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THE IMPORTANCE OF PAD ASPECT RATIO IN THE THERMAL ANALYSIS OF A REDUCED SCALE BRAKE

[1]Prabhu, Varun S.; [2]Alnaqi, Abdulwahab A.; [1]Brooks, Peter C.

[1]University of Leeds, United Kingdom; [2]Dept. of Automotive and Marine Engineering Technology, College of Technological Studies - PAAET, Kuwait.

KEYWORDS – Aspect ratio, drag braking, thermal imaging camera, two – dimensional finite element model.

ABSTRACT - The performance of a brake system can be assessed in its early stages through reduced scale testing. An established scaling methodology was used in this study to scale the brake pads based on constant energy density. The primary aim was to investigate the impact of the aspect ratio of a scaled brake pad on the thermal characteristics of a scaled disc during drag braking conditions.

Brake pads with four different aspect ratios were used against the scaled disc and these were scaled relative to the brake disc and pads of a medium sized passenger car. Using relevant scaling relationships, a speed–pressure matrix, consisting of nine tests, was derived and utilised in the experimental analysis. The tests were conducted using a conventional brake dynamometer. Temperatures were measured using thermocouples on the disc and pad and, for specific tests a thermal imaging camera was also used. The experimental results showed that a strong dependency exists between the maximum rotor temperature and the pad aspect ratio. This suggests that consideration of pad aspect ratio be reflected within the scaling process. A relationship between the circumferential and radial dimensions of full scale and reduced scale brake pads was established based on their measured thermal performance.

The experimental results were also used to validate a finite element model of the system. This was subsequently able to highlight important limitations associated with the model that apply equally to simulations of full and small scale brakes.

INTRODUCTION

A combination of change in potential and kinetic energy associated with vehicle motion is converted into heat energy at the pad – disc interface during a braking process [1]. The brake system therefore permits control of the vehicle speed (maintenance of a constant speed or its progressive reduction) across the vehicle duty cycle. Extreme braking conditions generate considerable quantities of thermal energy that transfer to different parts of the brake assembly which might cause catastrophic failure if not properly managed. In order to minimise the likelihood of such failure, the performance of brake system can be assessed using a reduced scale testing technique. The main advantages of small scale testing is the low cost and short operation time [2]. To conduct the reduced scale test on a brake dynamometer, the friction pair (brake pad and rotor) along with other system parameters must be appropriately scaled in accordance with established scaling theory [2-6]. Alnaqi *et. al* [2] furnished scaling relationships between full scale and reduced scale parameters. In addition, the full scale values of other parameters such as sliding velocity, pad pressure and energy density were preserved within the reduced scale setup. This work makes use of the methodology presented in [2] along with the same experimental setup that successfully demonstrated good correlation between experiment and simulation.

Work by Desplanques *et. al* [4] goes further in that they suggest the scaling should follow a similitude rule. This states that the ratio of the area of the frictional rubbing surface on the disc to that of the pad should also be preserved between the full and reduced scale brakes. During their investigations on frictional heating due to third body particles at the pad – disc interface, Talati *et. al* [7] and Mazidi *et. al* [8] showed that when loaded with a uniform pressure, the disc and pad surface temperatures increase with increase in radius which is due to increase in sliding velocity. However, when uniform wear was introduced there was no increase in temperature in the radial direction. In reality, insufficient cooling away from the outer perimeter of the brake pad and change in pad arc length from the inside to outside rubbing radius may be two reasons that lead to an increase in temperature. These observations provided motivation to the authors of this paper for investigating the influence of pad aspect ratio on temperature rise on a disc surface when using a reduced scale testing technique.

Thus, this study aims to determine the significance of pad aspect ratio in a thermal analysis when scaled brake pads are used against a scaled disc during a drag braking event. The vehicle was assumed to move with three estimated speeds over three different gradients. Three different estimated line pressures were applied during this motion to maintain constant velocity, thereby simulating drag braking. The tests were conducted using a brake dynamometer with scaled disc rubbing against four sets of pads, each having a different aspect ratio. For safety reasons, the maximum rotor temperature was limited to 300°C. A two dimensional finite element model, built using ABAQUS, was validated using the experimental results

EXPERIMENTAL ANALYSIS

The experiments in this study were conducted using a scaled version of a medium sized passenger car disc brake in which the scaling factor was 5.5. Table 1 shows some of the properties of the scaled disc being used alongside those of the actual full scale disc.

