

CRANFIELD UNIVERSITY

Zicheng Wang

Investigation of Disc Brake Interface Strain Distributions
Using Fibre Bragg Grating Sensors

School of Aerospace, Transport and Manufacturing

PhD in Transport Systems

PhD

Academic Year: 2019 - 2022

Supervisor: Prof. Stephen James

Associate Supervisor: Dr. Marko Tirović

November 2022

CRANFIELD UNIVERSITY

School of Aerospace, Transport and Manufacturing

PhD in Transport Systems

PhD

Academic Year 2019 - 2022

Zicheng Wang

Investigation of Disc Brake Interface Strain Distributions
Using Fibre Bragg Grating Sensors

Supervisor: Prof. Stephen James

Associate Supervisor: Dr. Marko Tirović

November 2022

© Cranfield University 2022. All rights reserved. No part of this publication may be reproduced without the written permission of the copyright owner.

ABSTRACT

The processes occurring at the friction brake interface are complex due to the high interface pressure, and heat and wear, which change continually and are governed by the thermo-elastic instability phenomenon. Complexities in modelling are mainly related to the inability to establish all necessary material properties. Experimental investigations, on the other hand, are difficult due to the challenging working environment and limited space, whilst the sensors used should not disturb the very process that is being monitored.

The Thesis uses optical Fibre Bragg Grating (FBG) strain sensors, having a small diameter of order 250 μm and temperature resistance of several hundred degrees. The brake pads are modified by creating shallow and narrow grooves in which the optical fibre sensors are installed. In such a manner, the strains are measured in the close proximity to the friction surface, with minimum modifications and influence on the interface contact. Two special rigs were used, one for static and quasi-dynamic testing, and the other for dynamic brake applications, employing the complete brake assembly and accurately controlled test environment.

The static loading phase involved step-wise hydraulic pressure changes (0bar to 120bar) and variations (pressure applied to the piston sets differed by $\pm 10\text{bar}$) to understand the brake pad interface performance, while the quasi-dynamic measurement phase involved the application of torque levels (200Nm, 600Nm & 1000Nm) at varying hydraulic pressure levels (40bar, 80bar and 120bar). The measurements confirmed the FBG sensors' suitability for interface strain measurements, with high level of accuracy and repeatability, within a wide range of brake operating conditions. Comparisons with finite element (FE) analyses in the static and quasi-dynamic conditions showed very good agreement.

In the dynamic phase, the disc brake system is exposed to a (approx.) 50rpm disc rotational speed and 30bar applied hydraulic pressure, which represents the first reported attempt to apply the optical sensing technique on a disc brake under dynamic loading conditions. FBG sensors were able to follow the fast-changing strain distributions at the pad/disc interface, and a means of correcting the

influence of temperature on the sensor response was implemented. The possibility of direct measurement and monitoring of a truly dynamic change in the strain distributions at the entire pad frictional surface during braking lays the foundations for dynamic monitoring of pressure and temperature distributions at the brake pad frictional interface.

Keywords:

Interface Strain Distributions, Fibre Bragg Grating (FBG), Finite Element Analysis (FEA), Interface Pressure Distributions (IPD), Disc Brake

ACKNOWLEDGEMENTS

I want to express my sincere appreciation to my supervisors, Dr. Marko Tirović and Prof. Stephen James, for their sedulously patient guidance and continuous help throughout the duration of the whole PhD research project. Their advices and recommendations were extremely valuable for accomplishing such a challenge project. They always gave me plenty of time to discuss for various topics, and encouraged me with powerful words to make me confidence on both the research project and the human being.

I would like to thank the people that were provided different views to ensure the feasibility of the research project, in particular, Prof. Ralph Tatam and Dr. James Barrington. In addition, I would also like to thank Kelvin White who helped me and worded with me in the laboratories of Building 43. Meanwhile, I would like to give my deep appreciate to Prof. David Bryant, who provided a well-prepared laboratory and full support at the University of Bradford to complete dynamic measurements successfully.

Furthermore, I would show my gratitude to my parents for everything they offered me. They gave me a hand without any hesitation whenever I needed and encouraged me for making any decision to have enough confidence to finish this research project ideally.

Last but not least, I would like to provide my deeply grateful to my friends for all of laughs, advices and memories over these years. You have always been there for me through all the ups and downs during this project. I'm so lucky to have such best friends to defeat so many challenges when doing this research project.

In the end, without any of these people, this thesis could be hardly to complete, and the whole PhD research project would be impossible to come true.

TABLE OF CONTENTS

ABSTRACT	i
ACKNOWLEDGEMENTS.....	iii
LIST OF FIGURES.....	x
LIST OF TABLES	xvii
LIST OF EQUATIONS.....	xviii
LIST OF ABBREVIATIONS	xix
NOMENCLATURE	xxi
1 INTRODUCTION.....	1
1.1 Background.....	1
1.2 Aims and Objectives	3
1.3 Thesis Structure.....	3
2 LITERATURE REVIEW	5
2.1 Introduction	5
2.2 Recent Brake Publications.....	5
2.2.1 Brake Noise.....	5
2.2.2 Brake Vibrations.....	9
2.3 Interface Pressure Distributions.....	12
2.4 Optical Measurement Technology	16
2.4.1 Optical Fibre Sensing Applications.....	17
2.4.2 Applications in the Automotive Field.....	27
2.5 Summary	31
3 EXPERIMENTAL FACILITIES.....	33
3.1 The Test Rig	33
3.1.1 Brembo Land Rover Disc Brake Assembly	33
3.1.2 Two-Channel Control Hydraulic Pressure System	45
3.1.3 Initial Test: Pneumatic & Hydraulic Pressure Relationship.....	50
3.2 Brake Pad Modification	54
3.3 FBG Sensors Deployment	56
3.3.1 Sensor Location Concept.....	57
3.3.2 Fabrication and Installation	59
3.4 Summary	63
4 EXPERIMENTAL PHASE 1 – STATIC LOADING ANALYSIS	65
4.1 Static Loading Test System Set-up.....	65
4.2 Fibre Bragg Grating (FBG) Test Plan and Validation	69
4.2.1 Main Test Conditions.....	69
4.2.2 Additional Test Conditions.....	71
4.3 Static Loading Test Results and Analysis	73
4.3.1 Data Processing.....	73
4.3.2 Results and Analysis: Recording and Distribution Graphs	75
4.3.3 Results and Analysis: FBG Sensor Contour Map.....	78

4.3.4 Uncertainty Analysis: Experimental Phase	85
4.4 Summary	90
5 FE ANALYSIS PHASE 1 – STATIC LOADING ANALYSIS	93
5.1 Finite Element Model	93
5.1.1 FE Model Concept.....	93
5.1.2 FE Model Optimisation	98
5.2 Finite Element Analysis.....	105
5.2.1 Approaches in Processing the Results.....	105
5.2.2 FEA Results and Analysis	106
5.3 Comparison of FBG Test and FEA Simulation Results.....	112
5.3.1 Approaches in Result Comparisons	113
5.3.2 Analysis of The Results.....	115
5.4 Summary	124
6 EXPERIMENTAL PHASE 2 – QUASI DYNAMIC ANALYSIS.....	127
6.1 Quasi Dynamic Test System Set-up	127
6.2 Fibre Bragg Grating (FBG) Test Plan and Validation	129
6.2.1 Main Test Conditions.....	129
6.2.2 Additional Test Conditions.....	131
6.3 Quasi Dynamic Loading Test Results and Analysis.....	132
6.3.1 Results and Analysis: Python Graphs	132
6.3.2 Results and Analysis: Recording and Distribution Graphs	134
6.3.3 Results and Analysis: FBG Sensor Contour Map.....	137
6.3.4 Uncertainty Analysis: Experimental Phase.....	144
6.4 Summary	147
7 FE ANALYSIS PHASE 2 – QUASI DYNAMIC ANALYSIS	149
7.1 Finite Element Model	149
7.1.1 FE Model Update	149
7.1.2 FE Model Optimisation	154
7.2 FEA Results and Analysis.....	163
7.2.1 Influence of Different Hydraulic Pressure and Torque Levels	164
7.2.2 Influence of Hydraulic Pressure Variations between Piston Sets ...	168
7.3 Comparison of FBG Test and FEA Simulation Results.....	171
7.3.1 Approaches in Result Comparisons	171
7.3.2 Analysis of The Results.....	175
7.4 Summary	191
8 EXPERIMENTAL PHASE 3 – DYNAMIC ANALYSIS.....	195
8.1 Dynamic Test Preparations.....	195
8.1.1 Brake Pad Modification (Cranfield University)	195
8.1.2 Dynamic Test System and Schedule (University of Bradford).....	198
8.2 Dynamic Loading Test 1	201
8.2.1 Raw Recorded Data Graphs	201
8.2.2 Data Processing.....	203

8.2.3 Processed Data Graphs and Contour Map	203
8.2.4 Strain Differences between Initial and Final Static Applications	210
8.3 Dynamic Loading Tests 2 & 3	213
8.3.1 Processed Data Graphs	213
8.3.2 Strain Differences of Dynamic Loading Applications	214
8.3.3 Strain Differences between Initial and Final Static Applications	217
8.4 Dynamic Cooling Test 4	222
8.4.1 Raw Recorded Data Graphs	223
8.4.2 Strain Differences between Initial and Final Static Applications	225
8.5 Dynamic Consecutive Test 5	227
8.5.1 Raw Recorded Data Graphs and Temperature Measurement	227
8.5.2 Processed Data Graphs and Repeated Dynamic Results	229
8.6 Brake Pad Surface Condition Analysis	233
8.7 Summary	238
9 DISCUSSION, CONCLUSION AND FUTURE WORKS	241
9.1 Discussion	241
9.2 Conclusion	246
9.2.1 Research Project Conclusions	246
9.2.2 Intellectual Contributions	246
9.3 Future Work	247
RESEARCH PUBLICATIONS	249
REFERENCES	251
APPENDICES	265
Appendix Chapter 3-A: CAD Drawings of Designed Components	265
Appendix Chapter 3-B: Manufacturing Machines	269
Appendix Chapter 3-C: Completed Assembly Phases	271
Appendix Chapter 3-D: The Assembly Process of The Two-Channel Control Hydraulic Pressure System	275
Appendix Chapter 3-E: The Pneumatic Pressure and Hydraulic Pressure Initial Measurement	277
Appendix Chapter 3-F: The Brake Pad Modification	279
Appendix Chapter 4-A: MATLAB Program Code	283
Appendix Chapter 4-B: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Direction Comparisons between Three Pressure Variations at 40bar Pressure	287
Appendix Chapter 4-C: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Direction Comparisons between Three Pressure Variations at 120bar Pressure	289
Appendix Chapter 4-D: Visco-elasticity with Different Pressure Level Test Results	291
Appendix Chapter 5-A: FEA Results for Different Mesh (Element) Size (CPRESS, E22)	293

Appendix Chapter 5-B: The Full Data for Full and Reduced Integration Types.....	295
Appendix Chapter 5-C: FEA Results for Different Friction Coefficients (CPRESS, E22)	301
Appendix Chapter 5-D: FEA Results for Different Young's Modulus Values (CPRESS, E22)	303
Appendix Chapter 5-E: Processed FEA Contour Map in Vertical (Y) and Horizontal (X) Directions for Three Hydraulic Pressure Variations (40bar) .	305
Appendix Chapter 5-F: Processed FEA Contour Map in Vertical (Y) and Horizontal (X) Directions for Three Hydraulic Pressure Variations (120bar)	307
Appendix Chapter 5-G: E11 and E22 Strain Results and Differences for Three Hydraulic Pressure Variations (40bar).....	309
Appendix Chapter 5-H: E11 and E22 Strain Results and Differences for Three Hydraulic Pressure Variations (120bar).....	313
Appendix Chapter 6-A: Recording and Distribution Graphs at the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface with 600Nm Toque	317
Appendix Chapter 6-B: Recording and Distribution Graphs at the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface with 1000Nm Toque	319
Appendix Chapter 6-C: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions between Three Test Stages with Equal Pressure (p1 = p2)	321
Appendix Chapter 6-D: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions between Three Test Stages with Equal Pressure (p1 = p2)	323
Appendix Chapter 6-E: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions Comparison between Three Pressure Variations with 40bar/200Nm.....	325
Appendix Chapter 6-F: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions Comparison between Three Pressure Variations with 120bar/1000Nm.....	327
Appendix Chapter 7-A: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.4$ (40bar/200Nm).....	329
Appendix Chapter 7-B: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.46$ (120bar/1000Nm).....	331
Appendix Chapter 7-C: E11 & E22 Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations (40bar/200Nm) ..	333
Appendix Chapter 7-D: E22 Strain Results for ± 10 bar Hydraulic Pressure Variations.....	335

Appendix Chapter 7-E: E11 Strain Results for ± 10 bar Hydraulic Pressure Variations.....	337
Appendix Chapter 8-A: The CAD drawings of The New Modified Brake Pad	339
Appendix Chapter 8-B: The Test Result Graphs of The Initial Static Tests	341
Appendix Chapter 8-C: The Raw Recorded Data Graphs of The Dynamic Loading Test 2 & 3.....	343

LIST OF FIGURES

Figure 2-1: Effect of the Young's modulus of the friction material on the brake squeal generation (Belhocine and Ghazaly, 2015).....	6
Figure 2-2: Complex eigenvalues for all configurations considered (Denimal et al., 2017)	7
Figure 2-3: Experimental Set-Up Scheme (3) Air Bushings, (5) Test Sample (Lazzari et al., 2018).....	8
Figure 2-4: The Full Vehicle Experimental Rig (Lindberg et al., 2013)	9
Figure 2-5: Comparisons of Distortions, DTV and Pressure during Braking Event (Bryant, 2021).....	10
Figure 2-6: Hot Spots on a Brake Disc Rubbing Surface (Bryant, 2010).....	10
Figure 2-7: Hot Spot Formation Types (Panier et al., 2004).....	11
Figure 2-8: Disc Brake Pad Showing Deformation under Actuation Force (Tirovic and Day, 1991)	13
Figure 2-9: Disc Brake Pad Friction Interface Pressure Distribution Predicted by FE Analysis (Tirovic and Day, 1991)	14
Figure 2-10: Pad–disc static interface pressures for 100 bar hydraulic actuating pressure: (a) FEA results, (b) PSP imprints and (c) and (d) processed results with increased level of filtering (Tirovic et al., 2019)	15
Figure 2-11: Diagram of FBG Principle and Strain Response (FBGS, 2022) ...	18
Figure 2-12: Passive Interrogation Methods.....	20
Figure 2-13: Active Interrogation Methods	21
Figure 2-14: Complex Optical Sources Used in FBG Interrogation	22
Figure 2-15: Photograph of Mode Shapes Using Laser Holographic Interferometry (Fieldhouse, 2013)	28
Figure 2-16: Linear Relationship Between Wavelength Change and Hydraulic Brake Actuation Pressure (Major et al., 2017)	29
Figure 3-1: Different Series of Cars with the Same Disc Brake System (a) Land Rover Discovery, (b) Range Rover Sport and (c) Range Rover Velar (Jaguar Land Rover Limited, 2022)	34
Figure 3-2: Actual Components of the Land Rover Disc Brake	35
Figure 3-3: Schematic Diagram of The Disc Brake Components and Adaptors: (1) Adaptor 2, (2) Brake Disc, (3) Outside Ring, (4) Adaptor 1, (5) Original Adaptor B, (6) M12x1.75x30 Bolt, (7) M10x1.5 Bolt, (8) M12x1.5x25 Bolt, (9)	

M12x1.5x35 Bolt, (10) M8x1.5 Bolt, (11) M16x1.5 Bolt, (12) Brake Calliper, (13) Original Adaptor A	36
Figure 3-4: Original Adaptor A and B for Test Table and Component Connections	36
Figure 3-5: The Calculation and Selection of Adaptors 1 & 2	37
Figure 3-6: The Concept of The Adaptor 1	38
Figure 3-7: The Concept of The Adaptor 2	40
Figure 3-8: The Concept of The Brake Disc Modification	41
Figure 3-9: The Concept of The Outside Ring	42
Figure 3-10: Final Productions of Designed Components	43
Figure 3-11: Exploded and Assembled Views of Land Rover Disc Brake Components	43
Figure 3-12: The Land Rover Disc Brake Assembly	44
Figure 3-13: The Symbol Diagram of The Two-Channel Control Hydraulic Pressure System	45
Figure 3-14: The Brake Calliper Modification Concept	47
Figure 3-15: The Assembly of The Two-Channel Control Hydraulic Pressure System	49
Figure 3-16: The Brake Calliper with The Protective Shield	50
Figure 3-17: Two Samples of The System Initial Test	51
Figure 3-18: Disc Brake Rig System Pressure Graph	53
Figure 3-19: The Brake Pad Modification Concept: Groove in Y Direction (a) and (b), Groove in X Direction (c), Groove Detail (d)	55
Figure 3-20: Modified Brake Pads with Designed Grooves	56
Figure 3-21: The Design Concept of FBG Sensor Locations	58
Figure 3-22: Phase Mask FBG Writing Schematic (Laser Focus World, 2022) [Left] and Optical Fibre Array Specifications (Corning Incorporate, 2022) [Right]	59
Figure 3-23: The Final Brake Pads with FBG Sensors	61
Figure 3-24: The Final Assembly of The Brake Pad Installation	62
Figure 4-1: Static Loading Test System Set-up	66
Figure 4-2: General SmartSoft Instrument Set-up before Experiments	67
Figure 4-3: Y-Direction Brake Pad Installation at The Outboard Side	73

Figure 4-4: The Python Program Screenshots	74
Figure 4-5: Recording Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface	76
Figure 4-6: Strain Distribution Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface	77
Figure 4-7: FBG Sensor Contour Maps for the Vertical (Y) Direction Comparison between Three Pressure Levels with Equal Pressure ($p_1 = p_2$)	80
Figure 4-8: FBG Sensor Contour Maps for the Horizontal (X) Direction Comparison between Three Pressure Levels with Equal Pressure ($p_1 = p_2$)	81
Figure 4-9: FBG Sensor Contour Maps for the Vertical (Y) Direction Comparison between Three Pressure Variations	83
Figure 4-10: FBG Sensor Contour Map for the Horizontal (X) Direction Comparison between Three Pressure Variations	84
Figure 4-11: The Repeatability Test Results	86
Figure 4-12: Visco-elastic Behaviour: Concept and Test Result.....	87
Figure 4-13: Inboard and Outboard Comparison: Test Results	89
Figure 5-1: The FE Model Components: (a) Brake Disc Section, (b) Brake Pad, and (c) Two Pistons	93
Figure 5-2: The FE Model Assembly: (a) Piston Side View and (b) Disc Side View	94
Figure 5-3: The Brake Pad Mesh: (a) Frictional Surface and (b) Backplate	95
Figure 5-4: The Disc Section Mesh (a) and The Piston Mesh (b).....	95
Figure 5-5: The FE Assembly Model: Boundary Conditions and Loads	96
Figure 5-6: The Definition of Contact Interactions in the FE Model	97
Figure 5-7: Different Element Sizes: (a) 2mm, (b) 2.5mm, and (c) 5mm	98
Figure 5-8: E11 (X Direction) FE Analysis Result for Different Element Sizes	100
Figure 5-9: The Comparison between Full and Reduced Integration Types ..	101
Figure 5-10: E11 (X Direction) Comparisons for Different Friction Coefficients	102
Figure 5-11: E11 (X Direction) Comparisons for Different Young's Modulus Values.....	104
Figure 5-12: FEA Result Element Selections (E22).....	105

Figure 5-13: Processed FEA Contour Maps in Vertical (Y) Direction for Three Hydraulic Pressure Levels with Equal Pressure ($p_1=p_2$)	107
Figure 5-14: Processed FEA Contour Maps in Horizontal (X) Direction for Three Hydraulic Pressure Levels with Equal Pressure ($p_1=p_2$)	108
Figure 5-15: E22 (Y Direction) Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations	110
Figure 5-16: E11 (X Direction) Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations	111
Figure 5-17: E22 (Y Direction) Result Comparisons between FBG and FEA for Equal Pressure Variation, $p_1=p_2=40\text{bar}$	114
Figure 5-18: E22 (Y Direction) Strain Results for Two Hydraulic Pressure Levels	115
Figure 5-19: FBG and FEA E22 (Y Direction) Strain Differences for Two Hydraulic Pressure Levels	117
Figure 5-20: E11 (X Direction) Strain Results for Three Hydraulic Pressure Levels	118
Figure 5-21: FBG and FEA E11 (X Direction) Strain Differences for Three Hydraulic Pressures	119
Figure 5-22: E22 (Y Direction) Strain Results for Two Hydraulic Pressure Variations	121
Figure 5-23: FBG and FEA E22 (Y Direction) Strain Differences for Two Pressure Variations	122
Figure 5-24: E11 (X Direction) Strain Results for Two Hydraulic Pressure Variations	123
Figure 5-25: FBG and FEA E11 (X Direction) Strain Differences for Two Hydraulic Pressure Variations	124
Figure 6-1: Quasi Dynamic Test System Set-up	128
Figure 6-2: The Slipping Phenomenon during Quasi Dynamic Loading Tests	133
Figure 6-3: Recording Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface	135
Figure 6-4: Strain Distribution Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface	136
Figure 6-5: FBG Sensor Contour Maps for Vertical (Y) Direction Comparison between Three Test Stages with Equal Pressure ($p_1=p_2$)	139
Figure 6-6: FBG Sensor Contour Maps for Horizontal (X) Direction Comparison between Three Test Stages with Equal Pressure ($p_1=p_2$)	140

Figure 6-7: FBG Sensor Contour Maps for Vertical (Y) Direction Comparison between Three Pressure Variations Applied with 600Nm Torque Level..	142
Figure 6-8: FBG Sensor Contour Maps for Horizontal (X) Direction Comparison between Three Pressure Variations Applied with 600Nm Torque Level..	143
Figure 6-9: Additional Tests of Three Pressure and Torque Level Conditions	146
Figure 7-1: The Disc Model: (a) Static Analysis and (b) Quasi Dynamic Analysis	149
Figure 7-2: The Element Size: (a) Static Analysis and (b) Quasi Dynamic Analysis	150
Figure 7-3: The Assembly View: (a) Piston Side View and (b) Disc Side View	150
Figure 7-4: Boundary Conditions (BCs) of The Quasi-Dynamic FE Model (Pistons and Disc Settings)	151
Figure 7-5: Boundary Conditions (BCs) of The Quasi-Dynamic FE Model (Disc and Brake Pad Settings).....	152
Figure 7-6: Loads of The Quasi-Dynamic FE Model	153
Figure 7-7: FE Analysis Results for Different Element Sizes with 40bar (X Direction)	154
Figure 7-8: FE Analysis Results for Different Element Sizes with 40bar (Y Direction)	155
Figure 7-9: E11 (X Direction) FEA Results for Clockwise and Anticlockwise Moment Loads applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level.....	156
Figure 7-10: E22 (Y Direction) FEA Results for Clockwise and Anticlockwise Moment Loads applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level.....	157
Figure 7-11: E11 (X Direction) FEA Results for 0.4 and 0.5 Friction Coefficient applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level.....	158
Figure 7-12: E22 (Y Direction) FEA Results for 0.4 and 0.5 Friction Coefficient applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level.....	159
Figure 7-13: Three Design Concepts of Backplate Boundary Conditions.....	161
Figure 7-14: E11 (X Direction) FEA Results: Three Design Concepts applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level.....	162
Figure 7-15: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.46$	165

Figure 7-16: Processed FEA Contour Maps in Vertical (Y) Direction for Three Hydraulic Pressure and Torque Levels.....	166
Figure 7-17: Processed FEA Contour Maps in Horizontal (X) Direction for Three Hydraulic Pressure and Torque Levels.....	167
Figure 7-18: E22 (Y Direction) Processed FEA Strain Contour Map Comparisons for Three Hydraulic Pressure Variations; $\mu=0.46$	169
Figure 7-19: E11 (X Direction) Processed FEA Strain Contour Map Comparisons for Three Hydraulic Pressure Variations; $\mu=0.46$	170
Figure 7-20: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=40\text{bar}$	173
Figure 7-21: Processed E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=40\text{bar}$ with 200Nm Torque	174
Figure 7-22: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=80\text{bar}$	176
Figure 7-23: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=120\text{bar}$	177
Figure 7-24: E22 (Y Direction) Strain Differences for Static to Quasi-Dynamic differences of Two Hydraulic Pressure and Torque Levels; $\mu=0.46$	178
Figure 7-25: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure, $p_1=p_2=40\text{bar}$	180
Figure 7-26: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure, $p_1=p_2=80\text{bar}$	181
Figure 7-27: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure, $p_1=p_2=120\text{bar}$	182
Figure 7-28: E11 (X Direction) Strain Differences for Three Hydraulic Pressure and Torque Levels.....	183
Figure 7-29: E22 (Y Direction) Strain Results for +10bar Pressure Variation, 600Nm	185
Figure 7-30: E22 (Y Direction) Strain Results for -10bar Pressure Variation, 600Nm	186
Figure 7-31: FBG and FEA E22 (Y Direction) Strain Differences for Two Hydraulic Pressure Variations, 600Nm Torque Level	187
Figure 7-32: E11 (X Direction) Strain Results for +10bar Pressure Variation, 600Nm	188
Figure 7-33: E11 (X Direction) Strain Results for -10bar Pressure Variation, 600Nm	189

Figure 7-34: FBG and FEA E11 (X Direction) Strain Differences for Two Pressure Variations.....	191
Figure 8-1: New Modified X&Y Direction Brake Pad	197
Figure 8-2: The Dynamic Test System at The University of Bradford.....	199
Figure 8-3: Dynamic Test Schedule and Validation.....	200
Figure 8-4: “Dynamic Loading Test 1” Raw Recorded Data Graphs	202
Figure 8-5: Dynamic Loading Test 1 Processed Data Graphs	205
Figure 8-6: Dynamic Loading Test 1 Strain Distributions at the End of Brake Dynamic Loading Application (t=270s)	206
Figure 8-7: Dynamic Loading Test 1 Process from 240s to 255s	208
Figure 8-8: Dynamic Loading Test 1 Process from 255s to 270s	209
Figure 8-9: Strain Differences between Initial and Final Static Applications...	212
Figure 8-10: Comparisons of Dynamic Loading Tests 2 & 3 Processed Data Graphs.....	214
Figure 8-11: Tests 2 & 3 Strain Differences of Dynamic Loading Applications	216
Figure 8-12: Test 2 Strain Differences between Initial and Final Static Applications	218
Figure 8-13: Test 3 Strain Differences between Initial and Final Static Applications	219
Figure 8-14: Static Strain Differences: $\Delta Diff = \Delta Diff_{Test3} - \Delta Diff_{Test2}$	220
Figure 8-15: Test 2 & 3 Strain Differences between Initial or Final Static Applications	221
Figure 8-16: “Dynamic Cooling Test 4” Raw Recorded Data Graphs.....	224
Figure 8-17: Strain Differences between Initial and Final Static Applications .	226
Figure 8-18: Dynamic Consecutive Test 5 Raw Data Graphs	228
Figure 8-19: Dynamic Consecutive Test 5 Temperature Measurement	229
Figure 8-20: Dynamic Consecutive Test 5 Processed Data Graphs	230
Figure 8-21: Repeated Dynamic Loading Application Strain Results	232
Figure 8-22: The Brake Pad Wear Condition Differences	234
Figure 8-23: Different Views of The Brake Pad Wear Condition after Dynamic Tests.....	236

LIST OF TABLES

Table 3-1: The Part List of The Two-Channel Control Hydraulic Pressure System 46

Table 3-2: Pneumatic Pressure (P_p) and Hydraulic Pressure (P_H) Relationship52

Table 4-1: FBG Static Loading Test Plan 70

Table 5-1: The Specification of Brake Components (Vianello, 2015) 94

Table 6-1: Load Weight and Test Component Values 129

Table 6-2: Quasi Dynamic Loading Test Plan 130

Table 6-3: Applied Torque Level Combination 130

Table 6-4: Additional Test Plan 131

LIST OF EQUATIONS

(2-1)..... 19

(2-2)..... 19

(3-1)..... 54

(4-1)..... 73

(8-1)..... 203

(8-2)..... 203

LIST OF ABBREVIATIONS

ABS	Anti-Lock Braking System
BC(s)	Boundary Condition(s)
BPV	Brake Pressure Variation
BTV	Brake Torque Variation
CFBG	Chirped Fibre Bragg Grating
CNC	Computer Numerical Control
CoP	Centre of Pressure
DFOSS	Direct Fibre Optic Shape Sensing
DIC	Digital Image Correlation
DTV	Disc Thickness Variation
ECU	Electronic Control Unit
EDF	Erbium Duped Fibre
EDX	Electron Dispersive X-Ray
EHA	Electro-Hydrostatic Actuators
EMA	Electro-Mechanical Actuators
FBG	Fibre Bragg Grating
FC	Fibre Channel
FE	Finite Element
FEA	Finite Element Analysis
FMCW	Frequency Modulated Continuous Wave
FSI	Fibre Segment Interferometry
HC	High Carbon
ID	Inboard / Inner Diameter
IPD	Interface Pressure Distribution
IR	Infrared Radiation
LW	Low Pressure
MHS	Macroscopic Hot Spots
MOST	Media Oriented Systems Transport
NVH	Noise, Vibration and Harshness
OD	Outboard / Outer Diameter
OSA	Optical Spectrum Analyser
PCD	Pitch Circle Diameter

POF	Plastic Optical Fibre
PSP	Pressure Sensitive Papers
RNN	Recurrent Neural Network
SDM	Spatial Division Multiplexing
SHM	Structural Health Monitoring
TDM	Time Division Multiplexing
TEI	Thermoelastic Instability
TFBG	Tilted Fibre Bragg Grating
UV	Ultraviolet
WDM	Wavelength Division Multiplexing
2D	Two Dimensional/Two Dimensions
3D	Three Dimensional/Three Dimensions

NOMENCLATURE

Symbol	Description	Unit
λ_{Bragg}	Bragg Wavelength	nm
n	Refraction Index (Chapter 2)	
n	Rotational Speed (Chapter 9)	rpm
Λ	Index Period of FBG Refraction Variation	
D	Diameter	mm
ϵ	Strain	$\mu\epsilon$
t	Time	s
p	Pressure	bar
T	Temperature	$^{\circ}\text{C}$

1 INTRODUCTION

1.1 Background

Braking systems are of utmost importance for the safety of any vehicle. Therefore, a number of research projects have investigated the vehicle braking system, in order to make them safer in terms of both performance and reliability. Furthermore, braking systems play an important role in drivers and passengers' overall comfort. Undesired effects, such as brake noise, can affect surroundings, for example passers-by as well as people leaving in the proximity of the streets and roads. Last but not least, brake dust is a serious health hazard, apart from excluding asbestos from friction material formulations, not much further has been done to reduce or eliminate these negative effects.

Brake research areas are therefore numerous and the focus can be different. The research presented in this thesis addresses brake performance, reliability and Noise Vibration and Harshness (NVH) aspects concerning to the brake friction interface, in particular, interface pressure distributions (IPD) and more specifically the strain distribution on pad friction surface. This is investigated under a range of brake operating conditions, including static loading, quasi-dynamic and dynamic applications. In these early stages, it is important to gain a good fundamental understanding of strain distributions and influencing factors, which will provide the basis to improve the brake design and materials to achieve higher and more stable friction functions. The research project also intends to make several innovative research applications in terms of both scientific and industrial contributions in this regard.

Experimental investigations of friction brake strains, temperature and deflection are notoriously difficult because of high forces, temperatures, complex contact interactions, the rotating motion and considerable component wear. Furthermore, the determination of the brake operation's fundamentals, namely the interface pressure distributions, friction forces, component deflections and their interactions, are particularly challenging due to the limited space in which the sensors are incorporated. It is also of great importance that the installed sensors

should not alter component characteristics and interface properties, for example by adding mass or disrupting frictional surfaces. Consequently, optical methods and techniques offer a great potential in achieving these goals.

For instance, Laser Holographic Interferometry, can provide insight into the dynamic component behaviour, however, due to the unavailability of the optical access to the friction interface between pad and disc, only visible surfaces can be monitored, which has been tested by Talbot and Fieldhouse (2002) to determine the in-plane vibration modes.

Similarly, Digital Image Correlation (DIC), as presented by Sergent (2010), can provide useful information on component deflections and strains, however, the technique application is also limited to the visible surfaces.

The Centre of Engineering Photonics at Cranfield University has developed a range of optical sensors for strain and shape monitoring, which have been successfully applied in aeronautical and other challenging environments. It has also demonstrated that optical sensing technology, in particular the Fibre Bragg Grating (FBG) sensors, can be effectively used in disc brakes, for measuring friction material strains (Vianello, 2015). The results were encouraging but much further research is needed in order to gain a deeper understanding of a range of operating conditions. In particular, the results need to be combined with corresponding Finite Element Analysis (FEA) to assist in improving the understanding of the component loading and behaviour during brake operations.

The research gap identified in this research project is the development of the means to measure the brake pad or disc interface strain distributions in a reliable and effective way. Until now, there is no publication of any work in this area, except for the aforementioned Tirovic et al. (2019), which provided encouraging results but with many limitations. In this research, the experimental results will be obtained by using FBG strain sensors mounted on a modified brake pad in a specifically designed test rig, then compared with the FEA simulations to gain deeper insights into interface conditions during static and quasi-dynamic loading. Finally, FBG sensors will be used in actual, dynamic brake applications, in another specifically designed brake rig.

1.2 Aims and Objectives

There are two overall scientific aims of the project. One is to experimentally determine interface strain distributions at the brake pad under static and quasi-dynamic loading conditions, and to apply the finite element (FE) model analysis to support and complement the test results. The other is to measure interface strain distributions and the brake performances under dynamic loading conditions.

In order to achieve these aims, a special brake test rig will be designed, commissioned and used for the static and quasi-dynamic brake loading. A separate rig will be employed for the dynamic brake testing.

The research objectives can therefore be defined as following:

- To develop and validate a Finite Element (FE) model to simulate various test conditions and brake performances, including interface pressure distributions, strain distributions and friction material properties.
- Develop Fibre Bragg Grating (FBG) strain sensors suitable for brake strain measurements.
- Develop techniques for FBG sensor installations into brake pads, minimising their effect on pad and brake behaviour and maximising the ability of obtaining accurate, repeatable and reliable strain results at pad friction interface. This is to be achieved under static, quasi-dynamic and dynamic test conditions.
- Compare and validate test results obtained using FBG sensors with the outputs from the FEA simulations in static and quasi-dynamic conditions.

1.3 Thesis Structure

The thesis is divided into 9 chapters, structured in a most suitable way to present the research in a clear form, with sufficient level of detail and most appropriate interrelationships between various Thesis parts and research methods. Following this introductory Chapter, which gives a brief Introduction and defines the Aims and Objectives, Chapter 2 will focus on the literature review, including the literature on the research into brake systems, in particular interface pressure distributions, as well as the optical measurement technology. The experimental

facilities will be introduced in Chapter 3. Chapters 4 and 5 will analyse the static loading conditions from both the experimental and the finite element analysis (FEA) perspectives, whereas Chapters 6 and 7 concern the analysis of the response of the brake pad to the quasi-dynamic conditions. The dynamic analysis will be described in Chapter 8, and Chapter 9 includes discussion, conclusion, and potential future work.

2 LITERATURE REVIEW

2.1 Introduction

Noise, vibration and interface pressure distribution (IPD) of vehicle braking systems have been the subject of numerous investigations. The following chapter will review recent publications in these areas in order to understand the phenomena under investigation and the test approaches that have been employed. As the research presented within this thesis is focused on the application of optical sensors, in particular Fibre Bragg Grating (FBG) sensors, the chapter will also include an introduction to the optical sensing technologies used and a review of previously reported applications of the optical techniques.

2.2 Recent Brake Publications

The braking system of a modern vehicle represents the merging of advanced technologies, including material science, mechanical engineering and control systems, to ensure vehicle safety during driving activities. In recent years, substantial investigations have been undertaken to understand and improve braking systems. This section will mainly focus on studies concerning brake noise and vibrations.

2.2.1 Brake Noise

Noise from vehicle braking systems is one of the major issues plaguing both manufacturers and customers. Brake noise is caused by the airborne transmission of acoustic waves arising from structural vibrations that are excited during driving actions (Bryant, 2021). There were several classes of noise that have been defined by North (1976), Lang and Smales (1983) and other researchers, including groan noise (<100 Hz), low frequency squeal (2 – 4 kHz), high frequency squeak (4 – 15 kHz), as well as wire brush (up to 20 kHz). Even low frequency noise will attract attention from the users and can become a reason for the owner to seek repairs of the vehicle under warranty. The eradication of brake noise means significant cost and time to automotive manufacturers. This is the case within both the USA and the Europe, where customers have high

expectations of vehicle performance and requirement (Chen et al., 2006; Day, 2014).

The history of brake noise research starts with Mills (1938), who reported the earliest systematic determination of drum brake noise and noted some characteristics, including the 'trailing action' in the brake linings from the friction forces and the friction material effects, both of which become noise source parameters. The use of mass damping and friction damping equipment was proposed to control noise. Meanwhile, North (1969) also became a pioneer in the brake noise investigation and modelling fields, creating an 8 degree of freedom mass/spring/damper dynamic model with the lumped rigid body of a disc brake. He might be the first person to have correlated theoretical predictions against data obtained from a real squealing brake system. (Kinkaid et al., 2003; Day, 2014).

Over the last 10 to 20 years, brake squeal has remained an important area of brake analysis and research. Belhocine and Ghazaly (2015) developed a finite element model to improve the understanding of the effect of material properties, especially the Young's modulus. A model of the whole disc brake assembly was created, and the Young's modulus of each part was altered to explore its influence on the squeal noise. The effect of changes in the Young's modulus of the friction material is presented in Figure 2-1, which indicates that the brake system would be more stable and have reduced the squeal noise when the friction material has a higher value of the Young's modulus.

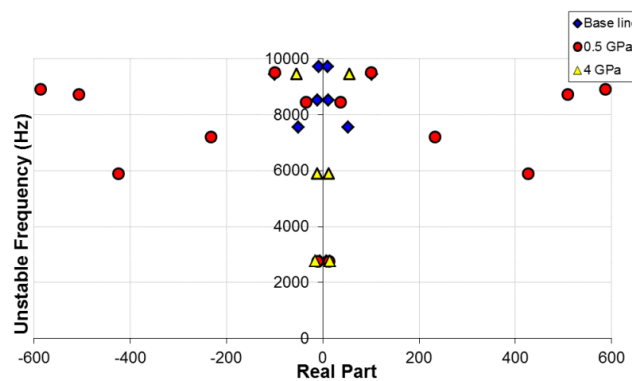


Figure 2-1: Effect of the Young's modulus of the friction material on the brake squeal generation (Belhocine and Ghazaly, 2015)

The Belhocine and Ghazaly study agreed with Papinniemi's (2007) conclusion that the squeal phenomena could be reduced and the whole brake system would be more stable when the Young's modulus' value of the brake pad friction material was increased. The rationale behind this conclusion is that the brake pad displacement will be reduced and hence so would the related excitation and physical effects. The research completed by Liu (2022) presented a new method via the MSC/Nastran solution 200 to simulate and analyse friction material properties on a higher level, in particular natural frequencies and mode shapes.

Another stability finite element analysis of multiple contact conditions was reported by Denimal et al. (2017), where they tried to simulate the interface of the frictional surface, and three potential contact interactions, between the piston and pad, between the bracket and pad, as well as between the calliper and pad, which could influence the brake stability behaviour. This attempt concluded that one specific condition would be difficult to predict. All factors needed to be considered for different configurations to reduce brake squeal noise. The predicted unstable modes of frequencies with a high dispersion can be observed in Figure 2-2.

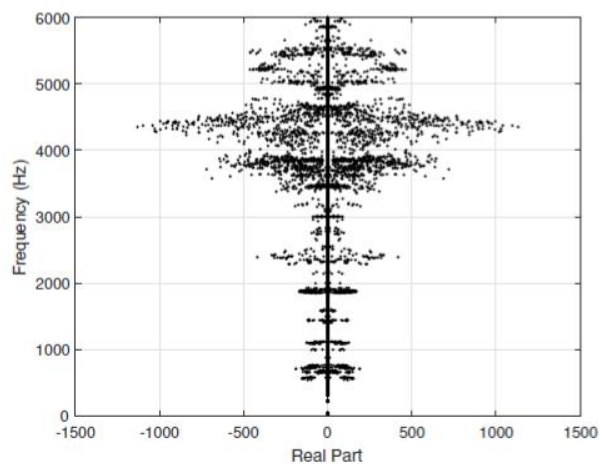


Figure 2-2: Complex eigenvalues for all configurations considered (Denimal et al., 2017)

Tirovic, Vianello and Bannister (2019) provided an improved method to predict squeal noise propensity via a complex eigenvalue analysis. There were 4 stages of the FE analysis, including free-free modal analysis, mechanical FEA of the

brake system assembly, thermo-mechanical simulations, and stability analyses for the brake assembly. According to this methodology, the validation of actual brake system can be assembled with improved NVH characteristics, while reducing design costs and prototype manufacture requirements are still feasible. The approach predicated that brake squeal noise can be generated as a result of the generation of thermal loads during a braking event. It is influenced by several parameters, including equipment geometries, friction coefficient and wear pattern at the interface of disc and pad. Meanwhile, an interesting approach has been attempted by Stender et al. (2021), who recorded noise via convolutional neural networks (CNN), and determined the brake system dynamic stability by using a recurrent neural network (RNN). This approach could detect the brake noise structural generation and provided a high accuracy in predicting the onset of the brake squeal noise.

Groan noise is a semi-resonant effect that occurs at lower speeds and frequencies, usually caused by the stick-slip response of the brake system (Day, 2014; Bryant, 2021). Lazzari et al. (2018) presented experimental results to evaluate the groan noise of the brake pad material (Figure 2-3). The frictional response and the stick-slip generation were measured with several lining materials. It allowed comparison of the tested materials with the stick-slip phenomena, and achieved positive feedback on the groan noise.

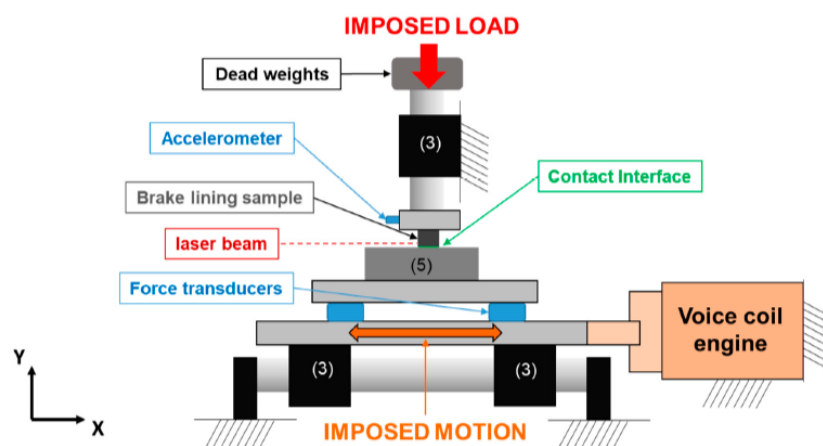


Figure 2-3: Experimental Set-Up Scheme (3) Air Bushings, (5) Test Sample (Lazzari et al., 2018)

Wire brush or 'roughness noise' is a non-resonant noise at up to 20kHz and happens prior to a true unstable squeak when heard immediately (Day, 2014; Bryant, 2021). Lindberg et al. (2013) created a new test method and tried to measure the influence of vehicle speed and brake pressure on the wire brush noise within a laboratory where the test conditions could be accurately controlled (Figure 2-4). A vibro-acoustic structure-borne phenomenon was defined and test results showed a strong relationship of increased interior roughness noise with increasing vehicle speed and brake pressure.

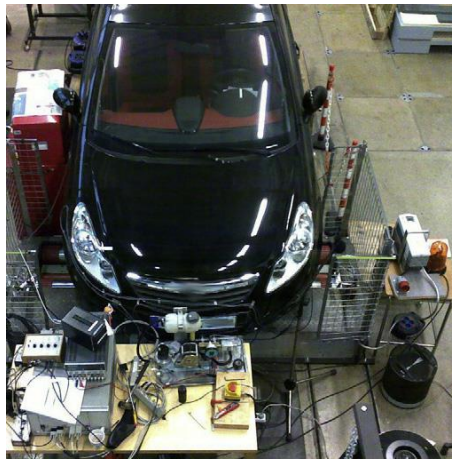


Figure 2-4: The Full Vehicle Experimental Rig (Lindberg et al., 2013)

2.2.2 Brake Vibrations

Brake vibrations are caused by variable forces and displacements, and oscillate at a fixed reference position at various frequencies that are proportional to vehicle speed (Bryant, 2021). Brake judder and stick-slip phenomenon have been the focus of research in recent years.

Brake judder is a low frequency vibration, under 100Hz, and usually manifests at several positions, including brake pedal, steering wheel, floor pan and seat rail. Such a vibration is either a result of brake torque variation (BTV) or a result of brake pressure variation (BPV). Cold and hot (thermal) judder are two types of brake judder, where the cold judder typically happens between 5 Hz to 50 Hz, while hot judder (100Hz – 250Hz) is a thermal issue generated in large, heavy-duty brake systems.

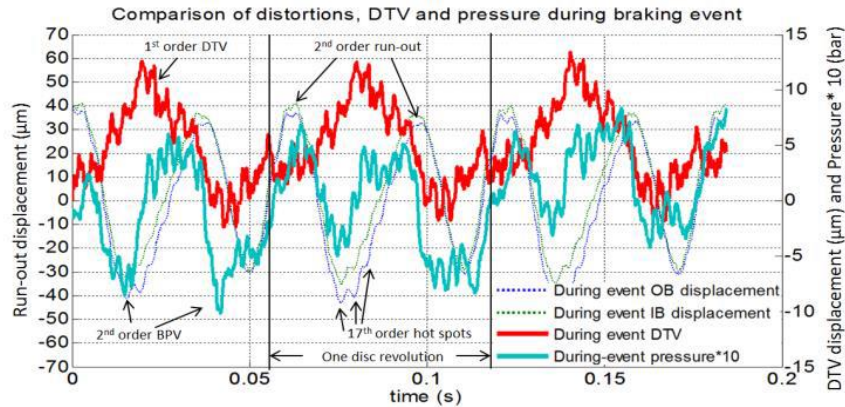


Figure 2-5: Comparisons of Distortions, DTV and Pressure during Braking Event
(Bryant, 2021)

Disc thickness variation (DTV) is a factor that causes cold judder, and arises from poor disc manufacturing processes or from the uneven 'off-brake wear' during the use of brake (Day, 2014). The generation of cold judder has been identified for DTV between 10 - 15 μm , which is considered to be the critical threshold of DTV (De Vires and Wagner, 1992; Jacobsson, 2003). Although increasing DTV and piston seal roll-back can help to reduce the judder, the pad/disc clearance and brake actuation response would also need to be increased as an expense of the changes (Barton and Haigh, 1998). Dynamic run-out and DTV have been tested by Bryant (2021), who found that the BPV arose from both run-out and DTV (Figure 2-5).

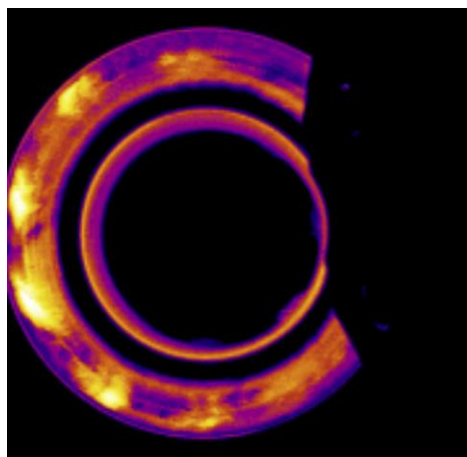


Figure 2-6: Hot Spots on a Brake Disc Rubbing Surface (Bryant, 2010)

Hot judder can be associated with all vehicles, but in particular with high performance cars, and commercial vehicles. The issue highlighted by Figure 2 6 is 'hot spotting' or 'blue spotting', which occurs when the brakes are not properly broken in (bedded). The brake rotors have uneven brake pad deposits which accumulate and finally turn into a compound called cementite. This generally occurs at 700°C. This causes hot judder and is associated with the cast-iron material phase change of the rotor frictional surface due to high working temperatures (Bryant, 2010). An infrared radiation (IR) camera was applied to measure the hot spot displacement on a railway brake disc system by Panier et al. (2004), who found that the energy could go into the brake pads via the macroscopic hot spots (MHS) formation system (Figure 2-7).

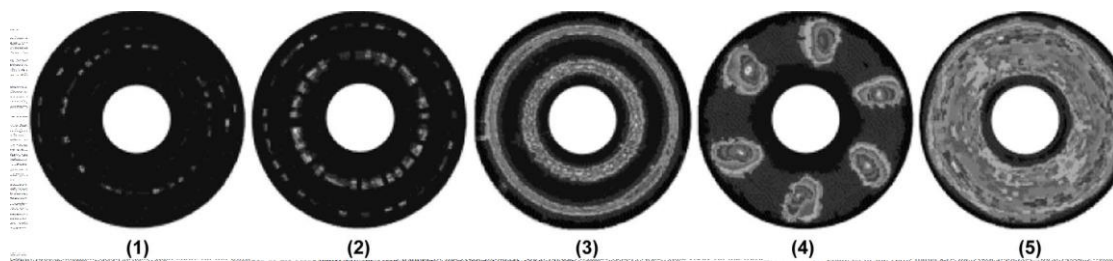


Figure 2-7: Hot Spot Formation Types (Panier et al., 2004)

Section 2.2.1 noted that groan noise happens at lower vehicle speeds, has a frequency lower than 100Hz, and is usually caused by stick-slip vibrations. Some researchers have investigated such vibrations under different conditions. A stick-slip friction model was created by Paliwal et al. (2005) to investigate brake noise and vibration. Parameters, such as friction material damping, mass ratios and coupling stiffness, influenced the brake assembly behaviour, especially the pad friction layer stiffness. The stick-slip motion could stabilize or destabilize the system under certain braking, contact pressure and environment conditions, which could affect the variation value and become the influence factor. Ashraf et al. (2017) considered the use of several experimental methods to understand brake pad and calliper characteristics when applying under different brake conditions. They created a test system that connected a commercial vehicle brake system with the trailer suspension system by using a beam axle. Optical displacement sensors were applied to determine the displacement of both the

calliper body and the pads during braking events. They concluded that the brake calliper deformation and the brake pad friction material predominantly influenced the stick-slip vibration phenomenon. They also pointed out that a 'low compression' set of brake pads could help to reduce such a phenomenon.

Shi et al. (2018) applied dry and wet working conditions to analyse brake pad/disc interface vibration phenomena. A result was observed when applying the brake under wet conditions, using the time-frequency vibration signal analysis method to determine the friction coefficient efficiency. Meanwhile, Otto and Ostermeyer (2019) focused on the high-frequency vibrations of the disc brake contact components, and highly specialized tribometers have been improved during experiments, where the friction boundary layer generated ultrasonic frequencies. They considered that installed parking sensors were influenced or even failed to work properly due to exposure to the same range of frequencies.

2.3 Interface Pressure Distributions

Brake pressure distributions at the friction interface between the disc and pad friction material, are often referred to as the 'interface pressure distributions' or IPD (Day, 2014). The non-uniformity and variations of IPD are influenced by several factors predominately, friction material properties, thickness and shape of the pad friction material and backplate surface, the means of the application of the actuation force, such as the piston diameter, and the piston number. Furthermore, the piston position in the brake calliper, as well as the calliper stiffness, and the disc coupling during braking also change the IPD considerably.

IPD was studied experimentally and theoretically using the available tools and software. The theory of beams on elastic foundation was initially used to theoretically assess IPD in static loading conditions. Although the approach was limited to 2D analysis, the results were helpful in assessing the suitable piston diameter, the pad length, the backplate thickness and most in particular, the friction material adopting Young's modulus and its compressibility. However, only with the development of numerical methods, could real advances be made in simulating and analysing IPD, in static, quasi-dynamic and dynamic conditions. Francavilla and Zienkiewicz (1975), who were the pioneers in this research field,

first investigated contact problems using numerical computation. They created a simple procedure to obtain the flexibility matrices at contact points for pressure distributions. Their novel method was able to solve these issues on a quite inexpensive level.

Drum brakes are sensitive to the changes in IPD, in particular the position of centre of pressure (CoP) along the brake shoe, as this will cause considerable changes in brake torque. Consequently, much of the early work was related to this brake type (Day, 2014). Disc brakes are much less sensitive to this phenomenon from the torque level point of view, as circumferential movement of CoP has no influence on performance, and radial change is limited by a relative narrow friction path from inboard (ID) to outboard (OD). Nevertheless, the change of braking torque can still be unacceptably high, in particular, if it is the result of other undesirable influencing factors, such as the reduction of friction coefficient, the disc coupling, and the flexibility of the calliper.

A single resultant normal force can be used to represent the pressure distributions as the CoP when the force crossed over the interface of the pad friction material (Day et al., 1979). Day et al. (1991) reported test results and FE analyses on a drum brake system, which presented the shoe and thermal expansion effects on the distributions. This could also be caused by the geometric conformity on the brake shoe between the drum and lining. Ito and Nakata (1990) have also analysed the dynamic behaviour of the IPD for a track when braking during a turning motion.

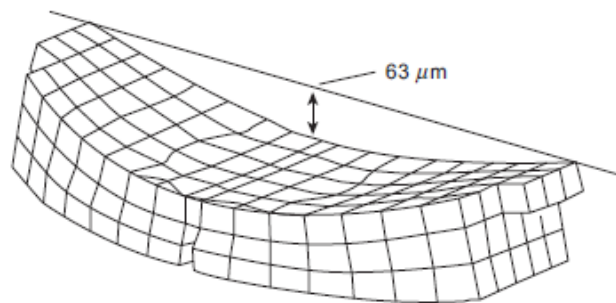


Figure 2-8: Disc Brake Pad Showing Deformation under Actuation Force (Tirovic and Day, 1991)

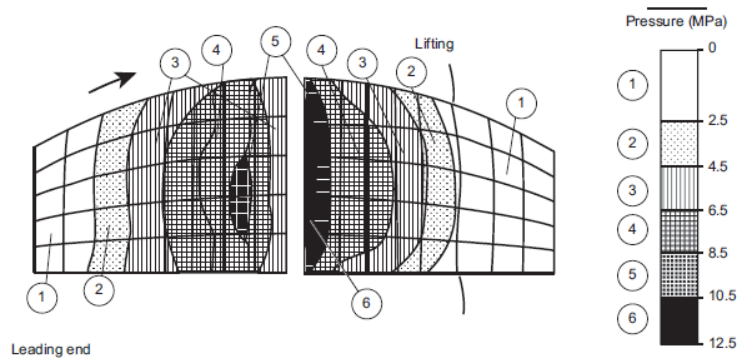


Figure 2-9: Disc Brake Pad Friction Interface Pressure Distribution Predicted by FE Analysis (Tirovic and Day, 1991)

A study completed by Tirovic and Day (1991) revealed the initial contact between the brake pad and the brake disc, using three-dimensional finite element analysis under static and dynamic braking conditions. Using Figure 2-8, they tried to determine the brake pad compression and deformation. Both compression and IPD were observed to have a uniform level for a range of brake pad characteristics. In addition, they successfully predicted the IPD at the brake pad friction surface via finite element analysis (FEA), which is illustrated in Figure 2-9. It could be observed that the highest pressure was located in the centre of the brake pad, and a weak pressure is predicted at the edges of the pad, where there might be zero pressure due to the loss of contact with the brake disc when applying the brake functions (Tirovic and Day, 1991; Day, 2014).

Meanwhile, a thermal stress analysis of a railway vehicle disc brake system has been performed by Kim et al. (2008), who created a 3D model for both variable and uniform pressure application on the contact area. They found that the locations where the highest thermal stresses happened with different pressure distributions are similar to the places where the fatigue cracks appeared on the actual brake assembly. Söderberg and Andersson (2009) used a general FEA software, ANSYS, to simulate the wear and pressure distribution at the contact interface between the pad and the disc. They focused the simulation on steady-state drag conditions, which were based on the Archard's wear law and the Euler integration.

The CoP was calculated in an improved analytical 3D model by Budinsky et al. (2018) to make the CoP level more reliable and accurate by determining the CoP position in both the radial and circumferential directions and providing the clamp and friction force on a more realistic level. They found that the new model provided about 8% higher friction coefficient value than that predicted by the traditional 2D model or by a complex numerical model. Fieldhouse and Steel (2003) paid more attention to the CoP offset in the interface area where the friction coefficient and the calliper mounting geometry would become the critical influencing factors.

According to the previous theoretical analysis, a number of experimental methods have also been used, such as pressure sensitive papers (PSP) and pressure mapping pads (e.g. Tekscan). PSP has the advantages of being low cost and easy to use without the requirement for any specific equipment. Tirovic et al. (2019) applied the method via the pre-scale low pressure (LW) paper, which could measure over a wide range between 25bar to 100bar, enough to investigate the brake interface pressure (Figure 2-10).

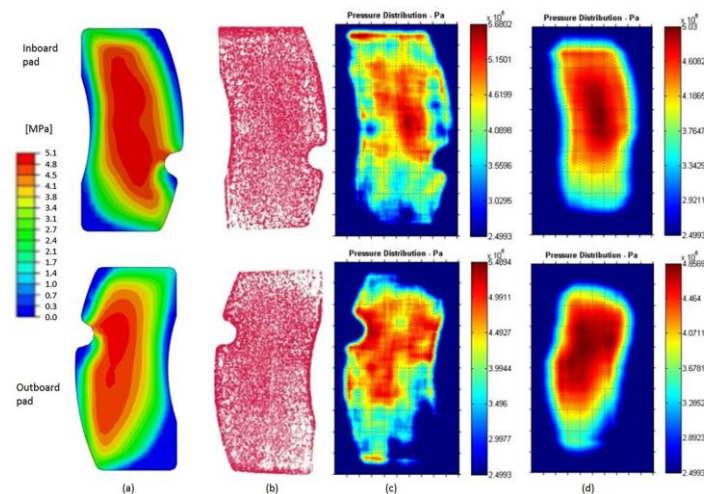


Figure 2-10: Pad–disc static interface pressures for 100 bar hydraulic actuating pressure: (a) FEA results, (b) PSP imprints and (c) and (d) processed results with increased level of filtering (Tirovic et al., 2019)

According to their experiments, the IPD could be directly observed on the paper itself. An understanding of IPD was obtained when the results were compared with FEA to establish if the PSP method is suitable for the measurement of such

brake system characteristics. However, the PSP technique has the limitation that each test paper could only be used for one pressure level under the static conditions, and the test environment should be maintained at the ambient temperature and humidity to ensure that the papers would not be damaged or invalid. An ideal IPD measurement system would be capable of providing a continuous measurement of IPD under a wide range of pressure levels and environment conditions, and operate under static, quasi-dynamic and dynamic conditions. A brief review in section 2.4.2 will explain the optical fibre sensor technology in the automotive research and industry fields.

In addition to direct measuring of interface pressure distributions, investigations of the real contact area have been attempted. Limmer et al. (2020) tried to use different methods to characterise brake pad surfaces, in particular 'contact plateaus'. They applied two measuring methods to capture the brake pad contact plateaus; image segmentation via a light microscope, and the EDX (Electron Dispersive X-Ray) method by using a focused ion beam. Both methods can provide a high contrast image where the contact plateaus are displayed in white, and non-contact areas in black. The image segmentation allows a fast and automated scanning of surfaces in large areas for providing more global properties. The EDX method captures the same area but can also obtain information on the chemical composition, so that this can fully display the contact area and allow the tribological processes during braking activities to be analysed. Due to these experiments, a more targeted friction and wear results, as well as the new friction materials' development can be undertaken to satisfy automotive industry's future requirements.

2.4 Optical Measurement Technology

Fibre optics were first developed and researched for use in telecommunications in 1970s. Then, the first optical fibre amplifier appeared in 1990, which allowed the extension of the distance over which the data signal could be transferred, and wavelength division multiplexing was used to improve the efficiency and capacity of the data transmission between optical fibre network paths (Kokaje et al., 2015). Due to these developments, the optical fibre technology is widely deployed in

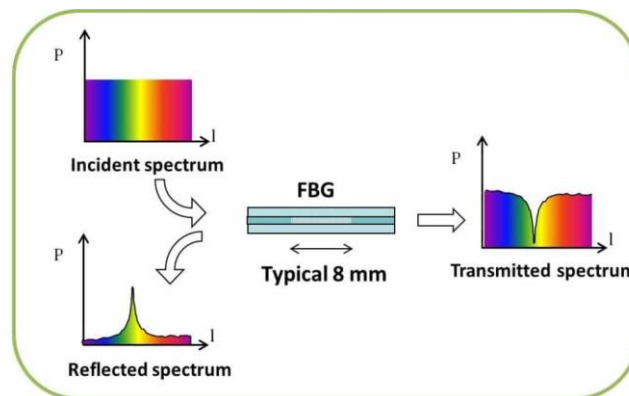
telecommunications. It is also known that the properties of light propagating in the optical fibre are influenced by the external environment, and this sensitivity has been exploited in the development of the new sensing techniques, which have benefitted from the ready availability of low-cost components, laser sources, detectors, filters, directional couplers and modulators developed for the telecommunications industry. This section will mainly focus on the recent development of optical fibre sensing technology, in particular the Fibre Bragg Grating (FBG), which is now a mature sensing technology that is widely commercially available, and is used in the work presented in this thesis.

2.4.1 Optical Fibre Sensing Applications

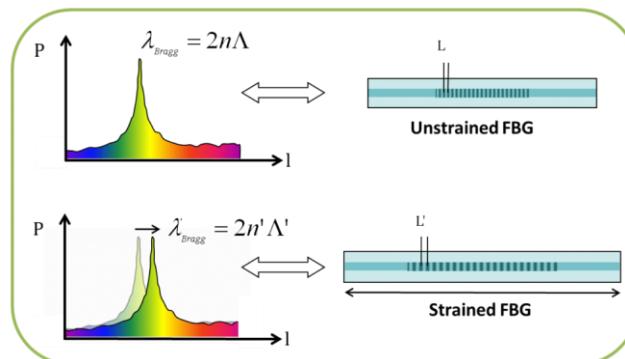
Optical fibre sensors have gained significant interest in recent years due to the small diameter of the fibre, its flexibility and suitability for the use in difficult-to-reach areas, and the ability to multiplex sensors within a single optical fibre. The optical fibre sensors and interrogation instrumentation are now commercially available and widely deployed across a range of industries, including oil and gas, aerospace, civil engineering and wind power (Dakin and Culshaw, 1997). There are many types of optical sensors used to measure different parameters in various research fields, such as point and distributed sensors, as well as extrinsic and intrinsic sensors. The point sensor performs a measurement at a single, specific location along an optical fibre, where the measurand is transduced into a change in phase, attenuation, polarisation or wavelength of the propagating light, however, the distributed sensors are operated alongside with the optical array or various point distributions. Fibre Bragg gratings (FBGs) are a specific example, where perturbations of the fibre that cause a change in the length and/or refractive index at the location of the FBG (typical gauge length 5 mm), induce a change in the wavelength reflected from the grating. Point sensors can also be multiplexed with a single optical fibre. For example, FBG sensors can be readily multiplexed in the wavelength domain by ensuring that each reflects a distinct wavelength under quiescent conditions. The difference between extrinsic and intrinsic sensors is where measurand influences the properties of the light. The extrinsic type has the light beam emitting from the optical fibre and changing

before the beam is directed back down onto the path, while the intrinsic type has the light beam maintained and changed inside the fibre (RF Wireless World, 2022).

A Fibre Bragg Grating (FBG) sensor is an intrinsic optical fibre sensor, which is fabricated by exposing single-mode fibre core to intense laser light with a periodic pattern. The exposure makes a permanent increase in the fibre core's refractive index, and establishes a fixed index modulation, which is called the 'grating'. The working principle is shown in Figure 2-11 (a), where each grating plane reflects a small portion of the propagating mode back along the fibre. The light reflected from each grating plane interferes constructively with an incident wavelength that is half of the grating period, termed the 'Bragg Wavelength', at which a reflected signal is observed. All signals not matching the Bragg condition (Equation (2-1)), experience destructive interference and pass through the FBG.



(a) Working Principle of FBG



(b) FBG Response as Function of Strain

Figure 2-11: Diagram of FBG Principle and Strain Response (FBGS, 2022)

According to Figure 2-11 (b), the relationship of the reflected central wavelength and the Bragg wavelength is listed below:

$$\lambda_{Bragg} = 2n\Lambda \quad (2-1)$$

where n is refraction index, and Λ is index period of FBG refraction variation.

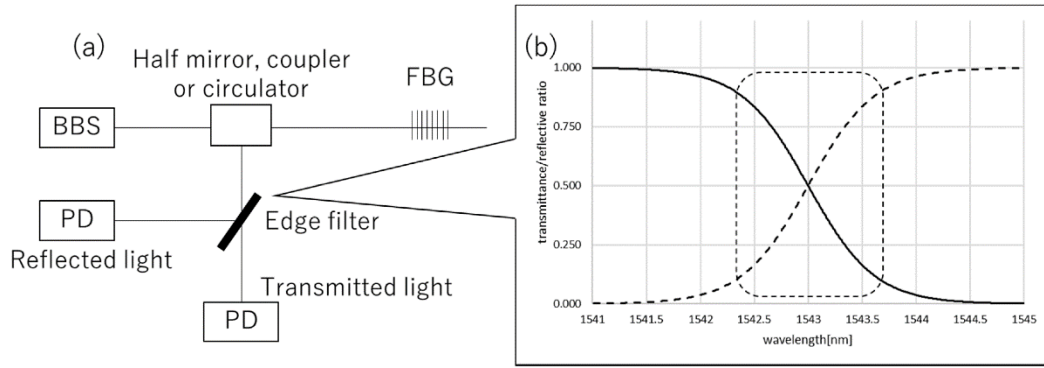
According to the principle of FBG sensor operation is based on the wavelength shift, an external parameter “X”, called the measurand quantity, has been considered to understand the sensing mechanism of FBG, which may be caused by temperature, pressure, strain or refractive index. The Bragg wavelength is influenced by such an external parameter and differentiates the Equation (2-1) with this measurand parameter “X”, which is listed below:

$$\frac{d\lambda_{Bragg}}{dX} = 2 \frac{d(n\Lambda)}{dX} = 2\Lambda \frac{dn}{dX} + 2n \frac{d\Lambda}{dX} \quad (2-2)$$

As the grating period and the effective refractive index of FBG are affected the Bragg wavelength, so that the parameter “X” will change either the grating period or the effective refractive index or depending on the measurement types (Kashyap, 2010).

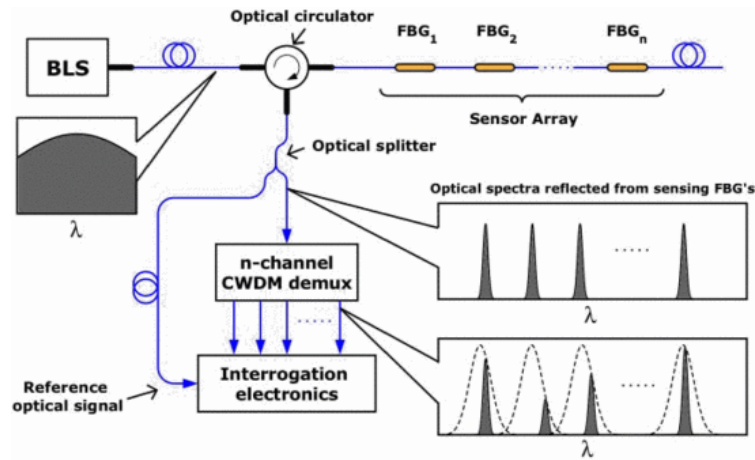
The recovery of the parameter signal encoded in the Bragg wavelength shift is the main challenge of FBG-based sensing approaches. Under laboratory conditions, the most often reported means for monitoring the Bragg wavelength involves the use of an optical spectrum analyser (OSA), with a broad band optical source employed to address the sensors. The OSA offers a slow response speed and low resolution, which cannot meet the demands of the accurate determination of the small and dynamic variations of the Bragg wavelength (Sahota et al., 2020). In addition, OSAs are bulky and sensitive optical instruments that are not suited to real world measurement environments. Therefore, several other interrogation methods have been developed to provide high data rates (kHz) and improved resolution to offer dynamic sensing capabilities with portable, rugged instrumentation. Passive and active interrogations are the two main approaches to be used. The passive interrogation method involves the use of a broad band optical source in conjunction with

passive devices such as edge filters (Ogawa et al., 2019) and coarse wavelength division multiplexing (CWDM) filters (Silveira et al., 2018), that are Bragg wavelength dependent optical filters (Figure 2-12). Here, the Bragg wavelength shift is transduced into a change intensity of the light transmitted by the filter.



(a) Typical setup of the edge filter-based FBG interrogation; (b) Conceptual diagram of the wavelength dependence of the transmission/reflection ratio of the edge filter.

(Ogawa et al., 2019)

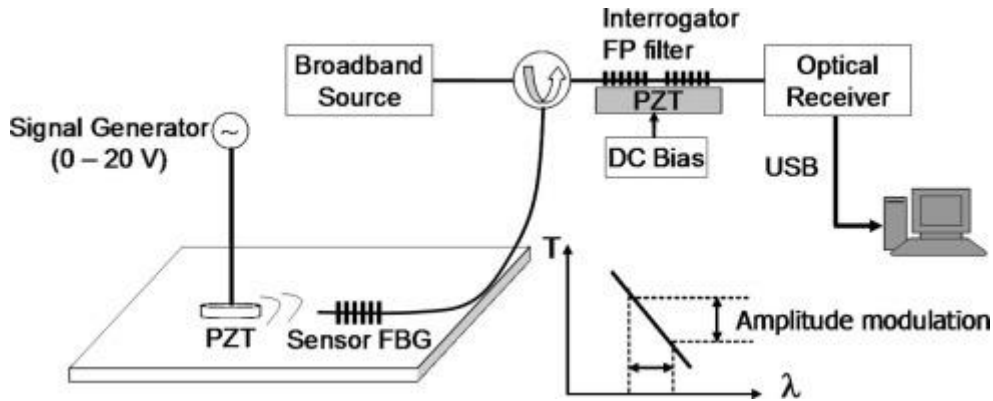


(c) Interrogation scheme for an array of FBG sensors based on their reflected optical power modulated by an n-channel CWDM demultiplexer. (Silveira et al., 2018)

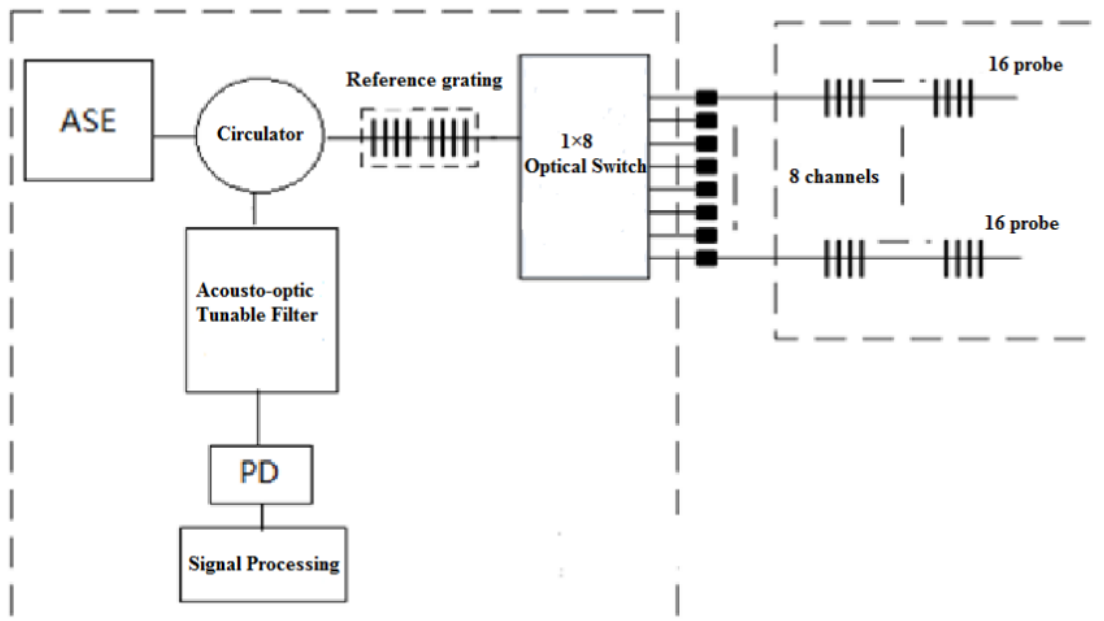
Figure 2-12: Passive Interrogation Methods

Active interrogation methods employ scanned wavelength filters, such as Fabry Perot (Harish et al., 2015) and acousto-optic tuneable filters (Li et al., 2014), in combination with a broadband optical source (Figure 2-13). Systems employing complex optical sources, such as wavelength tuneable lasers, Fourier-domain

mode-locked (FDML) laser fibre (Park et al., 2017) and dual optical frequency combs (DOFC) (Posada-Roman et al., 2016) have also been investigated to measure the Bragg wavelength shift (Figure 2-14).

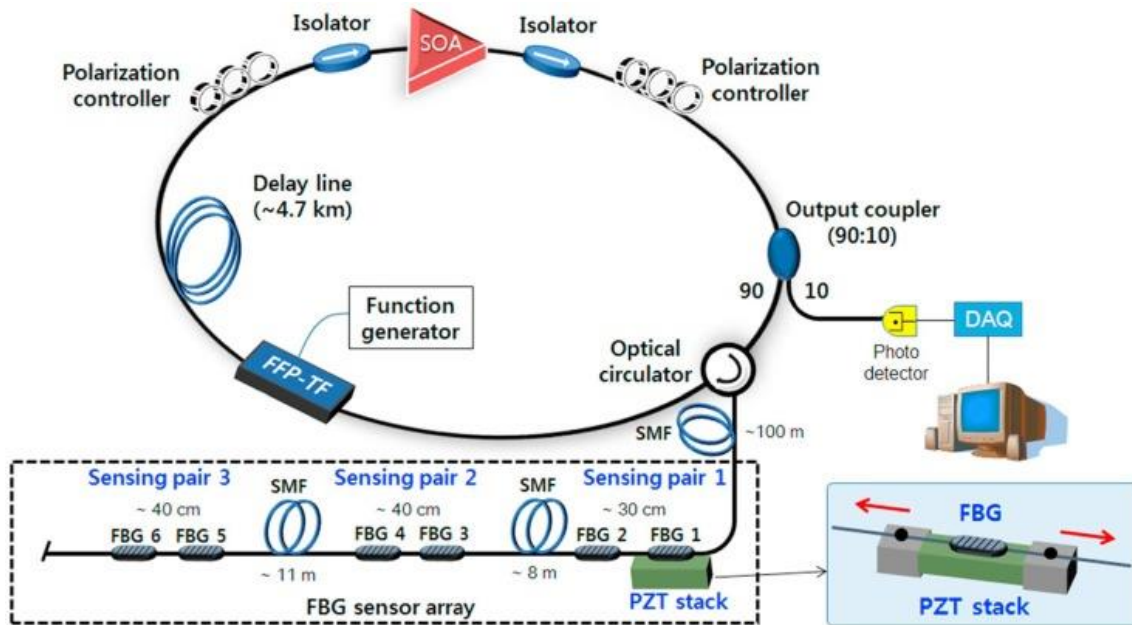


(a) Schematic diagram of a matched FBG-based sensor/interrogator system. The test bed consists of a FBG sensor glued to an aluminium plate, in which elastic waves are generated using a piezo transducer element (PZT). (Harish et al., 2015)

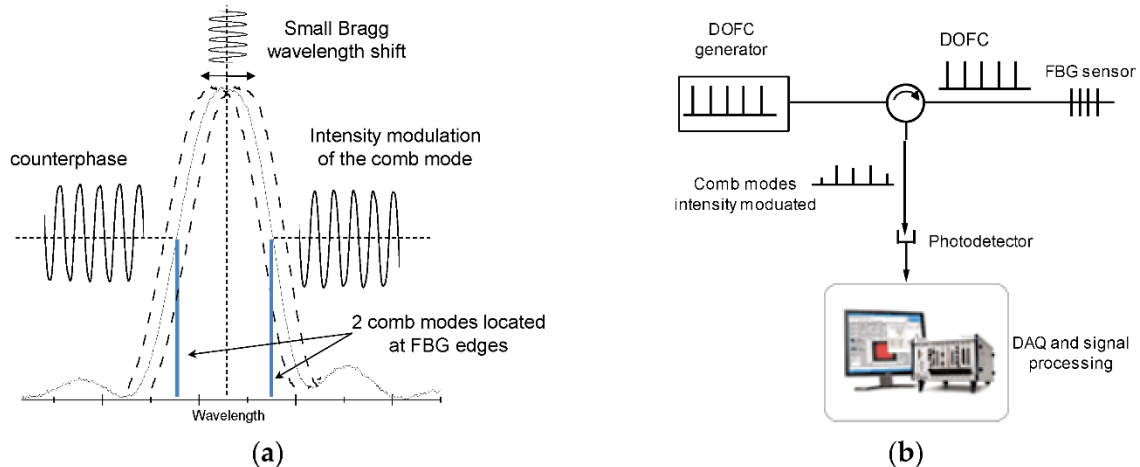


(b) High-speed large-scale FBG sensing demodulation system (Li et al., 2014)

Figure 2-13: Active Interrogation Methods



(a) Schematic Diagram of The Experimental Setup for A Resonance FDML Fiber Laser with FBG Array, Including A Semiconductor Optical Amplifier (SOA), Single Mode Fiber (SMF), Fiber Bragg Gratings (Fbgs), And A Fiber Fabry–Perot Tunable Filter (FFP-TF).
(Park Et Al., 2017)



(b) FBG interrogation system based on a dual optical frequency comb (DOFC). (a) Principle of the proposed interrogation method. (b) Simplified optical scheme of the proposed interrogation system. (Posada-Roman et al., 2016)

Figure 2-14: Complex Optical Sources Used in FBG Interrogation

A feature of FBGs that makes them an attractive sensing technology is the ease with which an array of sensors can be multiplexed within a single optical fibre.

Multiplexing approaches that have been reported include time division multiplexing (TDM), wavelength division multiplexing (WDM), spatial division multiplexing (SDM), frequency modulated continuous wave (FMCW), and some other hybrid multiplexing methods (Sahota et al., 2020).

WDM and TDM are two most commonly used approaches to multiplex FBG sensors. In wavelength division multiplexing (WDM), each FBG in the array is encoded with a unique and distinct Bragg wavelength. The FBG array can then be interrogated using a broadband light source covering all wavelengths and demultiplexed and demodulated using, for example, a tuneable filter. Alternatively, the array could be addressed by wavelength tuneable laser and the return light monitored on a photodiode. The WDM approach can be used in cases where adjacent FBG sensors have arbitrary physical separation. In TDM, an array of identical FBGs is generally addressed using a pulsed laser (Yu and Yin, 2002) or switched semiconductor optical amplifier, with the returned pulses processed based on the wavelength shift dependent reflected power, or using phase sensitive detection schemes. (Dai et al., 2008; Dai et al., 2009). Finally, the frequency modulated continuous wave (FMCW) approach, also known as optical frequency-domain reflectometry, can provide the high-spatial resolution with large multiplexing capacity using a source with narrow line-width and fast acquisition setup (Liu et al., 2016).

Meanwhile, the FBG has a low intrinsic sensitivity to pressure, which must be enhanced for most applications by packaging to transduce pressure to strain values. Aime et al. (2021) operated a pressure sensor based on optical fibre strain sensors with several aerodynamic conditions. Two techniques have been applied to measure different parameters, fibre segment interferometry (FSI) and fibre Bragg gratings (FBG), and considered thermal and pressure sensing sensitivities, particularly the pressure and temperature separations. The pressure responses of FSI sensors were characterised by placing the sensor package within a pressure chamber under a stable thermal condition (22°C). Here it was noted that the pressure responses of the FSI elements bonded to the outer annulus of the composite membrane experienced a compressive load, while the

FSI element at the centre of the diaphragm experienced tension. The thermal responses were then separately characterised by placing them in an oven, where the temperature was varied from 37°C to 82°C, where the segments all showed the same thermal sensitivities. Under the assumption that the pressure and thermal responses were independent, processing the responses of the FSI elements at the edge and centre of the diaphragm in a simple matrix operation could separate the pressure and temperature responses of the membrane with low errors. And the sensitivity of the measured pressure was decreased due to the testing temperature increase when the fibre membrane stiffness is increased as well. A further study completed by Chehura et al. (2007) has investigated temperature and strain discrimination with FBGs. They applied a single FBG sensor to separate between temperature and strain, which coupled a tilted fibre Bragg grating (TFBG) within the core-cladding mode. Such core and cladding mode experienced different thermal sensitivities but the strain sensitivity was about the same. Meanwhile, a small tilt angle has been applied to a single cladding mode where the spectrum required a narrower range than TFBGs occupied. Thus, the multiplexing division trends of the wavelength has been observed that allowed the temperature and strain separation within the same spectral range.

Fibre Bragg gratings are primarily sensitive to temperature and strain. FBG based pressure sensors operate by transducing the pressure to a strain acting on the fibre. As with most sensing technologies the temperature cross sensitivity issue introduces a requirement to compensate for, or separate, the response to temperature. Several approaches have been explored by researchers to solve this issue (Sahota et al., 2020). Generally, the approach adopted involves the use of two FBGs, one experiencing both strain and temperature, the other isolated from strain. Other approaches have adopted FBGs written in fibres with different diameters or of different composition, such that they produce distinct responses to strain and temperature, which may be separated using a simple matrix operation. Liang et al. (2018) created a temperature compensated FBG pressure sensor by using a plane diaphragm. Two FBG sensors were used, one experiencing both the pressure induced strain and the temperature. The other

was the temperature compensation FBG, which was isolated from pressure to experience only temperature, allowing the thermal influence to be removed from the pressure measurement. The two sensors were located near each other to ensure that they were isothermal. A discrimination technology for temperature and strain was easily employed reported by Tiwari et al. (2014), who fabricated the FBG sensor in the core of an erbium doped fibre (EDF). They observed that the amplified spontaneous emission power varied linearly with temperature, and the strain was then estimated by subtracting the wavelength shift due to the temperature change, from the measured Bragg wavelength shift. According to their measurements, this approach allowed independent strain and temperature measurements with root-mean-square errors of $21.2 \mu\epsilon$ and 1°C over ranges of $0\text{--}800 \mu\epsilon$ and $40\text{--}95^\circ\text{C}$.

Another method to compensate temperature effect has been attempted by Tian et al. (2005). They fabricated an FBG strain sensor based on the cantilever beam which was made of the bimetallic material. Such a specifically designed sensor can reduce the wavelength shift because the induced temperature was from more than one magnitude order. Two different coefficients of thermal expansion (CTE) bimetallic strips were placed together in parallel, and the FBG was placed at two positions onto the metal where the lower CTE obtained. The strain value was transferred from higher CTE metal to lower CTE metal, then, further moved to the FBG sensor when the temperature was applied. According to this attempt, the temperature dependence of Bragg wavelength was reduced by $-0.4 \text{ pm}/^\circ\text{C}$.

Another method to compensate temperature effect was reported by Tian et al. (2005). They fabricated an FBG strain sensor based on a cantilever beam which was made of the bimetallic material. The sensor design reduced the temperature induced wavelength shift by more than one order of the magnitude, from approximately $12 \text{ pm}/^\circ\text{C}$ to $-0.4 \text{ pm}/^\circ\text{C}$. Additionally, Kim et al. (2000) used a chirped fibre Bragg grating (CFBG) partially embedded in a SiO_2 glass tube. The embedded section was thus isolated from strain transfer of temperature under test. The uncovered grating portion responded to both strain and temperature.

Considering these findings, FBGs seem to be suitable for this research project as the test environment is required to have a low intrinsic pressure sensitivity and the temperature around the ambient level, so that the pressure data can be transduced to strain results and then to be illustrated and analysed during experiments. The in-fibre device is sensitive to the strain, and the diameter of fibre is of order $100\mu\text{m}$ with the gauge length of a typical FBG sensor lying between 1mm to 5mm. Also, the fact that multiplexed sensors can contain 10s of sensors in one single optical fibre using a commercially available interrogator is an attractive feature. In particular, FBGs can be considered to be a label formed within a short section of optical fibre, they can determine local parameters, strain and temperature respectively (Wang et al., 2021).

Due to the wide employment of FBG sensors in energy and power, aerospace, manufacturing, health care fields (Wang et al., 2022), several studies are reviewed here to illustrate the applications of FBGs in both academic and industrial areas. Berri et al. (2019) investigated the use of FBG sensors for the monitoring of Electro-Mechanical Actuators (EMA) and Electro-Hydrostatic Actuators (EHA) in aerospace applications, where they showed that it was possible to determine the aerodynamic hinge loads and to measure the local invasive temperature levels. A dedicated test bench was used to calibrate the FBG sensors, allowing the analysis of the strain and temperature responses, as well as considering the proper thermo-mechanical compensation strategy, so that such technology could be evaluated with different installation methods and defined the suitability of the mechatronic system configurations. Meanwhile, the crack monitoring on the aircraft structure has also been demonstrated using FBG sensors (Liang et al., 2019) during fatigue tests. According to their tests, FBGs can monitor the crack growth stages and can also define the occurrence time of fatigue cracks and the general location where the phenomenon happened. Furthermore, James et al. (2022) and Kissinger et al. (2022) have applied the FBG sensors to measure the strain and shape on the 5m length rotor blade during a 6.6Hz rotation rate helicopter ground run. They used FBG and FSI approaches to investigate the strain sensing ability (James et al., 2022), and the FSI method was particularly operated on the measurement of the blade shape (Kissinger et

al., 2022). They recorded the blade dynamic data through rotor hub-mounted sensor interrogators, which came from the controlled pilot input order, and analysed these results in time and spectral directions. Finally, they concluded that this study provided proceptions to the blade aeroelastic characteristics.

Structural health monitoring (SHM) is an area of application for FBGs that has been extensively investigated. Wu, Okabe and Yu (2018) reported the use of FBG sensors in ultrasonic SHM research on a plate-like structure, and noted that FBGs exhibited a higher sensitivity and broader bandwidth to ensure the high-fidelity measurement of the ultrasonic wave for subsequent analysis of the subsequent waveform. They also suggested that the test system, FBG installations, as well as the data processing should be designed carefully. Also, use of FBGs for SHM applications in composite materials was undertaken by Kinet et al. (2014). FBG were selected due to their lightweight and small dimensions, as well as the multiplexing ability. There were some challenges encountered when using FBGs in SHM tests, including the discrimination of strain and temperature effects, the amplitude spectrum demodulation, and the interface connections between embedded FBGs and surroundings. According to their research works, the connection and the curing cycle monitor aspects needed to be improved, especially in the industrial process area. Furthermore, the application of FBG strain sensors has been operated on monitoring the railway track component condition (Buggy et al., 2016). They installed the FBG sensor arrays on fishplates, switchblades and stretcher bars to investigate the safety condition of these critical rail parts. This study illustrated that in applying FBGs technology, the safety condition changes of those components could be observed and the continuous monitor allowed, providing remarkable safety and budget advantages.

2.4.2 Applications in the Automotive Field

According to the section 2.4.1, the wide-ranging use of optical sensors has spread quickly to various fields. The technology has also found application in automotive research and in the vehicle industry. Aulakh (2013) reported that most European vehicle manufacturers have installed MOST (Media Oriented Systems Transport) system, which is used to communicate vehicle systems via a high-

speed network. BMW was the first company using the fibre optical technique in the airbag control system under the MOST standards, called 'Byte Flight System'. BMW has established a plastic optical fibre (POF) network, which is a 10 Mb/s star network used to connect with airbag sensors. The force optical fibre sensor's data information is transmitted using the POF network to control the air bags' inflation when a collision is sensed, with the fibre communication systems offering fast communication capability.

A study has applied the laser holographic interferometry to determine the shape of vibration modes, completed by Talbot and Fieldhouse (2002). Figure 2-15 illustrated the result of such a study where the diagonal fringes observed along the brake disc edge correspond to a combination of the rotational motion, and the in-plane and out-of-plane vibrations. They observed that the backplate trailing end of the brake pad has occurred a 'tension' abutment design tended to vibrate in the audible noise and compared both pad and associated disc vibrations. They also suggested that the 'triggering' noise instability of the brake pad abutment was critical happened and summarised some conclusions. It was found that the pad effective length could be lower than its design length, and that the antinode would generally compress to fit under the brake pad frictional interface with a maximum antinode compression that was $2/3$ of the existing free mode angle. Finally, it was noted that the free mode antinode subtended angle must be greater than that of the pad.



Figure 2-15: Photograph of Mode Shapes Using Laser Holographic Interferometry
(Fieldhouse, 2013)

Due to the development of fibre optical sensors, the measurement of various parameters in automotive fields have become possible under various challenging environments. A fibre optical interferometric sensor has been applied to real-time monitoring of the dynamic pressure and temperature in a vehicle engine at the exhaust pipe (Bock et al., 2004). They used a highly birefringent fibre sensor, which monitors the measurand induced phase shifts between two pairs of polarized modes of the optical fibre. They reported that the system ran with a data rate of 200kHz and could measure pressure and temperature in a range of 140Mbar and 100°C, with a 2Mbar and 0.4°C accuracy level, respectively. Their test results indicated that the sensor data determined the exhaust gases flow and the mass emission information, which displayed the possibility of the engine operation diagnosis via those measurements.

There are several attempts to install FBG sensors on the brake pads for braking research aims. Major (2016) completed tests to demonstrate the strain transfer from the friction material to a fibre optic strain sensor under static loading conditions, and also compared the measured strain distribution with the data from PSP and FEA. Major et al. (2017) then explored the possibility of using the FBG sensors to measure the pressure distributions at the pad/disc interface. They measured strain transfer with a variety of pressure levels, and found a linear response relationship with the pressure level increasing (Figure 2-16).

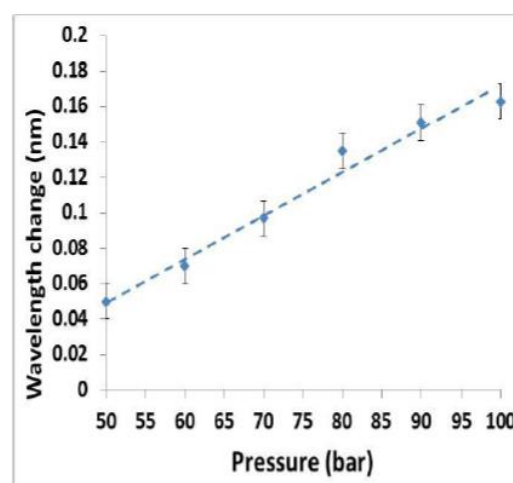


Figure 2-16: Linear Relationship Between Wavelength Change and Hydraulic Brake Actuation Pressure (Major et al., 2017)

It was shown that the FBG sensors respond to application of the static pressure and torque on the brake pad but there was a need to improve the result accuracy as their FBG sensors were installed deep into the grooves in the friction material, 5mm away from the contact surface. In addition, measurements were undertaken only under static and quasi-dynamic conditions. Therefore, this research project will be attempted to measure the interface strain distributions on the exact brake pad frictional surface. Meanwhile, Lenormand (2017) reported a further attempt using FBGs to characterise the interface pressure distribution under the same test conditions explored by Major (2016). Here, the visco-elastic phenomenon was investigated under the applications of the static pressure and the slip characteristic of the brake disc under the quasi-dynamic conditions was observed.

These two attempts showed that FBG sensors could provide unique insights in to the interface between the friction material and the disc, but a number of aspects of the experimental procedure and conditions needed to be improved and extended in order to obtain a deep understanding of the interface strain distributions under different test conditions. Firstly, the installed location of the optical fibres should be moved as close as possible to the brake pad interface so that more accurate results could be obtained and analysed. Then, the number of FBG sensors should also be increased in order to provide more data to build a full image of the interface strain distributions. Last but not least, the experimental and FE analysis phases need to be designed and validated more comprehensive, not only considering the loading conditions, from static to quasi-dynamic and to dynamic, but also thinking about the test conditions, including the applied hydraulic pressure and torque levels, and the provision of a means to adjust the pressure distribution under these conditions by redesigning the experimental rig. The previous work had also not considered the experimental uncertainties. In addition to fully understand the suitability of FBGs for monitoring automotive braking systems, there was a requirement to extend the experimental phase to undertake measurements under dynamic conditions.

2.5 Summary

The chapter represents transdisciplinary nature of this project, discussing automotive brake system issues and optical measurement technology, thereby connecting the brake research with the FBG sensor technique.

A brief overview of the brake noise and vibration field has been presented in section 2.2.1 and 2.2.2, where several types of noise and vibration studies were reviewed. The brake noise and vibration, including squeal noise, groan noise and stick-slip vibrations, are still a problem attracting researchers' attention using both theoretical and experimental methods, which could determine the phenomenon with increased accuracy aiming to reduce NVH issues and improve the vehicle performances. Therefore, the brake noise and vibration issues will be considered as well when taking the dynamic tests and trying to measure it in a different test method, even providing feasible solutions to improve the reduction and prediction of noise and vibration from a novel perspective.

Then, the interface pressure distributions (IPD) have been reviewed and discussed in section 2.3, containing the theoretical finite element analysis (FEA) method and the experimental PSP determinations. Such technologies have improved the understanding of the IPD characteristics. Due to the lack of attempts in the actual measuring level, therefore, this project will pay more attention to both simulating the IPD, in particular, interface strain distributions in static and quasi-dynamic conditions, as well as maintaining various test conditions, and also the ultimate aim, which has been mentioned in Chapter 1, is to determine the interface strain distributions during the dynamic tests by using the FBGs mounted on the modified brake pad.

The review of the optical measurement technology has focused on optical fibre sensing applications, and such attempts were made in automotive fields in sections 2.4.1 and 2.4.2. The FBG sensors have been applied in academic and industrial areas, which showed the great potential to measure the specific characteristics. However, the lack of attempts in automotive fields was noteworthy inasmuch as such technology can play an essential role and fulfil use to improve the development of the automotive design and manufacture,

especially in the brake research and industry. Therefore, this project intends to apply such creative technology to brake research fields demonstrating the bright future in static, quasi-dynamic, and dynamic phases.

According to these reviews, a further experimental design concept can be devised, and the practical tests will also consider existing works in relevant research areas to ensure the project's keeping on the right track.

3 EXPERIMENTAL FACILITIES

The research presented in this thesis employed several experimental facilities, including the designated test rig, the modification of brake pads, and the FBG sensors and interrogator. Some of the test equipment are entirely new designs, taken from the initial concept to the final product, while the others are based on existing systems, which only need to be installed and adjusted.

3.1 The Test Rig

A reliable and efficient test rig was designed and manufactured in order to simulate actual disc brake operating cases, and to facilitate the investigation of interface strain distributions on the brake pad/disc surfaces. Generally, the test rig contains two sub-assemblies, Brembo Land Rover disc brake components and a two-channel control hydraulic pressure system. Each section of this chapter refers to a specific design concept, leading to the final product manufacture and commissioning. In such a way, the role of each component, sub-assembly, assembly and system can be clearly understood, as well as their role in meeting the research objectives.

3.1.1 Brembo Land Rover Disc Brake Assembly

The incorporation of the Brembo Land Rover disc brake into the test rig includes three aspects, the disc brake selection, adaptor design concepts and the final assembly. Those three aspects will ensure the Brembo Land Rover disc brake working properly within the laboratory environment, and that it could meet the full range of the test demands.

3.1.1.1 Disc Brake Selection

The disc brake system applied during these experiments would require high braking performance and capacity. It should have a wide range of applications in various vehicles, hence meeting stringent design, performance and reliability requirements. Furthermore, such a system should enable measurement of a range of disc brake characteristics and be able to simulate actual working conditions. Consequently, components manufactured by Brembo, which is an

acknowledged world leader in the design, development and production of automotive brake technology, were considered. The Brembo company supplies the high-performance brake systems for many car manufacturers, commercial vehicles and motorbikes worldwide (Brembo S.p.A., 2022). Considering the research demands of one of their customers, Jaguar Land Rover, a global automotive manufacturer and leading technology company, the application is related to one of their vehicles. Jaguar Land Rover consists of two car brands: Land Rover, the world's leading manufacturer of premium all-wheel-drive vehicles; and Jaguar, one of the world's premier luxury sports saloon and sports car marques (Jaguar Land Rover Automotive Plc., 2022).

Based on those considerations, three components needed to be selected for the experiments, the brake calliper, the brake pad and the brake disc. Such disc brake systems have been installed on different series of cars, including Land Rover Discovery, Range Rover Sport, and Range Rover Velar (Figure 3-1). Therefore, it was considered that the Land Rover disc brake provides stable and efficient working conditions to support these experiments and is suitable for the investigations of the interface strain distributions and the measurements of brake properties.



Figure 3-1: Different Series of Cars with the Same Disc Brake System (a) Land Rover Discovery, (b) Range Rover Sport and (c) Range Rover Velar (Jaguar Land Rover Limited, 2022)

The calliper selected (Brembo DB0.T8.ED) is shown in Figure 3-2 (a), used on the front axle. It is the 'four pot fixed calliper', manufactured from an aluminium alloy. The steel pistons are typically solid (there are applications with 'hollow', i.e.

scalloped pistons) and there is a single hydraulic connection. The brake disc (Brembo 09.8877.30, Figure 3-2 (b)) is a 'swan-neck-type' disc, ventilated using pillars rather than vanes, in order to improve noise, vibration and harshness (NVH) characteristics and cooling at low speeds. The brake disc is made from high carbon (HC) cast iron GG150 that provides high thermal conductivity and high vibrational damping, reducing possibilities of NVH issues arising in the vehicle operation. The brake pad (Brembo P44018, Figure 3-2 (c) and (d)) is composed of an organic (composite) frictional material bonded to a steel backplate. A 'shim' is bonded to the backplate 'free' surface (in contact with the piston) to improve brake NVH characteristics (Figure 3-2 (d)). (Brembo S.p.A., 2022). Another important aspect when selecting brake assembly is to have a good research base regarding the characteristics of all brake parts, and the entire assembly. The PhD thesis by Vianello (2015) provides such a foundation.

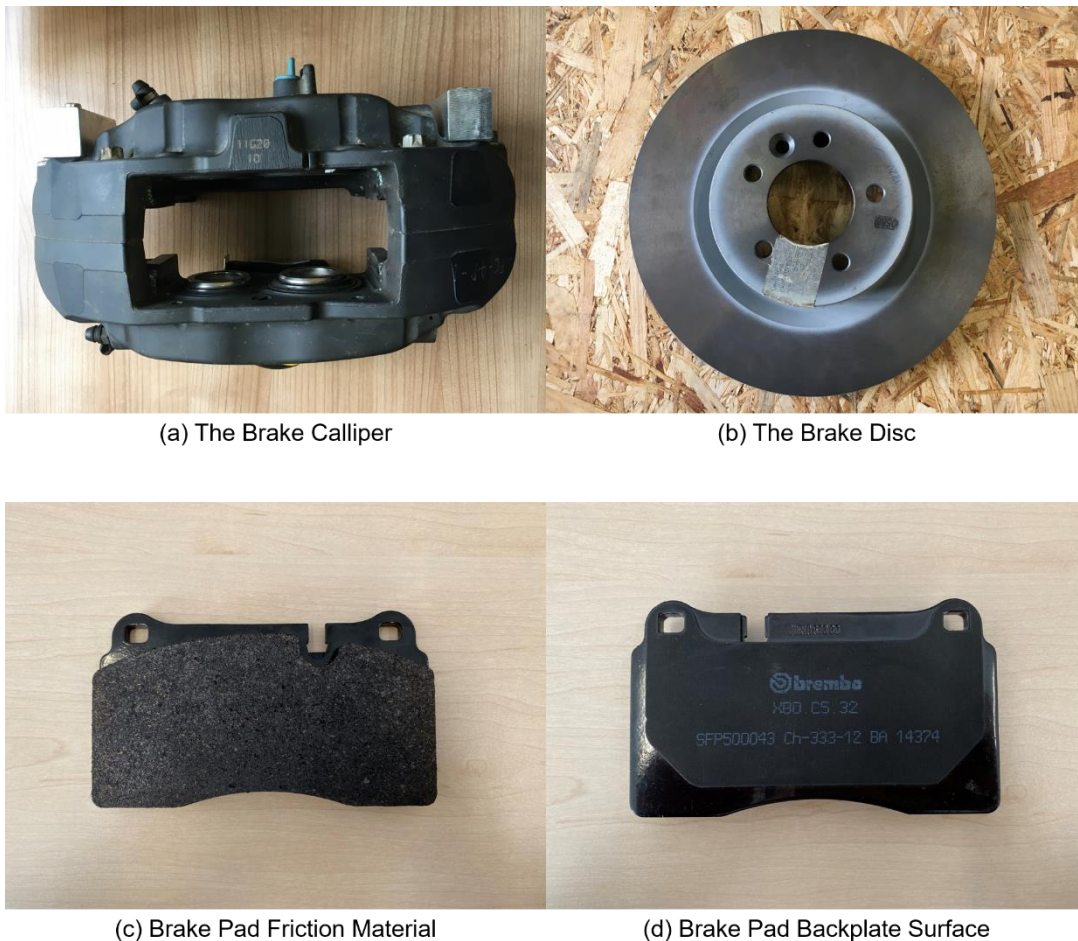


Figure 3-2: Actual Components of the Land Rover Disc Brake

3.1.1.2 Design Concept of Adaptors

To mount the braking system components in a way that was suitable for the planned experiments, several adaptors were required. According to Figure 3-3, Three specific adaptors were designed and created to combine with those components, including the adaptor 1 (4), the adaptor 2 (1) and the outside ring (3), respectively. Meanwhile, three original components have been placed as well, the original adaptor A (13), the original adaptor B (5), and the brake calliper (12), other numbers in Figure 3-3 were different size of bolts used to connect with each of the other component, except number 2, which was the modified brake disc.

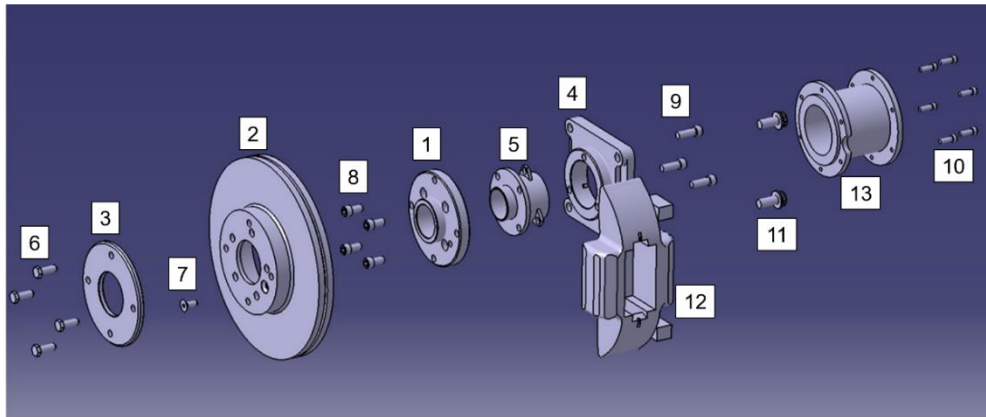


Figure 3-3: Schematic Diagram of The Disc Brake Components and Adaptors: (1) Adaptor 2, (2) Brake Disc, (3) Outside Ring, (4) Adaptor 1, (5) Original Adaptor B, (6) M12x1.75x30 Bolt, (7) M10x1.5 Bolt, (8) M12x1.5x25 Bolt, (9) M12x1.5x35 Bolt, (10) M8x1.5 Bolt, (11) M16x1.5 Bolt, (12) Brake Calliper, (13) Original Adaptor A

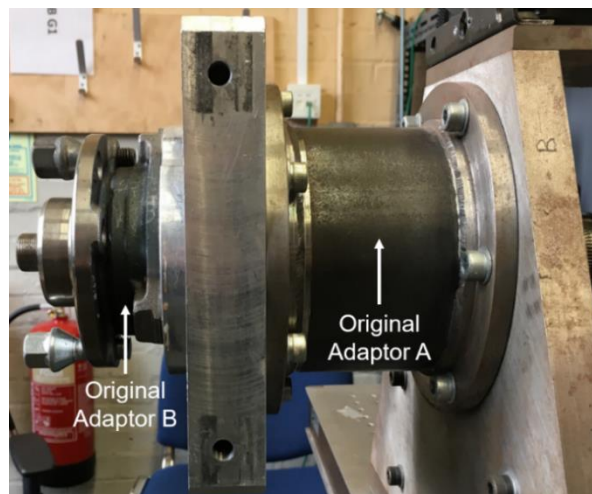
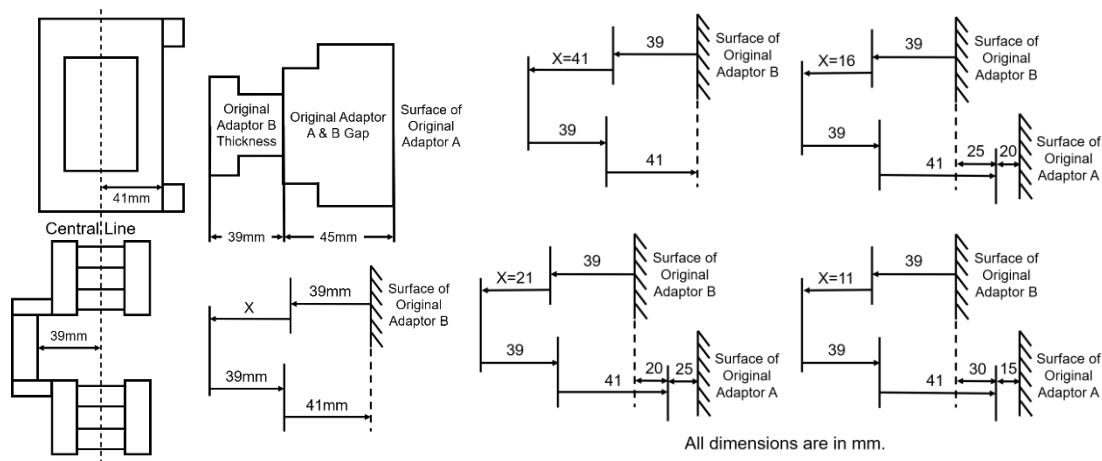


Figure 3-4: Original Adaptor A and B for Test Table and Component Connections

Two original adaptors (Adaptor A, a steel ‘spool’ and Adaptor B, wheel hub bearing from Lotus S2) were taken from another project to be used on the rig, which was to be further developed (Figure 3-4). The new adaptors were designed to connect brake components with the original adaptors and ultimately the brake rig.

The first step involved calculating each adaptor’s thickness. The surface of the original adaptor A would be the reference plane and other components would be assembled onto this. The gap between the original adaptors A and B was measured to be 45mm, which would be the thickness of the adaptor 1, 25mm would act as the supporting plate, to fit in and connect those relevant parts (Figure 3-5 (a)). The central line of the brake calliper to the reference plane was 41mm, while the distance from the central line of the brake disc to the inner surface has also been measured and checked to be 39mm.



(a) Measurements of Thickness (b) Thickness Calculations and Optimisations

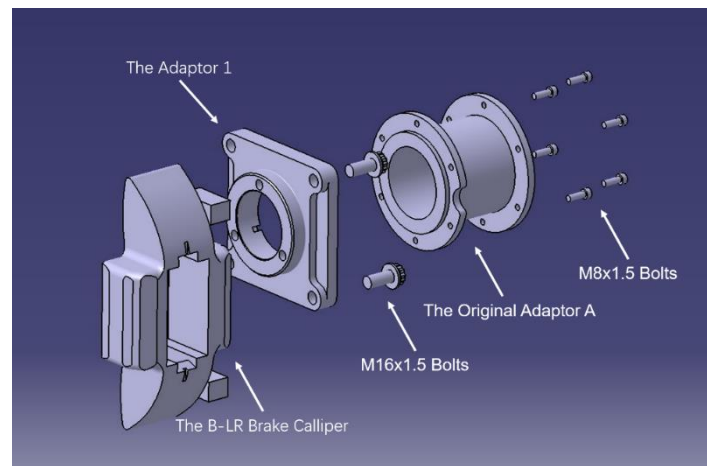
Figure 3-5: The Calculation and Selection of Adaptors 1 & 2

Therefore, the thickness of the adaptor 2 has been set as X in Figure 3-5 (a) and in the further calculation and optimisation illustrated in Figure 3-5 (b). In order to ensure that the designed adaptor 1 has enough strength and stiffness, calculations were performed that provided several options for the possible thickness of the adaptor 2, being 41mm, 21mm, 16mm and 11mm respectively. Considering the reasonable thickness of aluminium plate and the drilled holes on the plate, the adaptor 2’s thickness has been optimised to be 21mm, so that the

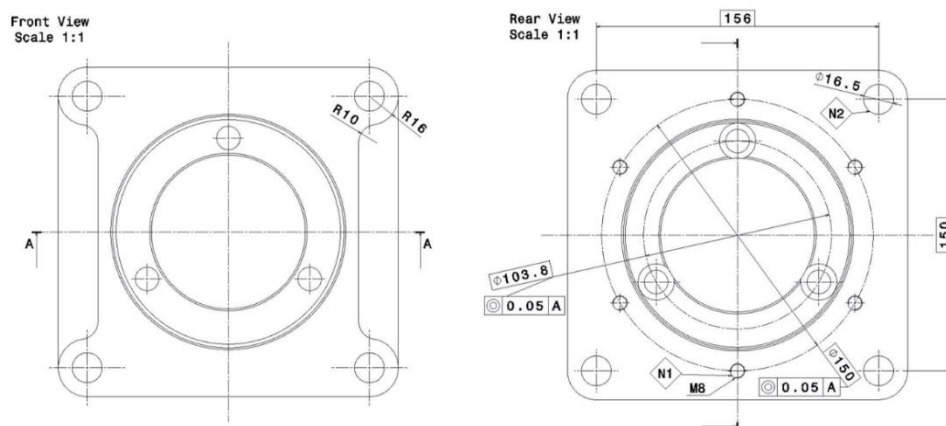
adaptor can be installed between the brake disc and the original adaptor B safely and reliably.

3.1.1.2.1 The Adaptor 1

Adaptor 1 (Figure 3-6) was devised to connect the brake calliper and the original Adaptor A. Therefore, the general concept was based on an aluminium plate with several sizes and location holes for different types of bolts to fixate the original adaptor A and the brake calliper together. CAD models are shown in Figure 3-6 (a) and the manufacturing drawing in Figure 3-6 (b). Six M8x1.5 bolts were used to connect the original adaptor A with the Adaptor 1, and two M16x1.5 bolts were used to link the Adaptor 1 to the brake calliper.



(a) Connections between Brake Calliper, Adaptor 1 and Original Adaptor A



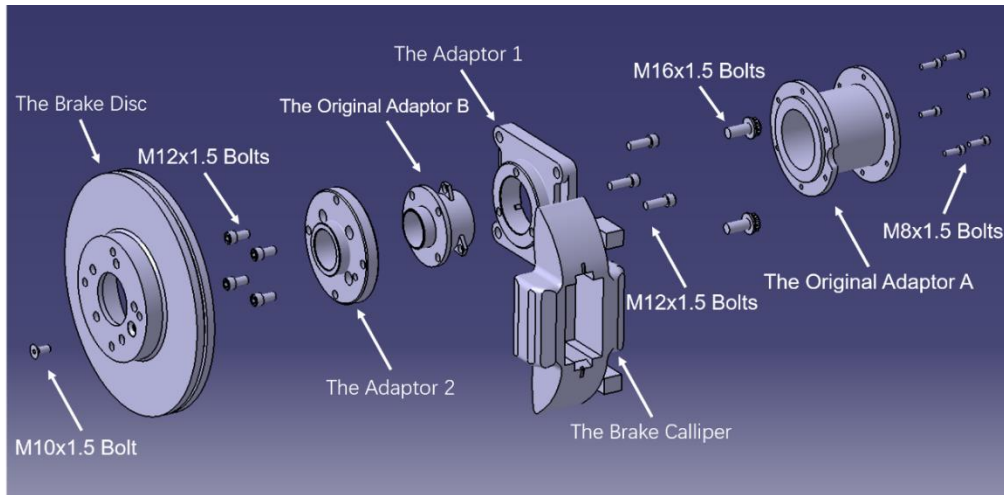
(b) CAD Drawing of The Adaptor 1

Figure 3-6: The Concept of The Adaptor 1

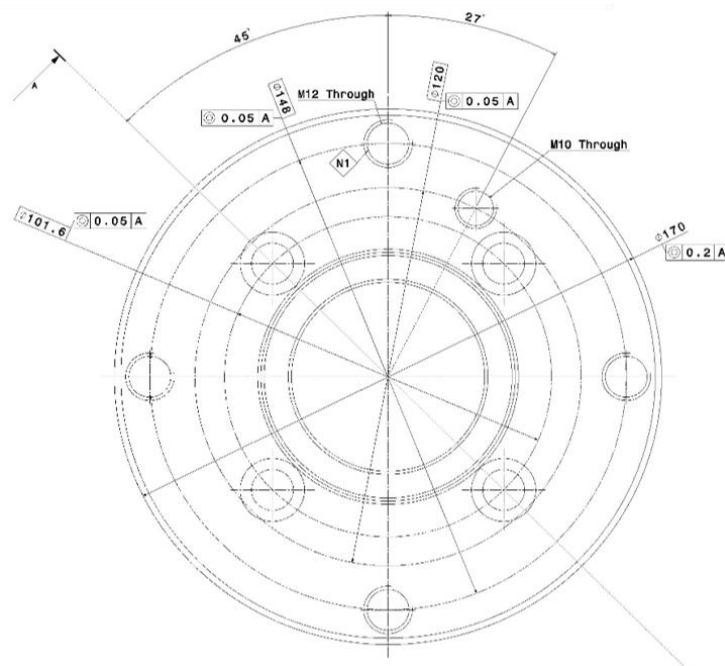
According to Figure 3-6 (b) and Appendix Chapter 3-A, the symmetric design of the adaptor (in relation to the mid-vertical plane) will ensure that the brake calliper can be placed in two orientations to suit the demands of the assembly and further planned tests. There were four clearance holes ($D=16.5\text{mm}$) at each corner on the adaptor that were to allow the M16x1.5 bolts to pass through and screw into the brake calliper. Six M8 thread holes at the backplate have been created to firmly fix the adaptor to the original Adaptor A. Three clearance holes ($D=13\text{mm}$) with the 103.8mm pitch circle diameter (PCD) suitably counterbored were machined to connect Adaptor 1 to Adaptor B. A shaped form groove has been created on each side of the adaptor, which was to ensure that the brake calliper only gets in contact with Adaptor 1 at the designed contact areas.

