

Heat Transfer for Fusion Power Plant Divertors



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Abstract

Exhausting the thermal power from a fusion tokamak is a critical engineering challenge. The life of components designed for these conditions has a strong influence on the availability of the machine. For a fusion power plant this dependence becomes increasingly important, as it will influence the cost of electricity.

The most extreme thermal loading for a fusion power plant will occur in the divertor region, where components will be expected to survive heat fluxes in excess of 10 MW/m^2 over a number of years. This research focussed on the development of a heat sink module for operation under such conditions, drawing on advanced cooling strategies from the aerospace industry.

A reference concept was developed using conjugate **Computational Fluid Dynamics**. The results were experimentally validated by matching Reynolds numbers on a scaled model. Heat transfer data was captured using a transient thermochromic liquid crystal technique. The results showed excellent agreement with the corresponding numerical simulations.

To facilitate comparison against other divertor heat sink proposals, a non-dimensional figure of merit for cooling performance was developed. When plotted against a non-dimensional mass flow rate, the reference heat sink was shown to have superior cooling performance to all other divertor proposals to date.

Results from **Finite Element Analysis** were used in conjunction with the **ITER** structural design criteria to life the heat sink. The sensitivity of life to both

boundary conditions, and local geometric features, were explored. The reference design was shown to be capable of exceeding the life requirements for heat fluxes in excess of 15 MW/m^2 .

A number of heat sinks, based on the reference design, were fabricated. These underwent non-destructive testing, before experimentation in a high-heat flux facility developed by the author. The heat transfer performance of the tested modules was found to exceed that predicted by numerical modelling, which was concluded to be caused by the fabrication processes used.

To my parents:
for their unwavering support and encouragement.

To Deb:
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To family departed:
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Nomenclature

Acronyms

APS	Air Plasma Spray
BoP	Balance of Plant
BPP	Basic Performance Phase
BS	British Standards
BSL	Baseline
BSL-EARSM	Baseline Explicit Algebraic Reynolds Stress Model
CAD	Computer Aided Design
CCC	Committee on Climate Change
CCFE	Culham Centre for Fusion Energy
CFD	Computational Fluid Dynamics
CFETR	China Fusion Engineering Test Reactor
CHF	Critical Heat Flux
CMOS	Complementary Metal-Oxide-Semiconductor
CPS	Capillary-Pore System
CVD	Chemical Vapour Deposition
CWHF	Critical Wall Heat Flux
DAQ	Data Acquisition System
DBTT	Ductile to Brittle Transition Temperature
DEMO	DEMONstration power plant
DIC	Digital Image Correlation
DN	Double Null
dpa	displacements per atom
DS	Dispersion Strengthened
EDM	Electrical Discharge Machining
EIA	Energy Information Administration
ELM	Edge Localised Mode
EPC	Engineered Performance Coatings
EU	European Union

FEA	Finite Element Analysis
FGM	Functionally Graded Material
fpv	full-power year
GPR	Gaussian Process Regression
HEMJ	HElium-cooled Modular divertor with Jet cooling
HEMP	HElium-cooled Modular divertor with integrated Pin array
HETS	High Efficiency Thermal Shield
HHF	High-Heat Flux
HIP	Hot Isostatic Pressing
HPJC	High Pressure Jet Cascade
HVOF	High Velocity Oxide Fuel
IACS	International Annealed Copper Standard
IAPWS	International Association for the Properties of Water and Steam
ICF	Inertial Confinement Fusion
ICHF	Incident Critical Heat Flux
IFMIF	International Fusion Material Irradiation Facility
IR	Infra-Red
ITB	Internal Transport Barrier
ITER	International Thermonuclear Experimental Reactor
IVT	Inner Vertical Target
JAERI	Japanese Atomic Energy Research Institute
JET	Joint European Torus
KINH	KINematic Hardening
KIT	Karlsruhe Institute of Technology
LES	Large Eddy Simulation
LiMIT	Lithium-Metal Infused Trenches
LLNL	Lawrence Livermore National Lab
MCF	Magnetic Confinement Fusion
MIM	Metal Injection Moulding
NASA	National Aeronautics and Space Administration
NDT	Non-Destructive Testing
NIST	National Institute for Standards and Technology
NPL	National Physical Laboratory
NTU	Number of Transfer Units
OAR	Open Area Ratio
ODS	Oxide Dispersion Strengthened
OECD	Organisation for Economic Cooperation and Development
OFHC	Oxygen Free High Conductivity
OMCS	Oxford Materials Characterisation Service
ORNL	Oak Ridge National Laboratory
OTFL	Osney Thermo-Fluids Laboratory

OVT	Outer Vertical Target
PAL	Plasma Arc Lamp
PCHE	Printed Circuit Heat Exchanger
PCM	Photo-Chemical Machining
PD	Positive Displacement
PFC	Plasma Facing Component
PFIT	Porous Foam in Tube
PFU	Plasma Facing Unit
PLC	Programmable Logic Controller
PPCS	Power Plant Conceptual Studies
PPI	Pores per Inch
PVD	Physical Vapour Deposition
PWI	Plasma Wall Interactions
PWR	Pressurised Water Reactor
RANS	Reynolds Averaged Navier Stokes
RE	Runaway Electrons
SDC-IC	Structural Design Criteria for ITER In-vessel Components
SEM	Scanning Electron Microscope
SMD	Sauter Mean Diameter
SN	Single Null
SNL	Sandia National Laboratory
SOL	Scrape Off Layer
SST	Shear Stress Transport
TBC	Turbine Blade Coating
TRL	Technology Readiness Level
VDE	Vertical Displacement Event
VPS	Vacuum Plasma Spray
VT TJ	Vacuum Tight Threaded Junction

Symbols

A	Input area for heat flux, [m ²]
Bi	Biot number, $\frac{h_c t}{k_s}$
C	Constant in Chaboche KINH model, [MPa]
D	Jet diameter, [m]
Eu	Euler number, $\frac{\Delta P}{\rho_f V^2}$
F	Peak stress, [MPa]
H	Jet entry length, [m]
L	Length across triangular cavity median, [m]
Nu	Nusselt number, $\frac{h_c D}{k_f}$
P	Primary stress, [MPa]
Pr	Prandtl number, $\frac{c_{p_f} \mu_f}{k_f}$

Q	Secondary stress, [MPa]
R	Resistance, [$\text{K m}^2 \text{W}^{-1}$]
Re	Reynolds number, $\frac{\rho_f V D}{\mu_f}$
S	Gaussian heat flux width, [m]
St	Stanton number, $\frac{Nu}{Re \cdot Pr}$
T	Temperature, [$^{\circ}\text{C}$]
V	Volume, [m^3]
Z	Jet impingement plate spacing, [m]
$\mathbf{s}$	Deviatoric stress tensor, [MPa]
$\dot{Q}$	Power, [W]
$\dot{m}$	Mass flow rate, [kg s^{-1}]
$\dot{q}$	Internal heat generation, [W/m^3]
c	Specific heat capacity, [$\text{J kg}^{-1} \text{K}^{-1}$]
d	Mesh length scale, [m]
f	Reserve factor
g	Magnetic flux expansion
h	Heat transfer coefficient, [$\text{W m}^{-2} \text{K}^{-1}$]
j_H	Chilton j-factor, $\frac{Nu}{Re \cdot Pr^{\frac{1}{3}}}$
k	Thermal conductivity, [$\text{W m}^{-1} \text{K}^{-1}$]
m^*	Non-dimensional mass flow rate, $\frac{\dot{m} c_p}{k_s \sqrt{A}}$
q''	Heat flux, [W/m^2]
r	Fillet radius, [m]
t	Material length scale, [m]
v	Fluid velocity, [m s^{-1}]
v^*	Friction velocity at wall, $\sqrt{\frac{\tau_w}{\rho}}$
a	Plasma minor radius, [m]
R	Plasma major radius, [m]

Subscripts

L	Membrane, when in combination with bending
T	Thermal
$avg.$	Average
b	Bending
c	Convective
ce	Coolant exit
cf	Film
ci	Coolant inlet
eff	Effective
f	Fluid
g	Gas
i	i^{th} component
j	Jet

<i>m</i>	Membrane
<i>norm.</i>	Normalised
<i>p</i>	Constant pressure
<i>pl</i>	Plastic
<i>por.</i>	Porosity
<i>q</i>	Heat transfer
<i>ref.</i>	Reference
<i>s</i>	Solid
<i>sep</i>	Separatrix
<i>t</i>	Thermal
<i>tot</i>	Total
<i>w</i>	Internal wall
<i>wet</i>	Internal wetted area
<i>y</i>	Yield

Greek

Λ	Accumulated plastic strain
β	Back stress tensor, [MPa]
χ	Uncertainty analysis variable
δ	Gaussian width parameter
ϵ	Percentage uncertainty
γ	Rate of decrease of hardening modulus, [GPa]
λ	Plasma length scale, [m]
μ	Dynamic viscosity, [Pa s]
ν	Kinematic viscosity, [m ² s ⁻¹]
ω	Cascade mass flow rate ratio, $\frac{\dot{m}_1}{\dot{m}_n}$
ρ	Density, [kg m ⁻³]
σ	Stress, [MPa]
τ	Non-dimensional overheat parameter, $\frac{T-T_{ci}}{T_{ce}-T_{ci}}$

Chapter 1

Introduction

“Never express yourself more clearly than you are able to think.”

— Neils Bohr

This chapter begins with a brief overview of the fundamentals of nuclear fusion, and the criteria that need to be met to generate net power from it. One particular approach to achieving this, called the tokamak, is then discussed in more detail. Finally, the critical issue of thermal power exhaust in such machines is considered.

1.1 The Fundamentals of Nuclear Fusion Power

The mass of a nucleus is always lower than the sum of the individual nucleons that constitute it^I. This mass difference is called the binding energy of the nucleus. In Fig.1.1 the binding energy per nucleon is plotted against the mass number.

A nucleus becomes more stable as its binding energy per nucleon is increased. The shape of this curve can be exploited in two ways to convert nuclear potential, into kinetic, energy: by splitting heavy, or fusing light, nuclei [81, Chpt.3]. This former process is known as nuclear fission, whilst the latter as nuclear fusion.

For light nuclei to fuse they must have sufficient kinetic energy to overcome the Coulomb barrier: the energy associated with the electric repulsion between positively charged nuclei. This requires temperatures of the order of 10 MK for isotopes of hydrogen [263, Chpt. 1].

^I ^1H being the exception.

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Figure 1.1: Binding energy per nucleon [MeV] plotted against the atomic mass number [200].

At such temperatures the fuel becomes a fully ionized gas called a plasma. In contrast to the collisional nature of a gas, the motion of a fusion plasma is dictated by long range electromagnetic forces [81, Chpt. 6].

A 50:50 fuel mixture of deuterium and tritium (D-T) will be used in a near term fusion reactor. This is because it has the highest average reactivity² at temperatures accessible using current technology. The reactivities of a number of other potential fusion fuels are shown in image (b) of Fig. 1.2.

The by-products of a D-T fusion reaction are helium-4 (alpha) and a neutron - see image (a) of Fig. 1.2. Severe damage to structural materials is caused by the neutrons. When compared to a fission reactor core, the irradiation doses are ~ 5 times greater [310].

Achieving net power from fusion involves being able to confine the plasma for long enough

²The average product of the fusion reaction cross-section and relative velocities, denoted $\langle \sigma v \rangle$ [81, Chpt.3].

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Figure 1.2: (a) D-T fusion reaction and associated energies of by-products. (b) Average fusion reactivity against ion temperature for a range of possible fusion fuels [1 keV $\rightarrow$ 11.6 MK].

at the required thermal conditions. In written form this necessitates [263, Chpt.1]

$$\text{fusion heating} + \text{external heating} \geq \text{radiation loss} + \text{transport loss}.$$

The ratio of output fusion power³ to external heating is called the amplification factor, and designated Q . A value of $Q > 1$ is required for a fusion reactor to ‘break-even.’ When $Q = \infty$ the heating power needed to sustain the fusion reaction is supplied solely by the alpha particles. At this point the plasma is said to be ignited. Lawson [155] set down a limiting criteria for ignition which, for a magnetically confined D-T plasma, is given by the inequality [65],

$$nT\tau_E > 3 \times 10^{21} \text{ m}^{-3}\text{keVs}.$$

Here τ_E is the plasma confinement time related to transport losses, n the ion density, and T the ion temperature. The product $nT\tau_E$ is referred to as the fusion triple product. For

³This includes both the plasma heating component from the alpha particles, and the neutrons.

typical ion densities of $10^{19} - 10^{20} \text{ m}^{-3}$, and ion temperatures of $\sim 10 \text{ keV}$, confinement times in the range of $\sim 1\text{-}10 \text{ sec}$ are required [293, Chpt.1].

1.2 The Tokamak

Though other approaches do exist [98, 141, 181, 254], current research in fusion power is dominated by just two: Inertial Confinement Fusion (ICF) and Magnetic Confinement Fusion (MCF). The former generates dense plasmas, for short time scales, to achieve high fusion triple products. In contrast, MCF uses magnetic fields to confine a relatively diffuse plasma over longer time scales.

The magnetic field configurations used in MCF generally fall into two categories: 1) open - mirror-type; 2) closed - toroidal-type. Mirror-type schemes operate by forming a magnetic bottle. As charged particles approach the high field regions their velocity parallel to the field lines is converted to angular momentum, and they are reflected back into the core. However, this reflection only occurs for certain ratios of velocity parallel and perpendicular to the field lines⁴. Though enhancements were made to try and reduce the amount of escaped particles, the superior performance achieved by closed concepts around the same time period resulted in them falling out of favour. An example of one of the most advanced mirror-type devices constructed is shown in Fig.1.3.

Employing a toroidal magnetic geometry addresses the issue of end losses found in open configurations. However, the curvature of the field leads to a non-zero radial gradient, which causes ions and electrons to drift from their orbits. This generates an electric field in the plasma due to the charge dependence of the drift motion. The cross product of this electric field with the applied toroidal magnetic field results in an outward radial force on the charged particles, inhibiting confinement. To cancel this radial drift towards the vessel wall, a poloidal field is applied [263, Chpt.4]. The resulting combined magnetic field con-

⁴The particular particle velocity perpendicular to the field line ($v_{\perp}$) is given by $v_{\perp}(s_0) = \pm \left(\frac{B_{max}}{B_{min}} - 1 \right)^{-\frac{1}{2}} v_{\parallel}(s_0)$ [263, Chpt.4]. Here B_{max} and B_{min} are the maximum and minimum magnetic field strengths, respectively. The quantity $v_{\parallel}$ is the velocity parallel to the field line.

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Figure 1.3: MFTF-B advanced mirror-type fusion device constructed at the Lawrence Livermore National Lab (LLNL) in the mid 1980s [93]

figuration results in particles following helical orbits around the toroid such that the $E \times B$ drifts cancel out over a full poloidal rotation.

1.2.1 Key Characteristics

The most advanced toroidal MCF concept is the tokamak, which was invented in the Soviet Union during the 1950s [293, Chpt.1]. It employs a number of coils around the torus to generate a toroidal magnetic field at the plasma centre of the order of 1 T [263, Chpt. 1]. The poloidal field is generated by passing a current through the plasma using a transformer action. A cross-section of the plasma geometry (a), and a typical configuration (b), for this device are shown in Fig.1.4

The ratio of the plasma minor radius, a , to the toroid major radius, R , is termed the aspect ratio. This value plays an important role in determining the operational characteristics of the machine. Under standard physics and engineering constraints a value of 2-3 is found to be optimum for instances where superconducting magnetic field coils are used [294].

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Figure 1.4: (a) Cross-section of tokamak plasma showing typical geometric parameters. (b) Tokamak magnetic coil architecture and plasma streamlines, credit: Max-Planck Institute for Plasma Physics.

The transformer action used to generate the poloidal magnetic field also has the secondary benefit of resistively heating the plasma. However, as the plasma resistivity decreases with temperature, ohmic heating alone is generally not sufficient to achieve the temperatures required. Therefore, additional heating methods such as neutral beams or Radio Frequency waves are often used [263, Chpt.5].

1.2.2 Progress to Date & Future Machines

There are currently 23 tokamaks [124] operating worldwide, researching both the plasma physics and engineering aspects of fusion. Of these, the Joint European Torus (JET) at Culham, UK, is the largest by vacuum vessel volume. This machine has broken new ground, and set a number of fusion records, during its operational life: (1) the world's first controlled release of fusion power [227]; (2) currently the only tokamak to operate with D-T [138]; (3) the highest amplification factor (0.65) [138]; (4) the largest fusion power output (16.1 MW) [138].

From the late 1960s to the early 2000s the fusion triple product achieved in tokamaks has followed a Moore's law⁵ type improvement [117]. This is evidenced in Fig. 1.5, where the fusion triple product is plotted for a range of tokamaks during different experimental campaigns.

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Figure 1.5: Improvement in fusion triple product achieved by a range of tokamaks over the period 1965-1990s. Note that the majority of the results are with D-D fuel [65].

In 1985, the idea of building a tokamak to achieve break-even was set in motion. By 2001 a final design for the machine, called the International Thermonuclear Experimental Reactor (ITER), had been approved. It is the biggest science experiment ever attempted on Earth, involving a host of nations: US, EU, Japan, Russia, China, India and South Korea. A specification for the machine is shown in Fig. 1.6

Construction of ITER began in Cadarache, France, in 2010. Images of the construction progress to date are shown in Fig. 1.7. When completed, it is expected to produce D-T plasmas for ~ 400 sec at an amplification factor ≥ 10 . The first plasma is expected in 2025 [125].

⁵The number of transistors that can be placed on an integrated circuit will double each year [188].

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Figure 1.6: Plasma parameters and geometry for ITER [185].

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Figure 1.7: The ITER program from top right to bottom left: nations involved; site in Cadarache; torus hall; artists conception of completed machine [49, 184, 208].

ITER is a science experiment, with the primary purpose of showing that the tokamak concept is capable of producing net power. In progressing to a power plant there are a number of critical engineering issues that need to be addressed: material irradiation damage; a closed tritium cycle; remote handling; thermal power exhaust [236, 292].

Within the [EU](#) fusion program, a near-term [DEMONstration power plant](#) ([DEMO](#)) will succeed [ITER](#) in the late 2030s. It will operate with pulse lengths in excess of 2 hours, and supply several 100s of MWe to the grid [\[267, 292\]](#). Its purpose will be to demonstrate all of the required technologies for a fusion power plant, so that wide spread deployment could occur during the second half of the 21st century. A comparison between the design parameters for [ITER](#) and two potential [DEMO](#) machines are shown in [Fig.1.8](#).

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Figure 1.8: A comparison of the design parameters for a pulsed ([DEMO1](#)), and steady-state ([DEMO2](#)), [DEMO](#) with the [ITER](#) baseline [\[78\]](#).

1.3 Thermal Power Exhaust in a Tokamak

The cost of electricity for a tokamak fusion power plant will be dominated by capital costs [\[289\]](#). This influence can be diluted by achieving long component lifetimes. As the heat exhaust system experiences some of the most extreme operating conditions within the machine, developing robust solutions for this region is regarded as critical to the realisation of commercial fusion power [\[236\]](#).

1.3.1 Plasma Energy Loss Mechanisms

There are three sources of thermal power loss from a fusion plasma: mass flow (direct loss of kinetic energy), due to neutrons and charge exchange particles; radiation; diffusive, via

conductive and convective heat transport [104]. The different routes of heat flow from the plasma core to vessel wall are summarised in Fig.1.9.

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Figure 1.9: Breakdown of power input and loss mechanisms in a fusion plasma, together with the regions in which they occur. Adapted from [251].

The first two of these mechanisms distribute their power approximately uniformly over the inner surface of the vacuum vessel. The high kinetic energy of the neutrons, combined with their zero charge, means that their power deposition becomes a volume heating source, rather than a surface flux [251].

Heat transport in the plasma edge region depends on four key factors [81, Chpt.14]: core plasma density; operational mode; Edge Localised Modes (ELMs); internal transport barriers. The dominant transport mode, which determines the confinement time, is conduction [81, Chpt.14].

There are two distinct operational modes in a tokamak plasma where external heating power is applied: Low confinement (L-mode) and High confinement (H-mode). Operating in the latter mode leads to an increase in the confinement time of ~ 2 [263, Chpt.4]. The mode in which the plasma operates depends upon the amount of non-radiative power cross-

ing the last closed magnetic flux surface (separatrix), and the plasma-to-surface contact [81, Chpt.14]. Direct impingement of the particle flux on the vessel wall leads to severe material erosion, and subsequent contamination of the plasma. To avoid this operating scenario, two widely used plasma-to-surface contact methods are employed: a limiter, or a divertor. A limiter is a surface extruding from the vessel wall into the plasma that acts to break the field lines. As particles diffuse radially from the core region they eventually pass the separatrix and are deposited onto the limiter.

To operate with a divertor, additional coils are used to create a null point in the poloidal magnetic field close to the plasma edge - see Fig.1.10. Particles that are transported across the separatrix are then swept down into the divertor region. A major benefit of this approach over the limiter is that impurities in the core plasma are minimised. The improvements in confinement time that have been achieved by tokamaks operating with a divertor has led to its widespread adoption.

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Figure 1.10: Schematic of limiter [left] and divertor [right] magnetic geometries [81].

In a H-mode plasma turbulence at the edge is suppressed, forming an effective barrier to heat transport [266]. Associated with H-mode is the build-up of density in the region between the separatrix and the vessel wall, which is called the Scrape Off Layer (SOL) [81,

[104], [144]. Left unchecked, this would lead to a disruption. The plasma self-regulates itself to avoid this through **ELMs**. These are localised instabilities in the plasma edge that act to discharge particles and energy akin to a pressure relief valve [81, Chpt.14]. If the correct combination of amplitude and frequency for the **ELMs** is achieved, the average density in the **SOL** can be maximised. This leads to the greatest enhancement of the confinement time. In **ITER**, **ELMs** will release ~ 20 MJ of energy into the **SOL** at frequencies of 1-2 Hz during normal operation [213].

Large-scale plasma instabilities can lead to the loss of magnetic confinement. For large machines, such as **ITER**, the subsequent release of stored energy can result in high transient heat, and mechanical, loads on components in contact with the plasma. As an example, large amplitude **ELMs** in **ITER** will produce heat fluxes of $0.6\text{--}3.5$ GW/m², on time scales of milliseconds [35]. Such power loading levels lead to localised melting of plasma contacting surfaces, which enhances the erosion rate and dilutes the plasma [33]. In Fig.1.11 the impact of various transient thermal loads on two candidate plasma contacting materials is shown.

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Figure 1.11: The response of tungsten and beryllium to different transient thermal loads [282].

There are a number of other plasma disruptions that can occur: Runaway Electrons; Vertical Displacement Events; thermal and current quenches [38]. **ITER** will employ a number of strategies to mitigate the impact of these events, and prolong the lifetime of components. The basic approach is to enhance the level of radiation from the plasma by increasing the

impurity level using gas or pellet injection [156].

1.3.2 The ITER Divertor

A number of tokamaks worldwide now operate with a divertor: Alcator C-mod, ASDEX-U, DIII-D, JET, JT-60U and JFT-2M [163]. However, only Tore Supra currently has an actively cooled divertor to allow longer pulse durations.

ITER's actively cooled divertor will be composed of 54 cassettes, each weighing approximately 9 tonnes, and consisting of three Plasma Facing Components (PFCs): the Inner Vertical Target (IVT) and Outer Vertical Target (OVT), together with the dome [213]. The cassettes will be mounted ~ 20 mm apart on a toroidal rail that allows the number of access ports for remote maintenance to be minimised [40, 143]. Cryogenic pumps located in the shielded region under the dome operate at flow rates of ~ 170 m³/s to maintain the He concentration in the core below 10% during a D-T pulse [164]. A schematic of a cassette, and their location within the tokamak, are shown in Fig. 1.12.

The body of each cassette is constructed from 316LN stainless steel, onto which the PFCs are mounted. They consist of a copper alloy heat sink bonded to a tungsten armour tile. Tungsten was chosen as the plasma surface contact material due to its high sputtering threshold⁶ and low tritium inventory [173, 238].

The PFCs for each cassette have a total plasma wetted area of ~ 2.8 m²; composed of 72 Plasma Facing Units (PFUs) and 60,000 tungsten monoblocks/flat tiles [213]. To further reduce electromagnetic loading from eddy and halo currents during disruptions the vertical targets are split in two poloidally [40]. The PFUs themselves are separated by gaps of ~ 0.5 mm due to manufacturing tolerances [40].

Cooling water is fed to each cassette at 140°C and a pressure of 4 MPa [222]. Under the ITER baseline operating scenario this coolant will be supplied to each cassette at a rate

⁶220 eV under Deuterium ion bombardment [173]

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Figure 1.12: (a) ITER divertor cassette construction and size. (b) Placement of cassettes within the ITER tokamak [72].

of ~ 30 kg/s [224]. The resulting temperature rise will be 25-30°C. A cross-section through the cassette showing the manifold structure, and a breakdown of the mass flow distribution, are shown in Fig. 1.13.

The thermal power loading of the divertor is comprised of both volume and surface heating terms. As the divertor covers approximately 22% [213, 268] of the ITER vacuum vessel, the estimated volume heating from neutrons is ~ 90 MW - see Fig. 1.6. Surface heating is principally determined by the power entering the SOL from the plasma core [264]. For the ITER baseline operating scenario this will be ~ 100 MW, including both alpha, and auxiliary heating, power components [213, 268]. Of this, 50-75% will fall on the OVT [213].

The IVT and OVT are themselves comprised of two regions: the (lower) straight target plate and the (upper) curved baffle [40]. Where the separatrix contacts the target plates is known as the strike point. As the magnetic flux over the target plates is non-uniform, so too is the heat flux profile.

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Figure 1.13: (a) Cross-section of ITER cassette showing coolant manifolding. (b) Mass flow rate breakdown to different segments of the divertor cassette, along with coolant temperature changes [222].

In the SOL heat transport is assumed to take the form of an exponential radial decay with an e-folding length, λ_q [61]. In recent research by [61, 291] it has been shown that the divertor target plate profile is more accurately represented if the exponential decay is convoluted with a Gaussian of width S . This leads to a heat flux profile on the target plate of the form

$$q''(\bar{s}) \propto \exp\left(\left(\frac{S}{2\lambda_q}\right)^2 - \frac{\bar{s}}{\lambda_q \cdot g}\right) \cdot \operatorname{erfc}\left(\left(\frac{S}{2\lambda_q}\right)^2 - \frac{\bar{s}}{S \cdot g}\right), \quad (1.1)$$

where $\bar{s} = (R_{sep} - R)g$ and g is a measure of the magnetic flux expansion. Heat flux profiles based on this formulation have been fitted to experimental data from a range of machines in Fig. 1.14 by [61].

Heat fluxes along open field lines in ITER will be up to 1 GW/m² [104]. This is beyond the

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Figure 1.14: [Left] Typical poloidal cross-section of tokamak with the region on which equation (1.1) is valid highlighted [224]. [Right] Experimental, and predicted, heat flux profiles on of a number of tokamaks used in the study by [61].

power handling capability of solid divertor target components. To reduce the loading the angle of incidence between the open field lines and the target plates is minimised, which increases the plasma wetted area. In practice, this is limited to values $>1^\circ$ due to the manufacturing tolerances of the divertor target plate components [104]. Taking ITER as an example, the maximum allowed cumulative misalignment between adjacent components is 0.4 mm, over a toroidal gap of 1 mm [213]. This ensures any edges exposed to the full parallel heat flux do not melt.

Injecting gases into the divertor region will also be used in ITER to radiate the power from the SOL more evenly over the target plates. This is most effective when the divertor plasma is operated in a high density and low temperature regime [163]. To achieve this a closed divertor is utilised, whereby the escape fraction for recycled neutrals⁷ is kept small [285].

Even with these schemes to reduce non-uniform power loading, the quasi steady-state heat fluxes near the strike point on the ITER outer target plate will be 10 MW/m^2 [109]. During slow transients (seconds) this will increase to 20 MW/m^2 [222]. The minimum life require-

⁷Ions that become neutralised in the divertor region and migrate back towards the plasma core.

ments for these components has been set as follows: 5000 cycles at quasi steady-state levels; 300 cycles for transients [107].

1.4 Summary of Research Achievements

The research reported in this thesis has studied an essential technology required for the successful generation of power from nuclear fusion. At the time of writing, a viable active cooling system for the divertor region of a future fusion power plant has yet to be developed. The present work has analysed the thermal boundary conditions for this region, where the heat fluxes within the machine are a maximum, and researched in detail convective cooling technologies, derived from those used for turbine blades in jet engines, as potential solutions. The research has succeeded in introducing a new cooling performance parameter that allows cooling approaches to be ranked. A heat sink module has also been developed using a combination of experimental and numerical methods. Fabricated versions of this heat sink have been tested in a facility designed and developed by the author. The research results show that the developed heat sink module has superior cooling performance to all proposals put forward to date.

Chapter 2

Literature Review

“There are two kinds of people in the world: Johnny Von Neumann, and the rest of us.”

— Eugene Wigner

The majority of tokamak research machines do not employ active convective cooling for in-vessel components. This limits their peak pulse times to ~ 5 s [104], for heat fluxes in the region of 10 MW/m^2 . Only Tore Supra [46] incorporated active cooling elements from its inception. As part of the WEST project [66], this machine now lies at the heart of the EU research effort on the development of High-Heat Flux (HHF) components for ITER.

During the ITER design phase, a range of heat sink components were explored for use in the divertor. Water was chosen as the coolant based on the operational experience from Pressurised Water Reactors (PWRs), and its capacity for removing high heat loads and fluxes. This development process for ITER highlighted the large number of difficulties associated with HHF component design, and a program for future machines has been in place ever since.

In this chapter the requirements for a DEMO divertor target plate are first introduced. A range of past and present concepts that have been proposed for this role are then assessed. Finally, the aims of the research are presented.

2.1 DEMO Target Plate Design Criteria

The total thermal power loading of a near-term **EU DEMO** divertor is calculated to be 260 MW, ~43% of which will be deposited on the **PFU**'s surfaces [300]. The resultant peak heat fluxes in the strike point region will be similar to those in **ITER** [299].

Neutron damage, measured in **displacements per atom (dpa)**[†], will be an order of magnitude greater than in **ITER** [266], which is driven by the longer component life requirements: two **full-power years (fpys)**. This eliminates the viability of some materials, whilst reducing the operational temperature windows of others.

As **DEMO** is intended to be a stepping stone towards a fusion power plant, consideration also needs to be given to thermal cycle efficiency [78]. This was not a requirement for **ITER**, or previous machines, where being able to exhaust the heat load was the primary concern. In the literature, maintaining the coolant pumping power below 10% of the incident heat load has been used as a design selection criterion in this regard [274].

In Table 2.1 the requirements for the outer target plate for **ITER** and **DEMO** are compared. The higher thermal power loading in **DEMO** is a result of the increased plasma volume. It is worth noting that the total number of thermal cycles is of similar magnitude, which implies that the increased irradiation damage is driven by the longer fusion pulses in **DEMO**.

2.2 DEMO Divertor Target Candidates

The current options for a **DEMO** divertor target component can be classified according to the flow type: internal channel; jet impingement; porous; capillary; external channel. Table 2.2 provides a summary of the literature survey using these designations. Specific designs utilising each of these flow types is discussed in more detail in what follows.

[†]The average number of times an atom in the lattice is displaced, which depends on the neutron fluence.

	ITER	DEMO
Total Power [MW]		
- Volume	89	147
- Surface	100	112
Peak Heat Flux [MW/m ²]		
- Steady-State	10	10
- Transient	20	17-21
Life (cycles)		
- Steady-State	5000	5000-10000
- Transient	300	-
Damage (dpa)		
- Tungsten	0.3-0.5	2.8-6
- Copper	0.6-1	6.5-13

Table 2.1: Comparison of divertor outer vertical target requirements for ITER and a near-term DEMO [21, 78, 107, 266, 267, 297, 299, 300].

Flow Type	Heat Transfer Mode(s)	Heat Transfer Enhancement(s)	Coolant(s)	Reference(s)
Internal	Forced Convection, Forced Convection Boiling	Swirl Tape/Rod, Screw Tube, Roughened Wall, HyperVapotron	Water, Helium	[23, 25, 41, 57, 70, 71, 88, 90, 221, 222, 284, 287, 306]
Jet Impingement	Forced Convection, Forced Convection Boiling	Roughness	Water, Helium	[16, 115, 121, 137, 147, 194, 197, 215, 230, 286, 288]
Porous	Forced Convection	-	Helium	[55, 103, 140, 219, 301]
Capillary	Pool Boiling	-	Sodium, Lithium	[69, 172, 183, 237]
External	-	-	Lithium	[243]

Table 2.2: Summary of HHE cooling methods that have been proposed for operation in the divertor region.

2.2.1 Internal Channel Flow

Internal channel flow concepts have been extensively researched for HHE applications by the fusion community. For water cooled variants the focus has been on using sub-cooled nucleate boiling as the heat transfer mode. Consequently, heat transfer enhancement has revolved around increasing the Critical Heat Flux (CHF) over that of a smooth tube. As

the majority of the research in this area was performed as part of the development of the **ITER** divertor target plate, these designs form the focus of this section.

The phenomena of **CHF** occurs when vapor escape, or generation, near the heated surface during nucleate boiling is inhibited [120, Chpt. 10]. This produces vapor columns and slugs that block the liquid from wetting the surface. The vapor blanket formed at the wall due to this mechanism leads to a sharp decrease in the heat transfer coefficient, which results in a sharp increase in the wall temperature. Components that operate with two-phase flows near the **CHF** run the risk of catastrophic failure due to the increased wall operating temperature.

2.2.1.1 Swirl Tape/Rod

The final design selected for the **ITER** divertor target is comprised of a CuCrZr pipe ($\varnothing 12 \times 1$ mm) bonded to a tungsten monoblock via an **Oxygen Free High Conductivity (OFHC)** copper interlayer (0.5 mm) [284]. The monoblock itself has dimensions of 28.5 mm x 25 mm x 12 mm. Within the pipe, a swirl tape is used to enhance the **CHF** over the smooth tube case. This has a twist ratio² of two and is 0.8 mm thick. The tape reduces the effective hydraulic diameter of the pipe, inducing mixing of the bulk and near wall flows, leading to a reduction in the convective boundary layer thickness [25]. In Fig. 2.1 the geometry of the **ITER** design is shown, along with a target plate prototype and alternative tile concepts.

The inclusion of a swirl tape with a twist ratio of two doubles the **CHF** but results in a four-fold increase in the pressure drop [23, 222]. As the twist ratio of the swirl tape is reduced, both the pressure drop and **CHF** increase [222].

Small scale **ITER** prototypes were experimentally tested by **Japanese Atomic Energy Research Institute (JAERI)** in a particle beam facility [88]. Nominal **ITER** inlet conditions were used: specifically a flow velocity of 12 m/s; a coolant pressure of 3.3 MPa; 120°C of

²The number of pipe diameters travelled axially required to achieve a 180° twist in the tape.

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Figure 2.1: (a) Schematic of the ITER divertor geometry [51]. (b) Large scale prototype for vertical target plate [234]. (c) Various proposed tungsten armour tile geometries [108].

sub-cooling³. For cyclic testing with 10 sec of power on, followed by 10 sec of dwell, these workers [88] concluded the following: no damage at 10 MW/m²; longitudinal cracks in the tungsten for some parts under 500 cycles at 20 MW/m²; significant melting on some of the monoblocks after 1000 cycles at 20 MW/m².

The ratio of heat flux incident on the monoblock surface, to the maximum value at the tube wall, is called the peaking factor. For an ITER-like monoblock geometry the peaking factor has been consistently found to lie in the region of ~ 1.4 -1.5 [23]. A safety margin of 1.4 is also used in ITER against a CHF [223]. Taken together, these two factors mean that the Incident Critical Heat Flux (ICHF) on the monoblock is approximately half the CHF. For the ITER design under nominal inlet conditions this results in an ICHF of 25-30 MW/m² [23].

³The difference between the saturation and bulk coolant temperatures.

To use the [ITER](#) design in a [DEMO](#) divertor target plate [\[284\]](#) proposed increasing the coolant inlet temperature to 200°C, and the pressure to 5 MPa. A higher inlet temperature reduces the embrittlement of the CuCrZr pipe under the higher neutron damage levels in a [DEMO](#), whilst the increased pressure aids in maintaining the margin to [CHF](#). The [OFHC](#) copper interlayer was also increased to 1 mm, and the thickness of the tungsten monoblock reduced to 4 mm. Numerical simulations were performed with these alterations at an increased flow velocity of 20 m/s. The calculated peak temperature of the CuCrZr pipe was 370°C for a heat flux of 10 MW/m². As this temperature would lead to significant softening of the CuCrZr [\[75, 158\]](#), additional changes were proposed by [\[221\]](#): a reduction in the width of the monoblock to 21 mm; altering the CuCrZr tube dimensions to $\varnothing 10 \times 1$ mm. These changes reduced the peak temperature of the CuCrZr pipe to 313°C. However, the peak von Mises stress was still above the $3S_m$ ⁴ limit to avoid ratcheting.

The higher flow velocities proposed for an [ITER](#)-like [DEMO](#) design cause two important issues: 1) increased erosion of the pipe wall [\[309\]](#); 2) an increase in pressure drop from the [ITER](#) nominal conditions of ~ 0.5 - 1.5 MPa/m [\[222, 300\]](#). A study by [\[201\]](#) found that erosion from flow velocity became increasingly important as the coolant temperature rose above 150°C.

Copper alloy heat sinks are limited by thermal creep and radiation softening to maximum operating temperatures of ~ 300 - 400 °C [\[133, 312\]](#). This restricts the coolant outlet temperature to relatively low values, which has a negative impact on plant thermal cycle efficiency. To address this [\[90\]](#) proposed employing EUROFER 97⁵ as the tube structural material. In this design a tube ($\varnothing 11 \times 0.5$ mm) made of EUROFER 97 is bonded to a tungsten monoblock via two carbon-based interlayers. One of these interlayer materials was 0.1 mm thick and acted as a compliance layer. The second layer was tapered from 0.075 mm at the top of the pipe, to zero at the sides, to redistribute the heat load around the tube.

⁴Limit for $P_L + P_b + \Delta(P + Q)$ stresses to ensure shakedown after initial loading and unloading cycle. This ensures no large scale deformation will occur in the structure. The terms in the expression are as follows: P_L - local primary stress; P_b - primary bending stress; $\Delta(P + Q)$ - primary plus secondary stress variation.

⁵A low activation ferritic-martensitic steel developed by the [EU](#) fusion program.

A 3D view of the design, and a cross-section, are shown in Fig. 2.2. For an inlet pressure of 15.5 MPa, and 25°C of sub-cooling, this design had an ICHF of 15 MW/m², at a coolant flow rate of 20 m/s.

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Figure 2.2: A 3D view and cross-section of the ITER-like EUROFER 97 swirl tape concept [90].

The major concern with the EUROFER 97 based ITER-like concept was irradiation swelling of the carbon interlayer materials. A more recent geometry based on this concept replaced the carbon interlayers with OFHC copper (1 mm), and reduced the pipe size to $\varnothing 6 \times 0.42$ mm. Numerical simulations by [284] on this design, using the previous flow conditions, gave a maximum temperature of 554°C for the EUROFER 97, at a heat flux of 10 MW/m². However, the design failed both the $3S_m$ and time dependent fatigue limits set out in the Structural Design Criteria for ITER In-vessel Components (SDC-IC) [123, App. B]. In addition, recrystallisation and grain growth in the tungsten monoblock, due to the increased operating temperature, reduce its mechanical strength [214].

Internal channel flow systems using helium as the coolant have received limited attention in the fusion community for HHF cooling applications. The only documented case found by the author was for a swirl rod insert concept developed by General Atomics⁶. This design replaces the swirl tape with a finned rod, which reduces the hydraulic diameter of the pipe

⁶<http://www.ga.com/>

and simplifies the fabrication [24]. Additional 2D wall roughness was incorporated into this design to further enhance the heat transfer. Various studies were performed on this concept looking at the impact of varying wall roughness and swirl rod insert geometries. The researchers concluded that it was not possible to meet the 10% pumping power requirement at a heat flux of 10 MW/m².

2.2.1.2 HyperVapotron

Another concept that received serious consideration for the ITER divertor was the HyperVapotron [29]. It employs a number of grooves, inclined to the flow direction, to act as sights for nucleate boiling. Vapor formed in the grooves is eventually expelled, and recombines with the liquid in the bulk. A cross-section through the concept is shown in image (a) of Fig. 2.3.

An analytical comparison was made by [25] between the HyperVapotron and swirl tape concepts using correlations. The author assumed an equivalent heated area, with copper as the structural material. Fluid inlet conditions for both concepts were set at 4 MPa and 100°C of sub-cooling. At an equivalent mass flow rate both designs achieved a similar ICHF. However, the pumping power required for the HyperVapotron was calculated to be approximately five times greater.

To determine whether the high pressure drops in the HyperVapotron were due to entry length effects, an experimental campaign was undertaken by [306]. The grooves were pitched 6 mm apart with a height of 4 mm and a width of 3 mm. Prototypes fabricated via two different routes can be seen in image (b) of Fig. 2.3. For a 5 mm open channel height the experimental pressure drop was found to be approximately half that predicted by the correlations used by [25]. ICHF measurements were also performed using a flow velocity of 8.5 m/s and inlet pressure of 3 MPa. For 60°C of sub-cooling the design achieved an ICHF of 16 MW/m².

Studies have also been performed to determine the impact of fin geometry on the ICHF of

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Figure 2.3: (a) A section through the JET MkI Hypervapotron [182] [mm]. (b) Front plates with and without vapotron grooves connected to side walls [222].

the HyperVapotron. In [41] they looked at fin heights of 3-7 mm, widths of 2-4 mm and pitches of 2-4 mm. The maximum ICHF was obtained for a fin height of 7 mm, with corresponding widths and pitches of 2 mm. For an inlet pressure of 0.9 MPa, and a flow velocity of 9 m/s, the ICHF was 29.4 MW/m². This was obtained with 115°C of sub-cooling. A complimentary study by [287] looked at the impact of fin shape. They found that a rectangular fin was more effective than a triangular one in maximising ICHF.

The current ICHF values achieved with a HyperVapotron offer no significant advantage over the simpler swirl tape design. However, there is greater scope for optimisation. The flat plate structure also simplifies armour tile attachment, but has been shown to result in de-lamination at lower heat flux values than the monoblock concept [249]. Further, the high pumping powers, in comparison to the swirl tape, will reduce plant thermal cycle efficiency.

2.2.1.3 Screw Tube & Roughened Wall

Other concepts considered for the ITER divertor include screw tubes and porous metal coatings. In the Japanese fusion program this former concept has received significant attention. At the JAERI particle beam facility, screw tubes (M10 x 1.5) machined into copper

blocks were tested with inlet conditions of 1 MPa and 155°C of sub-cooling [222] - see image (a) of Fig. 2.4. At a flow rate of 10 m/s this configuration was able to handle an ICHF of 46 MW/m². The corresponding pressure drop was ~0.5 MPa/m, 20-30% higher than the swirl tape at an equivalent velocity. For an alternative screw geometry of M10 x 1 this dropped to ~33 MW/m², and the pressure drop to 90% of the swirl tape value.

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Figure 2.4: (a) Diagram of key geometric features of screw-tube concept [71]. (b) ICHF for one-sided heating of tubes with and without porous coatings. Coolant pressure of 2 MPa and sub-cooling of 160C. [222].

One of the original concerns with utilising the screw tube was the potential for the geometry to act as a crack initiator. Numerical and experimental studies by [70] showed that fatigue cracks actually initiate from the outer surface of the geometry and propagate inwards. This occurred even for an ICHF of 30 MW/m².

The increased level of sub-cooling, and reduced effective hydraulic diameter, in the screw tube experimental results of [71] account for the difference in ICHF values when compared to the ITER design. As with the HyperVapotron, fabrication and fatigue life then become the primary differentiators. Machining large lengths of screw tube is more manufacturing intensive than inserting a swirl tape. Further, if the screw tube is machined into a block of heat sink material, then a monoblock tile geometry can not be used.

Limited data is available on the wall roughening technique developed by the Russian fusion program. The basic principle was to bake quasi-spherical metal particles (50-100 nm) onto the inner wall of the channel [57, 222]. Experiments using a CuCrZr tube ($\varnothing 10$ mm), with copper particles for the porous layer, achieved an ICHF of 20.7 MW/m² at ITER inlet conditions. The corresponding pressure drop was ~ 0.19 MPa/m. Some results from these experiments that show the influence of the porous coating on the ICHF are shown in image (b) of Fig. 2.4.

2.2.2 Jet Impingement

Jet impingement has received relatively little attention from the fusion community with water as the coolant. An early study by [121] considered the impact of the impingement surface geometry on the CHF. They investigated surfaces with different radii of curvature under impingement from a free slot jet. The slot had a diameter of 2.5 mm and was located 46 mm from the heated section. For a reference flat plate case they found that, for radial distances greater than ~ 5 jet diameters from the stagnation region, the CHF halved. This became more pronounced at higher jet velocities and levels of sub-cooling. For the curved surfaces the centrifugal force increased the pressure gradient normal to the surface, promoting bubble departure and delaying CHF. This benefit increased as the radius of curvature decreased. A Froude⁷ number above 100 was defined as an approximate limit above which the benefits could be observed. The highest CHF achieved in the stagnation region was 39 MW/m², with 80°C of sub-cooling and a jet velocity of 14.6 m/s.

A later jet impingement concept, called the High Efficiency Thermal Shield (HETS), adopted this curved geometry [137]. The design was comprised of a hemispherical shell ($\varnothing 14 \times 1$ mm) made from Dispersion Strengthened (DS) copper, together with an internal 7.5 mm diameter submerged impinging jet. The geometry confined the wall jet to the surface of the hemispherical shell. In image (a) of Fig. 2.5 a HETS module is shown with arrows indicating

⁷The ratio of inertial forces to external forces. In the case of gravity $Fr = \frac{v}{\sqrt{gD}}$, where v is the flow velocity, g the acceleration due to gravity, and D a characteristic length scale [151].

fluid inflow (blue) and outflow (red).

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Figure 2.5: [Left] A cross-section of the HETS module used for HHF measurements. [Right] Fabricated HETS modules used for testing [137].

An experimental campaign was carried out at FE-200⁸ on a prototype module - see image (b) of Fig. 2.5 - using an inlet pressure of 0.116 MPa, and 66°C of sub-cooling. At a heat flux of ~ 24 MW/m² the peak solid temperature was 350°C, for a jet velocity of 10 m/s [215]. The corresponding pressure drop at these conditions was ~ 6 kPa.

For a single, submerged, impinging jet the optimum ratio of jet diameter, to spacing from the impingement surface, is in the region of 4-6 [313]. The water jet impingement concepts discussed here have not operated near this condition, indicating that performance could be improved. In addition, for the same jet Reynolds number, a smaller jet diameter leads to higher heat transfer coefficients [127, 290]. The 7.5 mm jet diameter used in the HETS appears excessive when compared to the ~ 1 mm values used in gas turbine cooling schemes [95]. Confining the jet is also known to degrade the heat transfer performance [27, 157].

