



University of Wales Trinity Saint David

School of Engineering

Drive Cycle Modelling and Engine
Analysis

Dissertation

M6X00417

Jade Mellors

P141265

BENG Automotive Engineering

07/04/2017

Declaration of Authenticity

The thesis is the result of the authors original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

Signed:

Date:

Table of Contents

1	<u>Table of Figures</u>	<u>iv</u>
2	<u>Table of Tables.....</u>	<u>xii</u>
3	<u>Table of Equations</u>	<u>xiii</u>
4	<u>Acknowledgements</u>	<u>1</u>
5	<u>Abstract</u>	<u>2</u>
6	<u>Nomenclature</u>	<u>3</u>
7	<u>Abbreviations</u>	<u>6</u>
8	<u>Constants</u>	<u>7</u>
9	<u>Introduction</u>	<u>8</u>
10	<u>Drive Cycles.....</u>	<u>10</u>
10.1	Vehicle Constants – 0	11
10.1.1	VW Passat.....	11
10.1.2	Ford Fox	11
10.2	NEDC	12
10.2.1	The Model.....	13
10.2.2	Fuel Flow	17
10.2.3	Results.....	20
10.3	WLTP.....	27
10.3.1	Gear Shift Profile Calculator.....	29
10.3.2	Results.....	31
10.4	Analysis	33
10.4.1	NEDC	33
10.4.2	Gear Calculator.....	42
10.4.3	Comparison	43
11	<u>Combustion</u>	<u>50</u>
11.1	The Model.....	50

11.2	Results.....	54
11.3	Analysis	59
12	<u>Friction</u>	<u>62</u>
12.1	Model Fundamentals	62
12.2	Results.....	66
12.2.1	Piston Skirt	72
12.2.2	Piston Rings	75
12.3	Analysis of Model.....	81
13	<u>Conclusions.....</u>	<u>83</u>
14	<u>Further Work.....</u>	<u>84</u>
15	<u>References</u>	<u>85</u>
16	<u>Appendix</u>	<u>88</u>
16.1	Appendix A – Conversions.....	88
16.2	Appendix B- Drive Cycle Model.....	89
16.2.1	Fuel	89
16.2.2	Results.....	103
16.2.3	Gear Calculator.....	108
16.3	Appendix C- Combustion Model.....	122
16.3.1	Logarithmic Pressure and Volume	122
16.3.2	Extra Pressure From Combustion	128
16.3.3	MFB	134
16.4	Appendix D – Friction Model	140
16.4.1	Gas Force	140
16.4.2	Inertia Force	146
16.4.3	Net Force	152
16.4.4	Connecting Rod Force	158
16.4.5	Work Done by the Pressure	164
16.4.6	Side Force	170

16.4.7	Friction Force.....	176
16.4.8	Work Done by the Piston Skirt	182
16.4.9	Radial Force	188
16.4.10	Friction Force	194
16.4.11	Work Done by the Piston Rings.....	200
16.5	Appendix E – Supervisor Notes	206
16.5.1	18 th October 2017	209
16.5.2	25 th October 2017	210
16.5.3	8 th November 2017	211
16.5.4	15 th November 2017	215
16.5.5	22 nd November 2017.....	216
16.5.6	29 th November 2017	216
16.5.7	6 th December 2017	217
16.5.8	17 th January 2018	217
16.5.9	24 th January 2018	218
16.5.10	31 st January 2018.....	219
16.5.11	14 th and 21 st February 2018.....	220
16.5.12	28 th February 2018	221
16.5.13	7 th March 2018	222
16.5.14	14 th March 2018	224
16.5.15	28 th March 2018	225

1 Table of Figures

Figure 10-1: NEDC Drive Cycle	13
Figure 10-2: Forces acting on a Flat Road	14
Figure 10-3: Graph at 4000RPM of the Fuel Flow (g/s) for the VW Passat.....	18
Figure 10-4: Graph at 4000RPM of the Fuel Flow (g/s) for the Ford Fox.....	19
Figure 10-5: WLTP Showing the Low, Medium, High and Extra High-Speed Sections	27
Figure 10-6: Iteration 1; Gear Selection.....	31
Figure 10-7: Iteration 2; Gear Selection.....	32

Figure 10-8: Iteration 3; Gear Selection.....	32
Figure 10-9: Iteration 4; Gear Selection.....	33
Figure 10-10: RPM Comparison during the UDC Section of the NEDC.....	34
Figure 10-11: RPM Comparison during the EUDC Section of the NEDC.....	34
Figure 10-12: BMEP Comparison during the UDC Section of the NEDC.....	35
Figure 10-13: BMEP Comparison during the EUDC Section of the NEDC	36
Figure 10-14: Fuel Flow Comparison during the UDC of the NEDC	37
Figure 10-15: Fuel Flow Comparison during the EUDC of the NEDC.....	37
Figure 10-16: Cumulative Fuel Consumption Comparison	38
Figure 10-17: Power Used by the VW Passat, during the UDC Section of the NEDC.....	39
Figure 10-18: Power Used by the Ford Fox, during the UDC Section of the NEDC.....	40
Figure 10-19: Power Used by the VW Passat, during the EUDC Section of the NEDC	40
Figure 10-20: Power Used by the Ford Fox, during the EUDC Section of the NEDC.....	41
Figure 10-21: Comparison of the NEDC and Drive Cycle Velocities during Testing	43
Figure 10-22: Comparison of RPM Between the NEDC and WLTP.....	44
Figure 10-23 BMEP for the VW Passat in the NEDC.....	45
Figure 10-24: BMEP of the VW Passat in the WLTP	45
Figure 10-25: Comparison of the Fuel Flow per Second between the NEDC and WLTP....	46
Figure 10-26: Fuel Consumption Comparison between the NEDC and WLTP	47
Figure 10-27: Comparison of the Power Used during the NEDC and WLTP Drive Cycles ..	48
Figure 11-1: Piston Displacement	51
Figure 11-2: Example of Log Pressure Vs Log Volume (Ferguson & Kiskpatrick, 2000, p. 122)	53
Figure 11-3: Ford Fox Piston Displacement from TDC	55
Figure 11-4: Logarithmic Pressure and Volume at 4000RPM, with Gradient Placement	56
Figure 11-5: Extra Pressure from Combustion at 4000RPM.....	57
Figure 11-6: Mass Fraction Burned at 4000RPM	58
Figure 11-7: Comparisons of Mass Fraction Burned of the RPM Range 1000-2500	60
Figure 11-8: Comparisons of Mass Fraction Burned for the RPM Range 3000-4500	60
Figure 11-9: Comparison of Mass Fraction Burned of the RPM Range 5000-6500	61
Figure 12-1: Force Diagram Acting on the Piston.....	64
Figure 12-2: Maximum Gas Force at 4000RPM	68
Figure 12-3: Inertia Force at 4000RPM	70
Figure 12-4: Net Force at 4000RPM	70
Figure 12-5: Force Acting Through the Connecting Rod at 4000RPM.....	71
Figure 12-6: Work Done by the Pressure from Combustion at 4000RPM.....	72
Figure 12-7: Side Force Acting on the Piston Skirt at 4000RPM	73

Figure 12-8: Friction Force from the Piston Skirt against the Cylinder Wall at 4000RPM.....	74
Figure 12-9: Work Done Against Combustion by the Piston Skirt at 4000RPM	74
Figure 12-10: Piston rings	75
Figure 12-11: Diagram of Radial and Friction Force of the First Piston Ring	76
Figure 12-12: Radial Force on the Top Piston Ring at 4000RPM	77
Figure 12-13: Friction Force from the Piston Rings against the Cylinder Wall at 4000RPM.	78
Figure 12-14: Work Done by the Piston Rings against Combustion at 4000RPM	79
Figure 16-1: Fuel Flow Rate for VW Passat at 500RPM.....	89
Figure 16-2: Fuel Flow Rate for VW Passat at 850 RPM.....	89
Figure 16-3: Fuel Flow Rate for VW Passat at 1000 RPM.....	90
Figure 16-4: Fuel Flow Rate for VW Passat at 1250 RPM.....	90
Figure 16-5: Fuel Flow Rate for VW Passat at 1500 RPM.....	91
Figure 16-6: Fuel Flow Rate for VW Passat at 1750 RPM.....	91
Figure 16-7: Fuel Flow Rate for VW Passat at 2000 RPM.....	92
Figure 16-8: Fuel Flow Rate for VW Passat at 2500 RPM.....	92
Figure 16-9: Fuel Flow Rate for VW Passat at 3000 RPM.....	93
Figure 16-10: Fuel Flow Rate for VW Passat at 3500 RPM.....	93
Figure 16-11: Fuel Flow Rate for VW Passat at 4500 RPM.....	94
Figure 16-12: Fuel Flow Rate for VW Passat at 5000 RPM.....	94
Figure 16-13: Fuel Flow Rate for VW Passat at 6000 RPM.....	95
Figure 16-14: Fuel Flow Rate for VW Passat at 7000 RPM.....	95
Figure 16-15: Fuel Flow Rate for Ford Fox Engine at 800RPM	96
Figure 16-16: Fuel Flow Rate for Ford Fox Engine at 1000RPM	96
Figure 16-17: Fuel Flow Rate for Ford Fox Engine at 1250RPM	97
Figure 16-18: Fuel Flow Rate for Ford Fox Engine at 1500RPM	97
Figure 16-19: Fuel Flow Rate for Ford Fox Engine at 1750RPM	98
Figure 16-20: Fuel Flow Rate for Ford Fox Engine at 2000RPM	98
Figure 16-21: Fuel Flow Rate for Ford Fox Engine at 2250RPM	99
Figure 16-22: Fuel Flow Rate for Ford Fox Engine at 2500RPM	99
Figure 16-23: Fuel Flow Rate for Ford Fox Engine at 3000RPM	100
Figure 16-24: Fuel Flow Rate for Ford Fox Engine at 3500RPM	100
Figure 16-25: Fuel Flow Rate for Ford Fox Engine at 4500RPM	101
Figure 16-26: Fuel Flow Rate for Ford Fox Engine at 5000RPM	101
Figure 16-27: Fuel Flow Rate for Ford Fox Engine at 5500RPM	102
Figure 16-28: Fuel Flow Rate for Ford Fox Engine at 6000RPM	102
Figure 16-29: Fuel Flow Rate for Ford Fox Engine at 6500RPM	103

Figure 16-30: Comparison of the Differential Input Torque during the UDC Section of the NEDC	103
Figure 16-31: Comparison of the Differential Input Torque during the EUDC Section of the NEDC	104
Figure 16-32: Comparison of the Gearbox Input Torque during the UDC Section of the NEDC	104
Figure 16-33: Comparison of the Gearbox Input Torque during the EUDC Section of the NEDC	105
Figure 16-34: Comparison of Engine Angular Acceleration during the UDC Section of the NEDC	105
Figure 16-35: Comparison of Engine Angular Acceleration during the EUDC Section of the NEDC	106
Figure 16-36: Comparison of the Engine Torque Lost during the UDC Section of the NEDC	106
Figure 16-37: Comparison of the Engine Torque Lost during the EUDC Section of the NEDC	107
Figure 16-38: Comparison of the Engine Internal Torque during the UDC Section of the NEDC	107
Figure 16-39: Comparison of the Engine Internal Torque during the EUDC Section of the NEDC	108
Figure 16-40: Logarithmic Pressure and Volume at 1000RPM.....	122
Figure 16-41: Logarithmic Pressure and Volume at 1250RPM.....	122
Figure 16-42: Logarithmic Pressure and Volume at 1500RPM.....	123
Figure 16-43: Logarithmic Pressure and Volume at 2000RPM.....	123
Figure 16-44: Logarithmic Pressure and Volume at 2500RPM.....	124
Figure 16-45: Logarithmic Pressure and Volume at 3000RPM.....	124
Figure 16-46: Logarithmic Pressure and Volume at 3500RPM.....	125
Figure 16-47: Logarithmic Pressure and Volume at 4500RPM.....	125
Figure 16-48: Logarithmic Pressure and Volume at 5000RPM.....	126
Figure 16-49: Logarithmic Pressure and Volume at 5500RPM.....	126
Figure 16-50: Logarithmic Pressure and Volume at 6000RPM.....	127
Figure 16-51: Logarithmic Pressure and Volume at 6500RPM.....	127
Figure 16-52: Extra Pressure from Combustion at 1000RPM.....	128
Figure 16-53: Extra Pressure from Combustion at 1250RPM.....	128
Figure 16-54: Extra Pressure from Combustion at 1500RPM.....	129
Figure 16-55: Extra Pressure from Combustion at 2000RPM.....	129
Figure 16-56: Extra Pressure from Combustion at 2500RPM.....	130

Figure 16-57: Extra Pressure from Combustion at 3000RPM.....	130
Figure 16-58: Extra Pressure from Combustion at 3500RPM.....	131
Figure 16-59: Extra Pressure from Combustion at 4500RPM.....	131
Figure 16-60: Extra Pressure from Combustion at 5000RPM.....	132
Figure 16-61: Extra Pressure from Combustion at 5500RPM.....	132
Figure 16-62: Extra Pressure from Combustion at 6000RPM.....	133
Figure 16-63: Extra Pressure from Combustion at 6500RPM.....	133
Figure 16-64: Mass Fraction Burned at 1000RPM	134
Figure 16-65: Mass Fraction Burned at 1250RPM	134
Figure 16-66: Mass Fraction Burned at 1500RPM	135
Figure 16-67: Mass Fraction Burned at 2000RPM	135
Figure 16-68: Mass Fraction Burned at 2500RPM	136
Figure 16-69: Mass Fraction Burned at 3000RPM	136
Figure 16-70: Mass Fraction Burned at 3500RPM	137
Figure 16-71: Mass Fraction Burned at 4500RPM	137
Figure 16-72: Mass Fraction Burned at 5000RPM	138
Figure 16-73: Mass Fraction Burned at 5500RPM	138
Figure 16-74: Mass Fraction Burned at 6000RPM	139
Figure 16-75: Mass Fraction Burned at 6500RPM	139
Figure 16-76: Maximum Gas Force at 1000RPM	140
Figure 16-77: Maximum Gas Force at 1250RPM	140
Figure 16-78: Maximum Gas Force at 1500RPM	141
Figure 16-79: Maximum Gas Force at 2000RPM	141
Figure 16-80: Maximum Gas Force at 2500RPM	142
Figure 16-81: Maximum Gas Force at 3000RPM	142
Figure 16-82: Maximum Gas Force at 3500RPM	143
Figure 16-83: Maximum Gas Force at 4500RPM	143
Figure 16-84: Maximum Gas Force at 5000RPM	144
Figure 16-85: Maximum Gas Force at 5500RPM	144
Figure 16-86: Maximum Gas Force at 6000RPM	145
Figure 16-87: Maximum Gas Force at 6500RPM	145
Figure 16-88: Inertia Force at 1000RPM	146
Figure 16-89: Inertia Force at 1250RPM	146
Figure 16-90: Inertia Force at 1500RPM	147
Figure 16-91: Inertia Force at 2000RPM	147
Figure 16-92: Inertia Force at 2500RPM	148
Figure 16-93: Inertia Force at 3000RPM	148

Figure 16-94: Inertia Force at 3500RPM	149
Figure 16-95: Inertia Force at 4500RPM	149
Figure 16-96: Inertia Force at 5000RPM	150
Figure 16-97: Inertia Force at 5500RPM	150
Figure 16-98: Inertia Force at 6000RPM	151
Figure 16-99: Inertia Force at 6500RPM	151
Figure 16-100 Net Force at 1000RPM	152
Figure 16-101: Net Force at 1250RPM	152
Figure 16-102: Net Force at 1500RPM	153
Figure 16-103: Net Force at 2000RPM	153
Figure 16-104: Net Force at 2500RPM	154
Figure 16-105: Net Force at 3000RPM	154
Figure 16-106: Net Force at 3500RPM	155
Figure 16-107: Net Force at 4500RPM	155
Figure 16-108: Net Force at 5000RPM	156
Figure 16-109: Net Force at 5500RPM	156
Figure 16-110: Net Force at 6000RPM	157
Figure 16-111: Net Force at 6500RPM	157
Figure 16-112: Force Acting Through the Connecting Rod at 1000RPM	158
Figure 16-113: Force Acting Through the Connecting Rod at 1250RPM	158
Figure 16-114: Force Acting Through the Connecting Rod at 1500RPM	159
Figure 16-115: Force Acting Through the Connecting Rod at 2000RPM	159
Figure 16-116: Force Acting Through the Connecting Rod at 2500RPM	160
Figure 16-117: Force Acting Through the Connecting Rod at 3000RPM	160
Figure 16-118: Force Acting Through the Connecting Rod at 3500RPM	161
Figure 16-119: Force Acting Through the Connecting Rod at 4500RPM	161
Figure 16-120: Force Acting Through the Connecting Rod at 5000RPM	162
Figure 16-121: Force Acting Through the Connecting Rod at 5500RPM	162
Figure 16-122: Force Acting Through the Connecting Rod at 6000RPM	163
Figure 16-123: Force Acting Through the Connecting Rod at 6500RPM	163
Figure 16-124: Work Done by the Pressure from Combustion at 1000RPM	164
Figure 16-125: Work Done by the Pressure from Combustion at 1250RPM	164
Figure 16-126: Work Done by the Pressure from Combustion at 1500RPM	165
Figure 16-127: Work Done by the Pressure from Combustion at 2000RPM	165
Figure 16-128: Work Done by the Pressure from Combustion at 2500RPM	166
Figure 16-129: Work Done by the Pressure from Combustion at 3000RPM	166
Figure 16-130: Work Done by the Pressure from Combustion at 3500RPM	167

Figure 16-131: Work Done by the Pressure from Combustion at 4500RPM.....	167
Figure 16-132: Work Done by the Pressure from Combustion at 5000RPM.....	168
Figure 16-133: Work Done by the Pressure from Combustion at 5500RPM.....	168
Figure 16-134: Work Done by the Pressure from Combustion at 6000RPM.....	169
Figure 16-135: Work Done by the Pressure from Combustion at 6500RPM.....	169
Figure 16-136: Side Force Acting on the Piston Skirt at 1000RPM.....	170
Figure 16-137: Side Force Acting on the Piston Skirt at 1250RPM.....	170
Figure 16-138: Side Force Acting on the Piston Skirt at 1500RPM.....	171
Figure 16-139: Side Force Acting on the Piston Skirt at 2000RPM.....	171
Figure 16-140: Side Force Acting on the Piston Skirt at 2500RPM.....	172
Figure 16-141: Side Force Acting on the Piston Skirt at 3000RPM.....	172
Figure 16-142: Side Force Acting on the Piston Skirt at 3500RPM.....	173
Figure 16-143: Side Force Acting on the Piston Skirt at 4500RPM.....	173
Figure 16-144: Side Force Acting on the Piston Skirt at 5000RPM.....	174
Figure 16-145: Side Force Acting on the Piston Skirt at 5500RPM.....	174
Figure 16-146: Side Force Acting on the Piston Skirt at 6000RPM.....	175
Figure 16-147: Side Force Acting on the Piston Skirt at 6500RPM.....	175
Figure 16-148: Friction Force from the Piston Skirt against the Cylinder Wall at 1000RPM	176
Figure 16-149: Friction Force from the Piston Skirt against the Cylinder Wall at 1250RPM	176
Figure 16-150: Friction Force from the Piston Skirt against the Cylinder Wall at 1500RPM	177
Figure 16-151: Friction Force from the Piston Skirt against the Cylinder Wall at 2000RPM	177
Figure 16-152: Friction Force from the Piston Skirt against the Cylinder Wall at 2500RPM	178
Figure 16-153: Friction Force from the Piston Skirt against the Cylinder Wall at 3000RPM	178
Figure 16-154: Friction Force from the Piston Skirt against the Cylinder Wall at 3500RPM	179
Figure 16-155: Friction Force from the Piston Skirt against the Cylinder Wall at 4500RPM	179
Figure 16-156: Friction Force from the Piston Skirt against the Cylinder Wall at 5000RPM	180
Figure 16-157: Friction Force from the Piston Skirt against the Cylinder Wall at 5500RPM	180
Figure 16-158: Friction Force from the Piston Skirt against the Cylinder Wall at 6000RPM	181
Figure 16-159: Friction Force from the Piston Skirt against the Cylinder Wall at 6500RPM	181
Figure 16-160: Work Done Against Combustion by the Piston Skirt at 1000RPM	182
Figure 16-161: Work Done Against Combustion by the Piston Skirt at 1250RPM	182
Figure 16-162: Work Done Against Combustion by the Piston Skirt at 1500RPM	183
Figure 16-163: Work Done Against Combustion by the Piston Skirt at 2000RPM	183
Figure 16-164: Work Done Against Combustion by the Piston Skirt at 2500RPM	184
Figure 16-165: Work Done Against Combustion by the Piston Skirt at 3000RPM	184
Figure 16-166: Work Done Against Combustion by the Piston Skirt at 3500RPM	185
Figure 16-167: Work Done Against Combustion by the Piston Skirt at 4500RPM	185

Figure 16-168: Work Done Against Combustion by the Piston Skirt at 5000RPM	186
Figure 16-169: Work Done Against Combustion by the Piston Skirt at 5500RPM	186
Figure 16-170: Work Done Against Combustion by the Piston Skirt at 6000RPM	187
Figure 16-171: Work Done Against Combustion by the Piston Skirt at 6500RPM	187
Figure 16-172: Radial Force on the Top Piston Ring at 1000RPM	188
Figure 16-173: Radial Force on the Top Piston Ring at 1250RPM	188
Figure 16-174: Radial Force on the Top Piston Ring at 1500RPM	189
Figure 16-175: Radial Force on the Top Piston Ring at 2000RPM	189
Figure 16-176: Radial Force on the Top Piston Ring at 2500RPM	190
Figure 16-177: Radial Force on the Top Piston Ring at 3000RPM	190
Figure 16-178: Radial Force on the Top Piston Ring at 3500RPM	191
Figure 16-179: Radial Force on the Top Piston Ring at 4500RPM	191
Figure 16-180: Radial Force on the Top Piston Ring at 5000RPM	192
Figure 16-181: Radial Force on the Top Piston Ring at 5500RPM	192
Figure 16-182: Radial Force on the Top Piston Ring at 6000RPM	193
Figure 16-183: Radial Force on the Top Piston Ring at 6500RPM	193
Figure 16-184: Friction Force from the Piston Rings against the Cylinder Wall at 1000RPM	194
Figure 16-185: Friction Force from the Piston Rings against the Cylinder Wall at 1250RPM	194
Figure 16-186: Friction Force from the Piston Rings against the Cylinder Wall at 1500RPM	195
Figure 16-187: Friction Force from the Piston Rings against the Cylinder Wall at 2000RPM	195
Figure 16-188: Friction Force from the Piston Rings against the Cylinder Wall at 2500RPM	196
Figure 16-189: Friction Force from the Piston Rings against the Cylinder Wall at 3000RPM	196
Figure 16-190: Friction Force from the Piston Rings against the Cylinder Wall at 3500RPM	197
Figure 16-191: Friction Force from the Piston Rings against the Cylinder Wall at 4500RPM	197
Figure 16-192: Friction Force from the Piston Rings against the Cylinder Wall at 5000RPM	198
Figure 16-193: Friction Force from the Piston Rings against the Cylinder Wall at 5500RPM	198

Figure 16-194: Friction Force from the Piston Rings against the Cylinder Wall at 6000RPM	199
Figure 16-195: Friction Force from the Piston Rings against the Cylinder Wall at 6500RPM	199
Figure 16-196: Work Done by the Piston Rings against Combustion at 1000RPM	200
Figure 16-197: Work Done by the Piston Rings against Combustion at 1250RPM	200
Figure 16-198: Work Done by the Piston Rings against Combustion at 1500RPM	201
Figure 16-199: Work Done by the Piston Rings against Combustion at 2000RPM	201
Figure 16-200: Work Done by the Piston Rings against Combustion at 2500RPM	202
Figure 16-201: Work Done by the Piston Rings against Combustion at 3000RPM	202
Figure 16-202: Work Done by the Piston Rings against Combustion at 3500RPM	203
Figure 16-203: Work Done by the Piston Rings against Combustion at 4500RPM	203
Figure 16-204: Work Done by the Piston Rings against Combustion at 5000RPM	204
Figure 16-205: Work Done by the Piston Rings against Combustion at 5500RPM	204
Figure 16-206: Work Done by the Piston Rings against Combustion at 6000RPM	205
Figure 16-207: Work Done by the Piston Rings against Combustion at 6500RPM	205

2 Table of Tables

Table 10-1: Main Difference between the Two Test Procedures (ACEA, 2017).....	10
Table 10-2: Engine Displacement for the VW Passat	11
Table 10-3: Gear Efficiencies and Ratios for the VW Passat	11
Table 10-4: Engine Displacement for the Ford Fox Engine	11
Table 10-5: Gear Efficiencies and Ratios for the Ford Fox (Ford Motor Company, 2012)....	11
Table 10-6: NEDC Testing Parameters (Barlow, et al., 2009)	13
Table 10-7: Fuel Flow (g/s) for the VW Passat	17
Table 10-8: Fuel Flow (g/s) for the Ford Fox	18
Table 10-9: Inputs for Time at 150s	20
Table 10-10: Distance and Speed Specifications for each Urban, Rural and Motorway Part of the RDE Test (ICCT, 2017)	28
Table 10-11: Boundary Conditions for RDE tests (ICCT, 2017)	28
Table 10-12: Process of Implementing the WLTP (Lloyd, 2018)	29
Table 10-13: Cumulative Fuel Consumption by the VW Passat in the NEDC	38
Table 10-14: Cumulative Fuel Consumption by the Ford Fox in the NEDC	39

Table 10-15: Maximum; Power, Engine RPM and BMEP Produced by the VW Passat in the NEDC	41
Table 10-16: Maximum; Power, Engine RPM and BMEP Produced by the Ford Fox in the NEDC	42
Table 10-17: Cumulative Fuel Used in the NEDC	47
Table 10-18: Cumulative Fuel Used in the WLTP.....	48
Table 10-19: Maximum; Power, Engine RPM and BMEP in the NEDC test.....	49
Table 10-20: Maximum; Power, Engine RPM and BMEP in the WLTP	49
Table 11-1: Engine Parameters for the Ford Fox	51
Table 11-2: Connecting Rod Parameters	51
Table 11-3: Initial Pressure Values at 4000RPM and 25 Degrees ATDC	55
Table 11-4: Calculated Polytropic Index for the Range of RPM.....	59
Table 12-1: Ford Fox Engine Parameters	63
Table 12-2: Ford Fox Connecting Rod Parameters	63
Table 12-3: - Piston Ring Measurements	63
Table 12-4: Given Values for 4000RPM.....	63
Table 12-5: Peak Cylinder Gas Forces and the Degrees ATDC at which they occur for the RPM Range	68
Table 12-6: Measured Frictional Forces.....	78
Table 12-7: Work Done by the Piston Skirt at Each RPM.....	79
Table 12-8: Work Done by the Piston Rings at Each RPM.....	80
Table 12-9: The Combined FMEP of the Piston Rings and Skirt	81
Table 12-10: Total Engine FMEP	82
Table 12-11: Percentage of Mechanical Loss from Frictional Forces	82
Table 16-1: RPM to Rads/s Conversion Chart.....	88

3 Table of Equations

Equation 10-1: Vehicle Acceleration.....	14
Equation 10-2: Wheel Force.....	14
Equation 10-3: Wheel Force.....	14
Equation 10-4: Wheel Velocity	15
Equation 10-5: Wheel Torque	15
Equation 10-6: Wheel Acceleration	15
Equation 10-7: Wheel RPM.....	15
Equation 10-8: Drive Line Torque	15

Equation 10-9: Differential Acceleration	15
Equation 10-10: Differential Torque Output	15
Equation 10-11: Differential Torque Input.....	15
Equation 10-12: Gear Box Torque Output	16
Equation 10-13: Gear Box Torque Input.....	16
Equation 10-14: Engine Acceleration	16
Equation 10-15: Engine Torque Lost.....	16
Equation 10-16: Engine Torque Output.....	16
Equation 10-17: Engine RPM.....	16
Equation 10-18: Engine Internal Torque.....	17
Equation 10-19: Engine BMEP.....	17
Equation 10-20: Power.....	17
Equation 10-21: Polynomial Equation for Fuel Flow at 4000RPM below 10 Bar for the VW Passat.....	19
Equation 10-22: Polynomial Equation for Fuel Flow at 4000RPM above 10 Bar for the VW Passat.....	19
Equation 10-23: Polynomial Equation for the Fuel Flow at 4000RPM for the Ford Fox.....	20
Equation 10-24: Interpolation Equation	20
Equation 10-25: Maximum Allowed Engine Speed	29
Equation 10-26: Minimum Engine Speed for First Gear	30
Equation 10-27: Minimum Engine Speed for Second Gear	30
Equation 10-28: Minimum Engine Speed for Third Gear and Above	30
Equation 11-1: Piston Displacement	52
Equation 11-2: Connecting Rod Angle	52
Equation 11-3: Bore Area.....	52
Equation 11-4: Cylinder Volume.....	52
Equation 11-5: Derivation of the Polytropic Index.....	53
Equation 11-6: Polytropic Index	53
Equation 11-7: Pressure due to Piston Motion	54
Equation 11-8: Pressure due to Combustion.....	54
Equation 11-9 Mass Fraction Burned	54
Equation 11-10: Work Done by the Cylinder Pressure	54
Equation 11-11: Indicated Mean Effective Pressure	54
Equation 12-1: Gas Force	64
Equation 12-2: First Order Inertial Force	64
Equation 12-3: Second Order Inertial Force	64
Equation 12-4: Inertial Acceleration.....	64

Equation 12-5: Inertia Force.....	65
Equation 12-6: Reciprocating Mass.....	65
Equation 12-7: Net Force Acting on Piston.....	65
Equation 12-8: Force in the Connecting Rod.....	65
Equation 12-9: Work Done by the Combustion Pressure	65
Equation 12-10: Side Force on the Piston	65
Equation 12-11: Friction Force on the Piston Skirt	65
Equation 12-12: Work Done by the Piston Skirt.....	66
Equation 12-13: Radial Piston Ring Force.....	66
Equation 12-14: Piston Ring Friction Force	66
Equation 12-15: Total Frictional Force from the Piston Rings.....	66
Equation 12-16: Work Done by the Piston Rings.....	66
Equation 12-17: FMEP	66
Equation 12-18: Reciprocating Mass Value.....	69
Equation 16-1: Conversion Degrees to Radians.....	88
Equation 16-2: Conversion RPM to Radians per Second	88
Equation 16-3: Conversion Kilometres per Hour to Metres per Second.....	88

4 Acknowledgements

I would like to thank my family Nick Mellors, Rachel Mellors, Natalie Mellors, Derek Wilkinson and Jane Goodeve Hampson, for which without their unwavering support and love I would never have got this far. I would also like to thank Malcolm McDonald who without his support though my university career this report would not have been written.

5 Abstract

This project looks in to the NEDC and WLTP test procedures, creating mathematical models of both with focus on optimising fuel usage. The WLTP model will be focused around a gear selector as, unlike the NEDC, this is not predefined, but user selected within legislation. These models will be paired with combustion and friction models to look in-to the efficiency of the engine and where the losses occur. These models give an insight into what is occurring in the engine during running in the drive cycle.

6 Nomenclature

<i>Unit Symbol</i>	<i>Quantity</i>	<i>Unit</i>
A_B	Bore Area	m^2
A_{fr}	Frontal Area	m^2
Acc_{Dif}	Differential Acceleration	$Rads/s^2$
Acc_{Eng}	Engine Acceleration	$Rads/s^2$
Acc_V	Vehicle Acceleration	m/s^2
Acc_{Wh}	Wheel Acceleration	$Rads/s^2$
CD	Drag Coefficient	-
CV	Clearance Volume	m^3
D_B	Bore diameter	m
D_{PR}	Inner Diameter of the Piston Ring	m
F_D	Drag Force	N
F_F	Friction Force	N
F_{FPR}	Friction Force of the Piston Ring	N
F_{FPS}	Friction Force of the Piston Skirt	N
F_G	Gradient Force	N
F_{Gas}	Gas Force	N
$F_{Inertia}$	Inertia Force	N
F_{Net}	Net Force	N
F_R	Rolling Force	N
F_{Rad}	Radial Force	N
F_{Rod}	Connecting Rod Force	N
F_{Side}	Piston Side Force	N
F_{WF}	Net Forward Force	N
F_{Wh}	Wheel Force	N
FD_{in}	Final Drive	-
FD_R	Final Drive Ratio	-
g	Gravity	m/s^2
G_R	Gear Ratio	-
I_{Eng}	Engine Inertia	Kgm^2
I_{Wh}	Wheel Inertia	Kgm^2
L_{ConRod}	Connecting Rod Length	m
M_{ConRod}	Connecting Rod Mass	kg

M_G	Gudgeon Pin Mass	kg
M_P	Piston Mass	kg
M_{PR}	Piston Ring Mass	kg
M_{Rec}	Reciprocating Mass	kg
M_V	Vehicle Mass	kg
n	Polytropic Index	-
P	Pressure	Bar
P_c	Pressure due to Combustion	Bar
P_{Cyl}	Cylinder Pressure	Bar
P_{CylMax}	Maximum Cylinder Pressure	Bar
P_{mp}	Pressure due to Piston Motion	Bar
P_w	Power	Kw
R_{Crank}	Crank Throw	m
RPM_{Eng}	Engine RPM	Revs/min
RPM_{idle}	Idle RPM	Revs/min
RPM_{Max}	Maximum RPM	Revs/min
RPM_{Min}	Minimum RPM	Revs/min
RPM_{Wh}	Wheel RPM	Revs/min
RRc	Rolling Resistance Coefficient	-
s	RPM at which Maximum Power is Reached	
S	Displacement	m
t_{PR}	Piston Ring Thickness	m
T_{Din}	Differential Input Torque	Nm
T_{DL}	Drive Line Torque	Nm
T_{Do}	Differential Output Torque	Nm
T_{Engo}	Engine Output Torque	Nm
T_{GBin}	Gearbox Input Torque	Nm
T_{GBo}	Gearbox Output Torque	Nm
T_{inEng}	Internal Engine Torque	Nm
T_{Ls}	Engine Torque Lost	Nm
T_{Wh}	Wheel Torque	Nm
v_s	Vehicle Speed	m/s
v_{Wh}	Wheel Velocity	Rads/s
V	Volume	m^3
V_{Cyl}	Cylinder Volume	m^3

WD_{PR}	Work Done by the Piston Ring	Nm
WD_{Press}	Work Done by the Pressure	Nm
WD_{PS}	Work Done by the Piston Skirt	Nm
W_{rr}	Wheel Rolling Radius	m
$\alpha_{Inertia}$	Rotational Acceleration	Rads/s ²
η_G	Gear Efficiency	%
θ	Crank Angle	Radians
μ_{Cyl}	Cylinder Wall Coefficient of Friction	-
ρ	Air Density	Kg/m ³
ϕ	Connecting Rod Angle	Radians
ω_{Eng}	Rotational Engine Speed	Rads/s

7 Abbreviations

- ATDC –After Top Dead Centre
- BMEP- Brake Mean Effective Pressure
- BTDC – Before Top Dead Centre
- CO – Carbon Monoxide
- CO₂ – Carbon Dioxide
- EUDC – Extra Urban Driving Cycle
- FMEP – Frictional Mean Effective Pressure
- HC – Hydrocarbons
- IMEP – Indicated Mean Effective Pressure
- MFB – Mass Fraction Burned
- NEDC- New European Drive Cycle
- NO_x – Nitrous Oxides
- RDE – Real Driving Emissions
- RPM- Revolutions Per Minutes
- TDC – Top Dead Centre
- UDC – Urban Driving Cycle
- UTS – Ultimate Tensile Strength
- WLTP – Worldwide Harmonized Light-Duty Vehicles Test Procedure

8 Constants

<i>Quantity</i>	<i>Value</i>	<i>Units</i>
Gravity	9.81	m/s ²
Air Density	1.2	Kg/m ³
Vehicle Mass	1650	Kg
Wheel Rolling Radius	0.32	m
Engine Idle	800	RPM
Engine Inertia	0.15	Kgm ²
Rolling Resistance Coefficient	0.015	-
Frontal Area	2.235	m ²
Drag Coefficient	0.4	-
Wheel Inertia	0.8	Kgm ²

9 Introduction

Vehicle testing for Emissions has been a prominent feature in the automotive industry since the mid- 1960s when California introduced the first legislation to control emissions (Bosch, 2014, p. 532). This followed studies in the 1950's and 1960's in-to the sources of air pollution and the causes of smog when interactions of hydrocarbons and oxides of nitrogen discovered by Dr Arie Haagen-Smit, were identified (Kovarik, 2017).

Testing procedures were developed, as the technology to detect emissions and manufacturing advanced, with the developments leading to more stringent limits on allowable emissions.

Many procedures have been criticised as not being realistic and outdated, such as the NEDC which came under criticism when the 'VW Emissions Scandal' emerged (Tovey & Huggler, 2015). This prompted the United Nations Economic Commission for Europe World Forum for Harmonization of Vehicle Regulations to develop the WLTP and accompanying RDE.

The development of a new testing procedure is long overdue as the gap between real world and official miles per gallon figures has grown rapidly since 2001, when it was only 8%. Only 2% of drivers can now match the official figures on the road compared with 14% in 2001. (Webster, 2014)

This project looks in to the NEDC and WLTP test procedures, creating mathematical models of both with focus on optimising fuel usage. The WLTP model will be focused around a gear selector as, unlike the NEDC, this is not predefined, but user selected within legislation. These models will be paired with combustion and friction models to look in-to the efficiency of the engine and where the losses occur. These models give an insight into what is occurring in the engine during running in the drive cycle.

Emissions and fuel consumption are intrinsically linked, establishing how losses in the engine affect fuel consumption gives an insight into where future development needs to be focused and what can be done to reduce emissions, also preventing more scandals from damaging the industry further.

The chapters in this dissertation will cover; Drive cycles, specifically the NEDC and WLTP, how these were modelled and the effectiveness and accuracy of them; the internal combustion process and the losses that occur during engine running; the friction rubbing forces acting on the cylinder from the pistons and where losses occur; results and analysis of each of the models focus on efficiency, fuel consumption; a summary of the study and assessment of the projects' successes and opportunities for further developments.

10 Drive Cycles

Vehicle emissions are responsible for 12% of Europe's emissions. In the UK, London traffic alone is responsible for 80% of pollution and 25% nation-wide (Foster, 1997). By 2021, all new cars must meet an EU emissions limit of 95 grams of CO₂ per km (Nielsen, 2015).

The drive cycle is a fundamental part in the testing of vehicles; it provides a standardized test that can be used for performance comparison. The drive cycle tests have predetermined gear shifts, braking and idle phases along with corresponding velocities and accelerations; they are intended as a replication of 'typical' driving. The emissions are collected and analysed to calculate the pollution mass and emissions (CO, HC & NO_x and other pollutants) that have been produced. In Europe the test procedure was the NEDC up until September 2017 (Lea, 2017) when it was superseded by the WLTP.

Table 10-1: Main Difference between the Two Test Procedures (ACEA, 2017)

	<i>NEDC</i>	<i>WLTP</i>
Test Cycle	Single Test Cycle	Dynamic Cycle more Representative of real driving
Cycle Time	20 minutes	30 minutes
Cycle Distance	11 Kilometres	23.25 Kilometres
Driving Phases	2 Phases, 66% urban and 34% non-urban driving	4 more dynamic phases, 52% urban and 48% non-urban
Average Speed	34 km/h	46.5 km/h
Maximum Speed	120km/h	131km/h
Influence of optional equipment	Impact on CO ₂ and Fuel performance not considered under NEDC	Additional Features (differing per car) are taken in to account
Gear Shifts	Vehicles have fixed gear shift points	Different gear shift points for each vehicle
Test temperatures	Measurements 10-30°	Measurements at 23°C, CO ₂ values corrected to 14°C

10.1 Vehicle Constants – 0

10.1.1 VW Passat

Table 10-2: Engine Displacement for the VW Passat

Engine Displacement	1984	cc
Engine Displacement	0.001984	m^3

Table 10-3: Gear Efficiencies and Ratios for the VW Passat

Gearing		
Gear	Efficiency	Ratio
1	94%	3.36
2	94%	2.09
3	95%	1.47
4	97%	1.1
5	97%	0.87
6	96%	0.73
Final Drive	94%	3.94

10.1.2 Ford Fox

Table 10-4: Engine Displacement for the Ford Fox Engine

Engine Displacement	999	cc
Engine Displacement	0.000999	m^3

Table 10-5: Gear Efficiencies and Ratios for the Ford Fox (Ford Motor Company, 2012)

Gearing		
Gear	Efficiency	Ratio
1	94%	3.727
2	94%	2.048
3	95%	1.357
4	97%	1.032
5	97%	0.821
6	96%	0.69
Final Drive	94%	4.07

10.2 NEDC

The NEDC test was implemented in 1996. It was developed from the UDC which had a maximum speed of 50km/hr and lasted 195 seconds. The NEDC was intended to more accurately replicate an urban driving environment, involving accelerating and slowing down several times, steady speeds and engine idling to simulate a vehicle in traffic. The test is 1km long with an average speed of 18km/hr. It uses four repetitions of the UDC which can be seen in Figure 10-1, for the first 800 seconds. This is supplemented by the EUDC which involves 50% steady state speed driving, accelerating, slowing down and engine idling, with an average speed of 62km/hr and a maximum of 120km/hr it is supposed to simulate motorway driving, while lasting 400 seconds.

The official test was conducted in a laboratory test cell on a rolling road, on which the rolling resistance could be changed to simulate found frictional forces. The tests were conducted between 20-30°C with no inclines, and simulated wind resistance. It was not mandatory for non-essential electrical equipment such as air conditioning, fans, lights and the radio to be running.

Criticisms of the testing procedure came to the fore front of the industry in the wake of the VW emissions scandal in 2015, which revealed just how the test programme could be manipulated to show false readings. Many car manufacturers were also taking advantage of loopholes in official tests by, for example; over inflating tyres and altering wheel alignment to reduce rolling resistance, removing the alternator to prevent battery charging and so reduce energy use, using higher gears to save fuel, taping over door gaps to reduce drag, removing fittings such as car stereos and spare tyre to reduce weight. (Webster, 2014)

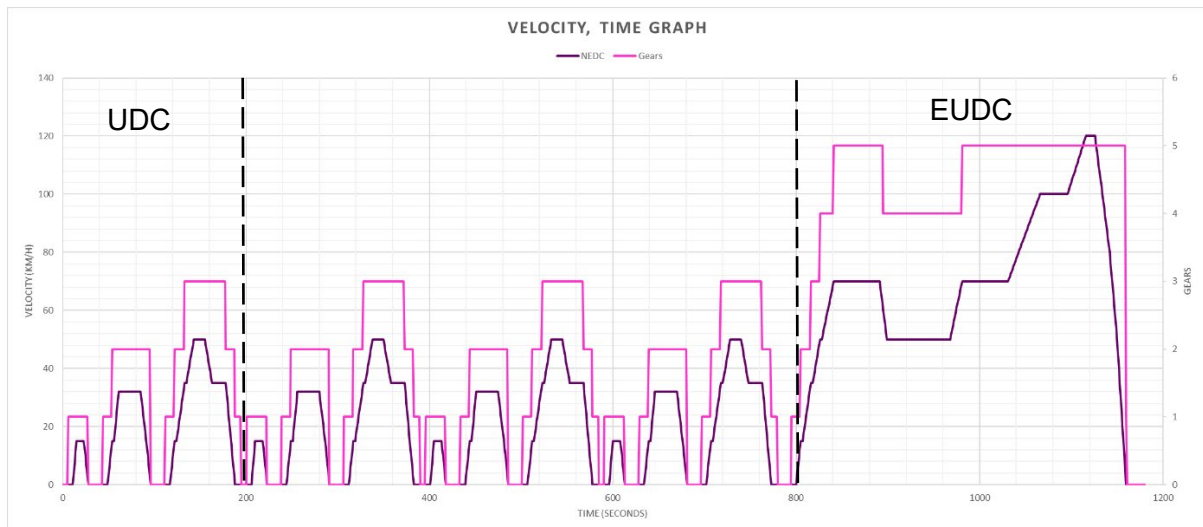


Figure 10-1: NEDC Drive Cycle

Table 10-6: NEDC Testing Parameters (Barlow, et al., 2009)

	<i>Value</i>	<i>Units</i>
Total Distance	11016.63	m
Total Time	1180	s
Time Stationary	241	S
Time Accelerating	278	S
Time Decelerating	204	S
Time braking	200	S
Average speed (cycle)	33.6	Km/hr
Average speed (driving)	42.24	Km/hr
Maximum speed	120.09	Km/hr
Number of stops	14	-

10.2.1 The Model

The drive cycle model developed by the author uses the given gears and velocities per second, with the velocity converted to meters per second (Equation 16-3). This was then used to find; the vehicle acceleration, force, torque, velocity, acceleration and RPM at the wheels. This was then worked through the drive line to find the input and output torque of the differential, gear box and engine; as well as the engine angular acceleration, RPM and BMEP. These were then used along with fuel polynomials (16.2.1 Fuel) to find the fuel flow for each second and subsequently power.

As the data is for every second, the acceleration can be found with the current velocity at that second and the velocity from the previous second.

$$Acc_V = v_S - v_{S1}$$

Equation 10-1: Vehicle Acceleration

The forward force of a vehicle is calculated from the wheel force, aerodynamic force, wheel rolling force and gradient force. As the NEDC test is conducted on a flat surface, the gradient force is negated.

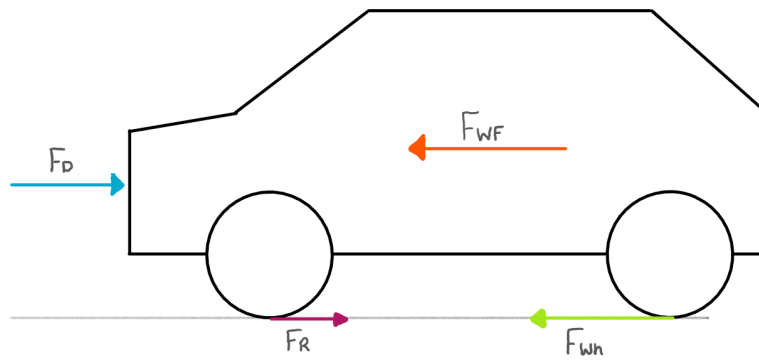


Figure 10-2: Forces acting on a Flat Road

$$F_{Wh} = F_{WF} + F_D + F_R + F_G$$

Equation 10-2: Wheel Force

The net forward force is found from the vehicle acceleration and mass. The aerodynamic force incorporates the frontal area of the vehicle, found in the design process; air density, the drag coefficient and the velocity. The rolling force is a constant force acting against the motion of the wheels calculated from the vehicle mass, gravity and the rolling resistance.

$$F_{Wh} = (Acc_V \times M_V) + \left(\frac{1}{2} \times A_{FR} \times \rho \times CD \times v_S^2 \right) + (M_V \times g \times RRC)$$

Equation 10-3: Wheel Force

As the given velocity is that of the linear velocity acting at the wheels, the angular velocity of the wheels needs to be found, as well as the wheel torque and wheel acceleration, for the calculations for the rest of the driveline.

$$v_{Wh} = \frac{v_s}{Wrr}$$

Equation 10-4: Wheel Velocity

$$T_{Wh} = F_{Wh} \times Wrr$$

Equation 10-5: Wheel Torque

$$Acc_W = v_{Wh} - v_{Wh1}$$

Equation 10-6: Wheel Acceleration

$$RPM_{Wh} = \frac{v_{Wh}}{(2 \times \pi)} \times 60$$

Equation 10-7: Wheel RPM

When the variables were calculated at the wheel, they could be translated along the driveline, for the model the efficiency between: the gearbox and differential, wheel and driveline, differential and driveline, engine and gearbox, was presumed to be 100%, as the actual vehicle efficiencies for these values was unknown.

$$T_{DL} = T_{Wh} + (2 \times I_W \times Acc_W)$$

Equation 10-8: Drive Line Torque

$$Acc_{Dif} = Acc_W$$

Equation 10-9: Differential Acceleration

$$T_{Do} = T_{DL}$$

Equation 10-10: Differential Torque Output

$$T_{Din} = \frac{T_{Do}}{FD_{in}}$$

Equation 10-11: Differential Torque Input

$$T_{GBo} = T_{Din}$$

Equation 10-12: Gear Box Torque Output

The torque input for the gearbox was calculated from the gearbox torque output, gear ratio and gear efficiency (see Table 10-3 & Table 10-5). Working backwards through the driveline, the input torque of the gearbox is dependent on; the gear prescribed at that second and the corresponding ratios and efficiencies.

$$T_{GBin} = \frac{T_{GBo}}{G_R \times \eta_G}$$

Equation 10-13: Gear Box Torque Input

As the driveline efficiency is assumed to be 100% the engine angular acceleration can be calculated from the wheel acceleration, the final drive ratio from the differential, and the ratio of the gear selected.

$$Acc_{Eng} = G_R \times FD_R \times Acc_W$$

Equation 10-14: Engine Acceleration

The torque 'lost' from the engine, is that used to accelerate the engine found using the engine inertia and engine angular acceleration.

$$T_{Ls} = Acc_{Eng} \times I_{Eng}$$

Equation 10-15: Engine Torque Lost

$$T_{Engo} = T_{GBin}$$

Equation 10-16: Engine Torque Output

With the efficiency assumption for the driveline, the engine RPM is found from the wheel RPM, final drive ratio and the ratio of the selected gear.

$$RPM_{Eng} = RPM_W \times G_R \times FD_R$$

Equation 10-17: Engine RPM

The torque produced by the engine, is the summation of the torque lost from accelerating the engine and the output torque of the engine.

$$T_{InEng} = T_{Ls} + T_{Engo}$$

The BMEP is a measure of useful pressure produced in the cylinder by the engine to overcome mechanical inefficiencies and produce the power required, calculated using: the internal engine torque, engine displacement and number of strokes in one full cycle.

$$BMEP = \left(\frac{T_{InEng} \times 4 \times \pi}{Engine\ Displacement \times 10^{-6}} \right) \times 10^{-5}$$

The power produced by the engine is calculated from the torque produced inside the engine and the running engine RPM.

$$P_w = \frac{2 \times \pi \times T_{InEng} \times RPM_{Eng}}{60000}$$

10.2.2 Fuel Flow

The data for the fuel flow in grams per second was provided for both engines.

Table 10-7: Fuel Flow (g/s) for the VW Passat

Fuel Flow (g/s)		RPM														
		500	850	1000	1250	1500	1750	2000	2500	3000	3500	4000	4500	5000	6000	7000
TORQUE (NM)	315.76	1.098674	1.97057	2.369509	3.066999	3.800913	4.565326	5.354184	6.967979	8.628862	10.44457	12.57564	15.12651	18.15548	25.88612	34.49552
	299.98	1.027175	1.836469	2.206691	2.855742	3.546792	4.279078	5.04337	6.53651	7.95915	9.427374	11.25446	13.82776	16.99886	23.94106	31.58216
	284.19	0.962433	1.711496	2.051422	2.64346	3.271133	3.946159	4.695541	6.068165	7.311274	8.513743	9.983747	12.59976	15.8435	21.83464	28.70197
	268.40	0.904734	1.602721	1.918558	2.46768	3.045857	3.651059	4.274751	5.514681	6.681385	7.875844	9.384215	11.56413	14.23041	19.39536	25.92094
	252.61	0.855388	1.504334	1.79504	2.296054	2.815921	3.351104	3.896301	5.001389	6.124177	7.342631	8.800455	10.58919	12.65105	16.63801	23.30943
	236.82	0.811612	1.411349	1.676206	2.127843	2.591441	3.066131	3.551477	4.554075	5.62334	6.802496	8.13193	9.611797	11.20573	14.49508	20.89782
	221.03	0.771273	1.324892	1.565783	1.972422	2.386903	2.81131	3.246784	4.160973	5.153364	6.238872	7.417026	8.676375	10.02393	13.26888	18.56689
	205.25	0.733705	1.247665	1.468174	1.836465	2.208632	2.59137	2.984275	3.810575	4.706378	5.675998	6.71421	7.821488	9.024059	12.00447	16.0094
	189.46	0.696535	1.178807	1.384294	1.725539	2.067112	2.411631	2.761464	3.493452	4.285351	5.139999	6.057395	7.045566	8.125834	10.73095	13.80802
	173.67	0.655856	1.108972	1.301017	1.616757	1.929041	2.240464	2.55343	3.199896	3.894332	4.644657	5.455972	6.335692	7.292778	9.504593	11.92185
	157.88	0.606346	1.022747	1.198899	1.489054	1.775518	2.059591	2.342908	2.91922	3.530918	4.191203	4.907585	5.683324	6.517444	8.366423	10.30011
	142.09	0.549309	0.925744	1.085071	1.348032	1.608183	1.866248	2.123244	2.642487	3.188277	3.774229	4.406744	5.084729	5.802295	7.334517	8.895117
	126.31	0.488919	0.8241	0.966104	1.200771	1.433529	1.665143	1.896544	2.366674	2.861242	3.387967	3.947749	4.53657	5.148107	6.408468	7.665798
	110.52	0.428979	0.722722	0.847173	1.052981	1.25779	1.462799	1.669216	2.094455	2.54962	3.029468	3.524157	4.032607	4.549641	5.57606	6.577583
	94.73	0.371886	0.62592	0.733317	0.910647	1.087542	1.266928	1.452684	1.832666	2.256306	2.695774	3.127243	3.563544	3.995653	4.819039	5.601339
	78.94	0.317849	0.536582	0.629158	0.781615	0.932963	1.09252	1.257957	1.599179	1.979266	2.370064	2.751784	3.118087	3.470264	4.117653	4.712267
	63.15	0.265944	0.452501	0.532894	0.667924	0.806301	0.948591	1.094301	1.365217	1.691594	2.037879	2.3755	2.67567	2.956615	3.453975	3.888916
	47.36	0.217516	0.371758	0.438161	0.549639	0.664048	0.78506	0.910231	1.153974	1.417834	1.699804	1.970855	2.217807	2.442176	2.813043	3.112365
	31.58	0.170399	0.292735	0.345907	0.435556	0.527139	0.623157	0.721758	0.927	1.138434	1.353202	1.55996	1.750287	1.921084	2.182885	2.365811
	23.68	0.147183	0.25407	0.300918	0.380513	0.462454	0.546402	0.632431	0.812593	0.996695	1.179807	1.35492	1.515757	1.658919	1.868607	1.996862
	15.79	0.124133	0.215716	0.256212	0.325423	0.396716	0.469477	0.543891	0.698289	0.854352	1.006487	1.150186	1.281142	1.396225	1.554408	1.629087
	7.89	0.101219	0.177511	0.211569	0.27009	0.330489	0.392326	0.455437	0.584187	0.711742	0.83323	0.945691	1.04655	1.133226	1.240178	1.261908
	0.00	0.078368	0.139343	0.166917	0.214657	0.264189	0.315079	0.366909	0.470191	0.569067	0.66	0.741301	0.811986	0.870115	0.925914	0.89493

Table 10-8: Fuel Flow (g/s) for the Ford Fox

Fuel Flow (g/s)		RPM															
		800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
Torque (Nm)	202		1.689921	2.098446	2.523423	2.851448	3.21414	3.622518	4.008861	5.015116	5.725518	6.421229	7.168355	8.652324	9.549874	10.5767	11.17163
	200						3.184644	3.597124	3.954625	4.930555	5.483534						
	190						2.99004	3.342655	3.711297	4.70724	5.155194						
	180					2.530815	2.792876	3.124314	3.478005	4.238008	4.949841	5.721887					
	170				2.117739	2.436945	2.610024	2.926263	3.251403	3.901684	4.737158	5.253674	5.948213				
	160				1.96943	2.244726	2.428567	2.713289	3.02291	3.6247	4.181576	4.853412	5.50407	6.574074			
	150				1.816233	2.070833	2.26631	2.532909	2.831797	3.328779	3.88663	4.48026	5.042256	5.827218	6.796312		
	140			1.448769	1.683545	1.899995	2.107892	2.348471	2.62163	3.060435	3.590462	4.148997	4.621806	5.251405	6.011728	7.105584	
	130			1.315033	1.512808	1.710015	1.931614	2.14669	2.374811	2.790781	3.300912	3.839026	4.27466	4.804455	5.474167	6.362074	
	120			1.207332	1.372876	1.598634	1.785123	1.965066	2.162463	2.540501	3.042109	3.561868	3.953694	4.406956	5.003161	5.68	6.468714
	110		0.923134	1.085922	1.25271	1.457015	1.606285	1.784752	1.979058	2.337432	2.770681	3.250967	3.6271	4.04611	4.529915	5.177868	5.802783
	100		0.836013	0.951932	1.147554	1.293507	1.436406	1.603521	1.798416	2.142392	2.509638	2.910046	3.296054	3.710279	4.164501	4.653048	5.331981
	90		0.715236	0.833177	1.001383	1.141249	1.270774	1.426087	1.604176	1.918728	2.23485	2.610663	2.972339	3.356268	3.775409	4.206593	4.705189
	80		0.61296	0.738274	0.871268	0.998561	1.12213	1.262397	1.423607	1.697856	1.980832	2.330829	2.65674	3.042691	3.421194	3.783874	4.23382
	70		0.527788	0.643954	0.771828	0.89227	1.001819	1.128879	1.265509	1.499063	1.755321	2.059023	2.344806	2.699006	3.033862	3.384193	3.798564
	60		0.453437	0.559814	0.669159	0.771523	0.87057	0.984104	1.116138	1.318422	1.538159	1.809557	2.050675	2.364921	2.667648	2.986607	3.342131
	50		0.385427	0.479966	0.579667	0.66228	0.749328	0.842012	0.946841	1.134464	1.333722	1.559743	1.778927	2.0304	2.291036	2.590069	2.898556
	40	0.279625	0.32882	0.404771	0.485202	0.558331	0.632275	0.715236	0.796452	0.962724	1.139467	1.322261	1.523149	1.740093	1.967219	2.208889	2.482964
	30	0.219702	0.27105	0.332703	0.399637	0.46273	0.52517	0.59003	0.655589	0.790896	0.943481	1.095369	1.262528	1.446442	1.644842	1.836785	2.08401
	20	0.168669	0.215257	0.258891	0.317999	0.361429	0.413643	0.459196	0.514581	0.614356	0.740572	0.869639	1.011855	1.161808	1.323425	1.480737	1.694889
	10	0.113958	0.162577	0.189005	0.224886	0.25916	0.300022	0.333663	0.37241	0.43066	0.535322	0.663458	0.785005	0.891863	0.97769	1.094147	1.243373
	5	0.088814	0.139146	0.156007	0.189412	0.185932	0.24039	0.288241	0.306669	0.420537	0.449291	0.557342	0.653059	0.777762	0.781064	0.939423	1.02877
	2	0.084022	0.085079	0.265625	0.154654	0.202675	0.188146	0.234913	0.258803	0.308225	0.404861	0.506355	0.573053				

The data was graphed as fuel flow against BMEP for both engines and the trendline for the data found using polynomials. These polynomials were compared against the values given in the above tables, and if the fuel flow calculated from the equations didn't align, the data was split to find multiple polynomials for the RPM, to improve the accuracy of the calculated fuel flow in the drive cycle.

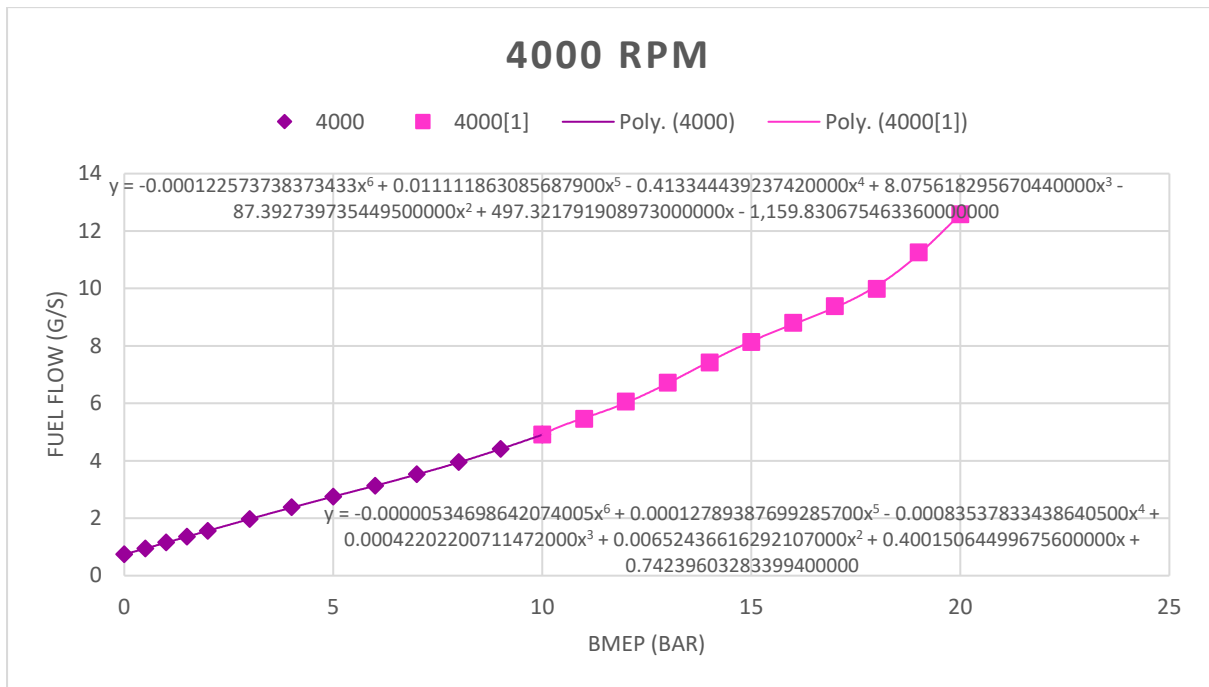


Figure 10-3: Graph at 4000RPM of the Fuel Flow (g/s) for the VW Passat

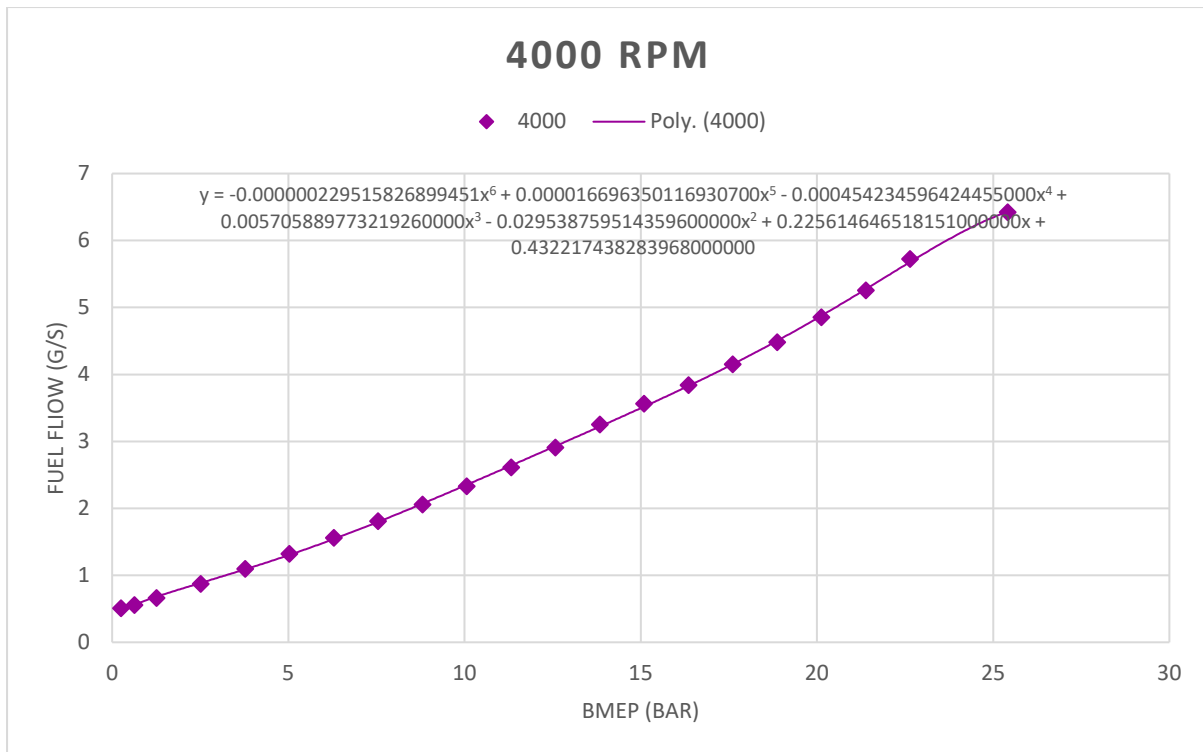


Figure 10-4: Graph at 4000RPM of the Fuel Flow (g/s) for the Ford Fox

Due to the high orders available in the equations, the trendlines were selected as polynomials, allowing for closer representation to the data. Also, for accuracy of calculations of the fuel flow in the drive cycle, the number of significant figures in the equation is important; this model uses 25 significant figures.

$$y = -0.00000534698642074005x^6 + 0.00012789387699285700x^5 - 0.00083537833438640500x^4 + 0.00042202200711472000x^3 + 0.00652436616292107000x^2 + 0.40015064499675600000x + 0.74239603283399400000$$

Equation 10-21: Polynomial Equation for Fuel Flow at 4000RPM below 10 Bar for the VW Passat

$$y = -0.000122573738373433x^6 + 0.011111863085687900x^5 - 0.413344439237420000x^4 + 8.075618295670440000x^3 - 87.392739735449500000x^2 + 497.321791908973000000x - 1,159.830675463360000000$$

Equation 10-22: Polynomial Equation for Fuel Flow at 4000RPM above 10 Bar for the VW Passat

$$\begin{aligned}
y = & -0.000000229515826899451x^6 + 0.000016696350116930700x^5 \\
& - 0.000454234596424455000x^4 + 0.005705889773219260000x^3 \\
& - 0.029538759514359600000x^2 + 0.225614646518151000000x \\
& + 0.432217438283968000000
\end{aligned}$$

Equation 10-23: Polynomial Equation for the Fuel Flow at 4000RPM for the Ford Fox

The polynomials find the values of the fuel flow at the calculated BMEP and given RPMs. This enables the values to be interpolated with regards to the RPM calculated in the drive cycle, to find the corresponding fuel flow.

$$y = \left(y_1 + \left(\left(\frac{(x - x_1)}{(x_2 - x_1)} \right) \times (y_2 - y_1) \right) \right)$$

Equation 10-24: Interpolation Equation

- $y =$ Fuel Flow for Calculated RPM
- $y_1 =$ Fuel Flow at Lower Bound RPM
- $y_2 =$ Fuel Flow at Upper Bound RPM
- $x =$ Calculated RPM
- $x_1 =$ Lower Bound RPM
- $x_2 =$ Upper Bound RPM

10.2.3 Results

The results were calculated for every second of the NEDC drive cycle using the values that can be found in 8 Constants & 10.1 Vehicle Constants – 0

Table 10-9: Inputs for Time at 150s

<i>Inputs</i>		
Time	150	s
Previous Velocity	50	Km/hr
Current Velocity	50	Km/hr
Gear	3	-

VW

$$\begin{aligned}
Acc_V &= 50 - 50 \\
Acc_V &= 0m/s^2
\end{aligned}$$

Ford Fox

$$\begin{aligned}
Acc_V &= 50 - 50 \\
Acc_V &= 0m/s^2
\end{aligned}$$

$$\begin{aligned}
F_{Wh} &= (0 \times 1650) \\
&\quad + \left(\frac{1}{2} \times 2.235 \times 1.2 \times 0.4 \right. \\
&\quad \left. \times 13.88^2 \right) \\
&\quad + (1650 \times 9.81 \times 0.015) \\
F_{Wh} &= (0) + (103.34) + (242.798) \\
F_{Wh} &= 346.269N
\end{aligned}$$

$$\begin{aligned}
v_{Wh} &= \frac{13.88}{0.32} \\
v_{Wh} &= 43.402 \text{ rads/s}
\end{aligned}$$

$$\begin{aligned}
T_{Wh} &= 346.269 \times 0.32 \\
T_{Wh} &= 110.806Nm
\end{aligned}$$

$$\begin{aligned}
Acc_W &= 43.402 - 43.402 \\
Acc_W &= 0 \text{ rads/s}^2
\end{aligned}$$

$$\begin{aligned}
RPM_{Wh} &= \frac{43.402}{(2 \times \pi)} \times 60 \\
RPM_{Wh} &= 414.459 \text{ revs/min}
\end{aligned}$$

$$\begin{aligned}
T_{DL} &= 110.806 + (2 \times 0.8 \times 0) \\
T_{DL} &= 110.806 + (0) \\
T_{DL} &= 110.806Nm
\end{aligned}$$

$$Acc_{Dif} = 0 \text{ rads/s}$$

$$T_{Do} = 110.806Nm$$

$$\begin{aligned}
T_{Din} &= \frac{110.806}{0.94 \times 3.94} \\
T_{Din} &= \frac{110.806}{3.704} \\
T_{Din} &= 29.919Nm
\end{aligned}$$

$$\begin{aligned}
F_{Wh} &= (0 \times 1650) \\
&\quad + \left(\frac{1}{2} \times 2.235 \times 1.2 \times 0.4 \right. \\
&\quad \left. \times 13.88^2 \right) \\
&\quad + (1650 \times 9.81 \times 0.015) \\
F_{Wh} &= (0) + (103.34) + (242.798) \\
F_{Wh} &= 346.269N
\end{aligned}$$

$$\begin{aligned}
v_{Wh} &= \frac{13.88}{0.32} \\
v_{Wh} &= 43.402 \text{ rads/s}
\end{aligned}$$

$$\begin{aligned}
T_{Wh} &= 346.269 \times 0.32 \\
T_{Wh} &= 110.806Nm
\end{aligned}$$

$$\begin{aligned}
Acc_W &= 43.402 - 43.402 \\
Acc_W &= 0 \text{ rads/s}^2
\end{aligned}$$

$$\begin{aligned}
RPM_{Wh} &= \frac{43.402}{(2 \times \pi)} \times 60 \\
RPM_{Wh} &= 414.459 \text{ revs/min}
\end{aligned}$$

$$\begin{aligned}
T_{DL} &= 110.806 + (2 \times 0.8 \times 0) \\
T_{DL} &= 110.806 + (0) \\
T_{DL} &= 110.806Nm
\end{aligned}$$

$$Acc_{Dif} = 0 \text{ rads/s}$$

$$T_{Do} = 110.806Nm$$

$$\begin{aligned}
T_{Din} &= \frac{110.806}{0.94 \times 4.07} \\
T_{Din} &= \frac{110.806}{3.826} \\
T_{Din} &= 28.962Nm
\end{aligned}$$

$$T_{GBo} = 29.919Nm$$

$$T_{GBin} = \frac{29.919}{1.47 \times 0.95}$$

$$T_{GBin} = \frac{29.919}{1.397}$$

$$T_{GBin} = 21.424Nm$$

$$Acc_{Eng} = (1.47 \times 0.95) \times (3.94 \times 0.94) \times 0$$

$$Acc_{Eng} = 0 \text{ rads/s}^2$$

$$T_{Ls} = 0 \times 0.15$$

$$T_{Ls} = 0Nm$$

$$T_{Engo} = 21.424Nm$$

$$RPM_{Eng} = 414.459 \times 1.47 \times 3.94$$

$$RPM_{Eng} = 2400.504 \text{ revs/min}$$

$$T_{InEng} = 0 + 21.424$$

$$T_{InEng} = 21.424Nm$$

$$BMEP = \left(\frac{21.424 \times 4 \times \pi}{1984 \times 10^{-6}} \right) \times 10^{-5}$$

$$BMEP = \left(\frac{269.222}{0.001984} \right) \times 10^{-5}$$

$$BMEP = 135696.534 \times 10^{-5}$$

$$BMEP = 1.357Bar$$

$$P_w = \frac{2 \times \pi \times 21.424 \times 2400.504}{60000}$$

$$P_w = \frac{323134.153}{60000}$$

$$P_w = 5.386kW$$

$$T_{GBo} = 28.962Nm$$

$$T_{GBin} = \frac{28.962}{1.357 \times 0.95}$$

$$T_{GBin} = \frac{28.962}{1.289}$$

$$T_{GBin} = 22.466Nm$$

$$Acc_{Eng} = (1.357 \times 0.95) \times (4.07 \times 0.94) \times 0$$

$$Acc_{Eng} = 0 \text{ rads/s}^2$$

$$T_{Ls} = 0 \times 0.15$$

$$T_{Ls} = 0Nm$$

$$T_{Engo} = 22.466Nm$$

$$RPM_{Eng} = 414.459 \times 1.357 \times 4.07$$

$$RPM_{Eng} = 2289.092 \text{ revs/min}$$

$$T_{InEng} = 0 + 22.466$$

$$T_{InEng} = 22.466Nm$$

$$BMEP = \left(\frac{22.466 \times 4 \times \pi}{999 \times 10^{-6}} \right) \times 10^{-5}$$

$$BMEP = \left(\frac{282.316}{0.000999} \right) \times 10^{-5}$$

$$BMEP = 282607.158 \times 10^{-5}$$

$$BMEP = 2.826Bar$$

$$P_w = \frac{2 \times \pi \times 22.466 \times 2289.092}{60000}$$

$$P_w = \frac{323123.743}{60000}$$

$$P_w = 5.385kW$$

10.2.3.1 Fuel

VW Passat

As can be seen above in the VW column, the RPM for the engine is 2400.504 revs/min and BMEP is 1.357 Bar. To calculate the fuel flow, the equations where the BMEP is under 10 Bar from Figure 16-7 & Figure 16-8 would be needed.

2000RPM

$$\begin{aligned} y = & 0.00000244874641586179x^6 - 0.00011252899746239100x^5 \\ & + 0.00171776699176007000x^4 - 0.01083310972899200000x^3 \\ & + 0.02909374022056000000x^2 + 0.15231141430740500000x \\ & + 0.36926522987425900000 \end{aligned}$$

2500RPM

$$\begin{aligned} y = & 0.00000418558761428045x^6 - 0.00014551851771216200x^5 \\ & + 0.00181558864755971000x^4 - 0.00939127169749554000x^3 \\ & + 0.01888814410449410000x^2 + 0.21610010699441800000x \\ & + 0.47088640936658500000 \end{aligned}$$

The values of 'x' are the BMEP, in this case it is 1.357 Bar.

2000RPM

$$\begin{aligned} y = & 0.00000244874641586179(1.357)^6 - 0.00011252899746239100(1.357)^5 \\ & + 0.00171776699176007000(1.357)^4 - 0.01083310972899200000(1.357)^3 \\ & + 0.02909374022056000000(1.357)^2 + 0.15231141430740500000(1.357) \\ & + 0.36926522987425900000 \end{aligned}$$

$$\begin{aligned} y = & 0.00001529 - 0.000517801 + 0.005824835 - 0.02707027609 + 0.05357463884 \\ & + 0.2066865892 + 0.369265229874259 \end{aligned}$$

$$y = 0.6077785747 \text{ g/s}$$

2500RPM

$$\begin{aligned} y = & 0.00000418558761428045(1.357)^6 - 0.00014551851771216200(1.357)^5 \\ & + 0.00181558864755971000(1.357)^4 - 0.00939127169749554000(1.357)^3 \\ & + 0.01888814410449410000(1.357)^2 + 0.21610010699441800000(1.357) \\ & + 0.47088640936658500000 \end{aligned}$$

$$\begin{aligned} y = & 0.000026135 - 0.000669603 + 0.00615642 - 0.02346734447 + 0.03478155407 \\ & + 0.2932478452 + 0.470886409366585 \end{aligned}$$

$$y = 0.7809614162 \text{ g/s}$$

These values were then used to interpolate to find the fuel flow at 2400.504 revs/min.

$$y = \left(y_1 + \left(\left(\frac{(x - x_1)}{(x_2 - x_1)} \right) \times (y_2 - y_1) \right) \right)$$

y=	Fuel flow at 2400.504 RPM	
y ₁ =	Fuel flow at 2000 RPM (g/s)	0.6077785747
y ₂ =	Fuel flow at 2500 RPM (g/s)	0.7809614162
x=	Calculated RPM	2400.504
x ₁ =	Lower Bound RPM	2000
x ₂ =	Upper Bound RPM	2500

$$y = \left(0.6077785747 + \left(\left(\frac{(2400.504 - 2000)}{(2500 - 2000)} \right) \times (0.7809614162 - 0.6077785747) \right) \right)$$

$$y = \left(0.6077785747 + \left(\left(\frac{(400.504)}{(500)} \right) \times (0.1731828415) \right) \right)$$

$$y = (0.6077785747 + ((0.801008) \times (0.1731828415)))$$

$$y = (0.6077785747 + 0.1387208415)$$

$$y = 0.7464994162 \text{ g/s}$$

Ford Fox

As can be seen in the Ford Fox column above the RPM was calculated to 2289.092 revs/min with a corresponding BMEP of 2.826 Bar. For this example, the equations from Figure 16-21 & Figure 16-22 are needed.

2250RPM

$$y = 0.000000032723033265708x^6 - 0.000001951587810233310x^5 \\ + 0.000036195233753790400x^4 - 0.000141419529919228000x^3 \\ - 0.000013340055078974100x^2 + 0.099396261104508100000x \\ + 0.214614842611049000000$$

2500RPM

$$y = 0.0000000462972531457828x^6 - 0.0000033222107073738500x^5 \\ + 0.0000854417353416137000x^4 - 0.0009234577691681420000x^3 \\ + 0.0054287708165856100000x^2 + 0.0994349967543215000000x \\ + 0.2383937941422870000000$$

The values of 'x' are the BMEP, in this case it is 2.826 Bar.

2250 RPM

$$y = 0.000000032723033265708(2.826)^6 - 0.000001951587810233310(2.826)^5 \\ + 0.000036195233753790400(2.826)^4 \\ - 0.000141419529919228000(2.826)^3 \\ - 0.000013340055078974100(2.826)^2 \\ + 0.099396261104508100000(2.826) + 0.214614842611049000000$$

$$y = 0.00001666811541 - 0.0003517619696 + 0.002308553883 - 0.003191727914 \\ - 0.00001065373617 + 0.2808938339 + 0.214614842611049$$

$$y = 0.4942797549 \text{ gms/sec}$$

2500RPM

$$y = 0.0000000462972531457828(2.826)^6 - 0.0000033222107073738500(2.826)^5 \\ + 0.0000854417353416137000(2.826)^4 \\ - 0.0009234577691681420000(2.826)^3 \\ + 0.0054287708165856100000(2.826)^2 \\ + 0.0994349967543215000000(2.826) + 0.2383937941422870000000$$

$$y = 0.00002358240914 - 0.0005988085065 + 0.005449525517 - 0.02084171784 \\ + 0.04335566208 + 0.2810033008 + 0.238393794142287$$

$$y = 0.5467853386 \text{ gms/sec}$$

These values were then used to interpolate to find the fuel flow at 2289.092 revs/min.

$$y = \left(y_1 + \left(\left(\frac{(x - x_1)}{(x_2 - x_1)} \right) \times (y_2 - y_1) \right) \right)$$

y=	Fuel flow at 2289.092 RPM	
y ₁ =	Fuel flow at 2250 RPM (g/s)	0.4942797549
y ₂ =	Fuel flow at 2500 RPM (g/s)	0.5467853386
x=	Calculated RPM	2289.092
x ₁ =	Lower Bound RPM	2250
x ₂ =	Upper Bound RPM	2500

$$y = \left(0.4942797549 + \left(\left(\frac{(2289.092 - 2250)}{(2500 - 2250)} \right) \times (0.5467853386 - 0.4942797549) \right) \right)$$

$$y = \left(0.4942797549 + \left(\left(\frac{(39.092)}{(250)} \right) \times (0.0525055837) \right) \right)$$

$$y = (0.4942797549 + ((0.156368) \times (0.0525055837)))$$

$$y = (0.4942797549 + 0.008210193112)$$

$$y = 0.502489948 \text{ g/s}$$

VW Passat/ Ford Fox Comparison

The fuel flow per second at 150 seconds in the NEDC test were therefore found to be 0.7465g/s for the VW Passat and 0.5025g/s for the Ford Fox

10.3 WLTP

The concept for the WLTP was introduced in 2007; the approach to the design was to collect data on driving behaviours in the EU, India, Japan, Korea and the USA. The data collected was initially filtered into; urban, rural and motorway, but due to differing definitions for these road types for each country, they were redefined by speed classes of: low, medium, high and extra high.

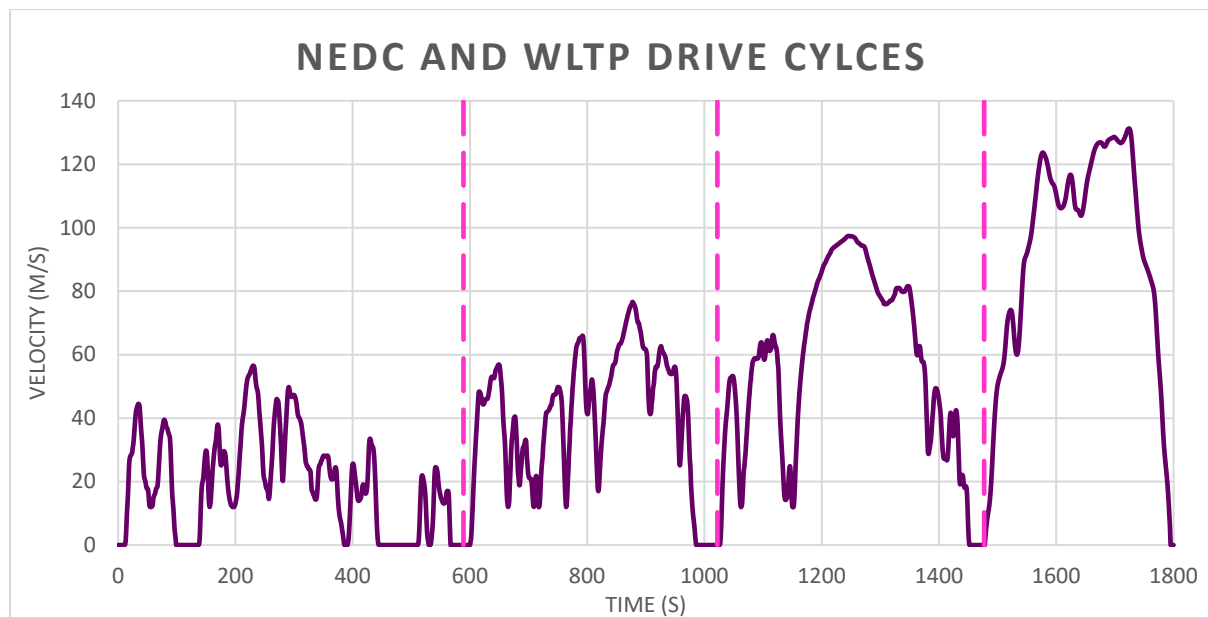


Figure 10-5: WLTP Showing the Low, Medium, High and Extra High-Speed Sections

After testing and retesting procedure designs the final design is; 23km long with a maximum speed of 130km/hr, the whole test lasts 1800 seconds, only 158 of which are spent stationary at idle. The test is designed to make the figures more representative of real world driving, involving shorter stopping periods, more rapid acceleration and deceleration with a greater spread of engine load and gears used (SNJ , 2017).

For the WLTP, the gear shift points are not predefined, this should allow for closer figures produced for fuel consumption and emissions production. As the gear shift points are open, the profile can be chosen to suit the vehicle, which should produce figures closer to those found in everyday driving. With the gear shift points open, manufacturers will be able to find optimal shifting profiles, with the aim of minimising fuel consumption and emission levels.

To accompany the WLTP there is a RDE test, which uses a mobile exhaust monitoring system, that measures the NO_x and other particulate emissions. The RDE incorporates tests to

simulate life driving such as; freezing temperatures, uphill driving, continuous high speed, heavy loading of the vehicle or combinations of. The RDE doesn't have a set time and velocity profile, though there is legislation, to characterize the trips and allow accurate comparisons between vehicles to be made.

Table 10-10: Distance and Speed Specifications for each Urban, Rural and Motorway Part of the RDE Test (ICCT, 2017)

<i>Trip Specifics</i>		<i>Provisions set in the Legal Text</i>
Total Trip Duration		Between 90 and 120 minutes
Distance	Urban	>16km
	Rural	>16km
	Motorway	>16km
Trip Composition	Urban	29% to 44% of distance
	Rural	23% to 43% of distance
	Motorway	23% to 43% of distance
Average Speeds	Urban	15 to 40km/hr
	Rural	Between 60km/hr and 90km/hr
	Motorway	>90km/hr(>100km/hr for at least 5 minutes)

As the test is conducted on public roads, it is intended to be the most 'real' vehicle test, with the most accurate fuel and emissions data recorded. However, to ensure fair testing, boundary conditions have been set.

Table 10-11: Boundary Conditions for RDE tests (ICCT, 2017)

<i>Parameter</i>		<i>Provisions set in the legal text</i>
Payload		<=90% of maximum vehicle weight
Altitude	Moderate	0 to 700 m
	Extended	Between 700 and 1300m
Altitude difference		No more than a 100m altitude difference between start and finish
Cumulative Altitude gain		1200m/100km
Ambient Temperature	Moderate	0°C to 30°C
	Extended	From - 7°C to 0°C and 30°C to 35°C
Stop Percentage		Between 6% and 30% of urban time
Maximum Speed		145km/h (160km/h for 3% of motorway driving time)
Use of Auxiliary systems		Free to use as in real life (operation not recorded.)

Full implementation of the WLTP is currently underway with the WLTP fully replacing the NEDC (apart from for vehicles registered before 31st August 2017) from 2021

Table 10-12: Process of Implementing the WLTP (Lloyd, 2018)

<i>Implementation Date</i>	<i>Effects</i>
1 st September 2017	All newly-certified cars emissions tested under the new WLTP system. However, at the time of writing, the NEDC figure is still being calculated from the WLTP test figure and continues to be shown as the legally required value
September 2018	All models to be certified under the WLTP rules with the RDE test in force. NEDC figures, backwards calculated from the WLTP numbers, remain the ones included in sales documents and other publications until April 2020 in the UK
April 2020	Figures used for tax purposes and in adverts switch from NEDC to WLTP
End of 2020	NEDC economy and CO2 figures continue to be calculated for all vehicles
From 2021	WLTP values fully replace NEDC figures, bar those applying to vehicles registered before 31 August 2017

10.3.1 Gear Shift Profile Calculator

10.3.1.1 Legislation-193

For the gear shift selection to be legal in the WLTP, there is strict legislation that the selection must adhere to. The engine speed must be between a minimum and maximum for the gear to be selected. The maximum is calculated using the engine idle speed and the engine speed at which the maximum power is produced.

$$RPM_{Max} = 0.9(s - RPM_{idle}) + RPM_{idle}$$

Equation 10-25: Maximum Allowed Engine Speed

The minimum RPM differs for the gear that is selected, for first gear:

$$RPM_{Min} = RPM_{idle}$$

Equation 10-26: Minimum Engine Speed for First Gear

For second gear:

$$RPM_{Min} = 1.25 \times RPM_{idle}$$

Equation 10-27: Minimum Engine Speed for Second Gear

For third gear and above:

$$RPM_{Min} = RPM_{idle} + 0.125 \times (s - RPM_{idle})$$

Equation 10-28: Minimum Engine Speed for Third Gear and Above

With the initial gear selection being defined from the engine speed further legislation was agreed upon to further replicate real world driving conditions. Regarding skipping gears, it is permitted during deceleration, but not during acceleration and any gear selected during acceleration must be in use for a minimum of 3 seconds. If the gear has been shifted up and then back down for a period of 5 seconds or less, then this is corrected to the original gear, before the shift up. During transition between acceleration and deceleration the gears must stay the same, and first gear must be selected 1 second before acceleration from the vehicle being stationary. (Tutuianu, et al., 2013, pp. 54-60)

10.3.1.2 Modelling -334

The author's model for a gear calculator was designed to try to optimise fuel usage, while still conforming to the legislation. The final iteration of code can be seen in 16.2.3 Gear Calculator

Iteration one: the construct of the basic model using the variables 8 Constants and 10.1 Constants – 0 and the velocity and time profile from (Tutuianu, et al., 2013, pp. 68-79) following the same calculation process as seen in 10.2.1 The Model - 288 up until the gearbox output torque Equation 10-12. As the gears were 'unknown' values the code was changed to a hypothetical, e.g. if this was the gear selected then the relevant ratios would be used to calculate: gearbox input torque, engine angular acceleration, torque lost, engine output torque and RPM. The RPM value was then compared to the maximum and the minimum allowed RPM for the gear above the selected gear, if the RPM was between the allowed limits, the process would be done again as if the gear above had been selected. The modelling process

was then continued the same way as the earlier NEDC model; finding the: internal engine torque, BMEP, power, and fuel flow (done using the polynomials).

Iteration two: followed on from the previous iteration, with the inclusion of the piece of gear shifting legislation, that states that the vehicle cannot shift up a gear for five seconds or less and then back down to the original gear. This was done, by implementing the legislation second by second, to ensure that the coding process was running correctly.

Iteration three: was a continuation from iteration two, with the implementation of the legislation stating that the gears cannot change during the transition between acceleration and deceleration.

Iteration four: includes the legislation for selecting first gear one second before the acceleration phase begins from a stationary position.

With each new iteration, potentially creating new gear sequences, which may conflict with the legislation, the gears were checked twice in the code, and the process after selection was recalculated.

10.3.2 Results

The results show the gear shift profile, for each iteration. This can be used to compare the effect each section of legislation had on the profile, with no optimisation made.