3.1.1.2.2 The Adaptor 2

Adaptor 2 (Figure 3-7) was created to connect the brake disc to the original Adaptor B. As mentioned before, Adaptor B is actually a wheel hub assembly from Lotus S2 vehicle (identical at front and rear axle, on both sides of the vehicle). This adaptor can directly accommodate Lotus brake discs but the Land Rover disc design is very different in terms of OD (outer diameter) and ID (inner diameter), locating diameters, offset, thickness as well as the number, size and PCD of (wheel) mounting holes. Keeping Original Adaptor B offered big advantages in terms of space and cost but an adaptor was needed to suitably connect Land Rover brake disc to the rig (i.e. the Original Adaptor B). Therefore, Adaptor 2 was created, as shown in the exploded CAD assembly view in Figure 3-7 (a) and manufacturing drawing (front view) shown in Figure 3-7 (b). Four M12x1.5 bolts were used to connect the Adaptor 2 with the original Adaptor B. A countersunk screw M10x1.5 bolt was used as a positioning bolt to keep the brake disc in a fixed position to the Adaptor B in order to install the adaptors and other units.



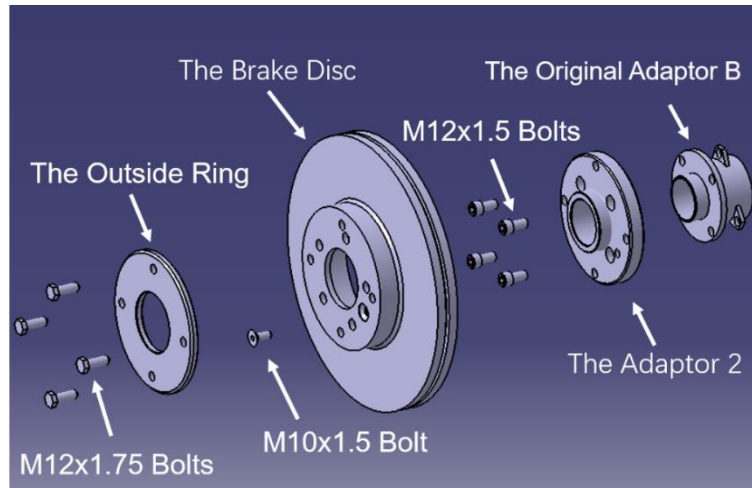
(a) Exploded View of The Assembly Process from Original Adaptor A to Brake Disc



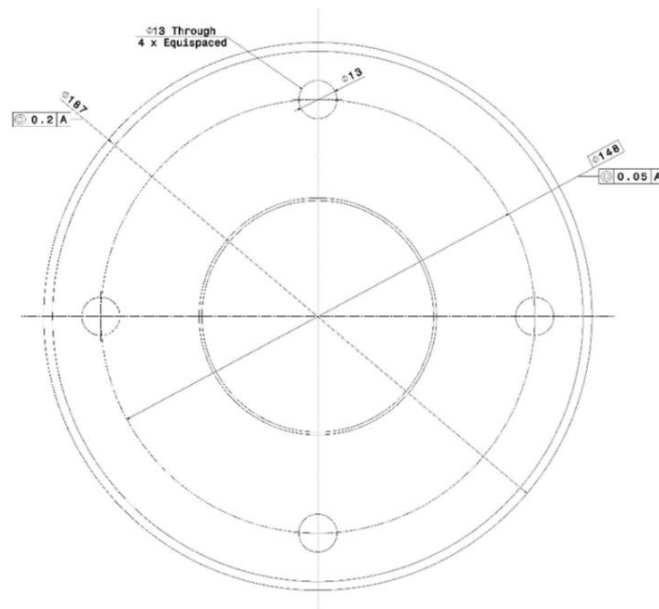
(b) CAD Drawing of The Adaptor 2

Figure 3-7: The Concept of The Adaptor 2

According to Figure 3-7 (b) and Appendix Chapter 3-A, there were four clearance holes ($D=13\text{mm}$) with the 101.6mm PCD being machined to allow $M12\times 1.5$ bolts to pass through and screw onto the original adaptor B. Four $M12$ thread holes with the 148mm PCD were designed to ensure that the brake disc can be installed properly. In addition, a $M10$ thread hole with the 120mm PCD has been drilled as the location hole of the brake disc to assemble in the right direction. The whole



(a) Connections between Brake Disc and Outside Ring

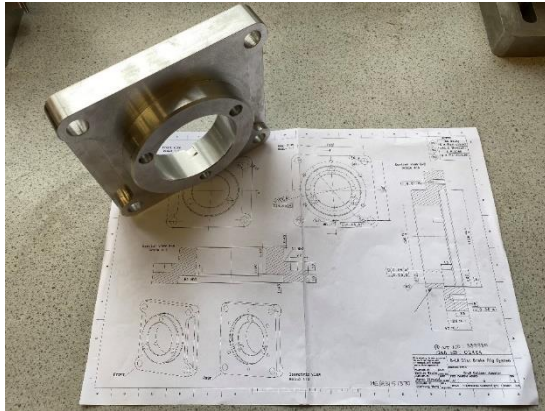


(b) CAD Drawing of The Outside Ring

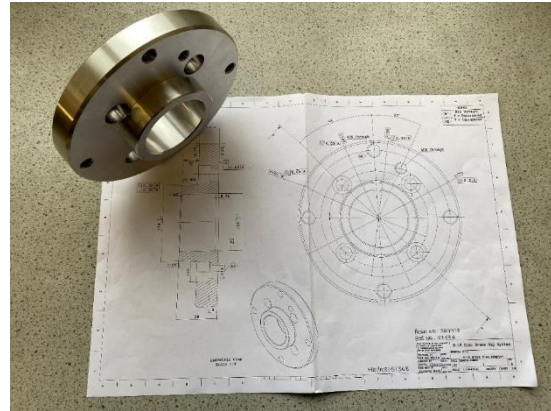
Figure 3-9: The Concept of The Outside Ring

3.1.1.3 Manufacture, Products and Assembly

All of the designed components were manufactured in the Cranfield University workshop and were draft-assembled in order to check the production quality and accuracy. Several types of machines were used during the manufacturing process and displayed in Appendix Chapter 3-B, including the mill table, the centre lathe and the CNC lathe to assist manufacturing from CAD drawings. Each component with associated CAD drawing is displayed in Figure 3-10.



(a) The Adaptor 1



(b) The Adaptor 2



(c) The Brake Disc Modification



(d) The Outside Ring

Figure 3-10: Final Productions of Designed Components

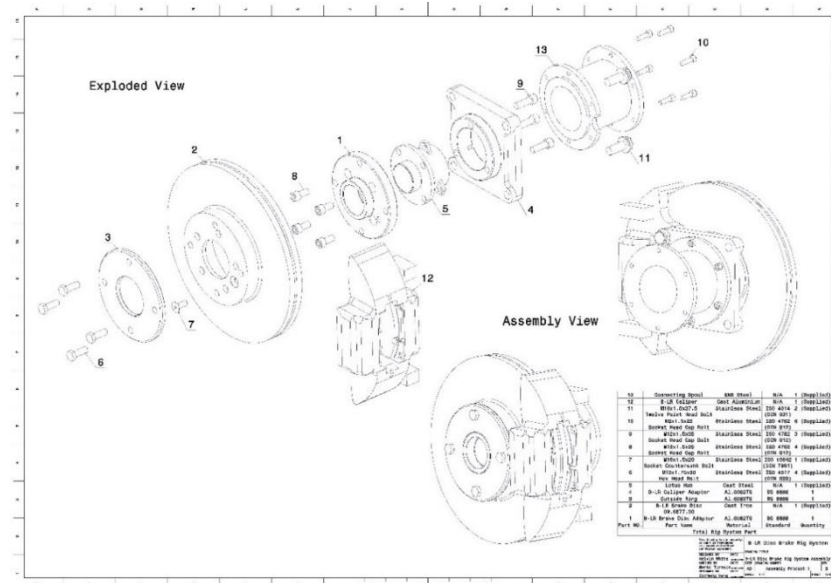
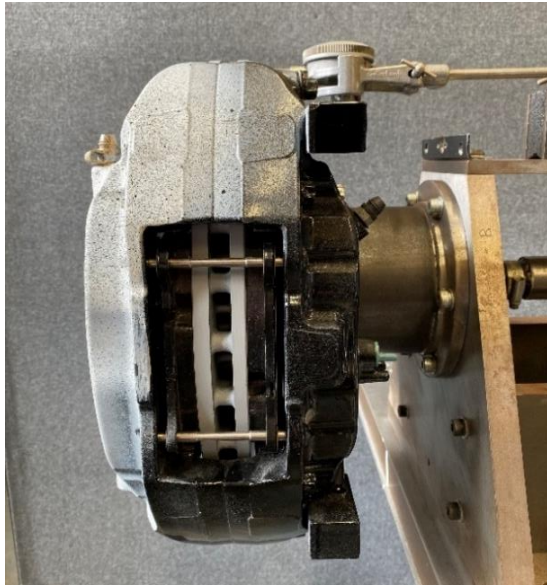


Figure 3-11: Exploded and Assembled Views of Land Rover Disc Brake Components

Finally, Figure 3-11, the drawing of all components of the Brembo Land Rover disc brake assembly are shown in both exploded and assembled views. Detailed description of the assembly procedures is provided in Appendix Chapter 3-C. The final step involves the installation of the brake calliper to the Brembo Land Rover disc brake system, with photographs of the complete assembly shown in Figure 3-12, in three side views.



(a) The Front View



(b) The Rear View



(c) The Isometric View

Figure 3-12: The Land Rover Disc Brake Assembly

3.1.2 Two-Channel Control Hydraulic Pressure System

In previous sections, all mechanical (rotating and stationary) components of the brake assembly were suitably modified ensuring the vehicle-like installation on the test rig. In addition to this, a suitable hydraulic actuating system is needed. In order to enable change of the strain and interface pressure distribution along the brake pad (leading edge to trailing end), and the position of the centre of pressure, the original calliper needed to be modified as well. The calliper has one hydraulic connection resulting in the equal pressure being supplied to all 4 pistons. The modification was required to enable different pressures to be applied to the two sets of opposing pistons. In addition to calliper modification, it was necessary for a suitable two-channel control hydraulic pressure system to be designed and manufactured.

3.1.2.1 Design Concept

The initial concept is to develop a system to transfer the hydraulic pressure into two drive lines with manual control. The ability to control the pistons separately was required to allow more flexibility in the investigation of interface strain distributions. The manual control of this test system would provide more options for the design of experiments. Safety was assured via the emergency stop valve. The symbol diagram (Figure 3-13) has been created to elaborate the initial design concept and the part list is provided in Table 3-1.

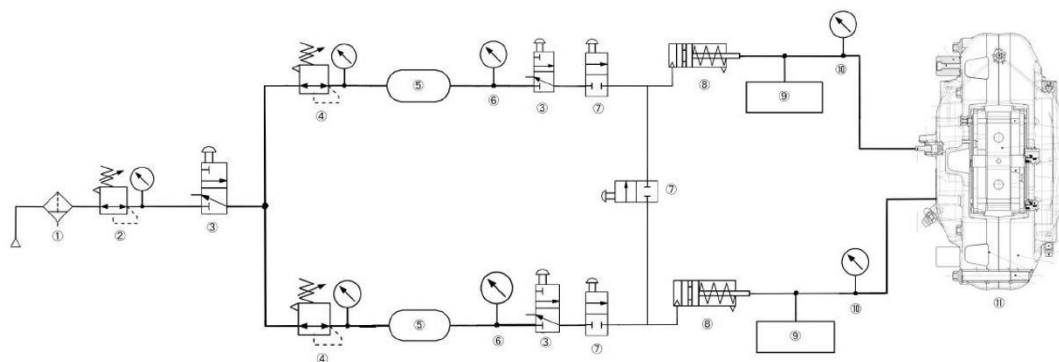


Figure 3-13: The Symbol Diagram of The Two-Channel Control Hydraulic Pressure System

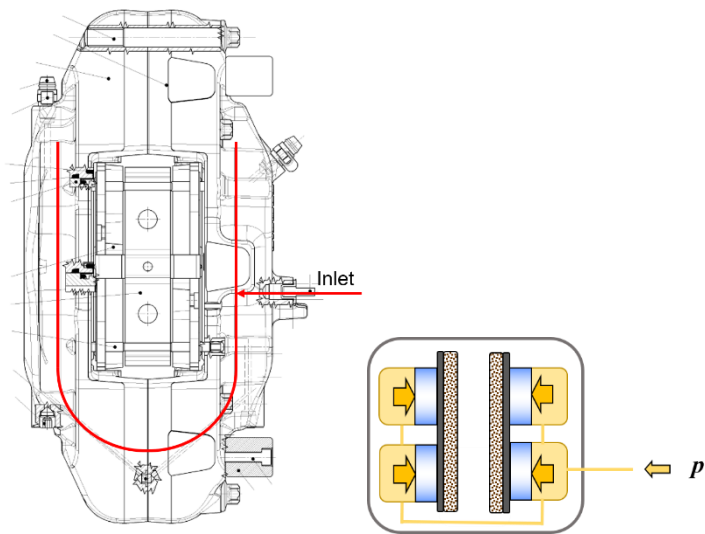
Table 3-1: The Part List of The Two-Channel Control Hydraulic Pressure System

Part NO.	Part Name
1	Manual Drain Filter
2	Air Regulator with Gauge
3	3 Way – 2 Position (3/2) Manual Valve (NC)
4	Manual Pressure Regulator with Gauge
5	5l Air Reservoir/Tank
6	Air Pressure Gauge
7	2 Way – 2 Position (2/2) Manual Valve (NC)
8	Pneumatic Cylinder
9	Brake Fluid Tank
10	Hydraulic Pressure Gauge
11	Brembo Land Rover Brake Caliper

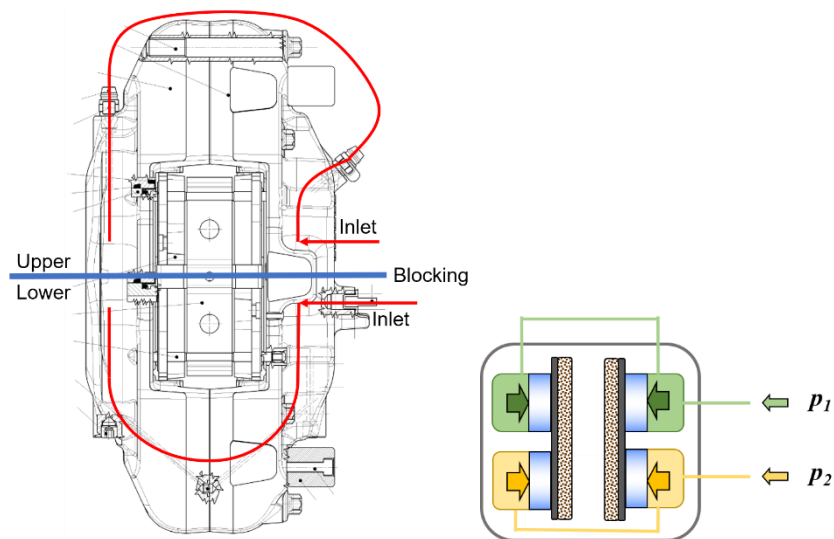
According to Table 3-1, such system consisted of several key parts, including 5L air reservoirs (5), manual air regulators with gauges (4), two types of manual valves (3 and 7), and hydraulic pressure gauges (10). Air reservoirs (5) were used to store sufficient air to ensure that the experimental tests could be repeated several times without the air supplying system being switched on and off every time. Manual air regulators (4) were used to control the air pressure levels, which displayed meter readings via the air pressure gauges (6).

Two types of manual valves, 3 Way/2 Position (3/2) (3) and 2 Way/2 Position (2/2) (7) respectively, were deployed in this system. The 3/2 manual valves (3) were installed after the air supply to satisfy the emergency stop demand such that the stored air can be exhausted immediately in times of accident. A 2/2 manual valve (7) was installed to link the upper and lower channels, such that this valve controls the air flow either from one single channel or two channels in charge of themselves. The hydraulic pressure section contained several parts, pneumatic cylinders (8), brake fluid tanks (9) and the Land Rover brake calliper (11), respectively. Those components were used to change air pressure into hydraulic pressure, actuating the brake calliper function.

Due to the concept of the pressure system, the modification of the brake calliper was necessary to allow the experimental objectives to be achieved. The original brake calliper circuit had only one inlet connection on the inboard side, and the brake fluid is guided through an internal U-circuit through the brake calliper supplying the brake pressure to the outboard piston bores (Figure 3-14 (a)). In this way all four pistons are experiencing the same pressure.



(a) The Original Brake Calliper Hydraulic Circuit



(b) The Modified Brake Calliper Hydraulic Circuits

Figure 3-14: The Brake Calliper Modification Concept

In order to create independent circuits, as shown in Figure 3-14 (b), the brake calliper was modified by:

- a) Creating an additional inlet. The calliper body was drilled, and suitable thread was created to supply pressure p_1 .
- b) An external hydraulic pressure connection was created between newly established hydraulic input on the inboard side and the corresponding bore on the outboard calliper side. This was done by removing the existing

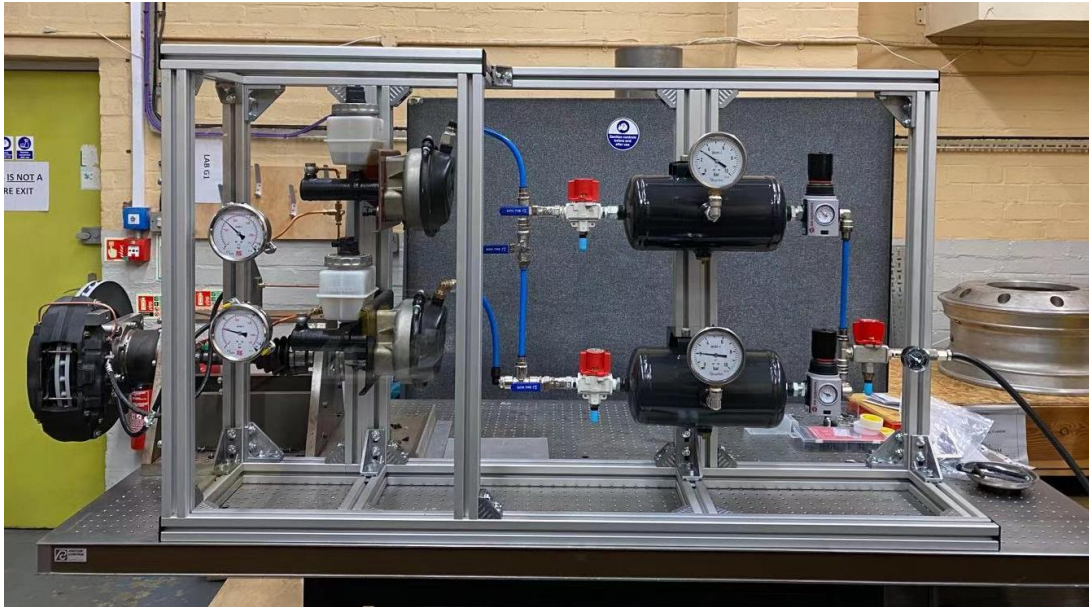
bleeding screws and installing a suitable hydraulic (copper) line with adequate fittings. No doubt, a special procedure would be needed when bleeding the system but this was achievable without exerting excessive efforts.

- c) Finally, the channel connecting upper and lower bore on the outboard calliper side had to be blocked. This was done by installing a small 'piston' and bonding it with 'liquid metal' at room temperature.

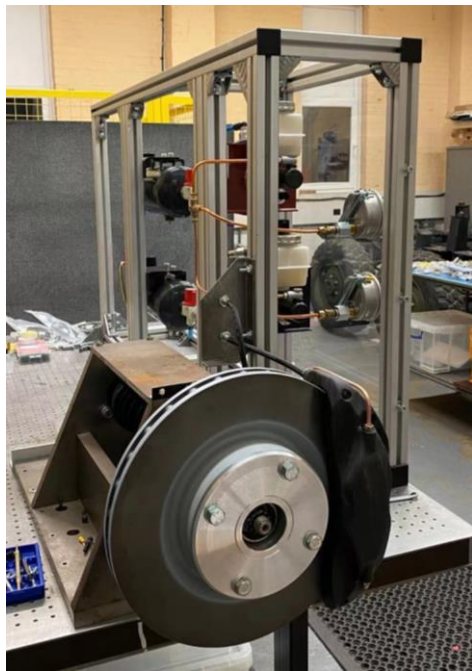
3.1.2.2 Production and Assembly

Starting from the diagram of the hydraulic pressure system shown in Figure 3-13, suitable components were specified, ordered and assembled, as detailed in Appendix Chapter 3-D. By using Bosch Rexroth profiles Aluminium, a frame structure (Figure 3-15), was created to support the components and junctions. Such frame structure contains the entire pneumatic and hydraulic pressure systems, with the only external connection needed for compressed air. Obviously the two independent hydraulic outputs are to be connected to the brake calliper (Figure 3-15 (b) and (c)). The frame is self-supported and could be easily lifted onto the test table.

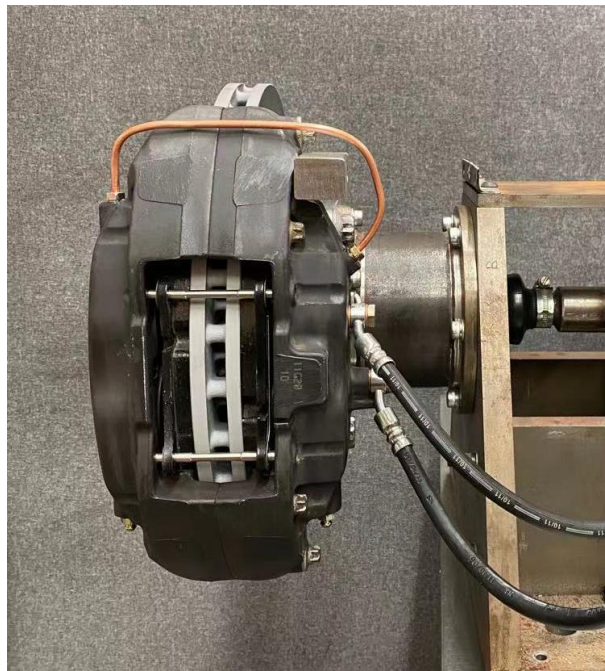
Meanwhile, another modification made to the calliper involved the placement of a small protective shield on one side of the brake calliper, which was used to prevent connecting pipelines splitting off from inlet holes and splashing brake fluid from the hydraulic pressure system accidentally (Figure 3-16).



(a) The Front View



(b) The Side View



(c) The Zoom View

Figure 3-15: The Assembly of The Two-Channel Control Hydraulic Pressure System

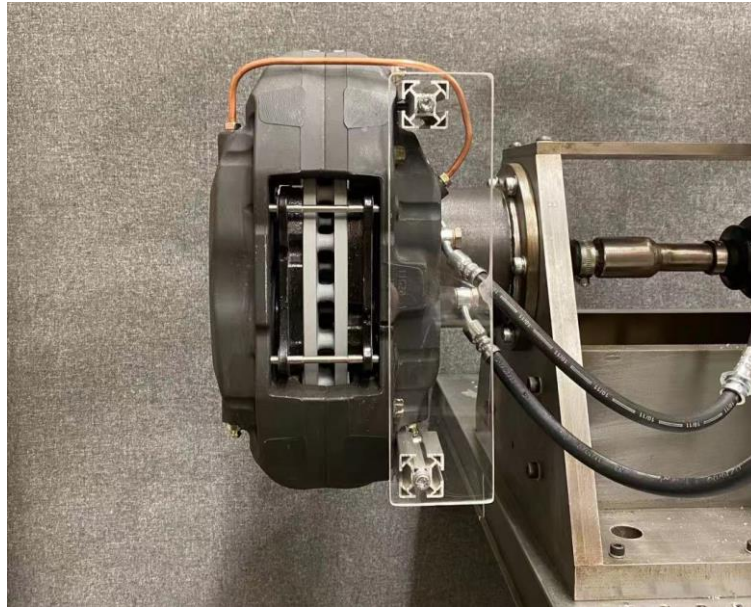
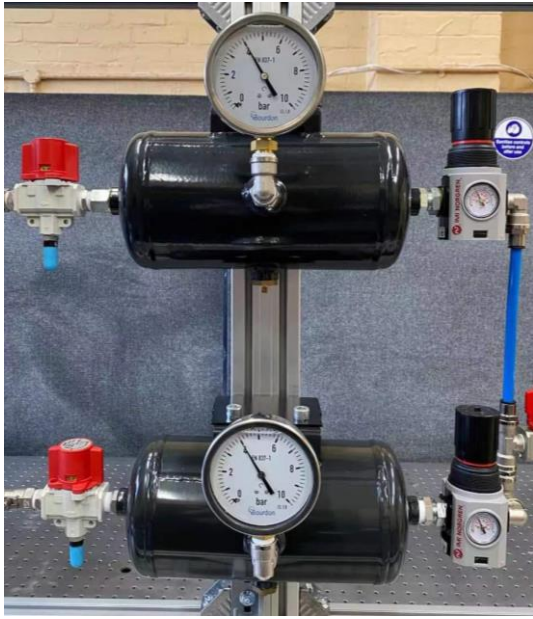


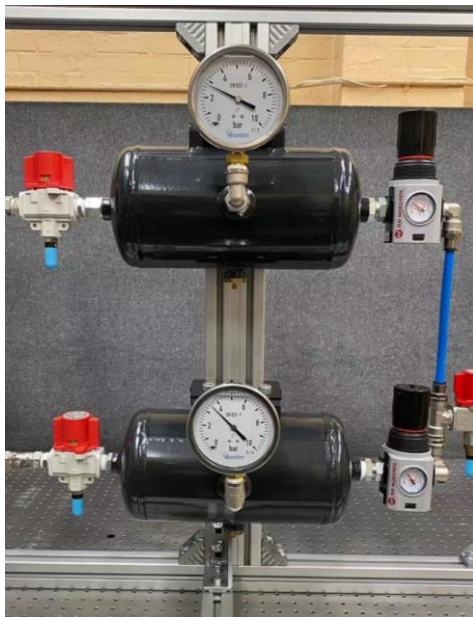
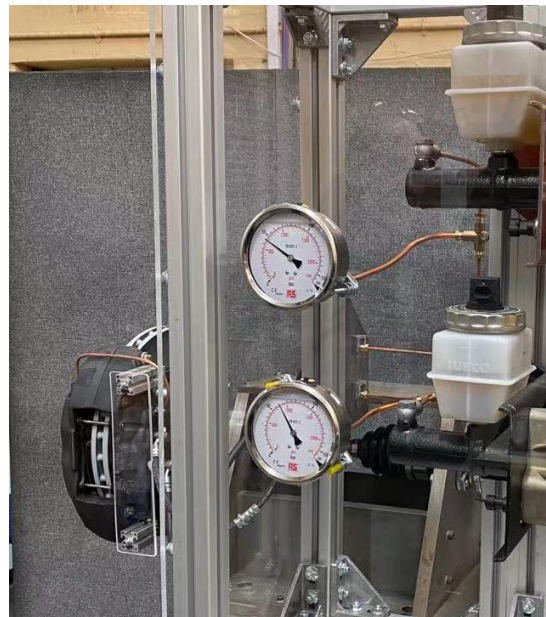
Figure 3-16: The Brake Calliper with The Protective Shield

3.1.3 Initial Test: Pneumatic & Hydraulic Pressure Relationship

Before starting the experiments, commissioning trials were undertaken to ensure that the system operated as intended. Air pressure was applied in the range from 0 bar to 6 bar in a step-by-step fashion to observe the system performance and also test different pressure levels between the two channels. Images of the pressure gauges readings for two example settings are shown in Figure 3-17 indicating that the system was capable of operating under the conditions envisioned for the subsequent experiments, with either equal pressure in both channels (Figure 3-17 (a)) or with different pressure variations (Figure 3-17 (b)).



(a) The Same Pressure Level on Two Channels



(b) Different Pressure Levels on Two Channels

Figure 3-17: Two Samples of The System Initial Test

In order to obtain the reliable and accurate relationship graph between the pneumatic pressure (P_p) and the hydraulic pressure (P_H), three repeated measurements were performed, measuring the input pneumatic pressure applied in response to each hydraulic pressure level. Due to the copper pipe pressure limitation adopted in this system, which could operate at a maximum pressure of

122 bar, thus the maximum hydraulic pressure level has been set as 120 bar, corresponding to the 6.6 bar pneumatic pressure. Therefore, pneumatic pressure levels were selected every 0.5 bar from 0 bar to 6.6 bar, with the exception of the ranges from 0 bar to 1 bar, and from 6 bar to 6.6 bar, which would be measured every 0.2 bar to gain the detailed results at both lower and higher levels. Each pressure level was tested and recorded three times to calculate the average value and the recorded data can be found in Appendix Chapter 3-E. Such measurement processes ensure the accuracy of test results (Table 3-2) and allow the plotting of a calibration graph to act as the reference benchmark during subsequent experiments (Figure 3-18).

Table 3-2: Pneumatic Pressure (P_P) and Hydraulic Pressure (P_H) Relationship

Pneumatic Pressure and Hydraulic Pressure with Fitting Result ($y = 18.87x - 4.0554$; $R^2 = 0.9999$)	
Pneumatic Pressure (bar)	Hydraulic Pressure (bar)
0	0
0.4	3
0.6	6
0.8	10
1	15
1.5	24
2	34
2.5	43
3	53
3.5	62
4	71
4.5	81
5	90
5.5	100
6	107
6.2	113
6.4	117
6.6	120

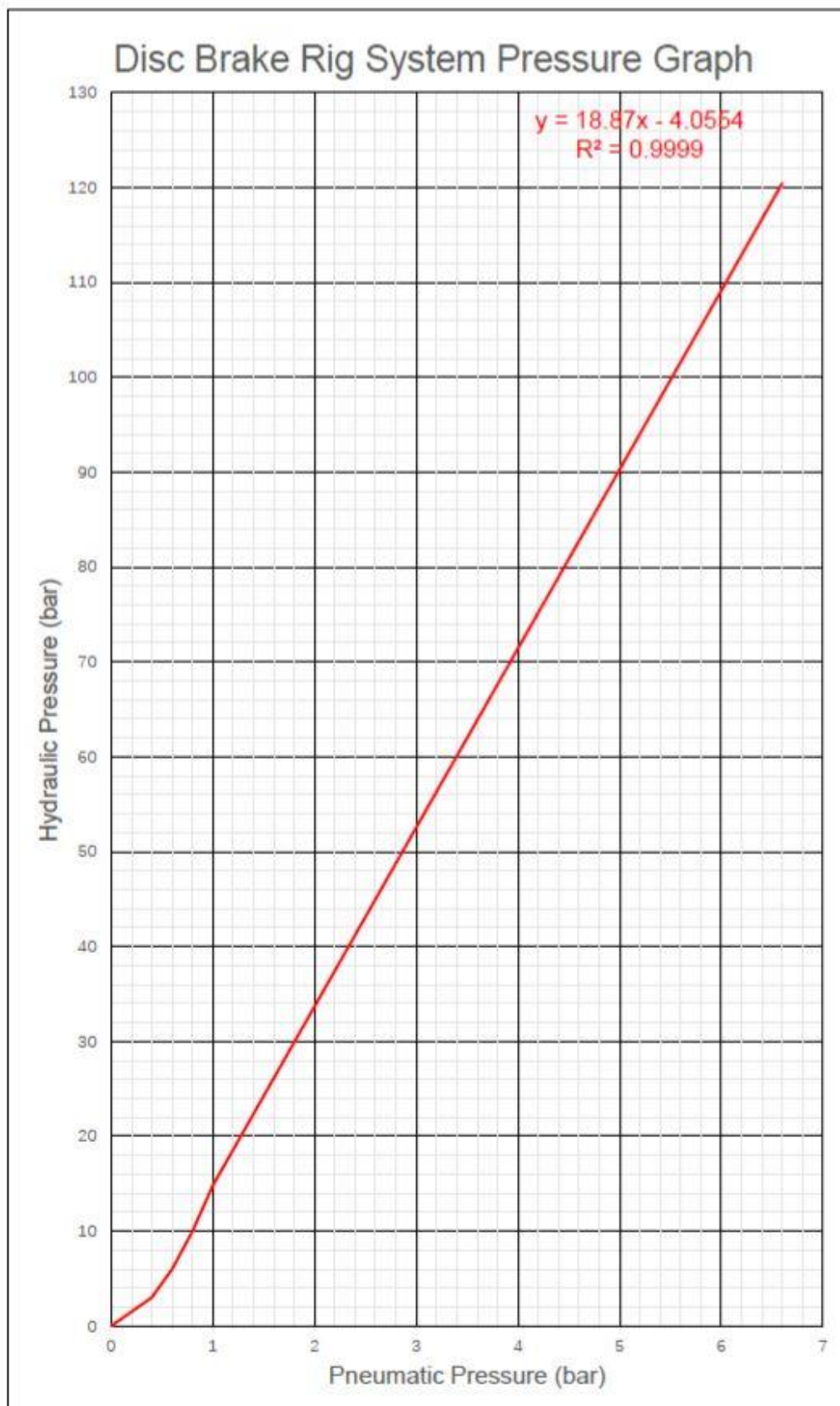


Figure 3-18: Disc Brake Rig System Pressure Graph

Figure 3-18 indicated that there were two relationships between pneumatic and hydraulic pressure, nonlinear from 0 bar to 1 bar input (pneumatic) pressure, and linear from 1 bar to 6.6 bar. The nonlinear relationship at the range between 0 bar to 1 bar was the result of the air-over-hydraulic chamber, as the spring with rubber membrane in the pneumatic cylinder requires an initial pressure (force) to be deformed. Corresponding hydraulic pressure is under 15 bar, which is related to very gentle braking and not within the scope of the experiments conducted in this research.

A linear relationship fit to the measurements between 0.8 bar and 6.6 bar yield:

$$y = 18.87x - 4.0554, R^2 = 0.9999 \quad (3-1)$$

Meanwhile, Table 3-2 also shows the relationship between the input pneumatic pressure and the brake hydraulic pressure after fitting calculations.

3.2 Brake Pad Modification

The design concept concerned the development of the means to find a way to install the optical sensors on the brake pads, and without significantly changing component characteristics and interface properties of the system. Due to the small diameter of an optical fibre, typically of the order of 0.25mm and considering the brake pad structure and stiffness, it was concluded that the creation of small grooves on the brake pad in which to place the fibres could achieve the test requirements. In view of manufacturing limitation and cost, the groove size was set to be a width of 1mm and a depth of 1mm, as illustrated in Figure 3-19 (d), such that the fibre could be installed with its surface below the brake pad surface, ensuring that the operation of the disc braking system would not be adversely influenced. Figure 3-19 shows the designed brake pad modifications on both the vertical (a & b) and horizontal (c) directions, and the completed view can be seen in Appendix Chapter 3-F, which details the locations of each groove.

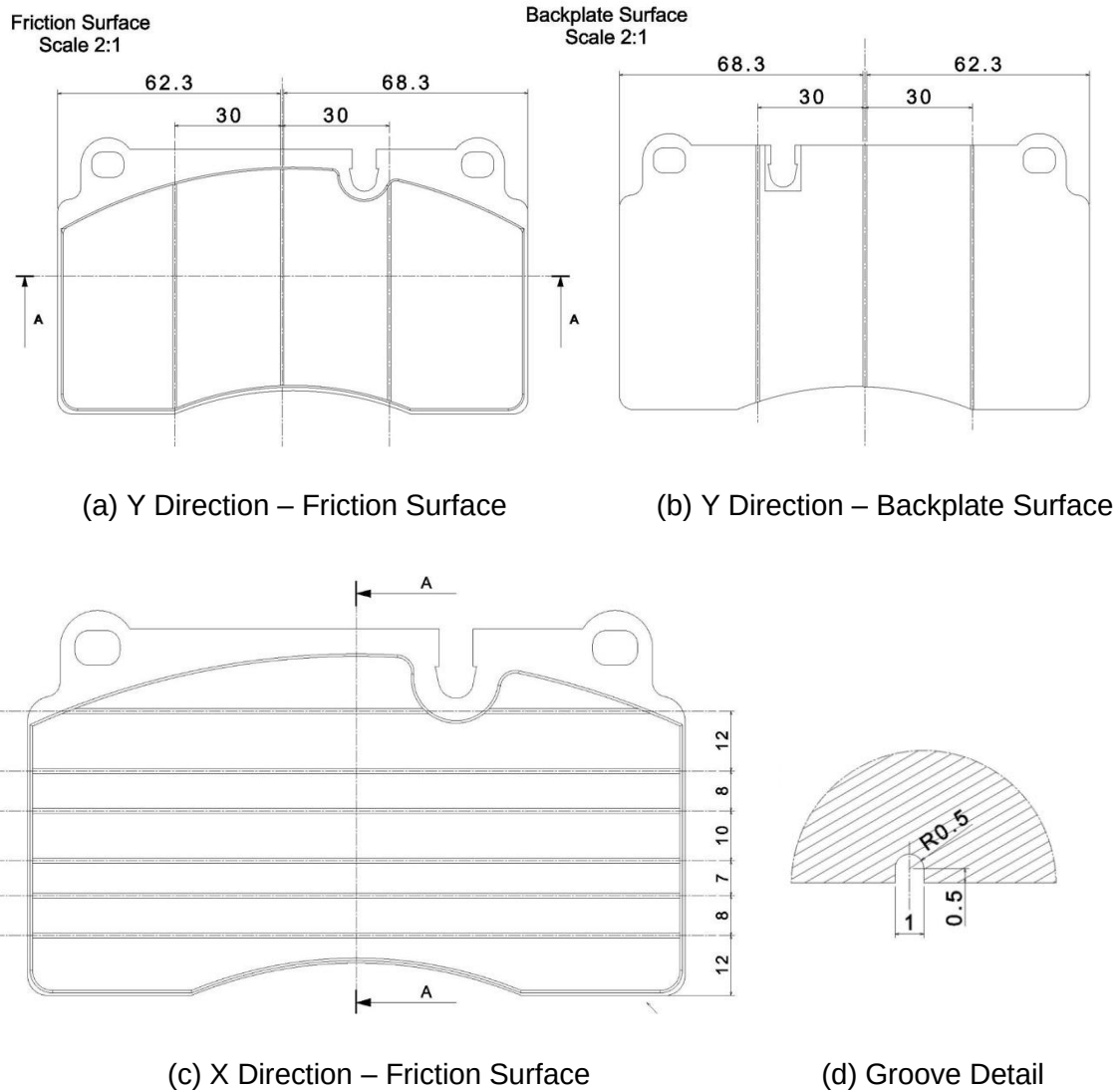


Figure 3-19: The Brake Pad Modification Concept: Groove in Y Direction (a) and (b), Groove in X Direction (c), Groove Detail (d)

According to the drawings shown in Figure 3-19, two brake pads were modified by the Cranfield University workshop using the CNC machine. The shims (NVH stickers) as well as paint were removed from the brake pad backplates and both friction and backplate surfaces were finely ground to a good finish and parallelism (Figure 3-20). These preparations were to ensure good pad/disc and pad/piston contacts under all test conditions, repeatable measurements and the easy pad installation and removal from the calliper.



(a) Y Direction – Friction Surface

(b) Y Direction – Backplate Surface



(c) X Direction – Friction Surface

Figure 3-20: Modified Brake Pads with Designed Grooves

3.3 FBG Sensors Deployment

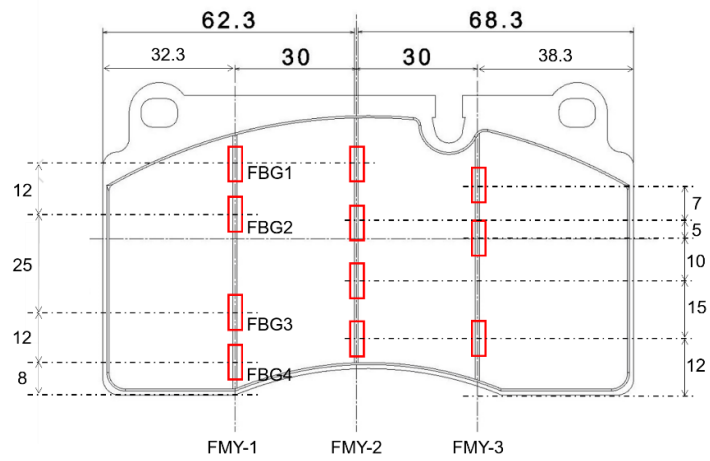
The use of FBG strain sensors reflects the innovation in the work presented in this thesis. The exploitation of the FBG technique's inherent advantages, its small size, high sensitivity, and multiplexing capabilities, coupled with the designed grooves, would allow interface strain distributions to be investigated in static, quasi-dynamic and dynamic conditions. The measurement results would be compared with corresponding FE analysis in order to deepen the current understanding of the relevant brake component loading and dynamic behaviours. In the following sections, the sensor location concept, and its fabrication and installation will be described.

3.3.1 Sensor Location Concept

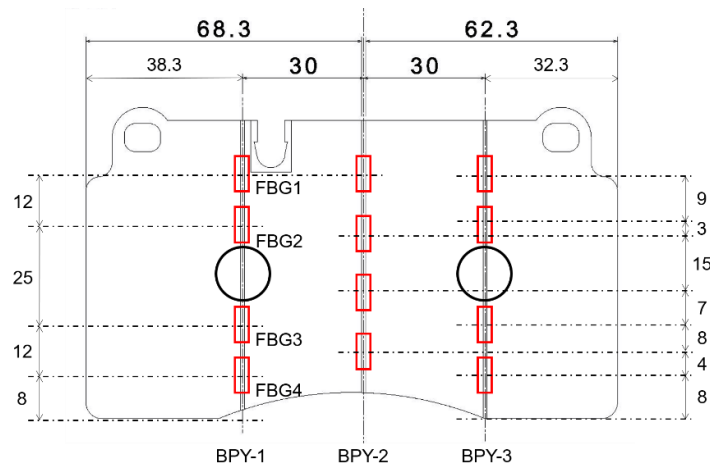
Having reasonable FBG sensor locations are necessary in order to ensure that the recorded data can provide enough strain results on the brake pad surfaces for the further analysis under different test conditions, and making comparisons with FE analysis results. The initial design concept was to use as many sensor elements as possible to obtain the comprehensive view of the disc brake performance, which includes every edge, corner and location on both the friction material and steel backplate surfaces of the brake pad. However, considering the cost of the fabrication and test time, the experimental equipment price, and the measurement and data analysis efficiency, two FBG sensor placement options have been put forward for the actual test applications, the vertical (Y) direction (Figure 3-21 (a), (b)), and the horizontal (X) direction (Figure 3-21 (c)).

The grooves in the vertical (Y) direction, on both the friction material and the backplate surfaces, were designed to each contain one optical fibre with either three or four FBG sensors. Based on the discussion in section 3.2, there were two location holes on the backplate surface (Figure 3-20 (b), Figure 3-21 (b)), which were used to couple the friction material with the steel backplate during the manufacturing process. Meanwhile, the location of both holes was the contact area of pistons when operating the brake applications. Therefore, the FBG sensors tried to avoid such area to ensure that the fibre would not be destroyed, and the recorded data could be observed during different tests.

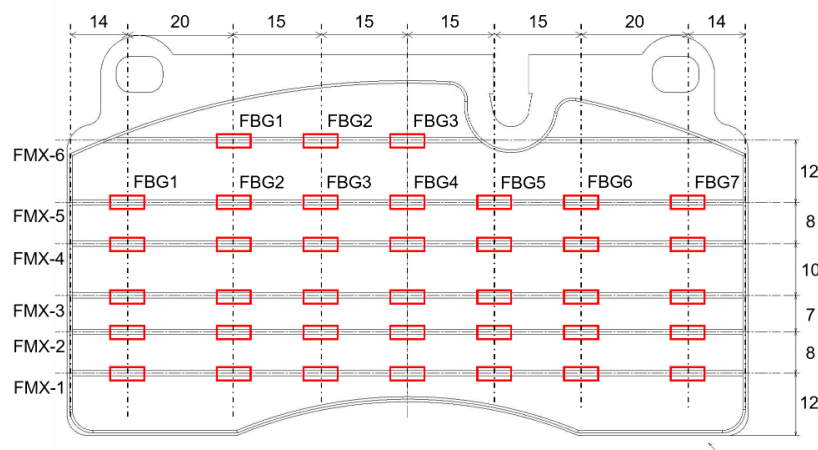
The 5 grooves oriented in the horizontal (X) direction on the friction material surface were designed to house fibres with 7 sensors. However, as a result of space limitations of the friction material imposed by the brake calliper and the brake pad assembly, one of the optical fibres was to only contain 3 sensors. The total number of FBG sensors was 38, which would ensure sufficient measurement locations to meet the initial research aims and objectives in order to measure and determine interface strain distributions and other relevant brake characteristics.



(a) The Vertical (Y) Direction at The Friction Material



(b) The Vertical (Y) Direction at The Backplate



(c) The Horizontal (X) Direction at The Friction Material

Figure 3-21: The Design Concept of FBG Sensor Locations

3.3.2 Fabrication and Installation

The fabrication of the FBG sensors was completed at the Cranfield University Engineering Photonics Centre, where the team has facilities for fabrication of custom optical fibre sensor arrays, and has extensive experience of design, build and field deployment of sensor systems. A general manufacturing process is to expose an optical fibre to an interference fringe pattern under the ultraviolet (UV) light (Figure 3-22). An excimer laser has been launched through a phase mask, and such a phase mask diffracts the incident laser into different orders, where the laser overlaps and interferes each other within the mask vicinity. The interference creates stationary, alternating zones of the high and low laser intensity. The spatial period of the interference pattern is equal to the phase-mask period. Therefore, such a FBG fabrication method will produce an interference pattern, which then projects downward through space onto the fibre, providing an evenly spaced FBG pattern into the optical fibre (Laser Focus World, 2022). Also, the polyacrylate coating, which is designed to be removed from the fibre easily and thus does not allow high fidelity strain transfer, is removed prior to the fabrication of the FBG. The coating is UV cured and thus highly absorbent of the UV, so the grating will not be formed if it is present.

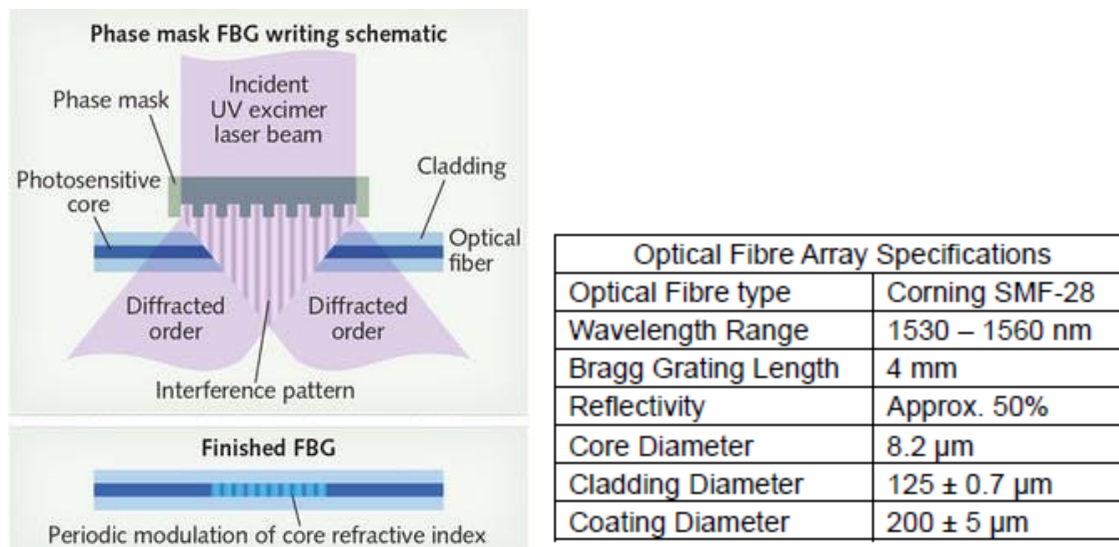
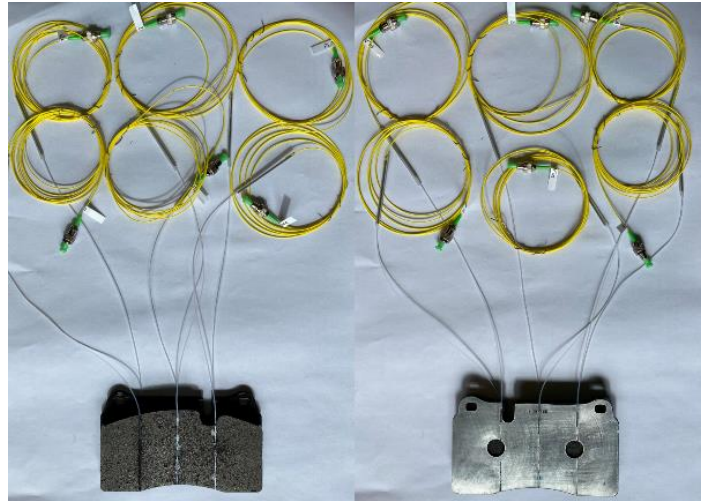


Figure 3-22: Phase Mask FBG Writing Schematic (Laser Focus World, 2022) [Left] and Optical Fibre Array Specifications (Corning Incorporate, 2022) [Right]

The optical fibre used in this research project is Corning SMF-28, which is a single-mode fibre with 125 μ m cladding diameter and 9.2 μ m mode field diameter (MFD). The fibre is used widely in telecommunications, in regional, long-haul, metro, access, mobile, and fibre to the home (FTTH) applications (Corning Incorporate, 2022). Meanwhile, the multiplexing of the FBG sensors was achieved via wavelength division multiplexing (WDM), which is a straightforward multiplexing technique, utilizing the wavelength-encoding feature of an FBG-based sensor. Each FBG sensor is encoded with a unique wavelength. The sensor arrays were interrogated using a commercially available WDM interrogator, SmartScan by SmartFibres (described in Chapter 4).

After these fabrications, the optical fibres with FBG sensors were installed on the modified brake pads (Figure 3-23). At first, a layer of cyanoacrylate was placed to seal the groove surface, and then the optical fibre was in position, after which the fibre was to be bonded to the surface using the adhesive (Permabond Adhesive 920). Prior to the fabrication of each FBG, a 5 mm length of the polyacrylate buffer coating was removed from the fibre, as the polyacrylate has a large absorption at the UV wavelengths used in the FBG fabrication. The fibre was not recoated after FBG fabrication to ensure high fidelity strain transfer when the cladding was subsequently bonded to the friction material. A length of nylon sleaving (OD=1mm) was placed over the optical fibre leading to the FBGs, terminating 1mm into the grooves in the friction material, to protect the fibre as it exited the pad (Figure 3-23 (C)). Such an adhesive has a low viscosity, and thus it was found that some residue was present at the edges of the grooves as well as on the top surface of the friction material. As a result, the brake pad was polished carefully to remove the residue, while ensuring that optical fibres would not be damaged and that the surface contact between brake disc and pads was not affected during the measurements.



(a) Installed FBG Sensors at The Vertical (Y) Direction

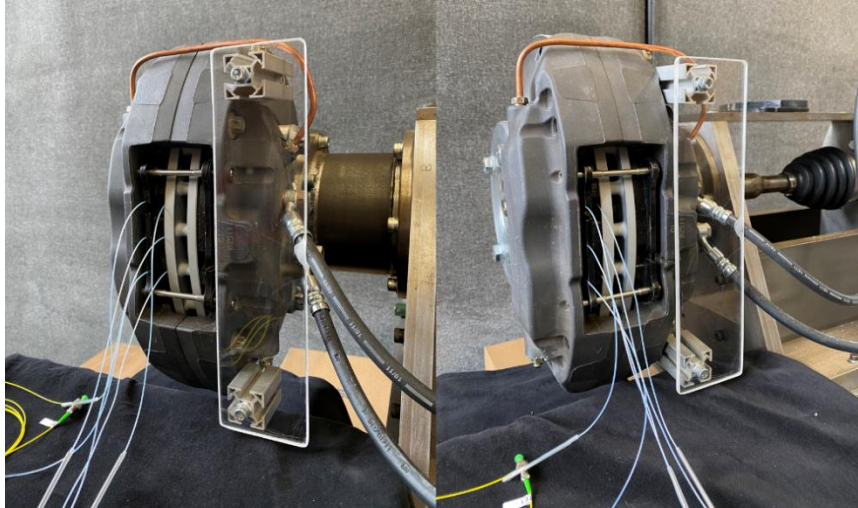


(b) Installed FBG Sensors at The Horizontal (X) Direction

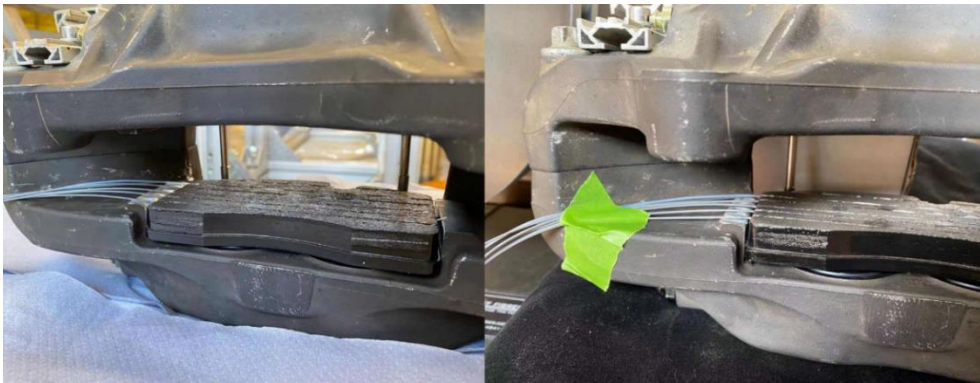


(c) The Fibre Array Detailed Diagram

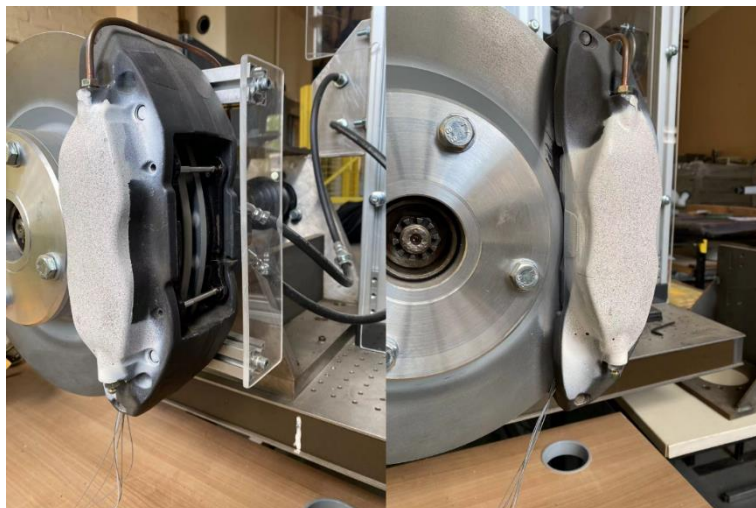
Figure 3-23: The Final Brake Pads with FBG Sensors



(a) The Vertical (Y) Direction Brake Pad Installation



(b) The Horizontal (X) Direction Brake Pad Installation (Brake Calliper)



(c) The Horizontal (X) Direction Brake Pad Installation (The Rig)

Figure 3-24: The Final Assembly of The Brake Pad Installation

Images showing the instrumented brake pad mounted in the Brembo Land Rover disc brake system are shown in Figure 3-24. The Y-direction brake pad was installed immediately after putting the brake calliper on the test rig, as the fibres were visible from the middle area of the calliper, where no obstacle was encountered in connecting these fibres with the interrogator (Figure 3-24 (a)). However, there were some challenges during the brake pad installation of the horizontal (X) direction fibres. This occurred as the X-direction fibres were extended from the edges of the brake pads to the inner wall of the brake calliper, which had been adhered to the inner wall via the tape (Figure 3-24 (b)). Therefore, the X-direction modified brake pad installation should be completed prior to the brake calliper, bolting to the test rig, so that the fibres were laid to the ground smoothly without any sharp bending (Figure 3-24 (c)).

3.4 Summary

This chapter describes the main experimental facilities, including the design, manufacture and commissioning of the test rig, modifications of the brake calliper and pads, as well as FBG sensors' fabrication and installation. The construction of the test rig enables an independent two-channel control hydraulic pressure system, with the calliper modifications enabling different pressure levels to be applied to the two sets of pistons. This can change hydraulic pressure and strain distributions along the brake pad, which should be monitored using FBG sensors.

Meanwhile, the brake pads underwent a number of modifications in preparation for the installation of the FBG sensor arrays, including the grooves created on the horizontal (X) and vertical (Y) directions, and the polished surfaces on both friction material and steel backplate. Two brake pads with different orientations of optical fibres have been fabricated and bonded steadily, each fibre on the brake pad contained reasonable amounts of FBG sensors in order to reflect the completed image of interface strain distributions on either the friction material or the steel backplate surface.

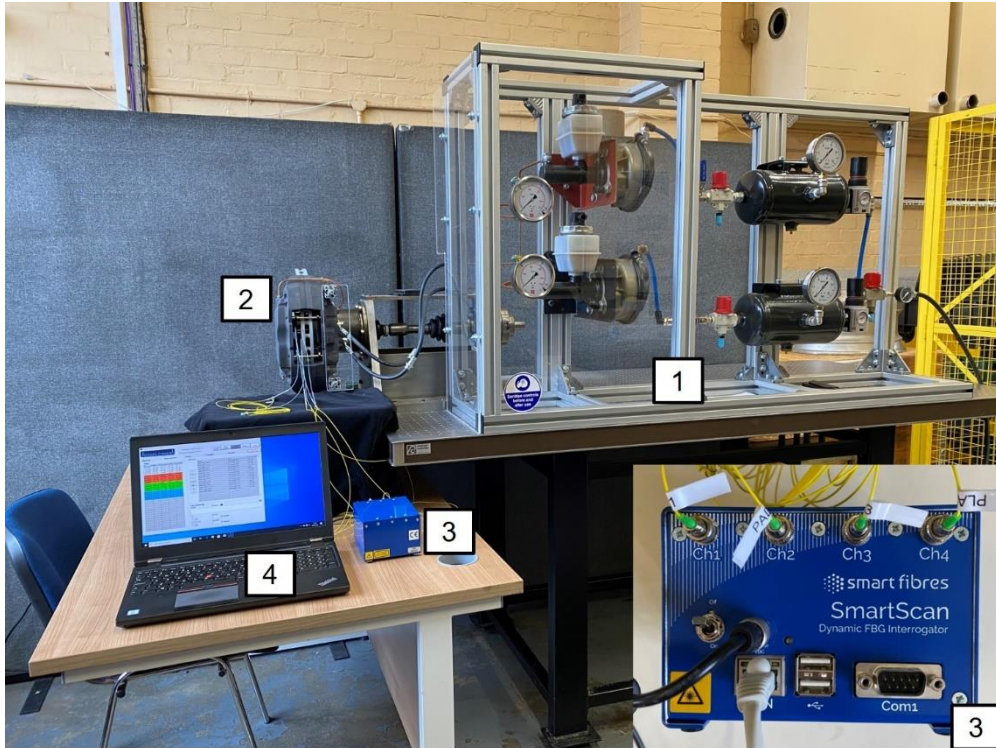
4 EXPERIMENTAL PHASE 1 – STATIC LOADING ANALYSIS

The studied static loading cases will be introduced in this chapter, which will include descriptions of the experimental set-up, and of the test conditions, along with an analysis of the test results. This chapter will provide only relevant details about the static loading tests. Comparisons with FEA will be discussed in detail in Chapter 5, following the presentation and analysis of the FE simulations.

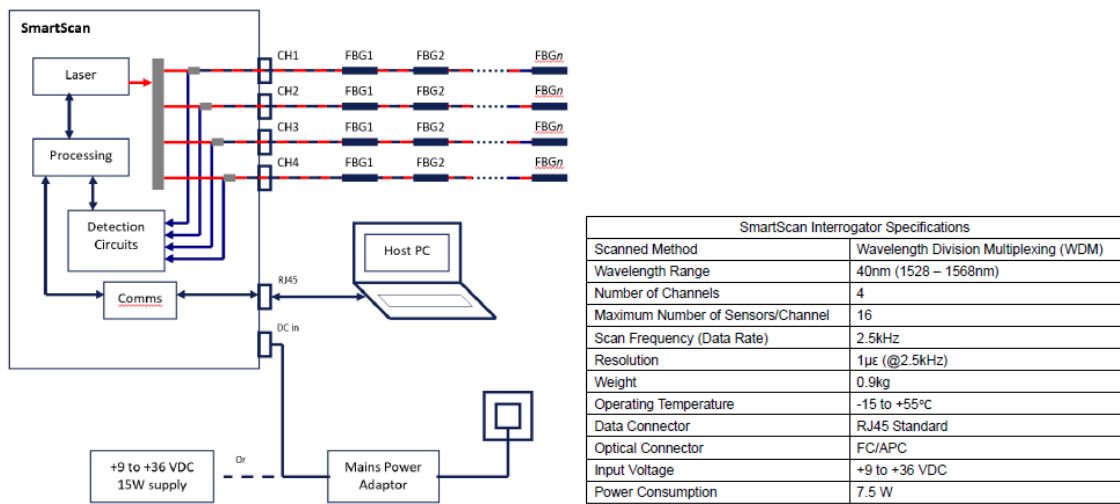
4.1 Static Loading Test System Set-up

In Chapter 3, the test rig and the FBG-instrumented brake pads were described. The test rig used for the static and quasi-dynamic loading cases comprised four main units that are illustrated in Figure 4-1; (1) the two-channel control hydraulic pressure system, designed and constructed specifically for this project (2), the modified Land Rover disc brake, (3) the SmartScan FBG interrogator, and (4) the SmartSoft application software.

The SmartSoft application software is used to configure and control the SmartScan FBG interrogator, and allows the FBG wavelengths to be acquired from the interrogator and logged in the laptop. The optical fibres containing the FBG sensors were each terminated with a fibre channel (FC) connector, which allowed the fibre connection with the four channels of the SmartScan interrogator. The interrogator was connected to the laptop, where the SmartSoft application software was installed, using the RJ45 Ethernet Lan cable (Figure 4-1 (a)). The interrogator uses a wavelength scanned laser, which scans a wavelength range of 1528 nm – 1568 nm at a repetition rate of 2.5 kHz (Figure 4-1 (b)). The reflected light is directed to a photodiode, yielding spectrum such as that shown in Figure 4-2, which is that of a length of SMF 28 optical fibre in which an array of WDM FBGS had been fabricated. The peaks correspond to the Bragg reflections from the FBGs in the arrays. The SmartScan interrogator allows changes in the Bragg wavelengths to be detected with a resolution of 1pm.

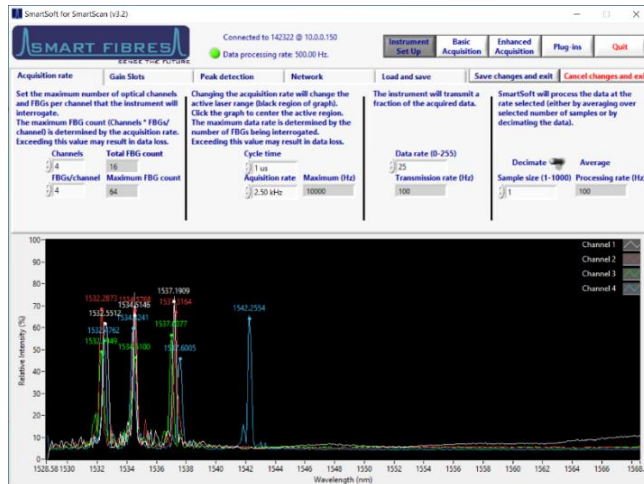


(a) Four Main Test Units: (1) the two-channel control hydraulic pressure system; (2) the modified Land Rover disc brake; (3) the SmartScan FBG interrogator; (4) the SmartSoft application software



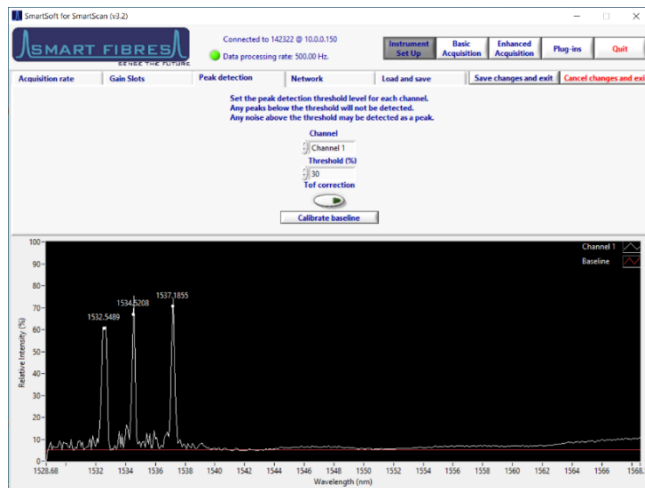
(b) SmartScan System Diagram and Specifications (Smart Fibres, 2022)

Figure 4-1: Static Loading Test System Set-up



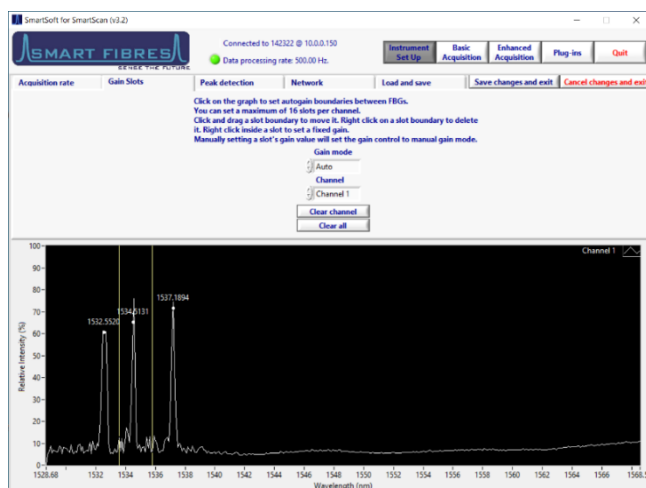
Acquisition Rate Settings	
Channels	4
FBGs/Channel	3 or 4 or 7
Cycle Time	1 μ s
Aquisition Rate	2.50 kHz
Data Rate	25 Hz
Transmission Rate	100 Hz
Sample Size	1
Processing Rate	100 Hz

(a) The Acquisition Rate Setup



Peak Detection Setup	
Threshold	15%~30%

(b) The Fibre Peak Detection Setup



Gain Slot Settings	
Gain Mode	Auto

(c) The Sensor Gain Slots Setup

Figure 4-2: General SmartSoft Instrument Set-up before Experiments

The SmartSoft instrument set-up tab was used to configure the data acquisition before starting experiments or when connecting a different sensor array to the interrogator. Figure 4-2 (a) shows the parameters that needed to be set and optimised, including the number of channels, the number of FBGs per channel, the acquisition rate, the data transmission rate, as well as the sample size and processing rate. The peak detection tab allows the setting of a threshold reflectivity level, above which the system searched for peaks in the spectrum. This ensures that pronounced sidelobes around the Bragg peaks are not misinterpreted as separate Bragg reflections.

The SmartScan system offers a number of features to take into account variability in the FBG reflection spectrum, which may arise from the fabrication of the sensors, or may occur during the measurements. Each optical fibre used in experiments described in this thesis contained several WDM sensors. In the default setting, for each channel, the system determines the central wavelengths of the Bragg peaks lying above a threshold set in the “peak detection” tab (Figure 4-2 (b)) and returns them in an array for each time sample, numbered from the longest to the shortest wavelength. This approach exhibits a number of challenges in practical applications. The non-uniform loading of an FBG in the array could result in, for example, the imposition of chirp on the grating period of FBG_i , broadening the peak and lowering the peak reflectivity below the threshold. This causes the wavelengths of FBG_{i+1} and subsequent FBGs to be displaced within the returned array, which complicates subsequent processing. Loading the FBG results in a splitting of the peak, perhaps due to the introduction of birefringence via transverse loading, or a discrete change in the strain transfer that results in the introduction of a phase shift in the grating period, introduces an extra peak into the spectrum, again disrupting the order of the wavelengths returned, and causing the wavelength of the final FBG in the array to be absent from the array as the number of peaks exceeds the set number of FBGs/channel. The software allows the setting of gain boundaries with-in the optical spectrum so that each sensor is assigned a specific region of the spectrum, avoiding some of these issues by ensuring that, should the reflected signal from an individual sensor fall below the set threshold, or an extra peak appear, the logging data

from the remaining sensors would not be confused (Figure 4-2 (c)). Additionally, the gain within each of these sections may be set independently, which can help when the reflectivity of the FBGS within the array differ significantly. The gain can be set as a fixed value, or can be automatically set by the software depending on the signal strength within each individual gain boundary. While the issues discussed above can still have an adverse influence on the data, changes in the spectrum of one FBG will not affect the logging of the Bragg wavelengths of other elements of the FBG array. The width of the gain boundaries should be set such that the anticipated changes in Bragg wavelengths arising from the loading of the object will not cause the wavelength to move beyond the boundaries.

4.2 Fibre Bragg Grating (FBG) Test Plan and Validation

Following the preparation steps described in section 4.1, the characterisation of the FBG-instrumented brake pads was undertaken under the conditions described in the following paragraphs. In order to ensure that each test was performed under similar environmental conditions, the lab temperature and humidity were recorded, and the SmartScan FBG interrogator was allowed to warm up for about 30 minutes before starting the experiments to ensure that the laser wavelength scan was stabilised.

4.2.1 Main Test Conditions

Several test conditions were considered for the static loading cases, including different pressure levels and controls. The test under each condition was repeated three times as to allow the repeatability to be assessed, and to allow the test results to be averaged, as well as to provide sufficient data for the subsequent analysis step. A complete static loading test plan is provided in Table 4-1.

Table 4-1: FBG Static Loading Test Plan

FBGs Test Plan – Static Loading Conditions (20s for each step)		
Hydraulic Pressure Range	Nominal Pressure Variations	Optical Fibre Direction
0bar – 120bar – 0bar (10bar Increment or Decrement)	Equal Pressure ($p_1 = p_2$) +10bar ($p_1 = p_2 + 10\text{bar}$) -10bar ($p_1 = p_2 - 10\text{bar}$)	Vertical Direction (Y) Horizontal Direction (X)

The hydraulic pressure was varied from 0 bar to 120 bar and returned to 0 bar, in steps of 10 bar, with the pressure held at each value for 20 seconds. Due to the specific design and modifications made to the brake calliper, described in section 3.1.2.1, the nominal pressure variations could be divided into three test series, containing equal pressure ($p_1 = p_2$), +10bar ($p_1 = p_2 + 10\text{bar}$) and -10bar ($p_1 = p_2 - 10\text{bar}$), where p_1 was the pressure applied to piston 1, and p_2 was the pressure applied to piston 2, to allow the investigation of the changes in the brake pad/disc interface distributions on the measured strain levels.

The 10bar hydraulic pressure difference between the pistons was selected as FBG sensors have sufficient sensitivity to allow determination of the changes in the micro strain ($\mu\epsilon$) level under these conditions, where the following sections showed that a 10bar pressure change could be observed and reflected in a measurable strain change. In addition, the brake pad itself might experience significant bending if the applied pressure difference for both pairs of pistons was too large. The test rig's hydraulic control system offered sufficient control to provide a 10bar difference, due to the characteristics of the pressure gauges employed.

As discussed in section 3.2, brake pads were prepared with slots on both the frictional surface and backplate, which allowed the FBGs to oriented in either the horizontal (X) or vertical (Y) direction. The test schedule had to take into account the limitation that the available interrogator could only monitor four FBG arrays simultaneously, where each array could contain up to 16 sensors. In the vertical (Y) direction, three fibres were attached to the friction material surface, and

another three to the backplate. Therefore, to characterise the pressure induced strains in the Y direction, the measurements at each pressure condition were undertaken with different combinations of FBG sensor arrays connected to the interrogator. The three fibres on the surface of interest (backplate or friction material), and the fibre attached in the middle slot of the opposite surface would be connected to the interrogator. The monitoring of the fibre on the opposite surface provided a benchmark to ensure that the two surfaces were characterised under the same test conditions, assessed by the repeatability of the central FBG array readings. In the horizontal (X) direction, six FBG arrays were attached to the friction surface, so that each experiment would use four of the fibres, with two of these, usually the centrally mounted fibres, always connected to the interrogator, acting as the benchmark to ensure that the measurements and thus conditions were repeatable.

4.2.2 Additional Test Conditions

Section 4.2.1 detailed the basic process used in the step-by-step static loading tests. The following section describes other experimental considerations, which are divided into three aspects, repeatability, visco-elastic behaviour, and inboard and outboard comparisons.

4.2.2.1 Repeatability

To assess repeatability, the measurements were taken by the same person using the same equipment, under the same test conditions, and within a short period of time. Comparing the test results with the average result can allow assessment of the percentage error to judge if it is suitable for analysis or discussion. Here, the tests were all undertaken under similar conditions, with the test temperature maintained constant to within 0.5°C, and humidity to within 0.5% difference. The test series was completed within two hours. The test conditions were designed to be as simple as possible to reduce unnecessary influences, and considered only the case in which the pressure was applied equally to two pistons from the range 0bar-120bar-0bar, monitoring only the vertical fibres bonded to the friction material.

4.2.2.2 Visco-elastic Behaviour

Visco-elastic behaviour is a material property observed when a material that exhibits both viscous and elastic characteristics undergoes deformation (Choi et al., 2008). The friction material on a brake pad has such characteristics and might be observed within the measurements when the hydraulic pressure was rapidly increased or decreased. A four-pressure-level series of test was performed, comprising pressures of 25 bar, 50 bar, 75 bar and 100 bar applied equally to both pairs of pistons instantaneously when opening the pressure regulator. The pressure level was maintained for 10 minutes and then released, and the response monitored for a further 10 minutes. This was repeated three times for each pressure level. It is considered that these four pressure levels could provide sufficient information to investigate the influence of the friction material's visco-elastic phenomenon during the experiments.

4.2.2.3 Inboard and Outboard Comparisons

Generally, all static loading tests were completed with the instrumented pad mounted on the outboard side of the disc brake system. This was because the pad with the horizontally (X) mounted fibres could only be installed and tested when located on the outboard side due to the limited space available to allow routing of the fibres out of the brake calliper (Figure 4-3) to the interrogator. However, the fibres mounted in the vertical (Y) direction could be tested on both the outboard and the inboard sides of the disc brake system, so that the static loading cases were repeated with the pad mounted on the inboard side. These tests were used to assess the differences between the pads mounted on either side of the disc, to discover if the side on which the pad was mounted could influence the test results and the disc brake performance.

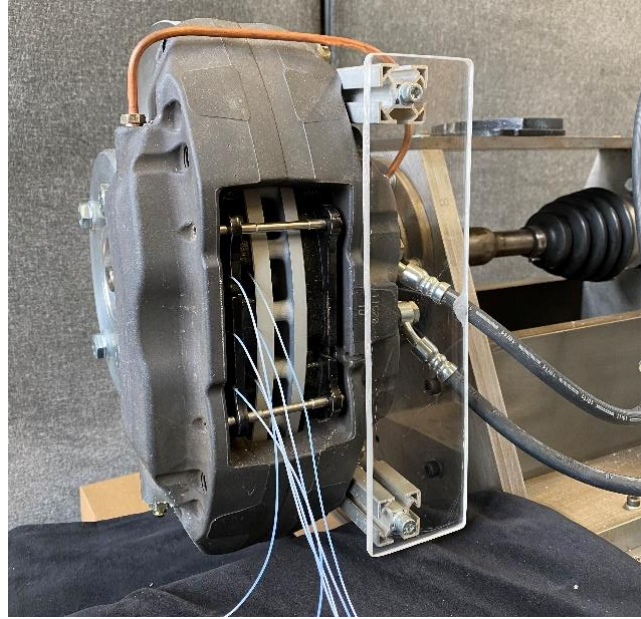


Figure 4-3: Y-Direction Brake Pad Installation at The Outboard Side

4.3 Static Loading Test Results and Analysis

4.3.1 Data Processing

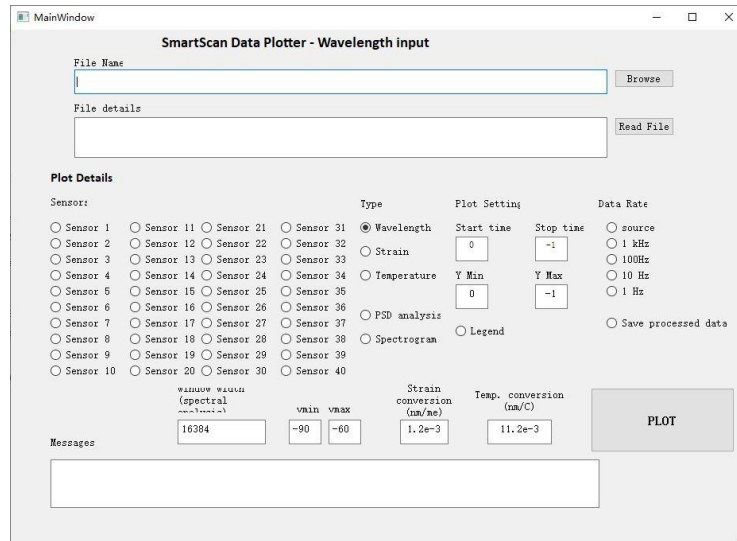
The SmartSoft software was configured to record the measured data at a rate of 100 Hz, providing a large amount of data to be stored and processed for further analysis and comparisons. Generally, the SmartScan was used to record the change of the Bragg wavelengths of the connected sensors during each experiment.

A program written in the Python language (Figure 4-4 (a)) by Prof. Stephen James was used to convert the wavelength data into strain:

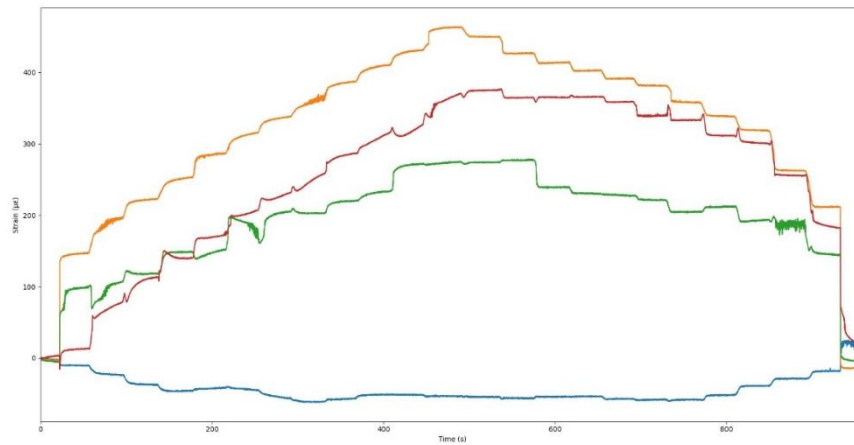
$$Strain (\mu\epsilon) = \frac{Test\ Wavelength\ (nm) - Initial\ Wavelength\ (nm)}{Strain\ Sensitivity\ (nm/\mu\epsilon)} \quad (4-1)$$

Where *Strain Sensitivity* = 0.0012 nm/ $\mu\epsilon$ (Othonos & Kalli, 1999).

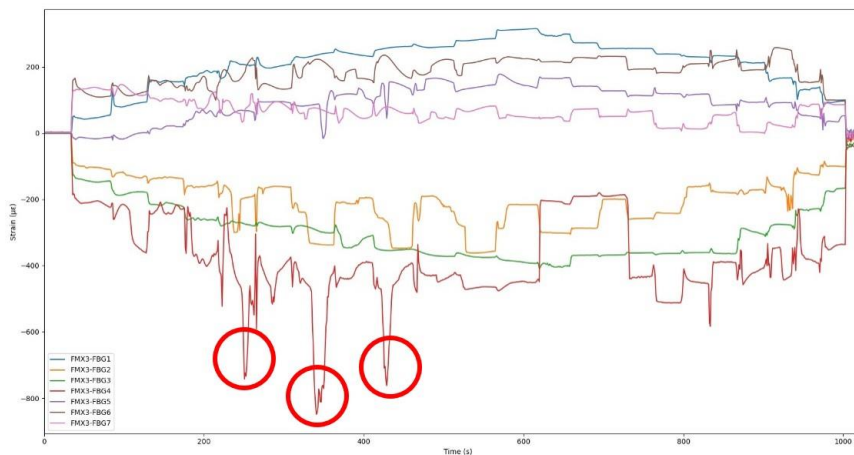
The strain data was then stored in a csv file for subsequent processing and analysis. The programme also allowed the time series of the recorded data to be plotted to check the data quality (Figure 4-4 (b)), as there were some issues that arose occasionally during the test, for example, losing sensor signals due to the reflected power dropping below the set threshold (Figure 4-4 (c)).



(a) The Python Program Window with Different Parameters



(b) The Sample Recording Graph



(c) The Sample Data Recorded Issue

Figure 4-4: The Python Program Screenshots

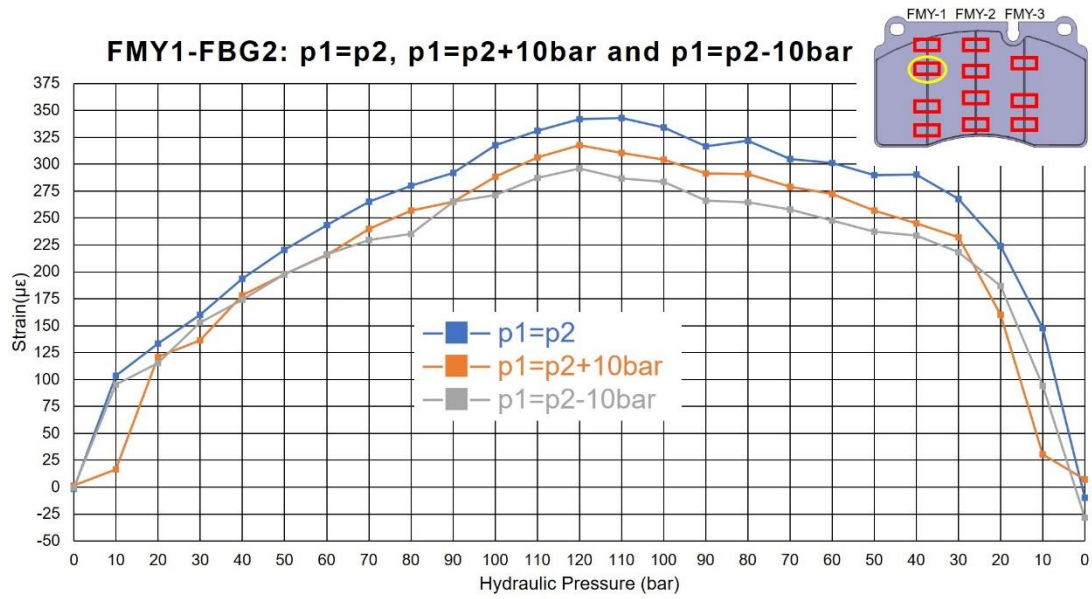
To process the data a MATLAB programme has been created (Appendix Chapter 4-A). As each hydraulic pressure level test step was maintained for about 20 seconds, corresponding to 2000 data points, the data was averaged over each step and saved as an Excel file for plotting graphs and making comparisons. As each test condition has been repeated three times, the test results discussed in following sections are the results of the data from those three tests.

4.3.2 Results and Analysis: Recording and Distribution Graphs

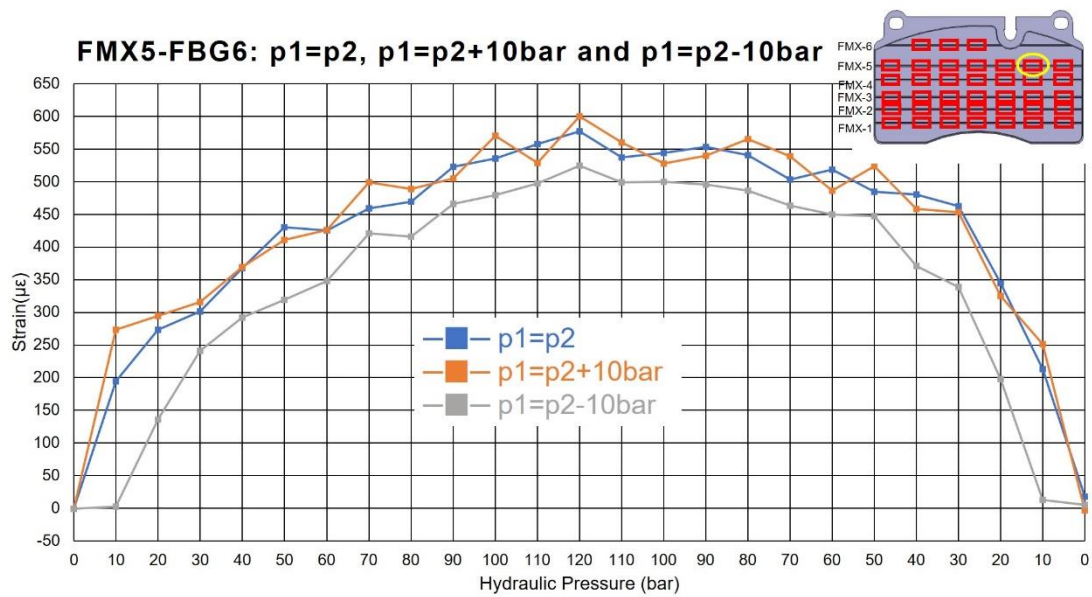
According to the data processing step, the initial test results were visualised using two approaches: the test recording graph and the strain distribution graph.

To assess the response of each sensor to three static loading conditions, equal pressure, +10bar and -10bar respectively, recording graphs such as those shown in Figure 4-5 (a) and Figure 4-5 (b) were plotted. The icon at the top right allows identification of the location of the sensor on the brake pad and indicates its orientation. The graph shows the measured strain for each of the pressure steps for the three test configurations, with the data being the average of three measurements. The oscillations visible in these figures might be caused by measurement variations, where a detailed uncertainty analysis and discussion will be provided in section 4.3.4.

The orange and grey lines in Figure 4-5 (a) and Figure 4-5 (b), corresponding to measurements where there was a 10bar difference between the pressures applied to the two pairs of pistons, demonstrate that the sensitivity achieved by the optical fibre strain sensors, especially at the beginning and the end of tests, was sufficient to allow assessment of the different pressure variations. Although the measured strain for three pressure variations under the same baseline pressure was different, the total strain has not significantly changed, which the maximum difference was under 100 $\mu\epsilon$. The measurements at lower-pressure levels (0bar-20bar) reflect influences on the data value arising from the pressure levels at the beginning of the braking phase. Such pressures corresponding to the light braking actions, are likely to be used when driving slowly in busy traffic conditions (<10km/h), or moving at a car park (<5km/h).

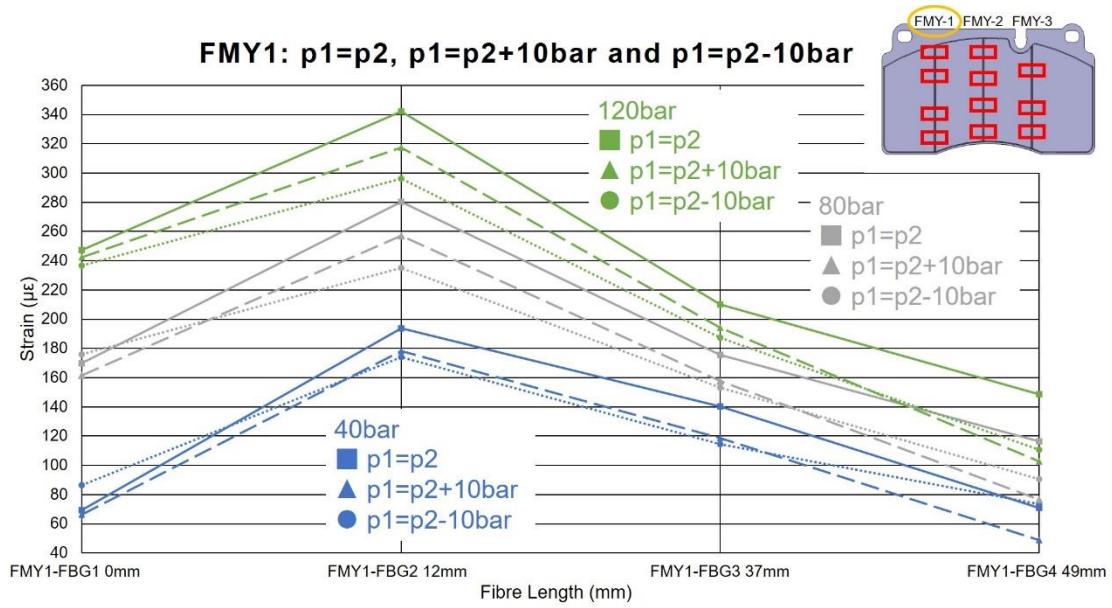


(a) Vertical (Y) Direction: Hydraulic Pressure – Strain (FMY1-FBG2)

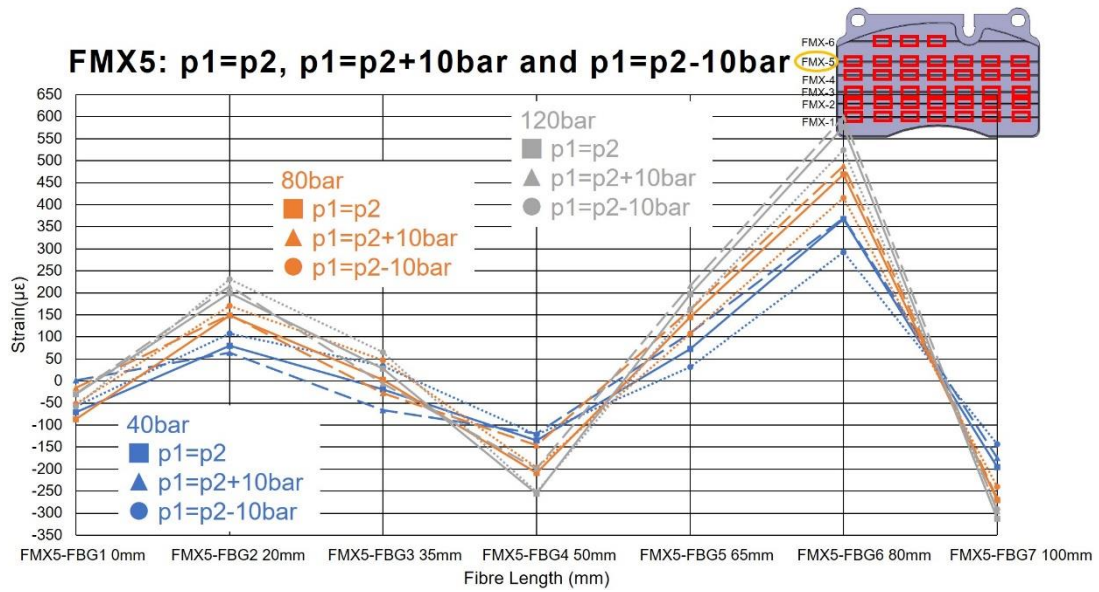


(b) Horizontal (X) Direction: Hydraulic Pressure – Strain (FMX5-FBG6)

Figure 4-5: Recording Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface



(a) Vertical (Y) Direction: Fibre Length — Strain (FMY1)



(b) Horizontal (X) Direction: Fibre Length — Strain (FMX5)

Figure 4-6: Strain Distribution Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface

An example of the visualisation of the strain distributions along a single array of sensors is shown in Figure 4-6 (a) for FMY1, oriented in the vertical (Y) direction, and Figure 4-6 (b) for FMX5, oriented in the horizontal (X) direction. Here, the icons in the top right indicates the FBG sensor array under considerations. The plots allow the strain distributions for the three pressure variations (equal

pressure, +10bar and -10bar) at three pressure levels (40 bar, 80 bar and 120 bar) to be compared. In Figure 4-6 (a), the optical fibre mounted in the vertical direction on the friction surface shows clearly the dependence of the strain on the pressure level. The second sensor (FMY1-FBG2) in the array exhibits the highest strain value, as the sensor location is close to the piston area. The higher overall pressure level is the cause of the large strain differences between the three pressure variations, with small differences observed at the lower pressure level (40 bar).

For the array oriented in the horizontal direction, the results shown in Figure 4-6 (b) were obtained under the same test condition at three sensors on FMX5 (FBG2, FBG4 and FBG6) showing different trends, which could be analysed to establish what happened at these sensor positions. The second (FMX5-FBG2) and sixth (FMX5-FBG6) sensors were both located near a piston, and the strain levels were always positive. The sixth (FMX5-FBG6) sensor exhibited the highest strain level, as the pistons are not symmetrically positioned, having a 3mm offset from the central line to the right side of the pad. The fourth (FMX5-FBG4) sensor exhibited negative strain values for every pressure level, likely caused by the squashing phenomenon of the brake pad due to the changes of the applied pressure levels and variations. The sensors adjacent to the second (FMX5-FBG2) and sixth (FMX5-FBG6) sensors also show the same tendency under these circumstances, where the higher the strain level on these specific sensors, the lower the strain level measured by the surrounding sensors. In particular, the fourth sensor (FBG4) on each fibre, located at the middle of the brake pad, experienced negative strain due to the pairs of piston reflections. Thus, the strain of the brake pad could keep balance, and exhibited the friction material properties through these tests as well.

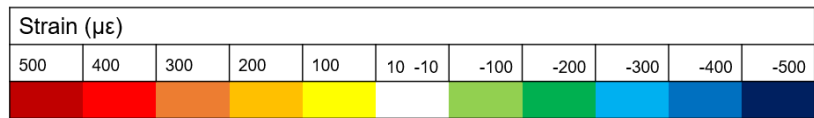
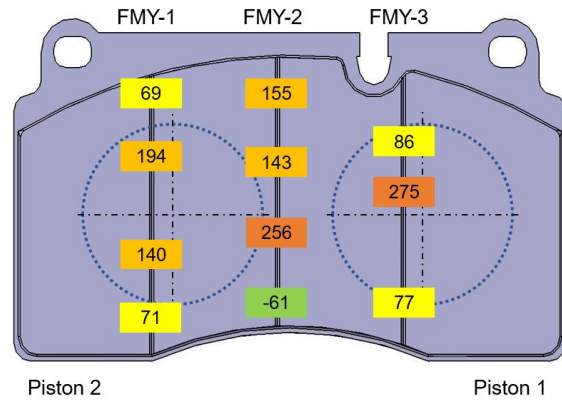
4.3.3 Results and Analysis: FBG Sensor Contour Map

The line graph is a common way to plot test results. However, the large number of sensors, 11 for the vertical (Y) direction and 38 for the horizontal (X) direction means that only a limited part of the data could be clearly visualised in this way. Therefore, an innovative visualisation approach has been employed to explain test results more easily and clearly, termed the FBG sensor contour map.

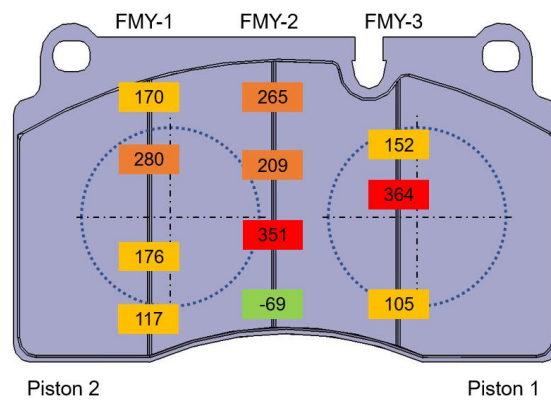
The contour maps allow the display of the data from all sensors under the same test pressure level and over the same time period, easing the comparison and analysis steps and clarifying the discussion. Generally, following comparisons are divided into two groups for the vertical (Y) and horizontal (X) directions, including three pressure levels with equal pressure, and three pressure variations with the same pressure level.

As an example, the data for three pressure levels (40 bar, 80 bar and 120 bar) with the same pressure variation, equal pressure ($p_1 = p_2$) applied to the pairs of pistons, are shown in Figure 4-7 for the vertical (Y) direction case and in Figure 4-8 for the horizontal (X) direction case. The spectrum colour map has been defined for every $100\mu\epsilon$, and all contour maps would continue to use the same definition throughout the thesis.

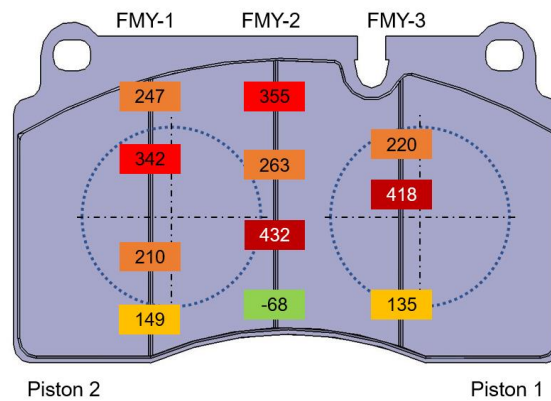
Figure 4-7 and Figure 4-8 show the FBG strain data obtained when measuring different pressure levels with equal pressure variation. Three pressure levels have been selected to represent typical driving environments: 40 bar is a normal pressure level in daily driving, 80 bar corresponds to heavy braking when driving in the country-side or on a high-speed road, while 120 bar is a rarely encountered level, which might occur during an emergency stop.



(a) 40bar Hydraulic Pressure



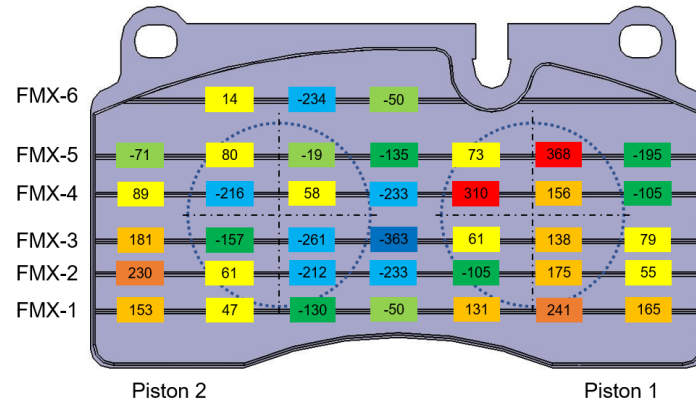
(b) 80bar Hydraulic Pressure



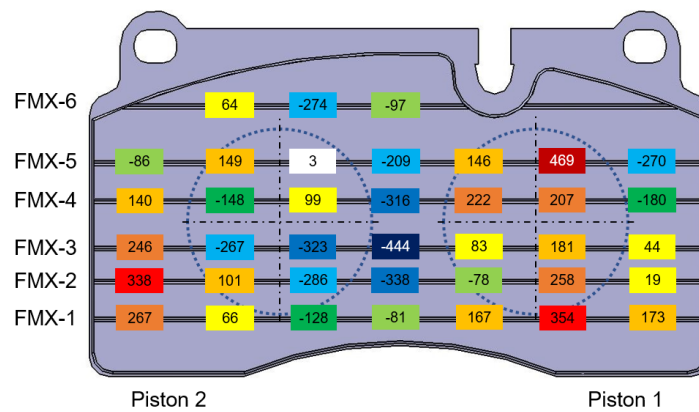
(c) 120bar Hydraulic Pressure

Figure 4-7: FBG Sensor Contour Maps for the Vertical (Y) Direction Comparison between Three Pressure Levels with Equal Pressure ($p_1 = p_2$)

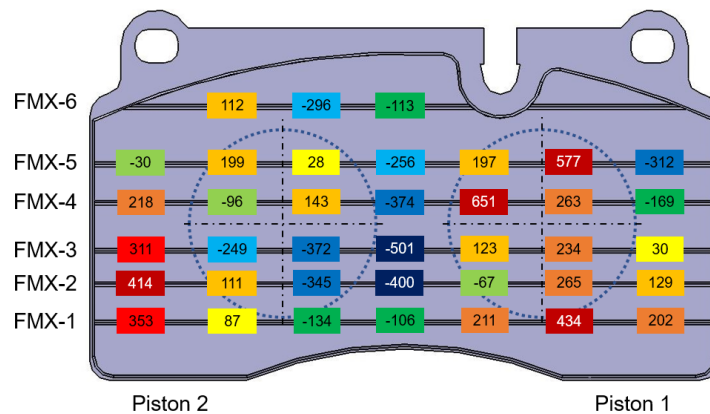
Strain ($\mu\epsilon$)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) 40bar Hydraulic Pressure



(b) 80bar Hydraulic Pressure

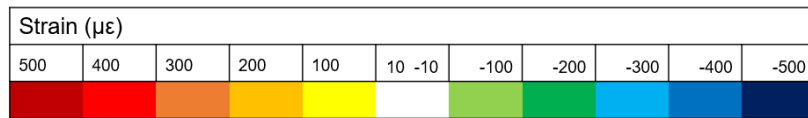
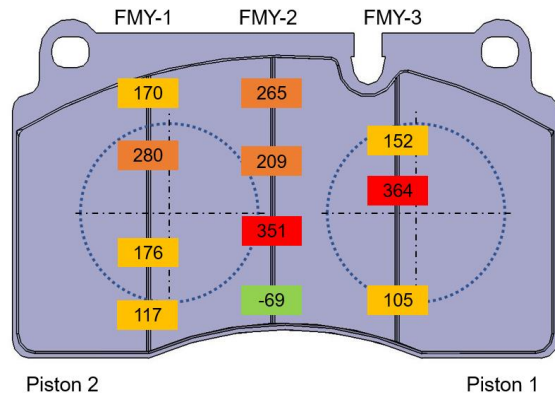


(c) 120bar Hydraulic Pressure

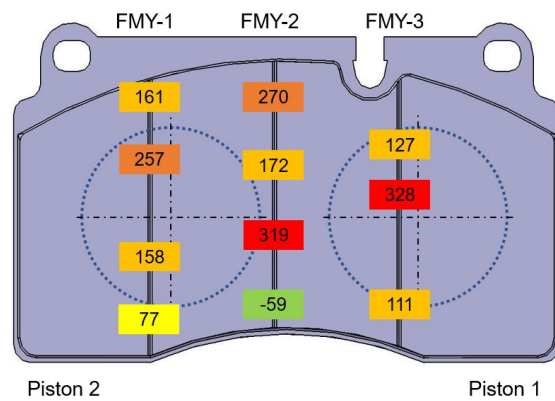
Figure 4-8: FBG Sensor Contour Maps for the Horizontal (X) Direction Comparison between Three Pressure Levels with Equal Pressure ($p_1 = p_2$)

For both the vertical and horizontal directions, in general the strain levels increase when the applied pressure increases. However, the middle sensor at the bottom of the pad in the vertical case exhibits negative strain levels, which might be caused by the brake pad lifting up or by the squashing of the friction material. At the same time, the middle region of the pad in the horizontal case shows negative strain levels, which can be explained in the way similar to that described in the previous sentence. It is noted that the third (FBG3) and fourth (FBG4) sensors in each fibre, in general, are affected by the brake pad squashing.

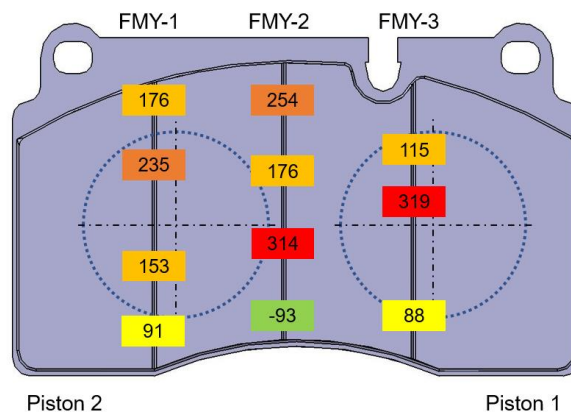
Another comparison was undertaken for three hydraulic pressure variations (equal pressure, +10bar and -10bar) with the same pressure level (80 bar). The vertical (Y) direction data is shown in Figure 4-9 and the horizontal (X) direction is shown in Figure 4-10. Other pressure level comparisons are presented in Appendix Chapter 4-B and Appendix Chapter 4-C. Generally, the vertical data does not display significant strain differences (under $50\mu\epsilon$) in response to the three pressure variations. The horizontal data shows a clear movement in the strain level between the left- and right-hand regions of the pad when activating different pressure variations under the same pressure level. Data presented in this way will be used to compare with the output from FE model simulations, and will be the basis of the discussion in Chapter 5.



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)



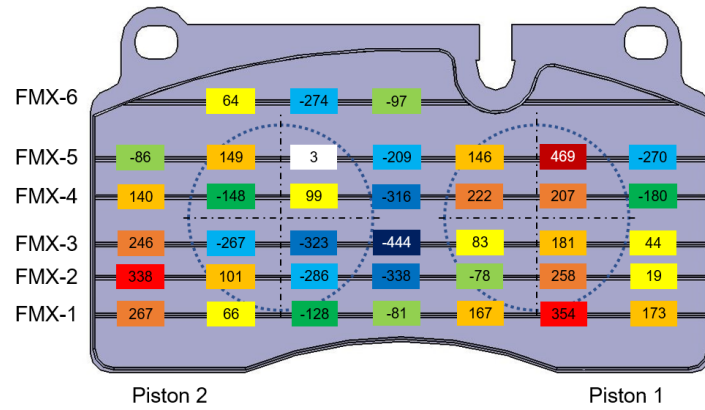
(b) +10bar Variation ($p_1=80\text{bar}; p_2=70\text{bar}$)



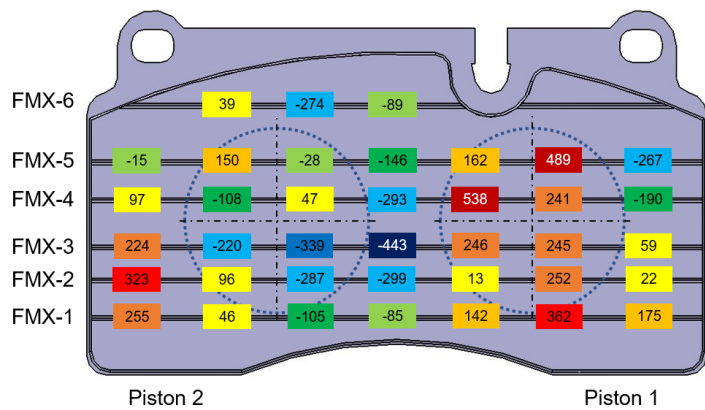
(c) -10bar Variation ($p_1=70\text{bar}; p_2=80\text{bar}$)

Figure 4-9: FBG Sensor Contour Maps for the Vertical (Y) Direction Comparison between Three Pressure Variations

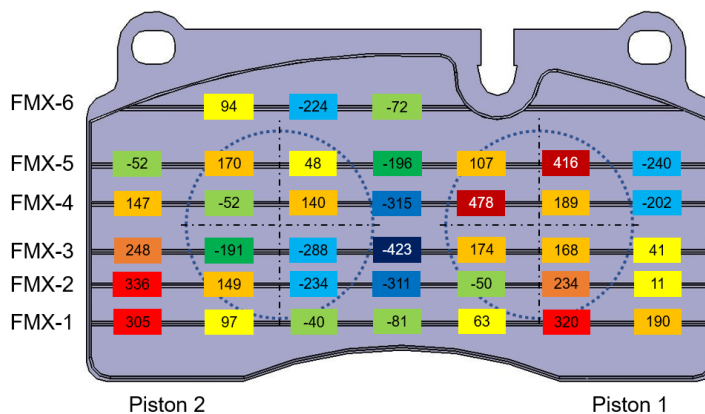
Strain ($\mu\epsilon$)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)



(b) +10bar Variation ($p_1=80\text{bar}$; $p_2=70\text{bar}$)



(c) -10bar Variation ($p_1=70\text{bar}$; $p_2=80\text{bar}$)

Figure 4-10: FBG Sensor Contour Map for the Horizontal (X) Direction Comparison between Three Pressure Variations

In the vertical (Y) case, sensors around the piston area show lower strain than sensors covered by the piston area, which is probably caused by the use of solid pistons in those tests. The edge of the pistons might not transfer all hydraulic pressure from the pad backplate to the friction surface and such phenomenon is also apparent in the horizontal (X) case. However, the horizontal (X) data might be influenced by other factors, in particular the optical fibre installation, where bending and micro-bend losses could reduce the signal returned to the detector (the SmartScan interrogator).

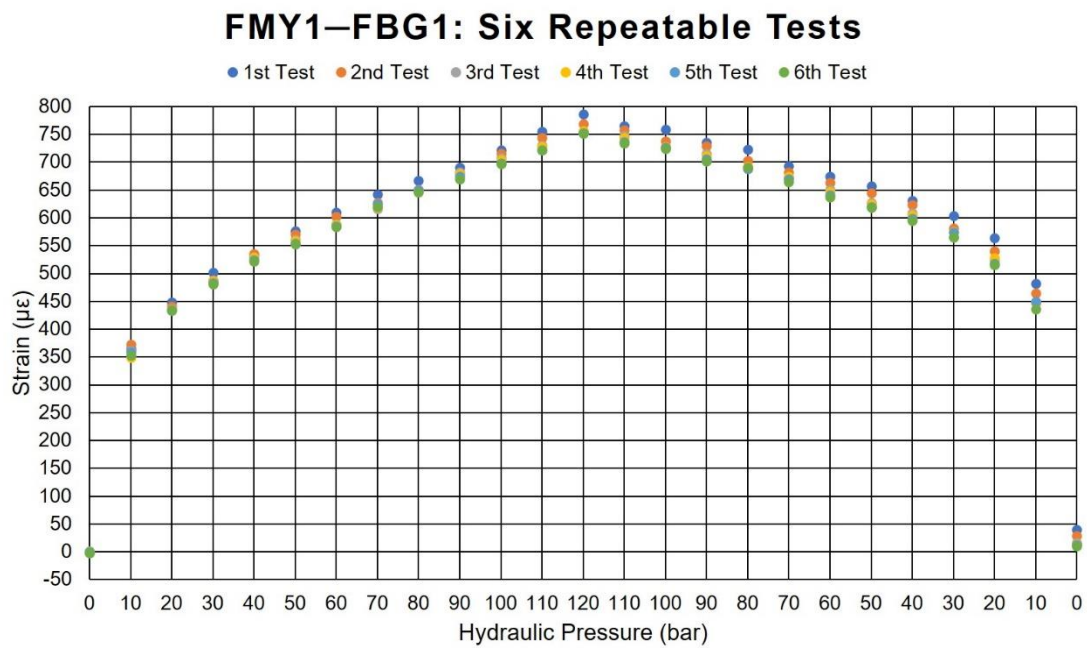
Meanwhile, the fibre installation could also provide unexpected results, as those optical fibres are bonded in grooves in the brake pad that were formed using a CNC machine. There might be loose fragments remaining in the grooves, so that the sensor bonding in these regions could be unreliable, compromising the strain transfer and the measurements. Such effects would be expected to be more apparent at high pressure levels and further discussions will be provided when the quasi-dynamic data is analysed where the pad was exposed to high pressure levels and a large applied torque.