Table 1: Properties of Scaled Disc and full scale disc.

Properties	Full scale disc	Scaled disc
Disc Material	Grey Cast Iron	Grey Cast Iron
Diameter of the Disc	240mm	124mm
Mean Rubbing Radius	102.8mm	50mm (estimated)

Four sets of brake pads, manufactured by Wilwood, were used against the scaled grey cast iron disc. According to Wilwood, these brake pads are manufactured for high performance braking with details available through their catalogue [9]. All the four pairs of brake pad retained the same mean rubbing radius of 50 mm and had approximately the same area, controlled to within $\pm$ 3 mm^2 but had different aspect ratios. These four brake pads are designated as Pad A, B, C and D. Figure 1 illustrates the brake pads whilst Table 2 provides the details of their geometry and aspect ratio along with the full scale pad.

Aspect Ratio (AR) is given by:

$$\mathbf{AR} = \frac{\textbf{Width of the scaled brake pad (W)}}{\textbf{Length of the scaled brake pad (L)}}$$

Table 2: Essential Pad geometry.

Pad designation	Pad length (L) [mm]	Pad width (W) [mm]	Aspect Ratio (W/L)	Pad area [mm^2]	Equivalent arc angle [°]
Full scale	74	39	0.52	2886	43
Pad A	24	22	0.91	528	27
Pad B	26.5	20	0.75	530	30
Pad C	29.5	18	0.61	531	33
Pad D	44	12	0.27	528	51

During the experiments, the brake pads were supported within a two opposed piston Wilwood caliper. The details of the rig along with other components used are shown in Figure 2. The brake is actuated by an electric actuator whose displacement output energises a conventional master cylinder. To assist in the replication of an actual braking event, air was drawn over the brake assembly using available overhead ducting as shown in the Figure 2. Two thermocouples, one each on the disc and pad were used to measure the temperature attained by these components. The disc thermocouple was placed diametrically opposite to the caliper on the mean rubbing radius of the disc as shown in Figure 3. The reason for placing it in this location was to allow the thermal imaging camera to capture the radial temperature distribution immediately after the trailing edge of the brake pad. The pad thermocouple was bonded to the rear of the friction material through a hole drilled into the back plate. The thermal imaging camera was used to image the temperature profile across the brake rotor at the trailing edge of the pad. A LabVIEW® program was written to control the movement of the actuator and to log the temperatures attained by the disc and pad [3].

To facilitate the modelling exercise and generate an appropriate test matrix, the gross weight of the vehicle at full scale was assumed to be 1400 kg and the proportion of the braking effort dissipated by the front brake was assumed to be 0.75 of the total. Initial tests revealed that the magnitude of the interface coefficient of friction had a value of around 0.5. The equivalent gradients down which the full scale vehicle would have driven were calculated to be 4.5, 5 and 6° and the resulting speed-pressure matrix is shown in Table 3. A total of 36 tests were conducted (9 for each aspect ratio). The time taken to reach the target temperature of 300°C by each of the disc-pad combinations was noted for each test. The mean rubbing radius for all tests was maintained at 50 mm. The thermal imaging camera in conjunction with the thermocouples was used to record data during the high pressure tests.