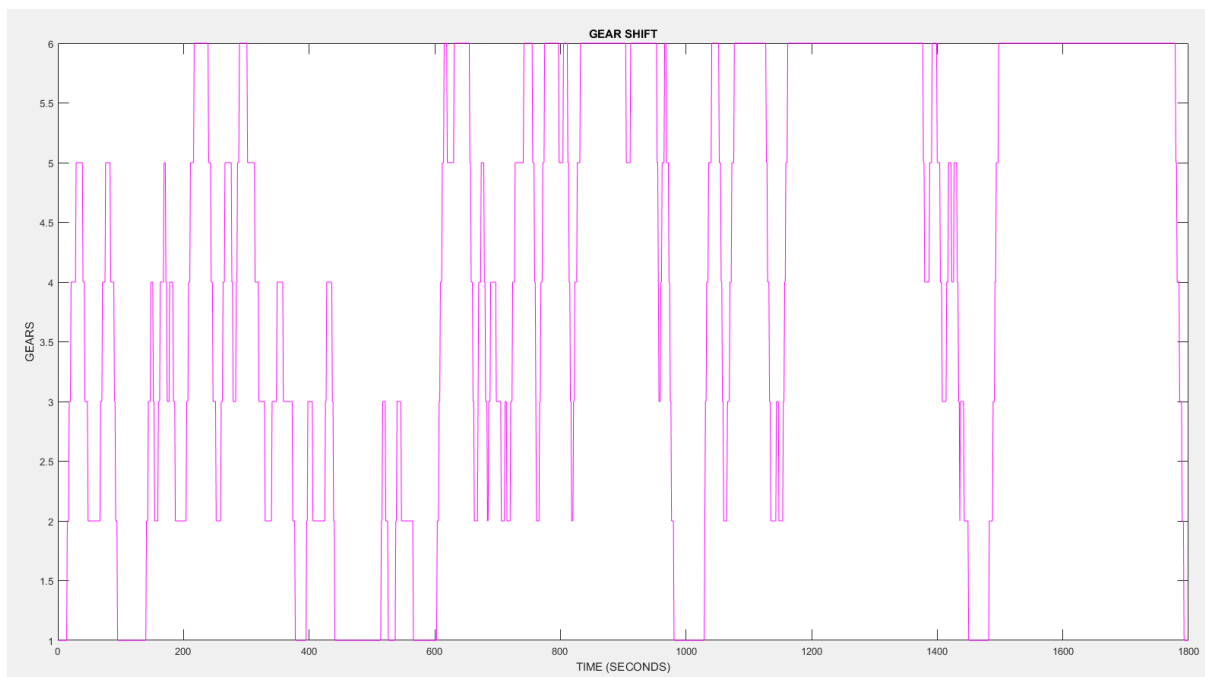


Figure 10-6: Iteration 1; Gear Selection

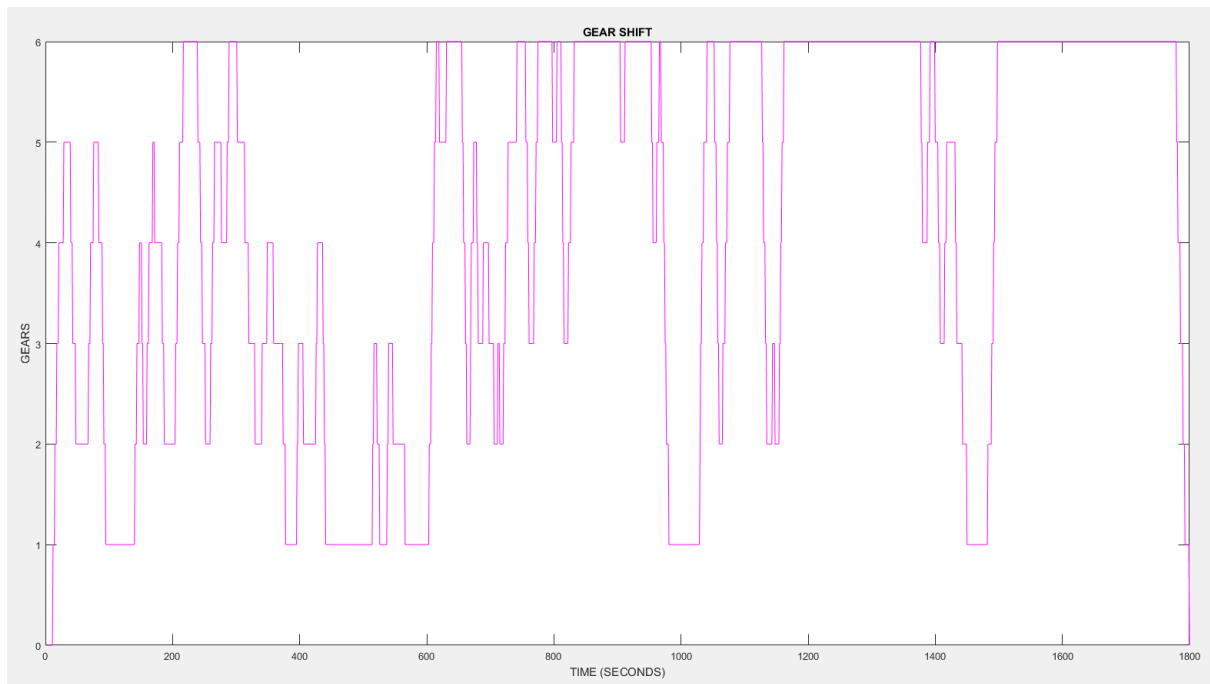


Figure 10-7: Iteration 2; Gear Selection

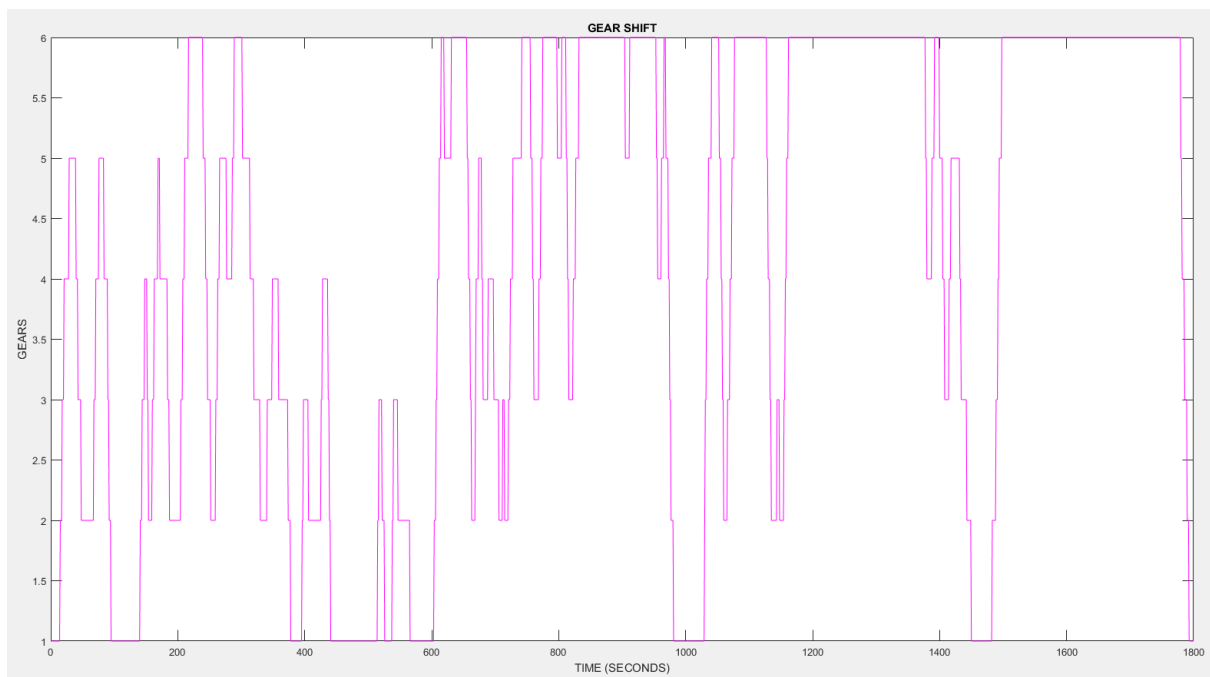


Figure 10-8: Iteration 3; Gear Selection

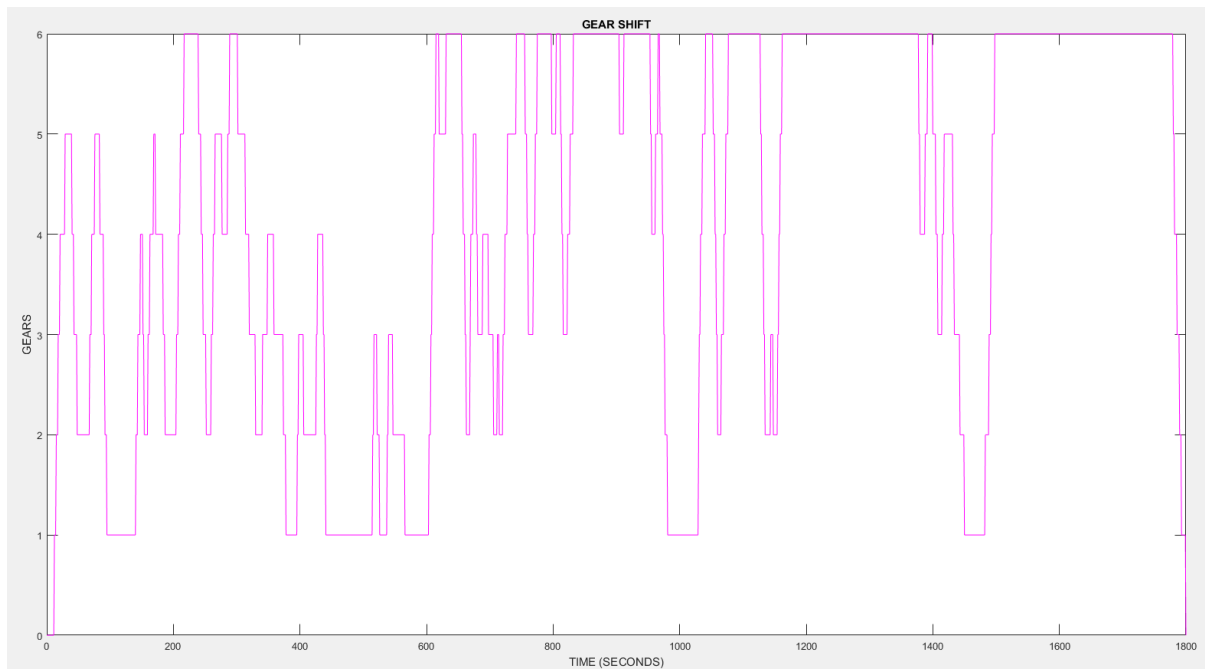


Figure 10-9: Iteration 4; Gear Selection

The implication between iteration one and two, can be seen clearly at around 1400s where the profile has changed with the addition of the legislation prohibiting, changing up to a gear for 5 seconds or less and then back to the original gear. Iteration 4 changes the gearshift profile at the start and end when the vehicle is stationary, keeping the vehicle in neutral, in this case gear zero.

10.4 Analysis

10.4.1 NEDC

The engines were being compared with engine size and gear ratios, to see if the NEDC favoured smaller engines. The differences in parameters used can be seen in 10.1 Vehicle Constants – 0. These parameters in the model do not come into effect until the differential torque input is being calculated (Equation 10-11). The smaller final drive ratio in the VW Passat means that there is a higher torque value going in to the differential. This difference in torque for the differential, has an effect for further calculations. Looking at the effect the gear ratios (see Figure 16-32 & Figure 16-33) have on the calculated value for the gearbox torque input. The Ford Fox's first gear and final drive ratio is greater than that of the Passat, this larger gear ratio creates a lower input torque for the gearbox, which can be seen in the first 20 seconds of the UDC.

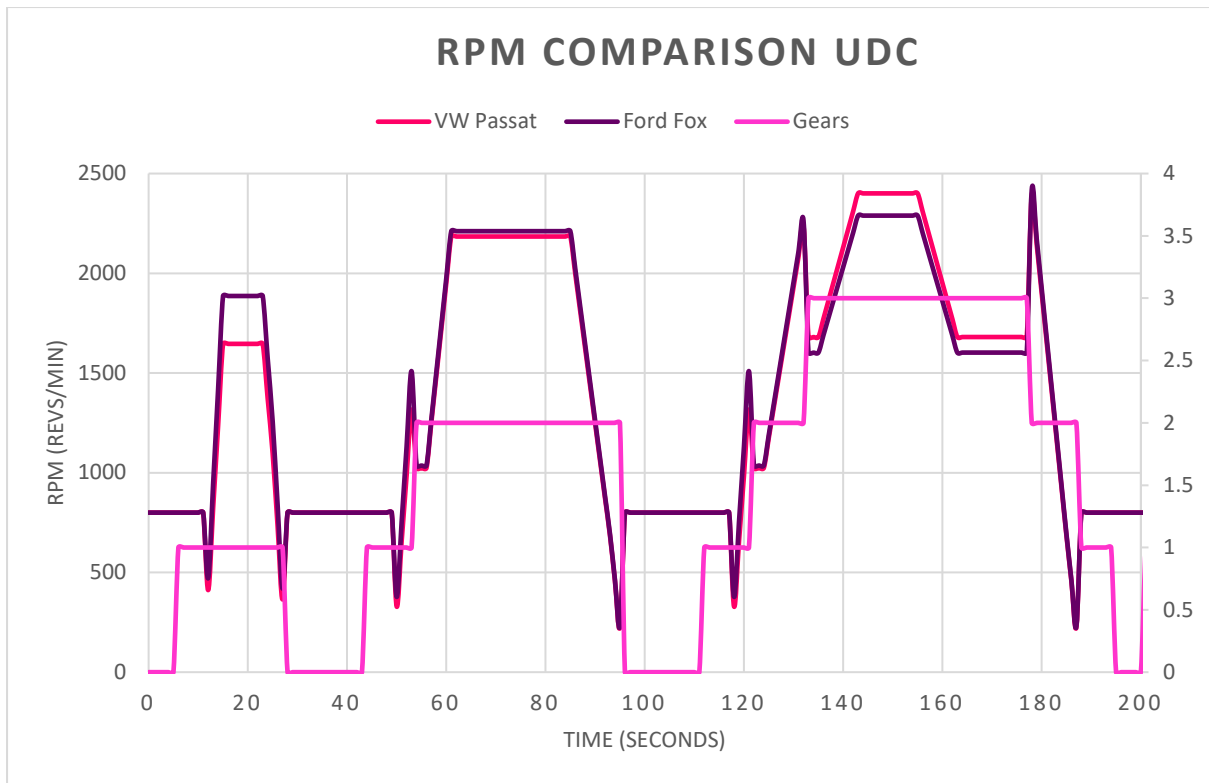


Figure 10-10: RPM Comparison during the UDC Section of the NEDC

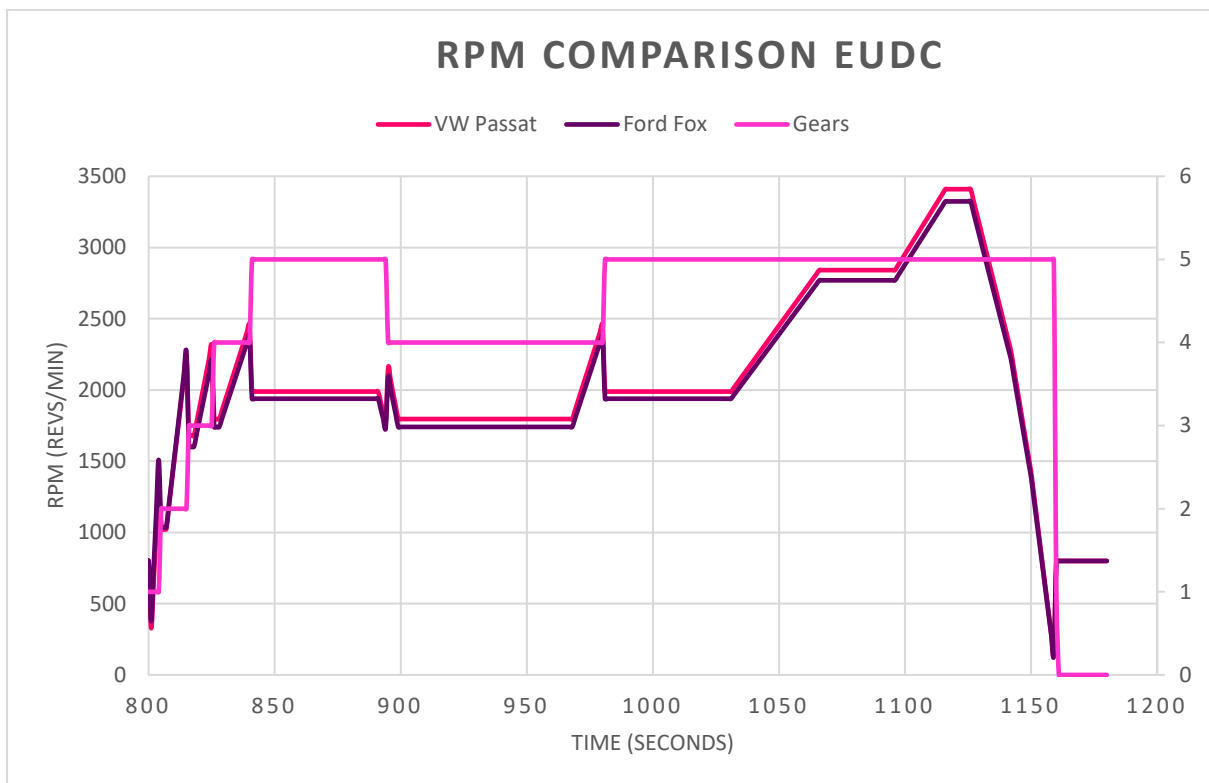


Figure 10-11: RPM Comparison during the EUDC Section of the NEDC

The gear ratios also affect the engine RPM as the engine RPM is based on the wheel RPM, gear ratios and final drive ratio. Due to the wheel RPM being constant with both engines being tested, when first gear is selected, the RPM for the Ford Fox is higher. At lower vehicle speeds and lower engine torque, the higher RPM enables the engine operating zone to be more ideal for stoichiometric combustion, reducing the emissions and need for over fuelling.

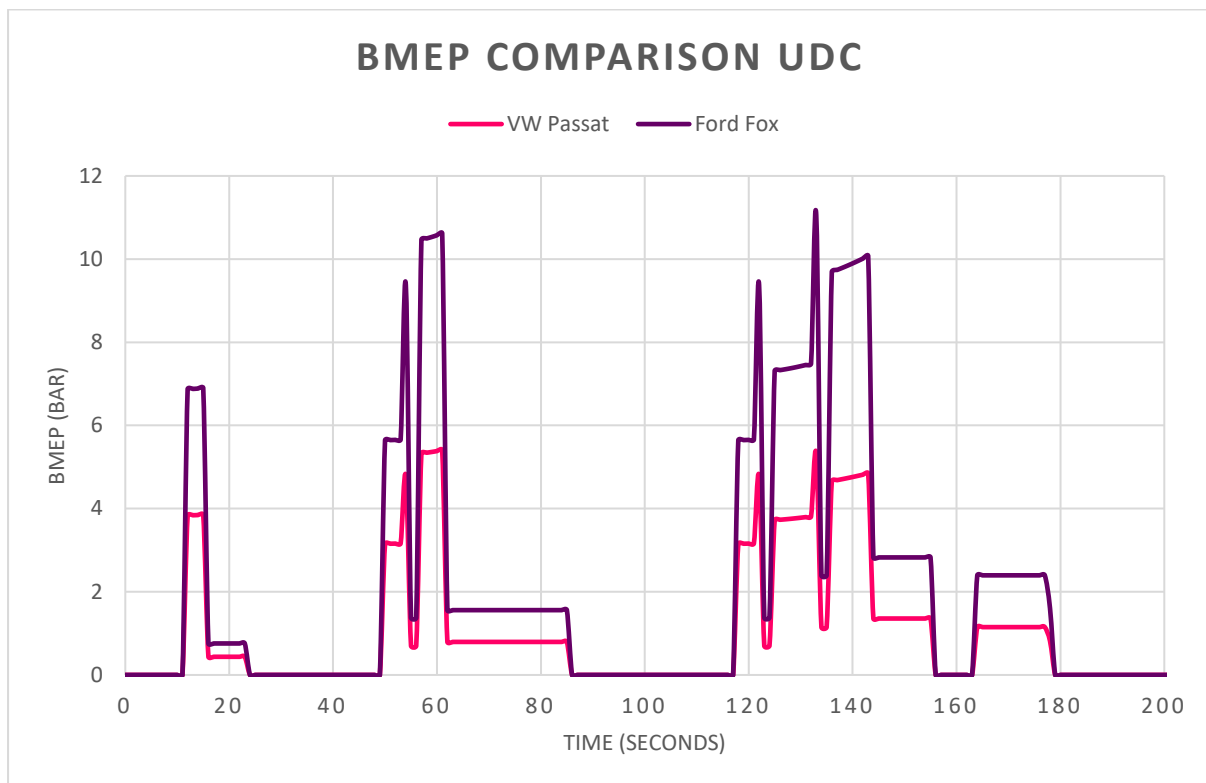


Figure 10-12: BMEP Comparison during the UDC Section of the NEDC

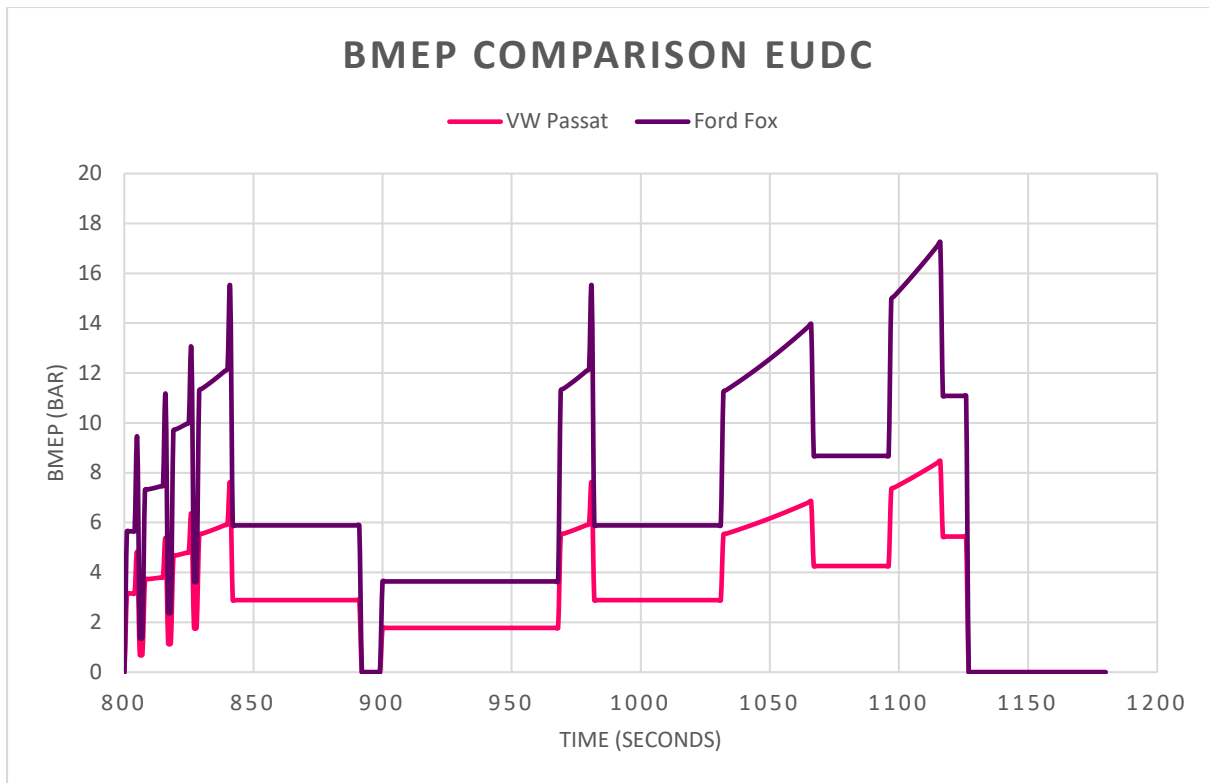


Figure 10-13: BMEP Comparison during the EUDC Section of the NEDC

As the work done by the engines (Figure 16-38 & Figure 16-39) is similar throughout the test, the engine size has a significant effect on the values for BMEP. With the BMEP being the work done per engine cycle divided by the engine displacement. The advantage is for the Ford Fox; with an engine displacement of just over half that of the VW Passat, this results in the calculated BMEP for the Ford Fox being double that of the Passat. This higher operating range for the BMEP enables the engine to work in a more ideal range for fuel efficiency.

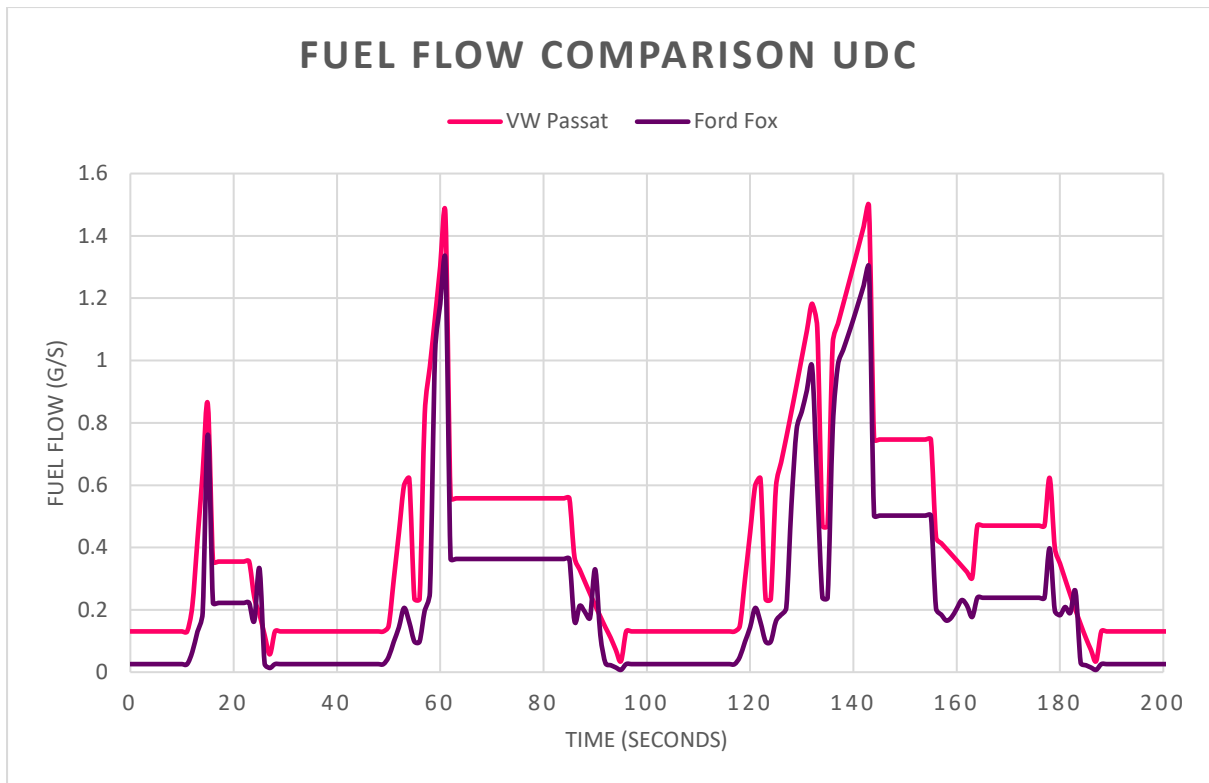


Figure 10-14: Fuel Flow Comparison during the UDC of the NEDC

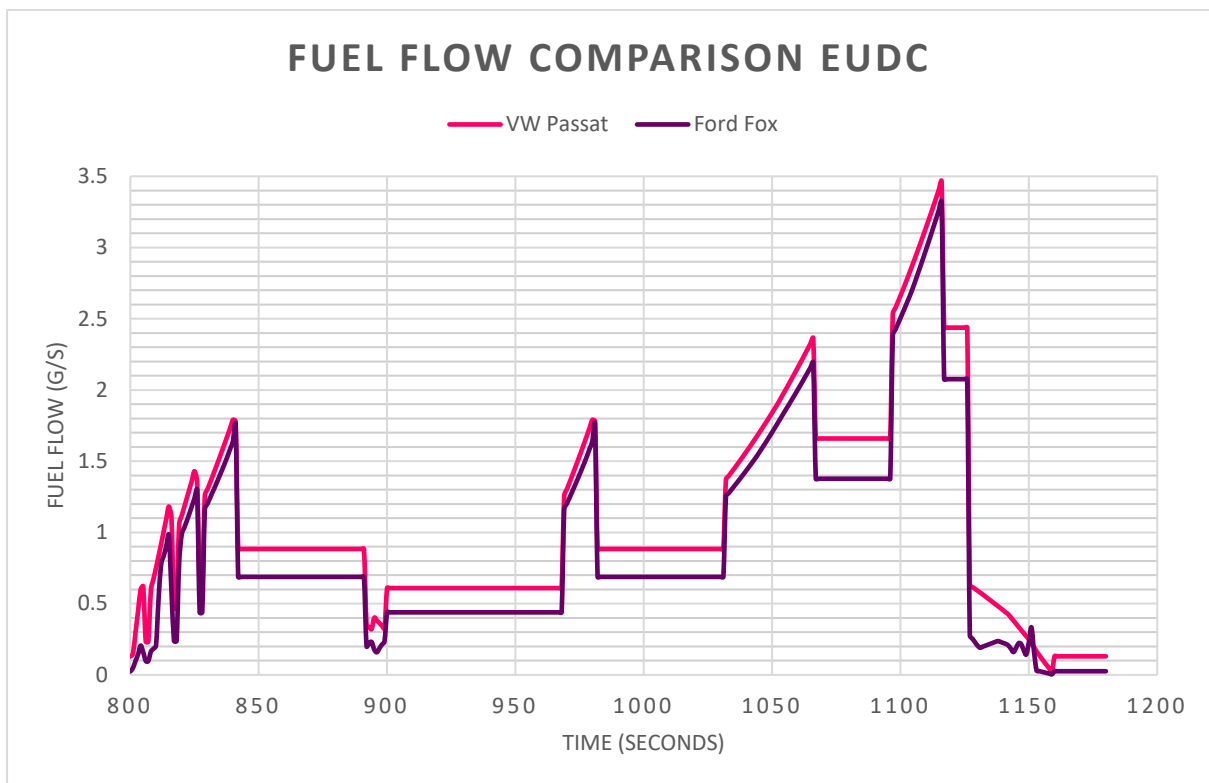


Figure 10-15: Fuel Flow Comparison during the EUDC of the NEDC

The fuel usage (16.2.1 Fuel) for the Ford Fox, is consistently lower than that of the VW Passat. This is due to the engine size of the Ford Fox being smaller and needing less fuel, for combustion to occur and power to be produced. This can be seen to have a positive knock on effect when the fuel consumption during the drive cycle is being calculated. The higher the operating BMEP range for the Ford Fox, the more positive the effect on the fuel consumption calculated in the drive cycle.

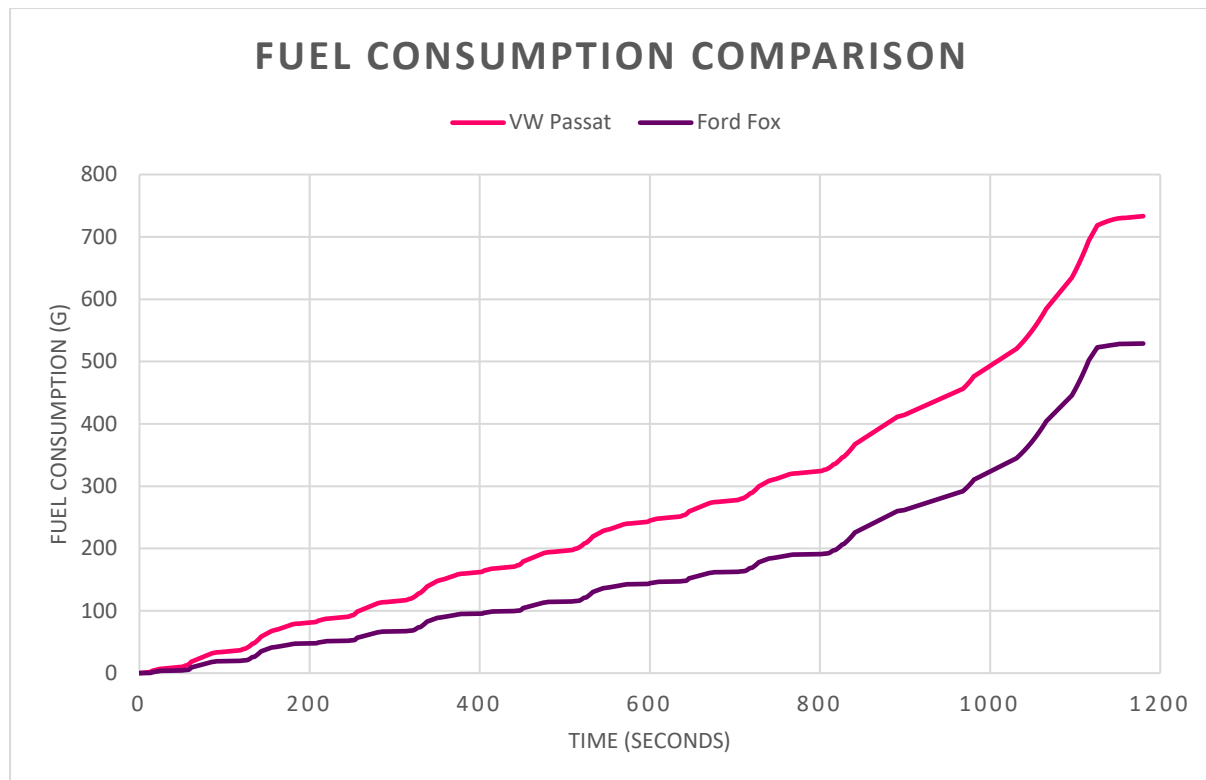


Figure 10-16: Cumulative Fuel Consumption Comparison

Table 10-13: Cumulative Fuel Consumption by the VW Passat in the NEDC

Fuel Usage VW		
Urban (0-800)	324.1768525	g
Extra Urban (801-1100)	409.0379617	g
Total Fuel Used	733.2148142	g
Fuel Per Km	66.6558922	g/km

Table 10-14: Cumulative Fuel Consumption by the Ford Fox in the NEDC

<i>Fuel Usage Fox</i>		
Urban (0-800)	191.0119951	g
Extra Urban (801-1100)	337.8385201	g
Total Fuel Used	528.8505152	g
Fuel Per km	48.07731957	g/km

For the cumulative fuel consumption figures, the smaller engine of the Ford Fox has a significant advantage over that of the VW Passat. In the urban section of the drive cycle, the Ford Fox consumes 41% less fuel. In the extra urban section of the drive cycle, the Ford Fox sees a fuel consumption decrease of 17%. This leads to a total fuel consumption decrease over the whole drive cycle by the Ford Fox of 27.9%.

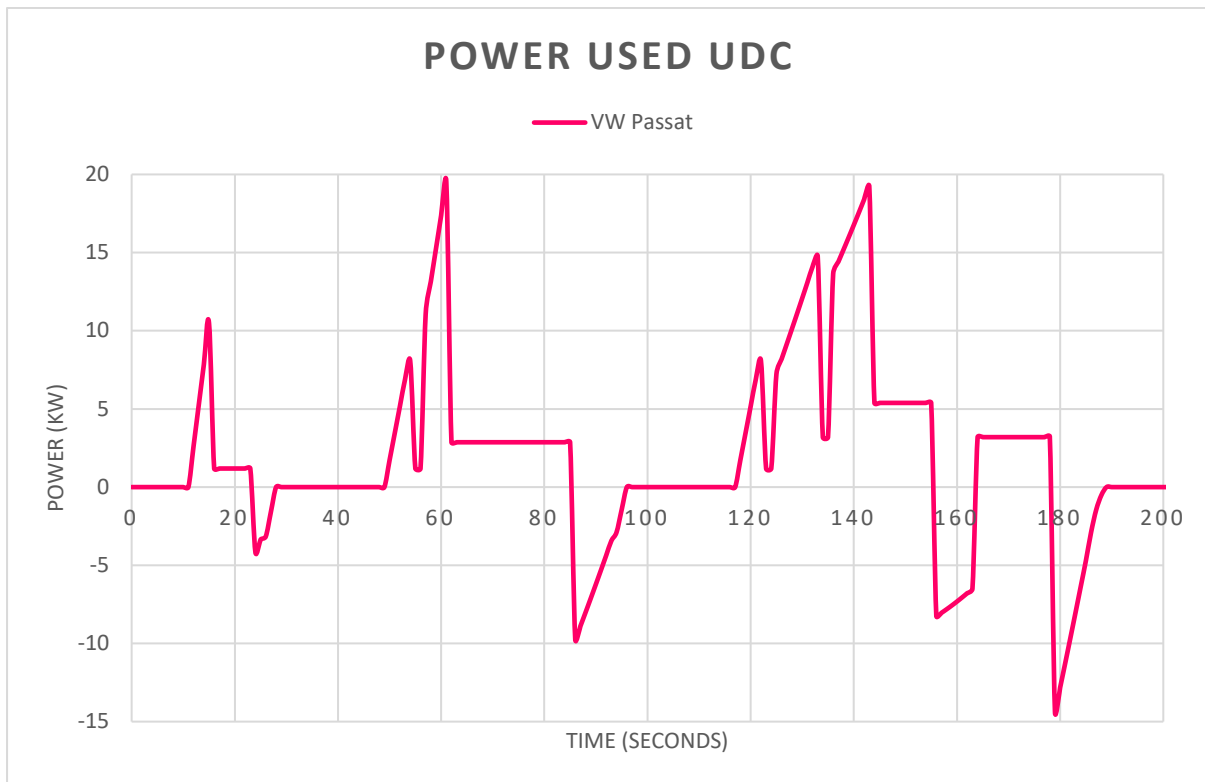


Figure 10-17: Power Used by the VW Passat, during the UDC Section of the NEDC

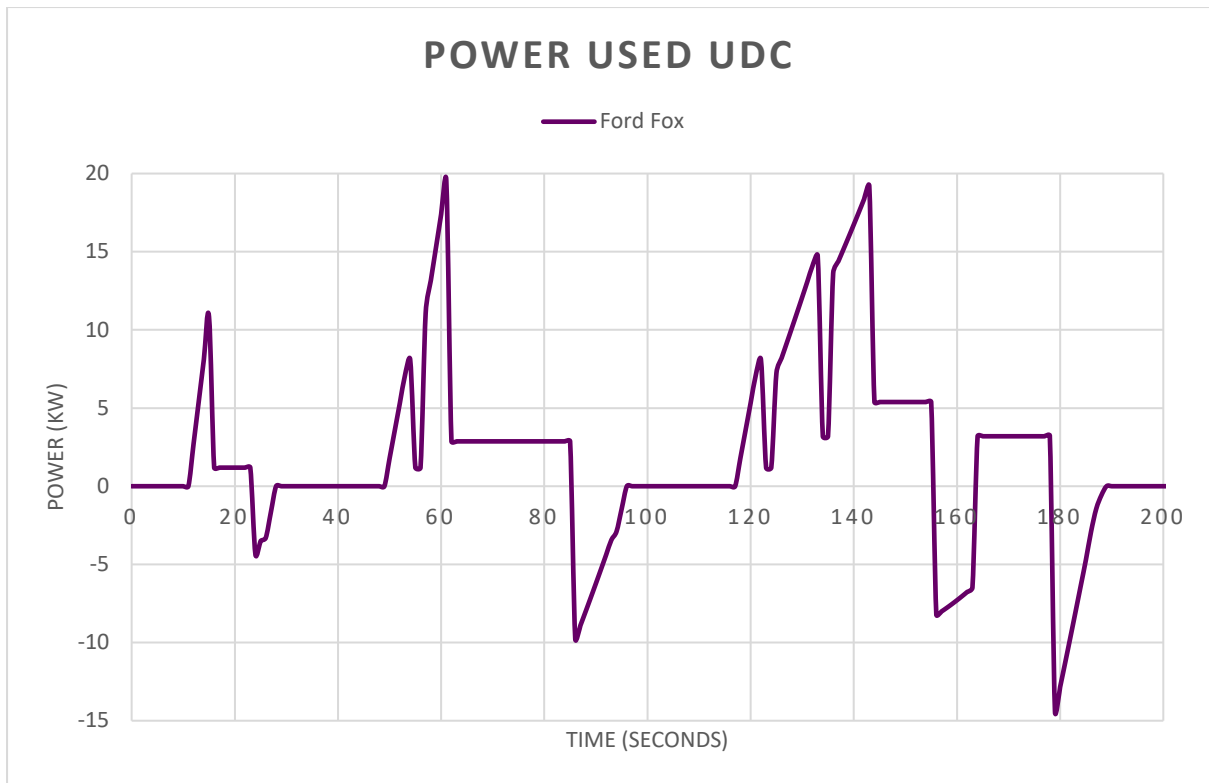


Figure 10-18: Power Used by the Ford Fox, during the UDC Section of the NEDC

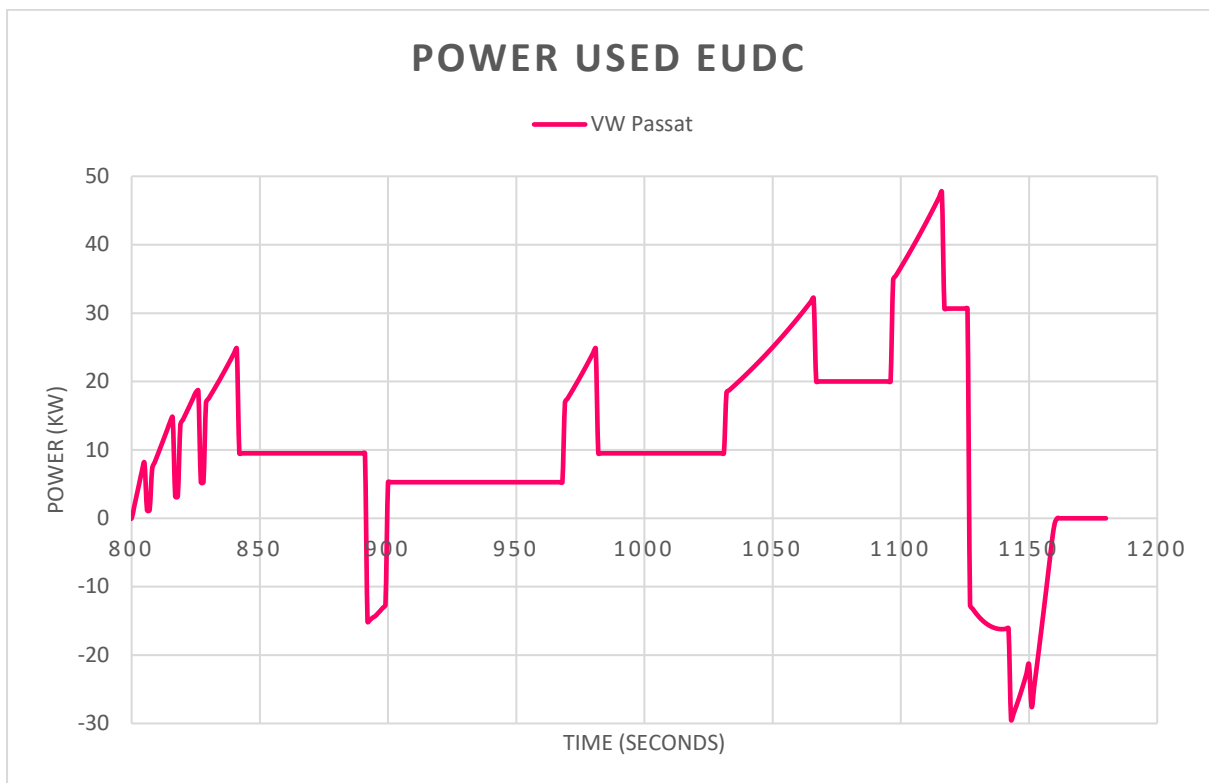


Figure 10-19: Power Used by the VW Passat, during the EUDC Section of the NEDC

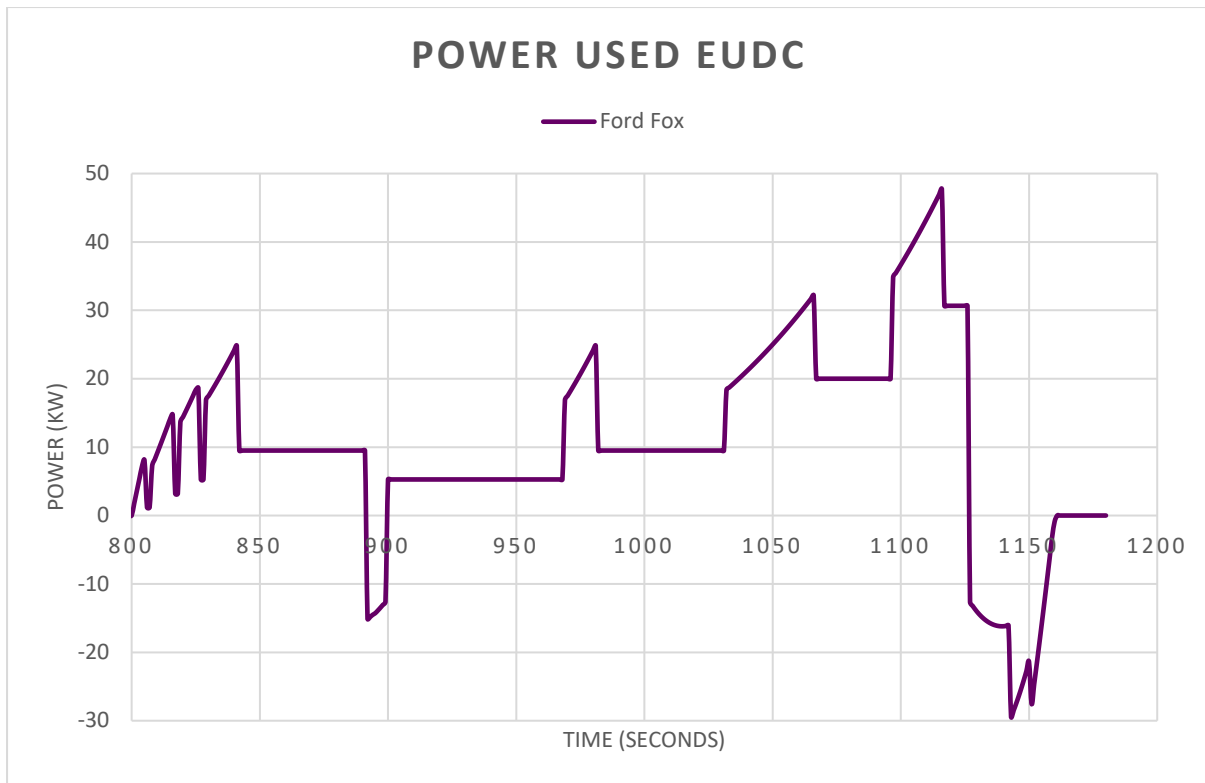


Figure 10-20: Power Used by the Ford Fox, during the EUDC Section of the NEDC

The values found for power were calculated from the engine RPM and internal torque, for this reason the power output of both engines is very similar. For the VW Passat a total of 7071.22kW of power was used to run the drive cycle. If the engine was hybridised and could recover power through braking, 1505.296kW of power could be recovered over the course of the drive cycle.

For the Ford Fox a total of 7078.696kW of power was used to run the drive cycle. If the engine was hybridised and could recover energy through braking, 1507.208kW of power could be recovered over the course of the drive cycle.

Table 10-15: Maximum; Power, Engine RPM and BMEP Produced by the VW Passat in the NEDC

VW		
Maximum Power	47.73818955	Kw
Max Engine RPM	3409.695712	RPM
Maximum BMEP	8.468176462	Bar

Table 10-16: Maximum; Power, Engine RPM and BMEP Produced by the Ford Fox in the NEDC

<i>FOX</i>		
<i>Maximum Power</i>	47.73026324	<i>Kw</i>
<i>Max Engine RPM</i>	3323.821673	<i>RPM</i>
<i>Maximum BMEP</i>	17.24931575	<i>Bar</i>

According to (Heywood, 1988), typical maximum BMEP values for a naturally aspirated spark-ignition engine (VW Passat) are in the range of 8.5 Bar to 10.5 Bar. For a turbo charged spark-ignition engine (Ford Fox) the maximum BMEP is between 12.5 Bar and 17 Bar. The marginal difference between the figures found, could be down to the driveline efficiencies being kept to 100%, which in the real world is not possible.

10.4.2 *Gear Calculator*

The author's model for the gear calculator works as a fundamental tool to predict the gear shift pattern during the running of the WLTP. During development additional improvements have been identified that could enhance its use.

One such improvement would be to incorporate the final piece of legislation, which states that there is to be no skipping of gears during the acceleration phases of the WLTP and any gear selected during these periods must be held for at least three seconds. However, the data does not appear to indicate that the gears are skipped during any of the acceleration phases, although on some occasions the selected gear did not stay engaged for a minimum of three seconds.

Another improvement would be to introduce the maximum RPM speed while selecting each gear. This was part of the initial code before a full set of results was gathered, however problems occurred in the model when trying to select fifth and sixth gear in the extra high-speed section of the WLTP. The calculated RPM exceeded that of the maximum, so the gear selection process stopped. Issues also occurred in (European Commission, 2017) model when the maximum RPM for all gears resulted in 'clipped' velocity profiles.

After the implementation of all the legislation, the main improvement to the model would be to optimise the fuel consumption, as was the initial premise for the model. This was unable to be achieved due to the author's lack of understanding of the software capabilities.

A final improvement that could be made would be to automate the calculations based on user inputted variables. This would make the model applicable to any type of vehicle; however, issues would occur with the current model when calculating the fuel flow, as the current code requires the fuel equations, for a variety of RPM ranges, to allow for interpolation to occur.

10.4.3 Comparison

The WLTP test is 30 minutes long, one and a half times the length of the NEDC, and covers 23 km, over twice the distance of the NEDC. In comparison with the NEDC, the WLTP more accurately reflects real driving conditions. The variety of hard and soft acceleration and deceleration allows for the full range of the vehicle to be tested. The continuous change in velocity in the WLTP is much closer to real driving conditions and should give more accurate emissions and fuel consumption results compared to those which would be seen on the public road.

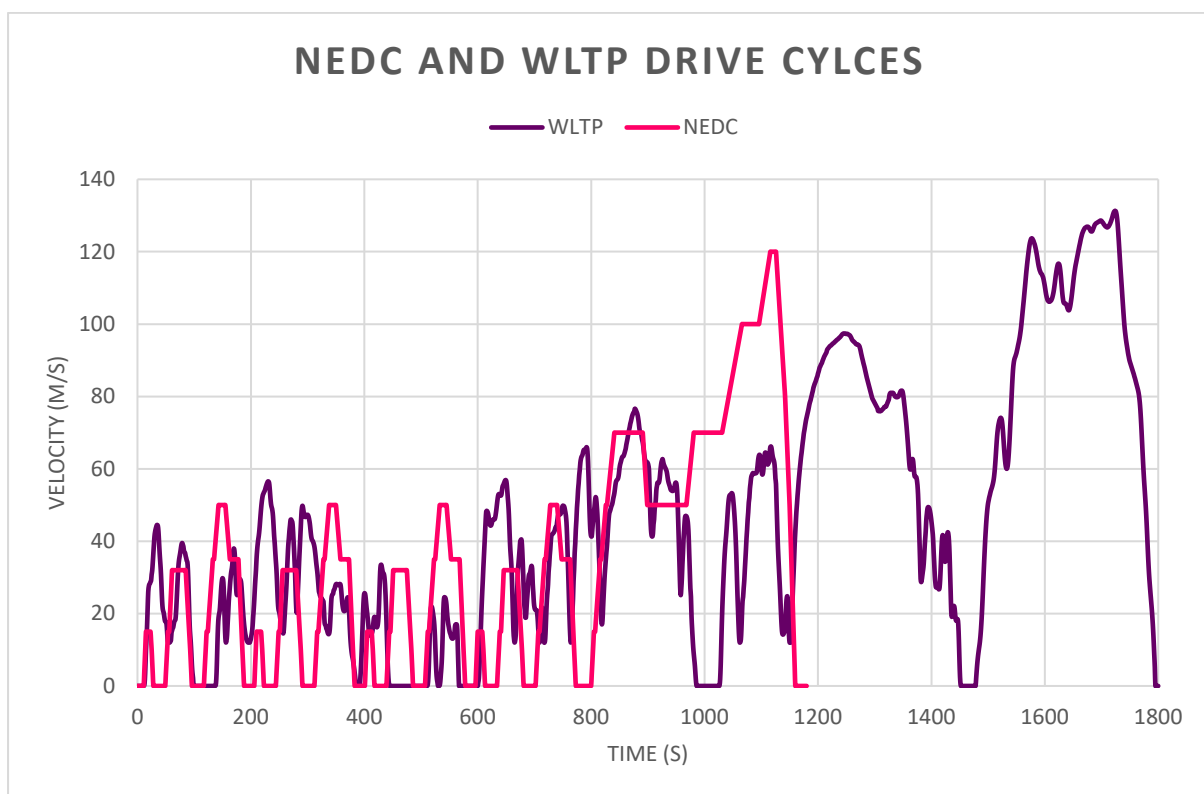


Figure 10-21: Comparison of the NEDC and Drive Cycle Velocities during Testing

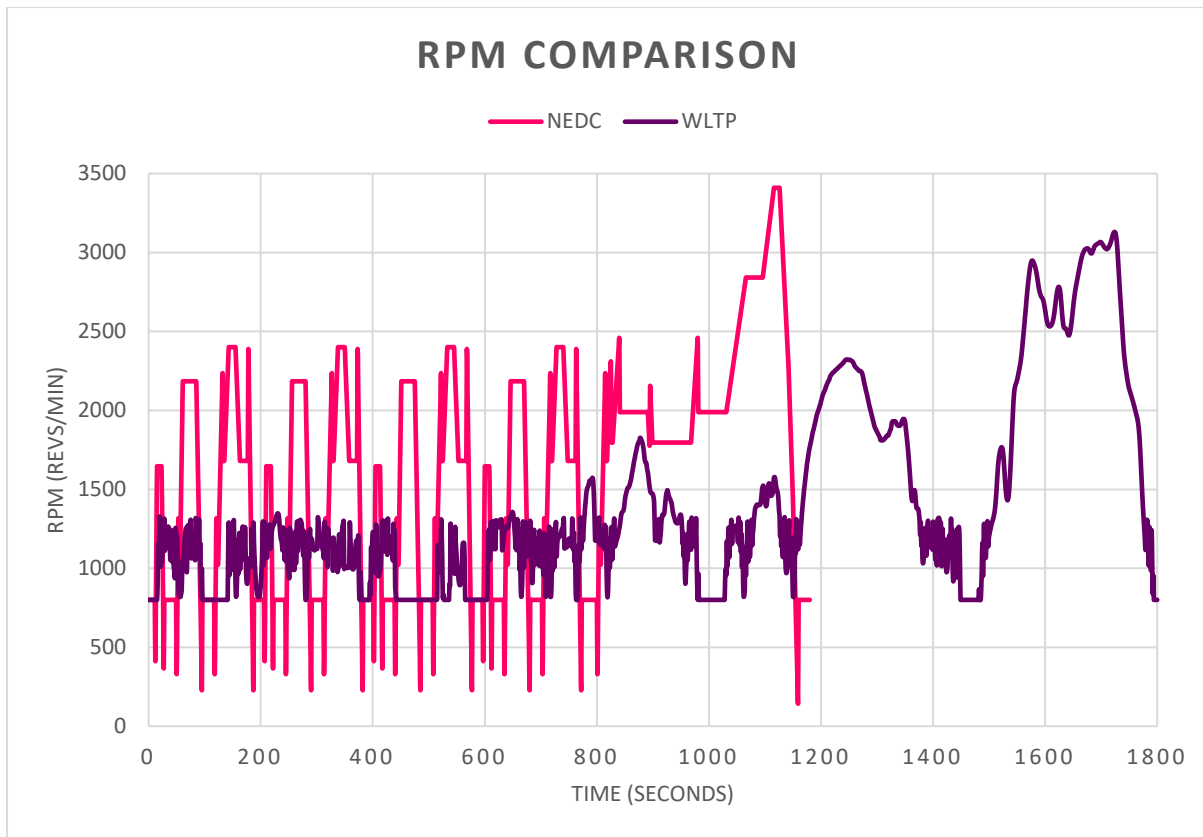


Figure 10-22: Comparison of RPM Between the NEDC and WLTP.

The engine RPM is dependent on the wheel RPM, gear ratio and final drive ratio. The openness of the gear shift positions means that the engine RPM can be kept at a more consistent and lower range than in the NEDC. This is beneficial to larger engines, unlike in the NEDC where the higher RPM from the prescribed gear shifting position would produce more emissions and use more fuel for the same velocities. The lower the RPM, the fewer combustion cycles occur, producing lower emissions and requiring less fuel.

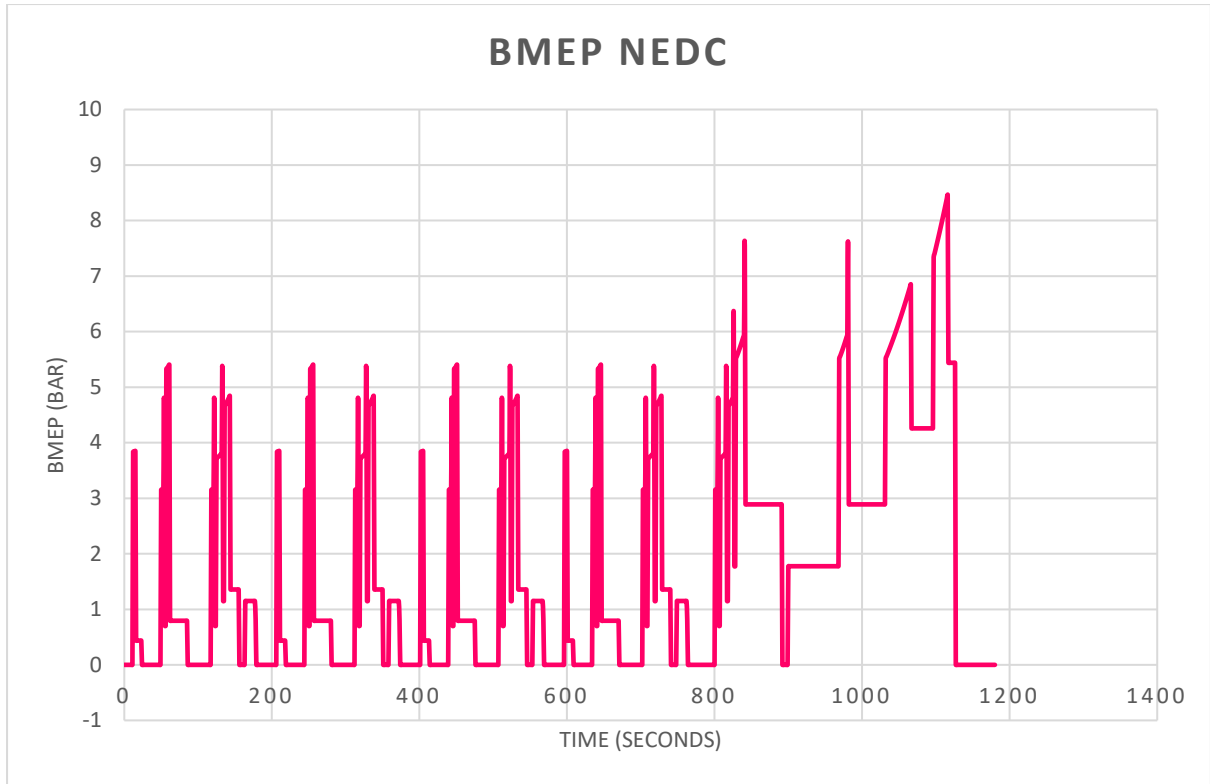


Figure 10-23 BMEP for the VW Passat in the NEDC

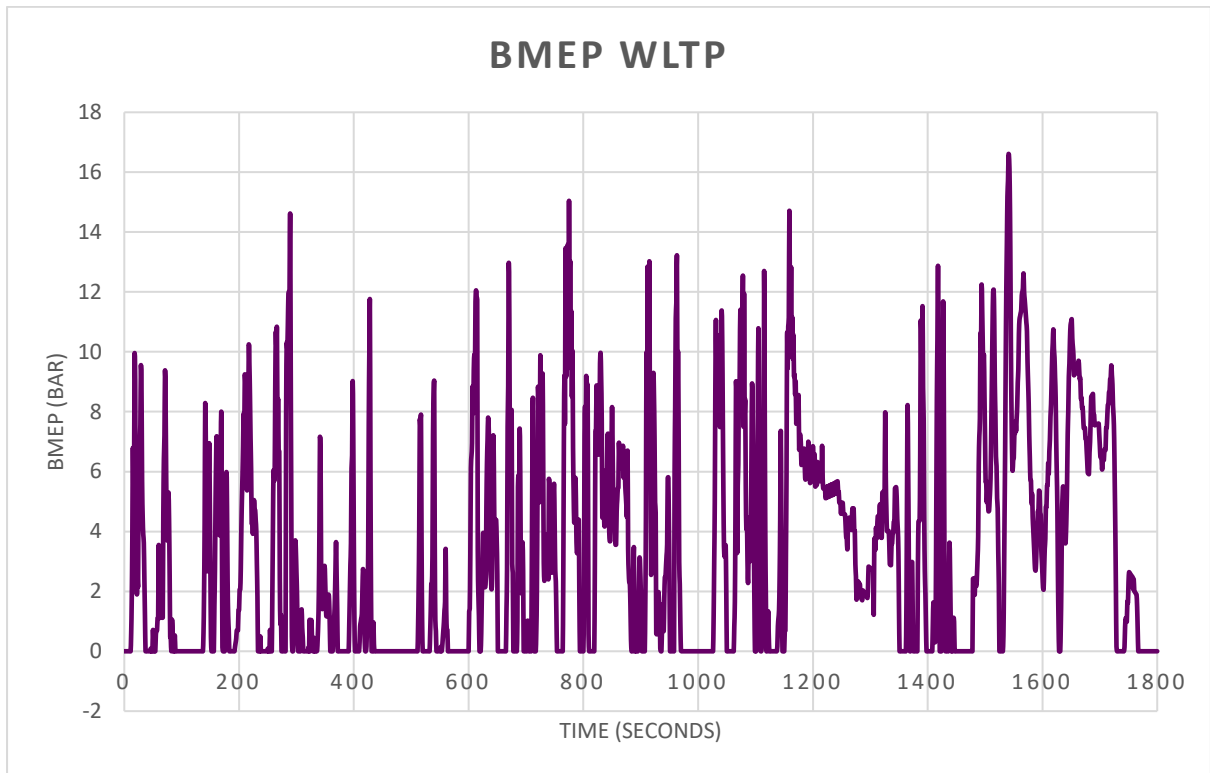


Figure 10-24: BMEP of the VW Passat in the WLTP

For the comparison of the BMEP, the engine displacement is constant, the change in BMEP is reliant of the values for RPM, which can be seen to be considerably lower for the most part of the WLTP, and the internal engine torque, which is higher through the WLTP. The gear ratios in each model stayed constant. The openness of the gear shifting profile for the WLTP allows the most appropriate gear to be selected for the engine load. The higher load in the WLTP allows for the engine to be operating in a more ideal load to RPM range for stoichiometric combustion to occur, resulting in lower emissions.

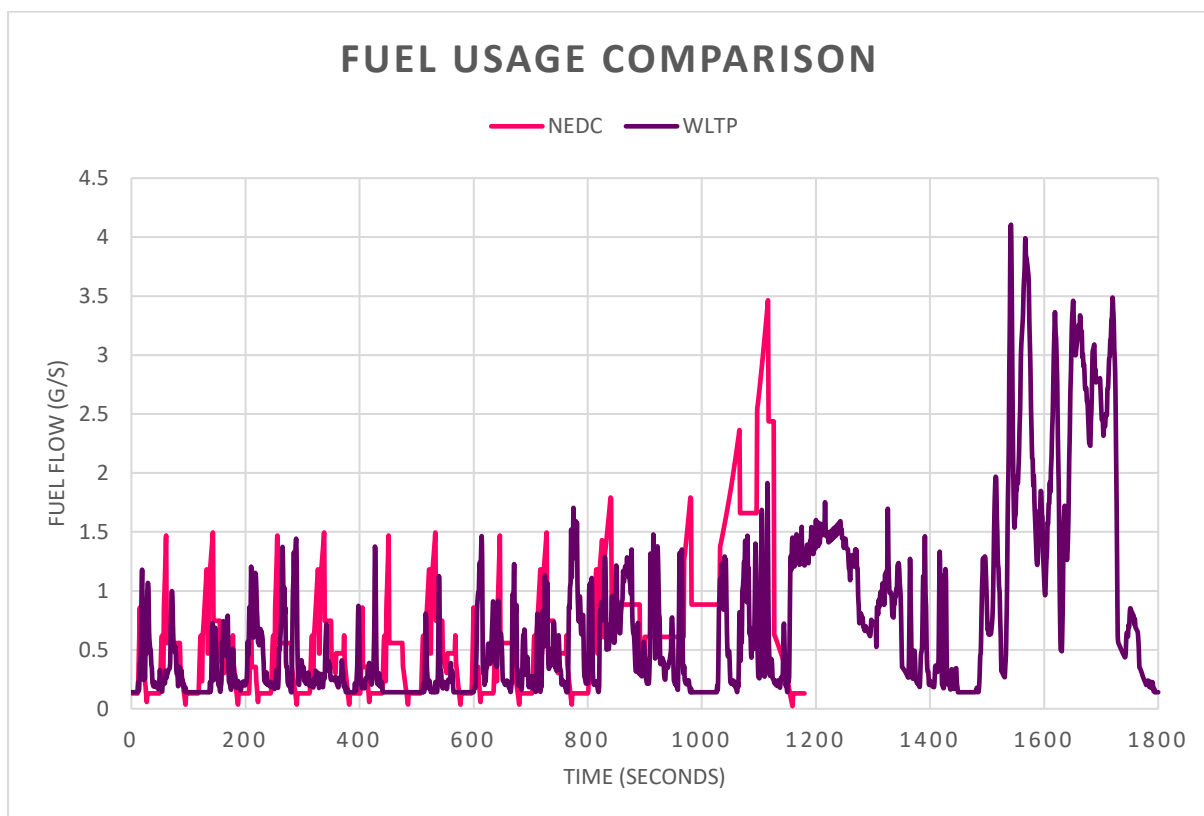


Figure 10-25: Comparison of the Fuel Flow per Second between the NEDC and WLTP.

With the higher calculated values for BMEP and lower calculated RPMS, the fuel consumption stays relatively consistent for the lower vehicle speeds.

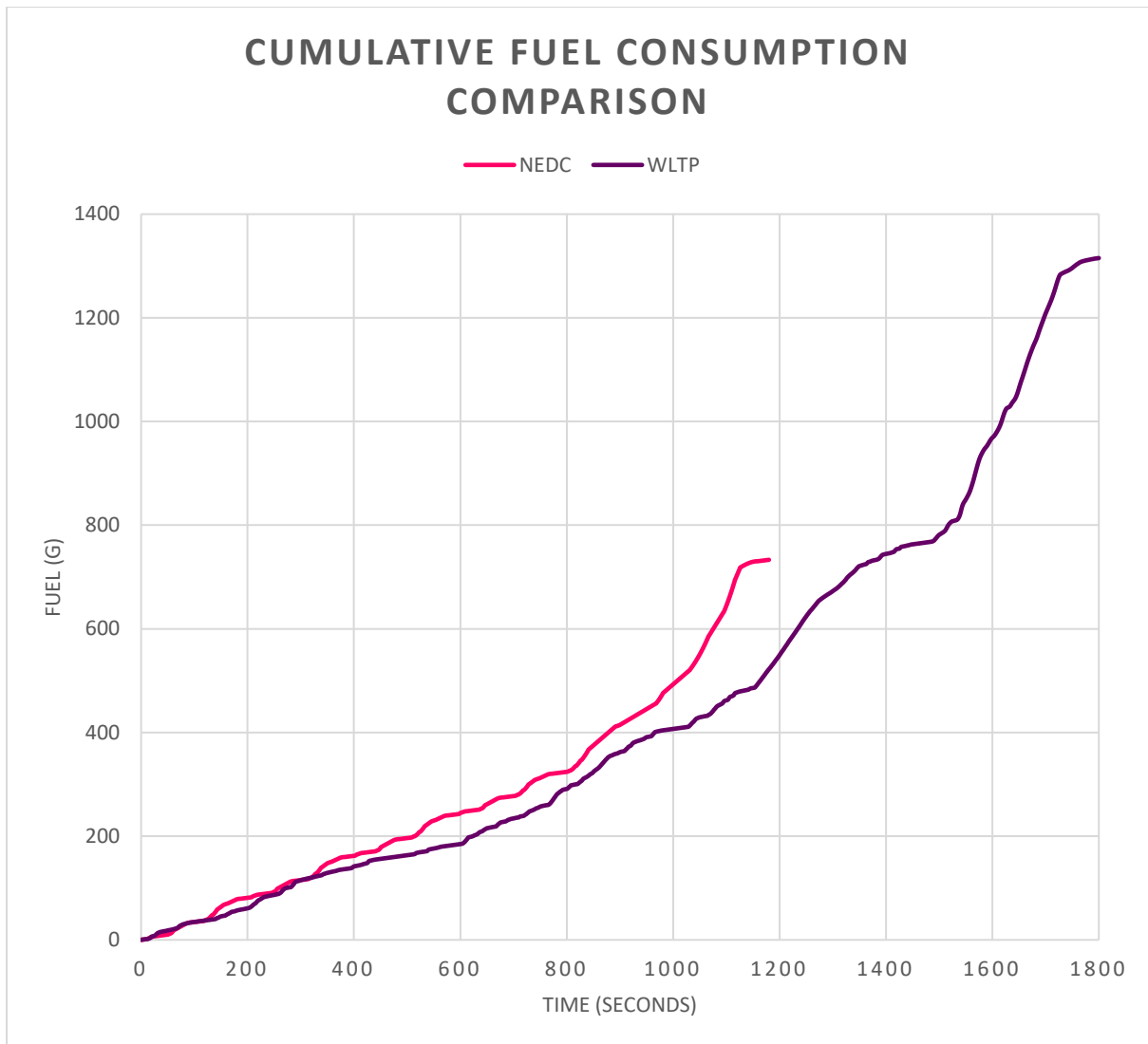


Figure 10-26: Fuel Consumption Comparison between the NEDC and WLTP

As the WLTP runs at a slower velocity for longer than the NEDC, the total fuel consumption for the NEDC, is higher than that of the WLTP, with a significant difference in velocities at those seconds.

Table 10-17: Cumulative Fuel Used in the NEDC

<i>Fuel Usage NEDC</i>		
Urban (0-800)	324.1768525	<i>g</i>
Extra Urban (801-1100)	409.0379617	<i>g</i>
Total Fuel Used	733.2148142	<i>g</i>
Fuel Per Km	66.6558922	<i>g/km</i>

Table 10-18: Cumulative Fuel Used in the WLTP

<i>Fuel Usage WLTP</i>		
<i>Total Fuel Used</i>	1315.3	<i>g</i>
<i>Fuel Per Km</i>	57.186	<i>g/km</i>

The total fuel consumption for the WLTP is 1315g, this is a 44% increase from the 733.21g used in the NEDC. Due to the extended length of the WLTP, the grams of fuel per kilometre in the WLTP is 57.186g/km, in the NEDC the grams of fuel per kilometre was 66.65g/km. This is a decrease of fuel used in the WLTP of 14.2%.

It has been reported that the fuel economy figures with the introduction of the WLTP would be reduced by about 20% (Lloyd, 2018). If the model had the full legislative parameters for the test, and was optimised correctly, the percentage decrease of fuel used would be closer to the 20% reduction found in other studies (Holder, 2018).

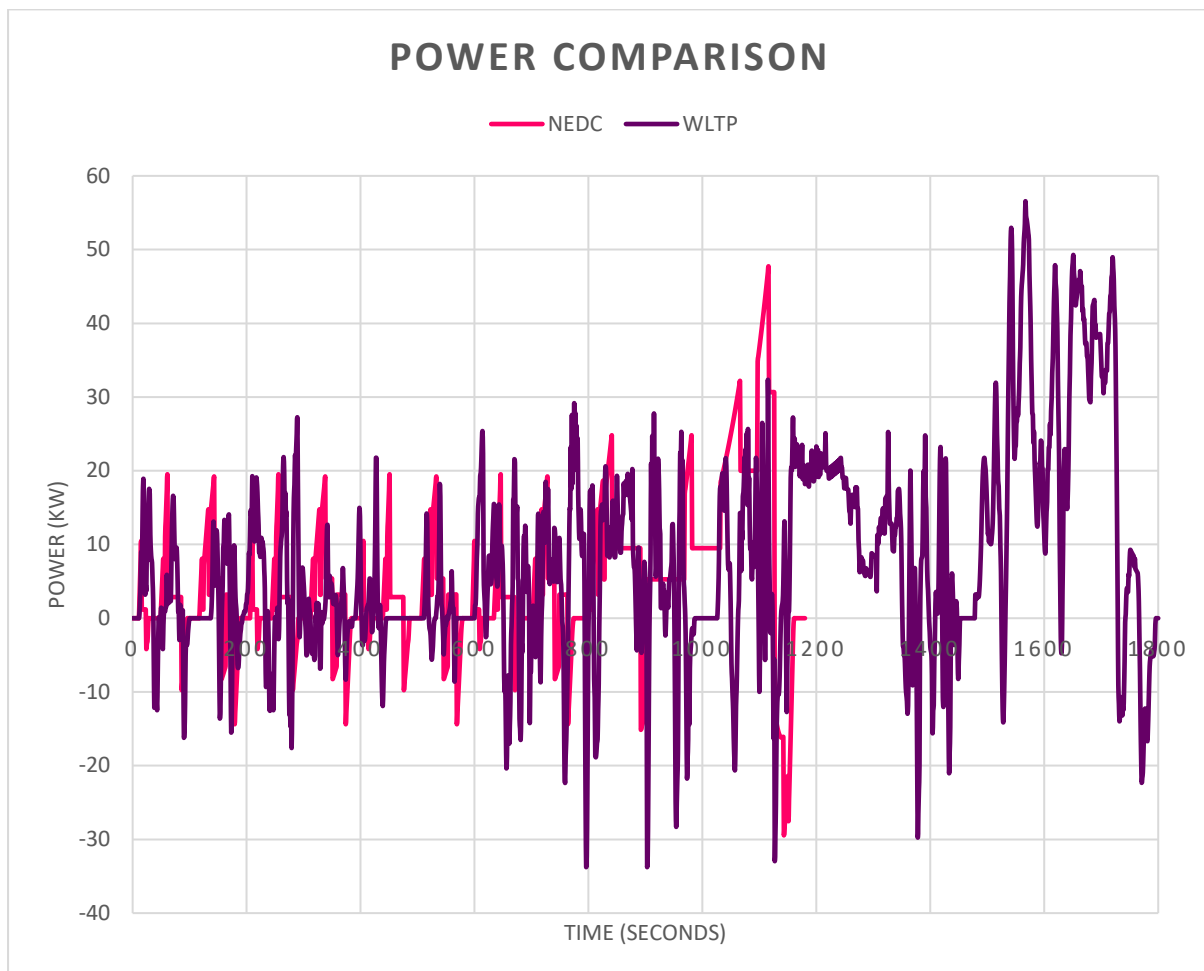


Figure 10-27: Comparison of the Power Used during the NEDC and WLTP Drive Cycles

The values for power were found using the RPM and the engine torque. With a decrease in engine speed and an increase in engine torque, it would be predicted that the power used in the NEDC and WLTP would be similar, with the increases in the WLTP from the heavier acceleration and deceleration.

Table 10-19: Maximum; Power, Engine RPM and BMEP in the NEDC test

<i>NEDC</i>		
Maximum Power	47.73818955	Kw
Max Engine RPM	3409.695712	RPM
Maximum BMEP	8.468176462	Bar

Table 10-20: Maximum; Power, Engine RPM and BMEP in the WLTP

<i>WLTP</i>		
Maximum Power	56.5637	Kw
Max Engine RPM	3130.4	RPM
Maximum BMEP	16.6114	Bar

For the NEDC the total power used over the cycle was 7071.22kW (642.84 kW/km). if the engine could recover power from braking, 1505.296kW of power would be available for recovery.

The total power used in the WLTP is 16945.3kW (736.75 kW/km) if the engine was able to recover power through braking, 3178.15kW would be available. The power used per kilometre has increased by 12.7% from the NEDC to the WLTP.

11 Combustion

The modern combustion engine has been around since 1876. The process of combustion is made up of four strokes; intake, compression, combustion and exhaust. The intake allows for the gaseous air fuel mixture to be introduced in-to the cylinder. The compression stroke increases the pressure from the gas, while also raising the temperature in the cylinder as the volume of air fuel mixture stays constant. The combustion (power or expansion) stroke begins with the ignition of the air fuel mixture. This spark increases the pressure from the gas, pushing the piston down and expanding the cylinder, this in turn produces power, by rotating the crankshaft. The exhaust stroke pushes the burnt gas out of the cylinder for the process to start again, this burnt gas is made up of the combustion emissions.

The emissions from combustion in a vehicle include: NO_x, which is formed at high temperatures, the quantity produced increasing with the increase of peak combustion temperatures; unburned HC, caused by pockets of the air fuel mixture cooling; CO, formed when the engine is over fuelled and there are not enough free oxygen molecules available for the reaction to create CO₂; and particulate matter which is the result of incomplete combustion of fuel droplets. (Taylor, 1985, p. 33)

The efficiency of the engine shows how much energy from the fuel burnt is used as mechanical energy (producing power). The ideal fuel ratio is stoichiometric, this is the ratio that manufacturers run through the load and RPM ranges seen in the drive cycle. the engine can also be run lean, which reduces the amount of fuel initially mixed with the air in the cylinder. This is one way to reduce fuel consumption, however it increases the emissions produced. Running the engine rich for combustion, over fuels the mixture in the cylinder, increasing the fuel consumption, however running the engine rich, decreases the production of some of the emissions. Manufacturers run the engine stoichiometric as this provides a balance between fuel economy and emissions levels (Heywood, 1988).

11.1 *The Model*

The model has been made from the measured cylinder pressure at each degree of the crank angle over one full cycle. This was then used to find: piston displacement, cylinder volume, logarithmic pressure and volume, polytropic index, pressure from piston motion, pressure from combustion, MFB and the work done by the pressure at each degree of crankshaft rotation.

The measured pressures are from the 1L Ford Fox engine. The example of the application of the equations was done with the data from 15 degrees ATDC (intake) at 4000RPM

Table 11-1: Engine Parameters for the Ford Fox

Engine Displacement	0.000995	m ³
Clearance Volume	0.00003705	m ³
Compression Ratio	11:01	-
Bore	0.0719	m
Stroke	0.082	m
Throw	0.041	m

Table 11-2: Connecting Rod Parameters

Connecting Rod Length	0.1375	m
Connecting Rod Width	0.025	m
Connecting Rod Depth	0.0126	m
Connecting Rod Area	0.0001257	m ²
UTS	750	n/mm ²
Youngs' Modulus	200	Gpa

The displacement of the piston is the distance travelled by the piston in relation to the rotation of the crankshaft. It is calculated using the angle of the crankshaft from TDC, crankshaft radius (crank throw), the connecting rod angle and connecting rod length.

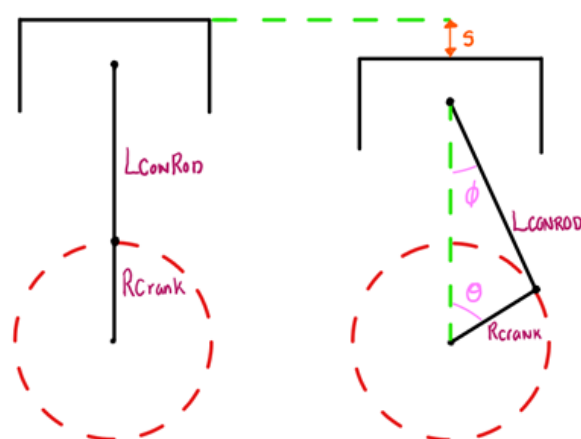


Figure 11-1: Piston Displacement

$$S = R_{Crank} + L_{ConRod} - (R_{Crank} \cos \theta + L_{ConRod} \cos \varphi)$$

Equation 11-1: Piston Displacement

The connecting rod angle is found from using the crank throw, the angle rotated from TDC in radians (the conversion can be found in the 16.1 Appendix A – Conversions) and the connecting rod length.

$$\varphi = \sin^{-1} \left\{ \frac{R_{Crank}}{L_{ConRod}} \times \sin(\theta) \right\}$$

Equation 11-2: Connecting Rod Angle

The volume of the cylinder during the cycle is calculated using the displacement, clearance volume and bore area.

$$A_B = \frac{\pi D_B^2}{4}$$

Equation 11-3: Bore Area

$$V_{Cyl} = CV + \left(\left(\frac{\pi D_B^2}{4} \right) \times S \right)$$

Equation 11-4: Cylinder Volume

The logarithmic values of the pressure and volume produces a graph that can be used to find the polytropic index. The polytropic index is used to define the process taking place and can be found from the gradient of the non-reacting portions of the compression and expansion strokes. The polytropic index relates to the thermodynamic process that is taking place; this can be at: constant pressure, constant volume, constant temperature or without heat transfer occurring

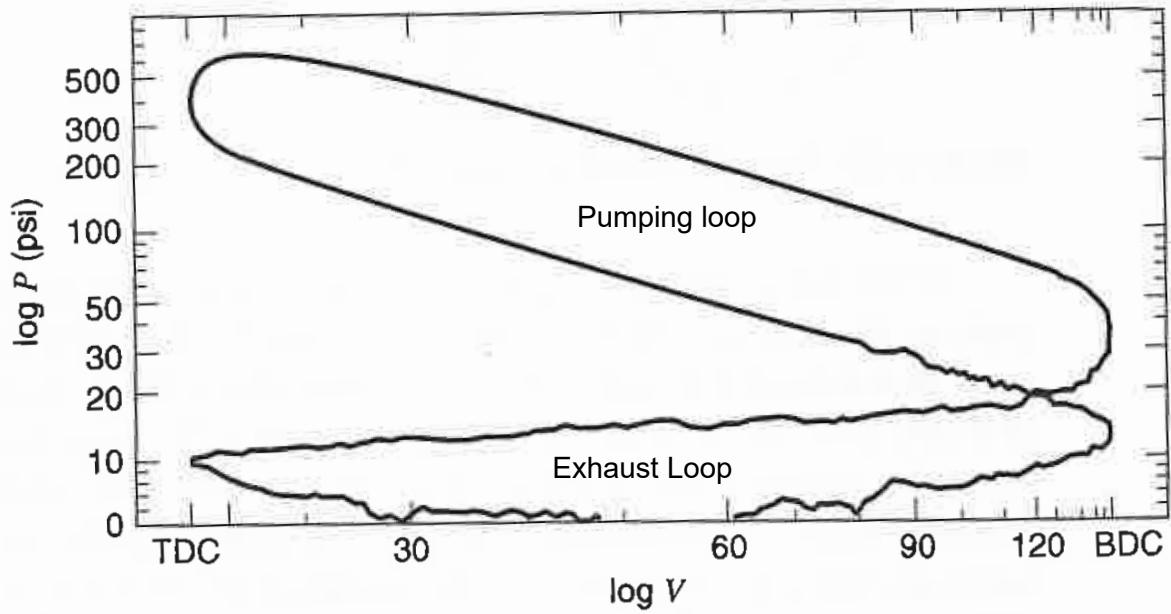


Figure 11-2: Example of Log Pressure Vs Log Volume (Ferguson & Kiskpatrick, 2000, p. 122)

$$PV^n = \text{Constant}$$

$$\text{Log}(PV^n) = \text{Constant}$$

$$\text{Log}P + \text{Log}V^n = \text{Constant}$$

$$\text{Log}P + n\text{Log}V = \text{Constant}$$

$$\text{Log}P = -n\text{Log}V + \text{Constant}$$

Equation 11-5: Derivation of the Polytropic Index

Using the basis of the last equation, the polytropic index can be found from the gradient of the pumping loop.

$$n = \frac{(\text{Log}(P) - \text{Log}(P)_1)}{(\text{Log}(V) - \text{Log}(V)_1)}$$

Equation 11-6: Polytropic Index

The cylinder pressure can be split in to two criteria: that caused by piston motion and that due to combustion. The pressure due to piston motion is found using the polytropic index, the previous and current volume, and previous pressure in the cylinder. The pressure due to combustion, is the remainder of pressure in the cylinder.

$$P_{mp} = P_{Cyl1} \times \left(\left(\frac{V_{Cyl1}}{V_{Cyl}} \right)^n \right)$$

Equation 11-7: Pressure due to Piston Motion

$$P_C = P_{Cyl} - P_{mp}$$

Equation 11-8: Pressure due to Combustion

The mass fraction burned is a ratio found during the combustion process using the extra pressure found from combustion. It is derived by accumulating the pressure from combustion from the start of combustion to the end of combustion and using the total pressure of the combustion process and the pressure from combustion at each degree from the start of combustion.

$$MFB = \frac{\sum_{SOC \rightarrow \theta} P_{C\theta}}{\sum_{SOC \rightarrow EOC} P_C}$$

Equation 11-9 Mass Fraction Burned

The work done by the pressure on the piston is calculated from the pressure and the change in volume.

$$WD_{Press} = P_{Cyl} \times (V_{Cyl} - V_{Cyl1})$$

Equation 11-10: Work Done by the Cylinder Pressure

The IMEP is the net work per unit displacement volume done by the gas during compressions and expansion (Ferguson & Kiskpatrick, 2000).

$$IMEP = \frac{\sum_{RPM} WD_{Press}}{Engine\ Displacement \times 10^5}$$

Equation 11-11: Indicated Mean Effective Pressure

11.2 Results

The results below are for 25 degrees ATDC in the combustion stroke, at 4000 RPM, the full set of results can be seen 16.3 Appendix C- Combustion Model.

Table 11-3: Initial Pressure Values at 4000RPM and 25 Degrees ATDC

Crank Angle	25	Degrees
Cylinder 1 Pressure	72.671	Bar
Cylinder 2 Pressure	76.067	Bar
Cylinder 3 Pressure	74.907	Bar

$$\varphi = \sin^{-1} \left\{ \frac{0.041}{0.1375} \times \sin(0.436) \right\}$$

$$\varphi = 0.126 \text{ Radians}$$

$$S = 0.041 + 0.1375 - \left((0.041 \times \cos(0.436)) + (0.1375 \times \cos(0.126)) \right)$$

$$S = 4.495 \times 10^{-3} m$$

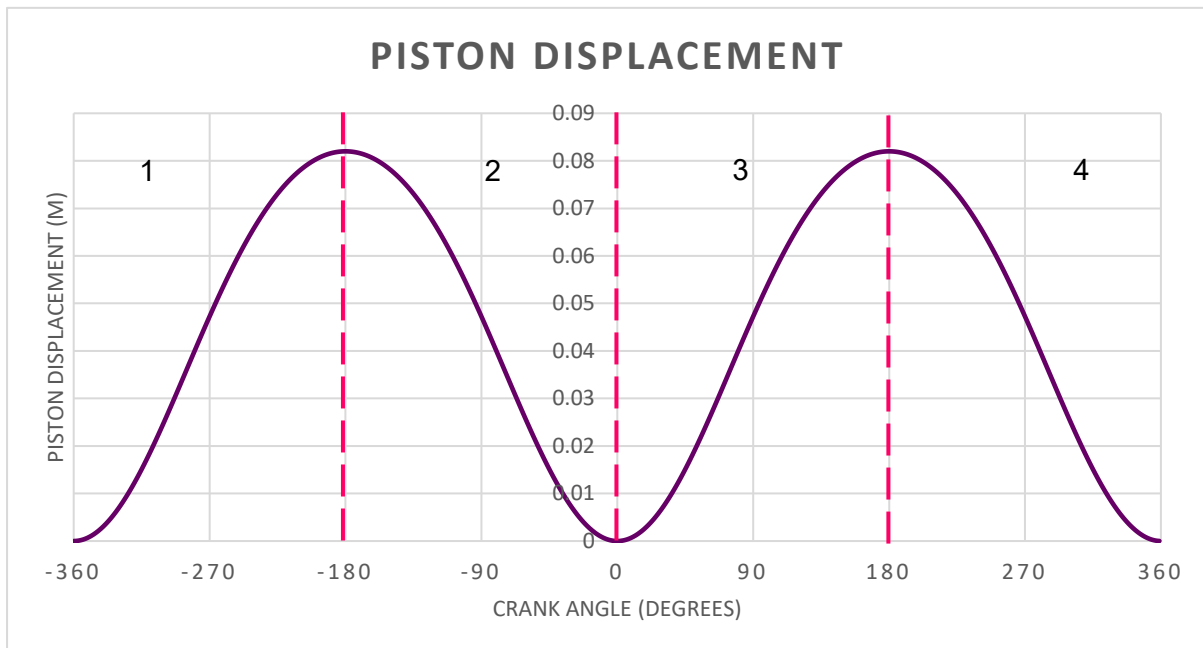


Figure 11-3: Ford Fox Piston Displacement from TDC

The piston measurements were taken from TDC which occurs at 360 and 0 degrees. The graph is separated to account for the piston movement on different strokes. 1; intake, 2; compression, 3; expansion (ignition) and 4; exhaust. The maximum distance that the piston travels is that of the length of the stroke, this is defined by cylinder and connecting rod length.

The cylinder volume is found from the clearance volume, bore area and piston displacement. When the piston is at TDC, the volume is equal to the clearance volume; when the piston is at BDC the volume is at its maximum.

$$A_B = \frac{\pi 0.0719^2}{4}$$

$$A_B = 4.06 \times 10^{-3} m^2$$

$$V_{Cyl} = 0.00003705 + (4.06 \times 10^{-3} \times 4.495 \times 10^{-3})$$

$$V_{Cyl} = 0.00003750 + 1.82497 \times 10^{-5}$$

$$V_{Cyl} = 5.53 \times 10^{-5} m^3$$

The polytropic index was derived by calculating the gradient of the logarithmic pressure vs volume graph between -26 and -1 degrees

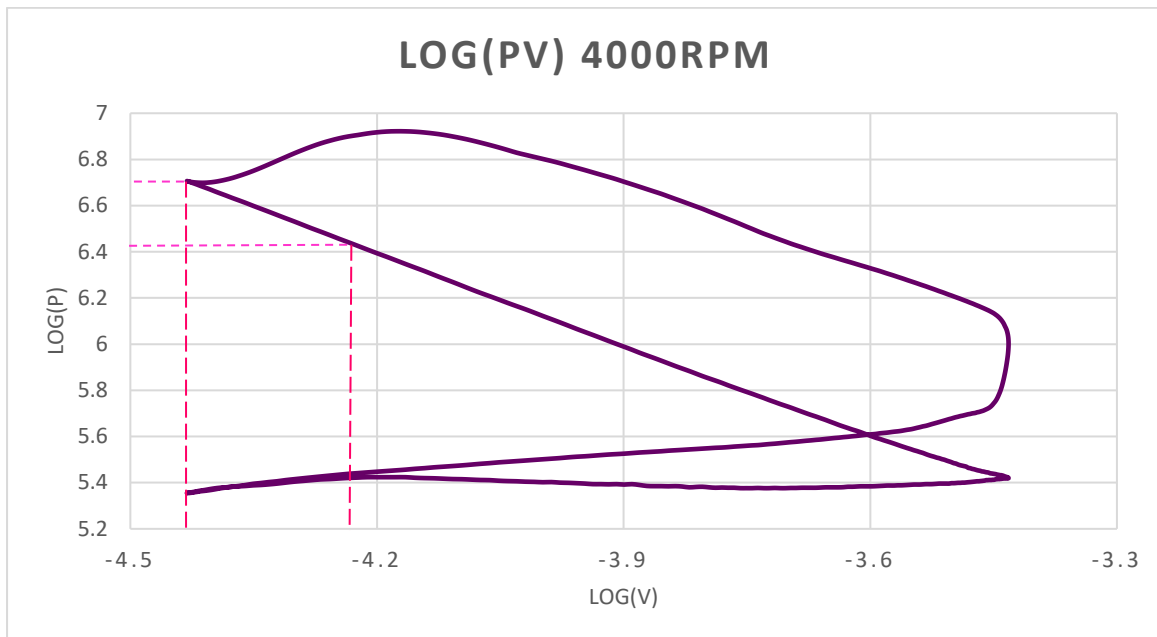


Figure 11-4: Logarithmic Pressure and Volume at 4000RPM, with Gradient Placement

$$n = - \frac{(\text{Log}(P) - \text{Log}(P)_1)}{(\text{Log}(V) - \text{Log}(V)_1)}$$

$$n = - \frac{(\text{Log}(2760500) - \text{Log}(5066700))}{\text{Log}(5.826 \times 10^{-5}) - \text{Log}(3.717 \times 10^{-5})}$$

$$n = -\frac{-0.26374}{0.195178}$$

$$n = 1.339$$

$$P_{mp} = 74.056 \times \left(\left(\frac{5.39 \times 10^{-5}}{5.53 \times 10^{-5}} \right)^{1.339} \right)$$

$$P_{mp} = 74.056 \times ((0.97468)^{1.339})$$

$$P_{mp} = 74.056 \times (0.96625)$$

$$P_{mp} = 71.555 \text{ Bar}$$

$$P_C = 76.067 - 71.555$$

$$P_C = 4.512 \text{ Bar}$$

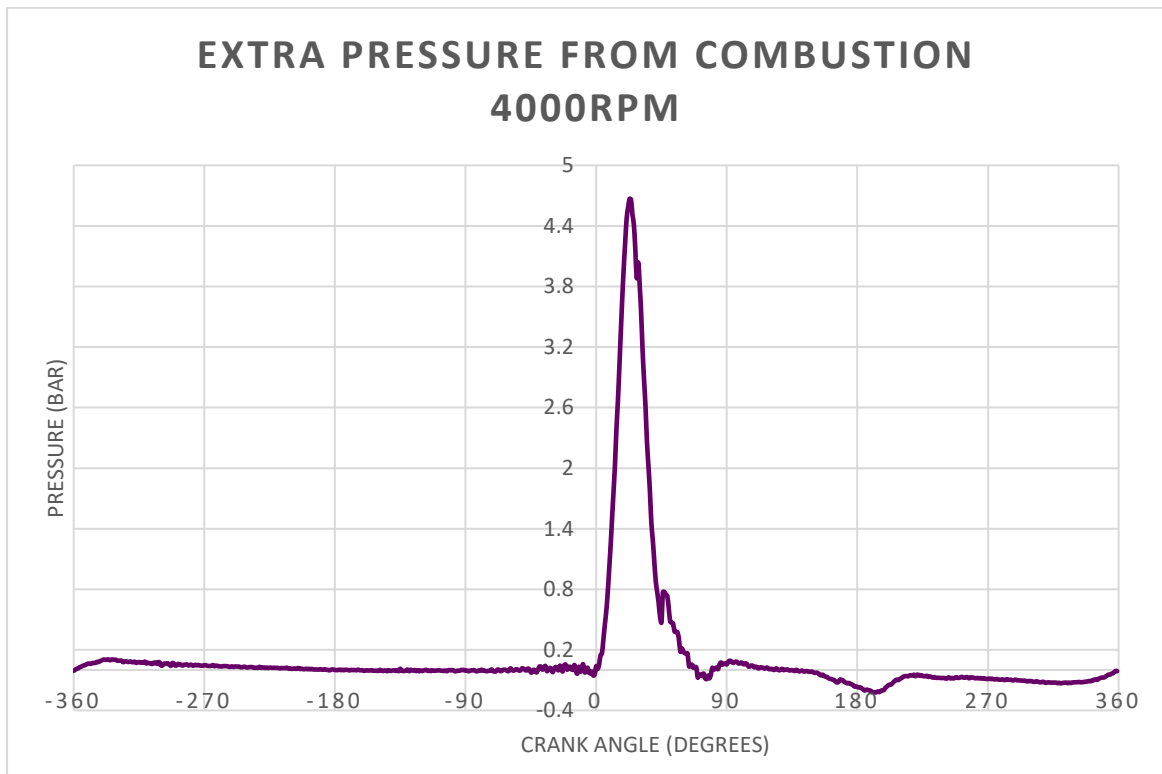


Figure 11-5: Extra Pressure from Combustion at 4000RPM

The mass fraction burned is the cumulative heat released during the burn duration.

At 25 degrees ATDC the cumulative extra pressure from combustion from SOC to the crank angle is 56.294 Bar. The total additional pressure from combustion at 4000RPM was found to be 109.297 Bar.

$$MFB = \frac{56.294}{109.297}$$

$$MFB = 0.515$$

The start of combustion at 4000RPM occurs at 1-degree BTDC and ends at 41 degrees ATDC, where 90% of the mass has been burned, creating a burn duration of 42 degrees.

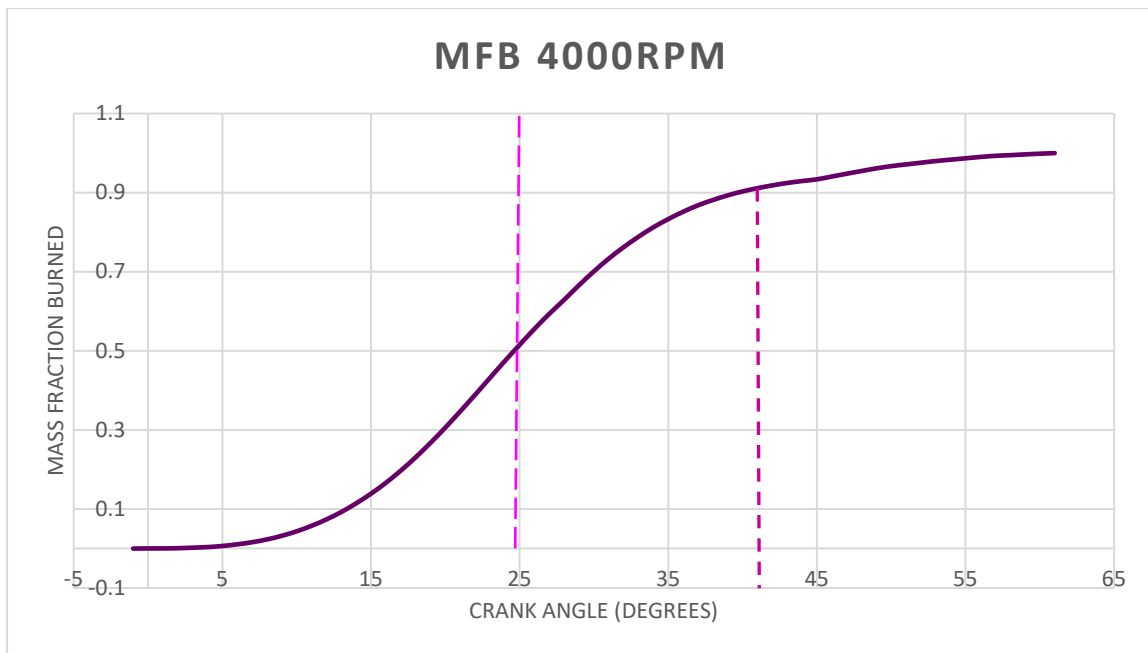


Figure 11-6: Mass Fraction Burned at 4000RPM

$$WD_{press} = 7606700 \times (5.67 \times 10^{-5} - 5.53 \times 10^{-5})$$

$$WD_{press} = 7606700 \times 1.4 \times 10^{-6}$$

$$WD_{press} = 10.659 Nm$$

The IMEP is the pressure that would act on the piston to produce the power required, with the inclusion from mechanical and frictional losses.

$$IMEP = \frac{1708.449 \times 3}{0.000995 \times 10^5}$$

$$IMEP = \frac{5125.347}{99.5}$$

$$IMEP = 51.511 \text{ Bar}$$

11.3 Analysis

The graphs in 16.3.1 Logarithmic Pressure and Volume show that as the RPM increases the pumping losses increase. During the exhaust stroke, work is done on the gas, this 'negative' work is the useful work lost which increases with RPM, as the cylinder pressures are higher.

