

CRANFIELD UNIVERSITY

Stergios Topouris

Design and Optimisation of a High Performance Lightweight
Monoblock Cast Iron Brake Disc

School of Aerospace, Transport and Manufacturing

PhD

Academic Year: 2012 - 2017

Supervisor: Dr. Marko Tirović

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This thesis is submitted in partial fulfilment of the requirements for
the degree of PhD

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ABSTRACT

In the past few decades, part weight reduction that would yield lower fuel consumption and decrease CO₂ emissions has been one of the main challenges in the automotive industry. In brake rotors, the downsizing is even more difficult because mass reductions may result in lower thermal capacity which will compromise rotors performance. This research work focuses on developing a design methodology for a novel concept of a lightweight monoblock brake disc with fingered hub. Discs of such a design can feature reduced mass, lower cost, improved cooling and are particularly suitable for Electric and Hybrid Electric Vehicles.

The described methodology comprises a number of steps, each investigating a different attribute of this novel disc design and ensuring its feasibility. The use of Finite Element Analysis was incorporated to study the mechanical and thermomechanical loading on the disc with the results indicating acceptable behaviour. Physical testing with an actual prototype disc confirmed the findings of the modelling work. Thereafter, structural optimisation was conducted where significant reduction in mechanical stress was achieved and also further weight reductions. Finally, in a newly developed experimental facility cooling tests were conducted that revealed substantially improved cooling characteristics for the brake disc with fingered hub when compared to the baseline solid hub disc.

Keywords:

Finite Element Analysis, Structural Optimisation, Lightweight Brake Disc, Fingered Hub, Brake Disc Cooling, Disc Cooling Rig.

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‘The art of engineering lies not in solving the equations but in picking the “right” equations and installing the “right” values so the result of our mathematics answers the question we had in mind at the beginning’.

Robert J. Moffat

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NOMENCLATURE

Greek Symbols

$\bar{\theta}_D$	Average Surface Temperature of the Disc	[°C]
$\bar{\theta}_D$	Average Surface Temperature of the Disc	[°C]
$\bar{\theta}_s$	Average Surface Temperature	[°C]
θ_∞	Ambient Temperature	[°C]
κ_{air}	Thermal Conductivity of Air	[W/mK]
ν_{air}	Kinematic Viscosity of Air	[m²/s]
α	Coefficient of Thermal Expansion	[1/K]
ε	Surface Emissivity ($0 \leq \varepsilon \leq 0.99$)	
κ	Thermal Conductivity	[W/mK]
μ	Coefficient of Friction	-
μ	Dynamic Viscosity of the Fluid	[kg/ms]
ρ	Density	[kg/m³]
σ	Stefan-Boltzmann Constant ($5.67 \times 10^{-8} \text{ W/m}^2\text{K}^2$)	
τ	Tightening Torque	[Nm]
ω	Rotational Speed	[RPM]

Latin Symbols

$\bar{h}_{Dconv}$	Average Disc Convective Heat Transfer Coefficient	[W/m²K]
$\bar{h}_{Drad}$	Average Disc Radiative Heat Transfer Coefficient	[W/m²K]
$\bar{h}_{Dt}$	Average Disc Total Heat Transfer Coefficient	[W/m²K]
h_{cond}	Conductive Heat Transfer Coefficient	[W/m²K]
h_{conv}	Convective Heat Transfer Coefficient	[W/m²K]
h_{rad}	Radiative Heat Transfer Coefficient	[W/m²K]
A_{CS}	Cross Sectional Area	[m²]
A_{cond}	Area of Conduction	[m²]
A_{rad}	Area of Radiation	[m²]
A_{rad}	Area of Radiation	[m²]
A_w	Wetted Area	[m²]

Cp_D	Specific Heat Capacity of the Disc	[J/kgK]
$\overline{Nu}$	Mean Nusselt Number	-
Q_{cond}	Conductive Heat Dissipation	[W]
Q_{conv}	Convective Heat Dissipation	[W]
Q_{rad}	Radiative Heat Dissipation	[W]
R_a	Roughness Average	[μm]
Re_t	Crossflow Reynolds Number	-
Re_ω	Rotational Reynolds Number	-
R_o	Outer Radius	[mm]
T_∞	Ambient Temperature	[K]
T_D	Surface Temperature of the Disc	[K]
T_w	Temperature of the Wall	[K]
d_{ID}	Disc Inner Diameter [mm]	[mm]
d_{OD}	Disc Outer Diameter [mm]	[mm]
l_{cond}	Length of Conduction	[m]
m_D	Mass of the Disc	[kg]
r_r	Rolling Radius	[m]
A_{ch}	Flow Channel Cross Sectional Area	[m ²]
Gr	Grashof Number	-
ID	Inner Diameter	[mm]
Nu	Nusselt Number	-
OD	Outer Diameter	[mm]
Pr	Prandlt Number	-
Q_{air}	Air Volume Flow	[m ³ /s]
Ra	Rayleigh Number	-
Re	Reynolds Number	-
U	Linear Speed	[m/s]
V_{air}	Air Velocity	[m/s]

LIST OF ABBREVIATIONS

ARC	Active Resonance Control
BC	Boundary Conditions
BIW	Body in White
CAE	Computer Aided Engineering
CFD	Computational Fluid Dynamics
CGI	Compacted Graphite Iron
DoF	Degrees of Freedom
DTV	Disc Thickness Variation
EV	Electric Vehicle
FBG	Fibre Bragg grating
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Method
FFT	Fast Fourier Transform
FOD	Foreign Object Debris
GI	Grey Cast Iron
GVM	Gross Vehicle Mass
HEV	Hybrid Electric Vehicle
HTC	Heat Transfer Coefficient
ID	Inner Diameter
LES	Large-eddy-simulations
MHS	Macroscopic Hot Spots
MMC	Metal Matrix Composite
MPC	Multi Point Constrains
MY	Model Year
NVH	Noise Vibration and Harshness
OD	Outer Diameter
PDGE	Partial Deferential Governing Equation
RPM	Revolutions Per Minute
SG	Spheroidal Graphite
SIMP	Solid Isotropic Material with Penalization

TCR	Thermal Contact Resistance
TEI	Thermoelastic Instability
UKAS	United Kingdom Accreditation Service
UTS	Ultimate Tensile Strength

1 INTRODUCTION

1.1 Research Project Background and Structure

This research project focuses on developing a methodology for designing a novel monoblock ventilated brake disc that would feature a fingered hub in place of the traditional solid hub (Figure 1-2). Such designs offer a number of potential benefits: reduced mass, lower cost, improved cooling characteristics and more flexibility in directing the cool air towards their inner vanes. The last advantage is particularly important for Electric Vehicles (EV) and Hybrid Electric Vehicles (HEV), with in-wheel motors and also for more conventional designs. Furthermore, it should be possible to use the disc with solid wheels that improve vehicle aerodynamics and reduce fuel consumption. These potential advantages are vital in today's highly competitive market, with strong demands for cost reduction, lower fuel consumption and CO₂ emissions.

There are without a doubt a number of challenges to make such a design a practical alternative to conventional discs. To start with, the design must be structurally sound under mechanical and thermal loads. Moreover, a manufacturing process had to be developed. Via private communications, it was appreciated that such process can be established for suitably high manufacturing volumes leading to competitive prices of the final fully machined discs.

Initially, the study was focused on investigating the concept feasibility for a brake rotor intended for a mid-size family sedan. After developing a number of concepts, the thermomechanical analyses using the Finite Element Method (FEM) indicated that the thermal stresses are not likely to cause any problems for the disc safe operation. However, the mechanical analyses that followed showed that discs cannot withstand the mechanical loading caused by high braking torque.

This resulted in the study exploiting the possibility of having a brake disc with a fingered hub design for a lightweight vehicle that can develop a relatively low braking torque. The lightweight vehicle identified was the Lotus Elise, Model Year

(MY) 2013, and its standard disc, the S2 model, became the basis for the new investigation. The original disc is shown in Figure 1-1.

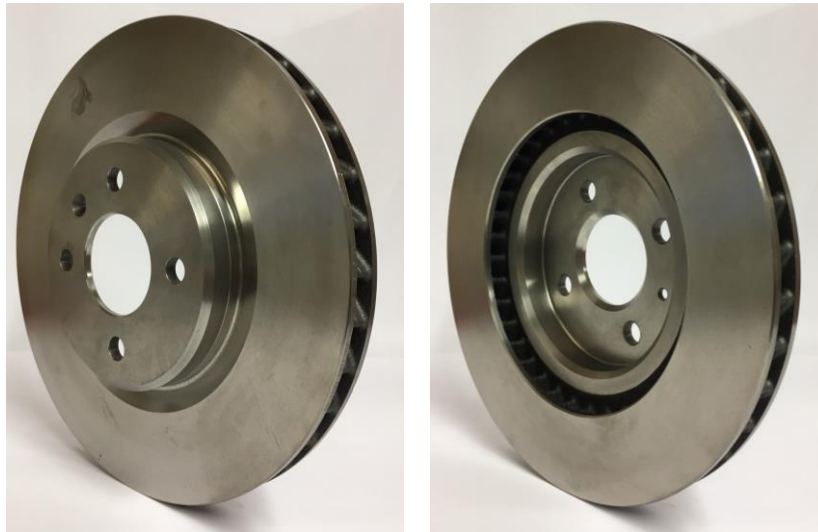


Figure 1-1 Lotus Elise S2 Brake Disc



Figure 1-2 Prototype Elise Brake Disc with Fingered Hub

The first step of the study led to the generation of three conceptual designs. In the second step, different modelling approaches to simulate the mechanical modelling were investigated and in the third step, initial mechanical analyses with simplified material data were conducted in order to get an initial estimate whether the concept of having brake discs with fingered hub design is feasible for lightweight vehicles. The results were promising, and so in the next step

mechanical analyses were conducted with more complex material data. The results again indicated that it was feasible for the alternative hub design to work effectively. In the fifth step, thermomechanical analyses were carried out with the use of simple and complex material data. The results from the disc with fingered design compared favourably with the results from the standard Elise disc; that gave more confidence to the feasibility of the fingered hub concept.

Experimental investigations with regards to mechanical loading were conducted in the sixth step of this research project. A prototype disc based on one of the conceptual designs was manufactured and subjected to torsion testing. The performed experiments served as validation for the mechanical Finite Element Analyses (FEA). In addition, the true limits of the design were identified by loading the prototype rotor to the point of fracture and also a separate test provided information about the limit of cast iron in bending.

In the following steps, structural optimisation was performed to the fingered hub conceptual design disc. In the first part, the technique of shape optimisation was used to reduce the mechanical stresses and then in the second part, the method of topology optimisation was incorporated aiming to investigate the possibility of further weight reductions. Both techniques yielded good results especially in the case of shape optimisation.

Thereafter, a brake disc cooling facility was developed, manufactured and commissioned with its purpose being the investigation of cooling performance and cooling behaviour for the prototype disc with the fingered hub design. In the final steps of this study, the aforementioned cooling facility was used to compare the convective heat transfer coefficients between the prototype disc and the standard production rotor. The results showed that the prototype Elise disc with the fingered hub cools better than the existing S2 model.

1.2 Aims and Objectives

The aims of the research are:

- To establish a methodology designing and optimising monoblock discs with fingered hub,
- To validate the methodology through Finite Element (FE) analysis, structural optimisation, and physical testing in specific disc designs,
- To establish gains in terms of mass reduction and performance increase via FE modelling and physical tests,
- To demonstrate a viable design solution at a proof of concept stage.

The objectives of this research are to:

- Encompass disc shapes that are fully interchangeable with the existing solid hub design (Elise S2),
- Investigate modelling methods to most effectively address the mechanical/thermal loading and research and evaluate the material characteristics,
- Validate the results through sets of tests for loading conditions considered most critical,
- Structurally optimise the design aiming in lower stresses and further mass reduction,
- Investigate cooling characteristics experimentally and establish improvements when compared to the original solid hub design,
- Conduct the tests for a range of crossflow air velocities, disc rotational speeds and angles in a controlled environment,
- Based on modelling and experimental work, confirm that the fingered monoblock disc design is a viable proposition, at least at the proof of concept level.

2 LITERATURE REVIEW

2.1 Introduction

In the present Chapter, a review of the existing publications concerning the topic of the thesis is presented. Due to the multiple areas of investigation during the project, the literature review has been divided in four main sections. In the first section, the aspects of brake system analysis are presented and discussed with the main focus on the brake discs. In the second section, the characteristics and properties of cast iron are investigated since this is the most commonly used material for the brake rotors and it is the material that the brake rotors studied in this thesis were made out of. The third section presents automotive wind tunnels: design aspects, design approaches, features, possible issues and ways to overcome them. The final section of this Chapter pertains to the subject of brake disc cooling, the different modes of heat transfer, early analytical investigations and more modern numerical analysis methods.

2.2 Brake Disc Analysis

In the early years of the brake disc analysis, between the decades of 1950s and 1960s when the FEM was still not developed sufficiently, the analyses were considering mainly the temperature distributions within the brake rotors. Newcomb presented an analytical approach to determine the temperatures on disc brakes during braking by using Laplace transformations (Newcomb, 1959). The author assumed the brake pair to be three parallel infinite slabs of solid with the two interfaces being subjected to thermal flux linearly decreasing with time and the two outer interfaces being subjected to no heat flow.

For his calculations he approximated 99% of the heat energy to go to the disc and 1% to the pads. His predicted temperatures when compared to values obtained in physical tests showed agreement in the level of 15%. The author finally noted that if the cooling losses have been taken into account the level of accuracy between the calculated values and the experimental values would be lower.

Newcomb continued his work, by including the cooling losses in his calculations for determining temperatures on disc brakes (Newcomb, 1960). That resulted in predicted temperature values being closer to experimental values, but the percentage of deviation was not mentioned. It was recognised though, that the cooling of convection and radiation combined has an influence of only 6% for discs subjected to a single stop with constant deceleration.

Moreover, by utilising his analytical method he calculated the temperatures in the disc/pad interface for asbestos-resin and sintered pads assuming 99% and 95% of the thermal energy going to the disc respectively. The calculated values were very close to the measured values especially for the asbestos pads. Finally, he applied the Newton's law of cooling for the case of repetitive brake applications with the calculated values having good correlation to the measured values for physical experiments.

Ling and Pu presented the mechanism of Thermal Contact Resistance (TCR) (Ling & Pu, 1964). The authors conducted a novel experiment considering the position of points of contact between two solids which are sliding vary randomly with time (Figure 2-1).

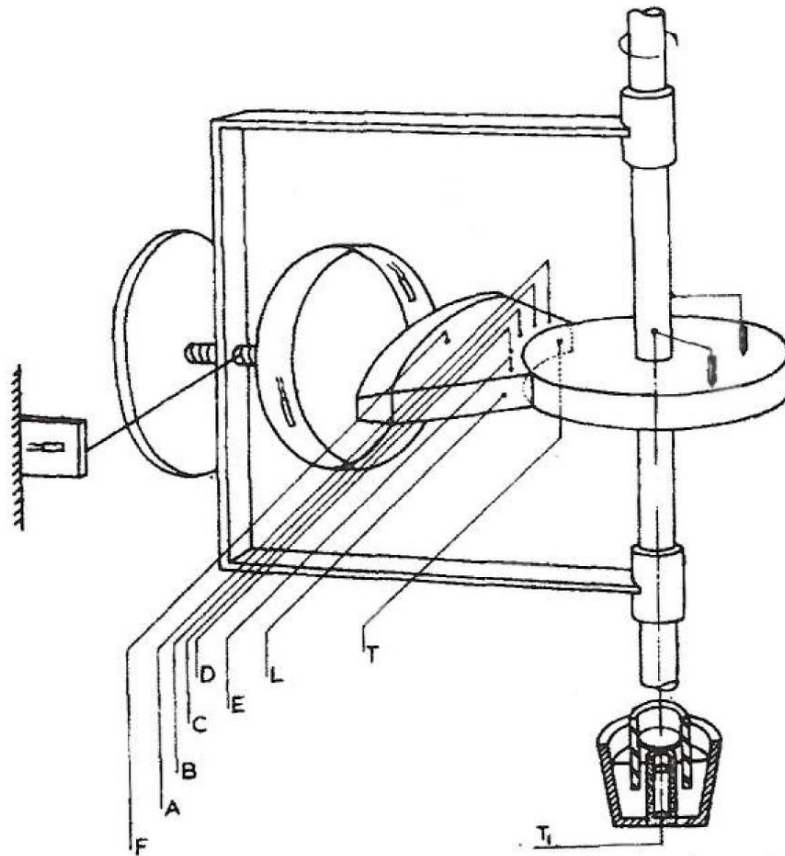


Figure 2-1 F. F. Ling and S. L. Pu Experimental Set-up (Ling & Pu, 1964)

They suggest that heat is not generated on the interface of the two solids in contact but on one or both the surfaces of these solids. This heat will flow from one solid to the other depending on the coefficient of thermal conductivity of each of the solids, and the heat transfer mechanism is conduction.

The authors additionally stated that between the two sliding solids a thermal resistance mechanism exists which resists the heat transfer via conduction. This is supported by the “temperature jumps” that were observed in macroscopic scale from one solid to the other. This mechanism is known as Thermal Contact Resistance.

One of the first numerical analyses on disc brakes was cited in (Abbas, et al., 1969), where they investigate ways to create a tool in order to predict the temperature distributions on the bell area of brake discs. After explaining that due to the complex geometry of brake discs the analytical method is not effective, they presented a numerical model to predict the temperature distributions. They

simplified their model to one-dimensional problem by assuming that the temperature does not vary through the thickness of the disc and then presented each station of temperature calculation.

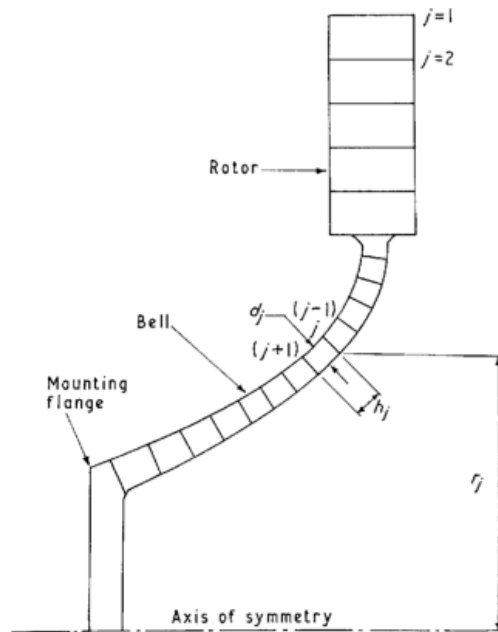


Figure 2-2 S.A. Abbas et al. Half Section Model of a Brake Disc (Abbas, et al., 1969)

Then they modelled two braking cases: a constant deceleration to a stop and a drug application. Thereafter, the predicted temperature distributions on the disc's bell were compared against experimental values that the authors measured on physical tests they conducted replicating the two cases that they modelled. By using the trial and error approach they identified values of convective heat transfer coefficients for which the numerical model gave results similar to the measured temperature values for both braking cases.

Whereas the majority of published work investigates the heat transfer due to convection, Dike recognised that in an emergency brake application due to the short time duration of the event, heat transfer by convection and radiation are small compared to the conduction within the disc (Dike, 1974). Motivated by this, he investigated analytically the conductive heat transfer by reducing the 3D geometry of a disc with pads to 2D, by assuming uniform pressure distribution, uniform heat flow and uniform temperature distribution on the disc. Also, the

values of specific heat and thermal conductivity were assumed temperature independent.

Furthermore, it was stated that several limiting factors, such as available space, highest allowable rotor weight and/or moment of inertia are impacting the brake rotor design. Finally, guidelines for disc sizing based on the limiting factors and the maximum allowable disc temperature were proposed.

A categorisation of the thermal analyses of brake rotors found in the literature in exact analytical solutions and numerical difference solution was conducted in (Sisson, 1978). The author after explaining the shortcomings of both these methods, he presented how by using the Duhamel's theorem and combining analytical and numerical solutions, he developed an algorithm to predict temperatures and thermal stresses on ventilated brake discs.

The heating rates input for the algorithm were easily identified but the cooling rates for the air moving around the disc and the air pumped through the vanes had to be obtained experimentally. For that he conducted two types of bench tests, where in the first he measured the average vent air speed and in the second he measured the vent heat transfer coefficients. In both tests the author derived the expressions, equations (2-1) and (2-2), with which the desired inputs for the algorithm were calculated within the range of disc rotational speed used.

$$\begin{aligned} & \text{average vent air speed} & \text{(2-1) Air} \\ & = \frac{\pi \times d_{OD} \times RPM}{60} \sqrt{-0.0201 + 0.2769 \times d_{ID} - 0.0188 \times d_{ID}^2} & \text{Vent} \\ & & \text{Speed} \end{aligned}$$

$$h_{vent} = h_o + \alpha U_{vent}^N \quad \text{(2-2) Vent Heat Transfer Coefficient}$$

Where:

d_{ID} : Disc inner diameter,

h_{vent} : Vent convective heat transfer coefficient,

α : Coefficient of thermal expansion,

U_{vent} : Vent air velocity.

In equation (2-2) the terms h_o , a and N are constant and U_{vent} is the average vent airspeed. Moreover, the author experimentally derived an expression for the convective heat transfer coefficient where the parameters “ref” subscript are known or measured.

$$h = \left[\frac{\kappa_{air}}{\kappa_{air\ ref}} \right] \left[\frac{\nu_{air\ ref}}{\nu_{air}} \right]^{0.8} \times h_{ref} \quad \text{(2-3) Sission's Convective Heat Transfer}$$

Where:

κ_{air} : Air thermal conductivity,

ν_{air} : Air kinematic viscosity

It was shown that predicted temperatures and stresses correlated very well with the equivalent experimental data especially when 55% of the energy was assumed to go in the inboard rim of the disc brake. Finally, the author mentioned the importance of the accurate identification of the heat transfer coefficients and the impact that they have on the accurate predictions of his model. Additionally, it was noticed that the influence of the residual brake torque should not be neglected in the thermal analyses of brake discs.

With the improvement of the computer hardware and the advancements in the FEM algorithms, the use of FEA became a useful tool for the analysis of disc brakes. Day and Newcomb investigated the thermal dissipation of the interface between the disc and the pad by using a simple 2D FE model representing the disc rotor and the brake pad (Figure 2-3). By performing a numerical analysis and a physical experiment the authors intend to study the behaviour of the friction pair under a certain duty. They were mainly interested in the interface pressure, temperature and wear (rate), their values and how these three are linked to each other (Day & Newcomb, 1984).

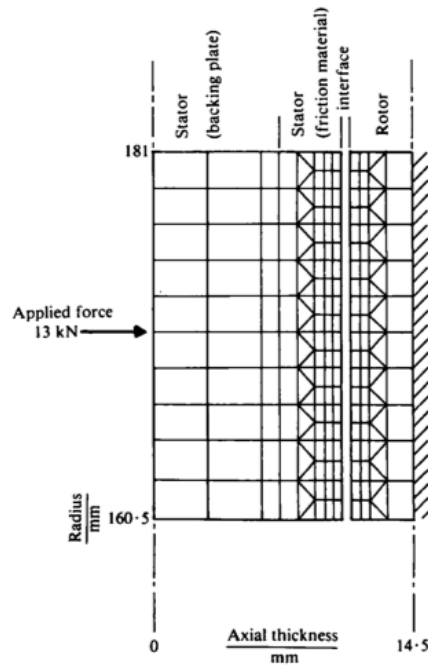


Figure 2-3 A. J. Day and T. P. Newcomb Axisymmetric FE Mesh (Day & Newcomb, 1984)

In both approaches the pads had higher temperatures than the disc, which shows that the assumption that adjacent surfaces will have the same temperatures is not valid. This happens due to contact the resistance on the transfer of frictional heat across the interface. In the experimental work it was noticed that the interface pressure, the temperature and wear of the pads increase towards the outer diameter.

Finally, Day and Newcomb concluded that the performance of resin bonded composite friction materials in a particular brake design is strongly dependent upon the dissipation of frictional heat from the interface. Also, under dynamic braking conditions interface pressure is seldom uniform. Finally, the general wear rate of resin bonded friction material may be realistically considered as being directly proportional to interface pressure and exponentially related to temperature.

In general, the major design target when developing a disc brake is the maximum allowable temperature on the rotor. This is influenced by the combination of

some, if not all, of the following factors and also the duty a disc is subjected to (Sheridan, et al., 1988):

- Lining fade
- Lining ware
- Lining property changes
- Rotor corrosion
- Rotor strength deterioration
- Fluid boil
- Elastomer durability

It was also stated by the Sheridan et al., that for different needs, such as initial sizing, temperature, mass trade off evaluations, space, wheel or wheel trim effects, on cooling and braking operating temperatures, thermal deflection determination of system development or problem solving, there are different modelling techniques. Because of this, they presented four different modelling approaches for the thermal modelling of disc brakes (Sheridan, et al., 1988).

The first was a lump model for the prediction of transient bulk temperatures and is useful for the initial sizing of the disc. The second was a 1D model that could predict peak surface temperatures and also bulk temperatures. A model like this gives enough accuracy for preliminary design processes. The third was a 2D steady-state model able to predict the temperature distributions on the frictions faces and the vane walls for multiple brake applications. Finally, a 3D model of a ventilated disc was developed. In order to reduce the computational time, power and storage space required the authors took only a segment of the full disc that included half of the vane wall and half of the vane channel.

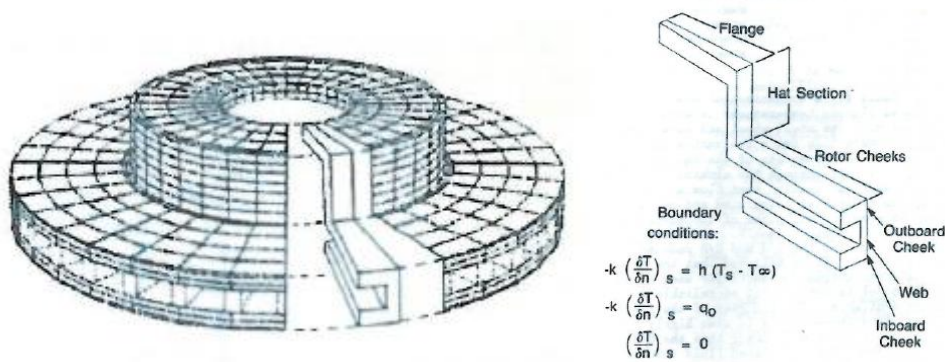


Figure 2-4 D. Sheridan et al., 3D Model (Sheridan, et al., 1988)

With the appropriate boundary conditions, the assumption of uniform heat generation and proper material properties they found that this modelling approach can give the temperature distributions, stresses and deformations of an entire brake disc.

The consequences of the Thermoelastic Instability (TEI) regime with regards to the normal pressure distributions and surface temperatures on clutch and braking systems comprising of multiple discs in parallel were studied in (Zagrodzki, 1990). By creating simplified analytical and numerical models Zagrodzki concluded that the normal pressure distributions are affected by the discs' material (steel or friction material) and the friction material's Young's Modulus. In addition, the discs' geometry, manner of supporting the opposing plate and the shape of the hydraulic piston are influential aspects in the normal pressure distributions which have impact on the temperature gradients and thermal stresses on the discs' surfaces.

Thermal effects and pressure distributions were studied as they are considered to be crucial for the design of disc brakes. More specifically, the two aforementioned aspects are linked with potential problems in performance variation and rotor damage. This led Day et al., to study these two phenomena. They divided the thermal effects into two categories: the bulk temperature effects and the macroscopic thermal effects (Day, et al., 1991).

In bulk temperature effects, the influence of pressure distribution on interface temperature, the influence of coning on pressure distribution and the influence of temperature on the coefficient of friction (μ) were studied. In macroscopic thermal effects, the influence of TEI, in the braking process was discussed, how the cast iron is affected by the thermal effects, what causes heat spots and heat bands on rotors, their consequences, and finally how cracks are initiated and their results were presented.

This work concluded that interface pressure, interface temperature and wear are interdependent. High interface pressures can occur in light duties under low actuation forces. Disc coning has a direct influence on interface pressure. Finally,

the “chill effect” was mentioned, followed by ways to reduce high interface pressures between the linings and the mating surfaces.

In relation to the main topic of this thesis, which is the design of lightweight brake discs, Grieve et al. present a design methodology to develop lightweight brake rotors made out of Al-MMC (Grieve, et al., 1998). In the first step of their methodology, they conducted numerical simulations to evaluate the feasibility of using Al-MMC in the brake discs of small to medium passenger vehicle by modelling a 10° segment of a disc/wheel/hub assembly (Figure 2-5). They simulated two braking scenarios: a) an alpine decent and b) an emergency stop from the top speed of the vehicle. The results showed that it was possible to have Al-MMC rotors but the weight of the vehicles had to be low.

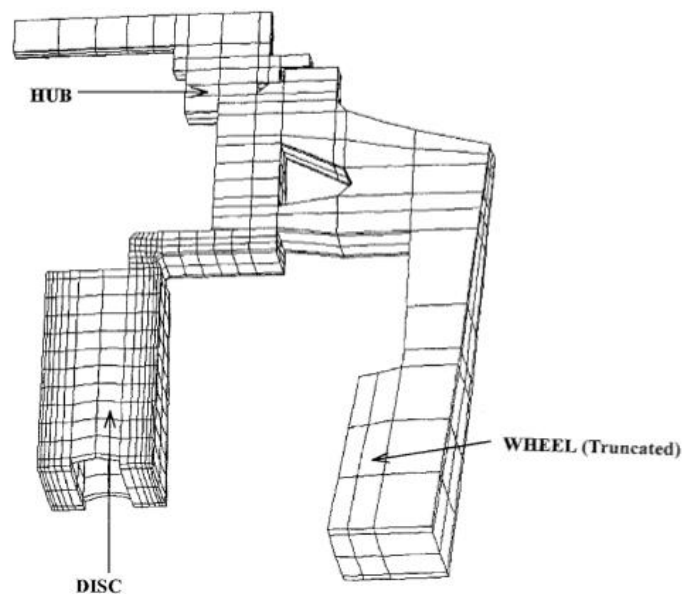


Figure 2-5 D.G. Grieve et al. 10° Segment FE Model (Grieve, et al., 1998)

In the second step, they wanted to investigate the most influential factors on the brake disc design. By using the Taguchi technique, they concluded that the cheek thickness, cooling rate and conductivity are the most important design factors. In the third and final step of their methodology, they performed a parametric study on the disc's cheek thickness, vane wall height and different Al-MMC types to see which combination would allow acceptable temperature levels for the two braking scenarios mentioned in step one and for a vehicle with gross vehicle mass (GVM)

of 1830 kg. They found that a disc with cheek thickness of 12 mm, vanes of 3 mm made out of 68% SiC MMC or 7% TiB2 MMC reinforced Al-MMC to operate under the maximum operating temperatures of the materials.

It has to be mentioned that the authors based their study on the temperature prediction and not on thermal stress predictions or thermal deformations. No validation with physical tests took place as at that stage the described methodology was planned to be used for a creation of a rotor prototype.

Around the beginning of the 21st century, the use of the FEM was well established. Techniques were integrated or were used side by side with the FEM aiming to introduce more accurate boundary conditions or reduce the computational time for more complex modelling simulations. Valvano and Lee recognised that when modelling long heat cycles on brake discs, due to prominent temperature rise and developed thermal crack effects, the heat inputs to FE models can be inaccurate resulting in outputs with unreliable results (Valvano & Lee, 2000). To overcome this issue, they developed a modelling tool comprising of a PC-based algorithm and a FE model that predicted brake component temperatures, stresses and deformations of a modelled braking system. Friction heat inputs and heat transfer coefficients were input to the PC-algorithm and the outputs included predicted rotor, pads and brake fluid temperatures. These outputs were then the inputs to the FE, model which by conducting an uncoupled thermal stress analysis, could predict the disc coning, stresses or cracks as a result of thermal fatigue.

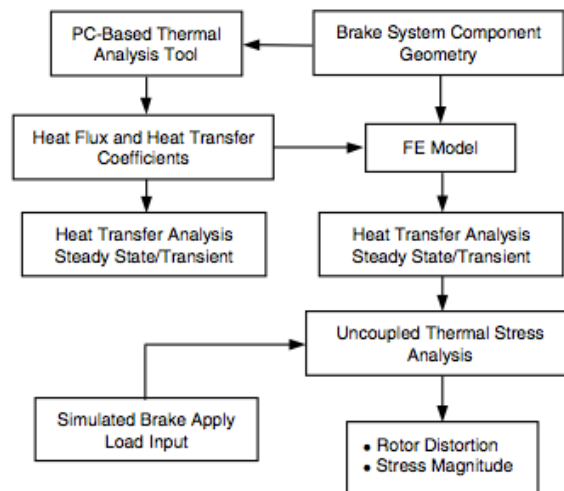


Figure 2-6 T. Valvano and K. Lee's Proposed Analysis Process (Valvano & Lee, 2000)

The results from the developed modelling tool had good correlation against physical tests. However, there is no reference on the type of material data used or the type of analysis conducted by the FE model.

Cho and Ahn were motivated by the fact that modelling the thermoelastic instability with the FE method required at that time very complex models with long computational times and using analytical methods was not possible they created a method that combined the Fast Fourier transform (FFT) and the FEM (Cho & Ahn, 2002). With the FFT, they reduced the transient three-dimensional partial differential governing equation (PDGE) describing the TEI, rotation of the disc and the boundary conditions of the braking situation modelled into a two-dimensional PDGE. Thereafter, they used the FEM and the derived results were then inverted to the three-dimensional space by using the inverse FFT technique. This approach resulted in an efficient modelling methodology of the TEI regime. It has to be mentioned though, that the results produced by this technique were not validated against physical tests or data in the literature.

Choi and Lee performed transient thermoelastic analyses for sliding disc brakes due to the importance of the phenomenon in the brake system performance and the brake disc fatigue (Choi & Lee, 2003). They created a simplified axisymmetric model with disc and pads. The analysis was linear elastic and included frictional

contact, heat transfer with convection but no wear. It was mentioned that even though linear properties were used, the contact problem is non-linear because the contact area does not change linearly with the applied load and lead to the use of the fully implicit transient scheme for better accuracy in the results at every step of the simulation. Their results indicated that the hoop stresses on the disc had the highest compressive values and they are the crucial ones in terms of thermal fatigue. In addition, it was shown that the friction material properties, especially the thermal expansion and coefficient of elastic modulus, had considerable influence in the pressure distributions of the friction faces leading to large temperature variations and hot spots that can cause cracks. Thereafter, the authors conducted the analyses with the use of orthotropic material data for the friction pair, found in carbon-carbon systems, that showed that due to the material's low thermal expansion, the pressure distributions between the friction pair were very uniform leading to uniform temperature distributions. Lastly, with regards to validation, the authors compared the results from their transient model with the results of a steady state thermoelastic model.

Okamura and Yumoto conducted a large-scale performance comparison for the different designs of ventilated automotive brake discs (Okamura & Yumoto, 2006). The designs were compared against each other on criteria like the surface temperatures and top-hat temperatures reached during braking, axial deformations and cooling behaviour.

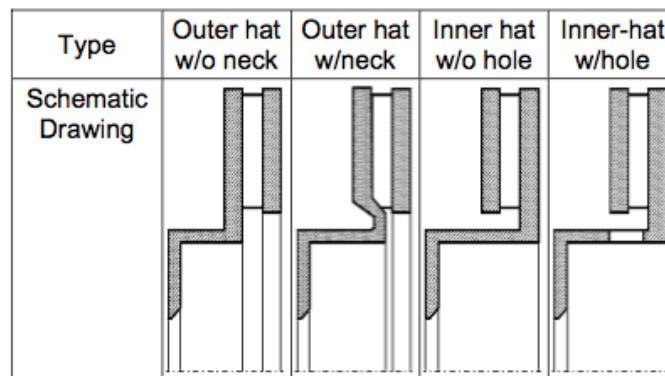


Figure 2-7 Types of Ventilated Brake Discs Studied (Okamura & Yumoto, 2006)

Thereafter, the authors by using a conventional linear FE modelling approach, conducted a parametric study based on the Taguchi Method in order to identify the critical geometrical parameters influencing the brake disc coning. The geometrical parameters investigated are presented in Figure 2-8. For each of these parameters minimum and maximum values were assigned.

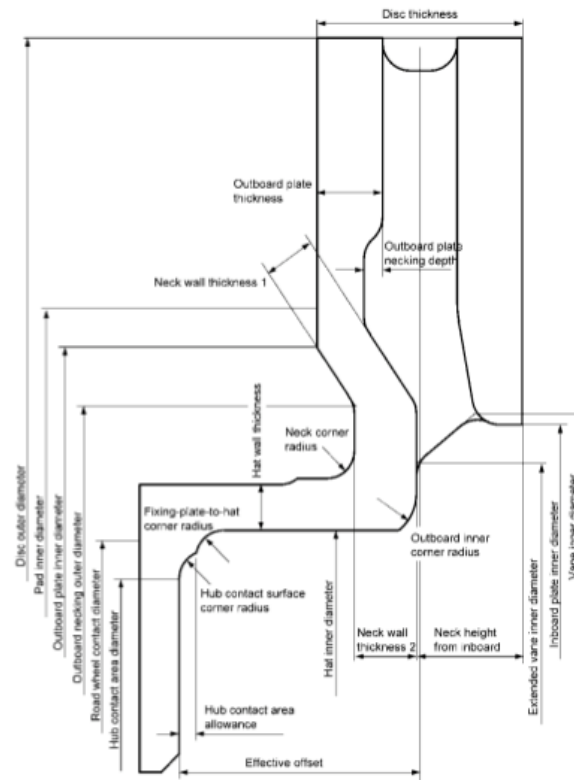


Figure 2-8 Geometrical Parameters Investigated (Okamura & Yumoto, 2006)

They concluded that large offsets and thick friction faces generally improve the coning behaviour of a disc and also highlighted the areas of the disc that have to be stiffened and the ones that have to be weakened in order to achieve the desired results. Based on these conclusions, a guideline was proposed on how to produce disc designs with minimum coning

Apte and Ravi described a numerical method to predict the thermal performance of a braking system that includes the brake disc, the hub and the mounting fasteners but not the pads for the braking scenario of repetitive brake application – fade test (Apte & Ravi, 2006). Initially, they used custom made software, based on analytical equations, to calculate the heat flux and the heat transfer

coefficients to be used in the numerical simulation. Thereafter they created an axisymmetric 3D FE model with linear material properties. Interestingly, they divided the friction surfaces of the brake disc into four sections and they applied different heat transfer coefficients on each that included convection and radiation but their values were assumed to stay constant with temperature (Figure 2-9).

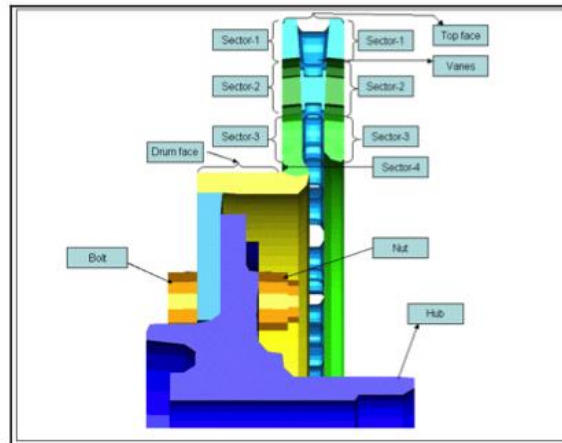


Figure 2-9 Sectors of Different Heat Transfer Coefficients (Apte & Ravi, 2006)

The MSC Nastran was used as solver with the transient analysis module SOL 159. A solid and a ventilated disc were modelled with the predicted values of temperature showing good correlation against values from physical tests especially for the solid disc. Thereafter, the authors used the same modelling approach to create a non-linear static 3D FE model to help them create a new brake disc design with improved thermal performance.

Gao et al. aimed to investigate the effect of hot spots on brake discs' friction surfaces during braking and how these can result to crack generation and propagation (Gao, et al., 2007). They created a 3D numerical model including the braking pair of a disc and two pads. The analysis was linear elastic, sequentially coupled and since the disc modelled was solid no forced convection was included. In addition, the model was simplified by assuming symmetrical loading on the friction surfaces, no wear and uniform pressure application on the backplates of the pads. The duty simulated was from 100 km/h to zero under constant deceleration. The results did show stresses above the Ultimate Tensile Strength (UTS) and yield strength of the disc material but this is expected for a

linear elastic analysis. In addition, the temperature results, peaking at 144°C, indicated much lower values than the ones quoted to be the thresholds for the creation of martensite in ferrous materials (750°C or more). It is the author's view that Gao et al. should have modelled the TEI using more detailed material data as it leads to hot spots and then disc brake material failure.

For the purpose of identifying the aspects leading to achieve optimal design in brake discs, Cho and Cho conducted a study covering the areas of thermomechanical aspects, TEI and also Noise Vibration and Harshness (NVH) by performing a modal analysis on three disc designs (Cho & Cho, 2008). They used the FE method to study the thermomechanical phenomena on disc and pads for a number of braking scenarios and assumed the use of one-pot and two-pot calipers. Furthermore, the model the TEI regime with the use of a custom made algorithm to generate hot spots and compared the numerical predictions against experimental results from an IR camera. In the last step of their investigation, they predicted the natural modes and frequencies. They concluded that:

- The shape and the stiffness of the brake disc has a more significant role than the pads' shape or the pressurisation type in the structural behaviour of the braking system,
- The thermal performance of a brake disc with thinner friction faces than the pads was improved as shown in TEI analyses,
- The single-piston caliper had better performance with regards to TEI than the two-piston one,
- However, the two-piston caliper had better braking stability due to a more uniform pressure distribution than the single-piston caliper,
- All these factors and their interaction have to be considered in order to reach the optimal design for brake discs.

According to Hwang et al., in order to perform a thermal analysis on a ventilated disc brake a 3D model is necessary (Hwang, et al., 2007). However, they realised that the computational power required and storage space is significant. As a result, they presented a technique of creating a 2D axisymmetric model of a ventilated disc to predict temperature and coning. Attention was paid to the boundary conditions of the model, more specifically to the variable heat flux input, the calculation of the convective heat transfer coefficient and also the area where the h_{Dconv} was applied. Linear material properties were used and four different

brake scenarios were modelled. The predicted temperature and coning values from the 2D axisymmetric model were compared against the equivalent predicted values from a 3D model that indicated reasonable matching. Thereafter the predicted temperatures from the 2D model were validated against temperature results from physical tests.

Hwang et al. continued the work documented in (Hwang, et al., 2007) by investigating ways to produce simplified FE models simulating the thermal behaviour of ventilated disc rotors. They conducted a simulation for a ventilated disc when subjected to repetitive brake applications on a downhill road (Hwang, et al., 2008). Instead of making a 3D model of the exact geometry of the analysed ventilated disc, they created only a solid ring without the top-hat section or the undercut but kept the actual Outer Diameter (OD), Inner Diameter (ID) and thickness.

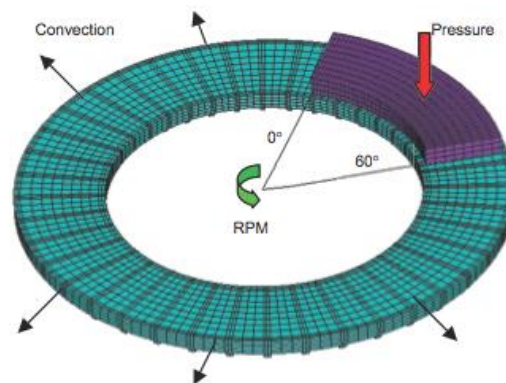


Figure 2-10 P. Hwang Simplified 3D FE Model (Hwang, et al., 2008)

Thereafter, they concentrated on calculating analytically the values of h_{conv} for both the solid areas of the disc (e.g. friction faces) and inside the vanes. Then, they applied the two different h_{conv} values (which were also time dependent) to appropriate areas of the solid ring. Finally, they included pads rubbing against the discs' friction faces, contact between pads and disc. The material properties used were not temperature dependent and the boundary condition to rotate the disc was angular velocity applied at a node on the centre of the solid ring. The predicted temperature values correlated well with the results from a test bench.

Dufrenoy and Brunel highlighted the effects that the hot spots and hot bands can have on the brake discs; more specifically the impact on performance, fatigue and NVH. This led the authors to classify the effects into five categories (Dufrenoy & Brunel, 2008):

- Asperity type
- Gradients on hot bands
- Hot Bands
- Macroscopic hot spots (MHS)
- Regional hot sport

They stated that the most critical category of thermal gradients in terms of crack initiation and damage to the brake rotor is the macroscopic hot spot. Thereafter, the writers developed numerical macroscopic models aiming to understand and explain the origin of each category. These models gave good results when compared to physical tests and/or results published on the literature. Finally, they found that the numerical models could assist the brake designs to perform design alterations in the braking system components and cure the phenomena of hot bands and hot spots.

Zhang et al. analysed the performance of a ventilated brake disc with inboard and outboard friction faces of different thicknesses under thermomechanical loading (Zhang, et al., 2010). The duty modelled was a stop from 130 km/h to 0 km/h at constant deceleration. They developed a 3D linear elastic model including both a brake disc and two brake pads. Heat transfer with conduction between the disc and pads was simulated along with heat convection. It has to be mentioned that the authors applied the convective heat transfer on a number of disc and pad areas including the pad areas rubbing against the disc something that is not realistic. Also, the assumption that uniform pressure is applied on the pads' backplates was non-realistic. Their results showed lower bulk temperature rise, higher stress concentrations and smaller axial deformations on the outboard disc friction surface compared to the inboard because the outboard face was the thickest of the two. These results were validated with physical test.

Belhocine and Bouchetara conducted a study on thermomechanical aspects on automotive brake discs. They used Computational Fluid Dynamics (CFD) to

acquire the heat transfer coefficients and then they applied the values to FE sequentially coupled models simulating an emergency stop for a solid and a ventilated disc. The properties used were linear elastic apart from the thermal conductivity and specific heat, which were temperature dependent (Belhocine & Bouchetara, 2013).

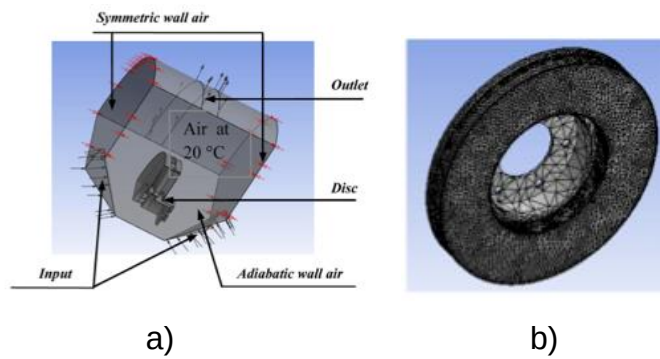


Figure 2-11 a) CFD Model, b) FE Model (Belhocine & Bouchetara, 2013)

They conducted analyses, with uniformly applied heat flux, predicting the temperatures and stresses on the discs. Furthermore, they used three different grades of cast iron to see their influence in the results. Finally, they developed FE models where pads were pressing against rotating discs aiming to predict thermal stresses and thermal deformations and the discs and pressure distributions on the pads' friction surfaces.

Kim et al. presented a design approach to create a lightweight brake rotor for automotive applications (Kim, et al., 2014). The top hat of the disc was made out of aluminium and the friction ring out of cast iron. Two ways were developed to connect the friction ring with the top hat (Figure 2-12): in the first, there was a lip in the friction ring that was inserted in the top hat. In the second, the friction ring was connected to the top hat with steel bolts.

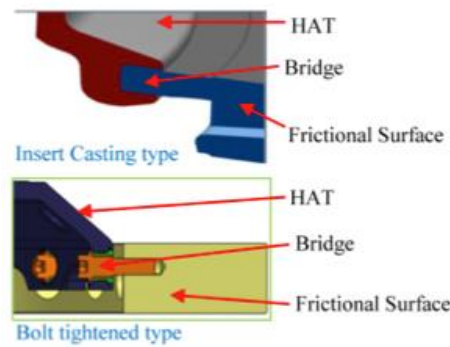


Figure 2-12 Ways to Connect Friction Ring and Top Hat (Kim, et al., 2014)

FE simulations were conducted that proved the feasibility of each of the two connecting ways. In Figure 2-13 the von Mises stress distributions are presented for the design using bolts to connect the friction ring and the top hat.

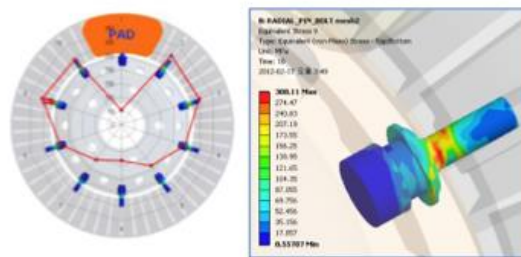


Figure 2-13 von Mises Stress Variations on the Radial Bolts and Stress Distributions on the Bolts with the Highest Stress Values (Kim, et al., 2014)

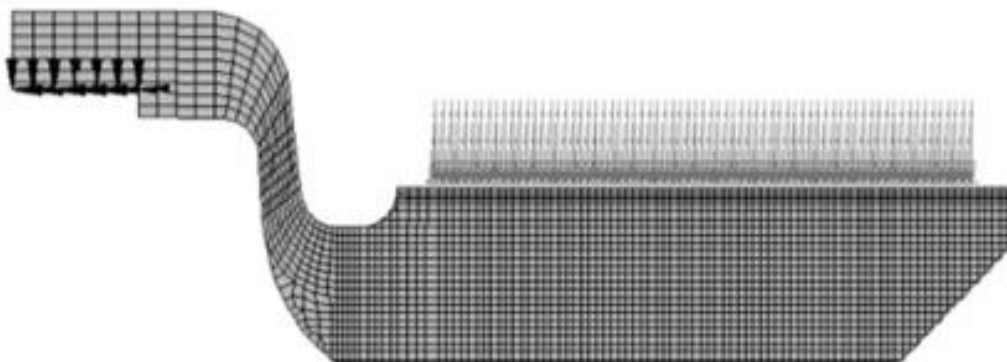
Finally, dynamometer tests were performed to investigate the durability of the concepts. The results showed crack on the friction ring for the disc prototype using the first connecting technique. The issue was solved by increasing the “cast bridges” that held the two parts of the disc together during the brake applications. Other aspects, such as the friction coefficient and the temperature profile during thermal capacity test, were similar to the monoblock cast iron discs. The weight savings achieved were between 15% and 20%.

A group of publications was identified containing valuable information of the analysis and design methodologies for railway discs. Although some boundary conditions of railway discs are sometimes different to the ones in automotive discs, the analysis steps described are universally applicable for brake disc analysis. The study by Tirovic and Ali is the first part of three publications that

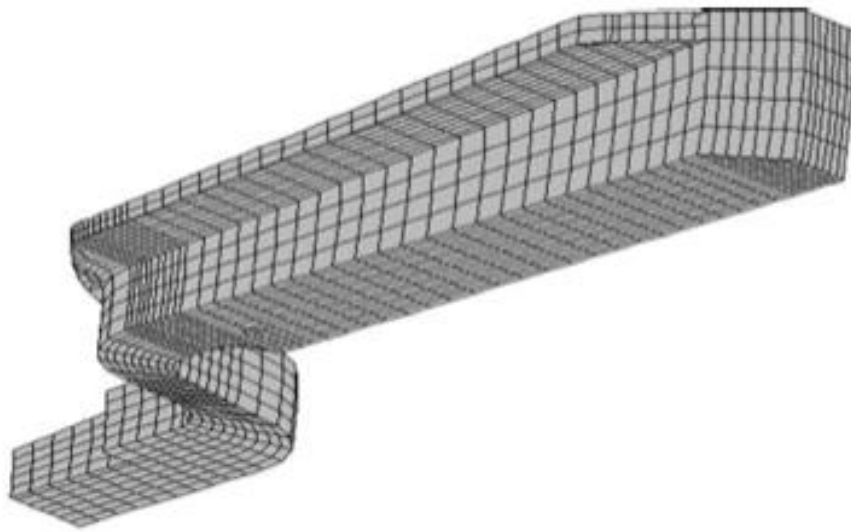
described a design methodology of two similar wheel hub-mounted discs for railway applications (Tirovic & Ali, 2001). From the initial stage of the design the FEA method was used, but the analyses were non-linear elastic and considered only the bulk thermal effects. Due to the nature of the particular discs, the thermal load was applied only on one side. Designs of such discs are very sensitive to small geometric alterations. This is not the case for the discs used in the present thesis as symmetrical loading was assumed, meaning that the thermal loads are applied on both sides of their friction faces.

Tirovic and Sarwar presented the second stage of the design process for the two aforementioned railway disc brakes. In this stage the FE models included macro-thermal effects, and more specifically the TEI. Additionally, disc plasticity was also modelled. The authors noted that for accurate modelling of TEI and disc plasticity good material data is required, and more specifically, the stress-strain relationship and the coefficient of thermal expansion (α) (Tirovic & Sarwar, 2004).

The TEI and the disc plasticity were modelled separately offering a simplified but time accurate procedure. The disc deflections were calculated using a variety of FE models and analysed including 2-D axisymmetric sequentially coupled models, 2-D axisymmetric fully coupled models (Figure 2-14 – a) and 3-D analyses (Figure 2-14 – b). The resulting deflection results were similar in all cases.



a) 2D Axisymmetric Model



b) 3D Model of the Disc Segment

Figure 2-14 M. Tirovic and G. A. Sarwar FE Models (Tirovic & Sarwar, 2004)

In the final part of the design work for the two railway discs (Tirovic & Sarwar, 2004) the verification procedure was presented. Good correlation between the experimental results and the numerical analyses described in the first part (Tirovic & Ali, 2001) and the second (Tirovic & Sarwar, 2004) was achieved. The authors noted the lack of experimental data, for example relating to actual duty, discs' permanent stresses, permanent strains, permanent coning and pads used set limitations in their work. Nevertheless, the manual optimisation process that was presented at the end resulted in a successful new disc design. For the optimisation, they modified the parameters P1-P11 accordingly and kept fixed the boxed dimensions due to space constraints (Figure 2-15).

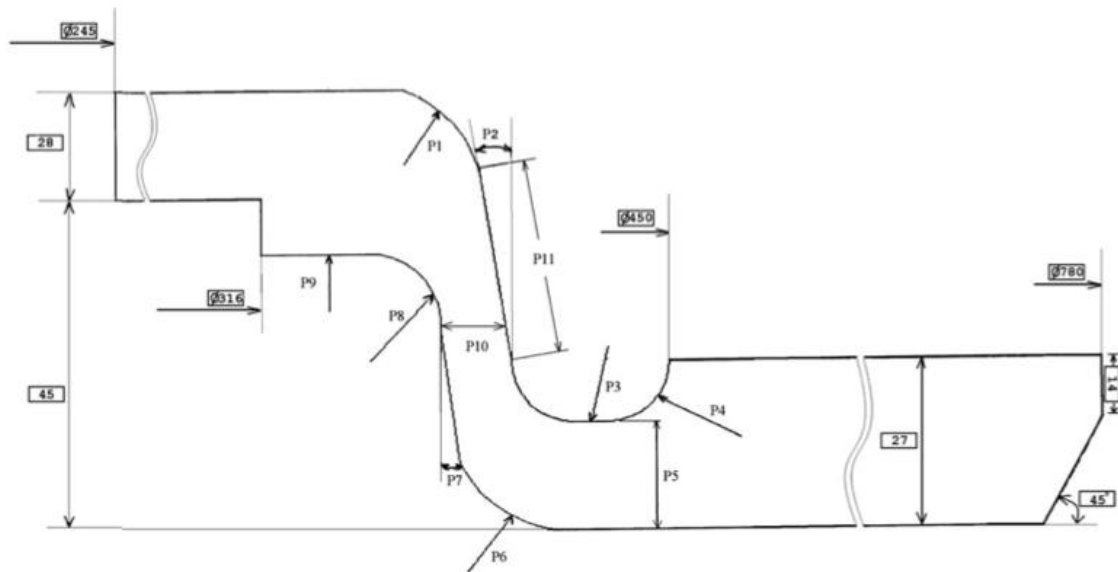


Figure 2-15 Optimisation Parameters (Tirovic & Sarwar, 2004)

A valuable conclusion for the whole design procedure was that by only using non-linear elastic analyses in the beginning to model the bulk thermal effects the global shape optimisation is feasible. More sophisticated analyses are only required to accurately define the limits of the design (Tirovic, 1998).

2.3 Cast Iron Attributes and Properties

In this Section of the literature review, properties, attributes and behaviour of cast iron are presented. The high carbon grey cast iron offers high thermal conductivity, high damping capacity, low thermal expansion and low modulus of elasticity. Due to these attributes it is the preferred material for braking components, such as disc brakes and drums for automotive applications. British Leyland was the first manufacturer to introduce it in 1961 followed by DAF for commercial vehicle in 1965 to prevent thermal cracking. In the 1970s, it was introduced in the United States of America mainly to solve squealing and judder problems. The high carbon grey cast iron with UTS 150 N/mm^2 generally contains 3.6% to 4.0% C and 1.8% to 2.4% Si. These proportions change according to the needs of each vehicle manufacturer. The report contains temperature dependent material properties for cast irons with UTS of 150 N/mm^2 , 175 N/mm^2 and 200 N/mm^2 .

Palmer studied the physical properties of cast iron grades for engineering applications, stated that when selecting material for components in high-temperature applications, the engineer should consider primarily two principle properties (Palmer, 1986):

- The iron to be stable with no changes in its microstructure or undue oxidation which can change the component's dimensions at the design temperatures,
- The maximum stress to be withstood by the iron at the design temperature before failure has to be known, as well as any dimensional instability caused by creep.

Thereafter, he presented the material properties for a number of grades of cast iron at the temperature range between 20°C and 500°C.

With regards purely to automotive brake disc made out of cast iron, Chatterley and Macnaughtan conducted a survey to find the attributes of the discs used at the time in European cars (Chatterley & Macnaughtan, 1999). They used a selection of forty-nine ventilated discs mounted on the front axels and performed tests to identify their chemical composition. They found that the majority of the discs were made of medium carbon cast irons while very few were made out of low carbon cast irons and even fewer out of high carbon cast irons equipped in expensive automobiles.

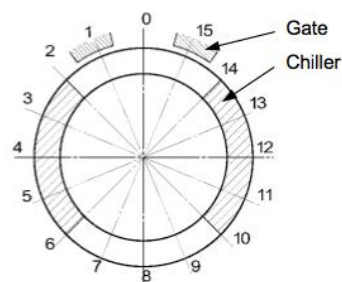
Additionally, they manufactured four identical discs made out of four different cast iron alloys and test them on an inertia dynamometer. The results showed that the disc made out of high carbon cast iron with additions of Cu performed superior to the others. They also noted that when they used organic non-asbestos pads in the tests, the wear on all discs was very high at a point that no performance comparison could be made. The comparisons were possible only with the use of semi-metallic pads.

Furthermore, the possibility to introduce metal matrix composites (MMC) as materials for brake rotors was mentioned. Compared to cast iron grades, the improved thermal diffusivity, thermal conductivity and weight reductions offered by the MMC materials are counterbalanced by the difficulty in manufacturing,

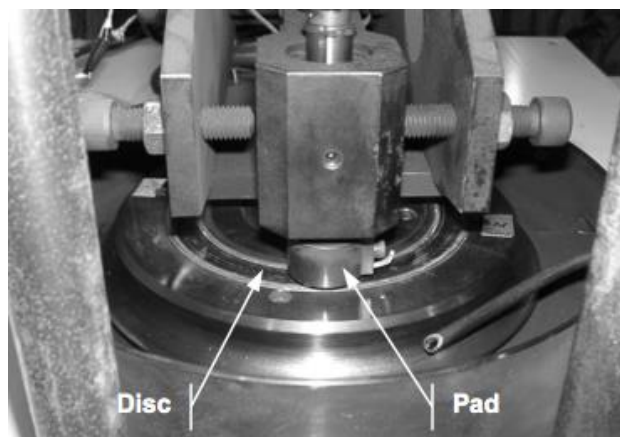
lower maximum operating temperature, need of cooling ducts and cost making them an unfavourable choice as a brake rotor material at that time.

Finally, the authors stated that the future of brake disc material of choice would be the cast iron grades with high carbon content due to their optimal thermal properties, castability, integrity, good resistance to distortion and thermal cracks and better damping properties when compared against the medium carbon grades.

Okamura and Imasaki studied the impact of disc thickness variation (DTV) and material properties on the local wear of a brake disc. They created surrogate discs with intentionally non-uniform properties (Figure 2-16 – a) and tested them on a chip-on-disc friction tester (Figure 2-16 – b) (Okamura & Imasaki, 2003).



a) Angular Positions Where Material Properties Were Examined



b) Chip-on-Disc Set up

Figure 2-16 T. Experimental Set up (Okamura & Imasaki, 2003)

With regards to material properties, they found that cast iron grades with high carbon contents offer less local and total wear. It was also noticed that the organic non-asbestos pads tend to cause larger Disc Thickness Variations (DTV) to the disc rotors than the low-steel pads also used in the test. These findings are in agreement with previous work published on the subject (Chatterley & Macnaughtan, 1999).

In engineering applications, where conventional materials like grey iron cannot cope with due to durability limits and the use of materials, such as ductile iron is not necessary due to excess of requirement, Dawson and Schroeder introduced the Compacted Graphite Iron (CGI) as the material appropriate for use (Dawson & Schroeder, 2004). The authors discussed its microstructure, properties, machinability, suitability for applications, such as train brake discs, its weight advantage over grey iron, strength and also some design considerations. All these make CGI the ideal material for components subjected to combined mechanical and thermal loads. They also cited the CGI's advantages over the conventional grey cast iron:

- Reduced wall thickness at similar operating loads
- Increased operating loads
- Reduced safety factors due to less variation in as-cast properties
- Reduced brittle failure due to higher ductility
- Reduced hot cracking
- Higher strength without resorting to alloying
- Shorter thread engagement depth and therefore shorter bolts.

In (Gusser, et al., 2005), the authors conducted a physical test to measure the thermal conductivity of two grey iron grades (GI 250 CuSnCr and GI 300 CuSnCrMo) and compare it against the thermal conductivity of two types of CGI (CGI 350 and CGI 450) at a range of temperatures between 0°C and 400°C. They noted that the thermal conductivity of both the grey irons tested to be higher than the thermal conductivity of the CGI specimens. This was due to the graphite form of the grey iron. However, the values of thermal conductivity in the grey iron specimens were reducing with the temperature increase, whilst the values of CGI thermal conductivity remained stable.

Mace et al. discussed the phenomenon that bedded cast iron brake discs have higher resistance to crack failures (Figure 2-17). They mentioned that this is a phenomenon that has been noticed only empirically but no explanation has been published in the literature (Mace, et al., 2006).



Figure 2-17 Left: Sudden Crack in a Non-bedded Disc Early in its Life, Right: Acceptable Microcracks During a Discs' Service Life (Mace, et al., 2006)

Their investigation included a series of bedding tests of different cast iron brake discs used in racing cars ranging from light F3000 single-seater car to heavy Nascar car. They used an inertia dynamometer and recreated the bedding process that a race car driver follows for new brake discs and then simulations of entire races. Thereafter, the discs' microstructure was investigated visually and with scanning techniques.

During the bedding tests, the authors noticed that the local disc temperatures were high enough to exceed the phase transformation temperature of cast iron. In the analyses after the tests, they found that the flake structure of carbon was unaffected by the bedding treatment. However, the pearlite matrix broke down due to high temperature on the friction surfaces. During this breakdown, the carbide particles around the graphite flakes agglomerated leaving large areas of ductile ferrite around the graphite/carbide couples. This phenomenon made the discs more ductile and increased their resistance to crack propagation.

This finding of Mace et al. is interesting, as within the present thesis the structural strength of a disc rotor was investigated and it is known that the brittle nature of cast iron sets the limit on the mechanical loading the material can withstand. A more ductile behaviour achieved by the method described would result in a material that will be able to withstand higher loading and elongate more before fracture takes place.

Okamura and Imasaki studied ways of reducing the weight of the brake disc in an effort towards more efficient and eco-friendly vehicles (Okamura & Imasaki, 2009). After discussing some alternatives to grey cast iron, such as Al-MMC, silicon-carbide composites and thinner steel rotors similar to ones found in motorcycles they stated that cast iron will be the material of choice for the present and the medium term future, mainly due to its price and thermomechanical properties, and the research should be focused on optimising it. The focus of their study was on manufacturing discs with improved material homogeneity which increased the thermal strength of the component. They concluded that with improved thermal strength and the use of optimisation techniques, weight reductions are possible.

As stated by the authors in (Han, et al., 2011), the advantages of cast iron for use in automotive applications lay on its self-lubricating characteristics, the casting design flexibility that offers and also the significant cost saving in its production. In ball-on-disc conducted tests they found that crack initiation might occur due to graphite flakes creating stress concentration points when the material is loaded mechanically.

According to Vadiraj and Tiwari high carbon grey cast iron offers good thermal properties, castability and damping for NHV problems (Vadiraj & Tiwari, 2013). On the other hand, the increase in carbon equivalent produces more graphite flakes that can act as weak points on the rotors reducing their strength and at the same time increasing the wear. In order to overcome these issues, cast iron alloys have been created with addition of elements like Al, Si and Mn that can modify the microstructure of the material and its properties. The authors analysed three alloyed specimens with inclusions of Ni, Mo, Cu and compared them to a fourth alloy specimen with Al, something that had not been documented in the literature at that point. They found that the wear rates for the specimens with Ni and Mo were worse to the Al specimen something which they describe as “very peculiar trend”. They also found strange that the specimen with Al and 3% Ni was harder than the specimen with 2% Al. Finally, they concluded that the UTS

increases with the decrease of graphite flake content and wear increases with increase in hardness.

Grabiec investigated the influence of the grey cast iron composition in the friction coefficient and wear (Grabiec, 2014). He created specimens from twelve discs, analysed their microstructure and tested their hardness but not their tensile strength. Thereafter, by using a dynamometer he subjected each of the twelve brake discs to the braking scenario where a country route was simulated in order to see how each disc performed in terms of friction coefficient and wear. Although pads from the same batch were used for all the tests, the author noticed that the wear levels differ from disc to disc to a percentage as high as 100%. Similarly, significant differences were found in the friction coefficient levels as a result of differences in graphite levels. Other findings included the absence of carbon content influence and surface hardness in wear performance, the materials with low wear rate also had low friction coefficient, the materials with higher amount of graphite B showed higher wear resistance but were more prone to micro crack propagation.

Vadiraj et al. focused their work on comparing the performance of grey cast iron alloys with the addition of Mo (Molybdenum) against grades of cast iron with Al (Vadiraj, et al., 2015). They prepared four cast iron alloy specimens one with 1% Mo and three with Al ranging between 0.5% and 1%. They performed qualitative analyses of their macrostructure, they measured their hardness, their tensile strength, coefficient of friction and their wear properties. The results showed that the Mo alloy had 30% higher tensile strength, had higher hardness, its coefficient of friction was between 0.4 and 0.5 and 6 to 7 times improved wear behaviour.

2.4 Automotive Wind Tunnels

In Chapter 7 the design and development of a crossflow brake cooling rig is presented. Due to lack of previous experience and in order to understand basic principles for the development of the rig, this part of the literature survey was dedicated in reviewing publications of automotive wind tunnels. Although the developed crossflow rig varies from the wind tunnels identified in the open literature, with the only one close in terms of shape was the MIRA wind tunnel

(Figure 2-18), appreciation of the wind tunnels' basic characteristics and airflow patterns was gained.

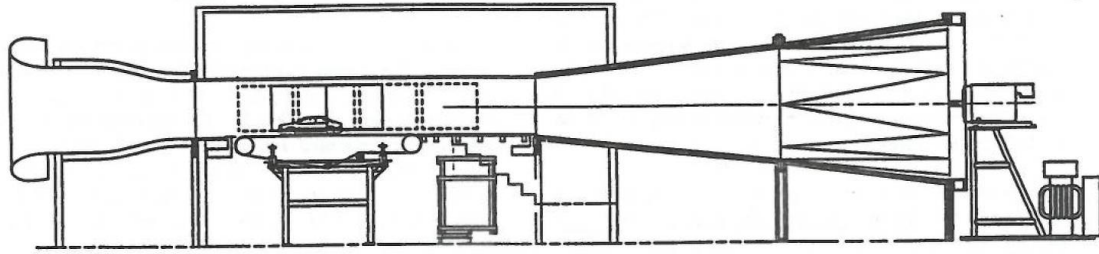


Figure 2-18 The MIRA Wind Tunnel (Carr & Eckert, 1994)

According to Tadakuma et al., modern wind tunnels should (Tadakuma, et al., 2014):

- Be capable of simulating road flows enabling highly accurate and repeatable measurements of aerodynamic characteristics,
- Suppress pulsations in order to evaluate unsteady flows around the vehicle,
- Be capable to simulate flow fields similar to the ones found in on-road environments
- Have low noise levels during operation for evaluation noise generated by the vehicles tested,
- Incorporate efficient measurement system capable of capturing the flow fields around the vehicle in full extend,
- Enable the cooperation between engineers and designers that aim to develop smooth vehicle aerodynamic characteristics.

Garry et al. described the development of a 3/8th scale wind tunnel (Garry, et al., 1994). The commission process is discussed as well as aspects, such as the boundary layer thickness, the acoustic performance and flow angularity. Thereafter, the performance of the semi-open jet configuration and the effectiveness the slotted-wall (closed jet configuration) during piloting tests were investigated. Having examined how all the systems operated, the authors concluded that the 3/8th tunnel functions adequately and can be the basis of a full-scale wind tunnel.

Some automotive wind tunnels incorporate the conventional stationary ground-plane, for positioning the tested models or vehicles. However, other automotive wind tunnels however, use moving ground-planes or the technique of boundary layer suction to simulate the road rolling under the vehicles when they are moving (Figure 2-19).

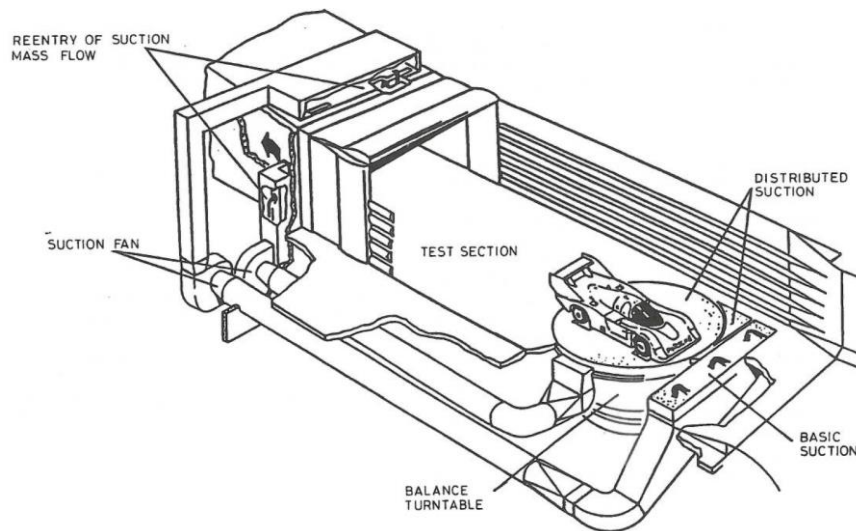


Figure 2-19 Suction System (Carr & Eckert, 1994)

Carr and Eckert wanted to investigate how the aerodynamic drag and lift are affected by each of these three installations (Carr & Eckert, 1994). They found that the moving ground considerably increases the downforce and the drag, but reduces the lift on both the front and rear axles. Similarly, the distributed suction reduced the lift especially at the front axles but its impact was not as great as the moving ground-plate's.

Künstner et al. presented the open-jet automotive wind tunnel of Stuttgart University, its arrangement and its specifications when first started to operate in 1988 (Künstner, et al., 1995). However, because at that time the noise coming from the engine, tyres and exhausts of the automobiles was continuously reducing the focus of the automotive manufacturers was on the noise levels coming from the aerodynamics of the cars. Due to this reason the authors of the paper presented the conducted modifications in the wind tunnel in order to reduce the noise levels produced during its operation. By installing U-bent silencers for the upstream and downstream and placing sound absorbing lining in the plenum

chamber, the noise levels during the operation of the tunnel empty were reduced by 20 dB with small impact on the achieved top air speed.

Rout described the problem of simulating the airflow around a vehicle during an outdoor driving test when conducting tests in a wind tunnel environment (Rout, 1998). The fixed dimensions of the wind tunnels (either open jet or close jet) influence the airflow impacting the air velocity distributions on the frontal area of the tested vehicles. In order to overcome this situation, the author created a technique to correlate the air velocities achieved in real world testing conditions and the air velocities in wind channels. He then derived a mathematical relationship that was implemented in the speed control software of the wind tunnels and acted as correction factor in order to achieve the desired airflow. The main findings of this research were that the blockage factor is function of the frontal area and of the car and the distance from the nozzle exit, non-dimensionalised single relationships can be developed to represent the blockage effects in the wind tunnels for any vehicle and the open and closed jet types wind tunnels have different blockage factors (non-linear of the closed and linear for the open). Finally, it is supported by Rout that the correlation developed depends on functional characteristics, such as the boundary layer growth, tunnel test-section size and data used for the correlation making the process unique for each vehicle type and tunnel type.

The use of slotted walls to produce a more realistic flow in the test section is a common approach if an increase of the section's dimensions is not possible (Flay & Vuletich, 1995). The authors described the modification of a wind tunnel aiming to accommodate larger scale models of racing yachts. Interestingly, they mentioned the use of slotted walls in the test section in order to achieve more realistic airflow in their open jet tunnel. That seems to be applicable for wind tunnels designated for automotive purposes.

The work of Arnette et al. was an effort to eliminate the low frequency pressure pulsations and achieve uniform static pressure distributions in the test section of an automotive open jet wind tunnel (Arnette, et al., 1999). The low-pressure fluctuations are undesirable because they can lead to unstable air velocity

conditions. The physical mechanism that initiates the low-frequency fluctuations was not well understood at that time, an aspect that served as an extra motivation for their research. On the other hand, adverse pressure gradients in the test section result into artificially low drag measurements. In order to understand and eliminate these phenomena, the authors investigated the effect of key wind tunnel components, such as the length of the open jet, the collector and plenum geometries and the collector throat area. After conducting experimental measurements into two existing wind tunnels and by creating models to predict the pressure frequencies, they found that the plenum fluctuation levels depend on the length of the open jet, the collector geometry and the plenum dimensions. In addition, it was noted that the collector configuration had no influence in the axial static pressure, but the open jet length had. Overall, they concluded that increasing the open jet length highlights a trade-of as it increases the low-pressure fluctuations but achieves a more uniform static axial pressure distribution.

The aeroacoustic wind tunnel establishment of Audi AG which had an open as well as a close jet type wind tunnel was presented in (Wickern & Lindener, 2000). Figure 2-20 illustrates a schematic of the entire establishment. The area with the number one is the open jet wind tunnel and the area marked with number four is the close jet tunnel.

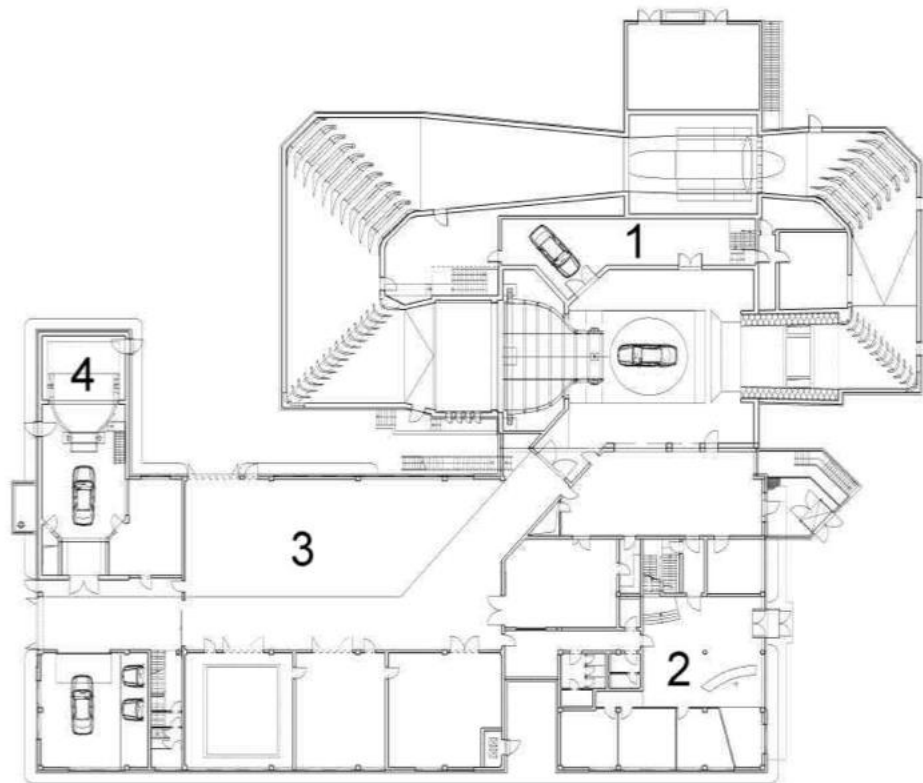


Figure 2-20 Audi AG's Aeroacoustic Establishment (Wickern & Lindener, 2000)

There were two main considerations during the development of the tunnel: to effectively simulate the effect of road and wheels rotation and to achieve quiet wind tunnel operation in order to be able to conduct aeroacoustic measurements. Both these demands were addressed. The first by using boundary layer suction, a moving belt between the wheels, belt wheel drives for each wheel and a lifting device to examine the underbody aerodynamic effects. The second by a specially designed fan, acoustic treatment in the plenum chamber walls and the use of an active resonance control system that eliminated the resonance of all wind velocities resulting in 60 dB at 160 km/h. Calibration of the tunnel as well as first test results were also discussed.

The Active Resonance Control (ARC) System of the Audi's wind tunnel was presented by the authors in (Wickern, et al., 2000). Motivated by the fact that classical acoustic wall absorption linings cannot address low-frequency pressure fluctuations, they created an active system to eliminate them (Figure 2-21).

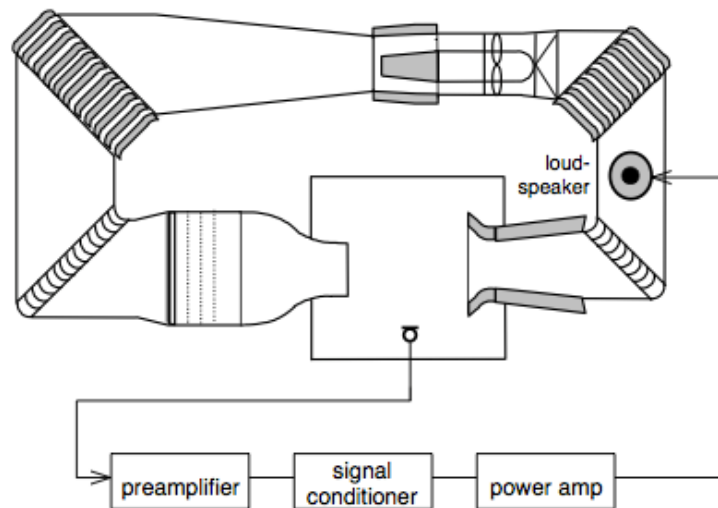


Figure 2-21 The ARC System (Wickern, et al., 2000)

The system comprised of a microphone to detect the pressure pulsation in the plenum, a loudspeaker on the tunnel wall and a time delay algorithm to adjust the phase relation between the input signal of the microphone and the output of the loudspeaker. As a result, the pressure fluctuations caused by the speaker where in opposite phase to the ones generated by the airflow, and as a result they cancelled each other.

Kim et al. demonstrated the Goettigen type full-scale aeroacoustic wind tunnel of Hyundai Motor Company (Kim, et al., 2001). Capable to reach air velocities of up to 200 km/h the tunnel was equipped with two turntables in the test section; one purely for the aerodynamic investigations and the other could be used as dynamometer for two-wheel drive cars.

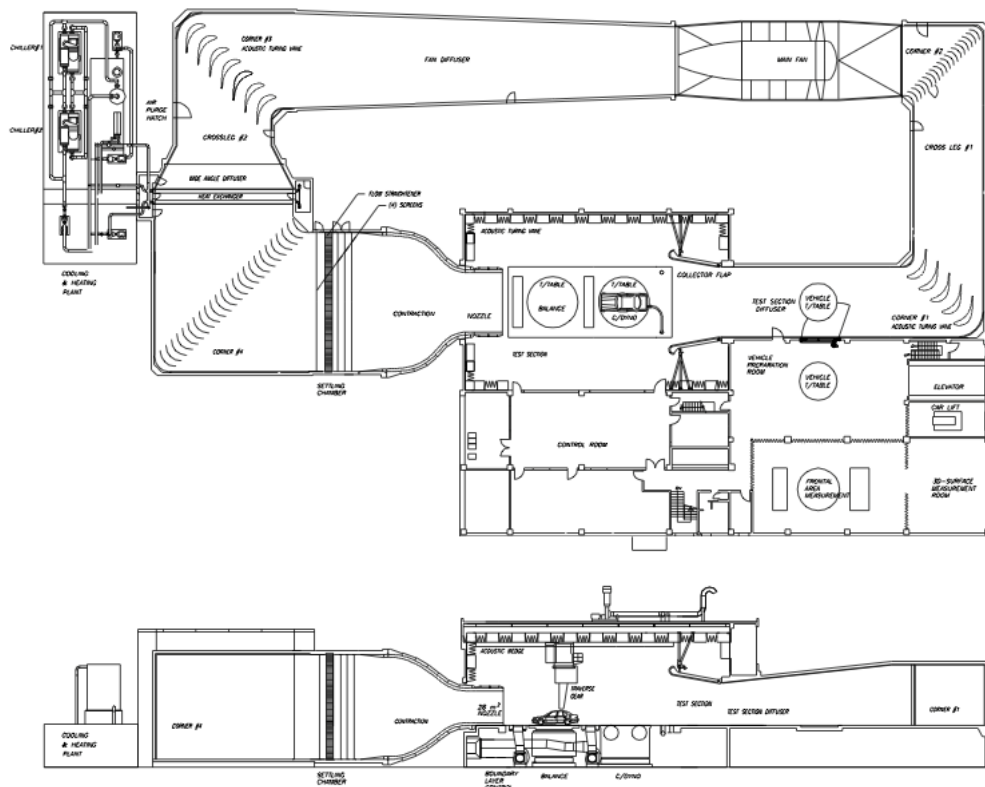


Figure 2-22 Lay Out of the Hyundai Aeroacoustic Establishment (Kim, et al., 2001)

The second turntable could also pivot around its yaw axle in order to simulate crosswind conditions. In front of both the turntables, boundary layer suctions were installed to minimise the boundary layer during the tests. Interestingly, it was decided for the test section to be of a semi-open design rather than having slotted walls. The decision was based on the engineers' past experience that with slotted walls there was a compromise in wind noise measurements and airflow visualisation around the vehicles.

Ubertini and Desideri described the experimental procedure to measure the aerodynamic attributes of a maxi-scooter (500cc) in an open jet wind tunnel (Ubertini & Desideri, 2002). Apart from the usual major components found in wind tunnels of this type like the nozzle, the collector, the turning vanes, the diffuser after the fan and the honeycomb that reduce swirl and lateral mean velocity variations, a specially developed balancing base was used to keep the scooter

and the dummy rider in position and also measure the forces and the moments in the three directions of the coordinate system.

Wiedermann and Potthoff presented a turntable system aiming to represent as closely as possible the effect of the road underneath the vehicles when tested in wind tunnels (Wiedermann & Potthoff, 2003). It incorporates 5 moving belts (Figure 2-23); one in the middle of the systems with width of 1.000/1.105/1.210 m depending on the vehicle tested. The other four belts are in contact with the wheels of the cars with width of 280/360/410 mm. All the belts have a top speed of 70 m/s (252 km/h). In order to eliminate the boundary layer from the nozzle a combination of suction before the belts and tangential air blowing is used. The experimental results, when vehicles were tested in these turntable, indicated decrease in drag up to 0.020 and reduction in lift as well.

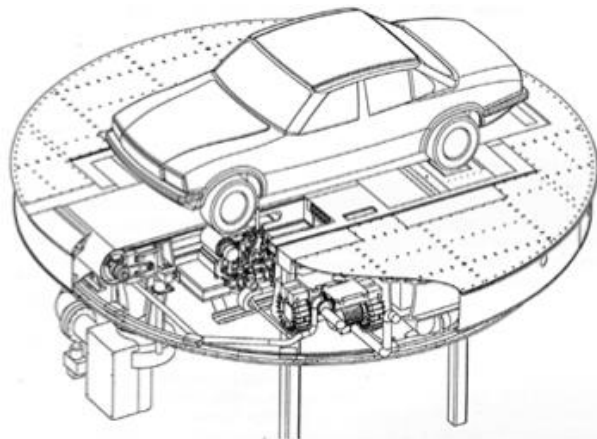


Figure 2-23 The 5-Belt Road Simulation System (Wiedermann & Potthoff, 2003)

The static pressure after the fan in the crossflow ring, presented in Section 8.2.3, was investigated as high values would suffocate the flow which will result in reduced air velocities in the working section (Nijhof & Wickern, 2003). The authors developed two methods to measure reference static pressures in automotive wind tunnels: the nozzle method and the plenum one. They found that the plenum method was less dependent on nozzle blockage and also less sensitive to the vehicle's position from the nozzle compared to the nozzle method. As a result, they stated that the plenum method should be the preferred method for practical measurements. They also categorised the sources of experimental

error into non-blockage and blockage related. The non-blockage errors can be eliminated by technical efforts, but the blockage related errors are more difficult to be dealt with.

The DaimlerChrysler aeroacoustic wind tunnel (Figure 2-24) was presented in (Walter, et al., 2003). Having as initial targets the excellent airflow, low background noise and to complete the work within the given budget, the authors describe the design methodology followed for each of the main tunnel components. For the test section, it was decided to increase the nozzle size instead of using slotted walls. The decision for this was based on studies that showed that slotted walls present little advantage over open jets for the aerodynamic boundary interference corrections when there is a considerable blockage of the flow due to testing of large vehicles.

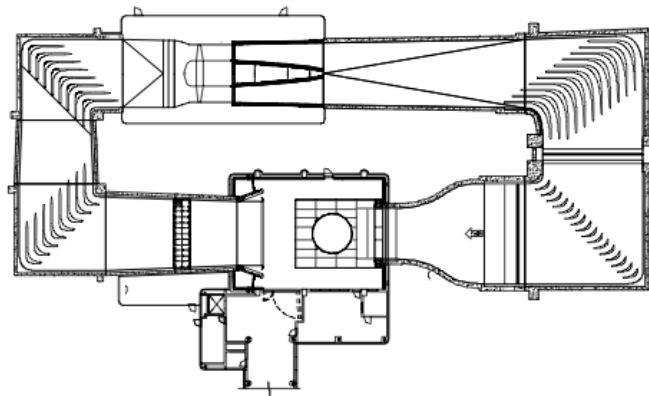


Figure 2-24 The DaimlerChrysler Wind Tunnel Schematic (Walter, et al., 2003)

Also, the strategy of the company at the time was towards cars with larger size like SUVs and minivans. The cost of the bigger nozzle was balanced by not having slotted walls. The interaction of the plenum, collector and open jet length were thoroughly studied so that low-frequency pressure fluctuations were avoided. With regards to the turning vane passages' design, CFD was used and then experiments in small-scale wind tunnel for cost reduction purposes. CFD studies were also performed for the acoustic design of the tunnel. Finally, the boundary layer was controlled by a suction and tangential blowing system.

The development and performance characteristics of the GIE S2A wind tunnel were discussed in (Waudby-Smith, et al., 2004). For the test section It was initially planned that the test section would incorporate a $\frac{3}{4}$ open jet and slotted walls for the benefits in the acoustic measurements due to the open jet and the flow quality, and stability thanks to the slotted walls. For financial reasons though, the installation of slotted walls did not happen but the design of the wind tunnel was such that slotted walls could be placed in the future. The turning vanes in corners 1, 2 and 3 were acoustically treated but not the vanes in corner 4 in order for the airflow not to be compromised (Figure 2-25).

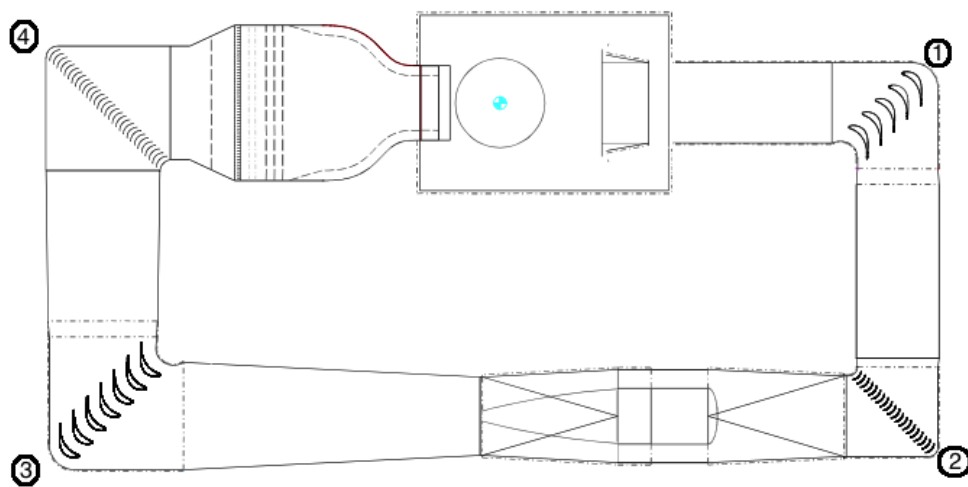


Figure 2-25 The GIE S2A Corner Lay Out (Waudby-Smith, et al., 2004)

The fan section of the tunnel was also acoustically treated so that the tunnel would be able to operate at night next to a residential area. The settling chamber was preceded by a wide-angle diffuser for rapid controlled diffusion of the airflow and had three fine mesh screens to straighten the flow. Future plans were made for the installation of a honeycomb flow straightener, as it was not financially possible at that time. Finally, the turntable of the wind tunnel included a balancing mechanism, a 5-belt rolling system with top speed of 200 km/h and a boundary layer control system with two suctions that could eliminate the boundary layer.

The study presented by Tortosa et al., was focused on investigating the feasibility of correlation between experiment data from aerodynamic tests conducted in three different wind tunnels; those of Chrysler, Ford and General Motors (GM)

(Tortosa, et al., 2008). Although there were considerable differences in the type of the tunnels (open-closed jet), the balance systems and the boundary layer control systems of the tunnels, it was found that there was a linear correlation between the results recorded.

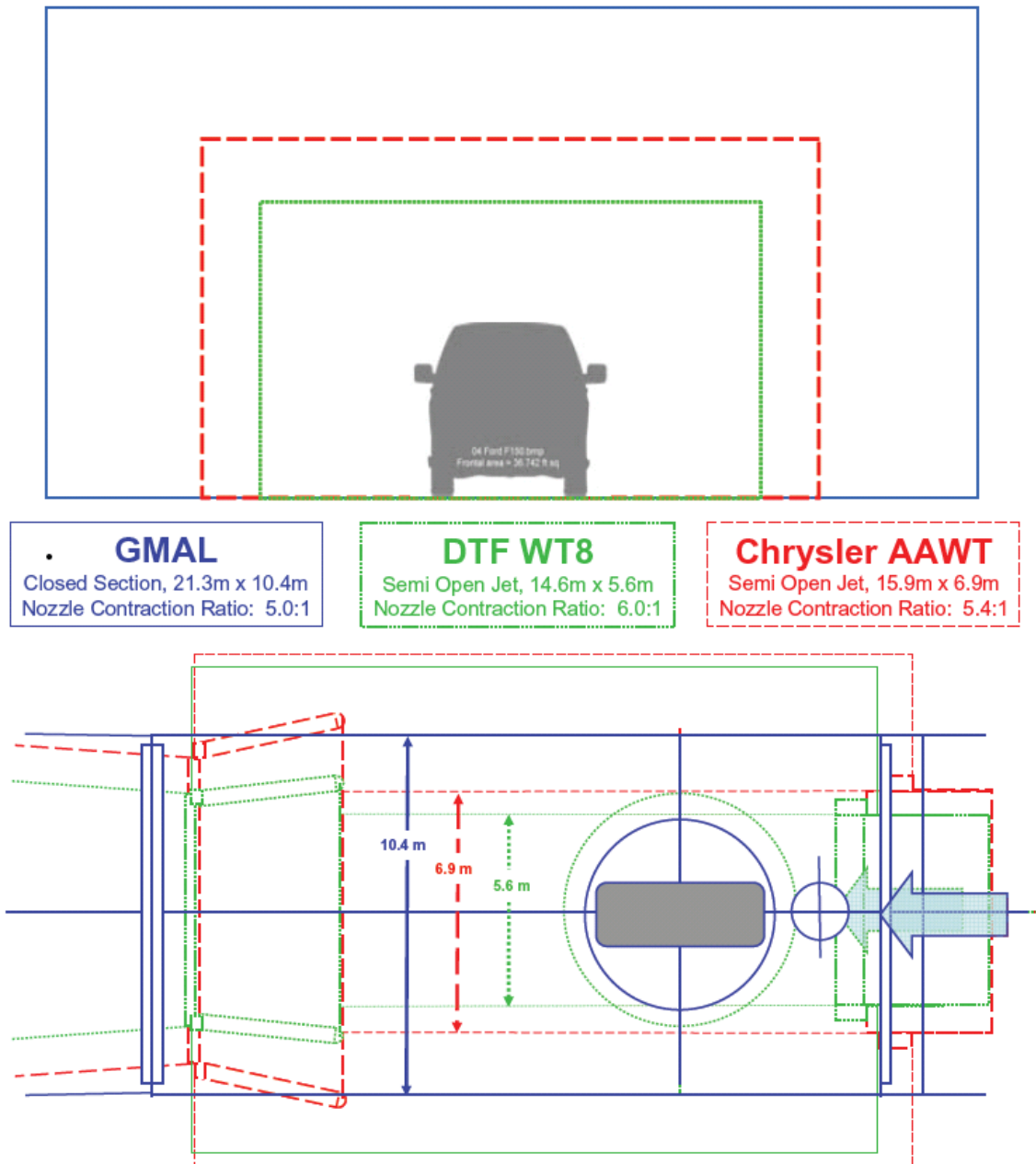


Figure 2-26 Major Dimensions of the GM, Ford and DaimlerChrysler Wind Tunnels (Tortosa, et al., 2008)

The study was beneficial to the participating automotive manufacturers with regards to the levels of confidence each one had on the measurements

conducted in their wind tunnels. It also enabled them to identify shortcomings in their testing and data recording methods.

Duell et al. described the aerodynamic test facilities of BMW which incorporated a full-scale wind tunnel and an aerodynamic wind tunnel laboratory (Duell, et al., 2010). Both the tunnels were open jet type and attention was given to simulate as closely as possible the “on-road” dynamic forces by having uniform static pressure distribution. In order to achieve this, the collector’s geometry was meticulously designed and also it had some degrees of freedom to adjust to the vehicle tested (Figure 2-27).

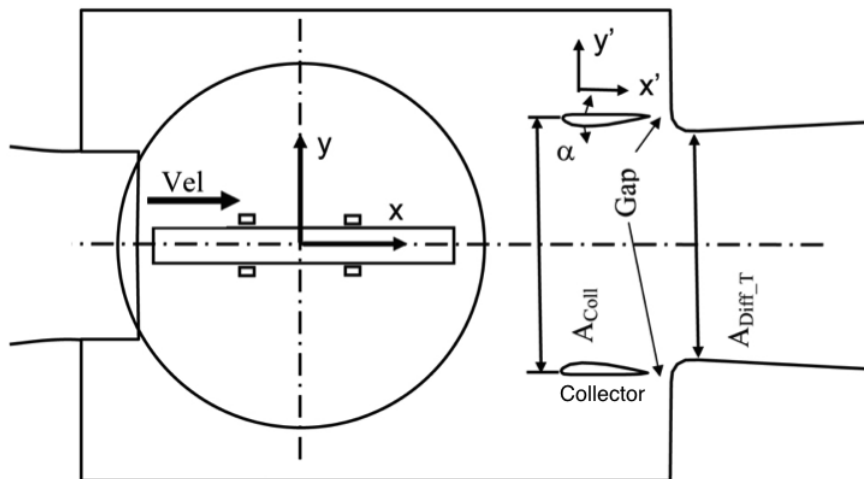


Figure 2-27 Collector in the BMW Full-Scale Wind Tunnel (Duell, et al., 2010)

Furthermore, the full-scale tunnel was equipped with a 5-belt system, whereas the aerodynamic laboratory had a single belt one. In both tunnels boundary layer reduction systems were installed and it was possible for the engineers to investigate the correlation of the two systems. The nozzle in the plenum was adjustable, able to achieve airflows of up to 300 km/h. All the turning vanes were acoustically treated with Foreign Object Debris (FOD) screen being placed before the vanes of corner two in order to protect the electric fan. The whole facility was developed with the aid of CFD and the utilisation of small scale wind tunnels.

The most recent publication on automotive wind tunnel was with regards to Toyota’s tunnel (Tadakuma, et al., 2014). The authors set the following performance targets for their wind tunnel:

- Substantial reduction in pulsation,
- Flat static pressure distribution in the axial direction and within each cross section,
- Wide and uniform velocity distribution,
- A wind tunnel able of evaluating both aerodynamic and wind noise characteristics.

Thereafter, they discussed how they achieved each of these targets. More specifically:

- Pulsation was reduced by creating openings above the wind tunnel and acoustic transition structure was applied on the collector,
- The nozzle outlet angle, the suction rate from the flow installing system and the nozzle/collector area ratio aided the creation of flat static pressure distribution,
- The use of 1:8 contraction ratio, honeycomb and mesh screens influenced the flow to have wide and uniform velocity distribution.

Application of acoustic absorption structures throughout the wind tunnel and the features presented in the bullet points above made the evaluation of both aerodynamic and wind noise characteristics possible and accurate.

2.5 Heat Dissipation From Brake Discs

During the brake application, as the brake pads are pressed against the brake discs to retard or slow down the vehicle, heat is generated in the interface of the two components as a result of friction. The heat must be dissipated efficiently (Voller, et al., 2003) for the sake of avoiding the overheating of the braking components, such as the friction pair, the electronic sensors or even the brake fluid. Failure to achieve this will result in a compromise of the braking system's performance, which in a worst-case scenario can put human lives at risk. At best case, it can influence negatively the drivability of the vehicle, its refinement levels and the feel transmitted to the driver, all of which are critical selling points for vehicles. In addition, extended exposure of the braking components in high temperature may lead to component failures that can even occur in components with very short service life (Mace, et al., 2006). Due to these risks, the thermal and the heat dissipation aspects have always been and are still the focal point of the braking system's design.

In general terms, heat can be dissipated by three modes: conduction, radiation and convection. In disc brakes, for the majority of operating conditions, convection is the dominant mode of dissipating heat. In the early studies of disc brake cooling, conduction and radiation were usually neglected in order to reduce the complexity of calculations. The derived results included some errors and inaccuracies, but they were considered acceptable because the braking systems were performing under lower temperatures. The modern braking systems operate under high temperatures, mainly due to the increase in vehicles' achievable speeds, and as a result accurate brake cooling prediction techniques had to be introduced.

For a single emergency stop brake application it can be assumed that no cooling takes place and the heat is only transferred within the disc and pads via conduction (Dike, 1974). For drag or repeated brake applications all the modes of heat dissipation must be included in the calculations.

To predict disc brake temperatures either analytical or numerical methods can be used. For the analytical methods, a number of simplifications have to be used to reduce the complexity of the problem. However, the analytical methods are useful for initial calculations in the early development stages. Examples of analytical methods can be found in (Newcomb & Spurr, 1967) and (Limpert, 2011). The numerical methods incorporate the use of FEA and CFD for brake temperature and brake cooling characteristics prediction. With FEA, the braking components can be modelled and their cooling performance can be approximated by inputting heat transfer coefficients values found either numerically, experimentally or from the literature. The CFD analysis models the fluid flow around the braking components and can predict the values of convective heat transfer. At this point it has to be mentioned that CFD analysis was not used in the current work.

2.5.1 Conductive Heat Dissipation

Heat transfer by conduction takes place when there is temperature difference within a single solid or between several solids in contact. Heat will be transferred from the areas of high temperature to the areas of lower temperature until thermal equilibrium is reached. In the case of braking systems, the investigations focus

on the conduction between the disc rotor and the neighbouring parts, i.e. like the pads, the hub, the pistons, the caliper, the brake fluid and even the wheels. In Figure 2-28 the temperature gradient (difference) between two solids in contact can be seen at “steady-state”.

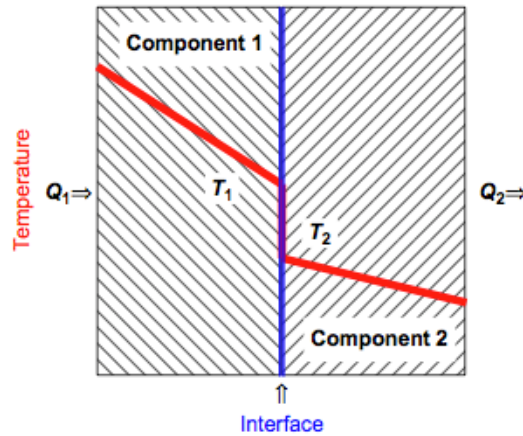


Figure 2-28 Conductive Thermal Gradient (Tirovic, 2011)

The Fourier's law for heat conduction defines the expression for the rate of conductive heat transfer as:

$$Q_{cond} = -\kappa A_{CS} \frac{dT}{dx} \quad (2-4)$$

**Convective
Heat
Transfer**

Where:

Q_{cond} : Conductive heat transfer,

κ : Conductivity,

A_{CS} : Cross sectional area,

dT : Temperature gradient,

dx : Thickness.

The difference in the heat gradients in Components 1 and 2 in Figure 2-28 is due to the different value of thermal conductivity (κ) that each component has.

Component 1 has lower thermal conductivity than component 2, resulting in lower heat transfer between its warm and cold areas and a steep temperature slop. The sudden drop of temperature in the interface between components 1 and 2 is the result of the TCR phenomenon. TCR (Ling & Pu, 1964) is the result of a non-perfect contact between two or more solids with the heat transfer taking place only in the asperities of the contacting surfaces reducing the effective A_{CS} considerably. Heat transfer is also achieved with radiation between the valleys of the asperities and conduction via the trapped air (Figure 2-29).

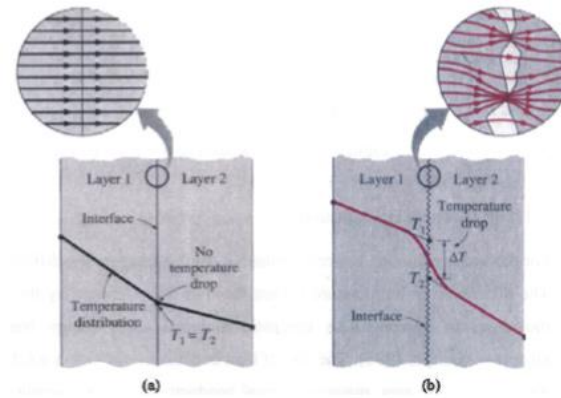


Figure 2-29 a) Ideal, b) Actual Thermal Contact Between Mating Solids, Çengel (Cengel, 2002)

Equation (2-4) can be written as follows to include the thermal conductance:

$$Q_{cond} = h_{cond}A_{CS}\Delta T_{int} \quad (2-5)$$

**Thermal
Conductance**

Where h_{cond} is the thermal conductance and ΔT_{int} is the temperature difference on the interface.

The only efficient way to determine the values of thermal conductance is by carrying out experiments and measuring temperatures close to the interface of the components with known material properties. Factors influencing the thermal conductance are (Tirovic, 2011):

- The materials in contact,
- The surface finish and condition,
- The ambient medium,

- The interface pressure.

Extensive experimental work in the area of conductive heat transfer on automotive brake disc has been conducted by Voller (Voller, 2003). The author developed an experimental rig, where a cast iron brake disc from a heavy-duty commercial vehicle was attached by bolts to a Spheroidal Graphite (SG) wheel carrier and a steel wheel (Figure 2-30). Four thermocouples were embedded on each side of the interface between the brake disc and the wheel carries. The surrounding surfaces were properly insulated to avoid any influence of convective and radiative heat transfer. By recording and plotting the temperatures from the eight thermocouples the author found that the relationship between temperature and distance along the conductive path was linear for each of interfaces sides. The conductive gradient slopes however were different in each of the components, but that was the result of the different values of thermal conductivity between GI and SG.

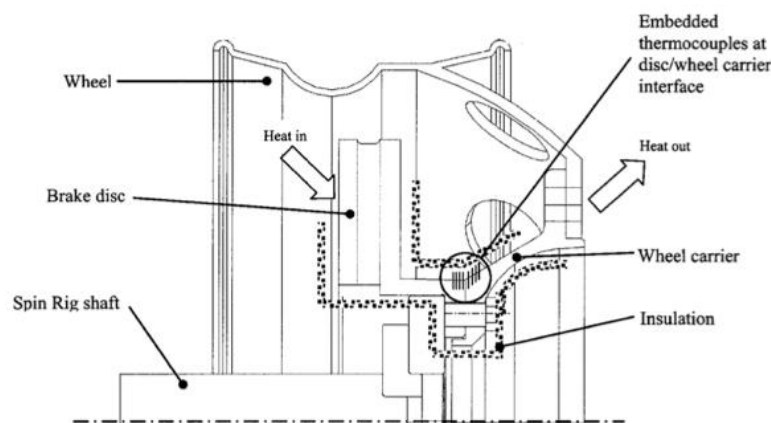


Figure 2-30 G. P. Voller et al. Experimental Set Up (Voller, et al., 2003)

Typical average values of thermal conductance were quoted in Voller's work (Voller, 2003) and are presented in Figure 2-31. For corroded components the values of h_{conv} are lower by approximately 40% compared to new components values that are about 12 KW/m²K. When Voller inserted paste of high thermal conductivity or thin aluminium foil in the interface, the thermal conductance values considerably increased to 59 and 67 KW/m²K respectively.

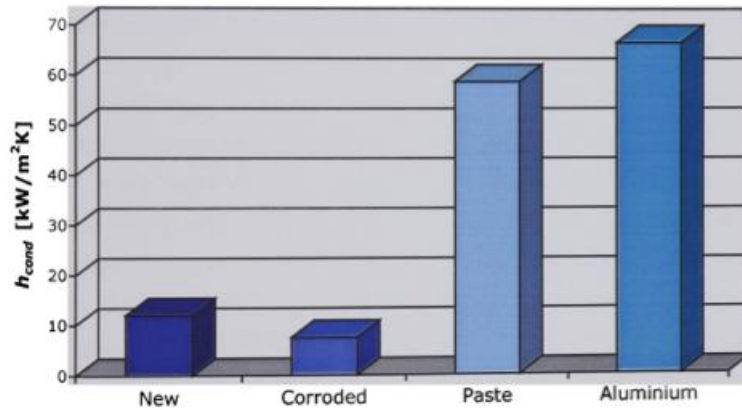


Figure 2-31 Values of Thermal Conductance, G. P. Voller (Voller, 2003)

Tirovic and Voller studied the influence of interface contact pressure in thermal conductance (Tirovic & Voller, 2005). By using the experimental set up of Figure 2-30 they adjusted tightening torque of the bolts and proved that the pressure-dependency of the TCR. They also found that the average thermal conductance has a linear relationship with the interface pressure. For the materials they used, with $Ra = 1 - 3.3 \mu m$, the expression derived was:

$$h_{cond(avg)} = 80P_{avg} + 2300 \quad \text{(2-6) Thermal Conductance Relation}$$

The expression given for the average interface pressure P_{avg} , which dependent on the number of bolts, the tightening torque (τ) and the contact area, was:

$$P_{avg} = 0.107426\tau \quad \text{(2-7) Average Contact Pressure}$$

2.5.2 Radiative Heat Dissipation

Radiative heat transfer, or thermal radiation, takes place when thermal energy is emitted from a solid body towards its surroundings in the form of electromagnetic waves largely in the infrared region (Modest, 2013). For the case of a solid at an absolute temperature being by gas, such as air, the rate of heat dissipated by radiation is given by the equation:

$$Q_{rad} = \varepsilon \sigma A_{rad} (T_D^4 - T_\infty^4) \quad (2-8) \quad \text{Radiative Heat Rate}$$

Where:

Q_{rad} : Radiative heat transfer,

ε : Emissivity,

σ : Stefan-Boltzmann constant,

A_{rad} , Area of radiation,

T_D : Average disc temperature,

T_∞ : Ambient temperature.

By applying Newton's law of cooling the derived equation is:

$$Q_{rad} = h_{rad} A_{rad} (T_D - T_\infty) \quad (2-9) \quad \text{Radiative Heat Rate}$$

By equating (2-8) and (2-9), the following expression for radiative heat transfer coefficient h_{rad} can be obtained:

$$h_{rad} = \varepsilon \sigma (T_D^3 + T_\infty^3 + T_D T_\infty^2 + T_\infty T_D^2) \quad (2-10) \quad \text{Radiative Heat Transfer Coefficient}$$

Or:

$$h_{rad} = \frac{\varepsilon \sigma (T_D^4 - T_\infty^4)}{T_D - T_\infty} \quad (2-11) \quad \text{Radiative Heat Transfer Coefficient}$$

As it can be seen from equations (2-8), (2-9), (2-10) and (2-11), the radiative heat transfer coefficient has a non-linear relation with temperature. It is also evident that thermal radiation is very influential in high temperatures and can even become the dominant mode of heat dissipation when the vehicle is stationary with its brakes at high temperature.

The determination of radiative heat transfer coefficient is heavily dependent on the values of surface emissivity for the solid in question. For any material, the emissivity is defined as the ratio of the emitted radiation of the material over the emitted radiation of the “black” body.

A number of studies have been dedicated on identifying emissivity values. Eisengrüber et al. used a number of different approaches to measure disc temperatures after having been frictionally heated (Eisengrüber, et al., 1999). It was observed that as a result of the hot spots on the discs’ surfaces, the emissivity values could be between 0.15 and 0.9 which represents a 500% increase. During a drag application the authors found that emissivity can start from 0.4 and reach the value of 0.7; this is a 75% increase. Noyes and Vickers assumed the emissivity value of 0.8 for their FE modelling (Noyes & Vickers, 1969) and Limpert suggested the use of 0.55 (Limpert, 1975). Stevens quantified the emissivity on a heavy-duty commercial vehicle brake disc (Stevens, 2013). The author found that for a heavily corroded disc after multiple cycles of heating and cooling, the emissivity had reached a value of 0.92 and was not affected by temperature variations (Figure 2-32).

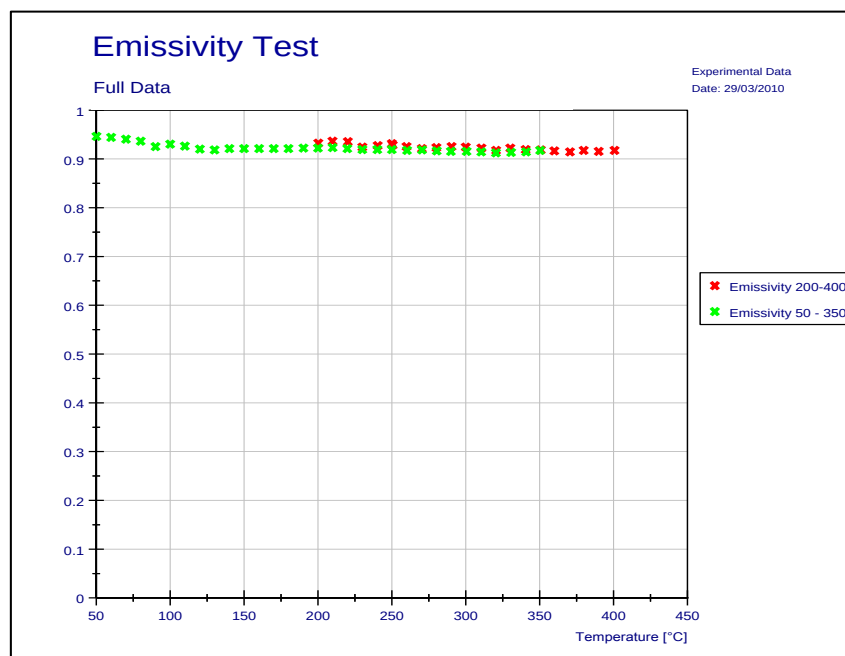


Figure 2-32 Emissivity VS Disc Temperature, (Stevens, 2013)

Voller demonstrated the impact of emissivity in radiative heat transfer coefficient by using different values found in the literature for the same set of experimental temperature measurements (Figure 2-33) (Voller, 2003).

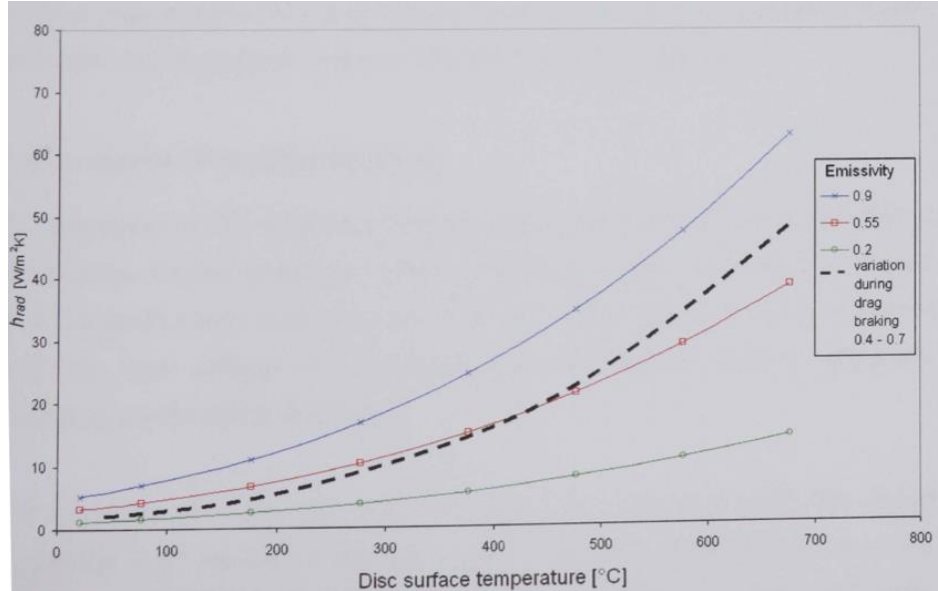


Figure 2-33 Influence of Emissivity on h_{rad} , Voller (Voller, 2003)

2.5.3 Convective Heat Transfer

In the mode of convective heat transfer, or more simplified convection, the heat transport process is influenced by the flow of fluids (Bejan, 2013). For the majority of braking scenarios convection is considered the dominant mode of heat transfer and unlike the previous two modes, conduction and radiation, it is speed dependent. Mathematically, the Newton's law of cooling can express the convective heat rate as:

$$Q_{conv} = h_{conv}A_w(T_D - T_{\infty}) \quad (2-12) \quad \text{Convective Heat Rate}$$

Where:

Q_{conv} : Convective heat transfer,

h_{conv} : Convective heat transfer coefficient,

A_w : Wetted area.

Convection can be categorised as “natural” or “forced”. When a disc is stationary in still air, heat is dissipated by natural convection. Due to heat exchange from the warmer disc to the colder air, the air density adjacent to the disc is reduced causing an upward movement of air streams around the disc, which gives way to new colder streams. This airflow is known as buoyancy flow and natural convection is based on this phenomenon. When a disc is rotated in still air or in a crossflow, then forced convection takes place. In forced convection, external means supply energy to the flow resulting in higher values of h_{conv} .

The convective heat transfer potentials heavily depend on the flow patterns which are found by utilising a set of dimensionless numbers. In the following Sections these numbers will be presented and discussed.

2.5.3.1 Reynolds Number

The Reynolds number is the ratio between the forces of inertia and viscosity (Isachenko, et al., 1987). For a flow over a plate with characteristic length l Reynolds number is described mathematically as:

$$Re = \frac{\rho U l}{\mu} = \frac{U l}{\nu} \quad (2-13) \text{ Reynolds Number}$$

Where:

ρ : Density,

U : Fluid velocity,

μ : Dynamic viscosity,

ν : Kinematic viscosity.

In problems with disc brakes, the expression for the Reynolds number is slightly different due to fluid velocity. Morgan and Dennis (Morgan & Dennis, 1972) proposed equation (2-14) to calculate Reynolds number when a disc is subjected to crossflow and Sheridan et al. introduced equation (2-15) for the rotating disc (Sheridan, et al., 1988).

$$Re_t = \frac{\rho U R_o}{\mu} = \frac{U R_o}{\nu}$$

**(2-14) Crossflow
Reynolds Number**

Where:

**(2-15) Rotational
Reynolds Number**

R_o : Channel outer radius.

$$Re_\omega = \frac{\rho \omega R_o^2}{\mu} = \frac{\omega R_o^2}{\nu}$$

Where:

ω : Rotational speed,

R_o : Disc outer radius.

The practical use of Reynolds number is to determine whether the flow is laminar or turbulent. According to aus der Wiesche the transition from laminar to turbulent flow is when $Re_t = 5 \times 10^4$ and the flow becomes fully turbulent when $Re_t > 10^5$ (aus der Wiesche, 2007).

When the disc is rotating, because the rotational Reynolds number is influenced by the disc's diameter, as shown in equation (2-15) the flow may not be the same over the entire disc (Harmand, et al., 2013). For the rotational Reynolds number, a variety of different limits for the laminar, the transition and the turbulent flow have been published in the literature. According to Shevchuk the majority of these limits have been experimentally identified and the differences between them are a result of differences in experimental apparatus and measuring techniques (Shevchuk, 2009).

The lowest value for the upper limit of the laminar flow was introduced by Tien and Campbell, who suggested $Re_\omega = 1.8 \times 10^5$ to be the beginning of transition to turbulent flow (Tien & Campbell, 1963). Liang et al. used the value of $Re_\omega = 2.4 \times 10^5$ (Liang, et al., 2005), Cardone et al. the value of $Re_\omega = 2.6 \times 10^5$ (Cardone, et al., 1996), and the highest transition value was quoted by Elkins and Eaton, $Re_\omega = 3.6 \times 10^5$ (Elkins & Eaton, 1994).

2.5.3.2 Prandtl Number

The Prandtl number is composed entirely of physical parameters and it represents the ratio of the fluid's momentum diffusivity over its thermal diffusivity. When the momentum diffusion overcomes the thermal diffusion the Prandtl number is high. This is usually found in oils where the dominant mode of heat transfer is the thermal conductivity. When convection is the predominant heat transfer mode, the Prandtl number is low. Liquid metals are substances with a small Prandtl number (Özisik, 1989). The Prandtl number is not dependent on the fluid – only on the property of the surface interaction.

$$Pr = \frac{\mu C_p}{\kappa} = \frac{\nu}{\alpha} \quad (2-16) \text{ Prandtl Number}$$

Where:

C_p : Specific heat,

α : Thermal diffusivity.

2.5.3.3 Grashof Number

The Grashof number is used in natural convection to characterise the flow pattern. It is similar to Reynolds number as it defines the ratio of inertia forces of the fluid to its viscous forces. For laminar flow, the viscous forces are prevailing not allowing eddy currents to develop. With the increase of the inertial forces the fluid transitions from laminar to turbulent flow, with the viscous forces not being able to diminish inertial forces (Özisik, 1989). When the Grashof number is very small, the buoyancy of the fluid is neglected. Equation (2-17) allows the calculation of Grashof number for a vertical flat plate with characteristic length l .

$$Gr = \frac{g\alpha(T_w - T_\infty)l^3}{\nu^2} \quad (2-17) \text{ Grashof Number}$$

Where:

g : Acceleration due to Earth's gravity,

α : Coefficient of thermal expansion,

T_w : Surface temperature.

2.5.3.4 Rayleigh Number

The Rayleigh number is used instead of the Grashof number for the investigation of heat transfer for a body exposed to buoyant airflow. It characterises the influence of the conductive and convective heat transfer. There is a critical Rayleigh value below which the predominant heat transfer mode is conduction, whilst above the critical value convection becomes the primary mode. The critical value depends on aspects such as the shape and the surface of the solid but also on cooling situation. Different critical values were found in the literature. For instance, McAdams proposed the value of $Ra = 10^7$ for a flat horizontal plate (McAdams, 1954) and Özisik the value of $Ra = 10^9$ for a cylinder vertically placed in free air (Özisik, 1989). Equation (2-18) allows the calculation of Rayleigh number:

$$Ra = Gr \times Pr \quad (2-18) \text{ Rayleigh Number}$$

2.5.3.5 Nusselt Number

The Nusselt number, or criterion of heat transfer, is a dimensionless number that distinguishes the heat transfer process at the “wall-fluid” boundary (Isachenko, et al., 1987). In other words, it represents the ratio of convective and conductive heat transfer between the solid surface boundary and the fluid that flows across its thickness in the normal to the surface of the solid direction. When $Nu = 1$, the airflows are laminar with the conduction and convection having an even distribution, whereas for $Nu > 100$, the airflows are turbulent and dominated by convective heat transfer. The convective heat transfer coefficient for a body with characteristic length of surface l , can be calculated by the equation:

$$Nu = \frac{h_{conv} l}{\kappa} \quad (2-19) \text{ Nusselt Number}$$

Calculating the Nusselt number from the equation (2-19) is not always straightforward, as some of the parameters might not be known. That is especially

true for the convective heat transfer coefficient, which in most cases has to be identified experimentally.

2.5.3.6 Convective Cooling Characterisation of Brake Discs

As mentioned in Section 2.5, the convective cooling heavily depends on the fluids flow patterns, which can be described by the dimensionless numbers presented above. Extended research has been conducted over the years by researches that incorporated these dimensionless numbers to characterise the convective cooling of different solids. This part of the literature survey has been focused on studying previous work in the area of brake disc cooling. The early investigations included flat disks rotating in still air. Von Kármán examined the airflow near a single rotating disk and identified that the developing boundary layer, near the disk, had its thickness depending on the disk's angular velocity and kinematic viscosity (Kármán, 1921). The author also calculated the torque resisting the rotation, because of the surface friction of the fluid, for both laminar and turbulent flows.

Goldstein studied the heat transfer from a rotating disk in the laminar region (Goldstein, 1935). By using analytical and experimental methods, he derived a relation between the coefficient of resisting moment for both sides of the disc and the Reynolds number. In addition, expressions for the turbulent velocity distributions were presented.

$$Nu = 0.38Re^{0.5} \quad \textbf{(2-20) Goldstein's Equation}$$

Wagner investigated the heat transfer from a rotating disc to the ambient air (Wagner, 1948). Based on distributions of temperature and velocity, he derived the following equations for convective heat transfer, where the fluid has $Pr = 0.74$:

$$h = 0.339Re^{0.5} \quad \textbf{(2-21) Wagner's Convective Heat Transfer Expression (1)}$$

$$h = 0.335Re^{0.5} \quad \textbf{(2-22) Wagner's Convective Heat Transfer Expression (2)}$$

Equation (2-22) represents a closer approximation than equation (2-21), because the flow of enthalpy was taken into consideration. It was mentioned that these equations were true for laminar flows which occur when $Re_\omega < 5 \times 10^5$.

Millsaps and Pohlhausen examined analytically the heat transfer of a uniformly heated circular plate rotated about its vertical axis at a constant angular velocity within a viscous fluid (Figure 2-34). The derived equation when $Pr = 0.74$ is shown below (Millsaps & Pohlhausen, 1952).

$$Nu = 0.28Re^{0.5}$$

(2-23) Millsaps and Pohlhausen Equation

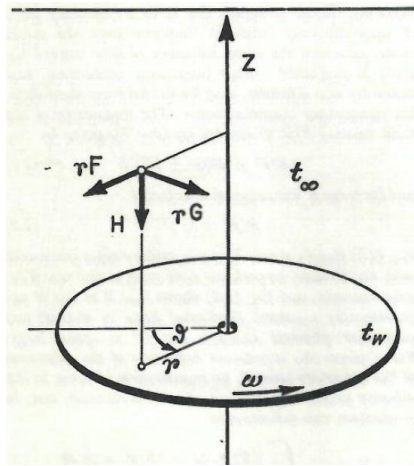


Figure 2-34 Schematic Diagram With the Coordinate System and the Velocity Components (Millsaps & Pohlhausen, 1952)

Experimental investigations for convective heat transfer were conducted in (Anderson & Saunders, 1953). Their experimental set up consisted of electrically heated cylinder positioned horizontally able to rotate. Three different cylinder diameters were used for the experiments: 1, 1.8 and 3.9-inch of outer diameter. The cylinders were able to rotate, the temperature on their surface was measured by thermocouples and the energy input was measured electrically. The cylinders were heated up to 60°C higher than ambient temperature and their ends were heated separately by end-sections for the sake of end-loses minimisation. The heat dissipation from radiation was taken into consideration. The whole

apparatus was isolated from draughts in order to avoid influence of forced convection.