4.3.4 Uncertainty Analysis: Experimental Phase

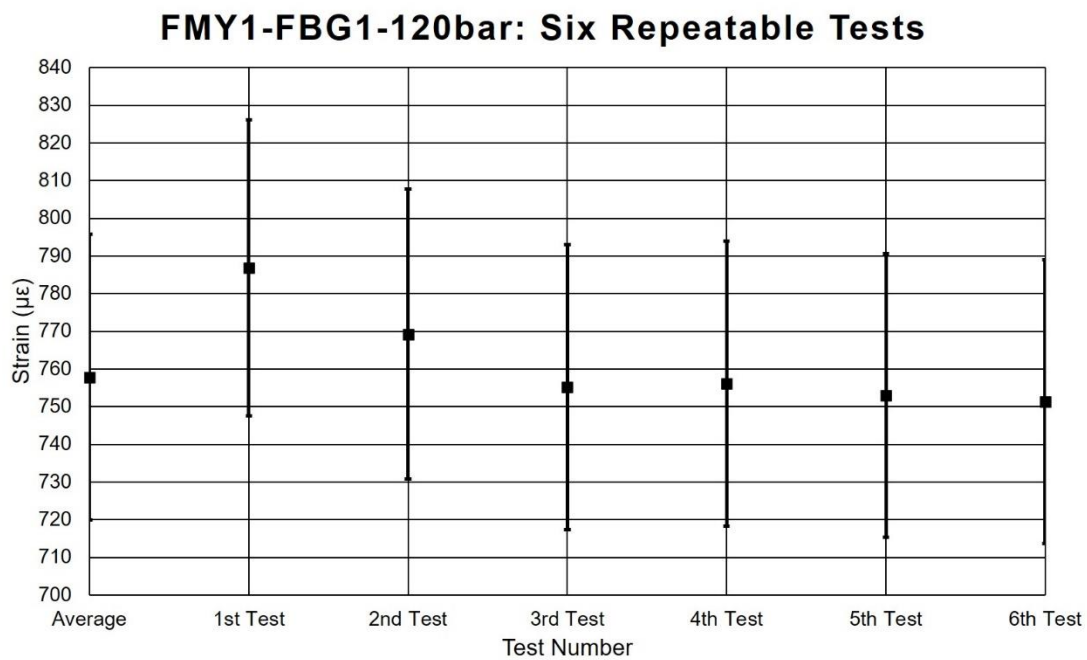
The uncertainty analysis for the static loading analysis will be important to the discussion, which includes repeatability, visco-elastic behaviour, inboard and outboard comparisons, and other possible factors. Some of those experimental uncertainties cannot be avoided, but it is essential to discuss these issues in detail to ensure the reliability of the test results.

4.3.4.1 Repeatability

The repeatability was mentioned in section 4.2.2.1 and here is based on the data obtained from a repeated series of 6 tests under the same test conditions, namely equal pressure on both pistons in the vertical (Y) direction. All six tests were completed in about three hours, each individual test taking about 30 minutes, and in the same test environment; the temperature of the laboratory varied by 0.5°C, between 16.9°C and 17.4°C, and the humidity varied by only 0.50%, from 51.00% to 51.50% (Figure 4-11).



(a) All Six Repeated Test Results (FMY1-FBG1)



(b) Six Repeated Test Results at 120bar pressure level (FMY1-FBG1)

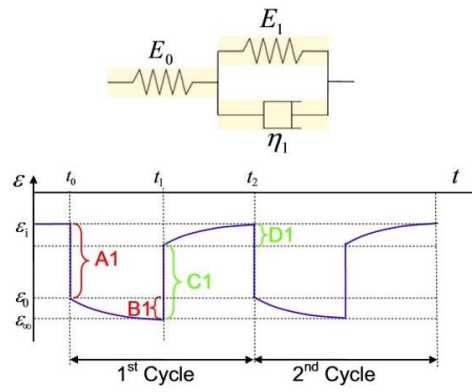
Figure 4-11: The Repeatability Test Results

Generally, all six test results were repeatable within the error range at the highest pressure (120bar) of around 3% ($25\mu\epsilon$), where the maximum difference was $36\mu\epsilon$ (between 1st and 6th tests) and the minimum difference was $1\mu\epsilon$ (between 3rd

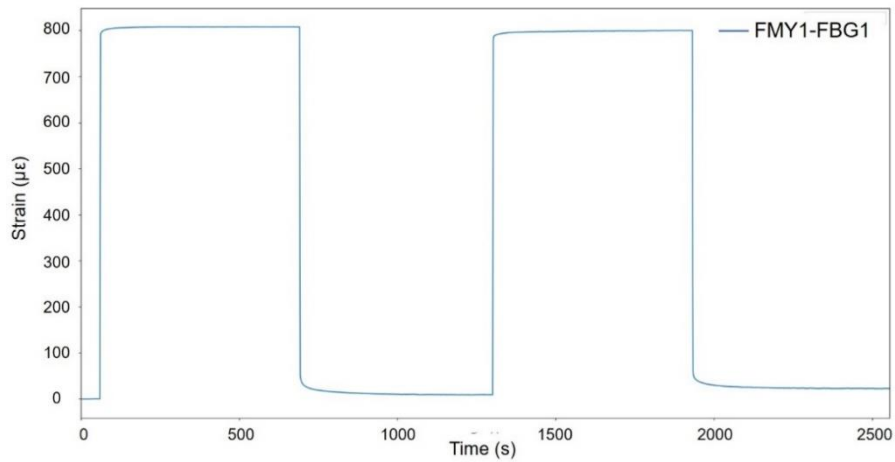
and 4th tests). Based on this observation, the subsequent step-by-step static loading analyses used three repeats under the same test conditions.

4.3.4.2 Visco-elastic Behaviour

Following the explanation of the visco-elastic behaviour in section 4.2.2.2, several tests were performed using equal pressure variation at four pressure levels (25 bar, 50 bar, 75 bar and 100 bar). In this section only the data obtained at the highest pressure (100 bar) will be discussed, with other results to be found in Appendix Chapter 4-D.



(a) Visco-elastic mathematical model and creep behaviour of a friction material (Wegmann et al., 2009)



(b) Visco-elastic Test Results at 100bar pressure level (FMY1-FBG1)

Figure 4-12: Visco-elastic Behaviour: Concept and Test Result

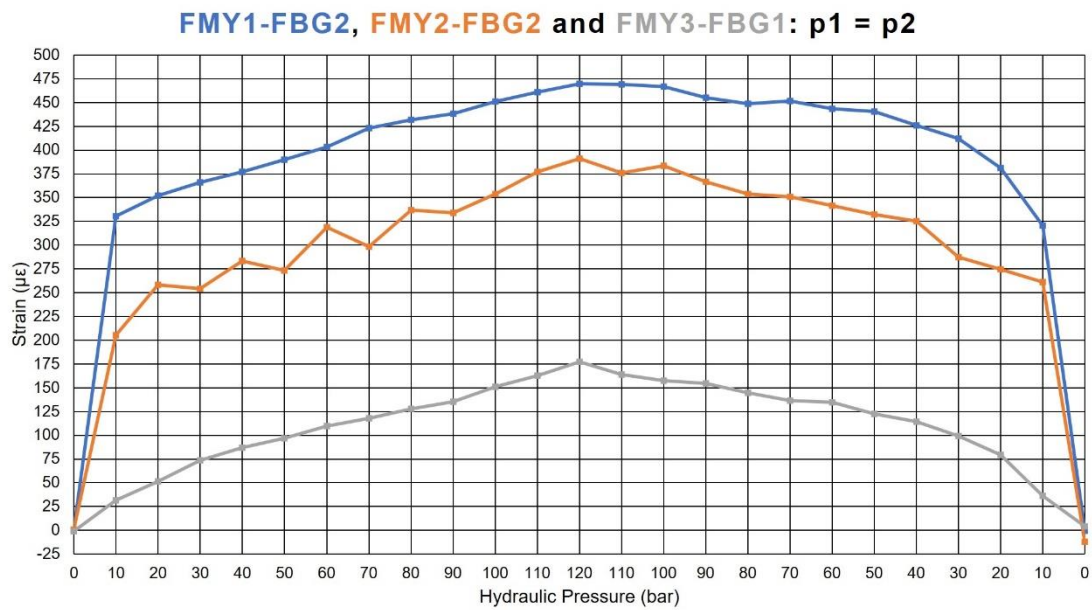
Figure 4-12 (a) illustrated the visco-elastic mathematical model and creep behaviour, where E_0 and E_1 were elastic elements combined with the viscous

element η_1 (Wegmann et al., 2009). They provided the creep parameters, A1, B1 C1 and D1, to determine each phase condition of two repeatable cycle runs. Such a mathematical model was used to display the material reflections of elastic (A1 and C1) and visco-elastic (B1 and D1) behaviours (Lenormand, 2017).

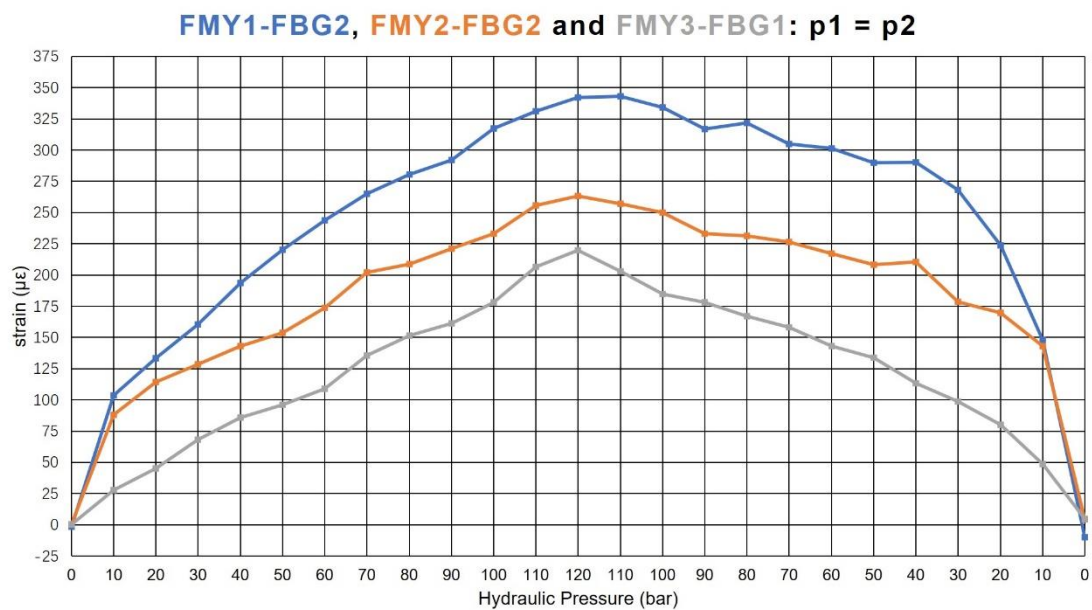
According to the test results provided in Figure 4-12 (b), the visco-elastic behaviour was revealed by a slow change in the strain when the hydraulic pressure was released from 100 bar to 0 bar. The material takes about 10 minutes to return to its original condition. However, considering time cost and test efficiency, the magnitude of the change would not significantly influence the data obtained when changing the pressure step-by-step. Usually, each test condition is repeated three times within a short period of time (about 1.5 hours), and then the pad is allowed to recover overnight due to the friction material's visco-elastic characteristic.

4.3.4.3 Inboard and Outboard Comparisons

Due to the space limitation discussed in section 4.2.2.3, it was impossible to make the comparison as the horizontal (X) direction pad could not be mounted in the inboard side, therefore, such tests were only focused on the vertical (Y) direction brake pad. A simple comparison is plotted in Figure 4-13, obtaining the data from three sensors, which located in the same horizontal line. The test results for the inboard and outboard sides differ significantly, as the pistons on each side of calliper have a 3mm offset, as mentioned in section 4.2.2.3, due to the disc brake system design. When swapping the instrumented pad from inboard to outboard, the offset of the piston location relative to the FBG arrays became 6mm in total, which was a significant offset in the context of the disc brake system, as the length of the brake pad displayed in Figure 3-22 was only around 130mm, with the 46mm piston diameter, which will be determined in Chapter 5. Thus, differences between the measurements are to be expected. Meanwhile, the horizontally oriented FBG arrays could only be monitored with the pad installed on the outboard side, so that the inboard side tests will be the initial practise.



(a) Inboard Sample Test Results at Equal Pressure Control



(b) Outboard Sample Test Results at Equal Pressure Control

Figure 4-13: Inboard and Outboard Comparison: Test Results

4.3.4.4 Other Possible Factors

There are several factors which might cause uncertainties observed during the tests, including temperature, humidity and human inputs.

Due to the thermal sensitivity of FBGs, temperature is an important factor to be considered, as a 1°C temperature change causes a change in the Bragg wavelength equivalent to that shown for a strain of 10µε. Therefore, as long as possible, each series of the test was completed under the same temperature environment, where the temperature was controlled to about 1°C in difference. As a result, most of the static loading tests have been done with the temperature ranged from 15°C to 25°C, around the ambient temperature level (20°C).

Meanwhile, it is known that the friction material on brake pad could be influenced by humidity. The coefficient of friction changes between wet and dry air, which might influence the test results. As a result, all of these tests were performed under air humidity conditions between 45%~50% and 65%~70%. The friction coefficient is known to change when increasing or decreasing the humidity, affecting the coefficient by about ± 0.1 to 0.2, where the normal coefficient was between 0.4 and 0.5. The brake pad would be exposed under these air conditions, where the braking performances of the friction material might be influenced by these changes. In particular, the brake function applied under the wet condition might increase the braking distance, which was considered dangerous during the real driving activities.

The human inputs concern the application of different hydraulic pressure levels from the test rig system. The pressure supply system has been designed to be controlled manually when applying various hydraulic pressures. Given the quality of the pressure gauges available, the pressure was typically set within a 2bar pressure error, but the level was constant once being set. This was judged to be preferable to the use of an automated control system, where the computer-controlled system would set and monitor the pressure, constantly adjusting the value to maintain the aimed point, creating a level of instability in the applied pressure level.

4.4 Summary

This chapter has described the experimentally determination of interface strain distributions at the brake pad frictional surface under static loading conditions,

which was part of the research aims. The results show that FBG strain sensors can be used to measure brake interface strain. The techniques used for the FBG sensor installations, while minimising their influences on the brake pad and its braking performance, also allows accurate, repeatable and reliable strain results to be obtained at the frictional interface of the brake pad.

Generally, the static loading cases focused on understanding the response to different applied pressure levels and variations. Some initial tests were performed prior to the actual test cases to ensure that the experimental facilities worked properly. Besides the static loading conditions, several conditions were applied to enable an understanding of the uncertainties and causes of inconsistent results.

The static loading analysis investigated brake pad strain performances under various test conditions, including pressure levels and variations, with sensors oriented along both the vertical and horizontal directions on the brake friction surface. The test results show high levels of repeatability and sensitivity, and could be used to understand and explain how brake interface pressure distributions would be captured through local strain levels mapped across the surfaces. Meanwhile, the application of FBG sensors could be effectively used to measure strain in disc brake and was easy to deploy and monitor. Meanwhile, the test result analysis and discussions considered the data processing and visualisation.

5 FE ANALYSIS PHASE 1 – STATIC LOADING ANALYSIS

The finite element analysis (FEA) is employed to simulate static test conditions by using Abaqus CAE software. These analyses will address several aspects, including model concepts and optimisation, simulations and analysis types. Finally, comparisons between FBG tests and FEA simulations will be conducted.

5.1 Finite Element Model

5.1.1 FE Model Concept

For static analyses, the created FE model of the brake assembly consists of, as shown in Figure 5-1, the disc section (a), the brake pad (b), and two pistons (c). Although the whole brake system has the brake calliper and the other (half) side of these parts, the static loading tests and analyses are focused on the outboard side of the brake, as the assembly can be considered (for the purpose of this work) symmetric to the disc mid-plane. The component material properties and dimensions are included in Table 5-1, as per Vianello's thesis (2015). The FE model assembly is shown in Figure 5-2, each component is carefully modelled to have practically identical geometry to the actual parts.

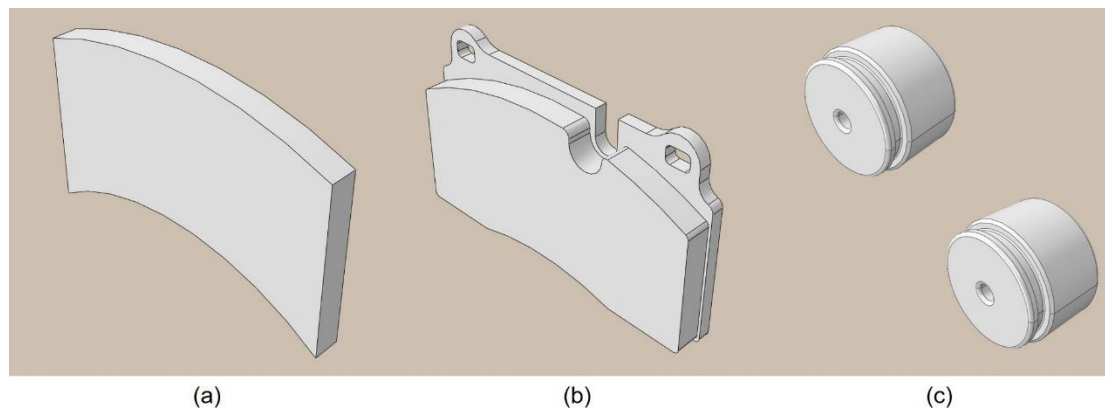


Figure 5-1: The FE Model Components: (a) Brake Disc Section, (b) Brake Pad, and (c) Two Pistons

Table 5-1: The Specification of Brake Components (Vianello, 2015)

Component Name	Material Type	Young's Modulus [GPa]	Poisson's Ratio [-]	Density [kg/m^3]	Dimension [mm]
Brake Disc	Grey Cast Iron GG150	110.7	0.245	6900	Ø360x30 (Thickness 9+12+9)
Brake Pistons	Grey Cast Iron G4000	200	0.21	7600	Ø46x32
Frictional Material	Organic Material	2.6	0.30	2670	131.6x77.3x10 (LxWxT)
Pad Backplate	Mild Steel	210	0.30	7800	131.6x77.3x5 (LxWxT)

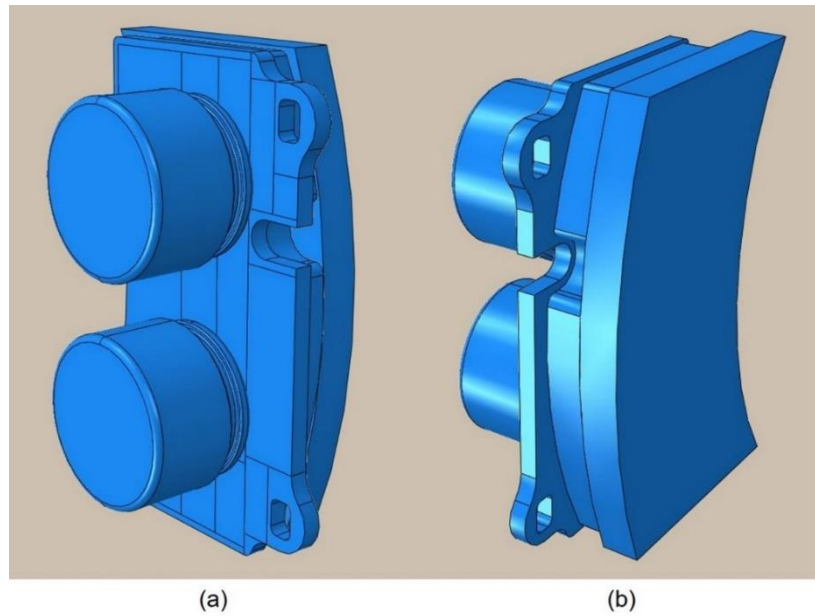


Figure 5-2: The FE Model Assembly: (a) Piston Side View and (b) Disc Side View

The mesh of the brake pad needed to be considered very carefully to ensure that the final meshed component could submit to FE analysis successfully. In order to create the uniform mesh on both the friction surface and the backplate, the partitions have been suitably created to ensure smooth, well-structure mesh. The friction surface has been divided into 16 areas so that each area would be meshed separately as hexahedron type elements, to ensure that the friction surface mesh is as uniform as possible (Figure 5-3 (a)).

A different approach was needed for the backplate, which has two holes infilled with the friction material and discrete circular contacts with two pistons. The backplate was therefore partitioned into various sections to capture the holes and

complex shape. The tetrahedron element type was used for meshing (Figure 5-3 (b)). Both friction material and backplate surfaces were discretised with 2.5mm in mesh size, four layers of the frictional surface and two layers of the backplate respectively, to allow specific loading conditions and to be accurately modelled. The mesh (element) size will be evaluated later in section 5.1.2.1.

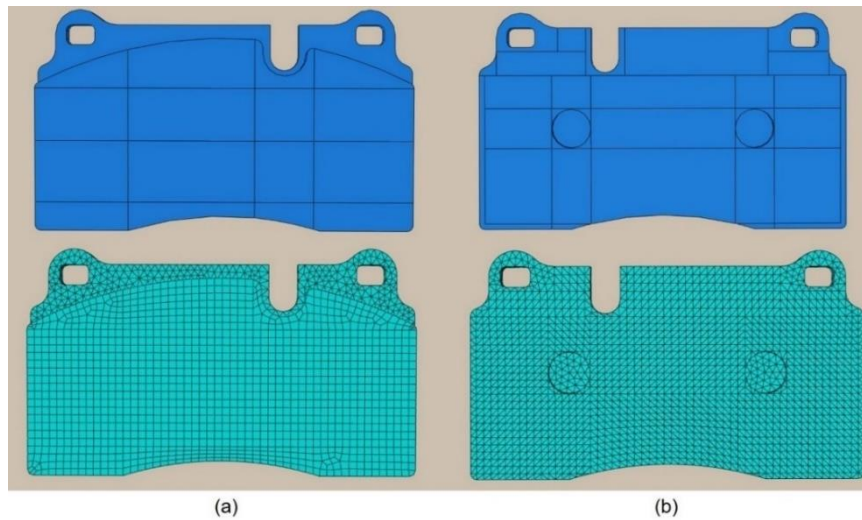


Figure 5-3: The Brake Pad Mesh: (a) Frictional Surface and (b) Backplate

The mesh of the disc section and pistons can be simplified but still ensuring they are effectively executing their duties (Figure 5-4). Both components have relatively simple geometry, are quite rigid and loaded in compression. Therefore, the disc section was set 3mm in element size and four layers. Meanwhile, the two pistons were discretized with the 5mm in nominal element size with tetrahedron element type.

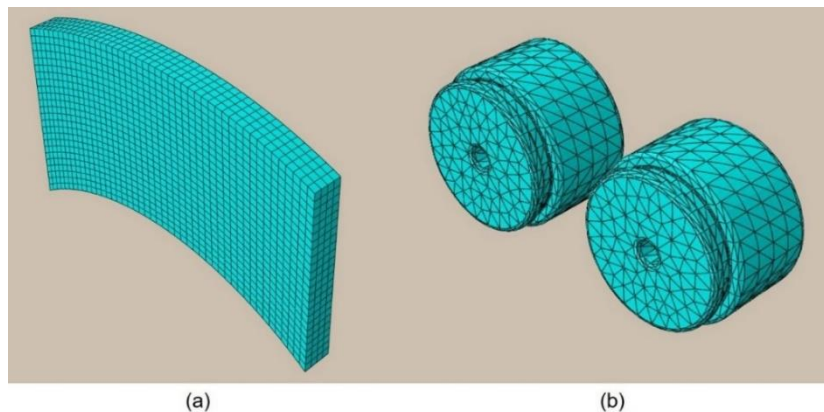
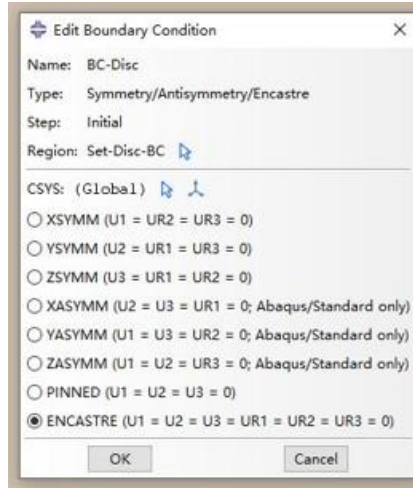
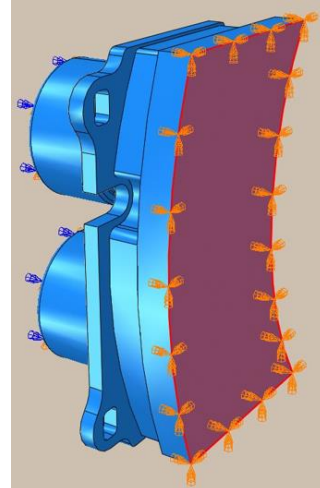
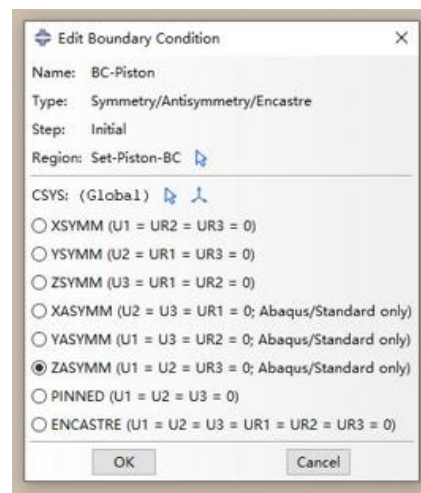
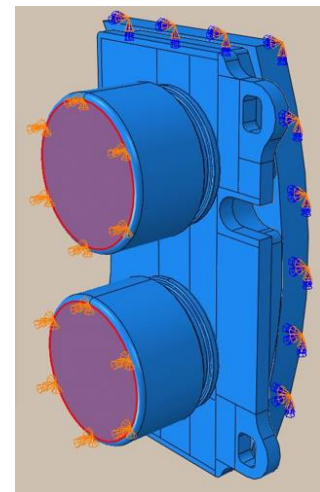


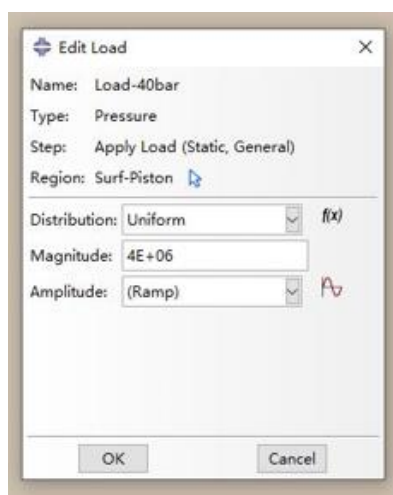
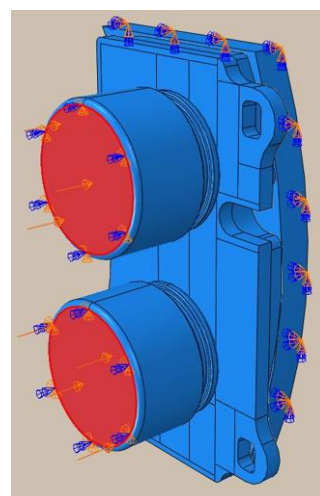
Figure 5-4: The Disc Section Mesh (a) and The Piston Mesh (b)



(a) Disc Section Boundary Condition: Mid-plane Displacement Constraint



(b) Piston Boundary Condition: Displacement Constraint



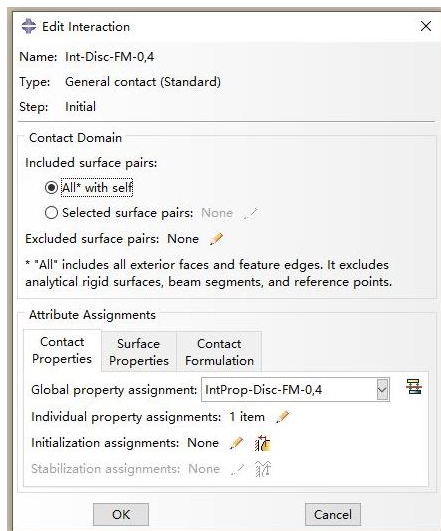
(c) Piston Load: Hydraulic Pressure Application

Figure 5-5: The FE Assembly Model: Boundary Conditions and Loads

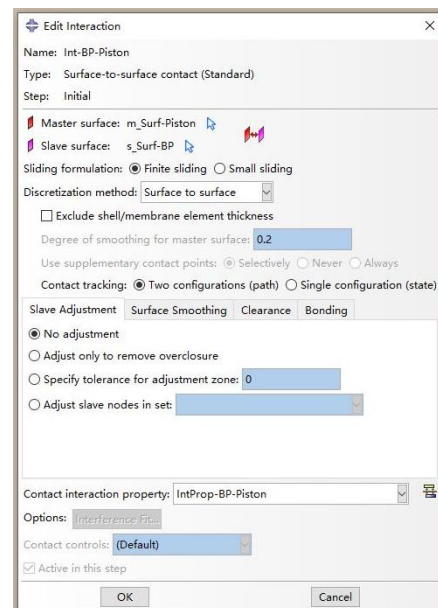
Boundary conditions (BCs), loads and interactions were carefully established to ensure the actual brake operation is suitably modelled. BCs and loads have been applied on both disc section and pistons in Figure 5-5. The disc section has been defined with encastre condition in the non-contact disc area to make sure that the whole disc section cannot move in any direction (Figure 5-5 (a)).

Pistons have contained both constraints and loads on the same surfaces. Constraints only allow pistons to move along their axes (Figure 5-5 (b)) and the hydraulic pressure is applied on the top of the pistons (Figure 5-5 (c)).

The previously explained boundary conditions were relatively easy to introduce and ensure the realistic constraints and correct component loading in static conditions. However, modelling of brake pad (friction material) contacted with the disc (at friction interface), and pistons contacted with the pad backplate required much more effort. After studying the problem carefully and Abaqus modelling options, the most suitable approach was established and validated.



(a) The General Contact



(b) The Surface-To-Surface Contact

Figure 5-6: The Definition of Contact Interactions in the FE Model

This included 'general contact' at the frictional surface between the friction material and the disc section. The friction coefficient has been set at 0.4 as the common value which is shown in Figure 5-6 (a), but will be further investigated

later in section 5.1.2.3. The contact between the backplate and the piston area was defined as ‘surface-to-surface contact’. Both surfaces are made of metal so that the hard contact has been selected as the normal behaviour of the contact interaction property in Figure 5-6 (b). The friction at here is very low and of practically no influence on the results obtained.

5.1.2 FE Model Optimisation

The previous section 5.1.1 established the initial FE model concept for the static loading analyses. However, several modelling aspects have been assumed so that the following sections will discuss these characteristics in detail, including mesh (element) size, integration type, friction coefficient and Young’s modulus of the friction material respectively.

5.1.2.1 Mesh (Element) Size

The element nominal size for the brake pad was selected at 2.5mm. However, nominal element size of 2mm and 5mm were also considered, as shown in Figure 5-7. Generally, the 5mm meshed pad only has two layers of elements for the frictional material and one layer for backplate (Figure 5-7 (c)). For the nominal element size of 2.5mm (Figure 5-7 (b)), the friction material is discretised with four layers and two layers for backplate. With 2mm nominal element size, the layers become of unequal thickness, but more layers have been applied (Figure 5-7 (a)).

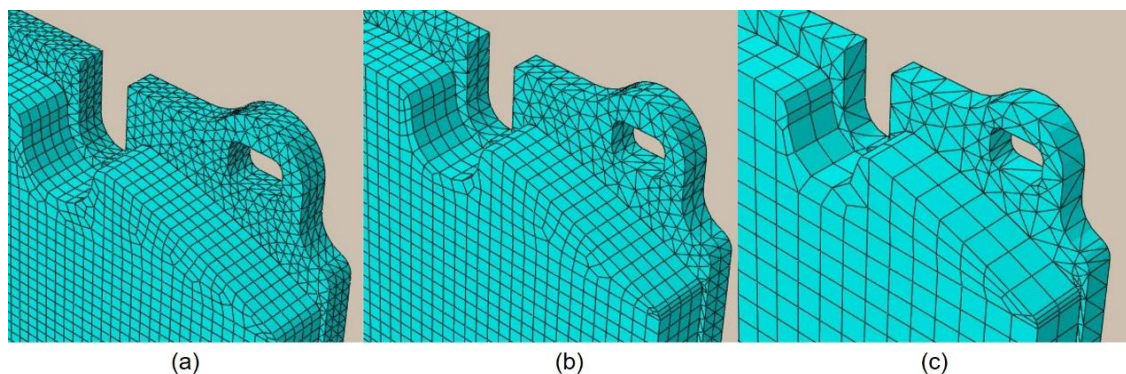
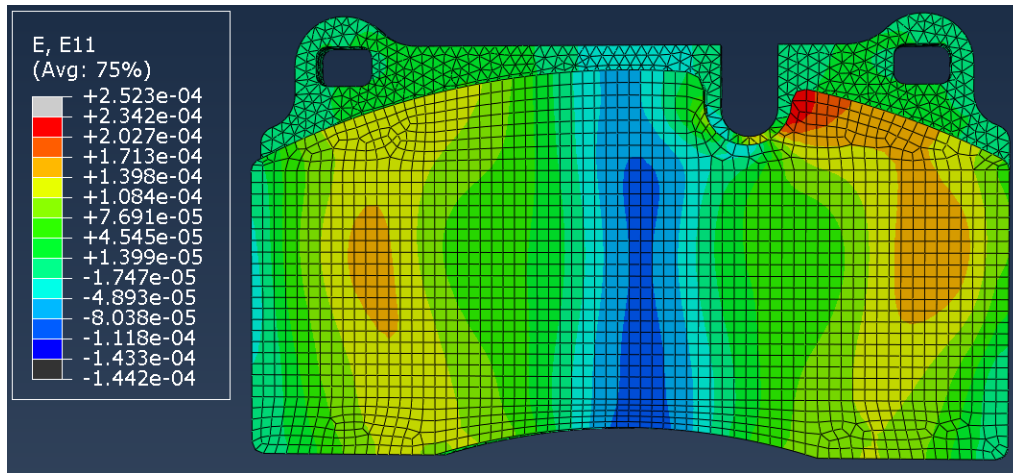


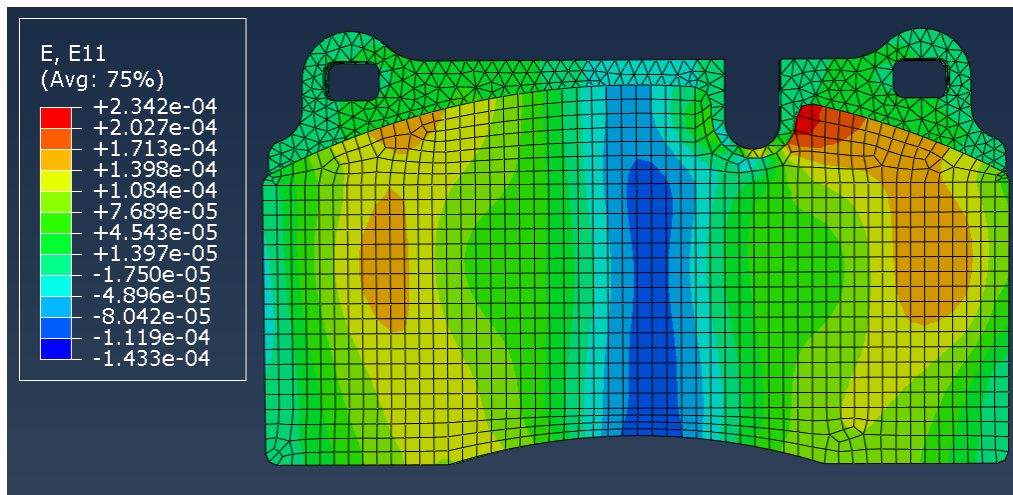
Figure 5-7: Different Element Sizes: (a) 2mm, (b) 2.5mm, and (c) 5mm

For selecting the most suitable element size, it is necessary to conduct relevant FE analyses under identical BCs and other parameters. Therefore, all three mesh (element) sizes of the pad were analysed under the identical conditions and the results were compared. The comparisons included contact pressure (CPRESS), strain in X direction (E11) and strain in Y direction (E22). These variables are selected as they are most indicative of the contact conditions and changes at the frictional interface, with E11 and E22, also measured using FBG strain sensors. For the result illustration, the X direction strain (E, E11) has been displayed in Figure 5-8, other comparisons are included in Appendix Chapter 5-A.

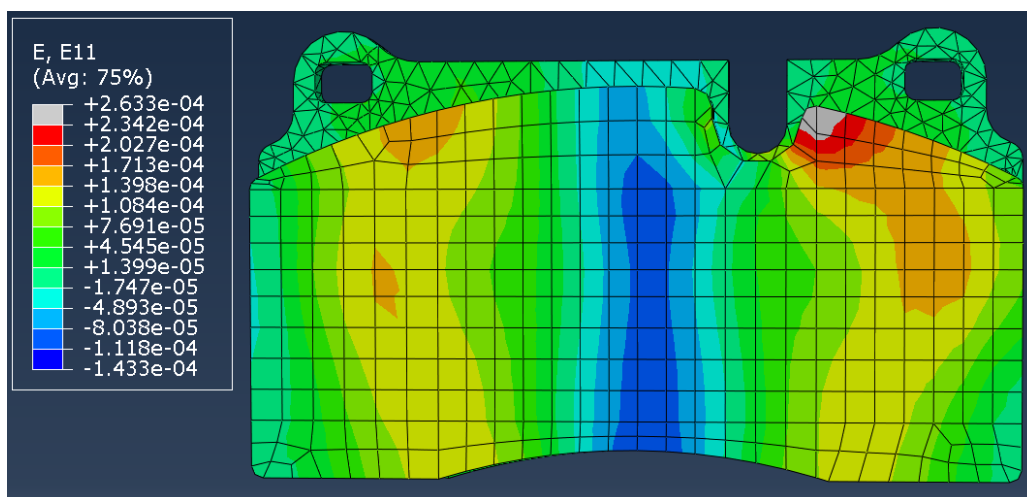
Figure 5-8 clearly shows that a 5mm meshed size would not provide strain results in sufficient details. However, there are no significant differences in strain values and distributions between 2mm and 2.5mm mesh (element) sizes. Considering the number and position of FBG strain sensors installed, as well as their nominal size of 5mm, FE mesh (element) size of 2.5mm is considered to be perfectly adequate, offering further advantages in averaging the strains across two elements, for direct comparisons with FBG test results.



(a) 2mm



(b) 2.5mm



(c) 5mm

Figure 5-8: E11 (X Direction) FE Analysis Result for Different Element Sizes

5.1.2.2 Integration Type

The Abaqus CAE software for element type offers full (F) and reduced (R) integration for 8 noded hexagon elements, i.e. C3D8 and C3D8R respectively. The full integration element uses 8 (Gauss) points for numerical integration, whereas reduced integration uses only 4 points. In some conditions (i.e. beam bending) considerable differences in the results (deflection, strain and stress) can be predicted when using full or reduced integration, with full integration leading to, so-called 'shear locking' phenomenon for the first order elements.

However, in the case of the brake pad, as shown in Figure 5-9, the results are almost identical for full and reduced integration element types, with maximum differences being under 3%. More detailed comparisons have been included in Appendix Chapter 5-B. As far as the analysis efficiency is concerned (shorter CPU times), the reduced integration type (C3D8R) was used for the FE model.

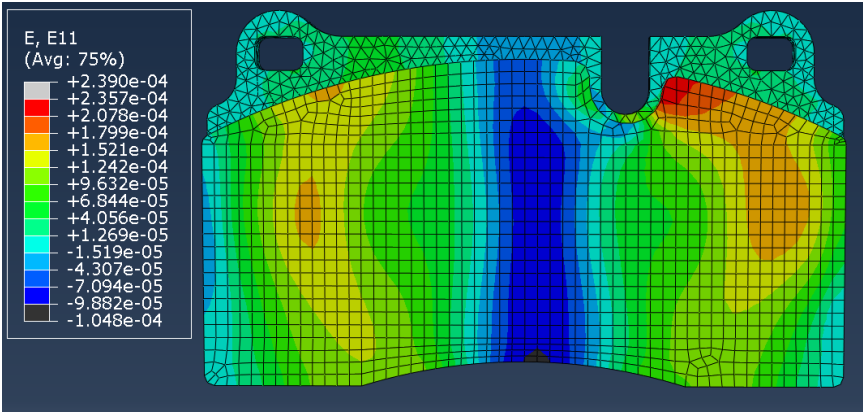
40bar-2.5mm-2.6Gpa-0.3-Full				
Part Instance	Element ID	Type	Int. Pt.	E, E22
Brake Pad Modelling-1	1194	C3D8	1	201.23E-06
Brake Pad Modelling-1	1194	C3D8	2	212.278E-06
Brake Pad Modelling-1	1194	C3D8	3	200.152E-06
Brake Pad Modelling-1	1194	C3D8	4	210.633E-06
Brake Pad Modelling-1	1194	C3D8	5	200.608E-06
Brake Pad Modelling-1	1194	C3D8	6	212.654E-06
Brake Pad Modelling-1	1194	C3D8	7	200.491E-06
Brake Pad Modelling-1	1194	C3D8	8	211.898E-06
Average				206.243E-06
40bar-2.5mm-2.6Gpa-0.3-Reduced				
Part Instance	Element ID	Type	Int. Pt.	E, E22
Brake Pad Modelling-1	1194	C3D8R	1	212.127E-06

Figure 5-9: The Comparison between Full and Reduced Integration Types

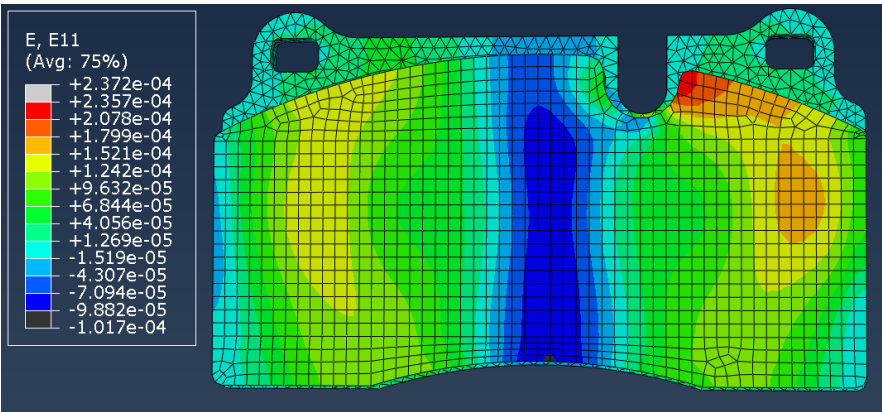
5.1.2.3 The Coefficient of Friction

The coefficient of friction is a very complex parameter, influenced by many aspects, in particular temperature, pressure and sliding speed, but also prone to changes with ambient conditions such as humidity, and prone to 'Memory Effects'. Static and dynamic loading friction coefficients can also differ and the relationship

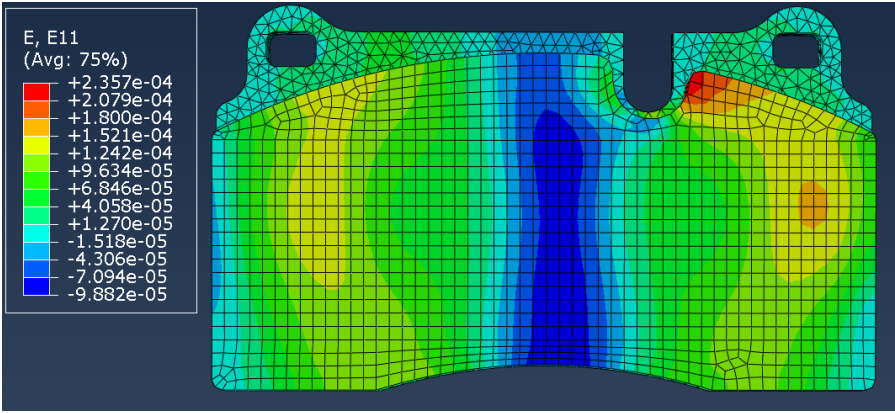
can vary, from one measurement to next, even when all conditions are practically identical. For most organic friction materials used in automotive industry, the most commonly used value for performing braking calculations is 0.4. This value was also used by Vianello (2015) for this very friction material.



(a) Friction Coefficient 0.2



(b) Friction Coefficient 0.3



(c) Friction Coefficient 0.4

Figure 5-10: E11 (X Direction) Comparisons for Different Friction Coefficients

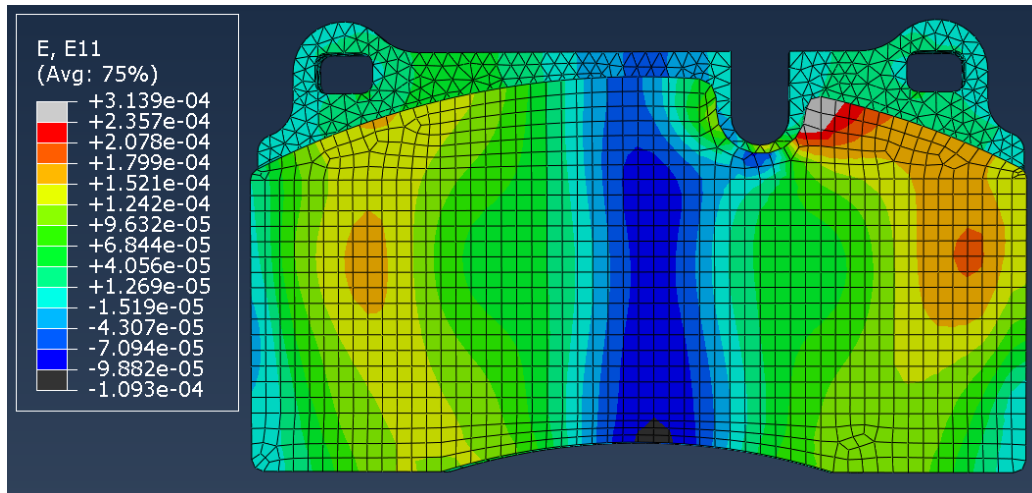
However, in order to investigate the influence of the coefficient of friction on predicted strain values, in addition to 0.4, the values of 0.2 and 0.3 were also used for the friction coefficient in the FE analyses conducted. The obtained strain values at the friction interface in X direction (E11) are shown in Figure 5-10, with other results placed in Appendix Chapter 5-C. It should be noted that all analyses conducted were only for static loading conditions.

According to Figure 5-10, the strain results are almost identical for the three values used for the friction coefficients. The differences are very small and only local. It would not be possible to detect such changes with the applied FBG strain sensors. Consequently, the value for the friction coefficient for all FE analyses will be set at 0.4, which is the most commonly used value.

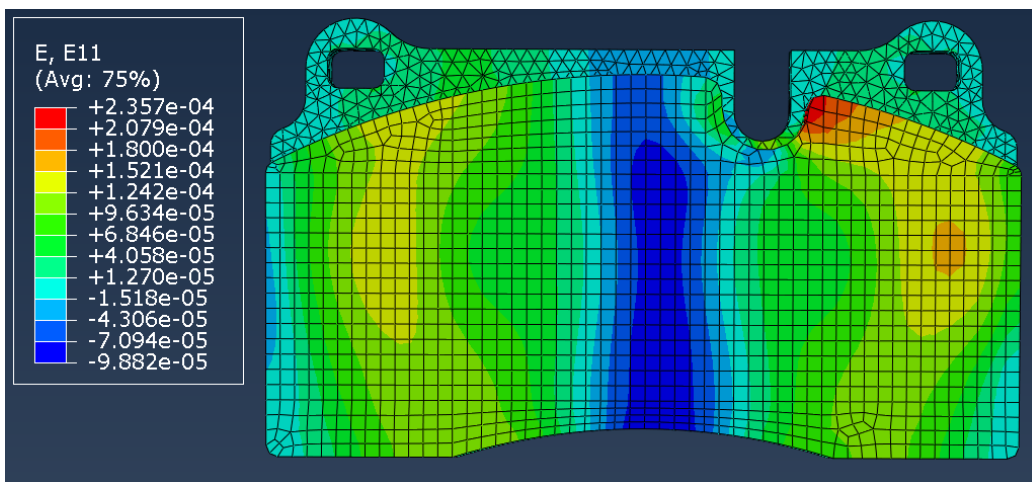
5.1.2.4 Friction Material Young's Modulus

The Young's modulus of 2.6GPa was selected for the setting friction material definition that based on the Vianello's measurement (Vianello, 2015). As the properties of the organic material would not always be stable during different test conditions, 25% differences of the Young's modulus have been investigated, 1.95GPa (-25%) and 3.25GPa (+25%) respectively. As in previous sections 5.1.2.1 and 5.1.2.3, strain results in X direction (E11) at the friction surface were used for the comparisons. Contour plots are displayed in Figure 5-11 and other results are included in Appendix Chapter 5-D.

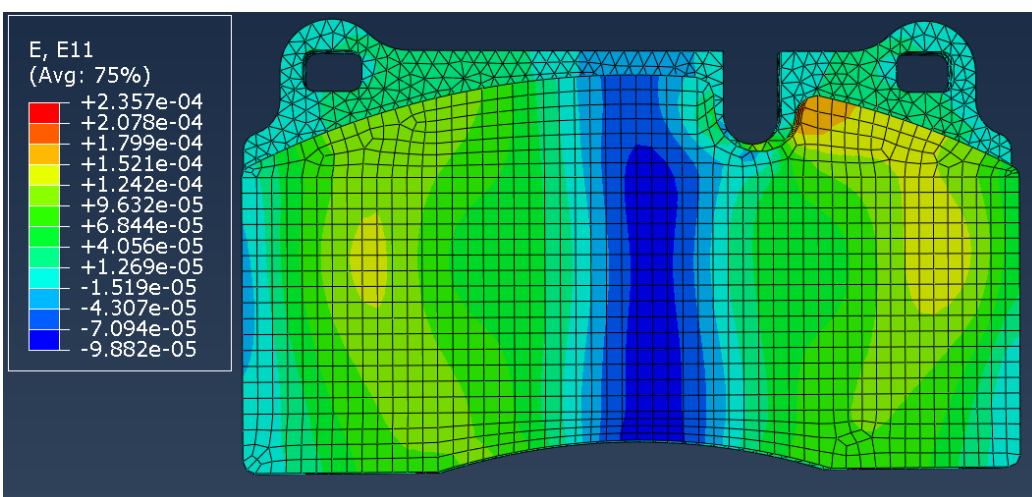
Some differences in actual strain values can be observed in Figure 5-11, for the three values of the Young's modulus. However, the result effects are relatively localized, and the nature of strain distributions is almost identical. Considering FBG strain sensor size and position, and relative differences being of particular importance, it can be considered that the nominal value Young's modulus is 2.6GPa, as measured by Vianello (2015) makes most sense to be used.



(a) Young's Modulus 1.95GPa (-25%)



(b) Young's Modulus 2.6GPa (Nominal)



(c) Young's Modulus 3.25GPa (+25%)

Figure 5-11: E11 (X Direction) Comparisons for Different Young's Modulus Values

In conclusion, according to previous analyses and considerations, the static loading FE brake assembly model characteristics can be set as follows:

- Nominal Element Size: 2.5mm (pad), 3mm (disc), 5mm (pistons)
- Integration Type: Reduced Integration
- Friction Coefficient: 0.4
- Young's Modulus: 2.6GPa

5.2 Finite Element Analysis

5.2.1 Approaches in Processing the Results

FE analysis results displayed in contour plots are difficult to compare with FBG test results shown in section 4.3.3, therefore, further processing was needed for FE analysis results. As each FBG strain sensor was about 5mm in length (to measure wavelength changes), two elements (2.5mm each) were selected at the sensor location and strains were averaged to define the specific strain result and created the FEA sensor contour map. An example is given in Figure 5-12 for strains in Y direction (E22).

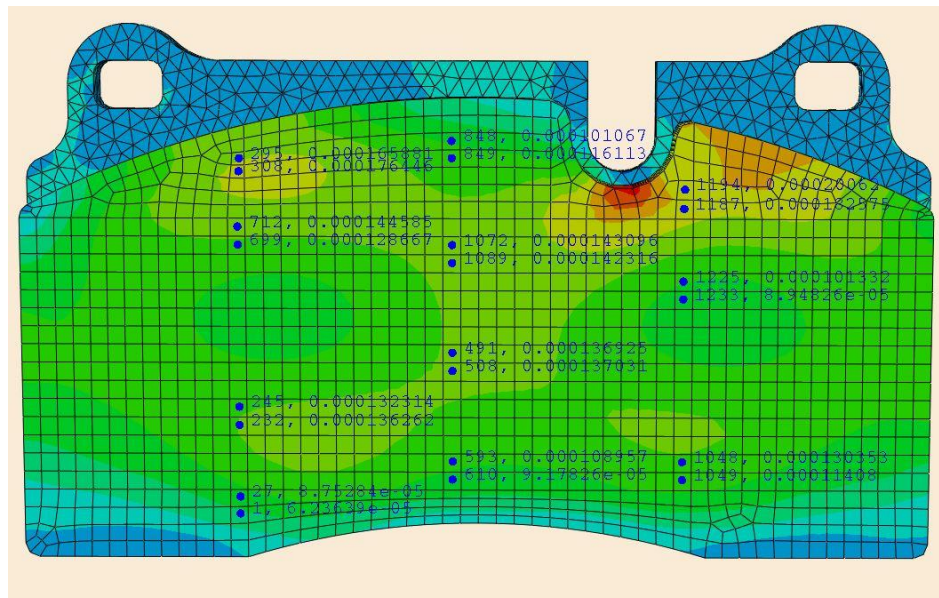


Figure 5-12: FEA Result Element Selections (E22)

Although the FE elements cannot always be exactly picked at the same position as the FBG strain sensors, the chosen elements are in a very close proximity to cover the corresponding sensors' locations. Furthermore, since there are no high

strain gradients in these areas, such an approach is viable. FEA strain results were transferred onto the 'FEA sensor contour map' enabling direct comparisons with FBG test results, as shown in the following section 5.3.

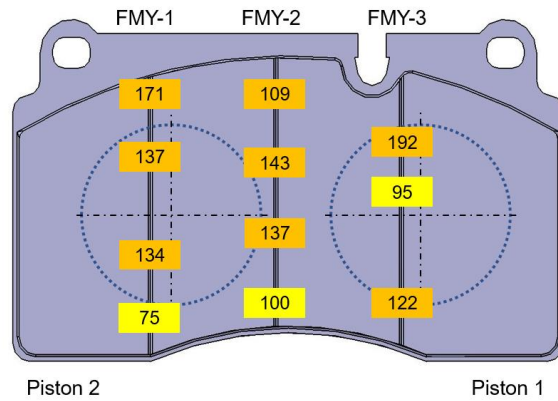
5.2.2 FEA Results and Analysis

5.2.2.1 Influence of Different Hydraulic Pressure Levels

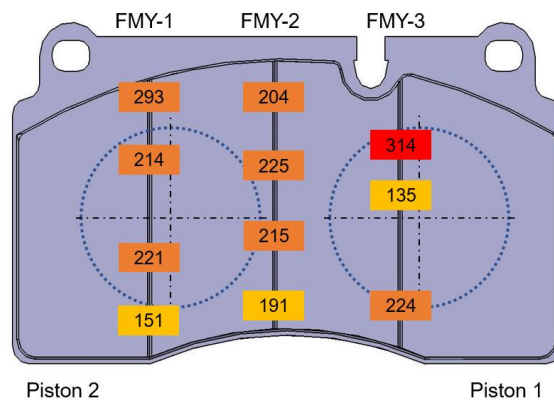
Three hydraulic pressure levels were selected, identical to the static loading experimental phase, being 40 bar, 80 bar and 120 bar respectively. Each level of the strain results in horizontal (circumferential, X) and vertical (radial, Y) directions were processed and hence displayed the same spectrum colour legends as in section 4.3.3. Figure 5-13 and Figure 5-14 show FEA predicted strain values at FBG sensor locations in Y and X directions. The maps are generated for actuating 40 bar, 80 bar and 120 bar hydraulic pressure levels. Pressures are applied with the equal pressure variation on both piston sets, $p_1 = p_2$.

According to Figure 5-13 and Figure 5-14, both directions have shown similar trends that the strain values change as expected when the applied hydraulic pressure levels increase or decrease. Especially, it is found along the X direction that the strain level changed significantly around the piston areas with different hydraulic pressure levels. In the middle of the frictional surface, strains have become negative values, indicating the material compression.

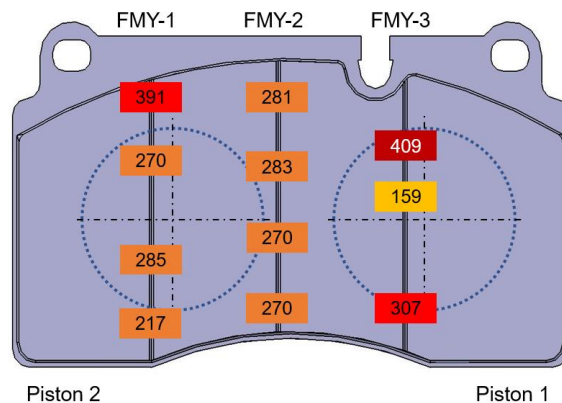
Strain ($\mu\epsilon$)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



(a) 40bar Hydraulic Pressure

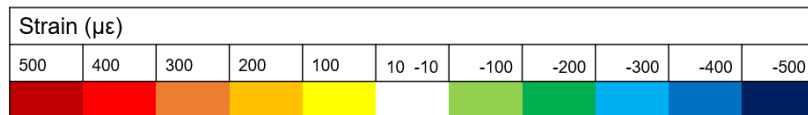
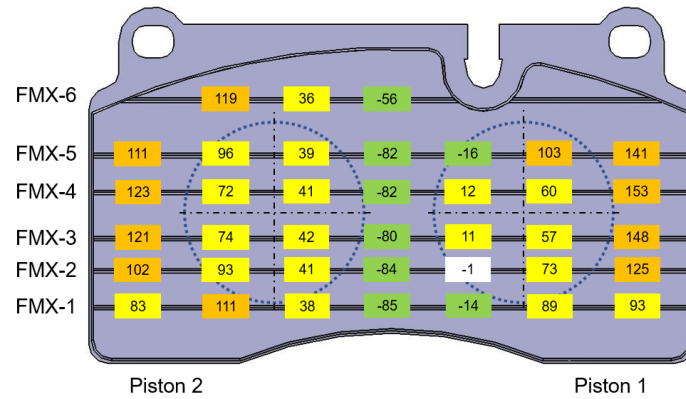


(b) 80bar Hydraulic Pressure

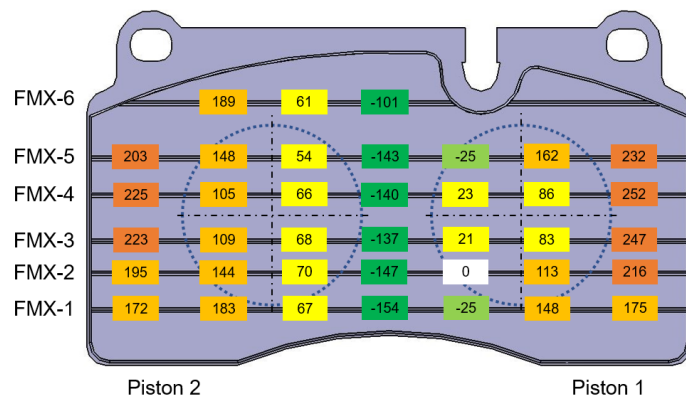


(c) 120bar Hydraulic Pressure

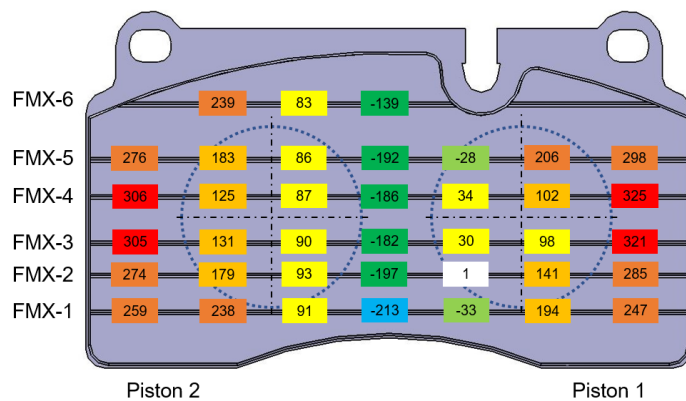
Figure 5-13: Processed FEA Contour Maps in Vertical (Y) Direction for Three Hydraulic Pressure Levels with Equal Pressure ($p_1=p_2$)



(a) 40bar Hydraulic Pressure



(b) 80bar Hydraulic Pressure



(c) 120bar Hydraulic Pressure

Figure 5-14: Processed FEA Contour Maps in Horizontal (X) Direction for Three Hydraulic Pressure Levels with Equal Pressure ($p_1=p_2$)

5.2.2.2 Influence of Hydraulic Pressure Variations between Piston Sets

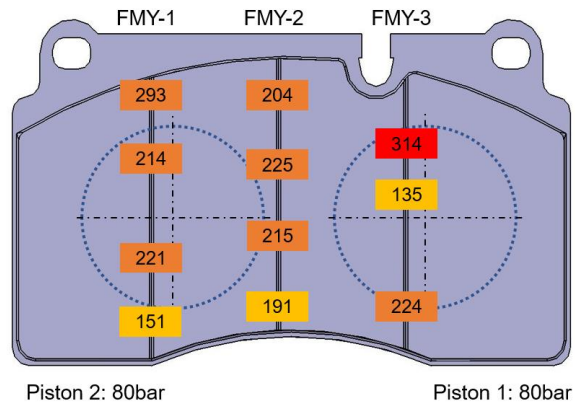
In addition to the equal pressure variation on both piston sets ($p_1=p_2$), the hydraulic pressure was also varied by ± 10 bar in order to investigate how these changes affect the strain distributions on the frictional surface.

Figure 5-15 and Figure 5-16 show strain distributions in Y and X directions, respectively. The results refer to 80 bar nominal pressure, with the results for 40 bar illustrated in Appendix Chapter 5-E and for 120 bar placed in Appendix Chapter 5-F. First, the 'baseline' strain distribution is shown, for equal pressure of 80 bar on both piston sets (Figure 5-15 (a) and Figure 5-16 (a)). In the next step, the pressure was reduced on piston 2 to 70 bar, whilst maintaining 80 bar at piston 1 (Figure 5-15 (b) and Figure 5-16 (b)). Finally, in the third step, the pressure was set 80 bar at piston 2 and 70 bar at piston 1 (Figure 5-15 (c) and Figure 5-16 (c)). It needed to be pointed out that pistons 1 and 2 are not positioned symmetrically in relation to the pad axis (mid-plane) of symmetry, 3mm offset. The hydraulic pressure variations are aimed at investigating changes of strain distributions, hence the position of the centre of pressure, which is of crucial importance for the brake performance, NVH and wear characteristics.

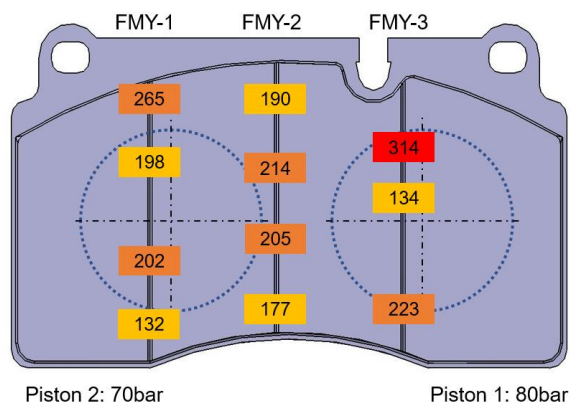
Looking at Figure 5-15, starting with the equal pressure (Figure 5-15 (a)), E22 strain distribution is relatively equal and symmetrical. The cut-out above piston 1, for the wear sensor installation, causes increase of the local strain ($314\mu\epsilon$ at the top of piston 1), however, the strain then drops to the lowest value predicted ($135\mu\epsilon$) in the proximity of piston 1 axis. When the pressure applied in piston 2 is reduced to 70 bar (still maintaining 80 bar in piston 1, Figure 5-15 (b)), there is an expected drop of strain values under piston 2 and in the proximity to the middle of the pad. Strain under piston 1 remain practically unchanged.

Similarly, Figure 5-15 (c), when pressure in piston 1 is reduced to 70 bar (and 80 bar maintained at piston 2), strains in the area under piston 1 are reduced but strains under piston 2 remain unaffected. Interestingly, reduction of p_1 pressure has less effect on the strain reduction in the groove between the two piston sets, as the groove is positioned closer to the piston 2.

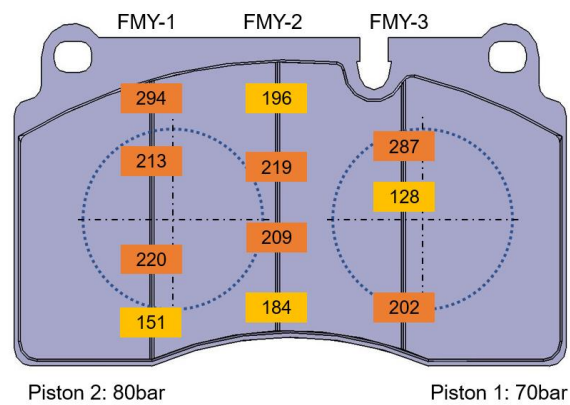
Strain ($\mu\epsilon$)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)

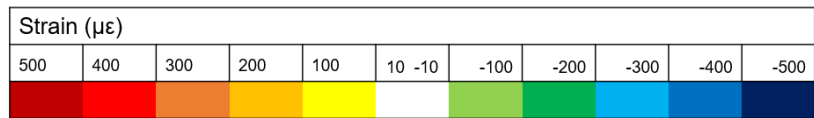
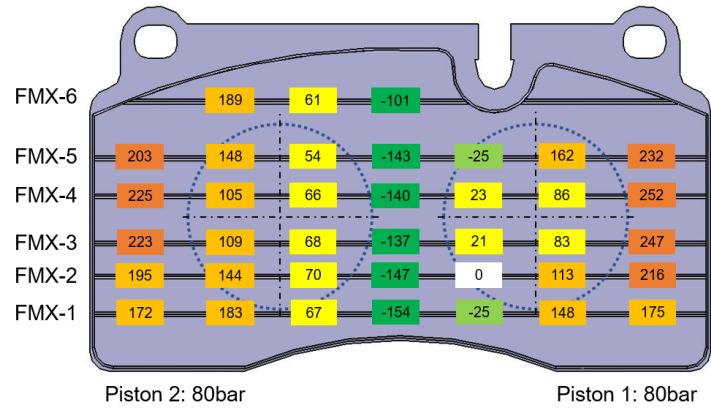


(b) +10bar ($p_1=80\text{bar}$; $p_2=70\text{bar}$)

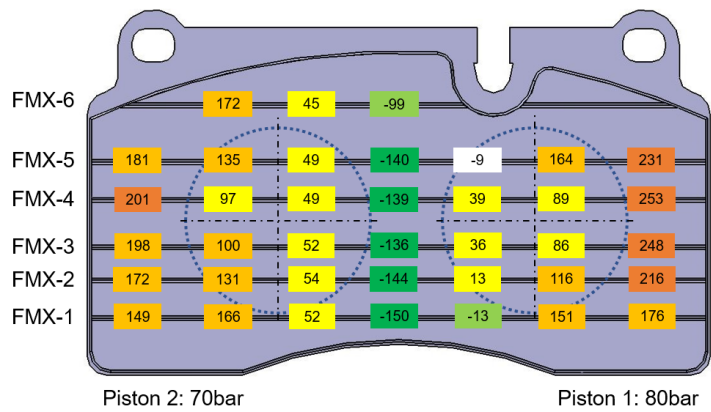


(c) -10bar ($p_1=70\text{bar}$; $p_2=80\text{bar}$)

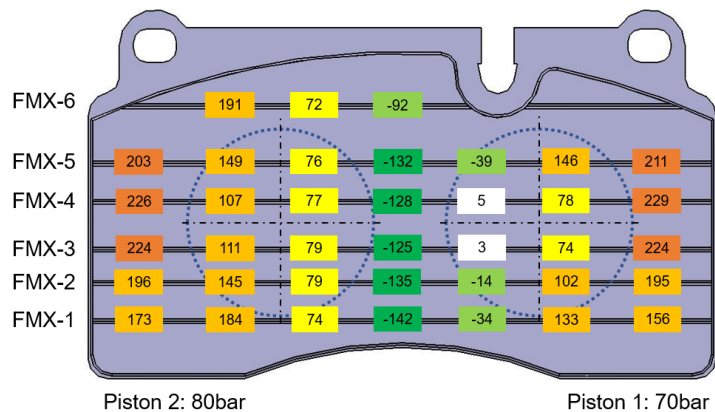
Figure 5-15: E22 (Y Direction) Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)



(b) +10bar ($p_1=80\text{bar}$; $p_2=70\text{bar}$)



(c) -10bar ($p_1=70\text{bar}$; $p_2=80\text{bar}$)

Figure 5-16: E11 (X Direction) Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations

Figure 5-16 shows predicted E11 strain distributions in horizontal (X) direction. There are 38 sensor positions here and the results are more numerous, nevertheless, FEA predictions show for equal pressure (Figure 5-16 (a)) quite symmetrical distributions. The highest tensile strains are at the pad's leading and trailing edges, with the maximum value of $252\mu\epsilon$. Strain values are quite low around the piston edges towards the middle of the brake pad, where one prediction shows $0\mu\epsilon$, changing to compression in the area between the pistons, up to the value of $-154\mu\epsilon$.

When the pressure in piston 2 is reduced to 70 bar (Figure 5-16 (b)), still maintaining 80 bar in piston 1, there is a reduction of E11 strain values under piston 2, which is to be expected. There is a very slight reduction of strain values in the middle of the brake pad, whereas the strain values in the area under piston 1 remain practically unchanged.

Finally, Figure 5-16 (c), when the pressure in piston 1 is reduced to 70 bar and pressure under piston 2 kept at 80 bar, there is a reverse effect. The strain under piston 2 remains unchanged in comparison to equal pressure condition shown in Figure 6-16 (a), but there is a small change of strain values under piston 1. There is also a small but noticeable strain reduction in the middle of the pad.

The results shown in Figure 5-15 and Figure 5-16 look very logical and strain distributions are smooth, without sudden changes. This is the result of the component geometry and loading, as well as adequate modelling approach and assumption of a homogeneous friction material. In the next section, FEA predicted strains will be compared with measured values using FBG sensors.

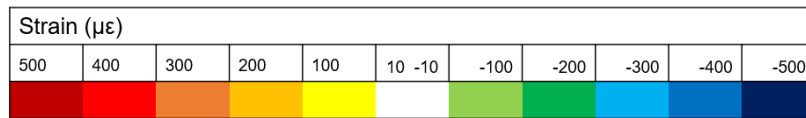
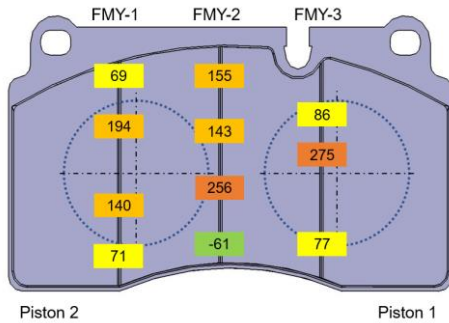
5.3 Comparison of FBG Test and FEA Simulation Results

The previous section 5.2 in this chapter shows FEA simulation results for different static loading conditions, with Chapter 4 detailing FBG static loading test results. Both sets of results have been presented and analysed in detail and in the same layout. That forms a base for direct comparisons of FBG test and FEA simulation results. Hence, a further investigation into a clear, most informative comparison format is necessary.

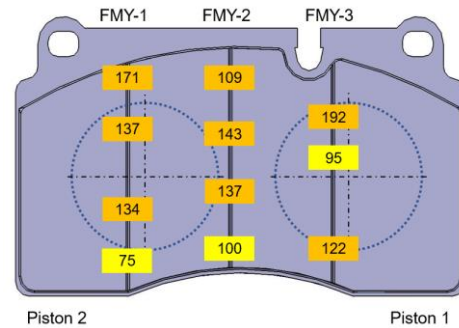
5.3.1 Approaches in Result Comparisons

Both FBG and FEA results have been displayed on sensor contour maps, which were an effective way to find strain value at specific locations. In order to compare these results, several steps have been taken to establish strain differences between FBG test and FEA simulation results in Y direction (E22) for the same hydraulic pressure level of 40 bar. Both sets of results will keep the same colour legend to ensure comparisons can be easily traced. Figure 5-17 begins by showing FBG test results (a), then FEA simulations (b), continuing with both sets of FBG and FEA side-by-side (c), and finally the strain differences (d).

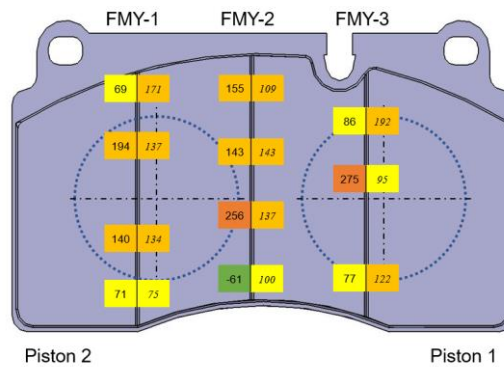
Figure 5-17 (d) is found to be most suitable for comparing FBG and FEA results. It shows calculated strain differences between FBG tests and FEA simulations, hence if all values were $0\mu\epsilon$, the 'perfect match' would have been achieved. In three places, FMY1-FBG3, FMY1-FBG4, and FMY2-FBG2, strain levels of FBG and FEA are practically identical. FMY1-FBG2, FMY2-FBG1 and FMY3-FBG3 are three sensor positions where strain differences are mainly under $100\mu\epsilon$, even under $50\mu\epsilon$. As for the other locations, the differences are between $100\mu\epsilon$ and $200\mu\epsilon$. The exceptions are FMY2-FBG4, where the difference is $-161\mu\epsilon$, and FMY3-FBG2 with the difference of $180\mu\epsilon$. The general summary table has also been created, which is 'standardised' and will be used for all further comparisons. As stated, in 55% of places (6/11), the differences are under $100\mu\epsilon$, with the remaining 45% being between $100\mu\epsilon$ and $200\mu\epsilon$ (5/11).



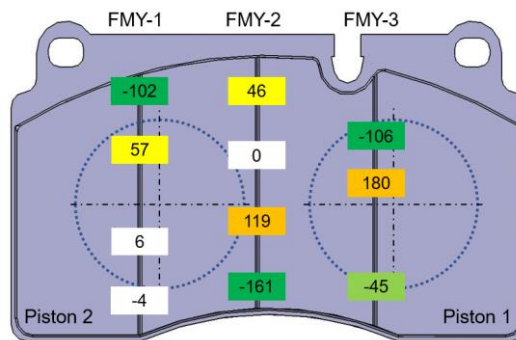
(a) FBG Test Results



(b) FEA Simulation Results



(c) Comparisons of FBG and FEA



$\Delta\epsilon$	$[\mu\epsilon]$	Range	Percentage
Minimum	0	$\Delta\epsilon \leq 100\mu\epsilon$	55% (6/11)
Maximum	180	$100\mu\epsilon < \Delta\epsilon \leq 200\mu\epsilon$	45% (5/11)
Absolute Average	75	$200\mu\epsilon < \Delta\epsilon \leq 300\mu\epsilon$	0% (0/11)
		$\Delta\epsilon > 300\mu\epsilon$	0% (0/11)

(d) Strain Differences: $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$

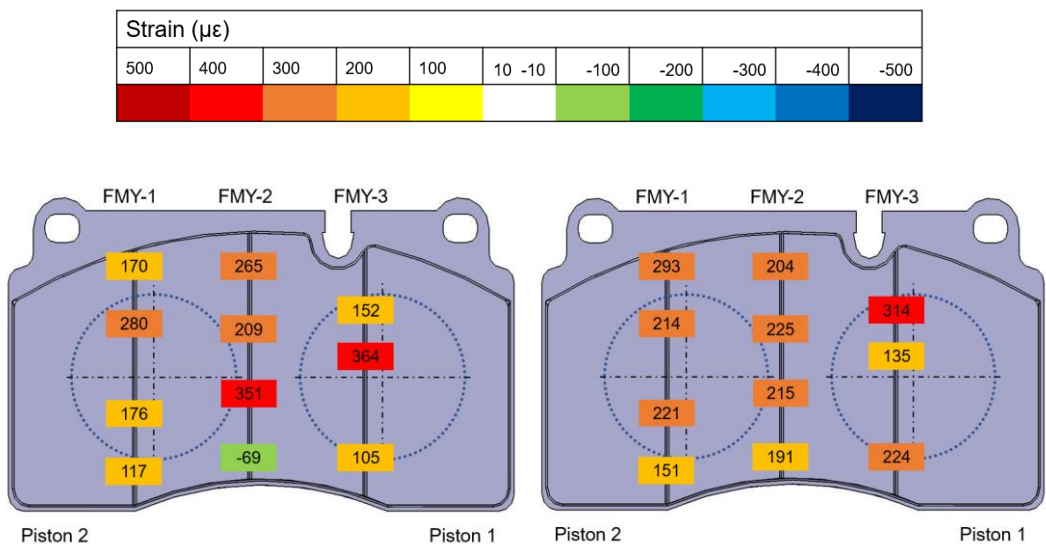
Figure 5-17: E22 (Y Direction) Result Comparisons between FBG and FEA for Equal Pressure Variation, $p_1=p_2=40\text{bar}$

5.3.2 Analysis of The Results

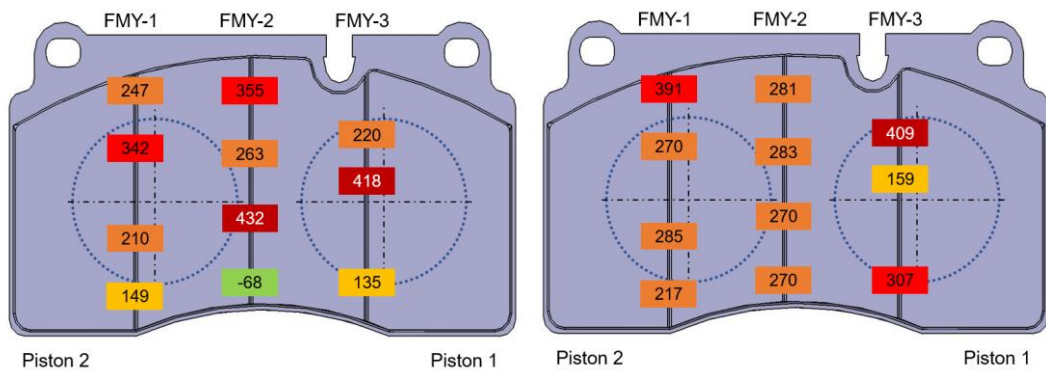
The previous section 5.3.1 established the approach in FBG and FEA result comparisons. This will now be used to investigate strain differences at various hydraulic pressure levels, as well as when unequal hydraulic pressure variation is applied between the two piston sets.

5.3.2.1 Hydraulic Pressure Levels

Both FBG tests and FEA simulations have been conducted for a range of hydraulic pressure levels, with the results more closely analysed for the actuating hydraulic pressures of 40 bar, 80 bar and 120 bar. In this section, hydraulic pressures will be identical for all pistons (i.e. equal pressure condition, $p_1 = p_2$).



(a) FBG (Left) and FEA (Right) Results at 80 bar Level



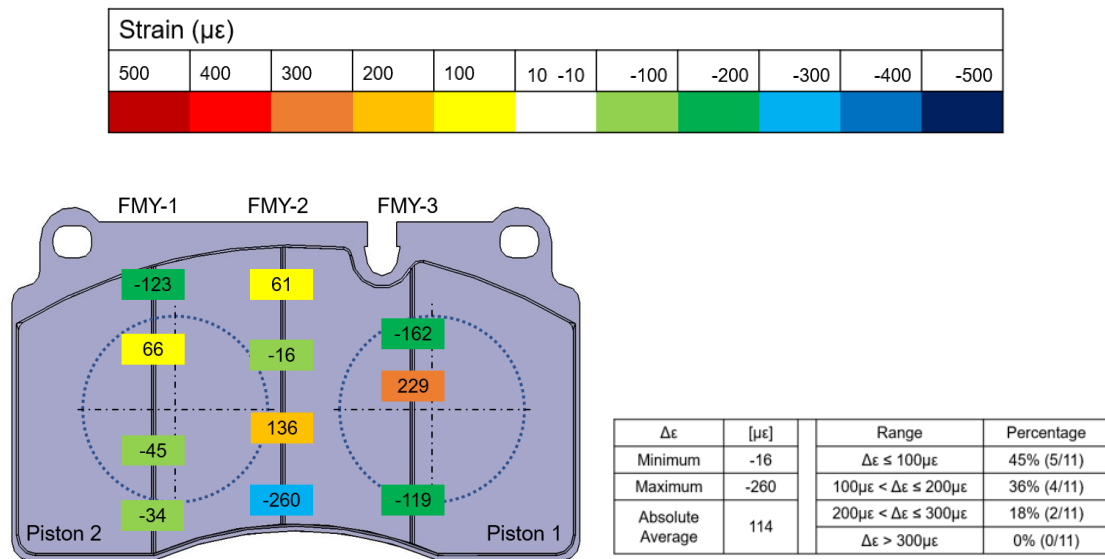
(b) FBG (Left) and FEA (Right) Results at 120 bar Level

Figure 5-18: E22 (Y Direction) Strain Results for Two Hydraulic Pressure Levels

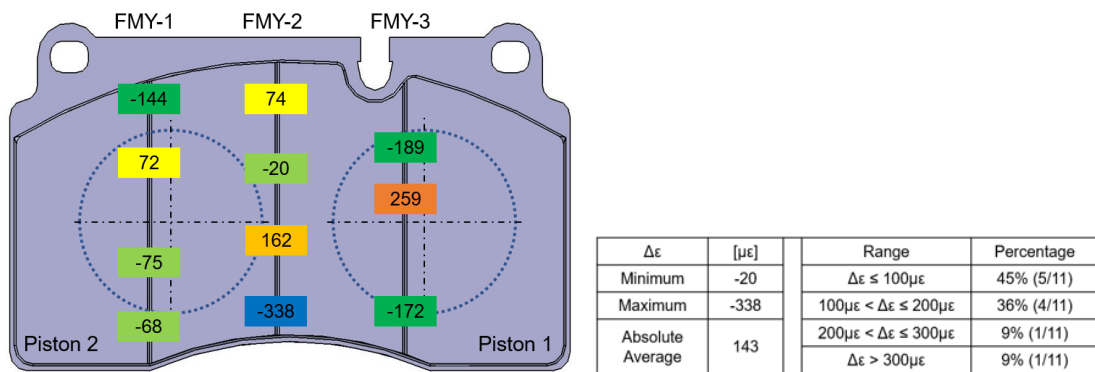
As the results for the actuating pressure of 40 bar were investigated in the previous section 5.3.1, pressures of 80 bar and 120 bar will be observed in more detail. Figure 5-18 shows FBG and FEA strain results in Y direction (E22) for 80 bar (Figure 5-18 (a)) and 120 bar (Figure 5-18 (b)). It can be seen that some results are relatively close, with others being quite far apart.

In order to gain a deeper insight, as proposed in the previous section, strain differences are plotted in Figure 5-19 for each location and both pressure levels. For 80 bar (Figure 5-19 (a)), the good agreement between test and simulation results was achieved in the area of piston 2 (except for the top position towards disc OD), as well as for the upper part of the middle groove (fibre position). In the lower part of the middle groove and in the area under piston 1, the differences are more pronounced. Interestingly, looking back on Figure 5-18 (a), at the top position (towards the slot for wear sensor installation) and the centre of the piston 1, FBG and FEA results give opposite picture. FBG measurement indicates $152\mu\epsilon$ close to the groove and $364\mu\epsilon$ close to the centre of piston 1, whereas FE analyses predict almost exactly reversed values, i.e. $314\mu\epsilon$ at the proximity of the groove and $135\mu\epsilon$ in the centre of piston 1. Statistical table associated with Figure 5-19 (a) indicates strain differences of under $100\mu\epsilon$ for 45% of the positions, with only 18% being over $200\mu\epsilon$.

The situation is very similar for 120 bar actuating pressure (Figure 5-19 (b)), with some higher strain differences owing to the higher mechanical load (actuating pressure). The reasons for these differences can be numerous and will be investigated later. Uncertainties with FBG tests and FEA simulations have already been discussed.



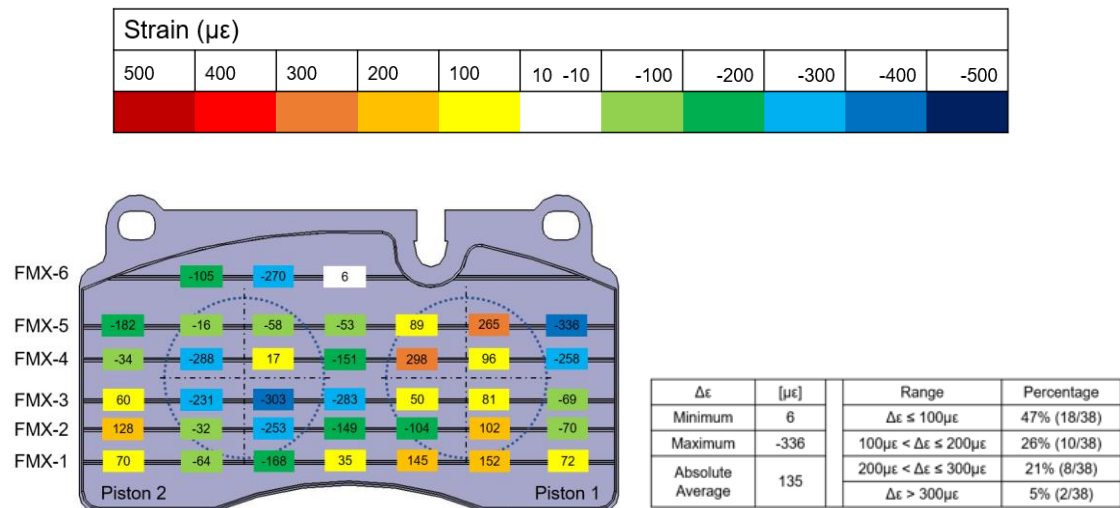
(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: Equal Pressure ($p_1=p_2$) at 80 bar Level



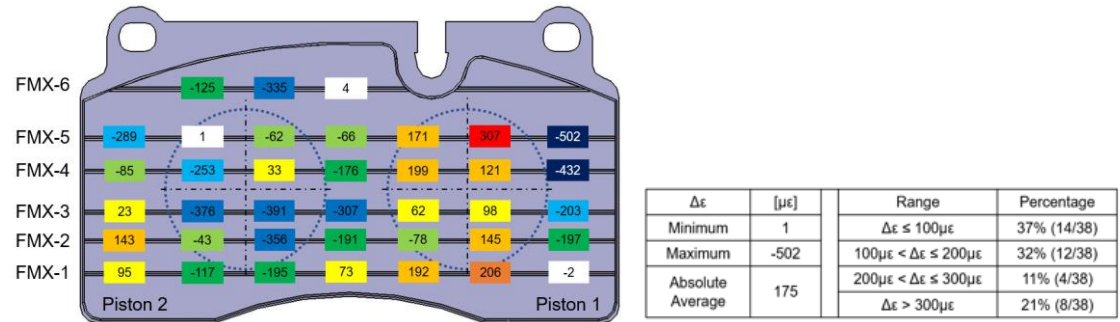
(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: Equal Pressure ($p_1=p_2$) at 120 bar Level

Figure 5-19: FBG and FEA E22 (Y Direction) Strain Differences for Two Hydraulic Pressure Levels

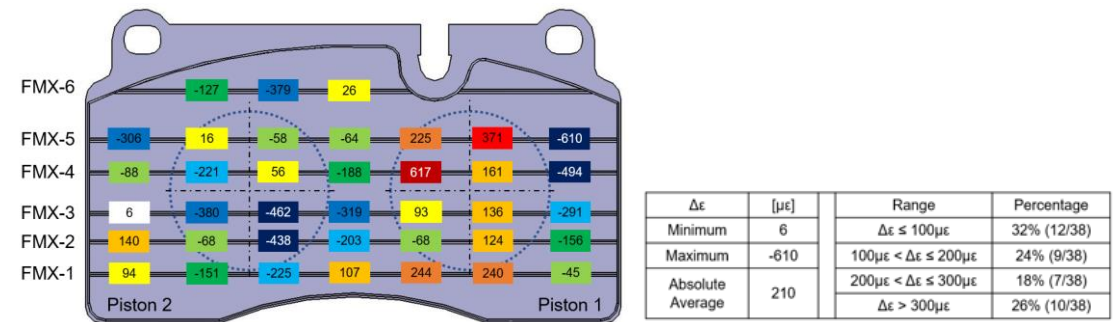
Figure 5-20 shows E11 strain data, obtained through tests and analyses, for pressure levels of 40 bar (a), 80 bar (b) and 120 bar (c). As before, FE analysis results seem more consistent, showing 'smoother' transitions and greater strain symmetry between the areas associated with the pair of two pistons. Such results would be 'expected' but a care should be exercised as the experimental uncertainties can be manifold and the friction materials are also well known to be inhomogeneous and have the granular structure. Saying all this, similarities between FBG measurements and FEA simulations are apparent, with higher



(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: Equal Pressure ($p_1=p_2$) at 40 bar Level



(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: Equal Pressure ($p_1=p_2$) at 80 bar Level



(c) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: Equal Pressure ($p_1=p_2$) at 120 bar Level

Figure 5-21: FBG and FEA E11 (X Direction) Strain Differences for Three Hydraulic Pressures

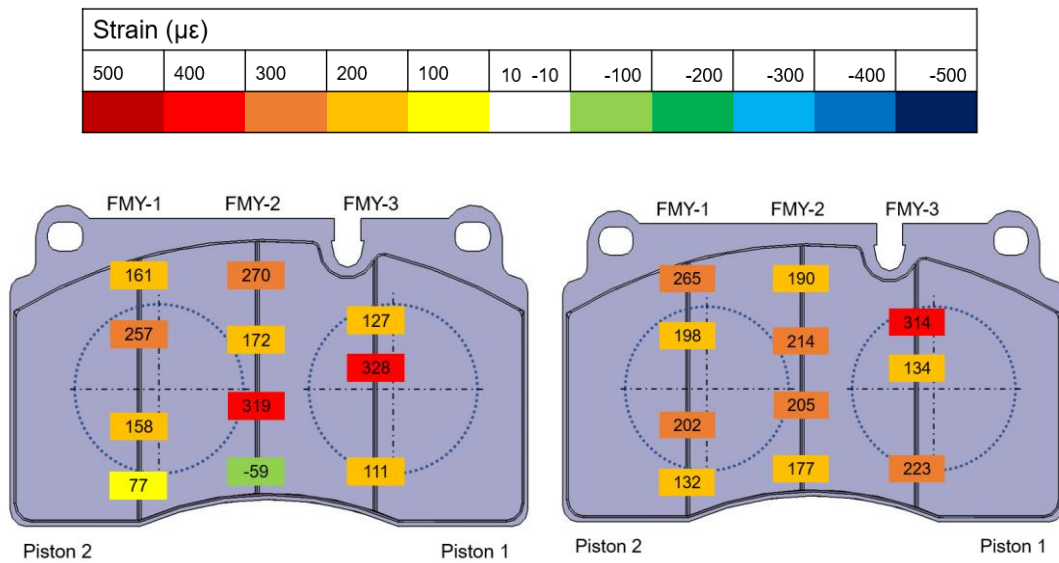
When looking at the maps and tables of E11 (X Direction) strain differences (Figure 5-21), they are higher than for E22 (Y Direction) strain (Figure 5-17 (d))

and Figure 5-19). However, the differences do not increase proportionally with the increment of the hydraulic pressure, from 40 bar to 120 bar. There are areas where the results are quite close and some with medium and large differences, but it is difficult to establish any pattern for such a phenomenon. Nearly identical behaviour can be noticed for all actuating pressures, confirming the robustness and reputability of the experimental work. For 40 bar hydraulic pressure, in 47% of the locations strain differences are under $100\mu\epsilon$, which reduces to 37% for 80 bar condition and further to 32% for 120 bar level, as shown in Figure 5-21.

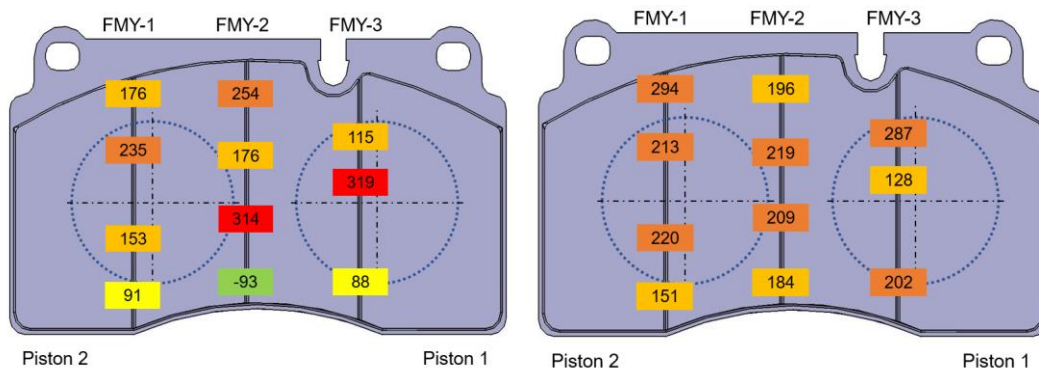
5.3.2.2 Hydraulic Pressure Variations

In section 5.2.2.2, the background of different hydraulic pressure variations between piston pairs has been discussed, from the FEA simulation standpoint. Here, FBG and FEA results will be compared for the nominal actuating hydraulic pressure of 80 bar, with the further results included in Appendix Chapter 5-G (40 bar) and Appendix Chapter 5-H (120 bar).

E22 (Y Direction) strain results of FBG and FEA are shown in Figure 5-18 (a) for the identical pressure of 80 bar for both sets of pistons. In Figure 5-22 (a), FBG and FEA comparisons of E22 (Y Direction) strain has been illustrated for 80 bar at piston 1 and 70 bar at piston 2, with Figure 5-22 (b) showing the reverse pressure levels ($p_1 = 70\text{bar}$ and $p_2 = 80\text{bar}$). In all cases, higher pressure results in higher strain levels, as it would be expected. The FBG and FEA results appear to be similar, except for several areas where larger differences can be observed.



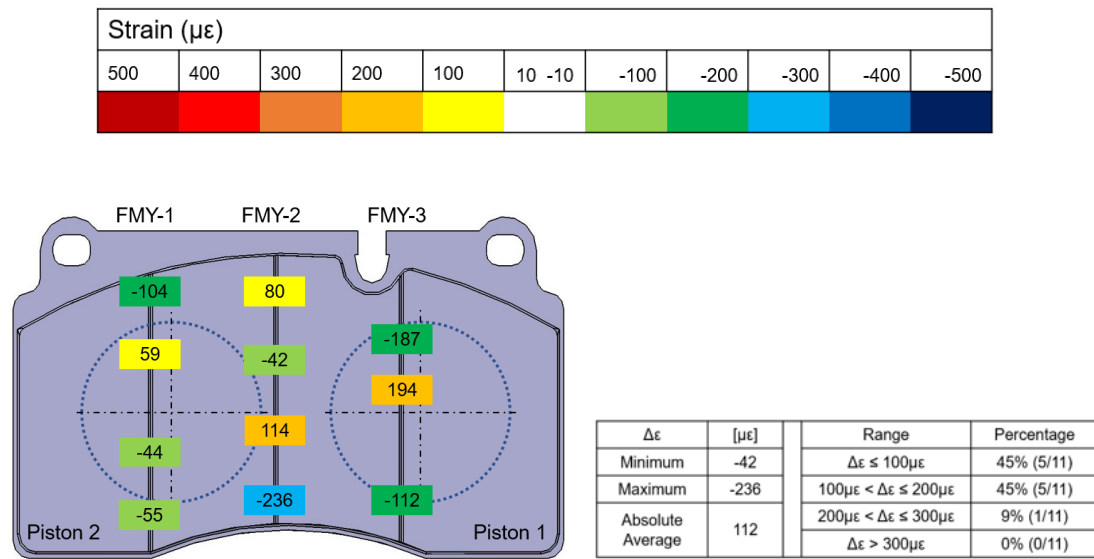
(a) FBG (Left) and FEA (Right) Results at $p_1 = 80\text{bar}$ and $p_2 = 70\text{bar}$



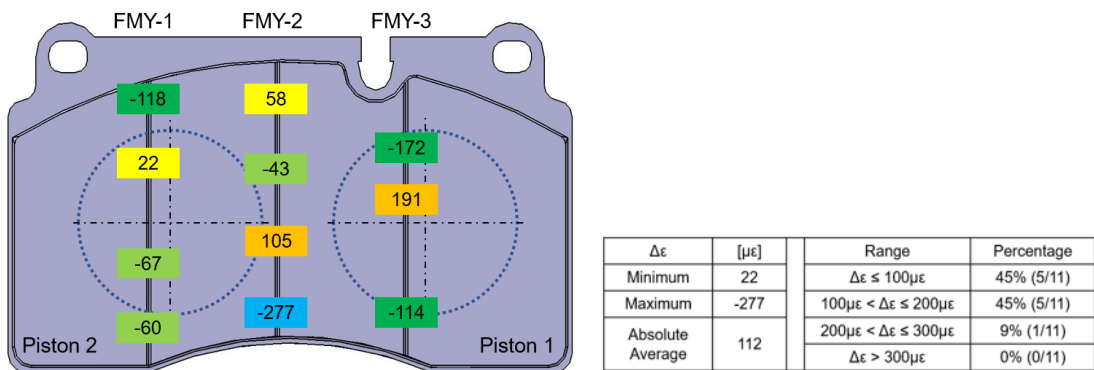
(b) FBG (Left) and FEA (Right) Results at $p_1=70\text{bar}$ and $p_2=80\text{bar}$

Figure 5-22: E22 (Y Direction) Strain Results for Two Hydraulic Pressure Variations

As before, this is most effectively analysed by looking at strain differences, as shown in Figure 5-23 (a) ($p_1 = 80\text{bar}$, $p_2 = 70\text{bar}$) and Figure 5-23 (b) ($p_1 = 70\text{bar}$, $p_2 = 80\text{bar}$). In both cases, looking from left to right, at piston 2 (FMY-1), FBG and FEA results are very similar, with the exception of the top positioned sensor, where differences are just over $100\mu\epsilon$. In the middle of the pad (FMY-2), good agreement is achieved towards the disc OD but this deteriorates towards disc inner diameter (ID) where the difference is over $-200\mu\epsilon$. At FMY-3, strain differences are quite pronounced, from around $100\mu\epsilon$ at disc ID, to nearly $200\mu\epsilon$ in the piston centre.



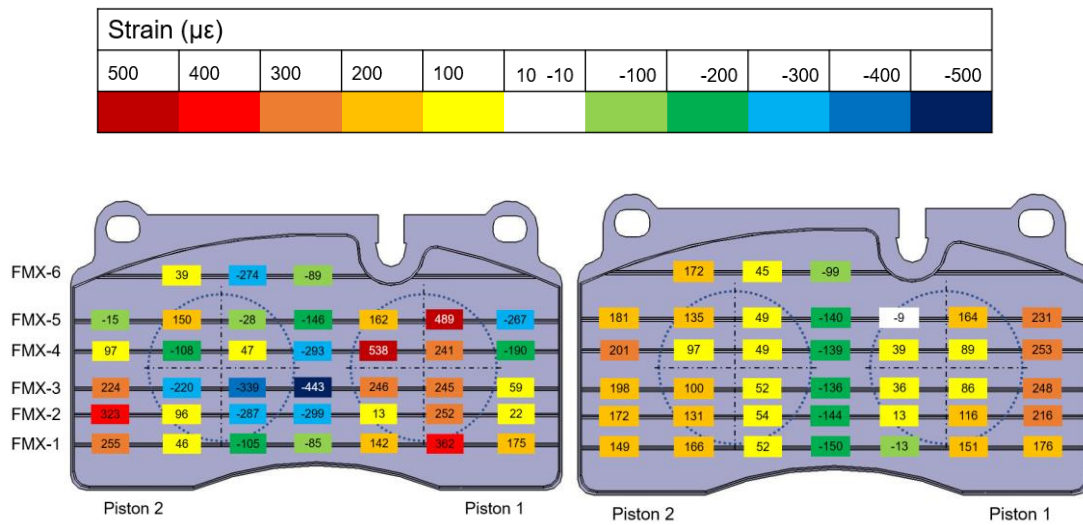
(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=80bar; p2=70bar



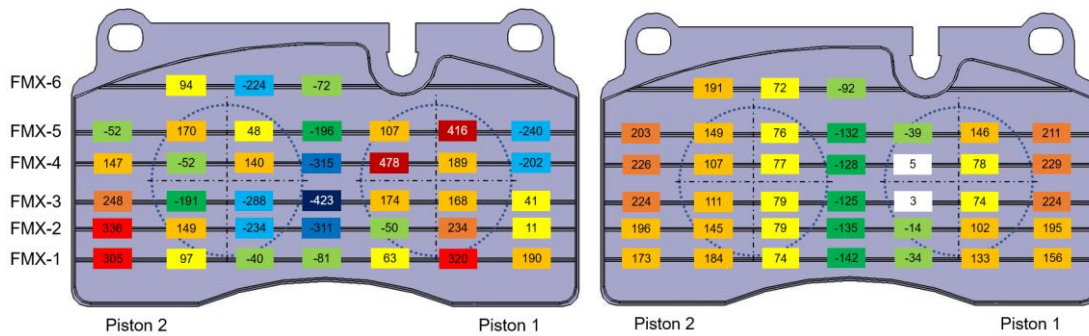
(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=70bar; p2=80bar

Figure 5-23: FBG and FEA E22 (Y Direction) Strain Differences for Two Pressure Variations

The analyses were also conducted for E11 strain (X direction), with FBG and FEA strain values shown in Figure 5-24 for two pressure variations. Again, the FEA results look more 'as expected', with FBG test data giving locally high values in some locations. This is particularly noticeable at the leading edge of the brake pad and under the piston 1.



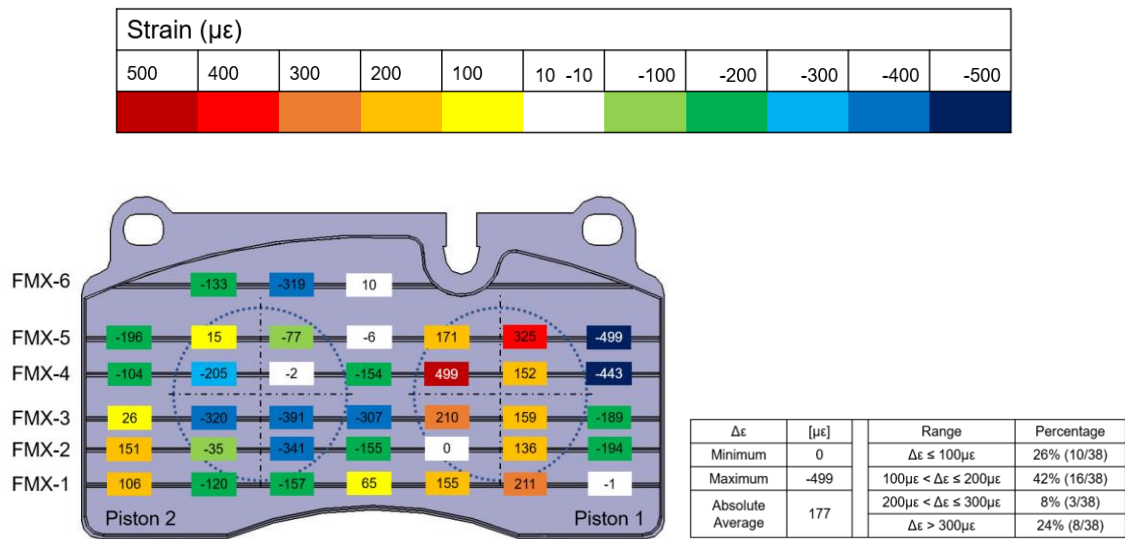
(a) FBG (Left) and FEA (Right) Results at $p_1=80\text{bar}$ and $p_2=70\text{bar}$



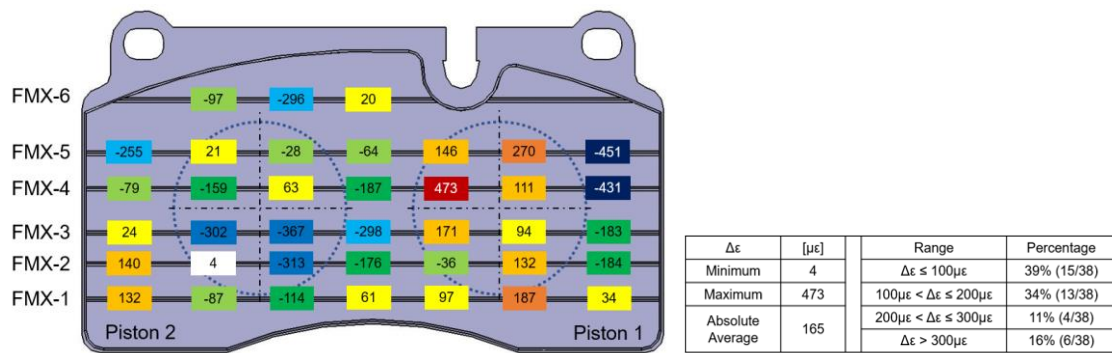
(b) FBG (Left) and FEA (Right) Results at $p_1=70\text{bar}$ and $p_2=80\text{bar}$

Figure 5-24: E11 (X Direction) Strain Results for Two Hydraulic Pressure Variations

Again, best insight into the discrepancies between FBG test and FEA simulation results can be obtained by showing the actual differences at all sensor locations (Figure 5-25). Despite the first visual impression indicating large differences, in 26% of the locations, the differences are under $100\mu\epsilon$ and in 42% between $100\mu\epsilon$ and $200\mu\epsilon$, when $p_1 = 80\text{bar}$ and $p_2 = 70\text{bar}$ (Figure 5-25 (a)). When $p_1 = 70\text{bar}$ and $p_2 = 80\text{bar}$ (Figure 5-25 (b)), the agreement is more favourable with 39% locations showing differences under $100\mu\epsilon$ and in 34% between $100\mu\epsilon$ and $200\mu\epsilon$.



(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=80bar; p2=70bar



(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=70bar; p2=80bar

Figure 5-25: FBG and FEA E11 (X Direction) Strain Differences for Two Hydraulic Pressure Variations

5.4 Summary

This chapter has described the application of the finite element (FE) analysis to support and complement the FBG test results under static loading conditions, which was part of the research aims. It developed and validated the FE model to simulate different test conditions and brake performances as the static experimental phase has done. In addition, the test results obtained using FBG strain sensors were compared and validated with the outcomes from the FEA simulations for static loading conditions. Typically, at approximately 45% of the

sensor positions, the differences between measured and predicted strain are under $100\mu\epsilon$.

The Finite Element (FE) model was established, with several major material characteristics, mesh, boundary conditions and analysis procedures investigated into some details. The results showed very good sensitivity to change of the brake pad loading (hydraulic pressure). In addition to strain values, the FEA simulation results were evaluated in terms of interface pressure distributions, stress components and pad deflection. All these are far too numerous to be included in the thesis but helped to build a confidence in the FE model and analyses, which can be therefore considered optimised and substantiated.

In order to compare FEA simulation results with FBG tests, the FEA obtained strain values have been processed and plotted at FBG sensor locations. In such a manner, very precise, direct result comparisons could be made. FBG and FEA strains were compared for a range of operating conditions, as previously tested on the test rig using FBG strain sensors. Hydraulic pressures of 40 bar, 80 bar and 120 bar were modelled to be both equal pressure variation (on two pairs of pistons), as well as to vary by ± 10 bar pressure variations.

The general observation can be made that the FEA results are much 'smoother', without sudden changes in strain values, as the brake pad geometry is relatively simple and uniform, and the applied loading is equally distributed over the piston/backplate contact areas. As discussed before, various parameters of model and analysis were varied to gain confidence in FEA results but the components were considered to have perfect geometry and the friction material to be homogeneous. FBG results vary more and in some locations the values are considerably lower or higher without an obvious reason. The uncertainties with FBG sensors and experimental works were discussed in detail in the previous section 4.3.4. It should be noted here that the measurements were very repeatable, typically within $\pm 3\%$, and strain values quoted are averages from three repeatable measurements.

Overall, it can be stated that FBG and FEA strain results are quite similar, with exception of the areas mentioned above. Some values are almost identical, in

most areas, the strain differences are relatively low to medium, with higher discrepancies limited to smaller areas. Both FBG and FEA results show the identical trends when loading (hydraulic pressure) is changed, being equally on all four pistons or varied between two piston sets.

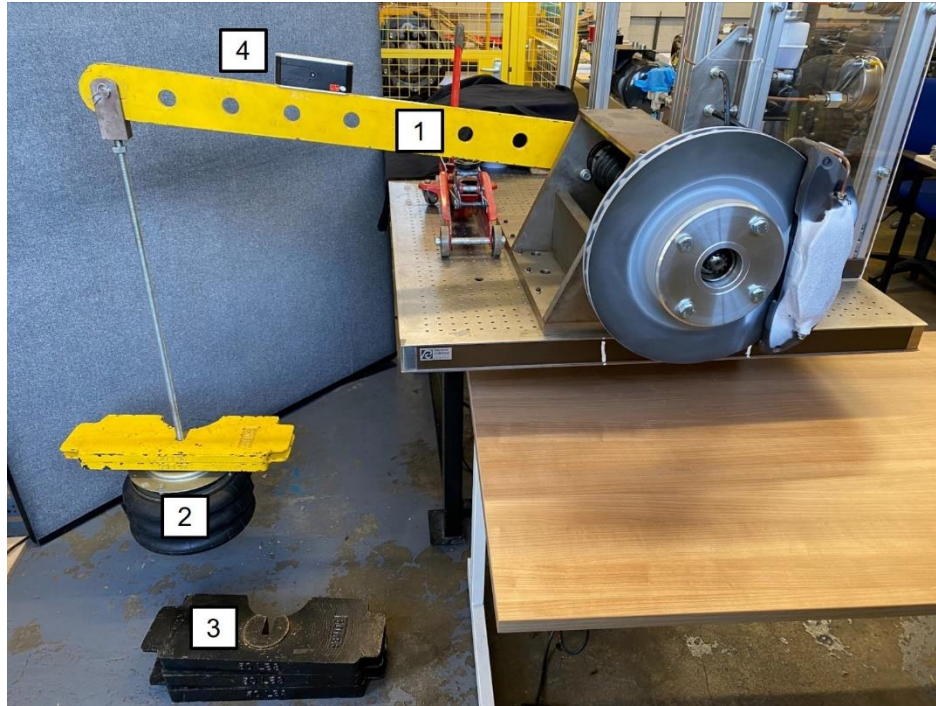
The above finding is very encouraging, as the changes in strain values can be further investigated in quasi-dynamic conditions and particularly in dynamic conditions. Tracing these changes in the dynamic conditions are crucial for monitoring processes at friction interface in actual service duties, which are used to determine brake performance, wear and NVH characteristics.

6 EXPERIMENTAL PHASE 2 – QUASI DYNAMIC ANALYSIS

This chapter will have a similar structure to that of Chapter 4, containing the system set-up, test plan and validation, and an analysis of the results. In this chapter, only the analysis of FBG test results will be presented. The data will be compared with the output from FE analysis results in Chapter 7.

6.1 Quasi Dynamic Test System Set-up

The test system set-up was based on the original static loading test rig, with a number of components added to allow the application of different torque levels. The torque application system included steel bar (1), load-bearing platform (2), load weights (3), and digital protractor (4) in Figure 6-1 (a). The steel bar used to apply the torque was 1m in length and weighted 10.65kg. The load-bearing platform weighted 8.35kg and acted as the platform to hold the load weights stably during these tests. There was an upward degree of the steel bar before applying the torque level to the Brembo Land Rover disc brake system to ensure once the torque has been applied, the load-bearing platform could still hang in the air until the load weights became stable to achieve the test requests. Figure 6-1 (b) shows a digital protractor with $\pm 0.05^\circ$ accuracy, which was stably positioned on top of the steel bar using magnets, as illustrated in Figure 6-1 (a). Such a component was used to observe the stability of the torque application system when load weights were added to the load-bearing platform, and the onset of the slip phenomenon happened in some test conditions, which might apply too much torque or low hydraulic pressure level.