Table 11-4: Calculated Polytopic Index for the Range of RPM

<i>RPM</i>	<i>Polytopic Index</i>
1000	1.30710125
1250	1.296743863
1500	1.303393589
2000	1.307196663
2500	1.316049254
3000	1.326059917
3500	1.326616789
4000	1.33854746
4500	1.335306809
5000	1.332258472
5500	1.337249875
6000	1.344741293
6500	1.34254206

Heywood states that the polytropic index for compression is 1.3 (Heywood, 1988, p. 384). As the values found from the author's model are in the region of 1.3, the development of the model was successful. At higher RPM where the polytropic index has risen to 1.34, this could be due to issues with the identification of the crank angle at which is the start of combustion.

The extra pressure due to combustion (16.3.2 Extra Pressure From Combustion) is greatest during the spark and combustion period. The peak combustion pressure aligns with that of maximum cylinder pressure. As the cylinder volume is constant across the RPM range, the extra pressure from combustion is dependent on the pressure measured in the cylinder, the pressure from piston motion and the polytropic index. At higher RPMs the increased calculated polytropic index would result in slightly lowered pressure due to piston motion. However, with the initial pressures already greater than those at lower RPMs, the pressure due to combustion does not increase any more than it does at lower RPMs.

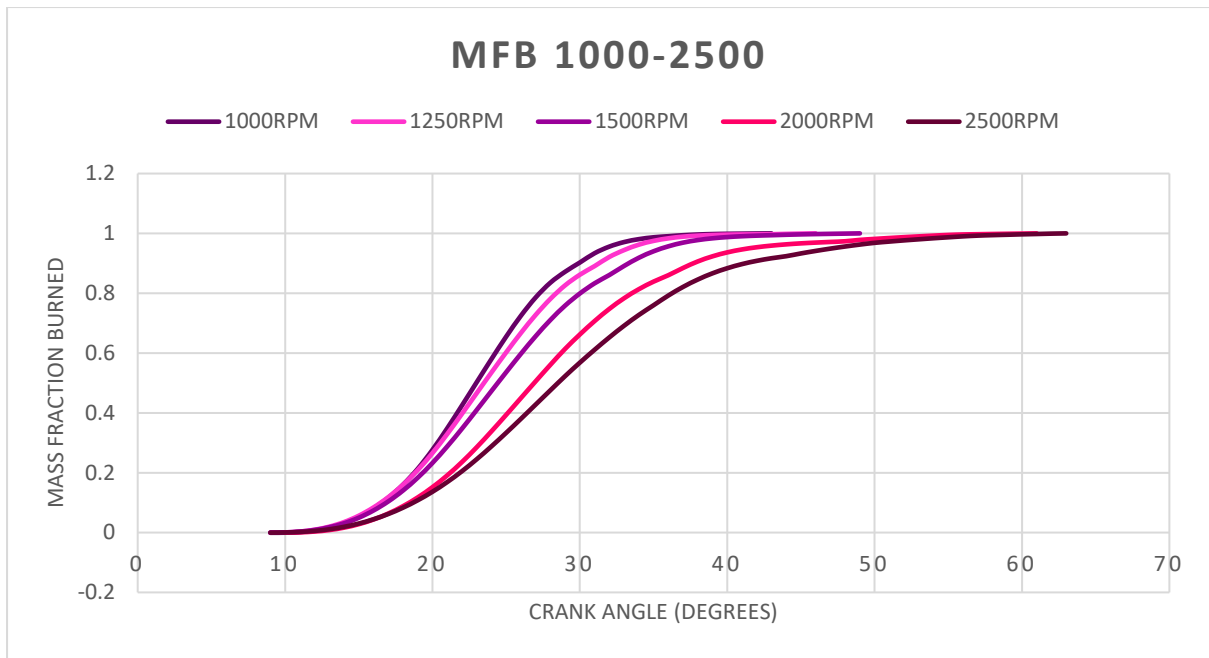


Figure 11-7: Comparisons of Mass Fraction Burned of the RPM Range 1000-2500

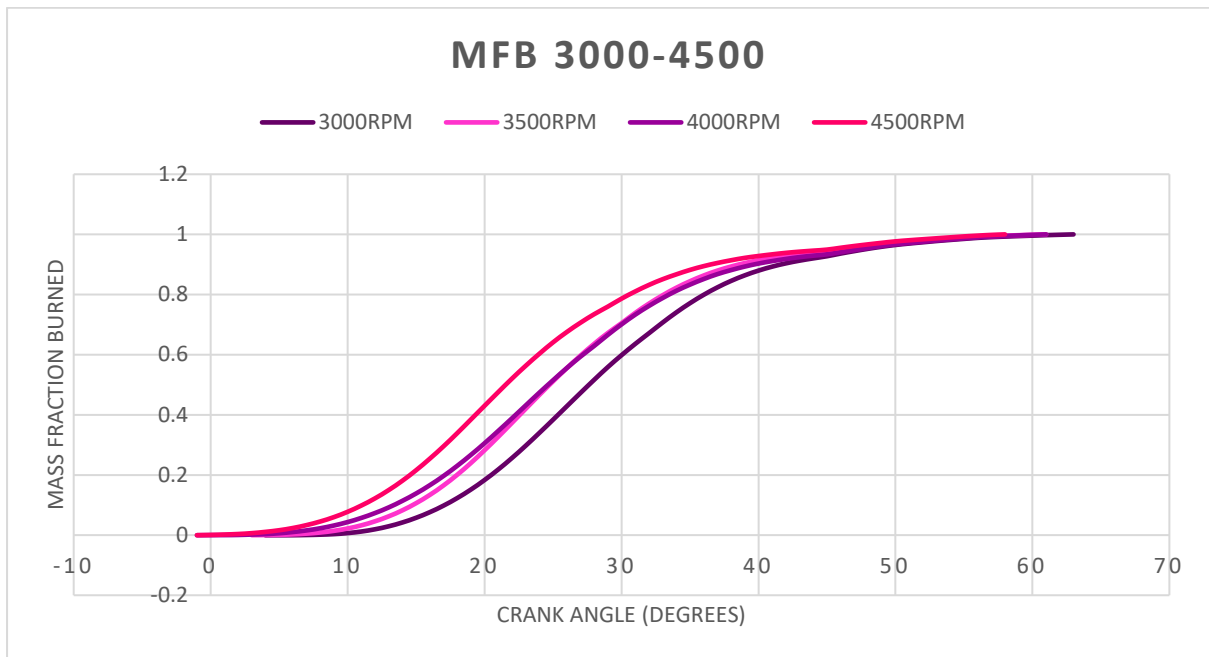


Figure 11-8: Comparisons of Mass Fraction Burned for the RPM Range 3000-4500

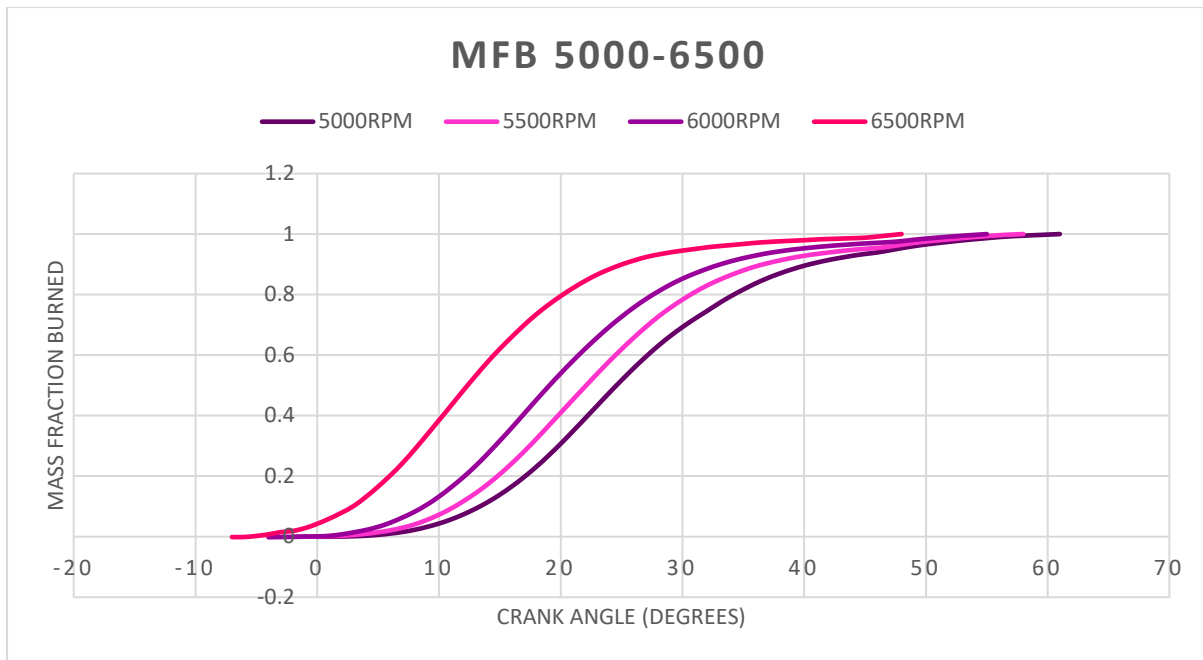


Figure 11-9: Comparison of Mass Fraction Burned of the RPM Range 5000-6500

As flame speed stays proportional to the piston speed, the burn duration (degrees) stays consistent when the RPM is increased. However, the increase in RPM decreases the available time (seconds) for combustion to occur. With the decrease in time (seconds) for combustion to occur the position for start of combustion changes with RPM. The higher the RPM the earlier the start of combustion, to ensure there is enough time for full combustion.

The IMEP during the cycle ranges from 14-27 Bar peaking at 3000RPM. Over 3000RPM the IMEP decreases, this is due to the higher initial cylinder pressures and larger pumping losses caused by exhaust port size restrictions.

12 Friction

The frictional work from an engine is the energy required to overcome the frictional forces produced by the engine. There are three main sections in which frictional losses occur: pumping friction, from drawing the mixture through the intake and pushing the burned gas through the exhaust; rubbing friction, the motion friction which affects the piston assembly, crankshaft and valve mechanisms; and accessory work, the friction from the driving of essential extras such as fans and pumps (Heywood, 1988). This chapter looks at modelling the rubbing friction occurring in the cylinder, from the piston assembly.

Rezeka and Henein identified that the piston assembly friction counted towards 60-70% of the overall engine losses (Rezeka & Henein, 1984). However later research contradicted this with friction losses due to piston assembly measured at only 20-30% (Patterson & Mehment Uras, 1987). Following advancements in technology and design, frictional losses are now believed to be as low as 15-20% (Nagar & Miers, 2011). Despite these reductions, under idle conditions the frictional forces contribute to 100% of engine losses. (Heywood, 1988)

The calculation techniques for piston friction are the floating liner method (Gore, et al., 2014) and the use of instantaneous IMEP (Patterson & Mehment Uras, 1987) to determine the forces acting on the piston.

As stated above 15-20% of the power from the engine, is used to overcome the frictional forces being produced in the engine. Engine friction is a major factor in an engines efficiency. Developing material properties such as: roughness, plasticity, sheer strength and hardness can reduce the friction coefficient, in turn reducing the energy from combustion needed to overcome the frictional forces.

12.1 Model Fundamentals

The model created by the author has been made from the measured cylinder pressure at each degree of the crank angle over one full cycle. This was used to find the gas force in each cylinder, inertia force of the piston assembly, net force, connecting rod force, piston displacement, cylinder volume and work done by the pressure at each degree of the crankshaft rotation.

The measured pressures are from the 1L Ford Fox engine. The example of the application of the equations was done with the data from 15 degrees ATDC (intake) at 4000RPM

Table 12-1: Ford Fox Engine Parameters

Engine Displacement	995	cc
Bore	0.0719	m
Stroke	0.082	m
Throw	0.041	m
Friction Coefficient	0.1	-

Table 12-2: Ford Fox Connecting Rod Parameters

Connecting Rod Length	0.1375	m
Connecting Rod Width	0.025	m
Connecting Rod Depth	0.0126	m
Connecting Rod Area	0.0001257	m ²
UTS	750	N/mm ²
Youngs' Modulus	200	GPa

<i>Masses</i>		
Connecting Rod	0.355	kg
Piston	0.22	kg
Gudgeon Pin	0.067	kg
Piston Ring	0.0008	kg
Reciprocating	0.4077333	kg

Table 12-3: - Piston Ring Measurements

<i>Piston Ring</i>		
Diameter	0.068	m
Thickness	0.001	m

Table 12-4: Given Values for 4000RPM

<i>Crank Angle</i>	-345	Degrees
<i>Cylinder 1 Pressure</i>	1.954	Bar
<i>Cylinder 2 Pressure</i>	2.431	Bar
<i>Cylinder 3 Pressure</i>	2.369	Bar
<i>Cylinder Volume</i>	44.38	cc

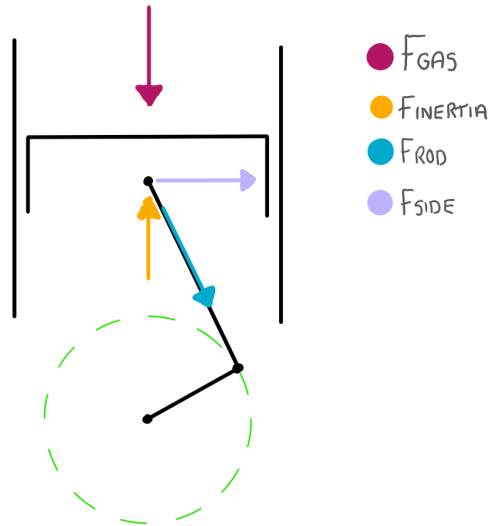


Figure 12-1: Force Diagram Acting on the Piston

The gas force (Pink, Figure 12-1) is the force produced from combustion. It is the downward force that acts on the pistons top surface and the first piston ring, creating a seal against the cylinder wall. The peak gas force occurs just after TDC (expansion) on the ignition stroke and calculated from the cylinder pressure and bore area.

$$F_{Gas} = P_{Cyl} \times A_B$$

Equation 12-1: Gas Force

The inertia force (Orange, Figure 12-1) acting on the piston assembly is the acceleration from the oscillating motion of the piston it reaches its maximum at TDC.

$$First\ Order\ Acceleration = (R_{Crank} \omega_{Eng}^2 \times \cos(\theta))$$

Equation 12-2: First Order Inertial Force

$$Second\ Order\ Acceleration = \left(\frac{R_{Crank}}{L_{ConRod}} \times R_{Crank} \times \omega_{Eng}^2 \times \cos(2\theta) \right)$$

Equation 12-3: Second Order Inertial Force

$$\alpha_{In} = First\ Order\ Acceleration + Second\ Order\ Acceleration$$

Equation 12-4: Inertial Acceleration

$$F_{Inertia} = M_{Rec} \times \alpha_{Inertia}$$

$$F_{Inertia} = M_{Rec} \times \left\{ (R_C \omega^2 \times \cos(\theta)) + \left(\frac{R_C}{L_C} \times R_C \times \omega^2 \times \cos(2\theta) \right) \right\}$$

Equation 12-5: Inertia Force

$$M_{Rec} = M_P + M_G + M_{PR} + \frac{1}{3} M_{Conrod}$$

Equation 12-6: Reciprocating Mass

The net force is the overall force acting on the piston.

$$F_{Net} = F_{Gas} - F_{Inertia}$$

Equation 12-7: Net Force Acting on Piston

The rod force (Teal, Figure 12-1) is the downward force acting through the centre of the connecting rod at each degree of the crankshaft.

$$F_{Rod} = \frac{F_{Net}}{\cos(\varphi)}$$

Equation 12-8: Force in the Connecting Rod

The work done by the pressure shows the available energy from the combustion process; using the average in the change of pressures (Nm) between degrees and the difference in volume.

$$WD_{Press} = \left(\frac{P_{CylMax} + P_{CylMax1}}{2} \right) \times (V_{Cyl} - V_{Cyl1})$$

Equation 12-9: Work Done by the Combustion Pressure

The side force (Lilac, Figure 12-1) acting on the piston skirt, is dependent on the crank angle, cylinder pressure, piston speed, acceleration and connecting rod angle.

$$F_{Side} = F_{Rod} \times \sin(\varphi)$$

Equation 12-10: Side Force on the Piston

The friction force in the cylinder is the force acting on the side of the piston while it is rubbing against the cylinder wall.

$$F_{FPS} = \mu_{Cyl} \times F_{Side}$$

Equation 12-11: Friction Force on the Piston Skirt

The work done by the piston is the energy that is required from the energy in the combustion to process to overcome the frictional force created by the piston skirt.

$$WD_{PS} = F_{FPS} \times (S - S_1)$$

Equation 12-12: Work Done by the Piston Skirt

The radial force (F_{Rad} , Figure 12-11) is created from the pressure of the combustion gas pushing against the inner area of the piston ring, creating the seal against the cylinder wall.

$$F_{Rad} = P_{CylMax} \pi D_{PR} t_{PR}$$

Equation 12-13: Radial Piston Ring Force

$$F_{FPR} = F_{Rad} \times \mu_{Cyl}$$

Equation 12-14: Piston Ring Friction Force

The values are then used to find the total frictional force.

$$F_F = F_{FPR} + \text{Measured Friction Force}$$

Equation 12-15: Total Frictional Force from the Piston Rings

The work done by the piston rings is the energy from combustion needed to overcome the frictional forces produced. The work done was calculated each for degree and is dependent on the distance moved by the piston and the frictional force from the piston rings per degree.

$$WD_{PR} = F_F \times (S - S_1)$$

Equation 12-16: Work Done by the Piston Rings

FMEP calculates the effective pressure produced by the piston.

$$FMEP = \frac{\sum WD_{Piston} \times \text{No of Cylinders}}{\text{Engine Displacement} \times 10^5}$$

Equation 12-17: FMEP

12.2 Results

The cylinder displacement at 15 degrees ATDC (expansion)

$$\varphi = \sin^{-1} \left\{ \frac{0.041}{0.1375} \times \sin(-6.02) \right\}$$

$$\varphi = 0.07725 \text{ Radians}$$

$$S = 0.041 + 0.1375 - (0.041\cos(-6.02) + 0.1375\cos(0.07725))$$

$$S = 0.1785 - 0.1766$$

$$S = 1.82 \times 10^{-3}$$

The gas force was calculated for each cylinder and the maximum across the engine used for future calculations. The bore area found Equation 11-3.

$$F_{Gas1} = (1.954 \times 10^5) \times 4.06 \times 10^{-3}$$

$$F_{Gas1} = 793.36N$$

$$F_{Gas2} = (2.431 \times 10^5) \times 4.06 \times 10^{-3}$$

$$F_{Gas2} = 987.04N$$

$$F_{Gas3} = (2.369 \times 10^5) \times 4.06 \times 10^{-3}$$

$$F_{Gas3} = 961.86N$$

The average gas force across the cylinders is 914.09N and the maximum gas force across the cylinders at 15° TDC (intake) is 987.04N.

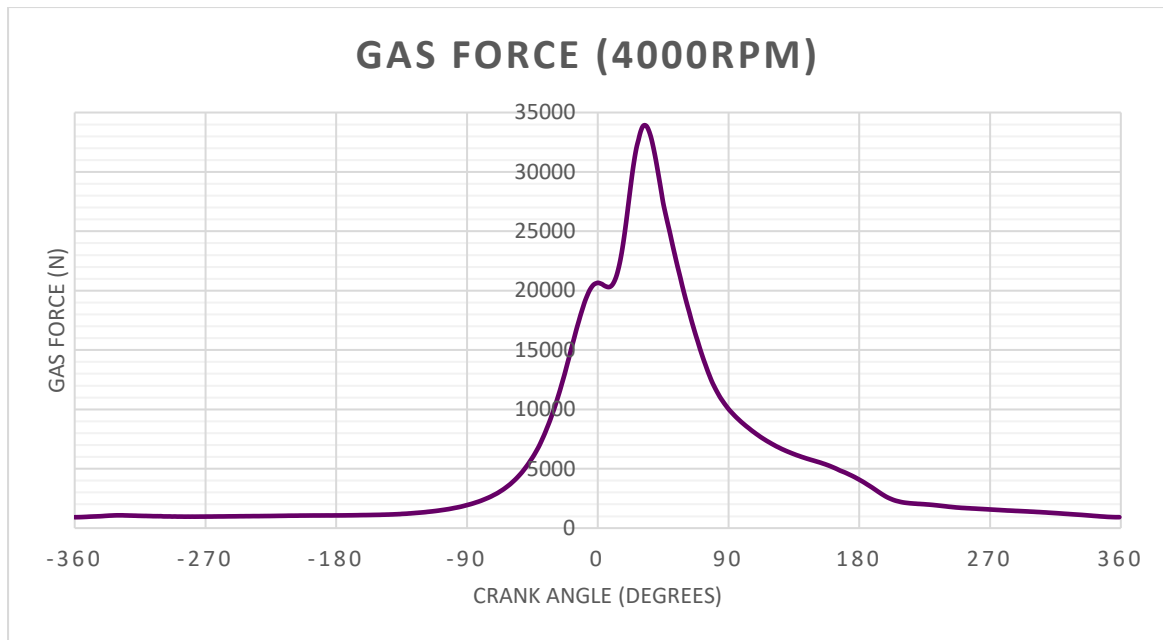


Figure 12-2: Maximum Gas Force at 4000RPM

The peak gas force across the range of RPM occurs between 19-37 degrees ATDC (expansion).

Table 12-5: Peak Cylinder Gas Forces and the Degrees ATDC at which they occur for the RPM Range

<i>RPM</i>	<i>Degrees ATDC</i>	<i>Peak Gas Force (N)</i>
1000	29	22178.85
1250	30	27114.03
1500	31	30595.25
2000	34	33796.72
2500	37	33435.77
3000	36	33363.09
3500	33	35822.35
4000	32	33927.86
4500	28	33993.64
5000	31	30723.96
5500	29	32564.04
6000	26	33161.3
6500	19	34833.29

The peak gas forces for one cycle at each RPM can be seen in the table above. These are important for engine design, as the force acts outwards putting force also on the cylinder walls and cylinder head. The engine design therefore needs to ensure that the peak forces can be endured and that any fluctuations that may occur in engine running wouldn't potentially cause any damage.

$$M_{Rec} = 0.22 + 0.067 + (3 \times 0.0008) + \left(\frac{1}{3} \times 0.355\right)$$

$$M_{Rec} = 0.408kg$$

Equation 12-18: Reciprocating Mass Value

The reciprocating mass is that of the piston assembly that is assumed to be moving up and down in relation to the crank rotation.

$$First\ Order\ Acceleration = (0.041 \times 418.88^2 \times \cos(-6.02))$$

$$First\ Order\ Acceleration = 6948.72\ Rads/s^2$$

$$Second\ Order\ Acceleration = \left(\frac{0.041}{0.1375} \times 0.041 \times 418.88^2 \times \cos(2 \times -6.02)\right)$$

$$Second\ Order\ Acceleration = 1875.69\ Rads/s^2$$

$$\alpha_{In} = 6948.72 + 1875.69$$

$$\alpha_{In} = 8806.41\ Rads/s^2$$

$$F_{Inertia} = 0.408 \times \{8806.41\}$$

$$F_{Inertia} = 2590.67N$$

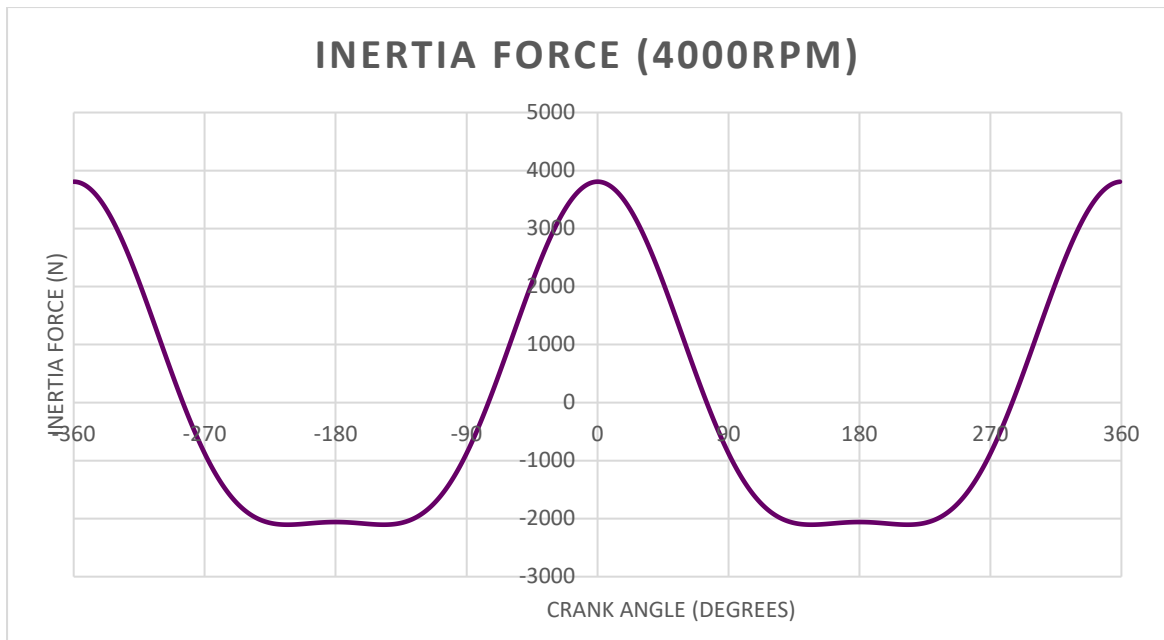


Figure 12-3: Inertia Force at 4000RPM

As the inertia is dependent on the vertical piston acceleration and subsequently the angular velocity of the engine, the inertia forces increase for higher engine speeds. The graph above shows the fluctuations in the load of the piston from positive to negative.

$$F_{Net} = 961.86 - 2590.67$$

$$F_{Net} = -2603.63$$

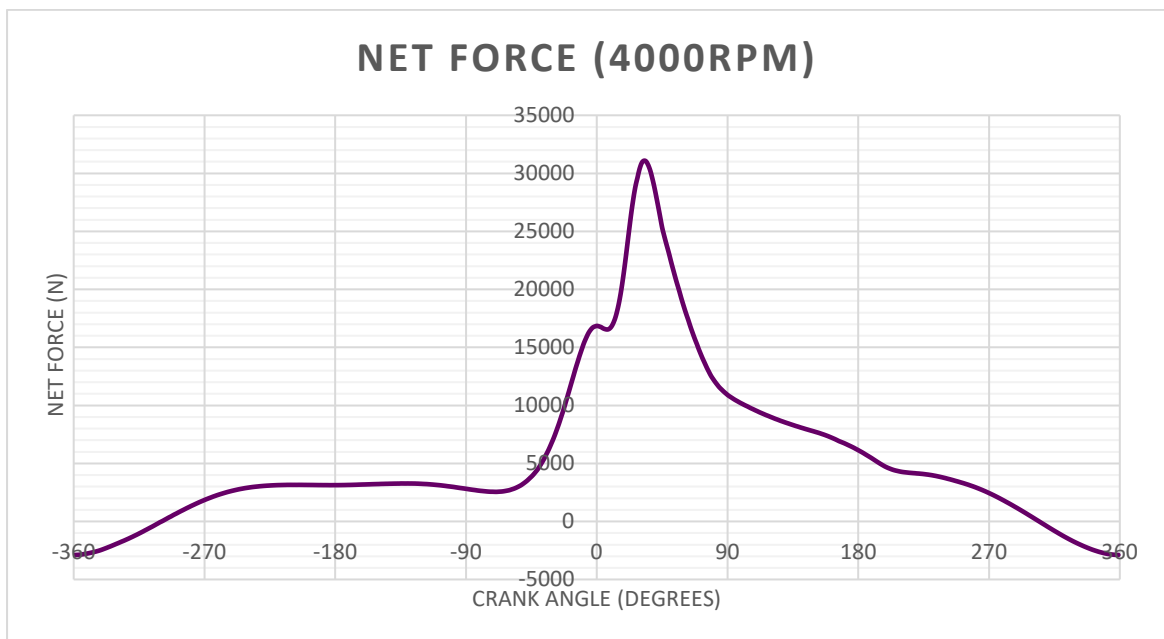


Figure 12-4: Net Force at 4000RPM

$$F_{Rod} = \frac{-2603.63}{\cos(0.077)}$$

$$F_{Rod} = -2611.42$$

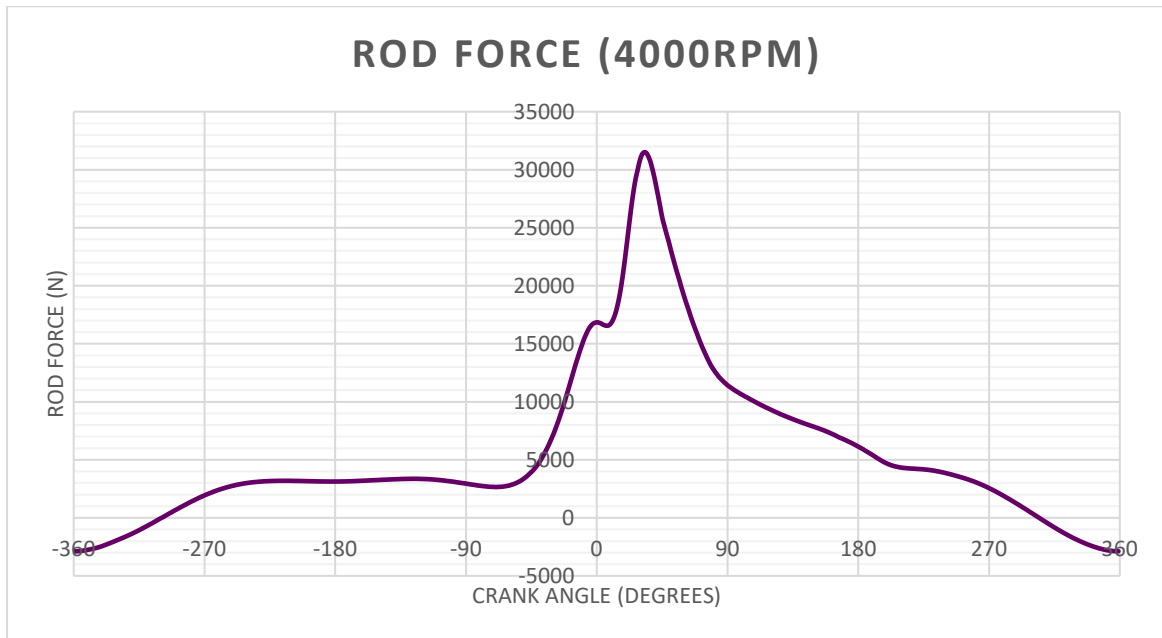


Figure 12-5: Force Acting Through the Connecting Rod at 4000RPM

A negative net and rod force shows that the inertia force is doing more work and that the compression force and the connecting rod is being put under tensile stress.

$$WD_{Press} = \left(\left(\frac{2.431 + 2.422}{2} \right) \times 10^5 \right) \times ((44.38 - 43.44) \times 10^{-6})$$

$$WD_{Press} = 0.23 \text{ Nm}$$

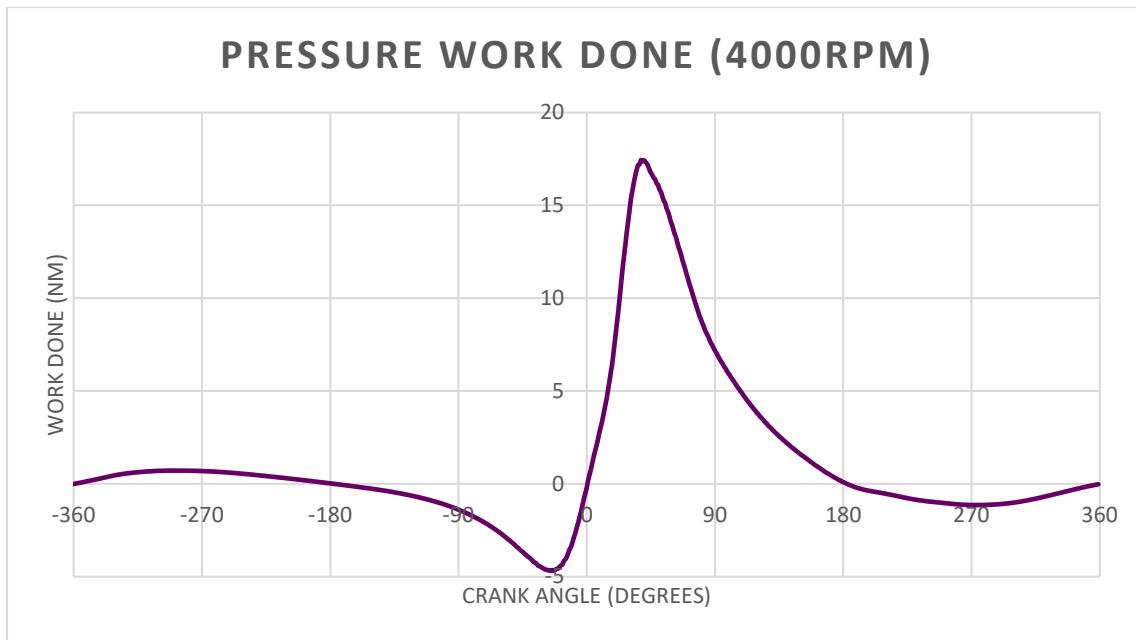


Figure 12-6: Work Done by the Pressure from Combustion at 4000RPM

The work done by the pressure from combustion, is the potential work to be transferred to power. Where the work done is negative this shows where the engine is having to use power to compress the gas, this occurs where the gas force is increasing quickest.

12.2.1 Piston Skirt

The side force presses the piston skirt against the cylinder wall, changing the direction of application throughout the cycle.

$$F_{Side} = -2611.42 \times \sin(0.077)$$

$$F_{Side} = -201.54N$$

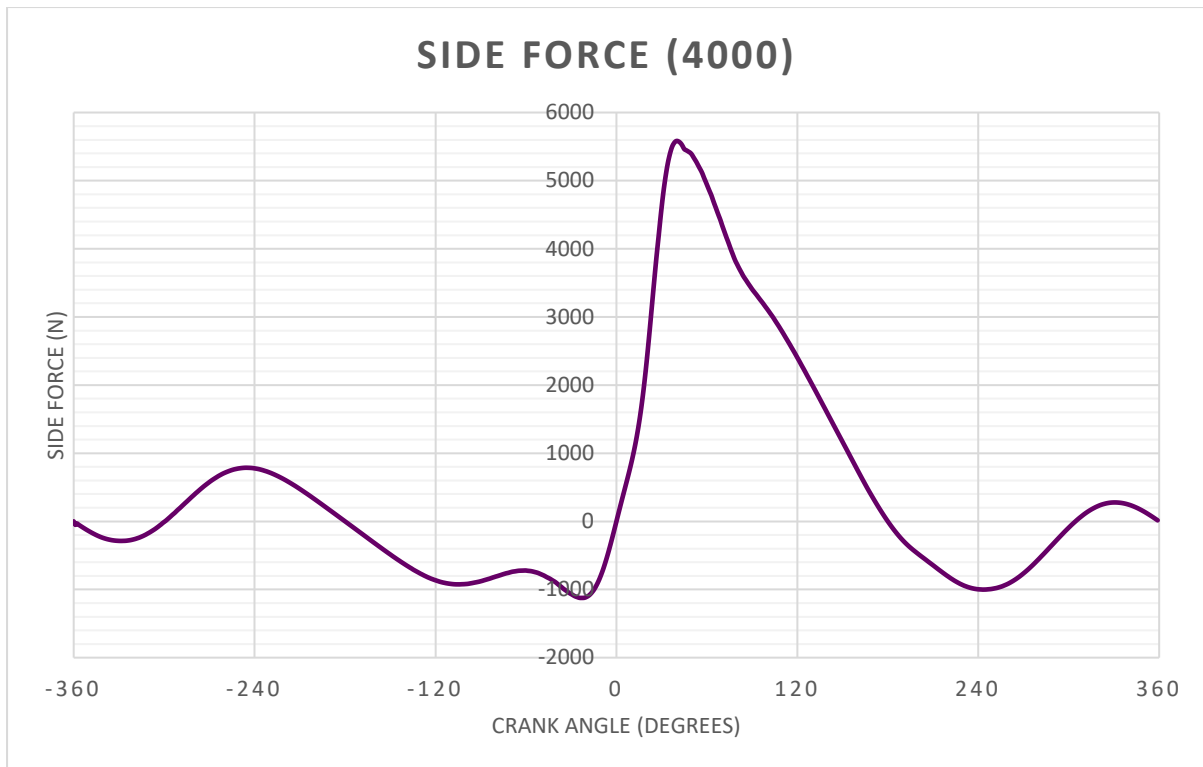


Figure 12-7: Side Force Acting on the Piston Skirt at 4000RPM

The largest side force occurs during the expansion stroke, when the cylinder pressures are the largest. In a clockwise rotating engine, the positive forces are acting to the right.

At lower speeds the gas force is predominant even in the early compression phase, thereby inducing contact on the anti-thrust (Left) side as the piston is thrust upwards. Not until the crank angle ranges 30-10 BTDC fired does the bottom end of the skirt change contact from the anti-thrust side to the thrust (Right) side. (MAHLE GmbH, 2016, pp. 211-212)

$$F_{FPS} = 0.1 \times -201.54$$

$$F_{FPS} = -20.15N$$

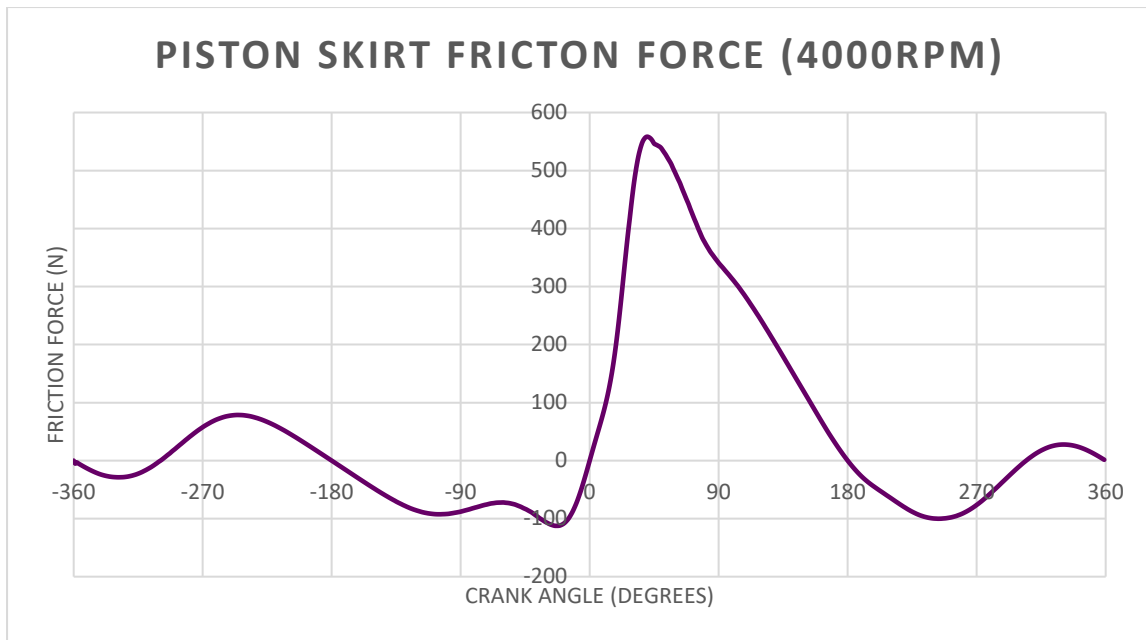


Figure 12-8: Friction Force from the Piston Skirt against the Cylinder Wall at 4000RPM

The frictional force from the piston against the cylinder wall, changes from negative to positive according to which side the force is acting upon. The 'negative' force acts to the left of the piston causing the piston skirt on the right to be rubbing against the cylinder wall.

$$WD_{PS} = -20.15 \times (1.81 \times 10^{-3} - 1.58 \times 10^{-3})$$

$$WD_{PS} = -4.66 \times 10^{-3} Nm$$

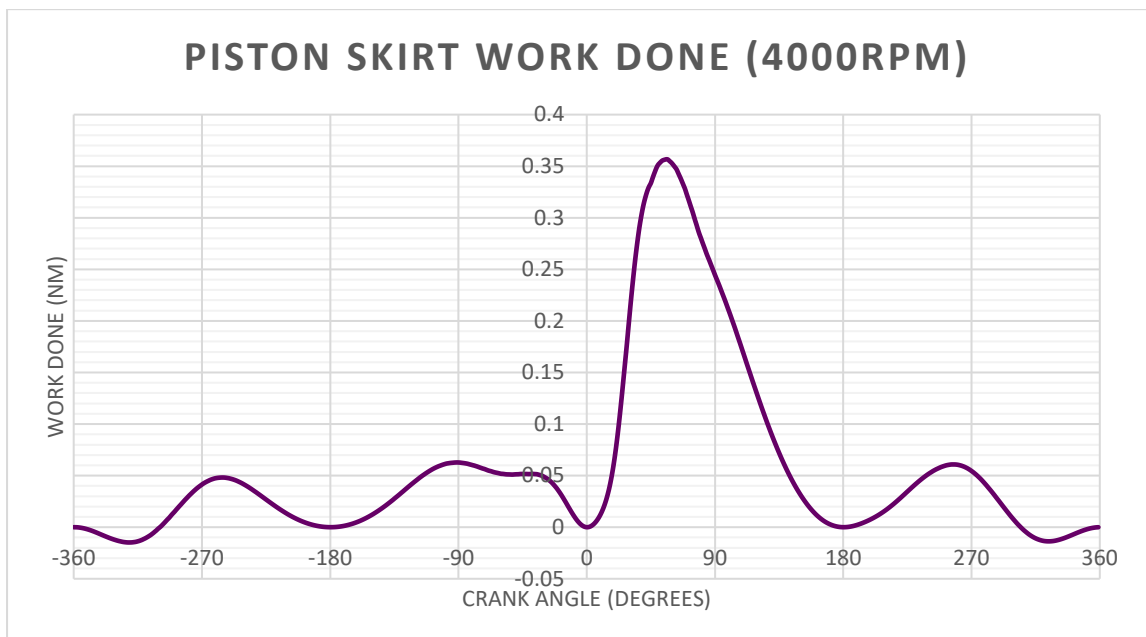


Figure 12-9: Work Done Against Combustion by the Piston Skirt at 4000RPM

The work done by the piston skirt, is taken from the work done by combustion to over-come the frictional force produced against the cylinder wall. Keeping these values small, improves the efficiency of the engine by reducing the losses from friction.

12.2.2 Piston Rings

The piston ring pack has three main functions. The rings seal the combustion chamber, control the lubrication oil flow, and transfer heat from the piston to the cylinder. (Ferguson & Kiskpatrick, 2000)

It is assumed that the piston ring has a flat outside edge with no curvature across the thickness.

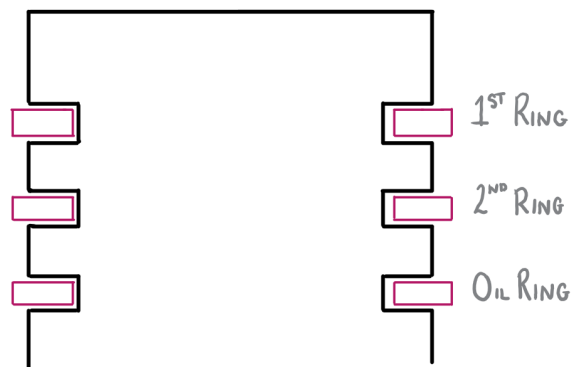


Figure 12-10: Piston rings

The top piston ring (1st Ring Figure 12-10) acts as an extension to the piston surface, creating a seal between the piston and cylinder wall while compressing the gas mixture in the cylinder. If the piston ring fails this can cause blow-by, in which combustion gas can leak in to the crank case. The middle piston ring (2nd Ring Figure 12-10) controls the pressure ratios between the piston rings, while also being a secondary protection against blow by and scraping any excess oil from the cylinder wall. The third piston ring (Oil Ring Figure 12-10) is designed to homogenously distribute oil to lubricate the cylinder wall reducing friction and absorbing heat while also scraping away any excess. All three piston rings are responsible for heat conduction in to the cylinder wall (MAHLE GmbH, 2016). The forces calculated in the model look at the top piston ring, and the effects of the cylinder pressures.

The radial force and the friction force acting on the top piston ring were calculated using the maximum pressure from the cylinder, measurements from the piston ring of the inner diameter and thickness and the friction coefficient.

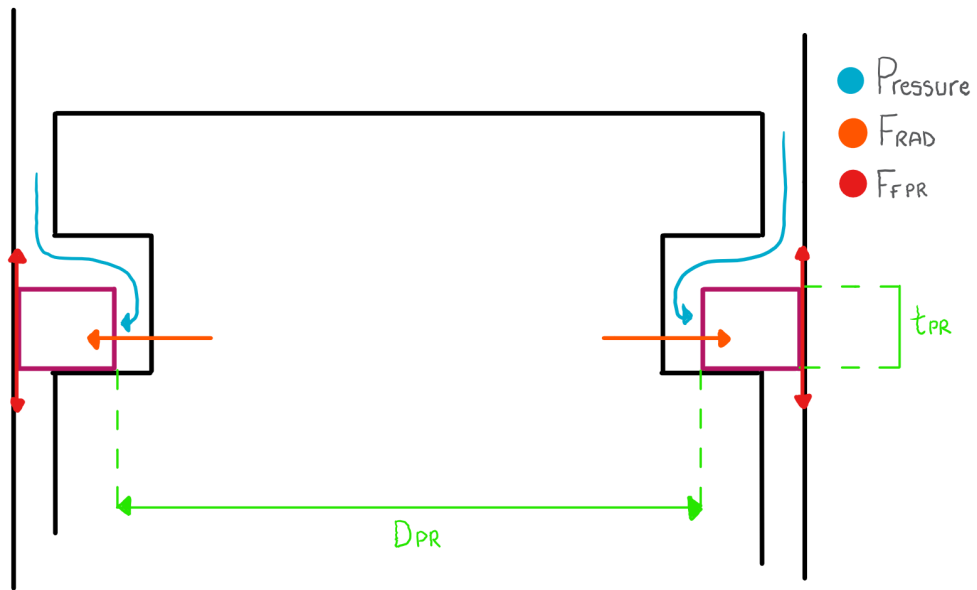


Figure 12-11: Diagram of Radial and Friction Force of the First Piston Ring

The figure above shows how the pressure created from the combustion gas pushes against the inside area of the piston ring creating a seal against the cylinder wall and, in turn creating a frictional force.

The maximum pressure across the cylinders is 2.431 Bar, from Table 13-3 the piston ring inner diameter is 0.068m and a thickness of 0.001.

$$F_{Rad} = (2.431 \times 10^5) \times \pi \times 0.068 \times 0.001$$

$$F_{Rad} = 51.93N$$

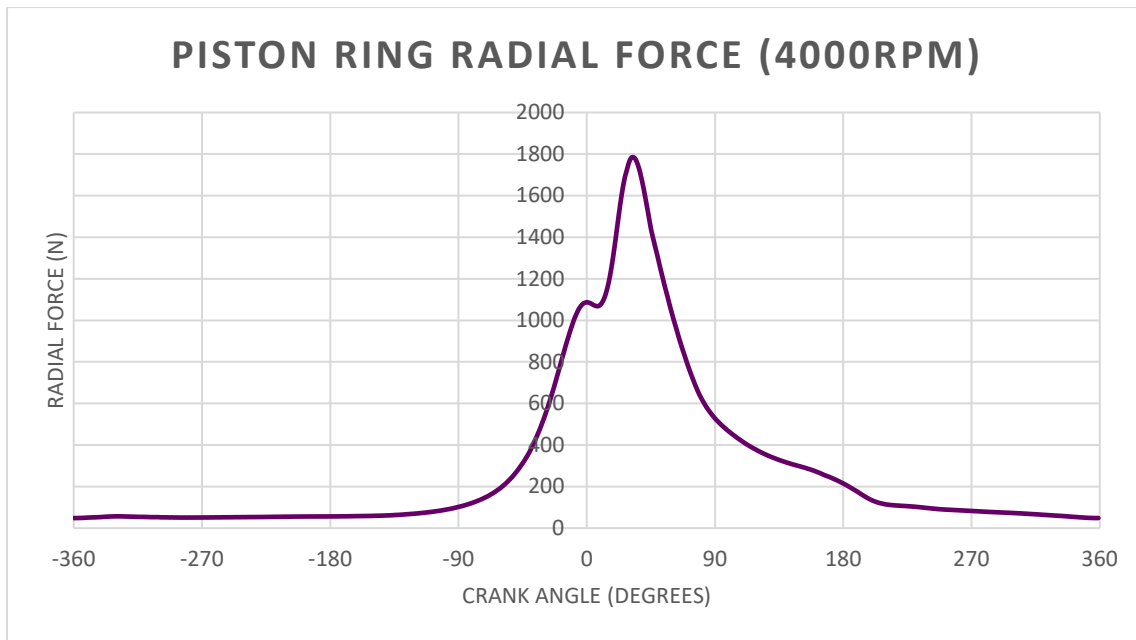


Figure 12-12: Radial Force on the Top Piston Ring at 4000RPM

The friction force (F_{FPR} , Figure 12-11) is dependent on the radial force and the seal created by the pressure against the cylinder wall. It is caused by the outer edge of the piston ring rubbing against the cylinder wall; with the motion of the piston affected by lubrication and surface roughness. These factors make up the coefficient of friction of the cylinder.

This gives the friction force of the top piston ring.

$$F_{FPR} = 51.93 \times 0.1$$

$$F_{FPR} = 5.19N$$

12.2.2.1 Measuring Piston Ring Friction - 122

The frictional forces produced by the second and oil rings can be measured physically. This is done by removing the top piston ring and compressing the next two, so the piston can be put in to the cylinder. The weight applied to the top of the piston is then increased incrementally, until the piston moves. The test was done on each of the cylinders, with no lubrication.

Table 12-6: Measured Frictional Forces

	Cylinder 1	Cylinder 2	Cylinder 3	Units
Test 1	2866	4874	4840	g
Test 2	2410	4466	4429	g
Test 3	2734	5790	3144	g
Test 4	2735	4465	3603	g
Test 5	2374	4054	4104	g
Average	2623.8	4729.8	4024	g
Average Cylinder Friction			3792.533333	g
Average Cylinder Friction			3.792533333	kg
Average Cylinder Friction			43.526316	N

$$F_F = 5.19 + 43.52$$

$$F_F = 48.72N$$

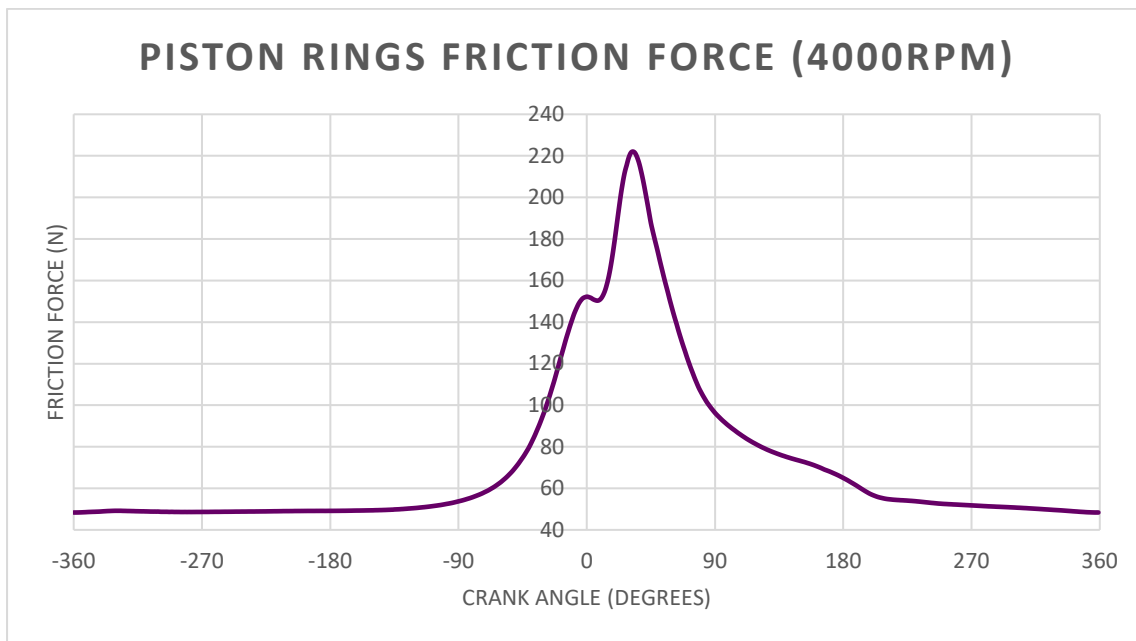


Figure 12-13: Friction Force from the Piston Rings against the Cylinder Wall at 4000RPM

The friction force produced from the piston ring, stays positive as the gas pressure is always expanding the top piston ring, therefore contact with the cylinder wall is consistent.

$$WD_{PR} = 48.72 \times (1.81 \times 10^{-3} - 1.58 \times 10^{-3})$$

$$WD_{PR} = 1.13 \times 10^{-2} Nm$$

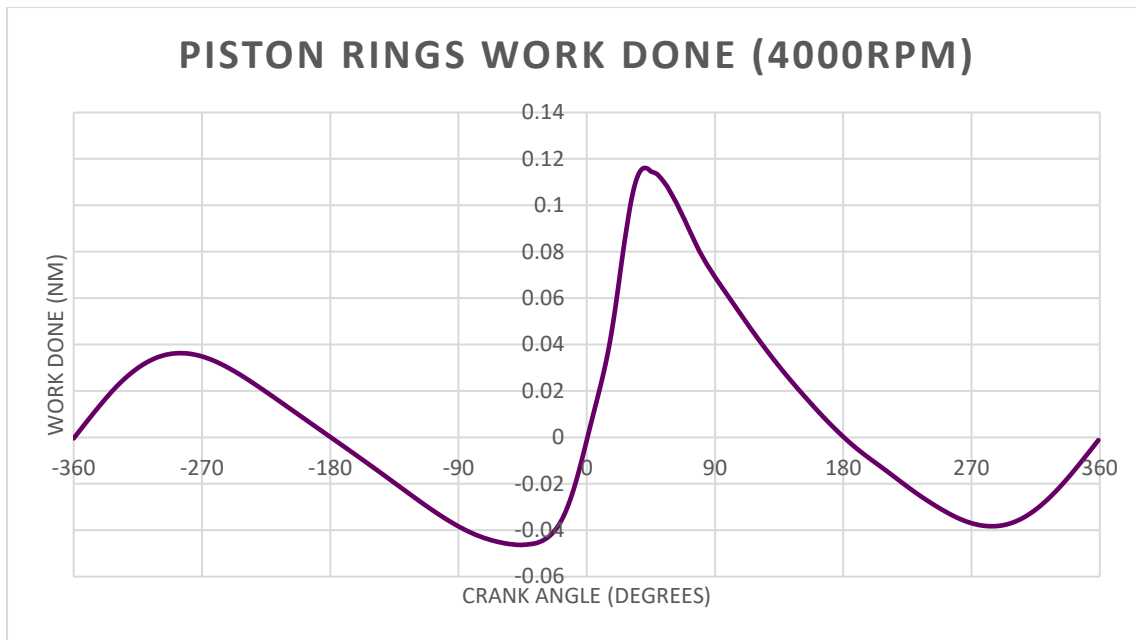


Figure 12-14: Work Done by the Piston Rings against Combustion at 4000RPM

The work done by the piston rings is taken from the work done by combustion to overcome the frictional force produced by the contact with the outside edges of the piston rings and the cylinder wall.

The FMEP for the piston skirt at 4000RPM,

$$FMEP = \frac{42.99 \times 3}{0.00995 \times 10^5}$$

$$FMEP = \frac{128.97}{99.5}$$

$$FMEP = 1.296 \text{ Bar}$$

Table 12-7: Work Done by the Piston Skirt at Each RPM

	Total Work Done (Piston Skirt)	FMEP (Piston Skirt)
	Joules	Bar
1000	61.35	0.6166
1250	78.43	0.7883
1500	94.26	0.9473
2000	114.77	1.1534
2500	123.04	1.2366
3000	127.01	1.2765
3500	129.16	1.2981

4000	128.97	1.2962
4500	125.49	1.2612
5000	132.36	1.3302
5500	134.40	1.3508
6000	134.21	1.3488
6500	129.38	1.3003

The FMEP of the piston rings

$$FMEP = \frac{23.56 \times 3}{0.00995 \times 10^5}$$

$$FMEP = \frac{70.68}{99.5}$$

$$FMEP = 0.71 \text{ Bar}$$

Table 12-8: Work Done by the Piston Rings at Each RPM

	Total Work Done (Piston Rings)	FMEP (Piston Rings)
	Joules	Bar
1000	57.40	0.5769
1250	61.06	0.6137
1500	64.98	0.6531
2000	69.37	0.6972
2500	70.89	0.7124
3000	71.45	0.7181
3500	71.55	0.7191
4000	70.68	0.7103
4500	69.00	0.6935
5000	69.21	0.6956
5500	68.50	0.6884
6000	67.01	0.6735
6500	63.99	0.6431

12.3 Analysis of Model

The gas and inertia forces increase with RPM. This is due to the initial pressure in the cylinder being greater at higher RPMs. The increase in RPM sees an increase in the angular acceleration of the engine, which effects the first and second order acceleration used to find the inertia force. These increases in force put the connecting rod under higher fluctuations of tensile and compressive stress.

The side force acting on the piston skirt increases with RPM (see 16.4.6 Side Force). It can also be seen that the force changes the side that it is acting on with a greater force, this has the potential to cause damage to the piston. This is due to the increased gas force acting on the surface of the piston.

In the model made by the author the friction coefficient was kept constant at 0.1. This is inaccurate, as the coefficient of friction can be dependent on RPM, oil lubrication and contact force. This coefficient when compared to real data is too high in a representation at lower RPMs. This would mean the frictional forces found at the piston skirt and rings are over-predicted at lower RPMs.

Table 12-9: The Combined FMEP of the Piston Rings and Skirt

RPM	FMEP (total)
	Bar
1000	1.193515747
1250	1.401962599
1500	1.600378741
2000	1.850663461
2500	1.94902003
3000	1.994631908
3500	2.017226822
4000	2.006504671
4500	1.954731865
5000	2.025838846
5500	2.039206728
6000	2.022269346
6500	1.943469756

The friction between the piston and cylinder wall accounts for 35% of mechanical losses (SynLube, 2016). From the values for FMEP above, the total mechanical losses of the engine would be,

Table 12-10: Total Engine FMEP

<i>RPM</i>	<i>Total Engine FMEP</i>
	Bar
1000	3.410044992
1250	4.005607425
1500	4.572510688
2000	5.287609888
2500	5.568628658
3000	5.698948309
3500	5.763505206
4000	5.732870489
4500	5.584948186
5000	5.788110988
5500	5.826304938
6000	5.777912417
6500	5.552770731

As stated previously frictional losses in the engine account for between 15% and 20% of losses in the engine, the author's model predicts the losses to be;

Table 12-11: Percentage of Mechanical Loss from Frictional Forces

<i>RPM</i>	<i>% Mechanical Losses</i>
1000	12.26149413
1250	11.3071165
1500	10.81918712
2000	10.44172813
2500	10.40322577
3000	10.4320061
3500	10.50710731
4000	10.77248234
4500	11.15387854
5000	11.46450647
5500	11.84729064
6000	12.45457066
6500	13.63785421

The percentage of mechanical losses for each of the RPM range from 10% to 13%, from this it can be seen that, despite the short comings of the model such as that of the coefficient of friction, it can still be used to predict the friction losses, with relative accuracy.

13 Conclusions

Looking first at the drive cycles, the models developed by the author show that the NEDC test, benefits smaller engines as predicted. The smaller engine capacity allows for a higher BMEP to be achieved, improving the fuel efficiency across the test. This is seen in the model as a decrease of 27% in fuel used per kilometre from the 1984cc VW Passat and the 995cc Ford Fox.

The models also show the WLTP to be a fairer test when it comes to measuring vehicle performance. This is due to the gear shift position being open. This change to testing procedures will aid in the accuracy of emissions and fuel consumption figures. The model made by the author provides a close representation of the comparative values of fuel consumption over both tests.

The combustion and friction models made by the author, concur with the mechanical losses found in actual efficiency testing. With the introduction of the WLTP, the combustion process needs to become 'cleaner', reducing the frictional forces and allowing for more efficient combustion, with less fuel used for the same power produced.

Overall the models created are representative of both the drive cycles and the engine process that is occurring during these test procedures.

14 Further Work

For the drive cycles, within the WLTP, the model could be further developed to include the final piece of legislation, affecting the gear changes. With this addition the model could be used to model the optimum gear shift position, with the aim of minimising fuel consumption.

Both drive cycle models (NEDC and WLTP) could be further developed to calculate battery power and energy recovery, this would enable the model to also be used for hybrid and electric vehicles.

The friction model could be developed to include modelling of the friction between; the connecting rod bearings and crankshaft pin and piston pin, the main bearings and crank shaft and the valve train friction. This extra modelling would accurately predict the friction losses experienced in the engine.

15References

- ACEA, 2017. *WLTP FACTS*. [Online]
Available at: <http://wltpfacts.eu/from-nedc-to-wltp-change/>
[Accessed 12 April 2018].
- Barlow, T. J., Latham, S., McCrae, S. & Boulter, P. G., 2009. *A Reference Book of Driving Cycles for Use in the Measurement of Road Vehicle Emissions*, Wokingham: TRL.
- Bosch, 2014. *Automotive Handbook*. 9th ed. Postfach: Robert Bosch GmbH.
- European Commission, 2017. *WLTP Gear Shifts Calculator*. s.l.:European Commission.
- Fenger, J., Hertel, O. & Palmgren, F., 2013. *Urban Air Pollution - European Aspects*. 1st ed. Berlin: Springer Science & Business Media.
- Ferguson, C. R. & Kiskpatrick, A. T., 2000. *Internal Combustion Engines: Applied Thermosciences*. 2nd ed. New York: John Wiley & Sons.
- Ford Motor Company, 2012. *Ford Focus 1.0-Litre Ecoboost Specification*. Michigan: Ford Motor Company.
- Foster, P., 1997. Air Pollution 'exceeds limits every five days'. *The Times*, 26 September, p. 4.
- Gore, M. et al., 2014. Direct Measurement of Piston Friction of Internal Combustion Engines Using the Floating-Liner Principle. *Journal of Automobile Engineering*, 228(3), pp. 344-354.
- Heywood, J. B., 1988. *Internal Combustion Engine Fundamentals*. 1st ed. New York: McGraw-Hill.
- Holder, J., 2018. Deadline Looms for Emissions Shake-up. *AutoCar*, 7th February, p. 18 & 19.
- ICCT, 2017. *Real-Driving Emissions Test Procedure for Exhaust Gas Pollutant Emissions of Cars and Light Commercial Vehicles in Europe*, Brussels: ICCT.
- Kovarik, B., 2017. *Air Pollution*. [Online]
Available at: <http://environmentalhistory.org/about/airpollution/>
[Accessed April 2018].
- Lea, R., 2017. Emissions Test To Mimic Real World Driving. *The Times*, 1 September, p. 46.
- Lette van Oostvoore, N. & Ingram, A., 2017. *What is mpg? How is it Calculated*. [Online]
Available at: <https://www.carwow.co.uk/guides/running/What-is-MPG-0255>
[Accessed April 2018].

Lloyd, C., 2018. *Fuel Economy Test Changes: What You Need To Know About WLTP*. [Online] Available at: <https://www.parkers.co.uk/car-advice/fuel-economy-test-changes-what-you-need-to-know-about-wltp/>

[Accessed April 2018].

MAHLE GmbH, 2016. *Pistons and Engine Testing*. 2nd ed. Stuttgart: Springer Vieweg.

Mehmet Uras, H. & Patterson, D. J., 1985. *Oil and Ring Effects on Piston-Ring Assembly Friction by the Instantaneous IMEP Method*, Detroit: SAE.

Mehmet Uras, H. & Patterson, D. J., 1983. *Measurement of Piston and Ring Assembly Friction Instantaneous IMEP Method*, Detroit: SAE.

Nagar, P. & Miers, S., 2011. *Friction between Piston and Cylinder of an IC Engine: a Review*, Michigan: SAE International .

Ngo, V. D., Hofman, T., Steinbuch, M. & Serrarens, A., 2013. Gear Shift Map Design Methodology for Automotive Transmissions. *Journal of Automobile Engineering*, 228(1), pp. 50-72.

Nielsen, A., 2015. UK, France and Germany lobbied for flawed car emission test, documents reveal - Exclusive Countries publicly calling for investigations into VW emissions digging scandal have privately fought to keep loopholes in car tests for carbon emissions. *The Guardian*, 24 September, p. 4.

Patel, P., Mourelatos, Z. P. & Shah, P., 2007. *A Comprehensive Method for Piston Secondary Dynamics and Piston-Bore Contact*, Detroit: SAE.

Patterson, D. J. & Mehmet Uras, H., 1987. *Effect of Some Piston Variables on Piston and Ring Assembly Friction*, Detroit: SAE.

Patton, K. J., Nitschke, R. G. & Heywood, J. B., 1989. *Development and Evaluation of a Friction Model for Spark Ignition Engines*, Massachusetts: SAE.

Rao, J. S. & Dukkupati, R. V., 1992. *Mechanism and Machine Theory*. 2nd ed. New Delhi: New Age International.

Rezeka, S. F. & Henein, N. A., 1984. *a New Approach to Evaluate Instantaneous Friction and its Components in Internal Combustion Engines*, Detroit: SAE .

Singh, T., Balzer, D., Leertouwer, K. & Wise, J., n.d. WLTP Regulations and Advanced Recuperation Strategies. *ATZ worldwide* , pp. 50-53.

SNJ , 2017. Motor News: Keeping Emissions Under Control. *Stroud News and Journal*, 10 Septemeber.

Stanley, R., Taraza, D. & Henein, N., 1999. *A Simplified Friction Model of the Piston Ring Assembly*, Detroit: SAE International.

SynLube, 2016. *Reduced Friction and Wear*. [Online] Available at: [http://www.synlube.com/SynLube Magic2.htm](http://www.synlube.com/SynLube_Magic2.htm) [Accessed April 2018].

Taylor, C. F., 1985. *The Internal Combustion Engine in Theory and Practice*. 2 ed. Cambridge: Massachusetts Institue of Technology.

Tovey, A. & Huggler, J., 2015. UK car tests 'inadequate in wake of VW scandal. *The Daily Telegrapg*, 17 November, p. 1.

Tutuianu, M. et al., 2013. *Development of a World-wide Harmonized Light Duty Driving Test Cycle (WLTC)*, s.l.: UN.

Wakuri, Y. et al., 1995. *Studies on Friction Characteristics of Reciprocating Engines*, Toronto: SAE.

Webster, B., 2014. Carmaker Cheating in fuel tests'. *The Times*, 7 November, p. 16.

Zubaydi, A. et al., n.d. *Analysis of Dynamic Inertia Forces at Main Bearing of Ship Reciprocating Engines*, Surabaya: s.n.

16 Appendix

16.1 Appendix A – Conversions

Degrees to radians

$$\text{Degrees} \times \frac{\pi}{180} = \text{Radians}$$

Equation 16-1: Conversion Degrees to Radians

RPM to radians per second

$$\text{RPM} \times \frac{2\pi}{60} = \text{Rads/s}$$

Equation 16-2: Conversion RPM to Radians per Second

Table 16-1: RPM to Rads/s Conversion Chart

<i>RPM</i>	<i>Radians/s</i>
1000	104.7197551
1250	130.8996939
1500	157.0796327
2000	209.4395102
2500	261.7993878
3000	314.1592654
3500	366.5191429
4000	418.8790205
4500	471.238898
5000	523.5987756
5500	575.9586532
6000	628.3185307
6500	680.6784083