The HETS results indicate that a water jet impingement system could be capable of exhausting higher heat fluxes than an ITER-like design, particularly from a thermo-fluid perspective. This is corroborated in the literature, where CHF correlations predict an ap-

⁸A 200 kW electron beam test facility resulting from a partnership between AREVA and the Tore Supra team [30].

proximate doubling⁹ over the swirl tube case for equivalent characteristic flow velocities [96, 187, 222].

Jet impingement was first explored as a means of enhancing the heat transfer performance of helium cooled concepts by adapting the HETS design [137]. The thimble material was changed from DS copper to tungsten, and the outer diameter increased to 32.5 mm. A 7 mm diameter jet was set 1.8 mm from the top of the hemispherical shell. Numerical simulations were performed using an inlet temperature of 600°C and pressure of 14 MPa. For a coolant mass flow rate of 0.0549 kg/s the peak thimble temperature was 2047.2°C, at a heat flux of 10 MW/m². The corresponding pressure drop was 0.1631 MPa.

Within the EU fusion program the Helium-cooled Modular divertor with Jet cooling (HEMJ) [16, 147, 194–197], developed at the Karlsruhe Institute of Technology (KIT), is the jet impingement concept that has undergone the most development. The design is comprised of a hexagonal tungsten armour tile (18 mm across flats) brazed to a WL10¹⁰ thimble. An internal cartridge, constructed from Oxide Dispersion Strengthened (ODS) or EUROFER 97, contains an array of circular jets ranging from ~0.6–1 mm in diameter. A number of images of the concept are shown in Fig. 2.6.

Fatigue studies were carried out at the Efremov institute on a specific HEMJ design. For a coolant flow rate of 13 g/s the module survived 1112 cycles at a heat flux of 10 MW/m² [198]. This was for inlet conditions of 10 MPa and 500°C respectively. Through further optimisation the HEMJ has been able to handle a peak heat flux of 12 MW/m² [115].

The leading helium cooled jet impingement concept proposed by the US fusion programme is the T-tube [288] - see Fig. 2.7. The design employs a tungsten alloy tube ($\varnothing 15 \times 1$ mm), 100 mm in length, as the heat sink, which is cooled by a 0.4 mm slot jet issuing from a tungsten alloy cartridge. A saddle tungsten tile is brazed on top of the outer tube. At

⁹Calculated using a jet diameter of 1 mm and an Open Area Ratio (OAR) of 3%. The OAR is the ratio of the jet-to-cooled area.

¹⁰An alloy of tungsten with 1% weight of La₂O₃.

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Figure 2.6: [Left] [HEMJ](#) cross-section and multi-module construction. [Right] Jet cartridge and 3D cut-section [\[115\]](#), [\[197\]](#).

coolant inlet conditions of 600°C and 10 MPa, a maximum heat flux of ~ 11 MW/m² can be removed, whilst keeping the pumping power below 10% of the heat input. Under these conditions the peak temperature of the heat sink was 1200°C, and the $3S_m$ ratcheting limited was also satisfied. Further optimisation studies [\[288\]](#) showed that the performance could be increased to ~ 13 MW/m².

A disadvantage of the [HEMJ](#) concept is the number of units that would be required to cover the target plates of a [DEMO](#) divertor, which is estimated to be 500,000 [\[274\]](#). To simplify the mechanical design a plate-type concept was developed by [\[286\]](#). The structure has a rectangular cross-section (22 x 67 mm) and is made from a tungsten alloy. Internally, two [ODS](#) steel manifolds are used to supply and exhaust the flow. A slot in the inlet manifold produces a cooling jet along the device length. A distinguishing feature of this design was the increased back plate thickness to maximise the amount of neutron volume heating. The purpose of this was to try and keep the temperature distribution within the structure as uniform as possible. For a poloidal length of 1000 mm the design was able to meet the

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Figure 2.7: [Left] Major components and helium flow route for T-tube. [Right] Dimensioned cross-section through impingement region of T-tube concept [116].

pumping power and $3S_m$ limits up to a maximum heat flux of 10 MW/m^2 . This was for coolant inlet conditions of 600°C and 10 MPa , respectively.

A more recent plate concept looked at incorporating HEMJ-like finger modules into the plate design for a $\sim 20 \text{ cm}$ poloidal section near the target strike point [225, 274, 286]. This allowed the number of required finger modules to be reduced to $\sim 90,000$ [225], whilst also increasing the heat flux handling capability above that of the plate concept alone. Further, the manufacturing process for the finger is simplified by removing the need to bond the tungsten thimble to EUROFER 97 or ODS steel. A diagram of the combined construction is shown in Fig. 2.8.

Extensive testing of the HEMJ has proven its viability at heat fluxes up to 10 MW/m^2 . This meets the quasi steady-state requirements for a DEMO divertor target plate, but no data is available to indicate whether it would survive typical transient loads. The required number of thermal cycles before failure has also not been achieved.

Limited work has been done to date on the manifolding of multi-module jet impingement systems. Apart from the hybrid plate concept incorporating HEMJ-like finger modules, no

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Figure 2.8: [Left] Integration of plate and finger type units. [Right] Breakdown of assembly for plate/finger hybrid [225].

other candidates exist.

2.2.2.1 Roughness

Employing surface roughness features as a means to enhance the heat transfer of a jet impingement system has recently been explored by [230, 231]. The design is comprised of a HEMJ-like finger module, but with an additional roughened plate attached to the impingement surface. A single, 2 mm diameter, jet impinges centrally and then flows outwards through the roughness elements, called a sectorial extended surface by the authors. This disrupts the thermal boundary layer, enhancing the heat transfer. A 3D image and cross-section of the design are shown in Fig. 2.9.

Conjugate CFD simulations were performed on this design using helium as the coolant, at inlet conditions of 600°C and 10 MPa respectively. The results indicated that the maximum heat flux handling capability is limited to $\sim 10 \text{ MW/m}^2$ by the 10% pumping power criteria [230]. This lack of improvement over the HEMJ-like design is driven by the fact that any increase in area averaged heat transfer is offset by an equivalent increase in the pressure drop. In addition, the current fabrication of the roughness feature greatly increases the conduction resistance in the system, which further degrades any benefit from its implemen-

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Figure 2.9: 3D image [left] and cross-section [right] of Sectorial Extended Surface HEMJ-like concept [230].

tation.

Under neutron irradiation the Ductile to Brittle Transition Temperature (DBTT) of tungsten shifts to $\sim 800^{\circ}\text{C}$ [312]. At temperatures above $\sim 850\text{-}1250^{\circ}\text{C}$ grain growth and recrystallisation reduce its strength and hardness [212]. This poses a fundamental problem for all helium cooled jet impingement concepts that use current alloys of this material as both a heat sink and armour tile. For a heat flux of 10 MW/m^2 the thermal gradient through tungsten is $\sim 100^{\circ}\text{C/mm}$, at DEMO relevant operating temperatures. If the peak temperature of the 5 mm thick armour tile is to be kept below the recrystallisation limit, it would mean operating with a brittle heat sink. The lower operating temperature of tungsten under neutron irradiation also poses the secondary problem of joining to standard manifold materials like EUROFER 97, that are limited to upper operating temperatures of $550\text{-}600^{\circ}\text{C}$ by thermal creep [312].

2.2.3 Porous

Micro-porous media have received considerable interest as a means of enhancing the power extraction capability of helium cooled concepts. The ARIES-ST project proposed the use

of concentric pipes with a porous medium between them [219]. In this design a split inner pipe ($\varnothing 16 \times 1$ mm) served as both the inlet and outlet manifold. A cross-section of the structure is given in image (a) of Fig. 2.10.

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Figure 2.10: (a) A cross-section of a split concentric tube porous system [219]. (b) Construction of PFIT and magnified image of porous structure [301].

Numerical simulations were performed on this design using a porosity value of 95%, and assuming tungsten material properties for the pipes. At inlet conditions of 4 MPa, and 500°C, a heat flux of 5 MW/m² resulted in a peak pipe temperature of 1400°C. This is at the upper operating temperature range of tungsten, indicating that 5 MW/m² is an approximate performance limit. These results were for a flow velocity of 53 m/s, and corresponding pressure drop of 0.07 MPa. The researches noted that higher thermal conductivity porous media were more effective at lower porosities. Consequently, conduction through the solid was found to quickly become the limiting factor in heat transfer performance over flow velocity. Channels with small dimensions and high porosity therefore offered the highest heat flux handling capability.

Helium cooled linear porous concepts have also been explored [140]. One, called the Porous Foam in Tube (PFIT) [301], used a Chemical Vapour Deposition (CVD) technique developed by Ultramet Inc.^[1] to create an integrated tungsten tube and porous foam structure. In contrast to sintered particle foams, this design contained relatively large volumes around the ligaments - see image (b) of Fig 2.10. This minimised the pressure drop, whilst still retaining a large contact surface area for heat transfer. Helium at 4 MPa and 40°C was flowed through a PFIT pipe (12.7øx 2.05 mm) for a porous length of 38 mm. At a flow rate of 27 g/s the peak structural temperature was 2300°C, for an incident heat flux of 22.4 MW/m². The corresponding pressure drop was equivalent to 1.84 MPa/m.

Micro-porous structures combine large internal wetted areas with small hydraulic diameters. This results in systems with low coolant mass flow rates and high pressure drops. The high coolant temperature changes mean that heat sinks have to be limited in size to maintain relatively uniform temperatures.

As conduction through the solid limits heat transfer performance, micro-porous heat sinks have a sharp heat flux limit. This is because large changes in coolant flow rate only have a limited impact on solid temperatures. Consequently, the system pumping power can quickly reach values above the 10% limit with gaseous coolants.

The small hydraulic diameters in a micro-porous structure are also susceptible to blockage. This can drastically increase the pressure drop of the system, and makes producing robust components based on this mechanism challenging. When a gas is used as the coolant, large density changes can also have a negative impact on the coolant mass flow distribution. This particular effect was noted by [305].

At the KIT porous systems with macro features have also been developed. One concept, termed HELium-cooled Modular divertor with integrated Pin array (HEMP), had a thimble

¹¹<http://www.ultramet.com/index.html>

structure formed from TZM¹² ($\varnothing 16 \times 1$ mm) into which a pin-fin array was brazed [55]. On top of the thimble a 5 mm thick tungsten tile was bonded. The design is shown in Fig. 2.11. Helium flows from the outer region inwards and is collected in a manifold made of ODS steel.

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Figure 2.11: Details of HEMP geometric design [55].

A numerical study was performed on this design using heat transfer coefficient and pressure drop correlations for heat exchanger tube bundles. The inlet conditions were set to 620°C and 10 MPa respectively. At a heat flux of 15 MW/m² the peak thimble temperature was calculated to be 1400°C, and the fractional pumping power 5% of the heat input.

Another pin-fin concept with tubular flow was explored by [103]. This design housed rectangular inflow and outflow ODS steel manifolds inside a tungsten alloy heat sink (12 x 24 mm). A slot at the top of the inlet manifold fed into a pin-fin array brazed onto the tungsten structure. By reducing the height of the inlet manifold, and increasing the height of the outlet manifold, along the flow direction a constant flow velocity from the slot was obtained. Numerical simulations gave a peak heat sink temperature of 1400°C at 10 MW/m², with

¹²An alloy of 0.5% Titanium, 0.08% Zirconium, 0.02% Carbon and with the balance Molybdenum

coolant inlet conditions of 620°C and 10 MPa respectively. Heat exchanger tube bundle correlations were again used to predict heat transfer coefficients and pressure drops. A novel feature of this concept was the use of the spent helium from the pin-fin as a temperature buffer between the ODS steel and tungsten alloy.

There has been no experimental validation of the applicability of heat exchanger tube bundle correlations to predict the performance of these macro-porous concepts. The TZM alloy used in the HEMP concept has a DBTT under neutron irradiation in the region of $\sim 700\text{-}750^\circ\text{C}$, and a maximum operating temperature of 1200°C [281]. This means that it will suffer from similar operational temperature window constraints to the helium cooled jet impingement concepts discussed in Sec.2.2.2.

Macro-porous systems operate at higher mass flow rates than micro-porous designs. This necessitated a switch to higher operational pressures to avoid excessive pumping powers. As a result, these designs suffer from the same fabrication and manifolding difficulties as many of the designs discussed in Sec.2.2.2.

2.2.4 Capillary

The use of liquid metals for first wall and divertor cooling applications in a tokamak has received considerable interest in the literature [7, 39, 128, 145, 172, 183, 191, 220]. Early work with these coolants for HHF applications in the divertor focused on heat pipes [172, 237]. This interest was driven by results in the aerospace industry where such structures had been researched for hypersonic applications. As part of the National Aerospace Program, Thermacore Plc.[13] developed a lithium heat pipe for cooling the leading edge of a hypersonic engine cowl that was able to remove a peak vapor heat flux[14] of 200 MW/m^2 , operating at a temperature of 1650°C [237].

Some of the first experimental studies on heat pipes for fusion applications were performed

¹³<https://www.thermacore.com/>

¹⁴The heat flow of the vapor divided by the cross-sectional area. This is not the surface heat flux.

by [172]. Using sodium as the coolant, they were able to achieve a CHF of 5.3 MW/m² in a horizontal orientation. Operation in the direction of gravity increased performance by ~20%. The same study also recorded the performance under a magnetic field of 1.5 T transverse to the flow direction. This was shown to reduce the CHF by 18%.

As heat pipes only act to redistribute heat, a divertor target geometry was proposed by [172] in which radiation in the condensation region of the heat pipe is exchanged with a convection cooled system. This design proposal is shown in Fig 2.12.

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Figure 2.12: [Left] Location of heat pipe modules in divertor. [Right] Duel heat pipe design to re-distribute divertor heat load over an increased area [172].

Experimental validation of heat pipes operating at DEMO relevant heat fluxes has not been achieved to date. Calculations indicate that such heat fluxes would be at the limit of what is feasible with a heat pipe design [296]. As the presence of a strong magnetic field would deteriorate performance further, there has been limited recent interest in such concepts.

The limited space in the divertor region makes a radiatively cooled heat pipe system unattractive. Such a design would also require running the heat sink at low operational temperatures to ensure enough radiative power is transferred, which would have a negative impact on the thermal cycle efficiency.

More recently, a liquid metal design called the **Capillary-Pore System (CPS)** was proposed by [69, 183] as an alternative to a solid armour tile. It operates by continually wetting the **PFC** surface with lithium using capillary action - see Fig. 2.13. The small hydraulic diameter of the capillary structure minimises the Hartmann number¹⁵, which limits the impact of the magnetic field on the flow. Though a heat pipe also uses a capillary action to wet the heated end, the resulting vapour flow can be effected by magnetic fields.

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Figure 2.13: **CPS** with and without lithium wicking. Potential divertor wall structure also shown [183].

The role of the lithium is to remove a fraction of the incident heat flux through evaporation, and ensure minimal erosion to the divertor target surface. A convection cooling system installed below the wicking structure then only needs to remove a much reduced heat flux. Under testing in the T-11M tokamak this system was able to remove a peak heat flux of 25

¹⁵Ratio of the electromagnetic force to the viscous force. For a magnetic field, B , it is given by $Ha = BL\sqrt{\frac{\sigma}{\mu}}$. Here L is a characteristic length for the flow, σ the surface tension, and μ the dynamic viscosity.

MW/m² over 3 cm².

The **CPS** system can be seen as complimentary, rather than competitive, to current **HHF** cooling concepts. To allow flexibility in the plasma facing surface, it points towards a future design strategy that focuses on **HHF** heat sink development, rather than full **PFU** components.

2.2.5 External Flow

A design called the **Lithium-Metal Infused Trenches (LiMIT)** was proposed by [243] that has liquid lithium running through channels on the **PFC** surface. The temperature gradient through the lithium generates a current via the thermoelectric effect which, when combined with the toroidal magnetic field, drives the flow without any external pumping requirements. Fig. 2.14 shows how this concept operates.

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Figure 2.14: Diagram of how the **LiMIT** concept would operate in the divertor [243].

The flow velocity in the channels of the **LiMIT** are linked directly to the heat flux. This means that the concept is also self regulating. Experiments were performed on a stainless steel mock-up with trenches 2 mm in width, 10 mm in depth and 90 mm in length. Extrapolating from the results, [243] calculated that the flowing lithium alone would be able

to handle a heat flux of $\sim 20 \text{ MW/m}^2$.

There are a number of key issues that need to be resolved before serious consideration can be given to utilising liquid metal as plasma facing surfaces: stability of the liquid metal flow under magnetic fields and particle bombardment; mass transport and material migration; integrating such components into a machine to give desirable plasma scenarios [128].

2.3 Key Findings & Research Gaps

A range of divertor target concepts have been discussed. All currently fail to fulfil at least one of the criteria for a DEMO divertor target component set out in Table 2.1. The maturity of the ITER divertor swirl-tape concept has meant that iterations of this design are currently most favoured for a DEMO in the EU fusion community. The key findings from the literature review, and the related research gaps for divertor cooling are given below:

- Key Findings
 - Maintaining the peak tungsten armour tile temperature below the recrystallisation limit at DEMO steady-state heat fluxes, and relevant coolant inlet conditions, has not been achieved by any concept;
 - Using tungsten as both an armour tile and heat sink material is a flawed approach with currently available alloys. Under neutron irradiation, the whole operational temperature window of tungsten can be passed through in the armour tile alone at DEMO steady-state heat flux levels;
 - Current helium cooled concepts can not fulfil the DEMO transient heat flux requirements without either: a) exceeding the 10% pumping power limit; b) using structural materials outside of their operational temperature windows;
 - No concept has been able to meet the life requirements for a DEMO target component under relevant inlet coolant conditions;

- The manifolding of multiple small modules remains an outstanding challenge;
 - Liquid metal **PFC** research is complimentary to **HHF** heat sink development. The **Technology Readiness Level (TRL)** - see App.**A** - of these concepts is still in the 1-3 range;
- Research Gaps in the Field of Divertor Cooling
 - Liquid, single-phase, jet impingement and porous flows have not been considered for divertor target plate cooling;
 - There are no water cooled concepts that combine high operational pressures with low coolant flow rate requirements in order to maximise thermal cycle efficiency;
 - For forced convection cooling systems, no experimental techniques have been used to report local heat transfer coefficient data;
 - All of the heat transfer enhancement techniques reviewed in the literature revolved around convection and boiling. In **HHF** component design conduction heat transfer can be just as important. Designing components that consider all aspects of thermal resistance is currently lacking.

The author acknowledges that there is existing literature, outside of the fusion field, on the identified research gaps. However, to the author's best knowledge, there has been no research in these areas under the specific boundary conditions found in the divertor region of a fusion power plant. Additionally, the cross-disciplinary nature of the divertor challenge promotes combining knowledge across fields, which offers a further means of achieving novelty.

Chapter 3

The High Pressure Jet Cascade Concept

“It is a profound and necessary truth that the deep things in science are not found because they are useful; they were found because it was possible to find them.”

— Julius Openheimer

This chapter begins by establishing the main aims of the research. The author then introduces a concept to address them. Numerical thermo-fluid simulations are performed to compare its performance against an **ITER**-like **DEMO** target. The methodology of this approach is discussed in detail.

3.1 Research Aims

In Chpt. 2 it was established that there is currently no solution to the **DEMO** divertor target plate problem. Each proposal to date fails either the heat flux handling, or life, requirements. Those that currently achieve the heat flux specification, do so at the expense of high coolant flow rates. This leads to low coolant outlet temperatures, which has a negative impact on plant thermal efficiency. Taking the **ITER** divertor as an example: if the coolant outlet temperature was increased by $\sim 40^\circ\text{C}$, it would improve the overall thermal efficiency by 1-2%.

HHF component design can be seen as a compromise between the following four quantities: 1) solid component temperatures; 2) heat flux loading; 3) coolant mass flow rates; 4) coolant inlet temperatures. The higher the heat transfer performance of a component, the more flexibility the designer has within this range of variables. For this reason the author decided to make the aim of this research the development of a **Plasma Facing Unit (PFU)** with enhanced heat transfer performance relative to current **DEMO** divertor target concepts.

The heat transfer performance of a **PFU** is generally driven by the heat sink design. Consequently, the author decided to focus on this part of the **PFU** in their research. As there is still a high degree of uncertainty about the magnitude and nature of the transient heat loads in a **DEMO** [292], focus was given to designing a heat sink that met the steady-state heat flux, and life, requirements.

An efficient cooling geometry is beneficial when used with any material and coolant combination. However, the author chose to use the **EU DEMO** divertor target reference configuration of a water cooled CuCrZr heat sink in this research [299]. This heat sink material has well documented properties in a fusion environment, which reduces modelling uncertainty [62, 73, 74, 76, 158, 189, 271]. Using this reference material and coolant configuration also facilitates comparison.

Taking an **ITER**-like **DEMO** divertor target as a reference, the research aims can be stated more precisely as:

- (1) A reduction in the mass flow rate required to obtain a set component temperature, at a given heat flux;
- (2) An increase in the operational pressure to ensure higher coolant outlet temperatures can be achieved without the risk of a phase change; above the critical heat flux this can lead to a catastrophic increase in the wall temperature.

Point (1) is a practical statement of the requirement for enhancing heat transfer. In fusion **HHF** systems this involves considering both conduction and convection, as Biot numbers are generally of the order of 10^{-1} . The convective resistance is minimised when the product of the area averaged heat transfer coefficient and the internal wetted area is maximised. For a given material, conductive resistance is minimised by using short conduction paths lengths and avoiding geometric heat flux peaking.

Water coolant circuits at pressures in excess of 15 MPa are routinely used in fossil fuel power stations. These pressures are also frequently encountered in the oil and gas extraction sectors. This latter industry has developed **Printed Circuit Heat Exchangers (PCHes)** that are able to operate at working fluid pressures and temperatures of 60 MPa and 900°C respectively [102]. The application of high pressure technologies developed in these two sectors offers a potential route to fulfilling aim (2).

3.2 A High-Heat Flux Cooling Module

In Sec.2.3 of Chpt.2 the author noted that single phase jet impingement with a liquid coolant has not been researched for applications in the divertor target region. In gas turbines, jet impingement is used to combat the most extreme heat loads at the leading edge of the turbine blade, where heat fluxes can be in excess of 3 MW/m^2 [99]. Convective heat transfer coefficients produced by this method are over three times those achievable in parallel wall flow concepts, at equivalent flow speeds [313].

A novel means of reducing cross-flow when feeding multiple jets was developed for power electronic applications by [47]. It involves deploying the coolant in a cascade arrangement, whereby spent flow from one jet/array is used to feed another jet/array. A wall separates adjacent cascade elements, stopping the build-up of any cross-flow. This flow arrangement also offers a means of exchanging additional pressure drop in a system for a reduced mass flow rate. Such a configuration is particularly effective when the coolant is a liquid, as high pressures can be achieved for minimal pumping work.

To fulfil the research aims set out in Sec.3.1 the author decided to pursue a high pressure cooling system employing cascade jet impingement. Component manufacturing complexity was reduced by using a flat plate geometry. This also kept the heat conduction paths small, and has the potential to increase temperature uniformity relative to other designs. By stacking multiple sheets into a laminate structure, three dimensional flow paths can be incorporated into a compact design.

An hexagonal geometry was chosen for the in-plane structure to maximise the plasma wetted area for a given component length scale. This shape also allows tight packing of multiple modules on the divertor target plates. The internal surface of the hexagon was sub-divided into a number of cavities to accommodate high coolant pressures. This general design structure was termed the High Pressure Jet Cascade (HPJC).

A baseline design, called the HPJC-1a, has an hexagonal side length of approximately 10 mm, and is sub-divided internally into six equilateral triangles, each with a side length of 6.8 mm. These triangular cavities are cooled by a 0.8 mm diameter impinging jet¹. This set of length scales was chosen to achieve an optimum Open Area Ratio (OAR)² in the range of 1-3% [290, 313], and ensure that pressure stresses in the impingement plate were below $\sim 0.5\sigma_y$. A plan view, cross-section, and the fluid region of this design are shown in Fig.3.1.

To develop a quantitative understanding of the merits of the cascade concept, only a single cascade was used in the baseline design. Jets were grouped, and then labelled, according to their position within the cascade. An ordinal numbering system was used, with the first set of jets in the cascade called the primary jets.

Three primary jets are first fed with coolant from a manifold - (1) in image (c) of Fig.3.1

¹The minimum jet diameters employed in turbine blade cooling that avoid blockage are ~ 0.5 mm [154].

²The ratio of the area of the jet to the surface to be cooled

Following jet impingement nomenclature these were called the impingement plate, jet plate and cascade plate. This structure is depicted in Fig. 3.2, with arrows representing fluid inflow (blue) and outflow (red) locations.

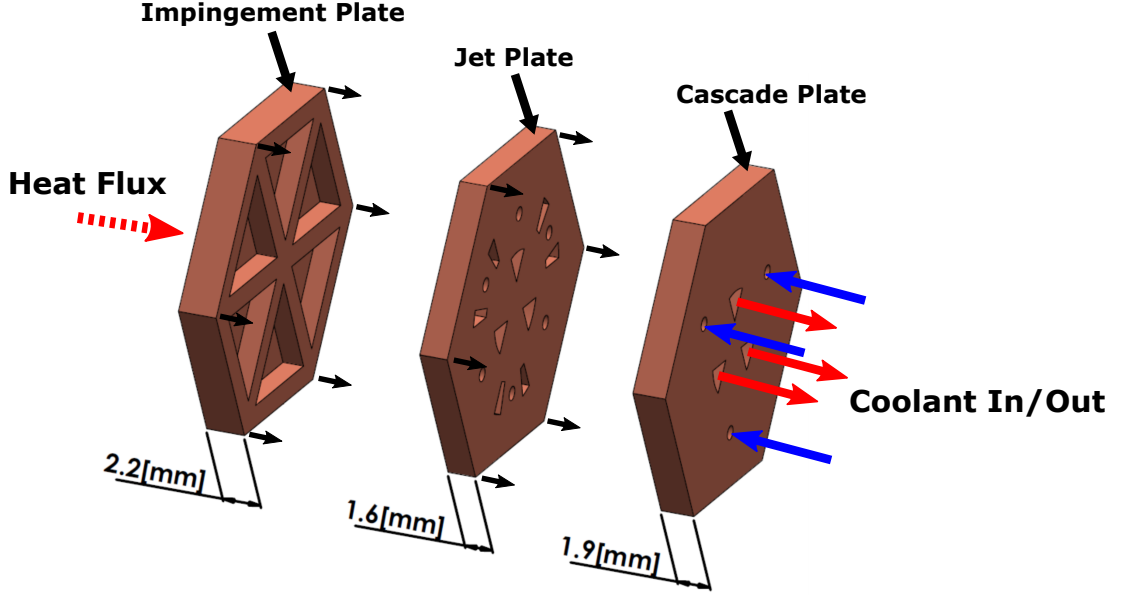


Figure 3.2: Constituent laminate CuCrZr sheets for HPJC-1a construction indicating fluid inflow (blue) and outflow (red) locations.

The jet exit plane was located 1.2 mm away from the impingement surface. In the literature, the ratio of this distance to the jet diameter (Z/D) has been shown to have a significant impact on the heat transfer performance [127, 290]. For a single jet the optimum value is found to be in the range of 4-6 [313], whilst for multiple jets it reduces to 1-3 [290]. A value of 1.5 is used for the HPJC-1a. This was a compromise between optimising heat transfer performance and minimising pressure stresses.

3.3 Numerical Simulations

3.3.1 Methodology

Steady-state conjugate CFD was performed on a 120° periodic segment of the HPJC-1a using the ANSYS 15.0 CFX software. The thermal resistance of the structure was assumed to be dominated by that through the impingement plate, and at its interface with the coolant.

This allowed the laminate structure to be modelled as a single solid body.

An unstructured mesh was created using ICEM in ANSYS 15.0. Approximately two million cells were used to model the fluid region, and one million in the solid. The mesh at the fluid-to-solid interface is shown in Fig. 3.3, along with a cross-section through the fluid domain. A number of prism layers were used to capture the gradients in the boundary layer at the fluid-to-solid interface. The thickness of the boundary layer was estimated to be $30\text{ }\mu\text{m}$, based on a correlation for turbulent flow over a flat plate [120]. Prism layers were grown to this height using a growth ratio of 1.25.

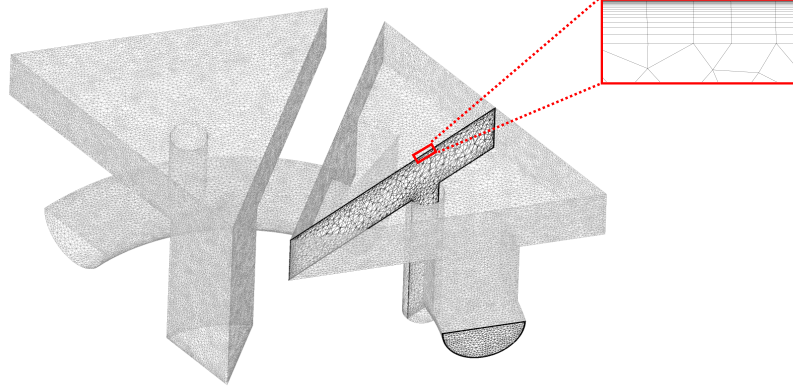


Figure 3.3: Fluid-to-solid boundary mesh for HPJC-1a geometry. Cross-section through fluid section shows boundary layer prisms.

Water was modelled using the in-built IAPWS-97 thermodynamic properties library, and assumed to be single phase. The minimum operating temperature of CuCrZr is dictated by neutron embrittlement. However, the exact value ranges from $150\text{--}200^\circ\text{C}$, depending on whether a minimum value of uniform, or total, elongation is used as the criteria [300]. In this research the author chose to use the upper bound. To meet this condition the coolant inlet temperature was set to 180°C , which accounts for the fact that there will be a finite convective thermal resistance between the coolant and wall. The reference pressure was set to 20 MPa based on discussions with colleagues at Rolls-Royce nuclear on industrially available high pressure water equipment [232]. The thermal properties of CuCrZr were taken from the ITER reference heat treatment in [311] and assumed constant; a reference

temperature of 300°C was used.

As there is still uncertainty in the exact heat flux distribution over a DEMO divertor target, the fusion community currently compares designs assuming a uniform heat flux [116, 274]. This same approach is taken here. The remaining external surfaces of the module were all set to be adiabatic.

Interpolation of quantities between the mesh faces on the periodic boundaries was automated by ANSYS CFX 15.0 for both fluid and solid bodies. The area averaged static pressure was set to 0 Pa at the fluid outlet boundary, and a mass flow rate condition applied at the inlet. These boundary conditions are summarised in Fig. 3.4

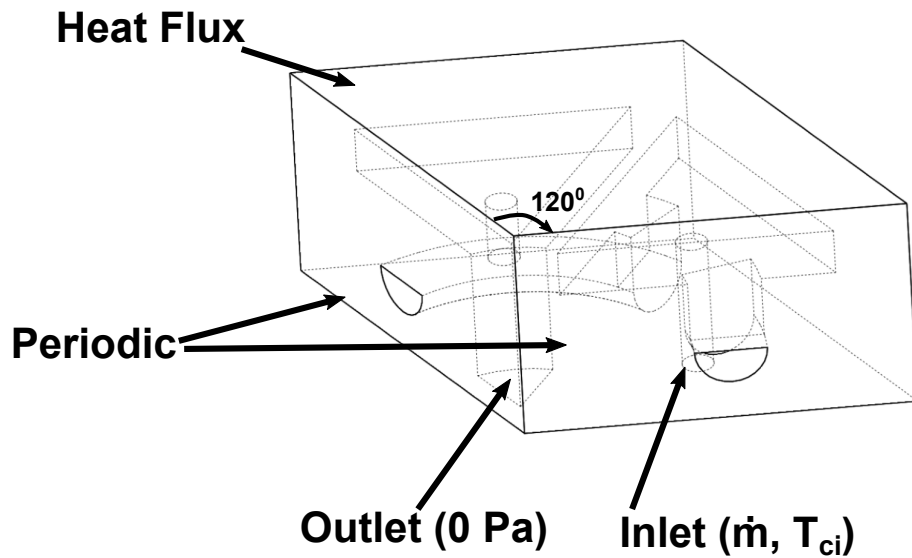


Figure 3.4: Thermo-fluid boundary conditions used for CFD on HPJC-1a module.

Convergence of the mass, momentum and energy residuals below 10^{-5} was achieved for all simulations. Additionally, energy imbalance for the solid and fluid domains was below 1% at the solution end.

3.3.2 Turbulence Modelling

The Reynolds numbers of the jets were of the order of 10^4 ; in the turbulent flow regime. To capture the heat transfer at the impingement surface this turbulence needs to be modelled accurately. The author chose to use two-equation eddy viscosity turbulence models to close the Reynolds Averaged Navier Stokes (RANS) equations [14] in this research. These models have been used extensively in other fields to simulate jet impingement flows [313]. Of the two-equation models available in the ANSYS 15.0 CFX software, those based on the k- Ω ³ formulation were chosen for this study due to their ability to fully resolve the viscous sublayer. Though these low turbulent Reynolds number (low-Re) models involve higher computational effort than employing a full wall function treatment, the accuracy of the heat transfer results is generally superior as long as the boundary layer is adequately resolved [313].

A total of four models were compared for the work: k- Ω ; Shear Stress Transport (SST); Baseline (BSL); Baseline Explicit Algebraic Reynolds Stress Model (BSL-EARSM). This latter model is an extension of BSL to account for streamline curvature and secondary flow effects [3]. Both of these effects are important in jet impingement flow fields [26].

An automatic wall treatment was used for all of the models. This allows CFX to switch from a wall function, to a low-Re approach, where the boundary layer mesh is adequately refined [19]. The accuracy of these models in predicting convective heat transfer and pressure drop results for the HPJC-1a is compared with experimental data in Sec. 3.5.3.

3.3.3 Mesh Sensitivity & Independence

To take advantage of the capability of k- Ω turbulence models to resolve the viscous sublayer, y^+ values of less than unity should be targeted at the fluid-to-solid interface [19]. The formula for y^+ is given by

³These models assume that the turbulence viscosity (μ_t), turbulence kinetic energy (k) and turbulent frequency (ω) are related by the following: $\mu_t = \rho \frac{k}{\omega}$, where ρ is the density [2].

$$y^+ = \frac{v^* y}{\nu}, \quad (3.1)$$

where v^* is the friction velocity at the wall, ν the kinematic viscosity, and y is the spacing of the first node from the wall. The friction velocity is dependent on the square root of the wall shear stress.

A sensitivity study was performed on the HPJC-1a mesh. The variation in peak y^+ with jet Reynolds number, and near-wall mesh refinement, were both investigated. The impact of this latter effect was captured by varying the number of prisms layers from 14-20, whilst keeping the total height and growth ratio constant. At each level of refinement, simulations were run at a range of Reynolds numbers. A fixed heat flux of 10 MW/m^2 was applied to the impingement surface, and the turbulence modelled using SST. In Fig. 3.5 the peak y^+ is plotted against Reynolds number for varying levels of mesh refinement.

At a fixed Reynolds number the peak y^+ is seen to decrease with the number of prism layers. This is caused by the reduction in the first node spacing from the wall. If the number of prism layers is kept constant, and the Reynolds number increased, the value of y^+ also increases. This is due to the fact that the Reynolds number is proportional to the wall shear stress.

From Fig. 3.5 it is evident that 20 prism layers are required to achieve a peak $y^+ < 1$ over the range of Reynolds numbers of interest. The corresponding first node spacing from the wall was $\sim 0.12 \mu\text{m}$.

Mesh independence at a local level was investigated by plotting the variation of the heat transfer along a line across the triangular cavity. This was done at a fixed Reynolds number for each level of mesh refinement. The heat transfer coefficient, h_c , was defined as,

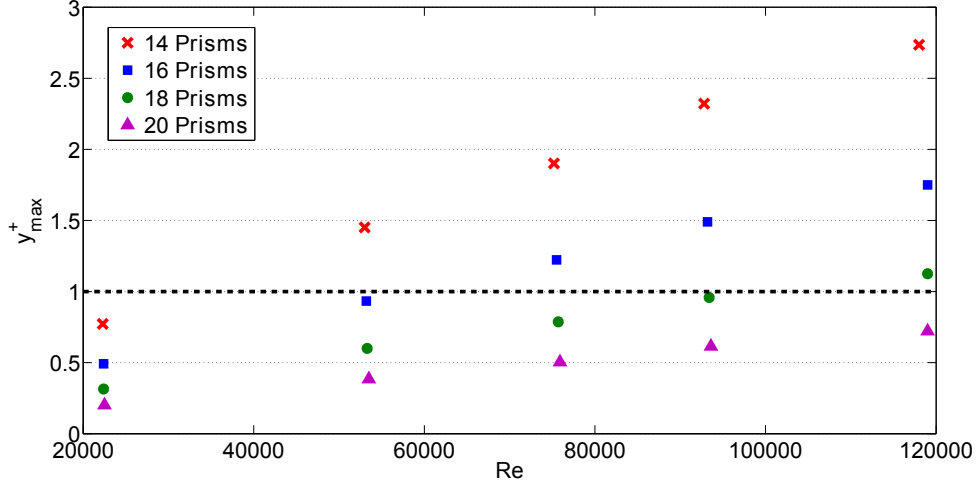


Figure 3.5: Maximum y^+ on primary triangular surface for different levels of prism layer refinement.

$$h_c = \frac{q''}{(T_w - T_j)}, \quad (3.2)$$

where q'' represents the local wall heat flux, and T_w the wall temperature at the same location. The reference value T_j is taken as the area averaged temperature of the fluid, one jet diameter upstream of the jet exit plane.

The Chilton j-factor [120], j_H , is a useful non-dimensional measure of convective heat transfer. It is defined as

$$j_H = St \cdot Pr^{\frac{2}{3}} = \frac{Nu}{Re \cdot Pr^{\frac{1}{3}}}. \quad (3.3)$$

It is an empirical relation valid for turbulent flows with $0.6 < Pr < 60$. The form of this non-dimensional heat transfer parameter means that it is only weakly dependent on both the Reynolds and Prandtl numbers for a broad range of coolants and flow conditions. The author chose to use this parameter as a measure of convective heat transfer to facilitate application of the HPJC-1a architecture to alternative coolants.

Reference length scales and areas for j_H were based on the jet diameter, and its flow cross-section, respectively. Fluid properties were taken at the film temperature, T_{cf} , defined as

$$T_{cf} = \frac{T_j + T_{w_{avg.}}}{2}, \quad (3.4)$$

where $T_{w_{avg.}}$ is the average temperature of the heat transfer surface of interest.

In Fig. 3.6 j_H is plotted against the normalised distance along the primary triangular cavity altitude⁴ for varying levels of mesh refinement.

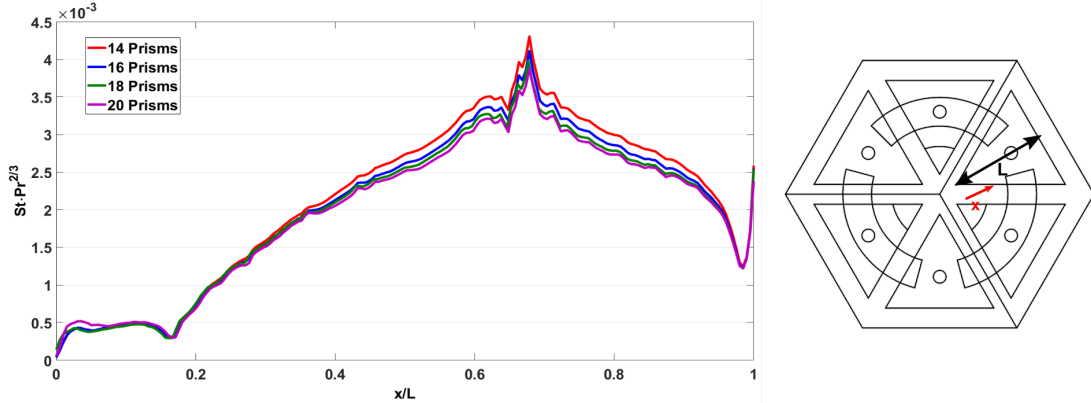


Figure 3.6: Variation in local j-factor along altitude path in primary cavity for different levels of boundary layer refinement.

As the number of prism layers is increased from 14-20, there is a reduction in the peak heat transfer predicted in the stagnation region of $\sim 10\%$. The variation between different levels of refinement diminishes as the number of prism layers is increased. Between the 18 and 20 prism layer cases the peak value differs by only $\sim 2.5\%$. As this value is at least an order of magnitude smaller than the experimental measurement uncertainty - see Sec. 3.5.2 - further refinement would have a negligible impact on the results. Therefore, a final value of 20 prism layers was chosen for this research.

⁴The altitude of a triangle is a line segment through a vertex that meets the opposite side at 90 degrees.

3.4 Reference Case

A reference simulation of the HPJC-1a was performed using the SST turbulence model at a heat flux of 10 MW/m². The coolant mass flow rate was adjusted until the peak solid temperature was 350°C. This is regarded as the maximum permitted operating temperature of CuCrZr to avoid severe thermal/irradiation creep and softening [298]. The purpose of this study was to determine whether the concept was capable of operating under DEMO level steady-state heat flux levels, and to compare it with an ITER-like DEMO design. A summary of the results is shown in Table 3.1.

Concept	$\dot{m}_A$ [kg/m ² s]	$\Delta P_{\text{tot.}}$ [kPa]	ΔT_{fluid} [°C]	P_w [kW/m ²]	v [m/s]
HPJC-1a	56	140	41	8.6	10
ITER-like	5300	8.0	0.4	49	20

Table 3.1: Comparison of the HPJC-1a and ITER-like DEMO divertor [222, 284] concepts for a 10 MW/m² heat flux. The HPJC-1a results come from conjugate CFD simulations at a fixed peak solid temperature of 350°C. P_w is the pumping power per square metre, and v the characteristic flow velocity.

The ITER-like values in Table 3.1 were based on the results of [284]. It was deemed a fair comparison by the author as both the incident heat flux, and difference between the coolant inlet temperature and peak heat sink temperature, was equivalent for both concepts. To calculate the pressure drop the swirl tape was assumed to have a twist ratio of two and be 0.8 mm thick [222].

Evidently the HPJC-1a operates at considerably lower mass flow rates than the ITER-like DEMO design. This results in increased coolant outlet temperatures. This benefit could be offset if the pumping powers are excessive. However, it is found that the HPJC-1a pumping powers are over five-fold smaller than those for an ITER-like DEMO. Proponents of the ITER-like design may argue that the swirl tape will only be employed in the region around the strike point, so that the actual pumping powers will be considerably lower. This neglects the modular nature of the HPJC-1a, which would allow an even finer degree of

control over optimising the mass flow distribution to minimise the total pumping power.

In the following sections local results from this simulation are considered in detail.

3.4.1 Impingement Surface Heat Transfer

A contour plot of j_H over the impingement surfaces is shown in Fig. 3.7

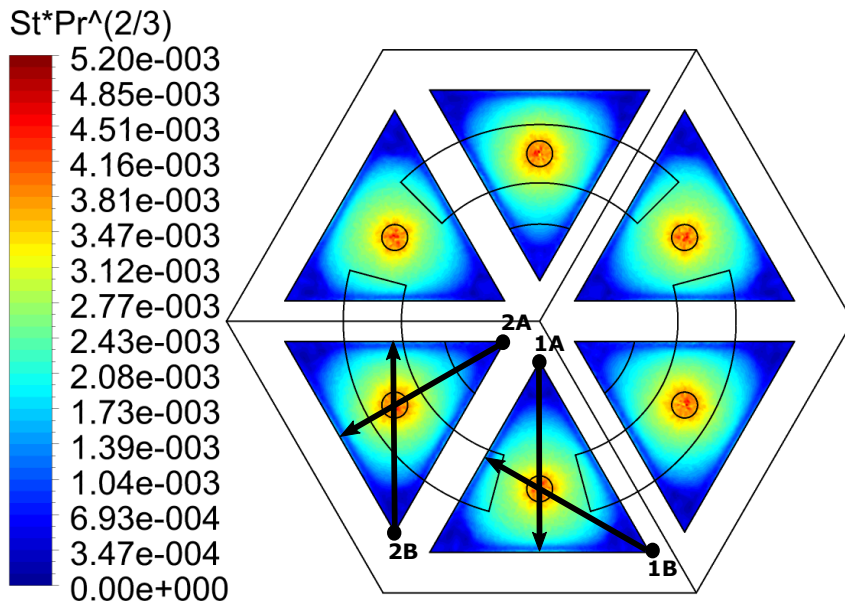


Figure 3.7: A contour plot of j_H over the triangular impingement surfaces for the reference case. Local data altitudes are also shown emanating from vertices 1A, 1B, 2A and 2B.

For each jet the convective heat transfer is a maximum in the stagnation region, and deteriorates moving outwards along the impingement surface. The maximum j_H values are $\sim 3.8\%$ larger for the secondary jets. When the j_H values are averaged over the impingement surfaces, minimal difference is found between the cavities. As the only geometric variation between the primary and secondary jet cavities is the outlet locations and jet entry length, these results imply that the influence of these factors on the jet flow field is minimal.

Convective heat transfer near the triangular vertices on the impingement surface is negligible in comparison to the stagnation region. This non-uniformity in the distribution

leads to corresponding non-uniformities in the solid temperature field, which generate thermal stresses.

To illustrate the variation in the local convective heat transfer over the cavity surfaces, line data were taken along altitudes originating at vertices 1A, 1B, 2A and 2B - see Fig. 3.7. These results are shown in Fig. 3.8.

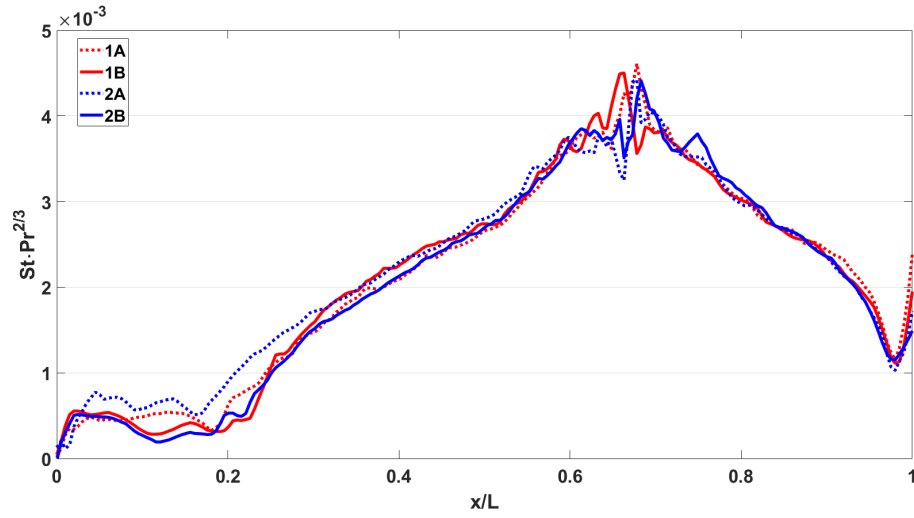


Figure 3.8: Local variation of j_H along altitudes in both primary and secondary jet cavities. These altitudes are depicted in Fig. 3.7.