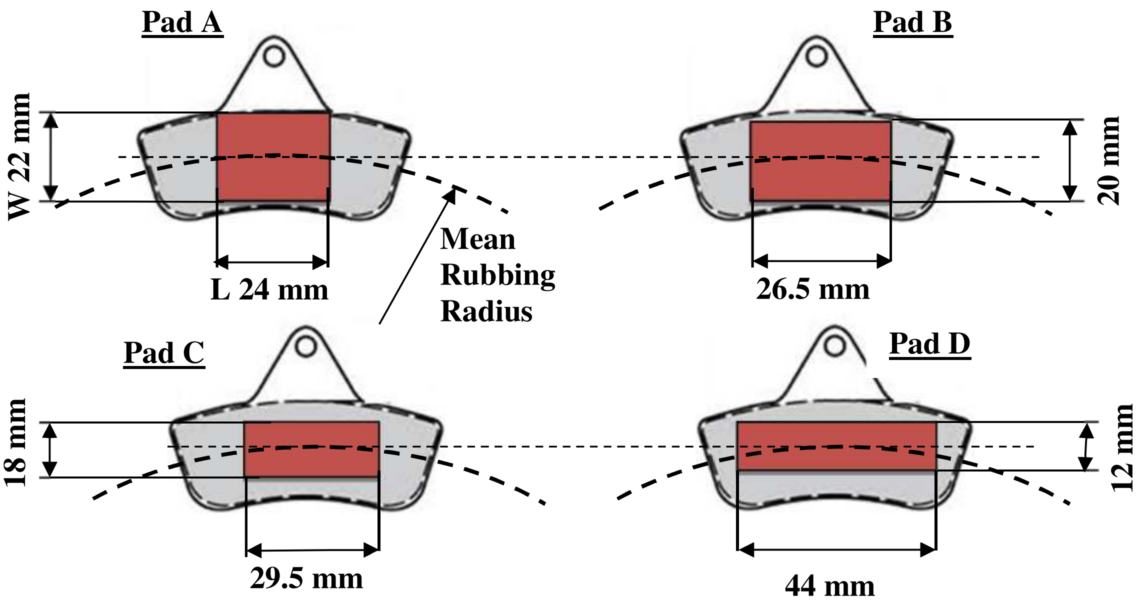


Figure 1: Brake Pads with Different Aspect Ratios (edited [9]).

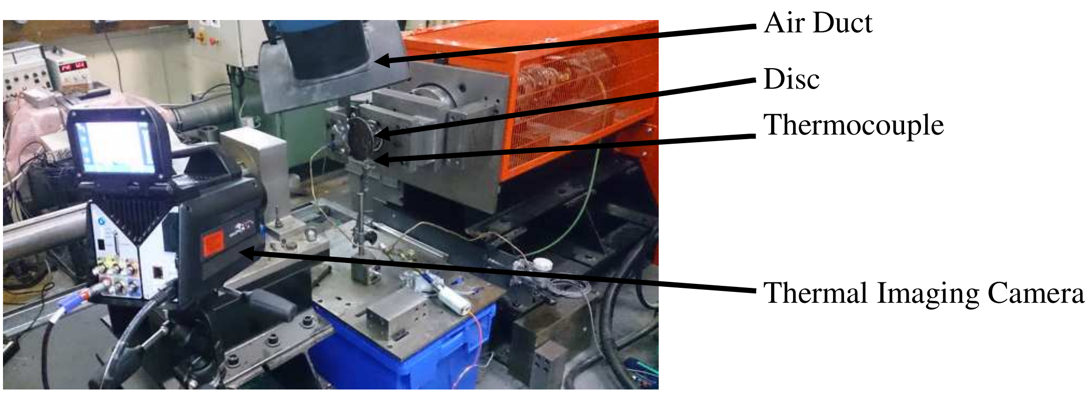


Figure 2: Full Scale Brake Dynamometer Test Rig.

Table 3: Speed – Pressure Matrix.

Speed			Pressure		
Level	**kmph**	**rpm**	**Low (1.5 bar)**	**Medium (2.5 bar)**	**High (4.5 bar)**
Low	**5.2**	**185**			
Medium	**15.5**	**545**			
High	**25.8**	**925**			

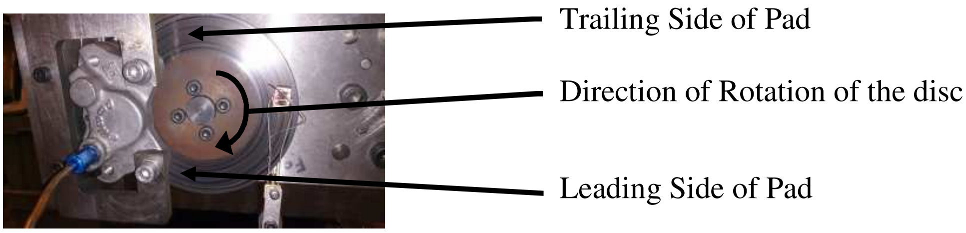


Figure 3: Position of Thermocouple on the disc.