Km/h to m/s

$$\text{Km/h} \times \frac{1000}{3600} = \text{m/s}$$

Equation 16-3: Conversion Kilometres per Hour to Metres per Second

16.2 Appendix B- Drive Cycle Model

16.2.1 Fuel

16.2.1.1 VW Passat

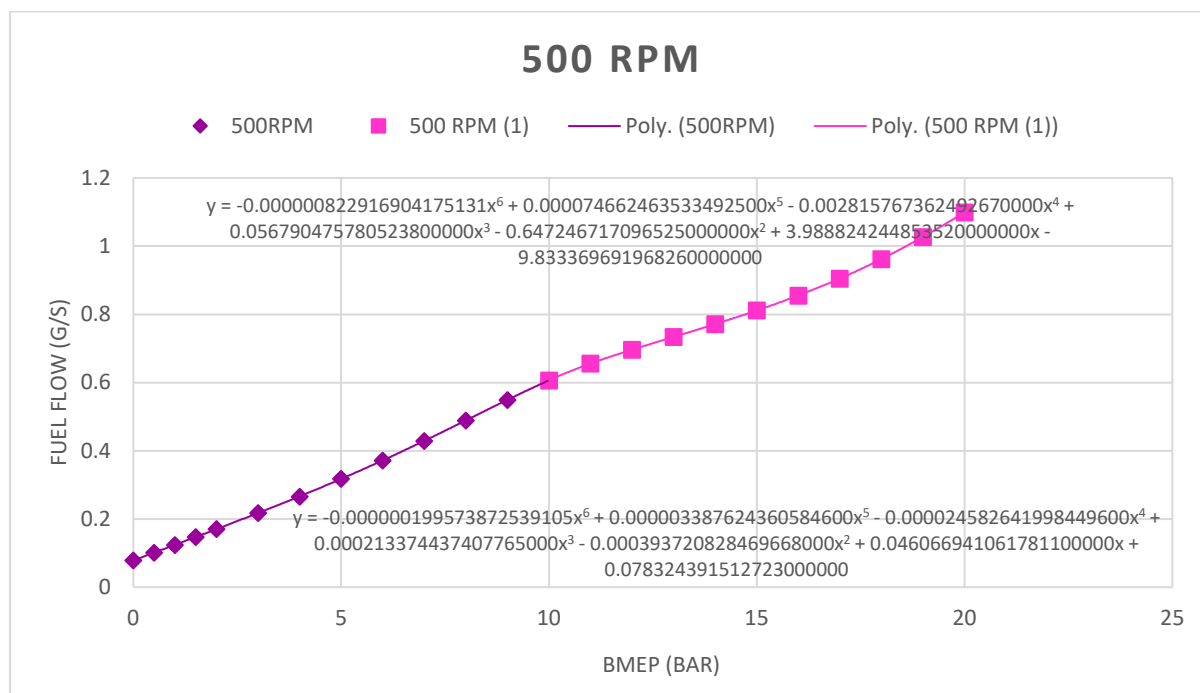


Figure 16-1: Fuel Flow Rate for VW Passat at 500RPM

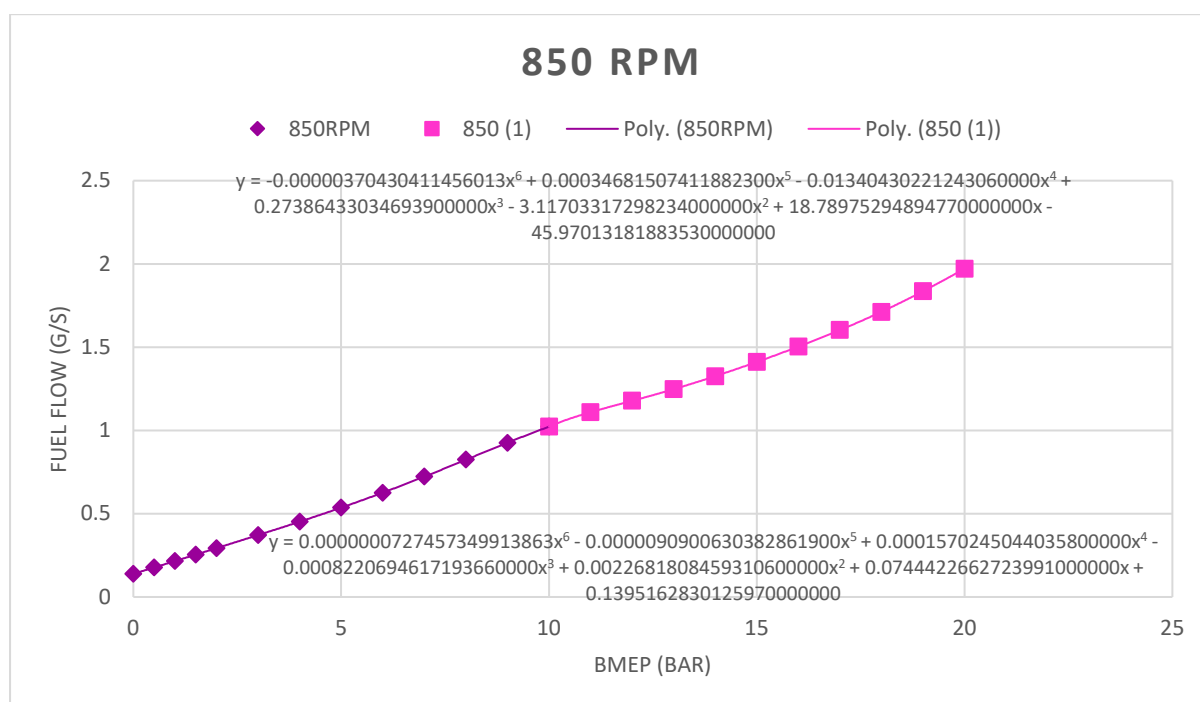


Figure 16-2: Fuel Flow Rate for VW Passat at 850 RPM

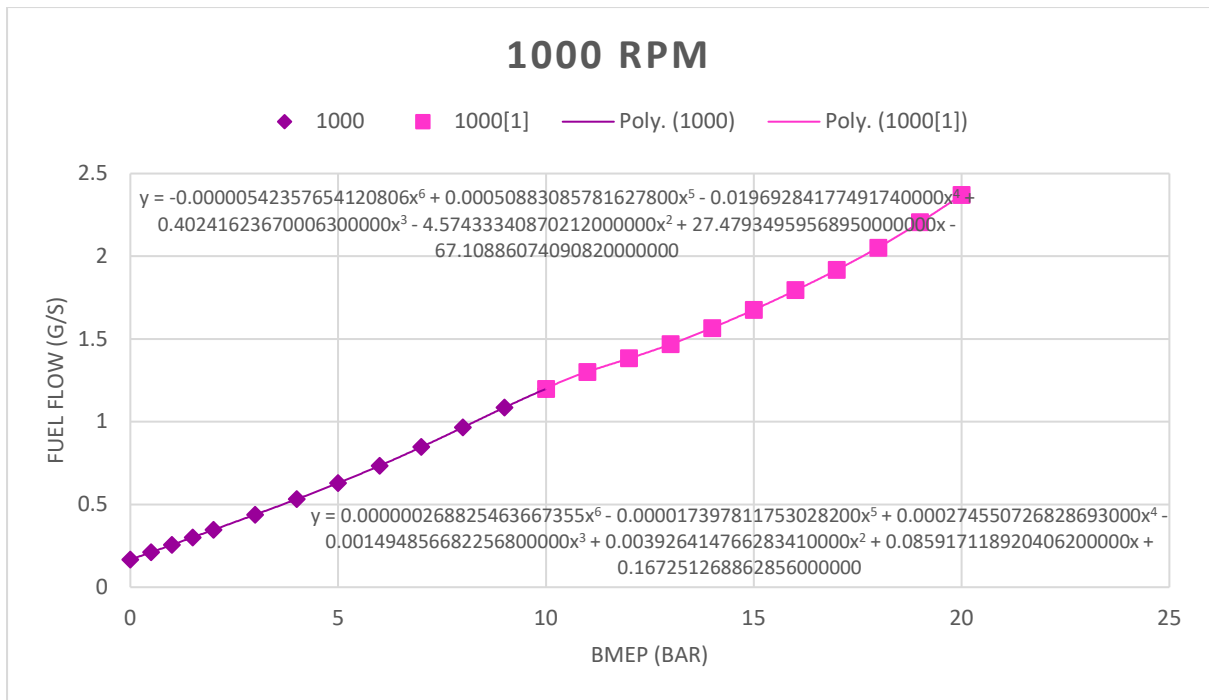


Figure 16-3: Fuel Flow Rate for VW Passat at 1000 RPM

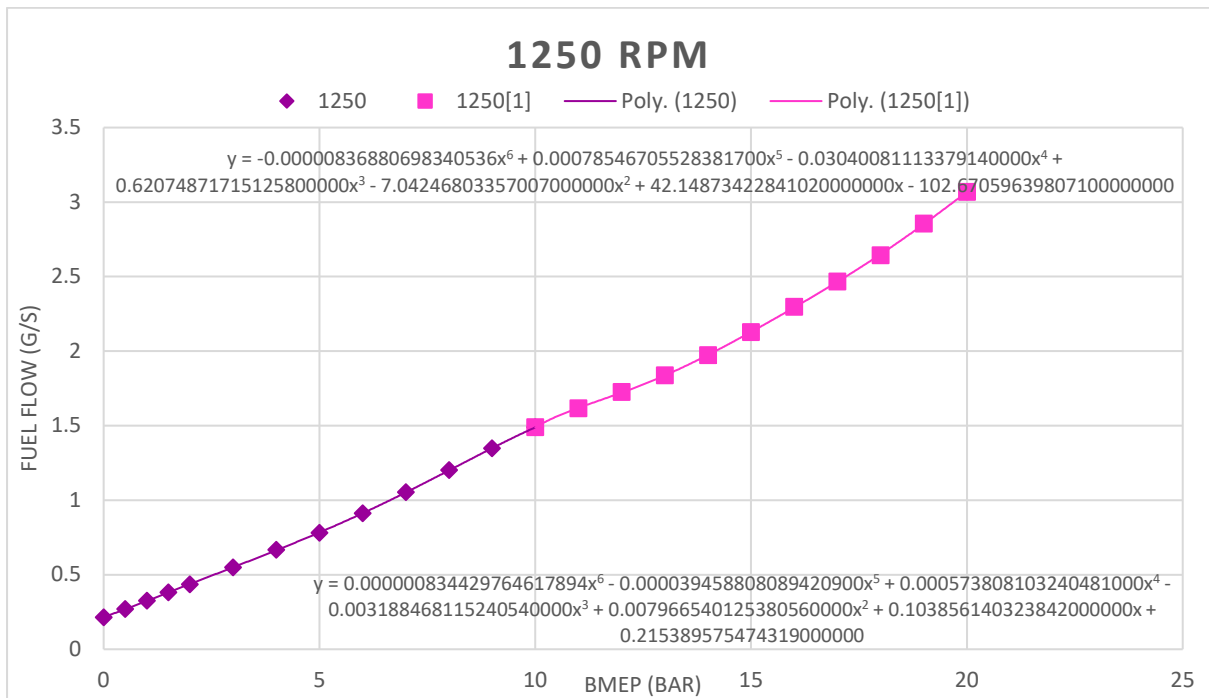


Figure 16-4: Fuel Flow Rate for VW Passat at 1250 RPM

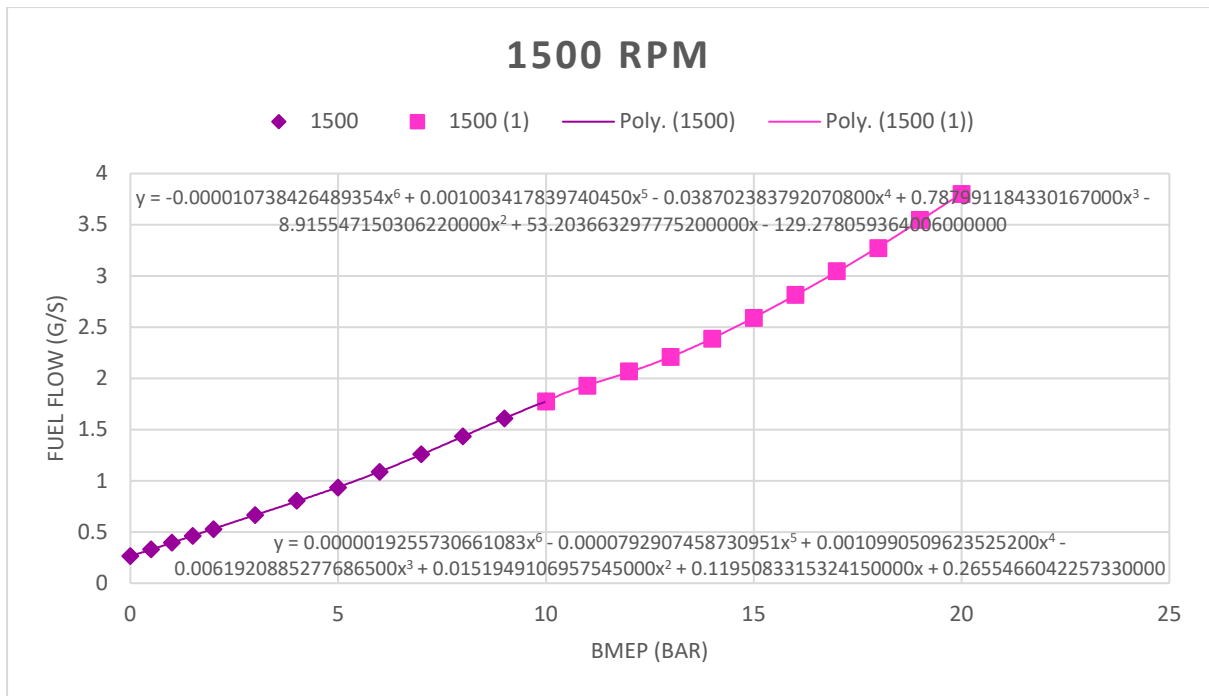


Figure 16-5: Fuel Flow Rate for VW Passat at 1500 RPM

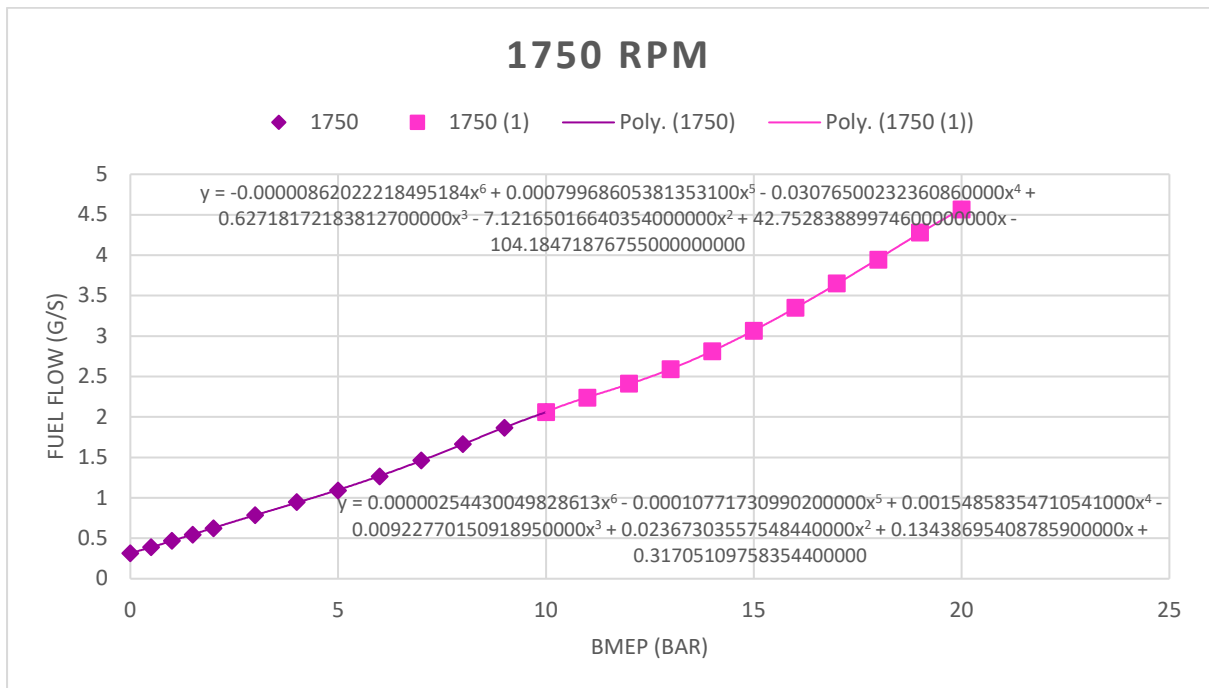


Figure 16-6: Fuel Flow Rate for VW Passat at 1750 RPM

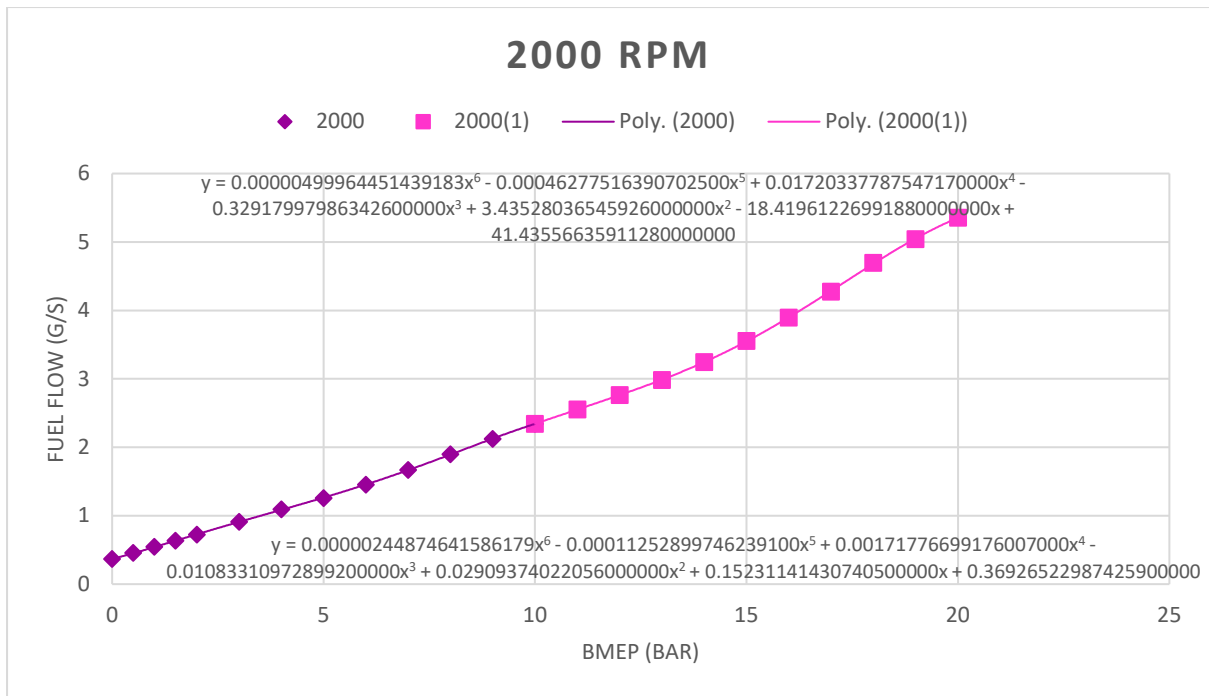


Figure 16-7: Fuel Flow Rate for VW Passat at 2000 RPM

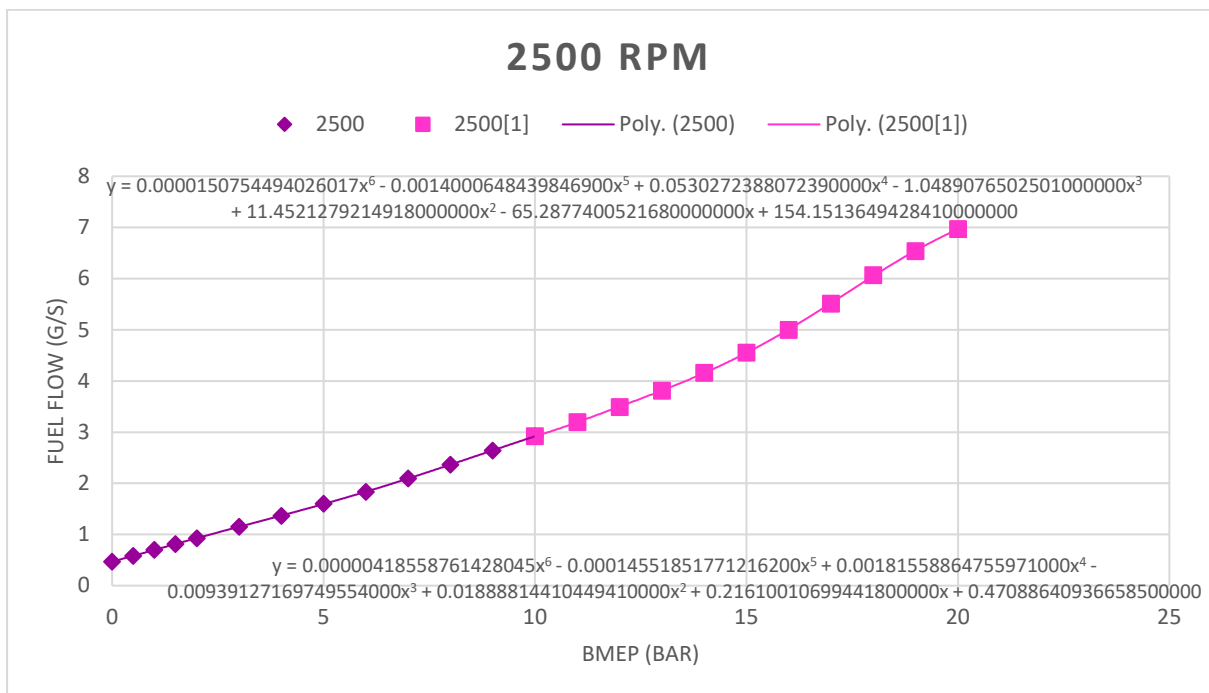


Figure 16-8: Fuel Flow Rate for VW Passat at 2500 RPM

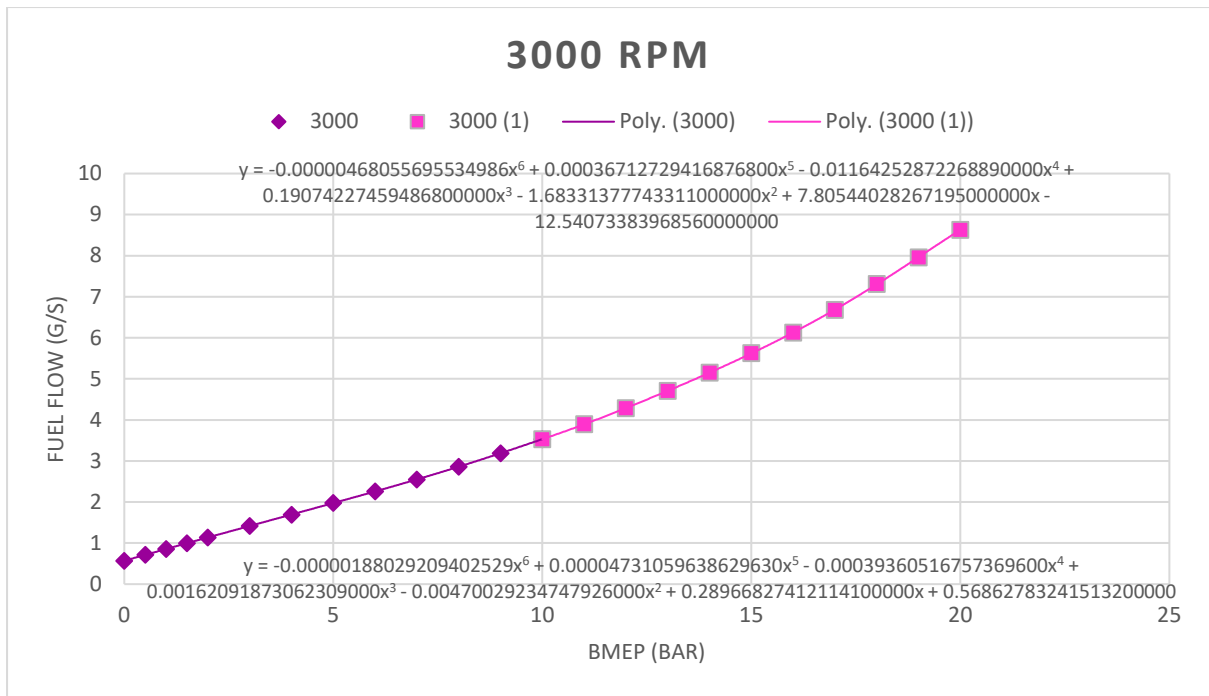


Figure 16-9: Fuel Flow Rate for VW Passat at 3000 RPM

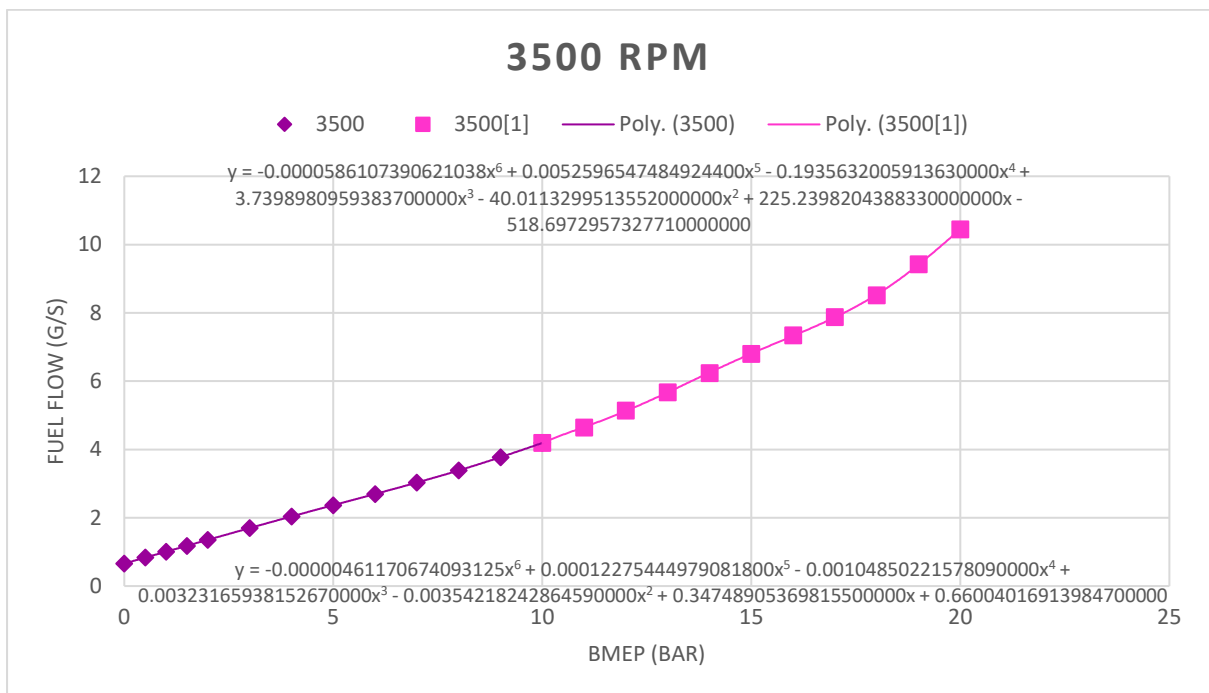


Figure 16-10: Fuel Flow Rate for VW Passat at 3500 RPM

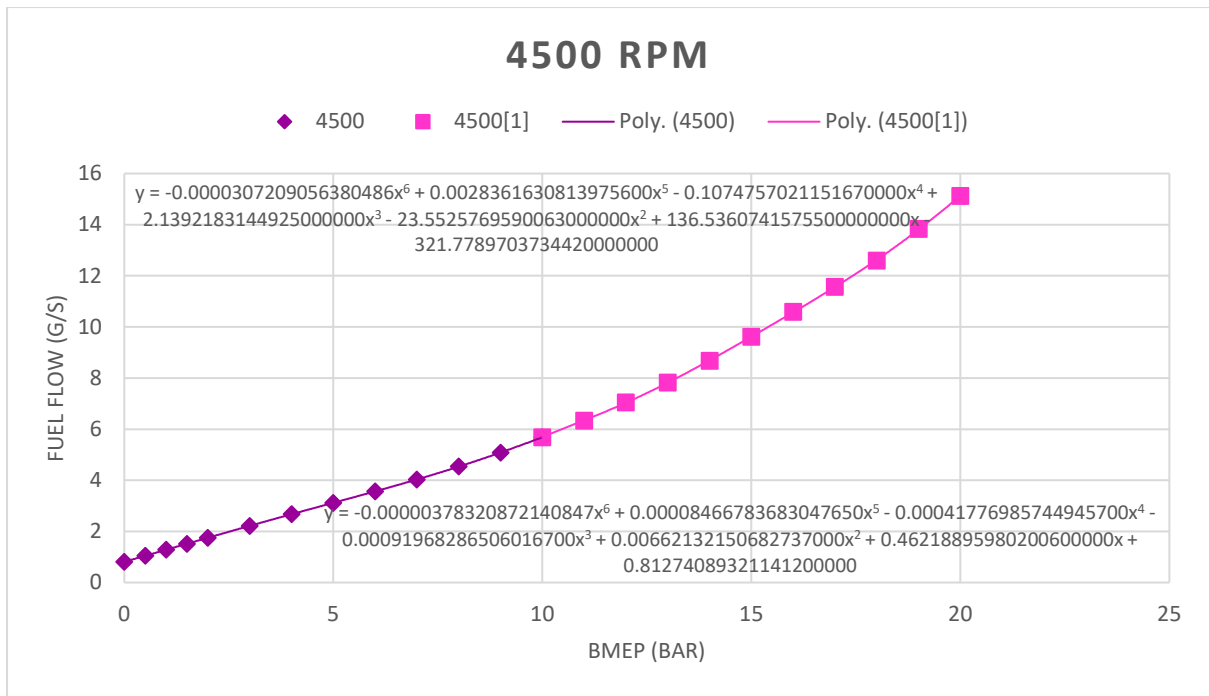


Figure 16-11: Fuel Flow Rate for VW Passat at 4500 RPM

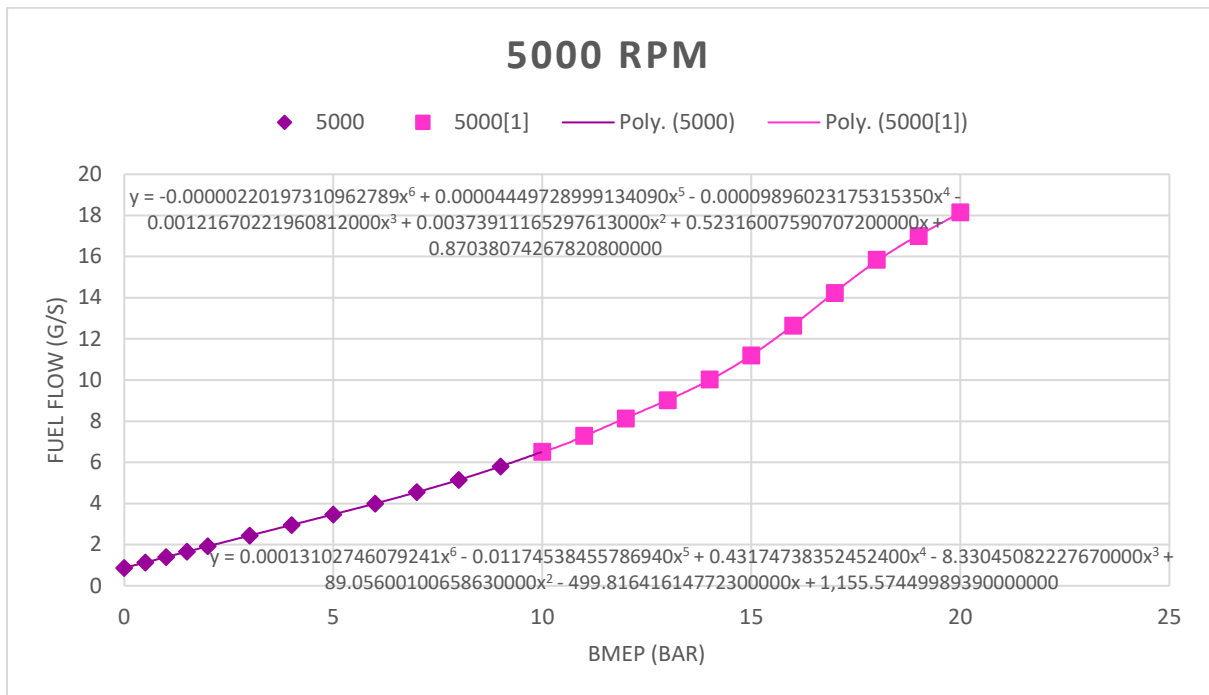


Figure 16-12: Fuel Flow Rate for VW Passat at 5000 RPM

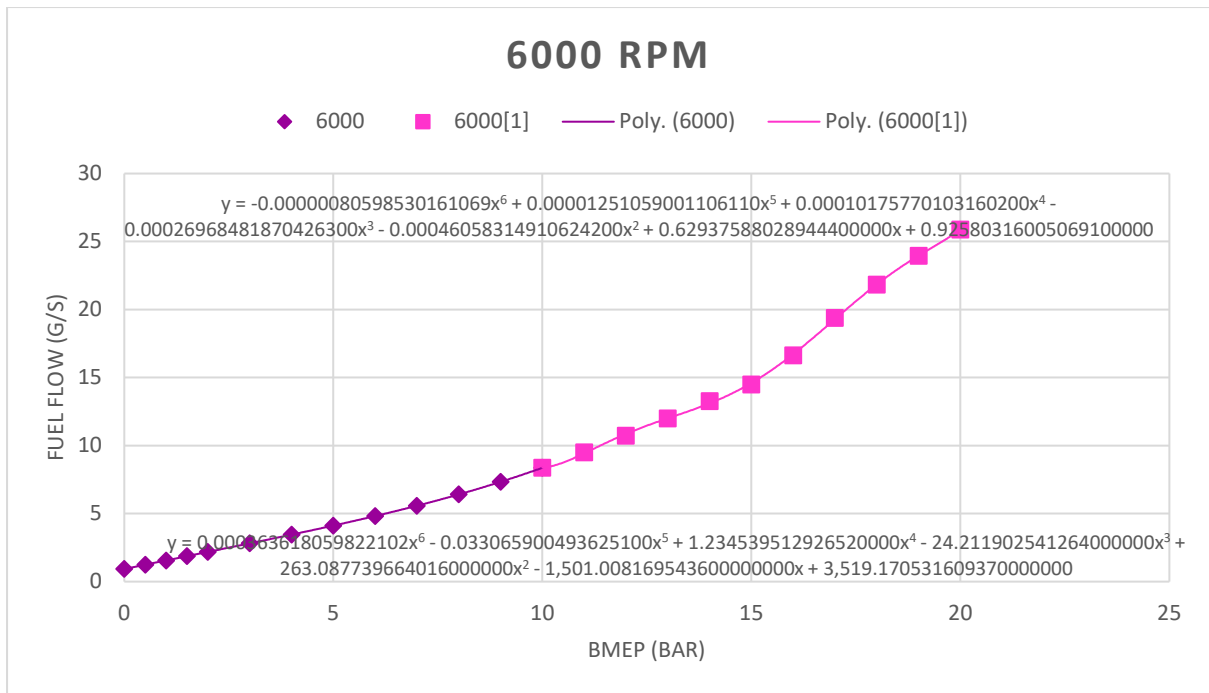


Figure 16-13: Fuel Flow Rate for VW Passat at 6000 RPM

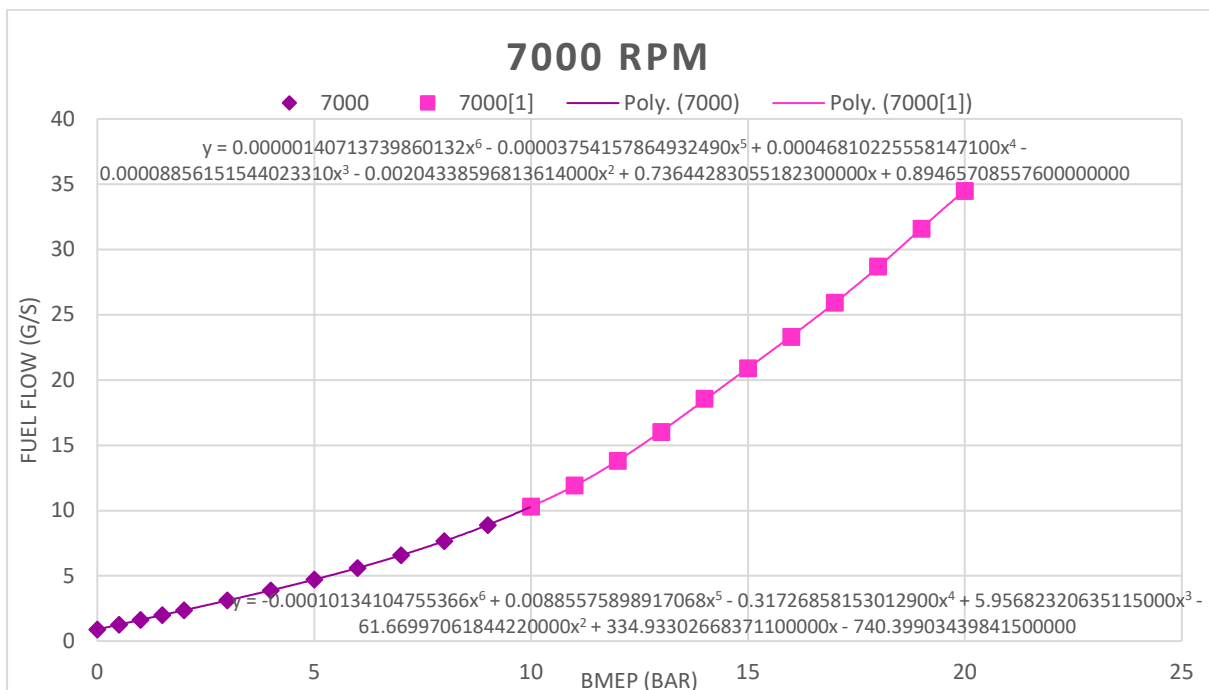


Figure 16-14: Fuel Flow Rate for VW Passat at 7000 RPM

16.2.1.2 Ford Fox

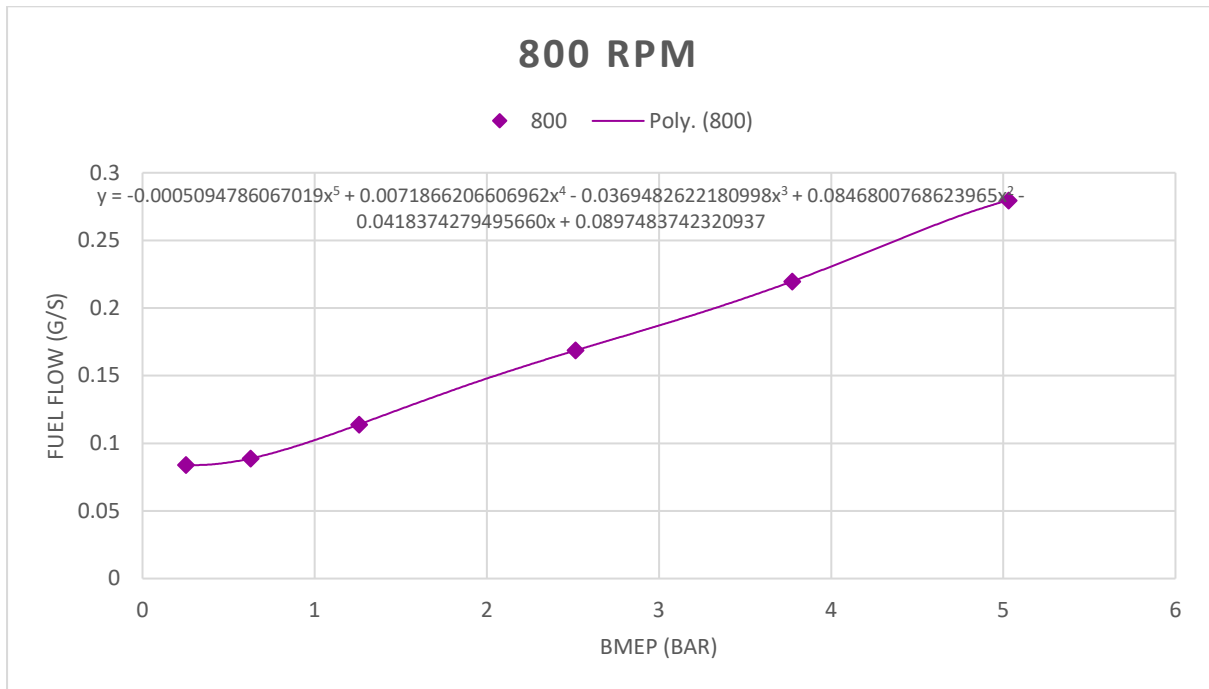


Figure 16-15: Fuel Flow Rate for Ford Fox Engine at 800RPM

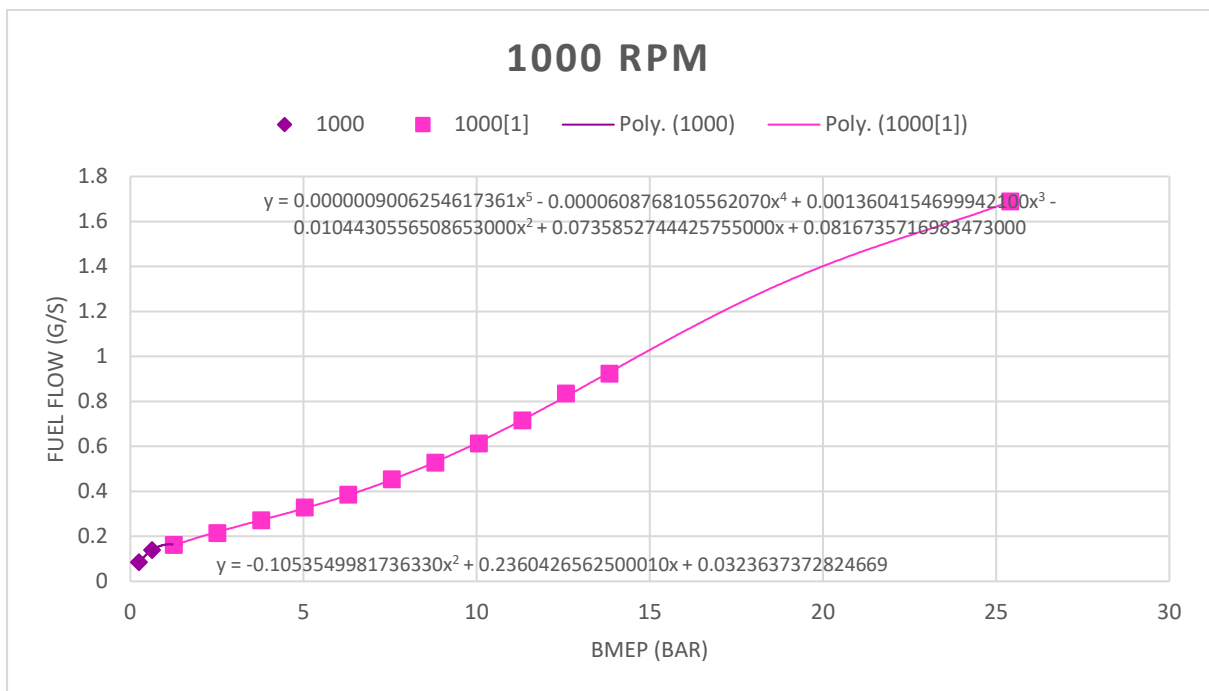


Figure 16-16: Fuel Flow Rate for Ford Fox Engine at 1000RPM

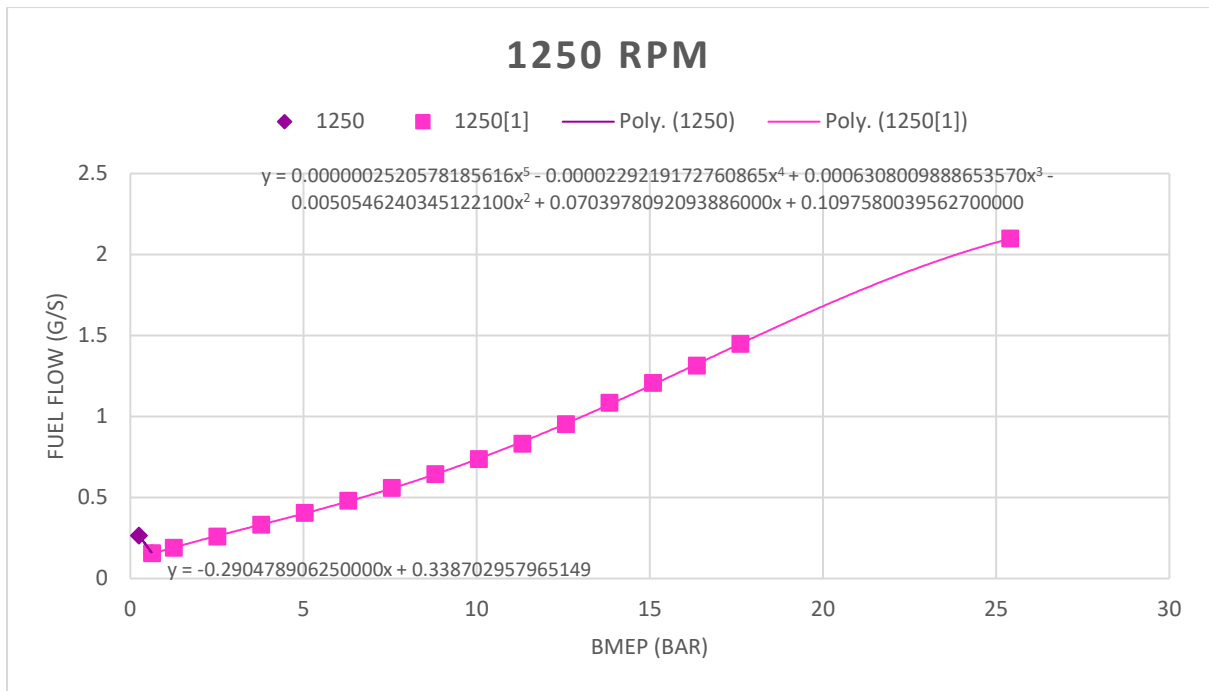


Figure 16-17: Fuel Flow Rate for Ford Fox Engine at 1250RPM

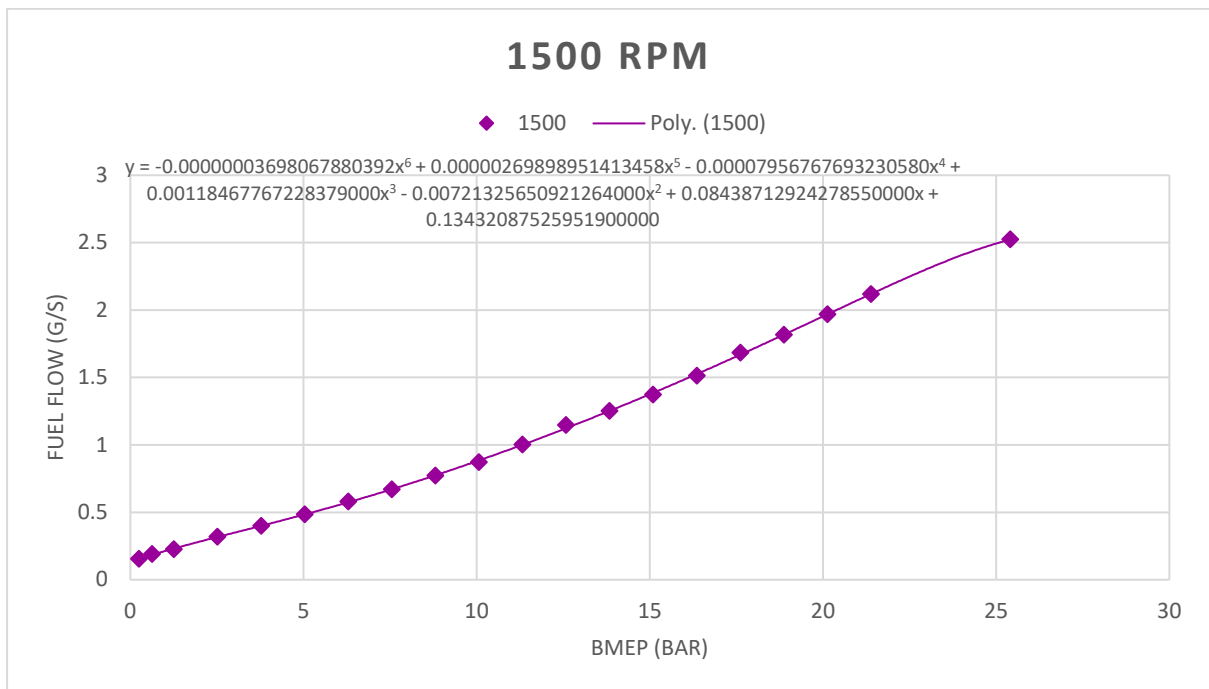


Figure 16-18: Fuel Flow Rate for Ford Fox Engine at 1500RPM

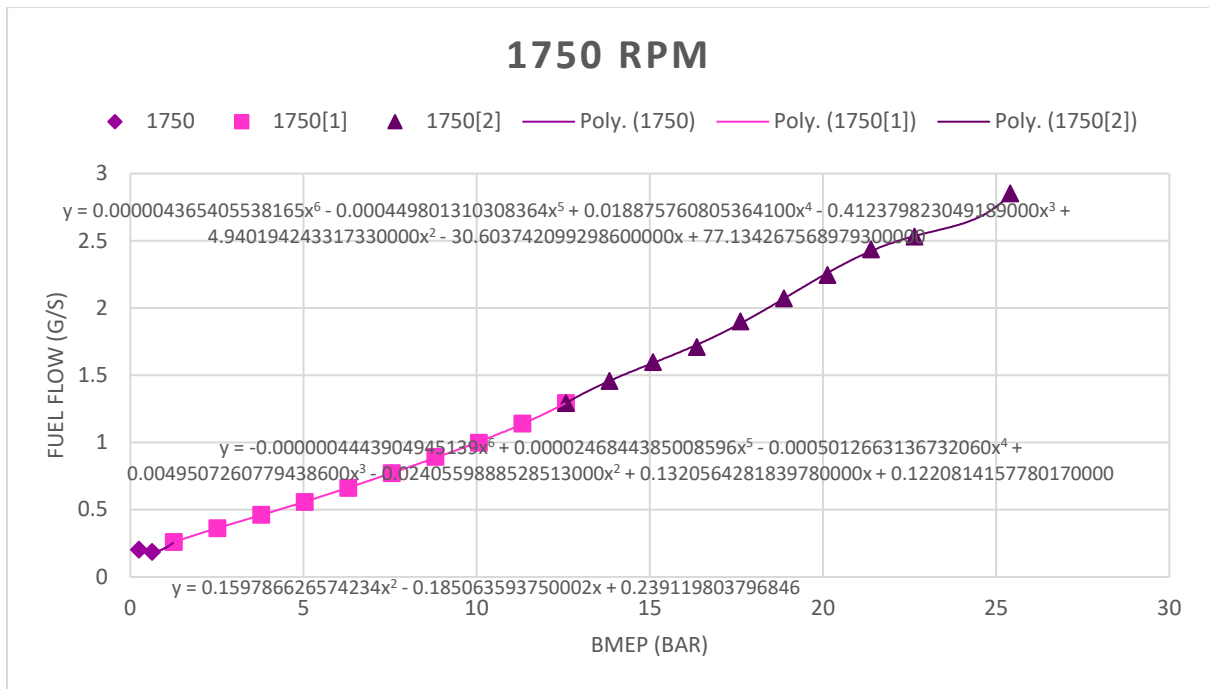


Figure 16-19: Fuel Flow Rate for Ford Fox Engine at 1750RPM

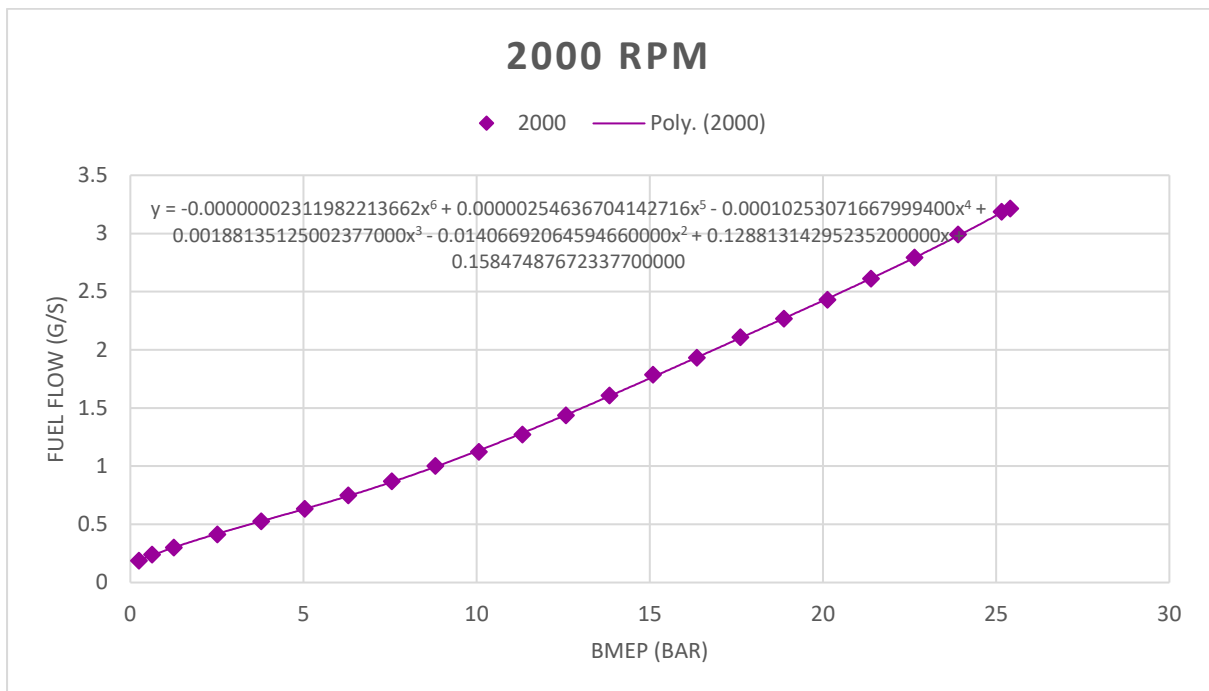


Figure 16-20: Fuel Flow Rate for Ford Fox Engine at 2000RPM

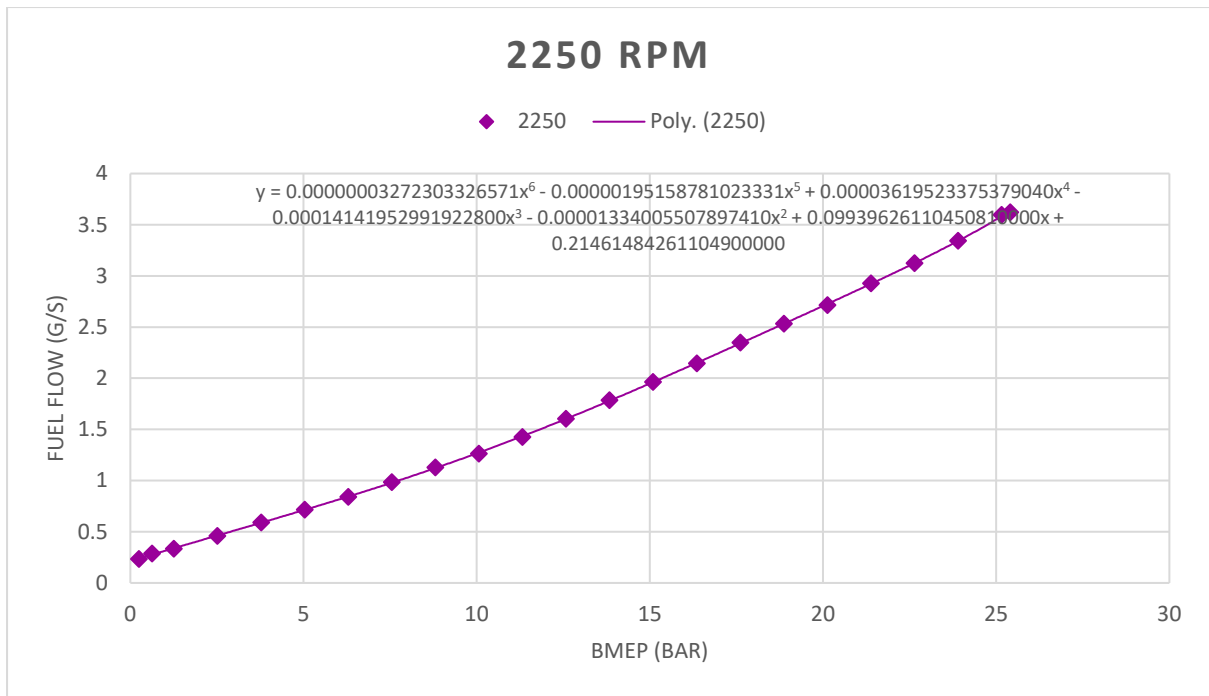


Figure 16-21: Fuel Flow Rate for Ford Fox Engine at 2250RPM

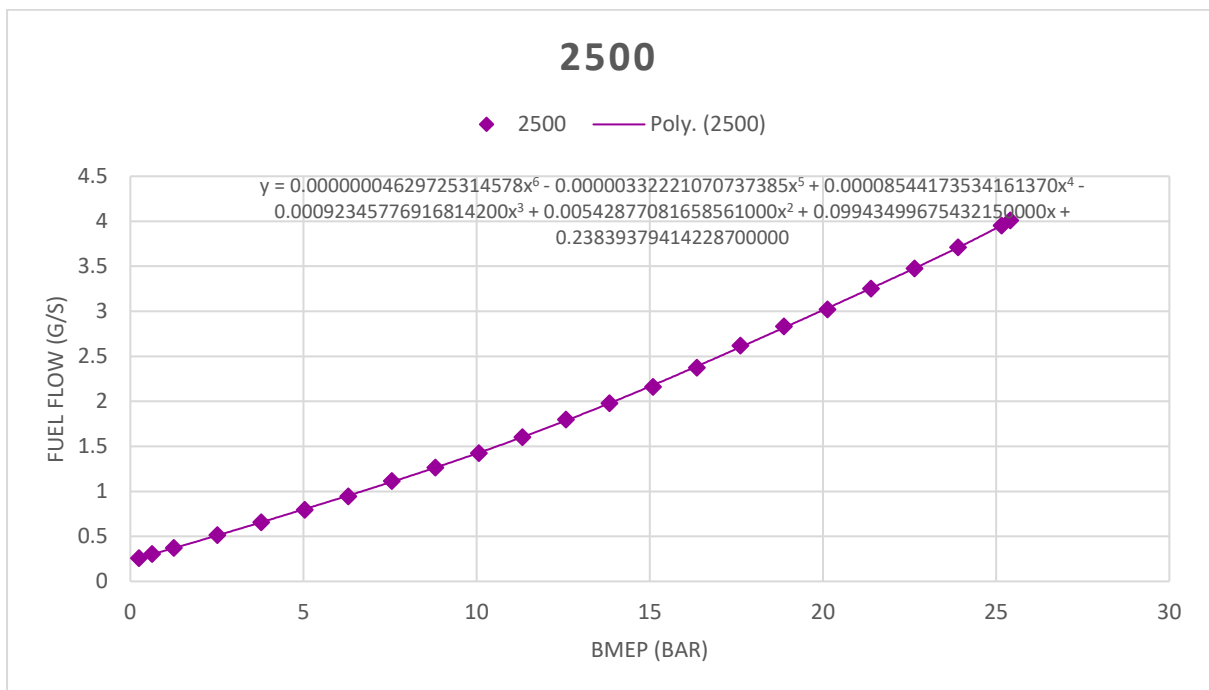


Figure 16-22: Fuel Flow Rate for Ford Fox Engine at 2500RPM

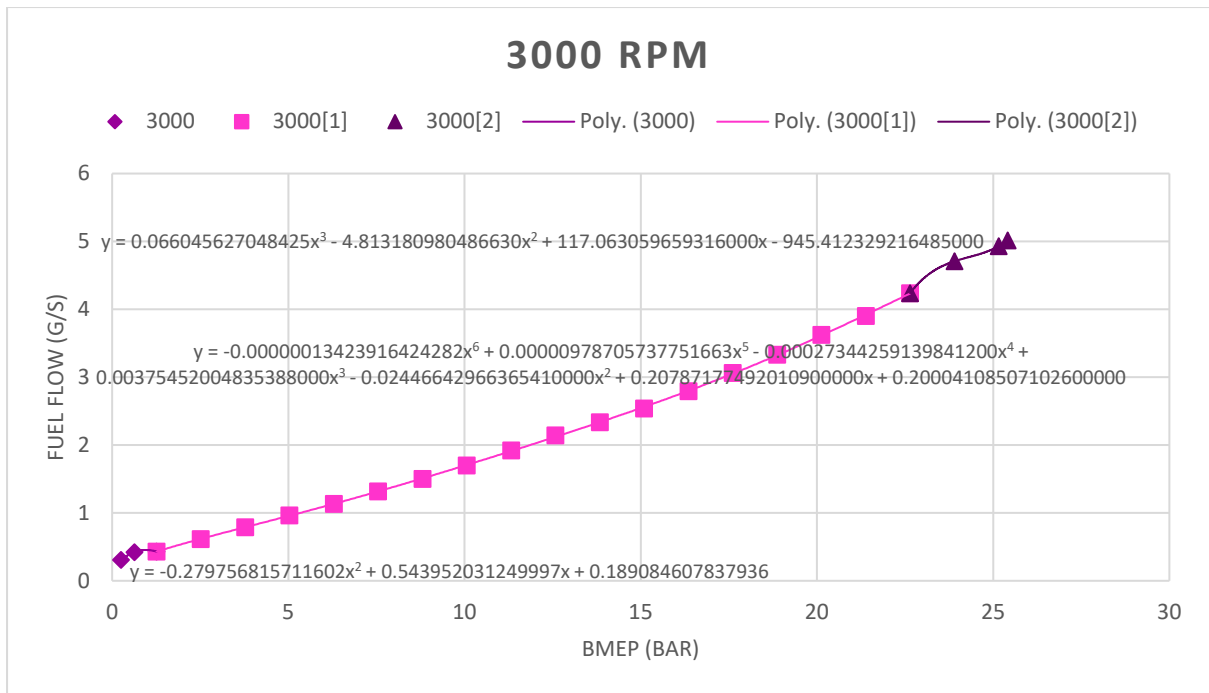


Figure 16-23: Fuel Flow Rate for Ford Fox Engine at 3000RPM

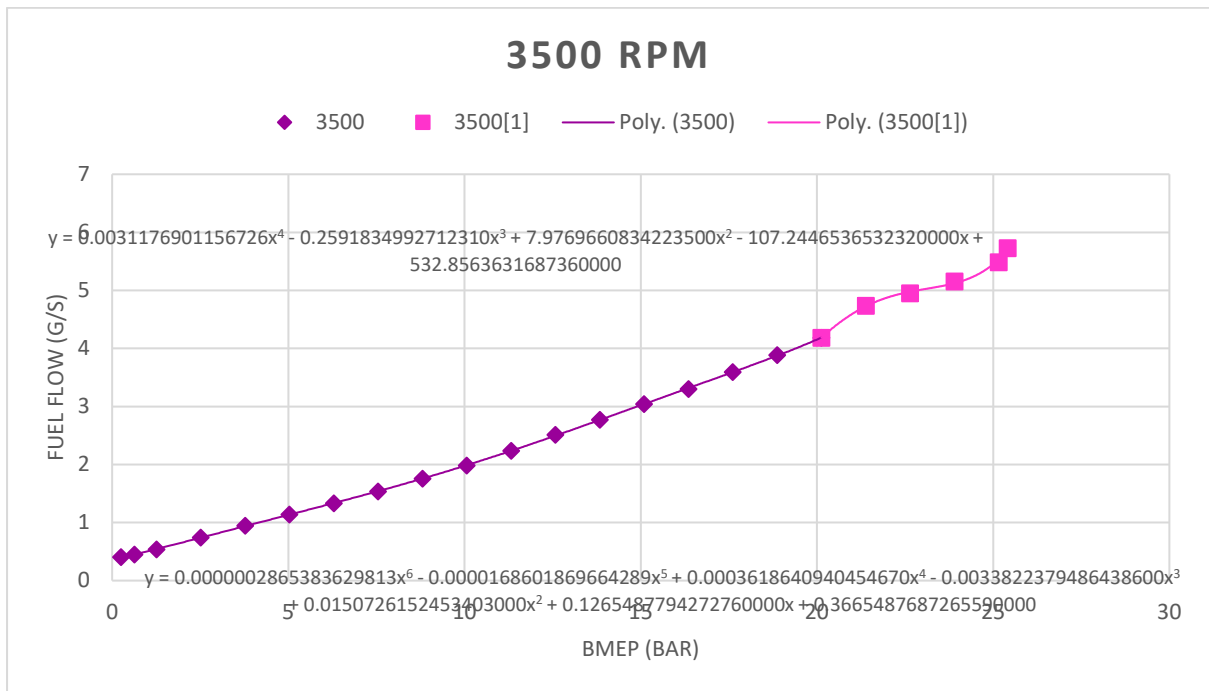


Figure 16-24: Fuel Flow Rate for Ford Fox Engine at 3500RPM

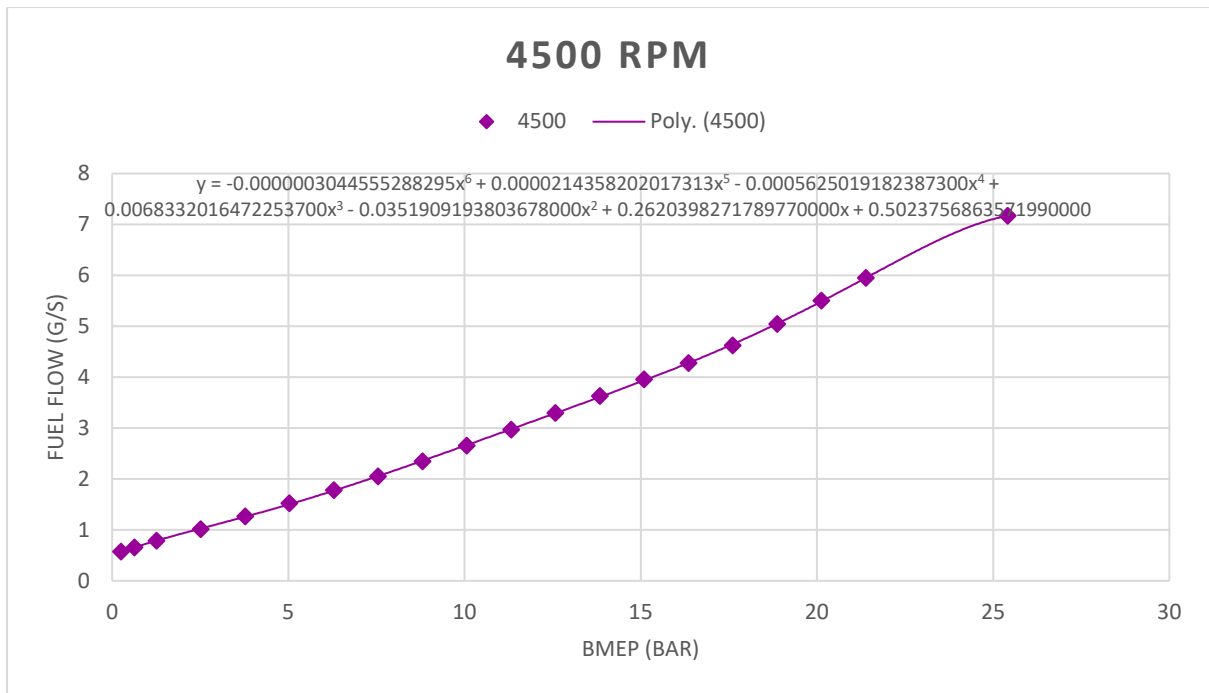


Figure 16-25: Fuel Flow Rate for Ford Fox Engine at 4500RPM

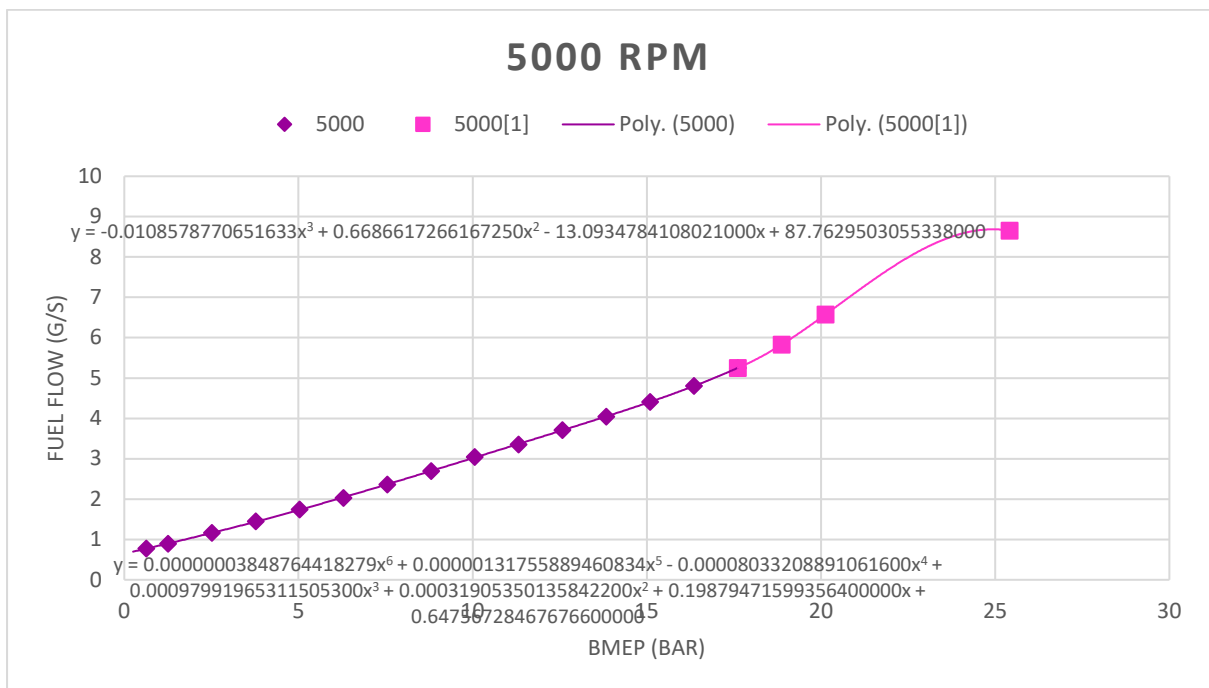


Figure 16-26: Fuel Flow Rate for Ford Fox Engine at 5000RPM

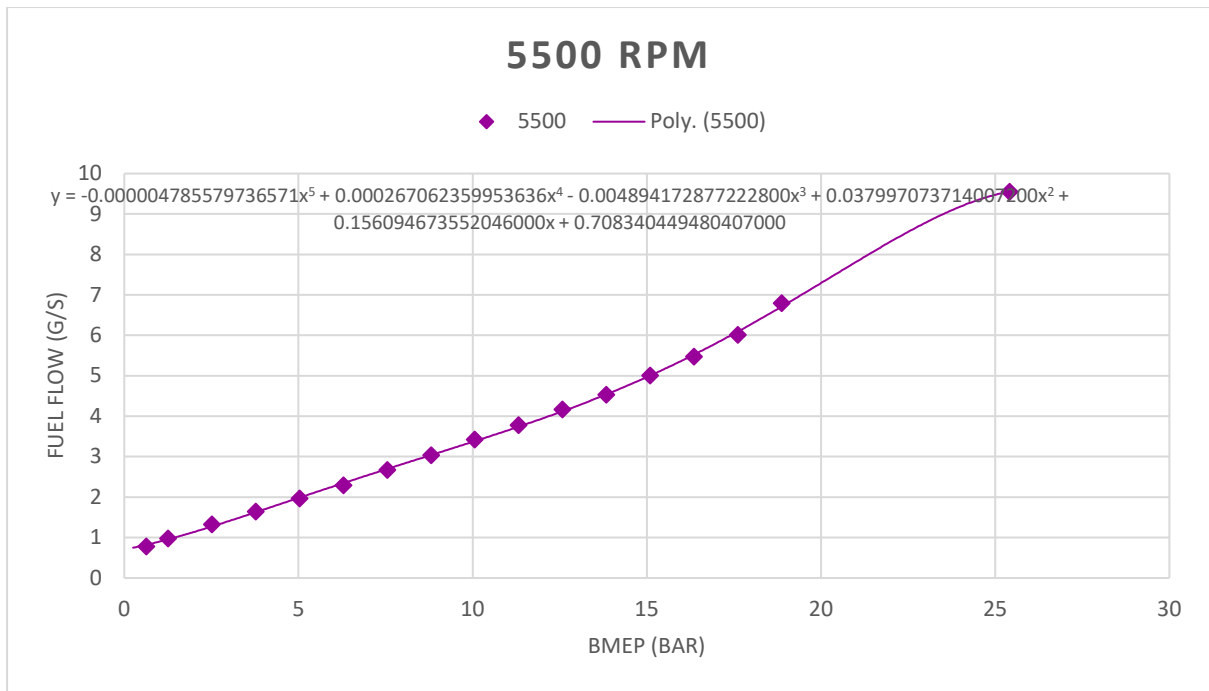


Figure 16-27: Fuel Flow Rate for Ford Fox Engine at 5500RPM

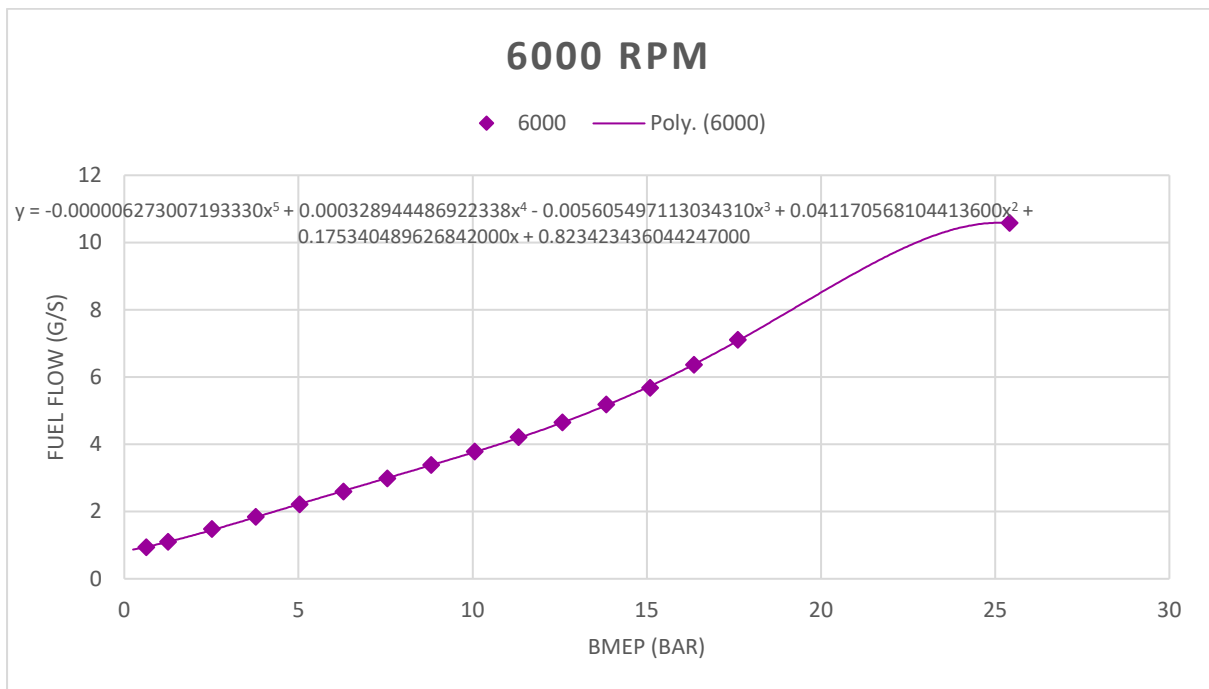


Figure 16-28: Fuel Flow Rate for Ford Fox Engine at 6000RPM

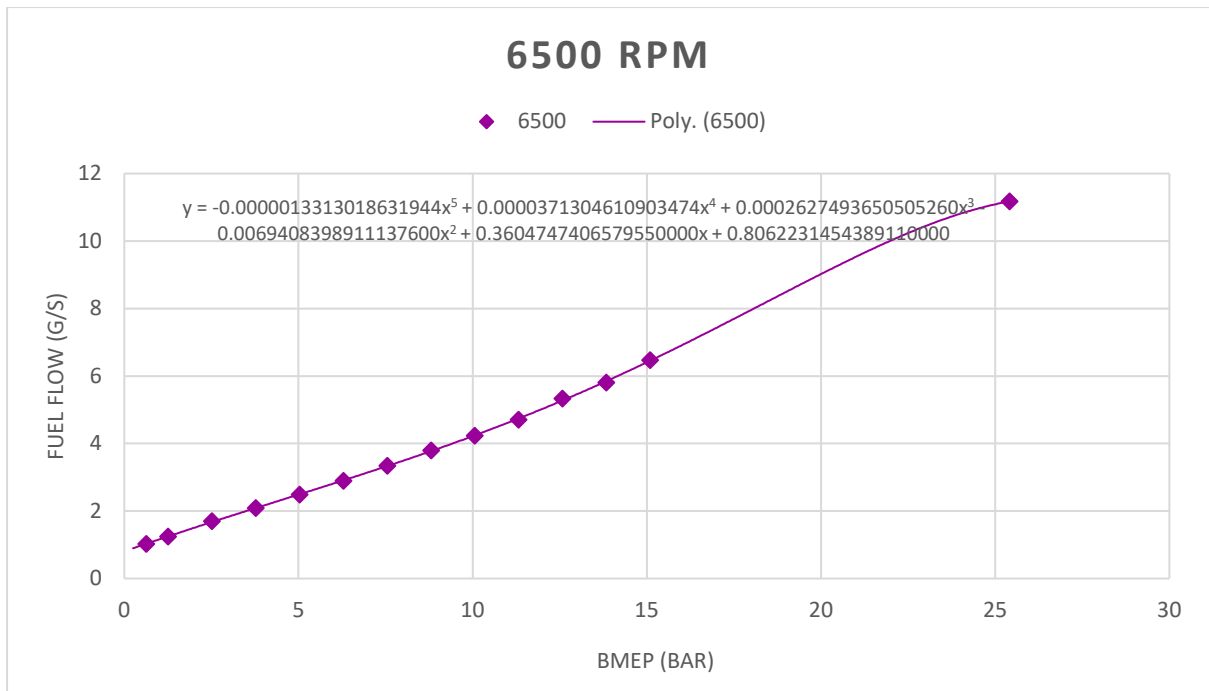


Figure 16-29: Fuel Flow Rate for Ford Fox Engine at 6500RPM

16.2.2 Results

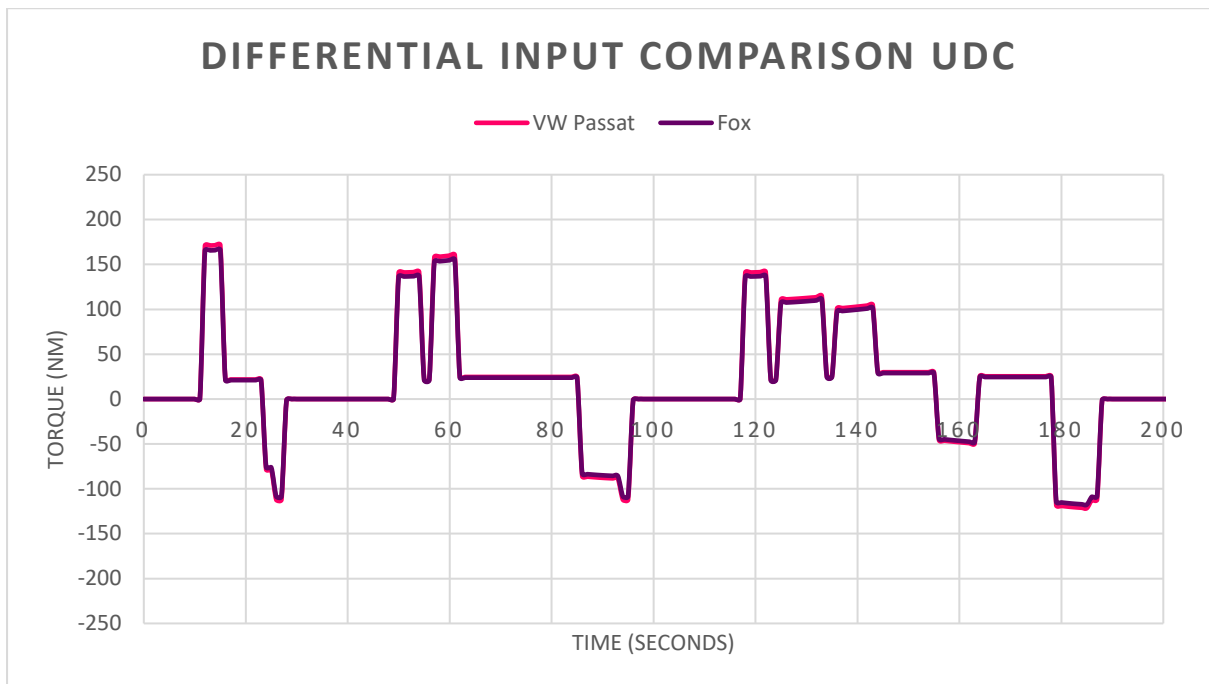


Figure 16-30: Comparison of the Differential Input Torque during the UDC Section of the NEDC

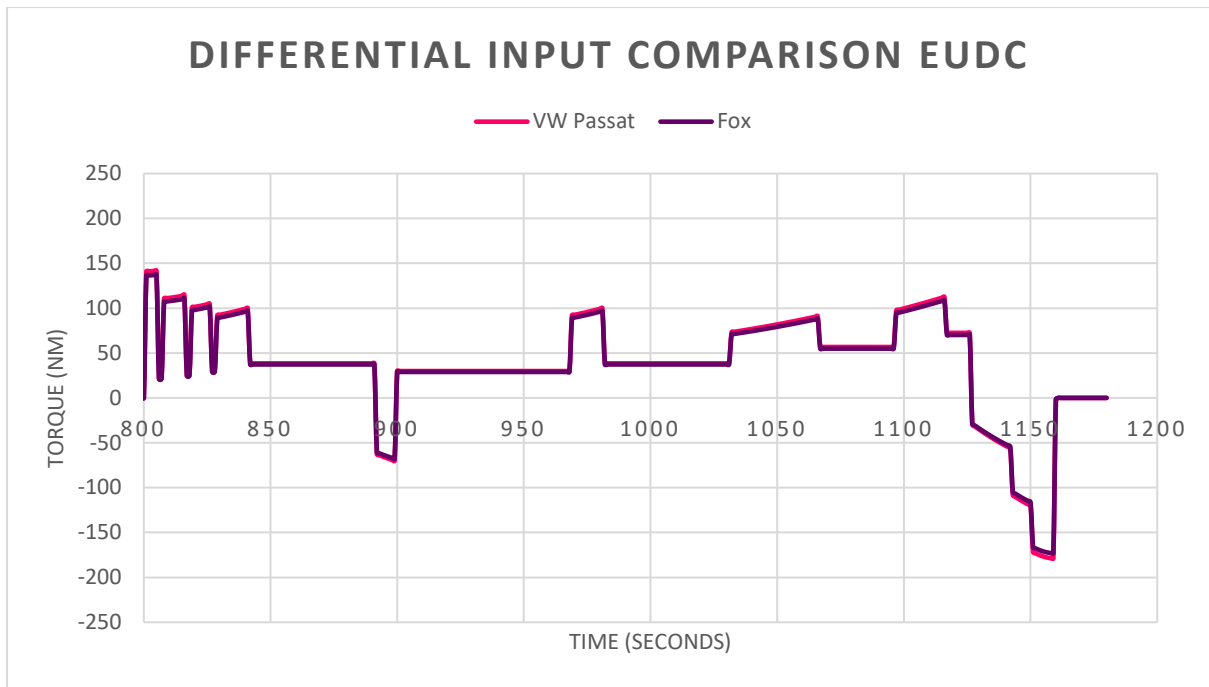


Figure 16-31: Comparison of the Differential Input Torque during the EUDC Section of the NEDC

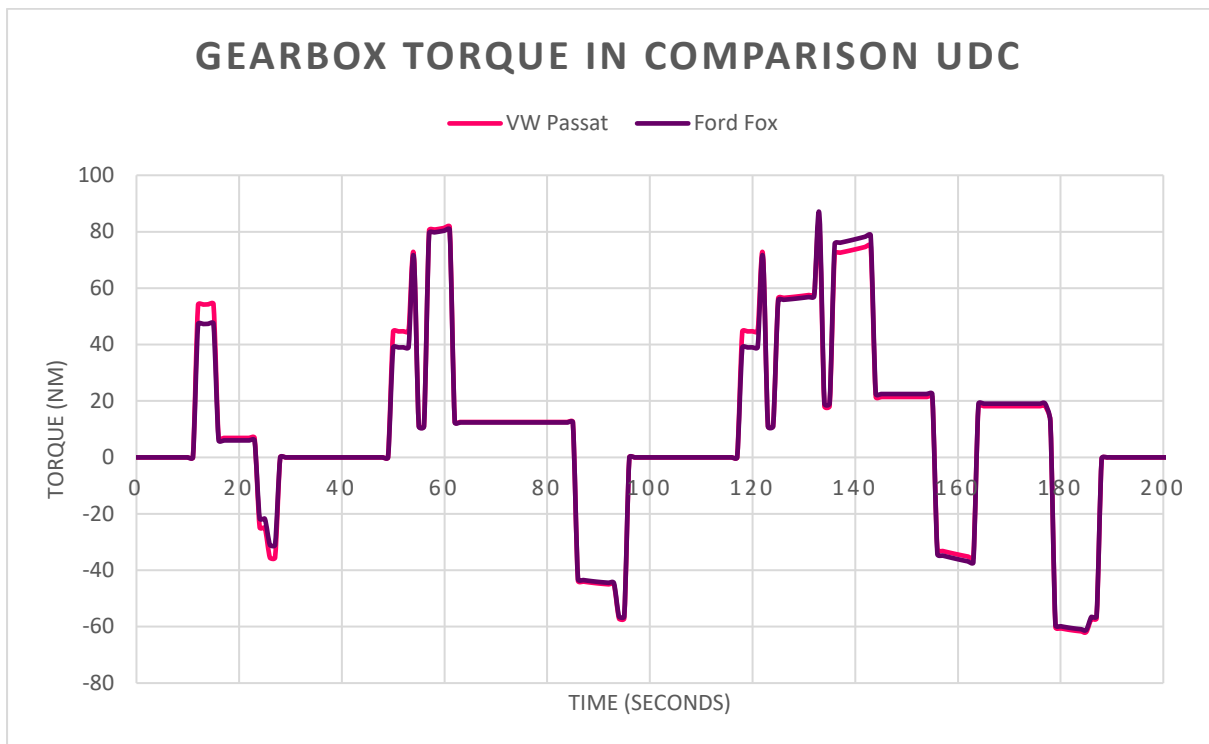


Figure 16-32: Comparison of the Gearbox Input Torque during the UDC Section of the NEDC

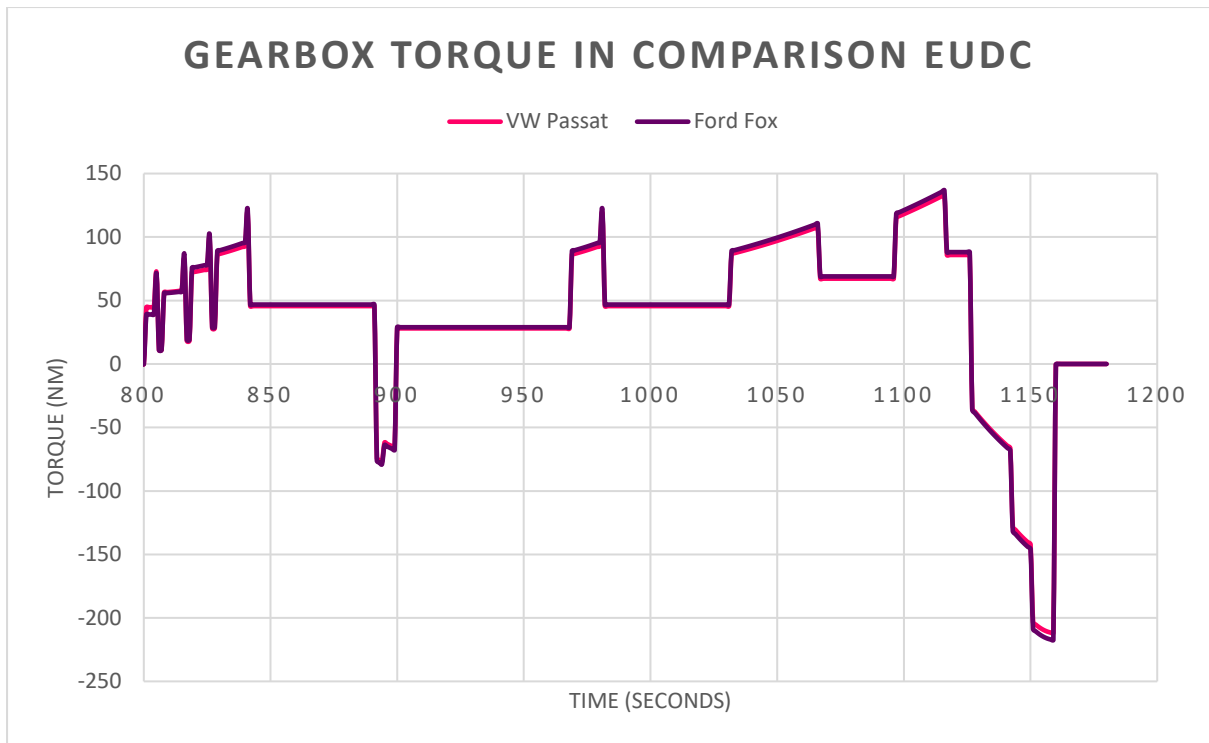


Figure 16-33: Comparison of the Gearbox Input Torque during the EUDC Section of the NEDC

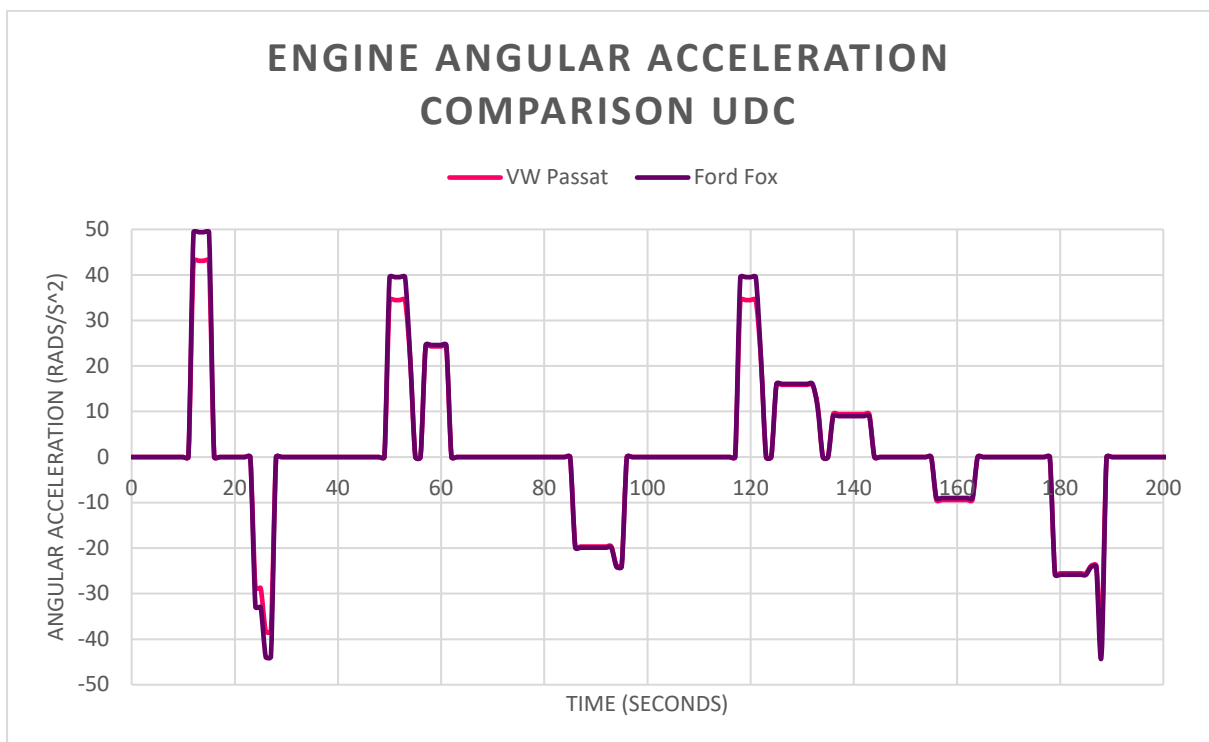


Figure 16-34: Comparison of Engine Angular Acceleration during the UDC Section of the NEDC

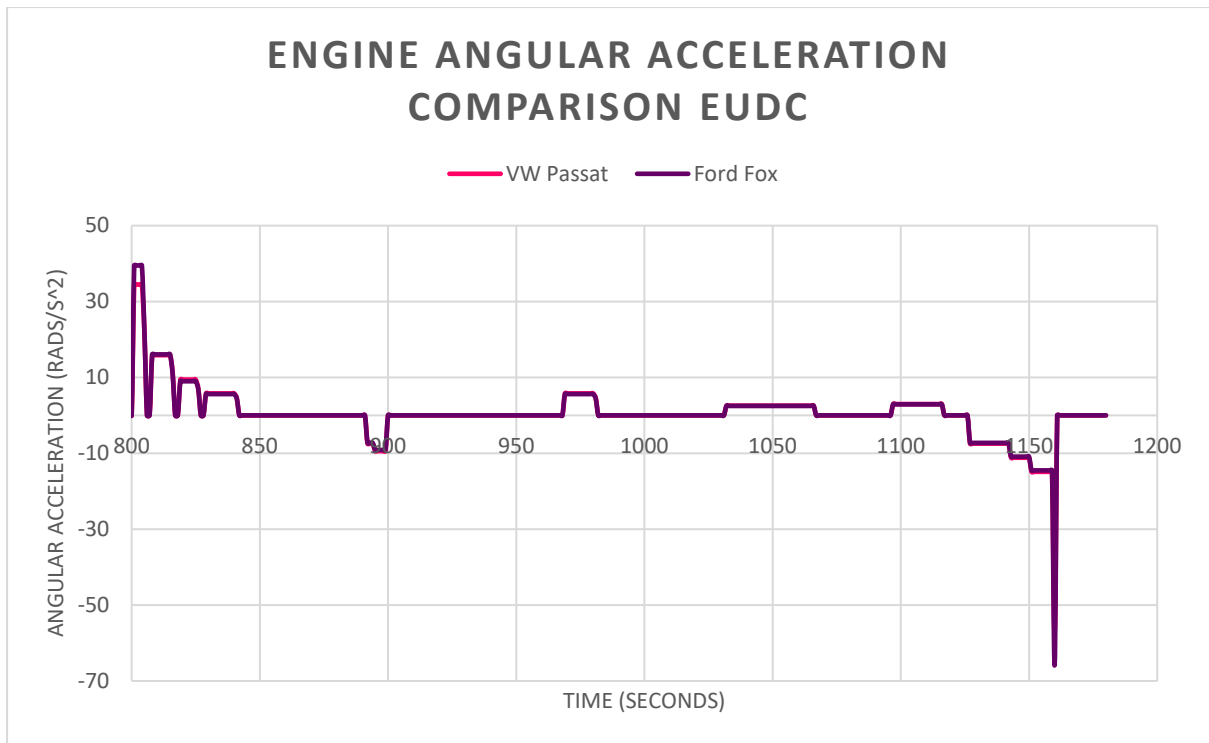


Figure 16-35: Comparison of Engine Angular Acceleration during the EUDC Section of the NEDC

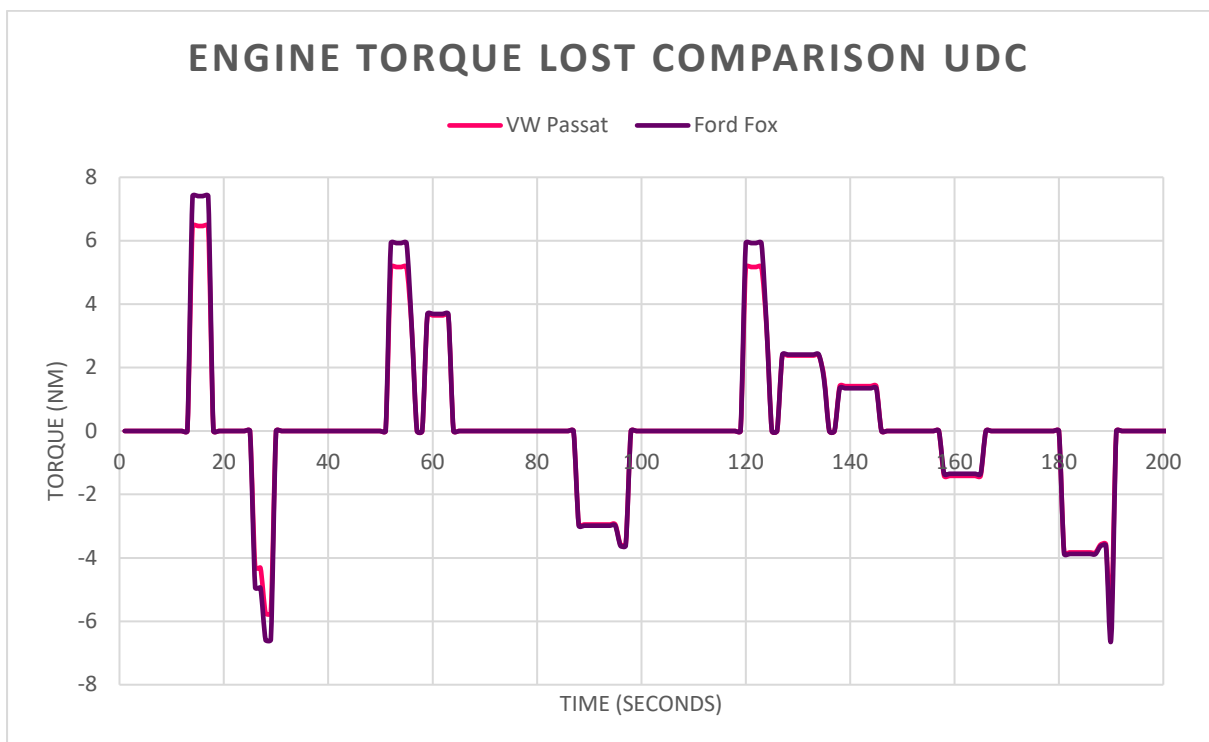


Figure 16-36: Comparison of the Engine Torque Lost during the UDC Section of the NEDC

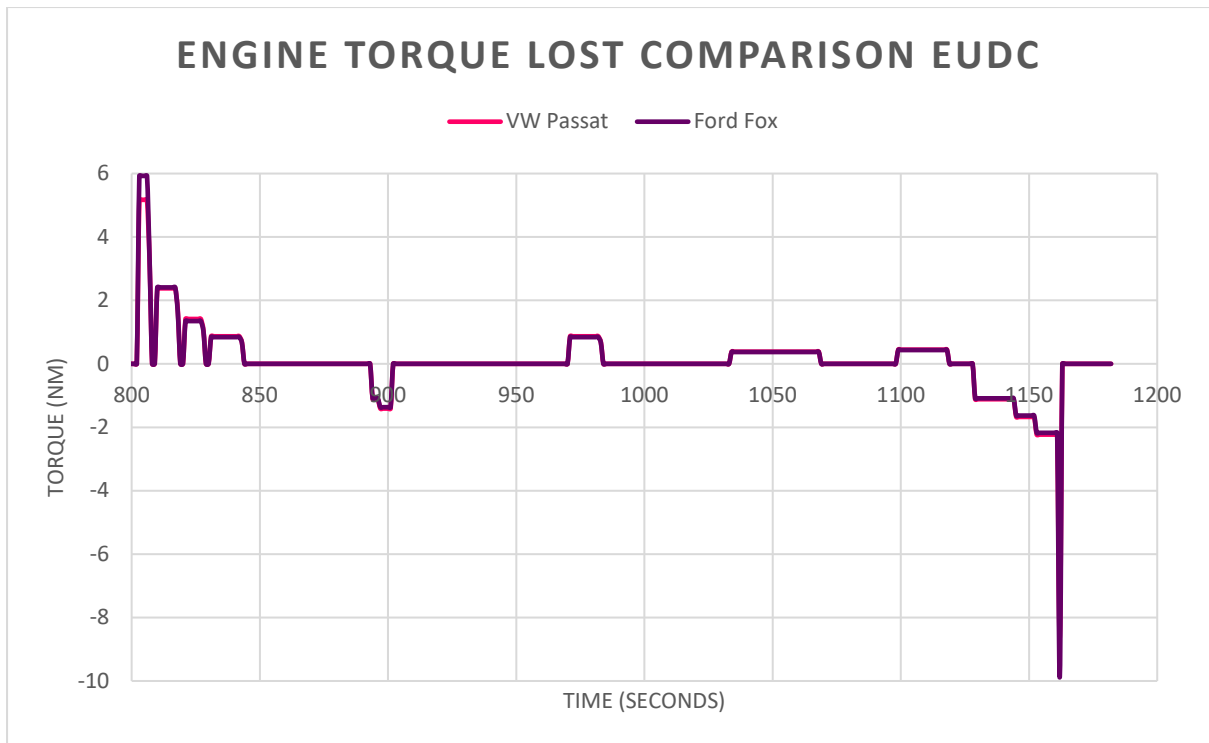


Figure 16-37: Comparison of the Engine Torque Lost during the EUDC Section of the NEDC

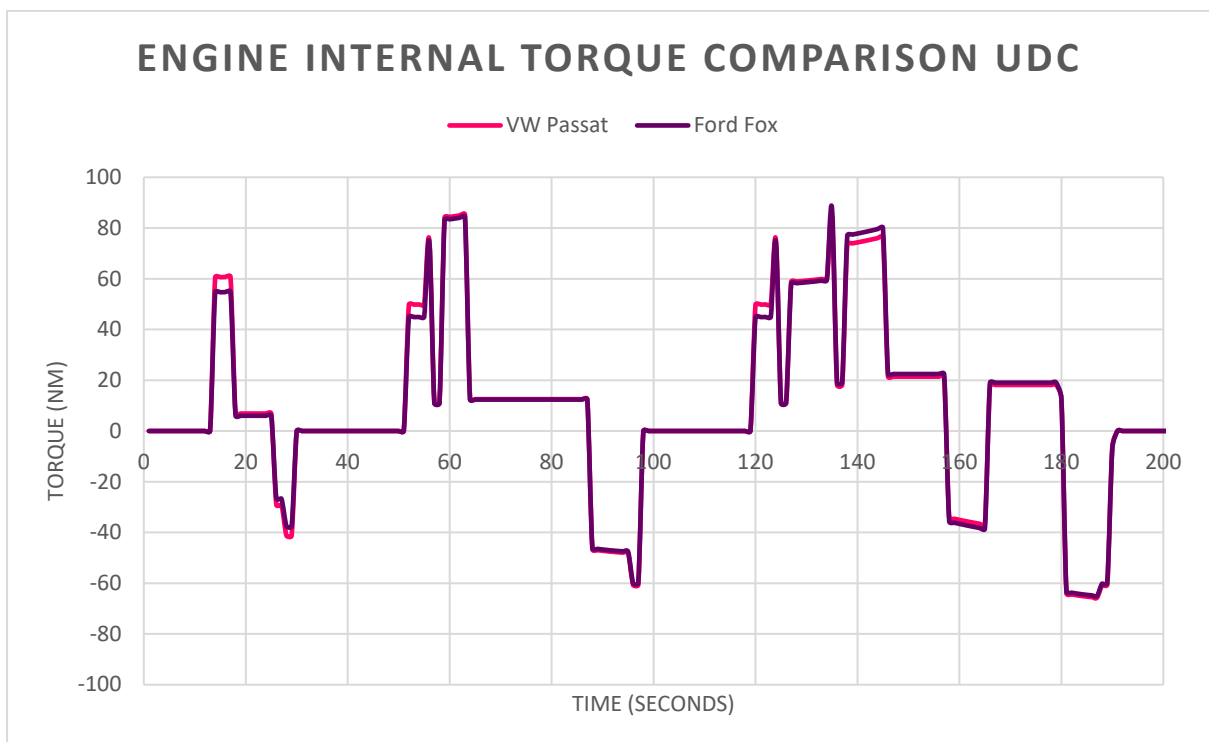


Figure 16-38: Comparison of the Engine Internal Torque during the UDC Section of the NEDC

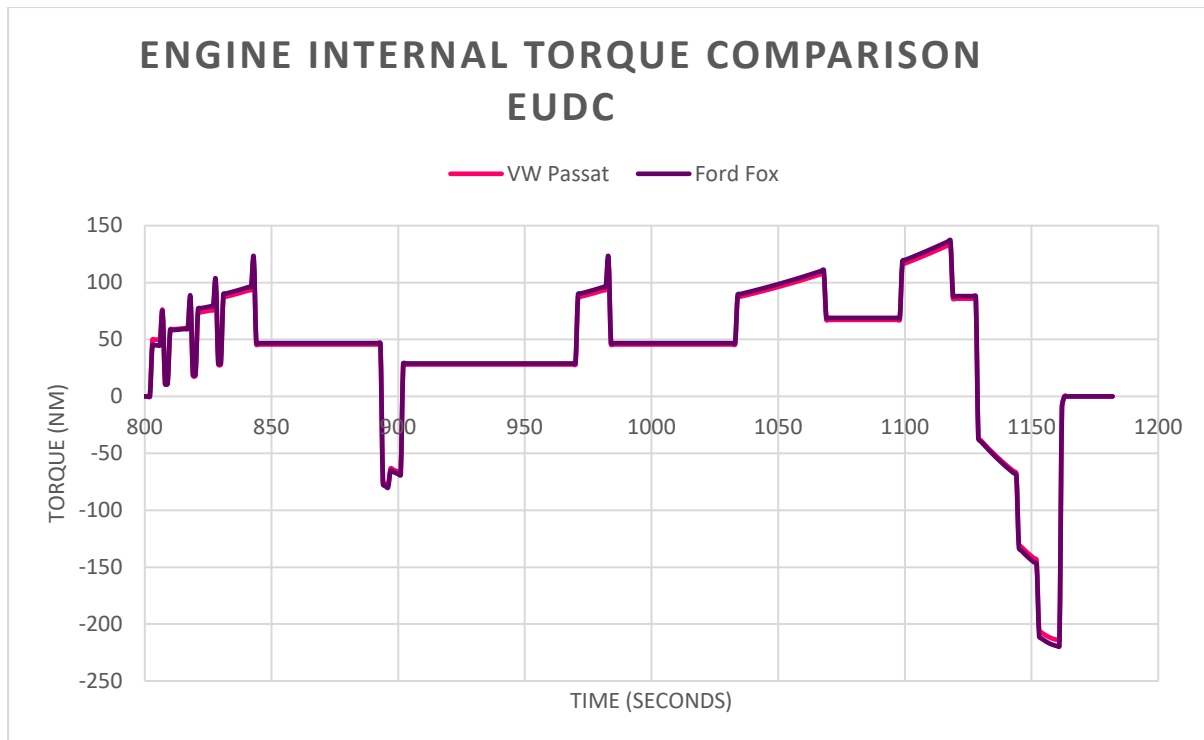


Figure 16-39: Comparison of the Engine Internal Torque during the EUDC Section of the NEDC

16.2.3 Gear Calculator

```
clc
clear all %#ok<CLALL>
% Vehicle Parameters (VW Passat)
disp('Vehicle Parameters:')
VM=1650; % Vehicle Mass (kg)
fprintf('Vehicle Mass: %d',VM)
disp(' ')
Wrr=0.32; %Wheel Rolling Radius (m)
fprintf('Wheel Rolling Radius: %d',Wrr)
disp(' ')
IDLE=800; %Engine Idle (RPM)
fprintf('Engine Idle Speed: %d',IDLE)
disp(' ')
EIn=0.15; %Engine Inertia (kgm^2)
fprintf('Engine Inertia: %d',EIn)
disp(' ')
Afr=2.235; %Frontal Area (m^2)
fprintf('Frontal Area: %d',Afr)
disp(' ')
RRc=0.015; %Rolling Resistance Coefficient
fprintf('Rolling Resistance Coefficient: %d',RRc)
disp(' ')

```

```

EDis=1984; % Engine Displacement
    fprintf('Engine Displacement: %d',EDis)
    disp(' ')
IW=0.8; %Wheel Inertia (kgm^2)
    fprintf('Wheel Inertia: %d',IW)
    disp(' ')
PwMax=110.325; %Maximum Power of the Engine (kW) (found
on web)
    fprintf('Maximum Power: %d',PwMax)
    disp(' ')
s=5000; % Engine Speed at Maximum Power (RPM) (found on
web)
    fprintf('Engine Speed at Maximum Power: %d',s)
    disp(' ')
    disp(' ')

% Constants
disp('Constants:')
g=9.81; % Gravity (m/s^2)
    fprintf('Gravity: %d',g)
    disp(' ')
CD=0.4; % Coefficient of Drag
    fprintf('Drag Coefficient: %d',CD)
    disp(' ')
Roe=1.2; % Air Density (kg/m^3)
    fprintf('Air Density: %d',Roe)
    disp(' ')

% Gear Ratios
NG=6; %Number of Gears
fprintf('Number of Gears: %d',NG)
disp(' ')
    %1st Gear
    eff1=0.94;
    R1=3.36;
    G1=eff1*R1;
    %2nd Gear
    eff2=0.94;
    R2=2.09;
    G2=eff2*R2;
    %3rd Gear
    eff3=0.95;
    R3=1.47;
    G3=eff3*R3;
    %4th Gear
    eff4=0.97;
    R4=1.1;

```

```

G4=eff4*R4;
%5th Gear
eff5=0.97;
R5=0.87;
G5=eff5*R5;
%6th Gear
eff6=0.96;
R6=0.73;
G6=eff6*R6;
%Final Drive
effFin=0.94;
RFin=3.94;
Fin=effFin*RFin;

%Initial Time and Velocity
tS=[0:1800]; %#ok<NBRAK>
ts=im2col(tS,[1 1]);
[r,c]=size(ts);
% load('WLTvariables.mat', 'Ts')
load('WLTvariables.mat', 'Vkmh')
Vms=Vkmh*0.277777778;

a=1; %#ok<NASGU>
b=1;
% Acceleration
Acc=zeros(r,c);
for a=1 b=1 %#ok<SEPEX,NOPTS>
    Acc(a,b)=0;
end
a=2;
while a<=r
    Acc(a,b)=Vms(a,b)-Vms(a-1,b);
    a=a+1;
end

%Wheel Force, Velocity, Torque
a=1;
FWh=zeros(r,c);
VWh=zeros(r,c);
TWh=zeros(r,c);
while a<=r

FWh(a,b)=(Acc(a,b)*VM)+(VM*g*RRc)+(0.5*Afr*Roe*CD*(Vms(a,
b)^2));
    if Vms(a,b)==0
        FWh(a,b)=0;
    end

```

```

    VWh(a,b)=Vms(a,b)/Wrr;
    TWh(a,b)=FWh(a,b)*Wrr;

    a=a+1;
end

%Wheel Acceleration
AccW=zeros(r,c);
for a=1
    AccW(a,b)=0;
    a=2; %#ok<FXSET>
    while a<=r
        AccW(a,b)=VWh(a,b)-VWh(a-1,b);
        a=a+1; %#ok<FXSET>
    end

end

%Wheel RPM
a=1;
RPMW=zeros(r,c);
while a<=r
    RPMW(a,b)=(VWh(a,b)/(2*pi))*60;
    a=a+1;
end

%Driveline Torque
a=1;
TDrLn=zeros(r,c);
while a<=r
    TDrLn(a,b)=TWh(a,b)+(2*IW*AccW(a,b));
    a=a+1;
end

%Differential
AccDif=AccW;
TDifO=TDrLn;
TDifI=zeros(r,c);
a=1;
while a<=r
    TDifI(a,b)=TDifO(a,b)/Fin;
    a=a+1;
end

%Gear Out Torque
TGrO=TDifI;

%Gears

```

```

a=1;
b=1;
Gears=zeros(r,c);
while a<=r
    Gears(a,b)=1;
    if Gears(a,b)==1
        TGbI(a,b)=TGrO(a,b)/G1; %#ok<*SAGROW>
        AccEn(a,b)=R1*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGbI(a,b);
        RPM(a,b)=RPMW(a,b)*R1*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
        if RPM(a,b)<=IDLE
            RPM(a,b)=IDLE;
        end
    end
    if RPM(a,b)>=1.25*IDLE
        Gears(a,b)=2;
        if Gears(a,b)==2
            TGbI(a,b)=TGrO(a,b)/G2; %#ok<*SAGROW>
            AccEn(a,b)=R2*RFin*AccW(a,b);
            Tlost(a,b)=AccEn(a,b)*EIn;
            TEngO(a,b)=TGbI(a,b);
            RPM(a,b)=RPMW(a,b)*R2*RFin;
            ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
            if RPM(a,b)<=IDLE
                RPM(a,b)=IDLE;
            end
        end
    end
    if RPM(a,b)>=(IDLE+(0.125*(s-IDLE)))
        Gears(a,b)=3;
        if Gears(a,b)==3
            TGbI(a,b)=TGrO(a,b)/G3; %#ok<*SAGROW>
            AccEn(a,b)=R3*RFin*AccW(a,b);
            Tlost(a,b)=AccEn(a,b)*EIn;
            TEngO(a,b)=TGbI(a,b);
            RPM(a,b)=RPMW(a,b)*R3*RFin;
            ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
            if RPM(a,b)<=IDLE
                RPM(a,b)=IDLE;
            end
        end
    end
    if RPM(a,b)>=(IDLE+(0.125*(s-IDLE)))
        Gears(a,b)=4;
        if Gears(a,b)==4

```

```

    TGbI(a,b)=TGrO(a,b)/G4; %#ok<*SAGROW>
    AccEn(a,b)=R4*RFin*AccW(a,b);
    Tlost(a,b)=AccEn(a,b)*EIn;
    TEngO(a,b)=TGbI(a,b);
    RPM(a,b)=RPMW(a,b)*R4*RFin;
    ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
    if RPM(a,b)<=IDLE
        RPM(a,b)=IDLE;
    end
end
end
if RPM(a,b)>=(IDLE+(0.125*(s-IDLE)))
    Gears(a,b)=5;
    if Gears(a,b)==5
        TGbI(a,b)=TGrO(a,b)/G5; %#ok<*SAGROW>
        AccEn(a,b)=R5*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGbI(a,b);
        RPM(a,b)=RPMW(a,b)*R5*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
        if RPM(a,b)<=IDLE
            RPM(a,b)=IDLE;
        end
    end
end
end
if RPM(a,b)>=(IDLE+(0.125*(s-IDLE)))
    Gears(a,b)=6;
    if Gears(a,b)==6
        TGbI(a,b)=TGrO(a,b)/G6; %#ok<*SAGROW>
        AccEn(a,b)=R6*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGbI(a,b);
        RPM(a,b)=RPMW(a,b)*R6*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
        if RPM(a,b)<=IDLE
            RPM(a,b)=IDLE;
        end
    end
end
end

% Section E of Legislation
% (1)
if a>=3 && a<=r
    if Gears(a,b)==Gears(a-2,b)&&Gears(a,b)==Gears(a-
1,b)+1
        Gears(a-1,b)=Gears(a,b);
    end
end

```

```

end

% (2)
if a>=4 && a<=r
    if Gears(a,b)==Gears(a-3) && Gears(a,b)==Gears(a-
1,b)+1&&Gears(a,b)==Gears(a-2,b)+1
        Gears([a-1 a-2])=Gears(a,b);
    end
end

% (3)
if a>=5 && a<=r
    if Gears(a,b)==Gears(a-4,b) && Gears(a,b)==Gears (a-
1,b)+1 && Gears(a,b)==Gears(a-2,b)+1 &&
Gears(a,b)==Gears(a-3,b)+1
        Gears([a-1 a-2 a-3])=Gears(a,b);
    end
end

% (4)
if a>=6 && a<=r
    if Gears(a,b)==Gears(a-5,b) && Gears(a,b)==Gears (a-
1,b)+1 && Gears(a,b)==Gears(a-2,b)+1 &&
Gears(a,b)==Gears(a-3,b)+1 && Gears(a,b)==Gears(a-4,b)+1
        Gears([a-1 a-2 a-3 a-4])=Gears(a,b);
    end
end

% (5)
if a>=7 && a<=r
    if Gears(a,b)==Gears(a-6,b) && Gears(a,b)==Gears (a-
1,b)+1 && Gears(a,b)==Gears(a-2,b)+1 &&
Gears(a,b)==Gears(a-3,b)+1 && Gears(a,b)==Gears(a-4,b)+1
&& Gears(a,b)==Gears(a-5)+1
        Gears([a-1 a-2 a-3 a-4 a-5])=Gears(a,b);
    end
end

%Section D of Legislation
if a<=r-2
    if Vkmh(a+1,b)>Vkmh(a,b) && Vkmh(a+1)>Vkmh(a+2,b) &&
Gears(a,b)==Gears(a+1)
        Gears(a+2,b)=Gears(a+1);
    end
end

%Section A of Legislation

```

```

if Vms(a,b)==0
    Gears(a,b)=0;
    if a<=r-1 && Vkmh(a+1,b)>=1 && Vms(a,b)==0
        Gears(a,b)=1;
    end
end
a=a+1;
end

% Recalculation with new gears
a=1;
while a<=r
    if Gears(a,b)==1
        TGrI(a,b)=TGrO(a,b)/G1; %#ok<*SAGROW>
        AccEn(a,b)=R1*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGrI(a,b);
        RPM(a,b)=RPMW(a,b)*R1*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if Gears(a,b)==2
        TGrI(a,b)=TGrO(a,b)/G2; %#ok<*SAGROW>
        AccEn(a,b)=R2*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGrI(a,b);
        RPM(a,b)=RPMW(a,b)*R2*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if Gears(a,b)==3
        TGrI(a,b)=TGrO(a,b)/G3; %#ok<*SAGROW>
        AccEn(a,b)=R3*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGrI(a,b);
        RPM(a,b)=RPMW(a,b)*R3*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if Gears(a,b)==4
        TGrI(a,b)=TGrO(a,b)/G4; %#ok<*SAGROW>
        AccEn(a,b)=R4*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGrI(a,b);
        RPM(a,b)=RPMW(a,b)*R4*RFin;
        ndv1(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if Gears(a,b)==5
        TGrI(a,b)=TGrO(a,b)/G5; %#ok<*SAGROW>
        AccEn(a,b)=R5*RFin*AccW(a,b);
    end
end

```

```

        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGbI(a,b);
        RPM(a,b)=RPMW(a,b)*R5*RFin;
        ndvl(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if Gears(a,b)==6
        TGbI(a,b)=TGrO(a,b)/G6; %#ok<*SAGROW>
        AccEn(a,b)=R6*RFin*AccW(a,b);
        Tlost(a,b)=AccEn(a,b)*EIn;
        TEngO(a,b)=TGbI(a,b);
        RPM(a,b)=RPMW(a,b)*R6*RFin;
        ndvl(a,b)=RPM(a,b)/Vkmh(a,b);
    end
    if RPM(a,b)<=IDLE
        RPM(a,b)=IDLE;
    end
    a=a+1;
end

%Engine Internal Torque
a=1;
while a<=r
    EnITq(a,b)=Tlost(a,b)+TEngO(a,b);
    a=a+1;
end

%BMEP
a=1;
while a<=r
    BMEP(a,b)=( (EnITq(a,b)*4*pi)/(EDis*10^-6))*(10^-5);
    if BMEP(a,b)<=0
        BMEP(a,b)=0;
    end
    a=a+1;
end

%Power (kW)
a=1;
while a<=r
    Pw(a,b)=( (2*pi*EnITq(a,b)*RPM(a,b))/60000);
    a=a+1;
end

%Fuel Flow
a=1;
while a<=r