Fig. 3.8 corroborates the previous statement that the convective heat transfer appears to be largely independent of the chosen cavity. The figure also shows that, away from the side walls, the heat transfer is approximately symmetric about the cavity orthocentre⁵. This implies that there is minimal cross-flow being generated.

3.4.2 Solid Temperature Distribution

Approximately 51% of the impingement plate surface area in the HPJC-1a is covered with the webs that form the internal cavities. This facilitates conduction heat transfer to the cascade plate. If the heat transfer coefficients in the cascade plate are also low, then the minimum solid temperature may be maintained significantly above the coolant inlet temper-

⁵The point where all three altitude lines of a triangle intersect.

ature. This would mean the coolant inlet temperature could be reduced, and the minimum temperature of the CuCrZr still maintained above the embrittlement limit. At the same coolant flow rate, this would increase the heat flux handling capacity, but at the expense of a larger temperature gradient through the structure.

The lowest solid temperatures in the HPJC-1a occur near the primary jet inlet ports, at the fluid interface. A contour plot of this surface is shown in image (a) of Fig. 3.9.

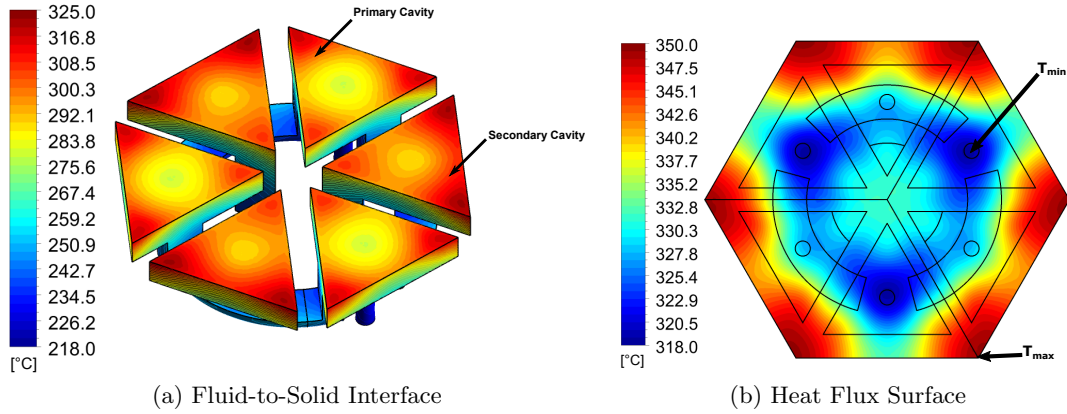


Figure 3.9: Contour plots of temperature distributions in the HPJC-1a at (a) the fluid-to-solid interface; (b) the heat flux surface.

The minimum temperature of the CuCrZr is 218°C, which is 38°C above the coolant inlet temperature. This validates the hypothesis, and provides a simple means of improving the heat flux handling capacity of the HPJC-1a.

In image (b) of Fig. 3.9 the temperature distribution on the heat flux surface is shown. The global peak solid temperatures occur on this surface at the hexagonal vertices. There are two mechanisms that drive the higher temperatures at these points: (1) the increased conduction path length relative to the plate thickness; (2) the low convective heat transfer in these regions. Point (1) could be addressed by reducing the thickness of the outer walls of the hexagon. However, doing this would increase the pressure stresses and reduce the bonding area. Adapting the shape of the cavity or the location of the jets would address (2).

3.5 Experimental Comparison

3.5.1 Methodology

An experimental campaign was undertaken to validate the numerical approach. Reynolds numbers were matched, with air as the working fluid. This required the HPJC-1a design to be scaled to ten times the diverter target plate application size. At the inlet conditions used in the numerical simulations the Prandtl number of water is ~ 1 , which compares to a value of ~ 0.7 for air. By scaling the heat transfer data by $Pr^{-\frac{1}{3}}$, the influence of any difference between the two Prandtl numbers on the results was minimised. Perspex was used as the construction material to allow easy visual access to the rig, and because of its well documented thermal properties.

A regulated 100 psi air line was used to feed the three primary jets. Orifice meters were installed within each of the three inlet channels to measure the flow rates. These had all previously been calibrated against a [British Standards](#) 1042 orifice meter. A single channel was used for the combined outlet port exhaust. This was restricted with an orifice to provide back-pressure to stabilise the flow. In Fig. [3.10](#) a flow schematic for the rig is shown.

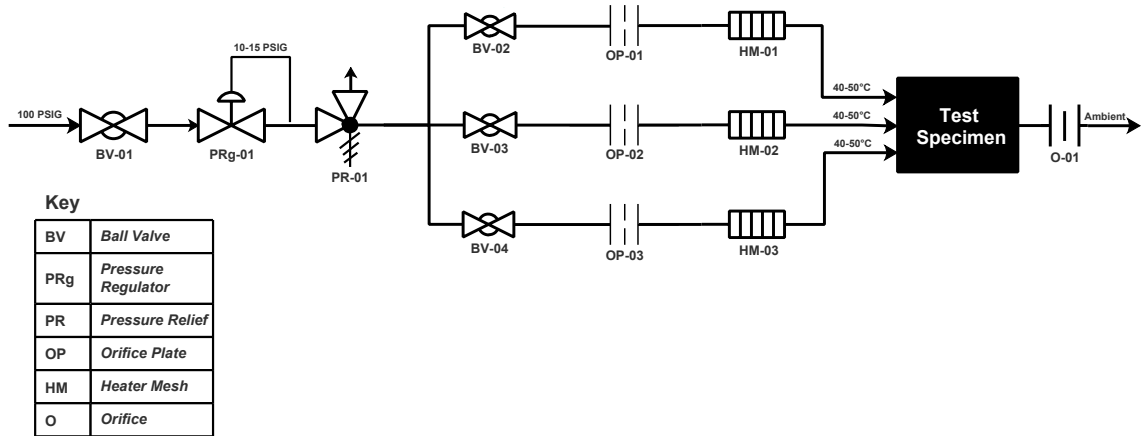


Figure 3.10: Flow diagram for the experimental rig used to validate the [CFD](#) results.

The transient heat transfer method, together with a coating of thermochromic liquid crys-

tals, were used to measure the heat transfer coefficients over the impingement surface [131], [132], [265]. These crystals respond to a change in temperature by altering their optical properties. For the narrow band variety used, this occurs over a small temperature interval (0.5°C), allowing an accurate map of temperature to reflected light to be achieved; often involving the green signal. Crystals with differing colour play temperatures can be mixed and sprayed onto the surface from which heat transfer coefficient measurements are to be made. The crystals are encapsulated in polymer micro-capsules by the manufacturer and then sprayed onto a black background to enhance visible contrast. For this research, crystals with quoted band temperatures centred around 25 and 30°C were used.

The response of the perspex surface to heating is modelled using the solution to the transient one-dimensional Fourier equation for a semi-infinite solid under conditions of a step change in the driving gas temperature. Electrically heated meshes are placed in each inlet channel to achieve this temperature step change. The DC power to these meshes was supplied using a 600 W inverter powered from the mains. For complete information about the data processing for this established experimental method see [122].

3.5.1.1 Data Capture

Gas temperature thermocouples were placed in the jet orifices to record the variation in driving gas temperature throughout the experiment. Additional gas thermocouples were also placed at other locations in the assembly, and thin-foil thermocouples fixed to the cavity impingement surfaces. These latter thermocouples were used to obtain a temperature against green intensity calibration curve for the thermochromic liquid crystals. The importance of performing this calibration is evident from Fig. 3.11. If the colour play temperature quoted by the manufacture had been used there would have been a discrepancy at the lower temperature of $\sim 1.5^{\circ}\text{C}$. Both thermocouples and mesh voltage readings were recorded using a data logger at a frequency of 40 Hz ⁶.

Video data for the liquid crystals was recorded using a Complementary Metal-Oxide-

⁶OM-DAQ-USB-2400

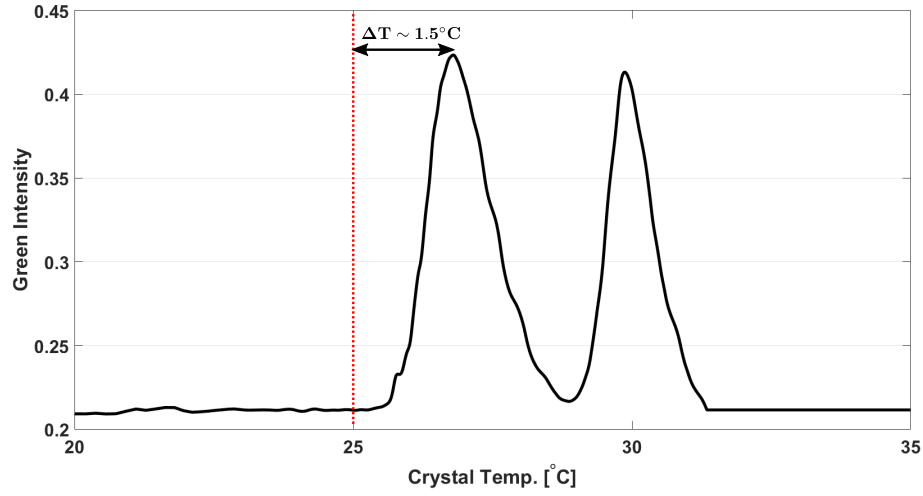


Figure 3.11: Calibration curve for the thermochromic liquid crystals sprayed onto the heat transfer surface. Green intensity is plotted against the temperature recorded by the surface thermocouple.

Semiconductor (CMOS) camera⁷ at a rate of 32 frames per second. An LED was placed in the capture frame that was activated once power was supplied to the heater meshes. This allowed time synchronisation of the video and thermocouple data for subsequent analysis. The experiment was limited to a maximum run-time of 50 sec to ensure the validity of the transient assumption [177]. A CAD drawing of the test section, along with a typical image of the impingement surface during a run, is shown in image (a) of Fig. 3.12

In image (b) of Fig. 3.12 local data across each of the cavities is displayed for a jet Reynolds number of $\sim 40,000$. For either the primary or secondary cavities the variation between results was found to be smaller than $\sim 2\%$. The author found that this also held for other jet Reynolds numbers, and when the heat transfer was averaged over the cavity surfaces. For this reason, in the following sections, results are only presented for the primary cavity 1a and the secondary cavity 2a.

The cavity invariance observed in image (b) of Fig. 3.12 seems contradictory to the liquid crystal distribution in image (a). However, this can be explained for the primary cavities as follows: 1) each cavity has a separate feed, which can lead to variations in the mass flow

⁷Point Grey Blackfly PGE-13E4C-CS

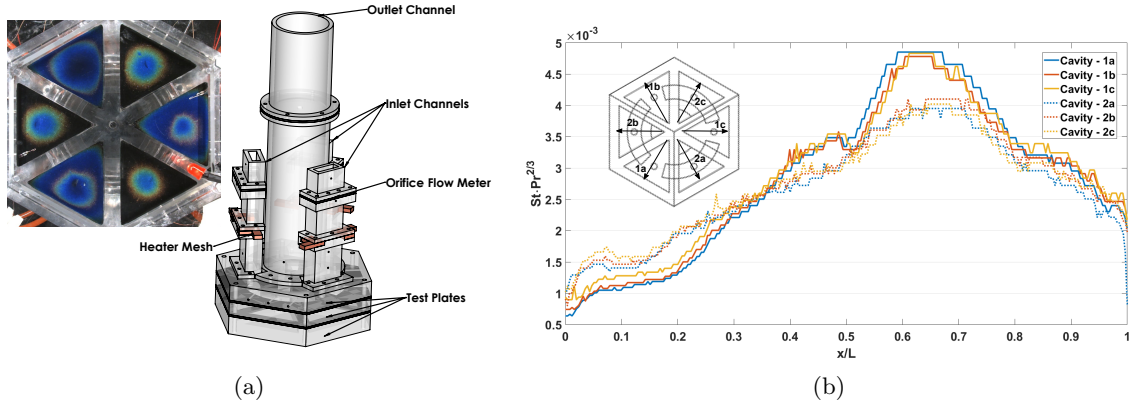


Figure 3.12: (a) [left] Typical image of the liquid crystal coating during an experimental run. As the liquid crystal transitions through its colour play temperature it reflects red, green and then blue light. The green intensity is measured in these experiments to determine the change in surface temperature with time. [right] Experimental test section set-up for the thermochromic liquid crystal heat transfer measurements. (b) Comparison of the spatial variation in the heat transfer across each of the cavities for a jet Reynolds number of $\sim 40,000$.

rates and driving temperatures for each jet; 2) this changes the power deposited into the liquid crystals and, consequently, the temperature rise time. For the secondary jets a similar argument holds, but the variation is now solely caused by different driving temperatures. When these factors are taken into account, the variation of heat transfer with mass flow rate is found to only depend on whether it is a primary or secondary cavity.

Separate steady-state tests were run to gather pressure drop data. Static pressure tapings were placed throughout the assembly and readings taken using a digital manometer.

3.5.2 Uncertainty Analysis

To estimate the uncertainty in the heat transfer coefficient data the small perturbations method of Moffat [186] was used. The partial derivatives for each variable were found numerically using the central differencing method. This is expressed mathematically for a variable, χ_i , as [44]

$$\frac{\partial h_c}{\partial \chi_i} = \frac{h_c(\chi_i + \delta \chi_i) - h_c(\chi_i - \delta \chi_i)}{2\delta \chi_i},$$

where $\delta\chi_i$ is a small perturbation in the chosen variable. The expected percentage uncertainty in the heat transfer coefficient, ϵ_i , for each χ_i can be estimated from

$$\epsilon_i = \left(\frac{\partial h_c}{\partial \chi_i} \right) \frac{\Delta \chi_i}{h_c}.$$

Here $\Delta\chi$ is the uncertainty in the variable measurement, and the value used for h_c should be taken as the most likely result. An estimate of the overall uncertainty was then made using the root sum square method, such that

$$\epsilon_{tot} = \sqrt{\sum_{i=1}^n \left[\left(\frac{\partial h_c}{\partial \chi_i} \right) \frac{\Delta \chi_i}{h_c} \right]^2}.$$

The results of these analyses for the area averaged heat transfer coefficients in the primary and secondary jet cavities are shown in Table [3.2](#)

The measurement uncertainties obtained for the temperatures were achieved by using a reference platinum/palladium thermocouple, that was calibrated by the [National Physical Laboratory](#) (NPL). For the measurement uncertainty in the product of the volumetric heat capacity and thermal conductivity of the Perspex, data from [\[131\]](#) was used. The data acquisition sampling period was used for the time measurement uncertainty.

The largest uncertainty for both cavity jets is in the temperature measured by the liquid crystal coating. This uncertainty is approximately 38% higher for the cavities fed by the secondary jets, which is caused by the reduced jet driving temperature - the heat transfer coefficient is dependent on the temperature difference between the wall and jet. As this difference decreases, uncertainties in either can cause a large percentage change in the calculated heat transfer coefficient.

Parameter	Typical Value	Measurement Uncertainty	$\frac{\partial h_c}{\partial X}$	$\frac{\partial h_c}{\partial X} \frac{\Delta X}{h_c}$
Primary Cavity				
T_{crystal}	35°C	0.3°C	44	0.06
T_j	40°C	0.2°C	-51	-0.046
$\sqrt{\rho c k}$	569 Jm ⁻² K ⁻¹ s ^{-0.5}	29 Jm ⁻² K ⁻¹ s ^{-0.5}	0.34	0.047
t	50 s	0.03 s	44	0.006
RSS Uncertainty [%]				8.9
Secondary Cavity				
T_{crystal}	30°C	0.3°C	62	0.082
T_j	35°C	0.2°C	-86	-0.075
$\sqrt{\rho c k}$	569 Jm ⁻² K ⁻¹ s ^{-0.5}	29 Jm ⁻² K ⁻¹ s ^{-0.5}	0.54	0.069
t	50 s	0.03 s	-25	-0.0035
RSS Uncertainty [%]				13

Table 3.2: Uncertainty analysis for the area averaged heat transfer coefficients in the cavities for the primary and secondary jets. Area averaged reference heat transfer coefficient ~ 225 W/m²K.

3.5.3 Results

3.5.3.1 Area Averaged and Stagnation Point Convective Heat Transfer

Experimental heat transfer coefficient data were processed using a MATLAB code developed by [177]. As flow rates through the secondary jets could not be measured individually, an even split of the total mass flow was assumed. This was considered reasonable, as the model was carefully assembled and its components machined to tight tolerances; ~ 0.1 mm for features.

The thermo-physical properties of air were based on the driving gas temperature, rather than the film temperature. As the step change from the heater mesh was limited to $\sim 40^\circ\text{C}$, this introduced negligible uncertainty. The Prandtl number was fixed at a value of 0.71.

Comparative CFD simulations were run for each of the turbulence models discussed in Sec. 3.3.2. The heat flux was reduced to 2 MW/m² to ensure single-phase flow for the range

of inlet mass flow rates considered. All other data were processed using the methodology set out in Sec. 3.4

The area averaged j_H were calculated from the experimental results for both the primary and secondary jet cavities. These results are compared against the CFD in Fig. 3.13 for varying Reynolds numbers.

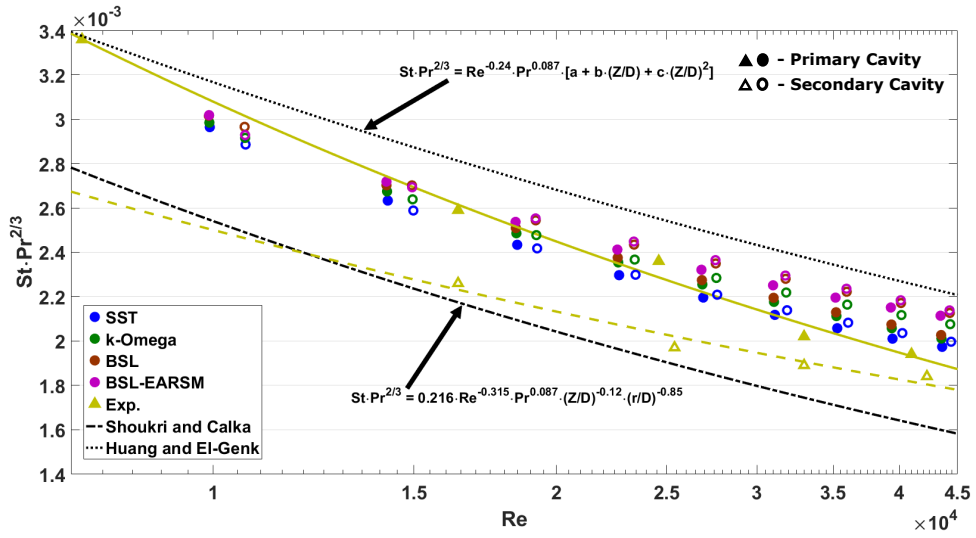


Figure 3.13: Variation of the area averaged j_H with Reynolds number. Primary jet cavity CFD and experimental data shown with solid markers, hollow markers used for secondary jet cavity data. Correlations from the literature have also been plotted [113, 253].

Focussing initially on the variation between turbulence models; there is a trend of increasing spread in the results with Reynolds number for both jet cavities. This is caused by two mechanisms: (a) the jet turbulence intensity; (b) the entry length. For all turbulence models the primary jets had their turbulence intensity set to 5% at the inlet boundary. Area averaged values of this quantity for the SST model over a plane one jet diameter from the exit decreased by only 22% as the Reynolds number increased from $\sim 10,000$ -40,000. In contrast, for the secondary jets, the area averaged turbulence intensity over an equivalent plane increased almost three-fold for the same Reynolds number range. This indicates that the spread in the primary jet results is driven by (b). In contrast, for the secondary jets, the strong dependence of the turbulence intensity on the specific turbulence model leads to

the increased spread observed. Experimentally it has been found that a higher turbulence intensity reduces the jet core length, which reduces the optimum Z/D [313].

For a particular turbulence model, there is minimal difference between the area averaged convective heat transfer in the primary and secondary jet cavities. The lowest absolute values are obtained using the SST model, and the highest with BSL-EARSM. The dependence of the convective heat transfer on the Reynolds number is predicted to be slightly reduced for the secondary jet cavity.

All of the turbulence models are able to correctly predict the area averaged heat transfer for the primary jet cavity to within experimental uncertainty. The experimental results show an increased dependence on the Reynolds number in comparison to the CFD.

Only the SST model is able to correctly predict the area averaged convective heat transfer in the secondary jet cavity to within experimental uncertainty. There is also seen to be a more significant drop in Reynolds number dependence than predicted by the CFD.

The two correlations plotted in Fig. 3.13 are for a single jet in both confined [253], and un-confined [113], configurations. The literature shows that confining a jet reduces its heat transfer performance [202], which is corroborated here. The correlation by Shoukri and Calka is found to be accurate to within $\sim 20\text{-}25\%$ for both sets of cavity results.

A comparison between the CFD and experimental results for the stagnation point convective heat transfer in the primary and secondary jet cavities is shown in Fig. 3.14. In both cases it is evident that the numerical results over-predict the experimental values. This trend has been observed in other experimental research on jet impingement heat transfer [259].

All of the turbulence models over-predict the peak convective heat transfer in the primary jet cavity by 50-70%. For the secondary jet cavity the difference reduces to 30-50%. The

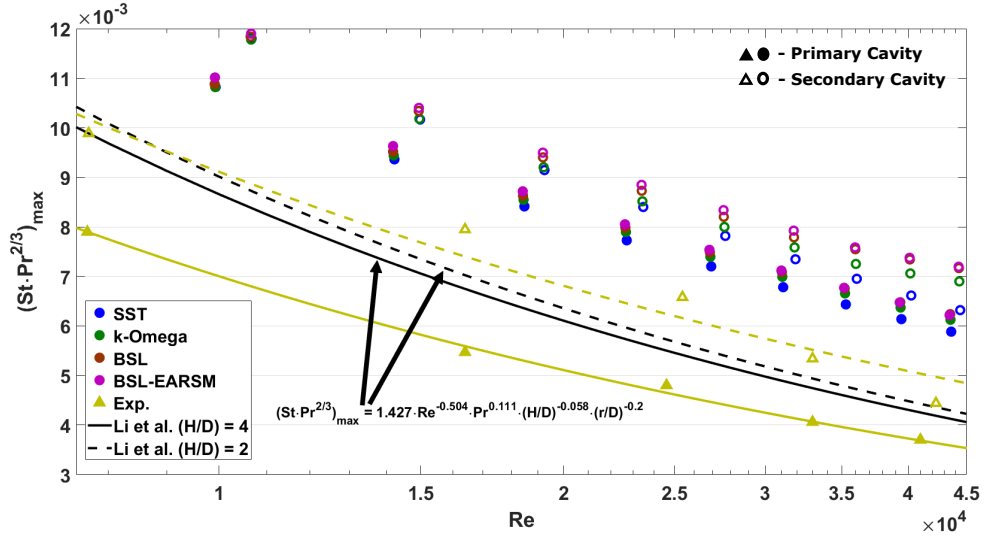


Figure 3.14: Variation of the stagnation point j_H with Reynolds number. Primary jet cavity **CFD** and experimental data shown with solid markers, hollow markers used for secondary jet cavity data. A correlation from the literature has also been plotted for different jet entry length ratios (H/D) [157].

percentage discrepancy increases for the primary jet cavity, but decreases for the secondary jet cavity, at higher Reynolds numbers. The **SST** turbulence model again offers the best match to the experimental results.

The **CFD** successfully predicts that the stagnation point heat transfer will be higher in the secondary jet cavities. However, it underestimates the percentage difference by 10-20%.

A correlation by [157] was also plotted in Fig. 3.14 for two different jet entry length ratios. It is able to predict the stagnation point heat transfer for both jet cavities to within $\sim 20\%$. Additionally, it captures the behaviour of increasing stagnation point heat transfer with reducing jet entry length found in the experimental results.

3.5.3.2 Local Heat Transfer

To investigate the differences between the experimental and **CFD** results further, local data were plotted along altitude lines for each jet cavity at a jet Reynolds number of $\sim 40,000$. Two altitude lines were considered for each jet cavity starting at vertices 1A, 2A, 1B and

2B - see Fig. 3.7 from Sec. 3.4. Fig. 3.15 shows the results for altitude lines running from vertices 1A and 2A.

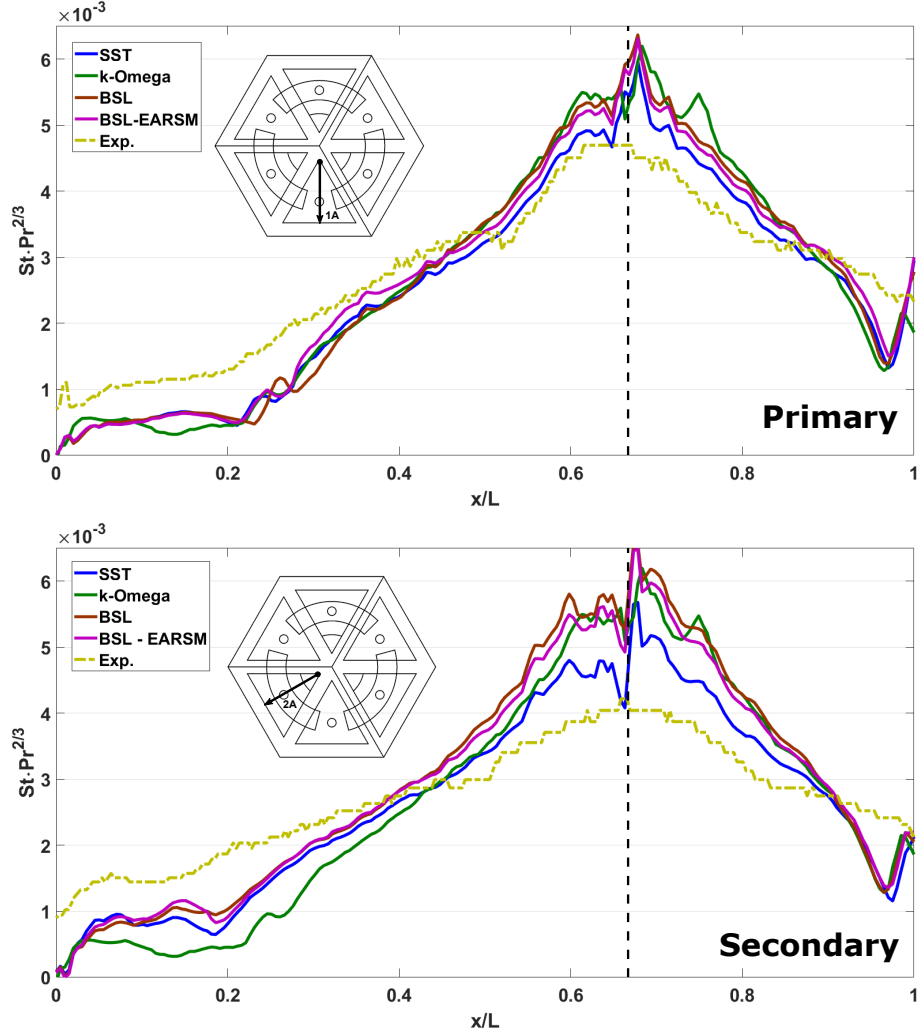


Figure 3.15: Local distribution of j_H for both the primary and secondary jet cavities along altitudes originating from vertices 1A and 2A. Data is shown for a range of turbulence models against the experimental results. A dotted line is shown to represent the jet centreline.

Considering the primary jet cavity results first: the experimental distribution agrees closely with the **CFD** in the stagnation region for all of the turbulence models. However, at the stagnation point, the **CFD** over-predicts the heat transfer, which leads to the observed spikes in the profiles. This is a result of the difficulty of accurately resolving the large velocity gradients at this point.

In the secondary jet cavity stagnation region there is a more significant discrepancy between the experimental and [CFD](#) profiles. The experimental distribution is broader than in the primary jet cavity, and the [CFD](#) only partially captures this. Such broadening of the stagnation region has been observed by [\[313\]](#) and is due to the reduced jet entry length.

In both experimental profiles there is a noticeable transition from a laminar to turbulent boundary layer in the wall jet region at a normalised distance of approximately 0.5. For normalised lengths below 0.5 the [CFD](#) under-predicts the recorded heat transfer for all turbulence models by up to $\sim 250\%$. This change in profile, caused by boundary layer transition was studied in detail by [\[91\]](#).

Above a normalised distance of ~ 0.95 there is a drop in the heat transfer predicted by the [CFD](#) for both jet cavities, followed by a recovery. This is due to flow separation, and subsequent re-attachment, caused by the un-filleted corner at the edge of the cavity. In the experimental results this is not observed because the fabricated perspex components had filleted edges. The influence of impingement surface roughness fillet radii on convective heat transfer has been investigated thoroughly by [\[260\]](#).

The close agreement between [CFD](#) and experimental area averaged heat transfer values for both jet cavities indicates that over predictions and under predictions must approximately cancel out. Local heat transfer profiles are most accurately re-produced by the [SST](#) turbulence model.

Results for the local profiles along altitudes originating from vertices 1B and 2B are shown in [Fig.3.16](#).

In comparison with the previous results for the primary jet cavity, there is a more significant under-prediction of the heat transfer in the wall jet region - up to $\sim 300\%$ in some cases. The situation is reversed for the secondary jet cavity where the [BSL-EARSM](#) turbulence model matches the experimental profile to within approximately 40% for $0.1 < x/L < 0.4$.

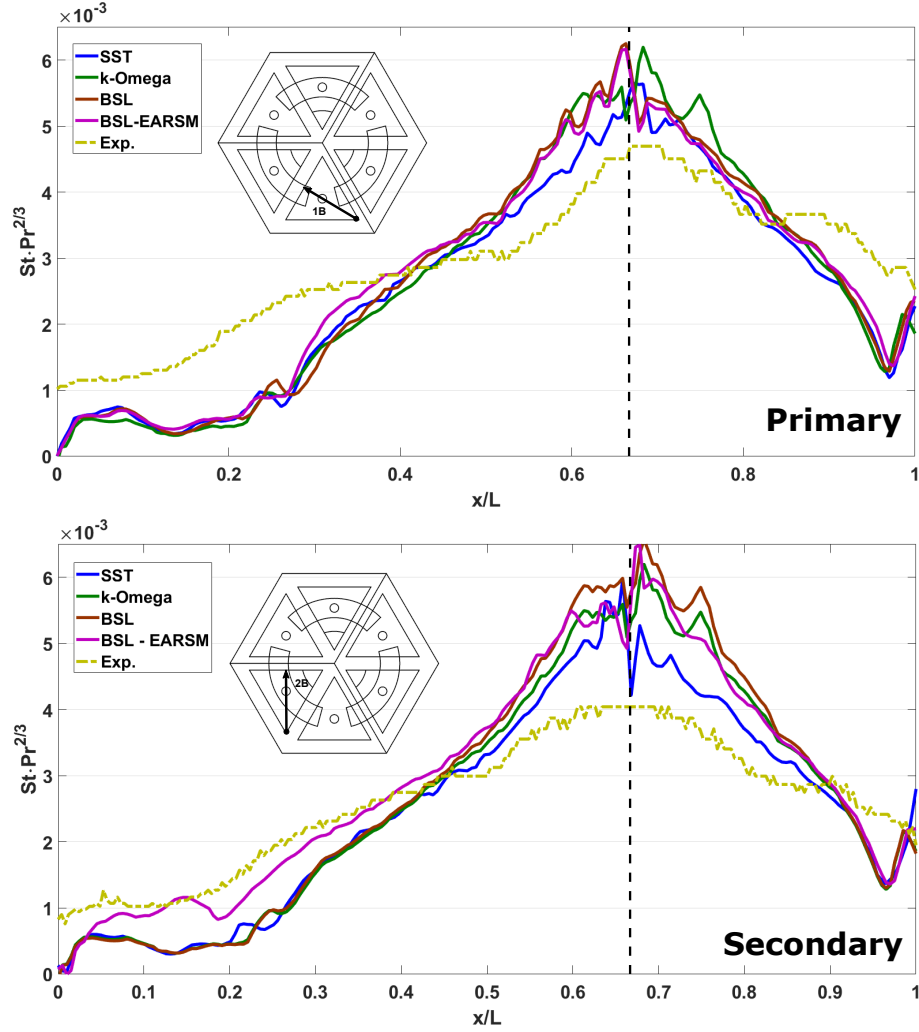


Figure 3.16: Local distribution of j_H for both the primary and secondary jet cavities along altitudes originating from vertices 1B and 2B. Data is shown for a range of turbulence models against experimental results. A dotted line is shown to represent the jet centreline.

In both Figs. 3.15 & 3.16 the jet centreline has also been indicated with a vertical dashed line at an $x/L \approx 0.66$. As this co-insides with the location of peak heat transfer for the experimental results, it confirms that there is no significant cross-flow influencing the impact location and trajectory of the jets.

Fig. 3.17 shows contour plots of j_H at a fixed Reynolds number of $\sim 40,000$ for the CFD and experimental results.

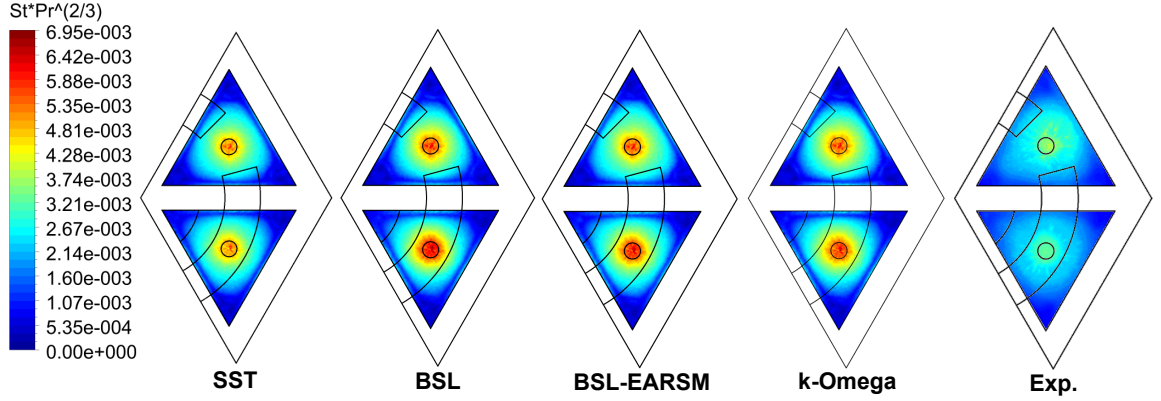


Figure 3.17: Comparative contour plots of j_H for **CFD** and experimental results. Data is for a fixed fixed Reynolds number of 40,000. Turbulence models: k- Ω , **SST**, **BSL** and **BSL-EARSM**.

The experimental heat transfer profiles are flatter than those predicted by the **CFD**. This is beneficial from a thermal stress perspective as it will result in reduced solid temperature non-uniformity.

3.5.3.3 Pressure Drop

In this section the pressure drop performance of the HPJC-1a is considered. The Euler number, Eu , is plotted against the Reynolds for the experimental and **CFD** results in Fig. 3.18. Thermo-physical properties were taken at the average bulk temperature of the fluid, and the cross-sectional area of the jet used as the reference area.

The average discrepancy between experiment and **CFD** is in the region of $\sim 13\%$. This is above the measurement uncertainty of 0.1%. From the **CFD**, approximately 91% of the total pressure drop through the system occurs across the jets, and in the impingement cavity regions. A study by [211] considered the influence of manufacturing tolerances on the total pressure drop of a jet impingement system. They found that an uncertainty in the jet diameter of $\pm 2.5\%$ resulted in a corresponding change in the total pressure drop of $\pm 35\%$. The uncertainty in the jet diameter from the machining methods used in this research is $\pm 1\%$. It is therefore conceivable that the discrepancy between the numerical and experimental pressure drop measurements is caused by this uncertainty.

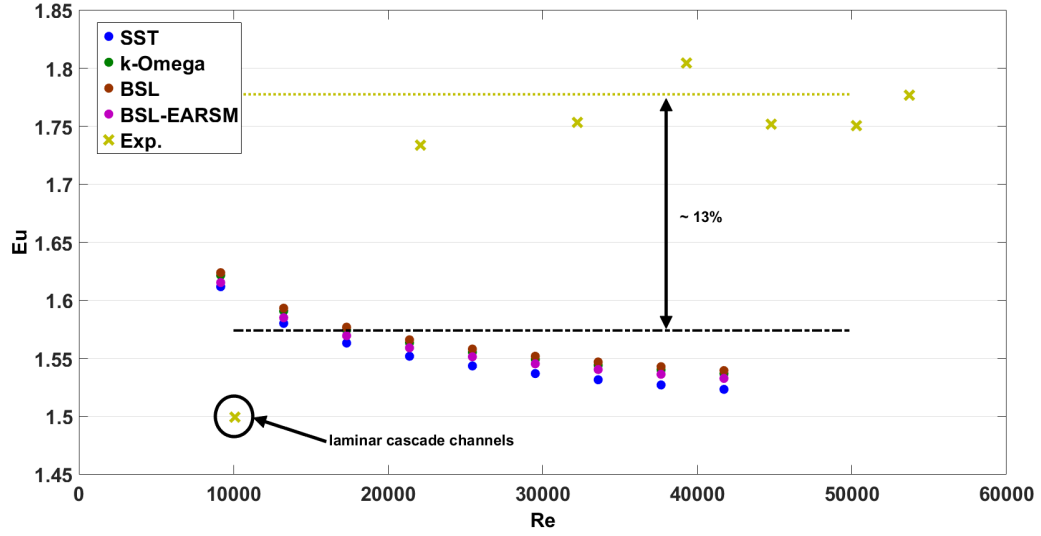


Figure 3.18: Variation of Euler number with Reynolds number for experimental and CFD results. Turbulence models: k- Ω , SST, BSL and BSL-EARSM.

There is little variation in the results from the different turbulence models. All predict a slight reduction in Euler number with increasing Reynolds number. This is in contrast to the experimental results, which predict a Euler number that is relatively Reynolds number invariant above 10,000. The experimental point at 10,000 is due to regions of flow through the cascade plate being in the laminar regime. This step change in Euler number, due to transition in the cascade flow from laminar to turbulent, has been observed in later work involving un-scaled models of a HPJC-type design - see Sec. 9.3.2 of Chpt. 9.

3.6 Summary

A heat sink component has been developed for HHF applications in a first stage DEMO divertor target plate. The design is composed of a number of laminate sheets that are individually machined and bonded together in a stack. This greatly facilitates manufacture and allows complex flow paths to be produced in a simplistic manner.

Water at 20 MPa is employed as the coolant in a cascade jet impingement flow path arrangement. This method of deploying the flow allows increased pressure drop to be exchanged for a reduced mass flow rate. By operating the coolant in this manner the temperature change

of the fluid can be increased, which can aid in improving the thermal cycle efficiency.

The general conclusions from this chapter on the numerical modelling, and experimental validation, of the thermo-fluid performance of the HPJC-1a design are as follows:

- For DEMO relevant coolant inlet conditions the numerical results indicate that the design is capable of handling divertor target plate steady-state heat flux magnitudes;
- When compared at an equivalent heat flux, the HPJC-1a was shown to have a reduced pumping power and mass flow rate in comparison to an ITER-like DEMO. This was for a matched value of the peak heat sink temperature minus the coolant inlet temperature;
- The CFD was validated against experiment by performing heat transfer and pressure drop measurements. The results of this comparison were as follows:
 - The SST turbulence model correctly predicted the experimental area averaged convective heat transfer to within the measurement uncertainty of 9-13%;
 - Turbulence intensity was over predicted by the CFD in the secondary jet. The discrepancy was most pronounced at low Reynolds numbers;
 - The stagnation point heat transfer levels were over predicted by the CFD by up to 70%;
 - In the wall jet region the CFD under predicted the experimental results by up to 300%;
 - The SST turbulence model provided the most accurate fit to the experimental convective heat transfer data, particularly at a local level around the stagnation point;
 - Experimental heat transfer profiles were found to be flatter than those predicted by the CFD;
 - All of the turbulence models failed to predict the transition from a laminar to turbulent boundary layer in the wall jet region, which was found in the experimental results;

- The experimental Euler number was $\sim 13\%$ above the CFD results. This was attributed to the uncertainty in the machined jet diameter.

Chapter 4

A Performance Figure of Merit

“The first principle is that you must not fool yourself, and you are the easiest person to fool.”

— Richard Feynman

Figures of merit to facilitate comparison between **HHF** components, and simplify their development, are currently lacking in the field of nuclear fusion. The majority of effort to date has focused around specific designs.

In the field of gas turbines, a cooling efficiency figure of merit was developed by colleagues Holland and Thake [112]. They considered the case of a gas turbine blade supplied with coolant at a temperature T_{ci} , and subject to a convective heat flux driven by a hot gas at a temperature T_g . Their analysis showed that the thermal performance of the cooling system can be measured by a cooling effectiveness parameter, ϵ_s , given by

$$\epsilon_s = \frac{T_g - T_s}{T_g - T_{ci}}.$$

When the component temperature, T_s , is equal to the coolant inlet temperature, the cooling effectiveness is unity. In contrast, as the component temperature approaches the gas temperature it tends to zero.

The cooling effectiveness is often used as an initial metric to determine whether new turbine blade cooling designs offer a performance advantage over current concepts, or as a means of estimating the required coolant flow for new engine cycle conditions, i.e. T_{ci} and T_g . In their paper, Holland and Thake [112] showed how increasing cooling effectiveness coincided with the development of improved gas turbines. The figure from their paper is reproduced in Fig. 4.1.

The image(s) originally presented here cannot be made freely available by ORA due to copyright.

Figure 4.1: Turbine blade cooling effectiveness (ϵ_m) plotted against a non-dimensional mass flow rate (w^+) for various jet engines. Lines of constant cooling efficiency (η_c) have also been added. Reproduced from [112].

The cooling effectiveness serves as a valuable performance measure for cases where the flow of heat into the component is convective in nature. Although a plasma temperature can be defined in a tokamak, it is so high that any variation in the component wall temperature has little impact on the heat flux. In this chapter, the author derives a figure of merit for cooling performance for the case of a heat flux boundary condition. The derivation closely follows the work of Holland and Thake, but differs due to the fact that there is no natural

maximum temperature for a system with a heat flux boundary condition. This means that a poor cooling system could, theoretically, result in an infinite component temperature. No attempt is made to incorporate pumping power into this figure of merit at this stage.

4.1 Figure of Merit Derivation

4.1.1 The Thermally Ideal Heat Sink

To determine the merit of a cooling system a point of reference is required. To fulfil this role the author proposed the notion of a thermally ideal heat sink component that has the following characteristics:

- i A structure composed of a material(s) with an infinite thermal conductivity; ensuring that it is isothermal;
- ii An internal flow geometry such that the coolant exits at the component temperature; maximising the heat transfer coefficient for a given coolant mass flow rate.

The isothermal temperature profile for the ideal component means that there is no thermal resistance due to conduction in the solid. For real components this is not the case, and **HHFs** can lead to significant temperature non-uniformities in the material.

The first law of thermodynamics ensures that the required heat capacity rate, defined as the mass flow rate multiplied by the specific heat capacity, is minimised when the coolant leaves at the maximum component temperature. In turn, the product of the heat transfer coefficient and the area needs to be adequate to ensure the required temperature increase.

4.1.2 Introduction of a Dimensionless Temperature Difference Ratio

The development of the figure of merit begins with consideration of statement (ii) from the definition of the thermally ideal heat sink component in Sec. **4.1.1**. To simplify the analysis, it is assumed that the thermally ideal heat sink operates with a single-phase coolant, and that steady-state conditions apply. For a given heat input, $\dot{Q}$, to the system the first law

gives

$$\dot{Q} = \dot{m}_{id} c_{p_{id}} (T_s - T_{ci}). \quad (4.1)$$

Here T_s represents the component temperature, which is uniform since the thermal conductivity is infinite. The subscript ‘*id*’ indicates ideal component quantities with $\dot{m}_{id} c_{p_{id}}$ being the heat capacity rate for the ideal case considered.

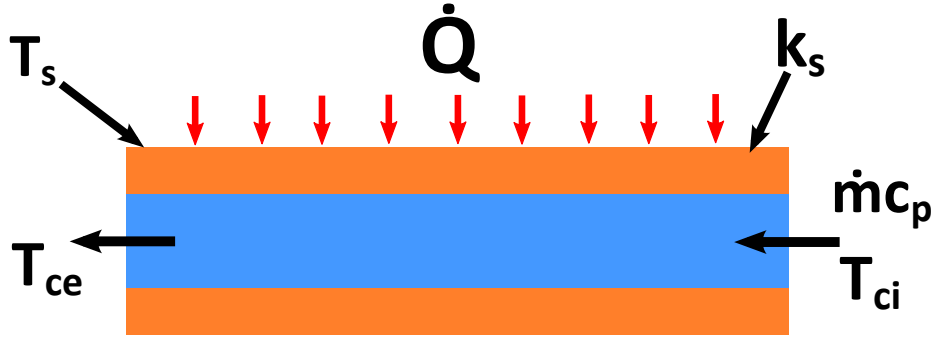


Figure 4.2: Real heat sink with key thermo-fluid variables indicated.

The figure of merit compares a real system, depicted in Fig. 4.2, to the ideal case. To make the comparison, the following constraints are applied:

- i A reference component temperature in the real case is equal to the uniform component temperature in the ideal case;
- ii The coolant inlet temperature is the same for both the ideal and real cases;
- iii The heat input is equivalent in the ideal and real systems.

If it is also assumed that the real system is cooled by a single-phase coolant, then statement (iii) can be expressed as

$$\dot{m} c_p (T_{ce} - T_{ci}) = \dot{m}_{id} c_{p_{id}} (T_s - T_{ci}),$$

where T_{ce} is the coolant exit temperature in the real case.

A high performing cooling system will minimise the heat capacity rate required to obtain a specified T_s , for a given heat input. This statement is encapsulated in a figure of merit termed the wall overheat (τ_s), defined by

$$\tau_s = \frac{\dot{m}c_p}{\dot{m}_{id}c_{p_{id}}} = \frac{T_s - T_{ci}}{T_{ce} - T_{ci}} = \frac{\text{maximum possible heat extraction}}{\text{actual heat removal}}. \quad (4.2)$$

The following observations can be made about the wall overheat:

1. τ_s is used to render dimensionless the component reference temperature, T_s , of choice. Sensible options could be the maximum, volume or surface averaged temperature;
2. A good cooling system will have a low wall overheat, and so will plot low on a chart of τ_s versus coolant flow rate;
3. The value of τ_s is unity only for the ideal component, and will normally¹ exceed this for real components.

There are obvious parallels between heat exchanger, and heat sink, design. However, the wall overheat differs from the effectiveness parameter used in the Number of Transfer Units (NTU) method [120], Chpt.11] for heat exchanger design in an important way. A high effectiveness is associated with the efficient transfer of heat from one fluid to another. In contrast, a low wall overheat corresponds to minimising the mass flow rate required to maintain a specified component temperature for a given heat input.

¹If the component reference temperature selected is not the maximum, then there is the possibility that the wall overheat could be less than unity.