In Table 3, the arrows indicate the order in which the experiments linked to a particular disc-pad combination were conducted, starting with the low speed/low pressure conditions. The low pressure/low speed experiments also facilitated the bedding-in process for the friction pair as the window of time in which to conduct the experimental work was limited.

NUMERICAL MODELING

A series of four similar finite element models, one associated with each pad, that capture the essential features of the experiment were constructed using Abaqus CAE software and a typical structure is shown in Figure 4. The 2D model represents only the disc and utilises 4 noded quadrilateral heat transfer elements, arranged in a regular mesh. Symmetry permits the thickness of the model to be half that of the actual disc. Free surfaces were assumed to be flat. The model is able to reflect the change in pad width, W, through adjustment of this parameter as indicated in Figure 4. The impact of change in pad length, L, on the thermal response of the disc is linked to the concept of the rotating heat source and is reflected by change in the duration of the pulse of heat into the model per revolution of the disc. This is a function of the equivalent pad arc angle, Table 2, expressed as a proportion of 2π and the rotational velocity of the dynamometer or the combination of vehicle speed and wheel radius. The equivalent pad arc angle is defined as the angle subtended between radial lines struck from the disc centre that pass through the intersection of the leading and trailing edges of the pad at the mean rubbing radius. The transient response of the model therefore consists of alternate heating and cooling steps. The convective heat transfer coefficient was estimated to be 30 W/m^2K and heat transfer due to radiation was assumed to be negligible. The heat partition ratio between the disc and pad was determined to be 0.9 (90% of the total heat flux transferring to the disc). During the heating phases within the transient solution, a heat flux was applied across the rubbing surface (the pad width in Figure 4) of the disc. In this study, a uniform heat flux was assumed despite several other workers, for example [7, 8], demonstrating this to be a simplification of the real world behaviour. A mesh sensitivity analysis was performed prior to validation and it was found that a mesh size of 0.25 mm yielded near mesh independent temperature fields. The model was validated for one particular case, Pad C running the high speed/high pressure combination. The models follow the same trend for all the other tests [11].

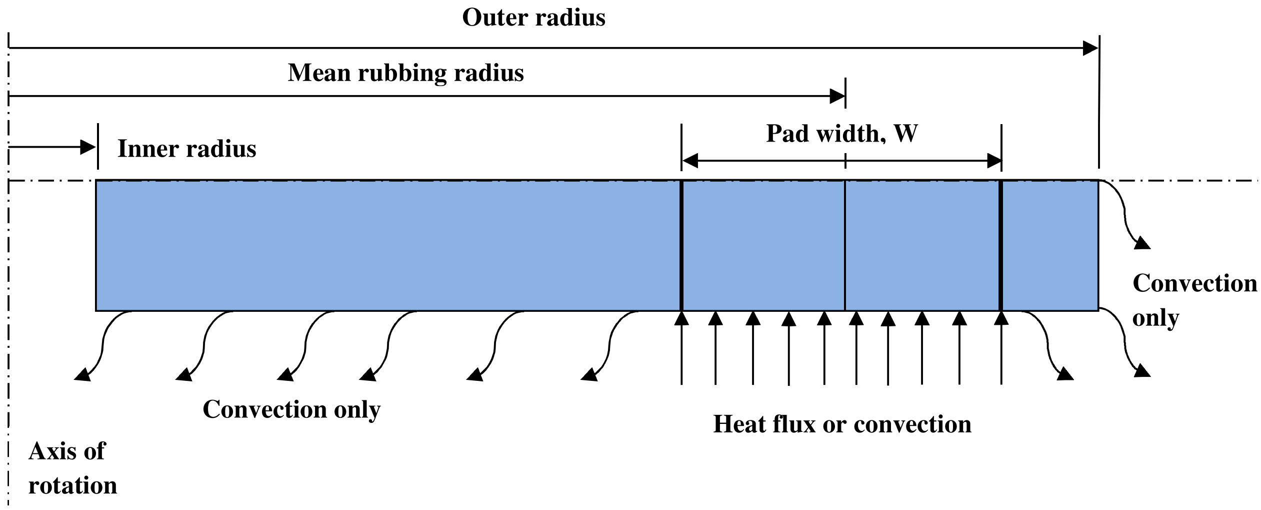


Figure 4 Outline of finite element model.