```

```

    if RPM(a,b)<500
        FF(a,b)=((0)+((RPM(a,b)-0)/(500-0))*((1.733e-
10*BMEP(a,b)^9 + -1.306e-08*BMEP(a,b)^8 + 3.706e-
07*BMEP(a,b)^7 + -4.631e-06 *BMEP(a,b)^6 +1.789e-
05*BMEP(a,b)^5 + 0.0001092*BMEP(a,b)^4 + -
0.001028*BMEP(a,b)^3 + 0.002996*BMEP(a,b)^2 +
0.04316*BMEP(a,b) + 0.07863)-(0)))));
    end

    if RPM(a,b)>=500 && RPM(a,b)<850
        FF(a,b)=((1.733e-10*BMEP(a,b)^9 + -1.306e-
08*BMEP(a,b)^8 + 3.706e-07*BMEP(a,b)^7 + -4.631e-06
*BMEP(a,b)^6 +1.789e-05*BMEP(a,b)^5 +
0.0001092*BMEP(a,b)^4 + -0.001028*BMEP(a,b)^3 +
0.002996*BMEP(a,b)^2 + 0.04316*BMEP(a,b) +
0.07863)+((RPM(a,b)-500)/(850-500))*((1.677e-
10*BMEP(a,b)^9 + -8.674e-09*BMEP(a,b)^8 + 4.667e-
08*BMEP(a,b)^7 + 5.356e-06*BMEP(a,b)^6 + -
0.0001404*BMEP(a,b)^5 + 0.001455*BMEP(a,b)^4 + -
0.006926*BMEP(a,b)^3 + 0.01528*BMEP(a,b)^2 +
0.06465*BMEP(a,b) + 0.1404)-(1.733e-10*BMEP(a,b)^9 + -
1.306e-08*BMEP(a,b)^8 + 3.706e-07*BMEP(a,b)^7 + -4.631e-
06 *BMEP(a,b)^6 +1.789e-05*BMEP(a,b)^5 +
0.0001092*BMEP(a,b)^4 + -0.001028*BMEP(a,b)^3 +
0.002996*BMEP(a,b)^2 + 0.04316*BMEP(a,b) + 0.07863)))));
    end

    if RPM(a,b)>=850 || RPM(a,b)<1000
        FF(a,b)=((1.677e-10*BMEP(a,b)^9 + -8.674e-
09*BMEP(a,b)^8 + 4.667e-08*BMEP(a,b)^7 + 5.356e-
06*BMEP(a,b)^6 + -0.0001404*BMEP(a,b)^5 +
0.001455*BMEP(a,b)^4 + -0.006926*BMEP(a,b)^3 +
0.01528*BMEP(a,b)^2 + 0.06465*BMEP(a,b) +
0.1404)+((RPM(a,b)-850)/(1000-850))*((1.108e-
10*BMEP(a,b)^9 + -1.594e-09*BMEP(a,b)^8 + -2.971e-
07*BMEP(a,b)^7 + 1.394e-05*BMEP(a,b)^6 + -
0.00026*BMEP(a,b)^5 + 0.002391*BMEP(a,b)^4 + -
0.01083*BMEP(a,b)^3 + 0.02304*BMEP(a,b)^2 +
0.07192*BMEP(a,b) + 0.1685)-(1.677e-10*BMEP(a,b)^9 + -
8.674e-09*BMEP(a,b)^8 + 4.667e-08*BMEP(a,b)^7 + 5.356e-
06*BMEP(a,b)^6 + -0.0001404*BMEP(a,b)^5 +
0.001455*BMEP(a,b)^4 + -0.006926*BMEP(a,b)^3 +
0.01528*BMEP(a,b)^2 + 0.06465*BMEP(a,b) + 0.1404)))));
    end

    if RPM(a,b)>=1000 || RPM(a,b)<1250

```

```

        FF(a,b)=((1.108e-10*BMEP(a,b)^9 + -1.594e-
09*BMEP(a,b)^8 + -2.971e-07*BMEP(a,b)^7 + 1.394e-
05*BMEP(a,b)^6 + -0.00026*BMEP(a,b)^5 +
0.002391*BMEP(a,b)^4 + -0.01083*BMEP(a,b)^3 +
0.02304*BMEP(a,b)^2 + 0.07192*BMEP(a,b) +
0.1685)+((RPM(a,b)-1000)/(1250-1000))*((-9.99e-
11*BMEP(a,b)^9 + 2.068e-08*BMEP(a,b)^8 + -1.263e-
06*BMEP(a,b)^7 + 3.612e-05*BMEP(a,b)^6 + -
0.0005503*BMEP(a,b)^5 + 0.00456*BMEP(a,b)^4 + -
0.01956*BMEP(a,b)^3 + 0.0399*BMEP(a,b)^2 + 0.0813
*BMEP(a,b) + 0.2173)-(1.108e-10*BMEP(a,b)^9 + -1.594e-
09*BMEP(a,b)^8 + -2.971e-07*BMEP(a,b)^7 + 1.394e-
05*BMEP(a,b)^6 + -0.00026*BMEP(a,b)^5 +
0.002391*BMEP(a,b)^4 + -0.01083*BMEP(a,b)^3 +
0.02304*BMEP(a,b)^2 + 0.07192*BMEP(a,b) + 0.1685)))));
    end

    if RPM(a,b)>=1250 || RPM(a,b)<1500
        FF(a,b)=((-9.99e-11*BMEP(a,b)^9 + 2.068e-
08*BMEP(a,b)^8 + -1.263e-06*BMEP(a,b)^7 + 3.612e-
05*BMEP(a,b)^6 + -0.0005503*BMEP(a,b)^5 +
0.00456*BMEP(a,b)^4 + -0.01956*BMEP(a,b)^3 +
0.0399*BMEP(a,b)^2 + 0.0813 *BMEP(a,b) +
0.2173)+((RPM(a,b)-1250)/(1500-1250))*((-3.867e-
10*BMEP(a,b)^9 + 4.931e-08*BMEP(a,b)^8 + -2.451e-
06*BMEP(a,b)^7 + 6.254e-05*BMEP(a,b)^6 + -
0.0008886*BMEP(a,b)^5 + 0.007059*BMEP(a,b)^4 + -
0.02962*BMEP(a,b)^3 + 0.05932*BMEP(a,b)^2 +
0.08925*BMEP(a,b) + 0.268)-(-9.99e-11*BMEP(a,b)^9 +
2.068e-08*BMEP(a,b)^8 + -1.263e-06*BMEP(a,b)^7 + 3.612e-
05*BMEP(a,b)^6 + -0.0005503*BMEP(a,b)^5 +
0.00456*BMEP(a,b)^4 + -0.01956*BMEP(a,b)^3 +
0.0399*BMEP(a,b)^2 + 0.0813 *BMEP(a,b) + 0.2173)))));
    end

    if RPM(a,b)>=1500 || RPM(a,b)<1750
        FF(a,b)=((-3.867e-10*BMEP(a,b)^9 + 4.931e-
08*BMEP(a,b)^8 + -2.451e-06*BMEP(a,b)^7 + 6.254e-
05*BMEP(a,b)^6 + -0.0008886*BMEP(a,b)^5 +
0.007059*BMEP(a,b)^4 + -0.02962*BMEP(a,b)^3 +
0.05932*BMEP(a,b)^2 + 0.08925*BMEP(a,b) +
0.268)+((RPM(a,b)-1500)/(1750-1500))*((-3.956e-
10*BMEP(a,b)^9 + 5.161e-08*BMEP(a,b)^8 + -2.617e-
06*BMEP(a,b)^7 + 6.793e-05*BMEP(a,b)^6 + -
0.0009813*BMEP(a,b)^5 + 0.007937*BMEP(a,b)^4 + -
0.03403*BMEP(a,b)^3 + 0.06962*BMEP(a,b)^2 +
0.1035*BMEP(a,b) + 0.3194)-(-3.867e-10*BMEP(a,b)^9 +

```

```

4.931e-08*BMEP(a,b)^8 + -2.451e-06*BMEP(a,b)^7 + 6.254e-
05*BMEP(a,b)^6 + -0.0008886*BMEP(a,b)^5 +
0.007059*BMEP(a,b)^4 + -0.02962*BMEP(a,b)^3 +
0.05932*BMEP(a,b)^2 + 0.08925*BMEP(a,b) + 0.268))));

```

```

end

```

```

if RPM(a,b)>=1750 || RPM(a,b)<2000
    FF(a,b)=((-3.956e-10*BMEP(a,b)^9 + 5.161e-
08*BMEP(a,b)^8 + -2.617e-06*BMEP(a,b)^7 + 6.793e-
05*BMEP(a,b)^6 + -0.0009813*BMEP(a,b)^5 +
0.007937*BMEP(a,b)^4 + -0.03403*BMEP(a,b)^3 +
0.06962*BMEP(a,b)^2 + 0.1035*BMEP(a,b) +
0.3194)+((RPM(a,b)-1750)/(2000-1750))*((5.275e-
10*BMEP(a,b)^9 + -3.114e-08*BMEP(a,b)^8 + 4.316e-
07*BMEP(a,b)^7 + 8.301e-06*BMEP(a,b)^6 +-
0.0003139*BMEP(a,b)^5 + 0.003657*BMEP(a,b)^4 + -
0.01912*BMEP(a,b)^3 + 0.04454*BMEP(a,b)^2 +
0.1425*BMEP(a,b) + 0.3698)-(-3.956e-10*BMEP(a,b)^9 +
5.161e-08*BMEP(a,b)^8 + -2.617e-06*BMEP(a,b)^7 + 6.793e-
05*BMEP(a,b)^6 + -0.0009813*BMEP(a,b)^5 +
0.007937*BMEP(a,b)^4 + -0.03403*BMEP(a,b)^3 +
0.06962*BMEP(a,b)^2 + 0.1035*BMEP(a,b) + 0.3194))));

```

```

end

```

```

if RPM(a,b)>=2000 || RPM(a,b)<2500
    FF(a,b)=((5.275e-10*BMEP(a,b)^9 + -3.114e-
08*BMEP(a,b)^8 + 4.316e-07*BMEP(a,b)^7 + 8.301e-
06*BMEP(a,b)^6 +-0.0003139*BMEP(a,b)^5 +
0.003657*BMEP(a,b)^4 + -0.01912*BMEP(a,b)^3 +
0.04454*BMEP(a,b)^2 + 0.1425*BMEP(a,b) +
0.3698)+((RPM(a,b)-2000)/(2500-2000))*((8.046e-
10*BMEP(a,b)^9 + -6.267e-08*BMEP(a,b)^8 + 1.881e-06
*BMEP(a,b)^7 + -2.687e-05*BMEP(a,b)^6
+0.0001757*BMEP(a,b)^5 + -0.000298*BMEP(a,b)^4 + -
0.001155*BMEP(a,b)^3 + 0.002533*BMEP(a,b)^2 +
0.2283*BMEP(a,b) + 0.4697)-(5.275e-10*BMEP(a,b)^9 + -
3.114e-08*BMEP(a,b)^8 + 4.316e-07*BMEP(a,b)^7 + 8.301e-
06*BMEP(a,b)^6 +-0.0003139*BMEP(a,b)^5 +
0.003657*BMEP(a,b)^4 + -0.01912*BMEP(a,b)^3 +
0.04454*BMEP(a,b)^2 + 0.1425*BMEP(a,b) + 0.3698))));

```

```

end

```

```

if RPM(a,b)>=2500 || RPM(a,b)<3000
    FF(a,b)=((8.046e-10*BMEP(a,b)^9 + -6.267e-
08*BMEP(a,b)^8 + 1.881e-06 *BMEP(a,b)^7 + -2.687e-
05*BMEP(a,b)^6 +0.0001757*BMEP(a,b)^5 + -
0.000298*BMEP(a,b)^4 + -0.001155*BMEP(a,b)^3 +

```

```

0.002533*BMEP(a,b)^2 + 0.2283*BMEP(a,b) +
0.4697)+(((RPM(a,b)-2500)/(3000-2500))*((-4.689e-
10*BMEP(a,b)^9 + 3.619e-08*BMEP(a,b)^8 + -1.152e-
06*BMEP(a,b)^7 + 1.975e-05*BMEP(a,b)^6 + -
0.0002016*BMEP(a,b)^5 + 0.001276*BMEP(a,b)^4 + -
0.004439*BMEP(a,b)^3 + 0.005787*BMEP(a,b)^2 +
0.2831*BMEP(a,b) + 0.5691)-(8.046e-10*BMEP(a,b)^9 + -
6.267e-08*BMEP(a,b)^8 + 1.881e-06*BMEP(a,b)^7 + -2.687e-
05*BMEP(a,b)^6 +0.0001757*BMEP(a,b)^5 + -
0.000298*BMEP(a,b)^4 + -0.001155*BMEP(a,b)^3 +
0.002533*BMEP(a,b)^2 + 0.2283*BMEP(a,b) + 0.4697))));

```

```

end

```

```

if RPM(a,b)>=3000 || RPM(a,b)<3500
    FF(a,b)=((-4.689e-10*BMEP(a,b)^9 + 3.619e-
08*BMEP(a,b)^8 + -1.152e-06*BMEP(a,b)^7 + 1.975e-
05*BMEP(a,b)^6 + -0.0002016*BMEP(a,b)^5 +
0.001276*BMEP(a,b)^4 + -0.004439*BMEP(a,b)^3 +
0.005787*BMEP(a,b)^2 + 0.2831*BMEP(a,b) +
0.5691)+(((RPM(a,b)-3000)/(3500-3000))*((-4.734e-
09*BMEP(a,b)^9 + 4.041e-07*BMEP(a,b)^8 + -1.42e-
05*BMEP(a,b)^7 + 0.0002663*BMEP(a,b)^6 + -
0.002896*BMEP(a,b)^5 + 0.01859*BMEP(a,b)^4 + -
0.06767*BMEP(a,b)^3 + 0.1224*BMEP(a,b)^2 +
0.2644*BMEP(a,b) + 0.6665)-(-4.689e-10*BMEP(a,b)^9 +
3.619e-08*BMEP(a,b)^8 + -1.152e-06*BMEP(a,b)^7 + 1.975e-
05*BMEP(a,b)^6 + -0.0002016*BMEP(a,b)^5 +
0.001276*BMEP(a,b)^4 + -0.004439*BMEP(a,b)^3 +
0.005787*BMEP(a,b)^2 + 0.2831*BMEP(a,b) + 0.5691))));

```

```

end

```

```

a=a+1;

```

```

end

```

```

%Cumulative Fuel

```

```

for a=1

```

```

    FFtot(a,b)=FF(a,b);

```

```

end

```

```

a=2;

```

```

while a<=r

```

```

    FFtot(a,b)=FFtot(a-1)+FF(a,b);

```

```

    a=a+1;

```

```

end

```

```

%Graphs

```

```

figure (19)

```

```

plot(ts,Gears,'m')

```

```

title('GEAR SHIFT')
xlabel('TIME (SECONDS)')
ylabel('GEARS')

figure(20)
plot(ts,FF,'m')
title('FUEL FLOW')
xlabel('TIME (SECONDS)')
ylabel('FUEL FLOW (G/S)')

figure (21)
plot(ts,BMEP,'m')
title('BMEP')
xlabel('TIME (SECONDS)')
ylabel('BMEP (BAR)')

figure (22)
plot(ts,RPM,'m')
title('RPM')
xlabel('TIME (SECONDS)')
ylabel('RPM (REVS/MIN)')

figure (23)
plot(ts,FFtot,'m')
title('Cumulative Fuel Flow')
xlabel('TIME (SECONDS)')
ylabel('FUEL FLOW (G)')

figure (24)
plot(ts,Pw,'m')
title('Power Used')
xlabel('TIME (SECONDS)')
ylabel('POWER (KW)')

% Fuel Total (grams)
FTot=sum(FF) %#ok<NOPTS>
MAXPw=max(Pw) %#ok<NOPTS>
MAXBMEP=max(BMEP) %#ok<NOPTS>
MAXRPM=max(RPM) %#ok<NOPTS>