It was observed by the authors that for low angular velocities the ascending side of the cylinders had higher values for heat transfer than the natural convection, but the descending side had lower values resulting in an overall constant convective heat transfer. They also observed the existence of critical rotational speeds at which the heat transfer was becoming dependent on the Re_ω . The critical values for Re_ω when the $Pr = 0.75$ were defined by:

$$Re_\omega = 1.09\sqrt{Gr} \quad (2-24) \text{ Critical } Re_\omega \text{ Anderson and Saunders}$$

Above this critical Re_ω , the equation governing the convective heat transfer was:

$$Nu = 0.10Re_\omega^{2/3}\sqrt{Gr} \quad (2-25) \text{ Cr Anderson and Saunders Equation}$$

Cobb and Saunders were aiming to study the heat transfer from a rotating disk for angular velocities in the turbulent region (Cobb & Saunders, 1956). They constructed an experimental rig where a disk with outer diameter of 18-inch was mounted on a shaft with outer diameter of 1-inch (Figure 2-35). The disk could be spun up to 2000 rpm which would result in 80% of the disk area to be within the turbulent region. The disk comprised two aluminium disks and an electric heating disk in between the two aluminium disks.

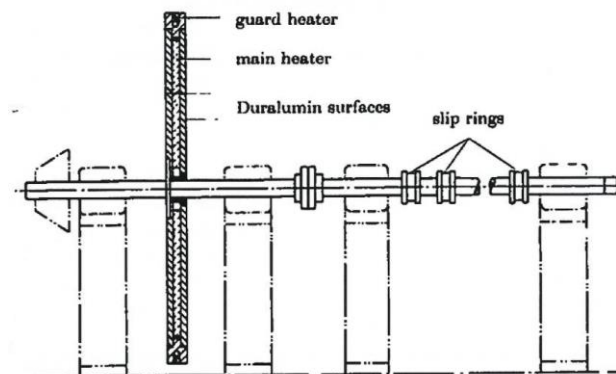


Figure 2-35 Schematic Representation of E. C. Cobb and O. A. Saunders (Cobb & Saunders, 1956)

In order to avoid heat transfer from the disk's outer diameter surface, the authors installed a guard heater. The result was a uniformly heated disk with $\pm 5\%$ of temperature distribution as given by thermocouples positioned close to the outer diameter's edge and the centre of the disk. For the elimination of experimental errors, correction coefficients were applied for the heat transfer from the outer diameter surface of the disk and also from the inner diameter towards the rotating shaft.

The heat transfer when the disk angular velocities are within the laminar region was firstly investigated. The equation derived for the mean Nusselt number from experiments conducted for Reynolds numbers between 10^5 and 2×10^5 was:

$$Nu = 0.36Re_{\omega}^{0.5} \quad \textbf{(2-26) Cobb and Saunders Nu for Laminar Flow}$$

Thereafter, experiments were conducted for turbulent angular velocities which, according to the authors, begin at $Re_v = 2.4 \times 10^5$. The areas on the edge of the outer diameter of the disk are the first to be in the turbulent region and as the angular velocity increases the areas towards the inner diameter are included as well. The derived equation for the mean Nusselt number in turbulent flow was:

$$Nu = 0.015Re_{\omega}^{0.8} \quad \textbf{(2-27) Cobb and Saunders Nu for Turbulent Flow}$$

Richardson and Saunders experimentally investigated the heat transfer from a rotating disk in both the laminar and turbulent regions (Richardson & Saunders, 1963). Roughly, the same experimental set up used in (Cobb & Saunders, 1956) was used in this study as well. The derived experimental formula for the laminar region, indicated dependence of the Nusselt number on both the Reynolds and Grashof numbers:

$$Nu = 0.40(Re_{\omega}^{0.8} + Gr)^{1/4} \quad \textbf{(2-28) Richardson and Saunders Nu for Laminar Flow}$$

The equation for the turbulent region was:

$$Nu = 0.0138Re_{\omega}^{0.8}$$

**(2-29) Richardson and
Saunders Nu for Turbulent
Flow**

According to the authors, equation (2-29) gave results far closer to the experimental values than the formula for turbulent flow presented in (Cobb & Saunders, 1956) equation (2-27).

Newcomb and Millner conducted experimental investigations with regards to the cooling rates of brake discs and brake drums (Newcomb & Millner, 1965). They installed rubbing thermocouples in fourteen passenger cars and one lorry to measure the temperatures in brake discs and drums. By drag braking they increased the temperature up to 300°C-400°C and then cooled the drums and discs at constant vehicle speeds between 0-90 miles per hour. Their experiments were conducted during warm summer days with maximum wind of 7 miles per hour.

They started their analysis by identifying that the total cooling rate b_u consists of two components: the b_o , which is the conductive cooling rate that is independent of the vehicle speed, and b_c , which is the convective cooling rate that is speed dependent.

$$b_u = b_o + b_c$$

**(2-30) Newcomb and
Millner's Cooling Rate**

After conducting the tests and using $Nu = 0.015Re_{\omega}^{0.8}$ to derive the convective heat transfer when the flow was turbulent, they proposed the following equation for the total cooling rate:

$$b_u = (1.17 \times 10^{-3}A_c + 0.00094 \times 10^{-3}Au^{0.8})/w$$

**(2-31) Newcomb and
Millner's Cooling Rate**

Where:

b_u : The area of disc and drum in contact with the hub in [in^2],

b_u : The total area of the rubbing path in $[in^2]$,

b_u : The vehicle speed in $[\frac{miles}{hr}]$,

w : The weight of the disc or the drum in $[lbs]$.

Furthermore, they reached the following conclusions:

- The front drums were cooling 10-20% quicker than the rear drums,
- The ratio of cooling rates for cars with drums carrying circumferential fins in the front and bimetallic drums at rear was 1.1:1,
- In vehicles equipped with disc brakes in the front and drums at the rear, the values of b_u were 20-40% higher for the brake discs,
- When mounted on the front axle, disc brakes had approximately 25% improved cooling than drums also mounted on the front axle,
- No great difference in cooling was noticed when instead of solid wheels, wire wheels were used,
- The addition of dust shields reduced the cooling rates by approximately 30%.

Dennis et al. created an experimental set up to investigate the heat transfer from discs when rotating in still air, when they were stationary in air crossflow and when they rotated in air crossflow (Dennis, et al., 1970). Their set up comprised an electrically heated disc with 450 mm of outer diameter. The disc could rotate from 30 to 3000 rpm. For the air crossflow tests they used a 0.7 m X 0.25 m test section wind tunnel and air velocities between 5 and 32 m/s.

The equation describing the heat transfer for a rotating disc in still for laminar rotational Reynolds numbers was:

$$Nu = CRe_{\omega}^{0.5}$$

**(2-32) Dennis et al. General
Nu Equation Form for
Laminar Flow**

The values of C varied between 0.31 and 0.4 with the value of 0.4 giving the best correlation. For the transition to turbulent region, which began for $Re_{\omega} > 2.4 \times 10^5$, the equation was:

$$Nu = 0.015Re_{\omega}^{0.8} \quad (2-33) \text{ Dennis et al. Nu Equation for Turbulent Flow}$$

For the transition region between the laminar and turbulent flows the following equation was derived for the outer radius of the disc:

$$Nu = 0.4 \frac{Re_{cr}}{\sqrt{Re}} + 0.015 \left[Re^{0.8} - Re_{cr}^{0.8} \sqrt{\frac{Re_{cr}}{Re}} \right] \quad (2-34) \text{ Dennis et al. Nu Equation for Transition Flow}$$

In equation (2-34) the term Re_{cr} is the critical Reynolds number for the transition to turbulent flow. For the second case, where the disc was stationary in an air crossflow, for laminar flow the equation for heat transfer was:

$$Nu = 0.364Re_t^{0.5} \quad (2-35) \text{ Dennis et al. Nu Equation for Stationary Disc in Laminar Flow}$$

and for turbulent flow:

$$Nu = 0.027Re_t^{0.8} \quad (2-36) \text{ Dennis et al. Nu Equation for Stationary Disc in Turbulent Flow}$$

Finally, for the case of the disc rotating in a crossflow, the authors assumed the heat transfer coefficients as vector quantities instead of scalar quantities with direction being the direction of the freestream. Thereafter, they vectorially added their experimental results from the heat transfer of the rotating disc in still air and the results when the disc was stationary in the crossflow. With this method they derived the equation for the local heat transfer when the disc rotated in air crossflow:

$$Nu = 0.0199Re_r^{0.8}$$

(2-37) Dennis et al. Nu Equation for Rotating Disc in Crossflow Flow

In equation (2-37) the term Re_r is the Reynolds number that is dependent on the radius of the disc.

Sheridan et al. mentioned the existence of empirical relationships for the calculation of convective heat transfer coefficient available in the open literature. Despite the fact these relationships were derived for geometrical shapes, like disks and cylinders, they were found to be useful in practical applications, such as the characterisation of brake disc cooling (Sheridan, et al., 1988). The expression for a disc rotating in crossflow is given by:

$$Nu = 0.436Re_{\omega}^{0.8} \left(\frac{Re_t}{Re_{\omega}} \right)^{0.74}$$

(2-38) Sheridan et al., Rotating Disc in Crossflow

For a turbulent rotation of a disc in still air, the expression relating the Nusselt number and the Reynolds number that was found to be useful by the authors was:

$$Nu = 0.0195Re_{\omega}^{0.8}$$

(2-39) Sheridan et al., Rotating Disk in Still Air

aus der Wiesche investigated numerically the aerodynamics of a thin disk when stationary and also rotating in a parallel flow of air with constant mean velocity (aus der Wiesche, 2002). For the case of the stationary disk subjected to laminar flow the derived correlation for heat transfer was:

$$\overline{Nu} = 0.470Re_t^{1/2}Pr^{1/3}$$

(2-40) Wiesche Nu Equation for Stationary Disk in Laminar Flow

Equation (2-40) is true for: $10^3 \leq Re_t \leq 5 \times 10^4$. For the case of the stationary disk subjected to turbulent flow, where $10^5 \leq Re_t$, the derived correlation for heat transfer was:

$$\overline{Nu} = 0.012 Re_t^{4/5} \frac{Pr}{1 + 2.443 Re_t^{-\frac{1}{10}} (Pr^{\frac{2}{3}} - 1)} \quad (2-41) \text{ Wiesche Nu Equation for Stationary Disc in Turbulent Flow}$$

When the angular velocity of the disk is low, for $Re_\omega \ll Re_t$ the, heat transfer is not increased by the rotation of the disc. On the ascending side of the disk the heat transfer is higher, but the reduced heat transfer on the descending side counters this.

$$\overline{Nu} = \overline{Nu}_{\omega=0} \quad (2-42) \text{ Wiesche Mean Nu for } Re_\omega \ll Re_t$$

For increased values of disk angular velocity, when $Re_\omega \geq Re_t$, flow separation takes place and leads to increased local values of heat transfer. The high heat transfer values of the ascending side are not counterbalanced by the low values of the descending side due to the symmetry being broken by the critical value of Re_ω . The author suggested that the Landau theory can offer simple correlation.

$$\overline{Nu} = \overline{Nu}_{\omega=0} + 0.34 \overline{Nu}_{\omega=0} \left(\frac{Re_\omega - Re_{\omega,cr}}{Re_t} \right) \quad (2-43) \text{ Wiesche Mean Nu for } Re_\omega \geq Re_t$$

Finally, for the case of very high disk angular velocities, when $Re_\omega \gg Re_t$ the author proposed a logarithmic correlation as a substitute of the power-law for the Nusselt number.

$$\overline{Nu} = c + m \log \frac{Re_\omega}{Re_t} \quad (2-44) \text{ Wiesche Mean Nu for } Re_\omega \gg Re_t$$

In equation (2-44), term c is a correlation factor and term m is an augmentation gradient in corresponding to logarithmic plots.

aus der Wiesche continued his study on the heat transfer coefficient for disks. By applying the numerical method of large-eddy-simulations (LES) using the Smagorinsky sub-filter (aus der Wiesche, 2004), he derived relations for mean Nusselt numbers for three cases: a) when the disk rotates in still air, b) when the disk is stationary in a parallel crossflow and c) when the disk rotates in a crossflow. For the first case, the derived equations were:

$$\overline{Nu} = 0.33Re_{\omega}^{1/2} \quad \text{(2-45) Wiesche Nu Equation for Disc Laminar Rotation in Still Air}$$

for $10^3 \leq Re_{\omega} \leq 2 \times 10^5$ and $Pr = 0.7$.

$$\overline{Nu} = 0.015Re_{\omega}^{0.8} \quad \text{(2-46) Wiesche Nu Equation for Disc Turbulent Rotation in Still Air}$$

for $5 \times 10^5 \leq Re_{\omega}$ and $Pr = 0.7$.

For the second case the equations derived were:

$$\overline{Nu} = 0.417Re_t^{1/2} \quad \text{(2-47) Wiesche Nu Equation for Stationary Disc in Laminar Crossflow}$$

for $10^3 \leq Re_t \leq 2 \times 10^5$ and $Pr = 0.7$.

$$\overline{Nu} = 0.0127Re_t^{0.8} \quad \text{(2-48) Wiesche Nu Equation for Stationary Disc in Turbulent Crossflow}$$

for $5 \times 10^5 \leq Re_t$ and $Pr = 0.7$.

The author explained that the difference in results between equation (2-48) and the experimental results in (Dennis, et al., 1970) was due to the large amount of turbulence that took place during the experiments.

Thereafter, for the case of a rotating disk subjected to a crossflow of air, it was found that the air velocity has an impact on the heat transfer in the following ways: when the air velocity is low this results to the translational Reynolds number to be low. Thus the rate of heat transfer is governed by the angular velocity of the disk. In Figure 2-36, the horizontal lines in the low values of Re_t signify that the mean Nusselt number is independent of the Reynolds for crossflow. When the crossflow velocity increases the translational Reynolds number increases; this results in the heat transfer being governed by the crossflow. That is why in Figure 2-36 for high Re_t numbers the curves follow the curve when $Re_\omega = 0$.

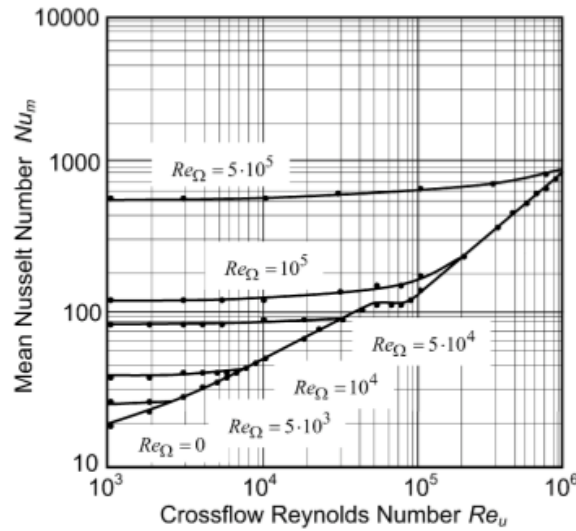


Figure 2-36 Nu, Re_ω and Re_t Numbers. (aus der Wiesche, 2007)

In addition, for crossflows with laminar Reynolds number it was found that there is a critical number for the ratio of Re_ω/Re_t , which is $Re_\omega/Re_t \approx 1.4$. This finding allowed the author to the limits for equation (2-42) which are: $0 \leq Re_\omega/Re_t \leq 2 \times 10^5$. It also allowed him to revise equation (2-43) to the following expression

$$\overline{Nu} = \overline{Nu}_{\omega=0} + 0.32 \overline{Nu}_{\omega=0} \left(\frac{Re_\omega}{Re_t} - 1.4 \right)^{1/2} \quad \text{(2-49) Wiesche Mean Nu for } Re_\omega \geq Re_t \text{ (Revised)}$$

Finally, the author derived the mean Nusselt number equations for a rotating disk in crossflow parallel to the disk plane for gases with $Pr = 0.7$:

$$\overline{Nu} = \begin{cases} 0.417Re_t^{1/2}, & \text{for } Re_\omega/Re_t \leq 1.4 \\ 0.330Re_\omega^{1/2}, & \text{for } Re_\omega/Re_t > 1.4 \end{cases} \quad \text{(2-50) Wiesche Nu For Rotating Disk in Laminar Crossflow}$$

when $10^3 \leq Re_t \leq 5 \times 10^4$.

$$\overline{Nu} = \begin{cases} \sqrt{(0.0127Re_t^{0.8})^2 + (0.015Re_\omega^{0.8})^2} & \text{for } Re_\omega > 2 \times 10^5 \\ \sqrt{(0.0127Re_t^{0.8})^2 + (0.33Re_\omega^{0.8})^2} & \text{for } Re_\omega < 2 \times 10^5 \end{cases} \quad \text{(2-51) Wiesche Mean Nu For Rotating Disk Turbulent Crossflow}$$

when $5 \times 10^4 < Re_t$.

Latour et al. calculated experimentally the convective heat transfer coefficient from a disc by using a thermal image camera (Latour, et al., 2011). Their experimental set up (Figure 2-37) included a disk mounted on a cylinder placed in a wind tunnel that could be rotated by an electric motor, a radiant panel that emitted short infrared waves to heat up the disc up to 120°C, a pyrometer to measure the air temperature in the tunnel and a thermal image camera.

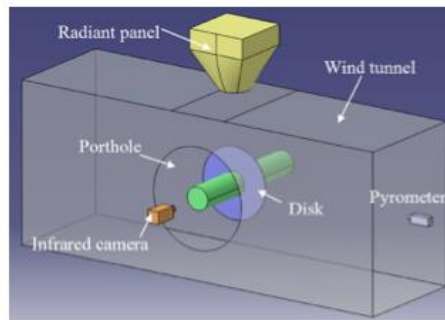


Figure 2-37 B. Latour et al., Experimental Set Up

It is worth mentioning that the surface of the disk was covered by a thin layer of black paint coating that gave the disk constant and known emissivity. They conducted the experiments under three different conditions: a) rotating disk in still air, b) stationary disk in crossflow and c) rotating disk in crossflow. When they processed the results, they derived relationships for the Nusselt and Reynolds numbers.

$$\overline{Nu} = 0.556Re_{\omega}^{0.5}$$

(2-52) Latour et al., Nu for Rotating Disk in Still Air

Equation (2-52) is the relation when Re_{ω} is in the laminar region and the disk is rotating in still air.

$$\overline{Nu} = 0.036Re_t^{0.8}$$

(2-53) Latour et al., Nu for Still Disk in Crossflow

Equation (2-53) is for the case of stationary disk in a crossflow of air and also the same equation applies when the disk is rotating in still air with the ratio of Reynolds number being $Re_{\omega}/Re_t < 0.18$.

$$\overline{Nu} = \sqrt{(0.036Re_t^{0.8})^2 + (0.556Re_{\omega}^{0.5})^2}$$

(2-54) Latour et al., Nu for Rotating Disk in Crossflow

For a rotating disk in a crossflow and a ratio of Reynolds number $Re_{\omega}/Re_t > 0.18$, equation (2-54) was proposed.

2.6 Chapter Summary

The literature review was focused on a number of thematic areas starting from the brake disc analysis, then moving to cast iron as it is the material of choice for the brake rotors, the automotive wind tunnels and finally studying the heat dissipation from the brake discs. In the first thematic area, early methods that formed the basis of the brake disc analysis using analytical approaches were identified and then modelling techniques using the Finite Element method were reviewed.

In the second thematic area, the literature survey was focused on understanding the attributes of cast iron and its behaviour when used in engineering applications, in general, and also when utilised in automotive brake discs. In the part dedicated to the automotive wind tunnels best practices in developing such facilities were comprehended and the basic principles in achieving stable air flow were appreciated.

In the last section of this chapter, an in-depth review of the brake disc heat dissipation took place. The application of all three modes of heat transfer, conduction, convection and radiation, was discussed. Particular focus was given on convective heat transfer as it is the dominant mode of brake disc heat dissipation in the majority of operating conditions. Additionally, dimensionless numbers that are used to study the convective heat transfer, such as the Reynolds number and the Nusselt number were presented followed by analytical and experimental equations that use these numbers in order to characterise the flow conditions and/or derive the convective heat transfer coefficients.

By reviewing the findings of the literature survey, it was evident that there was a gap in the area of lightweight brake rotors. More specifically, no work was identified in the area of mechanical analysis, experimental validation of the mechanical analysis, structural optimisation and experimental investigation of cooling aspects for such discs. All these areas are discussed in the ten-step methodology presented in the following chapters.

3 LIGHTWEIGHT FINGERED BRAKE DISC INVESTIGATION – MECHANICAL MODELLING

3.1 Background

The efforts to investigate the feasibility of an automotive brake rotor with a hub comprising spokes or fingers, that would provide weight reductions and improve cooling, started with a rotor intended for a mid-size sedan vehicle. The requirements of that initial study were for the brake disc to:

- Be made out of cast iron,
- Be installed in the front axle of the vehicle,
- Offer weight reductions of at least 300 gr compared to the baseline disc,
- Be manufacturable with a casting process.

A number of concepts were created with each of the discs featuring top hats of a different design (Figure 3-1). All these designs were analysed for mechanical and thermomechanical stresses using the methods that are presented in this Chapter and in Chapter 4.

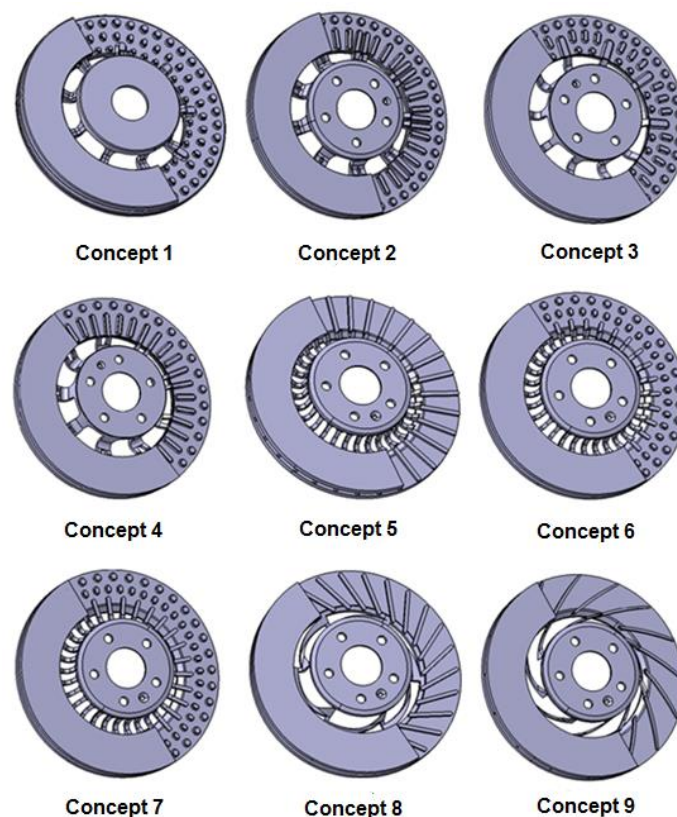


Figure 3-1 Fingered Hub Conceptual Designs for a Mid-Size Sedan

The main conclusions of this initial investigation were that a brake rotor with fingered hub design would be able to withstand the thermomechanical loading resulting by the heat flux entering the rotor. However, because of the hub not being a stiff uninterrupted tube anymore, the real challenge for a brake disc incorporating such a design was the mechanical loading. In other words, the brake disc is incapable of transferring the developed braking torque without being fractured.

It was soon realised that the major limiting factors were the maximum allowable weight of a mid-size sedan, which can be in the area of 2200 kg, and the cast iron mechanical strength. Since the cast iron properties cannot change drastically and the objective from the beginning was to create a lightweight brake disc made out of cast iron, the investigation was focused on more lightweight vehicles to evaluate the feasibility of the fingered hub design. Still, some designs showed promising characteristics such as the Concept 5 which featured low values of stresses.

3.2 Baseline Vehicle and Disc Selection

The vehicle selected as the basis for further investigations was the Lotus Elise MY 2013. It is a lightweight vehicle with almost half the mass of a mid-size sedan when fully laden. In addition, its discs (Figure 1-1) had considerably smaller inner and outer diameters compared to discs used in mid-size family sedans and smaller axial offset.

Table 3-1 Lotus Elise Specifications (Lotus Cars, 2013)

Top Speed [m/s]	57
Mass (fully laden) [kg]	1141
Wheelbase [mm]	2300
Length / Width / Height [mm]	3824 / 1719 / 1117
Static Load Distribution (front / rear)	50:50

Height of CoG [mm]	450
Disc OD/ID [mm]	288 / 185
Rolling Radius [m]	0.3

3.3 Step 1 – Concept Generation

The first step in the study was a brainstorming session aiming to generate viable conceptual designs. Although there are at least three different factory designs for Lotus Elise brakes discs, including one for the Opel/ Vauxhall Speedster/VX220, the design which the new concepts were based on was the series two brake disc, also known as S2. This is used in models from 2001 and onwards and it is the one used in the Lotus Elise MY 2013.

During the brainstorming, three different conceptual designs were generated. The first incorporated eighteen spokes of trapezoidal shape (Figure 3-2). It was named “Elise Version 1”.

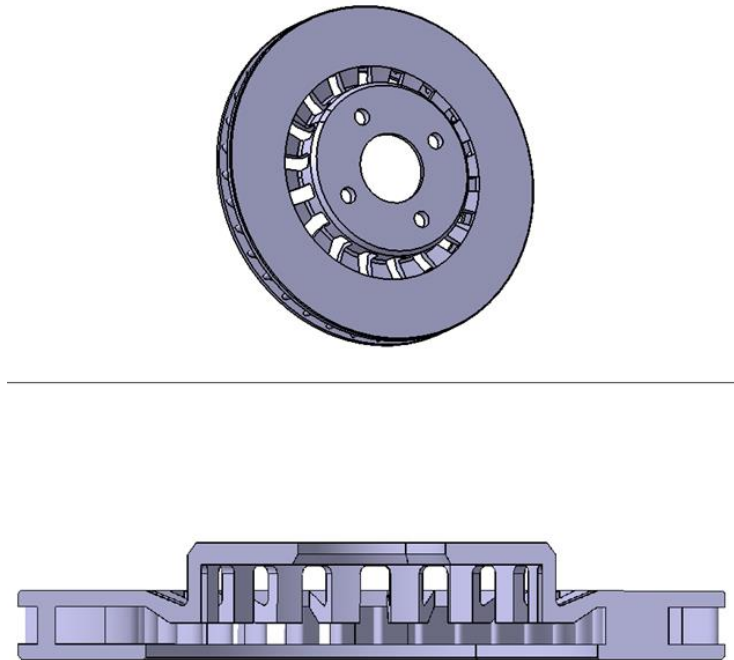


Figure 3-2 Elise Version 1

The second conceptual design featured eighteen straight spokes (Figure 3-3). This was the “Elise Version 2”.

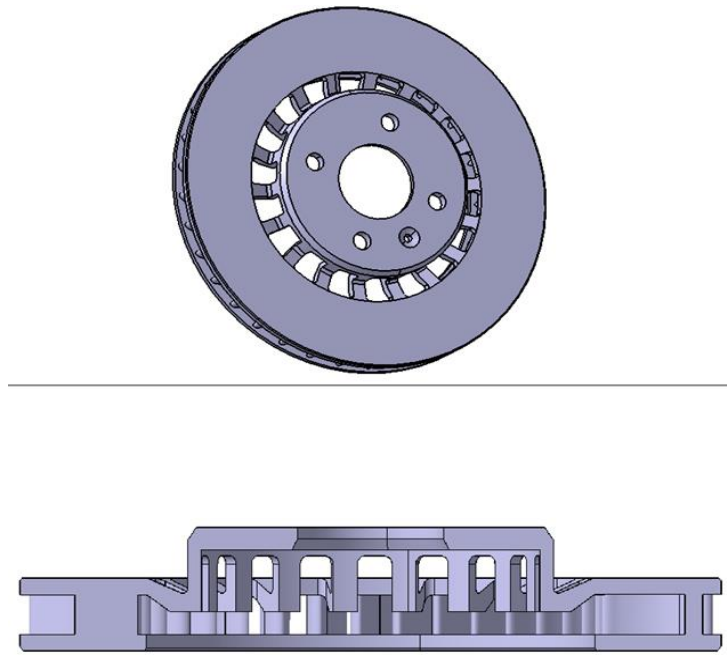


Figure 3-3 Elise Version 2

The third and final conceptual design featured eighteen spokes of round shape (Figure 3-4). This was the “Elise Version 3”. Detailed drawings of the three concepts can be found in Appendix A.

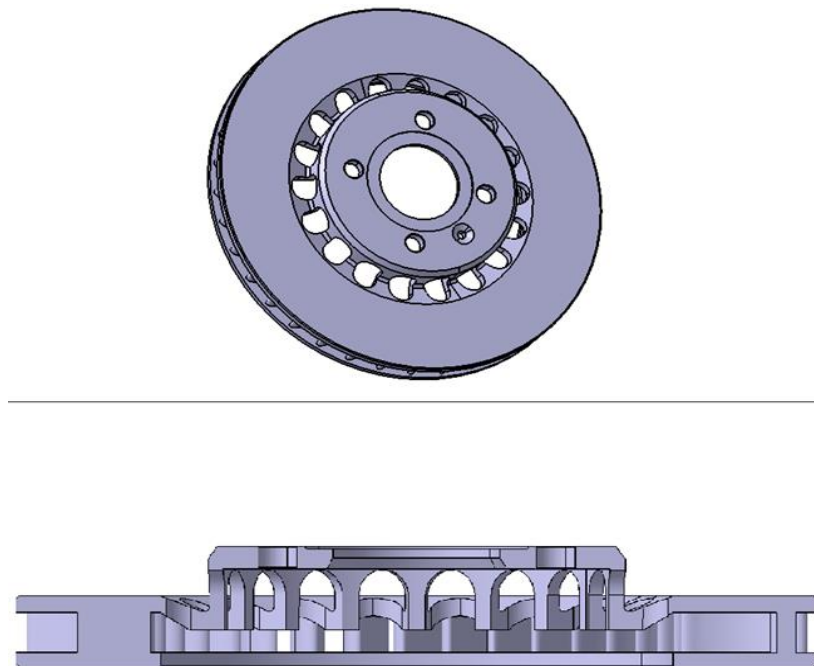


Figure 3-4 Elise Version 3

All the conceptual designs were thought to be manufacturable and in terms of mass savings all the designs achieved reduction 400 g or more (Table 3-2). The mass calculations were performed using Catia V5 and the value of density used was 7200 kg/m³.

Table 3-2 Mass Comparisons – Lotus Elise Case Study

Brake Disc Model	Mass [kg]	Mass Reduction [%]
Original Elise S2	6.0	-
Elise Version 1	5.6	-6.6
Elise Version 2	5.5	-8.3
Elise Version 3	5.5	-8.3

3.4 Step 2 – Development of Mechanical Modelling Techniques

At this stage of the study, mechanical FE analyses were conducted on all three concepts described in Section 3.3. The aim was to model the application of braking torque which occurs during braking as closely as possible and then evaluate whether the proposed design concepts are capable of withstanding such a torque. As it was found from the analyses in the disc for the mid-size family sedan, the real challenge for this project was the mechanical loading on discs with fingered hub and not the thermal loading. This is happening as a result of the torque being transferred from the friction faces to the hub via a number of spokes (fingers) with small cross section instead of the solid tube connection between the friction faces and the hub found in ordinary brake discs.

The purely mechanical loading has never been an issue in brake design due to the stiff cylindrical hub. As a result, no published work in the area is in existence meaning that ways to introduce the braking torque had to be investigated. Two different techniques were developed and they are presented in the next Sections (Topouris, et al., 2015). It has to be mentioned that these modelling cases were established during the feasibility study for the fingered disc design on a mid-size sedan, and then applied on the Lotus Elise concepts.

3.4.1 First Modelling Case

In this modelling technique, also referred to as “Case 1”, the friction faces were constrained and torque was applied on the hubs of the brake disc concepts.

An area in the shape of a brake pad imprint was created on each of the friction surfaces. The dimensions were arbitrary, but the overall area was similar to the areas of brake pads used with brake discs with dimensions (OD 288 mm, ID 185 mm). All the Degrees of Freedom (DoF) on these partitioned areas of the friction faces were blocked with “Encastré” mechanical constraint.

In order to apply the required torque on each disc, the idea was to simulate how the studs tend to rotate the wheel of the car and at the same time the brake disc. It was envisaged that point loads acting on the holes of the disc’s hub would force the hub to rotate around the centre of rotation of the disc. In order to realise this idea, the upcoming methodology was followed. A reference point was created in the middle of each of the holes of the hub. Afterwards, a cylindrical coordinate system (ρ , θ , z) was created with its origin at the centre of rotation, ρ was aligned with one of the reference points, θ was the swept angle and (+ve) z the axis of rotation. The position of the reference points and their related cylindrical coordinate systems are presented in Figure 3-5.

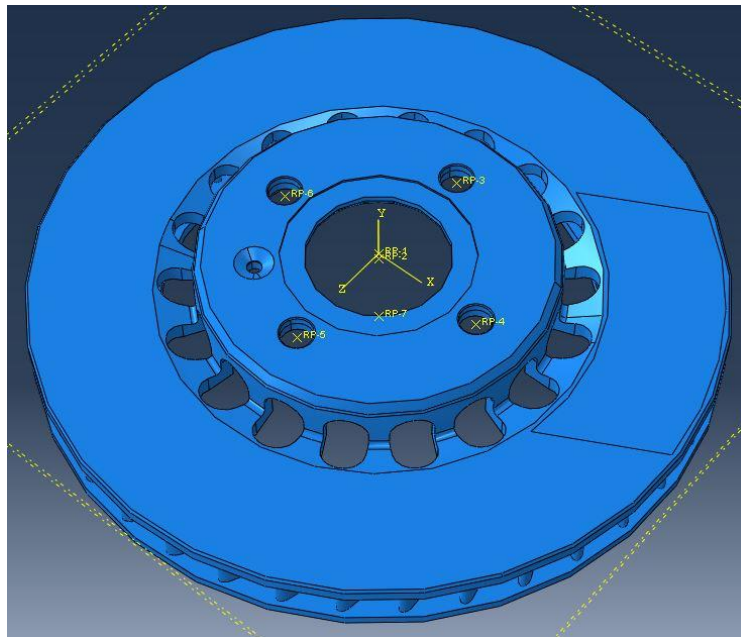


Figure 3-5 Reference Points and Cylindrical Coordinate Systems of Case 1

Thereafter, Multi Point Constraints (MPC) were created (Figure 3-6). The number of MPCs was equal to the number of reference points. The purpose of the MPCs was to establish a linkage between nodes on the disc and their closest reference point. The control point was always selected to be the reference point created in the previous step and the slave nodes were the nodes in the inner diameter of the holes of the disc located at the same axial level as the correlated reference point. The MPC type selected was “link” and the coordinate system for each MPC was the cylindrical system.

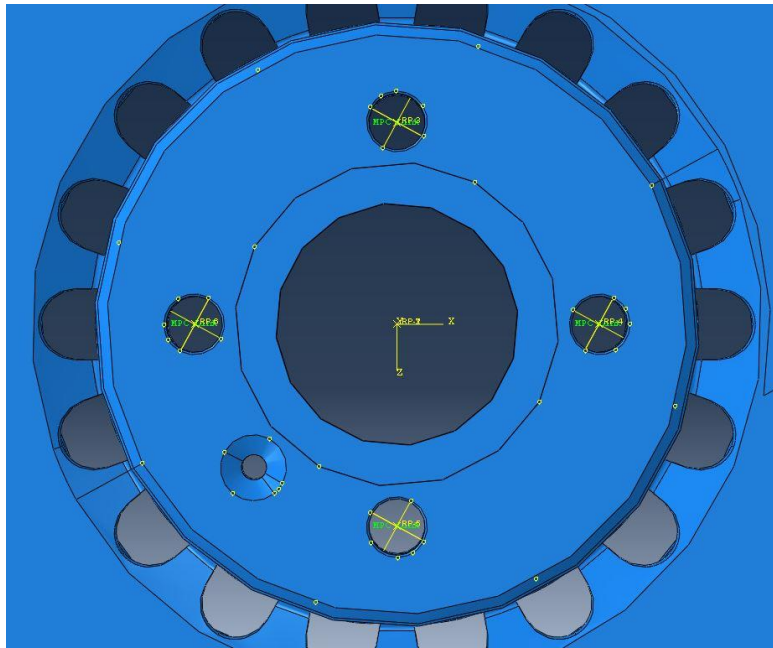


Figure 3-6 Multi Point Constraints

Upon the creation of all the MPCs, point loads were created in order to apply torque on the hub. The number of point loads was the same as the number of reference points created. Each point load was applied on a reference point and its direction was normal to the plane of the cylindrical coordinate system (Figure 3-7). Every point load created a torque and by superimposing all the torque values the total braking torque could be applied on the brake disc.

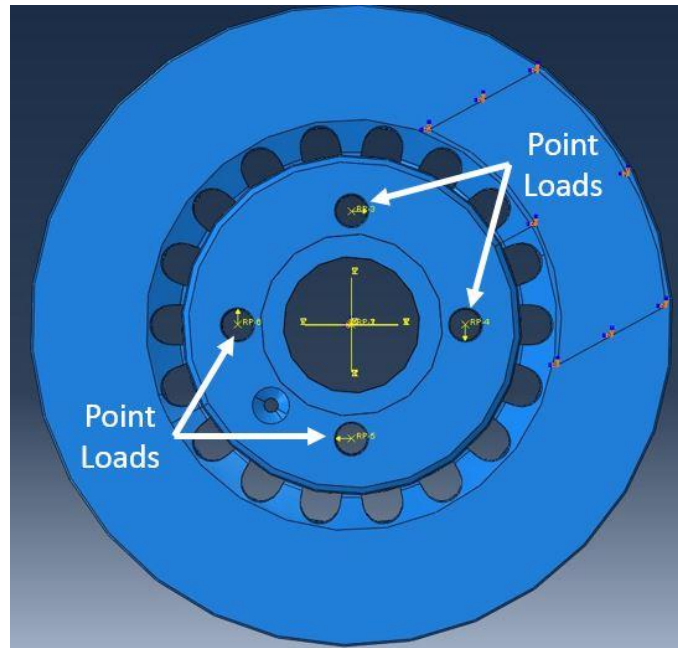


Figure 3-7 Point Loads – Case 1

Finally, boundary conditions not allowing the translation normal to their plane (in axial direction) were applied on the inboard and outboard surfaces of the hub (Figure 3-8). These boundary conditions ensure the realistic loading of the hub.

By viewing the modelling approach followed in this modelling case it can be said that the FE model represents the situation when the wheels of the car are locked under heavy braking.

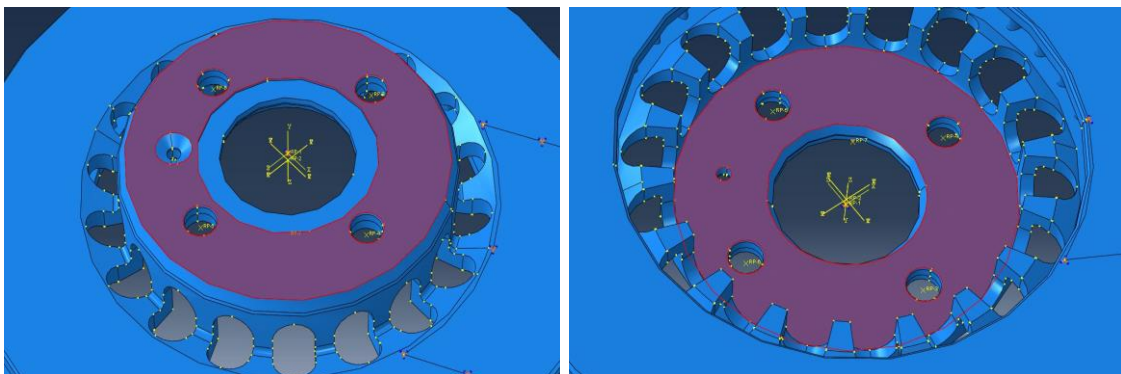


Figure 3-8 Boundary Conditions on the Hub – Case 2

3.4.2 Second Modelling Case

The motivation behind the second modelling case, also referred to as “Case 2”, was the same as in modelling Case 1; to create a FE model close to the real mechanical load which the brake discs are subjected to. Case 1 only modelled the scenario where the disc is clamped by the brake pads, but no rotation takes place. For the second modelling Case, more advanced modelling was envisaged. The idea was to apply the braking torque on the brake discs by modelling two brake pads being forced against the brake rotor while it is rotating (Figure 3-9).

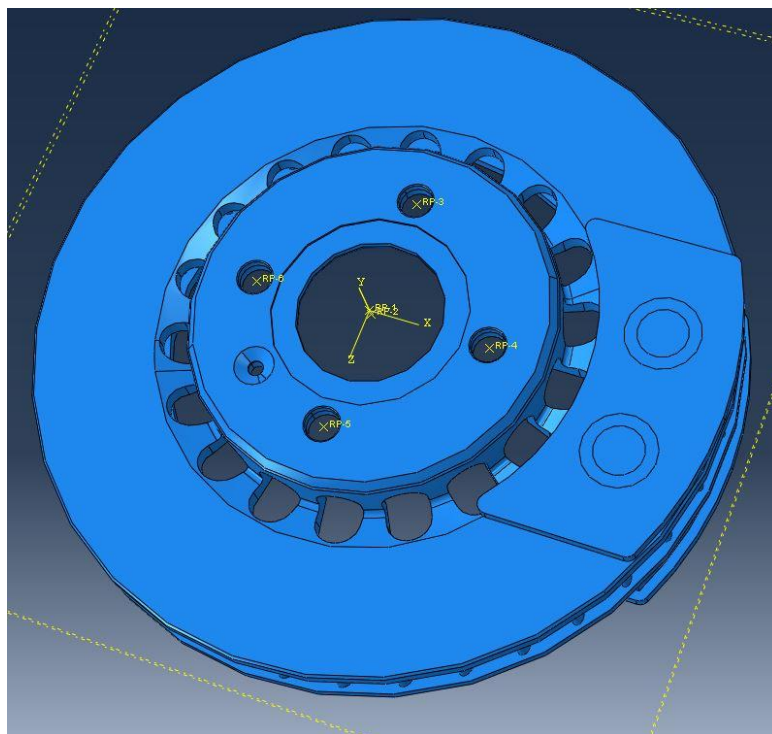


Figure 3-9 Model Set Up – Case 2

As it can be seen in Figure 3-9, in addition to the concept designs, brake pads were also included in the FE models. For the part of the pads with the frictional material, the same dimensions were used as the pad imprint of Case 1. The thickness of the friction material was 10mm, which corresponds to the new pads. No slots were created to reduce the complexity of the models. For the analyses, material properties found in the literature (Hassan, et al., 2013) were used and can be found in Appendix B.

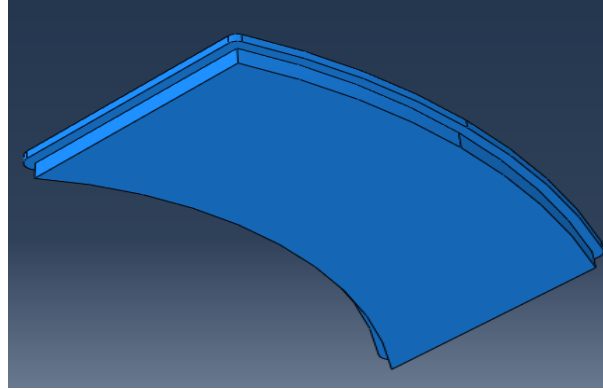


Figure 3-10 Brake Pads Model

The backplate of the pads also featured a simplified geometry. The thickness was 5mm, also a typical value of thickness, and there were no shims. The imprints of two pistons were sketched on each of the pads' backplates. The inner diameter of each piston imprint was 30 mm and the outer diameter was 40 mm. The distance between the centres of the pistons was 60 mm (Figure 3-11). The properties of steel used are shown in Appendix B.

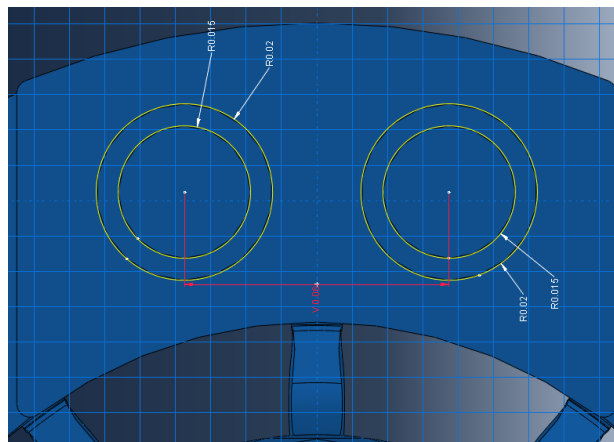


Figure 3-11 Brake Pads' Backplate Sketch

This modelling approach consisted of four steps that were created in Abaqus Pre-Post processor. These steps were: the Initial step, the Contact step, the Pressure Application step and finally the Rotation step. The way modelling Case 2 was created will be described by following these four steps.

Before starting with the Initial step though, the three components of the FE model, the brake disc concept design and the two brake pads, were positioned relatively

to each other. This was performed in the Assembly option of Abaqus CAE. The pads were placed in such a way that their inner edges had 2mm distance inwards from the inner diameter of the disc friction surfaces in radial direction and their outer edges had 2mm distance inwards from the outer diameter of the disc friction surfaces (Figure 3-12).

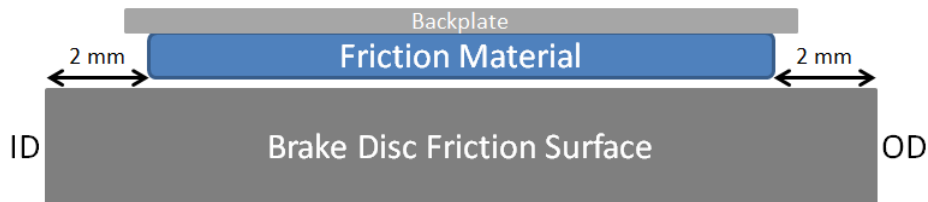


Figure 3-12 Radial Position of the Brake Pads to the Brake Disc

Additionally, the inboard pad and the outboard pad had 1 mm axial distance (gap) from the inboard friction surface and the outer friction surface respectively. Finally, both brake pads were set to be parallel to the friction surfaces of the disc concept designs.

In the Initial Step all the initial boundary conditions for Case 2 were created. The first boundary condition was applied on the brake disc design concept and blocked the translation on all directions and the rotation of the concept around its longitudinal and lateral axis. The rotation around its vertical axis though was not blocked. In order to create this boundary condition the following process was followed. A reference point was created on the axis of rotation of the brake disc at the same vertical level as the outboard surface of the disc's hub. Then, a "Rigid Body" constraint was created. The "Tie (nodes)" option was used (Hassan, 2009) and the region selected was the outboard hub surface of the disc. Finally, the reference point, which was created previously, was selected as "Reference Point". The boundary condition that blocked all the DoF apart from the rotation around the vertical axis was applied on the reference point used for the rigid body constraint.

Thereafter, the boundary conditions for the brake pads were created to keep them in the desired position and also simulate the pads being held by the abutments of the brake calipers. Boundary conditions were applied on the two lower corners

of the pad backplates to restrain the lateral translation (z direction). Boundary conditions were placed on the two upper corners to restrain both the lateral and the longitudinal (x direction) translation of the pad. The boundary conditions on the brake pads can be visualised in Figure 3-13.

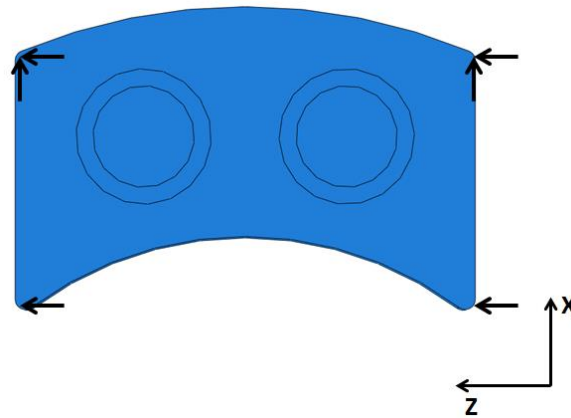


Figure 3-13 Brake Pads Boundary Conditions – Initial Step

Finally, for the Initial step, an interaction was created to model the contact between the brake discs and the pads. Case 2 incorporated both contact modelling and rotation as it was aiming to be as realistic as possible. In order to model contact, the interaction “Surface-to-surface contact” was used. The master surfaces were selected to be the friction surfaces of the brake disc design concepts and the slave surfaces were selected to be the brake pads’ surfaces. For the interaction to work, contact properties were defined. Since only mechanical analyses were taking place at this point, only the mechanical properties of “Tangential Behaviour” and “Normal Behaviour” were included in the contact properties. It is important to note that in “tangential behaviour” the value for the coefficient of friction used was 0.4. This is a typical value for the friction between cast iron disc and asbestos free brake pads (Tirovic, 2015). By defining the contact between the disc concept designs and the brake pads, all the parameters for the Initial step were created.

In the second step, called “Contact Step”, the brake pads were forced to come into contact with the disc friction faces. This was achieved by introducing a new boundary condition that was applied on the four points of the brake pads where

the boundary conditions of the Initial step were acting on. The type of boundary condition was “Displacement/Rotation”, the value was 1 mm and the direction of displacement was vertically towards the disc concepts (Figure 3-14). No other actions took place in Contact step.

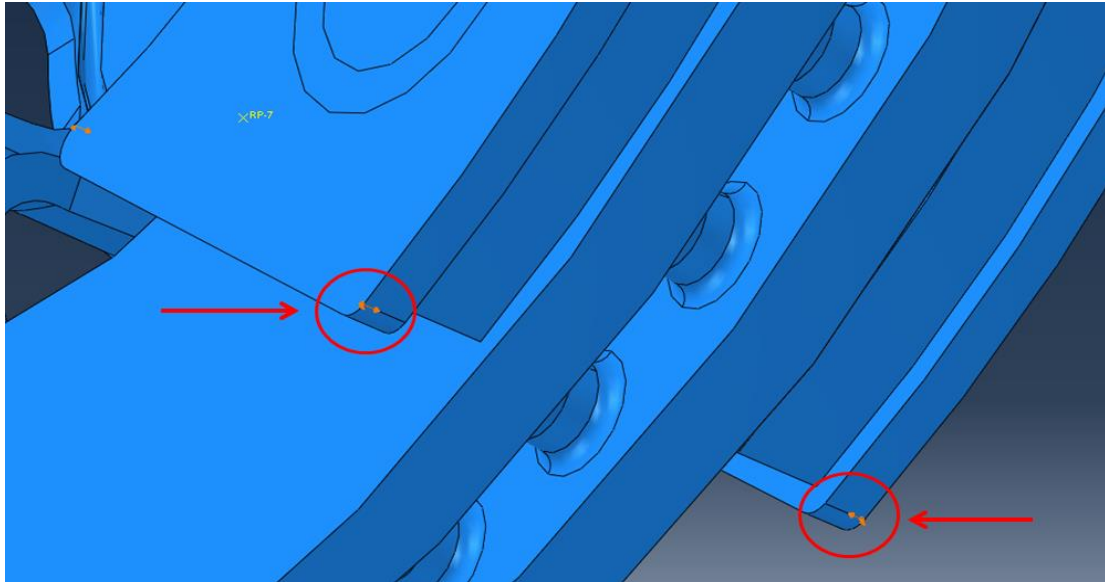


Figure 3-14 Brake Pads Boundary Conditions – Contact Step

The third step was named “Pressure Application”. In this step, hydraulic pressure was applied on the backplates of the brake pads in order to clamp the disc. Two actions took place in this step. The first was to apply a pressure type load on the backplates. Usually, the pressure in a vehicle hydraulic system when the brakes are applied is up to 100 bar (Tirovic, 2015). Taking this into consideration and also the pistons imprint areas, it was worked out that if 100 bar was the pressure on the hydraulic system, the pressure on the backplates contact surfaces should be 228 bar (Figure 3-15). The calculations can be found in Appendix C.

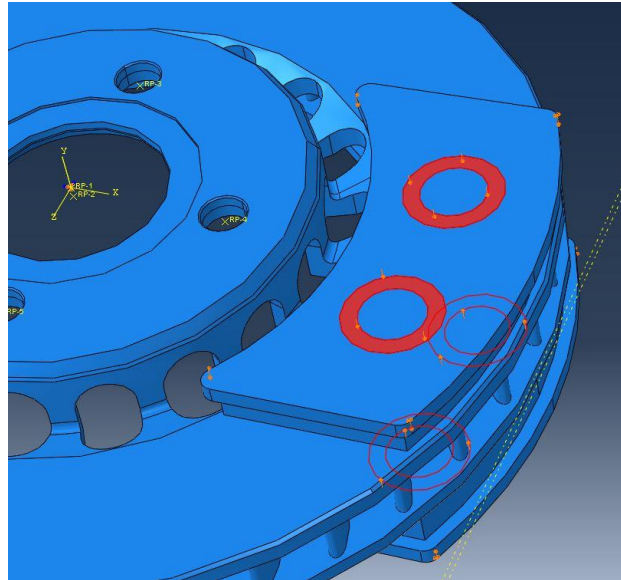


Figure 3-15 Application Area of Hydraulic Pressure

The second action in this step was to deactivate the boundary condition created in the second step. It is considered unrealistic to keep forcing the pads to have contact with the disc. The contact should be realised by the application of hydraulic pressure and the pad can freely move in axial direction constrained only by the abutments of the brake caliper.

The fourth and final step was the “Rotation Step”. In this step, the brake disc was forced to rotate while the brake pads were pressed against the friction surfaces. Two different methods to rotate the discs were identified. The first was to apply the “Rotation” boundary condition on the reference point created in the Initial step and input the value of rotational speed in rads/sec. The second method was to apply torque on the hub of the disc concept and force it to rotate. It was decided for the torque to be applied with the same method as in Case 1, described in Section 3.4.1 (Figure 3-16).

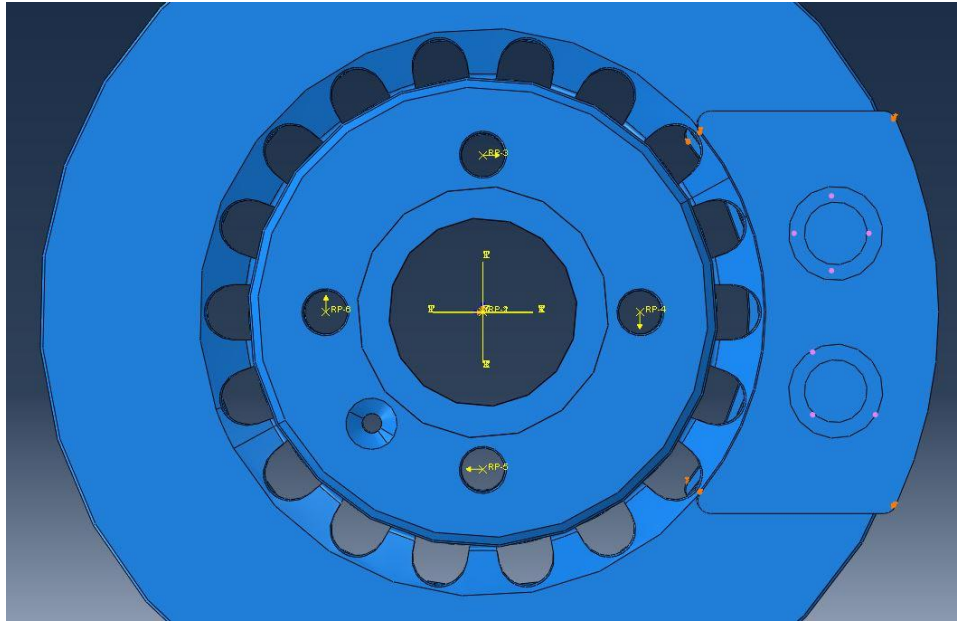


Figure 3-16 Torque Application – Case 2

At this point, it has to be mentioned that the size of the FE mesh was not approximately 3 mm as it was in the first modelling case. Due to the fact that Case 2 required high computational power, mainly as a result of contact modelling and application of rotation, and taking into consideration that the simulations ran on a local PC, there was a necessity to increase the size of elements. They are approximately 10 mm in the concept design and 5 mm on the brake pads. Smaller size elements were chosen on the brake pads as their friction surfaces were selected to be the “Slave surfaces” when the contact was defined in the Initial step.

3.5 Step 3 – Simplified Mechanical Analyses

Initially, simplified mechanical analyses took place in order to have the first indications of how the concept designs perform under mechanical loading. The assumptions made before the start of the modelling were the following:

- Resistances to motion and rotary inertia were neglected,
- Tyre adhesion coefficient was $\varphi = 0.8$,
- Disc was mounted on the front axle,
- Only mechanical load was simulated,
- The entire disc (360°) was used.

The worst case scenario was simulated with the torque input being the maximum braking torque the vehicle was able to produce at the limit of adhesion.

For the analyses, the two modelling cases presented in Section 3.4.1 and Section 3.4.2 were used. The braking torque generated by Lotus Elise at the limit of adhesion is presented in Table 3-3. The actual calculations are omitted here for the sake of brevity; they are presented in Appendix D.

Table 3-3 Braking Torque Input – Lotus Elise

Braking Torque	882 [Nm]
----------------	----------

General static linear analyses were performed with non-temperature dependent properties. All the material properties used in this stage of the analysis are presented in Appendix B. At this point of the investigation the free meshing technique was used. The elements' type was C3D4 tetrahedra elements and their size was approximately 3 mm.

3.5.1 Results and Discussion

The evaluation of the three conceptual designs for the Lotus Elise disc was conducted with regards to their predicted values of the equivalent von Mises stress and Maximum Principal stress. The benchmark arbitrarily was selected to be the Elise Version 1 and the equation used to calculate the difference between that conceptual design and the Elise Versions 2 and 3 was:

$$\begin{aligned} \text{Difference (\%)} &= \frac{\text{Final Value} - \text{Initial Value}}{\text{Initial Value}} \times 100 \end{aligned} \quad \begin{array}{l} \text{(3-1) Difference} \\ \text{Calculation} \end{array}$$

Table 3-4 presents a summary of the results when the first modelling case was used.

Table 3-4 Stress Results Summary – Case 1

Model	von Mises [MPa]	Diff. [%]	Max. Principal [MPa]	Diff. [%]
Elise Version 1	57	0.0	69	0.0
Elise Version 2	105	84.2	126	82.6
Elise Version 3	104	82.4	102	47.8

By reviewing the results, it can be said that the concept of brake discs with fingered hub design seems to be possible at least in terms of the mechanical loading. The Elise Version 1 features very low stress values for both von Mises and Maximum Principal stresses. Elise versions 2 and 3 feature higher stresses but they are below the UTS of grey cast iron by some margin.

Table 3-5 Stress Results Summary – Case 2

Model	von Mises [MPa]	Diff. [%]	Max. Principal [MPa]	Diff. [%]
Elise Version 1	104	0.0	117	0.0
Elise Version 2	167	60.5	198	69.2
Elise Version 3	167	60.5	163	39.3

The results of the second modelling case are illustrated in Table 3-5. Similarly to the first modelling case, Elise Version 1 exhibited the best mechanical behaviour with the Elise Version 3 being the second best with predicted stress levels slightly higher than the theoretical UTS of cast iron. The worst performing design was the Elise Version 2 with Maximum Principal stress 32% higher than the cast iron's UTS. However, the prominent conclusion from the results processing of Step 3 indicated the feasibility of the fingered hub disc design.

At that point of the investigation, one concept design was selected in order to be submitted to more advanced modelling with the use of linear-elastic properties,

material properties that introduced cast iron plasticity and a selection of different element types. Although Elise Version 1 featured the most promising performance compared to the other concepts, it was the Elise Version 3 which was selected to be the basis of further modelling. The creation of an actual working prototype for physical testing and validation of the FE models was deemed to be very important for the fingered hub concept feasibility investigation. The Elise Version 3 concept was the easiest to be manufactured compared to the other designs and this was the driving factor behind its selection. Also, the author considered the most robust design for service use.

3.6 Step 4 – Advanced Mechanical Modelling

In the fourth step of the investigation, alternative modelling techniques were used. Different ways of meshing the Elise Version 3, different element types and material properties that introduced plasticity were used.

The computational time for Case 2 was much longer than Case 1. Ultimately, a physical testing will be presented in Sections 5.3 and 5.4 and the setup of that testing was very similar to the setup of Case 1. As a result of all the above, Case 1 was selected to be used for the advance modelling of mechanical loading and for comparing the results from the FE modelling against the physical test.

The spokes connecting the hub with the friction faces is the most crucial area for the mechanical loading and became the focus of the investigation. Up to that point, simulations were conducted using the free meshing technique. Now, a more uniformly distributed mesh on the spokes was envisaged to capture more accurately the stress concentrations and stress gradients.

For the creation of a more uniform mesh, the spokes were partitioned, as it can be seen in Figure 3-17. All the other areas of the disc concept were not partitioned, as they were not crucial for the mechanical load modelling.

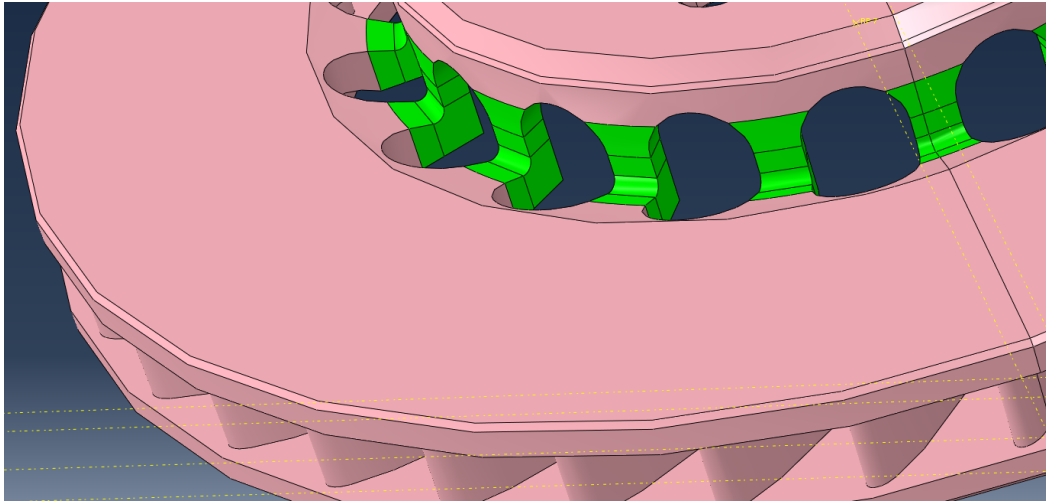


Figure 3-17 Partitioned Spokes

Two types of elements were applied on the spokes. Initially, tetrahedra C3D4 (Tet) elements were used. Although these elements were applied using again the free meshing technique, this time because the spokes were partitioned, the elements followed the geometry in a more homogeneous way (Figure 3-18). The use of elements with approximate size of 1 mm also aided the software to generate a uniform mesh. All the other areas of the disc concepts were meshed with elements of 3 mm approximate size to reduce the computational time.

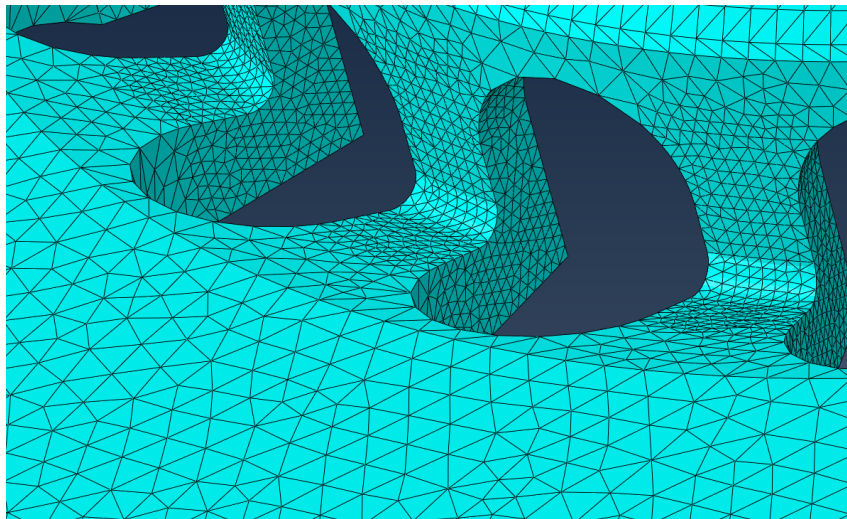


Figure 3-18 Tetrahedra 1mm Elements on the Spokes

The other type of elements used was the C3D8R hexahedra (Hex) elements with reduced integration, as bending is the dominant load. Similar to tetrahedra

elements, 1 mm size was selected for these elements as well. Since the spokes were partitioned, the meshing technique used this time was the “structured” technique. The combination of elements type, elements size and meshing technique resulted in elements following the geometry in a uniform way without any distortions (Figure 3-19).

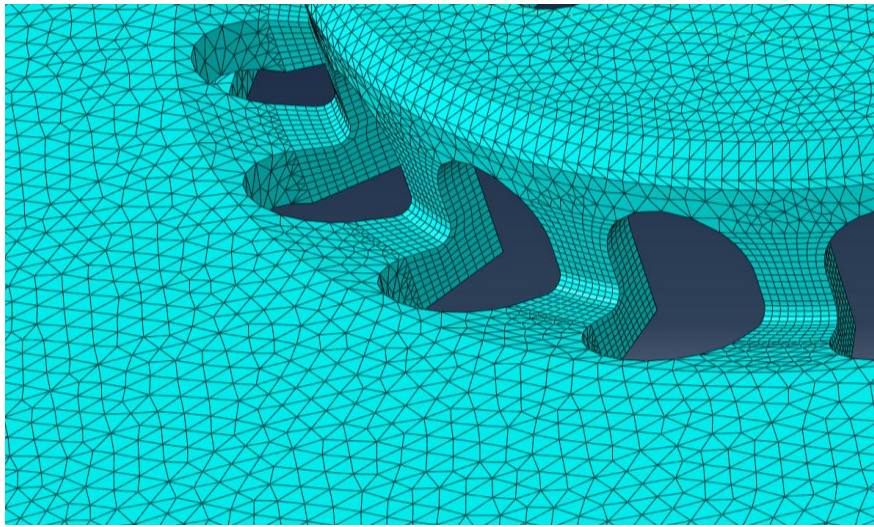


Figure 3-19 Hexahedra 1mm Elements on the Spokes

The C3D8R elements of approximately 1 mm of size and the structured technique were used only on the spokes. The rest of the disc was meshed with tetrahedra C3D4 elements of 3 mm size applied by using the free mesh technique.

Two sets of material properties were used. For the first set of simulations with tetrahedra and hexahedra elements the linear properties in the Appendix B were used. For the second set of simulations, plasticity was introduced in the models. The properties for these models came from data of limited circulation and can be found in Appendix E. The load input for the simulations was the maximum braking torque of 882 Nm that the Lotus Elise is able to produce. The load was applied in ambient temperature of 20° C.

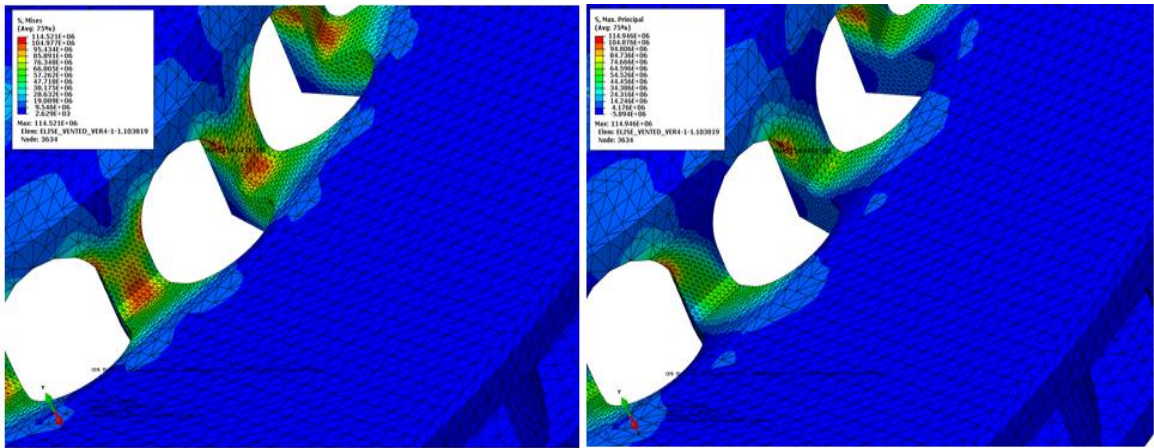
3.6.1 Results and Discussion on Advanced Mechanical Modelling

In this Section, the results of the advanced mechanical modelling are presented and discussed. The results for the tetrahedra elements with the linear and non-

linear set of properties are presented first. Then, the results for the hexahedra elements follow.

Table 3-6 Tetrahedra Elements with Linear Properties

Max. von Mises Stress [MPa]	Maximum Principal Stress [MPa]
114.5	114.9



a)

b)

Figure 3-20 von Mises Stress (a), Maximum Principal Stress (a) – Tetrahedra Elements with Linear Properties

The results in Table 3-6 differ from the results in Table 3-4 of Section 3.5.1, where again tetrahedra elements were used, but their approximate size was 3 mm. The results' difference between the different size of elements is relatively small, in the area of 10%. Most of the times elements of small size are used to better follow the solid geometry and capture the results more accurately. However, because smaller size of elements means larger number of elements in the same area, the stresses are overestimated by multiplying the numerical error. This is why there is an increase in both maximum values of von Mises and Maximum Principal stresses. This does not mean that the results coming from the 1 mm elements are wrong. The size of the elements that gives the most accurate results will have to be verified by comparing the FE results against the results from the actual testing.

Thereafter, the results from the tetrahedra elements with the incorporation of plasticity are presented in Table 3-7 and the stress plots for von Mises and Maximum Principal stresses are illustrated in Figure 3-21.

Table 3-7 Tetrahedra Elements with Non-linear Properties

Max. von Mises Stress [MPa]	Maximum Principal Stress [MPa]
112.6	113

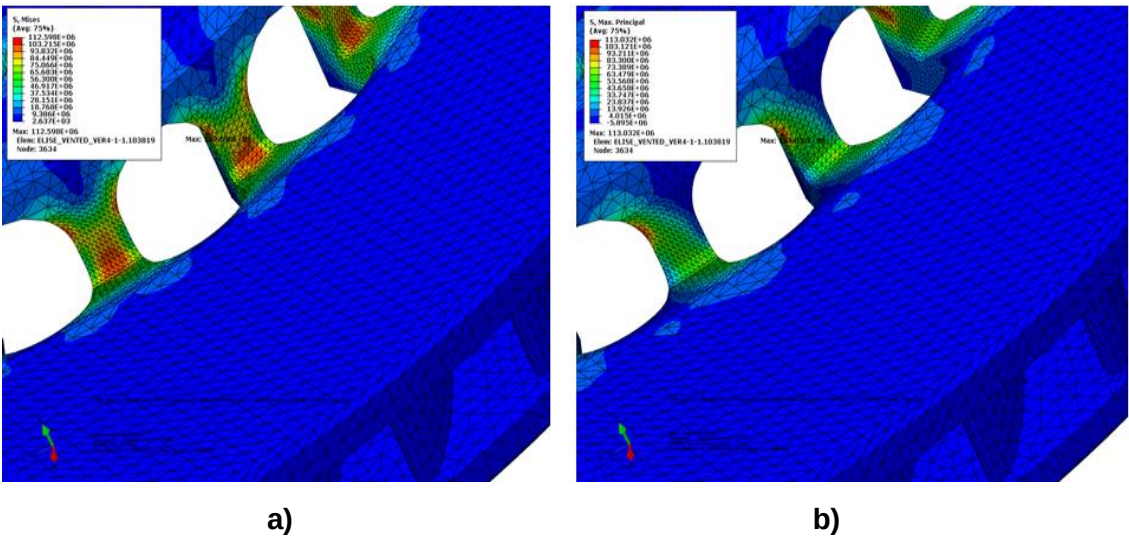


Figure 3-21 von Mises Stress (a), Maximum Principal Stress (b) – Tetrahedra Elements with Non-linear Properties

The stress values in Table 3-7 are marginally lower than the values of Table 3-6. This slight difference was expected with the introduction of plasticity and will be greater for higher values of braking torque.

Subsequently, the results of the hexahedra elements will be presented. Firstly, the outcomes from the use of linear properties and then, the outcomes from the set of non-linear properties. The maximum stress values of von Mises and Maximum Principal stress are displayed in Table 3-8. The areas of these values in the Elise Version 3 can be seen in Figure 3-22.

Table 3-8 Hexahedral Elements with Linear Properties

Max. von Mises Stress [MPa]	Maximum Principal Stress [MPa]
109.5	99

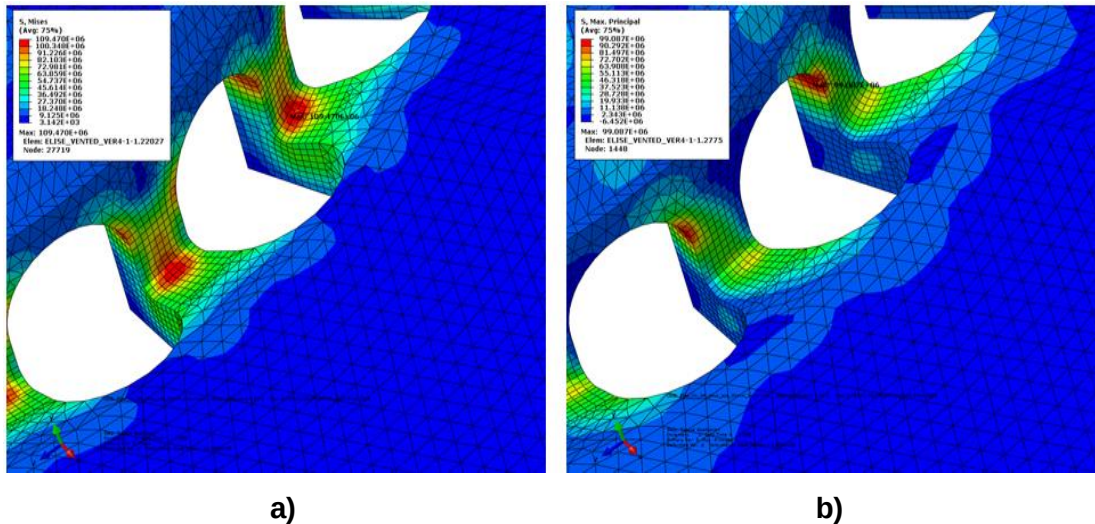


Figure 3-22 von Mises Stress (a), Maximum Principal Stress (b) – Hexahedral Elements with Linear Properties

Finally, the predicted maximum values for the von Mises and Maximum Principal stresses for the case of hexahedra elements with non-linear properties are presented in Table 3-9. These peak values of stress were acting on the areas shown in Figure 3-23.

Table 3-9 Hexahedral Elements with Non-linear Properties

Max. von Mises Stress [MPa]	Maximum Principal Stress [MPa]
109.5	98.2

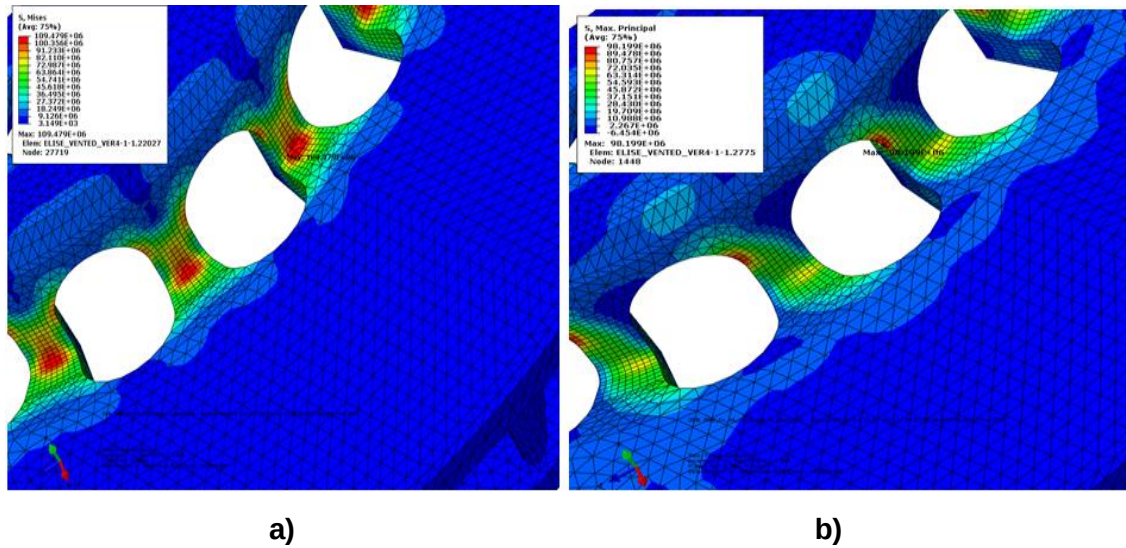


Figure 3-23 von Mises Stress (a), Maximum Principal Stress (b) – Hexahedral Elements with Non-linear Properties

No use of hexahedra elements had taken place prior to this set of simulations and the results can only be compared with the maximum stress values obtained using the tetrahedra elements. Table 3-10 features a comparison between the results from the tetrahedra elements and the hexahedra elements both for the linear analyses.

Table 3-10 Results Comparison Tetrahedra-Hexahedral Elements – Linear Properties

Simulation	von Mises [MPa]	Diff. [%]	Max. Principal [MPa]	Diff. [%]
Tet – Linear	114.5	0	114.9	0
Hex – Linear	109.5	-4.3	99	-13.8

The same comparison, but for the use of non-linear material properties, is illustrated in Table 3-11.

Table 3-11 Results Comparison Tetrahedra-Hexahedral – Non-linear Properties

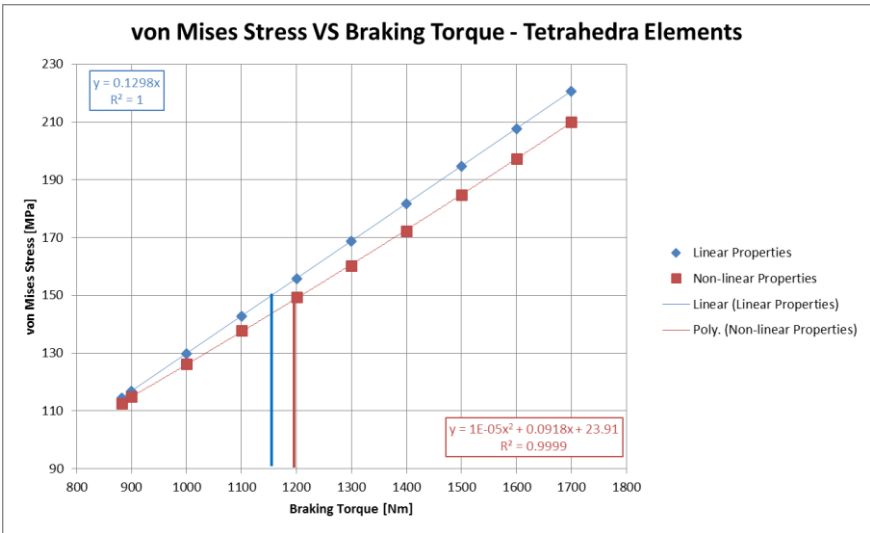
Simulation	von Mises [MPa]	Diff. [%]	Max. Principal [MPa]	Diff. [%]
Tet – Non-linear Plastic	112.6	0	113	0
Hex – Non-linear Plastic	109.5	-2.7	98.2	-13.1

From the results of Table 3-10 and Table 3-11 it can be said that a marginal decrease in the maximum von Mises criterion is observed. For the maximum value of Maximum Principal stress a higher reduction is noticed, but not at a major level. If a comparison is to take place between the results from the simulations with linear and non-linear properties using hexahedra elements, the differences are insignificant.

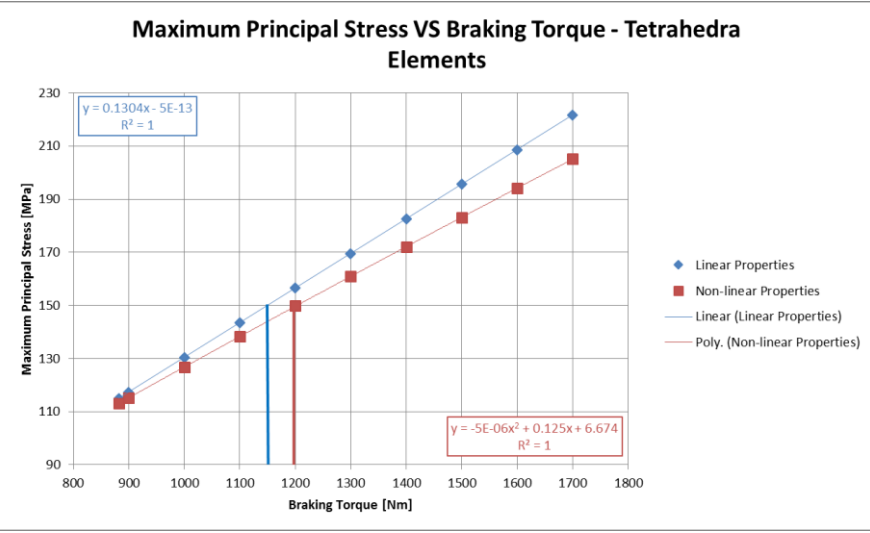
By reviewing all the results of this Section, it can be concluded that the concept of brake discs with fingered hub design seems to be feasible at least for the case of lightweight vehicles. This is the outcome from the set of simulations that were described in Section 3.6. The focus that was given on the critical area of the spokes, the investigation of different mesh techniques, mesh types, element sizes and finally, the different sets of material properties resulted in stress values that gave confidence that the Elise Version 3 concept would be able to withstand the mechanical loading of the Lotus Elise at the limit of adhesion. In all models, the von Mises and Maximum Principal stress concentrations were found to be in the same areas. The areas where the maximum values of von Mises stresses were acting are areas that a brake disc is not prone to fail. The areas, where the maximum values of the Maximum Principal stresses (the stress criterion which is the most critical for causing mechanical failure) were acting are prone to cause failure of brake discs, but the values recorded are below the UTS of cast iron, so failure is unlikely to occur.

The final investigation within the framework of mechanical modelling was to re-run the simulations of this Section by gradually increasing the braking torque applied on Elise Version 3. This investigation took place in order to predict at

which value of braking torque the disc concept is more likely to fail mechanically. Figure 3-24 presents the results of this investigation for the use of tetrahedra elements. The von Mises maximum stresses versus the braking torque applied are illustrated on the upper graph and the Maximum Principal stresses versus the braking torque applied are displayed in the lower graph.



a)



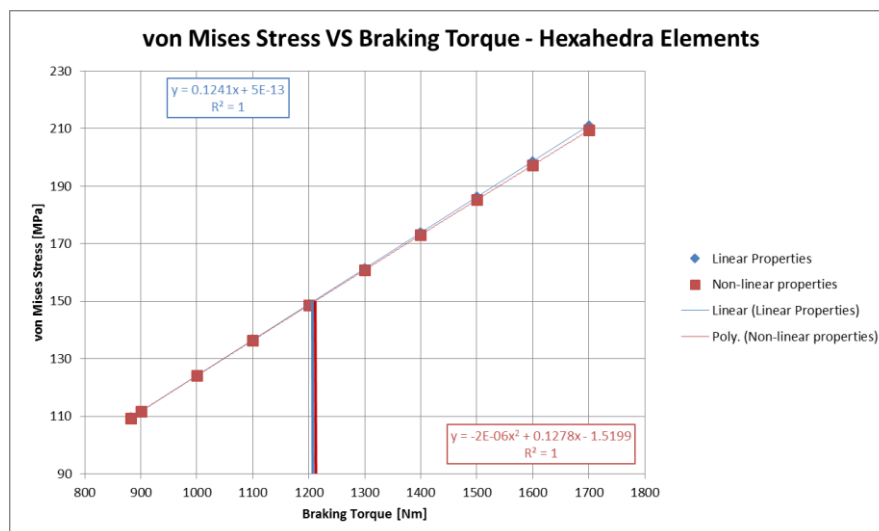
b)

Figure 3-24 von Mises Stress vs Braking Torque (a), Maximum Principal Stress vs Braking Torque (b) – Tetrahedra Elements

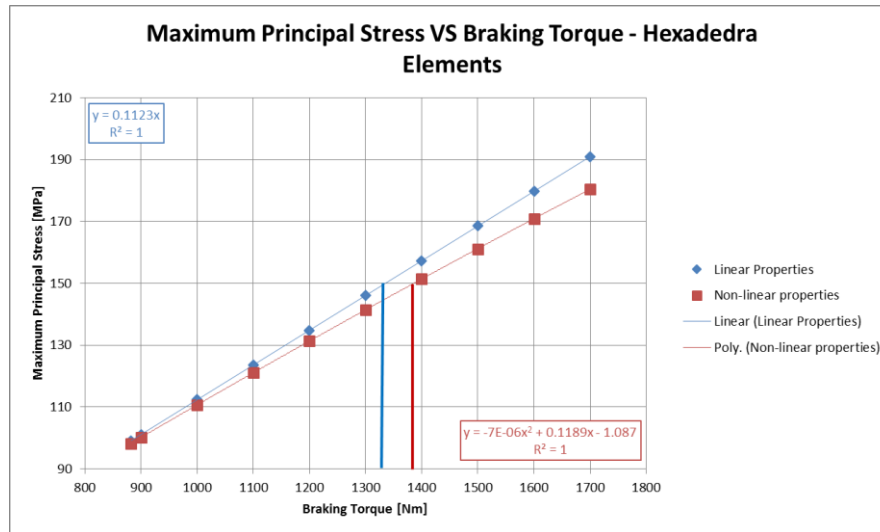
The curves of the predicted von Mises and Maximum Principal stresses are similar. The stress criterion most likely to cause failure, as stated previously, is the Maximum Principal stress. If the UTS value of 150 MPa for grey cast iron is to be used as the benchmark for material failure, the Elise Version 3 will fail at around 1150 Nm of braking torque when linear properties are used, and 1200 Nm, when non-linear elastoplastic properties are used.

Figure 3-25 illustrates the results for the simulations where hexahedra elements were used.

The curves of the von Mises stresses when using linear and non-linear properties are almost identical. For the Maximum Principal stresses the curves slopes are slightly different with the stresses being lower for the simulations with non-linear elastoplastic properties. According to the predictive Maximum Principal stresses, the Elise Version 3 will fail under a braking torque of around 1350 Nm, when linear properties are used and around 1390 Nm for the non-linear elastoplastic set of properties. This last value of braking torque is 57% higher than the maximum braking torque a fully laden Lotus Elise can produce at the limit of adhesion.



a)



b)

Figure 3-25 von Mises Stress vs Braking Torque (a), Maximum Principal Stress vs Braking Torque (b) – Hexahedral Elements

3.7 Chapter Summary

In this Chapter, the approach to identify the feasibility of lightweight brake discs with fingered hub design was described. Attempts were initially conducted in brake rotors for a mid-size family sedan, but the predicted stresses showed that it was not possible to have lightweight discs of such a design in vehicles of this segment.

Thereafter, the study was focused on investigating the feasibility of the fingered hub design for brake discs in lightweight sport vehicles like the Lotus Elise. Three conceptual designs were created, all featuring weight reductions compared to the standard Elise disc, close to 10%. Furthermore, two FE modelling techniques were developed, aiming to simulate the torque application and predict the maximum stresses on the prototypes caused by the maximum braking torque that the vehicle was able to produce. The conducted simulations included linear material properties, properties with plasticity, different element sizes as well as tetrahedra and hexahedra element types.

By evaluating the FE results, it was concluded that brake discs with fingered hub design are feasible for lightweight vehicles in terms of mechanical loading.

Additionally, it was realised that the criterion most likely to cause fracture to the discs was the Maximum Principal stress due to the areas where its maximum values were predicted to be and because it is the criterion of failure for brittle materials, such as cast iron. Further FE simulations were planned to evaluate the behaviour of the Elise Version 3 prototype when subjected to thermal loading and will be presented in the following Chapter.

4 LIGHTWEIGHT FINGERED BRAKE DISC INVESTIGATION – THERMOMECHANICAL MODELLING

4.1 Background

In Section 3.6 it was shown that it is feasible for brake discs with fingered hub design to withstand mechanical loads if these discs are used on lightweight vehicles, which produce relatively low braking torque. In this Chapter, advanced thermal analyses will be presented and their results will be discussed in order to cover both loading cases (mechanical and thermal) that the brake discs are subjected to.

4.2 Step 5 – Thermomechanical Analyses

At this stage of the investigation, non-linear elastic and non-linear elastoplastic analyses were conducted. Tetrahedra elements with the free meshing technique and also hexahedra elements with the structured and sweep techniques were used. The size of the elements in all cases was approximately 1mm and the following assumptions were made:

- Resistances to motion and rotary inertia are neglected,
- All braking energy is absorbed by the disc,
- Heat is generated uniformly over the disc friction surfaces,
- The initial disc temperature is 20°C,
- Heat dissipation is neglected due to short braking times,
- Tyre adhesion coefficient: $\varphi = 0.8$,
- Maximum deceleration of $\alpha = 7.85 \text{ m/s}^2$,
- Disc is mounted on the front axle.

In order to decrease the computational demand for the simulations, the analyses were carried out using only the smallest repeating segment in terms of the disc's hub. The smallest repeating segment was found to be 10 degrees. In this case, this method may imply a degree of inaccuracy for the thermal capacity of the segment because the discs featured curved vanes, but the important thermal stresses on the spokes will be accurately predicted.

The data from the Lotus Elise MY 2013 (Table 3-1) was used to compute the load inputs for the thermal modelling. The braking energy and the heat flux input

values are presented in Table 4-1, whilst the actual calculations were omitted here for the sake of brevity. They can be found in Appendix F.

Table 4-1 Load Inputs – Elise Version 3

Braking Force [kN]	7.3
Braking Duration [s]	7.26
Kinetic Energy [MJ]	1.2
Max. Power (per disc at the front axle) [kW]	335
Max. Heat Flux (per disc at the front axle) [MW/m ²]	2.21

The heat flux input used for the analyses has a bi-linear profile (Figure 4-1). Taking into account the time delay in both the driver's reaction and the pressure increase in the hydraulic circuit, the peak flux assumed to take place after 0.5 sec into the braking event. Then, it is decreasing linearly to zero after 7.3 seconds due to the vehicle velocity decrease to zero. The more rounded value of 7.3 seconds was used in the FE modelling instead of 7.26 seconds in order to create time steps of 0.1 sec resulting in the simulations being more stable.

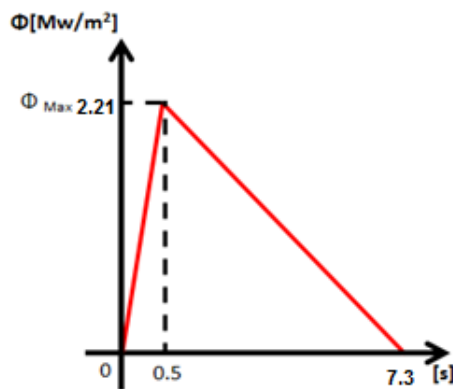


Figure 4-1 Bi-Linear Heat Flux Input – Elise Case Study

Heat flux input was assumed to be uniformly distributed over the swept area of the friction faces, as shown in Figure 4-2.

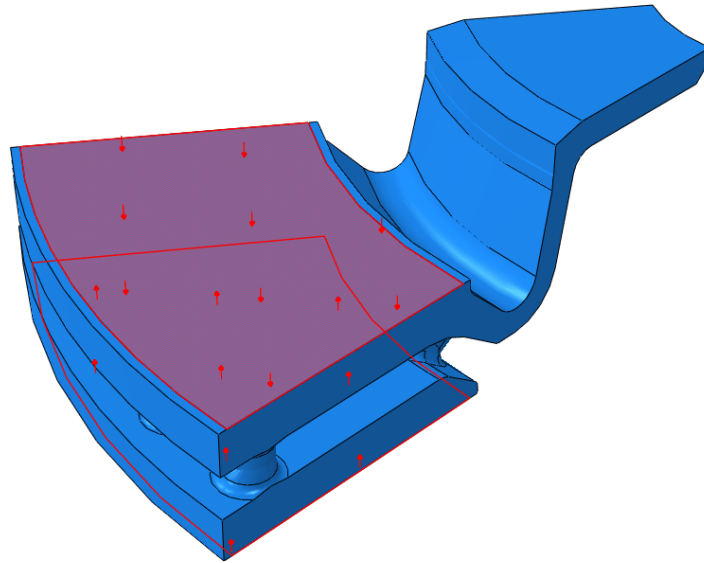


Figure 4-2 Uniform Heat Input

Boundary Conditions had to be implemented properly in order to compensate for the fact that only a segment of the entire geometry was modelled. In all cases the hub was assumed to be rigidly constrained for all three Degrees of Freedom (DoF). Hence, the mounting faces were chosen for the application of this constraint (Figure 4-3, a).

Thereafter, in order to set up symmetry boundaries, which will enable each modelled segment to behave as a representative portion of the whole disc, one cylindrical (ρ, θ, z) coordinate system was created with its origin at the centre of rotation, ρ was its friction face radius, θ the swept angle and (+ve) z the axis of rotation.

In both coordinate systems, the translation normal to their plane was constrained, and additionally, the rotation for all DoF was also constrained, apart from the one normal to their faces (Figure 4-3, b and c).

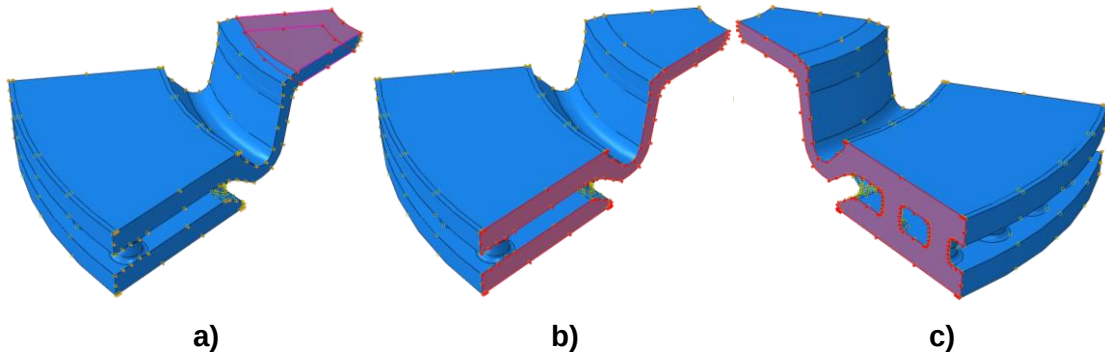


Figure 4-3 Boundary Conditions on the Modelled Segments, a)Flange, b) and c) Segment Boundaries

All the conducted analyses were fully-coupled thermal and mechanical for temperature, thermal stress and thermal displacements prediction. With regards to the models using tetrahedra elements, the “free mesh” meshing technique was used, created by the algorithms of the software. The elements were linear tetrahedra C3D4T and their approximate size was 1mm. For the analyses with hexahedra elements, after properly partitioning each part, the “structured” meshing technique was used with C3D8TR elements of 1 mm approximate size.

4.3 Results and Discussion on Thermal Modelling

The results for the thermal modelling will be presented in the following order. Firstly, the results from the use of tetrahedra elements will be shown for the three different sets of properties that were used: the linear elastic, the non-linear elastic and the non-linear elastoplastic. Thereafter, the results obtained with the use of hexahedra elements will be illustrated again for the linear elastic, the non-linear elastic and the non-linear elastoplastic sets of properties.

The comparison criteria used were the maximum temperature at the end of braking, the maximum von Mises stress at the end of braking, the maximum value of von Mises stress during the entire brake application and the moment in time when the peak stress occurred. Furthermore, the radial displacement and coning at the end of brake application were recorded and will be presented below.

The results of Elise Version 3 concept with the use of tetrahedra elements and linear properties are shown in Table 4-2.

Table 4-2 Elise Version 3 – Tetrahedra Linear Properties

Elise Version 3 – Tetrahedra Elements and Linear Properties	
Temperature [°C]	316.3
von Mises [MPa]	867.8
Maximum von Mises [MPa]	878 @ 6.5 sec
Radial Displacement [mm]	0.45
Coning [mm]	0.16

The values of temperature, radial displacement and coning are considered acceptable. On the contrary, the values of von Mises at the end of braking and the maximum von Mises during the braking are considered quite high. The peak von Mises stress was found at the area where the hub is connected to the fingers in the inboard side of the disc (Figure 4-4).

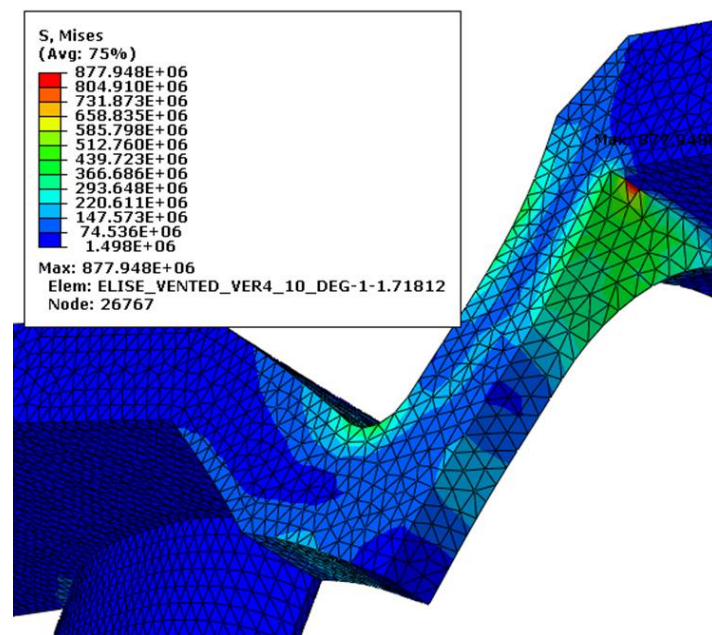


Figure 4-4 Elise Version 3 (Tet-Elements Linear Properties) Maximum von Mises

For the creation of CAD designs, which formed the basis of FE analyses, no manufacturing tolerances were used like fillets or chamfers. To investigate whether or not the Maximum Principal stress predicted was a result of the lack of

manufacturing tolerances, a chamfer of 0.5 mm was added between the hub and the fingers in the inboard side of the Elise Version 3 concept and the simulation was conducted again. The predicted maximum von Mises during the brake application decreased by 20% (700 MPa) compared to the value of von Mises without the chamfer (Figure 4-5). This result led to the conclusion that the existence or not of the chamfer was not the primary factor for the presence of such an increased value for von Mises stress. Keeping this in mind, the simulations that followed did not incorporate any chamfer between the hub and the fingers in the inboard side of the design concept.

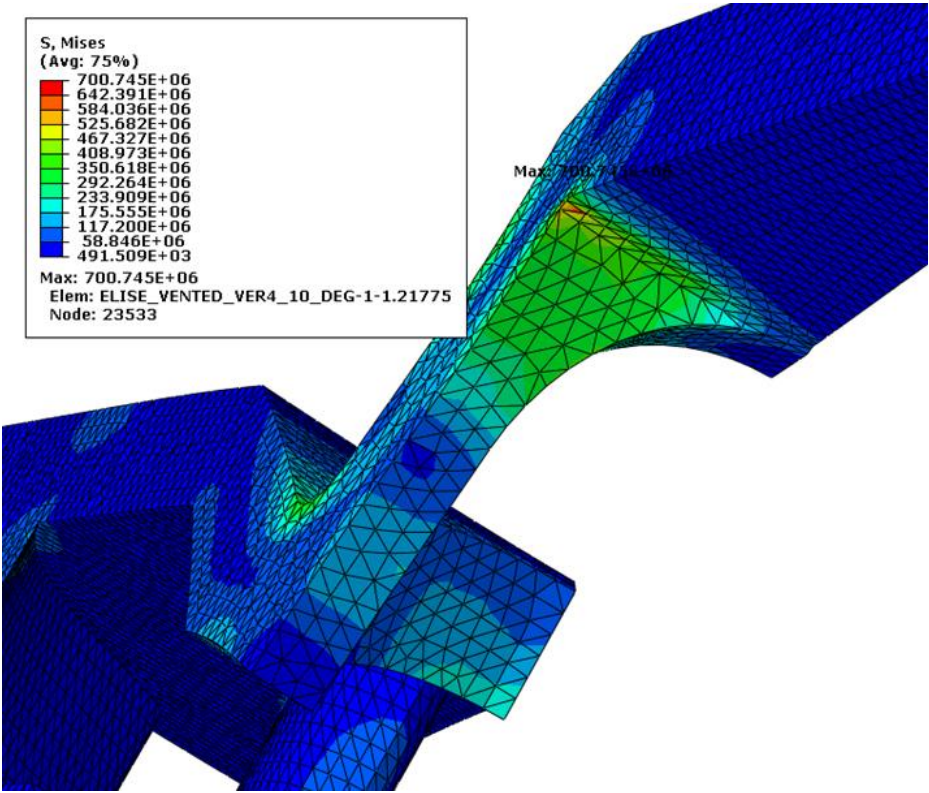


Figure 4-5 Elise Version 3 (Tet-Elements Linear Properties w/ Chamfer) Maximum von Mises

The results with the use of tetrahedra elements with non-linear elastic properties are shown in Table 4-3.

Table 4-3 Elise Version 3 – Tetrahedra Non-linear Elastic Properties

Elise Version 3 – Tetrahedra Elements and Non-linear Elastic Properties	
Temperature [°C]	291.7
von Mises [MPa]	892.1
Maximum von Mises [MPa]	903.4 @ 6.4 sec
Radial Displacement [mm]	0.47
Coning [mm]	0.17

The reduction in the maximum temperature for the set of properties that are temperature dependent compared to the temperature of Table 4-2 with temperature independent properties was expected, resulting from the thermal capacity being increased as the temperature increases. The value of von Mises stress at both the end of braking and the maximum von Mises during braking was increased remaining at considerably high levels. The area where the peak values of von Mises stresses were predicted was the same as with the linear analyses. The values of radial displacement and coning were similar to the ones from the previous analysis and considered as good performance from the Elise Version 3.

Finally, for the use of tetrahedra elements, the results with the use of non-linear elastoplastic properties are presented in Table 4-4.

Table 4-4 Elise Version 3 – Tet-Elements Non-linear Elastoplastic Properties

Elise Version 3 – Tet Elements and Non-linear Elastoplastic Properties	
Temperature [°C]	291.7
von Mises [MPa]	324.4
Maximum von Mises [MPa]	331.5 @ 6.4 sec
Radial Displacement [mm]	0.47
Coning [mm]	0.16

The set of properties used to predict the results in Table 4-4, being temperature dependent, gave a predicted maximum temperature exactly the same as in Table 4-3.

The noticeable difference between the results from the non-linear elastoplastic compared to the results from the linear and non-linear elastic analyses was for the prediction of von Mises stresses at the end of braking and the maximum von Mises during the braking application (Table 4-5).

Table 4-5 von Mises Comparison for Different Properties (Tet Elements)

Simulation	von Mises [MPa]	Diff. [%]	Max. von Mises [MPa]	Diff. [%]
Tet – Linear Elastic	867.8	-	878	-
Tet – Non-linear Elastic	892.1	2.8	903.4	2.9
Tet – Non-linear Elastoplastic	324.4	-62.6	331.5	-62.2

As a result of the introduction of plasticity, the von Mises stresses in Table 4-4 are considerably lower. In addition to the reduction of von Mises values, the area where the peak stress was predicted changed compared to the previous analyses. For the non-linear elastoplastic analysis, the maximum stress was predicted to occur on the upper part of the finger close to the hub and on the outboard side of the disc (Figure 4-6). The values of radial displacement and coning remained similar to the values from the previous analyses, as the most influencing factor for them is the geometry of the design. Since the geometry remains unaltered, the differences for radial displacement and coning for the different sets of properties are marginal.

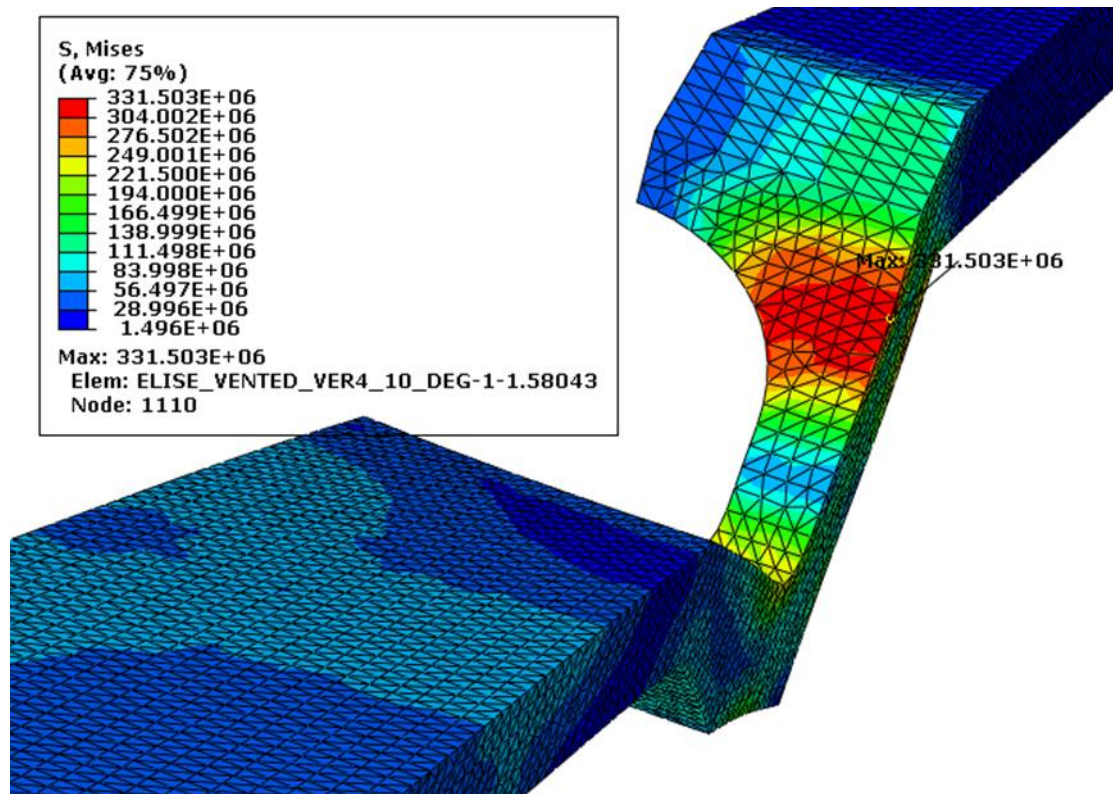


**Figure 4-6 Elise Version 3 (Tetrahedra Non-linear Elastoplastic Properties)
Maximum von Mises**

With regards to the use of hexahedra elements the results from the analysis using linear elastic set of properties are illustrated in Table 4-6.

Table 4-6 Elise Version 3 – Hexahedra Elements Linear Properties

Elise Version 3 – Hexahedra Elements and Linear Properties	
Temperature [°C]	333.6
von Mises [MPa]	957.9
Maximum von Mises [MPa]	970.4 @ 6.5 sec
Radial Displacement [mm]	0.45
Coning [mm]	0.16

Comparing the results when using tetrahedra elements to the results where hexahedra elements were applied (Table 4-7), an increase in maximum

temperature and von Mises stress is observed. The maximum values for von Mises stress appeared on the same spot in which they were found when the tetrahedra elements were used and this is the connection of the disc's hub and fingers on the inboard side (Figure 4-7).

Table 4-7 Thermal Analyses Comparison Tet-Hex Elements – Linear Properties

Parameter	Tet – Linear	Hex – Linear
Maximum Temp. [°C]	316.3	333.6
Difference [%]	-	5.4
von Mises [MPa]	867.8	957.9
Difference [%]	-	10.4
Max von Mises [MPa]	878	970.4
Difference [%]	-	10.5
Radial Displacement [mm]	0.45	0.45
Difference [%]	-	0.0
Coning [mm]	0.16	0.16
Difference [%]	-	0.0

No alteration in the geometry of the design allowed the values of radial displacement and coning to remain identical.

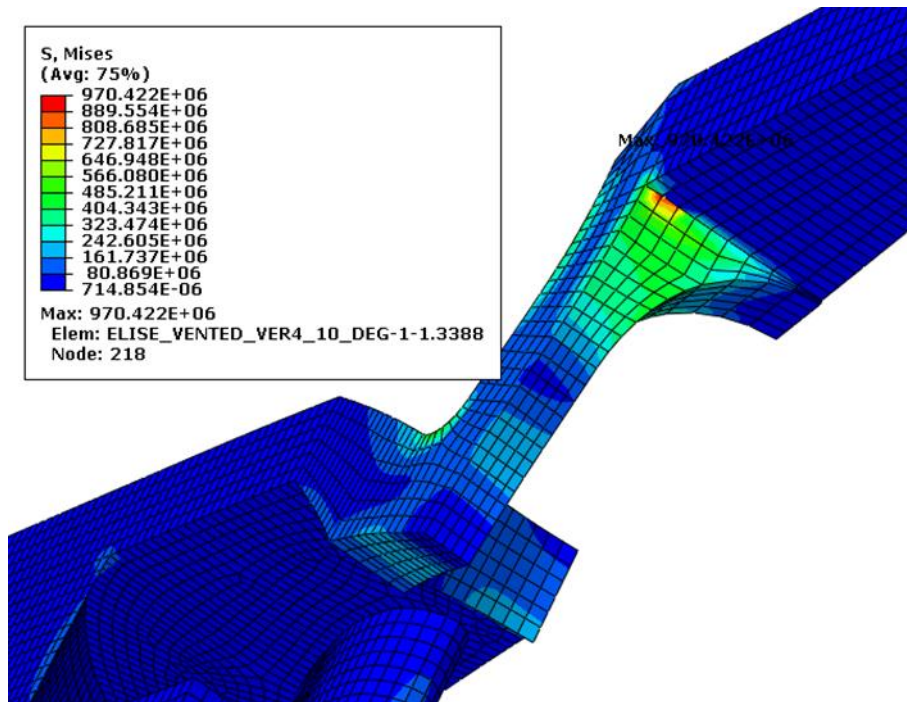


Figure 4-7 Elise Version 3 (Hex-Elements Linear Properties) Maximum von Mises

The results from the analysis using the non-linear elastic set of properties are shown in Table 4-8.

Table 4-8 Elise Version 3 – Hexahedra Elements Non-linear Elastic Properties

Temperature [°C]	306.3
von Mises [MPa]	1099
Maximum von Mises [MPa]	1114 @ 6.5 sec
Radial Displacement [mm]	0.45
Coning [mm]	0.17

As happened when tetrahedra elements were used, with the introduction of temperature dependent properties the value of temperature at the end of the braking application was reduced compared to the linear elastic analysis.

On the other hand, when comparing the results of Table 4-8 and the ones of

Table 4-3 where non-linear elastic properties were used again but with tetrahedra elements, a noticeable increase in von Mises stress both at the end of braking and also the maximum value of von Mises stress during braking is observed (Table 4-9).

Table 4-9 Thermal Analyses Comparison Tet-Hex Elements – Non-linear Elastic Properties

Parameter	Tet – Non-linear Elastic	Hex – Non-linear Elastic
Maximum Temp. [°C]	291.7	306.3
Difference [%]	-	5.0
von Mises [MPa]	892.1	1099.9
Difference [%]	-	23.2
Max von Mises [MPa]	903.4	1114
Difference [%]	-	23.3
Radial Displacement [mm]	0.47	0.45
Difference [%]	-	-4.2
Coning [mm]	0.17	0.17
Difference [%]	-	0.0

The peak von Mises stress is found in the same area as in linear analysis, where the hub and the finger are connected at the inboard side of the disc. The predicted values for radial displacement featured a marginal increase compared to the linear analysis with hexahedra elements, and coning remained unchanged.

Finally, the results from the non-linear elastoplastic analysis are presented in Table 4-10.

Table 4-10 Elise Version 3 – Hexahedra Elements, Non-linear Elastoplastic Properties

Temperature [°C]	306.4
von Mises [MPa]	351.8
Maximum von Mises [MPa]	359.7 @ 6 sec
Radial Displacement [mm]	0.47
Coning [mm]	0.16

The maximum temperature at the end of braking was predicted to be almost identical to the analysis using non-linear elastic properties since the same temperature dependent properties were used.

As expected with the introduction of plasticity there was a considerable reduction in von Mises stresses. In Table 4-11, the von Mises stresses are compared for the three different sets of properties with the use of hexahedra elements.

Table 4-11 von Mises Comparison for Different Properties (Hex Elements)

Simulation	von Mises [MPa]	Diff. [%]	Max. von Mises [MPa]	Diff. [%]
Hex – Linear Elastic	957.9	-	970.4	-
Hex – Non-linear Elastic	1099	14.7	1114	14.8
Hex – Non-linear Elastoplastic	351.8	-63.27	359.7	62.9

The area where the highest values of von Mises stress were predicted was on the upper parts of the fingers close to the hub at the outboard side of the disc (Figure 4-8). Similar prediction was made for the non-linear elastoplastic analysis, but with the use of tetrahedra elements.

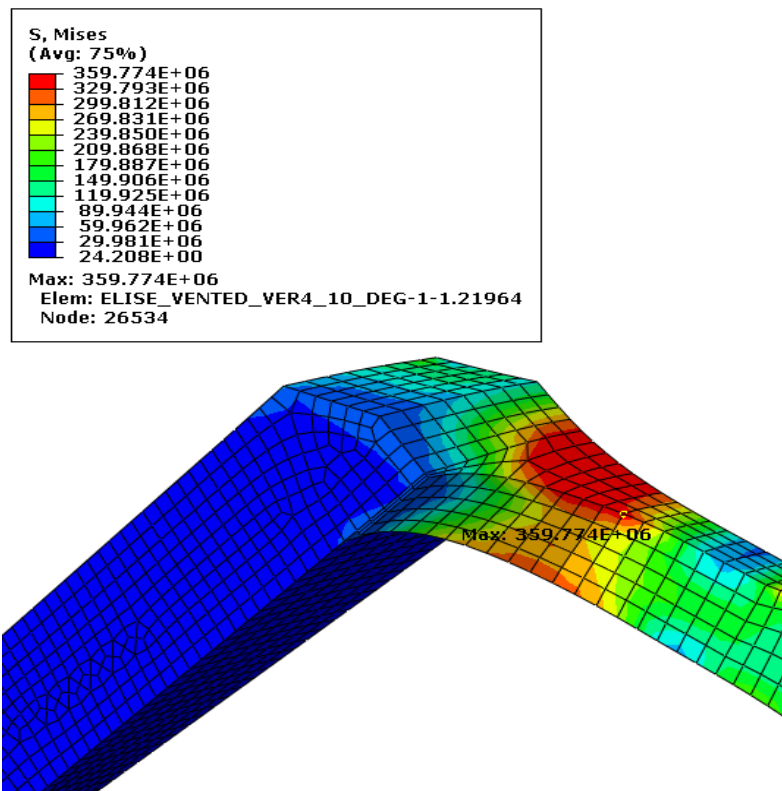


Figure 4-8 Elise Version 3 (Hexahedra Elements, Non-linear Elastoplastic Properties) Maximum von Mises

With regards to the values of radial displacement and coning, these were similar to the results from all the previous thermal simulations of this Section. With the Elise Version 3's geometry remaining unaltered it was expected that only marginal differences to the values of radial displacement and coning would take place.

By comparing the results from the two non-linear elastoplastic analyses; the one where tetrahedra elements were used against the one with hexahedra elements, it is observed that the discrepancies between the results are small. In fact, they are the smallest compared to the two previous similar comparisons (Table 4-7 and Table 4-9). The temperature values differ by 5%, the von Mises stresses by less than 10% and the values of radial displacement and coning are the same.

Table 4-12 Thermal Analyses Comparison Tet-Hex Elements – Non-linear Elastoplastic Properties

Parameter	Tet – Non-linear Elastic	Hex – Non-linear Elastic
Maximum Temp. [°C]	291.7	306.4
Difference [%]	-	5
von Mises [MPa]	324.4	351.8
Difference [%]	-	8.4
Max von Mises [MPa]	331.5	359.7
Difference [%]	-	8.5
Radial Displacement [mm]	0.47	0.47
Difference [%]	-	0
Coning [mm]	0.16	0.16
Difference [%]	-	0

By examining all the results from the thermal analyses of this Section, the following can be concluded for the Elise Version 3:

- In terms of thermal capacity, radial displacement and coning, the behaviour of the concept can be considered expectable,
- In terms of the thermal stresses, there were concerns that the concept disc might not be able to withstand the thermal loading due to the high values of von Mises stress.

As a result of the high von Mises values and due to the fact that the Elise Version 3 was actually a slight modification of the series production Elise S2 brake disc, it was considered useful to conduct thermal analyses on the original Elise disc and see how it behaves under thermal load. Then, compare it against the behaviour of the Elise Version 3 concept design.

A 10-degree segment from the CAD model taken from the exact same area that was used for the fingered disc simulations was also used for the Elise S2 simulations. The assumptions, inputs and boundary conditions used for modelling were the same as with the Elise Version 3 (see Section 4.2). Tetrahedra and hexahedra elements were used with the sets of properties being linear, non-linear elastic and non-linear elastoplastic for each type of elements. Finally, the size of elements was approximately 1 mm. Table 4-13, Table 4-14 and Table 4-15 present the results using tetrahedra elements for the different material properties.

Table 4-13 Elise S2 – Tetrahedra Elements Linear Elastic Properties

Temperature [°C]	336
von Mises [MPa]	551
Maximum von Mises [MPa]	552.5 @ 7 sec
Radial Displacement [mm]	0.44
Coning [mm]	0.3

Table 4-14 Elise S2 – Tetrahedra Elements Non-linear Elastic Properties

Elise S2 – Tetrahedra Elements and Non-linear Elastic Properties	
Temperature [°C]	308
von Mises [MPa]	567
Maximum von Mises [MPa]	569 @ 7 sec
Radial Displacement [mm]	0.46
Coning [mm]	0.31

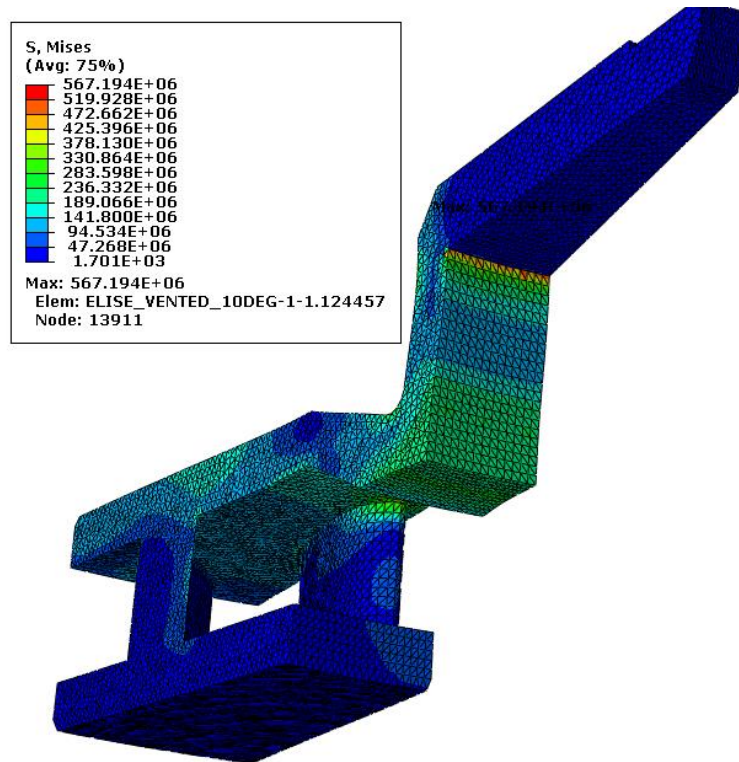


Figure 4-9 Elise S2 (Tetrahedra Elements Non-linear Properties) von Mises at the End of Braking

In both the linear and non-linear elastic analyses, the high von Mises stresses at the end of braking were almost equally distributed in the area where the swan neck was connected to the flange of the hub at the inboard side of the disc (Figure 4-9).

Table 4-15 Elise S2 – Tetrahedra Elements Non-linear Elastoplastic Properties

Elise S2 – Tetrahedra Elements and Non-linear Elastoplastic Properties	
Temperature [°C]	308
von Mises [MPa]	278.7
Maximum von Mises [MPa]	279 @ 6.8 sec
Radial Displacement [mm]	0.46
Coning [mm]	0.3

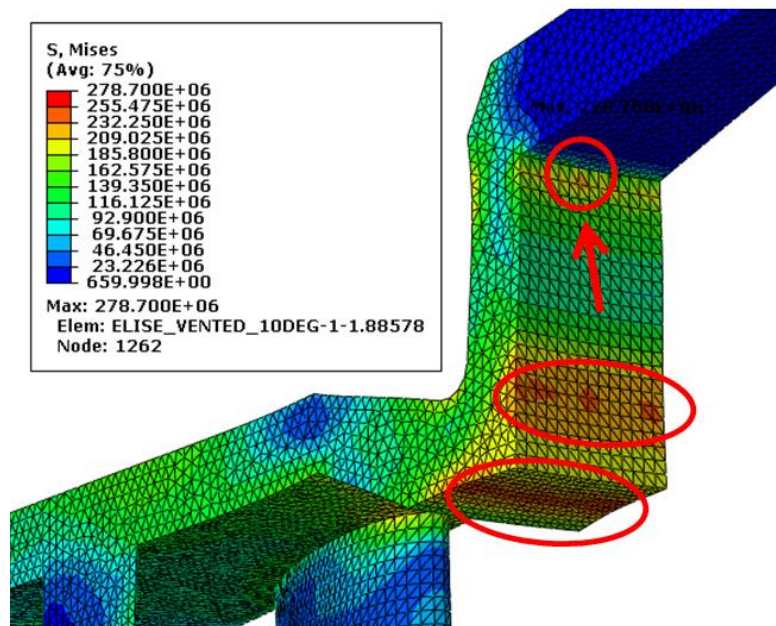


**Figure 4-10 Elise S2 (Tetrahedra Elements, Non-linear Elastoplastic Properties)
von Mises at the End of Braking**

With regards to the analysis with non-linear elastoplastic properties, the maximum von Mises stress values at the end of braking were predicted to be at the inboard side of the disc in the connection of the swan neck and the hub's flange. Nevertheless, high stress concentrations occurred on the swan neck as well (Figure 4-10). The peak value of von Mises stress was reduced by 50% with the introduction of plasticity.

By comparing the predictions of the simulations with the use of tetrahedra elements, the following can be concluded:

- The values of maximum temperature in the case of Elise S2 are somehow higher compared to the Elise Version 3. This prediction contrasts with what was expected as the Elise S2 with the solid hub has more material to absorb the heat. This is an area that further investigation is required,
- The values of von Mises stress both the maximum during braking and the highest at the end of braking are lower for the case of the Elise S2. Although the difference is considerable in the case of the linear and non-linear elastic analyses (about 36%), in the case of non-linear elastoplastic, which can give predictions closer to reality, the difference was 14%,
- The values of radial displacement for both Elise S2 and Elise Version 3 in all simulations were almost identical,
- With regards to coning, the Elise S2 features 87.5% increased coning compared to the Elise Version 3. This is due to the fact that the solid hub

tends to affect more the axial displacement of the outboard friction surface compared to the fingers.