4.1.2.1 Dependencies of the Wall Overheat

Using equation (4.2) in equation (4.1) we can also express the wall overheat as

$$\tau_s = \frac{\dot{m}c_p(T_s - T_{ci})}{\dot{Q}}. \quad (4.3)$$

In a real component the temperature varies through the solid material, and a temperature jump occurs at the convective heat transfer surface between the wall and the fluid. For the steady-state 1D heat equation, with convective and heat flux boundary conditions, this thermal resistance, R_t , can be represented as

$$R_t = \frac{1}{h_c} + \frac{t}{k_s}. \quad (4.4)$$

Here the convective heat transfer coefficient is referenced to the coolant inlet temperature. The quantity k_s is the thermal conductivity of the component material and t is an associated material thickness.

The thermal resistance in equation (4.4) can be used to express the temperature difference in the numerator of (4.3) via

$$T_s - T_{ci} = \frac{\dot{Q}}{A}R_t, \quad (4.5)$$

where the variable A is the area through which the heat input to the real component is applied.

Equation (4.5) can be re-arranged into a more useful form by introducing a Biot number, Bi , given by

$$Bi = \frac{h_c t}{k_s}. \quad (4.6)$$

Combining equations (4.6) and (4.5) gives

$$\begin{aligned} T_s - T_{ci} &= \frac{\dot{Q}}{A} \left(\frac{1}{h_c} + \frac{t}{k_s} \right) \\ &= \frac{\dot{Q}}{A} \frac{t}{k_s} \frac{(1 + Bi)}{Bi}. \end{aligned} \quad (4.7)$$

Returning to equation (4.3), and using (4.7), the wall overheat becomes

$$\tau_s = \frac{\dot{m} c_p t}{k_s A} \frac{(1 + Bi)}{Bi}. \quad (4.8)$$

This expression can be re-arranged by introducing a non-dimensional mass flow rate, m^* , given by

$$m^* = \frac{\dot{m} c_p}{k_s \sqrt{A}} = \frac{\text{potential of fluid for heat extraction}}{\text{potential of solid for heat conduction}}. \quad (4.9)$$

If equation (4.9) is then substituted into equation (4.8), the wall overheat becomes

$$\tau_s = \frac{t}{\sqrt{A}} \frac{(1 + Bi)}{Bi} m^*. \quad (4.10)$$

It is also useful to determine a form for the wall overheat that relates it directly to the convective heat transfer performance. To do this a Stanton number, St , is defined as

$$St = \frac{h_c A}{\dot{m} c_p}. \quad (4.11)$$

In combination with m^* , equation (4.11) can be used to express the Biot number as

$$Bi = \frac{t}{\sqrt{A}} St \cdot m^*. \quad (4.12)$$

Combining equations (4.11) and (4.12) into (4.10) the wall overheat becomes

$$\tau_s = \frac{1 + Bi}{St} = \frac{t}{\sqrt{A}} m^* + \frac{1}{St}. \quad (4.13)$$

From equation (4.13) it can be seen that a low wall overheat corresponds to a minimum value of m^* , and a maximum value of the Stanton number. The Stanton number is a measure of the heat transferred to the fluid in comparison to its thermal capacity. By achieving a high heat transfer coefficient, at a low thermal capacity rate, this quantity can be maximised.

The form of the wall overheat parameter given in equations (4.10) & (4.13) is only applicable for the 1D model considered. However, it can be shown via dimensional analysis that, for a given component geometry

$$\tau_s = \mathcal{F}(Bi, m^*), \quad (4.14)$$

where $\mathcal{F}$ represents a general functional dependence. This allows the wall overheat parameter to be used to compare the cooling performance of any component by producing a plot of m^* against τ_s . This is done in the proceeding section for heat sink components from a range of fields.

4.2 Application of the Wall Overheat as a Means of Comparing Cooling Systems

The wall overheat parameter provides a simple means of comparing different HHF cooling concepts. However, to apply it effectively, there must be consistency in how the reference

values are chosen across designs.

In this work the peak temperature of the heat sink was used for T_s , and as the reference value for solid material properties. As a heat sink often only forms a part of a composite component, this will be an oversimplification. However, it enables the wall overhear to be used to compare a much broader range of HHF designs. In cases where the engineer is interested in a specific family of designs, an adapted version of the wall overhear could be used. This may involve a length-scale weighted thermal conductivity for composite structures, or an alternative selection for T_s .

The reference area for the heat flux is set as the surface of the component through which the heat input initially passes. For guidance on what is meant by this, the reader is referred to the fusion divertor component shown in Fig. 4.3. The reference area for the heat flux in this case would be the top of the monoblock, and the value of T_s would be taken from the external surface of the CuCrZr tube.

The image(s) originally presented here
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Figure 4.3: Typical HHF component geometry used for the divertor target plate region of the ITER fusion tokamak [221].

In the whole of this work the thermo-physical properties of the coolant are taken at the inlet state. If large fluid property changes occur through the component, averaged values may be more appropriate.

As an introduction to the utility of the wall overheat as a measure of thermal performance, Fig 4.4 was produced. It compares a range of HHF cooling methods from a number of fields [94, 105, 110, 126, 129, 135, 152, 167, 207, 216, 240, 255, 261, 308].

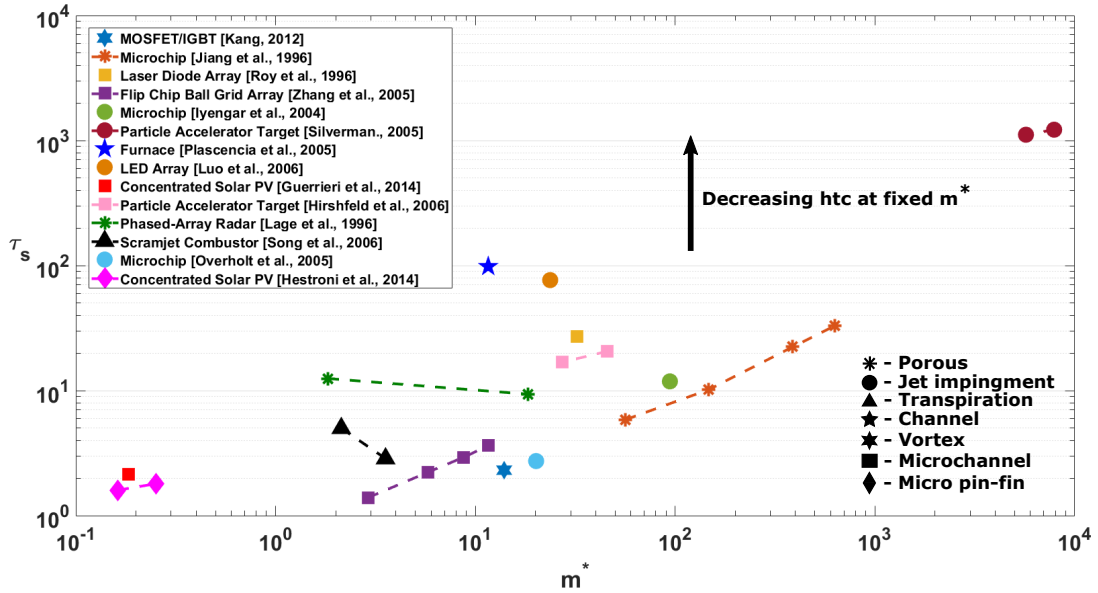


Figure 4.4: Variation of the wall overheat with non-dimensional mass flow rate for a range of HHF cooling methods and fields of application [94, 105, 110, 126, 129, 135, 152, 207, 216, 240, 255, 261, 308]

For the majority of components there is an increase in wall overheat with m^* . The two components that do not follow this trend - phased-array radar, scramjet combustor - both have high effective through-thickness thermal conductivities. In the scramjet lamilloy geometry the coolant travels towards the heated surface through a macro-porous structure. As the coolant flow rate is increased, the temperature change of the coolant through the lower layers of this porous structure will decrease. This reduces the coolant temperature near the heat flux boundary wall. When this effect is combined with the higher heat transfer coefficients from the increased mass flow rate, the reduction in peak heat sink temperature

more than compensates for the reduced coolant temperature change.

Though direct comparison at equivalent m^* values is not possible for all the cooling methods, Fig. 4.4 does point towards Pareto law type behaviour. This indicates there is a limit to the minimum τ_s a particular design can achieve, for a given m^* . By letting $St \rightarrow \infty$ in equation (4.13), this minimum for the 1D model case becomes $\tau_s = \frac{t}{\sqrt{A}}m^*$.

It is worthwhile considering at this stage what makes a good HHF heat sink based on the wall overheat methodology. From the 1D component model in Sec. 4.1.2.1 a good HHF heat sink is one which minimises the thermal resistance both through, and at, the wall using the minimal amount of coolant. In any real component we can minimise thermal resistance in the solid by: 1) minimising conduction path lengths from the heat flux boundary to the fluid; 2) employing high thermal conductivity materials; 3) avoiding geometric heat flux peaking. For the thermal resistance at the internal wall, the main criteria is to maximise the heat extraction at a given coolant flow rate, which can be achieved geometrically via the following two routes: 1) maximising the internal wetted area; 2) maximising the heat transfer coefficient for a given coolant flow rate. A uniform heat transfer coefficient over the internal wetted area is also desirable to limit temperature non-uniformities in the solid.

The concentrated solar photovoltaic system that employs a micro pin-fin array achieves a low wall overheat through a combination of large internal wetted area, small hydraulic diameters, and the use of a copper alloy structural material. In contrast to this, the jet impingement system designed to cool a particle accelerator target has a much larger wall overheat. From a convective heat transfer perspective this is caused by the larger hydraulic diameters employed in this system. Conductive resistance is also increased in this design due to the use of a thicker, and less thermally conductive, solid material.

The choice of coolant for a single-phase system also plays a role in determining the wall overheat. For a fixed mass flow rate, the fluid should preferably have a low specific heat to allow a high temperature change, coupled with a high thermal conductivity and low

dynamic viscosity. In fact, reducing the specific heat is simply equivalent to traversing to lower m^* values on the same curve.

4.3 Investigation of the Effect of Varying Thermal Conductivity and Heat Flux on the Wall Overheat

4.3.1 Methodology

A **HHF** component in a tokamak may see considerable variation in heat flux boundary conditions, and material properties, during its lifetime. It would be invaluable to the engineer to be able to predict the thermo-fluid performance for such varying conditions using a single design curve. To study this possibility for the wall overheat parameter, the HPJC-1a concept from Chpt.3 was used as a reference design.

The numerical methodology used was the same as that outlined in Sec.3.3.1 of Chpt.3. For the heat flux boundary, five values were taken in the range of 5-15 MW/m², with each run at a number of different mass flow rates. These values lie in the typical range expected for cooling applications in a **DEMO** divertor target [79, 170, 247].

The neutron energy spectrum present in a deuterium-tritium burning plasma can lead to severe degradation of material thermal conductivity. There are four primary mechanisms by which this degradation occurs in copper alloys: radiation defects; ballistic dissolution; void swelling; transmutation [60, 74, 75, 83]. These mechanisms are dependent on the accumulated level of irradiation and the material temperature.

Transmutation and ballistic dissolution will be the dominant thermal conductivity degradation mechanisms for a CuCrZr heat sink component in a fusion power plant [74, 75]. Data from [75] estimates an increase in electrical resistivity for copper due to transmutation of $\sim 0.28 \text{ n}\Omega\text{m}/\text{displacements per atom (dpa)}$. Taking an upper limit² of 6 dpa/full-power year

²In the strike-point region the damage has been calculated to be 3 dpa/fpy.

(fpy) for copper in the divertor region from [267], it is possible to estimate the impact on thermal conductivity over a specified time span. If it is also assumed that ballistic dissolution causes a fixed 10% reduction [75], then it is possible to estimate a total reduction in thermal conductivity over the component lifetime. Taking the irradiation damage out to 3 fpy, which is at the upper limit of the life expected for the EU DEMO divertor [300], results in a predicted thermal conductivity drop of 39%.

The heat treatment history of the material also has an influence on the thermal conductivity for precipitation hardened copper alloys, so two different process routes were considered from [311]: the ITER reference (83.4% International Annealed Copper Standard (IACS)), and Hot Isostatic Pressing (HIP) followed by rapid cooling and then ageing (66.5% International Annealed Copper Standard (IACS)).

A range of thermal conductivity values were selected for the thermo-fluid simulations, considering both the heat treatment and total dpa damage. In each case the thermal conductivity was assumed to be temperature independent.

4.3.2 Results

Calculations of τ_s as a function of m^* for the HPJC-1a were based on the method set out in Sec. 4.2. The results of the influence of heat flux magnitude on the value of the wall overheat are shown in Fig. 4.5.

To an excellent approximation it is evident that the wall overheat is sensibly independent of heat flux, for a given m^* . The results can be physically explained as follows: as the heat flux increases, the fluid temperature change will also increase for a given value of m^* . Concurrently, the peak component temperature will also rise. The resulting increases in T_s and T_{ce} have opposite influences on τ_s , leaving the wall overheat approximately unchanged.

The simulations for varying heat flux were run with coolant flow rates that achieved similar peak component temperatures. This allows the results in Fig. 4.5 to be used to compare

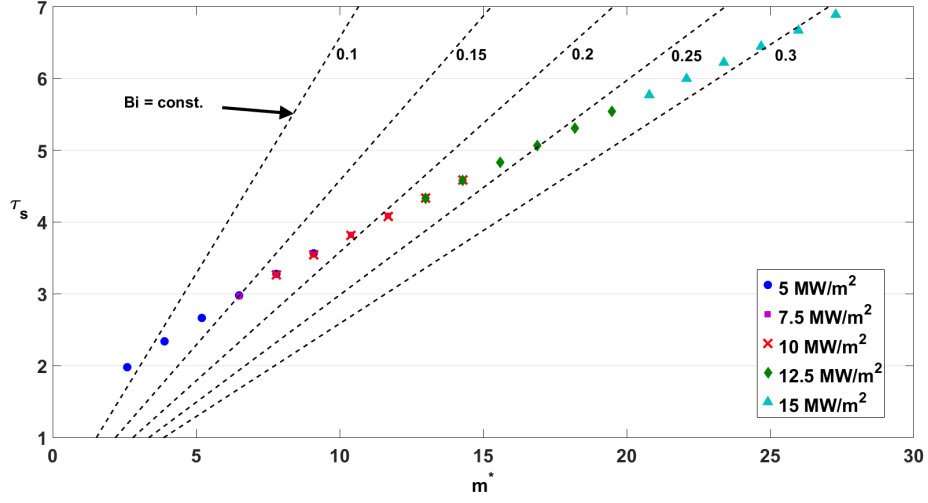


Figure 4.5: Dependence of the wall overshoot on non-dimensional mass flow rate for varying heat fluxes - material thermal conductivity fixed at 320 W/mK. Lines of constant Biot number have also been added using equation (4.10) and setting $\frac{t}{\sqrt{A}} = 0.064$ based on the HPJC-1a geometry.

coolant requirements. For instance, Fig 4.5 indicates that, to operate with the same peak component temperature at 15 MW/m² as 10 MW/m², the coolant flow rate must be at least doubled.

In Fig 4.6 the influence of varying material thermal conductivity on the functional relationship between τ_s and m^* has been plotted. As with the heat flux variation, it is evident that τ_s is sensibly independent of material thermal conductivity for a given m^* .

To physically explain this result, it is first useful to note that varying the thermal conductivity at a fixed m^* is equivalent to changing the mass flow rate between simulations. This alters the temperature change of the fluid for an equivalent heat input, and the peak component temperature as well. As in the heat flux scenario, these two factors effectively cancel, leading to the same value of τ_s .

The importance of a high thermal conductivity heat sink material for HHF applications can be clearly elucidated from Fig 4.6. At the end of 3 fpy in a DEMO divertor, the coolant flow rate would need to be increased three-fold to maintain the same peak temperature.

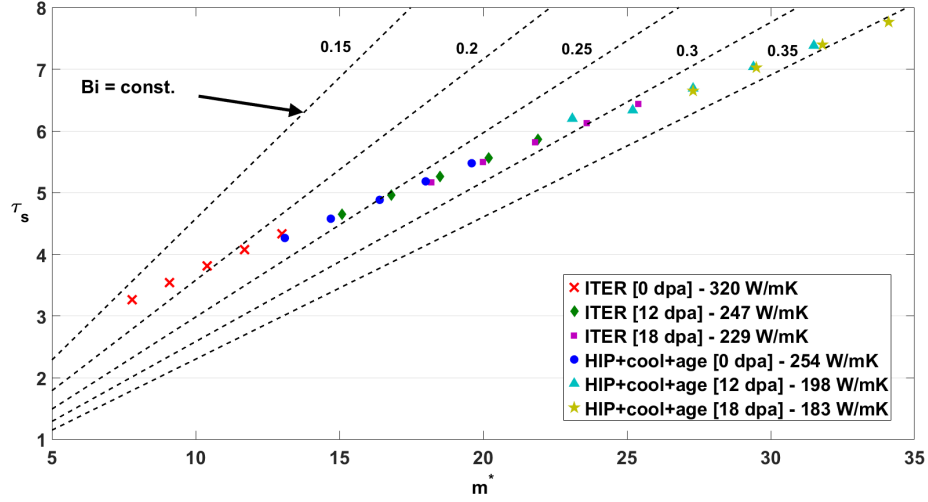


Figure 4.6: Dependence of the wall overshoot on non-dimensional mass flow rate for varying material thermal conductivity - constant heat flux of 10 MW/m^2 . Lines of constant Biot number have also been added using equation (4.10) and setting $\frac{t}{\sqrt{A}} = 0.064$ based on the HPJC-1a geometry.

A powerful feature of τ_s is that it allows the engineer to extrapolate the performance of a concept to a wide range of conditions. This would be invaluable to preliminary tokamak design, where component performance at end-of-life is expected to be vastly different to that at start-of-life. It also negates the need to perform expensive HHHF thermo-fluid experiments. As long as data is acquired for the m^* range of interest, the results can be extrapolated to other relevant conditions.

Artificial lines of constant Biot number have also been added to Figs 4.5 & 4.6 using equation (4.10) from Sec. 4.1.2.1, and a value for $t/A = 0.064$ based on the geometry of the HPJC-1a. These lines show that the wall overshoot for the HPJC-1a tends to higher Biot numbers as m^* is increased. At a fixed m^* the wall overshoot decreases with increasing Biot number, as a higher heat transfer coefficient is being achieved for the same heat capacity rate.

4.4 Comparison of Divertor Cooling Concepts using the Wall Overheat

The unique combination of required life-time (typically years), irradiation, and **HHFs** makes component design a tough challenge in tokamaks. This is particularly true of the divertor region, where the heat load conditions are most severe. Many proposals have been put forward over the past decades - see Sec. 2.2 in Chpt. 2 - to solve this engineering challenge, though no clear candidate has emerged as the definitive answer for a future fusion power plant.

In Fig. 4.7 the wall overheat has been plotted for a number of these candidates against m^* , using the approach set out in Sec. 4.2.

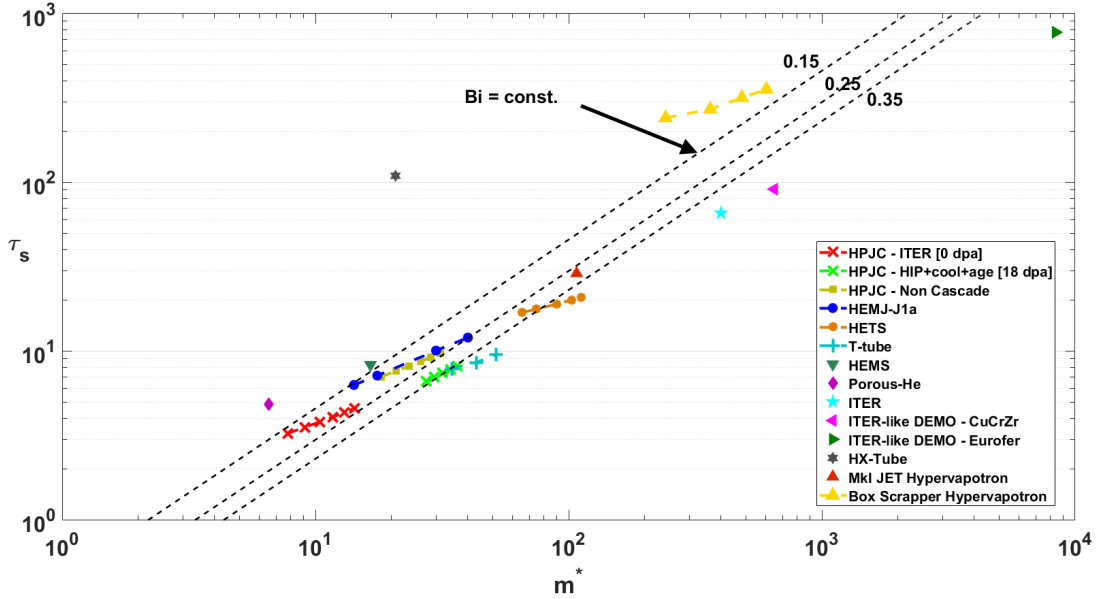


Figure 4.7: Variation of the wall overheat with non-dimensional mass flow rate for a range of past and present divertor proposals [6, 31, 58, 136, 147, 199, 250, 284, 302, 303]. Lines of constant Biot number have also been added using equation (4.10) and setting $\frac{t}{\sqrt{A}} = 0.064$ based on the HPJC-1a geometry.

A general observation from Fig. 4.7 is that jet impingement and porous flow concepts have achieved lower wall overheats than the **ITER** and **ITER**-like, tubular flow designs. This is because the former flow systems maximise the power extracted at a given coolant mass flow

rate. In jet impingement systems, this is achieved through high heat transfer coefficients whilst, for porous flow concepts, it is the large internal wetted area.

As the wall overheat against m^* methodology was developed assuming single-phase flow, there is a question over the validity of including **ITER**-like concepts, that involve two-phase flow, in the results. It is actually the case that, as long as the inlet and outlet conditions are taken at locations where the vapour fraction is zero, the wall overheat is valid in its current form. This could be achieved by assuming that there are small pipe sections either side of the monoblock that have adiabatic walls; any vapour formed in the heated monoblock section condenses in the bulk flow along the adiabatic pipe section. The effects of nucleate boiling are incorporated into the value of T_s .

The HPJC-1a offers the lowest wall overheat of all the concepts considered. This is achieved due to its combination of cascade jet impingement, which maximises the Stanton number for a given thermal capacity rate, together with the application of the coolant as close to the heat flux surface as possible. Combining these flow advantages with a high thermal conductivity copper alloy, and relatively high thermal conductivity coolant, is what gives the design its distinct performance advantage.

Of the helium cooled jet impingement concepts, the HEMJ-J1a has a marginal advantage over the T-tube under equivalent heat flux and peak component temperature boundary conditions. The **ITER** and **ITER**-like tubular flow concepts plot poorly due to a combination of heat flux peaking, and high coolant mass flow rate requirements for a desired heat transfer coefficient. However, their simpler manufacture, particularly for large assemblies, does offer some advantages over smaller heat sink designs, such as the HEMJ-J1a.

If the HPJC-1a concept is operated without cascading the jets the wall overheat is seen to rise, along with the required m^* to achieve an equivalent peak component temperature. Also shown in Fig. 4.7 is the wall overheat for the HPJC-1a under the most detrimental heat treatment and irradiation damage. This result indicates that the mass flow rate would

have to be approximately doubled to retain the same peak component temperature under equivalent boundary conditions. For the plant designer, this type of information is pivotal in ensuring successful operation throughout a reactors lifetime.

4.5 Summary

A figure of merit has been established by the author that can aid the engineer in the development and comparison of **HHF** cooling components. Its derivation was shown to compliment similar work in the field of gas turbine blade cooling where the thermal boundary conditions are specified by a convective heat transfer coefficient, rather than a fixed heat flux level. This figure of merit was termed the wall overhear, and shown to have a minimum value of unity for the ideal case when the peak component temperature is used as a reference value.

The basic parameter dependency of the wall overhear was established through a 1D model of the thermal resistance across the conducting material, and at the convective heat transfer surface. A summary of this chapter's findings are as follows:

- For **HHF** heat sink designs from a range of fields, the wall overhear was shown to increase with the non-dimensional mass flow rate;
 - An exception to this rule were components that had a high effect through-thickness thermal conductivity. They showed a trend of decreasing wall overhear over the the range of m^* values considered;
- Where multiple m^* plot points were available, Pareto line type behaviour was observed, indicating that there is a limit to the minimum τ_s that can be achieved by a particular component;
- Microchannel, jet impingement, vortex and macro-porous flows displayed the lowest wall overhear performance;
- A good cooling component, based on the wall overhear methodology, was shown to have the following characteristics:

- Short conduction path lengths;
 - Constructed from a high thermal conductivity material;
 - A large internal wetted area for the coolant;
 - Applies the coolant as close to the heat source as feasible;
 - Maximises the heat transfer coefficient for a given coolant mass flow rate;
 - Employs a coolant with high thermal conductivity, but low specific heat and dynamic viscosity.
- Curves of wall overhear against non-dimensional mass flow rate were shown to collapse onto a single master curve under varying heat flux and solid thermal conductivity;
 - This master curve could be utilised by the tokamak designer to estimate the performance of a divertor component over the course of its life;
 - The HPJC-1a was shown to have the lowest wall overhear when compared to current, and past, divertor cooling proposals;
 - ITER, and ITER-like, tubular flow concepts were shown to operate at high wall overheats due to the large mass flow rates required to obtain the desired peak solid temperatures.

Chapter 5

Component Life

“There are two possible outcomes to an experiment: If the result confirms the hypothesis, then you’ve made a measurement. If the result is contrary to the hypothesis, then you’ve made a discovery.”

— Enrico Fermi

Preceding chapters have focussed on the heat transfer performance of the HPJC-1a. However, the benefits of improving this aspect of a DEMO divertor target component can only be realised if it also fulfils the life requirements. In this chapter, the author makes an initial assessment of the life of the HPJC-1a based on criteria set out in the Structural Design Criteria for ITER In-vessel Components (SDC-IC) [123].

5.1 Numerical Modelling Methodology

A number of Finite Element Analysis (FEA) simulations were performed on the HPJC-1a using ANSYS Workbench 16.2. A combination of both elastic and elastic-plastic material models were used to simulate shakedown¹ in the component, before comparing the results to the SDC-IC. The way in which the component life changed with varying thermal and mechanical boundary conditions was considered in detail.

The effect of neutron irradiation on material properties was ignored in these analyses, as it

¹Stabilisation of plastic deformation after a finite number of loading cycles so that further cyclic stresses remain within the elastic range [248].

was not possible to incorporate time(flouence)-dependent properties into the elastic-plastic material models, as irradiated true stress against strain curves were not available. This same approach was used by [84] in analysing an ITER-like DEMO concept.

The mismatch in thermal expansion coefficients between the tungsten armour tile and Cu-CrZr heat sink creates additional thermal stresses in a full divertor target component. As only the CuCrZr heat sink is considered in this research, the results of this section should be treated as those for the case where these stresses are minimised or eliminated. Modern manufacturing techniques, such as Functionally Graded Materials (FGMs) [146], indicate that this may be possible. In addition, liquid metal Plasma Facing Component (PFC) could eliminate this problem entirely - see Chpt.2

Finally, there is also a number of disruption events that can occur in a tokamak that can influence component life, some of which have been discussed in Chpt.1. There is intent within the fusion community to suppress these disruptions in a DEMO [20]. The author therefore chose to focus on determining the life of the HPJC-1a assuming that the thermal power loading only varied because of the pulsed nature of a tokamak plasma burn [266].

5.1.1 Model Geometry & Meshing

5.1.1.1 Overview

As ANSYS Mechanical 16.2 does not allow the use of stress linearisation² with periodic geometries, the 120° model from the CFD in Chpt.3 could not be employed in these analyses. Instead, a 180° segment was utilised, together with a symmetry plane. As in Chpt.3, the solid was treated as a single body, rather than a number of separate plates. A cross-section of this geometry is shown in Fig.5.1, together with the computational mesh.

For the mesh, a proximity and curvature tetrahedral method was used with 3-4 elements across gaps. External edges and face sizes were set at 0.01-0.5 mm, with all internal edges

²Calculates the membrane, bending, peak and total stress along a straight line path [159].

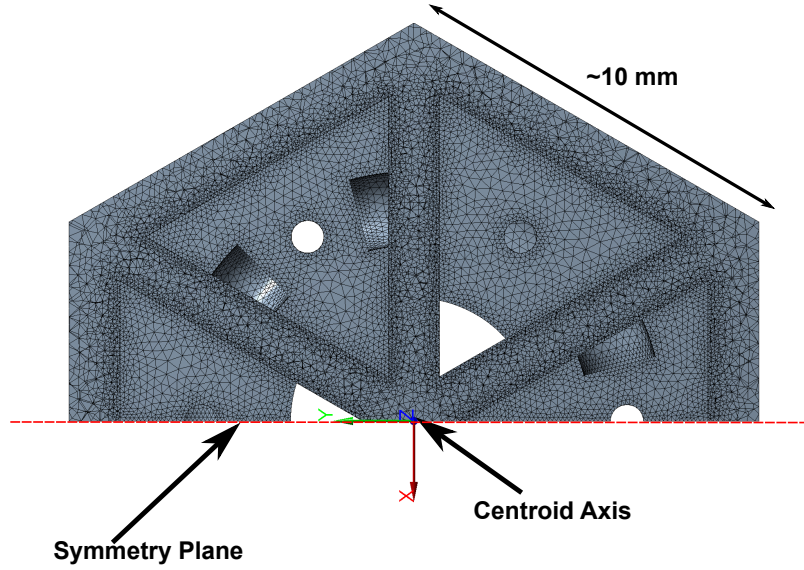


Figure 5.1: Cross-section of geometry used for **FEA** simulations showing symmetry plane and origin at the centroid. The associated mesh used for thermal and structural analyses is also shown.

at the fluid-to-solid interface fixed at 0.1 mm. These settings were found to produce stress values away from geometric discontinuities that varied minimally with further refinement.

5.1.1.2 Sub-Modelling

The sub-modelling technique available in ANSYS Mechanical 16.2 allows smaller segments of a geometry to be analysed in higher fidelity. This can be employed in two ways: 1) to increase the local mesh density in an important region of the geometry; 2) to explore the influence of small changes to the geometry, such as fillets. To use sub-modelling, a simulation of the full geometry must first be performed to obtain a displacement field. These displacements are then applied as a boundary condition to the cut-plane of the sub-model. From St. Venant's principle [101], the location of this cut-plane must be sufficiently far away from the region of interest to avoid invalidating the displacement values.

The geometric discontinuities in the HPJC-1a model will act as stress concentrators. Accurate calculation of the stresses in these regions with **FEA** is not possible, as they are mesh-dependent. This makes application of the **SDC-IC** design criteria in these regions highly subjective. In reality, both bonding and machining methods will leave fillets in these

areas. Rather than performing a computationally expensive simulation on the full model with fillets, key information can be gained by analysing a sub-model instead.

Even with fillets, there is also the possibility that stresses could be higher than the yield strength of the material. Sub-modelling can also be used in these cases to greatly reduce the mesh size for computationally expensive elastic-plastic analyses. However, this involves applying elastic displacement boundary conditions to an elastic-plastic model. For this research, where the plastic deformation is limited to small regions surrounded by elastic material, this assumption will produce minimal errors. A similar approach was used by [229] to model fatigue in ball grid array packages in electronics.

A scoping simulation of the full geometry, using the reference case boundary conditions from Chpt. 3, showed little variation between the stress fields in each impingement cavity. Based on these results, it was decided that the sub-modelling should be performed on a secondary cavity region, where the higher operating temperatures will lead to reduced material strength. The choice of cut-plane for the sub-model, and the corresponding segment, are shown in Fig. 5.2

The suitability of the sub-model cut-plane boundary was determined by comparing the results of elastic simulations for both the full model, and sub-model, without fillets. Temperature profiles were imported from the full model onto the sub-model, and the same mechanical boundary conditions applied to both. Local and global von Mises stresses are shown in Fig. 5.3 for the reference loading conditions, namely: a heat flux of 10 MW/m^2 ; a coolant pressure of 20 MPa. The maximum discrepancy between the local von Mises distributions is less than 4%. It is therefore evident that the chosen cut-plane enables the sub-model to accurately capture the stress distribution in the regions of interest.

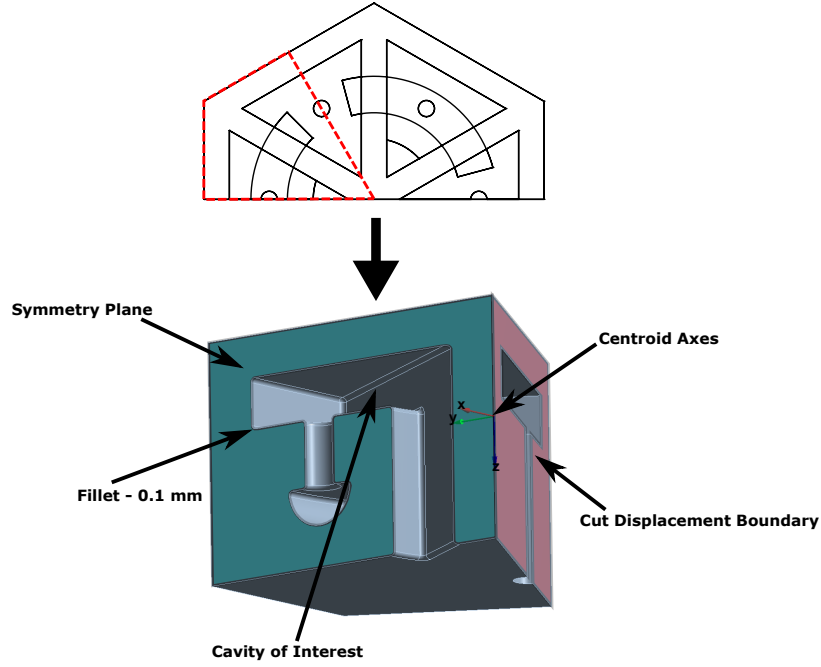


Figure 5.2: Segment, shown by dashed red line, of full model chosen for sub-modelling, together with associated 3D geometry.

5.1.1.3 Fillet Mesh Refinement

In fillet regions the mesh must be adequately resolved to obtain accurate stress values. To determine the mesh resolution required for this research, a number of elastic simulations were performed with varying surface element length scales (d). A reference fillet radius, r , of 0.1 mm was chosen based on cross-sections from manufactured geometries - see Chpt. 7. The corresponding r/d quotients simulated ranged from 1.3 to 10.

In Fig. 5.4 the peak von Mises stress in the indicated fillet region is plotted against the ratio of the fillet radius to the maximum mesh size. To estimate the peak von Mises stress at higher values of r/d the Richardson extrapolation method was used [228]. Calculations were made using step sizes for r/d of 0.1 and 0.2, in conjunction with the Euler method and least square best fit curve in Fig. 5.4. Above $r/d = 28.2$ the fractional change in the peak von Mises stress for the 0.2 step size was estimated to be below $1e^{-5}$. The estimated peak von Mises stress at this r/d value using the Richardson extrapolation method was calculated to be $641.5 \text{ MPa} + \mathcal{O}\left(\left(\frac{d}{r}\right)^2\right)$.

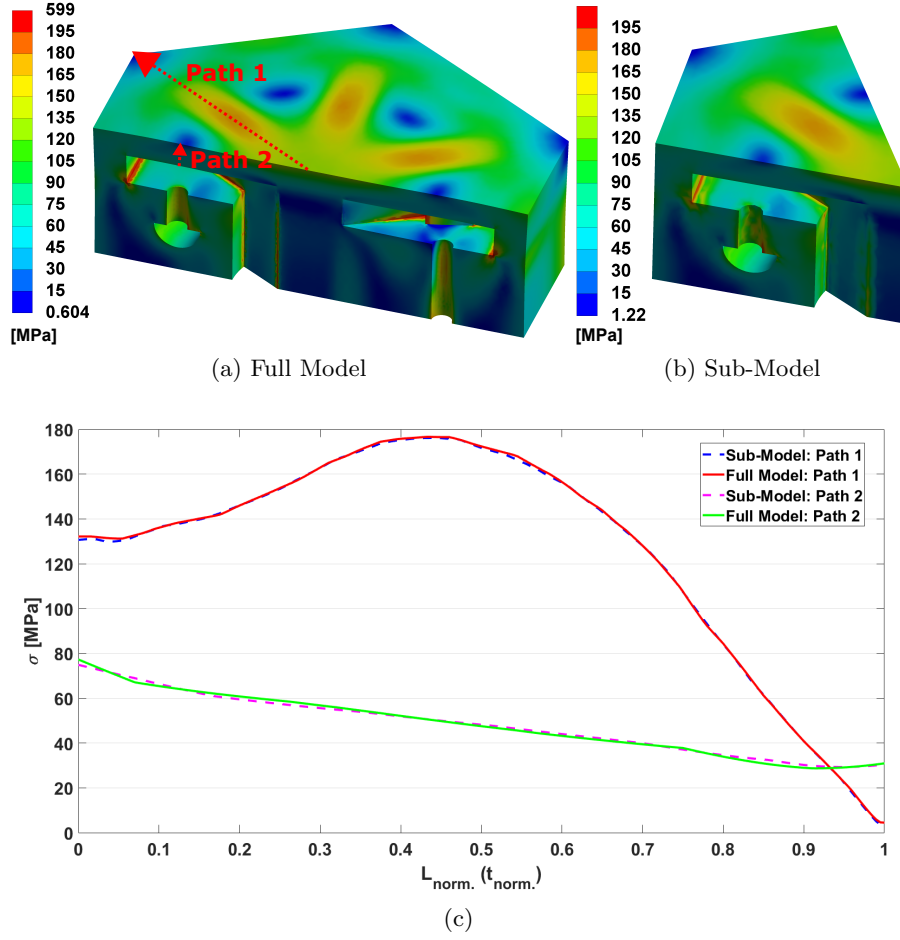


Figure 5.3: Local and global comparison of von Mises stress distributions for full and sub-model geometry cases - elastic analysis.

At a scaled mesh size of ~ 3 the von Mises stress is 97.5% of the estimated maximum value. This value of r/d was deemed a good compromise between computational accuracy and effort. For all alternative fillet radii in this chapter the same r/d value was used.

5.1.2 Determination of Heat Sink Temperature Distribution

Steady-state thermal simulations were performed in ANSYS Workbench 16.2 to determine the solid temperature distributions for a range of boundary conditions. This approach was chosen as an alternative to importing the results directly from the conjugate CFD analy-

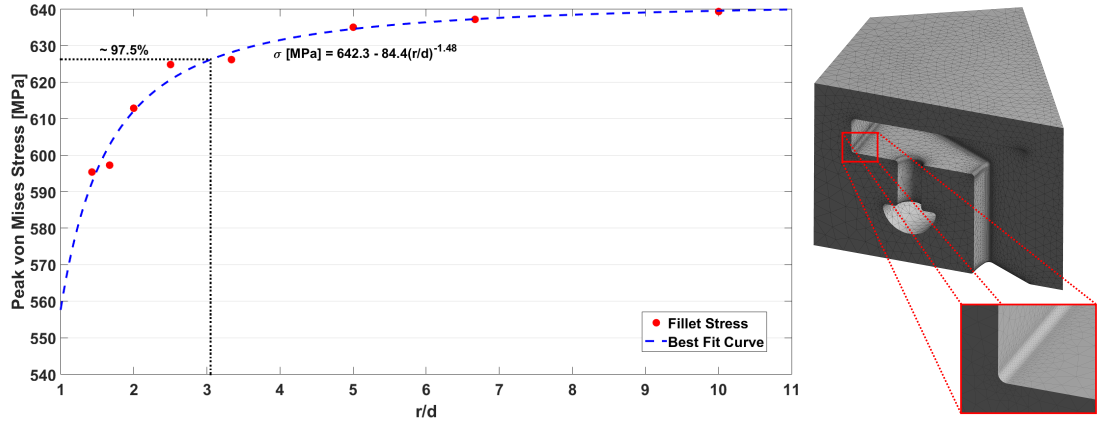


Figure 5.4: Comparison of peak von Mises stress in fillet region against scaled maximum surface mesh linear dimension. Fillet radius (r) fixed at 0.1 mm and surface mesh element linear dimension (d) varied.

ses in Chpt.4, as two additional features needed to be included; specifically: 1) a uniform volume heat source to replicate neutron heating; 2) temperature dependent material properties. The former used a value of 10 MW/m^3 , taken from data in [118, 149, 150], whilst material properties for CuCrZr were extracted from App. A of the SDC-IC [123, App. A].

A uniform heat flux was applied to the plasma facing side of the impingement plate, whilst all other external surfaces were set as adiabatic. At the fluid-to-solid interface, heat transfer coefficients were imported based on data extracted from the CFD simulations in Sec.4.3 of Chpt.4; these distributions all resulted in a peak heat sink temperature of 350°C in the CFD simulations, for each heat flux loading case. A coolant inlet temperature of 180°C was used as the reference value.

Applying the CFD derived heat transfer coefficients to the thermal simulations required the nodal values to be manipulated in MATLAB R2016a to account for the differing model geometries. As the meshes also differed between the FEA and CFD models, a profile preserving mapping tool was used within ANSYS Workbench 16.2 to import the data. Weighting of the nodal values from the CFD mesh was carried out using a distance based averaging technique. Up to eight nodal points from the CFD mesh were selected from within a sphere of 0.35 mm radius around a given FEA mesh node.

The heat transfer coefficients were assumed to be temperature independent in these analyses. This is justifiable, as the peak temperature difference between the **CFD** and **FEA** was found to be $<10^{\circ}\text{C}$ for all cases, which would only change the thermal properties of water by less than $\sim 2\%$.

5.1.3 Mechanical Boundary Conditions

The temperature fields generated by the thermal simulations were imported into ANSYS Mechanical 16.2 to perform subsequent structural analyses. To ensure accurate mapping of these temperature fields, the meshing parameters were kept equivalent to those used in the thermal analyses.

An additional uniform pressure boundary condition was applied at the fluid-to-solid interface to represent the coolant operating pressure. This was deemed sufficiently accurate, as the variation in static pressure over the equivalent surface in the **CFD** was less than 5%.

To replicate realistic mechanical constraints a number of displacement conditions were applied for all loading conditions, which are shown in Fig. 5.5 for the full geometry.

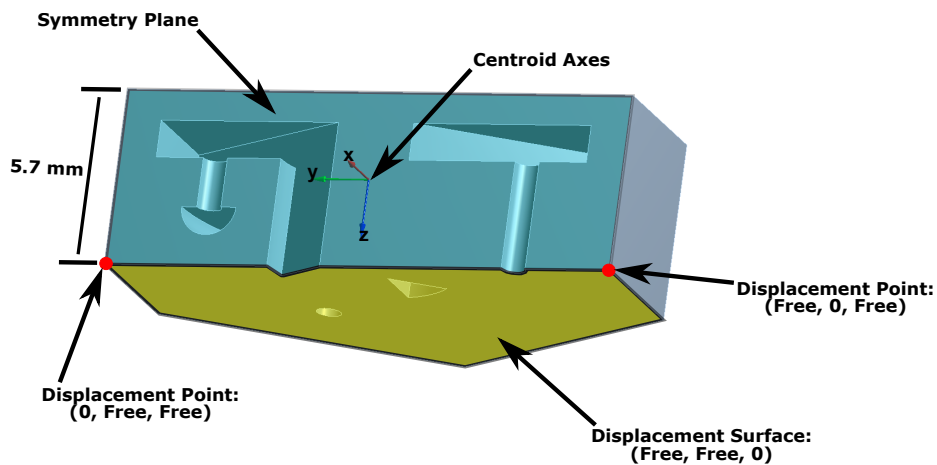


Figure 5.5: Displacement conditions and symmetry plane used for mechanical analysis. Note: the restraints in the parentheses are of the form (x, y, z) .

On the bottom face of the heat sink the nodes were constrained to move in the x-y plane only. This simulates the cooling module being bonded to a manifold that is rigid, but that has an equivalent thermal expansion coefficient to the heat sink material. Such a situation would arise if CuCrZr were bonded to a manifold constructed of 316L stainless steel. Two additional displacement conditions were also used to inhibit rotation about the z-axis. For the sub-model, the displacement point that is constrained to the x-z plane is replaced by the cut boundary displacements.

5.1.4 Material Mechanical Properties

For all of the static-structural simulations of the full model, CuCrZr was assumed to be perfectly elastic, with large deflections ignored, but thermal strain effects included.

The selection of an elastic-plastic material model depends on the behaviour one wishes to investigate. For shakedown, a model that incorporates the Bauschinger effect³ is required [256]. Only **KINematic Hardening (KINH)** models in ANSYS Mechanical 16.2 are able to do this. They differ from isotropic (work) hardening models because the yielding surface translates in stress space under plastic deformation, rather than increasing in size. This difference is depicted graphically in Fig. 5.6.

The image(s) originally presented here cannot be made freely available by ORA due to copyright.

Figure 5.6: Comparison of the hardening behaviour of (a) isotropic and (b) kinematic elastic-plastic model [4].

³Lowering of the yield stress of a material when the direction of deformation is changed [56]

Following previous studies [52, 53, 84, 134] on divertor components the author chose to use the Chaboche KINH model in this work. Unaxial plastic strain data for a Chaboche KINH model of CuCrZr was taken from the SDC-IC App. A, where it was extracted from the true stress-strain curves for heat treatment B - see Fig. 5.7. This heat treatment involves solution annealing at 980°C for 1 hour, followed by rapid quenching, and then ageing at 475°C for 2 hours [311]. Given the small modular nature of the HPJC-1a design, it was judged that such a process path could be implemented. In addition, heat treatment B also has the most complete data set, making it a useful starting point for a mechanical analysis.

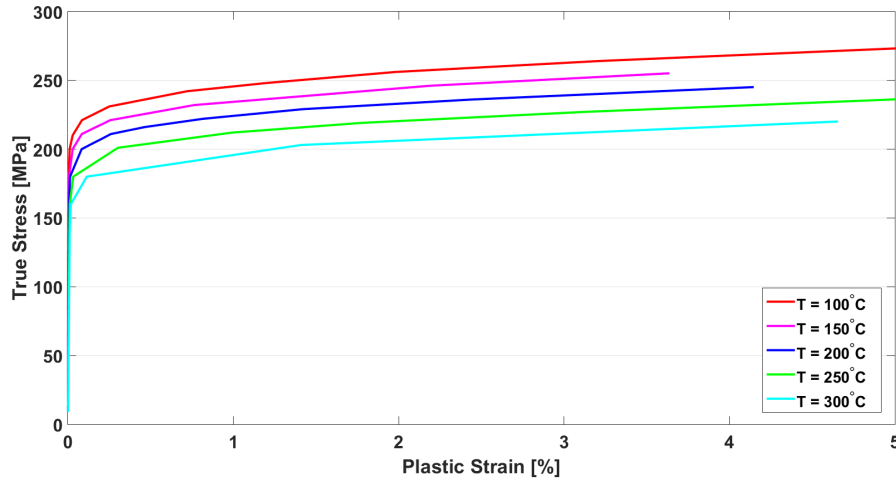


Figure 5.7: Variation of true stress with plastic strain extracted from total strain data in the SDC-IC App. A, for heat treatment B [123, App. A] of CuCrZr.