```

16.3 Appendix C- Combustion Model

16.3.1 Logarithmic Pressure and Volume

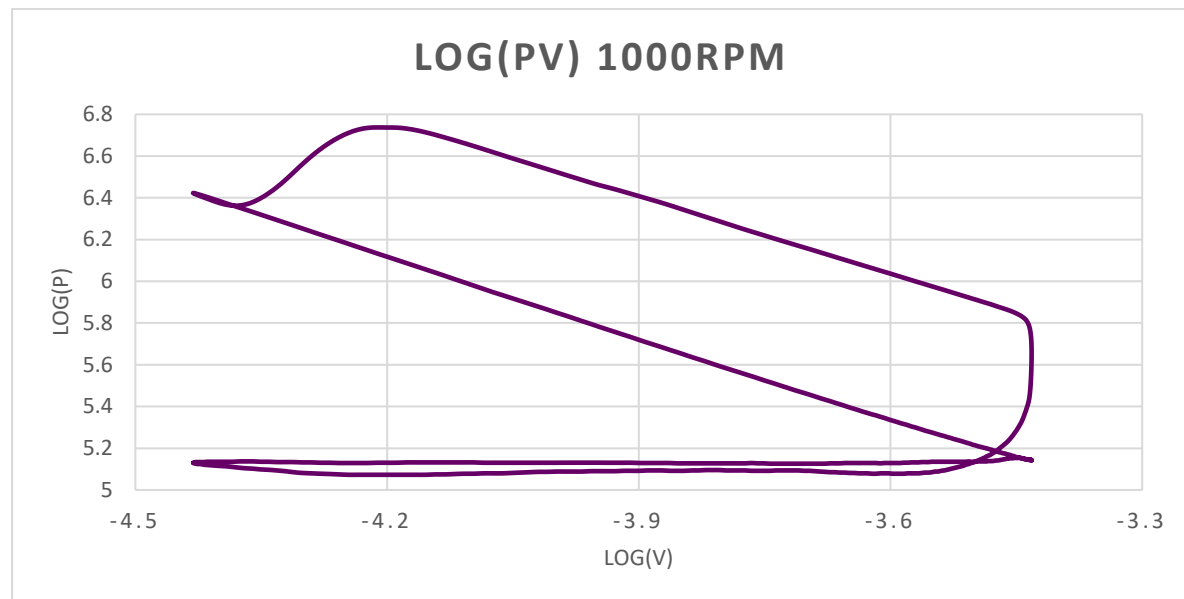


Figure 16-40: Logarithmic Pressure and Volume at 1000RPM

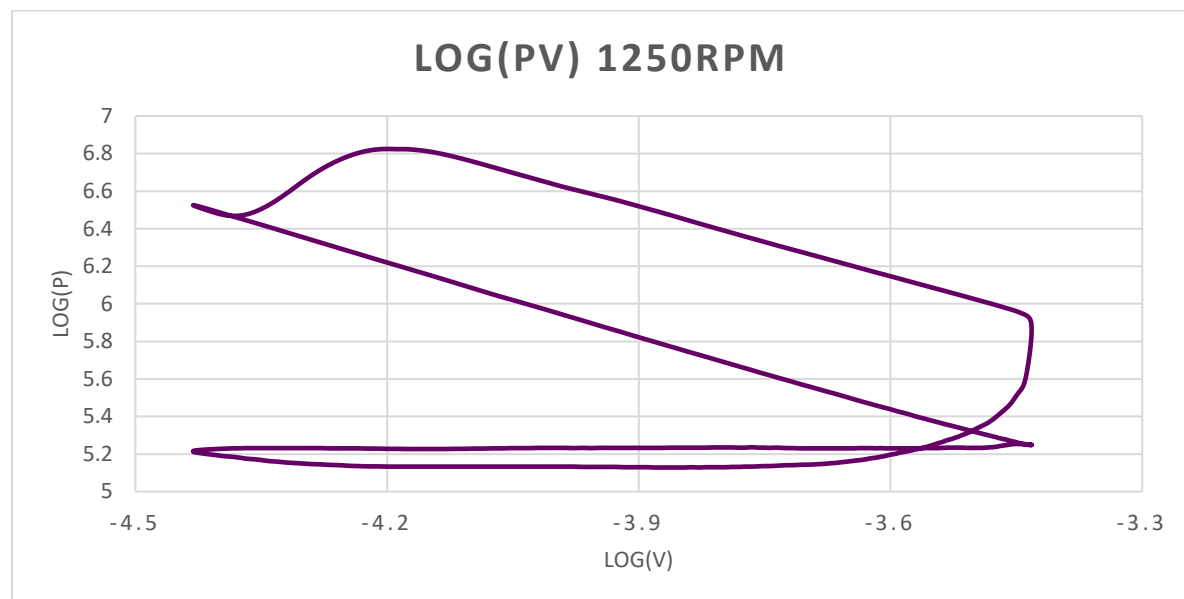


Figure 16-41: Logarithmic Pressure and Volume at 1250RPM

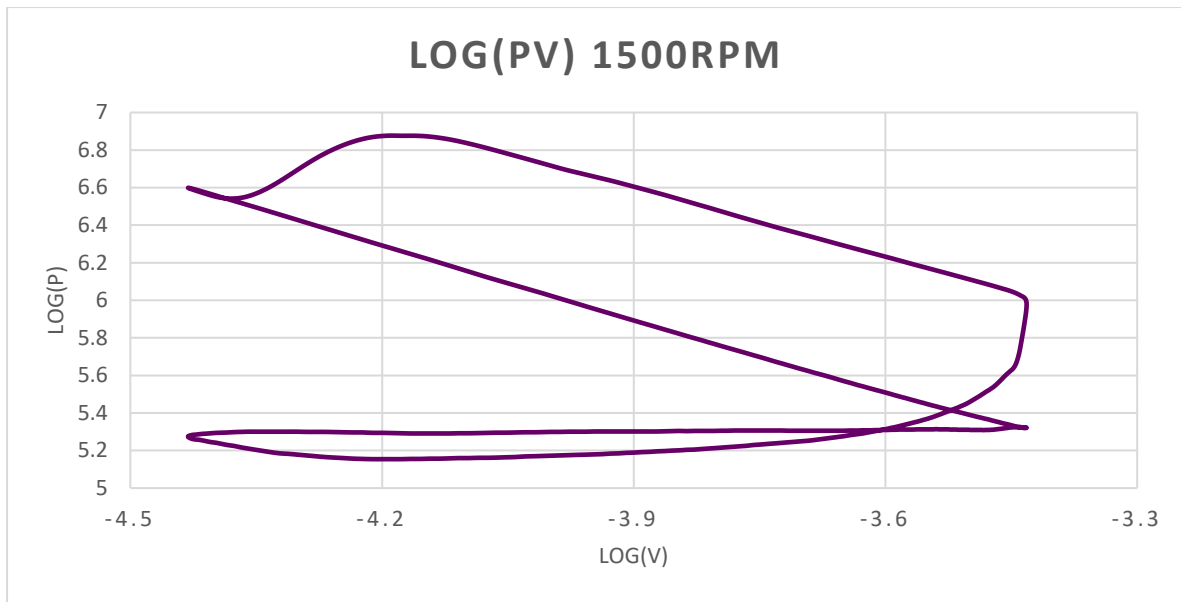


Figure 16-42: Logarithmic Pressure and Volume at 1500RPM

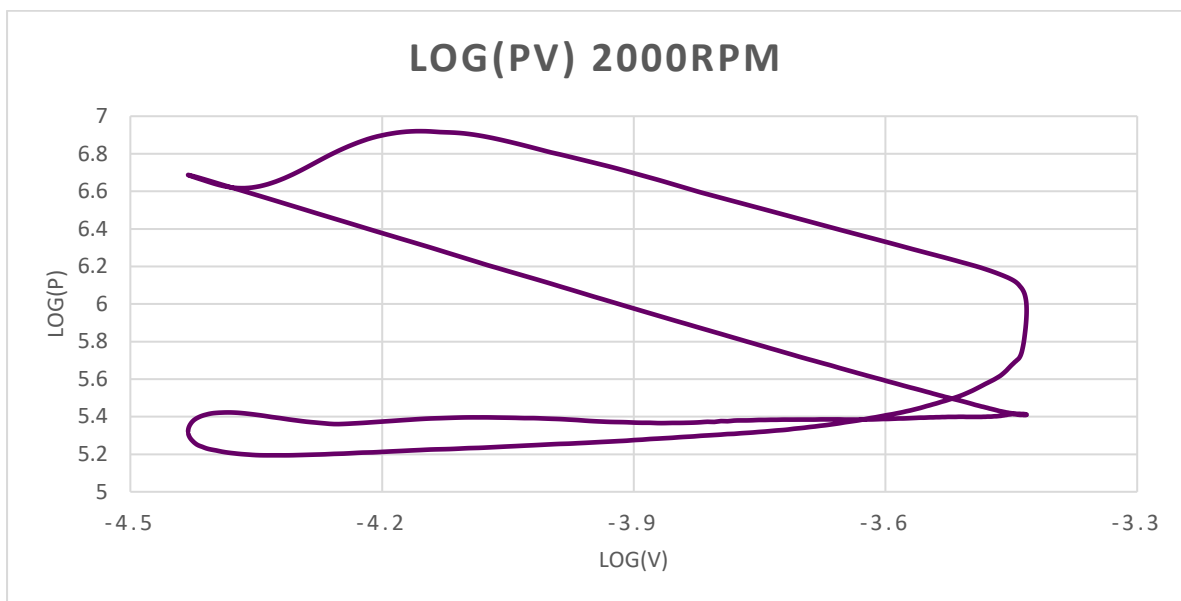


Figure 16-43: Logarithmic Pressure and Volume at 2000RPM

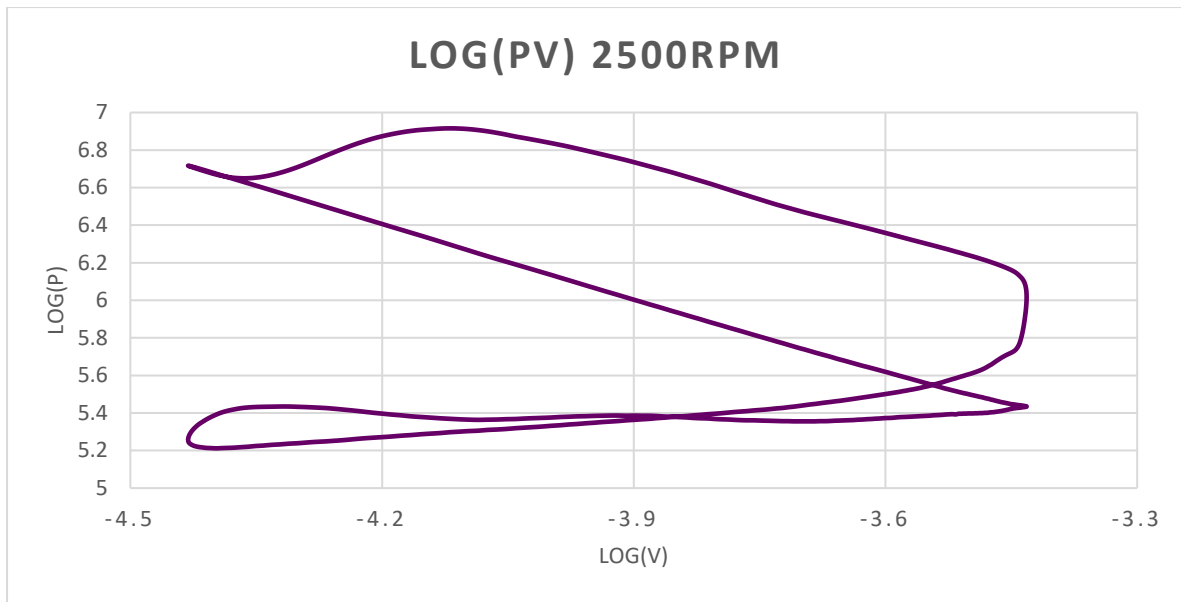


Figure 16-44: Logarithmic Pressure and Volume at 2500RPM

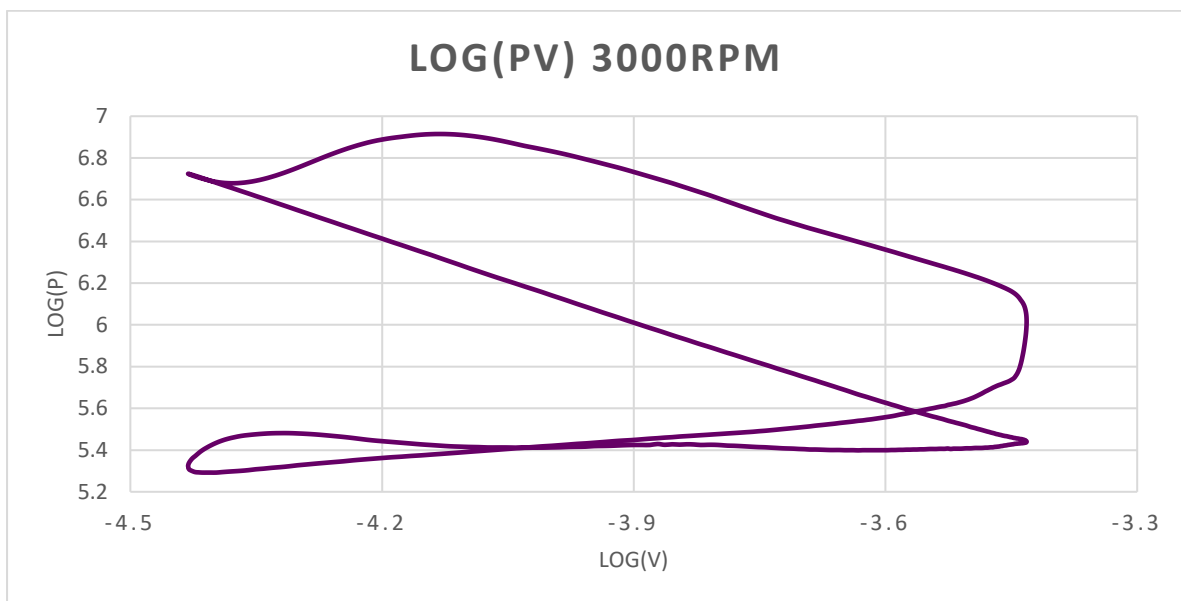


Figure 16-45: Logarithmic Pressure and Volume at 3000RPM

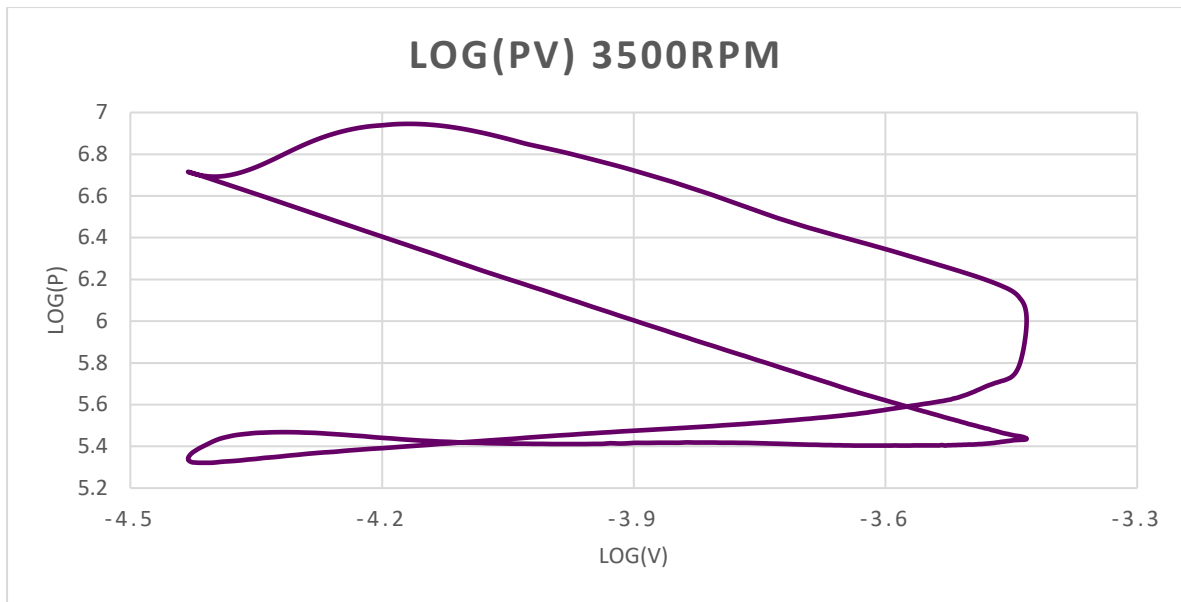


Figure 16-46: Logarithmic Pressure and Volume at 3500RPM

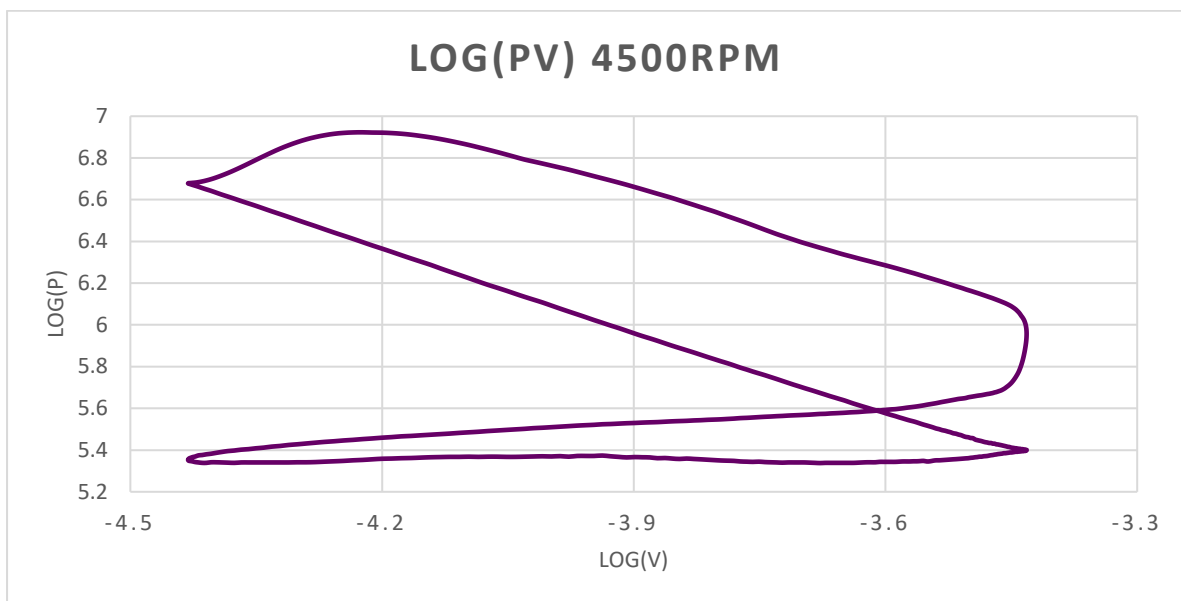


Figure 16-47: Logarithmic Pressure and Volume at 4500RPM

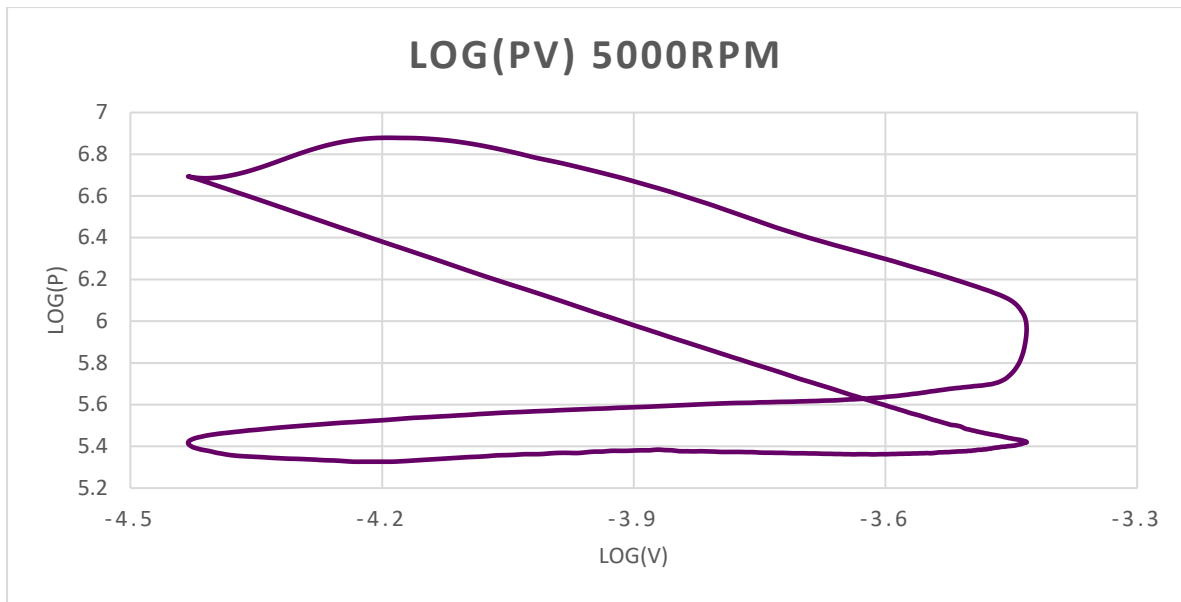


Figure 16-48: Logarithmic Pressure and Volume at 5000RPM

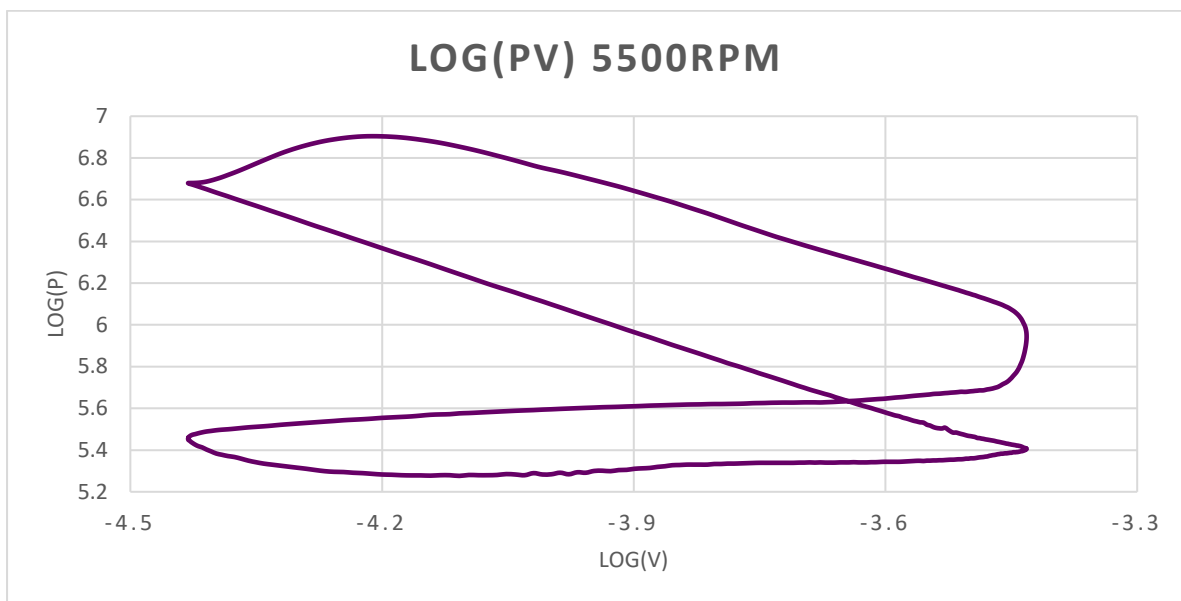


Figure 16-49: Logarithmic Pressure and Volume at 5500RPM

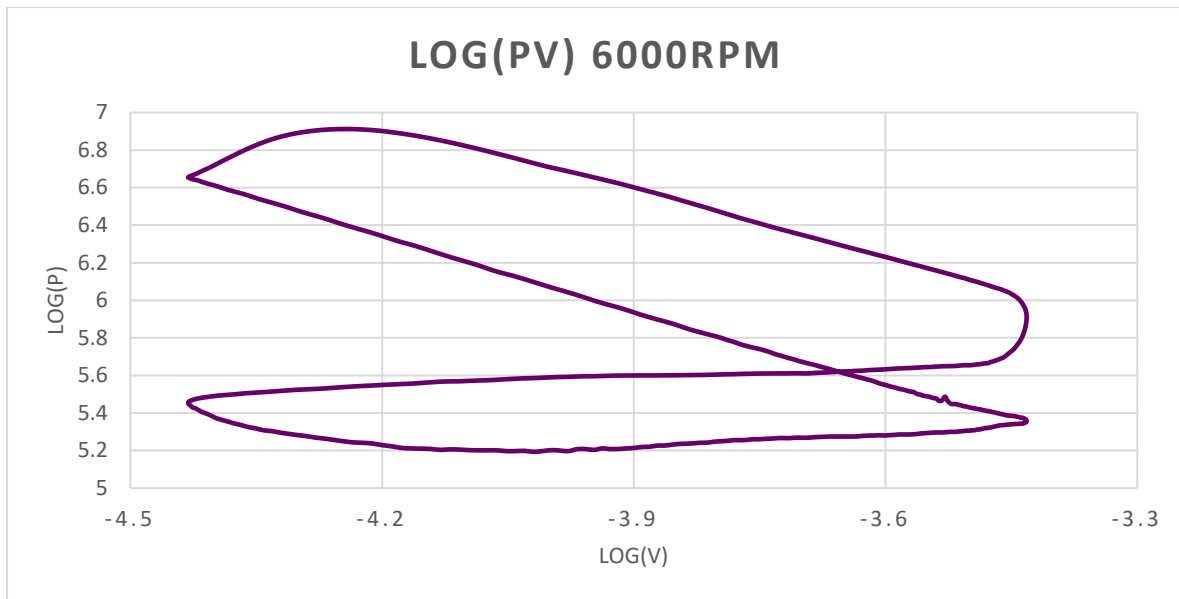


Figure 16-50: Logarithmic Pressure and Volume at 6000RPM

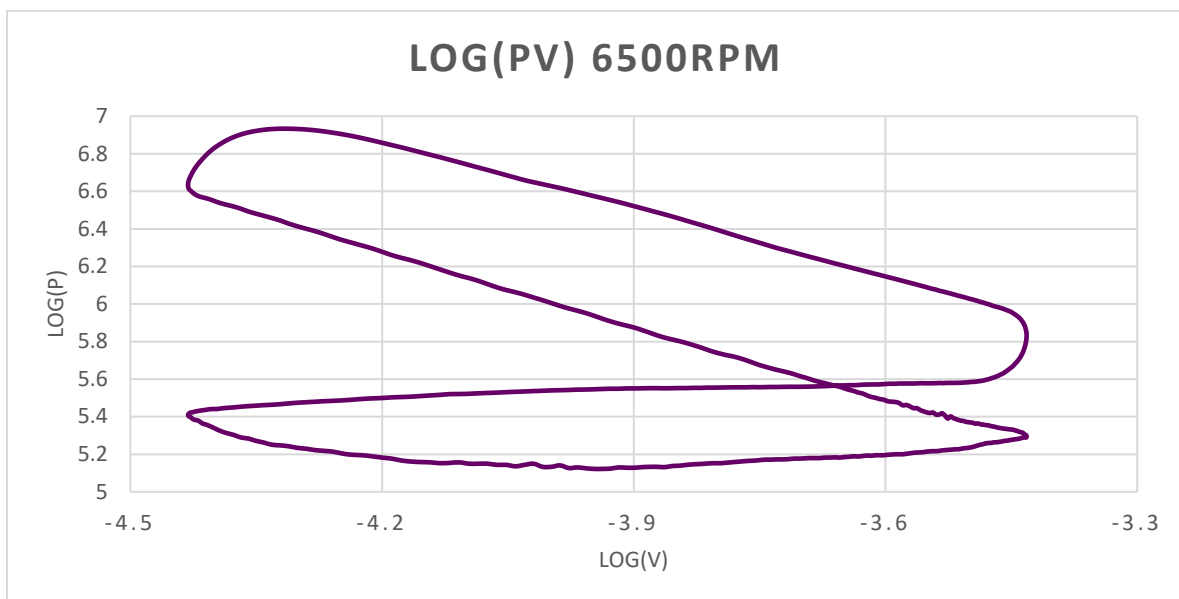


Figure 16-51: Logarithmic Pressure and Volume at 6500RPM

16.3.2 Extra Pressure From Combustion

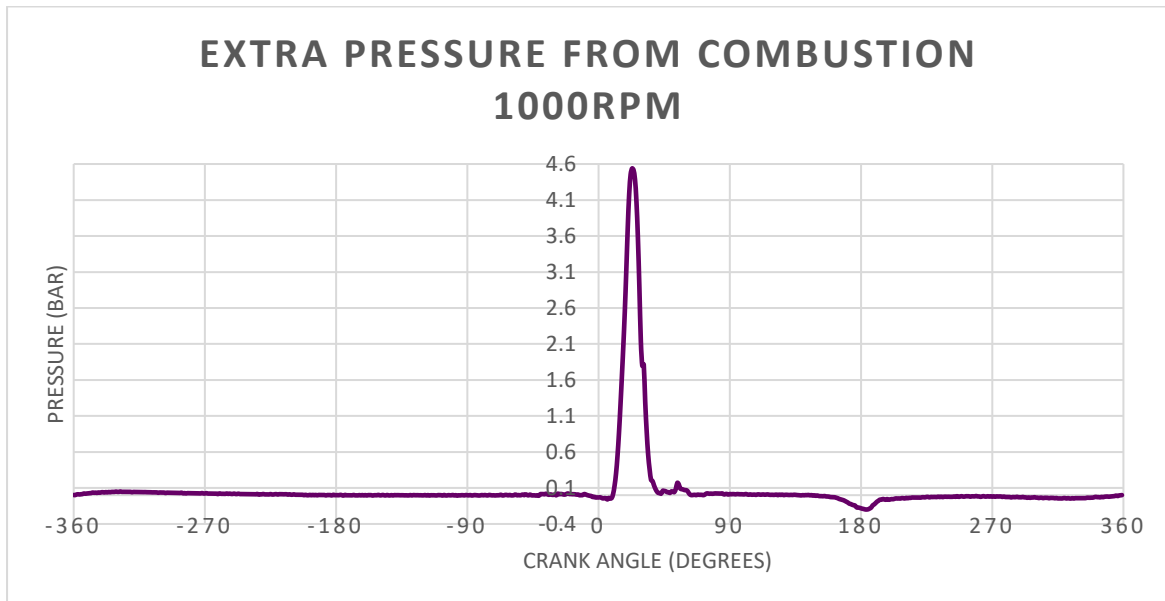


Figure 16-52: Extra Pressure from Combustion at 1000RPM

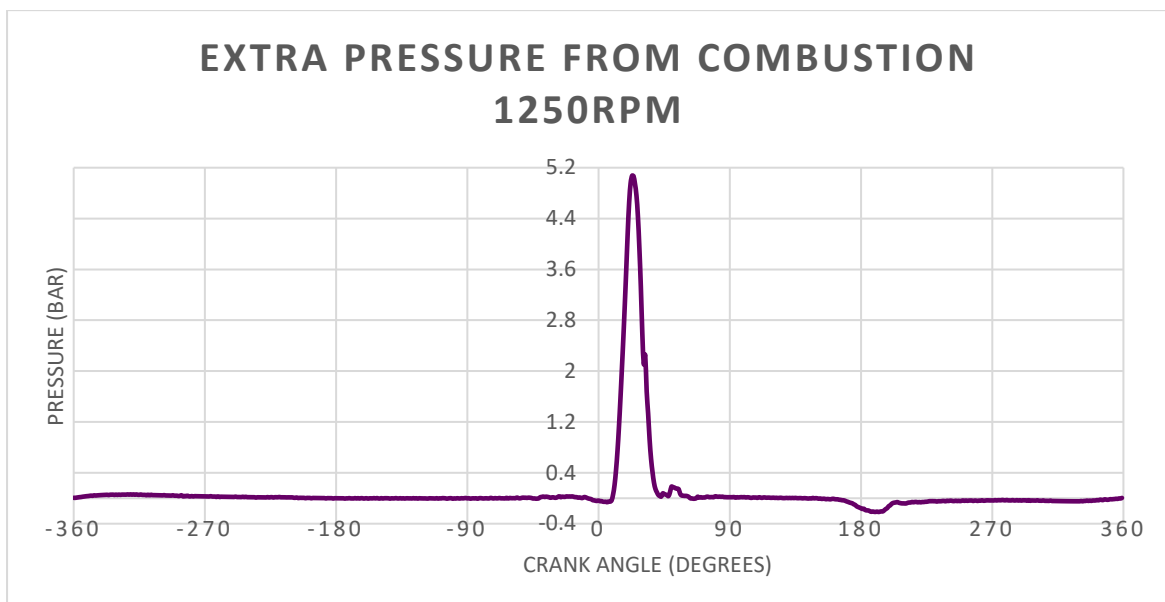


Figure 16-53: Extra Pressure from Combustion at 1250RPM

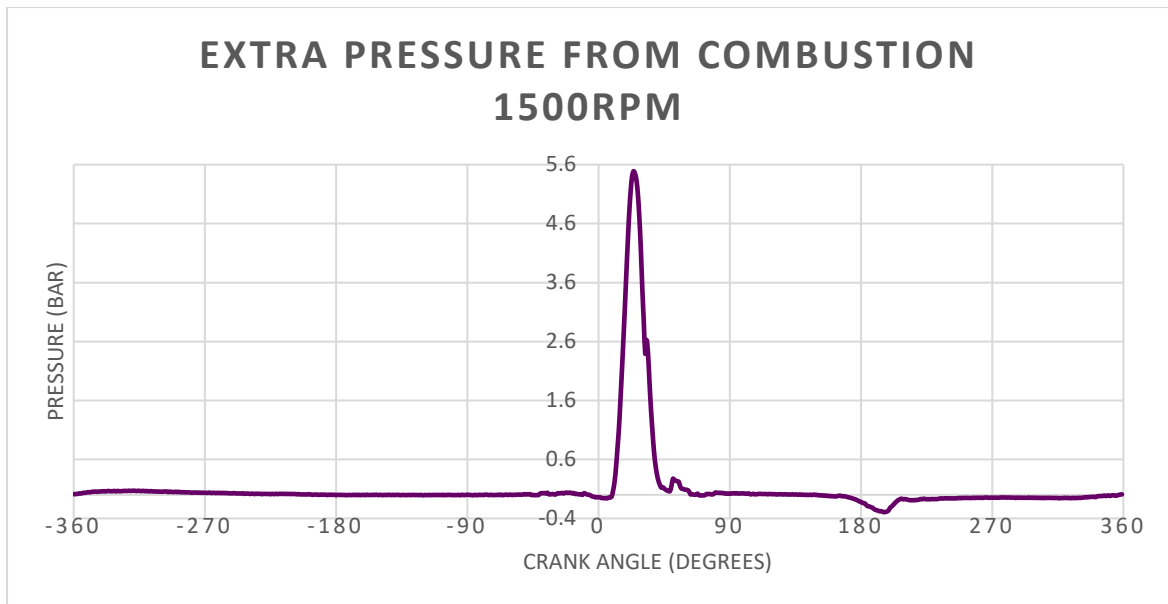


Figure 16-54: Extra Pressure from Combustion at 1500RPM

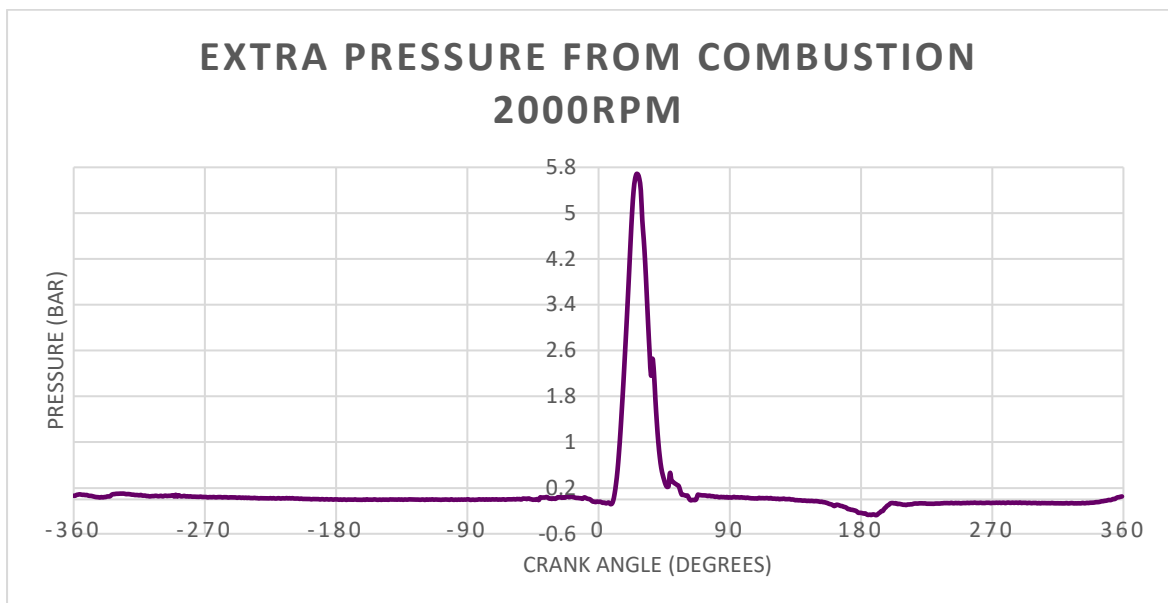


Figure 16-55: Extra Pressure from Combustion at 2000RPM

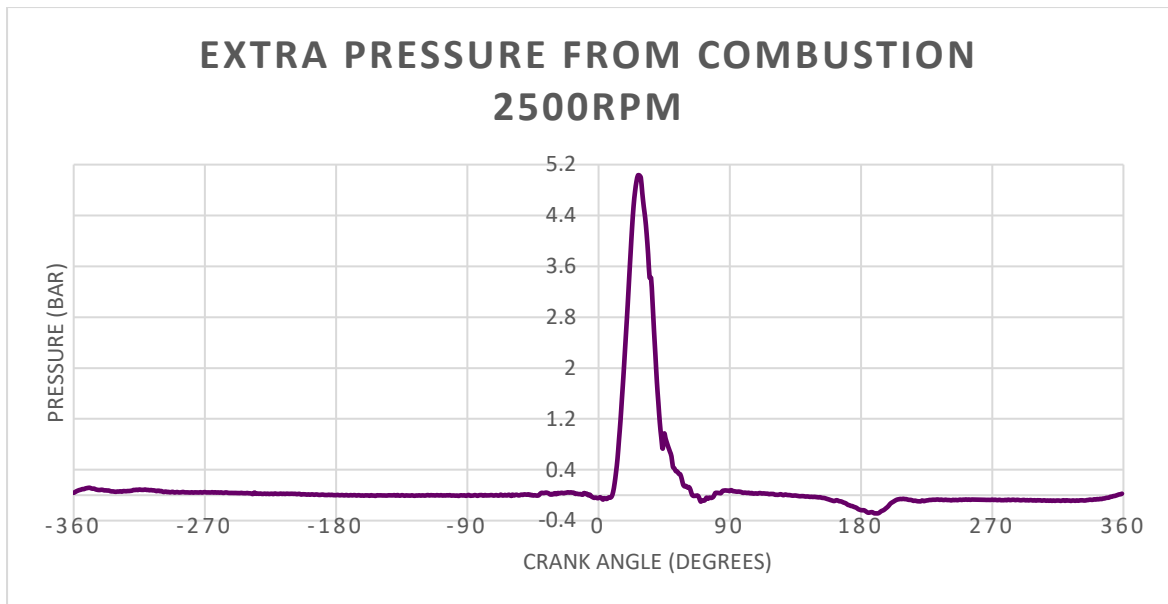


Figure 16-56: Extra Pressure from Combustion at 2500RPM

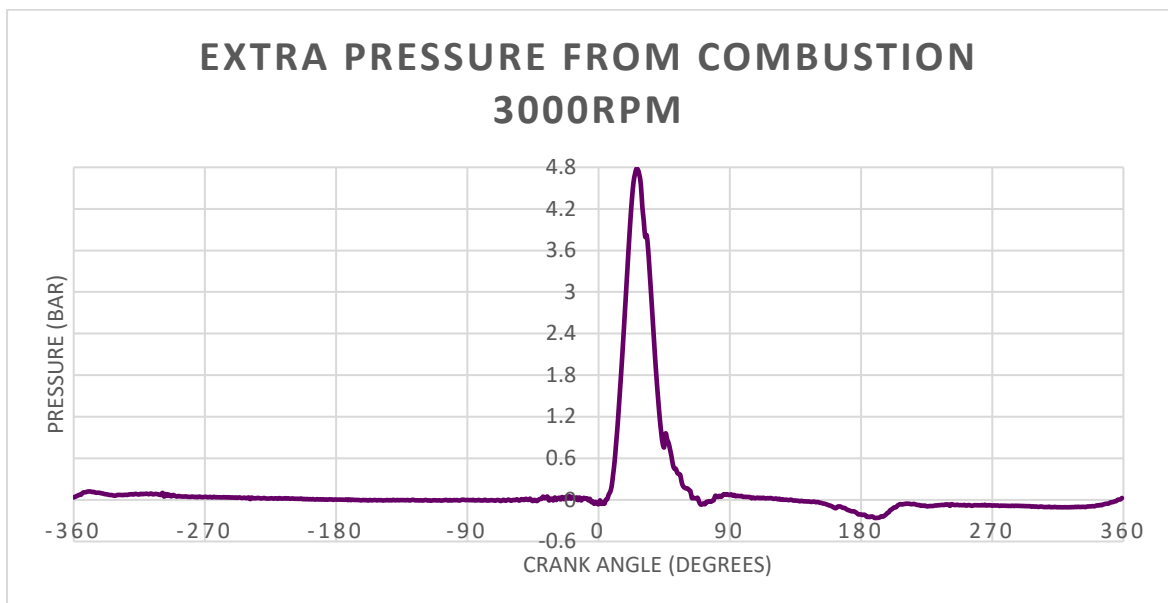


Figure 16-57: Extra Pressure from Combustion at 3000RPM

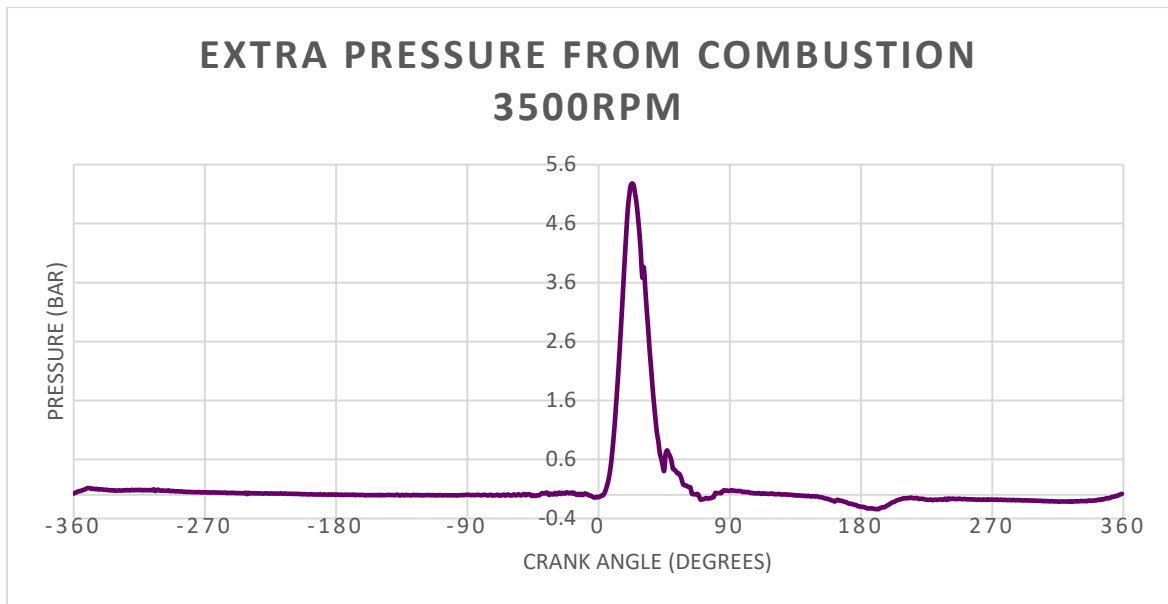


Figure 16-58: Extra Pressure from Combustion at 3500RPM

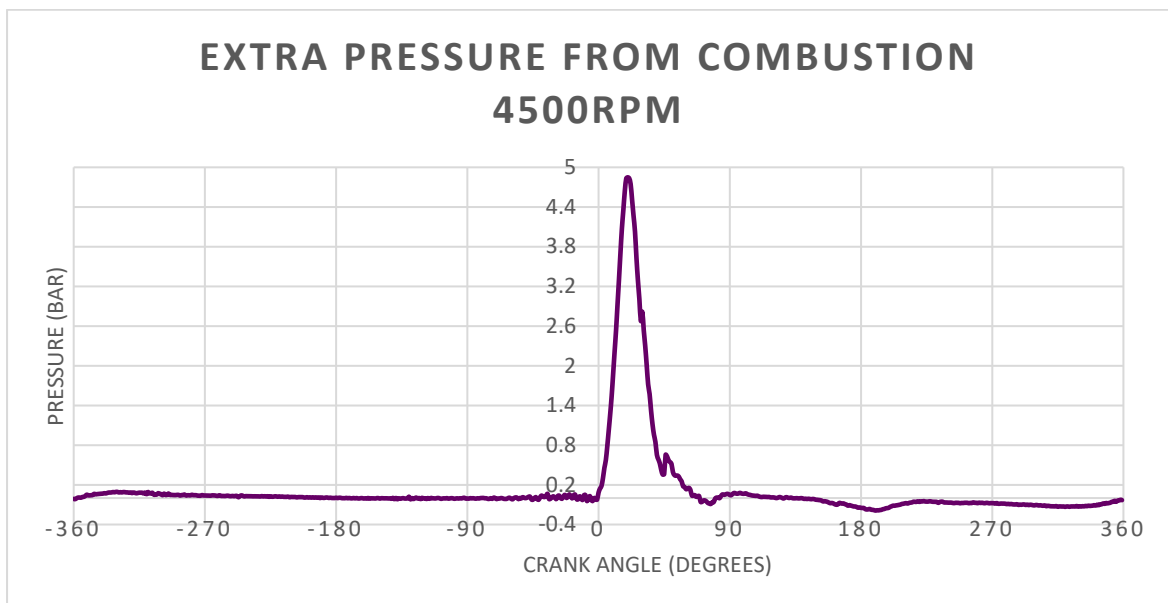


Figure 16-59: Extra Pressure from Combustion at 4500RPM

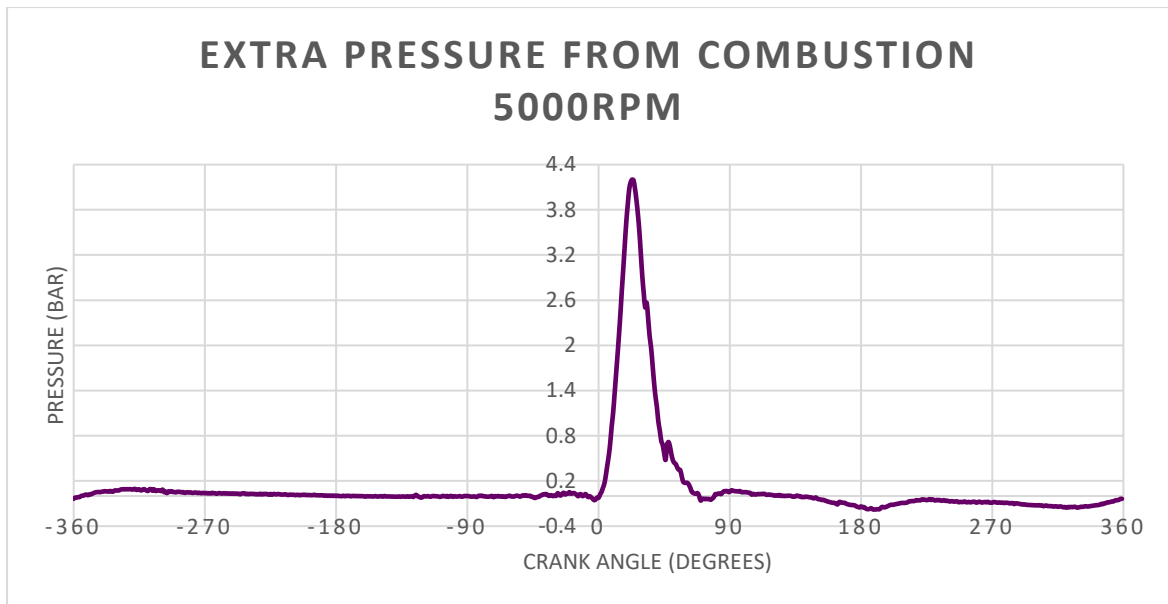


Figure 16-60: Extra Pressure from Combustion at 5000RPM

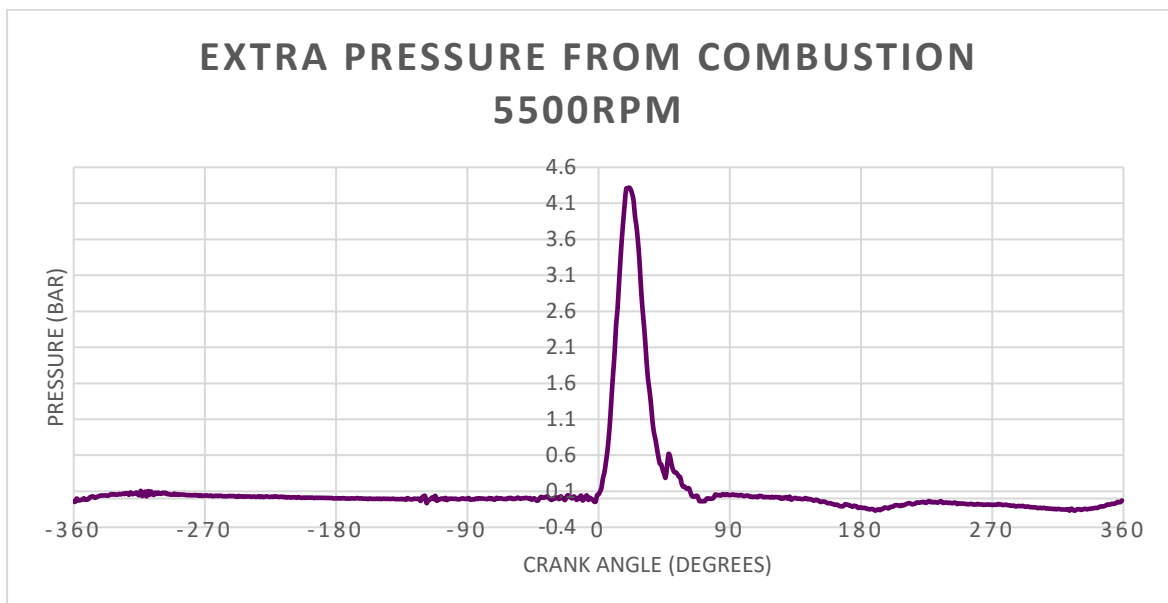


Figure 16-61: Extra Pressure from Combustion at 5500RPM

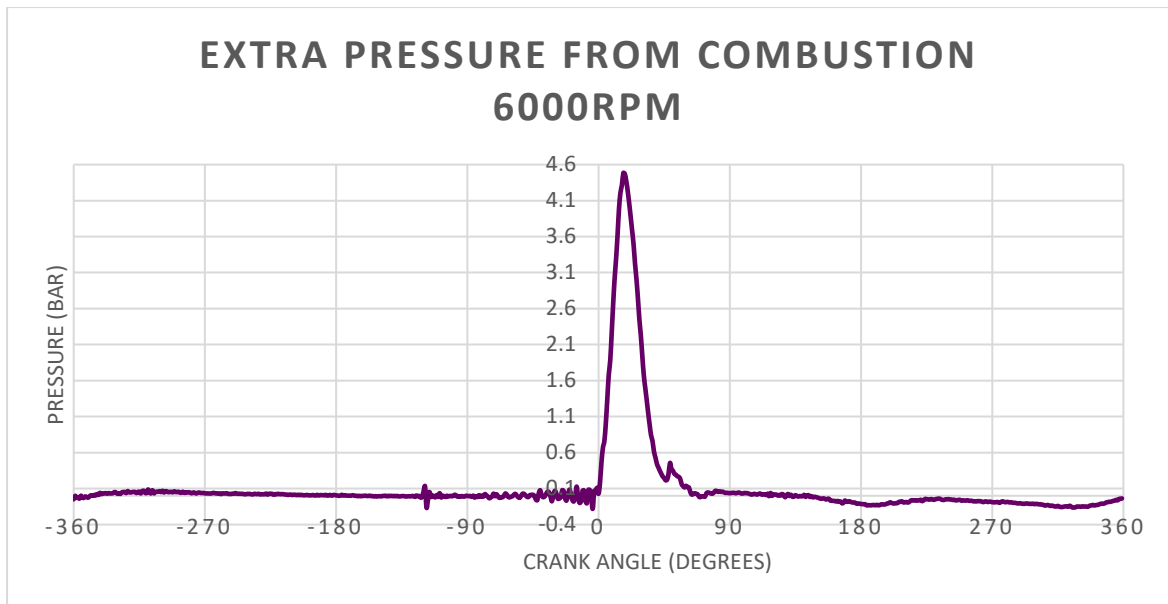


Figure 16-62: Extra Pressure from Combustion at 6000RPM

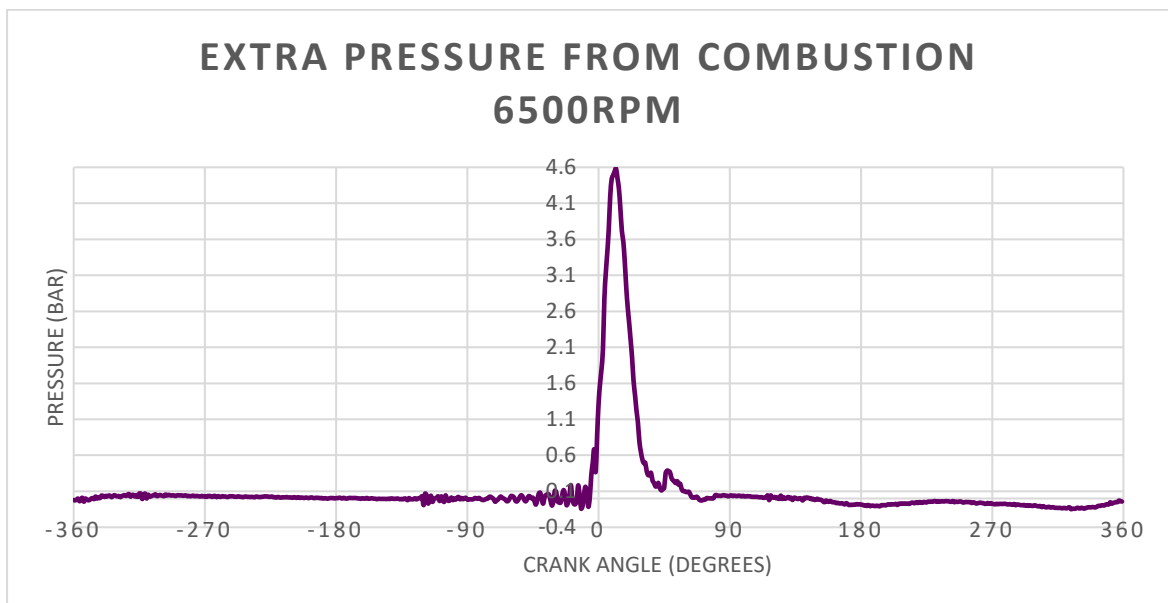


Figure 16-63: Extra Pressure from Combustion at 6500RPM

16.3.3 MFB

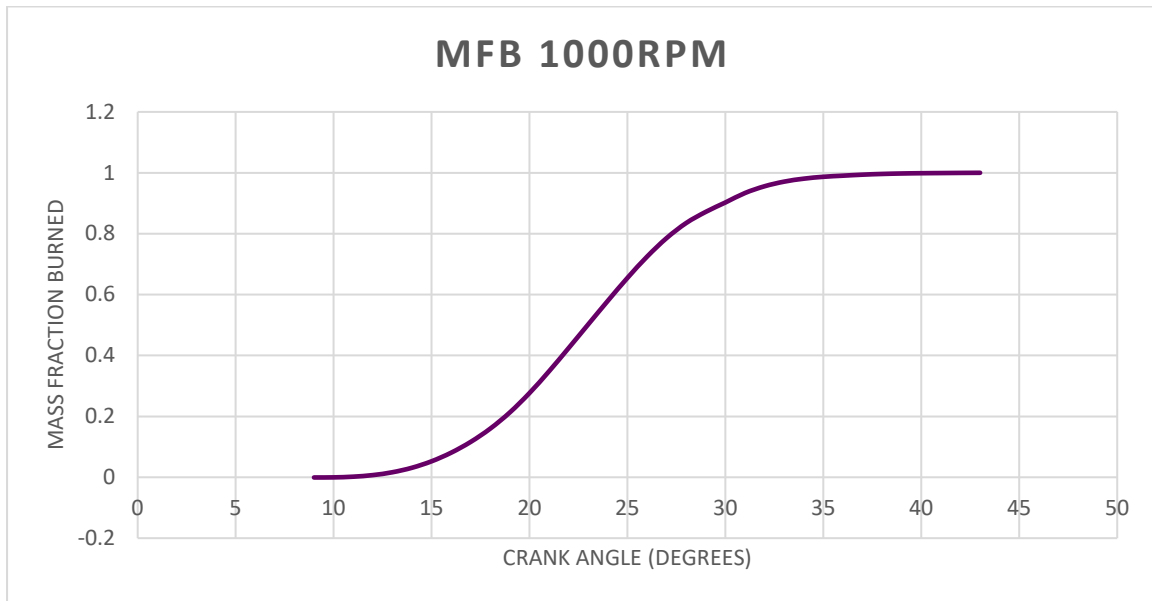


Figure 16-64: Mass Fraction Burned at 1000RPM

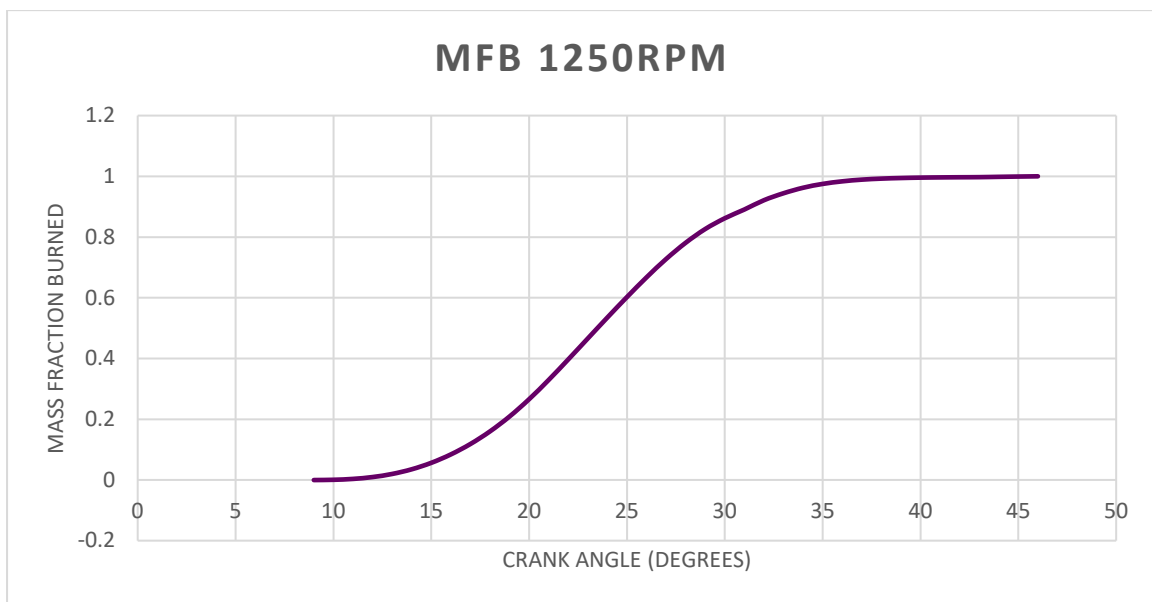


Figure 16-65: Mass Fraction Burned at 1250RPM

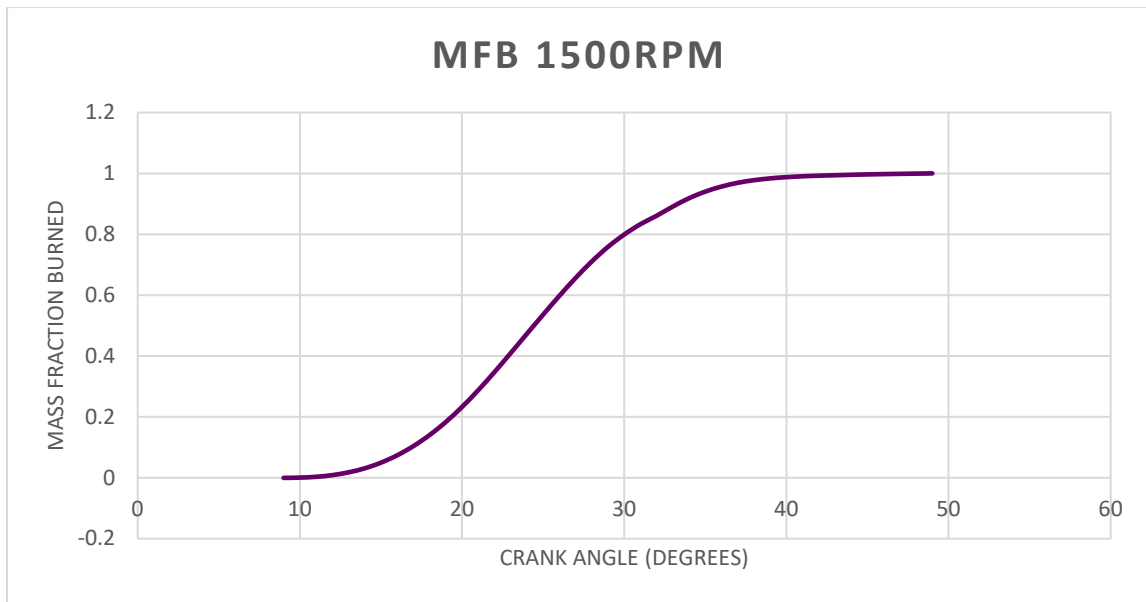


Figure 16-66: Mass Fraction Burned at 1500RPM

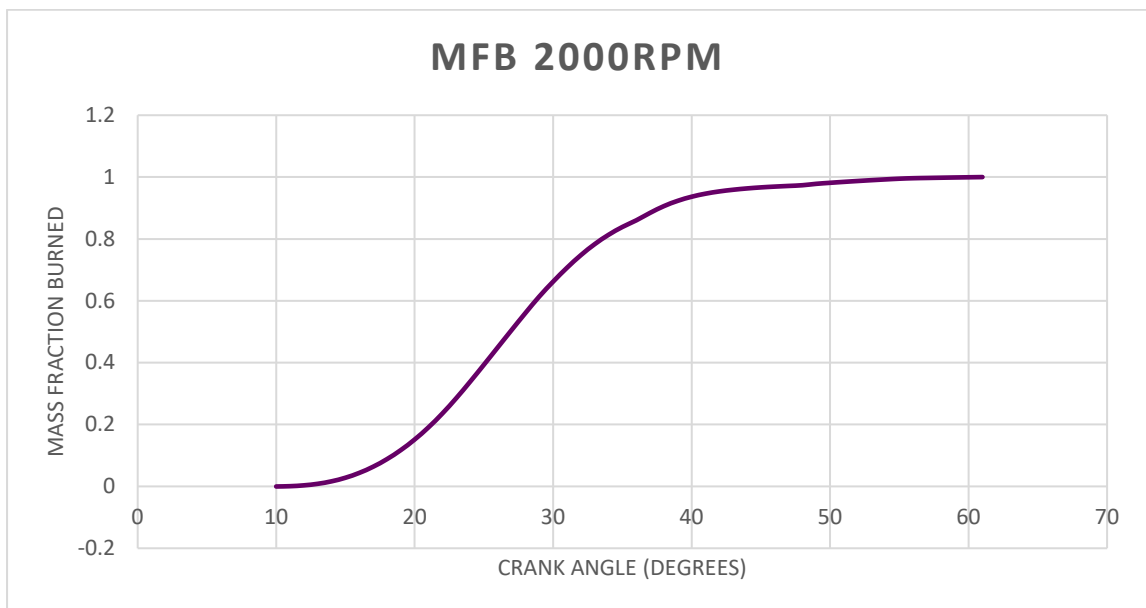


Figure 16-67: Mass Fraction Burned at 2000RPM

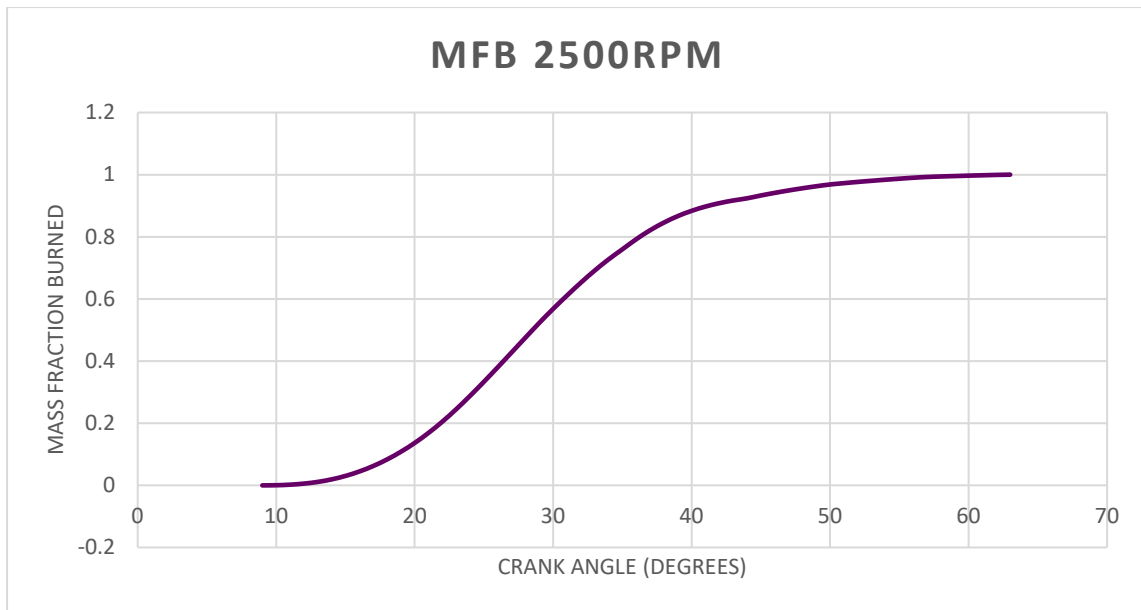


Figure 16-68: Mass Fraction Burned at 2500RPM

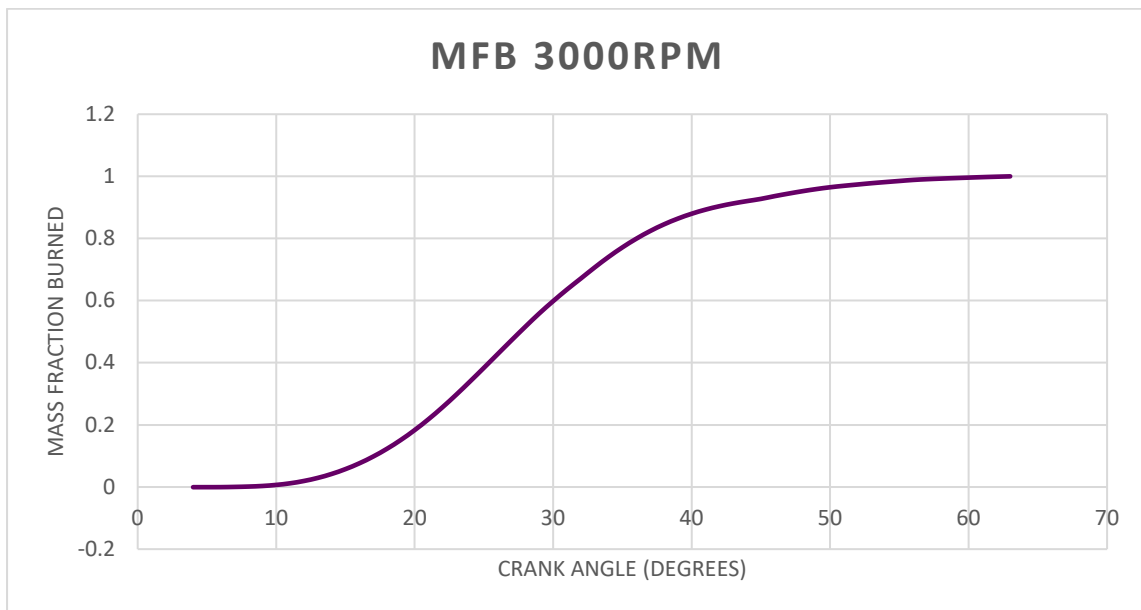


Figure 16-69: Mass Fraction Burned at 3000RPM

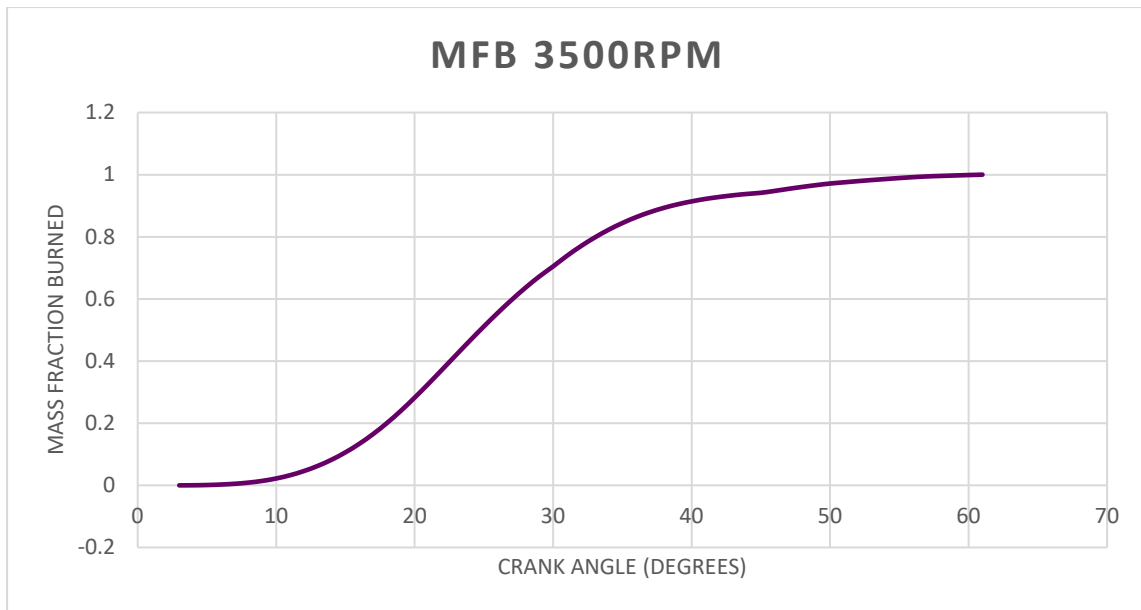


Figure 16-70: Mass Fraction Burned at 3500RPM

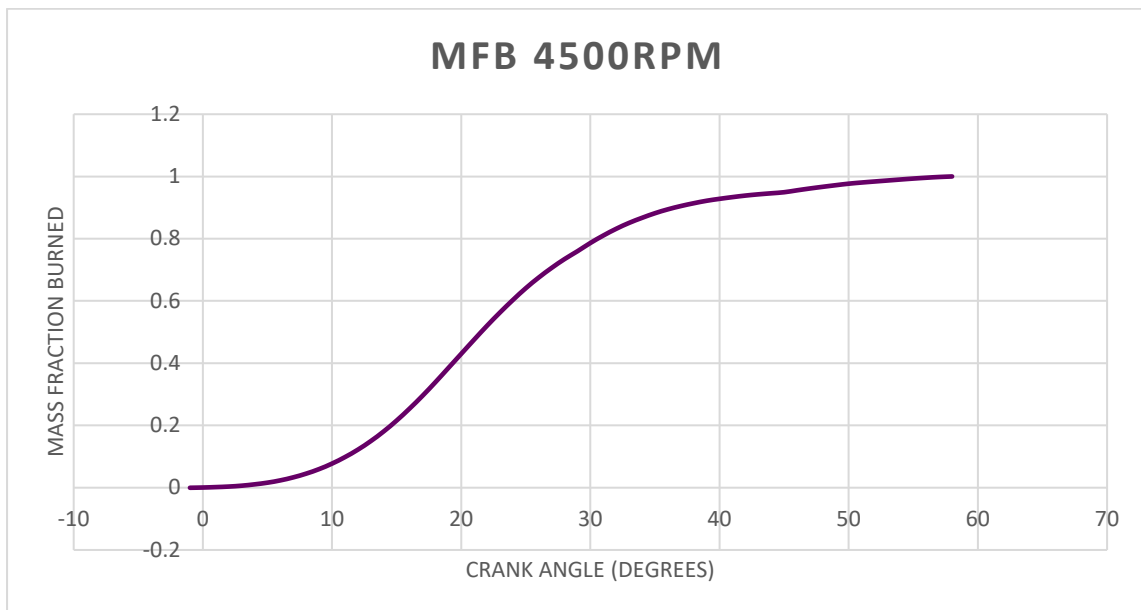


Figure 16-71: Mass Fraction Burned at 4500RPM

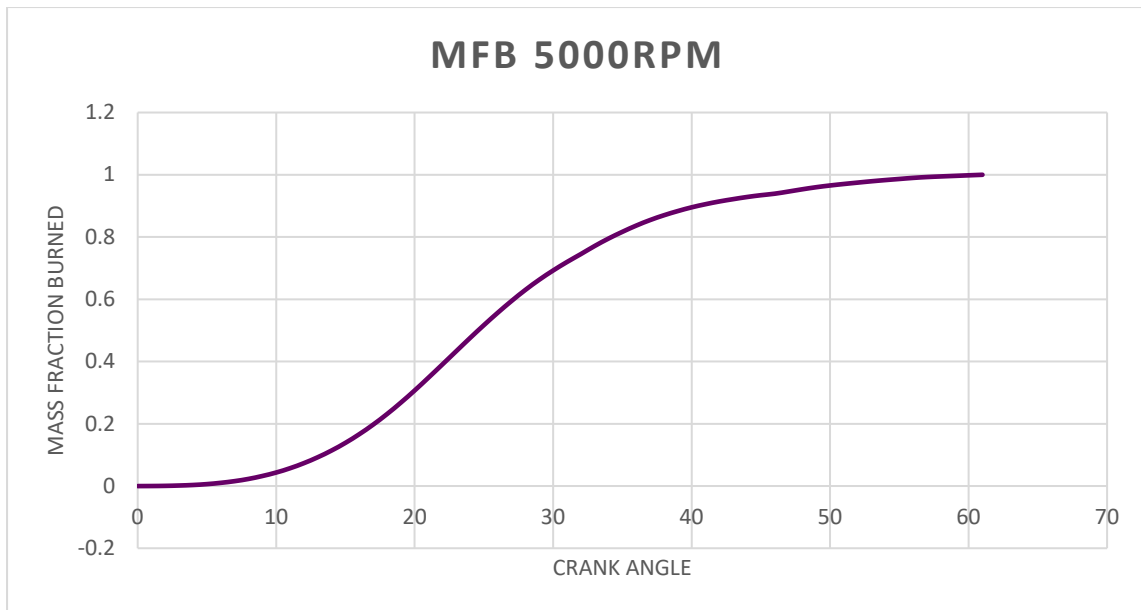


Figure 16-72: Mass Fraction Burned at 5000RPM

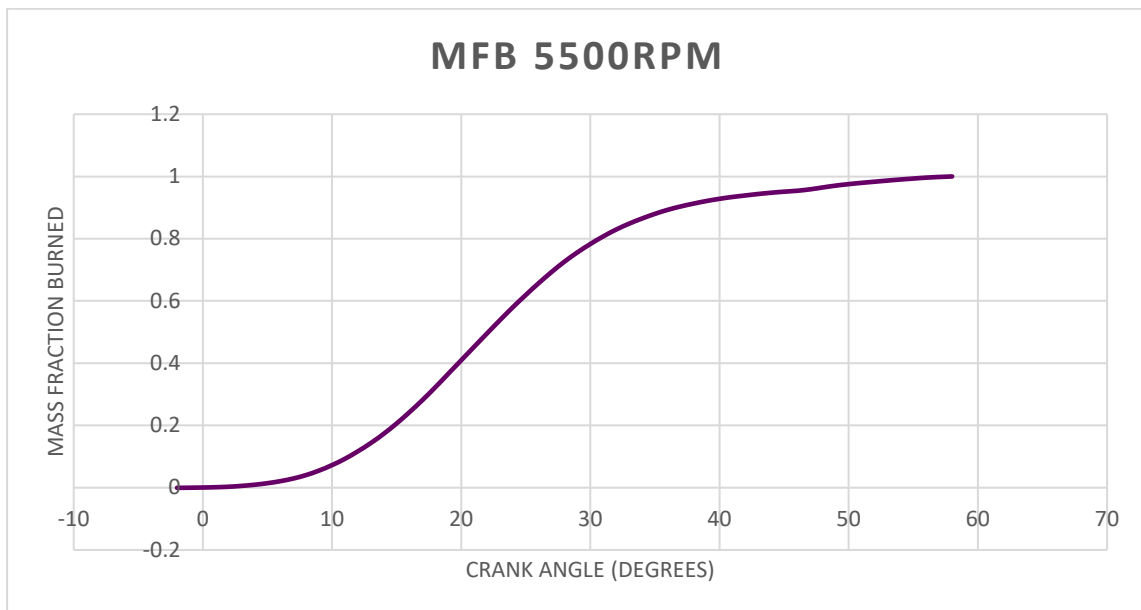


Figure 16-73: Mass Fraction Burned at 5500RPM

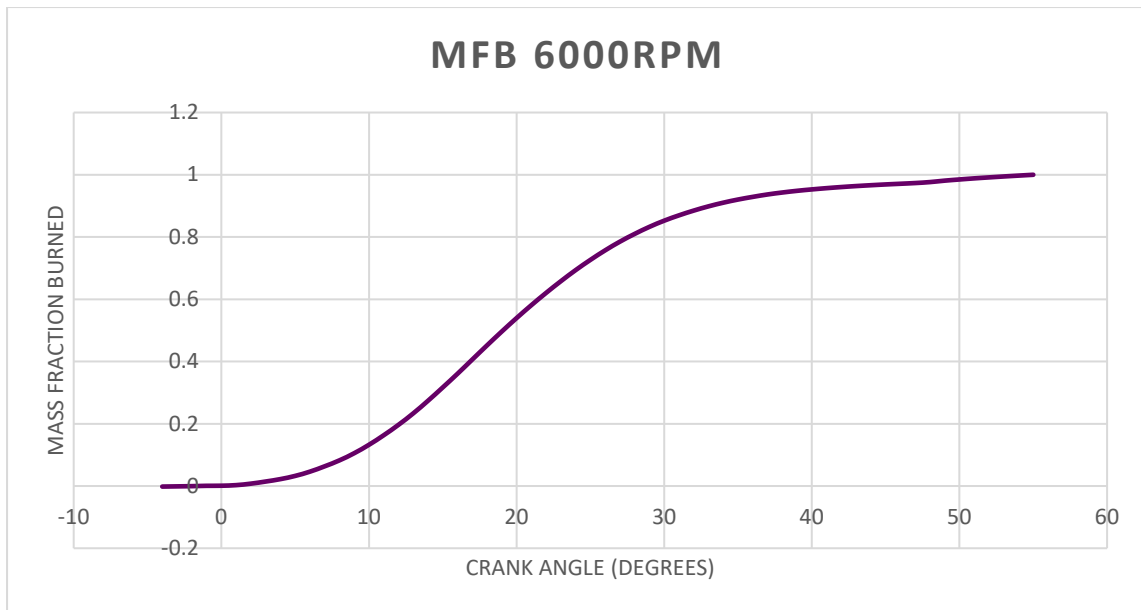


Figure 16-74: Mass Fraction Burned at 6000RPM

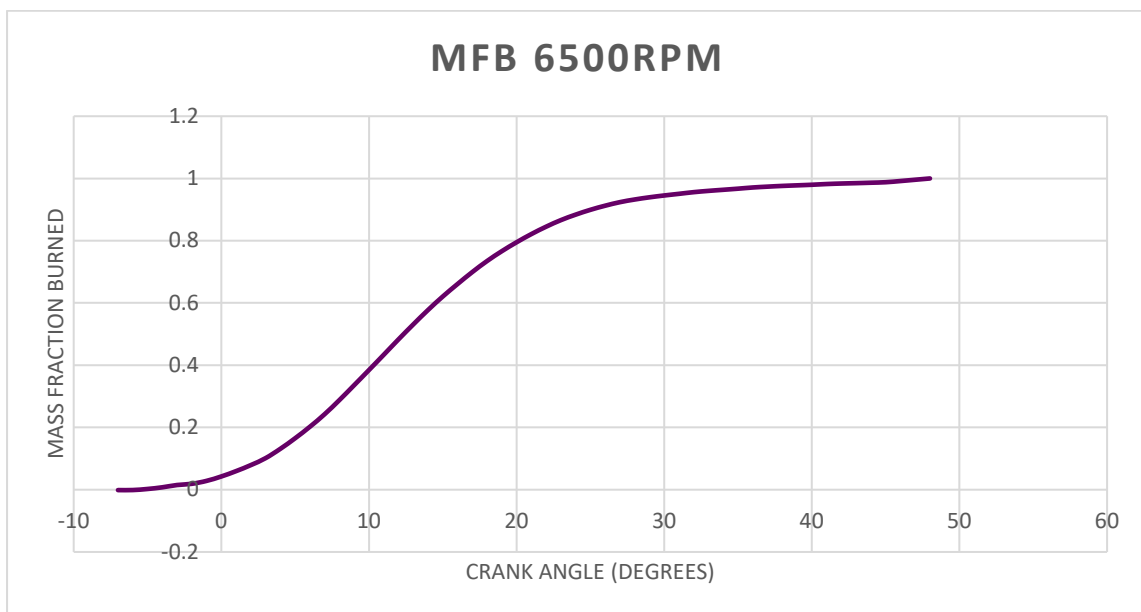


Figure 16-75: Mass Fraction Burned at 6500RPM

16.4 Appendix D – Friction Model

16.4.1 Gas Force

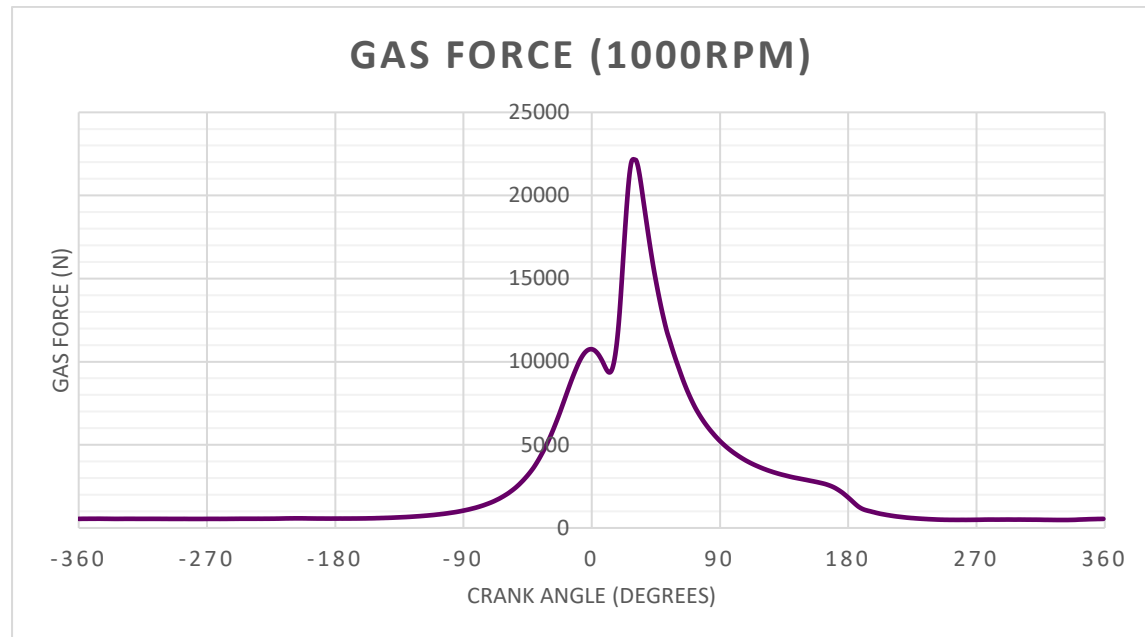


Figure 16-76: Maximum Gas Force at 1000RPM

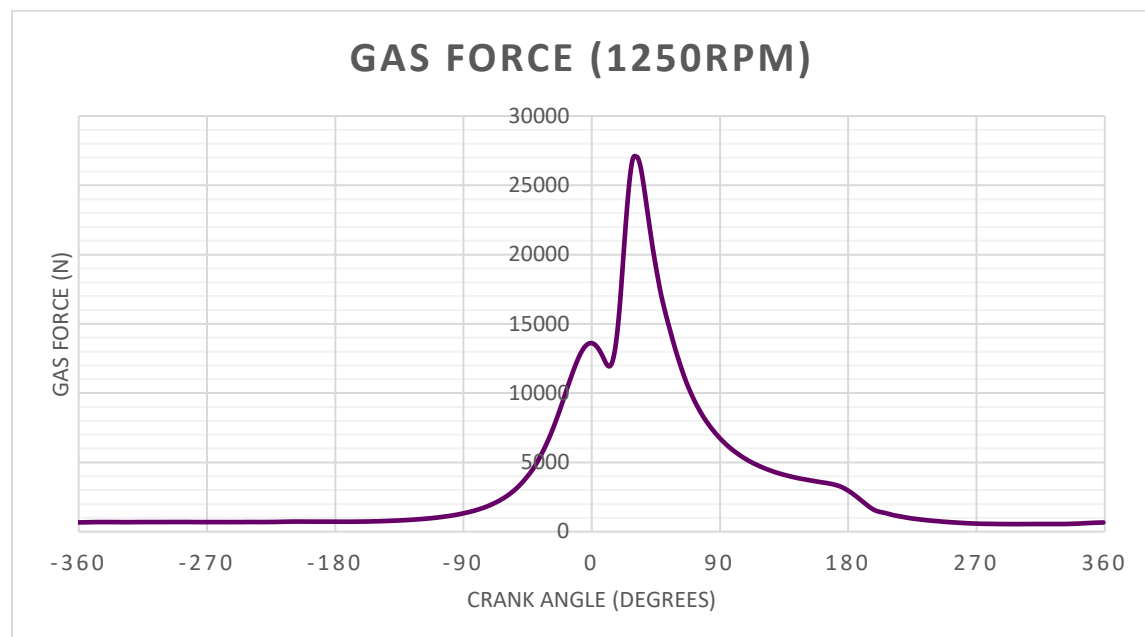


Figure 16-77: Maximum Gas Force at 1250RPM

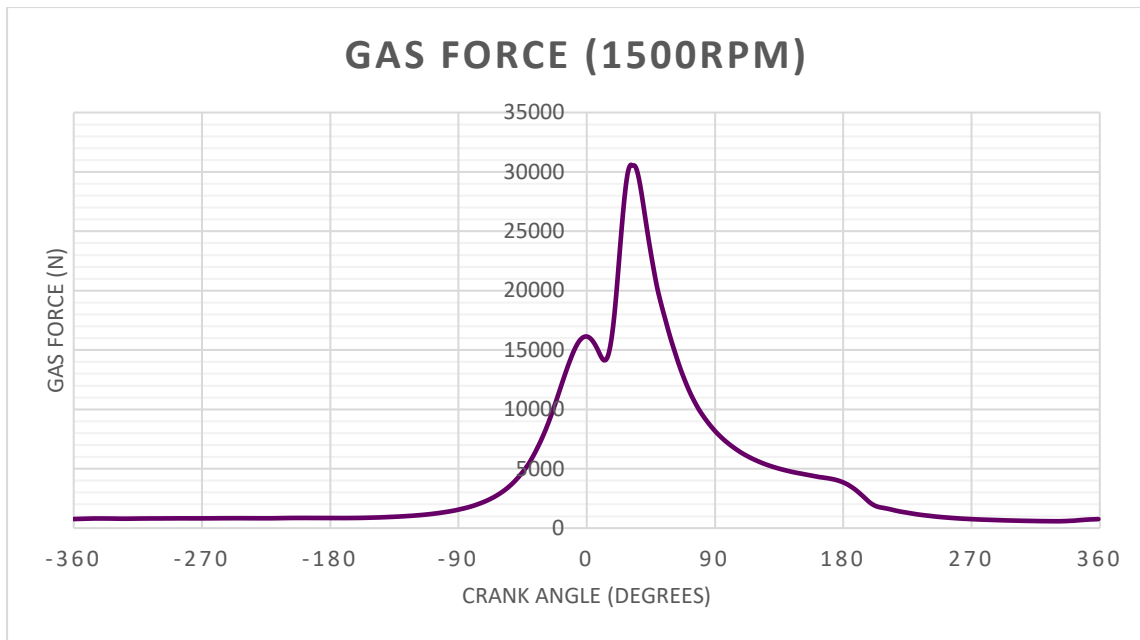


Figure 16-78: Maximum Gas Force at 1500RPM

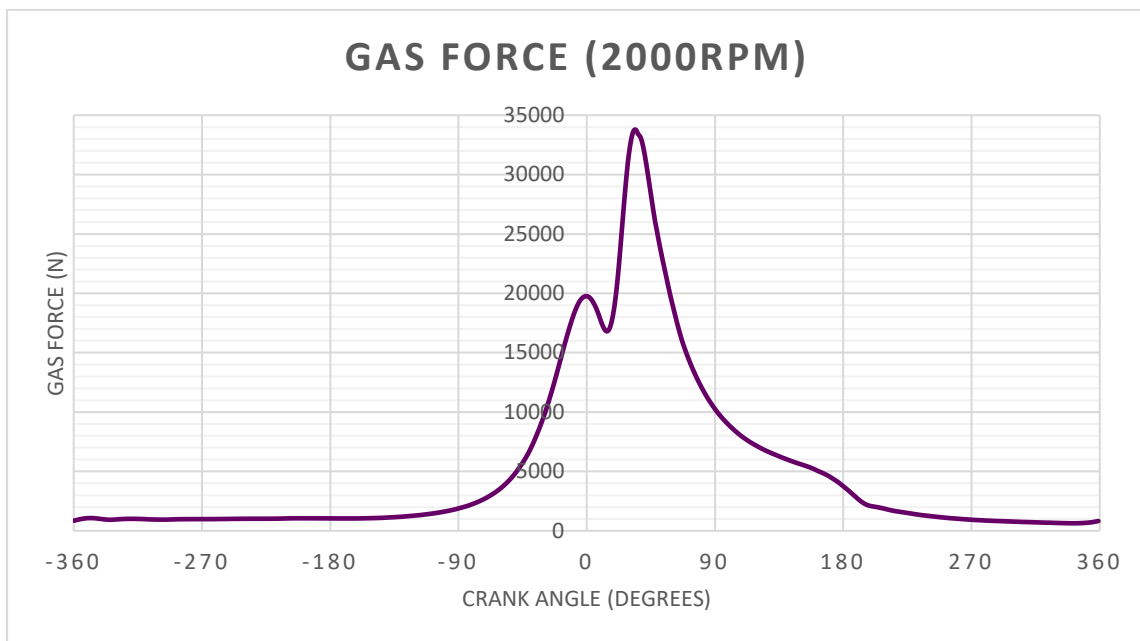


Figure 16-79: Maximum Gas Force at 2000RPM

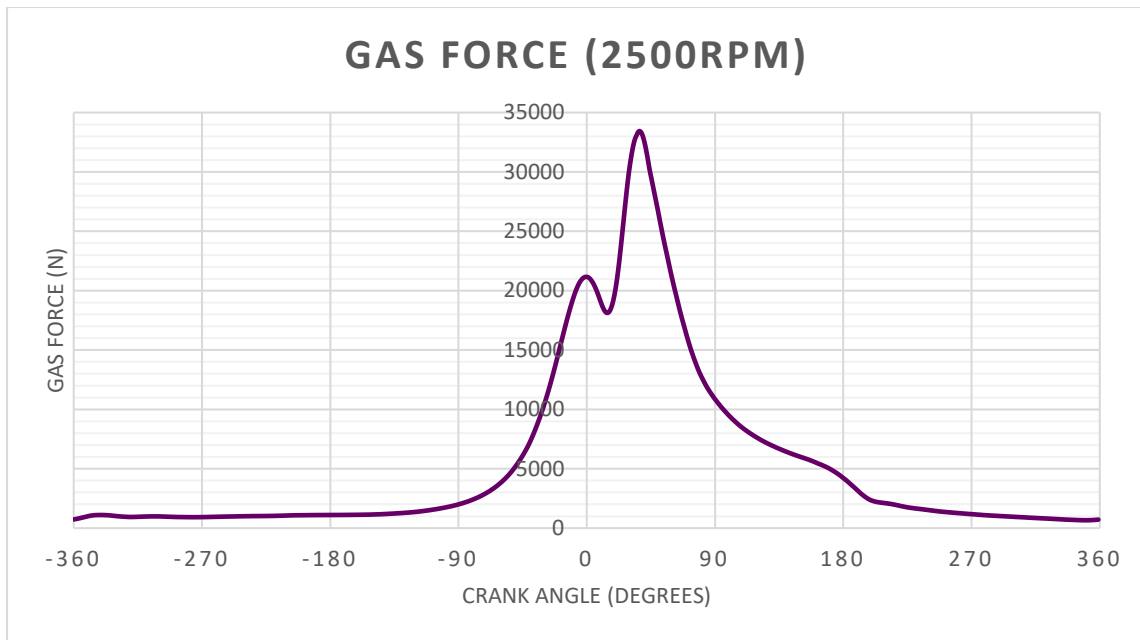


Figure 16-80: Maximum Gas Force at 2500RPM

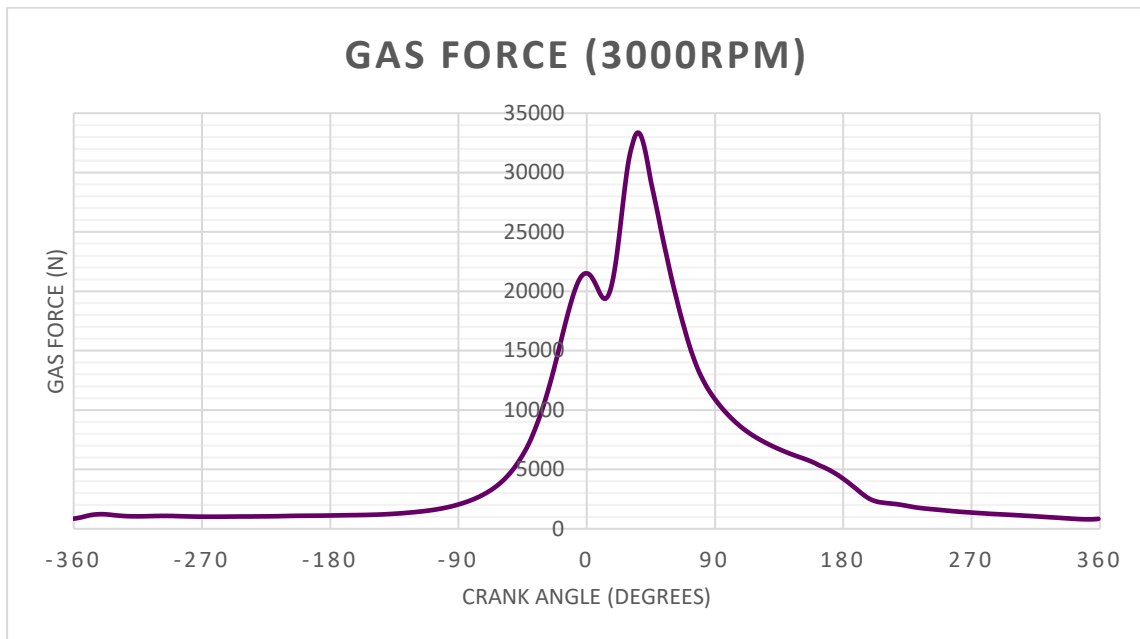


Figure 16-81: Maximum Gas Force at 3000RPM

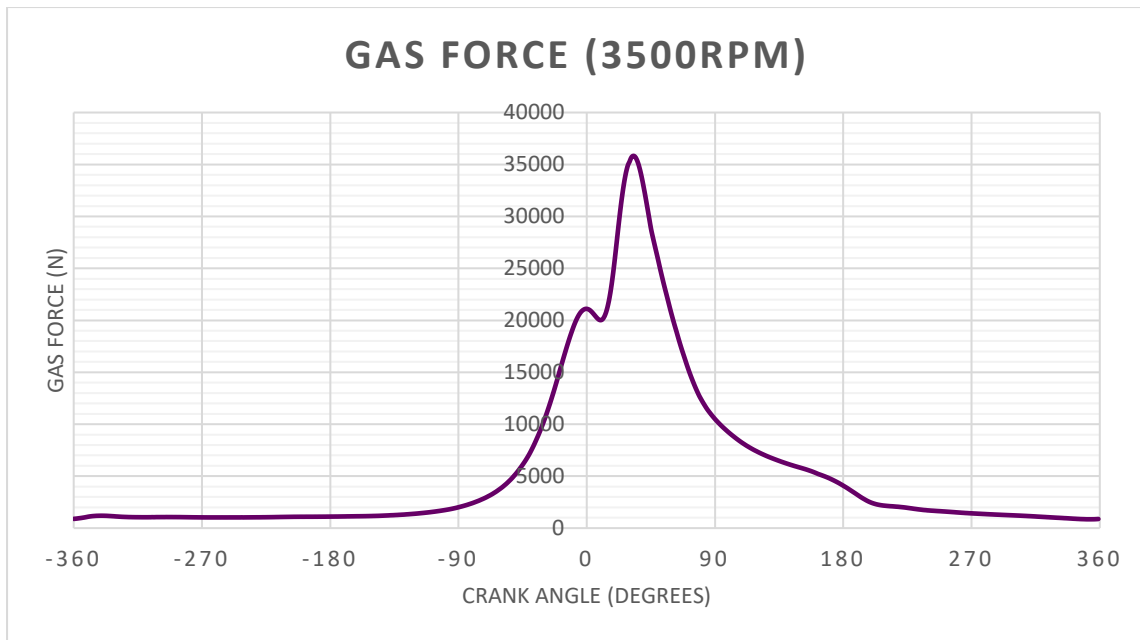


Figure 16-82: Maximum Gas Force at 3500RPM

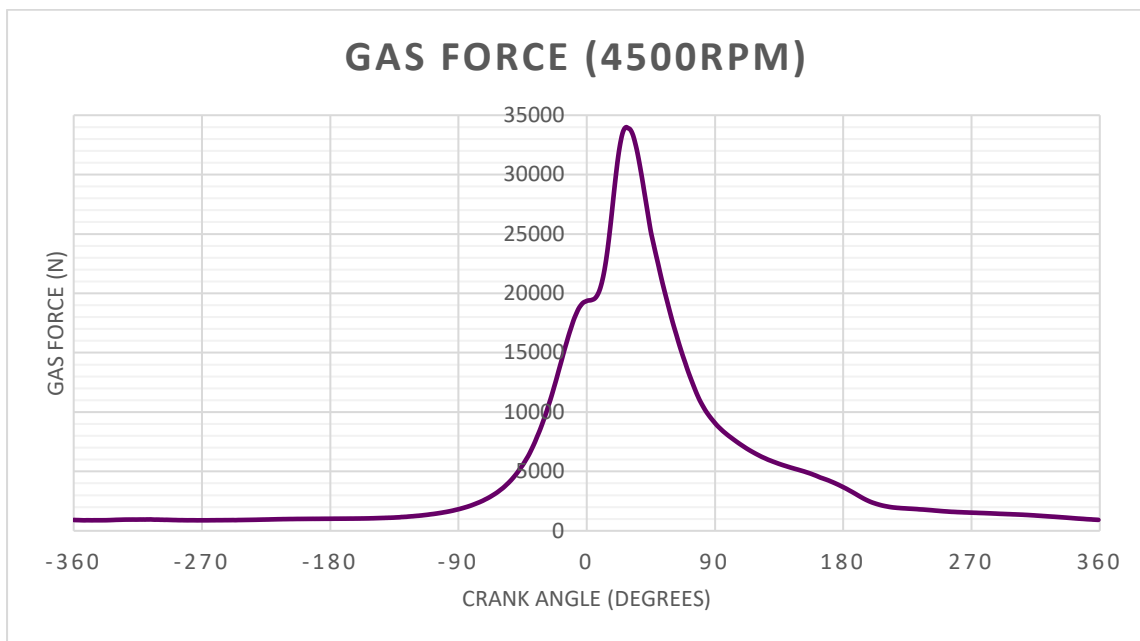


Figure 16-83: Maximum Gas Force at 4500RPM

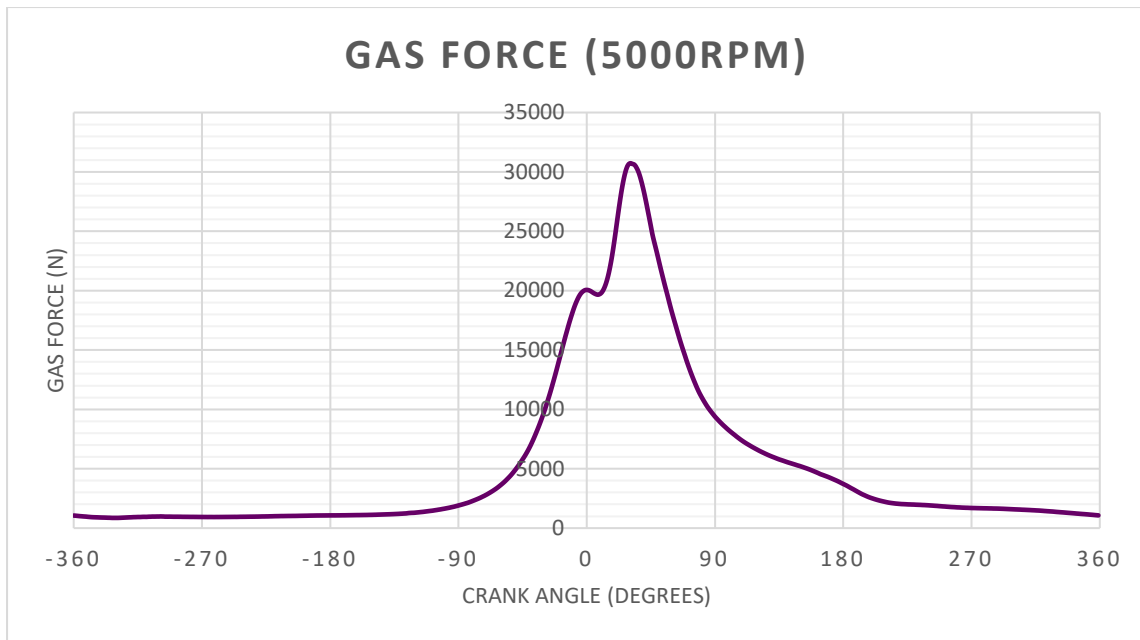


Figure 16-84: Maximum Gas Force at 5000RPM

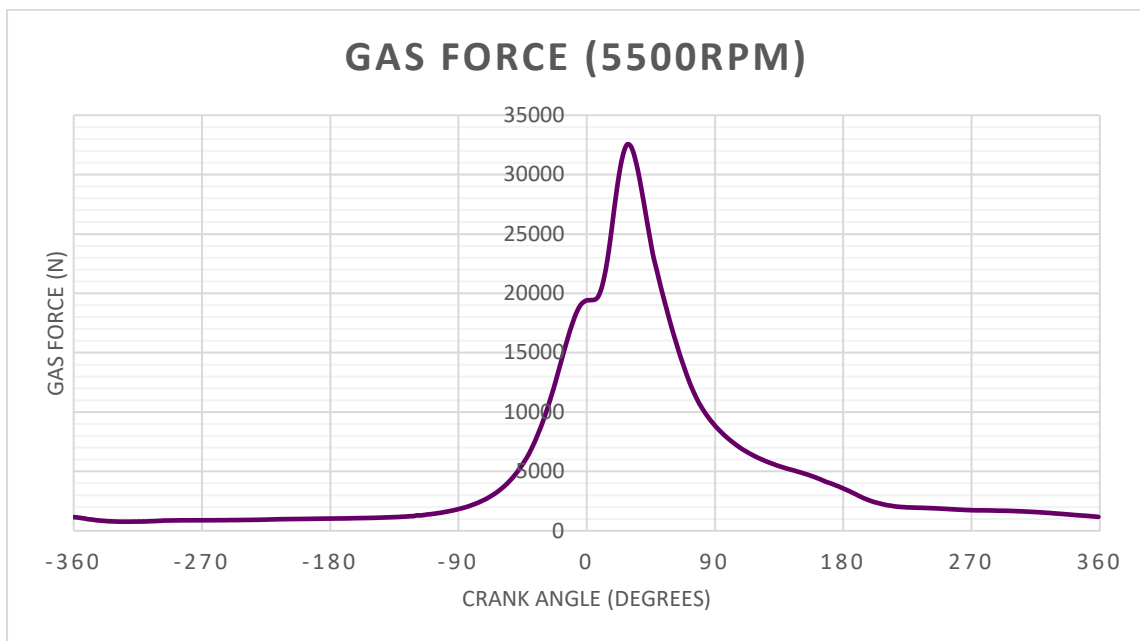


Figure 16-85: Maximum Gas Force at 5500RPM

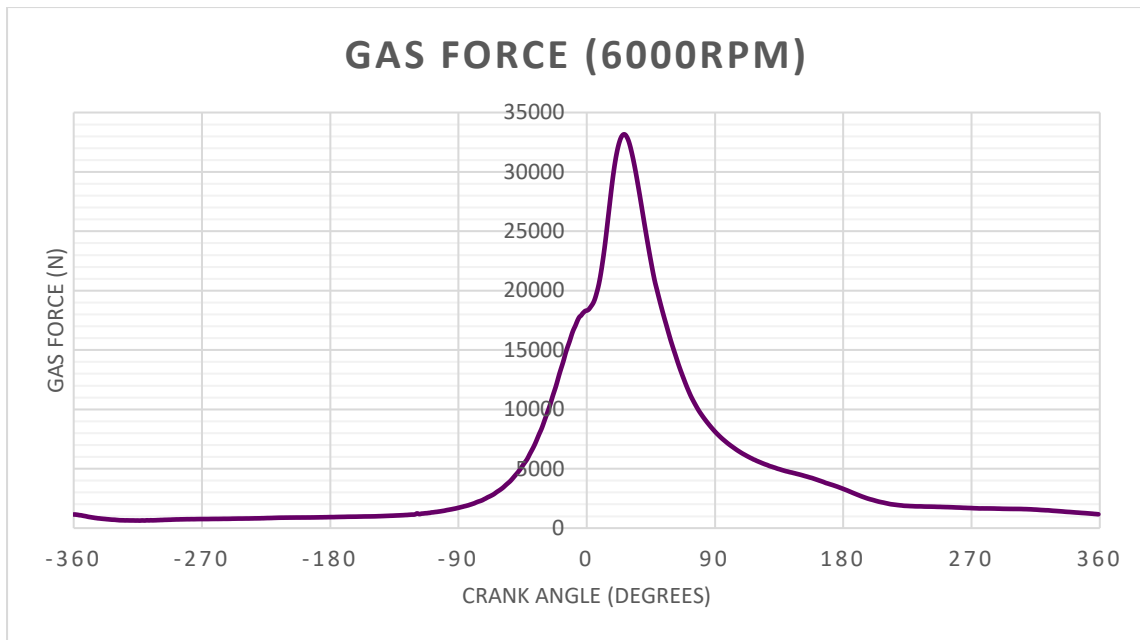


Figure 16-86: Maximum Gas Force at 6000RPM

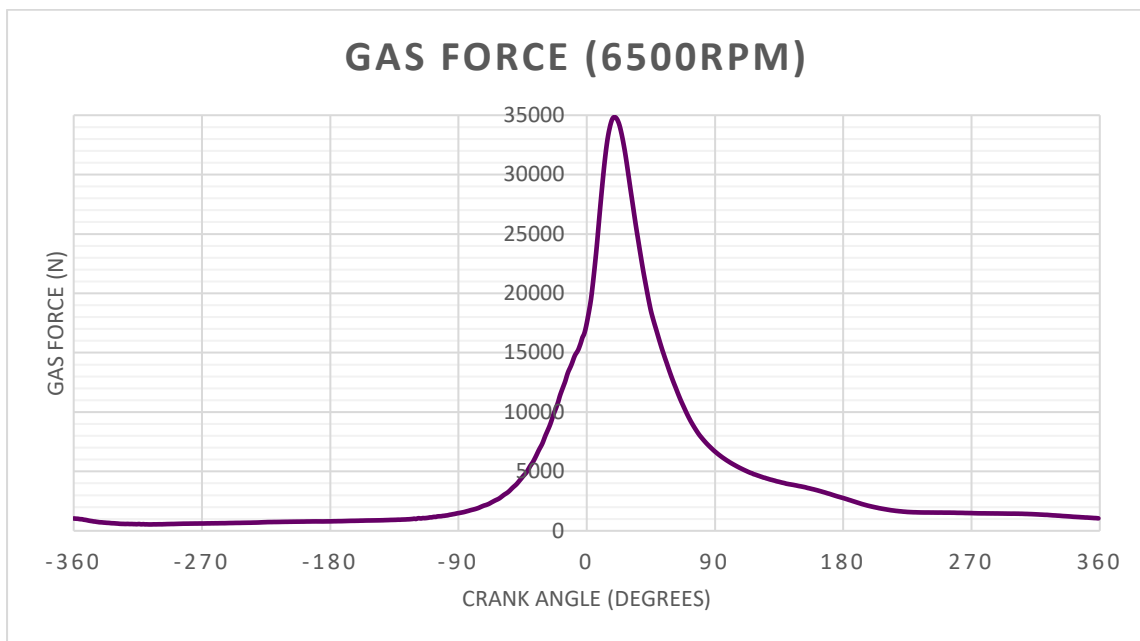


Figure 16-87: Maximum Gas Force at 6500RPM

16.4.2 Inertia Force

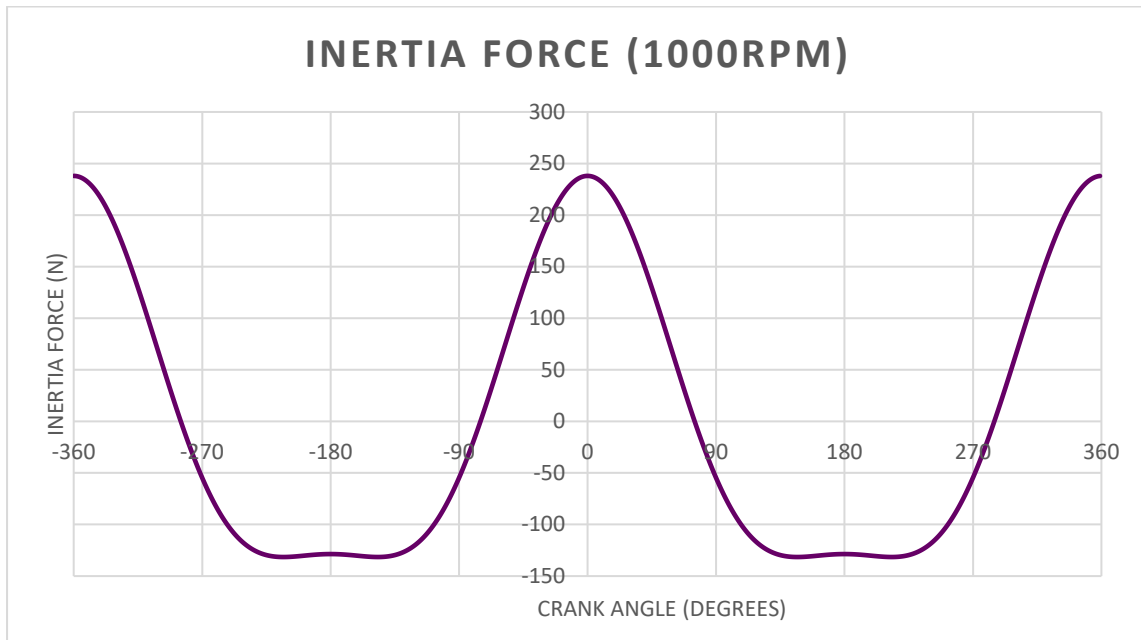


Figure 16-88: Inertia Force at 1000RPM

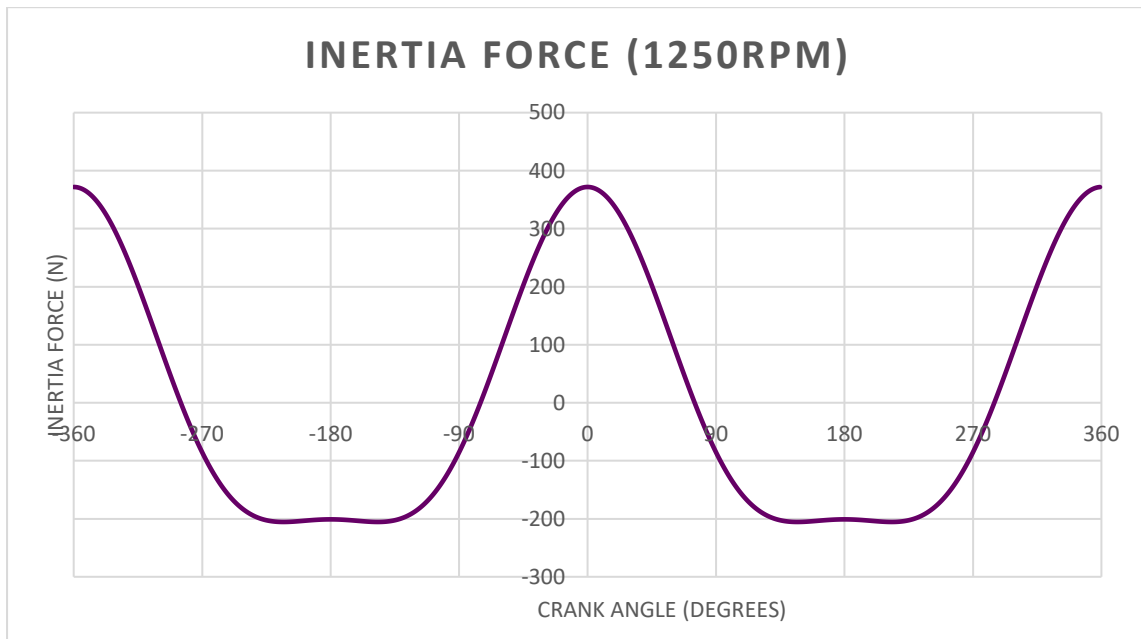


Figure 16-89: Inertia Force at 1250RPM

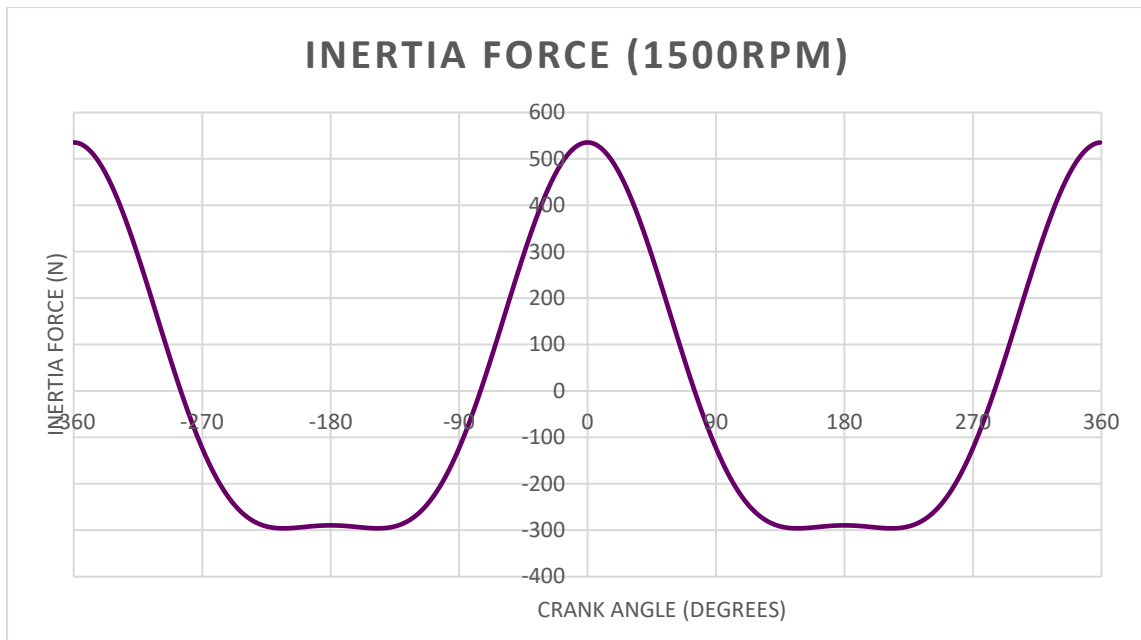


Figure 16-90: Inertia Force at 1500RPM

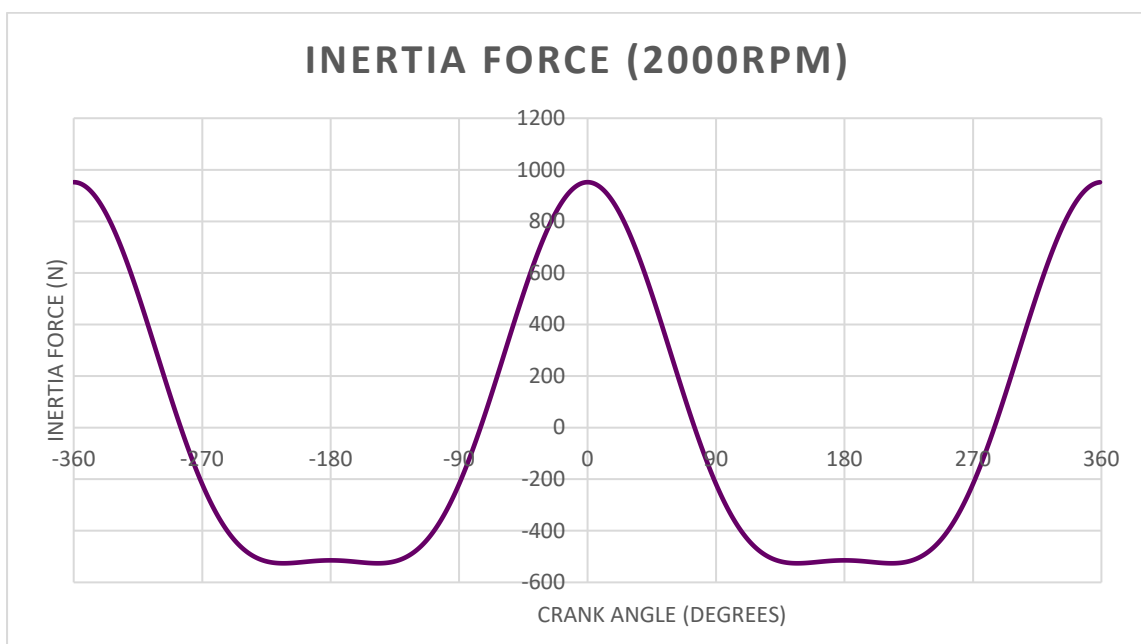


Figure 16-91: Inertia Force at 2000RPM

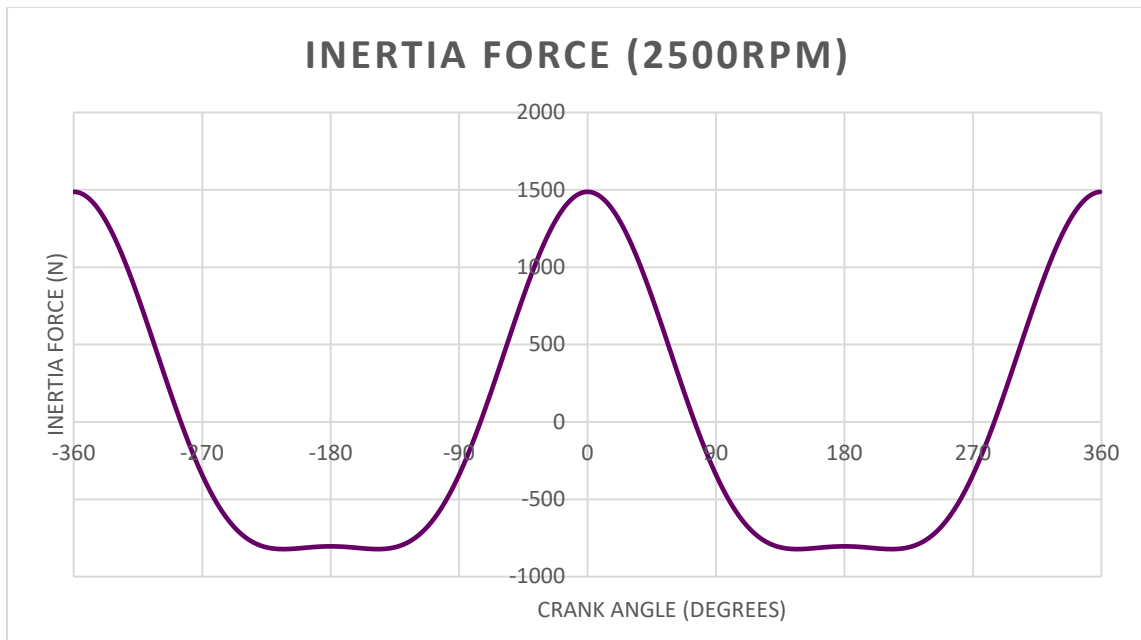


Figure 16-92: Inertia Force at 2500RPM

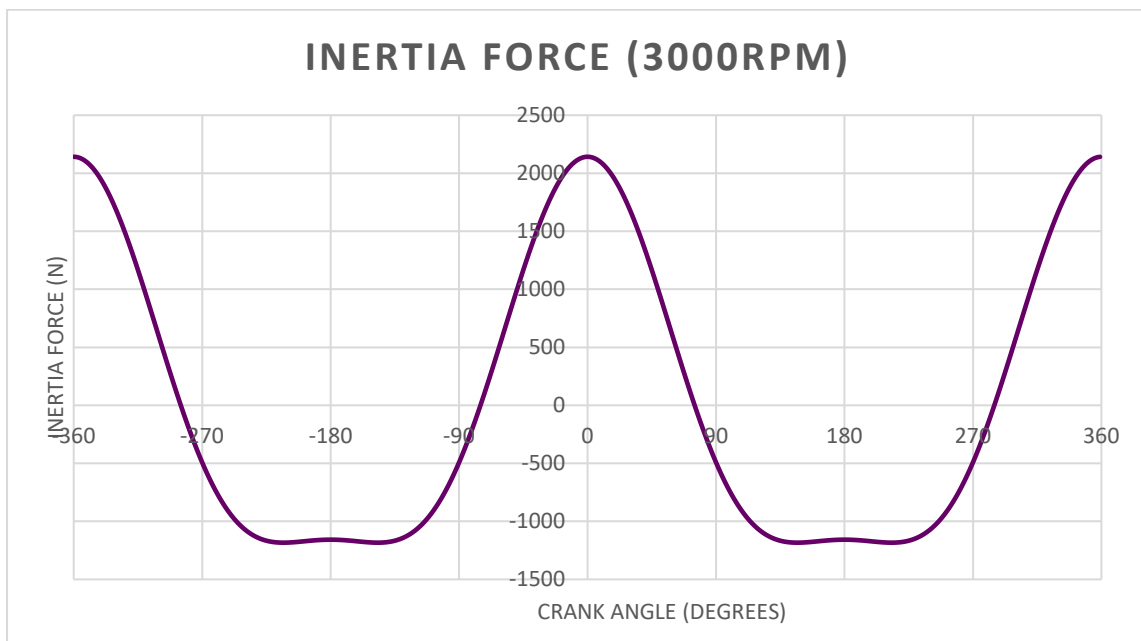


Figure 16-93: Inertia Force at 3000RPM

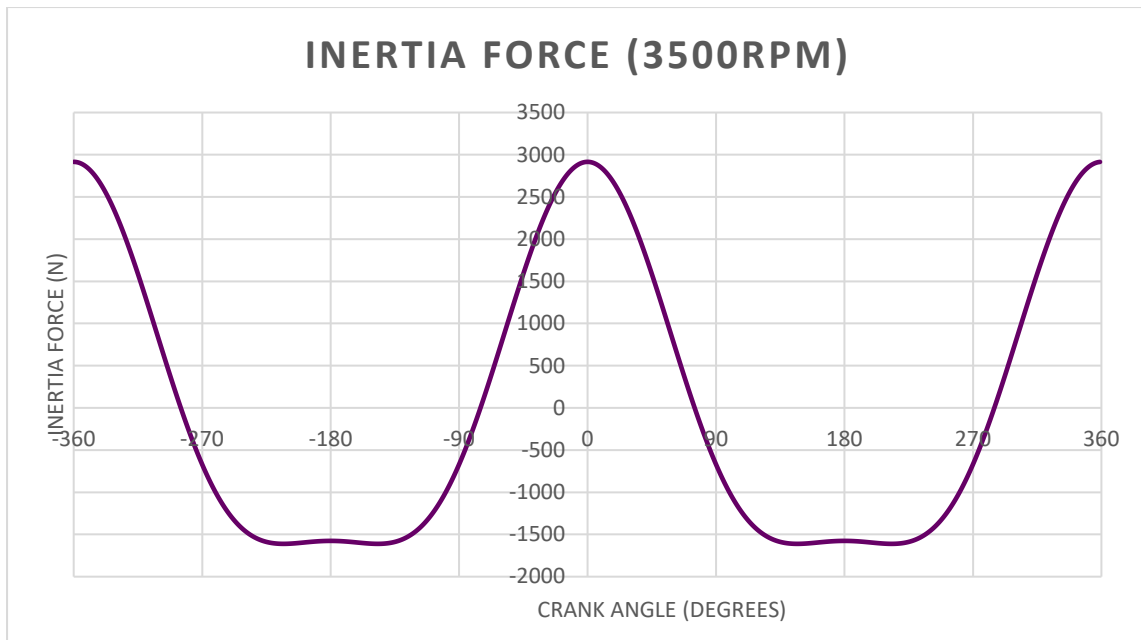


Figure 16-94: Inertia Force at 3500RPM

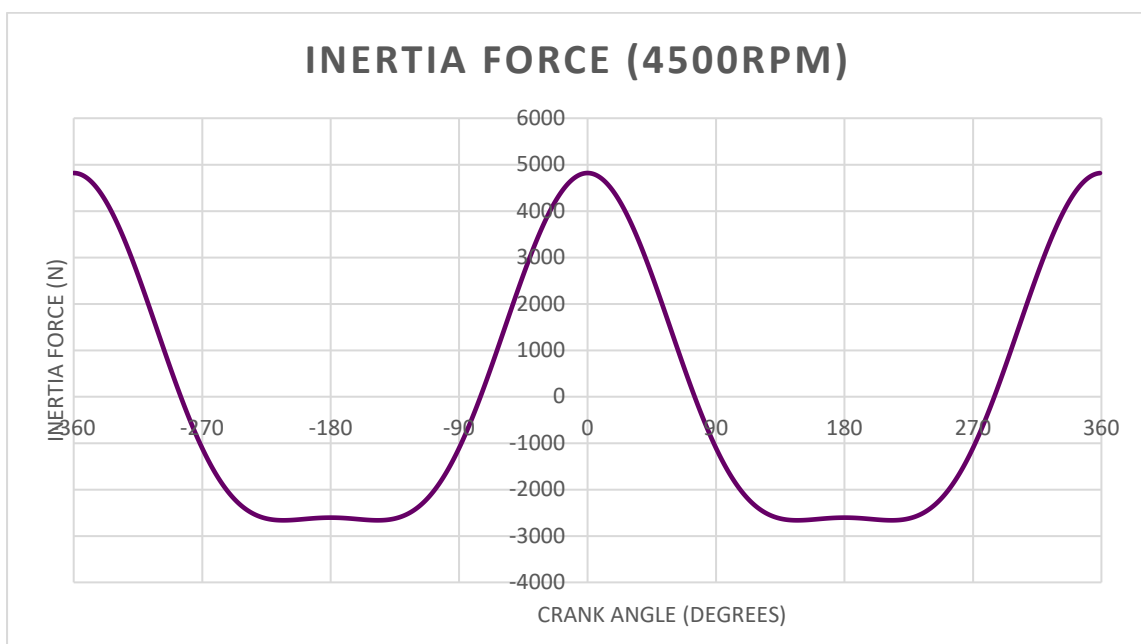


Figure 16-95: Inertia Force at 4500RPM

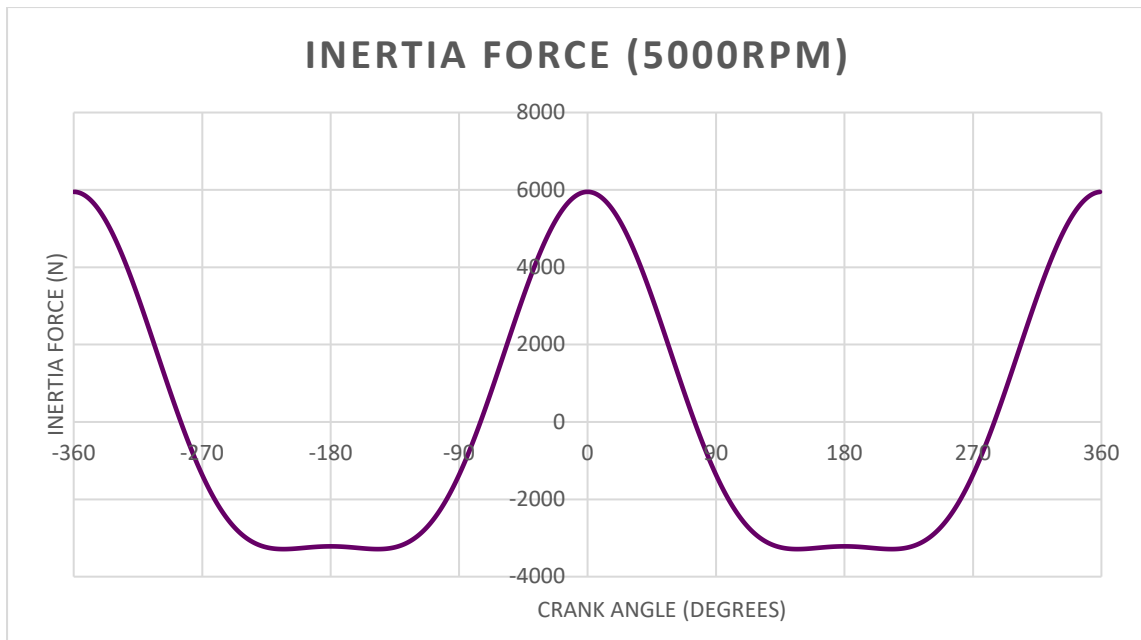


Figure 16-96: Inertia Force at 5000RPM

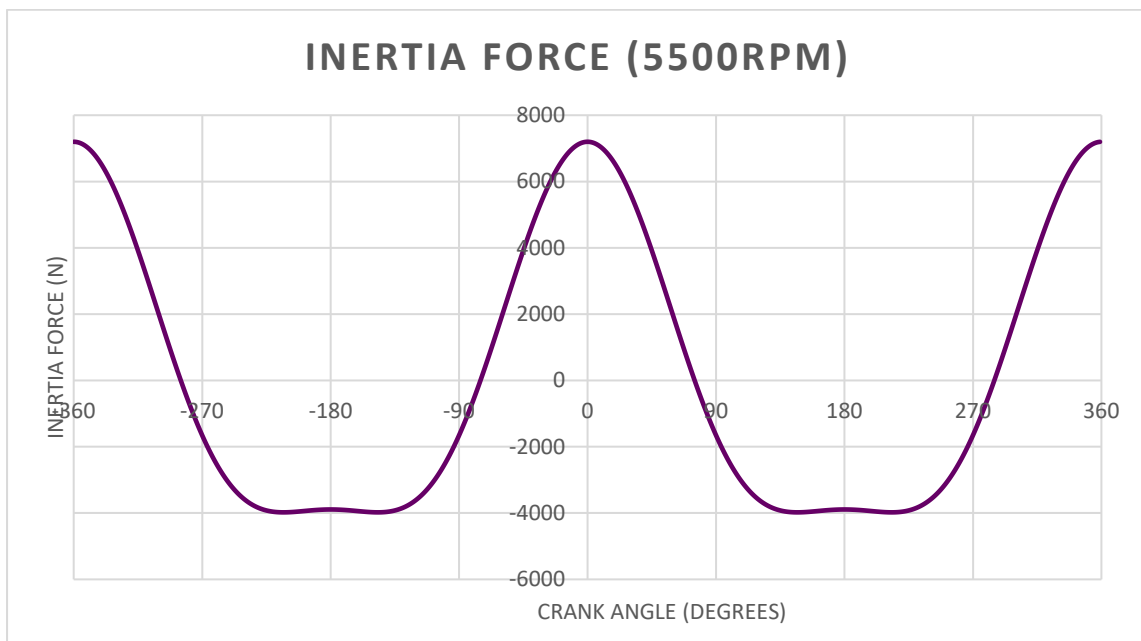


Figure 16-97: Inertia Force at 5500RPM

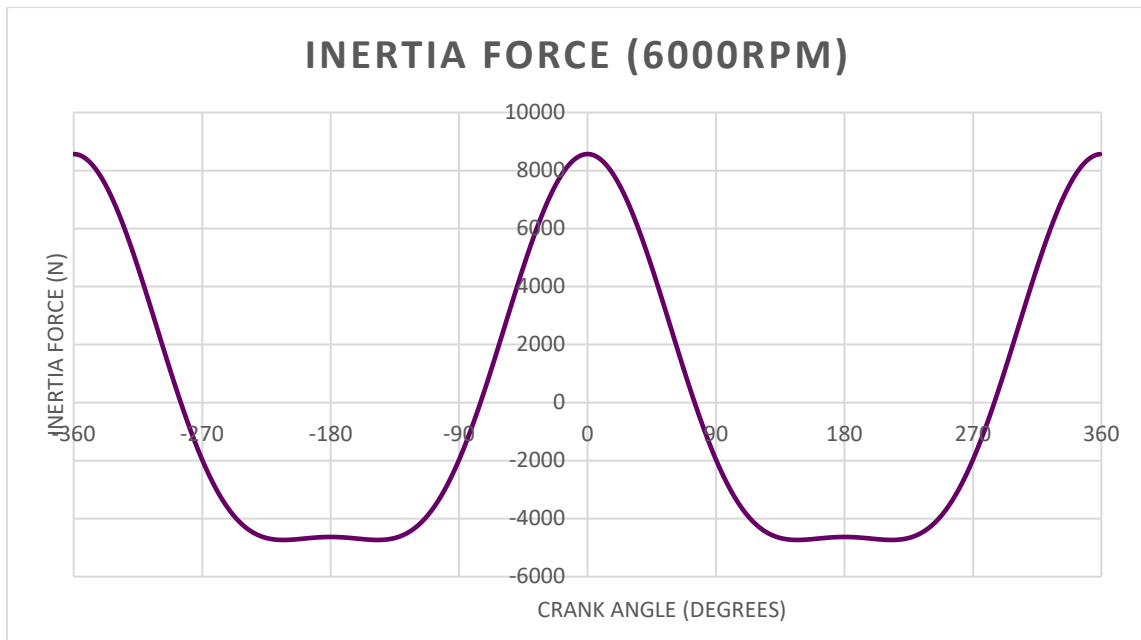


Figure 16-98: Inertia Force at 6000RPM

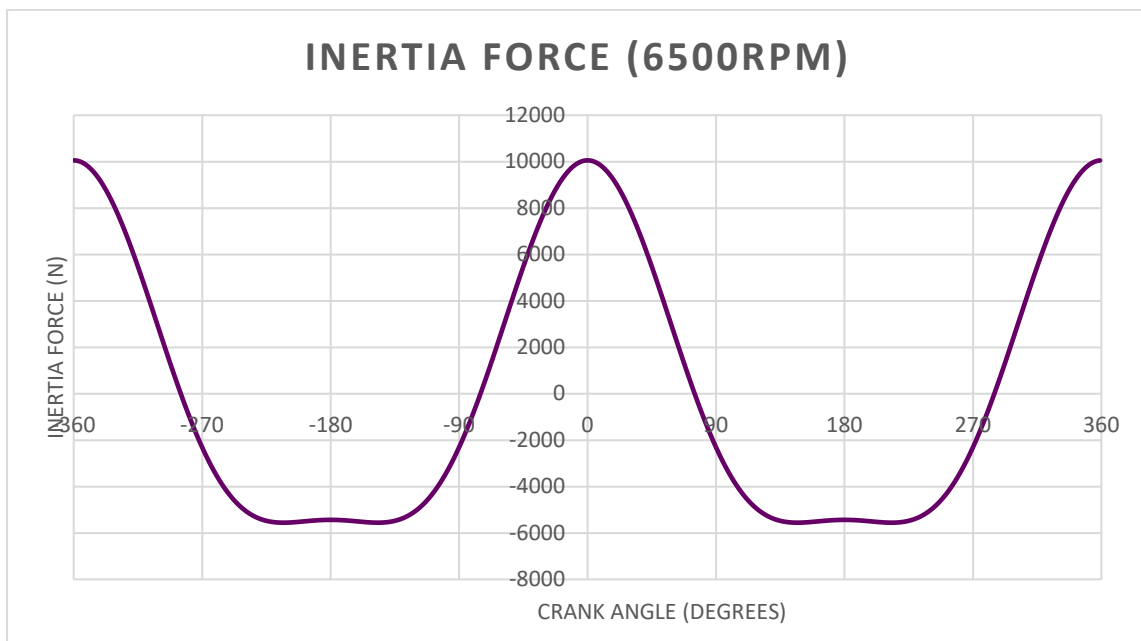


Figure 16-99: Inertia Force at 6500RPM

16.4.3 Net Force

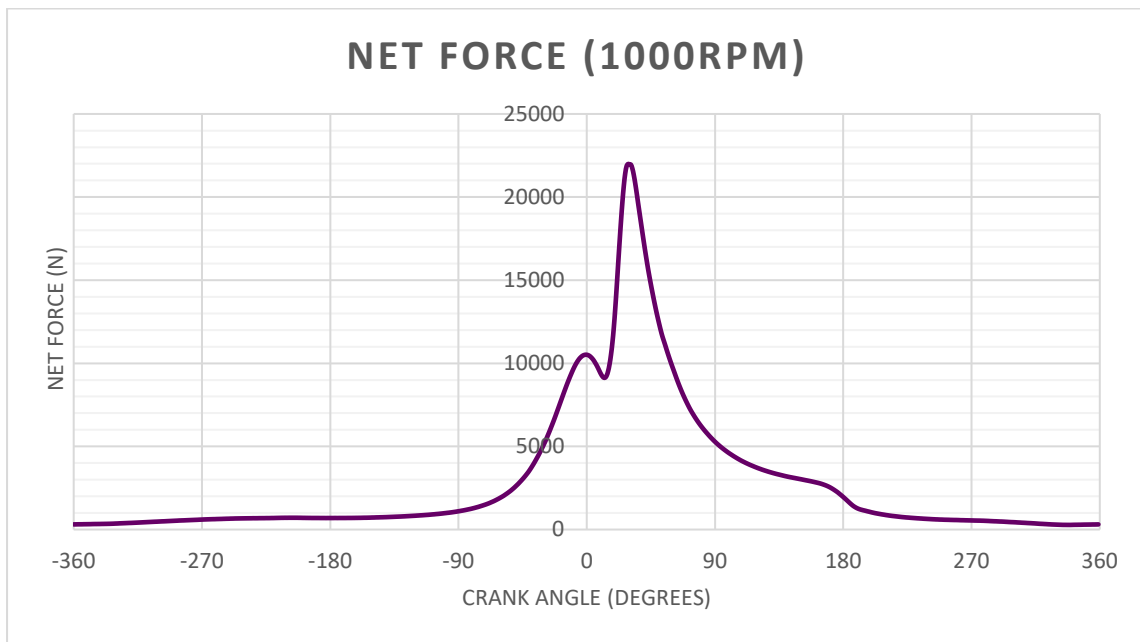


Figure 16-100 Net Force at 1000RPM

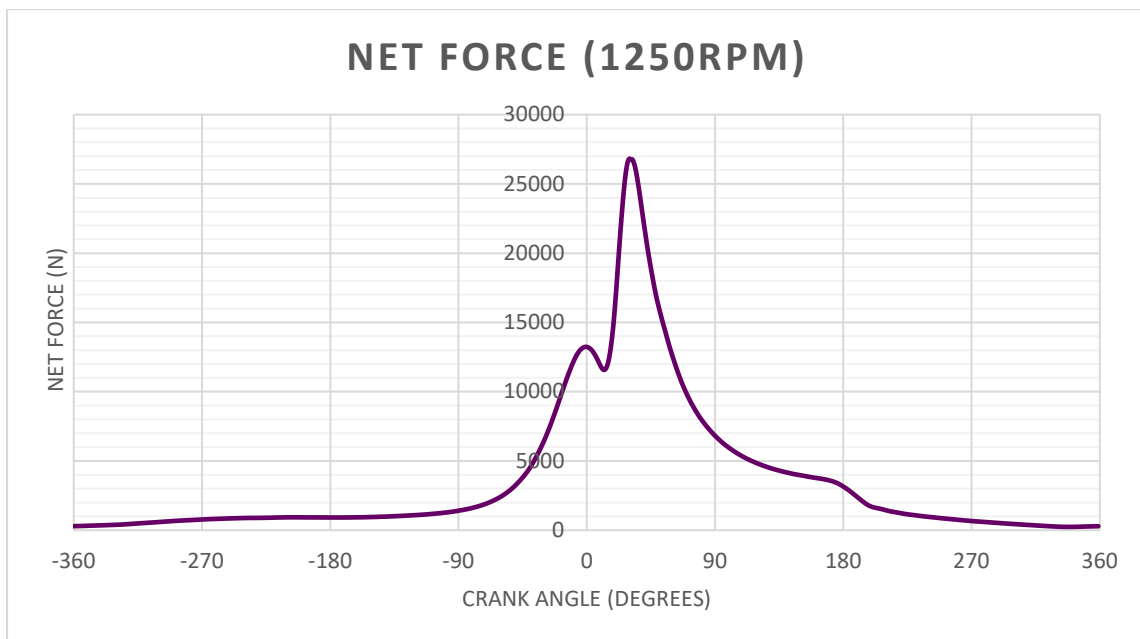


Figure 16-101: Net Force at 1250RPM