(a) The Torque Application System: (1) Steel Bar; (2) load-bearing platform; (3) Load Weights and (4) Digital Protractor



(b) Digital Protractor

Figure 6-1: Quasi Dynamic Test System Set-up

The test components and load weights were weighed using weight scale and the values were recorded in Table 6-1 so that the specific torque level could be selected and combined to meet the test requirement. The steel bar was weighted 10.65kg where the 5.45kg was used in the torque calculation due to the central gravity of the steel bar. Such a steel bar was measured under a specific condition using a hanging hook weight meter where the steel bar was connected to the brake disc via the driveshaft and placed horizontally to the ground using the digital

protractor to maintain the degree level at 0° and then recorded the weight at that time period. The weight recorded from such a free initial condition could then be used as the reference to calculate the total torque level. Other set-ups would be the same as in the static loading test but needed to be calibrated each time before preparing for different experimental status, including the hydraulic pressure control system, the SmartScan FBG interrogator and the SmartSoft software.

Table 6-1: Load Weight and Test Component Values

Load Weight Values (1lbs = 0.454kg; Accuracy ± 0.05 kg)		
Nominal Value	Quantity	Actual Value
0.51kg	4	0.50kg
1kg	3	1.00kg
1.02kg	2	1.00kg
2kg	6	2.00kg
5kg	2	5.00kg
6kg	1	5.25kg
20lbs	1	9.05kg
10kg	2	1 for 10.00kg; 1 for 10.05kg
25lbs	2	11.35kg
50lbs	3	1 for 22.55kg; 2 for 22.60kg
Stand Carrier	1	8.35kg
Steel Bar with Inclinometer	1	10.65kg & 1m 5.45kg (Torque)

6.2 Fibre Bragg Grating (FBG) Test Plan and Validation

The following section will explain the main test conditions, along with the additional conditions used for further uncertainty analysis.

6.2.1 Main Test Conditions

Three hydraulic pressure levels were exerted during the static loading tests, 40 bar, 80 bar and 120 bar, with three pressure variations, equal pressure ($p_1 = p_2$), +10bar ($p_1 = p_2 + 10\text{bar}$) and -10bar ($p_1 = p_2 - 10\text{bar}$). Torques of 200 Nm, 600 Nm and 1000 Nm were applied under each condition as detailed in Table 6-2. Several initial attempts have been made to decide whether each pressure level

is corresponding to the torque value in general. The hydraulic pressure was applied with decrement trends for every 10 bar, and each level was maintained for 30 seconds. The responses of the brake pads with FBGs mounted in the vertical (Y) and horizontal (X) directions were investigated.

Table 6-2: Quasi Dynamic Loading Test Plan

FBGs Test Plan – Quasi Dynamic Loading Conditions (30s for each step)			
Hydraulic Pressure Range	Torque Level	Nominal Pressure Variations	Fibre Direction
40bar – 10bar (10bar Decrement)	200Nm (Actual 203Nm)	Equal Pressure ($p_1 = p_2$) +10bar ($p_1 = p_2 + 10\text{bar}$) -10bar ($p_1 = p_2 - 10\text{bar}$)	Vertical Direction (Y) Horizontal Direction (X)
80bar – 40bar (10bar Decrement)	600Nm (Actual 600Nm)		
120bar – 80bar (10bar Decrement)	1000Nm (Actual 1002Nm)		

Table 6-3: Applied Torque Level Combination

FBGs Test Plan – Applied Torque Level Combination				
Nominal Torque	Steel Bar Weight	Stand Carrier Weight	Load Weights	Measured Torque
200Nm	5.45kg	8.35kg	2kg*2; 1kg*2; 0.5kg*1	203Nm
600Nm			22.6kg*2; 1kg*1	600Nm
1000Nm			22.6kg*2; 11.35kg*2; 10kg*1; 5kg*1; 2kg*1; 1kg*1; 0.5kg*1	1002Nm

Those torque levels were applied by combining different load weights, which was illustrated in Table 6-3. The minimum applied torque was 138 Nm, as the combined weight of the steel bar (5.45kg) and the load-bearing platform (8.35kg) was 13.8kg. Therefore, a minimum applied torque level was determined to be 200 Nm to ensure that the whole torque application system could work properly and the load-bearing platform was able to maintain balance with load weights during different test conditions. The selected torque levels reflected the real brake performances as closely as possible. A 40 bar pressure with 200 Nm torque was corresponded to the daily driving behaviour during commuter time. An 80 bar pressure with 600 Nm torque simulates heavy brake activity, while a 120 bar pressure with 1000 Nm torque represents an unusual situation, for example being in an accident, where the brake application contains both the emergency stop

applied from the human activity and the anti-lock braking system (ABS) activated by the electronic control unit (ECU) of the on-board computer.

Each test began with the specified torque level for each test condition laid on the load-bearing platform in advance, lifting up the steel bar with these load weights to the upward degree (around 5°) by a jack. The hydraulic pressure was then applied but without the torque released in order to ensure that the test system and measurement instrumentation were operating correctly. After that, the torque was imposed by withdrawing the jack from the holding position of the steel bar, and the hydraulic pressure reduced in the 10 bar process as indicated earlier. Each test condition was repeated for three times, and the digital protractor was used to observe the brake slipping. A detailed discussion is provided in section 6.3.1 when presenting the test results and analysis from the python graph.

6.2.2 Additional Test Conditions

According to the main test conditions focused on the pressure decrement with a fixed torque level, an additional process was set with total inverse conditions, where the torque level was increased step by step under a constant hydraulic pressure.

Table 6-4: Additional Test Plan

FBGs Test Plan – Additional Quasi Dynamic Loading Conditions (30s for each step)			
Nominal Torque Level	Hydraulic Pressure	Nominal Pressure Variations	Fibre Direction
140Nm – 200Nm (20Nm Increment)	10bar	Equal Pressure ($p_1 = p_2$)	Vertical Direction (Y) Horizontal Direction (X)
150Nm – 600Nm (150Nm Increment)	40bar		
200Nm - 1000Nm (300Nm Increment)	80bar		

Generally, such tests operated the constant hydraulic pressure level as the first step, then added load weights to increase the torque level in a certain range, 20 Nm increment for total 200 Nm, 150 Nm increment for total 600 Nm, and 300 Nm increment for total 1000 Nm, respectively. At the same time, hydraulic pressure was maintained at 10 bar, 40 bar and 80 bar to correspond with each torque level, which is shown in Table 6-4. Therefore, these additional test results will be

discussed and analysed in section 6.3.4, which are used to investigate the reasonability of those selected test conditions provided in section 6.2.1.

6.3 Quasi Dynamic Loading Test Results and Analysis

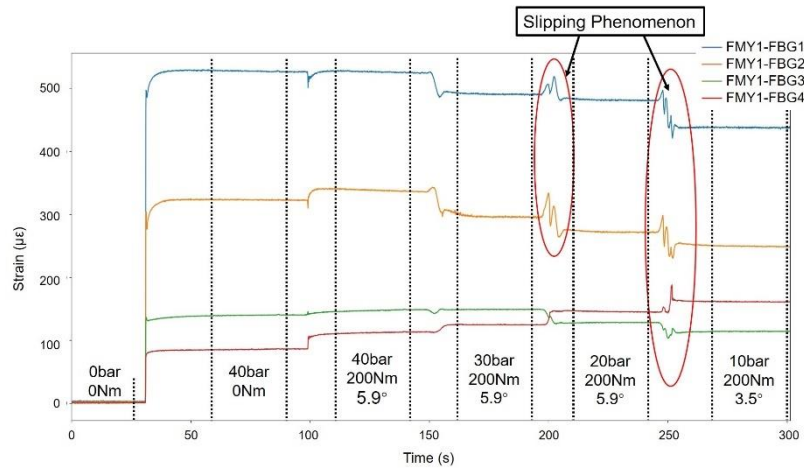
6.3.1 Results and Analysis: Python Graphs

As mentioned in sections 6.1 and 6.2.1, the slip phenomenon could be observed by monitoring the digital protractor. An error range of the equipment was $\pm 0.05^\circ$, which could be used to detect the beginning of the brake system slips when reducing the hydraulic pressure. The data from the FBG sensors on the fibre FMY1 at a number of pressure and torque levels are shown in Figure 6-2, with the red circle added to highlight the slipping. Each of these graphs shows the completed time series of one test, adding with the vertical dotted lines at the times to illustrate the change of the hydraulic pressure level.

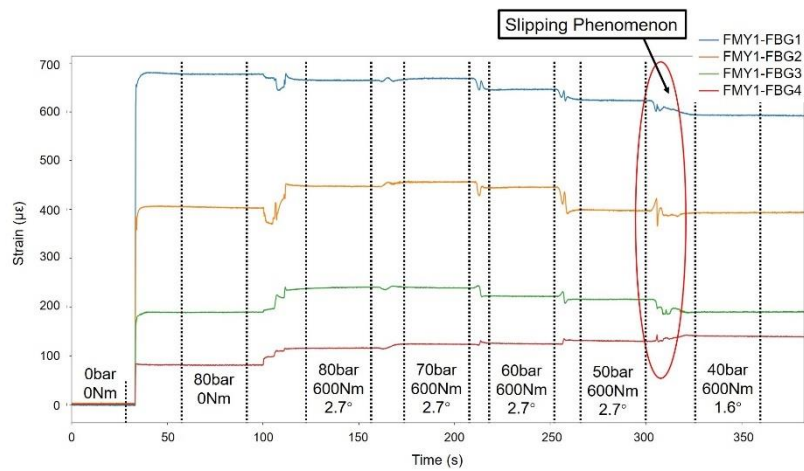
The Figure 6-2 (a) displays clearly that a slipping event occurred before applying 10 bar hydraulic pressure with 200 Nm torque for all four FBG sensors of the FMY1 fibre. Also, the data from FMY1-FBG1 and FMY1-FBG2 exhibits the signature of the slipping when the hydraulic pressure was reduced from 30 bar to 20 bar with the same torque level, which suggests that the top half of the brake pad slips first, then affecting the rest of the area of the pad.

In Figure 6-2 (b), with an applied torque of 600 Nm, slipping is observed on all sensors along the optical fibre only when the hydraulic pressure is decreased to 40 bar. There is no evidence of the slippage in the data where the initial hydraulic pressure was 120 bar with 1000 Nm applied torque (Figure 6-2 (c)), however, this test condition might not add enough torque to make the slippage happen to be observed at either the digital protractor or the result data level.

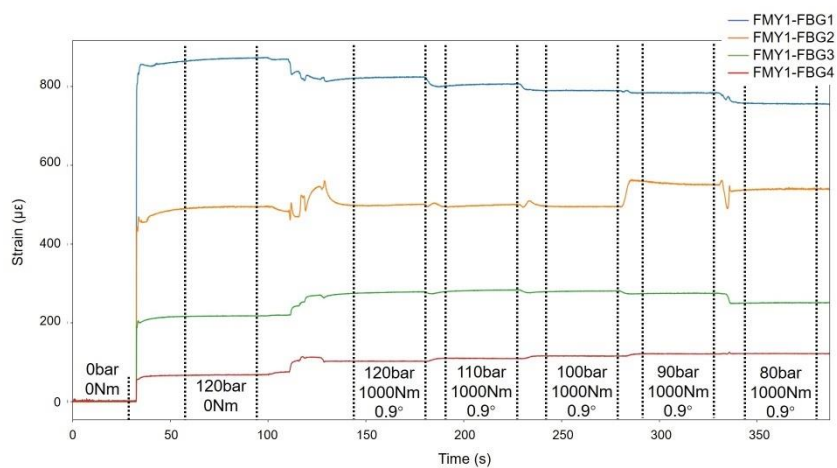
According to the time series shown in Figure 6-2, the slipping phenomenon is most likely to happen at lower pressures with the FBG strain sensors being able to capture this at an early stage. Both digital protractor changes and the measured data results are used to ensure such pressure and torque applications are suitable for the existing test environments and conditions.



(a) Quasi-dynamic FMY1: 40bar with 200Nm



(b) Quasi-dynamic FMY1: 80bar with 600Nm



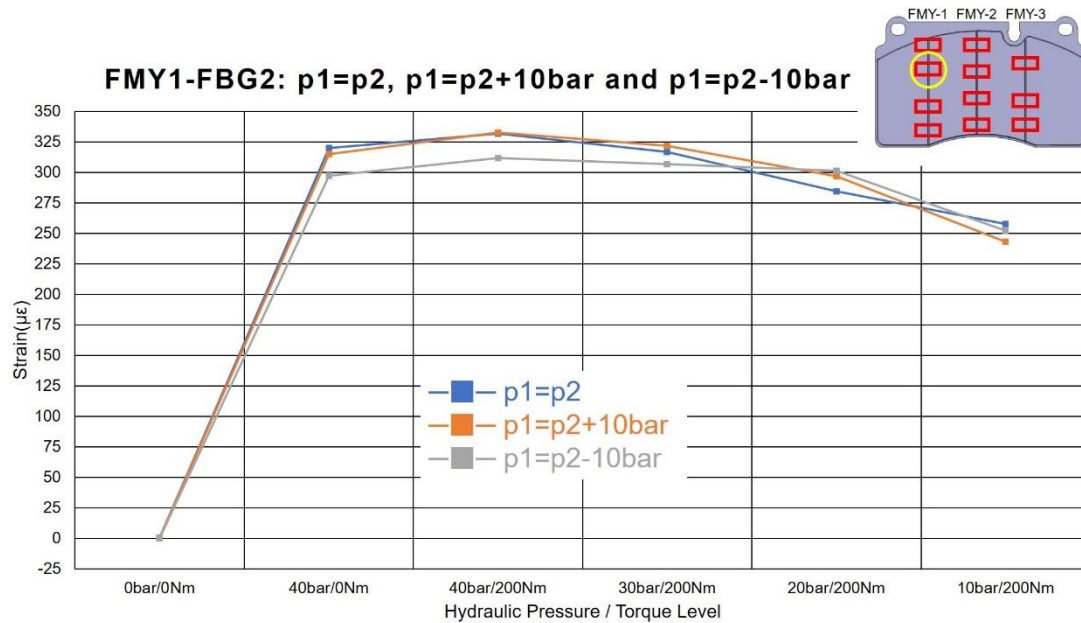
(c) Quasi-dynamic FMY1: 120bar with 1000Nm

Figure 6-2: The Slipping Phenomenon during Quasi Dynamic Loading Tests

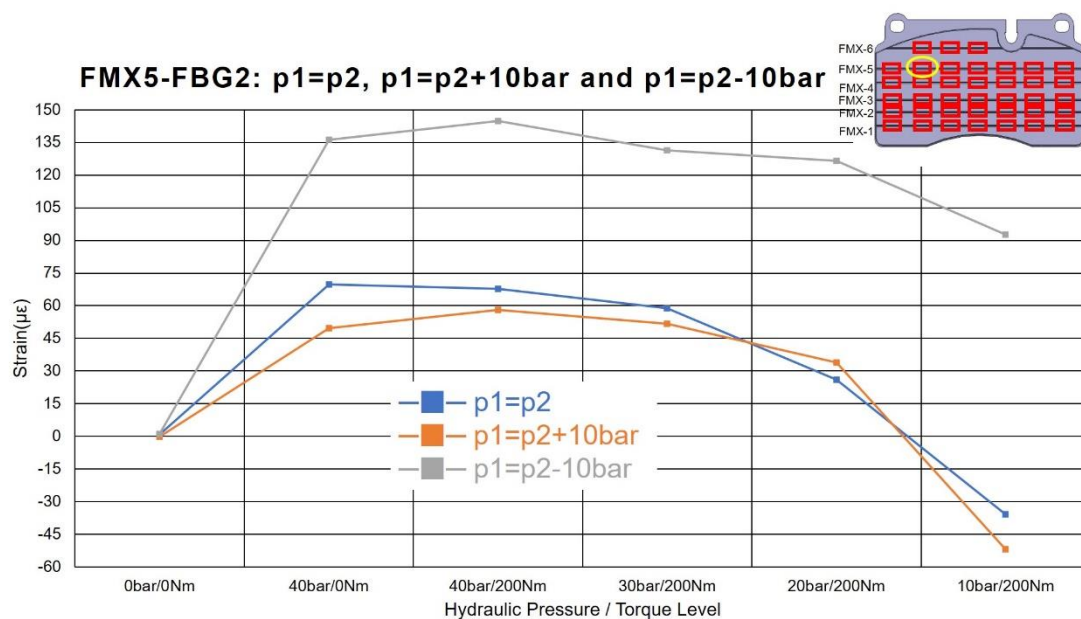
6.3.2 Results and Analysis: Recording and Distribution Graphs

According to the data processing step described in section 4.3.1, there are two ways to observe the test results, which are the same as section 4.3.2 illustrated, the test recording graph and the strain distribution graphs. In order to assess each sensor response to quasi-dynamic loading conditions with three pressure variations, equal pressure, +10bar and -10bar, the data obtained from the FBG arrays with the test condition, 40 bar pressure with 200 Nm torque, are shown in Figure 6-3. The data acquired under the other test conditions can be found in Appendix Chapter 6-A (600 Nm) and Appendix Chapter 6-B (1000 Nm). The icon located at the top right points of the sensor location on the brake pad indicates its orientation, while the graph displayed the measured strain for each test step for three test demands, with an average data of three repeated measurements.

The data for FBGs oriented in the vertical (Y) direction showed similar values of all pressure variation conditions with the maximum difference being about $20\mu\epsilon$. However, the data for the FBGs mounted in the horizontal (X) direction displayed a significant difference for the pressure variation settings. In particular, the data for -10bar ($p_1 = p_2 - 10\text{bar}$) condition, which was affected by the sensor selection as sensor FMX5-FBG2 was close to the location of the piston 2 area. Data obtained for the other two variation conditions showed similar trends with the orange line (+10bar condition) being lower than the blue line (equal pressure condition) for almost all stages of the test, with the exception of the 20bar baseline pressure level where the slipping had occurred, combining with the red circle illustrated in Figure 6-2 (a). Such a sample suggested that the FBG sensor application can also be used to determine interface strain distributions under the quasi-dynamic loading conditions.



(a) Vertical (Y) Direction: Hydraulic Pressure – Strain (FMY1-FBG2)

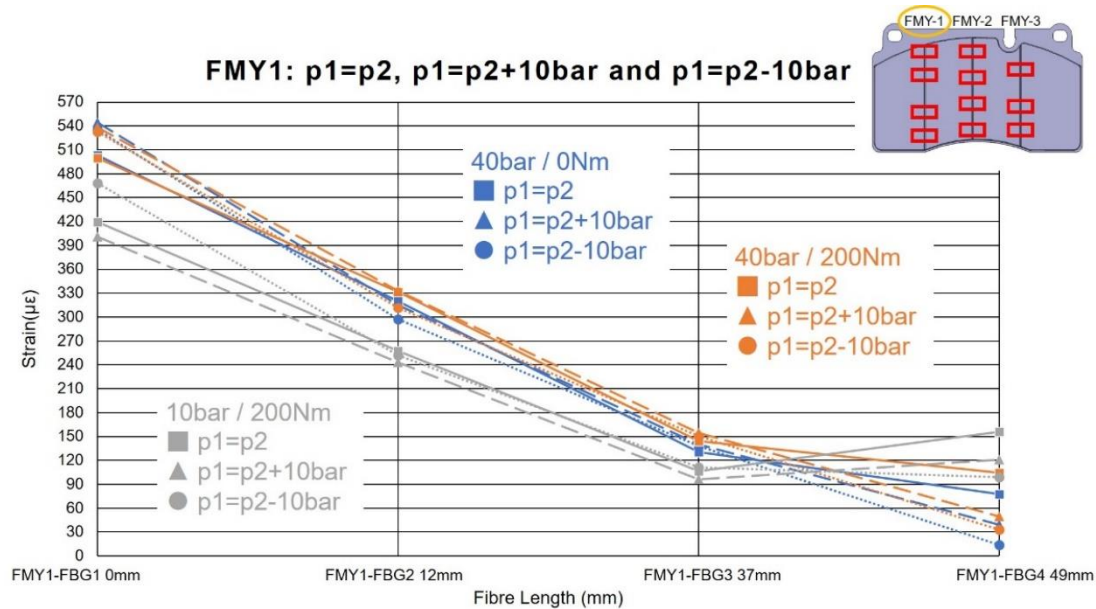


(b) Horizontal (X) Direction: Hydraulic Pressure – Strain (FMX5-FBG2)

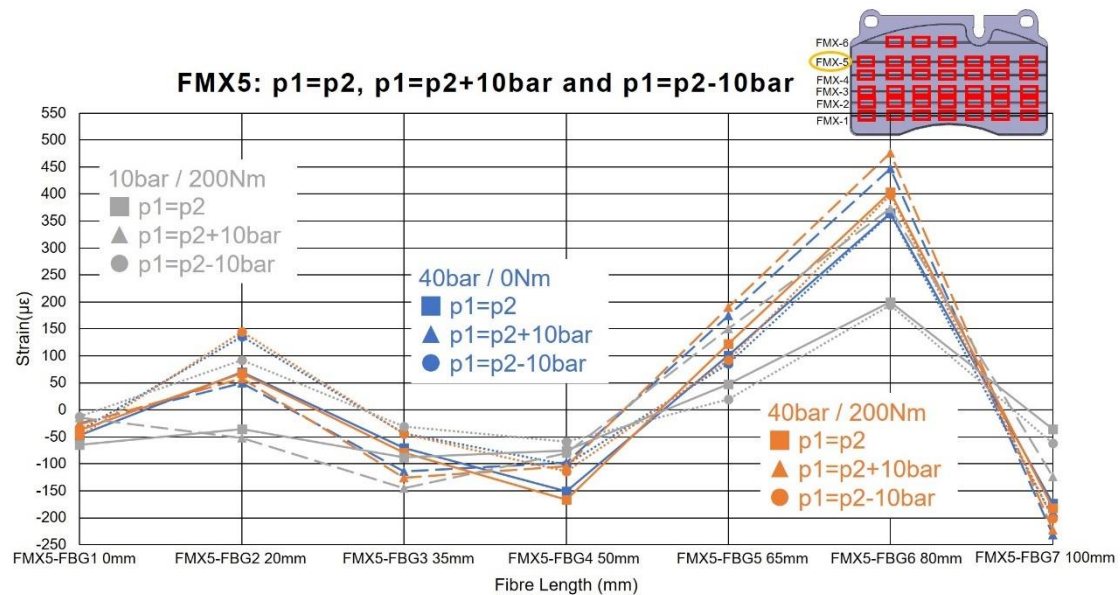
Figure 6-3: Recording Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface

An example of the visualisation of the strain distributions along a single array of sensors is shown in Figure 6-4 (a) for FMY1, oriented in the vertical (Y) direction, and Figure 6-4 (b) for FMX5, oriented in the horizontal (X) direction. Here, the top

right icon indicates the FBG sensor array under considerations. The plots allow the strain distributions for the three pressure variations (equal pressure, ± 10 bar) at three pressure and torque levels (40bar/0Nm, 40bar/200Nm and 10bar/200Nm) to be compared.



(a) Vertical (Y) Direction: Fibre Length – Strain (FMY1)



(b) Horizontal (X) Direction: Fibre Length – Strain (FMX5)

Figure 6-4: Strain Distribution Graphs for the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface

In Figure 6-4 (a), the data for sensors mounted in the vertical (Y) direction illustrated with a trend that strain level changed from high to low along with the sensor array (FBG1 to FBG4), and three pressure variations obtained the similar strain results. Only sensor FBG4 at the 10 bar pressure with 200 Nm torque showed a different change that the strain rebounded higher level than the FBG3 strain data.

For the array oriented in the horizontal (X) direction, the results shown in Figure 6-4 (b) were obtained under the same test condition at three sensors on FMX5 (FBG2, FBG4 and FBG6) showing different trends, which could be discussed to establish what happened at these sensor locations. Both FMX5-FBG2 and FMX5-FBG6 sensors were positioned around the piston area, where the strain results were continuously in positive trends. In particular the sixth sensor (FBG6) obtained the highest strain level when applying the 200 Nm torque to the 40 bar baseline hydraulic pressure level, as the explanation presented in section 4.3.2 that there is a 3mm offset from the centre to the right edge of the brake pad. Also, the FBG4 sensor always displayed the negative trends due to the squashing phenomenon of the brake pad when applying pressure and torque levels with different variation conditions, and maintained the same performance illustrated in Figure 4-6 for static loading test results.

Meanwhile, considering the performance shown in Figure 6-4, the slip occurred when the hydraulic pressure was reduced to 10 bar. The FBG strain results under this condition correspond to only the influence of the pressure as, following the slippage, the load-bearing platform has in contact with the floor, and thus no torque would be experienced by the pad.

6.3.3 Results and Analysis: FBG Sensor Contour Map

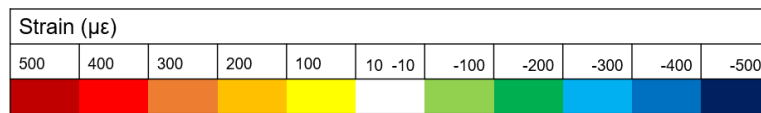
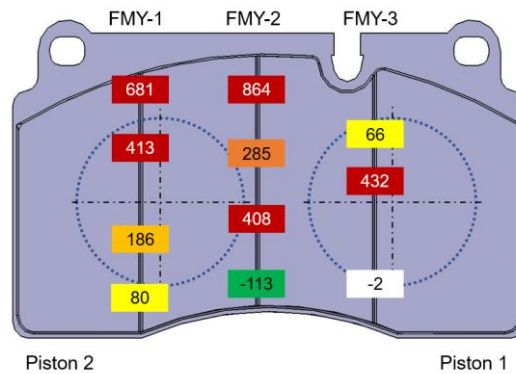
The FBG sensor contour maps employ the same stages to present test results as those graphs used in section 6.3.2; the static stage, the loading stage, and the final stage. The test condition used to analyse and discuss these plots is the 80 bar hydraulic pressure and 600 Nm torque level, which are considered to keep the same condition presented in section 4.3.3 when taking the static loading tests, and those test conditions are also corresponding to heavy brake applications,

which have enough representativeness in the real working conditions. The spectrum bar used here is the same as that used for the static loading analysis in section 4.3.3.

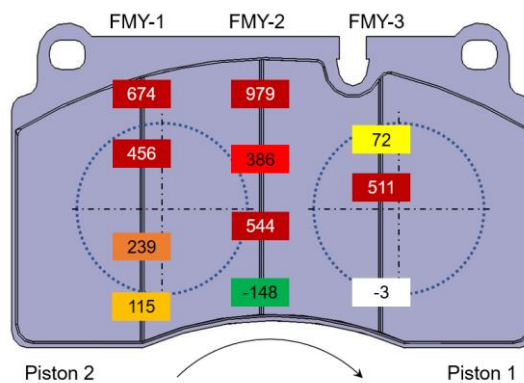
Generally, the following comparisons are divided into three groups for the vertical (Y) and horizontal (X) directions, including three test stages with equal pressure, three pressure variations with the same pressure and torque level, and three test conditions with equal pressure. Figure 6-5 and Figure 6-6 show three stages of the data from the sensors mounted in the vertical and horizontal direction for the specific case of 80 bar equal pressure ($p_1 = p_2$) applied to each pair of pistons with 600 Nm torque. The data obtained from other test conditions can be found in Appendix Chapter 6-C (40bar/200Nm). and Appendix Chapter 6-D (120bar/1000Nm).

The data illustrated in Figure 6-5 and Figure 6-6 show the strain levels for both vertical (Y) and horizontal (X) direction at three stages during a single quasi-dynamic test, including 80 bar with 0 Nm (the static stage), 80 bar with 600 Nm (the loading stage), and 40 bar with 600 Nm (the final stage), respectively. The strain results in both figures were increased when adding the torque application to the disc brake system, and then reduced during the hydraulic pressure release process until the finished 40 bar pressure level.

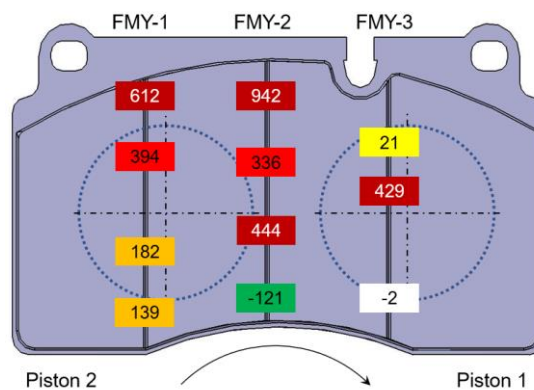
The maximum increment in vertical (Y) direction was about $140\mu\epsilon$ at the FMY2-FBG3 sensor, meanwhile, the maximum decrement was $100\mu\epsilon$ at the same sensor, such a sensor position was located in the middle of the brake pad, containing the main loadings during the applied test. The strain changes in horizontal (X) direction were not significant when the torque has been applied to the system, which was generally around $50\mu\epsilon$ in several positions, while the maximum strain reductions were observed at the left side of the brake pad, containing three sensors: FMX1-FBG1, FMX2-FBG1 and FMX3-FBG1, and these changes were from about $110\mu\epsilon$ to $140\mu\epsilon$.



(a) 80bar Hydraulic Pressure with 0Nm Torque Level

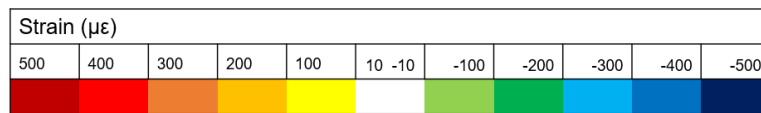
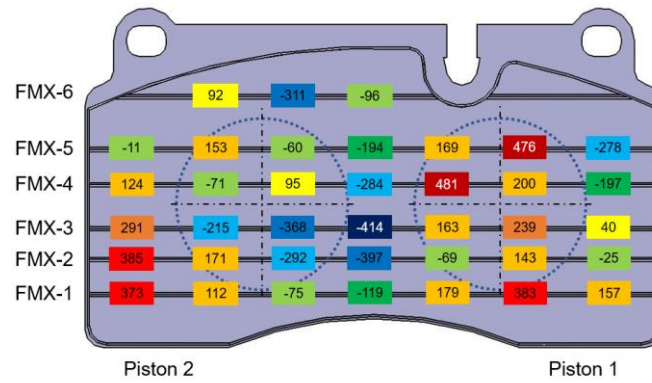


(b) 80bar Hydraulic Pressure with 600Nm Torque Level

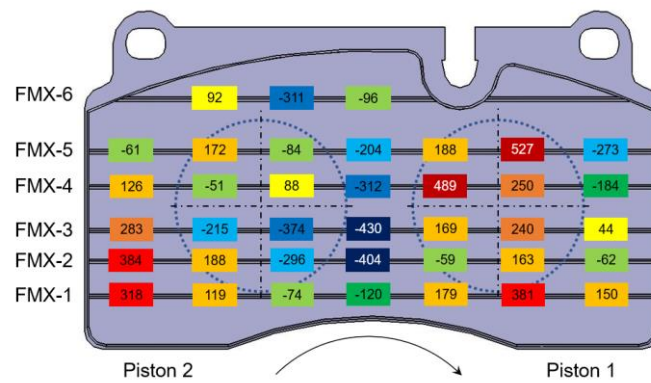


(c) 40bar Hydraulic Pressure with 600Nm Torque Level

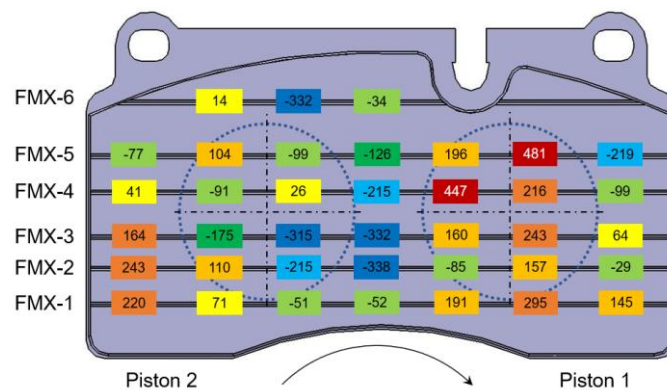
Figure 6-5: FBG Sensor Contour Maps for Vertical (Y) Direction Comparison between Three Test Stages with Equal Pressure ($p_1=p_2$)



(a) 80bar Hydraulic Pressure with 0Nm Torque Level



(b) 80bar Hydraulic Pressure with 600Nm Torque Level



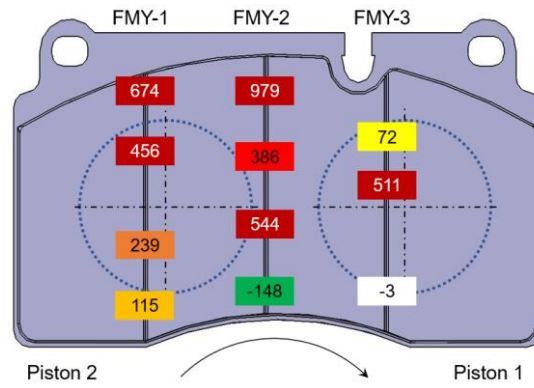
(c) 40bar Hydraulic Pressure with 600Nm Torque Level

Figure 6-6: FBG Sensor Contour Maps for Horizontal (X) Direction Comparison between Three Test Stages with Equal Pressure ($p_1=p_2$)

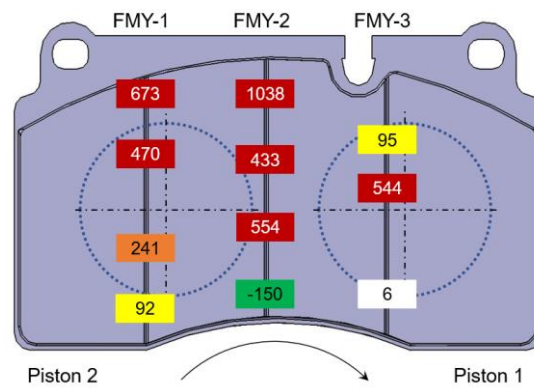
The data obtained from the sensors oriented in the horizontal (X) direction reveal further information on the influence of pistons on the measured strain under torque applications. The contour maps indicated that the FBG sensors near piston 1 area always showed higher strain levels than those near piston 2 area. One reason for this could be the design of the Brembo Land Rover disc brake system, where the brake pad is not located symmetrically to the pistons and has a 3mm offset from the middle to the right end of the brake pad. The other reason would be the disc rotation direction, which was clockwise, or from piston 2 to piston 1. Therefore, the torque application first contacted with the left edge of the brake pad and then was extended to the whole friction surface. Such applications might be lifted or bent the brake pad during different test conditions due to squash trends of the friction material.

Figure 6-7 and Figure 6-8 compare the strains measured under the different pressure variations with the same test condition, 80 bar hydraulic pressure with 600 Nm, for the vertical (Y) and horizontal (X) directions, respectively. The contour map results of other test conditions are provided in Appendix Chapter 6-E (40bar/200Nm) and Appendix Chapter 6-F (120bar/1000Nm). Generally, the data from the vertical (Y) direction does not show significant strain changes, in particular, the maximum difference was around $70\mu\epsilon$ observed at FMY2-FBG1, in response to these three pressure variations. While the data obtained in the horizontal (X) direction displays an obvious movement between the left- and right-hand regions of the brake pad when applying different pressure variations under the same pressure and torque levels. Several sensor positions located around piston 1 area has shown the maximum strain changes from about $140\mu\epsilon$ (FMX5-FBG6) to $180\mu\epsilon$ (FMX4-FBG5).

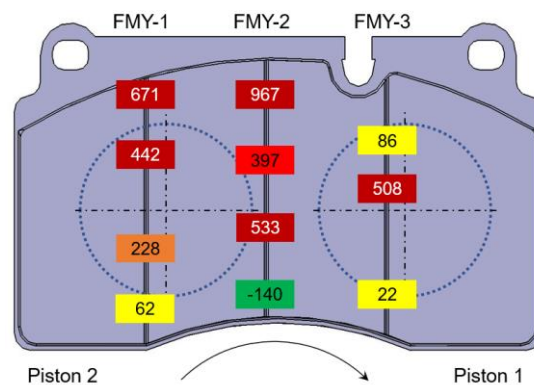
Strain ($\mu\epsilon$)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)

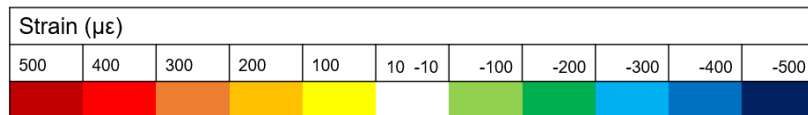
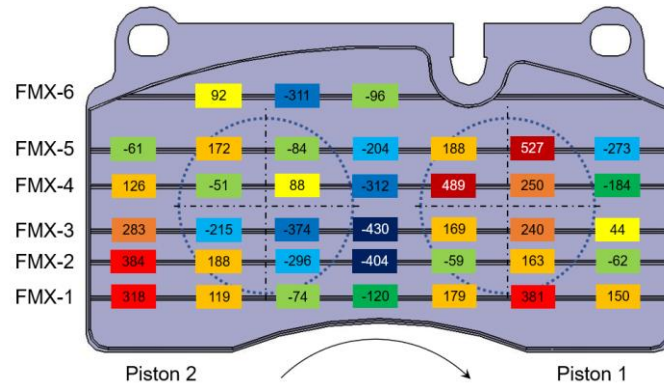


(b) +10bar Variation ($p_1=80\text{bar}$; $p_2=70\text{bar}$)

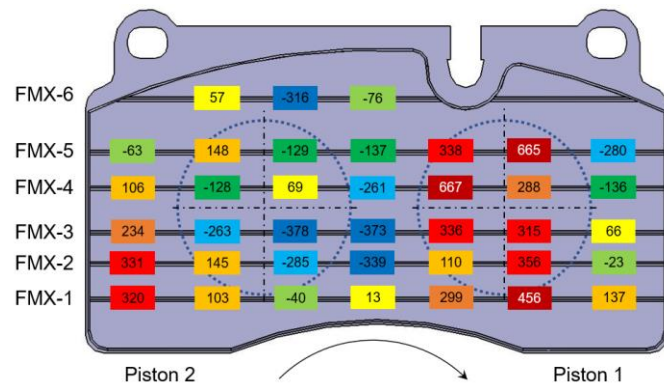


(c) -10bar Variation ($p_1=70\text{bar}$; $p_2=80\text{bar}$)

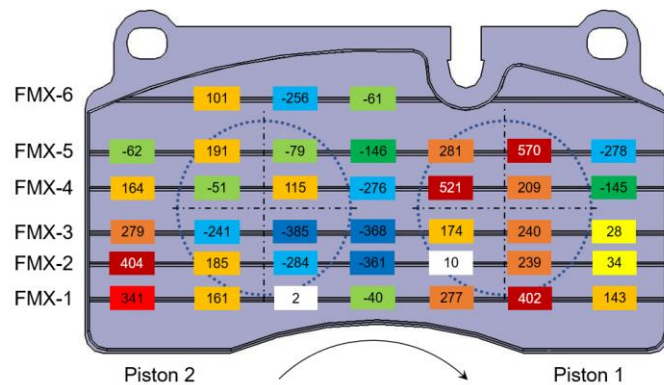
Figure 6-7: FBG Sensor Contour Maps for Vertical (Y) Direction Comparison between Three Pressure Variations Applied with 600Nm Torque Level



(a) Equal Pressure ($p_1=p_2=80\text{bar}$)



(b) +10bar Variation ($p_1=80\text{bar}$; $p_2=70\text{bar}$)



(c) -10bar Variation ($p_1=70\text{bar}$; $p_2=80\text{bar}$)

Figure 6-8: FBG Sensor Contour Maps for Horizontal (X) Direction Comparison between Three Pressure Variations Applied with 600Nm Torque Level

The data from the vertical (Y) direction has been viewed where small changes occurred, but it is insufficient to discuss these variations with the same test condition due to the application of torque in the clockwise direction. Therefore, the analysis and discussion presented here will then mainly focus on the data obtained from the horizontal (X) direction.

Looking into the horizontal data results, the measured strain levels show the influence of the three pressure variations applied during the load stage. The sensors mounted around piston1 provide a clear view that strain levels were higher under the +10bar variation than they were under the equal pressure condition. The data at the -10bar variation condition was not increased significant, comparing with the data at the equal pressure condition, but the interface strain distributions were still moved slightly to the left side of the brake pad, in particular, these sensors around the central area of the brake pad contain FBG3, FBG4 and FBG5 on each fibre array.

Therefore, these quasi-dynamic experiments have shown that FBG strain sensors can be allowed to measure and distinguish interface strain distributions between different test conditions. Thus, it can be concluded that optical fibre sensor technique can be used to make measurements and determinations under quasi-dynamic loading conditions. The data illustrated in this way will be compared with the simulation results from FE analysis, which will be discussed in Chapter 7.

6.3.4 Uncertainty Analysis: Experimental Phase

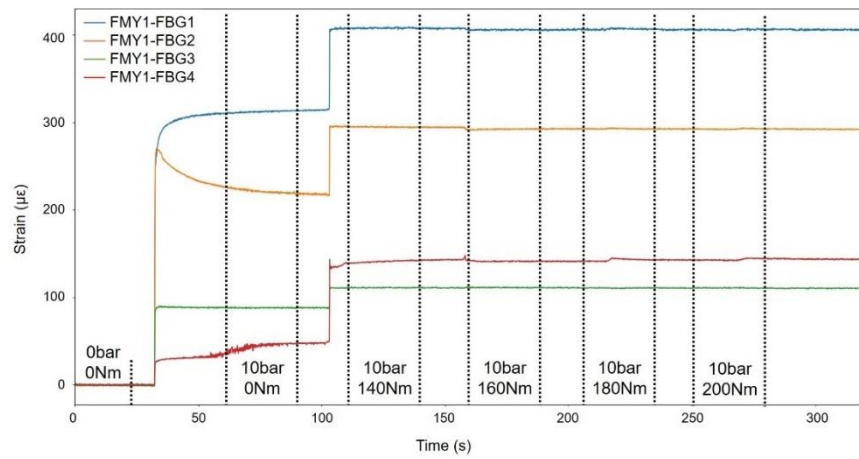
Several aspects of the uncertainties in the FBG strain measurements have been discussed at the static loading cases in section 4.3.4, therefore, this section will mainly focus on those factors revealed by the quasi-dynamic loading tests.

One factor was the test condition selection, where there were mainly two approaches to apply the quasi-dynamic loading tests, either fixing the hydraulic pressure with changing torque levels, or fixing the torque level with changing hydraulic pressures. The following discussion will illustrate the results of the additional test condition, which has been described in section 6.2.2.

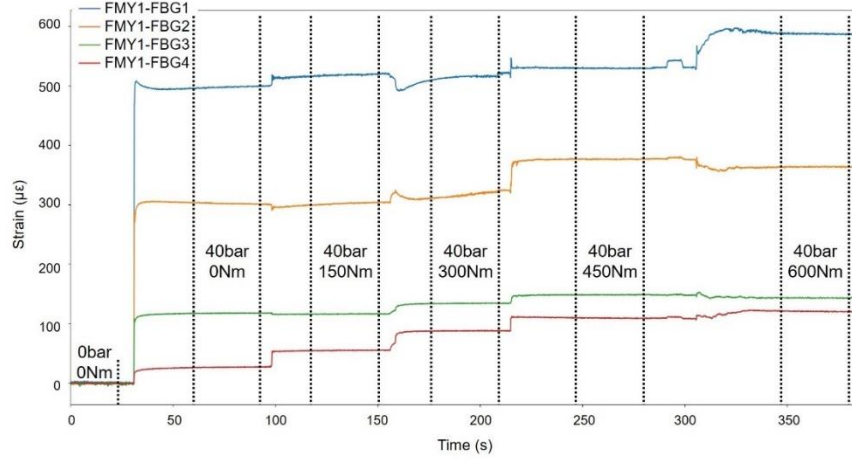
Generally, the additional tests could be divided into three conditions, 140 Nm to 200 Nm at the 10 bar hydraulic pressure level for each 20 Nm torque increment, 150 Nm to 600 Nm at the 40 bar hydraulic pressure level for each 150 Nm torque increment, and 200 Nm to 1000 Nm at the 80 bar hydraulic pressure level for each 300 Nm torque increment respectively. Figure 6-9 shows the data obtained under these conditions, with equal pressure ($p_1=p_2$) variation for the fibre array (FMY1).

According to the three graphs provided in Figure 6-9, it was impossible to observe that the change of the torque did not have a significant influence on the data. The recorded data did not increase step by step when the load weights were added, and even the 10 bar pressure with 200 Nm torque and the 80 bar pressure with 1000 Nm torque maintained similar strain level trends during the whole test process. While the 40 bar pressure with 600 Nm torque was observed an increase trend when adding the torque level from 450 Nm to 600 Nm, where the brake slip might happen at such a load testing stage.

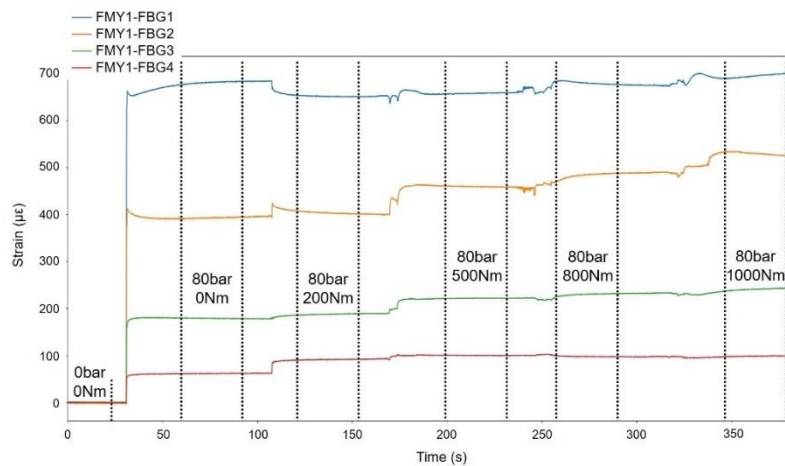
Therefore, such strain results of the additional tests can hardly be used to analysis and compare with other test conditions, when uncertainty for the quasi-dynamic loading requirements might occur. Therefore, the main test analysis and discussion only focused on the parameter changes of the pressure level and variation with the constant torque level to ensure that the data could be obtained and used to present the application of the optical fibre sensor technology in the disc brake research field.



(a) FMY1: 200Nm Torque Level with 10bar Hydraulic Pressure



(b) FMY1: 600Nm Torque Level with 40bar Hydraulic Pressure



(c) FMY1: 1000Nm Torque Level with 80bar Hydraulic Pressure

Figure 6-9: Additional Tests of Three Pressure and Torque Level Conditions

Other factors influencing the uncertainties include the bonding of the sensor issue and the material properties. The FBG strain sensors were bonded in the designed grooves using cyanoacrylate adhesive. Due to the small dimensions of the grooves (1mm in width and depth), areas in these grooves might contain small humps as the friction material was made with various kinds of raw materials, the strain at some of the sensor positions therefore might be influenced by these matters. This would be a particular problem under quasi-dynamic conditions where the torque applied to the brake disc would squeeze the friction surface in both the applied pressure direction and the torque load direction, unlike the static loading condition where there was one single force direction. Meanwhile, the visco-elastic characteristic also has to be considered during the quasi-dynamic tests when reducing pressure levels to the final aimed stage, where the detailed discussion has already been presented in section 4.3.4.2 when analysing the static loading analysis.

6.4 Summary

This measurement of the interface strain distributions at the brake pad frictional surface under quasi-dynamic conditions, which was part of the research aims, has been described in this chapter. The data reinforces the conclusion that FBG strain sensors are capable of the measurement of brake interface strain, and shows that the techniques used for the installation of the FBG sensors was appropriate to allow measurements under quasi-dynamic test conditions.

Generally, the quasi-dynamic loading conditions focused on determining the response to different pressure and torque level, as well as various pressure variations. Some optimisations have been selected to simulate the actual working conditions to make sure that the loading conditions applied in this chapter were suitable and reasonable. Besides the main test conditions, additional tests have been applied to enable an understanding of the uncertainties and causes of inconsistent results.

The quasi-dynamic loading analysis investigated the brake pad strain characteristics under several test conditions, including various pressure and torque levels, and three pressure variations, with FBG sensors oriented along

both the vertical and horizontal directions on the brake pad friction material. The test results provide high ranges of sensitivity, and could be used to simulate the brake functions and explain brake applications in normal driving activities. Although most of these FBG strain sensors worked properly, fewer sensors occasionally reflected abnormal data under some test conditions due to the fact that these sensors might be located in some areas where the grooves support the optical fibre array poorly.

In summary, applications of FBG strain sensors could be used in both static and quasi-dynamic loading conditions. Such an innovative technique provides enough possibilities to be applied during these fundamental brake research works, and can even be applied during the manufacturing process in industry.

7 FE ANALYSIS PHASE 2 – QUASI DYNAMIC ANALYSIS

The quasi-dynamic FEA presented in this chapter follows a similar approach and format to that presented in Chapter 5 on the FEA for static loading. This chapter will address three main aspects, FE model update and optimisation, simulations and analysis types, and comparisons between FBG tests and FEA simulations. The analyses were set up similarly to those in Chapter 5, to ensure a coherent approach, obviously with necessary modifications to account for a different type of loading. Namely, in addition to the actuating hydraulic pressure applied to the calliper pistons, a torque is applied to the disc.

7.1 Finite Element Model

7.1.1 FE Model Update

The FE model created for the static loading simulations was explained in detail, from the design concept to the optimisation, in section 5.1. Here, the static FE model was updated in several aspects to make it suitable for the quasi-dynamic analysis discussed in this section.

The FE model updates were mainly related to the brake disc and the corresponding boundary conditions (BCs) were differently formulated. The disc section used in the static analysis only covered the pad contact area (Figure 7-1 (a)). However, for quasi-dynamic conditions, a complete brake disc has been created, as shown in Figure 7-1 (b).

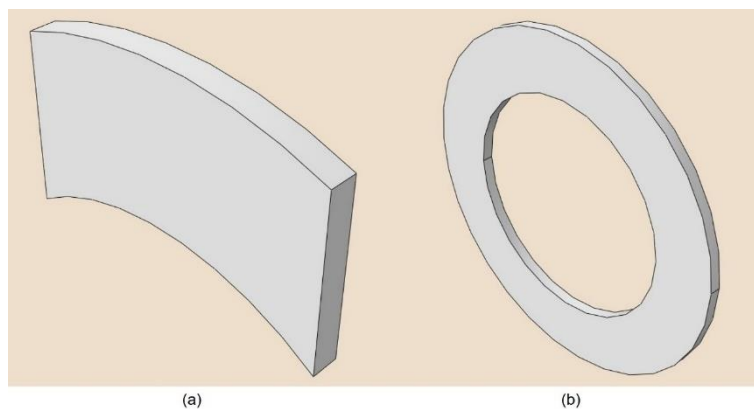


Figure 7-1: The Disc Model: (a) Static Analysis and (b) Quasi Dynamic Analysis

For the disc applied in quasi-dynamic analysis, the disc thickness was kept the same as for the static model (Figure 7-2 (a)), while the nominal element size was increased from 3mm to 4.5mm (Figure 7-2 (b)).

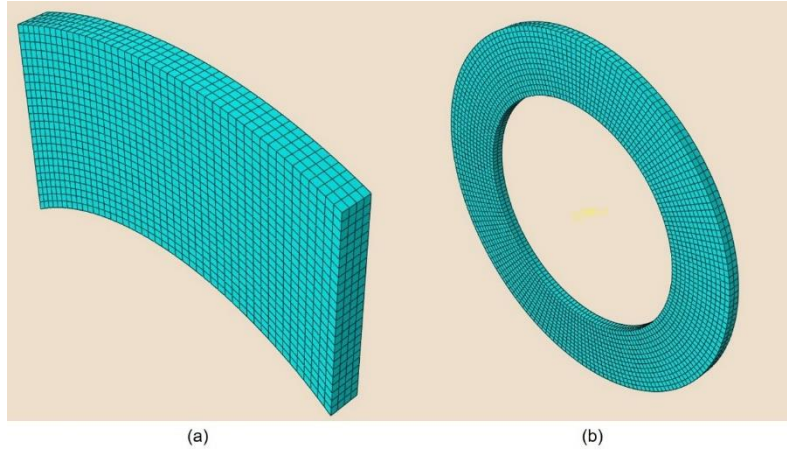


Figure 7-2: The Element Size: (a) Static Analysis and (b) Quasi Dynamic Analysis

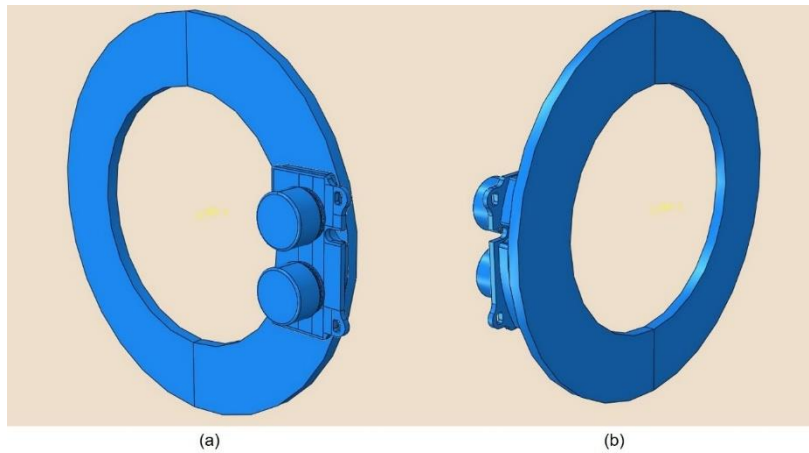
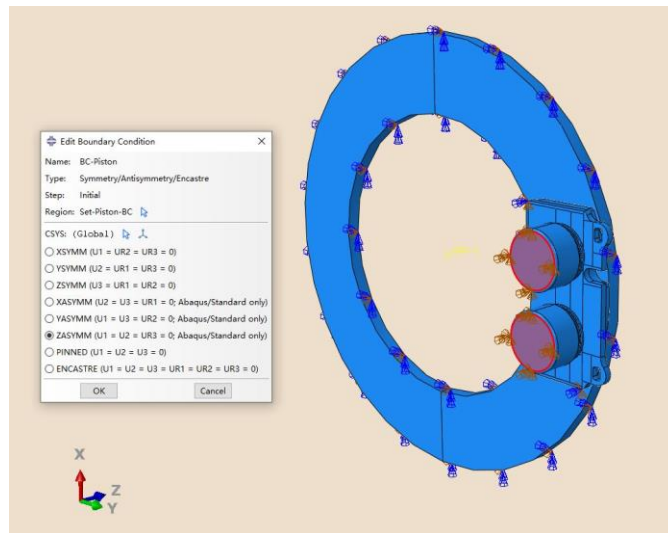


Figure 7-3: The Assembly View: (a) Piston Side View and (b) Disc Side View

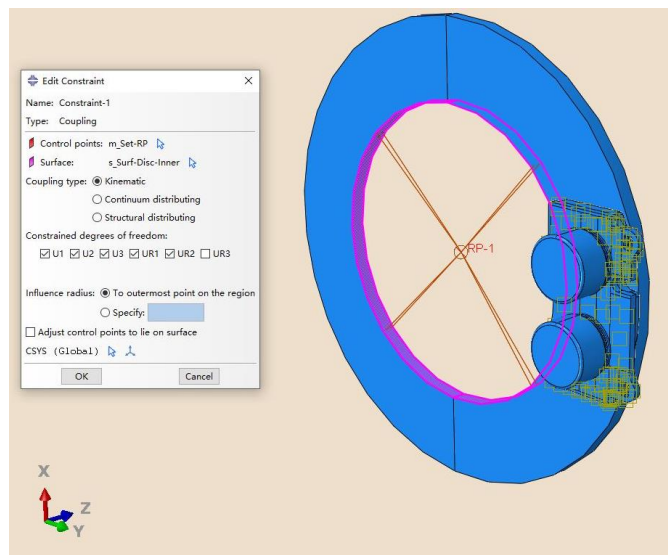
The updated quasi-dynamic FE model is displayed in Figure 7-3. The brake pad and pistons being the same as the static FE model, while the disc has been replaced by the complete brake disc, as discussed above. Again, only half of the brake assembly is modelled, assuming symmetry in relation to the brake disc mid-plane.

The main BCs and loads were similar to the original static FE model explained in section 5.1.1, and some updates were made to suit the quasi-dynamic analysis demands. The BCs on the pistons were the same as the original static model,

where the pistons could only move in the Z direction of the applied pressure (Figure 7-4 (a)). The BCs of the disc included two aspects, the constraint point (axis) and the surface BCs. The constraint point was the central point of the brake disc model, i.e. in the disc axis, allowing the disc only to rotate around its axis (Figure 7-4 (b)).



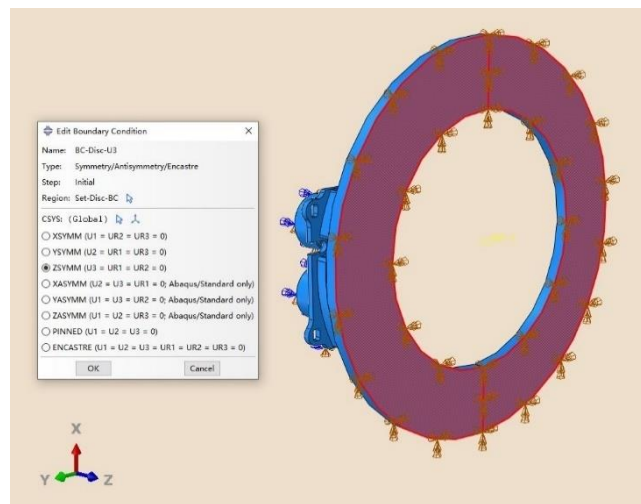
(a) BCs on The Pistons



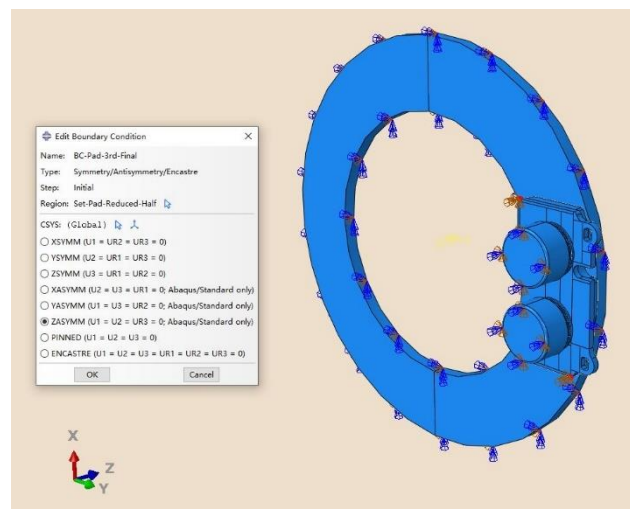
(b) Constraint on The Disc

Figure 7-4: Boundary Conditions (BCs) of The Quasi-Dynamic FE Model (Pistons and Disc Settings)

As for the static analyses, the disc mid-plane displacement was restricted in Z direction (Figure 7-5 (a)), allowing the disc to ‘slide’ in this plane, i.e. rotate around its axis (the constraint point). This enabled various torque levels to be applied to the disc, as explained later on. Extra BCs were imposed on the backplate of the brake pad, so that the pad was not free in both the X and Y directions, which meant the brake pad only moved in the same direction as the two pistons (Figure 7-5 (b)). This was necessary in order for the backplate to react to braking torque applied to the disc.



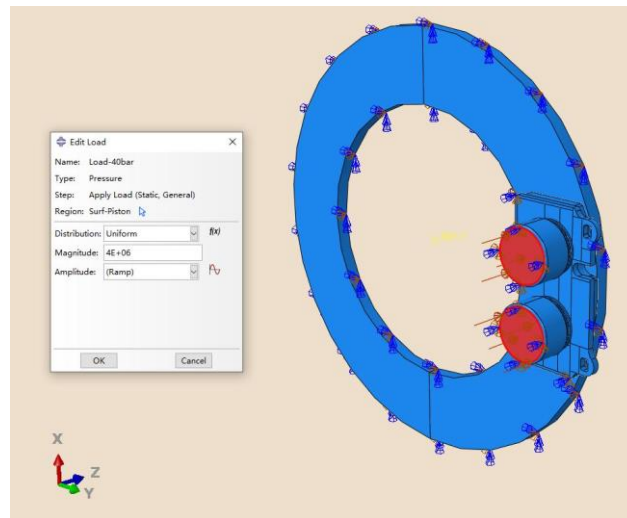
(a) BCs on The Disc



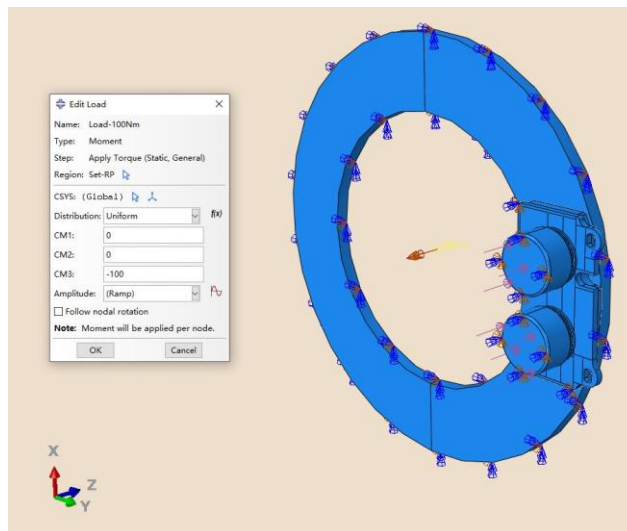
(b) BCs on The Backplate of The Brake Pad

Figure 7-5: Boundary Conditions (BCs) of The Quasi-Dynamic FE Model (Disc and Brake Pad Settings)

The loads applied in the updated quasi-dynamic FE model were placed in two areas, hydraulic pressure was applied to the pistons (as per the original static FE model, Figure 7-6 (a)), in addition, a torque was applied to the disc by using the moment load type (Figure 7-6 (b)). This caused disc rotation in the anticlockwise direction, matching the actual testing condition. The applied torque level was set at half of the required test value as the FE model contained only half of the disc brake assembly. The optimisation of the BCs and loads are explained and discussed later in the next section 7.1.2.



(a) Loads on Pistons



(b) Loads on The Disc

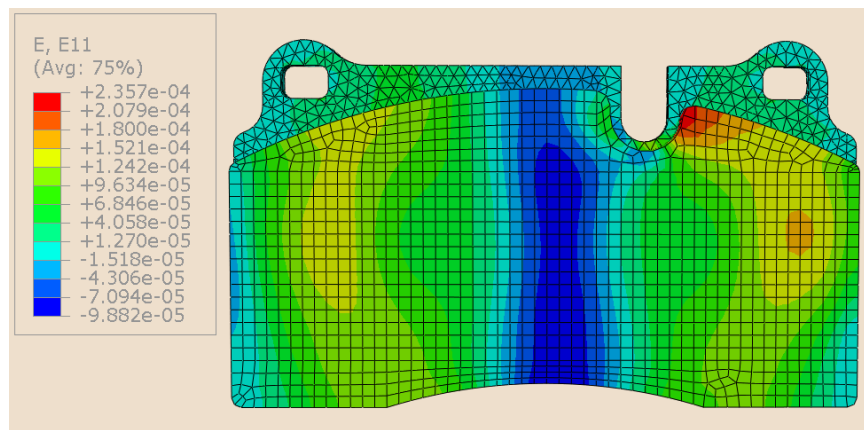
Figure 7-6: Loads of The Quasi-Dynamic FE Model

7.1.2 FE Model Optimisation

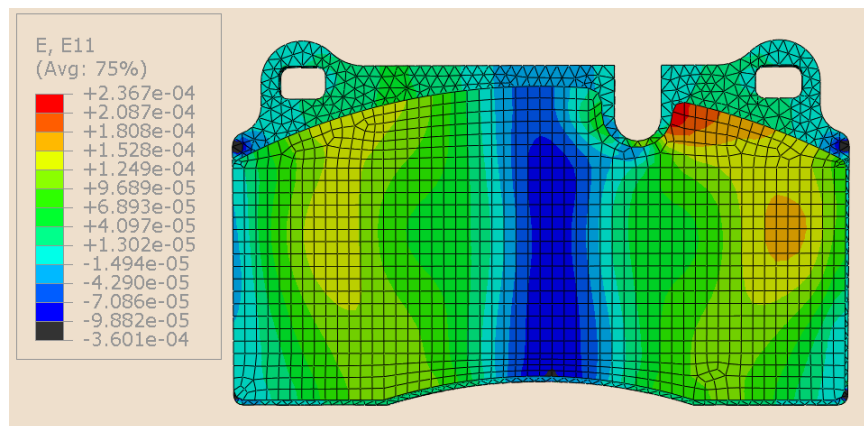
The previous section illustrated the updated FE model for the quasi-dynamic loading analyses. Therefore, the following sections will present the modelling optimisation aspects, disc mesh (element) size, torque loading direction, friction coefficient, and backplate boundary conditions, respectively.

7.1.2.1 Disc Mesh (Element) Size

As disc element size was increased from 3mm for static to 4.5mm for quasi-dynamic analyses, comparative analyses were conducted, and the obtained results evaluated. For the actuating pressure of 40 bar, under static conditions, Figure 7-7 (a) and Figure 7-7 (b) show strains in X direction (E11), with Figure 7-8 (a) and Figure 7-8 (b) in Y direction (E22).

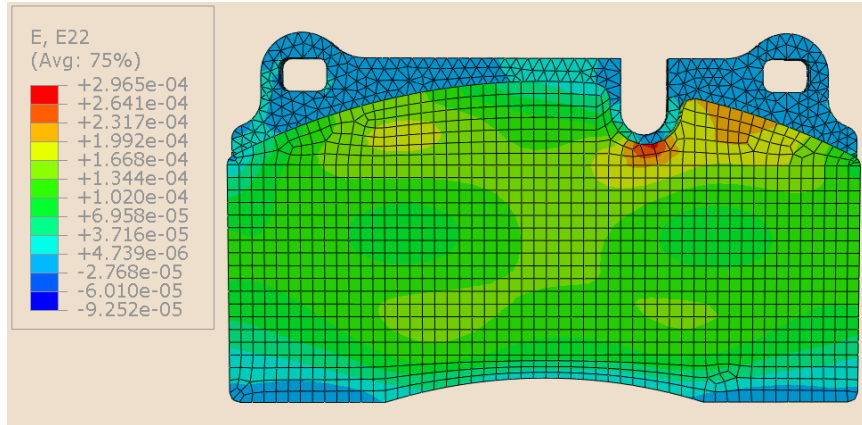


(a) 3mm Disc Section

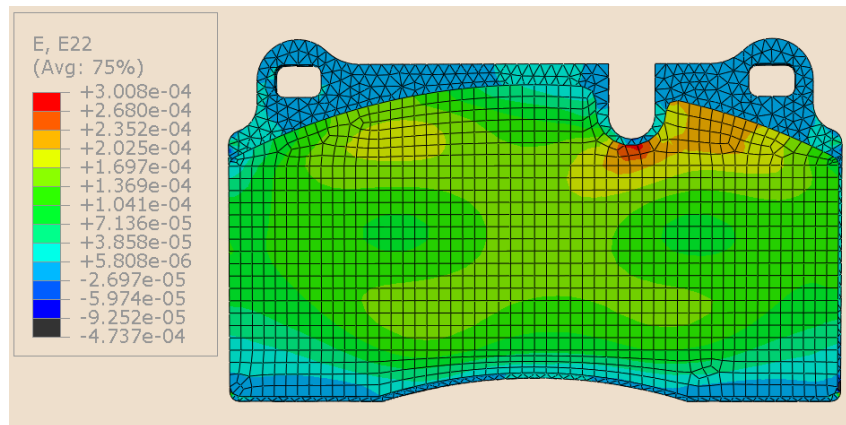


(b) 4.5mm Complete Disc

Figure 7-7: FE Analysis Results for Different Element Sizes with 40bar (X Direction)



(a) 3mm Disc Section



(b) 4.5mm Complete Disc

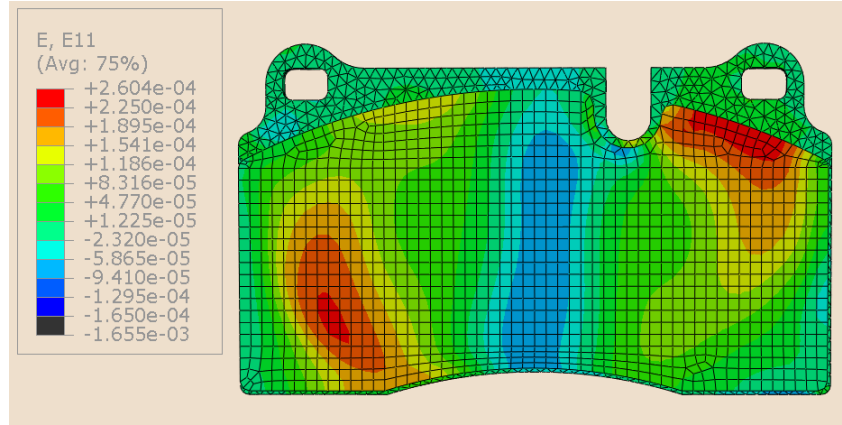
Figure 7-8: FE Analysis Results for Different Element Sizes with 40bar (Y Direction)

The colour bar values are slightly different but it can be safely concluded that the increase of disc mesh (element) size has no detrimental effects on the calculated strain results. The maximum difference was only of the order $10\mu\epsilon$ in the X direction (E11), and $40\mu\epsilon$ in the Y direction (E22).

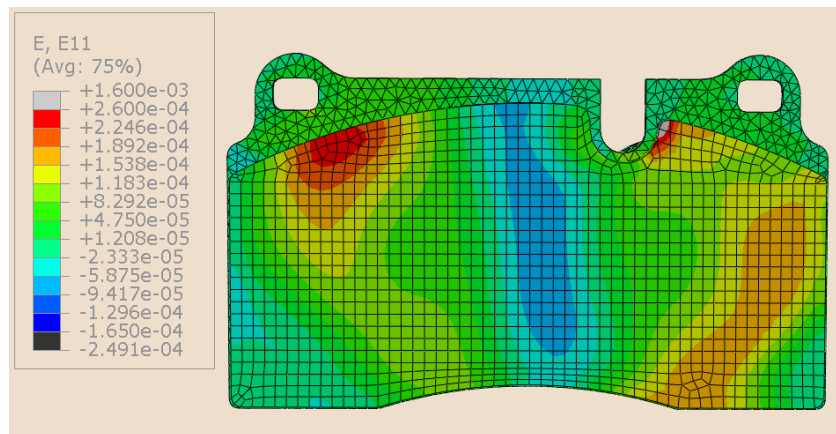
7.1.2.2 Torque Loading Direction

Figure 7-9 (E11, X Direction) and Figure 7-10 (E22, Y Direction) show the strain contours for 40 bar hydraulic pressure and 200 Nm torque level, with two different torque directions, clockwise and anticlockwise directions respectively. The predicted trends were reasonably symmetrical to each other, which were expected. All quasi-dynamic loading measurements were completed with the instrumented brake pad installed on the outboard side of the disc brake system,

i.e. the disc rotated in the anticlockwise direction, but the analyses shown (Figure 7-9 and Figure 7-10) were considered necessary in order to build confidence in the FE model and analyses.

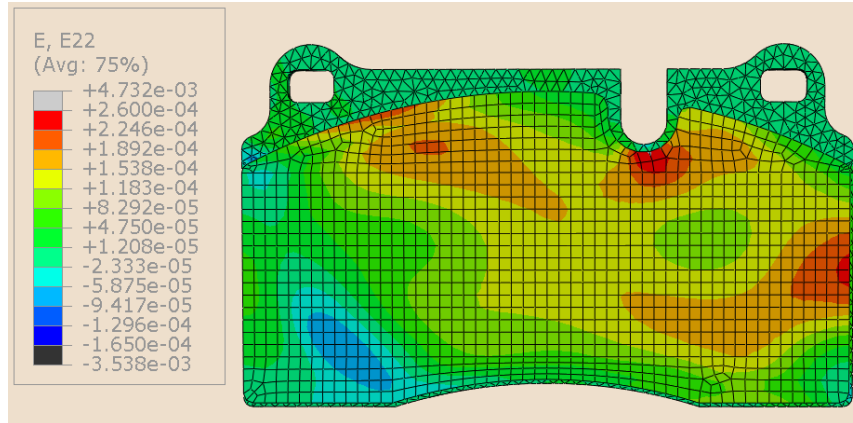


(a) Clockwise Moment Load

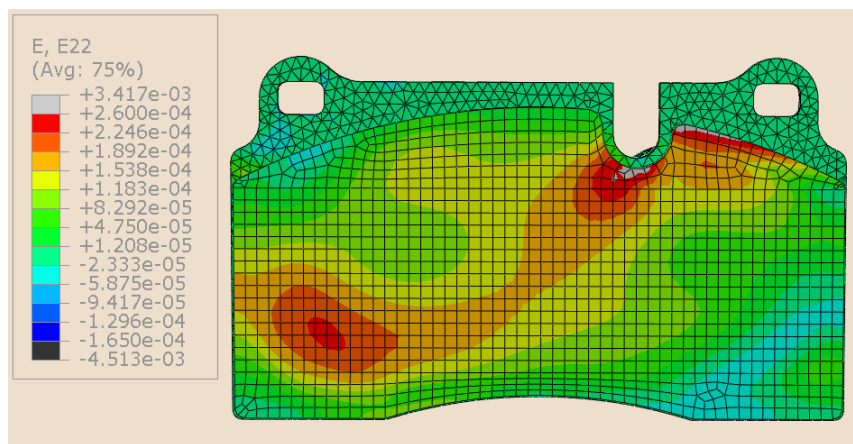


(b) Anticlockwise Moment Load

Figure 7-9: E11 (X Direction) FEA Results for Clockwise and Anticlockwise Moment Loads applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level



(a) Clockwise Moment Load

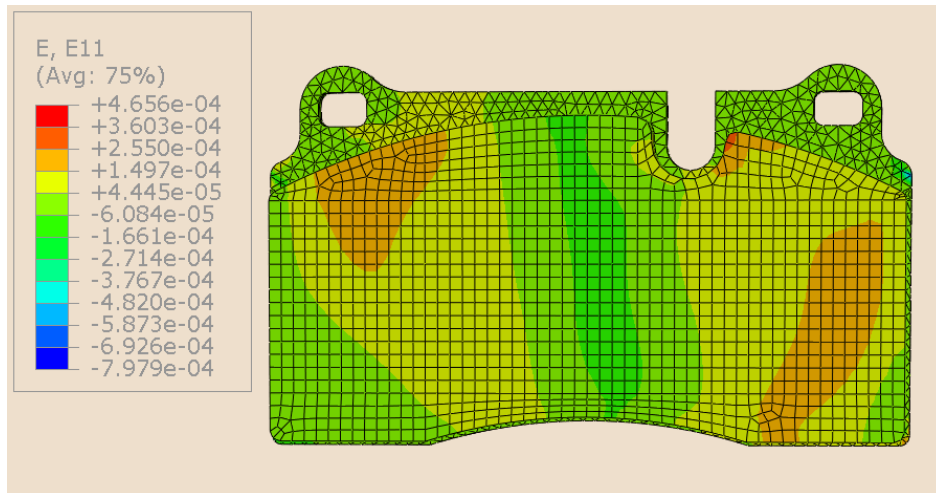


(b) Anticlockwise Moment Load

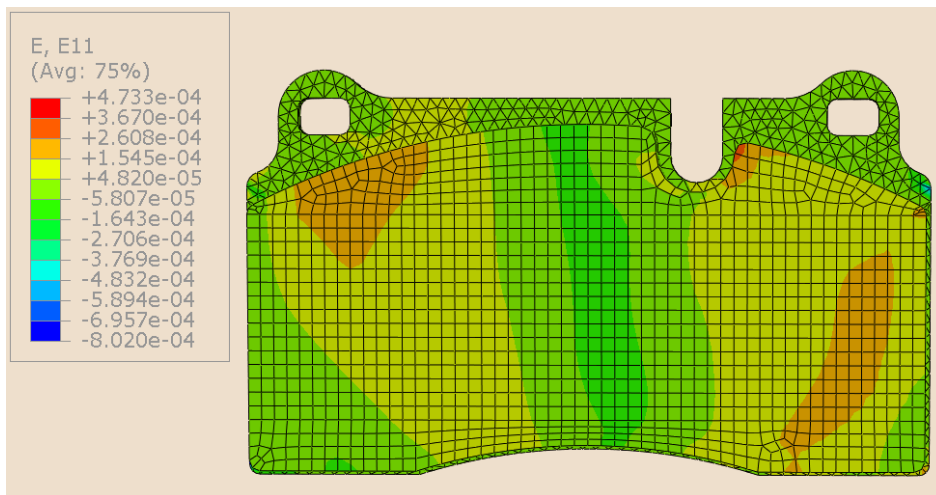
Figure 7-10: E22 (Y Direction) FEA Results for Clockwise and Anticlockwise Moment Loads applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level

7.1.2.3 The Coefficient of Friction

Although the friction coefficient was discussed in detail in static loading (section 5.1.2.3), the quasi-dynamic analyses have BCs and loads with the further consideration will be presented in this section. The coefficient of friction for the friction pair used is assumed to be 0.4 under a wide range of operating conditions, with values increasing up to 0.5 (Vianello, 2015). Strain contour plots (E11 and E22) obtained for the friction coefficient values of 0.4 and 0.5 are shown in Figure 7-11 (E11, X Direction) and Figure 7-12 (E22, Y Direction), for the actuating hydraulic pressure of 40 bar and applied torque of 200 Nm.

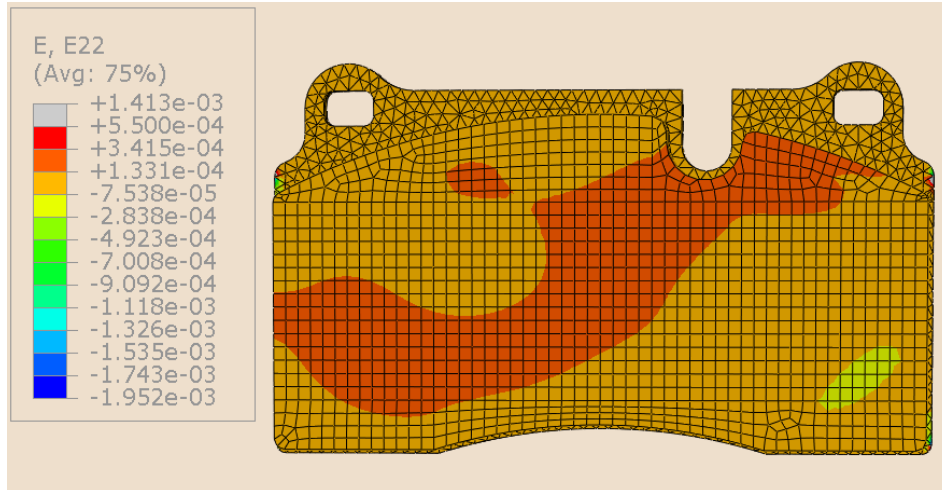


(a) Friction Coefficient 0.4

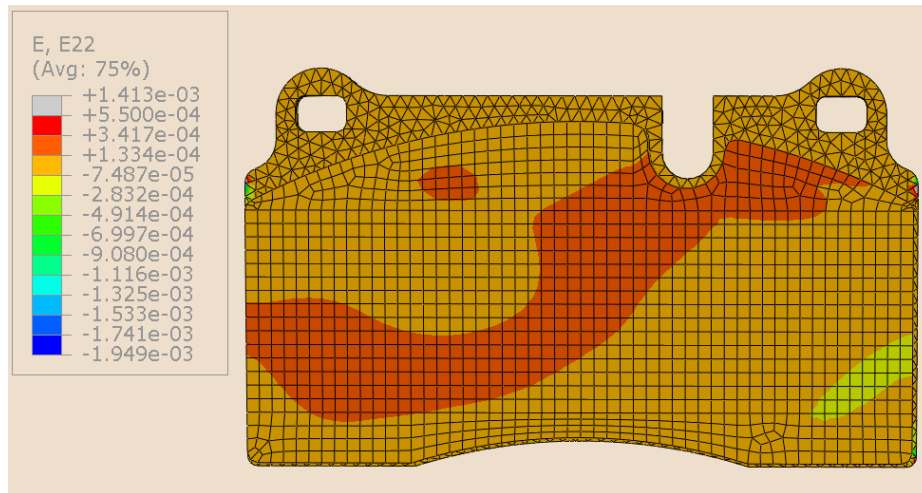


(b) Friction Coefficient 0.5

Figure 7-11: E11 (X Direction) FEA Results for 0.4 and 0.5 Friction Coefficient applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level



(a) Friction Coefficient 0.4



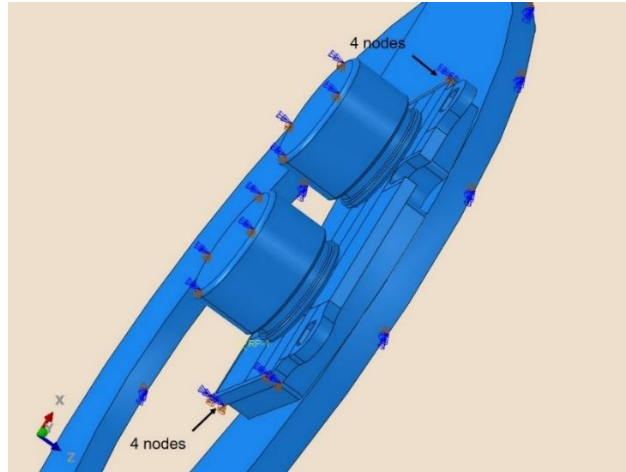
(b) Friction Coefficient 0.5

Figure 7-12: E22 (Y Direction) FEA Results for 0.4 and 0.5 Friction Coefficient applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level

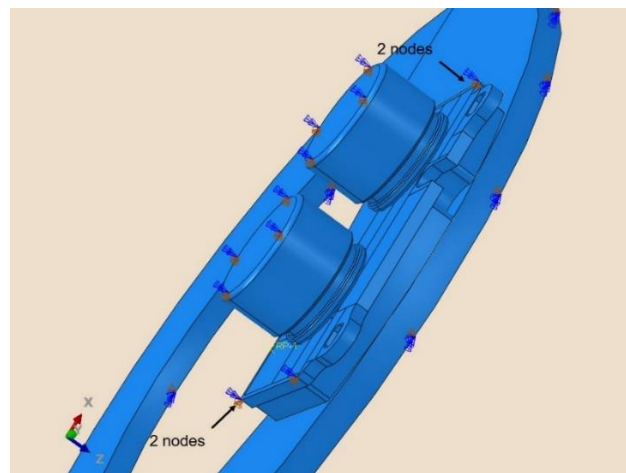
The differences in predicted strain values do not differ significantly for the two values of the friction coefficient used. Several pressure and torque levels will be simulated as quasi-dynamic loading conditions in this chapter, so that the friction coefficient will be selected within a range from 0.4 to 0.5. All FE analyses commenced using 0.4 as the nominal friction coefficient, and then increased by 0.01 each time when taking different loading conditions.

7.1.2.4 Backplate Boundary Conditions

The brake pad is located within the calliper with free space as it is necessary to have a 'loose' clearance fitting in order for the pad to move freely when the brake is actuated and released. Having said that, braking torque and road excitations will move the pad and retaining springs are necessary to ensure pad stability (prevent 'rattle') and avoid NVH issues (squeal in particular). A lubricant is also used on pad abutments. For all these reasons, modelling the backplate boundary conditions is a complex task, as in different brake applications, the backplate or calliper contacts can be accomplished in different areas (i.e. corners). In order to investigate this phenomenon, three boundary condition concepts, illustrated in Figure 7-13, were analysed and compared under the same simulation conditions. These include (Figure 7-13 (a)) constraining all backplate corners (8 nodes), (Figure 7-13 (b)) only half of the backplate corners (4 nodes), and (Figure 7-13 (c)) only the diagonal backplate corners (2 nodes). The diagonal backplate corners (2 nodes) have been selected for the left top corner and the right bottom corner from the front view of the brake pad due to the brake disc anticlockwise rotation, so that these selected nodes will be corresponded to the FEA simulation of the real test conditions.



(a) 1st Concept: Whole Backplate Corners (8 nodes)

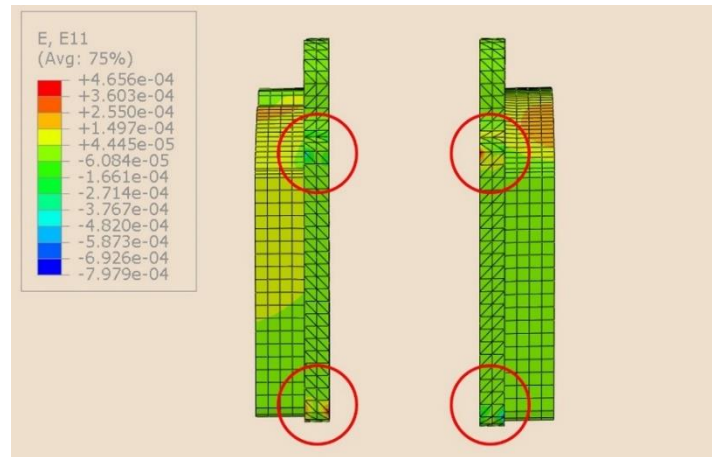


(b) 2nd Concept: Half Backplate Corners (4 nodes)

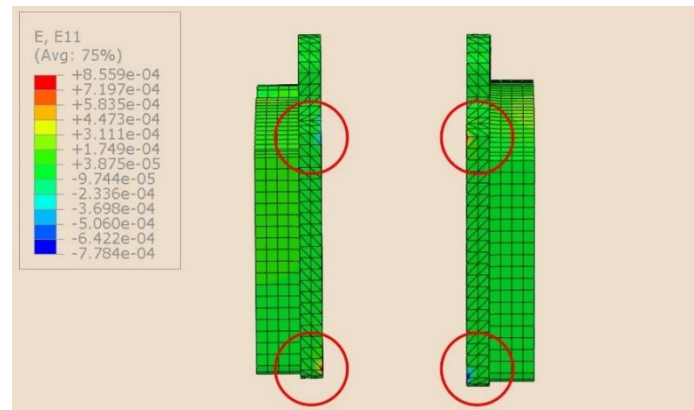


(c) 3rd Concept: Diagonal Backplate Corners (2 nodes)

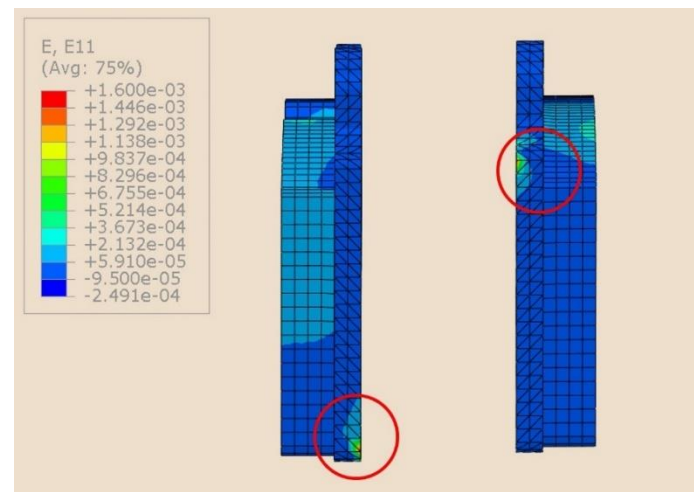
Figure 7-13: Three Design Concepts of Backplate Boundary Conditions



(a) 1st Concept



(b) 2nd Concept



(c) 3rd Concept

Figure 7-14: E11 (X Direction) FEA Results: Three Design Concepts applied with 40 bar Hydraulic Pressure and 200 Nm Torque Level

The obtained results are compared in Figure 7-14 to help decide which BCs should be used for the quasi-dynamic analyses. These comparisons are only focused on the horizontal strain (E_{11}), for the hydraulic pressure of 40 bar and 200 Nm torque.

According to Figure 7-14 (c), the third concept, which is diagonal backplate corners (2 nodes) exhibits the positive strain values when applying the 40 bar hydraulic pressure with 200 Nm torque level on both left top and right bottom corners. While other two concepts have several nodes, four for the first concept in Figure 7-14 (a) and two for the second concept in Figure 7-14 (b), showed negative values, indicating that those areas suffered the unexpected material compression, which would not physically happen in real working environment. The strain at the backplate of the brake pad should never be compressed for the applied BCs. As a result, it is decided that the third concept, the diagonal backplate corners (2 nodes), offered the best solution to be used for the further simulations and analyses.

In conclusion, according to previous analyses and considerations, the updated quasi-dynamic FE brake assembly model characteristics can be set as follows:

- Nominal Element Size: 2.5mm (pad), 4.5mm (disc), 5mm (pistons)
- Moment Load Type: Anticlockwise Direction
- Friction Coefficient: 0.4 ~ 0.5
- Backplate BCs: Diagonal Backplate Corners

7.2 FEA Results and Analysis

The quasi-dynamic FE analysis employs a similar approach to that described for the static loading (section 5.2), displaying the output of the FEA results at FBG sensor locations as a contour map rather than as a screenshot from the Abaqus. The spectrum colour legend is consistent with that used for FBG results and FEA simulations presented in Chapters 4, 5 and 6, to ensure consistency and facilitate easy comparisons.

7.2.1 Influence of Different Hydraulic Pressure and Torque Levels

Two aspects were analysed and compared in this section, the strain results in horizontal (circumferential, X) and vertical (radial, Y) directions. Different values of the applied hydraulic pressure and brake torque were used, with the analysis set-up corresponding to the actual test conditions. Initially, hydraulic pressure was kept the same in both sets of pistons and the results were shown alongside static strain distributions, in order to detect and evaluate change of strain distributions as a result of the torque applied.

Figure 7-15 shows the comparison of the strains applied with equal 80 bar hydraulic pressure, for static (0 Nm) and quasi-dynamic loading with 600 Nm torque, assuming a coefficient of friction being 0.46. In vehicle operation, such condition is representative of a heavy-duty (but not maximum possible) brake application. The results obtained for other test conditions are listed in Appendix Chapter 7-A (40 bar / 200 Nm, light brake application), and Appendix Chapter 7-B (120 bar / 1000 Nm, maximum duty application).

Figure 7-15 clearly indicates that strain values change considerably when the braking torque is applied, which was to be expected. The Y-direction strain decreased at the disc OD area (the top side of the pad), which might be caused by the brake pad 'bending or lifting' when the torque is applied. The disc ID area (the bottom side of the pad) shows higher strains at the piston 2 area than at the piston 1 area, due to the torque clockwise direction (looking from the disc surface).

In addition, the X direction is observed to have the strain level changed significantly at both leading and trailing edges with the introduction of the braking torque of 600 Nm. In the middle of the brake pad friction material, strain values have become negative, indicating the material compression again, which is the similar trend observed at the static loading analysis in section 5.2.2.1. The other two loading conditions (40 bar / 200 Nm and 120 bar / 1000 Nm) have also shown similar trends of strain changes at those areas, as illustrated in Appendix Chapter 7-A and Appendix Chapter 7-B.

Whilst Figure 7-15 shows changes of the strain distributions on pad friction surface as a result of introducing the braking torque, in relation to the static distributions, for 80 bar pressure and 600 Nm torque, it is important to examine the influence of different hydraulic pressures and corresponding braking torques on strain distributions.

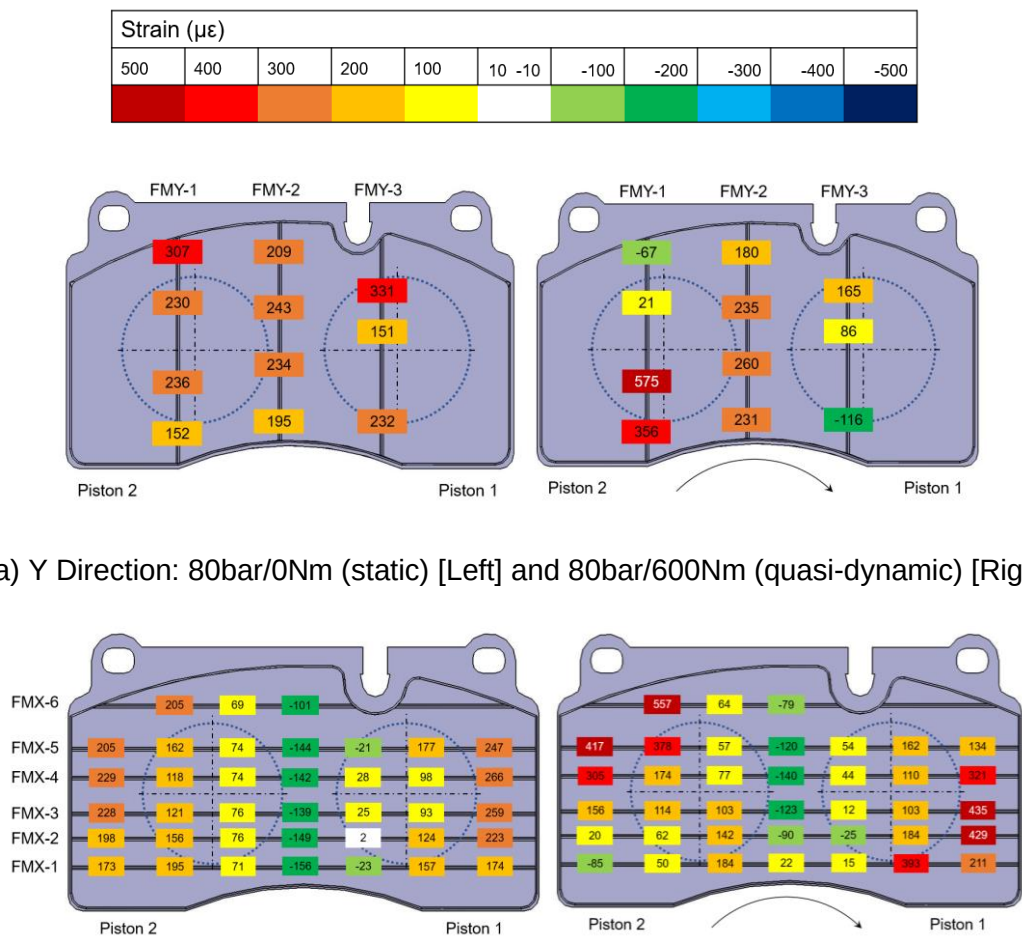
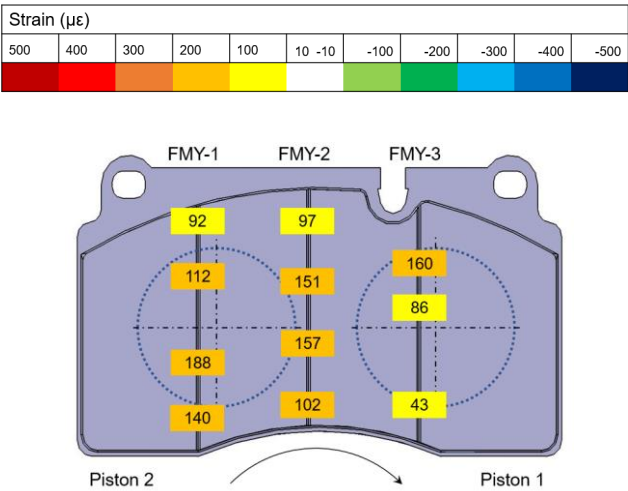


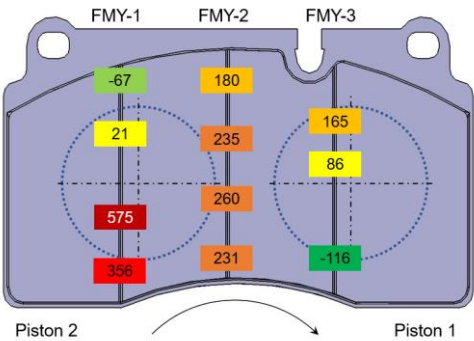
Figure 7-15: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.46$

Three conditions were selected, identical to the quasi-dynamic experimental phase, being 40 bar pressure with 200 Nm torque, 80 bar pressure with 600 Nm torque, and 120 bar pressure with 1000 Nm torque. Figure 7-16 and Figure 7-17 show FEA predicted strain values (at FBG sensor locations) in Y and X directions respectively. Pressure values are equal on both piston sets, i.e. $p_1 = p_2$, and the

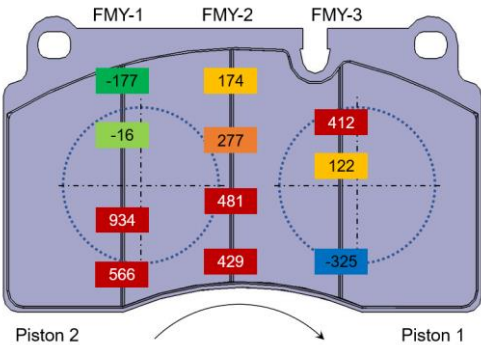
same spectrum colour legend is used as in section 4.3.3 and elsewhere. Disc rotation direction is also shown.



(a) 40bar/200Nm with Equal Pressure ($p_1=p_2$); $\mu=0.4$

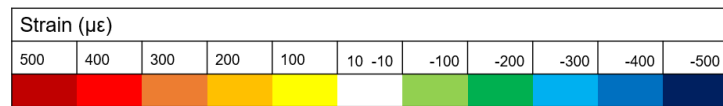
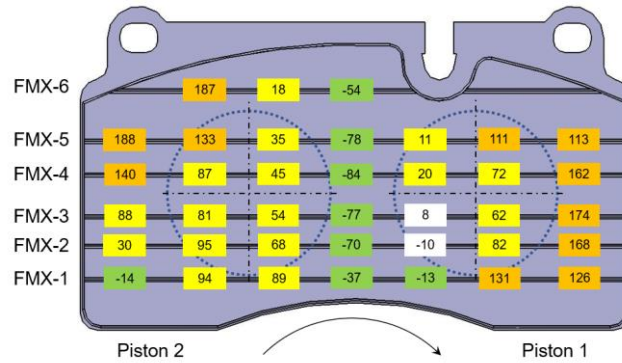


(b) 80bar/600Nm with Equal Pressure ($p_1=p_2$); $\mu=0.46$

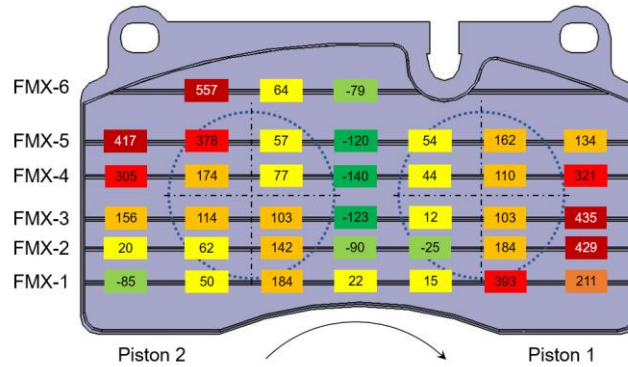


(c) 120bar/1000Nm with Equal Pressure ($p_1=p_2$); $\mu=0.46$

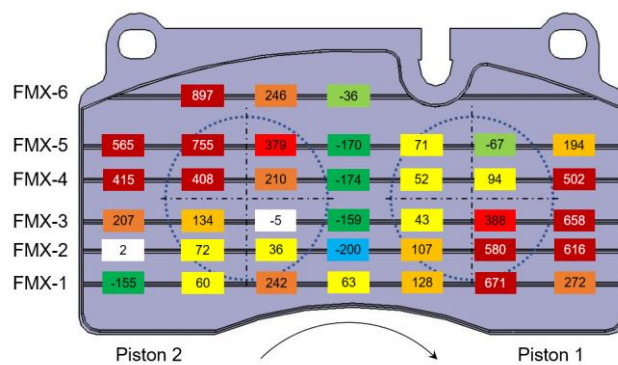
Figure 7-16: Processed FEA Contour Maps in Vertical (Y) Direction for Three Hydraulic Pressure and Torque Levels



(a) 40bar/200Nm with Equal Pressure ($p_1=p_2$); $\mu=0.4$



(b) 80bar/600Nm with Equal Pressure ($p_1=p_2$); $\mu=0.46$



(c) 120bar/1000Nm with Equal Pressure ($p_1=p_2$); $\mu=0.46$

Figure 7-17: Processed FEA Contour Maps in Horizontal (X) Direction for Three Hydraulic Pressure and Torque Levels

From Figure 7-16 it can be concluded that there is no linear increase (or decrease) in E22 strain with the pressure and torque increase. Nevertheless, the trends in strain changes remain consistent. Similarly, Figure 7-17 shows that E11 strain increases under the piston with the increase of pressure and torque, and decreases in the area between the pistons, indicating the friction material compression. Such responses seem logical, taking into consideration non-symmetrical brake pad loading due to disc rotation and constraints on backplate abutments placed in diagonally opposite corners.

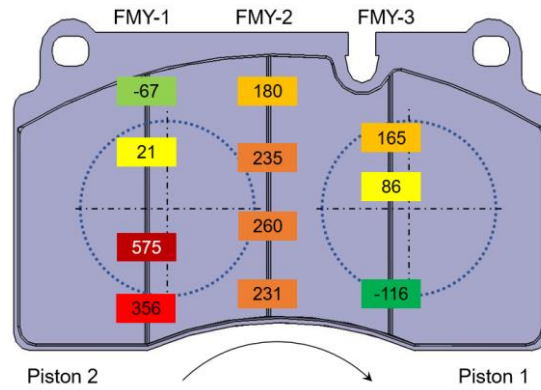
7.2.2 Influence of Hydraulic Pressure Variations between Piston Sets

In addition to the equal pressure on both piston sets ($p_1 = p_2$), the hydraulic pressure was also varied by ± 10 bar between the piston sets, in order to investigate how these changes influenced the strain distributions on the pad frictional surface under quasi-dynamic conditions (static loading was analysed in section 5.2.2.2).

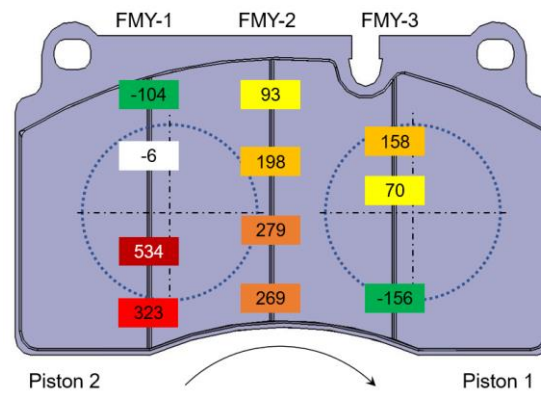
Initially, the nominal pressure was set at 80 bar with ± 10 bar and 600 Nm torque. Figure 7-18 and Figure 7-19 illustrate strain changes in Y and X directions, respectively. The results for 40 bar nominal pressure with 200 Nm torque are included in Appendix Chapter 7-C. It is worth pointing out that the condition of 120 bar hydraulic pressure with 1000 Nm torque level did not yield any meaningful results to be observed after several analyses. The pressure is too high for the change of ± 10 bar to influence strain distributions in a significant way.

In Figure 7-18 (a) and Figure 7-19 (a), the 'baseline' strain distribution is shown, for equal pressure of 80 bar on both piston sets. In the next step, the pressure was reduced on piston 2 to 70 bar, whilst maintaining 80 bar at piston 1 (Figure 7-18 (b) and Figure 7-19 (b)). Finally, in the third step, the pressure was set 80 bar at piston 2 and 70 bar at piston 1 (Figure 7-18 (c) and Figure 7-19 (c)). It needed to be pointed out that pistons 1 and 2 are not positioned symmetrically in relation to the pad axis (mid-plane) of symmetry, as there is a 3mm offset. In all these steps 600 Nm of torque was applied to the brake disc.

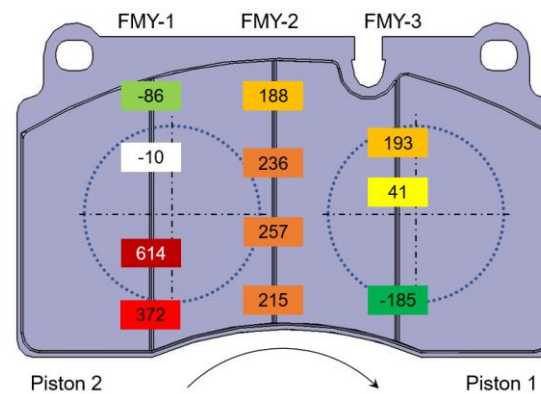
Strain ($\mu\epsilon$)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) Equal Pressure ($p_1=p_2=80\text{bar}$; 600Nm)



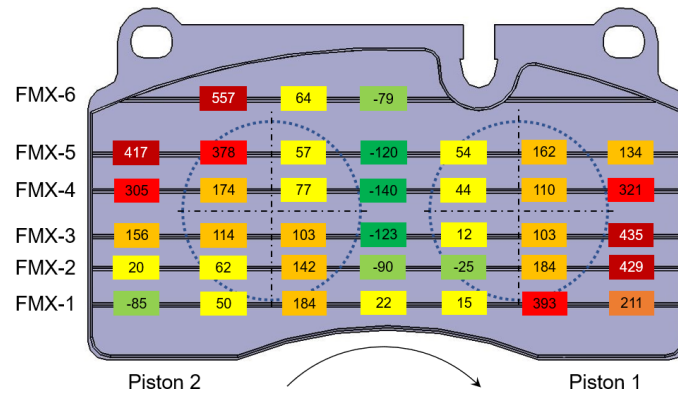
(b) +10bar ($p_1=80\text{bar}$; $p_2=70\text{bar}$; 600Nm)



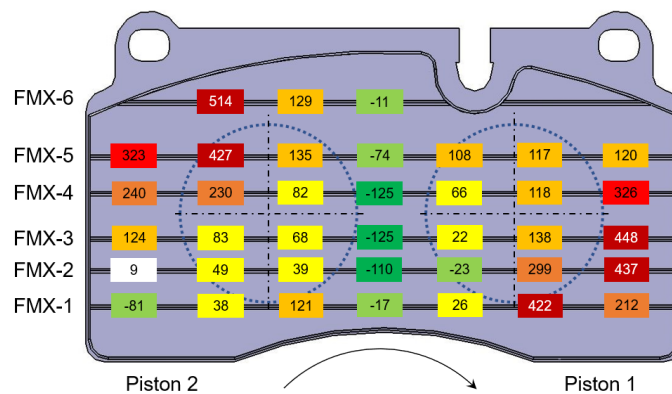
(c) -10bar ($p_1=70\text{bar}$; $p_2=80\text{bar}$; 600Nm)

Figure 7-18: E22 (Y Direction) Processed FEA Strain Contour Map Comparisons for Three Hydraulic Pressure Variations; $\mu=0.46$

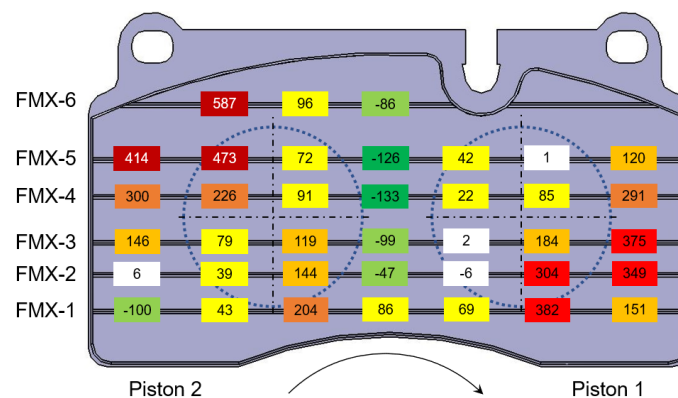
Strain ($\mu\epsilon$)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



(a) Equal Pressure ($p_1=p_2=80\text{bar}$; 600Nm)



(b) +10bar ($p_1=80\text{bar}$; $p_2=70\text{bar}$; 600Nm)



(c) -10bar ($p_1=70\text{bar}$; $p_2=80\text{bar}$; 600Nm)

Figure 7-19: E11 (X Direction) Processed FEA Strain Contour Map Comparisons for Three Hydraulic Pressure Variations; $\mu=0.46$

Generally, both E11 (X Direction) and E22 (Y Direction) strain results show similar trends, with hydraulic pressure variations not significantly influencing the strain values under such a loading application (80 bar / 600 Nm). Looking at Figure 7-18, starting with equal pressure and +10bar variation, the strain levels reduced around the piston 2 area, where the pressure level decreased to 70 bar (still maintaining 80 bar in piston 1, Figure 7-18 (b)). The maximum strain decrement was $87\mu\epsilon$ at FMY2-FBG1, at the disc OD contact area. However, the increased strain levels can be observed at the disc ID contact area, FMY2-FBG3 and FMY2-FBG4, about $40\mu\epsilon$. Then, the -10bar variation condition in Figure 7-18 (c), the largest strain changes appeared at FMY1-FBG3, which increased by $39\mu\epsilon$. Other sensor locations showed similar trends, increase or decrease, based on their position in relation to the piston areas. However, most of these strain changes were under $50\mu\epsilon$.

Looking at the E11 (X Direction) strain, Figure 7-19, similar trends were obtained, with three hydraulic pressure variations. In Figure 7-19 (b), the maximum increase was obtained at FMX2-FBG6 ($115\mu\epsilon$), and the biggest decrease was $94\mu\epsilon$ located at FMX5-FBG1. Such strain changes were expected and could be considered logical, as the FMX2-FBG6 was positioned around piston 1 area (the pressure maintaining 80 bar in piston 1), and the FMX5-FBG1 was near piston 2 edge (the pressure reducing to 70 bar in piston 2). On the contrary, Figure 7-19 (c), when the pressure in piston 1 is reduced to 70 bar and pressure under piston 2 kept at 80 bar, the maximum increment appeared at FMX5-FBG2 ($95\mu\epsilon$), and FMY5-FBG6 is showing the largest decrease of $161\mu\epsilon$.

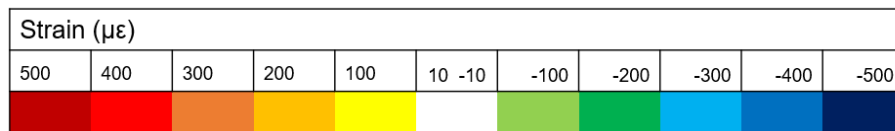
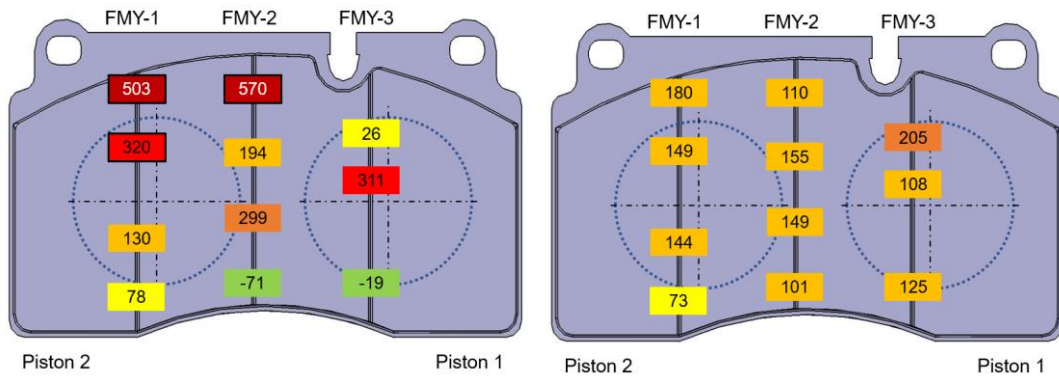
These simulations provided a clear view of strain values and their variations for the quasi-dynamic conditions, and will provide a basis for better analysing and understanding dynamic test conditions.