For the KINH models, the yield criterion for a ductile material is given by [42, 43]

$$\sqrt{\frac{3}{2} (\mathbf{s} - \boldsymbol{\beta}) : (\mathbf{s} - \boldsymbol{\beta})} - \sigma_y = 0, \quad (5.1)$$

where $\mathbf{s}$ is the deviatoric stress⁴, $\boldsymbol{\beta}$ the back stress⁵ and σ_y the yield stress. For the Chaboche KINH model the evolution of the back stress in equation (5.1) is non-linear, having the form

⁴The stress that remains once those stresses that only produce a change in volume have been removed. These latter stresses are referred to as the hydrodynamic, or dilational stresses [101].

⁵Location of the centre of the yield surface [256].

$$\Delta\beta_i = \frac{2}{3}C_i\Delta\epsilon_{pl} - \gamma_i\beta_i\Lambda. \quad (5.2)$$

Here ϵ_{pl} is the equivalent, and Λ the accumulated, plastic strain. The constants C_i and γ_i are material parameters, with the latter being the rate of decrease of the hardening modulus. Each i^{th} term represents a separate Chaboche **KINH** model, which are superimposed to determine the total back stress during each solver iteration.

Manual curve fitting of a 3 parameter Chaboche **KINH** model was done using the ANSYS Workbench 16.2 software. In order to simulate shakedown it is imperative that one of the γ_i values be set to zero so that there is not a limiting value on the back stress **[119]**. In this case the author choose to set $\gamma_3 = 0$, which gave the results for the remaining parameters shown in Table**5.1**.

Temp. [°C]	C₁ [MPa]	γ₁	C₂ [MPa]	γ₂	C₃ [MPa]	σ_y [MPa]
100	4.1E+4	1.7E+03	3.0E+03	1.2E+02	5.0E+02	2.0E+02
150	4.6E+4	1.9E+03	3.4E+03	1.6E+02	5.8E+02	1.9E+02
200	5.9E+4	2.0E+03	3.1E+03	1.5E+02	4.9E+02	1.7E+02
250	9.9E+4	2.8E+03	2.5E+03	1.3E+02	4.7E+02	1.6E+02
300	10E+5	3.2E+03	2.6E+03	7.8E+01	2.1E+02	1.5E+02

Table 5.1: Three parameter Chaboche **KINH** model parameters to simulate shakedown in CuCrZr heat treatment B from the **SDC-IC** App. A.

To determine the accuracy of the Chaboche **KINH** model parameters, a simulation of a uni-axial tensile test was carried out. The comparison of these results with data from the **SDC-IC** App. A for CuCrZr, after heat treatment B, is shown in Fig**5.8**. It is evident that the model parameters provide an excellent fit to the test data.

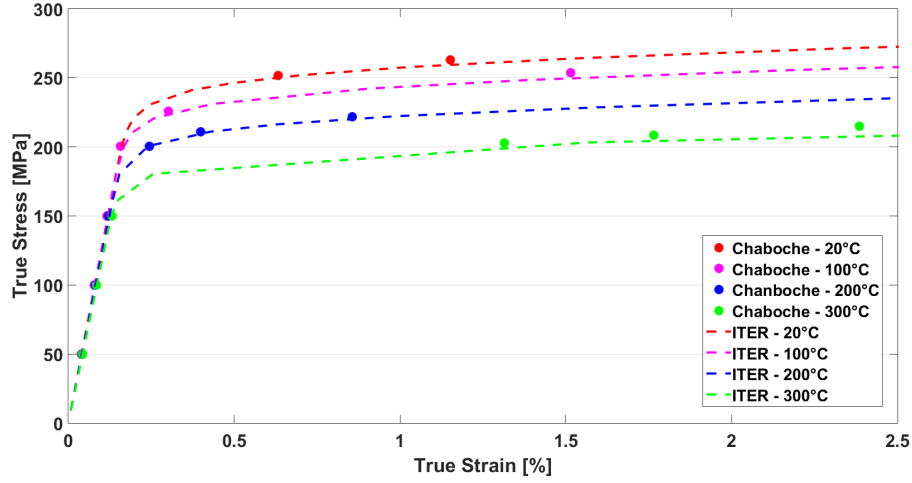


Figure 5.8: Comparison of simulated uni-axial tensile test results using Chaboche KINH model against data in the SDC-IC App. A for CuCrZr heat treatment B.

5.1.5 Cyclic Load States

Two different loading states were considered for the FEA analyses, based on a similar work by [84]: standby and pulse. For the former case, the solid was set to be isothermal, with a temperature equal to that of the coolant inlet. A pressure boundary condition at the fluid-to-solid interface was also applied, with five different values simulated in total: 10, 15, 20, 25 and 30 MPa. These are envisaged as the conditions the HPJC-1a would experience between fusion pulses.

The pulse state represents the flat-top burn period of the plasma, where both thermal and pressure loads occur simultaneously. In the case of the former, five different heat fluxes were considered: 5, 7.5, 10, 12.5 and 15 MW/m². When combined with the possible pressure loadings, this gave a total of twenty five cases to simulate. A summary table of the boundary conditions considered for each case is shown in Table 5.2.

State	Pressure Range [MPa]	Interval [MPa]	Heat Flux Range [MW/m ²]	Interval [MW/m ²]
Standby	10-30	5	-	-
Pulse	10-30	5	5-15	2.5

Table 5.2: Details of the boundary conditions considered in the mechanical analyses.

In the elastic-plastic simulations the reference fillet radius was set to 0.1 mm. Additional simulations were performed with fillet radii of 0.05 and 0.2 mm to determine its influence on component life. Care must be taken to avoid making the fillet size too large, as any temperature data is mapped from the full, un-filletted, model. For large fillet radii this may lead to incorrect predictions of the temperature distribution within the fillet region.

5.1.5.1 Cycles to Shakedown

To bound the impact of cyclic damage it is common in an elastic-plastic analysis to cycle between minimum and maximum stress states [101, Chpt.11]. In the case of ITER, the plasma start-up and cooling phases are expected to take ~ 200 sec [166]. During start-up a poloidal limiter configuration is used until the magnetic divertor is formed after ~ 30 sec. These transient time-scales indicate that peak stresses will occur during the plasma burn. Assuming the same will hold for a DEMO, the author chose in the present work to mimic shakedown by cycling between the standby and pulse stress states.

Shakedown was simulated for each of the thermal and mechanical loading states outlined in Table 5.2, with a reference fillet radius of 0.1 mm. To determine the number of cycles required to achieve shakedown the author used the results from Fig 5.9, which show the percentage change in equivalent plastic strain between one cycle and the next for heat fluxes of 7.5-15 MW/m². The results for the 5 MW/m² case are not shown as the behaviour became purely elastic after one cycle.

To determine the number of cycles to achieve the $\Delta\epsilon_{pl} = 0$ condition the Richardson extrapolation method was again used. The step sizes were kept the same as those in Sec. 5.1.1.3. As the data for the 10 MW/m² case implied that it would take the largest number of cycles to achieve shakedown, its least square best fit curve was used for the extrapolation. The results showed that a minimum of 18 cycles would be required to achieve $\Delta\epsilon_{pl} < 0$. However, the change in equivalent plastic strain from 10 to 18 cycles is

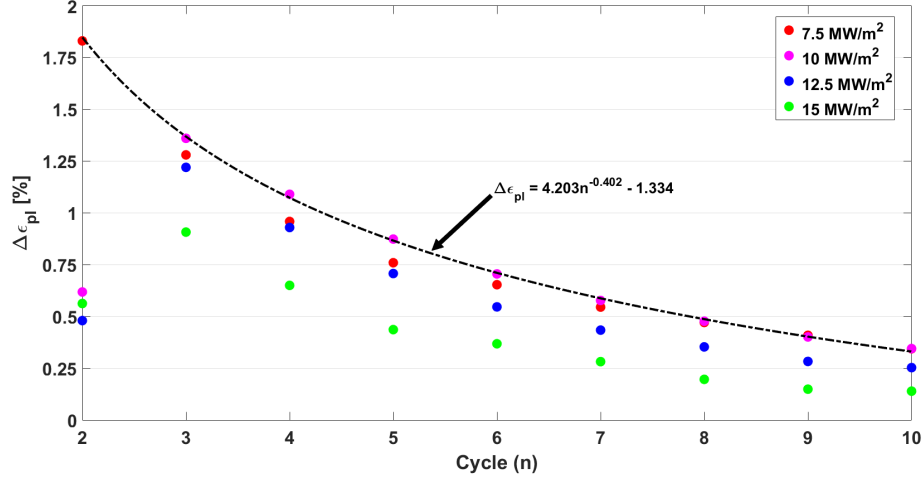


Figure 5.9: Percentage variation of peak equivalent plastic strain in fillet regions of sub-model, with cycle number, for different heat flux loading conditions. The coolant pressure in all cases was 20 MPa.

$$\frac{\epsilon_{pl18}}{\epsilon_{pl10}} - 1 = \left[\prod_{i=10}^{18} (1 + \Delta\epsilon_{pl_i}) \right] - 1 \approx 1\%.$$

Therefore a value of 10 cycles was chosen to provide a good compromise between computational effort and accuracy.

Above 5 MW/m² there is a trend towards a reduced shakedown period with increasing heat flux. This is caused by two mechanisms: 1) the higher initial tensile stresses in the fillet region, which leads to a greater rate of strain hardening during the first pulse; 2) the increased strength of the material due to the lower operating temperatures.

5.2 Comparison with the SDC-IC

In this section the low temperature elastic design rules from the [SDC-IC](#) Apps. B & C [123, App. B & C] are used to determine the life of the HPJC-1a. Neglecting thermal creep was deemed a reasonable assumption as peak metal temperatures were only $\sim 30\%$ of the melting point value [18]. The main purpose of this section of the work was to determine any operational limits and their associated failure modes. A summary of the process path

used to obtain the shakedown stress states, and the design criteria that will be applied, are shown in Fig. 5.10.

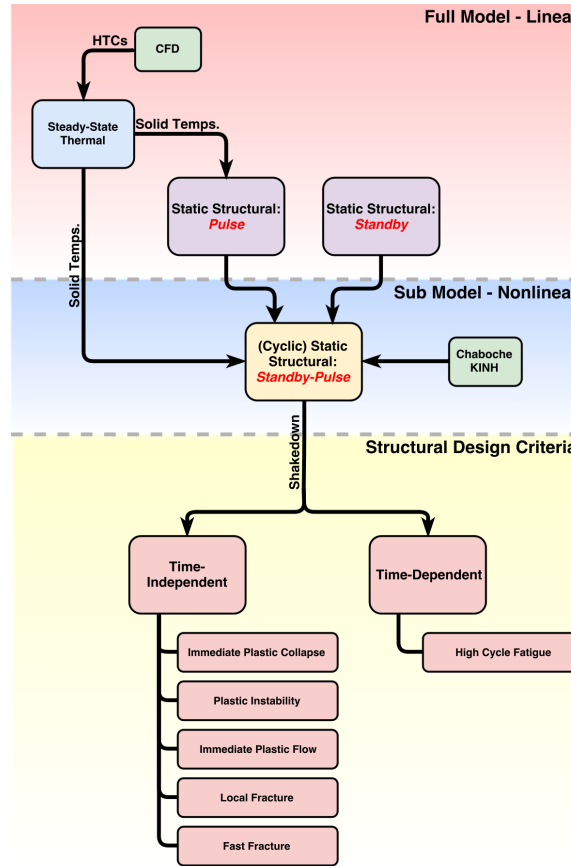


Figure 5.10: Process steps used to obtain final states used for assessment against the SDC-IC. The exact design criteria applied during the life analysis is also shown.

The elastic design rules from the SDC-IC were utilised, assuming the operational loading criteria. However, as shakedown has already been achieved in the structure, the potential for failure through the accumulation of excessive plastic strain (ratcheting) was not considered. Primary stresses were taken from the standby state, and primary plus secondary from the pulse state.

In a complex three dimensional body stress linearisation is required to apply the design rules. The choice of paths along which to linearise the stresses is arbitrary, and must be chosen by the designer to capture the important facets of the stress distribution [13]. To

help achieve this, contour plots of the von Mises stresses after shakedown were first produced for the standby and pulse cases under the reference loading conditions of a 10 MW/m^2 heat flux and 20 MPa coolant pressure. These results are shown in Fig. 5.11

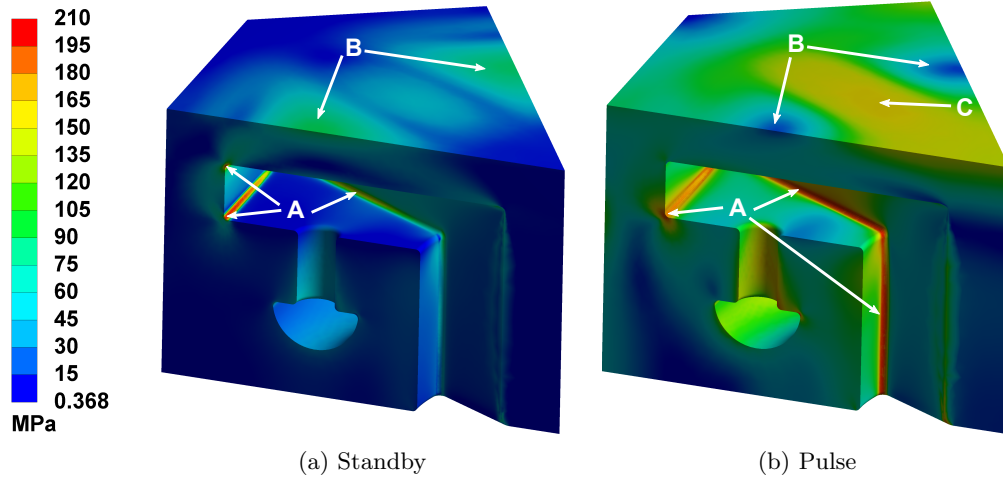


Figure 5.11: von Mises stress distributions at (a) standby and (b) pulse after shakedown for baseline loading case of 10 MW/m^2 heat flux and 20 MPa of coolant pressure.

High stresses occur for the standby case in the fillet regions, labelled A, as well as the mid-point of the cavity, labelled B. The high stresses at the latter location are driven by the peak moment at this point, caused by the uniform pressure loading.

During a plasma pulse, high stresses are again found in the fillet regions (A). On the top of the impingement plate the area above the internal webbing, labelled C, is also highly stressed due to the reduction in flexibility at this cross-section. Interestingly, the high stresses at B have now become negligible. This is because the tensile stress generated by the coolant pressure are offset by compressive thermal stresses.

Based on the distributions in Fig. 5.11, nine paths were chosen for stress linearisation, which are displayed in Fig. 5.12. Focus has been placed on the high stresses in the fillet regions, and the impingement plate.

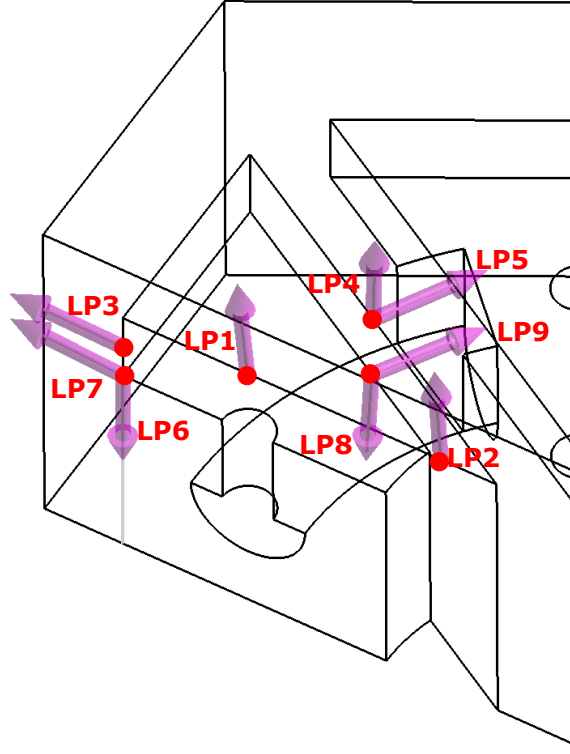


Figure 5.12: Location of paths used for stress linearisation for comparison with the SDC-IC design criteria.

The author began the analysis against the SDC-IC criteria by considering the potential for failure through monotonic⁶ type damage. Results are displayed in the form of a reserve factor, f , defined as

$$f = \frac{\sigma_{ref.}}{\sigma}. \quad (5.3)$$

Here $\sigma_{ref.}$ represents the maximum allowable stress for the particular failure mode being considered, which is dependent on the local material strength. The value of $\sigma_{ref.}$ can vary with both temperature and irradiation history. Path averaged temperatures were used to assess $\sigma_{ref.}$ in this work, except when peak stresses were being considered. In this instance the temperature at the location of the peak stress was used instead.

⁶A load which evolves in a constantly increasing or decreasing manner [123], App. B].

5.2.1 Immediate Plastic Collapse & Plastic Instability

Reserve factors against immediate plastic collapse (P_m) and plastic instability ($P_L + P_b$) from primary stresses are depicted in Fig 5.13 for a range of different coolant pressure conditions. For the plastic instability, an effective bending shape factor⁷ of 1.5 was used when determining σ_{ref} ⁸

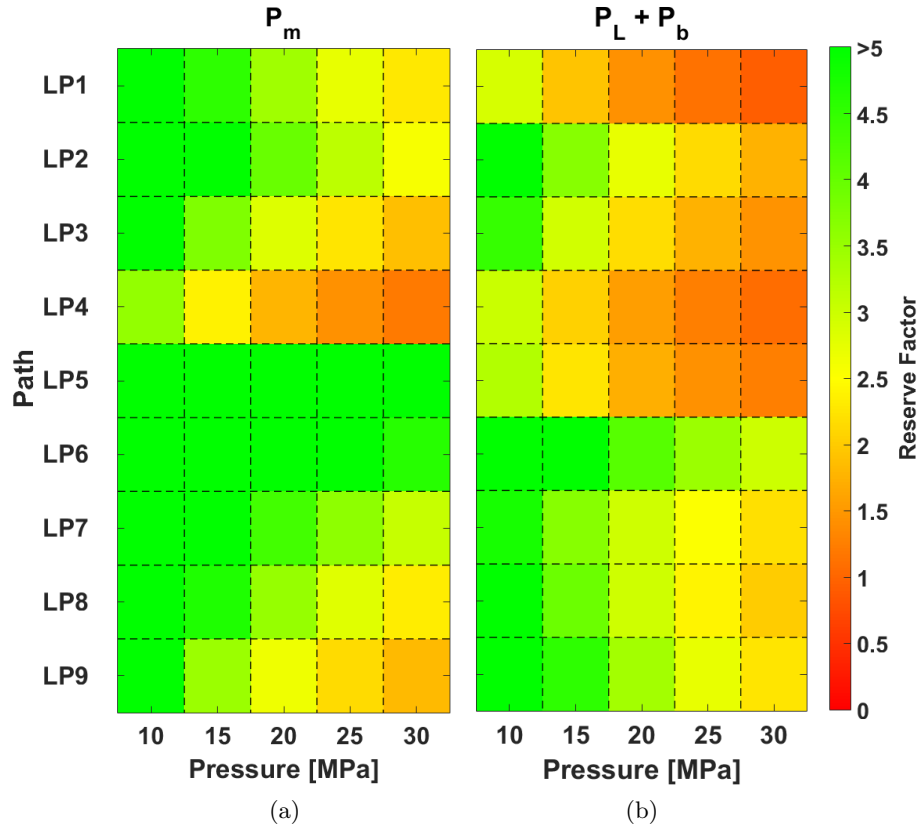


Figure 5.13: Reserve factors, f , for (a) immediate plastic collapse and (b) plastic instability along various stress linearisation paths and coolant pressures.

The lowest reserve factor for immediate plastic collapse is along LP4, which is 1.2 at a coolant pressure of 30 MPa. This path runs from the fillet region closest to the webs, through to the heat flux surface. The stresses are compressive in nature, and caused by the moment generated by the pressure loading within the cavity.

⁷Accounts for the redistribution of stress due to plasticity [123, App. B].

⁸Whilst the standby distribution has a fixed temperature of 180°C, values for σ_{ref} were still based on the temperature distribution under pulse.

It is interesting to note that the reserve factor for immediate plastic collapse along LP4 is lower than LP1 at all pressure loadings. This means failure via this mode will occur in the cavity side walls before the impingement plate.

In general, the reserve factors for plastic instability are lower than those for immediate plastic collapse. For LP1 the reserve factor drops to 0.95 at a pressure of 30 MPa, and is only 1.08 for LP4 under the same conditions. The low reserve factor at LP1 is caused by the lack of support in the central region of the cavity.

For both failure modes, the reserve factors decrease with increasing coolant pressure. However, the maximum difference between the reserve factors along each path, at a given coolant pressure, is less than three. This indicates that there is not one significant weak spot in the design. To obtain a minimum reserve factor of two along all the paths, the coolant pressure must be below ~ 15 MPa.

5.2.2 Immediate Plastic Flow Localisation & Fracture

The reserve factor is plotted against heat flux in Fig. 5.14 for three different structural criteria: immediate plastic flow localisation ($P_L + Q_L$); local fracture without peak stress ($P_L + P_b + Q$); local fracture with peak stress ($P_L + P_b + Q + F$). A triaxiality factor⁹ of 2 was used, together with an elastic follow-up factor¹⁰ of 4, to calculate the allowable stress intensities.¹¹ These results are for a coolant pressure loading of 20 MPa. Further simulations at different coolant pressures showed minimal variation.

Considering the failure criteria from immediate plastic flow localisation first, it can be seen that the paths of greatest concern are LP4 and LP9. An interesting feature of the results

⁹The ratio between the hydrostatic stress and the von Mises stress, normalised to unity for uni-axial loading [123, 2541.1 App. B].

¹⁰The maximum value of the quantity $R = \frac{E\epsilon - \sigma}{\sigma_{el} - \sigma}$ in the structure. Here σ_{el} is the elastically calculated stress, σ the actual stress and ϵ the total strain [123, 3024.1.3 App. B].

¹¹In locations where the uniform elongation dropped below 2% the value of r_3 was set to ∞ [123, 3212.1 App. B].

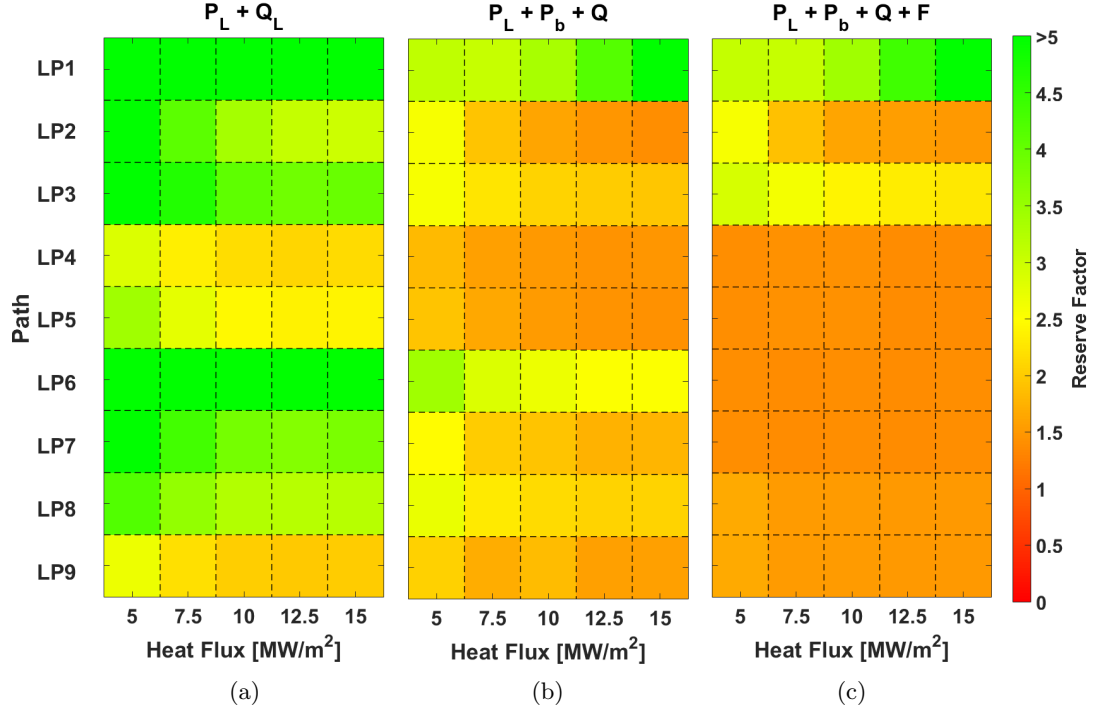


Figure 5.14: Reserve factors, f , for (a) immediate plastic flow localisation and (b,c) local fracture due the exhaustion of ductility. (b) is without peak stress included, whilst (c) includes peak stress. These results are shown for a number of different heat flux boundary conditions.

for these two paths is that there is a small increase in both reserve factors going from 12.5-15 MW/m², which is in contrast to the general decrease observed for the rest of the paths. At a fixed peak component temperature, as the heat flux is increased, the average solid temperature decreases. In most instances this translates into an increase in material strength. If the benefit of the attendant increase in strength exceeds the deleterious effect of the higher stress, then the reserve factor will increase.

The results presented in (b) and (c) of Fig. 5.14 concern local fracture due to the exhaustion of ductility with, and without, peak stress (F). Considering (b) first, paths LP1 and LP7 exhibit interesting behaviours. In the case of LP1, the reserve factor actually increases with heat flux. This is driven not only by the drop in average temperature through the impingement plate, but by a shift in the region of highest compressive stress towards the centre of the hexagon. This effect is evident for LP2, which has the most significant variation in

reserve factor with heat flux. This highlights the importance of a buckling analysis in future research. For LP7 the cause is undoubtedly similar to that discussed previously regarding increasing material strength against higher thermal stress.

Results in (c) of Fig. 5.14 show a similar trend to (b). For paths LP1-LP3 the only significant variation with heat flux is seen for LP3, which actually has a larger reserve factor than for local fracture without peak stress. This is because the reference stress, $\sigma_{ref.}$, is different for each loading case. In this instance, the increase in $\sigma_{ref.}$ is sufficient to offset the impact of adding the peak stress.

In both (b) and (c) it can be seen that paths emanating from fillet regions have low reserve factors. In these cases the peak equivalent plastic strain after shakedown is a more appropriate indicator as to whether the component will fail. This has been plotted for a number of boundary conditions in Fig. 5.15.

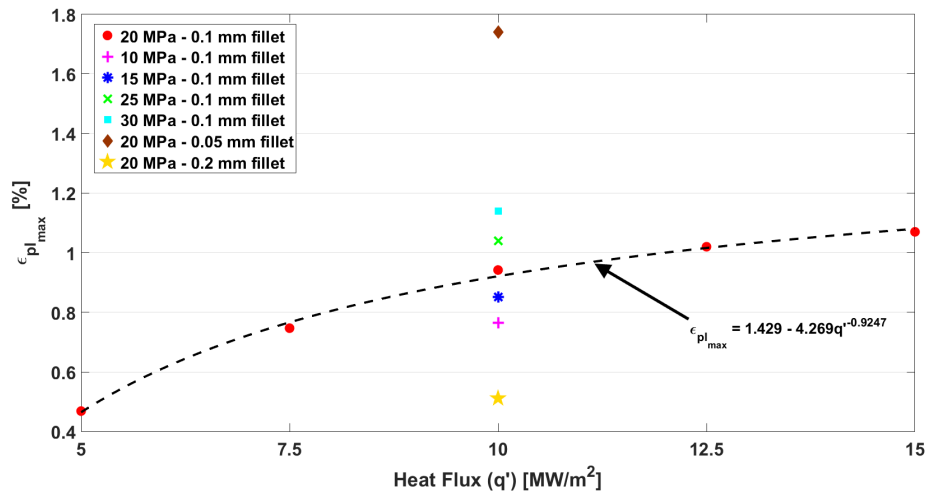


Figure 5.15: Peak equivalent plastic strain in fillet regions - see region A of Fig. 5.11 - of HPJC-1a module after shakedown for various boundary conditions and fillet sizes.

In the ASME Boiler and Pressure Vessel Code [203] the maximum allowable accumulated plastic strain is 3%. Using this as a guideline we can see that in all cases considered the deformation in the fillet regions of the HPJC-1a are within acceptable limits. With increasing

heat flux it tends towards a maximum value of $\sim 1.4\%$ for a fillet radius of 0.1 mm, which is based on the equation in Fig. 5.15.

Fig. 5.15 also highlights the impact fillet size has on the peak equivalent plastic strain, in comparison to variations in the coolant pressure. The results show that it is important to ensure an adequate fillet size is employed in the design.

Reserve factors against both local and global fast fracture were found to be above four for all paths, and so have not been plotted here. The postulated initial crack depth in each case was assumed to be a quarter of the corresponding path length, which is based on methodology set out in the SDC-IC.

5.2.3 Fatigue Life

In preceding sections the HPJC-1a was shown to only fail via monotonic damage for coolant pressures above 30 MPa. In this section, the author considers failure from cyclic loading. In particular, the fatigue life of the component was calculated using the peak equivalent elastic strain range after shakedown.

In Sec. 2.1 of Chpt. 2 the design criteria for a DEMO divertor target plate component were given. For two full-power years (fpys) of operation it is expected to survive 5,000-10,000 thermal cycles. These figures are based on the number of fusion pulses, which translates directly to the required fatigue life in this instance. The reason for the variation in the fatigue life is the uncertainty in a DEMO pulse length [275]. In Fig. 5.16 the results for the HPJC-1a are plotted for a range of different boundary conditions - the 5 MW/m² heat flux case was excluded as the fatigue life was in excess of 10⁹ cycles.

There is only a 21% reduction in fatigue life as the heat flux is increased from 7.5-15 MW/m². This is a consequence of the insensitivity of the peak equivalent elastic strain during a pulse to the heat flux magnitude. As this strain dominates the strain range, it leads to the observed variation in fatigue life with heat flux.

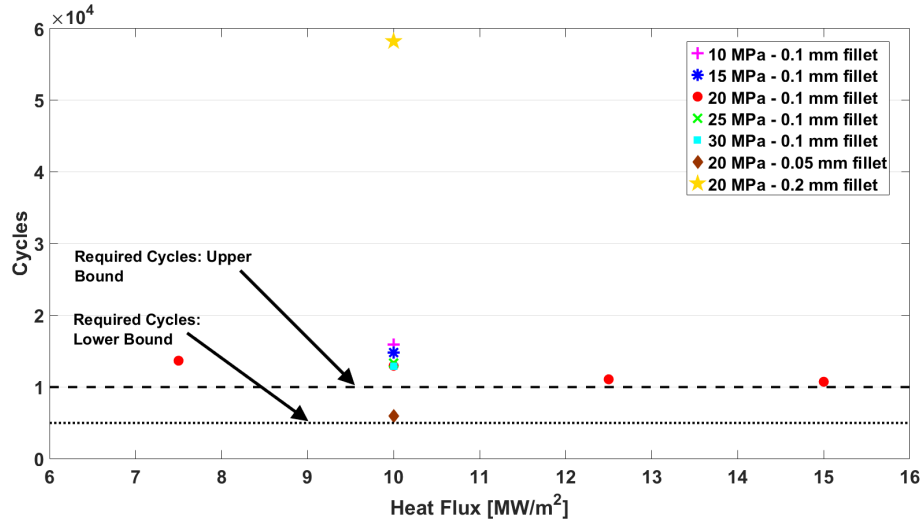


Figure 5.16: Variation of fatigue life of the reference HPJC-1a module under varying boundary conditions.

The results show that changes in the coolant pressure have minimal impact on fatigue life: the difference between a coolant pressure of 10 MPa and 30 MPa is $\sim 24\%$.

By far the strongest influence on fatigue life is observed with changes to the fillet radii. An increase from 0.1 to 0.2 mm is seen to improve the fatigue life more than five-fold. In contrast, halving the fillet radius to 0.05 mm only results in an equivalent halving of the fatigue life. When the fillet radius is reduced to 0.05 mm it leads to the life dropping below 10,000 cycles. For all geometries and boundary conditions considered the fatigue life is above the lower bound of 5,000 cycles.

5.3 Summary

CFD and **FEA** simulations have been performed to allow assessment of the HPJC-1a against the **SDC-IC**. The results indicate that the design is capable of handling heat fluxes in excess of 15 MW/m^2 , at a maximum coolant pressure of 25 MPa. For cases satisfying these conditions both the monotonic damage, and fatigue life, criteria for a **DEMO** target component were met. More detailed information is given below:

- Using the sub-modelling capability within ANSYS Mechanical 16.2 was shown to be an efficient way of simulating small regions of plasticity, and the impact of local geometry changes, within a larger elastic body;
- Parameters were determined for a Chaboche **KINH** model of CuCrZr that could be used to simulate shakedown. The values were validated by simulating a uni-axial tensile test, and comparing the results against the material data in the **SDC-IC** App. A. An excellent agreement was found;
- Approximately 99% of the total equivalent plastic deformation during shakedown was found to occur during the first ten thermal cycles;
- The comparison against the **SDC-IC** revealed the following:
 - Reserve factors against immediate plastic collapse remained above one up to coolant pressures of 30 MPa;
 - Minimum reserve factors against plastic instability occurred at the mid-point of the cavity in the impingement plate. For a coolant pressure of 30 MPa, the reserve factor in this region was below 1;
 - To obtain a minimum reserve factor of two against plastic instability, coolant pressures must be smaller than ~ 15 MPa;
 - All the reserve factors against immediate plastic flow localisation, and local fracture, were above one for heat fluxes up to at least 15 MW/m^2 . Minimum values occurred in fillet regions, and on the surface of the impingement plate near the centre of the hexagon;
 - The peak equivalent plastic strain after shakedown was below 1.8% for all mechanical and thermal boundary conditions considered. The fillet size was found to have the strongest influence on this value;
 - A fatigue life above 5,000 cycles was achieved for all thermal and mechanical boundary conditions;

- The fatigue life dropped by 21% going from a heat flux of 7.5-15 MW/m²;
- The most significant variation in fatigue life was with fillet radius: when increased from 0.1 mm to 0.2 mm, the fatigue life improved five-fold;

Chapter 6

Module Development

“I’m all in favour of the democratic principle that one idiot is as good as one genius, but I draw the line when someone takes the next step and concludes that two idiots are better than one genius.”

— Leo Szilard

In preceding chapters the viability and benefits of the **HPJC** concept have been shown using both numerical and experimental methods. These results have also indicated areas where improvements to the reference HPJC-1a design could be made. In addition, to increase the **Technology Readiness Level** (**TRL**) of the **HPJC** concept it is important to consider how it could be manufactured and deployed in a divertor target plate.

This chapter focusses on the development of a second generation **HPJC** concept. After discussing the motivation behind changes to the HPJC-1a, numerical simulations are performed on the new design to ascertain its thermo-fluid performance.

6.1 Module Sizing Considerations

It would require $\sim 1,850$ HPJC-1a modules to cover the **Outer Vertical Target** (**OVT**) of a single **ITER** divertor cassette, which would correspond to $\sim 200,000$ units for the whole divertor area. This number could be reduced by a factor of four if they were only used in a poloidal region ± 10 cm about the strike-point. However, this number of units would

still pose a significant challenge from both a manufacturing, and manifolding, perspective. **DEMO** will likely be bigger than **ITER**, which means that this figure only serves as a lower bound. In this section, the author considers the practicalities of increasing the size of the HPJC-1a.

The size of the HPJC-1a impingement plate can be increased by adding additional triangular cavities. This can be done without compromising the high pressure handling capability of the design. To feed these additional cavities with coolant there are two options: 1) increase the total mass flow rate to the module; 2) employ additional cascades. The drawback of 1), is that feeding a higher total jet flow area requires increasing the size of the channels in the cascade plate to maintain the same flow area ratio. Employing additional cascades would avoid this issue, but could be detrimental to the heat transfer performance.

6.1.1 Cascade Impingement Modelling

To determine the impact of additional cascades on the heat transfer performance of a **HPJC**-type concept an analytical model was developed based on the following idealised assumptions about the system:

- The geometry of the impingement and jet plates are equivalent for each jet in the cascade;
- All of the jets are exposed to an equal portion of the applied heat load;
- The total mass flow is split evenly between the first jets in the cascade.

Given a local mass flow rate of $\dot{m}_n$ to an individual jet, the total mass flow rate to the module is given by

$$\dot{m}_{n_{\text{tot}}} = \frac{\dot{m}_n s_j}{n}, \quad (6.1)$$

where s_j is the total number of jets in the system, and n the number of cascades; the case

$n = 1$ corresponds to the non-cascading case. It is assumed that n is a factor of s_j .

The wall overhear parameter from Chpt.4 provides a measure of the heat transfer performance of a system. To derive this quantity for the cascading model, an expression is required for the fluid temperature change through a module with n cascades. Using the assumption that each jet is exposed to an equal portion of the heat load, $\dot{Q}$, this is given by

$$T_{ce} - T_{ci} = \frac{\dot{Q}}{s_j \dot{m}_n} \sum_{i=1}^n \frac{1}{c_{p_i}}, \quad (6.2)$$

where c_{p_i} is the specific heat capacity for the i^{th} cascade, and the other variables have the same definitions as in Chpt.4. The jet mass flow rate in equation (6.2) can be expressed in terms of the total mass flow rate using equation (6.1) to give

$$T_{ce} - T_{ci} = \frac{\dot{Q}}{\dot{m}_{n_{\text{tot}}} n} \sum_{i=1}^n \frac{1}{c_{p_i}}. \quad (6.3)$$

For a reference heat sink temperature, T_s , the wall overhear can be expressed using equation (6.3) as

$$\begin{aligned} \tau_{s_n} &= \frac{T_s - T_{ci}}{T_{ce} - T_{ci}} \\ &= \frac{n \dot{m}_{n_{\text{tot}}} (T_s - T_{ci})}{\dot{Q} \sum_{i=1}^n \frac{1}{c_{p_i}}}. \end{aligned} \quad (6.4)$$

Equation (6.4) can be used to develop an expression for the ratio of the non-cascade (τ_{s_1}), to cascading, cases. This is given by

$$\begin{aligned}\frac{\tau_{s1}}{\tau_{sn}} &= \frac{\dot{m}_{1\text{tot}}}{\dot{m}_{n\text{tot}}} \frac{c_{p1} \sum_{i=1}^n \frac{1}{c_{pi}}}{n} \\ &= \omega \frac{\left(1 + c_{p1} \sum_{i=2}^n \frac{1}{c_{pi}}\right)}{n},\end{aligned}\tag{6.5}$$

where $\omega = \frac{\dot{m}_{1\text{tot}}}{\dot{m}_{n\text{tot}}}$. To produce equation (6.5) the following assumptions have been made about the non-cascade and cascading systems: 1) they both operate at the same value of $T_s - T_{ci}$; 2) the heat input is the same; 3) the total number of jets are the same. A value of $\frac{\tau_{s1}}{\tau_{sn}}$ greater than unity indicates that cascading improves the heat transfer performance.

From equation (6.5) it can be seen that the variation of specific heat capacity with both temperature and pressure will influence the benefits of cascading. If the specific heat capacity increases as the flow cascades, then the following occurs in relation to the constant specific heat capacity case: a) the value of ω reduces because the increasing Prandtl number enhances the heat transfer coefficient; b) the fraction in equation (6.5) becomes smaller than unity. As b) dominates the overall influence, an increasing specific heat capacity is seen as detrimental to the benefits of employing a cascade system.

For the case of a constant specific heat capacity, equation (6.5) reduces to

$$\frac{\tau_{s1}}{\tau_{sn}} = \omega.$$

In this instance, the wall overhear decreases with cascading for cases in which $\omega > 1$.

6.1.1.1 Model Validation

To determine the validity of the analytical model it was compared against the simulation results for the cascade and non-cascade HPJC-1a designs from Chpt.4. Solving the expressions derived in the previous section for the case of a variable specific heat capacity required

an iterative approach. A program was written in MATLAB 2016b to carry out this task, with fluid properties for water interpolated from an input file using data from the [National Institute for Standards and Technology \(NIST\)](#) fluid library [\[192\]](#). It was assumed that the flow through each cavity could be represented by a pressure drop across the jet, followed by heat addition at the impingement surface at constant pressure. A 1D model for the thermal resistance was used to determine T_s based on material properties for CuCrZr from the [Structural Design Criteria for ITER In-vessel Components \(SDC-IC\)](#) App. A, and the heat transfer correlation of [\[253\]](#) plotted in Fig. [3.13](#) of Chpt. [3](#). Fluid properties for the correlation were taken at the film temperature, and the inlet temperature of 180°C used as a reference value.

To limit discrepancies caused by 2D thermal effects, area averaged values of temperature on the heat flux surface were used in equation [\(6.4\)](#). This was defined for a system with n cascades as

$$T_s = \frac{1}{n} \sum_{i=1}^n T_{s_n}, \quad (6.6)$$

where T_{s_n} is the temperature of the surface in the n^{th} cascade. To differentiate this from the wall overhear discussed in Chpt. [4](#) it was given the symbol τ_s^+ . The results of this analysis are shown in Fig. [6.1](#).

There is excellent agreement with the numerical simulations for both cascade systems over the range of m^* values considered. These results indicate that the analytical model is capable of estimating the heat transfer performance of a [HPJC](#)-type design for varying numbers of cascades.

6.1.1.2 Influence of Cascading on the Wall Overheat

The MATLAB 2016b program from the previous section was run for the additional cases of three and six cascades, under the same thermal boundary conditions. These cascade

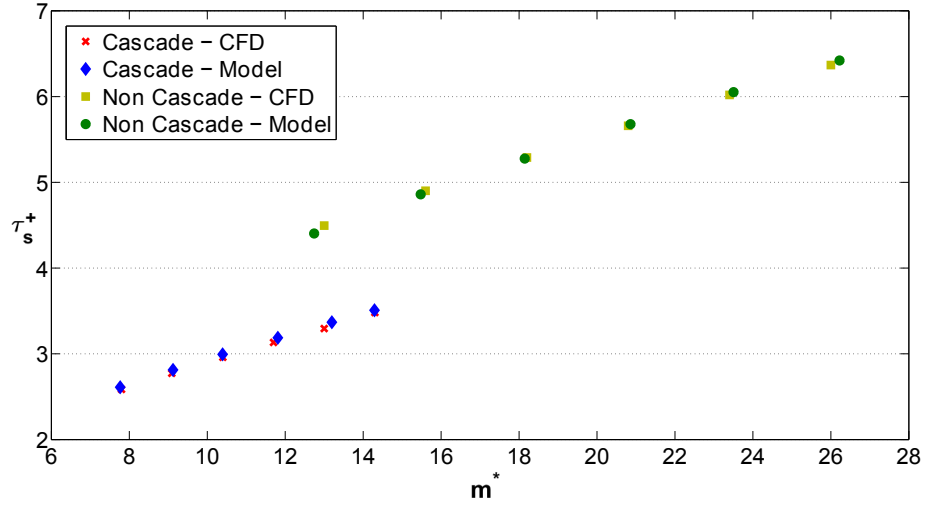


Figure 6.1: Comparison between analytical cascade model and numerical simulations of the modified wall overhead (τ_s^+) for the HPJC-1a. Data is shown for both the non-cascade and cascade version of the HPJC-1a.

values were chosen to allow a total of six jets; the number used in the HPJC-1a. It was also assumed that the cavity and jet dimensions were equivalent to those of the HPJC-1a. Additionally, the Reynolds number exponent in the heat transfer correlation was varied to determine if it has a significant influence on performance. All of the results were referenced to the HPJC-1a non-cascade case (τ_{s1}), and plotted against ω - see Fig 6.2

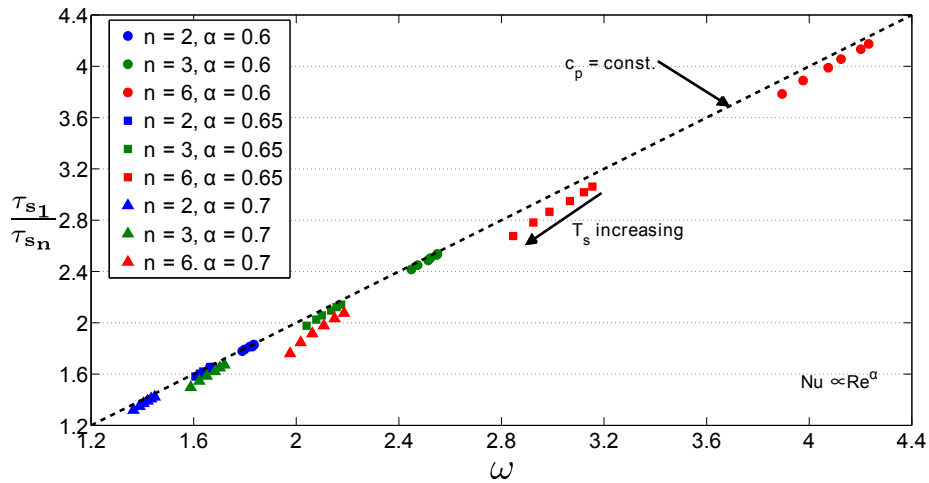


Figure 6.2: Influence of increasing the number of cascades on the wall overhead, scaled with respect to the non-cascade ($n = 1$) case.

Fig 6.2 shows that using additional cascades in the HPJC-1a reduces the wall overhead. If

data for the different number of cascades is compared at equivalent values of T_s ($\alpha = \text{const.}$) it can be shown that the reduction in relative wall overhear with the number of cascades is approximately $n^{-0.75}$. Above four cascades, the variation drops below 5% percent. This value was chosen by the author for the next generation design as it provides a good compromise between heat transfer performance and cascade plate design complexity.

The value of the Reynolds number exponent has a significant influence on the benefits of employing cascading. It reveals that it is best to employ cascading in systems where the heat transfer coefficient has a small mass flow rate dependence. Additionally, this dependence on the Reynolds number exponent becomes more pronounced as the number of cascades is increased.

The variation from the constant specific heat case becomes more prominent as both the Reynolds number exponent, and the number of cascades, is increased. This is due to the larger changes in coolant pressure and temperature which occur for these systems.

6.2 The HPJC-1b

Based on the results of the previous section a second generation HPJC-1a was designed. Termed the HPJC-1b, dimensioned drawings of both the impingement plate cavity structure and cascade channel architecture are shown in Fig. 6.3

In comparison to the HPJC-1a, the hexagonal side length was doubled, and the number of cavities increased from six to twenty four. This would reduce the total number of units required for a region ± 10 cm about the ITER divertor strike-point to $\sim 12,500$, compared to 50,000 for the HPJC-1a.