Thereafter, the results of the simulations using hexahedra elements are presented. The outcomes from the analysis using linear elastic properties are presented in Table 4-16, the results from the analysis using non-linear elastic properties are illustrated in Table 4-17, and finally the predicted temperature, von Mises stress, radial displacement and coning for non-linear elastoplastic analyses are shown in Table 4-18.

Table 4-16 Elise S2 – Hexahedra Elements Linear Elastic Properties

Temperature [°C]	336
von Mises [MPa]	656
Maximum von Mises [MPa]	657 @ 6.9 sec
Radial Displacement [mm]	0.44
Coning [mm]	0.29

Table 4-17 Elise S2 – Hexahedra Non-linear Elastic Properties

Temperature [°C]	308
von Mises [MPa]	675.2
Maximum von Mises [MPa]	677 @ 6.9 sec
Radial Displacement [mm]	0.46
Coning [mm]	0.31

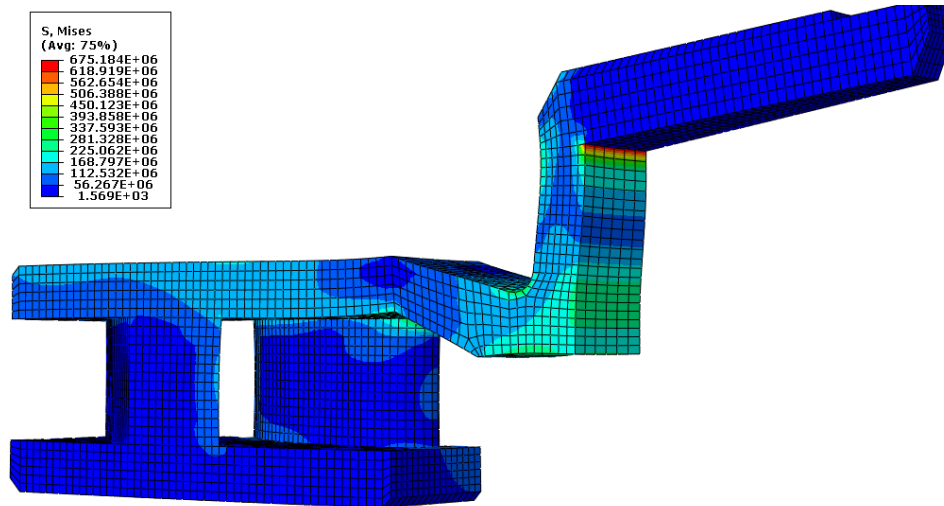


Figure 4-11 Elise S2 (Hexahedra Elements, Non-linear Properties) von Mises at the End of Braking

A drop in temperature was noticed between the analysis with non-linear elastic properties and the analysis with linear elastic properties. This was expected since, as mentioned previously, the specific heat capacity increases with the increase in temperature.

With regards to the von Mises stresses, the values are increased on the non-linear elastoplastic analysis compared to the linear elastic one. The peak values at the end of braking acted on the area of connection between the swan neck and the flange of the hub (Figure 4-11). In the same manner to what the analyses with tetrahedra elements showed, the peak values are equally distributed around the connection of the flange and the swan neck. This distribution is different from the stress distribution of the Elise Version 3 (Figure 4-7), where the pick values of von Mises stress were acting on one spot.

Table 4-18 Elise S2 – Hexahedra Elements, Non-linear Elastoplastic Properties

Temperature [°C]	308
von Mises [MPa]	321.4
Maximum von Mises [MPa]	323 @ 6.9 sec
Radial Displacement [mm]	0.46
Coning [mm]	0.3

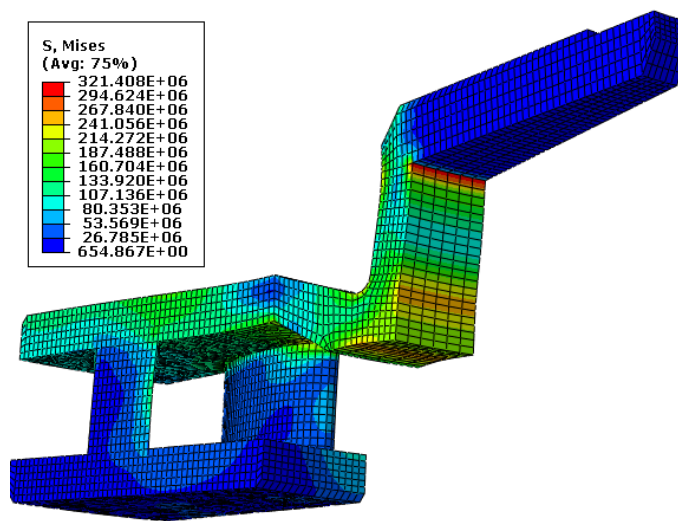


**Figure 4-12 Elise S2 (Hexahedra Elements, Non-linear Elastoplastic Properties)
von Mises at the End of Braking**

Evaluating the results for the non-linear elastoplastic analysis, it can be concluded that there is no actual difference for the values of temperature prediction, radial displacement and coning compared to the non-linear elastic analyses. There is, however, a reduction, which was expected, in both von Mises stress at the end of braking and the maximum von Mises during the braking event. This reduction is at about 52% for each case.

By analysing the results for the analyses using hexahedra elements for the Elise S2 and the Elise Version 3, the following conclusions can be made:

- There is a marginal difference in terms of predicted temperature at the end of braking in favour of the Elise Version 3 for the analyses where temperature dependent properties were used. It was expected that the two

designs have a similar maximum temperature prediction as the segment used had the same size and was taken from the same area of the disc, but the lower value was foreseen to be for the Elise S2 as it has more material and can absorb more heat. Instead, it was for the Elise Version 3. This trend was also observed for the use of tetrahedra elements,

- With regards to von Mises stress, the Elise S2 disc was predicted to have lower values for all the simulations compared to the Elise Version 3 design. For the analyses without the introduction of plasticity, the Elise S2 featured lower stresses ranging from 31% to 39%. Nevertheless, for the analyses with plasticity, where the actual limit of the designs can be more accurately predicted, this reduction was only 8.75% for the value of von Mises at the end of braking and 10.2% for the value of maximum von Mises during braking. This relatively small difference gave confidence that the Elise Version 3 would be able to withstand the thermal load of the Lotus Elise. Furthermore, in private communications with a major automotive manufacturer the author was made aware that a brake disc can be considered to have acceptable thermal performance if under the FE analysis, using non-linear elastoplastic properties, the design features von Mises stresses lower than 450 MPa. The Elise Version 3 showcased with a safety margin of more than 90 MPa,
- The radial displacement is similar for the Elise S2 and the Elise Version 3. It was noticed to be similar for all the series of simulations that took place in this investigation. This was the case for the use of both element types and for all three different sets of simulation,
- Finally, in terms of coning the Elise S2 had an increase of up to 87.5% compared to the Elise Version 3. This was the exact same percentage found with the use of tetrahedra elements. This performance from the Elise S2 is believed to be due to the fact that the solid hub tends to affect more the axial displacement of the outboard friction surface compared to the fingers.

As a general conclusion from the analyses that were conducted, it seems that the concept of a brake disc with fingered hub design is feasible for a lightweight car without an excessive top speed. Physical testing needs to be performed for the findings from the numerical analyses to be validated.

4.4 Chapter Summary

In this Chapter, the thermomechanical modelling process for the fingered hub brake disc feasibility study was presented. For the analyses, linear elastic, non-linear elastic and non-linear elastoplastic material properties were used. Additionally, tetrahedral and hexahedra elements were used.

The analyses were focused on predicting the maximum disc temperatures, the maximum von Mises and Maximum Principal stresses, the maximum radial disc displacements and the maximum coning during braking.

When comparing the results of the Elise Version 3 against the results of the standard Elise S2 disc, it was observed that the aspects of maximum disc temperatures and the maximum disc radial displacements were very close. Furthermore, it was seen that when non-linear elastoplastic properties are used, which can more accurately predict the limit of performance, the maximum predicted values of the von Mises stress for the Elise Version 3 are approximately 10% higher than the Elise S2, something that can be considered acceptable. Finally, the predicted coning for the two discs showed a considerable advantage of the prototype Elise Version 3 disc when compared to the standard Elise S2 disc as it featured almost 90% reduced coning. This is a great improvement in terms of any potential NVH issues, such as squeal and in particular judder.

5 LIGHTWEIGHT FINGERED BRAKE DISC INVESTIGATION – EXPERIMENTAL WORK

In this Chapter, the experimental work conducted on the fingered hub brake disc will be presented. Three different physical tests took place. The first test was a 3-point bending test of cast iron specimens, conducted with the intention to identify the value of strain that would fracture the specimens. In order to validate the mechanical FE modelling presented in Chapter 3, in the second test, an instrumented Elise Version 3 prototype rotor was subjected to torsional loading simulating the brake torque applied during braking. Finally, the third test was similar to the second, but the Elise Version 3 prototype rotor was not instrumented and the aim of the test was to find the actual torsional limit of the design by causing it to fail. All tests will be discussed in more depth in the following Sections.

5.1 3-Point Flexural Test

In the experimental part of the investigation for the feasibility of a brake with fingered hub, a 3-point bending test was conducted. Although values for cast iron properties are available in the literature, they vary considerably. Consequently, the aim of the test, which is described below, was to create an understanding of material properties with regards to fracture stress, strain and Young's Modulus for cast iron when subjected to bending and compare them to the material data used in the modelling part (Chapters 3 and 4).

Specimens were manufactured out of a VX220 brake disc made of cast iron and casted by the same company that manufactured the Elise discs used for making the Elise Version 3 prototype. Two types of specimens were manufactured. The first was 100 mm X 12 mm X 6 mm (length/width/height) and the second was 60 mm X 12 mm X 6 mm. They were taken from different areas of the disc and from both the inboard side and the outboard side (Figure 5-1).

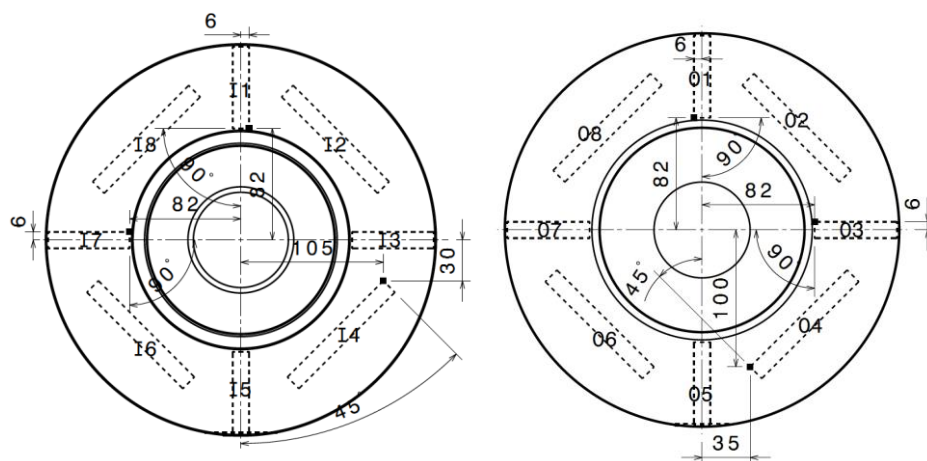


Figure 5-1 Specimens' Position (Left inboard side, Right outboard side)

The specimens were taken from different areas of the disc, because these discs are casted vertically and the density of cast iron is usually slightly higher in the lower part of the discs where the balancing mark is. Obviously, mechanical and thermal properties can vary as well. To account for these potential variations, the samples were taken from different areas and with different orientation (e.g. circumferential and radial). The specimens were taken from the disc with reference to the balancing mark. The exact dimensions of each of the specimens of the test can be found in Table 5-1.

Table 5-1 Specimens Dimensions

Specimen	Length [mm]	Width [mm]	Height [mm]
Inboard 1 (I1)	60.47	12.00	5.99
Inboard 3 (I3)	60.70	12.01	6.02
Inboard 5 (I5)	59.80	12.02	6.05
Inboard 7 (I7)	60.54	12.00	5.97
Outboard 1 (O1)	60.50	12.00	5.99
Outboard 3 (O3)	60.56	12.01	5.94
Outboard 5 (O5)	59.77	12.01	5.94

Outboard 7 (O7)	60.63	12.02	5.97
Inboard 2 (I2)	99.47	11.95	6.04
Inboard 4 (I4)	99.50	11.95	6.01
Inboard 6 (I6)	99.52	11.94	6.03
Inboard 8 (I8)	99.50	11.95	6.04
Outboard 2 (O2)	99.51	11.96	5.99
Outboard 4 (O4)	99.52	11.97	5.98
Outboard 6 (O6)	99.56	11.97	5.98
Outboard 8 (O8)	99.57	11.96	6.05

The tests were conducted in the laboratory of the Applied Mechanics Department at Cranfield University using an Instron 5965 machine able to produce compressive force of 5 kN. The specimens were supported by two rollers on their bottom side with the span for Specimens I1, I3, I5, I7, O1, O3, O5 and O7 being 50 mm. The span for the rollers supporting the Specimens I2, I4, I6, I8, O2, O4, O6 and O8 was 80 mm. A plunger with a roller applied force on the top side of the specimens in a position which was half the distance of the edges of each specimen forcing it to bend.

The Instron machine was able to record the applied force on each specimen during the test and the deflection of the plunger. By knowing these two aspects and by using the equations below from the Engineers Bending Theory, the fracture stress, strain and the elastic modulus in bending were computed.

$$\sigma_f = \frac{M_y}{I} \quad \textbf{(5-1) Flexural Stress}$$

Where:

M_y : Moment around y axis,

I : Second moment of area.

$$\varepsilon_f = \frac{6\delta h}{L_s^2}$$

(5-2) Flexural Strain

Where:

δ : Maximum deflection,

h : Thickness of the beam,

L_s : Span of the support.

$$E_f = \frac{F L_s^3}{48\delta I}$$

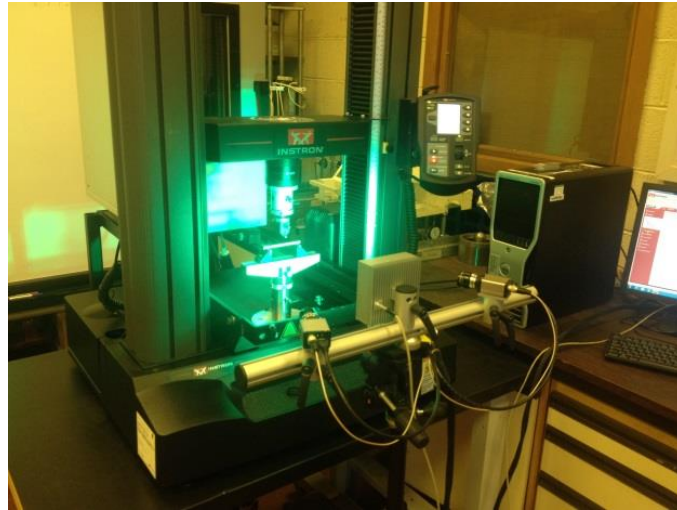
(5-3) Elastic Modulus in Bending

Where:

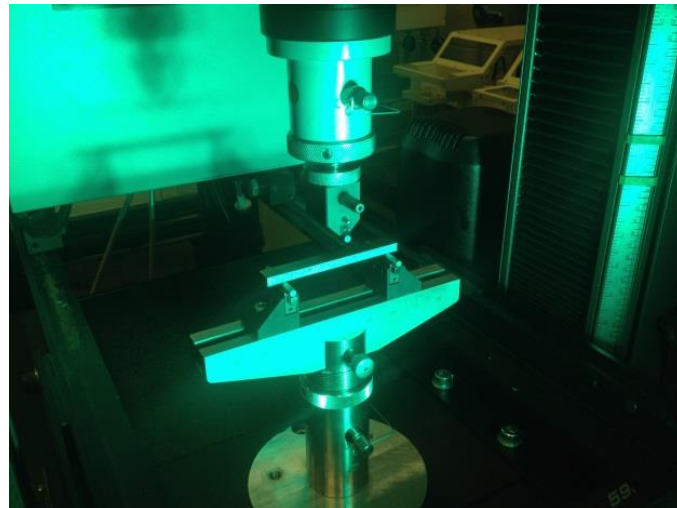
F : Applied load.

Furthermore, and in order to validate the calculations of flexural stress, strain and elastic modulus in bending, the technique of Digital Image Correlation (DIC) was utilised. The DIC is able to measure deformations and strains by recording pictures with two cameras before the tests when the specimens were intact, and during the tests, when the specimens were forced to bend. Then, the software that operates the two cameras can spot the changes in individual pixel positions before and after each test and give the specimens' deformation and strain. In order for this technique to work, one side of the specimens (the one facing the cameras) had to be initially painted white and then painted with a spray that only leaves black marks. These black marks are crucial as they are the ones that the software uses to correlate the pictures before, during and after the test in order to give the values of deformation and strain.

The setup of the test is shown in Figure 5-2 where the two cameras can be seen on the left part of the figure and the strong green light that illuminates the painted side of the specimens in order to make the pictures recorded by the cameras clearer and easy to process. The two supports, the plungers and their rollers, can be seen in the right part.



a)



b)

Figure 5-2 a) 3-Point Flexural Test Setup, b) Detail of Specimen Position

Three sizes of rollers were used during the tests aiming to investigate whether the size can have an influence on the results or not. The sizes used were 3 mm, 5 mm and 8 mm and the size used in each specimen is presented in Table 5-2 along with the speed that the plunger applied the force to each specimen and the span of their supports.

Table 5-2 Flexural Test Parameters

Specimen Name	Roller Diameter [mm]	Speed [mm/min]	Support Span [mm]
Inboard 1 (I1)	5	4.5	50
Inboard 3 (I3)	5	4.5	50
Inboard 5 (I5)	5	4.5	50
Inboard 7 (I7)	5	4.5	50
Outboard 1 (O1)	5	4.5	50
Outboard 3 (O3)	5	4.5	50
Outboard 5 (O5)	5	4.5	50
Outboard 7 (O7)	-	-	-
Inboard 2 (I2)	5	4.5	60
Inboard 4 (I4)	5	4.5	60
Inboard 6 (I6)	8	4.5	60
Inboard 8 (I8)	8	4.5	60
Outboard 2 (O2)	3	4.5	50
Outboard 4 (O4)	5	4.5	60
Outboard 6 (O6)	3	4.5	60
Outboard 8 (O8)	-	-	-

It should be mentioned that Specimen O2, although initially had dimensions of 100 mm X 12 mm X 6 mm, was cut in order to be 60 mm in length and serve as a comparison tool between itself and Specimen I2. The comparison would be made to check if the change in length affected the properties. Furthermore, it was decided for Specimens O7 and O8 not to be tested and rather be kept for future use.

5.2 3-Point Bending FE Analysis

In addition to the physical 3-point flexural test presented in the section above, FE simulations were created to replicate the actual tests. The aim was to compare the test results against the results from the numerical models and understand the behaviour of cast iron under mechanical loading.

The Abaqus CAE package was used for the analyses and both specimen types were modelled by using the Specimen I1 to represent the 100 mm X 12 mm X 6 mm specimens and the Specimen I2 to represent the 60 mm X 12 mm X 6 mm specimens. A number of methods to replicate the physical tests were used; however, only the method that converged successfully giving repeatable results will be presented. Figure 5-3 presents the FE model for the 100 mm X 12 mm X 6 mm specimens.

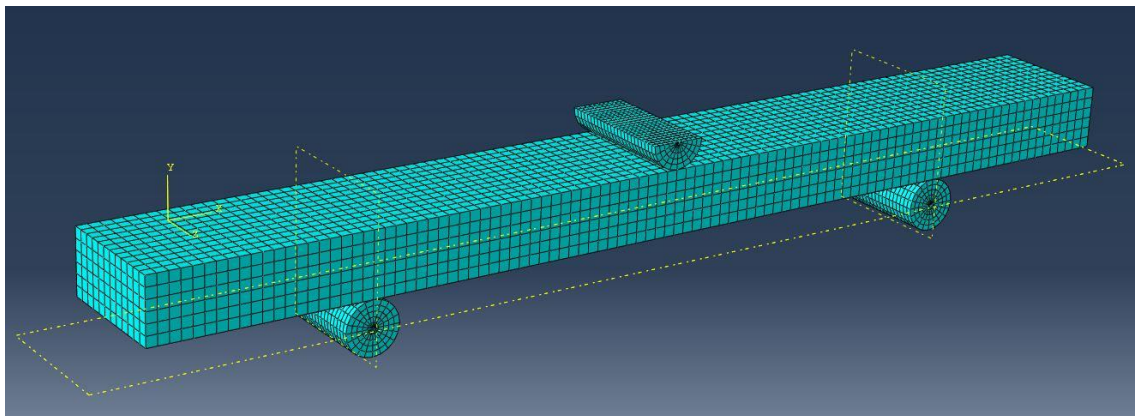


Figure 5-3 3-Point Flexural Bending FE Model

Linear 3D stress hexahedra C3D8R with reduced integration were applied. Their size was 1 mm which resulted in their total number being 4320 with 5551 nodes for the I1 Specimen, and 7200 with 9191 nodes for the Specimen I2. The same non-linear cast iron properties that were used for the advance mechanical modelling of section 3.6 were used for the specimens in these analyses and can be found in Appendices E.6, E.7 and E.8.

Two boundary conditions were applied on the specimens to ensure realistic symmetrical loading and also to avoid any rigid body motions. The first was to

constrain the lateral translation on the centreline around the specimen (Figure 5-4).

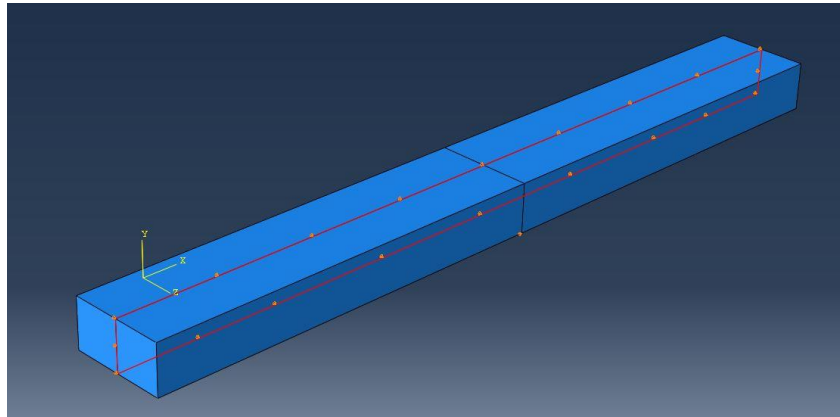


Figure 5-4 Specimen Boundary Condition - Lateral

The second was to constrain the longitudinal translation of the middle nodes on the bottom side of the specimen (Figure 5-5).

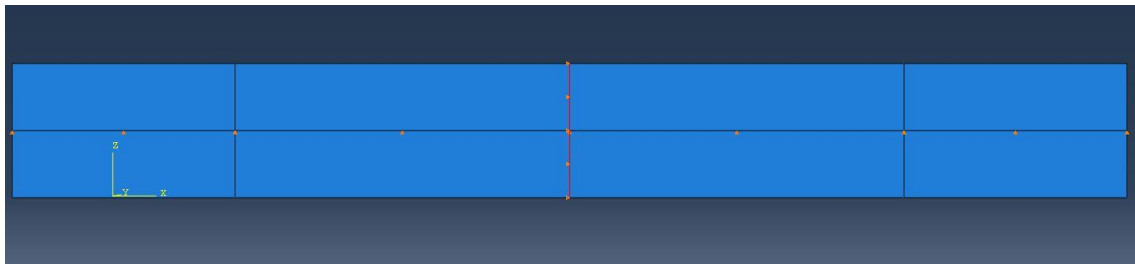


Figure 5-5 Specimen Boundary Condition - Longitudinal

There are two rollers of 5 mm outer diameter and 12 mm of length under the specimens that support them. Linear 3D stress hexahedra C3D8R with reduced integration and wedge C3D6 elements were used for the rollers. In order to achieve uniform mesh, the rollers were partitioned and the elements were applied using the sweep technique with the medial axis algorithm and by minimising the mesh transition. The size of the elements was 0.5 mm, which resulted in 1920 C3D8R and 480 C3D6 elements. The vertical side surfaces of the rollers were constrained using the Encasté boundary condition (Figure 5-6).

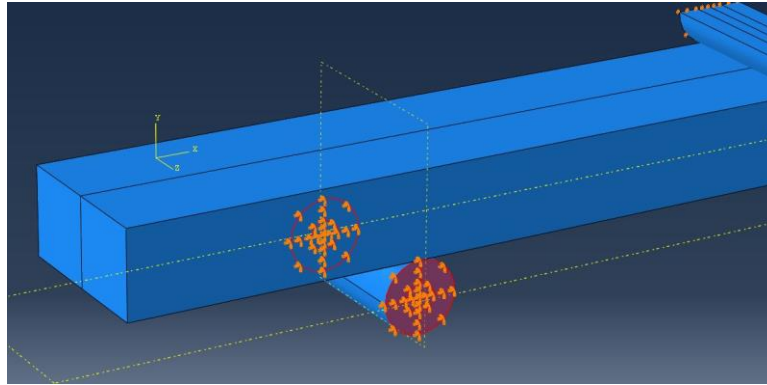


Figure 5-6 Rollers Boundary Condition

The plunger applying load to the specimen was modelled as a half roller using 960 C3D8R and 240 C3D6 elements. Boundary conditions were applied on the two side surfaces of the plunger constraining the longitudinal and lateral translations, but allowing the vertical translation (Figure 5-7).

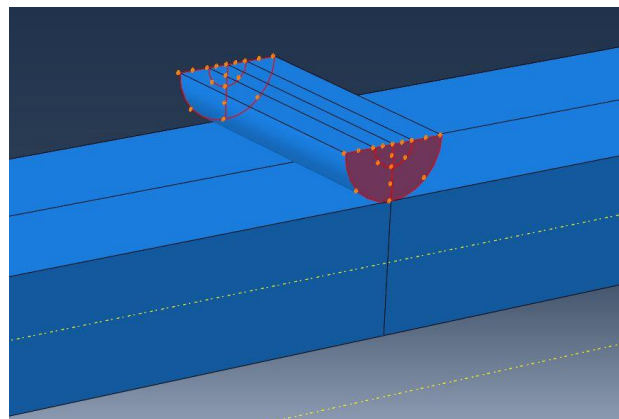


Figure 5-7 Plunger Boundary Conditions

The load input for the plunger was the force values against time that were recorded during the tests of Specimens I1 and I2. Figure 5-8 and Figure 5-9 show the load profile for Specimens I1 and I2 respectively.

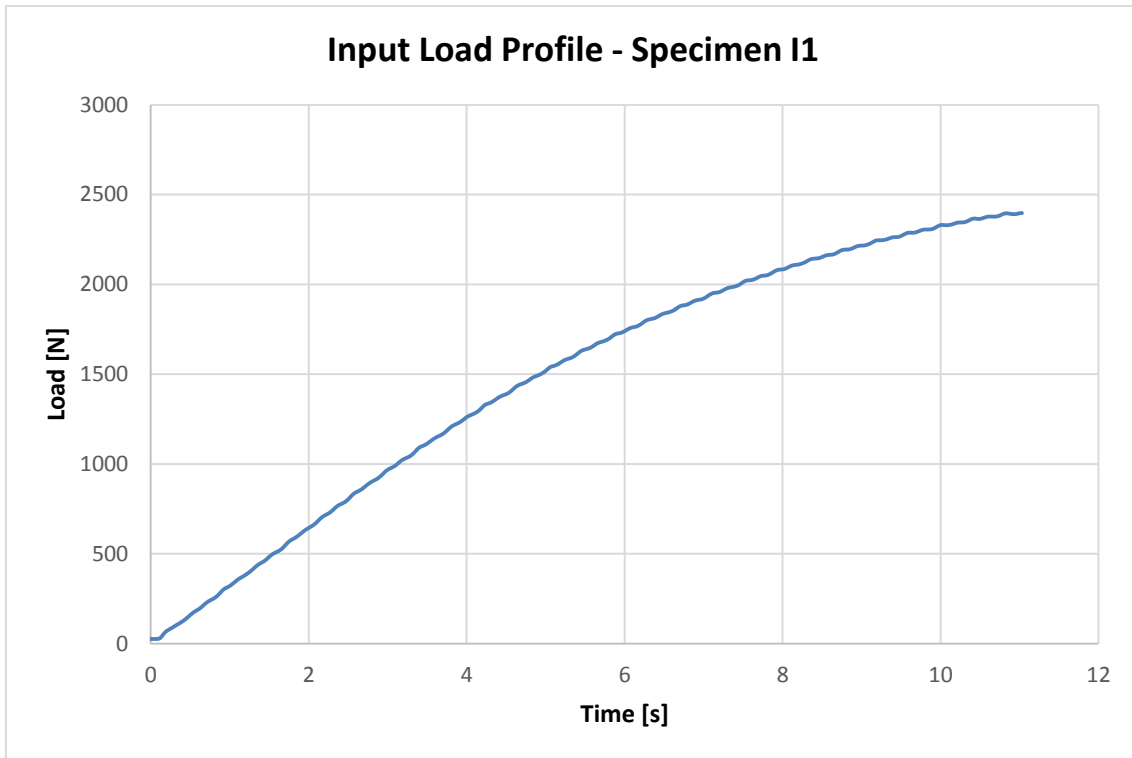


Figure 5-8 Load Input – Specimen I1

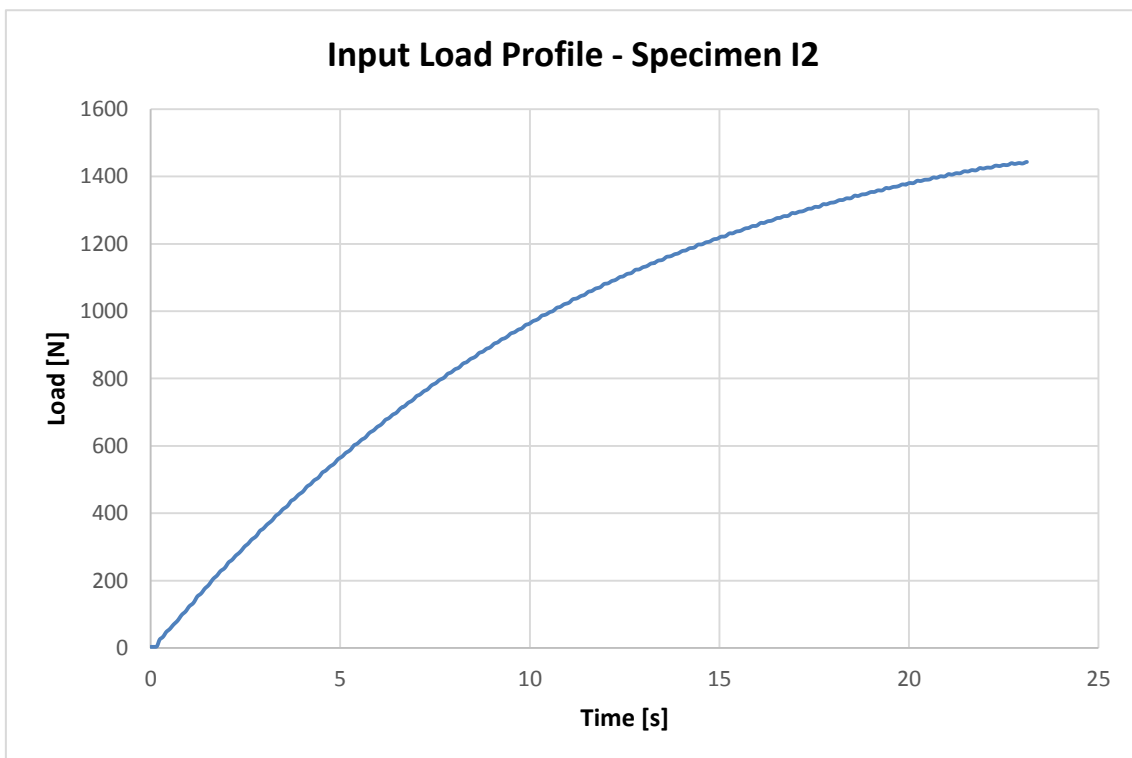


Figure 5-9 Load Input – Specimen I2

Finally, the material applied on both the two rollers and the plunger was steel and its properties can be found in Appendix B.2. In addition, contact was defined between the specimens and the rollers and plunger. The properties were the same as in the Second Modelling Case of the mechanical loading (section 3.4.2) with the only difference being the value of friction coefficient between steel and cast iron, which was set to 0.23 (EngineeringToolbox, 2016).

5.2.1 3-Point Bending Test and FEA Results

The flexural stress, strain and elastic modulus in bending were calculated using the data from the Instron machine and the formulae (5-1), (5-2) and (5-3). Figure 5-10 shows the stress/strain curves for the specimens on the inboard side of the disc, while the ones for the outboard side are presented in Figure 5-11. Finally, the elastic modulus in bending for each of the specimens of the test can be found in Table 5-3. It should be noted that the specimens of 60 mm X 12 mm X 6 mm dimensions are in the left column and the specimens of 100 mm X 12 mm X 6 mm dimensions are in the right column.

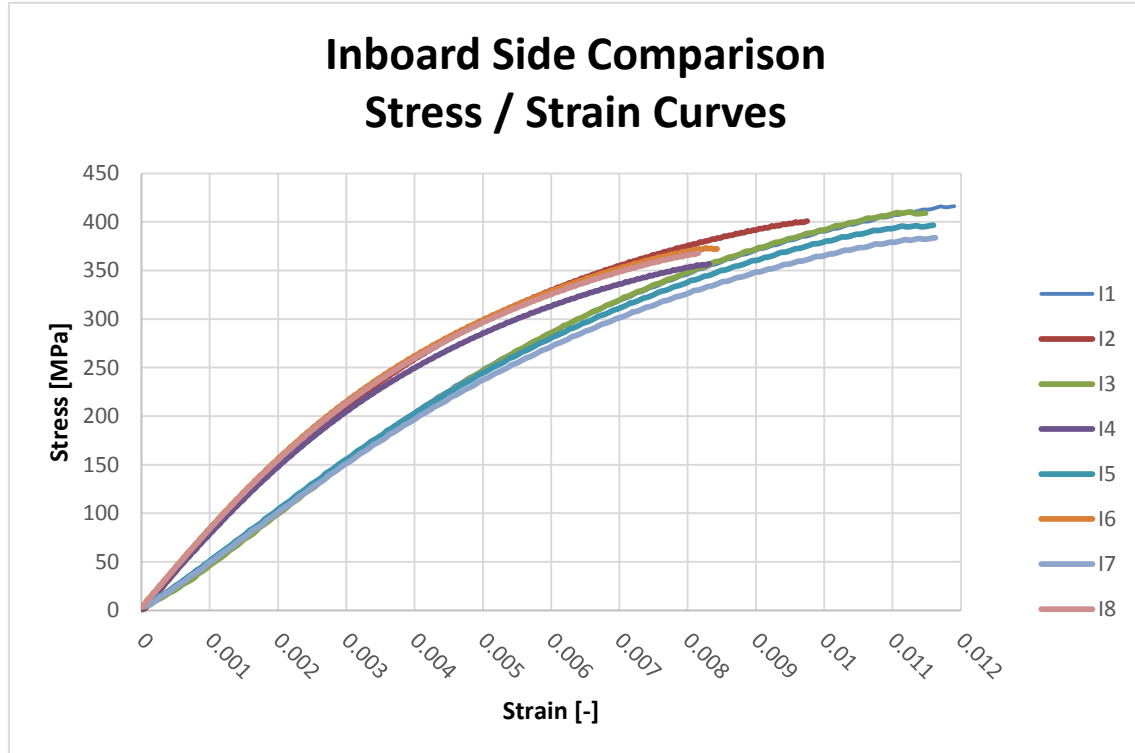


Figure 5-10 Stress/Strain Curves of the Specimens on the Inboard Side

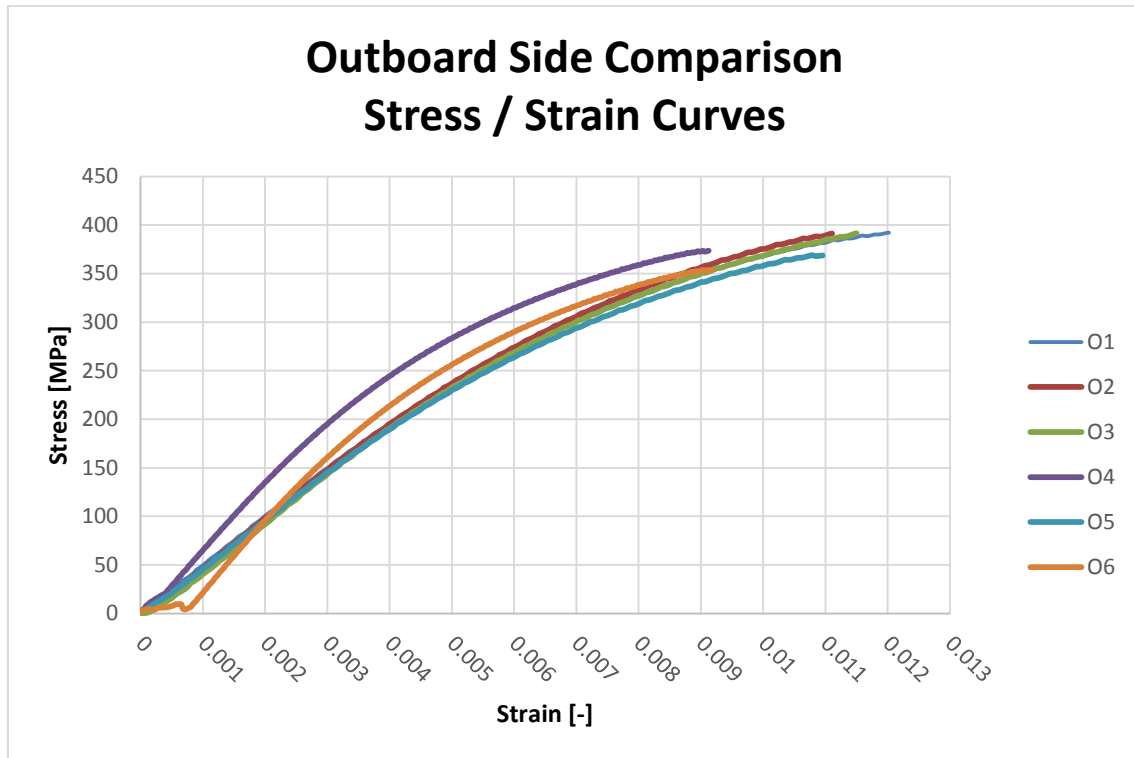


Figure 5-11 Stress/Strain Curves of the Specimens on the Outboard Side

Table 5-3 Specimens' Elastic Modulus in Bending

Specimen Name	Elastic Modulus [GPa]	Specimen Name	Elastic Modulus [GPa]
Inboard 1	35.5	Inboard 2	40.7
Inboard 3	36.9	Inboard 4	42.9
Inboard 5	34.9	Inboard 6	44.1
Inboard 7	33.0	Inboard 8	43.3
Outboard 1	32.1	Outboard 2	35.2
Outboard 3	32.5	Outboard 4	41.3
Outboard 5	34.2	Outboard 6	39.1

By evaluating the results of the stress/strain curves, it can be seen that all the specimens had flexural stress of more than 350 MPa before failure. In addition, the hypothesis that the specimens at the bottom of the disc would have higher strength compared to those on the top of the disc due to higher density was not valid. Specimen I1 has higher strength than I5, and similarly specimen O1 has higher flexural stress than O5. Of course, there is a possibility that the milling machine of the workshop that machined the disc from which the specimens were taken had influenced the specimens' surfaces, but this could not be verified or controlled. The last observation on these initial results is with regards to the elastic modulus in bending. The values are much lower than the typical value used for linear elastic analyses (110 GPa) and also it seems that they have been influenced by the length of the specimens. The specimens with length of 100 mm gave higher values compared to the specimens of 60 mm in all cases.

With regards to the FE analyses, their predicted results were compared against the results obtained from the Instron data and the DIC data. The aim was to understand how the material behaves and assess if the material data of Appendix E that was used in the FE analyses was the most appropriate to be used.

In Figure 5-12 the stress/strain results for Specimen I1 are illustrated. It can be seen that the curve derived from the Instron machine data features almost the same fracture stress as the data used in the FE simulations which is approximately 400 Mpa at 20°C. The slope is certainly different; this is because the data from the Instron machine is only for bending. The curve predicted in the FE simulation follows partly the Tension Hardening data, but its slope differs slightly. This is happening because in the FE modelling, the cast iron plasticity was included and both tension and compression hardening were implemented. This enables the software to predict the stresses, strains and displacements of complex loading cases.

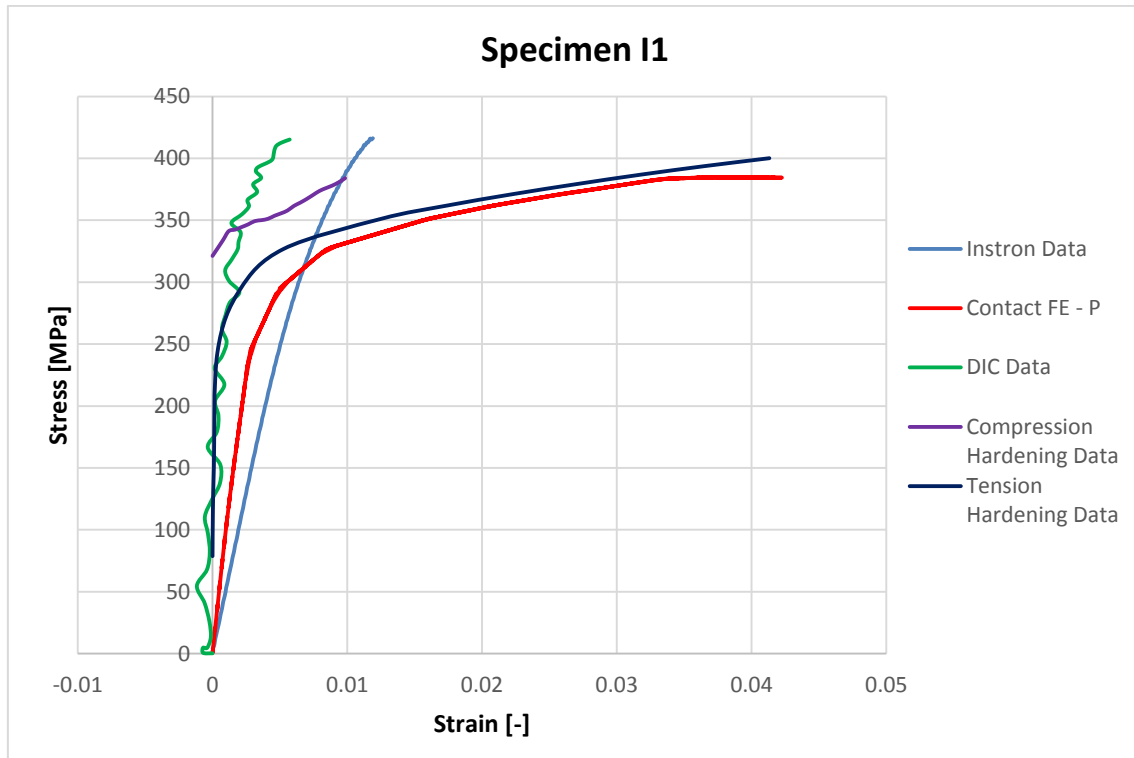


Figure 5-12 Stress/Strain Results for Specimen I1

The findings for Specimen I2 are similar. The results are depicted in Figure 5-13. The DIC data seems to follow the data derived from the Instron machine; however, for Specimen I1 it shows a significant offset, whereas in Specimen 2 it indicates that the material has yielded at a much lower point than it actually did. In both cases, it incorporated noteworthy noise.

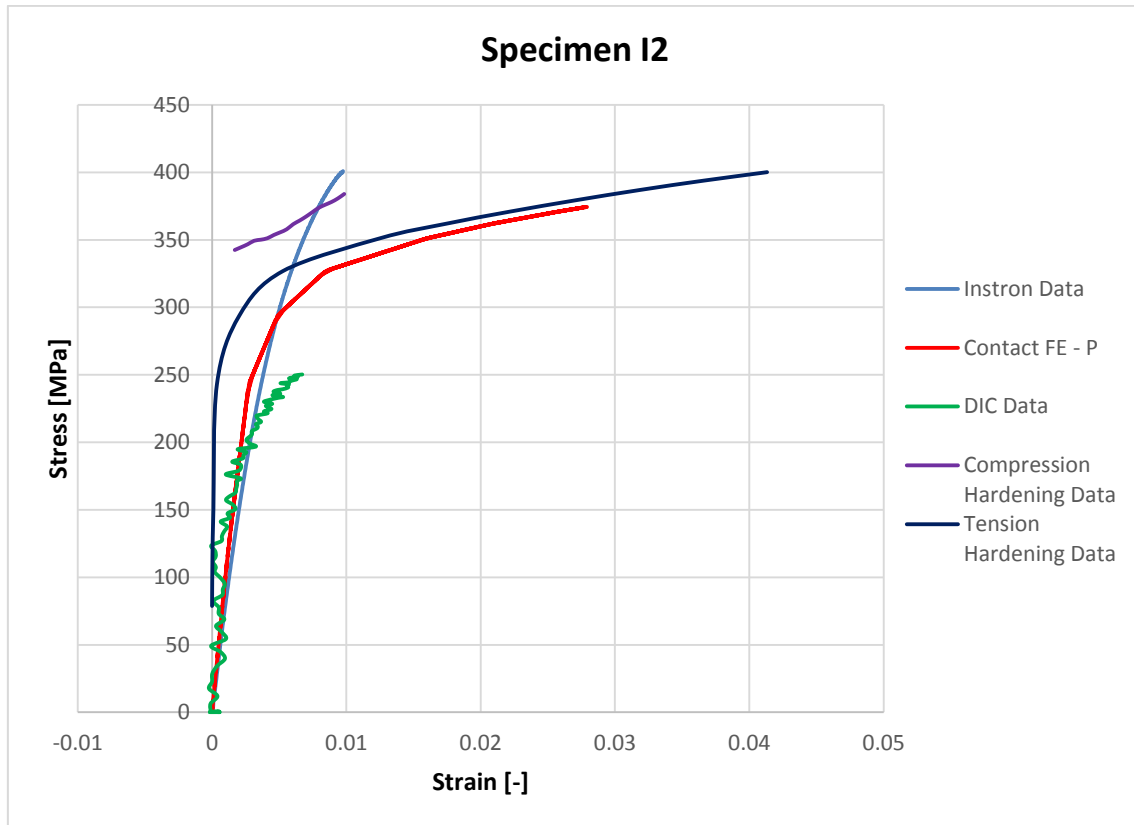


Figure 5-13 Stress/Strain Results for Specimen I2

The conclusions after analysing the data from the Instron machine, the DIC and the FE analyses are the following:

- In all cases the maximum bending strength of cast iron was measured to be more than 350 MPa,
- Although the specimens in the lower areas of the disc were expected to be stronger than those in the higher, results showed the opposite,
- The specimens with length of 100 mm gave constantly elastic modulus in bending higher than those with length of 60 mm,
- The limit of performance for cast iron that was derived from the Instron machine agrees with the material data used in the FE simulations,
- Accurate modelling can be achieved with the data used in the FE simulations resulting in detailed stress/strain curves derivation with high level of confidence.

5.3 Step 6 – Torsion Test with Instrumented Disc

Since the beginning of the investigation for the feasibility of the brake disc with a fingered hub, it was envisaged that physical testing would take place mainly to be used as validation for the FE modelling and to also see the actual limits of the concept. Bearing these two reasons in mind, torsional tests were designed to

replicate the mechanical modelling of the first modelling case in Chapter 3. An experimental rig was created in the brakes' laboratory of the Automotive Department at Cranfield University (Figure 5-20). The aim was to subject the prototype designs to mechanical torsional loading.

5.3.1 Instrumentation and Data Logging Equipment

High Maximum Principal stresses were predicted on the curved areas where the fingers of the hub are connected to the flange as detailed in Chapter 3 and shown in Figure 5-14. As a result, it was decided for chain strain gauges to be installed.

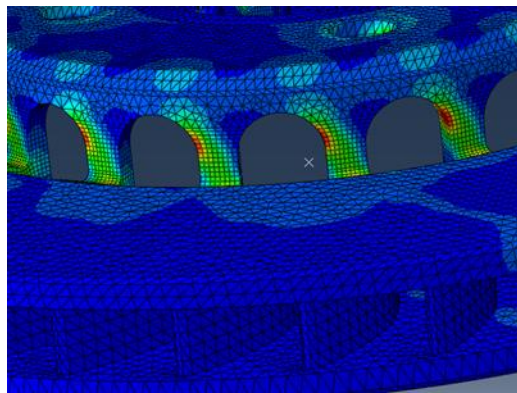


Figure 5-14 Areas of High Maximum Principal Stress

A finger of the prototype disc was selected and two chain gauges were attached to the area where the highest Maximum Principal stresses occur according to the FE simulations. A gauge on each side of the finger was installed as seen in Figure 5-15. They were positioned in such a way that they could capture both the tensile and also compressive strains.

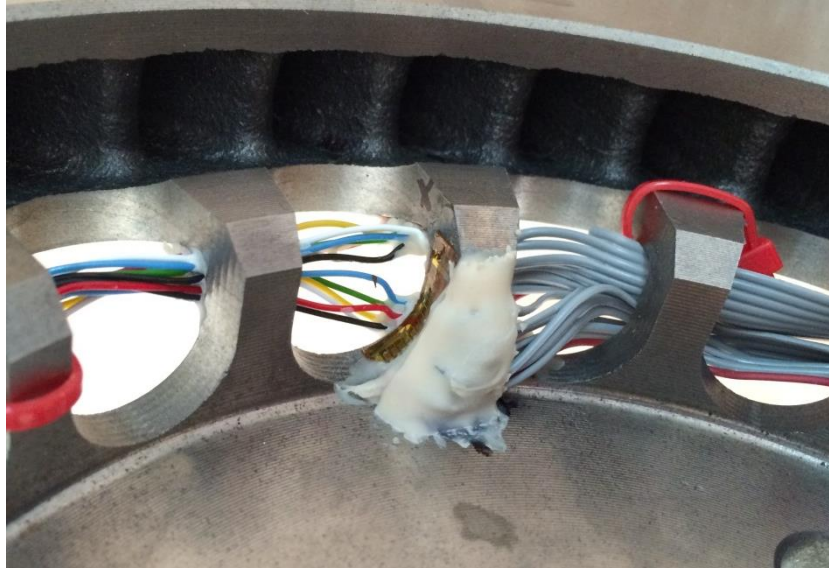


Figure 5-15 Chain Strain Gauge on One Side of the Finger

The gauges were provided by the company HMB. Their type was 1-KY53-1/120 and each comprised ten measuring grids of 120 Ohm resistance. The main selection criterion for this type of strain gauge was its ability to record ten strain measurements simultaneously. Thus, it would be able to capture the strain gradients along the finger, which it will be mounted on.

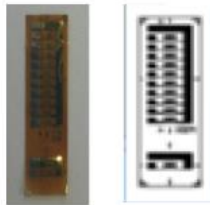
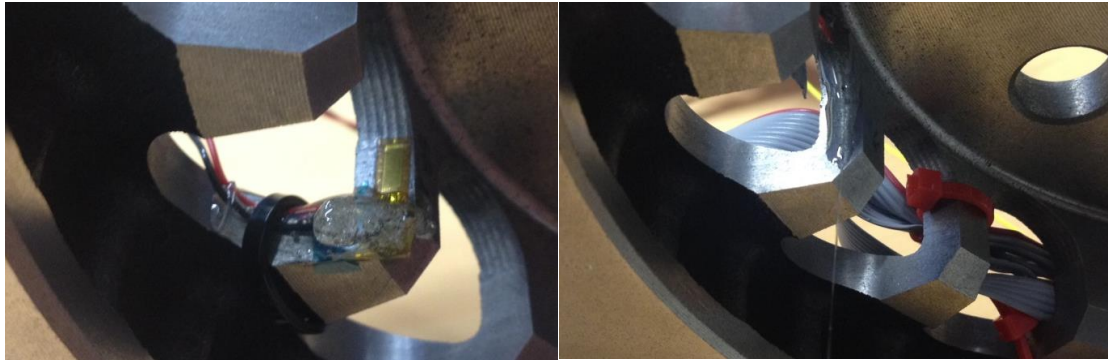


Figure 5-16 1-KY53-1/120 Type Strain Gauge

Moreover, in an effort to investigate alternative ways of measuring strains that would possibly improve the accuracy of the measurements from the torsion tests, a new Elise Version 3 prototype was manufactured and an optical fibre with a 5mm sensor was attached on one of its fingers. At a finger 180° from the optic fibre, an RS Pro strain gauge with a 5 mm X 1.8 mm grid was fitted aiming to conduct direct comparisons between the measurements of the two sensors.



a)

b)

Figure 5-17 Sensors for Strain Measurements, a) Single Strain Sensor, b) Optical Fibre

Figure 5-17 shows the sensor installation for the single strain gauge (Figure 5-17 – a) and the optical strain sensor (Figure 5-17 – b). Both were installed on the side of the fingers where tensile stresses were expected, since tension is most likely to cause failure in grey cast iron. Although the peak stress values were expected to occur on the curved surface of the finger, close to the flange of the top hat, the strain gauge was positioned on the relatively flat part of the finger with relatively constant strain in order to improve accuracy. The optical fibre strain sensor was installed in an identical place. The intention was therefore to have a direct comparison of the strain measured using the strain gauge and the optical strain sensor (in the same place) at fingers 180° apart (at opposite disc positions). This would enable a comparison of the results using the two sensors, and also the circumferential variation in the finger loading and repeatability/consistency of the measured strains as the disc is rotated and re-loaded in different positions.

All the strain gauges were connected to National Instruments SCC-SG01 120Ω connector modules installed in a National Instruments NI-SC2345 series connector block (Figure 5-18). The signals were conditioned there, and then they were fed to custom-made LabView software, running on a local desktop PC, that presented and logged the strain values in real-time.

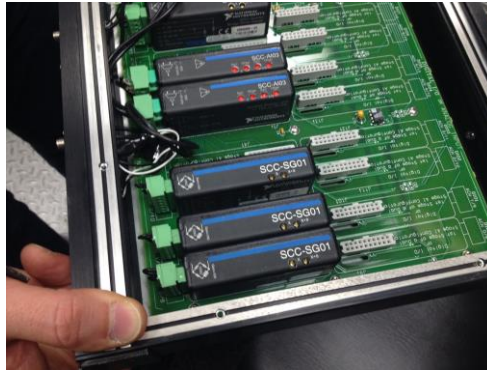


Figure 5-18 The SCC-SG01 Modules in the SC2345 Connector Box

In the case of the optical fibre, it is connected to a SmartScan Aero device with S/N 137514, which is an interrogator system manufactured by SMART FIBRES Ltd. UK (Figure 5-19). This type of interrogator is compact and robust, intended for dynamic measurements of Fibre Bragg grating (FBG) sensors. It incorporates a tuneable laser source that enables high-resolution interrogation at various frequencies.

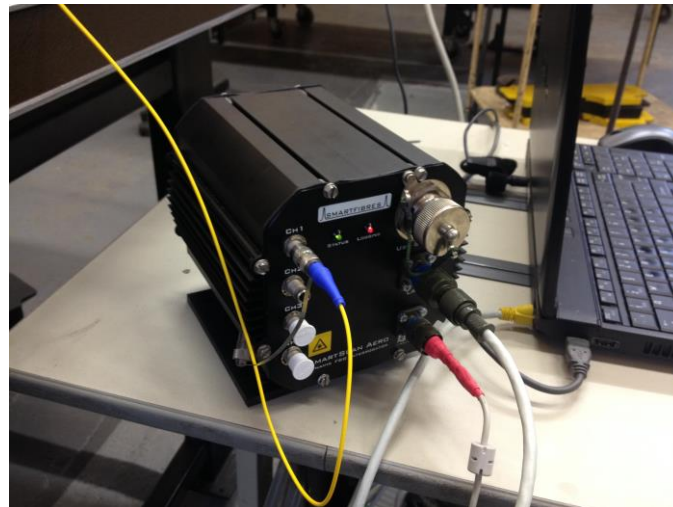


Figure 5-19 SmartScan Aero Interrogator

5.3.2 Experimental Process

An experimental rig was created in the brakes' laboratory of the Automotive Centre at Cranfield University (Figure 5-20). The aim was to replicate the first modelling case described in Section 3.4.2 and subject the prototype design of Elise Version 3 to mechanical torsional loading.

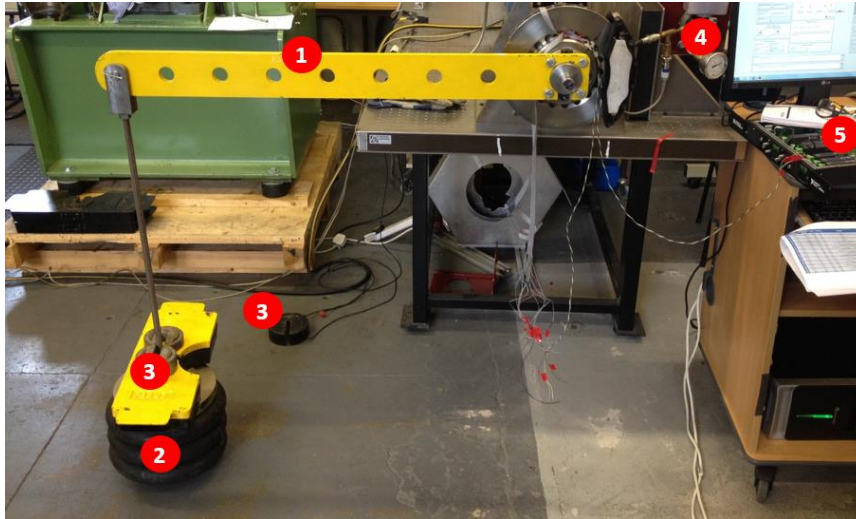


Figure 5-20 Torsion Rig Set up

The principle of operation was as follows: An air over hydraulic actuation system was used to apply hydraulic pressure to the brake caliper and as a result the brake disc was clamped. One side of a steel bar (1) was bolted on the hub of the Elise Version 3 prototype. On the other side of the bar, a weight carrier (2) was attached. Weights (3), were laid on the weight carrier and these would apply vertical force. This force would be transferred as a torque via the bar (which was kept in horizontal position and the distance between the centroids of the opposite far holes was known to be 1 m). In addition, there was a hydraulic pressure gauge (4) on the rig to measure the hydraulic pressure that actuated the caliper. Finally, the data reading and logging was performed by a custom made LabView software running on a local desktop along with the appropriate hardware (5), which will be presented separately in this Section.

In terms of the loading procedure, it was decided that the first step, and the beginning of the strain values logging, is with the disc without any torque applied and not even the brake applied. In the second step, only the brake was applied and strain values were logged. In the third step, the bar and the weight carrier were attached on the disc (the bar horizontally and the weight carrier vertically as shown in Figure 5-20). These two components, due to their own weight, applied torque to the disc equal to 133.3 Nm. In the fourth step, weights were placed on the weight carrier so that the gross weight of the bar, the weight carrier and the

weights would apply torque as close as possible to 200 Nm. From the fifth until the twelfth step, weights were placed on the weight carrier that increased the applied torque by 100 Nm every time until it reached the value of about 1000 Nm. For every step the values of strain were recorded. Then, in the last step all the weights were removed, including the bar and weight carrier, and additionally the brake was disengaged. The final values of strain were recorded. The steps of the process are presented in Table 5-4.

Table 5-4 Experimental Loading Process

Step	Brake Applied	Applied Torque [Nm]
1	No	0.0
2	Yes	0.0
3	Yes	133.3
4	Yes	197.0
5	Yes	295.0
6	Yes	400.2
7	Yes	496.0
8	Yes	597.3
9	Yes	699.3
10	Yes	800.5
11	Yes	898.6
12	Yes	1000.7
13	No	0.0

The procedure was conducted for one disc position in relation to the loading bar which had to be positioned horizontally. Once the procedure was complete, the bar and weight carrier were removed. Then the disc was rotated by 90°, the bar

and weight carrier were positioned again and the procedure was repeated. Consequently, three further loading and off-loading measurements were collated with the disc being indexed by 90° each time. Since there were four bolts connecting the bar to the disc and only four corresponding holes, the relative position could only be varied by 90°. The intention was to gain a better understanding of the strain pattern in the fingers around the disc, also to compare the strain measurements obtained by the two sensors (for identical positions on the disc in relation to the loading bar) and, most importantly, compare the results with the FE predictions.

The first position was named “0°”, the second “90°”, the third “180°”, and the last “270°”, as graphically presented relative to the caliper/pad position in Figure 5-21.

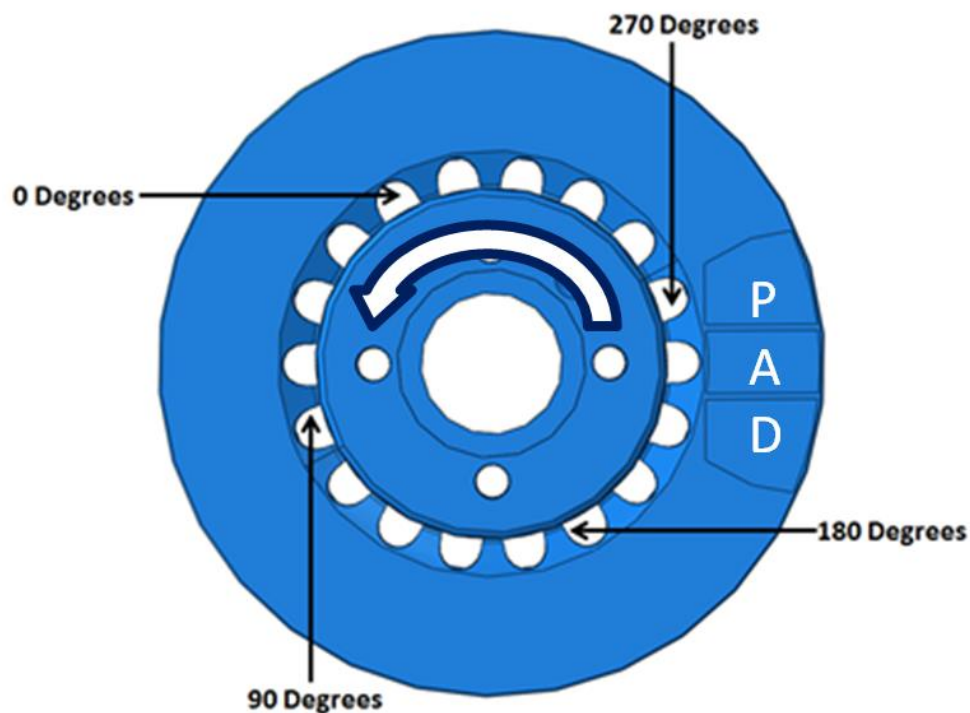


Figure 5-21 Positions of the Strain Gauges

5.3.3 Torsion Test Results

The analyses of the measured strains commenced with the values logged for the chain strain gauges. When reviewing the results, it was evident that the recorded values fluctuate and the results were deemed inconclusive and were not

processed further (Figure 5-22). Nevertheless, the results from the optical fibre and the single 5 mm X 1.8 mm strain gauge were very clear and repeatable and could be analysed. As a result, data from these two sensors will be presented and compared next.

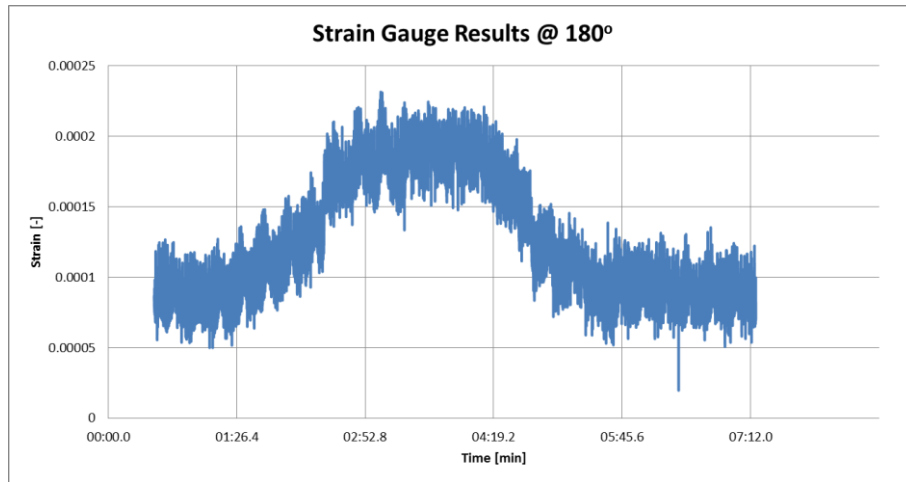


Figure 5-22 Chain SG Results at 180°

From both the FE predictions and the physical testing, it was observed that the peak values of strains (and therefore the stresses) were at the 270° position, that is in agreement with the findings of (Kim, et al., 2014). Thus, Figure 5-23 shows only the predicted and measured strains at this location. The values from the FE predicted strains were the average of the nodal strains for five nodes in a position adjacent to the strain gauge and the optical sensor. The values of the five nodes had to be averaged as both the strain gauge and the optical fibre had sensors 5 mm long that were averaging their recoded values.

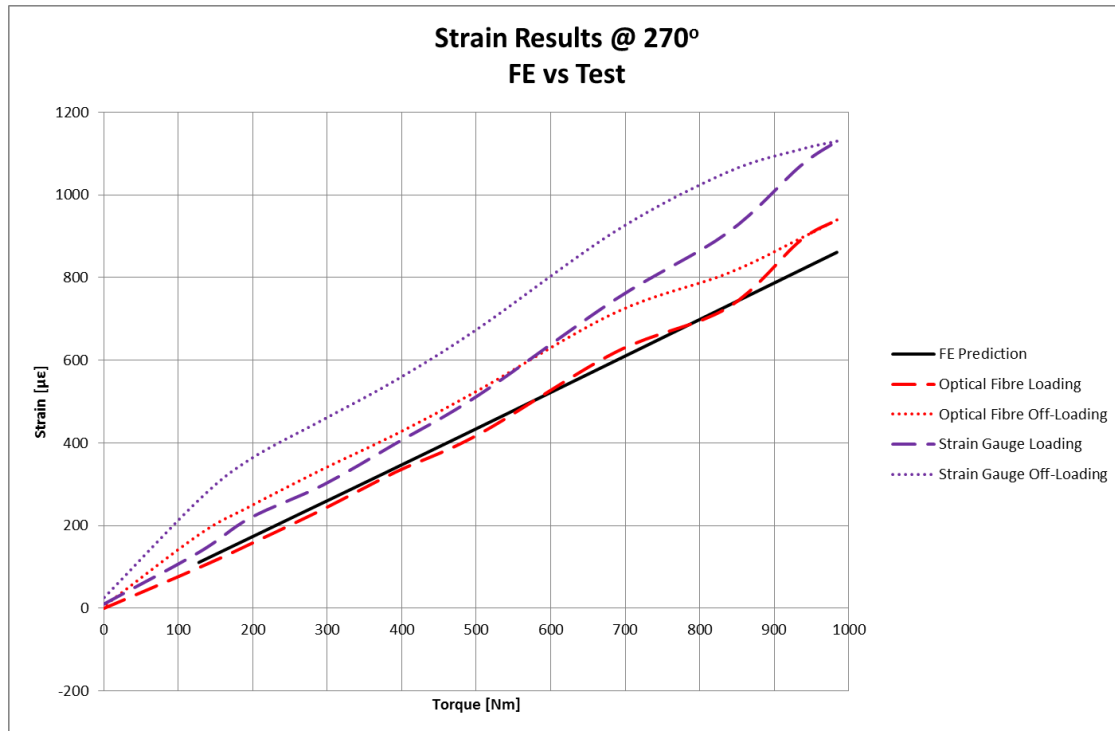


Figure 5-23 Comparison Between Predicted and Measured Strains

Figure 5-23 shows that a very good correlation has been achieved between the measurements using the optical strain sensor and the predicted strains from the finite element model where hexahedra elements with non-linear properties were used. The agreement is particularly good, with practically identical values obtained in the loading phase. Off-loading shows some hysteresis with strains being about 100 $\mu\epsilon$ higher. Interestingly, the strain gauge results are higher by 50 $\mu\epsilon$ at lower to medium torques, with differences increasing to 200 $\mu\epsilon$ at maximum torque. Strain gauge results also show higher hysteresis.

5.4 Fracture Torsion Test

The aim of the third test was to examine the true torque limit of the fingered hub design by loading the Elise Version 3 prototype disc to the point of fracture.

5.4.1 Fracture Test Process

The experimental set up from the torsion test with the instrumented disc (Figure 5-20) was the basis of this test as well, but included some modifications. Because the objective was to break the disc, a third Elise Version 3 prototype was

manufactured but no sensors to measure strains were installed in view of the fact that sensors are sensitive, expensive pieces of equipment and a broken disc would damage them irreversibly. To overcome the absence of strain sensors, a load cell (1) was used to measure the force applied on the steel bar (Figure 5-24).

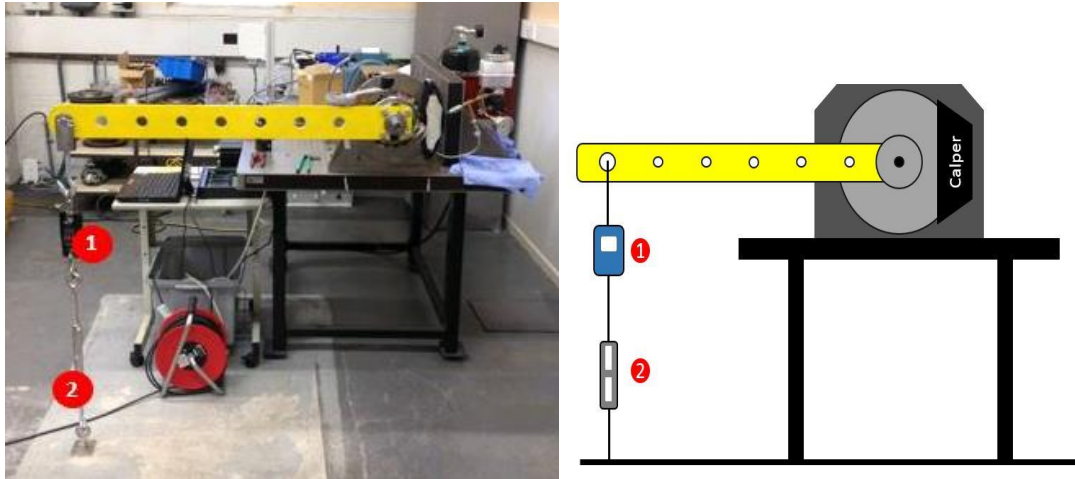


Figure 5-24 Fracture Torsion Rig Set up

Using a turnbuckle (2), which can be seen in Figure 5-24, the load was applied. The solution of the turnbuckle was chosen, because the available weights were not enough to cause the disc to break. Similarly to the process followed in Section 5.3.2, the torque bar was held in horizontal position and the prototype brake disc was clamped by the air over the hydraulic actuation system and a bolt as the friction torque was insufficient.

5.4.2 Fracture Test Results

By considering that the Maximum Principal Stress is the most prone to cause mechanical failure and that the disc was made out of cast iron with an approximate UTS of 150 MPa, the expected braking torque that would cause disc fracture was between 1200 Nm and 1490 Nm (Figure 3-24 and Figure 3-25). However, the measured force that resulted the disc to crack was 2481 N, which translates to 2481 Nm of torque. That is twice the anticipated fracture torque as can be seen in Figure 3-24 and Figure 3-25. Moreover, it is almost three times higher than the maximum braking torque (882 Nm) that the Lotus Elise can develop.

In order to establish the safety factor for the Elise Version 3 disc, FE analyses were conducted with gradually increasing the applied torque up to 2500 Nm. The model used for the simulations was the one with the hexahedra linear elements and the non-linear material properties that gave a good correlation with the physical test seen in Figure 5-23. The analyses showed that for 2481 Nm of braking torque, the fracture stress would be 250 MPa (Figure 5-25).

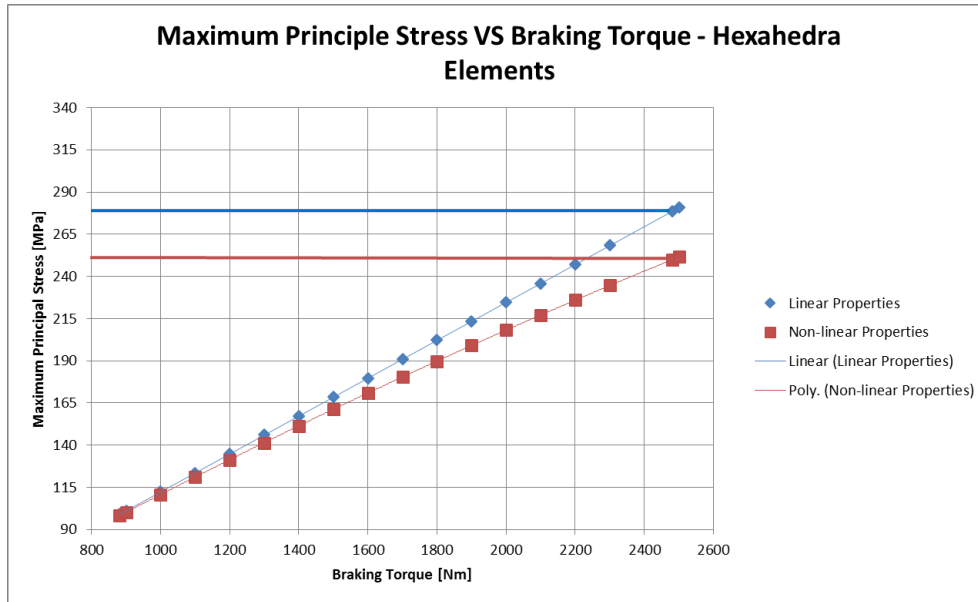


Figure 5-25 Predicted Fracture Stress

As shown in Figure 3-23, the predicted Maximum Principal stress resulting from the maximum braking torque developed by the vehicle is 98.2 MPa. By using equation (5-4), the safety factor for the prototype disc is 2.55.

$$Factor\ of\ Safety = \frac{yield\ stress}{working\ stress} \quad (5-4)\ Safety\ Factor$$

Lastly, in Figure 5-26 one of the cracks can be seen after the prototype disc was broken. The crack was in the area where the mechanical FE models predicted (Chapter 3) that the highest Maximum Principal stresses will occur. This also proved the accuracy of the mechanical modelling process followed.

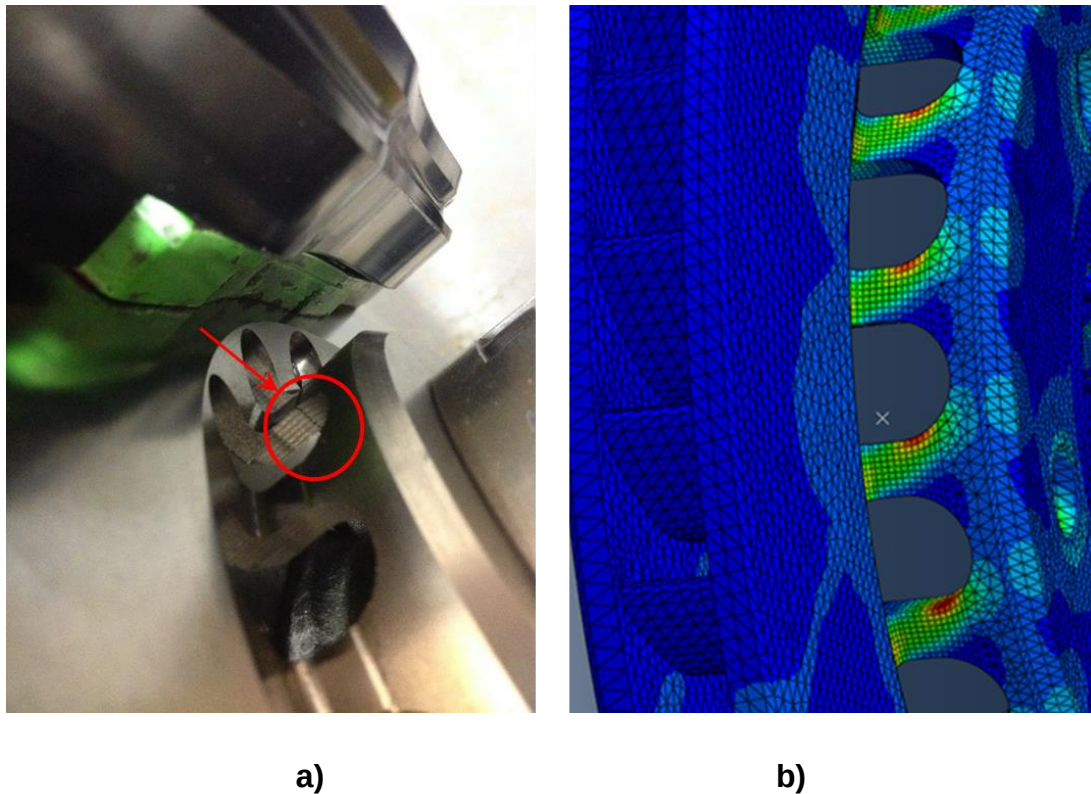


Figure 5-26 a) Crack on a Finger – Fracture Test b) Area of Highest Maximum Principal Stress – FE Modelling

5.5 Chapter Summary

Three physical tests were presented within this Chapter. The tests were aiming to improve the understanding of the grey cast iron's behaviour under mechanical loading, validate the first modelling case discussed in Chapter 3 and identify the true mechanical limit of the Elise Version 3 design.

The results of the first test showed that in ambient temperature the cast iron can withstand stresses of up to 350 MPa under bending. The second test validated the first modelling case of Chapter 3, since its results had very good correlation with the FE predicted strains.

Finally, the limit of the Elise Version 3's design was found experimentally in the third test and it turned out that the torque that caused fracture was 79% higher than the expected fracture torque. Moreover, it was 181% higher than the maximum braking torque that the Elise Version 3 can develop giving a safety factor of 2.5 to the fingered hub disc prototype.

6 STRUCTURAL OPTIMISATION

6.1 Introduction

In the current Chapter, the methodology to structurally optimise the Elise Version 3 prototype disc using Computer Aided Engineering (CAE) software will be presented. The results of the optimisation work will be exhibited and discussed.

Following the findings of Chapters 3 and 5, where it was observed numerically and experimentally that the Maximum Principal stress criterion on the fingers will cause the fracture of the disc, the first optimisation target was to reduce the levels of Maximum Principal stresses on the disc fingers. The second optimisation target was to investigate the possibility of further mass reduction, by removing material from areas where there are no load paths.

In order to achieve both optimisation targets, the techniques of Shape Optimisation and Topology Optimisation were utilised, each emphasising only on one target. Shape Optimisation was used to minimise the Maximum Principal stresses on the fingers and Topology Optimisation was employed to reduce the mass of the disc. The FE solver used for the aforementioned techniques was OptiStruct, the pre-processing software was HyperMesh and the post-processing software was HyperView. All these packages were provided by Altair Engineering.

6.2 Related Literature

In this Section, the principles of structural optimisation are reviewed followed by investigations of practical applications of the method, techniques and some shortcomings. As already presented in Section 1.2, one of the tasks in the present work was to structurally optimise the prototype of the Lotus Elise disc that was created aiming into even increased weight savings compared to the standard S2 disc. The CAE tools used were the Shape Optimisation and Topology Optimisation, two relatively modern techniques that optimise the shape of the geometry and the distribution of material in an area designated by the user as “design area/region/space/domain” in order to achieve most efficiently the

optimisation objectives. They combine the use of FE analysis and mathematical optimisation algorithms and they are applicable in isotropic materials.

Structural optimisation is based on two methods. Homogenisation is the first one introduced by Bendsøe and Kikuchi (Bendsøe & Kikuchi, 1988), where the authors aimed to introduce a method of generating both the optimal topology and the optimal shape of a structure. Doing that at the time was difficult as the usual approach of structural optimisation was to use parametric equations to describe the design boundaries. As a result, they created an alternative method, the homogenisation, where the target was to optimally distribute the void in a rough block of space by boundary variations. The introduced method comprises two steps. In the first, a non-smooth estimate of the structural member is created and in the second step the design computed is optimised. The second method is the density method introduced by (Mlejnek & Schirrmacher, 1993), where the Young's moduli of the elements are approximated by an energy approach. In Atom, the optimisation module of Abaqus CAE that was used for the topology optimisation, the homogenisation method is used.

Bakhtiary et al., presented the different types of optimisation available at the time of their paper publication by FE software (Bakhtiary, et al., 1996):

- Sizing optimisation (Figure 6-1 – a),
- Shape optimisation (Figure 6-1 – b),
- Topology optimisation (Figure 6-1 – c).

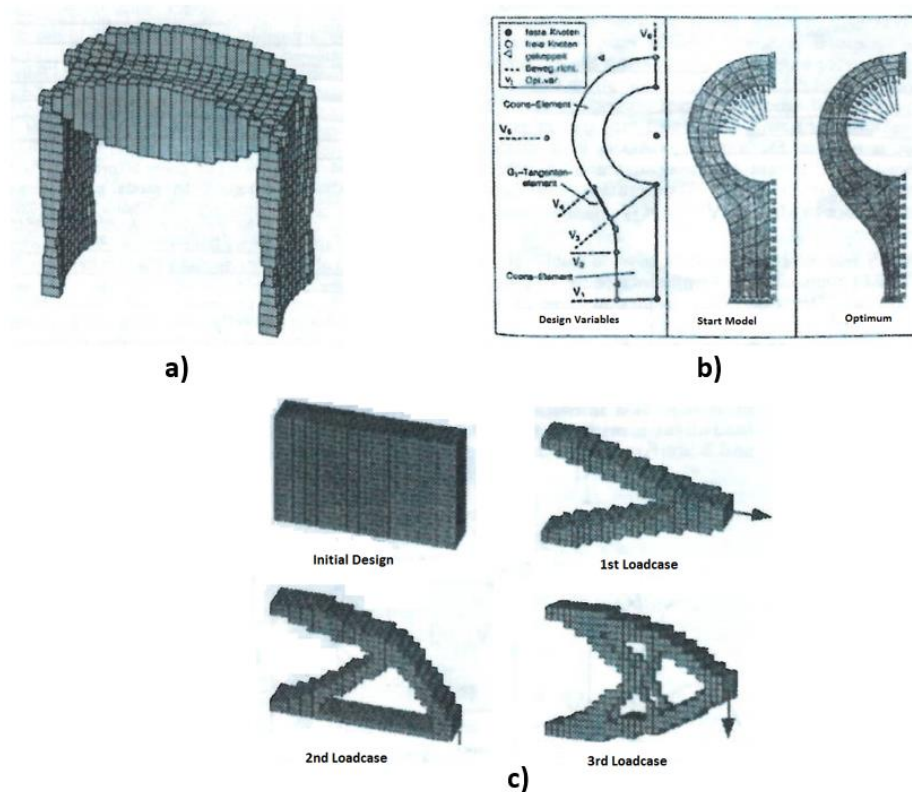


Figure 6-1 Optimisation Types (Bakhtiary, et al., 1996)

The sizing optimisation adjusts the thickness or cross section of the finite elements. In shape optimisation, two approaches were presented: the FE models based and the geometry based. In the first, the coordinates of the surface nodes are the design variables and are modified during the optimisation. In the second approach, the design variables are the parameters of the geometry. In topology optimisation, a 2D or 3D area is used with homogenous material distribution and depending on the boundary conditions the area becomes highly inhomogeneous in order for the optimal design to be created.

Strömberg worked on a method to automatically generate trade-off curves between compliance and weight when conducting numerical topology optimisation (Strömberg, 2011). It was shown that this approach was treating efficiently manufacturing constraints.

A number of publications were studied with practical applications of the topology optimisation technique for the purpose of gaining knowledge on good practises, learn troubleshooting approaches and explore the potentials of the method. One

of the early examples of topology optimisation is given by Xie and Stevens. By using the FE mesh created during the FE structural analysis, the authors imposed a rejection criterion (RC) based on the von Mises stress predicted values in order to neglect elements and only keep the ones necessary for the component's structural integrity. The RC includes a rejection ratio (RR) and an evolution ratio (ER). The software eliminates elements until it reaches a steady state according to the RR and thereafter it repeats the process until it satisfies the ER. They present the functionality of their software with four 2D examples and they conclude that the starting values for the RR and the ER should be as low as 1% in order to avoid over rejection of elements and that the usual value of converging for RR was 25% (Xie & Steven, 1993)..

Topology optimisation is widely applied for the optimisation of automotive Body in White (BIW). Wang et al., used the method in an automotive body aiming to increase its rigidity. On the FE of the body structured comprised by 2D shell elements another layer of shell elements was applied with the exact number of elements (Wang, et al., 2004).

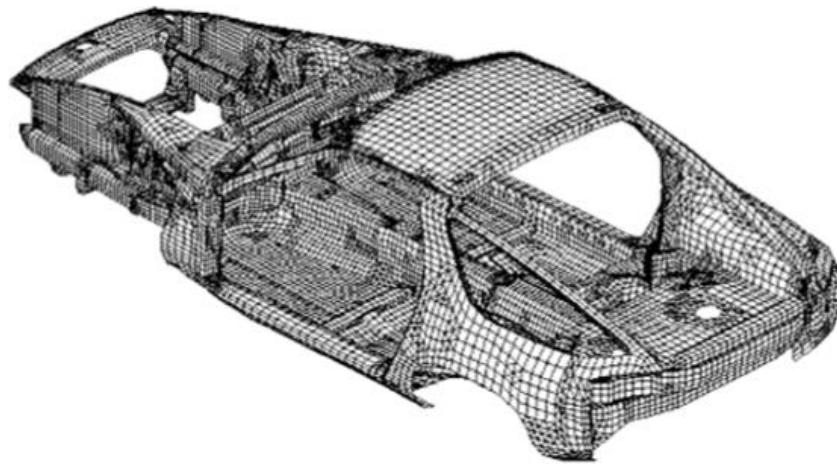


Figure 6-2 Topology Optimisation Model of the Full BIW (Wang, et al., 2004)

Then, by using topology optimisation, the authors investigated the areas of the structure where the additional material is required in order to maximise the first torsional frequency, the first bending frequency and the average of the two. Consequently, they used another optimisation technique, the “sizing optimisation”, in the initial FE mesh of the body structure investigating which

areas would increase the rigidity of the structure if there was an increase in their thickness.

Another example of topology optimisation application in an automotive structure was presented by Laxman et al. The authors started by categorising and discussing the optimisation techniques that can be used for the optimisation of BIW vehicle structures in an attempt to determine the best use of each of these techniques (Laxman, et al., 2009).

- Topology optimisation: can determine the optimal material distribution within a package space for a set of boundary conditions,
- Sizing optimisation: modification of the properties of a part's section to achieve the specified design target,
- Topometry optimisation: advanced sizing optimisation based on element sizing instead of property sizing.

They used these techniques in a BIW vehicle structure to quantify the effectiveness and usefulness of each method. The analyses were in the linear domain, the mechanical loadings applied were the bending and torsion loading cases, and also the first six non-rigid modes were investigated. They found that topology optimisation is useful in the early stages of BIW development where conceptual designs are generated. The other two methods are more effective when improvements are required in existing designs. Finally, they concluded that if enhancements on sheet metal designs already in use are sought, the technique of topometry optimisation is ideal as it indicated areas sensitive for the BIW design and identified weight saving opportunities by the application of tailor welded blanks in specific areas of the body structure.

Cavazzuti et al., presented an optimisation approach applied on an automotive body of a mid-engine car (Cavazzuti, et al., 2011). A large area was defined as design domain without taking into consideration the space required for the engine, the gearbox and the fuel tank of the car in order to provide as much space as possible for optimisation (Figure 6-3).

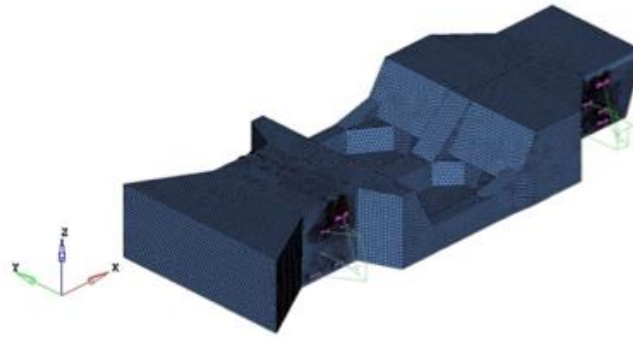


Figure 6-3 Design Domain (Cavazzuti, et al., 2011)

The physical/geometrical constraints applied were: the wheelbase and track of the vehicle, the suspension position and geometry and in addition, the passenger compartment were set as non-design domains. The loading cases considered were the global bending stiffness, the global torsional stiffness, the local stiffness of the engine, gearbox, suspension joints, the modal response and finally the frontal crush response. As a first step, they conducted topology optimisation to see the load paths and get the distribution of material density around the design domain (Figure 6-4).

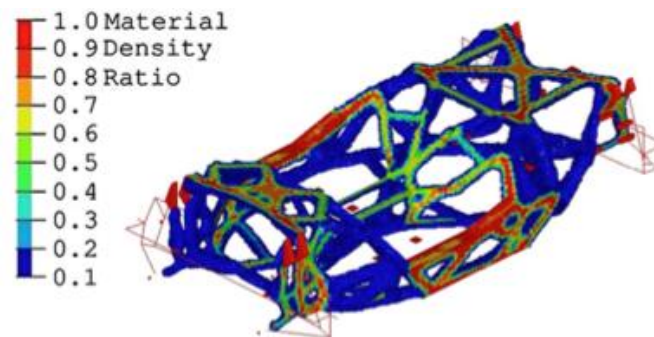


Figure 6-4 Optimised Structure Under the Torsion Loading Case (Cavazzuti, et al., 2011)

In the second step, they performed topometry optimisation aiming in mirror improvements from the first step in terms of parts that could be removed to reduce weight and components that should be re-oriented to maximise the stiffness response of the structure. In the final step, sizing optimisation was incorporated to optimise the thickness distribution of the structure's shell elements. In the findings, it was mentioned that experience is needed by the designer to decide

what truss-like parts of the structure with low thickness must be replaced with thin-walled structural components for the structure to retain its rigidity.

Numerous publications of topology optimisation applications in automotive components were studied. Shin et al., describe an optimisation method applied on the inner panel of an automotive door. They used an existing design as a starting point and during the conceptual phase they conducted topology optimisation to determine the thickness distribution on the parts (comprising the panel), the number of parts needed and the approximate welding lines. The objective in that phase was to reduce the strain energy as much as possible for the given loading conditions and reduce the material used. Thereafter, the number of parts was decided and size optimisation took place to identify the optimal thickness of each part. Moreover, shape optimisation was conducted to find the optimal welding line. It was mentioned that depending on the boundary conditions, size and shape optimisation can be done simultaneously, but it was not possible in the study described. By applying these optimisation steps (Figure 6-5), the weight of the inner panel was reduced by 8.72% and the total weight of the door was reduced by 3.23% (Shin, et al., 2002).

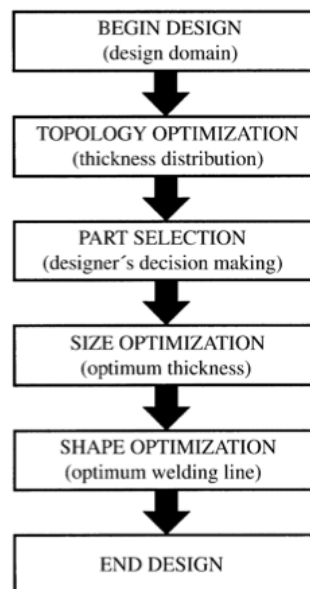


Figure 6-5 Optimisation Steps (Shin, et al., 2002)

Chiandussi et al., described an approach to optimise the geometry of a rear suspension sub frame of a mid-size vehicle. They used the power-law approach, Solid Isotropic Material with Penalization (SIMP), of topology optimisation and by calculating the Lagrange multiplier of the volume constraint before each new material density layout modification, they were able to indirectly control the density distribution of the finite elements. This method resulted in the number of elements with intermediate density (0-1) being lower than 1% for an optimisation based on maximum von Mises stresses and minimum first natural frequency (Chiandussi, et al., 2004).

A transmission housing unit was optimised aiming to achieve weight reductions and maintain or increase the rigidity of the structure by Ogata et al. From prior experience they found out that the topology optimisation method on components, which have to preserve air or water tightness, may remove material from the walls of these components leading them to lose their air or water tightness ability (Figure 6-6) (Ogata & Hiraoka, 2005).

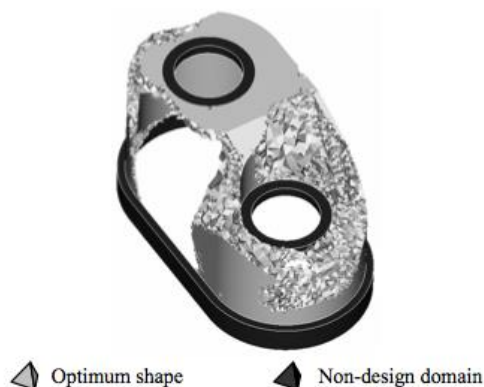


Figure 6-6 Holes on the Walls of Water-tight Component (Ogata & Hiraoka, 2005)

In order to overcome this situation, they created a wall of shell elements in the area where the structure had to be hermetic and assigned that area to be a non-design domain (Figure 6-7). They also stated that the appropriate optimisation method has to be implemented according to the maturity of the design. For instance, they applied topology optimisation in the first and second stages where conceptual designs were generated and external ribs were created, but the shape

optimisation method was used in the final stage to make minor adjustments on the thickness of the external ribs.

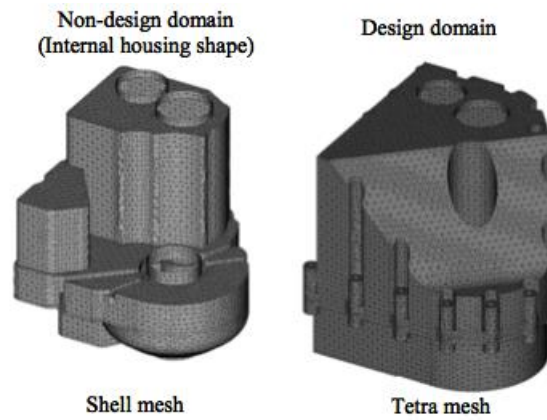


Figure 6-7 Shell and Solid Mesh (Ogata & Hiraoka, 2005)

Edara applied the topology optimisation method on several other automotive components (Edara, 2007). The author mentioned that the success of the optimisation lies in the good cooperation of the design, analysis and manufacturing teams in an industrial environment with the additional involvement of the suppliers. Thereafter, he presented a flow chart of the optimisation process (Figure 6-8).

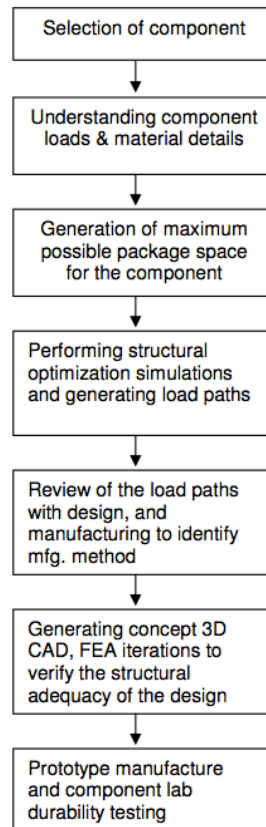


Figure 6-8 Edara's Topology Optimisation Flowchart (Edara, 2007)

The optimisation should start by conducting simple linear static FE analysis for the sake of verifying the setup of the FE model, its functionality and the validity of its predictions. Furthermore, the optimisation responses, objectives, targets and constraints were discussed followed by the recommendation to keep elements of density equal and higher than 0.5 in the post processing of the optimisation results. With regards to the cases where multiple loadings are applied, Edara that either separate analyses should be conducted with the different loadings and their results must be superimposed or analyses with all the different loadings with the addition of weighting factors for the predominant loads should take place.

Additionally, according to the author, the optimisation results have to be analysed with the use of FEA to see if the developed component satisfies the design targets and then tested in physical tests. The manufacturability is an area also to be considered along with the cost for new machinery and the manufacturing processes of the optimised components. Finally, the importance of boundary conditions is highlighted by an example of optimising a bearings housing under

the same load but using two different ways to support it (Figure 6-9 and Figure 6-10).

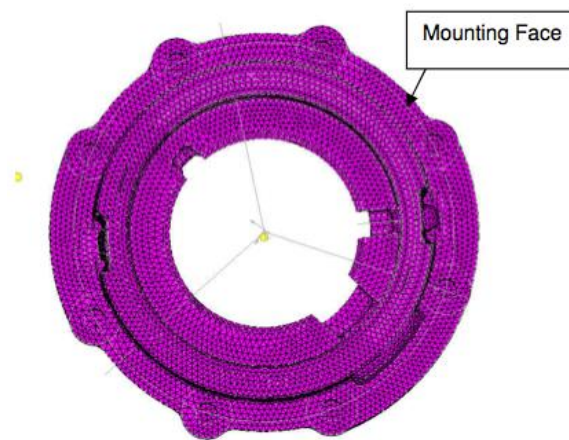


Figure 6-9 FE Model of the Bearings Housing (Edara, 2007)

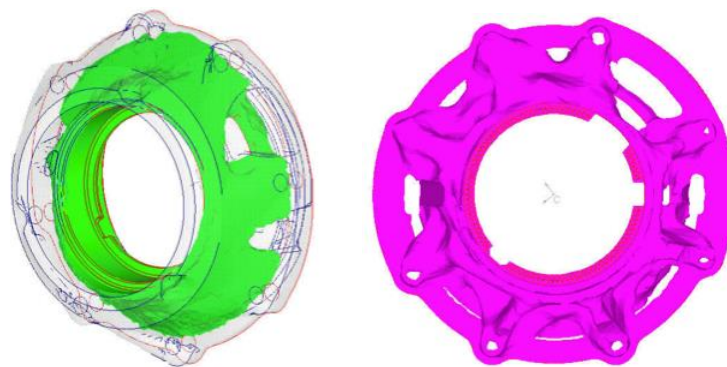


Figure 6-10 Optimised Bearings Housing Under Two different Sets of BCs (Edara, 2007)

Kumagai et al., presented a method to apply topology optimisation on a parking pawl component of an automatic transmission. They recognised that topology optimisation is a useful tool in the conceptual design stage of a product, but it can result to components that are difficult to be manufactured. In order to overcome this shortcoming, an algorithm that forced the optimisation software to keep the outer dimension of the component's cross section constant in, was created (Figure 6-11) (Kumagai, et al., 2016).



Figure 6-11 Optimised Component with Constant Cross Section (Kumagai, et al., 2016)

They compared the resulted optimised design using their method to a design derived by topology optimisation without their method. The comparison showed a decrease in mechanical performance for the component with constant cross section, but the authors stated that this was acceptable since the component was actually manufacturable.

Deng and Lan investigated an efficient method to conduct topology optimisation when multiple optimisation targets are set. They used an automotive door glass lifter as an example (Figure 6-12), for which they set specific stiffness and NVH targets (Deng & Lan, 2016).

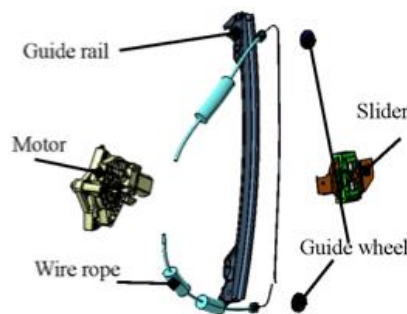


Figure 6-12 Door Glass Lifter Mechanism (Deng & Lan, 2016)

Initially, they conducted the optimisation only for the stiffness target and found that although it was very time-efficient, the resulted structure was not reasonable as there was almost no material in the centre of the component. Thereafter, they conducted the optimisation again by creating a compromising programming method that accounted both the stiffness and NVH targets. By using the linearity-weighted-sum method they transformed the multi-objective optimisation into a single optimisation problem. The results showed that although the second

optimisation approach was less time-efficient, as the number of iterations was higher compared to the first approach, the developed structure had reasonable material distribution and satisfied both the structural and NVH optimisation targets.

Although publications discussing topology optimisation on brake system components were identified, no documented work with regards to structural optimisation of brake disc rotors was found.

Edara used the method to optimise the performance and at the same time reduce the weight of a torque plate on a brake caliper of a heavy-duty commercial vehicle (Figure 6-13) (Edara, 2006).

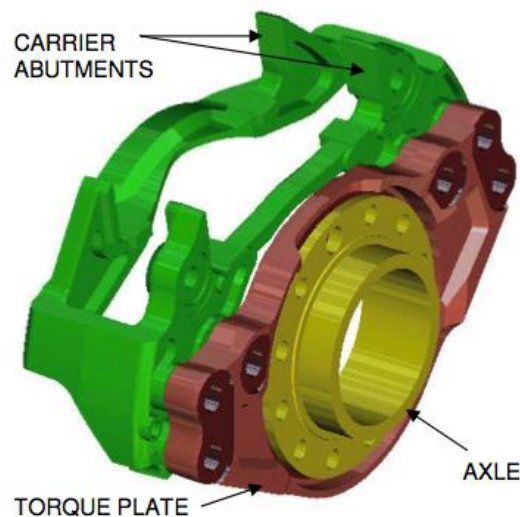


Figure 6-13 Brake Caliper Torque Plate (Edara, 2006)

The first step in the optimisation procedure included FE analysis of the existing torque plate for stresses and displacements. In the second step, the designable and non-designable spaces were defined and also since the plate was symmetrical only one half of the plate was modelled in order to reduce the computational power and time for the optimisation (Figure 6-14 – a).

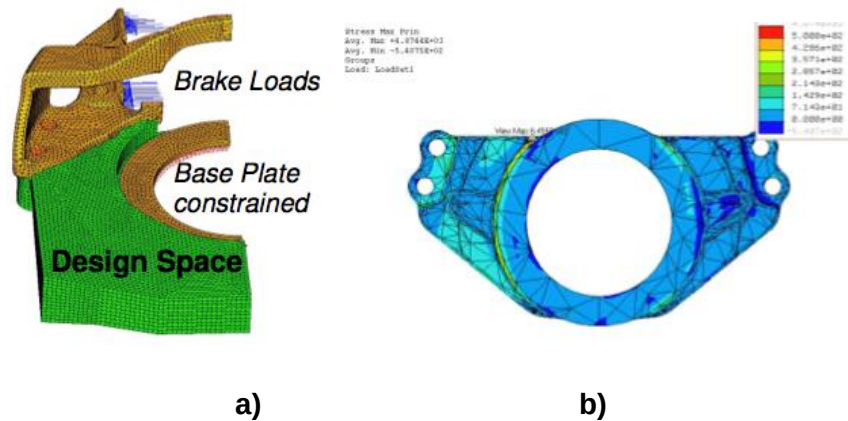


Figure 6-14 a) Half Plate Model with BCs, b) Optimised Plate (Edara, 2006)

Although the pad carried was not part of the torque plate, it was included in the analysis so that its influence in stiffness to be taken into consideration. In the third step, the boundary conditions were applied and optimisation targets based on maximum displacements and weight reductions were set. After 30 iterations the software produced an optimal design with distributions of density between 0 and 1 among the elements. The produced design was modified in step four in order to be manufacturable (Figure 6-14 – b). In the fifth step, FE analysis was conducted. The aim for the created design was to have similar or improved performance in terms of stresses and displacements compared to the baseline existing torque plate. The FE analysis showed better performance in comparison with the baseline design, something that was validated by physical tests conducted with a prototype plate. The newly designed component was 20% lighter than the previous.

Sergent et al., considered a design optimisation methodology aiming to reduce the weight of a fixed four-pot caliper, but at the same time keep its stiffness under the same load. They categorised the important parameters in topology optimisation (Sergent, et al., 2014):

- Designable volume,
- Non-designable volume,
- Boundary conditions,
- Optimisation constraints,
- Optimisation aim.

Following this, they presented the application of topology optimisation on a four-pot brake caliper (Figure 6-15). Attention was given in understanding the physics behind the operation of brake calipers, which would influence the boundary conditions (friction forces, actuating forces, constraints based on the way the caliper was mounted on the upright, etc.) in the numerical optimisations.

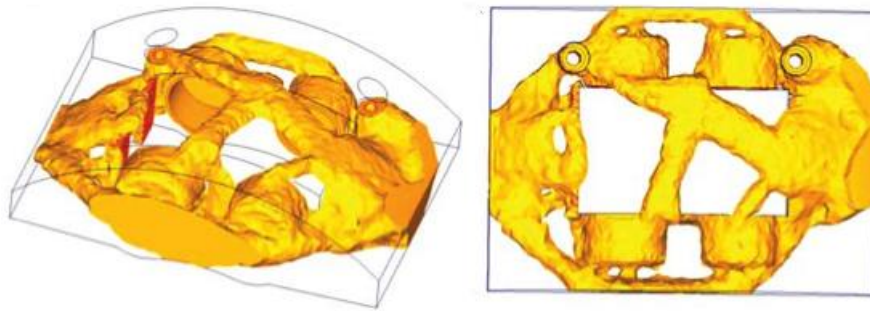


Figure 6-15 Material Density Distributions for Vales of 0.3 and Higher (Sergent, et al., 2014)

After optimising the baseline caliper, a second optimisation was conducted altering the designable volume to investigate the influence of this parameter. It was found that the resulted design did not differ greatly from the one obtained from the first optimisation procedure. Finally, the authors conducted a manual parametric study by manually adding or removing a number of design features aiming to quantify their impact on the performance of the caliper (Figure 6-16).

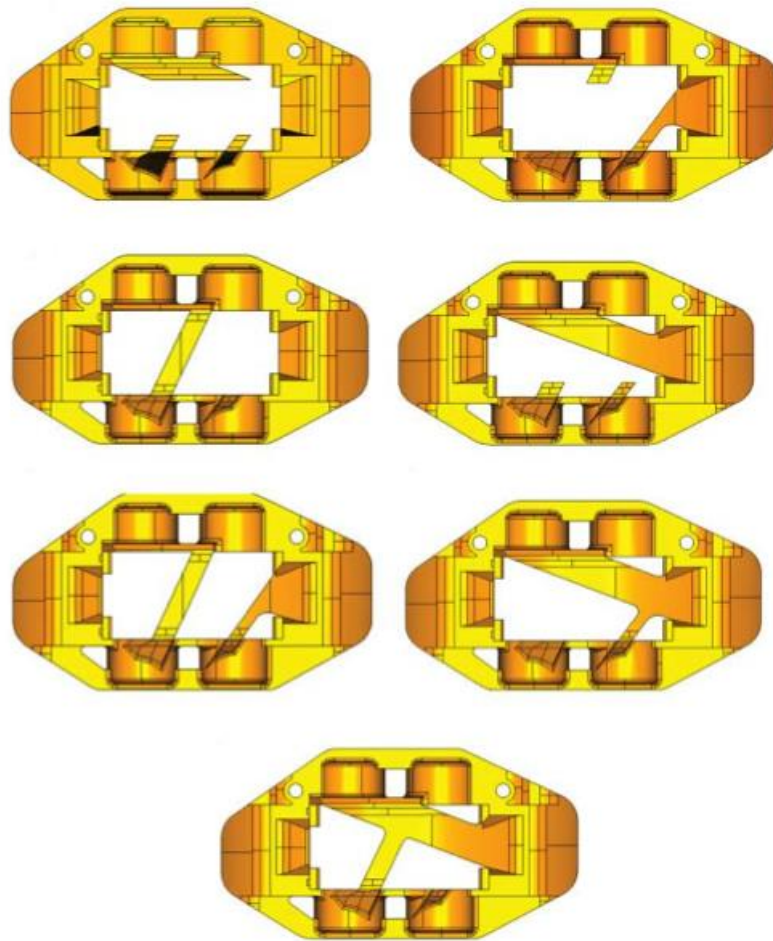


Figure 6-16 Parameters Manually Changed in the Parametric Study on Four-Pot Caliper (Sergent, et al., 2014)

They concluded that trying to optimise a design in such a way is incredibly complex and labour intensive compared to the use of FE topology optimisation software.

Wang et al., described the optimisation of the lever component on an air braking system for heavy-duty vehicles (Figure 6-17) (Wang, et al., 2014). By setting the reduction of von Mises stresses and displacements as the optimisation objectives, they achieved their stresses and displacements target values by marginally increasing the weight of the component.

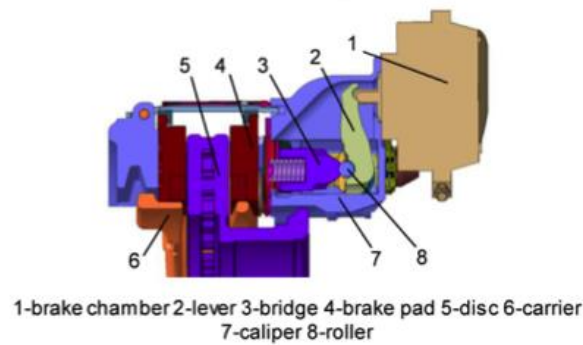


Figure 6-17 Cut-view of the Air-over-air Brake System (Wang, et al., 2014)

The topology optimisation method had been used on brake components optimisation for NVH purposes. Goto et al., studied ways to reduce the squeal from automotive braking components. Among the ways to achieve their aim, they investigated the application of topology optimisation for the sake of modifying the geometry of several components of the braking system (caliper, caliper carrier, pads, rotors) and achieving the target eigenvalues (Goto, et al., 2010). Inoue et al., investigated the possibility of using topology optimisation to reduce the weight and improve the NVH performance in braking systems. They created a simplified FE model of brake rotor and pad. The backplate thickness of the pad was set as a design parameter. The objective parameter was the minimisation of mass and the constraint condition was for the real parts of the eigenvalues to be lower than zero. They found that their optimisation method improved the plural unstable modes and at the same time offered weight reductions (Inoue, et al., 2015).

6.3 Step 7 – Elise Version 3 Shape Optimisation

The technique of Shape Optimisation was used for the optimisation target of reducing the Maximum Principal stresses on the Elise Version 3's fingers. The principle behind it for a 3D model is that for a given set of boundary conditions, the FE solver moves the elements' nodes into the three dimensional space accordingly, in order to achieve the optimisation target set by the user. Thus, with minor geometric adjustments, the desired results are achieved.

6.3.1 Modelling Approach

As with any optimisation technique, either it is a topology optimisation, a size optimisation or a shape optimisation, the influencing parameters when creating the FE model are:

- The definition of the Design and Non-Design spaces,
- The type of elements used,
- The material properties,
- The boundary conditions applied, which include the loads and the restrains,
- And finally the optimisation objectives and the optimisation constraints when needed.

All the parameters above will be presented and discussed thoroughly as they were vital to the model being efficiently built resulting to low running times and reliable results.

The Design and Non-Design spaces are defined in order to specify the area where the solver has to conduct the optimisation process. The entire FE model cannot be specified as Design space, because it will result in critical errors. As it is shown in Figure 6-18, for the Elise Version 3 model, the Design space was the area of the disc's fingers, highlighted in green colour. All the other areas, the hub, the friction faces and the cooling vanes, were considered not to be influential for the reduction of Maximum Principal stresses on the area of the fingers and so they were in the Non-Design space, highlighted in yellow.

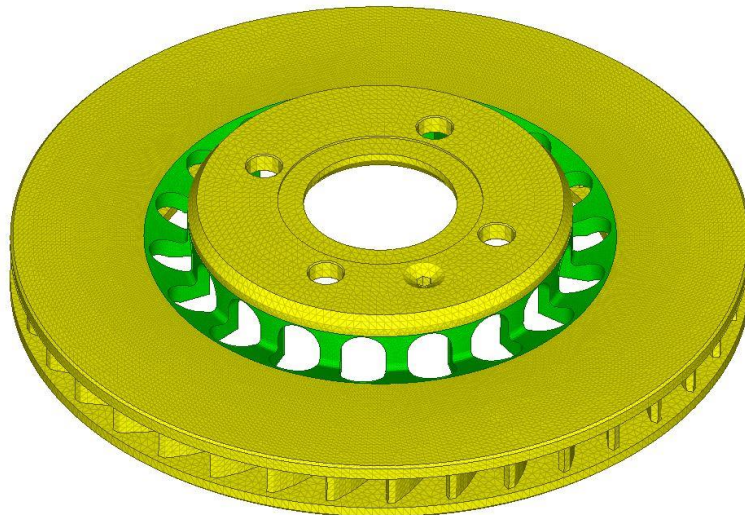


Figure 6-18 Design and Non-Design Spaces

Following the modelling approach of the mechanical simulations and aiming to capture the stress gradients with as great accuracy as possible, in the Design space the elements' size was set to 1 mm. In the Non-Design space, the elements' size was decided to be 3 mm for faster computational times. This resulted in the total amount of elements being 851,400, of which 425,695 were in the Design space, and the total number of nodes being 210,156.

For the Shape Optimisation study, simplified material properties were selected having in mind two major aspects. The first and most crucial was that as shown from the mechanical analyses of Chapter 3, when the maximum braking torque of the Lotus Elise is applied (which is 882 Nm), the highest von Mises and Maximum Principal stresses are below the yield threshold of the material. In addition, simplified material properties would result in faster simulation times. This is even more important for structural optimisation analyses, where a number of iterations takes place and within each iteration mechanical analyses and nodes or elements adjustments take place. As a result, it was considered reasonable to use linear elastic properties including only the Young's modulus and the Poisson's ratio (Appendix B.1).

The boundary conditions were applied in accordance to the first modelling case of Chapter 3. The mechanical loads were created as follows: in each of the four mounting holes situated on the disc's top hat, 1D rigid elements (RBE2) were created that connected all the holes' nodes to a single point element in the middle of the node (Figure 6-19).

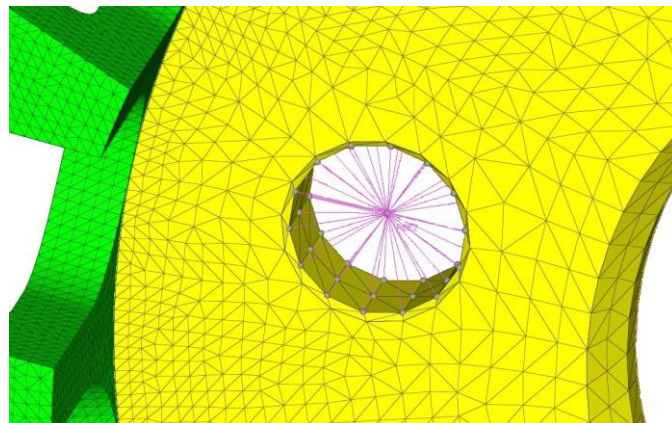


Figure 6-19 Rigid Elements (RBE2)

Thereafter, a total of four point loads were applied on each of the 1D point elements in the middle of each hole. The point loads individually applied a force equal to 4100 N which resulted in a total applied torque of 882 Nm. In order to allow only the rotational degree of freedom around the vertical axis of the disc, all the nodes located on the top hat's flange were connected to a single 1D point element in the middle of the top hat by rigid 1D elements (Figure 6-20). A cylindrical coordinate system was created in the middle of the top hat and it was used as a reference axis in order to block any translation and any rotation of the flange's nodes apart from the axial rotation (Figure 6-20).

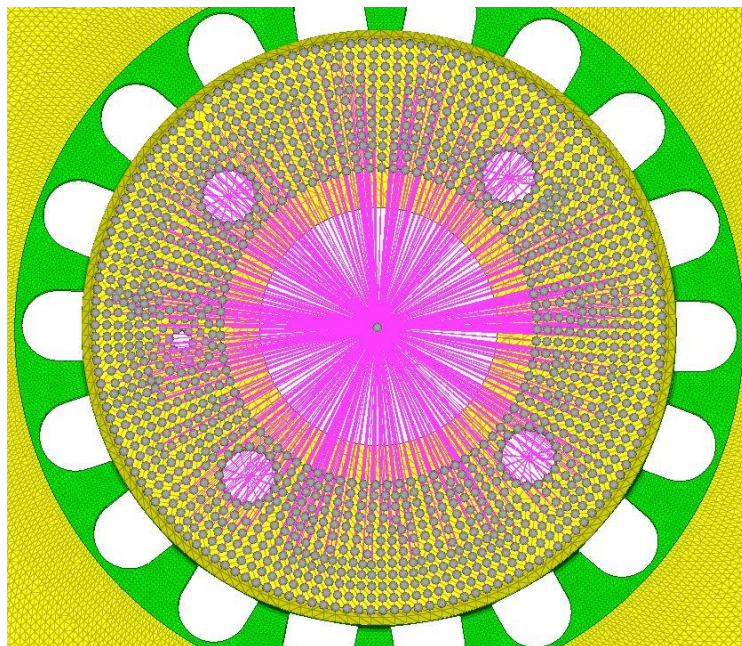


Figure 6-20 Rigid Elements on the Top Hat

The last boundary condition was to restraint the disc's friction faces. Whereas previously only a part of the friction faces, resembling the imprint of a pad, was restrained with the Encasté boundary condition, for the Shape Optimisation it was decided that the entire friction faces would be restrained (Figure 6-21). This decision was taken in order for the solver to create a circumferentially optimised geometry.

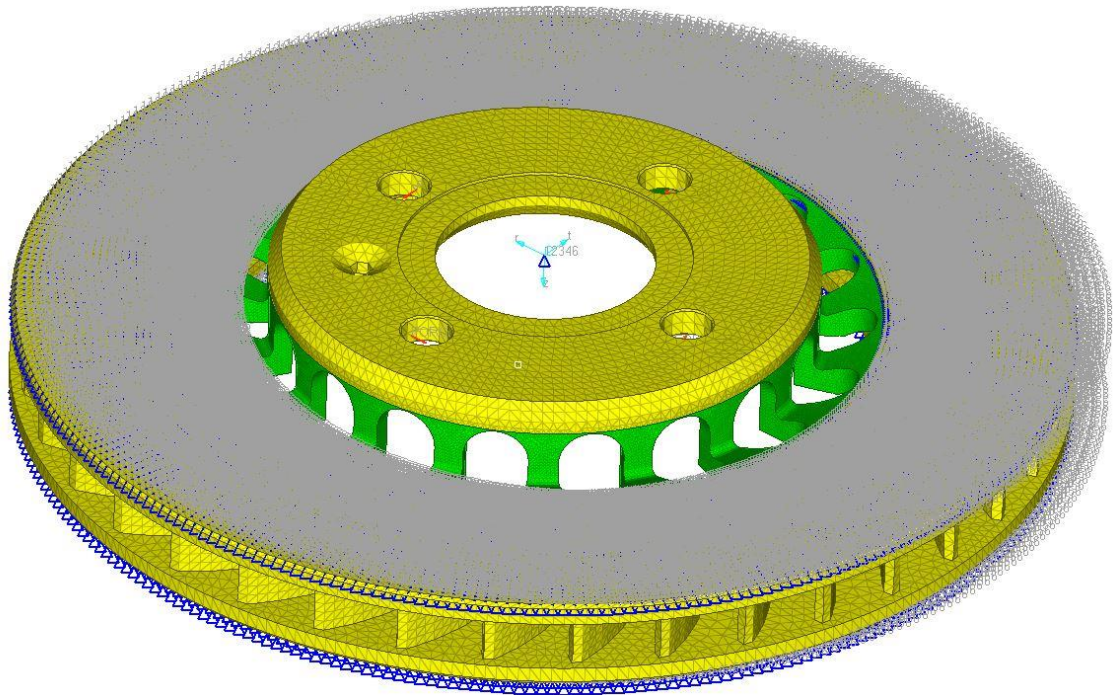


Figure 6-21 Encastré Boundary Condition on the Friction Faces

The final step in setting up the Shape Optimisation was to create the optimisation objectives. There was only one objective in the case studied and it was to minimise as much as possible the Maximum Principal stresses in the finger areas of the disc. So, the minimisation of the static Maximum Principal stress was selected only for the Design space of the FE model.

6.3.2 Shape Optimisation - Results

The analysis time lasted approximately 3 minutes and required 5 iterations to fully converge. At iteration “0”, when no optimisation was performed, the highest value of Maximum Principal stress on the fingers without averaging the nodal stresses was predicted to be 109.6 MPa (Figure 6-22). This compares favourable to the value of 108.8 MPa predicted when using Abaqus as solver for the same boundary conditions (Figure 6-23). This meant that although the mechanical analyses were conducted in two different FEA packages, the comparison between their results can be considered valid. This is due to the fact that the same solid geometry, the same material properties and most importantly the same boundary conditions were used in all the analyses.

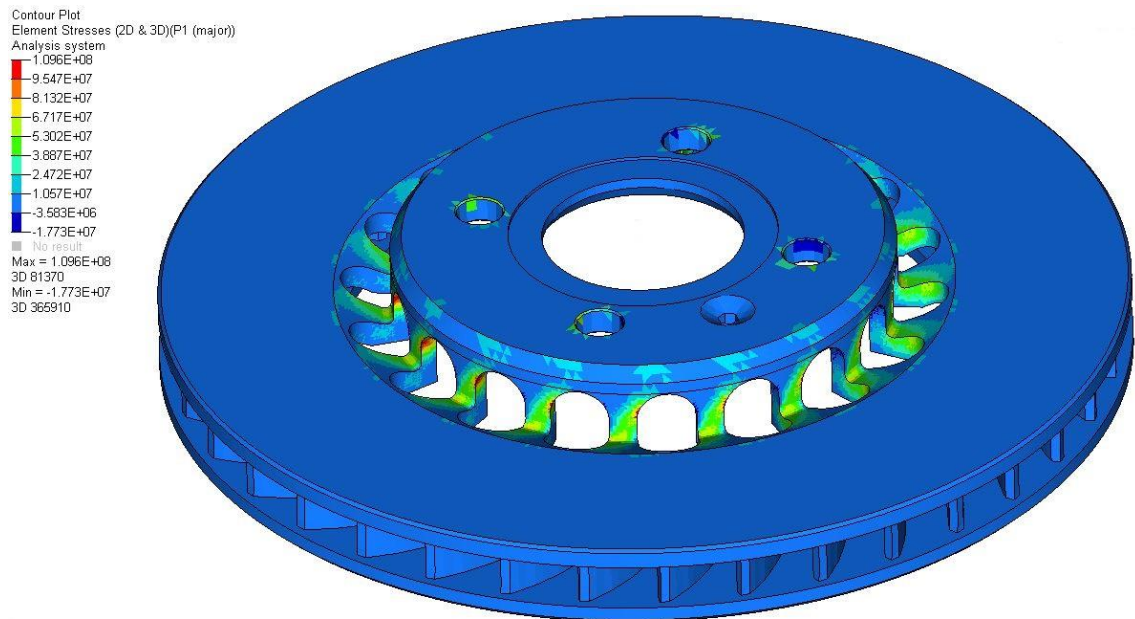


Figure 6-22 Non-Optimised Mechanical Analysis – OptiStruct

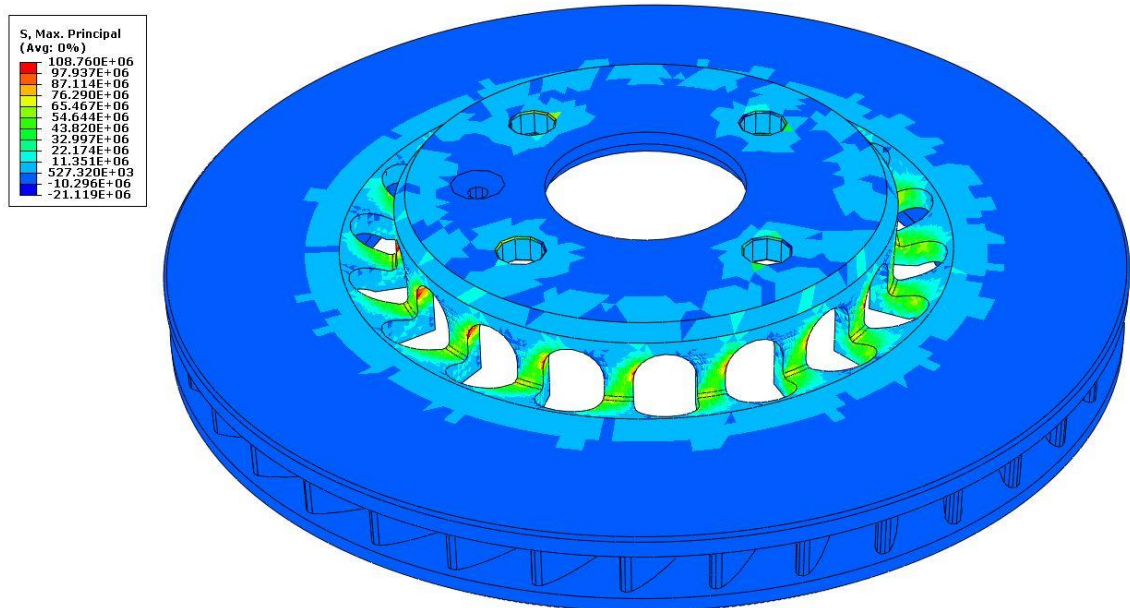


Figure 6-23 Non-Optimised Mechanical Analysis – Abaqus

At the fifth and final iteration, the highest Maximum Principal stress value predicted was 82.1 MPa (Figure 6-24). This result compared to the predicted value before the start of optimisation process entails a reduction of 25.1% (Table 6-1).