RESULTS

Temperature-time histories, showing the disc thermocouple response are shown in Figure 5. Each graph is for a specific speed-pressure combination and presents the thermal response of the rotor to each of the designated pads. It is evident that the target temperature is not attained in all tests. However, the rate of increase in disc surface temperature, particularly during the early stages of the braking event, proved sensitive to change in pad aspect ratio. Pads A and D were most sensitive whilst Pad C demonstrated the least aggressive response. Differences in the maximum temperature attained during each test are clear to see and are most notable in the high pressure tests. The disc ran coolest when partnered with Pad C. The corresponding time required to heat disc-pad combination to the target temperature also differs. These observations support the hypothesis that the choice of pad aspect ratio clearly impacts on the thermal performance of the scaled brake rotor.

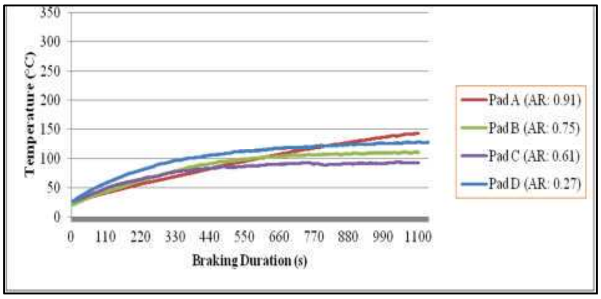
Further insight into the experimental results can be found through comparison of the radial (width) and circumferential (length) dimensions of the full scale pad to the scaled pads A to D. Two ratios are defined to permit this comparison:

$$\text{Length Ratio} = \frac{\text{Length of full scale brake pad } (L_F)}{\text{Length of scaled pad } (L_S)}$$

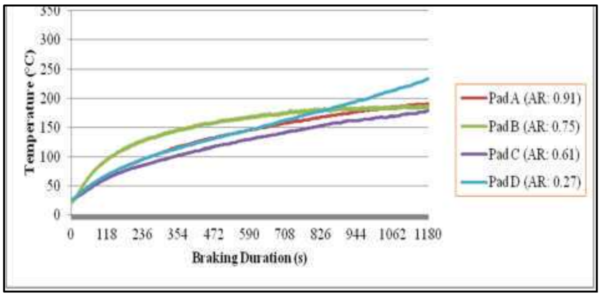
$$\text{Width Ratio} = \frac{\text{Width of full scale brake pad } (W_F)}{\text{Width of scaled pad } (W_S)}$$

and the absolute difference between these quantities is:

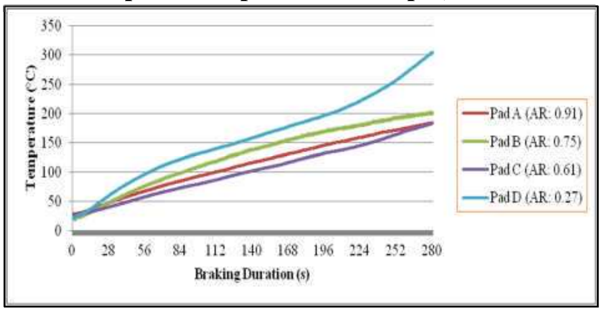
$$\Delta = \left| \frac{L_F}{L_S} - \frac{W_F}{W_S} \right|$$



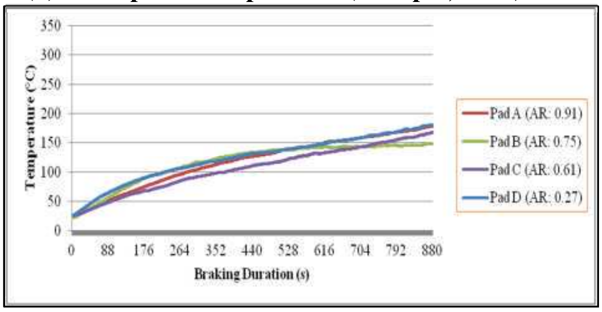
(a) Low speed-Low pressure (185 rpm, 1.5 b)