7.3 Comparison of FBG Test and FEA Simulation Results

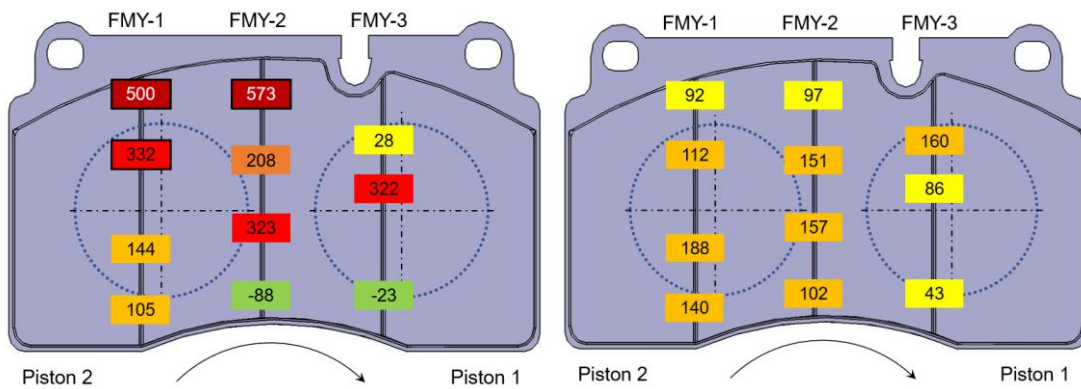
7.3.1 Approaches in Result Comparisons

In the next section, FEA predicted strains will be compared with FBG sensor measured values. The approach similar to the static loading case presented in

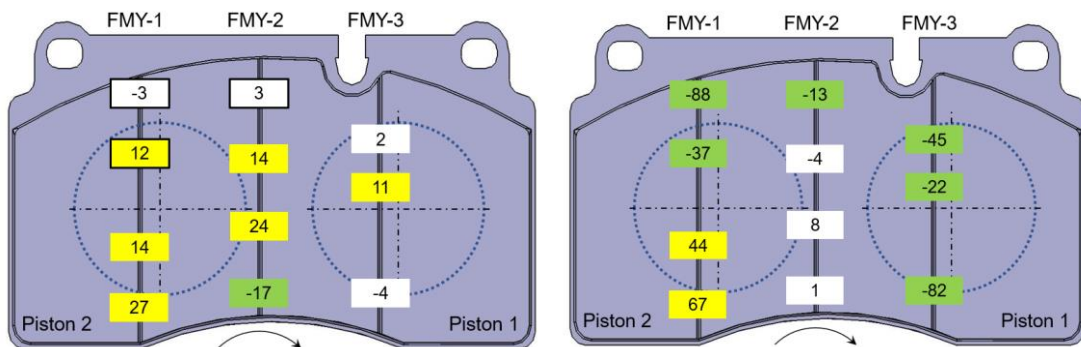
section 5.3 will be adopted. However, the quasi-dynamic conditions contained two variable parameters, hydraulic pressure and torque level. In order to easily compare these results, several steps have been taken to establish strain differences between FBG tests and FEA simulation results in Y direction (E22) for the loading condition of 40 bar pressure and 200 Nm torque. Both sets of results will also keep the same colour legend to ensure that comparisons can be easily traced. Figure 7-20 begins by showing FBG and FEA results for static condition, at 40 bar pressure with 0 Nm torque (a), then FBG and FEA results at 40 bar pressure with 200 Nm torque (b), continuing with the Static to Quasi-Dynamic strain differences (c). Reference to static results is considered important as FBG and FEA results is somewhat in static conditions and the effect of introducing braking torque can be different.



(a) 40bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.4$] (Right)



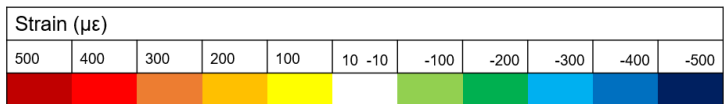
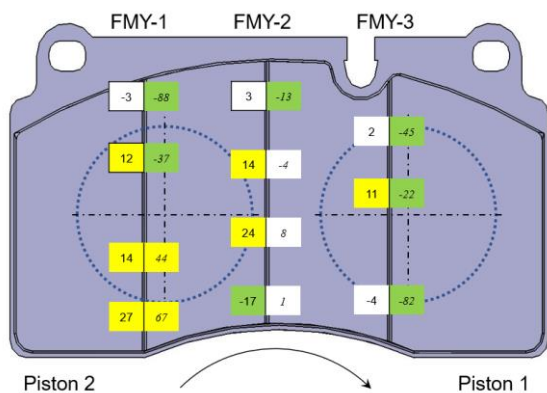
(b) 40bar with 200Nm: FBG Test (Left) and FEA Simulation [$\mu=0.4$] (Right)



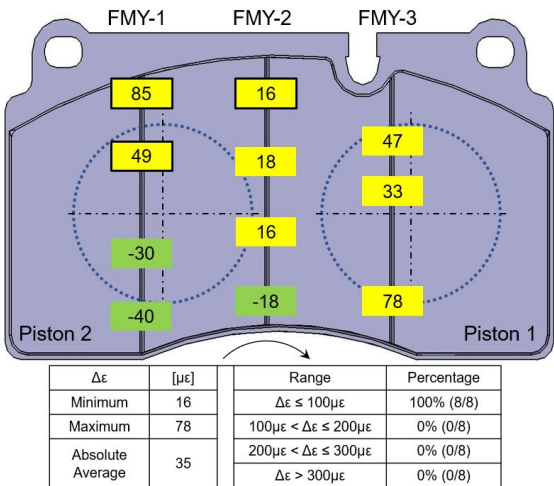
(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

Figure 7-20: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=40\text{bar}$

Figure 7-21 displays both sets of FBG and FEA side-by-side (a), and finally the strain differences of Static to Quasi-Dynamic differences (b), which are found to be the most suitable for result comparisons. Namely, if all values were $0\mu\epsilon$, the 'perfect match' would have been achieved. It can be seen (Figure 7-21 (b)) that in all sensor locations the strain differences are under $100\mu\epsilon$, mostly even under $50\mu\epsilon$. The highest strain difference is $78\mu\epsilon$ observed at FMY3-FBG3.



(a) Comparisons of FBG (left) and FEA [$\mu=0.4$] (right)



(b) Strain Differences $\Delta\epsilon_{\text{Static to Quasi-Dynamic Differences}} = \epsilon_{\text{Test}} - \epsilon_{\text{FEA}}$

Figure 7-21: Processed E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=40\text{bar}$ with 200Nm Torque

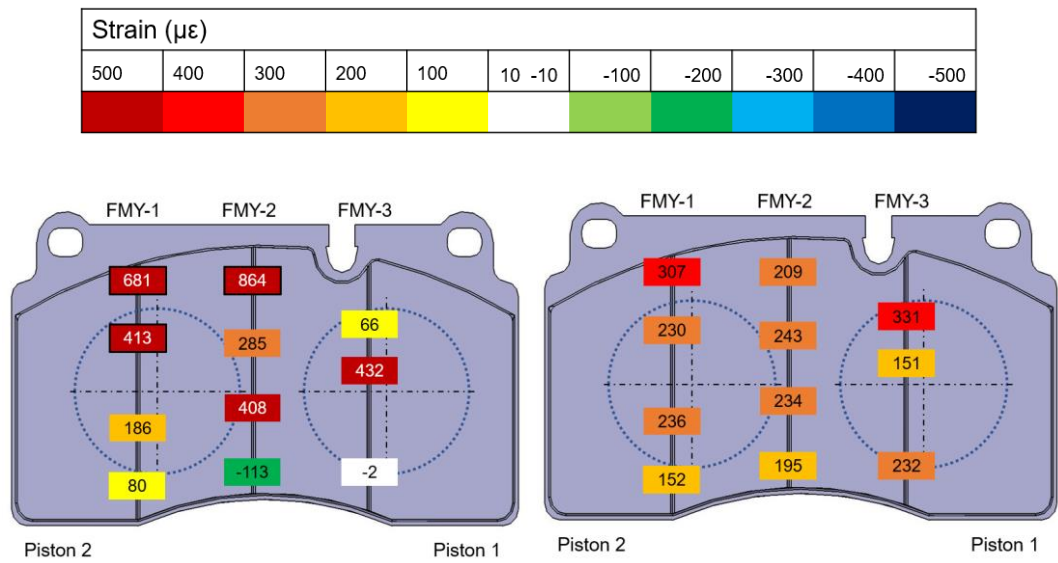
7.3.2 Analysis of The Results

The previous section 7.3.1 established the similar approach in FBG and FEA result comparisons, as presented in section 5.3.1. This will now be applied to investigate strain differences at various hydraulic pressure and torque levels, as well as when unequal hydraulic pressure is applied between the two piston sets.

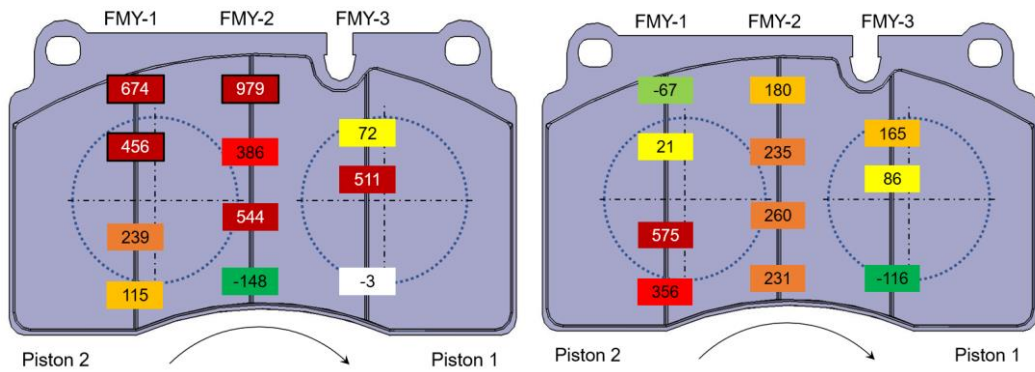
7.3.2.1 Hydraulic Pressure and Torque Levels

Both FBG tests and FEA simulations have been conducted for a range of hydraulic pressure and torque levels, with the results more closely analysed for the actuating pressures and torque levels of 40 bar / 200 Nm, 80 bar / 600 Nm and 120 bar / 1000 Nm. In this section, hydraulic pressures will be identical for all pistons (i.e. equal pressure condition, $p_1 = p_2$).

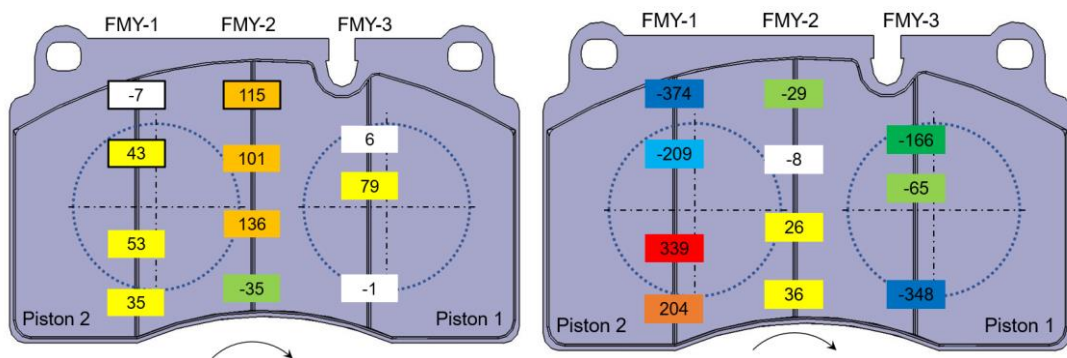
As the results for the actuating pressure of 40 bar with 200 Nm torque level were investigated in the previous section 7.3.1, hydraulic pressures with torque levels of 80 bar / 600 Nm and 120 bar / 1000 Nm will be observed in more detail. Figure 7-22 shows FBG and FEA strain results in Y direction (E22) for 80 bar with 600 Nm, and E22 results of 120 bar with 1000 Nm is included in Figure 7-23. It can be seen that some results are relatively close, with others being quite far apart.



(a) 80bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



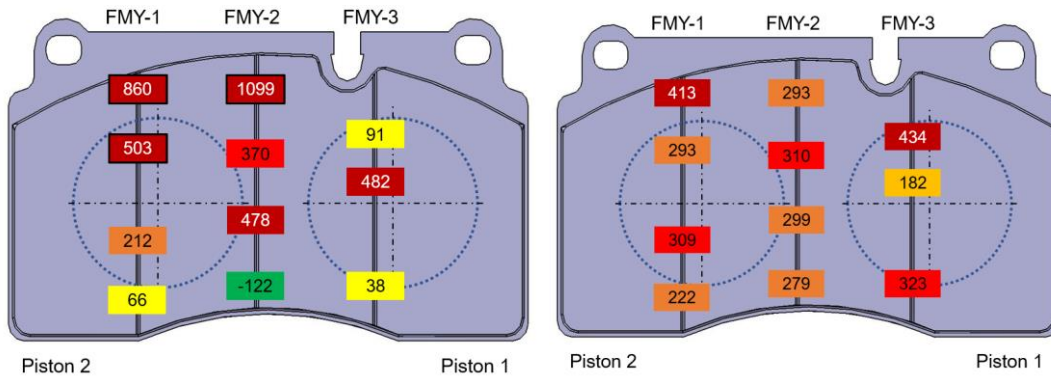
(b) 80bar with 600Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



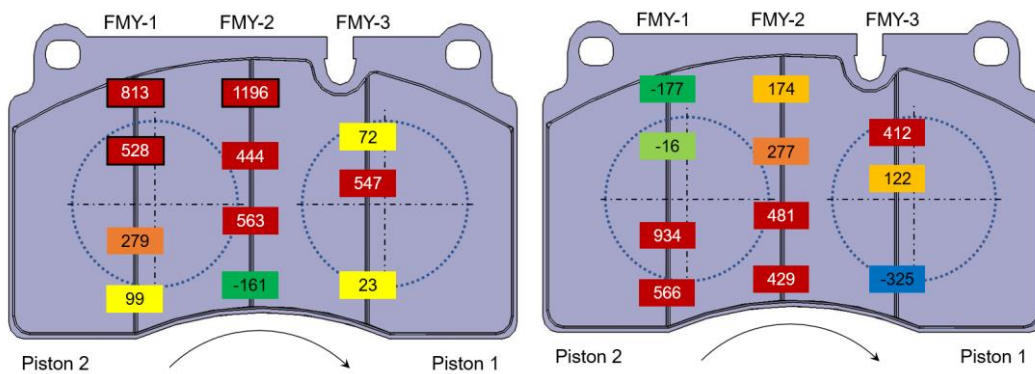
(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

Figure 7-22: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=80\text{bar}$

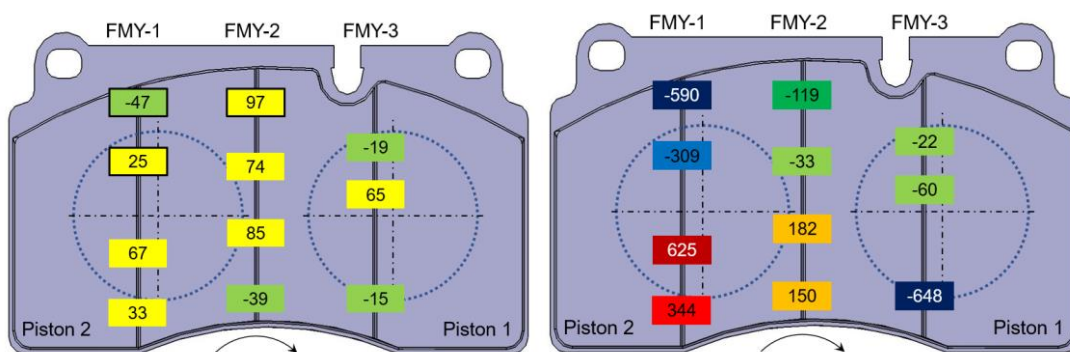
Strain ($\mu\epsilon$)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



(a) 120bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



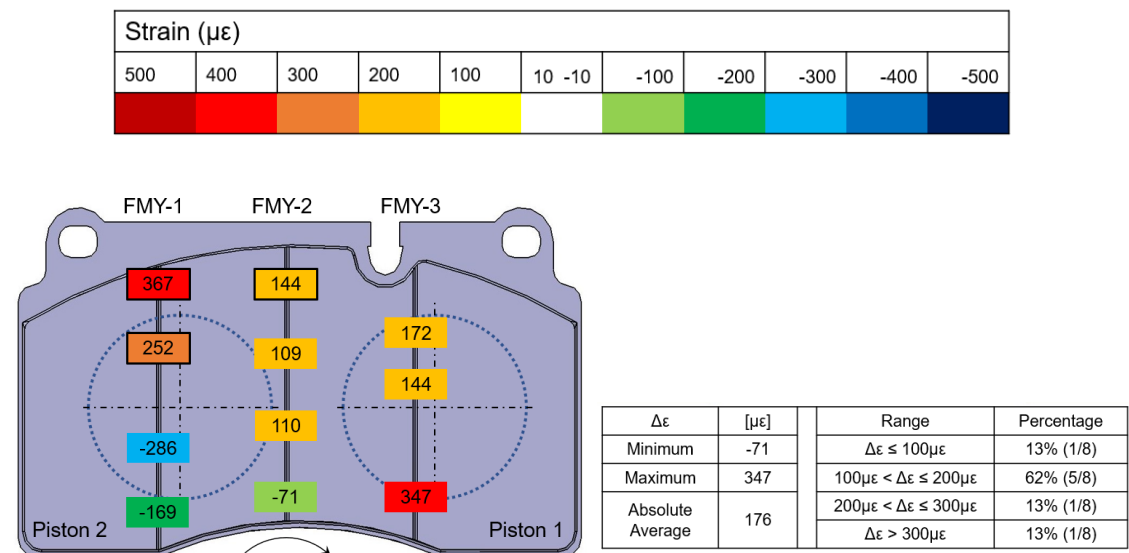
(b) 120bar with 1000Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



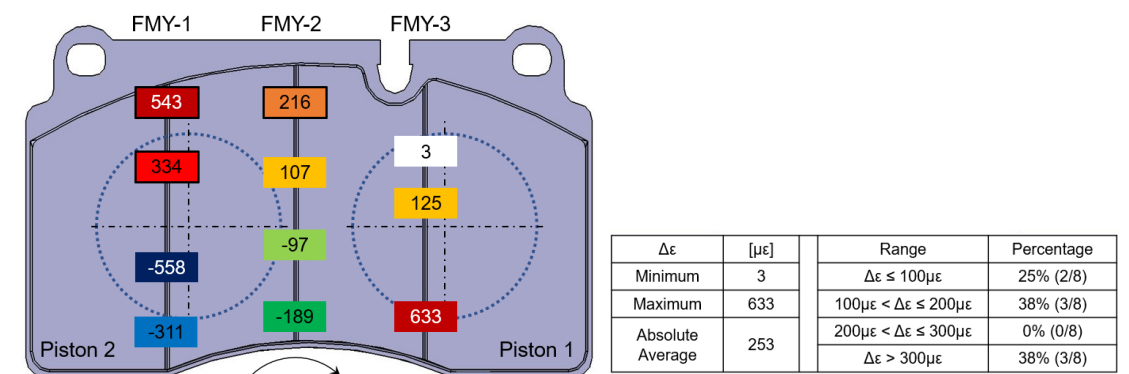
(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

Figure 7-23: E22 (Y Direction) FBG Test and FEA Simulation for $p_1=p_2=120\text{bar}$

In order to gain a deeper insight, as proposed in the previous section, strain differences of Static to Quasi-Dynamic strain differences are plotted in Figure 7-24 for each location and both pressure and torque levels (80 bar / 600 Nm and 120 bar / 1000 Nm).



(a) Strain Differences $\Delta\epsilon_{\text{Static to Quasi-Dynamic Differences}} = \epsilon_{\text{Test}} - \epsilon_{\text{FEA}}$: 80bar & 600Nm



(b) Strain Differences $\Delta\epsilon_{\text{Static to Quasi-Dynamic Differences}} = \epsilon_{\text{Test}} - \epsilon_{\text{FEA}}$: 120bar & 1000Nm

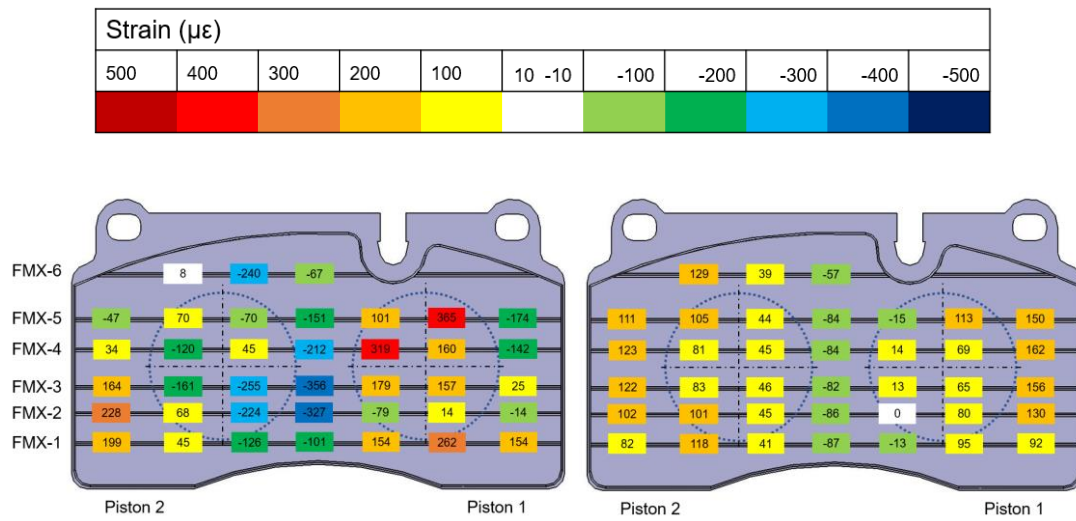
Figure 7-24: E22 (Y Direction) Strain Differences for Static to Quasi-Dynamic differences of Two Hydraulic Pressure and Torque Levels; $\mu=0.46$

For 80 bar with 600 Nm (Figure 7-24 (a)), the sensor locations around piston 1 area obtained a good agreement between test and simulation results, only the FMY3-FBG3 experienced the highest strain change, which was $347\mu\epsilon$. In the lower part of the piston 2 area, the differences are more pronounced with the

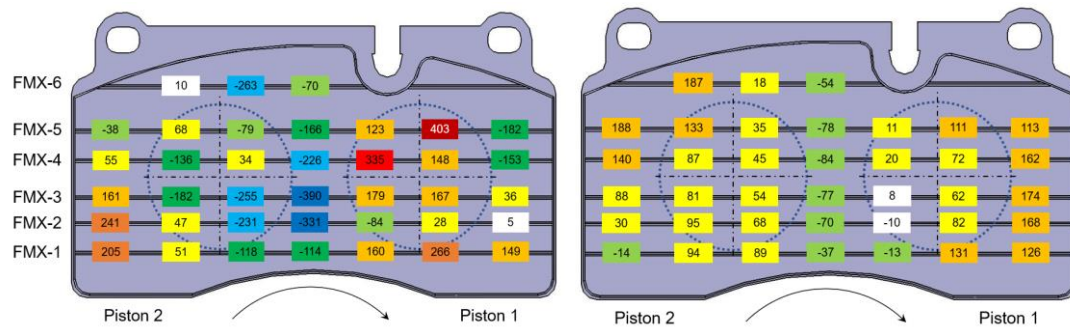
negative values, where the FMY1-FBG3 showed the biggest difference ($-286\mu\epsilon$). The summary table associated with Figure 7-24 (a) displays that strain differences of under $200\mu\epsilon$ for 75% of the sensor locations, with only 13% (1/8) being over $200\mu\epsilon$, and another 13% (1/8) over $300\mu\epsilon$. Three sensor locations, FMY1-FBG1, FMY1-FBG2 & FMY2-FBG1, are excluded due to the unpractical test results, as illustrated in Figure 7-22 and Figure 7-23.

The situation is very similar for 120 bar hydraulic pressure with 1000 Nm torque condition (Figure 7-24 (b)), with much higher strain differences owing to the much higher mechanical load (hydraulic pressure and torque level) compared with 40 bar / 200 Nm and 80 bar / 600 Nm loading conditions. The exception was FMY3-FBG1, which was located at the top position towards the slot for the wear sensor installation, with only $3\mu\epsilon$ difference between FBG and FEA results. The torque applied has direction from piston 2 to piston 1, which 'squeezes' the Y-direction grooves (as the friction force direction is approximately perpendicular to the groove direction). Consequently, some of the strain variations and differences were quite significant in these applications.

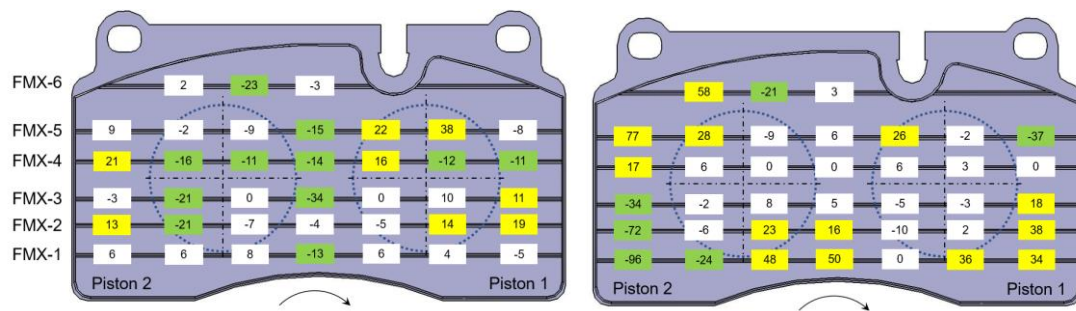
The E11 (X Direction) strain data obtained through FBG test and FEA simulation has been illustrated in Figure 7-25, for loading conditions of 40 bar / 200 Nm, in Figure 7-26 for 80 bar / 600 Nm, and in Figure 7-27 for 120 bar / 1000 Nm. It seems that E11 strain (X Direction) results show different trends to E22 strain (Y Direction) results. Generally, similarities between FBG tests and FEA simulations are quite apparent, with higher tensile strains at both the leading and trailing edges of the brake pad and compression strains in the middle areas.



(a) 40bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.4$] (Right)

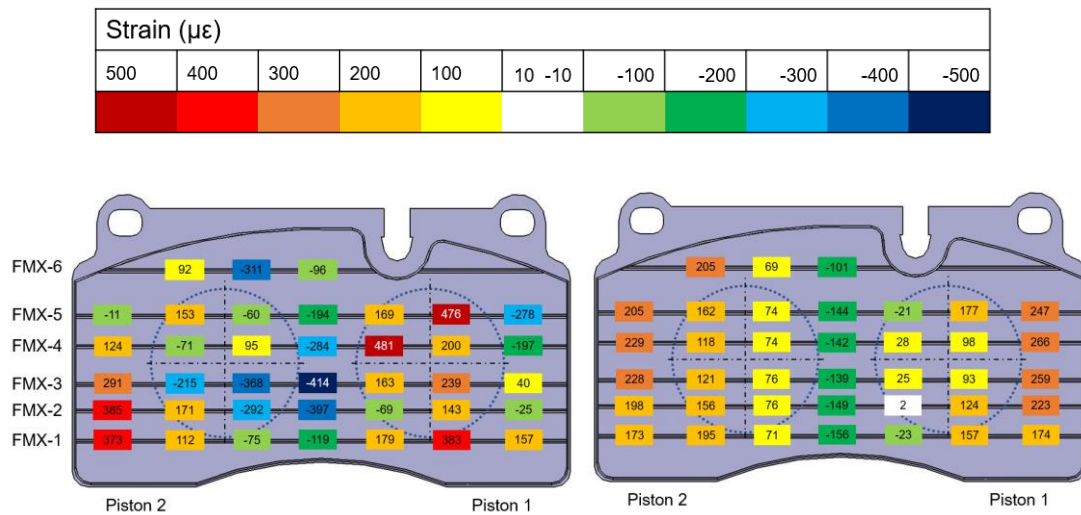


(b) 40bar with 200Nm: FBG Test (Left) and FEA Simulation [$\mu=0.4$] (Right)

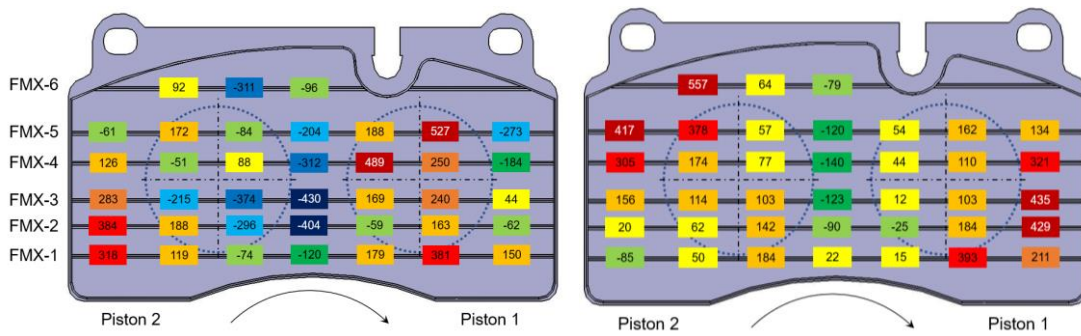


(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

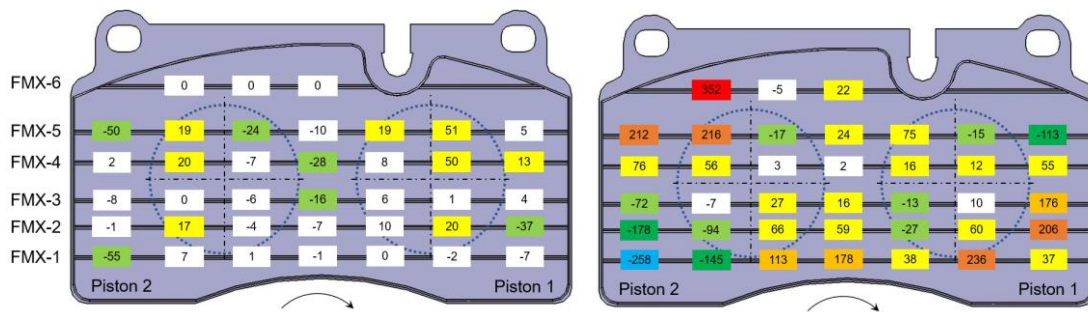
Figure 7-25: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure,
 $p_1=p_2=40\text{bar}$



(a) 80bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)

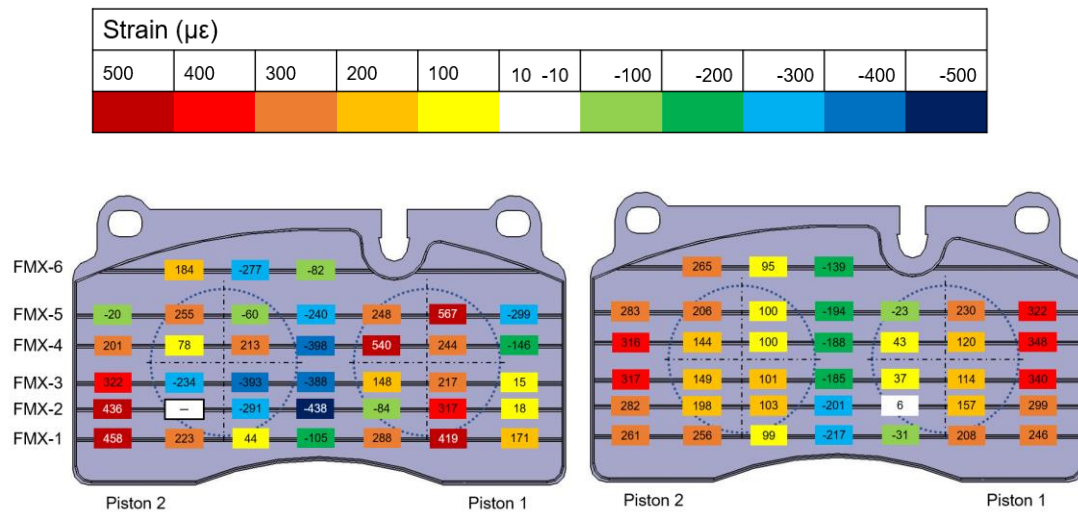


(b) 80bar with 600Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)

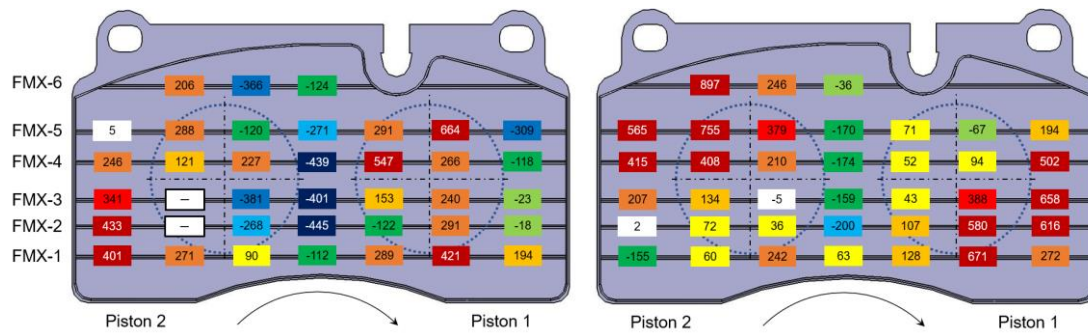


(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

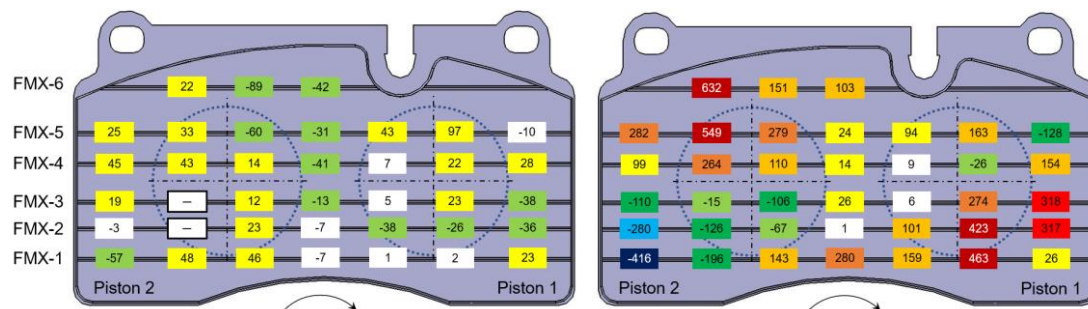
Figure 7-26: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure, $p_1=p_2=80\text{bar}$



(a) 120bar with 0Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



(b) 120bar with 1000Nm: FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right)



(c) Static to Quasi-Dynamic Differences: FBG Test (Left) and FEA Simulation (Right)

Figure 7-27: E11 (X Direction) FBG Test and FEA Simulation for Equal Pressure, $p_1=p_2=120\text{bar}$

When examining the maps and tables of E11 strain differences (X Direction) between tests and simulations (Figure 7-28), they are lower than E22 strain

differences (Y Direction) (Figure 7-21 (b) and Figure 7-24). As expected, the differences increase with the rise of hydraulic pressure and torque levels.

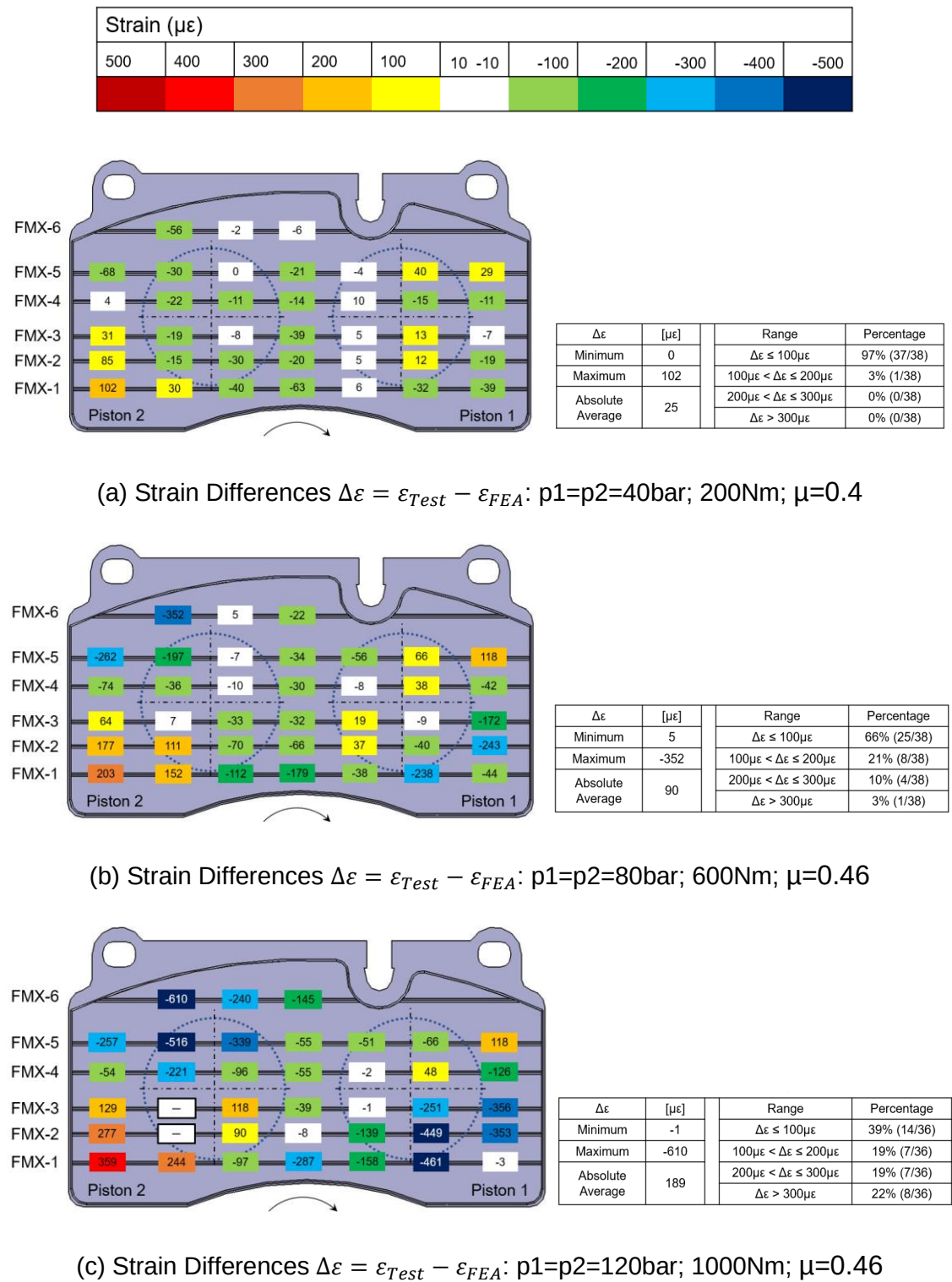


Figure 7-28: E11 (X Direction) Strain Differences for Three Hydraulic Pressure and Torque Levels

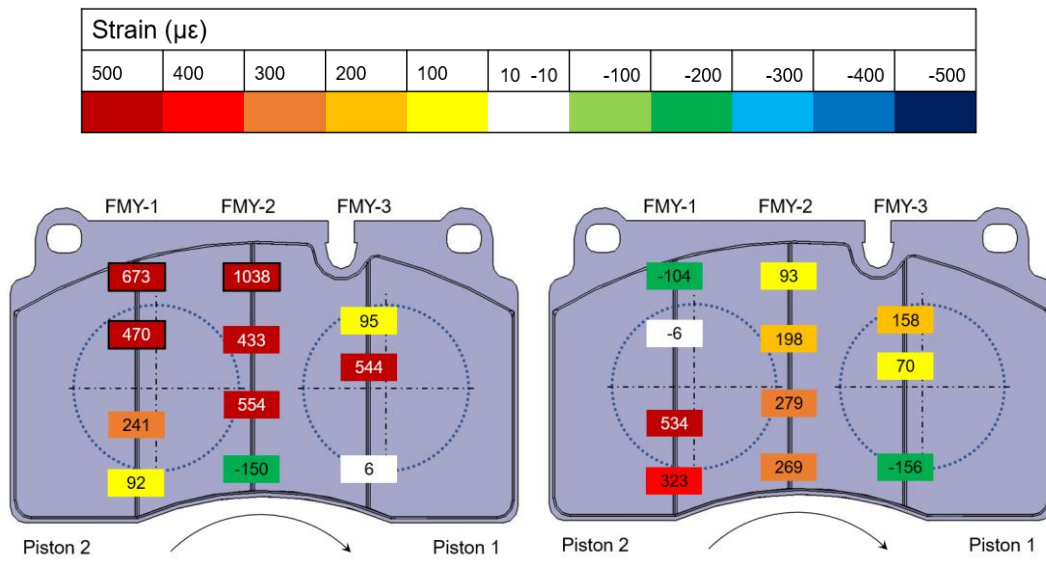
There are areas where the differences are quite small, even for the highest pressure and torque (Figure 7-28 (c)), and some with medium or extremely large changes but it is hard to build any pattern from such a phenomenon. Having said that, most of the positive differences can be observed around two areas, on the left lower side of the piston 2 areas, and the right upper side of piston 1. In other sensor locations the trends are opposite, as the differences are negative. Such results can have various explanations, with the experimental uncertainties being particularly pronounced, in particular related to pad friction materials' non-homogeneous and granular structure.

7.3.2.2 Hydraulic Pressure Variations

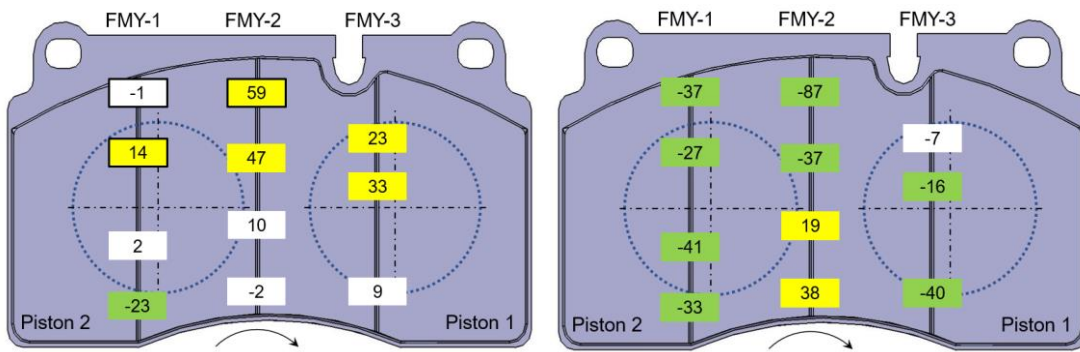
In section 7.2.2, the background of different hydraulic pressure variations between piston pairs has been discussed, from the FEA simulation standpoint. Here, FBG and FEA results will be compared for the nominal actuating hydraulic pressure of 80 bar with 600 Nm torque level, with the results for 40 bar hydraulic pressure with 200 Nm torque level included in Appendix Chapter 7-D.

Before comparing FBG and FEA strain results, the influence of pressure variations between two sets of pistons will be examined, i.e. the strain differences caused by reducing the pressure at piston 2 (p_2) from 80 bar to 70 bar. Obviously, for consistency, FBG measured strains will be compared with each other for equal and variable pressures, as will the corresponding FEA strains. FBG and FEA results for E22 strain (Y Direction) have already been shown, compared and discussed in Figure 7-22 for the identical pressure and same loading condition ($p_1 = p_2 = 80$ bar and 600 Nm torque).

In Figure 7-29, FBG and FEA comparisons of E22 strain (Y Direction) is shown for 80 bar at piston 1 and 70 bar at piston 2. When comparing FBG and FEA results (Figure 7-29 (a)), quite considerable differences can be noticed at practically all points. In comparison to the equal pressure (Figure 7-29 (b)), E22 strain (Y Direction) seems to be changing in opposite directions. FBG measurements recorded strain increase in almost all places, with FEA predicted decrease in majority of the locations.



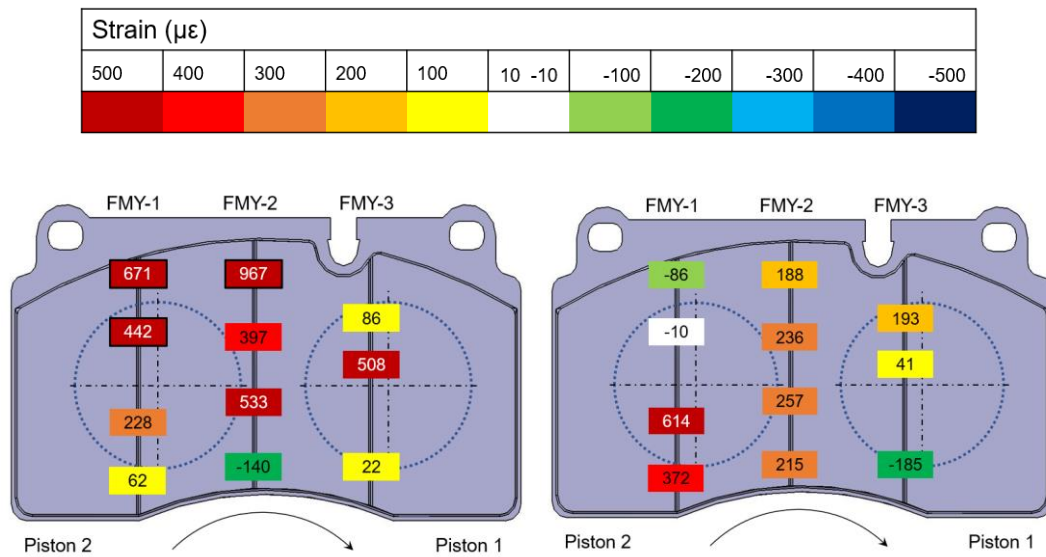
(a) FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right) ($p_1=80\text{bar}$ and $p_2=70\text{bar}$)



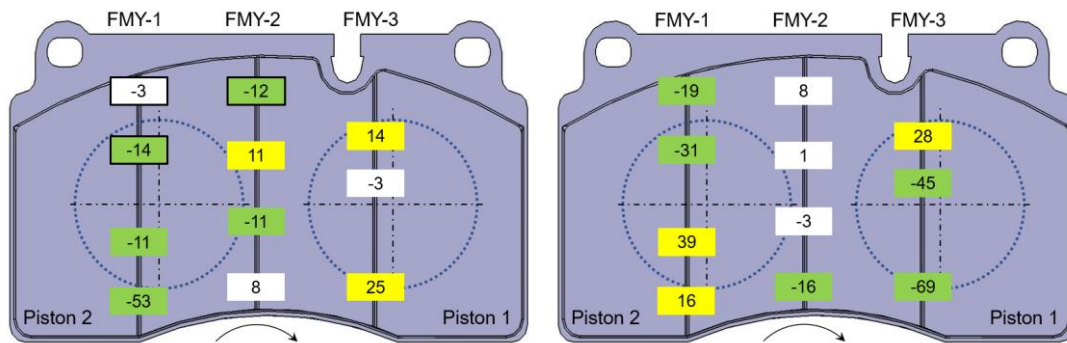
(b) Strain Differences: $\Delta\epsilon = \epsilon_{+10\text{bar}} - \epsilon_{\text{Equal}}$: FBG Test (Left) and FEA Simulation (Right) ($p_1=80\text{bar}$ and $p_2=70\text{bar}$)

Figure 7-29: E22 (Y Direction) Strain Results for +10bar Pressure Variation, 600Nm

Figure 7-30 shows a reverse pressure level variation ($p_1 = 70\text{ bar}$ and $p_2 = 80\text{ bar}$). Again, E22 strain differences (Y Direction) between FBG measurements and FEA predictions are quite pronounced in most locations (Figure 7-30 (a)). When comparing FBG test with equal pressure ($p_1 = p_2 = 80\text{ bar}$) the strain differences are small (Figure 7-30 (b), left side), some are positive, some negative and look randomly distributed, i.e. it would be difficult to relate them (location-wise) to the actual reduction of pressure under piston 1. FEA results (Figure 7-30 (b), right side) show very similar randomness and small strain changes.



(a) FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right) ($p_1=70\text{bar}$ and $p_2=80\text{bar}$)

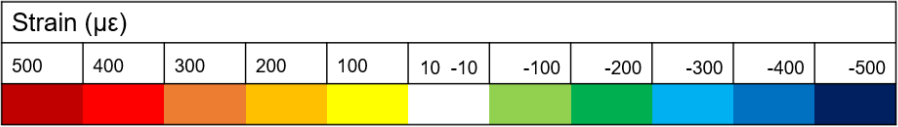
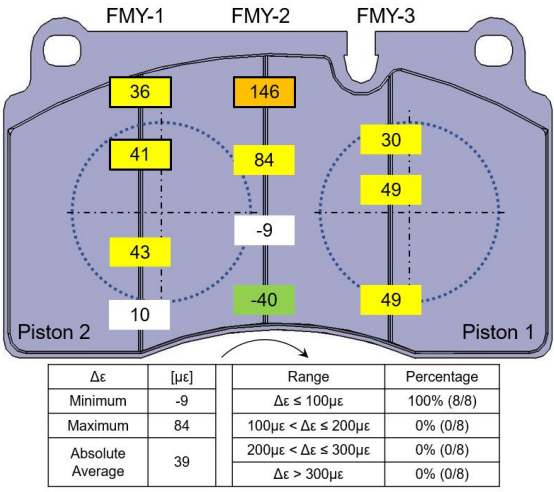


(b) Strain Differences: $\Delta\epsilon = \epsilon_{-10\text{bar}} - \epsilon_{\text{Equal}}$: FBG Test (Left) and FEA Simulation (Right) ($p_1=70\text{bar}$ and $p_2=80\text{bar}$)

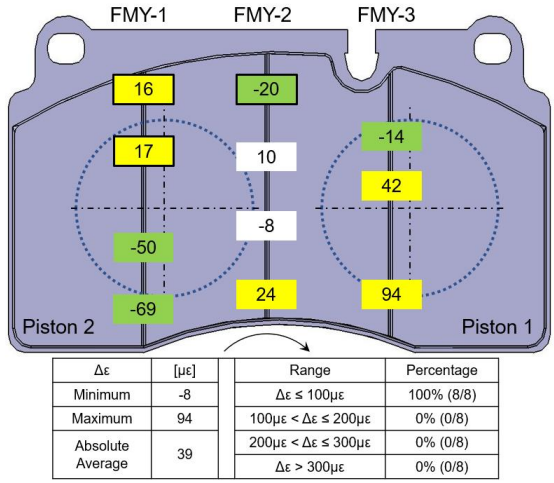
Figure 7-30: E22 (Y Direction) Strain Results for -10bar Pressure Variation, 600Nm

Figure 7-31 (a) presents calculated strain differences between FBG and FEA for $p_1 = 80\text{ bar}$ and $p_2 = 70\text{ bar}$, at 600 Nm torque (for strains shown in Figure 7-29 (b)). Figure 7-31 (b) presents calculated strain differences between FBG and FEA for $p_1 = 70\text{ bar}$ and $p_2 = 80\text{ bar}$, at 600 Nm torque (for strains shown in Figure 7-30 (b)). In both cases, looking from left to right, at piston 2 (FMY-1), the results are very similar, with differences being within $50\mu\epsilon$. However, in the middle area of the pad (FMY-2), for $p_1 = 80\text{ bar}$ and $p_2 = 70\text{ bar}$ (Figure 7-31 (a)), one upmost position (towards disc OD) has high differences (being $84\mu\epsilon$), with the strain differences being much smaller ($-9\mu\epsilon$ and $-40\mu\epsilon$) towards disc ID. Interestingly,

for the reverse pressure variation, $p_1 = 70$ bar and $p_2 = 80$ bar (Figure 7-31 (b)), strain differences are quite small, from OD towards ID being only $10\mu\epsilon$, $-8\mu\epsilon$ and $24\mu\epsilon$. Such strain comparisons prove that FBG tests and FEA simulations are in good agreement.



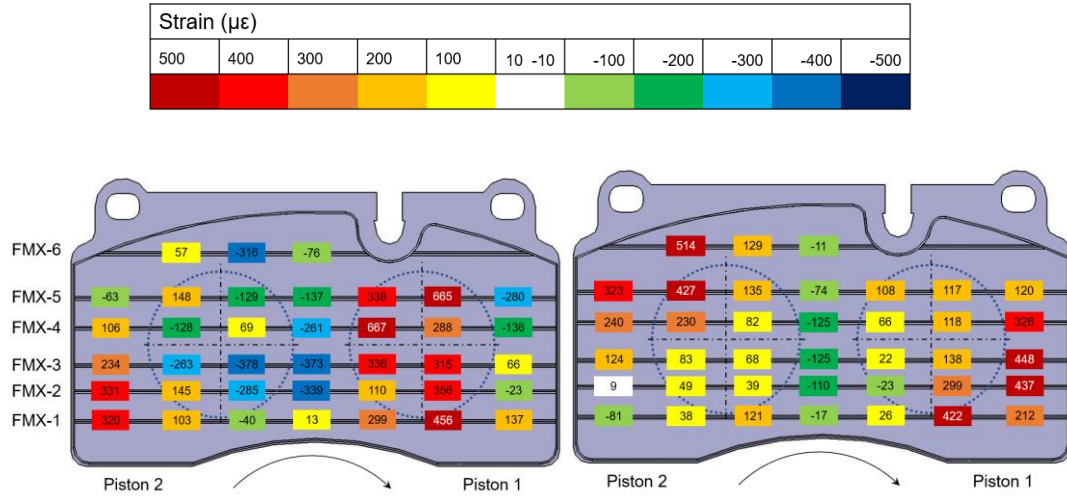
(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: $p_1=80$ bar; $p_2=70$ bar



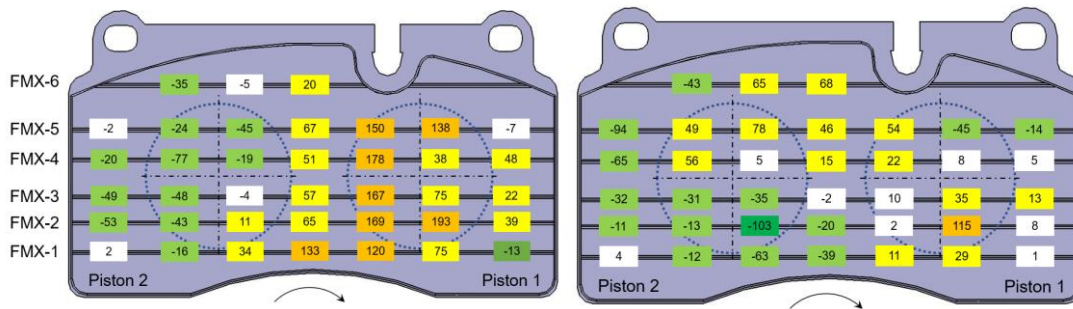
(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: $p_1=70$ bar; $p_2=80$ bar

Figure 7-31: FBG and FEA E22 (Y Direction) Strain Differences for Two Hydraulic Pressure Variations, 600Nm Torque Level

Furthermore, the analyses were also conducted of E11 strain (X direction) for the same nominal actuating hydraulic pressure of 80 bar with 600 Nm torque level, the results obtained for 40 bar hydraulic pressure with 200 Nm torque level are illustrated in Appendix Chapter 7-E.



(a) FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right) ($p_1=80\text{bar}$ and $p_2=70\text{bar}$)



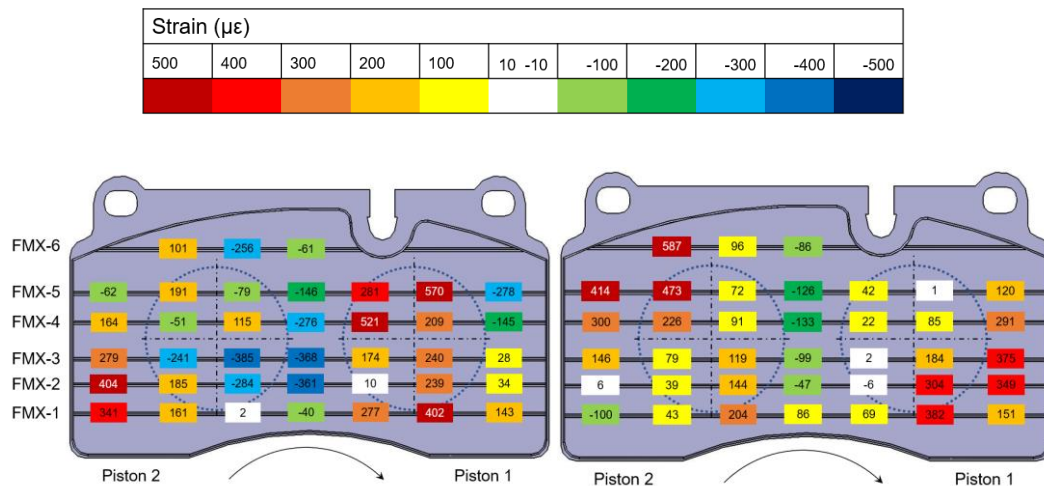
(b) Strain Differences: $\Delta\epsilon = \epsilon_{+10\text{bar}} - \epsilon_{\text{Equal}}$: FBG Test (Left) and FEA Simulation (Right) ($p_1=80\text{bar}$ and $p_2=70\text{bar}$)

Figure 7-32: E11 (X Direction) Strain Results for +10bar Pressure Variation, 600Nm

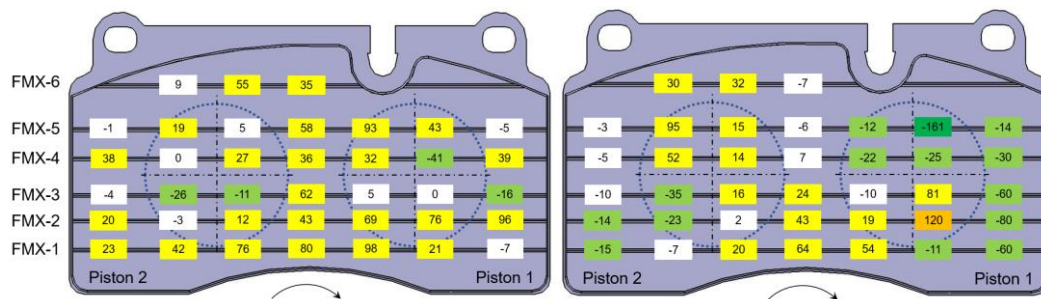
In Figure 7-32, FBG and FEA comparisons of E11 strain (X Direction) is shown for 80 bar at piston 1 and 70 bar at piston 2. When comparing FBG and FEA results (Figure 7-32 (a)), quite significant differences can be observed in practically all positions. In comparison to the equal pressure (Figure 7-32 (b)), E11 strain (X Direction) seems to be changing in a different trend, FBG tests recorded strain increase in right side of the brake pad (the piston 1 applied with

80 bar), and decrease in the left side, where $p_2 = 70$ bar. However, with FEA simulations, the increase was detected in the middle top side and right bottom side of the brake pad, while in other locations decreasing trends can be observed.

Figure 7-33 shows a reverse pressure level variation ($p_1 = 70$ bar and $p_2 = 80$ bar). Again, E11 strain differences (X Direction) between FBG measurements and FEA predictions are quite pronounced in most locations (Figure 7-33 (a)). When comparing FBG tests with equal pressure ($p_1 = p_2 = 80$ bar), the strain differences are mostly lower than $100\mu\epsilon$, even under $50\mu\epsilon$ (Figure 7-33 (b), left side), and only four locations display clear negative trends. However, FEA results (Figure 7-33 (b), right side) show a different trend, where the right side of the brake pad experiences the negative trend at piston 1 with 70 bar pressure applied.



(a) FBG Test (Left) and FEA Simulation [$\mu=0.46$] (Right) ($p_1=70$ bar and $p_2=80$ bar)



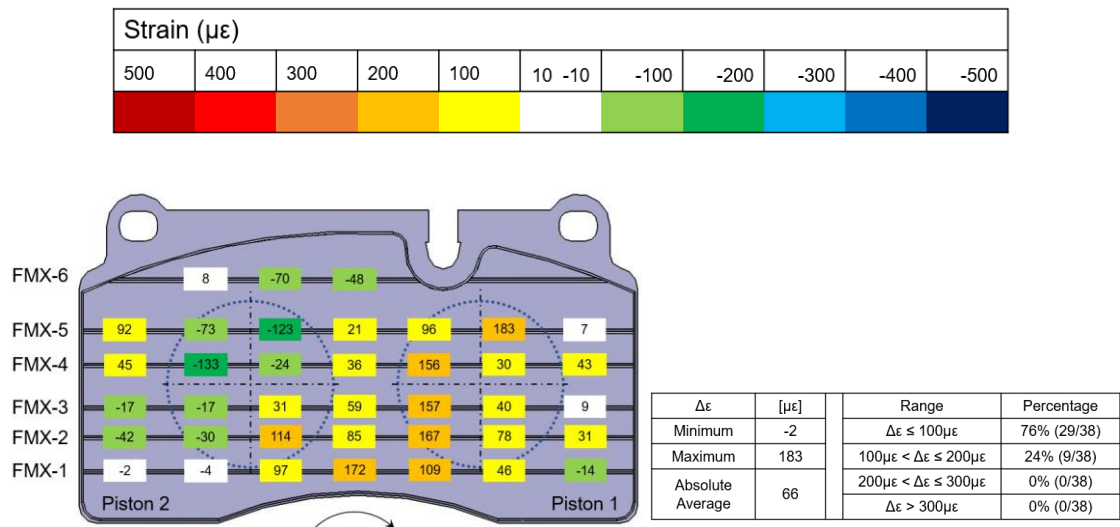
(b) Strain Differences: $\Delta\epsilon = \epsilon_{-10bar} - \epsilon_{Equal}$: FBG Test (Left) and FEA Simulation (Right) ($p_1=70$ bar and $p_2=80$ bar)

Figure 7-33: E11 (X Direction) Strain Results for -10bar Pressure Variation, 600Nm

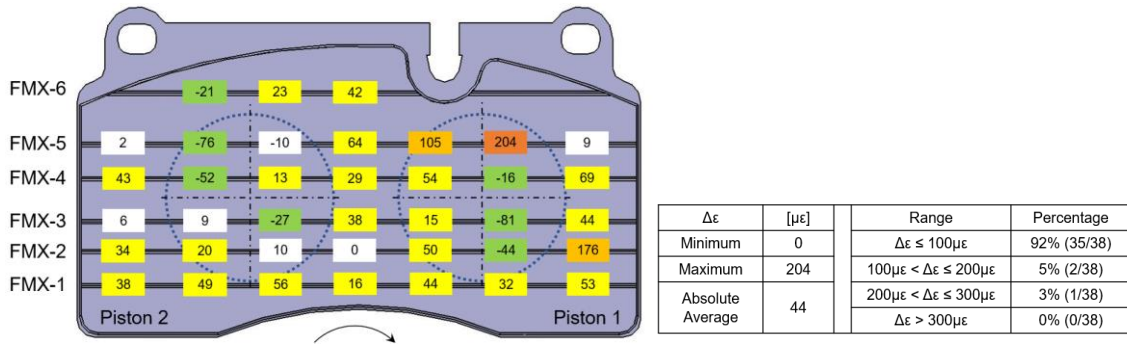
Strain comparisons for pressure variations (± 10 bar) to equal pressure ($p_1 = p_2$) was considered to be very useful in gaining deeper insights and confidence into test and simulation results, their absolute values at equal pressure, as well as changes with pressure variations. After these comparisons are made, the differences between test and simulation results can be calculated and a more informed discussions and conclusions made.

In Figure 7-34 (a), calculated differences between FBG and FEA are shown for $p_1 = 80$ bar and $p_2 = 70$ bar, with Figure 7-34 (b) showing the reverse pressure variation, $p_1 = 70$ bar and $p_2 = 80$ bar. In the first variation (Figure 7-34 (a)) strain differences are quite pronounced in some locations in the areas under the pistons. The maximum difference under piston 1 area reaching $-133\mu\epsilon$, with FBG results values being lower than FEA. Under piston 2, the differences are even more pronounced, with the maximum values being $183\mu\epsilon$ but test results are now higher than FE predictions. Despite some big differences, in many locations under the pistons, and in particular in the middle of the pad and both edges (leading and even more so on the trailing edge) strain differences are relatively small, and with only several exceptions, often under $50\mu\epsilon$. The summary table data best illustrates the above statements, with the differences in 76% of the sensor locations being under $100\mu\epsilon$, with the remaining 24% being between $100\mu\epsilon$ and $200\mu\epsilon$.

For the reverse pressure variation, $p_1 = 70$ bar and $p_2 = 80$ bar (Figure 7-34 (b)), there is much better agreement between test and simulation data. The differences are again most prominent in the areas under the pistons but the differences are typically smaller. The exception is FMX5-FBG6 with the strain difference of $204\mu\epsilon$. This is the location with the biggest difference in the previous pressure variation of $183\mu\epsilon$, which most probably indicates some friction material irregularity in this area. Interestingly, in the close proximity and on the same fibre, at the trailing edge of the pad, the differences are only $9\mu\epsilon$ and $7\mu\epsilon$ respectively. As stated in the summary table, in 92% of the places, the strain differences are under $100\mu\epsilon$, with 5% (2/38) being between $100\mu\epsilon$ and $200\mu\epsilon$, and only in one place (1/38, i.e. 3%,) being just over $200\mu\epsilon$.



(a) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=80bar; p2=70bar



(b) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=70bar; p2=80bar

Figure 7-34: FBG and FEA E11 (X Direction) Strain Differences for Two Pressure Variations

7.4 Summary

In this chapter, the finite element (FE) model analysis has been applied to support and complement the FBG test results under quasi-dynamic conditions, which was another part of the research aims. It updated the previous static FE model to be appropriate for the simulation of various test conditions and brake behaviours, including under quasi-dynamic conditions. The test results obtained using FBG strain sensors have compared and validated with the outcomes from the FEA simulations in quasi-dynamic conditions. Particularly, the strain differences are relatively low to medium, being under $100\mu\epsilon$ over 75%, or often over 90%, in the brake pad area.

The quasi-dynamic loading required an updated FE model, in terms of mesh, disc portion modelled and its mesh, and above all different boundary conditions. The model was established and tested to ensure validity of the results obtained. The model and analysis stability were ensured and repeatable results obtained under all conditions, which were modelled to correspond to the test conditions. As expected, the model (results) is sensitive to the loading, i.e. pressure and torque levels. The coefficient of friction, in the range considered between 0.4 and 0.5, showed to have a relatively little impact but nevertheless more pronounced impact than under static loading, where only actuating pressure (force) was applied. This is perfectly logical and expected, with the in-plane loading of the pad friction surface realised by friction from the disc surface. The most influence on strain values has boundary conditions in pad/calliper (abutments) contacts. A most realistic contact was modelled, with only two diagonally opposite nodes restrained.

FEA modelling was conducted for the same loading conditions as during tests, with a most specific focus on 40 bar with 200 Nm torque (corresponding to typical light braking), 80 bar with 600 Nm torque (severe braking) and 120 bar with 1000 Nm torque (maximum performance emergency braking). Quasi-dynamic strain results were first compared with the static FE results in order to investigate changes in strain distributions and build confidence in the modelling. Strain values were suitably extracted and presented in the same manner and formatted as FE static and FBG strains, at the physical position of FBG sensors. In this manner, fair and accurate comparisons can be made under all loading conditions. Introduction of braking torque in FEA models showed to have a relatively small but nevertheless detectable and important change in strain distributions. Even before comparisons with the FBG results, it can be safely concluded that the strain change is logical, and follows expected trends with the introduction of torque for all pressure and torque levels studied. Furthermore, pressure variations of ± 10 bar showed important, though sometimes settled changes in strain distributions. These were particularly obvious for lower to higher pressure and torque levels (40 bar / 200 Nm and 80 bar / 600 Nm), with the emergency braking (120 bar / 1000 Nm) showing only small changes. This was to be

expected as the pressure change of ± 10 bar in one pair of pistons equates to only 8% of the baseline pressure and the torque modelled remained the same.

Furthermore, strains were compared for quasi-dynamic conditions before FBG test and FEA modelling, static and quasi-dynamic strains for each method were compared side by side (FBG static with FBG quasi-dynamic and FEA static with FEA quasi-dynamic). This is to facilitate establishing the baseline static strain distributions and gain insights into how quasi-dynamic loading changes these distribution trends for each of the two methods used, experimental (FBG) and modelling (FEA). Overall, FEA showed more realistic and often smaller but 'well distributed' changed with FBG having sometimes unexplainable higher and lower strain changes, 'disturbing the smooth strain distribution'. This is to be expected, considering much higher uncertainties are related to the experimental works.

Essentially, a general observation can be made that FEA results are logical, without sudden changes in strain values, as the pad geometry is relatively simple and uniform, and the applied pressure and torque loads are equally distributed over piston contact areas. As discussed before, various parameters of model and analysis were varied to gain confidence in FEA results but the components were considered to have a perfect geometry and the friction material to be homogeneous. FBG results vary more and in some locations the values are considerably lower or higher without an obvious reason.

Finally, FBG experimental strains were compared with FEA modelling strains at the sensor locations, for both strain directions and all loading conditions. Again, plotting strain differences $\Delta\varepsilon = \varepsilon_{Test} - \varepsilon_{FEA}$ prove to be the best method in gaining a global insight and locating places of good and poor data agreement. Overall, it can be stated that FBG and FEA strain results are quite similar, with exception of some areas, which were discussed in detail above. These are typically, relatively small areas. In some locations the FBG and FEA strain values are practically identical, in majority of the brake pad friction surface, for most loading cases, the strain differences are relatively low to medium, being under $100\mu\text{m}$ in over 75% or often even over 90% of the pad area.

The above finding is very encouraging, as the changes in strain values can be then be further investigated in dynamic conditions. Tracing these changes is crucial for monitoring processes at friction interface in actual service duties, which are used to determine brake performance, wear and NVH characteristics.

8 EXPERIMENTAL PHASE 3 – DYNAMIC ANALYSIS

The static and quasi-dynamic experimental phases described in Chapters 4 and 6 provide a general view of the use of the FBG strain sensors in the brake research area. This chapter concerns the dynamic analysis, which represents the first attempt to determine the reliability and efficiency of the fibre optical sensing technology in a brake assembly under dynamic conditions, with the disc rotating. The chapter is organised into several sections, including test preparations, experimental methodology, analysis of test results, and brake pad surface condition analysis.

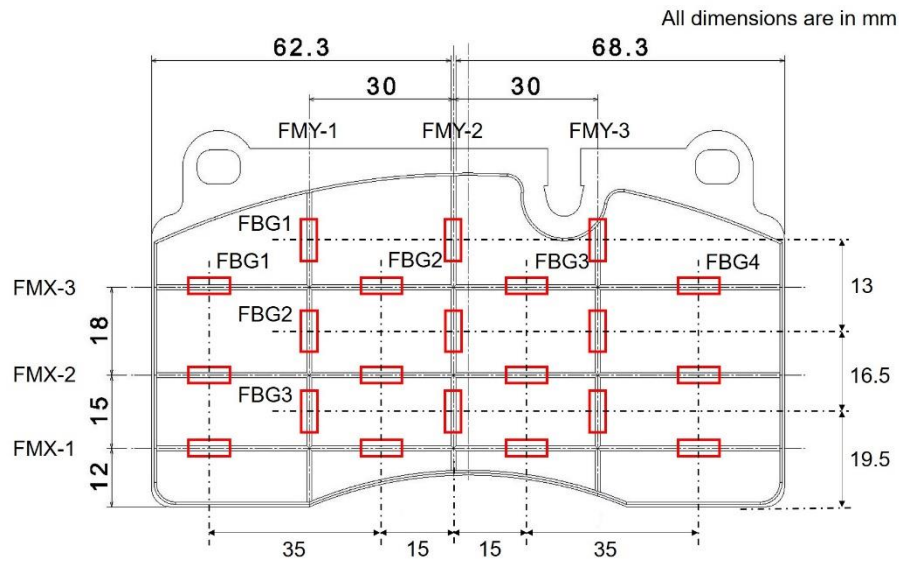
8.1 Dynamic Test Preparations

8.1.1 Brake Pad Modification (Cranfield University)

The modified brake pads used in the static and quasi-dynamic experiments contained either circumferential (X) or vertical (Y) oriented grooves, as illustrated in Figure 3-21. However, dynamic conditions are much more complicated. Sequential measurements in the X and Y directions with separate brake pads will not be sufficient to characterise the brake performance under dynamic conditions, as the condition of the brake pad will have changed after each brake application, so that using the X-direction pad in one brake application and then the Y-direction pad in a repeated measurement (nominally 'identical') will not allow the interface strain distributions to be reliably determined. In particular, wear debris will influence the contact, furthermore, the disc and particularly the pad will wear, heat generation will be altered, and inevitably pad and disc mechanical and thermal deformation will be different. All these effects, however small they may be individually, their cumulative effect may have considerable influence on the local pad strain and the distribution over the friction surface. After all, all brake applications under medium and higher loads operate under thermoelastic instability (TEI) regime, meaning continuous change in contact pressure, wear and temperature. Consequently, it is necessary to make measurements simultaneously in both the X and Y directions.

A new modified brake pad was designed and manufactured as shown in Figure 8-1. A detailed set of CAD drawings can be found in Appendix Chapter 8-A. On the friction material surface, the modified pad contains six grooves, with three grooves oriented in the X direction and three in the Y direction. The dimensions of each groove are 1mm in width and 2mm in depth. The design ensures that the optical fibres, oriented in the two directions, have enough space to overlap and cross over in the intersections between grooves, without inducing significant bend losses, and that the optical fibres can be installed with their surface below the frictional surface, ensuring that the operation of the disc brake would not be adversely influenced during dynamic applications. That means that friction material wear and accumulated wear debris, despite small number of brake applications, will not impose any additional load onto the fibres. At the intersections of the grooves, the optical fibres deployed in X direction grooves cross over the top of those deployed in the Y direction grooves. The optical fibres were bonded into the grooves using cyanoacrylate adhesive, as described in section 3.3.2.

Each optical fibre bonded into the X direction groove contained four FBG sensors, while those bonded into the Y direction grooves contained three FBGs. Thus, the interface strain distributions were characterised using 21 FBG strain sensors in total. The sensor numbering convention followed that for the modified brake pads used in the static and quasi-dynamic experiments, where the numbering of sensors oriented in the X direction start from the left (FBG1) to the right (FBG4), and in the Y direction numbering begins from the top (FBG1) to the bottom (FBG3). The sensor naming combines the fibre with the sensor (FMX1-FBG1, FMY1-FBG1) to ensure the recorded test data can be matched to the data acquired in the static and quasi-dynamic loading experiments.



(a) Design Concept



(b) Final Product

Figure 8-1: New Modified X&Y Direction Brake Pad

The new modified X and Y direction brake pad (Figure 8-1) was characterised under static loading conditions in the laboratory in Cranfield University. The initial tests employed a hydraulic pressure of 30 bar with equal pressure on all pistons. As before, the limitation on the number of measurement channels on the FBG interrogator meant that X and Y direction fibres were monitored in sequential experiments, with a total of 3 repeats for each fibre direction. The experiments were undertaken with the aim of ensuring that the installed FBG sensors worked

correctly. The initial (static) test results, shown in Appendix Chapter 8-B, exhibited a high-level of repeatability.

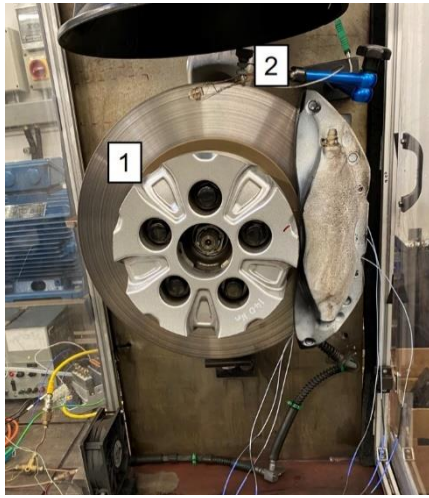
8.1.2 Dynamic Test System and Schedule (University of Bradford)

8.1.2.1 Test System

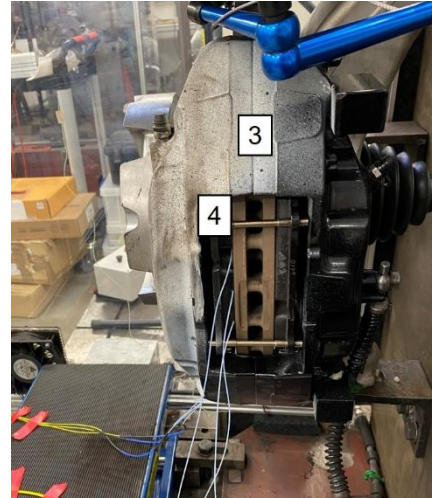
The dynamic tests were undertaken in the Braking Research Centre (2022) at the University of Bradford. Figure 8-2 illustrates the Rig and Brembo Land Rover disc brake assembly. The brake assembly (Figure 8-1 (a) and (b)), is identical to the assembly used in the Cranfield University. However, it was necessary to use another set of disc and calliper. On the friction surface of the brake disc (1), a rubbing thermocouple sensor (2), was installed to monitor surface temperature. This arrangement is widely used in automotive industry. In addition to providing information about the initial disc temperature and temperature changes during braking, disc temperature was necessary in order to compensate the wavelength shift (processed as the interface strain) for thermal gain. The brake calliper (3) accommodated modified brake pads (4) with FBG strain sensors installed. As in previous tests (static and quasi-dynamic), the pad's 'anti-rattle' spring was omitted in order to enable unobstructed exit of optical fibres.

The brake assembly (Figure 8-2 (c)) was installed within the enclosure (5) for safety (rotating parts) and health (dust extraction) reasons. Optical FBG measurement system (6) was set-up in the proximity of the enclosure. Figure 8-2 (d) gives overall view of the Rig, i.e. the test system for the dynamic brake application. The variable speed fan/extractor (7) enables dust extraction and brake cooling. The brake disc is driven by an electric motor (8), having a maximum rated power of 22 kW and maximum speed of 1460 rpm, controlled by an inverter. The gear box (9), having a ratio of 30:1, is used to increase electric motor torque (with inevitable drop in speed to about 48.7 rpm) and change of the shaft axis by 90°. The output from the gearbox is connected to the wheel flange (on which the disc is installed) by a vehicle driveshaft (with CV joints on both ends). The air-over-hydraulic system (10) is used to actuate the brake with a computer-controlled pressure level. For such a large disc and calliper, the maximum hydraulic pressure that can be used is about 30 bar. Higher pressure

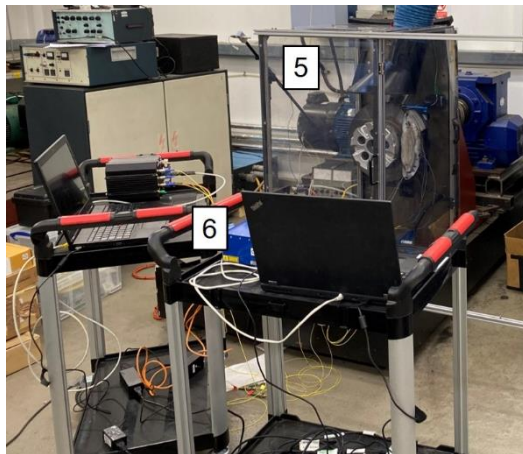
would generate too much clamp force and torque level that would stop the motor from rotating.



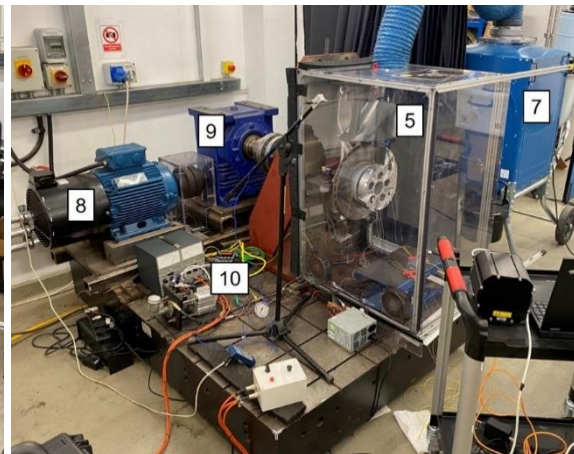
(a) Front View of Disc Brake System



(b) Side View of Disc Brake System



(c) Setup of The FBG Measurement



(d) Setup of The Dynamic Application

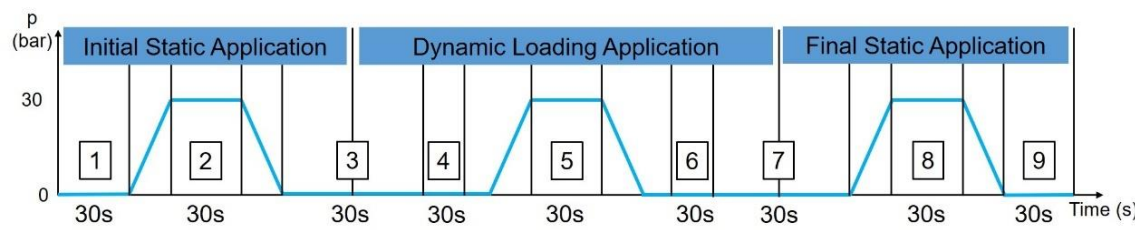
Figure 8-2: The Dynamic Test System at The University of Bradford

It is important to point out that FBG measurement system (position 6 in Figure 8-2 (c)) consists of two SmartScan FBG interrogators. They are the same as used for static and quasi-dynamic tests, having four channels each. For the static and quasi-dynamic experiments, where a total of 6 sensor arrays were installed on the modified brake pads, this meant that each measurement had to be repeated with different combinations of sensors attached in order to record both X and Y direction strain under the notionally same test conditions. This was no problem (except for longer test times) as the measurements were highly repeatable (and

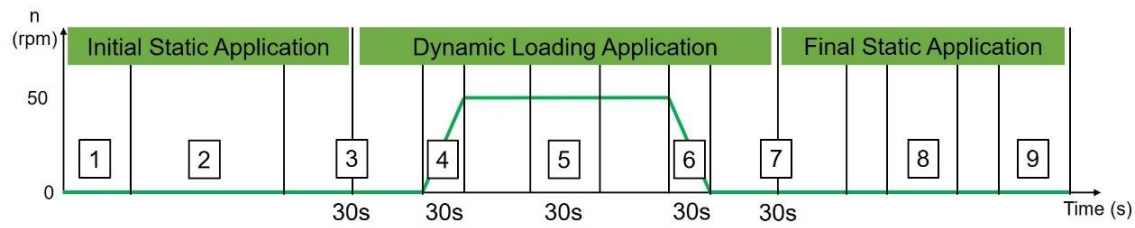
the results presented were averages from 3 measurements). However, the nature of the dynamic tests meant that it was necessary to monitor all 21 FBG strain sensors simultaneously. Consequently, two SmartScan FBG interrogators were used to acquire the data from a total of 8 sensor arrays simultaneously.

8.1.2.2 Test Schedule

Different tests were performed, as detailed later on but it is necessary to explain here the initial test plan. The dynamic tests were all conducted at the same input loading, i.e. disc rotational speed was (approx.) 50 rpm and hydraulic pressure 30 bar in all 4 pistons (no pressure variations per piston pairs were employed). Figure 8-3 shows the changes in pressure (Figure 8-3 (a)), speed (Figure 8-3 (b)), and details steps and periods (Figure 8-3 (c)).



(a) Hydraulic Pressure Level



(b) Electric Rotor Speed

Step	Initial Static Application			Dynamic Loading Application			Final Static Application		
Period	1	2	3	4	5	6	7	8	9
p (bar)	0	30	0	0	30	0	0	30	0
n (rpm)	0	0	0	0 - 50	50	50 - 0	0	0	0

(c) Full Steps and Periods

Figure 8-3: Dynamic Test Schedule and Validation

Each experiment consisted of three steps (Figure 8-3); an initial static pressure application, a dynamic load application, and a final static pressure application. Each step comprised three periods, where each period was of 30 seconds duration. The initial static pressure application step involved increasing the pressure from 0 bar to 30 bar (Period 1), maintaining the pressure at 30 bar for 30s (Period 2) and then reducing the pressure back to 0 bar (Period 3). This step was undertaken to ensure that the FBG sensors were operating correctly and consistently before each dynamic test run. The second step concerns the dynamic load application, where the electric motor rotation rate was increased to 50 rpm (Period 4), followed by the application of 30 bar hydraulic pressure for 30s (Period 5). The hydraulic pressure was then released to 0 bar, and the motor rotation rate then reduced to 0 rpm (Period 6). The final, the third step involved a static pressure application (Period 7 to Period 9), repeating the first step, in order to compare the strains before and after the dynamic loading application. The test plan explained here is the main loading condition discussed in the following sections 8.2 and 8.3. A further two specific test conditions will be explained in sections 8.4 and 8.5 respectively.

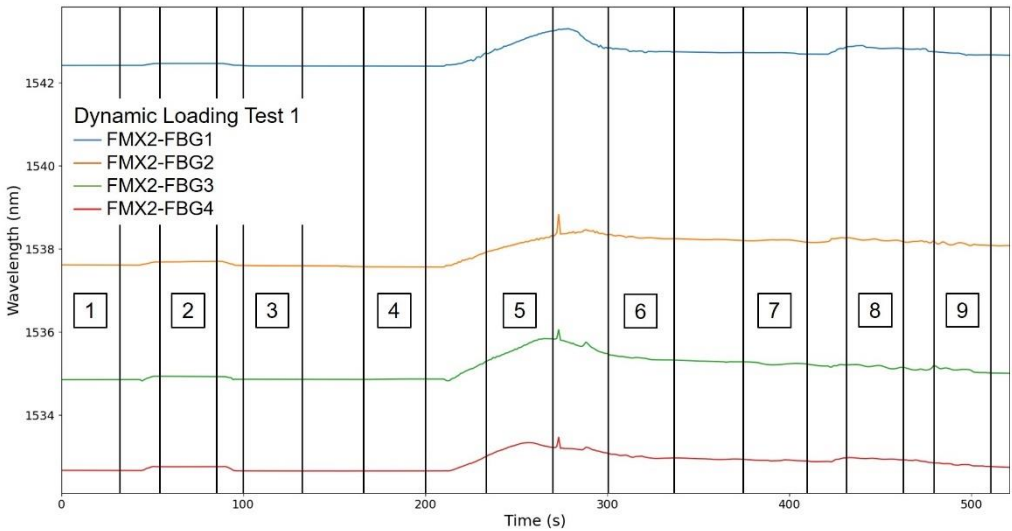
8.2 Dynamic Loading Test 1

8.2.1 Raw Recorded Data Graphs

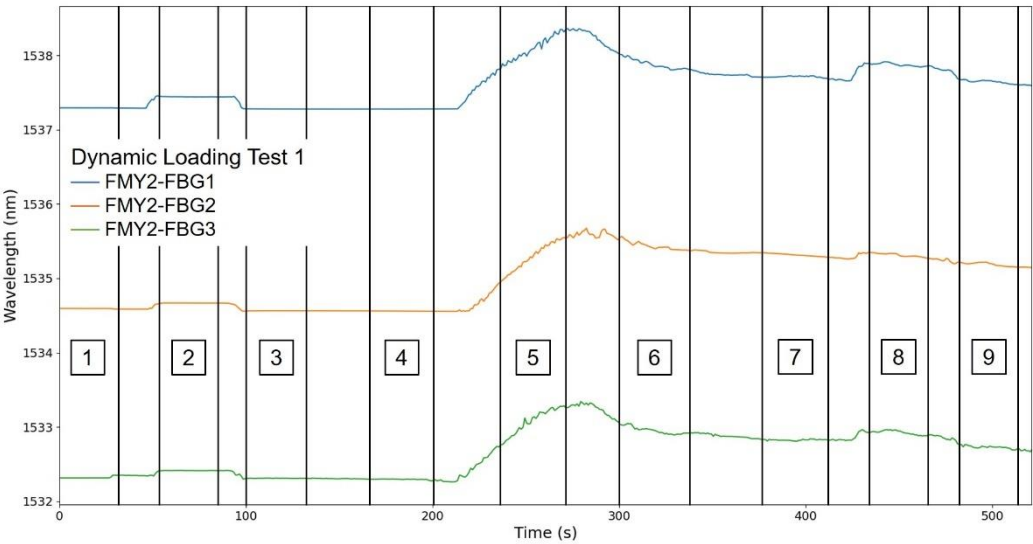
An initial run was undertaken before the formal test series to ensure that all relevant facilities operated correctly. The “Dynamic Loading Test 1” followed the test plan introduced in section 8.1.2, and with the raw recorded wavelength shift graphs plotted in Figure 8-4, including the data from FMX2 (a) and FMY2 (b). Each plot is marked with the period number to allow the test conditions to be linked to the data. At the first step, the initial static pressure application, the hydraulic pressure was maintained at 30 bar during Period 2, observing the wavelength increase as expected. Period 5 corresponds to the dynamic load application run, with 30 bar hydraulic pressure and 50 rpm rotation speed, where the wavelength shift can be seen to increase over the entire 30 seconds. During Period 6, the wavelength decreased as the hydraulic pressure was released and the electric rotor stopped completely. The hydraulic pressure applied during

Period 8 was the same as that in the first step (Period 2), and the comparison of measurement observed during Periods 2 and 8 will be discussed in section 8.2.4.

Step	Initial Static Application			Dynamic Loading Application				Final Static Application	
Period	1	2	3	4	5	6	7	8	9
p (bar)	0	30	0	0	30	0	0	30	0
n (rpm)	0	0	0	0 - 50	50	50 - 0	0	0	0



(a) Dynamic Loading Test 1: FMX2



(b) Dynamic Loading Test 1: FMY2

Figure 8-4: “Dynamic Loading Test 1” Raw Recorded Data Graphs

According to the raw recorded data shown in Figure 8-4 (a), all four FBG sensors in optical fibre FMX2 showed similar trends, the differences in values were a result of the differences in their locations on the brake pad. Meanwhile, the FBG sensors in optical fibre FMY2 also displayed similar trends and values over the duration of dynamic loading test 1, as shown in Figure 8-4 (b).

8.2.2 Data Processing

Considering the raw recorded data presented in section 8.2.1, the wavelength changes include contributions from the pressure-induced data and from the temperature changes. Therefore, further processing was needed to determine the actual strain for further analysis. Equation (8-1) reveals the general theory relating the wavelength changes to the influences of strain and temperature:

$$\Delta_{Wavelength} = k_{\varepsilon} \cdot \varepsilon + k_T \cdot T \quad (8-1)$$

Where $k_{\varepsilon} = 1 \text{ pm}/\mu\varepsilon$, and $k_T = 10 \text{ pm}/^{\circ}\text{C}$ (Othonos & Kalli, 1999).

Looking back on Equation (4-1) and developing Equation (8-1), which relates the wavelength changes to the strain, Equation (8-2) indicates how the strain can be recovered independently of the temperature, provided a measurement of temperature is available, as explained in section 8.5:

$$\varepsilon_{True} = \varepsilon_{Measured} - k \cdot (T_i - T_0) \quad (8-2)$$

Where $k = 10 \mu\varepsilon/^{\circ}\text{C}$ (Othonos & Kalli, 1999), T_i is the temperature at the selected time, and T_0 is the temperature at the beginning of the experiment.

The FBG sensor derived contour maps will be maintained with processed strains to be used in section 8.2.4 and further sections to compare the behaviour of the brake pad under different test conditions.

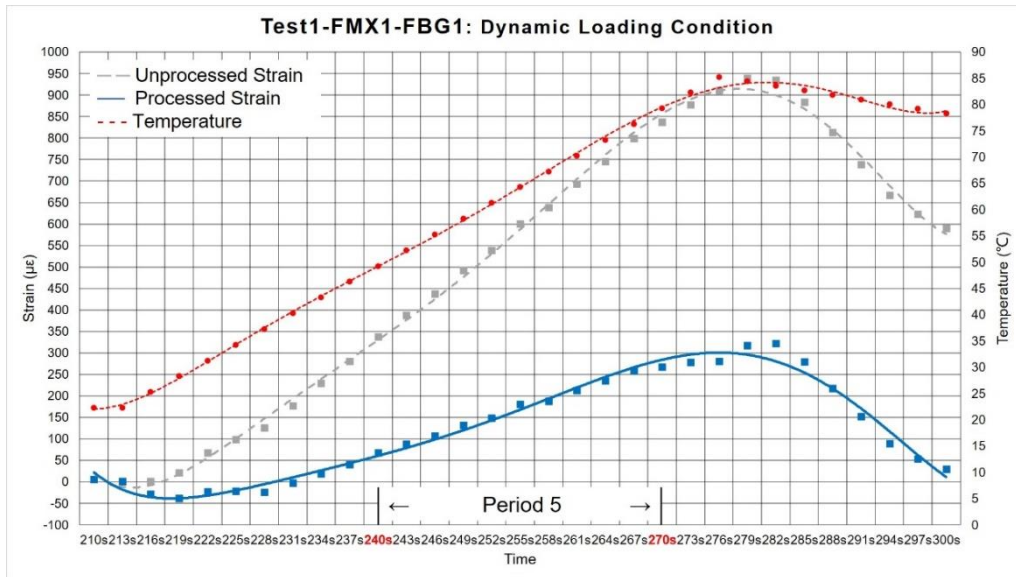
8.2.3 Processed Data Graphs and Contour Map

The processed data graphs plot the time series over the full dynamic loading application step, of a duration of 90 seconds with each 3 second as one data point, as shown in Figure 8-5, which presents the responses of sensor elements

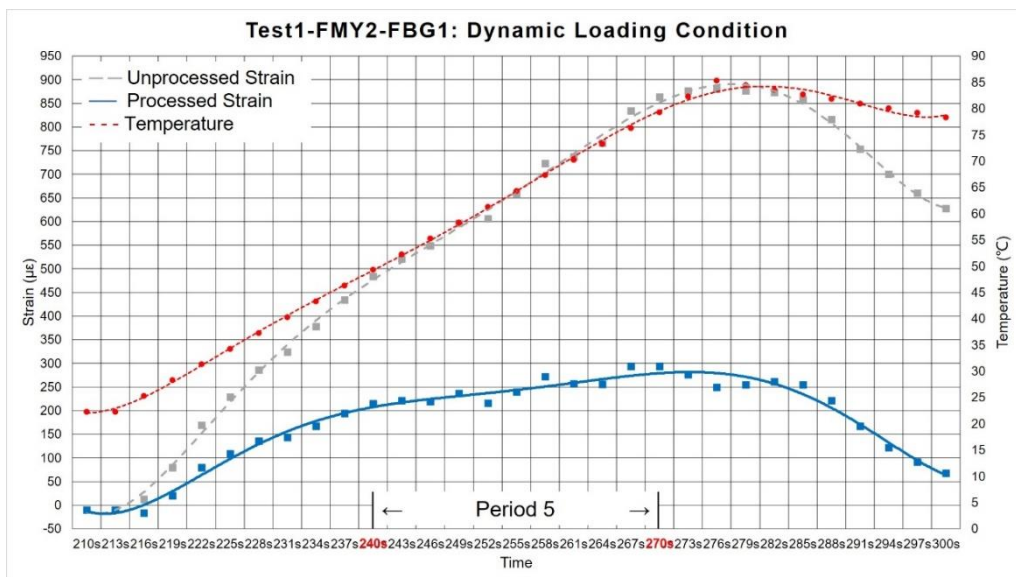
FMX1-FBG1 (Figure 8-5 (a)) and FMY2-FBG1 (Figure 8-5 (b)) as an example. Period 5, between 240 and 270 seconds, corresponds to the actual dynamic loading, the application of 30 bar hydraulic pressure at disc rotating at 50 rpm, as highlighted. Over the 30 seconds prior to this (210s-240s), the hydraulic pressure was increased from 0 bar to 30 bar in a gradual manner, while during the final 30 seconds (270s-300s) the hydraulic pressure was reduced gradually back to 0 bar. Red dashed line shows the temperature measured by the rubbing thermocouple.

The strain graphs in Figure 8-5 indicate that there is a considerable influence of temperature, which must be taken into account. The maximum unprocessed (raw) strain measured by the sensor at FMY2 (Figure 8-4 (b)) was $900\mu\epsilon$, which reduced to $300\mu\epsilon$ after the temperature compensation (Figure 8-5 (b)). Similarly, the maximum unprocessed strain measured by the sensor at FMX2 (Figure 8-4 (a)) was around $900\mu\epsilon$, becoming about $325\mu\epsilon$ after temperature compensation (Figure 8-5 (a)). Meanwhile, the peak strain of both sensors was observed at around 279s and 282s, which was a bit delay after completing Period 5 as the hydraulic pressure system needed about 10 to 20 seconds to reduce pressure back to 0 bar. The further analysis and discussion will be conducted to select a suitable time period for comparison in order to avert such an issue, which might influence test results.

Processed data graphs of the type presented in Figure 8-5 can display the trends of the measurements provided by a single sensor only. In order to gain a fuller understanding of the interface strain distributions across the frictional surface, the FBG sensor contour map was created, as explained earlier on for static (Chapters 4 and 5) and quasi-dynamic (Chapters 6 and 7) conditions. As the dynamic loading conditions were continuously changing, the last second of Period 5 was selected to compare and display the strain distributions (at the 270th second), as shown in Figure 8-6. The strain data illustrated as FBG sensor contour maps in this chapter is compensated for the influence of temperature.



(a) Test1-FMX1-FBG1



(b) Test1-FMY2-FBG1

Figure 8-5: Dynamic Loading Test 1 Processed Data Graphs

The contour map of the data captured at the final second (the 270th second) of the dynamic loading (Period 5) during Test 1 shows that most of the sensors worked well. Only one sensor, FMY1-FBG2, displayed spurious data, which may have been caused by debris or compression, where peak splitting occurred due to the transverse loading. The highest strain was exhibited at FMY3-FBG1, located at the edge of the pad, near the wear sensor installation hole, causing

the increased strain, as was explained in section 5.2.2.2. The lowest strain was observed at FMX3-FBG3, where the nearby sensors exhibited similar values. Generally, the sensors in the optical fibre FMY2 showed strain levels between 200 $\mu\epsilon$ and 300 $\mu\epsilon$.

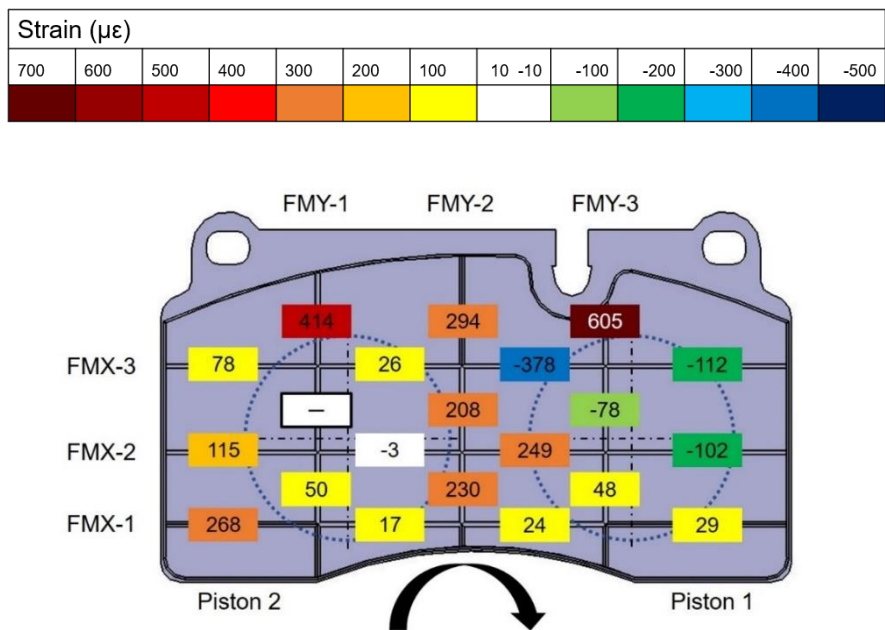


Figure 8-6: Dynamic Loading Test 1 Strain Distributions at the End of Brake Dynamic Loading Application (t=270s)

The left side of the brake pad, which was the leading edge due to the anti-clockwise electric rotor rotation, exhibited higher strains that were observed on the right side, the trailing edge, illustrating the same trends observed during both static (Chapter 5) and quasi-dynamic (Chapter 7) experiments. Meanwhile, the top-side of the brake pad, corresponding to the brake disc's outer diameter (OD), showed higher strain than the brake disc's inner diameter (ID), because the pad/disc contact surfaces of the brake disc ID and OD were different, where the outer diameter has touched a larger proportion of the total contact surfaces.

In addition to the contour map at the end of the dynamic loading (Period 5), as shown in Figure 8-6, it is interesting looking at the strain distribution throughout the (dynamic) brake application, the entire Period 5, from 240s to 270s. Figure 8-7 and Figure 8-8 show strain distributions during Period 5 every 3 seconds. It was concluded that this is the optimal timestep for displaying data as the change

is not very rapid and shorter time periods would not provide more information. Due to the large amount of information and smaller plots, in order to ease the dynamic overview of the strain changes over the entire pad and the entire Period 5, the actual strain values were not included in the sensor positions (as included in Figure 8-6 and elsewhere). The emphasis here is on the dynamic change, with the actual values having less importance (and being included elsewhere). The identical colour legend was used, making the comparisons easy. Furthermore, plot in Figure 8-7 (f) at the 15 seconds ($t = 255s$), is repeated in Figure 8-8 (a) to facilitate easier tracings of the changes in strain distributions in both halves of the brake application.

Looking at the results from the beginning of brake application (Period 5), the plots in Figure 8-7 (a) and Figure 8-7 (b), show little change in the first 3 seconds. It should be kept in mind that difference between neighbouring colours is $100\mu\epsilon$, which means that the strain was changing but under $100\mu\epsilon$ in all sensor positions, except for the position FMX1-FBG4, where the strain increased by about $400\mu\epsilon$ and the position FMX3-FBG4, where the increase was about $100\mu\epsilon$. Looking further into the braking event, it can be seen that 3 seconds later, at 6 seconds (Figure 8-7 (c)), at the position FMX1-FBG4, the strain drops by about $200\mu\epsilon$ but there are no substantial changes in any of the other positions. It may well be possible that at FMX1-FBG4, the sudden high increase in E11 (X direction) strain for 3 seconds into the braking was caused by wear debris being pushed over the sensor at that position, as 3 seconds later (at 6 seconds) the strain drop is also considerable, although not returning to the level at the beginning of the brake application.

From 6 to 9 seconds into brake application (Figure 8-7 (c) and Figure 8-7 (d)) strain increases by about $100\mu\epsilon$ at two positions, FMX2-FBG3 and FMY3-FBG1. Then, strain increases in three positions for about $100\mu\epsilon$, FMX1-FBG4, FMY1-FBG1 and FMY2-FBG3, from 9 to 12 seconds (Figure 8-7 (d) and Figure 8-7 (e)). Finally, Figure 8-7 (e) and Figure 8-7 (f), from 12 to 15 seconds, strains of two positions are increased, FMX1-FBG3 and FMX1-FBG4, and another two positions, FMX3-FBG3 and FMY1-FBG1, the strain decreases by about $100\mu\epsilon$.

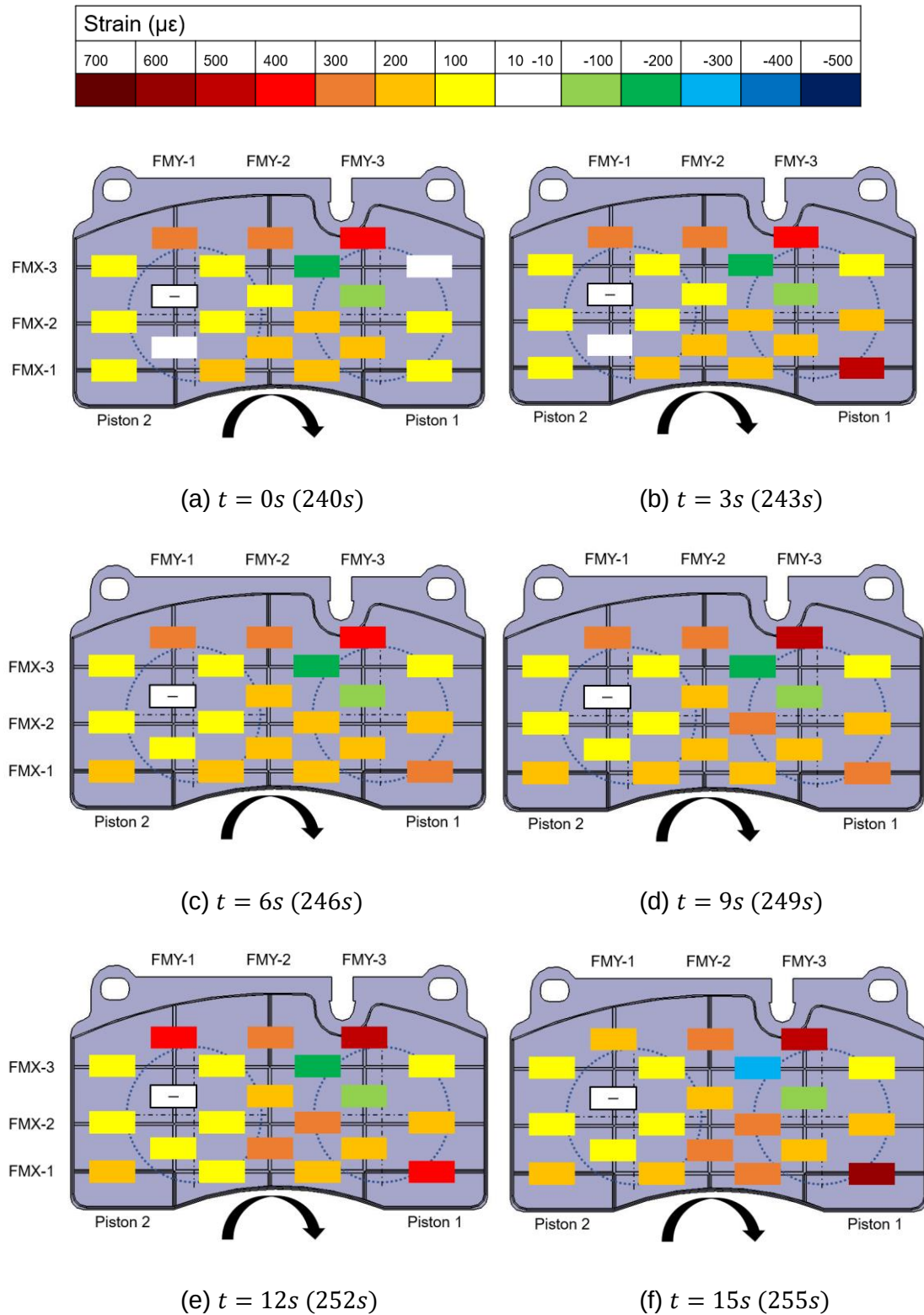


Figure 8-7: Dynamic Loading Test 1 Process from 240s to 255s

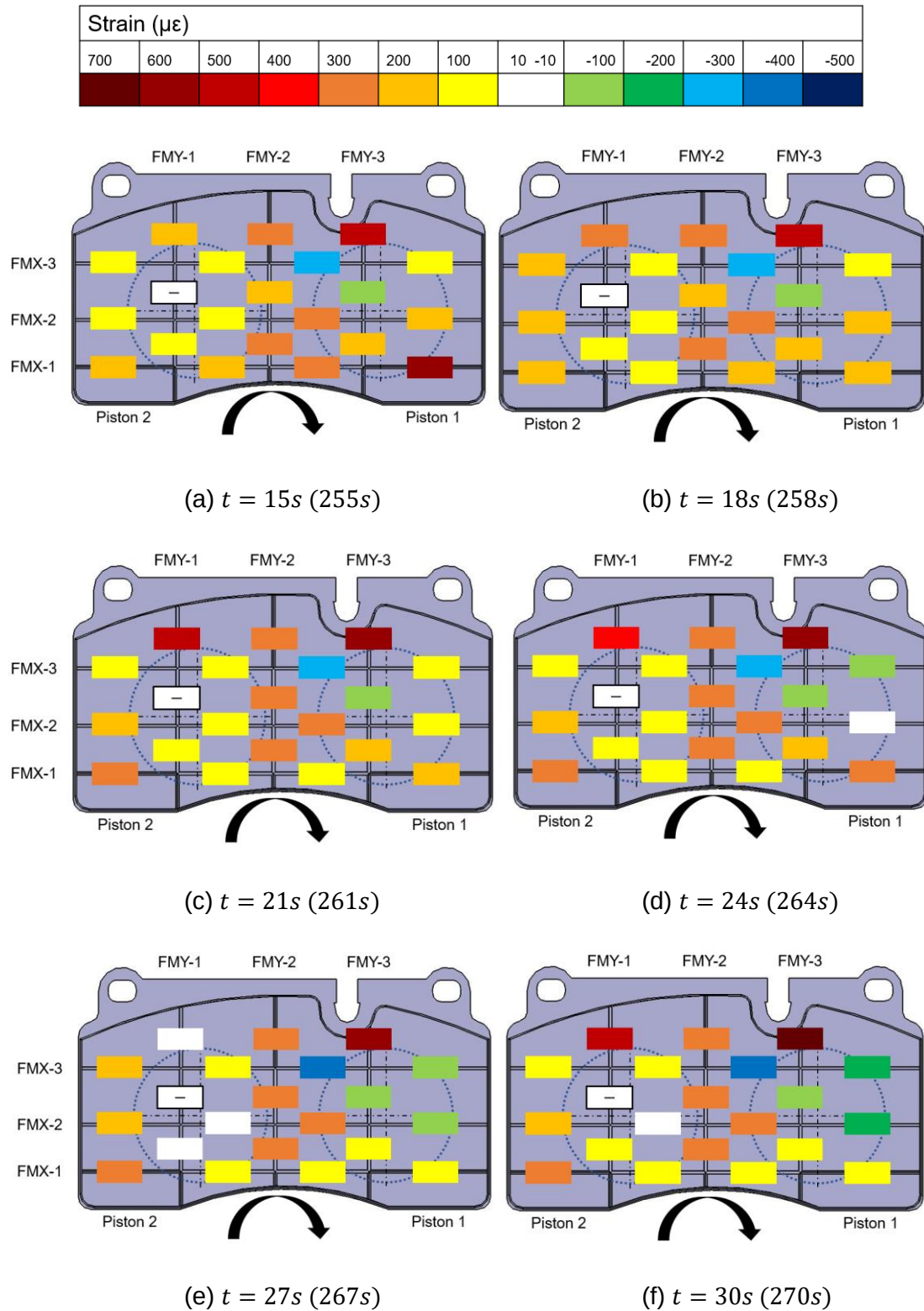


Figure 8-8: Dynamic Loading Test 1 Process from 255s to 270s

Looking further into the brake application, similar trends can be observed in Figure 8-8, i.e. the strain changes over $100\mu\epsilon$ happen on only several locations

over the 3 second periods, and typically the changes are between $100\mu\epsilon$ and $200\mu\epsilon$. The most prominent changes are observed at two positions, FMX1-FBG4 from 15 to 18 seconds, and FMY1-FBG1 from 24 to 27 seconds, the strain changes are around or over $300\mu\epsilon$. Comparing the first second ($t = 0s, 240s$, Figure 8-7 (a)) and the last second ($t = 30s, 270s$, Figure 8-8 (f)), most of the sensor positions, the strain changes are still between $100\mu\epsilon$ and $200\mu\epsilon$, while only two sensor positions are over $300\mu\epsilon$, FMX2-FBG4 and FMY3-FBG1 respectively.

Furthermore, strains of the outer diameter (OD) side are always higher than that of the inner diameter (ID) side, and also the leading edge of the brake pad (left areas) experiences lower strain values at the first 15 seconds than the trailing edge (right areas), then exhibits higher strain changes at the last 15 seconds. A true strain cycling displays among the positions of the middle brake pad, in particular, FBG2 of FMX1, FMX2 and FMX3, and FMY2 of FBG1, FBG2 and FBG3, these 6 positions are illustrated as the stable strain values during the total 30 seconds' application of the dynamic brake loading condition. The best method of getting the insight and 'practical feel' for the changes is to look 'dynamically' at this dynamic event. To enable this, a dynamic brake animation is created and included here ([Dynamic Brake Animation](#)).

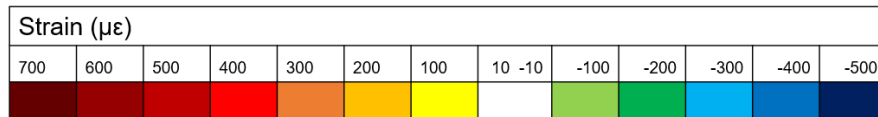
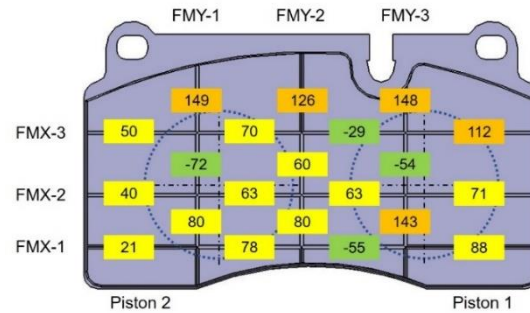
8.2.4 Strain Differences between Initial and Final Static Applications

According to the description of the test plan presented in section 8.1.2, in order to determine the strain changes before and after dynamic loading applications, the dynamic loading test procedure included two periods of static pressure application, the initial (Periods 1 to 3) and final (Periods 7 to 9) steps. Because of the stability of the static conditions, the test results displayed in the contour maps here are the average of 30s recording time. Meanwhile, the final static application results (Period 8) needed to be operated further data processing in advance. Generally, this was a simple calculation, where the average data of Period 7 was subtracted from the Period 8 data, with Period 7 acting as the benchmark for the final step, ensuring that final static strain results could be compared with initial static strain results under the same analysis conditions. Figure 8-9 shows three

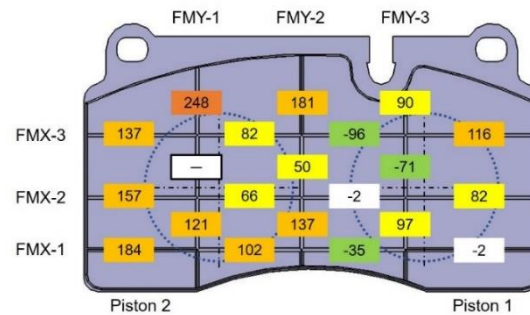
contour maps, Period 2 as the initial static pressure application (Figure 8-9 (a)), Period 8 as the final static pressure application (Figure 8-9 (b)), and strain differences between the two applications (Figure 8-9 (c)).