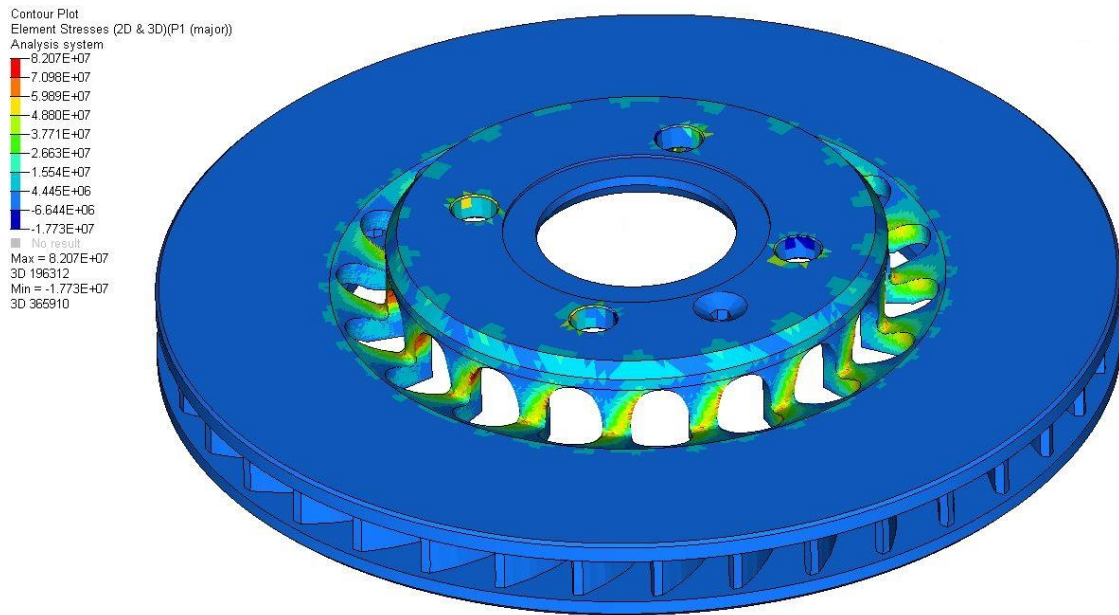


Figure 6-24 Shape-Optimised Mechanical Analysis – OptiStruct

In order to compare the safety factors between the optimised and the non-optimised designs, the results from OptiStruct were averaged by 75%. This was considered necessary in order to be consistent as the FE predictions from Abaqus that compared favourably with the result from the physical test, were also averaged by 75%.

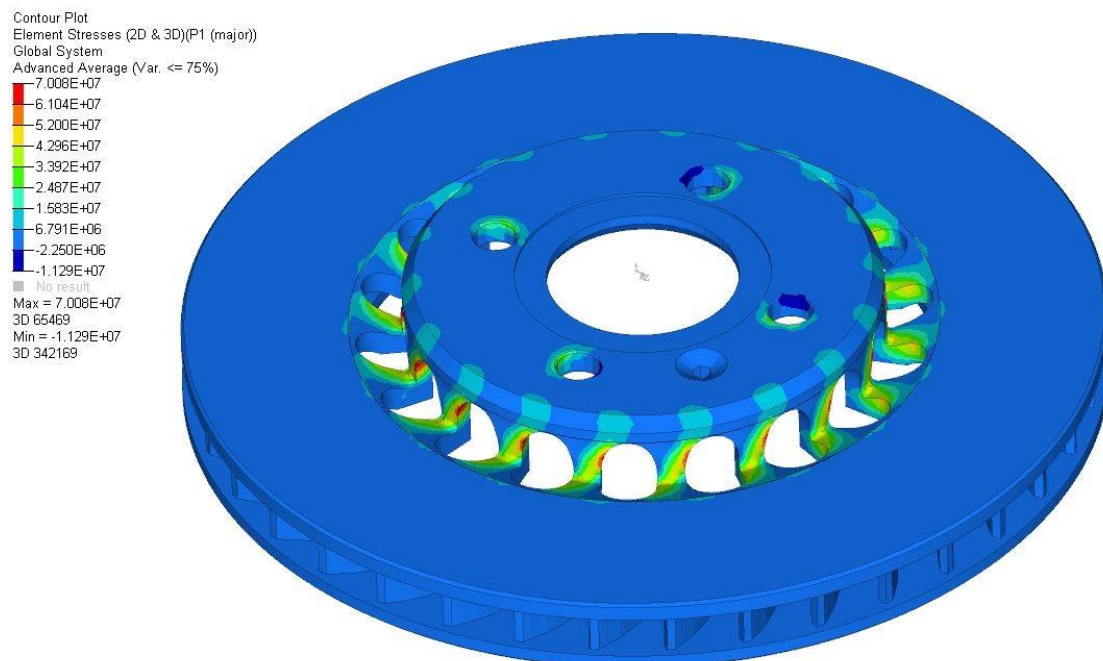


Figure 6-25 Shape-Optimised Mechanical Analysis – OptiStruct (75% Averaged)

Table 6-1 Shape Optimisation Results

Model	Max. Principal [MPa]	Difference [%]	Safety Factor
Non-Optimised - OptiStruct	103.0	-	2.43
Non-Optimised - Abaqus	98.2	-4.6	2.55
Shape-Optimised - OptiStruct	70.0	-32.0	3.57

In Table 6-1 it can be seen that after the Shape Optimisation, the safety factor has been increased by 1 unit. This was possible with the minor adjustments in the position of the elements' nodes conducted by the solver. In Figure 6-26, a contour plot of the shape change magnitude is presented. As it can be seen, the majority of the nodes were moved by less than 1 mm. Only in the middle of the finger's swan neck design there is a summit with the nodes moved by up to 1.55 mm.

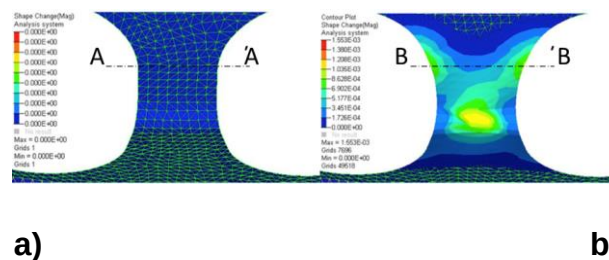


Figure 6-26 Shape Optimisation Effect on the Fingers – Outboard Side, a) Initial Geometry b) Optimised Geometry

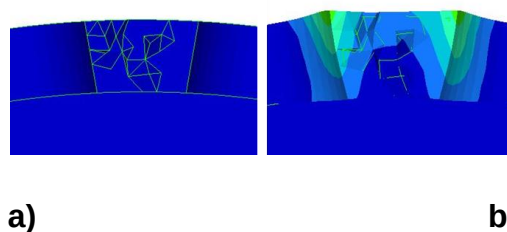


Figure 6-27 Shape Optimisation Effect on the Fingers – a) Section View A-A b) Section View B-B

The optimisation had resulted in each finger to be thickened, but mostly on the outboard side of the rotor. The inboard geometry mainly remained unaltered with the exception of the finger areas close to the hub as illustrated in Figure 6-28.

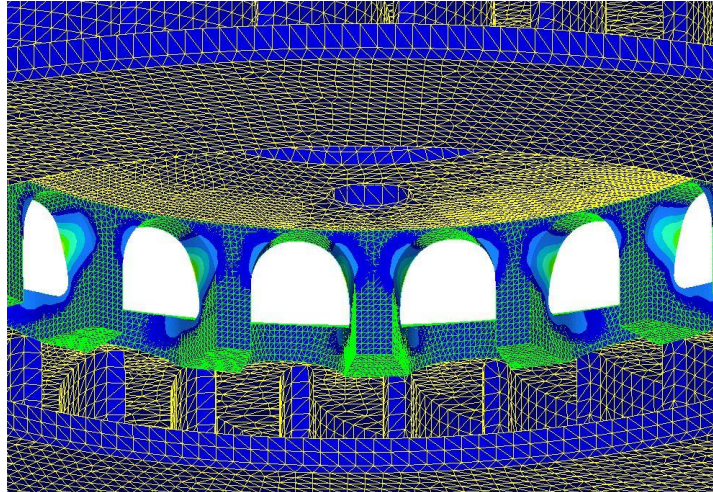


Figure 6-28 Shape Optimisation Effect on the Fingers – Inboard Side

6.4 Step 8 – Elise Version 3 Topology Optimisation

After achieving lower predicted Maximum Principal stresses on the Elise Version 3 fingers, the next step in the structural optimisation process was to investigate if it was possible to achieve further mass reduction. The CAE tool used in this step was Topology Optimisation, where the FE solver performs mechanical analysis and identifies the load paths in the Design space of a structure. Density values are assigned to the elements based on their position with regards to the load paths. Elements that are on the load paths or next to them are assigned with density value of “1” or close to “1”. On the contrary, elements in areas with no load paths or far from them are assigned with value of “0” or approximately “0”.

This process requires a number of iterations to converge and at the end the software presents the density value results. Thereafter, the user can decide the threshold value of density and neglect the elements below that value. If for example the threshold isodensity value is selected to be 0.5 (Edara, 2007), and all the elements below this density value will be removed from the FE model.

6.4.1 Modelling Approach

As a result of the optimisation regarding the minimisation of the Maximum Principal stresses in the area of the disc fingers being effectual in the previous step, it was decided in this step to import the optimised FE mesh and base the Topology Optimisation on it. As a result, the element type, size and numbers were the same as in the model for the Shape Optimisation. Furthermore, the assignment of Design and Non-Design spaces was exactly the same as in the previous step with the material properties and the boundary conditions being no exception (Section 6.3.1).

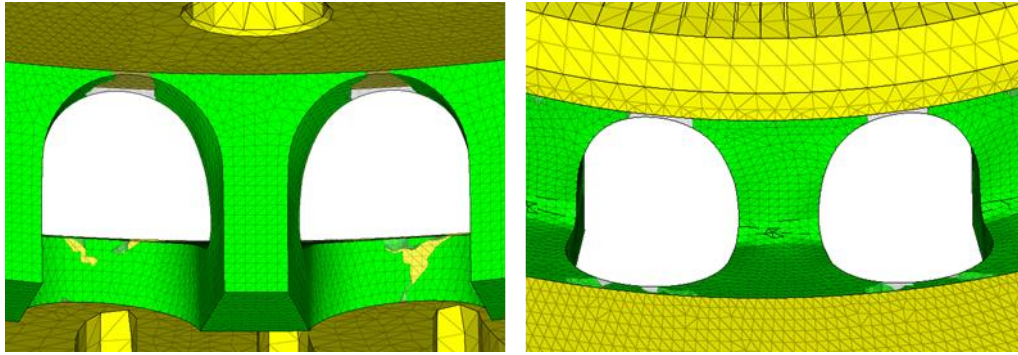
When the Topology Optimisation is created by the user, the following must be specified in order for the process to converge:

- The optimisation constraint,
- The optimisation objective.

Since the Maximum Principal stress results from the previous step were satisfying, it was decided to set the static Maximum Principal stresses as the optimisation constraint. Having constrained the stresses, the optimisation objective selected was to minimise the volume of the Design area. So, the solver had to work on minimising as much as possible the volume in the fingers of the disc while retaining the low stress levels obtained by Shape Optimisation.

6.4.2 Results

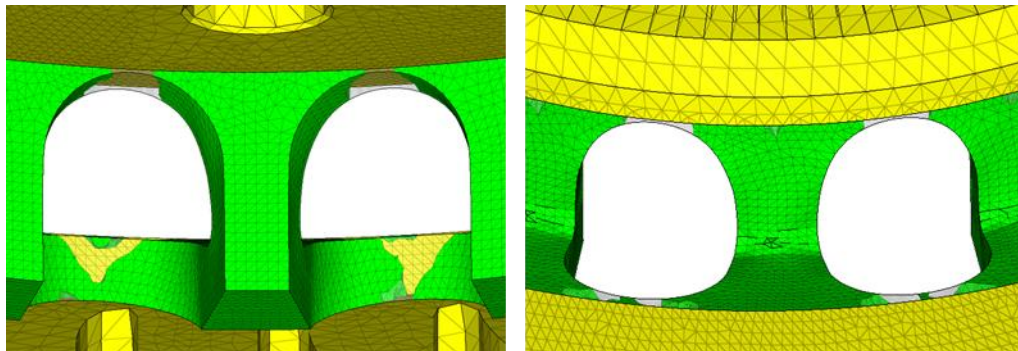
The analysis time lasted approximately 4 minutes and required a total number of 11 iterations to successfully converge. By visualising the results in HyperView, the areas where material was removed by OptiStruct can be identified. The simulation outcomes are illustrated in Figure 6-29 to Figure 6-35 based on their isodensities. It can be seen that the bulk amount of material removed by the software was in the areas where the fingers are connected to the hub, to the friction faces of the disc and at the rear and bottom part of the fingers themselves. Additionally, small portions of material were removed from the upper areas of the fingers close to the flange on the outboard side of the disc (Figure 6-29).



a)

b)

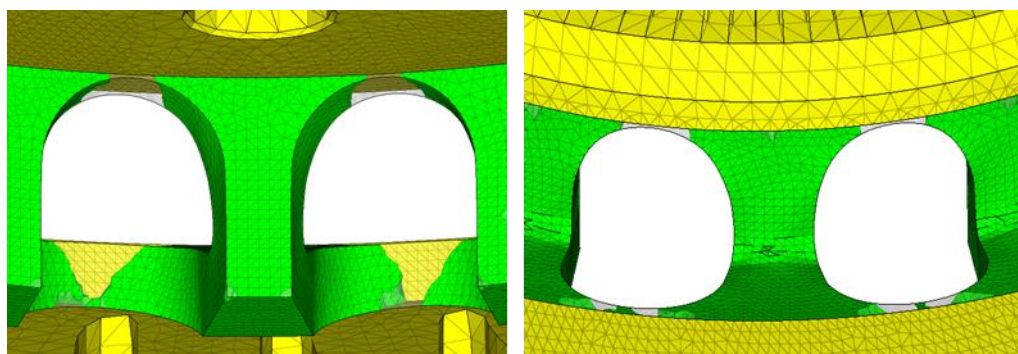
Figure 6-29 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.10



a)

b)

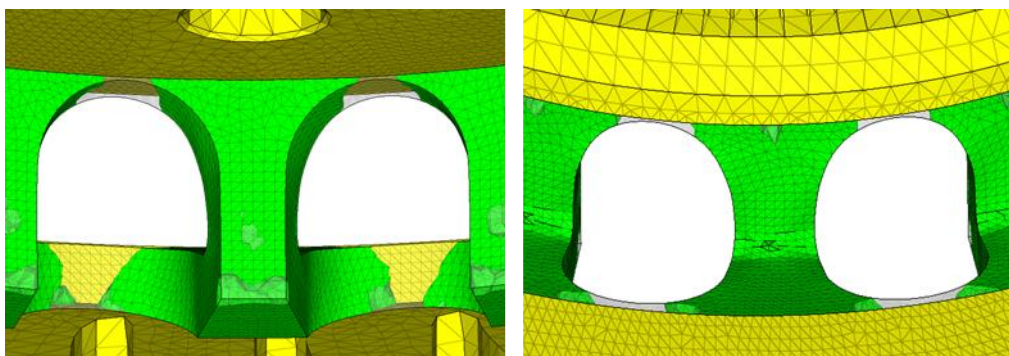
Figure 6-30 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.15



a)

b)

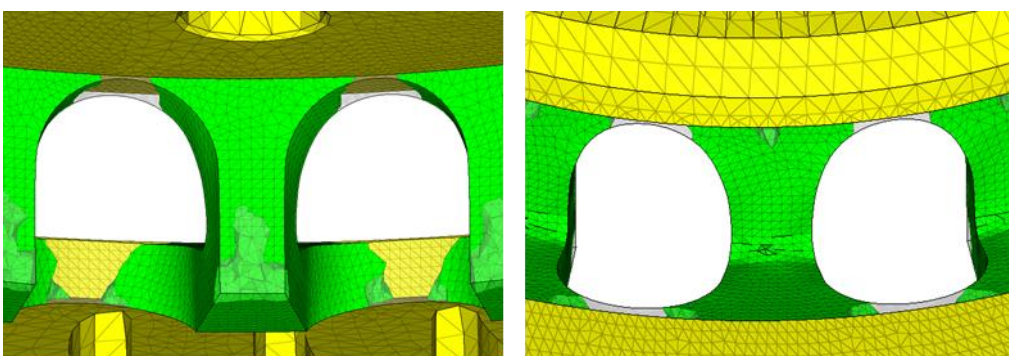
Figure 6-31 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.20



a)

b)

Figure 6-32 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.25



a)

b)

Figure 6-33 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.30

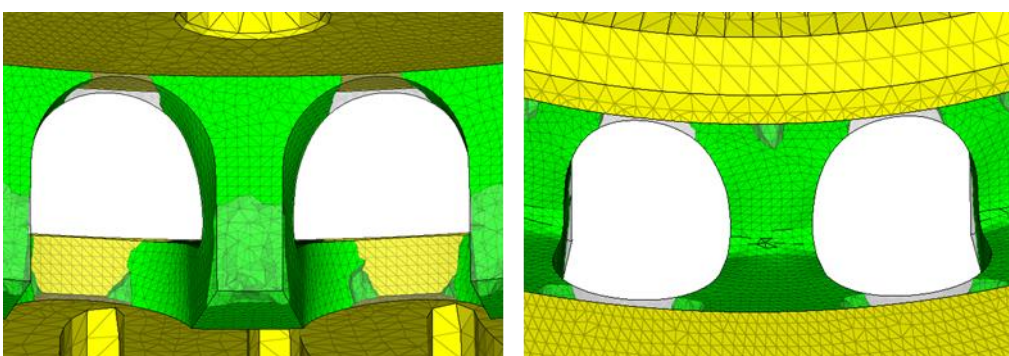


Figure 6-34 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.50

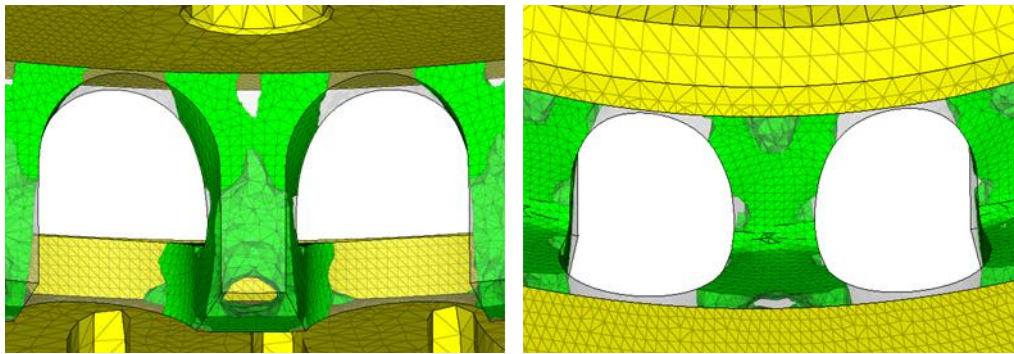


Figure 6-35 Topology Optimisation Effect on the Fingers, a) Inboard Side b) Inboard Side – Isodensity 0.90

Judging from the results of the figures above, it was considered reasonable to choose the shape when the isodensities were equal to 0.30 as the structural integrity is maintained and there are some weight savings.

With the Shape Optimisation, that preceded the Topology Optimisation, the total volume of the disc was increased since the fingers became slightly thicker. With the conduction of Topology Optimisation, and the decision to keep the results at 0.30 isodensity, the total disc volume including the Design and Non-Design spaces, was reduced by 1.1% compared to the initial volume of the Elise Version 3 prototype and by 1.5% compared to the volume of the disc after the Shape optimisation. The volume reduction percentages and the gains in terms of mass are illustrated in Table 6-2.

Table 6-2 Effect of Topology Optimisation

Disc Model	Volume [cm ³]	Volume Difference [%]	Mass Difference [g]
Elise V3	738.9	-	-
Elise V3 SO	741.8	0.4	22
Elise V3 TO	730.7	-1.1 (-1.5)*	60.5 (82.5)*
*the numbers in brackets are the comparison between the Shape Optimised volume and Topology Optimised volume.			

6.5 Discussion

The topic of structural optimisation for the case of the Elise V3 prototype disc can be divided into two areas: the Shape Optimisation and the Topology Optimisation. Each method was focused on a specific optimisation aspect. The Shape Optimisation was focused on minimising the Maximum Principal stresses and the Topology Optimisation on investigating possible further opportunities for weight reduction of the prototype disc.

The crucial achievement of Shape Optimisation was the reduction of the highest Maximum Principal stresses that were predicted in the mechanical analyses stage of the study, described in Chapter 3. It was realised that the disc will be fractured by this stress criterion, something that was verified by the fracture test explained in Section 5.4. The importance of the stress reduction by 32.0% is better understood by the increase in safety factor by almost one unit. By augmenting the safety factor from 2.55 to 3.57 the feasibility of the fingered disc concept increased. The small scale of the adjustments on the elements nodes indicates that the starting conceptual design of Elise Version 3 was developed in the right direction.

The Topology Optimisation further increased the mass reduction in the prototype disc. As with the minor modifications performed under Shape Optimisation, the somewhat limited volume reduction that was achieved with Topology Optimisation, highlighted the fact that the initial design of the prototype a very good starting point.

It is important to mention that Topology Optimisation was attempted on the Elise S2 disc with the solid hub. That was conducted in view of assessing whether the initial conceptual design of the Elise Version 3 was in agreement with a design solely created by the FE software or not. It was noticed that the analysis did not converge, meaning that a design direction needs to be given to the software, in the form of at least simple holes on the hub of the disc, in order to perform the optimisation process.

6.6 Chapter Summary

In this Chapter, the Elise Version 3 prototype disc was structurally optimised by using the techniques of Shape and Topology Optimisation. Slight alterations to the initial geometry of the disc were conducted during the Shape Optimisation that offered a stress reduction of 32%. Minor reduction of mass was achieved by Topology Optimisation proving that the initial design of the Elise Version 3 disc was reasonable in terms of its structural integrity.

7 DEVELOPMENT OF A BRAKE DISC COOLING FACILITY

7.1 Introduction

An opportunity arose with a project that ran in parallel with the study discussed in this thesis, for a Brake Disc Cooling Rig to be developed. It enables the determination of brake disc cooling characteristics in still air and in cross flow. Brake discs can be positioned parallel to the flow and also at different angles. The Brake Disc Cooling Rig comprises two main parts: the Flow Channel and the Disc Rotating and Heating Assembly. The first part provides the air cross flow that cools down the tested discs, whereas the second part provides support to the discs, heats them up to the desired temperature and rotates them as they would rotate on the axle of a vehicle. The specifications for each of the main parts are illustrated in Table 7-1 and Table 7-2.

Table 7-1 Flow Channel Specification

Inner diameter	1000 [mm]
Air speed range	2.5 – 13.5 [m/s]
Disc diameters	Up to 420 [mm] outer diameter

Table 7-2 Disc Rotating and Heating Assembly Specification

Maximum disc rotational speed	2643 [rpm]
Steering Knuckle maximum angles	-28° to 25°
Disc diameters	Up to 420 [mm] OD
Disc mass	Up to 25 [kg]
Base Plate travel adjustment	508 [mm]
Induction Heating System power	15 [kW]
Maximum heating temperature	450 [°C]

7.2 Flow Channel

7.2.1 Background and Conceptual Design

The major influential factors for the design of the Brake Cooling Rig were the requirements set for it and the constraints imposed during the conceptual design stage and the manufacturing stage. The requirements were:

- Good quality flow at the highest possible air velocity,
- Maximum outer diameter for the tested discs of 420 mm,
- The ability to have discs parallel to the air cross flow, but also at an angle relative to the flow.

It was deemed important to achieve stable air flow with minimum velocity fluctuations throughout the Flow Channel. In addition, as the vehicles become faster and heavier, the brake rotors become heavier and larger as well. As a result, it was considered beneficial to allow brake discs with outer diameters of up to 420 mm to be tested in the rig. The final requirement was about the discs being able to be steered when located inside the Flow Channel so that they can be positioned parallel to the flow, but also at an angle relative to it. The reason for this was that when the disc is mounted on the vehicle's axle it is never exposed to the free flow of air due to the wheel and several other components of the vehicle, like the dust shields and other suspension and body parts, surrounding it. The air always "hits" the disc at an angle and it was seen as useful to replicate this situation.

The constraints faced during the development of the rig were:

- The existing Fan-Motor specifications,
- The building dimensions,
- The integrity of the building.

The Brake Cooling Rig discussed here is based on an air extraction structure for a Powertrain Dynamometer (supplied by CP Engineering in 2004) in Building 43 of Cranfield University. The original extraction motor (11 kW) was removed and flow was provided by the most powerful blow fan and motor available rated at 21.3 kW. Hence, the first limiting factor was the power of the motor, and in combination with the existing fan, the limiting factor was their air volume flow performance, which according to the manufacturer's output curves was 10.5 m³/s

(Figure 7-1). The entire output curves specification sheet for the fan-motor can be found in Appendix G.

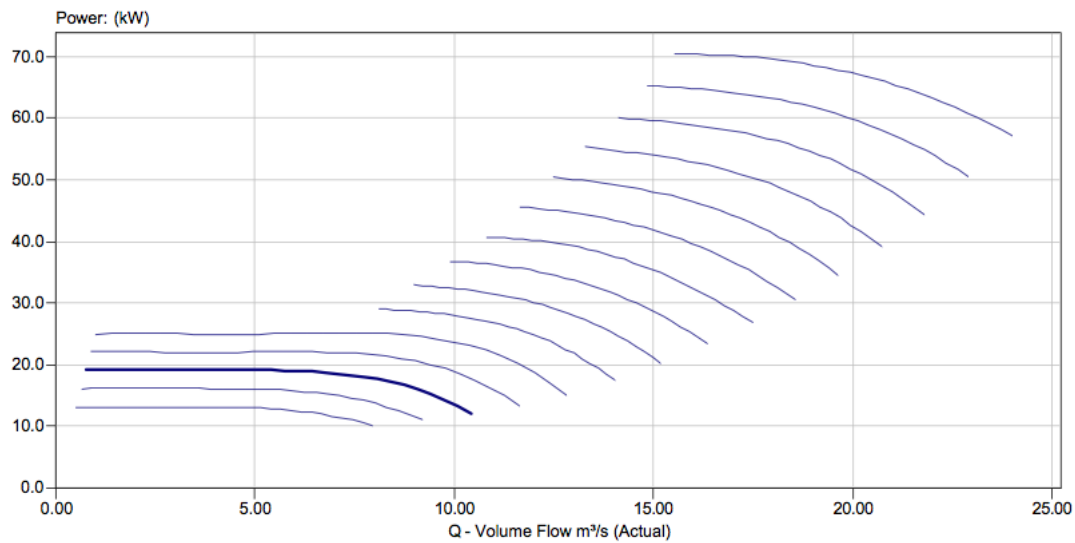


Figure 7-1 Fan Output Curves

The volume flow performance of the available fan in conjunction with the cross sectional area of the Flow Channel results in the theoretical air velocity that can be achieved. However, before deciding on the size of the Flow Channel's cross sectional area, its shape had to be decided first. If the tested discs were always positioned parallel to the cross flow, the Flow Channel could have a rectangular shape. Nevertheless, because the discs were planned to be positioned at different angles with regards to the flow, and not only parallel, the Flow Channel had to be circular in order to avoid the air turbulence.

Since the shape of the Flow Channel's cross sectional area was decided, its size had to be selected as it was a determining factor on the air velocity. The formula to calculate the air velocity is given by equation (7-1).

$$V_{air} = \frac{Q_{air}}{A_{ch}} \quad (7-1) \text{ Air Velocity}$$

The initial size considered was for the Flow Channel to have an inner diameter of 800 mm based on the fact that the inner diameter of the fan was 800 mm as well. This would result in an air velocity of 20.88 m/s.

$$V_{air} = \frac{Q_{air}}{A_{ch}} = \frac{10.5}{0.503} = 20.88 \text{ m/s} \quad \text{(7-2) Air Velocity for Flow Channel ID=0.8 m}$$

By having the Flow Channel with an inner diameter of 800 mm, the blockage caused by a brake disc with outer diameter of 420 mm will be 11.7% without taking into consideration any other components, such as the shaft and the upright, that would keep the disc into position and rotate it. In addition, a gap of 190 mm between the disc and the Channel's inner walls would exist on the top and bottom of the disc.

The second possible value for the inner diameter of the Flow Channel was 1 m. This would result in an air velocity of 13.37 m/s.

$$V_{air} = \frac{Q_{air}}{A_{ch}} = \frac{10.5}{0.785} = 13.37 \text{ m/s} \quad \text{(7-3) Air Velocity for Flow Channel ID=1 m}$$

With the inner diameter of 1 m, the blockage of a 420 mm disc, alone, would be 7.5%, whereas the gap between the disc and the Channel on the top and bottom of the disc would be 290 mm.

In view of the air flow quality being the prime requirement in the design phase, it was decided for the inner diameter of the Flow Channel to be 1m. Moreover, the theoretical achievable air velocity of 13.37 m/s was seen as adequate for the scope of the Rig.

The two remaining constraints, namely “the building dimensions” and “the fabric of the building” had an affiliation to each other. Although the room where the Brake Cooling Rig is positioned is used by the Centre of Automotive Engineering, the actual owner is the Estates department of Cranfield University. As a result, it was not allowed to change the fabric of the building in any way without prior permission from the Estates department and detailed blueprints of the changes to be made.

Having said that, the Rig had to fit inside building 43 which was 9.25 m long. The front edge of the fan-motor's casing had a distance of 2.3 m from the rear wall

partly due to its design and partly because it was attached to the Exhaust Outlet (Figure 7-2).

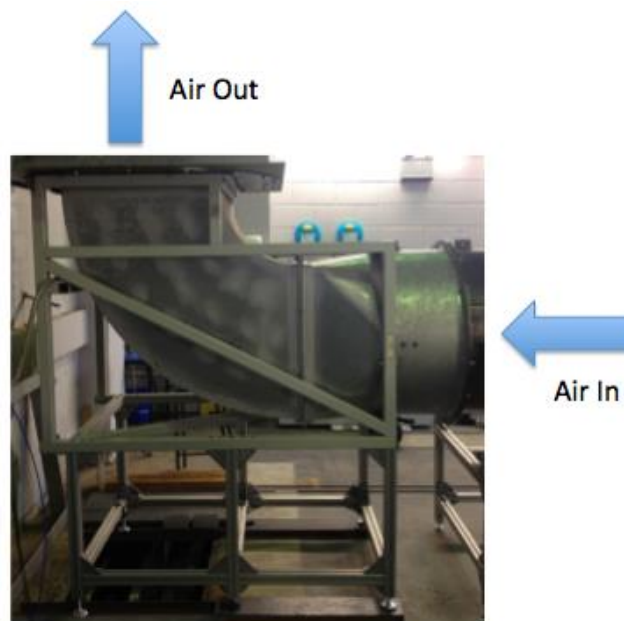


Figure 7-2 Fan Unit – Air Flow (R/H View)

In addition, at least 1 m of space had to be left for gangway in order to comply with the Health and Safety regulation with regards to the fire escape path width of the building. Therefore, the actual available length for the Rig was 5.95 m and in order to compensate for manufacturing tolerances, it was decided for the overall length of the Flow Channel to be 5.75 m. A CAD representation of the Flow Channel is presented in Figure 7-3 and a picture of the Channel is given in Figure 7-4. The numbers in Figure 7-4 are the numbers of the different sections of the Channel that will be discussed in the following Section.

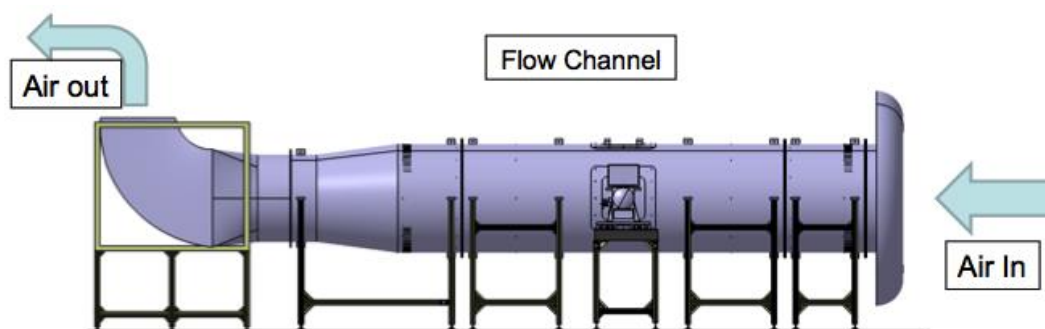


Figure 7-3 CAD Model of the Flow Channel (R/H View)



Figure 7-4 The Flow Channel

7.2.2 Flow Channel - Segments

The Flow Channel consists of seven major parts. Starting from the intake, these parts are:

1. Intake Bellmouth,
2. Segment 3,
3. Segment 2 – also called Working Segment,
4. Segment 1,
5. Fan Unit,
6. Segment 0,
7. Exhaust Outlet.

The major dimensions of the Channel are shown in Figure 7-5 and Table 7-3.

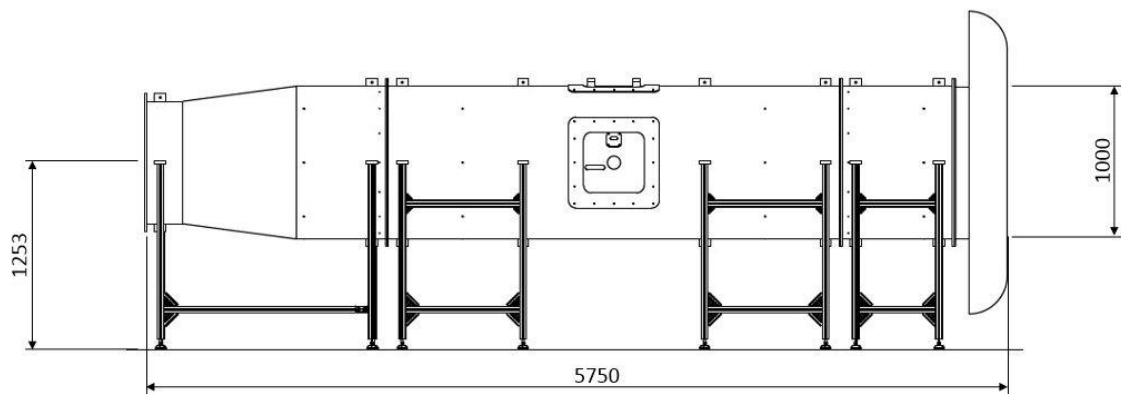


Figure 7-5 Channel Main Dimensions

Table 7-3 Channel Characteristics

Overall length	5750 [mm]
Distance of the Channel's centre line to the floor (nominal)	1253 [mm]
Inner diameter of the Channel	1000 [mm]

The main body of the Channel (Segments 1 to 3) is made of steel EN 10025-2 S275JR. The Bellmouth is made of aluminium 1050A. The whole Channel is supported by aluminium frames made by 45 x 45 mm Bosch Rexroth beams with the exception of the frame for the Fan Unit, where 50 x 50 mm beams were used to provide higher stiffness due to the weight of the Unit. The beams were joined together to form the supporting frames with aluminium Bosch Rexroth angle brackets. Rubber gaskets were placed among segment flanges in order to prevent any air leaks due to asperities of the different parts and to dampen any vibrations during the Flow Channel operation.

The Intake Bellmouth is one of the most important parts for the operation of the Flow Channel. A Bellmouth with a good design results in good straight flow, increased air mass rate and higher air velocity in the Channel.

Several potential designs were studied, but the design of the simple radius was chosen, as it was both effective and relatively easy to manufacture. It was decided for the Bellmouth to have a straight section of 100 mm and then a 250

mm radius. The manufacturing drawing is included in Appendix H.1. For manufacturing reasons the design was slightly altered. The 100 mm straight section remained, but the curved section was made of sheets of aluminium 2 mm thick and 100 mm long. Five strips of aluminium sheets comprised the curved section which was not 180° as originally planned, but slightly more than 90° (Figure 7-6).

As mentioned previously, the aluminium alloy used was 1050 A, a grade with high content of aluminium (around 98-99 %). This grade is ideal for forming a complex shape and also suitable for welding.

Figure 7-6 shows the actual Bellmouth assembled onto the Flow Channel. It is fastened to Segment 3 (Figure 7-6 – a).

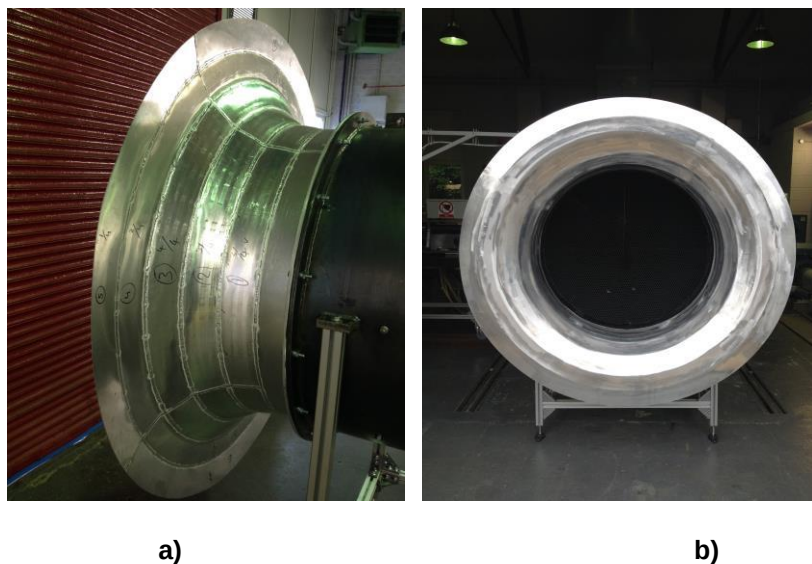


Figure 7-6 A Intake Bellbouth a)Side View, b)Front View

Despite some reduction in the “swept radius”, the arc at the Bellmouth is very effective in providing a smooth air intake attached to the wall. Figure 7-7 shows the experiment conducted using the Aerotech SGS-90 smoke generator. It can be seen that the flow remains attached to the wall. The experiment was conducted for a range of air speeds in the Flow Channel (2.5 – 13.5 m/s) and the flow was always stable and uniform.



Figure 7-7 Intake Air Flow at 7.5 m/s

Segment 3 is located between the Intake Bellmouth and Segment 2. Apart from providing extra length to the Flow Channel, Segment 3 houses two important parts, the first being a flow straightener (seen in the cut view of Figure 7-8) and the second, a thermometer/hydrometer. The Manufacturing drawing of Segment 2 is given in Appendix H.2. The flow straightener is a honeycomb structure made of 3003 aluminium alloy, being 75 mm thick. Each cell is approximately 12.7 mm of size and the foil thickness is 0.076 mm (Figure 7-9).

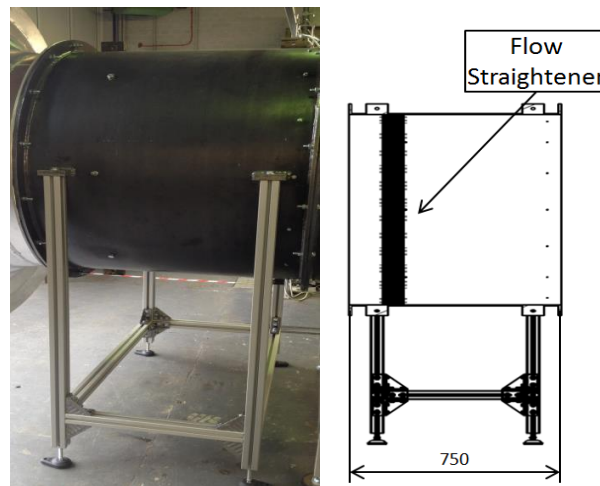


Figure 7-8 Segment 3 [L/H View]



Figure 7-9 Detail of the Flow Straightener Honeycomb

The thermometer/hydrometer is a Tempatron DP series capable of measuring temperature ranging from -20° C to 70° C and humidity from 0% to 100%. Its accuracy at ambient temperature is $\pm 0.2\%$ for the temperature sensor and $\pm 3\%$ for the humidity sensor.



Figure 7-10 The Tempatron DP Series Installed in Segment 3

Segment 2, also named Working Segment (Figure 7-11), is the largest (3000 mm) and most important in the operation of the Brake Disc Cooling Rig, as it houses the disc tested and the Disc Rotating Assembly enters in the Channel from this segment. It was decided to be that long in order to provide as much straight flow

as possible, ensuring repeatable experiments and producing reliable results. A detailed drawing of Segment 2 is presented in Appendix H.3.



Figure 7-11 Segment 2 (L/H View)

The main features of this Segment are the three access hatches, two on each side and one on the top. From the right hand side [R/H] hatch the Disc Rotating Assembly enters the Channel, whereas from the opposite side visual checks can be conducted to the disc/thermocouples assembly (Figure 7-12). The Left hand [L/H] side hatch is useful for having access into the Disc Rotating Assembly when it is placed within the Flow Channel and performs operations, such as adjusting the thermocouples and checking the heating coil. The top hatch is mostly useful for aligning the disc in its position in the middle.

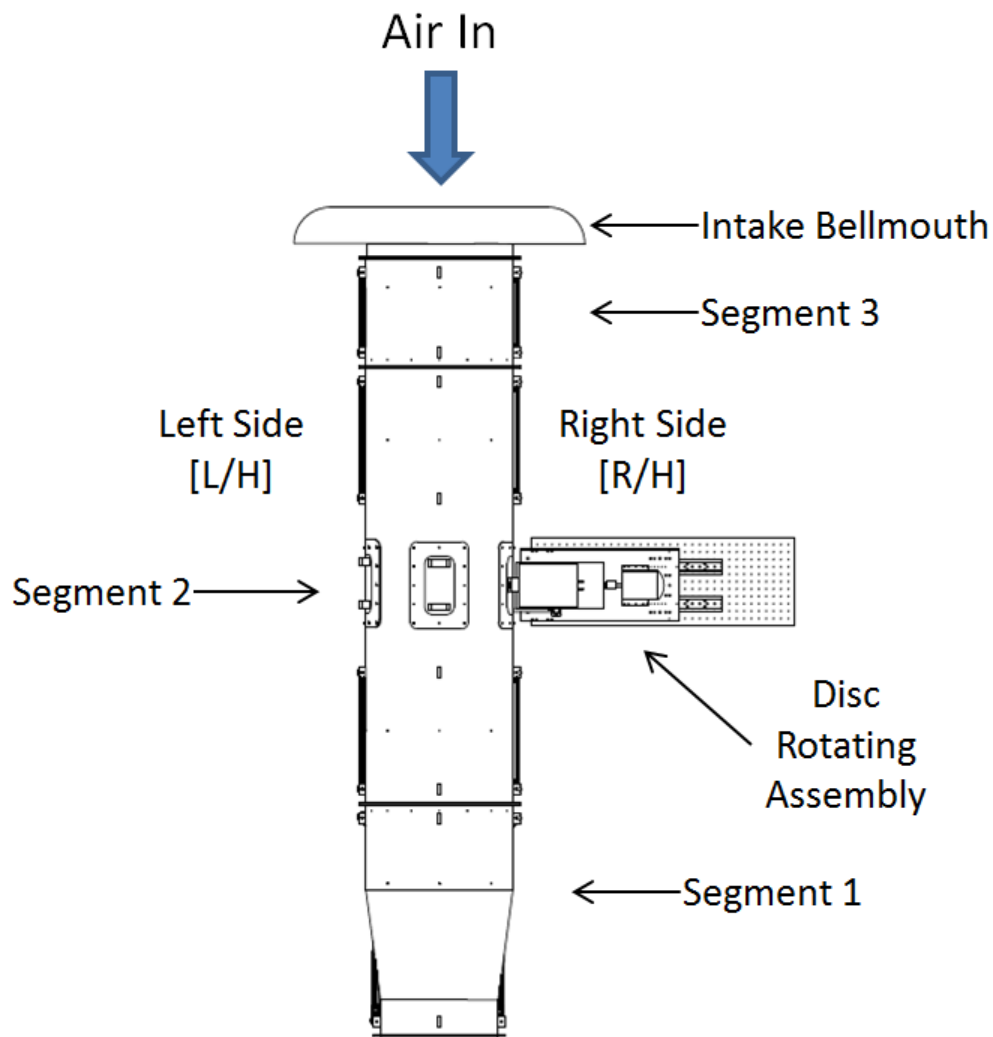


Figure 7-12 Top View of the Flow Channel/Disc Rotating Assembly

The right hand and the left hand side openings have the same dimensions, 500 x 500 mm, whereas the top hatch has a 500 x 300 mm opening (Appendix H.4.1 and H.4.2). The right hand side hatch door was modified in order for the Disc Rotating Assembly's axle to go through it (Figure 7-13). For that to happen the door was cut in two pieces and a number of holes were drilled. The drawing of the modified hatch can be found in Appendix H.4.3.

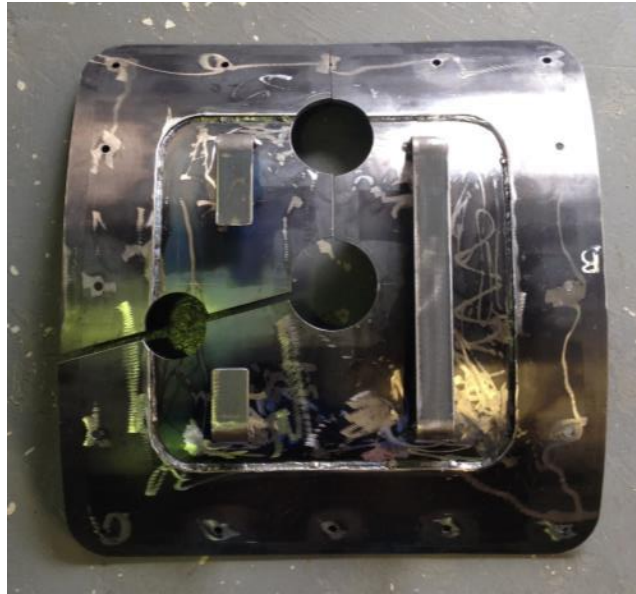


Figure 7-13 Modified Right Side Hatch

In addition to the three hatches, a number of threaded holes were created in Segment 2 in order to securely position pitot tubes.

Segment 1 connects Segment 2 with the Fan Unit. In this segment the inner diameter of the Flow Channel is reduced from 1000 mm to 800 mm due to the fixed dimensions of the Fan Unit (its propeller size and case). To make this happen, it was decided for Segment 1 to be as long as possible given the constraints of Building 43's space. As a result, Segment 1 is the second longest segment of the Brake Disc Cooling Rig with 1600 mm length. It consists of a part (the front part) which has an inner diameter of 1000 mm and length of 600 mm. Then, there is a part (middle part) of 750 mm length, where the inner diameter gradually decreases from 1000 mm to 800 mm. Finally, the last part (rear part) has an inner diameter of 800 mm and length of 250 mm (Figure 7-14). The detailed drawing of Segment 1 can be found in Appendix H.5.

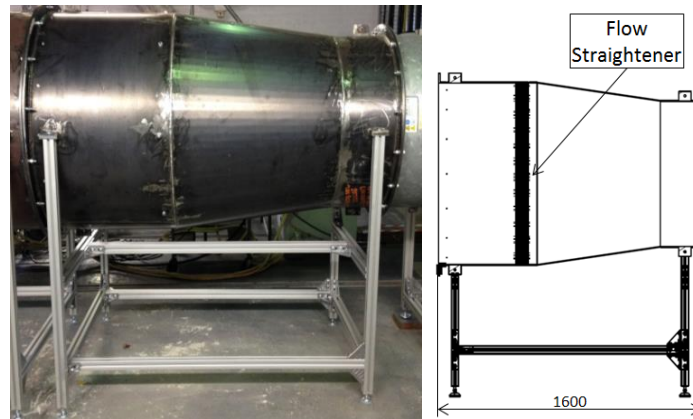


Figure 7-14 Segment 1 (L/H View)

Furthermore, Segment 1 houses the second flow straightener. The latter is identical to the flow straightener installed in Segment 3 with exactly the same specifications. It is positioned 474 mm from the beginning of the Segment (seen in the cut view in Figure 7-14).

The Fan Unit assembly consists of the electric motor, the propeller (fan) and the structure housing the latter two. The motor characteristics are presented in Table 7-4, the fan characteristics in Table 7-5, and finally the inverter characteristics in Table 7-6.

Table 7-4 Motor Characteristics

Manufacturer	Teco Westinghouse
2 pole electric motor	
Power	21.3 [kW]
Current	39.0/35.7 [A]
Voltage	380/420 [V]
Maximum frequency	50 [Hz]
Maximum Speed	2935 [rpm]

Table 7-5 Propeller Characteristics

Manufacturer	Fläkt Woods
Number of blades	9
Angle	12°

Table 7-6 Inverter Characteristics

Manufacturer	ABB
Model	ACS310-03E-48A4-4
Frequency	48/63 [Hz]
Voltage	380/420 [V]
Number of phases	3
Nominal current	48.4 [A]
Nominal power	22.0 [kW]

The way the Fan Unit forces the air to change direction by 90° upwards leading it to the Exhaust Outlet is shown in Figure 7-2.

Segment 0 is an adaptor plate to connect the Fan Unit to the Exhaust Outlet structure of Building 43. The connection is sealed with the use of two 5 mm thick rubber gaskets. The first is placed between the Fan Unit and Segment 0 and the second between the Segment 0 and the Exhaust Outlet. The detailed drawing is shown in Appendix H.6.

The final part of the Flow Channel is the Exhaust Outlet from the original Powertrain Dynamometer. This channel can direct the airflow either outside of the building or recirculate it inside the building by the appropriate flaps/grills (Figure 7-15 b).

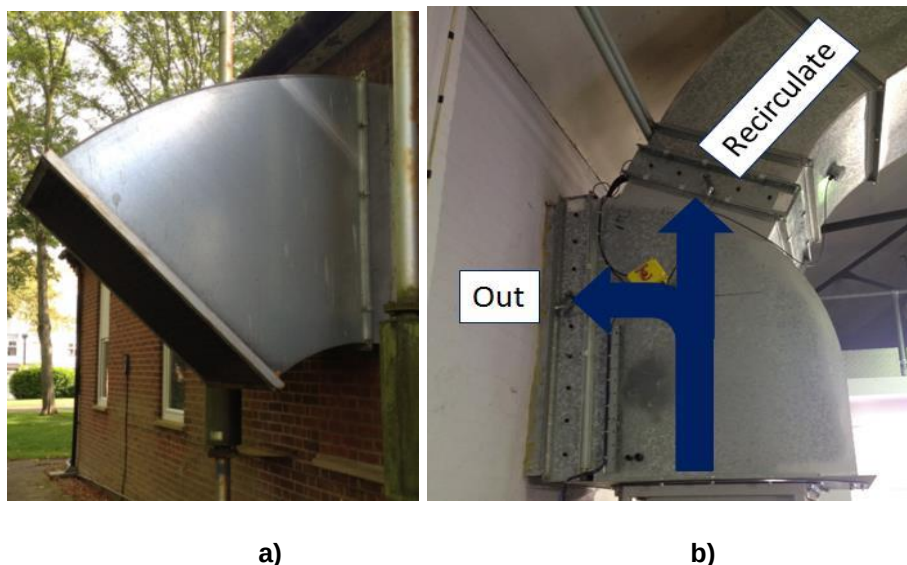


Figure 7-15 Exhaust Outlet a) Outside Section, b) Inside Section (R/H View)

After conducting air velocity trials it was evident that the air speed was considerably lower when the flaps in the Exhaust Outlet were recirculating the air, compared to the velocities achieved when the flaps were set to direct the air outside the building. This was due to the high air blockage inside the Exhaust Outlet in the area where the direction vents are. As a result, it was decided the flaps to direct the air 100% outside the building.

7.3 Disc Rotating and Heating Assembly

7.3.1 Background and Conceptual Design

Similarly to the Flow Channel, certain requirements were set during the conceptual stage for the Disc Rotating and Heating Assembly. The Assembly had to:

- Be able to position and support discs in the Flow Channel,
- Be able to spin discs up to 2500 rpm,
- Heat discs up to 450°C.

The Disc Rotating and Heating Assembly was designed and manufactured at Cranfield University specifically for the Brake Disc Cooling Rig. The original thought was to use an old rotating assembly, but it was considered inadequate in performing the duties required for the new Brake Cooling Test Rig. The new design was more demanding in functionality to carry the brake discs for cooling tests inside the Flow Channel, heat them up and spin them to cool them down. It uses an induction heating system to warm up the brake discs. The system comprises a Control Unit, a Cooler Unit, a Workhead and a Heating Coil with the Workhead and the Heating Coil designed and manufactured specifically for the Brake Disc Cooling Rig. The Control Unit and the Cooler pre-existed from previous projects. The discs are spun by a 3-phase AC electric motor driven by an inverter drive, both solely purchased for the new Disc Rotating and Heating Assembly. Figure 7-16 shows a CAD representation of the Assembly; the actual Assembly is presented along with its main components in Figure 7-17 (parts of the Induction Heating are not shown).

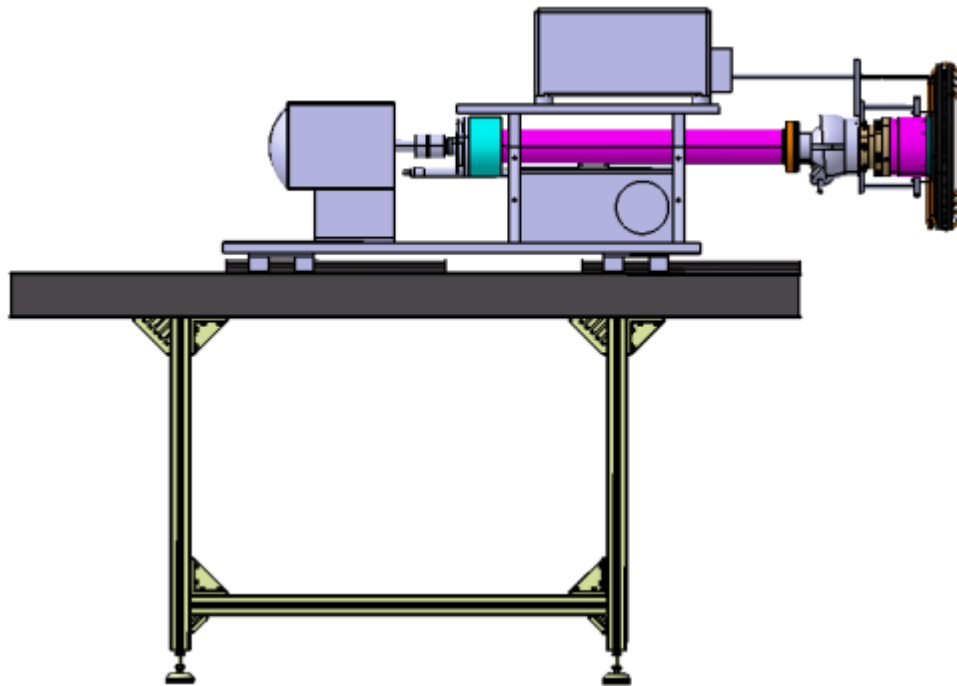


Figure 7-16 The Disc Rotating Assembly CAD Model

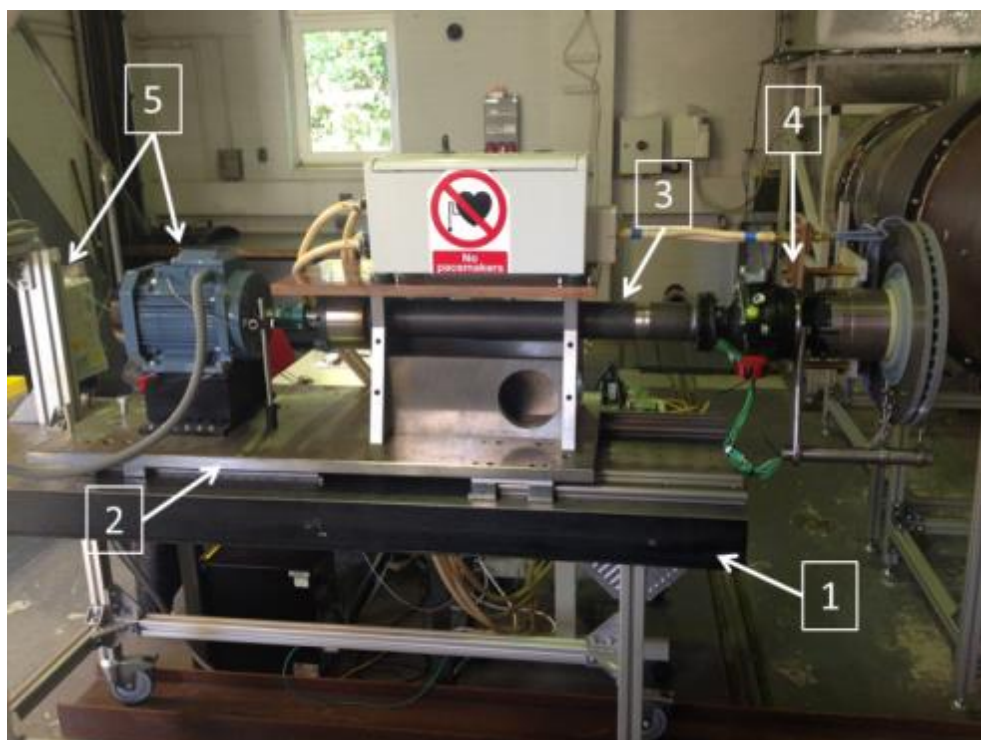


Figure 7-17 The Disc Rotating and Heating Assembly

7.3.2 The Assembly's Main Components

The Disc Rotating and Heating Assembly (Figure 7-16 and Figure 7-17) consists of:

1. Table,
2. Base Plate,
3. Axle Assembly,
4. Steering Knuckle,
5. Electric Motor and Inverter,
6. Sensors and Data Logging Equipment (not shown in Figure 7-17).

The Table (Position 1 in Figure 7-17) that was used to accommodate the Disc Rotating and Heating Assembly is an optical table 1813 x 594 mm, of an exceptionally flat and rigid design. Another very useful feature was that it incorporated vibration-dampening features. It acted as a low-pass filter preventing the high frequency vibrations. On its stainless steel top side it incorporates a grid of M6 treaded holes (2" x 2") that were used for placing the Base Plate's sliding mechanism which will be presented in the next Section.

To support the table, a frame was manufactured out of 45 x 45 mm Bosch Rexroth aluminium beams. The beams were joined together with aluminium Bosch Rexroth angle brackets. At the end of the frame legs, caster wheels were installed, allowing the table to be manoeuvrable (Figure 7-18). In addition, the wheels, made out of hard rubber, absorbed vibrations created during the Assembly's operation. Finally, the wheels were equipped with lock-up brakes that could hold the table stationary, preventing both wheel rotation and swivel.



Figure 7-18 Disc Rotating and Heating Assembly Table

The Base Plate (Position 2 in Figure 7-17) is a 1095 x 496 x 25.4 mm single steel piece acting as a support and location plate for all the parts and components of the Disc Rotating and Heating Assembly (Figure 7-19). Considerable effort was required in order to achieve the optimum packaging for the Assembly's parts on the top side of the Plate and also accommodate the Sliding Mechanism of the Assembly on the bottom side of the Plate. A number of contradicting requirements, such as the constraints of the optical table's threaded holes positions and fixed dimensions of the Axle's Assembly (discussed further down) were all successfully addressed within the final design of the Base Plate. The manufacturing drawing of the Base Plate can be found in Appendix I.1.

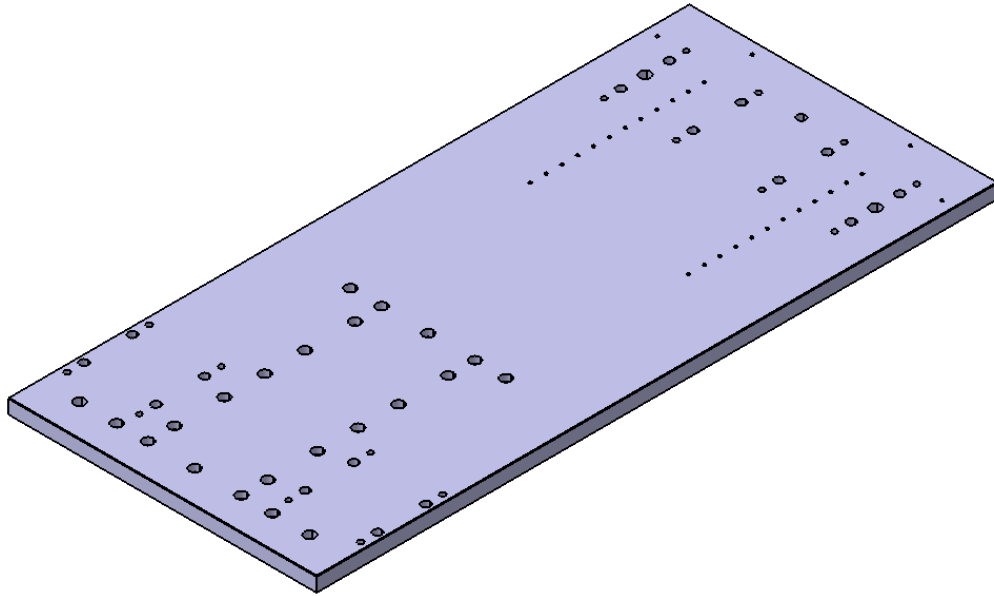


Figure 7-19 Base Plate CAD Model

The Sliding Mechanism (Figure 7-20) allows the Assembly to travel longitudinally a total distance of 508 mm and consists of platforms attached to the bottom side of the Base Plate and rails attached to the laser table. It is necessary to provide movement in order to ensure the brake discs are in the mid-plane of the Flow Channel. Although the longitudinal movement can be achieved by moving the entire table, the Sliding mechanism is required for fine-tuning and final positioning of the Assembly.

It consists of four sliding platforms and four sliding aluminium rails; all products are of the DryLin W range of Igus. The platforms were bolted on the Base Plate and the sliding rails were bolted on the Table. The platforms have special cylindrical slots on them for the rails to be attached to. In these slots, plastic inserts are installed in order to have aluminium to plastic contact resulting in a smooth and controllable forward and backward translation of the Disc Rotating and Heating Assembly.

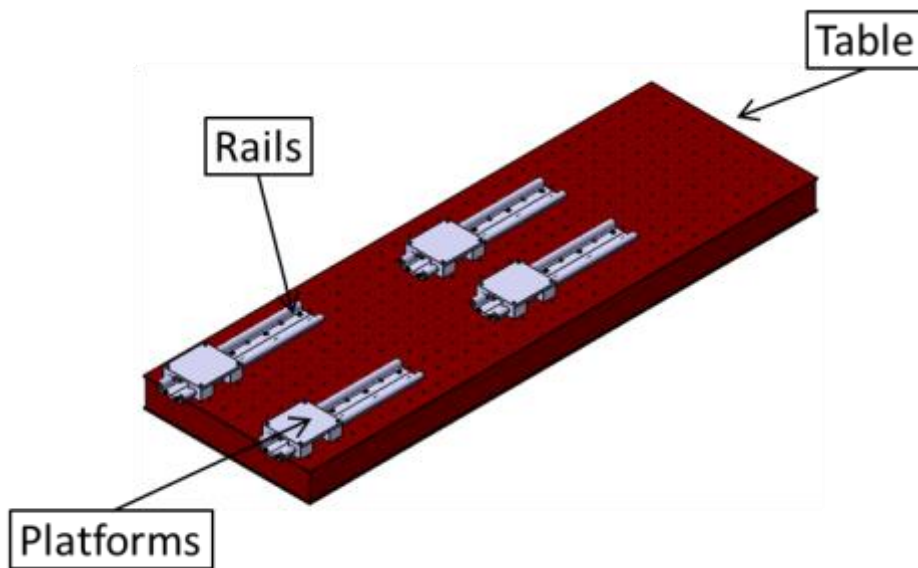


Figure 7-20 Base Plate Sliding Mechanism

The Axle Assembly (Position 3 - Figure 7-17) and the Steering Knuckle Assembly are the two major subassemblies of the Disc Rotating and Heating Assembly. The latter will be presented in the next Section, whereas in this Section the Axle Assembly is discussed.

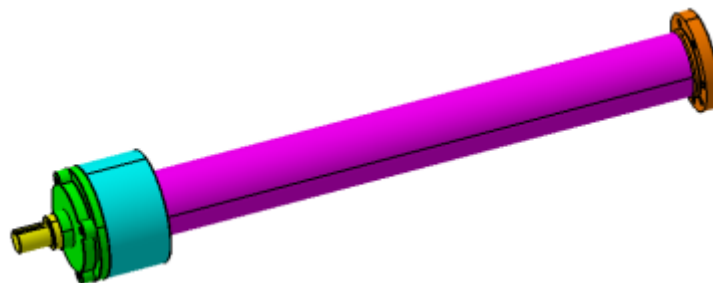


Figure 7-21 Axle Assembly CAD Model

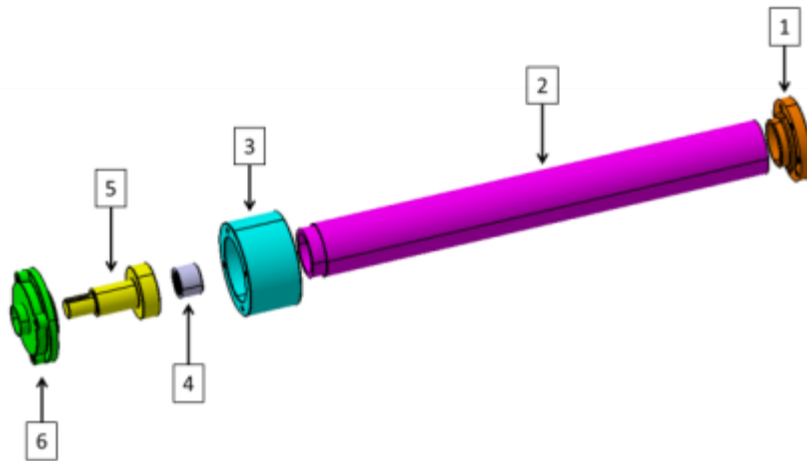


Figure 7-22 Axle Assembly Exploded View

The CAD model of the Axle Assembly is shown in Figure 7-21, with Figure 7-22 presenting an exploded view. Starting from the right end of Figure 7-22 the parts are:

1. Wheel Flange,
2. Tube,
3. Bearing Flange,
4. LH Defender Spline (drive shaft not shown),
5. Coupling Adaptor,
6. Bearing Unit.

The Wheel Flange (Position 1 in Figure 7-22) was cut from the hollow Tube of the Land Rover Defender axle. It was then cleaned and welded on the hollow Tube (Position 2 in Figure 7-22) of the Rotating Assembly. It is used for bolting the Axle and the Steering Knuckle Assemblies together. When on the Defender's axle, the Wheel Flange was positioned in a way to give 18.1° caster angle. Having a caster angle in the Disc Rotating and Heating Assembly was not desired, because the Steering Knuckle carrying the tested disc into the Flow Channel would tend to swing. Consequently, it was welded on the Tube ensuring 0° of caster angle.

At the other end of the Tube, the Bearing Flange (Position 3 - Figure 7-22) was welded. Its purpose was to be the mounting link of the Bearing Unit and the rest of the Assembly. Detailed drawings of the Bearing Flange can be found in

Appendix I.2 and of the Bearing Unit in Appendix I.3. Inside the Bearing Flange, the Coupling is positioned and inside the Coupling, a Spline from the Land Rover Defender has been adhesively bonded using Loctite. The Coupling and the Spline (Position 4 in Figure 7-22) are held in position with the Bearing being bolted to the Bearing Flange. The detailed drawing of the Coupling can be found in Appendix I.4 and the assembly drawing of the entire Axle Assembly is presented in Appendix I.5.

The Axle Assembly is suitably interconnected and fastened to the Base Plate by two Support and two Stiffening steel Plates (Figure 7-23). The manufacturing drawings of these Plates can be found in Appendix I.6 to Appendix I.9.



Figure 7-23 Support and Stiffening Plates

7.3.3 Steering Knuckle Assembly

The Steering Knuckle Assembly (Position 4 in Figure 7-17) accommodates the discs intended for testing in the Brake Disc Cooling Rig. Because the entire Knuckle Assembly is exposed to the cross flow, particular attention was paid in order to minimise blockage and disturbance of the flow. Furthermore, the disc had to be placed as much as possible in the free air flow. A number of design ideas were studied in order to develop the Steering Knuckle Assembly including creating custom assemblies with the use of a simple axle or with a double

wishbone configuration with wishbones of equal length in order to eliminate the camber angle. The final design solution is presented in Figure 7-24.

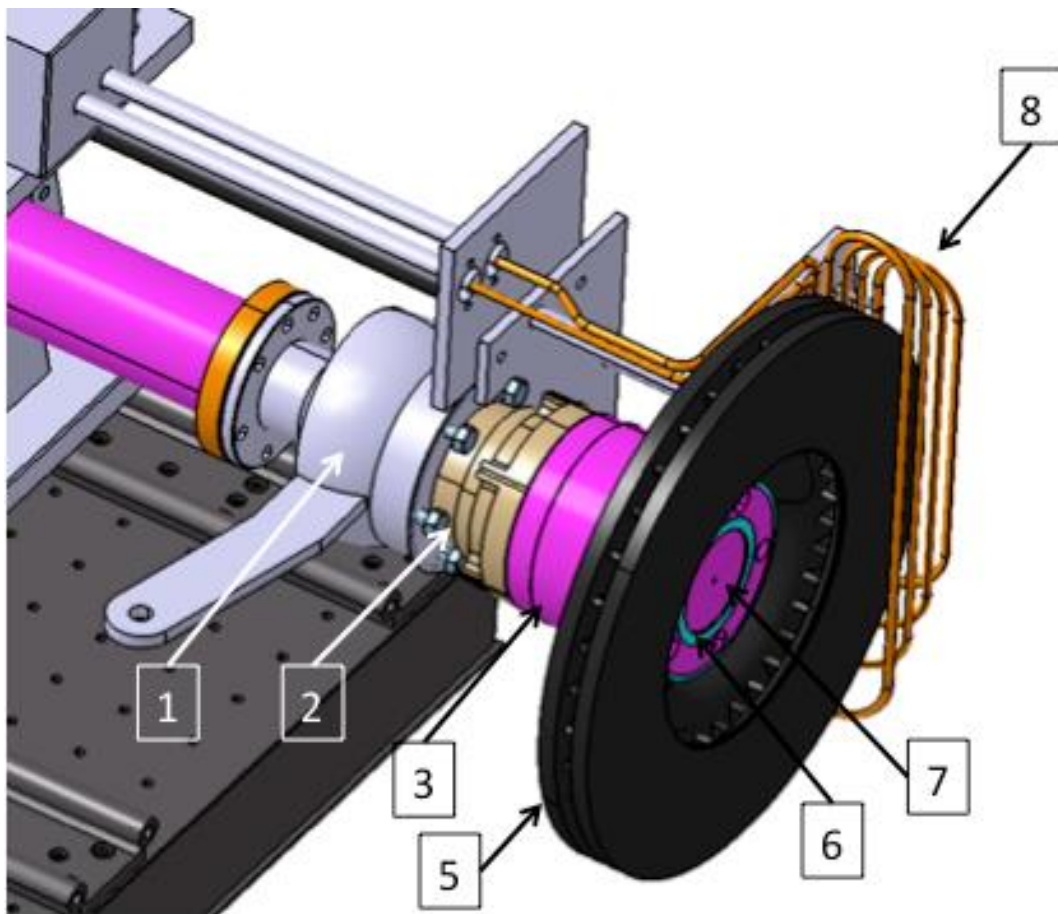


Figure 7-24 Steering Knuckle Assembly CAD Model

The Steering Knuckle Assembly consists of the following parts (Figure 7-24 and Figure 7-25):

1. Defender Knuckle (Near Side),
2. Modified Defender Hub,
3. Steel Flange,
4. Inner Plastic Flange,
5. Brake Disc,
6. Outer Plastic Flange,
7. Outer Steel Ring,
8. Induction Heater Coil.

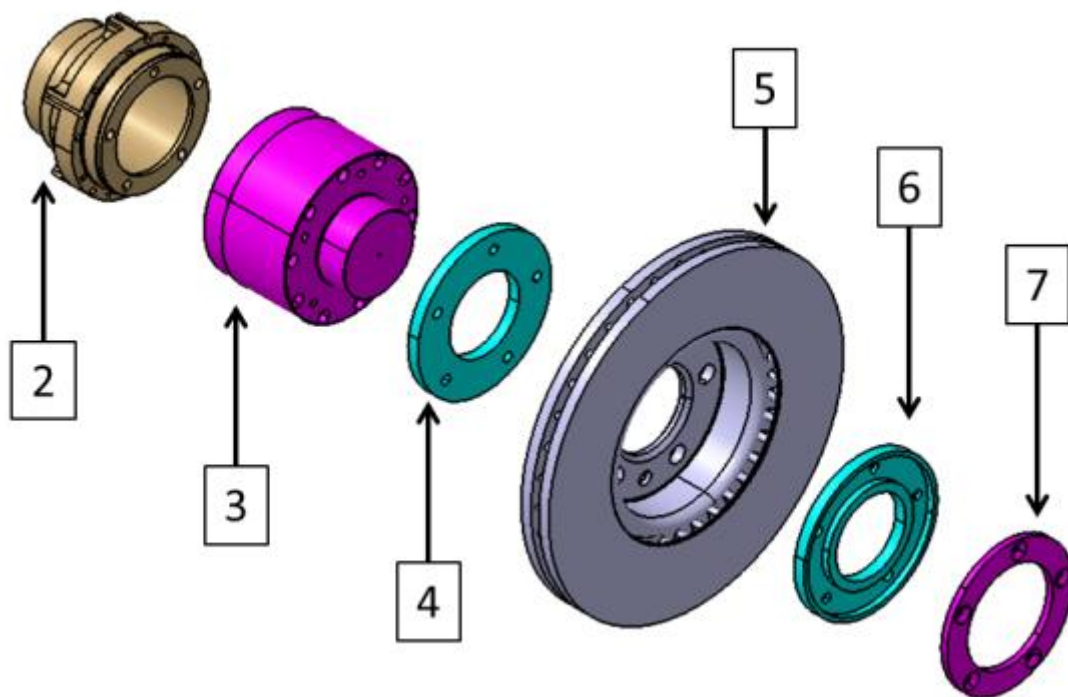


Figure 7-25 Steering Knuckle Assembly Exploded View

One of the requirements from the beginning of the project was that the brake discs tested were positioned within the Channel parallel to the cross flow, but to also allow change of angle relative to the flow. In the end, it was decided to base the Steering Knuckle Assembly on the front axle of the Land Rover Defender and modify its different parts depending on the needs of the Rig.

The outcome was for the Steering Knuckle to be able to travel from lock to lock a total distance of 150 mm (Figure 7-26), which corresponds to a $+25^\circ$ and -28° angles (Figure 7-27).

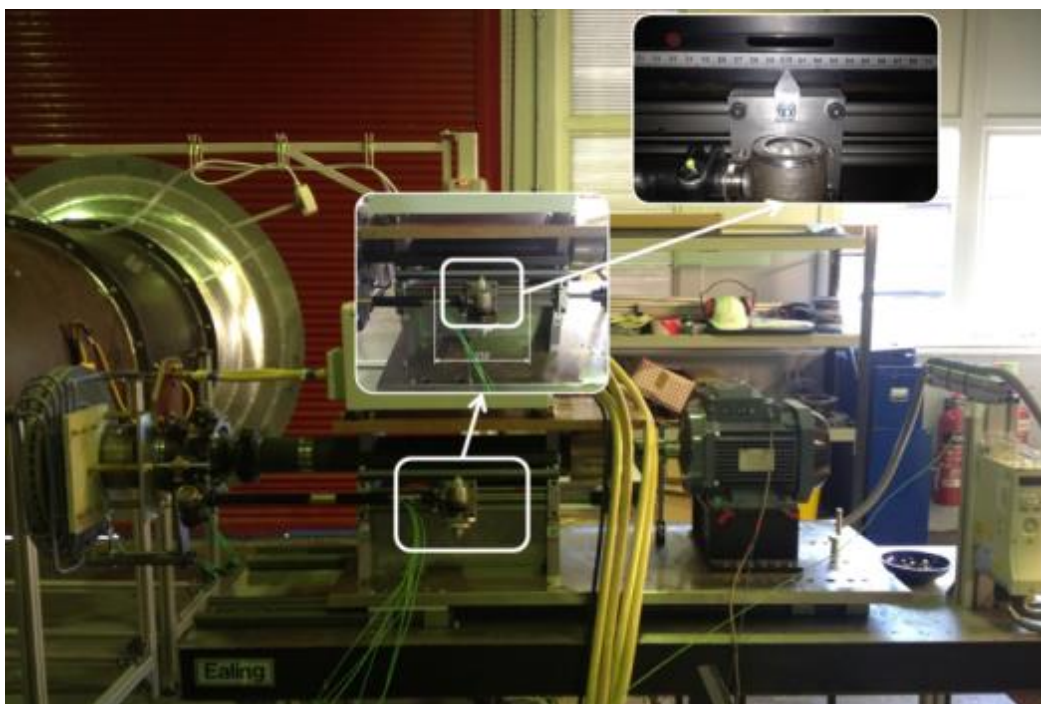


Figure 7-26 Track Rod Displacement Measurement

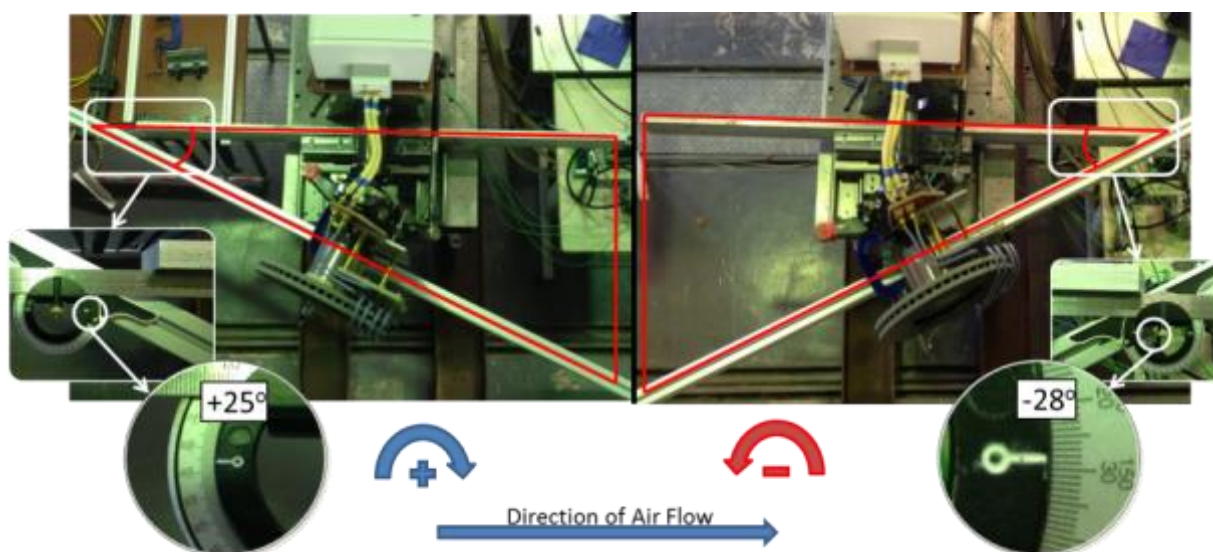


Figure 7-27 Steering Knuckle End to End Angles

The kingpin inclination was not possible to be eliminated (as mentioned the caster and camber angles were adjusted to 0°). The kingpin inclination resulted in the disc not been vertical when steered, which produced larger angles. Also, it caused the disc to “lift” when in extreme negative angle and “dive” in extreme positive angle. However, these deviations are small and could be measured and

suitably modelled for any “back to back” comparisons. In Figure 7-28 the relation between the disc angle and the track rod displacement measurement is shown, which is linear for small angles ($<5^\circ$) and then becomes non-linear.

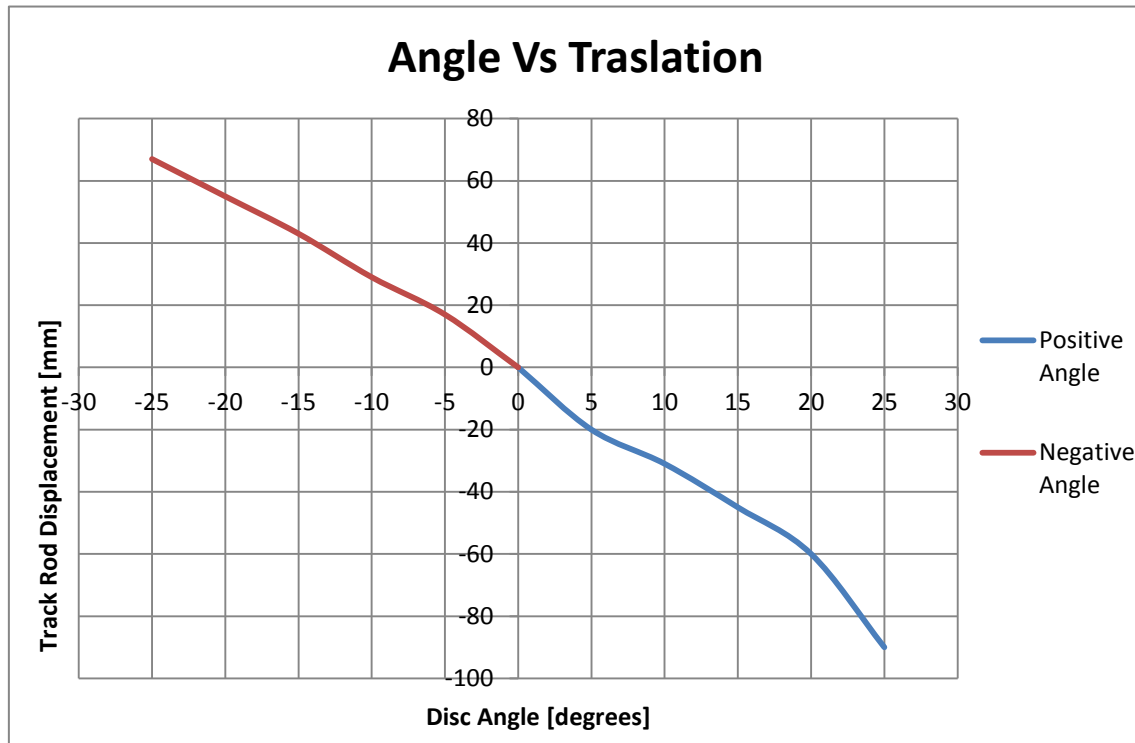


Figure 7-28 Disc Angle VS Track Rod Displacement Graph

The near side Defender knuckle (Figure 7-24) was used, as its driveshaft was longer. It was bolted to the Wheel Flange of the Axle Assembly and it accommodated the driveshaft that transfers the rotation from the Electric Motor to the brake disc. A modified version of the Defender's Wheel Hub was bolted on the Knuckle without the original brake disc of the vehicle. The modifications on the Wheel Hub were aimed to smoothen all the sharp edges and remove all the unnecessary material. Figure 7-29 shows a comparison between the original Wheel End and the Modified one. The drive parts required only minor modifications and are not shown. The detailed drawing of the Modified Wheel End can be found in Appendix I.10.

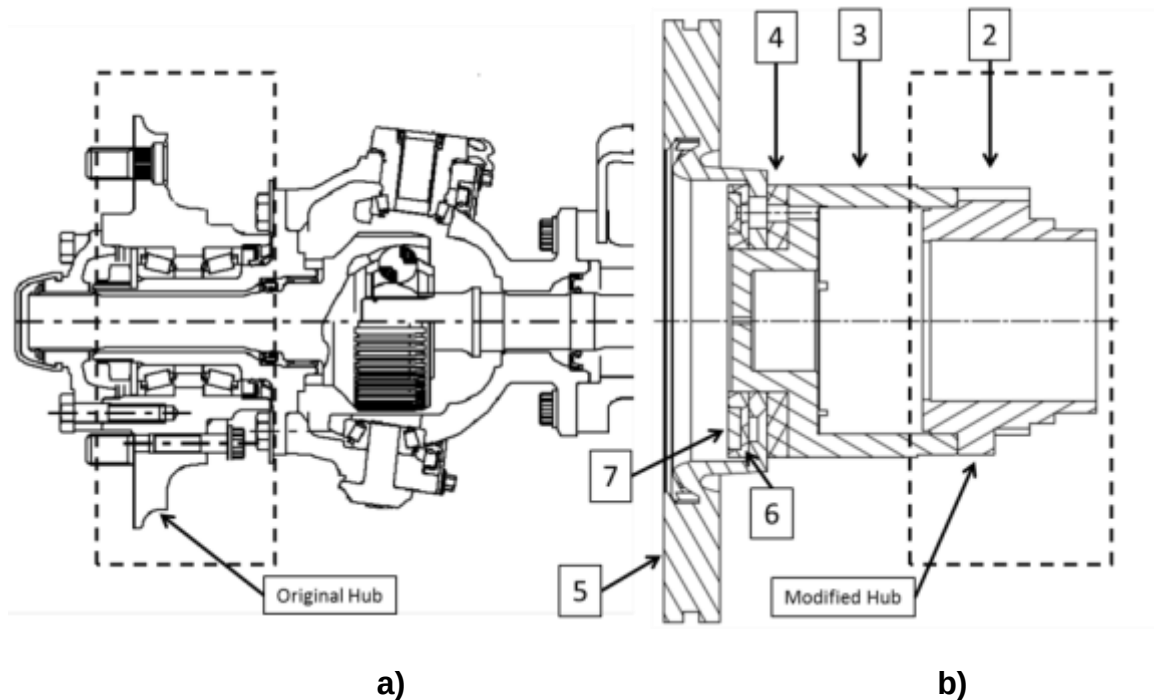


Figure 7-29 Hub Modification (a – original, b – modified)

Even with the modifications of the Wheel End, parts like the driveshaft's end tended to protrude and also there was no mounting point or surface for tested brake discs to be positioned on. This was addressed by designing the Steel Flange (Position 3 in Figure 7-29). The Flange is bolted on the Modified Wheel End and completely covers the part of the driveshaft that overhangs. In addition, it provides a surface for the discs tested to be mounted on the Flange. Its outer diameter is the same as the outer diameter of the Modified Wheel End for minimum blockage and disturbance in the cross flow. The manufacturing drawing for the Steel Flange is given in Appendix I.11.

Insulation flanges were designed and manufactured from Peek Plastic to prevent conductive heat transfer from the brake disc top hat to the Steel Flange. Named Inner Plastic Flange (Position 4 in Figure 7-29), it is located between the Steel Flange and the brake disc. The Peek Plastic is a colourless organic thermoplastic polymer, thermally stable with very low thermal conductivity ($k=0.25 \text{ W/m.K}$).

Finally, in Position 7 of Figure 7-29 there is the Outer Steel Ring with five counter shank holes (for five counter shank M8 screws) that holds the two insulation Plastic Flanges and the brake disc together on the Steel Flange. The Outer

Plastic Flange (Position 6 in Figure 7-29) prevents conductive heat transfer from the brake disc's top hat to the Outer Steel Ring and the screws, and the heat transfer to the Steel Flange. Manufacturing drawings of the Inner Plastic Flange, the Outer Plastic Flange and the Outer Steel Ring can be found in Appendix I.12 to Appendix I.14. Finally, the assembly drawing of the entire Assembly is in Appendix I.15.

The Steering Knuckle Assembly also carries the Heating Coil (Position 8 in Figure 7-24) of the induction heater and thermocouples with their mounting Support. These will be discussed separately in the Section for the Induction Heating System.

7.3.4 Electric Motor and Inverter

The rotation of the brake discs in the Disc Rotating and Heating Assembly is provided by the three-phase Electric Motor M2AA 1001-2 from ABB (Appendix J.1). The specification of the Motor is provided in Table 7-7.

Table 7-7 Electric Motor Specification

Manufacturer	ABB
Model	M2AA 1001-2
Power	3 [kW]
Current	6.2 [A]
Voltage	400 [V]
Maximum frequency	50 [Hz]
Maximum	2905 [rpm]

The Electric Motor is controlled by the Inverter Drive ACS150-03E-07A3-4 (Appendix J.2). The specification of the Inverter is presented in Table 7-8.

Table 7-8 Inverter Drive Specification

Manufacturer	ABB
Model	ACS150-03E-07A3-4
Frequency	48/63 [Hz]
Voltage	380/480 [V]
Number of phases	3
Nominal current	7.3 [A]
Nominal power	3 [kW]

7.3.5 Induction Heating System

The brake discs are heated up by an induction heating system. Induction heating is a method to heat up ferrous metallic objects made out of materials such as mild steel, stainless steel or cast iron by using electromagnetic induction. A typical induction heating system consists of an electromagnet and an electric oscillator that passes altering current at high frequency through the electromagnet. Electric currents are generated inside the metallic object (these are called eddy currents) as a result of the rapidly altering magnetic field. The metallic objects heat up from the inside due to the eddy currents flowing against the materials' electrical resistance. In ferromagnetic materials, the magnetic hysteresis also contributes to the temperature increase.

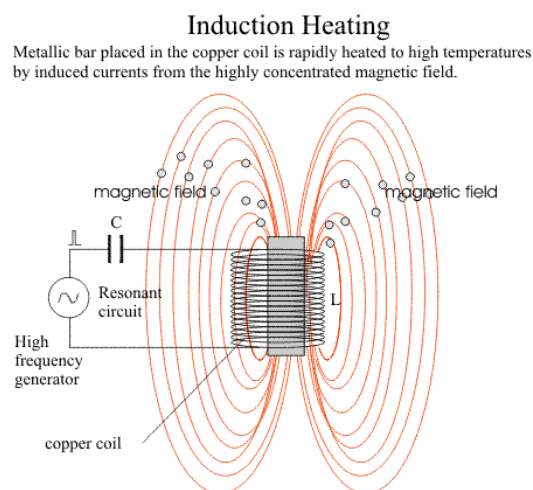


Figure 7-30 Induction Heating Process (source: www.dw-inductionheating.com)

The Induction Heating System for the Disc Rotating and Heating Assembly pre-existed in Cranfield University's Automotive Engineering Centre from earlier projects. It was provided by Precision Induction Heating Ltd and consists of four major parts, as schematically presented in Figure 7-31.

1. Heating Coil,
2. Workhead,
3. Control Unit,
4. Cooler.

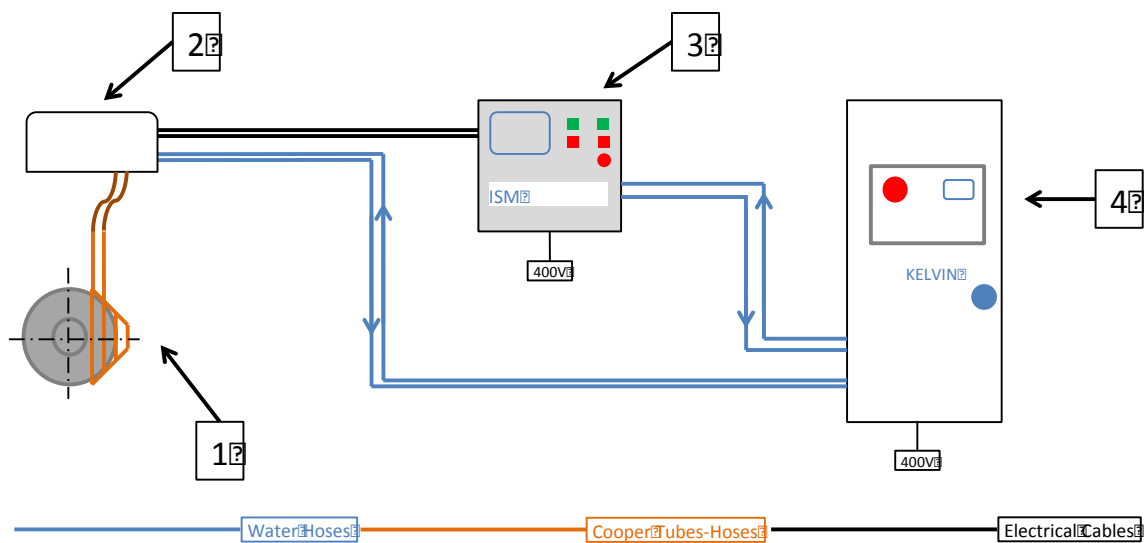


Figure 7-31 Induction Heating System Lay-out

The Heating Coil (Figure 7-32) is the electromagnet of the system generating the rapidly altering magnetic field causing the brake disc to heat up. The coil is an electric conductor made out of copper tube. Inside the Coil, water flows to keep its temperature at maximum 22°C. The water is coming from the Cooler and via the Workhead. The Coil's design is intended to heat up brake discs with outer diameters of up to 380 mm. The detailed drawing of the Heating Coil is illustrated in Appendix I.16.

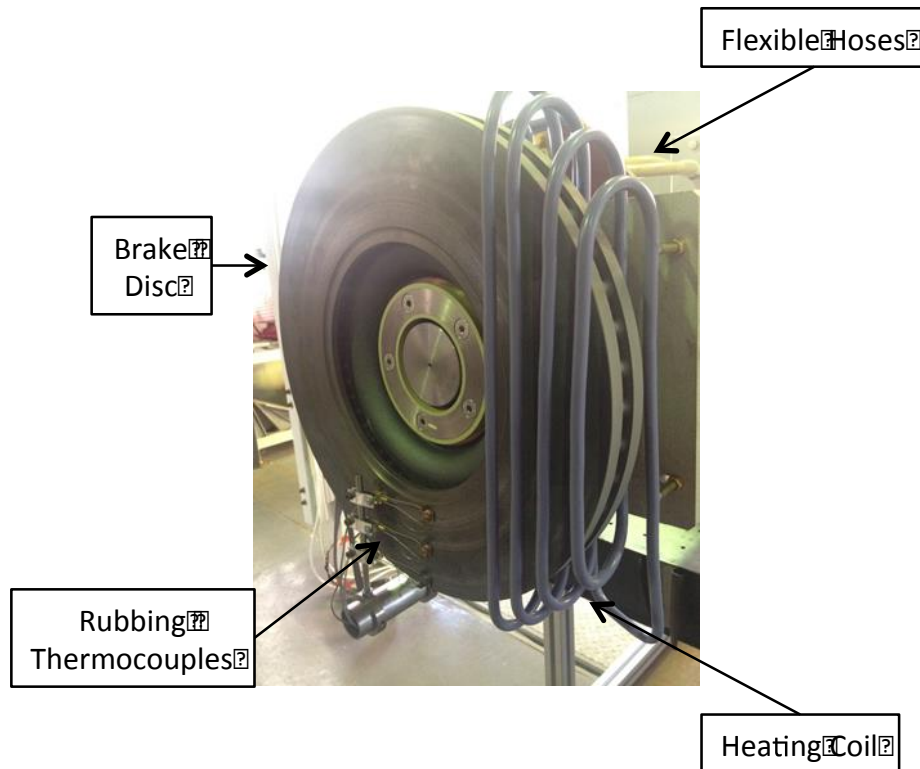


Figure 7-32 Heating Coil

The Heating coil is attached to the Workhead (Position 2 in Figure 7-31) via special flexible steel braided electric current conducting hoses permitting water flow to allow the steering knuckle movement. The Workhead includes a number of capacitors specifically selected to cover the operating band of the power supply and are tuned to deliver the rated power to the part being heated. The drawing of the Workhead is presented in Appendix I.16.

The electric oscillator of the system is the ISM Control Unit from Ambrell Induction Heating Solutions (Position 3 in Figure 7-31, picture in Appendix K.1). The Unit is connected to the Workhead and controls the voltage input in order to adjust the Heating System according to set heating requirements. The Unit's specification is shown in Table 7-9.

Table 7-9 ISM Control Unit Specification

Manufacturer	Ambrell
Model	ISM HF 15 170
Input voltage	360 – 520 [V]
Input frequency	45 – 65 [Hz]
Number of phases	3
Output power	15 [kW]
Output frequency	50 – 150 [kHz]

The water Cooler (illustrated in Appendix K.2), as the name suggests, cools the water and circulates it through the Control Unit, Workhead and Heating Coil. Although the Coil is made out of copper, which is a non-ferrous metal and does not heat up as quickly as ferrous metals, it would eventually heat up due to the altering magnetic field and the radiation from the brake disc. The Cooler keeps the temperature of the entire System between 19°C and 22 °C. The Cooler specification is presented in Table 7-10.

Table 7-10 Water Cooler Specification

Manufacturer	Kelvin S.r.l.
Model	KRNA70 400V 0012
PT/PS Pressure	6 [bar]
Temperature range	15 to 45 [°C]
Nominal voltage	400 V / 3 / 50 [Hz]
Running current	8.1 [A]
Starting current	35.5 [A]
Cooling capacity	7 [kW]
Pump available pressure	160 – 500 [kPa]

7.3.6 Sensors and Data Logging Equipment

The Disc Rotating and Heating Assembly is equipped with two types of sensors: thermocouples and rotational speed sensor.

There are six rubbing thermocouples (Figure 7-33) placed on the brake disc tested in order to measure its temperature during the heating up and cooling down phase. Their specification is listed in Table 7-11.

Table 7-11 Thermocouple Specification

Manufacturer	TC Direct
Type	K
Colour code (BS EN 60584.3 : 2007)	Green
Maximum temperature	850 [°C]
Features	Preloading spring with screw and locknut

Three thermocouples were used on the inboard side of the disc and another three on the outboard side. On each side, one thermocouple was positioned close to the inner diameter, one on the mean radius and one close to the outer diameter of the disc. Each thermocouple was given a specific index based on its position on the disc (Figure 7-34).

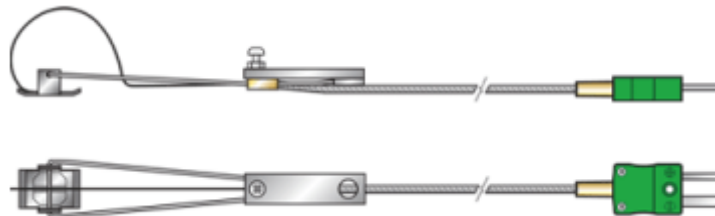


Figure 7-33 K Type Rubbing Thermocouple

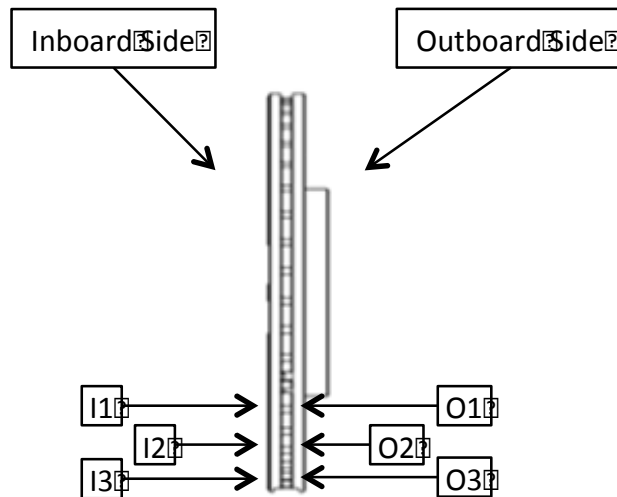


Figure 7-34 Thermocouple Positions

A special support (Figure 7-35 and Figure 7-36) for the thermocouples was designed and manufactured in order for them to be on the bottom side of the disc and parallel to the cross flow. It is mounted on the Defender's Steering Knuckle and is designed to be stiff, yet give freedom to the thermocouple positioning, for maximum adjustability with regards to their rubbing position. Manufacturing drawing of the thermocouples' support can be found in Appendix I.17.

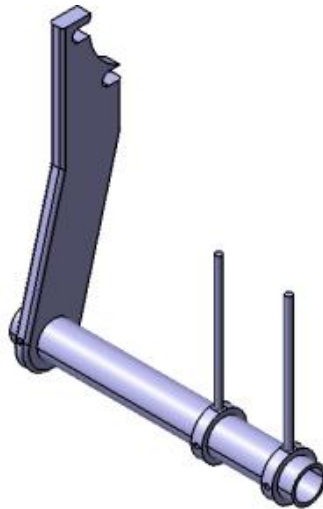


Figure 7-35 Thermocouple Support CAD Model



Figure 7-36 Thermocouple Support

As a result of having six thermocouples on the brake disc, their six cables require careful packaging as they are in the cross flow and the outboard rubbing thermocouples need to follow the steering movement. There is a further challenge in getting them out of the Flow Channel. The first issue was solved by using a hollow tube on the thermocouple support to avoid having the cables exposed to the cross flow and creating turbulences (Figure 7-36). Finally, a modified Defender's hollow steering track rod was used and further machined to pass the cables through it, and out of the Flow Channel (Figure 7-37).

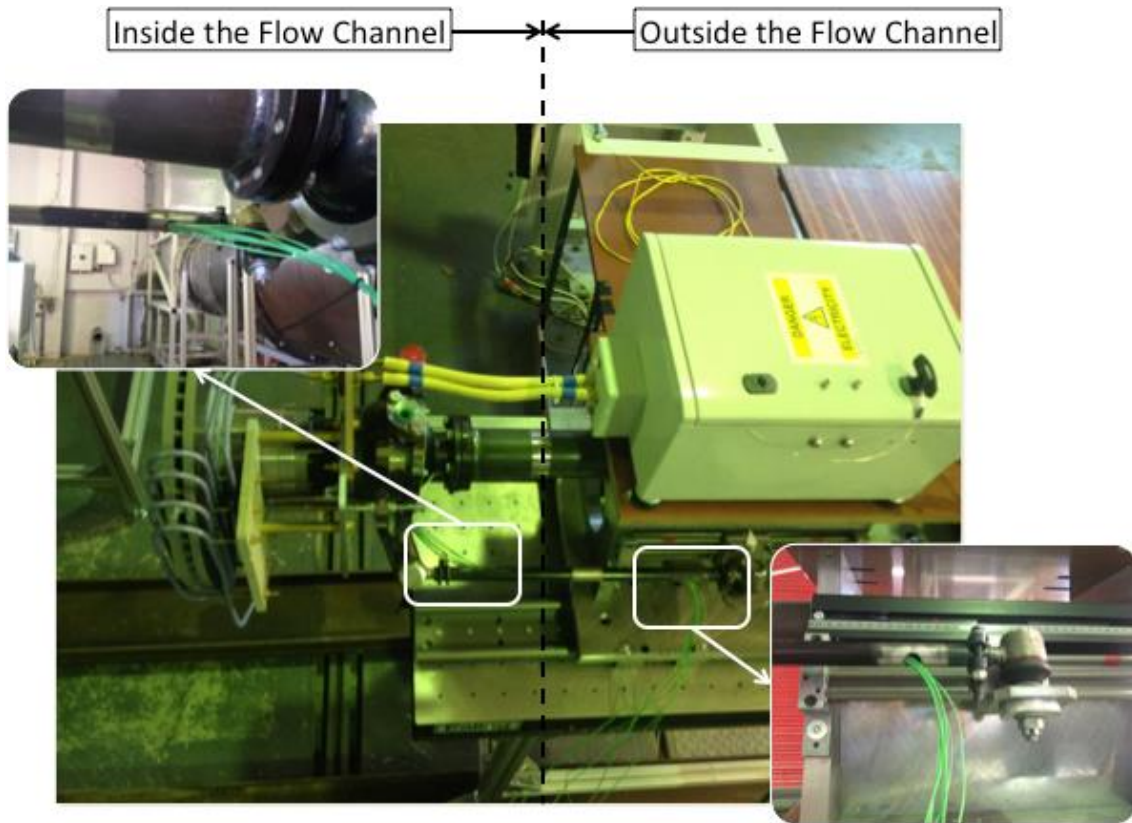


Figure 7-37 Modified Track Rod

In addition to the rubbing thermocouples, there is a probe thermocouple positioned on the support for the Inverter Drive (Figure 7-38). This thermocouple is used to measure the ambient temperature when cooling tests are conducted in still air and not in the Flow Channel. It is a K Type with characteristics similar to the rubbing thermocouples, but instead of the preloading spring with screw and locknut, it has a probe.



Figure 7-38 Probe Thermocouple

The rotational Hall speed sensor (Figure 7-39) of the Disc Rotating and Heating Assembly was installed to measure the angular velocity of the Electric Motor that is identical to the speed of the brake disc. The sensor works by picking up the key on the coupling between the Electric Motor and the Axle Assembly. This means that the resolution is low as there is only one reference point, but the tests are conducted at constant speed. The selected sensor provides a path for current to flow when activated that results in the sensor not being affected by noise coming from external sources. In Table 7-12, the specification of the Hall sensor is presented.



Figure 7-39 Rotational Speed Hall Sensor Installation

Table 7-12 Hall Sensor Specification

Manufacturer	Cherry Electrical Products
Model	GS100502
Operating voltage range	4.5 – 24 [V] DC
Supply current	6 [mA] max.
Output	Sink
Output saturation voltage	400 [mV] max.
Output current	25 [mA] max.
Operating temperature range	- 40 – 125 [°C]
Storage temperature range	- 40 – 125 [°C]
Barrel length	65 [mm]

While tuning the Inverter Drive to the required frequency during a test trial session, it was noticed that although the relation between the frequency and the Electric Motor Speed was considered to be linear, this was not actually true (Figure 7-40). This finding further justifies the use of the Hall speed sensor for the more accurate measurement of the rotational speed.

Finally, it was observed that when the Electric Motor's rotational speed exceeded 2634 rpm, excessive vibrations occurred that can make the Brake Disc Cooling Rig unsafe to use. As a result of that, Electric Motor's maximum speed is reduced from nominal 2905 rpm, to 2634 rpm. Nevertheless, this lower limit is acceptable, as it corresponds to vehicle speed of 340 km/h for a wheel with rolling radius of 0.347 m.

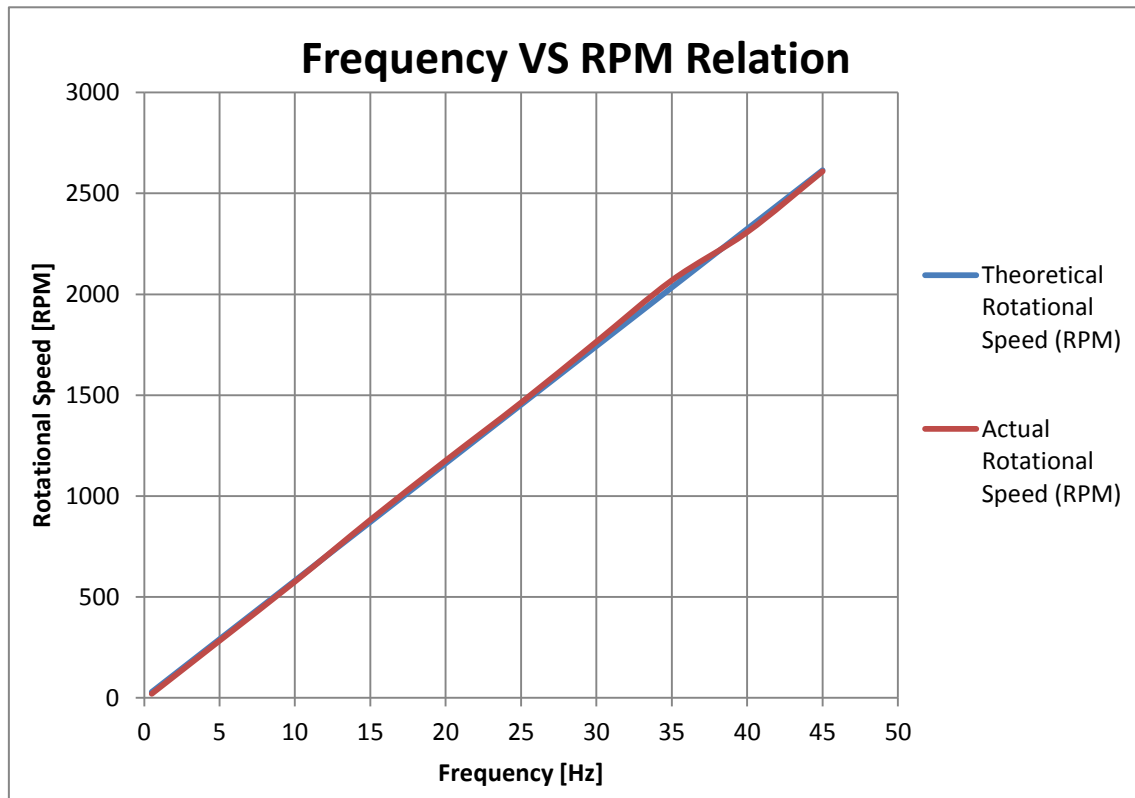


Figure 7-40 Inverter Frequency VS Electric Motor Speed Relation

The Data Logging Equipment of the Disc Rotating and Heating Assembly consists of two aspects: the hardware and the software. For the hardware, the core is the NI 9022 CompactRIO (cRIO) from National Instruments (picture in Appendix L.1). The cRIO is a reconfigurable embedded control and data acquisition system that

combines an embedded controller running a real-time operating system and a reprogrammable FPGA circuit. It is powered by a NI PS-15 power supply of 24 V DC. The specification of the NI cRIO is given in Table 7-13.

Table 7-13 CompactRIO Specification

Manufacturer	National Instruments
Model	NI CompactRIO 9022
Processor type	533 [MHz] PowerPC
CPU clock frequency	533 [MHz]
Operating system/target	Real-time
Nonvolatile memory	2 [GB]
System memory	256 [MB]
Number of Ethernet ports	2
Input voltage range	9 – 35 [V]
Operating temperature	-20 to +55 [°C]

A chassis is attached on the cRIO able to accommodate eight optional modules. Each of the modules has different functions and capabilities. For the Brake Disc Cooling Rig, three modules were selected:

- An analogue input module (NI 9205),
- A thermocouple module (NI 9214),
- A sourcing digital input module (NI 9426).

The NI 9205 module receives analogue input signals from the Tempatron DP thermometer/hydrometer for temperature and humidity (discussed in the Cross Flow Commissioning Report). It also receives the signals from the CMR-P Sensor with regards to the air velocity in the Flow Channel. The NI 9214 module receives the voltage signals from the six rubbing thermocouples on the brake disc. The NI 9426 module receives the digital signals from the Hall sensor with the rotational speed of the Electric Motor. The specifications for each of the modules used can be found in Appendix L.2.

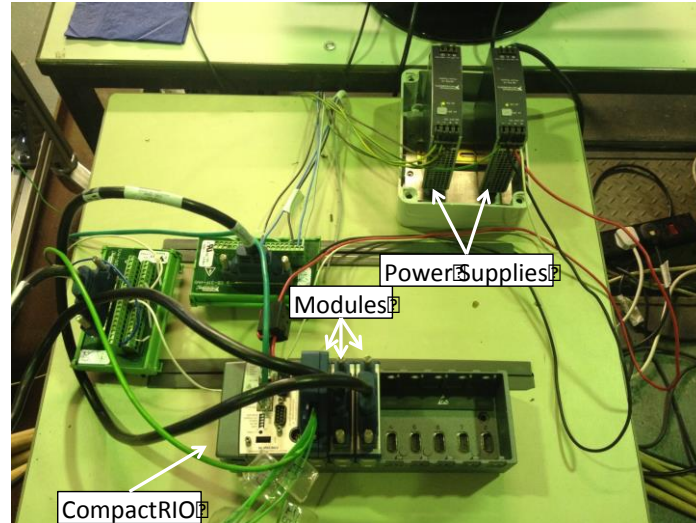


Figure 7-41 The Data Logging System

The data logging software system was developed specifically for the Brake Disc Cooling Rig by using the LabVIEW software from National Instruments. The software is installed in a Windows based PC. The cRIO is connected to the same PC and the developed software can gather all the measurements coming from the different sensors of the Rig and illustrate them on the screen in real-time. It can also log and store any of the measurements for processing after the cooling tests. It is robust with no “lagging” or “crashing”, which is important for the accuracy of the results and the effectiveness of the testing.

7.4 Chapter Summary

In the present Chapter, the developed Brake Disc Cooling Facility of Cranfield University’s Automotive Centre was presented. It offers unique ability to test the fingered disc cooling characteristics in a wide temperature range (up to 450°C), for a range of rotational speeds (0-2643 rpm), at air cross flow velocities between 0 m/s and 13.5m/s and steering angles -28° to +25°.

Its two main parts, the Flow Channel and the Disc Rotating and Heating Assembly has been thoroughly discussed with their individual components being described along with their specifications. Furthermore, the reasoning behind design decisions has been explained.

It is believed that all the initial design specifications have been achieved, something that will be proven in the commission Chapter that follows.

8 BRAKE DISC COOLING FACILITY COMMISSIONING

8.1 Introduction

Upon the completion of the Brake Cooling Rig, described in Chapter 7, and before initiating the operation of the facility, it was deemed necessary to conduct commissioning tests. The aim was to prove that all the Rig's systems and subsystems operate according to the design intent and also the quoted specifications. The additional aim was to identify system shortcomings or malfunctions.

The commissioning of the Flow Channel will be presented first, followed by the tests that took place for the Disc Rotating and Heating Assembly.

8.2 Flow Channel Commissioning

In the commissioning progress, the air velocities inside the Flow Channel were measured as it is important to ensure good quality air flow. In this Section, the methodology of setting up the measurements is discussed first. Thereafter, the measured air velocities are presented.

8.2.1 Methodology

The motor of the Fan Unit can be controlled by adjusting the frequency of the inverter, from 0 Hz to 50 Hz. It was decided to measure the air speeds inside the Flow Channel for the fan motor operating in steps of 10 Hz, starting from 10 Hz. The air measurements were taken in three different planes of the Flow Channel's Segment 2 (Figure 8-1); in Plane 1, which is 1300 mm in front of the Disc Rotating Assembly axis, in Plane 2, where the Assembly's axis is, and in Plane 3, 1300 mm after the axis of the Assembly.

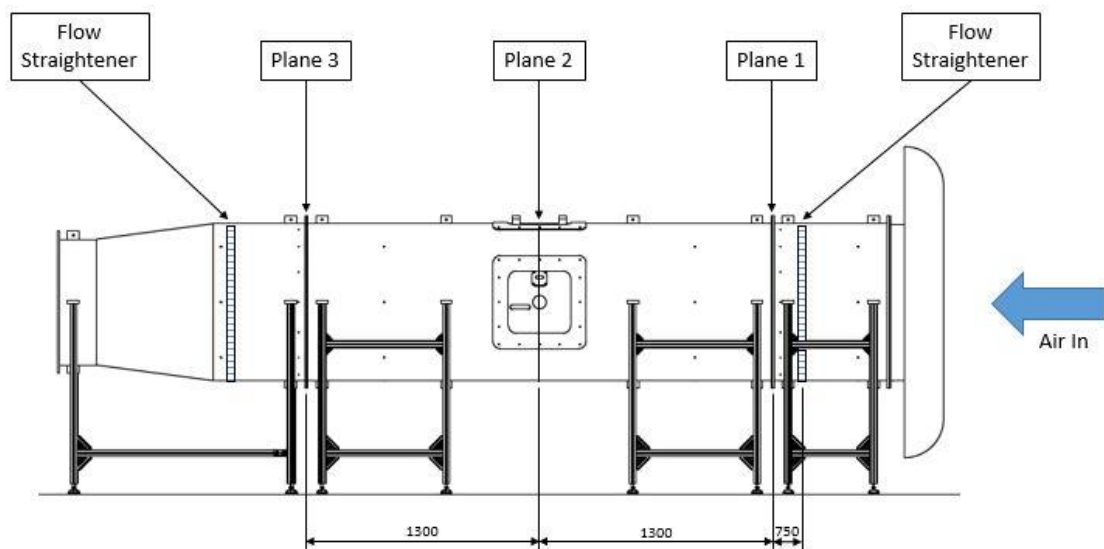


Figure 8-1 Measuring Planes (L/H View)

Furthermore, the air velocity was measured in each plane at several points for creating an air speed map and for having a good understanding of the air speed distribution inside the Flow Channel. The points and their position relative to the inner wall are shown in Figure 8-2. The pattern is following the ASHRAE Standard 111-1988, which is universally followed when conducting air speed measurements in ducts or closed channels. For making the results easier to understand, each point is given a specific index. For each point, the speed was measured for a time period of 2 minutes with the maximum, minimum and average values being recorded.

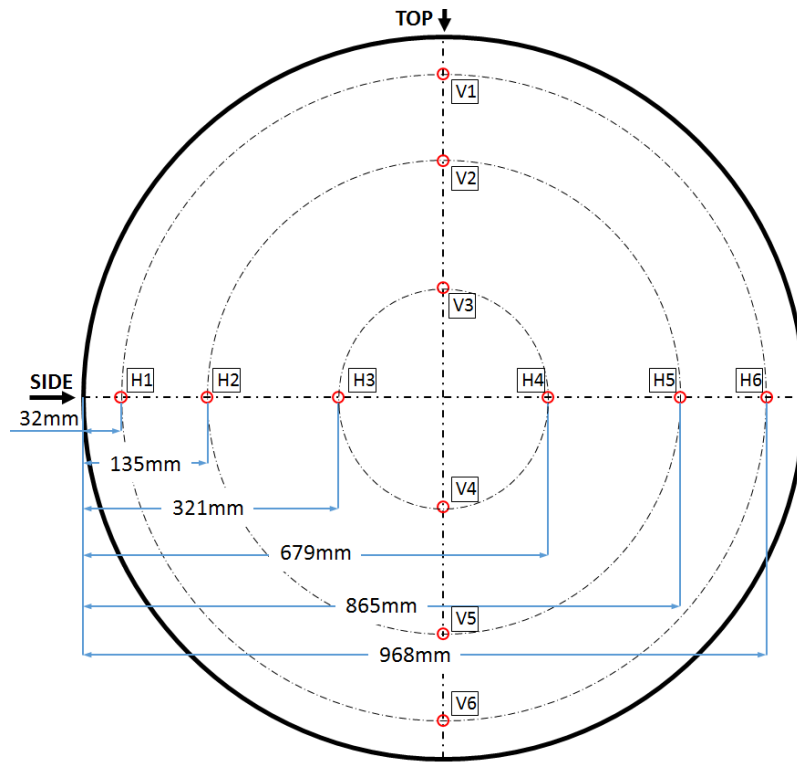


Figure 8-2 Measuring Points Inside the Channel (ASHRAE Std. 111 – 1988)

The measurements were conducted using the handheld Fluke 922 Airflow Meter (Figure 8-3 - a), which is convenient to use as it is portable and the CMR P-Sensor (Figure 8-3 -b). Two different pitot static “L-Type” tubes were used for the measurements. The first was 300 mm long (Fluke 922) with 8 mm outer diameter and 8 static pressure holes. It was used for measuring the air at points H1, H2, V1 and V2. The second Sensing Precision pitot tube used, was 1000 mm long with 8 mm outer dimension and 7 static pressure holes; it was used for all the other points. Particular care was taken in installing the pitot tubes in the Flow Channel providing air tight, mechanically strong and geometrically accurate position. Both tubes were tightened in the Flow Channel by special glands so that they are firmly and accurately positioned during operation. Finally, all hatches were sealed with duct tape to prevent any air leaks.

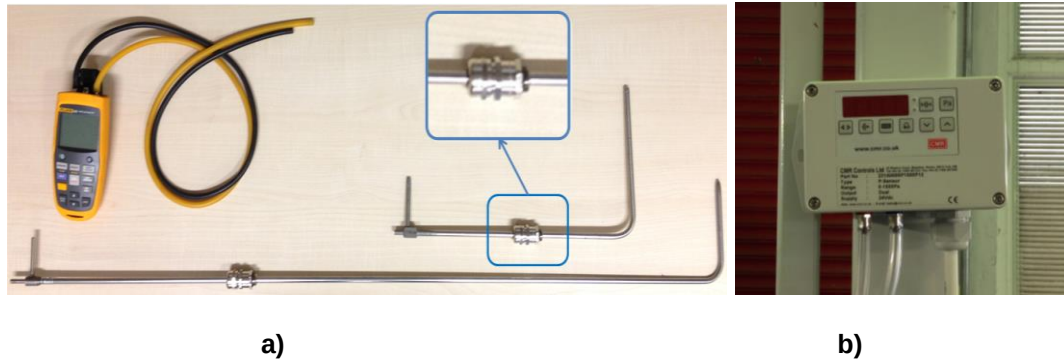


Figure 8-3 a) Fluke 922 and the Pitot Static Tubes, b) CMR P-Sensor

The technical specifications for the Fluke 922 are presented in Table 8-1. The calibration was done in the Turbomachinery Department of Cranfield University. For the calibration, a BETAGAUGE 330 calibrator from Martel Electronics was used.

Table 8-1 Fluke 922 Specifications

Parameter	Range	Accuracy	Resolution	Units Displayed
Air Pressure	± 4000 Pa ± 16 in H ₂ O ± 400 mm H ₂ O ± 40 mbar ± 0.6 PSI	$\pm 1\% + 1$ Pa $\pm 1\% + 0.01$ in H ₂ O $\pm 1\% + 0.1$ mm H ₂ O $\pm 1\% + 0.01$ mbar $\pm 1\% + 0.0001$ PSI	1 Pa 0.01 in H ₂ O 0.1 mm H ₂ O 0.01 mbar 0.0001 PSI	Pa H ₂ O mm H ₂ O mbar PSI
Air Velocity	250 – 16,000 fpm 1 – 80 m/s	$\pm 2.5\%$ of reading at 2000 (10.000 m/s)	1 fpm 0.001 m/s	fpm m/s
Air Flow (Volume)	0 – 99.999 cmf 0 – 99.999 m ³ /hr 0 – 99,999 l/s	Accuracy is function of velocity and duct size	1 cfm 1 m ³ /hr 1 l/s	cfm m ³ /hr l/s
Temperature	0 – 50 °C 32 to 122 °F	$\pm 1\% + 2$ °C $\pm 1\% + 4$ °F	0.1 °C 0.1 °F	°C °F

The specifications of the CMR P-Sensor airflow meter are given in Table 8-2. The instrument's calibration was performed by CMR Controls Ltd. satisfying the United Kingdom Accreditation Service (UKAS) standards.

Table 8-2 CMR P-Sensor Specifications

Range	± 7500 [Pa]
Accuracy	$\pm 0.25\%$ of full scale
Hysteresis/Repeatability	0.1% typical of full scale
Zero Drift	Automatically corrected
Operating Temperature	-10 [°C] to + 70 [°C]

8.2.2 Air Speed Measurements

As mentioned in Section 8.2.1, the air speeds were measured for five different inverter frequencies (10 Hz, 20 Hz, 30 Hz, 40 Hz and 50 Hz), at planes 1, 2 and 3 and in 12 points for each of these planes. For the sake of brevity, only the graphs of the air velocities achieved will be shown here. The precise air speed values can be found in Appendix M. It has to be noted that the velocities at points H6 and V6 could not be measured for Planes 1 and 3 due to the limited access in the Flow Channel and the limitation in the length of the pitot static tubes. Similarly, that the velocities at points H1 and V6 could not be measured for Plane 2. The air speed measurement results for Plane 1 are shown in Figure 8-4 and Figure 8-5.

