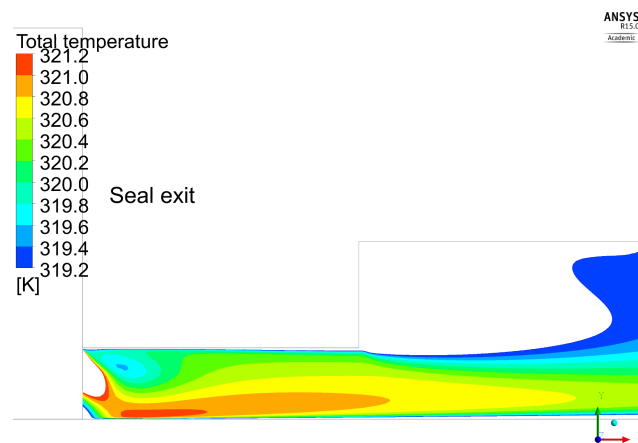
was used in the calculation of all the downstream *htcs*. The changes in inlet temperature result in small changes in the velocity field.

The final set of boundary conditions (Set 3) removes the constraint of adiabatic coverplates and clamping ring (yellow lines in Fig 4.7). Instead, a constant heat flux was imposed equal to that at the inner cylinder wall. This was done to investigate the effect of a further drop in fluid total temperature along the measurement regions.

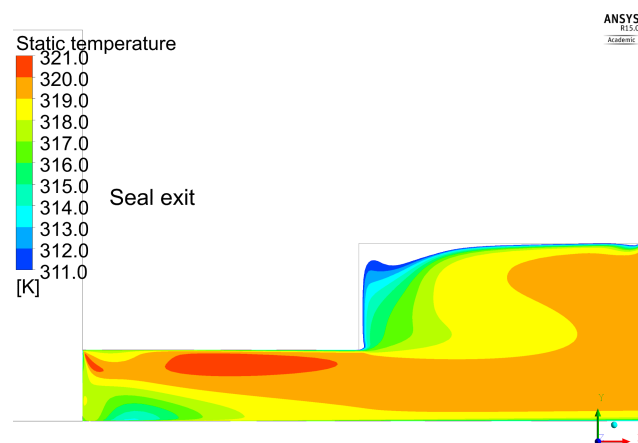
Set 1		Set 2		Set 3	
$T_{inlet}(K)$	$q_{rotor\ wall}$ W/m^2	$T_{inlet}(K)$	$q_{rotor\ wall}$ W/m^2	$T_{inlet}(K)$	$q_{rotor\ wall}$ W/m^2
325	500	303	500	303	500
325	1500	313	1500	313	1500
325	2000	318	2000	318	2000
325	2500	323	2500	323	2500
Outer Cylinder	Adiabatic	Outer Cylinder	Adiabatic	Outer Cylinder	As inner rotor
T_{ref}	0.5, 1, 1.5 mm from rotor surface, Points 1&2	T_{ref}	Points 1&2	T_{ref}	Points 1&2

Table 4.3: CFD heat flux and temperature boundary conditions

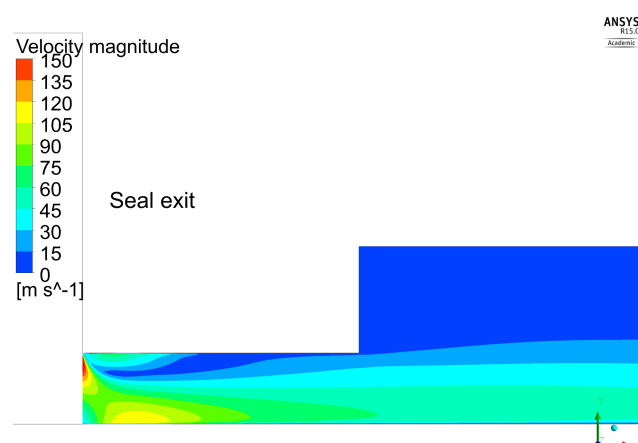
Fig 4.10 shows contours of velocity, static and total temperature downstream of the seal from the simulation with $q = 2500\ W/m^2$ of Set 3. There is a notable distribution of both total and static temperature associated with the flow into the downstream region. It can be seen that the variation in static temperature of $\sim 6\ ^\circ C$ is larger than that of the total temperature $\sim 2\ ^\circ C$ and is associated with regions of high and low velocity. Fig 4.11 shows the variation in static temperature with radial distance at different axial distances from the seal exit plane. Sensibly, both total and static temperature are lower than the inlet static temperature of 323 K. A gradual drop in total temperature with axial position is further associated with the heat loss through the walls of the model.



(a) Total temperature



(b) Static temperature



(c) Velocity

Figure 4.10: Total and static temperature and velocity distributions downstream of the seal, Set 3, $q = 2500 \text{ W/m}^2$, $T_{\text{inlet}} = 323 \text{ K}$

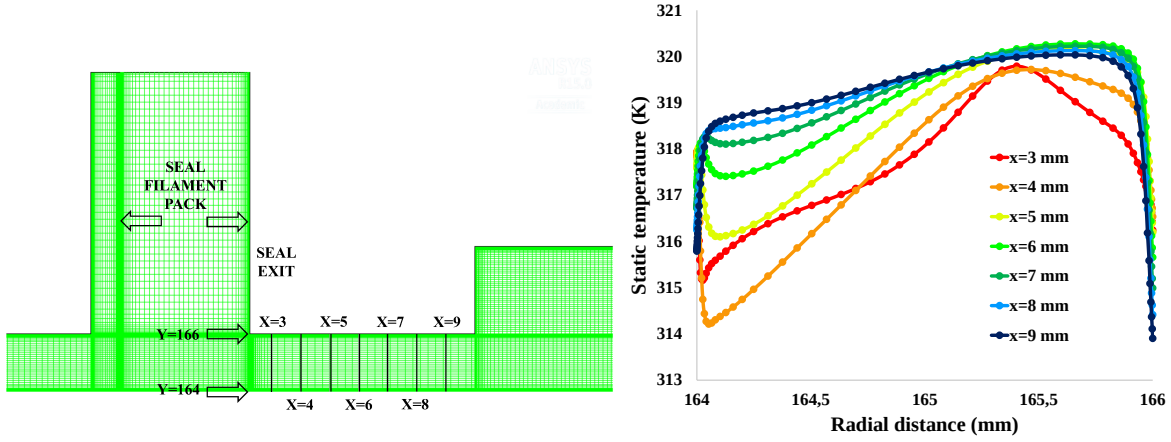


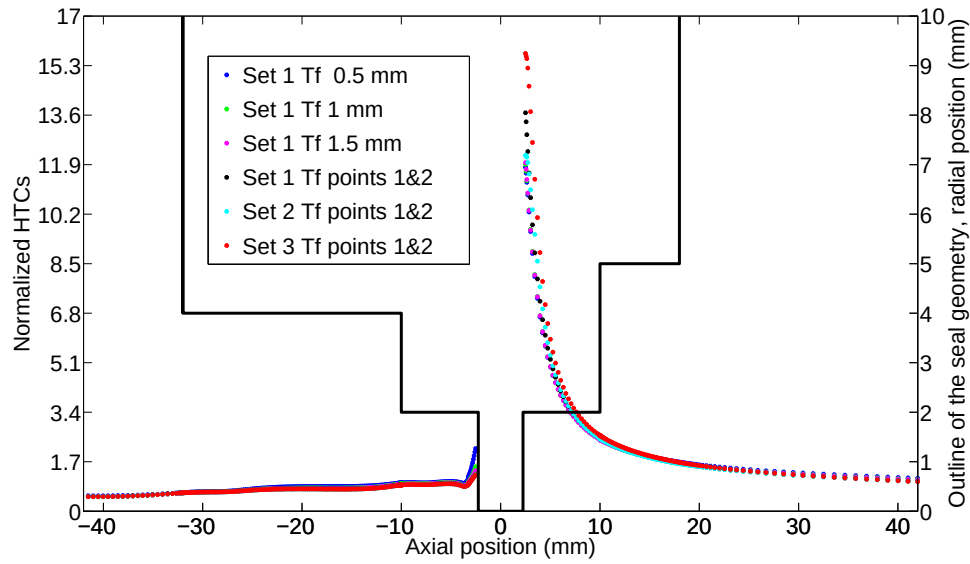
Figure 4.11: Static temperature variation downstream of the seal, Set 3, $q = 2500 \text{ W/m}^2$, $T_{\text{inlet}} = 323 \text{ K}$ (refer to Fig 4.10)

The htc distributions derived from all the simulations conducted are shown in Fig 4.12a. The htc data are all normalized by the same baseline value of 189.9 W/m^2 . This is found from the annular flow heat transfer coefficient correlation of Kays and Leung (1963). Here the annulus inner radius is matched to the rotor radius and the gap between the inner and outer radius is set to the coverplate gap height. The Reynolds number used is $Re = 10,000$, which is close to the experimental value when the seal is operated at 1 bar pressure difference with atmospheric outlet conditions.

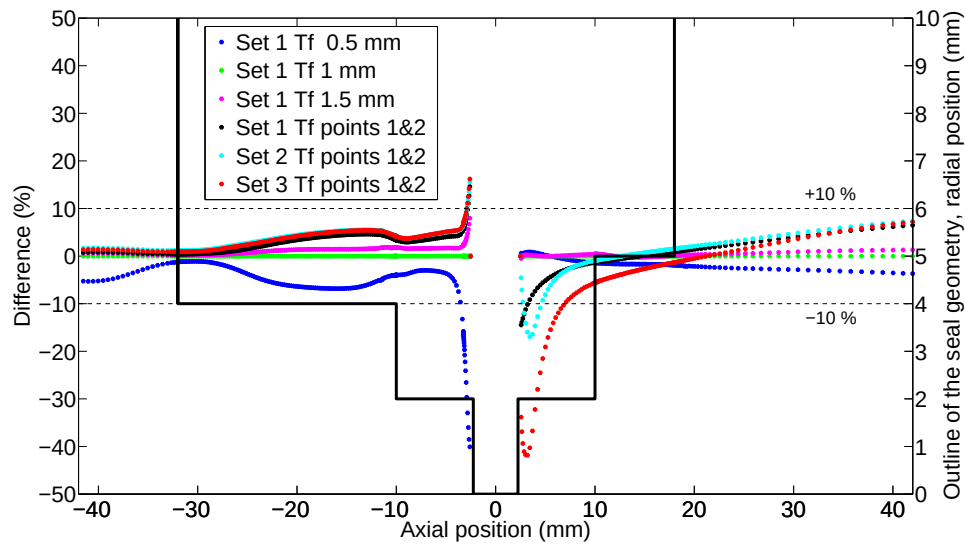
As the data in Fig 4.12a appear nearly coincident, Fig 4.12b is included to highlight the local differences in htc . Here, at each axial location, local htc values are normalised with respect to the local htc derived using $T_{ref} = T_f$ at 1 mm from the wall. This is then presented as a percentage deviation. Clearly errors at a high initial normalised htc are more significant.

All htc distributions shown in Fig 4.12a are very similar in level upstream of the seal, rising gradually as the flow velocity increases towards the seal, with a final sharp increase due to the flow acceleration occurring under the coverplate and then further to accommodate the changed porosity through the leaf pack. The htc level is initially an order of magnitude higher at the exit from the seal and follows an exponential-like trend downstream of the seal exit. Downstream of the rear coverplate, the htc is seen to be low, and steadily decreases, as would be expected for a thickening boundary layer. It is,

however, notable that in this region (from 10 to 41 mm), the level remains 2 – 3 times that on the equivalent upstream side of the seal (from -41 to -10 mm).



(a) Rotor htc distribution



(b) Normalised variation in reported rotor htc

Figure 4.12: Rotor htc distribution derived using linear regression, Set 1 – 3 (see table 4.3)

Considering the variation in htc , the largest difference is found when using the data in Set 3, and Points 1 and 2 as reference temperatures. This is the case where the

coverplates and clamping ring were not adiabatic. This results in larger variations in the total temperature of the system. However, in these simulations a heat flux equal to that on the inner cylinder wall was imposed on all of these surfaces. This is highly conservative since lower heat transfer would be expected here. Thus, the variation seen may be considered to be a worst case scenario.

In conclusion, the $htcs$ downstream of the seal strongly depends on the location where the reference fluid temperature is taken. However, the use of a linear regression can be used to remove this uncertainty largely.

4.3 Experimental Data Reduction

The equivalent htc distribution for a leaf seal in a strong blow-down configuration was tested experimentally. The htc distribution along the rotor surface was obtained at six seal pressure differences, ranging from 0.5 bar to 3 bar in steps of 0.5 bar. Fig 4.13 shows the sealing performance of the seal, normalized with respect to the mass flow rate at 1 bar pressure difference (0.1 kg/s).

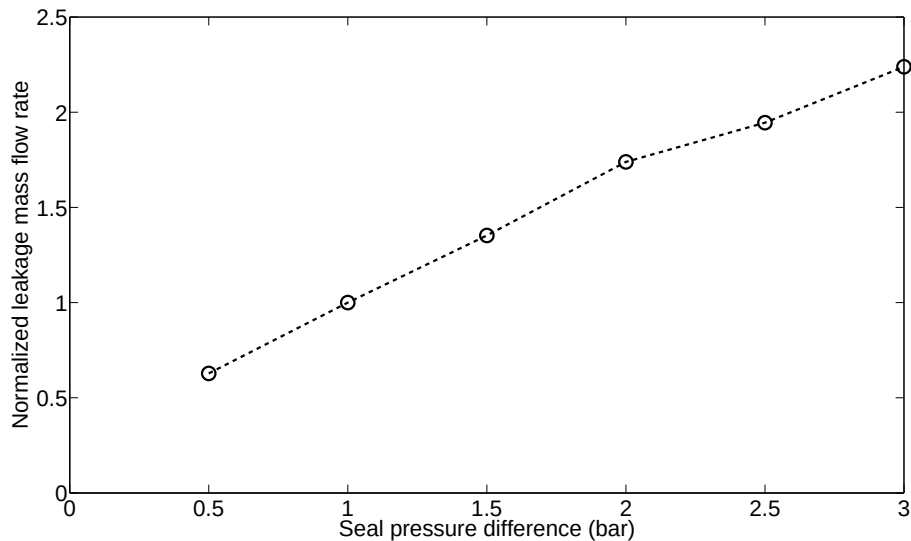


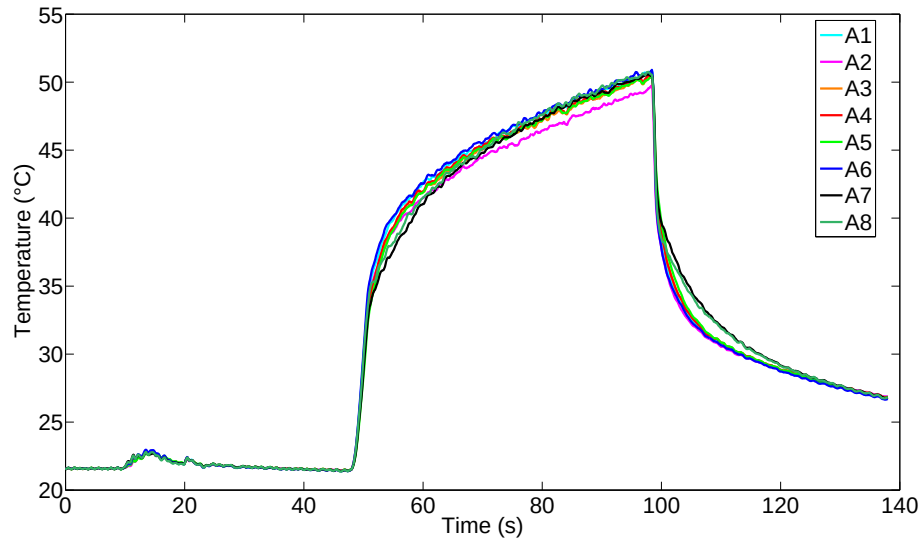
Figure 4.13: Sealing performance of the leaf seal tested

In a typical test, the pressure difference across the seal is first established using an electronic pressure regulator. This pressure difference remains constant throughout

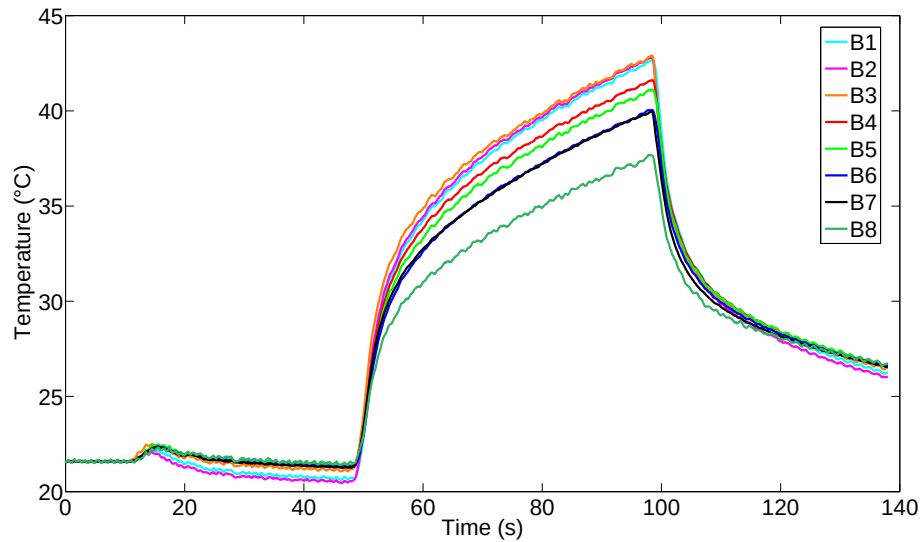
the test, which generates a steady-state flow field (if the transient thermal effects are neglected). Next, power is supplied to the heater mesh, resulting in a transient heating of the system, which provides us with a wide heat transfer data range that can be used to derive *htcs*. Typical gas temperature histories upstream and downstream of the seal are shown in Fig 4.14.

Notable features in the temperature field include the small temperature rise introduced by compression of the test gas when flow is instigated (~ 11 s), followed by a decrease in temperature from the expansion of the high pressure gas through the seal (especially on the downstream side). The heater is switched on at $t = 48$ s. While a step increase in temperature occurs at the heater, because of the considerable thermal mass of the test rig (see Fig 4.2), near the seal there is a slower temperature rise followed with an asymptotic approach to the inlet temperature. The temperature field is nearly spatially uniform upstream of the seal but downstream there is considerable variation, dependent on the radial and axial location of the thermocouple. This can primarily be accounted for by the large variation in static temperature driven by the high speed flow, as observed in the CFD study. Furthermore, in the experiment the seal itself absorbs heat, reducing, non-uniformly, the total temperature of the flow. Heat losses of 600 W – 1 kW are typical across the seal. Finally, as noted above, the thermocouples in the high speed flow are unlikely to report the true total temperature of the flow that drives convective heat transfer at the rotor surface.

Some circumferential non-uniformity was initially thought to be observed in thermocouples B1 to B5, which are located at the same axial position. However, clocking of the seal through 180° relative to the rig, and manipulation of the instrumentation system, showed no change in the distribution, suggesting that the thermocouples were in a region of high radial temperature gradient where small height variations resulted in temperature differences of up to 4°C occurring. It was therefore decided that the average of these gas thermocouples would be used in the calculation of the *htc* at these five surface thermocouple locations. Thermocouples B6, B7, and especially B8, are notably lower, as they are in flow which has lost more heat to the test rig, and has a lower total



(a)



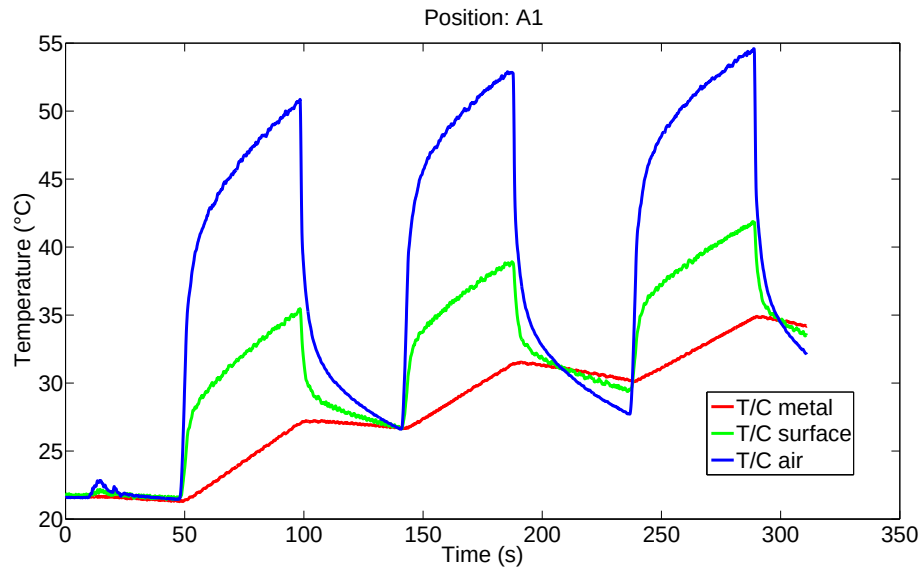
(b)

Figure 4.14: Gas temperature upstream (a) and downstream (b) of the seal (1 bar seal pressure difference)

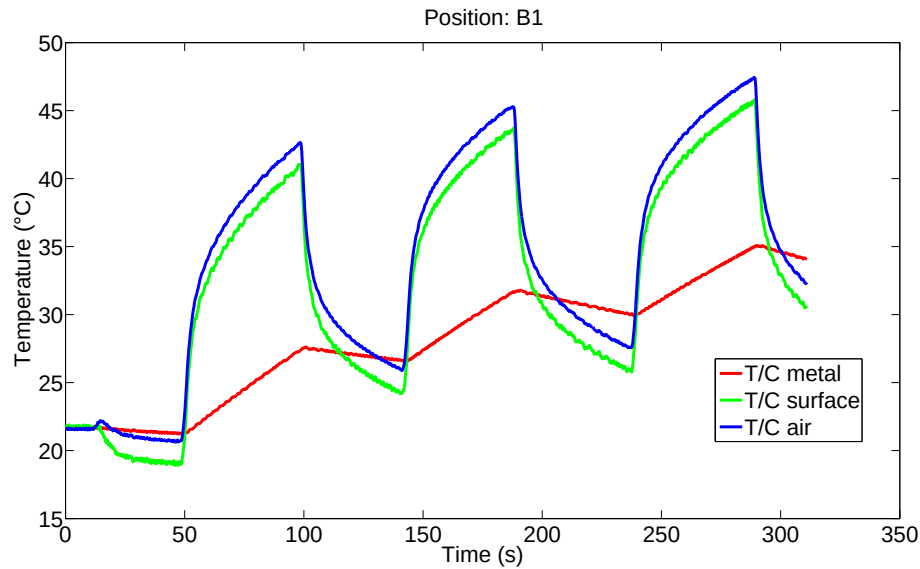
temperature.

Fig 4.15 shows typical gas, surface, and metal temperature histories at measurement locations upstream and downstream of the seal. The heater mesh was powered three times in this example. As the gas temperature rises, a temperature difference develops between the surface and the embedded wall thermocouples, which is proportional to the heat flux (Eq 4.2). When the heater mesh subsequently is powered off, the gas temperature drops towards the inlet temperature. Provided the gas thermocouple accurately reflects the temperature driving convective heat transfer, where the gas thermocouple temperature equals the surface temperature there is no heat flux to the surface. The corresponding embedded surface thermocouple is also then very close to the surface temperature, as observed in Fig 4.15a on the upstream side of the seal. Downstream of the seal, however, this is no longer the case and in Fig 4.15b, when the surface and embedded thermocouple temperatures are equal, there is still a positive temperature difference between the gas and surface thermocouple. Here, the local driving gas temperature must be lower than the recorded temperature, and if used without adjustment for the downstream thermocouples, the reported h_{tc} would vary between positive and negative values.

For the calculation of h_{tc} , linear regression of heat flux (determined by Eq 4.2) versus gas to surface thermocouple temperature difference was employed. The gradient of this regression is badly conditioned when the temperature difference range is very small. Hence, to improve the quality of the regression, the heater was switched repeatedly with different power settings during each test. Between heating events, the working section cools, and these intervals can also be used to gather heat transfer data. This is depicted in Fig 4.16a, where the heat flux history from a measurement location upstream of the seal is shown. In Fig 4.16b the data are plotted against gas to surface temperature difference and the regression used to calculate h_{tc} is indicated. It is notable that here, the intercept of the regression is approximately zero, indicating that the true driving temperature is captured in this example.

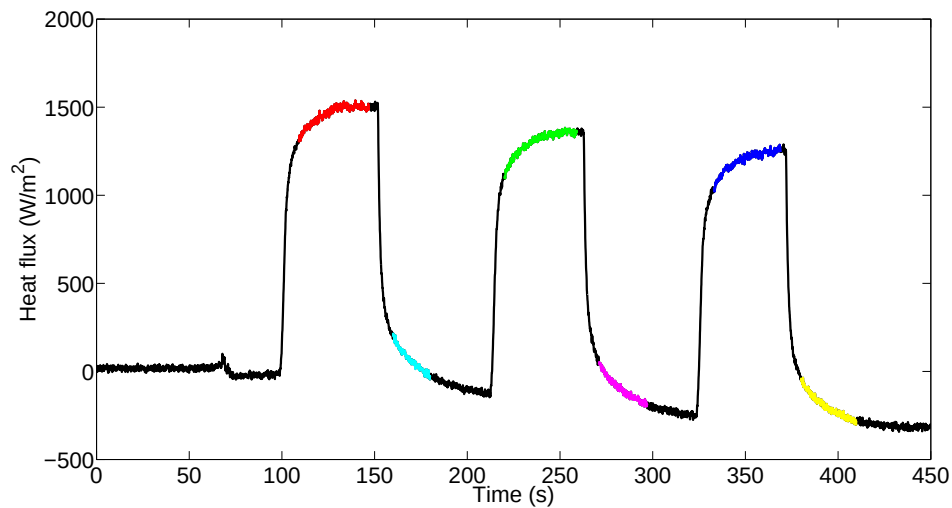


(a) Measurement location A1 (upstream side, see Fig 4.3)

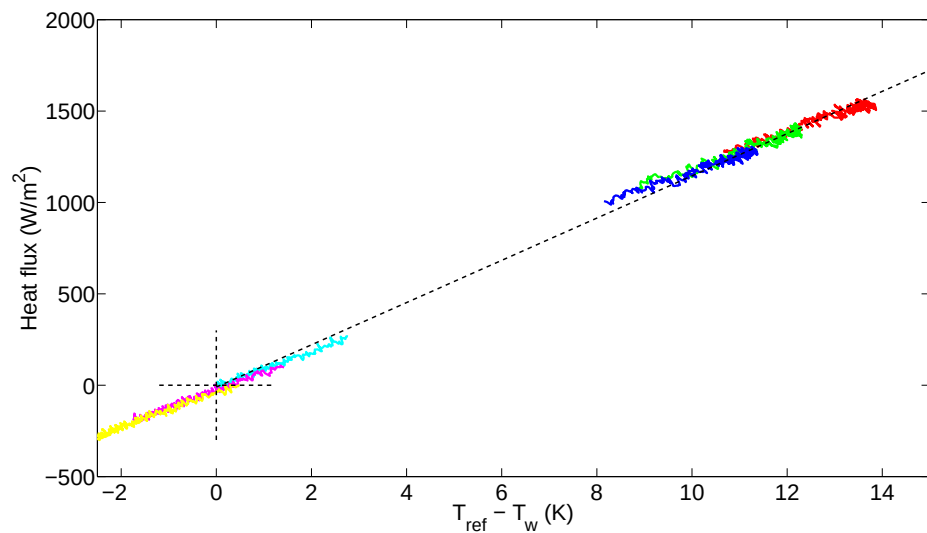


(b) Measurement location B1 (downstream side, see Fig 4.3)

Figure 4.15: Gas, surface and embedded thermocouple temperature traces



(a) Heat flux measured during a typical test run. Each coloured section forms a set of data points used in the htc derivation



(b) Typical regression fit, each coloured section corresponds to one of the sets of data points showed in 4.16a

Figure 4.16: Example of experimental strategy for measuring rotor htc

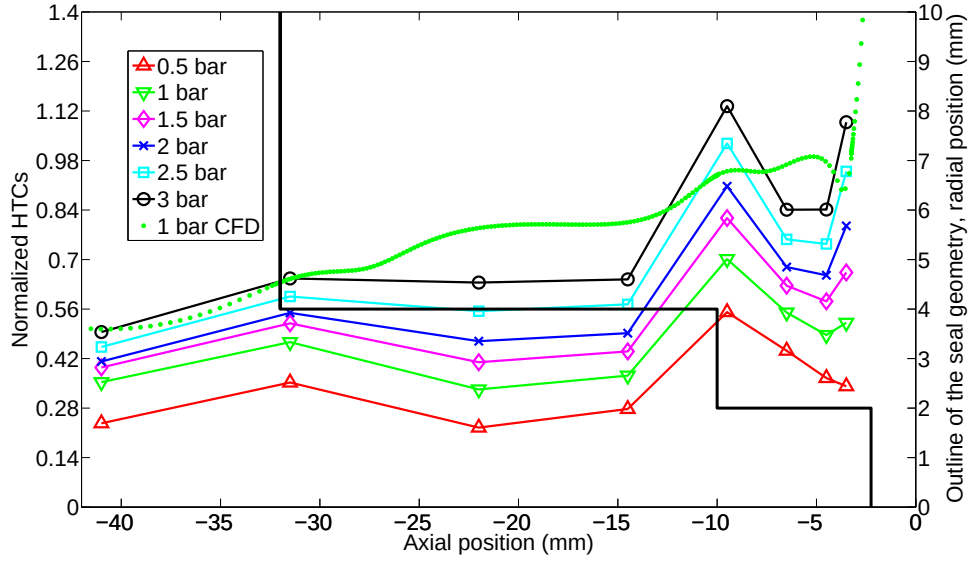
4.4 Experimental Data Results

Fig 4.17 shows the resulting h_{tc} distribution upstream and downstream of the seal over the range of pressure differences considered. A schematic of the seal installation is overlaid on the diagram, showing the piecewise flow contraction and expansion upstream and downstream from the seal. There are no data reported directly underneath the seal, where the heat transfer process is driven by conduction (which will be investigated in Chapter 5). Furthermore, in the first 5 mm downstream of the seal, the quality of the regression was insufficient for the h_{tc} to be considered reliable: very small temperature differences were achieved between the gas and surface temperatures. Similar difficulty begins to be seen at higher pressure ratio as the h_{tc} rapidly increases. Therefore, to obtain these larger values of h_{tc} , a higher temperature rise would need to be caused in the incoming flow using a more powerful heater mesh.

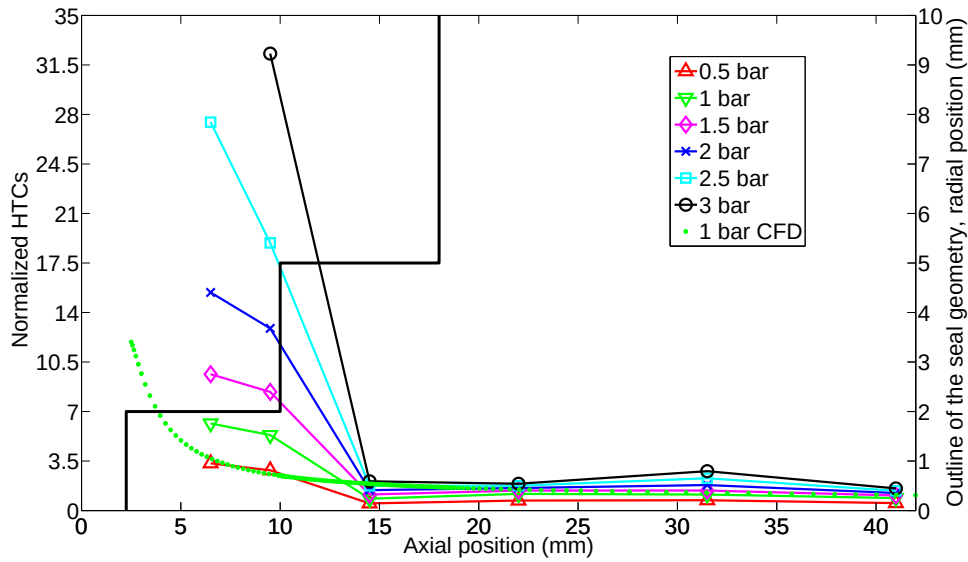
It is clear that upstream of the seal, the h_{tc} is initially low, but rises approaching the seal entrance. The stepwise contraction appears to generate an impinging flow (or separated accelerating flow) at -10 mm onto the rotor surface, followed by a drop and rise in h_{tc} as it is accelerated under the front coverplate of the seal.

Downstream, the highly accelerated and jetting flow from the exit of the seal causes extremely high heat transfer over the first 10 mm of the rotor surface. As discussed in Chapter 2 (see Fig 2.11), a stream exits the filament pack that contains a significant radial velocity component and thus impinges on the rotor surface in this zone. Note the vastly different scale on Figs 4.17a and 4.17b. The h_{tc} level subsequently drops, though the level, which is ~ 5 times higher than for the equivalent upstream cases, still shows evidence of enhancement from the jetting exiting flow. The level seen downstream is in good agreement with those reported in the CFD experiment for 1 bar differential pressure. Upstream, the CFD again at 1 bar pressure difference fails to capture the peak in h_{tc} at 10 mm and over predicts the heat transfer coefficient.

The h_{tc} data far upstream (thermocouples A8 – A6) are found to increase such that $h_{tc} \propto \dot{m}^{0.67}$; this appears to be driven by a laminar flow which transitions to turbulent

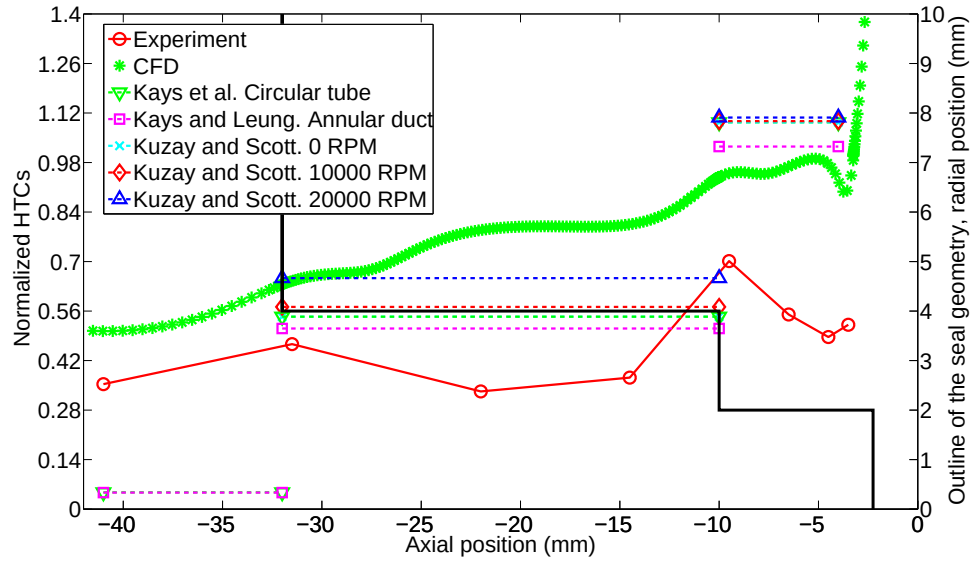


(a) Rotor htc distribution upstream of the seal

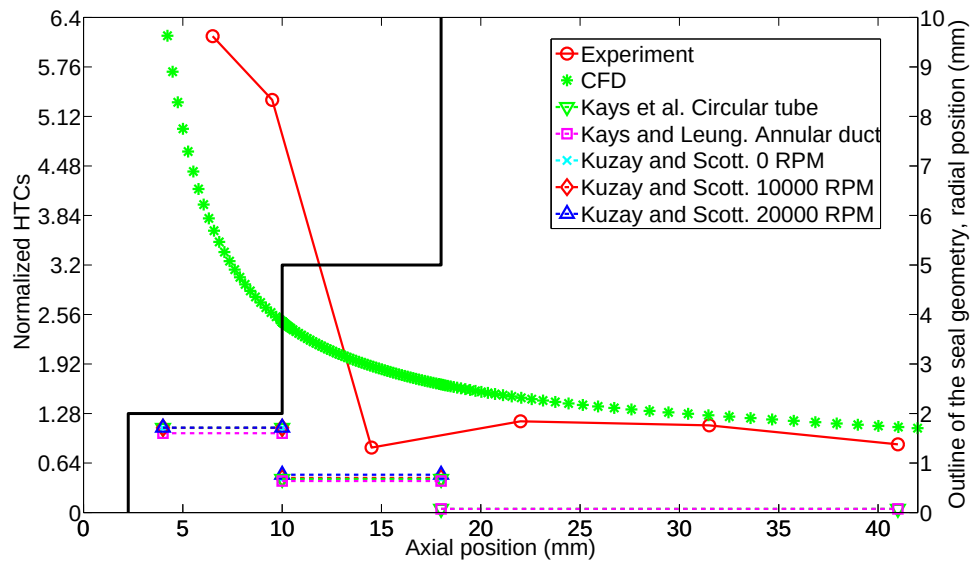


(b) Rotor htc distribution downstream of the seal

Figure 4.17: Rotor htc variation with seal differential pressure, values normalized by the same baseline value (189.9 W/m^2)



(a) Rotor htc distribution upstream of the seal



(b) Rotor htc distribution downstream of the seal

Figure 4.18: Comparison of experimental, CFD and correlated rotor htc distribution at 1 bar seal differential pressure

flow at the highest pressure ratios.

A comparison is made between the experimental and numerical data presented here and several well-known *htc* correlations for fully developed flow in ducts. The Reynolds number has been calculated using the leakage mass flow, cross sectional area and hydraulic diameter of the passage. A small change in *Re* occurs in the 3 different zones upstream and downstream of the seal: (from -41 to -32 mm, from -32 to -10 mm, from -10 to -3.5 mm, etc.). First, the correlation given by Kays et al. (2004) for turbulent flow in a circular tube is used:

$$Nu = 0.022 Pr^{0.5} Re^{0.8} \quad (4.5)$$

The heat transfer data presented in Kays and Leung (1963) for a turbulent flow in an annular passage is also considered. However, neither of these include rotation. Hence, the correlation proposed in Kuzay and Scott (1977) for turbulent flow in an annulus with inner cylinder rotation is employed to investigate the sensitivity of the results to the speed of rotation (swirl of the flow). These results are shown in Fig 4.18, where *htcs* have been normalized in the same manner as in Fig 4.12a. It is clear that the general trend of rising *htc* with decreasing gap height is seen in each case, however on the downstream side of the seal, the level is severely under predicted for all of the correlations employed. In contrast, the CFD and experiment are in quite good agreement. This is unsurprising as a wall jet develops along the rotor surface from the acceleration of the flow under the rear seal coverplate, which may be an order of magnitude higher in velocity than the average axial velocity in these regions.

The correlated effect of rotation on *htc* is insignificant over a representative range of shaft rotational speed from 0 – 20,000 RPM. Because of the rather low axial Reynolds number, and correspondingly high ratio of rotor circumferential velocity to through flow axial velocity, the correlation used was inapplicable in the - 41 to -32 mm and 18 to 41 mm regions. This also precluded the use of more recently developed *htc* correlations e.g. Childs and Turner (1994). It is however noted that the latter work shows the influence

of rotation on htc to be to cause a much stronger enhancement of the htc level.

4.5 Summary

As a summary of this Chapter, to predict the operating temperature of an installed seal and the rotor land on which it sits, knowledge is required not only of the heat transfer from the frictional contact to the seal, but also to the rotor. In this Chapter, an experimental technique used to obtain pointwise measurements along a full scale model of a rotor on which a seal is installed was described. The difficulty in obtaining an appropriate heat transfer reference temperature from gas thermocouples was investigated using CFD, and an appropriate, approximate, temperature proposed. Experimental heat transfer results were then presented over an engine representative range of pressure ratios for the first time. The results were obtained for a leaf seal in a strong blow-down configuration; however, they can be considered typical of any contacting filament seal. Far upstream, $htcs$ are largely unaffected by the presence of the seal, and thus, the swirling motion of the flow induced by the shaft rotation may be the dominant heat transfer mechanism and must be accounted for. The seal accelerates and straightens the flow, so that downstream $htcs$ are governed by the jetting axial flow and radial flow impingement on the rotor surface; and therefore, the effect of swirl is expected to be less important. The expansion of the flow through the seal is seen to cause a high speed flow downstream of the seal, with the htc in this area being an order of magnitude greater than that upstream. A CFD test case is found to be in reasonable agreement with the experimental data, particularly downstream of the seal element. This indicates that the porous medium approach employed is able to capture the main features of the flow field, and hence, can be used to estimate the level of htc in the seal vicinity. Nevertheless, more experimental testing and comparison with numerical predictions would be desirable in order to give more confidence to the CFD model developed.

Chapter 5

An Empirical Thermal Model for Contacting Filament Seals

As discussed in Chapter 3, the problem of frictional heat generation between a contacting filament seal and the rotor on which it is mounted can be regarded using an electrical analogy (Fig 5.1). In Chapter 4, the convective heat transfer characteristics of the flow in the vicinity of a leaf seal were investigated experimentally. The insight gained allows us to build more accurate heat transfer models of the rotor (R_{rotor}). The focus of this chapter is the development of a thermal model for brush and leaf seals (R_{seal} , in red in Fig 5.1). Essentially, what needs to be found out to calculate the partition of the frictional heat (Q_f) is the amount of heat absorbed by the seal (Q_{seal}) for a given nominal contact temperature rise ($\Delta T_{nominal}$).

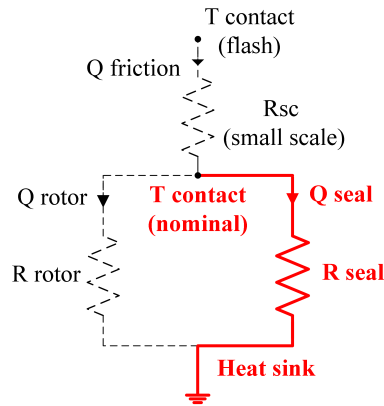


Figure 5.1: Thermal modelling of seal-rotor frictional heat generation using an electrical analogy

5.1 Brush/Leaf Seal Heat Transfer Model

Analytical, experimental and numerical approaches may be used to tackle any flow problem. The appropriateness of each of these for describing the thermal behaviour of brush and leaf seals is briefly considered below.

A thermal model of an idealised system may be conducted analytically. Fundamentally, heat transfer in a filament seal consists of the generation of heat due to friction at the bristle or leaf tips followed by its conduction into the filament pack. As this heat flow propagates along the leaves or bristles, it is convected to the surrounding leakage flow in a direction perpendicular to the principal direction of solid conduction. Consequently, if the variation in gas temperature is assumed to be small (i.e. near infinite heat capacity), a model based on a fin approach can be considered. As explained in the review presented in Chapter 2, one and two dimensional fin analytical models have been proposed in the past. For instance, Franceschini (2005) derived the following equation by performing an energy balance in the radial and axial plane of an individual leaf:

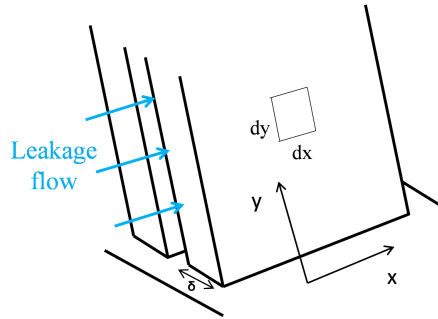


Figure 5.2: 2D-fin model for a leaf seal, Franceschini (2005)

$$\frac{\partial^2 T_m}{\partial x^2} + \frac{\partial^2 T_m}{\partial y^2} = \frac{htc_{pack}}{k_m \delta} (T_{gas} - T_m) \quad (5.1)$$

where T_m and k_m are the metal temperature and thermal conductivity respectively, and T_{gas} the gas temperature. Due to the small thickness (δ) of the leaf, conduction in the z-direction was neglected. The main limitation of the fin approach is the difficulty in evaluating the inter-filament convective heat transfer coefficient (htc_{pack}) on the right-

hand side term of Eq 5.1. The inter-filament flow can be considered internal flow, since the gas is confined between two planes. Due to the strong pressure gradient through the seal, the velocity profile changes with increasing axial position and the flow cannot be said to be fully developed. The significant radial velocity component of the flow and compressibility effects also make difficult to find a suitable correlation for $ht_{c_{pack}}$. As discussed in Section 2.1, the small dimension of the hydraulic diameter of the inter-filament passage ($\sim 10^{-5}$ m) suggests that microscale heat transfer considerations may be necessary. Previous investigations have demonstrated that many micro channel heat transfer phenomena cannot be explained by theories based on the continuum hypothesis. Transition from laminar to turbulent conditions, correlations for the Nusselt number, and fluid viscosity can differ considerably from those predicted by conventional theories (Morini and Yang (2013)). But the key consideration in this case is that the passages are so small that the gas temperature approaches the local metal temperature prior to the exit of the seal.