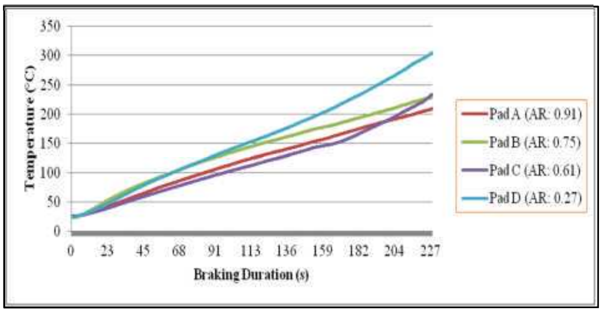
(b) Med speed-Low pressure (545 rpm, 1.5 b)



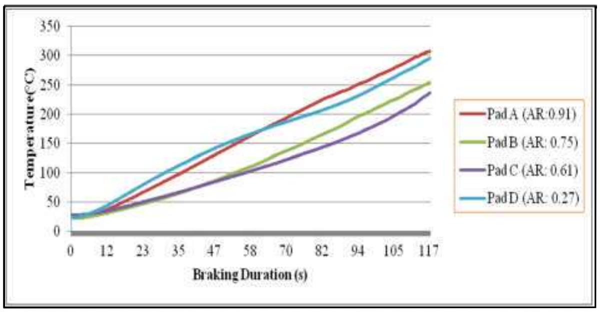
(c) High speed-Low pressure (925 rpm, 1.5 b)



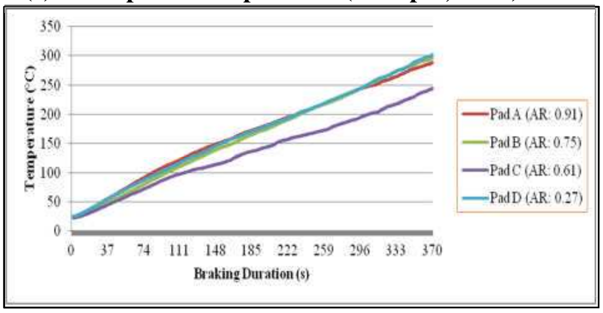
(d) Low speed-Med pressure (185 rpm, 2.5 b)



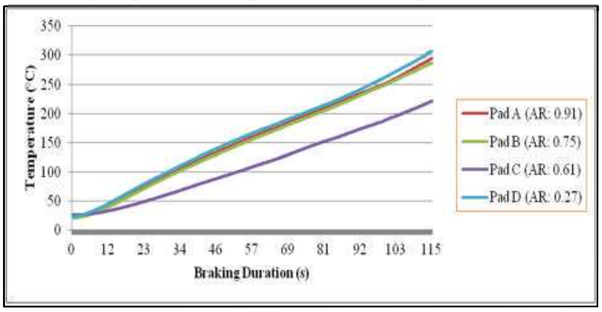
(e) Med speed-Med pressure (545 rpm, 2.5 b)



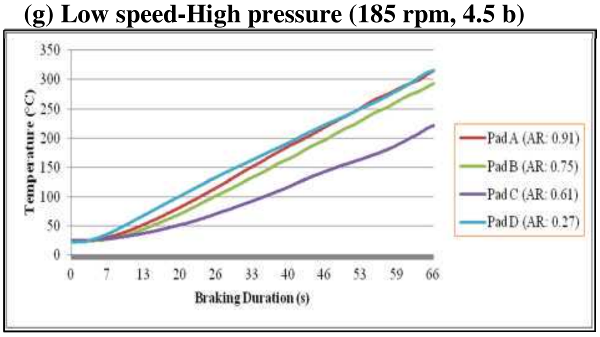
(f) High speed-Med pressure (925 rpm, 2.5 b)



(g) Low speed-High pressure (185 rpm, 4.5 b)



(h) Med speed-High pressure (545 rpm, 4.5 b)



(i) High speed-High pressure (925 rpm, 4.5 b

Figure 5: Experimental disc surface temperature time histories as a function of pad aspect ratio.