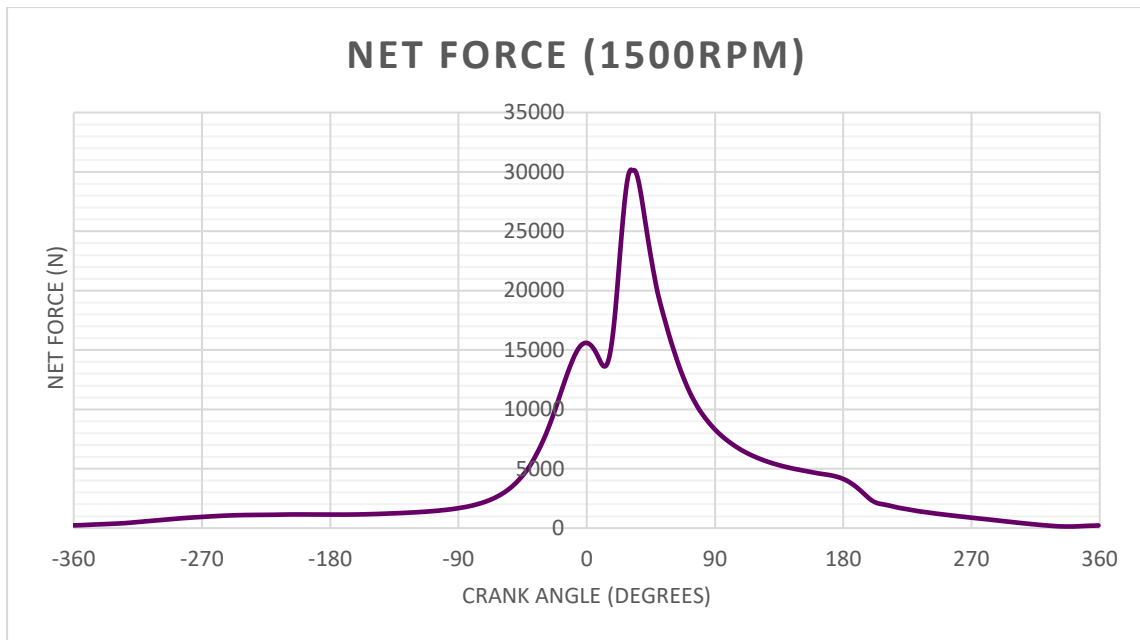


Figure 16-102: Net Force at 1500RPM

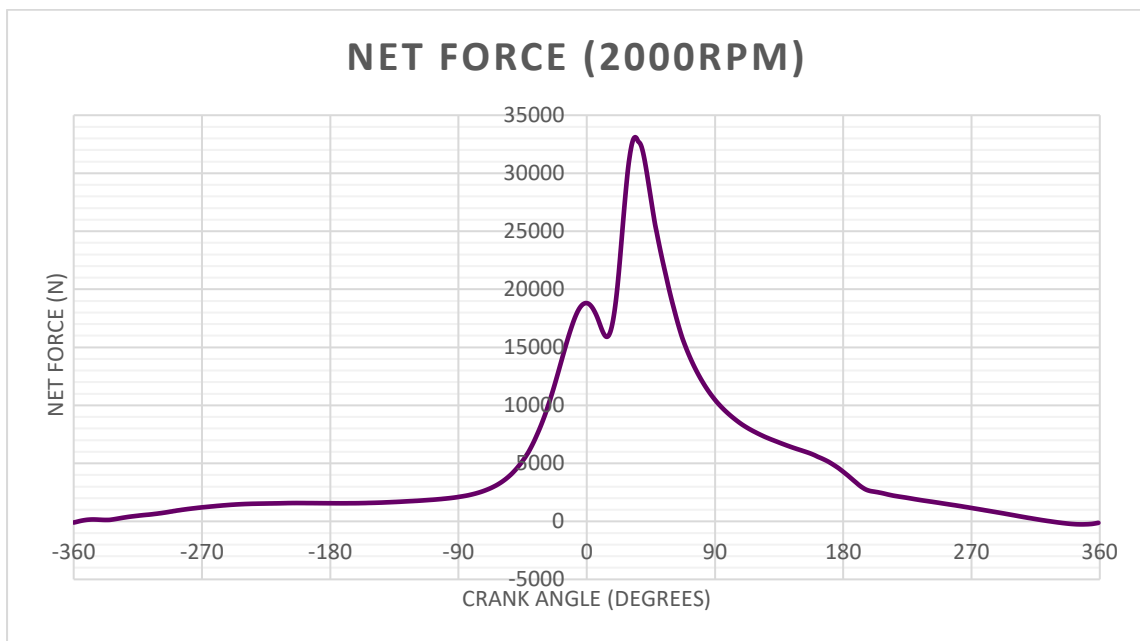


Figure 16-103: Net Force at 2000RPM

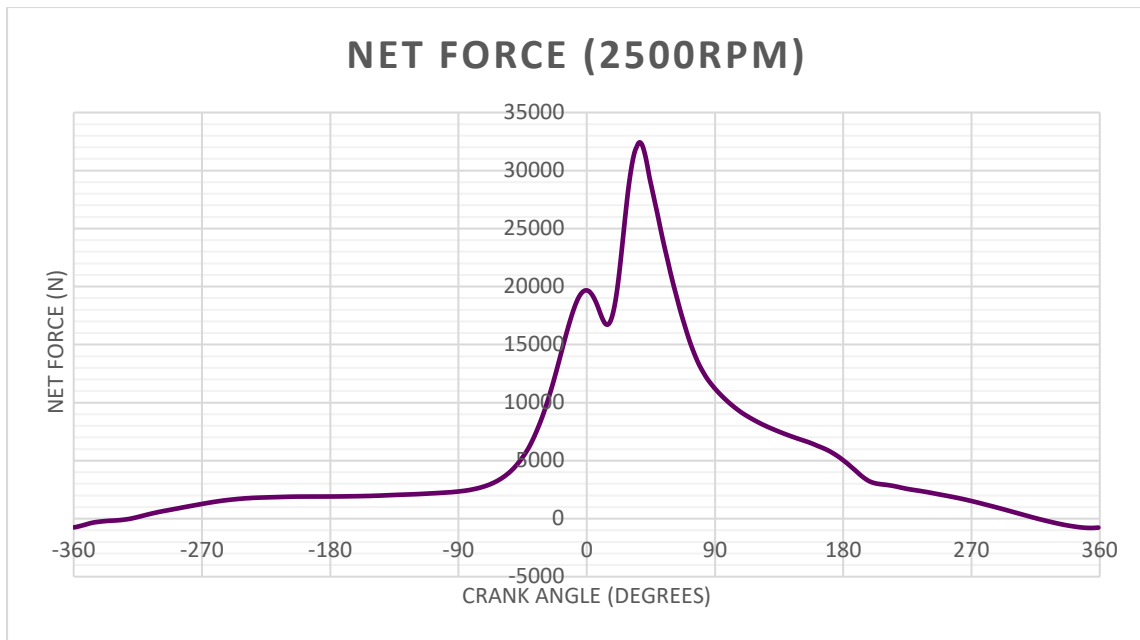


Figure 16-104: Net Force at 2500RPM

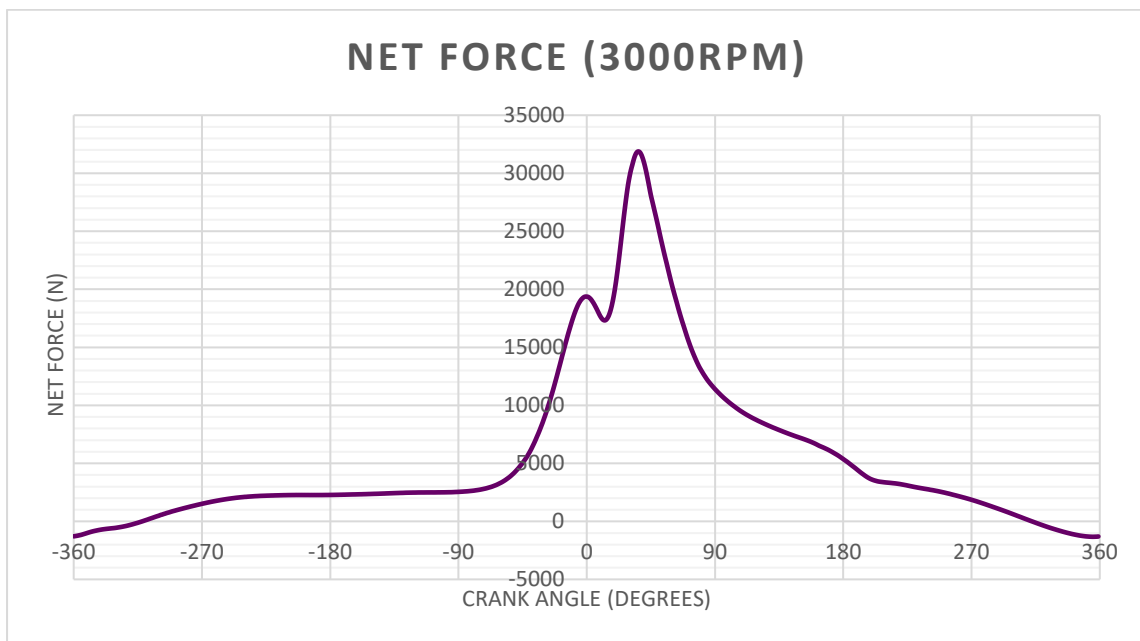


Figure 16-105: Net Force at 3000RPM

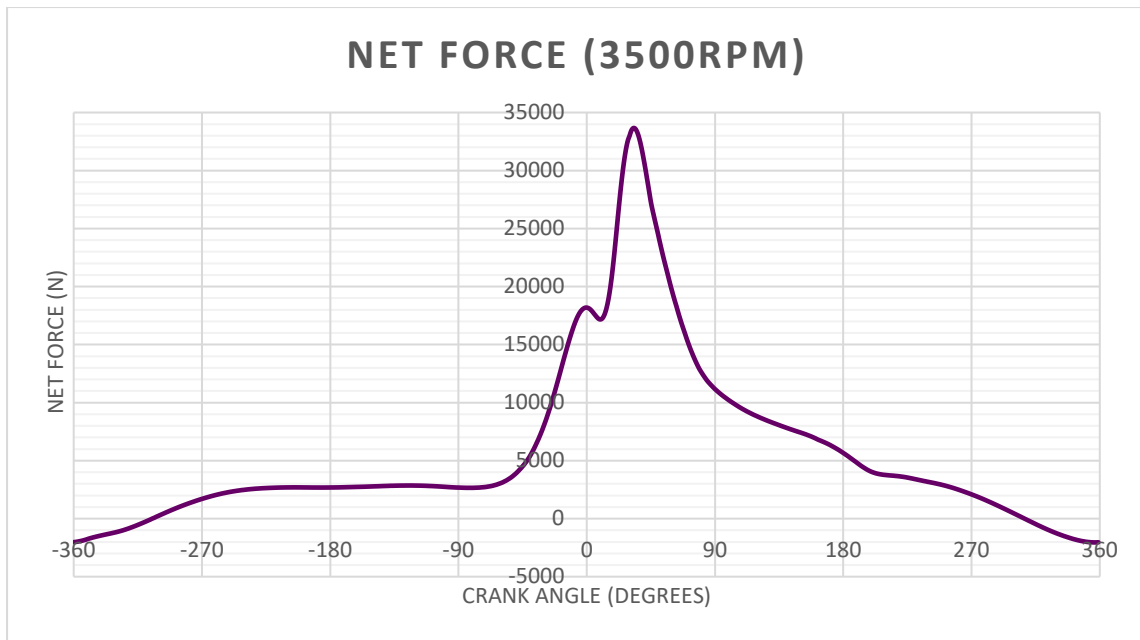


Figure 16-106: Net Force at 3500RPM

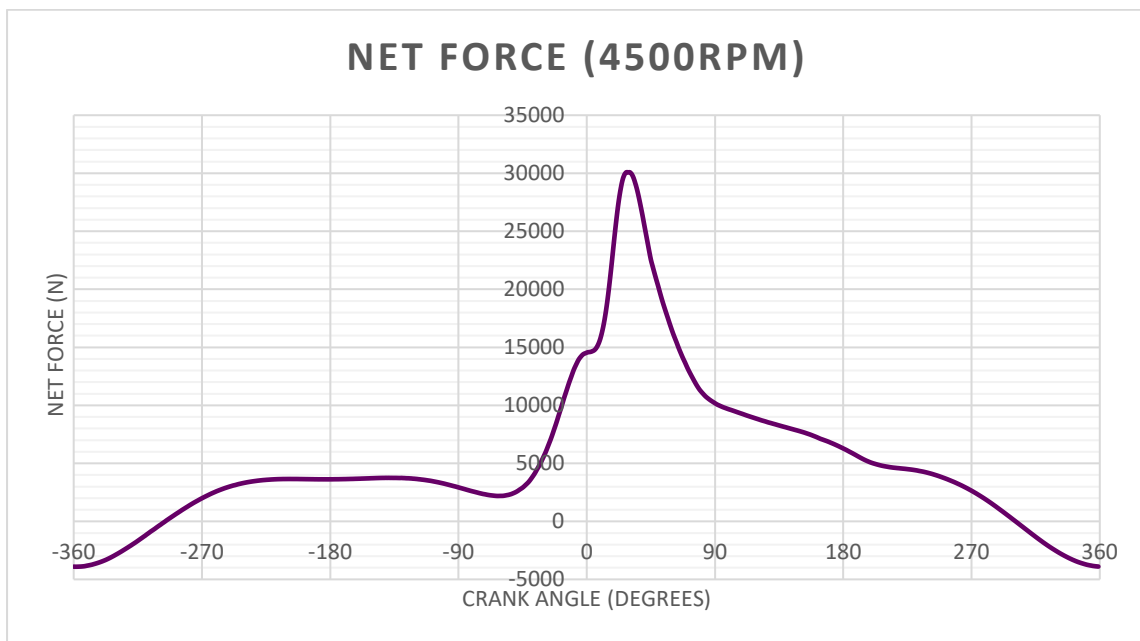


Figure 16-107: Net Force at 4500RPM

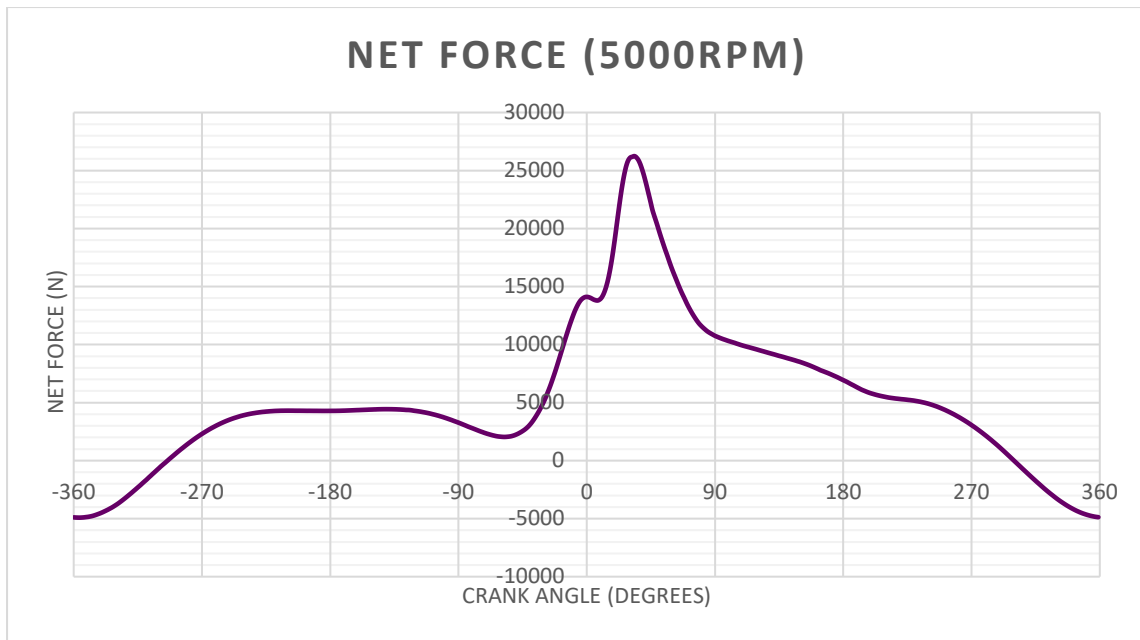


Figure 16-108: Net Force at 5000RPM

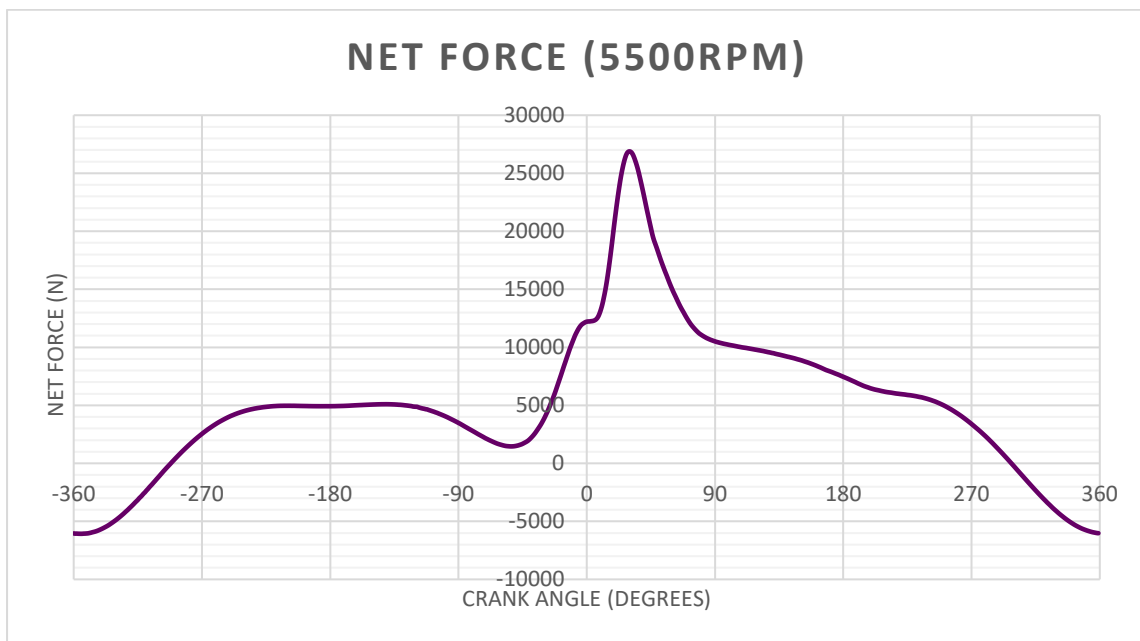


Figure 16-109: Net Force at 5500RPM

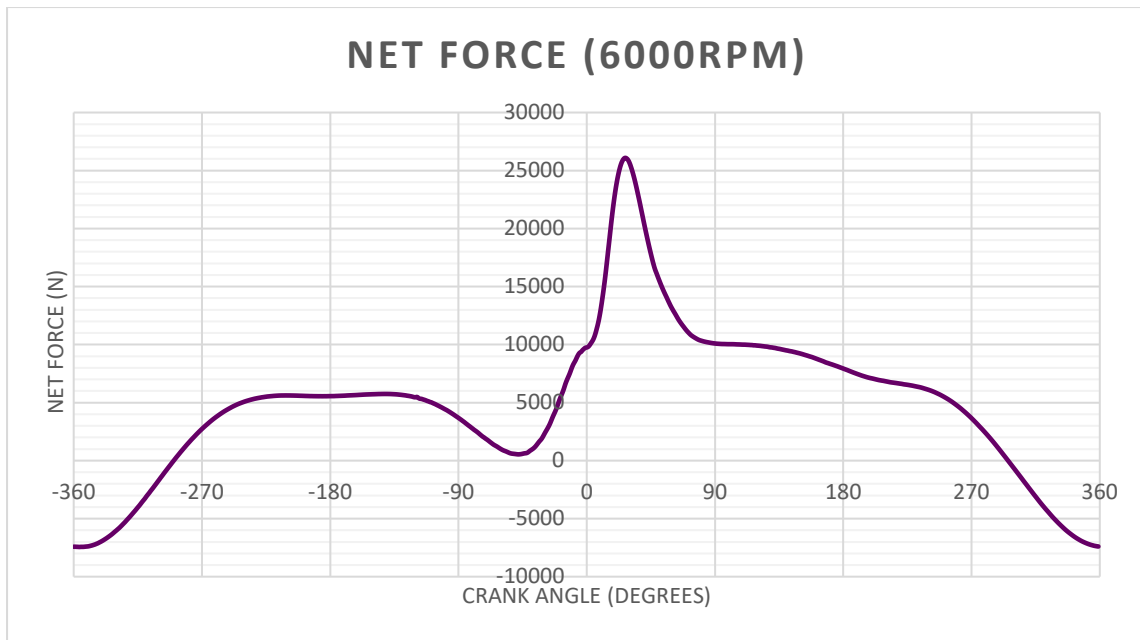


Figure 16-110: Net Force at 6000RPM

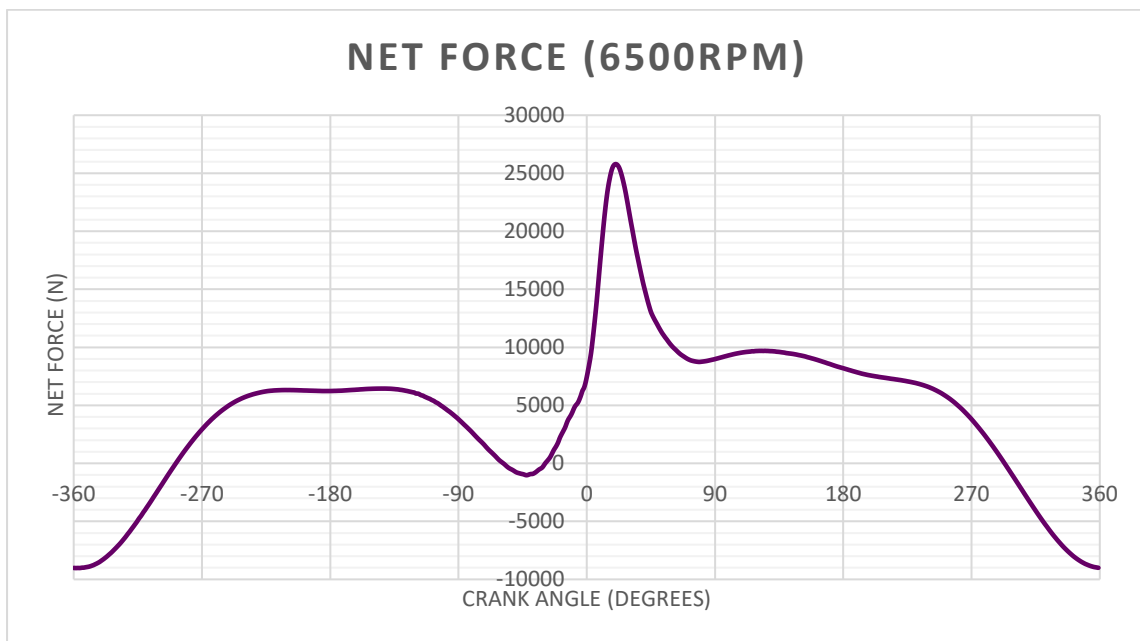


Figure 16-111: Net Force at 6500RPM

16.4.4 Connecting Rod Force

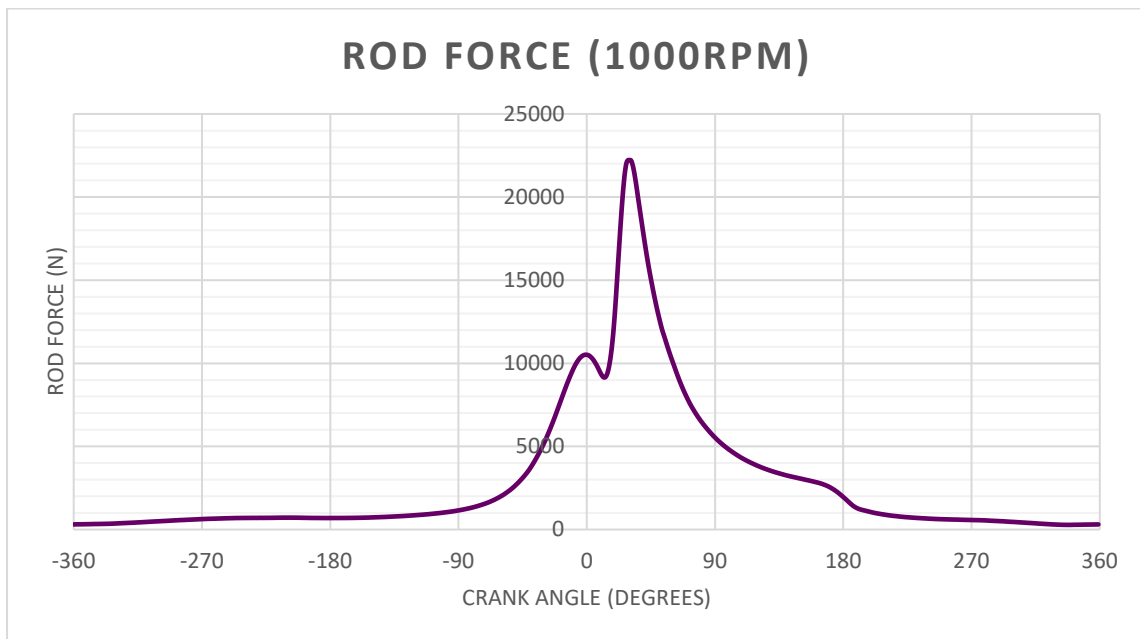


Figure 16-112: Force Acting Through the Connecting Rod at 1000RPM

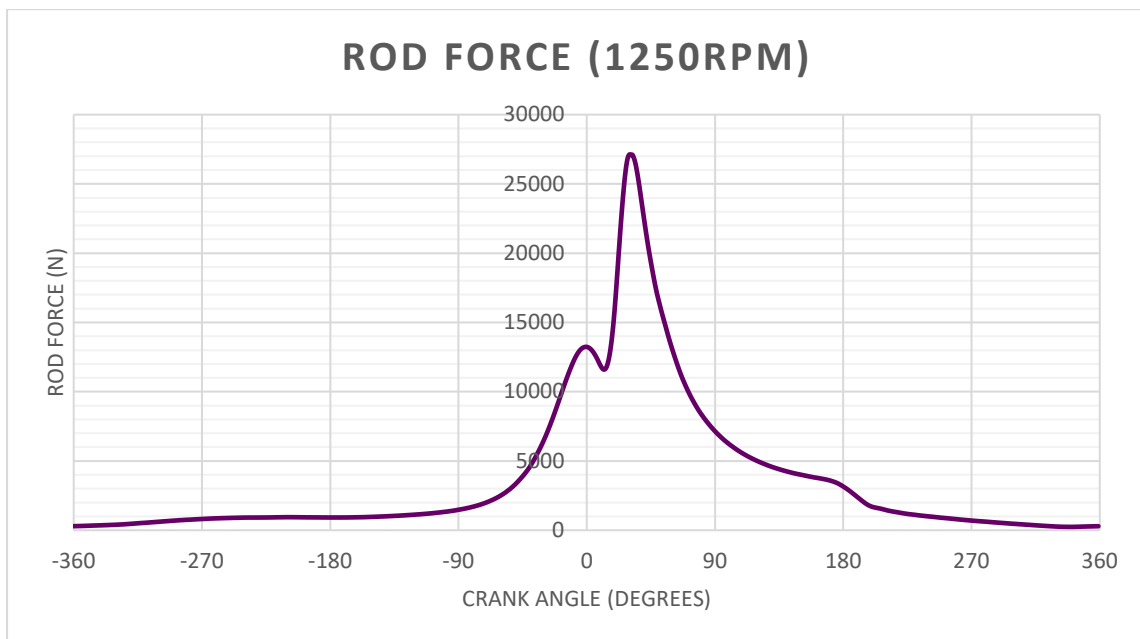


Figure 16-113: Force Acting Through the Connecting Rod at 1250RPM

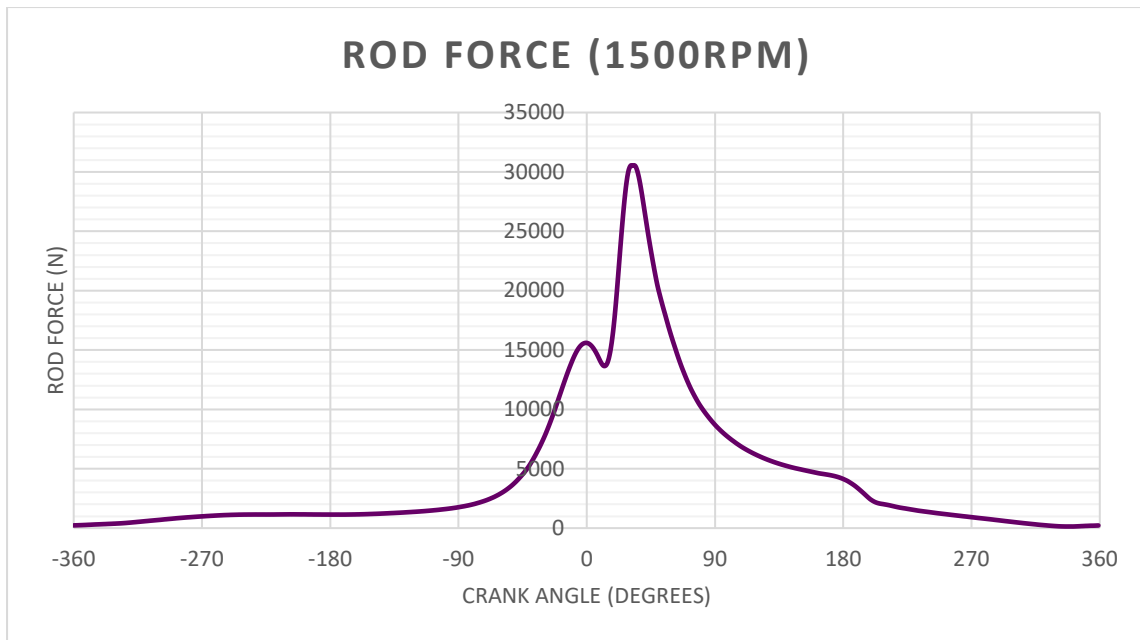


Figure 16-114: Force Acting Through the Connecting Rod at 1500RPM

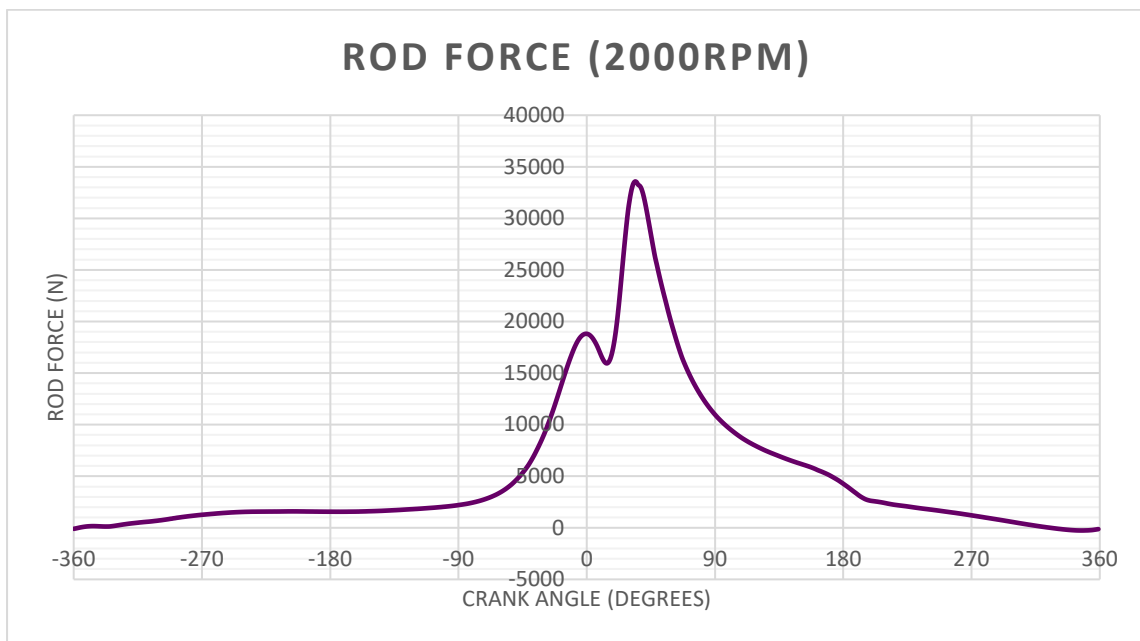


Figure 16-115: Force Acting Through the Connecting Rod at 2000RPM

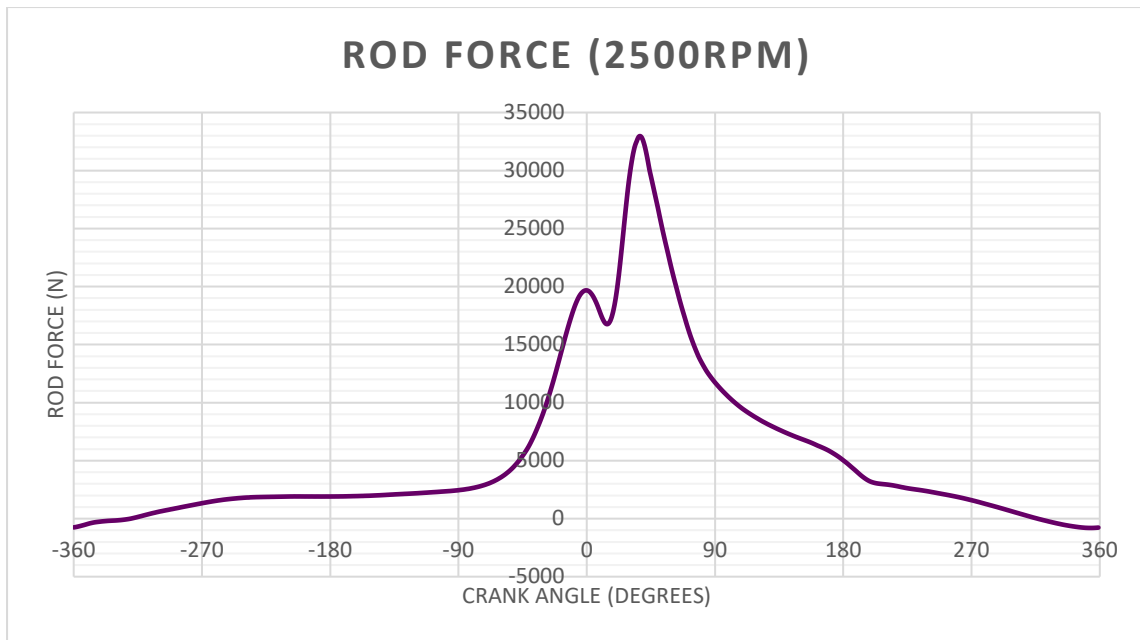


Figure 16-116: Force Acting Through the Connecting Rod at 2500RPM

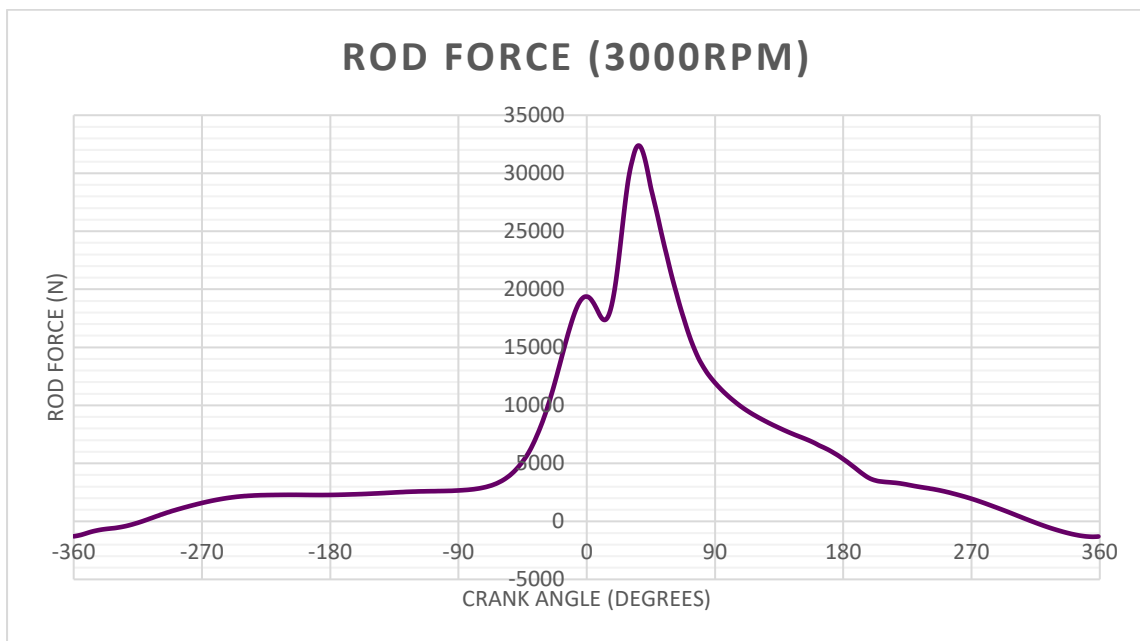


Figure 16-117: Force Acting Through the Connecting Rod at 3000RPM

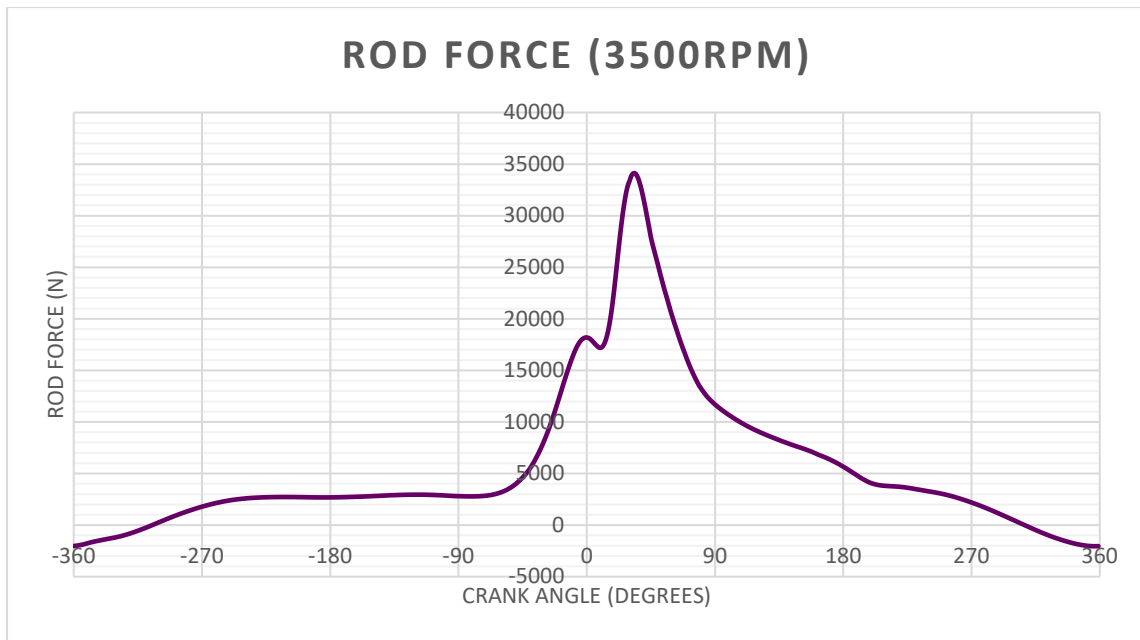


Figure 16-118: Force Acting Through the Connecting Rod at 3500RPM

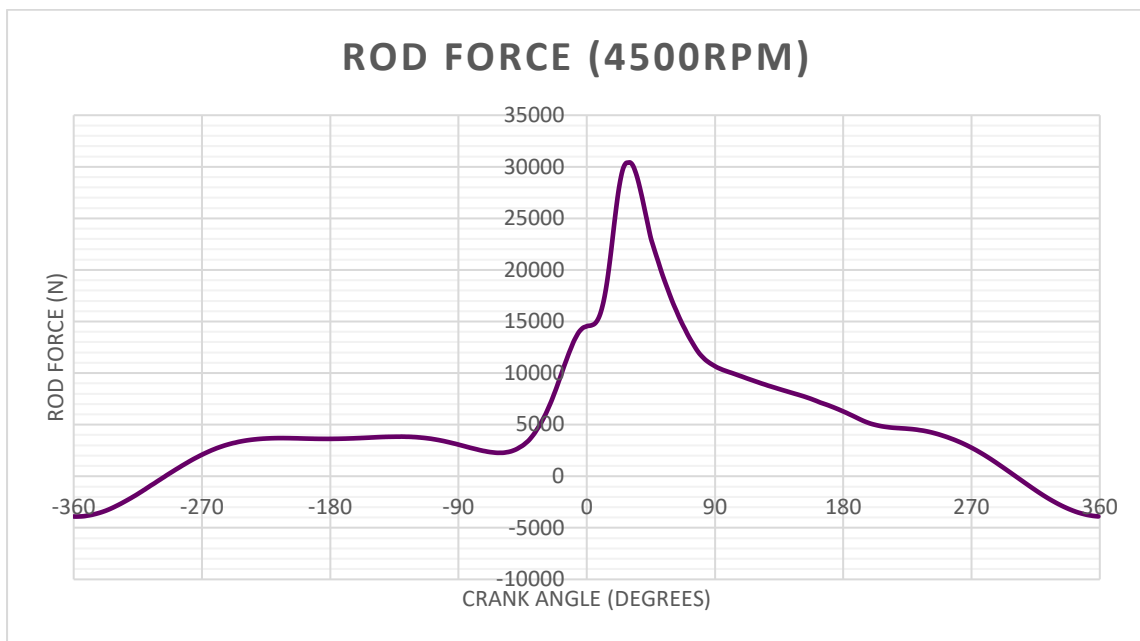


Figure 16-119: Force Acting Through the Connecting Rod at 4500RPM

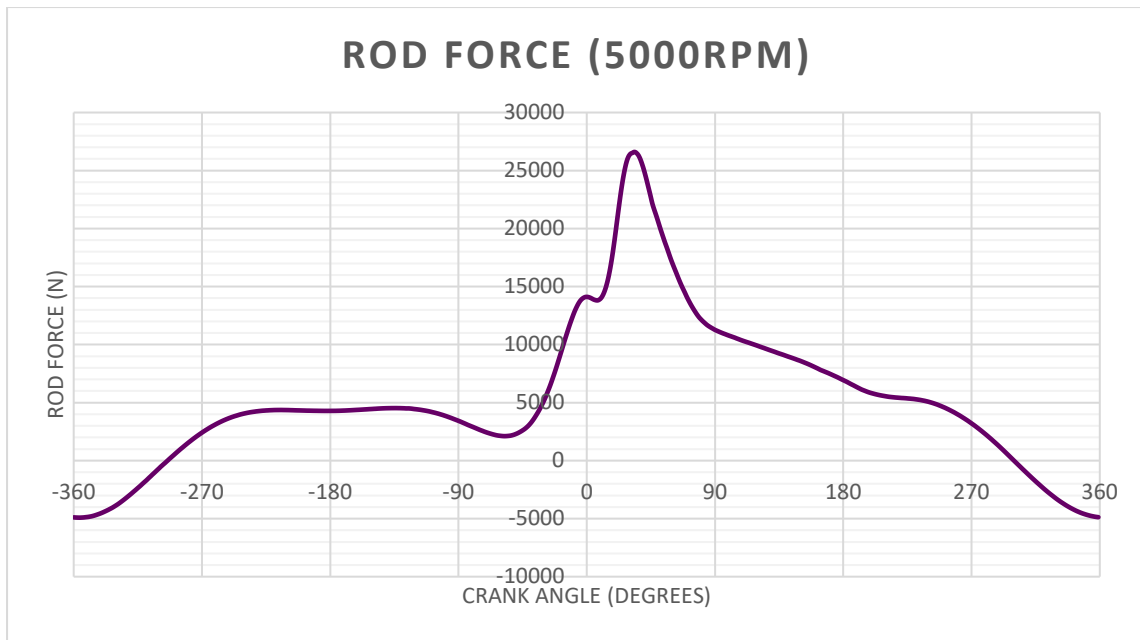


Figure 16-120: Force Acting Through the Connecting Rod at 5000RPM

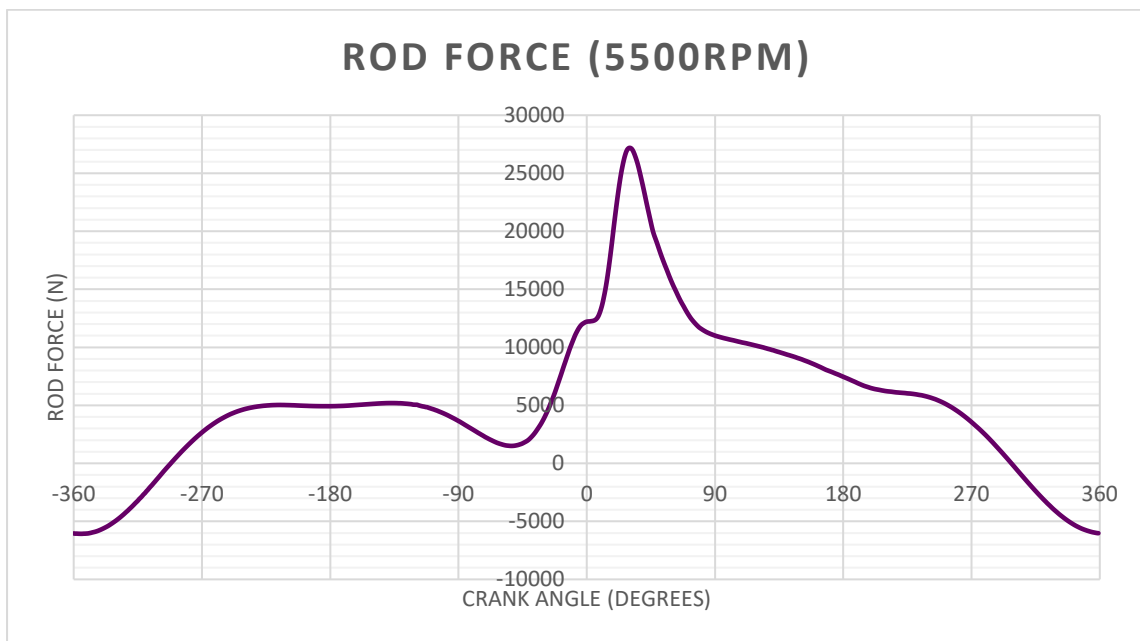


Figure 16-121: Force Acting Through the Connecting Rod at 5500RPM

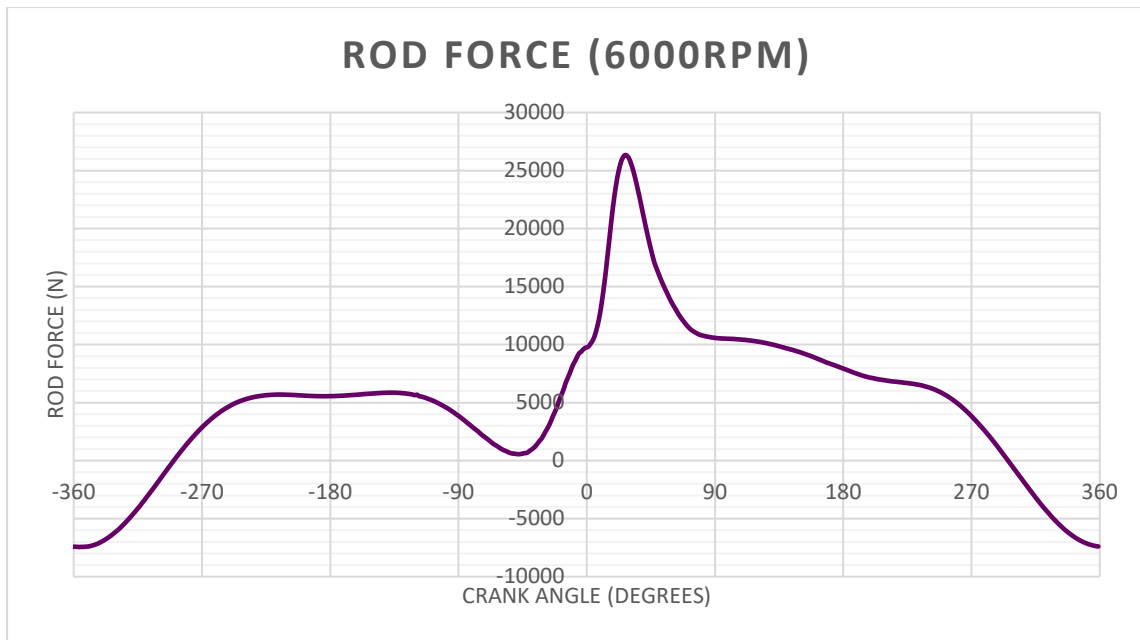


Figure 16-122: Force Acting Through the Connecting Rod at 6000RPM

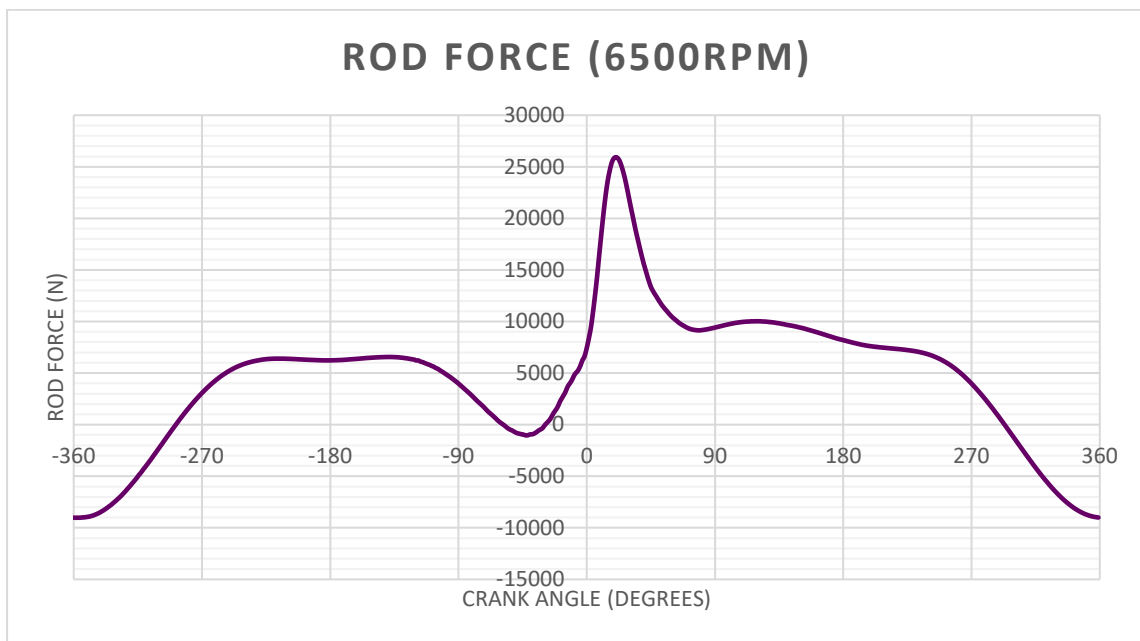


Figure 16-123: Force Acting Through the Connecting Rod at 6500RPM

16.4.5 Work Done by the Pressure

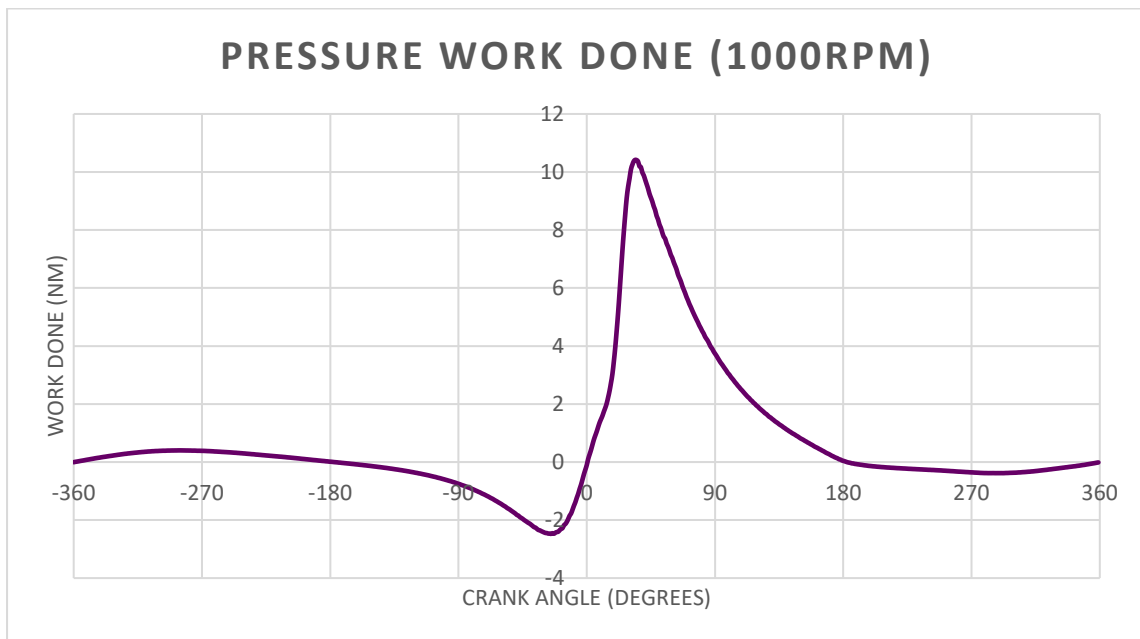


Figure 16-124: Work Done by the Pressure from Combustion at 1000RPM

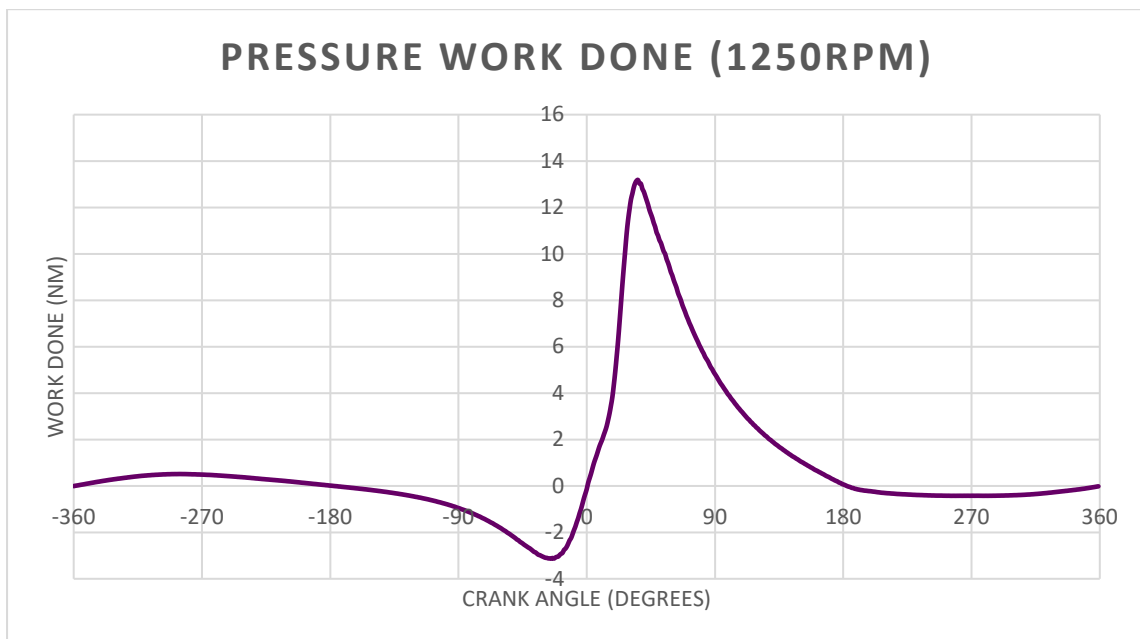


Figure 16-125: Work Done by the Pressure from Combustion at 1250RPM

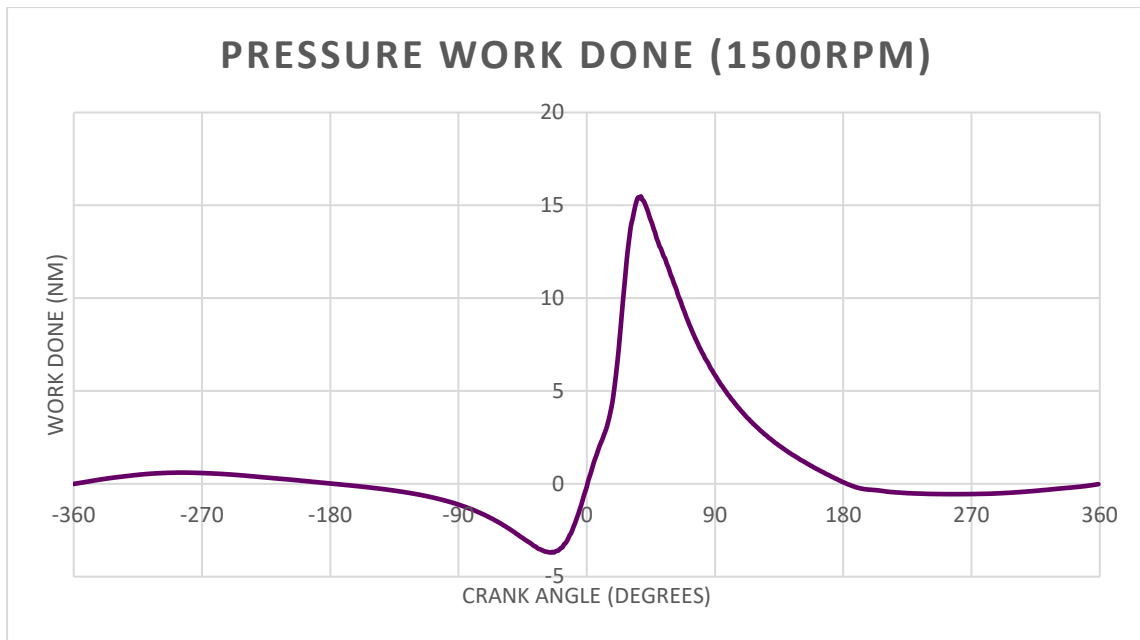


Figure 16-126: Work Done by the Pressure from Combustion at 1500RPM

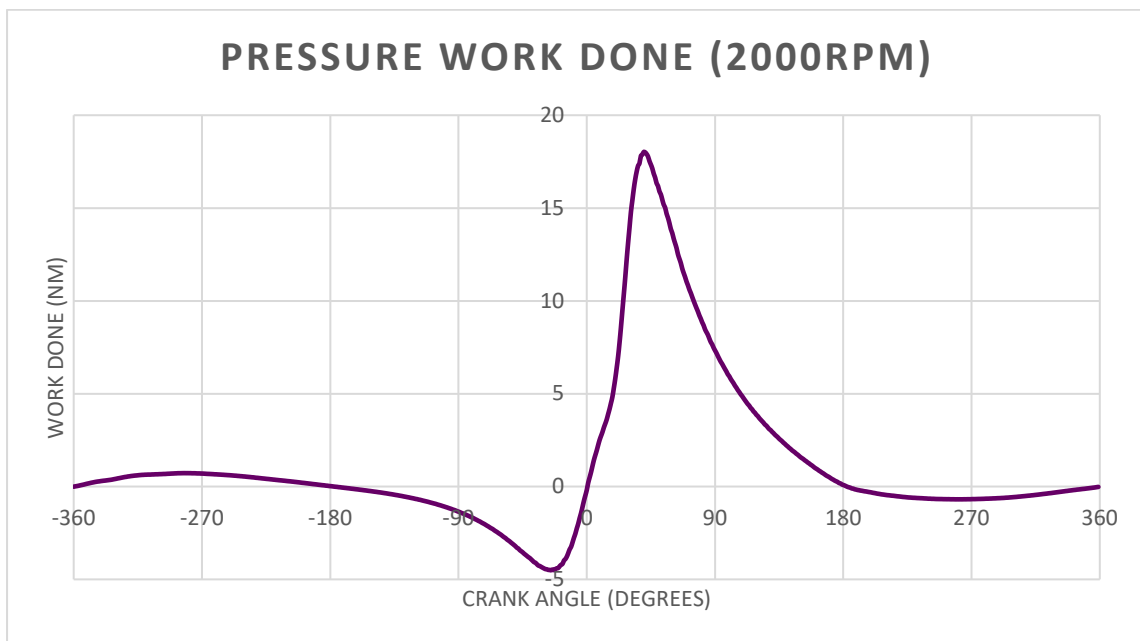


Figure 16-127: Work Done by the Pressure from Combustion at 2000RPM

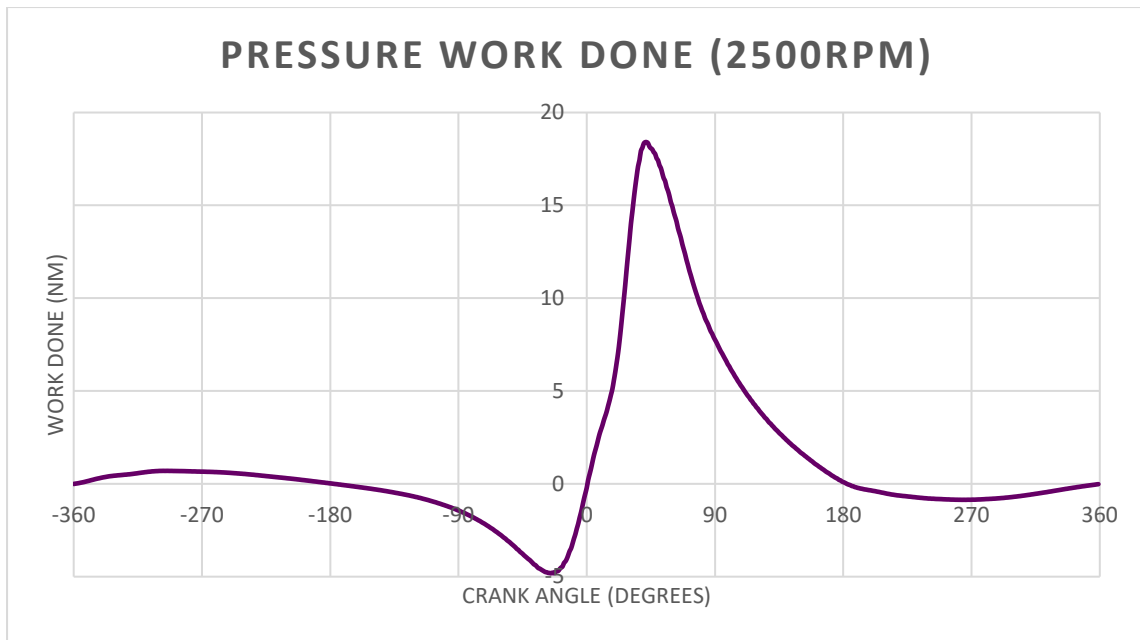


Figure 16-128: Work Done by the Pressure from Combustion at 2500RPM

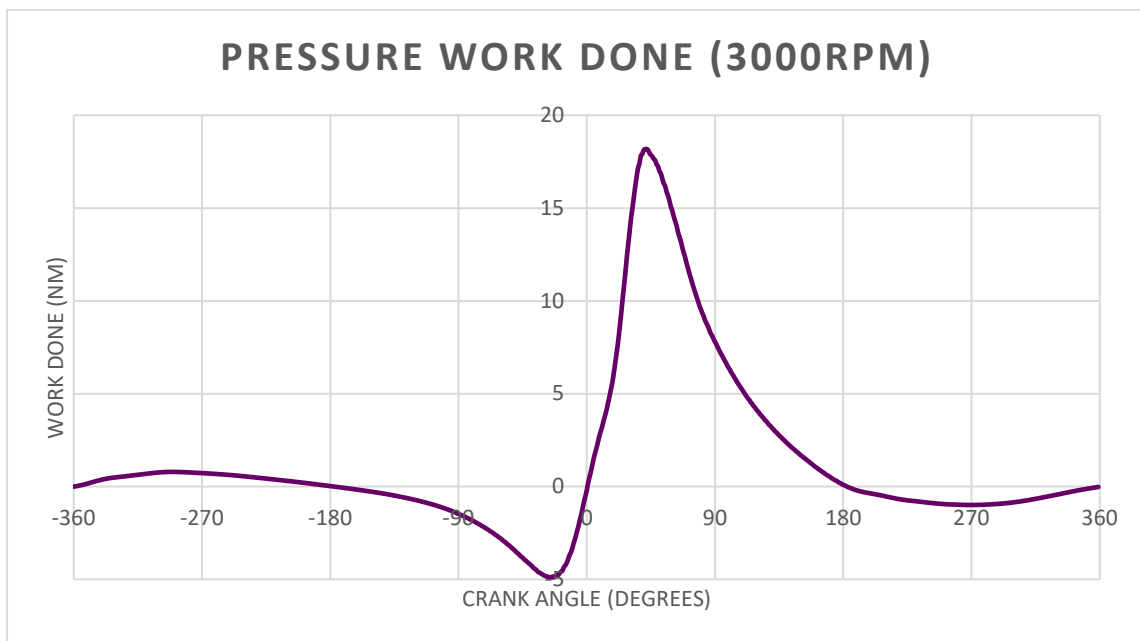


Figure 16-129: Work Done by the Pressure from Combustion at 3000RPM

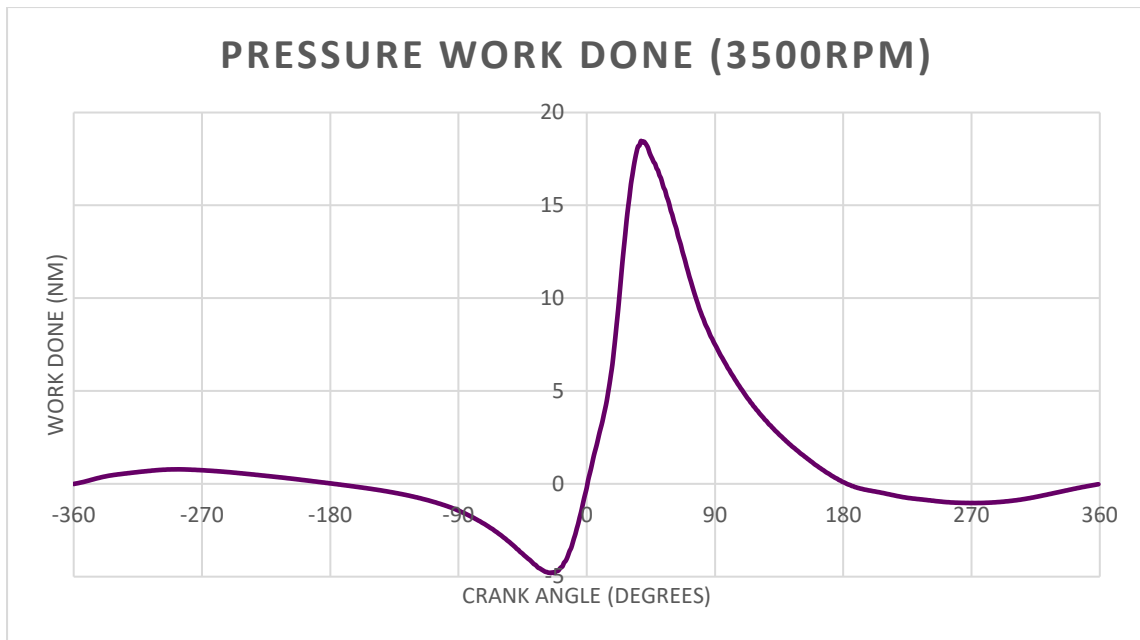


Figure 16-130: Work Done by the Pressure from Combustion at 3500RPM

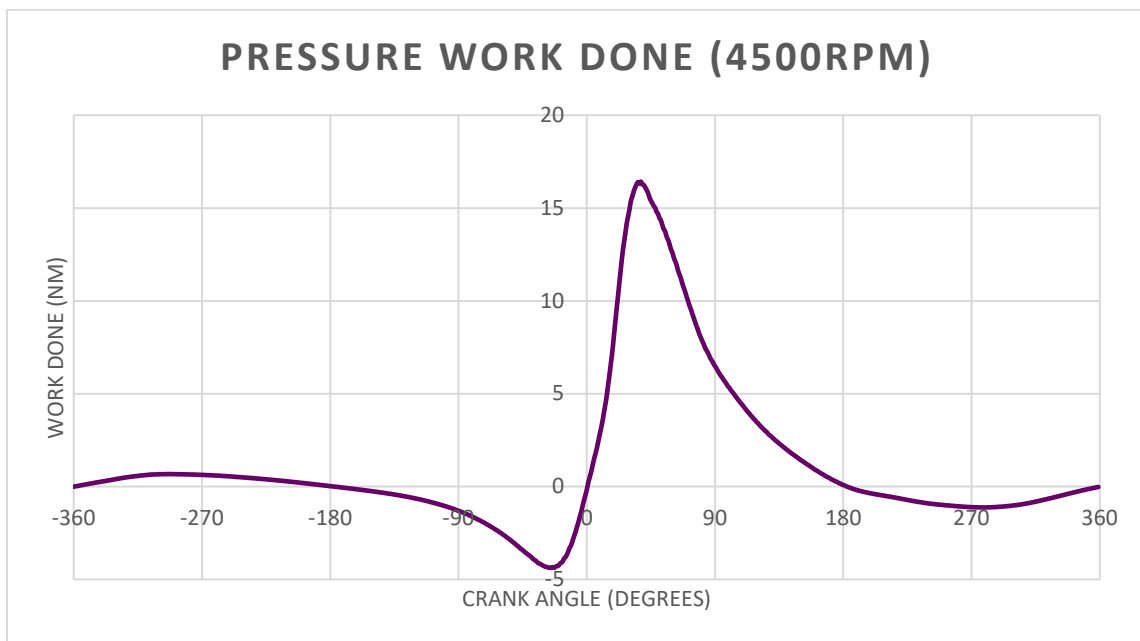


Figure 16-131: Work Done by the Pressure from Combustion at 4500RPM

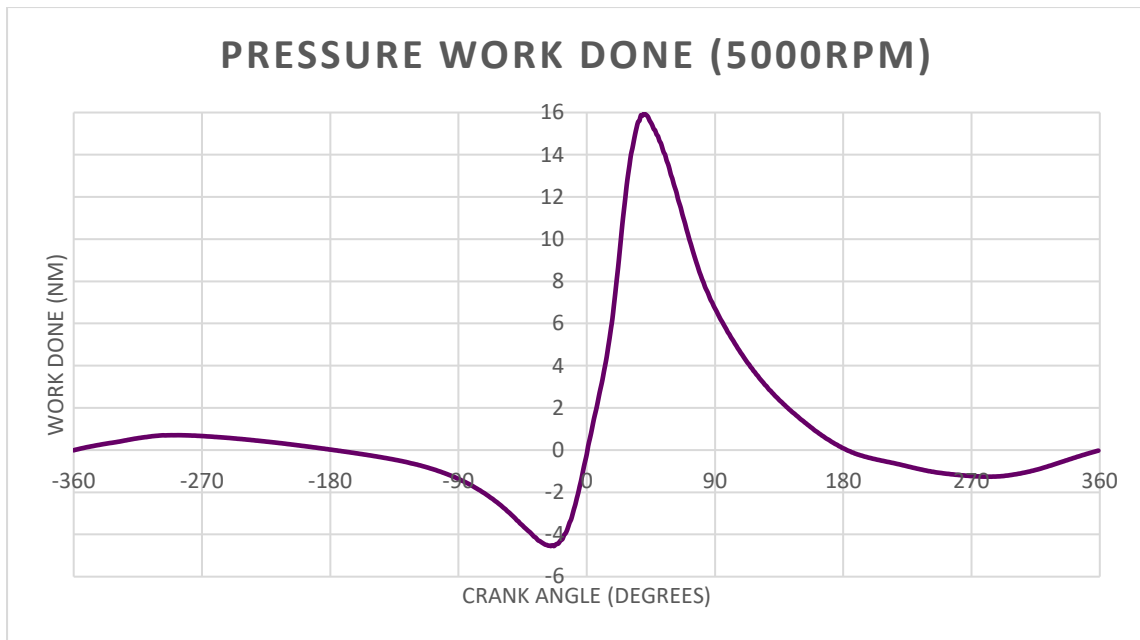


Figure 16-132: Work Done by the Pressure from Combustion at 5000RPM

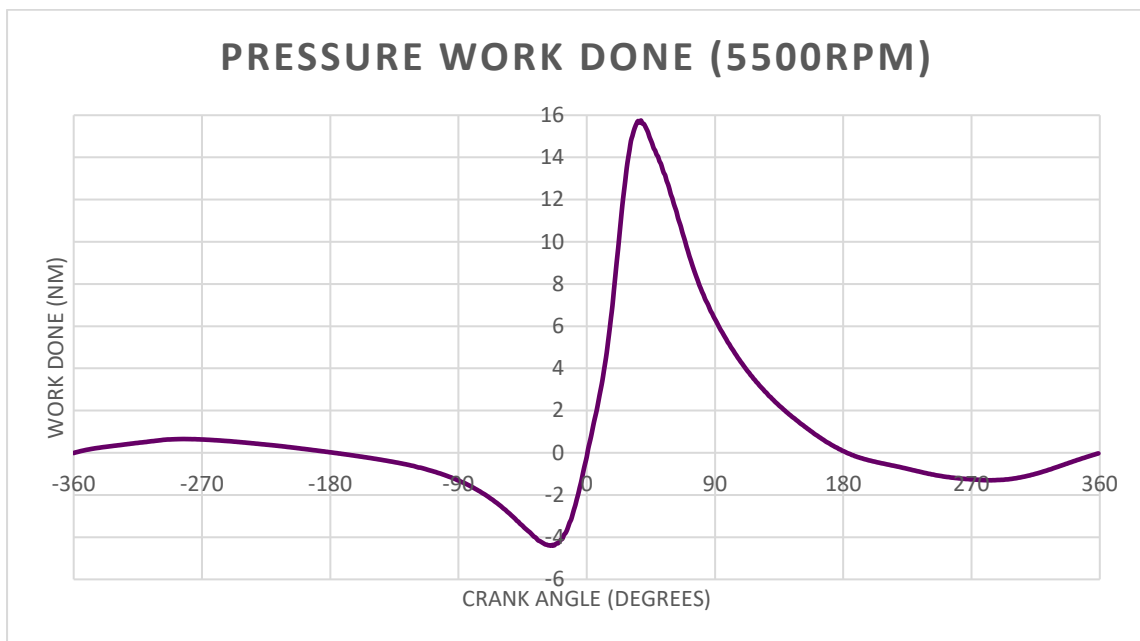


Figure 16-133: Work Done by the Pressure from Combustion at 5500RPM

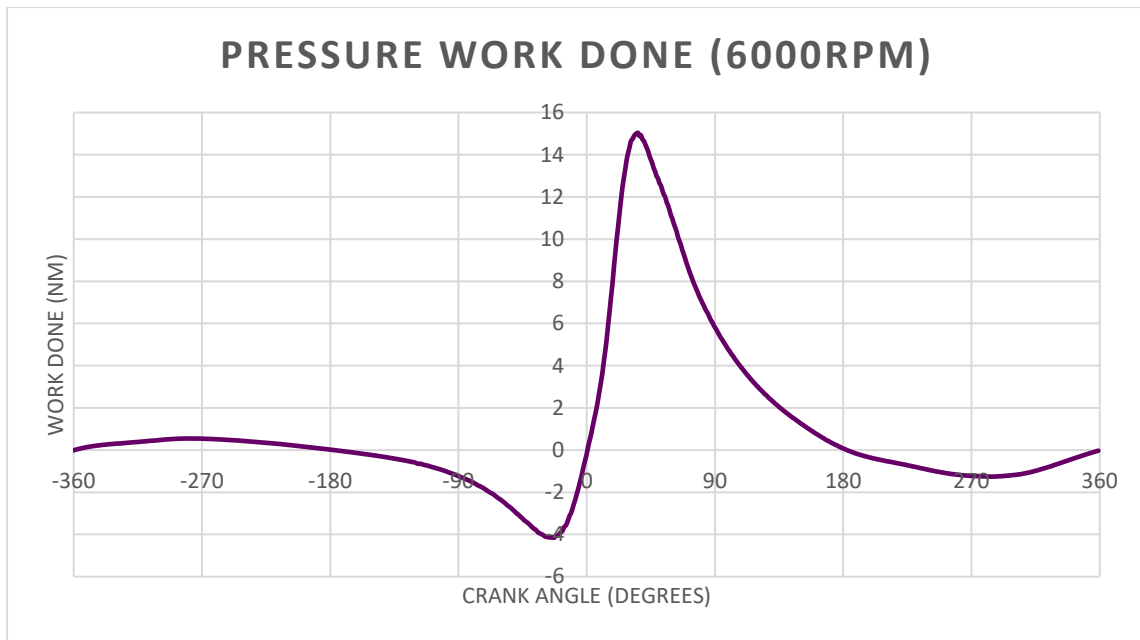


Figure 16-134: Work Done by the Pressure from Combustion at 6000RPM

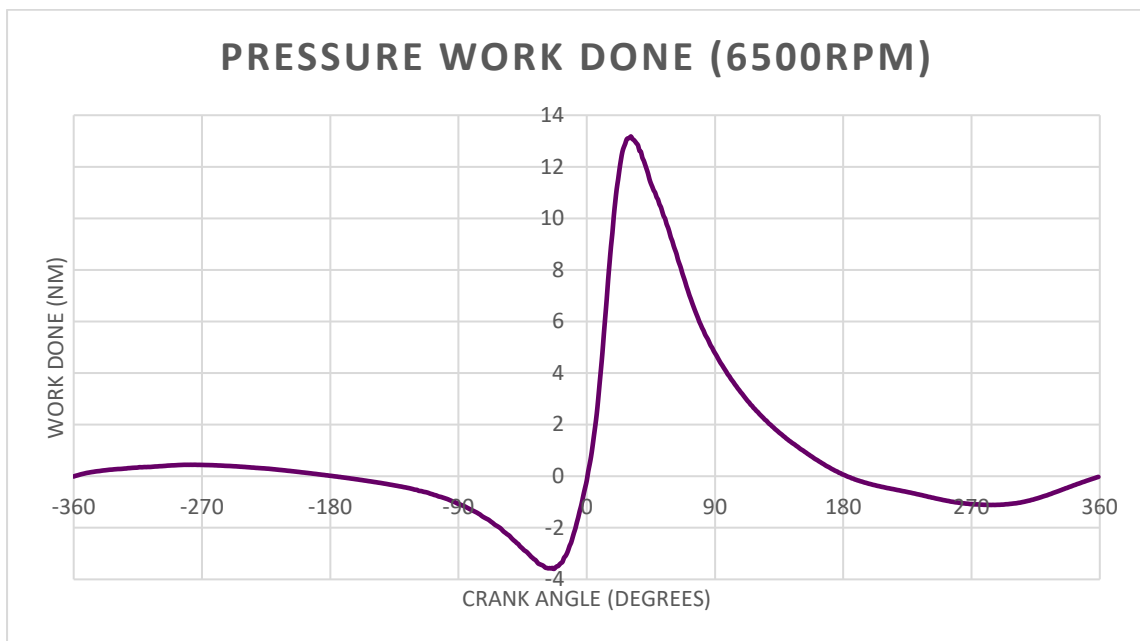


Figure 16-135: Work Done by the Pressure from Combustion at 6500RPM

16.4.6 Side Force

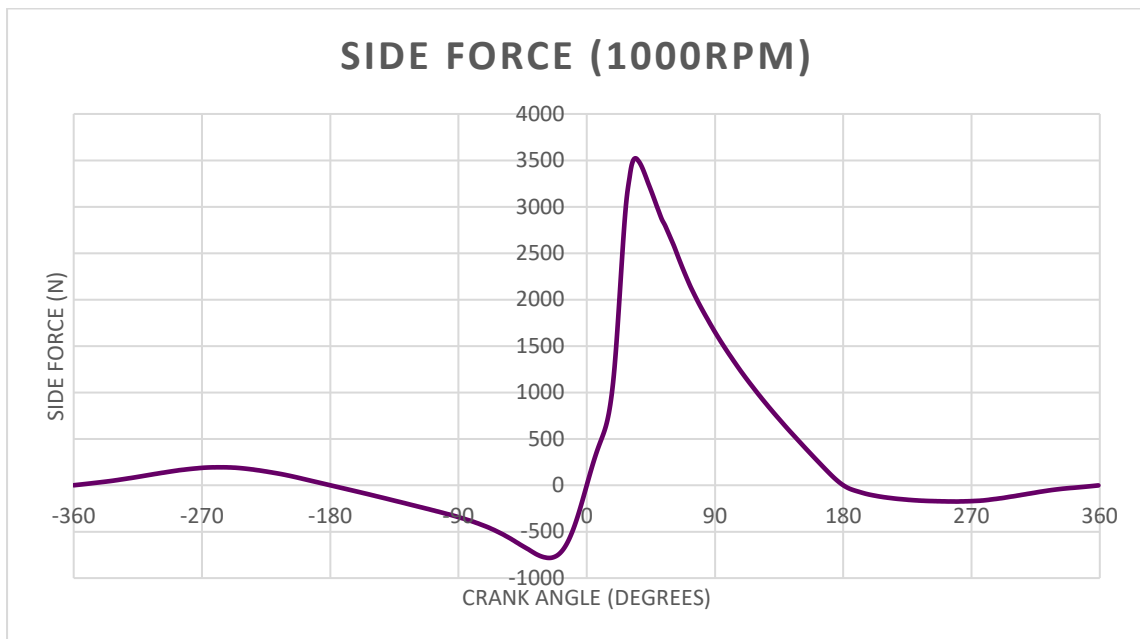


Figure 16-136: Side Force Acting on the Piston Skirt at 1000RPM

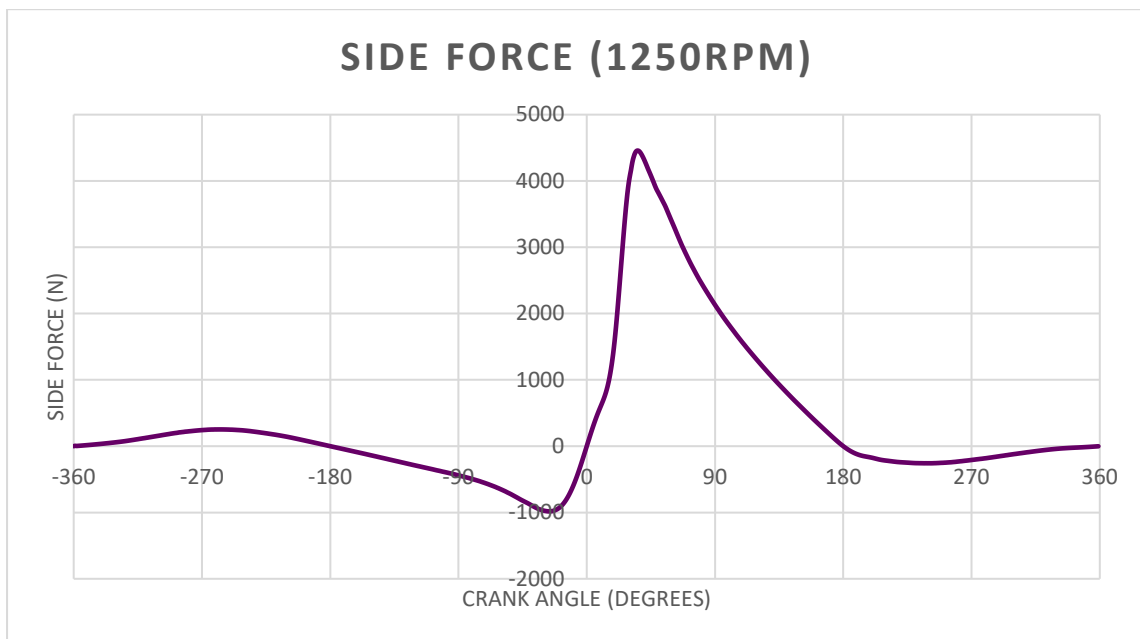


Figure 16-137: Side Force Acting on the Piston Skirt at 1250RPM

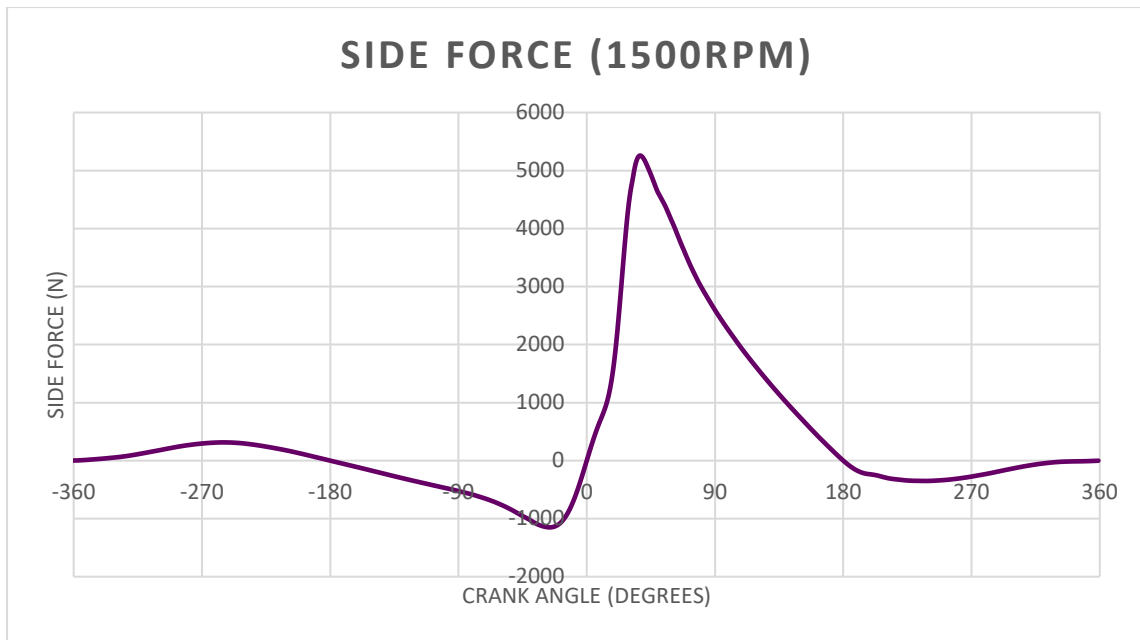


Figure 16-138: Side Force Acting on the Piston Skirt at 1500RPM

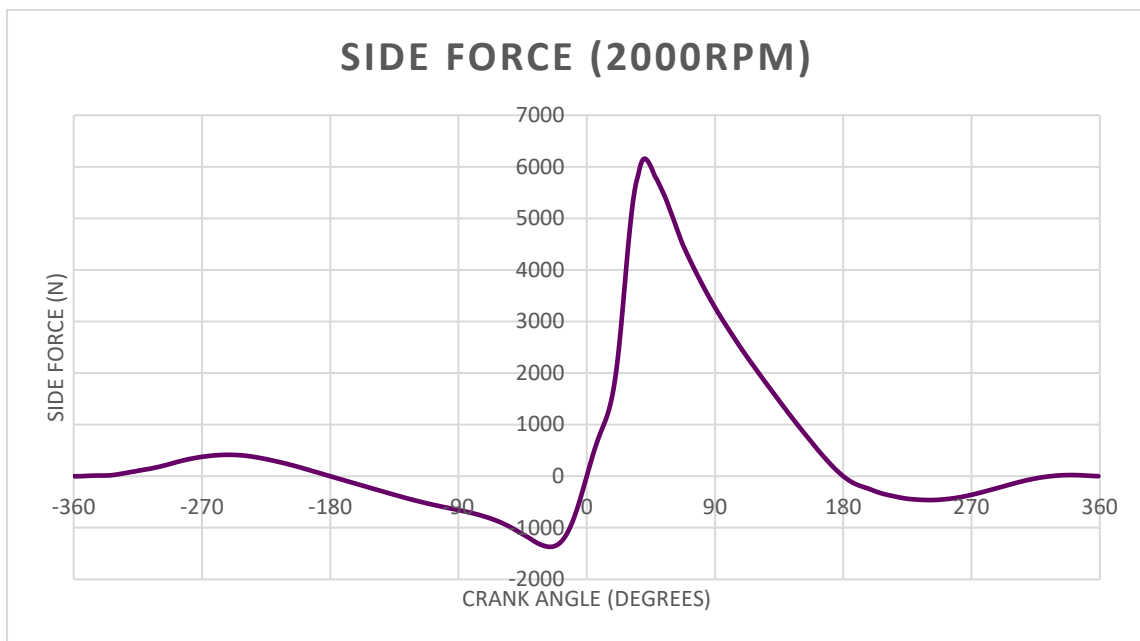


Figure 16-139: Side Force Acting on the Piston Skirt at 2000RPM

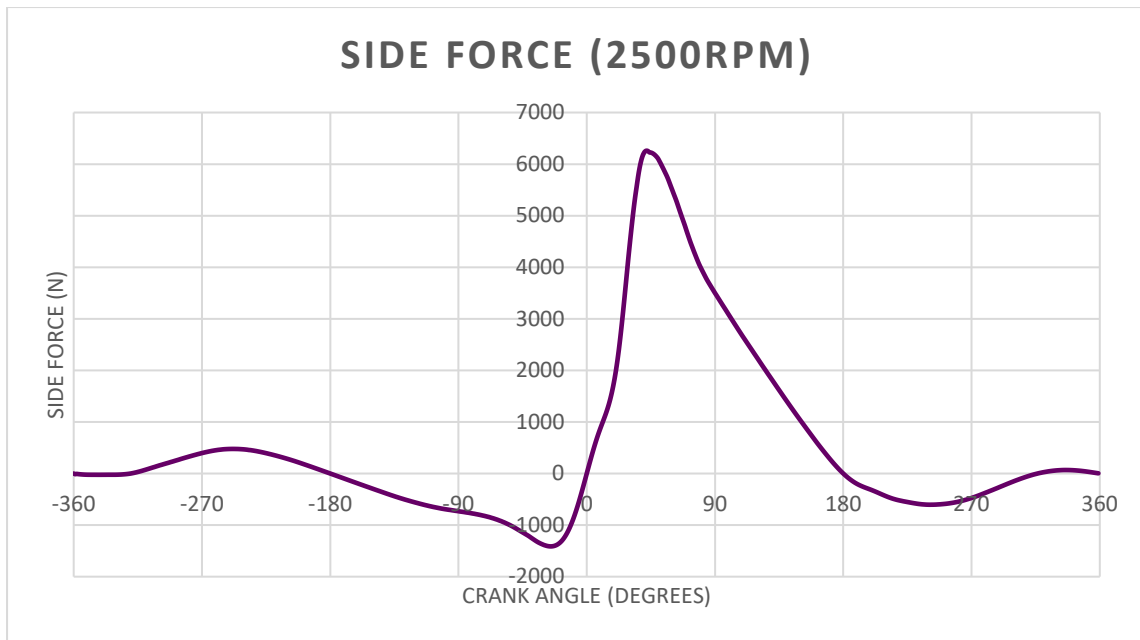


Figure 16-140: Side Force Acting on the Piston Skirt at 2500RPM

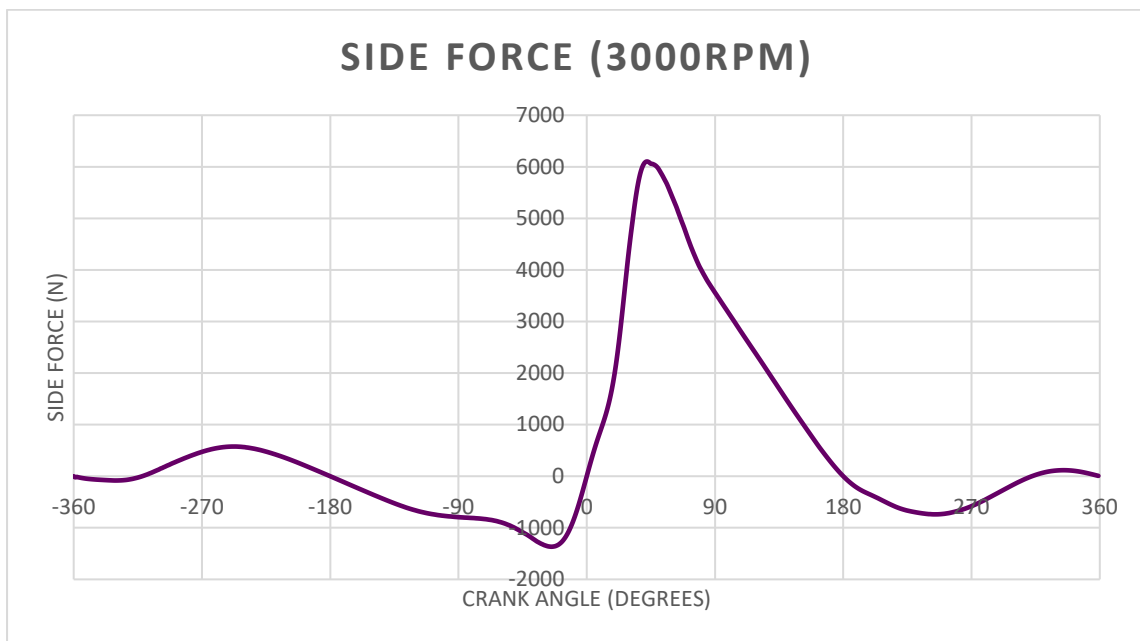


Figure 16-141: Side Force Acting on the Piston Skirt at 3000RPM

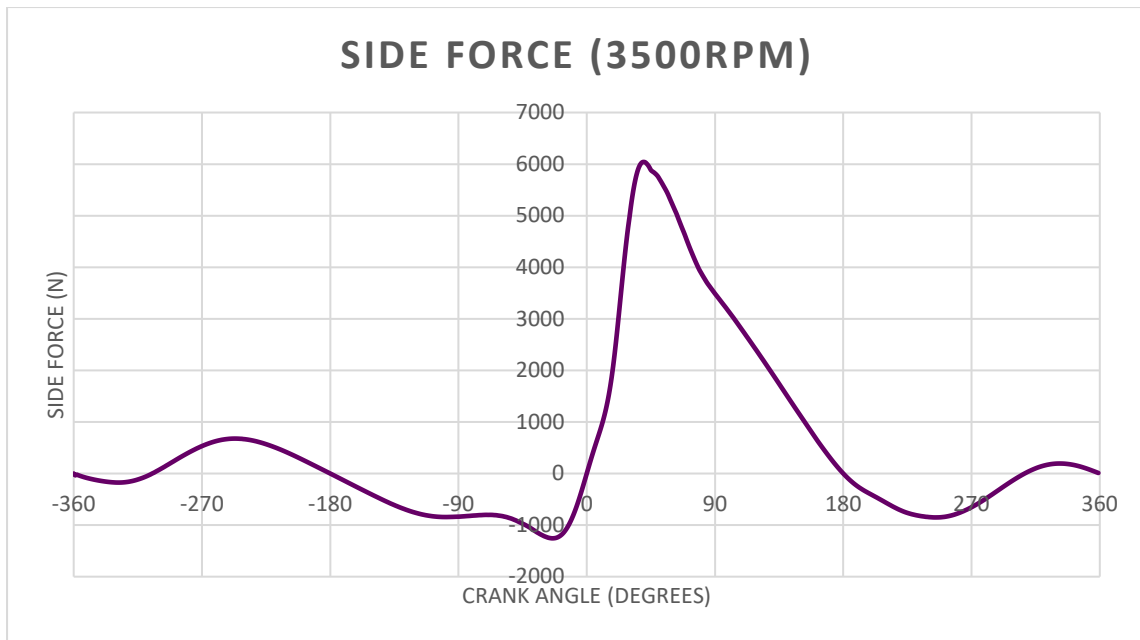


Figure 16-142: Side Force Acting on the Piston Skirt at 3500RPM

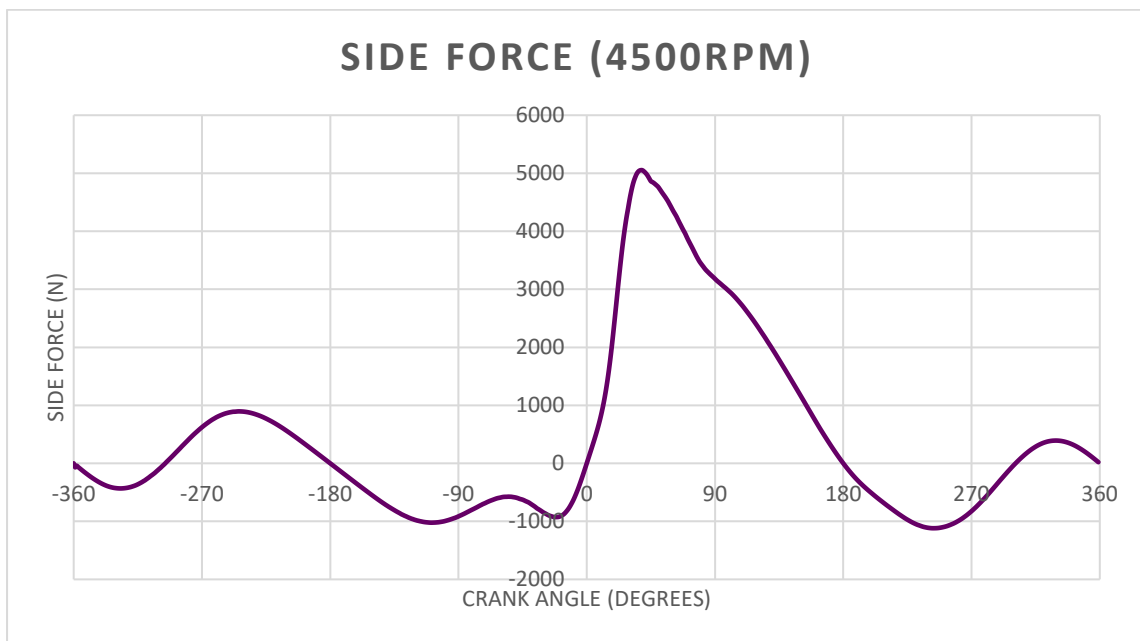


Figure 16-143: Side Force Acting on the Piston Skirt at 4500RPM

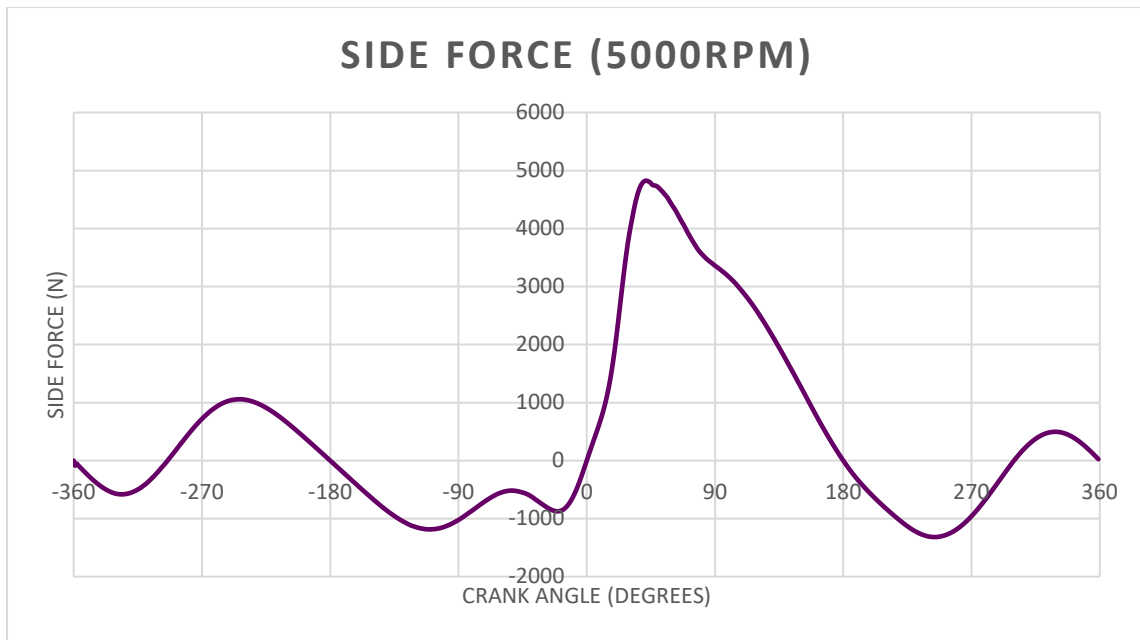


Figure 16-144: Side Force Acting on the Piston Skirt at 5000RPM

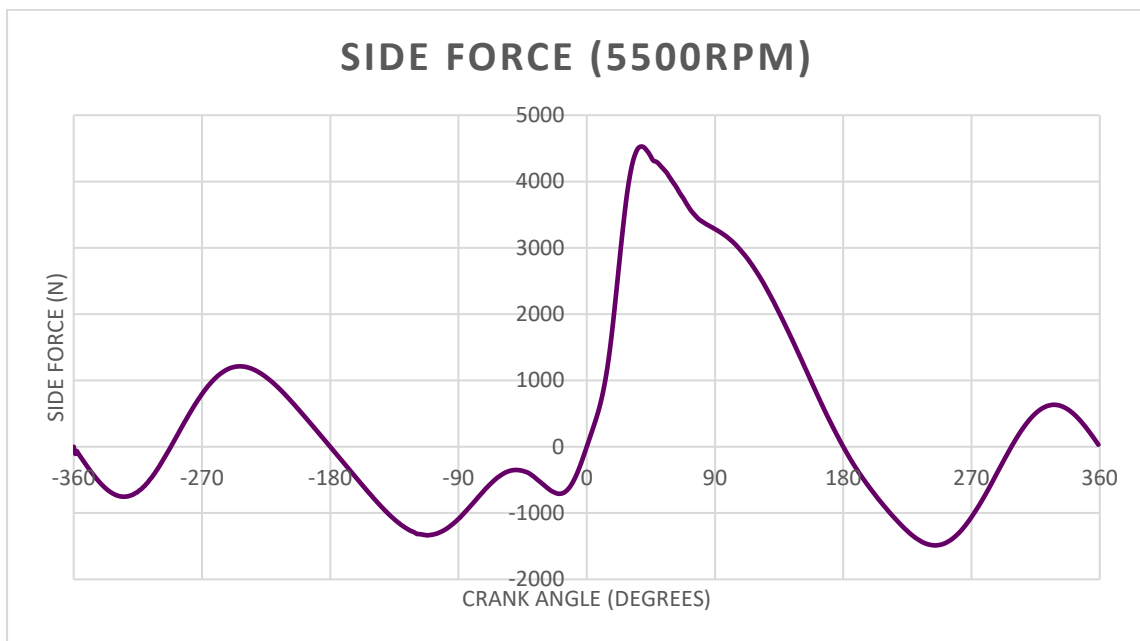


Figure 16-145: Side Force Acting on the Piston Skirt at 5500RPM

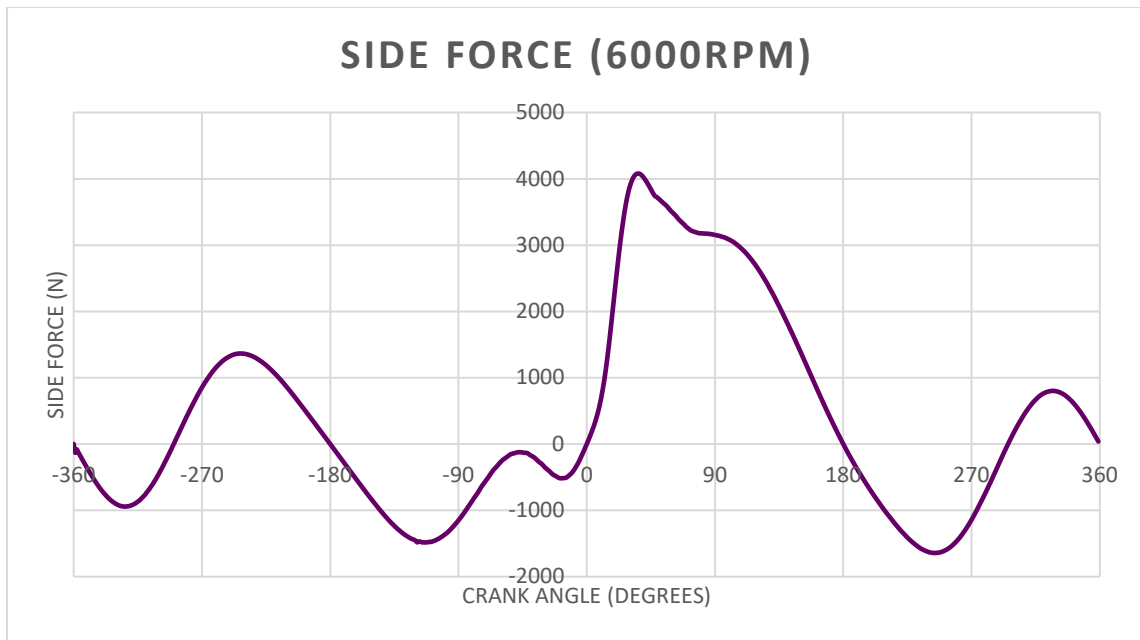


Figure 16-146: Side Force Acting on the Piston Skirt at 6000RPM

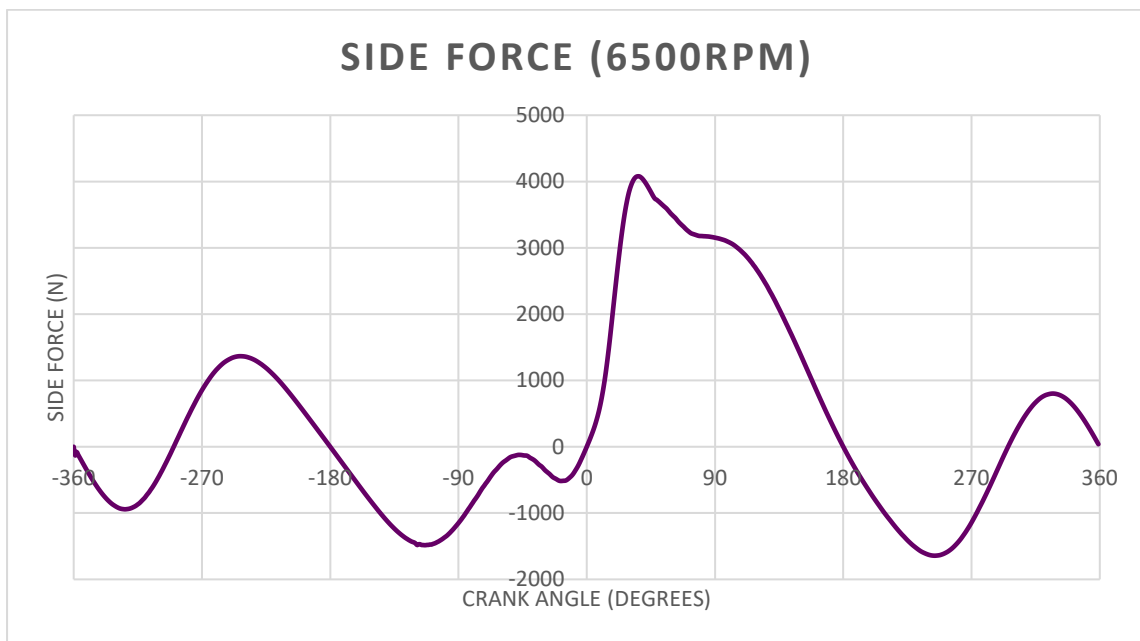


Figure 16-147: Side Force Acting on the Piston Skirt at 6500RPM

16.4.7 Friction Force

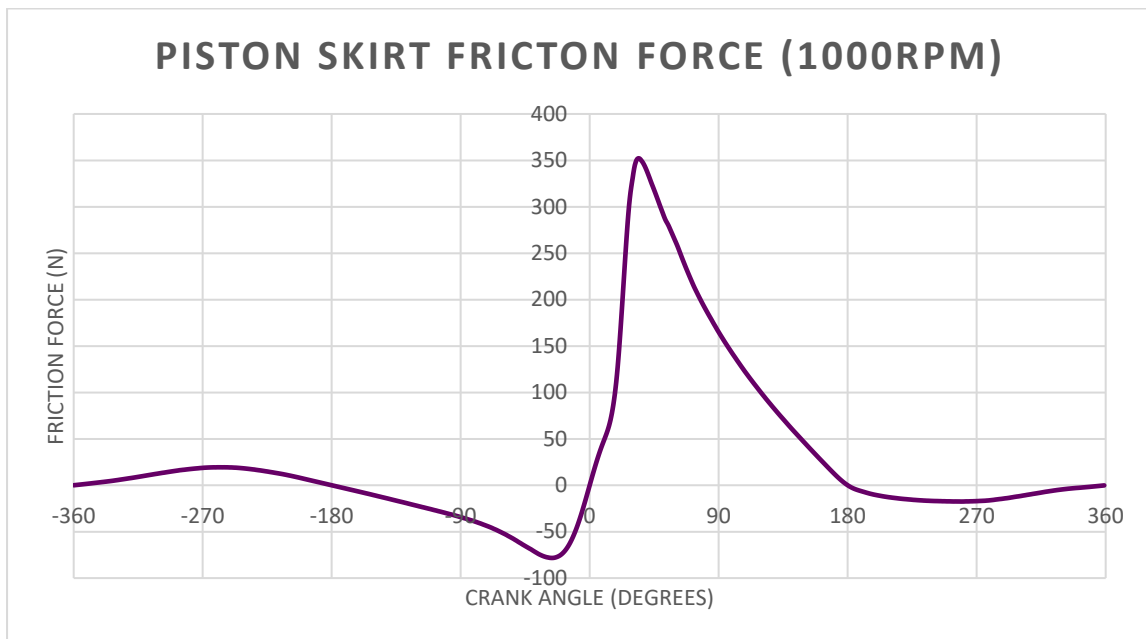


Figure 16-148: Friction Force from the Piston Skirt against the Cylinder Wall at 1000RPM

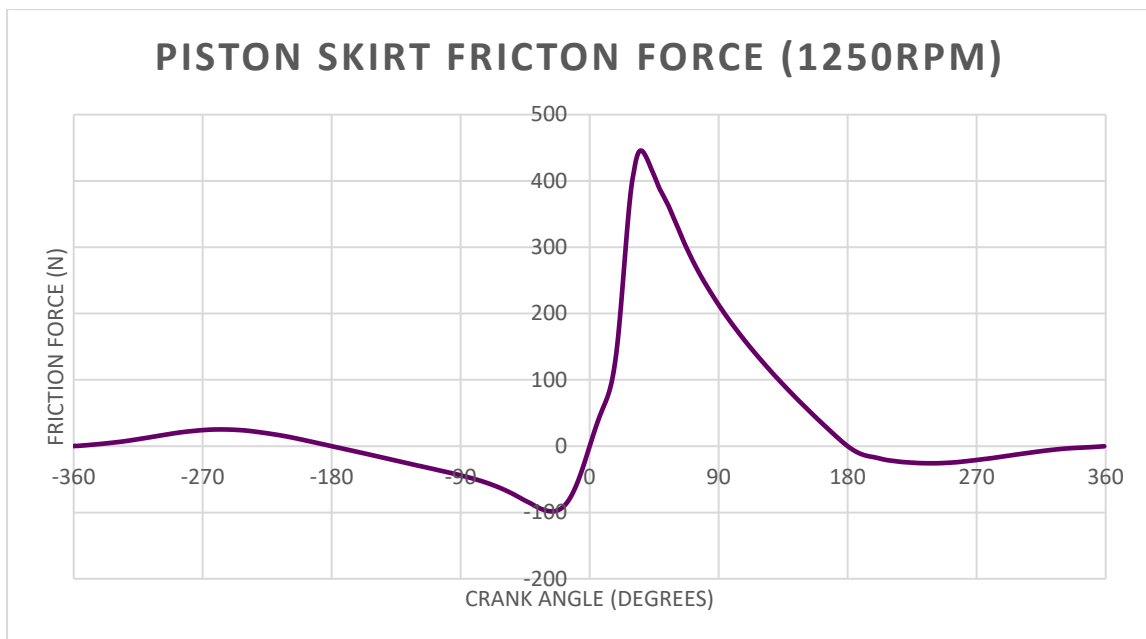


Figure 16-149: Friction Force from the Piston Skirt against the Cylinder Wall at 1250RPM

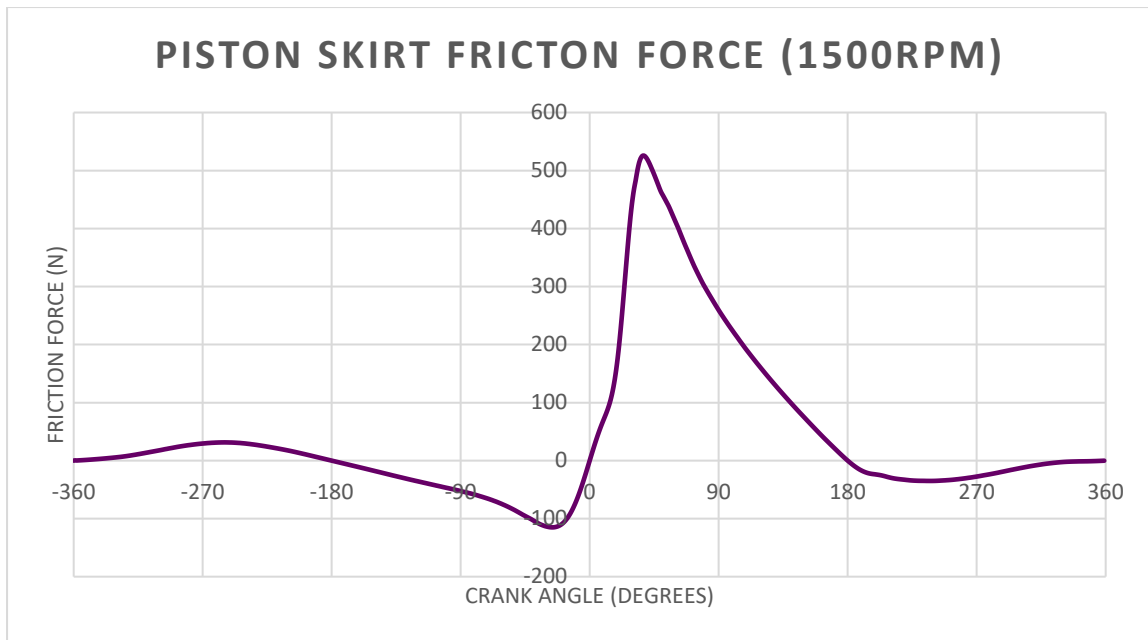


Figure 16-150: Friction Force from the Piston Skirt against the Cylinder Wall at 1500RPM

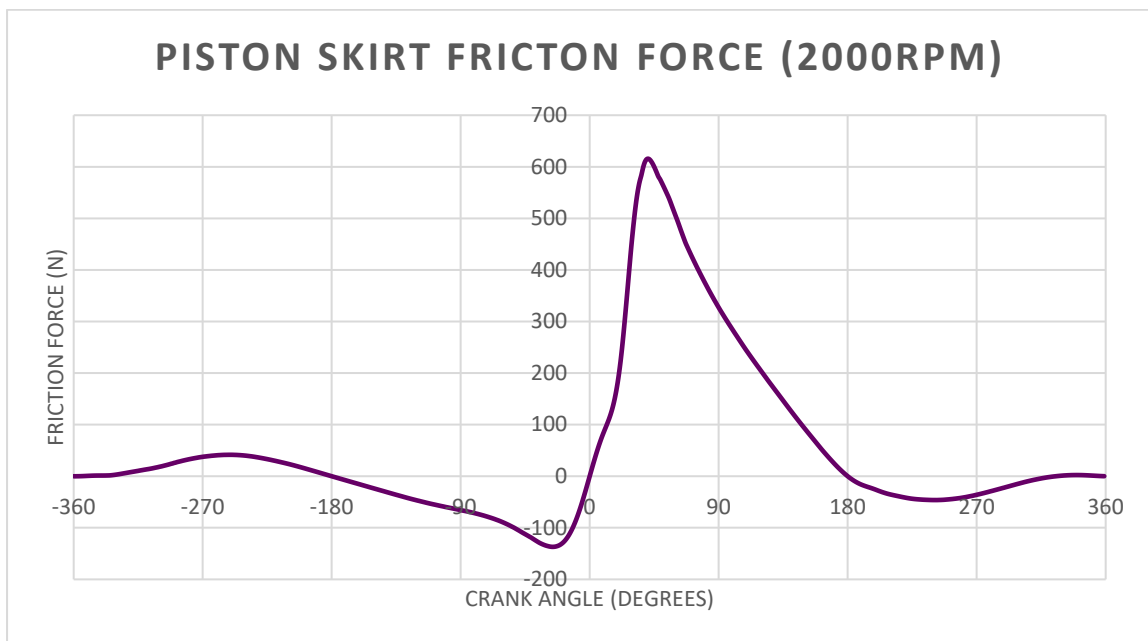


Figure 16-151: Friction Force from the Piston Skirt against the Cylinder Wall at 2000RPM

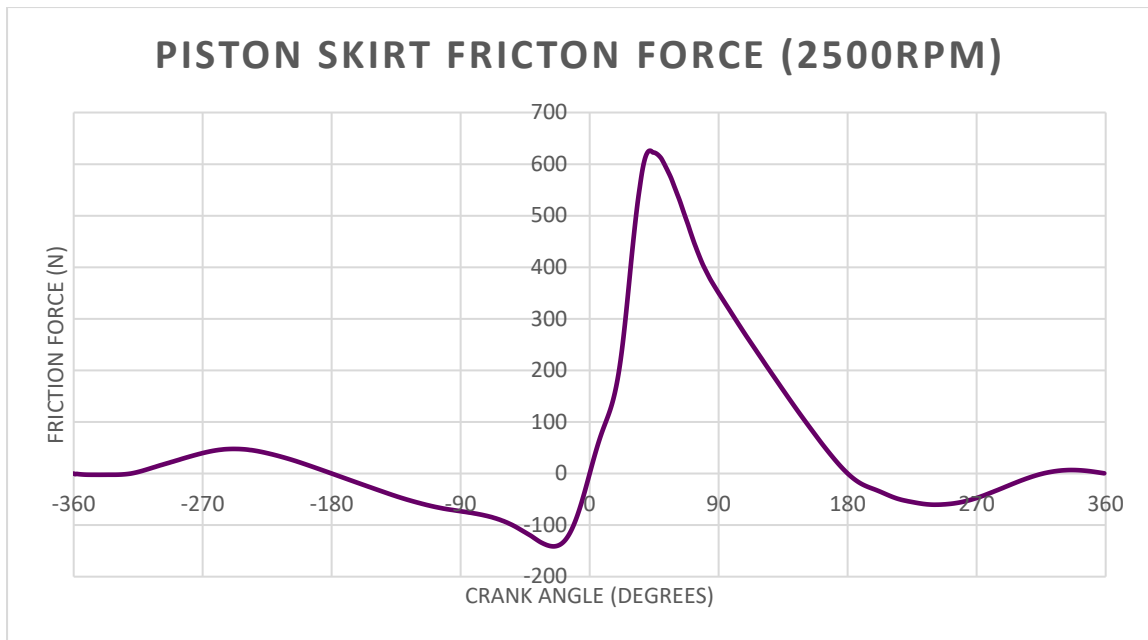


Figure 16-152: Friction Force from the Piston Skirt against the Cylinder Wall at 2500RPM

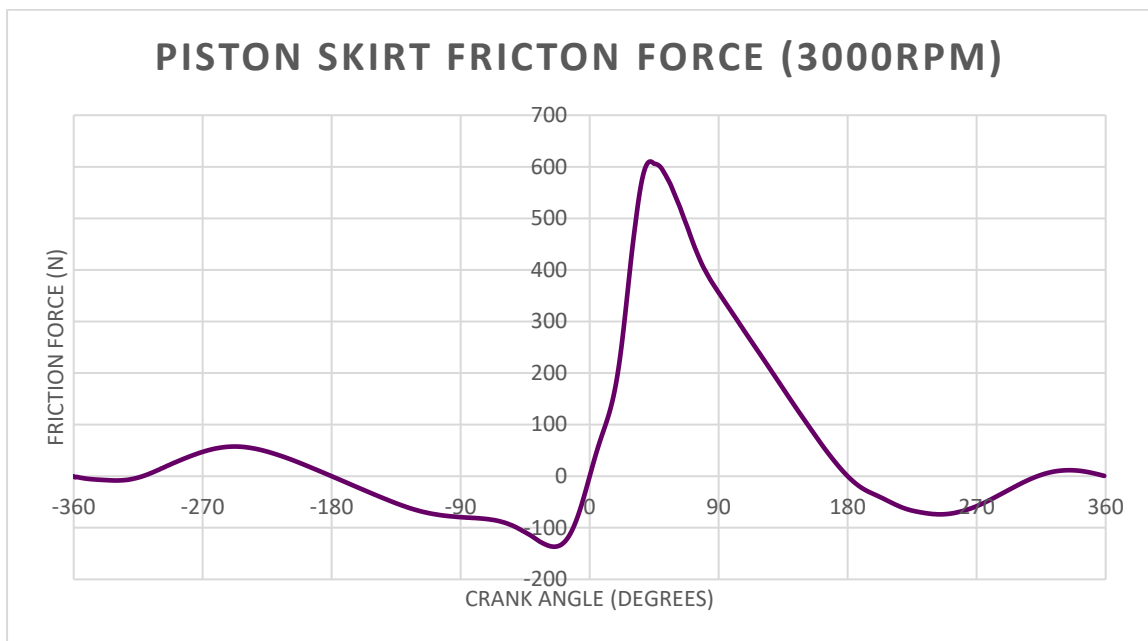


Figure 16-153: Friction Force from the Piston Skirt against the Cylinder Wall at 3000RPM

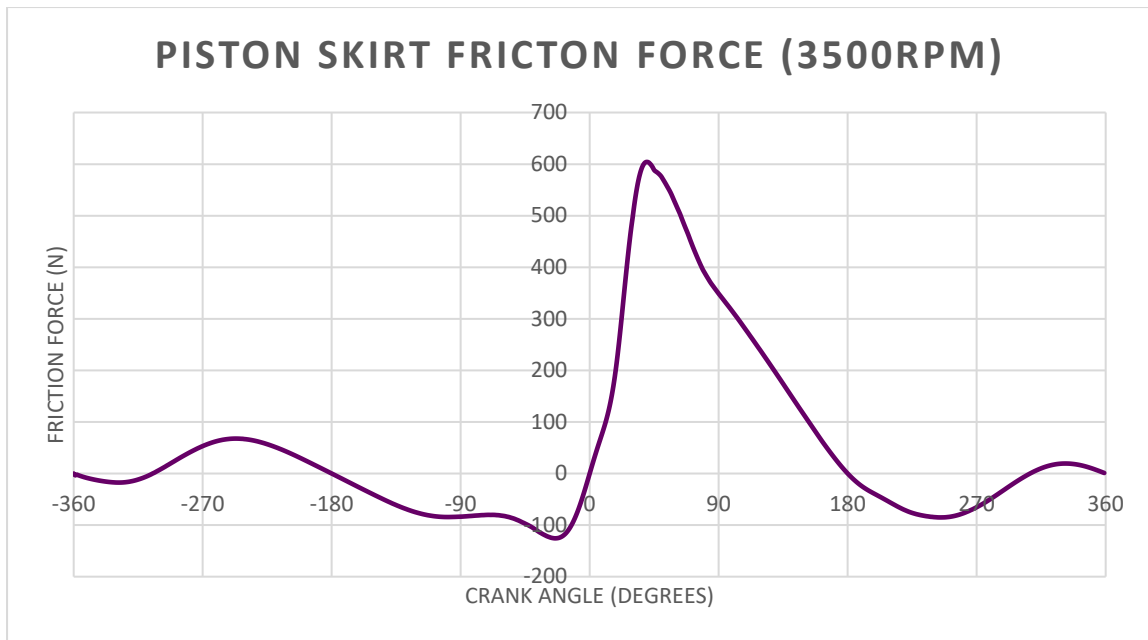


Figure 16-154: Friction Force from the Piston Skirt against the Cylinder Wall at 3500RPM

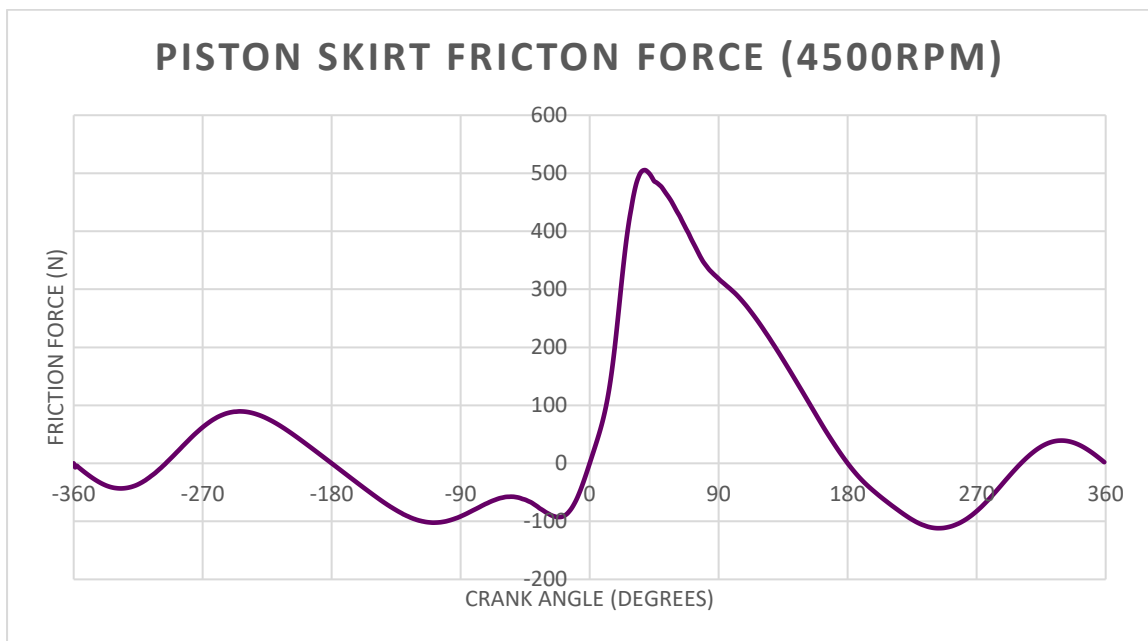


Figure 16-155: Friction Force from the Piston Skirt against the Cylinder Wall at 4500RPM