These complexities make the development of an accurate analytical model unfeasible. Recourse to high fidelity computational fluid dynamics (CFD) can be made, and a numerical solution of the either time averaged or unsteady full set of Navier-Stokes equations attempted. Nevertheless, this approach is not free of problems. The definition of the flow paths remains as the main obstacle for a reliable prediction of the seal thermal behaviour. While, numerical solutions of the cross flow through voids among uniformly positioned rigid cylinders (representing a bristle pack) have been obtained in the past (Pekris (2010)), in reality, the pressure gradient and aerodynamic forces exerted on the filaments (described in detail in Section 2.1) cause them to vibrate and move under pressure loading during operation. Therefore, the actual seal geometry is difficult to predict. Some researchers have modelled the seal filament pack as a porous medium using CFD. These models can be adequate for capturing the pressure loss caused by the presence of the seal. However, since the filament geometry and actual flow passage are not properly accounted, they fail to capture the implicit conjugate heat transfer mechanism occurring within the filament pack.

For the simple electrical resistance analogy heat transfer model the thermal resistance of the seal (R_{seal} in Fig 5.1) is required. To calculate the heat partition, we are primarily interested in the relationship between the nominal contact temperature rise and the heat transferred to the seal at different leakage flow rates, rather than in the temperature distribution along the filaments. The empirical method can be used to investigate this correlation. Due to the many uncertainties going into the mathematical models, experimental data are probably more reliable. In order to find a suitable functional relation for an empirical correlation, we need to ascertain which physical quantities determine the amount of heat conducted into the seal. To that end, the existing brush/leaf seal heat transfer model (as developed in Oxford, Owen et al. (2003)) can be considered. This was fully described in Chapter 2 (Section 2.2.2); for leaf and brush seals the following expressions result:

$$Q_{leaf} = \frac{3}{4} \sqrt{\frac{k c_p N L \pi \delta}{b}} \dot{m}^{1/2} (T_c - T_{gas}) = K_{seal} \dot{m}^{1/2} (T_c - T_{gas}) = \frac{\Delta T_{nominal}}{R_{seal}} \quad (5.2)$$

$$Q_{brush} = 2 \sqrt{\frac{k c_p L D}{b}} \dot{m}^{1/2} (T_c - T_{gas}) = K_{seal} \dot{m}^{1/2} (T_c - T_{gas}) = \frac{\Delta T_{nominal}}{R_{seal}} \quad (5.3)$$

where c_p is the specific heat of the gas, k the metal thermal conductivity, N the number of leaves, L the axial width of the filament pack, δ the leaf thickness, b the backing ring/downstream coverplate clearance, and D the inner diameter of the seal at the filament tips. All these parameters can be lumped into a constant (K_{seal}), which is an intrinsic characteristic of the seal application and does not depend on its operating conditions (there is an assumption of unchanging operational geometry here). The value of K_{seal} is determined by the geometry and thermal properties of the seal and the specific heat of the working fluid. The value of this seal thermal constant can be investigated through experimentation. In Eqs 5.2 and 5.3, the leakage mass flow rate is represented by $\dot{m}$, while T_c and T_{gas} stand for the nominal contact and upstream gas temperatures.

According to these formula, increasing the leakage flow through the seal reduces its thermal resistance, so that a larger fraction of the heat generated is conducted into the filament pack, and thus the cooling fluid.

The Oxford Engine Seal Test Facility (ESTF) has provided experimental data to validate the general form of the correlation expressed in Eqs 5.2 and 5.3 (Owen et al. (2003), Pekris (2010)). Furthermore, the functional dependency suggested in these equations (proportionality between Q_{seal} and $\dot{m}^{1/2} (T_c - T_{gas})$) was confirmed by Pekris (2010) using dimensional analysis. Nevertheless, the values of K_{seal} found experimentally using the ESTF were found to differ by a factor of ~ 2 compared to those predicted by Eqs 5.2 and 5.3.

To increase the confidence in the functional relationship set out in Eqs 5.2 and 5.3, an additional analytical analysis can be performed returning to the fin model discussed above. As a first approximation, the leaves or bristles of a seal can be modelled as 1D fins. As the heat is conducted in the radial direction, it is progressively convected to the fluid. We may assume that the metal temperature is only a function of the radial coordinate, neglecting axial and circumferential gradients. A bulk uniform convective heat transfer coefficient can be defined over the filament surfaces (htc_{pack}). Although the static temperature of the fluid changes due to the simultaneous heating and expansion through the filament pack, a reference temperature equal to the upstream temperature may be considered (T_{gas}). The heat absorbed by a single infinitely long 1D fin is then expressed as (Incropera et al. (2013)):

$$Q_{fin} = \sqrt{P k A} htc_{pack}^{1/2} (T_{tip} - T_{gas}) = K'_{seal} htc_{pack}^{1/2} (T_c - T_{gas}) \quad (5.4)$$

where P and A are the fin perimeter and cross sectional area respectively. T_{tip} is the temperature at the tip of the fin, which is analogous to the nominal contact temperature (T_c). To obtain the total heat absorbed by the entire filament pack, Eq 5.4 is multiplied by the total number of filaments. As in Eqs 5.2 and 5.3, the variables that depend on the geometry and thermal properties of the seal can be grouped into a constant term, K'_{seal} .

Eq 5.4 assumes that the temperature rise relative to the upstream gas temperature is negligible at the root of the filaments (infinitely long fin). This assumption has commonly been made in seal thermal modelling due to the highly convective heat transfer rates expected within filament pack, as concluded in Chapter 2. Clearly, the validity of this assumption is subjected to the operating conditions of the seal. For high frictional powers and low leakage flow rates, a temperature rise at the root region may exist. However, for the range of frictional power and seal differential pressure tested in the laboratory, leaves and bristles are expected to be able of dissipating all the heat along their length. This feature strongly simplifies the thermal modelling, since it allows treating the seal as a semi-infinite medium so that the heat absorbed only depends on a single temperature (the contact temperature). Nevertheless, a more general formulation can be obtained by imposing a temperature rise at the root of the fin (T_{root}), of length L . Then, Eq 5.4 results into (Incropera et al. (2013)):

$$Q_{fin} = \sqrt{P k A} htc_{pack}^{1/2} (T_{tip} - T_{gas}) \frac{\cosh(\sqrt{\frac{htc_{pack} P}{k A}} L) - \frac{(T_{root} - T_{gas})}{(T_{tip} - T_{gas})}}{\sinh(\sqrt{\frac{htc_{pack} P}{k A}} L)} \quad (5.5)$$

From the comparison between Eqs 5.2-5.3 and 5.4, it is evident that the inter-filament convective heat transfer coefficient (htc_{pack}) must be proportional to the leakage mass flow rate should both models hold true. To assess the validity of the previous statement a priori, we can refer to previous studies on convective heat transfer in micro channels. Despite the vast scientific literature on this topic, there are no generally accepted correlations for the calculation of the Nusselt number, and most proposed are in disagreement in form and value. What is universally accepted is that “classical” correlations obtained for conventional tubes, such as Dittus and Boelter’s equation, are not applicable to micro channels. One such htc correlation for micro channel heat transfer was developed by Wu and Little (1984) for nitrogen gas passing through fine channel heat exchangers. The dimensions of the channels were of the order of tens to hundreds of microns, i.e.

similar to inter-filament flow passages. They suggested a heat transfer correlation as:

$$Nu = 0.00222 Pr^{0.4} Re^{1.09} \quad (5.6)$$

For a given working fluid, assuming no changes in either fluid properties or channel geometry, Eq 5.6 implies:

$$htc \propto \dot{m}^{1.09} \quad (5.7)$$

Eq 5.7 shows that mass flow rate and convective heat transfer coefficient keep a relationship that is not far from linear in a micro channel of the same size as the inter-filament gap. This result should not be taken as a proof of the existence of proportionality between htc_{pack} and $\dot{m}$ in Eqs 5.4 and 5.2-5.3. The features of the inter-leaf flow are quite unique (no fully developed flow, compressibility effects, significant radial velocities, low thermal heat capacity) and thus Eq 5.6 is not entirely applicable. However, the functional dependency between Re and Nu here is notably different from either conventional laminar or turbulent passage flow.

In essence, the fin analysis yields a functional relationship (Eq 5.4) analogous to that of the Oxford seal thermal model (Eqs 5.2 and 5.3). This reinforces the idea of the suitability of the suggested functional relationship for an empirical correlation. We expect the heat transferred to the seal to be dependent on a constant term and two operational parameters; namely, the leakage mass flow rate and the nominal contact temperature rise. In the next section, the development of an extension of the Seal Static Thermal Test Facility (described in Chapter 4) is presented. The aim of this new rig is the experimental derivation of the proportionality constant (K_{seal}) required for the seal thermal correlation.

5.2 Seal Static Thermal Test Facility (SSTTF): Seal Thermal Characterization

5.2.1 Description of the experimental facility

According to Eqs 5.2 and 5.3, the heat absorbed by a brush or a leaf seal is a function of the leakage mass flow rate, the difference between the nominal contact and upstream gas temperatures, and the geometry and thermal properties of the seal. The latter are lumped into the seal thermal constant, K_{seal} , which will be inferred by experimentation. Therefore, the test facility to be developed needs to reproduce an environment where all these quantities can be easily controlled and measured. Fig 5.3 shows a schematic of the concept proposed for such a test rig.

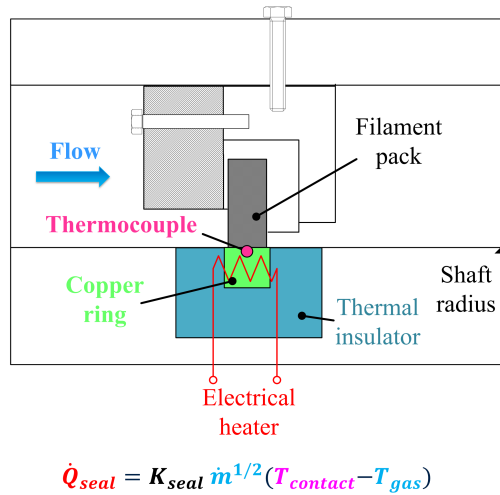


Figure 5.3: Concept of the test facility for the derivation of the seal thermal constant, K_{seal} . It should be noted that the heater element is located inside the copper ring; a more detailed representation of the heater assembly is included in Fig 5.4b

When friction occurs, mechanical energy is converted into heat at the seal-rotor contact interface. Building a real rubbing system, where the level of frictional heat generation can be adjusted precisely, involves many technical challenges. Also, controlling the amount of power dissipated with independence of the flow conditions can be difficult. Since the nature of the heat source is not relevant, an electrical heater may be used

instead. It is a simple matter to measure and adjust the voltage and current to specify the heat production.

It must be ensured that all the electrical power converted into heat, or at least a known proportion of it, is transferred to the seal. As shown in Fig 5.3 , the electrical heater is thus surrounded by a highly conductive metal structure (made of copper), which channels the heat generated towards the tips of the seal filaments. Using a material with a high thermal conductivity also helps to achieve a uniform and unambiguous temperature at the nominal contact area with the seal. This temperature is measured by a thermocouple embedded into the copper structure. Since the heater element and copper cannot be in contact, they are separated by a ceramic paste electrical insulator.

In order to minimize the heat losses to the surroundings, the copper structure is covered with a thermal insulator. *Tufnol* (epoxy-cotton fabric laminated plastic) has been chosen due to its mechanical properties, low thermal conductivity, and ease of machining. Since perfect thermal insulation is not possible, a set of thermocouples is installed on both sides of the insulator to estimate heat losses.

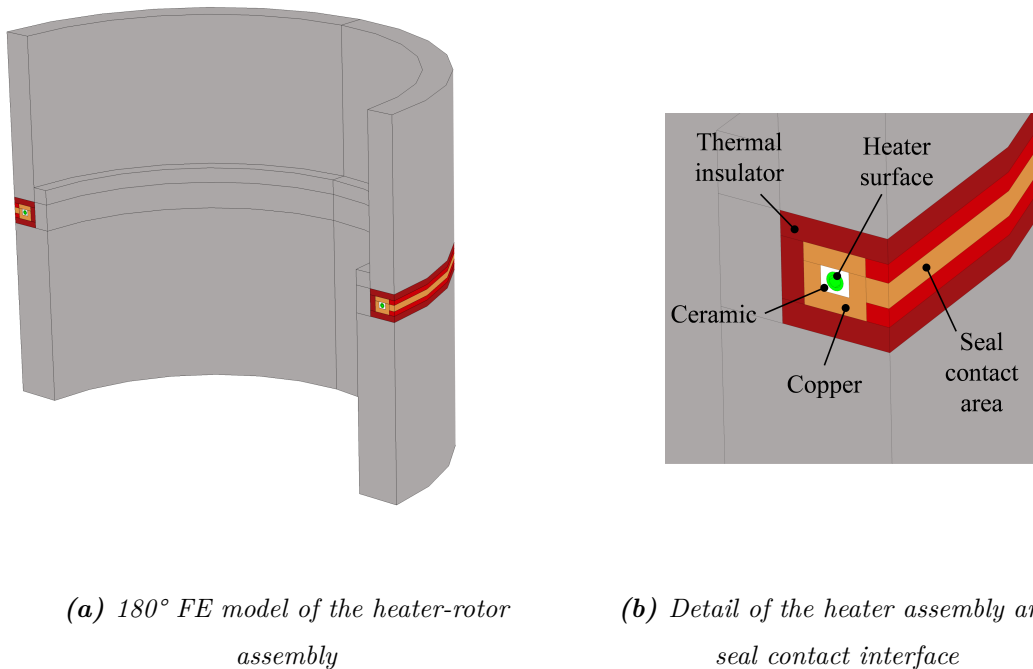


Figure 5.4: FE domain

Prior to manufacturing the final design, a basic finite element (FE) study was conducted to assess the correct operation of the system. The conceptual design depicted in Fig 5.3 was turned into the practical geometry shown in Fig 5.4. It represents an engine rotor in which the seal is in contact. The heater assembly described above is inserted in the rotor as shown in Fig 5.4b.

For the FE simulation, the following boundary conditions were employed. The power dissipated by the electrical heater was set to 1 kW, which was considered in the model as a surface heat source over the green area shown in Fig 5.4b. A uniform convective heat transfer coefficient of $10,000 \text{ W/m}^2\text{K}$ was imposed at the seal contact area. According to previous experience in Oxford, this value is representative of the cooling capacity of a filament seal (Pekris (2010)). Ambient temperature was applied to the top and bottom planes of the domain, and natural convection was allowed on the inner wall of the rotor representative static body (henceforth called the ‘rotor’ for ease of reference). Forced convection, with a heat transfer coefficient distribution based on the results obtained in Chapter 4, was used on the outer walls of the rotor.

Fig 5.5a shows the resulting heat flux distribution leaving the boundary surfaces of the domain. It can be seen that most of the heat input exits the system at the seal contact area. The ratio of heat flux at the seal contact area to electrical power was ~85% in this preliminary simulation. The corresponding temperature field obtained is displayed in Fig 5.5b. The uniformity in copper temperature is clearly seen, which indicates that a thermocouple embedded close to the interface with the seal is suitable for characterizing the nominal contact temperature. While the boundary conditions employed were necessarily approximate, the outcome of the FE analysis confirmed that the initial design of the test facility should be satisfactory.

A dimensioned drawing of the final heater-rotor assembly built is shown in Fig 5.6a. Aluminium alloy 2024 was used as the rotor material.

The heater-rotor assembly shown in Fig 5.6a was installed in the Seal Static Thermal Test Facility (SSTTF, described in Section 4.1.1, see Fig 4.2), replacing the existing rotor (instrumented for the measurement of the rotor convective heat transfer coefficients).

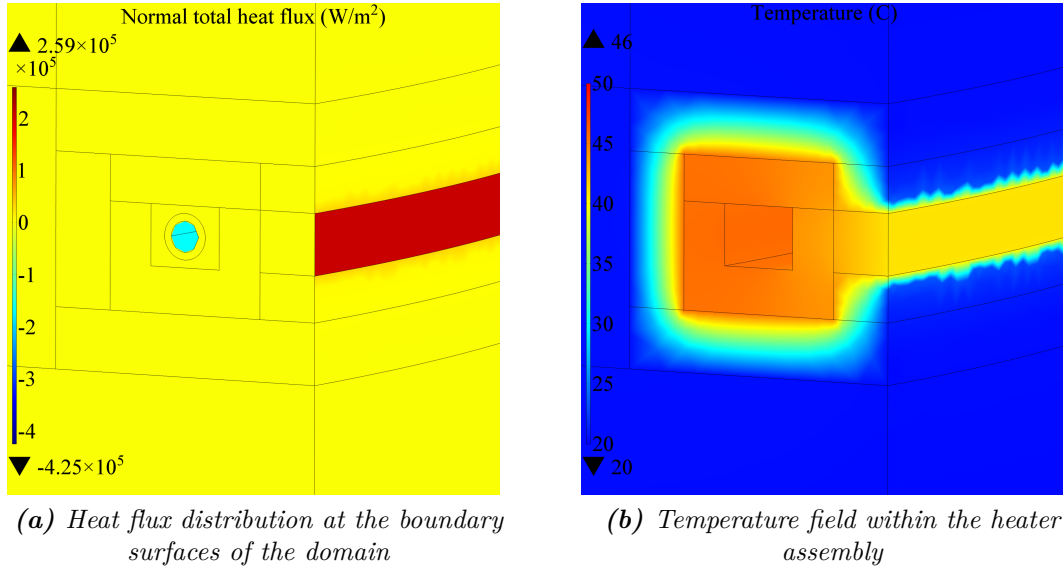
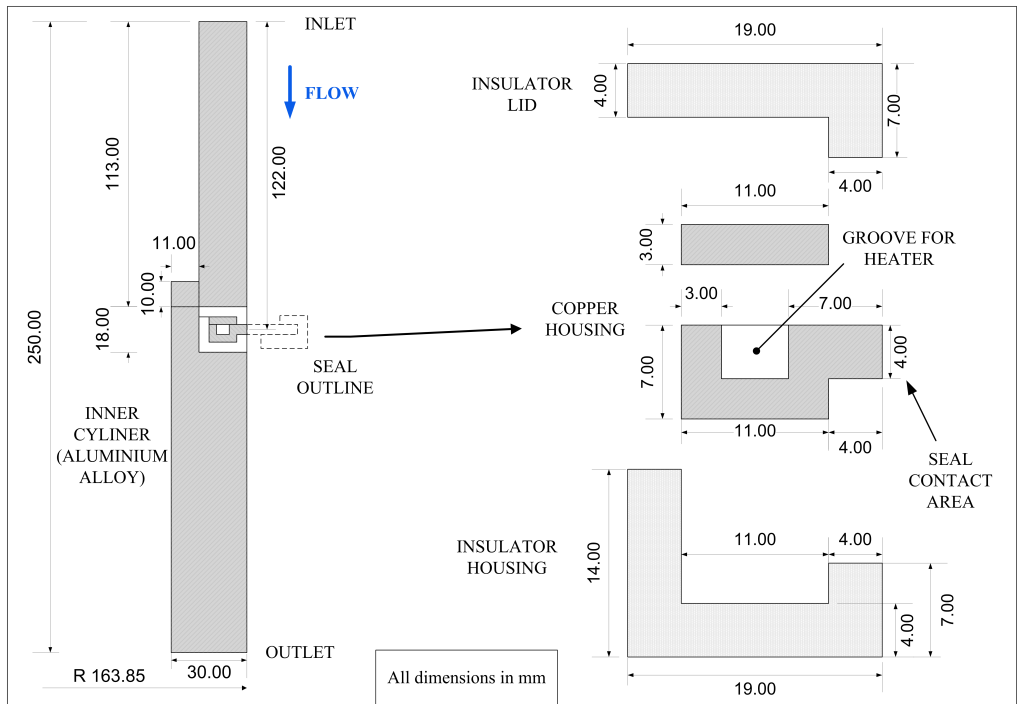


Figure 5.5: Key results from FE simulation

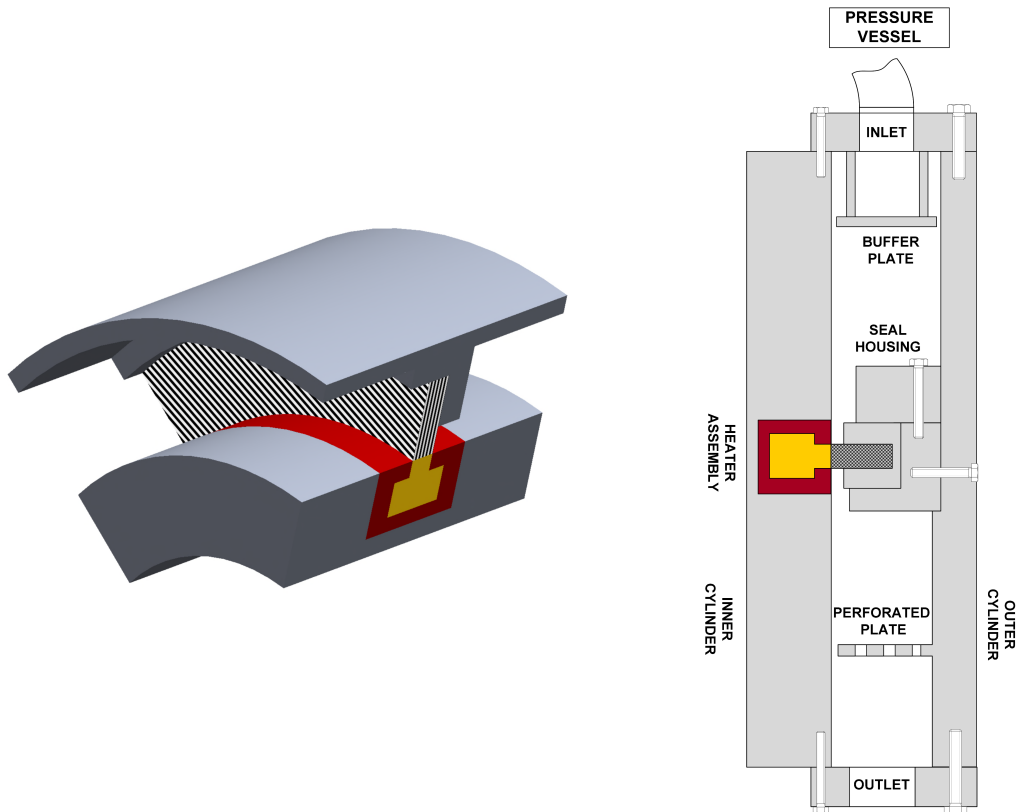
Hence, the same air supply and data acquisition systems of the SSTTF are employed here. Fig 5.6b shows a diagram of the pressure vessel formed with the new heater-rotor assembly. An adjustable seal housing system allows the installation of a brush or leaf seal in such a way that the centreline of the filament pack coincides with the midsection of the heater seal contact area, as depicted in the figure. The other two quantities required for the seal thermal correlation, namely, leakage mass flow rate and upstream gas temperature, are measured using the previously described instrumentation (see Chapter 4). It should be noted that, as in Chapter 4, the rotor or inner cylinder of the pressure vessel is static, and therefore, rotation is not included in the experimental setup.

5.2.2 Instrumentation and heater design

The instrumentation of the heater-rotor assembly consists of a set of K-type thermocouples. These are located in the copper structure to measure the contact temperature, and on both sides of the *Tufnol* walls to assess the heat losses to the surroundings. Although the rig can be considered axisymmetric, thermocouples were installed at 3 circumferential locations separated by 90 degrees. The position of these sensors is shown in Fig 5.7. The nominal contact temperature is assumed to be equal to that measured



(a) Drawing of the heater-rotor assembly built



(b) Schematic of the pressure vessel formed with the new heater-rotor assembly (inner cylinder) and existing hardware of the SSTTF (outer cylinder, buffer plate, perforated plate and, top and bottom lids); see Fig 4.2 for completeness

Figure 5.6: Experimental setup

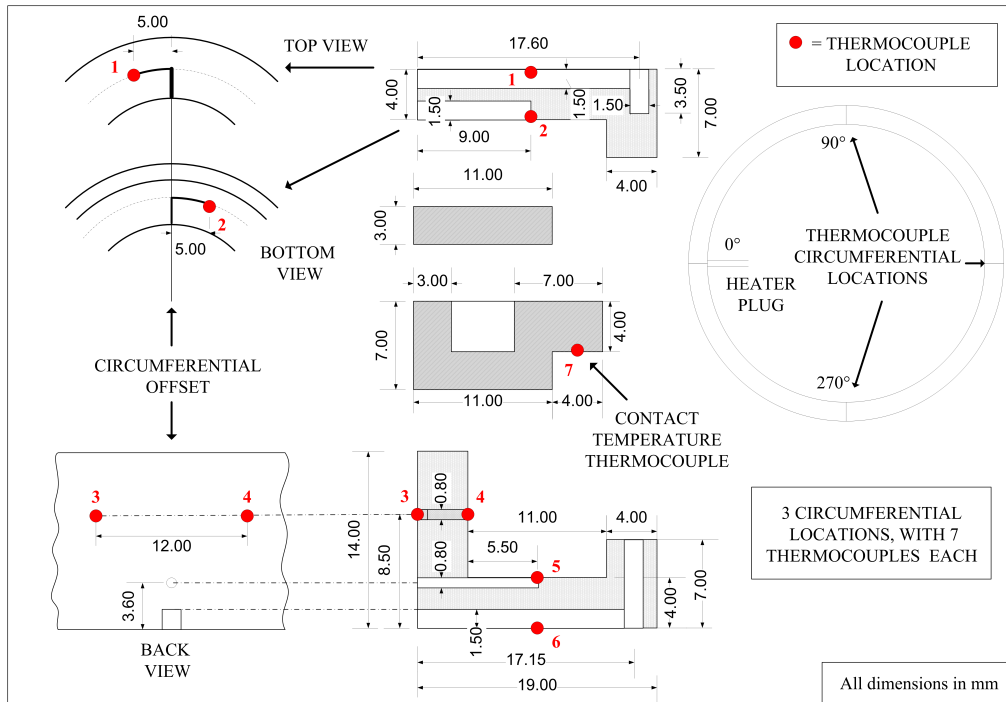


Figure 5.7: Location of the thermocouples in the heater assembly. The thermocouple positions are represented by the red circles

by the thermocouples embedded in the copper given the high thermal conductivity of this metal and the observations from the preliminary FE analysis (Fig 5.5b)). High conductivity silver paint was used to join the thermocouples to the copper substrate, in small thermal wells.

The temperature of the airflow upstream of the seal is measured by the gas thermocouples described in Chapter 4 (see diagram in Fig 4.3). In addition, the same orifice plate meter and pressure transducers are used for measuring the leakage mass flow rate through the seal and seal differential pressure.

As discussed before, frictional power is simulated using an electrical heat source. In the design process of the heater element, the first decision to be made was the maximum power aimed to be dissipated. In an engine, filament seals are not expected to generate frictional powers higher than 10 kW at normal operation. Only a fraction of this heat is conducted into the filaments of the seal, the remaining is transferred to the rotor according to a proportion that is unknown a priori. While engine representative

seal differential pressures can easily be attained in the laboratory, we are more limited in terms of temperature rises and frictional powers. As the target maximum power increases, the construction of the heater element and its instrumentation becomes more complicated and a bespoke electrical power supply is required. Literature review and previous studies have shown that the frictional heating and contact temperatures exhibit a linear relationship (refer to Chapter 3). Therefore, it is expected that results obtained at low-medium contact temperature rises in the laboratory can be extrapolated to engine operating conditions. Consequently, it was decided to design for a maximum power of 1 kW, which is a good trade-off between a representative value of frictional power and ease of manufacture.

Power is sourced from the single phase 230V AC supply, and first delivered through a variable transformer (*Variac*). This has a variable turns ratio that allows the adjustment of the output voltage. The *Variac* is subsequently connected to a step-down transformer, capable of delivering 1 kW (55 V, 18 A), which is power matched to the resistance wire employed. Finally, the heat is generated by passing a high current through the resistance wire. In order to produce 1kW using a 55 V power supply, the wire resistance was set at $\sim 3\Omega$ according to Eq 5.8:

$$P = \frac{V^2}{R} \quad R = \frac{55^2}{1000} = 3.03 \Omega \quad (5.8)$$

Nichrome (80% Nickel and 20% Chrome) was employed as the heating element because of its high resistivity ($1.03 \times 10^{-6} \Omega m$) and melting temperature ($1420^\circ C$), together with its relatively low coefficient of thermal expansion and resistance to oxidization. For a metallic wire, the cross sectional area (A) is related to the resistivity (ρ_Ω), length (l) and electrical resistance (R) by:

$$A = \frac{\rho_\Omega l}{R} \quad (5.9)$$

The length of the wire is fixed by the circumferential arc of the groove in the copper structure where the heater wire is housed (see Fig 5.6a, ~ 1 m). The resulting area required for the Nichrome wire is $\sim 3.4 \times 10^{-7} m^2$, and the corresponding wire diameter

0.66 mm. However, given the limitation in wire diameters available commercially, a different arrangement was used. Wires can be stranded to achieve a lower resistance. The resistance of a stranded wire (parallel circuit) is given by:

$$R_{stranded} = \frac{R_{single\ strand}}{n} \quad (5.10)$$

where n is the number of strands. For a Nichrome wire of 0.35 mm in diameter and 1 m in length, the electrical resistance is $\sim 10.4\ \Omega$. Stranding three wires of these characteristics results in an overall resistance of $3.47\ \Omega$, close to the desired value of $3.03\ \Omega$. Therefore, this configuration was used due to the availability of wires 0.35 mm in diameter. Also, in comparison with a single strand wire, a stranded wire of the same resistance has a larger surface area. This results in lower heat flux densities and wire temperatures, reducing the risk of failure. The lightly twisted stranded wire allows the wire to expand and contract during the operation of the test rig. Previous experience with single heating wires showed that they are subject to failure through expansion, heating and fracture upon contraction.

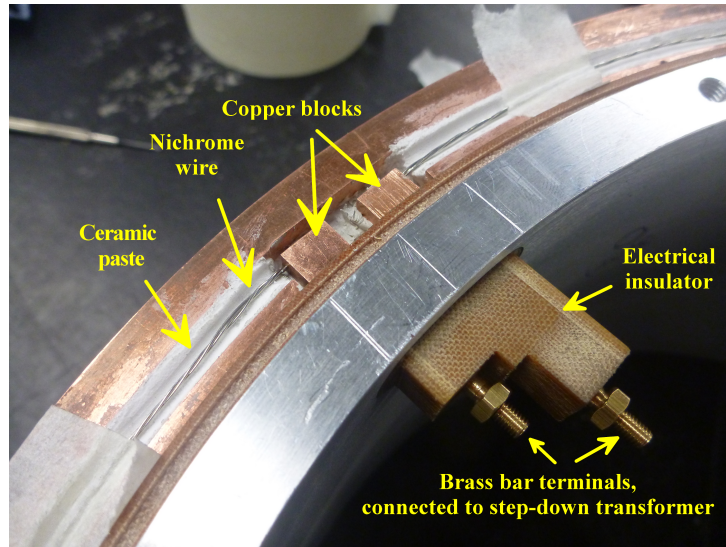


Figure 5.8: Picture of the heater assembly

Since the Nichrome wire has to be electrically insulated, it was embedded into a ceramic paste. Ceramics are good electrical insulators and cope well with high tem-

peratures and thermal fatigue. The two ends of the Nichrome wire were crimped into two small blocks of copper. These contain a threaded brass terminal for connection to the step-down transformer. A photograph of the heater assembly is shown in Fig 5.8 (taken during the fabrication process). The location of the brass bar terminals in the circumferential plane is labelled in Fig 5.7 as “heater plug”.

The final resistance of the Nichrome wire was measured after installation. The same value of $3.2\ \Omega$ was obtained using a 4-wire voltmeter measuring between the following three locations: the two brass bar terminals, the two copper blocks and the two ends of the wire. This final value is slightly above the target value, $3.03\ \Omega$ (1kW), which gives a certain security margin since at maximum voltage (55 V), the resulting current is lower than the maximum allowed (18 A), and showed that the contact resistance between elements of the heater assembly were insignificant.

5.2.3 Data acquisition system and accuracy considerations

The data acquisition system described in Chapter 4 is employed again in this experimental campaign. Pressure transducers and thermocouples were plugged into an expansion board connected to a PC, which remotely controls the rig. A *LabVIEW* routine simultaneously monitors and records the experimental data. In a test run, voltage and wire resistance were measured at the brass bar terminals (see Fig 5.8) using an *HP Hewlett-Packard 3478A* multimeter. Power was then calculated using Eq 5.8. An uncertainty analysis of the rig was included in Chapter 4, Section 4.1.3. For the physical quantities relevant to this new set of experiments, they were found a root-sum-square (RSS) error of 1.23% in mass flow rate (orifice plate meter), and a bias and precision error of 3.6°C and 0.3°C respectively in temperature (K-type thermocouples). The accuracy in the new quantities measured (voltage and wire resistance) is provided in the specifications of the multimeter employed as a percentage of the reading. Since these quantities are averaged with time throughout the test, only bias errors were considered. For the data range obtained in the test campaign, errors of 0.16% and 0.68% were estimated for the wire resistance and voltage respectively. The calculation of the net power

transferred to the seal by subtracting the heat losses from the total electrical power is explained in the next section. The uncertainty in this derived quantity was computed by perturbing the input variables involved in its derivation with their corresponding overall errors, as explained in Moffat (1988). The resulting typical RSS error band in net seal power was ± 12.2 W.

5.3 Experimental Data Reduction

To characterize the thermal behaviour of a particular seal design, several tests are run ranging the seal differential pressure (and therefore, the leakage mass flow rate) and the power delivered by the transformer. Fig 5.9 shows the evolution of the average contact temperature (mean value of the three thermocouples located at the different circumferential locations shown in Fig 5.7) during a typical experimental test. In this example, a leaf seal was installed in the rig. First, the valves of the air system upstream of the pressure vessel are opened, and high pressure supplied to the seal. In the figure, a slight initial drop in contact temperature occurring immediately after pressurization is observed due to the cooling effect induced by the expansion of the airflow through the seal. Second, the target electrical power is set by adjusting the turns ratio of the *Variac* transformer. This power setting is kept constant during the entire test, as indicated by the time interval comprised between the two red dotted lines in the figure (466 W in the example). After the heater element is powered, the contact temperature increases rapidly during the first hundred seconds, beyond which the rate of temperature increase reduces. When the variation in contact temperature with time is negligible, steady-state conditions are assumed, and a data point is obtained for the seal thermal correlation. Next, the seal differential pressure is changed, which results in another transient event that eventually provides a second data point. This process is repeated several times during the test. In general, for a given power setting of the transformer, the seal differential pressure is varied to obtain between 2 and 5 data points in a test. In the example of Fig 5.9, three steady-state data points were obtained, at seal differential pressures of 2.2, 3.2 and 4.4 bar (time intervals represented by the green segments).

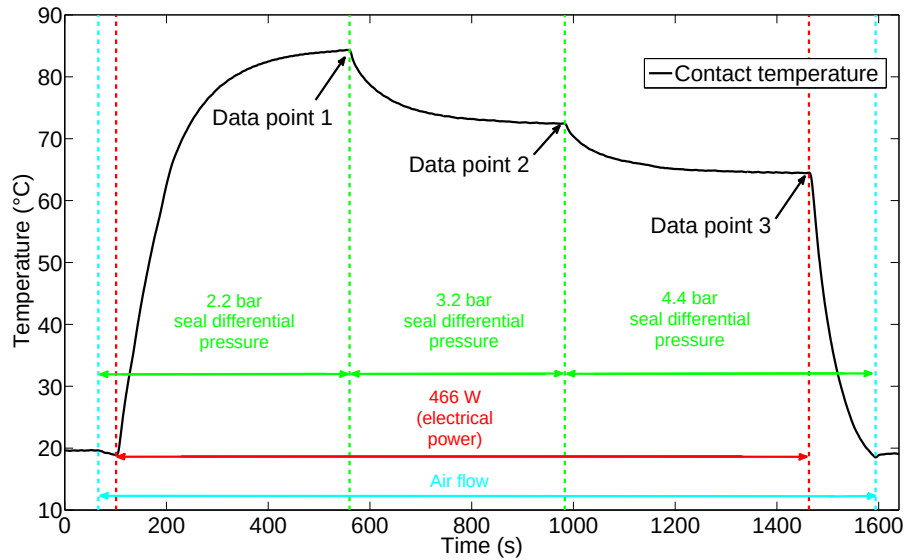


Figure 5.9: Evolution of the circumferential-average contact temperature during a typical experimental test

Finally, at time ~ 1480 s in the example, the transformer was shut down, however, air continued to blow through the seal to cool down the system. This caused the contact temperature to fall exponentially to ambient values; then, the airflow was stopped. The set of steps described forms the standard operating procedure of the rig.

Although for a given seal differential pressure the rig is run until changes in contact temperature with time are very low, the asymptotic steady-state value is inferred by fitting an exponential curve onto the experimental temperature trace. This is shown in the example of Fig 5.10, where the value of the contact temperature provided by one of the thermocouples is plotted together with its approximation by an exponential interpolation. This procedure is also used for the estimation of the steady-state values of the temperature signals from the thermocouples installed in the *Tufnol* insulator and from the upstream gas thermocouples. These asymptotic values of temperature are used to populate the seal thermal correlation.

Tests at different electrical powers are carried out to obtain a wide range of data points for the seal thermal correlation. Although the rig was designed to be operated at a maximum of 1 kW, the electrical power was limited so that contact temperatures

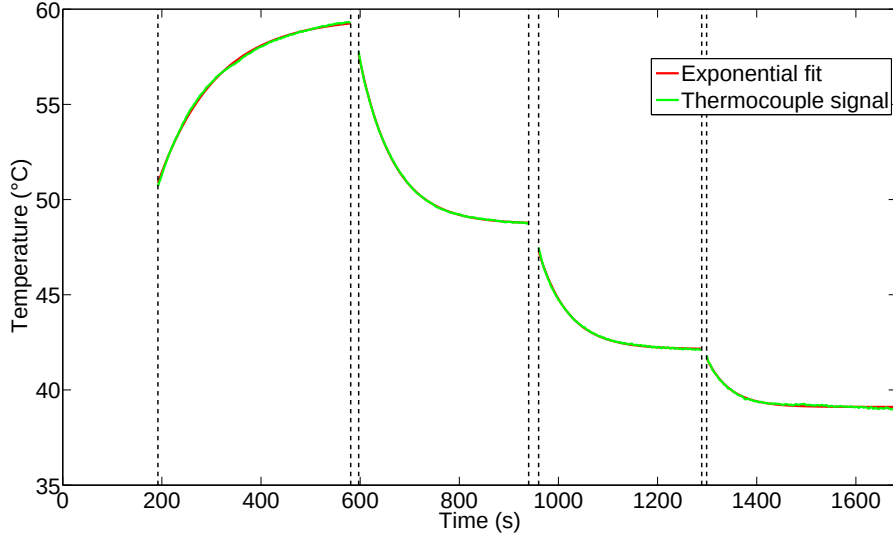


Figure 5.10: Estimation of the asymptotic steady-state temperatures via exponential fit

always remain below 100 °C. The silver paint and epoxy used to glue the thermocouples have a maximum operating temperature of 120 and 200 °C respectively. Also, the hazard of failure of the heater element was always present. The temperature of the Nichrome wire during rig operation was unknown and difficult to estimate, so melting of the wire could not be predicted. In addition, since the displacement of both ends of the wire was constrained, an excessive thermal expansion might cause wire buckling melting within the surrounding ceramic paste, followed by loss of electrical continuity on cooling. With a contact temperature of 100 °C set as the upper limit for rig operation, a relatively wide data range of contact temperature rises for the seal thermal correlation could still be obtained.

The voltage difference between the two ends of the heater wire is monitored throughout the test. Then, the average value with time is computed and used for the calculation of the electrical power generated. It should be noted that this is not the power absorbed by the seal. Despite the presence of an insulating material, the temperature difference that exists between the copper ring and the aluminium cylinder of the rotor always involve heat losses. These must be accounted and subtracted from the total electrical power to find out the amount of heat transferred to the filaments of the seal. The ther-

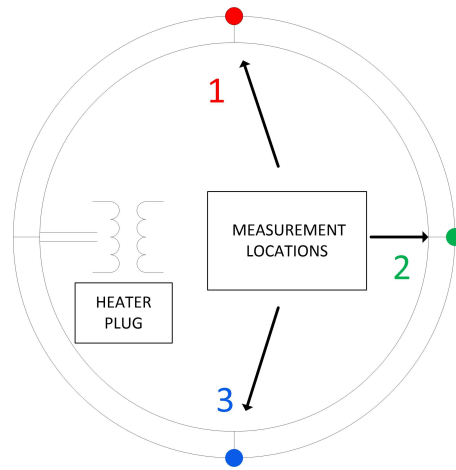
thermocouples installed on both sides of the insulator walls are employed to estimate the heat losses using the following expression:

$$q_{losses} = \frac{k_{tu} f_{nol}}{l_{wall}} (T_{int} - T_{ext}) \quad (5.11)$$

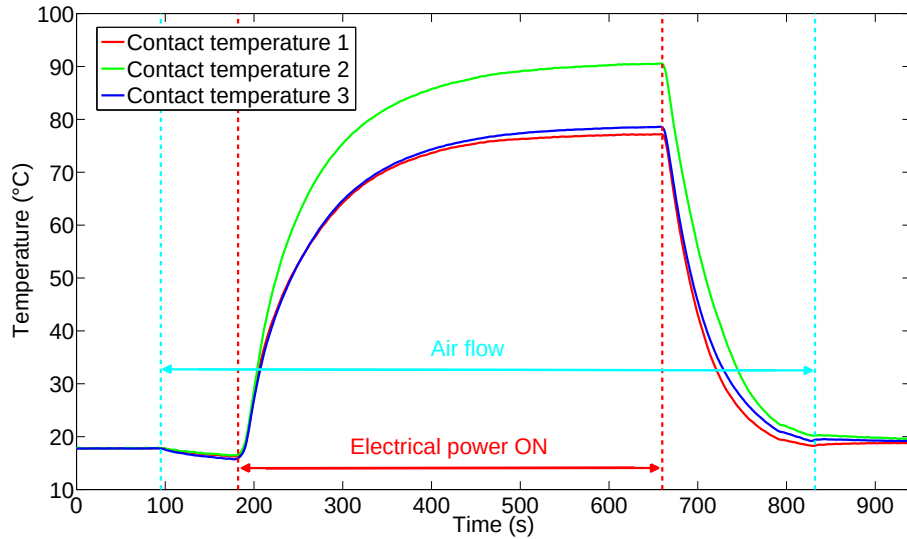
The location of these thermocouples was shown in Fig 5.7. They are placed at the three circumferential positions where the nominal contact temperature is also measured. At each circumferential position, there are three pairs of thermocouples, which measure the heat losses upwards, backwards and downwards. In Eq 5.11, T_{int} and T_{ext} refer respectively to the pair of internal and external thermocouples on both sides of an insulator wall. As for the contact temperature, the steady-state values of these thermocouple signals are inferred by fitting an exponential curve onto the experimental temperature trace. In each direction of heat loss (upwards, backwards, and downwards), an average is taken of the three heat fluxes (W/m^2) obtained using Eq 5.11 at the circumferential locations instrumented. These mean values of heat flux are then multiplied by the total area available for heat conduction in each direction to work out the heat loss (W) in that direction. The sum of the heat losses in the three directions considered is subsequently subtracted from the electrical power to derive the net heat conducted into the seal.

Although the experimental facility and the seals tested are nominally axisymmetric, differences were found between the temperatures measured at the three circumferential locations instrumented (see Fig 5.11a). This is demonstrated in Fig 5.11b, which shows the discrepancy between the three contact temperatures measured during a test where the seal was pressurized prior connection of the transformer.