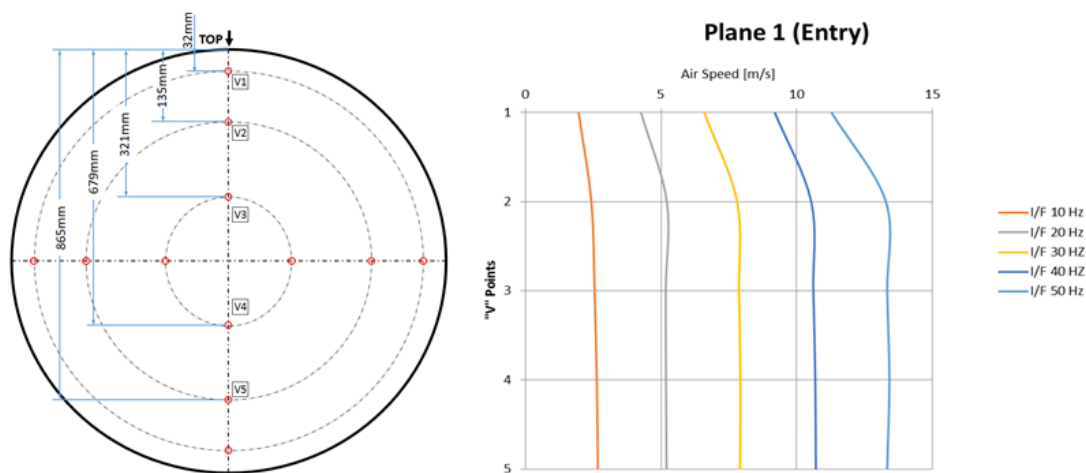


Figure 8-4 Plane 1, Average Air Velocities (Vertical)

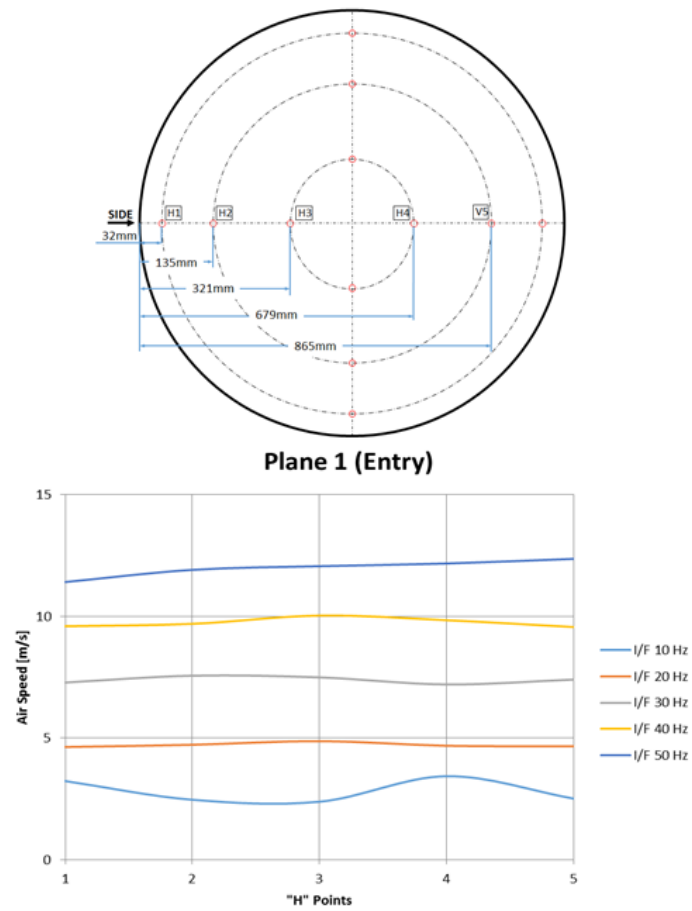


Figure 8-5 Plane 1, Average Air Velocities (Horizontal)

The results for the second plane are presented in Figure 8-6 and Figure 8-7.

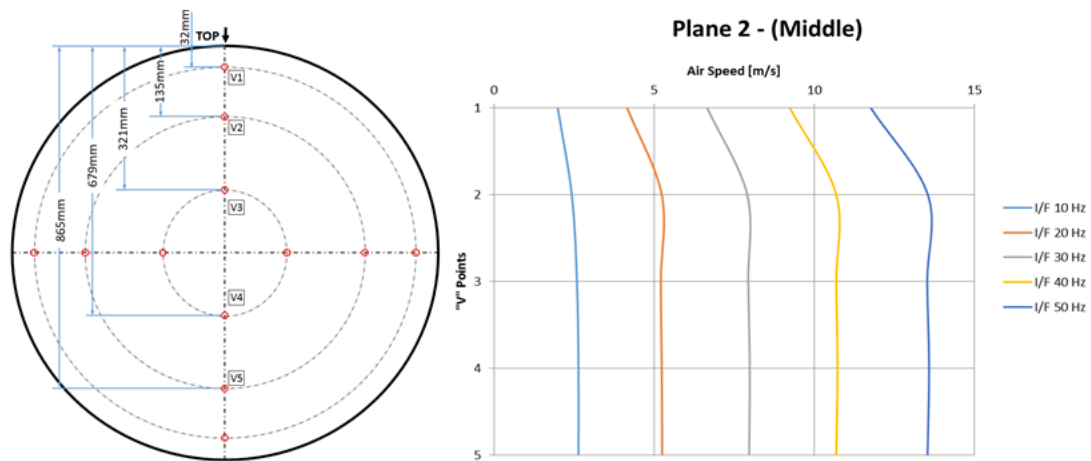


Figure 8-6 Plane 2, Average Air Velocities (Vertical)

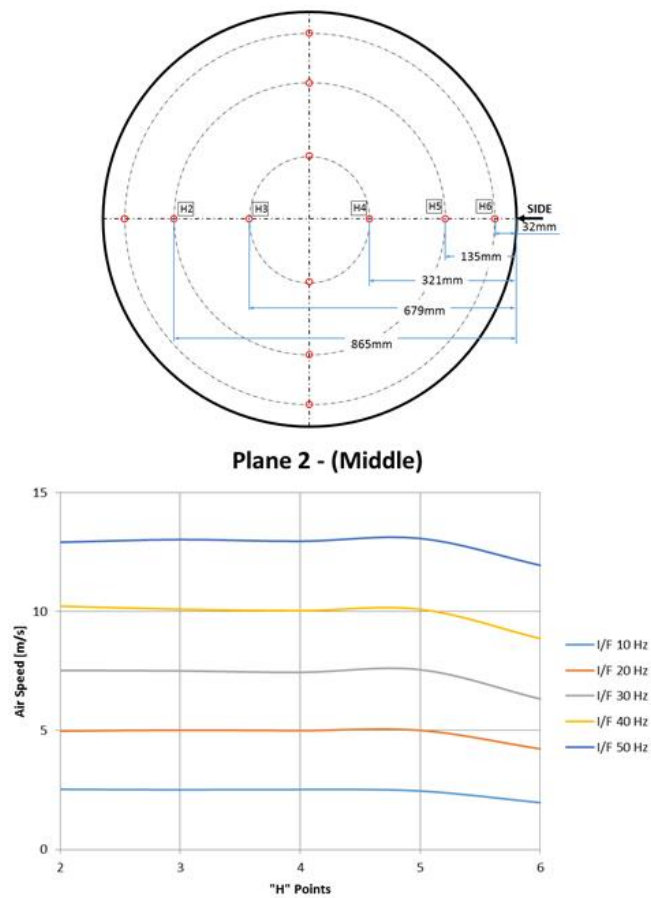


Figure 8-7 Plane 2, Average Air Velocities (Horizontal)

For the third and last plane, where the air velocities were measured, the results are shown in Figure 8-8 and Figure 8-9.

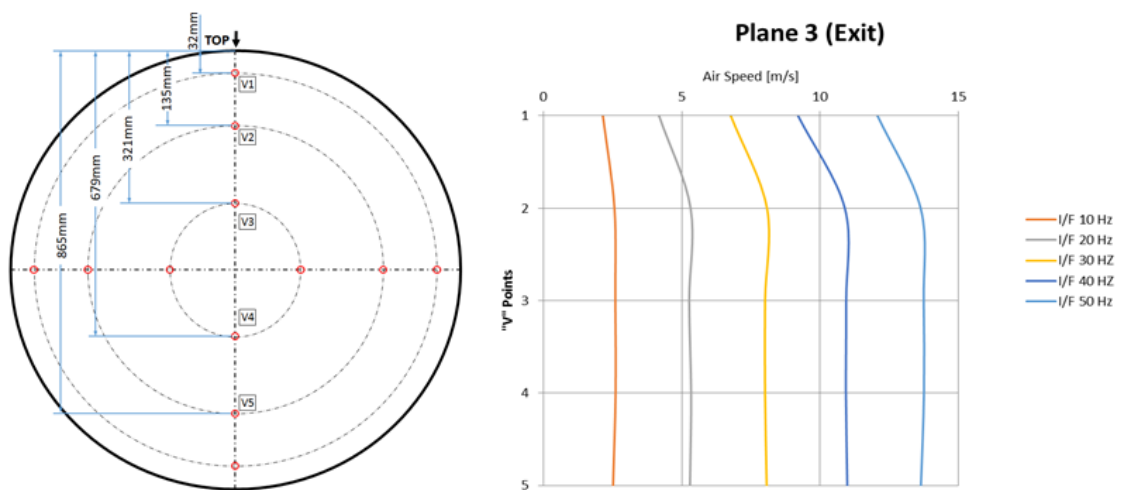


Figure 8-8 Plane 3, Average Air Velocities (Vertical)

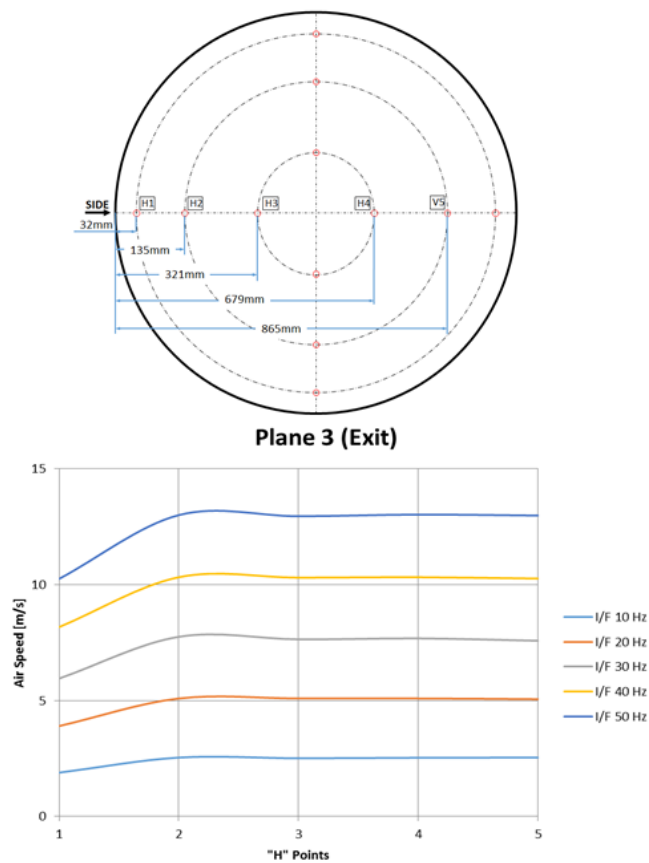


Figure 8-9 Plane 3, Average Air Velocities (Horizontal)

In order to visualise and better understand the speed achieved along the Working Section of the Flow Channel (planes 1, 2 and 3) for each measuring point, the values of air speed at the inverter frequency of 50 Hz were plotted in Figure 8-10. Similar graphs for 10 Hz, 20Hz, 30 Hz and 40 Hz can be found in Appendix N.

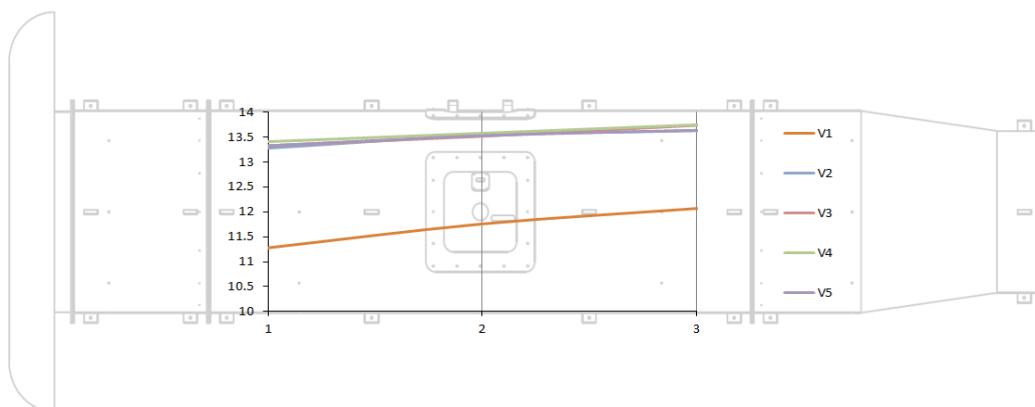


Figure 8-10 Air Speed Values Vs Measuring Planes at 50 Hz (Vertical)

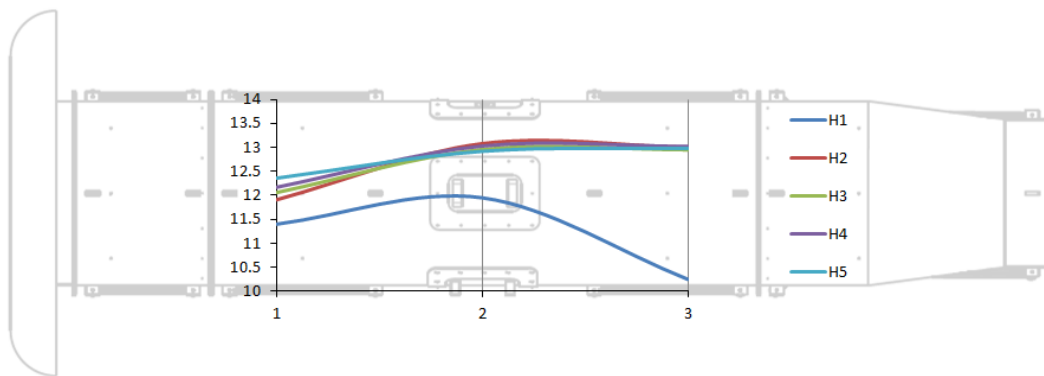


Figure 8-11 Air Speed Values Vs Measuring Planes at 50 Hz (Horizontal)

By reviewing Figure 8-10 and Figure 8-11 and also the figures in Appendix N, it can be concluded that the air velocities in plane 1 tend to deviate marginally, whereas the velocity values in planes 2 and 3 are very close to each other meaning that there is a reasonably uniform flow. In addition, as the frequency in the inverter and the air speed increase, the velocity values are more uniform.

Furthermore, in all cases the values at point H1 and V1 tend to fluctuate much more compared to the values in all the other points. This happens because these points are only 32 mm from the inner wall of the Channel meaning that most likely they are both inside the boundary layer. This is still well away from the disc OD and justifies the decision for selecting a larger channel inner diameter of 1 m instead of 0.8 m.

8.2.3 Static Pressure Measurements

The last test to verify that the achieved air velocities were the maximum possible, and not compromised by a static pressure increase behind the fan of the Flow Channel, entailed measurements of the static pressure at a number of points.

Two points before and two after the fan were selected. As it can be seen in Figure 8-12, point A is at a place where the inner diameter of the Channel is 1 m and just before it is gradually reduced to 0.8 m. Point B is just before the fan, at a horizontal distance of 575 mm. At an equal horizontal distance, but behind the fan, lays point C. Finally, point D was purposely selected to be in the beginning of the Exhaust Outlet structure as pulsation of the Exhaust Outlet's walls was

noticed during the Flow Channel operation at maximum inverter frequency, which may indicate a local increase of the static pressure.

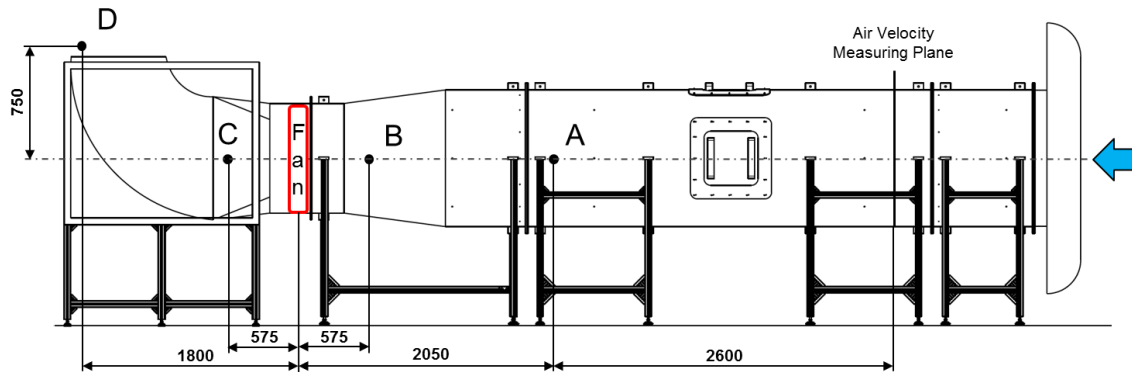


Figure 8-12 Static Pressure Measuring Points

At the points A, B, C and D clearance holes were drilled and brass tubes were installed by using Loctite super glue that did not allow any air leaks and kept the brass tubes steady in place. The brass tubes had an outer diameter of 1.5 mm and an inner diameter of 0.9 mm. Attention was given so that the brass tubes did not protrude into the air cross flow in order for the measurements to be accurate. The instrument used to take the pressure measurements was a Fluke 717 30G pressure calibrator. Its specifications are detailed in Table 8-3.

Table 8-3 Fluke 717 30G Specifications

Range	- 83000 to +206840 [Pa]
Resolution	10 [Pa]
Over Pressure	2 x full scale
Operating Temperature	-10 [°C] to +55 [°C]

The Fluke 717 30G was connected to the brass tubes via a flexible rubber tube of 2 mm outer diameter and 1.5 mm inner diameter. The test set up is graphically shown in Figure 8-13 and additional pictures of the tests can be found in Appendix O. In each measuring point, the static pressure was recorded at the air velocities of 2.5 m/s, 5 m/s, 7.5 m/s, 10 m/s, and finally 13.5 m/s.

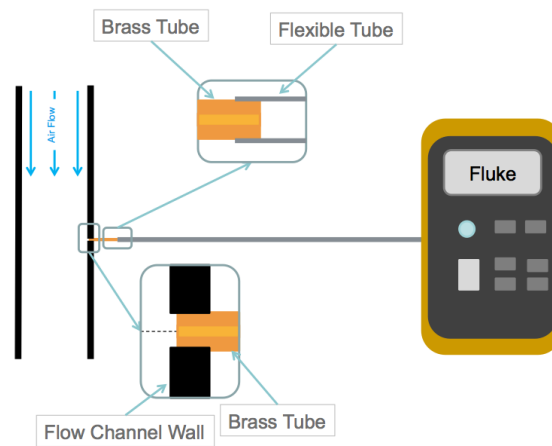


Figure 8-13 Static Pressure Measuring Test Set Up

The results of the static pressure test measurements are presented in Table 8-4 and by reviewing them it is noticed that at no point there is an increase in the value of static pressure. The findings led to the conclusion that during its operation, the Flow Channel achieves its theoretical maximum air velocity, as no increased back pressure exists.

Table 8-4 Static Pressure Measuring Results

Measuring Point	Static Pressures [Pa] <i>at air speed</i>				
	<i>2.5 m/s</i>	<i>5 m/s</i>	<i>7.5 m/s</i>	<i>10 m/s</i>	<i>13.5 m/s</i>
A	-10	-30	-60	-100	-160
B	-10	-40	-100	-170	-260
C	-10	-40	-100	-200	-330
D	-15	-25	-60	-105	-165

8.3 Disc Rotating and Heating Assembly Commissioning

The tests were conducted in still air, as the tests aimed to assess the Disc Rotating and Heating Assembly on its own and not, in the first instance, within the Flow Channel.

8.3.1 Methodology

Before conducting the tests in still air, the area where the tests were going to take place was isolated in order to avoid air draft currents that would compromise the test results. Enough obstacle free space was left around the Assembly for the pumping air from the rotating disc. A 19-inch brake disc was mounted on the Assembly. Finally, before starting the first test (and before the start of each test thereafter) the air velocity around the Assembly was measured at the points shown in Figure 8-14 to make sure there were no draft currents. Air movement could not be detected as all air velocities measured with the handheld TES 1340 anemometer were showing 0.0 m/s. The specification of the TES 1340 is presented in Table 8-5. The TES 1340 was calibrated using the airflow measurements of the Fluke 922 Airflow Meter as a reference.

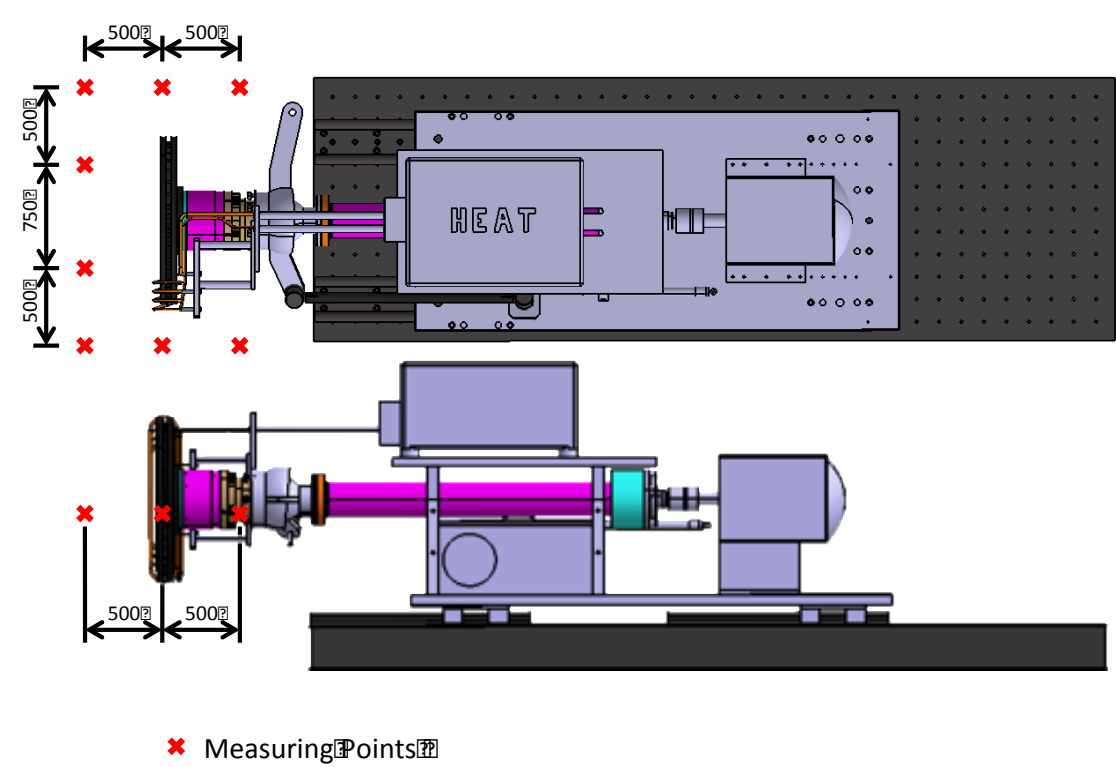


Figure 8-14 Measuring Points of Air Velocity

Table 8-5 Anemometer Specification

Manufacturer	RS Components
Model	TES 1340
Measuring range	0.0 – 30 [m/s]
Resolution	0.01 [m/s]
Response time	1 [s]
Operating temperature	0 – 50 [°C]

The following steps describe the test procedure:

1. Preparation

1.1. The desktop PC was switched on and the LabVIEW software for the Brake Disc Cooling Rig was run,

1.2. The Electric Motor and the Inverter Drive were switched on.

2. Disc Heating Up

2.1. The Inverter was set to 0.5Hz frequency that translated to disc rotational speed of 25 rpm,

2.2. The Induction Heater was switched on and set to 300 V to heat up the disc,

2.3. When the average measured disc temperature from the six rubbing thermocouples was higher than 450°C, the Induction Heater was switched off (the heating up phase took approximately 8 minutes).

3. Disc Cooling and Temperature Logging

3.1. The desired brake disc rotational speed was set by adjusting the frequency of the Inverter Drive,

3.2. The temperature logging from the six rubbing thermocouples on the disc and the probe thermocouple on the Disc Rotational Heating Assembly for the ambient temperature was initiated. In addition, the disc rotational speed was logged as well,

3.3. When the average disc temperature dropped below 100°C, the data logging and the cooling test were stopped.

The process above was repeated five times for five different rotational speeds that were (for wheel effective radius $r=0.347$ m):

- 133 rpm – 17.5 km/h,
- 265 rpm – 35 km/h,
- 398 rpm – 52.4km/h,
- 530 rpm – 70 km/h,
- 706 rpm – 100 km/h.

8.3.2 Cooling Curves for Test 3 @ 398 rpm

In Figure 8-15, an example of cooling curves from each thermocouple, when the disc was rotated at 398 rpm, is shown. It can be noticed that in the beginning of the cooling the disc temperature differs on the disc's friction faces, but gradually, as the cooling continues the temperature values become uniform. Also, due to the filtering of the thermocouples' signals from the software and the NI 9214 module, there is very little noise in the final results. Additional graphs with the results from all five tests can be found in Appendix P.

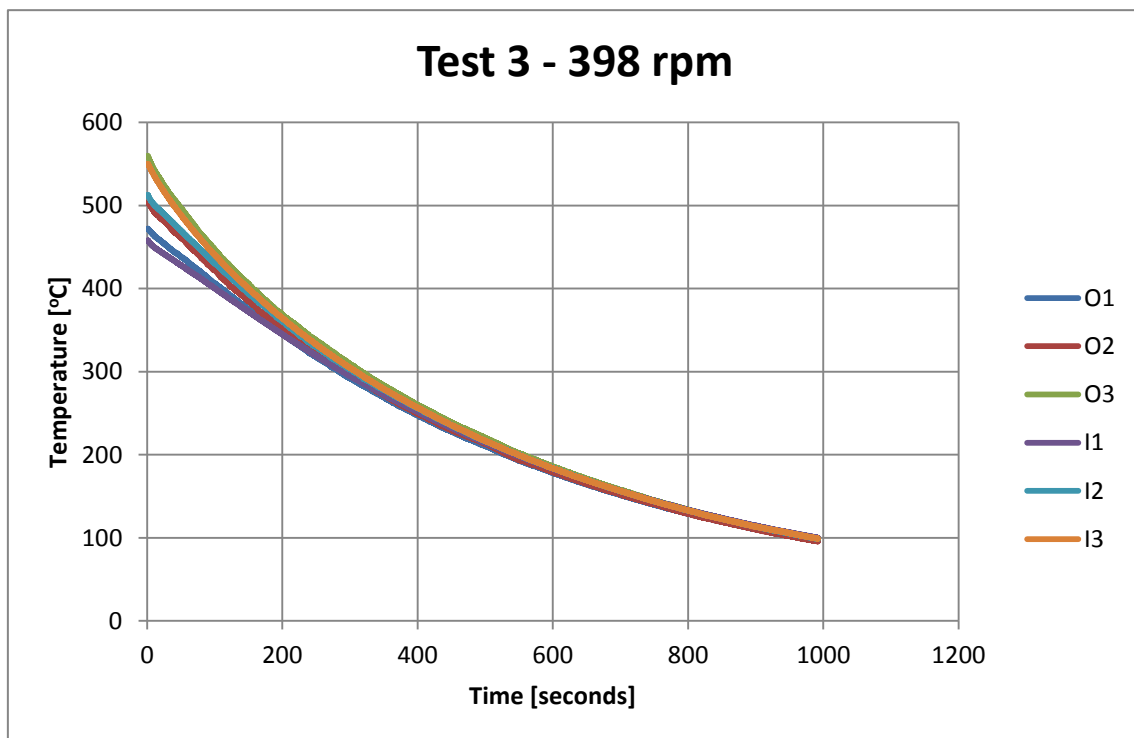


Figure 8-15 Cooling Curves @ 398 rpm

8.3.3 Cooling Curves for all Five Rotational Speeds

Figure 8-16 shows the average disc temperatures during the cooling period. Disc temperatures are averaged from all the six rubbing thermocouples on the disc. As expected, the faster the disc was spinning, the steeper the cooling curves were. For instance, when the disc was rotating at 133 rpm, it required almost 1770 seconds (29 and a half minutes) to reach an average temperature of 100 °C; whereas, when it was rotating at 706 rpm it only took 651 seconds (nearly 11 minutes) to reach the same temperature. Another example of the rotational speed's influence in the disc cooling was that by the time the disc in Test 5 (706 rpm) had an average temperature of 200°C, in Test 2 (265 rpm) it had 323°C and in Test 1 (133 rpm) it had 356°C.

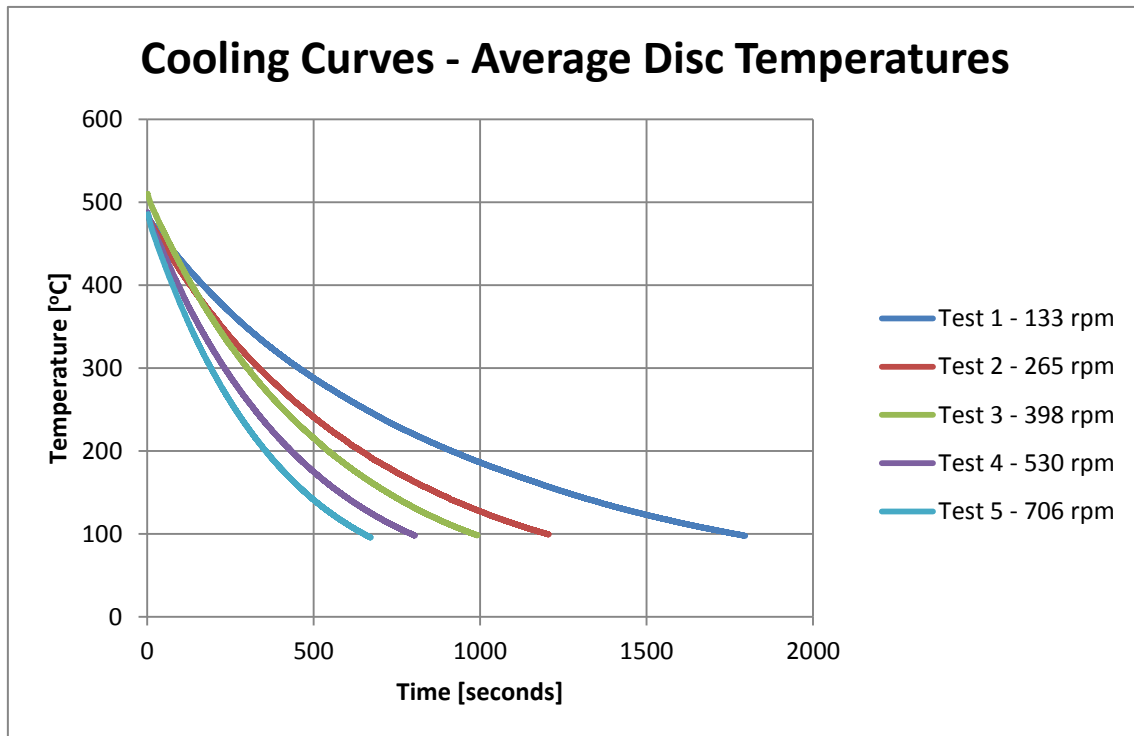


Figure 8-16 Average Disc Temperature Cooling Curves

From these cooling curves all three heat transfer coefficients [h] (for conduction, convection and radiation) can be calculated. The procedure for calculating h will be presented in Chapter 9, where the disc cooling within the Flow Channel will be discussed.

8.4 Chapter Summary

Within this Chapter, an extensive commissioning was conducted to both the main parts of the Brake Disc Cooling Facility that was presented in Chapter 7 aiming to prove that all the Rig systems and subsystems operate according to the design intent and also to the quoted specifications.

With regards to the Flow Channel, it was shown that the air speed distribution on both vertical and horizontal axis is uniform, especially in Plane 2 (at the Disc Rotating Assembly axis) and Plane 3 (after the Assembly's axis). From a distance of 135 mm to the inner wall and towards the centre of the Flow Channel the air velocities are uniform and stable. Cooling tests, with brake discs having an outer diameter of 420 mm, can be conducted as the discs are well within the clean air flow, justifying the selection of 1 m for the channel's inner diameter.

The air velocities achieved during the commissioning tests in the Flow Channel were 2.5 – 13.5 m/s with the air flow being stable with no pressure/speed variations. The air measurements in the Flow Channel will be taken in Plane 1 and more specifically in positions H2 and V3. Finally, the static pressure measurements indicate that there is no back pressure limiting the achieved air velocities.

The commissioning of the Disc Rotating and Heating Assembly showed that it can uniformly heat brake discs up to 450°C and that the cooling down is conducted in a repetitive manner with disc rotational speeds at up to 2643 rpm. Furthermore, the brake discs can be steered between + 25° and - 28° against the air cross flow and also that the entire Assembly can be accurately positioned in a position within the Flow Channel within an axial adjustment range of 508 mm.

As per the data logging equipment, the NI cRIO, the NI modules connected to it, and the developed software take reliable measurements and log the data accurately for further processing. As a result, the heat transfer coefficients can be calculated after having processed the data from the cooling tests and having derived the cooling curves.

9 DISC COOLING PERFORMANCE AND COOLING BEHAVIOUR CHARACTERISATION

9.1 Introduction

The main focus of this Chapter is to investigate the cooling performance of the Elise Version 3 prototype disc and compare it against the standard Elise S2 brake rotor. In order to achieve this, cooling tests were conducted in both still air and also in air cross flow. The cooling performance comparison of the two brake discs was based on the convective heat transfer coefficient $h_{Dconv} [\frac{W}{m^2K}]$. It has to be mentioned that no comparisons were conducted with regards to power or energy dissipated during braking because the disc areas were very similar. Thereafter, the cooling behaviour of each disc was characterised using the Nusselt Number (Nu), rotational Reynolds Number (Re_{ω}) and Reynolds Number for cross flow (Re_t).

9.2 Calculation of Heat Transfer Coefficients

When cooling tests take place, either in still air or in air cross flow, the raw measurements record the temperature of the disc at every increment of time throughout the duration of the test. By plotting the disc temperature against time, the cooling curves of each test can be derived and are similar to the curves presented in Sections 8.3.2, 8.3.3 and also in Appendix P. The disc temperatures are influenced by the three modes of heat dissipation: conduction, convection and radiation. Therefore, the heat transfer coefficients that can be extracted from the cooling curves are the total heat transfer coefficients that include the coefficients of conduction, convection and radiation. In order to calculate these coefficients individually, further analysis is required.

By using the energy balance equation below (Kreith & Bohn, 2001) (Moffat, 1988), the heat transfer coefficients can be derived from the cooling curves established by the logged raw data of the cooling tests.

$$Q_{in} = \left[\bar{h}_{Dconv} A_w (\bar{\theta}_D - \theta_\infty) + \sigma \varepsilon A_{rad} (\bar{T}_D^4 - T_\infty^4) + k \frac{A_{cond}}{l_{cond}} (\bar{\theta}_D - \theta_s) \right] = m_D C p_D \frac{d\bar{\theta}_D}{dt} \quad (9-1) \text{ Energy Balance}$$

The brake disc temperature used to calculate the heat transfer coefficients is the average temperature recorded from the six rubbing thermocouples I1, I2, I3, O1, O2 and O3 (see Section 7.3.6). In equation (9-1), the product $\bar{h}_{Dconv} A_w (\bar{\theta}_D - \theta_\infty)$ is the convection between the disc and the air that it surrounds it, the product $\sigma \varepsilon A_{rad} (\bar{T}_D^4 - T_\infty^4)$ is the amount of heat dissipated with radiation, and finally the product $k \frac{A_{cond}}{l_{cond}} (\bar{\theta}_D - \theta_s)$ is the conductance between the disc and the steering knuckle assembly. As mentioned in Section 7.3.3, an insulation flange made out of peek plastic was installed between the disc's top hat and the steering knuckle assembly to eliminate the heat transfer by conduction. Tests demonstrated that conductive losses are very small. Therefore, the conductive term of equation (9-1) was neglected and also Q_{in} is neglected because there is no heat added to the disc during the cooling process. By dividing equation (9-1) by $(\bar{\theta}_D - \theta_\infty)$, the resulting equation is:

$$\bar{h}_{Dt} A_w (\bar{\theta}_D - \theta_\infty) + m_D C p_D \frac{d\bar{\theta}_D}{dt} = 0 \quad (9-2) \text{ Lumped System Cooling}$$

The total heat transfer coefficient ($\bar{h}_{Dt}$) comprises the convective heat transfer coefficient ($\bar{h}_{Dconv}$) and the radiative heat transfer coefficient ($\bar{h}_{Drad}$) that contribute to the heat dissipated by the disc.

$$\bar{h}_{Dt} A_w = \bar{h}_{Dconv} A_w + \sigma \varepsilon A_{rad} \frac{\bar{T}_D^4 - T_\infty^4}{\bar{\theta}_D - \theta_\infty} \quad (9-3)$$

The differential equation (9-2) describes the cooling of a lumped system, a system where internal conductance resistance is very low and results to the temperature being distributed uniformly (Incropera & DeWitt, 2002). During the cooling process, by selecting two time increments, t_1 and t_2 , that correspond to temperatures $\bar{\theta}_1$ and $\bar{\theta}_2$ from the history plot (the cooling curves), and by using a logarithmic decay approximation, equation (9-2) can be written as:

$$\frac{1}{\Delta t} \ln \frac{\bar{\theta}_2 - \theta_\infty}{\bar{\theta}_1 - \theta_\infty} = - \frac{\bar{h}_{Dt} A_w}{m_D C p_D} \quad (9-4)$$

In equation (9-4), $\Delta t = t_2 - t_1$ and $\bar{\theta}_1 > \bar{\theta}_2$. With regards to the ambient temperature ($\bar{\theta}_\infty$), this is the average value of ambient temperature between the time increments t_1 and t_2 . From equation (9-4) the total heat transfer coefficient can be calculated by solving for $\bar{h}_{Dt}$:

$$\bar{h}_{Dt} = - \frac{m_D C p_D}{A_w \Delta t} \ln \frac{\bar{\theta}_2 - \theta_\infty}{\bar{\theta}_1 - \theta_\infty} \quad (9-5) \text{ Total Heat Transfer Coefficient}$$

For the calculation of the total heat transfer coefficient $\bar{h}_{Dt}$, attention should be given to the selection of the time increment Δt . It must cover a segment of the cooling curve that represents non-linear cooling behaviour, or else the calculation value would include error. A small time increment results in more calculations without necessarily increasing the accuracy of the results and large time increments results in inaccuracies as a linear temperature decrease is assumed for a long period of time, which is incorrect.

For all the conducted tests, the cooling process began at around 450°C and it was considered reasonable to allow the average temperature to drop by approximately 50°C in order for the temperature to equalise within the disc. For this reason, $\bar{\theta}_1 = 400^\circ\text{C}$ with the corresponding t_1 being recorded. It was also decided that the driving factor behind the selection of the time increment was the average temperature of the disc. In fact, the increments of time were taken for every 50°C of the average disc temperature drop until the temperature was equal to 100°C as shown in Figure 9-1.

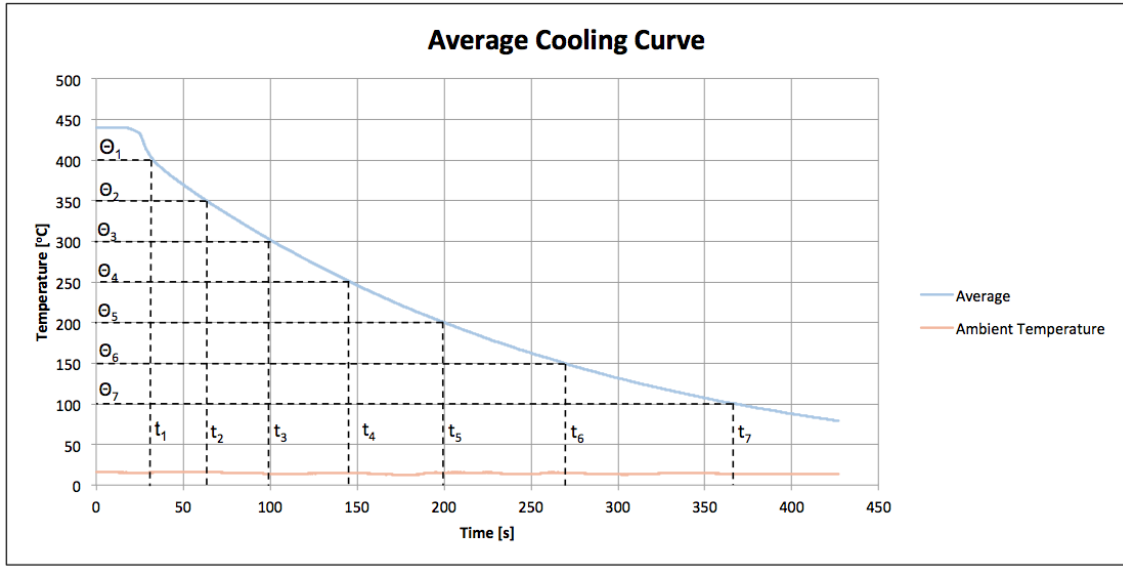


Figure 9-1 Time Increments for the Calculation of the Total Heat Transfer Coefficient

Since the total heat transfer coefficient ($\bar{h}_{Dt}$) is calculated using equation (9-5), the convective heat transfer coefficient ($\bar{h}_{Dconv}$) can be computed by subtracting the radiative heat transfer coefficient ($\bar{h}_{Drad}$), equation (9-6) from the total heat transfer coefficient.

$$\bar{h}_{Drad} = \sigma \varepsilon \frac{A_{rad}}{A_w} \frac{\bar{T}_D^4 - T_\infty^4}{\bar{\theta}_D - \theta_\infty} \quad (9-6) \quad \text{Radiative Heat Transfer Coefficient}$$

From equations (9-5) and (9-6), the formula for the average heat transfer coefficient for convection is:

$$\bar{h}_{Dconv} = \bar{h}_{Dt} - \bar{h}_{Drad} \quad (9-7) \quad \text{Convective Heat Transfer Coefficient}$$

The input data required for the calculations was extracted from the experimental results, private communications with industrial partners and own measurements. The disc surface temperatures $\bar{\theta}_{1-7}$, the average ambient temperatures $\bar{\theta}_\infty$ and the times t_{1-7} were taken from the cooling curves of each test. Temperature dependent values were used for specific heat capacity C_p and can be found in Appendix E.5.

Each disc was weighed to establish its mass. The values for the discs' wetted areas and areas of radiation were measured using the CAD models of the two discs. With regards to emissivity, the values for each disc were identified by comparing the logged temperature from each of the six rubbing thermocouples and then adjusting the emissivity setting on a Flir i3 thermal imaging camera (see Appendix R for specifications) in order to achieve same temperature readings. Inevitably, the identified emissivity values were an average of the different local emissivity values that exist in the disc areas. The measured input data is depicted in Table 9-1.

Table 9-1 Disc Input Data

	Elise Version 3	Elise S2
Mass [kg]	5.5	6.0
Wetted Area [m ²]	0.2372	0.2495
Area of Radiation [m ²]	0.1217	0.1321
Emissivity	0.82	0.80

Finally, the Stefan-Boltzmann constant used was $\sigma = 5.67 \times 10^{-8} \left[W/m^2K \right]$.

9.3 Calculation of the Nu, Re_t and Re_ω

Following the investigation of the cooling performance for the Elise Version 3 and the Elise S2 brake discs, the characterisation of their cooling behaviour was envisaged. This was possible by calculating the Nusselt Number (Nu) and the rotational Reynolds Number (Re_{ω}) for the case of the discs rotating in still air with the addition of the cross flow Reynolds Number (Re_t) when the discs were rotating while subjected to an air stream.

The equations that allow the computation of these dimensionless numbers are given in Section 2.5.3. More specifically, the Nusselt Number is given by equation (2-19), the rotational Reynolds Number by equation (2-15) and the cross flow Reynolds Number by equation (2-14).

The air properties have major impact on these numbers as the Nusselt Number is a function of the air thermal conductivity and the Reynolds Numbers are a function of the air kinematic viscosity. The data used for the calculations can be found in Appendix R. It has to be noted that the values of temperature at which the dimensionless numbers were computed were always the average disc temperatures minus the ambient temperature during the experiments.

The general form for characterising the cooling behaviour of a disc when rotating in still air is given by equation (9-8), which is the common form of equation to describe this test condition, as presented in Section 2.5.3.

$$Nu = aRe_{\omega}^b \quad \text{(9-8) General Equation Form – Still Air}$$

The equation to describe the cooling of a disc when it is rotating in a cross flow of air is equation (9-9), which was presented in (Sheridan, et al., 1988).

$$Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c \quad \text{(9-9) General Equation Form – Rotation in Cross Flow}$$

9.4 Step 9 – Cooling Tests in Still Air

The procedure followed for commissioning the Disc Rotating and Heating Assembly, discussed in Section 8.3.1, was also followed for the cooling tests in still air. Again, attention was given in eliminating any air drafts in the room where the tests were conducted. Additionally, the discs were positioned as far as possible from any obstacles that may have compromised the test outcomes with the primary obstacle being the Flow Channel itself (Figure 9-2).



Figure 9-2 Elise Version 3 Disc Ready to be Tested in Still Air

The test schedule that was compiled included seven tests at seven different rotational speeds starting at as low as 12.5 km/h and reaching up to 250 km/h in the seventh test. In Table 9-2, the test schedule is presented and it was followed for both Elise Version 3 and Elise S2 Discs.

Table 9-2 Test Schedule for Tests in Still Air

Test No	Equivalent Vehicle Speed [km/h]	Wheel/Disc Rotational Speed [RPM]
1	12.5	110.9
2	25	216.6
3	50	441.5
4	100	875.4
5	150	1333.8
6	200	1778.1
7	250	2219.3

For the conversion of linear to rotational speed, equation (9-10) was used and the value of rolling radius was 0.3 m as it has been previously shown in Table 3-1.

$$RPM = \frac{r_r \times 0.10472}{U} \quad (9-10) \quad \text{Linear to Rotational Speed}$$

9.4.1 Convective Heat Transfer Coefficient Results

In the current Section, the cooling performance of both the Elise Version 3 prototype and the Elise S2 disc are presented, compared and discussed. In Figure 9-3 the average convective heat transfer coefficient is plotted for both discs against the rotational speed. The values were derived for the average disc surface temperature of 275°C between points $\bar{\theta}_3$, $\bar{\theta}_4$ (see Figure 9-1). The results for the remaining average disc surface temperatures can be found in Figure 9-4 and in Appendix S.



Figure 9-3 Elise Version 3 VS Elise S2 Convective HTC at 275°C

Power trend lines were fitted to the points of the graph above, and in fact in all the graphs that plot the convective heat transfer coefficient against the rotational speed, in order to better understand the convective cooling and make comparisons easier. By examining Figure 9-3 it can be said that the Elise Version 3 has improved convective cooling compared to the Elise S2.

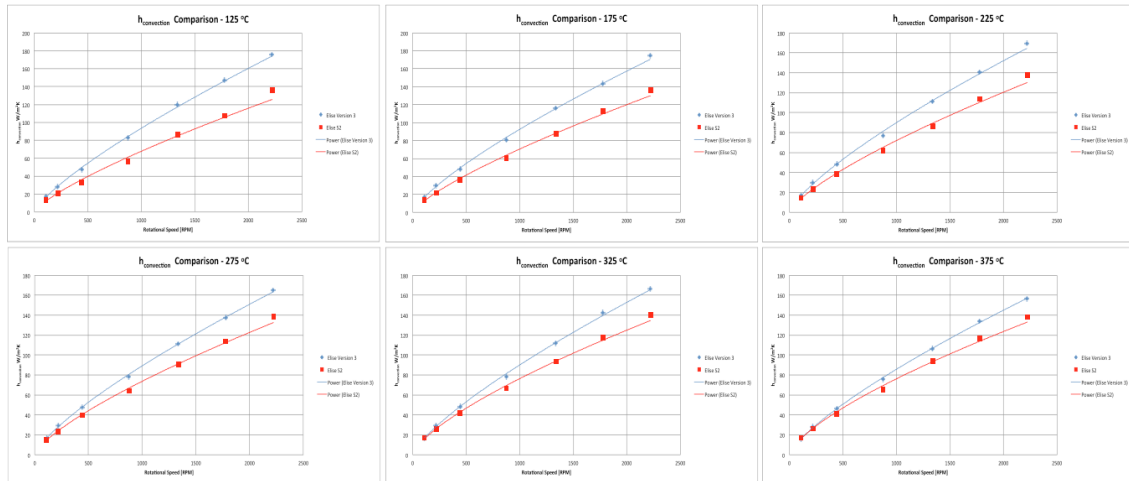


Figure 9-4 Elise Version 3 VS Elise S2 Convective HTC – All Temperatures

In fact, taking into consideration the results plotted in the graphs of Figure 9-4 and Appendix S it can be concluded that the Elise Version 3 features improved cooling compared to the Elise S2 for all the six average disc surface temperatures for which convective heat transfer coefficients were calculated.

Since both the tested discs are of the swan neck design it is expected that the cold air enters the disc from the inboard side. This is true for the Elise S2 disc; however, it was discovered that an additional amount of air was entering the disc from the outboard side in the case of the Elise Version 3 disc as well. This was happening due to the top hat having a fingered design allowing air to enter the disc via the hub's holes (Figure 9-5).

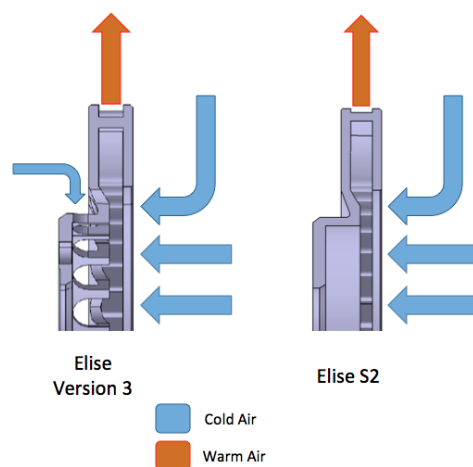


Figure 9-5 Air Flow in Elise Version 3 and Elise S2 Discs

From the graphs in Figure 9-4 and Appendix S it is evident that the Elise Version 3 features better cooling than the Elise S2. In Table 9-3 the difference of the convective heat transfer coefficient between the two discs is presented. At an average disc surface temperature of 125°C the Elise Version 3 has almost 37.5% better cooling than the Elise S2, whereas for the average disc surface temperature of 375°C, the difference in cooling was almost 10%. This indicates that at low rotational speeds and low average disc surface temperatures, the Elise Version 3 disc cools much better than the standard S2 disc, which means that the prototype disc is ideal for city driving.

Table 9-3 Elise Version 3's Convective HTC Difference to Elise S2

Equivalent Speed [km/h]	Elise Version 3 Convective HTC Difference [%]					
	<i>125°C</i>	<i>175°C</i>	<i>225°C</i>	<i>275°C</i>	<i>325°C</i>	<i>375°C</i>
<i>12.5</i>	30.61	25.14	15.15	6.86	-1.08	-8.53
<i>25</i>	35.94	35.68	27.97	26.01	14.74	8.40
<i>50</i>	43.75	34.55	25.56	20.66	16.87	12.89
<i>100</i>	46.51	33.34	23.82	21.92	17.95	15.42
<i>150</i>	39.19	32.35	28.43	23.00	19.75	12.92
<i>200</i>	37.36	27.28	23.78	21.14	20.99	14.92
<i>250</i>	29.10	28.51	23.35	18.96	18.81	13.29
	Average					
	37.49	31.01	24.01	19.79	15.43	9.90

Moreover, from the information of the table above it seems that at the very low rotational speed of 12.5 km/h the cooling performance difference in favour of the Elise Version 3 is limited compared to the difference at the speeds between 25 to 200 km/h. Furthermore, at the average surface temperatures of 325°C and 375°C, the Elise S2 disc is performing marginally and clearly better respectively.

In an effort to understand the reduction between the cooling performance of the two discs with the increase of the average disc surface temperatures, the convective heat transfer coefficients were plotted against the rotational speed for each of the six average disc surface temperatures.

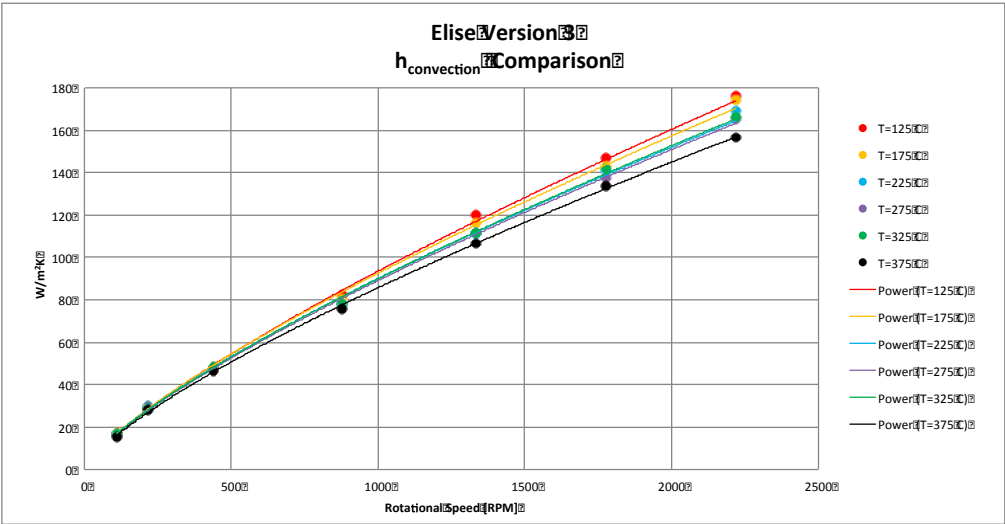


Figure 9-6 Convective HTC Curves at Different Average Disc Surface Temperatures – Elise Version 3

In Figure 9-6, the $\bar{h}_{D_{conv}}$ curves for the Elise Version 3 are presented and is evident that the disc is cooling better when the average surface temperatures are low. A similar graph, but for the Elise S2, is shown in Figure 9-7.

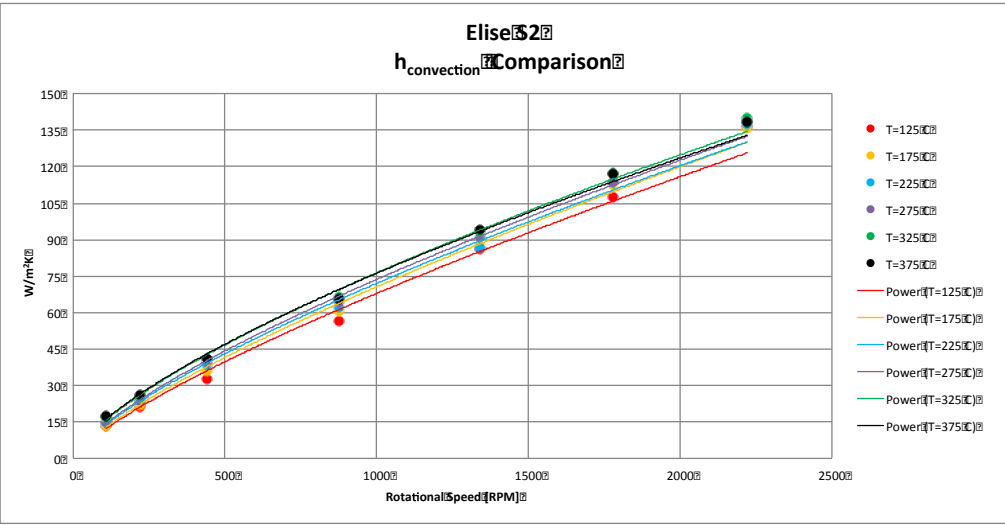


Figure 9-7 Convective HTC Curves at Different Average Disc Surface Temperatures – Elise S2

In contrast to what was observed in Figure 9-6, the Elise S2 disc features improved cooling when the average surface temperatures are high. The combination of the facts that the Elise Version 3 cools better at low temperatures and the Elise S2 has improved cooling at high temperatures results in the gradual decrease of the Elise Version 3 cooling advantage.

Despite this, the general outcome from the test results in still air is that the prototype Elise Version 3 disc certainly features improved cooling than the standard Lotus Elise disc.

9.4.2 Cooling Behaviour Characterisation in Still Air

With regards to the cooling behaviour characterisation of both discs in still air, the relation between the Nusselt number and the rotational Reynolds number for each of the six average disc surface temperatures was plotted. By fitting a power curve between the $\bar{h}_{D_{conv}}$ points, the identification of the (a) and (b) terms of equation (9-8) was possible. As an example, the Nu vs Re_{ω} graphs at 275°C average disc surface temperature are illustrated in Figure 9-8 for the Elise Version 3 and in Figure 9-9 for the Elise S2 respectively. The graphs for the remaining average temperatures can be found in Appendix T.

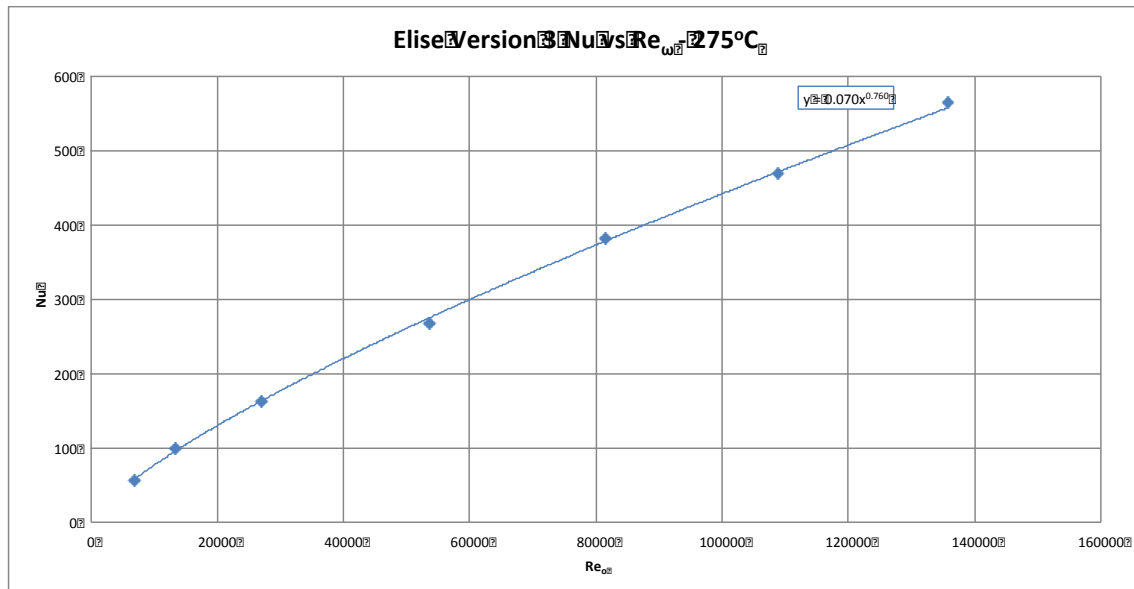


Figure 9-8 Nu vs Re_{ω} at 275°C – Elise Version 3

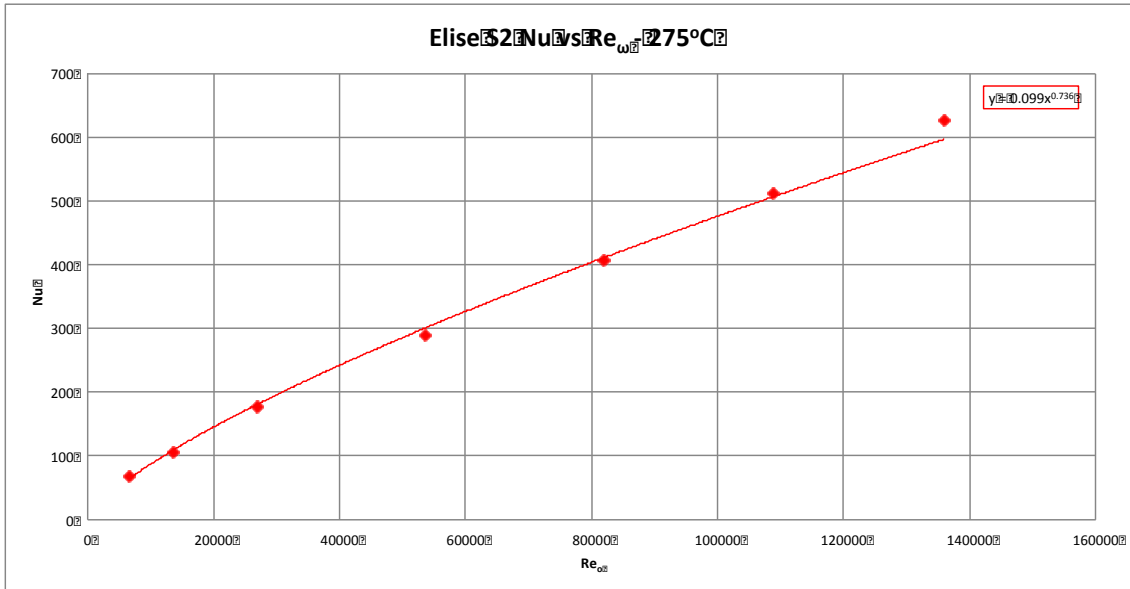


Figure 9-9 Nu vs Re_{ω} at 275°C – Elise S2

A summary of the (a) and (b) terms for both discs at the average disc surface temperatures and the general form of the equation describing the cooling in still air, are demonstrated in Table 9-4.

Table 9-4 Cooling Characterisation in Still Air

Temperature [°C]	Elise Version 3		Elise S2	
	<i>a term</i>	<i>b term</i>	<i>a term</i>	<i>b term</i>
125	0.052	0.78	0.053	0.77
175	0.063	0.77	0.064	0.76
225	0.071	0.76	0.089	0.74
275	0.070	0.76	0.099	0.74
325	0.071	0.76	0.140	0.71
375	0.072	0.76	0.160	0.70
General Form: $Nu = aRe_{\omega}^b$				

It needs to be mentioned that although the rotational speed in the tests reached 250 km/h, the relatively small outer diameter of the two discs did not allow for the flow to become fully turbulent at any point during the test (turbulent flow: $Re_{\omega} \geq 2.4 \times 10^5$), not even at low average disc surface temperatures. So, all the results are within the predominantly laminar flow region.

9.5 Step 10 – Cooling Tests in Air Cross Flow

Attention was given to position the discs in the middle of the Flow Channel and lock the caster wheels of the Disc Rotating and Heating Assembly in order to avoid any movement of the Assembly during the tests. Additionally, depending on whether the test schedule was planned to have the discs parallel to the cross flow or at an angle, the track rod position was set accordingly. Moreover, each of the six rubbing thermocouples was inspected to make sure that their contact with the disc was in order and that they gave reasonable readings. The final step in the process was to close all the hatches with the hatch doors and put insulating tape around them in order to avoid any air leaks. Furthermore, additional tape was used to cover the gap in the holes where the axle's shaft, the track rod and the Induction Heater's flexible hoses were entering the Flow Channel (Figure 9-10).



Figure 9-10 Sealing the Hatch Doors (Disc Rotating and Heating Assembly not in the Picture)

The test procedure followed in each test was similar to the procedure described in Section 8.3.1 with the addition of the frequency adjustment in the inverter that controls the fan of the Flow Channel in step 3.1, so as to achieve the desired air speed.

The test schedule envisaged for the tests in still air included a number of variables. The first was the rotational speed of the discs and its relation to the air velocity in the Channel. The starting set of tests was scheduled to have the rotational speed of the disc proportional (equal) to the air velocity in the Flow Channel. A total of five rotational speeds were selected corresponding to: 25, 31, 37.5, 44 and 50 km/h. Since the maximum air velocity achievable in the Flow Channel was 50 km/h, it was considered that the initial disc rotational speed should be half of the maximum air velocity and then have a gradual increase until it reached 50 km/h. Thereafter, in an effort to use the Nusselt and Reynolds numbers to create a map and characterise the cooling behaviour of the Elise Version 3 and the Elise S2, tests took place where the disc rotational speeds were 1.5 times higher than their equivalent air velocities in the Flow Channel. Subsequently, a set of tests took place with the disc rotational speeds being twice the air velocities in the Channel and finally, another set with the rotational speeds being three times compared to their equivalent air velocities.

During the aforementioned tests the discs were positioned parallel to the air flow. However, the influence of angle in the cooling of the discs was planned to be investigated and as of that, the tests were conducted at an angle $+10^\circ$ and also -10° in relation to the cross flow (Figure 9-11). This resulted in the total number of tests for both the discs to be 120 and the detailed test schedule is presented in Appendix V.

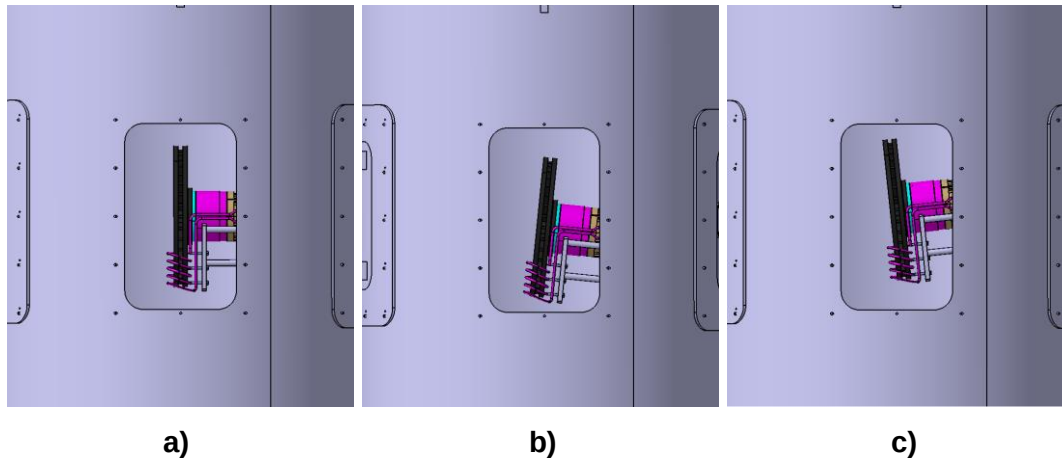


Figure 9-11 Top View of Disc Angles a) Parallel to the Flow, b) at +10°, c) at -10°

There is a number of parameters that influence the brake discs cooling behaviour. The dominant one with the most impact is the velocity of the air flow “hitting” the discs. However, in the current study this parameter was not studied because it is dependent on features like the suspension geometry, the dust shields, the under body design of the car and is not dependent on the design of the brake discs itself. The influence of other parameters was studied and, as explained above by the description of the test schedule, these parameters were the rotational speed of the discs and the angle at which the brake discs are positioned with regards to the airflow. In addition to these two parameters, a third parameter was identified when conducting the tests in still air, which was the average disc surface temperature. In the next Sections each of the three parameters are discussed separately. Initially, the influence that the individual parameter has on the cooling of each disc is considered and then how this parameter affects the two discs when compared to each other is discussed. Nonetheless, in order to have the complete picture of the cooling functioning of each of the discs and to identify which one has improved its performance, the combined effect of all three parameters must be studied. This will take place at the end of this Chapter.

9.5.1 The Influence of Disc Surface Temperature

In this Section the influence of the average disc surface temperature in the convective heat transfer coefficient will be discussed. In order to understand solely the impact that the surface temperature has, the results that will be

presented will be for rotational disc speeds that were proportional to their equivalent air velocities and with the discs positioned parallel to the air cross flow. As a result, the parameters of increased rotational speed and disc angle will not be examined here.

In Figure 9-12, the graph shows how the average disc surface temperature had influenced the convective heat transfer for the Elise Version 3. The calculated values for 125°C, 175°C, 225°C, 275°C and 325°C were pretty consistent and the application of power trend lines makes the results easier to be visualised. However, the values at the average temperature of 375°C were not as consistent as the others, something that was noticed in the groups of tests with increased rotational speeds and/or with discs at an angle to the flow. Similarly to what was noticed with the tests in still air, $\bar{h}_{Dconv}$ gradually tends to increase as the average surface temperature increases.

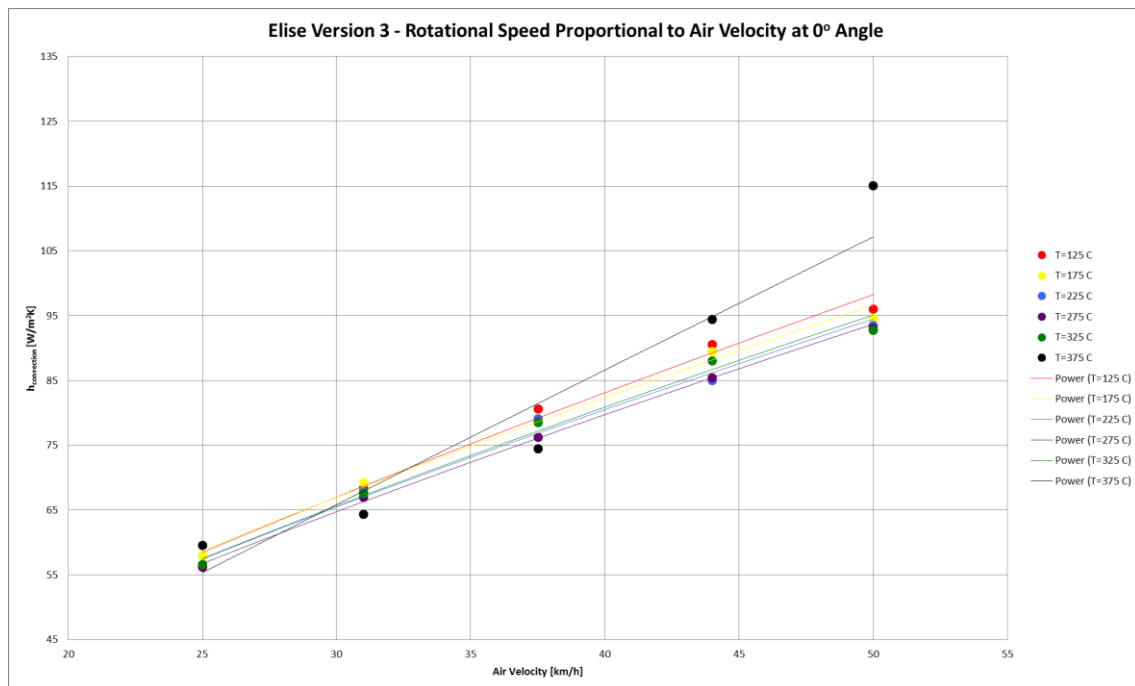


Figure 9-12 Influence of Average Disc Surface Temperature – Elise Version 3

With regards to the Elise S2 brake disc, the results are illustrated in Figure 9-13. The general trend is that the convective heat transfer coefficient increases with the increase of the disc average surface temperature, which was also noticed with the tests in still air. Nevertheless, it has to be mentioned that the convective

heat transfer coefficient values, for the average surface temperature of 375°C, tend to decrease with the increase of rotational speed and air velocity compared to the values of the other average surface temperatures.

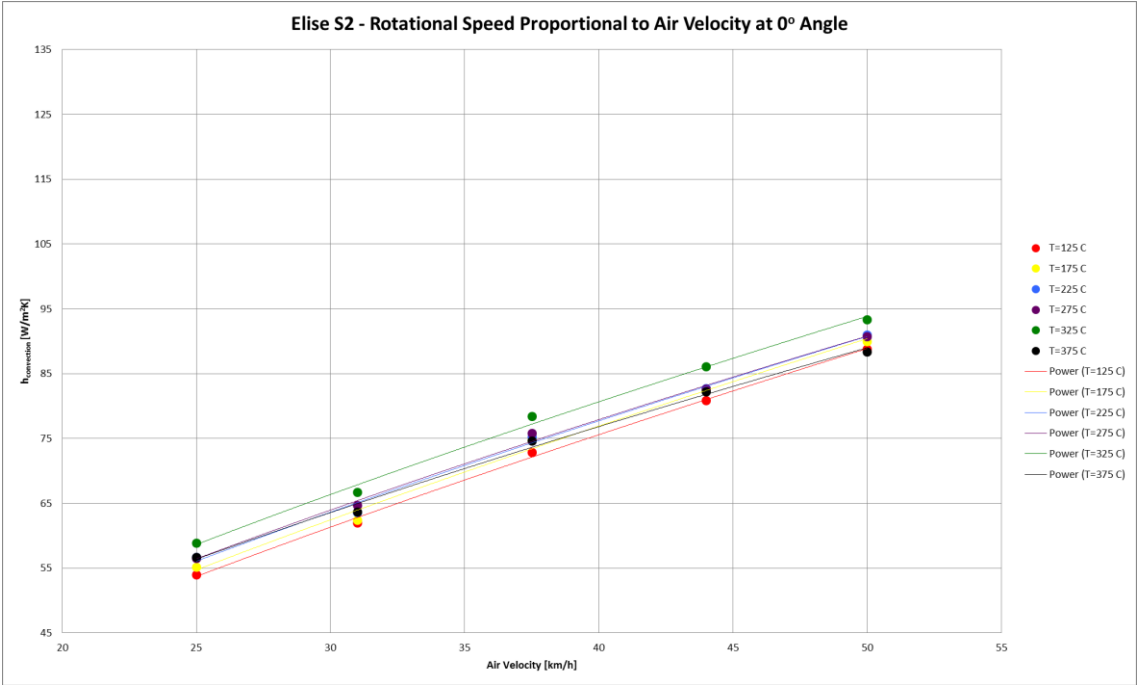


Figure 9-13 Influence of Average Disc Surface Temperature – Elise S2

If comparisons are to be made between the $\bar{h}_{Dconv}$ performance of the two discs based on the average disc surface temperatures, then it can be said that the Elise Version 3 features improved cooling compared to the standard Elise Disc. The graph in Figure 9-14 demonstrates the improvement in percentage for all the average disc surface temperatures.

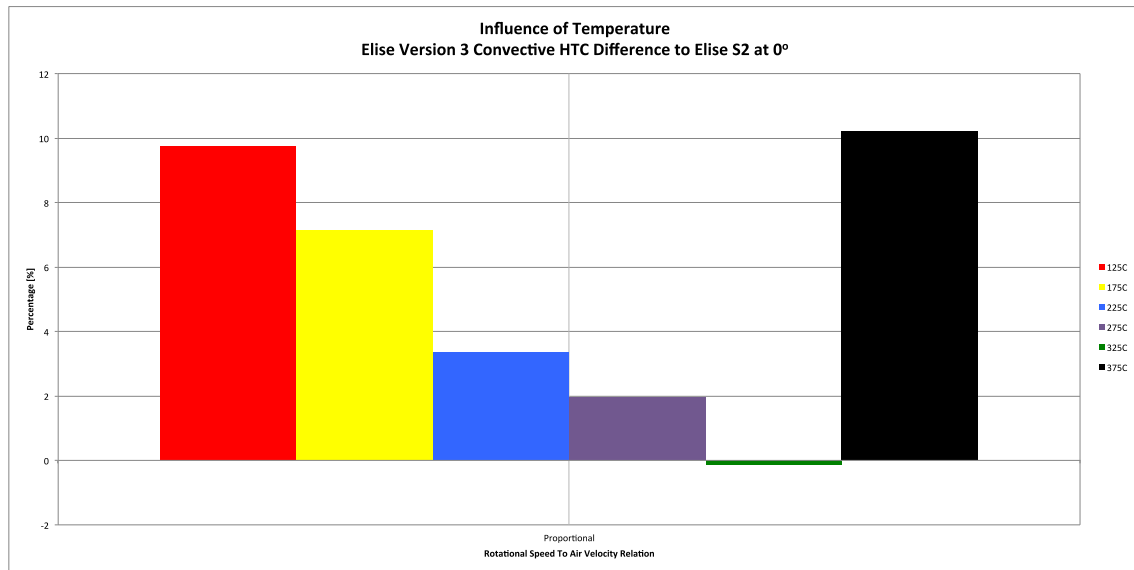


Figure 9-14 Performance Improvement of the Elise Version 3 Against the Elise S2

For the surface temperatures of 125°C, 175°C, 225°C and 275°C the Elise Version 3 has increased $\bar{h}_{Dconv}$ values ranging between 2% and almost 10%. Only at 325°C the Elise S2 has a marginally higher $\bar{h}_{Dconv}$ value for just 0.16%. At the average temperature of 375°C, the prototype Elise disc has convective coefficient values increased by more than 10% bringing the average percentage of improvement to 5.35%. Nonetheless, as it was mentioned before, the calculated values at 375°C for the Elise Version 3 were not consistent and can be neglected. Despite this, if the convective heat transfer coefficient results are not taken into account for the average disc surface temperature of 375°C, the prototype disc has an average increase in its cooling performance by 4.41% over the standard Elise disc.

9.5.2 The Influence of Disc Angle

When the brake disc is mounted on a car it is completely covered by the wheel and several components, such as the suspension parts or the dust shields, influence the direction of the air into the disc. As a result, the brake discs are rarely exposed to a clean air flow (unless they are axle mounted on purpose build vehicles) and in the vast majority of the cases the air is coming at an angle to cool them.

Since the Disc Rotating and Heating Assembly's design allowed the tested disc to be steered in the Flow Channel, this gave the opportunity to study the influence of the disc's position with regards to the air flow in the Flow Channel, and more specifically the impact of the discs' angle on $\bar{h}_{Dconv}$. Consequently, the tests were scheduled where both the Elise Version 3 and the Elise S2 were positioned parallel to the cross flow, at an angle of $+10^\circ$ and also at an angle of -10° . Similarly to when the influence of the average disc surface temperature was discussed, in order to isolate the effect of only the disc angle, the rotational speeds of the discs were proportional to the equivalent air velocities in this stage and the results were derived and studied in each of the six average disc surface temperatures, shown in Section 9.5.1, separately.

In Figure 9-15, the graph shows the influence of the disc angle when the average disc surface temperature of the discs was 275°C . For the sake of making the graph below easier to be understood, it has to be stated that all the markers and trend lines in blue colour describe the behaviour of the Elise Version 3 and all the markers and trend lines in red colour are with regards to the Elise S2. Furthermore, the markers that are of "X" style reference values of $\bar{h}_{Dconv}$ when the discs were parallel to the cross flow, the markers that are of star style reference values when the discs were at $+10^\circ$ angle, and finally the markers with a cross style reference values when the discs were positioned at -10° angle. Additionally, solid power trend lines are with regards to the discs being parallel to the air flow, the dashed power trend lines related the results when the discs were at $+10^\circ$ angle and the power trend lines with one dash and two dots related the results at -10° .

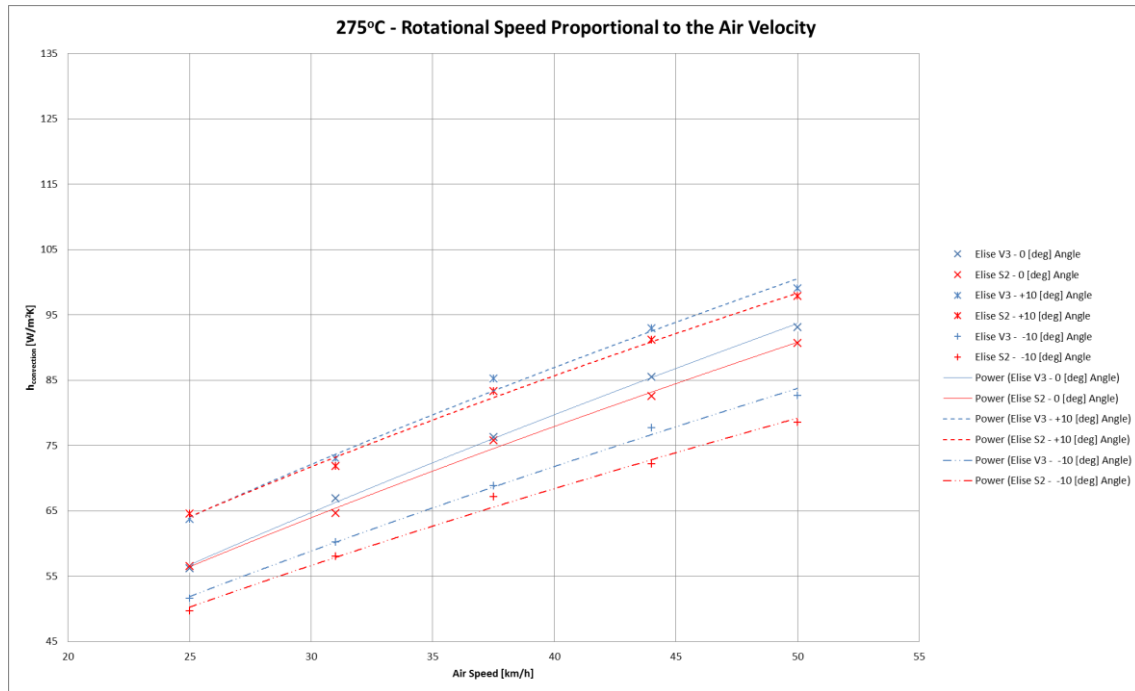


Figure 9-15 Influence of Disc Angle to Convective HTC

The graph in Figure 9-15 shows that the highest values of convective heat transfer coefficients occurred when the discs were positioned at $+10^\circ$ angle with regards to the air flow. When the discs were parallel to the cross flow, their cooling performance was lower than it was at $+10^\circ$ and higher than -10° . Subsequently, when the discs were positioned at -10° their $\bar{h}_{Dconv}$ were the lowest.

These results were expected because when the discs are at $+10^\circ$ their inboard sides are more exposed to the cross flow and swan neck discs, such as the Elise Version 3 and Elise S2 are designed to have the air entering from their inboard sides in order to cool them. As of this, the cold air had a clear path into the discs (graphically shown in Figure 9-16) which explains the high convective coefficient values. For the same reason, when the discs were at -10° to the cross flow, their cooling decreased considerably as the cold air did not have a clear path to enter the discs and had to go around them and in the process of going around, portion of the cold air was escaping resulting in less air entering the discs to cool them down (graphically indicated in Figure 9-17).

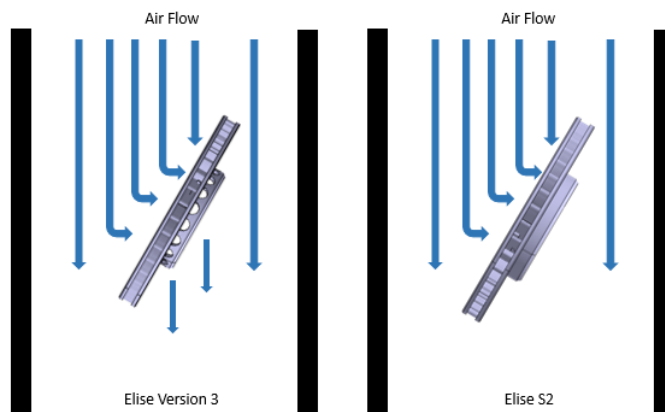


Figure 9-16 Discs at +10°

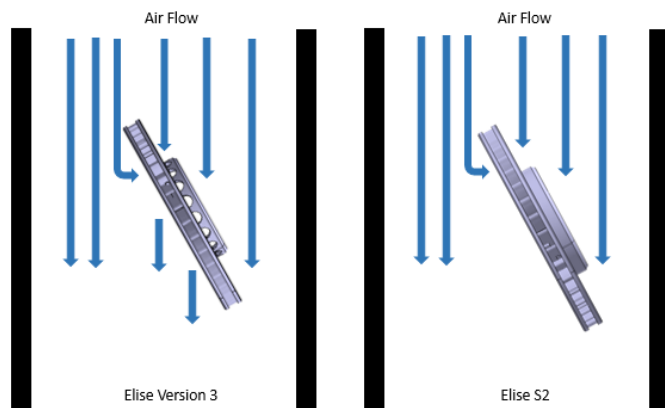


Figure 9-17 Discs at -10°

An interesting fact is that when the cooling performance of each disc at $\pm 10^\circ$ is compared to the performance of the same disc when it was positioned parallel to the air flow, the performance gain or loss is close. As a result, the Elise S2 at $+10^\circ$ gains slightly more when compared to the Elise S2 parallel to the flow than the Elise Version 3's gain if compared to the Elise Version 3 parallel to the flow, because the standard Elise rotor operates marginally better than the prototype disc. On the other hand, the Elise Version 3 loss in performance when positioned at -10° compared to its performance when positioned parallel to the flow is less than the equivalent loss of the Elise S2. This is indicated by the graph in Figure 9-18.

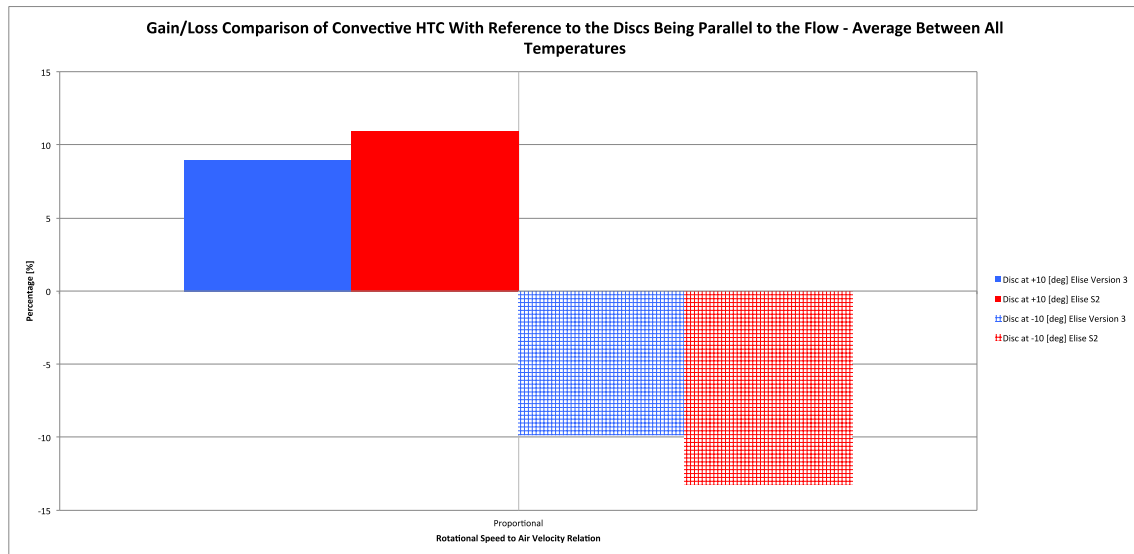


Figure 9-18 Gain/Loss Comparison of Both Discs Compared to Being Positioned Parallel to the Air Flow

Nevertheless, when the two discs are compared against each other, the Elise Version 3 performs better than the standard disc in all the positions relative to the cross flow. When the discs were positioned at $+10^\circ$ and parallel to the flow, the cooling of the two discs was similar at low rotational speeds/air velocities, whilst as the rotational speeds/air velocities were increasing, the cooling performance of the Elise Version 3 was gradually increasing compared to the Elise S2. On average, for the disc surface temperature at 275°C the cooling was 1.96% better for the prototype disc when both discs were parallel to the flow and 1.16% when the discs were at $+10^\circ$ angle. The difference in performance was noticeably greater in favour of the Elise Version 3 when the discs were positioned at -10° . The average improvement over the different rotational speeds/air velocities was 4.61%.

The remaining five average surface temperatures follow the same pattern as what was described for the average disc temperature of 275°C , as depicted in Figure 9-19. The Elise Version 3 improved on average more than 5%, 3% and 9% when the discs were positioned parallel, at $+10^\circ$ and -10° to the cross flow respectively.

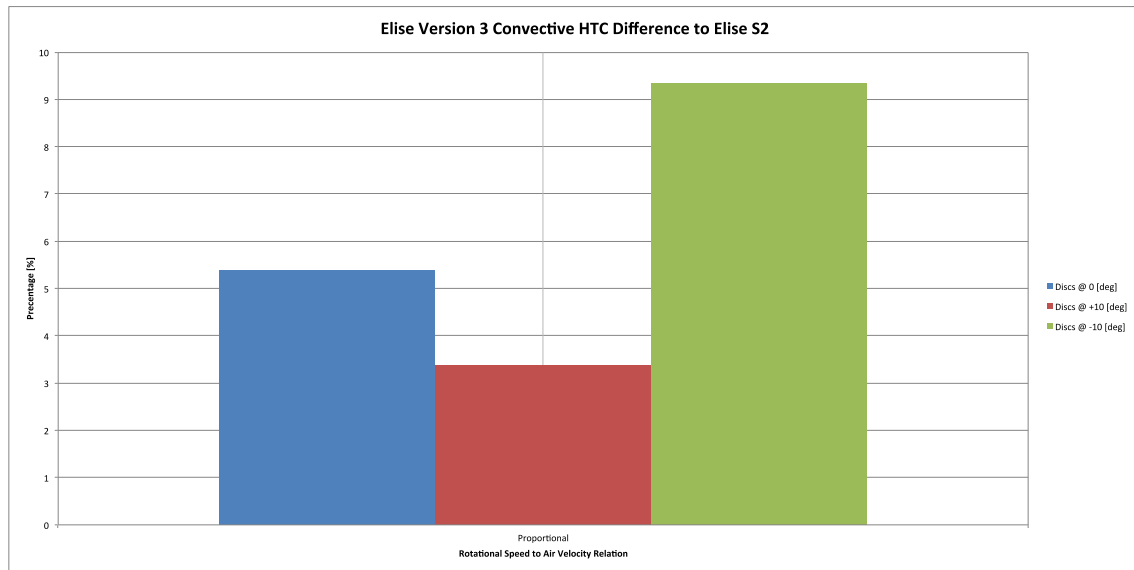


Figure 9-19 Elise Version 3 Gain at Different Angles

It is believed that the difference lays on what was explained in Section 9.4.1 of the tests in still air, and more specifically what was graphically shown in Figure 9-5. Because of the holes in the top hat of the prototype rotor, air is forced into the disc by mostly the inboard side, and some from the outboard side. So when the discs were positioned parallel to the flow, the Elise Version 3 was able to attract more air to cool down than the Elise S2. When the discs were positioned at $+10^\circ$, the Elise Version 3 lost some of its advantage because a) the path to the inboard side of both discs did not have any obstacles and b) some of the air was escaping from the holes of the prototype rotor and did not enter its vanes. Thereafter, the advantage of the Elise Version 3 was regained and increased when the discs were positioned at -10° with regards to the cross flow. Since the inboard sides of the discs were blocked for the main portion of the air flow close to the discs because of their relative position to it, the Elise Version 3 was benefited from its ability to force additional air from its outboard side and feature higher convective heat transfer coefficients.

Supporting graphs with the impact of the disc angle for all the average disc surface temperatures and increasing disc rotational speeds to air velocities can be found in Appendix V.1. In addition, graphs illustrating the gains or losses due to the disc angles are shown in Appendix V.2.

9.5.3 Influence of Disc Rotational Speed

Similarly to when the parameters of the average disc surface temperature and the disc angle were investigated in Sections 9.5.1 and 9.5.2, the results in this Section will exclude the impact of the two previous parameters and will focus only on the disc rotational speed parameter and its impact on the convective heat transfer coefficient. Subsequently, the values of $\bar{h}_{Dconv}$ presented here will be at 275°C average disc surface temperature and for the discs positioned parallel to the air cross flow.

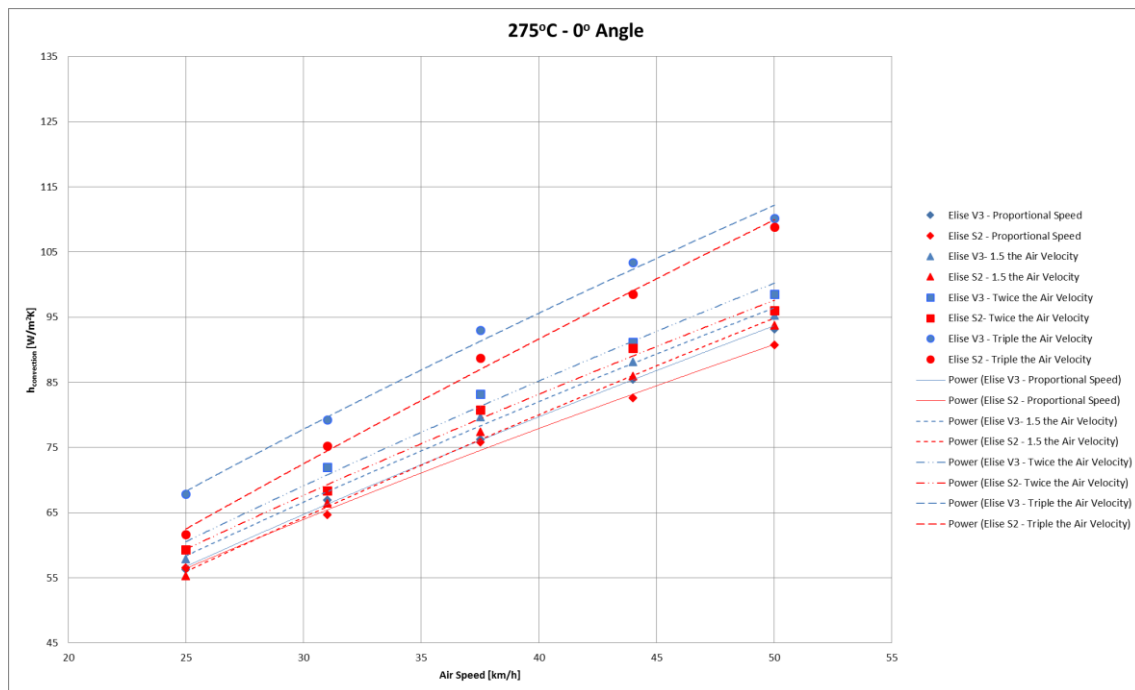


Figure 9-20 Influence of Disc Rotational Speed

The graph in Figure 9-20 shows how the increase of the disc rotational speed influences the convective heat transfer coefficient for the two tested brake discs. For clarification purposes, the markers and their trend lines in Figure 9-20 will be explained. The blue markers and blue power trend lines are for the Elise Version 3 disc and the red markers and red power trend lines are for the Elise S2 rotor. The diamond markers and the solid trend lines show the results when the discs' rotational speeds were proportional (equal) to the air velocities in the Flow Channel. The triangle markers and the dashed power trend lines represent the results when the disc rotational speeds were 1.5 times faster than their equivalent

air velocities. The square markers and the trend lines with a long dash followed by two dots are with regards to the $\bar{h}_{Dconv}$ values when the disc rotational speeds were twice the air velocities. Lastly, the circular markers and the long dashed lines signify the results when the disc rotational speeds were triple the air velocities.

By examining the graph in Figure 9-20 and the graphs for all the other average surface temperatures, the first main observation was the existence of a trend for the values of $\bar{h}_{Dconv}$ to gradually increase with the increase of the rotational speeds. The cases where this trend did not occur were very few for the different average disc surface temperatures and/or for the different disc angles (see Appendices W.1, W.2 and W.3). This increase is depicted in Figure 9-21 for the average disc surface temperature of 275°C.

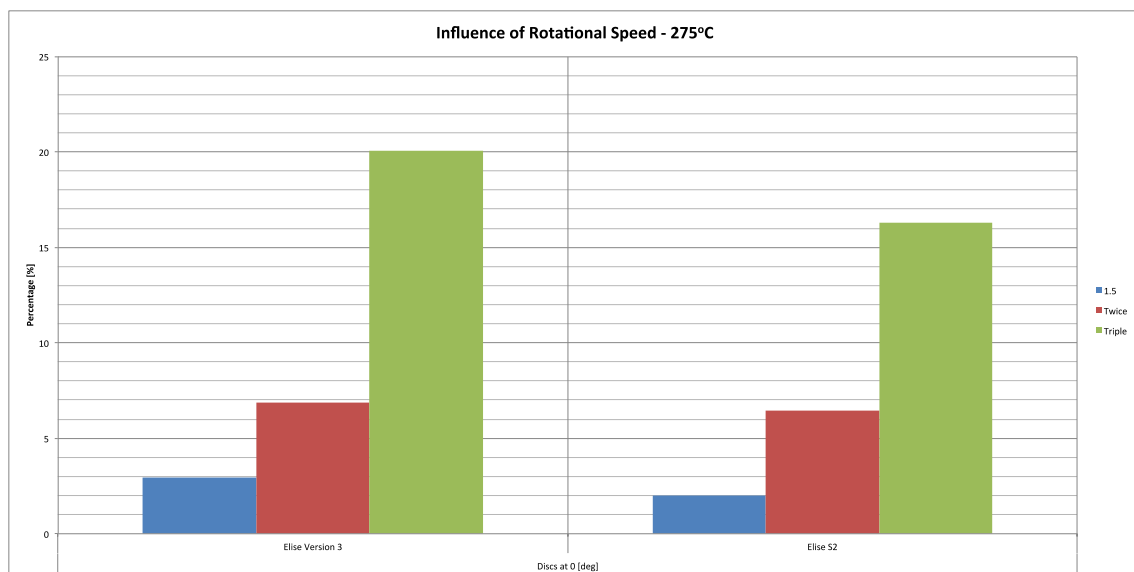


Figure 9-21 Comparison of the Increased Rotational Speeds to the Proportional Speed

The second main observation was that the Elise Version 3 featured an advantage over the standard Elise disc. The graph in Figure 9-22 illustrates this advantage for the different test cases and it can be said that overall, the convective heat transfer coefficient values calculated for the Elise Version 3 were approximately 6% higher than those of the Elise S2 when both discs were positioned parallel to the air flow.

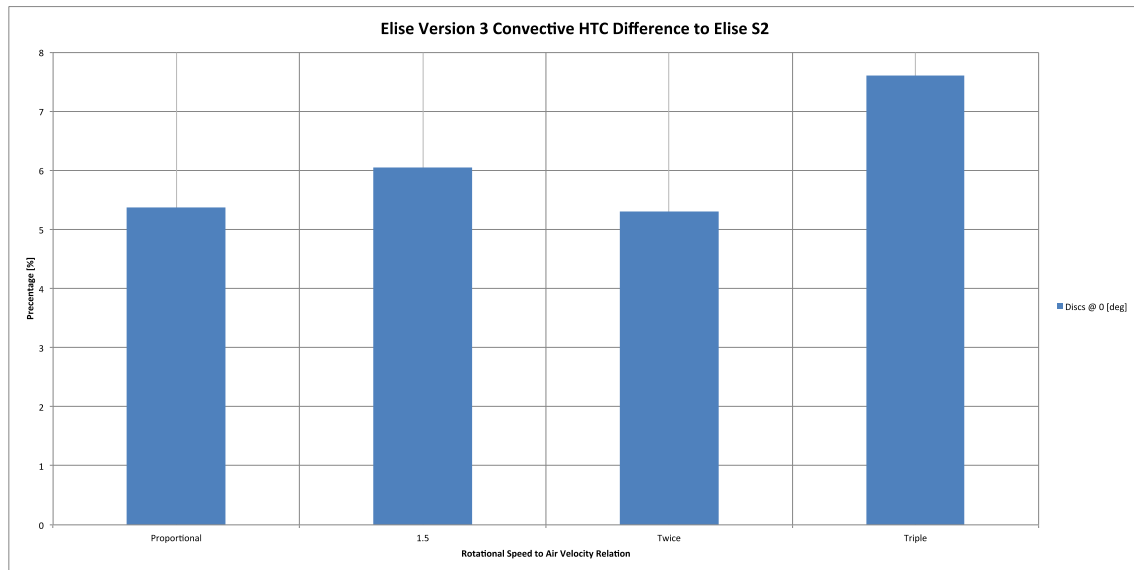


Figure 9-22 Percentage of Convective HTC Improvement of the Elise Version 3 to the Elise S2 with the Increase of Disc Rotational Speed at 0° Angle

9.5.4 Influence of all the Parameters Combined

As mentioned in Section 9.5, in order to draw conclusions on the two discs cooling behaviour and how they compare against each other, the combined influence of the three parameters, that were studied individually in Sections 9.5.1, 9.5.2 and 9.5.3, has to be investigated.

In Figure 9-23 the improved values of $\bar{h}_{Dconv}$ of the Elise Version 3 over the Elise S2 are illustrated from the perspective of the different average disc surface temperatures in conjunction with the different disc rotational speeds and for the average of the three different disc positions with regards to the cross flow. The graph is a combination of the graphs found in Appendix X.

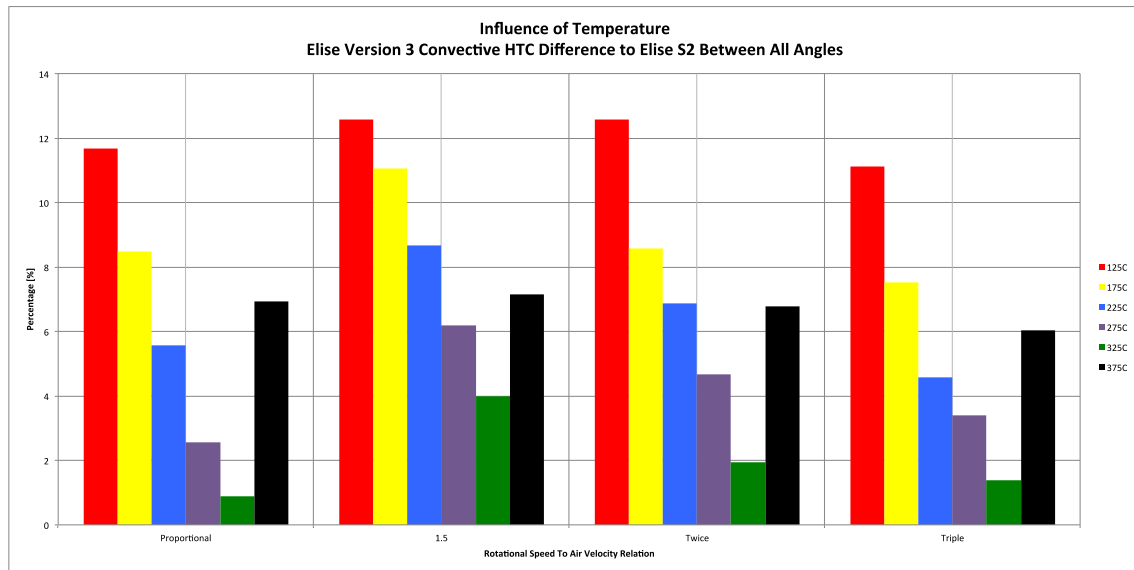


Figure 9-23 Elise Version 3 Performance Comparison Over the Elise S2 – All Parameters (Emphasis on Average Surface Temperature)

It can be seen that for all the different average surface temperatures, the prototype disc cools better than the standard Elise disc. Particular improvement is achieved for disc rotational speeds of 1.5 and twice the air velocities, which are the most common operating conditions for the brake rotors when mounted on vehicles. Overall, and if the effect of the 375°C average surface temperature is neglected due to inconsistencies, the Elise Version 3 features convective heat transfer coefficient values increased by around 6.5% compared to the Elise S2.

From the perspective of the different angles to the cross flow, the graph in Figure 9-24 shows how each disc has benefited from being positioned at +10° and how much its cooling was decreased when positioned at -10° as opposed to being parallel to the air flow. It can be concluded that at +10° angle, the Elise S2 generally improves more compared to the Elise Version 3; however, in the important case of the disc rotational speed being twice the air velocity, the prototype disc edges the standard disc. Conversely, when the discs were positioned at -10° the performance losses of the Elise Version 3 are far lower than those of the Elise S2, which results in better cooling for the prototype disc.

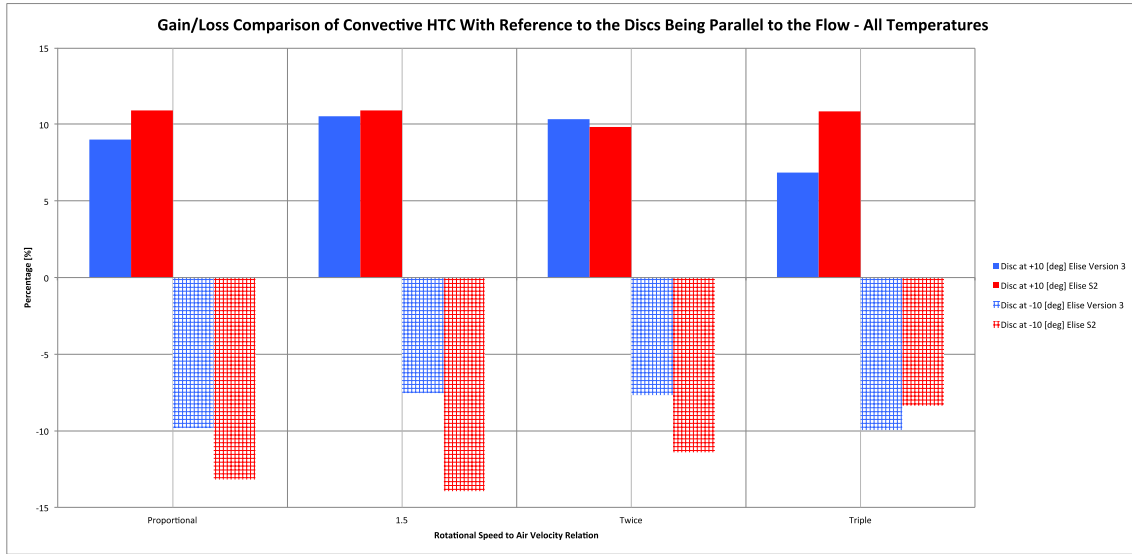


Figure 9-24 Gain/Loss in Performance for Each Tested Disc With Regards to Their Position Against the Cross Flow Between all Surface Temperatures

Finally, the perspective of the different disc rotational velocities and their impact on the cooling of each of the discs studied in this project is illustrated in Figure 9-25. As already mentioned, the greatest improvement in the values of $\bar{h}_{D_{conv}}$ was noticed when the disc rotational speed was triple the air speed.

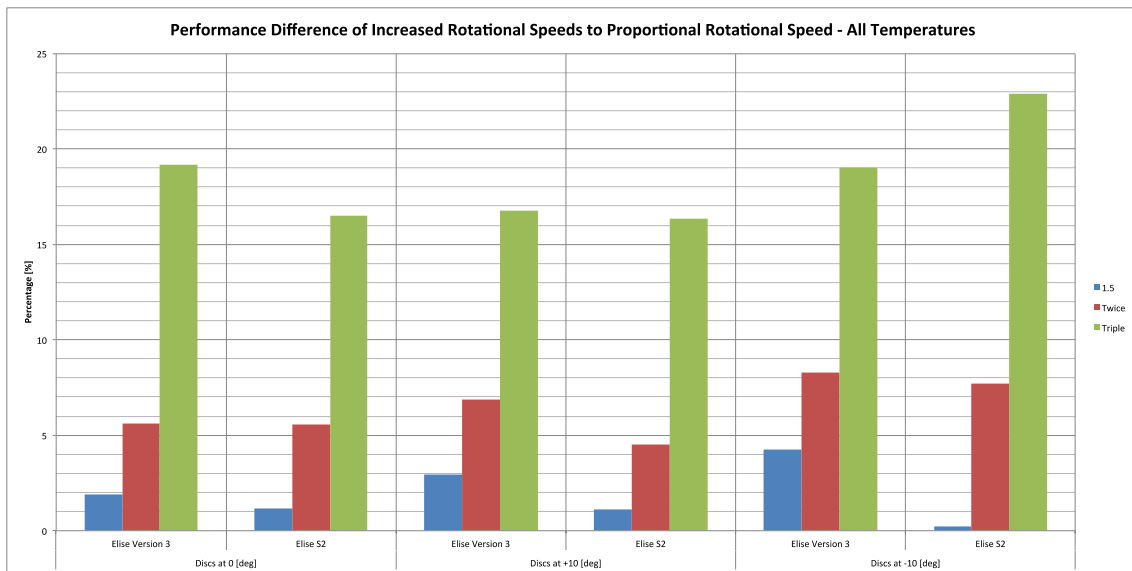


Figure 9-25 Convective HTC Improvement from the Increasing Disc Rotational Speeds

For the final comparison between the Elise Version 3 and the Elise S2, the results from the graphs in Figure 9-24 and Figure 9-25 can be combined resulting to the graph in Figure 9-26. The latter illustrates the improvement of the prototype disc to the standard Elise one, from the perspective of the disc rotational speeds at each of the three angles with the results of the six disc surface temperatures being averaged.

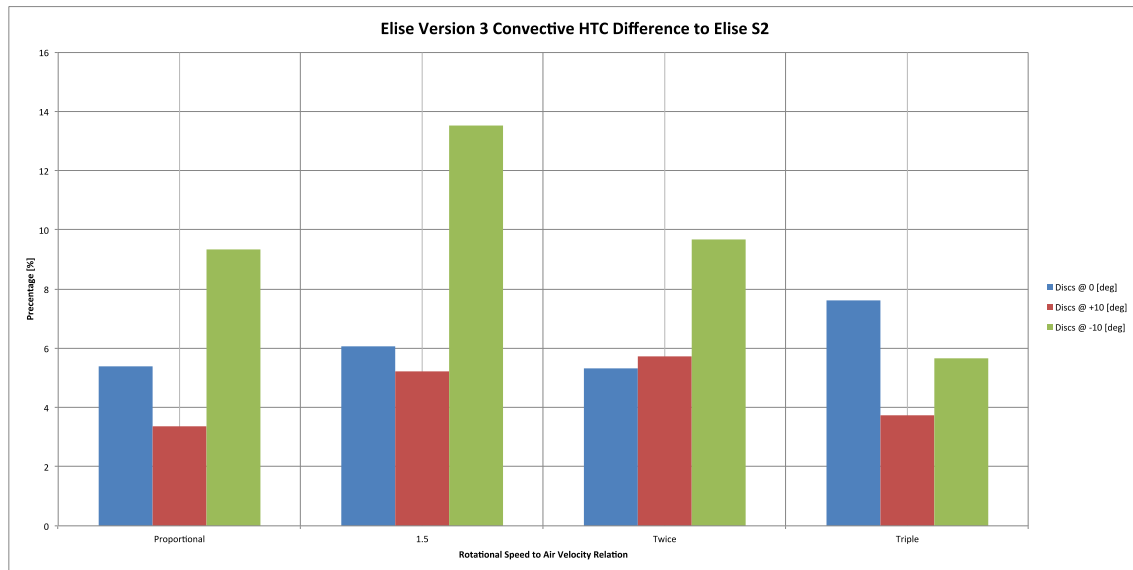


Figure 9-26 Elise Version 3 Improvement Over the Elise S2

In all the cases, the Elise Version 3's convective heat transfer coefficients are higher with the improvement being particularly increased when the rotational speeds were 1.5 and twice the air velocities. The overall improvement is estimated at slightly higher than 6.5%, which verifies what was concluded from studying the graph in Figure 9-23. This percentage is believed to be the average performance difference between the two discs in favour of the prototype Elise Version 3.

9.5.5 Cooling Behaviour Characterisation in Air Cross Flow

The characterisation of the two discs in cross flow was conducted in a similar way to when the discs were cooling in still air. However, compared to the cooling in still air, there was an additional parameter for the tests in cross flow and this was the ratio of the two Reynolds number, the number for the cross flow over the rotational number. As a result of having three parameters, three-dimensional

graphs were used to plot the relation between the dimensionless numbers as shown in Figure 9-27.

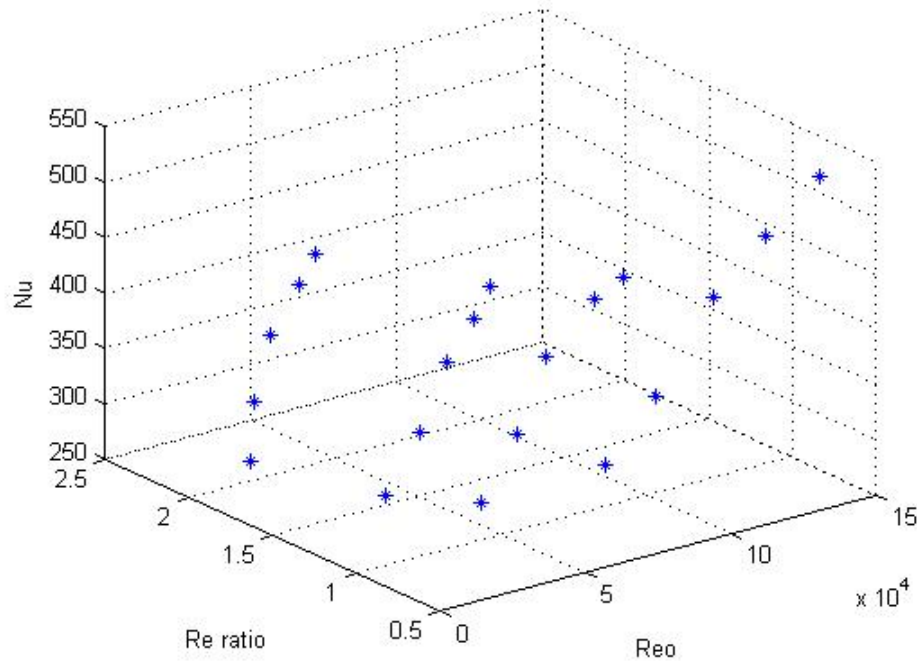


Figure 9-27 3D-Graph with the Relation of Nu, Re_t/Re_w and Re_w

The graph above represents the results for the Elise Version 3 when the disc was positioned parallel to the air flow and at 125°C average disc surface temperature. The points follow four lines which are a result of the four different relations between the disc rotational speeds and the air velocities.

The best fitting surface for the twenty points was found by using the Matlab software. In addition, the fitting surface was programmed to follow equation (9-9), which was the equation for characterising the disc cooling behaviour in cross flow. The surface that was fitted to the graph of Figure 9-27 is portrayed in Figure 9-28.

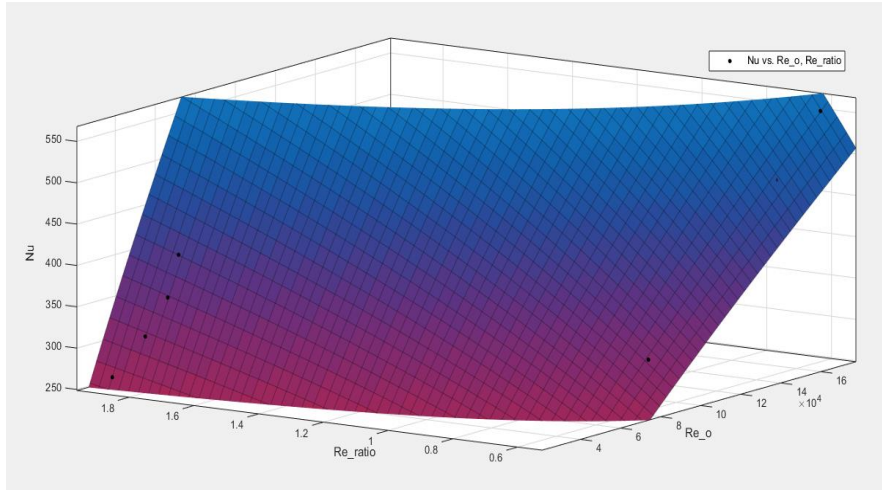


Figure 9-28 Fitting Surface

Graphs with the relations between Nu , Re_ω , and $\frac{Re_t}{Re_\omega}$ were plotted for all the cases tested and fitting surfaces similar to the one in Figure 9-28 and were used in each case in order to derive the a , b and c terms of equation (9-9). In Table 9-5, a summary of the terms for the Elise Version 3, when positioned parallel to the cross flow, is presented.

Table 9-5 Cooling Characterisation Elise Version 3, Parallel to the Flow

Angle [deg]	Temperature [°C]	<i>a term</i>	<i>b term</i>	<i>c term</i>
0	125	0.106	0.74	0.55
	175	0.106	0.74	0.56
	225	0.117	0.73	0.56
	275	0.121	0.73	0.56
	325	0.127	0.73	0.56
	375	0.044	0.83	0.72
General Form: $Nu = aRe_\omega^b \left[\frac{Re_t}{Re_\omega} \right]^c$				

The results in Table 9-5 indicate a particular consistency for the terms a , b and c for all the average disc surface temperatures apart from the one at 375°C. This is linked to the inconsistent $\bar{h}_{Dconv}$ values that were observed in Section 9.5.1 so it is better to neglect the results in this particular average disc surface temperature. The values of term a gradually increase with temperature, whilst terms b and c remain almost unchanged.

The terms, when the Elise Version 3 was positioned at +10° and -10° with regards to the cross flow, are presented in Table 9-6 and Table 9-7 respectively.

Table 9-6 Cooling Characterisation Elise Version 3 at +10° to the Flow

Angle [deg]	Temperature [°C]	a term	b term	c term
+10	125	0.146	0.71	0.56
	175	0.186	0.70	0.55
	225	0.160	0.71	0.58
	275	0.240	0.67	0.52
	325	0.244	0.67	0.54
	375	0.128	0.74	0.56
General Form: $Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c$				

The figures of Table 9-6 indicate that with the increase of temperature, term a gradually increases contrary to term b that gradually decreases and term c that stays constant. Again, the values at 375°C are inconsistent and they should not be included in the comparison; however, they are illustrated for completeness.

Table 9-7 Cooling Characterisation Elise Version 3 at -10° to the Flow

Angle [deg]	Temperature [°C]	<i>a term</i>	<i>b term</i>	<i>c term</i>
-10	125	0.110	0.72	0.54
	175	0.124	0.72	0.54
	225	0.135	0.71	0.55
	275	0.175	0.69	0.51
	325	0.217	0.67	0.50
	375	0.145	0.70	0.60
General Form: $Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c$				

The final table for the Elise Version 3 prototype disc, shows the values of term a increasing with the average disc surface temperature and the values of terms b and c gradually decreasing. Yet again, the results at 375°C average surface temperature do not follow the trends set by the results at 125°C, 175°C, 225°C, 275°C and 325°C.

The values derived for terms a , b and c , when the Elise S2 disc was tested, are demonstrated in Table 9-8, when the disc was positioned parallel to the cross flow, Table 9-9 when the disc was positioned parallel at +10° to the cross flow, and finally in Table 9-10 when the disc was positioned parallel at -10°.

Table 9-8 Cooling Characterisation Elise S2, Parallel to the Flow

Angle [deg]	Temperature [°C]	<i>a term</i>	<i>b term</i>	<i>c term</i>
0	125	0.036	0.82	0.65
	175	0.054	0.79	0.62
	225	0.065	0.78	0.63
	275	0.081	0.76	0.61
	325	0.102	0.75	0.61
	375	0.141	0.71	0.58
General Form: $Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c$				

Table 9-9 Cooling Characterisation Elise S2 at +10° to the Flow

Angle [deg]	Temperature [°C]	<i>a term</i>	<i>b term</i>	<i>c term</i>
+10	125	0.094	0.74	0.58
	175	0.140	0.73	0.56
	225	0.148	0.71	0.56
	275	0.207	0.69	0.53
	325	0.236	0.68	0.53
	375	0.287	0.66	0.53
General Form: $Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c$				

Table 9-10 Cooling Characterisation Elise S2 at -10° to the Flow

Angle [deg]	Temperature [°C]	<i>a term</i>	<i>b term</i>	<i>c term</i>
-10	125	0.097	0.73	0.51
	175	0.067	0.76	0.54
	225	0.103	0.73	0.51
	275	0.123	0.71	0.51
	325	0.159	0.69	0.49
	375	0.170	0.68	0.49
General Form: $Nu = aRe_{\omega}^b \left[\frac{Re_t}{Re_{\omega}} \right]^c$				

By reviewing the results from all three tables, for the Elise S2 disc, it can be said that the values of term *a* increase with the increase of disc surface temperature; whereas, the values of term *b* and term *c* decrease. Furthermore, by evaluating all the results presented in the six tables of this Section, it can be concluded that term *a* has an inverse relation to the other two terms. When it increases, the other two decrease and vice versa.

9.6 Chapter Summary

In this Chapter, both the Elise Version 3 and the Elise S2 discs were extensively tested with regards to their cooling performance. Tests were conducted in still air and in air cross flow under various conditions. In total, more than 127 tests were conducted and some of them had to be repeated in order to ensure accuracy. The general conclusion drawn was that the prototype brake disc with the fingered hub (Elise Version 3) featured improved cooling performance against the standard production rotor in practically all test conditions. In still air, the calculated convective heat transfer coefficient was on average around 23% higher for the Elise Version 3 and in the air cross flow it was improved by 6.5%. In particular,

improvement at low speeds and low average disc temperatures was noticed, aspect that makes the Elise Version 3 ideal for EVs and HEVs.

10 CONCLUSIONS AND FUTURE WORK

10.1 Conclusions

The developed methodology was validated through the analyses, manufacture and testing of the Elise Version 3 prototype disc. A total number of ten steps were followed with each step investigating a different attribute of the proposed design.

Initially, by using FE analyses, it was proven that the proposed concept is feasible with regards to mechanical strength. The predicted stress values were within acceptable limits. It was also understood that the criterion under which the disc is most likely to fail mechanically is the Maximum Principal stress.

The thermomechanical FE analyses that took place indicated that the Elise Version 3 performed similarly to the standard Elise S2 disc under mechanical loading. The thermal stresses were slightly higher for the conceptual design fingered disc, whereas the maximum disc temperature during braking was identical for the two disc. The disc coning though, was predicted to be considerably improved in the fingered disc.

The experimental work that focused on the mechanical loading, validated the findings of the numerical simulations. It was also demonstrated that the actual limit of the design is much higher than initially anticipated based on cast iron's properties. It was found that the Elise Version 3 disc could withstand two and a half times the maximum braking torque that the vehicle can develop.

The conducted structural optimisation yielded favourable results for the Elise Version 3. With the shape optimisation, a reduction of 32% in the Maximum Principal stresses was achieved that resulted in the safety factor increasing by one unit. The topology optimisation achieved some mass reduction proving the correctness of the starting conceptual design. The total mass saving was just under 10% compared to the original disc (Elise S2).

The newly developed Brake Disc Cooling facility was created to test the cooling performance and the cooling characteristics of the fingered disc prototype. The installation, is not only limited to the study of the Elise discs, but it is suitable for

the study of a broad range of brake discs. The conducted commissioning tests proved that the Rig's systems and sub-systems operated according to the design intent and also to the quoted specifications. Overall, the Brake Disc Cooling Rig offered good stable airflow and disc rotational speeds with highly repeatable test results.

The cooling tests, performed in both the Elise Version 3 and the Elise S2, showed that the prototype fingered disc features improved cooling under all test conditions. This is especially true for the important disc rotating speeds that are 1.5 and two times higher than their equivalent cross flow velocities. Overall, in still air the Elise Version 3 offers substantially better cooling of about 23%, whereas in air cross flow the gains were at approximately 6.5%.

Taking into consideration all the above, it can be concluded that with the Elise Version 3, the methodology developed satisfies all the aims at a proof of concept level for the disc analysed.

10.2 Recommendations for Future Work

Areas of expansion for the present study have been identified after the completion of the performed work. In order to fully validate the design, they are as per below:

- Conduct tests with the Elise Version 3 prototype in a brake dynamometer that combines mechanical, thermal loading and NVH characteristics,
- Casting the fully optimised design produced in Chapter 6 and repeat the brake dynamometer testing,
- Perform cooling tests with the fully optimised disc of Chapter 6 and compare its behaviour against the Elise Version 3,
- Create CFD models of the entire suspension assembly and the corner quarter vehicle in order to maximise the advantages of the fingered disc within the vehicle installation.
- Finally, conduct vehicle tests to validate the actual cooling performance behaviour and NVH characteristics of the design.