The term Δ tends to zero as the geometry of the scaled pad approaches that of the full scale. The results of this simple exercise, Table 4, demonstrate a connection between the full and scaled pads when viewed alongside the experimental data in Figure 5. When partnered with scaled pads A and D, the thermal response of the rotor is least favourable and the geometry of these pads deviates most from that of the full scale pad as indicate by the magnitude of Δ in Table 4. However, reduction in this deviation as seen with Pads B and C enhances the thermal response of the rotor and reduces the maximum temperature.

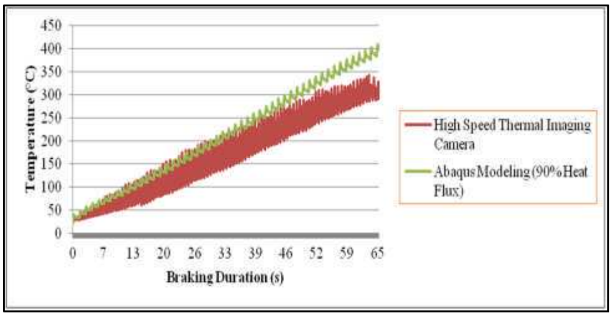
Table 4: Comparison between full and reduced scale pads

Pad designation	**Pad length (L) [mm]**	**Pad width (W) [mm]**	**Aspect Ratio (W/L)**	**Length Ratio (L_F/L_S)**	**Width Ratio (W_F/W_S)**	**Δ**
Full scale	74	39	0.52	-	-	-
Pad A	24	22	0.91	3.08	1.77	1.31
Pad B	26.5	20	0.75	2.79	1.95	0.84
Pad C	29.5	18	0.61	2.50	2.16	0.34
Pad D	44	12	0.27	1.68	3.25	1.57

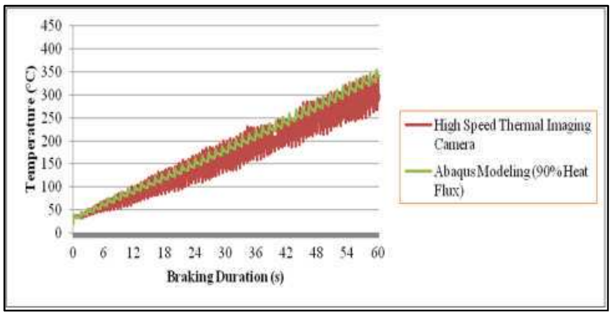
A comparison of the experimental temperature time histories with those of the equivalent simulations under the high speed/high pressure test conditions are shown in Figure 6. The simulated rotor response when running with either Pad B or C demonstrates good agreement with experimental data. Pads B and C have aspect ratios that tend towards those of the full scale component. Conversely, the simulated rotor response when running against Pads A or D tends to diverge from the experimental measurement with increase in time. The model overestimates the rotor temperature when running with Pad A and when working with Pad D, the temperature of the disc is underestimated.

These findings are illustrated graphically in Figure 7 which plots the quantity ΔT against pad aspect ratio. ΔT is defined as the absolute difference between the simulated disc temperature and the target temperature of 300°C at the time when the latter is achieved during the dynamometer test. The model response is clearly sensitive to change in the aspect ratio and this suggests the engineer should give due consideration to this when undertaking any modelling exercise in which pad aspect ratio is treated as a variable. The potential exists to specify user defined bounds on the extent to which aspect ratio can be changed before the need to re-validate the model becomes apparent. Examples of such limits are included in Figure 7 in which a ±15% change in aspect ratio has been assumed. As only a limited number of data points have so far been determined, the boundaries on Figure 7 are illustrative.