To investigate the origin of this unexpected circumferential temperature gradient, a test was run where the heater element was powered before opening the valves of the air supply system. The result is shown in Fig 5.12. It can be appreciated the uniformity in the three contact temperatures before the occurrence of airflow at time ~ 300 s. This implies that the rig itself operates in an axisymmetric manner. Then, when high pressure is delivered to the seal, the three temperature traces start to differ. This effect could be caused by a changing seal behaviour with angular position. Manufacturing



(a) Numbering employed for the circumferential locations of the heater assembly instrumented with thermocouples



(b) Contact temperatures measured at the circumferential locations shown in a)

Figure 5.11: Variation in measured temperatures with circumferential position during a test. In the example, the seal was pressurized throughout the time interval comprised between the two cyan dotted lines. Similarly, power was delivered to the heater element during the time interval represented by the red segment

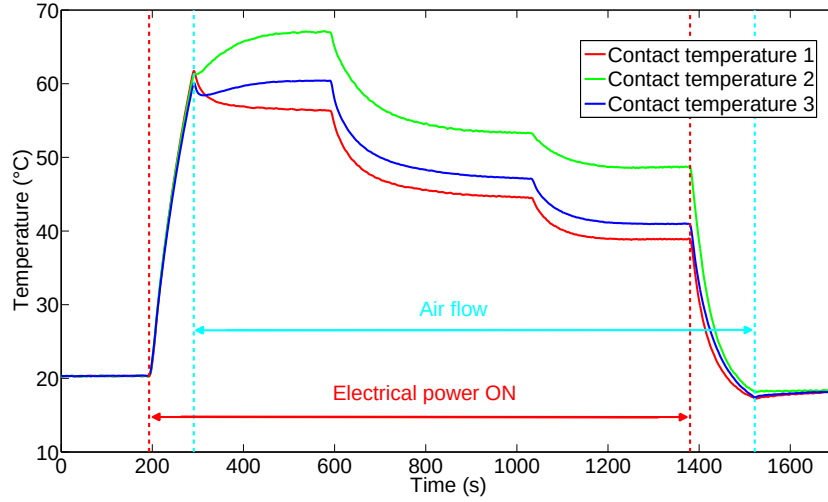


Figure 5.12: Variation in measured temperatures with circumferential position during a test (nomenclature explained in Fig 5.11a. In the example, the seal was pressurized throughout the time interval comprised between the two cyan dotted lines. Similarly, power was delivered to the heater element during the time interval represented by the red segment

tolerances and small deviations from nominal design values in seal geometric parameters can strongly affect the dynamics of the filaments. For instance, variations with angular position in the upstream coverplate gap leads to different levels of blow-down in a leaf seal. Consequently, heat transfer within the filament pack also varies circumferentially, which causes a non-uniform cooling of the contact area. As explained in Chapter 2, previous experience in brush seal testing has also indicated a variation with angular position in seal behaviour.

In spite of this circumferential variation in seal behaviour, the experimental values obtained at the three circumferential locations instrumented are averaged to characterize the overall seal heat transfer. For a case where the seal is in contact with a rotating component, it is expected that the nominal contact temperature presents a more uniform value. Although the behaviour of the seal may vary circumferentially, the effect of rotation contributes to attenuate any potential variation in nominal contact temperature. In the next section, the results derived from the data obtained in the test campaign are presented. It should be noted that they are calculated using values averaged circumferentially.

5.4 Experimental Data Results

A leaf and a brush seal were tested in the experimental facility. Different tests were carried out at power outputs from the transformer ranging between approximately 50-650 W. In each test, the electrical power was kept constant and the seal differential pressure varied to obtain several operating points as explained in the previous section. A total of 47 and 50 operating points were obtained for the leaf and brush seals respectively. By considering the ratio of the heat transferred to seal to electrical power generated, a form of heater efficiency (power split) can be defined as:

$$\varepsilon_{heater} = \frac{Q_{seal}}{Q_{electric}} \quad (5.12)$$

The term efficiency refers to the ability of the test rig to achieve the goal of transferring the majority of the heat generated towards the filaments of the seal. The variation of heater efficiency with leakage mass flow rate found during the experimental campaign is shown in Fig 5.13 for the leaf and brush seals investigated. In both cases, as the leakage mass flow rate increases, so does the heater efficiency. For an increment in leakage mass flow rate, an enhancement of the cooling of both seal and surroundings of the heater element occurs. However, the improvement in seal cooling is, in comparison, much larger than that of the surroundings. As shown in the figure, the net result is that increasing the leakage mass flow rate involves that a larger proportion of the heat generated is transferred to the seal. When the rig is operated at high seal differential pressures, and thus leakage mass flow rates, the efficiency found is ~80%. The seal geometry tested also has an influence on the heater efficiency. Seals with a high effectiveness at removing heat from the contact interface lead to higher efficiencies of the rig. As displayed in the figure, the brush seal tested results in greater efficiencies than the leaf seal.

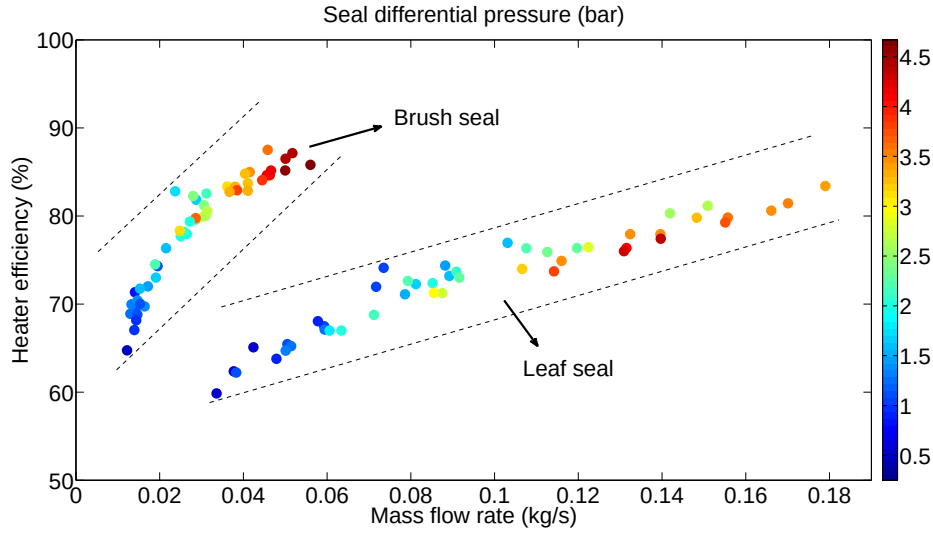
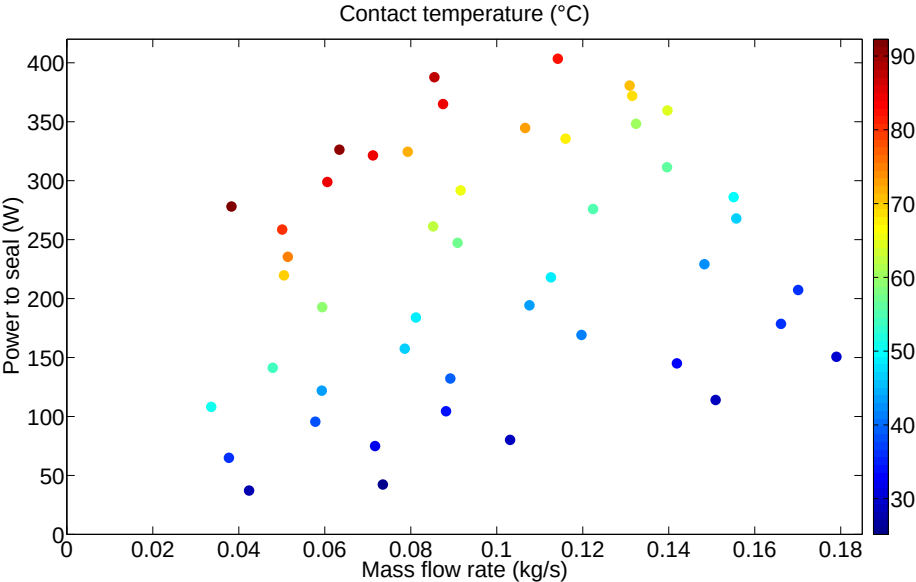
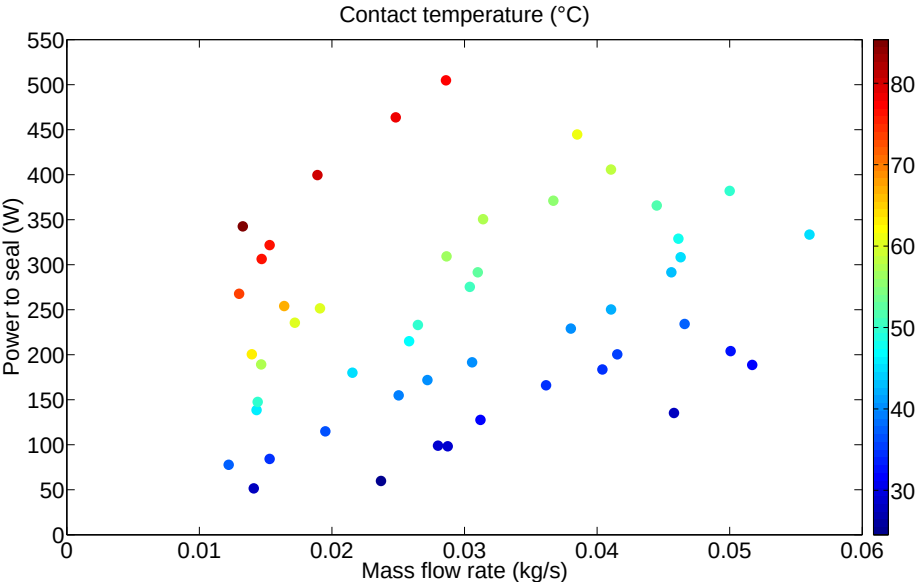


Figure 5.13: Heater efficiency versus leakage mass flow rate

As previously explained, to ensure the safe operation of the rig, the power output from the transformer was limited so that contact temperatures always remain below 100 °C. The contact temperatures obtained at the different seal powers (Q_{seal}) and leakage mass flow rates tested are shown in Fig 5.14. In all these experiments, the temperature of the incoming airflow was approximately ambient (18 °C). For both seals, the seal differential pressure was varied between 0-4.5 bar during the experimental campaign. However, as shown in the figures, the level of leakage mass flow rate found for the brush seal is much lower than that of the leaf seal, which indicates a better sealing performance of the former. In terms of power transferred to the seal, a similar range of values was obtained for both seals. Interestingly, the brush seal exhibits the same contact temperature as the leaf seal despite its lower leakage mass flow rates. This result is in line with a higher expected value of the seal thermal constant (K_{seal}) for the brush seal tested. In the figures, high contact temperatures occurring at low leakage mass flow rates can clearly be seen, where convective heat transfer within the filament pack is weaker. This indicates the risk of failure that exists at those engine operating conditions where high shaft speeds, and therefore frictional powers, and low seal differential pressures take place simultaneously. Although this regime may seem unlikely, a poorly fitted seal or a seal in a region of poorly controlled non-symmetrical thermal growth could be subject



(a) Leaf seal



(b) Brush seal

Figure 5.14: Contact temperatures measured at various leakage mass flow rates and seal powers

to such difficulties.

The main outcome of the experimental campaign carried out is a set of operating points where the heat conducted into the seal, leakage mass flow rate, and upstream gas and nominal contact temperatures are known. All these quantities can be plotted to consider the validity of the general form of seal thermal correlation, as proposed in Eqs 5.2 and 5.3. This is shown in Fig 5.15 for the leaf and brush seals tested.

According to the seal thermal correlation (Eqs 5.2 and 5.3), a linear relationship is expected to exist between the heat absorbed by the seal and the product of the square root of the leakage mass flow rate and the difference between the nominal contact and upstream gas temperatures. In Fig 5.15, it can be observed that for both brush and leaf seals, the experimental data points collapse remarkably well onto a straight line. This demonstrates that the linear dependence predicted by the correlation hold true. Therefore, linear regression was performed through the data points to find the value of the proportionality constant (K_{seal}). As discussed in Section 5.1, for a given working fluid, the seal thermal constant depends exclusively on the geometry and thermal properties of the seal. If we compare both seals tested, we can see that the brush seal presents a higher value of seal thermal constant, which implies that it is a more effective heat sink than the leaf seal. As such, for a given leakage and frictional heat generation rate, the brush seal is able to maintain a lower contact temperature. The final expressions for the thermal correlations of the two seals investigated are given by:

$$Q_{leaf} = 18.4 \dot{m}^{1/2} (T_c - T_{gas}) \quad (5.13)$$

$$Q_{brush} = 47.7 \dot{m}^{1/2} (T_c - T_{gas}) \quad (5.14)$$

As a means of comparison of the thermal performance of the two seal geometries, K_{seal} is not an appropriate indicator. Implicitly, it takes into account the size of the seal in terms of total area available for heat conduction through the filaments and convective heat transfer to the leakage flow. In an engine application, where heat is generated

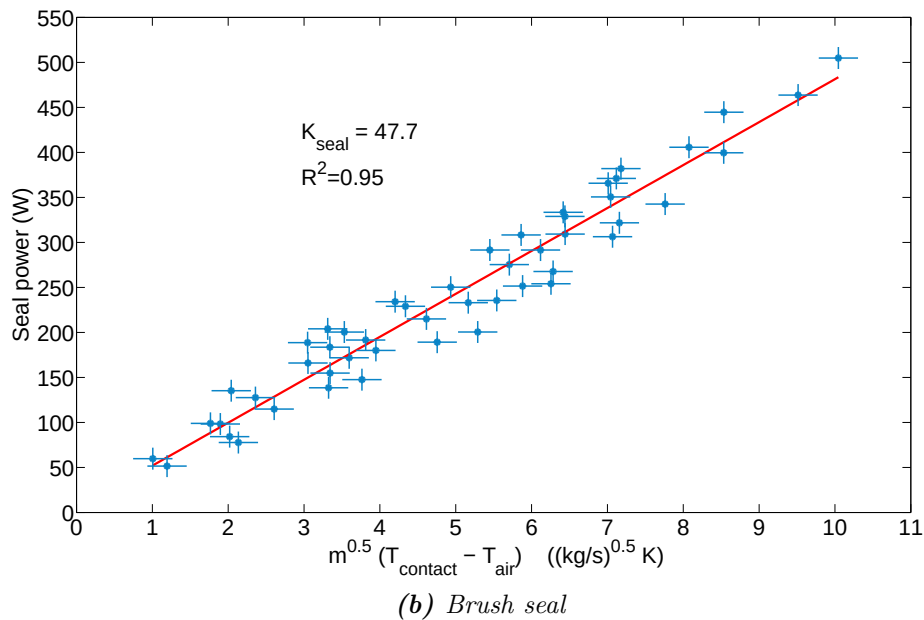
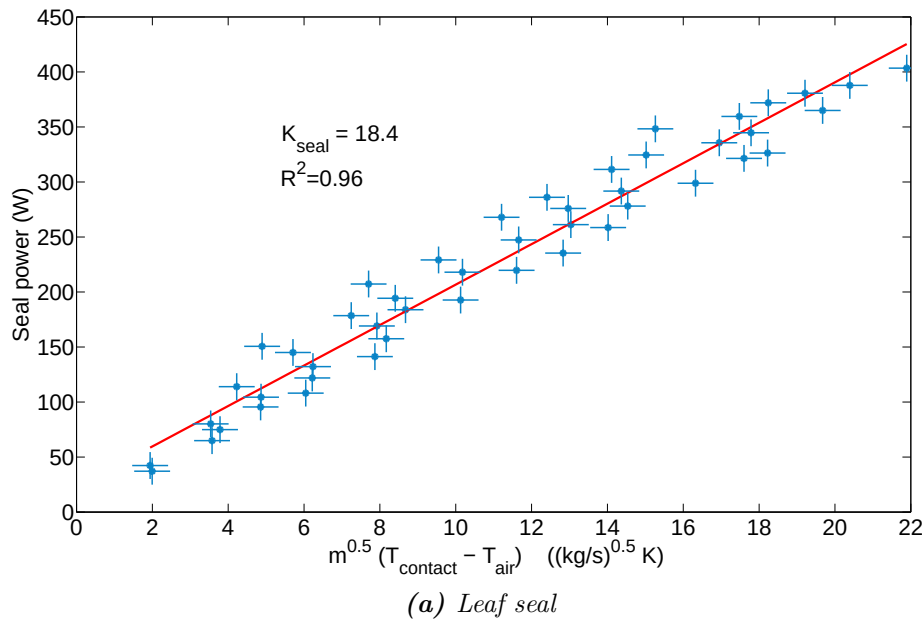


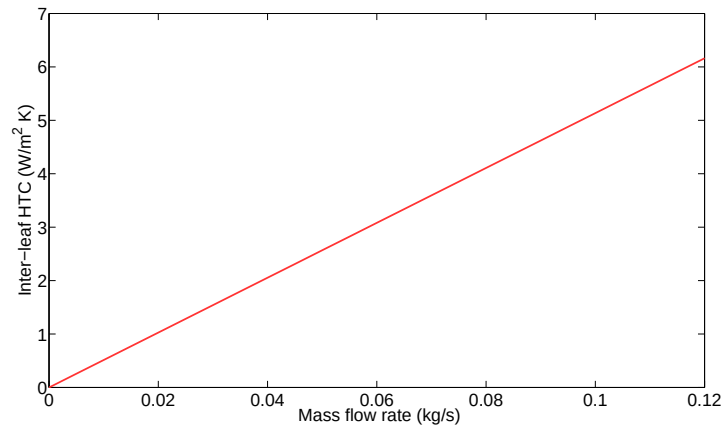
Figure 5.15: Seal thermal data with uncertainty bands, see Eqs 5.2-5.3

due to actual high speed rotary friction, the level of the frictional heat generation will depend on the seal type and size, with leaf seals typically being subject to much lower blowdown forces. Therefore, a higher value of K_{seal} does not ensure that a particular seal geometry will operate with a lower contact temperature rise - although it is desirable to maximise this value. Similarly, it does not necessarily mean that the seal geometry may be considered “thermally efficient”. If the actual aero-thermal characteristics of the inter-filament flow are the object of study, the size and thermal conductivity of the seal has to be accounted for. Consequently, to investigate the implicit convective heat transfer mechanisms behind Eqs 5.13 and 5.14, the 1D fin analytical model described in Section 5.1 can be employed. As previously discussed, the seal thermal correlation presents a functional dependency that is analogous to that obtained from a 1D fin analysis. By comparison of the fin equation (Eq 5.4) with the correlations derived (Eqs 5.13 and 5.14), a bulk inter-filament convective heat transfer coefficient (htc_{pack}) can be inferred from the experimental results. This parameter truly describes the convective heat transfer characteristics of the flow within the filament pack:

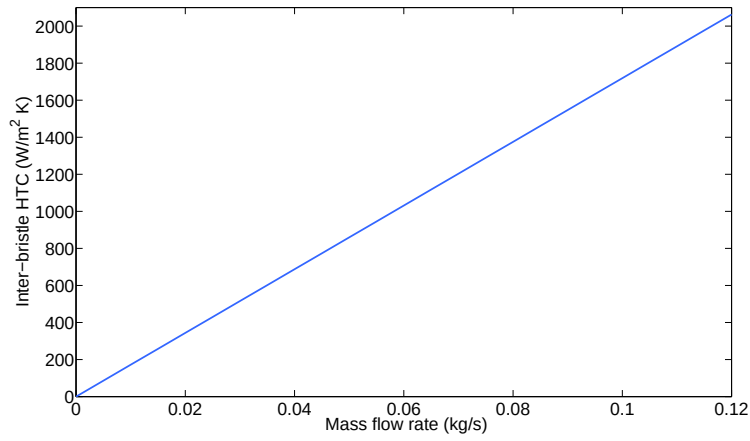
$$htc_{pack} = K_{seal}^2 \frac{1}{N^2 P A k} \dot{m} \quad (5.15)$$

where N is number of filaments, k the thermal conductivity of the seal material, and P and A the perimeter and cross sectional area of an individual bristle or leaf. For the geometry and thermal conductivity of the brush and leaf seals tested, the resulting htc_{pack} using Eq 5.15 is shown in Fig 5.16 over a representative range of leakage mass flow rates.

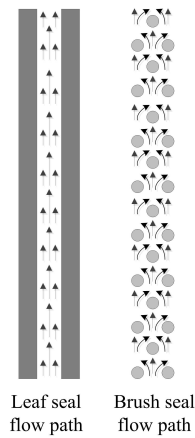
From the comparison between Figs 5.16a and 5.16b, it is evident that htc_{pack} in the brush seal is much higher than in the leaf seal. Fig 5.16c shows a schematic representation of the inter-filament flow path through leaves and bristles. In the leaf seal, the flow passage is essentially a rectangular duct with an infinite aspect ratio, while in the brush seal, the flow follows a more tortuous path, passing through several rows of staggered bristles and impinging continuously on their front surfaces. Intuitively, we expect convective heat transfer to be stronger in the latter scenario. In addition, as



(a) Leaf seal



(b) Brush seal

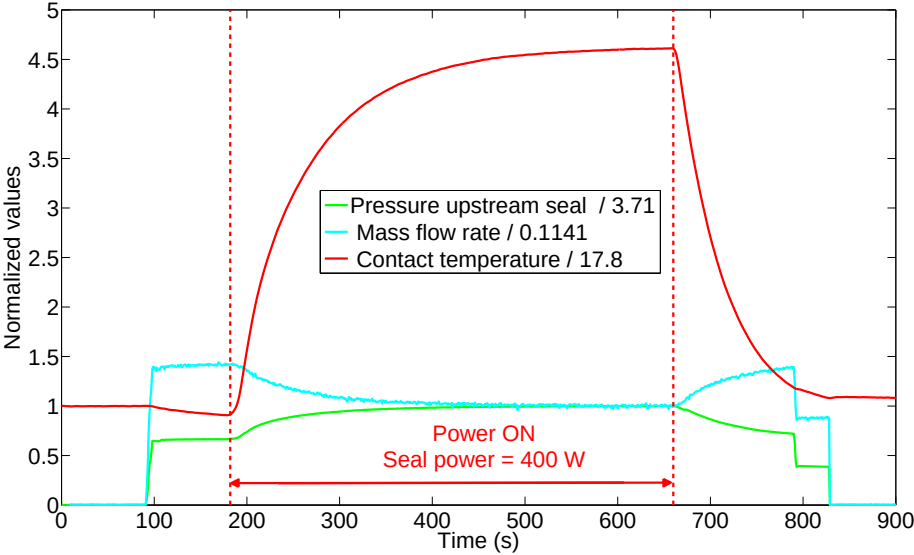


discussed previously, the level of leakage found for the brush seal was much lower than that of the leaf seal for the same range of differential pressures. This indicates a higher resistance to the passage of flow in the case of the brush seal. High pressure drops and thus, friction factors, are generally a clear symptom of heat transfer augmentation. Although the brush seal investigated is smaller than the leaf seal in terms of total filament perimeter and cross sectional area, the inherent higher htc_{pack} through bristles becomes a dominant factor, so that it can dissipate a larger amount of heat at the same operating conditions.

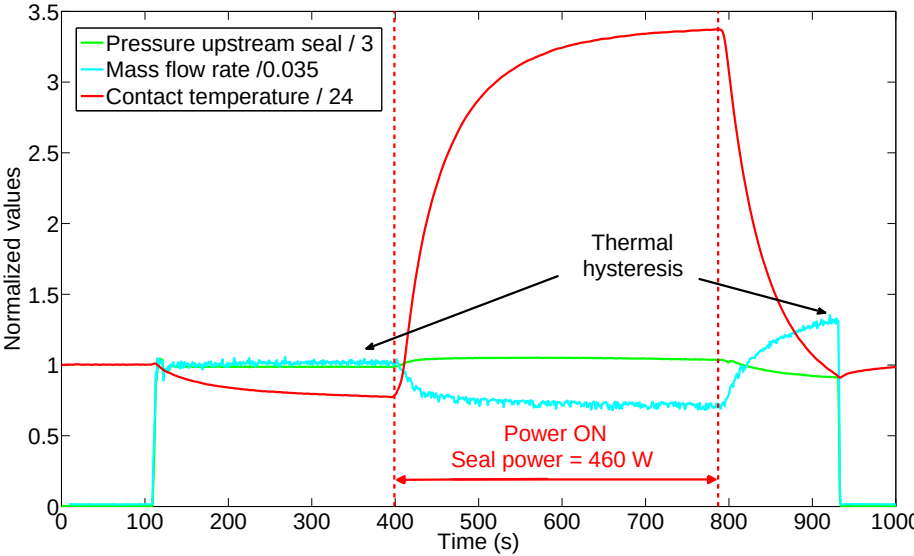
The htc_{pack} shown in Fig 5.16 represents an overall or mean value, locally, the convective heat transfer coefficient is likely to be higher in certain regions. Also, the actual area available for convective heat transfer might be smaller than the one estimated by the fin model, due to inter-filament contact with seal pressurization. According to Eq 5.15, this would result in higher htc_{pack} . The key result is the relative value found between the brush and leaf seal htc_{pack} (~ 333), rather than the absolute value. This demonstrates that the nature of the flow through bristles results in higher heat transfer rates than through a leaf pack. Nevertheless, in general brush seals present a higher radial stiffness than leaf seals (as explained in Chapter 2), so they tend to generate larger frictional powers. The resulting contact temperature rise may then be similar in both types of seals. For a mass flow rate of 0.06 kg/s , the empirical correlation Eq 5.6 predicts a htc in a circular micro channel of the same diameter as the inter-filament gap of $780 \text{ W/m}^2\text{K}$. This value is in line with the htc_{pack} levels found in the brush seal (Fig 5.16b).

5.5 Leakage Variation with Seal Heating

In the course of the experimental campaign, it was observed that, for both seals, the leakage mass flow rate tended to drop as the seal filaments were heated up. This behaviour is illustrated in Fig 5.17. In these examples, the pressure upstream of the seal was fixed first, which resulted in a certain constant leakage mass flow rate. In the figures, it can be observed the slight drop in contact temperature that exists during this portion of the test due to the cooling effect caused by the expanding flow through



(a) Leaf seal



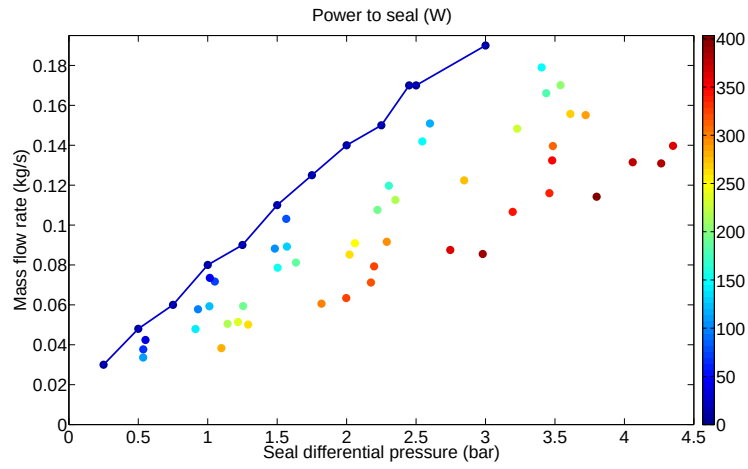
(b) Brush seal

Figure 5.17: Variation of leakage mass flow rate and seal upstream pressure with contact temperature

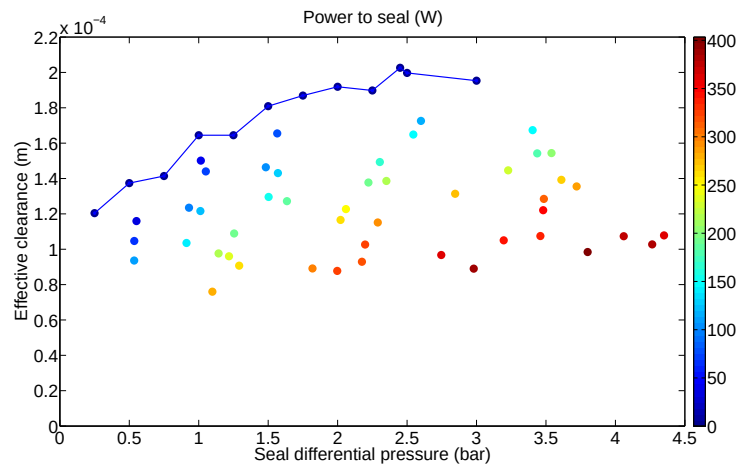
the system. As indicated by the red segments in the figures, the heater element was powered next. Consequently, the contact temperature rose. Simultaneously, in both cases, the mass flow rate started to decrease and the upstream seal pressure to increase. The rate of change of these three quantities reduced with time, until eventually a nearly constant value was reached. The subsequent shutdown of the transformer caused the system to cool down, and leakage mass flow rate and upstream pressure increased and decreased respectively towards the initial conditions. In the case of the leaf seal, it was found that the leakage mass flow rate at the end of the test was equal to that prior to connection of the transformer, as shown in Fig 5.17a. However, for the brush seal, a form of thermal hysteresis occurred, since the leakage mass flow rates obtained before and after the heating of the seal did not match, see Fig 5.17b.

For all the steady-state operating points obtained in the experimental campaign, the leakage mass flow rate can be plotted versus seal differential pressure to study the overall effect of the heat input to the seal on its sealing performance. This is shown in Figs 5.18a and 5.19a. For the leaf seal, it can clearly be seen that, at a given seal differential pressure, as the power absorbed by the seal increases the resulting leakage reduces. In the case of the brush seal, this effect is less pronounced, but the same general trend can be observed: at cold conditions, a lower seal differential pressure is required to attain a given leakage mass flow rate. The lower consistency in the brush seal results might be caused by the thermal hysteresis discussed before. As explained in Chapter 2, past experience has shown that brush seals are affected by mechanical hysteresis and therefore, their behaviour is less predictable. Moreover, in the set of tests carried out, it was observed that even at cold conditions (heater element off), the brush seal provided slightly different leakage rates at the same pressure setting in repeated runs.

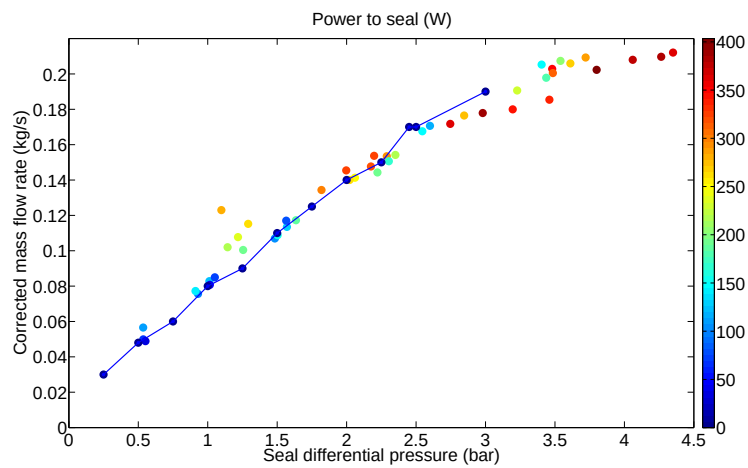
Different mechanisms could be involved in this unexpected behaviour of the seals. The heat transferred to the leakage flow increases its temperature and lowers its density, resulting in reduced leakage mass flow rates. This phenomenon was also reported by Qiu et al. (2015), as explained in Chapter 2 (see Fig 2.27). For a fixed seal differential pressure, increasing the rotor speed, and therefore the frictional heat generation rate, was



(a) Measured leakage mass flow rate

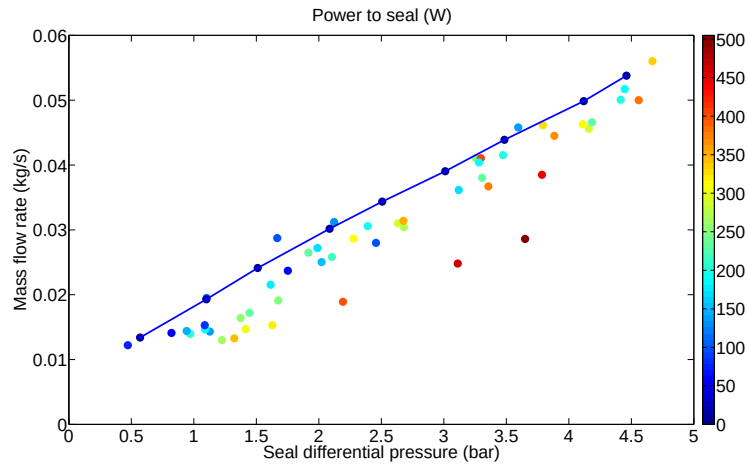


(b) Effective clearance (refer to Eq 5.16)

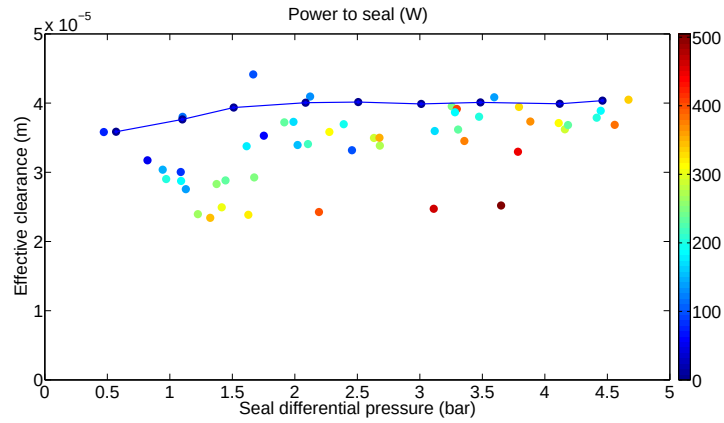


(c) Corrected leakage mass flow rate (refer to Eq 5.19)

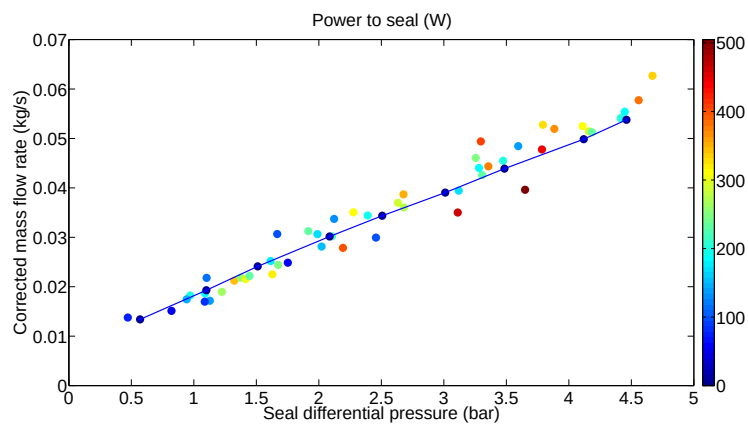
Figure 5.18: Sealing performance variation with seal heat input of the leaf seal tested



(a) Measured leakage mass flow rate



(b) Effective clearance (refer to Eq 5.16)



(c) Corrected leakage mass flow rate (refer to Eq 5.19)

Figure 5.19: Sealing performance variation with seal heat input of the brush seal tested

found by these authors to result in a drop in leakage. Moreover, the filaments of the seal suffer from thermal expansion when they are heated up, introducing an extra blockage that impairs the passage of the flow. In addition, the viscosity of the gas increases with temperature. The pressure drop through the seal is governed by the resistance induced by the viscous forces of the fluid. An increment in viscosity results in higher friction factors, and therefore, a higher upstream pressure is required to pump the same leakage mass flow rate. As explained in Chapter 2, the filament pack heat transfer and pressure drop mostly take place in the region around the filaments tips, especially at the rear of the pack. The backing ring and coverplate configuration of brush and leaf seals cause a convergence of the streamlines of the leakage flow towards that zone. A combination of the aforementioned effects, which are especially strong in the region described, could explain the phenomenon observed.

Generally, most seals are tested at ambient conditions, and then, scaled using effective clearance to engine operating conditions. Effective clearance (Morgan (2004)) represents the equivalent annular clearance, e , required of an ideal restriction to pass the same amount of flow as the seal:

$$e = \frac{\dot{m} \sqrt{T}}{\pi D F P_u} \quad (5.16)$$

where $\dot{m}$ is the leakage mass flow rate, T the leakage flow temperature (here the bulk temperature rise of the flow is considered, calculated as $T = T_u + \frac{Q_{seal}}{\dot{m} c_p}$, with T_u being the upstream gas temperature), D the seal inner diameter, and P_u the seal upstream pressure. F corresponds to the dimensionless flow function, given by:

$$F = \sqrt{\frac{2 \gamma}{R (\gamma - 1)} \left(\left(\frac{P_u}{P_d} \right)^{-2/\gamma} - \left(\frac{P_u}{P_d} \right)^{-(\gamma+1)/\gamma} \right)} \quad (5.17)$$

Above the critical pressure ratio (P_u/P_d) of approximately 1.9, $F = 0.0404$. In Figs 5.18b and 5.19b, the experimental values of effective clearance are plotted for the brush and leaf seals tested. As shown in the figures, the concept of effective clearance does not capture the thermal behaviour of the seals observed during the experimental campaign.

The change in leakage mass flow rate in the leaf seal, where we certainly do not

expect a significant variation in geometry, must be determined by changes in the viscous losses and changes in local velocity affecting the dynamic head. A simple model of the loss may be used to scale the leakage mass flow rates measured to standard conditions. If the pressure loss is assumed to follow a form such that

$$\Delta p = f \mu v \quad (5.18)$$

then viscous losses dominate, and the friction factor (f) may be expressed using the general form $f = C/Re^n$, with C and n being empirical constants. Eq 5.18 represents the pressure loss experienced by a laminar flow through a porous medium (this expression was used in Chapter 4, refer to Eq 4.4). In Eq 5.18, μ and v correspond respectively to the viscosity and velocity of the leakage flow. This equation can be used to scale the mass flow measured experimentally when the seal is heated ($\dot{m}$) to a certain reference cold condition (no seal heat input). If negligible changes in seal geometry with contact temperature are assumed, the resulting corrected (at reference conditions) mass flow, $\dot{m}_{corrected}$, can be shown to be approximately:

$$\dot{m}_{corrected} = \dot{m} \left[\left(\frac{\mu_{ref}}{\mu} \right)^{n+1} \frac{T_{ref}}{T} \right]^{1/(n-1)} \quad (5.19)$$

In Eq 5.19, the inlet gas temperature (T_u) is considered for μ_{ref} and T_{ref} , while μ is evaluated at the bulk temperature rise of the leakage flow ($T = T_u + \frac{Q_{seal}}{\dot{m}c_p}$). Figs 5.18c and 5.19c show the result of applying Eq 5.19 to the experimental data. For the leaf seal tested, it can be seen that when the mass flows measured are corrected, they collapse remarkably well onto the sealing performance curve at reference conditions (blue line, $Q_{seal} = 0$). For the brush seal, not as good a match is obtained, although the same trend is observed. As discussed before, brush seals are subjected to mechanical hysteresis, and therefore, they present a higher scatter in the leakage data, being their behaviour less predictable. The values of n used in Figs 5.18c and 5.19c were 0.94 and 0.6 for the leaf and brush seals respectively.

The leakage flow does not pick up heat uniformly. Thus the value of n required to

collapse the data if the bulk flow temperature is used for evaluating the fluid viscosity and density is a physically unrealistic value of 0.94 in the case of the leaf seal. A real temperature distribution shows heating of air near the rotor to values approaching the metal contact temperature, and this causes additional flow blockage in this area, reducing the mass flow rate more quickly. It has, however, been decided to base values on the bulk gas temperature increase, as this can easily be determined from a single energy balance. The effect of considering a higher gas temperature would be a drop in the value of n . Importantly, the variation in leakage with seal heating observed experimentally has been captured using Eq 5.19, which accounts for the changes in gas density and viscosity.

5.6 Summary

The suitability of different modelling strategies for a robust seal heat transfer model has been discussed. An empirical correlation, based on previous modelling and experimental work at Oxford, has then been proposed as the best tool for relating the amount of frictional heat absorbed by a filament seal and the two key operating parameters leakage mass flow rate and contact temperature. Using the existing hardware of the Seal Static Thermal Test Facility (introduced in Chapter 4), a new rig configuration has been built and implemented. Frictional heating is simulated using an electrical heater. This allows to separate the seal from the conductive / convective system through which heat is lost from the rotor, and control the amount of heat transferred to the seal (see Fig 5.3). The Oxford brush-leaf seal heat transfer model can thus be calibrated using this experimental setup. Experimental measurement of the electrical power, leakage mass flow rate, and contact temperature rise at various operating conditions generates heat transfer data to populate the semi-empirical model. A brush and a leaf seal geometries have been tested, and resulting contact temperature rise versus absorbed seal power and mass flow rate presented (see Fig 5.14). The experimental data obtained confirms that the functional dependency predicted by the existing Oxford seal thermal model holds true. Values of 47.7 and 18.4 $m^2 kg^{0.5} s^{-2.5} K^{-1}$ were found respectively for the empirical constant of the brush and leaf seal tested (see Fig 5.15). These figures, together with

further considerations into the nature of the filament pack heat transfer, reveal the inherent higher effectiveness of the brush seal at dissipating the frictional heat generated in comparison with the leaf seal. Circumferential variation in the behaviour of both seal geometries has been identified, which agrees with previous observed phenomena as discussed in Chapter 2. Notably, a key thermal effect has been observed during the experimental campaign: the drop in leakage rate with increasing contact temperature (see Fig 5.17 and 5.18). This variation in leakage performance must be accounted when designing a seal installation, which can be done using Eq 5.19.

Chapter 6

Modelling of Transient Experimental Tests Employing the Brush Seal Friction Heat Model

Brush and leaf seals are advanced filament seals employed in gas turbines to close clearances between a rotating shaft and a stationary land. They comprise thousands of bristles or thin plates that block the passage of the flow between zones where a pressure gradient exists. One of the challenges that these technologies face is the friction and wear caused by the contact between rotor and seal. The resulting heat generation can cause a failure of the sealing system, such as an excessive thermal growth of the rotor or the melting of the filaments. The accurate prediction of the temperature rise at the seal-rotor contact interface is therefore crucial to ensure the safe operation of the seal.

In Chapter 3, a methodology was proposed to model the temperature rise caused by the friction between a brush or leaf seal and the rotor. The problem of friction heat generation is regarded using an electrical analogy, where the thermal resistances of the two rubbing bodies, rotor and seal, must be determined. To that end, Chapter 4 and 5 investigate respectively the heat transfer characteristics of the flow in the seal vicinity and within the seal filament pack. In this Chapter, the knowledge gained throughout the previous chapters is brought together to build a model of an engine representative sealing

system. Predictions from the model are compared with experimental data, though this is necessarily only available for the case of the laboratory test rig.

The Oxford Engine Seal Test Facility (ESTF) is used to generate heat transfer data from a brush seal geometry running at engine representative rotor surface speeds and seal differential pressures (SDP). First, the instrumentation and operation of the experimental facility is described in detail. Infrared detectors installed in the rig allow the characterization of the rotor nominal contact temperature during the tests. The experimental results obtained are discussed, with a special focus on the thermal behaviour of the seal. Next, the transient operation of the rig is predicted using a finite element (FE) model of the rotor. This is combined with a separate computational fluid dynamics (CFD) model of the flow inside the rig cavity and the empirical correlation derived in Chapter 5. In the CFD simulations, the seal is modelled as a porous medium and the methodology outlined and validated in Chapter 4 is employed to derive the convective heat transfer coefficient distribution on the rotor surface. Thus, the CFD model provides the FE model with its convective boundary conditions. The partition of the total friction heat between seal and rotor is calculated using the empirical correlation derived in Chapter 5. Finally, a comparison is made between the model predictions and experimental measurements. This shows that the model can capture the thermal behaviour satisfactorily, and is a viable means of modelling the system. This methodology is represented in the block diagram showed in Fig 6.1.

6.1 Engine Seal Test Facility (ESTF)

6.1.1 Description of the experimental facility

The Engine Seal Test Facility (ESTF) allows the performance of engine scale brush and leaf seals at representative differential pressures, surface speeds and seal housing eccentricities to be investigated. Notably the rig runs at ambient temperature, with seals of 150 – 330 mm nominal bore diameter. Seal frictional power dissipation of up to 3 kW has been observed in experiments, which result in a typical temperature rise at

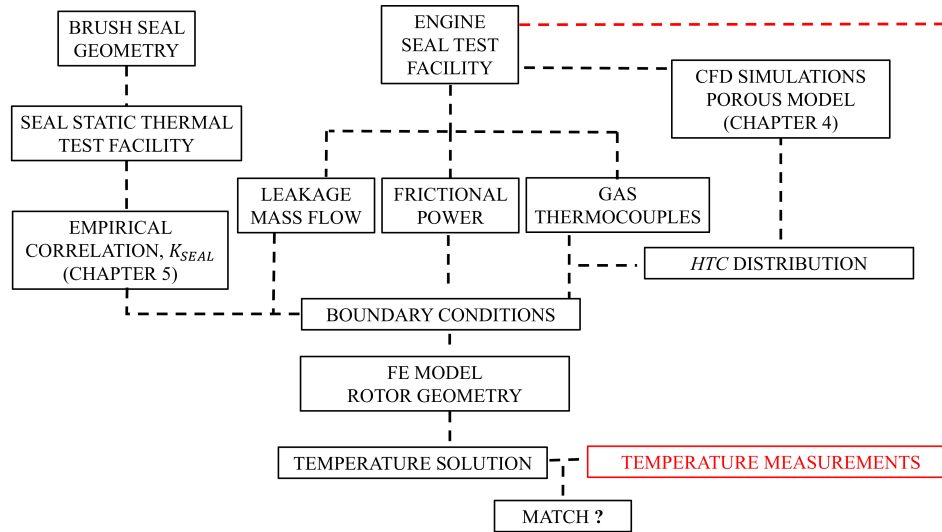


Figure 6.1: Flowchart representing the methodology proposed

the seal / rotor contact interface of about 70 K (averaged over the nominal seal contact area).