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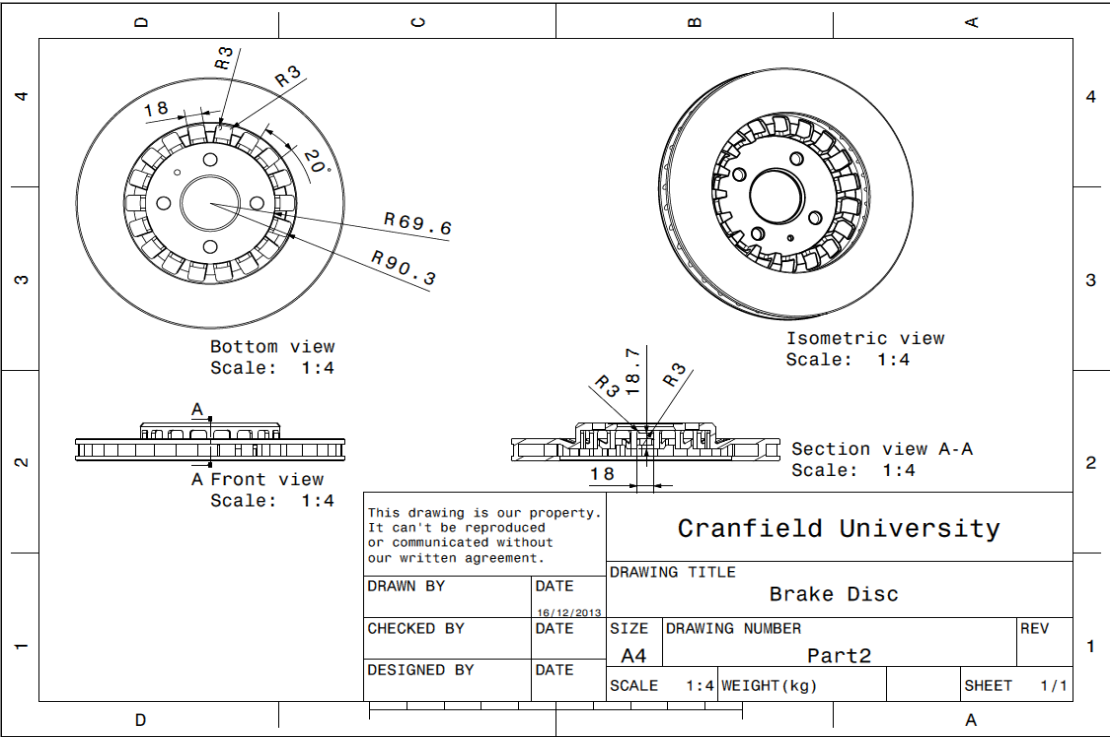
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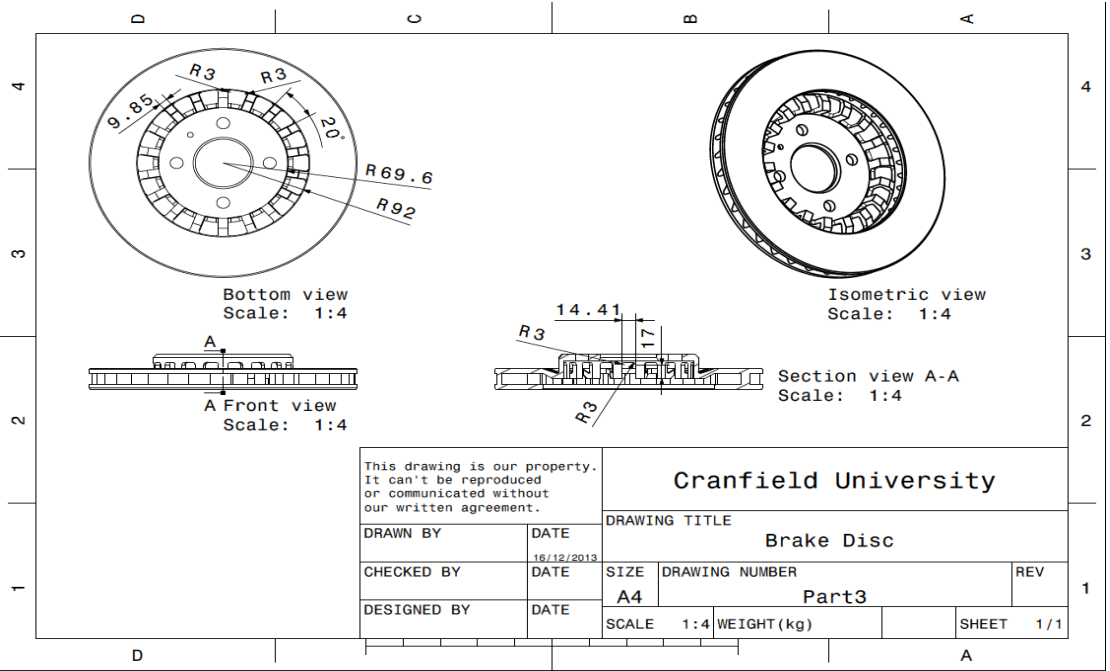
APPENDICES

Appendix A – Elise Disc Conceptual Designs

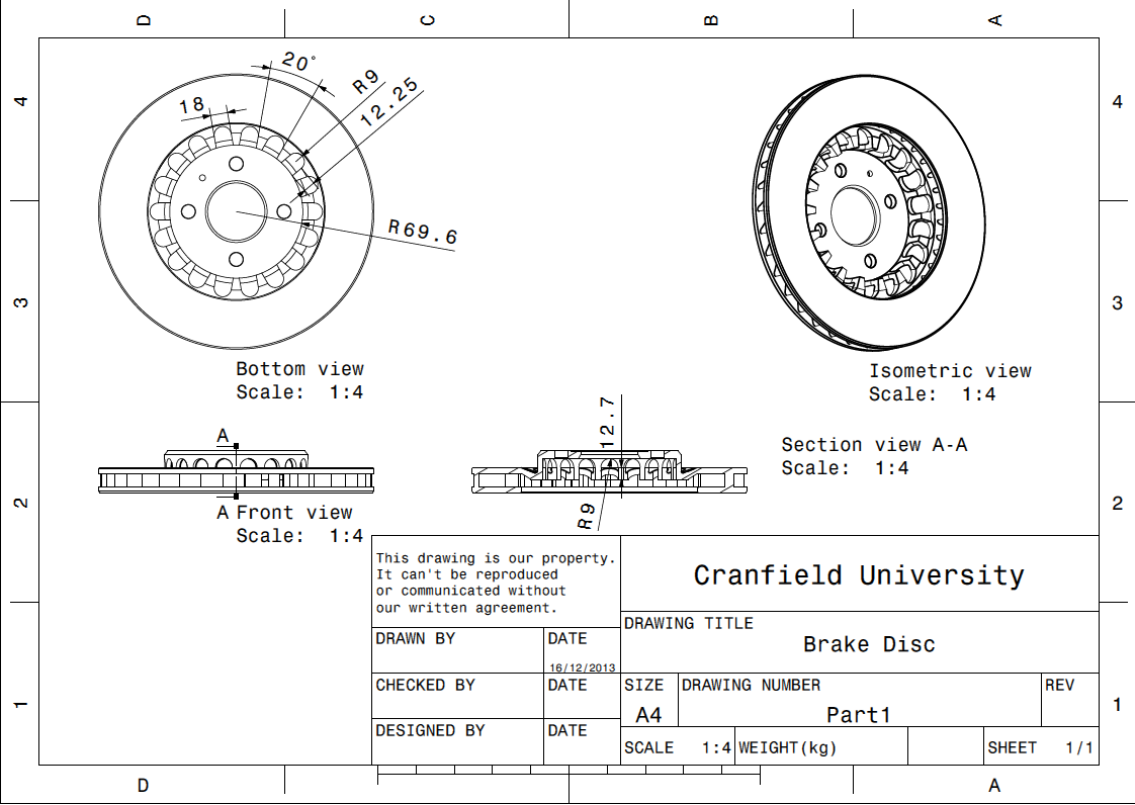
A.1 Elise Version 1



A.2 Elise Version 2



A.3 Elise Version 3



Appendix B – Linear Elastic Material Properties

B.1 Cast Iron

Property	Value
Density	7200 [kg/m ³]
Young's Modulus	116 [GPa]
Poisson's Ratio	0.25

B.2 Steel Backplate

Property	Value
Density	7800 [kg/m ³]
Young's Modulus	210 [GPa]
Poisson's Ratio	0.29

B.3 Friction Material

Property	Value
Density	2620 [kg/m ³]
Young's Modulus	1.25 [GPa]
Poisson's Ratio	0.37

Appendix C – Hydraulic Pressure Calculations

Piston radius (facing hydraulic fluid): $r_M = 0.02 \text{ m}$

Piston area (facing hydraulic fluid): $A = \pi \times r^2 = 0.0012 \text{ m}^2$

Force applied to the pad by the piston: $F = A \times P = 12566.37 \text{ N}$

Piston outer radius (facing pad): $r_{M1} = 0.02 \text{ m}$

Piston inner radius (facing pad): $r_{M2} = 0.0015 \text{ m}$

Piston total area (facing pad): $A_t = (\pi \times r_{M1}^2) - (\pi \times r_{M2}^2) = 0.0005 \text{ m}^2$

Pressure on the pad: $P = \frac{F}{A_t} = 22860415 \frac{\text{N}}{\text{m}^2} \rightarrow 228 \text{ bar}$

Appendix D – Lotus Elise Braking Torque Calculations

$$\text{Max deceleration: } \alpha = \varphi \times g = 0.8 \times 9.81 = 7.85 \text{ m/s}^2$$

$$\text{Braking Force: } F_B = X_B = m \times \alpha = 8.9 \text{ kN}$$

$$\text{Kinetic Energy: } E_k = \frac{1}{2} \times m \times V^2 = 1.8 \text{ MJ}$$

$$\text{Vertical Reaction Forces: } Z_1 = \frac{(\alpha h + gb)m}{l} = 7.3 \text{ kN}$$

$$Z_2 = mg - Z_1 = 3.8 \text{ kN}$$

$$\text{Braking Forces per Axle: } X_{B1} = \varphi \times Z_1 = 5.8 \text{ kN}$$

$$X_{B2} = \varphi \times Z_2 = 3 \text{ kN}$$

$$\text{Braking Forces per Wheel: } X_{BW1} = \frac{X_{B1}}{2} = 2.9 \text{ kN}$$

$$X_{BW2} = \frac{X_{B2}}{2} = 1.5 \text{ kN}$$

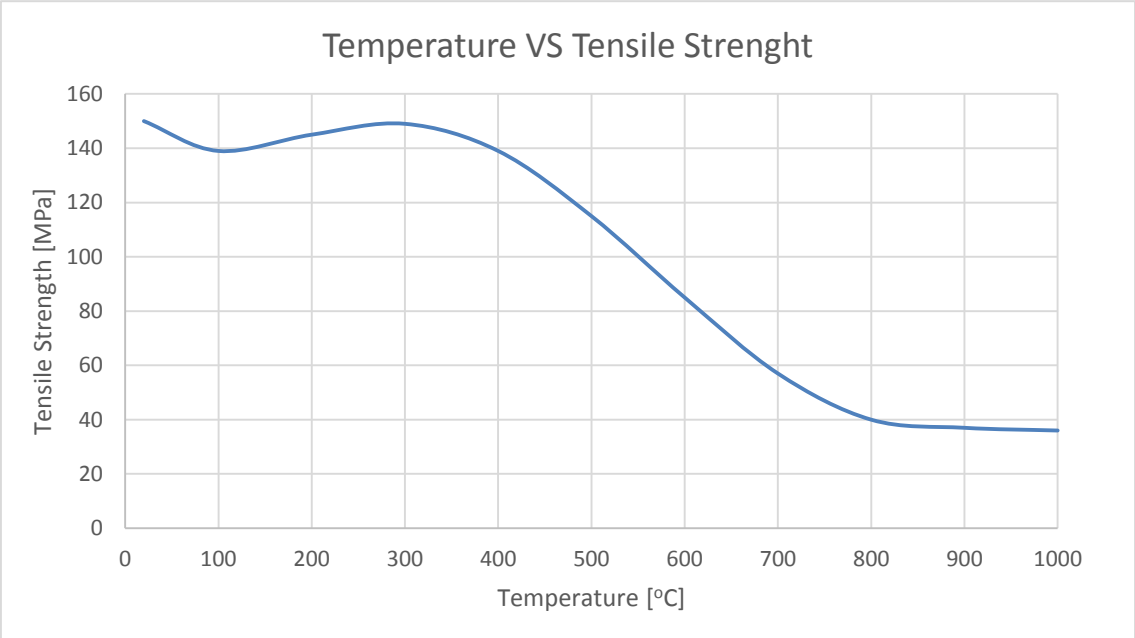
$$\text{Braking Torque per Wheel: } T_{B1} = X_{BW1} \times rd = \mathbf{882 \text{ Nm}}$$

$$\text{Mean Swept Radius of Disc: } r\mu = \frac{(OD + ID)}{4} = 0.118 \text{ m}$$

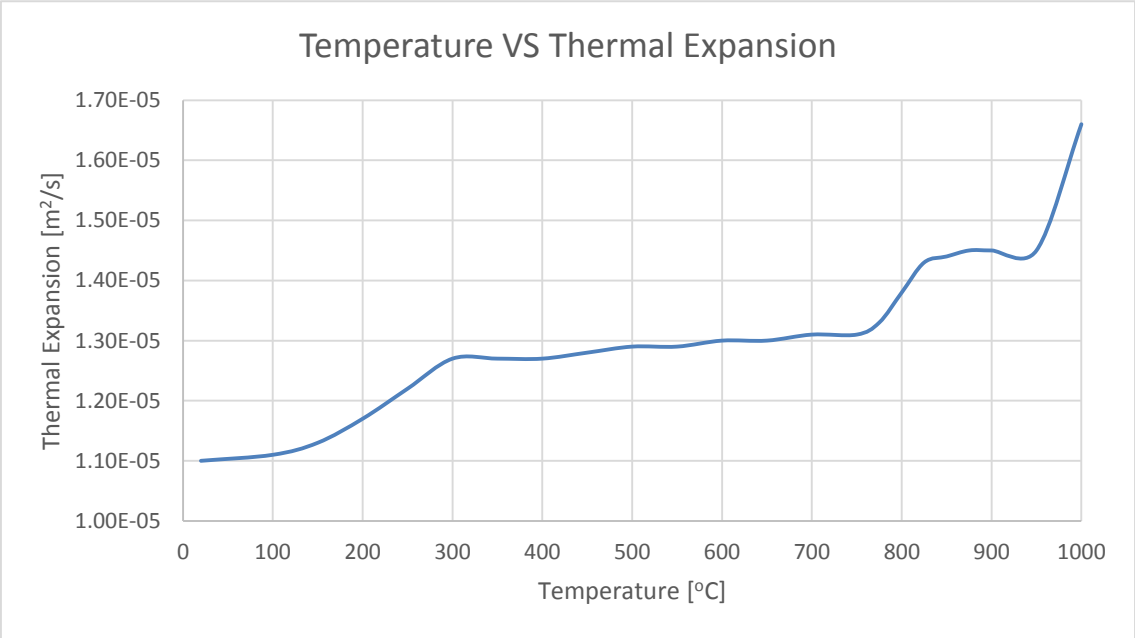
$$\text{Friction Forces: } F_{\mu 1} = \frac{T_{B1}}{r\mu} = \mathbf{7.4 \text{ kN}}$$

Appendix E – Temperature Dependent Properties with Plasticity for Cast Iron

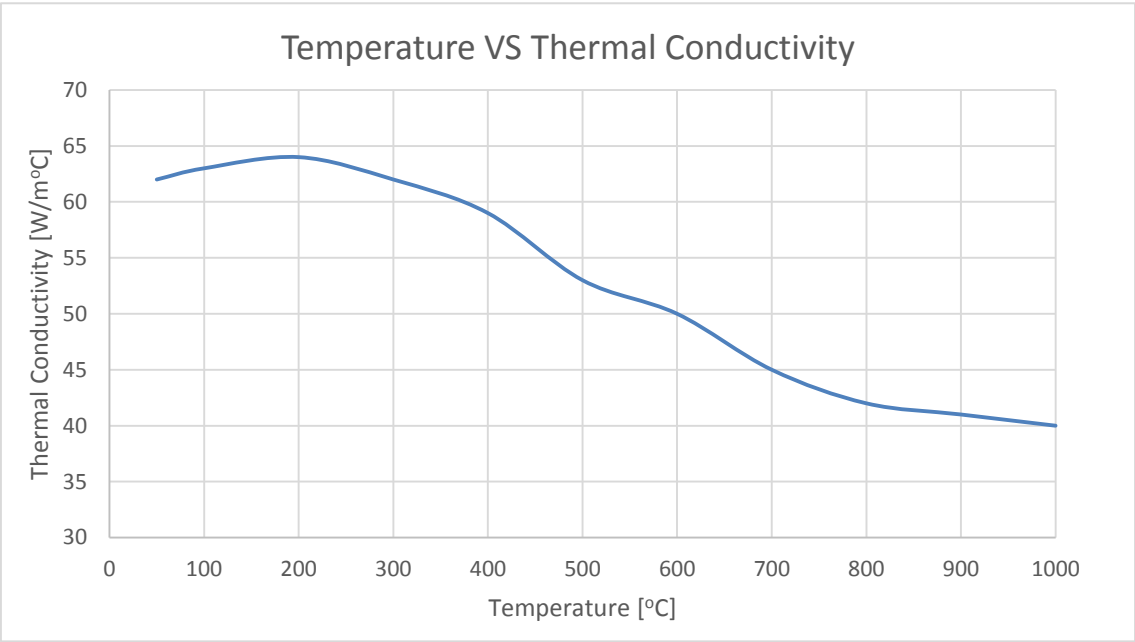
E.1 Temperature VS Tensile Strength



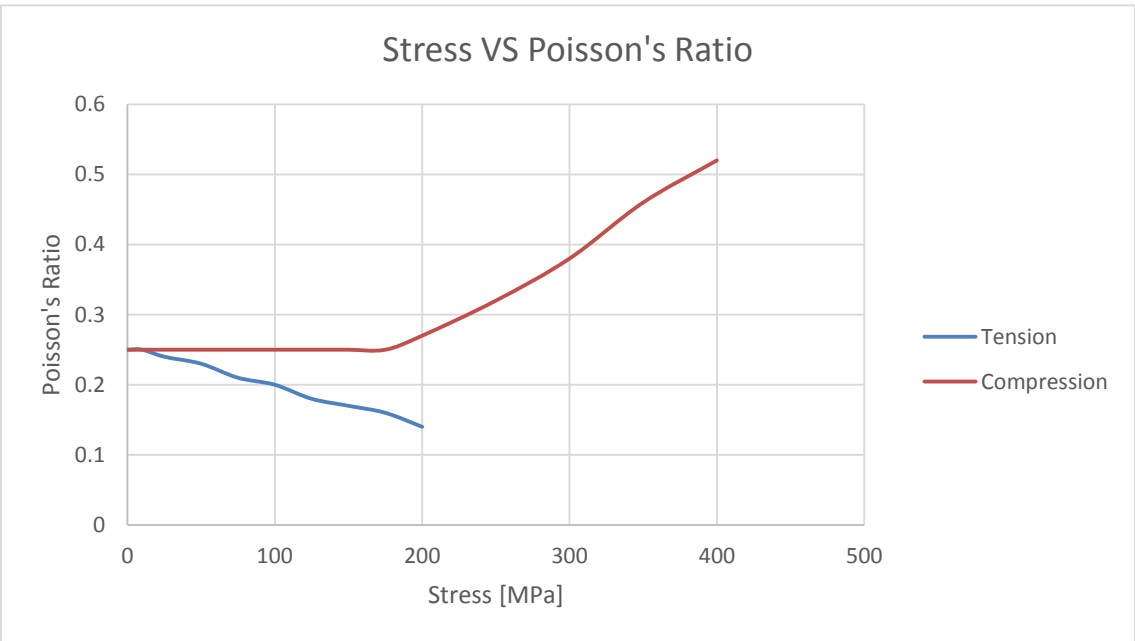
E.2 Temperature VS Thermal Expansion



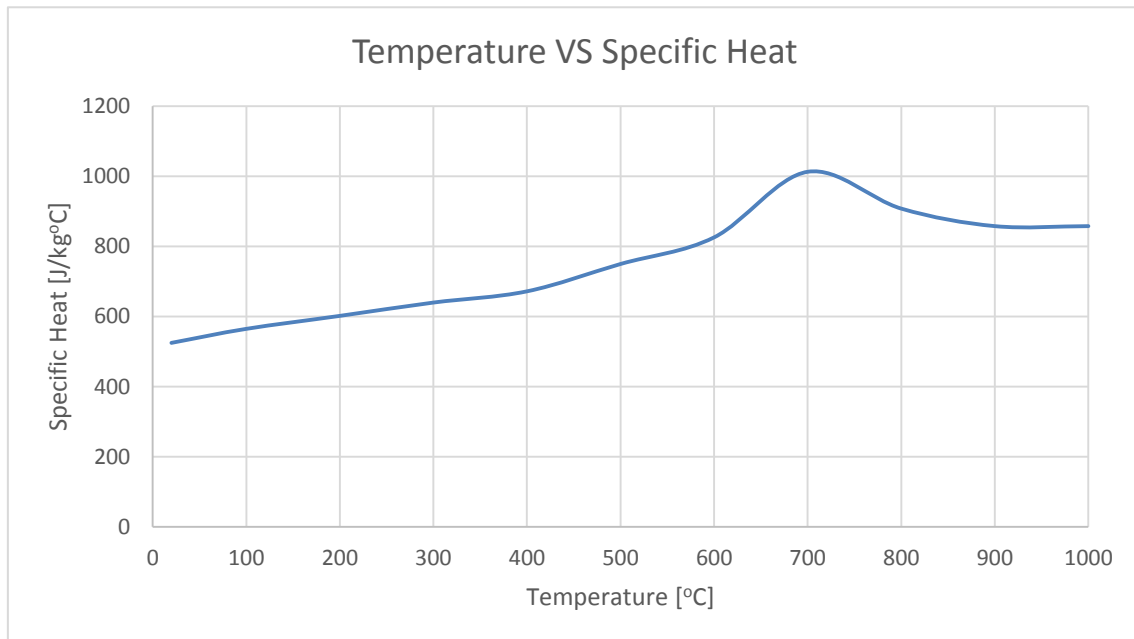
E.3 Temperature VS Thermal Conductivity



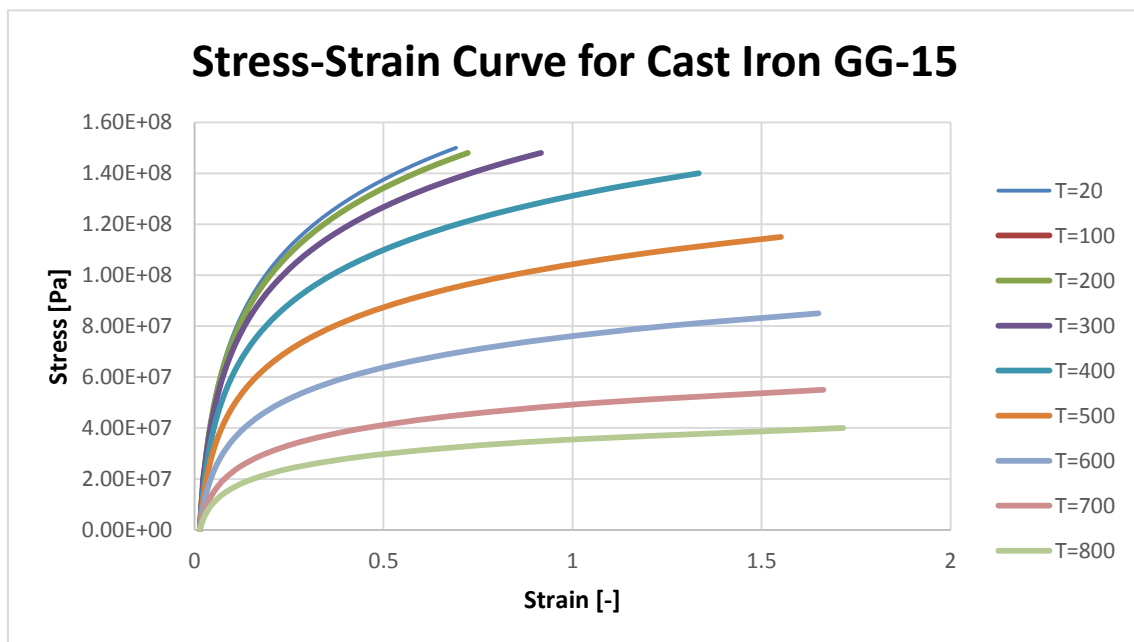
E.4 Stress VS Poisson Ratio



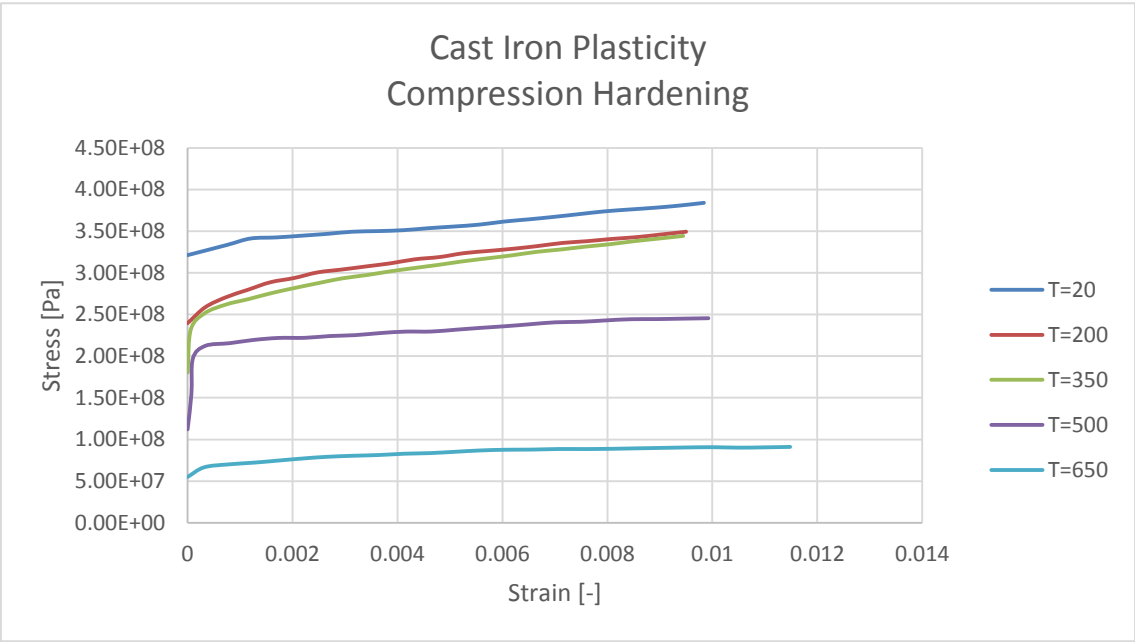
E.5 Temperature VS Specific Heat



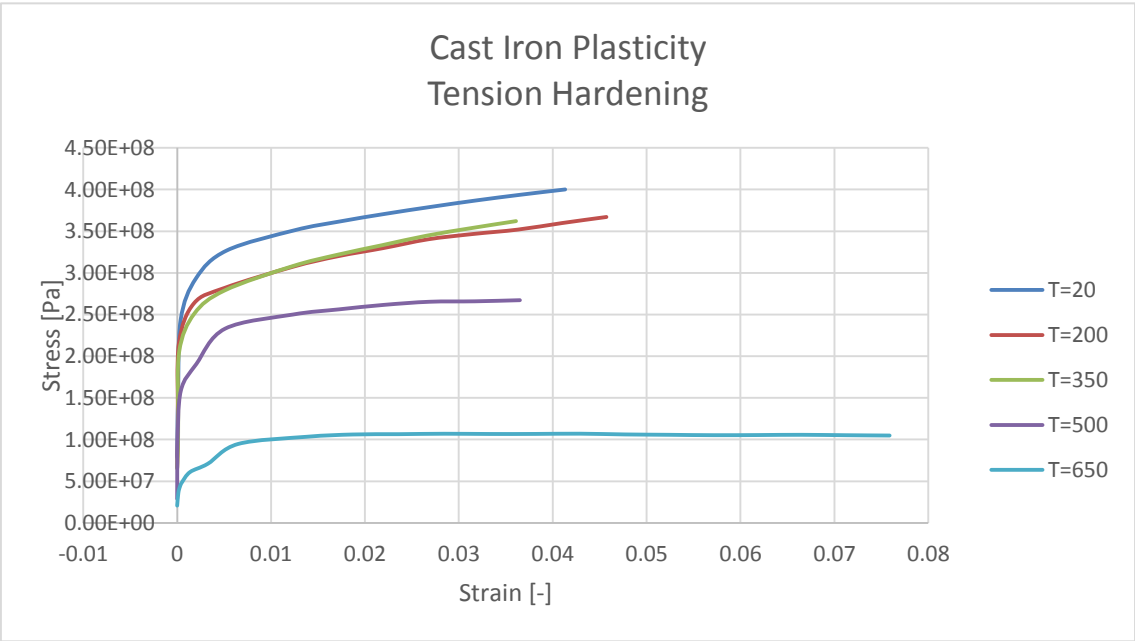
E.6 Stress/Strain Curves for Different Temperatures



E.7 Cast Iron Compression Hardening



E.8 Cast Iron Tension Hardening



Appendix F – Lotus Elise Heat Flux Calculations

Distance from the CoG (front): $\beta = \frac{l_{WB} \times M_R}{M_{FL}} = 1150 [mm]$

Distance from the CoG (rear): $\gamma = l_{WB} - \beta = 1150 [mm]$

Deceleration: $\alpha_{decel} = g \times \varphi = 7.85 [\frac{m}{s^2}]$

Dynamic Weight Distribution Front:

$$Z_{DF} = \frac{(M_{FL} \times \alpha_{decel} \times h_{CoG}) + (M_{FL} \times g \times \gamma)}{l_{WB}} = 7348.59 [N]$$

Dynamic Weight Distribution Rear:

$$Z_{DR} = (M_{FL} \times g) - Z_{DF} = 3844.62 [N]$$

Ratio: $Z_{DF}:Z_{DR} = 65.7\%:34.3\%$

Energy to be dissipated: $E_K = \frac{1}{2} \times M_{FL} \times V_{Max}^2 = 1.85 [MJ]$

Energy to be dissipated at the front axle: $E_{KF} = E_K \times \frac{Z_{DFratio}}{100} = 1.21 [MJ]$

Energy to be dissipated per disc (front axle): $E_{KFd} = \frac{E_{KF}}{2} = 0.63 [MJ]$

Braking event duration: $t_{Total} = \frac{V_{Max}}{\alpha_{decel}} = 7.26 [sec]$

Maximum braking power: $P_{Max} = \frac{E_K \times 2}{t_{Total}} = 0.51 [MW]$

Maximum braking power per disc (front): $P_{Maxd} = \frac{E_{KFd} \times 2}{t_{Total}} = 167 [kW]$

Disc swept area: $A_{SW} = \frac{\pi}{4} \times (OD^2 - ID^2) \times 2 = 0.075 [m^2]$

Maximum heat flux: $\Phi_{Max} = \frac{P_{Max}}{A_{SW}} = 2.21 [\frac{MW}{m^2}]$

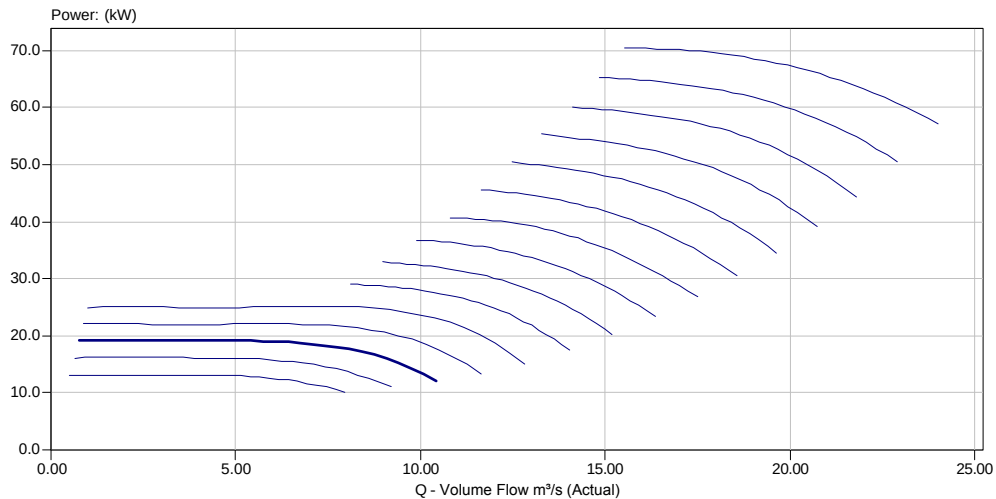
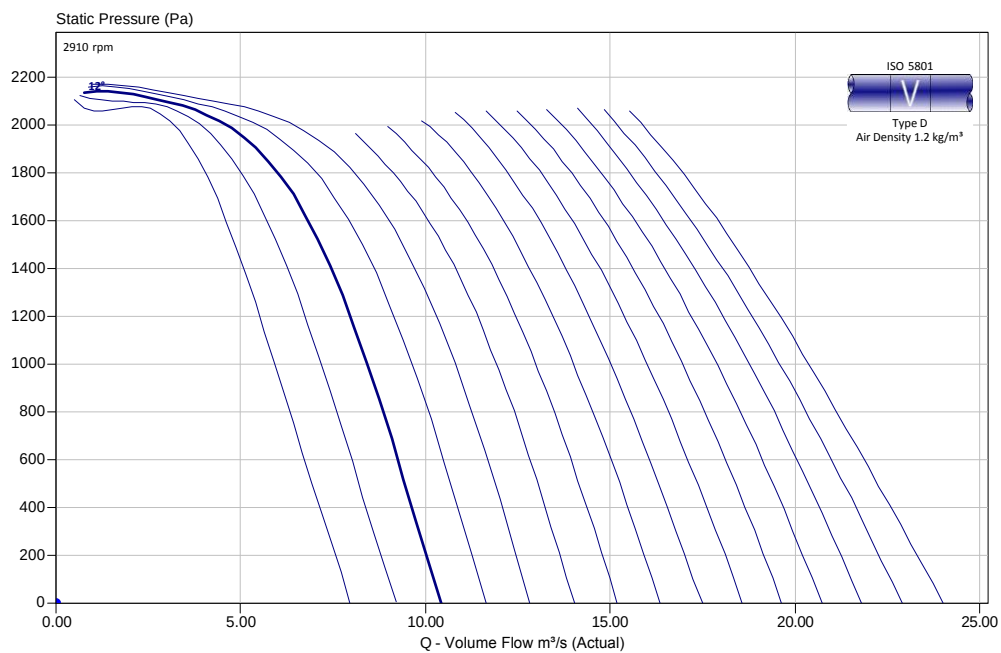
Appendix G – Fan-Motor Output Curves



Fläkt Woods Limited
Performance Chart
JM Aerofoil



Quotation Number :	Fan Code :	80JM/31/2/9/12
Project Name :	Customer :	
Item Reference: :	Date: :	Wednesday, August 19, 2015

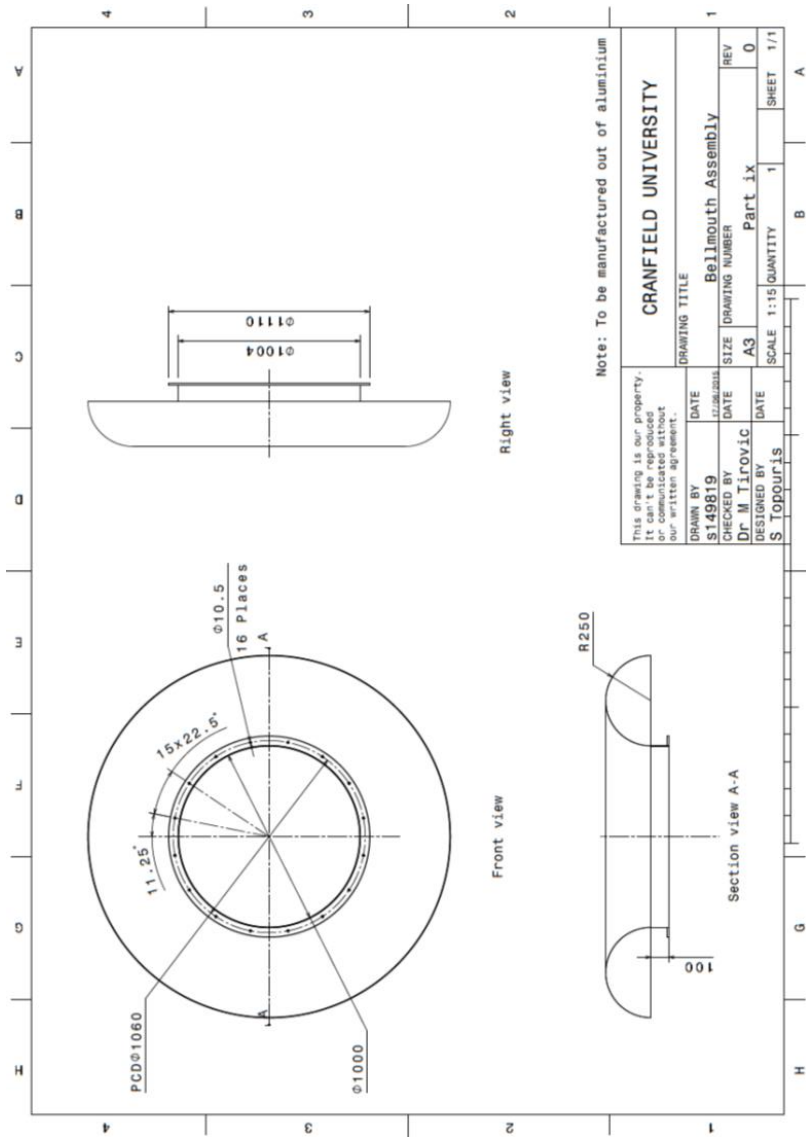


Axial Way
Colchester, Essex, CO4 5ZD
Tel: 01206 222621 Fax: 01206 222905
Printed on 19 August 2015

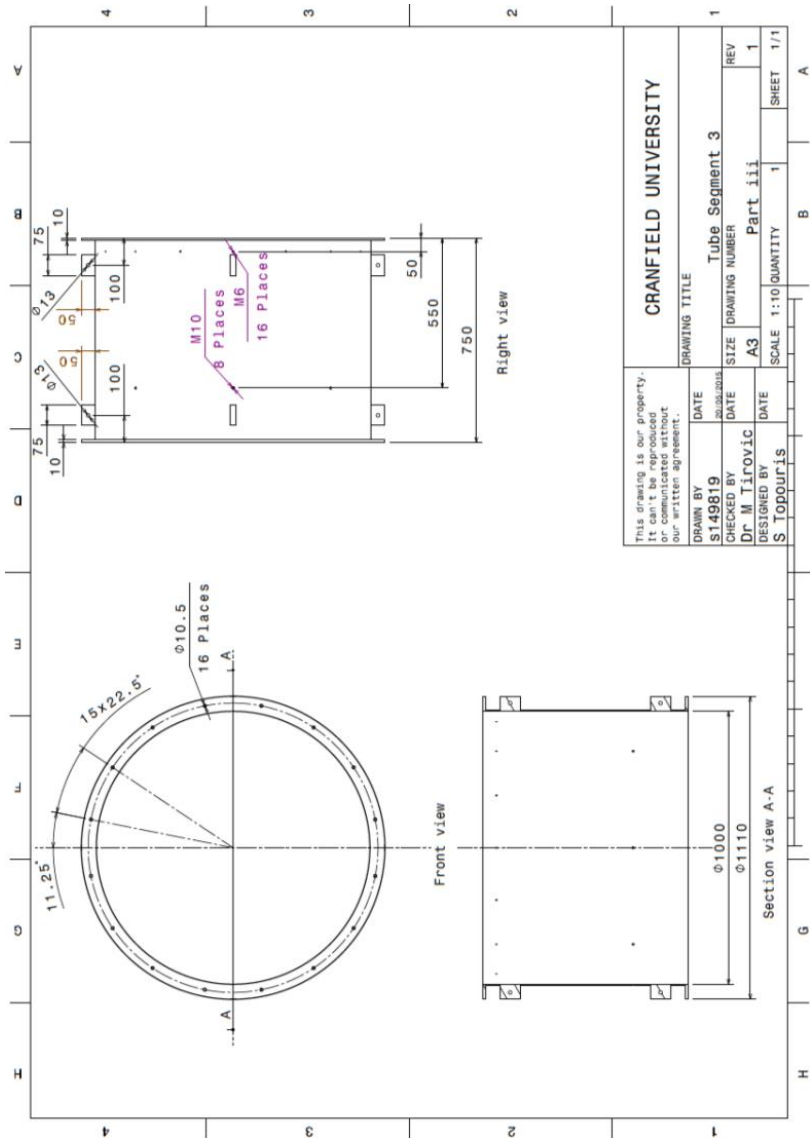
Website: www.flaktwoods.com
Email: samuel.perkins@flaktwoods.com
Copyright Fläkt Woods Group 2003 - 2015
Selection Engine: 3.0.0.4c

Appendix H – Flow Channel Drawings

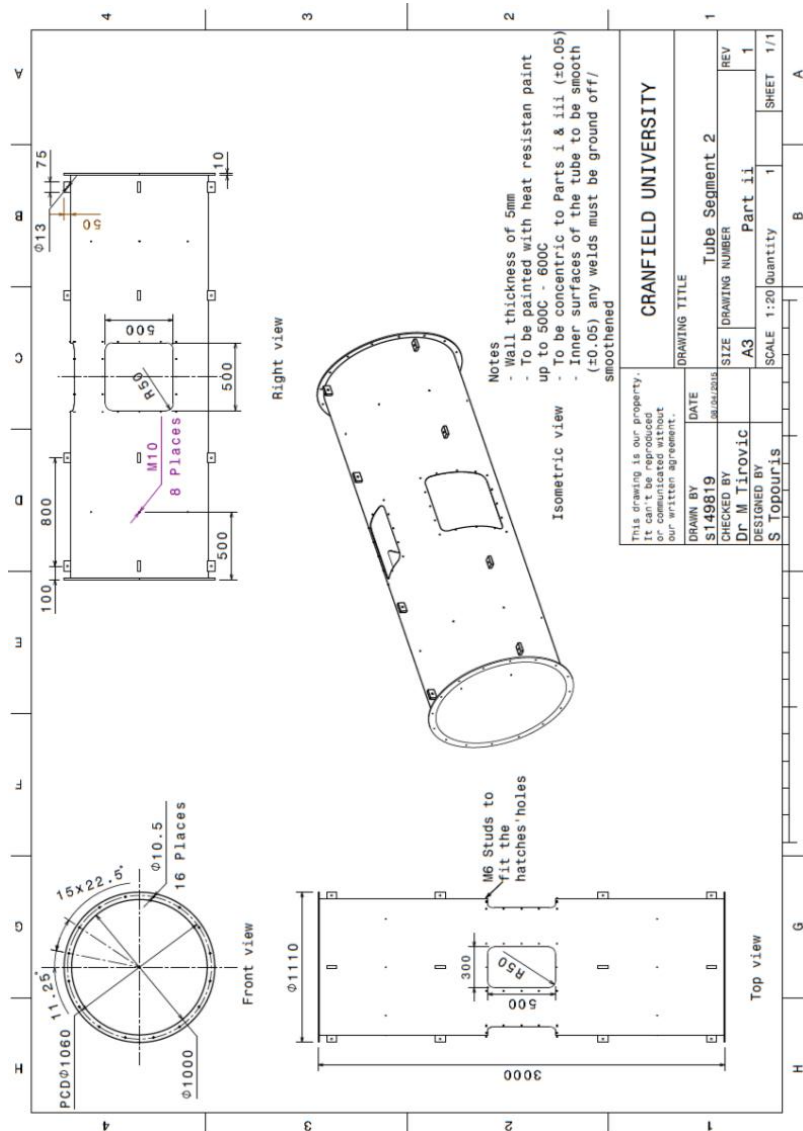
H.1 Bellmouth



H.2 Segment 3

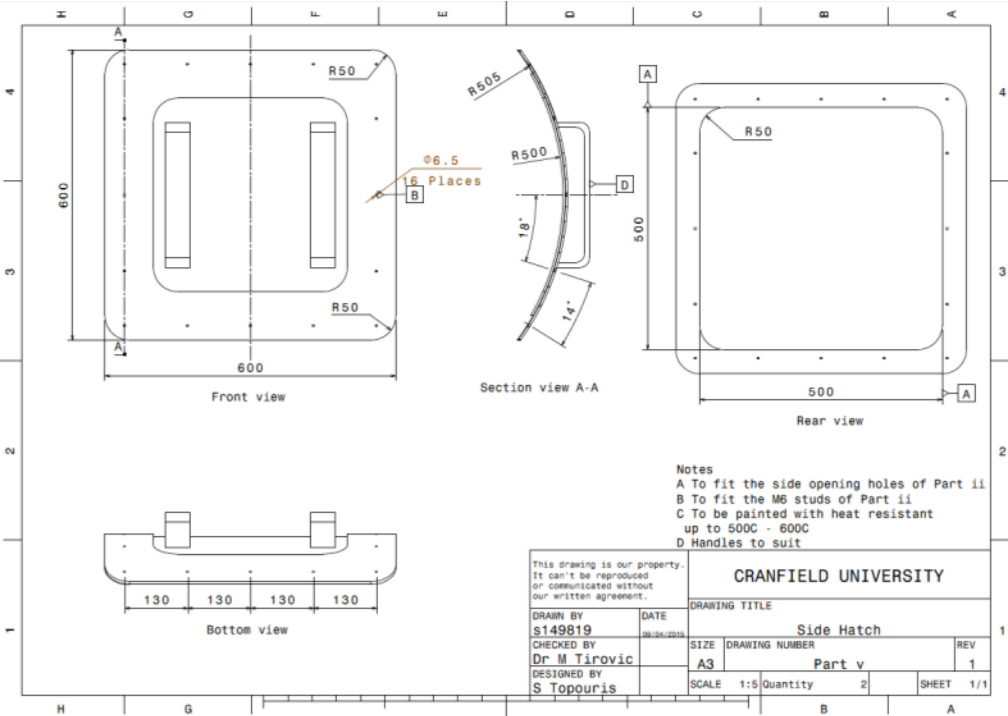


310

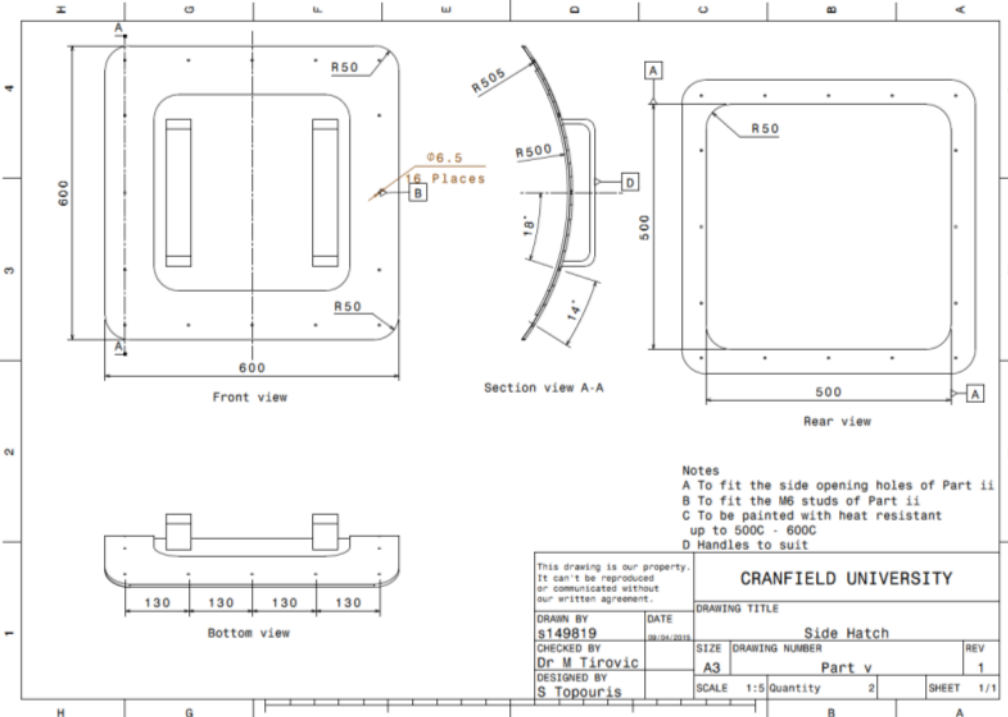


H.4 Drawings of Hatches

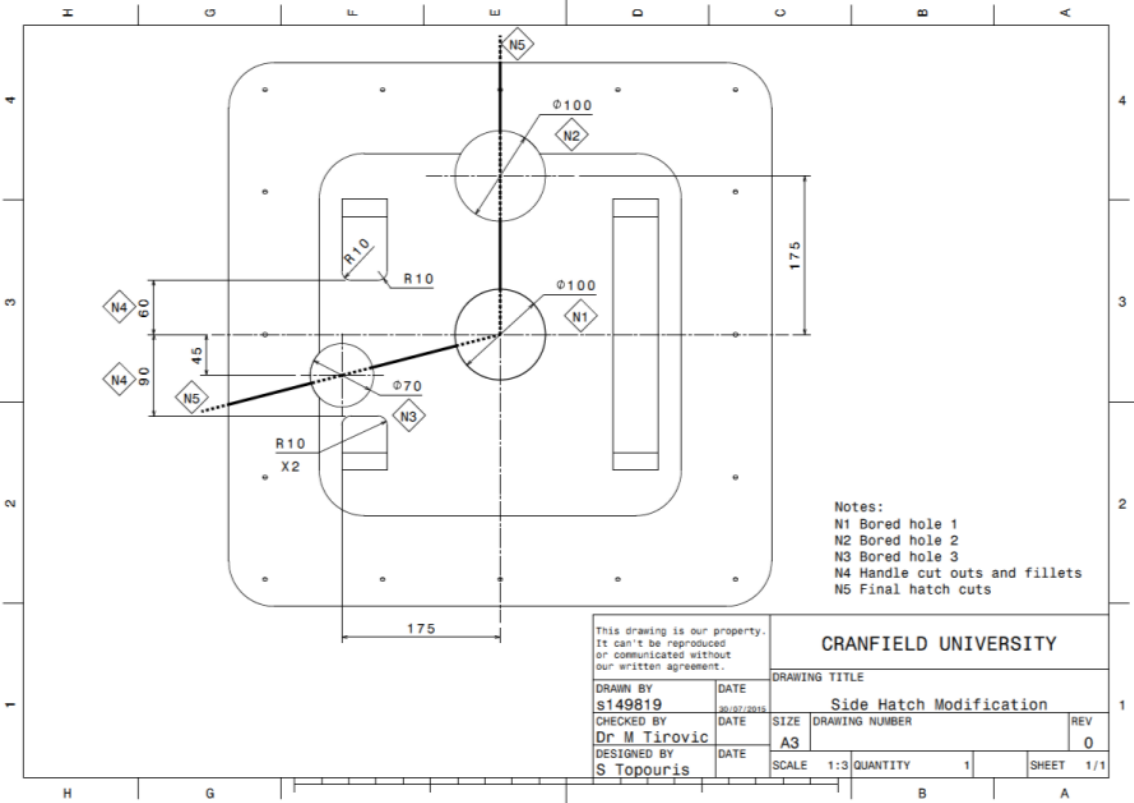
H.4.1 Side Hatches



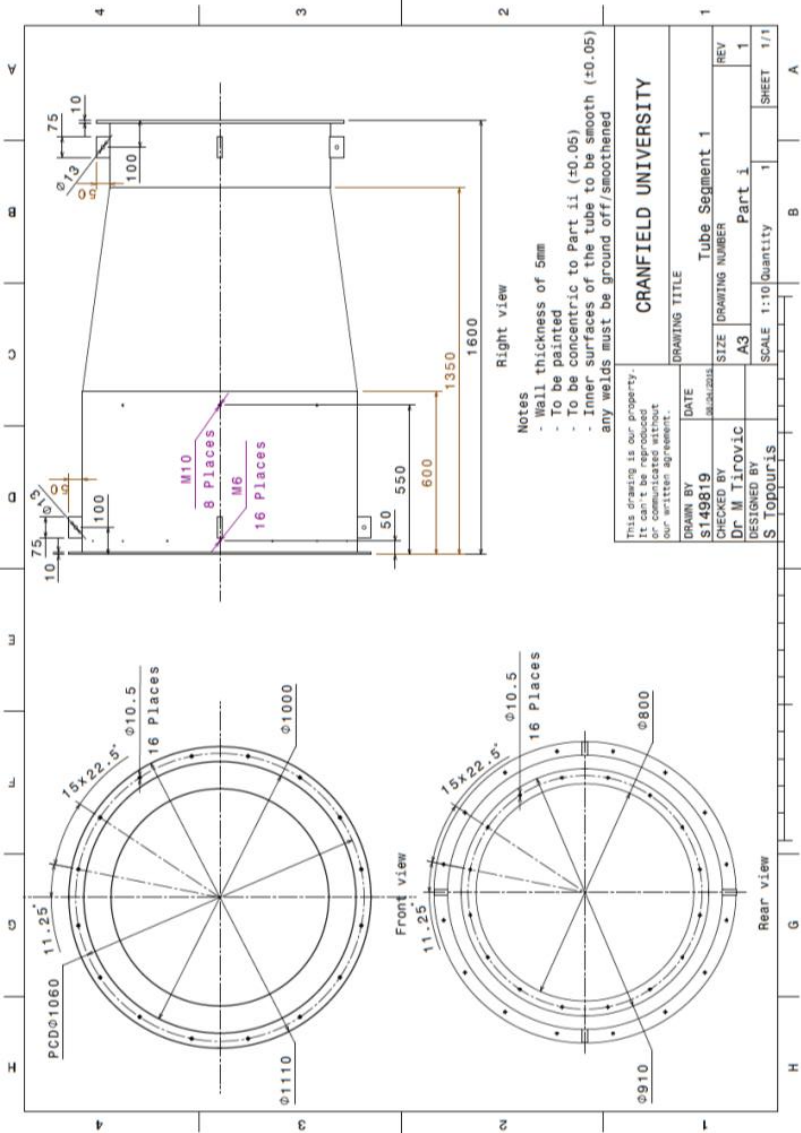
H.4.2 Top Hatch



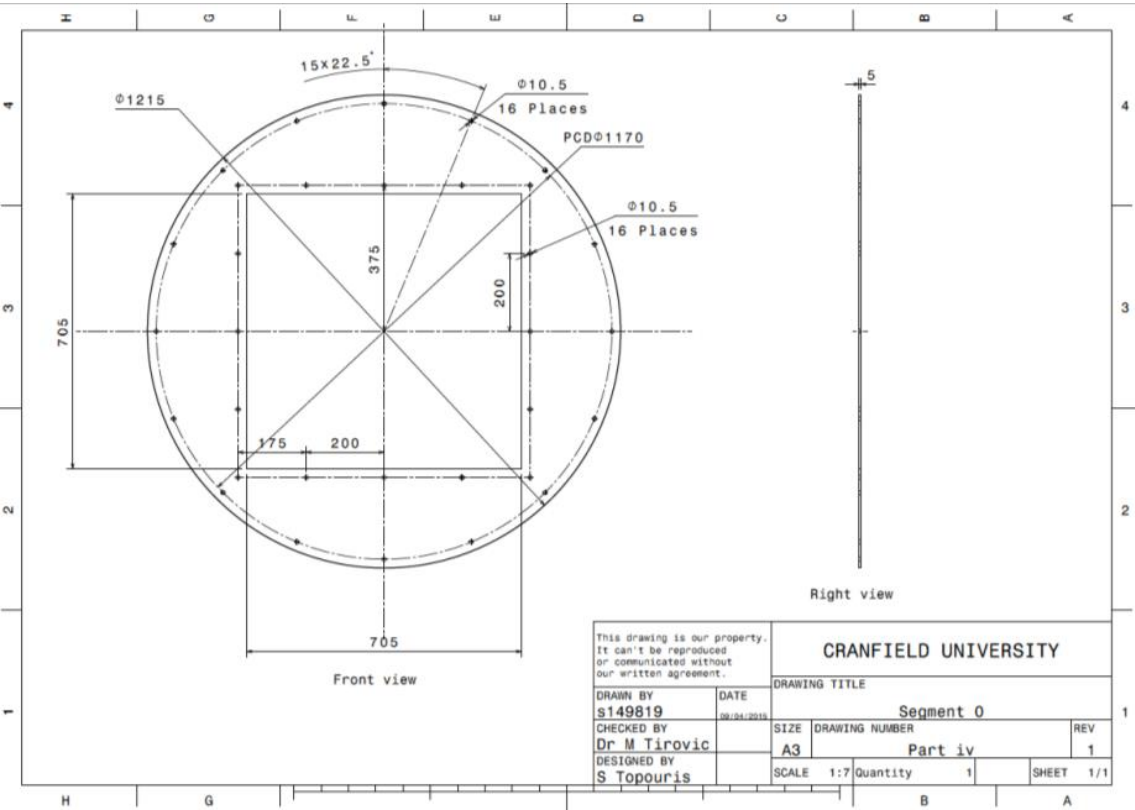
H.4.3 Modified Right Side Hatch



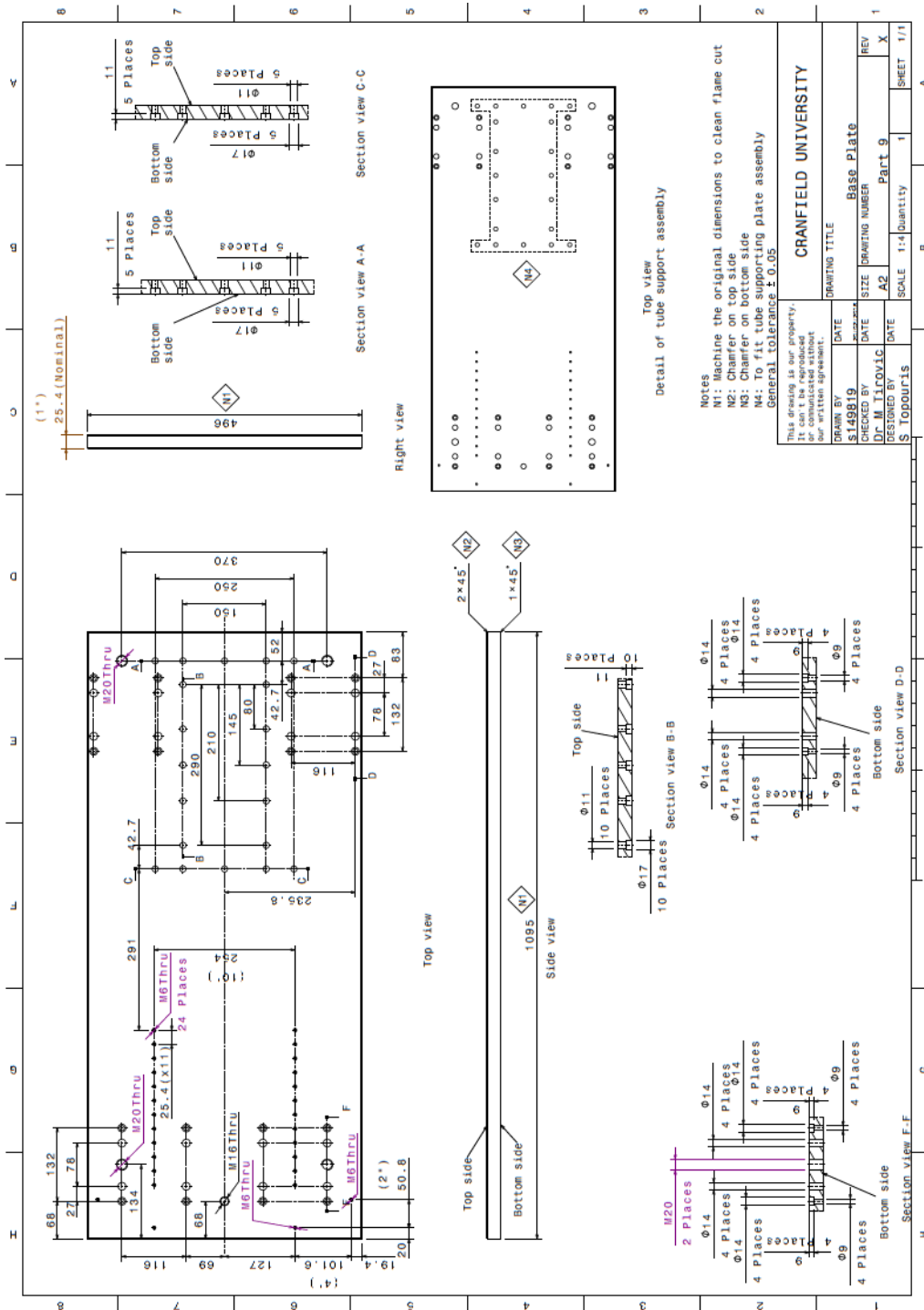
H.5 Segment 1



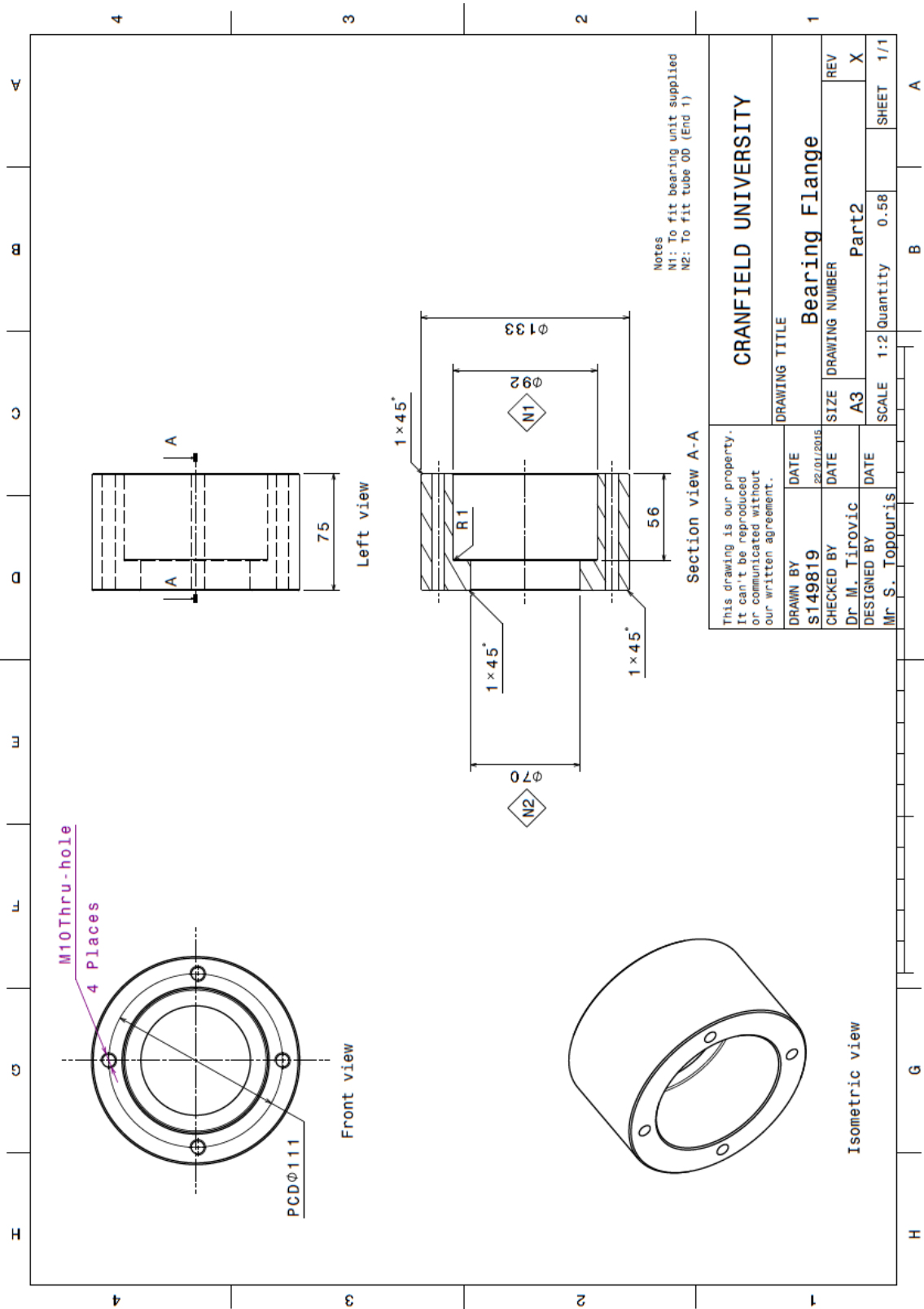
H.6 Segment 0



I.1 Base Plate

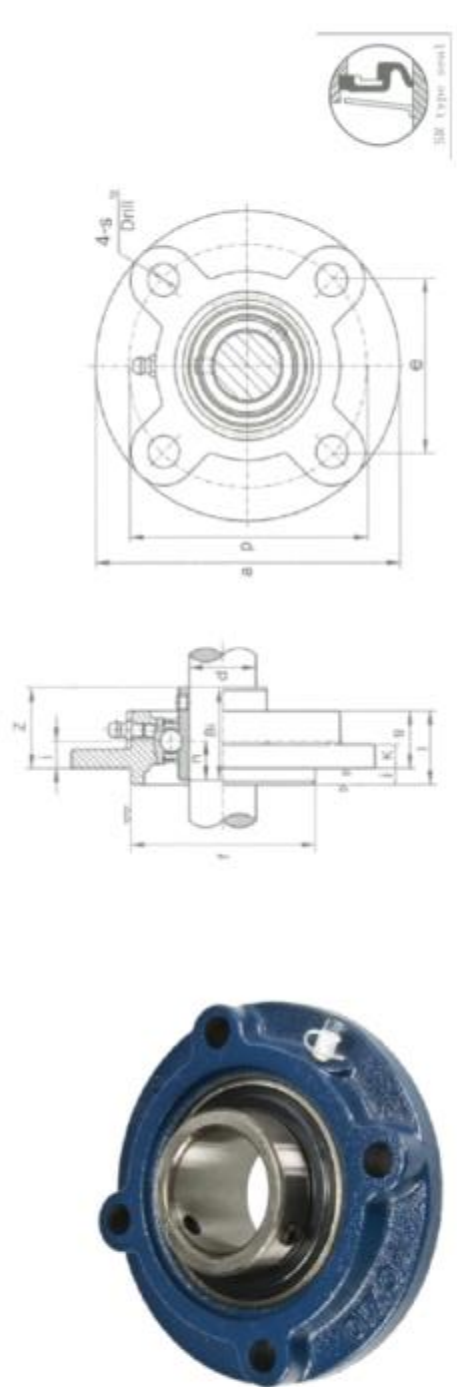


I.2 Bearing Flange



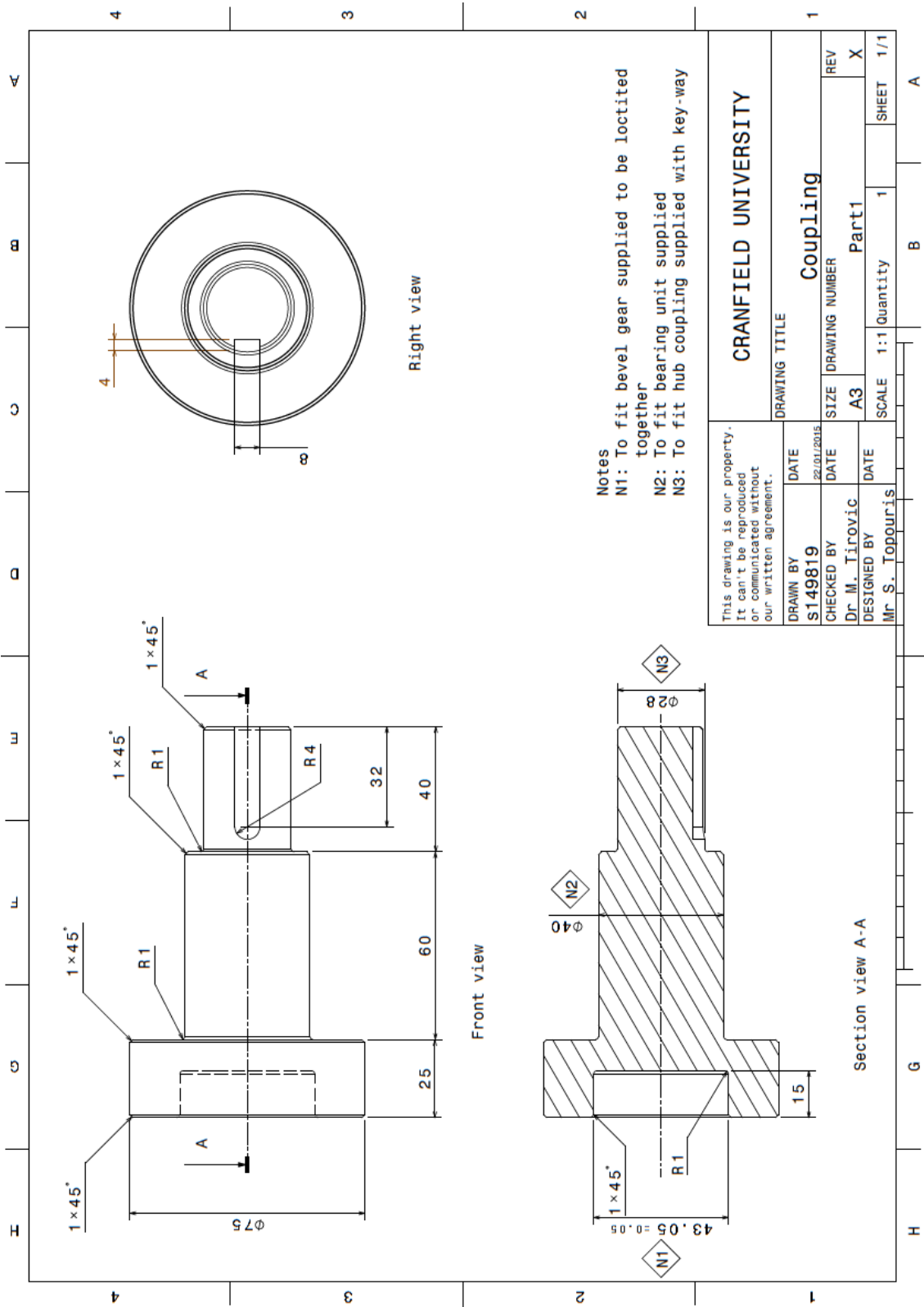
I.3 Bearing Unit

RS
Flange Cartridge Bearing unit MD – Metric

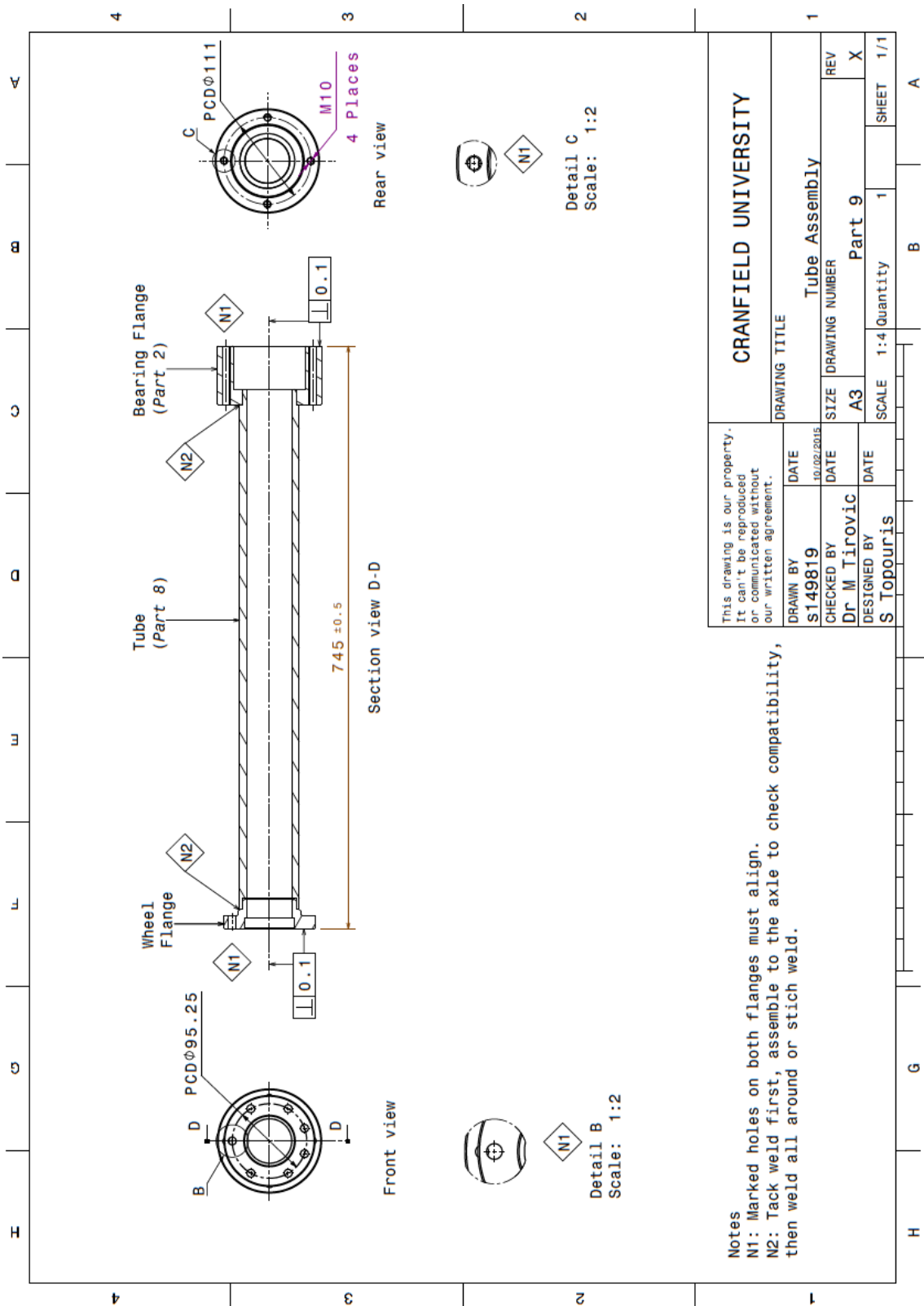


RS Article no.	Bearing unit No.	Shaft Dia. d (mm)	Dimension (mm)													Bolt used		Bearing No.	Housing No.	Weight (kg)
			a	p	e	i	l	s	j	k	g	f	z	Bi	n					
																(mm)				
7508948	UCFCX08	40	133	111	78.5	9	37	12	11	11	26	92	39.2	40.2	19	M10	UC X08	FCX 08	2	

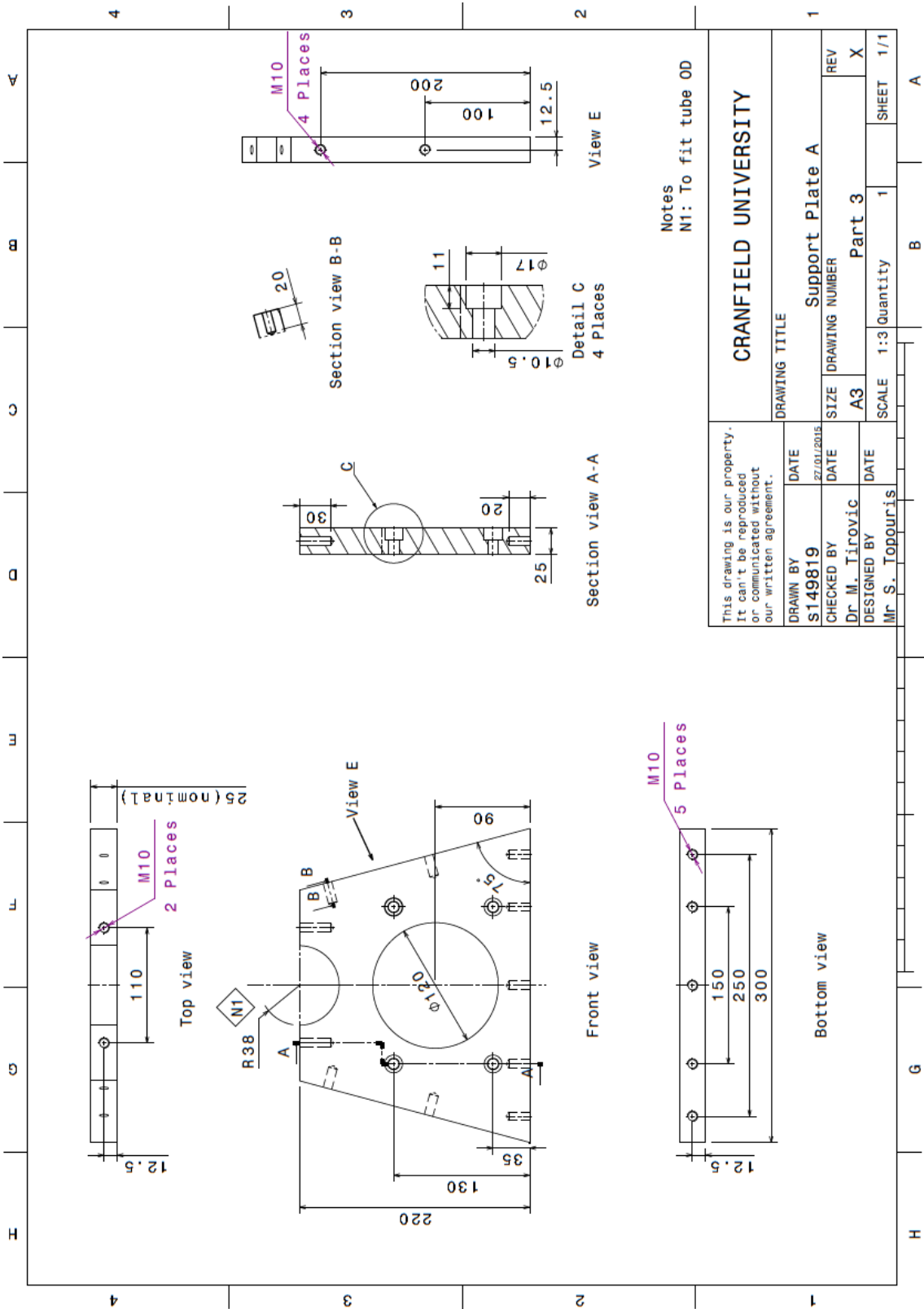
I.4 Coupling



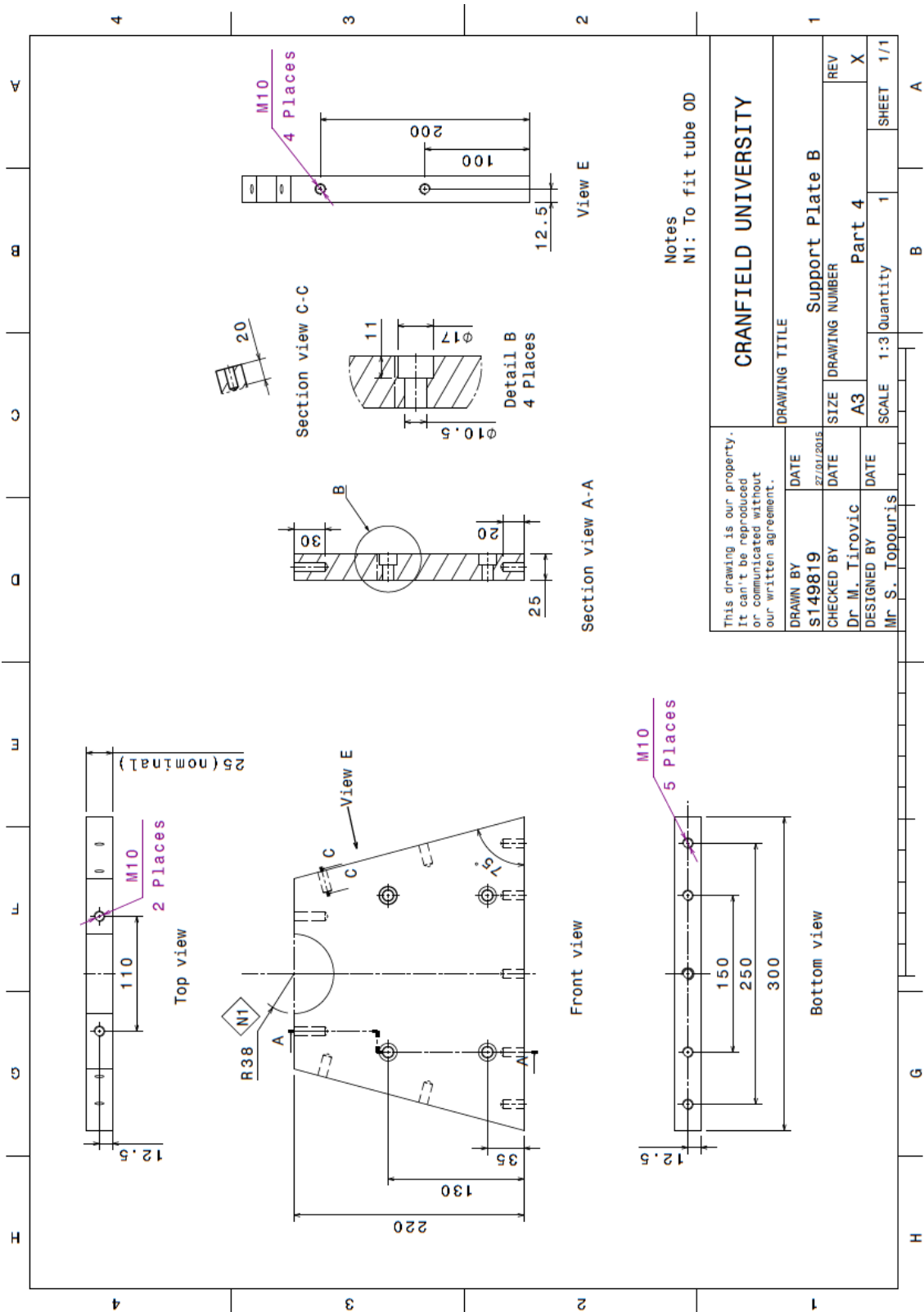
I.5 Axle Assembly Drawing



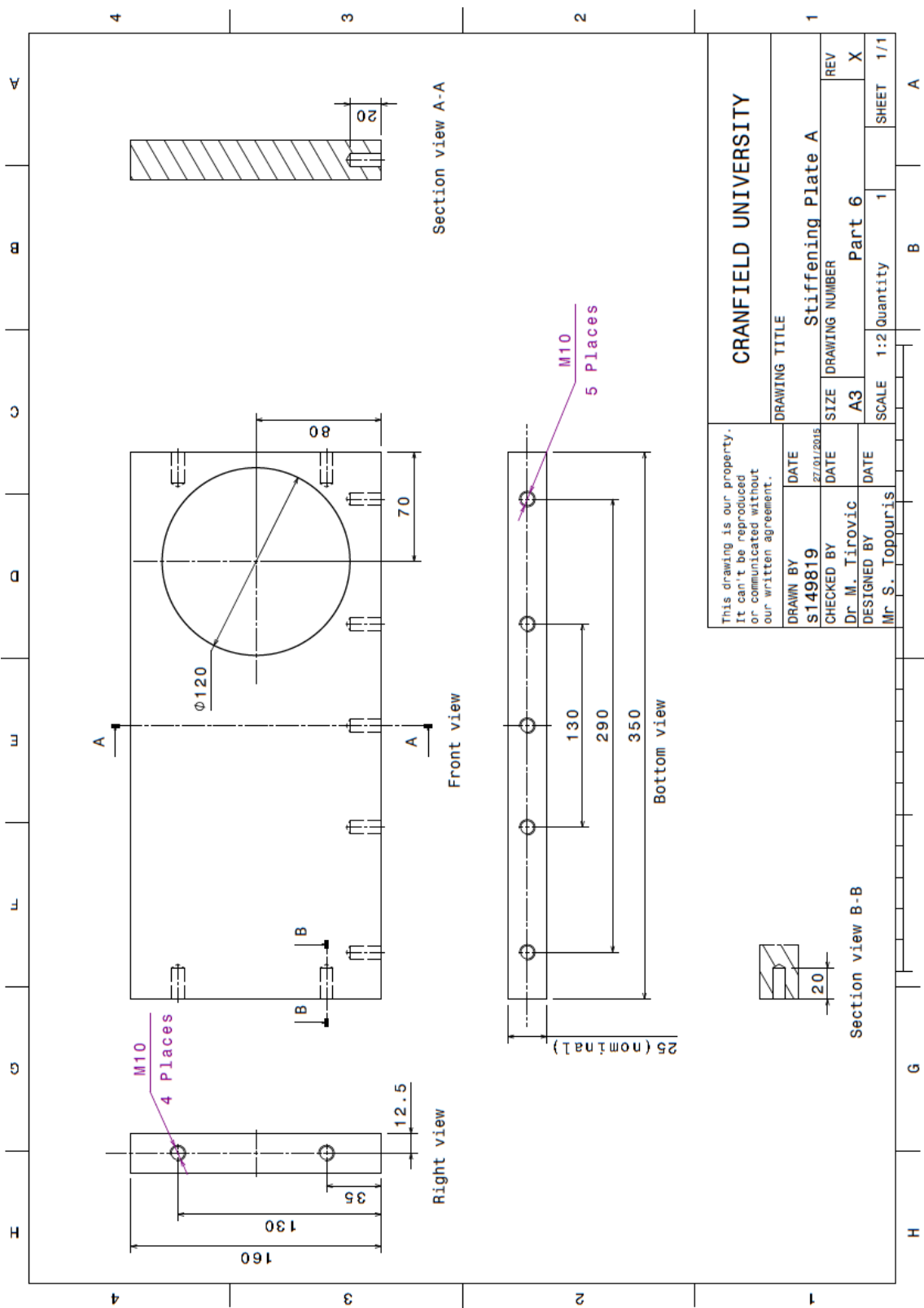
I.6 Support Plate A



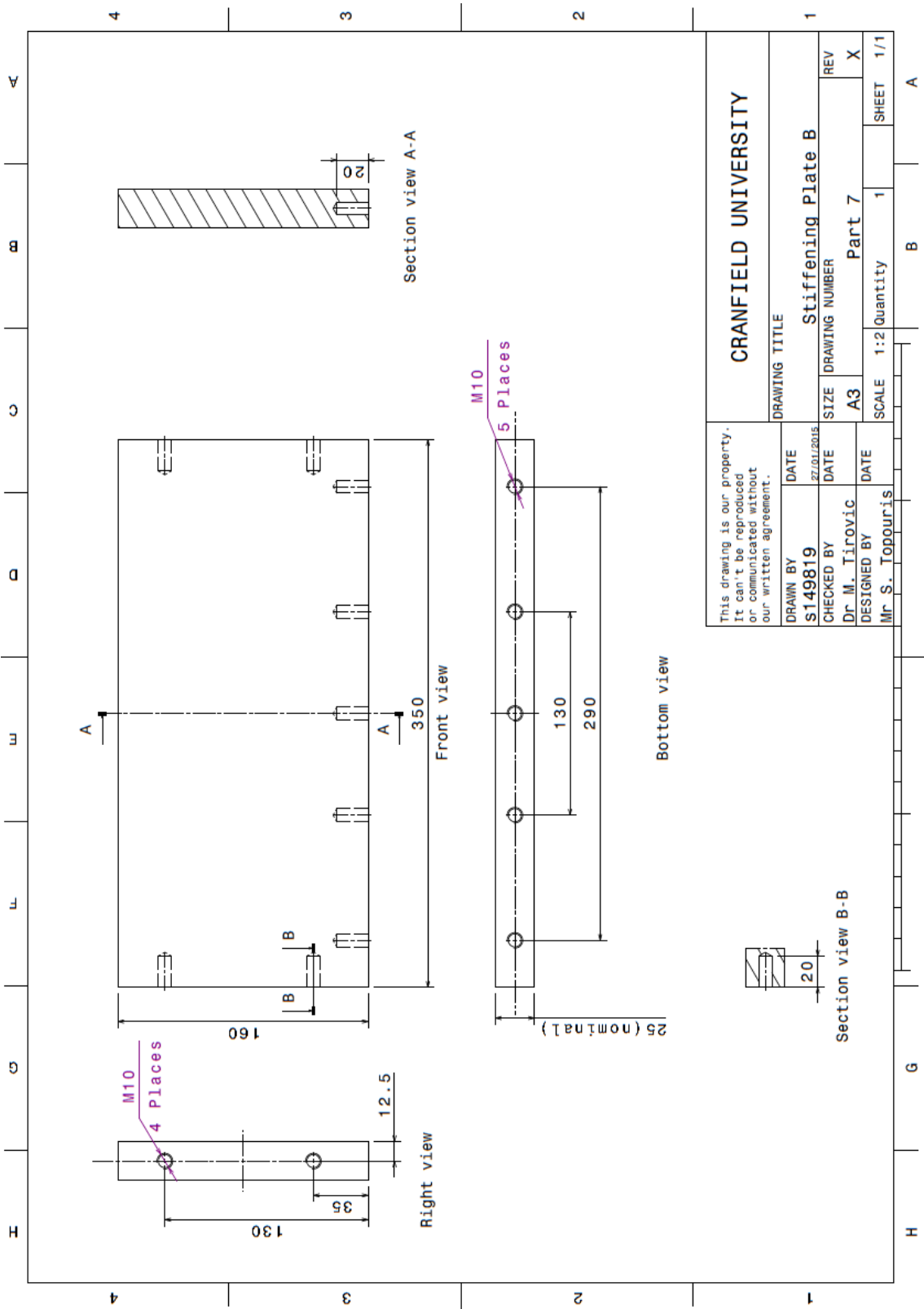
I.7 Support Plate B



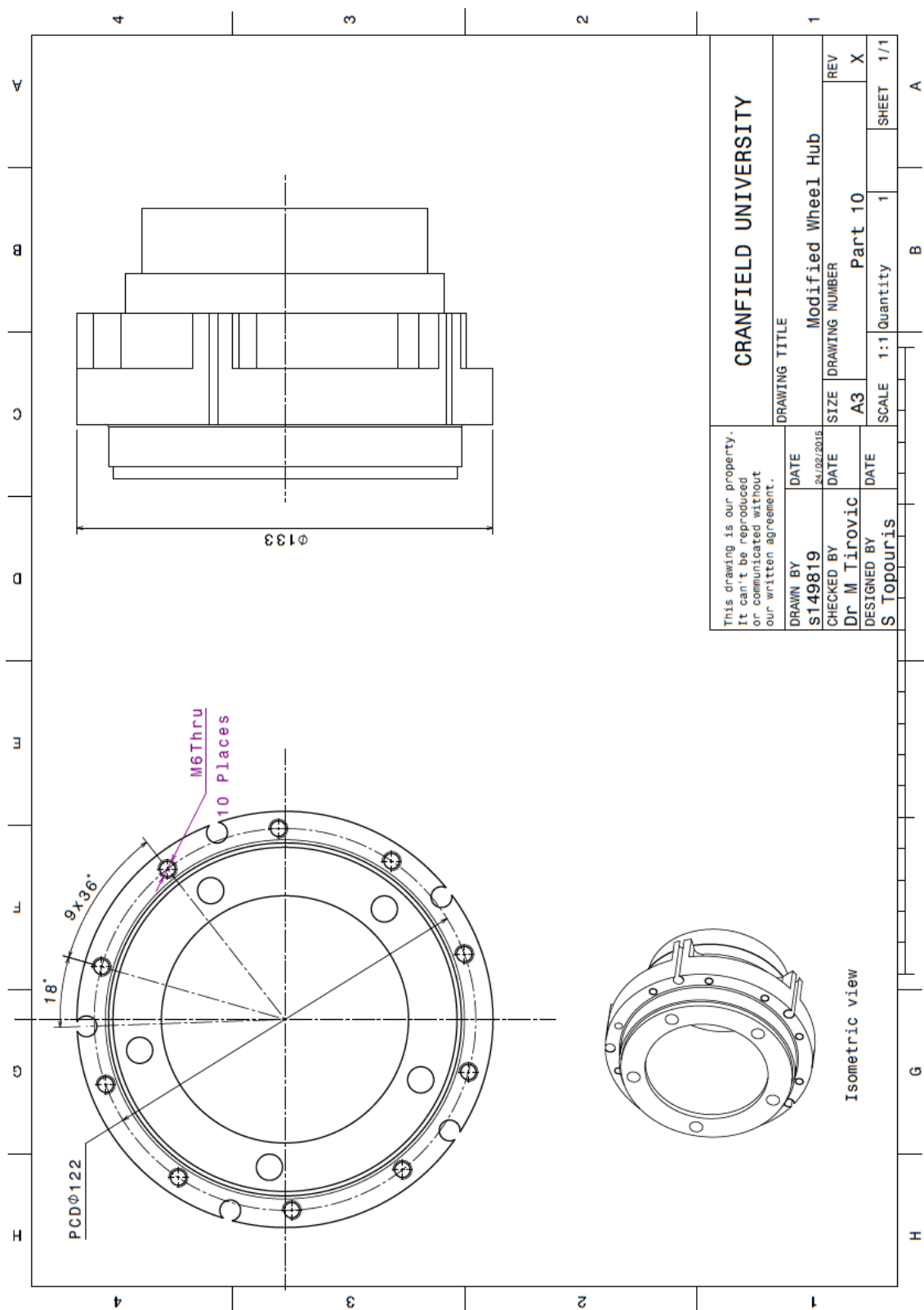
I.8 Stiffening Plate A



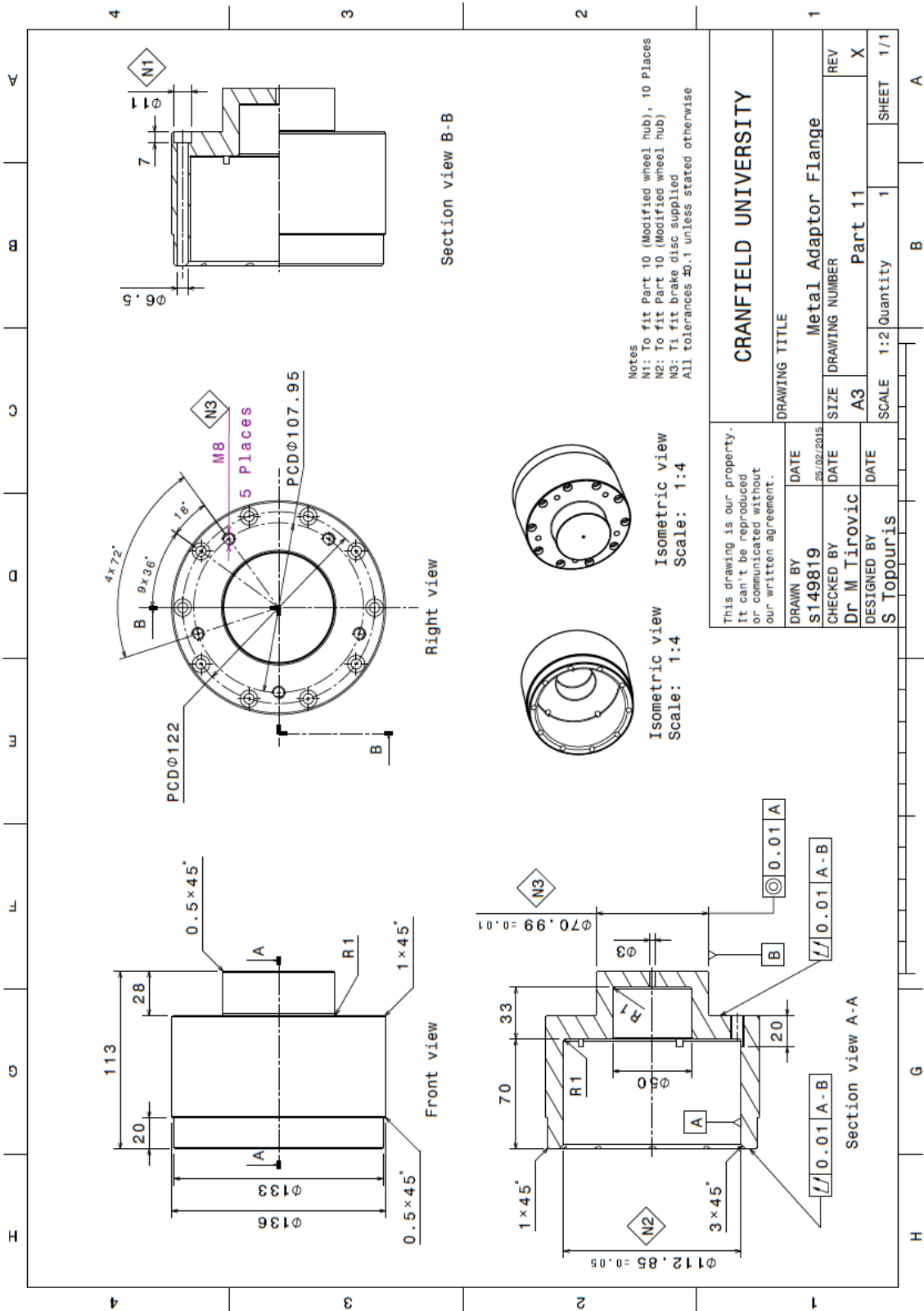
I.9 Stiffening Plate B



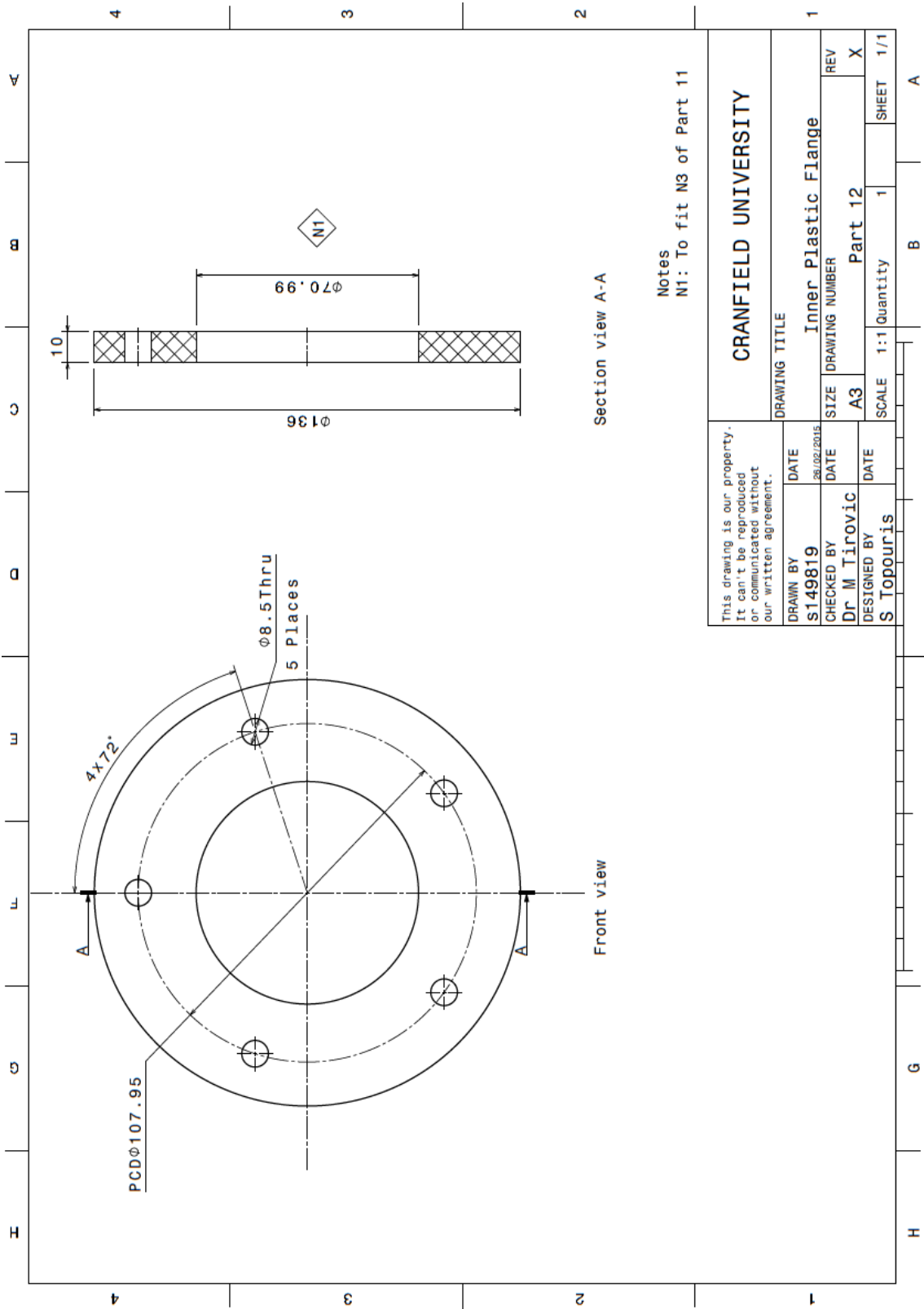
I.10 Modified Wheel End



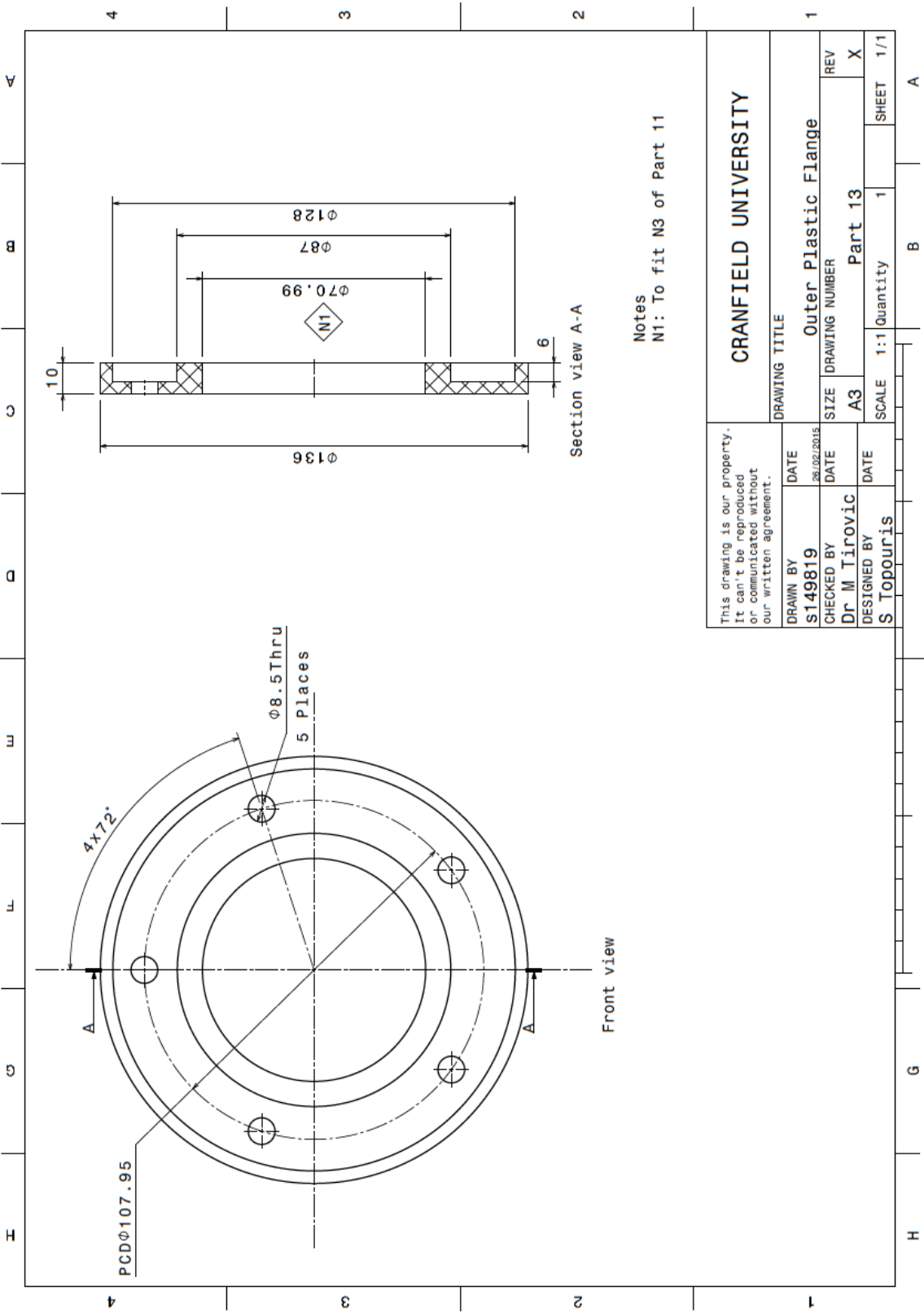
I.11 Steel Flange



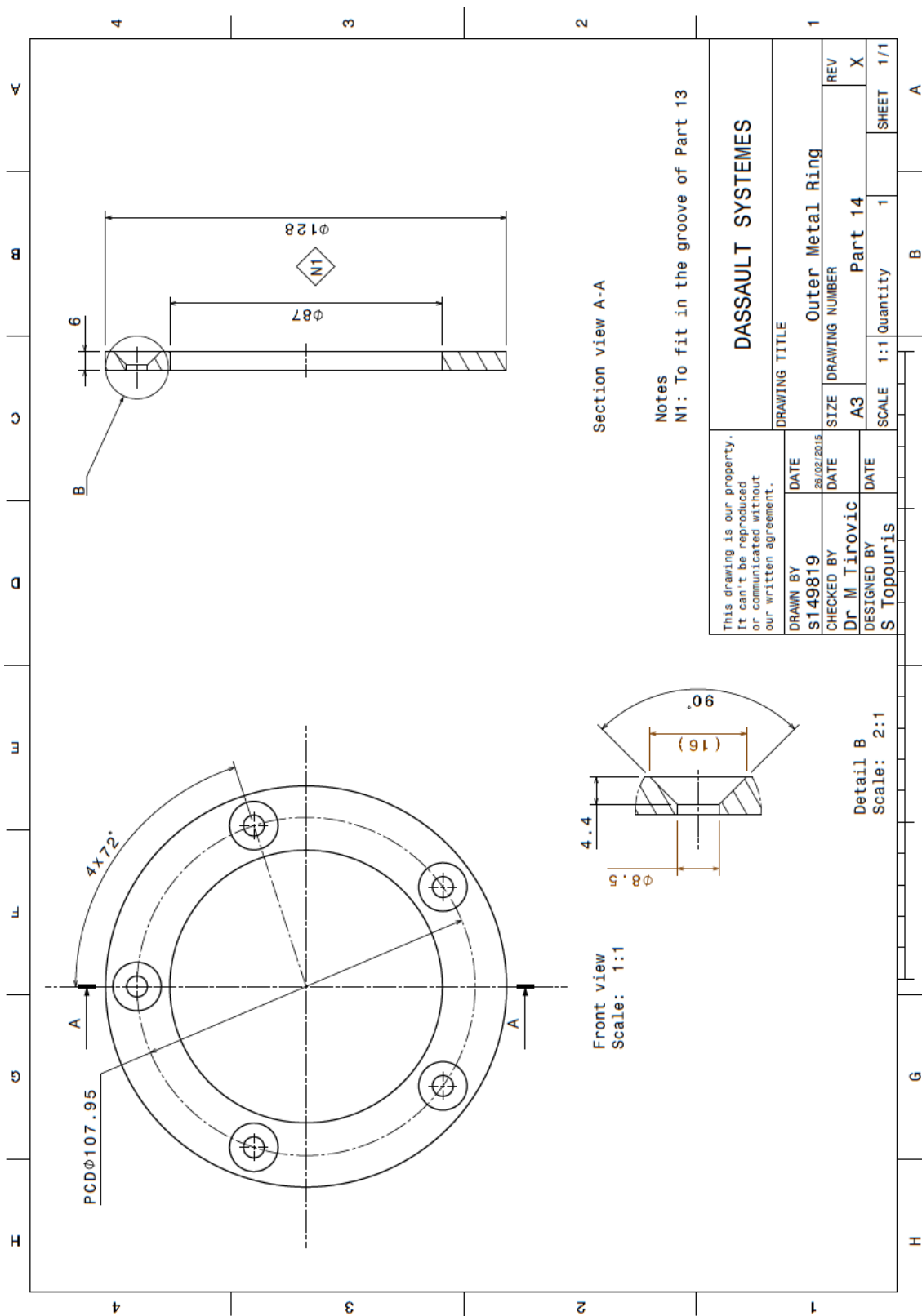
I.12 Inner Plastic Flange



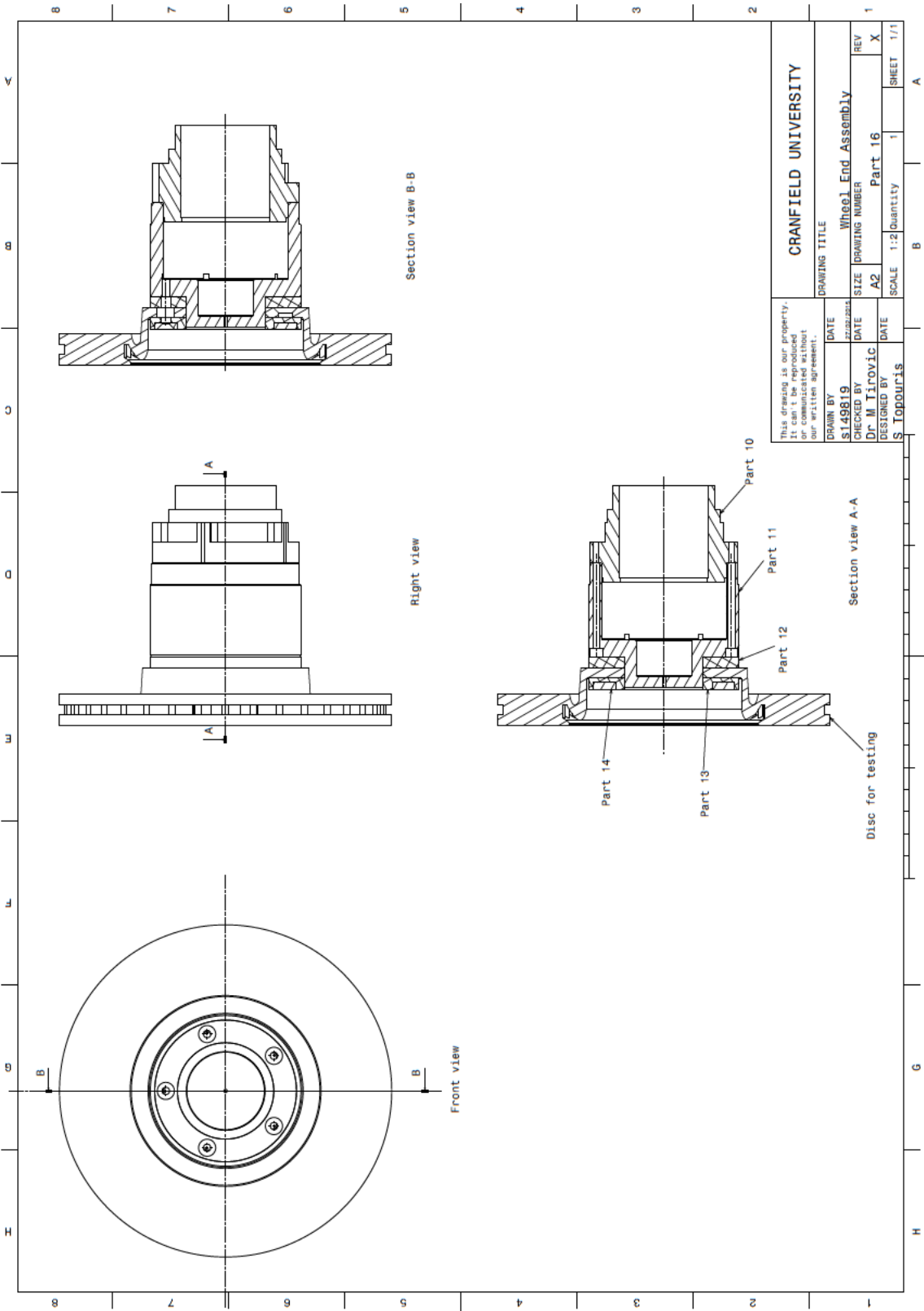
I.13 Outer Plastic Flange



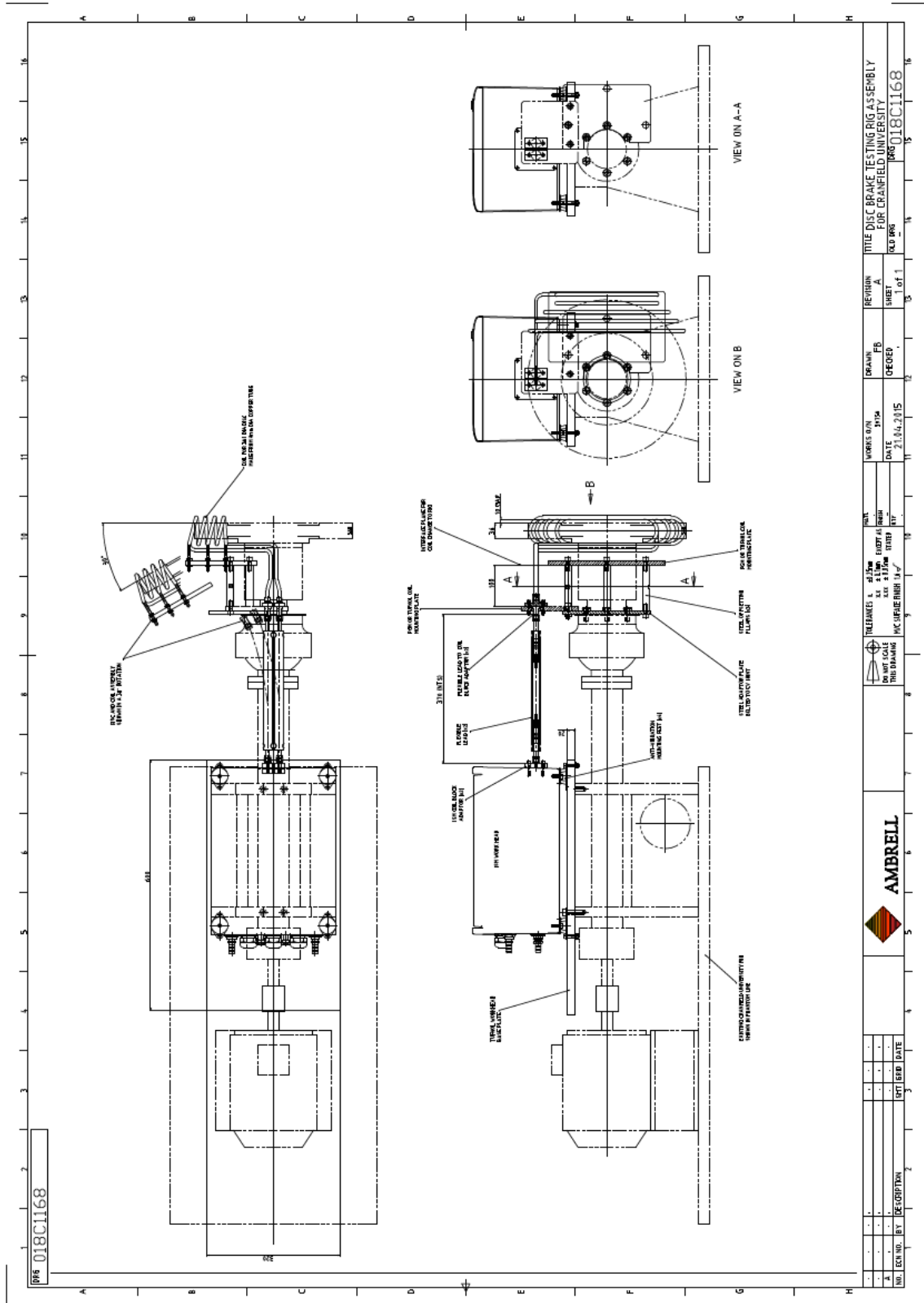
I.14 Outer Steel Ring



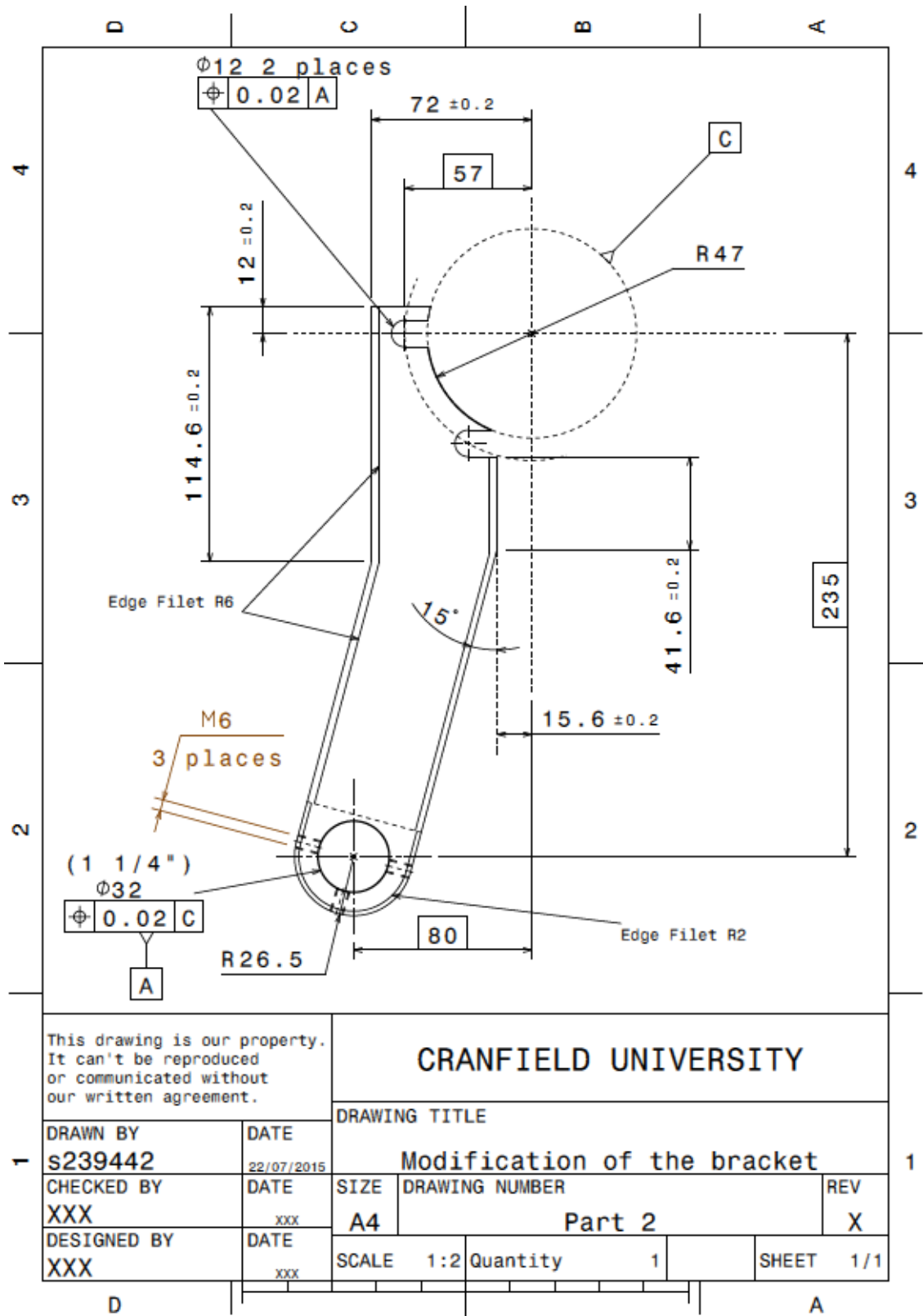
I.15 Steering Knuckle Assembly



I.16 IH Coil and Workhead



I.17 Thermocouple Support



Appendix J – Disc Rotating and Heating Assembly Electrical Components

J.1 ABB Electric Motor M2AA 1001-2



J.2 ABB Inverter Drive ACS150-03E-07A3-4



Appendix K – Induction Heater Components

K.1 ISM Control Unit



K.2 Water Cooler



Appendix L – National Instruments Modules and Specifications

L.1 CompactRio and Modules NI 9214, NI 9426 and NI 9205



L.2 Modules' Specifications

Table 0-1 NI 9205 Specification

Manufacturer	National Instruments
Model	Analogue input module 9205
Operating system/target	Real-time Windows
Single-ended channels	32
Differential channels	16
Analogue input resolution	16 bits
Maximum voltage range	± 10 V
Accuracy	6220 μ V
Minimum voltage range	± 0.2 V
Accuracy	157 μ V
Operating temperature	- 40 – 70 °C

Table 0-2 NI 9214 Specification

Manufacturer	National Instruments
Model	Thermocouple module 9214
Operating system/target	Real-time Windows
Ended channels	0
Differential channels	16
Analogue input resolution	24 bits
Maximum voltage range	± 78.125 V
Accuracy	25 μ V
Maximum bandwidth	68 Hz
Operating temperature	- 40 – 70 °C

Table 0-3 NI 9426 Specification

Manufacturer	National Instruments
Model	Digital input module 9426
Operating system/target	Real-time Windows
Digital input only channels	32
Maximum clock rate	140 kHz
Digital logic levels	24 V with external supply
Voltage range	0 – 24 V
Operating temperature	- 40 – 70 °C

Appendix M – Air Speed Measurements

M.1 Minimum – Maximum – Average Values

M.1.1 Plane 1

Plane 1		10Hz, 23C, 46.5% (7/8/2015)			20Hz, 23.5C, 46.6% (8/8/2015)			30Hz, 22.2C, 50% (8/8/2015)			40Hz, 20.3C, 76% (10/8/2015)			50Hz, 20.3C, 77.7% (10/8/2015)		
		max	min	avg	max	min	avg	max	min	avg	max	min	avg	max	min	avg
Side	1	3.38	2.92	3.23	4.91	4.33	4.63	7.65	6.82	7.28	10.19	8.69	9.59	12.14	9.06	11.41
	2	2.52	2.31	2.46	4.87	4.53	4.72	7.73	7.23	7.56	10.02	9.33	9.69	12.25	11.53	11.91
	3	2.64	2.08	2.38	4.97	4.75	4.87	7.71	7.12	7.49	10.28	9.79	10.03	12.36	11.7	12.06
	4	3.49	3.25	3.43	4.77	4.56	4.68	7.36	7.02	7.2	10.03	9.53	9.84	12.71	11.81	12.17
	5	2.45	2.32	2.51	4.75	4.46	4.66	7.57	7.09	7.4	9.92	9.14	9.56	12.93	11.64	12.36
Top	1	2.23	1.63	1.96	4.55	3.67	4.26	6.94	5.92	6.59	9.91	7.68	9.19	11.82	9.8	11.28
	2	2.69	1.24	2.43	5.29	4.97	5.21	7.99	7.48	7.8	10.83	9.56	10.53	13.87	12.72	13.28
	3	2.72	2.08	2.55	5.24	5.05	5.17	8.24	7.55	7.87	10.59	10.02	10.61	13.62	12.59	13.33
	4	2.74	2.52	2.63	5.33	4.75	5.18	8.06	7.63	7.92	10.89	10.1	10.68	13.69	13.02	13.41
	5	2.84	2.4	2.67	5.39	5.02	5.2	8.08	6.93	7.91	11.6	9.98	10.7	13.68	11.12	13.33