The Open Area Ratio (OAR) of the cavities was kept the same as HPJC-1a, but the Z/D ratio was increased to 2.25 by enlarging the separation from the target plate, in order to improve the convective heat transfer performance. Fillets were also added at the triangular

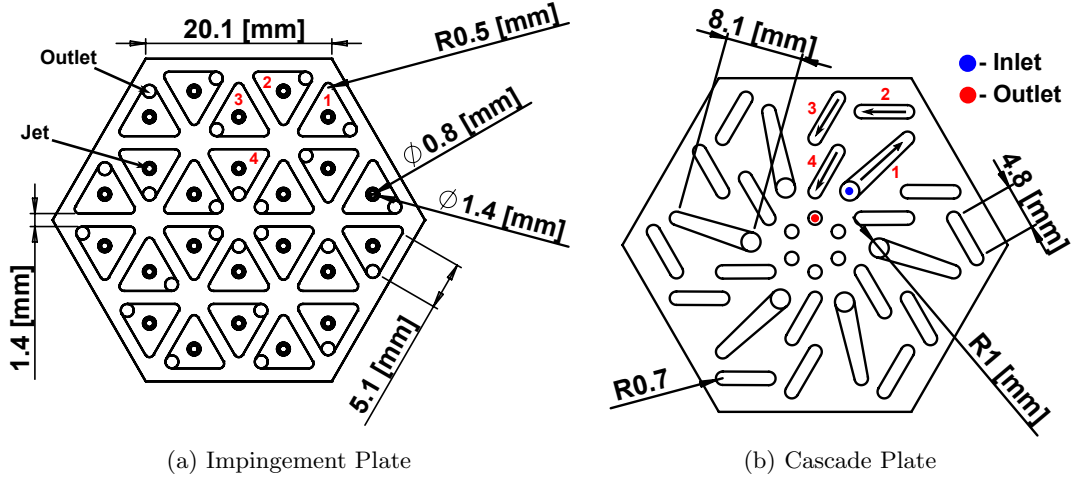


Figure 6.3: (a) Dimensional drawing of the HPJC-1b impingement plate with the cascade order indicated. (b) Dimensional drawing of the HPJC-1b cascade plate with flow arrows to indicate cascade path.

vertices to aid manufacture and reduce regions of low convective heat transfer. To increase the reserve factor against plastic instability in the cavity side walls above two, at a coolant pressure of 20 MPa, the material thickness was increased from 1.2 to 1.4 mm.

Flow is initially fed to six cavities at the outer edges of the module, labelled (1) in image (a) of Fig. 6.3. This was done to minimise the peak temperatures at the hexagon vertices that was noted in Sec. 3.4.2 of Chpt. 3. The flow then impinges a further three times, before exiting at the module centre. In contrast to the HPJC-1a, the flow is only split as it enters the module, which reduces the potential for non-uniform coolant flow distribution between subsequent jets in the cascade.

The total number of laminates in the stack was kept the same in order to minimise the number of bonding interfaces. An exploded CAD image of the plates is shown in Fig. 6.4 together with scaled temperature streamlines from an associated CFD analysis.

As the properties of CuCrZr are highly dependent on its heat treatment history [62, 311], the cascade plate was made from 316L austenitic stainless steel in order to avoid any sub-

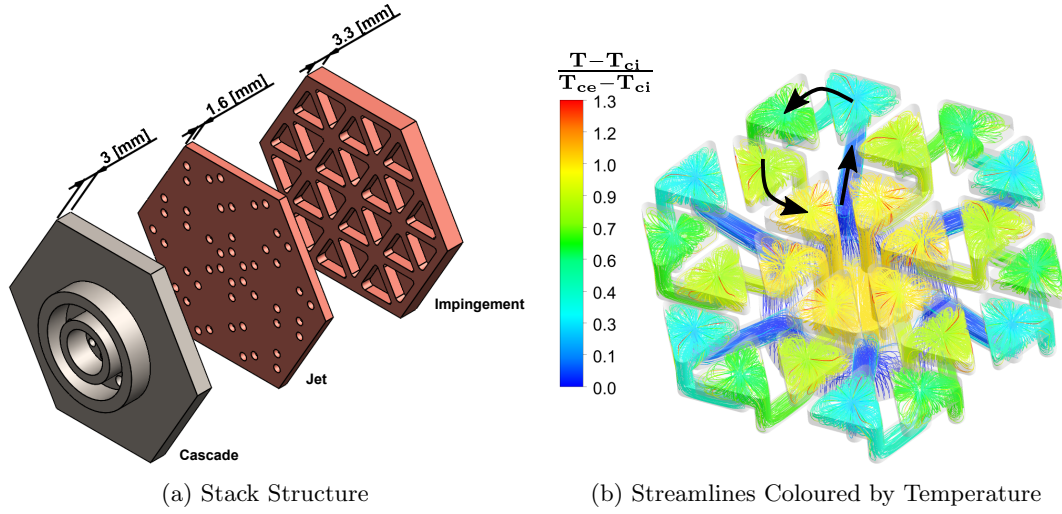


Figure 6.4: Exploded view of laminates in stack (a), together with scaled temperature streamlines from associated CFD (b).

sequent issues bonding the modules to a manifold structure. This approach makes use of the large number of joining techniques that have already been developed in the fusion community for stainless steels [246]. Over a temperature window of 100-300°C, the thermal expansion coefficient of 316L stainless steel is also within 10% of that for CuCrZr [123, App. A], which will minimise thermal stresses at the bonding interface. Further, the ITER Basic Performance Phase (BPP) study [239, 272, 273] showed that 316L stainless steel is capable of operating in a temperature window of 100-250°C up to ~ 10 displacements per atom (dpa). This upper dpa limit would be sufficient for a near-term DEMO divertor target component.

One area of concern for a CuCrZr/316L composite is galvanic corrosion. In such cells it is the less noble alloy that is depleted. For most operating conditions this will be the 316L stainless steel. If high purity water is employed in the coolant loop, then the electrical conductivity will be low. Under such conditions [28] calculated that the potential difference between CuCrZr and 316L stainless steel would be $< 1 \times 10^{-6}$ A/m². This result implies that, under typical ITER operating conditions, galvanic corrosion would not be a lifetime limiting factor [28].

6.3 Performance Comparison with the HPJC-1a

In this section conjugate **CFD** is used to compare the HPJC-1b against the HPJC-1a. The main aim is to determine whether the modifications made to the HPJC-1a design, to produce the HPJC-1b, lead to an improvement in the heat transfer performance.

6.3.1 Numerical Methodology

Steady-state conjugate **CFD** simulations were performed on a 60° periodic segment of the HPJC-1b design using ANSYS CFX 16.2. The laminate structure was modelled as two solid bodies: one body was comprised of the jet and impingement plates, whilst the other was the cascade plate. This allowed the simulation of a mixed material construction. All of the material properties were taken from the **SDC-IC** App. A and modelled as temperature dependent. Thermal contact resistance between the plates was assumed negligible. At the interface between the two solid bodies the normal component of heat flux was conserved.

The meshing procedure used was the same as that for the HPJC-1a, which is outlined in Sec. **3.3.1** of Chpt. **3**. The final mesh had approximately 3.5 million elements in the fluid region, and 1.5 million in the solid. In image (b) of Fig. **6.5** the mesh at the fluid-to-solid interface in the body incorporating the impingement plate is shown, together with a cross-section through an impingement region.

The boundary conditions applied to the **CFD** model are summarised in image (a) of Fig. **6.5**. These were the same as used for the numerical modelling of the HPJC-1a, namely: a mass flow rate and static temperature at the inlet; an area averaged static pressure at the outlet of 0 Pa; a uniform heat flux on the plasma facing side of the impingement plate. The remaining non-periodic external faces were all set as adiabatic.

The coolant was modelled as single-phase water using properties from the in-built **IAPWS-97** thermodynamic properties library. For the HPJC-1a, the **SST** turbulence model provided the most accurate fit to the experimental results - see Sec. **3.5.3** of Chpt. **3**. As the flow field

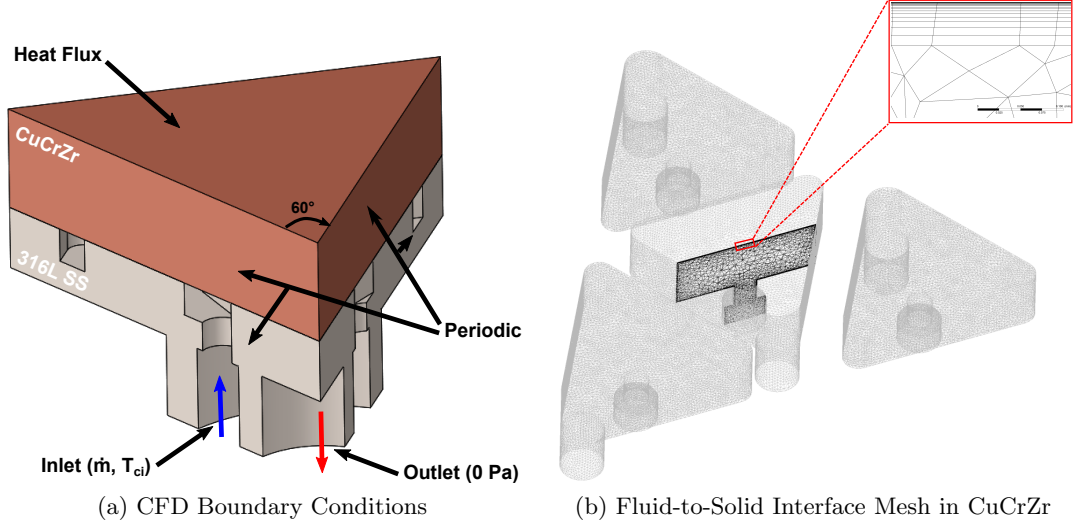


Figure 6.5: (a) Conjugate CFD boundary conditions. (b) Mesh at fluid-to-solid interface in the body incorporating the impingement plate. A magnified image of prism layers is also shown.

is expected to be of the same form in the HPJC-1b, it was also used for this work. All of the remaining methodology was the same as that outlined in Sec. 3.3 of Chpt. 3.

6.3.2 Reference Case Comparison of the HPJC-1b & HPJC-1a

The HPJC-1b was run at the reference case conditions from Sec. 3.4 of Chpt. 3, namely: a 10 MW/m² heat flux; a 20 MPa coolant inlet pressure; a coolant inlet temperature of 180°C. The body containing the impingement plate was set as CuCrZr, and the cascade plate body as 316L stainless steel. A maximum CuCrZr temperature on the heat flux surface of 350°C was achieved by varying the coolant inlet mass flow rate. In Table 6.1 key quantities from this simulation are compared against the results for the HPJC-1a under equivalent boundary conditions.

At an equivalent mass flow rate per unit area, increasing the number of cascades leads to an enhancement of the convective heat transfer coefficient due to the higher mass flow rate through an individual jet. This mechanism allows the HPJC-1b to run at ~29% lower mass flow rates per unit area than the HPJC-1a. Consequently, the coolant temperature change is also seen to be higher in the HPJC-1b.

Concept	$\frac{\dot{m}}{A}$ [kg/m ² s]	$\Delta P_{\text{tot.}}$ [kPa]	h_c [kW/m ² K]	ΔT_{fluid} [°C]	P_w [kW/m ²]	v_{jet} [m/s]
HPJC-1a	56	140	83	41	8.6	10
HPJC-1b	40	750	125	57	33	16

Table 6.1: Comparison of the HPJC-1a and HPJC-1b designs for a 10 MW/m² heat flux and equivalent peak heat sink temperature of 350°C. P_w is the pumping power per square metre.

The increased mass flow rate through each individual jet in the HPJC-1b leads to a higher total pressure drop for each cascade. This results in the total pressure drop across the whole HPJC-1b module being more than five times that of the HPJC-1a. Due to the reduced mass flow rate per unit area, the pumping power requirements are only approximately four-fold higher in the HPJC-1b. Whether this is offset by the increased coolant temperature change will depend on the efficiency of the thermal cycle used for power generation.

Contour plots of the Chilton j-factor [120] on the impingement surface of the HPJC-1b are shown in image (a) of Fig. 6.6. Even though the mass flow rate to an individual jet is approximately 50% higher in the HPJC-1b, the peak convective heat transfer values are 30% smaller than those in the HPJC-1a. However, the area averaged values were found to be ~5% higher. This is due to the increased Z/D ratio, which leads to additional entrainment of fluid into the jet core region. As a result, the jet velocity profile becomes broader, and the centreline velocity is reduced [8, Chpt.1]. The resulting broader convective heat transfer profile reduces temperature non-uniformity.

One of the potential challenges of operating a cascade system is that the increasing coolant inlet temperature to each jet in the cascade can result in large temperature non-uniformities in the solid. This feature becomes more prominent as the number of cascades is increased. It would therefore be expected that the HPJC-1b would have higher temperature non-uniformity on the heat flux surface in comparison to the HPJC-1a. However, image (b) in Fig. 6.6 actually shows that the HPJC-1b has a maximum temperature difference on the

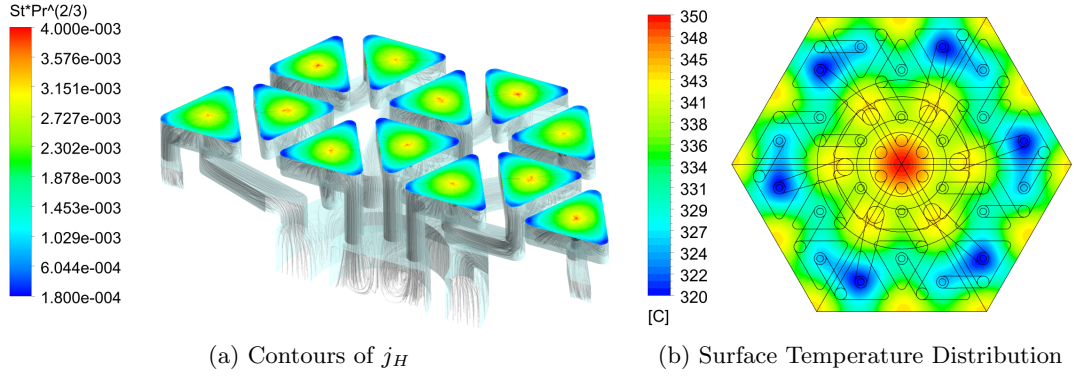


Figure 6.6: (a) Contour plot of j_H over the impingement surface for half of the HPJC-1b geometry. Flow streamlines also shown: darker regions of a streamline represent higher flow velocity. (b) Temperature distribution on heat flux surface.

heat flux surface that is 2°C smaller than that for the HPJC-1a- see Fig 3.9 in Chpt.3. This is due to two features of the HPJC-1b design: 1) the flatter convective heat transfer profile; 2) the arrangement of the cascading order such that the lowest jet inlet temperature is used as close to the location of maximum conductive resistance as possible. This latter point aids in reducing any peaking of the temperature distribution.

6.3.3 Wall Overheat

In Fig 6.7, the wall overheat performance of the HPJC-1b and HPJC-1a are compared for a range of m^* values. The non-cascade version of the HPJC-1a has also been added for completeness.

Over the range of m^* values considered, the HPJC-1b has a wall overheat that is approximately 36% smaller than that of the HPJC-1a. This is in spite of a 40% increase in the conductive thermal resistance due to the increased impingement plate thickness. There are three primary reasons for this improvement in the heat transfer performance, in addition to the increased number of cascades: 1) a 22% increase in the ratio of the internal wetted, to heat flux surface, area; 2) a $\sim 5\%$ enhancement of the area averaged convective heat transfer on the impingement surface; 3) application of the primary cascade as close to the location of maximum conductive resistance as possible to reduce temperature peaking.

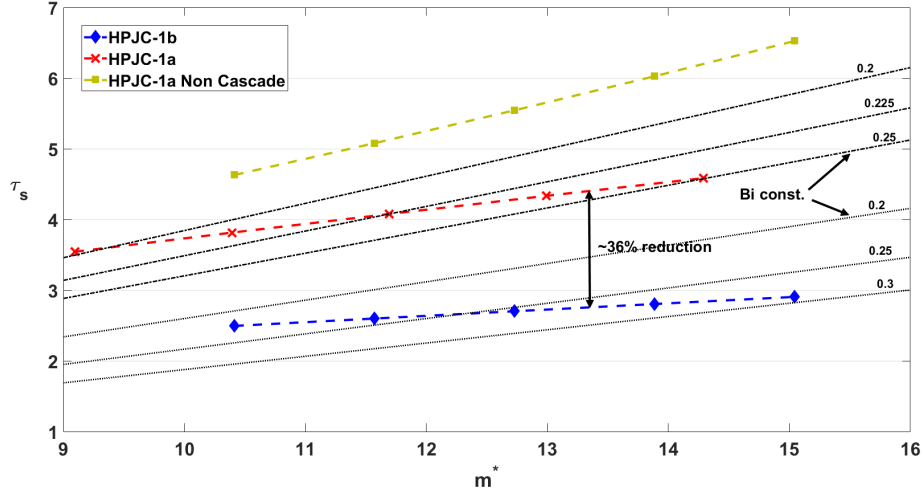


Figure 6.7: Comparison of the HPJC-1b wall overhear performance against the cascade, and non-cascade, versions of the HPJC-1a. Lines of constant Biot number have also been added using equation (4.10) from Chpt.4. The corresponding values were used for $\frac{t}{\sqrt{A}}$: 0.064 for the HPJC-1a designs; 0.043 for the HPJC-1b.

Lines of constant Biot number have also been added to Fig.6.7 using equation (4.10) from Chpt.4 and values of $\frac{t}{\sqrt{A}}$ from the corresponding geometries. At an equivalent m^* , the HPJC-1b is seen to operate at a higher Biot number than the HPJC-1a. This is predominantly caused by the increase in the impingement plate thickness, as operation at an equivalent m^* actually produces the same mass flow rate through an individual jet in both designs.

6.4 Summary

A second generation HPJC-type module, called the HPJC-1b, has been developed. The key findings of the chapter are as follows:

- Employing further cascading in a HPJC-1a-type architecture reduces the wall overhear. The percentage improvement reduces for each additional cascade, dropping below 5% for more than four;
- A fluid that has a specific heat capacity that increases with each cascade is most beneficial in reducing the wall overhear;

- For equivalent thermal boundary conditions, the HPJC-1b operates at 29% lower mass flow rates, and requires ~ 4 times the pumping power, of the HPJC-1a;
- The wall overheat of the HPJC-1b is $\sim 36\%$ lower than that for the HPJC-1a;
- Additional key aspects of the HPJC-1b that improved its heat transfer performance in relation to the HPJC-1a were as follows:
 1. A 22% increase in the ratio of internal wetted, to heat flux surface, area;
 2. A $\sim 5\%$ enhancement of the area averaged impingement plate convective heat transfer at an equivalent jet mass flow rate;
 3. Reduction of temperature peaking at the hexagonal vertices by placing the primary cavity as close to this location as possible;

Chapter 7

Fabrication & Non-Destructive Testing

“People must understand that science is inherently neither a potential for good nor for evil. It is a potential to be harnessed by man to do his bidding.”

— Glenn T. Seaborg

One of the key advantages of the **HPJC** design is that it combines a high performance convective fluid-flow architecture with a simple manufacturing process. All of the fabrication techniques required are mature, and available on an industrial scale. In this chapter the author discusses the fabrication and non-destructive testing of a batch of HPJC-1b modules. The purpose of this work was two-fold: 1) to determine steps in the manufacturing process that were important for achieving a high yield; 2) to produce a set of modules for subsequent **HHF** testing.

7.1 Fabrication

7.1.1 Overview

The process path used to fabricate a batch of eight HPJC-1b modules is shown in Fig. **7.1**. Material (C106C) for the jet and impingement plates - 200 mm x 200 mm x 10 mm - was

sourced from Luvata^[1], and had a chemical composition of 0.5-1.0% Cr, 0.05-0.15% Zr, Cu balance [168]. It had been solution annealed, water quenched, and then aged at 400-450°C. The quoted electrical conductivity of the material was 78% International Annealed Copper Standard (IACS).

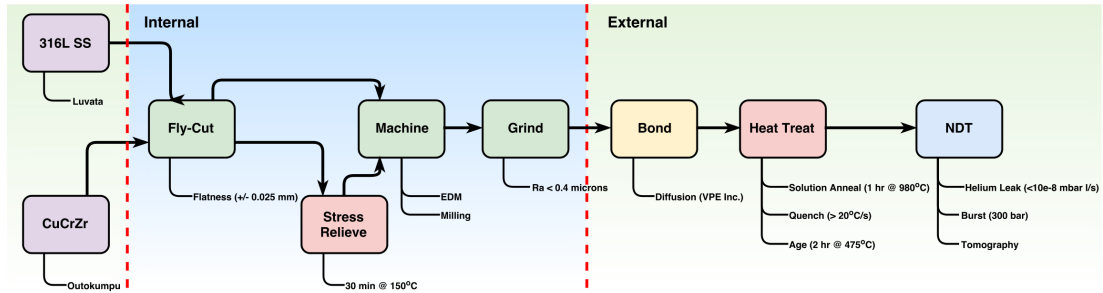


Figure 7.1: The process route used to manufacture an initial batch of eight HPJC-1b heat sink modules. Internal processes were carried out at the workshop in the Osney Thermo-Fluids Laboratory.

Type 316L (1.4404) stainless steel for the cascade plate - 200 x 200 x 30 mm - was procured from Outokumpu^[2]. This material had been cold rolled, and then solution annealed at a minimum of 1050°C, before water quenching.

Before flow features were machined into the plates, they were first fly cut to within 0.2 mm of the required thicknesses: 3.3 mm for the impingement plate; 1.6 mm for the jet plate; 17 mm for the cascade plate. The flatness of the plates during this operation was kept to within ± 0.025 mm to ensure good surface contact during bonding. To avoid the risk of distorting the CuCrZr plates during further machining, because of their reduced thicknesses, they were stress-relieved in atmosphere at 150°C for 30 min [10].

In Chpt. 6 it was estimated that a minimum of 12,500 HPJC-1b modules would be required for an ITER divertor, and that the number for DEMO would be even larger. Therefore, a means of efficiently fabricating multiple units is an important consideration. For the fabrication of the initial batch of HPJC-1b modules a ligature system was trialed for this

¹<http://www.luvata.com/wolverhampton>

²<https://www.outokumpu.com/en/Pages/default.aspx>

purpose. This is shown in Fig.7.2

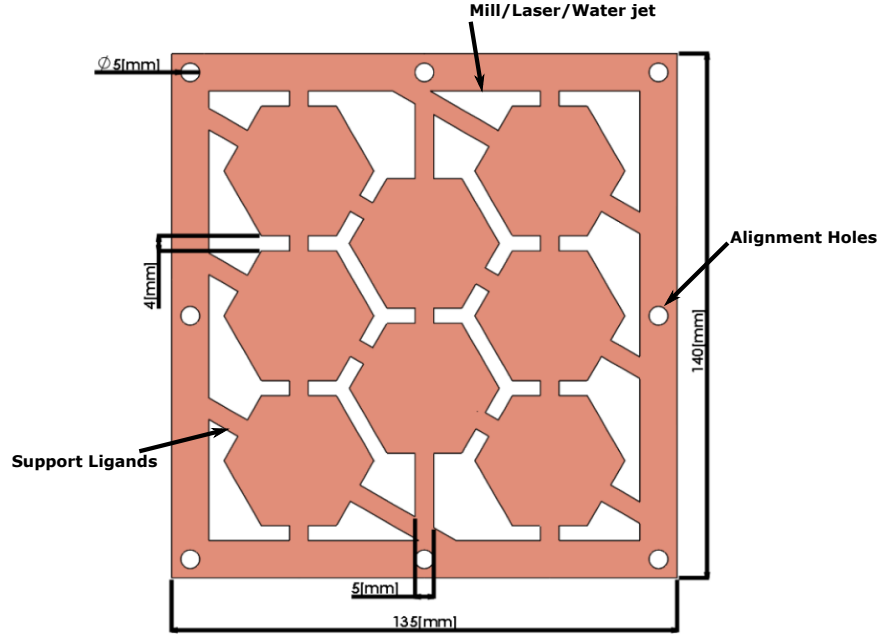


Figure 7.2: Proposed method for manufacturing multiple HPJC-type modules simultaneously.

The outer frame contained a number of holes, 5 mm in diameter, which were used to align the stack during bonding to within ± 0.1 mm.

7.1.2 Plate Machining

A number of different machining methods were considered for the initial batch of HPJC-1b modules: waterjet cutting; laser cutting; milling; Electrical Discharge Machining (EDM); Photo-Chemical Machining (PCM). In Fig.7.3 examples of trials with both EDM and waterjet cutting are shown.

Investment casting or Metal Injection Moulding (MIM) may provide a more economic long-term solution, especially since they would allow the elimination of some of the bonding steps. However, for only eight units, neither are cost-effective. In addition, casting CuCrZr with acceptable material properties is difficult due to the high affinity for oxygen of copper and chromium [244].



Figure 7.3: Machining methods investigated for HPJC-1b fabrication: (a) EDM electrode used to add channels to cascade plate and (b) waterjet cut triangular cavities.

The original intention was to use either waterjet, or laser, cutting to produce the features in the CuCrZr plates. However, the inability of these methods to machine to a fixed depth would have meant splitting the impingement plate; adding an additional bonding interface to the stack. Instead, the cavities were machined directly into the impingement plate. This choice had two additional benefits: 1) it ensured parent material strength in a region of high stress; 2) it allowed a fillet size to be chosen that ensured adequate fatigue life. The decision to machine the cavities into the impingement plate, rather than the jet plate, was based on the life analysis from Chpt. 5, which showed that the lowest reserve factors occurred in the web regions near the impingement surface.

Half-etch PCM was considered for creating the cavities, but was found to be limited to depths $\lesssim 1$ mm [64]. Therefore, milling was finally selected as the machining method for the CuCrZr plates. For the 316L stainless steel cascade plate, EDM was used to create the cascade channels, whilst milling was employed for the inlet and outlet ports. To ensure an even pressure distribution over the plate interfaces during bonding, only the central outlet channel of the concentric feed was milled at this stage. A breakdown of the machining methods used for each plate in the HPJC-1b stack is shown in Fig. 7.4. All of the methods used had a feature accuracy of ≤ 0.1 mm.

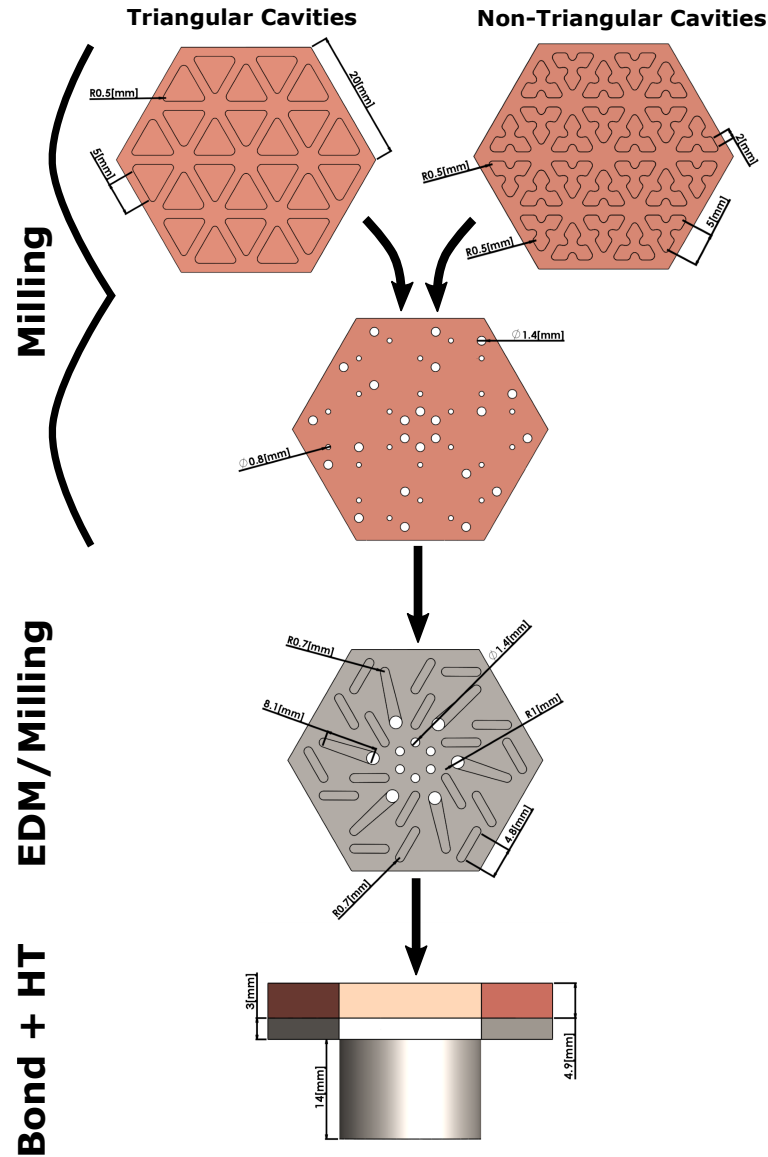


Figure 7.4: Breakdown of the machining methods used for each plate in the HPJC-1b stack.

The bonding method used generally requires that the maximum feature size in an individual layer (1) be no larger than the thickness of the subsequent layer (2) in the stack³. This ensures an even pressure distribution at the bonding interface between (2) and the next layer in the stack. The reference triangular cavity did not fulfil this criteria. For this reason a second impingement plate was designed, which had elliptical features added to the cavity side walls - see Fig. 7.4. These reduced the effective maximum feature size of the cavity to meet the bonding criteria.

³ Advice from staff at VPE Inc.

Once each of individual plates had been machined, the bonding interfaces were ground to a surface roughness smaller than 0.4 Ra^3 . Each of the plates after grinding is shown in Fig. 7.5.

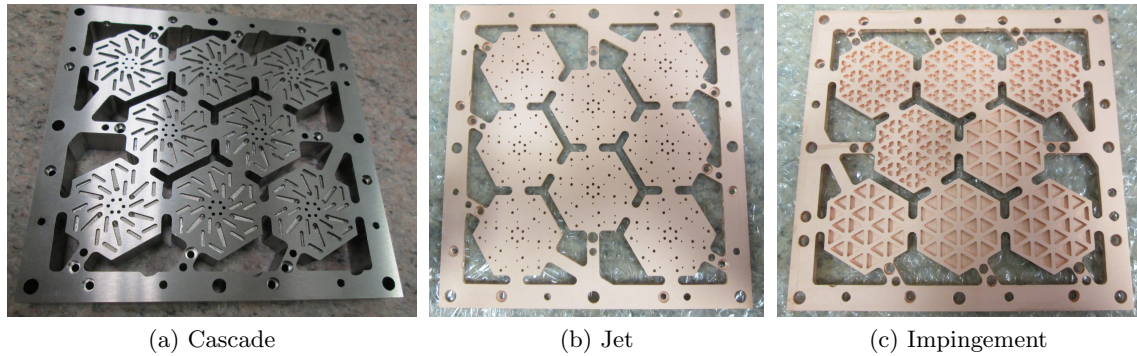


Figure 7.5: Machined, and ground, HPJC-1b multi-module plates prior to copper electroplating in preparation for diffusion bonding.

7.1.3 Bonding and Heat Treatment

Due to the high thermal, and pressure, loading of the HPJC-1b during operation, a near parent material strength bond is needed between the plates. Though there are a number of joining methods that could fulfil this criteria [22, 50, 148, 283], a un-axial pressure diffusion bonding technique was chosen for this work due to its prior success in Printed Circuit Heat Exchanger (PCHE) applications [102].

The bonding was carried out by VPE Inc.⁴ To minimise the risk of bond failure, each of the plates was first electroplated with copper, which has well defined solid-state diffusion characteristics.

After bonding, the stack underwent heat treatment. It was first solution annealed at 980°C for 1 hour. This was followed by a water quench, and then subsequent ageing for 2 hours at 475°C . Both steps of the heat treatment were carried out in a positive pressure argon

⁴<http://www.vpei.com/>

environment to minimise oxidation of the bonding interfaces.

Following heat treatment, the ligatures joining separate modules were removed. The concentric feed system was then milled for each module. An M22 x 1.5 mm thread was machined onto the outer ring of each concentric feed to allow attachment to a manifold. It is envisaged that a similar process could be used in the divertor target region itself, in conjunction with a technique such as the Vacuum Tight Threaded Junction (VTTJ) [11]. This method uses the plastic deformation of a thread, and subsequent electroplating of the joint, to create a seal that can operate at differential pressures of up to 50 MPa [12]. A complete HPJC-1b module is shown in Fig. 7.6.

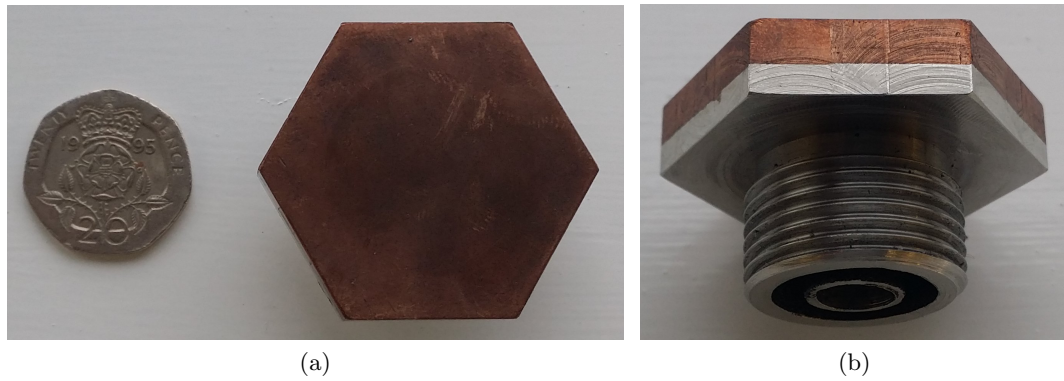


Figure 7.6: Completed HPJC-1b module with thread (b) for connection to a manifold.

7.2 Non-Destructive Testing

After the bonding and heat treatment had been performed, a number of quality control measurements were carried out on the fabricated HPJC-1b modules. The results from these tests were used to determine a manufacturing yield, and ensure safe operation during subsequent HHF testing.

7.2.1 Helium Leak Rate & Hydrostatic Pressure Tests

Leak rate measurements were carried out by VPE Inc. on the fabricated HPJC-1b modules. In [ITER](#), the maximum helium leak rate for an in-vessel component is $10\text{e-}8$ mbar l/s [\[87\]](#). Applying this as a pass criteria to the HPJC-1b modules gave a manufacturing yield of 50%. The full results are shown in [Fig. 7.7](#).

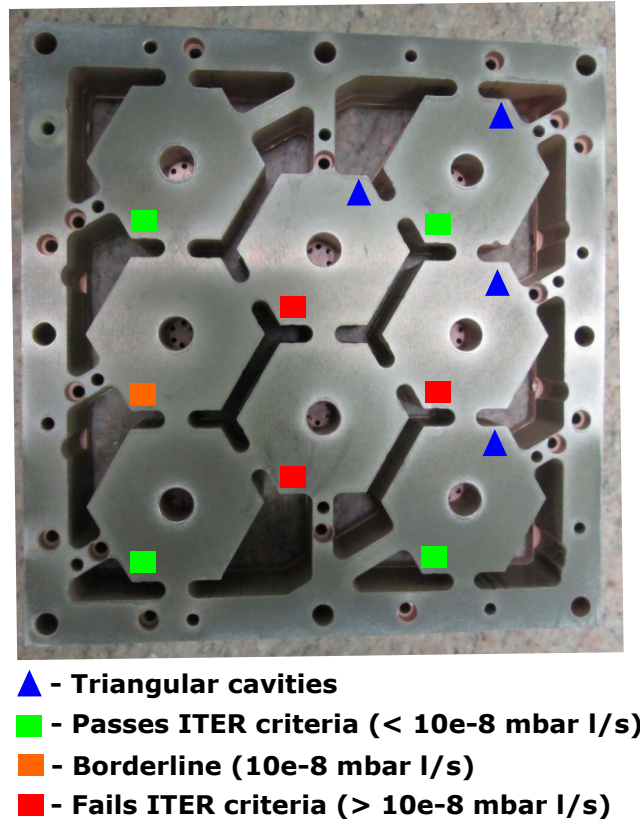


Figure 7.7: Results of the helium leak rate measurements with pass rates indicated based on the [ITER](#) in-vessel component criteria [\[87\]](#).

There is an equal numbers of triangular, and non-triangular, impingement plate designs that pass the [ITER](#) leak rate criteria. This suggests that either: a) the leak occurs at the bonding interface between the jet and impingement plates; b) the feature size on the impingement plate has a minimal influence on the bond quality.

All of the modules that passed the helium leak rate criteria lie at the corners of the plate. This raises the question of whether there is not a radial variation in the pressure, or tem-

perature, occurring during the bonding that influences the results. If the pressure were to increase with radial distance from the centre of the plate, this may explain why the corner modules have a reduced bond porosity.

The four modules that passed the leak rate criteria, and the borderline case, were subjected to hydrostatic pressure tests at 30 MPa. This pressure value was chosen such that

$$P_0 > \frac{\sigma_{y0}}{\sigma_{y1}} \times P_1 \quad (7.1)$$

where the subscripts zero and one represent the test and operational states respectively. Data for the yield strengths in equation (7.1) were taken from [123, 168]. All five modules passed with no visual indications of damage.

7.2.2 X-Ray Tomography & Magnified Bond Cross-Sections

The remaining non-triangular cavity design, that failed the ITER leak rate criteria, underwent x-ray tomography through colleagues at the Culham Centre for Fusion Energy (CCFE). This technique is being developed as a screening tool for the manufacture of DEMO HHF components [67, 68]. It operates by relating the level of x-ray absorption in a material to its density and thickness [130]. By recording the absorption characteristics at a number of different viewing angles, a complete 3D image of the internal structure can be re-produced from 2D cross-sections. It can detect flaws such as porosity and cracks, though the resolution is limited to defects larger than approximately 2% of the wall thickness [130]. The results obtained for the HPJC-1b, at three cross-sectional locations, are shown in Fig. 7.8.

In image (a) of Fig. 7.8 each of the cascade channels can easily be discerned, together with the six outlet holes. The apparent void at the centre of the hexagon, which appears in all of the images, is due to the low penetration of x-rays in this region.

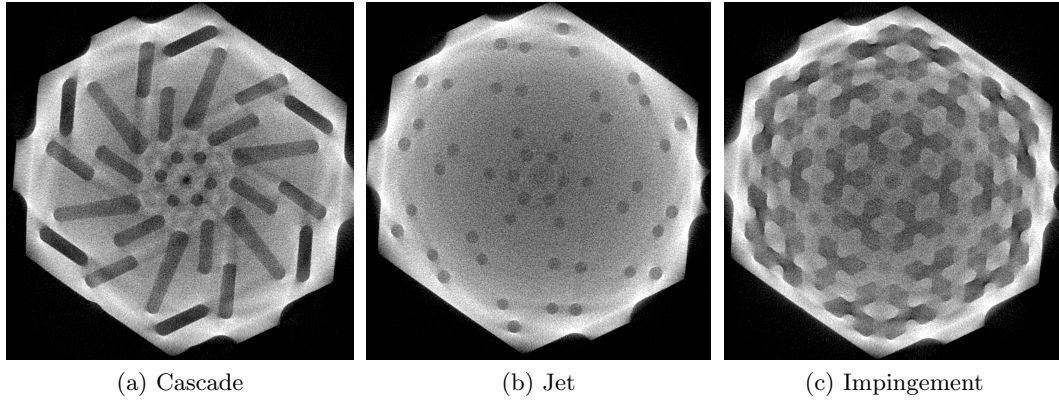


Figure 7.8: X-ray tomography images at the approximate locations of the bonding interfaces for a non-triangular HPJC-1b module.

One of the triangular cavity designs, that failed the ITER leak rate criteria, was cross-sectioned between hexagonal flats by VPE Inc. Magnified images of the bond interfaces between the plates on this cross-section are shown in Fig.7.9.

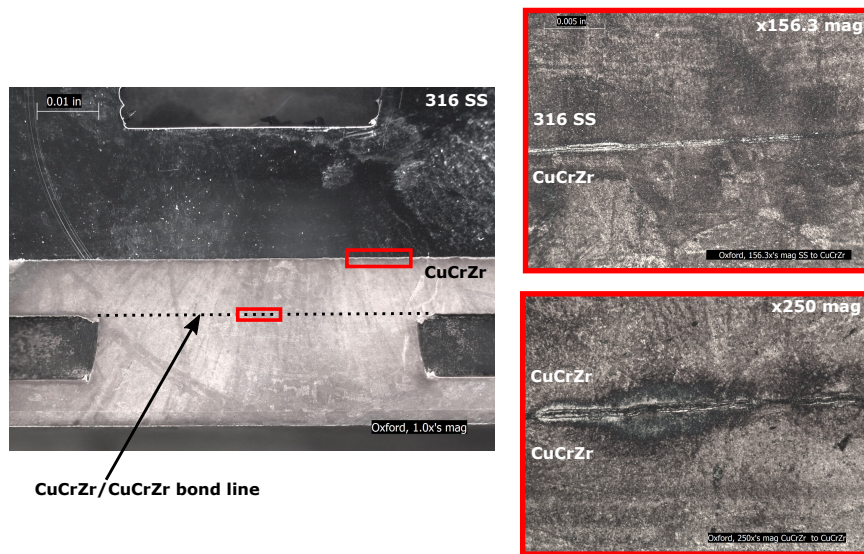


Figure 7.9: Optical images of cross-sectioned HPJC-1b module post bonding. Higher magnification images appear on right. The cross-section was between hexagonal flats.

A solid-state diffusion bond between parts made of the same material should eliminate any interface [59]. This appears to have occurred between the jet and impingement plates if viewed at x1 magnification. However, at closer magnification, it is evident that there is still at least some areas where this is not the case. The results are similar for the CuCrZr-316L

interface, though the lower magnification makes the diffusion appear more complete. These results may explain why this module failed to meet the helium leak rate criteria.

7.3 Summary

A number of HPJC-1b modules have been fabricated, and undergone subsequent non-destructive testing. As part of the development of the fabrication process, the most viable machining methods for the separate laminates were determined. The non-destructive testing identified key areas for improvement of the manufacturing yield. The results of the chapter can be summarised as follows:

- The number of bonding interfaces can be minimised if the triangular cavity structures of the HPJC-1b are machined directly into the impingement plate. This also results in parent material strength in regions of high stress;
- The most appropriate method for machining the triangular cavity architecture into the CuCrZr plates was found to be milling;
- Manufacturing yield was determined by the pass rate of the ITER helium leak rate criteria. For the current batch, a 50% yield was achieved;
- All of the modules that passed the leak rate criteria survived subsequent hydrostatic pressure tests at 30 MPa;
- Magnified images from a cross-section of a module that failed to meet the helium leak rate criteria showed areas of incomplete diffusion at the bond interfaces;

Chapter 8

Construction of a High-Heat Flux Measurement Facility

“The science of today is the technology of tomorrow.”

— Edward Teller

Experiments that match non-dimensional parameters provide a means of validating particular aspects of a design’s performance. However, it is the life of a component that determines its final viability for a particular application. As this depends on the materials and fabrication processes used, it can only be accurately established by experimentation on manufactured components under service-realistic boundary conditions.

In this chapter the design and construction of a **High-Heat Flux (HHF)** facility for testing of the HPJC-1b is discussed. Particular consideration is given to an experimental means of replicating the heat load boundary conditions a divertor target component would see during service.

Facility	Location	Method	Power [kW]	Peak Heat Flux [GW/m ²]	Max. Target Area [cm ²]	Max. Test Length [sec]	Reference
<i>Kinetic</i>							
JUDITH II	FZ Juelich, DE	e ⁻	200	10	5000	Cont.	171
JEBIS	JAERI, JP	e ⁻	400	2	1800	Cont.	106 161 233
OHBIS	JAERI, JP	e ⁻	50	1	0.625	Cont.	252
FE200	CEA, FR	e ⁻	200	60	10000	6000	30 233
EB1200	Sandia, US	e ⁻	1200	10.6	2700	Cont.	304
EBTS	Sandia, US	e ⁻	30	9.5	100	Cont.	233 304
TSEFEY	Efremov Inst., RF	e ⁻	60	0.2	2500	Cont.	176 233
ACT2	NIFS, JP	e ⁻	300	4.5	2500	Cont.	97
Electron Beam	SWIP, CN	e ⁻	60	3	0.25	Cont.	106 162
EBTH	Kyushu Univ., JP	e ⁻	3	0.35	0.5	Cont.	106
KoHLT-EB	Kaeri, KO	e ⁻	300	10	3500	Cont.	139 161
HHFTF	IPR, IN	e ⁻	200	0.7	2700	Cont.	161 270
EM60	SWIP, CN	e ⁻	60	10	100	Cont.	162
HELCZA	CVR, CZ	e ⁻	800	40	18000	Cont.	217
HHFF	Oak Ridge, US	He ⁺ , H ⁺	300	0.08	50	1	277
METF	Oak Ridge, US	H ⁺	3000	3	1000	30	277
PBEF	JAERI, JP	H ⁺ /He ⁺	1500	60/10	1000	1000	269
JET NBTB	JET, UK	D	7500	0.25	800	20	77
GLADIS	IPP, DE	H ⁺ /He ⁺	2200	0.05	3000	Cont.	92 161
<i>Radiative</i>							
FITWATKA	FZ Karlsruhe, DE	IR	500	8e ⁻⁴	1500	Cont.	106 111
CEF 1-2 EDA-BETA	Brasimone, IT	IR	100	8e ⁻⁴	88	~300	106
HHFT	Oak Ridge, US	IR	270	2.7e ⁻³	10	450	245
<i>Electrical Resistive</i>							
HIVE	CCFE, UK	Induction	30	30e ⁻³	25	Cont.	193
HHFTF	Univ. of Texas at El Paso, US	Cartridge	10	1e ⁻⁴	10	Cont.	114
ACTF	NASA, US	Cartridge	~40	8.67e ⁻³	5	Cont.	85
<i>Convective</i>							
MK200UG	TRINIRI, RF	Plasma Gun	1e ⁶	300	<0.39	5e ⁻⁵	106
HEG	DLR, DE	Shock Tunnel	~3e ⁵	2.2e ⁻³	~5000	1e ⁻³	36 100
TH2	RWTH, DE	Shock Tunnel	~1e ⁵	2e ⁻³	~3000	1e ⁻²	36 206

Table 8.1: List of experimental facilities that can achieve fusion relevant heat flux boundary conditions.

8.1 Experimental Methods for Replicating DEMO Divertor Target Plate Heat Loads

The fusion community currently lacks a dedicated tokamak to act as an engineering test bed for DEMO components. To rectify this problem, China is developing the China Fusion Engineering Test Reactor (CFETR) [262]. Until the CFETR becomes operational in the 2030s [45], a range of experimental facilities will be required to replicate the full spectrum of conditions a DEMO divertor will see during operation.

The life of a divertor target component will be determined by one of the following: 1) material erosion/deposition; 2) neutron irradiation damage; 3) steady-state/transient heat fluxes [178]. These phenomena can be inter-dependent. For instance, the erosion of the tungsten armour tile will be accelerated during transient heat loads [86].