The engine range of expected power dissipation can be found by considering expected brush seal stiffness, bristle friction, operating speed and seal radius. Dogu and Aksit (2006) use data from Chupp and Prior (1997) and Short et al. (1996) to define an expected operating seal stiffness envelope from 55 kPa/mm to as much as 1100 kPa/mm. Thus, for a typical engine seal with wire diameter 0.150 mm, 15 rows of bristles, a radius 500 mm, operating at a surface speed of 300 m/s and an interference of 0.25 mm the maximum power dissipated could be as high as 65kW. It is clear that the highest engine rotor temperature rises are not matched during experiments, however, only an appreciable and measurable rise is required to determine the physics underlying the heat transfer process.

The ESTF has been used continuously in brush and leaf seal testing campaigns over the last 24 years, being first developed by Hogg et al. (1991) for the characterisation of gas thrust bearings. The rig has been continuously upgraded, with the most recent test programmes being reported in Jahn (2008) and Pekris (2010), Pekris et al. (2016) for tests on leaf and brush seals respectively.

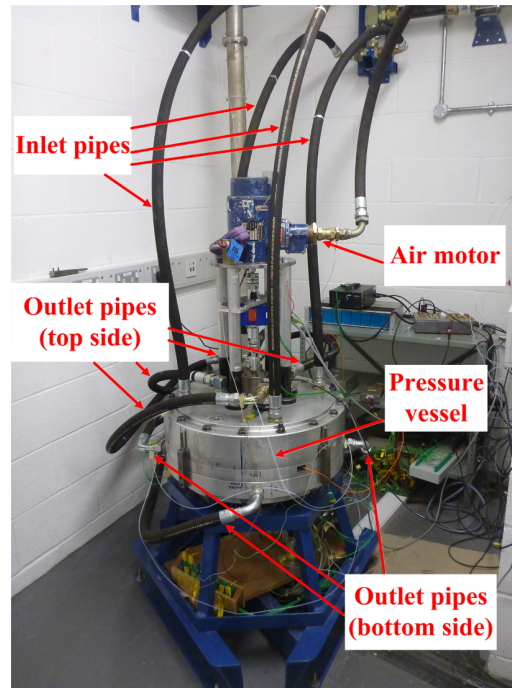


Figure 6.2: Engine Seal Test Facility (ESTF)

The original test facility was developed by O'Neill (1993), who modified a high speed rig designed by Hughes (1992), to study the mechanics of brush seals. Several modifications and improvements were made by Wood (1998), Owen et al. (2003), Morgan (2004), Jahn (2008) and Pekris (2010). Prior to running the experiments presented in this thesis, a major rig re-commissioning with improved instrumentation was conducted by Drs Andrew Owen and Simon Shun, and their contribution to the test operation is freely acknowledged. A photograph of the ESTF along with a schematic showing typical dimensions of the solid and fluid space when the test rig is equipped with brush seals are shown in Fig 6.2 and 6.3.

As shown in the cross sectional view in Fig 6.3, the working section of the rig is formed by a rotor in contact with a pair of brush seals (referred as top and bottom seals) at its outer diameter. The brush seals contact the rotor at two locations approximately equidistant from the rotor axial line of symmetry. The rotor presents two features which are key for the measurement and characterisation of the rotor temperature: the two annular wings or thin-walls (each is 3.2 mm in thickness and 14.5 mm in axial length)

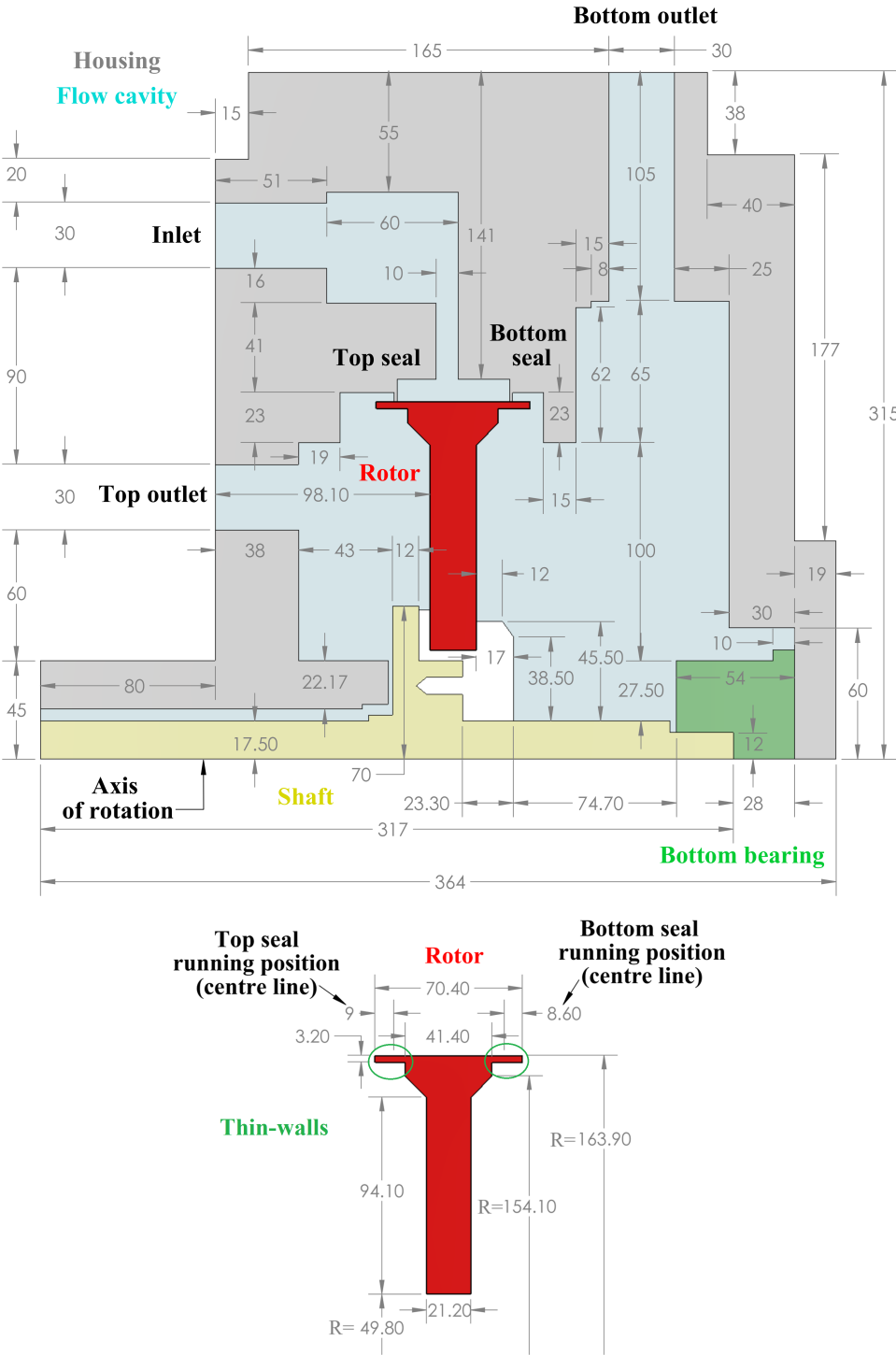


Figure 6.3: ESTF cross sectional schematic (all dimensions in mm). Note that in reality the axis of rotation is oriented vertically (see Fig 6.2)

halfway along which the seal interface lies. The importance of these features, referred to in this thesis as thin-walls, is explained in Section 6.4.1. The rotor sits in a cavity, the inlet of which is pressurized during the tests, after the rotor has been raised to its maximum operating speed. In Fig 6.2, the four inlet pipes that supply air at ambient temperature and a maximum of 10 barg to the seals are labelled. The cross-sectional schematic in Fig 6.3 shows the passages formed by the discrete inlet pipes (the 51 x 30 mm region), which then feed a continuous annular plenum upstream of the seals. Note that while the inlets are discrete, there is an annular gallery running around the test rig where flow is distributed prior to its introduction towards the rotor. The high pressure air expands through the seals into the two downstream cavities at the top and bottom sides of the rotor. The flow subsequently exits the rig via eight exit pipes (four pipes per seal). The performance of the seal is strongly dependent on the pressure difference, as opposed to pressure ratio, and so it is appropriate to maintain the pressure downstream of the seals equal to ambient. Previous work by Franceschini et al. (2010) has shown that the mechanical blowdown and bristle stiffening are entirely explained by the pressure difference across the seal, however at the seal exit over-expansion of the flow will depend on the pressure ratio across the final row of bristles, when this places the flow into a transonic or supersonic regime.

The test rig is designed to house leaf and brush seals, although labyrinth seals can also be mounted in the rig for calibration purposes i.e. to determine the windage. Two identical seal geometries are always installed and tested in the ESTF. Since the geometric features of the rig that may affect seal performance are common to both seals, it is expected that they present a similar mechanical and thermal behaviour. Therefore, where physical quantities such as friction torque or mass flow leakage, a total value is measured, and assumed to be due to both seals in equal part. This strategy, and transient rundown tests, reduces the axial loading on the rig bearings which reduces the complexity of the rig considerably. The dimensions of the rotor are shown in the bottom half of Fig 6.3. The rotor is axially symmetrical, although eccentricity can be achieved of up to 0.5 mm using brass bushing to simulate a rotor excursion event e.g. a sharp

turn of the aircraft causing gyroscopic forces on the shaft, or a rotor whirl transient. Similarly fixed seal housing offsets relative to the rotor axis can be applied to simulate poor build tolerance of the seals. The rotor is made from Inconel 718, and presents the same outer diameter of 328 mm as the seals installed in the SSTF described in Chapter 4. The inner diameter of the rotor is connected to a stub shaft through a clamped bushing which is coupled to an air-motor. A separate supply line provides high pressure air to the motor.

6.1.2 Instrumentation and data acquisition

The ESTF is designed to characterize the leakage, torque and thermal behaviour of the pair of seals tested. It allows for accurate measurements of the following performance parameters: seal pressure difference, rotor temperature, frictional torque and power, rotor speed and mass flow leakage. In addition, other quantities that are important for analysis and modelling of the experimental data, rig control and safety are: gas temperature, air motor mass flow rate, rotor vibration, and rotor centrifugal growth. In this chapter, only the primary rig data are discussed.

Gauge pressures are measured from static tapings located in the internal cavity upstream of the seals and also downstream of each seal (see Fig 6.4). Using these the seal differential pressure and target upstream pressure are set. The value of the pressure downstream of each seal is close to ambient, the main downstream flow restrictions being the flow in the exit pipework, a collecting plenum and an orifice meter situated in the exhaust line. The total mass flow rate that leaks through both seals is measured using a 75 mm diameter orifice meter designed to BS1042 located downstream of the facility. The calculation of the mass flow rate using this methodology requires the measurement of the upstream pressure, pressure drop across the orifice plate and the local gas temperature. The mass flow rate of air that expands in the turbine of the air motor is also measured for the calculation of the torque that accelerates the rotor, and thus the heating of the rotor prior to the “real” experiment. Differential pressure transducers are used across the orifice plates to improve the accuracy of mass flow rate calculations. All pressure

transducers were calibrated in-situ using a dead-weight tester calibrated to a national standard. Temperature dependent pressure transducer offset voltages were measured at the start of each test. Atmospheric pressure is measured using a Fortin barometer.

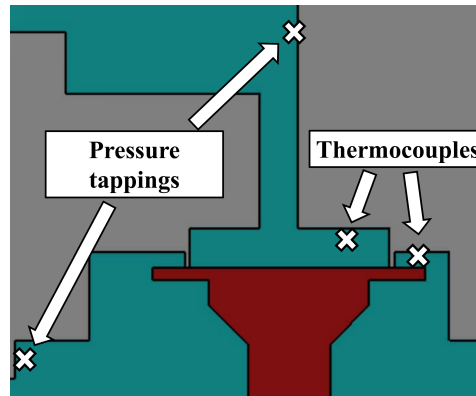


Figure 6.4: Locations of pressure tapings and thermocouples (ESTF)

The test facility is instrumented with K-type gas thermocouples. Gas thermocouples are installed in close proximity to the seals at eight circumferential locations and in the bearing housing. The gas thermocouples are located upstream of the seals and downstream of the bottom seal (see Fig 6.4). As previously discussed, a similar behaviour is expected between both seals tested and therefore the downstream temperature measured is assumed to be equal for both. The temperature at the inlet and outlet of the air motor and at the orifice plates is also recorded.

The angular speed of the ESTF rotor is measured using two optical rotary encoders. These devices consist of a disc with marks on its surface of different light reflection properties. The encoder disc rotates under a beam of light whose alternating reflection on the disc surface is sensed by a photodetector. The output of the sensor is dependent upon the speed of rotation. One of the encoders has a single non-reflective black line painted on its disc and is used for low angular speeds. The second encoder reads a disc with 16 non-reflective lines around its circumference and covers the range of high speeds.

The torque acting on the rotor is derived during post processing of the test data from the differentiation of the speed signal. The acceleration/deceleration of the rotor can be

related with torque by:

$$T_T = I_R \frac{d\omega}{dt} \quad (6.1)$$

where I_R is the rotor inertia and ω the rotational speed. The torque derived using Eq 6.1 corresponds to the total torque exerted on the rotor (T_T). Since only the torque caused by the friction with the seals is of interest (T_s), this contribution can be computed by subtracting all parasitic torques to the total torque. The friction resistance of the shaft bearings (T_b), and the shear stresses between the rotor surface and the surrounding air (windage torque, T_w) will always be part of the total torque. In addition, when the rotor is driven by the air motor, there is an additional torque term (T_m).

$$T_s = T_T - T_b - T_w - T_m \quad (6.2)$$

To determine the magnitude of T_b and T_w , separate tests were carried out. Labyrinth seals were installed in the facility to generate a flow field that is expected to be similar to that induced during a test with brush seals but without rotor contact. Total torque was computed from the deceleration of the rotor from 7000 RPM to rest, the air motor being unpowered during this transient. Since no friction with the seals exists in these tests, the torque obtained is caused by the aerodynamic windage and internal friction in the system. Several tests with a pressurized cavity were carried out to find the calibration constants required for the equation derived by Wood (1998):

$$T_b = T_0 + C_b \Delta P \quad (6.3)$$

$$T_w = C_w (\rho^{4/5} \omega^{9/5}) \quad (6.4)$$

where C_b and C_w are the calibration constants and ΔP is the pressure difference between the downstream rotor cavities and atmosphere. Wood's model assumes that the bearing torque can be split into a constant (T_0) and a term that linearly depends on the differential pressure across the drive shaft seals. In normal experimental tests, the seal-induced

torque is obtained by correcting the total torque using the windage and parasitic torque measured during these calibration runs. As will be explained in Section 6.1.4, the standard experimental procedure starts with the acceleration of the rotor driven by the air motor. The rotor is subsequently allowed to freely decelerate to rest, this event being where the most valuable data are recorded. The torque exerted by the air motor in the first part of a test is calculated using the mass flow rate and pressure and temperature ratio in the turbine of the motor. Finally, frictional heat is calculated as the product of T_s and rotor speed.

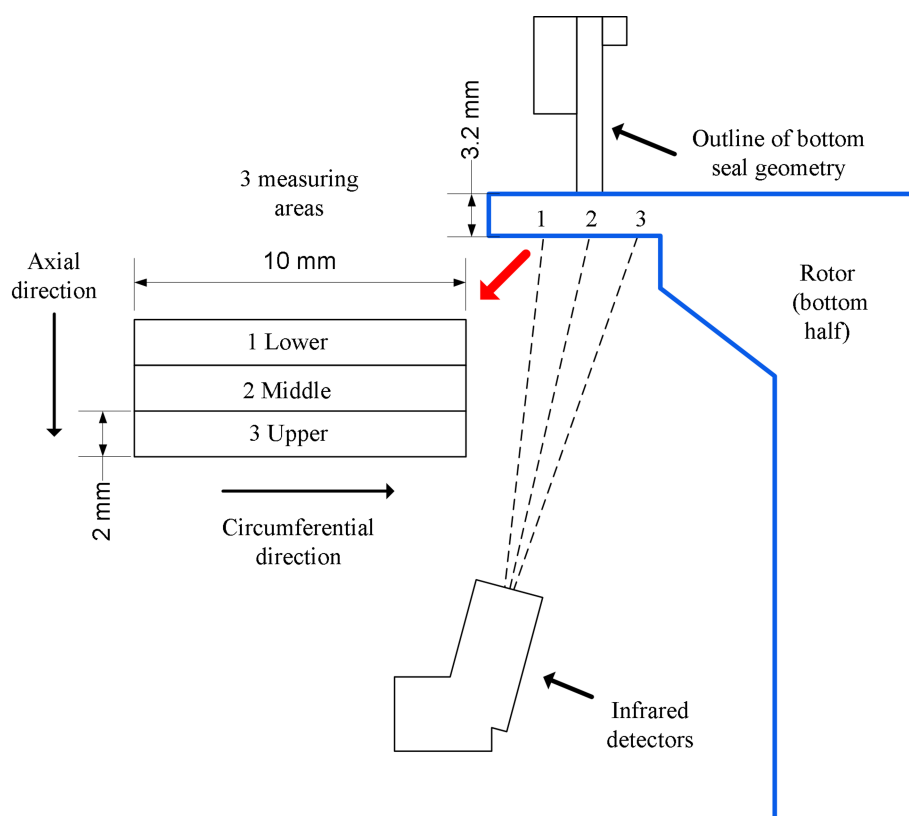


Figure 6.5: Measurement of rotor temperature (bottom side)

A fundamental parameter in friction heat generation is the nominal contact temperature attained. The temperature at the back of the rotor bottom thin-wall is measured using three infrared detectors. These are mounted on the inner wall of the containment pressure vessel, on the bottom side of the rotor. Given the small thickness of the wall and the high thermal conductivity of the rotor material, the temperature measured is a good

estimation of the nominal contact temperature, particularly at instances where the back wall of the rotor is momentarily adiabatic. A cartoon of the thermocouple installation is shown in 6.5. Each infrared sensor measures the temperature over a rectangular window that is 2 mm wide (axially) and 10 mm long (circumferentially). One of the sensors (referred to as the middle sensor) is pointing at the axial location that coincides with the running position of the bottom seal. The other two sensors focus on sections that are 2 mm (axially) above and below (referred to as the upper and lower sensors respectively). The surface of the rotor where the cameras are reading was sprayed with matt black paint of known emmissivity. The calibration of the infrared detectors was performed in situ. A thin aluminium soft-drink can, whose external surface was also painted black using the same paint and spraying technique, was filled with boiling water and allowed to cool to ambient temperature.

The can was placed at the foci of the three infrared sensors and their output voltages recorded. A K-type thermocouple glued to the can surface monitored the temperature during the cooling period, which spanned the range expected during experiments. By comparison of the thermocouple reading and the sensor output voltages, the data obtained allowed the derivation of correcting factors for the pseudo-temperature output from each sensor

The test facility is remotely controlled and monitored from a PC using a *LabVIEW* programme which also records the test data. All thermocouples and pressure transducers are plugged into an expansion board, whose output is connected onto the computer low-speed card. These channels are sampled at a frequency of 20 Hz. A high-speed card is installed in the PC capable of sampling up to 100 kHz. High speed sampling is required for the measurements of rotational speed and instantaneous position, and these are performed via the high-speed card.

6.1.3 Accuracy considerations

A detailed uncertainty analysis of the measurement system installed in the ESTF is presented in Jahn (2008) and Pekris (2010). The bias and precision errors in the quant-

ities that are directly measured were obtained from the specifications of the sensors employed and the data acquisition system. The errors in the derived values were estimated using a jitter-method as explained in Moffat (1988). The input quantities required for the calculation of a certain parameter are independently perturbed by their bias and precision errors to determine how they propagate to the solution. Tables 6.1 and 6.2 show the error in the measured and calculated quantities as a summary of the error analysis previously referenced. The parasitic torque calibration uses measurements from the current experimental campaign.

Measurement	Bias Error	Precision Error
Pressure – 1 barg transducer	250 Pa	110 Pa
Pressure – 16 barg transducer	3800 Pa	1800 Pa
Pressure – 20 barg transducer	4745 Pa	2500 Pa
Pressure – 1 bar differential	250 Pa	110 Pa
Temperature (K-type)	2.5 °C	0.2 °C
Temperature (infrared sensor)	2.5 °C	1.6 °C
Speed	± 1 %	4.7 % @ 100 RPM 0.11 % @ 3500 RPM 0.12% @ 7000 RPM

Table 6.1: Error in quantities directly measured, Pekris (2010)

Derived value	Bias Error	Precision Error
Mass flow rate	$1.91 \cdot 10^{-2}$ kg/s	$5.5 \cdot 10^{-3}$ kg/s
Torque	0.035 Nm @ 1000 RPM	0.035 Nm @ 1000 RPM
	0.121 Nm @ 3000 RPM	0.120 Nm @ 3000 RPM
	0.112 Nm @ 5000 RPM	0.110 Nm @ 5000 RPM
	0.370 Nm @ 7000 RPM	0.370 Nm @ 7000 RPM
Power	8 W @ 1000 RPM	8 W @ 1000 RPM
	41.6 W @ 3000 RPM	38.4 W @ 3000 RPM
	74.7 W @ 5000 RPM	57.6 W @ 5000 RPM
	267 W @ 7000 RPM	380 W @ 7000 RPM

Table 6.2: Error in derived values, Pekris (2010)

6.1.4 Test description

The experiments carried out with the ESTF follow an experimental procedure that consists of two main parts:

First, high pressure air is expanded in the turbine of the air motor, which in turn accelerates the ESTF rotor. Accordingly, this portion of the test is termed run-up. During this stage, the pressure difference across the seals and leakage flow are zero since the valves upstream of the rig cavity remain closed. The total torque acting on the rotor is composed of four different contributors: air motor, aerodynamic windage, shaft bearing friction and, in some cases, friction with seals due to an initial build interference. Although the rotor speed can be adjusted up to 11000 RPM, a maximum speed of 8000 RPM is set in order to avoid any requirement to conduct two axis balancing or a rotor failure due to vibrations or excessive deflections of the rotor thin-walls. Prior carrying out the test programme, the rotor was balanced by attaching counterweights at appropriate locations. In addition, rotor run-out (maximum radial displacement over a single revolution) was adjusted in situ to acceptable limits ($\pm 20 \mu\text{m}$) during rig assembly. Previous work at Oxford (Wood (1998), Pekris (2010)) suggests that during the run-up, the test rotor employed is likely to experience a centrifugal growth of the order of $60 \mu\text{m}$

at the thin-wall sections.

When the rotor reaches a speed slightly higher than 7000 RPM, the second part of the test starts. The air motor is shut down and high pressure delivered to the seals. For the seal tested, the leakage flow and possible contact caused by an aerodynamic force radially inwards on the filaments (blow-down) increases a retarding shear stress on the rotor. This seal frictional torque, together with the bearing and windage torques, decelerate the rotor to rest. Consequently, this portion of the test is named run-down and is the most relevant since the seals are pressurized. The terms run-up and run-down will frequently be used in the next sections.

6.2 Experimental Data Results

A pair of identical brush seals were installed and tested in the ESTF. Table 6.3 shows the key dimensions of the brush seal investigated. From all the tests carried out during the experimental campaign, six have been selected for analysis and modelling. These correspond with seal differential pressures (SDP) of 1, 1.5, 2, 2.5, 3 and 3.5 bar. Higher values of SDP resulted in unexpected high rotor vibrations, and the data obtained were difficult to interpret and of limited value. Thus, they have been discarded.

Bristle pack width (mm)	1.24 ± 0.10
Lay angle ($^{\circ}$)	45 ± 5
Seal diameter (mm)	327.74 ± 0.127
Upstream coverplate diameter (mm)	348.53 ± 0.241
Downstream coverplate diameter (mm)	330.14 ± 0.139
Bristle diameter (mm)	0.100
Downstream coverplate thickness (mm)	1.60 ± 0.10
Number of filaments	98,832 - 101,672

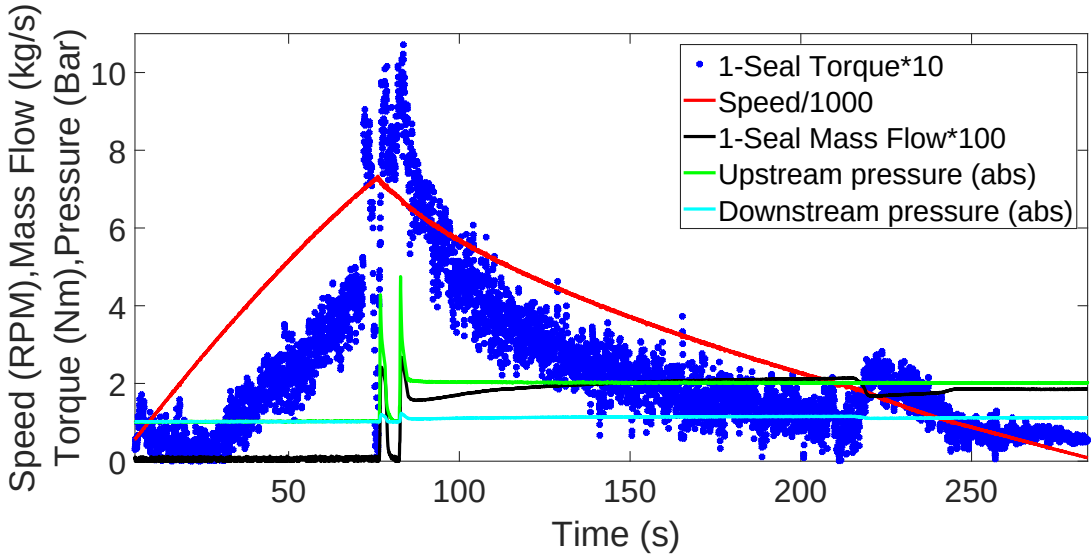
Table 6.3: Main dimensions of brush seal tested (manufacturer: EATON)

As explained in Section 6.1.4, a test consists of two main parts. First, the rotor, driven by an air motor, is accelerated to a speed of ~ 7500 RPM (run-up). Second, the

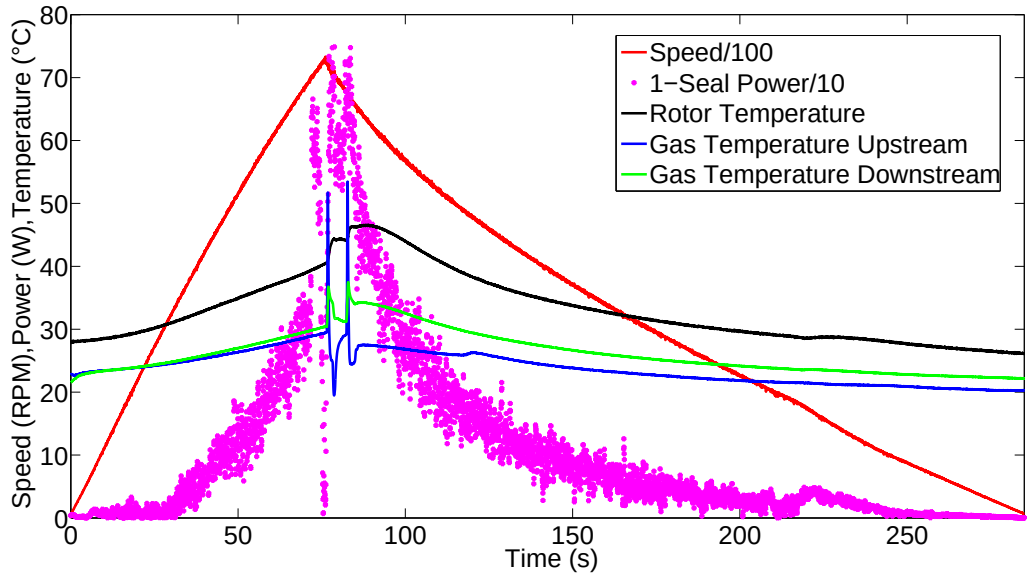
air motor is shut down and high pressure is supplied to the seals, which results in a deceleration of the rotor (run-down). During the entire test, rotor speed, pressure and temperature data are recorded to characterize the mechanical and thermal behaviour of the seals. Figs 6.6-6.11 show the time history of the main quantities measured during the six test cases under consideration. In the upper plot of each figure, the mechanical quantities rotor speed, torque, mass flow leakage, and the absolute pressure upstream and downstream of the seals are shown. The mass flow rate plotted corresponds to the leakage through a single seal. The torque trace represents the friction torque exerted on the rotor by a single seal. The upstream pressure is measured in the cavity upstream of both seals while the downstream pressure plotted is the average of the pressures downstream of the two seals. The thermal plot, at the bottom of each figure, illustrates the frictional power dissipated and the temperature of the rotor and air. The power curve is derived from the rotor speed and half the friction torque of the two seals. Thus, it corresponds to the frictional power generated by a single seal. The rotor temperature trace is obtained from the average of the infrared camera readings. As explained in 6.1.2, these temperature measurements are taken at the back of the thin walls of the rotor. The gas temperature is measured in the cavity upstream of both seals and downstream of the bottom seal at 8 circumferential positions. The gas temperature traces represented correspond to the average of the eight thermocouple signals. In order to fit all the quantities considered in a single plot with a reasonable scale of the axes, and visibility, some data are multiplied by a scaling factor as shown in the legends.

From the summary plots, it is clear that the run-up phase for the tests takes about 80 seconds and here two short spikes in the data are shown separated by 5 - 10 seconds. These correspond to the use of a short pulse in upstream pressure at the beginning of the run-down of each test (at ~ 7200 RPM), followed by the main flow being continuously introduced. This procedure, derived from previous experience in brush seal testing (Pekris (2010)), is carried out to overcome any residual hysteresis in the bristle pack from previous test runs or the run-up.

The short pulse is followed by a continuous supply of high pressure air at the target

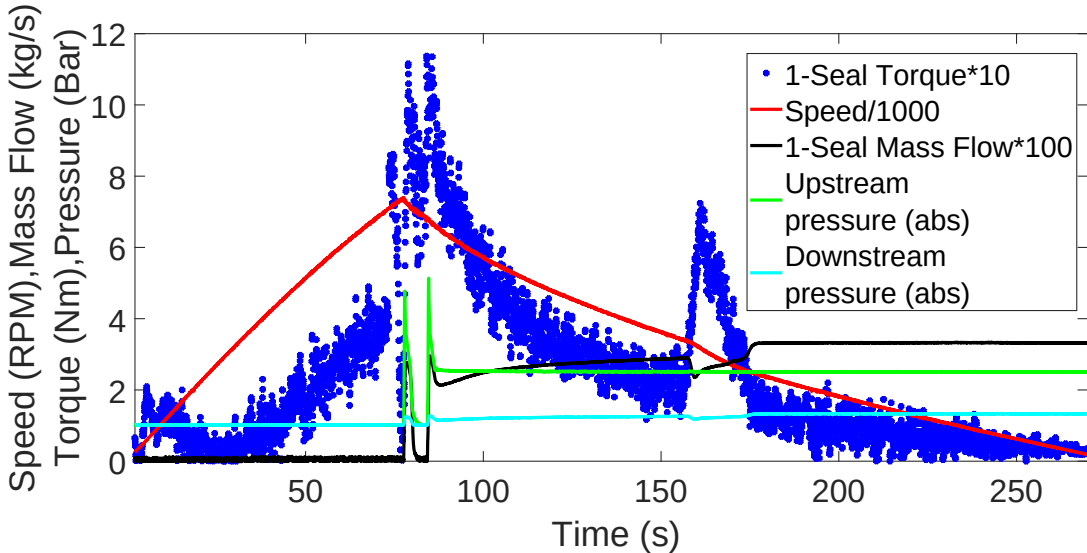


(a) Mechanical summary plot

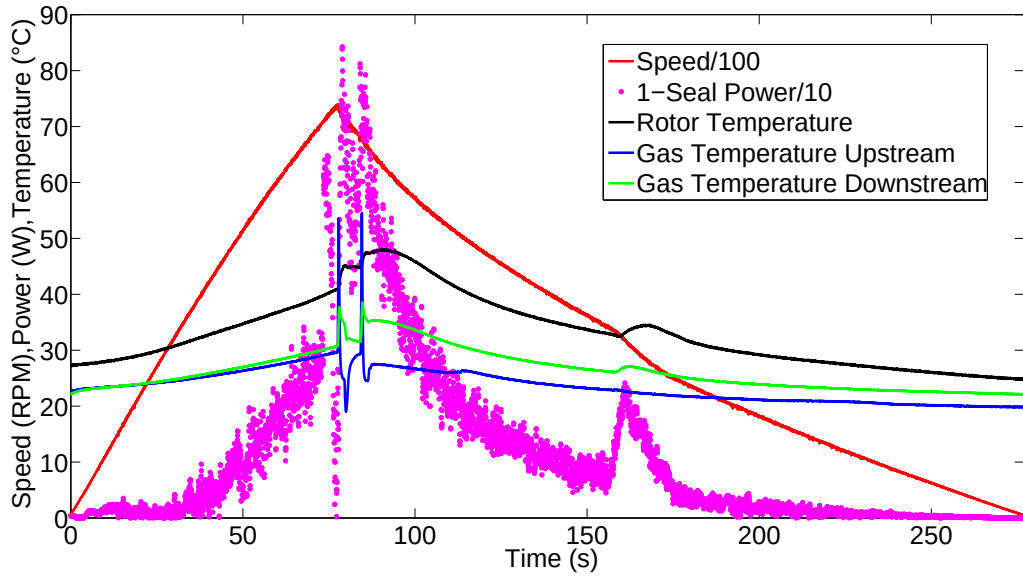


(b) Thermal summary plot

Figure 6.6: 1 bar SDP test

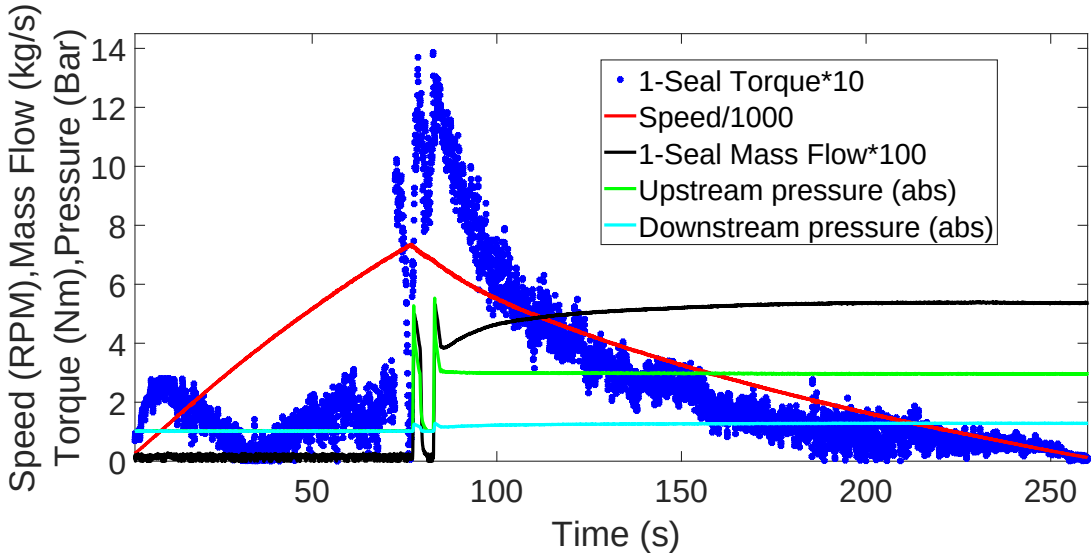


(a) Mechanical summary plot

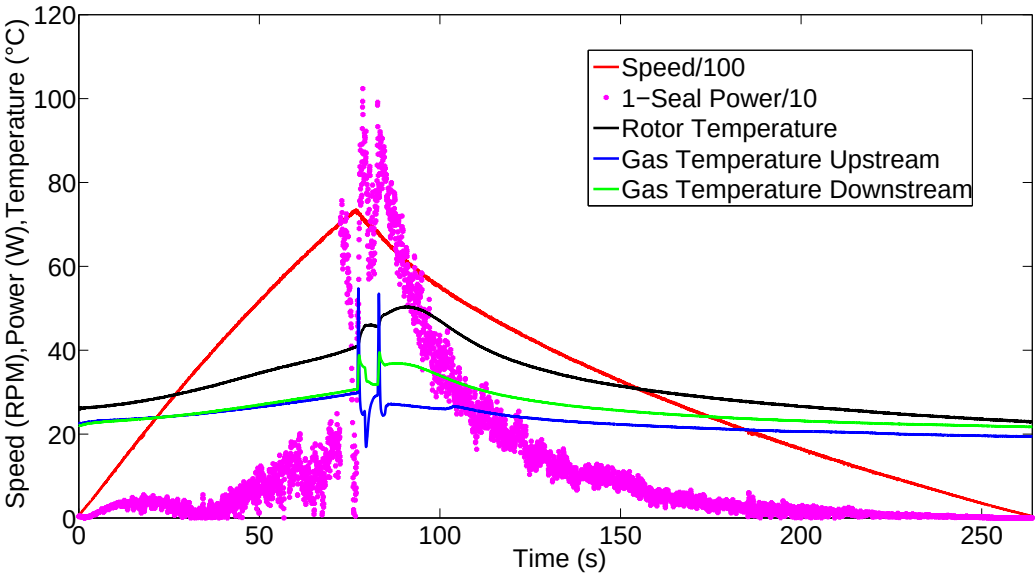


(b) Thermal summary plot

Figure 6.7: 1.5 bar SDP test

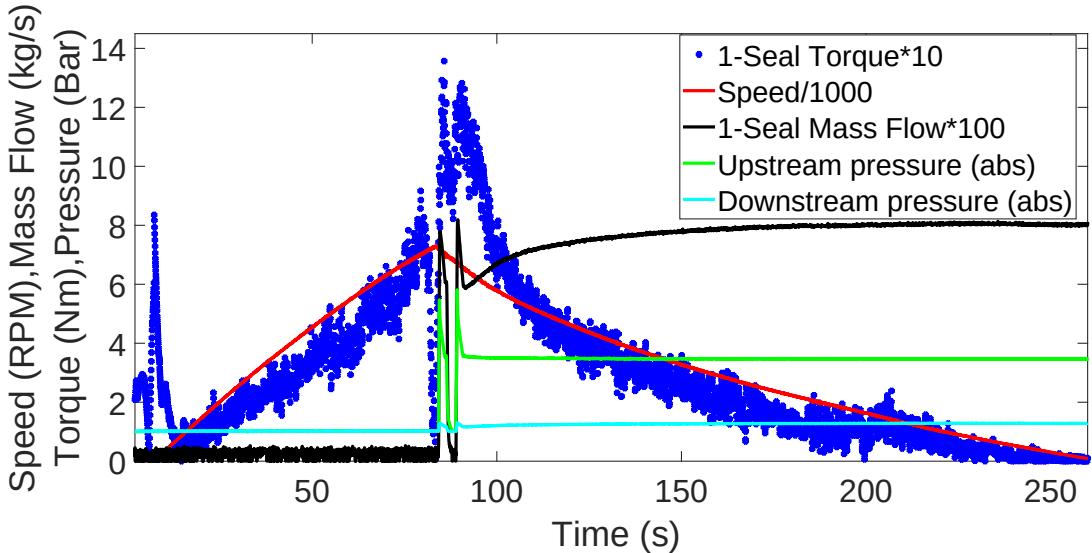


(a) Mechanical summary plot

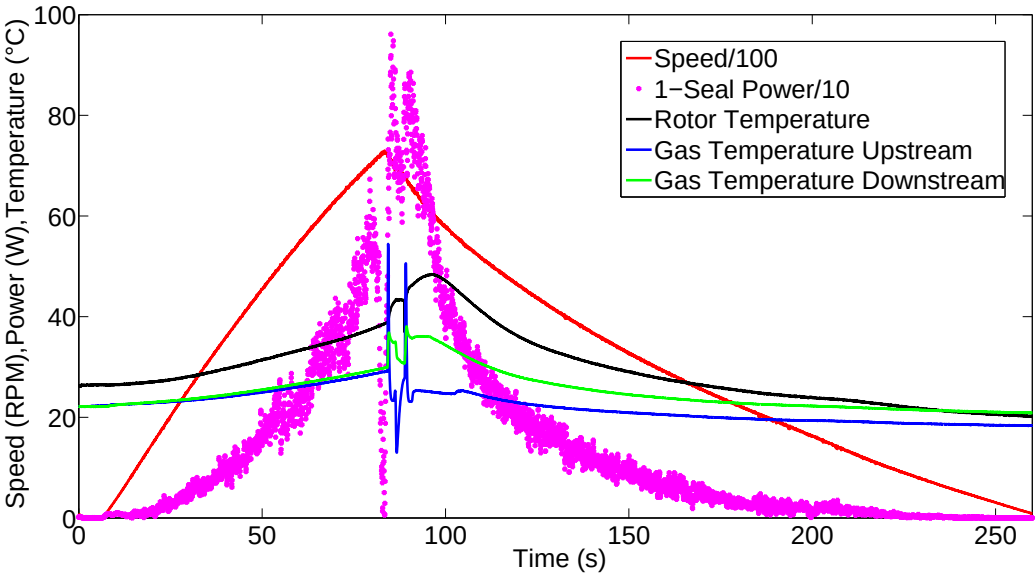


(b) Thermal summary plot

Figure 6.8: 2 bar SDP test

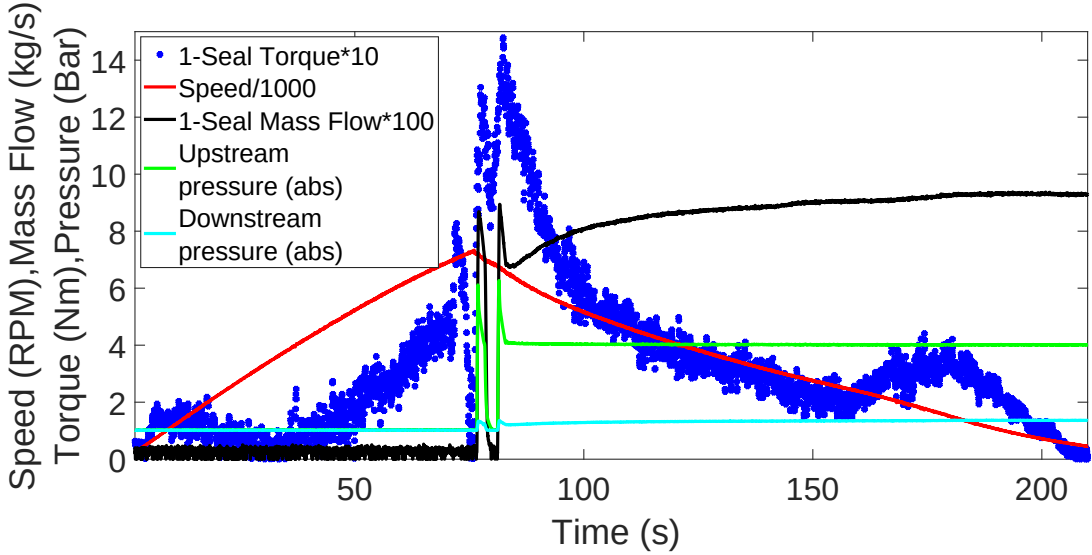


(a) Mechanical summary plot

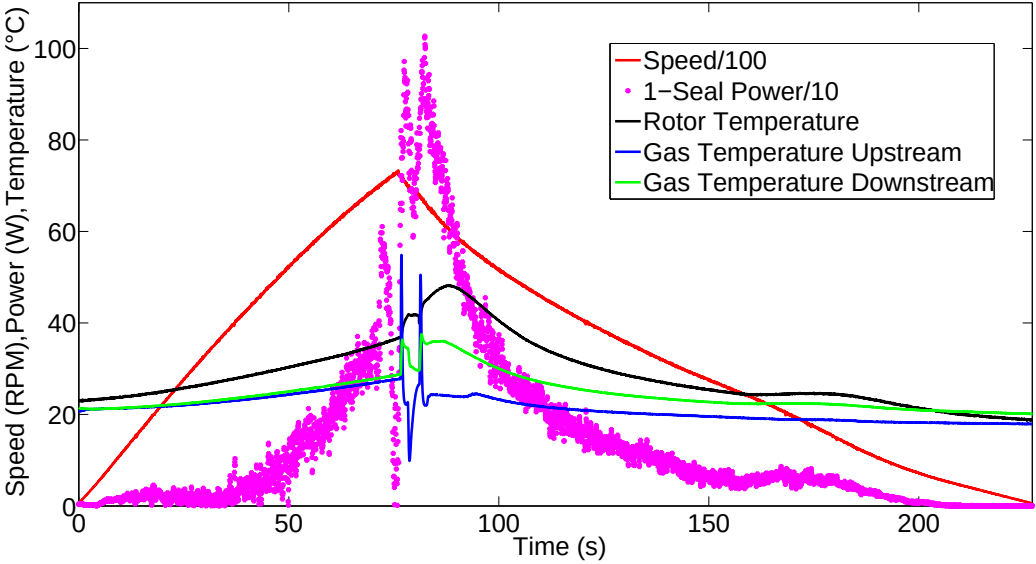


(b) Thermal summary plot

Figure 6.9: 2.5 bar SDP test

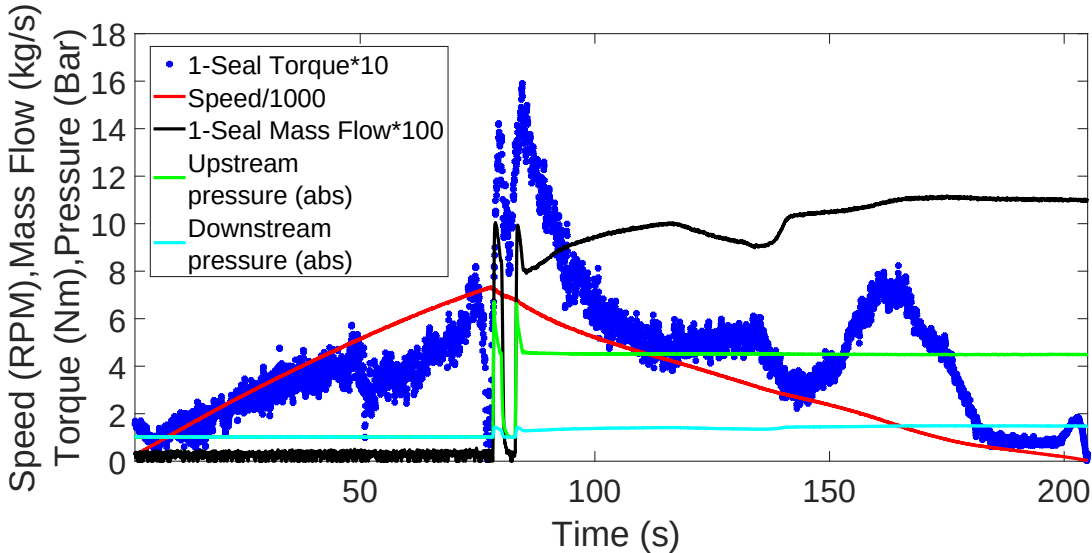


(a) Mechanical summary plot

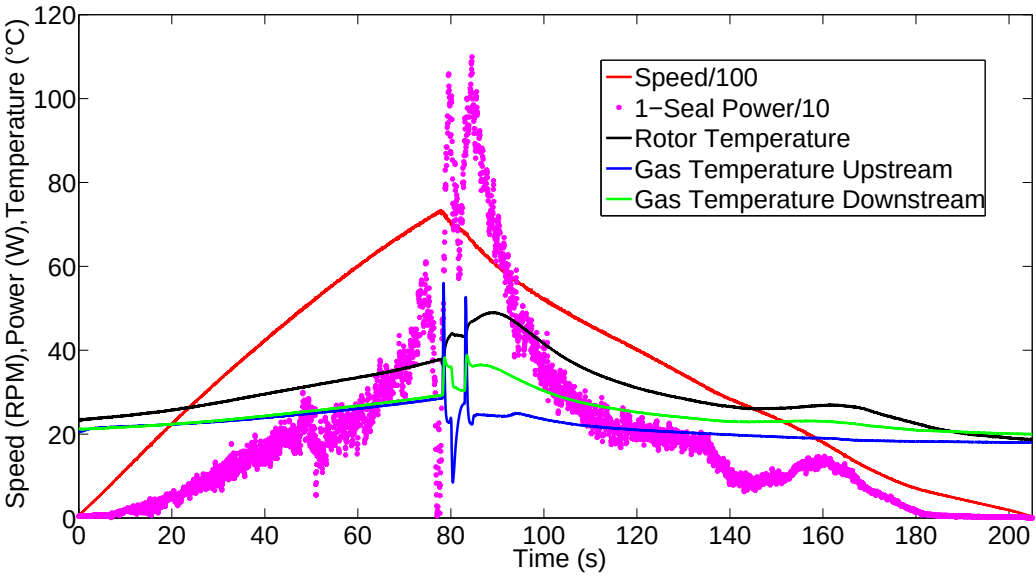


(b) Thermal summary plot

Figure 6.10: 3 bar SDP test



(a) Mechanical summary plot



(b) Thermal summary plot

Figure 6.11: 3.5 bar SDP test

pressure setting that extends during the entire run-down. It can be seen that the seal differential pressure is essentially steady during the tests, with the spikes associated with each pulse being caused by compression of the initial test gas within working section, and the time constant of the pressure regulators used.