M.1.2 Plane 2

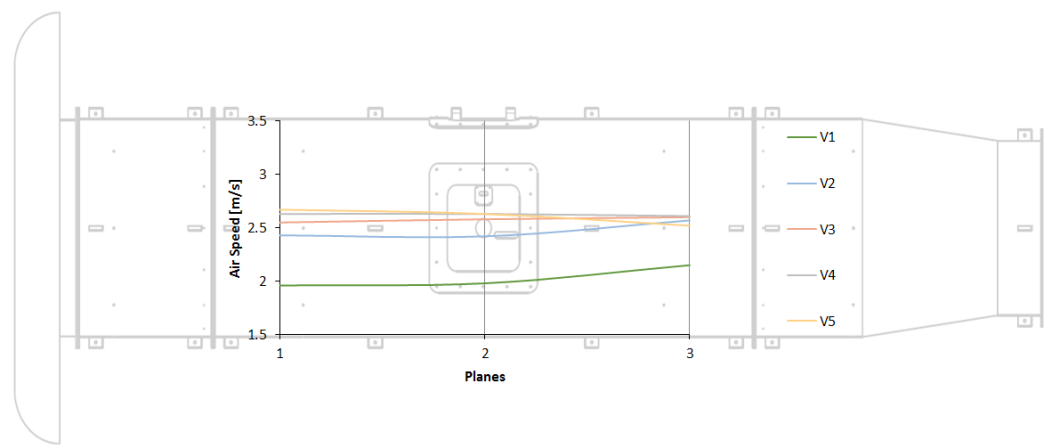
Plane 2		10Hz, 21C, 73% (10/8/2015)			20Hz, 22C, 71% (10/8/2015)			30Hz, 22.1C, 58% (10/8/2015)			40Hz, 22C, 68.5% (10/8/2015)			50Hz, 19C, 54% (11/8/2015)		
		max	min	avg	max	min	avg	max	min	avg	max	min	avg	max	min	avg
Side	2	2.78	2.19	2.53	5.14	4.49	4.98	7.79	6.78	7.53	10.43	9.95	10.23	13.07	12.61	12.92
	3	2.72	2.27	2.51	5.21	4.81	5.02	7.72	7.27	7.51	10.42	9.91	10.1	13.14	12.9	13.03
	4	2.89	2.09	2.52	5.46	4.62	5	7.68	6.69	7.45	10.2	9.74	10.04	13.16	12.56	12.96
	5	2.94	1.73	2.46	5.23	4.59	5.01	7.94	6.69	7.56	10.32	9.43	10.1	13.18	12.69	13.08
	6	2.24	1.33	1.97	4.6	3.49	4.23	6.95	5.14	6.33	9.78	8.24	8.87	12.55	11.22	11.95
Top	2	3.13	2.32	2.63	5.33	4.68	5.24	8.15	7.56	7.96	11	9.57	10.68	13.71	13.34	13.53
	3	3.22	2.26	2.63	5.51	4.91	5.23	8.33	7.42	7.98	10.9	10.4	10.72	13.69	13.41	13.58
	4	2.83	1.87	2.58	5.42	4.74	5.2	8.17	7.5	7.93	10.84	10.42	10.68	13.82	12.34	13.52
	5	2.66	1.62	2.42	5.52	4.52	5.23	8.26	7.31	7.9	10.91	9.96	10.68	13.07	13.4	13.54
	6	2.25	1.57	1.98	4.46	3.41	4.15	7.57	5.89	6.65	9.62	7.58	9.23	12.21	11.18	11.76

M.1.3 Plane 3

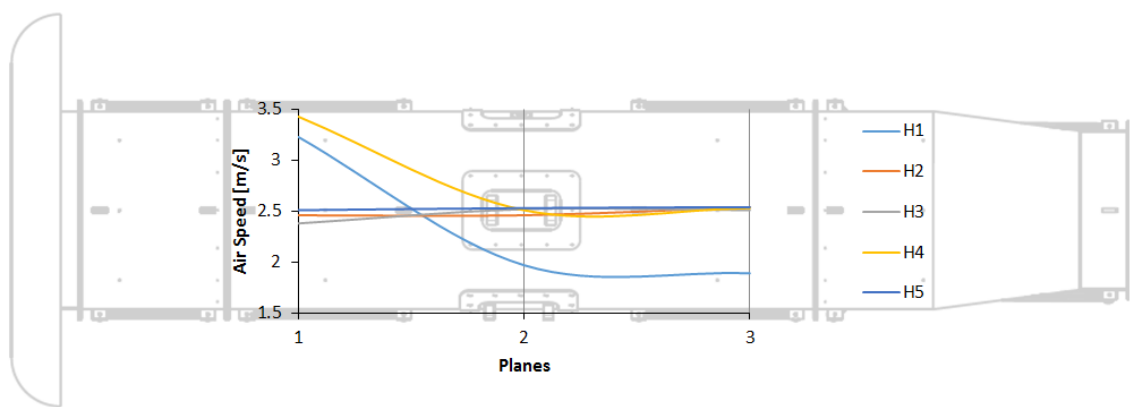
Plane 3 11/8/2015		10Hz, 21C, 48%			20Hz, 21C, 49%			30Hz, 21C, 49%			40Hz, 20.5C, 48%			50Hz, 20C, 52%		
		max	min	avg	max	min	avg	max	min	avg	max	min	avg	max	min	avg
Side	1	2.0														10.2
		7	1.76	1.89	4.07	3.56	3.9	6.15	5.63	5.95	8.35	7.85	8.17	10.52	10.089	5
	2	2.7	2.39	2.54	5.24	4.99	5.09	7.81	7.39	7.75	10.52	9.97	10.32	13.13	12.77	13
		2.6														12.9
	3	3	2.29	2.51	5.13	4.97	5.09	7.73	7.51	7.64	10.48	10.02	10.3	13.09	12.85	5
		2.6														13.0
	4	7	2.48	2.53	5.16	5.01	5.09	7.76	7.61	7.68	10.37	10.25	10.32	13.15	12.81	2
		2.6														12.9
	5	9	2.38	2.54	5.18	4.88	5.06	7.68	7.09	7.58	10.41	10.03	10.26	13.14	12.68	8
		2.3														12.0
Top	1	6	1.65	2.15	4.45	3.26	4.18	7.1	6.32	6.77	9.53	8.84	9.2	12.45	11.18	7
		2.6														13.6
	2	9	2.13	2.57	5.42	5.21	5.32	8.23	7.87	8.08	11	10.63	10.89	13.84	13.33	3
		2.6														13.7
	3	8	2.46	2.6	5.33	5.16	5.27	8.21	7.54	8.01	11.06	10.75	10.94	13.99	13.44	4
		2.8														13.7
	4	3	2.47	2.61	5.42	5.2	5.34	8.16	7.89	8.01	11.06	10.24	10.93	13.9	13.5	5
																13.6
	5	2.8	2.2	2.52	5.4	4.93	5.3	8.2	7.31	8.07	11	10.57	10.98	13.97	12.93	4

Appendix N – Flow Channel Air Speed Graphs

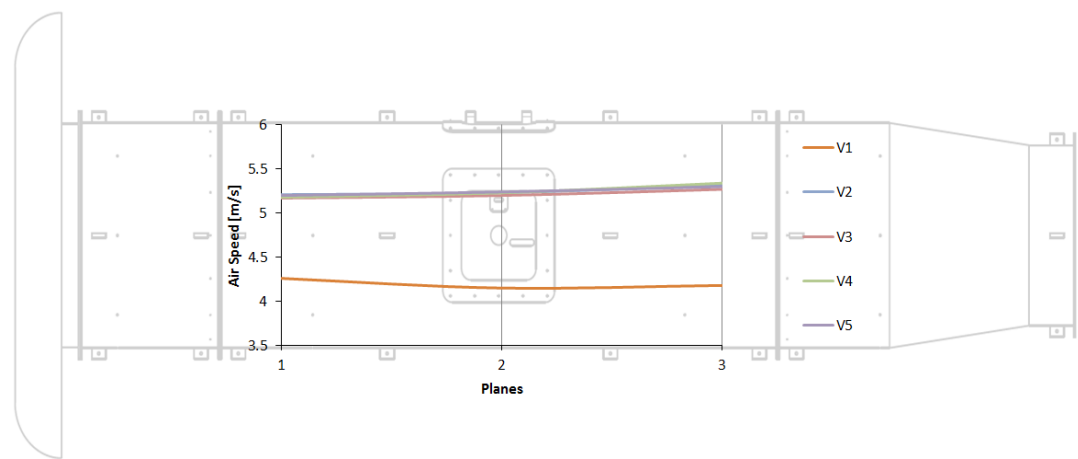
10 Hz – Average Values (In Vertical Plane)



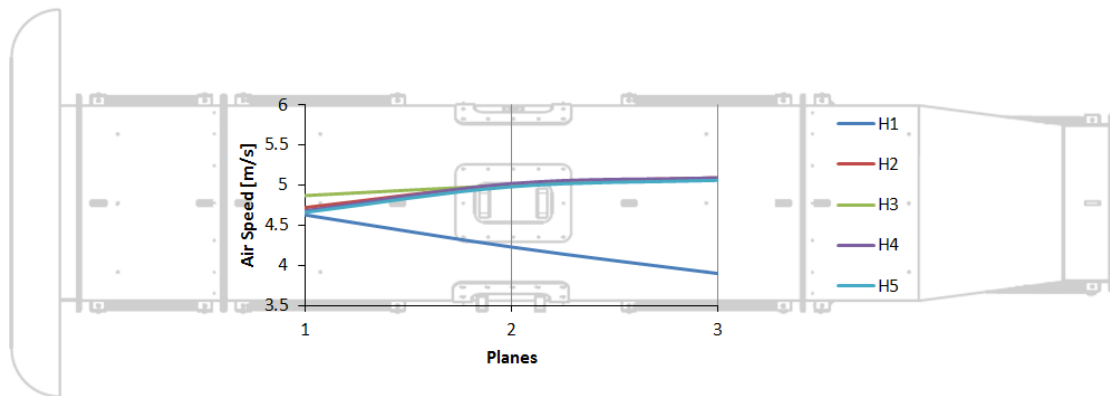
10 Hz – Average Values (In Horizontal Plane)



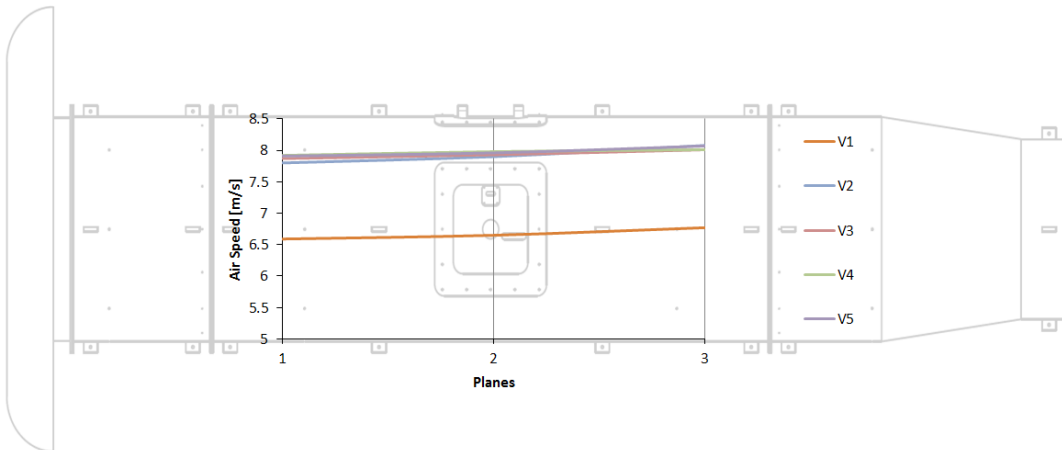
20 Hz – Average Values (In Vertical Plane)



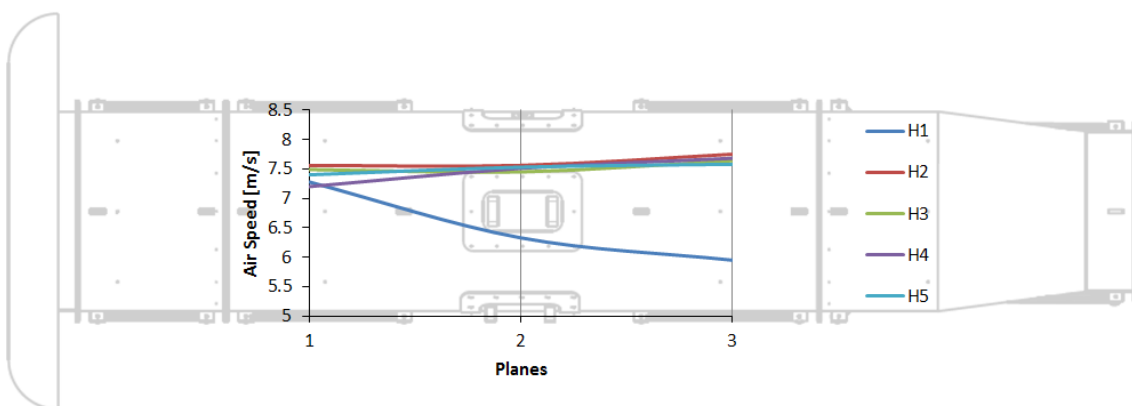
20 Hz – Average Values (In Horizontal Plane)



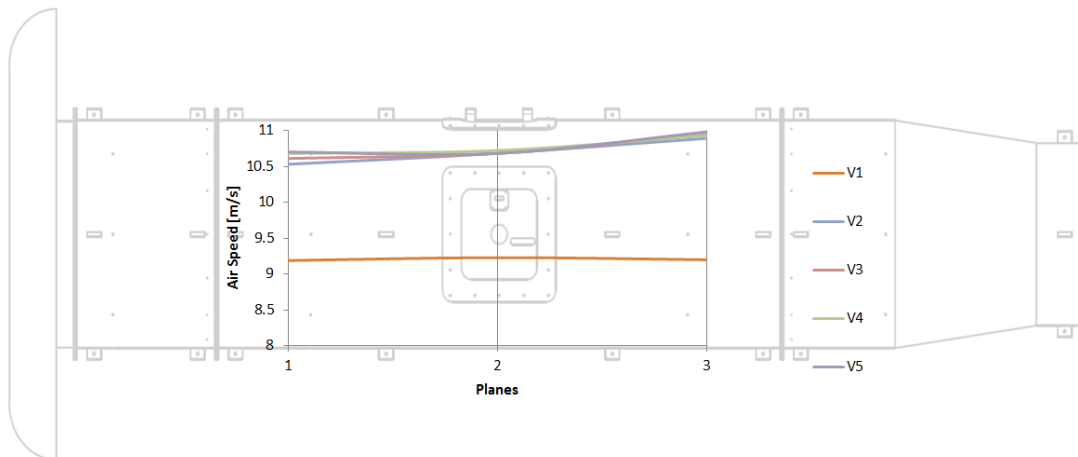
30 Hz – Average Values (In Vertical Plane)



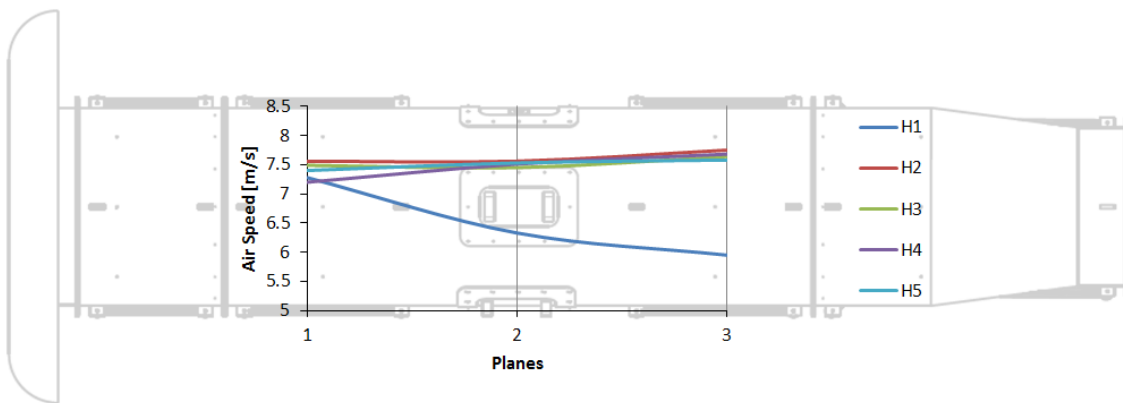
30 Hz – Average Values (In Horizontal Plane)



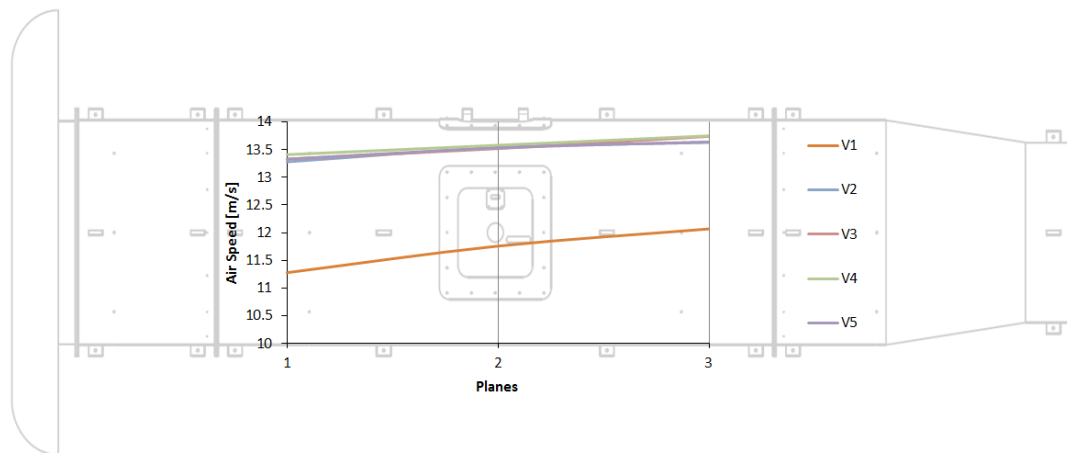
40 Hz – Average Values (In Vertical Plane)



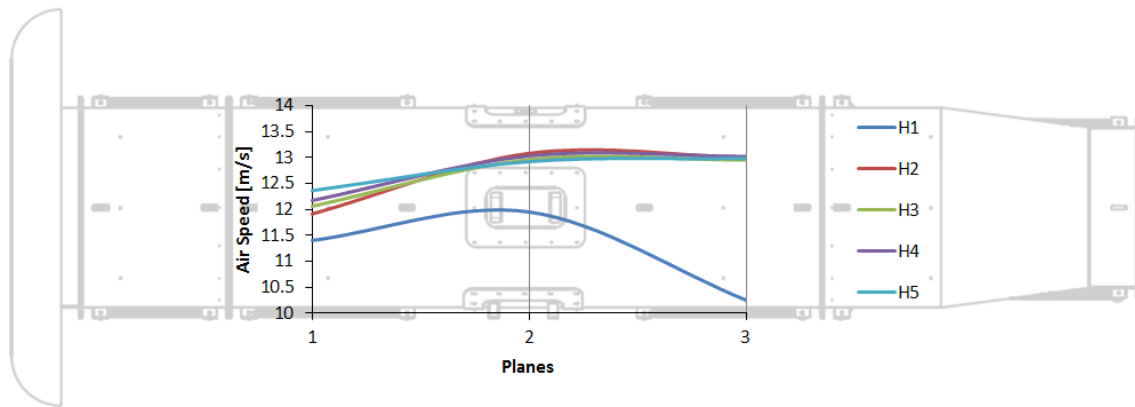
30 Hz – Average Values (In Horizontal Plane)



50 Hz – Average Values (In Vertical Plane)

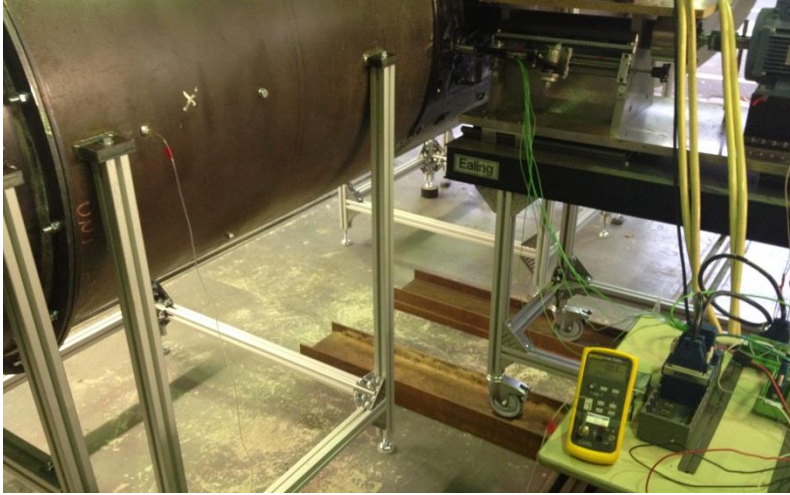


50 Hz – Average Values (In Horizontal Plane)



Appendix O – Static Pressure Tests

O.1 Point A



O.2 Point B



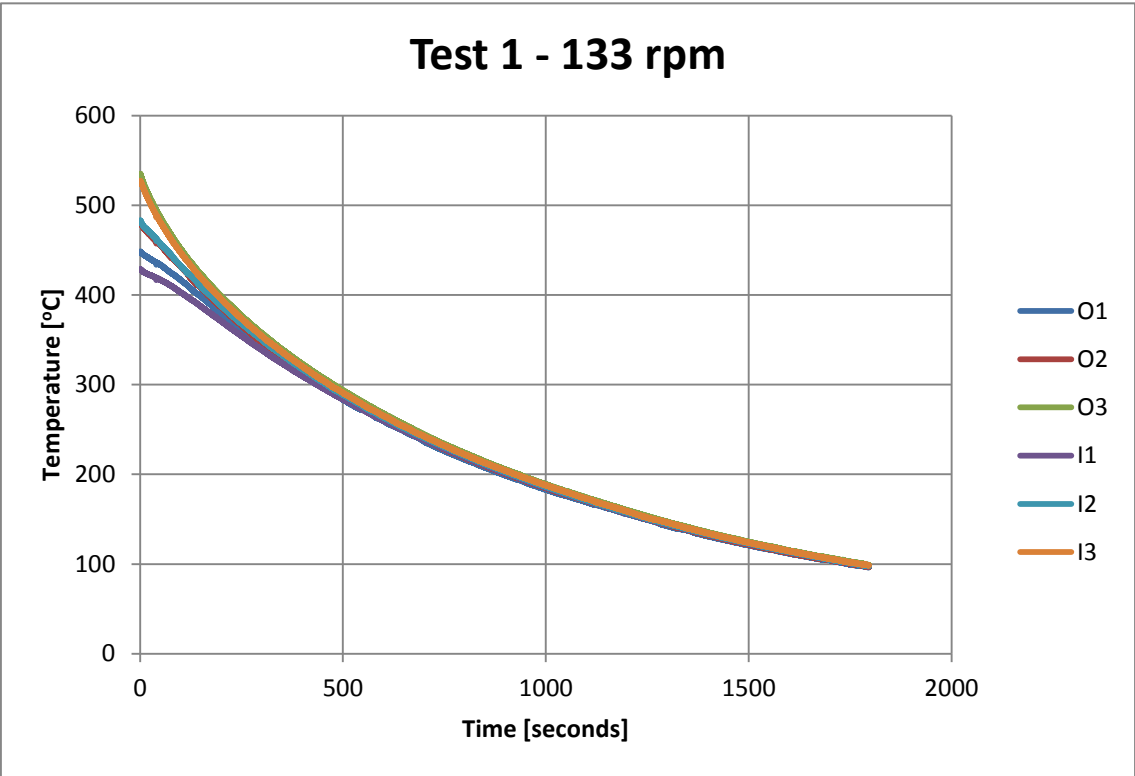
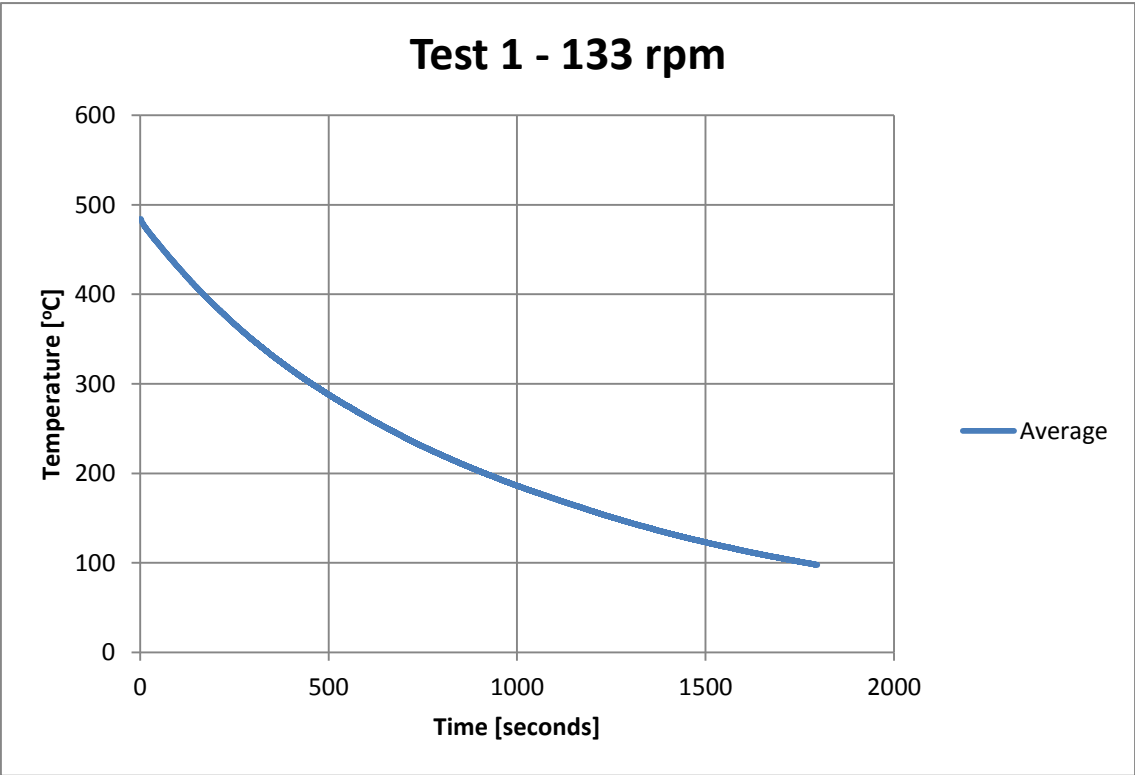
O.3 Point C

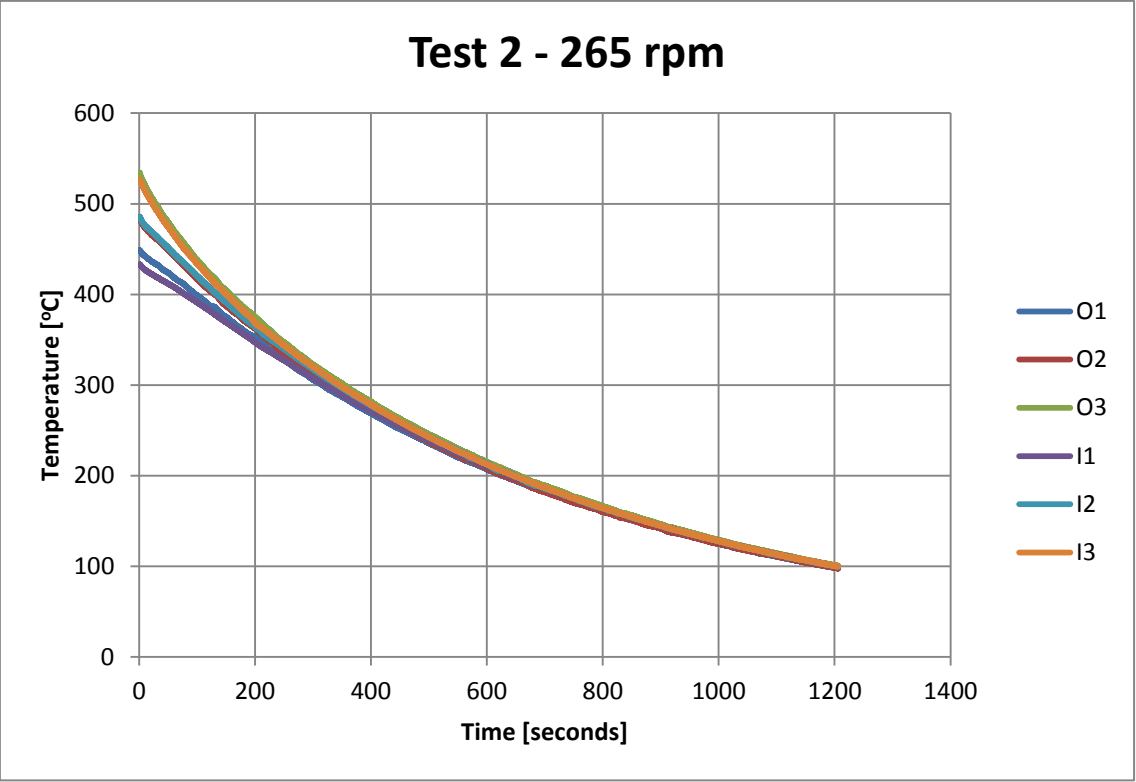
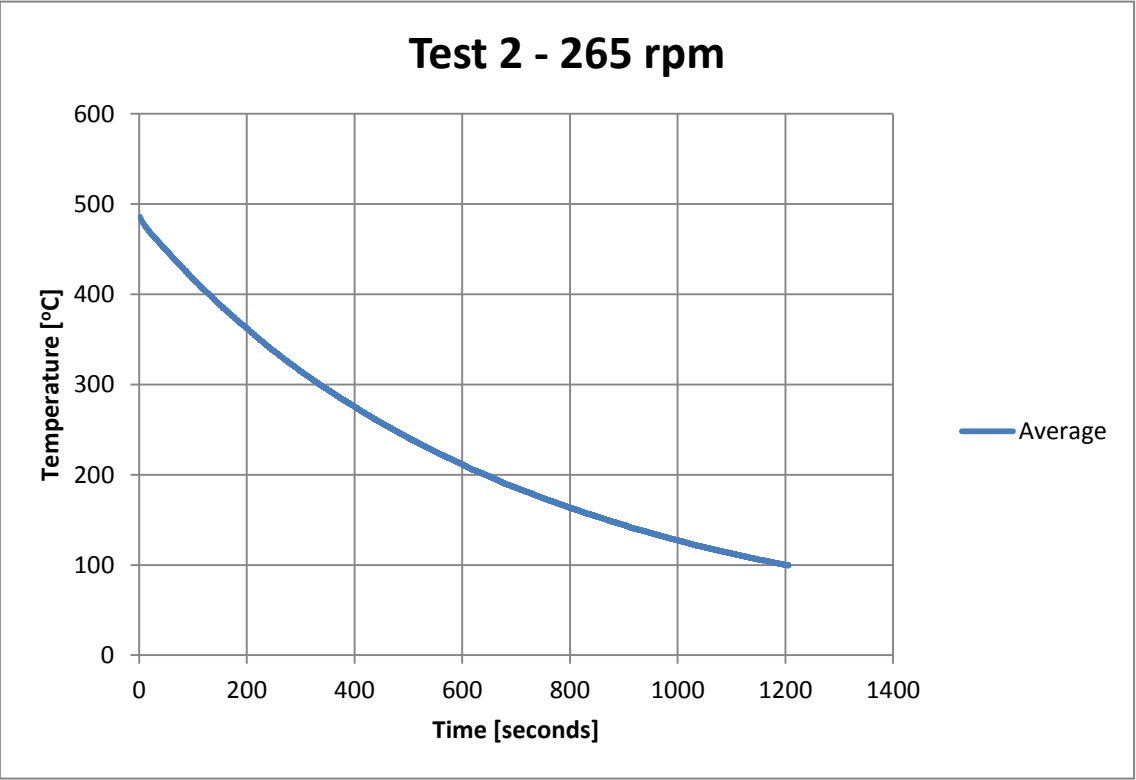


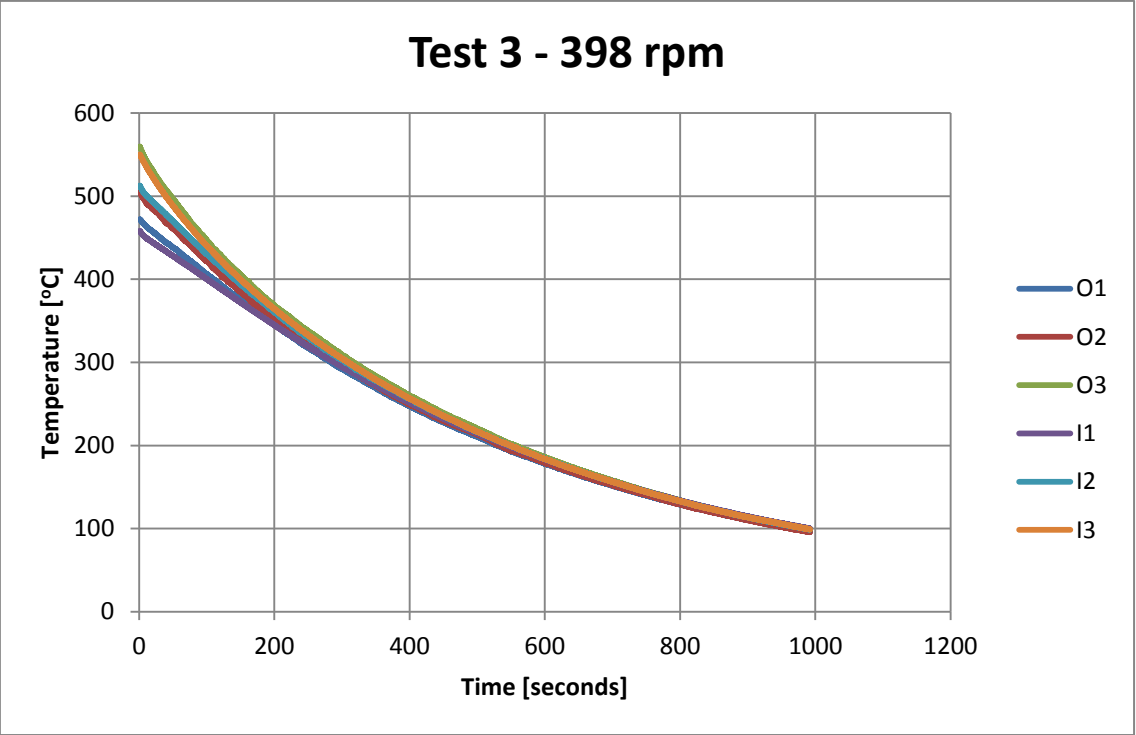
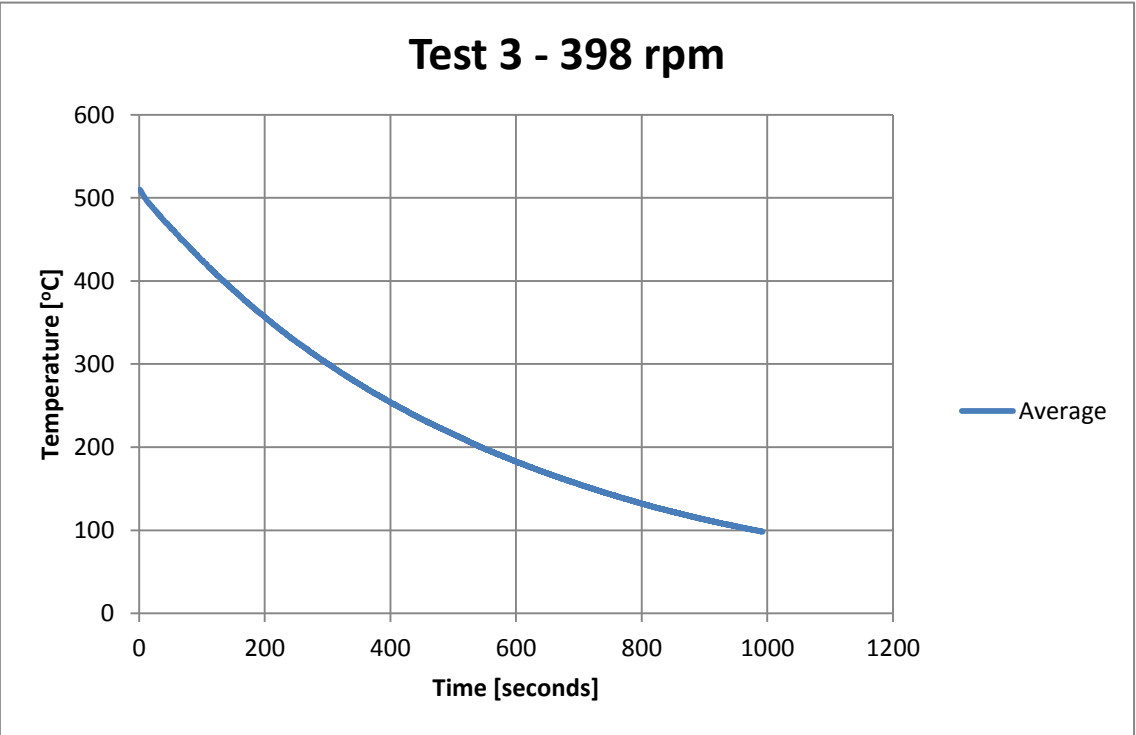
O.4 Point D

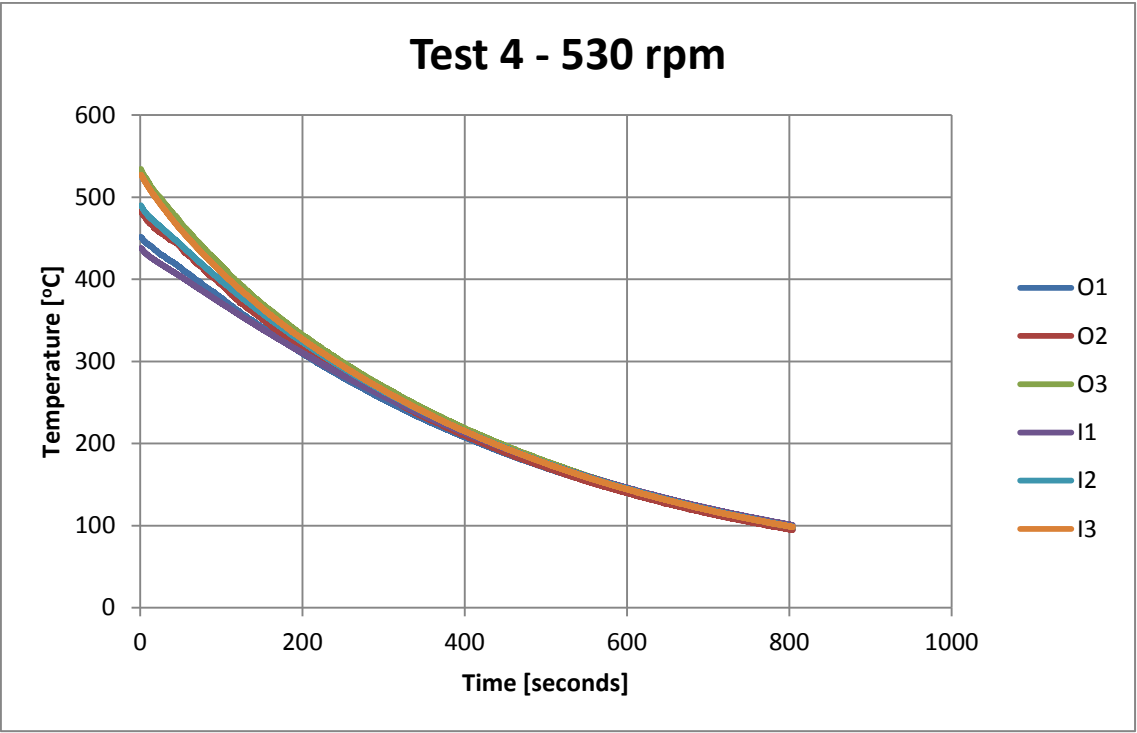
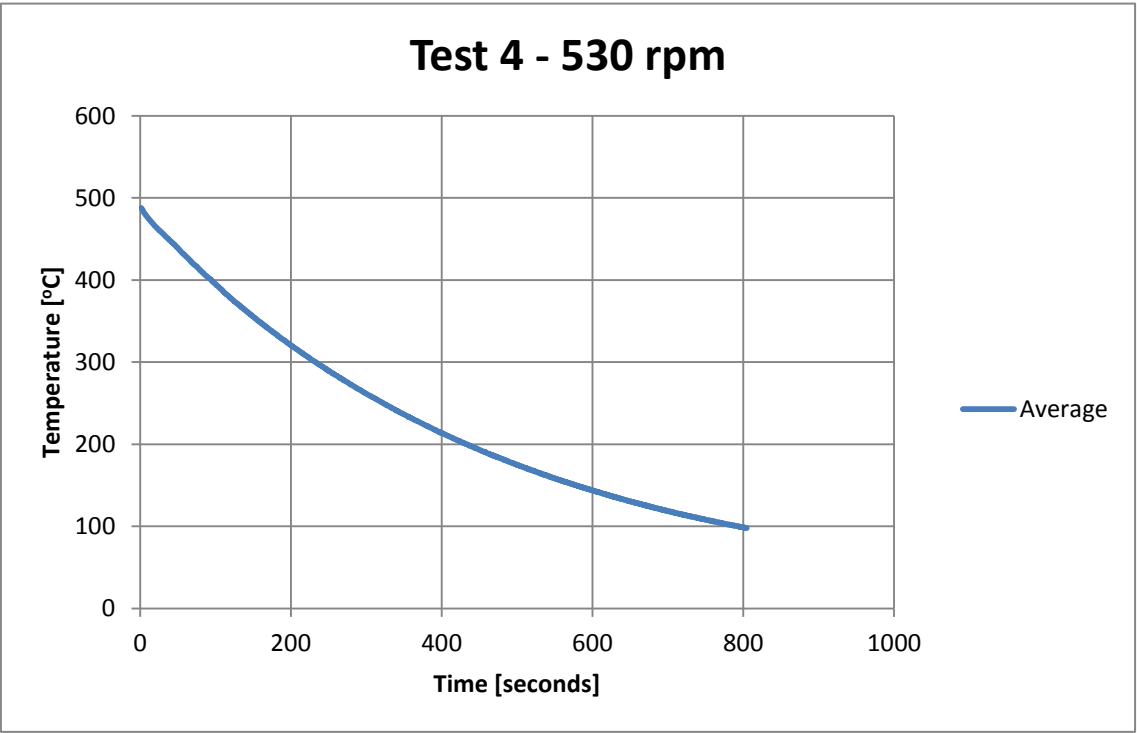


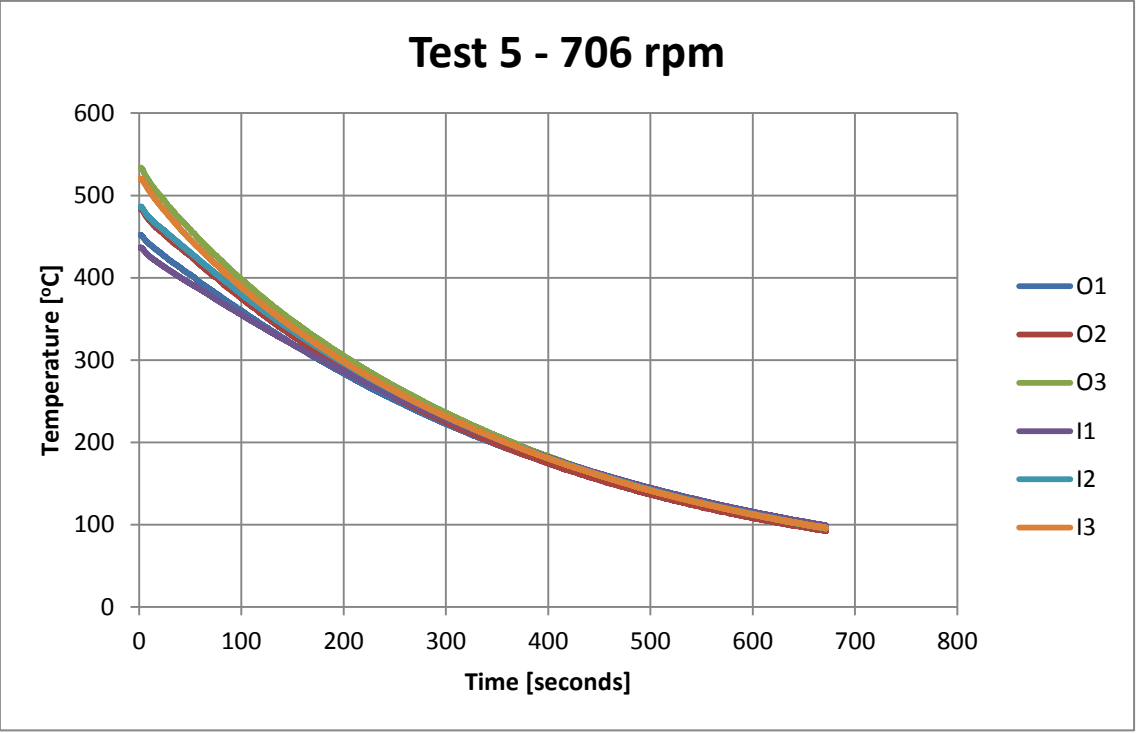
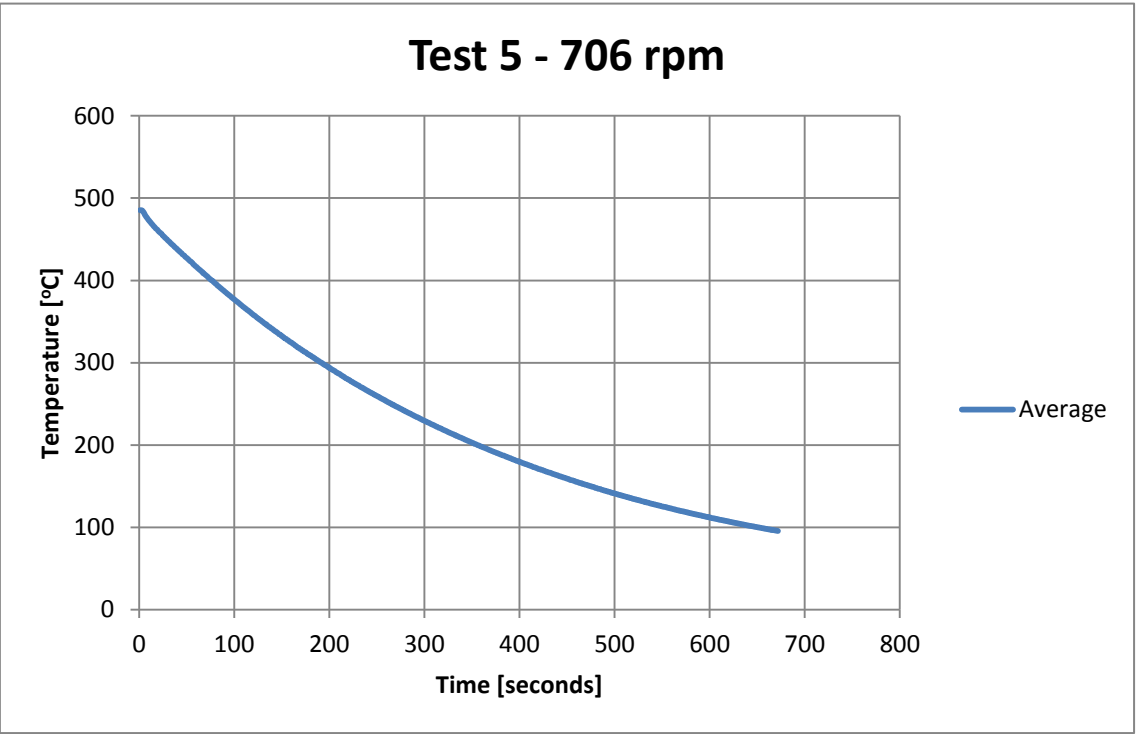
Appendix P – Cooling Curves Graphs











Appendix Q – Flir i3 Thermal Imaging Camera Specifications



Table 0-4 Flir i3 Specification

Manufacturer	Flir Systems
Model	i3 Infrared Camer
Resolution	60 x 60 pixels
Total Pixels	3600
Thermal Sensitivity	<0.15°C
Accuracy	± 2% or 2°C
Temperature Range	-20 °C to 250 °C
Field of View	12.5 ° x 12.5 °
Focus	Focus Free
Detector	Uncooled Microbolometer
Screen	2.8” Colour LCD
Frame Rate	9 Hz
Emissivity	0.1 to 1.0 Adjustable

Appendix R – Air Properties

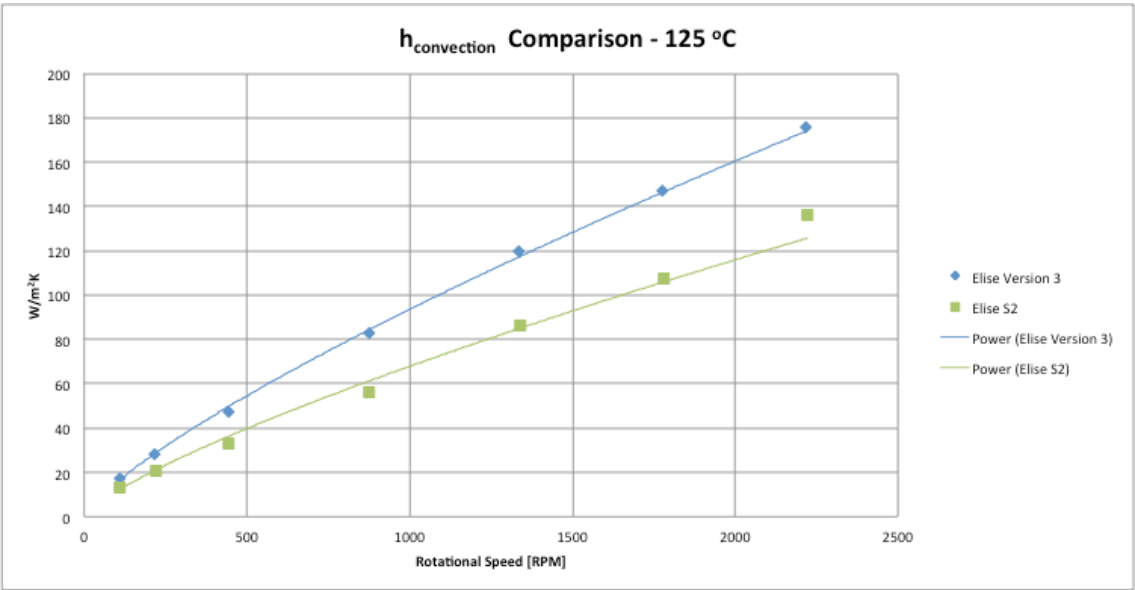
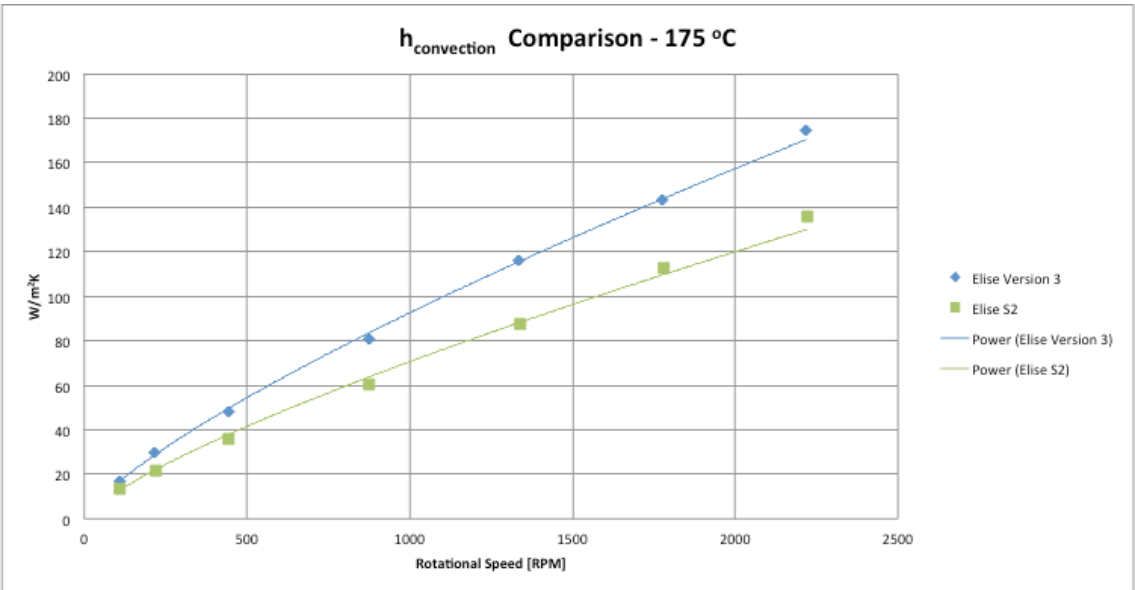
Table 0-5 Thermal Conductivity

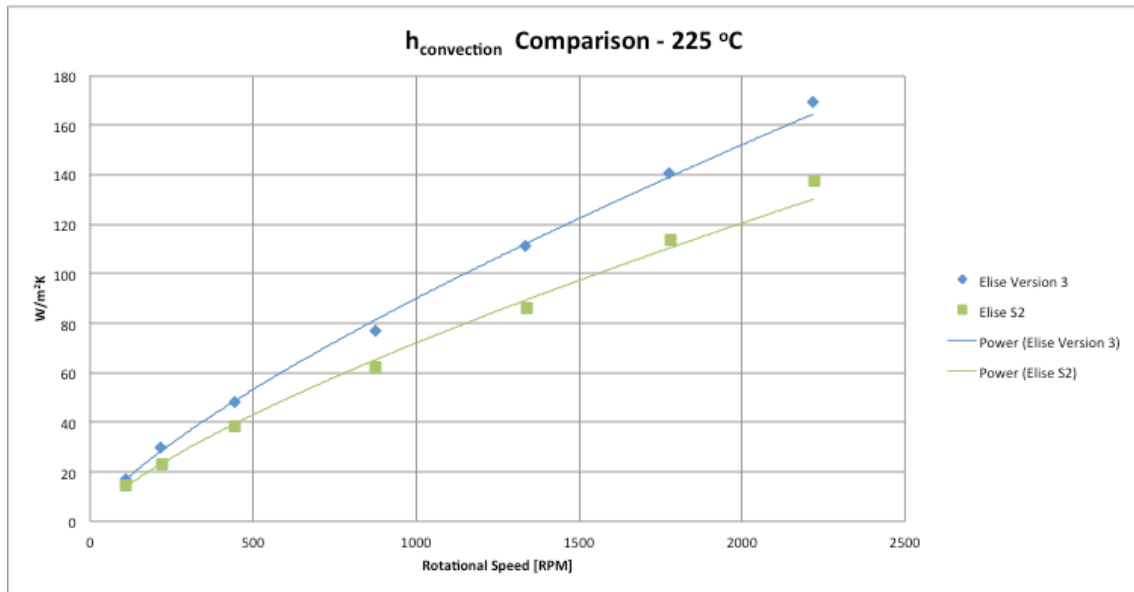
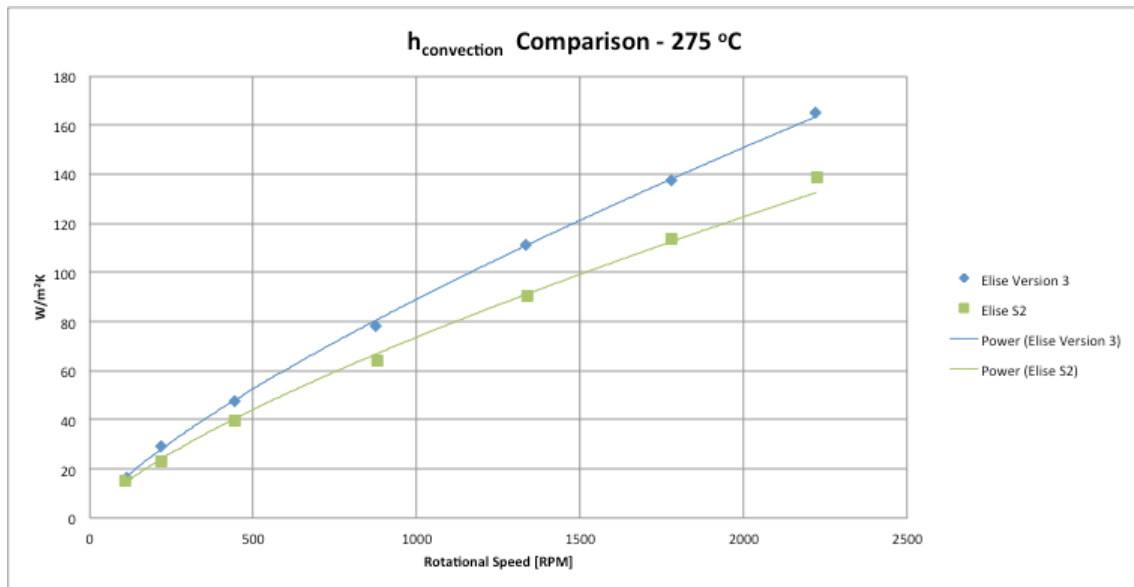
Temperature [°C]	Thermal Conductivity [W/mK]
105	0.0319
155	0.0354
205	0.0388
255	0.0422
305	0.0457
355	0.0491

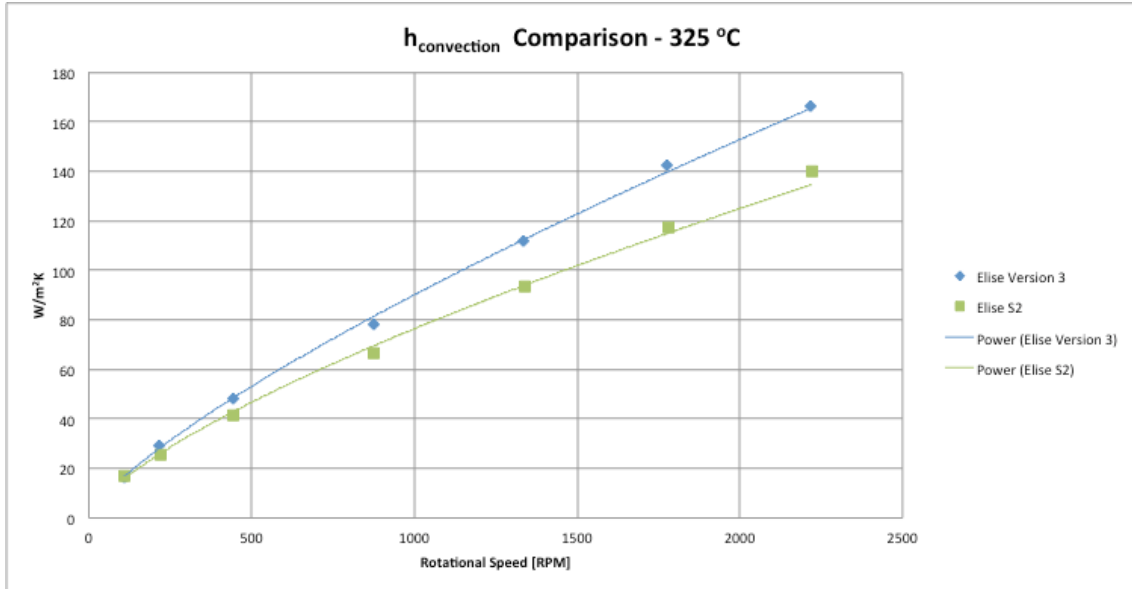
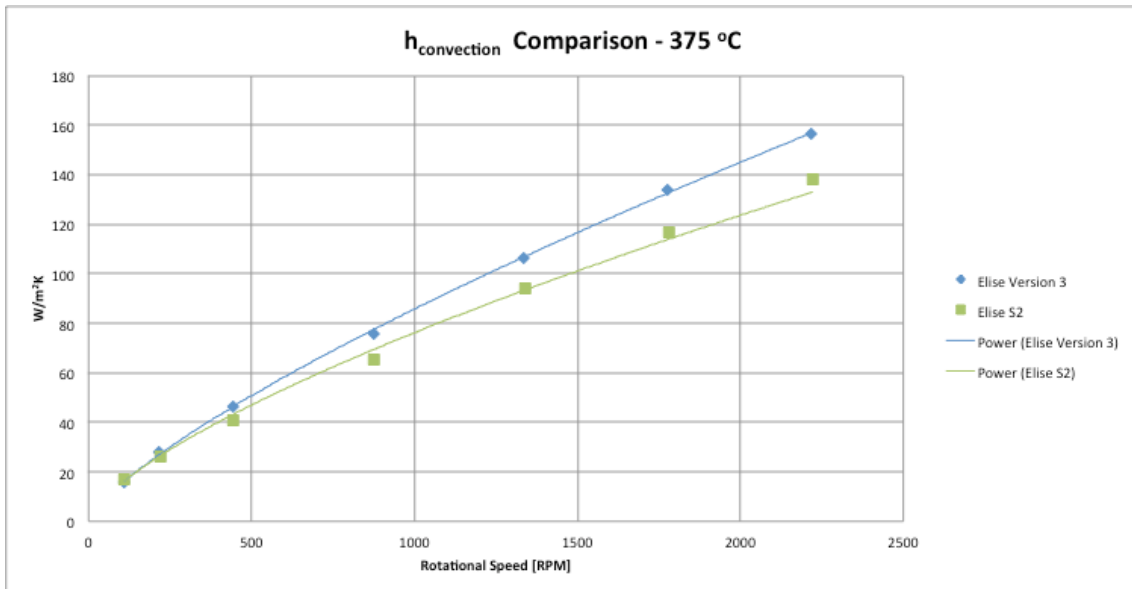
Table 0-6 Kinematic Viscosity

Temperature [°C]	Kinematic Viscosity [m ² /s]
105	0.0000205
155	0.0000255
205	0.0000305
255	0.0000355
305	0.0000405
355	0.0000455

Appendix S Convective Heat Transfer Coefficients in Still Air

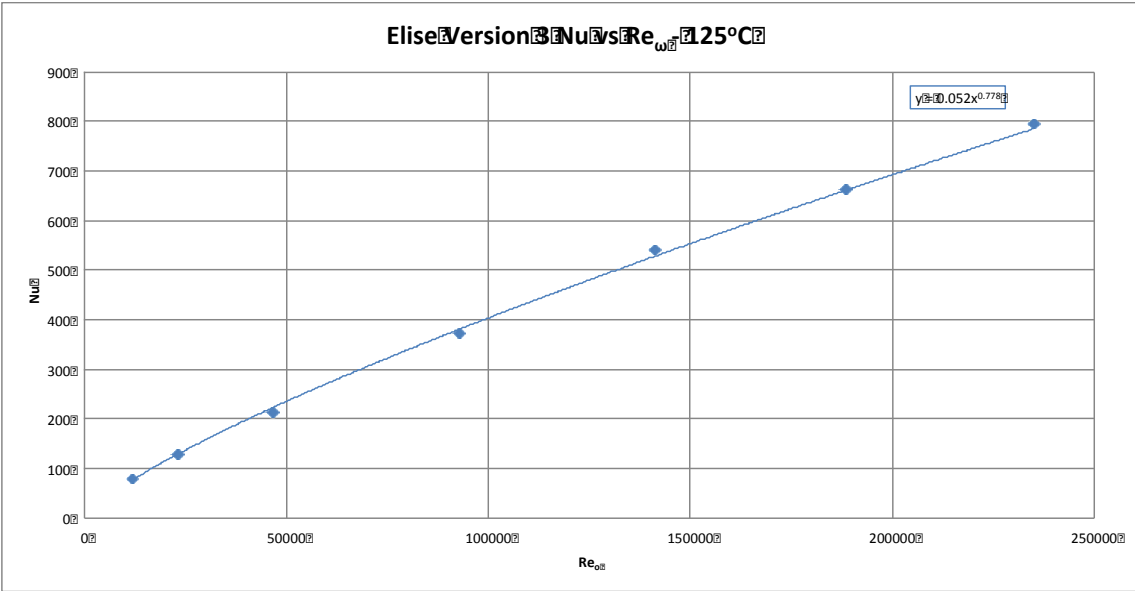
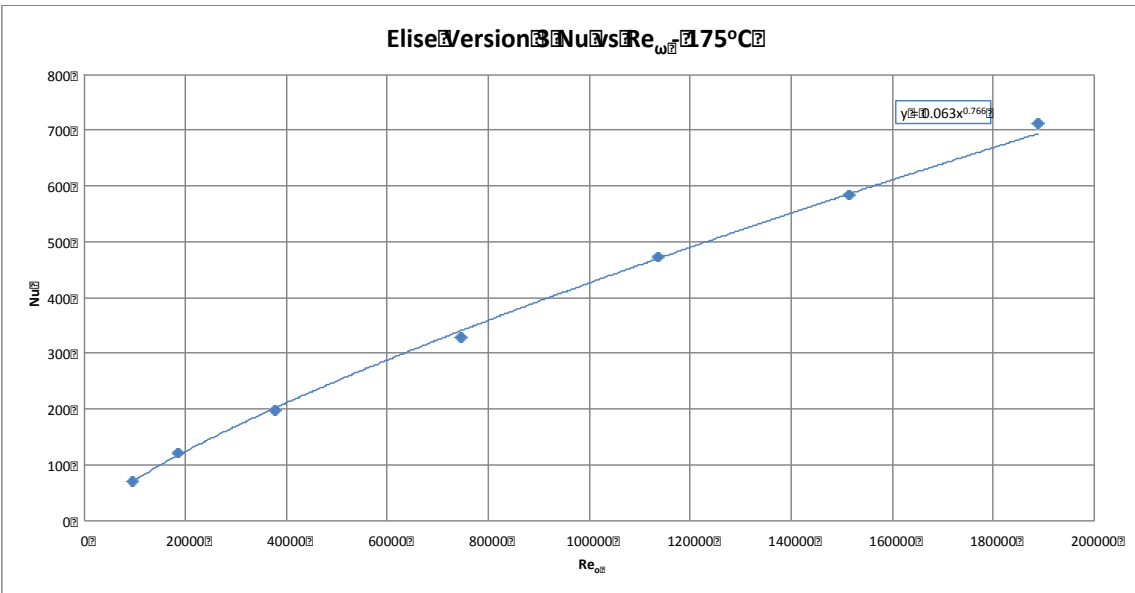


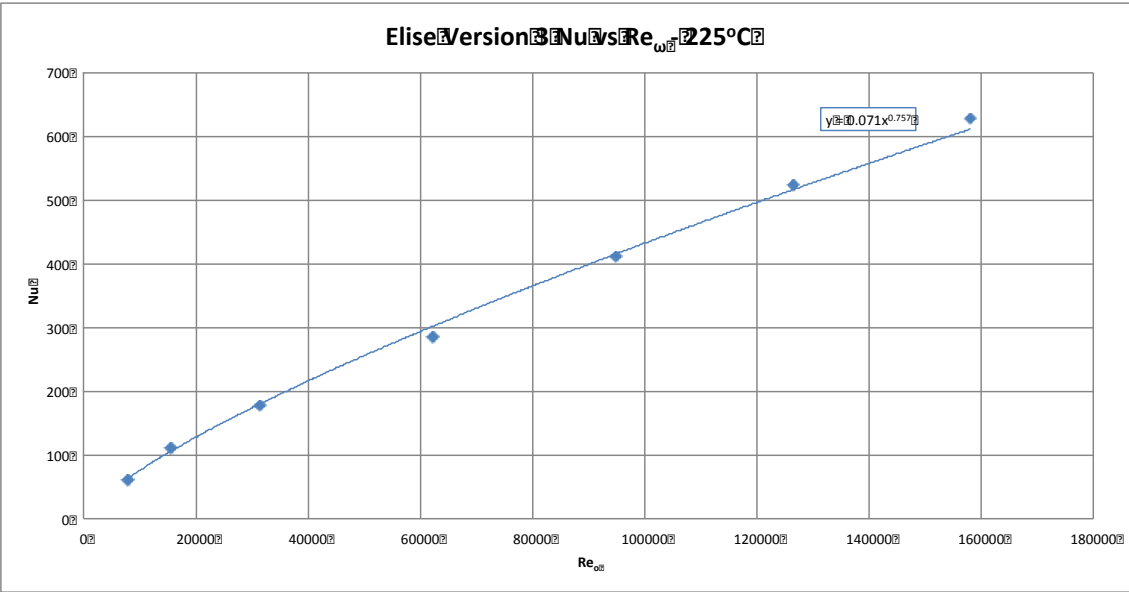
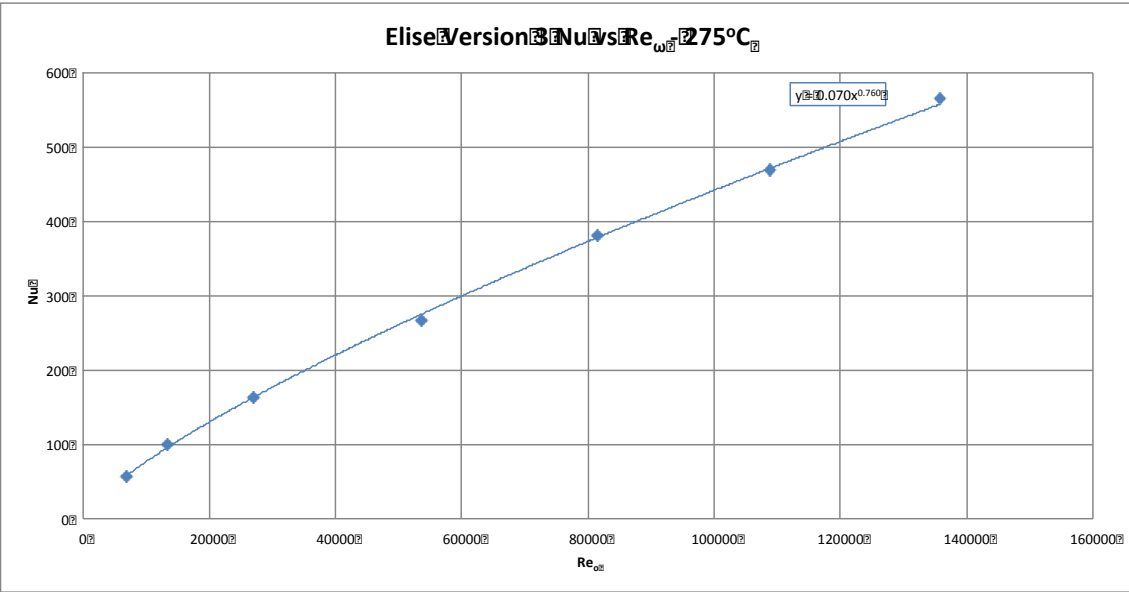


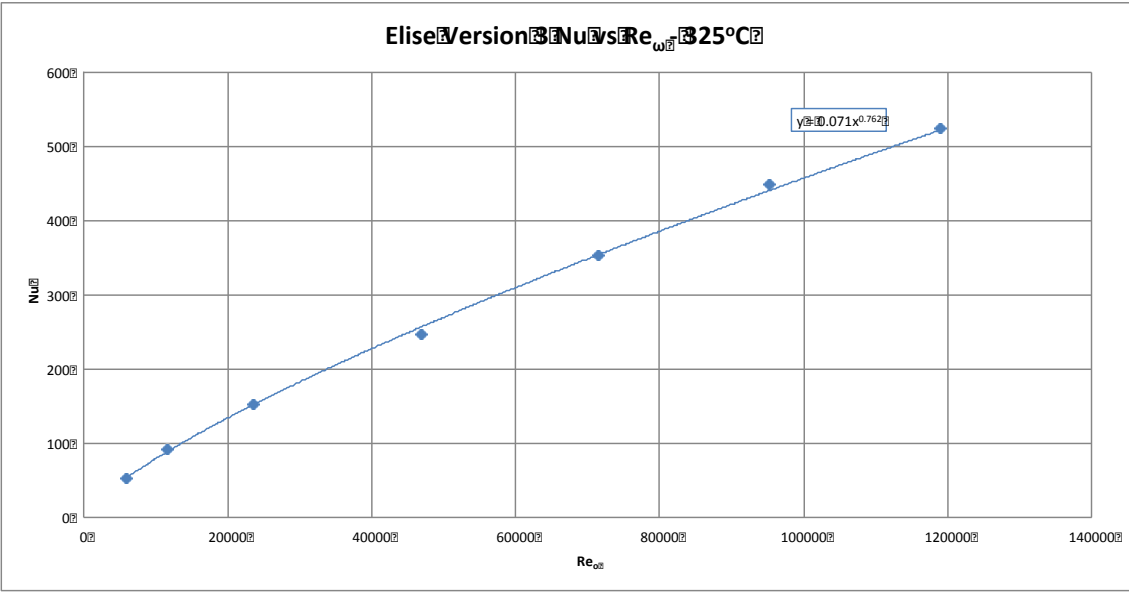
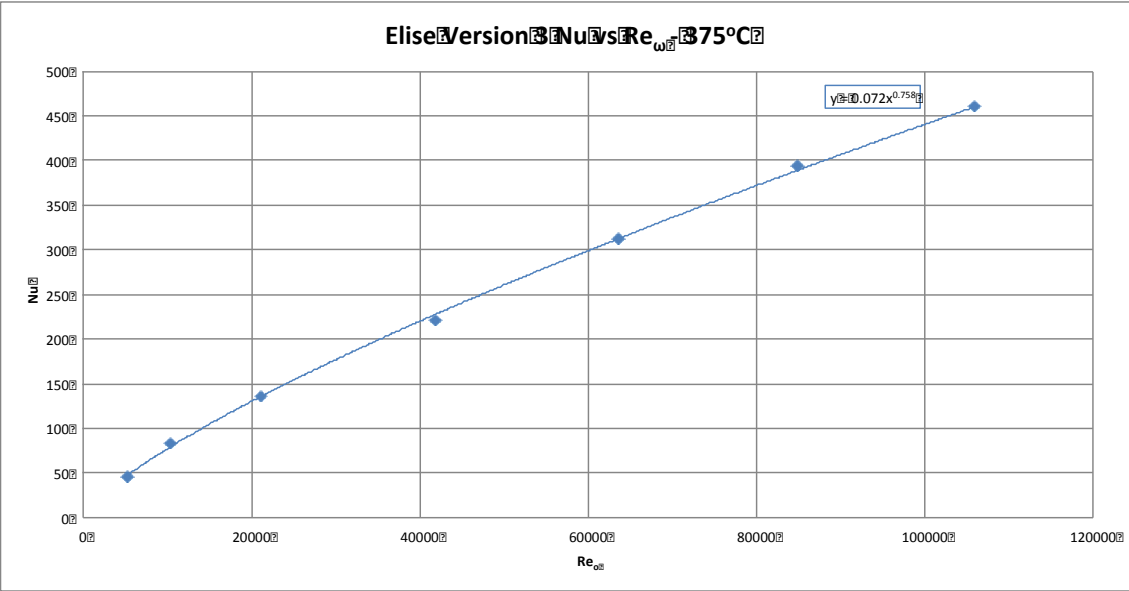


Appendix T – Nu vs Re_ω Graphs in Still Air

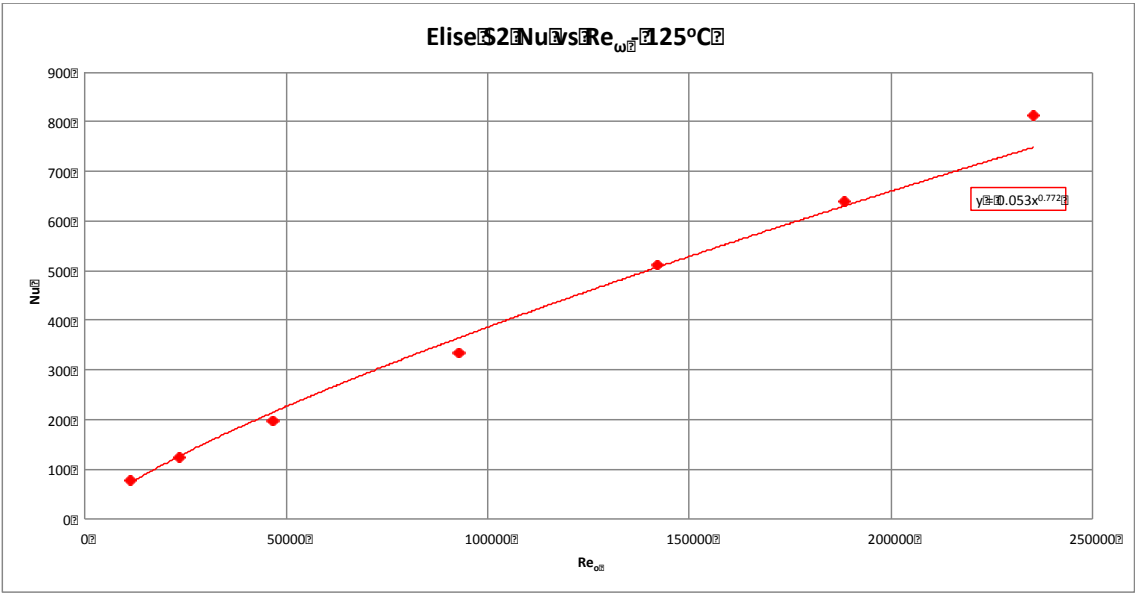
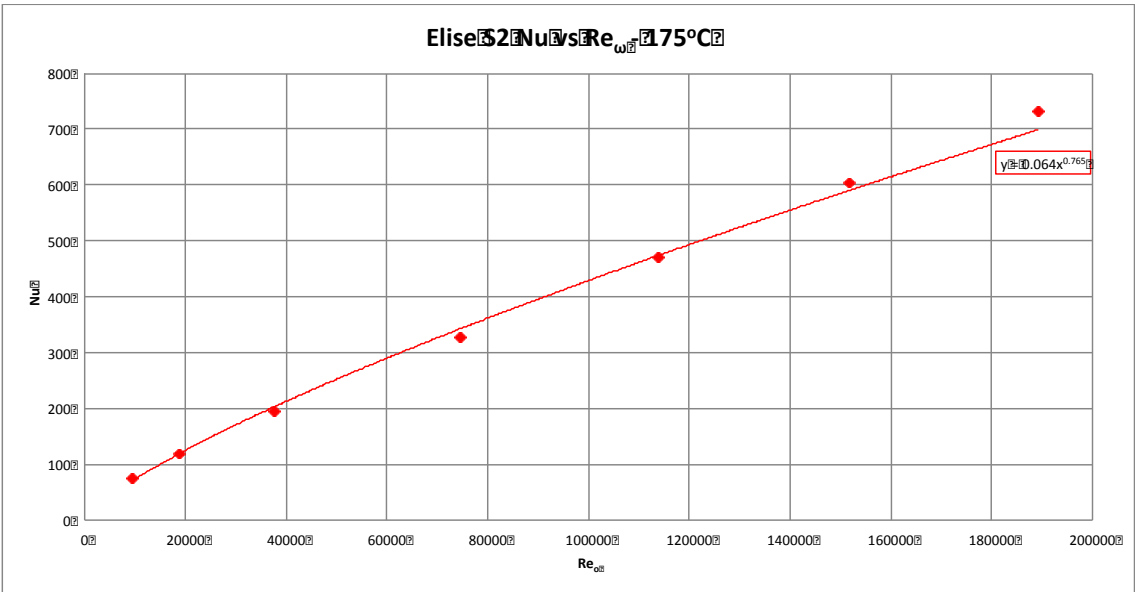
T.1 Elise Version 3

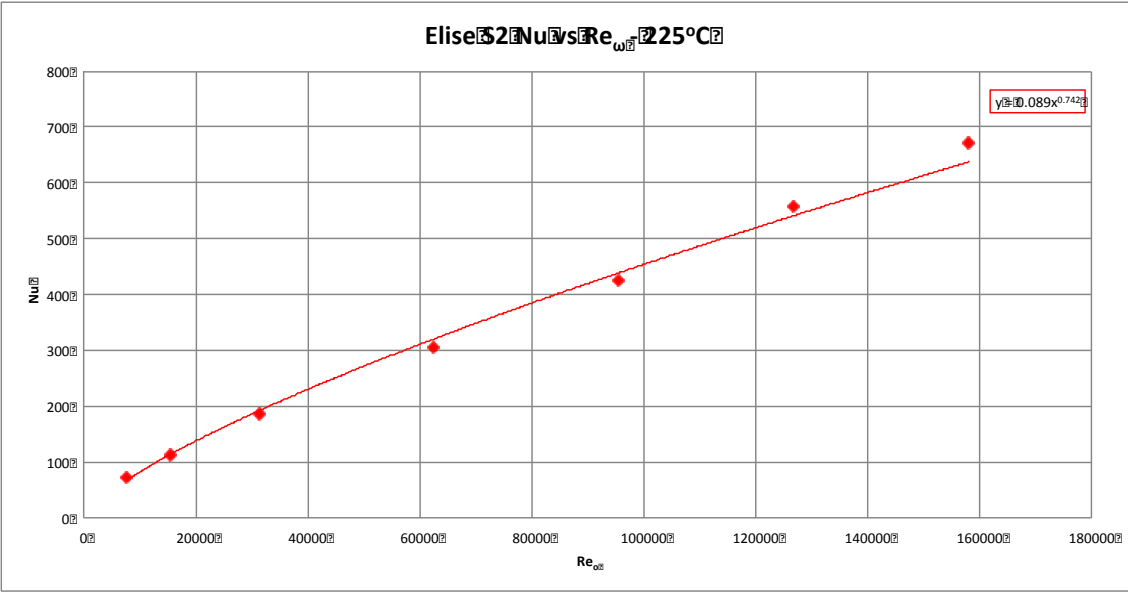
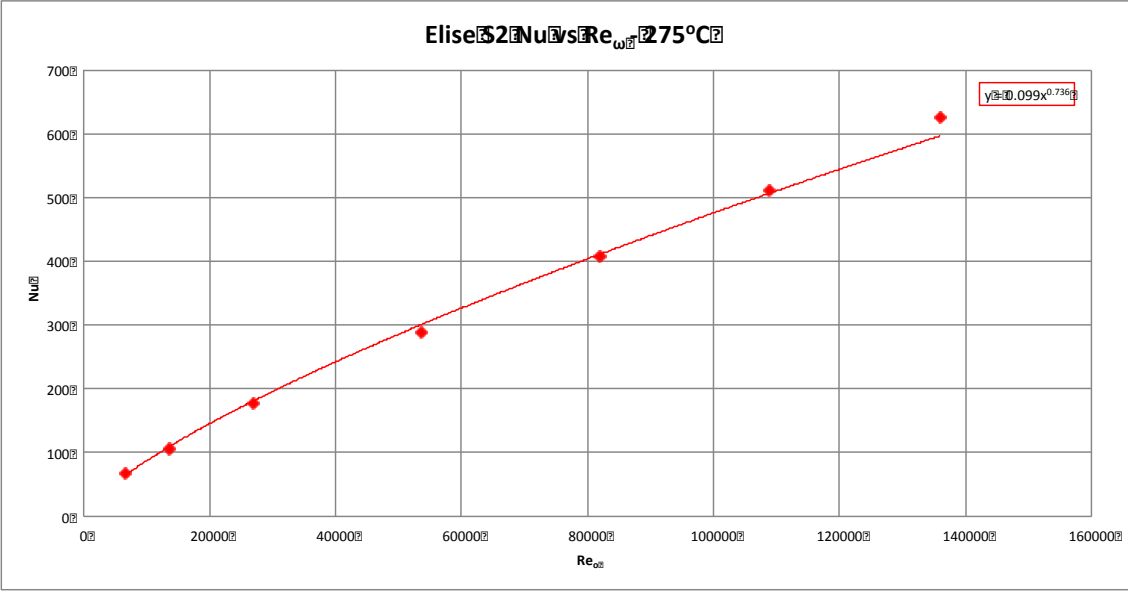


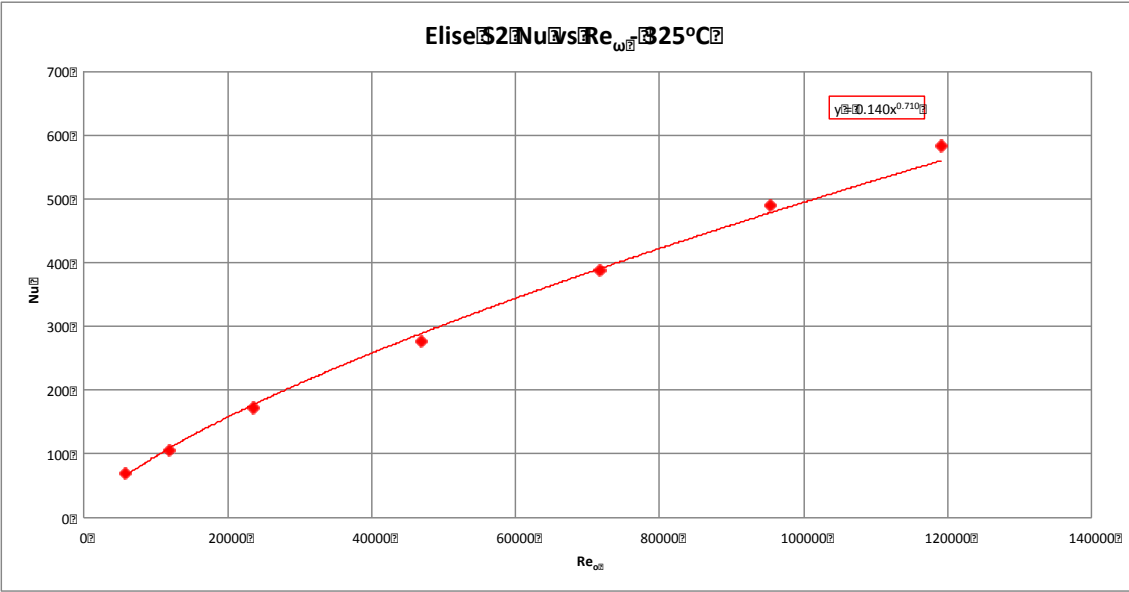
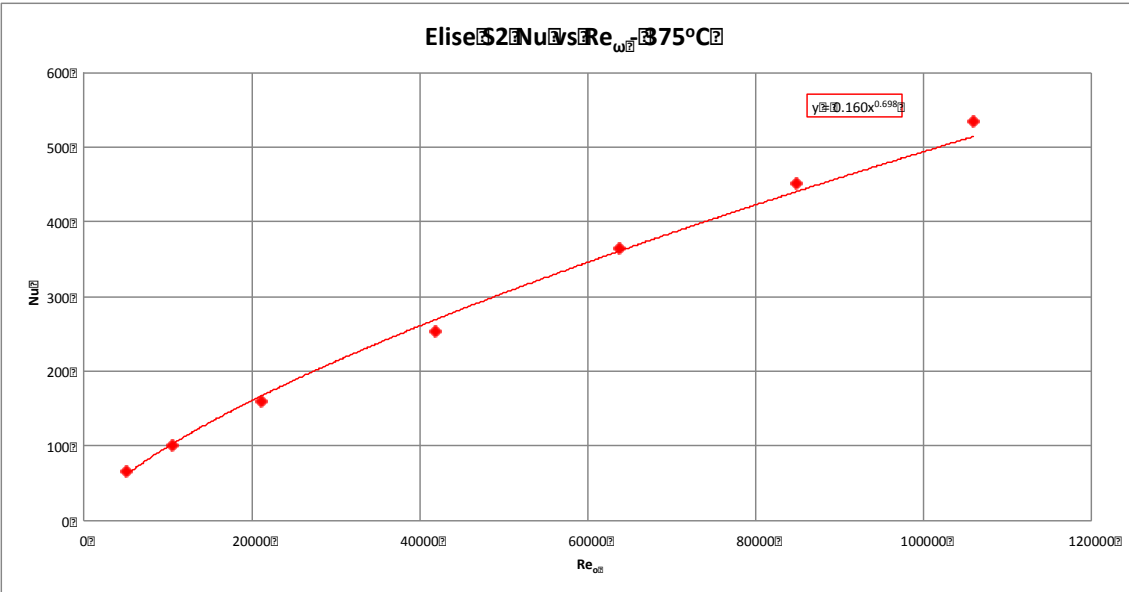




T.2 Elise S2







Appendix U – Cross Flow Test Schedule

Test No	Test Name	Nominal Air Velocity [km/h]	Nominal Disc/Wheel Speed [km/h]	Angle [deg]
1	25_25_0	25	25	0
2	25_37,5_0	25	37.5	0
3	25_50_0	25	50	0
4	25_75_0	25	75	0
5	31_31_0	31	31	0
6	31_46,5_0	31	46.5	0
7	31_62_0	31	62	0
8	31_93_0	31	93	0
9	37,5_37,5_0	37.5	37.5	0
10	37,5_56,2_0	37.5	56.2	0
11	37,5_75_0	37.5	75	0
12	37,5_112,5_0	37.5	112.5	0
13	44_44_0	44	44	0
14	44_66_0	44	66	0
15	44_88_0	44	88	0
16	44_134_0	44	132	0
17	50_50_0	50	50	0
18	50_75_0	50	75	0
19	50_100_0	50	100	0

Test No	Test Name	Nominal Air Velocity [km/h]	Nominal Disc/Wheel Speed [km/h]	Angle [deg]
20	50_150_0	50	150	0
21	25_25_10	25	25	10
22	25_37,5_10	25	37.5	10
23	25_50_10	25	50	10
24	25_75_10	25	75	10
25	31_31_10	31	31	10
26	31_46,5_10	31	46.5	10
27	31_62_10	31	62	10
28	31_93_10	31	93	10
29	37,5_37,5_0	37.5	37.5	10
30	37,5_56,2_0	37.5	56.2	10
31	37,5_75_0	37.5	75	10
32	37,5_112,5_0	37.5	112.5	10
33	44_44_10	44	44	10
34	44_66_10	44	66	10
35	44_88_10	44	88	10
36	44_134_10	44	132	10
37	50_50_10	50	50	10
38	50_75_10	50	75	10

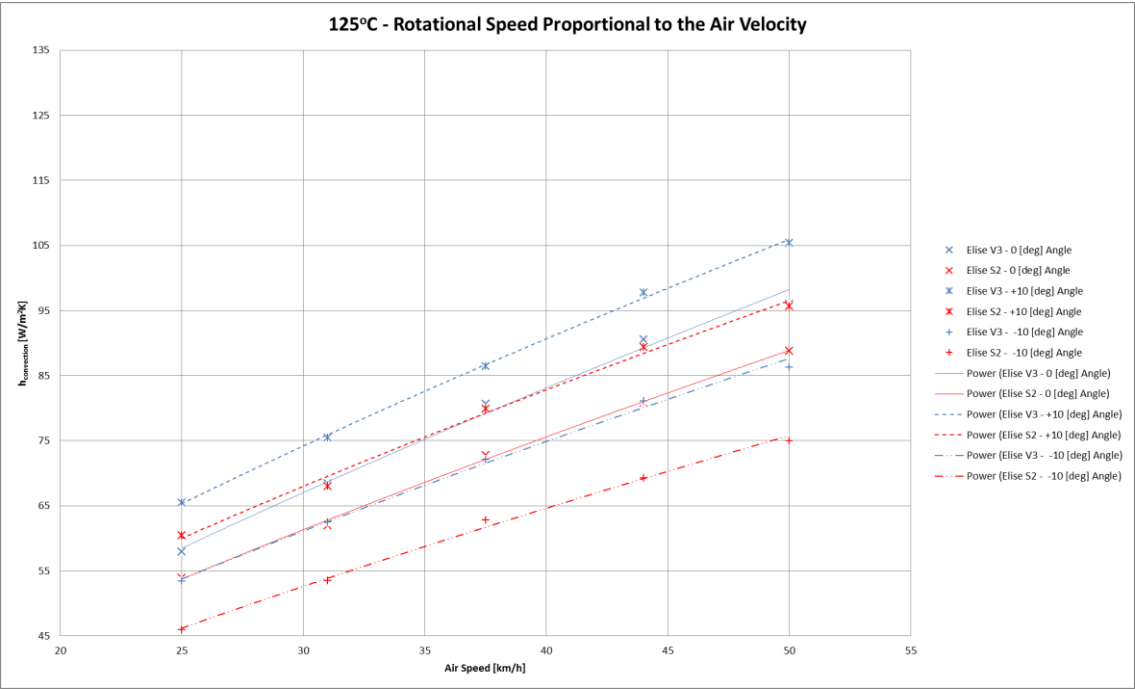
Test No	Test Name	Nominal Air Velocity [km/h]	Nominal Disc/Wheel Speed [km/h]	Angle [deg]
39	50_100_10	50	100	10
40	50_150_10	50	150	10
41	25_25_-10	25	25	-10
42	25_37,5_-10	25	37.5	-10
43	25_50_-10	25	50	-10
44	25_75_-10	25	75	-10
45	31_31_-10	31	31	-10
46	31_46,5_-10	31	46.5	-10
47	31_62_-10	31	62	-10
48	31_93_-10	31	93	-10
49	37,5_37,5_-10	37.5	37.5	-10
50	37,5_56,2_-10	37.5	56.2	-10
51	37,5_75_-10	37.5	75	-10
52	37,5_112,5_-10	37.5	112.5	-10
53	44_44_-10	44	44	-10
54	44_66_-10	44	66	-10
55	44_88_-10	44	88	-10
56	44_134_-10	44	132	-10
57	50_50_-10	50	50	-10

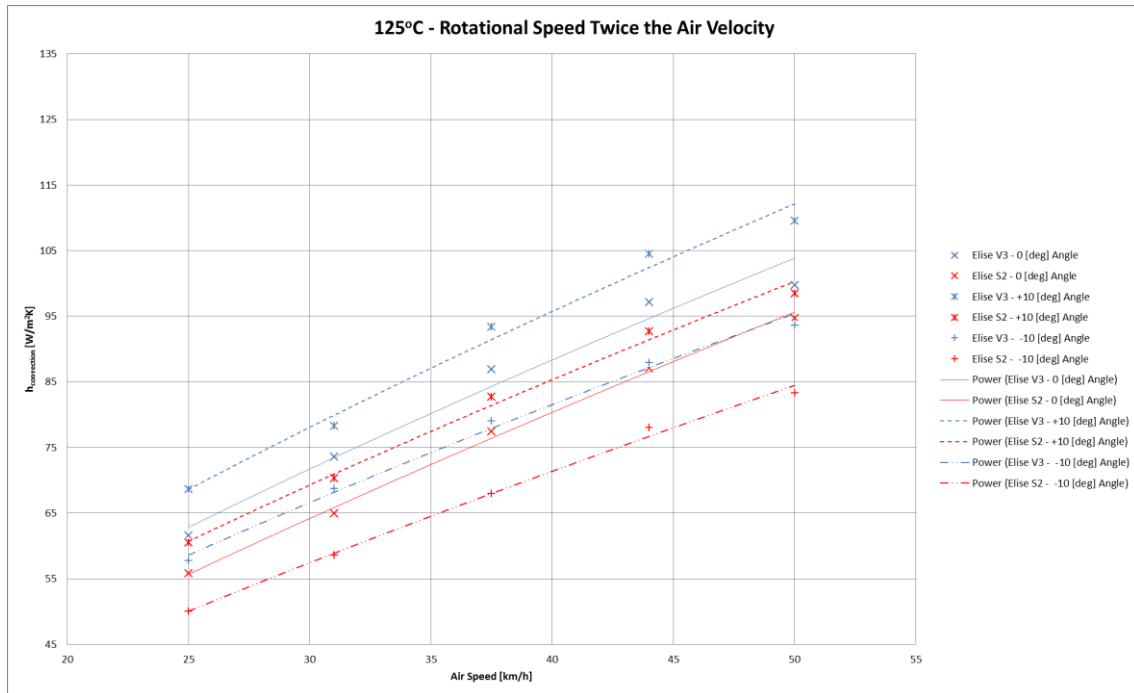
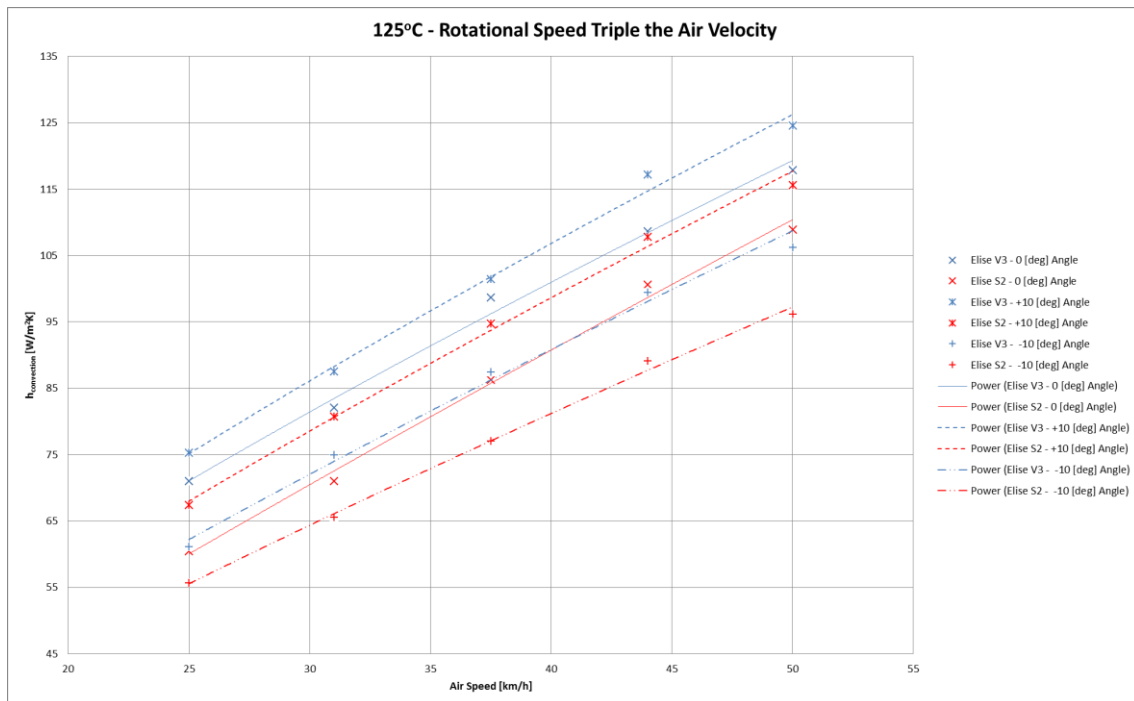
Test No	Test Name	Nominal Air Velocity [km/h]	Nominal Disc/Wheel Speed [km/h]	Angle [deg]
58	50_75_-10	50	75	-10
59	50_100_-10	50	100	-10
60	50_150_-10	50	150	-10

Appendix V – Influence of the Disc Angle

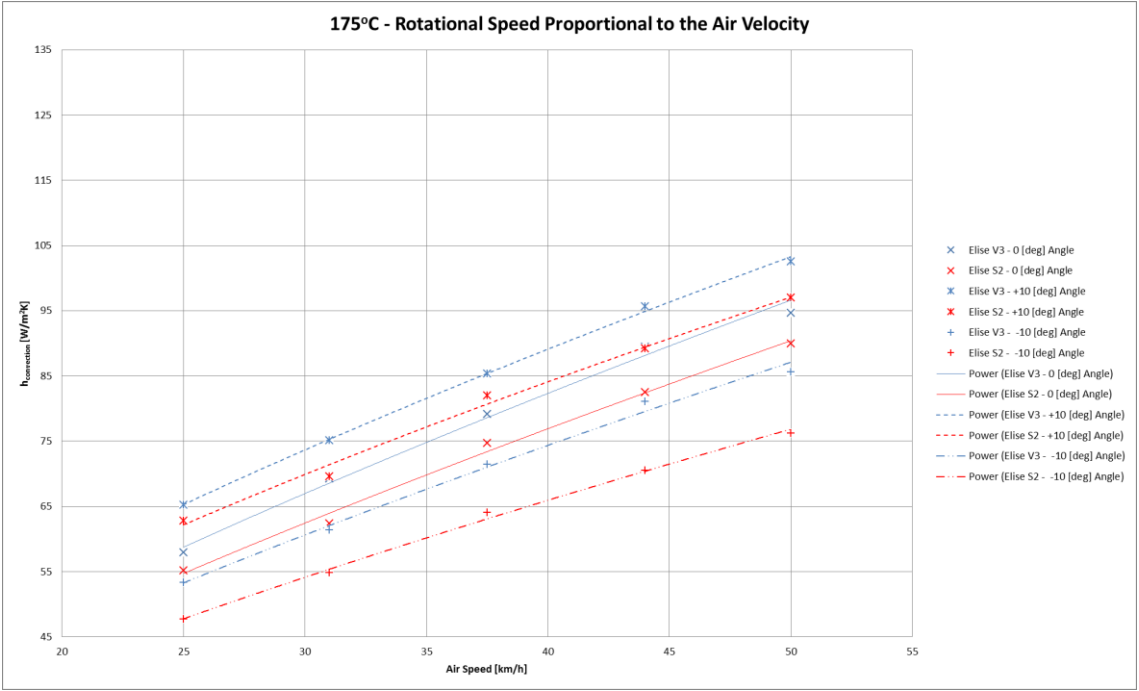
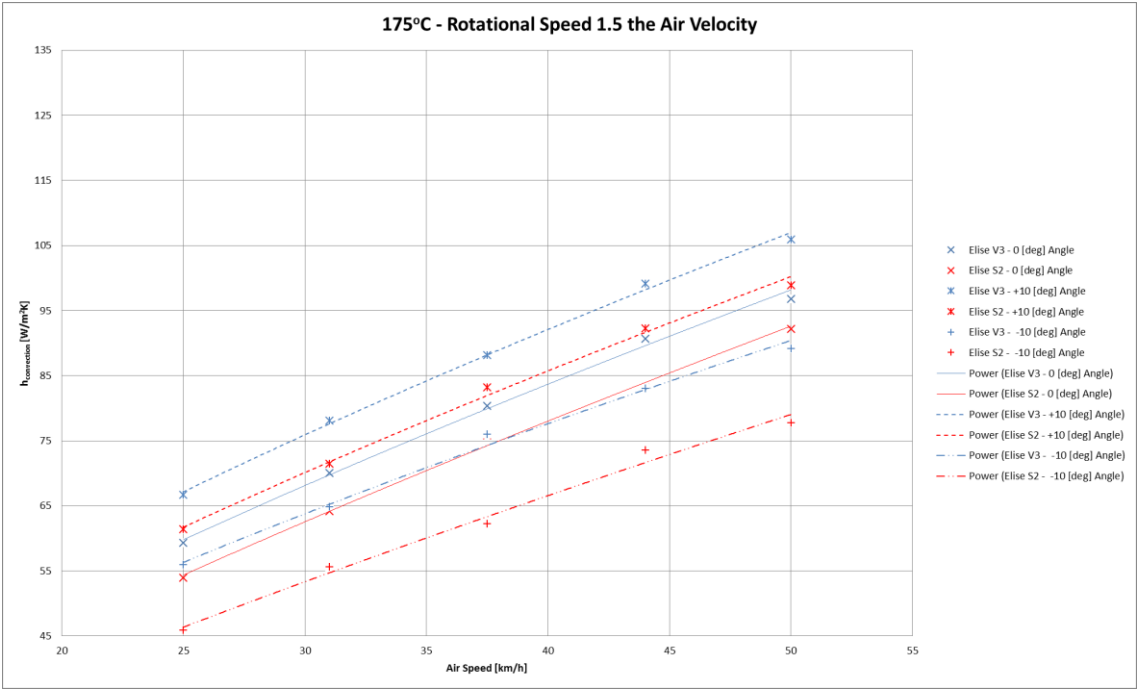
V.1 Impact of the Angle in the Convective HTC

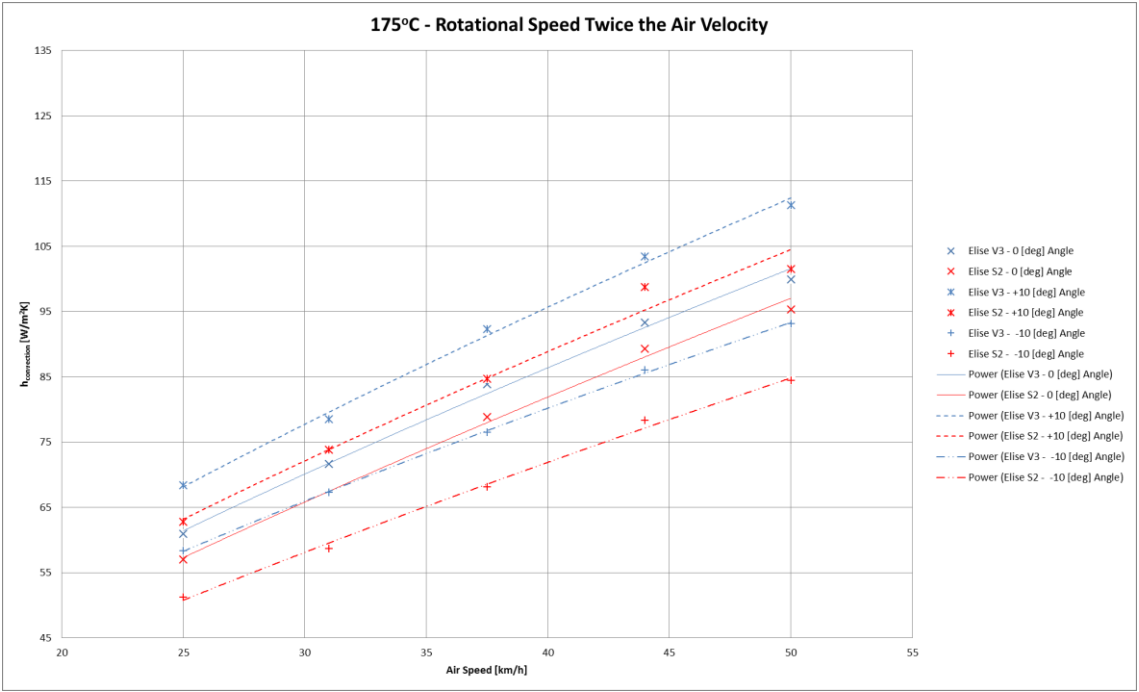
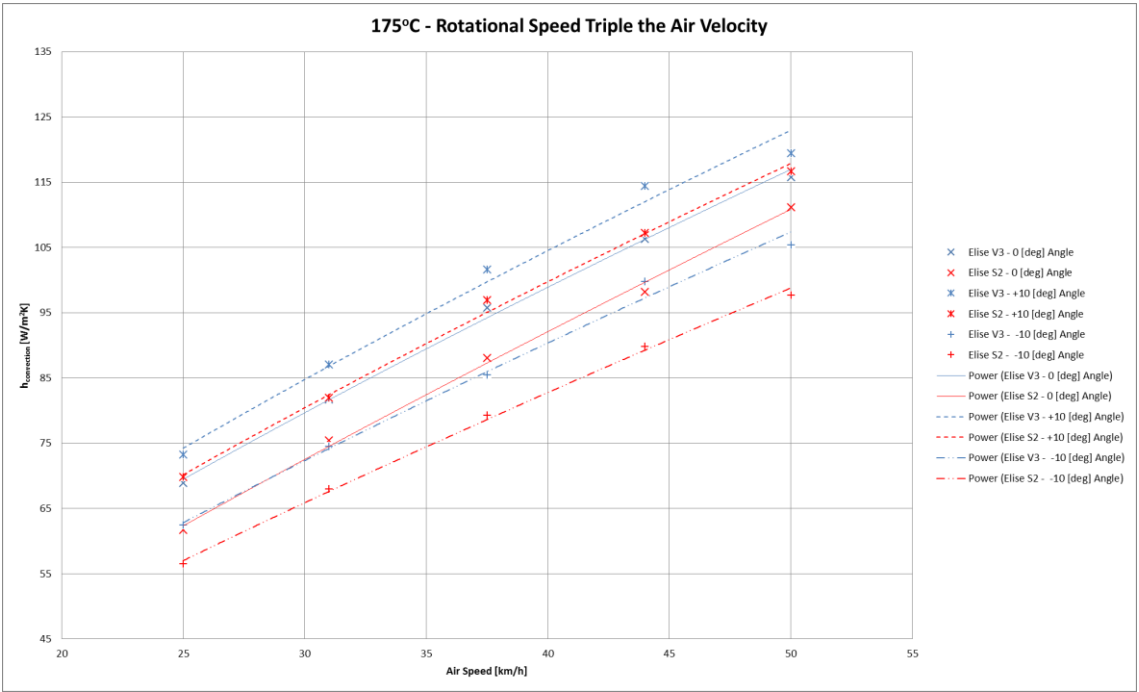
V.1.1 At 125°C Average Disc Surface Temperature



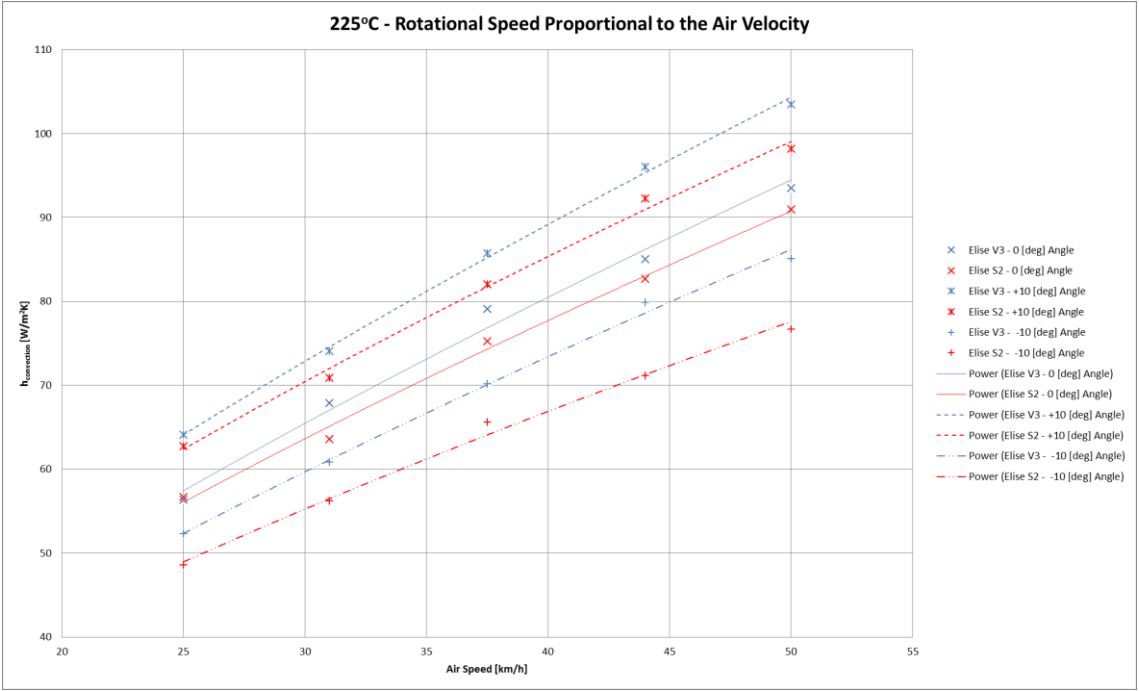


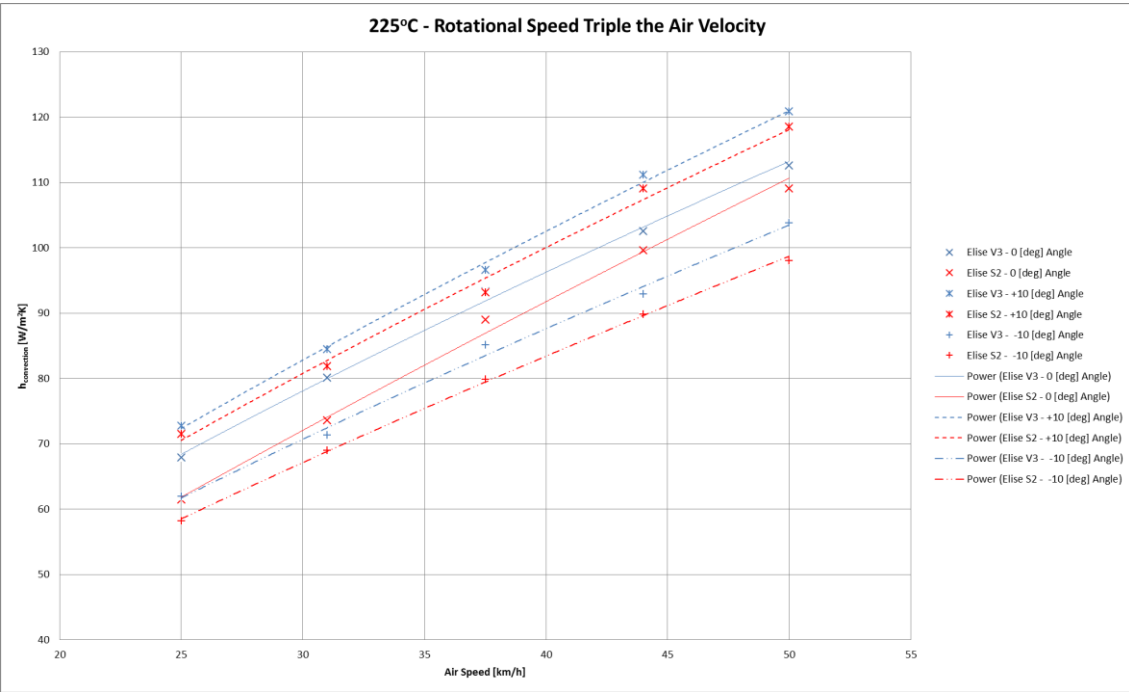
V.1.2 At 175°C Average Disc Surface Temperature



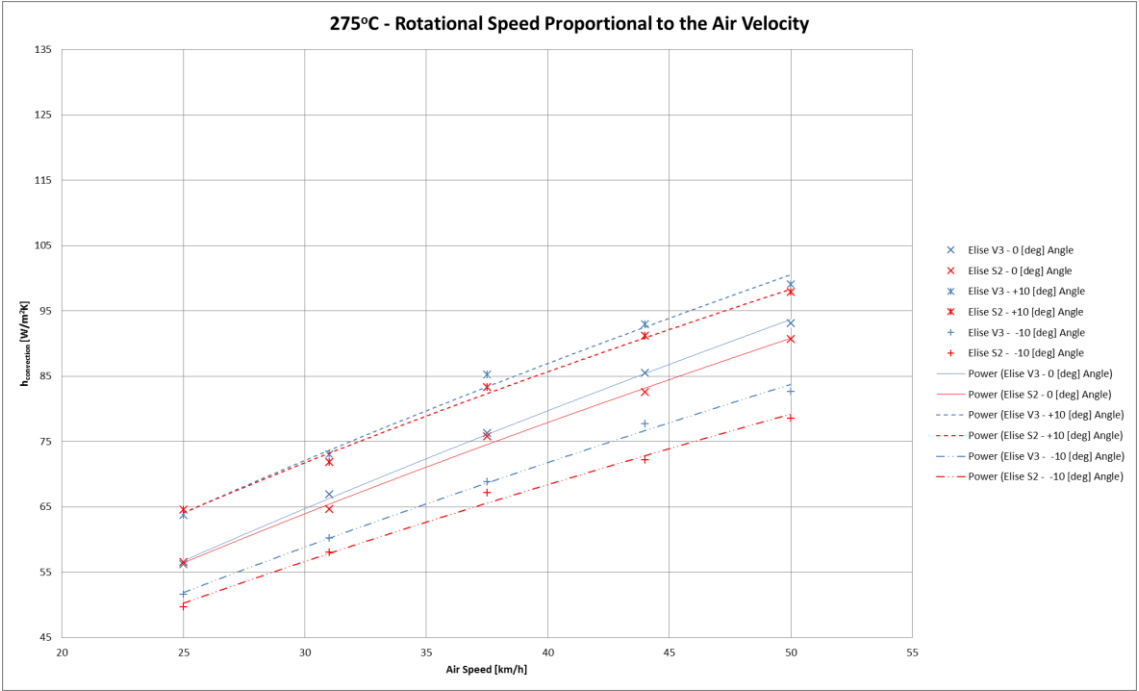


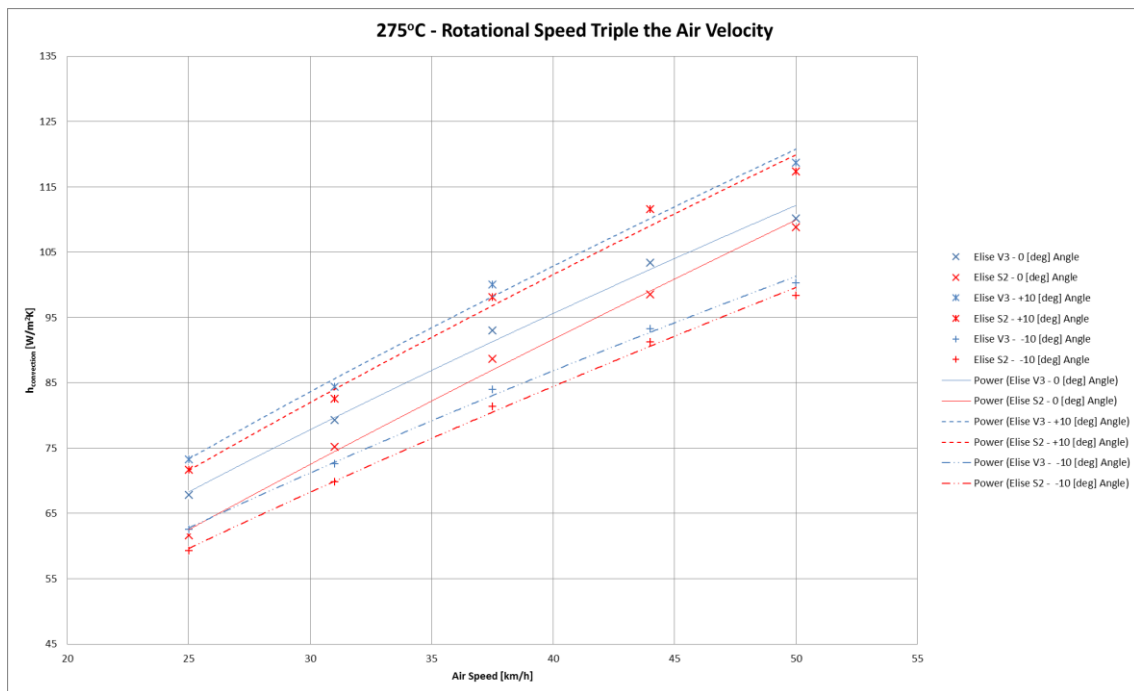
V.1.3 At 225°C Average Disc Surface Temperature



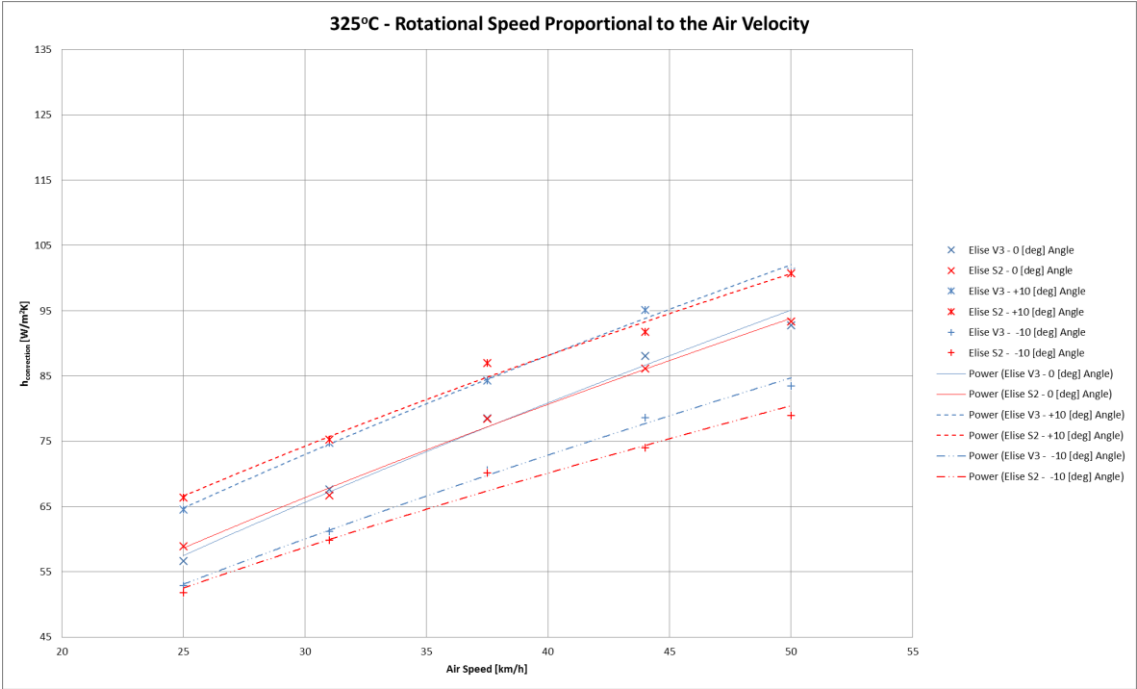
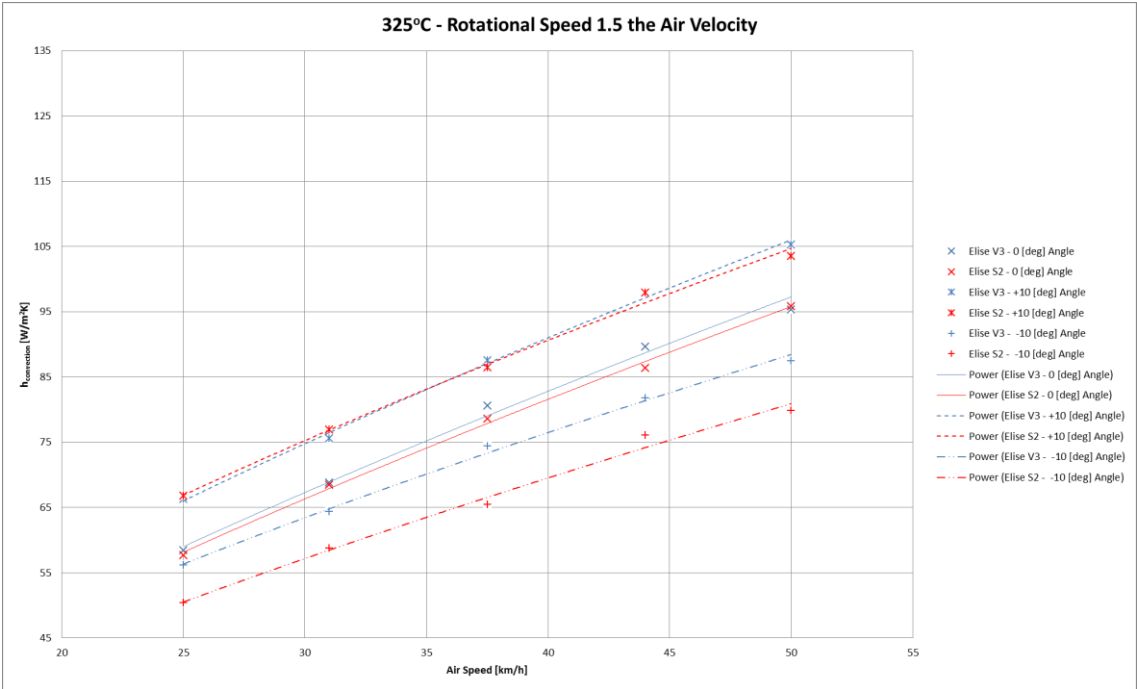