The impact of tritium inventory, and material erosion/deposition, on component life are being researched using current tokamaks [210, 235]. Until the International Fusion Material Irradiation Facility (IFMIF) is constructed [142], or an equivalent facility, the availability of data on material behaviour at DEMO levels of neutron damage will be limited.

There has been a range of facilities built by the fusion community to replicate the heat load conditions a divertor will experience during service. A number of these are shown in Table 8.1. This is the potentially life-limiting boundary condition that the Osney Thermo-Fluids Laboratory (OTFL)¹ HHF measurement facility will aim to reproduce. In particular, focus was given to replicating steady-state heat fluxes in the range of 5-10 MW/m² to allow thermo-fluid and life measurements to be made.

In the following sections, potential means of applying these steady-state heat fluxes experimentally are discussed in more detail.

¹<http://www.eng.ox.ac.uk/thermofluids>

8.1.1 Kinetic, Radiative & Convective Methods

Methods that input energy to a surface kinetically generally require beam rastering to achieve a desired heat flux profile. An example of rastering profiles used in the JUDITH II electron beam (e-beam) facility are shown in Fig. 8.1. This technique allows accurate replication of a spatially varying divertor heat load. However, the small beam cross-section can lead to local power densities in the GW/m^2 range, which can deform the material [245]. The availability of e-beam systems with gun powers in excess of 100 kW, and efficiencies above 80%, make them a particularly attractive kinetic method.

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Figure 8.1: Path followed by electron beam for thermal fatigue tests [left] and transient loads [right] in JUDITH II [160].

Though future experimentation of a HPJC-type concept may require such a facility, the current Technology Readiness Level (TRL) of the design calls for a much simpler approach.

Infra-Red (IR) heaters or plasma guns provide a more cost effective means of achieving similar power levels to kinetic methods. The former have good efficiencies, and optical access for instrumentation, but are diffusive. This means that their maximum heat flux levels are generally below those required for a divertor target plate [106]. However, the new Plasma Arc Lamp (PAL) systems, used at the Oak Ridge National Laboratory (ORNL) [161], show that it is possible to achieve heat fluxes of up to $27 \text{ MW}/\text{m}^2$ with such devices.

IR heaters are limited to replicating steady-state heat loads, as their rise times are of the order of 10 ms [245], which is an order of magnitude higher than some plasma disruption time scales [106].

A plasma gun allows the simulation of both heat fluxes and Plasma Wall Interactions (PWIs) simultaneously. However, the small experimental run times limit their application to simulating transient heat loads [106]. In addition, the methods used to generate the plasma are energy inefficient, typically achieving electrical energy conversion efficiencies of only 42-65% [54].

8.1.2 Electrical Resistive Methods

A power input of 10-20 kW would be required to achieve DEMO divertor target plate heat flux magnitudes on the HPJC-1b. The small dimensions of the module also mean that a constant heat flux boundary condition is a realistic replication of in-service conditions. Under these circumstances, electrical resistive heating methods are an attractive alternative due to their low cost and relatively simple operation.

An induction heater operates by passing a high frequency AC current through an electromagnetic coil to create a time-varying magnetic field. This field can induce strong eddy currents in conducting materials placed near the coil, which subsequently heat the material via the Joule effect [241]. The primary benefit of using induction, as opposed to other electrical resistive methods, to apply a heat flux boundary condition, is that no contact is required with the component. This simplifies thermal fatigue testing, as no modelling assumptions have to be made to account for electrical connections. However, the method can only be used to heat conductive materials, and is challenging to use with complex geometries. The efficiency is also highly dependent on achieving good inductive coupling between the electromagnetic coil and test specimen.

Cartridge heater designs operate on the principle of geometric concentration. A number of heater cartridges are placed into a block of conducting material with dimensions larger

than the test specimen. As the cross-sectional area for heat flow is reduced, the heat flux subsequently increases. By combining cartridges that have a high power density, with large geometric magnification, it is possible to generate heat fluxes of the order of 10 MW/m^2 using this method [114]. This technique was used in NASA's Aerojet Carbothermal Test Facility [85], shown in Fig. 8.2.

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Figure 8.2: Example of how cartridge heaters can be used in combination with conducting block to generate HHFs [85].

The major drawback of the cartridge heater method is the necessity for an intimate contact between the conducting block and test specimen to minimise thermal contact resistance. This additional mechanical constraint makes obtaining an accurate life for the component challenging, though thermo-fluid performance can still be obtained. The location of the heater block also limits optical access.

A resistive thin-film applied directly to the specimen surface provides a compromise between the induction and cartridge heater methods. The thermal contact resistance, and mechanical constraints, are reduced relative to the cartridge heater design, whilst the operational complexity of the induction method is removed. Commonly these films are applied

using techniques such as [Chemical Vapour Deposition \(CVD\)](#), [Physical Vapour Deposition \(PVD\)](#), thermal sprays or adhesives [\[82\]](#), in a layered stack featuring an electrical insulator and electrically resistive heating element as a minimum.

The resistive thin-film method was chosen for applying a steady-state heat flux boundary condition in the [OTFL HHF](#) measurement facility. In the proceeding sections the theory behind designing the heaters is developed, and results reported for a number of trial specimens.

8.2 Resistive Thin-Film Heater Design

In research by [\[204\]](#), [\[307\]](#) they used thermally sprayed thin-films to prove the viability of using jet impingement arrays to remove [HHFs](#). By trialling different thermal spray processes, and material combinations, they were eventually able to achieve heat fluxes in excess of 18 MW/m^2 [\[179\]](#). Given the success of this approach, the thermal spray method was also used to apply the films in this research.

8.2.1 Overview of the Thermal Spray Processes

There are a number of industrially available thermal spray processes. In this research [Air Plasma Spray \(APS\)](#) and [High Velocity Oxide Fuel \(HVOF\)](#) were used due to their wide scale availability, and relatively low cost. In the [APS](#) process an inert gas is fed through a torch in which an arc discharge occurs. This discharge heats the gas to temperatures in excess of $\sim 30,000^\circ\text{C}$ [\[280\]](#), dissociating it in the process to form a plasma. The plasma exists the torch through a nozzle, at which point the material to be deposited is introduced into the plume in powder or wire form. Molten splats of injected material impact the substrate surface and rapidly solidify, forming both mechanical and chemical bonds [\[280\]](#). To ensure adequate bond strength, the substrate is commonly roughened using grit blasting [\[257\]](#). By sweeping the torch over the surface thicker deposits can be formed.

The [HVOF](#) process uses combustion as a means of generating a high enthalpy flow, which

melts the injected material. This system operates at lower flame temperatures than APS ($<3000^{\circ}\text{C}$), but at higher molten particle velocities: 100-400 m/s for APS versus 400-1,000 m/s for HVOF [205]. These higher velocities allow for coating adhesive strengths frequently above 69 MPa, compared to typical values of 30-50 for APS [280]. However, the lower peak flame temperature in HVOF reduces the number of viable spray materials.

In Fig. 8.3, the flame temperatures and molten particle velocities are compared for a number of thermal spray processes. The Vacuum Plasma Spray (VPS) method overcomes the low densities of APS coatings caused by oxide pick-up, and retains the ability to deposit a wide range of materials because of the high flame temperatures. It operates by carrying out the spraying in a pressure and atmospherically controlled chamber. The drawback to using this process is that the additional equipment required reduces commercial availability, increasing coating costs.

APS is suited to materials with high melting points, such as ceramics and refractory metals. The higher adhesive strength, and reduced porosity, of HVOF coatings means that this process is preferred where possible.

8.2.2 Film Microstructure

As multiple splats impact the substrate surface and overlap, a lamellar microstructure is formed [257]. The thickness of each splat in this structure is dependent on a number of factors: 1) the molten particle velocity; 2) the molten particle temperature; 3) the substrate temperature. In general, the size is found to be in the region of $\sim 1\text{-}20$ microns [54].

The rapid cooling rate of the splats ($> 10^6$ K/s) results in a fine grained, uniform, polycrystalline coating, which has anisotropic properties parallel and perpendicular to the substrate surface [54, 257]. This high cooling rate generates shear stresses that can lead to de-lamination [279]. As more layers are deposited, residual stresses begin to accumulate. This limits thermally sprayed coatings to thicknesses of the order of 1 mm [279]. If there is a large difference in the thermal expansion coefficients of the substrate and deposit this

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Figure 8.3: Typical flame temperatures and molten deposit velocities for a range of thermal spray processes [205]. Note: CGS stands for Cold Gas Spray.

problem is exacerbated.

Porosities of up to 10% are common in thermally sprayed coatings, and are caused by the following: 1) un-melted/partially melted splats; 2) metastable phases; 3) oxidation from entrained air [54]. Postdeposition treatments such as fusion, sintering and Hot Iso-static Pressing (HIP) can heal splat boundaries, and reduce the porosity level [279]. A cross-section of a thermally sprayed coating with typical defects is shown in Fig. 8.4

To reduce coating de-lamination on flat surfaces, tapering can be used near free edges [276]. Additionally, bond coats are used to manage any thermal expansion mismatch between the substrate and deposited material [279]. This particular method finds application in Turbine Blade Coatings (TBCs), due the thermal expansion mismatch between the nickel superalloy and protective ceramic coating.

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Figure 8.4: Microstructure of thermally sprayed coating showing typical defects [15].

8.2.3 Thin-Film Heater Structure & Material Selection

8.2.3.1 Thin-Film Heater Structure

Only the merits, and potential fabrication process, of a resistive thin-film heater have been discussed so far. In this section the specific structure of the heater is developed in detail.

Fig 8.5 depicts the key parts that make up a resistive thin-film heater. The most general structure consists of four elements: 1) an environmental barrier; 2) a resistor; 3) an electrical insulator; 4) a bond coat. In most instances both the insulator and resistor films will be required as a minimum. Certain films are able to perform a dual role. For instance, a resistive layer with high temperature oxidation resistance may not require an environmental coating when operated in air.

Each layer in the heater stack must have properties suited to its particular role. These are as follows:

1. **Insulator:** Ensures electrical isolation of the substrate - high dielectric strength;
2. **Resistor:** Acts as the heat flux source - high electrical resistivity and service temperature;

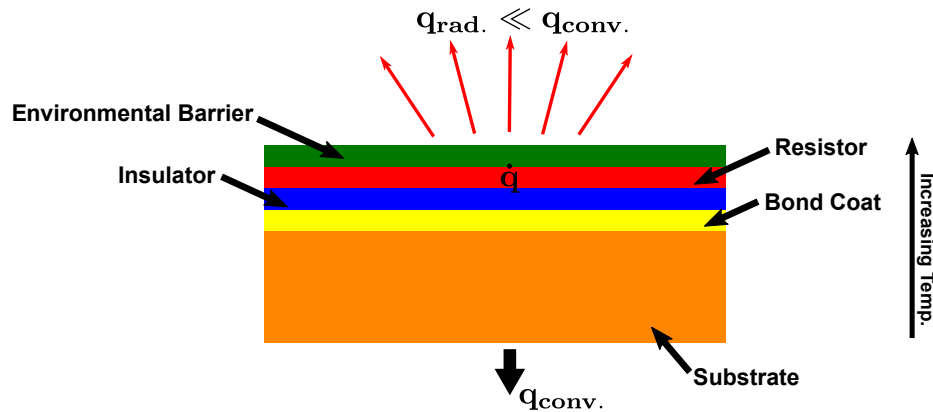


Figure 8.5: The basic elements and construction of a thin-film heater.

3. **Bond Coat:** Minimises thermal stress between the substrate and insulator - low elastic modulus and appropriate thermal expansion coefficient;
4. **Environmental Barrier:** Protects resistor film from environmental damage - generally requires high temperature oxidation resistance.

In addition to their role specific properties, a high thermal conductivity and service temperature is desirable for all of the films to maximise the heat fluxes that can be achieved.

8.2.3.2 Film Material Selection

The previous section established the desirable material properties for each layer in the heater stack. In Table 8.2 the relevant bulk properties of a number of candidate thermal spray materials are given. All of the entries were cross-referenced against databases from a number of thermal spray companies² to ensure that only materials industrially available were considered.

Molybdenum was explored as a resistor material in a thin-film heater concept by [307]. When operated in air, the design eventually failed due to high temperature oxidation of the molybdenum, which was operated without an environmental barrier. The high thermal conductivity of molybdenum is beneficial for reducing temperature gradients in the stack. However, the low electrical resistivity means that small film thicknesses, or long conduction

²Engineered Performance Coatings (EPC) <http://www.ep-coatings.co.uk/>; Bodycote <http://www.bodycote.com>; Orlikon Metco <https://www.oerlikon.com>.

Name	Type	T_m [°C]	E [GPa]	K_{Ic} [MPa m ^{0.5}]	ρ_e [1e ⁻⁴ Ohm cm]	E_m [MV/m]	k_s [W/mK]	α [1e ⁻⁶ K ⁻¹]	Process
<i>Environmental Barrier</i>									
MoSi ₂	Intermetallic	1900-2050	242-329	2.7-5.9	3.5	-	66.2	6.5	APS, VPS
Si	Metalloid	1412	112	-	100	-	105	2.5	APS, VPS
<i>Resistor</i>									
Mo	Metal	2617	330	20-40	0.057	-	138	5.4	APS, VPS
Ni20Cr	Alloy	1400	220	108	1.08	-	13.4	14.0	APS, HVOF, VPS
<i>Insulator</i>									
Al ₂ O ₃	Ceramic	2054	370	4.0	-	26	25.3	5.5	APS, VPS
Y ₂ O ₃	Ceramic	2439	170	1.4	-	300	10	9.1	APS, VPS
Zr ₂ O ₃	Ceramic	2681-2847	225	4.5	-	5	2.3	7-12	APS, VPS
<i>Bond Coat</i>									
NiCrAlY	Alloy	1400	-	-	-	-	20	-	APS, HVOF, VPS

Table 8.2: Candidate materials for thin-film heater construction. Data taken from [174, 226].

lengths, are required to avoid the need of a specialised high current power supply.

In the research by [179] on using thin-film heaters to achieve HHFs, Ni20Cr (nichrome) was used as the resistor layer. This material can be operated in air up to $\sim 1200^{\circ}\text{C}$ [307], and is used extensively in other resistive heating applications. Its increased resistivity (x5), relative to molybdenum, also allows operation at higher voltages. This material was chosen for the resistor layer in the thin-film heater test specimens.

Of the insulator materials in Table 8.2, Zirconium Oxide (Zr_2O_3) is immediately discounted as it has no discernible advantage over the other candidates. Alumina (Al_2O_3) is a widely used technical ceramic, and was one of the first materials to be industrially available via the thermal spray method [280]. Yttrium Oxide (Y_2O_3) is used as a strengthening agent in alloys or, in the electronics industry, as an electrical insulator [278]. From the data in Table 8.2, a layer of Yttrium Oxide would have a higher breakdown voltage, for an equivalent through-thickness temperature gradient, than alumina. However, there is more certainty in the properties that can be obtained when alumina is thermally sprayed and, for this reason, it was selected as the insulator for this initial batch of thin-film heater test specimens.

The heaters were applied to a copper substrate in order to ensure that the results would be valid for the fabricated HPJC-1b modules. To determine whether thermal expansion mismatch between the substrate and insulator would be a source of failure, a bond coat was not used for these initial specimens. The high temperature oxidation resistance of nichrome made the use of an environmental barrier superfluous.

8.2.4 Experimental Test Specimens

In the preceding sections the function of the individual layers in the heater stack have been discussed, and materials selected to fulfil their roles. The purpose of the test specimens was to determine the influence of film geometry and spray method on the heater performance. A total of six stack combinations were considered, all of which were deposited on to a 80

mm x 20 mm grit blasted copper³ surface by Engineered Performance Coatings (EPC)⁴. To avoid overheating of the substrate during operation of the resistive film, water was fed through a $\varnothing 13$ mm coolant channel milled into the substrate. A CAD model of the test specimen geometry, and corresponding fabricated components, are shown in Fig.8.6.

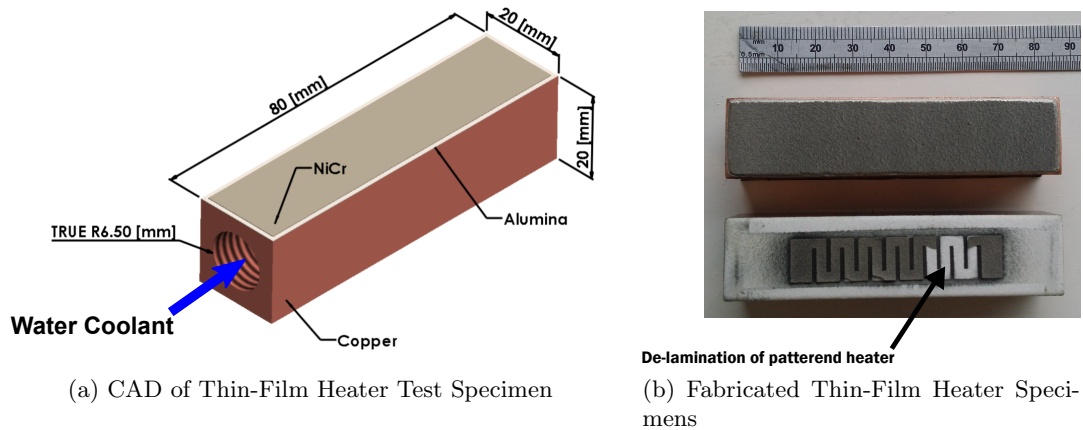


Figure 8.6: (a) A CAD image of the thin-film heater geometry. (b) Fabricated sample thin-film heaters. Numbers 1 & 4 in Table.8.3.

Based on the bulk properties of alumina from Table.8.2, less than a micron of material should be sufficient to prevent breakdown at ~ 10 V. However, in research by [153, 179] they found that values of the order of 100 microns were required when using APS alumina. The author subsequently chose to use values in the range of 100-150 microns in this research.

A thin layer of nichrome is desirable to reduce the thermal gradient through the heater stack, and increases its resistance. In reality, the minimum thickness is limited to 10-20 microns [54] by the thermal spray method. To avoid the potential of significant non-uniformity in the heat generation rate, the author decided to use thickness values in the range of 75-150 microns. Similar values were also used by [179, 307] in their successful application of nichrome in thin-film heaters.

The nichrome film thickness can be maintained, but its resistance increased, by using a

³Grade C101.

⁴<http://www.ep-coatings.co.uk/>

track design - see image (b) of Fig.8.6. This was attempted for one of the sample specimens using a laser cut mask to shield regions of the alumina during the nichrome spraying process. Unfortunately, the film de-laminated as it was sprayed due to the high free edge stresses generated from the mismatch in thermo-mechanical properties with alumina.

A summary of the six fabricated thin-film heater specimens is shown in Table.8.3.

Sample	Method	Nichrome		Method	Alumina	
		Thickness [μm]	Resistivity [$\mu\Omega\text{ cm}$]		Thickness [μm]	R [$\text{M}\Omega$]
1	APS	150	425	APS	150	0.54
2	APS	150	-	APS	100	-
3	APS	100	390	APS	150	0.47
4 (track)	APS	-	-	APS	150	-
5	APS	75	358	APS	150	0.45
6	HVOF	30 (150)	203	APS	150	0.57

Table 8.3: Thermal spray techniques, and material combinations, investigated for the thin-film heater samples.

Sample 2 could not be tested because 100 microns of alumina was found to provide insufficient electrical isolation. When the coating was viewed under a microscope, this appeared to be caused by areas of low alumina deposition. For sample 6, the intention was to have a nichrome layer of 150 microns. However, during the thermal spray process, the coating was found to de-laminate above a thickness of 30 microns.

Resistivity measurements of the nichrome were performed using a 4-point probe. The values displayed in Table.8.3 are the averages from ten measurements taken across each film. Similar room temperature resistivity values for APS and HVOF sprayed nichrome were found by [218].

8.3 The Osney High-Heat Flux Measurement Facility

8.3.1 Overview

An experimental facility for **HHF** measurements was constructed at the **OTFL** based on the thin-film heater method discussed in the previous sections. The purpose of this facility was two-fold: 1) to increase the **TRL** of HPJC-type concepts through experimentation on manufactured modules; 2) to develop the thin-film heater method. A CAD model of the installation, together with the completed structure, is shown in Fig. 8.7

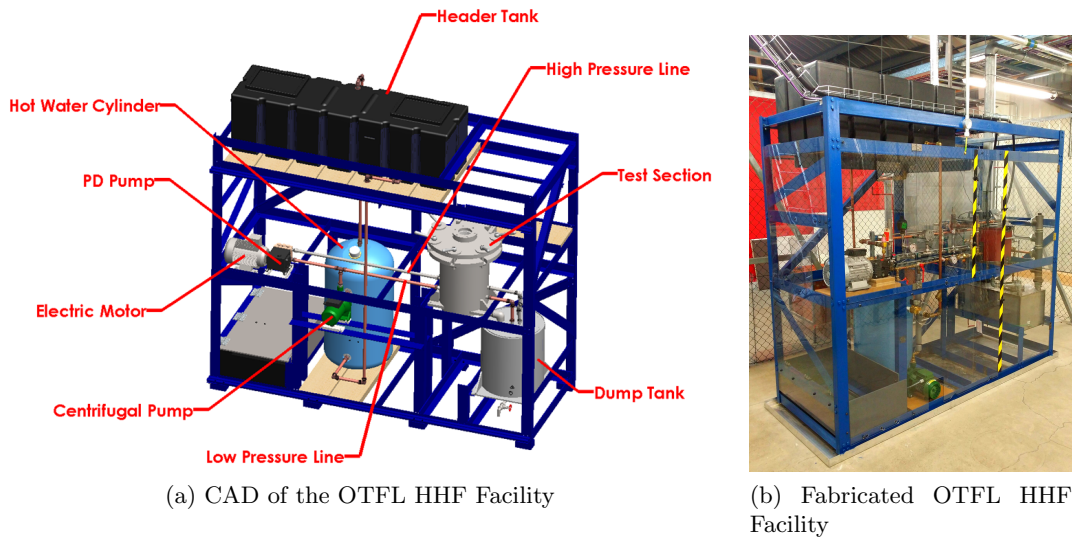


Figure 8.7: (a) A CAD image of the OTFL HHF experimental facility (credit: Peter Rodgers) (b) Completed OTFL HHF experimental facility.

Due to the low mass flow rate requirements of current HPJC-type concepts, a single-pass flow system was used for the **OTFL HHF** facility to simplify its design: the maximum flow rate requirements for the HPJC-1b are estimated to be 3 lpm.

Operation and monitoring of the facility was performed using a **Programmable Logic Controller (PLC)** built by MR Control Ltd.⁵ This system is able to independently shut-down the facility based on sensor data thresholds set by the user.

⁵<http://www.mrcontrol.co.uk/>

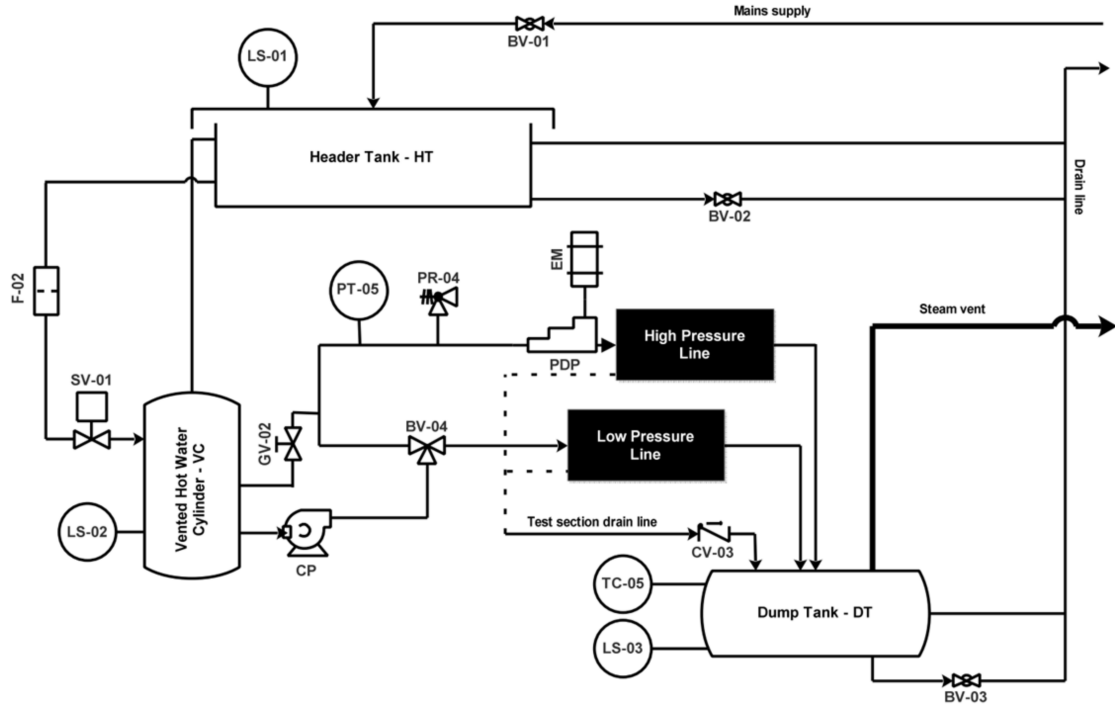
The facility operates by initially feeding water from the mains into a 225 L header tank. A solenoid valve is then used to control the flow from this tank into a 100 L hot water cylinder. When this valve is closed, a 3 kW immersion heater can be used to increase the coolant inlet temperature to the test section up to $\sim 70^{\circ}\text{C}$. However, this reduces the maximum continuous experimental run time to approximately 30 min; based on the mass flow rate requirements of the HPJC-1b.

There are two modes of operation for the OTFL HHF facility, which the author has chosen to call A and B. In mode A, fluid from the hot water cylinder is centrifugally pumped, at flow rates of up to 130 lpm, into the test section through $\varnothing 15$ mm OD x 0.8 mm copper piping. This mode was used for development of the thin-film heater concept, and is intended for high flow rate, low pressure, operation. In mode B, the centrifugal pump is used to supply fluid at sufficient pressure to avoid cavitation to a second, Positive Displacement (PD), pump. This pump feeds coolant into the test section through $\varnothing 8$ mm OD x 1 mm 316L stainless steel piping. It is able to provide coolant pressures of up to 20 MPa, at a maximum flow rate of 7 lpm. Mode B was used to perform thermo-fluid measurements on the HPJC-1b.

To avoid overheating of the centrifugal pump when operating in mode B, a recycle valve was placed between the two pumps to divert flow back into the hot water cylinder. An additional benefit of this system is the reduced stratification of the temperature distribution within the hot water cylinder. A flow diagram for the facility is shown in Fig. 8.8

The flow rate supplied by the PD pump was adjusted by varying the speed of a 3 kW electric motor, which itself was controlled using the PLC. To dampen oscillations in the flow rate, inherent with this type of pump, a 12 MPa, 0.16 L, nitrogen accumulator was installed. To adjust the coolant pressure in the mode B line, plates with differing orifice sizes were installed downstream of the test section.

After passing through the test section, the lines for both operational modes discharge into



Code	Description	Range	MWP [MPa]	MWT [°C]
BV-01/02	Ball Valve	n/a	4	170
BV-03	Ball Valve	n/a	1.6	120
BV-04	Ball Valve 3-way	n/a	2.5	120
LS-01	Level Sensor - Ultrasonic	65-600 mm	n/a	70
LS-02/03	Level Sensor - Capacitive	Digital	1	70
PT-05	Pressure Transducer	0-2 bar	0.2	85
GV-02	Gate Valve	n/a	1.6	120
SV-01	Solenoid Valve	2-230 psid (NO)	1.6	95
F-02	Filter	400 micron (pores)	1.6	110
CV-03	Check Valve	n/a	2.1	220
PR-04	Pressure Relief Valve 0.5"	10 barg (set)	n/a	195
TC-05	Thermocouple	-200 - 1250C	n/a	1250
CP	Centrifugal Pump	0-130 lpm	n/a	90
PDP	Positive Displacement Pump	0-7 lpm	20	60
EM	Electric Motor	0-1450 rpm	n/a	n/a

Figure 8.8: Process flow diagram for the OTFL HHF measurement facility. Table of components also shown with Maximum Working Pressures (MWP) and Maximum Working Temperatures (MWT).

a 100 L stainless steel dump tank. This drains manually, or via an overflow, to the outside of the building. As flashing across the orifice plate may produce significant quantities of steam during mode B operation, a separate vent line was also installed in the top of the tank.

The water level in the header tank was continuously monitored using an ultrasonic level sensor. Additionally, a capacitive level switch was used to ensure that there was a minimum quantity of water in the dump tank to act as a quench when operating in mode B.

8.3.2 Test Section & Instrumentation

The test section is comprised of a 30.5 cm ID x 33 cm mild steel cylinder with a 6.4 mm wall thickness. This is bolted to a base-plate during testing, but can be easily removed to gain access to the test specimen. A 33 cm ID x 5.1 cm steel cylinder was welded to the base-plate to act as a trough should leakages occur. In addition, a $\varnothing 1$ in drain line, that connected to the dump tank, was also installed in the base-plate.

Two, 5.1 cm x 21.6 cm, slots were machined into the cylindrical test section to allow coolant piping access. A removable cover on the top of the cylinder provided both optical access, through a 10 cm diameter port, and two slots for DC power cabling. The cover itself was secured via three M10 x 30 mm bolts during testing. A schematic of the test section layout, and associated images, are shown in Fig. 8.9.

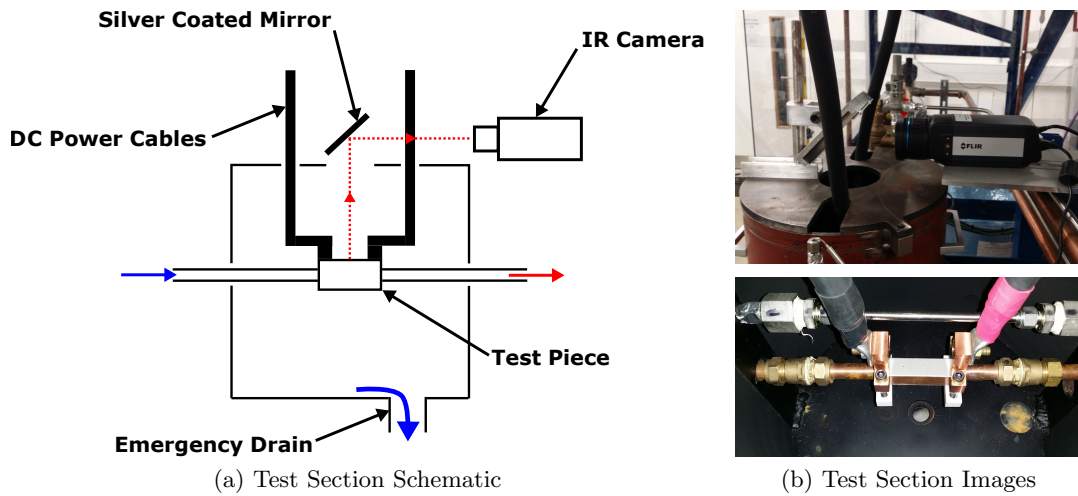


Figure 8.9: (a) Schematic of experimental set-up within test vessel. (b) Images of IR camera set-up and thin-film heater test piece.

Temperatures on the resistive thin-film surface were recorded using a FLIR⁶ A655sc IR camera. This camera has a resolution of 640 x 480, and the following accuracies: $\pm 2^{\circ}\text{C}$ when the temperature is less than 100°C ; $\pm 2\%$ of the reading when the temperature is above 100°C . To minimise the risk of damaging the camera should a component fail, a

⁶<http://www.flir.co.uk/>

mirror system was employed - see image (a) of Fig.8.9. This consisted of a 10 cm x 10 cm silver coated mirror, which was angled at 45° to the horizontal over the optical access port. The camera itself was then set ~15 cm from the mirror on the test section cover. Over the typical operational temperature range of nichrome (100-1000°C), Wien's displacement law predicts a peak wavelength range of ~2-7 μm [120]. In this wavelength range the reflectivity of the silver coated mirror is quoted by the manufacturer to be above 98% [1].

The base-plate and internal surfaces of the test section were also sprayed with matt black paint. This was done in order to minimise the impact of reflected radiation on the temperature recorded by the IR camera.

DC power was supplied to the resistive films using a Magna-Power Electronics Inc.⁷ 15 kW system, which is capable of delivering a maximum current of 375 A at peak power. Copper electrodes were secured to the heater using two M6 x 60 mm bolts, through a ceramic backing plate. These bolts were tightened to 15 Nm using a calibrated torque wrench, accurate to $\pm 4\%$. To minimise the contact resistance between the film and electrodes, silver paint was employed. A cross-sectioned CAD image of the electrode attachment assembly is shown in Fig.8.10.

Coolant temperatures and pressures were recorded in both operational mode lines before, and after, the test section. Flow rates were also recorded in both operational mode lines using turbine meters prior to the test section. All of the sensors gave output signals in the range of 4-20 mA, which were recorded by the PLC at a frequency of 1 Hz. Table.8.4 summarises the sensors used, together with their associated ranges and accuracies.

⁷<http://www.fmagna-power.com/>

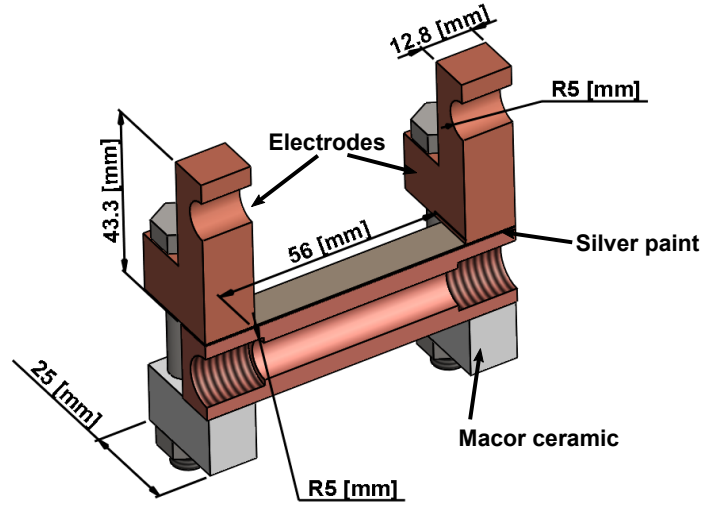


Figure 8.10: Cross-sectioned CAD image showing how electrodes are attached to the thin-film heater test specimen.

Code	Description	Mode Line	Range	Accuracy
PX309	Pressure Transducer	B	0-3000 psig	$\pm 0.25\%$
PX182B	Pressure Transducer	A	0-60 psig	$\pm 0.3\%$
T-type	Thermocouple	B	$-40-350^{\circ}\text{C}$	$\pm 0.5^{\circ}\text{C}$
Pt100	Resistance Thermometer	A	$-75-250^{\circ}\text{C}$	$\pm 0.03^{\circ}\text{C}$
FTB-1311	Turbine Meter	B	0.3-1.5 lpm	$\pm 1\%$
FTB372-R	Turbine Meter	A	4-160 lpm	$\pm 5\%$

Table 8.4: Instrumentation used to measure coolant properties in lines for both mode A & B operation.

8.4 Experimental Measurements on Sample Thin-Film Heaters

8.4.1 IR Camera Methodology

The IR camera was calibrated by the manufacturer against a black-body for three separate temperature ranges: $-40-150^{\circ}\text{C}$; $100-650^{\circ}\text{C}$; $300-2000^{\circ}\text{C}$. To associate these results to the resistive thin-film temperatures it is first assumed that the object and surroundings can be treated as diffuse gray bodies. The total radiative power incident on the camera sensor can then be expressed as [89]

$$\dot{Q}_{\text{tot}} = \epsilon_o \tau_{\text{atm}} \dot{Q}_o + (1 - \epsilon_o) \tau_{\text{atm}} \dot{Q}_e + (1 - \tau_{\text{atm}}) \dot{Q}_{\text{atm}}. \quad (8.1)$$

Here, the subscripts have the following associations: o - object; e - enclosure; atm - atmosphere. The symbols ϵ and τ represent emissivity and transmittance, respectively. Equation (8.1) neglects the radiative properties of the mirror and surroundings. However, the influence of these terms can be encapsulated in the object emissivity if it is determined whilst the component is in the test section.

The values of τ_{atm} and $\dot{Q}_{\text{atm}}$ are calculated internally by the FLIR Research IR software from the following user inputs: 1) camera-to-target separation; 2) atmospheric humidity; 3) atmospheric temperature. Number 1) was kept constant at 30 cm throughout the test campaign.

The value of $\dot{Q}_e$ is also calculated by the FLIR Research IR software based on a user input for the reflected temperature. To determine this, a crinkled piece of aluminium foil was placed over each thin-film heater sample, prior to testing, and the area averaged temperature recorded assuming an emissivity of one. This approach assumes that the surroundings will remain isothermal during any test run, which is valid for this research due to the high thermal mass of the test section.

A standard approach in IR thermography is to coat the surface of interest in a high emissivity material to enhance the ratio of emitted-to-reflected radiation from the object. However, as the failure temperature of the nichrome thin-film could be above 1200°C, this approach could not be taken by the author in this work due to the lack of a suitable coating material.

Bulk, oxidised, nichrome has an emissivity in the range of 0.87-0.9 [180]. Additionally, thermally sprayed Cr_2O_3 has been found to have an emissivity of ~ 0.8 [209]. The passive chromium oxide (Cr_2O_3) layer that protects nichrome from further oxidation normally forms during the first heating cycle above a temperature of $\sim 700^\circ\text{C}$ [295]. The calculated emissivity for each test specimen will therefore provide an indication as to whether the surface has been oxidised during the thermal spray process.

In calculating the emissivity of the nichrome thin-film it was assumed that the value would be temperature dependent, but wavelength independent. A single, k-type, thermocouple was attached to the surface of each sample using an electrically insulating, but thermally conductive, cermet paste. By running at a fixed coolant flow rate, and varying the input current to the film, the emissivity was calculated at discrete, steady-state, temperature values by matching the IR camera and thermocouple outputs. The results of this calibration procedure for each test sample are shown in Fig. 8.11

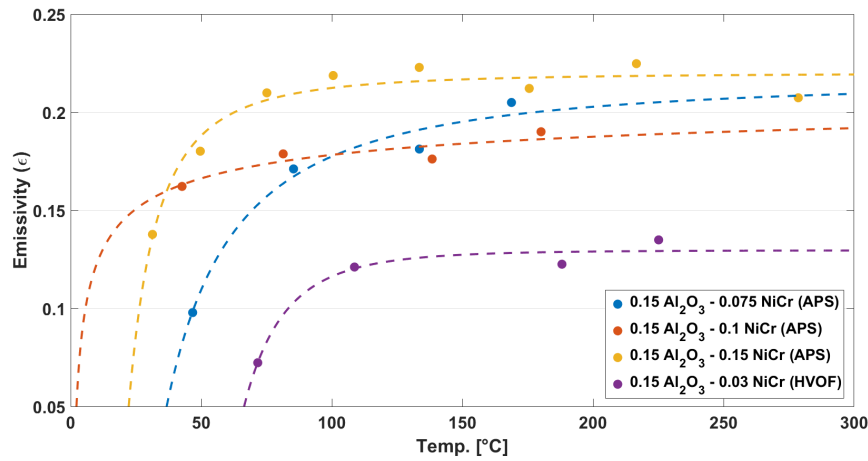


Figure 8.11: Calculated temperature dependent emissivity of thermally sprayed nichrome thin-films.

It appears from Fig. 8.11 that oxidation of the surface has not occurred during the thermal spray process. However, even un-oxidised bulk nichrome has an emissivity of approximately 0.6 [63]. Interestingly, bulk nickel in both oxidised, and un-oxidised, forms can have an emissivity in the range of 0.1-0.2 [63]. The lower emissivity of the nichrome film deposited by HVOF indicates that the thermal spray method also has an influence.

To avoid potentially damaging the thin-film heater, the emissivity of the nichrome was only measured up to a maximum temperature of 200-300°C. The minimal variation in emissivity above a temperature of ~100°C, for all of the samples, indicates that this range is sufficient.

As the FLIR Research IR software only allows the user to input a fixed emissivity, an algorithm was written in MATLAB 2106a to determine the true surface temperature distribution based on IR camera data taken with a fixed emissivity of one, and using the lines of best fit in Fig. 8.11. The dependence of the emissivity on the surface temperature meant that an iterative approach was required, with the actual temperature, T_a , given by the equation

$$T_a = \left[\frac{1}{\epsilon_o(T_a)} (T_m^4 - (1 - \epsilon_o(T_a))T_e^4) \right]^{\frac{1}{4}}. \quad (8.2)$$

Here the subscript m represents a measured quantity. If the actual temperature went above the range of the lines of best fit from Fig. 8.11, the value of the emissivity at 300°C was used.

Time synchronisation between the IR camera and thermocouple data was achieved by putting a small tungsten filament bulb in the field-of-view of the camera, which was triggered at the same instant the thermocouple began recording. IR camera data was captured at a frame rate of 6.25 Hz.

8.4.2 Uncertainty Analysis

To estimate the uncertainty in the area averaged heat flux produced by the thin-film heater, the same approach as in Sec. 3.5.2 of Chpt. 3 was used. The area averaged heat flux was calculated using the enthalpy change of the coolant, and the heated area of the thin-film between the electrodes. A summary of the results for test sample 1 are shown in Table. 8.5

The measurements uncertainty in the flow rate was based on separate uncertainties in the timing device (0.1 sec) and weight scales (0.01 kg) used to calibrate the turbine meter. This process involved running coolant through the system for three minutes into a vessel. By measuring the mass of coolant in the vessel after this time period it was then possible to compare the average measured flow rate against that from the turbine meter. For the parameters $T_{ce} - T_{ci}$, L and W , measurement uncertainties were extracted from supplier

Parameter	Typical Value	Measurement Uncertainty	$\frac{\partial q'}{\partial x}$	$\frac{\partial q'}{\partial x} \frac{\Delta x}{q'}$
$\dot{V}$	18 lpm	0.9 lpm	0.12	0.099
$T_{ce} - T_{ci}$	1.6°C	0.03°C	1.3	0.037
L	54 mm	0.01 mm	-39	-0.00019
W	18 mm	0.01 mm	-120	-0.00056
RSS Uncertainty [%]				11

Table 8.5: Uncertainty analysis for area averaged heat flux of thin-film heater sample 1. Reference area averaged heat flux $\sim 2.1 \text{ MW/m}^2$.

supplied calibration certificates.

From the results in Table 8.5 it is evident that the total uncertainty in the area averaged heat flux is driven by the uncertainty in the coolant flow rate measurement.

8.4.3 Results

During each experimental run the centrifugal pump was first activated, and the flow rate adjusted to the desired value. The system was then run in this state for a further 30 sec, before power was applied to the thin-film. To avoid thermal shock failure of the thin-film structure, the current was manually ramped up over a 30 sec period to the desired value. This time scale was based on achieving a Fourier number $\gg 1$, using the properties of the substrate. The system was then run for a further 60 sec under these conditions before shutdown. All of the data presented in this section was time-averaged over the final 30 sec of a run, unless otherwise stated. Subsequent experimental runs were carried out at increasing values of power input until the heater failed.

8.4.4 Temperature & Heat Flux Data

In Table 8.6, summary data is presented on the performance of the four sample thin-film heaters that underwent testing in the OTFL HHF facility. The highest area averaged heat flux was achieved by sample 1. All of the samples managed to produce maximum area averaged heat fluxes in the MW/m^2 range before failure.

Sample	Max. Heat Flux [MW/m ²]	Max. Temp. at Max. Heat Flux [°C]
1	4.1	543
3	2.4	273
5	1.7	389
6	1.5	464

Table 8.6: Maximum area averaged heat fluxes, and corresponding temperatures, for sample thin-film heaters.

An interesting feature of these results is that, for samples in which the nichrome was applied via [APS](#), the maximum area averaged heat flux achieved increased with nichrome film thickness. This is somewhat counter-intuitive, as reducing the nichrome thickness also reduces its operating temperature, which might be expected to increase the heater life. Additionally, thinner nichrome films should have reduced residual stresses and better adhesive strength [\[17\]](#). The author proposes that there are two potential mechanisms causing these results: 1) as the nichrome film thickness is reduced towards the value produced from a single pass of the thermal spray gun, the percentage variation in its thickness is likely to increase. This can cause local hot spots to form, leading to premature failure of the film; 2) during the thermal spray process micro-cracks will form in the nichrome film. For a thicker film it will require higher thermal stresses for this crack to fully propagate through the material. Mechanism 1) is somewhat validated by the results in Table [8.6](#), in that the maximum temperature of the sample 5 heater is above that of sample 2 at a lower heat flux. Given that sample 5 has a smaller nichrome film thickness, the inverse should apply.

Though sample 6 has the lowest area averaged heat flux, it was achieved using a [HVOF](#) nichrome film thickness of only 30 microns. If the same pattern applies as with the [APS](#) films, it implies that a 150 micron [HVOF](#) nichrome film would be able to achieve a maximum heat flux of ~ 11.7 MW/m². However, this would be conditional on developing a method to avoid de-lamination of the film during the thermal spray process.

In image (a) of Fig. [8.12](#) the surface temperature distribution at various area averaged heat fluxes is shown for sample 1.

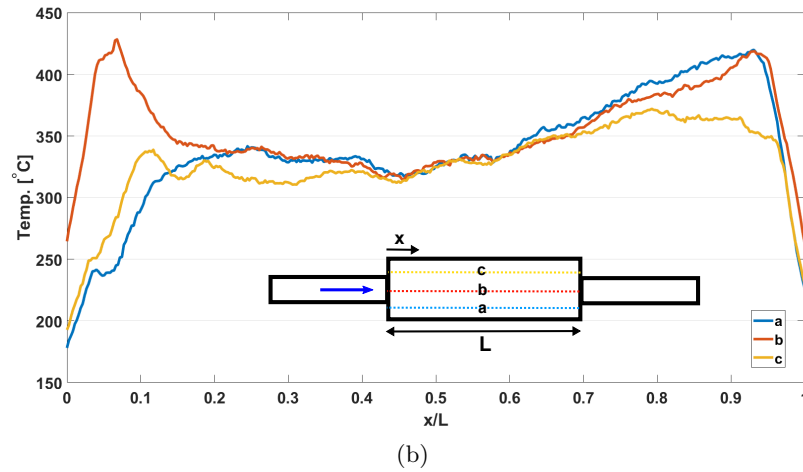
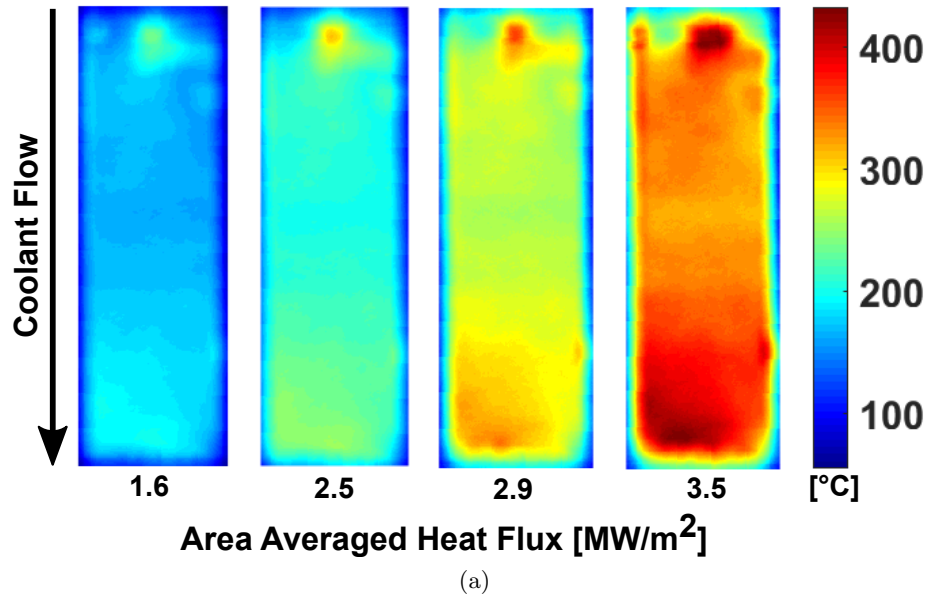


Figure 8.12: (a) Variation in sample 1 resistive thin-film surface temperature distribution for different area averaged heat fluxes. (b) Local variation in thin-film surface temperature along three lines across the specimen width. The results are for the 3.5 MW/m^2 area averaged heat flux case.