The deceleration of the rotor results in lower centrifugal stresses and a reduction in radius of the rotor during the test. The blow-down forces exerted by the leakage flow on the bristles make the seal partially compliant with these changes in radius and contribute to maintaining contact with the rotor. The presence of blow-down is definitively demonstrated by the higher level of frictional torque that exists when air is supplied to the seals. However, the shrinkage of the rotor and friction between the seal and the backing plate generally causes the frictional force to diminish with decreasing speed (increasing time). The mass flow rate is also seen to rise gradually from its initial steady value in most of the tests, with renewed partial reductions in the flow rate in the cases of the 1, 1.5 and 3.5 bar SDP tests during the rundown process. For example, in Fig 6.11, the mass flow rate initially rises as the speed decreases suggesting the bristles are largely unable to move in response to the rotor shrinkage. However, between 120 and 140 seconds there appears to be renewed blowdown of the seal which decreases the mass flow rate. The corresponding frictional torque and power fall with decreasing rotor speed, but where there is renewed blowdown there is an attenuation in the rate of decrease of torque and thus power. In these regions the blowdown force has overcome the frictional force, albeit temporarily, leading to reduced leakage but higher torque and heat input to the rotor, and one might conclude seal wear. Subsequently the bristles appear hang-up again against the backing ring, leading to reduced torque and power dissipation, more in line with the behaviour at lower differential pressures of 2 and 2.5 bar SDP. Overall, the notable increase in mass flow rate suggests that for this test seal, the pay-off between bristle movement being restricted by friction against the seal backing ring (bristle lock-up, see Chapter 2), and blowdown on the bristles caused by the differential pressure across them is strongly weighted towards the former.

Whether or not blowdown occurs or a gap opens underneath the bristle tips, a

consequence of the rotor contraction is an increase in seal porosity with more cross-sectional area becoming available for the passage of the flow.

Cases of blowdown followed by backing ring frictional behaviour, are exemplified by the plots from the cases at 2 and 2.5 bar SDP. Here continuous reduction in frictional torque with rotor speed and continuously increasing leakage mass flow rate are observed. In all cases, the thermal plots show that the temperature of the rotor and gas downstream of the seal are sensitive to the increments in frictional torque and power as expected.

The observations derived from the experiments presented in Chapter 5 can also explain the behaviour of the leakage mass flow rate found during the ESTF tests. In Chapter 5, it was shown how increments in contact temperature result in lower leakage rates. As the seal filaments and thus the air passing between them are heated-up, the sealing performance improves. In the ESTF tests, the maximum contact temperatures occur at the beginning of the run-down, which coincide with the minimum mass flow leakage for a given SDP. As the rotor is decelerated and the speed diminishes, so does the frictional power. The resulting contact temperature drops and the mass flow leaking through the seals increases.

As the SDP increases, the level of torque developed during the test rises due to a stronger blow-down force on the bristles. However, the resulting increment in power does not involve higher rotor temperatures, which are very similar in the six cases. Although higher SDPs cause larger level of friction and heat generation, the implicit increment in leakage mass flow rate improves the cooling of the seal and rotor. In the thermal plots, it can be appreciated that the evolution of the rotor temperature trace is essentially governed by the trend followed by frictional power. The gas temperature upstream of the seals is very similar to the downstream one during the run-up, where a steady rise can be observed. When the SDP is zero, the static mass of air confined in the ESTF cavity is heated at the rotor walls. This heating is similar upstream and downstream of the seals and therefore all gas thermocouples provide similar readings. However, when high pressure is delivered to the seals in the run-down, it can be seen the gap that opens between the upstream and downstream gas temperature traces, which corresponds to

the heating that the flow experiences as it leaks through the filament pack.

6.3 Thermal Modelling of the ESTF

The brush/leaf seal friction heat model developed throughout this thesis is employed to predict the thermal behaviour of the ESTF during the set of experiments presented in Section 6.2. A transient FE model of the ESTF rotor is built to solve the temperature field. This model applies as boundary conditions some of the measurements from the experimental data. The brush seal thermal correlation derived in Chapter 5 is used to infer the amount of heat transferred to the rotor when frictional heating occurs. A CFD model of the flow inside the rig cavity provides additional boundary conditions for the convective heat transfer on the rotor surfaces. Predictions can then be compared with the temperature measurements taken by the infrared sensors during the experimental tests. In the next two Sections 6.3.1 and 6.3.2, the FE and CFD models are described in detail, while the comparison between experimental and numerical results is presented in Section 6.4.

6.3.1 FE model of the ESTF rotor

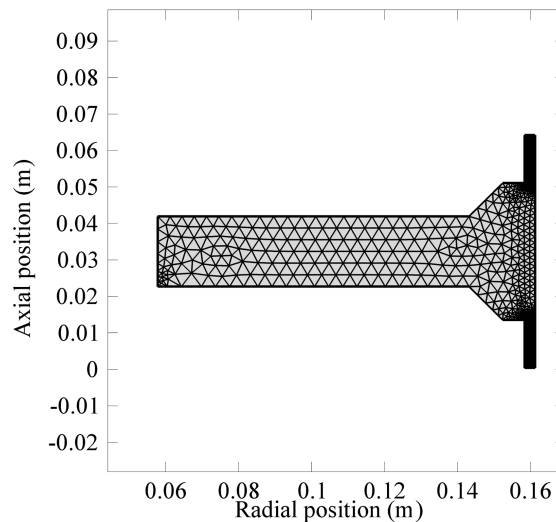


Figure 6.12: Computational mesh for the FE model of the ESTF rotor

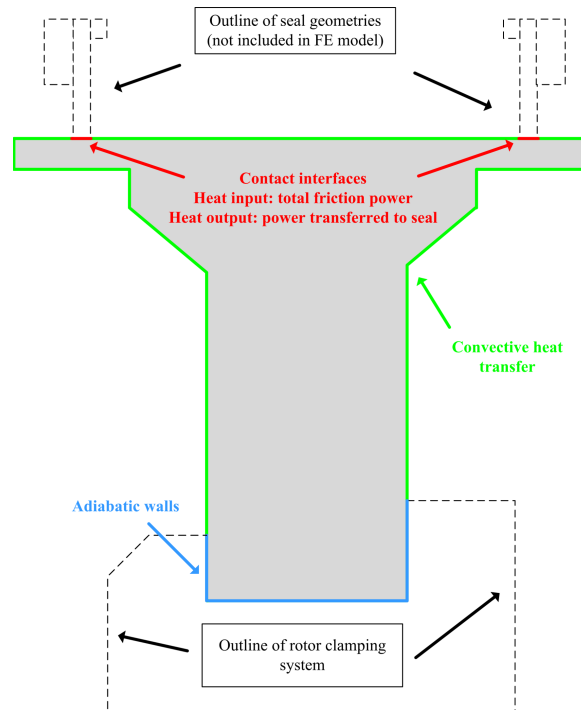
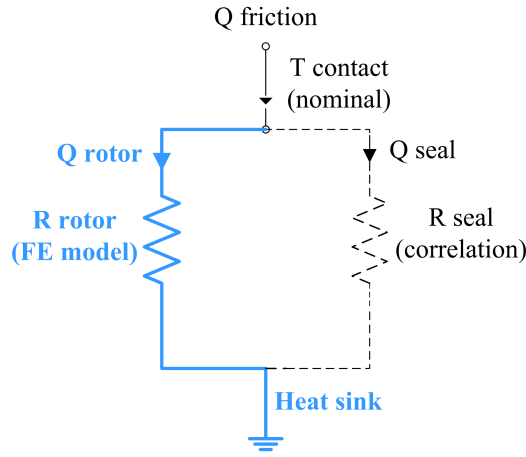


Figure 6.13: Schematic of the different boundary conditions imposed on the model (not to scale)

A 2D axisymmetric FE model of the ESTF rotor is built using *COMSOL Multiphysics*. This is used to run transient heat transfer studies that aim to reproduce the temperature field developed within the rotor during the experimental tests. The model simulates the entire test, including the run-up and run-down portions. The geometry and dimensions of the rotor were shown in Fig 6.3. Fig 6.12 shows the computational mesh employed for these simulations. The higher concentration of nodes in the regions close to the seal contact interfaces, where the temperature gradients are expected to be larger, are clearly visible. Further refinement of this mesh did not result in changes in the temperature field within the rotor, and therefore, the solution can be considered mesh independent. Four different boundary conditions are imposed at the various rotor surfaces: heat input, heat output, convective heat transfer and adiabatic walls. These are shown schematically in Fig 6.13 and are discussed in detail below.

In Section 6.2, it was shown the temperature rise experienced by the rotor during the experimental tests due to the heat generated by the friction with the seals. A certain portion of this heat is conducted into the rotor, and then, progressively convected to the

airflow at its surfaces. The remaining heat is absorbed by the seal filaments, which act as very efficient heat exchangers transferring it to the leakage flow. As discussed in Chapter 3, frictional heating can be studied using an electrical analogy. Fig 6.14 illustrates the principles behind the methodology adopted. The FE model developed represents the thermal resistance of the rotor (R_{rotor} , in blue in Fig 6.14). Total frictional power (Q_f) is calculated from the experimental data, as explained in Section 6.1.2. However, the FE model needs as boundary condition only the fraction of the total frictional power transferred to the rotor (Q_{rotor}).



$$Q_f = Q_{seal} + Q_{rotor} \quad (6.5)$$

$$\Delta T_c = Q_{seal} R_{seal} = Q_{rotor} R_{rotor} \quad (6.6)$$

Figure 6.14: Schematic of the modelling strategy

As explained in Chapter 5, the heat absorbed by a brush seal at a given nominal contact temperature is dependent upon the leakage flow. This relationship can be expressed by an empirical correlation (Eq 5.3), and experimental data used to derive the value of the thermal constant of the seal (K_{seal}):

$$Q_{seal} = K_{seal} \dot{m}^{1/2} (T_c - T_{gas}) \quad (6.7)$$

The domain of the FE model only considers the rotor geometry; the seals themselves are not included. However, their presence can be accounted using Eq 6.7 in the following manner. At the seal-rotor contact interfaces (red segments in Fig 6.13), a heat input equal to the frictional power derived from the experimental data is imposed as a boundary condition. This corresponds to the total frictional power generated (Q_f). On the same surface, a heat output representing the fraction absorbed by the seal (Q_{seal}) is subtracted. Eq 6.7 is used for the calculation of this heat removal. Consequently, the net heat flux that enters the domain of the model is only the fraction of the total frictional heat that is transferred to the rotor (Q_{rotor}).

From the modelling point of view, the proportionality between the temperature rise at the contact surface ($T_c - T_{gas}$) and the heat dissipated into the seal (Q_{seal}) in Eq 6.7 is analogous to the traditional convective heat transfer coefficient, as defined in Newton's law of cooling. Therefore, Eq 6.7 can be recast to give:

$$Q_{seal} = htc_{eq} A_c (T_c - T_{gas}) \quad (6.8)$$

$$htc_{eq} = \frac{K_{seal} \sqrt{\dot{m}}}{A_c} \quad (6.9)$$

where A_c is the nominal contact area between seal and rotor. The resulting equivalent convective heat transfer coefficient (htc_{eq}) is implemented in the *COMSOL* model as a convection boundary condition over the two contact areas (red lines in Fig 6.13). The gas temperature measured by the thermocouples installed in the cavity upstream of the seals is used by *COMSOL* as T_{gas} in Eq 6.8. Similarly, the value of the mass flow rate obtained experimentally is considered for $\dot{m}$. The nominal contact temperature, T_c in Eq 6.8, is calculated by *COMSOL* when solving the temperature field of the rotor. For the seal thermal constant (K_{seal}), the value derived in Chapter 5 for a brush seal is employed here (see Eq 5.14). Although the brush seal geometry installed in the ESTF is not exactly the same design as the one previously tested in the SSTTF, both have bristles of the same material (Haynes 25) and identical geometrical characteristics (lay angle,

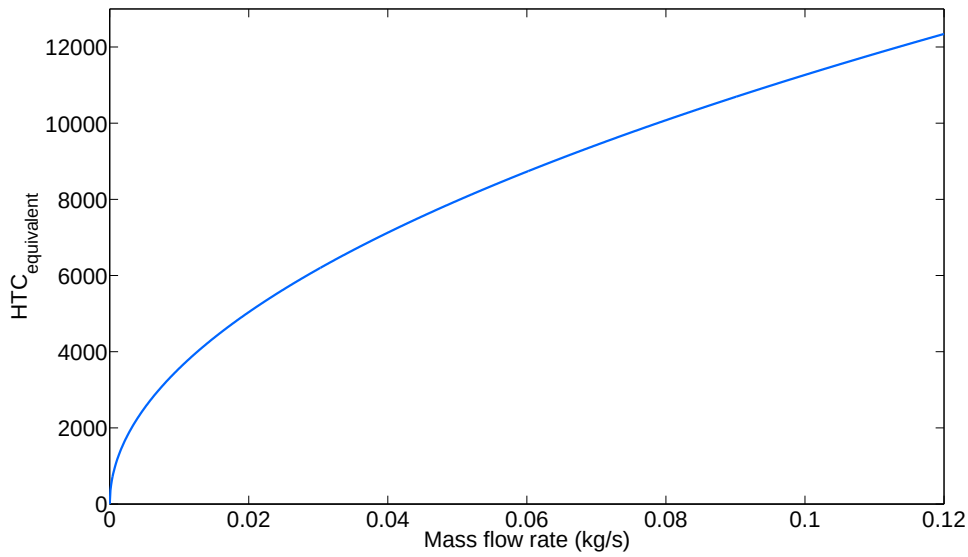


Figure 6.15: Variation of htc_{eq} with seal leakage mas flow rate, see Eq 6.9

number of row of bristles, inner diameter, bristle height). They only differ in bristle diameter (0.1 mm and 0.071 mm), and hence, axial width of the seal (1.5 mm and 1.3 mm). According to Owen et al. (2003), this change in axial width results in a variation in the heat absorbed by the seal of less than 7.4%, which is within the experimental uncertainty. To give an idea of the magnitude of the htc_{eq} obtained, Eq 6.9 is plotted in Fig 6.15 over the range of seal mass flow rates measured in the experiments.

During the run-up of a test, the leakage mass flow rate is zero and, according to Eq 6.7, the heat conducted into the seals would be zero. Although it is sensible that the capacity of the seal to absorb heat drops with mass flow rate, the result predicted at the limiting case of zero mass flow rate is not realistic. At this operating condition, the heat transfer mechanism within the bristle pack is different from the one assumed in the derivation of Eq 6.7. As a first approximation, it can be considered that when no air is leaking through the seal, heat transfer is mainly governed by solid conduction along the filaments, and that convection to the air confined between bristles is negligible. Since the geometry of the seals is not included in the FE model, their effect on the rotor can be modelled as follows. Assuming 1D conduction, the heat flux conducted into the seal

can be estimated using:

$$q_{\dot{m}=0} = \frac{k}{l} (T_1 - T_2) \quad (6.10)$$

where T_1 and T_2 are the temperatures at the tips and roots of the filaments respectively, and k and l the thermal conductivity and length of the bristles. The temperature at the root of the filaments is not measured during the experiments. However, given the low values of frictional power generated during the run-up and the large mass of the structure where the root of the filaments are welded, the temperature rise at this location is assumed to be negligible, and thus, T_2 can be assumed equal to ambient temperature (T_0). Eq 6.10 can then be compared with Newton's law of cooling (Eq 6.11) to obtain a dummy convection heat transfer coefficient ($htc_{\dot{m}=0}$):

$$q_{\dot{m}=0} = htc_{\dot{m}=0}(T_1 - T_0) \quad (6.11)$$

$$htc_{\dot{m}=0} = \frac{k}{l} \quad (6.12)$$

The filament length and thermal conductivity of the seal tested are substituted in Eq 6.12 to find a value for $htc_{\dot{m}=0}$ of $\sim 900 \text{ W/m}^2\text{K}$, which is similar to those showed in Fig 6.15 at leakage mass flow rates close to zero. This constant value is applied as boundary condition in the FE model (over the red lines in Fig 6.13) during the run-up portion of the tests. Although the modelling of the seal in the absence of leakage flow may seem poor, this operating condition is not of industrial or research interest. Furthermore, in Section 6.4.1, it will be shown that the temperature history developed in the run-up has little influence on the temperatures measured at the rotor thin-walls during the run-down. Note that thermal modelling of no-flow conditions is a valuable means to model rotor heating during windmilling or hot-shutdown events. In order to avoid the complexities involved in the modelling of the seals at zero leakage flow, it is recommended for future experiments that the test procedure is modified, so that a small leakage flow is imposed during the run-up portion of the tests.

On the remaining surfaces of the rotor, two different types of boundary conditions

are imposed. At the inner bore of the rotor, where it is clamped to the shaft, adiabatic walls are employed (blue lines in Fig 6.13). The heat that enters the rotor at the contact interfaces is convected to the surrounding flow well before reaching the inner radius of the rotor so the choice is somewhat arbitrary, and an isothermal condition might equally well be employed. Over this region, the temperature rise during the entire test is negligible. This will be confirmed later by the simulations performed with the model. On the surfaces of the rotor that are exposed to the flow, convective heat transfer is used as boundary condition (green lines in Fig 6.13). A steady-state CFD model of the flow inside the ESTF cavity (described in Section 6.3.2) provides the h_{tc} distribution on these surfaces at the different operating conditions. The h_{tc} at the rotor walls is dependent on the SDP, the rotational speed, and the temperatures within the cavity (the latter is a negligible driver in the test rig where the air has moderate changes in driving temperature compared to the walls of the disk). To generate the h_{tc} correlation, a limited number of cases at different rotational speeds are run with the CFD model for both run-up and run-down events. These correspond to the speeds of {500, 1500, 2500, 3500, 4500, 5500, 6500 RPM}. The FE model then switches between these over the course of the test according to the rotational speed of the rotor. An example of this methodology is shown in Fig 6.16, where the mass flow rate and rotational speed from the case at 2.5 bar seal differential pressure are plotted with time (this is considerably easier to implement than a functional form in COMSOL with limited diminution in accuracy). In this figure, different colours are used to indicate the time intervals over which each h_{tc} distribution is used. For instance, during the time interval comprised between 5000 RPM and 4000 RPM in the run-down (in cyan), the h_{tc} distribution obtained from the CFD simulation at 4500 RPM is applied. During the time interval comprised between the two vertical dotted lines in Fig 6.16, when a quick pulse in upstream pressure occurs, it is assumed that there is no mass flow through the system. Therefore, the h_{tc} distribution obtained from the CFD simulation at zero SDP and 6500 RPM is imposed during this interval. The temperature traces measured by the gas thermocouples located upstream and downstream of the seals are employed in the FE

model as the reference gas temperature required for the htc upstream and downstream of the seals respectively.

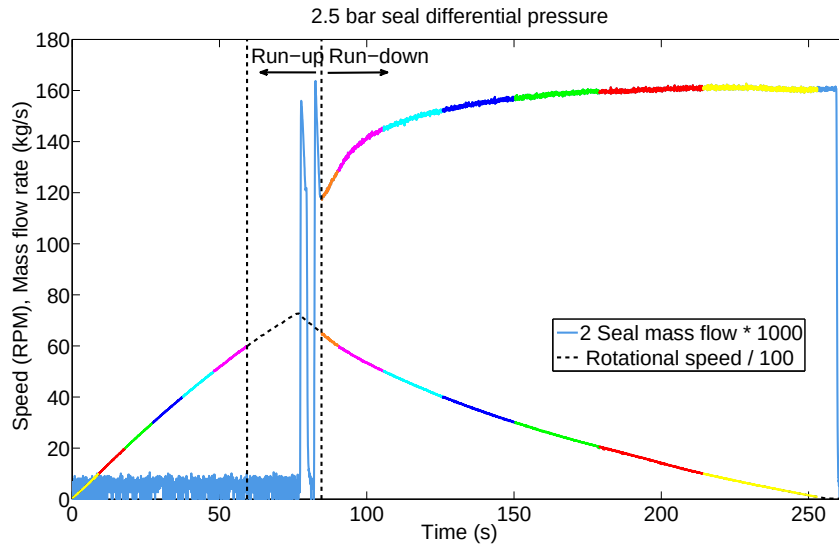


Figure 6.16: Example showing how the htc distribution applied to the FE model of the rotor changes with time according to the rotational speed. Each colour represents the time interval when a certain htc distribution is employed

6.3.2 CFD investigation of convective heat transfer on ESTF rotor surfaces

6.3.2.1 CFD model

To estimate the convective htc distribution on the ESTF rotor surfaces, a CFD study was conducted. The outcome of these simulations is a set of htc maps that are subsequently applied as boundary conditions to the FE model described in Section 6.3.1. A schematic of the CFD domain is shown in Fig 6.17. This represents the cavity inside the ESTF pressure vessel contained between the rotor and the casing. Except for the inlets and outlets of the pressure vessel (see Fig 6.2), the geometry of the rig is essentially axisymmetric. Hence, a simplified 2D axisymmetric model with swirl is used to reduce the time required for both the rig modelling and the execution of the simulations.

The filament packs of the two seal geometries installed in the ESTF are modelled in *ANSYS Fluent* as porous media. In Chapter 4, it was shown that the predictions in

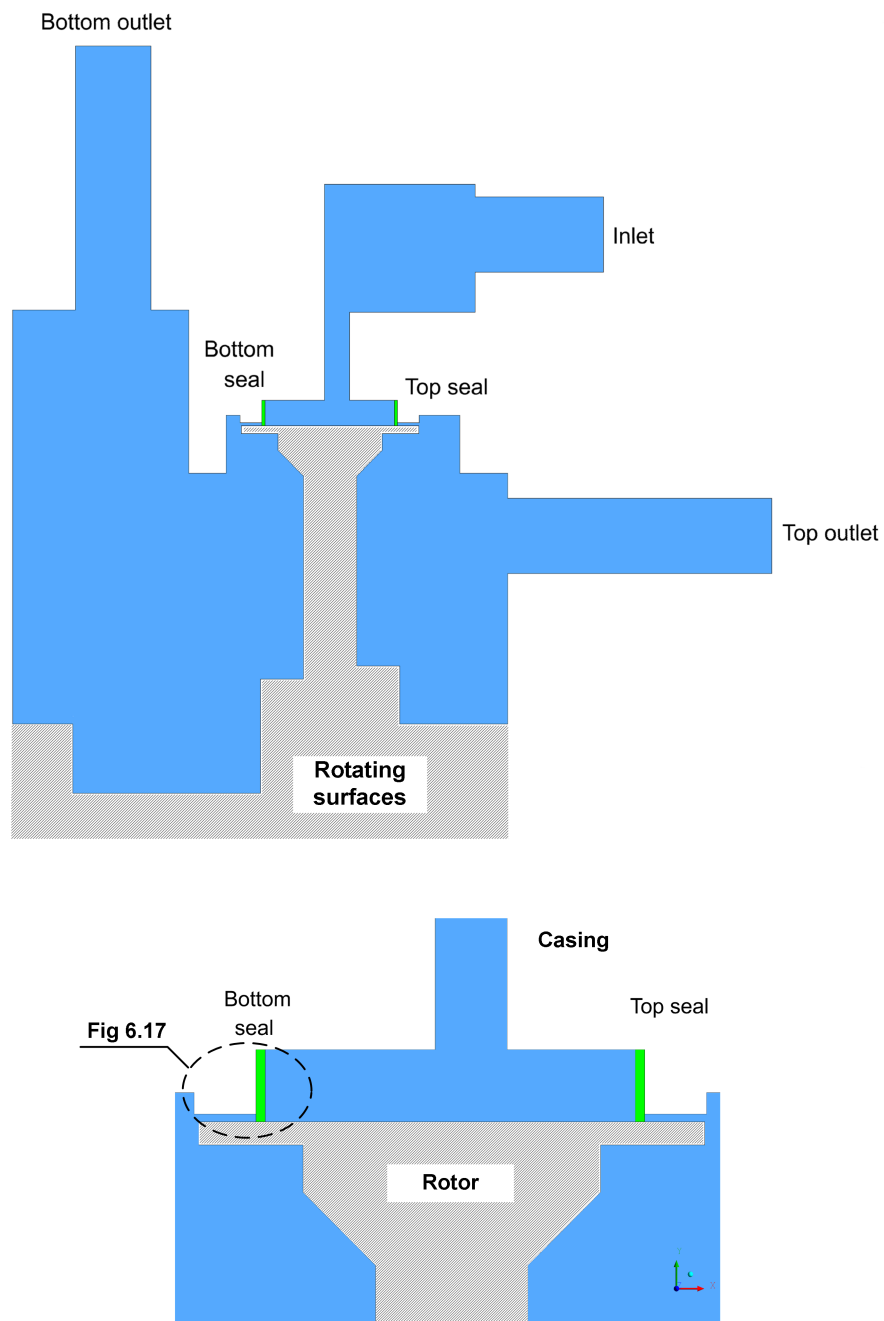


Figure 6.17: CFD domain of the flow inside ESTF cavity. The green regions correspond to the bristle packs of the seals, where the porous medium model is applied. Refer to Fig 6.3 for dimensions

h_{tc} using this approach are comparable with experimentally derived h_{tcs} . The pressure drop exerted by the seal on the flow and its effect on the velocity field are reasonably well captured by the porous medium model. Therefore, it can be used to obtain an estimation of the h_{tc} levels in the vicinity of a filament seal. In the areas of the domain corresponding to the bristle packs (green zones in Fig 6.17), a separate fluid zone is defined where resistance source terms are added to the flow momentum equations. A detailed description of the implementation and configuration of the porous medium model was provided in Section 4.2.3. The momentum source terms require as input the values of the viscous resistance coefficients in the axial, radial and circumferential (swirl) directions. These are determined by calibration, so that the leakage mass flow rate obtained with the numerical model at a given SDP matches the value found in the corresponding experimental test. Since the pressure gradients in the circumferential direction are negligible in comparison with the axial ones, the brush seal largely removes the swirl of the flow induced by the shaft rotation, as explained in Section 2.1. To account for this effect in the model, a resistance coefficient in the circumferential direction ten orders of magnitude higher than the axial one is imposed.

During the run-down portion of a test, the leakage mass flow rate slightly increases as the test progresses and the rotor decelerates. This effect is thought to be mainly caused by the shrinkage of the rotor at low RPM, as discussed in Section 6.2. The experimental leakage mass flow rate is thus averaged during the run-down portion, and the coefficients of the porous media are adjusted to match this mean value (see Fig 6.18). Table 6.4 summarizes the result of the calibration process of the CFD model.

The computational domain is discretized using a multi-block structured orthogonal mesh built in *ANSYS ICEM*. The mesh is especially refined at the regions close to the rotor walls. The vicinities of both seals and the transition regions between the non-porous and porous zones are also especially treated. *ANSYS Fluent* pressure-based solver with 2nd order discretization is employed to solve the steady RANS equations. The solver is configured following the same settings as those employed in the CFD model described in Section 4.2.3. The realizable k - ϵ turbulence model with enhanced

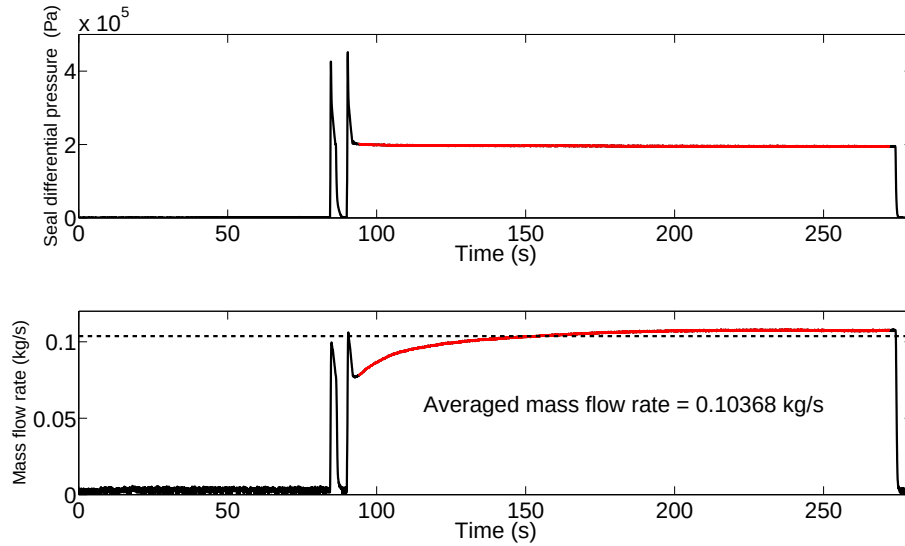


Figure 6.18: Example of the calculation of the average experimental leakage mass flow rate for the calibration of the CFD porous medium model

SDP (bar)	Experimental AVG 2-seal mass flow rate(kg/s)	CFD calibrated 2-seal mass flow rate (kg/s)	Difference (%)	Axial coefficient of porous medium (m^{-2})
1	0.0384	0.0388	-1.04	1.7 e12
1.5	0.0608	0.0605	0.49	1.8 e12
2	0.1037	0.1006	2.99	1.5 e12
2.5	0.1553	0.1548	0.32	1.3 e12
3	0.1773	0.1781	-0.45	1.35 e12
3.5	0.2040	0.2073	1.62	1.45 e12

Table 6.4: Calibration of the porous medium model with test data

wall treatment is used to solve the boundary layer and account for convective heat transfer. The distance to the wall of the first row of nodes at the rotor surfaces (where $htcs$ are calculated) is chosen to achieve y^+ values <1 for the entire range of SDPs tested. In addition, the mesh at the casing walls close to the seal is refined to result in y^+ values close to 1. To confirm the independence of the solution with mesh resolution, this was progressively increased until no further refinement resulted in changes in rotor wall temperature. Fig 6.19 illustrates the process of mesh refinement. The figure plots the temperature on the rotor surface downstream of the bottom seal for different meshes. The number of nodes in both the axial and radial directions was progressively increased, keeping the y^+ values fixed. When a variation of less than 1% in wall temperature was obtained, the lower mesh size was considered for the subsequent simulations (final mesh, in red in the figure). The final mesh contains a total of $\sim 323,300$ elements. Detail of the mesh in the vicinity of the bottom seal is shown in Fig 6.20.

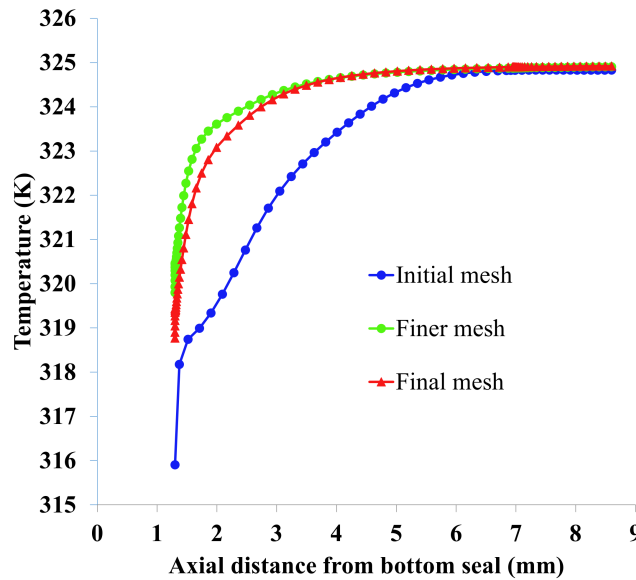


Figure 6.19: Effect of mesh refinement on wall temperature

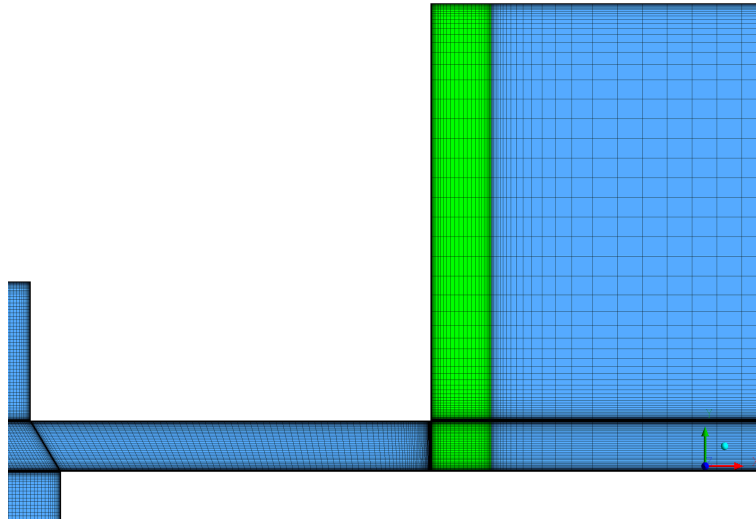


Figure 6.20: Detail of the computational mesh in the vicinity of the bottom seal (in green)

6.3.2.2 Boundary conditions and h_{tc} calculation method

Two distinct operation modes of the rig exist. First, during the run-up portion of a test, the gauge pressure upstream of the seals is zero, and therefore there is no airflow through the system. Second, when the air motor is shut down and the rotor starts to decelerate, high pressure air is supplied to the seals. The boundary conditions of the CFD model vary according to these operation modes.

Pressure boundary conditions are imposed at the inlet and outlets of the model (see Fig 6.17). Ambient pressure is always applied at the two outlets. For the cases that simulate the operation of the rig without airflow (run-up portion of the test), ambient pressure is also imposed at the inlet. This results in an inlet mass flow rate that is virtually zero.

The CFD model is used to estimate the h_{tc} distribution on the rotor surfaces at the different operating conditions that occur during an experimental test. Although the rotational speed changes continuously during a test, only a certain number of steady-state cases are simulated. Moving wall boundary conditions with rotational speed are imposed on the rotor and shaft assembly walls. For both the run-up and run-down, 7 cases are run with rotational speeds of 500, 1500, 2500, 3500, 4500, 5500, 6500 RPM. As explained

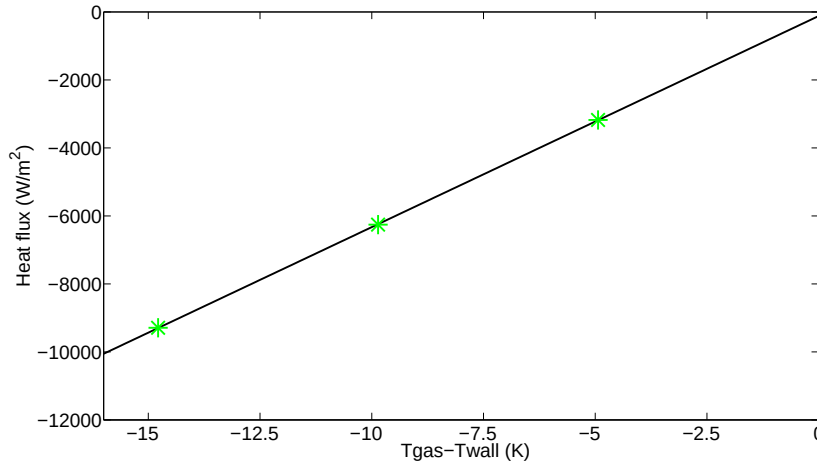


Figure 6.21: Example of h_{tc} calculation. Convective heat flux is plotted versus the difference between the rotor wall temperature and a local gas temperature. Linear regression is then performed, with the slope of the resulting fit corresponding to the value of the h_{tc} at the location considered

in Section 6.3.1, the FE model of the rotor, which is fully transient, extrapolates the resulting h_{tc} maps to intermediate speeds. The geometry of the rotor was not adjusted to take account for the rotor growth, as the purpose of these tests is to understand the bulk flow in the test rig, and not the precise flow in the vicinity of the seals.

A similar methodology as the one described in Chapter 4 is employed here to calculate the h_{tc} s from the CFD data. Contrarily to the SSTTF, in the ESTF the incoming air is at ambient temperature (no heater mesh) and cools the rotor, which is heated up by the seal frictional power. The temperature of the flow at the inlet of the CFD domain is set equal to 293 K. A constant temperature is imposed as boundary condition at all rotor walls. Three wall temperature cases are simulated: 295, 303 and 308 K for each SDP and rotational speed. Heat flux is extracted from these three simulations at every node on the walls and plotted against the difference between the rotor wall temperature and a local gas temperature. Linear regression is performed, with the slope of the best linear fit being the h_{tc} at that point (see example in Fig 6.21).

In the simulations at non-zero seal differential pressure and mass flow (run-down), the temperature of the fluid increases as it progresses through the domain and absorbs energy from the rotor surfaces. The remaining walls of the domain (shaft assembly and

casing) are adiabatic in these simulations. Fig 6.22a shows a typical flow temperature field from a case with 2.5 bar SDP.

When the gauge pressure at inlet is zero and no airflow occurs (run-up simulations), there is a mass of air confined inside the pressure vessel. Since the CFD simulations are steady-state, a temperature equal to ambient is imposed at the walls of the casing in order to generate a temperature gradient inside the flow field that allows the calculation of the $htcs$. An example of the typical temperature field resulting from a simulation at inlet pressure of 0 barg is shown in Fig 6.22b.

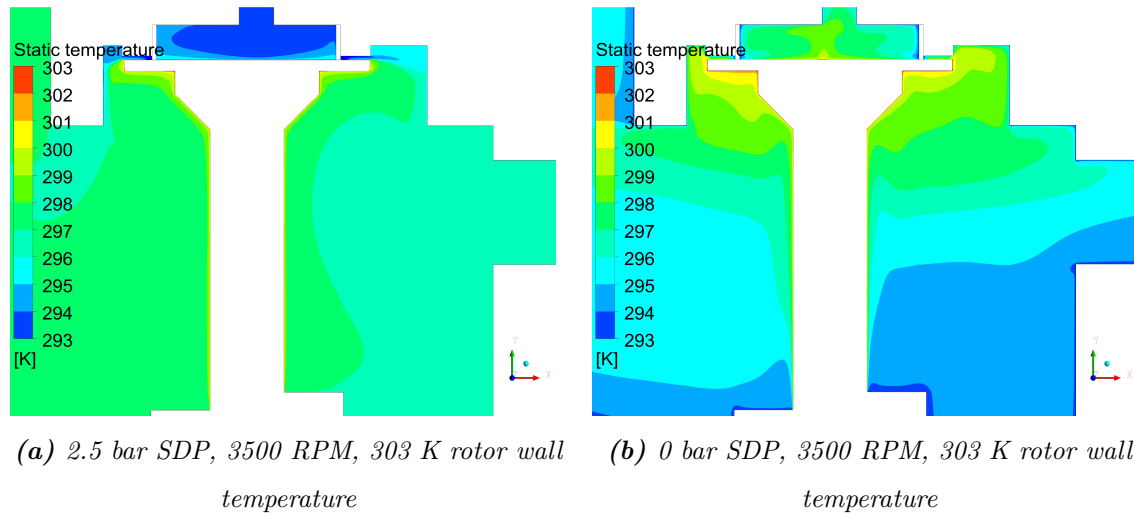


Figure 6.22: Typical temperature fields from run-down and run-up flow simulations

From Fig 6.22, it can be appreciated the strong gradients that exist in the temperature field of the flow in both types of simulations. Therefore, a local gas temperature must be included in the regression used to derive the htc (see Fig 6.21). Fig 6.23 shows the locations where the local gas temperature is considered for the calculation of the htc at the different surfaces of the rotor. For the region upstream of the seals and the vertical walls of the rotor, the gas temperature is taken at 5 mm perpendicularly from the rotor wall. Immediately downstream of the seal, where the coverplate gap is 1 mm, the gas temperature at 0.5 mm from the surface is chosen. The temperature of the gas at Points 1 and 2 in the figure is employed for the calculation of the htc on the horizontal and vertical surfaces forming a 90° angle (region at the back of the thin-walls). Finally,

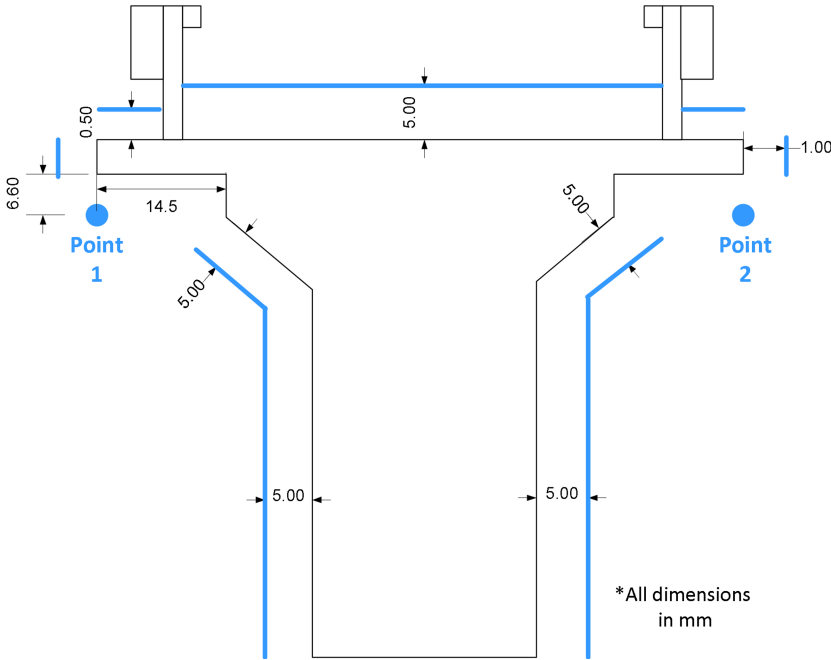


Figure 6.23: The blue lines and the two points represent the positions where the local gas temperature is considered for the calculation of the h_{tc} using linear regression (not to scale)

the h_{tc} along the thickness of the thin-walls is calculated using the gas temperature 1 mm perpendicularly from the wall.