Figure 8-9 (c) shows the strain differences between the measurements made during the two static pressure applications during Test 1, where 90% (19/21) sensor differences were under $100\mu\epsilon$. Only 2 sensors, FMX1-FBG1 and FMX2-FBG1, were showing an increase over $100\mu\epsilon$, but still lower than $200\mu\epsilon$. The absolute average of total 21 sensor strain differences was only $52\mu\epsilon$, so that it can be concluded that final static results showed small differences compared to the initial static results. Also, considering the full image of the friction material surface, the positive strain differences were located around Piston 2, on the leading edge, which meant that the final static step experienced higher strain than the initial static step.

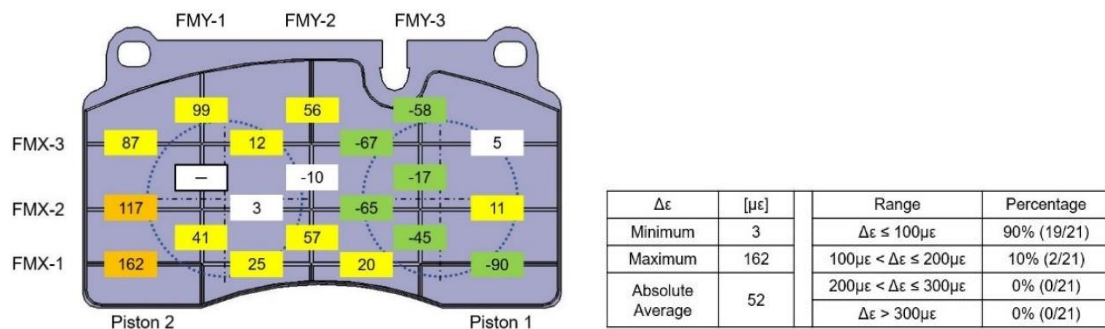
The strain differences revealed that the leading-edge experienced higher strain than that of the trailing-edge. Figure 8-9 (a) reveals that the initial static strain results of the trailing edge achieved higher strain levels. However, Figure 8-9 (b) illustrated dramatic changes after the dynamic loading application that the strain levels of the leading-edge side have increased, and that of the trailing side decreased, as Figure 8-9 (c) demonstrated. To further investigate static strain distributions before and after the dynamic test, "Dynamic Cooling Test 4" was conducted and the results will be discussed in section 8.4 to investigate in more detail the influence of temperature.



(a) Period 2: Initial Static Application ($T_0 = 22.3^\circ\text{C}$)



(b) Period 8: Final Static Application ($T_i = 55.5^\circ\text{C} - 54.0^\circ\text{C}$)



(c) Strain Differences: $\Delta_{Diff} = \epsilon_{Final} - \epsilon_{Initial}$

Figure 8-9: Strain Differences between Initial and Final Static Applications

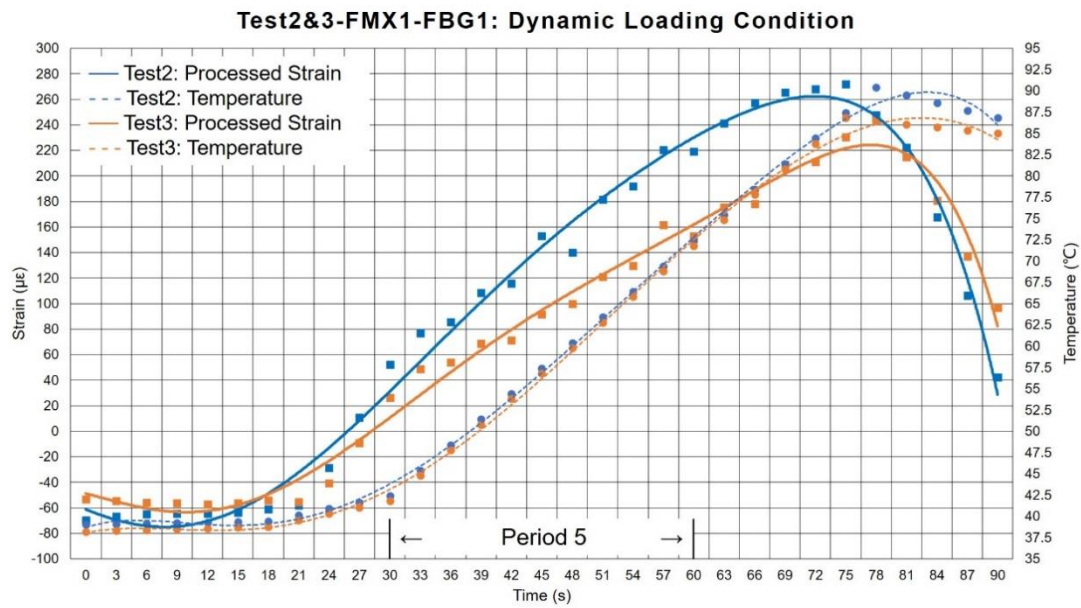
8.3 Dynamic Loading Tests 2 & 3

Two tests, “Dynamic Loading Tests 2 and 3”, were undertaken in order to observe the repeatability and sensitivity of the FBG strain sensing technology at the friction interface in dynamic conditions. Both tests involved the same operations and data processing as described for “Dynamic Loading Test 1”. Therefore, the raw data (wavelength) graphs in Appendix Chapter 8-C were broadly similar to those presented in section 8.2.1 for Test 1, hence the following sub-sections will mainly focus on processed data graphs and result comparisons with different loading conditions.

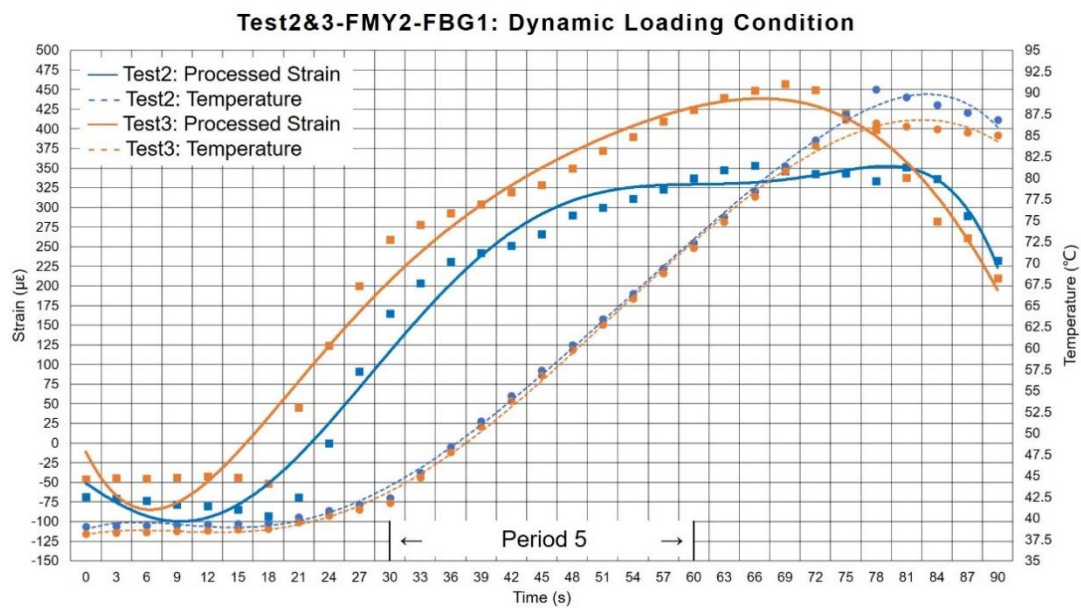
8.3.1 Processed Data Graphs

According to the explanation of the data processing provided in section 8.2.2, the data of Tests 2 and 3 was analysed using the same approach. In order to compare both test results in the same processed data graphs, establishing a new time scale (from 0 to 90 seconds) allows for a clear understanding of the strain differences between different tests. Figure 8-10 displays two FBG sensor measurements, FMX1-FBG1 (a) and FMY2-FBG1 (b). Each contains processed strain and temperature of both tests for the whole dynamic loading application.

The data obtained from the selected sensors in Figure 8-10 shows that the general trends of both tests were similar throughout all test periods. The temperature reached about the same level, while the temperature of each test was still slightly different due to the dynamic environments being always changeable. The trends of the strain measured in both directions were similar, where the maximum differences were about $100\mu\epsilon$ in the Y direction and about $60\mu\epsilon$ in the X direction. In both tests, the strains increased when the hydraulic pressure was applied, then began to decrease when the pressure decreased. The two processed data graphs exhibited approximately the same trends as observed during “Dynamic Loading Test 1” (Figure 8-5).



(a) Test2&3-FMX1-FBG1



(b) Test2&3-FMY2-FBG1

Figure 8-10: Comparisons of Dynamic Loading Tests 2 & 3 Processed Data Graphs

8.3.2 Strain Differences of Dynamic Loading Applications

In order to have a clear comparison of the two tests, the contour map method was used to compare the strain results for the last second of Period 5, the 280th

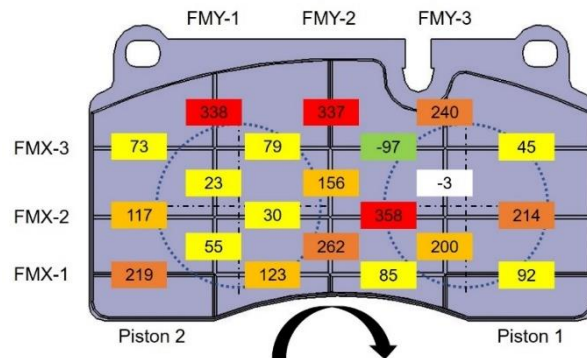
second of Test 2 (Figure 8-11 (a)), and the 255th second of Test 3 (Figure 8-11 (b)), coming with strain differences in Figure 8-11 (c).

Generally, the strain distributions measured during Tests 2 and 3 showed similar trends as displayed by the data from Test 1, shown in Figure 8-6. The strain results at the end of Period 5 of the test exhibited the highest strain levels, 358 $\mu\epsilon$ in Test 2, and 429 $\mu\epsilon$ in Test 3. The outer disc diameter (OD) area, the top side of the brake pad, still experienced higher strains than the inner diameter (ID) area. Furthermore, higher strains were observed in the central area of the brake pad than both left and right edges.

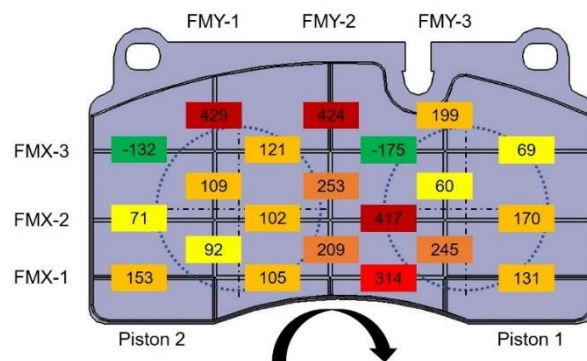
Strain differences shown in Figure 8-11 (c), conclude that 90% of the sensor results (19/21) were similar, with differences under 100 $\mu\epsilon$, 10 of which were even under 50 $\mu\epsilon$. Only 2 sensors, FMX1-FBG3 and FMX3-FBG1, exhibited differences between 200 $\mu\epsilon$ and 300 $\mu\epsilon$. Generally, the test conditions were similar and temperature differences were within $\pm 1^\circ\text{C}$, while the absolute average strain difference was 75 $\mu\epsilon$.

Considering dynamic environmental changes, such strain differences between two tests were acceptable, and it is noted that they in turn were similar to the results of Test 1. According to these three dynamic tests, it is concluded that the FBG sensors provided repeatable and sufficiently sensitive measurements when operating under dynamic conditions.

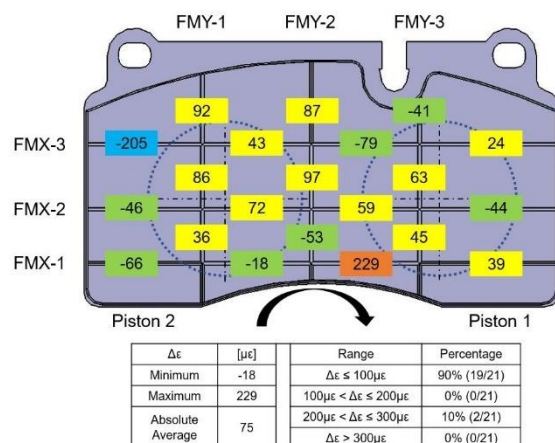
Strain ($\mu\epsilon$)													
700	600	500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



(a) Test 2: $T_0 = 39.0^\circ\text{C}$; $T_i = 72.4^\circ\text{C}$; $t = 280\text{s}$



(b) Test 3: $T_0 = 38.2^\circ\text{C}$; $T_i = 71.8^\circ\text{C}$; $t = 255\text{s}$



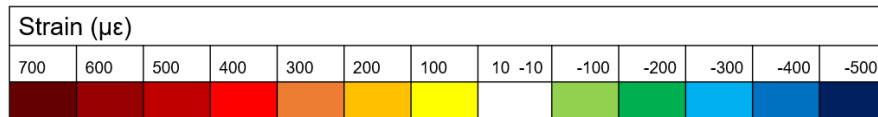
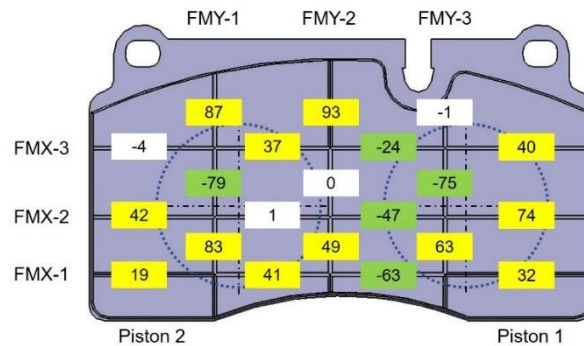
(c) Strain Differences $\Delta_{Diff} = \epsilon_{Test3} - \epsilon_{Test2}$

Figure 8-11: Tests 2 & 3 Strain Differences of Dynamic Loading Applications

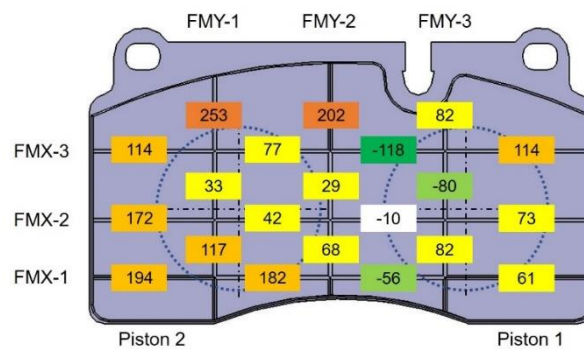
8.3.3 Strain Differences between Initial and Final Static Applications

Following the comparison made in section 8.2.4 for Test 1, this section will also compare strain differences observed in the measurements made during the initial and final static pressure applications for Tests 2 and 3. Meanwhile, additional comparisons were applied between Tests 2 and 3 of the initial (Period 2) or final (Period 8) static pressure application, in order to provide a full analysis, not only between Periods 2 and 8 for the same test, but also between Tests 2 and 3 for the same period. Therefore, a longitudinal analysis was used for the strain differences between the initial and final static pressure application of Test 2 (Figure 8-12). Figure 8-13 focuses on the result from Test 3, while Figure 8-14 shows a further comparison of the strain differences between dynamic loading Tests 2 and 3.

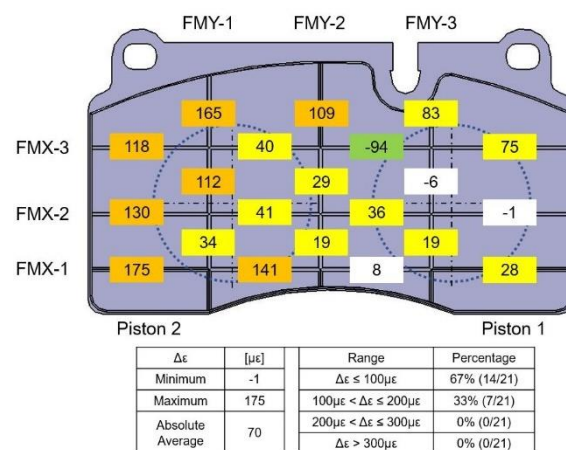
The strain differences in Figure 8-12 and Figure 8-13 show the same trends, with 67% of the sensor strain differences (14/21) being under $100\mu\epsilon$. The differences in the measurements from the other 7 sensors in Test 2 lay between $100\mu\epsilon$ and $200\mu\epsilon$, while Test 3 had 6 locations among such range value, and another sensor, which was FMX1-FBG2, was $292\mu\epsilon$. By observing the full image of the strain distributions, it is noted that both tests showed the same trends as Test 1 (Figure 8-9 (c)), where the leading-edge side (Piston 2) experienced higher strain levels than the trailing-edge side (Piston 1), reflecting the brake performance of the final static pressure application. Although such trends were not as significant as Test 1 demonstrated, the trends of Tests 2 and 3 were still clear enough and allowed the same conclusion to be reached.



(a) Period 2: Initial Static Application ($T_0 = 39.0^\circ\text{C}$)



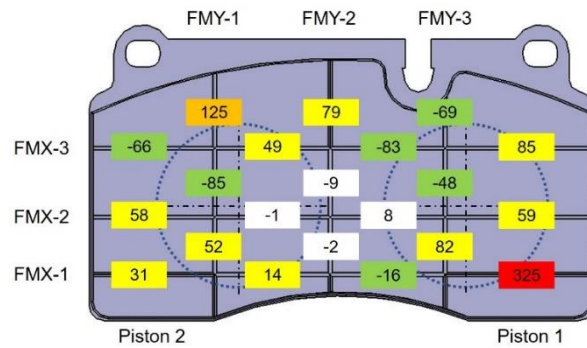
(b) Period 8: Final Static Application ($T_i = 78.9^\circ\text{C} - 77.4^\circ\text{C}$)



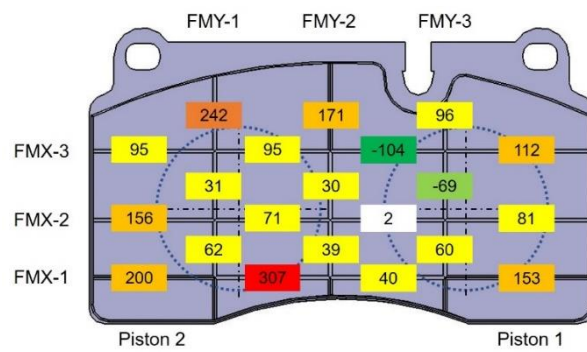
(c) Strain Differences: $\Delta_{Diff} = \epsilon_{Final} - \epsilon_{Initial}$

Figure 8-12: Test 2 Strain Differences between Initial and Final Static Applications

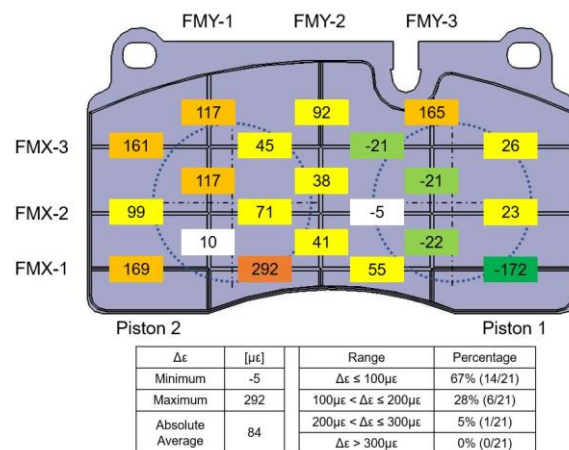
Strain ($\mu\epsilon$)												
700	600	500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



(a) Period 2: Initial Static Application ($T_0 = 38.2^\circ\text{C}$)



(b) Period 8: Final Static Application ($T_i = 78.7^\circ\text{C} - 77.2^\circ\text{C}$)



(c) Strain Differences: $\Delta_{Diff} = \epsilon_{Final} - \epsilon_{Initial}$

Figure 8-13: Test 3 Strain Differences between Initial and Final Static Applications

The strain difference comparison between the two tests (Figure 8-12 (c) and Figure 8-13 (c)) show strain differences that have a similar pattern to that shown in Figure 8-14, where 90% of the strain differences were under 100 $\mu\epsilon$, one sensor (FMX1-FBG2) difference was over 100 $\mu\epsilon$ at 151 $\mu\epsilon$, and FMX1-FBG4 was 201 $\mu\epsilon$, which was the only sensor detecting a strain difference of over 200 $\mu\epsilon$. The absolute average of strain differences was only 46 $\mu\epsilon$, suggesting that the comparison between initial and final static pressure applications of both tests displayed a high sensitivity under changing conditions.

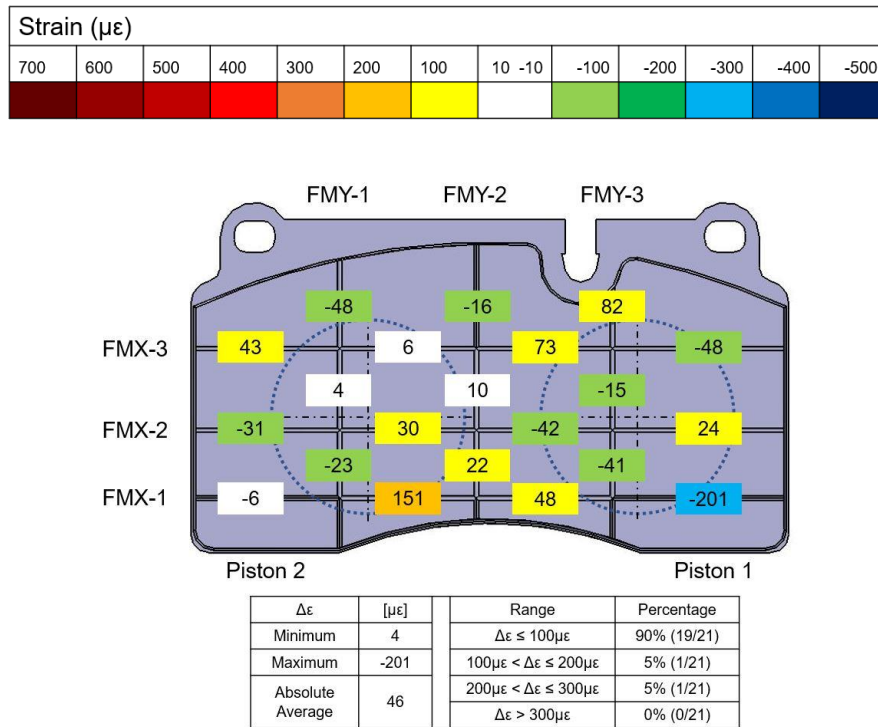
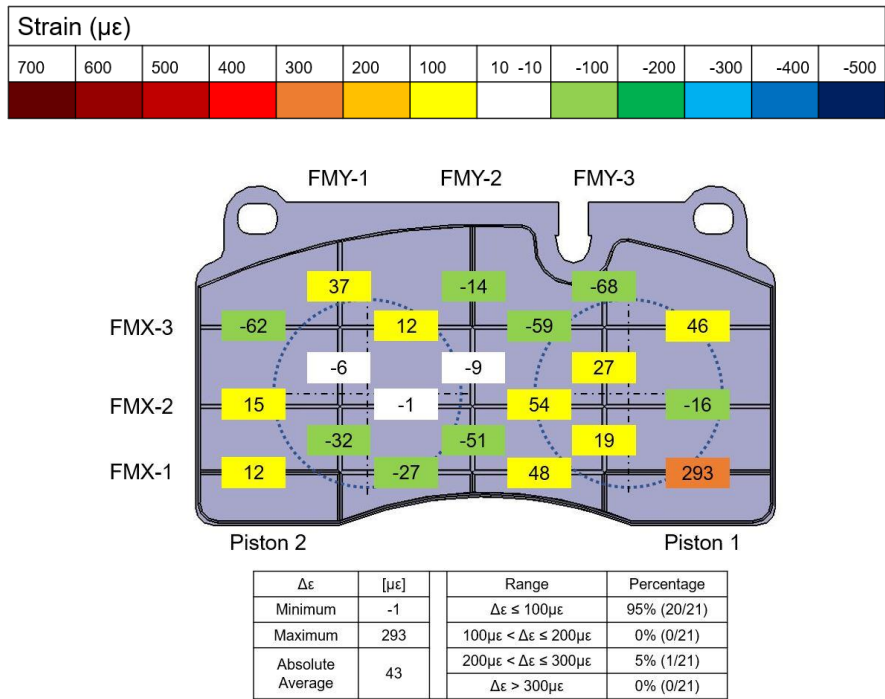


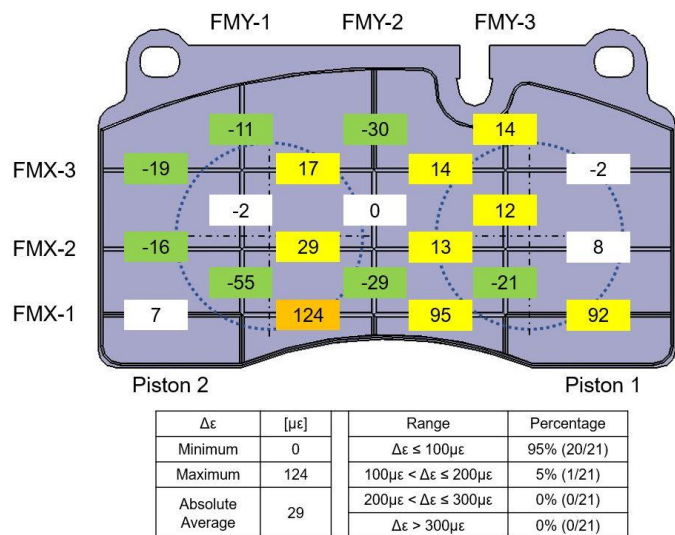
Figure 8-14: Static Strain Differences: $\Delta_{Diff} = \Delta_{Diff_{Test3}} - \Delta_{Diff_{Test2}}$

Furthermore, the crosswise analysis between the two dynamic tests during application Period 2 or Period 8 produces the static strain differences presented in Figure 8-15. Both comparisons indicate that 95% of the sensor strain differences are under 100 $\mu\epsilon$. Only sensor FMX1-FBG in Figure 8-15 (a) had a strain difference of over 200 $\mu\epsilon$, while FMX1-FBG2 in Figure 8-15 (b) exhibited a 124 $\mu\epsilon$ difference. Most of the strain differences in these two comparisons were even lower than 50 $\mu\epsilon$. In particular, FMX2-FBG2 in the initial static pressure application, and FMY2-FBG2 in the final static pressure application, showed -1 $\mu\epsilon$

and $0\mu\epsilon$ differences. These strain differences reflect the repeatability of the measurements between the two similar dynamic tests, and show that some sensor locations experienced the same strain level, which shows the high stability of FBG sensing technology applied to the complex dynamic test environments.



(a) Initial Strain Differences: $\Delta_{Diff} = \epsilon_{Test3} - \epsilon_{Test2}$



(b) Final Strain Differences: $\Delta_{Diff} = \epsilon_{Test3} - \epsilon_{Test2}$

Figure 8-15: Test 2 & 3 Strain Differences between Initial or Final Static Applications

In general, three dynamic loading tests have been completed under similar test conditions, in which most of the sensors operated correctly and recorded data well, and the measurements displayed repeatability and sensitivity. The strain trends during the dynamic loading application showed an increased strain when applying the hydraulic pressure to the rotating brake disc, and its decrease when releasing the pressure. Meanwhile, higher strains were recorded at the outer diameter than the inner diameter during dynamic loading periods.

Furthermore, two repeated static pressure applications were included before and after the dynamic loading step in order to discover strain changes under the same loading conditions in the same test. The leading-edge side always experiences larger strain differences than the trailing-edge side does. Again, the crosswise analysis and comparison illustrate the repeatability of the FBG strain measurements, ensuring that the test results, the raw recorded data, can be gathered and analysed reliably, after which several processing approaches may be adopted to interpret the meaning of the data and hence understand the underlying physical effects.

8.4 Dynamic Cooling Test 4

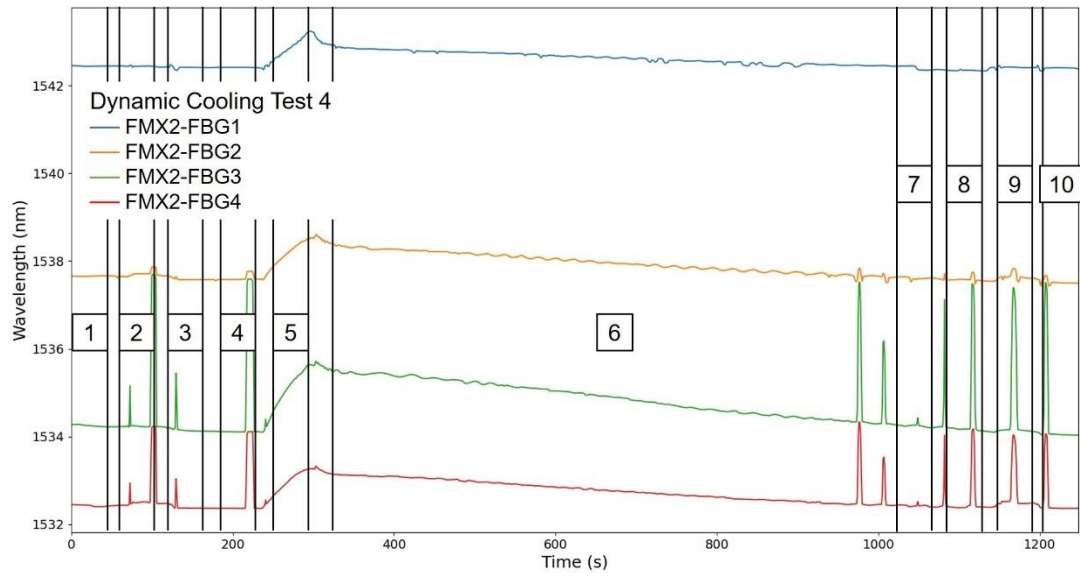
According to test results displayed in sections 8.2 and 8.3, the temperature of final static pressure application step did not return to its initial level. Although strains were not influenced as much as they were in sections 8.2.4 and 8.3.3, the idea of the cooling condition is still considered in order to detect the strain performance when both initial and final static pressure applications were imposed at the same temperature. In such a test, strain (wavelength shift) compensation for temperature will not be needed and a direct strain comparison can be made. Therefore, “Dynamic Cooling Test 4” includes the same test step as the previous dynamic loading tests, inserting an extra cooling period before the final static pressure application step, during which the brake disc continues to rotate at 50rpm to reduce the brake disc’s temperature until its value returns to that during the initial static pressure application.

8.4.1 Raw Recorded Data Graphs

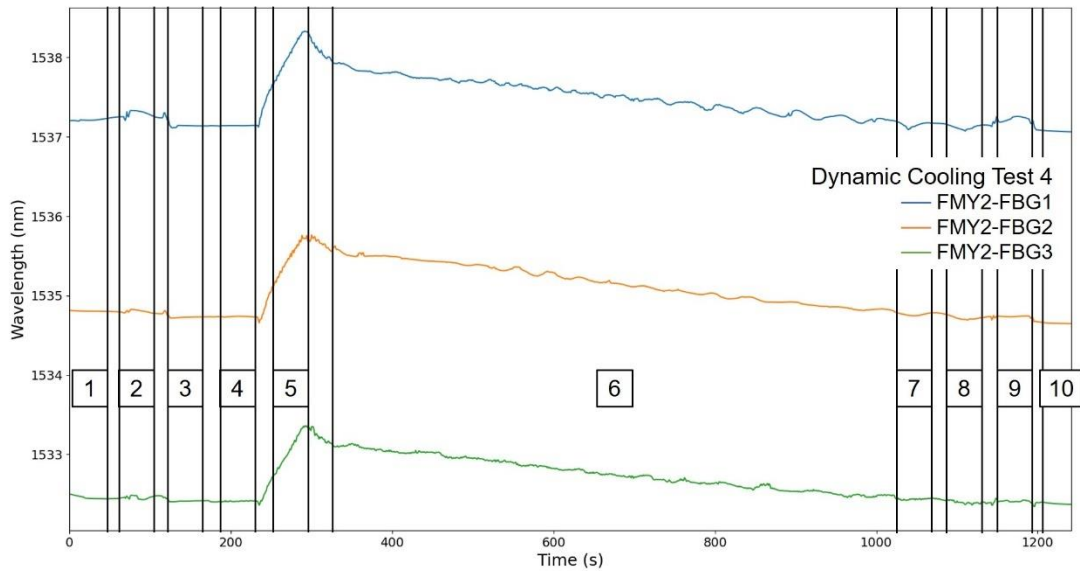
Two samples of raw recorded data (wavelength) graphs are shown in Figure 8-16, for sensors oriented in the X-direction (FMX2) and Y-direction (FMY2). Each graph is labelled to allow the identification of each measurement period, with a general comment on the test conditions, including hydraulic pressure level (p) and disc rotational speed (n), as detailed in section 8.2.1 and Appendix Chapter 8-C.

Period 6 in Figure 8-16 was the cooling step applied between the dynamic loading application (Period 5) and the final static pressure application (Periods 8 to 10). The strain levels increased during the dynamic loading application, as observed in Tests 1, 2 & 3. Then a clear decrease can be observed due to the pressure release, and after that, the wavelength values experienced a gentle decline trend during the cooling period. Meanwhile, the general detection of the initial static pressure application (Period 2), and the final static pressure application (Period 9) showed that wavelength levels were of similar values when the temperature level was about the same. The following section will then provide a comparison to display a clear view of the interface strain distributions on the friction material surface.

Step	Initial Static Application			Dynamic Loading and Cooling Application					Final Static Application	
Period	1	2	3	4	5	6	7	8	9	10
p (bar)	0	30	0	0	30	0	0	0	30	0
n (rpm)	0	0	0	0 - 50	50	50	50-0	0	0	0



(a) Dynamic Cooling Test 4: FMX2



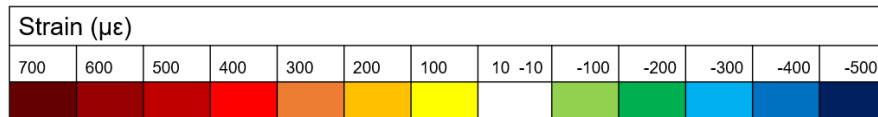
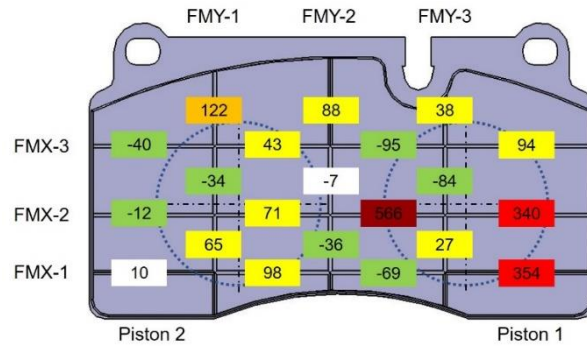
(b) Dynamic Cooling Test 4: FMY2

Figure 8-16: "Dynamic Cooling Test 4" Raw Recorded Data Graphs

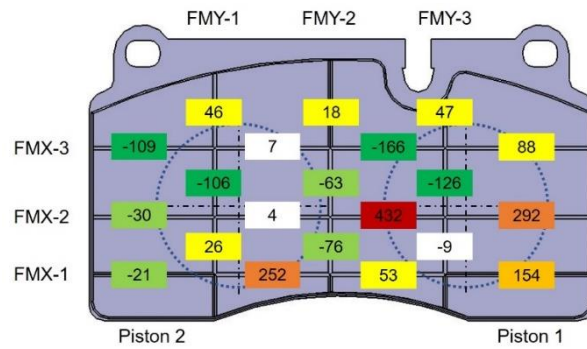
8.4.2 Strain Differences between Initial and Final Static Applications

According to raw recorded data (wavelength) graphs, both initial and final static pressure applications resulted in similar strain values under the same test conditions. Further detailed comparisons allow the strain differences between the measurements recorded during Periods 2 and 9 in Figure 8-17, which contains the processed static pressure application results, and the strain differences, with a general summary to illustrate the dynamic cooling effect.

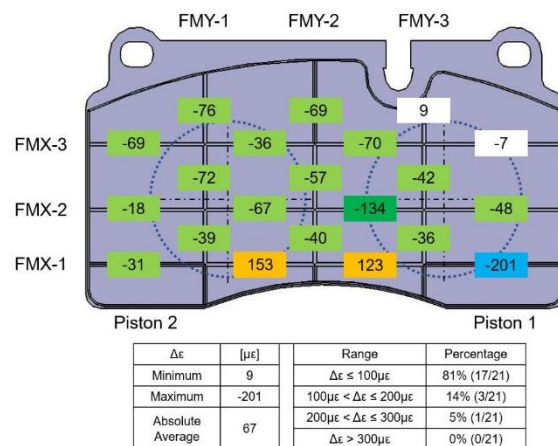
The temperature difference between these two periods was only 0.3°C, which was about the same in terms of the wavelength shift as that from the loading conditions to be compared. Figure 8-17 (c) showed that almost all FBG sensors (17/21) have strain differences lower than 100µε, and these strain differences all reflected that Period 9 obtained lower strains than that of Period 2. Only 2 FBG sensors, FMX1-FBG2 and FMX1-FBG3, displayed the positive trends. The absolute average of the strain differences was only 67µε, which allows it to be concluded that both periods illustrated similar strains when the brake disc cooled to the initial temperature. The maximum difference appeared at FMX1-FBG4, -201µε. Such a strain change might be influenced by various parameters, and further discussion will be presented in section 8.6 when analysing the brake pad surface condition before and after tests.



(a) Period 2: Initial Static Application ($T_0 = 38.2^\circ\text{C}$)



(b) Period 8: Final Static Application ($T_i = 38.5^\circ\text{C}$)



(c) Strain Differences: $\Delta_{Diff} = \epsilon_{Final} - \epsilon_{Initial}$

Figure 8-17: Strain Differences between Initial and Final Static Applications

In conclusion, “Dynamic Cooling Test 4” produced two repeated measurements under the same temperature conditions with the aim of investigating strain changes between Periods 2 and 9. The results are similar and Period 9 shows lower strain trends than that shown in Period 2. Also, such a long cooling time is unnecessary to be applied as these FBG strain sensors were not affected by the temperature change, and the analysis referenced the static strain to the value immediately before the pressure was applied, so it can ignore the effect of the temperature on the optical fibres. Therefore, “Dynamic Loading Tests 1, 2 & 3” were directly moved to the next step as the test plan validated, and there was no need to wait until the temperature was back to the initial level. The cooling test measurement provides persuasive evidence that the general dynamic test plan described in section 8.1.2 is a reasonable and appropriate means to achieve the chapter’s aim of measuring strain changes during various dynamic conditions.

8.5 Dynamic Consecutive Test 5

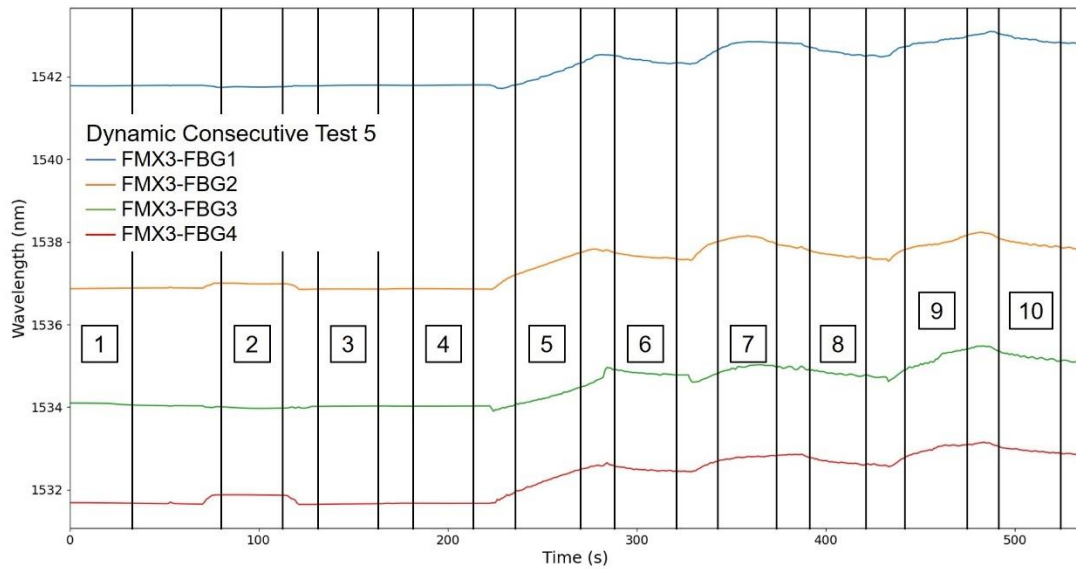
In addition to dynamic loading and cooling tests discussed in the previous sections, the additional consideration has been raised that brakes are usually applied with repeated applications under real driving conditions due to complicated traffic environments. “Dynamic Consecutive Test 5” simulates such loading conditions, where three repeated brake applications were conducted, each in 30 seconds duration (actuation), spaced by 30-second released brake.

8.5.1 Raw Recorded Data Graphs and Temperature Measurement

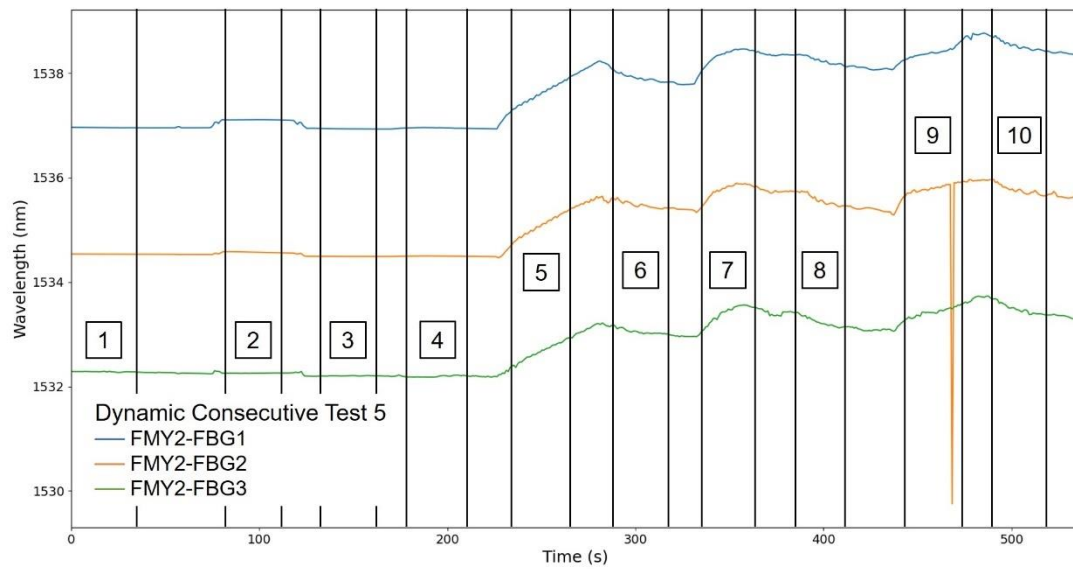
Raw recorded data (wavelength) graphs from FBG sensors in FMX3 and FMY2 are shown in Figure 8-18, with the specific test condition indicated along the time series. Figure 8-19 illustrates the recorded temperature via the rubbing thermocouple during the test time.

The test started with the initial static pressure application (Periods 1 to 3) to ensure that all sensors work well, before taking the further steps. The responses to three repeated dynamic loading applications can be observed in Periods 5, 7 and 9, and the subsequent pressure releasing step was illustrated in Periods 6, 8 and 10.

Step	Initial Static Application			Repeated Dynamic Loading Application				Final Static Application		
Period	1	2	3	4	5	6	7	8	9	10
p (bar)	0	30	0	0	30	0	30	0	30	0
n (rpm)	0	0	0	0 - 50	50	50	50	50	50	50



(a) Dynamic Consecutive Test 5: FMX3



(b) Dynamic Consecutive Test 5: FMY2

Figure 8-18: Dynamic Consecutive Test 5 Raw Data Graphs

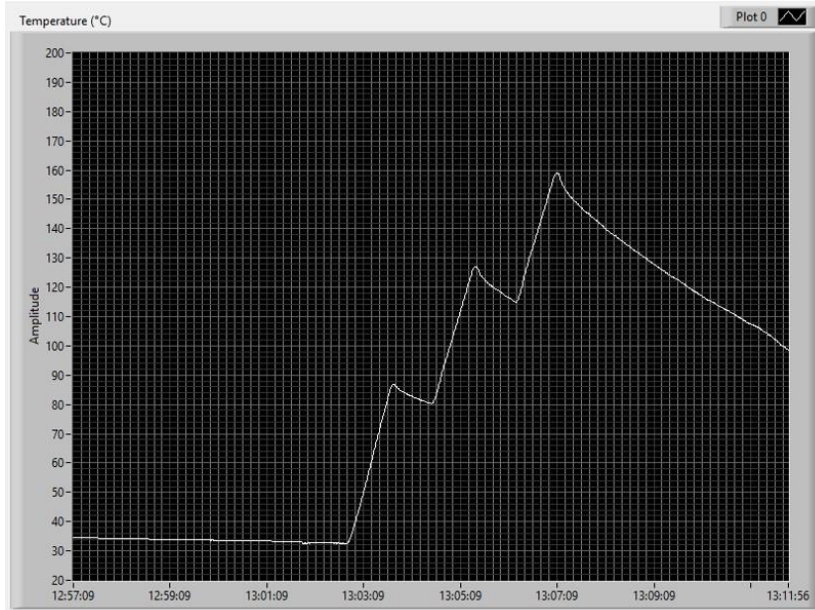
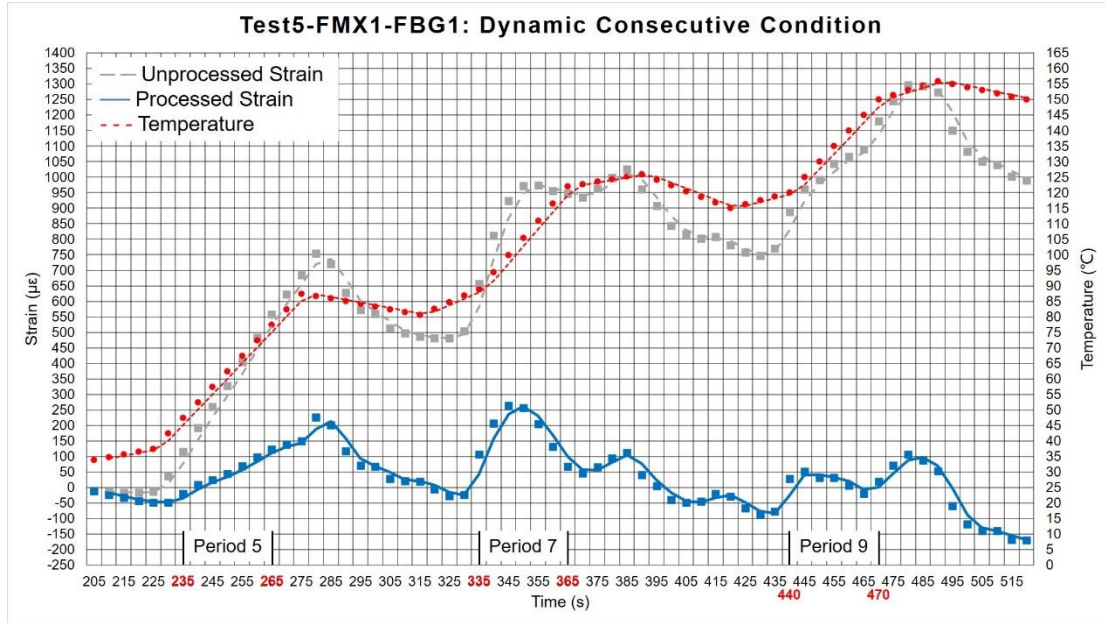


Figure 8-19: Dynamic Consecutive Test 5 Temperature Measurement

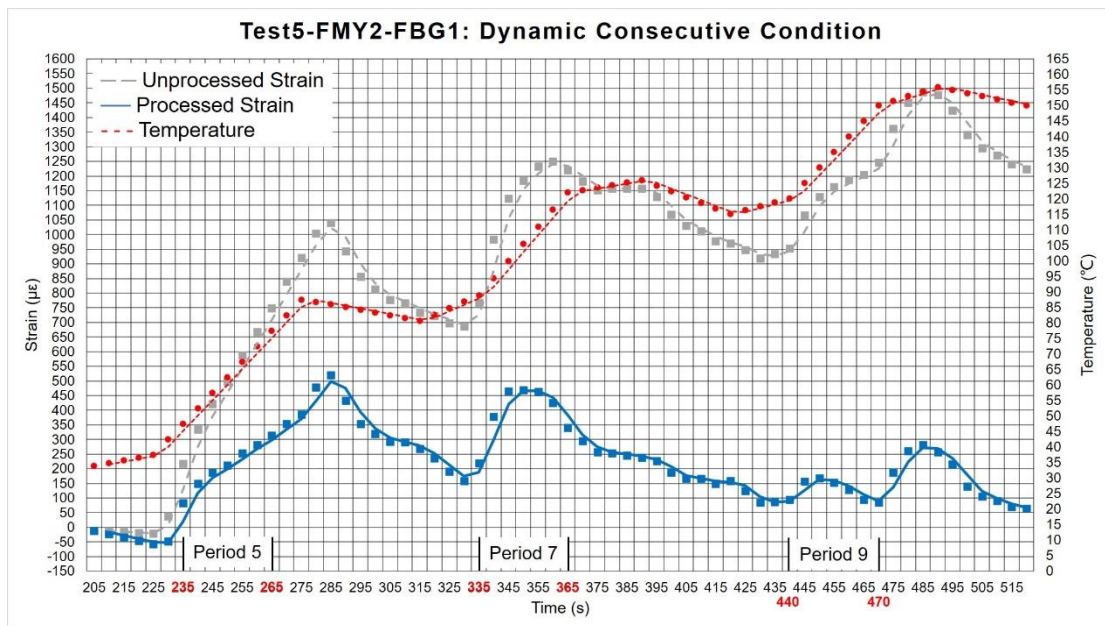
In Figure 8-19, it can be seen that the temperature of the brake disc responded to the different dynamic loading conditions. The temperature on the first brake application increased from 32°C to 88°C, followed by a slow cooling to 80°C. During the second brake application, it increased from 80°C to 128°C, followed by a slow cooling to 114°C, and then the third application from 114°C to 160°C. These temperature values will be used to compensate for the influence of temperature during data processing. The general relationship between temperature and test periods has been calculated, where the increasing trend was about 1°C/s, the decreasing trend with 50 rpm disc rotation rate was about 0.2°C/s, while the declining trend with a static disc was about 0.05°C/s. These relationships have been applied for all dynamic tests throughout this chapter.

8.5.2 Processed Data Graphs and Repeated Dynamic Results

Processed data graphs of Test 5 are displayed in Figure 8-20, containing the strain measured by two FBG sensors, FMX1-FBG1 and FMY2-FBG1 during three repeated pressure applications, from 205s to 520s. Each 5 seconds serves as a reference point to provide enough data to be displayed. Meanwhile, the time series of Periods 5, 7 and 9 was highlighted red on the timeline.



(a) Test5-FMX1-FBG1



(b) Test5-FMY2-FBG1

Figure 8-20: Dynamic Consecutive Test 5 Processed Data Graphs

Both sensors displayed the same trends, where the unprocessed strains increased with the temperature, however, processed strains reduced when temperature was increased up to around 155°C. The total strains declined after the second dynamic loading application due to Equation (8-2) that the higher the

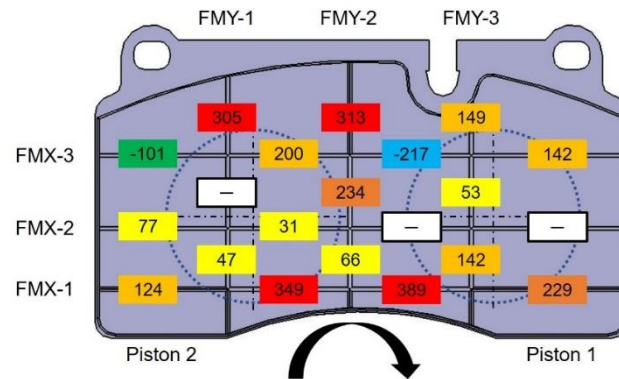
temperature was, the lower the strain obtained. Meanwhile, the maximum processed strain was around $250\mu\epsilon$ at FMX1-FBG1, and about $500\mu\epsilon$ at FMY2-FBG1, being observed between the first and second pressure applications.

However, the repeated dynamic results were not sufficient to determine the interface strain distributions as the values were always changeable, so that the contour map approach was again used to display the strains recorded at the last second of each dynamic loading application. Figure 8-21 includes the first dynamic application at 265s (a), the second dynamic application at 365s (b), and the third time dynamic application at 470s (c). Sensor locations labelled in the black dashed line show those where the strains were out of range or the signal was lost due to the brake pad surface condition changes, which will be discussed in the next section 9.6 in detail.

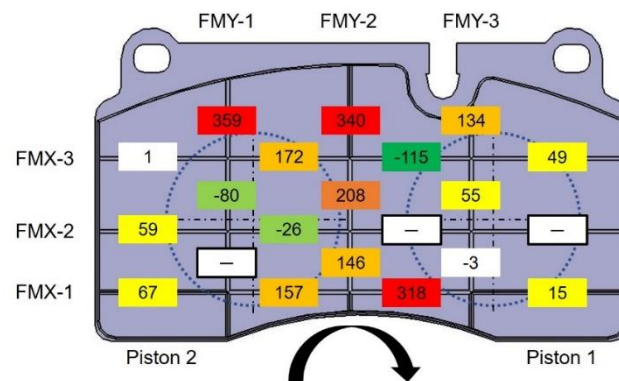
Figure 8-21 illustrates three repeated brake applications in separate images. The first application obtained similar strain trends to those of the dynamic loading tests 1, 2 and 3, where most of the FBG strain sensors (16/21) recorded properly and the strains were as expected, as the dynamic loading condition were similar to the previous tests mentioned in section 8.3. The results of the second application showed strains analogous to those measured during the first application, where the locations of the higher strains were similar.

Finally, the third application displayed the most significant changes, where half of the sensor locations (10/21) could not produce reliable results, and the other half displayed lower strain levels than those of the initial level. Although the optical fibre itself could operate at the temperatures up to 1000°C , it is likely that the cyanoacrylate adhesive (Permabond Adhesive 920) began to melt when the temperature reached 160°C (MatWeb, 2022), so that the strain transfer was not stable enough to provide dynamic loading measurements. A further analysis will be provided in section 8.6, which illustrates the brake pad performance to explain the non-ideal strain results of the third dynamic application.

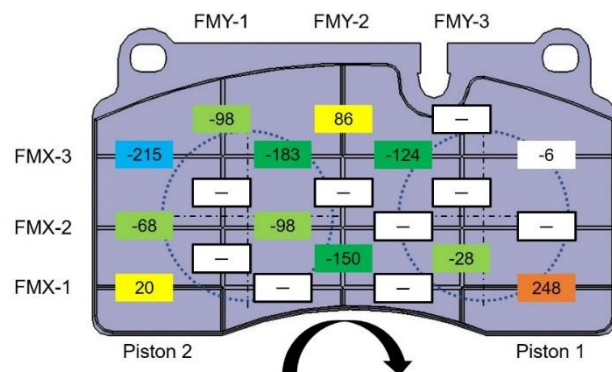
Strain ($\mu\epsilon$)													
700	600	500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



(a) 1st Dynamic Application: $T_i = 77.5^\circ\text{C}$; $t = 265\text{s}$



(b) 2nd Dynamic Application: $T_i = 122.0^\circ\text{C}$; $t = 365\text{s}$



(c) 3rd Dynamic Application: $T_i = 150.0^\circ\text{C}$; $t = 470\text{s}$

Figure 8-21: Repeated Dynamic Loading Application Strain Results

In conclusion, “Dynamic Consecutive Test 5” comprised three repeated brake applications to simulate the real driving conditions. FBG sensors worked well in most tests, however, at the highest temperature level (the third brake application), the adhesive started to weaken and melt, hence the strains could not be reliably measured. This resulted in the optical fibres shifting or falling out from the grooves, even being destroyed by rubbing at the pad/disc contact surface or ruining when touching the rotating brake disc. Meanwhile, these measurements may inspire future research to explore potential solutions to reduce these uncertain influences. The following section will provide analysis and discussion of the actual brake pad surface differences to explain the phenomena observed during Test 5.

8.6 Brake Pad Surface Condition Analysis

Following the five different dynamic tests, it is informative to observe the brake pad surface condition changes. This section analyses the brake pad friction surface and provides some possible factors to discuss the observed changes, while further potential solutions are provided to avoid and improve the brake pad wear condition. Therefore, Figure 8-22 illustrates the brake pad friction material conditions before and after several dynamic tests.

The images of the brake pad in Figure 8-22 (a) were recorded immediately after bonding the optical fibres onto the grooves by using the cyanoacrylate adhesive (Permabond Adhesive 920) with some adhesive marks in white on the frictional surface. The frictional surface had been polished following the machining of the grooves, prior to sensor installation. Some areas of the frictional surface can be seen to contain small voids arising from the loss of friction material during the manufacturing or the surface modification process. However, such issues were kept the same when taking all dynamic loading tests, so that test conditions were considered fair during all dynamic strain measurements.



(a) FM Surface before Dynamic Tests



(b) FM Surface after Dynamic Tests

Figure 8-22: The Brake Pad Wear Condition Differences

Then, the pad surface condition after several dynamic tests is shown in Figure 8-22 (b). Signs of friction surface wear can be easily observed, from applications of the pad on the rotating brake disc. Some friction material has been displaced into the machined grooves, and the friction material dust adhered to the protection jacket of the optical fibres. The adhesive marks disappeared due to the high-temperature levels generated by the friction that caused the cyanoacrylate adhesive to soften (around 70°C) and melt (around 160°C) (MatWeb, 2022).

Meanwhile, the flaked friction material pieces from the frictional surface have been wedged between grooves. This can impose a transverse load onto the optical fibres, causing micro bends, which would induce a reduction in the optical power reflected from the sensors, or could impose a transverse load on the optical fibre when the brake is activated, causing birefringence and a broadening or splitting of the peaks, which could compromise the measurements. In order to analyse potential factors of such a phenomenon, Figure 8-23 shows three close-up views of the pad friction surface.

Figure 8-23 (a) shows regions where the friction materials have been displaced into grooves. The edges of the grooves have been damaged during several dynamic runs. In particular, the intersections of the X and Y direction grooves exhibit significant damage and lose the bonded adhesive. The area in the red oval in the right-side view (Figure 8-23 (b)) has then been expanded in Figure 8-23 (c) to provide a detailed view of the optical fibre condition. The fibres oriented in both the X and Y directions have popped out of the grooves due to the loss of adhesive, which influenced the recorded data significantly during dynamic loading applications, for example, three FBG sensors in Tests 1 and 5, FMY1-FBG2, FMX2-FBG3 and FMX2-FBG4. During Test 1 of the normal dynamic loading condition, only FMY1-FBG2 lost the data result, while these three sensors were all out of range when taking Test 5 of three repeated brake applications.



(a) Left Side View



(b) Right Side View



(c) Detailed Side View

Figure 8-23: Different Views of The Brake Pad Wear Condition after Dynamic Tests

There are several factors which can explain the brake pad surface condition changes. The cyanoacrylate adhesive, Permabond Adhesive 920, applied in this research project has a softening point between 50°C and 150°C, and will start melting between 150°C and 250°C (Permabond Engineering Adhesives, 2022). “Dynamic Consecutive Test 5” involved three repeated brake applications, where the maximum disc temperature has reached around 160°C after the third application. It would thus be expected that the adhesive would begin to melt and exhaust from the rotating brake disc. Therefore, the strain transfer to the optical fibres was reduced, affecting the measurements as illustrated in Figure 8-21 (c), where 43% (9/21) of the FBG strain sensors did not record properly and even lost signals during the test time.

The modification of the frictional surface, in particular the X and Y direction grooves, was another factor. Although these grooves were used for installing optical fibres with FBG strain sensors, such modifications clearly involved the removal of material, and it would be expected that the edges of the grooves would be susceptible to damage. Although the grooves were filled with adhesive when bonding optical fibres, with the density of the adhesive being similar to that of the friction material, the adhesive became soft and melted after several dynamic tests, so that the frictional surface might be ruined and the fallen material pieces might be stuck between groove gaps, as Figure 8-23 (a) displayed.

Last but not least, the friction material characteristics, the material mechanical and thermal properties and the friction coefficient are variable within the same pad (mixture). The friction material composes of no fewer than 10 to 20 types of raw materials, mainly containing binders (phenol resin), reinforcements (aramid fibre and steel fibre), as well as friction modifiers (graphite and cashew particle) (Akebono Brake Industry, 2022). Therefore, strictly speaking, the friction material is not homogeneous and some material pieces (constituents) might be pulled out during the modification process, such phenomenon also happened when conducting dynamic tests. Furthermore, friction material compressibility reduces when the temperature rises. Such a property will then make the material squash so to affect the strain results, as displayed in Figure 8-21 (a) and (b), where some

FBG sensors stopped working. Also, the friction coefficient was affected by different dynamic test conditions. Although all tests have been kept in environments as similar as possible.

8.7 Summary

This chapter has investigated and measured interface strain distributions and the brake performances at the brake pad frictional surface under dynamic loading conditions, which was the other part of the research aims. FBG strain sensors were shown to be suitable for the brake interface strain measurements under dynamic test conditions, where most of sensors worked properly to monitor and record the strain changes under various dynamic braking conditions.

According to the previous sections that described the dynamic tests, the full view of the dynamic loading application has been explained from the test preparation, different types of test conditions, as well as the pad wear condition analysis. Several conclusions are hence listed below:

1. The FBG sensing technique can be used and functioned well under different dynamic loading conditions.
2. The data recorded during “Dynamic Loading Tests 1, 2 & 3” illustrated the high repeatability and sensitivity of the FBG strain sensors.
3. Accurate temperature measurement is vital in processing raw recorded data, i.e. accounting from wavelength shift.
4. Dynamic Cooling Test 4 proved that the strain level in static conditions, before and after a brake application, recovers to the initial level when the brake cools down.
5. Dynamic Consecutive Test 5 investigated the repeated brake application in a short time period, where most of the sensors were still active and were capable of recording the strains.
6. Further research elaborated in section 9.6 explained the factors that can reduce the influence of the optical fibres’ stability when deployed in challenging operation environments, including the stability of the cyanoacrylate adhesive, and the influence of the flaked friction material pieces wedged between grooves.

The possibility of directly measuring and monitoring a truly dynamic change of strain distribution at the entire pad frictional surface during braking applications (at this stage of research) laid foundations for dynamic monitoring of pressure and temperature distributions at the brake pad frictional interface. This achievement is of utmost importance, giving unique insight into the actual thermoelastic instability process and its influence on brake performance, NVH and life aspects. Such information will be a very powerful brake development tool, enabling brake optimisation at a much higher level.

Furthermore, FBGs were used successfully to perform measurements under 30 bar hydraulic pressure with 50 rpm rotational speed. Other hydraulic pressure levels (lower and higher), different pressure variations, as well as higher rotational speed, could be investigated using the technique presented. The main limiting factor encountered during the experiments conducted was the adhesive used. Thus, in order to account for higher temperatures in heavier duty brake applications, a more temperature resistant adhesive must be used.

9 DISCUSSION, CONCLUSION AND FUTURE WORKS

9.1 Discussion

An overall discussion is provided in this section, covering the literature review, experimental facilities, the results of the FEA simulation and the experimental data obtained from the three test phases, static loading, quasi-dynamic and dynamic conditions.

The literature review, Chapter 2, provided a brief overview of automotive brake system issues. The brake noise and vibration fields, and the interface pressure distributions (IPD) are found to be two main aspects of current brake researches' interests and to indicate the gap in these research works. Meanwhile, the review of the optical sensing technology applied in various industrial areas, including aerospace, manufacturing and structural health monitoring, as well as several attempts in the automotive field, which pointed to a measurement approach, optical fibre Bragg gratings (FBGs), to be adopted to offer new insights into these braking problems. The FBG measurement technology has been applied in a range of scientific and industrial fields, while limited, preliminary, work in the area of automotive brake has been conducted. Therefore, this research project applied the FBG sensing technique to determine interface strains and their distributions.

The experimental facilities, Chapter 3, described the designed and constructed test rig, the brake components' modifications, and the FBG sensors' fabrication and installation, which have been applied in both static loading and quasi-dynamic experiments. The test rig and the calliper modification enable different pressure levels and variations to be applied to the two piston sets, which can change the pressure distributions (and the centre of pressure) and thus influence the strain distributions across the brake pad. The brake pads underwent a number of modifications in preparation for the FBG sensor arrays' installation, including the machining of grooves oriented in the horizontal (X) and vertical (Y) directions, in order to provide enough space for these optical fibres to be bonded as close as possible to the brake pad surface. Then, FBG sensors were

fabricated in standard telecommunication optical fibres and bonded into the grooves.

The static loading analysis, detailed in Chapters 4 & 5, focused on gaining an understanding of the response of the brake pad to different applied pressure levels and variations. The static loading experimental phase in Chapter 4 demonstrated the repeatability of the measurements and the visco-elastic behaviour of the friction material. A comparison of the responses when the instrumented pad being mounted on the inboard and outboard sides of the calliper was undertaken as well. The results show high levels of repeatability and sensitivity, and can be used to understand and explain how brake interface pressure distributions will be captured throughout local strain levels mapped across the surfaces of the pad. The data shows that the FBG sensing technology can be used to measure strain in disc brake and is easy to deploy and monitor.

In the FE analysis phase of static conditions, Chapter 5, in which the FE model was established with major material characteristics, mesh, boundary conditions (BCs) and the analysis procedures investigated in detail. The results show sufficient sensitivity to changes of the brake pad loading (hydraulic pressure). In order to compare the FEA results with the experimental data from the FBG tests, the FEA strain values were processed and plotted at the FBG sensor locations, to allow direct comparison over a range of operating conditions. Generally, it can be stated that FBG and FEA strain results are quite similar, with exception of certain areas. Some values are almost identical, in most areas, the strain differences are relatively low to medium, with higher discrepancies limited to smaller areas. Both FBG and FEA results show identical trends when the loading (hydraulic pressure) is changed, being applied either equally on all four pistons or varied between two pairs of pistons.

More specific discussions and conclusions are to be drawn in regard to the similarities and differences between FBG and FEA strain values:

- Typically, at approximately 45% of the sensor positions, the differences between measured and predicted strain are under $100\mu\epsilon$.
- Differences in E22 (Y direction) strain tend to be lower than those in E11

(X direction) strain values.

- Although the differences in measured and predicted strain increase when the hydraulic pressure applied increases, they do not increase linearly. In other words, the result discrepancy at 120bar is not 3 times higher than that at 40bar, due to the material properties and boundary conditions applied in the FE model.
- Some geometrical features, such as the wear sensor slot, seem to cause similar effects in measured and predicted strain, but in certain conditions, the strain behaviour under the pistons seem to vary between the experimental results and the FE modelling.
- Although the difference between absolute values in certain places can be considered higher, both the FBG test data and the FEA simulation results in most cases show quite similar trends.

The quasi-dynamic analysis, detailed in Chapters 6 and 7, paid attention to determining the response to different pressure and torque levels, as well as pressure variations. In the quasi-dynamic experimental phase described in Chapter 6, the brake pad strain characteristics were investigated under several test conditions with FBG sensors oriented along both the vertical and horizontal directions on the brake pad frictional surface. Again, the results indicate high measurement sensitivity, under brake functions that simulate the brake application in normal driving activities. Here, some sensors occasionally produced abnormal data under some test conditions. It is suggested that these sensors might be located in areas where the grooves support the optical fibre sensor array poorly, and also the quality of the bonding might be sufficient to provide reliable, high fidelity strain transfer.

In the FE analysis phase of quasi-dynamic conditions, Chapter 7, an updated FE model has been established to assess the validity of the experimental results obtained, with the disc model replacement, mesh, boundary conditions (BCs) and the analysis procedures being investigated in detail. The model and analysis stability were ensured to produce repeatable results obtained under all conditions, which were modelled to correspond to the test conditions. As expected, the model

results were observed to be sensitive to the loading, i.e. pressure and torque levels. A hydraulic pressure changes of ± 10 bar in one pair of pistons equated to only 8% of the baseline pressure and the torque modelled remained the same.

Strains were compared for quasi-dynamic conditions before FBG test and FEA simulation, static and quasi-dynamic strains for each method were compared side by side (FBG static with FBG quasi-dynamic, and FEA static with FEA quasi-dynamic). This is to facilitate the establishment of baseline static strain distributions and to gain insight into how quasi-dynamic loading changes these distribution trends for each of the two methods used, experimental (FBG) and modelling (FEA).

Generally, it can be stated that FBG and FEA strain results are similar. In some locations, FBG and FEA strain values are practically identical, in majority of the brake pad friction surface, for most loading cases:

- The strain differences are relatively low to medium, being under $100\mu\epsilon$ over 75%, or often over 90%, in the brake pad area.
- Torque and pressure level changes of E22 (Y direction) results: 40 bar pressure and 200 Nm torque exhibited the maximum difference is of $78\mu\epsilon$. Under 80 bar pressure and 600 Nm torque, strain differences were under $200\mu\epsilon$ for 75% of the sensor locations, with only 13% being over $200\mu\epsilon$, and another 13% over $300\mu\epsilon$. For 120 bar pressure and 1000 Nm torque, the comparison revealed the minimum difference of only $3\mu\epsilon$.
- Torque and pressure level changes of E11 (X direction) results: Strain differences between experimental and model for three loading conditions are similar to those observed for the E22 (Y direction) results.
- Pressure variations of E22 (Y direction) results: Under 80 bar pressure with 600 Nm torque applied for ± 10 bar variations between the piston sets, the results of the experimental and model are very similar. As for the +10bar variation test condition, one upmost position (towards disc OD) shows the biggest differences ($84\mu\epsilon$), with the strain differences being much smaller ($-9\mu\epsilon$ and $-40\mu\epsilon$) towards disc ID. For the -10bar variation test condition, strain differences are quite small, from OD towards ID

being only $10\mu\epsilon$, $-8\mu\epsilon$ and $24\mu\epsilon$.

- Pressure variations of E11 (X direction) results: For the +10bar variation test condition, the differences in 76% of the sensor locations were under $100\mu\epsilon$, with the remaining 24% being between $100\mu\epsilon$ and $200\mu\epsilon$. For the -10bar variation test condition, the strain differences under $100\mu\epsilon$ are in 92% of the places, with 5% being between $100\mu\epsilon$ and $200\mu\epsilon$, and only in one place (3%) being just over $200\mu\epsilon$.

Such findings from both static loading and quasi-dynamic analyses are very encouraging, suggesting that the use of FBG strain sensors can be further investigated under dynamic braking conditions.

In the dynamic analysis, Chapter 8, the interface strain distributions under 30bar hydraulic pressure with 50rpm rotational speed were investigated. The FBG sensing technology was applied and functioned well under various dynamic loading conditions, and the recorded data again showed the high repeatability and sensitivity of the FBG strain sensors. The accurate temperature measurement was vital in processing the raw recorded data (wavelength shift) to strains, to compensate the influence of temperature. Meanwhile, the cooling test proved that the strain level in static conditions, before and after a brake application, resumes its initial level when the brake cools down. Furthermore, the consecutive tests investigated the repeated brake applications over a short time period, where most of the sensors were still active and were capable of recording the strains.

Generally, FBGs were successfully used to perform measurements under 30bar hydraulic pressure with 50rpm rotational speed. Therefore, other hydraulic pressure levels (lower and higher), different pressure variations, as well as higher rotational speed can be investigated using the technique presented. However, the main limiting factor in the experiments conducted was the adhesive used. To account for higher temperatures in heavier duty brake applications, a more temperature-resistant adhesive must be used. Further research into the brake pad surface condition explained the factors that could reduce the influence of the

optical fibres' stability when operating in such a challenging operational environment.

It should be noted that relatively low-duty brake applications were deliberately selected and compared back-to-back in dynamic tests, to avoid excessive pad wear and wear debris accumulations. This enabled direct strain comparisons and helped to validate and build confidence in the measuring system. Such a system and approach can be successfully used for a wider range of brake applications, under different conditions, in terms of initial temperature and speed, as well as hydraulic actuating pressure.

9.2 Conclusion

9.2.1 Research Project Conclusions

Some final summarised conclusions can be noted as below:

- The research project demonstrated that FBG sensors could be effectively used to measure strain on the brake pad interface under various test conditions, and these measurements were relatively easy to conduct.
- The test results showed a high-level of repeatability and sensitivity, and indicated that they could be applied to determine the brake pad interface strain distributions during static loading, quasi-dynamic and dynamic test phases.
- The designed and constructed test rig was an indispensable basis of the research project and could be used as a professional system for future research. Nevertheless, the FBG methodology developed can be easily installed and used on a vehicle to conduct various performance, endurance and NVH tests.

9.2.2 Intellectual Contributions

There are two intellectual contribution aspects; scientific and industrial fields.

The scientific aspect is the creation of a reliable and effective approach to detect interface strain distributions and to undertake proper validation to provide the possibility of applying the optical fibre sensing technique in the automotive field,

in particular in brake research areas. The use of FBG sensors can provide an insight into brake interface strain distributions clearly and in depth and the designed and constructed test rig can act as the basis for further research activities. Additionally, the combination of the disc brake system with the optical technology will not only create a reliable and effective experimental approach, but also provide a fresh view into interdisciplinary research.

Furthermore, the possibility of direct measuring and monitoring a truly dynamic change of strain distribution at the entire pad frictional surface during braking applications (at this stage of research) laid foundations for dynamic monitoring of pressure and temperature distributions at the brake pad frictional interface. This achievement is of utmost importance, providing unique insight into the actual thermo-elastic instability process and its influence on brake performance, NVH and pad life expectancy aspects. Such information can be a very powerful brake development tool, enabling brake optimisation at a much higher level.

Turning to the industrial aspect, the research points to a new methodology that can offer benefits to automotive manufacturers to create the improved products. The improvement of the brake design can influence further vehicle developments, including the disc brake system and the transmission system. Furthermore, the budget is the most important factor for both manufacturers and customers. Manufacturers can reduce production costs and time if they can solve the products' weaknesses in advance, and customers can avoid extra unwanted operating issues and maintenance costs.

9.3 Future Work

Some aspects of future works are considered to be improved and tried in further research. First of all, the means for bonding the FBGs to the friction surface should be investigated to ensure reliable high fidelity strain transfer and to reduce the influence of the installation techniques. This is a challenge, as the friction material is porous, and the temperature could increase to several hundreds of degrees under dynamic operation, requiring careful selection of adhesive. Ideally, the sensors could be installed during the manufacture of the pad, such that they are embedded within the friction material.

Then, the test conditions applied during the dynamic tests were limited by the rig used, and the pressure was not indicative of the usual braking applications, therefore, other hydraulic pressure levels (lower and higher), different pressure variations, as well as higher rotational speed can be investigated using the same measurement approach to investigate further views of the interface strain distributions under different working conditions.

The temperature measurements during these three test phases, in particular the dynamic conditions, should be investigated under the same test conditions applied in this research project, using other appropriate FBG temperature compensation approaches. The potential solution is to maintain the same brake pad modifications, but to deploy tilted FBG sensors to allow both the temperature and strain distributions on the frictional surface to be captured and thus provide an enhanced understanding of the brake performance when undertaking dynamic working conditions.

Furthermore, the applications of the direct fibre optic shape sensing (DFOSS), with fibres installed around the brake pad edges. They will be used to monitor the changes in the brake pad shape under various loading conditions in order to provide a fundamental view of the actual pad deformation and create better brake designs.

Last but not least, the investigation of the use of optical fibre sensors for vibration frequency measurement would be important to the brake system. Fibre optic sensors can be used to measure vibration frequencies and perhaps characterise the vibration mode shape in order to provide a deeper understanding of brake performances for further improvements. The possible approach is to apply a modified brake pad in a test rig, which can excite the NVH issues, especially Judder and Squeal. Although the FBG sensor is suitable for this task, Fibre Segment Interferometry (FSI) can offer a sensitivity up to 2 orders of magnitude, larger than that of FBGs, and the associated instrumentation can offer a higher measurement bandwidth (up to 200 kHz). Therefore, NVH issues can be identified early at low cost and reasonable solutions or improvements realised in order to reduce undesired effects.

RESEARCH PUBLICATIONS

Submitted Conference Papers:

Wang, Z.C., Tirovic, M., James, S.W. and Tatam, R.P. (2021). Investigation of Disc Brake Interface Pressure Distributions using FBG Sensors, EB2021-STP-021, Euro Brake, 2021.

<https://www.fisita.com/library/eurobrake/2021/6216eb2021-stp-021>

Wang, Z.C., Tirovic, M., James, S.W., Barrington, J.H. and Tatam, R.P. (2022). Investigation of Pad/Disc Interface using Optical Fibre Strain Sensors, EB2022-TSD-003, Euro Brake, 2022.

<https://www.fisita.com/library/eurobrake/2022/eb2022-tsd-003>

Journal Papers to be submitted to Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering:

Wang, Z.C., Tirovic, M., James, S.W., Barrington, J.H. and Tatam, R.P. (2022). Strain Distributions at Disc Brake Interface under Static and Quasi-Dynamic Loading Conditions using Optical Sensors. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*.

Wang, Z.C., Bryant, D., James, S.W., Tirovic, M., Barrington, J.H. and Tatam, R.P. (2022). Investigations of the Interface Strain Distributions under Disc Brake Dynamic Loading Conditions. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*.

REFERENCES

Aime, L., Kissinger, T., James, S., Chehura, E., Verzeletti, A. and Tatam, R. (2021). High sensitivity pressure measurement using optical fibre sensors mounted on a composite diaphragm. *Optics Express*, 29(3), p.4105.

Akebono Brake Industry Co., Ltd. (2022). Friction Materials: Brake Pads, Brake Linings. https://www.akebono-brake.com/english/product_technology/product/automotive/friction_material/

Ashraf, N., Bryant, D., & Fieldhouse, J. D. (2017). Investigation of Stick-Slip Vibration in A Commercial Vehicle Brake Assembly. *International Journal of Acoustics and Vibrations*, 22(3), 326–333. <https://doi.org/10.20855/ijav.2017.22.3478>

Aulakh, S. K. (2013). Applications of Optical Fibers in Automobiles. *International Journal of Scientific & Engineering Research*, Vol. 4, Issue 7, July-2013, 205. <https://www.ijser.org/paper/Applications-of-Optical-Fibers-in-Automobiles.html>

Barton, D. and Haigh, M. (1998). The Second International Seminar on Automotive Braking. *Bury St. Edmunds: Professional Engineering Pub.*

Belhocine, A., & Ghazaly, N. M. (2015). Effects of Material Properties on Generation of Brake Squeal Noise using Finite Element Method. *Latin American Journal of Solids and Structures*, 12(8), 1432–1447. <https://doi.org/10.1590/1679-78251520>

Berri, P. C., Dalla Vedova, M. D. L., Maggiore, P., & Scolpito, T. (2019). Feasibility Study of FBG-Based Sensors for Prognostics in Aerospace Applications. *Journal of Physics: Conference Series*, 1249(1). <https://doi.org/10.1088/1742-6596/1249/1/012015>

Bock, W. J., Nawrocka, M. S., Urbanczyk, W., Rostkowski, J., & Martynkien, T. (2004). Fiber-Optic Sensor for Automotive Applications. *Proceedings of IEEE Sensors. 1.* 248- 251 vol.1. 10.1109/ICSENS, 1426148.

Braking Research Centre. (2022). <https://www.bradford.ac.uk/automotive-research-centre/research-groups/>

Brembo S.p.A. (2022). <https://www.brembo.com/en/>

Brembo S.p.A. (2022). <https://www.bremboparts.com/europe/en/catalogue/disc/09-8877-30>

Bryant, D. (2010). Thermo-elastic Deformation of A Vented Brake Disc. *PhD Thesis. University of Huddersfield, UK.*

Bryant, D. (2021). Brake Noise Vibration & Harshness. *Braking of Road Vehicles Short Course, 2021. University of Bradford, UK.*

Budinsky, T., Brooks, P., & Barton, D. (2018). A New Analytical Model of the Brake Pad for Improved Calculation of the Centre of Pressure and Friction

Buggy, S., James, S., Staines, S., Carroll, R., Kitson, P., Farrington, D., Drewett, L., Jaiswal, J. and Tatam, R., 2016. Railway track component condition monitoring using optical fibre Bragg grating sensors. *Measurement Science and Technology*, 27(5), p.055201.

Chehura, E., James, S. and Tatam, R. (2007). Temperature and strain discrimination using a single tilted fibre Bragg grating. *Optics Communications*, 275(2), pp.344-347.

Chen, F., Tan, C. and Quaglia, R. (2006). Disc Brake Squeal Mechanism, Analysis, Evaluation and Reduction/Prevention. Warrendale, PA: SAE International.

Choi, H.R., Jung, K., Koo, J. C., Nam, J., & Lee, Y. (2008). Chapter 25 - Micro-Annelid-Like Robot Actuated by Artificial Muscles Based on Dielectric Elastomers. *Dielectric elastomers as electromechanical transducers: fundamentals, devices, models and applications of an emerging electroactive polymer technology*. Elsevier. pp. 259-269. <https://doi.org/10.1016/B978-0-08-047488-5.00025-3>.

Coefficient in A Multi-Piston Disc Brake. *International Journal of Engineering & Technology*, Vol. 7. www.sciencepubco.com/index.php/IJET

Corning Incorporate. (2022). Corning® SMF-28® Ultra Optical Fibers. <https://www.corning.com/optical-communications/emea/en/home/products/fiber/optical-fiber-products/smf-28-ultra.html>

Dai, Y. et al. (2008). Time Division Multiplexing of FBG sensor system. SPIE

Proceedings <https://doi.org/10.1117/12.839309>.

Dai, Y. et al. (2009) A novel time-division multiplexing fiber Bragg grating sensor interrogator for Structural Health Monitoring. *Optics and Lasers in Engineering*, 47(10), pp. 1028–1033. <https://doi.org/10.1016/j.optlaseng.2009.05.012>.

Dakin, J. and Culshaw, B., (1997). *Optical fibre sensors*. Boston: Artech House.

Day, A. (2014). Brake Design Analysis. *Braking of Road Vehicles*, 97–148. Elsevier. <https://doi.org/10.1016/b978-0-12-397314-6.00005-x>

Day, A. (2014). Brake Noise and Judder. *Braking of Road Vehicles*, 343–384. Elsevier. <https://doi.org/10.1016/b978-0-12-397314-6.00010-3>

Day, A. J., Tirovic, M., & Newcomb, T. P. (1991). Thermal Effects and Pressure Distributions in Brakes. *Proceedings of the Institution of Mechanical Engineers Vol. 205 Part D: J. Automobile Engineering*, 199-205.

Day, A.J., Harding, P. R. J., & Newcomb, T. P. (1979). A Finite Element Approach to Drum Brake Analysis. *Proceedings of the Institution of Mechanical Engineers Vol. 193 (No. 37) Part D: J. Automobile Engineering*, 401-406.

De Vries, A., & Wagner, M. (1992). The Brake Judder Phenomenon. *SAE Technical Paper*, 920554.

Denimal, E., Nacivet, S., Nechak, L., & Sinou, J. J. (2017). On The Influence of Multiple Contact Conditions on Brake Squeal. *Procedia Engineering*, 199, 3260–3265. <https://doi.org/10.1016/j.proeng.2017.09.355>

FBGS. (2022). FBG Principle. <https://fbgs.com/technology/fbg-principle/>

Fieldhouse, J. D., & Steel, W. P. (2003). A Study of Brake Noise and The Influence of The Centre of Pressure at The Disc / Pad Interface, The Coefficient of Friction and Calliper Mounting Geometry. *Proceedings of the Institution of Mechanical Engineers Vol. 217 Part D: J. Automobile Engineering*, 957-973.

Fieldhouse, J. D. (2013). Brake Noise and Vibration. *Braking of Road Vehicles Course. University of Bradford, UK.*

Francavilla, A., and Zienkiewicz, O. C. (1975). A Note on Numerical Computation of Elastic Contact Problems. *International Journal for Numerical Methods in Engineering*, Vol. 9, 913-924.

Harish, A.V. et al. (2015) “Dynamic interrogator for elastic wave sensing using Fabry Perot filters based on fiber Bragg gratings,” *Ultrasonics*, 60, pp. 103–108. <https://doi.org/10.1016/j.ultras.2015.02.018>.

Ito, N., & Nakata, M. (1990). Dynamic Behaviour of The Contact Pressure Distribution Beneath A Braked Track during Turning Motion. *Journal of Terramechanics*, Vol. 27, Issue 2, 2, 79-91.

Jacobsson, H. (2003). Aspects of Disc Brake Judder. *Proceedings of the Institution of Mechanical Engineers Vol. 217 Part D: J. Automobile Engineering*, 419-430.

Jaguar Land Rover Automotive Plc. (2022). <https://www.jaguarlandrover.com/>

Jaguar Land Rover Limited. (2022). Land Rover Discovery. <https://www.landrover.co.uk/vehicles/discovery/index.html>

Jaguar Land Rover Limited. (2022). Range Rover Sport. <https://www.landrover.co.uk/vehicles/range-rover-sport/index.html>

Jaguar Land Rover Limited. (2022). Range Rover Velar. <https://www.landrover.co.uk/vehicles/range-rover-velar-2020/index.html>

James, S., Kissinger, T., Weber, S., Mullaney, K., Chehura, E., Pekmezci, H., Barrington, J., E Staines, S., Charrett, T., Lawson, N., Lone, M., Attack, R. and Tatam, R. (2022). Fibre-optic measurement of strain and shape on a helicopter rotor blade during a ground run: 1. Measurement of strain. *Smart Materials and Structures*, 31(7), p.075014.

Kashyap, R. (2010) Principles of Optical Fiber Grating Sensors, 2nd ed., Academic Press, École Polytechnique de Montréal, University of Montréal, Montréal, Canada.

Kim, S. et al. (2000) Temperature-independent strain sensor using a chirped grating partially embedded in a glass tube. *IEEE Photonics Technology Letters*, 12(6), pp. 678–680. <https://doi.org/10.1109/68.849082>.

Kim, D. J., Lee, Y. M., Park, J. S., & Seok, C. S. (2008). Thermal Stress Analysis for A Disk Brake of Railway Vehicles with Consideration of The Pressure Distribution on A Frictional Surface. *Materials Science and Engineering A*, 483-484 (2008), 456–459. <https://doi.org/10.1016/j.msea.2007.01.170>

Kinet, D., Mégret, P., Goossen, K. W., Qiu, L., Heider, D., & Caucheteur, C. (2014). Fiber Bragg Grating Sensors toward Structural Health Monitoring in Composite Materials: Challenges and Solutions. *Sensors (Switzerland)*, Vol. 14, Issue 4, 7394–7419. MDPI AG. <https://doi.org/10.3390/s140407394>

Kinkaid, N. M., O'Reilly, O. M., & Papadopoulos, P. (2003). Automotive Disc Brake Squeal. *Journal of Sound and Vibration*, Vol. 267, Issue 1, 105–166. Academic Press. [https://doi.org/10.1016/S0022-460X\(02\)01573-0](https://doi.org/10.1016/S0022-460X(02)01573-0)

Kissinger, T., James, S., Weber, S., Mullaney, K., Chehura, E., H Pekmezci, H., H Barrington, J., E Staines, S., Charrett, T., J Lawson, N., Lone, M., Attack, R. and Tatam, R. (2022). Fibre-optic measurement of strain and shape on a helicopter rotor blade during a ground run: 2. Measurement of shape. *Smart Materials and Structures*, 31(7), p.075015.

Kokaje, P. G., Kawitkar, R.S., & Balan, M. S. (2015). Review of Recent Development in Optical Fiber Technology. *International Journal of Advanced Research in Electrical, Electronics and Instrumentation Engineering*, 04(03), 1546–1549. <https://doi.org/10.15662/ijareeie.2015.0403037>

Lang, A.M., Smales, H. (1983). An Approach to The Solution of Disc Brake Vibration Problems. *Paper C37/83, IMechE. Conference on Braking of Road Vehicles, 1983.*

Laser Focus World. (2022). Optics Fabrication: Fiber Bragg grating fabrication system is automated. <https://www.laserfocusworld.com/fiber-optics/article/16547082/optics-fabrication-fiber-bragg-grating-fabrication-system-is-automated>

Lazzari, A., Tonazzi, D., Conidi, G., Malmassari, C., Cerutti, A., & Massi, F. (2018). Experimental Evaluation of Brake Pad Material Propensity to Stick-Slip and Groan Noise Emission. *Lubricants*, 6, 107. <https://doi.org/10.3390/lubricants6040107>

Lenormand, P. P. H. (2017). Disc Brake/Pad Interface Pressure Distribution Investigation. *MSc Thesis. Cranfield University, UK.*

Li, X.J. et al. (2014) "FBG sensing demodulation technology based on acousto-optic tunable filter," *Applied Mechanics and Materials*, 513-517, pp. 4257–4260. <https://doi.org/10.4028/www.scientific.net/amm.513-517.4257>.

Liang, M., Fang, X. and Ning, Y. (2018) Temperature compensation fiber bragg grating pressure sensor based on plane diaphragm. *Photonic Sensors*, 8(2), pp. 157–167. <https://doi.org/10.1007/s13320-018-0417-9>.

Liang, L., Wang, R., Xu, G., Yi, Z., & Tu, B. (2019). Analysis and Study of FBG Sensor in Crack Monitoring of Aircraft Structure. *IOP Conference Series: Earth*

and *Environmental Science*, 300(2). <https://doi.org/10.1088/1755-1315/300/2/022111>

Limmer, F., Paulus, A., Barton, D., Brooks, P., Neville, A., & Kosarieh, S. (2020). A Comparison of Methods for Characterizing Brake Pad Surfaces. *Euro Brake 2020*, EB2020-FBR-027.

Lindberg, E., Hörlin, N. E., & Göransson, P. (2013). An Experimental Study of Interior Vehicle Roughness Noise from Disc Brake Systems. *Applied Acoustics*, 74(3), 396–406. <https://doi.org/10.1016/j.apacoust.2012.07.011>

Liu, D. et al. (2016) Research progress in the key device and technology for Fiber Optic Sensor Network. *Photonic Sensors*, 6(1), pp. 1–25. Available at: <https://doi.org/10.1007/s13320-015-0299-z>.

Liu, L.L. (2022). Theoretical and Experimental Study of Identification of Brake Pad Material Properties. *PhD Thesis. University of Liverpool, UK*.

Major, B. T. M. (2016). Disc Brake Interface Pressure Distribution Measurements. *MSc Thesis. Cranfield University, UK*.

Major, B. T., Tiwari, D., Correia, R., James, S. W., Tirovic, M., & Tatam, R. P. (2017). Fibre Bragg Grating Sensors for The Analysis of Pressure Distribution at A Disc Brake/Pad Interface. *25th International Conference on Optical Fiber Sensors*, 10323, 103232O. <https://doi.org/10.1117/12.2263455>

MatWeb. (2022). Overview of materials for Cyanoacrylate Adhesive. <https://www.matweb.com/search/DataSheet.aspx?MatGUID=d0d7dbec7666421caf8aa08724b634c5&ckck=1>

Mills, H.R. (1938). Brake Squeak. *Automobile Research Committee*.

North, M.R. (1969). A survey of published work on vibrations in braking systems. *MIRA*.

North, M.R. (1976). Disc Brake Squeal. Paper C38/76, IMechE. *Conference on Braking of Road Vehicles, 1976*.

Ogawa, K. et al. (2019) "Wireless, portable fiber Bragg grating interrogation system employing optical edge filter," *Sensors*, 19(14), p. 3222. <https://doi.org/10.3390/s19143222>.

Othonos, A. and Kalli, K. (1999) Fiber Bragg gratings: Fundamentals and applications in telecommunications and sensing. *Boston: Artech House*: <https://doi.org/10.1063/1.883086>

Otto, J., & Ostermeyer, G. P. (2019). High-Frequency Vibrations in The Contact of Brake Systems. *Facta Universitatis, Series: Mechanical Engineering*, 17(2), 103–112. <https://doi.org/10.22190/FUME190326017O>

Paliwal, M., Mahajan, A., Don, J., Chu, T., & Filip, P. (2005). Noise and Vibration Analysis of A Disc-Brake System using A Stick-Slip Friction Model involving

Coupling Stiffness. *Journal of Sound and Vibration*, 282(3–5), 1273–1284.
<https://doi.org/10.1016/j.jsv.2004.05.005>

Panier, S., Dufrénoy, P., Weichert, D., 2004. An Experimental Investigation of Hot Spots in Railway Disc Brakes. *Wear* 256 (7-8), 764-773.

Papinniemi, A. (2007). Vibro–Acoustic Studies of Brake Squeal. *PhD Thesis, The University of New South Wales, Australia*.

Park, J. et al. (2017) “Dynamic Fiber Bragg grating strain sensor interrogation with real-time measurement,” *Optical Fiber Technology*, 38, pp. 147–153.
<https://doi.org/10.1016/j.yofte.2017.09.011>.

Permabond Engineering Adhesives. (2022). Adhesive 920 Technical Data Sheet.
https://www.permabond.com/wp-content/uploads/2022/09/920_TDS.pdf

Posada-Roman, J. et al. (2016) “Fast interrogation of Fiber Bragg gratings with electro-optical dual optical frequency combs,” *Sensors*, 16(12), p. 2007.
<https://doi.org/10.3390/s16122007>.

RF Wireless World. (2022). Optical Sensor basics | optical sensor Types.
<https://www.rfwireless-world.com/Terminology/optical-sensor-basics-and-optical-sensor-types.html>

Sahota, J.K., Gupta, N. and Dhawan, D. (2020) “Fiber Bragg grating sensors for monitoring of physical parameters: A comprehensive review,” *Optical Engineering*, 59(06), p. 1. <https://doi.org/10.1117/1.oe.59.6.060901>

Sergeant, N. (2010). Analysis and Optimisation of Disc Brake Calipers. *PhD Thesis. Cranfield University, UK.*

Shi, L. B., Wang, F., Ma, L., Liu, Q. Y., Guo, J., & Wang, W. J. (2018). Study of The Friction and Vibration Characteristics of The Braking Disc/Pad Interface under Dry and Wet Conditions. *Tribology International*, 127, 533–544. <https://doi.org/10.1016/j.triboint.2018.07.012>

Silveira, P.C. et al. (2018) “Experimental evaluation of low-cost interrogation techniques for FBG Sensors,” 2018 IEEE International Instrumentation and Measurement Technology Conference <https://doi.org/10.1109/i2mtc.2018.8409722>.

Smart Fibres. (2022). SmartScan FBG Interrogator Data Sheet. <https://www.smartfibres.com/files/pdf/SmartScan.pdf>

Söderberg, A., & Andersson, S. (2009). Simulation of Wear and Contact Pressure Distribution at The Pad-to-Rotor Interface in A Disc Brake using General Purpose Finite Element Analysis Software. *Wear*, 267(12), 2243–2251. <https://doi.org/10.1016/j.wear.2009.09.004>

Stender, M., Tiedemann, M., Spieler, D., Schoepflin, D., Hoffmann, N., & Oberst, S. (2021). Deep Learning for Brake Squeal: Brake Noise Detection, Characterization and Prediction. *Mechanical Systems and Signal Processing*, 149, 107181. <https://doi.org/10.1016/j.ymssp.2020.107181>

Talbot, C., and Fieldhouse, J. D. (2002). Investigations of In-Plane Disc Vibration Using Laser Holography. *SAE Technical Paper 2002-01-2607*.

Tian, K., Liu, Y. and Wang, Q. (2005) "Temperature-independent fiber Bragg grating strain sensor using bimetal cantilever," *Optical Fiber Technology*, 11(4), pp. 370–377. <https://doi.org/10.1016/j.yofte.2005.03.001>.

Tirovic, M., & Day, A. J. (1991). Disc Brake Interface Pressure Distributions. *Proceedings of the Institution of Mechanical Engineers Vol. 205 Part D: J. Automobile Engineering*, 137-146.

Tirovic, M., Vianello, M., & Bannister, P. (2019). Methodology for Predicting Brake Squeal Propensity Using Complex Eigenvalue Analysis, Including Thermo-Mechanical Effects. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 233(8), 2048–2073. <https://doi.org/10.1177/0954407018765251>

Tiwari, U., Thyagarajan, K. and Shenoy, M.R. (2014). Strain and temperature discrimination technique by use of a FBG written in erbium doped fiber. *Optic*, 125(1), pp. 235–237. <https://doi.org/10.1016/j.jileo.2013.06.067>.

Vianello, M. (2015). Brake squeal propensity prediction methodology using complex eigenvalue analysis and including thermo-mechanical effects. *PhD Thesis. Cranfield University, UK*.

Wang, Z. C., Tirovic, M., James, S. W. and Tatam, R. P. (2021). Investigation of Disc Brake Interface Pressure Distributions using FBG Sensors. *Euro Brake 2021*,

EB2021-STP-021. <https://www.fisita.com/library/eurobrake/2021/6216eb2021-stp-021>

Wang, Z. C., Tirovic, M., James, S. W., Barrington, J. H., and Tatam, R. P. (2022). Investigation of Pad/Disc Interface Using Optical Fibre Strain Sensors. *Euro Brake* 2022, EB2022-TSD-003. <https://www.fisita.com/library/eurobrake/2022/eb2022-tsd-003>

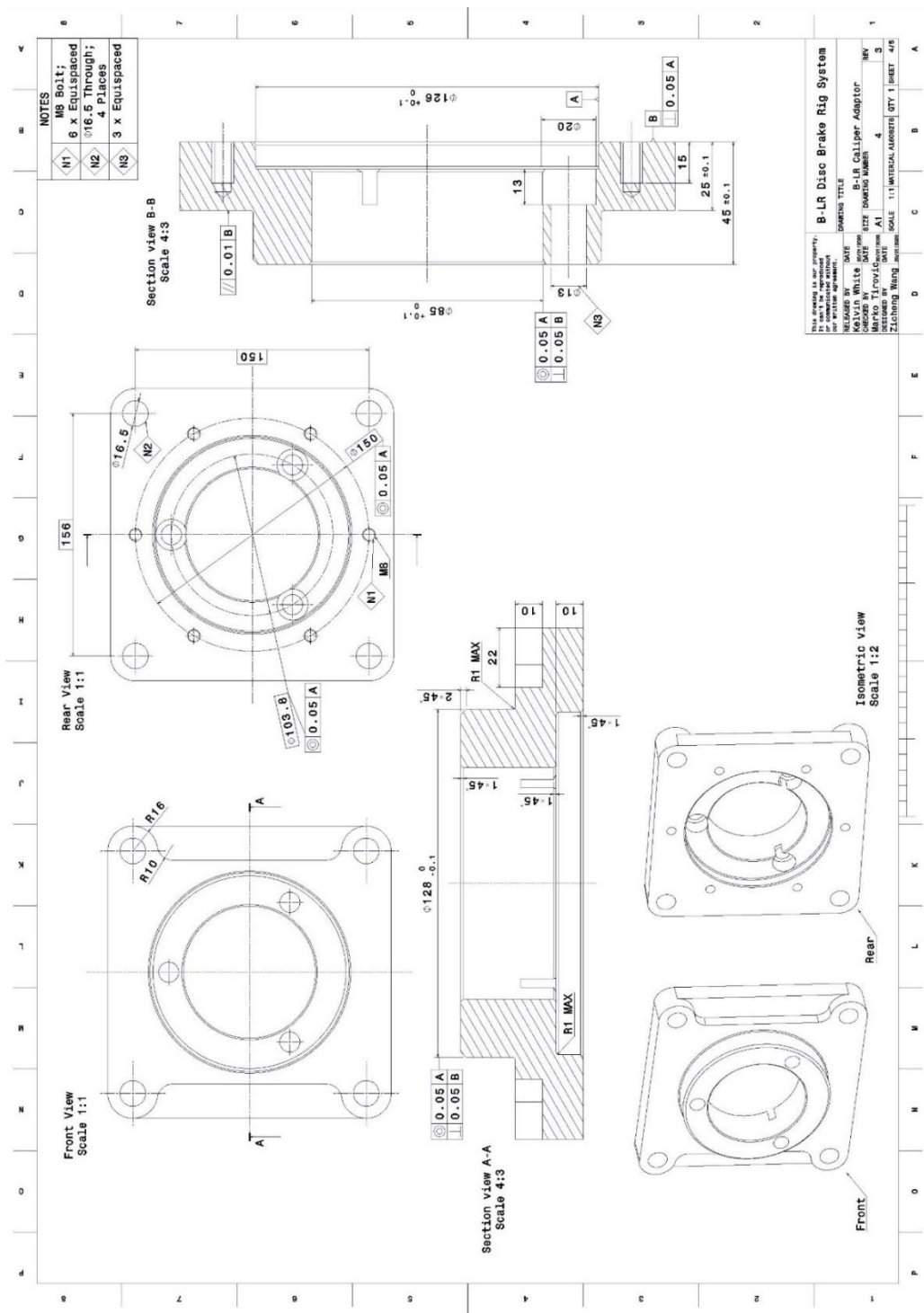
Wegmann, E., Stenkamp, A., & Dohle, A. (2009). Relation Between Compressibility and Viscoelastic Material Properties of a Brake Pad. *SAE International 2009-01-3017, 2009 Brake Colloquium and Exhibition.*

Wu, Q., Okabe, Y., & Yu, F. (2018). Ultrasonic Structural Health Monitoring using Fiber Bragg Grating. *Sensors (Switzerland), Vol. 18, Issue 10, 3395. MDPI AG.* <https://doi.org/10.3390/s18103395>

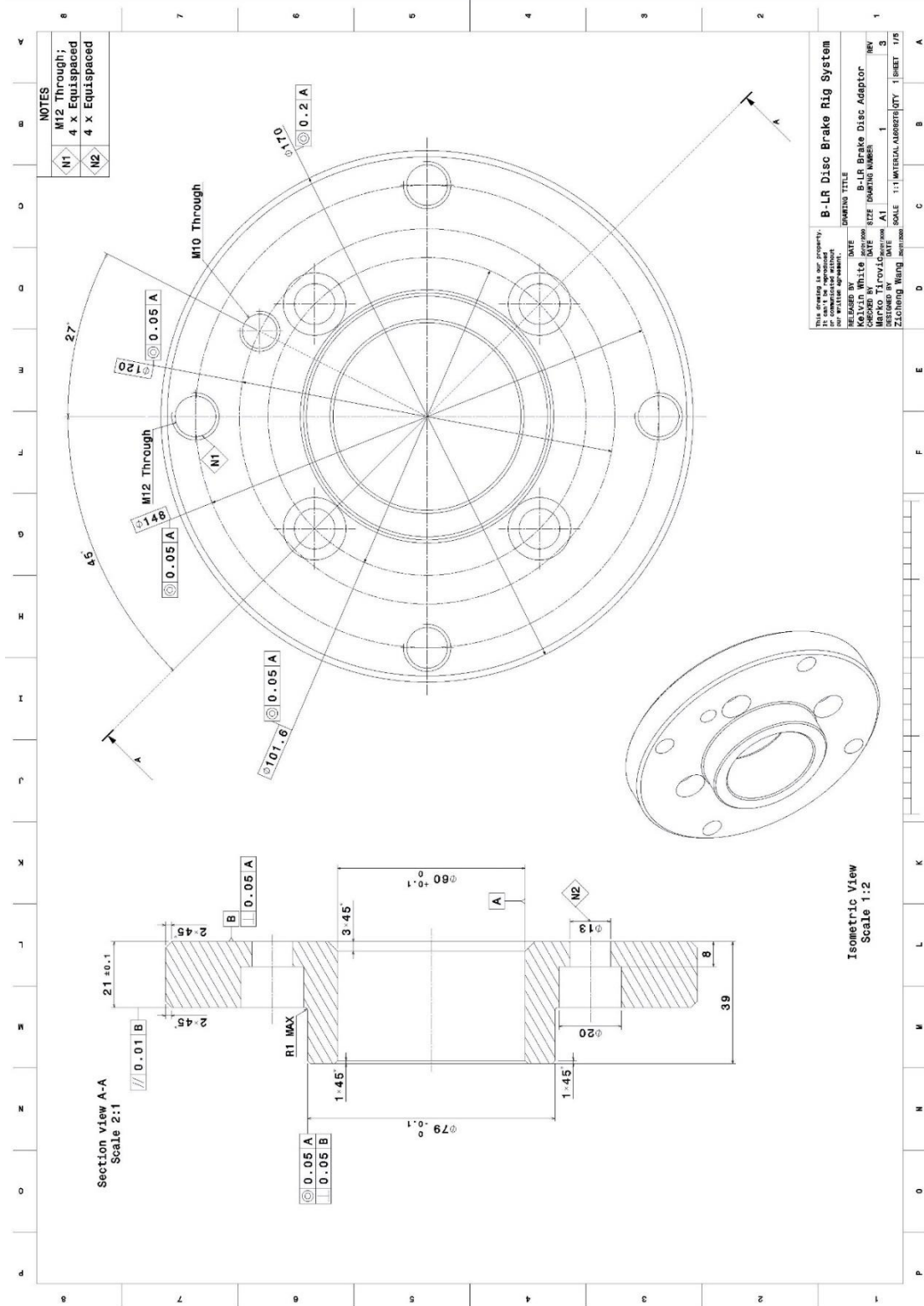
Yu, F. T. S. and Yin, S. (2002). *Fiber Optic Sensors*, Taylor and Francis, Boca Raton.