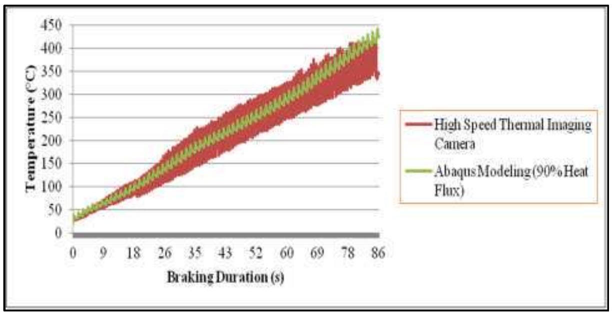
Further experimental work is underway to generate additional data points for inclusion within Figure 7 that will clarify the trend of this emerging relationship. This will enhance the confidence with which the above boundaries can be specified and the extent to which aspect ratio can be changed within the model before re-validation is required.



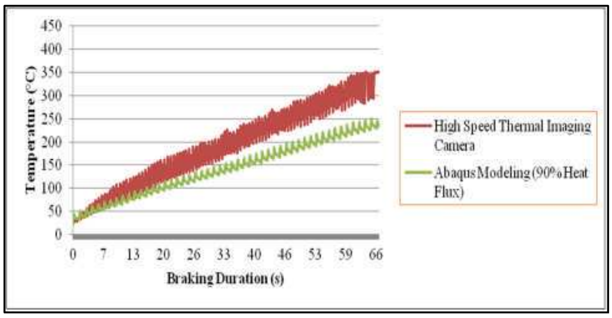
(a) Pad A



(b) Pad B



(c) Pad C



(d) Pad D

Figure 6: Comparison of simulation and experimental data at high speed and high pressure (925 rpm/4.5 Bar).

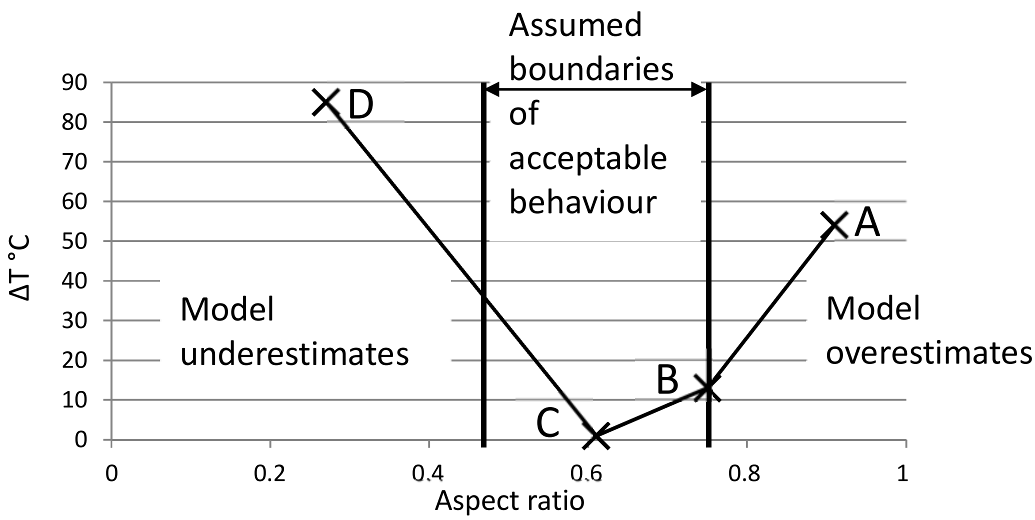


Figure 7: Performance of validated model as a function of aspect ratio.

CONCLUSIONS AND FUTURE WORK

This work has focused on the investigation of the importance of the aspect ratio of a scaled pad and its influence over the thermal response of the scaled disc. The work was conducted specifically for drag braking conditions. The thermal response of the scaled rotor was shown to be sensitive to change in the aspect ratio of the scaled pad which supports the findings of [4] and suggests that the methodology of [2] be refined to reflect this fact. From this, it follows that there exists a pad aspect ratio that minimises the thermal response of the rotor and this idea should apply equally to the full scale disc-pad combination. In this work, the magnitude of the full scale pad aspect ratio suggests that the minimisation of the maximum disc temperature has been addressed. The validated 2D finite element model of the reduced scale brake has been shown to exhibit limitations in its use linked to change in pad aspect ratio. This has implications with regard to the conduct of any parametric study that could be undertaken with the model. Further experimental work is required to better define nature the relationship between the model and aspect ratio. In addition, further case studies are required to strengthen these findings and these should be extended to include constant g stops.

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