To summarize all the cases run with the CFD model, Table 6.5 is included:

SDP (bar)	Rotational speed (RPM)						
	500	1500	2500	3500	4500	5500	6500
0	Rotor wall temperature 298, 303, 308 K						
1							
1.5							
2							
2.5							
3							
3.5							

Table 6.5: Summary of the cases run at different rotational speeds, SDPs, and rotor wall temperatures (147 simulations in total)

6.3.2.3 Results at zero SDP (run-up)

The *htc* distributions obtained for the cases at a pressure of 0 barg at the inlet and outlets of the domain are presented here. These cases represent the run-up portion of a experimental test, when the rotor is accelerated by the air motor and there is no mass flow leaking through the seals. The areas of the domain corresponding to the two seals still have the porous medium model activated in these simulations. This was found to improve the convergence of the solution. The resulting mass flow rate at the inlet of the domain was very low ($< 10^{-4}$ kg/s) in all the cases carried out at different rotor speeds. Therefore, heat transfer is mainly governed by the swirl induced into the flow by the motion of the rotor and the buoyancy forces generated by the density gradients.

Fig 6.24 shows a representative example of the velocity field found in these simulations. The axial and radial components of the flow velocity are nearly zero, and thus its magnitude practically corresponds to the value of the swirl component. The no-slip condition imposed at the rotating walls causes the flow to acquire locally the velocity of the rotor surface. The viscous forces and shear stresses generated inside the body of the flow transmit this swirl motion to adjacent layers of fluid. In Fig 6.24a it can be observed how the swirl of the flow increases with radius, due to the higher lineal speeds attained at the outer surfaces of the rotor. Fig 6.24b shows details of the velocity field in the vicinity of the bottom seal. Due to the resistance to any flow motion imposed by the porous model of the seal, gradients in swirl develop at its surroundings. On both sides of the seal, it can be seen that swirl tends to be low in the close vicinity of the bristle pack, while it rises with increasing distance from the seal. This increment in swirl occurs faster on the downstream side of the seal. On the upstream side, the highest levels of swirl occur in the central region of the cavity between both seals, where the flow is free to rotate with the disk. Here the seal acts as a flow straightener, imposing its direction at its close surroundings. As explained in Section 6.3.2.2, the inlet temperature of the flow is set equal to 293 K, while a higher temperature is imposed on the rotor walls to drive convective heat transfer. A typical flow temperature field from these simulations was shown in Fig 6.22b, indicating the large variation in static temperature and the

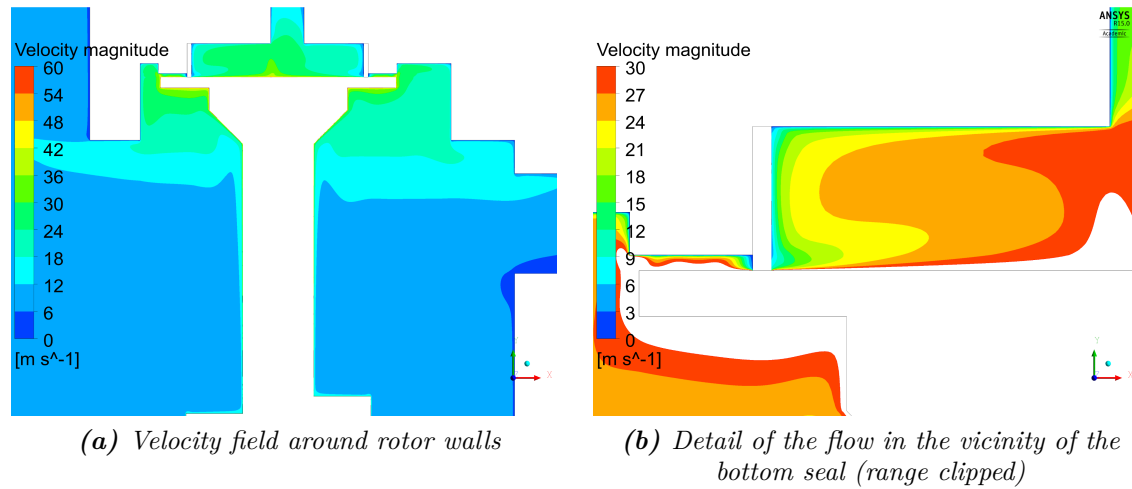


Figure 6.24: Typical velocity field, 0 bar SDP, rotational speed 3500 RPM, rotor wall temperature 303 K

need of considering the local gas temperature in the calculation of the htc .

The rotor surface is split into 6 different regions to ease the plotting of the htc distributions obtained. This is shown in Fig 6.25. The terms top and bottom are used to refer to the two different paths that the incoming air flow can follow. They correspond to the vertical orientation of the rig in reality (see Fig 6.2) and their locations in the CFD domain are also shown in the figure.

Figs 6.26-6.29 show the htc maps obtained at the different rotational speeds on the rotor surfaces depicted in Fig 6.25. Not surprisingly, the effect of increasing the rotor speed is always an enhancement in htc at all surfaces. In general, the same htc levels are found at positions that are at the same distance from the axial centre line of the rotor, the solution being quite symmetric. A comparison of Figs 6.26-6.29 indicates that the largest values of htc occur on the outer surfaces of the rotor (zones 1,2,3). The higher relative swirl that exists in these regions, where the radius presents its maximum value, results in greater heat transfer rates. The levels are especially high in the close vicinity to the seals as can be seen in Fig 6.26, which are acting essentially as solid walls in close proximity to the rotating surface in this run-up case. The flow in this region must be driven by the rotation of the disk and a pair of secondary vortices are set up in the inlet plenum resulting in flow across the axial length of the rotor (symmetric about the

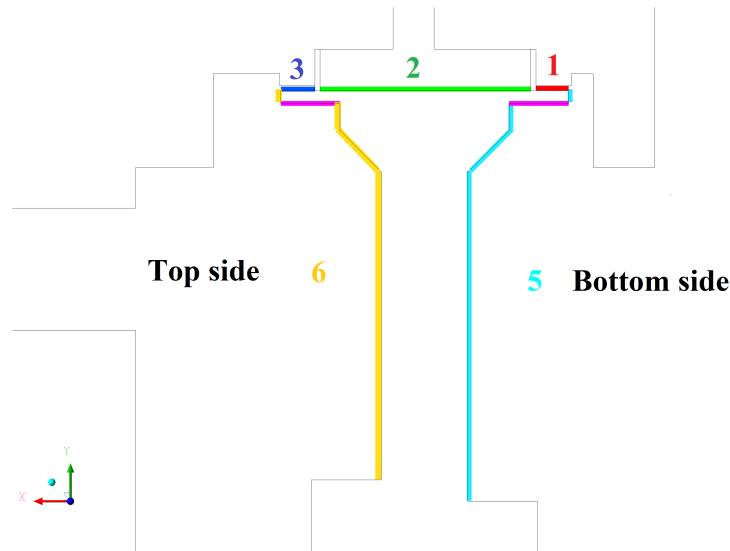


Figure 6.25: Subdivision of rotor surface into 6 zones

line of flow inlet), with flow turning onto the rotor surface close to the seals resulting in the higher $htcs$ observed. Of course in reality in spite of the high $htcs$, the low driving temperature differences mean that this is not necessarily associated with high heat flux, excepting in the case of a seal with a very high interface with the rotor. This effect sensibly scales with increasing rotational speed

Fig 6.27 shows the htc distribution on the back side of the rotor thin-walls (zone 4). At the root of the thin-walls, the flow is slow and probably separated, resulting in heat transfer close to zero. As we move closer to the tip of the thin-walls, the gradients in flow velocity increases and so does the htc . In Fig 6.28 and 6.29, the htc distributions on the vertical walls of the rotor are shown (zones 5 and 6). It can be seen the initial peak in htc that occurs in both cases at the lowest radius, which corresponds to recirculating flow impinging onto the face of the disk. The heat transfer up the face of the disk is associated with typical disk pumping behaviour causing a radial outward flow along the disk face. The increasing swirl velocity results in a slight net increase in htc with radial distance up to ~ 144 mm. Beyond this position (see Fig 6.24a) the impingement of the disk face flow and a series of flow separations cause some secondary peaks and rapid drops in the htc observed. On the outermost edge of the disk (> 160 mm) high htc is observed on both sides of the rotor.

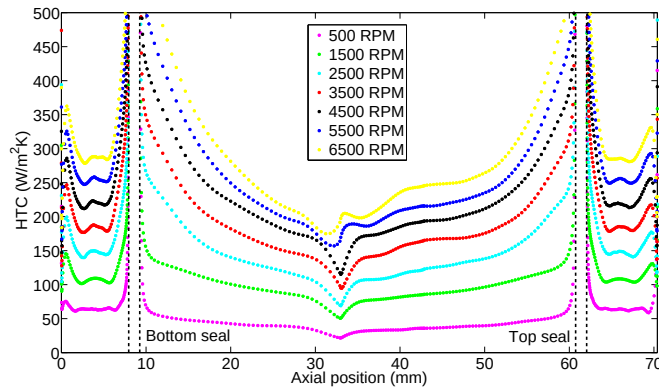


Figure 6.26: *Htc* distribution at the rotor outer surface (zones 1,2 and 3, see Fig 6.25), 0 bar SDP

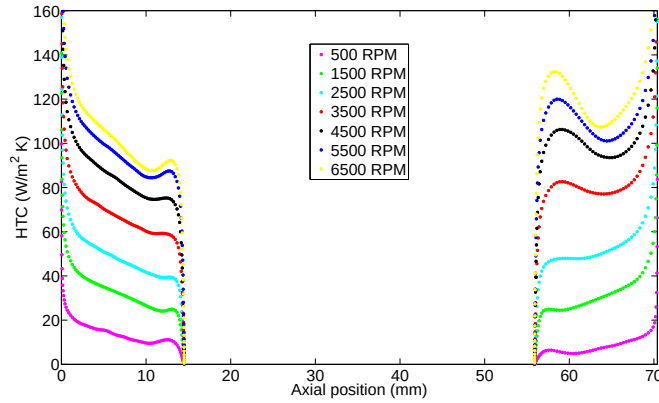


Figure 6.27: *Htc* distribution at the back of the rotor thin-walls (zone 4, see Fig 6.25), 0 bar SDP

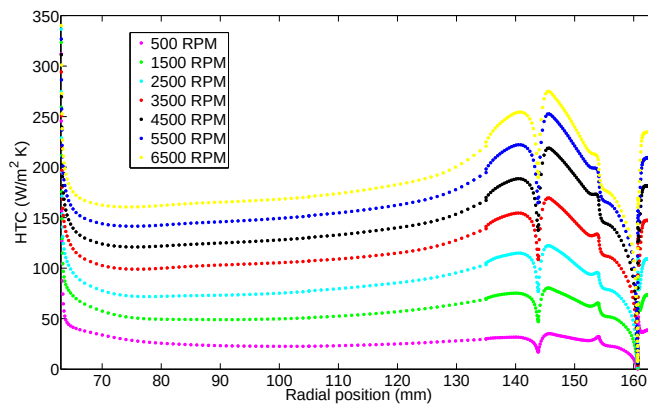


Figure 6.28: *Htc* distribution at the vertical walls of the rotor on the bottom side (zone 5, see Fig 6.25), 0 bar SDP

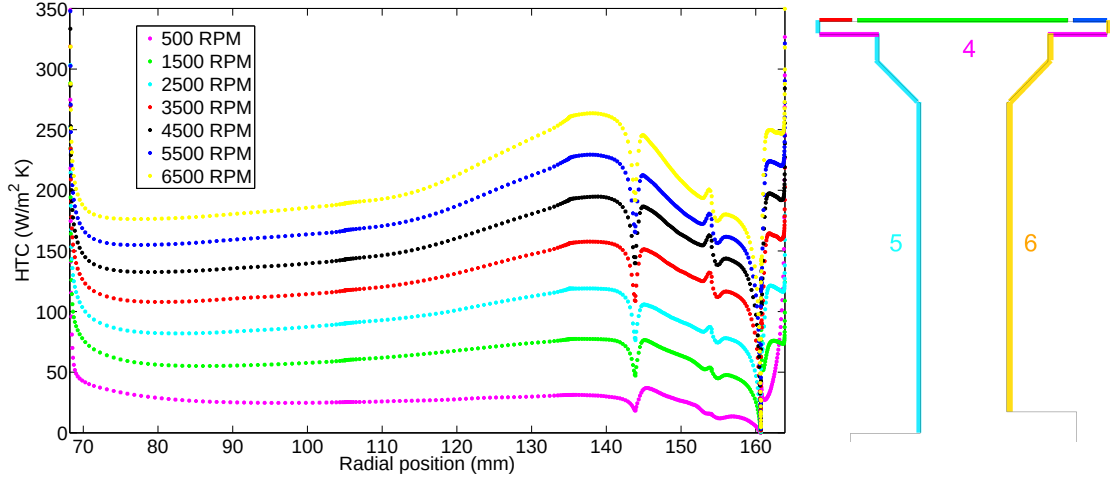


Figure 6.29: *Htc distribution at the vertical walls of the rotor on the top side (zone 6, see Fig 6.25), 0 bar SDP*

6.3.2.4 Influence of increasing SDP on rotor *htc* distribution

The characteristics of the flow within the rig cavity when pressure is supplied to the seals are discussed here. The resulting convective *htc* distribution on the rotor surface is shown at the different SDP tested in the experimental campaign. Each CFD simulation represents an instant of the run-down portion of a test, when a constant pressure is imposed upstream of the seals while the rotor decelerates. For each pressure setting, the resistance coefficients obtained by calibration of the model (see table 6.4) were applied to the porous zones. The simulations at rotational speed of 3500 RPM (mid-range) have been chosen as a representative example for displaying the *htc* distribution on the rotor surface. The main flow features found at 3500 RPM are also reproduced across all the range of speeds tested. In the following figures, the *htc* is plotted according to the division of the rotor surface into the different zones shown previously in Fig 6.25.

Fig 6.30 depicts the radial and axial components of the velocity vectors from the case at 2.5 bar SDP. The flow enters the cavity upstream of the two seals from a small duct perpendicular to the rotor axis. It impinges onto the rotor surface, resulting in a region of high *htcs* most notable at the highest inlet pressures tested, as shown in Fig 6.31. As the SDP and therefore the leakage mass flow rate decreases, the enhancement in *htc*

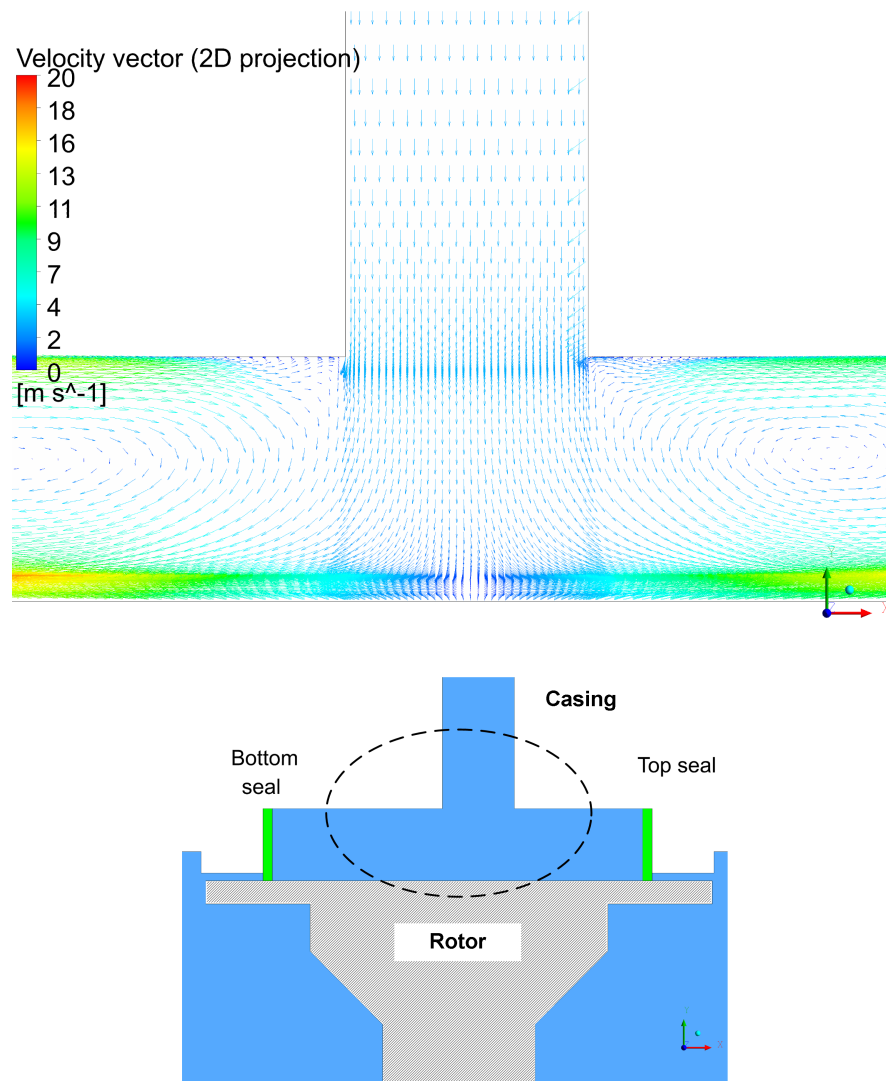


Figure 6.30: Velocity vectors (axial and radial components) in the cavity upstream of the seals (zone 2, see Fig 6.25). Example from simulation at 2.5 bar SDP and 3500 RPM

in the central region of zone 2 becomes less pronounced. Fig 6.30 clearly shows that recirculation vortices form upstream of each seal, and the highest velocity magnitude approaching the seal is not at the rotor surface. The growing boundary layer away from the point of flow impingement results in lower $htcs$ compared to those in the central region.

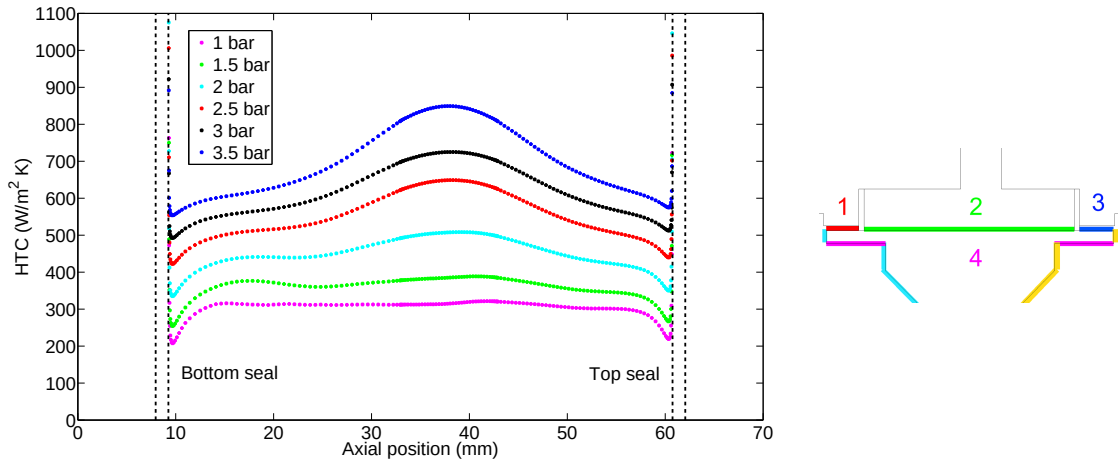


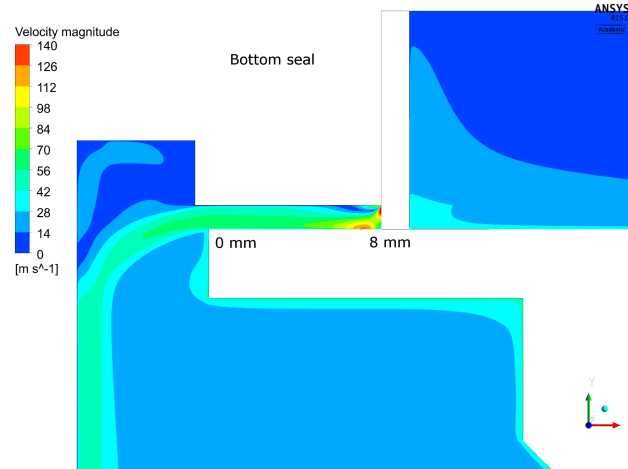
Figure 6.31: Variation of rotor htc distribution upstream of the seals with increasing SDP , 3500 RPM (zone 2, see Fig 6.25)

In contrast on the downstream side of the brush seal, Fig 6.32 shows a rapid increase in htc in the very close vicinity of the seals, caused by the local acceleration that the flow experiences to accommodate the change in porosity and cross-sectional area imposed by the presence of the seal backing ring. The experience acquired from the test campaign presented in Chapter 4 indicates that downstream of a filament seal, the flow structure consists in a high velocity jet that results in a region of high heat transfer. Fig 6.32a depicts the velocity field downstream of the bottom seal from the simulation at 2.5 bar SDP. It can be observed how the flow leaves the bristle pack at a very high speed, impinging radially on the rotor surface in the first millimetres after the seal exit. The resulting htc distribution is presented in Fig 6.32b, which is as expected given the velocity field. The htc exponentially decreases as the distance to the seal exit increases and the local flow velocity drops. Fig 6.32c shows the htc distribution downstream of the top seal; it can be seen that a similar behaviour is found for both seals. Increasing the SDP

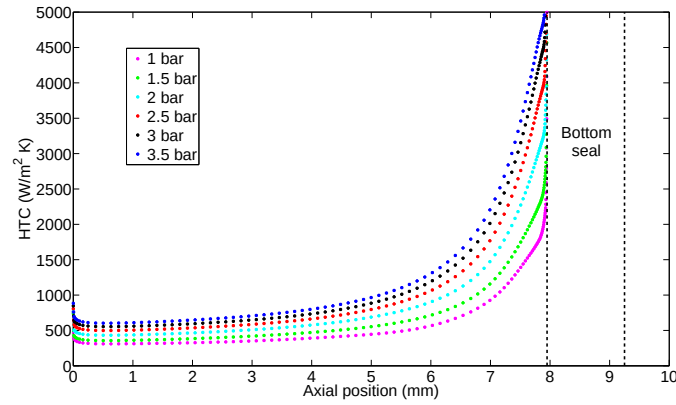
results in higher velocities of the jetting flow, and thus, larger values of rotor htc .

Fig 6.33 shows the htc distribution at zones 4, 5 and 6. Relatively far from the seal exits, the htc is determined by the manner in which the leakage flow expands and distributes into the downstream cavities. Most of the simulations at different SDPs result in an htc distribution that is very similar on the top and bottom sides, as the flow solution is quite symmetric. In zone 4, and at the highest radius in zones 5 and 6, increasing the SDP raises the htc level. However, as the airflow moves into the downstream cavities further away from the bristle pack exits, the effect on htc of increasing the upstream pressure is less marked. In these regions, heat transfer is mainly governed by the swirl generated by the disc rotation rather than the leakage flow. In Fig 6.33b and 6.33c, it can be observed that at the lower radius in zones 5 and 6, the htc is not very sensitive to the SDP. In addition, the values at these locations are very similar to those from the simulations at zero leakage flow presented in Fig 6.28 and 6.29 at the same rotational speed.

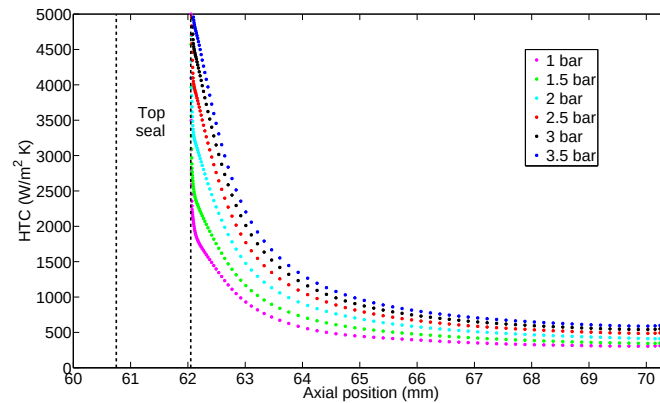
The simulation at a SDP of 3.5 bar is of special interest, since the behaviour of the flow found in this case differs from the previous arguments. In Fig 6.33b, it can be appreciated how the rotor htc in zone 5 greatly increases at 3.5 bar SDP, which indicates that the leakage flow is affecting the boundary layer at this rotor wall. Fig 6.34 shows the velocity and temperature fields from the solutions at 3 and 3.5 bar SDP. For SDPs lower than 3.5 bar, it was found that the jet that exits the bottom seal spreads into the downstream cavity and has little influence on the remaining rotor walls (see Fig 6.34b). However, the simulation at 3.5 bar predicts that the jet leaving the bottom seal hits the opposite casing wall and then impinges back onto the rotor walls (see Fig 6.34a), causing the great increment in htc shown in Fig 6.33b. This behaviour is, however, not found in zone 6 on the top side, where the rotor htc is less affected by the SDP.



(a) Velocity magnitude, bottom seal, SDP 2.5 bar, 3500 RPM, wall temperature 303 K

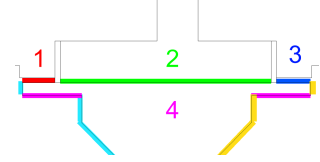
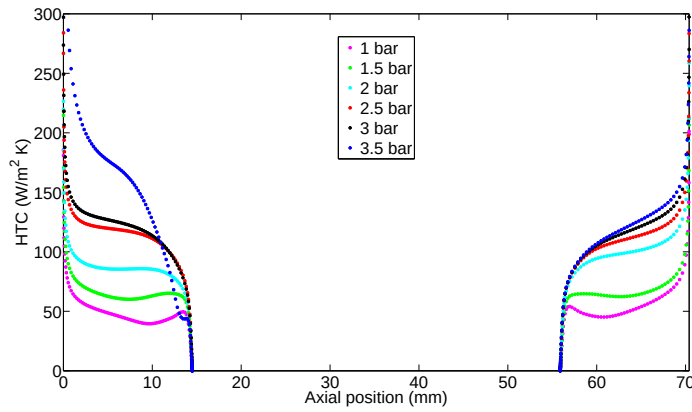


(b) Variation of rotor *htc* distribution downstream of the bottom seal with increasing SDP , 3500 RPM (zone 1, see Fig 6.25)

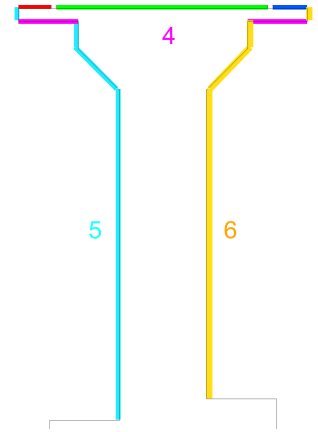
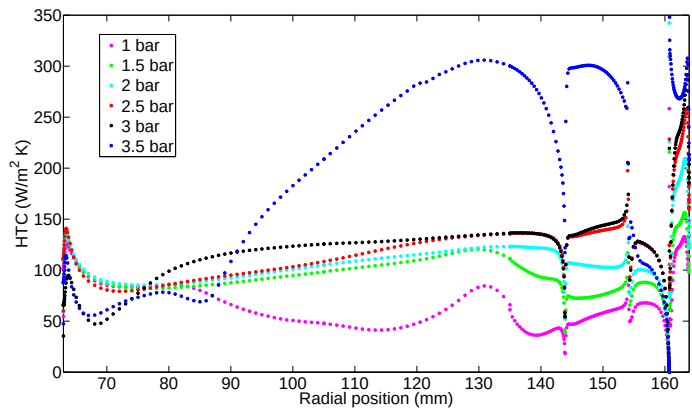


(c) Variation of rotor *htc* distribution downstream of the top seal with increasing SDP , 3500 RPM (zone 3, see Fig 6.25)

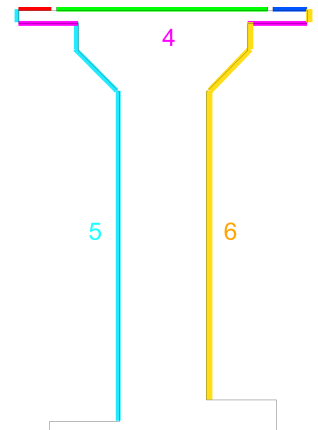
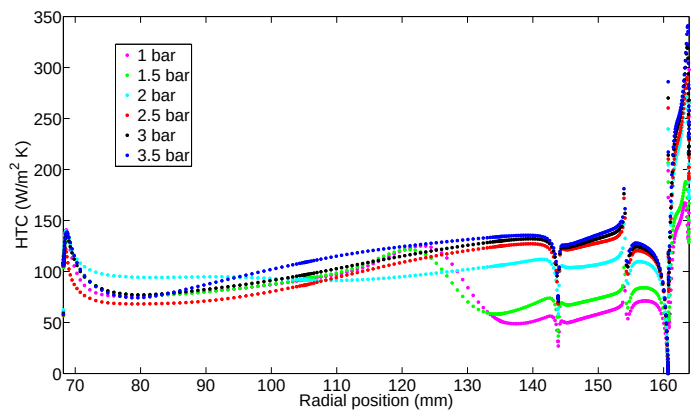
Figure 6.32: Flow characteristics downstream of the bristle pack



(a) Variation of rotor htc distribution at the back of thin-walls with increasing SDP , 3500 RPM (zone 4)



(b) Variation of rotor htc distribution at the vertical walls (bottom side) with increasing SDP , 3500 RPM (zone 5)



(c) Variation of rotor htc distribution at the vertical walls (top side) with increasing SDP , 3500 RPM (zone 6)

Figure 6.33: Rotor htc distributions at zones 4,5 and 6 (see Fig 6.25)

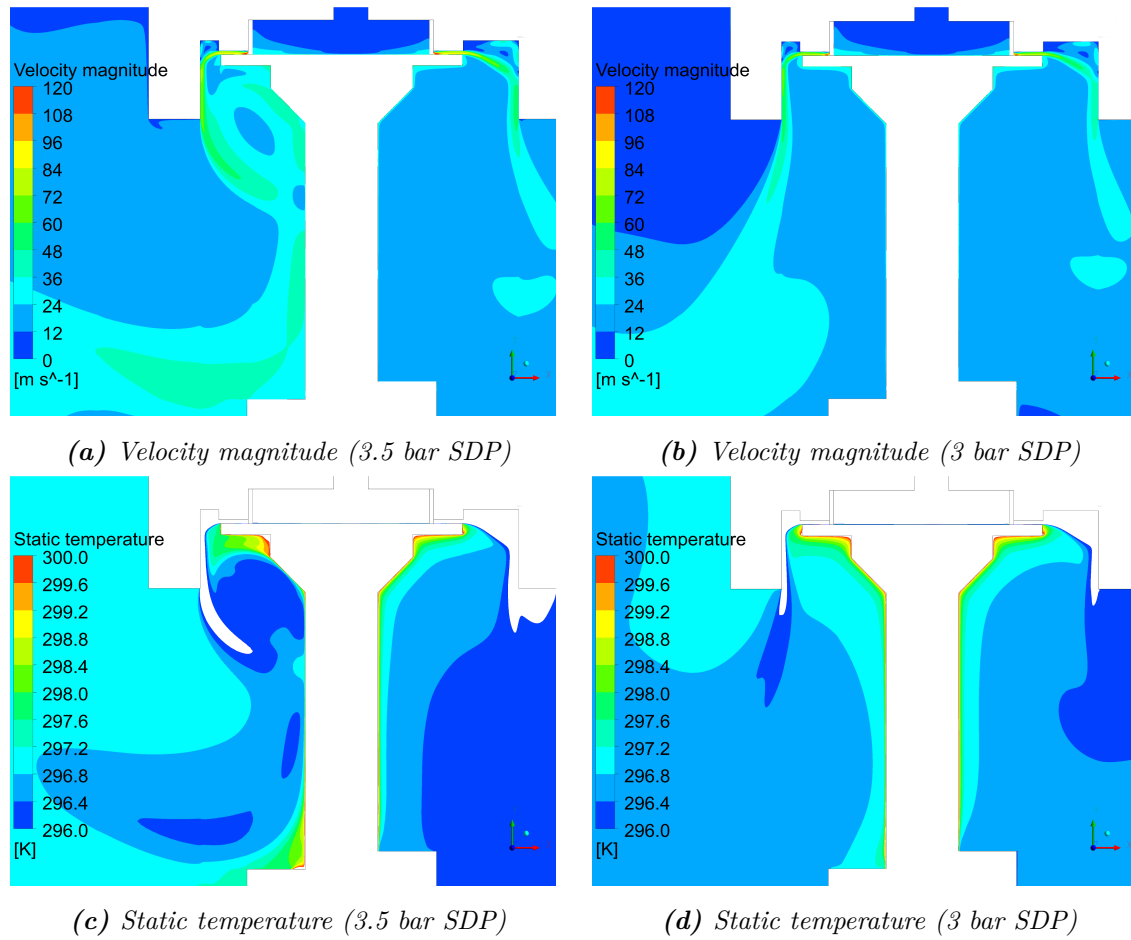
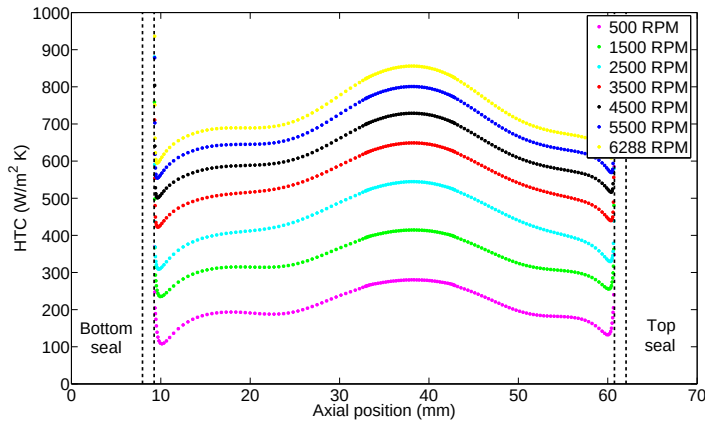


Figure 6.34: Velocity and temperature fields, 3500 RPM, rotor wall temperature 303 K

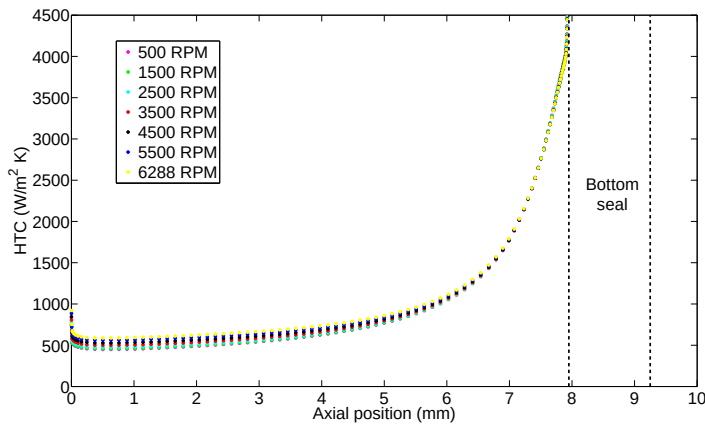
6.3.2.5 Influence of rotational speed and swirling flow on rotor *htc* distribution under seal pressurization

The effect on the *htc* of increasing the rotor speed keeping the SDP constant is shown in Fig 6.35 and 6.36. The *htc* distributions plotted were obtained from a representative example at 2.5 bar SDP. As for the simulations with zero gauge pressure upstream of the seals and no leakage flow, increasing rotational speed results in a rise in *htc* on all rotor surfaces. It is, however, noteworthy that on the downstream side of the seals (see Fig 6.35b and 6.35c), varying the rotational speed has a minimum influence on the *htc*. This is especially true in the first millimetres downstream of the seals, where the different *htc* distributions derived are nearly coincident. As the distance from the seal exit increases, the enhancement in *htc* caused by the increment in rotational speed starts

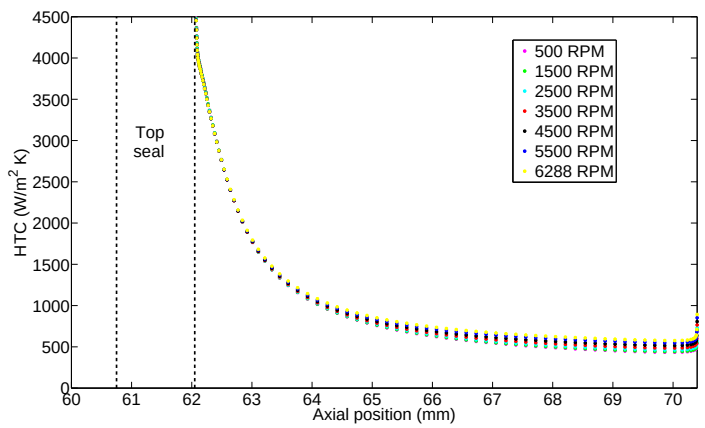
to be slightly noticeable. This indicates that downstream of the seal, heat transfer is mainly determined by the SDP, rather than the rotor speed. The seal accelerates and straightens the flow, so that downstream *htcs* are dominated by the jetting flow and swirl becomes less important.



(a) Variation of rotor htc distribution upstream of the seals with increasing rotor speed , 2.5 bar
SDP (zone 2)

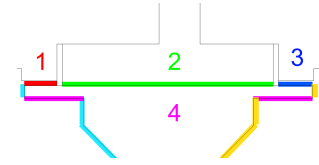
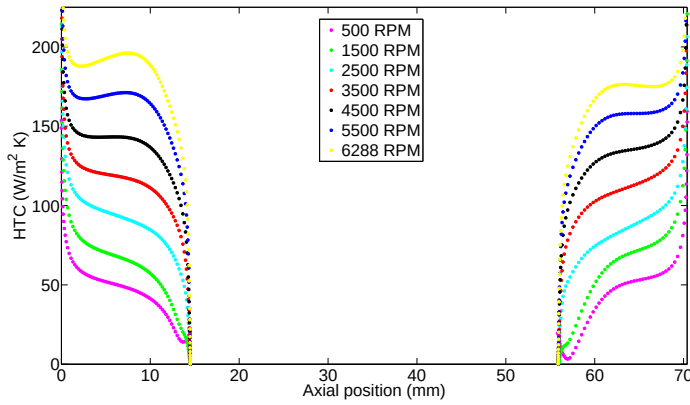


(b) Variation of rotor htc distribution downstream of the bottom seal with increasing rotor speed
, 2.5 bar SDP (zone 1)

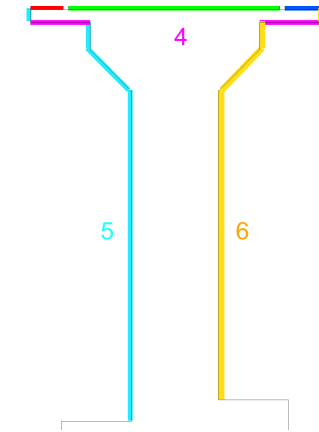
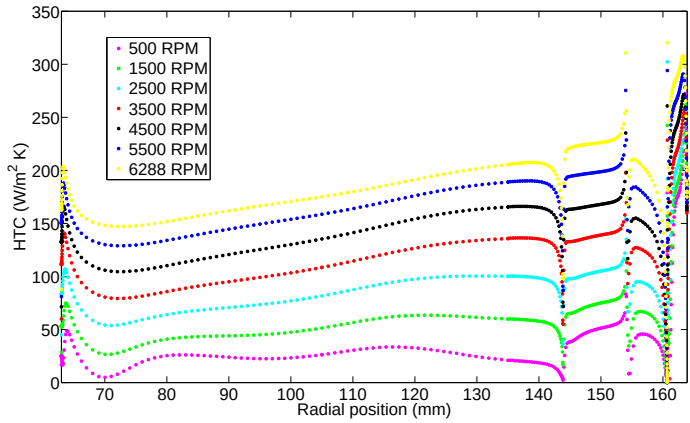


(c) Variation of rotor htc distribution downstream of the top seal with increasing rotor speed ,
2.5 bar SDP (zone 3)

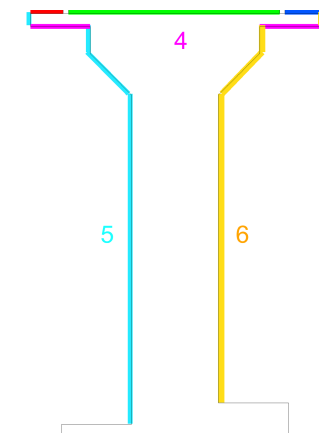
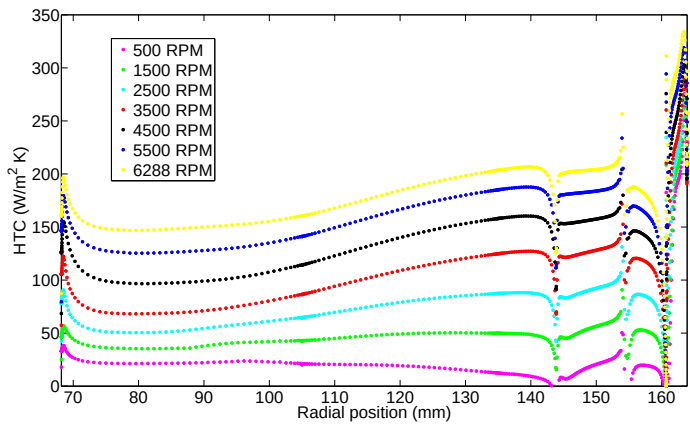
Figure 6.35: Rotor htc distributions at zones 1,2 and 3 (see Fig 6.25)



(a) Variation of rotor htc distribution at the back of the thin-walls with increasing rotor speed ,
2.5 bar SDP (zone 4)



(b) Variation of rotor htc distribution at the vertical walls (bottom side) with increasing rotor speed , 2.5 bar SDP (zone 5)



(c) Variation of rotor htc distribution at the vertical walls (top side) with increasing rotor speed , 2.5 bar SDP (zone 6)

Figure 6.36: Rotor htc distributions at zones 4,5 and 6 (see Fig 6.25)

6.4 ESTF Thermal Model Results and Comparison with Experimental Data

The outcome of simulating the ESTF tests described in Section 6.2 with the thermal model explained in Section 6.3 is presented here. First, some important features of the temperature field within the rotor are highlighted, which are key to understand the thermal behaviour of the system. Second, a comparison is made between the predictions of the model and the experimental data.

6.4.1 Key characteristics of the ESTF rotor temperature field

The results from a typical thermal FEA simulation highlight important features of the temperature field within the ESTF rotor. The boundary conditions of this simulation correspond to the 2 bar seal differential pressure (SDP) test. During the run-up, the seal was modelled as explained in Section 6.3.1. The predicted rotor temperature field at the time instant just at the beginning of the run-down is shown in Fig 6.37a for this case.