APPENDICES

Appendix Chapter 3-A: CAD Drawings of Designed Components



(a) The Adaptor 1



(b) The Adaptor 2

Appendix Chapter 3-B: Manufacturing Machines



(a) Bridgeport Mill Table

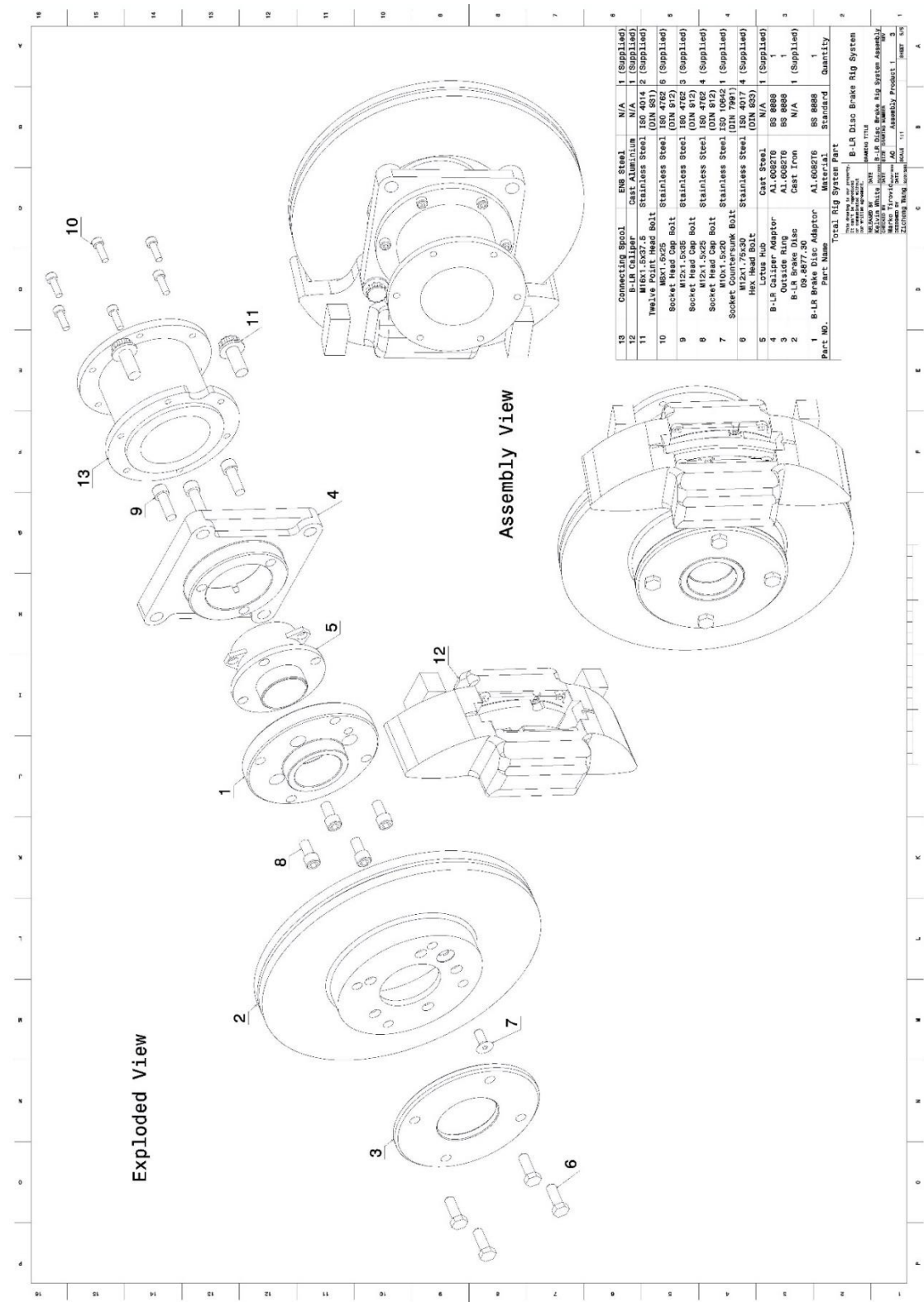


(b) Colchester Centre Lathe

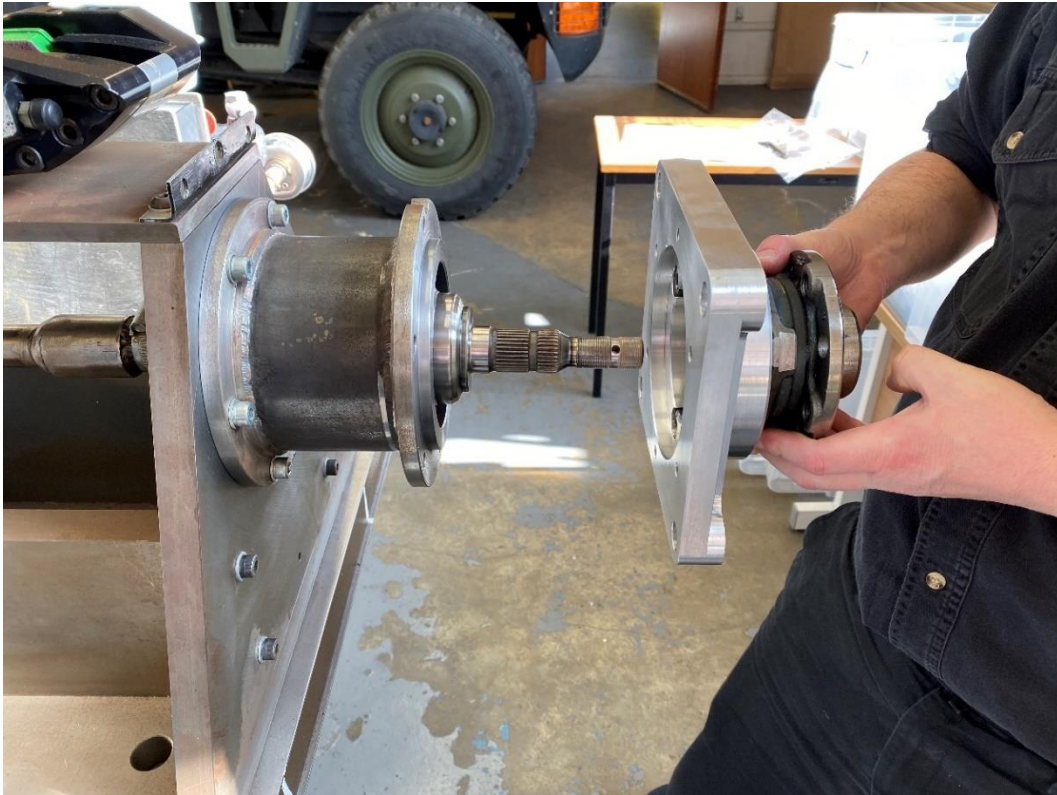


(c) XYZ CNC Lather

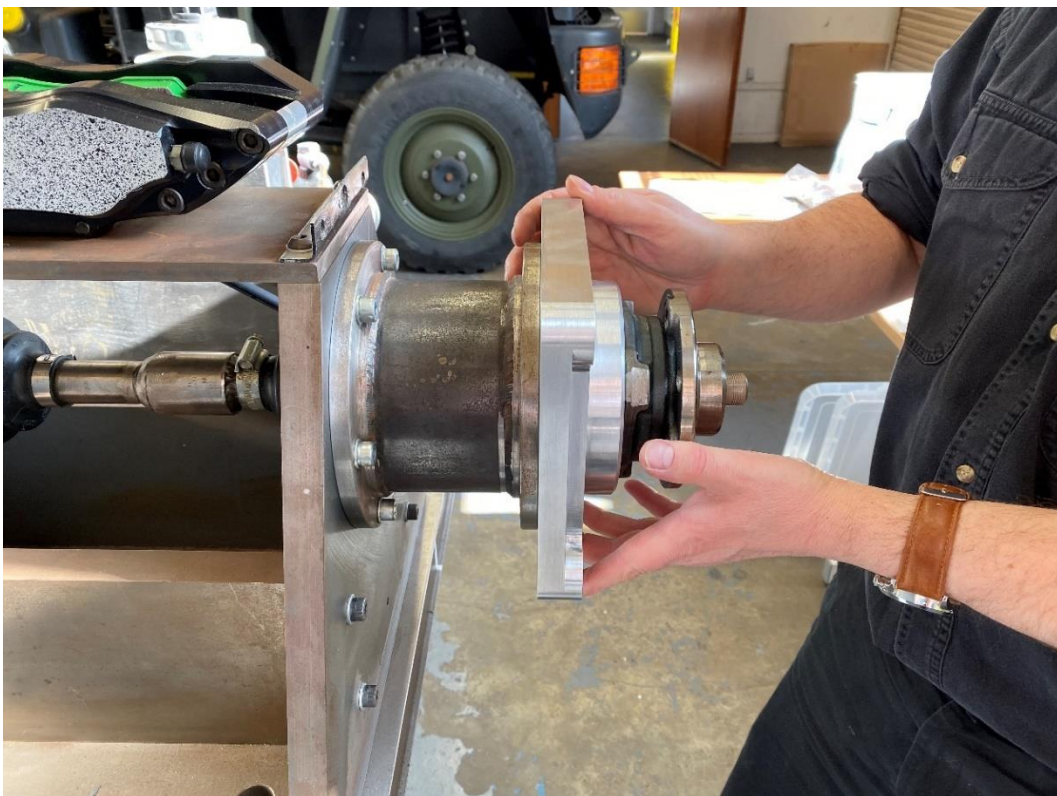
Appendix Chapter 3-C: Completed Assembly Phases



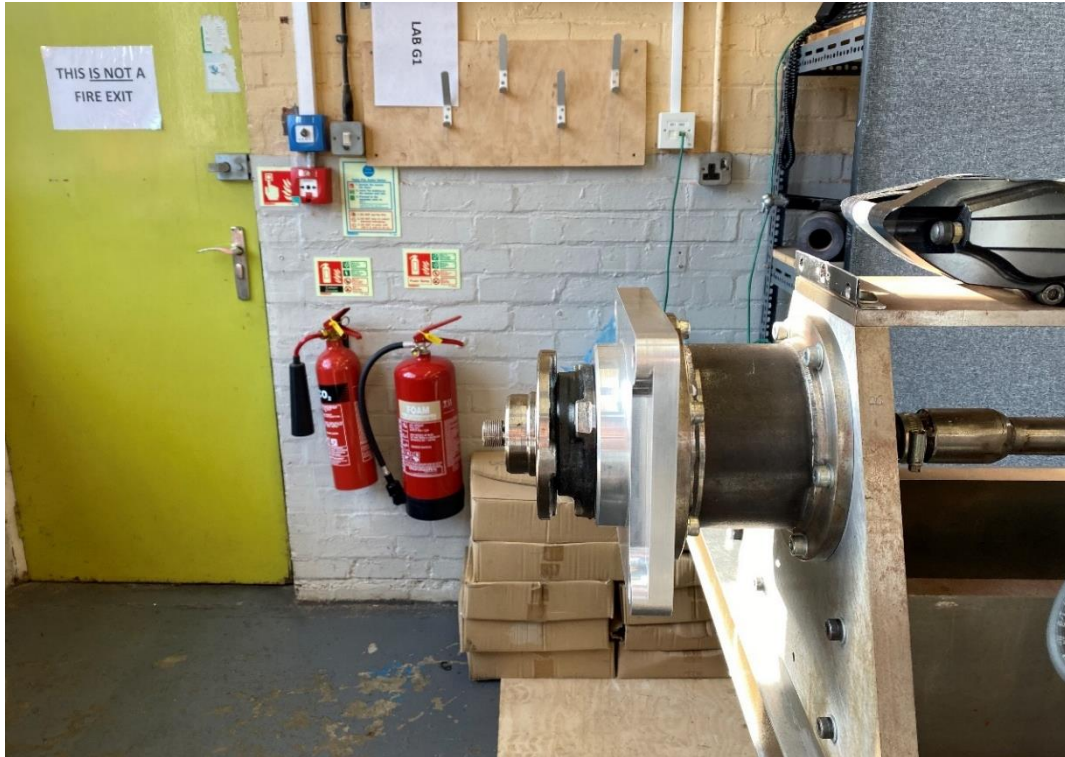
(a) Exploded and Assembled Views of The Land Rover Disc Brake Assembly



(b) The Assembly Phase: The Adaptor 1 and The Original Adaptor B



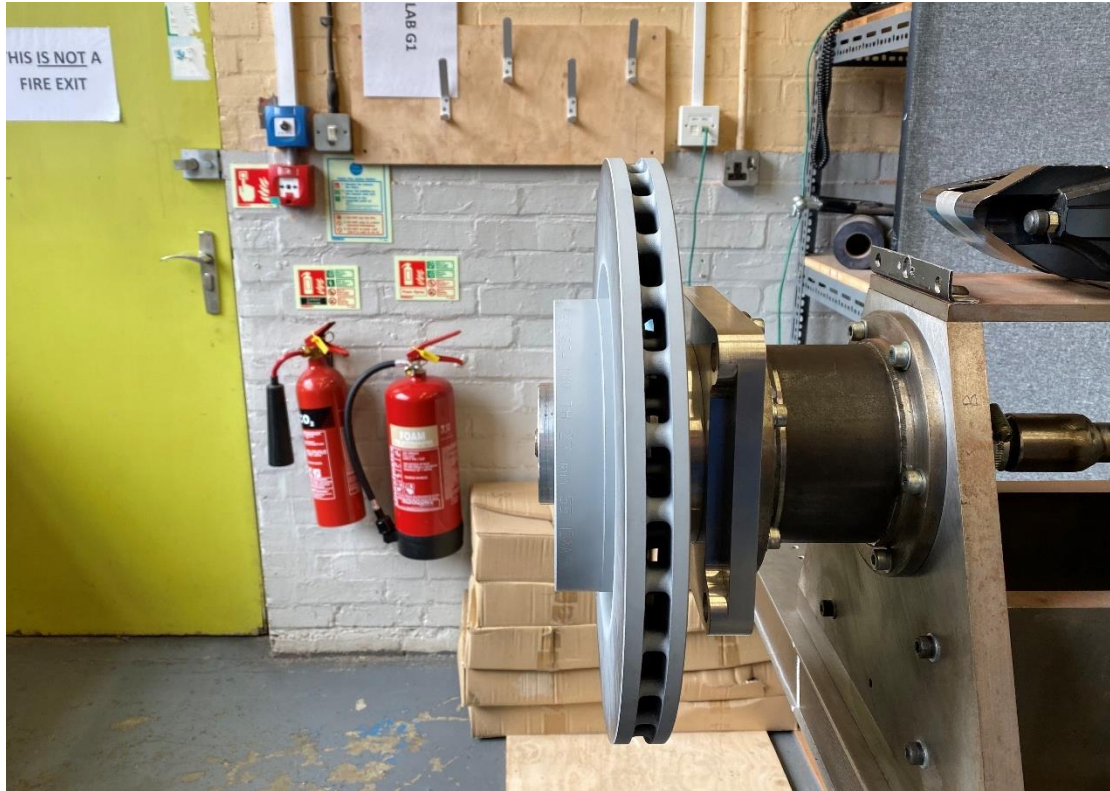
(c) The Assembly Phase: The Original Adaptor A



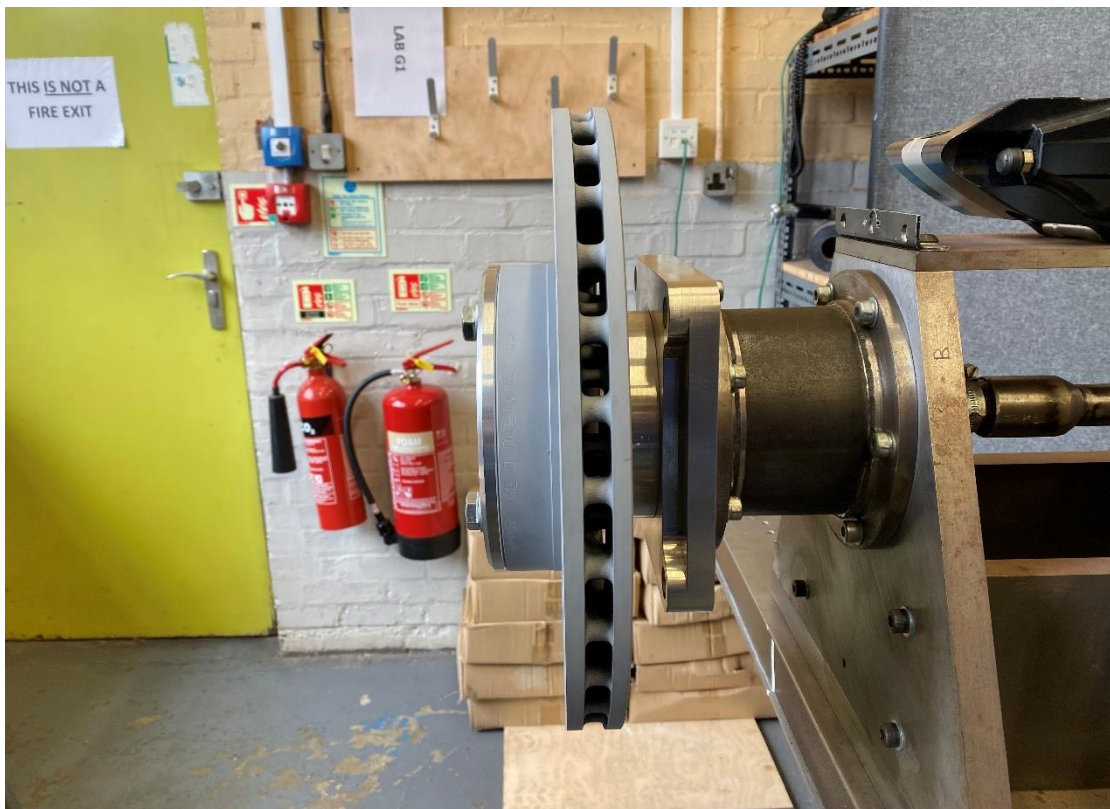
(d) The Assembly Phase: The Original Adaptor B, The Adaptor 1 and The Original Adaptor A



(e) The Assembly Phase: The Adaptor 2

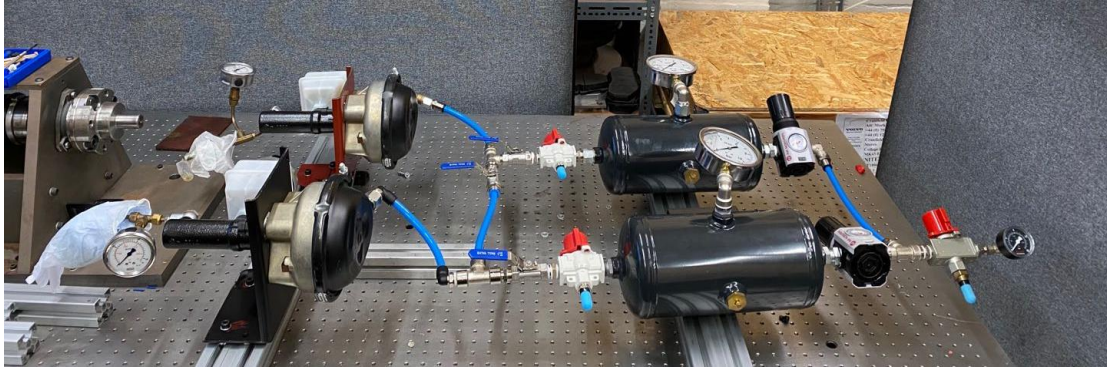


(f) The Assembly Phase: The Brake Disc

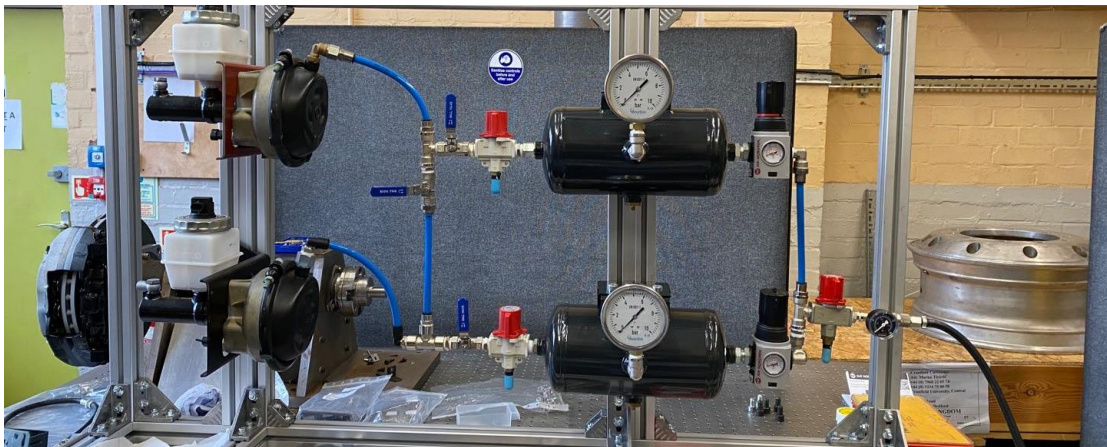


(g) The Assembly Phase: The Outside Ring

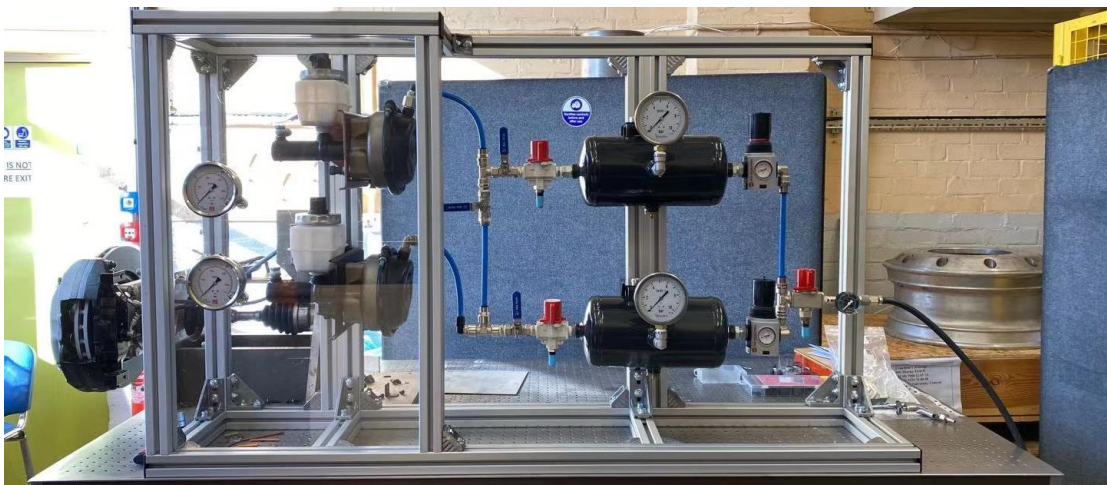
Appendix Chapter 3-D: The Assembly Process of The Two-Channel Control Hydraulic Pressure System



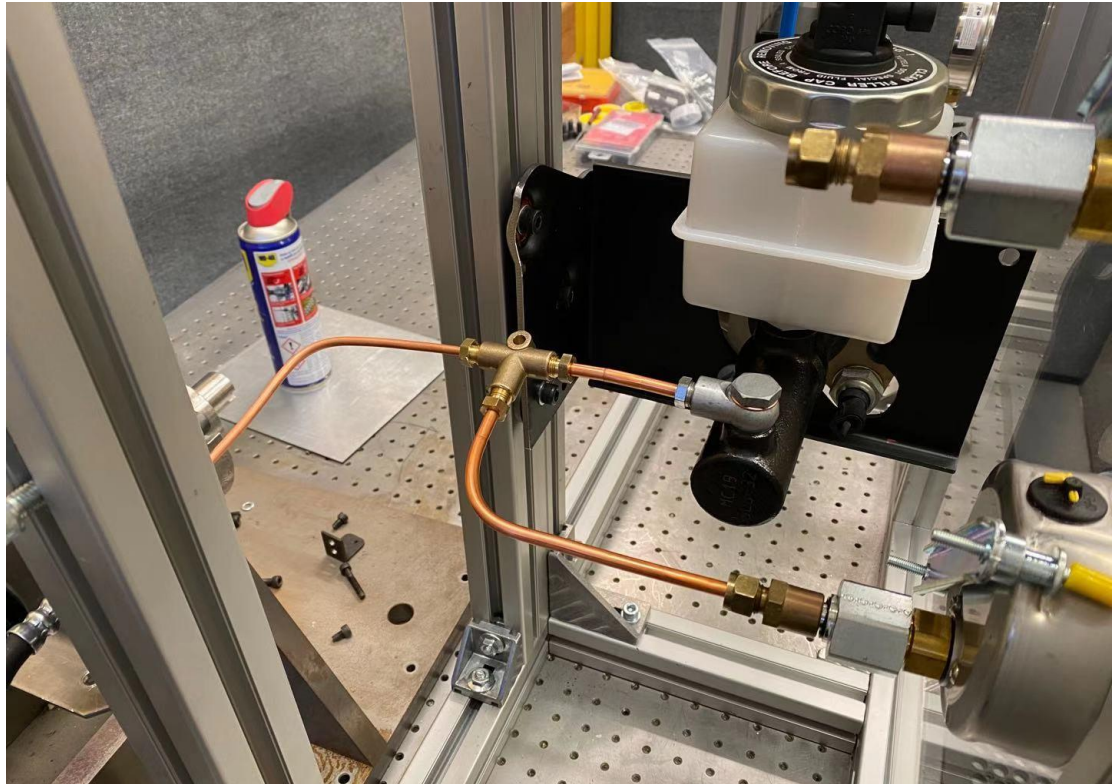
(a) The Two-Channel Draft Installation



(b) The Aluminium Frame Structure



(c) The Protective Shield



(d) The Connection Section with Several Components

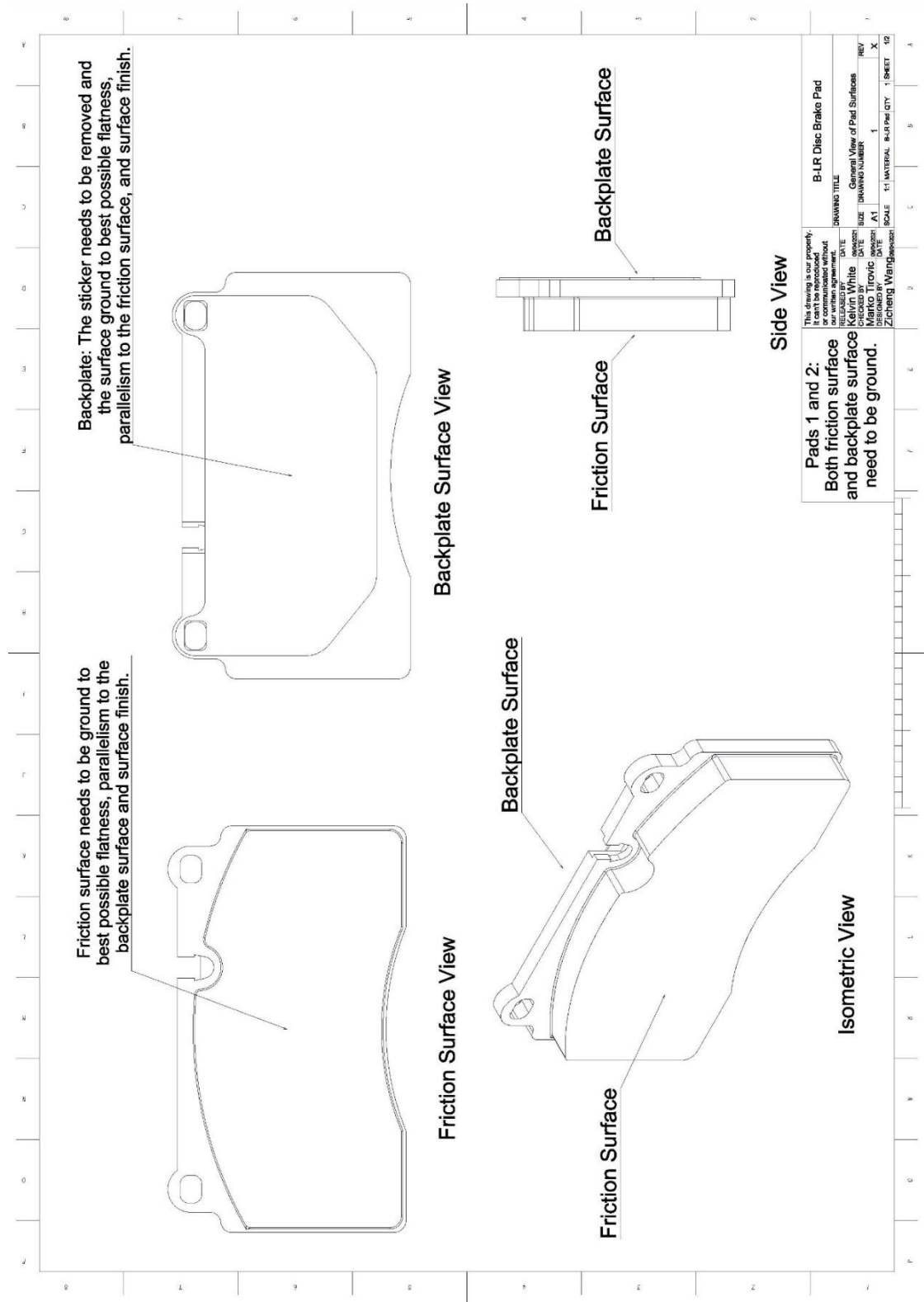


(e) The Final Copper Pipe Connection

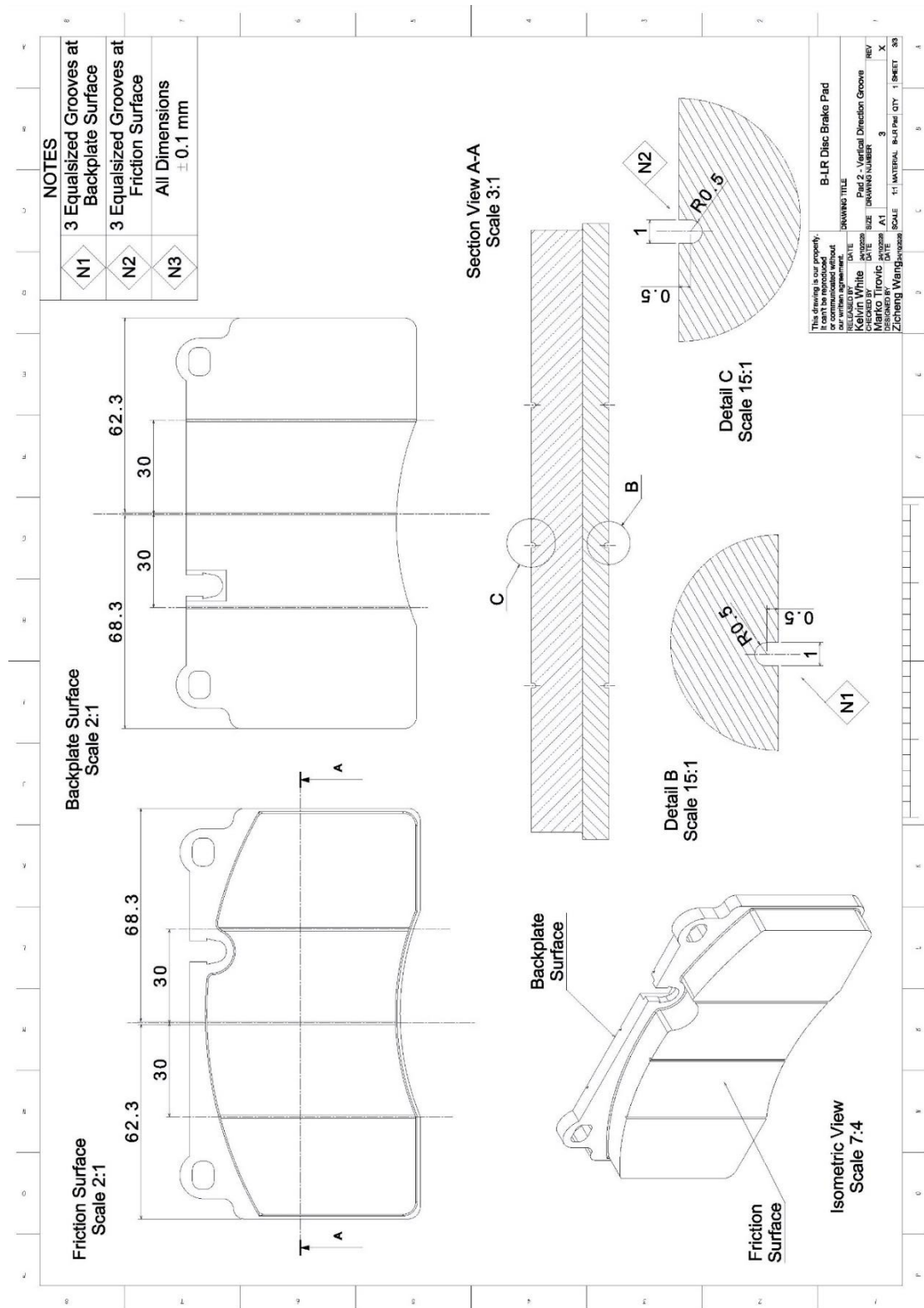
Appendix Chapter 3-E: The Pneumatic Pressure and Hydraulic Pressure Initial Measurement

Pneumatic Pressure and Hydraulic Pressure Measurement (18.2 °C; 55.9%; 05/02/2021)			
Pneumatic Pressure (bar)	Upper Line Hydraulic Pressure (bar)	Lower Line Hydraulic Pressure (bar)	Average Hydraulic Pressure (bar)
0	0	0	0
0.4	2/3/3	3/2/3	3
0.6	6/6/6	5/5/5	6
0.8	10/10/10	10/9/9	10
1	14/15/15	13/14/14	14
1.5	25/25/25	24/24/24	25
2	34/35/35	33/34/34	34
2.5	44/43/43	44/43/43	43
3	54/53/52	53/53/52	53
3.5	62/62/62	61/61/61	62
4	71/72/72	70/71/71	71
4.5	82/82/82	80/80/80	81
5	91/91/91	90/90/90	91
5.5	100/101/100	98/99/98	99
6	110/110/110	108/109/109	109
6.2	114/114/113	113/113/113	113
6.4	117/117/117	116/116/116	117
6.6	121/120/121	120/120/120	120

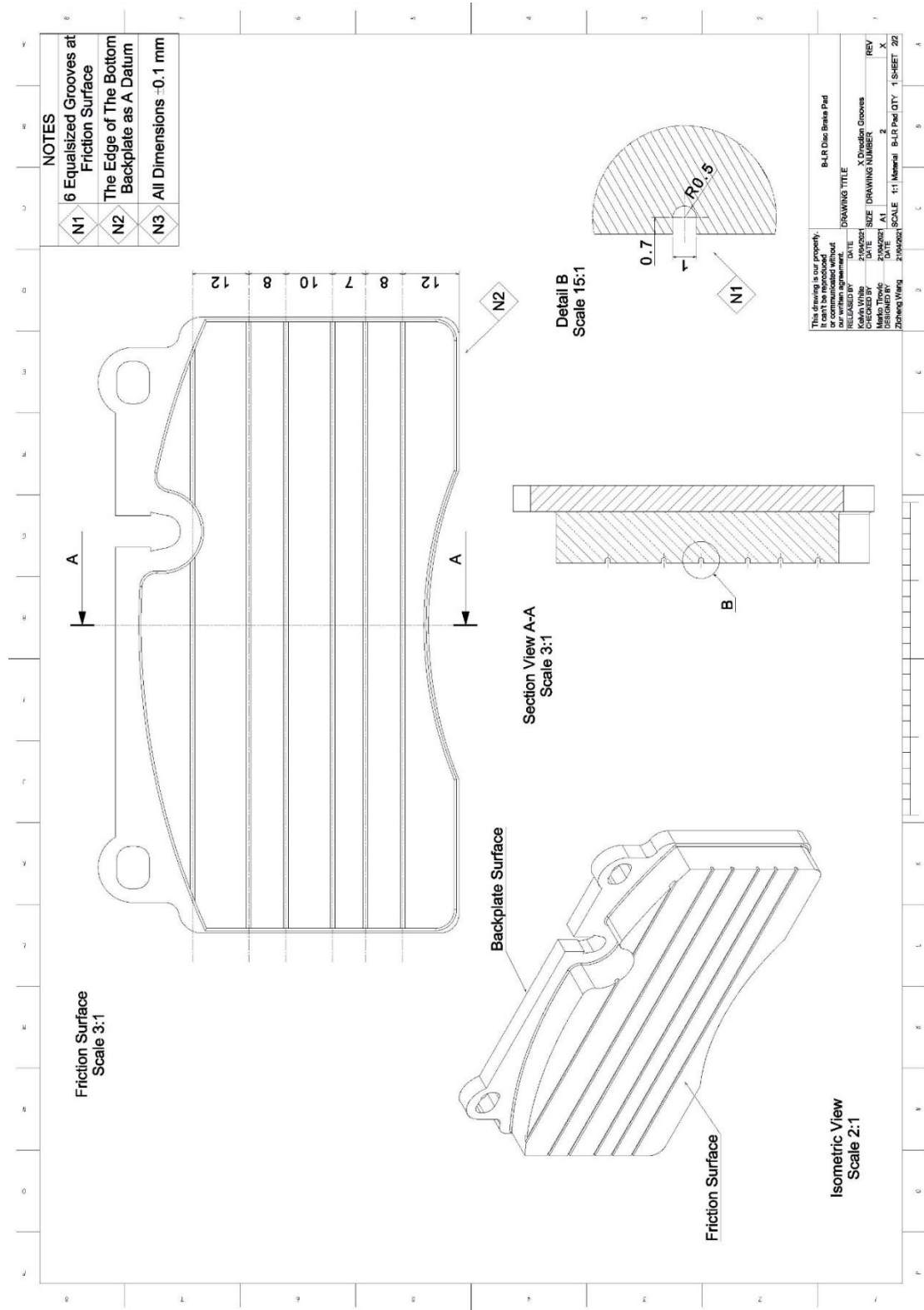
Appendix Chapter 3-F: The Brake Pad Modification



(a) The General View of The Brake Pad



(b) The Design Concept of The Brake Pad with Vertical (Y) Grooves



(c) The Design Concept of The Brake Pad with Horizontal (X) Groove

Appendix Chapter 4-A: MATLAB Program Code

```
clear all;
clc;
close all;

filename = ' ';
startline= ;
endline= ;
datarate=100;
strainsensitivity=0.0012;

delimiter = '\t';
startRow = 6;
formatSpec = '%f%f%f%f%f%f%f%f%f%f%f%f%f%f%f%f%f%[\n\r]';
fileID = fopen(filename,'r');
dataArray = textscan(fileID, formatSpec, 'Delimiter', delimiter,
'TextType', 'string', 'EmptyValue', NaN, 'HeaderLines' ,startRow-1,
'ReturnOnError', false, 'EndOfLine', '\r\n');
fclose(fileID);

Time = dataArray(:, 1);
Pad1_FBG4_49mm = dataArray(:, 2);
Pad1_FBG3_37mm = dataArray(:, 3);
Pad1_FBG2_12mm = dataArray(:, 4);
Pad1_FBG1_0mm = dataArray(:, 5);
Pad2_FBG4_45mm = dataArray(:, 6);
Pad2_FBG3_30mm = dataArray(:, 7);
Pad2_FBG2_15mm = dataArray(:, 8);
Pad2_FBG1_0mm = dataArray(:, 9);
Pad3_FBG3_37mm = dataArray(:, 10);
Pad3_FBG2_12mm = dataArray(:, 11);
Pad3_FBG1_0mm = dataArray(:, 12);
Plate2_FBG1_0mm = dataArray(:, 13);
Plate2_FBG2_15mm = dataArray(:, 14);
Plate2_FBG3_30mm = dataArray(:, 15);
Plate2_FBG4_45mm = dataArray(:, 16);
clearvars filename delimiter startRow formatSpec fileID dataArray ans;

[r,c]=size(Time);
for i = 1:l:r
    strain_Pad1_FBG4_49mm(i,1) = (Pad1_FBG4_49mm(i)-
Pad1_FBG4_49mm(1))/strainsensitivity;
    strain_Pad1_FBG3_37mm(i,1) = (Pad1_FBG3_37mm(i)-
Pad1_FBG3_37mm(1))/strainsensitivity;
    strain_Pad1_FBG2_12mm(i,1) = (Pad1_FBG2_12mm(i)-
Pad1_FBG2_12mm(1))/strainsensitivity;
    strain_Pad1_FBG1_0mm(i,1) = (Pad1_FBG1_0mm(i)-
Pad1_FBG1_0mm(1))/strainsensitivity;
    strain_Pad2_FBG4_45mm(i,1) = (Pad2_FBG4_45mm(i)-
Pad2_FBG4_45mm(1))/strainsensitivity;
    strain_Pad2_FBG3_30mm(i,1) = (Pad2_FBG3_30mm(i)-
Pad2_FBG3_30mm(1))/strainsensitivity;
    strain_Pad2_FBG2_15mm(i,1) = (Pad2_FBG2_15mm(i)-
Pad2_FBG2_15mm(1))/strainsensitivity;
    strain_Pad2_FBG1_0mm(i,1) = (Pad2_FBG1_0mm(i)-
Pad2_FBG1_0mm(1))/strainsensitivity;
    strain_Pad3_FBG3_37mm(i,1) = (Pad3_FBG3_37mm(i)-
```

```

Pad3_FBG3_37mm(1))/strainsensitivity;
    strain_Pad3_FBG2_12mm(i,1) = (Pad3_FBG2_12mm(i) -
Pad3_FBG2_12mm(1))/strainsensitivity;
    strain_Pad3_FBG1_0mm(i,1) = (Pad3_FBG1_0mm(i) -
Pad3_FBG1_0mm(1))/strainsensitivity;
    strain_Plate2_FBG1_0mm(i,1) = (Plate2_FBG1_0mm(i) -
Plate2_FBG1_0mm(1))/strainsensitivity;
    strain_Plate2_FBG2_15mm(i,1) = (Plate2_FBG2_15mm(i) -
Plate2_FBG2_15mm(1))/strainsensitivity;
    strain_Plate2_FBG3_30mm(i,1) = (Plate2_FBG3_30mm(i) -
Plate2_FBG3_30mm(1))/strainsensitivity;
    strain_Plate2_FBG4_45mm(i,1) = (Plate2_FBG4_45mm(i) -
Plate2_FBG4_45mm(1))/strainsensitivity;
end

numdps=1/datarate;
numstart=(startline/numdps)+1;
numend=(endline/numdps)+1;
numdata=numstart-numend;
sum_Pad1_FBG4_49mm = 0;
sum_Pad1_FBG3_37mm = 0;
sum_Pad1_FBG2_12mm = 0;
sum_Pad1_FBG1_0mm = 0;
sum_Pad2_FBG4_45mm = 0;
sum_Pad2_FBG3_30mm = 0;
sum_Pad2_FBG2_15mm = 0;
sum_Pad2_FBG1_0mm = 0;
sum_Pad3_FBG3_37mm = 0;
sum_Pad3_FBG2_12mm = 0;
sum_Pad3_FBG1_0mm = 0;
sum_Plate2_FBG1_0mm = 0;
sum_Plate2_FBG2_15mm = 0;
sum_Plate2_FBG3_30mm = 0;
sum_Plate2_FBG4_45mm = 0;
for i=numstart:1:numend
    sum_Pad1_FBG4_49mm = sum_Pad1_FBG4_49mm+strain_Pad1_FBG4_49mm(i,1);
    sum_Pad1_FBG3_37mm = sum_Pad1_FBG3_37mm+strain_Pad1_FBG3_37mm(i,1);
    sum_Pad1_FBG2_12mm = sum_Pad1_FBG2_12mm+strain_Pad1_FBG2_12mm(i,1);
    sum_Pad1_FBG1_0mm = sum_Pad1_FBG1_0mm+strain_Pad1_FBG1_0mm(i,1);
    sum_Pad2_FBG4_45mm = sum_Pad2_FBG4_45mm+strain_Pad2_FBG4_45mm(i,1);
    sum_Pad2_FBG3_30mm = sum_Pad2_FBG3_30mm+strain_Pad2_FBG3_30mm(i,1);
    sum_Pad2_FBG2_15mm = sum_Pad2_FBG2_15mm+strain_Pad2_FBG2_15mm(i,1);
    sum_Pad2_FBG1_0mm = sum_Pad2_FBG1_0mm+strain_Pad2_FBG1_0mm(i,1);
    sum_Pad3_FBG3_37mm = sum_Pad3_FBG3_37mm+strain_Pad3_FBG3_37mm(i,1);
    sum_Pad3_FBG2_12mm = sum_Pad3_FBG2_12mm+strain_Pad3_FBG2_12mm(i,1);
    sum_Pad3_FBG1_0mm = sum_Pad3_FBG1_0mm+strain_Pad3_FBG1_0mm(i,1);
    sum_Plate2_FBG1_0mm =
sum_Plate2_FBG1_0mm+strain_Plate2_FBG1_0mm(i,1);
    sum_Plate2_FBG2_15mm =
sum_Plate2_FBG2_15mm+strain_Plate2_FBG2_15mm(i,1);
    sum_Plate2_FBG3_30mm =
sum_Plate2_FBG3_30mm+strain_Plate2_FBG3_30mm(i,1);
    sum_Plate2_FBG4_45mm =
sum_Plate2_FBG4_45mm+strain_Plate2_FBG4_45mm(i,1);
end
average_Pad1_FBG4_49mm = -sum_Pad1_FBG4_49mm/numdata;
average_Pad1_FBG3_37mm = -sum_Pad1_FBG3_37mm/numdata;

```



```

average_Pad1_FBG2_12mm = -sum_Pad1_FBG2_12mm/numdata;
average_Pad1_FBG1_0mm = -sum_Pad1_FBG1_0mm/numdata;
average_Pad2_FBG4_45mm = -sum_Pad2_FBG4_45mm/numdata;
average_Pad2_FBG3_30mm = -sum_Pad2_FBG3_30mm/numdata;
average_Pad2_FBG2_15mm = -sum_Pad2_FBG2_15mm/numdata;
average_Pad2_FBG1_0mm = -sum_Pad2_FBG1_0mm/numdata;
average_Pad3_FBG3_37mm = -sum_Pad3_FBG3_37mm/numdata;
average_Pad3_FBG2_12mm = -sum_Pad3_FBG2_12mm/numdata;
average_Pad3_FBG1_0mm = -sum_Pad3_FBG1_0mm/numdata;
average_Plate2_FBG1_0mm = -sum_Plate2_FBG1_0mm/numdata;
average_Plate2_FBG2_15mm = -sum_Plate2_FBG2_15mm/numdata;
average_Plate2_FBG3_30mm = -sum_Plate2_FBG3_30mm/numdata;
average_Plate2_FBG4_45mm = -sum_Plate2_FBG4_45mm/numdata;
figure;
hkl = {'average_Pad1_FBG1_0mm'; 'average_Pad1_FBG2_12mm';

'average_Pad1_FBG3_37mm'; 'average_Pad1_FBG4_49mm'; 'average_Pad2_FBG1_0m
m'; 'average_Pad2_FBG2_15mm';

'average_Pad2_FBG3_30mm'; 'average_Pad2_FBG4_45mm'; 'average_Pad3_FBG1_0m
m'; 'average_Pad3_FBG2_12mm';

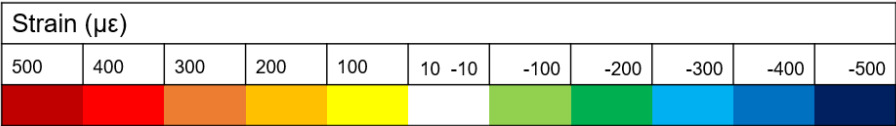
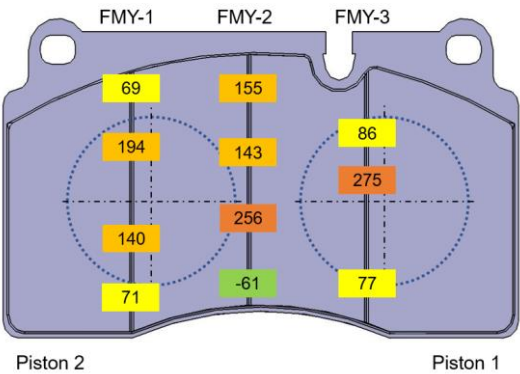
'average_Pad3_FBG3_37mm'; 'average_Plate2_FBG1_0mm'; 'average_Plate2_FBG2
_15mm'; 'average_Plate2_FBG3_30mm';
'average_Plate2_FBG4_45mm';};
Ehkl =
[average_Pad1_FBG1_0mm; average_Pad1_FBG2_12mm; average_Pad1_FBG3_37mm; av
erage_Pad1_FBG4_49mm;

average_Pad2_FBG1_0mm; average_Pad2_FBG2_15mm; average_Pad2_FBG3_30mm; ave
rage_Pad2_FBG4_45mm;
average_Pad3_FBG1_0mm; average_Pad3_FBG2_12mm; average_Pad3_FBG3_37mm;

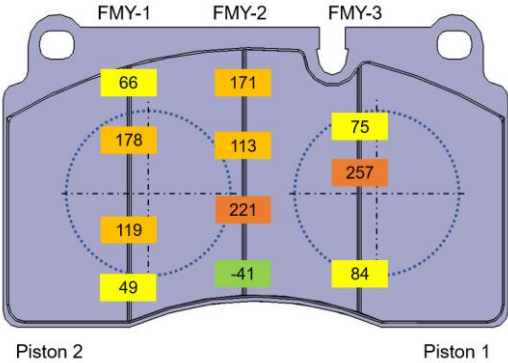
average_Plate2_FBG1_0mm; average_Plate2_FBG2_15mm; average_Plate2_FBG3_30
mm; average_Plate2_FBG4_45mm];
T = table(Ehkl, 'RowNames', hkl);
uitable('Data', T{:, :}, 'ColumnName', T.Properties.VariableNames, ...
'RowName', T.Properties.RowNames, 'Units', 'Normalized',
'Position', [0, 0, 1, 1]);

```

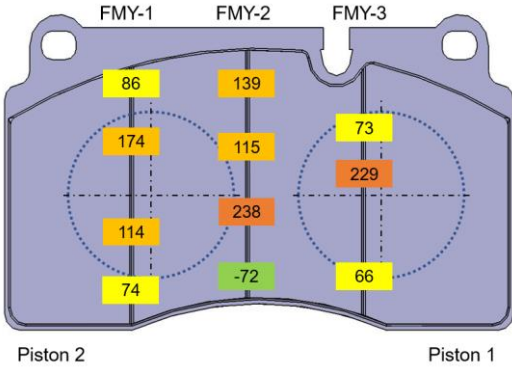

Appendix Chapter 4-B: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Direction Comparisons between Three Pressure Variations at 40bar Pressure



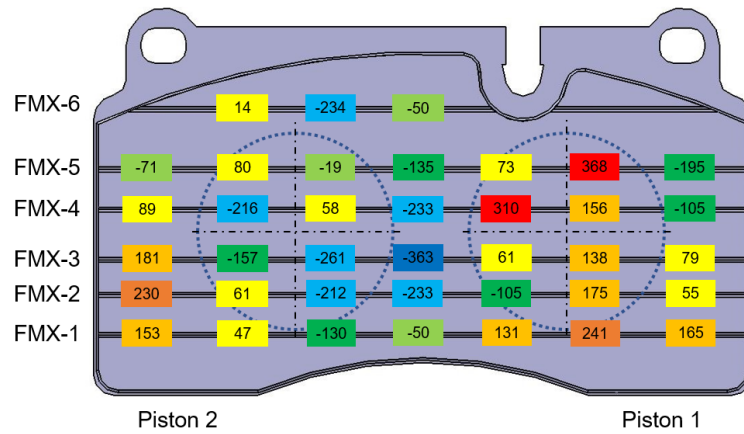
(a) Vertical (Y) Direction: Equal Pressure ($p_1=p_2$)



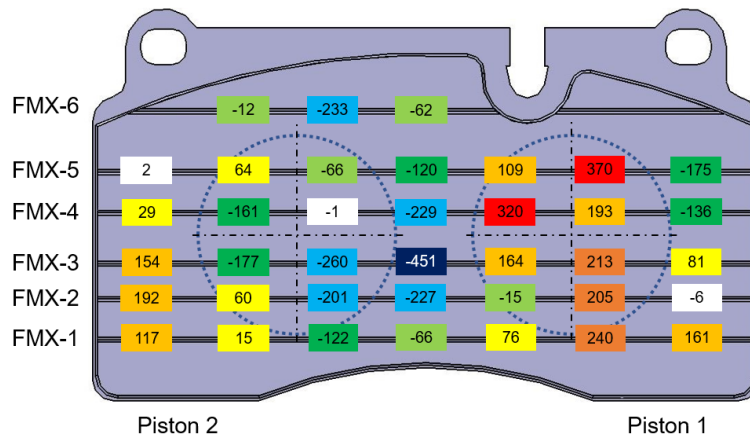
(b) Vertical (Y) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)



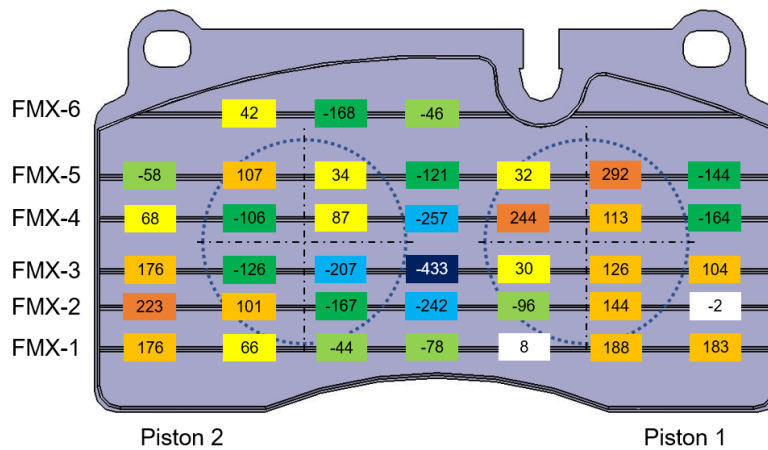
(c) Vertical (Y) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)



(d) Horizontal (X) Direction: Equal Pressure ($p_1=p_2$)



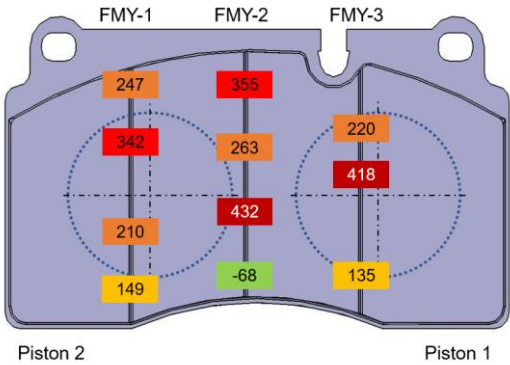
(e) Horizontal (X) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)



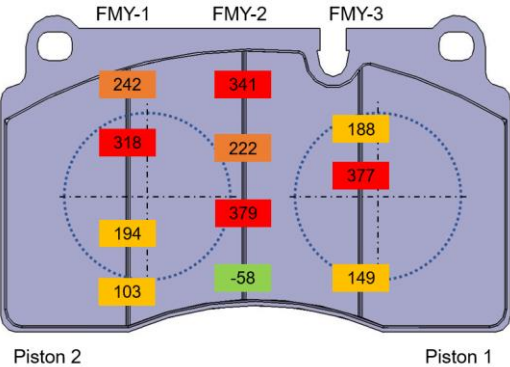
(f) Horizontal (X) Direction: -10bar ($p_1=p_2-10\text{bar}$)

Appendix Chapter 4-C: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Direction Comparisons between Three Pressure Variations at 120bar Pressure

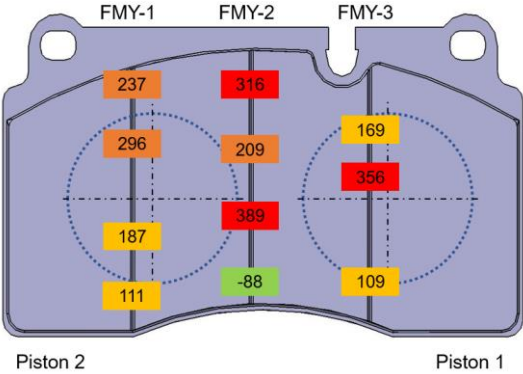
Strain (με)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



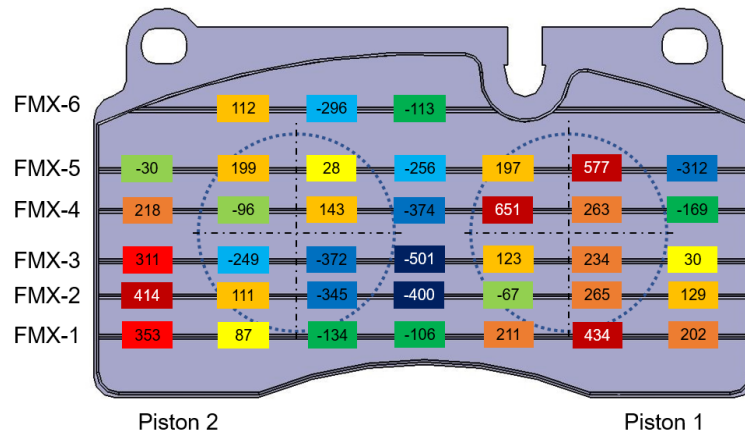
(a) Vertical (Y) Direction: Equal Pressure (p1=p2)



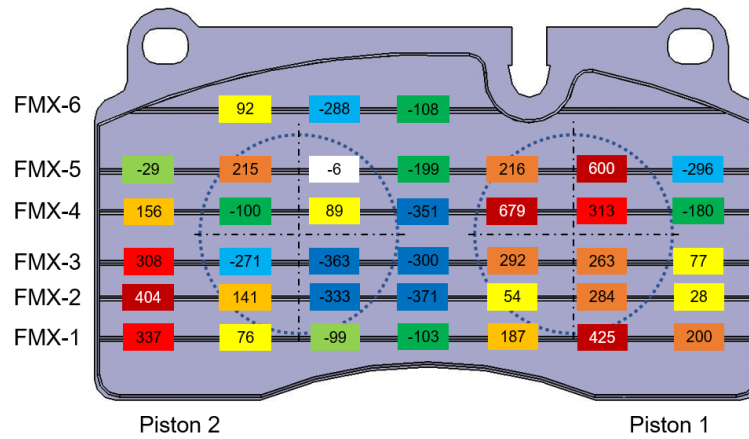
(b) Vertical (Y) Direction: +10bar Variation (p1=p2+10bar)



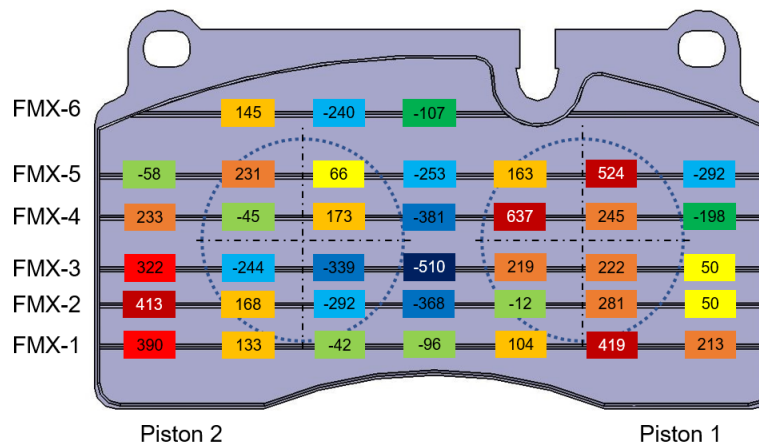
(c) Vertical (Y) Direction: -10bar Variation (p1=p2-10bar)



(d) Horizontal (X) Direction: Equal Pressure ($p_1=p_2$)

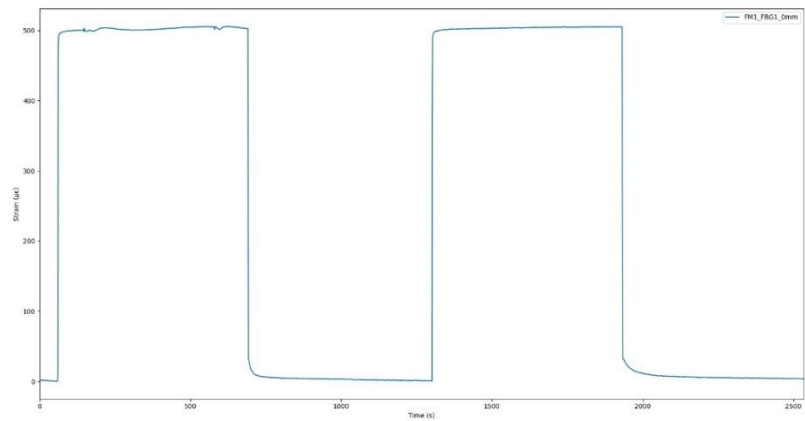


(e) Horizontal (X) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)

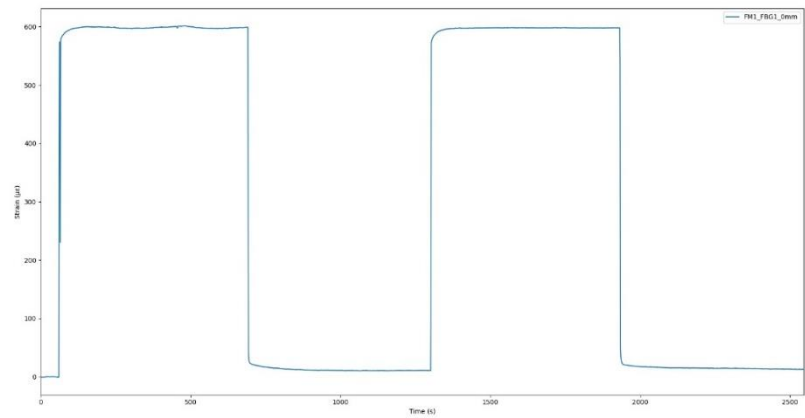


(f) Horizontal (X) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)

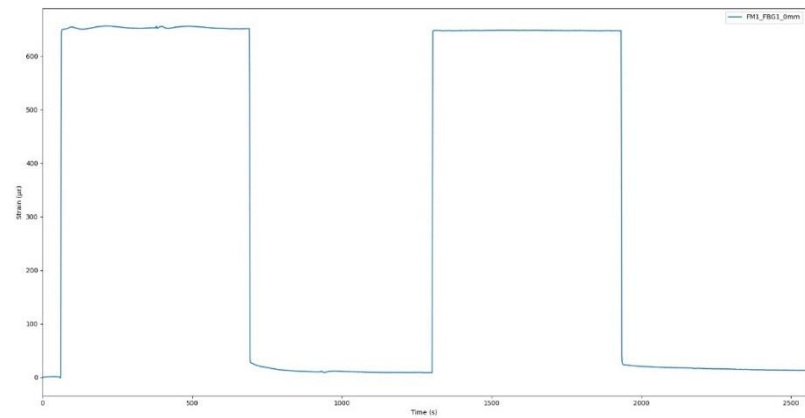
Appendix Chapter 4-D: Visco-elasticity with Different Pressure Level Test Results



(a) Visco-elastic Test Results at 25bar pressure level (FMY1-FBG1)

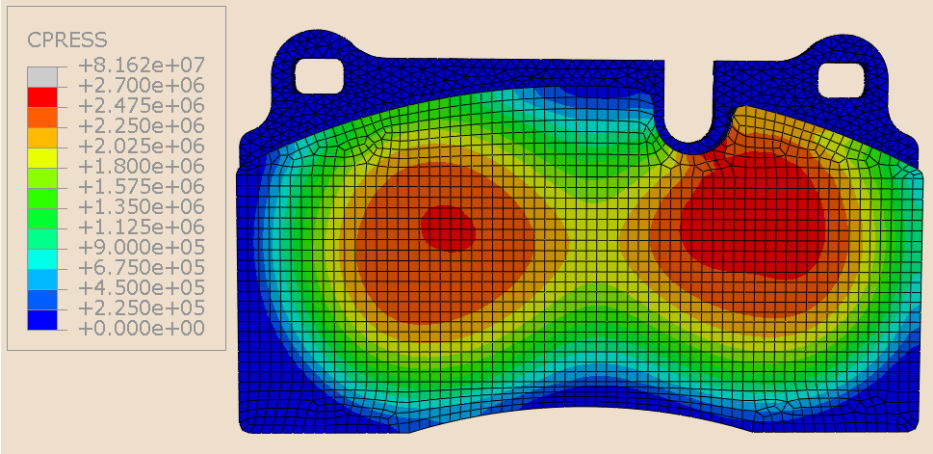


(b) Visco-elastic Test Results at 50bar pressure level (FMY1-FBG1)

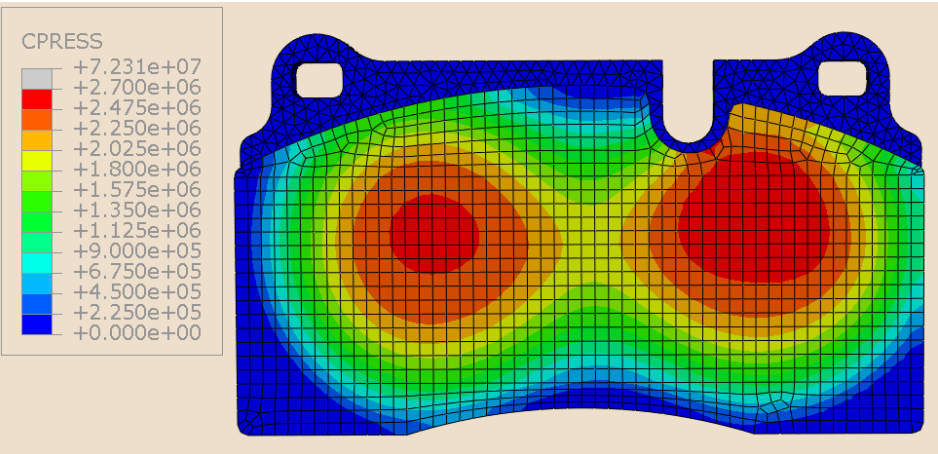


(c) Visco-elastic Test Results at 50bar pressure level (FMY1-FBG1)

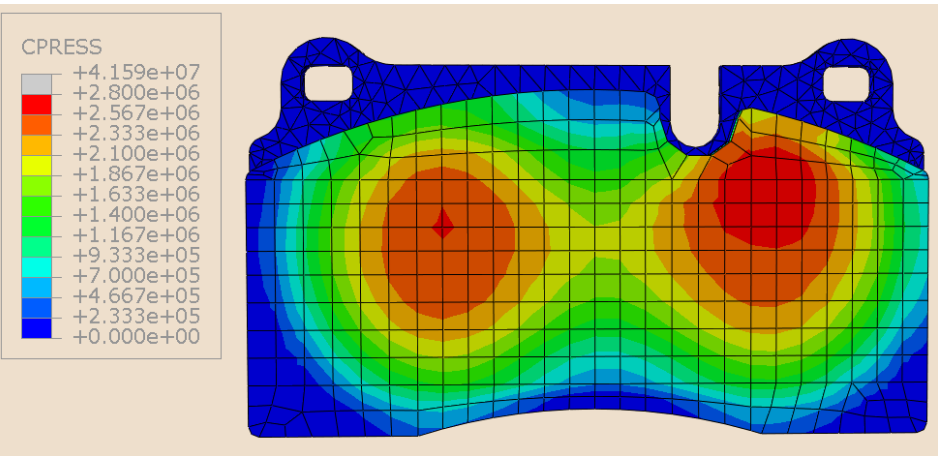
Appendix Chapter 5-A: FEA Results for Different Mesh (Element) Size (CPRESS, E22)



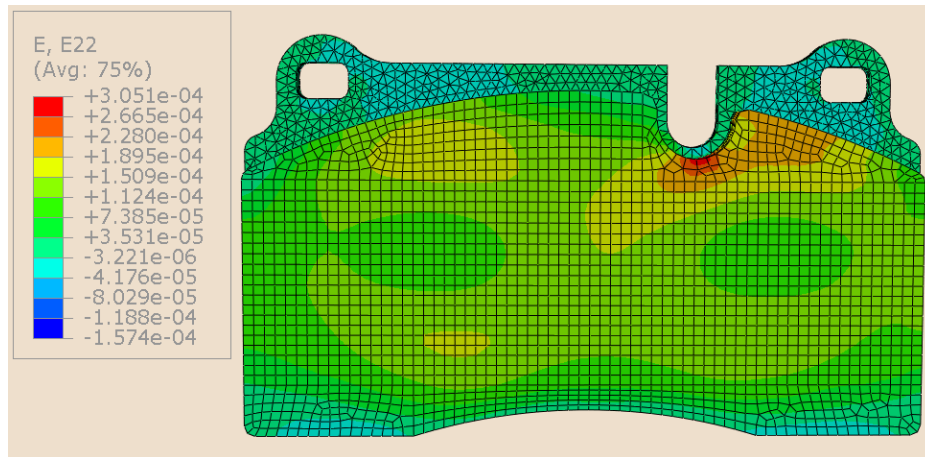
(a) 2mm: CPRESS



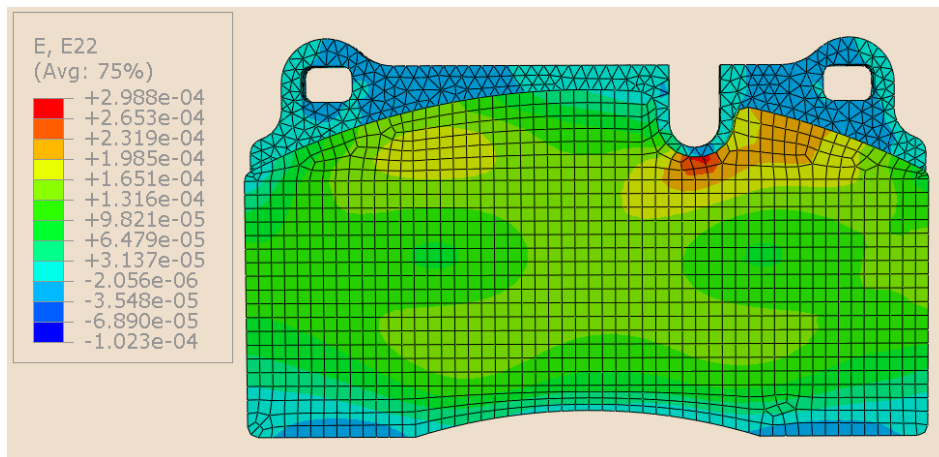
(b) 2.5mm: CPRESS



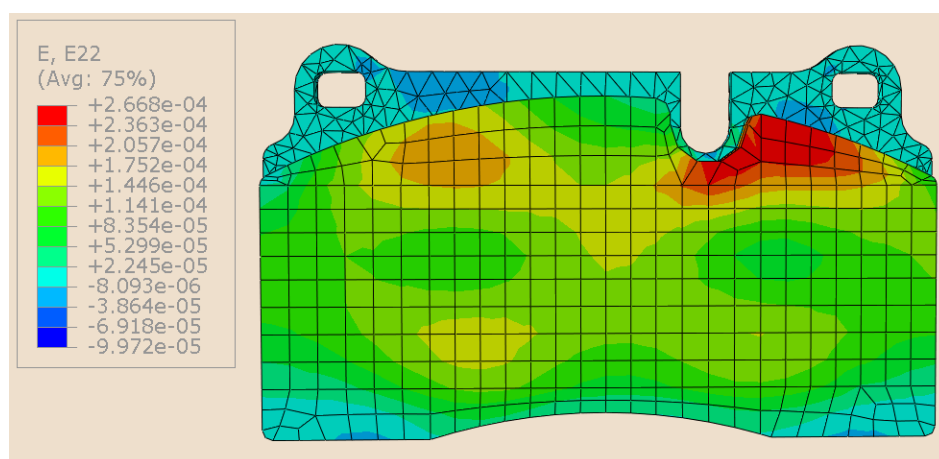
(c) 5mm: CPRESS



(d) 2mm: E22



(e) 2.5mm: E22



(f) 5mm: E22

Appendix Chapter 5-B: The Full Data for Full and Reduced Integration Types

40bar-2.5mm-2.6Gpa-0.3-Reduced

Part Instance	Element ID	Type	Int. Pt.	E, E22
Brake Pad Modelling-1	1	C3D8R	1	60.9989E-06
Brake Pad Modelling-1	27	C3D8R	1	87.8638E-06
Brake Pad Modelling-1	232	C3D8R	1	145.614E-06
Brake Pad Modelling-1	245	C3D8R	1	142.438E-06
Brake Pad Modelling-1	699	C3D8R	1	141.027E-06
Brake Pad Modelling-1	712	C3D8R	1	156.588E-06
Brake Pad Modelling-1	308	C3D8R	1	185.322E-06
Brake Pad Modelling-1	295	C3D8R	1	172.884E-06
Brake Pad Modelling-1	610	C3D8R	1	92.1718E-06
Brake Pad Modelling-1	593	C3D8R	1	110.911E-06
Brake Pad Modelling-1	508	C3D8R	1	148.201E-06
Brake Pad Modelling-1	491	C3D8R	1	148.981E-06
Brake Pad Modelling-1	1089	C3D8R	1	152.43E-06
Brake Pad Modelling-1	1072	C3D8R	1	151.745E-06
Brake Pad Modelling-1	849	C3D8R	1	116.971E-06
Brake Pad Modelling-1	848	C3D8R	1	100.869E-06
Brake Pad Modelling-1	1049	C3D8R	1	115.746E-06
Brake Pad Modelling-1	1048	C3D8R	1	135.045E-06
Brake Pad Modelling-1	1233	C3D8R	1	101.292E-06
Brake Pad Modelling-1	1225	C3D8R	1	113.486E-06
Brake Pad Modelling-1	1187	C3D8R	1	195.941E-06
Brake Pad Modelling-1	1194	C3D8R	1	212.127E-06

40bar-2.5mm-2.6Gpa-0.3-Full

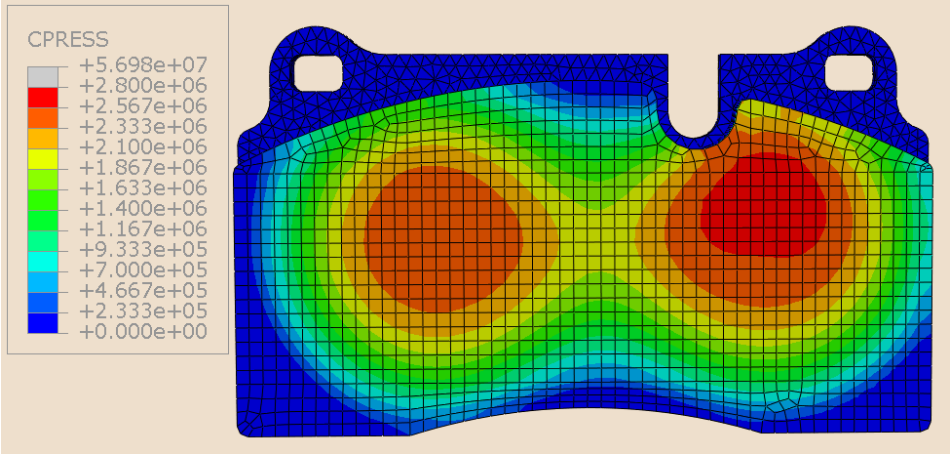
Part Instance	Element ID	Type	Int. Pt.	E, E22
Brake Pad Modelling-1	1	C3D8	1	51.6432E-06
Brake Pad Modelling-1	1	C3D8	2	70.0575E-06
Brake Pad Modelling-1	1	C3D8	3	58.7009E-06
Brake Pad Modelling-1	1	C3D8	4	77.3384E-06
Brake Pad Modelling-1	1	C3D8	5	44.6396E-06
Brake Pad Modelling-1	1	C3D8	6	62.89E-06
Brake Pad Modelling-1	1	C3D8	7	51.7017E-06
Brake Pad Modelling-1	1	C3D8	8	70.1743E-06
Brake Pad Modelling-1	27	C3D8	1	95.3455E-06
Brake Pad Modelling-1	27	C3D8	2	102.779E-06
Brake Pad Modelling-1	27	C3D8	3	77.4938E-06
Brake Pad Modelling-1	27	C3D8	4	84.9978E-06
Brake Pad Modelling-1	27	C3D8	5	87.7241E-06
Brake Pad Modelling-1	27	C3D8	6	95.1858E-06
Brake Pad Modelling-1	27	C3D8	7	70.1255E-06
Brake Pad Modelling-1	27	C3D8	8	77.6603E-06
Brake Pad Modelling-1	232	C3D8	1	147.959E-06
Brake Pad Modelling-1	232	C3D8	2	152.755E-06
Brake Pad Modelling-1	232	C3D8	3	137.817E-06
Brake Pad Modelling-1	232	C3D8	4	143.036E-06
Brake Pad Modelling-1	232	C3D8	5	147.637E-06
Brake Pad Modelling-1	232	C3D8	6	152.668E-06
Brake Pad Modelling-1	232	C3D8	7	137.595E-06
Brake Pad Modelling-1	232	C3D8	8	143.047E-06
Brake Pad Modelling-1	245	C3D8	1	143.544E-06
Brake Pad Modelling-1	245	C3D8	2	146.607E-06
Brake Pad Modelling-1	245	C3D8	3	136.52E-06
Brake Pad Modelling-1	245	C3D8	4	140.167E-06
Brake Pad Modelling-1	245	C3D8	5	144.308E-06
Brake Pad Modelling-1	245	C3D8	6	147.565E-06
Brake Pad Modelling-1	245	C3D8	7	137.328E-06
Brake Pad Modelling-1	245	C3D8	8	141.17E-06
Brake Pad Modelling-1	699	C3D8	1	138.35E-06
Brake Pad Modelling-1	699	C3D8	2	138.947E-06
Brake Pad Modelling-1	699	C3D8	3	139.92E-06
Brake Pad Modelling-1	699	C3D8	4	140.014E-06
Brake Pad Modelling-1	699	C3D8	5	140.303E-06
Brake Pad Modelling-1	699	C3D8	6	140.992E-06
Brake Pad Modelling-1	699	C3D8	7	141.909E-06
Brake Pad Modelling-1	699	C3D8	8	142.095E-06
Brake Pad Modelling-1	712	C3D8	1	151.325E-06

Brake Pad Modelling-1	712	C3D8	2	153.444E-06
Brake Pad Modelling-1	712	C3D8	3	156.166E-06
Brake Pad Modelling-1	712	C3D8	4	157.676E-06
Brake Pad Modelling-1	712	C3D8	5	152.78E-06
Brake Pad Modelling-1	712	C3D8	6	155.094E-06
Brake Pad Modelling-1	712	C3D8	7	157.582E-06
Brake Pad Modelling-1	712	C3D8	8	159.287E-06
Brake Pad Modelling-1	308	C3D8	1	179.554E-06
Brake Pad Modelling-1	308	C3D8	2	187.839E-06
Brake Pad Modelling-1	308	C3D8	3	174.907E-06
Brake Pad Modelling-1	308	C3D8	4	182.926E-06
Brake Pad Modelling-1	308	C3D8	5	179.188E-06
Brake Pad Modelling-1	308	C3D8	6	187.414E-06
Brake Pad Modelling-1	308	C3D8	7	174.901E-06
Brake Pad Modelling-1	308	C3D8	8	182.836E-06
Brake Pad Modelling-1	295	C3D8	1	178.28E-06
Brake Pad Modelling-1	295	C3D8	2	165.791E-06
Brake Pad Modelling-1	295	C3D8	3	182.26E-06
Brake Pad Modelling-1	295	C3D8	4	169.625E-06
Brake Pad Modelling-1	295	C3D8	5	182.646E-06
Brake Pad Modelling-1	295	C3D8	6	170.193E-06
Brake Pad Modelling-1	295	C3D8	7	186.054E-06
Brake Pad Modelling-1	295	C3D8	8	173.429E-06
Brake Pad Modelling-1	610	C3D8	1	83.6971E-06
Brake Pad Modelling-1	610	C3D8	2	91.1193E-06
Brake Pad Modelling-1	610	C3D8	3	101.08E-06
Brake Pad Modelling-1	610	C3D8	4	108.587E-06
Brake Pad Modelling-1	610	C3D8	5	74.4448E-06
Brake Pad Modelling-1	610	C3D8	6	82.4387E-06
Brake Pad Modelling-1	610	C3D8	7	91.9072E-06
Brake Pad Modelling-1	610	C3D8	8	99.9863E-06
Brake Pad Modelling-1	593	C3D8	1	101.929E-06
Brake Pad Modelling-1	593	C3D8	2	109.888E-06
Brake Pad Modelling-1	593	C3D8	3	117.036E-06
Brake Pad Modelling-1	593	C3D8	4	125.204E-06
Brake Pad Modelling-1	593	C3D8	5	95.0208E-06
Brake Pad Modelling-1	593	C3D8	6	103.152E-06
Brake Pad Modelling-1	593	C3D8	7	110.234E-06
Brake Pad Modelling-1	593	C3D8	8	118.575E-06
Brake Pad Modelling-1	508	C3D8	1	140.238E-06
Brake Pad Modelling-1	508	C3D8	2	146.971E-06
Brake Pad Modelling-1	508	C3D8	3	146.807E-06
Brake Pad Modelling-1	508	C3D8	4	153.397E-06
Brake Pad Modelling-1	508	C3D8	5	141.889E-06

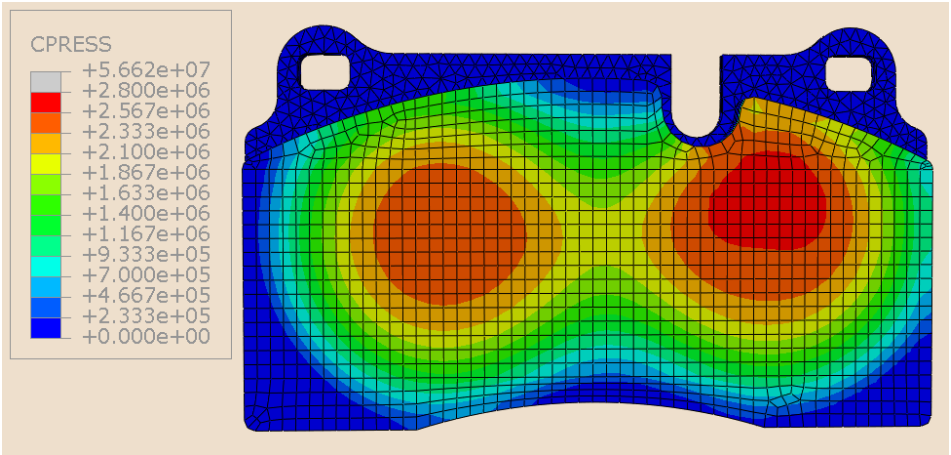
Brake Pad Modelling-1	508	C3D8	6	148.81E-06
Brake Pad Modelling-1	508	C3D8	7	148.457E-06
Brake Pad Modelling-1	508	C3D8	8	155.235E-06
Brake Pad Modelling-1	491	C3D8	1	141.939E-06
Brake Pad Modelling-1	491	C3D8	2	147.866E-06
Brake Pad Modelling-1	491	C3D8	3	146.552E-06
Brake Pad Modelling-1	491	C3D8	4	152.325E-06
Brake Pad Modelling-1	491	C3D8	5	144.447E-06
Brake Pad Modelling-1	491	C3D8	6	150.51E-06
Brake Pad Modelling-1	491	C3D8	7	149.057E-06
Brake Pad Modelling-1	491	C3D8	8	154.965E-06
Brake Pad Modelling-1	1089	C3D8	1	150.136E-06
Brake Pad Modelling-1	1089	C3D8	2	155.59E-06
Brake Pad Modelling-1	1089	C3D8	3	145.657E-06
Brake Pad Modelling-1	1089	C3D8	4	151.197E-06
Brake Pad Modelling-1	1089	C3D8	5	152.529E-06
Brake Pad Modelling-1	1089	C3D8	6	158.137E-06
Brake Pad Modelling-1	1089	C3D8	7	148.072E-06
Brake Pad Modelling-1	1089	C3D8	8	153.767E-06
Brake Pad Modelling-1	1072	C3D8	1	150.828E-06
Brake Pad Modelling-1	1072	C3D8	2	156.847E-06
Brake Pad Modelling-1	1072	C3D8	3	144.317E-06
Brake Pad Modelling-1	1072	C3D8	4	150.329E-06
Brake Pad Modelling-1	1072	C3D8	5	152.289E-06
Brake Pad Modelling-1	1072	C3D8	6	158.48E-06
Brake Pad Modelling-1	1072	C3D8	7	145.775E-06
Brake Pad Modelling-1	1072	C3D8	8	151.96E-06
Brake Pad Modelling-1	849	C3D8	1	107.953E-06
Brake Pad Modelling-1	849	C3D8	2	121.799E-06
Brake Pad Modelling-1	849	C3D8	3	115.58E-06
Brake Pad Modelling-1	849	C3D8	4	129.66E-06
Brake Pad Modelling-1	849	C3D8	5	101.223E-06
Brake Pad Modelling-1	849	C3D8	6	115.17E-06
Brake Pad Modelling-1	849	C3D8	7	108.928E-06
Brake Pad Modelling-1	849	C3D8	8	123.109E-06
Brake Pad Modelling-1	848	C3D8	1	92.2617E-06
Brake Pad Modelling-1	848	C3D8	2	107.358E-06
Brake Pad Modelling-1	848	C3D8	3	99.9513E-06
Brake Pad Modelling-1	848	C3D8	4	115.199E-06
Brake Pad Modelling-1	848	C3D8	5	84.8129E-06
Brake Pad Modelling-1	848	C3D8	6	100.004E-06
Brake Pad Modelling-1	848	C3D8	7	92.6627E-06
Brake Pad Modelling-1	848	C3D8	8	108.005E-06
Brake Pad Modelling-1	1049	C3D8	1	129.678E-06

Brake Pad Modelling-1	1049	C3D8	2	110.827E-06
Brake Pad Modelling-1	1049	C3D8	3	126.66E-06
Brake Pad Modelling-1	1049	C3D8	4	107.741E-06
Brake Pad Modelling-1	1049	C3D8	5	122.789E-06
Brake Pad Modelling-1	1049	C3D8	6	104.078E-06
Brake Pad Modelling-1	1049	C3D8	7	119.616E-06
Brake Pad Modelling-1	1049	C3D8	8	100.837E-06
Brake Pad Modelling-1	1048	C3D8	1	146.328E-06
Brake Pad Modelling-1	1048	C3D8	2	129.494E-06
Brake Pad Modelling-1	1048	C3D8	3	143.348E-06
Brake Pad Modelling-1	1048	C3D8	4	126.409E-06
Brake Pad Modelling-1	1048	C3D8	5	141.741E-06
Brake Pad Modelling-1	1048	C3D8	6	125.039E-06
Brake Pad Modelling-1	1048	C3D8	7	138.566E-06
Brake Pad Modelling-1	1048	C3D8	8	121.759E-06
Brake Pad Modelling-1	1233	C3D8	1	97.1478E-06
Brake Pad Modelling-1	1233	C3D8	2	99.5867E-06
Brake Pad Modelling-1	1233	C3D8	3	99.5775E-06
Brake Pad Modelling-1	1233	C3D8	4	102.028E-06
Brake Pad Modelling-1	1233	C3D8	5	98.6602E-06
Brake Pad Modelling-1	1233	C3D8	6	101.227E-06
Brake Pad Modelling-1	1233	C3D8	7	101.048E-06
Brake Pad Modelling-1	1233	C3D8	8	103.626E-06
Brake Pad Modelling-1	1225	C3D8	1	109.047E-06
Brake Pad Modelling-1	1225	C3D8	2	111.427E-06
Brake Pad Modelling-1	1225	C3D8	3	112.28E-06
Brake Pad Modelling-1	1225	C3D8	4	114.279E-06
Brake Pad Modelling-1	1225	C3D8	5	111.216E-06
Brake Pad Modelling-1	1225	C3D8	6	113.65E-06
Brake Pad Modelling-1	1225	C3D8	7	114.36E-06
Brake Pad Modelling-1	1225	C3D8	8	116.413E-06
Brake Pad Modelling-1	1187	C3D8	1	191.292E-06
Brake Pad Modelling-1	1187	C3D8	2	192.003E-06
Brake Pad Modelling-1	1187	C3D8	3	196.479E-06
Brake Pad Modelling-1	1187	C3D8	4	195.804E-06
Brake Pad Modelling-1	1187	C3D8	5	194.946E-06
Brake Pad Modelling-1	1187	C3D8	6	194.804E-06
Brake Pad Modelling-1	1187	C3D8	7	200.686E-06
Brake Pad Modelling-1	1187	C3D8	8	199.186E-06
Brake Pad Modelling-1	1194	C3D8	1	201.23E-06
Brake Pad Modelling-1	1194	C3D8	2	212.278E-06
Brake Pad Modelling-1	1194	C3D8	3	200.152E-06
Brake Pad Modelling-1	1194	C3D8	4	210.633E-06
Brake Pad Modelling-1	1194	C3D8	5	200.608E-06
Brake Pad Modelling-1	1194	C3D8	6	212.654E-06
Brake Pad Modelling-1	1194	C3D8	7	200.491E-06
Brake Pad Modelling-1	1194	C3D8	8	211.898E-06

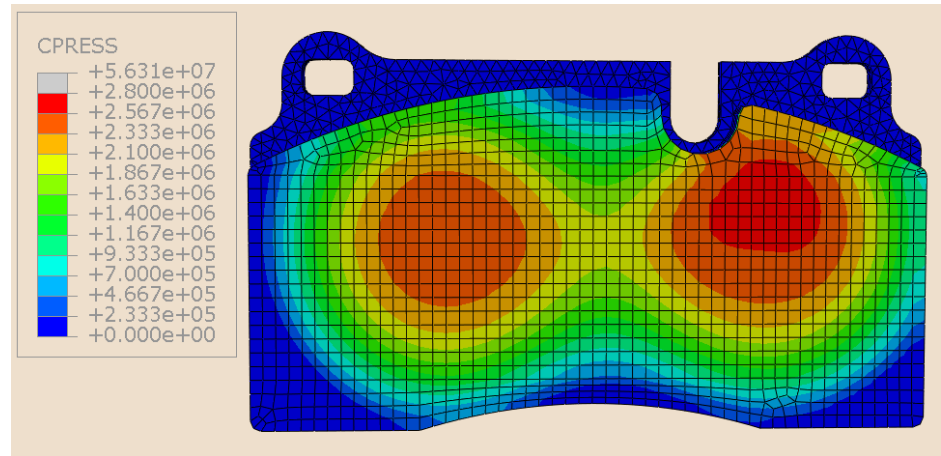
Appendix Chapter 5-C: FEA Results for Different Friction Coefficients (CPRESS, E22)



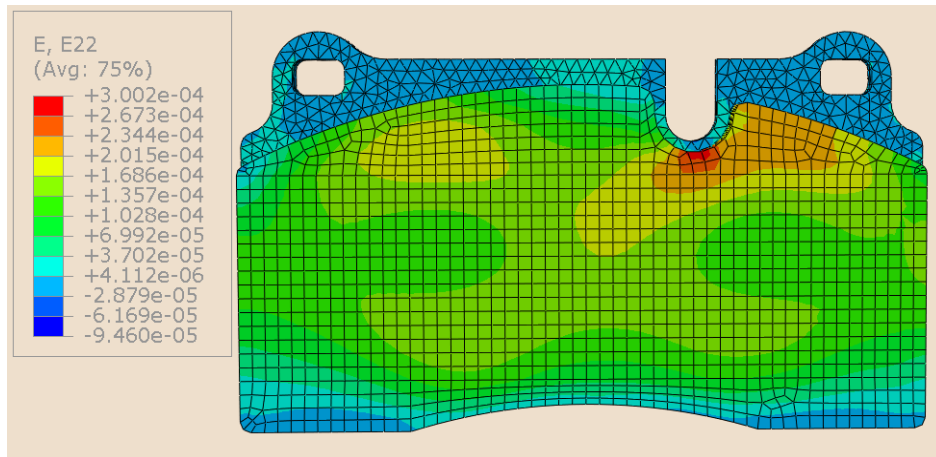
(a) Friction Coefficient 0.2: CPRESS



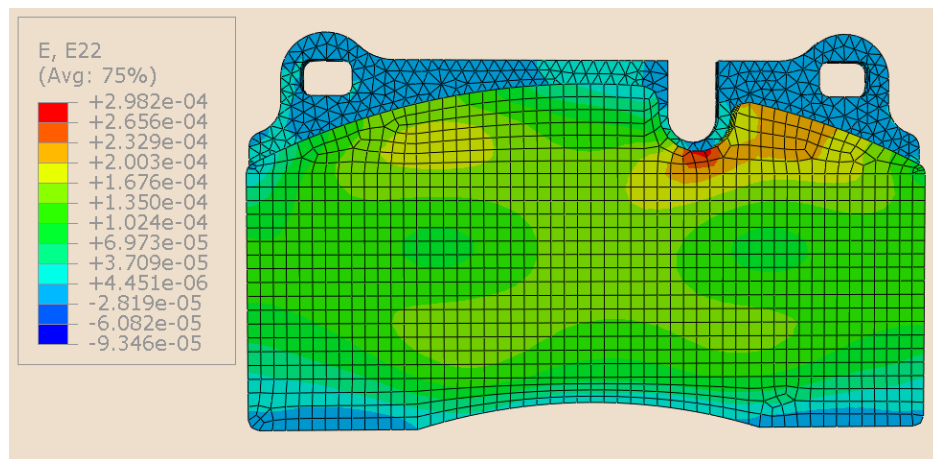
(b) Friction Coefficient 0.3: CPRESS



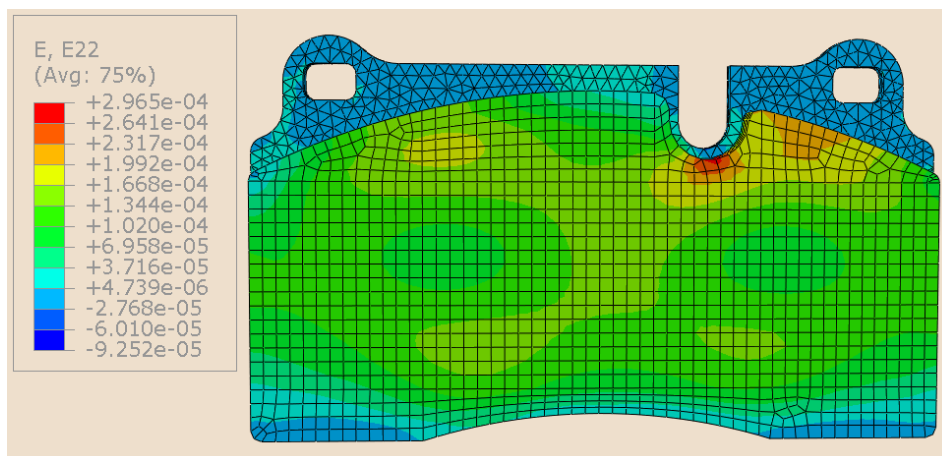
(c) Friction Coefficient 0.4: CPRESS



(d) Friction Coefficient 0.2: E22

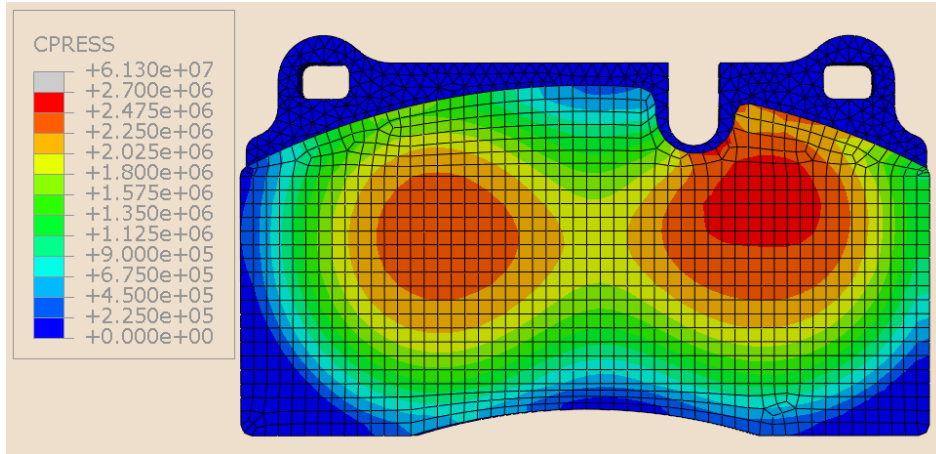


(e) Friction Coefficient 0.3: E22

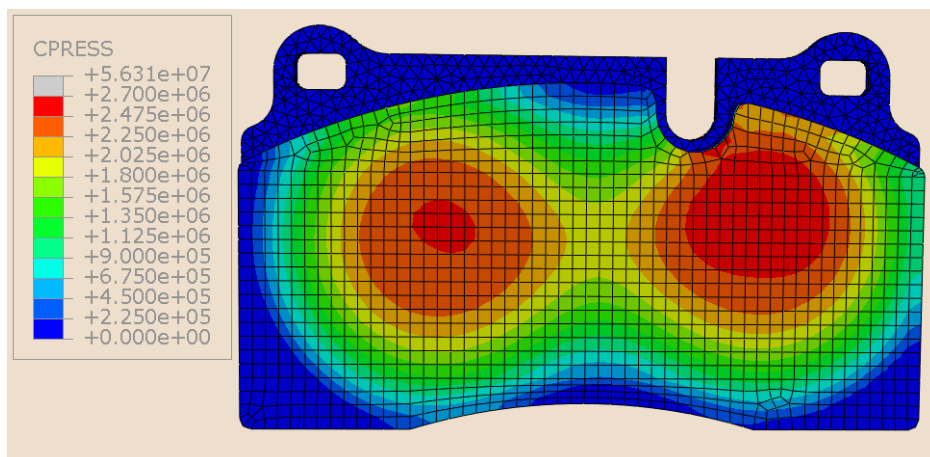


(f) Friction Coefficient 0.4: E22

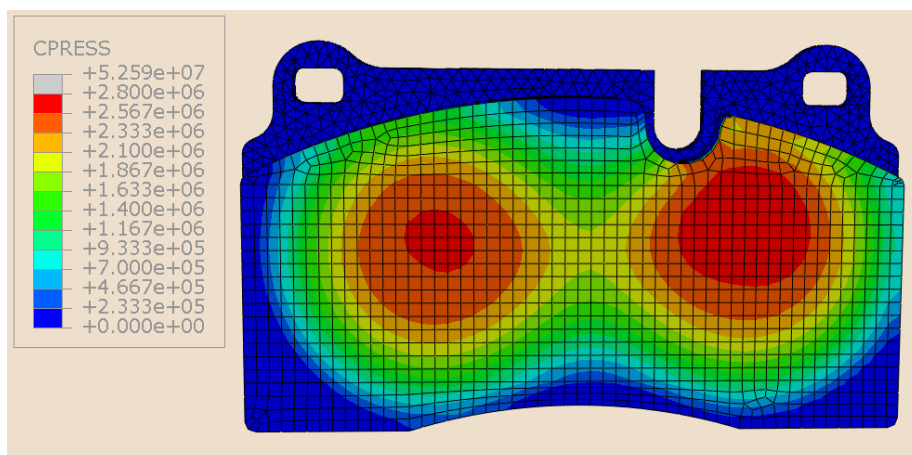
Appendix Chapter 5-D: FEA Results for Different Young's Modulus Values (CPRESS, E22)



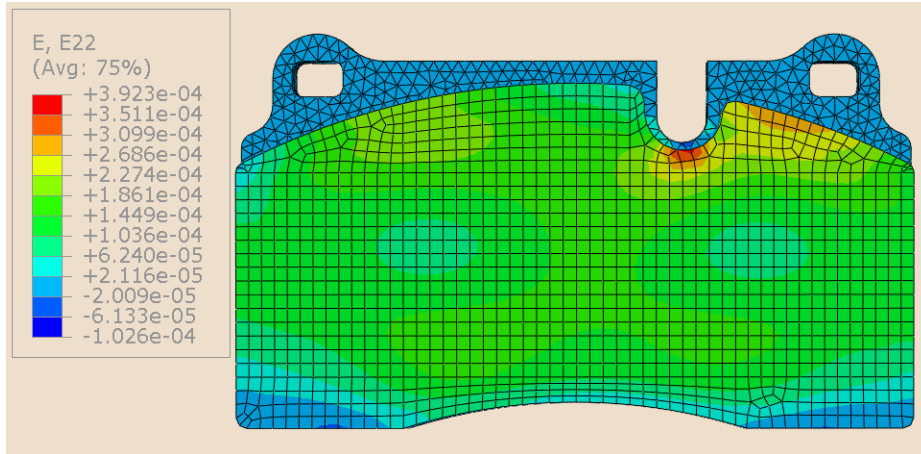
(a) Young's Modulus 1.95GPa (-25%): CPRESS



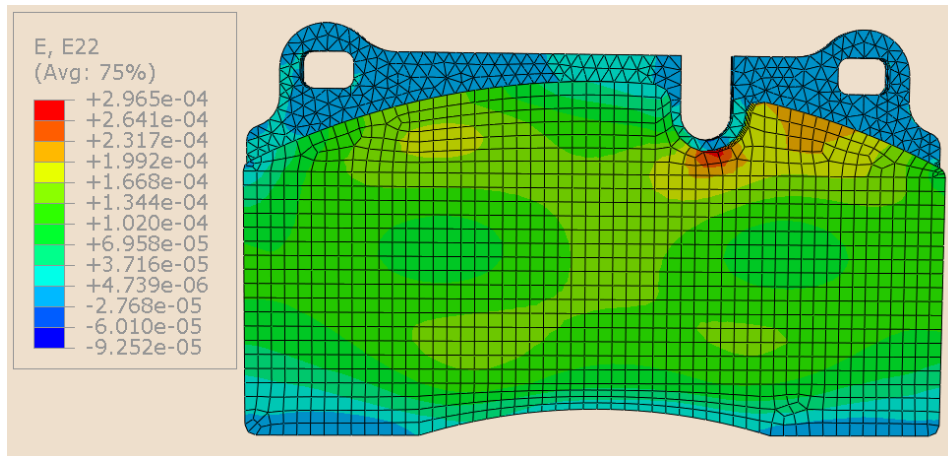
(b) Young's Modulus 2.6Gpa (Nominal): CPRESS



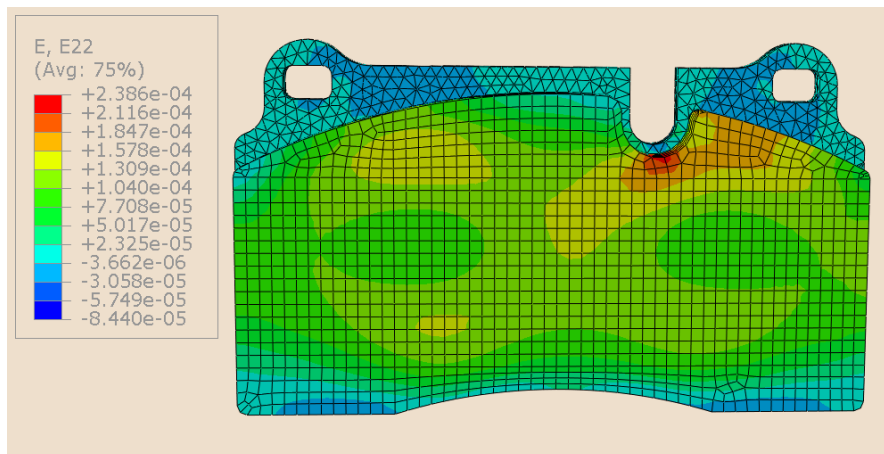
(c) Young's Modulus 3.25GPa (+25%): CPRESS



(d) Young's Modulus 1.95GPa (-25%): E22



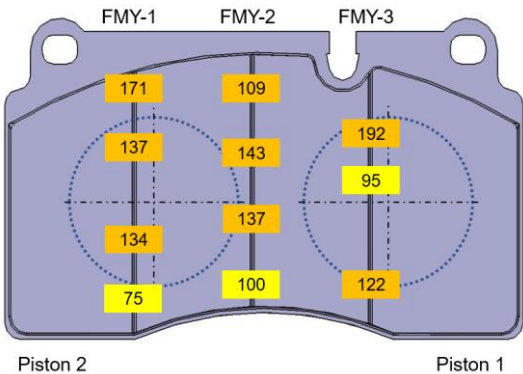
(e) Young's Modulus 2.6GPa (Nominal): E22



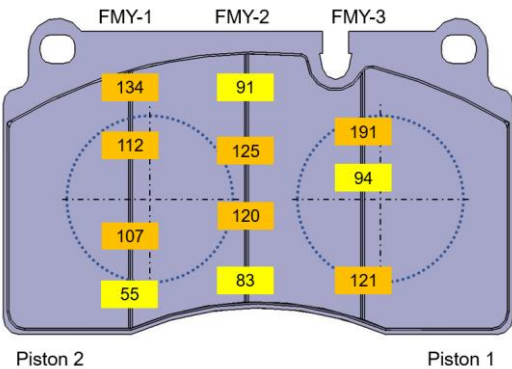
(f) Young's Modulus 3.25GPa (+25%): E22

Appendix Chapter 5-E: Processed FEA Contour Map in Vertical (Y) and Horizontal (X) Directions for Three Hydraulic Pressure Variations (40bar)

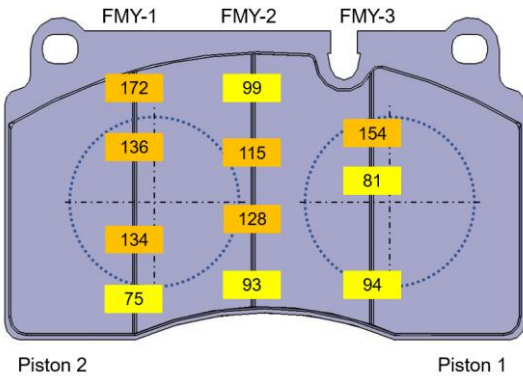
Strain (μϵ)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



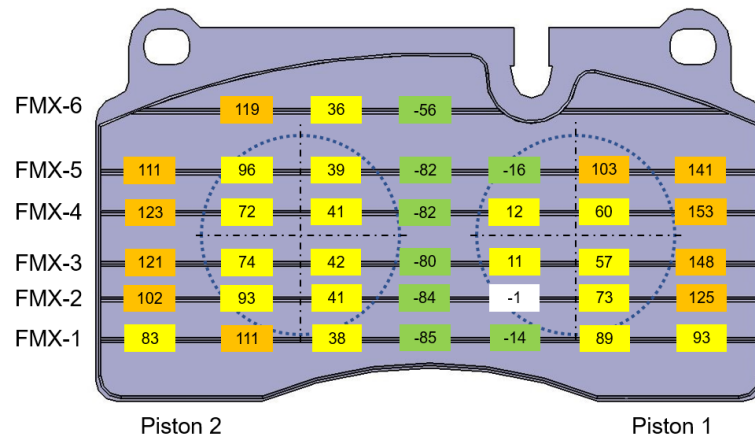
(a) E22: Equal Pressure ($p_1=p_2=40\text{bar}$)



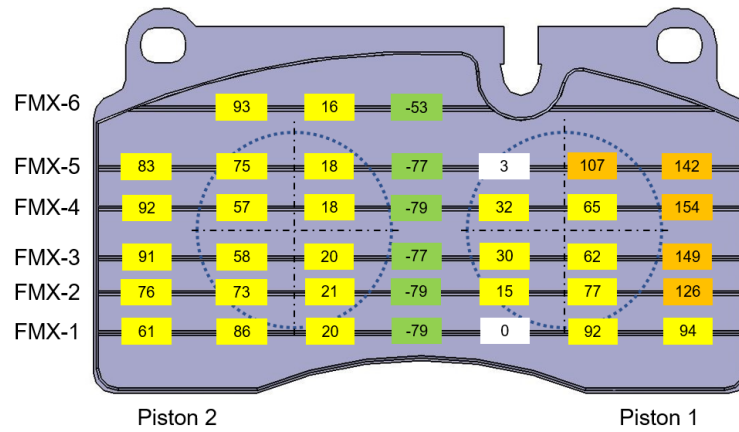
(b) E22: +10bar ($p_1=40\text{bar}$; $p_2=30\text{bar}$)



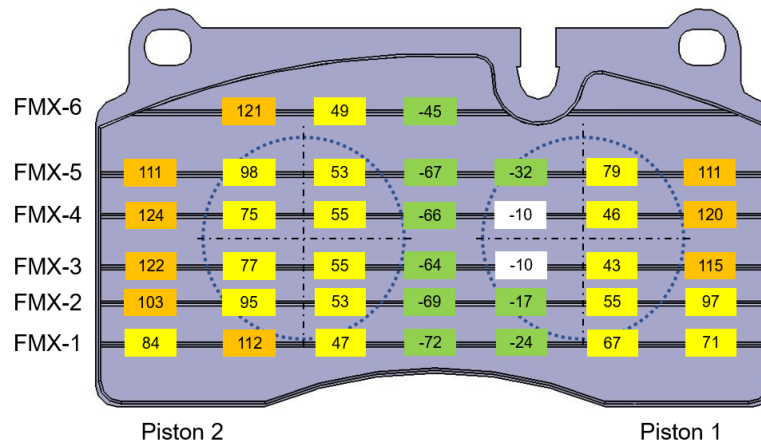
(c) E22: -10bar ($p_1=30\text{bar}$; $p_2=40\text{bar}$)



(d) E11: Equal Pressure ($p_1=p_2=40\text{bar}$)



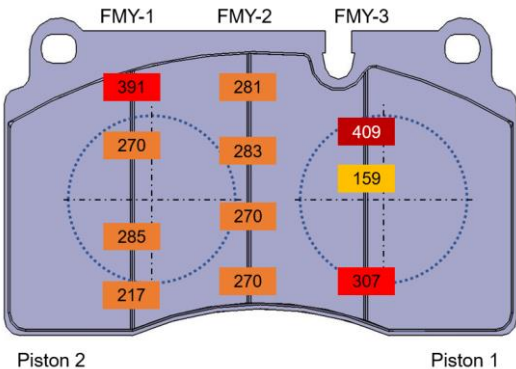
(e) E11: +10bar ($p_1=40\text{bar}$; $p_2=30\text{bar}$)



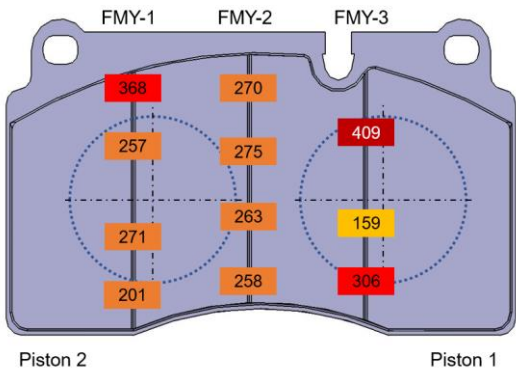
(f) E11: -10bar ($p_1=30\text{bar}$; $p_2=40\text{bar}$)

Appendix Chapter 5-F: Processed FEA Contour Map in Vertical (Y) and Horizontal (X) Directions for Three Hydraulic Pressure Variations (120bar)

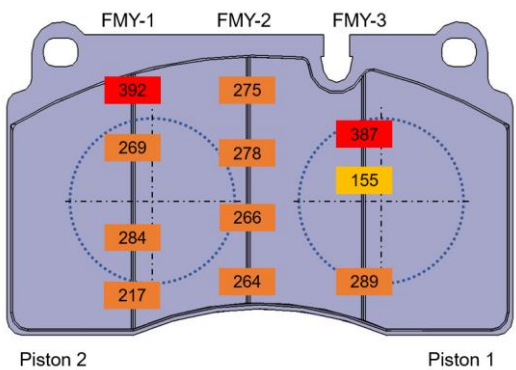
Strain (μɛ)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



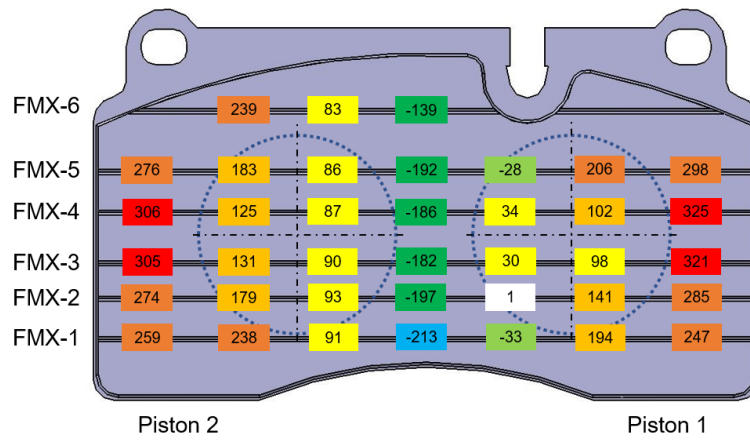
(a) E22: Equal Pressure (p1=p2=120bar)



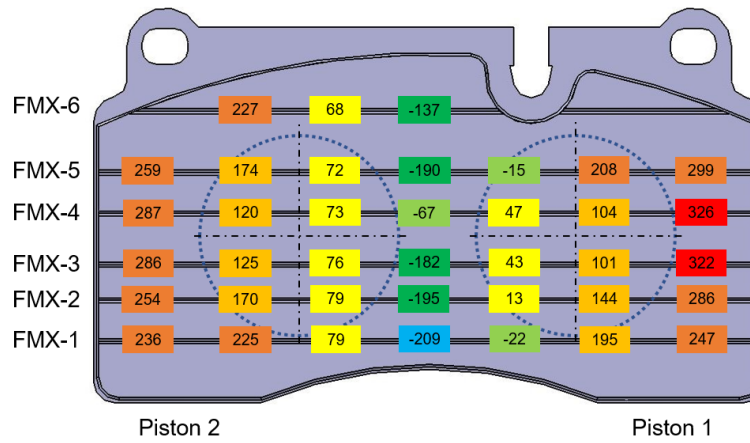
(b) E22: +10bar (p1=120bar; p2=110bar)



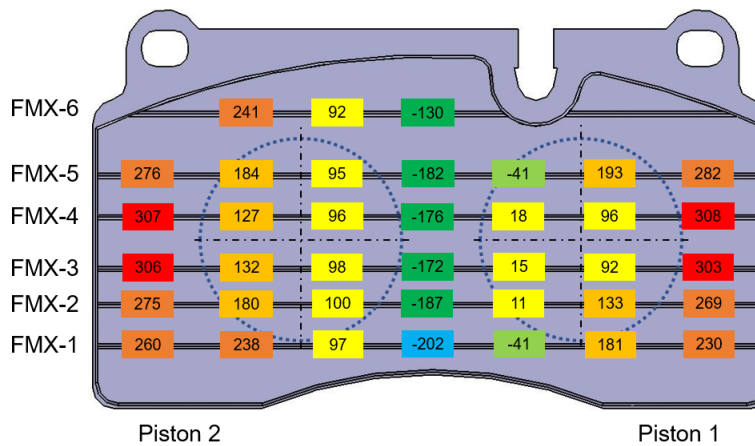
(c) E22: -10bar (p1=110bar; p2=120bar)



(d) E11: Equal Pressure ($p_1=p_2=120\text{bar}$)



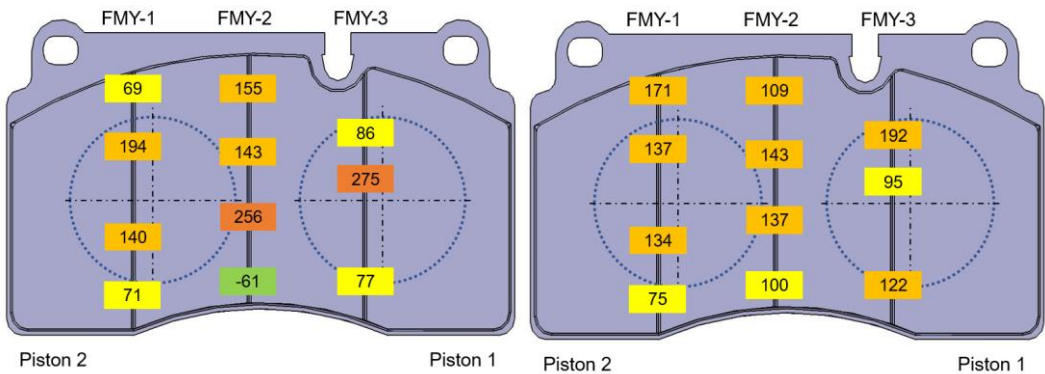
(e) E11: +10bar ($p_1=120\text{bar}$; $p_2=110\text{bar}$)



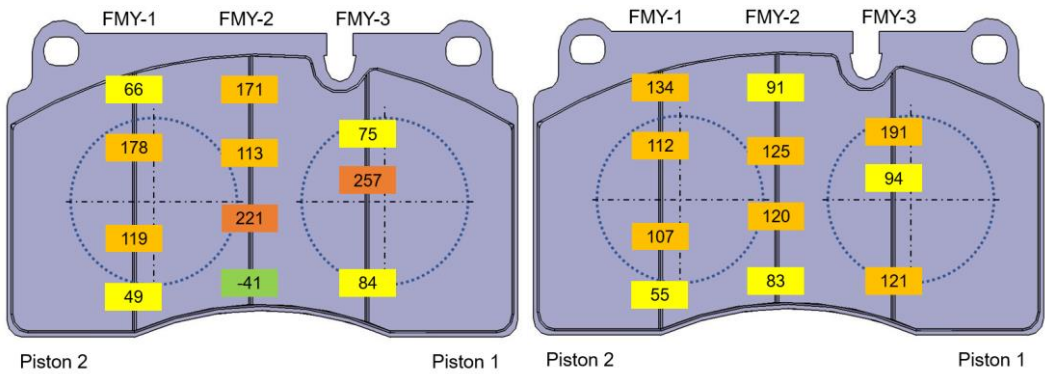
(f) E11: -10bar ($p_1=110\text{bar}$; $p_2=120\text{bar}$)

Appendix Chapter 5-G: E11 and E22 Strain Results and Differences for Three Hydraulic Pressure Variations (40bar)

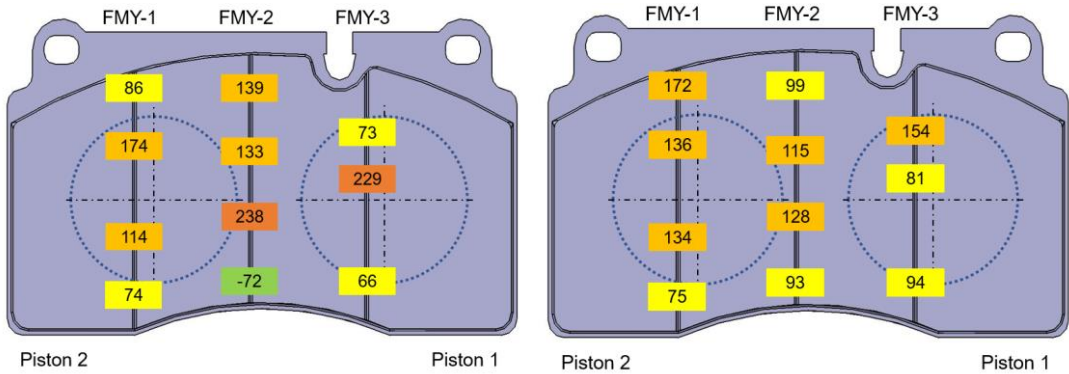
Strain (μϵ)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