It can be seen that the temperature profiles away from the electrodes show only a small variation with distance along the length of the sample for all of the heat flux values. Near the electrodes, temperature non-uniformities occur due to variations in the electrical contact resistance, which leads to subsequent non-uniformities in the current density across the width of the film. This is highlighted in image (b) of Fig. 8.12, where the temperature along

the length of the film has been plotted for three different locations along the sample width - area averaged heat flux of 3.5 MW/m^2 . When $0.2 < x/L < 0.8$ there is minimal difference in the temperature for each line, which indicates uniformity in the current density across the film width. Outside of this region the electrical contact resistance has a significant influence on the current density across the film width, leading to the observed peaking in the temperature profile. This temperature peaking in certain regions becomes less pronounced with increasing heat flux. Ensuring uniform electrical contact resistance to eliminate hot spots will be critical to maximising the heat fluxes that can be achieved with the thin-film method.

Absolute values of the local heat flux on the resistive thin-film surface can not be determined due to a lack of material data. However, via the steady-state 1D heat equation with heat flux and convective boundary conditions, it can be shown that $q' \propto T_s - T_{ci}$. If it is assumed that the thermal resistance is independent of temperature, this relation can be used to determine the normalised heat flux along the sample centreline for varying currents. These results are shown for sample 1 in Fig. 8.13. It was assumed that T_{ci} varied linearly along the length of the pipe.

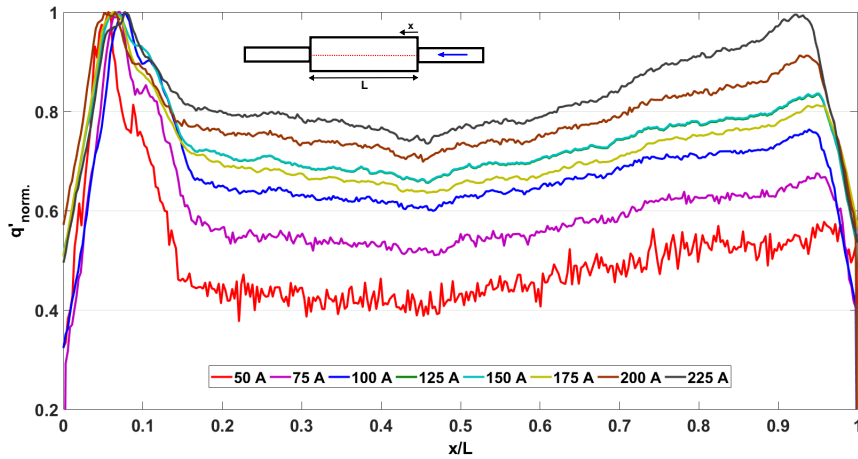


Figure 8.13: Local normalised heat flux distribution for sample 1 at varying power input levels.

The local heat flux is seen to peak near the electrode located next to the coolant inlet, due to the non-uniform distribution of the current in this region. As could be seen from the

temperature distributions in Fig. 8.12 the variation in heat flux across the sample becomes less pronounced as the power input to the resistive film is increased. At 225 A, the maximum variation in the heat flux across the centreline is $\sim 25\%$. Interestingly, the heat flux near the coolant exit shows a steady increase with increasing power input to the film. This is indicative of an increase in the conductive thermal resistance in this region.

If the properties of the films were truly temperature independent, the curves in Fig. 8.13 would be expected to collapse onto a single curve. The fact that this does not occur indicates that the film properties have a strong temperature dependence. Determining the exact nature of this relationship is left for future research.

All of the sample heaters that underwent testing failed via de-lamination of the nichrome layer from the alumina. There was no evidence in any of the samples of de-lamination of the alumina from the copper substrate.

Once the nichrome has de-laminated from the alumina it rapidly heats to its melting point in ~ 0.3 sec, before subsequently failing. An example of the hot-spot developing following de-lamination is shown for sample 1 in Fig. 8.14.

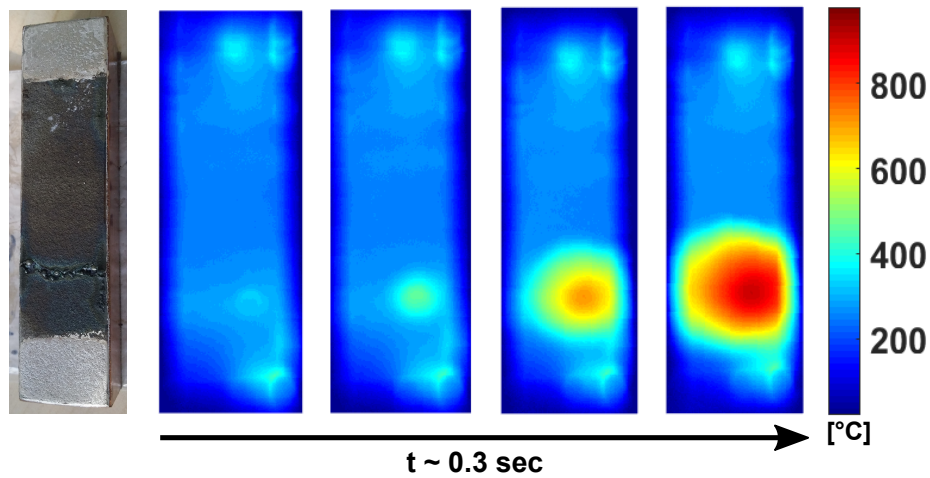


Figure 8.14: Failure of sample 1 thin-film heater caused by de-lamination of the resistive nichrome layer. Sequence of images show development of hot spot prior to failure.

These results indicate that minimising the thermal stresses, caused by thermal expansion mismatch between the insulator and resistive films, is important to achieving high heat fluxes via the thin-film method.

8.5 Summary

An experimental facility has been constructed at the **OTFL** to replicate the thermal-mechanical boundary conditions a HPJC-type divertor target component would see during operation in a **DEMO**. A detailed discussion was presented on the potential methods of applying the heat flux boundary condition, with resistive thin-films chosen as the preferred option. The constructed facility is capable of supplying coolant pressures of up to 20 MPa, at flow rates of 7 lpm, and inlet temperatures up to 70°C. A number of trial thin-film heaters were fabricated and tested in the facility. The results from this experimental campaign were as follows:

- To ensure electrical isolation of the substrate from the resistive thin-film a thickness of at least 100 microns of alumina is required. All successfully operated films had a thickness of 150 microns;
- The application of the resistive nichrome film via **HVOF** was only able to achieve a maximum thickness of 30 microns. Attempts to increase this thickness beyond this value resulted in de-lamination;
- A thicker nichrome resistive film was found to achieve higher area averaged heat fluxes before failure;
- The maximum area averaged heat flux achieved was 4.1 MW/m². This was for a thin-film structure consisting of 150 microns of alumina, and 150 microns of nichrome;
- Hot spots were principally driven by non-uniformities in the current distribution near the electrodes;

- The failure mode for all of the test samples was de-lamination of the nichrome film from the alumina.

Chapter 9

Initial High-Heat Flux

Measurements of the HPJC-1b

“When anybody contradicted Einstein he thought it over, and if he was found wrong he was delighted, because he felt that he had escaped from an error, and that now he knew better than before.”

— Otto Robert Frisch

In this chapter initial thermo-fluid measurements are presented for the manufactured HPJC-1b modules discussed in Chpt.7. The methodology for estimating the wall overheat from surface temperature data is discussed in detail.

9.1 Experimental Methodology

Thermo-fluid measurements of the fabricated HPJC-1b modules were carried out using the Osney Thermo-Fluids Laboratory (OTFL) High-Heat Flux (HHF) experimental facility, described in Chpt.8. To accurately replicate the design pressure stresses, and ensure single-phase flow, the modules were installed in the mode B test line. A special manifold structure was constructed to facilitate the concentric feed system of the modules. An image of a HPJC-1b module installed in the test section is shown in image (a) of Fig.9.1.

The heat flux boundary condition on the tested modules was replicated using the thin-

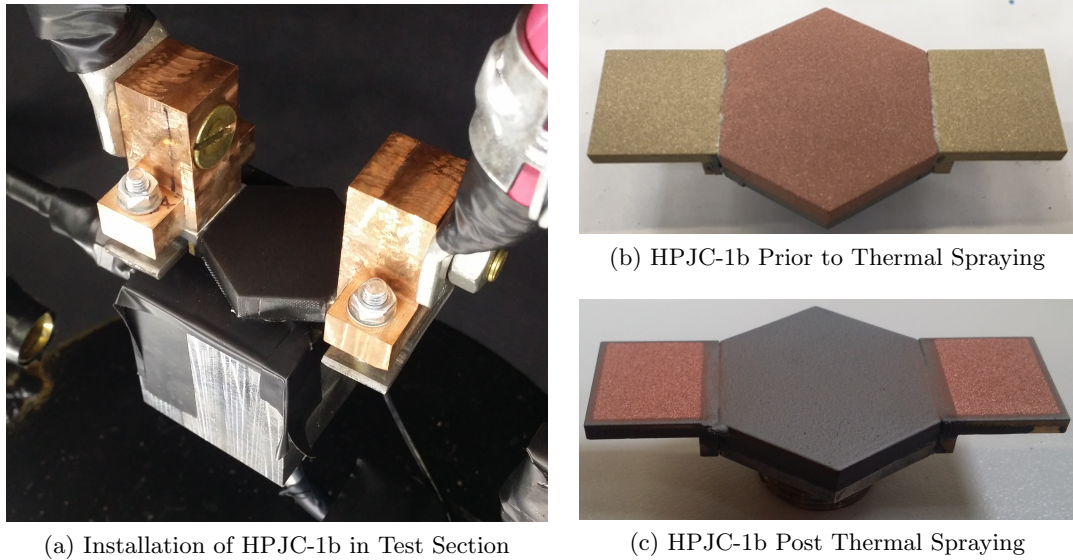


Figure 9.1: (a) HPJC-1b module installed in test section of the OTFL HHF experimental facility. (b) HPJC-1b with attached brass tabs after grit blasting. (c) HPJC-1b after thermal spraying of thin-film heater.

film heater method discussed in Chpt. 8. To avoid attaching the electrodes directly to the hexagonal surface, brass tabs were soldered onto two sides of each module - see image (b) of Fig. 9.1. Engineered Performance Coatings (EPC) then grit blasted the whole, extended, surface, before applying the thin-films.

Six of the fabricated HPJC-1b modules were considered for testing: three with triangular (1, 3, 5), and three with non-triangular (2, 4, 6), cavity designs. This selection was based on the fabricated modules having a helium leak rate $<10\text{e-}6$ mbar l/s¹.

Further development of the thin-film heater method led to the selection of two stack structures for the HPJC-1b experiments, which are summarised in Table 9.1. This approach was taken as there was no prior knowledge on how the thin-film heater technology would work with the HPJC-1b geometry. By using two different thin-film structures the author hoped to maximise the possibility of obtaining results.

¹Criteria used by VPE Inc. for water-cooled heat exchangers.

Triangular (1,3,5)				Non-Triangular (2,4,6)		
Layer	Material	Method	Thickness [μm]	Material	Method	Thickness [μm]
Bond Coat	MCrAlY	HVOF	50	MCrAlY	HVOF	50
Insulator	Al_2O_3	APS	150	Al_2O_3	APS	150
Compliance	$\text{Al}_2\text{O}_3/\text{MCrAlY}$	APS	50	-	-	-
Resistor	MCrAlY	HVOF	100	Mo	APS	150
Environmental	-	-	-	MoSi ₂	APS	50

Table 9.1: Summary of thin-film heater structures used to perform thermo-fluid measurements on fabricated HPJC-1b modules.

The resistive layer in each stack structure was comprised of discrete sub-layers, each 50 microns in thickness. As the total resistive layer thickness was increased, the planar dimensions of each sub-layer were reduced. This helped to alleviate the build-up of residual stresses, allowing thicker resistive layers to be sprayed.

A 50 micron thick layer of Air Plasma Spray (APS) copper was applied to the resistive/environmental coating in the brass tab regions of each test sample. This was done in order to reduce current density non-uniformity, caused by poor electrical contact between the electrode and thin-films. A typical test sample, showing the APS copper regions, can be seen in image (c) of Fig 9.1.

9.1.0.1 Thin-Film Heater Surface Emissivity

The experimental results from Chpt 8 indicated that failure of the thin-film heater stack is likely to occur at temperatures considerably below the maximum operating temperature of the resistor material, due to thermal stresses. The author therefore decided to use a high emissivity, matt black, spray paint by VITCAS² to increase the ratio of emitted-to-reflected power from the test specimen.

Determining the dependency of emissivity on temperature for any coating applied to the thin-film surface, using the thermocouple method, presents a number of challenges: 1) finding a high temperature adhesive that is electrically insulating, but thermally conducting,

²This paint is heat resistant up to temperatures of 800°C- <https://www.vitcas.com>.

to attach the thermocouple to the surface; 2) correctly locating the thermocouple so that the emissivity is obtained for the expected operational temperature range of the film; 3) performing the measurement as part of the test procedure so that, if a heater fails, experimental data is still obtained.

To avoid the challenges of measuring the emissivity temperature dependence during testing, a separate procedure was developed by the author. This involved applying the paint to one side of a steel substrate, and then heating the other with an oxy-acetylene torch. A thermocouple was located on the painted side, directly along the flame centreline. The sample was rotated into the flame, and heated to temperatures of $\sim 800^{\circ}\text{C}$. It was then rotated out of the flame to a secondary station, where the surface temperature on the painted side was recorded during cooling using an **Infra-Red (IR)** camera. Thermocouple data was recorded at a frequency of 20 Hz, whilst the IR camera capture rate was set to 12.5 fps. To synchronise the two data sets, the voltage across an optical switch was recorded on the same **Data Acquisition System (DAQ)** as the thermocouple. When the sample was rotated into the flame there was a voltage rise, which triggered the IR camera.

An emissivity measurement was performed for each black-body calibrated temperature range of the IR camera: -40 - 150°C ; 100 - 650°C ; 300 - 2000°C . The results of this procedure are plotted in Fig. 9.2, together with a polynomial best-fit curve across the whole temperature range.

The reflected temperature prior to each test run was calculated using the aluminium foil method discussed in Sec. 8.4.1 of Chpt. 8. A sensitivity study performed by the author using the FLIR Research IR software showed that the calculated emissivity changed by less than 1% for a $\pm 5^{\circ}\text{C}$ change in the ambient temperature, or a $\pm 10\%$ change in the humidity. By matching the experimental camera-to-object separation distance, the author was able to assume that the influence of varying transmittance on the results could be neglected.

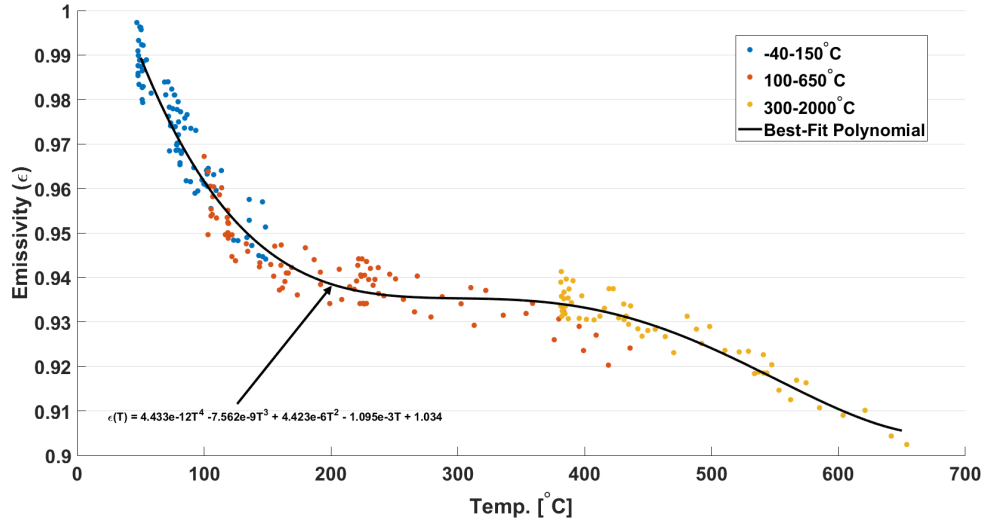


Figure 9.2: Temperature dependent emissivity of VITCAS heat resistant spray paint across black-body calibrated ranges of IR camera.

9.1.1 A Numerical Model to Determine Heat Transfer Performance

A steady-state Finite Element Analysis (FEA) model of the HPJC-1b, including the thin-film heater, was developed in ANSYS Workbench 16.2 to determine the heat transfer performance. The current densities at the interface between the brass tabs and the module were assumed to be uniform, allowing the former to be neglected from the geometry.

The material properties for the HPJC-1b were taken from the Structural Design Criteria for ITER In-vessel Components (SDC-IC) App. A, and assumed to be temperature dependent. For the heater stacks, all of the layers were modelled as homogeneous and isotropic. This was a simplification, but the influence of anisotropy should be minimal given the difference in scale between the thickness and planar dimensions of the films. Thermal contact resistance between all of the layers was also ignored. Given the high velocity impact of the splats during the thermal spray process, this is regarded as a realistic initial approximation.

All of the external surfaces of the module and films were set as adiabatic. To determine whether radiation heat transfer needed to be included, three assumptions were made: 1) the thin-film heater and test section enclosure behaved as diffuse gray bodies; 2) the large thermal mass and surface area of the test section enclosure meant that it could be treated

as isothermal; 3) the view factor between the thin-film heater and test-section was one. Under such conditions it can be shown that the net radiative heat transfer from the thin-film heater to the test section is given by [120]

$$\dot{Q}_{1-2} = \sigma A_1 (T_1^4 - T_2^4) \left[\frac{1}{\epsilon_1} + \left(\frac{1 - \epsilon_2}{\epsilon_2} \right) \frac{A_1}{A_2} \right]^{-1}, \quad (9.1)$$

Here, the subscript 1 is used to represent thin-film heater properties, and 2, those of the test section enclosure.

The internal surfaces of the test section enclosure were sprayed with matt black paint, which was taken to have an emissivity of 0.9 [180]. As the area ratio in equation (9.1) is ~ 300 , even with $\epsilon_1 = 1$ and $T_1 = 500^\circ\text{C}$, the net radiative heat transfer is only ~ 19 W; test section enclosure assumed to be at a temperature of 20°C . To produce a 1 MW/m^2 heat flux over the 10 cm^2 surface area of the HPJC-1b would require 1 kW. It is therefore evident that neglecting radiation will not have a significant impact on the accuracy of the model results. A similar calculation for natural convection showed that it too could be neglected.

To generate the HPJC-1b model mesh, the same approach was used as in Sec. 5.1.1.1 of Chpt. 5. For the thin-film bodies, a hex dominant method was applied to allow splitting in the thickness direction. The maximum element thickness for any thin-film was set as 75 microns. In Fig. 9.3, a typical mesh is shown through a cross-sectional between hexagonal flats, together with a magnified view of the thin-film heater region.

A normalised heat transfer coefficient distribution was imported from a conjugate CFD simulation performed on the HPJC-1b in ANSYS CFX 16.2 - see Chpt. 6 - and applied to the internal surfaces of the FEA model. Additional simulations showed that the form of this normalised distribution was largely invariant to the magnitude of the Reynolds number.

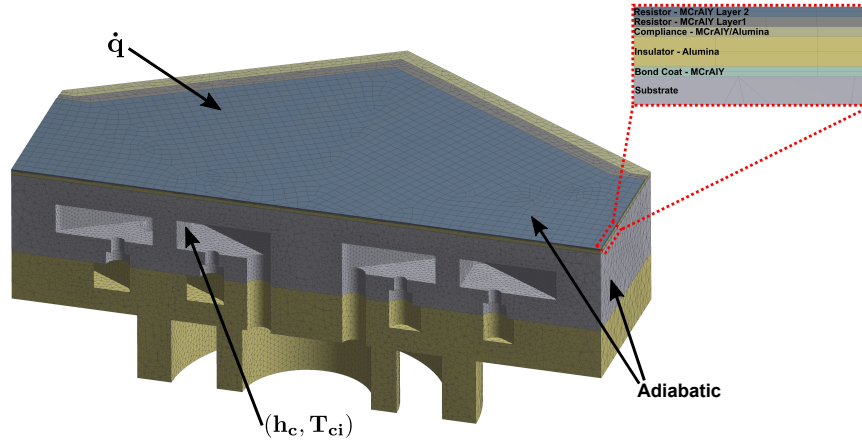


Figure 9.3: Cross-Section of numerical model showing mesh and thermal boundary conditions. A magnified image of thin-film region also shows the elements through the thin-film heater thickness.

9.1.1.1 Internal Heat Generation

The Joule heating in the resistive thin-films was modelled as a volume heating term. By assuming each 50 micron sub-layer of a resistive film had the same electrical resistivity, the ratio of power input into each could be determined by geometric quantities alone.

The local normalised distribution of the power in each sub-layer was calculated by solving the Laplace equation for the voltage distribution on a sheet with equivalent planar geometry. In Fig 9.4 a normalised profile obtained using this method is shown for a typical sub-layer, together with the boundary conditions used. It was assumed that this local power deposition did not change through the sub-layer thickness.

The total power input to a resistive film was based on the enthalpy change of the coolant. Current readings were taken from the DC power supply, and its division between resistive sub-layers based on the design dimensions of the films.

9.1.1.2 Properties of Thermally Sprayed Films

To perform steady-state modelling of the thin-film heater, thermal conductivity data is required. There is a significant spread in these values in the literature [32, 37, 80, 153, 179],

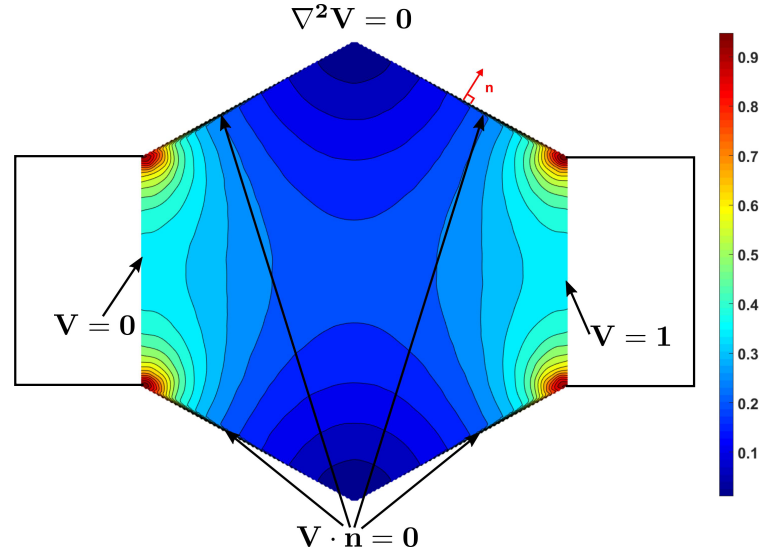


Figure 9.4: Normalised power deposition profile in a resistive thin-film sub-layer. Determined by solving the Laplace equation for voltage on a sheet with equivalent planar dimensions.

due to the large number of variables that influence the thermal spray process. To minimise the variability in this research the author took two steps: 1) all of the thermal spraying was performed by [EPC](#); 2) a number of coupons were prepared by [EPC](#) for each thermally sprayed material on steel substrates. These coupons were subsequently cross-sectioned, and analysed using a [Scanning Electron Microscope \(SEM\)](#), by the [Oxford Materials Characterisation Service \(OMCS\)](#)³. Thickness values were recorded at a number of cross-sections for each coupon, and compared to the target value. This allowed the film thickness accuracy to be determined. An example cross-section for [APS](#) alumina is shown in Fig. 9.5.

Area porosity measurements were also taken from the cross-sections using the ImageJ application. By taking an average of these porosity measurements from a number of cross-sections, an estimate of volume porosity was obtained. There are a number of formulae in the thermal spray literature that use this porosity value to estimate the thermal conductivity of the material perpendicular to the substrate surface [\[165, 242\]](#). The author chose to use the formula of [\[175\]](#) for porosities below 10%, and that of [\[34\]](#) for values above it. These are given by the equations below, which the authors claim are accurate to within 10% [\[209\]](#).

³<http://begbrokenano.materials.ox.ac.uk/>

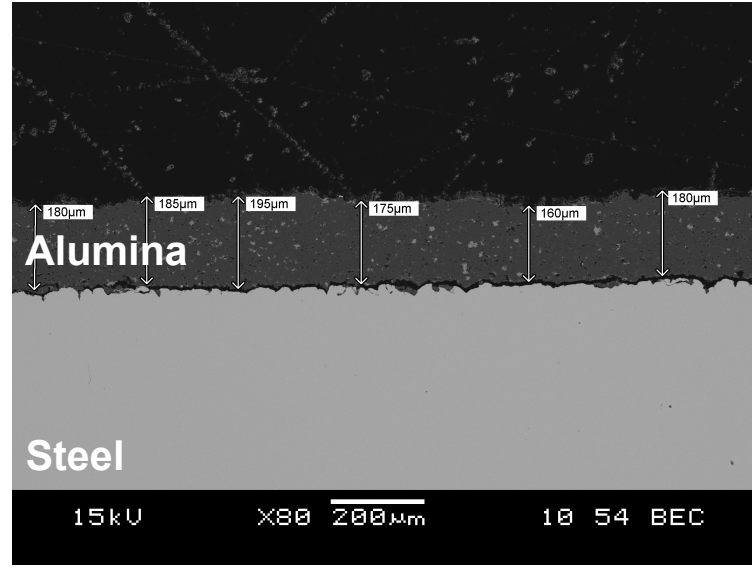


Figure 9.5: SEM cross-sectional image of APS alumina on a steel substrate. Film thickness values have also been displayed. These were compared against the quoted film thickness to determine the level of accuracy.

$$\frac{k_{\text{eff}}}{k_s} = \frac{(1 - V_{\text{por.}})}{(1 + 0.5V_{\text{por.}})} \quad V_{\text{por.}} < 10\%, \quad \frac{k_{\text{eff}}}{k_s} = (1 - V_{\text{por.}})^{1.5} \quad V_{\text{por.}} \geq 10\%, \quad (9.2)$$

Here $V_{\text{por.}}$ is the volume porosity, and k_{eff} the effective thermal conductivity of the film. The results of this analysis, for the heater materials and spray processes, are shown in Table 9.2.

As only a single sample coupon was produced for each material for cross-sectioning, the calculated thickness accuracy is signed. The biggest discrepancy between the requested film thickness, and that achieved, occurred for the HVOF deposition of MCrAlY.

It was assumed that the temperature variation in thermal conductivity for each thermally sprayed material would be the same as that for the corresponding bulk material. This is a valid approach for relatively low temperatures, where minimal amounts of sintering occur in the film [48].

Material	Spray Method	Area Porosities [%]	Volume Porosity [%]	k_s [W/mK]	k_{eff} [W/mK]	Thickness Accuracy [%]
Al ₂ O ₃	APS	7.6, 4.9, 8.3, 6.8, 8.2	7.2	25	23	+15
MCrAlY	HVOF	15, 12, 11, 11	12	20	16	-31
MCrAlY/ Al ₂ O ₃ (50:50 vol)	APS	24, 17, 20, 21	20	23	16	-5
Mo	APS	16, 16, 16, 18, 19	17	142	109	-9
MoSi ₂	APS	5.2, 5.8, 6.1, 7.2, 7.1	6.3	65	59	-15

Table 9.2: Calculated thermal conductivities of thermally sprayed thin-film heater materials based on cross-sectional data. The accuracy of the film thickness is also shown. Bulk thermal conductivities were taken from [174, 226].

9.1.2 Regression Analysis

For a fixed geometry, and material properties, the surface temperature distribution on the thin-film heater will depend on the following quantities: 1) the heat transfer coefficient magnitude and distribution; 2) the coolant inlet temperature; 3) the power input to the resistive thin-film. Instead of performing regression analyses for each of these parameters, a more efficient approach is to use non-dimensional analysis to first reduce their number. If this is done for the current case it can be shown that T^* , defined as

$$T^* = \frac{k_s (T - T_{ci})}{\dot{q} t^2}, \quad (9.3)$$

is dependent solely on the Biot number. Here the quantities in equation (9.3) have the following definitions: $\dot{q}$ - the volume averaged internal heat generation in the resistive thin-film; k_s - the thermal conductivity of the substrate; t - a reference length scale for the substrate; T_{ci} - the coolant inlet temperature. T is taken from the surface of the thin-film heater, viewed by the IR camera. It has been assumed for this work that any variation in

the thermal conductivities of the thin-films with temperature has a negligible impact on the results.

An algorithm was developed in MATLAB 2016a that took a number of points on the thin-film heater surface, together with data from a set of training simulations, and performed regression analyses to determine the relationship between the Biot number and T^* at each point. By minimising the sum of the squared differences between the simulated and experimental values of T^* at each point, a value for the Biot number was obtained. This algorithm was based on [Gaussian Process Regression](#) (GPR), which does not assume a specific functional form for the relationship between variables. More details of the application of this regression technique can be found in [\[190\]](#).

An advantage of using non-dimensional variables with [GPR](#) comes from the fact that the number of training points required scales with the number of parameters [\[258\]](#). This approach therefore greatly helps reduce the computational effort required to generate the training data. For this research, ten evenly spaced training Biot numbers were used over the expected range.

The points chosen for the regression analyses can have a significant impact on the final result. In this research, 1,200 points were used for the analyses, which were spread evenly between 60° segments of the hexagon. This number of points was based on a study performed by the author that looked at the percentage variation in the predicted Biot number, with the point density, for repeat analyses of the same data. A maximum percentage variation in the Biot number of 0.1% was set as the threshold criteria. These results are shown in image (b) of [Fig.9.6](#).

Within each segment, the radial distribution of points was such that the point density remained constant in the radial direction. A sample point cloud is shown in image (a) of [Fig.9.6](#).

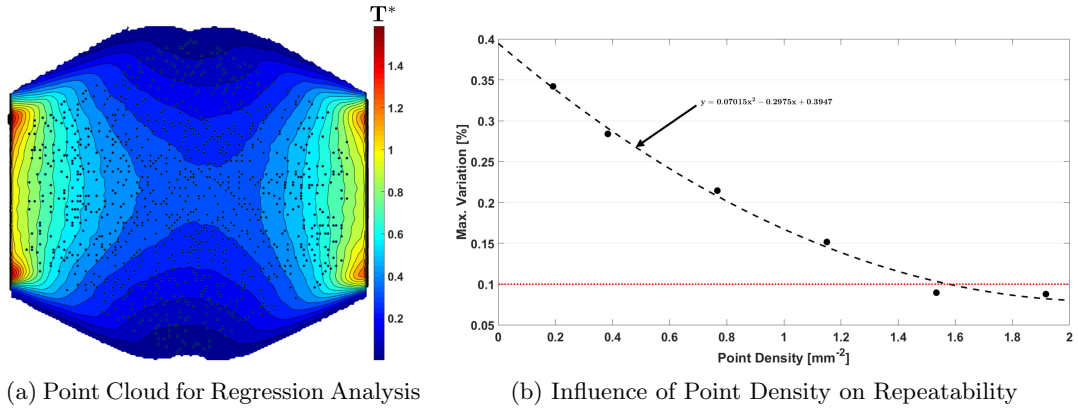


Figure 9.6: (a) Typical point cloud used for Gaussian regression. (b) Percentage variation in the predicted Biot number for repeat analyses of the same experimental T^* profile with sample point density.

The author plans to continue developing this regression technique to include additional non-dimensional parameters. For instance, separate Biot numbers could be assigned to sets of cavities to account for an un-even split of the coolant mass flow.

9.2 Uncertainty Analysis

To estimate the uncertainty in the wall overhear, the same method was used as in Sec. 3.5.2 of Chpt. 3. The results of this analysis are shown in Table 9.3.

In order to calculate the uncertainty in the wall overhear, an uncertainty was first calculated for the area averaged temperature on the HPJC-1b heat flux surface. These results are shown at the top of Table 9.3. Here, R_{th} represents the thermal resistance across the thin-film heater, and T_m the temperature measured by the IR camera.

The results indicate that the overall uncertainty in the heat transfer performance is dominated by that in the area averaged temperature on the heat flux surface of the HPJC-1b.

Parameter	Typical Value	Measurement Uncertainty	$\frac{\partial T_s(\tau_s)}{\partial \chi}$	$\frac{\partial T_s(\tau_s)}{\partial \chi} \frac{\Delta \chi}{T_s(\tau_s)}$
T_s				
T_m	40°C	2°C	0.62	0.031
$\dot{q}$	4.8 [GW/m ³]	0.2 [GW/m ³]	1.9e-9	0.0046
T_{ci}	31°C	0.5°C	1.0	0.00503
R_{th}	1.9e-5 [m ² K/W]	2e-6[m ² K/W]	-3.2e-5	-0.0015
RSS Uncertainty [%]				3.2
τ_s				
T_s	40°C	1.3°C	-0.099	0.14
T_{ci}	31°C	0.5°C	-0.099	-0.011
$T_{ce} - T_{ci}$	10°C	0.5°C	-0.092	-0.010
RSS Uncertainty [%]				14

Table 9.3: Uncertainty analysis for the area averaged surface temperature on the heat flux surface of the HPJC-1b, as well as the wall overhear. The reference values used were as follows: $T_s = 40^\circ\text{C}$; $\tau_s = 0.92$.

9.3 Results

Prior to experimentation, each module was first leak tested in the manifold system at pressures 10% above those expected during operation. All of the remaining procedures followed those in Sec. 8.4.3 of Chpt. 8.

Unfortunately, during attachment of the electrodes, the brass tabs on sample modules 5 & 6 were damaged to an extent that made testing them infeasible. The results presented below are therefore for the remaining four modules.

9.3.1 Wall Overheat

The wall overheat parameter developed by the author in Chpt. 4 was calculated for the fabricated modules using the methods outlined in previous sections. Due to the non-uniform power deposition in the thin-film heaters, the wall overheat was based on the area averaged, rather than peak, temperature on the heat flux surface of the HPJC-1b module. To avoid

confusion, this modified wall overheat parameter was given the symbol τ_s^+ . In contrast to τ_s , it is possible for τ_s^+ to take a value smaller than unity.

Initially, the flow rate to a module was set by adjusting the speed of the electric motor. The power input to the thin-film heater was then gradually increased up to ~ 500 W over a 30 sec period. This condition was maintained for a further 60 sec before shut-down. All data was time averaged over the last 30 sec of a test run. This process was then repeated at different coolant flow rates. The results of these experiments for the fabricated modules are shown in Fig. 9.7.

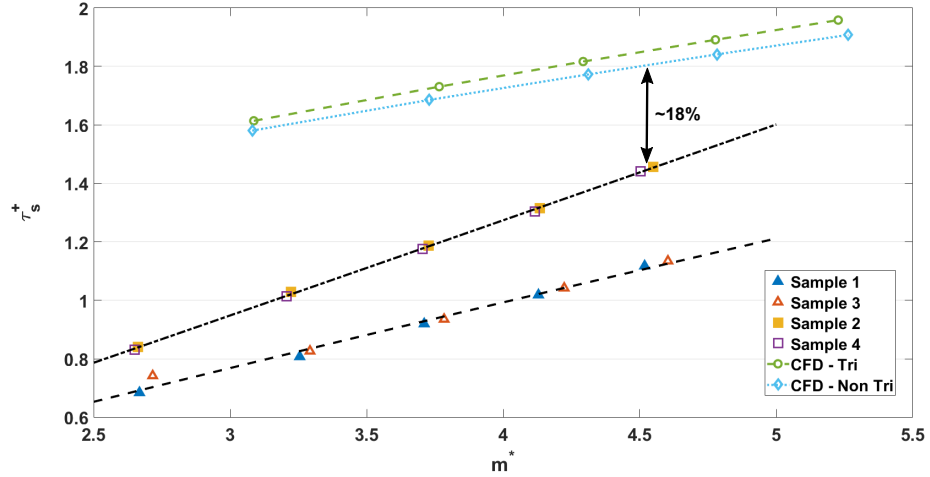


Figure 9.7: Comparison of the wall overheat performance of the fabricated HPJC-1b modules, with that predicted by conjugate CFD.

The fabricated HPJC-1b modules have wall overheats that are at least $\sim 18\%$ lower than those predicted by the **CFD**, which is above the 14% total measurement uncertainty. This difference is seen to increase with decreasing m^* . Whilst the **CFD** results indicate that the non-triangular design concept will have a reduced wall overheat, the opposite is found from the experimental results.

It is difficult to discern from these results whether it is the **CFD** itself that is incorrect, or that the fabrication process has introduced features that are not included in the numerical model. From a fabrication perspective, one potential explanation for the observed

behaviour is the addition of surface roughness, which is not accounted for in the **CFD**. This would enhance the area averaged heat transfer coefficient, lowering the wall overheat. Further research is required to determine the exact origin of the behaviour observed in Fig.9.7.

9.3.2 Pressure Drop

To determine the pressure drop performance of the modules, data was recorded over two minutes for a range of coolant flow rates. The experimental data presented in Fig.9.8 has been time-averaged over the last minute of each test run.

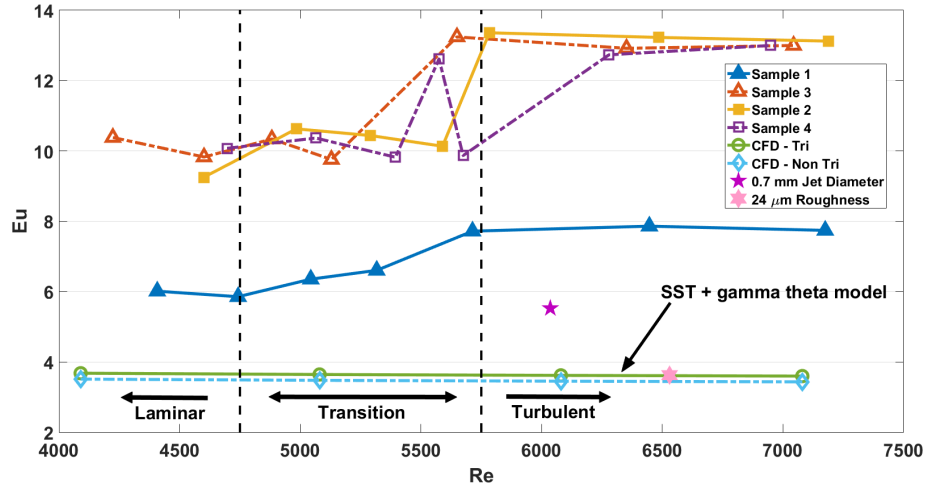


Figure 9.8: Comparison of the pressure drop performance of the fabricated HPJC-1b modules, with that predicted by CFD. Reynolds number based on a jet diameter of 0.8 mm.

It is evident that the **CFD** significantly under-predicts the pressure drop through both HPJC-1b module designs. In some cases the difference is as high as 3.5 times. There are two potential sources of this discrepancy: 1) the **CFD** fails to accurately capture the flow; 2) the fabrication process introduces geometric features that are not accounted for in the **CFD**. In order to determine the likelihood of 2), two additional simulations were performed. In one, shown by the star in Fig.9.8, the diameter of all of the jets were reduced to 0.7 mm to simulate the dimensional uncertainties of the fabrication processes (0.1 mm). Though this led to a significant increase in the Euler number, it still did not fully account for the observed results. The second simulation, shown by the hexagon, considered the influence

of introducing a wall roughness of 24 microns, which was determined from cross-sections of the modules after testing. In ANSYS CFX 16.2 a wall roughness is input through an equivalent sand grain height [5]. To relate this to the measured arithmetic average roughness of 24 microns the relation from [9] was used. The results in Fig.9.8 indicate that roughness is not the cause of the discrepancy between the CFD and experiment.

In Sec.3.5.3.3 of Chpt.3 the disparity in Euler number between the CFD and experiment was found to be $\sim 13\%$ for a system with two cascades. The results in Fig.9.8 indicate that there may be some form of compounding of the discrepancy with an increasing number of cascades. Additionally, given the approximate invariance of the CFD results to wall roughness, it is evident that the discrepancy is driven by a miscalculation of the pressure drop due to flow acceleration through bends and constrictions. This result is not surprising, as it is well known that accurately predicting pressure drop in impinging jet systems using CFD is challenging [91]. Further research is required to fully untangle the influence of factors 1) and 2) on the system pressure drop.

Another interesting feature of Fig.9.8 is the flow transition that occurs, starting at a jet Reynolds number of approximately 4,750. This same behaviour was observed in the pressure drop results for the scaled model in Chpt.3. As the jet would be turbulent at these Reynolds numbers, it indicates that the flow transition is most likely occurring in the cascade channels. As their flow area is ~ 4 times that of the jet, this indicates a transition to fully turbulent flow at a Reynolds number of $\sim 1,400$.

9.4 Summary

In this chapter initial thermo-fluid results have been presented for the fabricated HPJC-1b modules. The author has gone into detail regarding the methodology used to determine the heat transfer performance based on experimental data. A numerical model and regression technique have been specifically developed for this task. The results from the thermo-fluid measurements on the HPJC-1b modules are as follows:

- The experimental wall overheating was found to be at least 14% smaller than that predicted by CFD, which could not be accounted for solely by uncertainty;
- The triangular cavity design had a lower wall overheating than the non-triangular design. This was in contrast to the results predicted by the CFD;
- Experimental pressure drops were up to 3.5 times higher than predicted by CFD;
- There is an increase in the module pressure drop as the cascade channel transitions from laminar to turbulent flow.

Chapter 10

Conclusions

“Those of us who saw the dawn of the Atomic Age that early morning at Alamogordo know now that when man is willing to make the effort, he is capable of accomplishing virtually anything.”

— Leslie Richard Groves

This chapter describes the main accomplishments of the research and potential future areas of study. The overarching achievement has been the development of a **High-Heat Flux (HHF)** heat sink module capable of application in a near-term **DEMO** divertor target plate.

10.1 Main Research Contributions

1. Experimental validation of a conjugate **CFD** model of the heat sink: the results showed that area averaged heat transfer coefficients on the impingement surface could be accurately calculated by the numerical modelling to within $\sim 10\%$. Of the turbulence models investigated, **Shear Stress Transport (SST)** captured the local variation in convective heat transfer most accurately;
2. Derivation of a thermal figure of merit, termed the wall overheat, to facilitate comparison of **HHF** cooling components: when the wall overheat was plotted against a non-dimensional mass flow rate parameter, the resulting curve was found to be independent of material thermal conductivity and heat flux magnitude. The wall overheat values for the heat sink developed by the author were shown to be better than any

past, or present, divertor target proposal;

3. Determination of heat sink life: for monotonic loading, reserve factors below one were only found when the coolant pressure was increased above 250 bar. Fatigue life was established to be strongly dependent on fillet geometry. The heat sink achieved the DEMO divertor component life requirements of two full-power years (fpys) operation, for fillet radii above 0.05 mm;
4. Fabrication of sample heat sink modules: the laminate stack was formed using uni-axial pressure diffusion bonding. Subsequent helium leak rate measurements gave a manufacturing yield of 50% based on the ITER in-vessel criteria. All those modules that passed the leak rate criteria survived pressure testing at 300 bar;
5. Construction of a HHF experimental facility at the Osney Thermo-Fluids Laboratory (OTFL): able to operate at coolant pressures of up to 200 bar;
6. Development of the thin-film heater technique for experimentally applying HHFs: sample heaters were tested at the OTFL HHF facility developed by the author. Area averaged heat fluxes of up to 4 MW/m^2 were achieved. An alumina thickness of 150 microns was found to be sufficient to provide electrical isolation. Increasing the resistor thickness was found to improve the maximum heat flux attained before failure;
7. Thermo-fluid measurements on fabricated heat sink modules: the results showed significant discrepancy with the CFD predictions. Measured pressure drops were up to 3.5 times those predicted by the CFD. In addition, a step increase in the magnitude of the Euler number was observed as the cascade channels transitioned from laminar to turbulent flow over a jet Reynolds number range of 4,750-5,750. The wall overheat was found to be at least $\sim 18\%$ smaller for the fabricated modules, in comparison to the CFD. From the data available it was not possible to determine whether the discrepancies were driven by compounded errors in the CFD, due to the additional number of cascades in the HPJC-1b, or un-quantified influences from the fabrication process.

10.2 Future Research

If a HPJC-type component is to be deployed in a first stage DEMO divertor then there are a number of outstanding technical issues that need to be addressed. There are also methods that have been employed in this research that would benefit from further refinement.

- To progress further along the Technology Readiness Level (TRL) scale an HPJC-type module needs have its life determined experimentally. This would include attachment of an armour tile;
- A means of manifolding multiple HPJC-type modules onto a single divertor plate needs to be developed. This could focus on a small poloidal strip in the vicinity of the strike-point;
- Additional effort is required to optimise the fabrication process. Given that the manufacturing yield was determined by the leak rate test, the author suggests focussing on minimising the number of bond interfaces, or considering a process like investment casting to create the internal cavities;
- Thin-film Heater Method:
 - Development of an Finite Element Analysis (FEA) model to aid design;
 - Application to measuring component life, as well as thermo-fluid performance;
 - Determining performance repeatability;
 - Reduced uncertainties in material properties;
- Modelling
 - Extension of the Gaussian Process Regression (GPR) method to include additional parameters. Explore parametrising the heat transfer coefficient distribution;
 - Incorporation of neutron irradiation effects into life analyses;
 - Extension of the wall overheat methodology to fully incorporate two-phase flows.

Appendices

Appendix A

Technology Readiness Level

The **Technology Readiness Level** (**TRL**) scale is used as a measure of the maturity of a technology. As a concept is developed it progresses from a **TRL** of one, up to a final value of nine when it is successfully operated in its intended environment. Further details on the criterion for each level in the **TRL** scale are shown in Fig.A.1

The image(s) originally presented here cannot be made freely available by ORA due to copyright.

Figure A.1: Breakdown of the criteria required for a technology to reach a specific level in the **TRL** scale **169**.

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