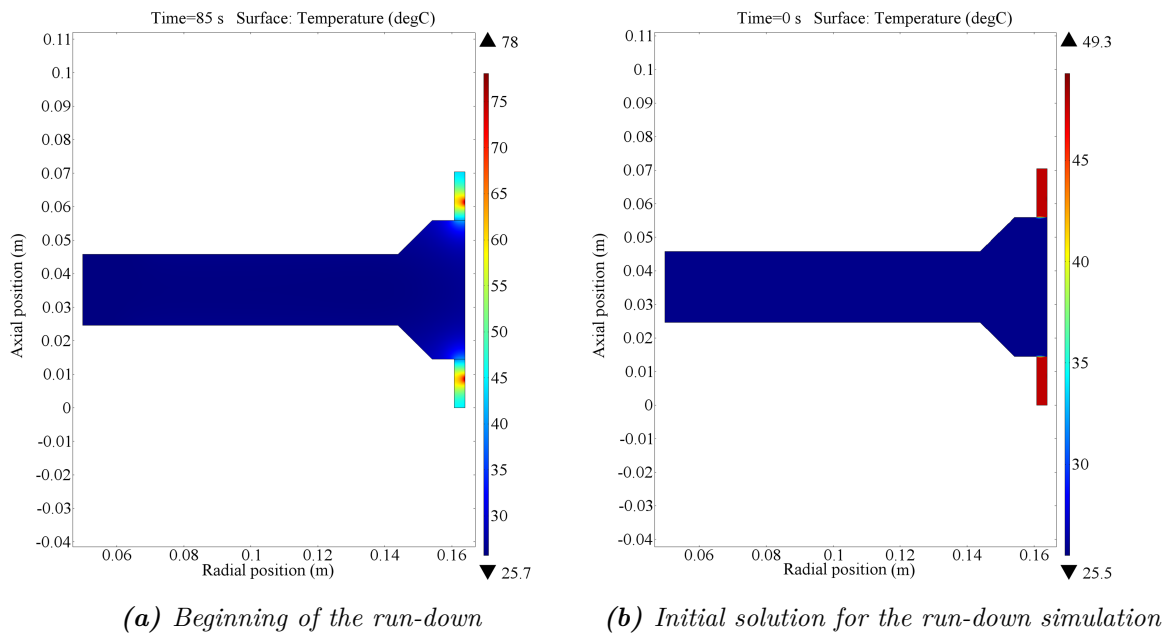


Figure 6.37: Numerical rotor temperature fields, 2 bar SDP test

The only source of heat input to the rotor is the heat generated by friction at the seal-rotor contact interfaces. A certain portion of the total frictional power enters the rotor at the thin-wall sections, and then, this heat flux is conducted further into the core of the rotor. As shown in Section 6.2, during the run-up, the rotor is pre-heated due to the initial build interference with the seals. Even in the absence of seal pressurization and blow-down forces, the bristles contact the rotor and frictional heating still occurs. It can be observed in Fig 6.37a that the heating of the rotor is practically limited to the thin-wall sections, while the central part of the rotor remains at the initial ambient temperature.

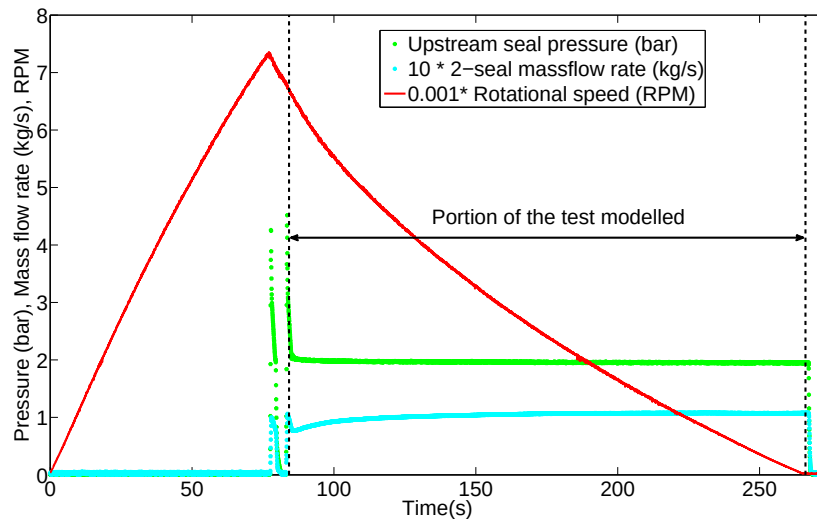


Figure 6.38: Portion of the test modelled in the run-down simulation, 2 bar SDP case

For the purpose of operational brush seal model validation, the only portion of the test that is of interest is the run-down regime, when air is leaking through the seals as in a real engine operating condition. Thus, it was desirable to omit the modelling and simulation of the run-up for subsequent test cases. To assess the validity of such strategy, a further simulation was performed to investigate the effect of the temperature history from the run-up on the run-down solution. The idea is to simulate only the run-down and compare the resulting solution with that from the simulation of the entire test. Fig 6.38 shows the portion of the test included in the run-down simulation, which starts at

the time instant immediately after the second and permanent pulse in seal upstream pressure to avoid portions with no air flow (time=85 sec in the figure). The brush seal thermal correlation obtained in Chapter 5 is employed during the run-down to estimate the amount of heat absorbed by the seals. However, during the run-up this correlation is no longer applicable since there is no leakage flow and the heat transfer mechanism is different. The thermal modelling of the seal employed for the run-up was explained in Section 6.3.1. The uncertainty in the calculation of the seal-induced frictional power is higher during the run-up, since the torque exerted by the air motor has to be accounted (see Section 6.1.2).

As previously discussed, the thin-walls and their close vicinity are the only regions that experience a rise in temperature, while the central part of the rotor remains very close to ambient temperature. The temperature field shown in Fig 6.37b can thus be used as an approximation of the rotor temperature distribution at the start of the run-down simulation. The area of the domain in blue is set at the temperature measured by the infrared detectors at the beginning of the run-up (ambient temperature). A uniform temperature rise equal to that measured by the infrared sensors at the beginning of the run-down (time=85 sec in Fig 6.38) is imposed over the area covered by the rotor thin-walls (red zones in the figure). This temperature distribution is used by the model as initial solution of the run-down simulation.

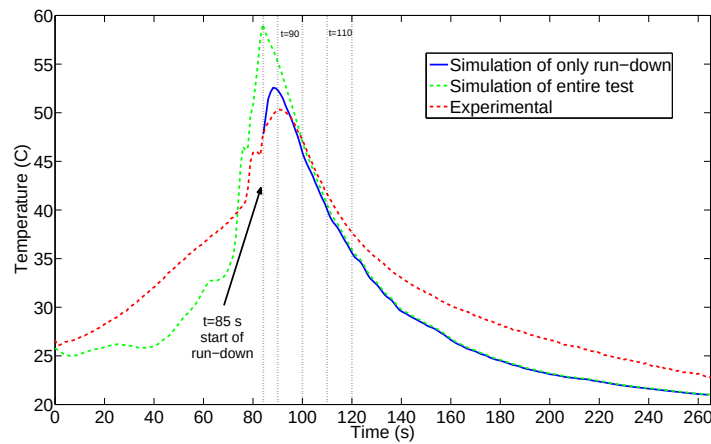


Figure 6.39: Comparison between rotor temperatures from experiment and two numerical simulations (entire test and only run-down), 2 bar SDP

A comparison between the temperature of the rotor measured experimentally and the predictions of both simulations (entire test and only run-down portion) is shown in Fig 6.39. The average of the temperatures measured by the infrared sensors is plotted together with the temperature predicted at the model at the equivalent location. It can be observed that the full FEA model fails to predict the rotor temperature during the run-up. Here the experimental rotor heats more quickly but reaches a lower peak temperature, this suggests that (i) more heat is transferred to the rotor than the bristles - which may be a function of the contact resistance at the bristle tips which is higher during run-up in the absence of blow-down on the bristles; (ii) there is some significant convective heat loss to the stagnant air in the cavity locally in the thin-walled area which is not captured in the model.

In the first 15 seconds after the start of the rundown (at $t = 85$ s), the experimental and foreshortened test both experience a turning point in reported rotor temperature near the contact area, although the temperature increase in the experiment is lower than for the simulation, and the dissipation of this temperature field is slower in the experiment than for either of the simulations. After the first 15 seconds temperatures predicted by both simulations are extremely self-consistent during for the remainder of run-down, and are in close agreement with the experimental values from 15 - 35 seconds, where the deviation in temperature rises to ~ 5 K. Both FEA models underpredict temperature by the end of the test.

Fig 6.40 shows the difference between the temperature fields within the bottom thin-wall from the two simulations at four time instants of the run-down (see time axis in Fig 6.39). It can be seen that the maximum deviation occurs at the interface between the thin-walls and the core of the rotor. In the zone where the cameras are pointing (i.e. under the seal contact surface), the maximum discrepancy is less than 3 K. The key conclusion from this comparison is that both simulations result in nearly the same rotor temperature field after ~ 10 seconds from the start of the run-down. The high thermal conductivity and low mass of the thin wall make its transient thermal response quite fast and essentially independent of previous temperature history. The model also suggests

that the convective and heat split performance of the seal is subject to some uncertainty and these might be determined through iteration of the boundary conditions within the FEA solution.

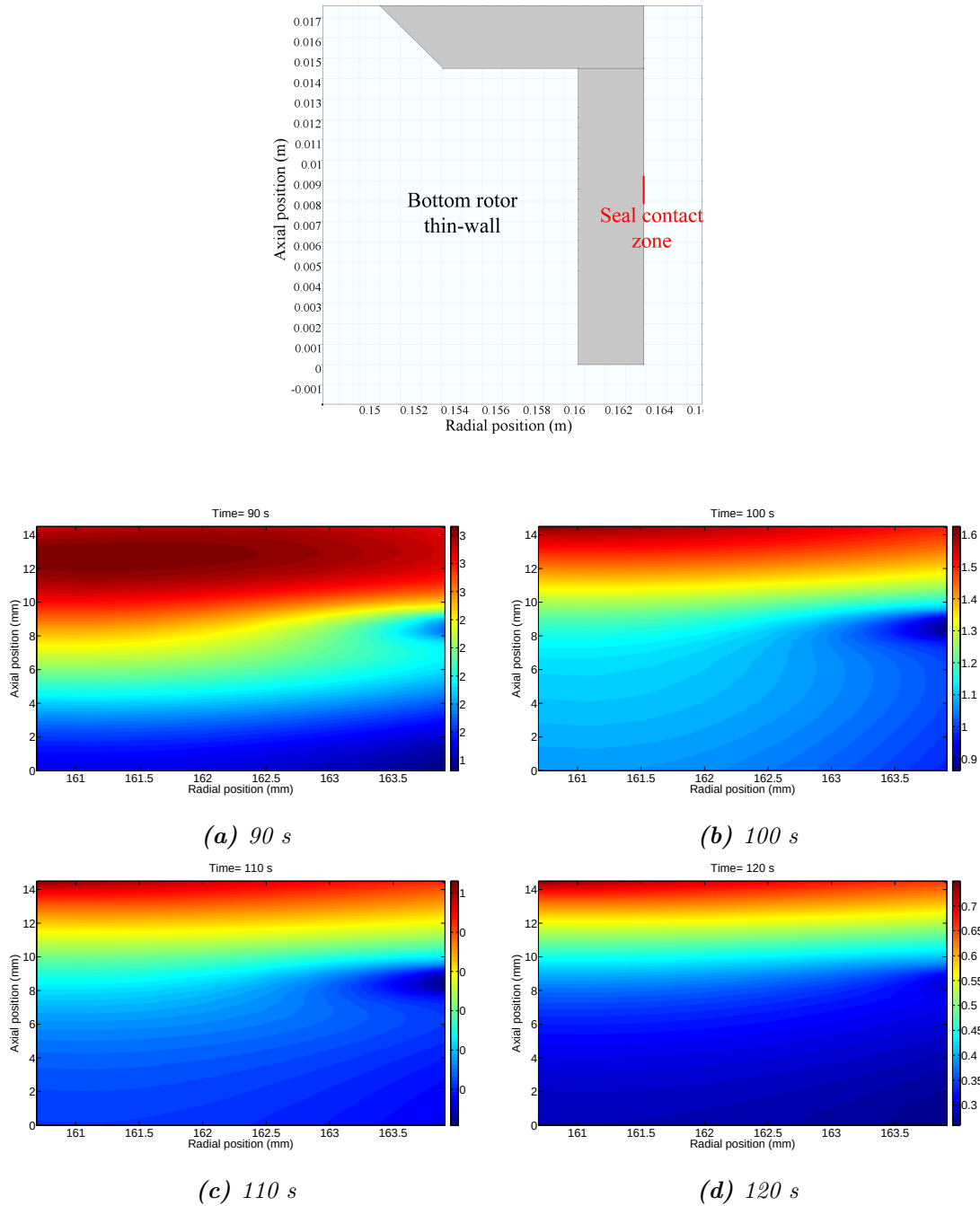


Figure 6.40: Difference in temperature (K) of the bottom thin-wall between the solutions of the entire test simulation (run-up and run-down) and only the run-down simulation at different time instants (90,100,110 and 120 s, see Fig 6.39). The x-axis refers to the radial thickness of the thin-wall, and the y-axis represents its axial length (refer to upper figure)

In light of these results, the run-up portion of the tests can be omitted if an adequate temperature distribution is imposed as initial solution for the run-down simulation. Since the heating of the rotor is mainly limited to the thin-wall sections, then the temperature field at the start of the run-down can be approximated by two regions. The first region corresponds to the core of the rotor, which stays cold and at a temperature equal to ambient. The second zone comprises the rotor thin-walls, which are approximated by a uniform temperature equal to that measured by the infrared sensors at the start of the run-down. It should be noted that simulating only the run-down with this initial condition could be more realistic than running the entire test with a poor modelling of the seal during the run-up. As shown in Fig 6.39, the solution of the run-down simulation is closer to the experimental value than that from the entire test simulation. Therefore, for the comparison between numerical and experimental results, only the run-down is simulated following this methodology. The beginning of the second pulse in seal upstream pressure is used to define the start of the run-down. The problems involved in the modelling of the run-up are thus circumvented.

The temperature distribution within the rotor during the run-down presents features similar to those found during the run-up. When the seals are pressurized, the rotor *htc* is especially high in the vicinity of the seals, as explained in Section 6.3.2.4. The frictional heat transferred to the rotor is quickly convected back into the surrounding flow. As a result, the rise in temperature mainly occurs at the rotor thin-walls, with the core remaining at a low temperature. Strong gradients in temperature must therefore exist at the interface between the thin-walls and central section of the rotor. This is shown in Fig 6.41, where a typical temperature field from a run-down simulation is depicted. In Fig 6.41b, the 2D temperature distribution 10 secs after the start of the run-down is illustrated, while Fig 6.41c and 6.41d show the variation in temperature with time at the measurement locations described in Fig 6.41a. It can be appreciated the uniformity in temperature on the lower side of the thin-wall (L side), while larger variations exist in the upper part (U side).

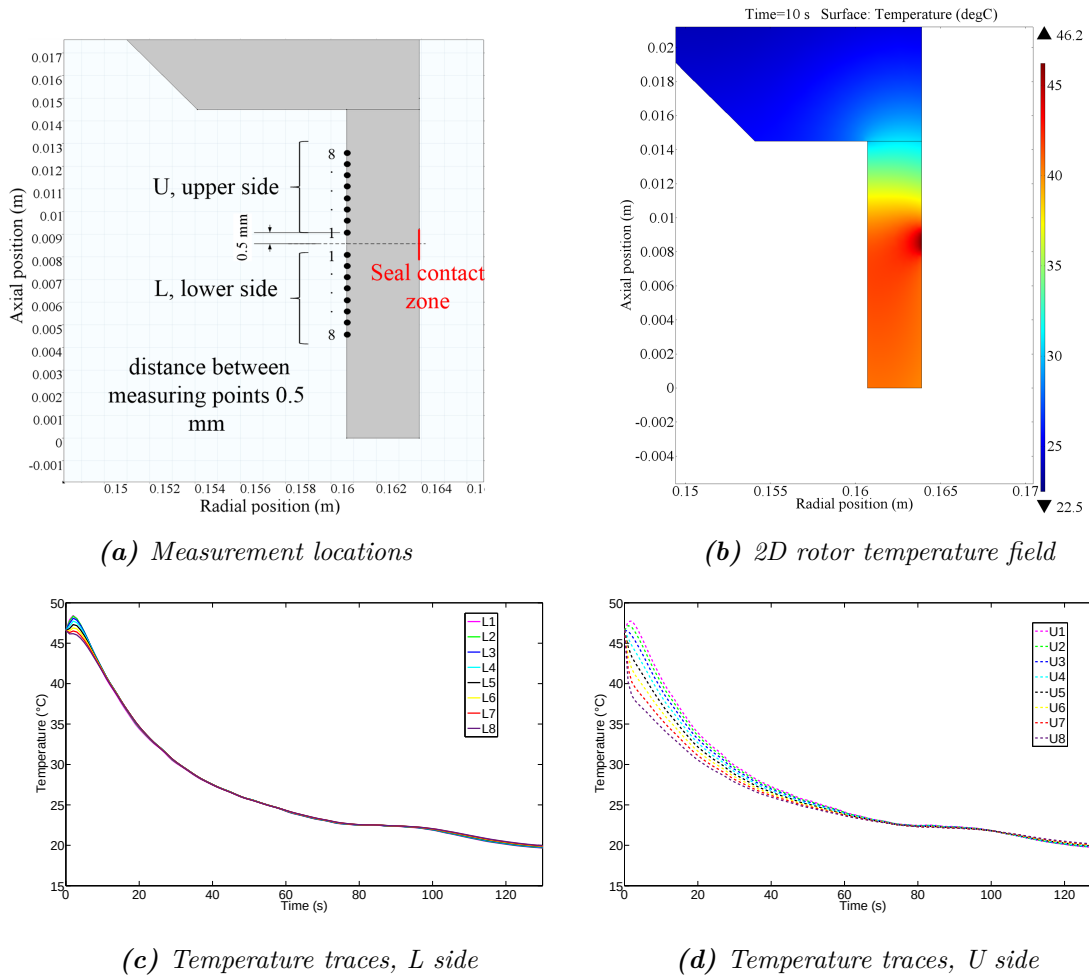


Figure 6.41: Example of temperature gradients within the bottom rotor thin-wall, example from a run-down simulation at 3 bar SDP

As explained in Section 6.1.2, the infrared sensors are installed in such a way that nominally they point at the centre line of the seal running position at the back of the bottom thin-wall (see schematic in Fig 6.5). They provide a value of the average rotor temperature over a measuring window of 6x10 mm. However, during the experimental campaign presented in Section 6.2, it was found that the middle infrared detector was measuring lower than the other two, which were reading a similar value. This effect is illustrated in Fig 6.42, which shows the rotor temperature traces measured by the 3 infrared sensors during the 3 bar SDP test. It can be appreciated the lower reading provided by the middle IR sensor. From the comparison between Fig 6.42 and Fig 6.41d, we can infer that the middle infrared sensor must be pointing slightly upper than

its nominal position, closer to the central section of the rotor. This could explain the discrepancy between sensors. Also, it can be appreciated how the difference between the temperatures at the different U locations in Fig 6.41d and between sensors in Fig 6.42 reduces as the run-down progresses.

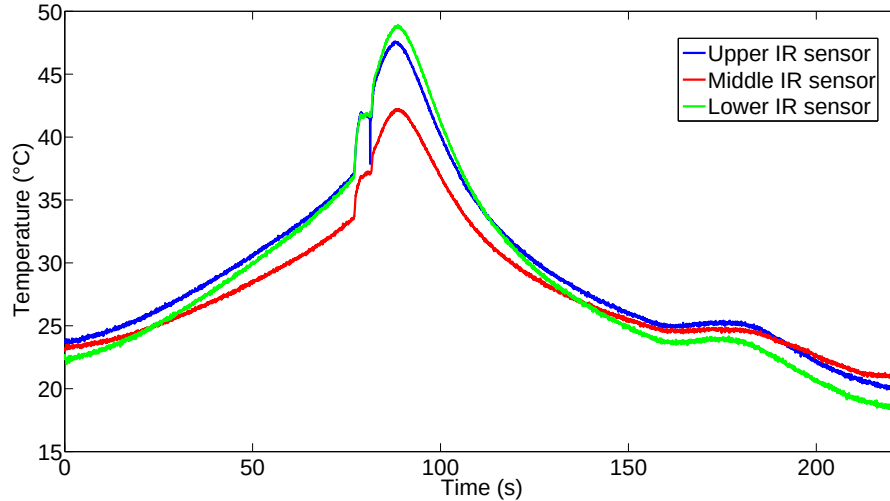


Figure 6.42: Rotor temperature measured by infrared (IR) sensors (3 bar SDP test)

For the comparison between the rotor temperatures measured experimentally and predicted by the model, the following approach is employed. The average between the upper and lower infrared sensors is chosen as the best indicator of the rotor experimental temperature. For the FE model, the temperature at the back surface of the bottom thin-wall is averaged along ± 3 mm axially centred around the seal running position (points L1 to L6 and U1 to U6 in Fig 6.41a). It is strongly recommended for future tests to set the cameras pointing slightly more towards the tip of the thin-wall than the root, in order to reduce the uncertainty in the measured rotor temperature.

6.4.2 Comparison between predicted and measured rotor temperatures

The numerical model described in Section 6.3 was run with boundary conditions from the experimental data presented in Section 6.2. Only the run-down portion of the tests was simulated, as discussed in Section 6.4.1. To assess the validity of the model, the

predicted temperature at the back of the rotor bottom thin-wall is compared with that measured by the infrared sensors during the tests. For such comparison, the approach explained at the end of Section 6.4.1 is employed.

In order to have different quantities in the same graph with an easily readable scale, they have been normalized in the following manner:

$$T^* = \frac{T - T_{min}}{T_{max} - T_{min}} \quad (6.13) \quad Q^* = \frac{Q}{Q_{max}} \quad (6.14) \quad \dot{m}^* = \frac{\dot{m}}{\dot{m}_{max}} \quad (6.15)$$

where the terms *max* and *min* correspond to the maximum and minimum values of the quantity considered developed during a test. Q refers to the power due to the seal-induced friction, T is the rotor temperature at the back of the thin-wall and $\dot{m}$ the leakage mass flow.

A fundamental result from these tests is the heat partition ratio, which indicates the proportion in which the frictional power is shared between the two contacting bodies, seal and rotor. The heat partition ratio η , expressed as a percentage, is thus defined as:

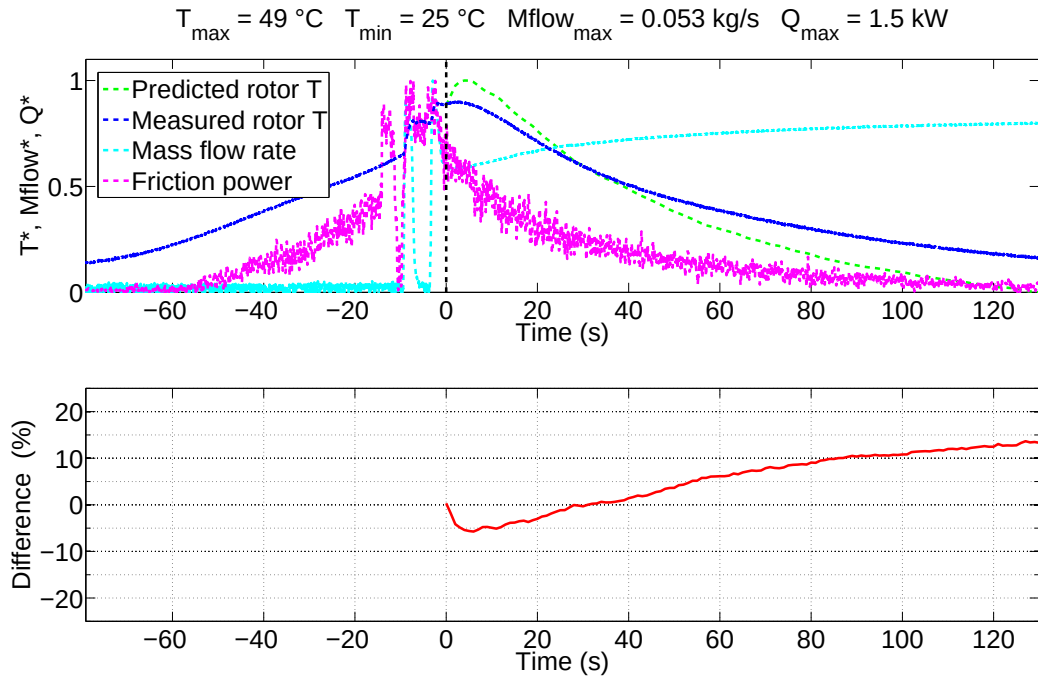
$$\eta = \frac{Q_{seal}}{Q_f} 100 \quad (6.16)$$

where Q_{seal} and Q_f are the heat transferred to the seal and the total frictional heat respectively. In the following Figs 6.43-6.48, the results from the six cases at different SDPs are shown. In these plots, the origin of the time axis corresponds with the start of the run-down simulation, and therefore, negative values of time refer to experimental data measured during the run-up. The power displayed is the heat generated by friction with the seals. In the top plot of each figure, both frictional power and leakage mass flow rate are the contribution of the two seals tested, while in the bottom plot the values of power correspond to a single seal. To quantify the difference between measured and predicted values of rotor temperature, the following formula was employed:

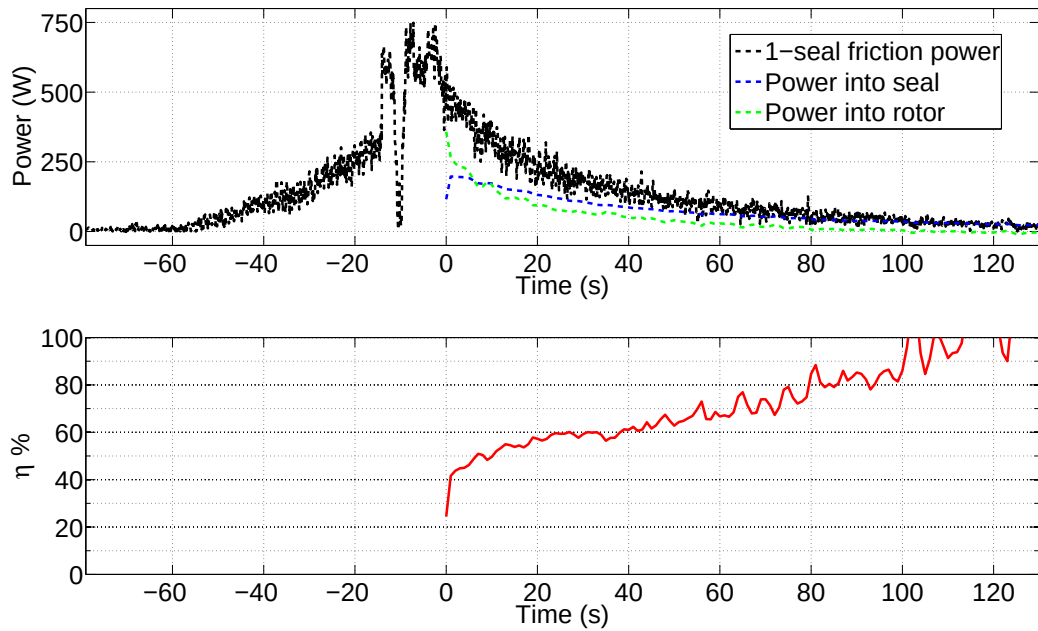
$$Difference (\%) = \frac{T_{measured} - T_{predicted}}{T_{measured}} 100 \quad (6.17)$$

Figs 6.43-6.48 confirm that the FE model predictions are in reasonable agreement with the experimental temperature traces, especially during the initial-medium portion of the run-down. The difference between predicted and measured values is lower than 15 % in all cases. In the case at 1 bar SDP, the model starts overpredicting the experimental temperature. As the run-down progresses, the predicted temperature approaches the experimental one until they cross, the experimental temperature being underpredicted for the remaining last part of the test. As discussed in Section 6.2, the mass flow leakage tends to increase with time during the run-down, due to the shrinkage of the rotor and reduction in bristle packing density as the rotor speed decreases. This is a clear symptom of a lighter contact between seal and rotor, especially at low SDP, where the blow-down forces are weaker and the bristles less compliant. This could explain why the model always predicts a rotor temperature lower than the experimental one at the end of the tests. If contact becomes very light or a gap opens, then the heat transfer mechanism assumed in the model would not be applicable anymore. Under such operating conditions, the cooling effect induced on the rotor by the seals would be less effective, but likewise the total power input would reduce. In consequence, a larger proportion of the total frictional power would be transferred to the rotor, resulting in the higher rotor temperatures found experimentally.

In the cases at 1.5 and 2 bar SDP, the same trend in rotor temperature is found as the one from the case at 1 bar SDP, although the initial portion during which the model over predicts the experimental temperature is much shorter. For 2.5, 3 and 3.5 bar SDP the model predicts a temperature lower than the experimental one during the entire run-down. Several reasons could explain this behaviour. For instance, it might be due to a low value of the seal thermal constant (K_{seal}) used, or an overprediction of the convective htc distribution on the rotor surface from the CFD simulations. In general, the difference between the model results and the experiments diminishes at the highest SDP tests ($< 10\%$), possibly because in these cases the stronger blow-down forces and resulting seal contact conditions are more in consonance with the assumptions behind the methodology employed. In any case, considering the uncertainties in the measured

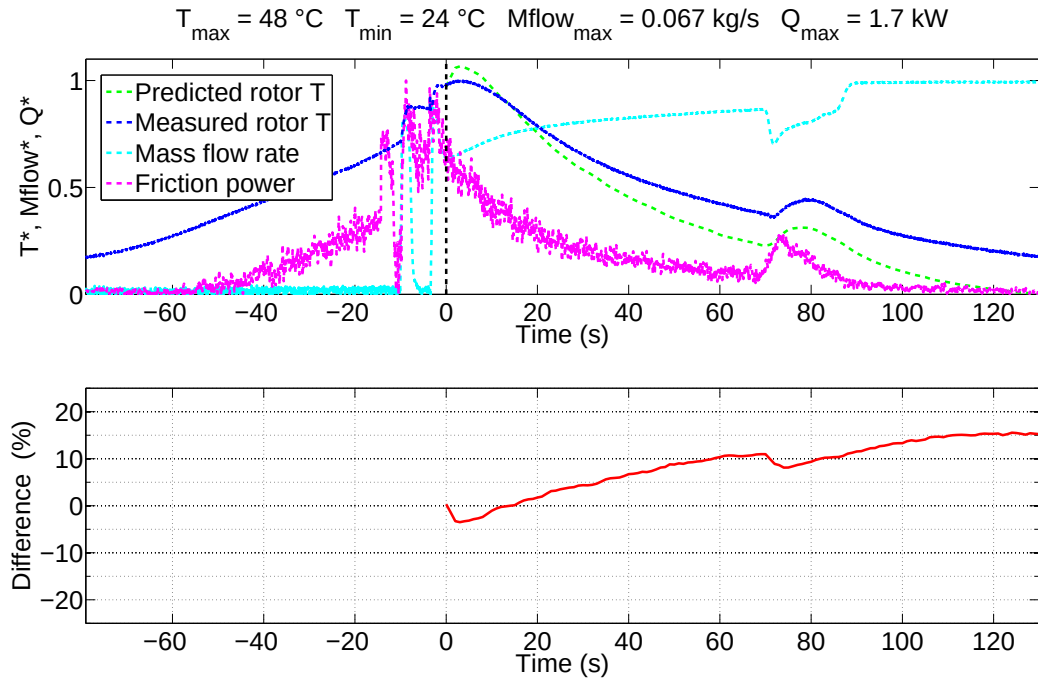


(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)

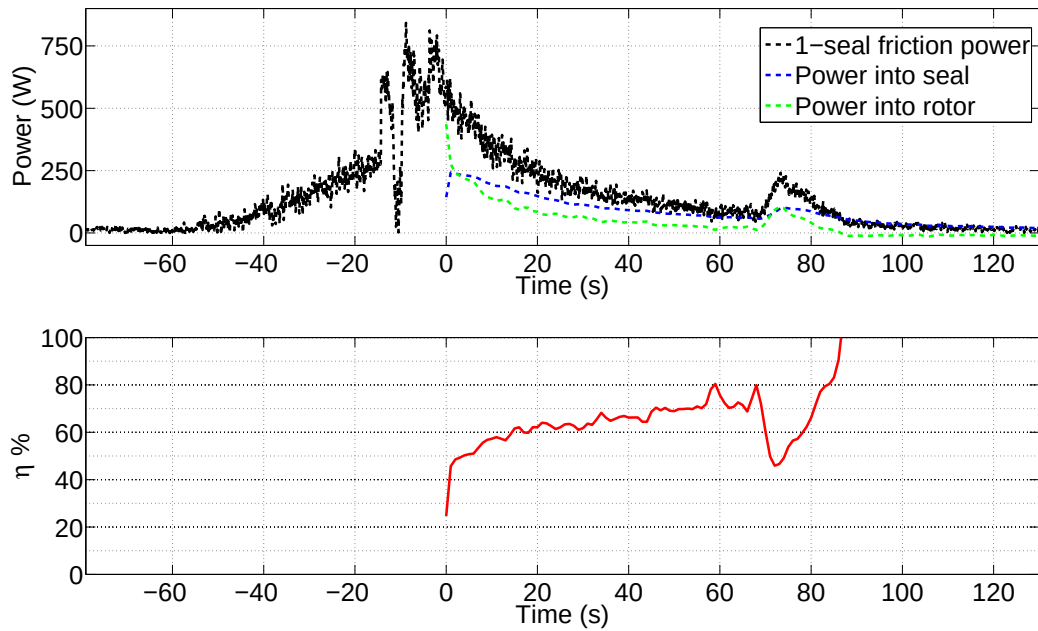


(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.43: 1 bar SDP

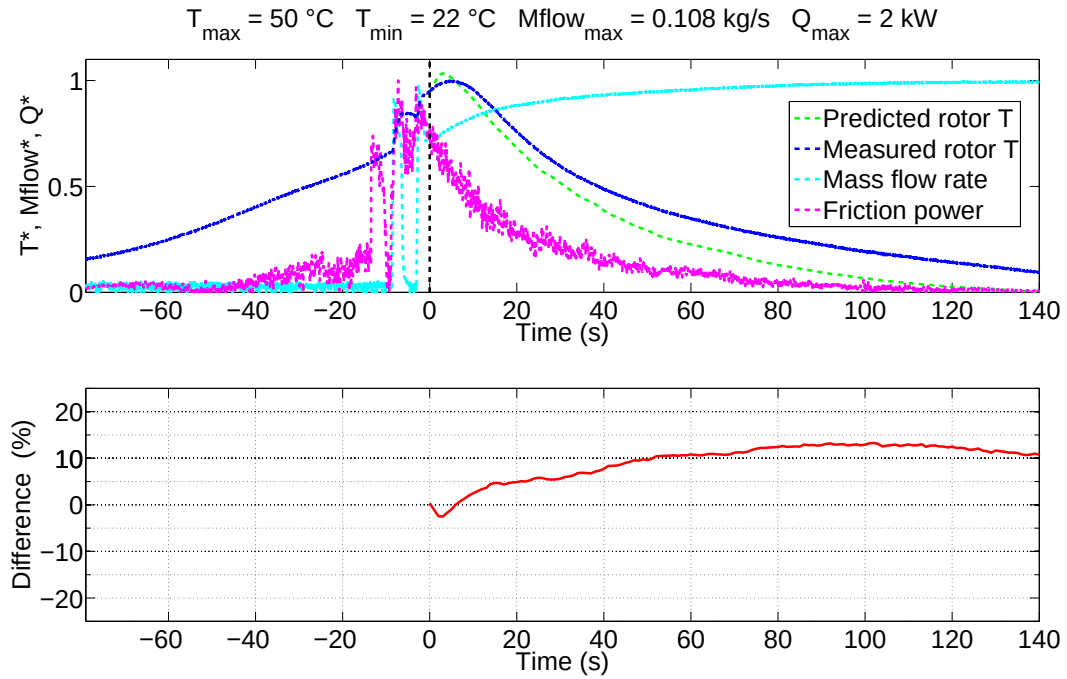


(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)

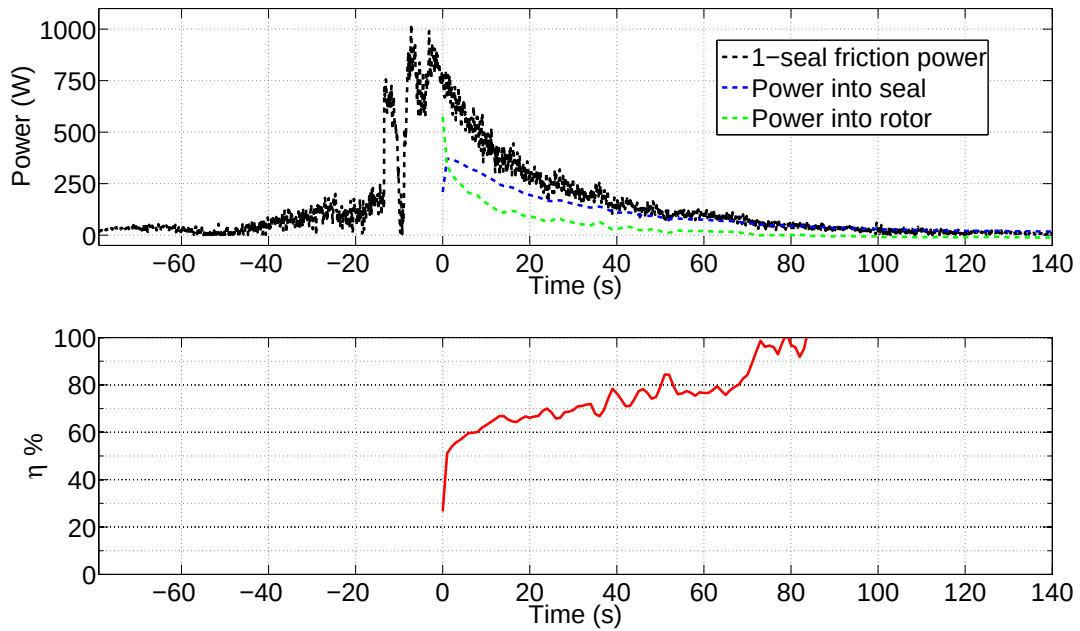


(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.44: 1.5 bar SDP

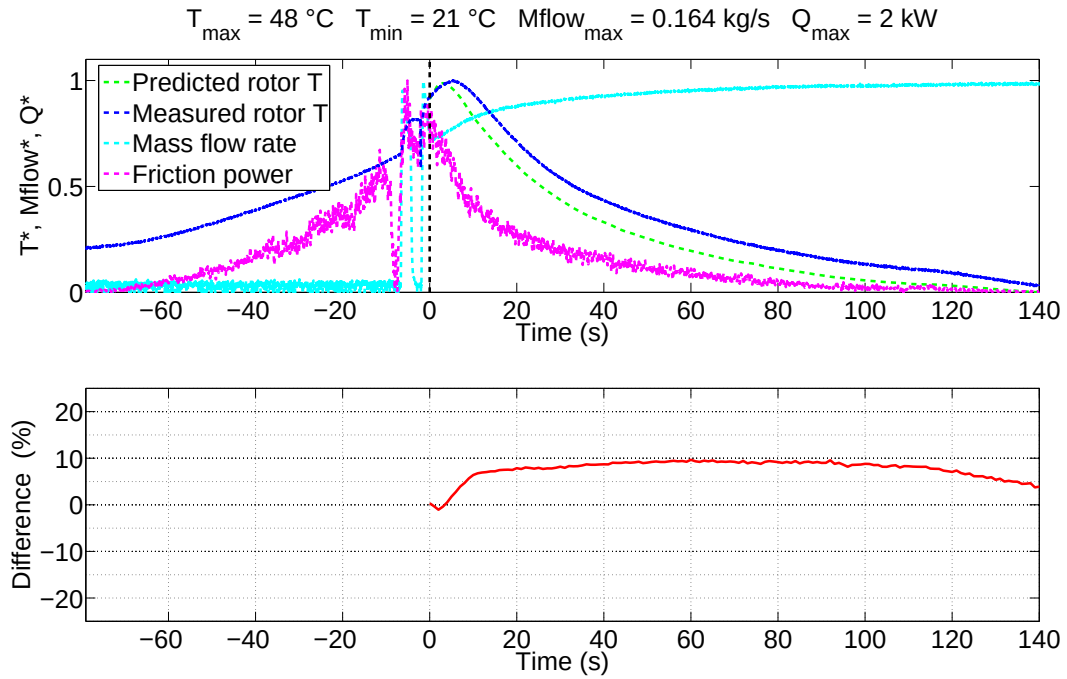


(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)

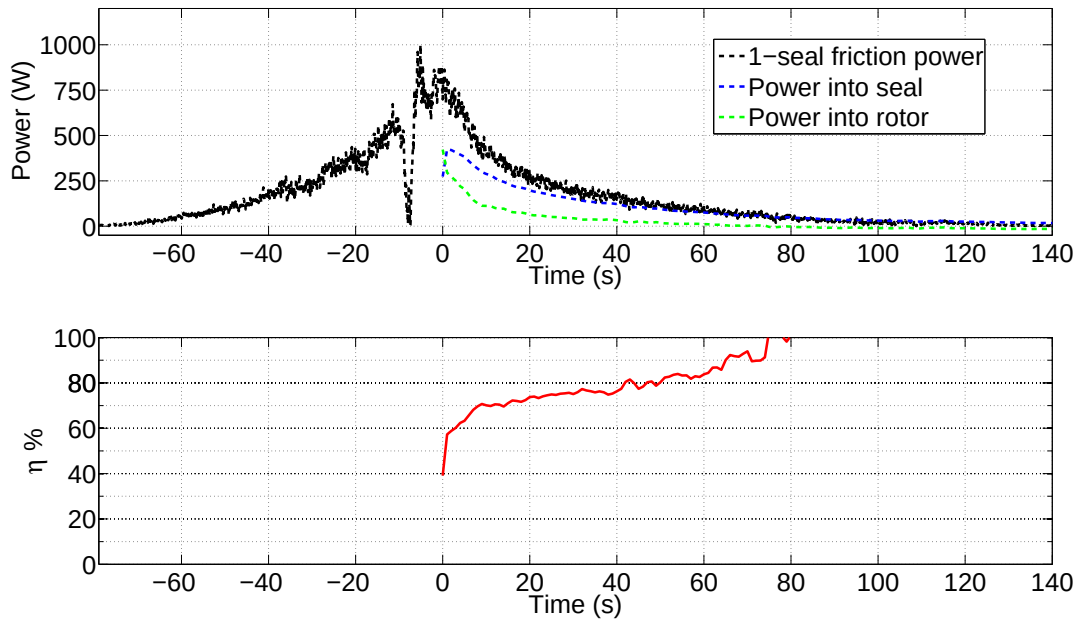


(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.45: 2 bar SDP

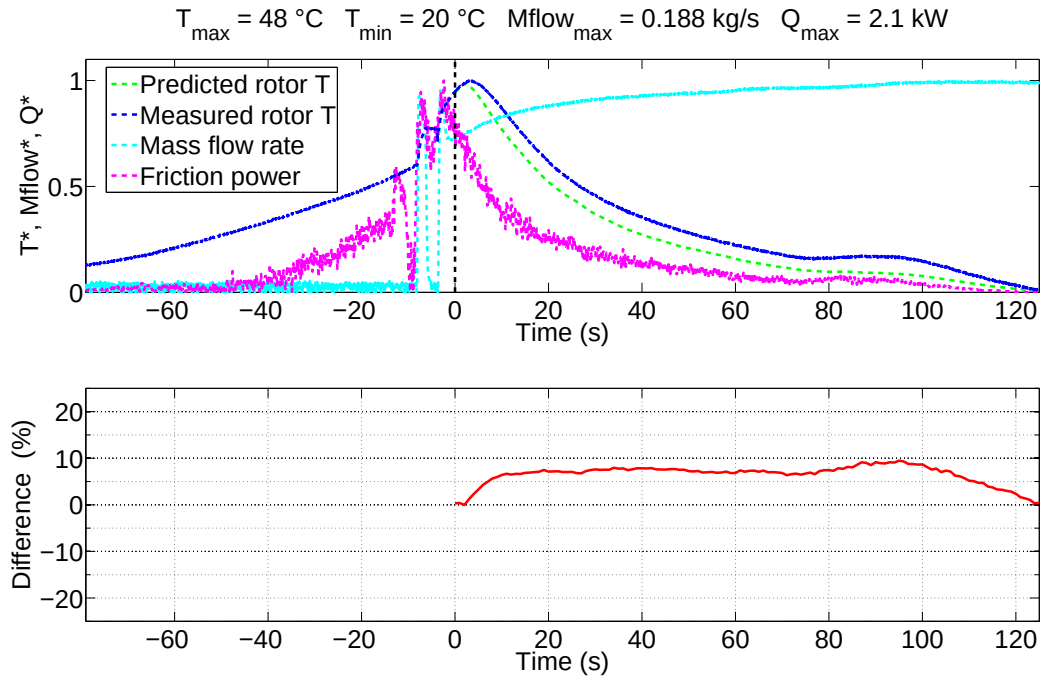


(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)