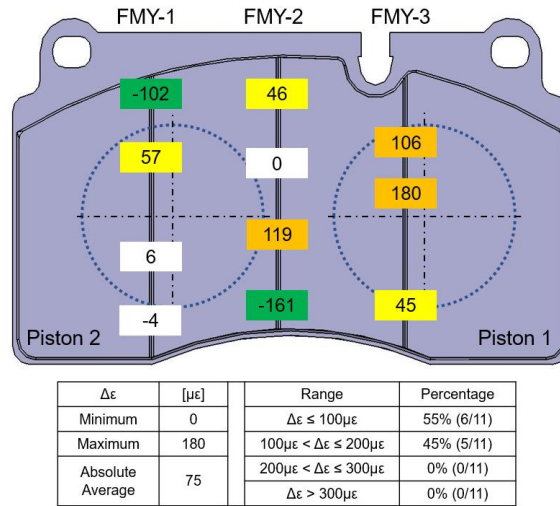
(a) E22: FBG Result (Left) and FEA Result (Right) at $p_1=p_2=40\text{bar}$



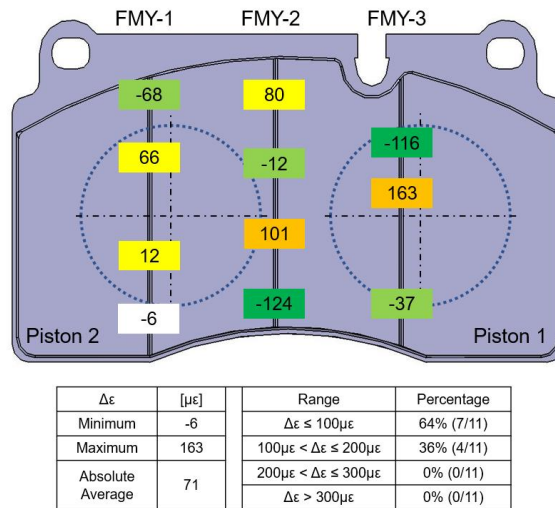
(b) E22: FBG Result (Left) and FEA Result (Right) at $p_1=40\text{bar}$ and $p_2=30\text{bar}$



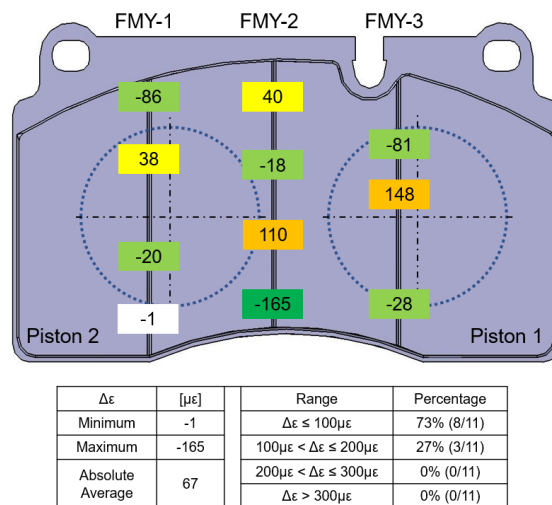
(c) E22: FBG Result (Left) and FEA Result (Right) at $p_1=30\text{bar}$ and $p_2=40\text{bar}$



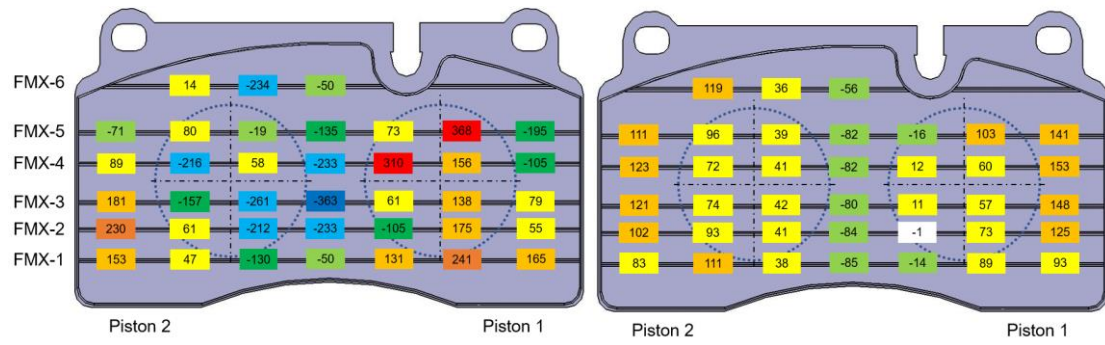
(d) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=p2=40bar



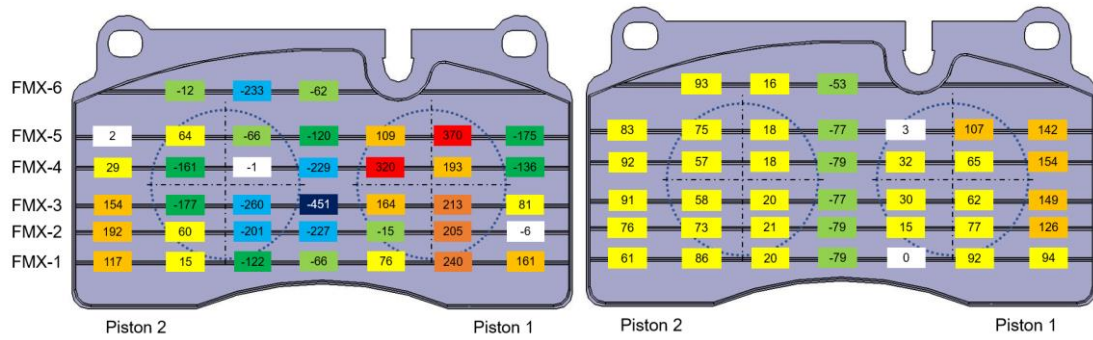
(e) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=40bar; p2=30bar



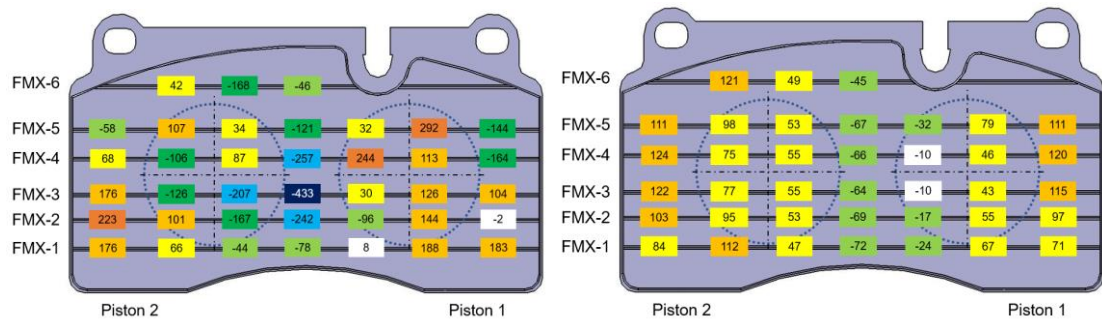
(f) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=30bar; p2=40bar



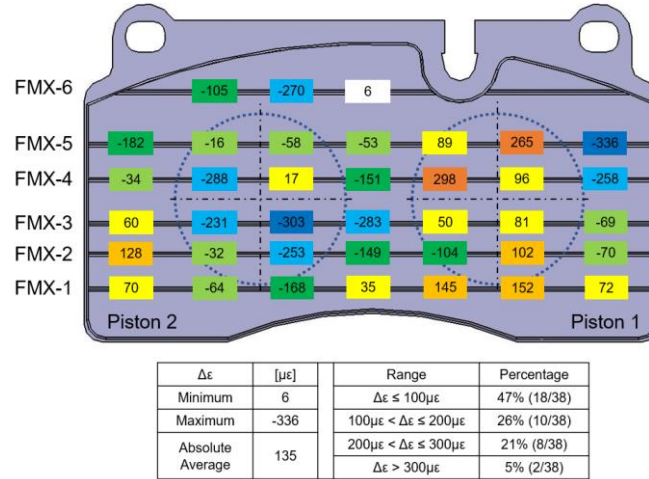
(g) E11: FBG Result (Left) and FEA Result (Right) at $p_1=p_2=40\text{bar}$



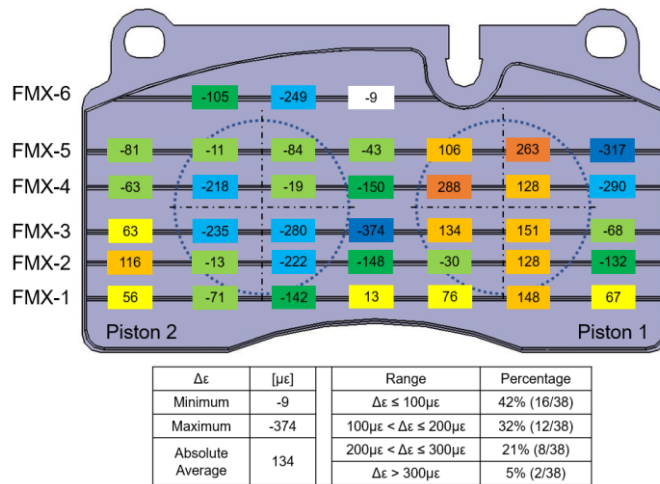
(h) E11: FBG Result (Left) and FEA Result (Right) at $p_1=40\text{bar}$ and $p_2=30\text{bar}$



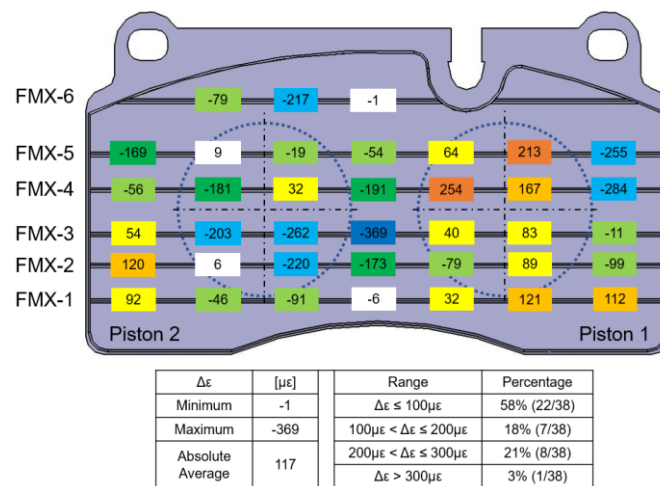
(i) E11: FBG Result (Left) and FEA Result (Right) at $p_1=30\text{bar}$ and $p_2=40\text{bar}$



(j) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=p2=40bar



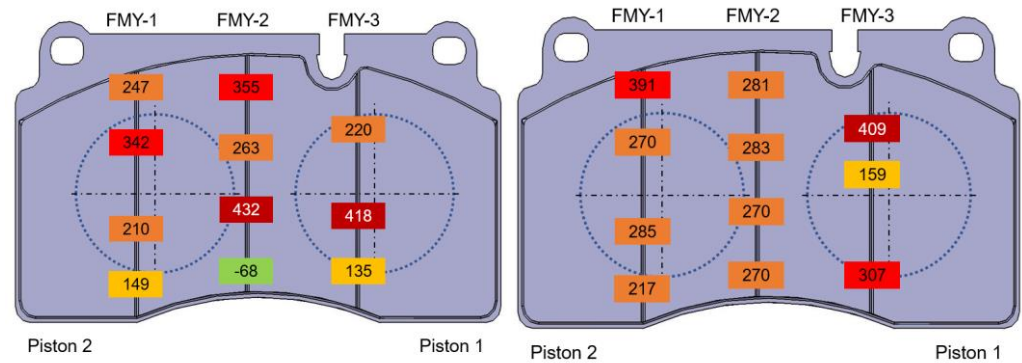
(k) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=40bar; p2=30bar



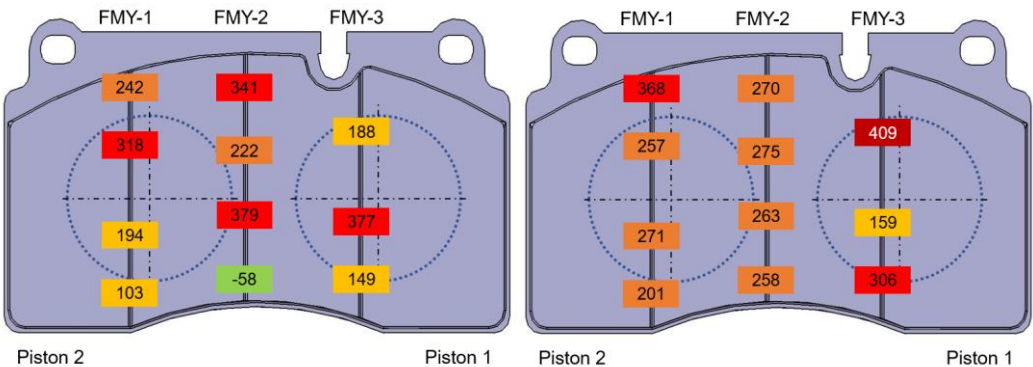
(l) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=30bar; p2=40bar

Appendix Chapter 5-H: E11 and E22 Strain Results and Differences for Three Hydraulic Pressure Variations (120bar)

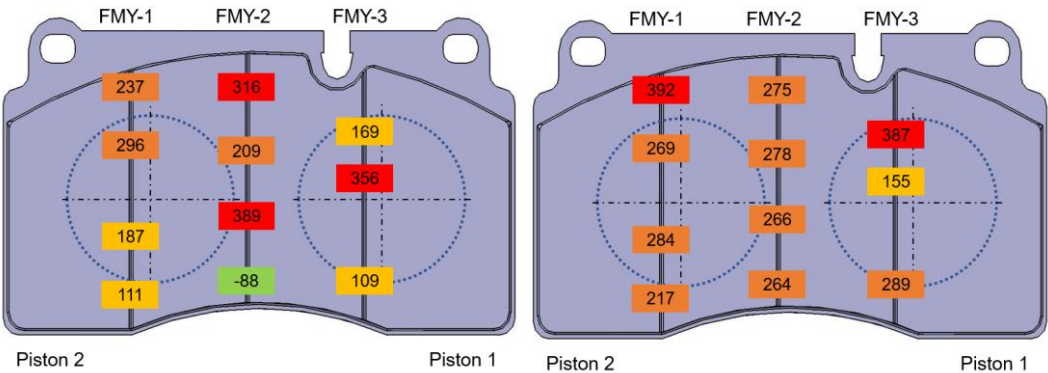
Strain (μϵ)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



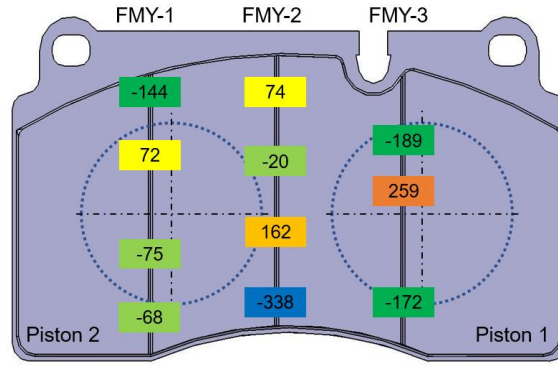
(a) E22: FBG Result (Left) and FEA Result (Right) at p1=p2=120bar



(b) E22: FBG Result (Left) and FEA Result (Right) at p1=120bar and p2=110bar

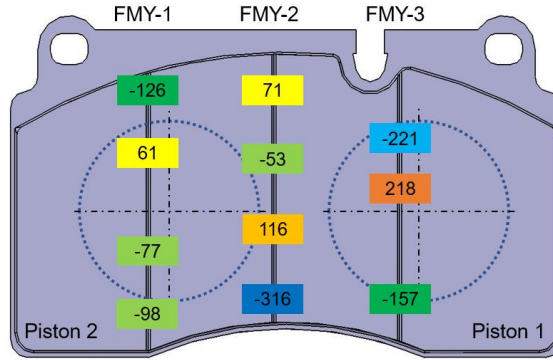


(c) E22: FBG Result (Left) and FEA Result (Right) at p1=110bar and p2=120bar



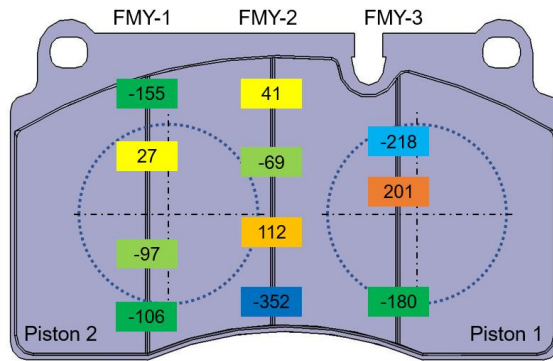
$\Delta\epsilon$	$[\mu\epsilon]$	Range	Percentage
Minimum	20	$\Delta\epsilon \leq 100\mu\epsilon$	45% (5/11)
Maximum	-338	$100\mu\epsilon < \Delta\epsilon \leq 200\mu\epsilon$	36% (4/11)
Absolute Average	143	$200\mu\epsilon < \Delta\epsilon \leq 300\mu\epsilon$	9% (1/11)
		$\Delta\epsilon > 300\mu\epsilon$	9% (1/11)

(d) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=p2=120bar



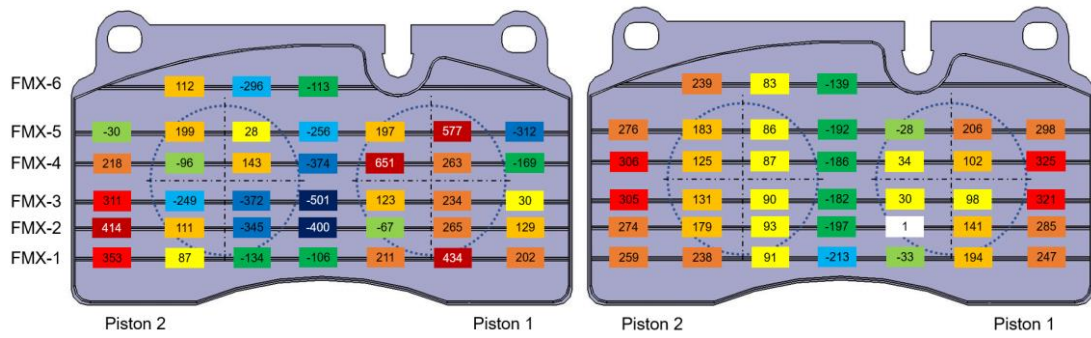
$\Delta\epsilon$	$[\mu\epsilon]$	Range	Percentage
Minimum	-53	$\Delta\epsilon \leq 100\mu\epsilon$	45% (5/11)
Maximum	-316	$100\mu\epsilon < \Delta\epsilon \leq 200\mu\epsilon$	27% (3/11)
Absolute Average	138	$200\mu\epsilon < \Delta\epsilon \leq 300\mu\epsilon$	18% (2/11)
		$\Delta\epsilon > 300\mu\epsilon$	9% (1/11)

(e) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=120bar; p2=110bar

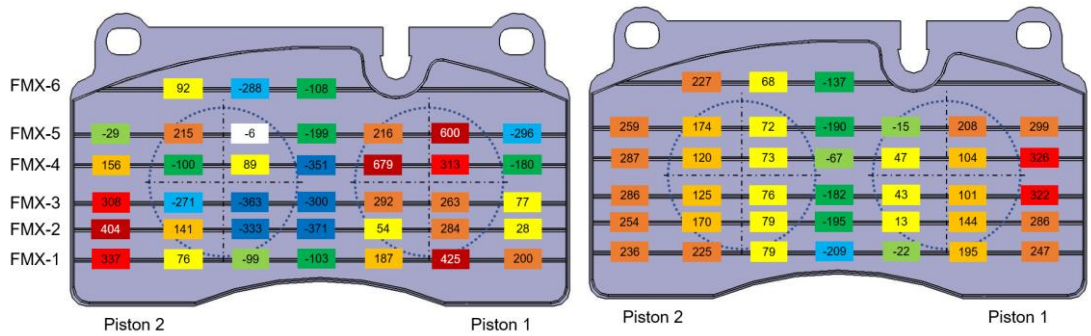


$\Delta\epsilon$	$[\mu\epsilon]$	Range	Percentage
Minimum	27	$\Delta\epsilon \leq 100\mu\epsilon$	36% (4/11)
Maximum	-352	$100\mu\epsilon < \Delta\epsilon \leq 200\mu\epsilon$	36% (4/11)
Absolute Average	143	$200\mu\epsilon < \Delta\epsilon \leq 300\mu\epsilon$	18% (2/11)
		$\Delta\epsilon > 300\mu\epsilon$	9% (1/11)

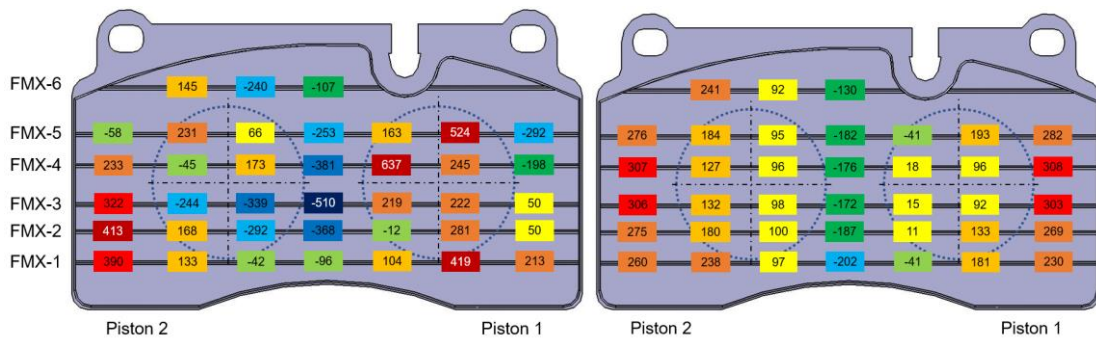
(f) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E22: p1=110bar; p2=120bar



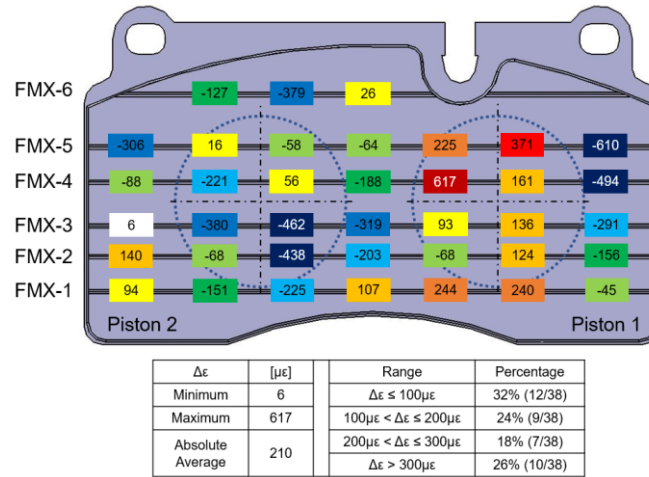
(g) E11: FBG Result (Left) and FEA Result (Right) at $p_1=p_2=120\text{bar}$



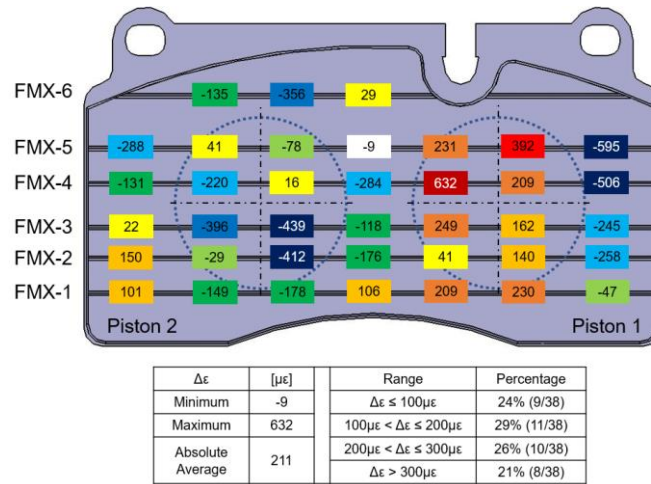
(h) E11: FBG Result (Left) and FEA Result (Right) at $+p_1=120\text{bar}$ and $p_2=110\text{bar}$



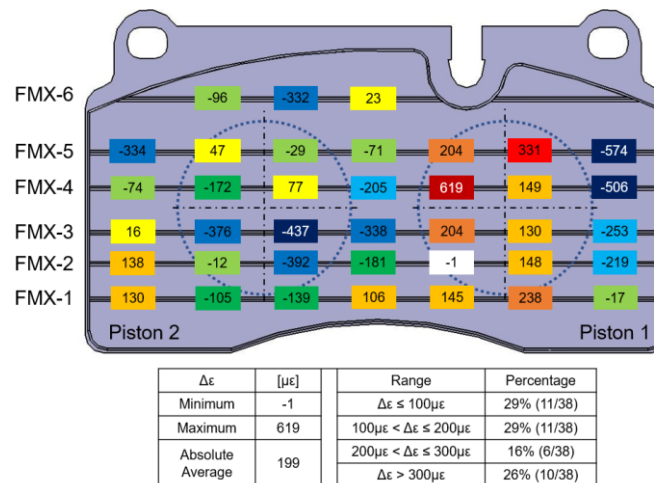
(i) E11: FBG Result (Left) and FEA Result (Right) at $p_1=110\text{bar}$ and $p_2=120\text{bar}$



(j) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=p2=120bar



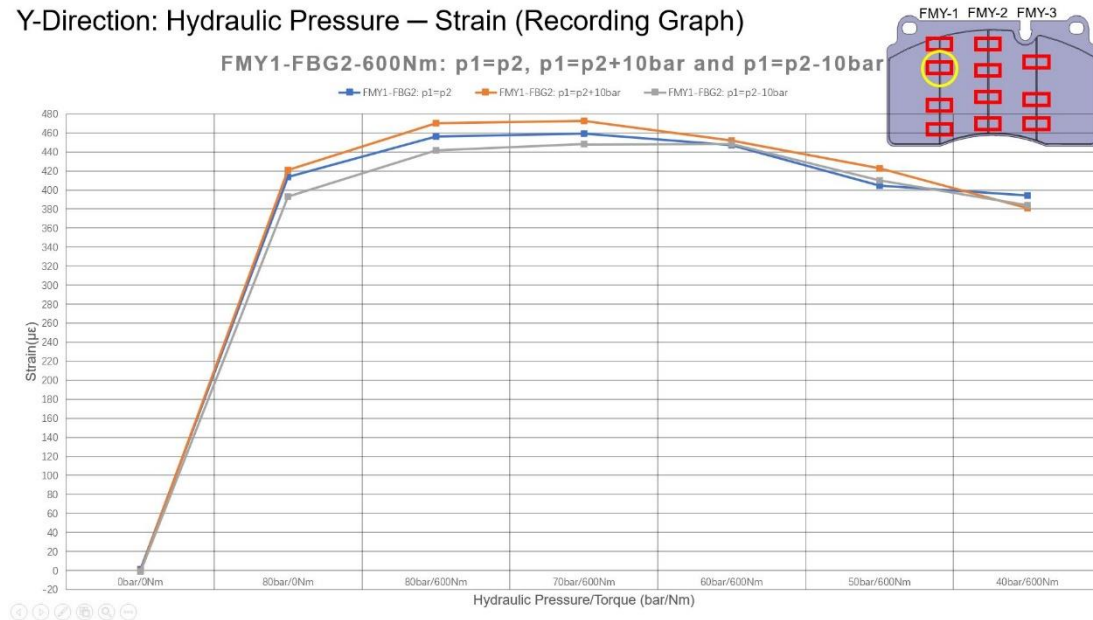
(k) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=120bar; p2=110bar



(l) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: E11: p1=110bar; p2=120bar

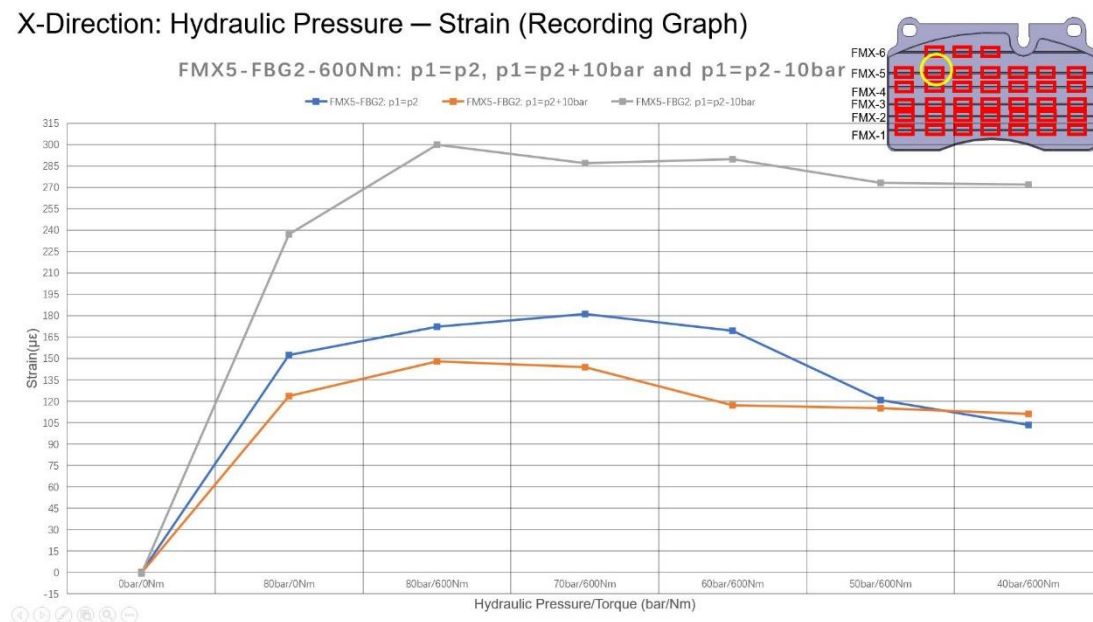
Appendix Chapter 6-A: Recording and Distribution Graphs at the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface with 600Nm Toque

Y-Direction: Hydraulic Pressure – Strain (Recording Graph)



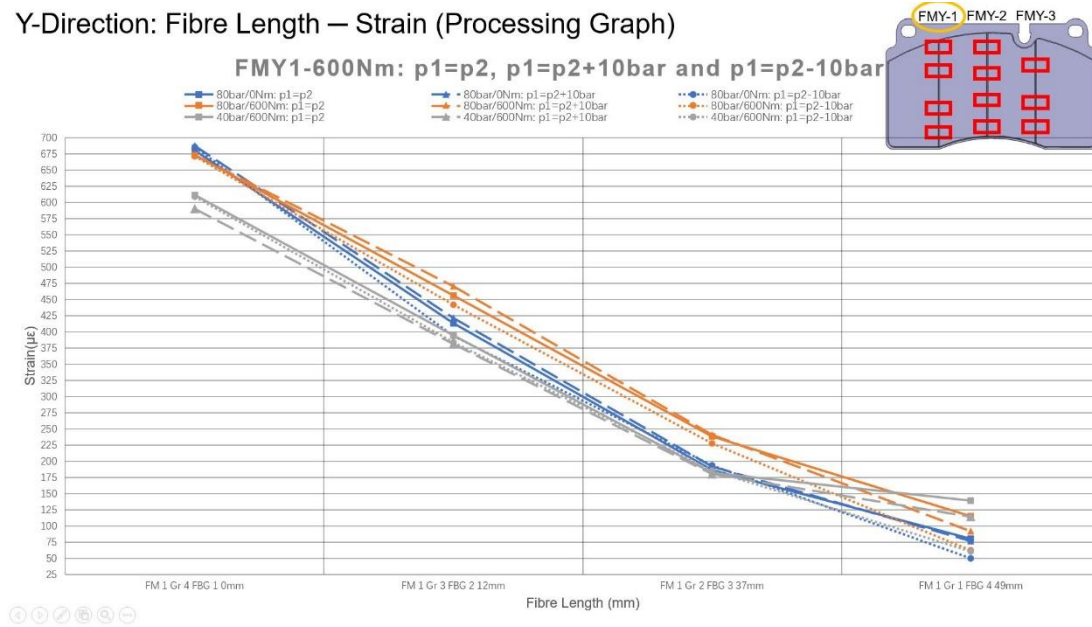
(a) Vertical (Y) Direction: Hydraulic Pressure – Strain (FMY1-FBG2)

X-Direction: Hydraulic Pressure – Strain (Recording Graph)



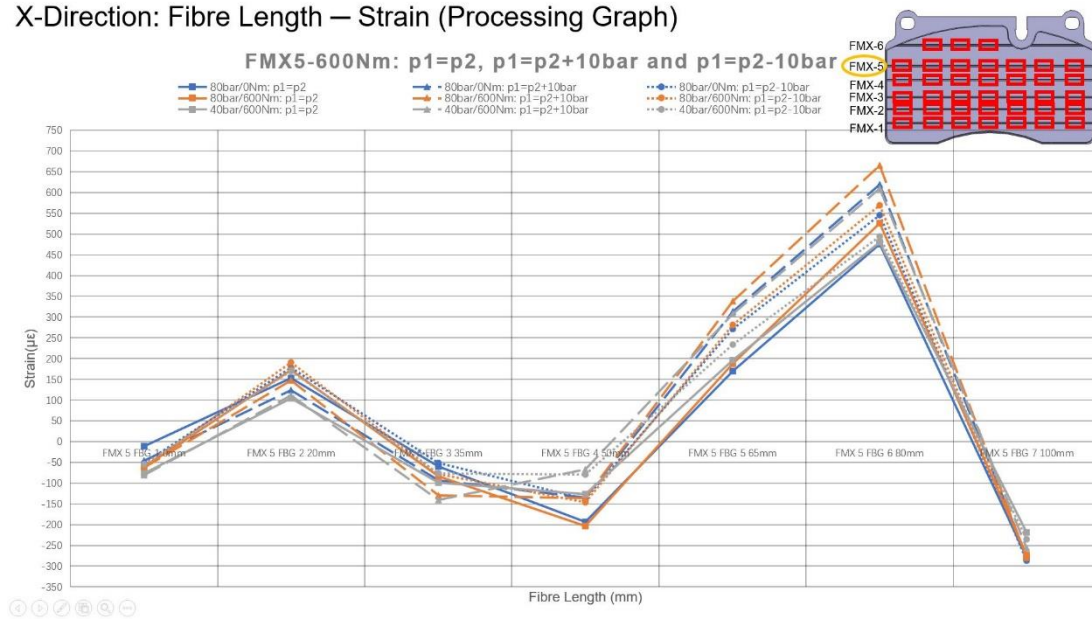
(b) Horizontal (X) Direction: Hydraulic Pressure – Strain (FMX5-FBG2)

Y-Direction: Fibre Length — Strain (Processing Graph)



(c) Vertical (Y) Direction: Fibre Length — Strain (FMY1)

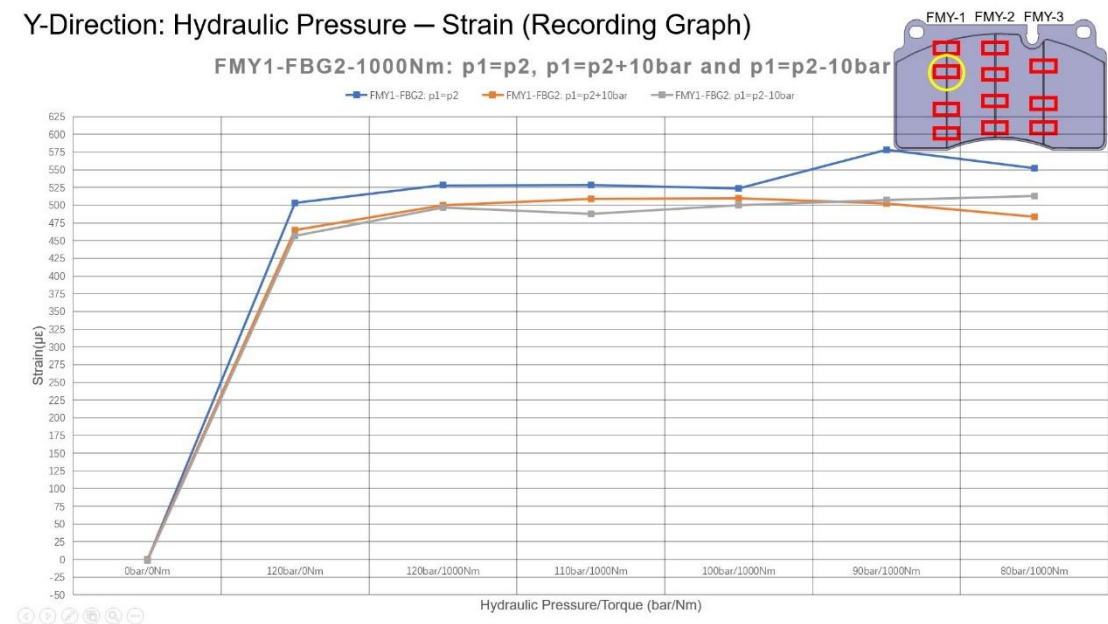
X-Direction: Fibre Length — Strain (Processing Graph)



(d) Horizontal (X) Direction: Fibre Length — Strain (FMX5)

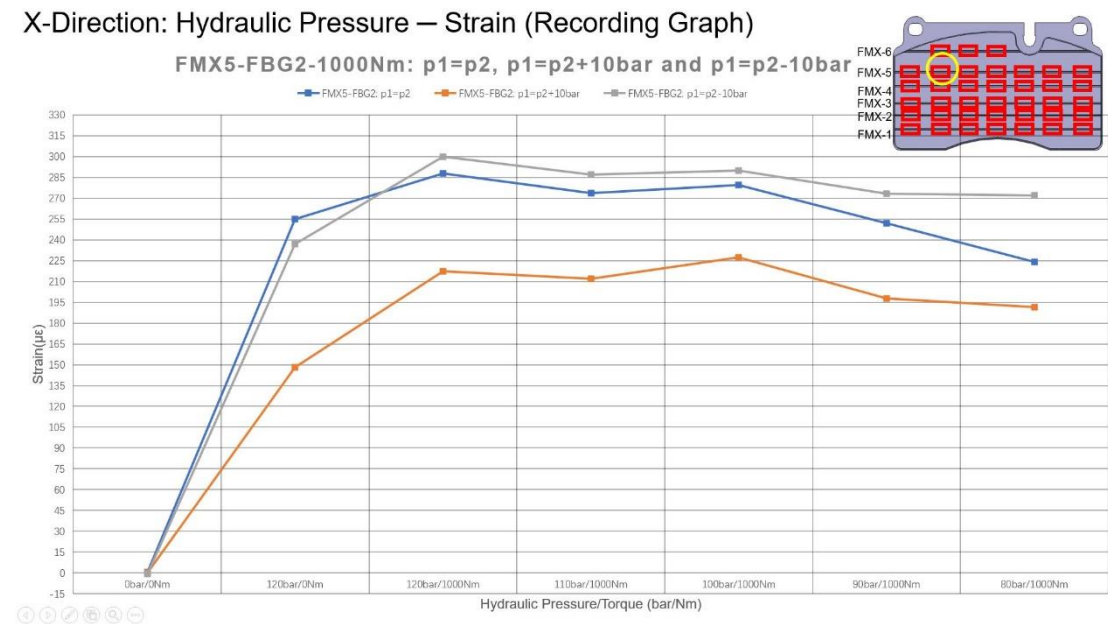
Appendix Chapter 6-B: Recording and Distribution Graphs at the Vertical (Y) and Horizontal (X) Directions on the Brake Pad Friction Surface with 1000Nm Toque

Y-Direction: Hydraulic Pressure – Strain (Recording Graph)



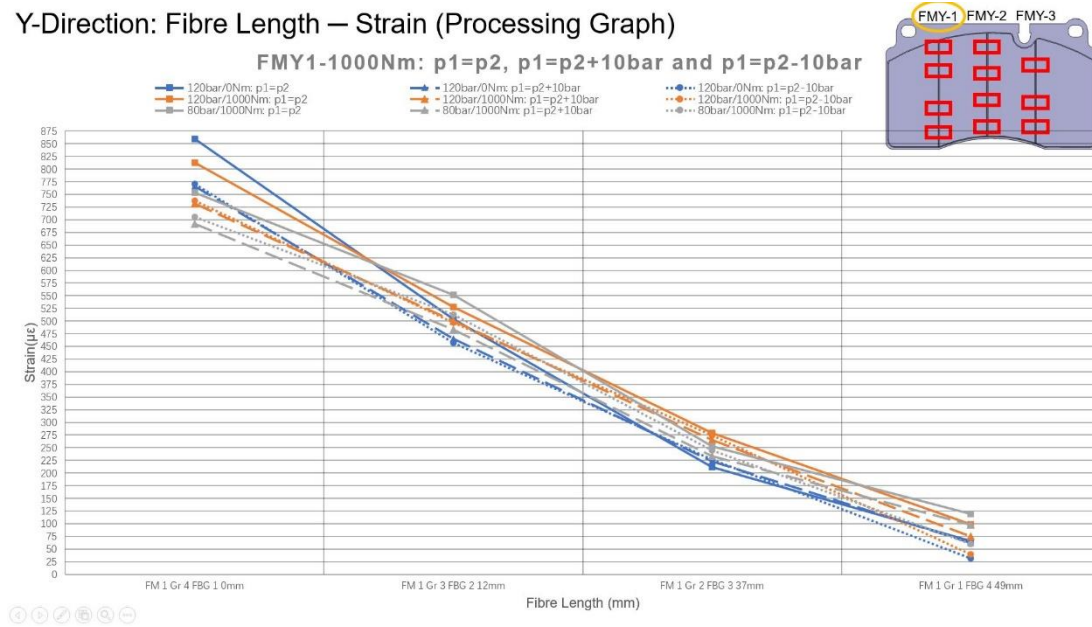
(a) Vertical (Y) Direction: Hydraulic Pressure – Strain (FMY1-FBG2)

X-Direction: Hydraulic Pressure – Strain (Recording Graph)



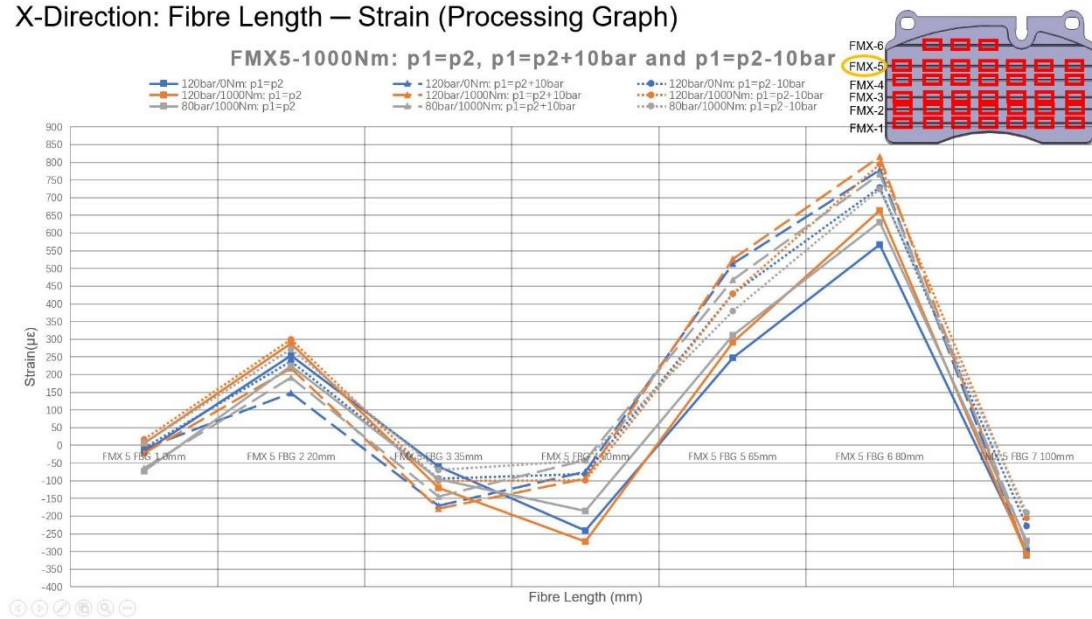
(b) Horizontal (X) Direction: Hydraulic Pressure – Strain (FMX5-FBG2)

Y-Direction: Fibre Length — Strain (Processing Graph)



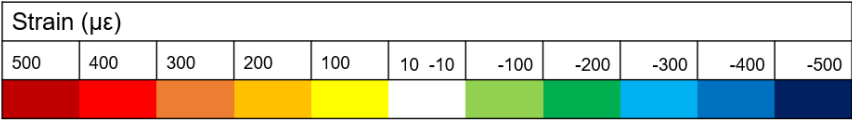
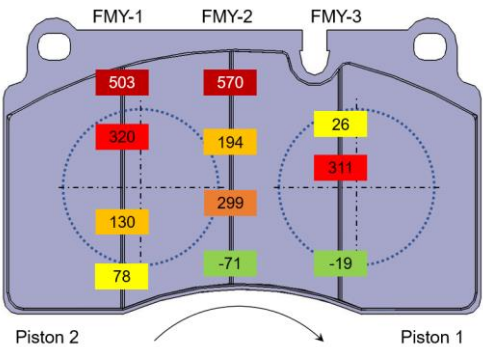
(c) Vertical (Y) Direction: Fibre Length — Strain (FMY1)

X-Direction: Fibre Length — Strain (Processing Graph)

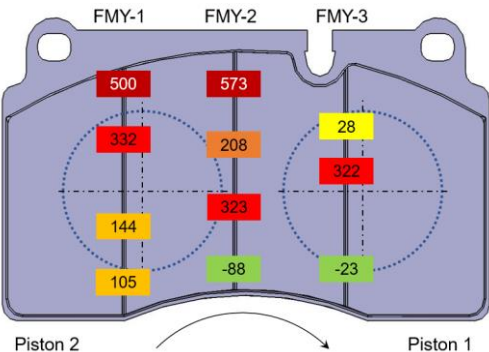


(d) Horizontal (X) Direction: Fibre Length — Strain (FMX5)

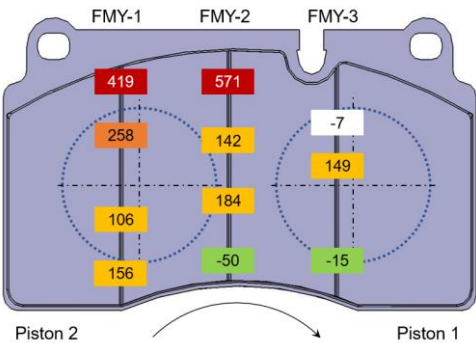
Appendix Chapter 6-C: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions between Three Test Stages with Equal Pressure (p1 = p2)



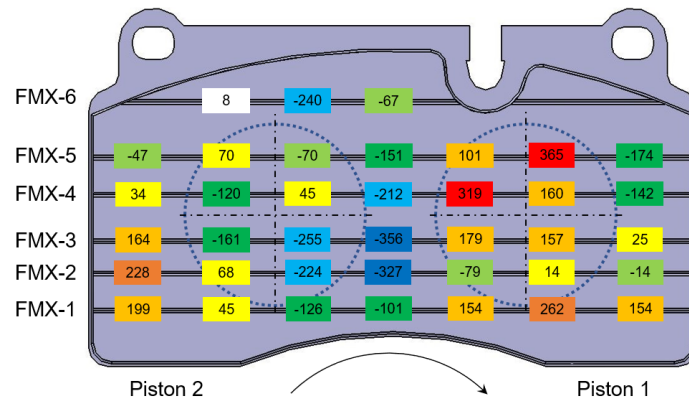
(a) Vertical (Y) Direction: 40bar Hydraulic Pressure with 0Nm Torque Level



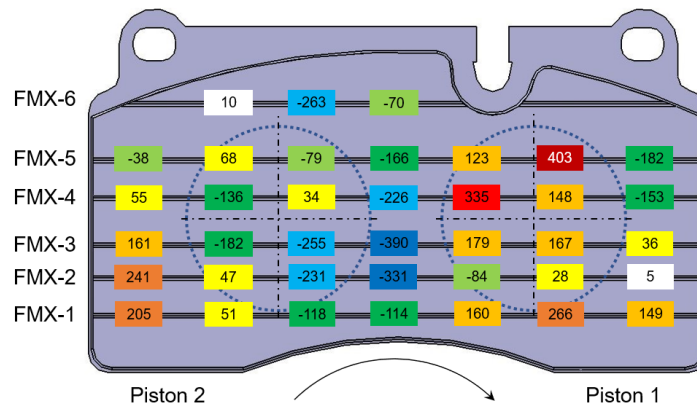
(b) Vertical (Y) Direction: 40bar Hydraulic Pressure with 200Nm Torque Level



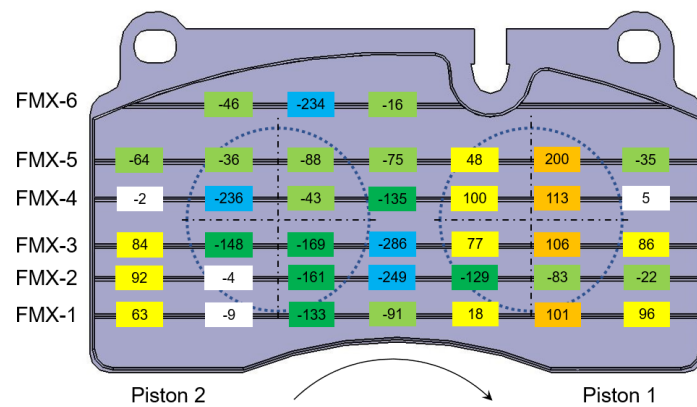
(c) Vertical (Y) Direction: 10bar Hydraulic Pressure with 200Nm Torque Level



(d) Horizontal (X) Direction: 40bar Hydraulic Pressure with 0Nm Torque Level

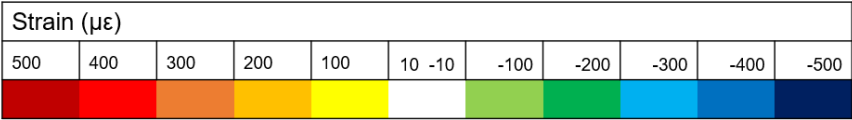
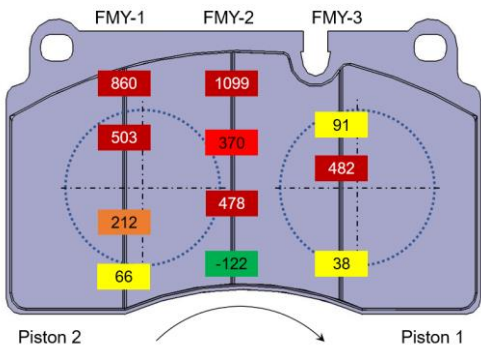


(e) Horizontal (X) Direction: 40bar Hydraulic Pressure with 200Nm Torque Level

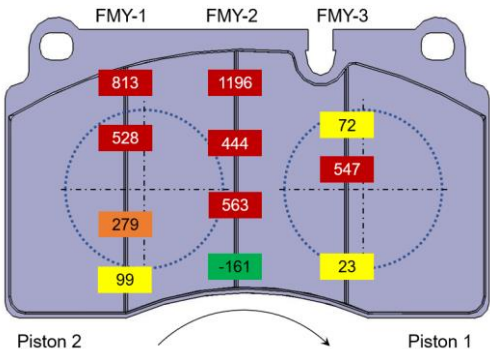


(f) Horizontal (X) Direction: 10bar Hydraulic Pressure with 200Nm Torque Level

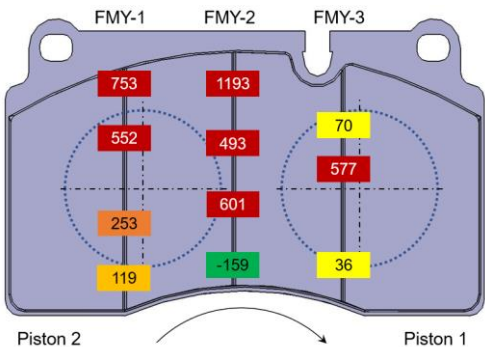
Appendix Chapter 6-D: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions between Three Test Stages with Equal Pressure (p1 = p2)



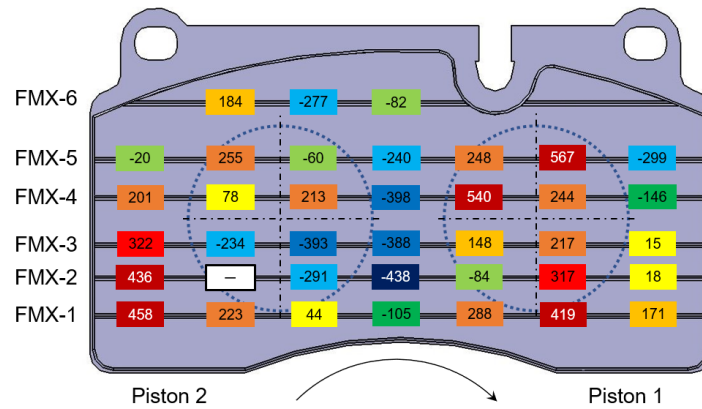
(a) Vertical (Y) Direction: 120bar Hydraulic Pressure with 0Nm Torque Level



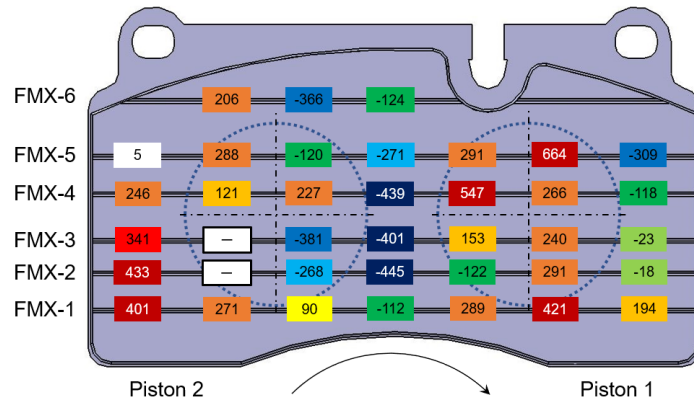
(b) Vertical (Y) Direction: 120bar Hydraulic Pressure with 1000Nm Torque Level



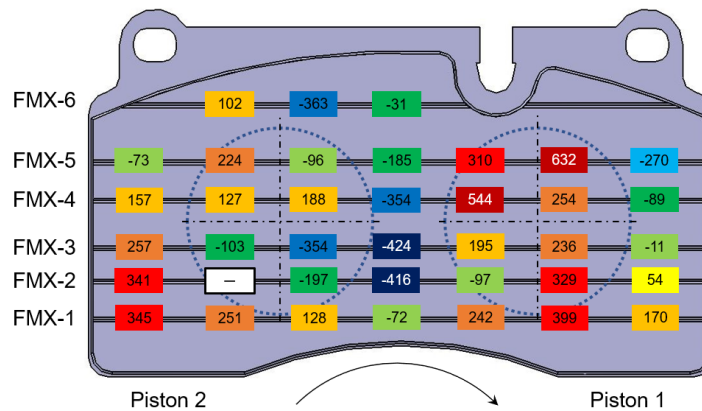
(c) Vertical (Y) Direction: 80bar Hydraulic Pressure with 1000Nm Torque Level



(d) Horizontal (X) Direction: 120bar Hydraulic Pressure with 0Nm Torque Level

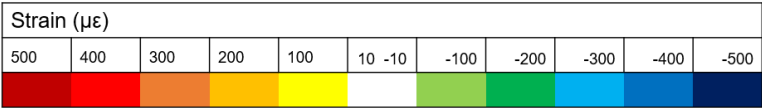
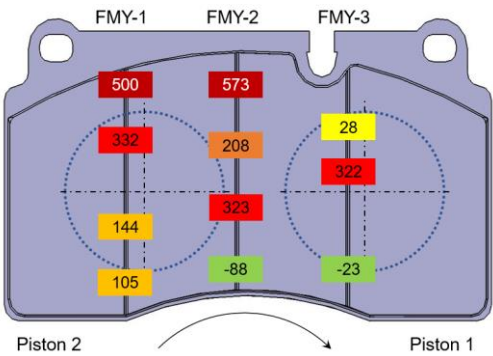


(e) Horizontal (X) Direction: 120bar Hydraulic Pressure with 1000Nm Torque Level

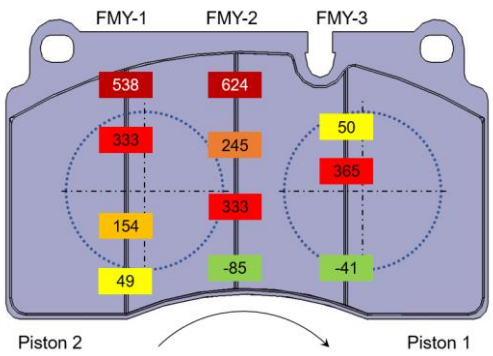


(f) Horizontal (X) Direction: 80bar Hydraulic Pressure with 1000Nm Torque Level

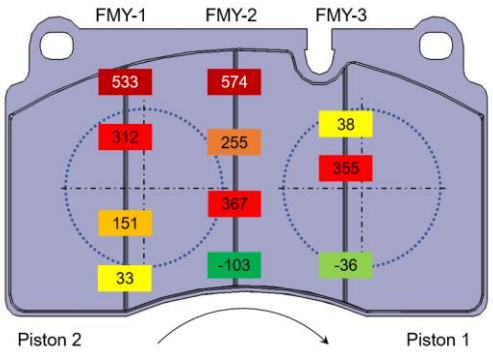
Appendix Chapter 6-E: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions Comparison between Three Pressure Variations with 40bar/200Nm



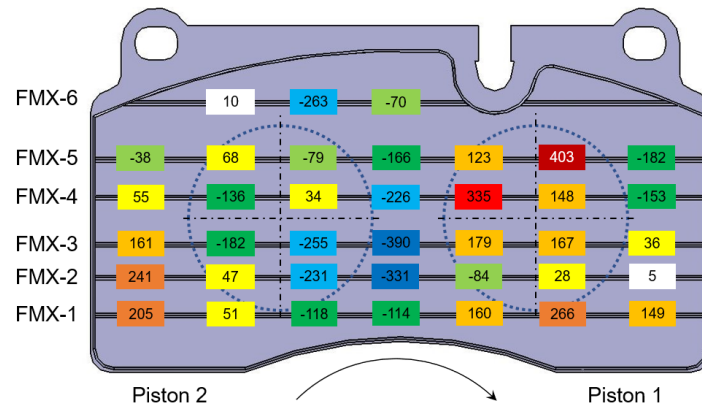
(a) Vertical (Y) Direction: Equal Pressure ($p_1=p_2$)



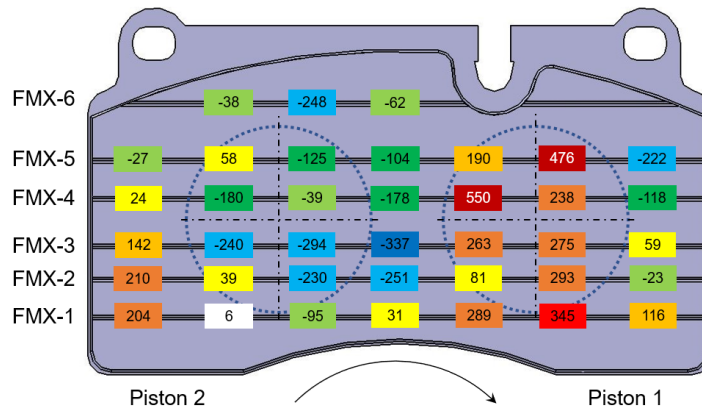
(b) Vertical (Y) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)



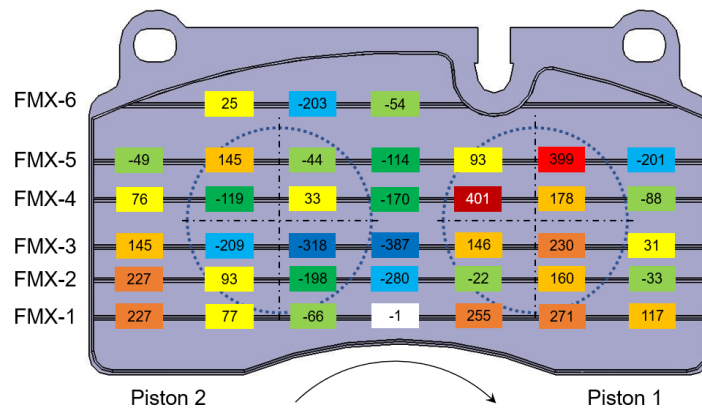
(c) Vertical (Y) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)



(d) Horizontal (X) Direction: Equal Pressure ($p_1=p_2$)

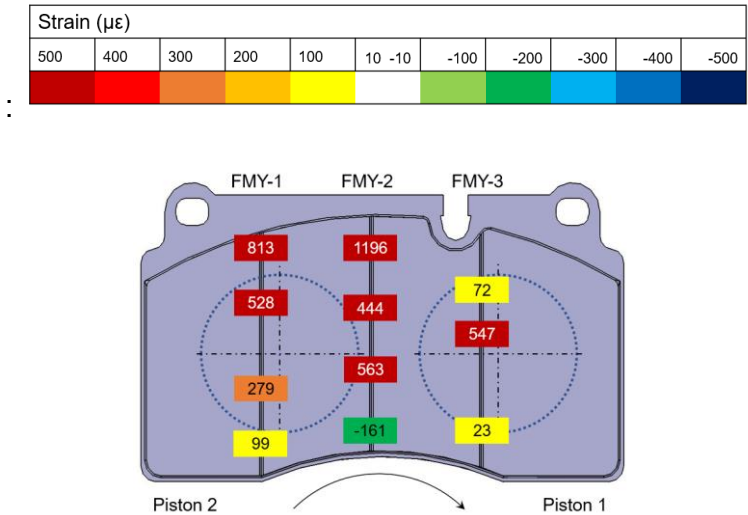


(e) Horizontal (X) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)

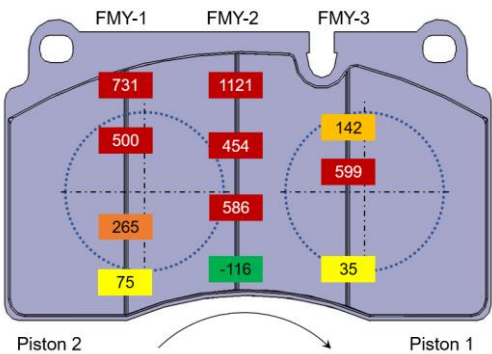


(f) Horizontal (X) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)

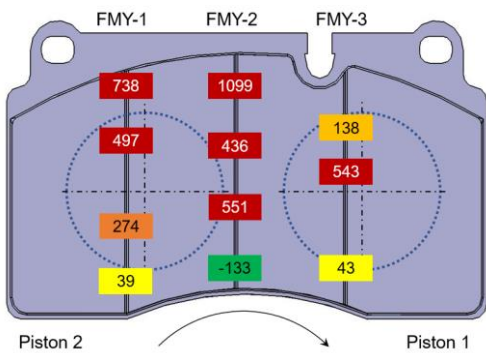
Appendix Chapter 6-F: FBG Sensor Contour Map for Vertical (Y) and Horizontal (X) Directions Comparison between Three Pressure Variations with 120bar/1000Nm



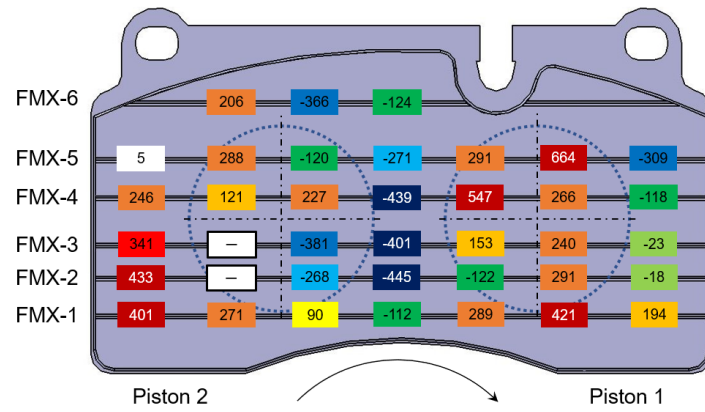
(a) Vertical (Y) Direction: Equal Pressure ($p_1=p_2$)



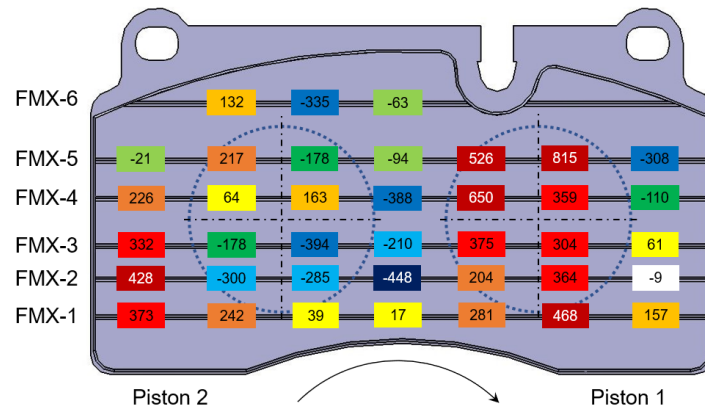
(b) Vertical (Y) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)



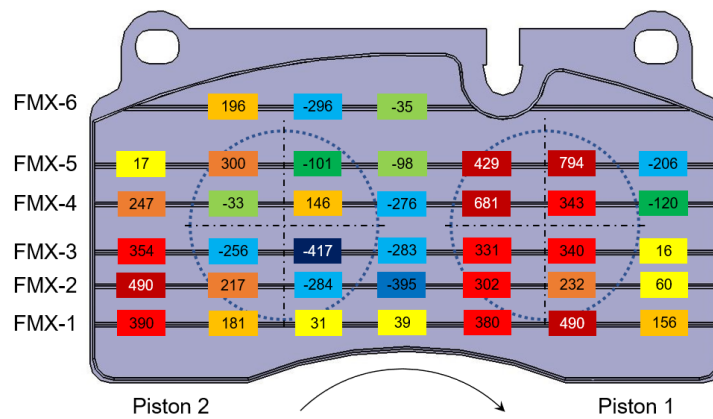
(c) Vertical (Y) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)



(d) Horizontal (X) Direction: Equal Pressure ($p_1=p_2$)

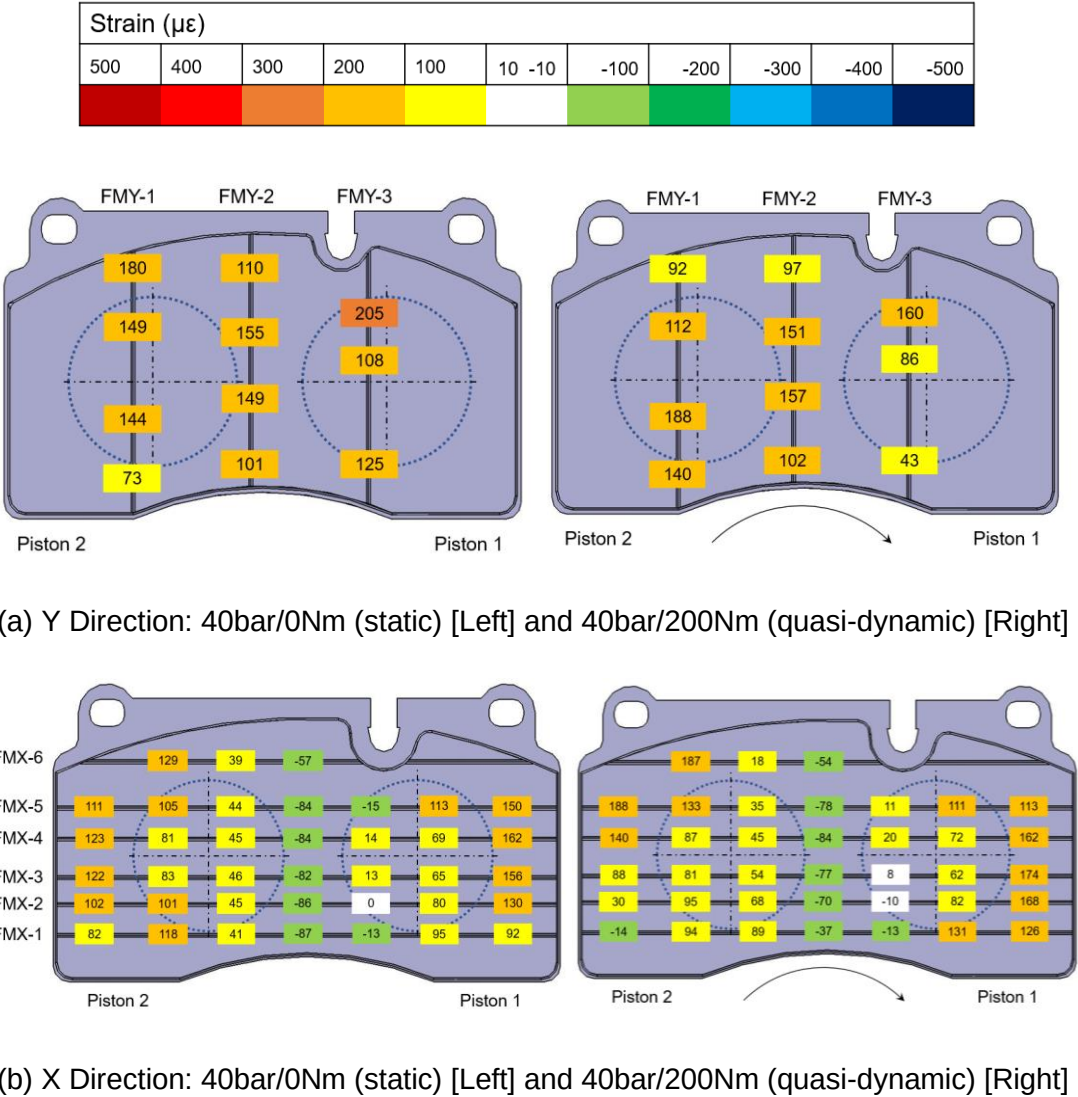


(e) Horizontal (X) Direction: +10bar Variation ($p_1=p_2+10\text{bar}$)



(f) Horizontal (X) Direction: -10bar Variation ($p_1=p_2-10\text{bar}$)

Appendix Chapter 7-A: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.4$ (40bar/200Nm)



FMX-6

129

39

-57

FMX-5

111

105

44

-84

-15

113

150

FMX-4

123

81

45

-84

14

69

162

FMX-3

122

83

46

-82

13

65

156

FMX-2

102

101

45

-86

0

80

130

FMX-1

82

118

41

-87

-13

95

92

Piston 2

Piston 1

FMX-6

187

18

-54

FMX-5

188

133

35

-78

11

111

113

FMX-4

140

87

45

-84

20

72

162

FMX-3

88

81

54

-77

8

62

174

FMX-2

30

95

68

-70

-10

82

168

FMX-1

-14

94

89

-37

-13

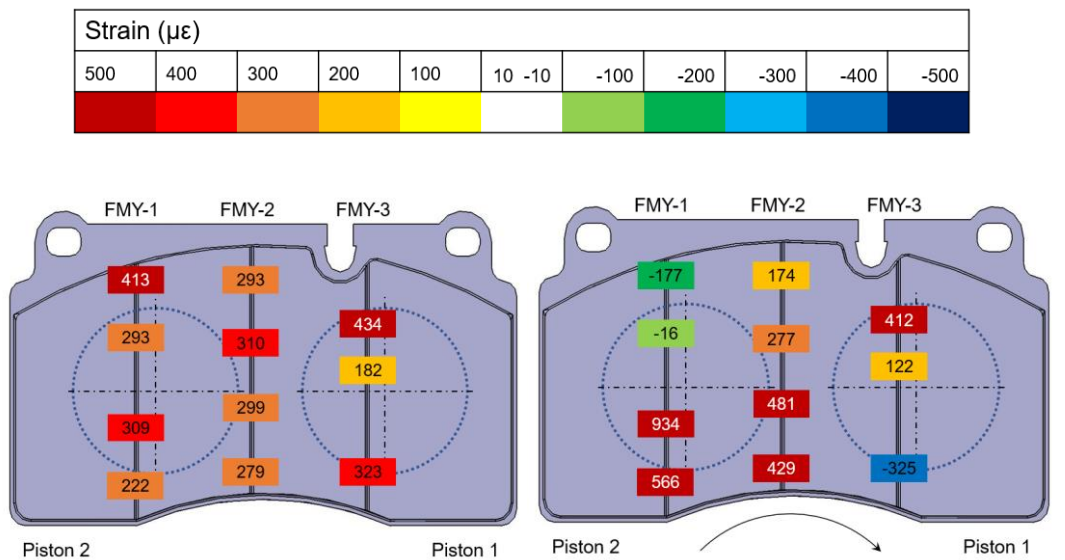
131

126

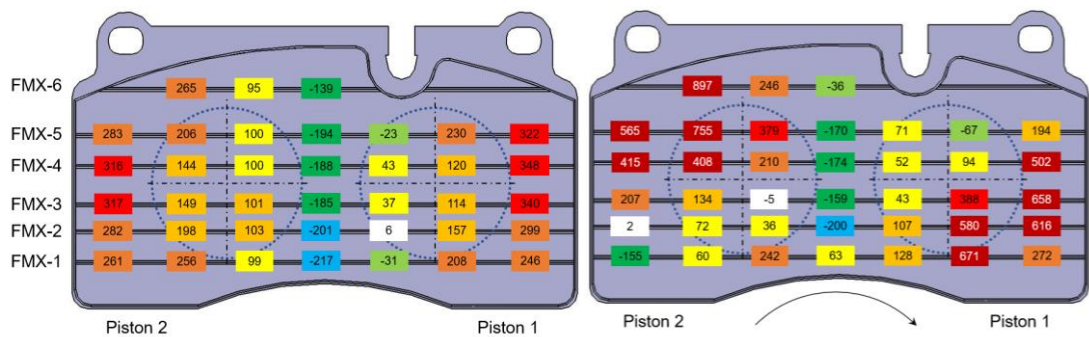
Piston 2

Piston 1

Appendix Chapter 7-B: FEA Results for Static and Quasi-Dynamic Loading Conditions with Friction Coefficient $\mu=0.46$ (120bar/1000Nm)



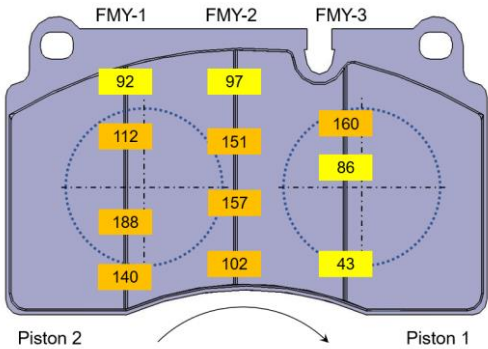
(a) Y Direction: 120bar/0Nm (static) [Left] and 120bar/1000Nm (quasi-dynamic) [Right]



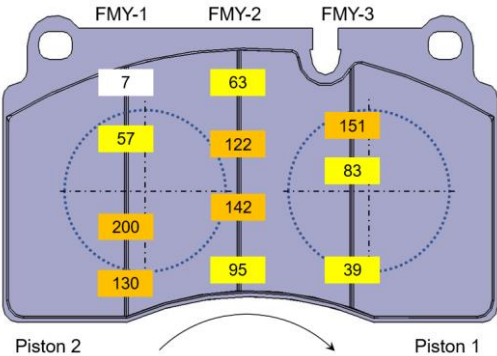
(b) X Direction: 120bar/0Nm (static) [Left] and 120bar/1000Nm (quasi-dynamic) [Right]

Appendix Chapter 7-C: E11 & E22 Processed FEA Strain Contour Map Comparisons for Different Hydraulic Pressure Variations (40bar/200Nm)

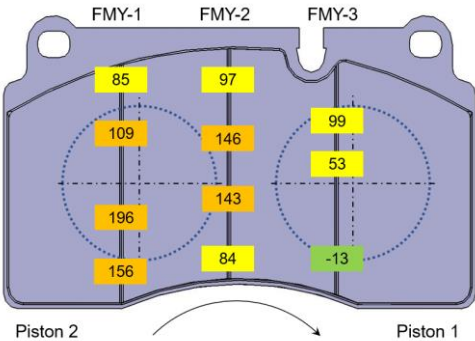
Strain (µε)											
500	400	300	200	100	10	-10	-100	-200	-300	-400	-500



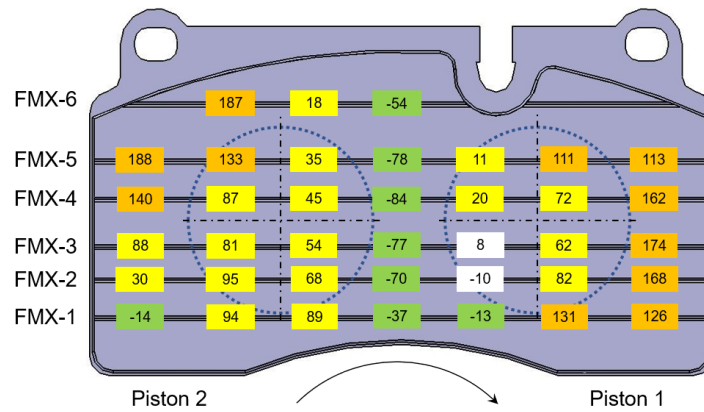
(a) Equal Pressure (p1=p2=40bar; 200Nm)



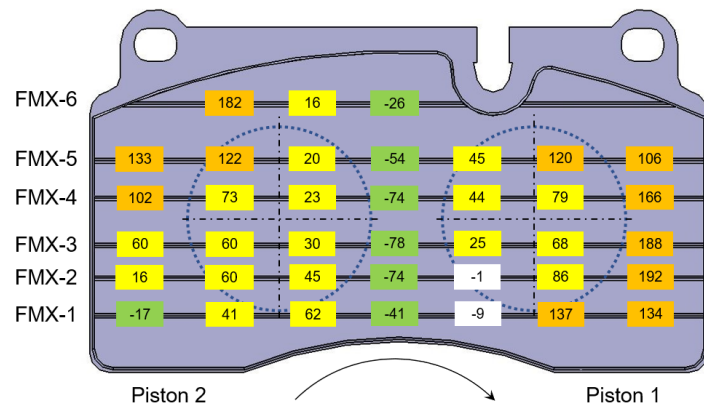
(b) +10bar (p1=40bar; p2=30bar; 200Nm)



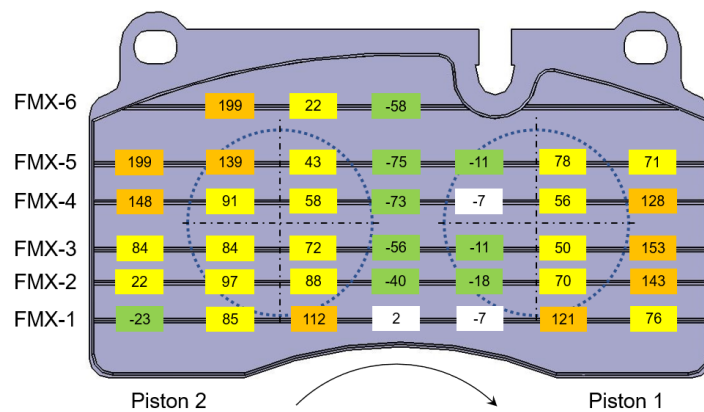
(c) -10bar (p1=30bar; p2=40bar; 200Nm)



(d) Equal Pressure ($p_1=p_2=40\text{bar}$; 200Nm)

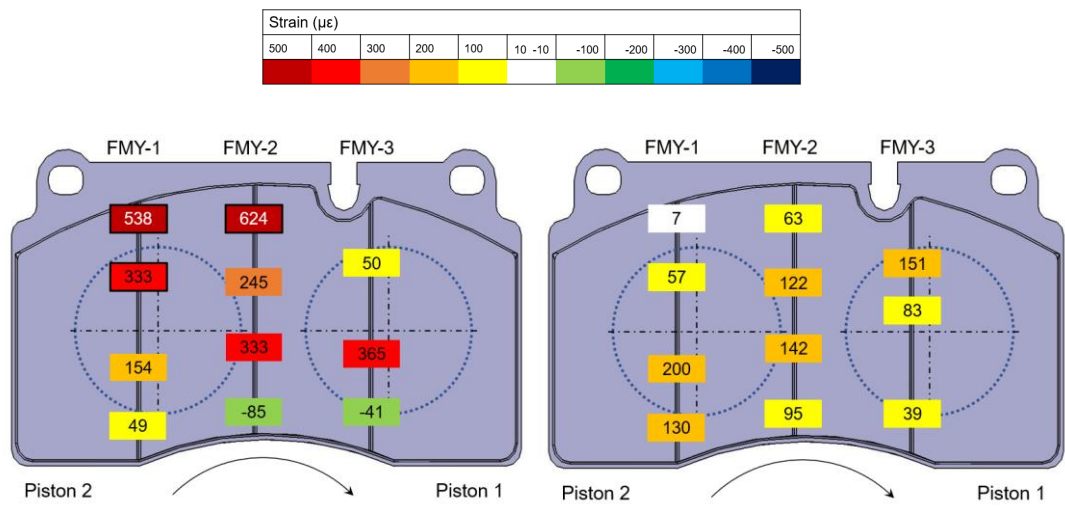


(e) +10bar ($p_1=40\text{bar}$; $p_2=30\text{bar}$; 200Nm)

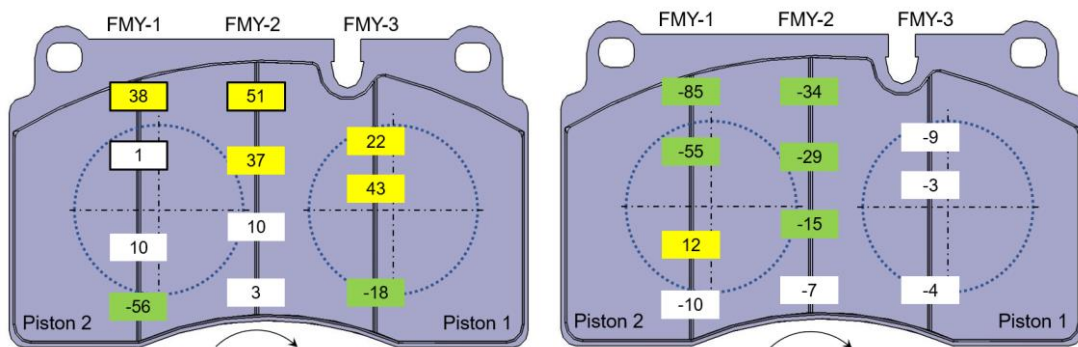


(f) -10bar ($p_1=30\text{bar}$; $p_2=40\text{bar}$; 200Nm)

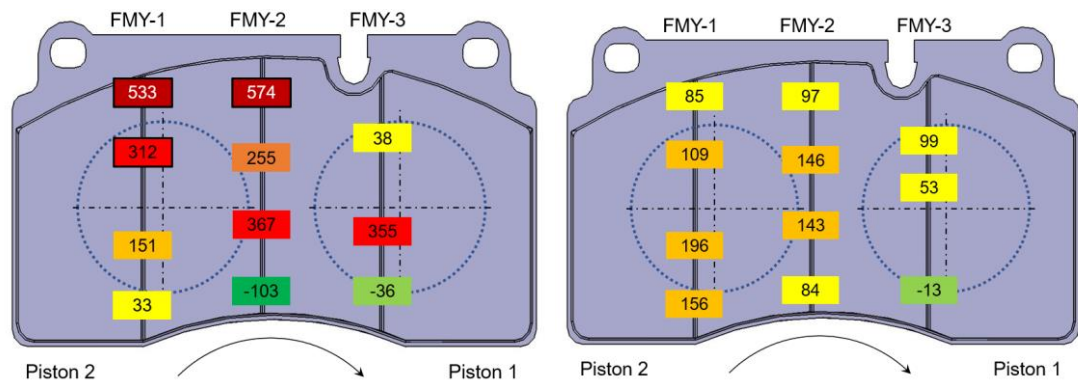
Appendix Chapter 7-D: E22 Strain Results for ±10bar Hydraulic Pressure Variations



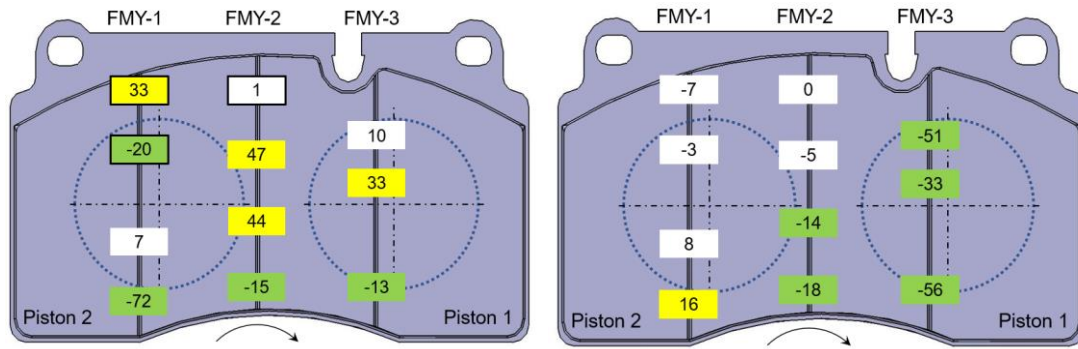
(a) FBG Test (Left) and FEA Simulation (Right) Results (p1=40bar and p2=30bar)



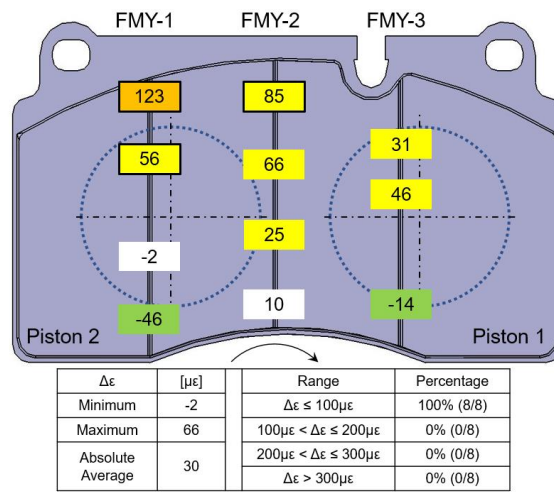
(b) Strain Differences: $\Delta\varepsilon = \varepsilon_{+10bar} - \varepsilon_{Equal}$: FBG Test (Left) and FEA Simulation (Right) (p1=40bar and p2=30bar)



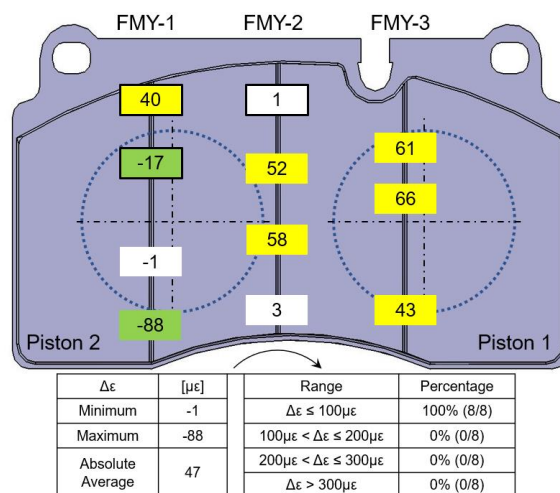
(c) FBG Test (Left) and FEA Simulation (Right) Results (p1=30bar and p2=40bar)



(d) Strain Differences: $\Delta\epsilon = \epsilon_{-10bar} - \epsilon_{Equal}$: FBG Test (Left) and FEA Simulation (Right) (p1=40bar and p2=30bar)



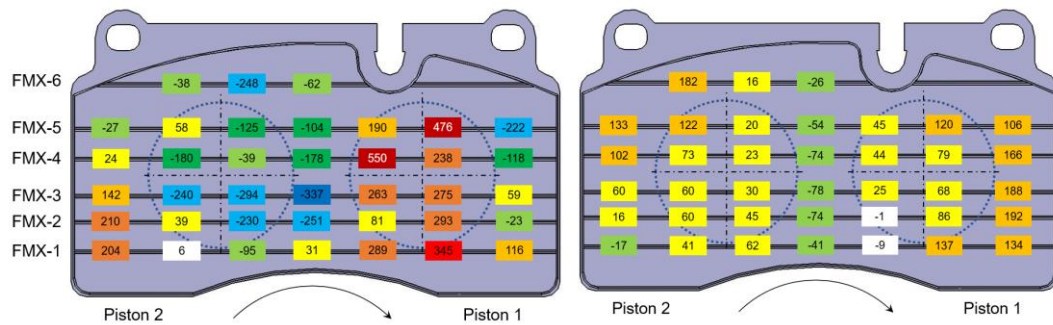
(e) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=40bar; p2=30bar



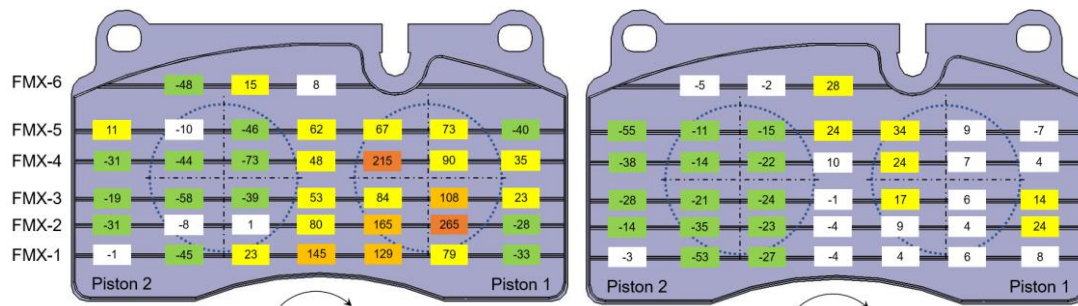
(f) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=30bar; p2=40bar

Appendix Chapter 7-E: E11 Strain Results for ± 10 bar Hydraulic Pressure Variations

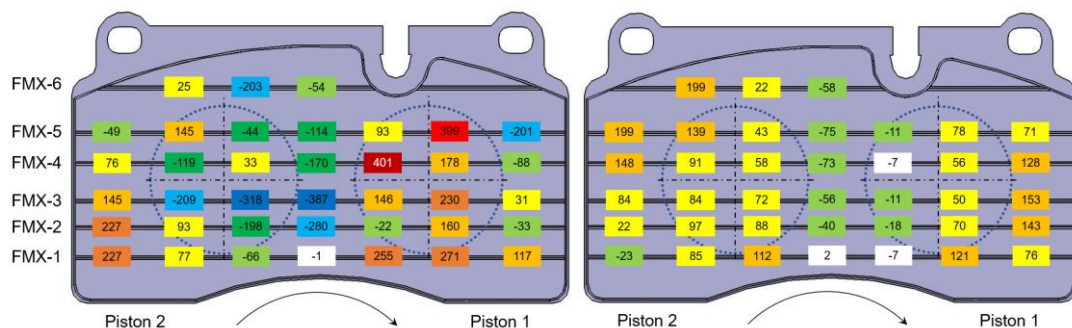
Strain (μϵ)										
500	400	300	200	100	10 -10	-100	-200	-300	-400	-500



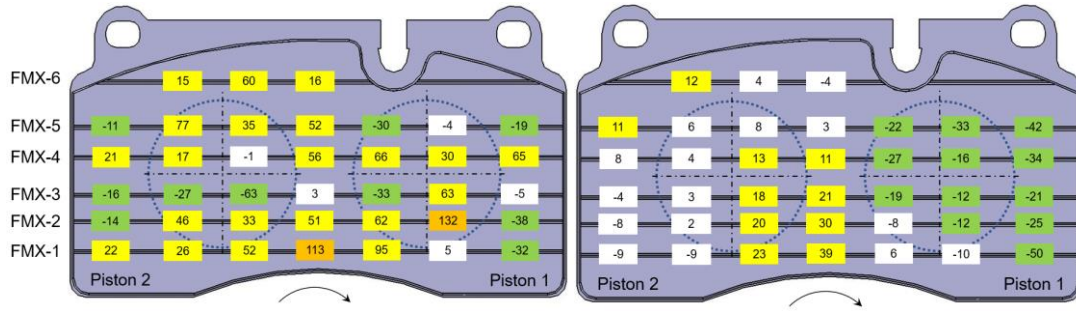
(a) FBG Test (Left) and FEA Simulation (Right) Results ($p_1=40\text{bar}$ and $p_2=30\text{bar}$)



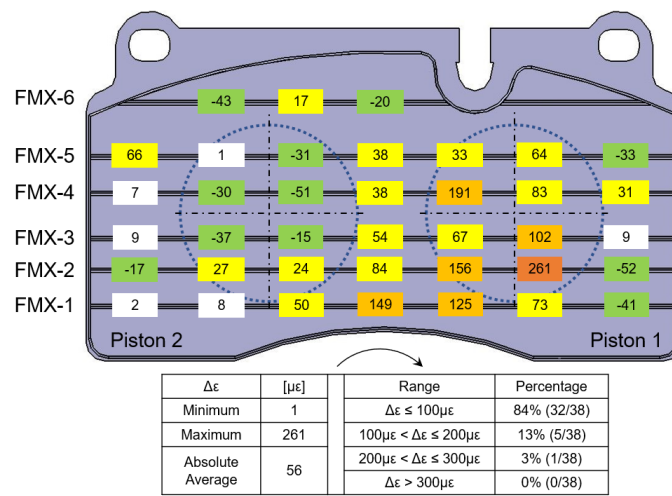
(b) Strain Differences: $\Delta\epsilon = \epsilon_{+10\text{bar}} - \epsilon_{\text{Equal}}$: FBG Test (Left) and FEA Simulation (Right) ($p_1=40\text{bar}$ and $p_2=30\text{bar}$)



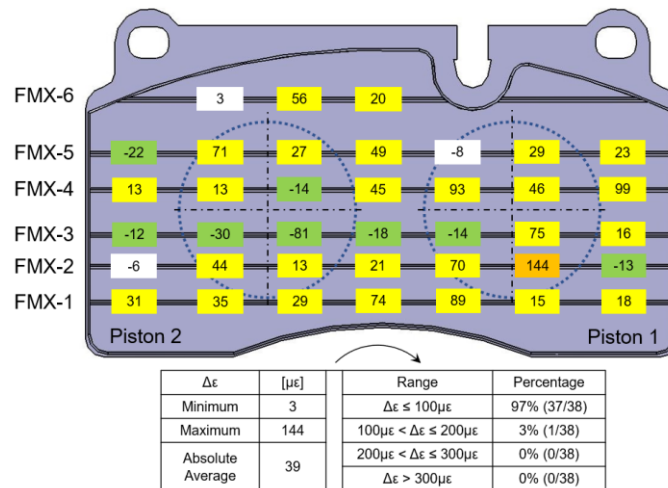
(c) FBG Test (Left) and FEA Simulation (Right) Results ($p_1=30\text{bar}$ and $p_2=40\text{bar}$)



(d) Strain Differences: $\Delta\epsilon = \epsilon_{-10bar} - \epsilon_{Equal}$: FBG Test (Left) and FEA Simulation (Right) (p1=40bar and p2=30bar)

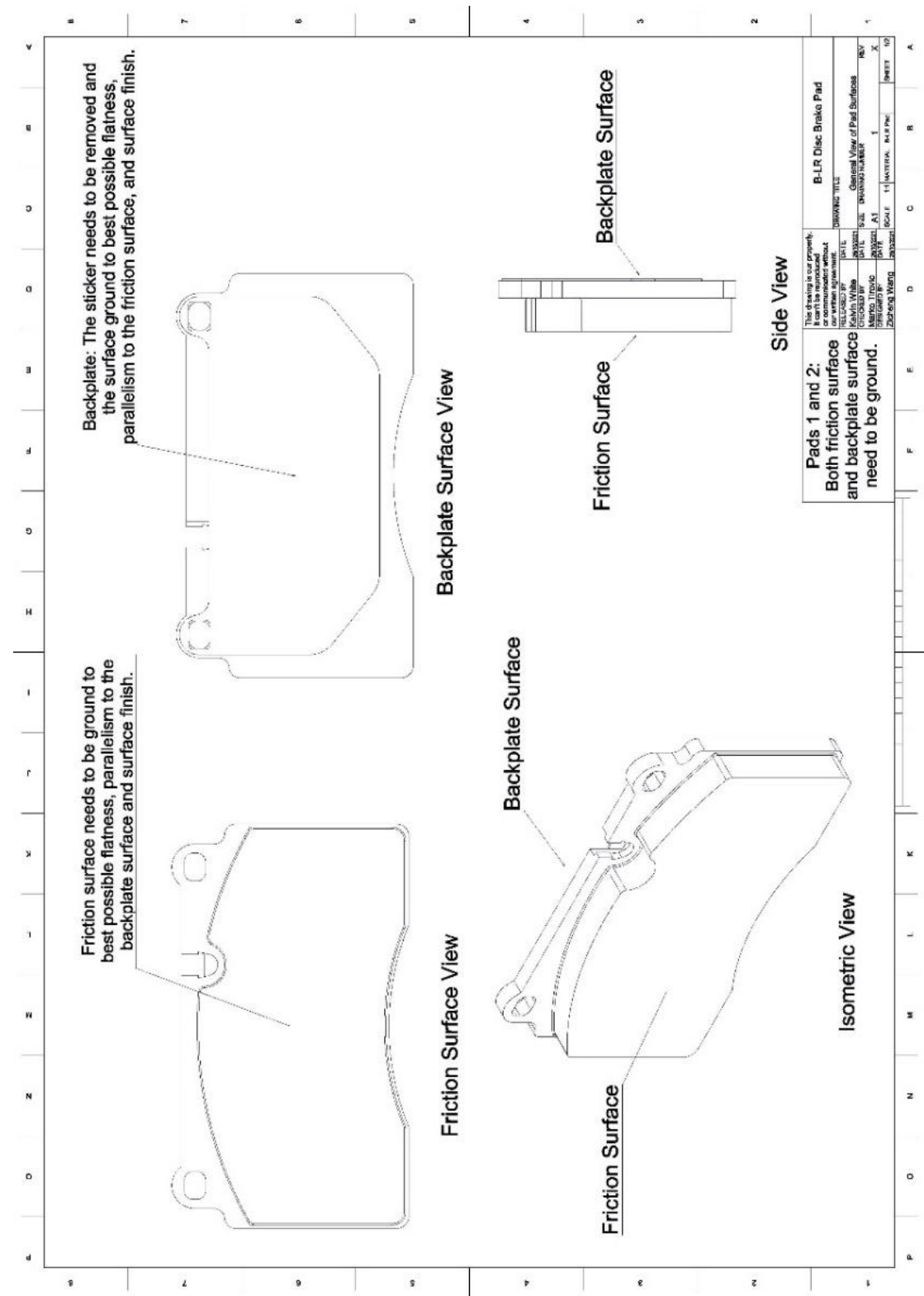


(e) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=40bar; p2=30bar



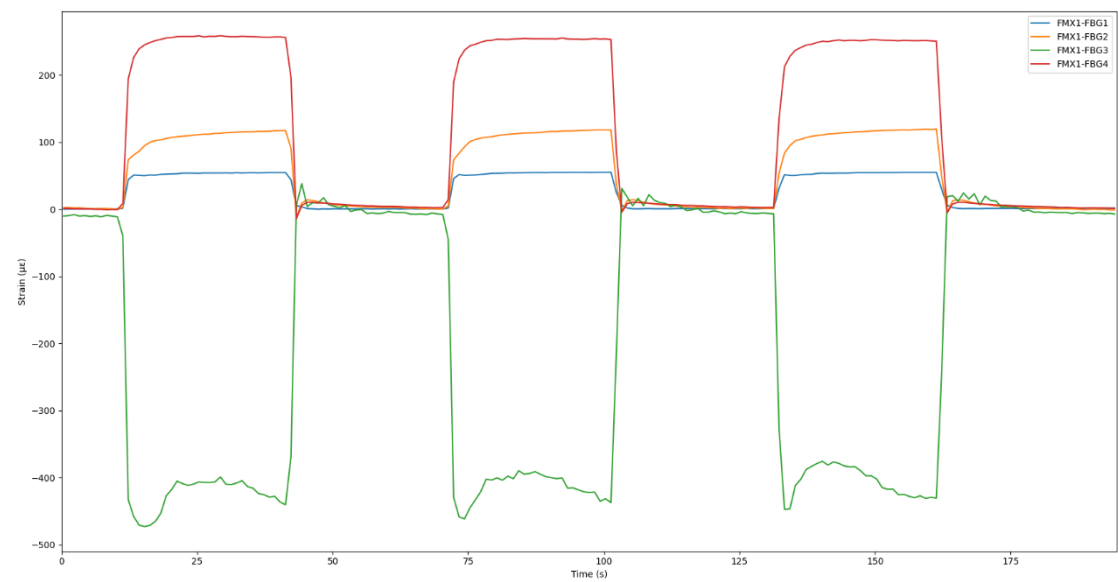
(f) Strain Differences $\Delta\epsilon = \epsilon_{Test} - \epsilon_{FEA}$: p1=30bar; p2=40bar

Appendix Chapter 8-A: The CAD drawings of The New Modified Brake Pad

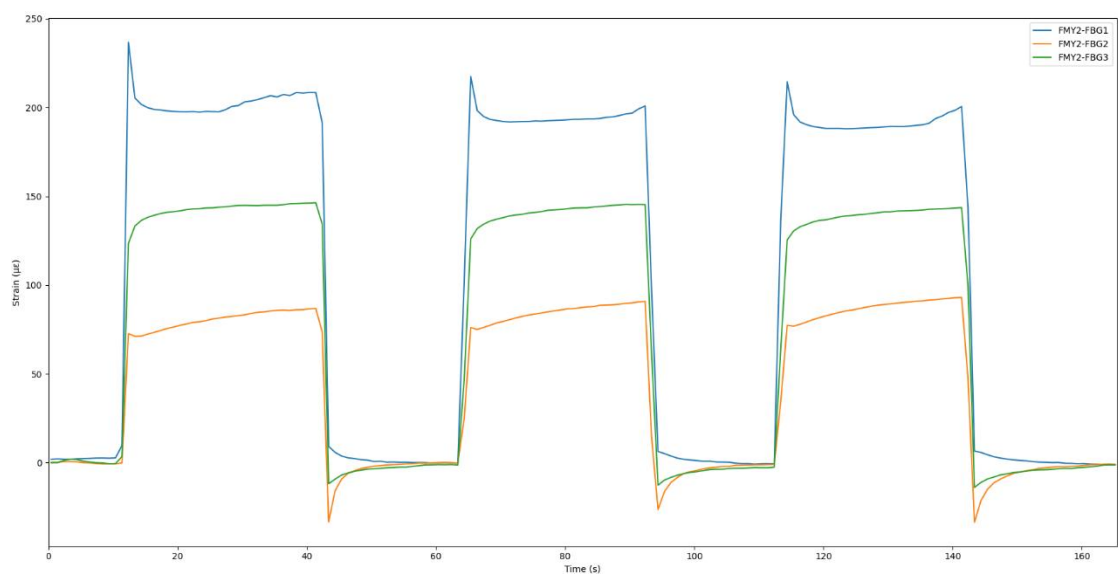


(a) The General View of The Brake Pad Surfaces

Appendix Chapter 8-B: The Test Result Graphs of The Initial Static Tests



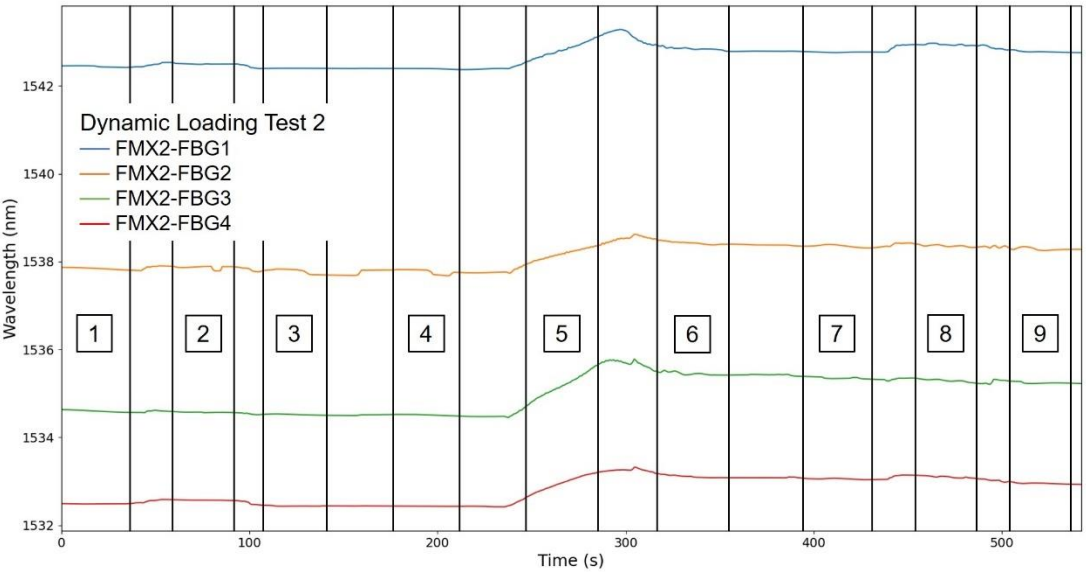
(a) FMX-1 Result Graph



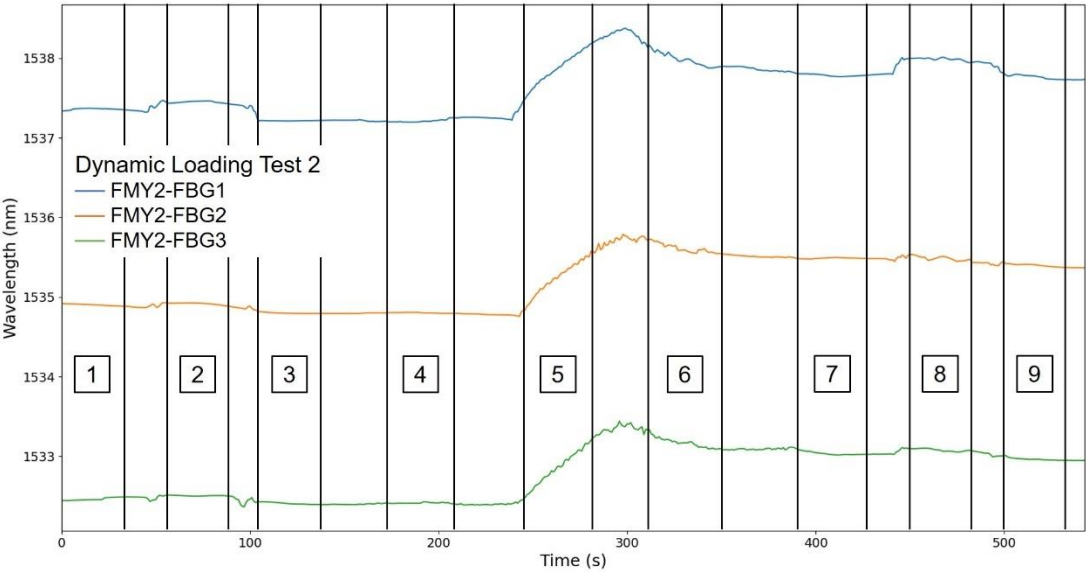
(b) FMY-2 Result Graph

Appendix Chapter 8-C: The Raw Recorded Data Graphs of The Dynamic Loading Test 2 & 3

Step	Initial Static Application			Dynamic Loading Application				Final Static Application	
Period	1	2	3	4	5	6	7	8	9
p (bar)	0	30	0	0	30	0	0	30	0
n (rpm)	0	0	0	0 - 50	50	50 - 0	0	0	0

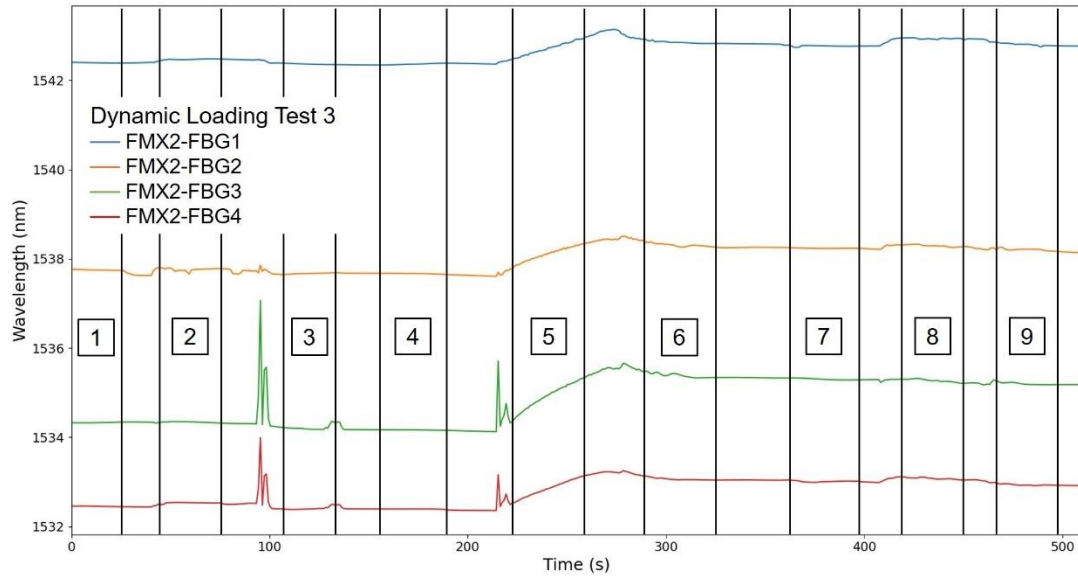


(a) Dynamic Loading Test 2: FMX2

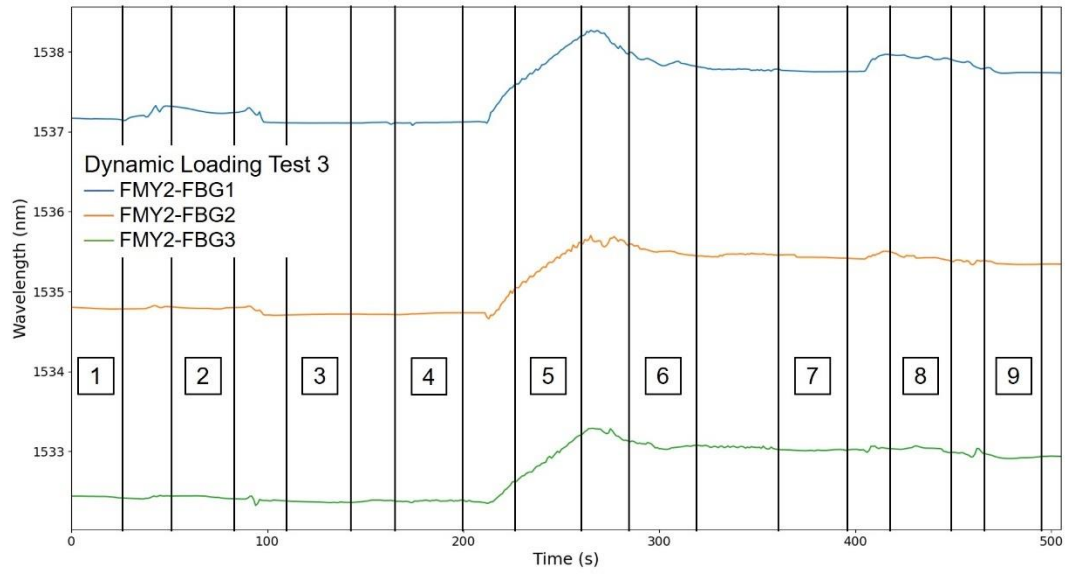


(b) Dynamic Loading Test 2: FMY2

Step	Initial Static Application			Dynamic Loading Application				Final Static Application	
Period	1	2	3	4	5	6	7	8	9
p (bar)	0	30	0	0	30	0	0	30	0
n (rpm)	0	0	0	0 - 50	50	50 - 0	0	0	0



(c) Dynamic Loading Test 3: FMX2



(d) Dynamic Loading Test 3: FMY2