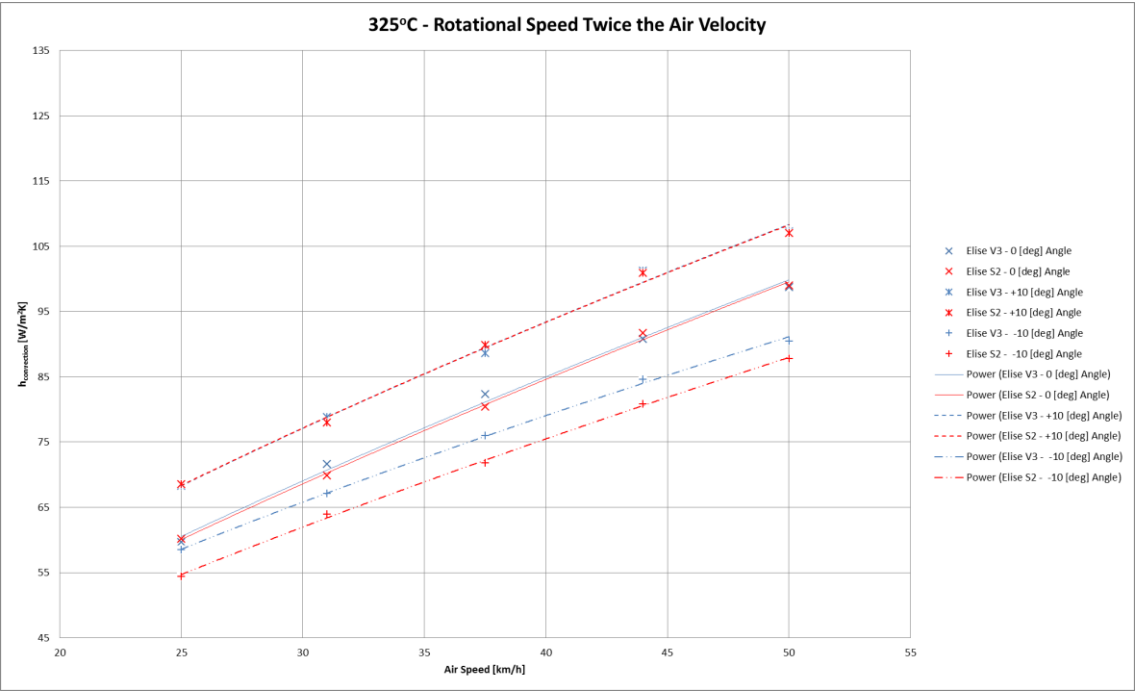
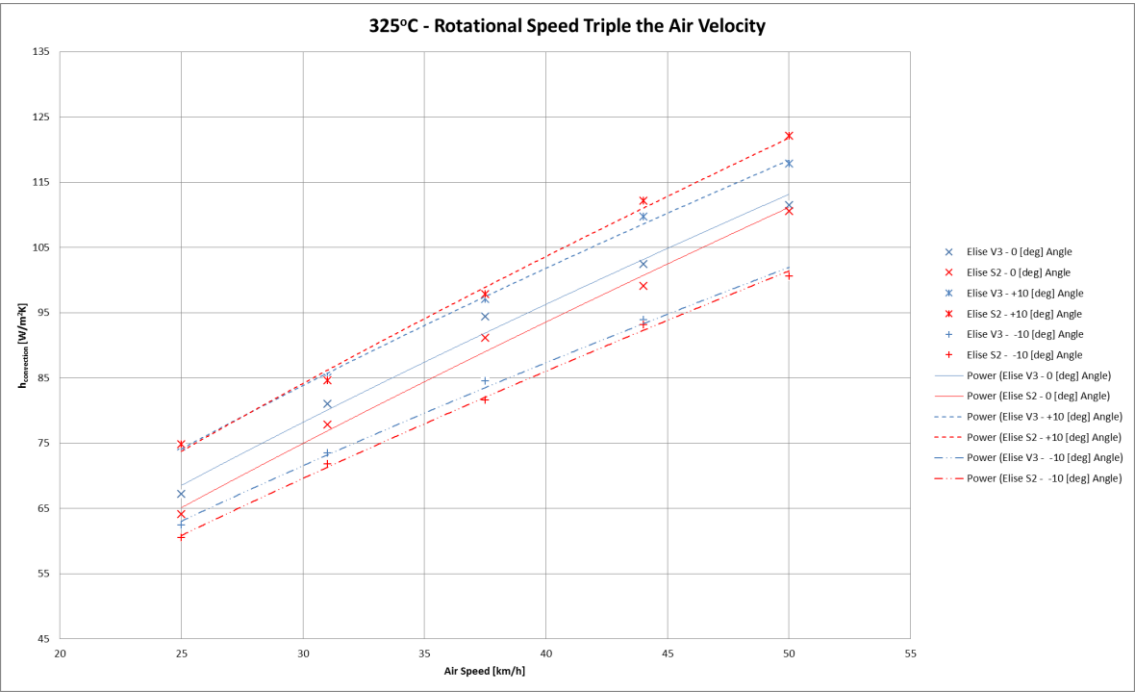
V.1.4 At 275°C Average Disc Surface Temperature



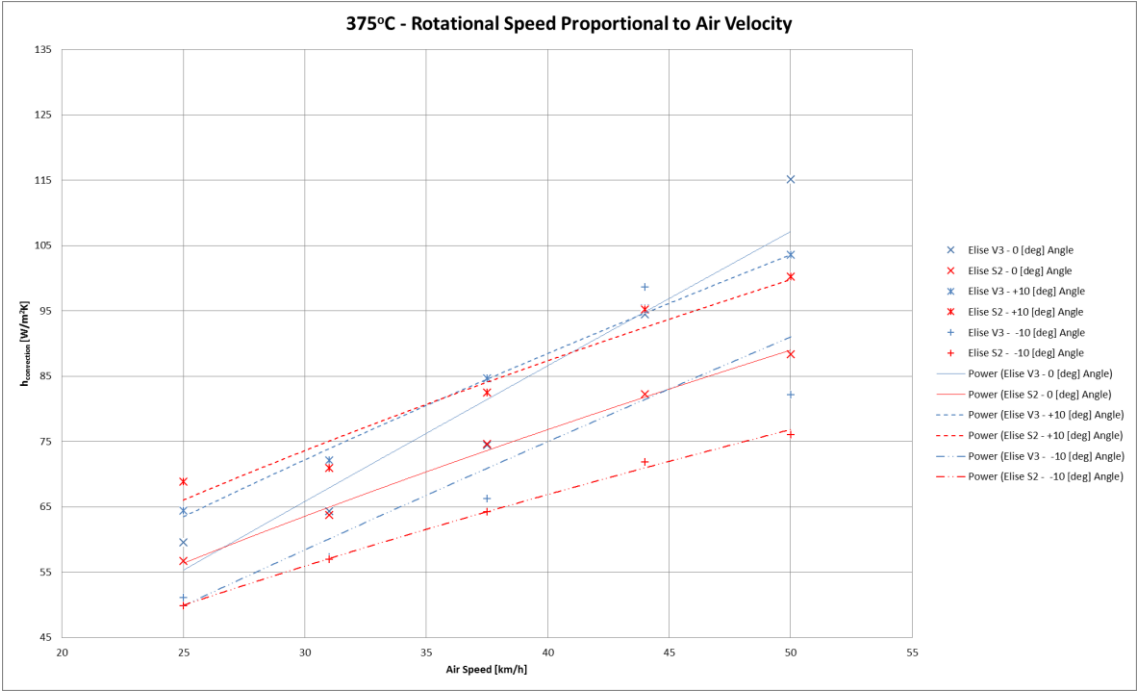


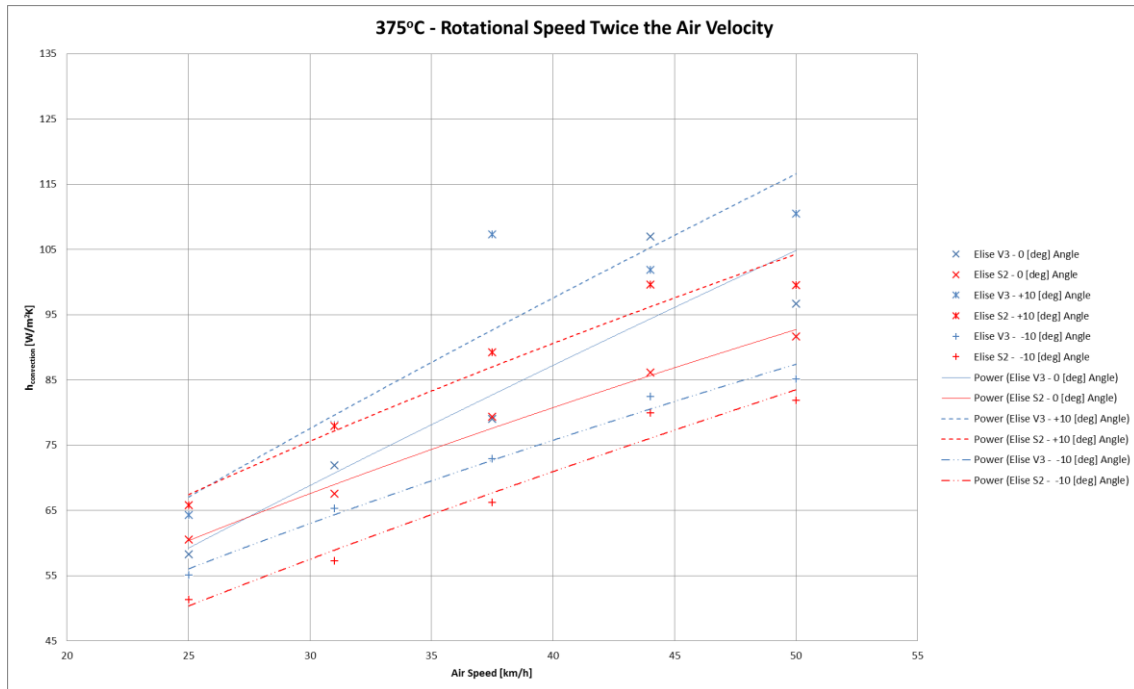
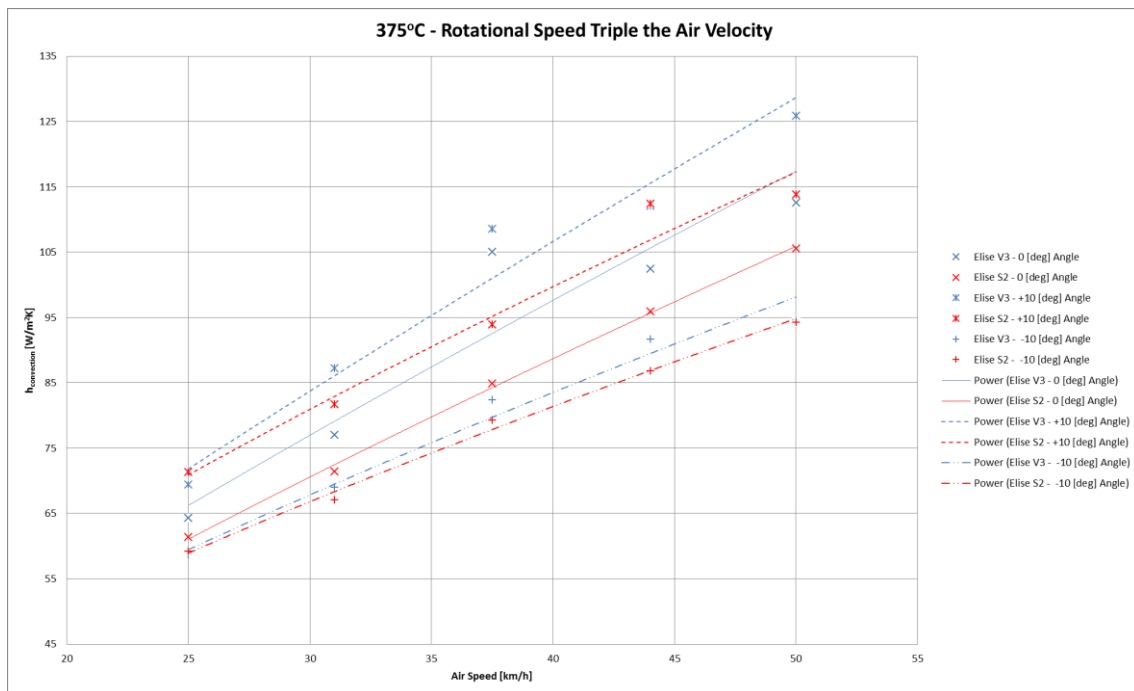
V.1.5 At 325°C Average Disc Surface Temperature



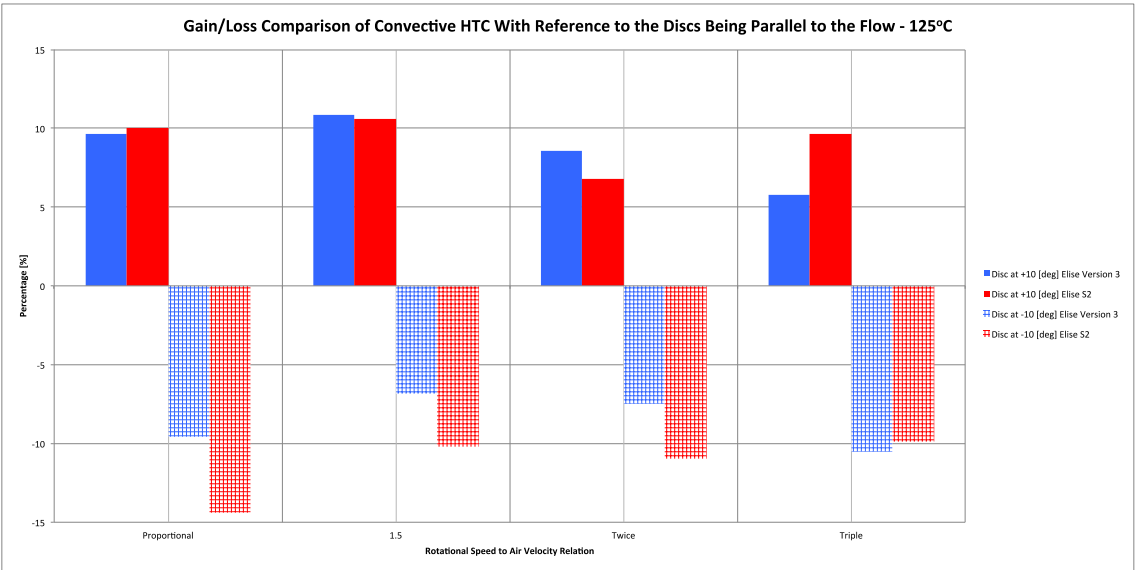
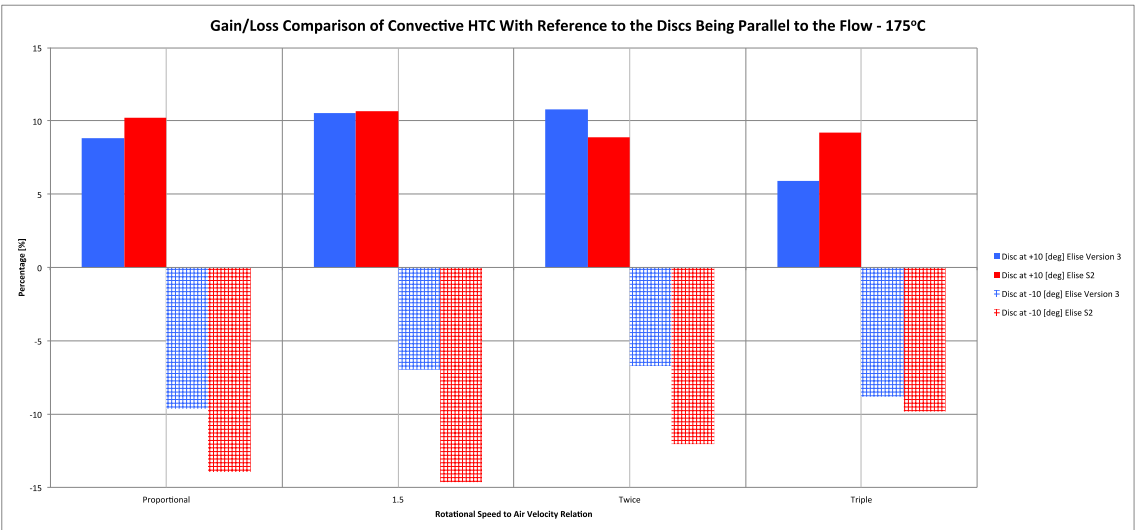


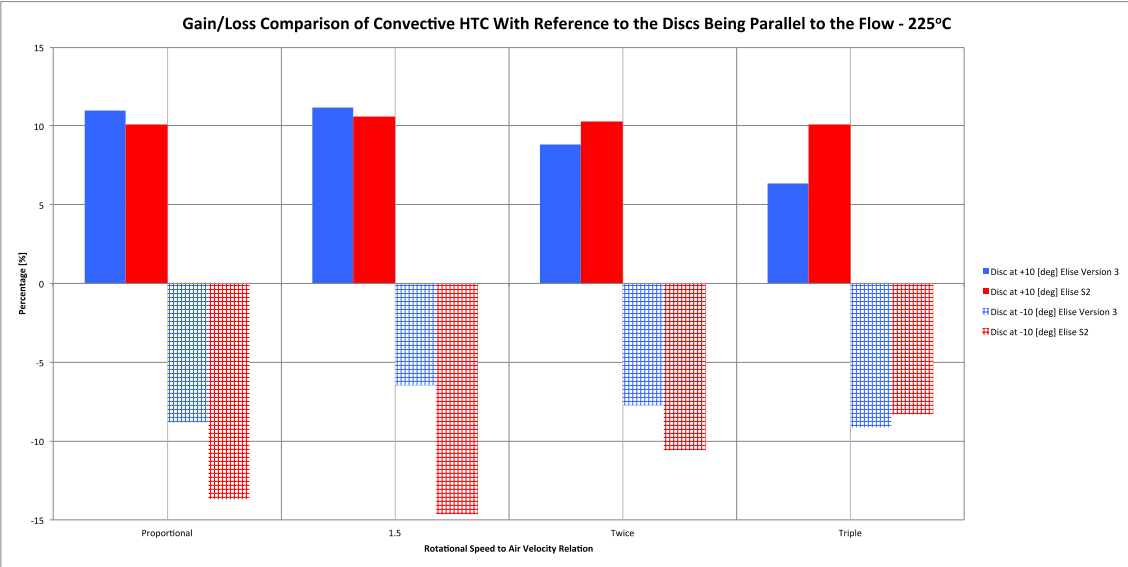
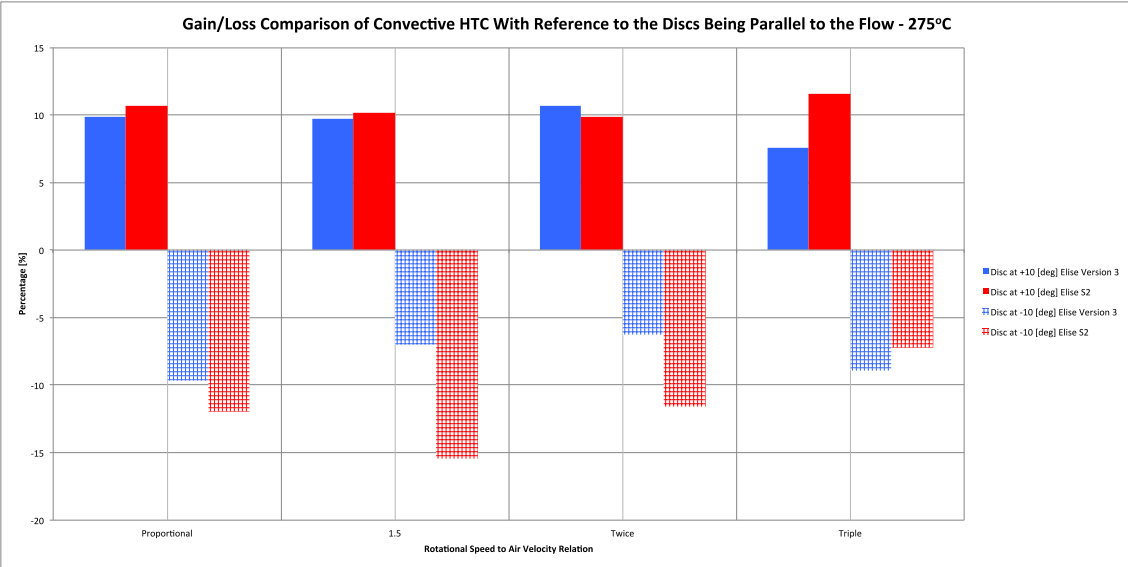
V.1.6 At 375°C Average Disc Surface Temperature

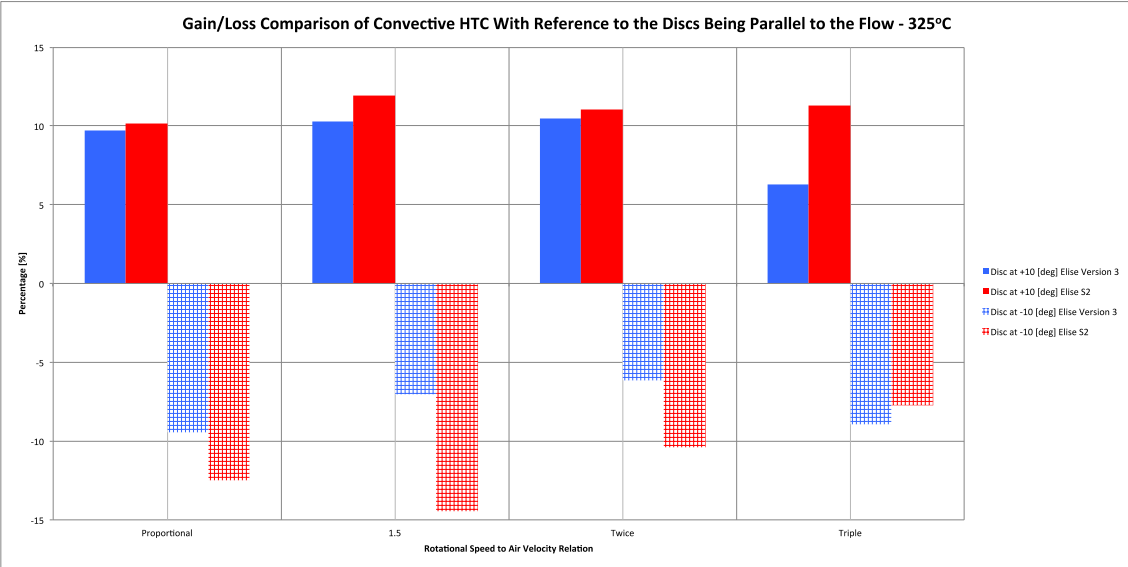
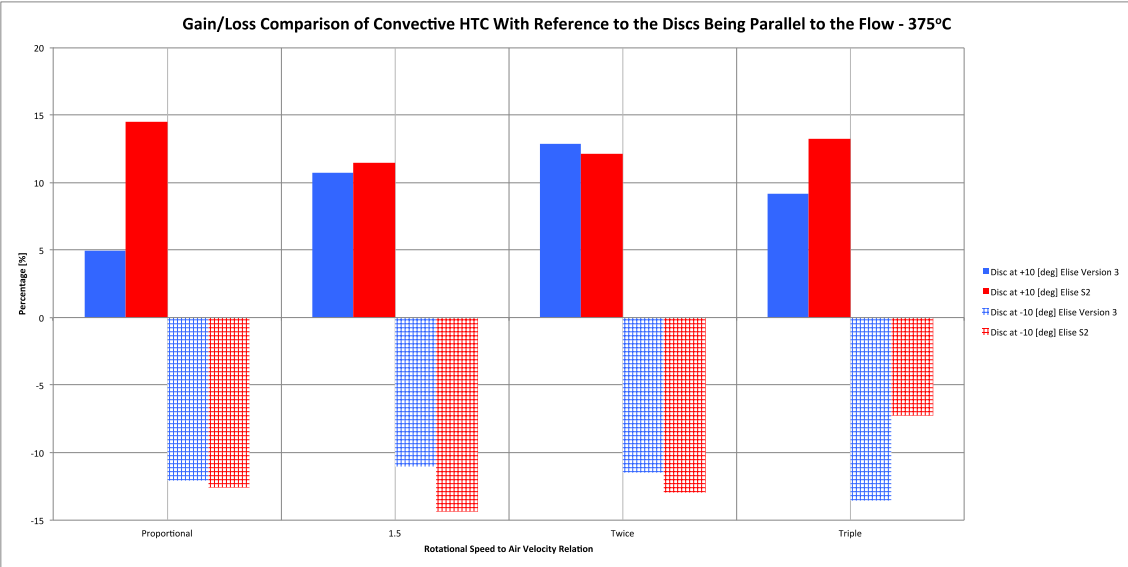




V.2 Gain/Loss in Convective HTC From the Disc Angle

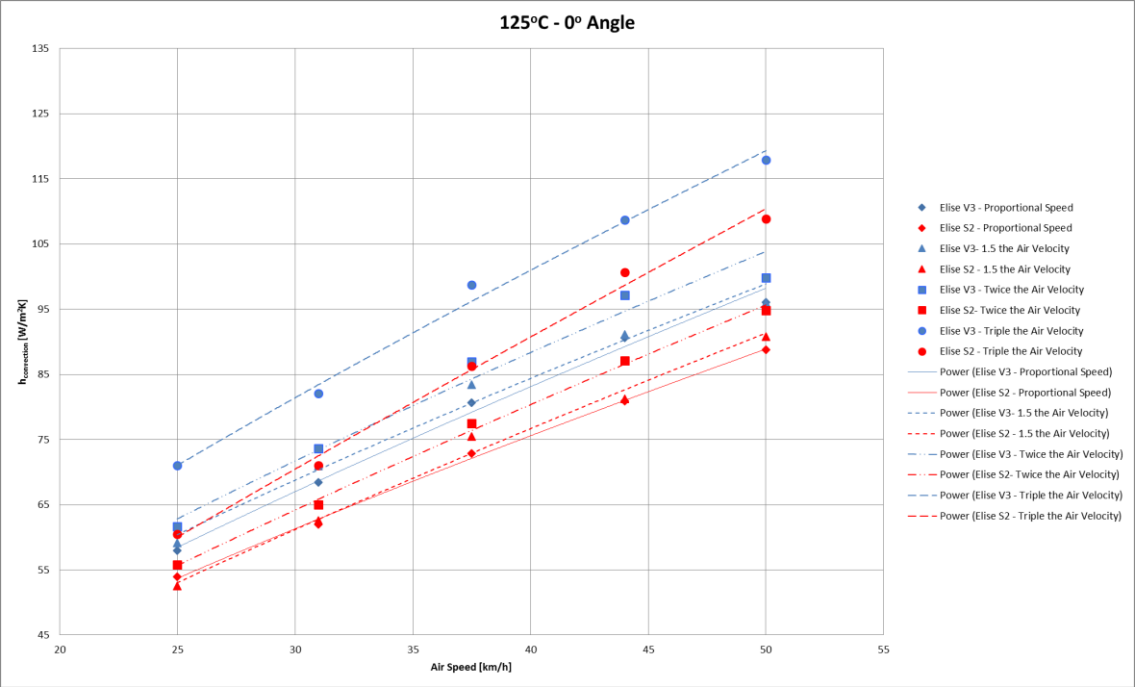
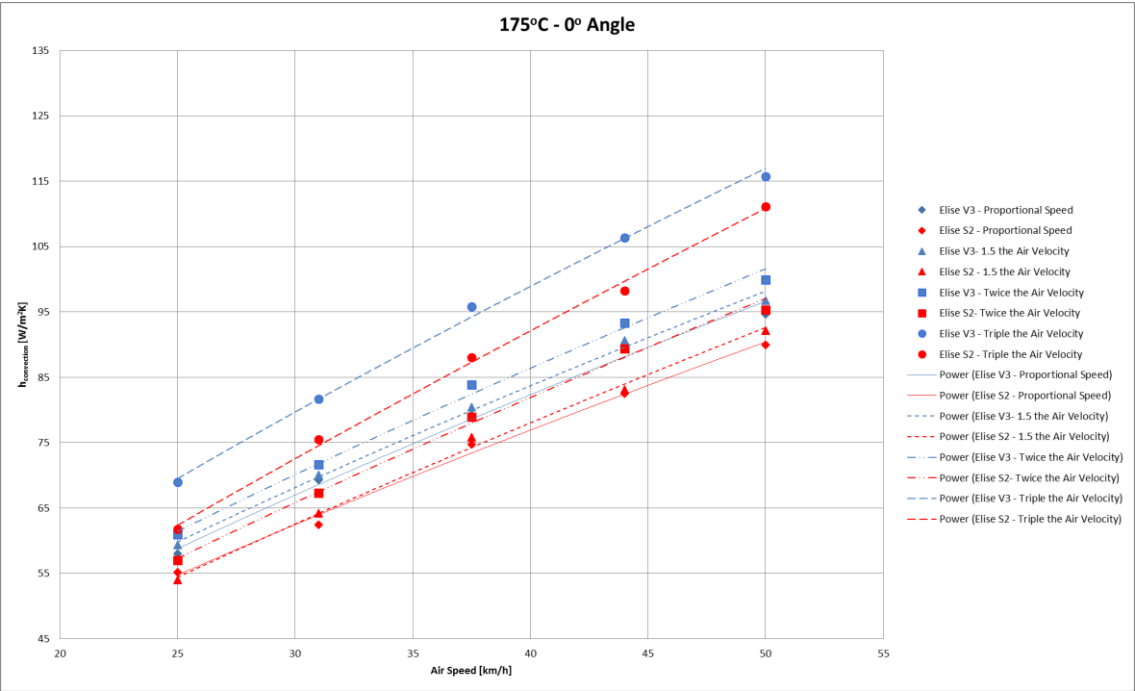


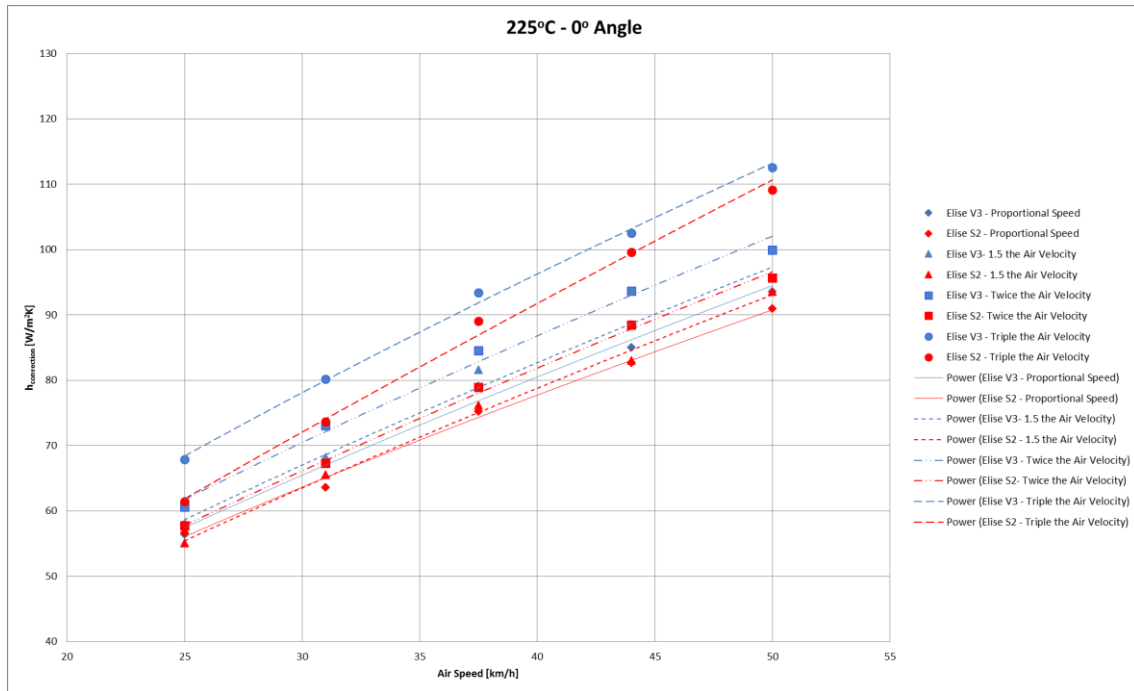
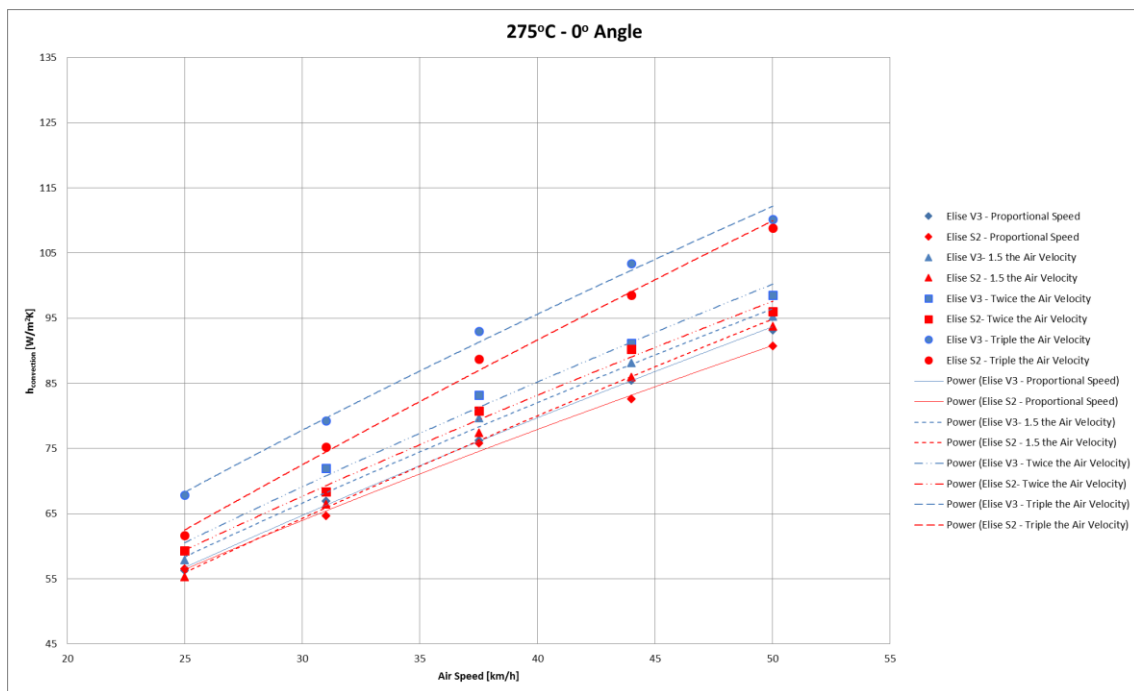


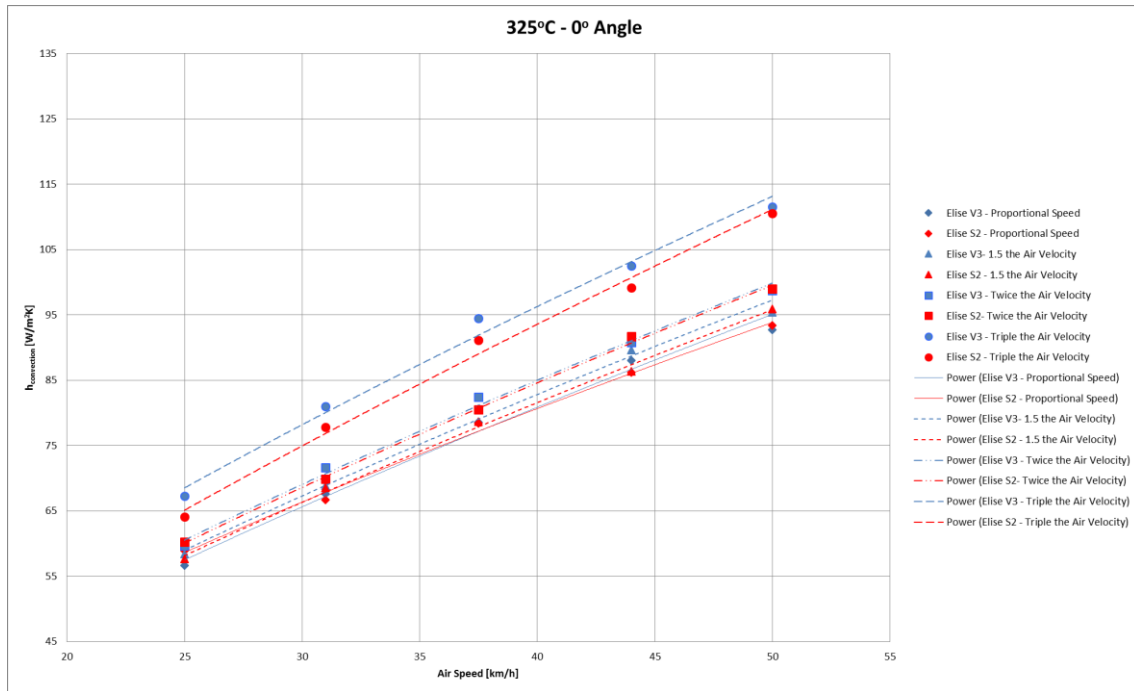
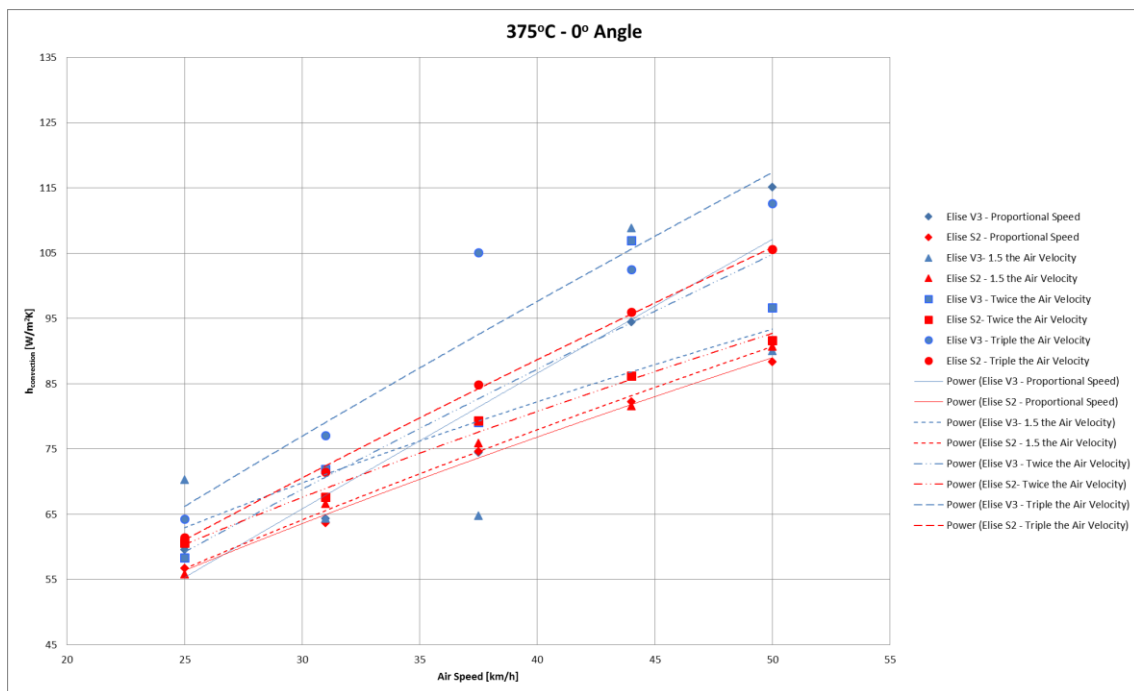


Appendix W – Influence of Disc Rotational Speed

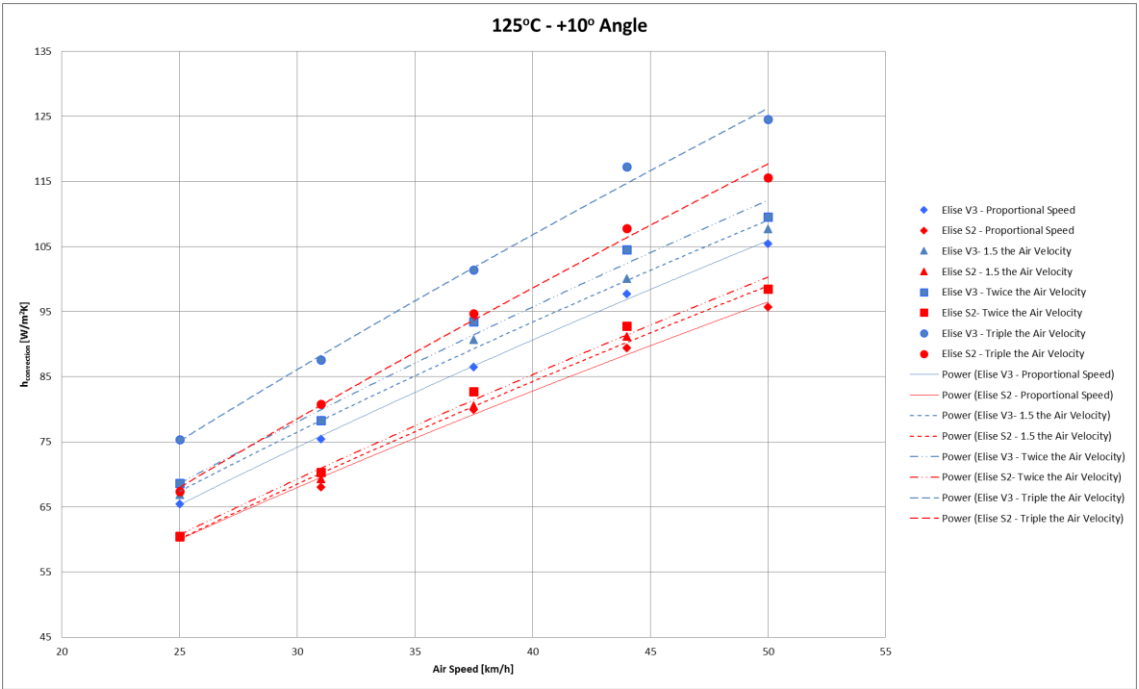
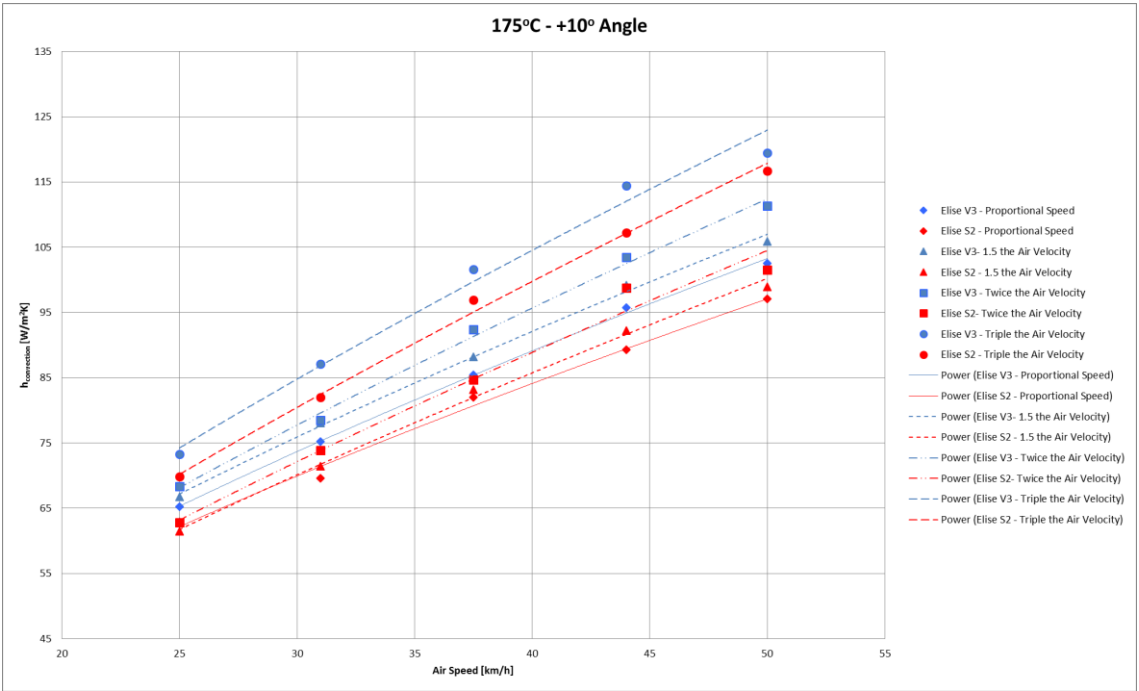
W.1 At 0° Angle

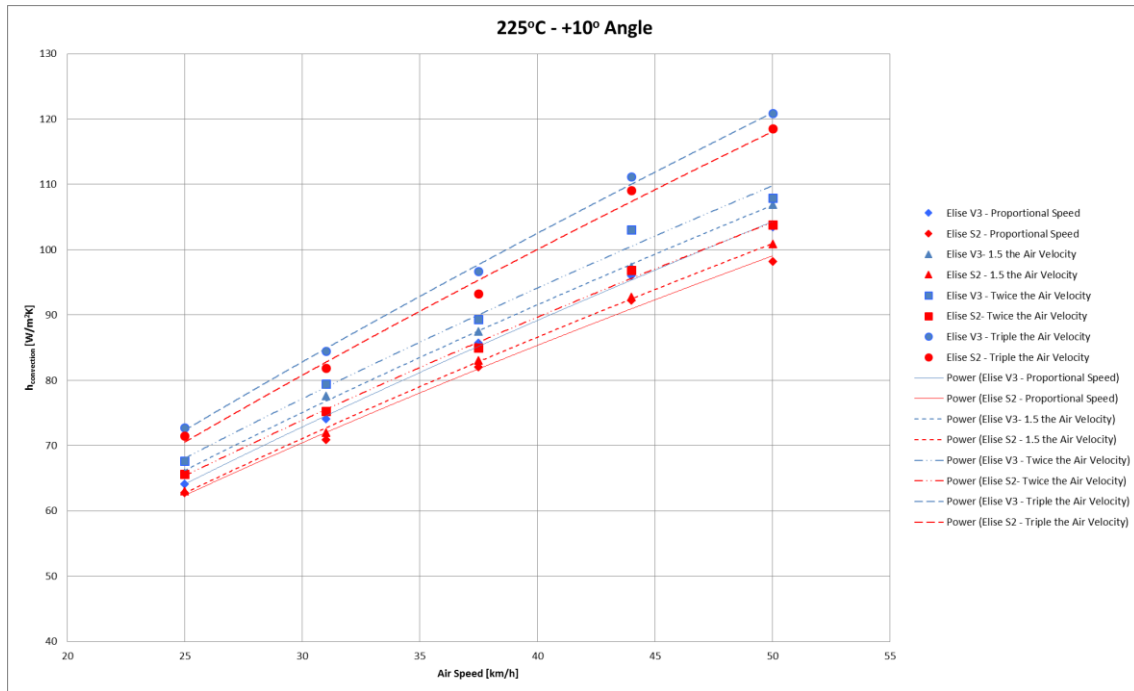
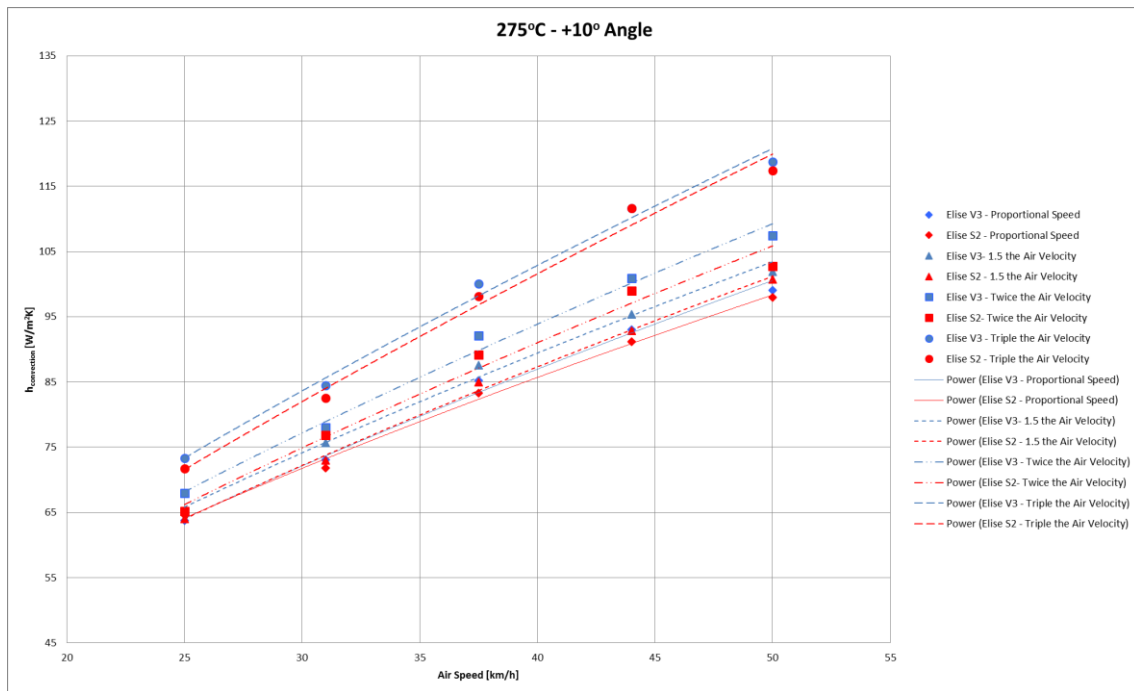


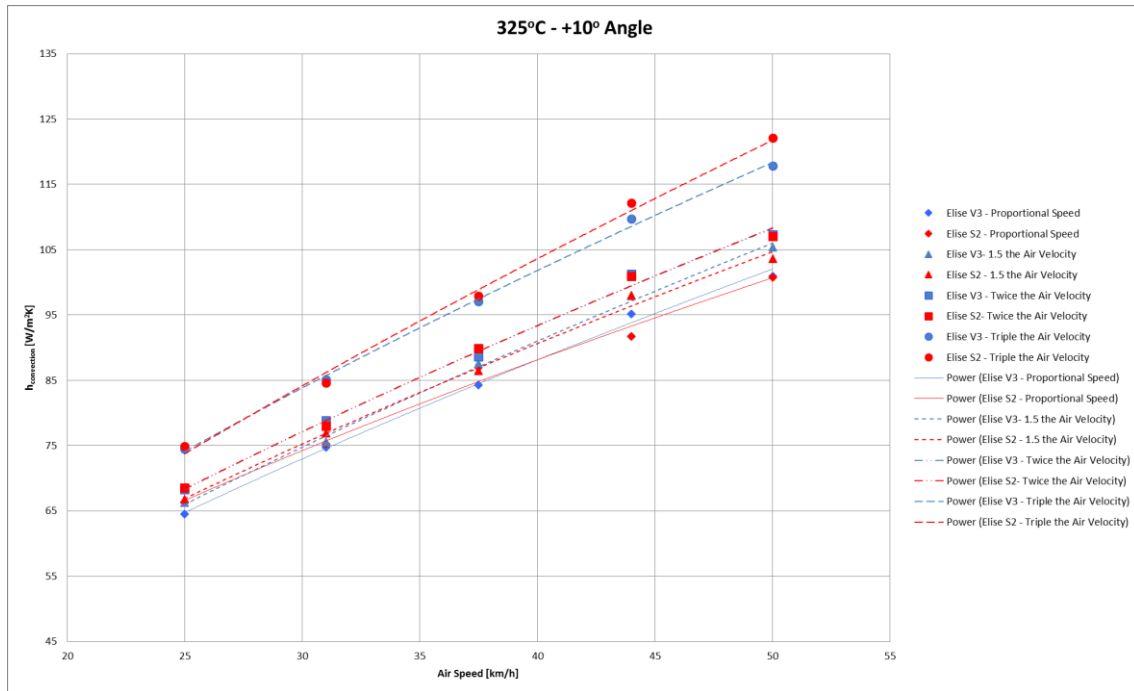
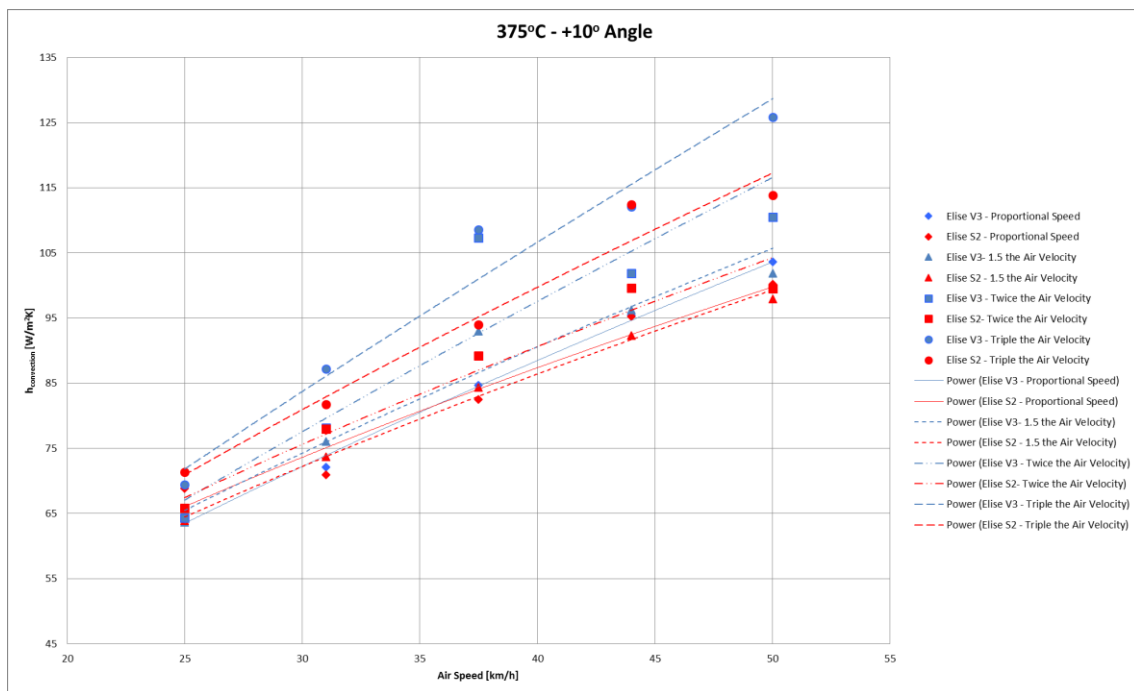




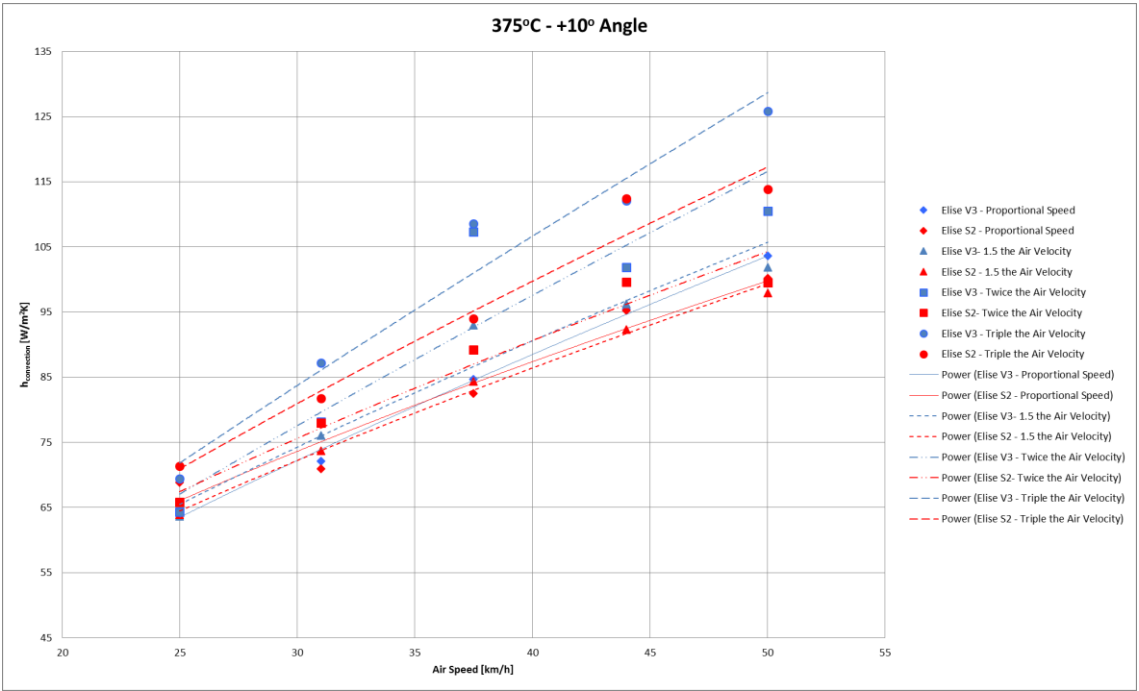
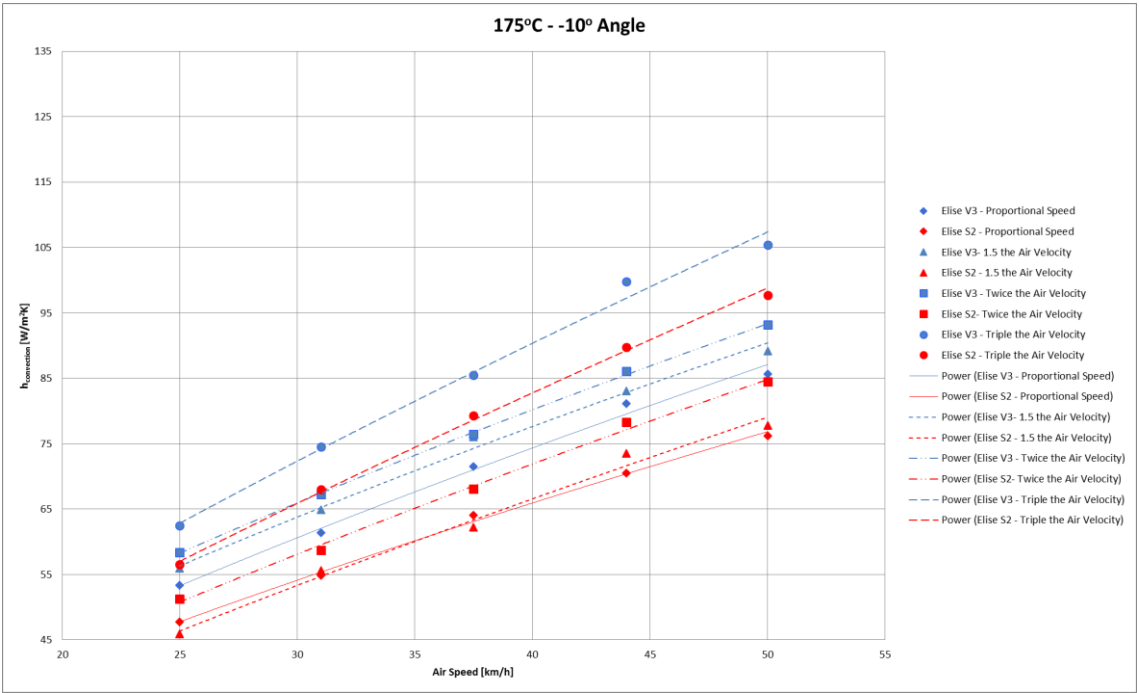
W.2 At +10° Angle

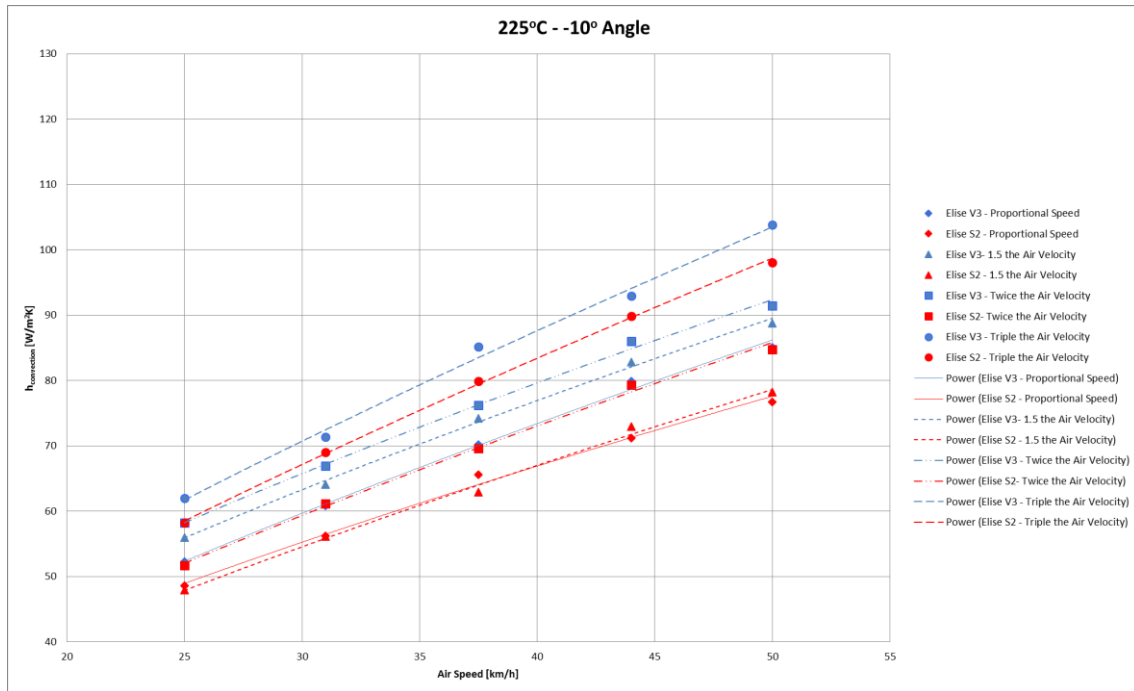
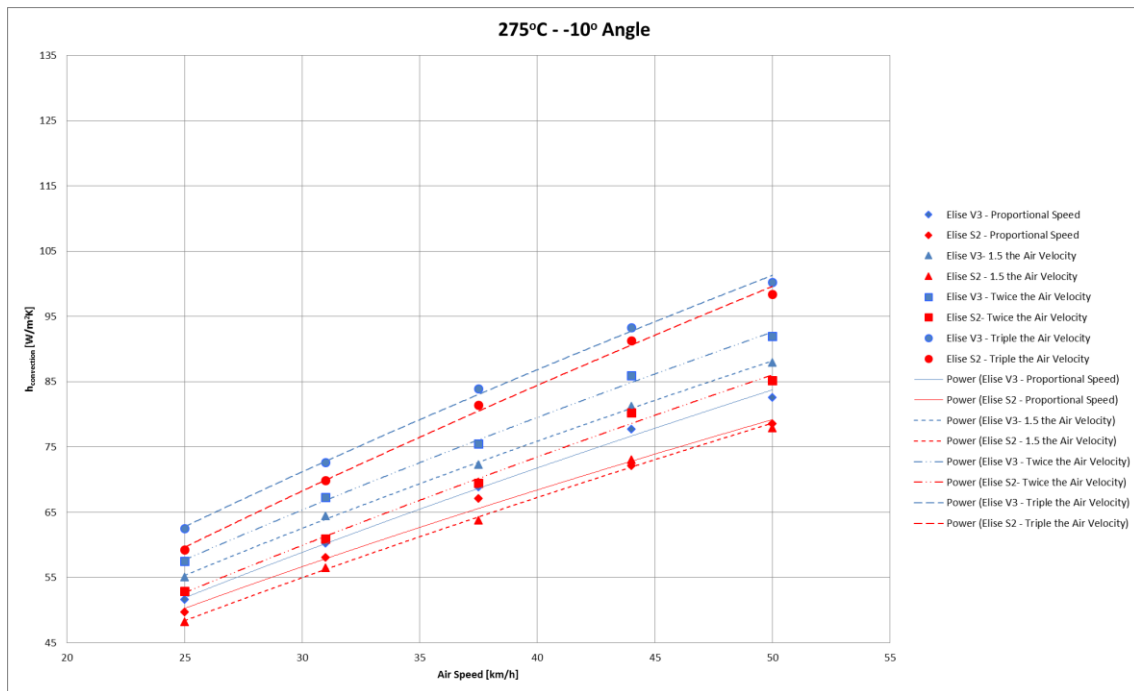


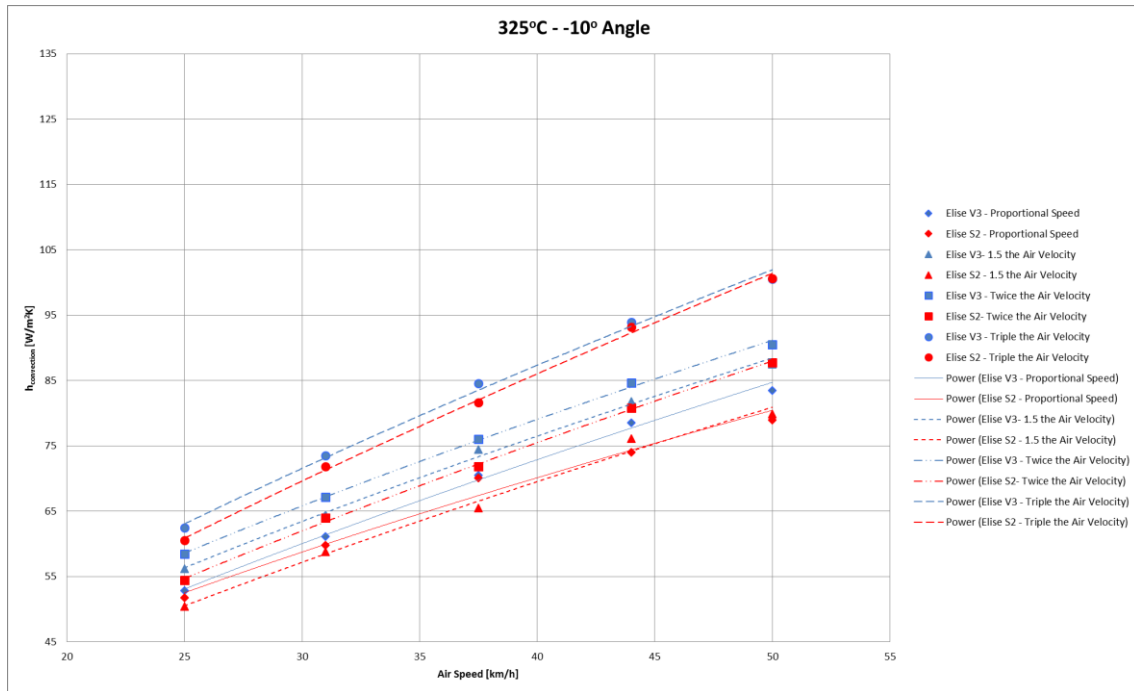
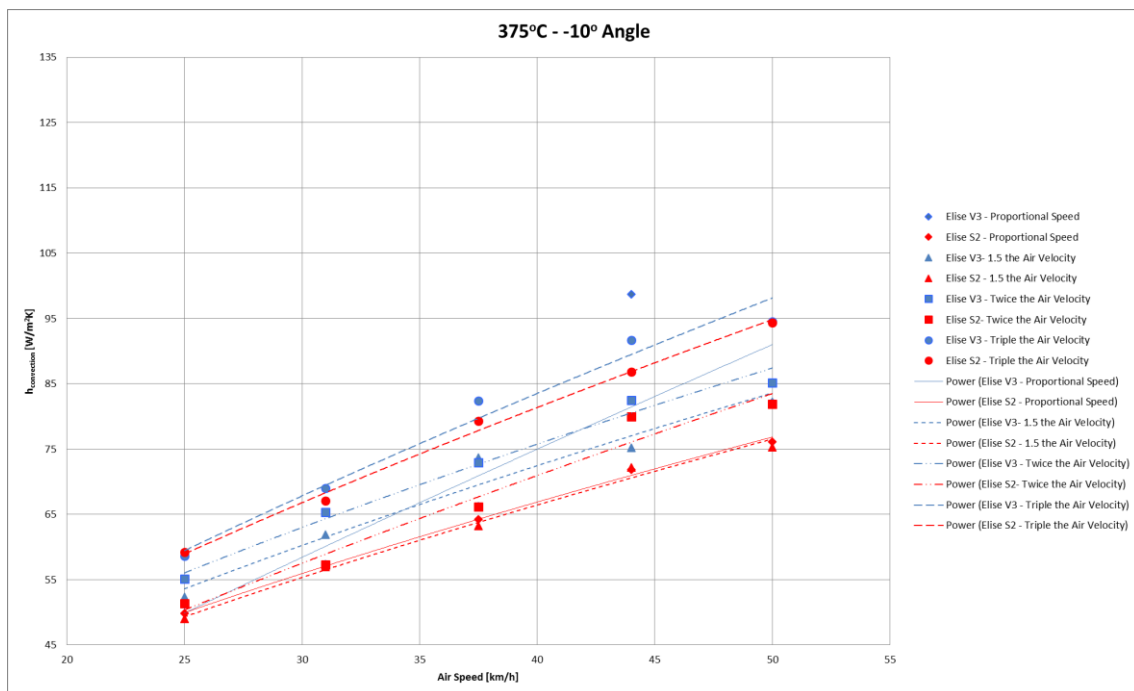




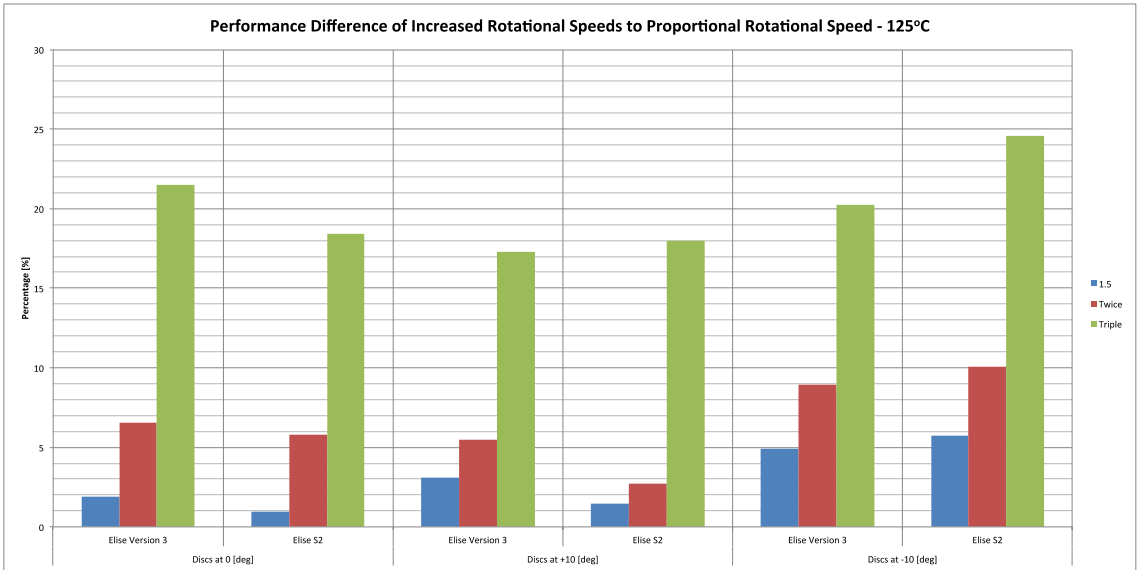
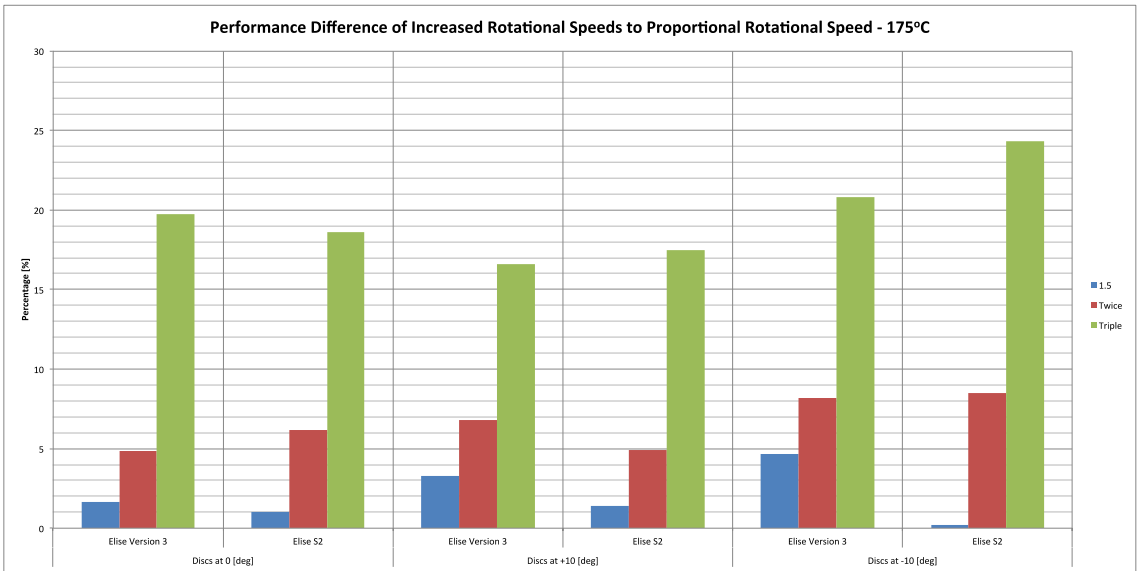
W.3 At -10° Angle

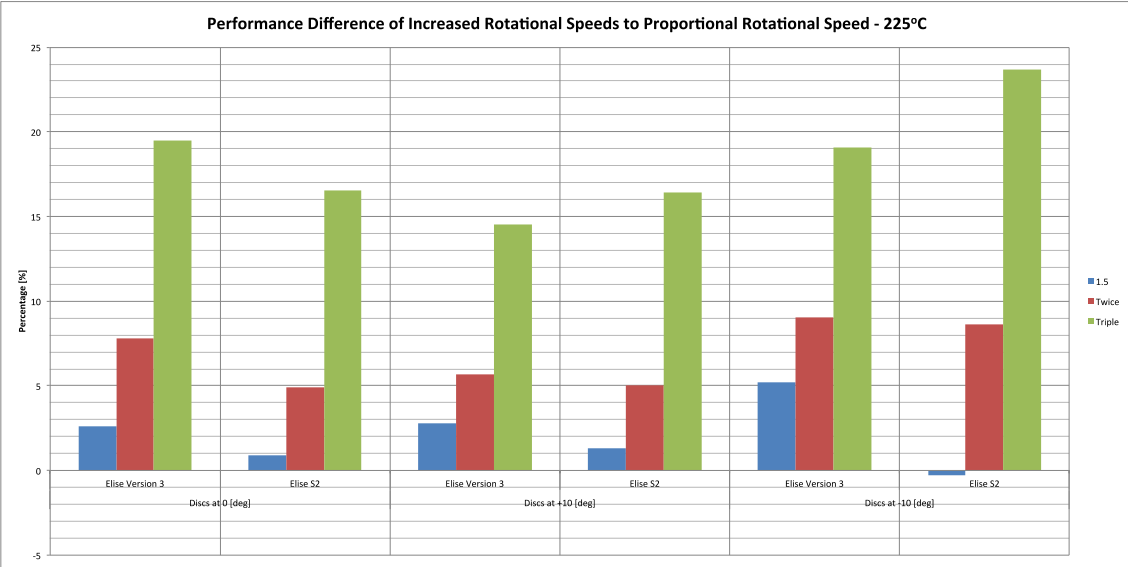
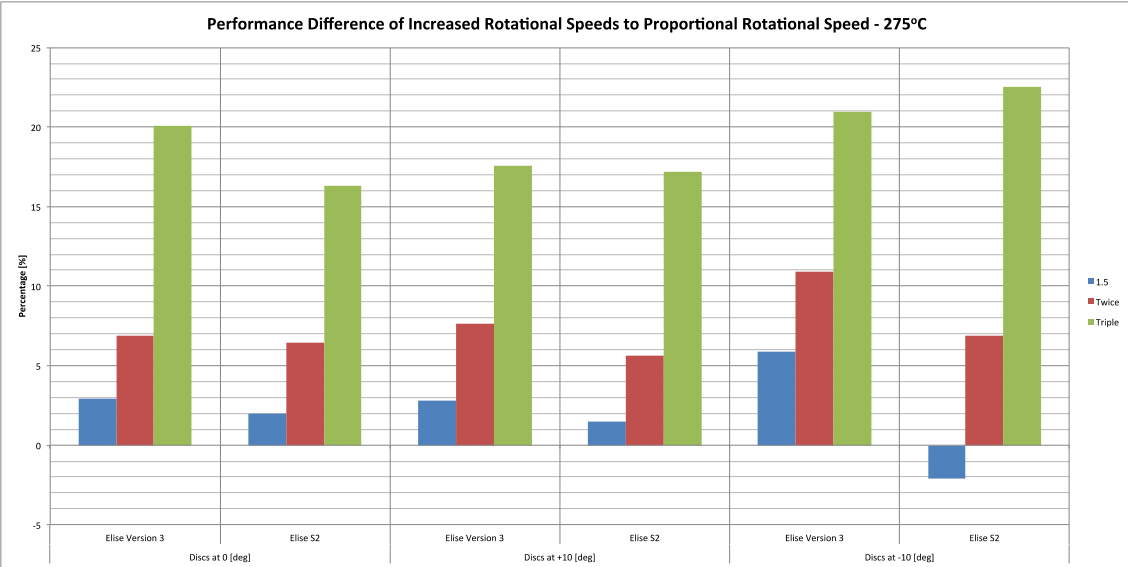


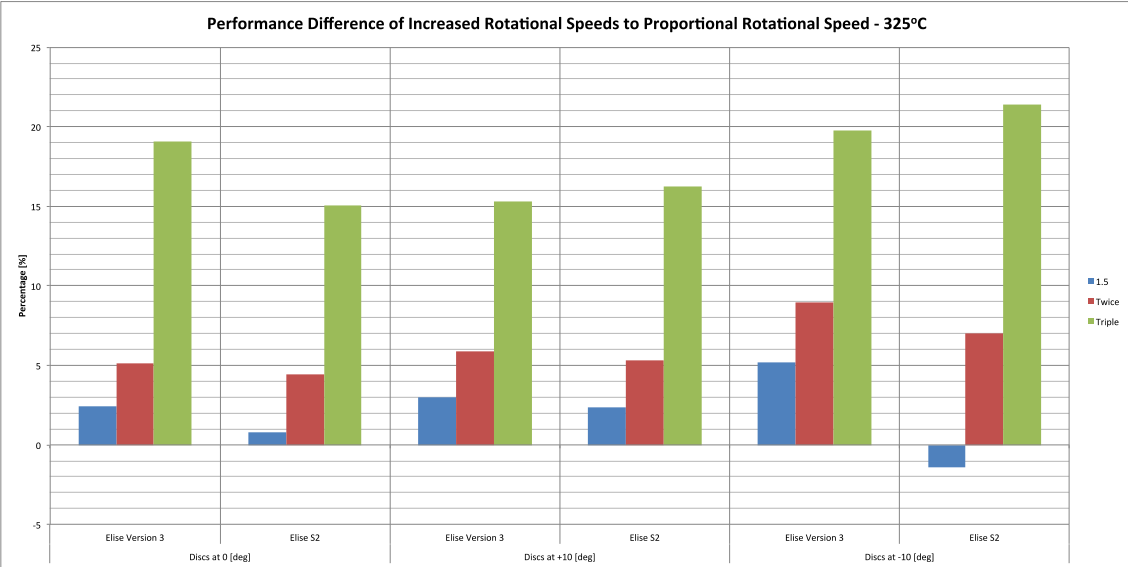
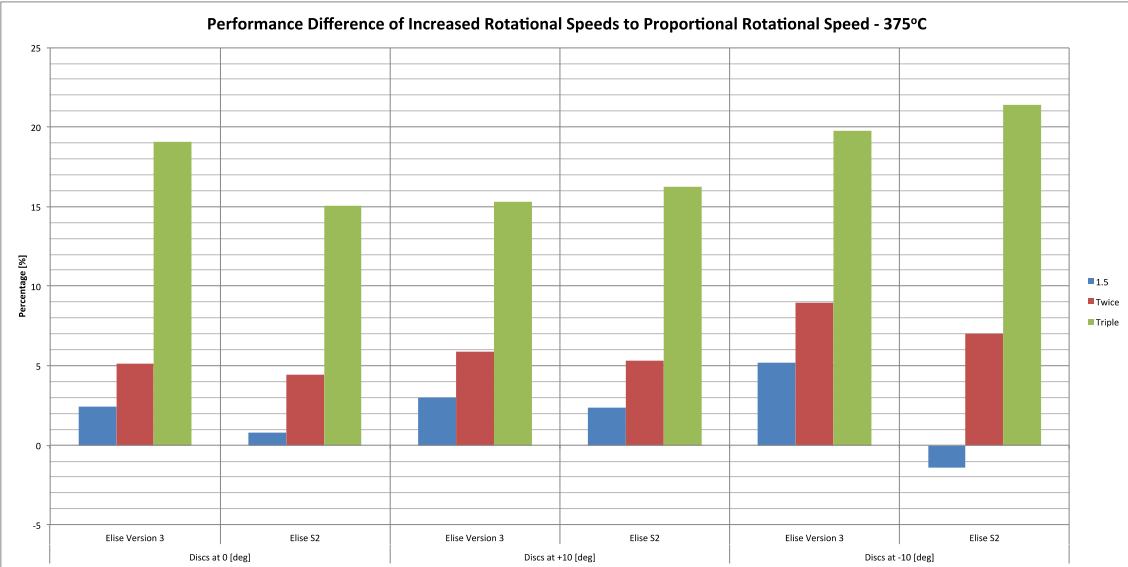




W.4 Performance Difference for Each Disc

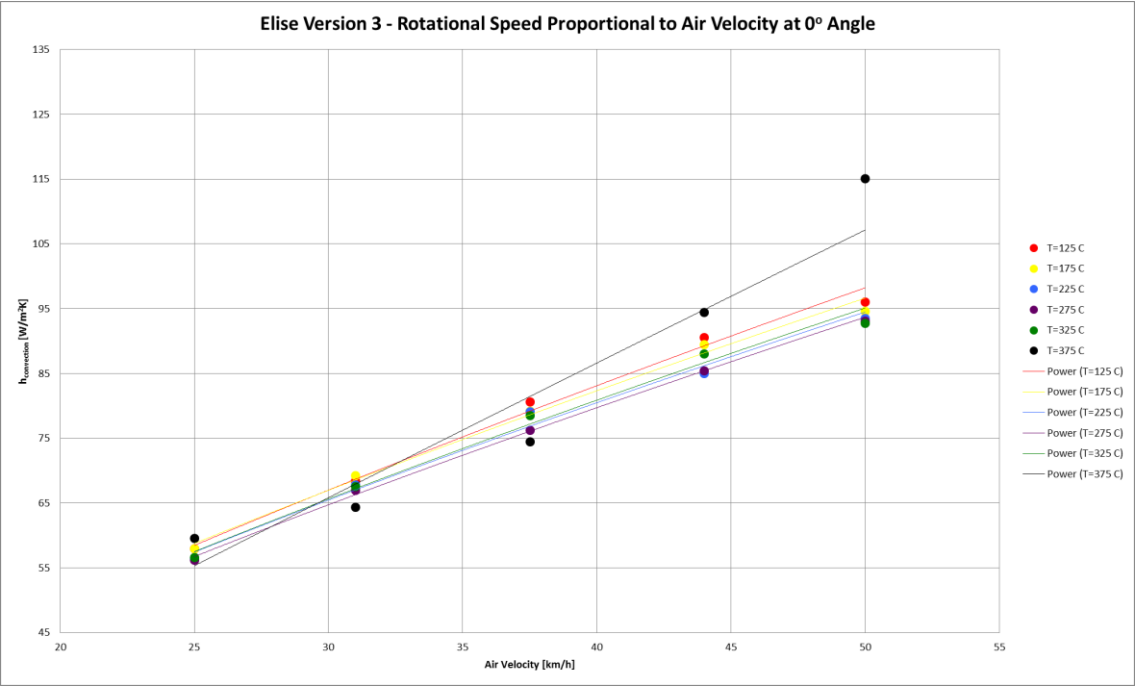
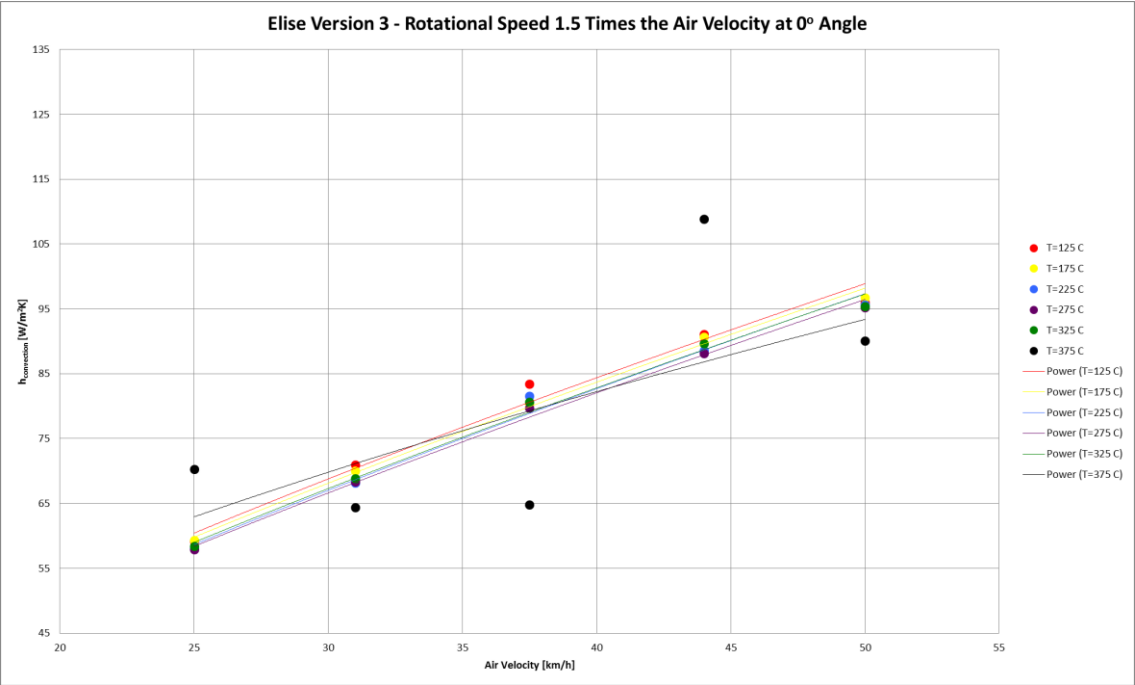


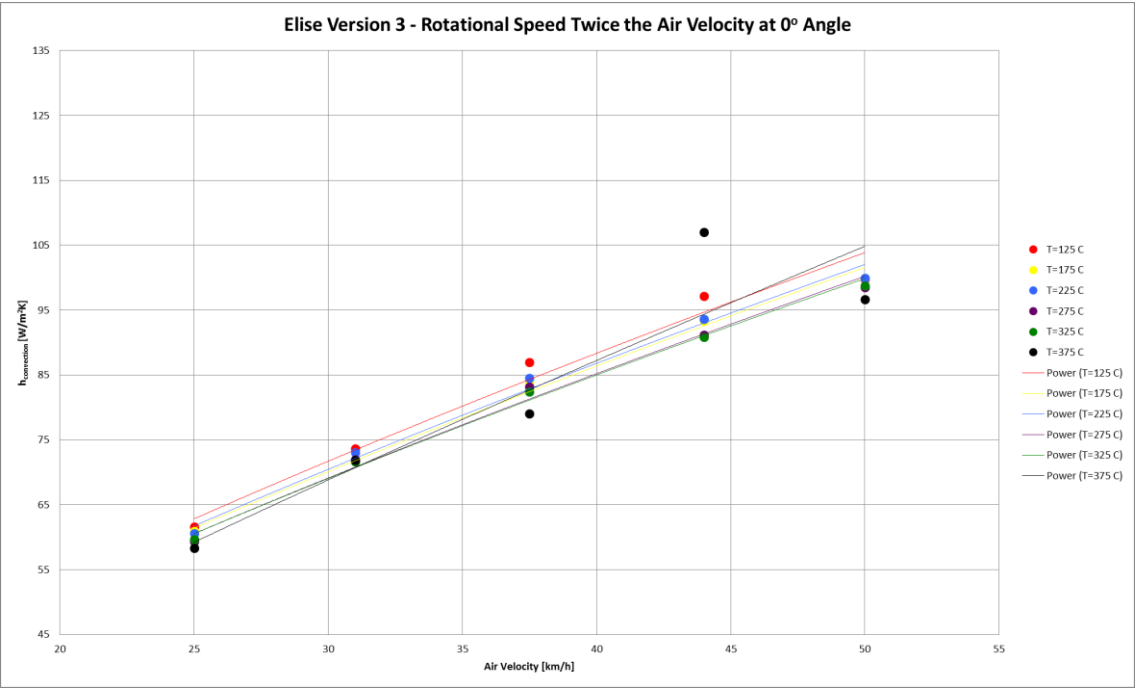
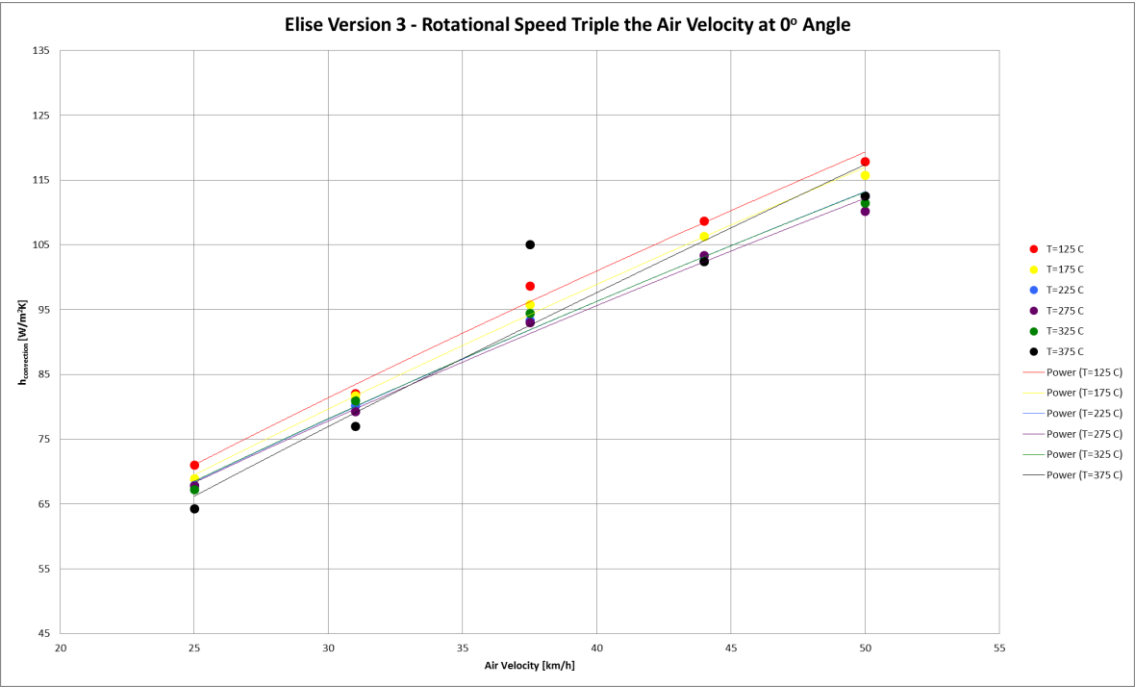


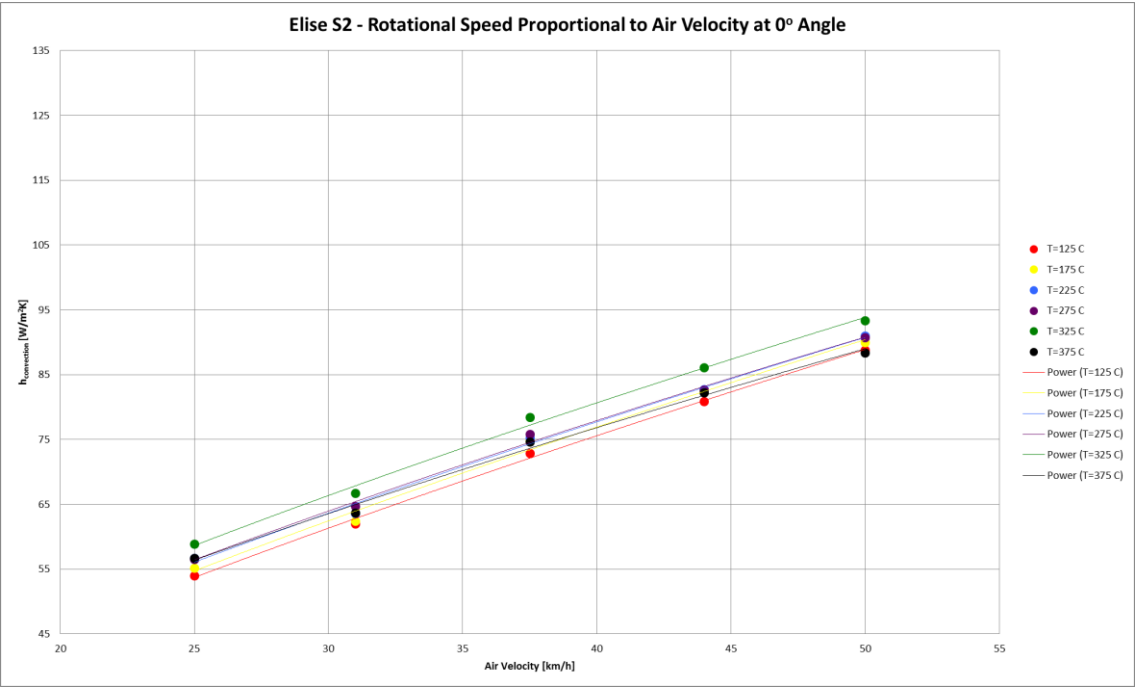
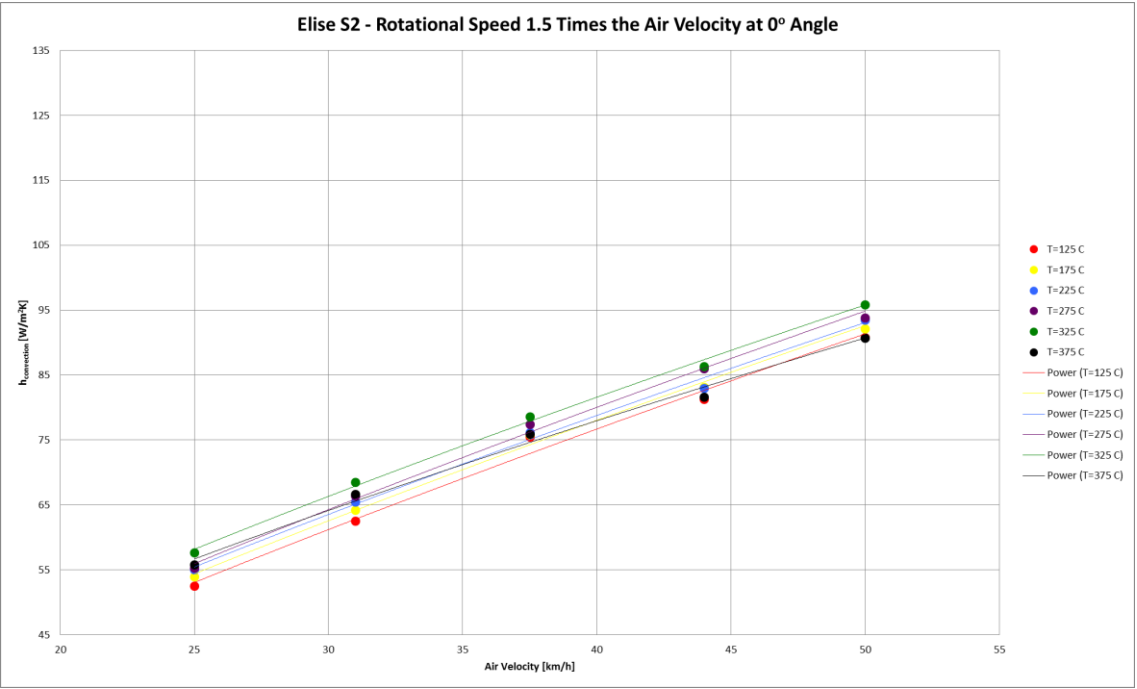


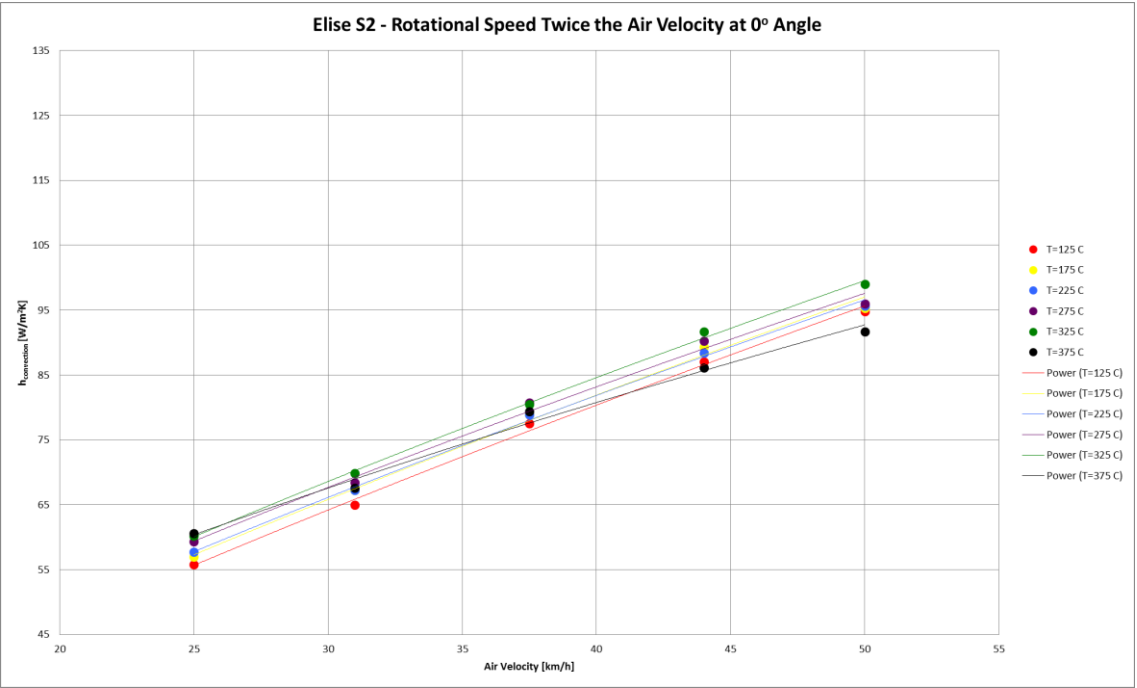
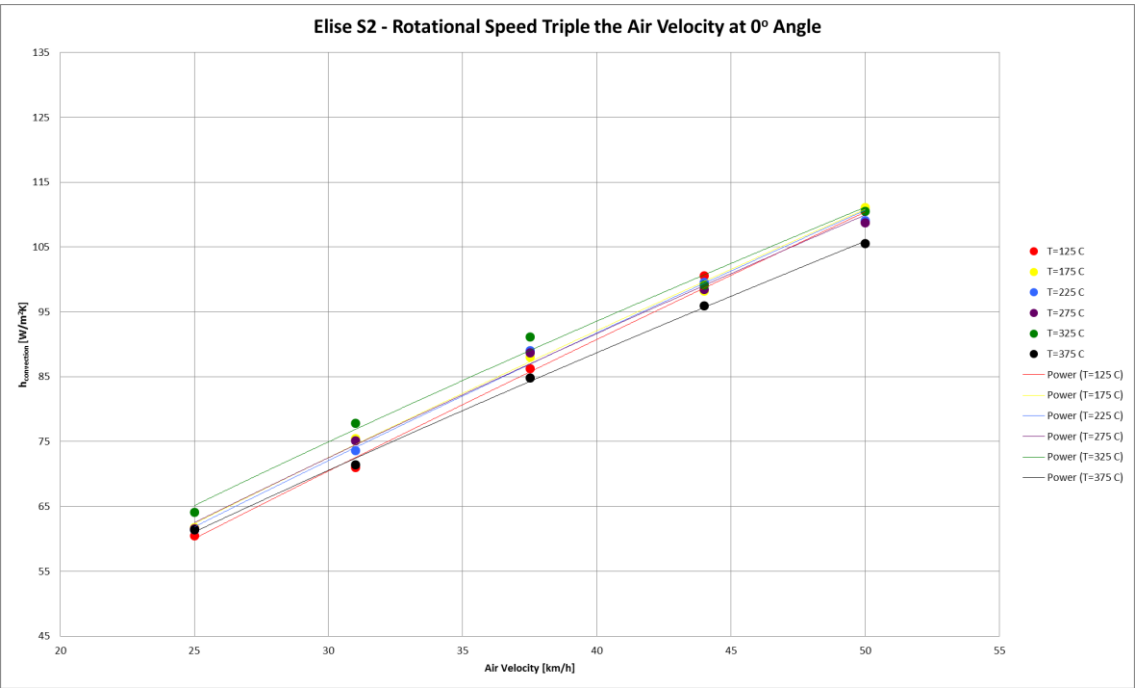
Appendix X – Influence of the Average Disc Surface Temperature

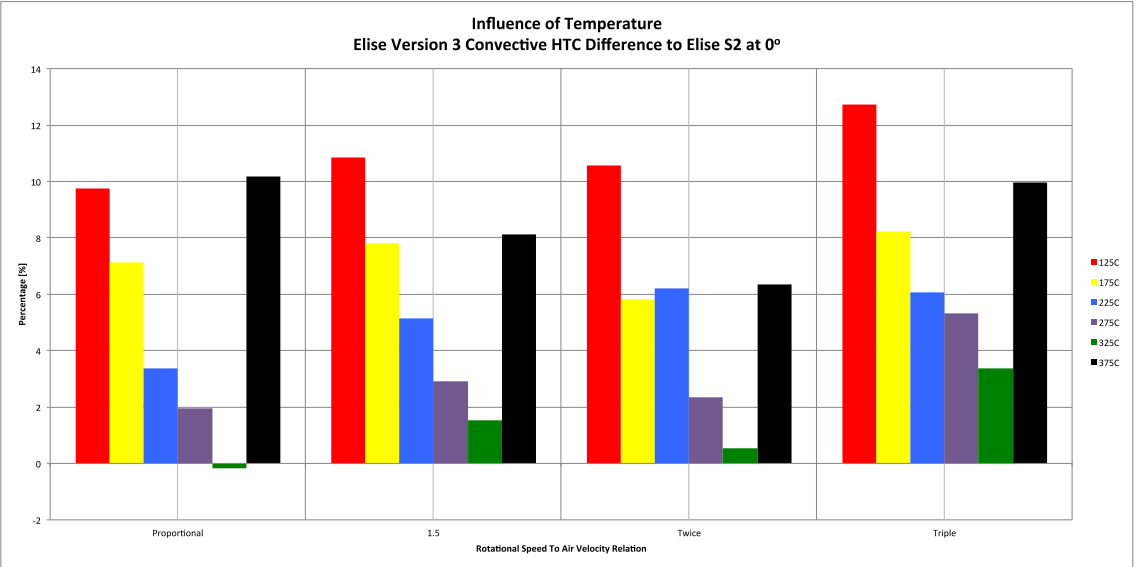
X.1 At 0° Angle



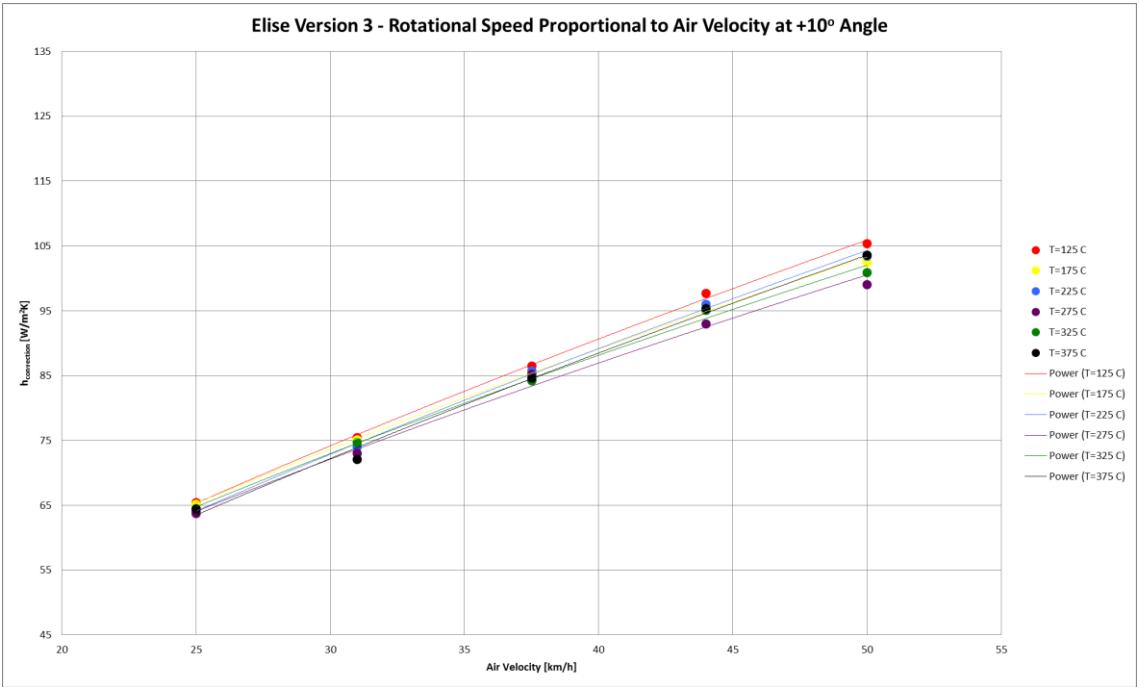
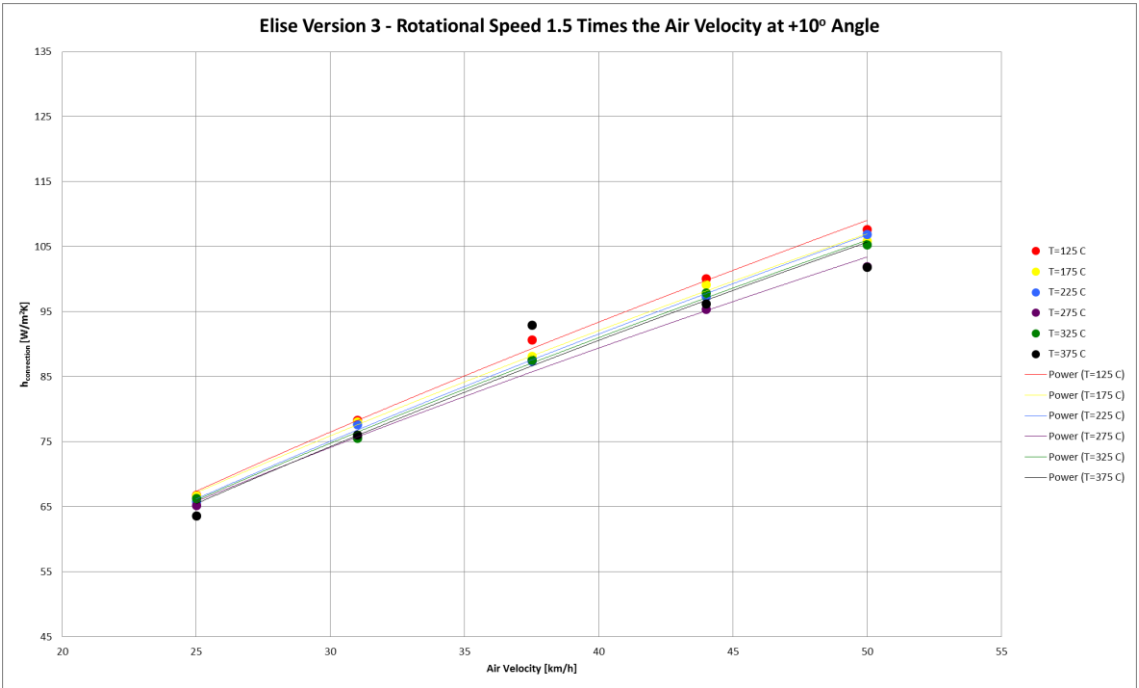


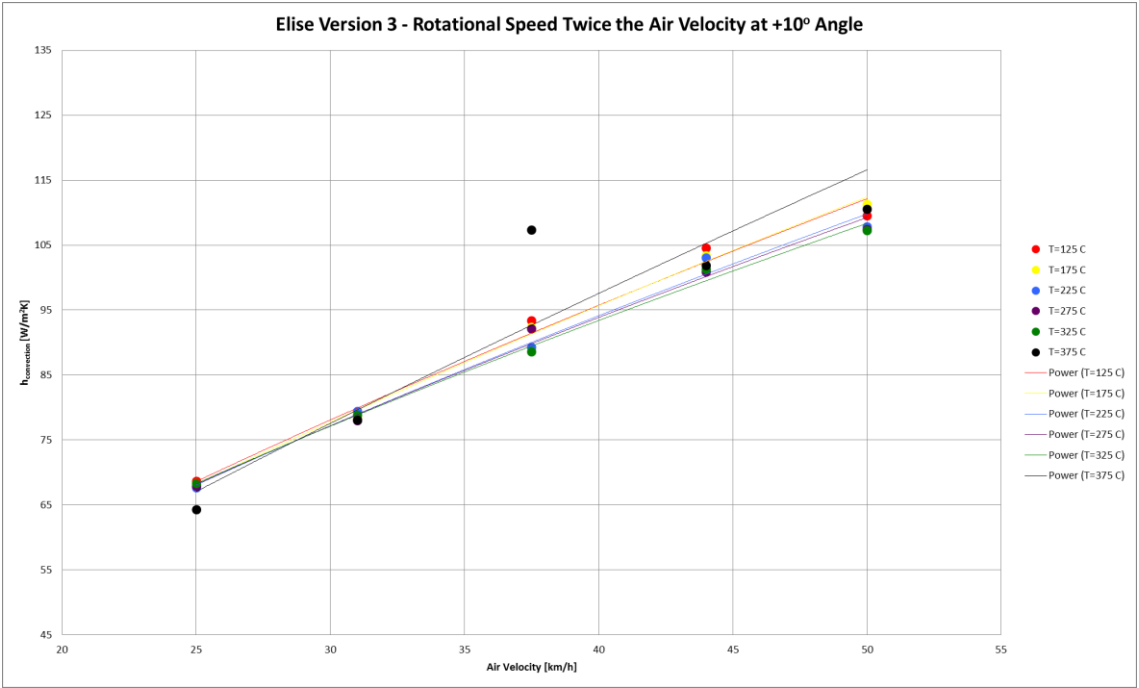
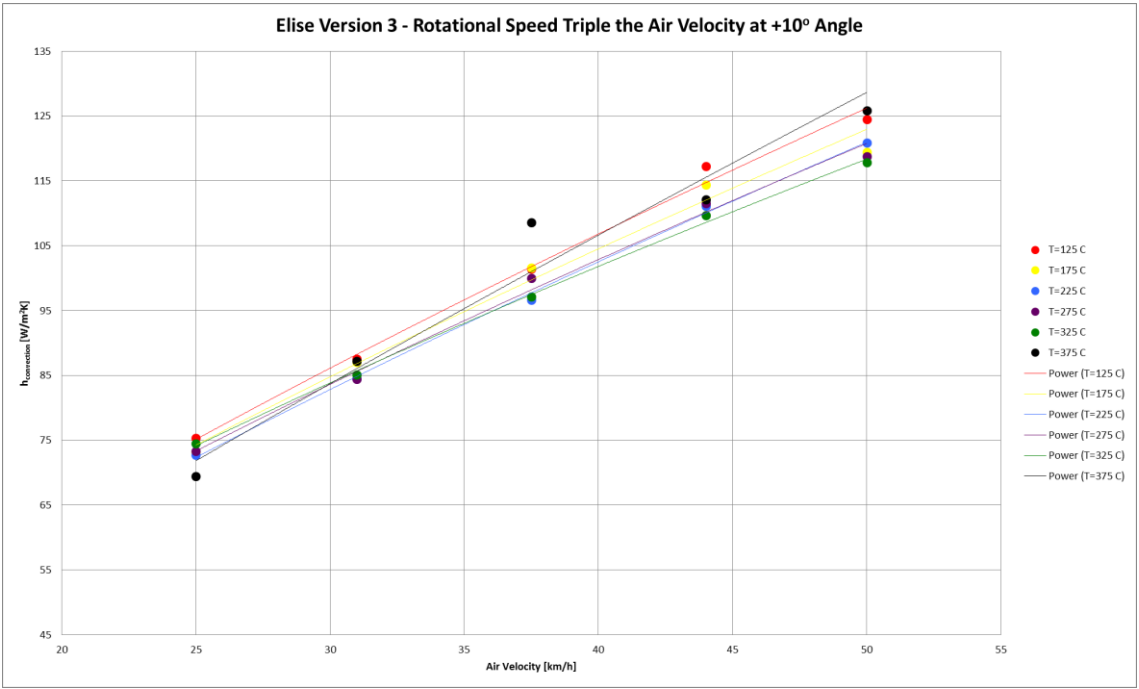


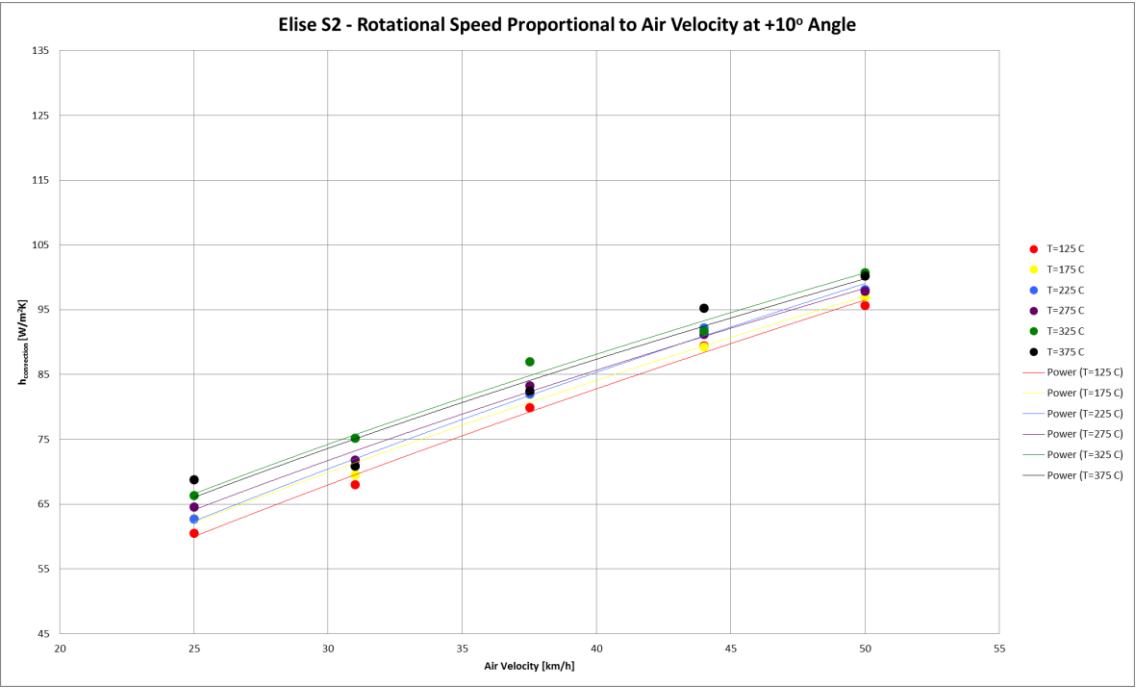
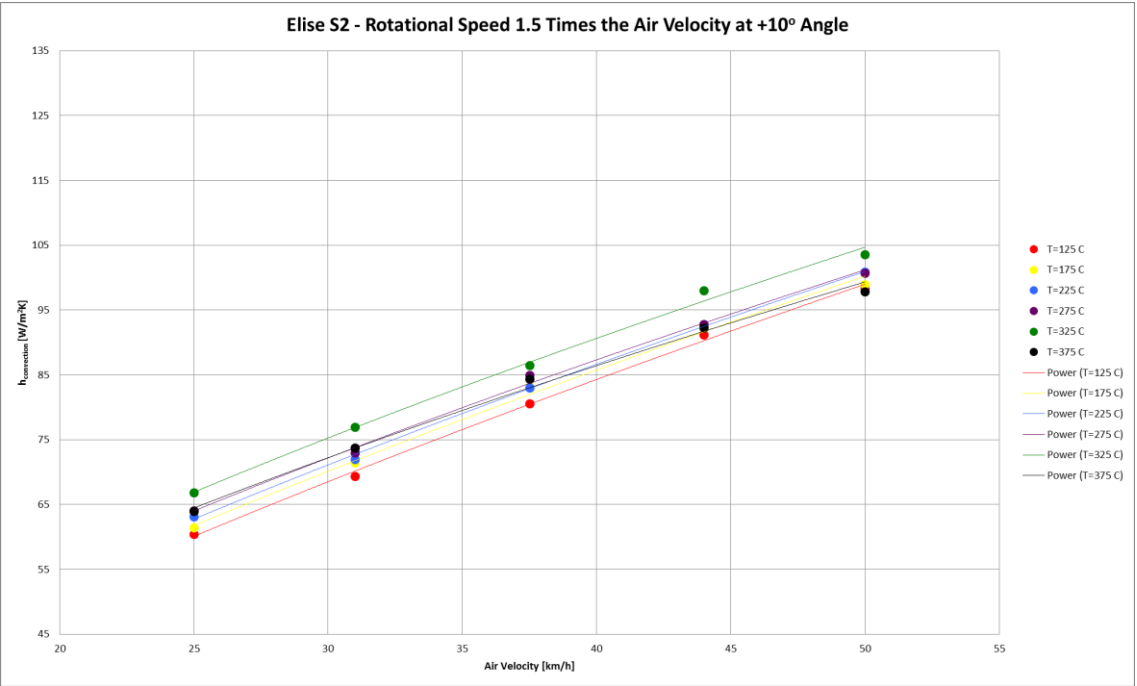


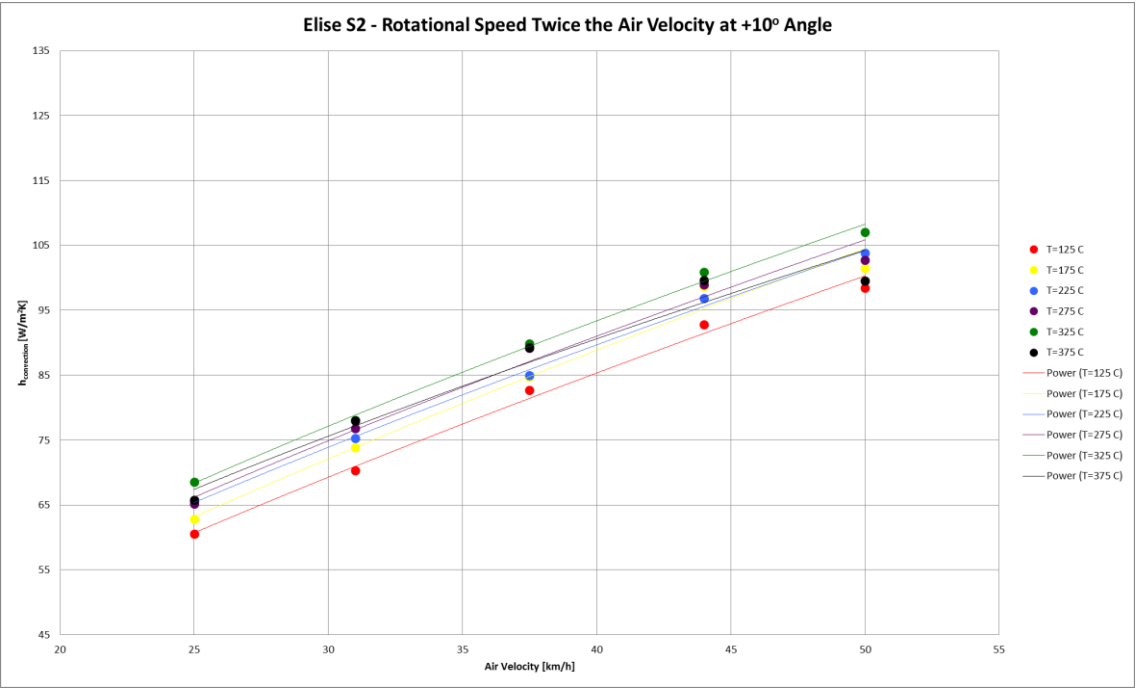
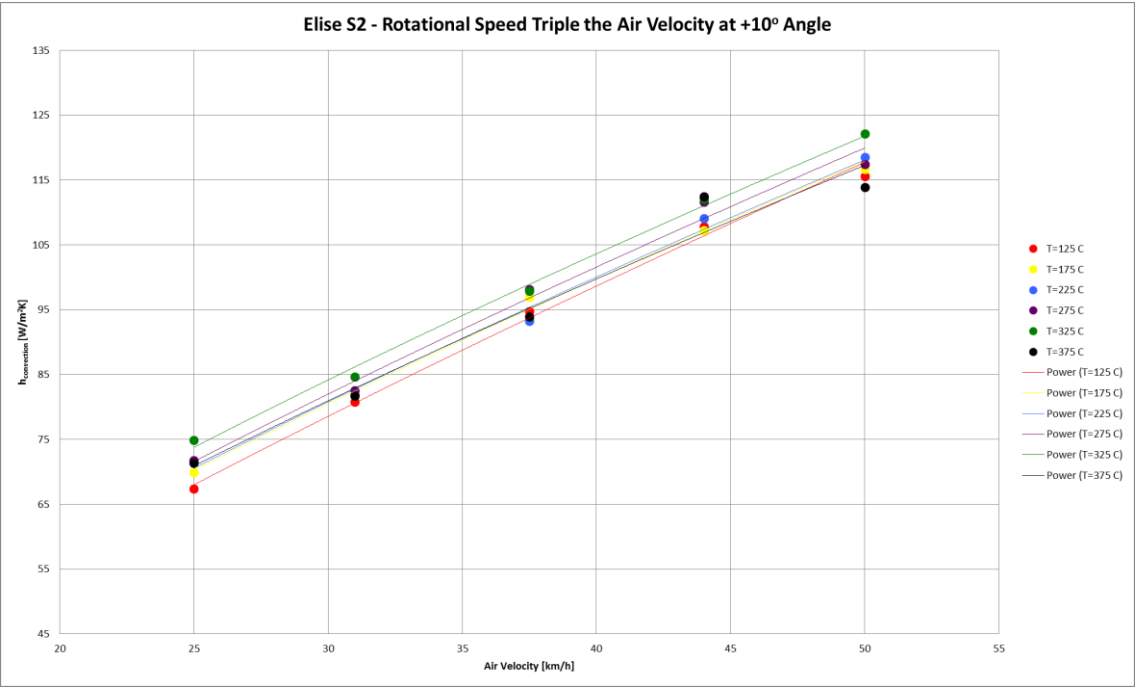


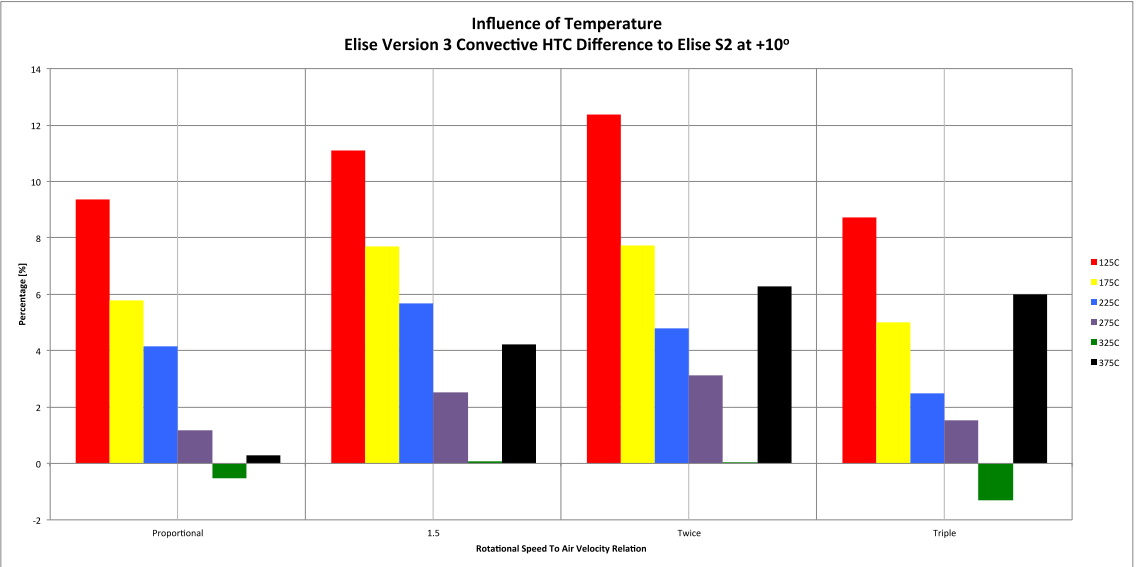
X.2 At +10° Angle











X.3 At -10° Angle