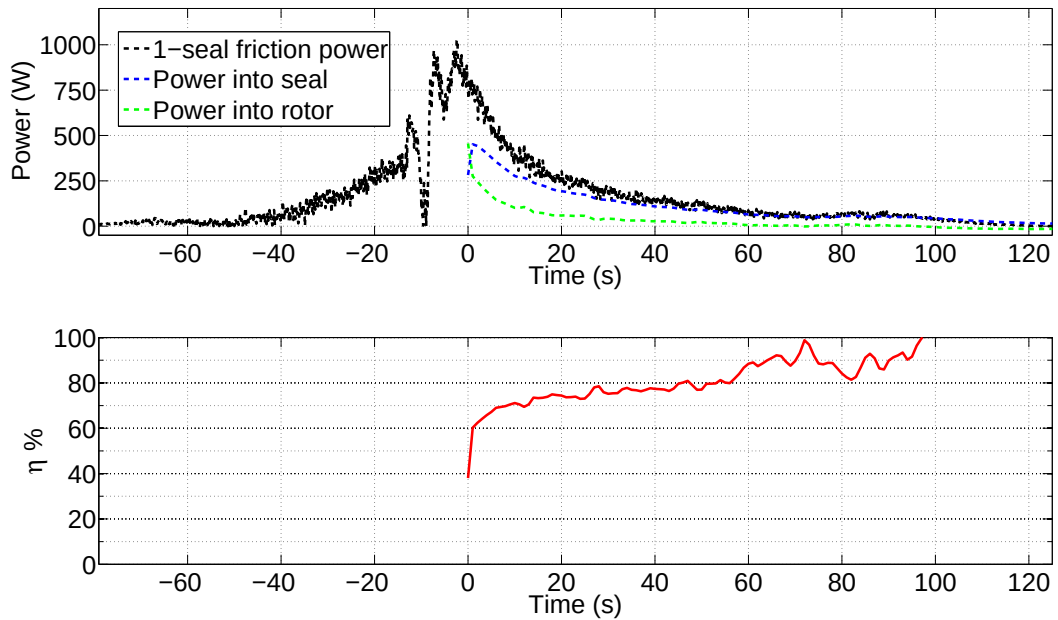


(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.46: 2.5 bar SDP

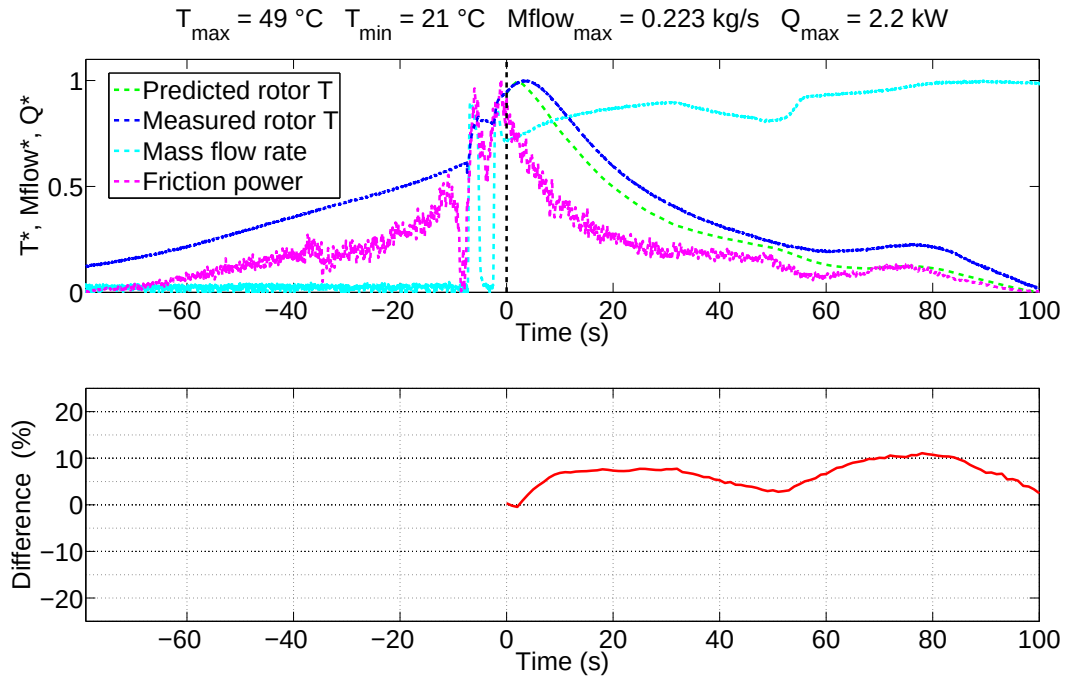


(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)

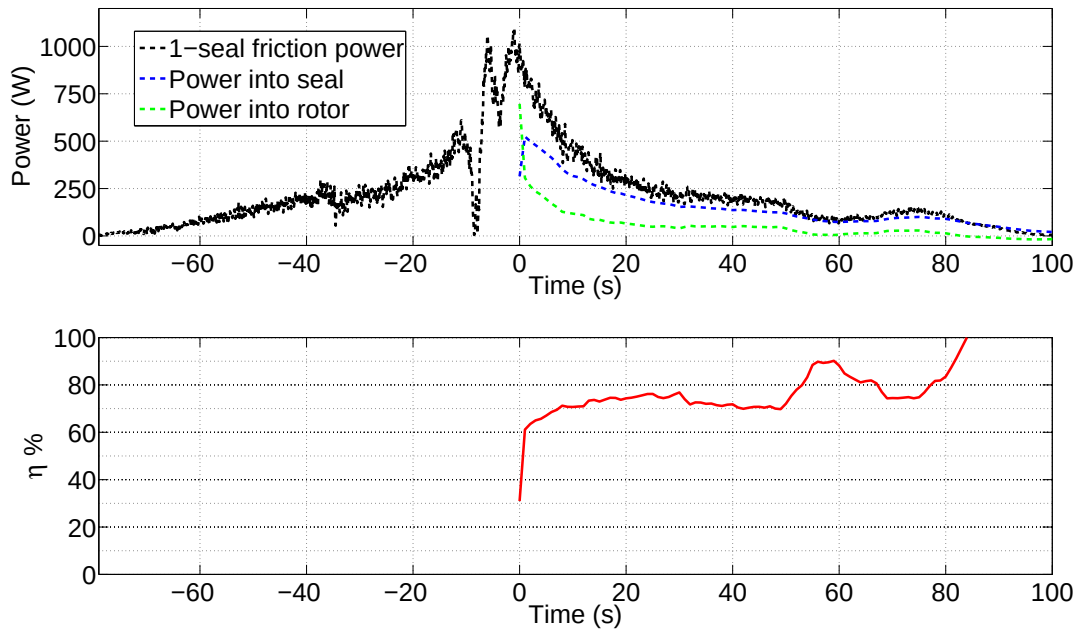


(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.47: 3 bar SDP



(a) 2-seal frictional power, 2-seal leakage mass flow and predicted and measured rotor temperatures (top), difference between predicted and measured rotor temperatures (bottom)



(b) 1-seal frictional power, power transferred to 1-seal, and power transferred to the rotor at one thin-wall section (top), evolution with time of heat partition ratio (bottom)

Figure 6.48: 3.5 bar SDP

quantities (power, air temperature, mass flow rate, rotor temperature, seal thermal constant) and the fact that the $htcs$ on the rotor surface are estimations from CFD simulations, the difference between predicted and measured rotor temperatures can be considered reasonable.

The heat partition ratio (η) presents the same trend in the six SDP cases. After a quick initial increment, it stays stable for approximately 30-40 seconds, which corresponds with the time interval where the model better matches the experimental rotor temperature. Then, it increases and eventually rises beyond 100%. A heat partition ratio beyond 100% can be physical. It means that the seal is absorbing not only all the heat generated by friction, but also it is acquiring some extra heat from the rotor. The rotor is heated up during the run-up, and contains certain thermal energy at the beginning of the run-down. The seal is such a strong heat sink that, at the low frictional powers occurring at the end of the run-down, it absorbs all the frictional power generated, and also, cools the rotor.

The evolution with time of the heat partition ratio can be explained referring to the aforementioned considerations on the comparison between predicted and measured rotor temperatures. As previously discussed, the model underestimates the rotor temperature at the end part of the run-down, which is likely to be due to a change in the contact conditions and seal heat transfer at low rotor speeds. Therefore, the model is implicitly assuming that a larger portion of the total frictional power than in reality is conducted into the seal, which results in the high heat partition ratios found at the end of the tests.

For the purpose of studying the effect of SDP on the heat partition ratio, its average value is calculated during 20 seconds after the first 10 seconds from the start of the run-down (in Figs 6.43-6.48 this corresponds to η between time 10s-30s). Fig 6.49 shows the relationship between leakage and heat partition ratio at the different SDPs tested. As the SDP and leakage mass flow rate increase, the cooling of both rotor and seal is enhanced due to the increment in htc on the rotor and bristle surfaces. However, the thermal resistance of the seal drops faster, and consequently a larger fraction of the total frictional power is transferred to the seal. The net result, as shown in the figure,

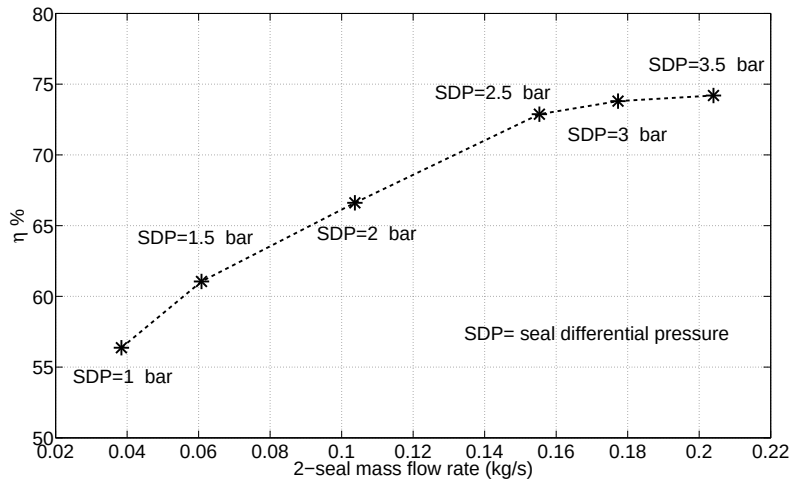


Figure 6.49: Heat partition ratio variation with seal leakage mass flow rate

is a rise in heat partition ratio with increasing SDP. It can be observed that between 1-2.5 bar SDP, the heat partitioning increases steadily, while beyond 2.5 bar SDP there is little variation and an asymptotic value of 75 % seems to be reached. In the case of no seal pressurization, it is expected that approximately 50% of the frictional heat conducts into the rotor and the other 50% into the bristles if both contacting bodies present similar values of thermal conductivity and surrounding sink temperature. As discussed in Chapter 2, previous studies have shown that the heat partition ratio rises in favour of the seal as the leakage flow and convection within the bristle pack increase. In the figure, it can be seen that at the lowest SDP tested (1 bar), the amount of frictional heat conducted into the seal is higher than that transferred to the rotor ($\eta \sim 57\%$).

It should be noted that the heat partition ratio is determined by the relative value of the thermal resistances of the seal and rotor, as explained in Chapter 3 with the electrical analogy. These thermal resistances depend on the physical properties of the two bodies in contact and the characteristics of the flow field that surrounds them. Also, the operating conditions, which in modelling terms correspond to the boundary conditions, strongly affect the heat partition ratio. Therefore, all these inputs have to be considered in the derivation of the heat partition ratio. Most of the previous work on this subject (see Chapter 2) have focused only on the investigation of the thermal

resistance of the seal, ignoring the heat transfer characteristics and boundary conditions of the rotor.

6.5 Summary

To assess the validity of the methodology proposed in this thesis, the existing Engine Seal Test Facility has been used to generate heat transfer data of a pair of brush seals running at engine representative seal differential pressures and rotor surface speeds. A finite element model of the test rotor has been built, which employs as boundary conditions *htc* distributions obtained with separate steady-state CFD model of the rig (based on the porous medium model previously developed, Chapter 4) and the empirical brush seal correlation (Chapter 5). Measured and predicted rotor temperatures were found to differ less than 15%, which gives credence to the seal friction thermal model presented in this thesis (see Fig 6.43-6.48). The agreement between numerical and experimental results was especially satisfactory at the highest rotor speeds and seal differential pressures, where the seal-rotor contact forces are greater and the assumptions behind the modelling strategy more valid. Reducing rotor speed and seal differential pressure diminishes the contact load due to the centrifugal shrinkage of the rotor and drop in blow-down forces. If seal-rotor contact becomes very light or even cease, then, the seal friction model employed is no longer applicable. In addition, the partition of the frictional heat during this tests was studied with the model built. It was found that the fraction of the total frictional heat transferred to the seal rapidly grows with increasing seal differential pressure and thus, leakage flow (see Fig 6.49). At the lowest seal differential pressure tested (1 bar), the fraction of the frictional heat absorbed by the seal was calculated to be approximately 57%, while above 2.5 bar this value increased over 70 %. Knowledge of the heat distribution in an engine seal installation is crucial for engine designers, since the heat generated by friction propagates to adjacent and downstream components, having a big impact on the overall engine performance. Surprisingly, as discussed in Chapter 2, previous authors have overlooked the partitioning of the frictional heat.

Chapter 7

Concluding Summary

Increasing the performance of the secondary fluid system of gas and steam turbines is vital to meeting improved emission and efficiency targets. Filament seals can break through into high pressure and temperature locations if their heat transfer characteristics over the full engine cycle can be better understood. The current work identifies a simple model for heat transfer from the seals which, with semi-empirical constants obtained in a laboratory environment, can model the temperature rise of both fluid and rotor in the seal vicinity with sufficient accuracy to ensure rotor integrity, and predict contact temperatures necessary for lifting activities.

A new test rig for brush and leaf seals, the Seal Static Thermal Test Facility, was developed, able to use both thermocouples and thin film gauges to measure temperature at low and high pressure in the air flow around the seal, but also on and within the rotor representative surface. Measurements were made of htc on a rotor representative surface in the close vicinity of seals for the first time by providing a heated flow to the test rig and measuring the transient response of the surface. The heat transfer coefficients were reported based on the local adiabatic wall temperature, allowing their use in streamwise energy balance models typically used in the gas turbine design process. The extent of the region of the rotor where the heat transfer coefficient was enhanced (lying in a region approximately 15 mm upstream and downstream of a 4 mm wide leaf seal) was considerably larger than assumed by previous workers who had considered this axial

length to be of the same order as the height of the seal cover plates (i.e. $<3\text{mm}$).

Whereas the upstream enhancement was generally in line with the levels expected for annular flow in a passage of similar geometry, downstream of the seal the high speed jet formed caused additional enhancement not captured by existing correlations, and the level of heat transfer was approximately 3-5 times higher. This is particularly significant as that existing flow will have an elevated driving gas temperature difference in an engine test. Comparison to existing annular flow correlations for co- and counter rotating annuli suggested that this enhancement dwarfed any changes in heat transfer likely due to the rotation of the flow. CFD simulations using a tuned porous medium somewhat surprisingly captured the heat transfer behaviour of the seal well in spite of not modelling the discrete jetting flow through the filament seal. Thus it is concluded that such a model might adequately capture the thermal behaviour of the seal in a more realistic engine environment.

In an extension to the experimental campaign, a simulated heat flux was supplied to both leaf and brush seals through the surface of the rotor representative surface. Here the purpose was to determine the contact temperature at the rotor-seal interface. With this knowledge it is possible to determine a heat transfer parameter for the filament packs of brush and leaf seals which can be used to determine the heat partition between the rotor and seal. The Oxford brush-leaf seal heat transfer model of Pekris (2010) can thus be calibrated using this experimental setup. Values of 47.7 and $18.4 \text{ m}^2\text{kg}^{0.5}\text{s}^{-2.5}\text{K}^{-1}$ were found respectively for brush and leaf seals. The functional form used was in line with that of previous workers, though here changing the leakage with changing local flow temperature and viscosity was observed. A simple equation was found to be able to correct cold flow leakage measurements to take account of this factor.

To assess the validity of a modelling strategy based on a porous CFD model with empirically determined seal heat transfer constants (K_{seal}), the Oxford Engine Seal Test Facility was used to generate heat transfer data for a pair of brush seals running at engine representative seal differential pressures and rotor surface speeds. A finite element model of the test rotor was built, which employs as boundary conditions *h**tc* distributions

obtained from a separate CDS model of the rig (based on the porous medium model previously developed) and the empirical brush seal correlation derived.

Measured and predicted rotor temperatures were found to differ less than 15%, which gives credence to the seal friction thermal model presented in this work (see Figs 6.43-6.48). The agreement between numerical and experimental results was especially satisfactory at the highest rotor speeds and seal differential pressures, where the seal-rotor contact forces are greater and the assumptions behind the modelling strategy more valid. Reducing rotor speed and seal differential pressure diminishes the contact load due to the centrifugal shrinkage of the rotor and drop in blow-down forces. If seal-rotor contact becomes very light or ceases, then, the seal friction model employed is no longer applicable. From these tests the partition of the frictional heat was studied. It was found that the fractions of the total frictional heat transferred to the seal rapidly grows with increasing seal differential pressure (or leakage flow, see Fig 6.49). At the lowest seal differential pressure tested (1 bar), the fraction of the frictional heat absorbed by the seal was calculated to be approximately 57%, while above 2.5 bar this value increased over 70%.

Overall, this thesis has improved the understanding of the thermal behaviour of brush and leaf seals in the fully contacting regime. The methodology developed throughout this thesis has proven to be suitable for building accurate and robust brush and leaf seal heat transfer models.

Further work is required to extend the work into a design tool. The first stage of this is to use data currently being generated on the Oxford ESTF on a wide variety of seals from numerous manufacturers to form a general correlation for the seal constant K_{seal} as a function of seal geometry. Opportunity then exists to compare results from the model to engine tests for a large civil engine brush seal to be conducted by the industrial partner of this work in either 2018 or 2019, where full engine cycles can be operated and the closure pinch points for the seal further investigated and the model validated.

Bibliography

- Archard, J. (1958). The temperature of rubbing surfaces. *Wear*, Vol. 2:pp. 438–455.
- Ashby, M., Abulawi, J., and Kong, H. S. (1991). Temperature maps for frictional heating in dry sliding. *Tribology Transactions*, Vol. 34:pp. 577–587.
- Bansal, D. and Streater, J. L. (2009). A method for obtaining the temperature distribution at the interface of sliding bodies. *Wear*, Vol.266:pp. 721–732.
- Barber, J. (1970). The conduction of heat from sliding solids. *International Journal of Heat and Mass Transfer*, Vol.13:pp. 857–869.
- Bayley, F. J. and Long, C. A. (1992). A combined experimental and theoretical study of flow and pressure distributions in a brush seal. *Proceedings of the International Gas Turbine and Aeroengine Congress and Exposition 1992, Cologne, Germany*, 92-GT-355.
- Billiard, N., Iliopoulou, V., Ferrara, F., and Dénos, R. (2002). Data reduction and thermal product determination for single and multi-layered substrates thin-film gauges. *The 16th Symposium on Measuring Techniques in Transonic and Supersonic Flow in Cascades and Turbomachines, Cambridge (UK)*.
- Blok, H. (1937). Theoretical study of temperature rise at surface of actual contact under oiliness lubricating conditions. *Proceedings of the General Discussion on Lubrication and Lubricants*, Vol.2:pp. 222–235.
- Blok, H. (1963). The flash temperature concept. *Wear*, Vol.6:pp. 483–494.

- Bos, J. and Moes, H. (1995). Frictional heating of tribological contacts. *Journal of Tribology*, Vol. 117.
- British Standards Institution, (1981). British standard measurement of fluid flow in closed conduits. *BS1042, Section 1.1*.
- Cameron, A., Gordon, A. N., and Symm, G. T. (1965). Contact temperatures in rolling/sliding surfaces. *Proceedings of the Royal Society of London*, Series A 286:pp. 45–61.
- Chen, W. and Wang, Q. (2008). Thermomechanical analysis of elastoplastic bodies in a sliding spherical contact and the effects of sliding speed, heat partition, and thermal softening. *Journal of Tribology*, Vol. 130.
- Chew, J. W. and Guardino, C. (2004). Simulation of flow and heat transfer in the tip region of a brush seal. *International Journal of Heat Transfer and Fluid Flow*, Vol. 25:pp. 649–658.
- Chew, J. W., Lapworth, B. L., and Millener, P. J. (1995). Mathematical modeling of brush seals. *International Journal of Heat and Fluid Flow*, Vol. 16:pp. 493–500.
- Childs, P. R. N. and Turner, A. B. (1994). Heat transfer on the surface of a cylinder rotating in an annulus at high axial and rotational reynolds numbers. *10th International Heat Transfer Conference, Paper 4-EC-5*.
- Chupp, R., Hendricks, R. C., Lattime, S. B., and Steinetz, B. M. (2006). Sealing in turbomachinery. *Journal of Propulsion and Power*, Vol. 22.
- Chupp, R. E. and Prior, R. J. (1997). Advanced seal development for large industrial gas turbines. *AIAA Paper No. AIAA-97-2731*.
- Clarke, A., Sharif, K., Evans, H. P., and Snidle, R. W. (2006). Heat partition in rolling/sliding elastohydrodynamic contacts. *Journal of Tribology*, Vol. 128:pp. 67–78.

- Colin, F. and Floquet, A. (1986). Combination of finite element and integral transform techniques in a heat conduction quasi-static problem. *International Journal of Numerical Methods in Engineering*, Vol. 23:pp 13–26.
- Demiroglu, M. and Tichy, J. A. (2007). An investigation of heat generation characteristic of brush seals. *Proceedings of the ASME Turbo Expo 2007, Montreal, Canada*, GT2007-28043.
- Dinc, S., Demiroglu, M., Turnquist, N., Mortzheim, J., Gayle, G., Maupin, J., Hopkins, J., Wolfe, C., and Florin, M. (2002). Fundamental design issues of brush seals for industrial applications. *ASME Journal of Turbomachinery*, Vol. 124:pp. 293 – 300.
- Dogu, Y. (2005). Investigation of brush seal flow characteristics using bulk porous medium approach. *ASME Journal of Engineering for Gas Turbine and Power*, Vol. 127:pp. 136–144.
- Dogu, Y. and Aksit, M. (2006). Brush seal temperature distribution analysis. *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 128:pp. 599–609.
- Energy UK, (2016). <http://www.energy-uk.org.uk>. accessed 19 Sep 2016.
- Epstein, A. H., Guenette, G. R., Norton, R. J. G., and Chao, Y. Z. (1986). High-frequency response heat-flux gauge. *Review of Scientific Instruments*, Vol. 57:pp. 639–649.
- Ferguson, J. G. (1988). Brushes as high performance gas turbine seals. *American Society of Mechanical Engineers, Paper 88-GT-182*.
- Ferguson, J. G. and Waters, G. (1975). Seals and method of manufacture thereof. *United States Patent (US0003917150)*. Rolls-Royce (1971) Limited.
- Fico, V., Pekris, M. J., Barnes, C. J., Jha, R. K., and Gillespie, D. (2016). CFD and thermal analysis of leaf seals for aero-engine application. *Proceedings of the ASME Turbo Expo 2016, Seoul, South Korea*, GT2016-56735.

- Flouros, M., Hendrick, P., Outirba, B., Cottier, F., and Proestler, S. (2014). Thermal and flow phenomena associated with the behaviour of brush seals in aero engine bearing chambers. *Proceedings of the ASME Turbo Expo 2014, Dusseldorf, Germany*, GT2014-25538.
- Flower, R. (1992). Brush seal with asymmetrical elements. *United States Patent (US0005135237)*. Cross Manufacturing Company (1938) Limited.
- Franceschini, G. (2005). *Shaft Seals for Turbine Engines*. PhD thesis, University of Oxford.
- Franceschini, G., Gillespie, D., and Jones, T. V. (2010). Improved understanding of blow-down in filament seals. *ASME Journal of Turbomachinery*, Vol. 132.
- Francis, H. A. (1970). Interfacial temperature distribution within a sliding hertzian contact. *ASLE Transactions*, Vol. 14:pp. 41–54.
- Furey, M. J. (1964). Surface temperatures in sliding contact. *ASLE Transactions*, Vol.7:pp. 133–146.
- Gao, J., Lee, S., Ai, X., and Nixon, H. (2000). An fft-based transient flash temperature model for general three-dimensional rough surface contacts. *ASME Journal of Tribology*, Vol. 122:pp. 519–523.
- Garimella, S. and Sobhan, C. (2003). Transport in microchannels, a critical review. *Annual Review on Heat Transfer*, Vol. 13.
- Gecim, B. and Winer, W. (1984). Steady temperature in a rotating cylinder subjected to surface heating and convective cooling. *Journal of Tribology*, Vol. 106:pp. 120–126.
- General Electric (2016). <https://powergen.gepower.com>. accessed 15 sept 2016.
- Greenwood, J. and Williamson, J. B. P. (1966). Contact of nominally flat surfaces. *Proceedings of the Royal Society of London, Series A* 295:pp. 300–319.

- Guo, S., Oldfield, M. L. G., and Rawlinson, A. J. (2002). Influence of discrete pin shaped surface roughness (p-pins) on heat transfer and aerodynamics of film cooled aerofoil. *Proceedings of the ASME Turbo Expo 2002, Amsterdam, The Netherlands*, GT2002-30179.
- Halliwell, I. (2014). An ultra-high bypass ratio turbofan engine for the future. *AIAA Air Breathing Propulsion Group and IGTI Aircraft Engines & Education Committees, Request for Proposal, Undergraduate Team Engine Student Design Competition 2014/15*.
- Hendricks, R., Steinetz, B. M., and Munson, S. J. (1998). Advanced seal technology role in meeting next generation turbine engine goals. *NASA Report NASA/TM-1998-206961*.
- Hendriks, R. C., Schlumberger, S., J., B. M., Choy, F., and Mullen, R. L. (1991). A bulk flow model of a brush seal system. *ASME Paper No. 91-GT-325*.
- Hogg, S., Hughes, S., and Jones, T. (1991). A test facility for the study of the thermofluid-dynamics of gas bearing lubrication films. *Isabe 10 Nottingham, England, Vols 1 And 2 Pages: 771-779*.
- Huang, S., Suo, S. Li, Y., and Wang, Y. (2014). Theoretical and experimental investigation on tip forces and temperature distributions of the brush seal coupled aerodynamic force. *ASME Journal of Engineering for Gas Turbine and Power*, Vol. 136:pp. 052502–1–12.
- Hughes, S. (1992). *A Study of the Distribution of Pressure and Flow in a Dynamic Thrust Bearing*. PhD thesis, University of Oxford.
- Incropera, F., Dewitt, D., Bergman, T., and Lavine, A. (2013). *Principles of Heat and Mass Transfer*. John Wiley & Sons, 7th edition.
- Ingo, I., Gillespie, D., Owen, A. K., and Franceschini, G. (2008). Experimental charac-

- terisation of the stiffness and leakage of a prototype leaf seal for turbine applications. *Proceedings of the ASME Turbo Expo 2008, Berlin, Germany*, GT2008-51206.
- Jaeger, J. C. (1942). Moving sources of heat and the temperature at sliding contacts. *Journal and Proceedings of the Royal Society of New South Wales*, Vol. 76:pp. 203–224.
- Jahn, I. H. (2008). *Leaf Seals For Gas and Steam Turbines*. PhD thesis, University of Oxford.
- Jahn, I. H., Gillespie, D. R. H., and Cooper, P. (2013). Hydrodynamic air-riding in leaf seals. *Proceedings of ASME Turbo Expo 2013, San Antonio, Texas, USA*, GT2013-95585.
- Jones, T. V., Guo, S. M., Lai, C. C., Oldfield, M. L. G., Lock, G. D., and Rawlinson, A. J. (2000). Influence of surface roughness on heat transfer and effectiveness for a fully film cooled nozzle guide vane measured by wide band liquid crystals and direct heat flux gauges. *ASME Journal of Turbomachinery*, Vol. 122:pp. 709–716.
- Justak, J. (2002). Robust hydrodynamic brush seal.
- Kadicic, A., Sayles, R. S., and Ioannides, E. (2008). Thermo-mechanical model for moving layered rough surface contacts. *Journal of Tribology*, Vol. 130.
- Kays, W., Crawford, M., and Weigand, B. (2004). *Convective Heat and Mass Transfer*. McGraw-Hill, 4th edition.
- Kays, W. M. and Leung, E. Y. (1963). Heat transfer in annular passages: Hydrodynamically developed turbulent flow with arbitrarily prescribed heat flux. *International Journal of Heat and Mass Transfer*.
- Kennedy, F. and Tian, X. (1993). Contact surface temperature models for finite bodies in dry and boundary lubricated sliding. *ASME Journal of Tribology*, Vol.115:pp. 411–418.

- Kennedy, F. and Tian, X. (1994). Maximum and average flash temperatures in sliding contacts. *ASME Journal of Tribology*, Vol. 116:pp. 167–174.
- Kennedy, F. E. (1981). Surface temperatures in sliding systems - a finite element analysis. *ASME Journal of Lubrication Technology*, Vol. 103:pp. 90–96.
- Kennedy, F. E. (1984). Thermal and thermo-mechanical effects in dry sliding. *Wear*, Vol. 100:pp. 453–476.
- Komanduri, R. and Hou, Z. (2001). Analysis of heat partition and temperature distribution in sliding systems. *Wear*, Vol. 251:pp. 925–938.
- Kuzay, T. M. and Scott, C. J. (1977). Turbulent heat transfer studies in an annulus with inner cylinder rotation. *ASME Journal of Heat Transfer*, Vol. 99:pp. 12–19.
- Laraqi, N., Alilat, N., Garcia de Maria, J., and Baïri, A. (2009). Temperature and division of heat in a pin-on-disc frictional device - exact analytical solution. *Wear*, Vol. 266:pp. 765–770.
- Lelli, D., Chew, J. W., and Cooper, P. (2006). Combined three-dimensional fluid dynamics and mechanical modelling of brush seals. *ASME Journal of Turbomachinery*, Vol. 128:pp. 188–195.
- Ling, F. and Pu, S. (1964). Probable interface temperatures of solids in sliding contact. *Wear*, Vol. 7:pp. 23–34.
- Ling, F. and Simkins, T. (1963). Measurement of pointwise juncture condition of temperature at the interface of two bodies in sliding contact. *Journal of Basic Engineering*, Vol. 85:pp. 481–487.
- Ling, F. F. (1959). A quasi-iterative method for computing interfacial temperature distribution. *Zeitschrift fur Angewandte Mathematik*, Vol. 10:pp. 461–474.

- Ling, J. P. C. W., Ireland, P. T., and Turner, L. (2004). A technique for processing transient heat transfer liquid crystal experiments in the presence of lateral conduction. *ASME Journal of Turbomachinery*, Vol. 126:pp. 247–258.
- Loizou, A., Qi, H., and Day, A. (2013). A fundamental study on the heat partition ratio of vehicle disk brakes. *ASME Journal of Heat Transfer*, Vol. 135.
- Majcherczak, D., Dufr  noy, P., and Nait-Abdelaziz, M. (2005). Third body influence on thermal friction contact problems: Application to braking. *ASME Journal of Tribology*, Vol. 127:pp. 89–95.
- Mitsubishi Hitachi Power Systems, (2015). <http://www.mhps.com>. accessed Nov 2015.
- Moffat, R. J. (1988). Describing the uncertainties in experimental results. *Experimental Thermal and Fluid Science*, Vol.107:pp. 3–17.
- Morgan, J. (2004). *Further Investigations into the Mechanics of Brush Seals*. PhD thesis, University of Oxford.
- Morini, G. and Yang, Y. (2013). Guidelines for the determination of single-phase forced convection coefficients in microchannels. *ASME Journal of Heat Transfer*, Vol. 135.
- Nakane, H., Maekawa, A., Akita, E., Akagi, K., Nakano, T., Nishimoto, S., Hashimoto, S., Shinohara, T., and Uehara, H. (2004). The development of high performance leaf seals. *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 126(2):pp–342–350.
- Nosko, O. (2013). Partition of friction heat between sliding semi-spaces due to adhesion-deformational heat generation. *International Journal of Heat and Mass Transfer*, Vol. 64:pp. 1189–1195.
- Oldfield, M. L. G. (2008). Impulse response processing of transient heat transfer gauge signals. *Journal of Turbomachinery*, Vol. 130.

- O'Neill, A. (1993). *Experiments on the Performance of Single and Multiple Brush Seals at Engine Representative Conditions*. PhD thesis, University of Oxford.
- Osterle, W. and Urban, I. (2006). Third body formation on brake pads and rotors. *Tribology International*, Vol. 39:pp. 401–408.
- Owen, A. K., Guo, S. M., and Jones, T. V. (2003). An experimental and theoretical study of brush seal and shaft thermal interaction. *Proceedings of ASME Turbo Expo 2003, Atlanta, Georgia, USA*, GT2003-38276.
- Patula, E. (1981). Steady-state temperature distribution in a rotating roll subjected to surface heat fluxes and convective cooling. *ASME Journal of Heat Transfer*, Vol. 103:pp. 36–41.
- Pekris, M. J. (2010). *Flow and Heat Transfer in Two Turbine Shaft Seals: The Brush and Leaf Seal*. PhD thesis, University of Oxford.
- Pekris, M. J., Franceschini, G., and Gillespie, D. R. H. (2011). Effect of geometric changes in an idealised contacting brush seal bristle pack on typical key performance measures. *Proceedings of the ASME Turbo Expo 2011, Vancouver, Canada*, GT2011-46492.
- Pekris, M. J., Franceschini, G., Owen, A. K., Jones, T. V., and Gillespie, D. (2016). Analytical modeling and experimental validation of heating at the leaf seal / rotor interface. *Proceedings of the ASME Turbo Expo 2016, Seoul, South Korea*, GT2016-57577.
- Piccini, E., Guo, S. M., and Jones, T. V. (2000). The development of a new direct-heat-flux gauge for heat-transfer facilities. *Measurement Science and Technology*, Vol. 11:pp. 342–349.
- Qiu, B. and Li, J. (2013). Numerical investigations on the heat transfer behaviour of brush seals using combined computational fluid dynamics and finite element method. *ASME Journal of Heat Transfer*, Vol. 135:pp. (122601) 1–10.

- Qiu, B., Li, J., and Feng, Z. (2015). Investigation of conjugate heat transfer in brush seals using porous media approach under local thermal non-equilibrium conditions. *Proceedings of ASME Turbo Expo 2015, Montréal, Canada*, GT2015-42550.
- Qiu, L. and Cheng, H. (1998). Temperature rise simulation of three-dimensional rough surface in mixed lubricated contact. *ASME Journal of Tribology*, Vol. 120:pp. 310–318.
- Rolls-Royce (2005). *The Jet Engine*. 5th edition. ISBN: 0902121235.
- Rolls-Royce (2016). <http://www.rolls-royce.com>. accessed 15 sept 2016.
- Ruggiero, E. J., Allen, J., and Lusted, M. (2009). Experimental testing techniques for kevlar fiber brush seals. *Proceedings of ASME Turbo Expo 2009, Orlando, Florida, USA*, GT2009-60172.
- Ryhming, I. L. (1979). On temperature and heat source distributions in sliding contact problems. *Acta Mechanica*, Vol. 32:pp. 261–274.
- Saravanamuttoo, H., Rogers, G., Cohen, H., and Straznicky, P. (2009). *Gas Turbine Theory*. Person Education Ltd., 6th edition. ISBN: 9780132224376.
- Shinohara, T., Akagi, K., Yuri, M., Toyoda, M., Ozawa, Y., Kawaguchi, A., Sakakibara, S., Yoshida, Z., Kunitake, N., Ohta, T., Nakane, H., Ito, E., Kawata, Y., and Takeshita, K. (2002). Shaft seal and turbine using the same. *United States Patent (US0006343792B1)*. Mitsubishi Heavy Industries, Ltd.
- Short, J. F., Basu, P., Datta, A., Loewenthal, R. G., and Prior, R. J. (1996). Advanced brush seal development. *AIAA Paper No. AIAA-96-2907*.
- Symm, G. (1967). Surface temperatures of two rubbing bodies. *The Quarterly Journal of Mechanics and Applied Mathematics*, Vol. 20:pp. 381–391.
- Thakare, M. R., Mason, J. F., Owen, A. K., Gillespie, D. R. H., Wilkinson, A. J., and Franceschini, G. (2016). Effect of sliding speed and counterface properties on the tribo-

- oxidation of brush seal material under dry sliding conditions. *Tribology International*, Vol. 96:pp. 373–381.
- Tudor, A. and Khonsari, M. M. (2006). Analysis of heat partitioning in wheel/rail and wheel/brake shoe friction contact: An analytical approach. *Tribology Transactions*, Vol. 49:pp. 635–642.
- Watanabe, E., Nakano, T., Tanaka, K., Tsutsumi, M., Tanaka, Y., Ohyama, H., Miyawaki, T., and Shinohara, T. (2003). Development of new high efficiency steam turbine. Technical report, Mitsubishi Heavy Industries.
- Wood, P. (1998). *An Investigation of Contact Forces, Flow, Pressure, Hysteresis and Frictional Effects in Brush Seals*. PhD thesis, University of Oxford.
- Wright, C. (2001). Resilient strip seal arrangement. *United States Patent (US0006267381B1)*. Rolls-Royce plc.
- Wu, P. and Little, W. (1984). Measurement of the heat transfer characteristics of gas flow in fine channel heat exchangers used for microminiature refrigerators. *Cryogenics*, pages pp. 415–420.

Appendix A

Risk Assessment and Standard Operating Procedure of the SSTTF

A risk assessment and a description of the standard operating procedure of the Seal Static Thermal Test Facility (SSTTF) developed during this programme is included here. The heater mesh employed to increase the temperature of the incoming flow is connected to a re-locatable power supply (*Magna-Power TSD 32-450*). The high currents required for the operation of the heater mesh involve two main risks: electrical shock and fire. Both the power supply itself and its connections to the heater mesh were properly insulated and earthed. The highest risk of fire was found to occur at the location where the power supply terminals are bolted to the heater mesh. If contact between these two elements is not good enough, an additional electrical resistance can be generated, leading to the formation of a hot spot under the high operating currents (up to 258 A). For this reason, the temperature at this critical location was assessed during commissioning using an infrared camera (see Fig A.1). A tightening torque of 45 Nm was found to be sufficient to guarantee the safe operation of the system.

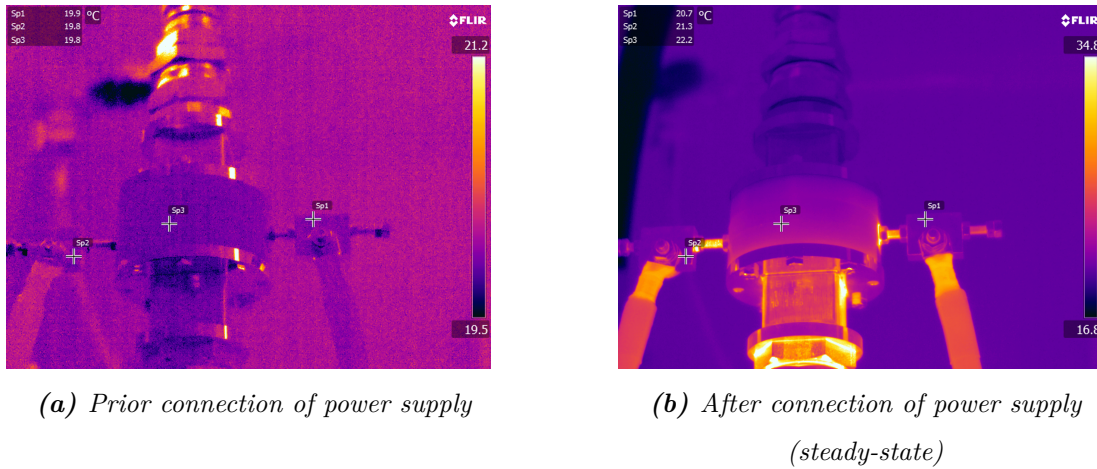


Figure A.1: Assessment of junction temperature during rig commissioning at maximum allowed current (258 A)

A.1 Standard Operating Procedure (S.O.P)

S.O.P. for connecting the DC power supply terminals to the heater mesh bus-bar

1. The connecting bolts are M10 stainless steel grade 8.8 bolts and should be tightened to a torque of 45 Nm
2. The tightening will be done using a calibrated torque wrench
3. Researcher must be present
4. The information of time, date, person, torque setting (on wrench) and witnessed by must be recorded in the test rig logbook
5. Once that the tightening is completed, connections must be appropriately covered to avoid accidental contact

S.O.P. for each experiment run

1. Obtain key to DC power supply cabinet from key safe and unlock ONLY the front door to give access to the front panel. The cabinet side panels and rear door should remain locked at all times

2. Researcher and supervisor must be present for tests and one of these must be available to disarm the power supply at all times. This person will also use a thermocouple reader to monitor the temperature in the power supply cabinet
3. Verify visually that the terminal covers for the DC connections to the heater mesh are attached correctly and securely to prevent accidental contact with the terminals
4. Verify that the last junction temperature and torque setting inspection was done less than one month ago according to the test rig logbook records
5. Open the air supply valve and establish the flow at the appropriate conditions for the experimental run
6. Test and ready the data acquisition system
7. On the front panel of the power supply, check that the voltage and current knobs are rotated fully anti-clockwise and that the PWR rocker switch is OFF
8. Verify that the 3-phase plug for the power supply is securely connected to the commando socket and switch on at the socket
9. Turn the PWR rocker switch on the front panel of the power supply to ON
10. Press the DISPLAY key on the front panel and use the voltage and current knobs to set the appropriate DC voltage and current to be supplied to the heater mesh. The current setting should not exceed 258 A, the maximum continuous operating current rating for the DC cables connected to the power supply
11. Start data acquisition
12. Press the START button on the front panel of the power supply to initiate flow heating
13. Press the STOP button on the front panel of the power supply after the appropriate length of time for the duration of the run to stop flow heating

14. Stop data acquisition
15. Continue to blow air through the heater mesh for at least 30 seconds to cool the mesh down
16. On the front panel of the power supply, rotate the Voltage and Current knobs fully anti-clockwise and turn the PWR switch to OFF
17. Close the air supply valve
18. Turn off the 3-phase commando socket
19. Close and lock the cabinet containing the power supply
20. Return key to key safe
21. Manually inspect the cabinet and area around mesh heater connections for any signs of hot spots or overheating before leaving the test area
22. In the event of any indication of overheating, mark the cabinet for non-use and inform the Osney lab electrician so that the system can be disabled prior to investigation

A.2 Risk Assessment

Table A.1 summarizes the risk assessment carried out for the SSTTF.

Hazard	Person at risk	Risk controls inplace	Further action necessary to control risk
Trailing power cables. People could be tripped up	All persons	Length of input 3-phase power leads minimised. Ensure that leads do not cross walkways where possible and are covered if unavoidable.	None.
Risk of electric shock from high current power to heaters	Researcher	Researcher trained in proper use of power supply. Power supply and connections are housed in locked cabinet, with largest holes being 4mm diameter for ventilation. Access to cabinet controlled by key and held by supervisor. Earth leads installed from cabinet to power supply chassis. Connections at heater mesh to be appropriately covered to avoid accidental contact. Power supply terminals and operator located in different rooms during tests. Thermocouple in interior of cabinet monitored during testing.	None.
Unauthorised operation	All persons	Front of control cabinet is lockable and keys held by supervisor, only to be issued to trained operators.	None.
Risk of high temperatures from high current power to heaters	Researcher	DC Cables appropriately rated for continuous operation, in free air or touching ground, in accordance with IET Wiring Regulations 17th Edition, 95 mm H07RN-F 95mm ² double insulated rubber cable, allowing maximum continuous current of 258 A from power supply. No run should take longer than 15 minutes. Appropriately trained electrician should make connections at both power supply and heater mesh and tightening torques to be recorded, as detailed in the Standard Operating Procedure. All flammable material to be kept clear of power supply cabinet and test rig.	Steady-state current must be limited to 258 A. Control unit of power supply can enable this. Researcher to monitor junction temperature during commissioning. Thereafter, monitoring of the junction temperature and inspection of the bolt torques to be performed monthly. The inspection dates will be recorded in the test rig logbook. The terminal bolts must also be checked after any period of non-use (i.e. decommissioned) before the rig is run again and also when any changes are made. Terminal temperature to be assessed during commissioning.

Hazard	Person at risk	Risk controls inplace	Further action necessary to control risk
Risk of overheating heater mesh	Researcher	Voltage should be set to an appropriate level prior to experiment to ensure that overheating does not occur. Researcher should be with experiment whilst the heater mesh is powered. Pressure/flow switch to be installed to allow activation of power supply to experiment. This can be controlled via LabVIEW software. For commissioning a manual control system must be used.	Researcher must ensure that the switch activation voltage is set to an appropriate level for the experiment.
Incorrect operation of test rig	Researcher	A Standard Operating Procedure has been written and is available for the rig. A hard copy of the SOP will be located by the rig and must be followed for every test run. In the event of changes to the rig setup or experimental process, the SOP must be updated to reflect the appropriate new test procedure.	None.

Table A.1: Risk assessment of re-locatable heater mesh power supply, Magna-Power TSD 32-450