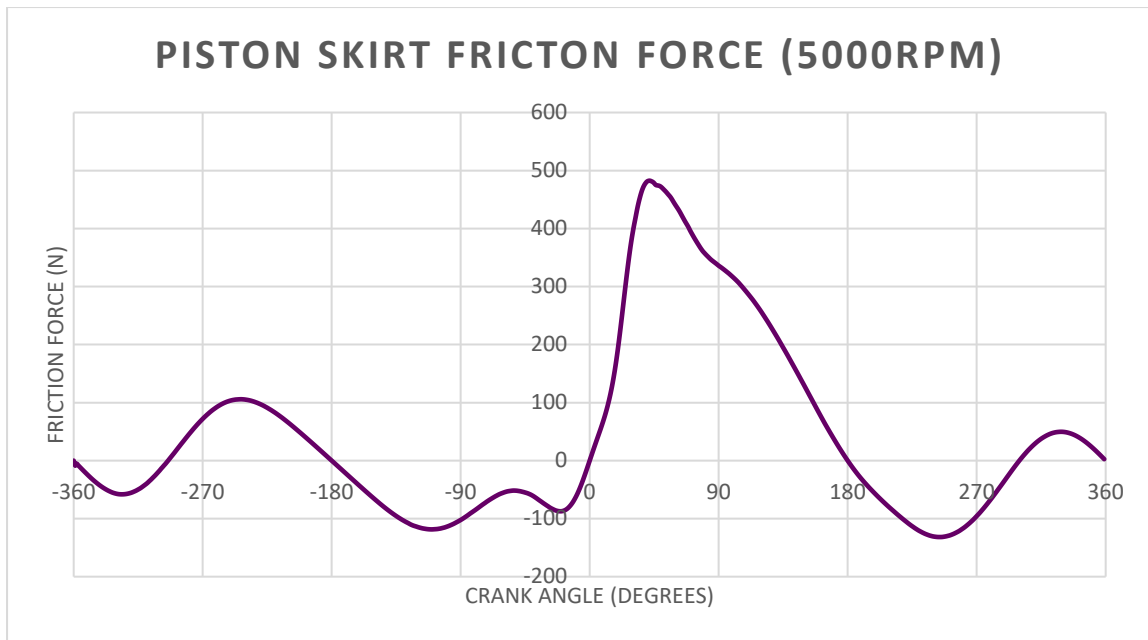


Figure 16-156: Friction Force from the Piston Skirt against the Cylinder Wall at 5000RPM

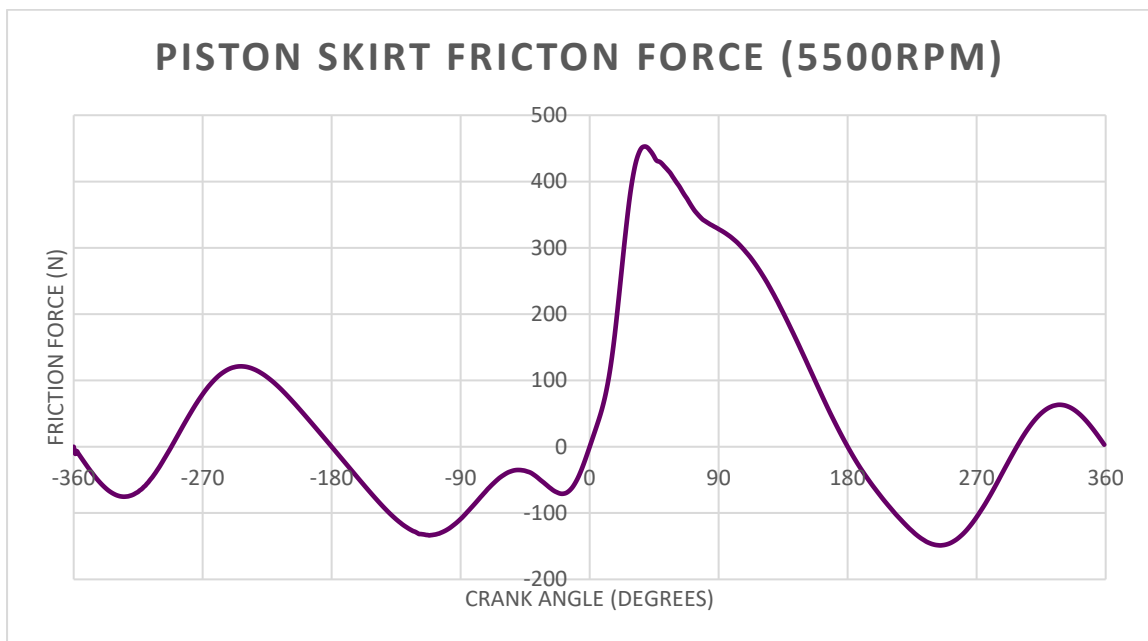


Figure 16-157: Friction Force from the Piston Skirt against the Cylinder Wall at 5500RPM

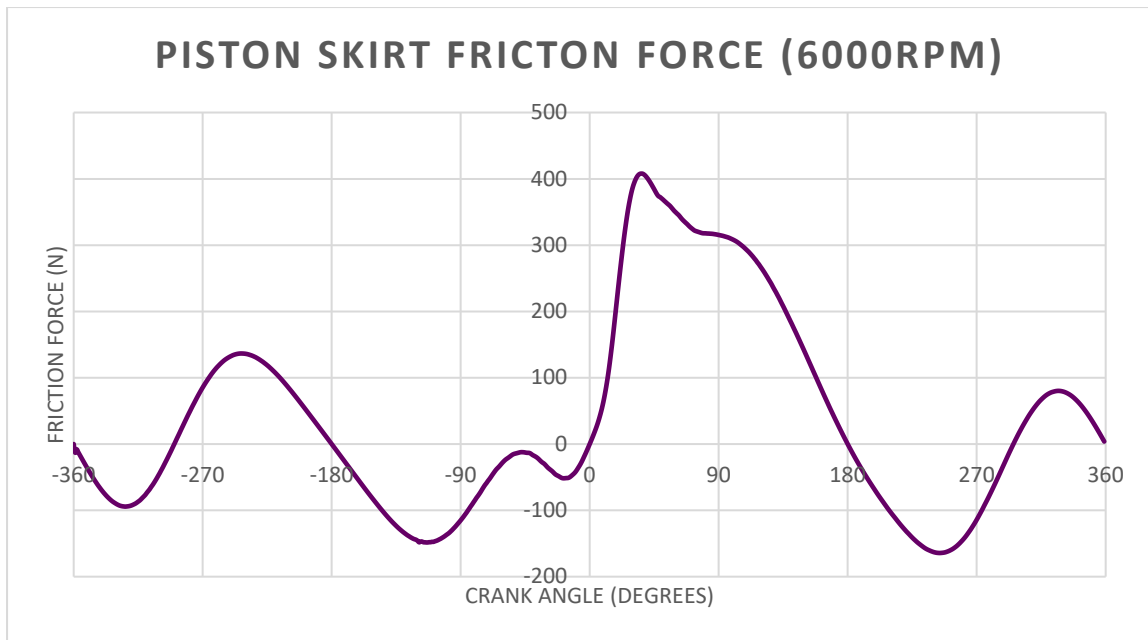


Figure 16-158: Friction Force from the Piston Skirt against the Cylinder Wall at 6000RPM

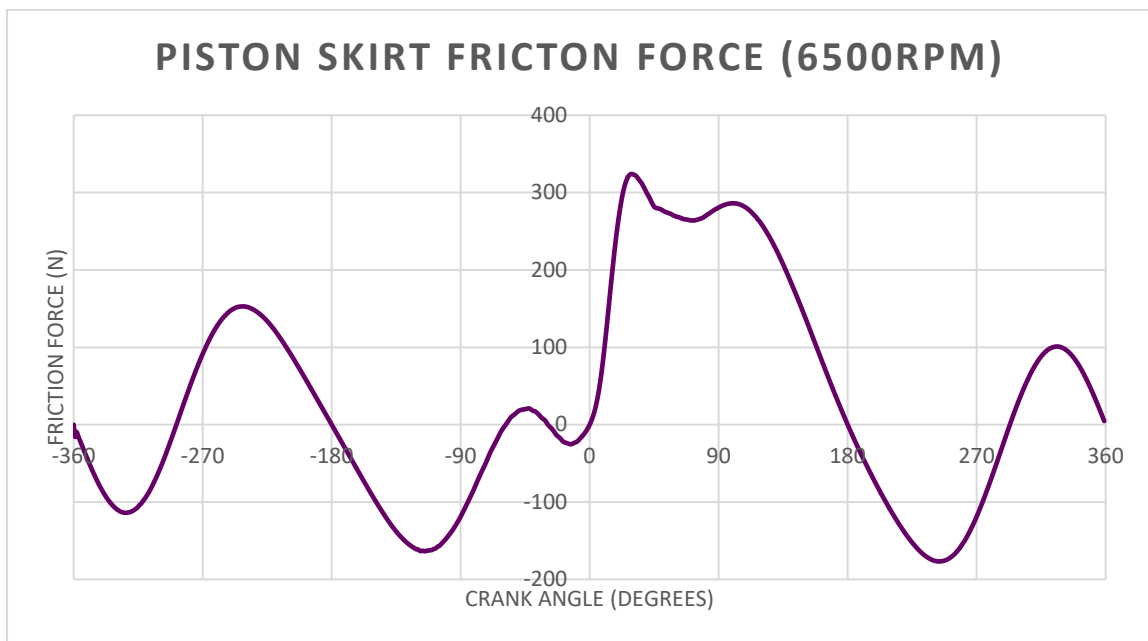


Figure 16-159: Friction Force from the Piston Skirt against the Cylinder Wall at 6500RPM

16.4.8 Work Done by the Piston Skirt

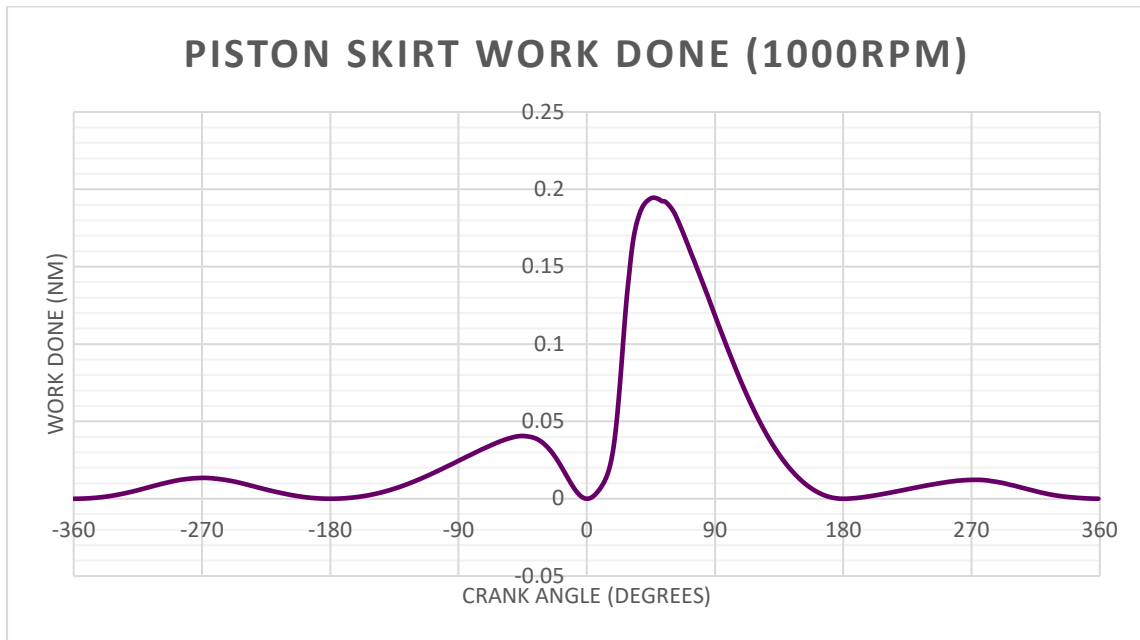


Figure 16-160: Work Done Against Combustion by the Piston Skirt at 1000RPM

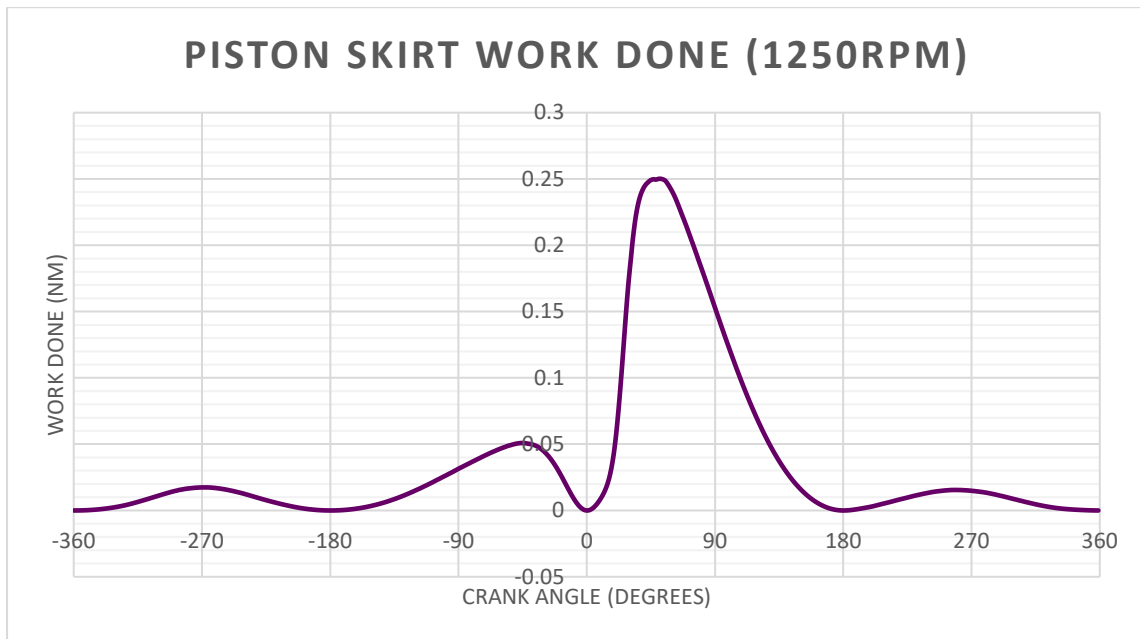


Figure 16-161: Work Done Against Combustion by the Piston Skirt at 1250RPM

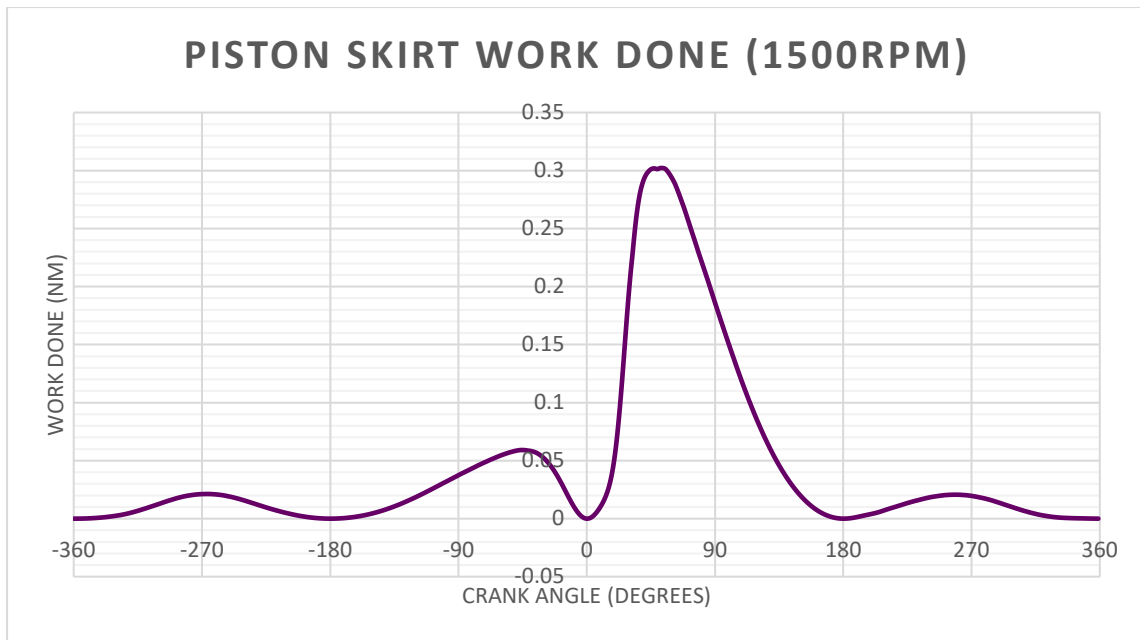


Figure 16-162: Work Done Against Combustion by the Piston Skirt at 1500RPM

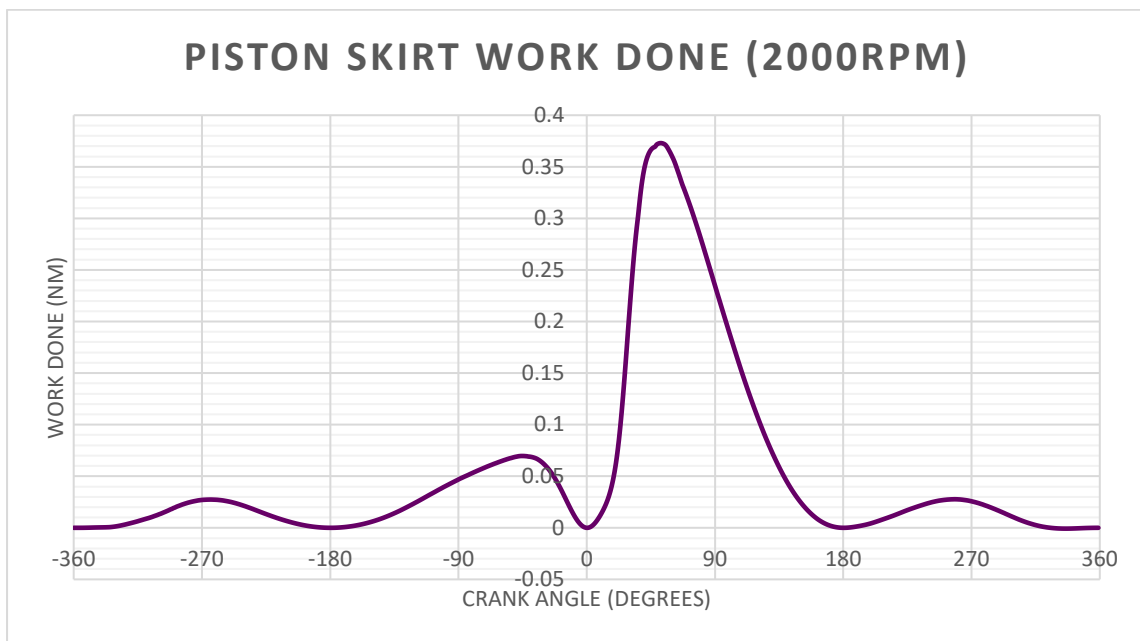


Figure 16-163: Work Done Against Combustion by the Piston Skirt at 2000RPM

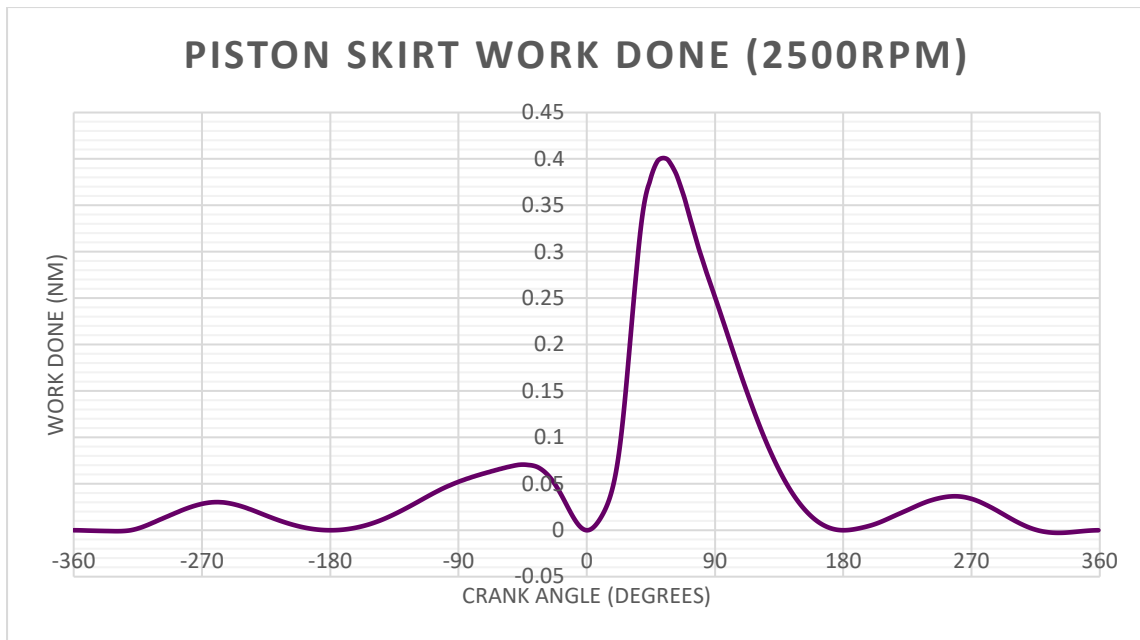


Figure 16-164: Work Done Against Combustion by the Piston Skirt at 2500RPM

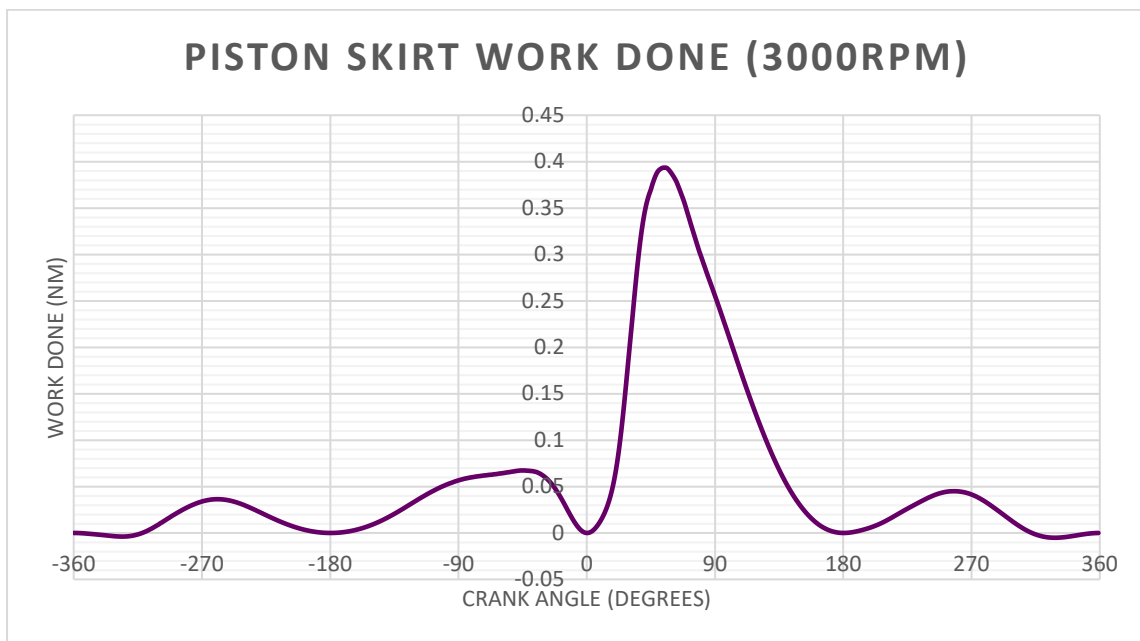


Figure 16-165: Work Done Against Combustion by the Piston Skirt at 3000RPM

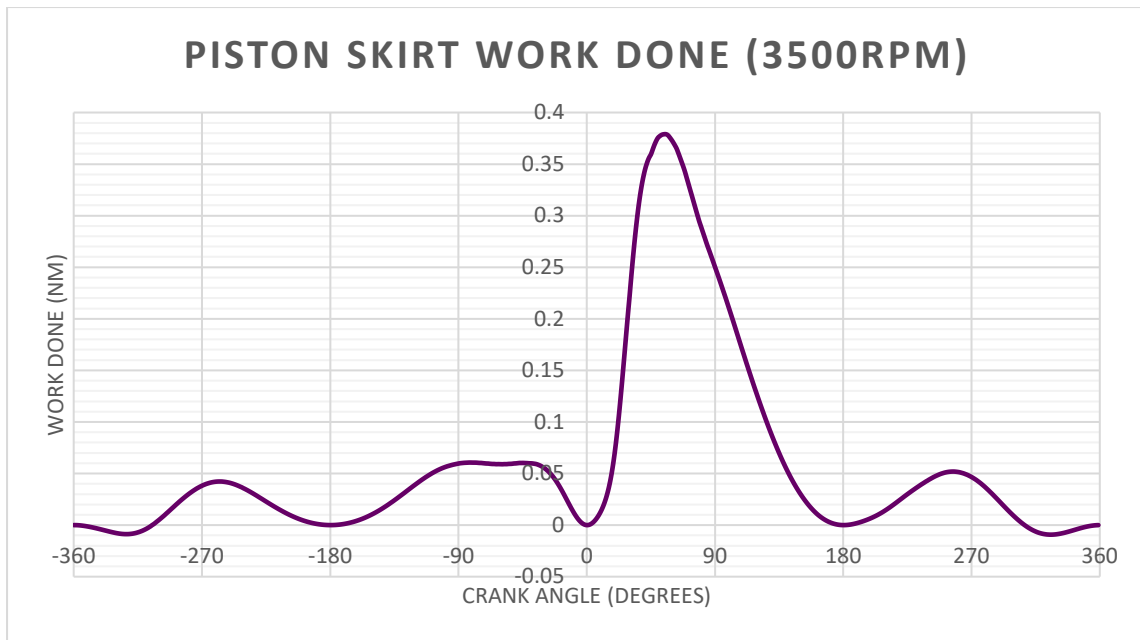


Figure 16-166: Work Done Against Combustion by the Piston Skirt at 3500RPM

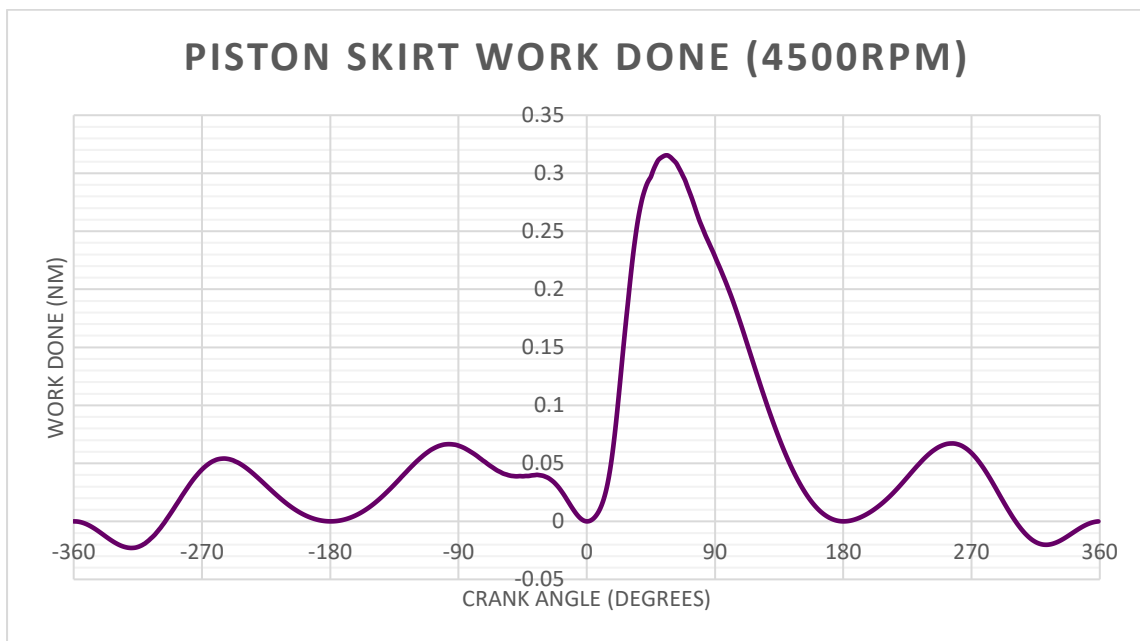


Figure 16-167: Work Done Against Combustion by the Piston Skirt at 4500RPM

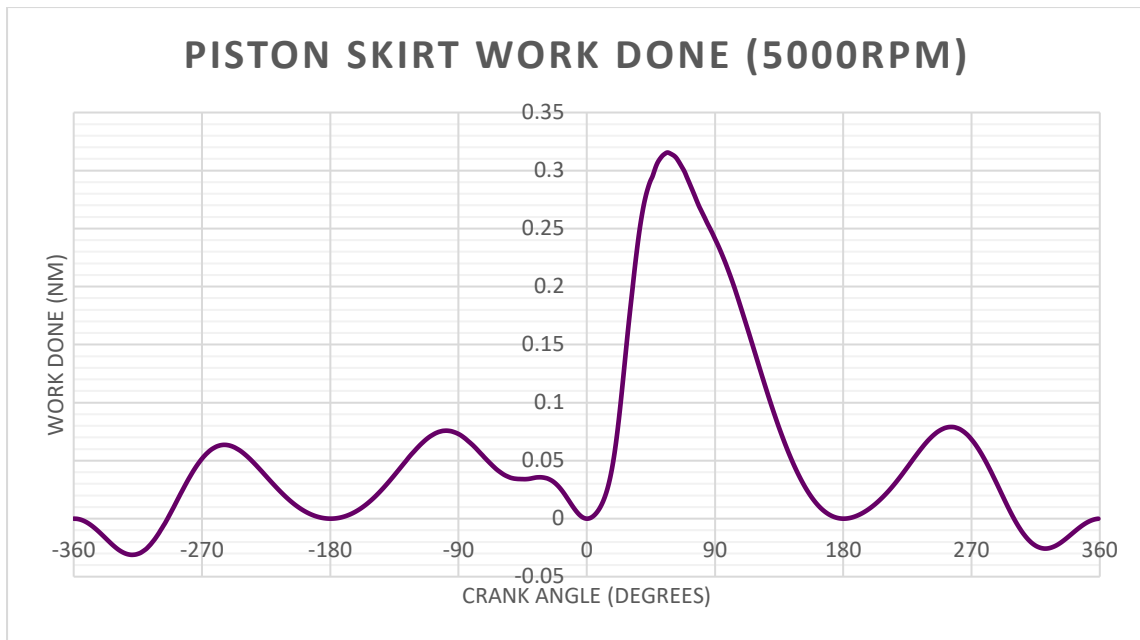


Figure 16-168: Work Done Against Combustion by the Piston Skirt at 5000RPM

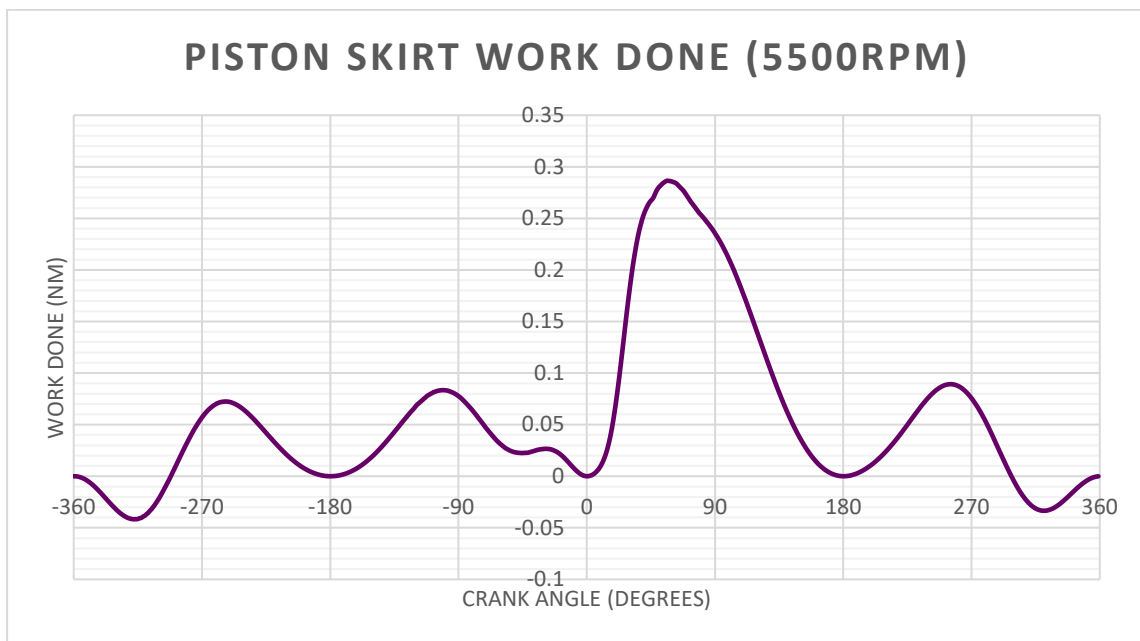


Figure 16-169: Work Done Against Combustion by the Piston Skirt at 5500RPM

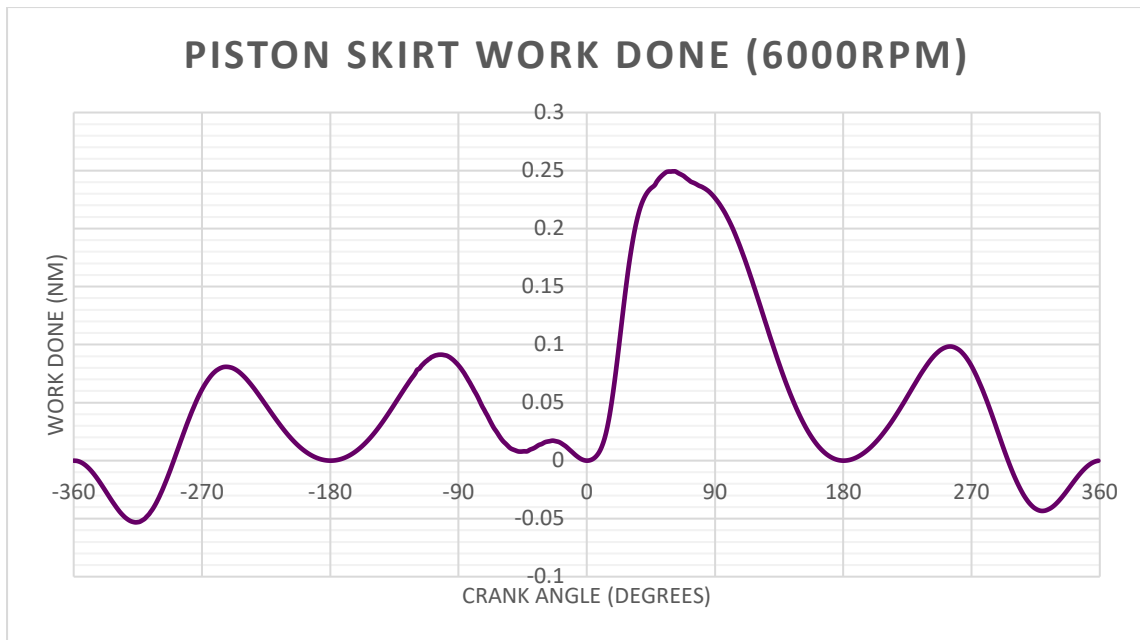


Figure 16-170: Work Done Against Combustion by the Piston Skirt at 6000RPM

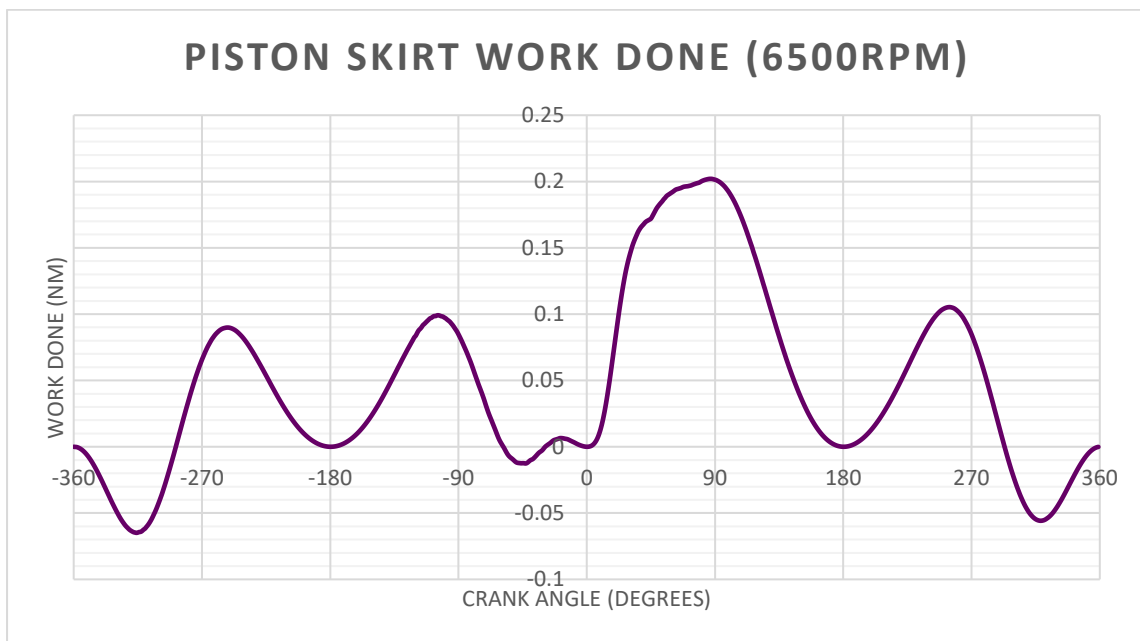


Figure 16-171: Work Done Against Combustion by the Piston Skirt at 6500RPM

16.4.9 Radial Force

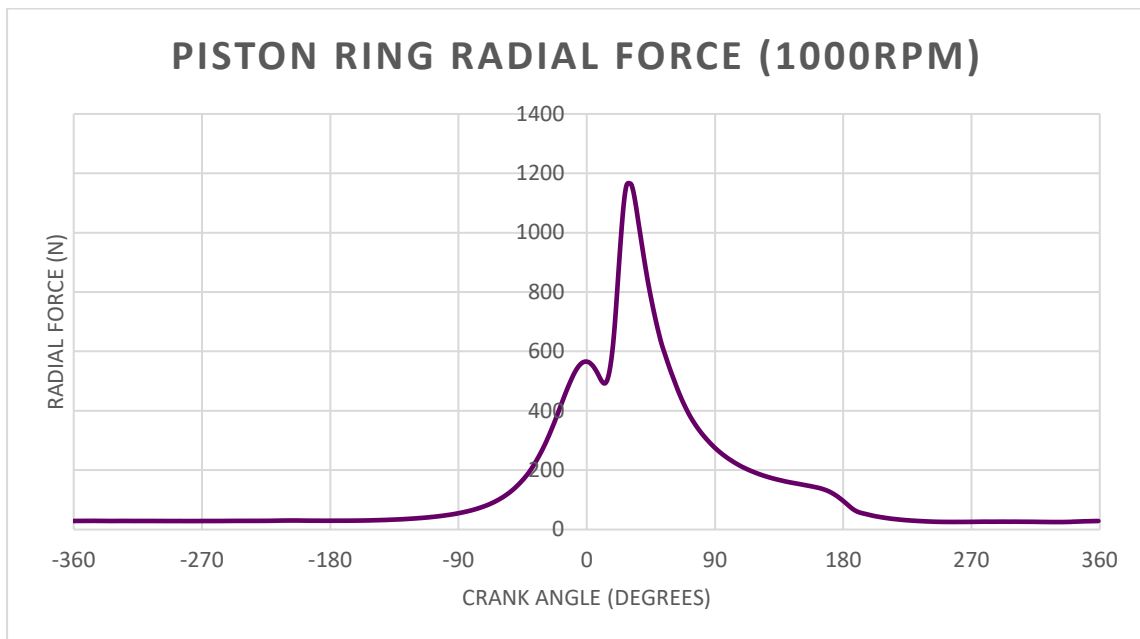


Figure 16-172: Radial Force on the Top Piston Ring at 1000RPM

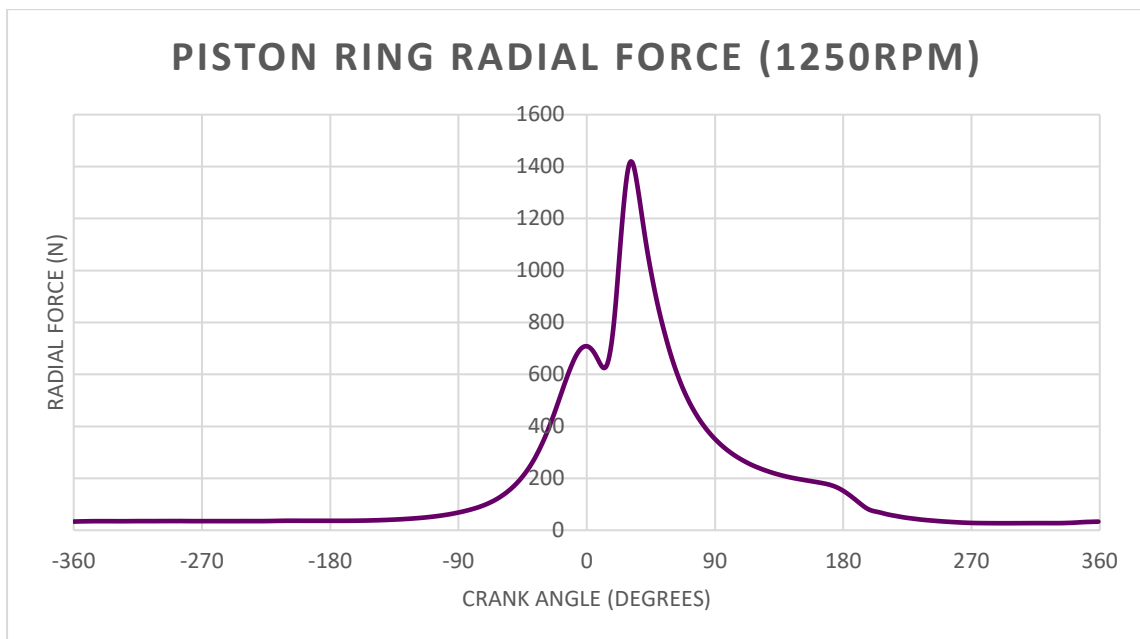


Figure 16-173: Radial Force on the Top Piston Ring at 1250RPM

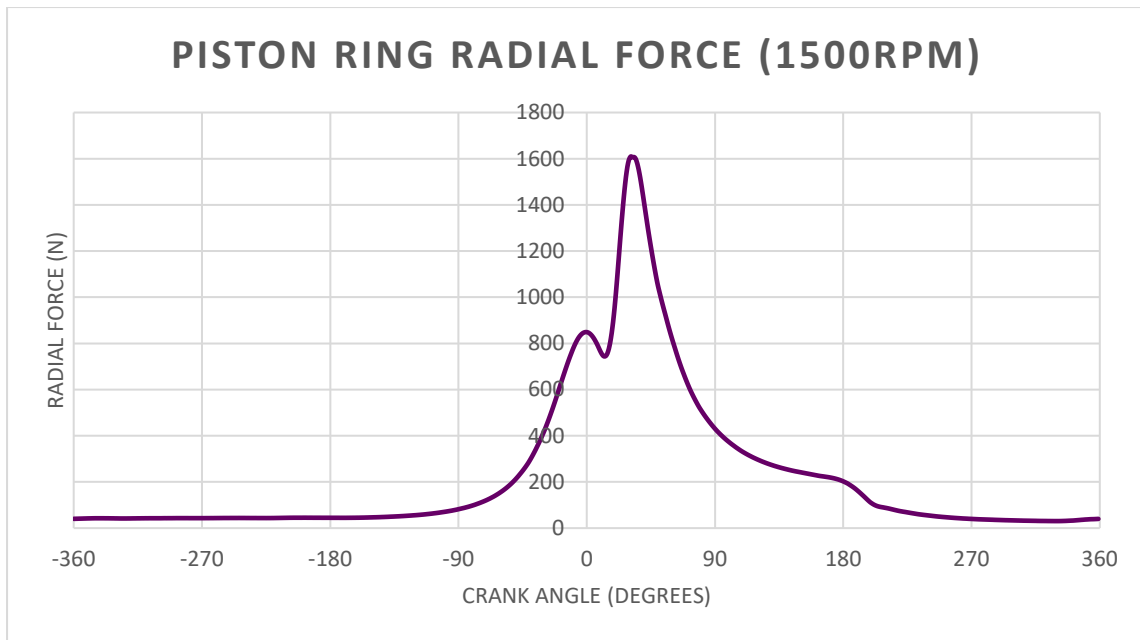


Figure 16-174: Radial Force on the Top Piston Ring at 1500RPM

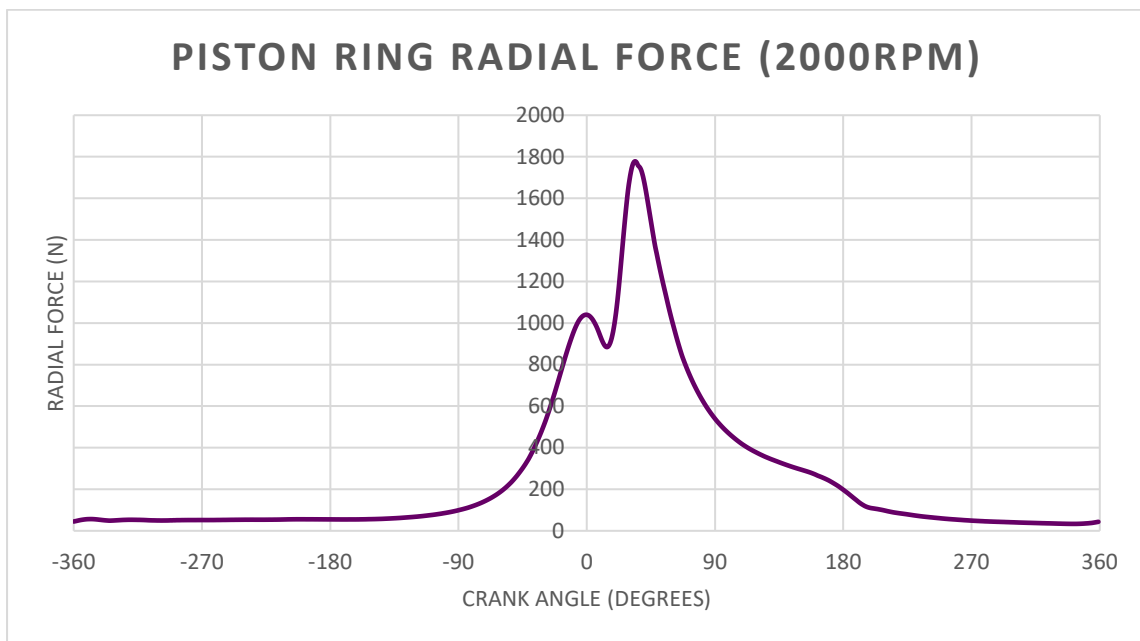


Figure 16-175: Radial Force on the Top Piston Ring at 2000RPM

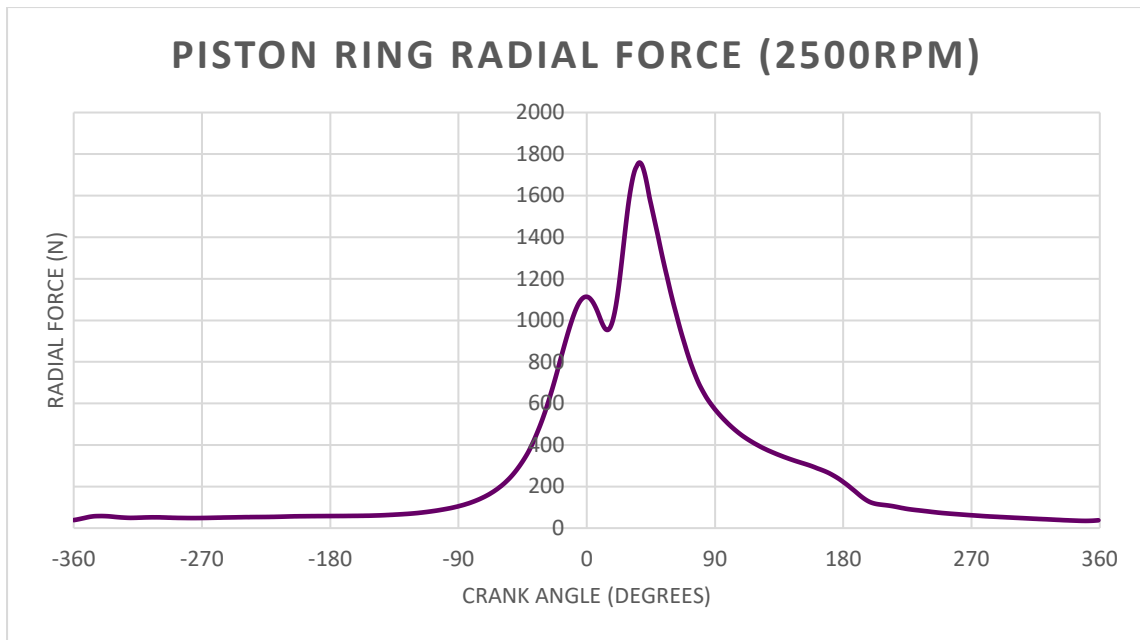


Figure 16-176: Radial Force on the Top Piston Ring at 2500RPM

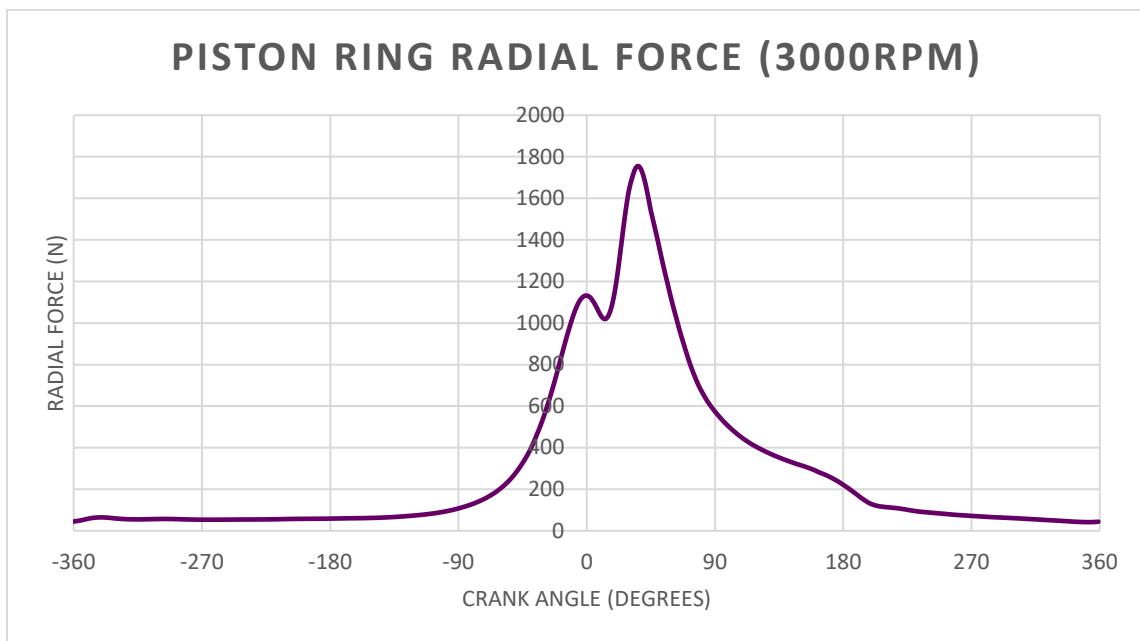


Figure 16-177: Radial Force on the Top Piston Ring at 3000RPM

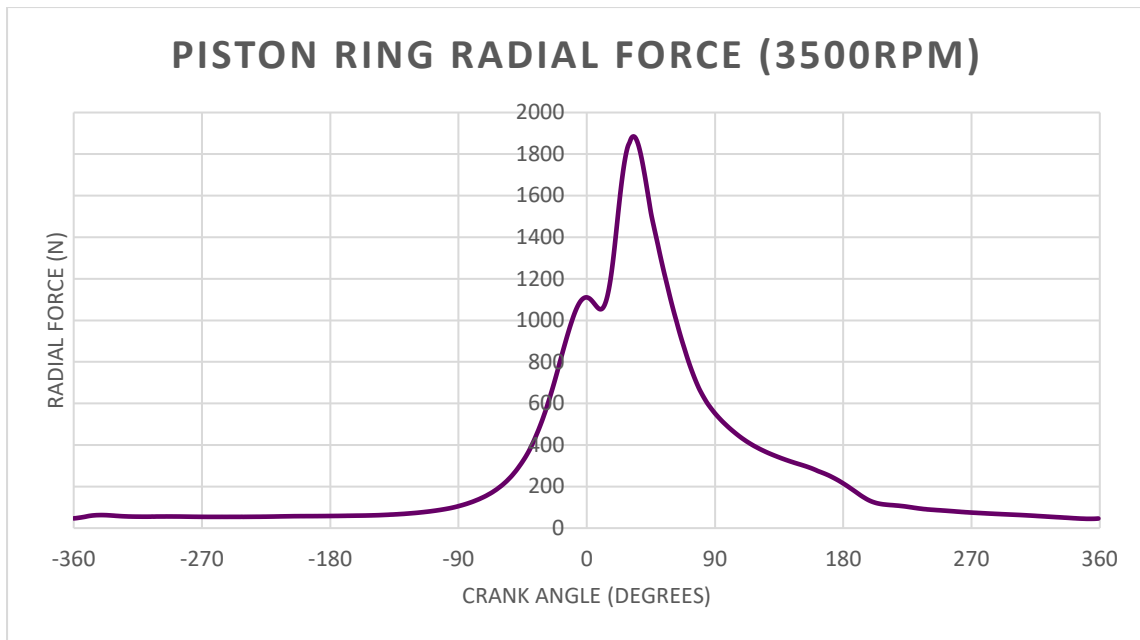


Figure 16-178: Radial Force on the Top Piston Ring at 3500RPM

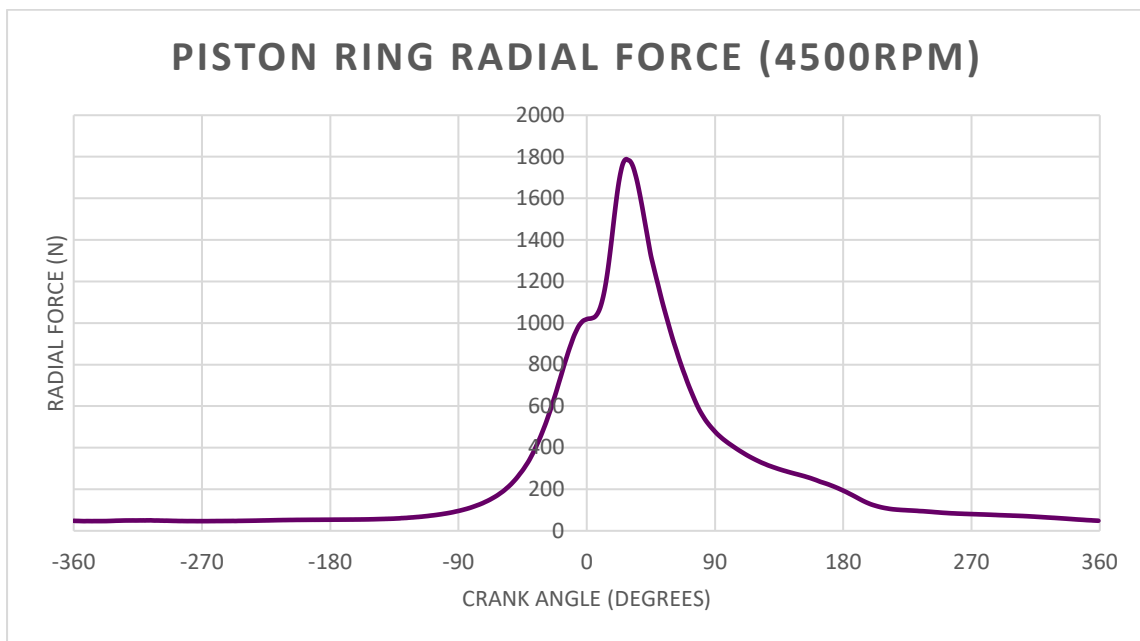


Figure 16-179: Radial Force on the Top Piston Ring at 4500RPM

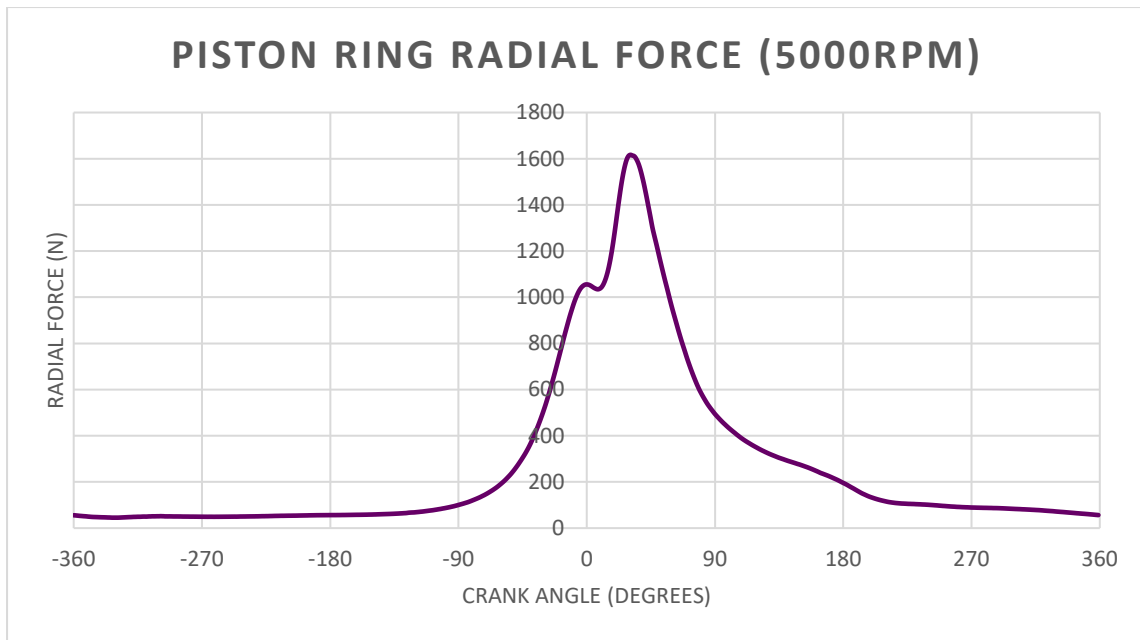


Figure 16-180: Radial Force on the Top Piston Ring at 5000RPM

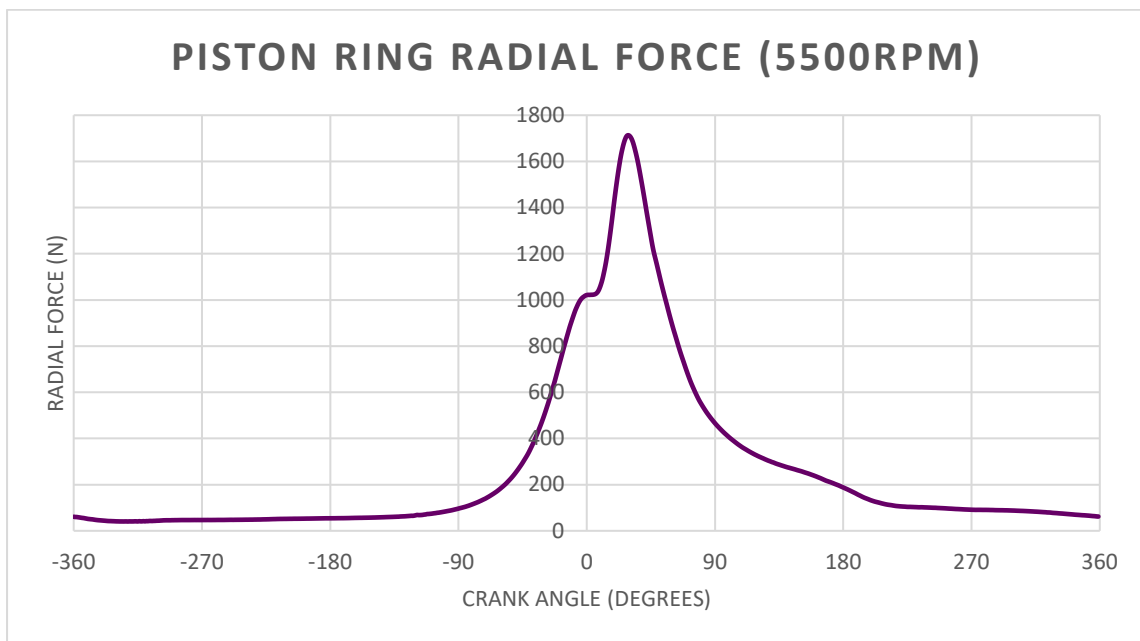


Figure 16-181: Radial Force on the Top Piston Ring at 5500RPM

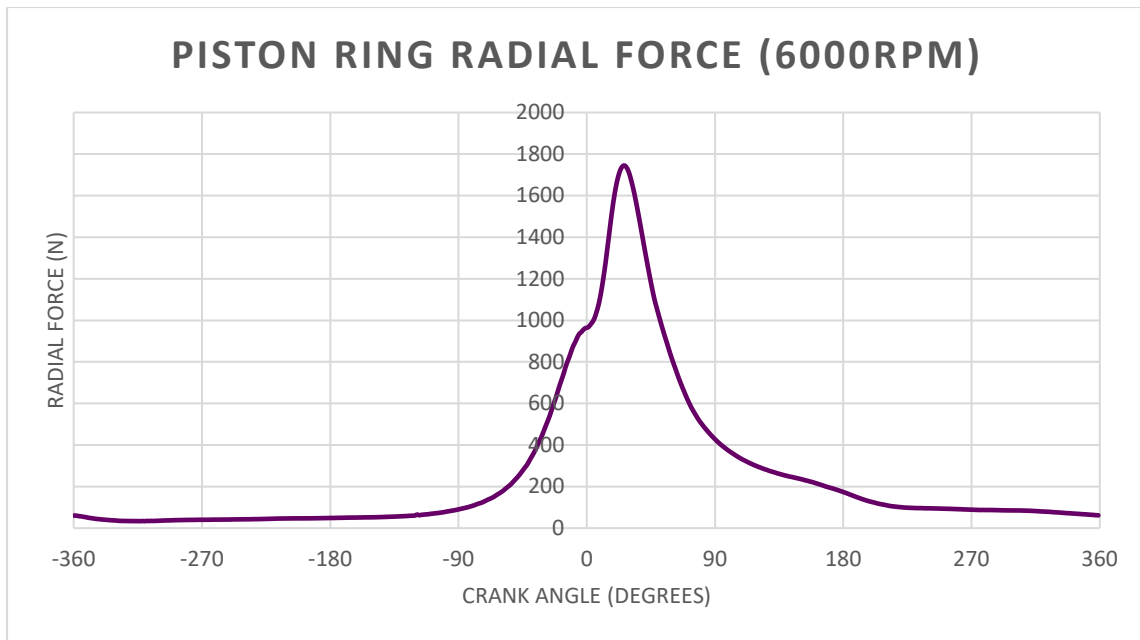


Figure 16-182: Radial Force on the Top Piston Ring at 6000RPM

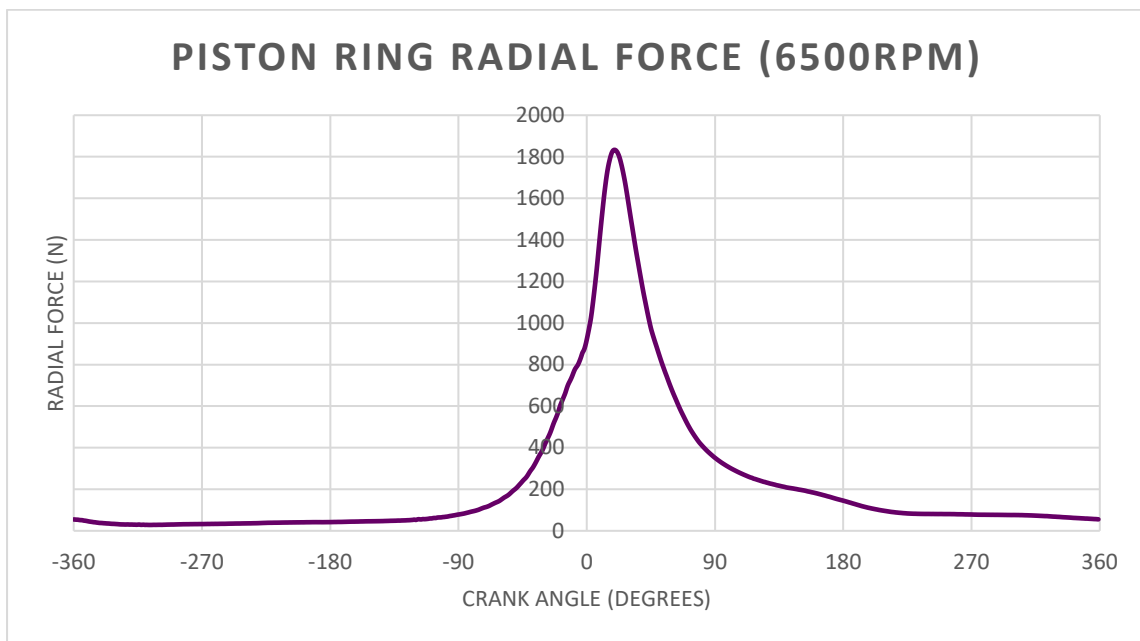


Figure 16-183: Radial Force on the Top Piston Ring at 6500RPM

16.4.10 Friction Force

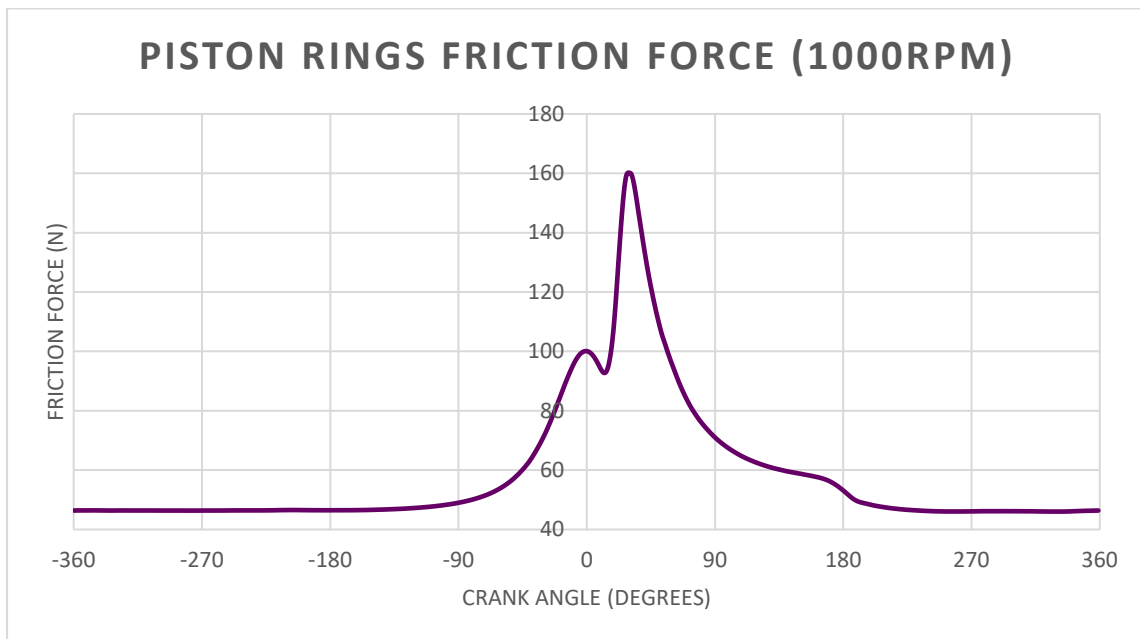


Figure 16-184: Friction Force from the Piston Rings against the Cylinder Wall at 1000RPM

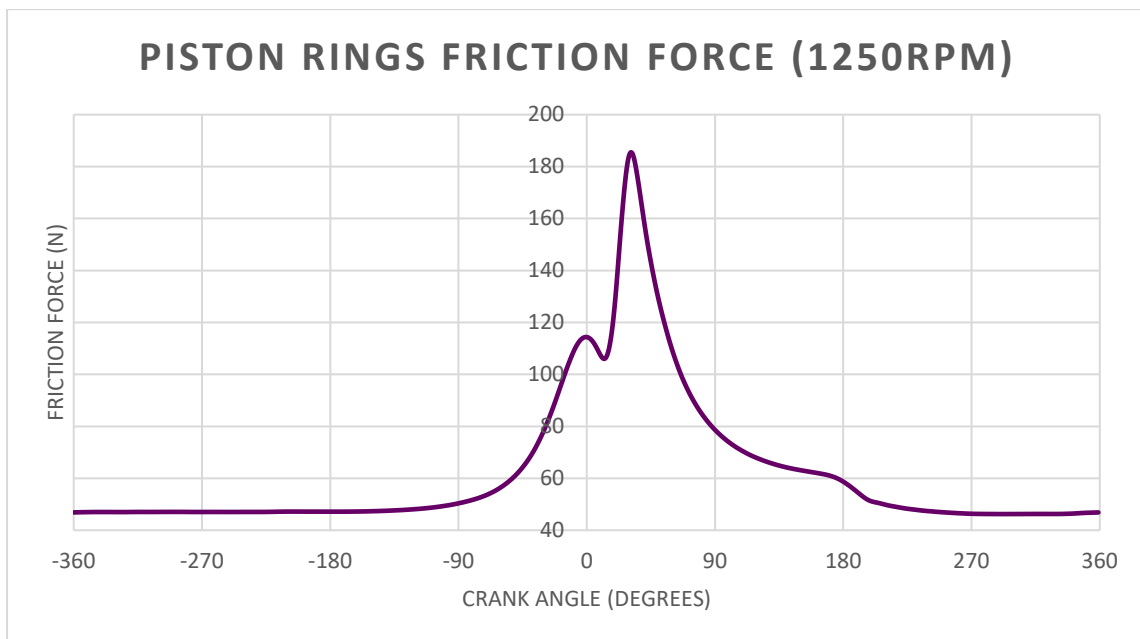


Figure 16-185: Friction Force from the Piston Rings against the Cylinder Wall at 1250RPM

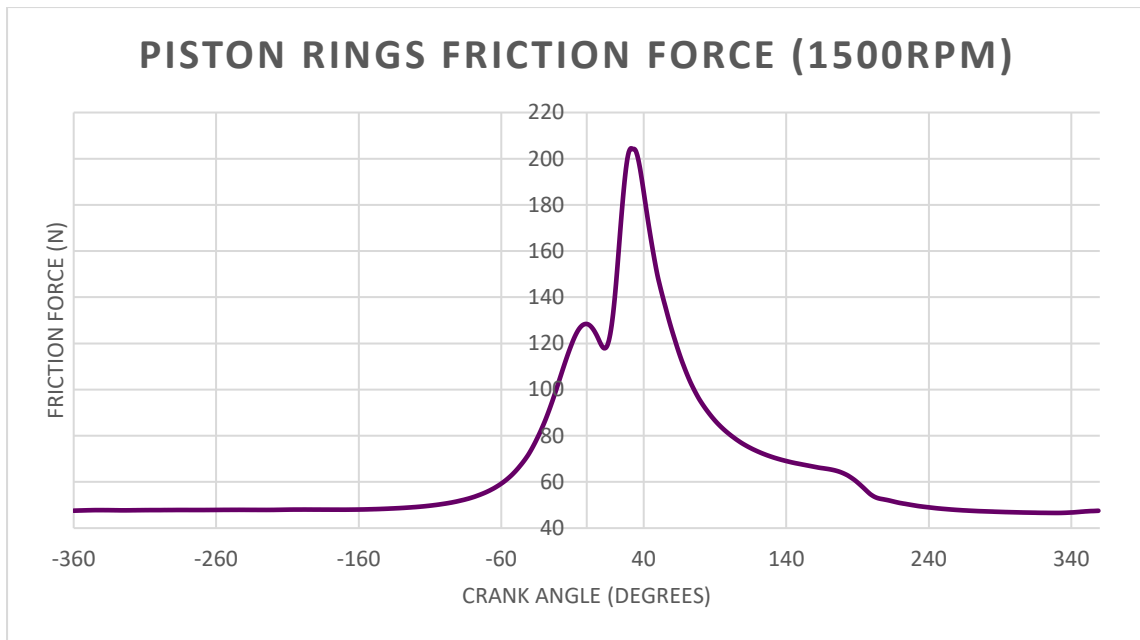


Figure 16-186: Friction Force from the Piston Rings against the Cylinder Wall at 1500RPM

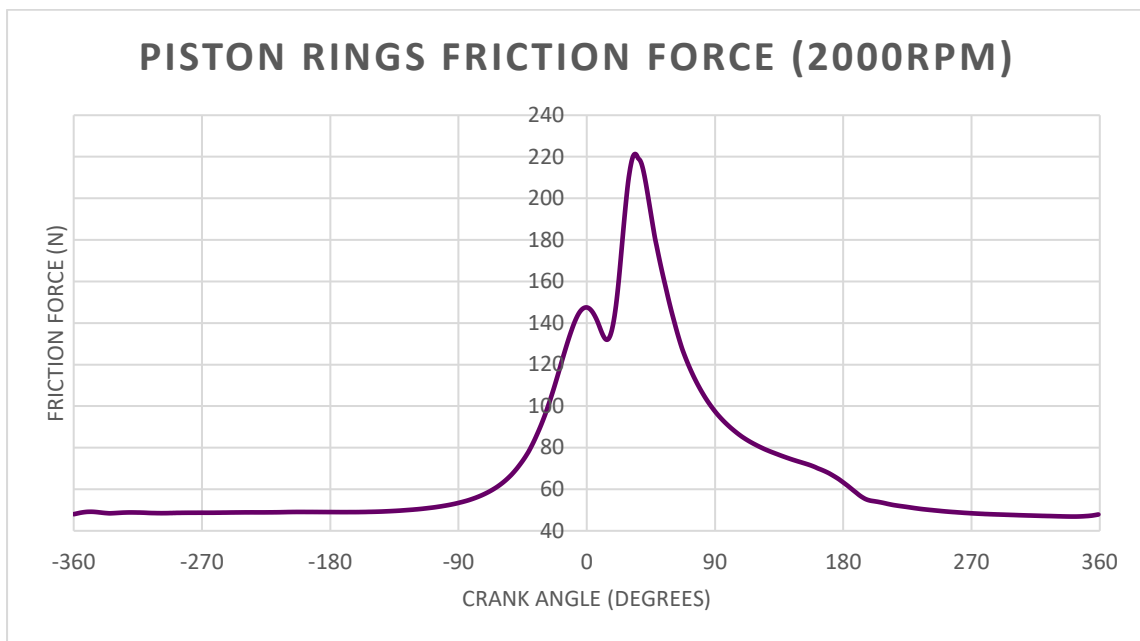


Figure 16-187: Friction Force from the Piston Rings against the Cylinder Wall at 2000RPM

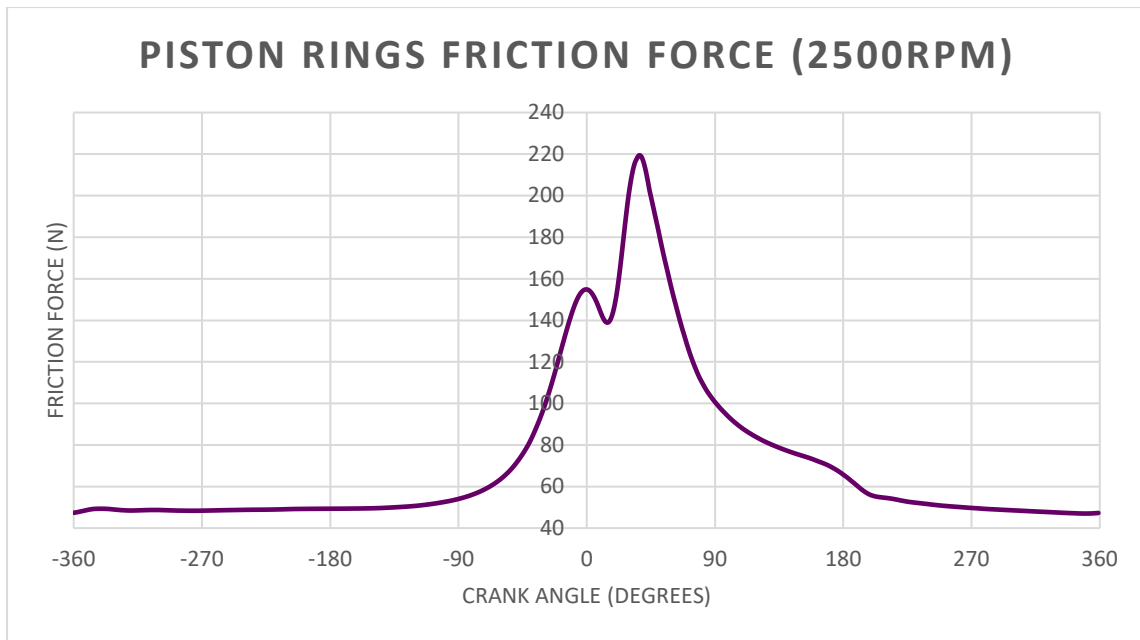


Figure 16-188: Friction Force from the Piston Rings against the Cylinder Wall at 2500RPM

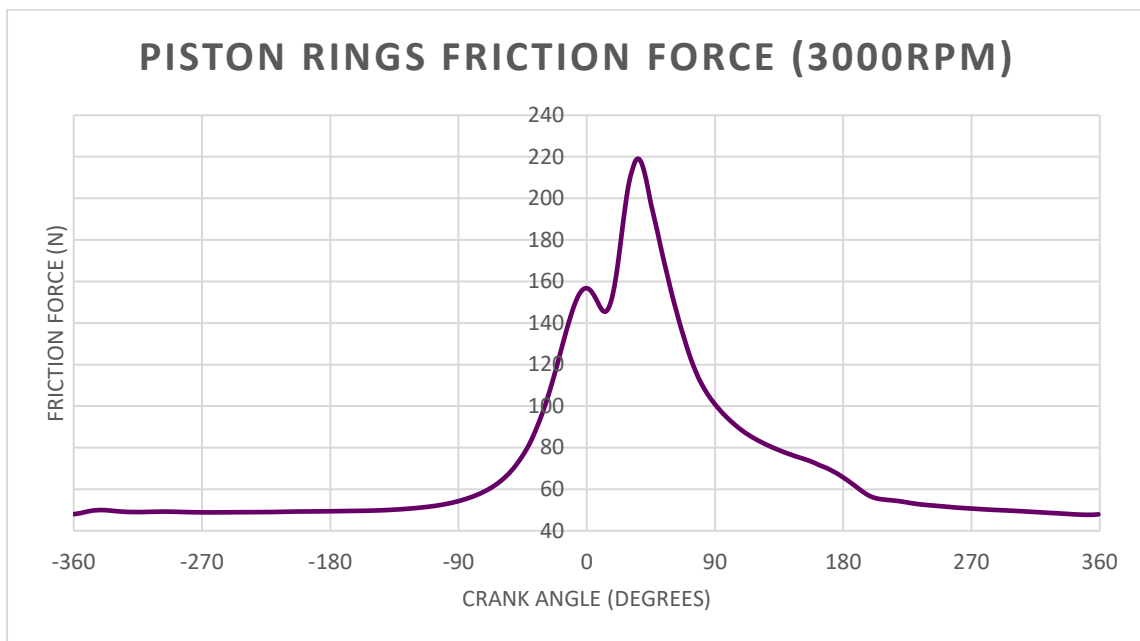


Figure 16-189: Friction Force from the Piston Rings against the Cylinder Wall at 3000RPM

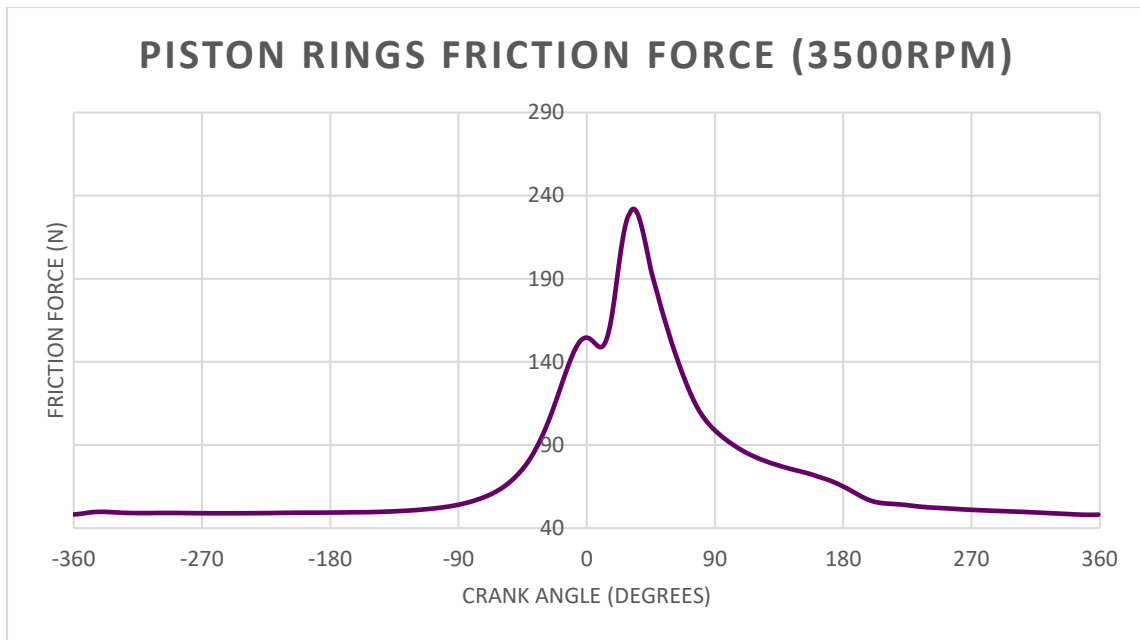


Figure 16-190: Friction Force from the Piston Rings against the Cylinder Wall at 3500RPM

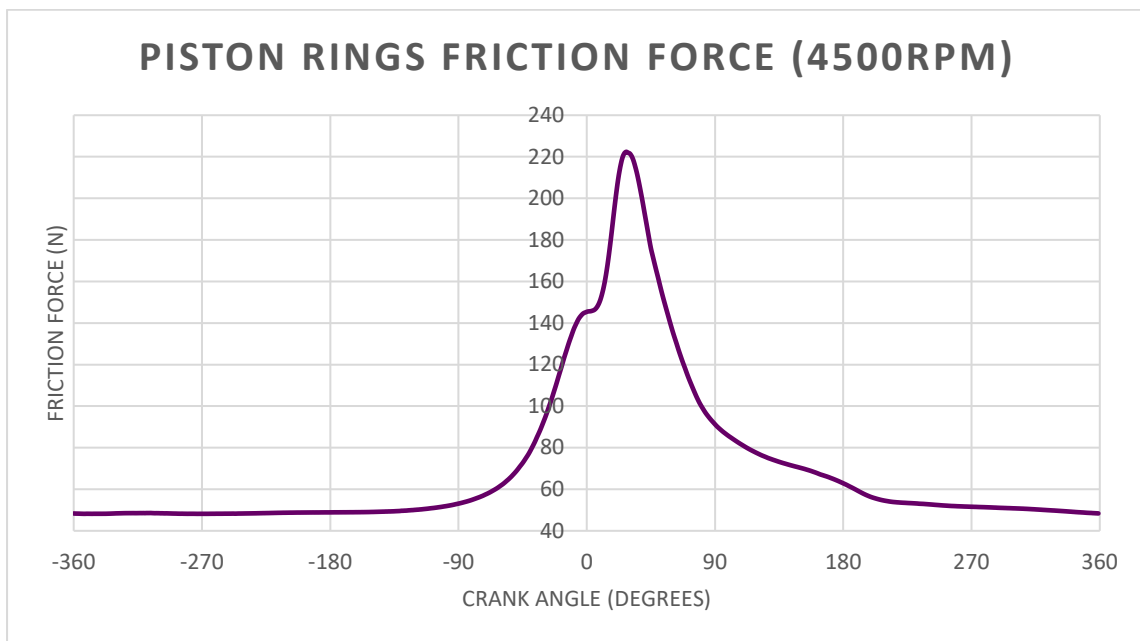


Figure 16-191: Friction Force from the Piston Rings against the Cylinder Wall at 4500RPM

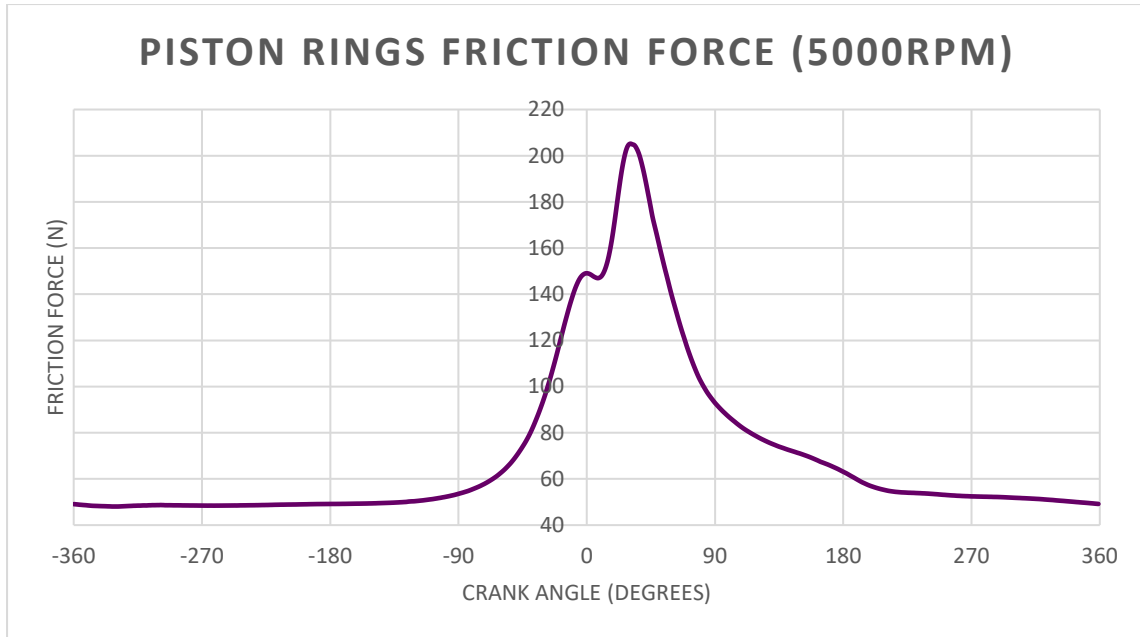


Figure 16-192: Friction Force from the Piston Rings against the Cylinder Wall at 5000RPM

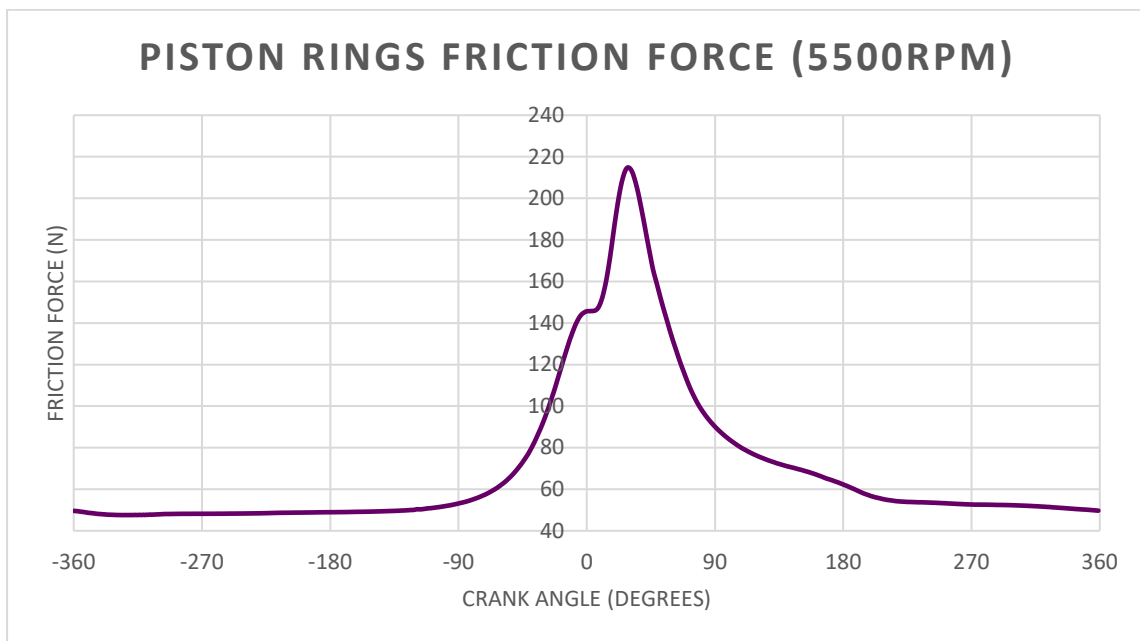


Figure 16-193: Friction Force from the Piston Rings against the Cylinder Wall at 5500RPM

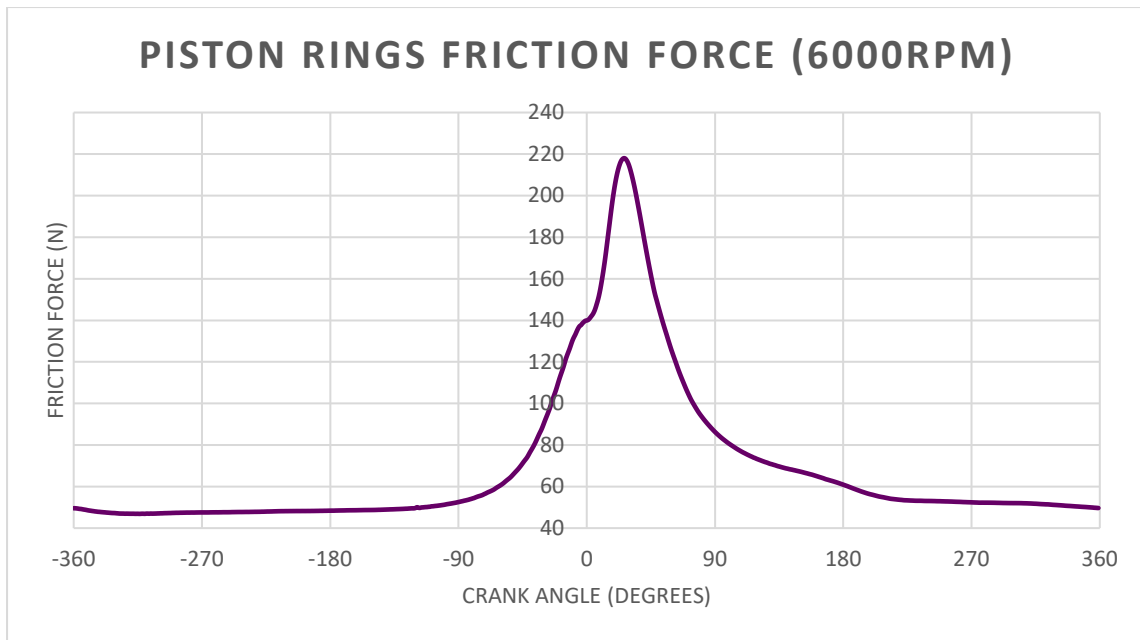


Figure 16-194: Friction Force from the Piston Rings against the Cylinder Wall at 6000RPM

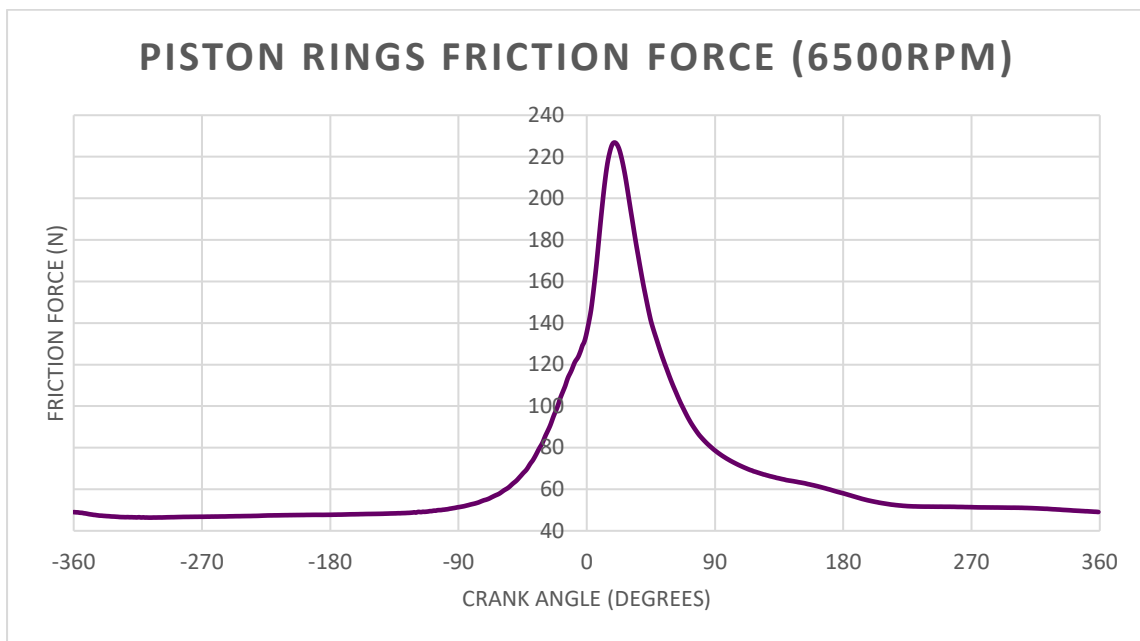


Figure 16-195: Friction Force from the Piston Rings against the Cylinder Wall at 6500RPM

16.4.11 Work Done by the Piston Rings

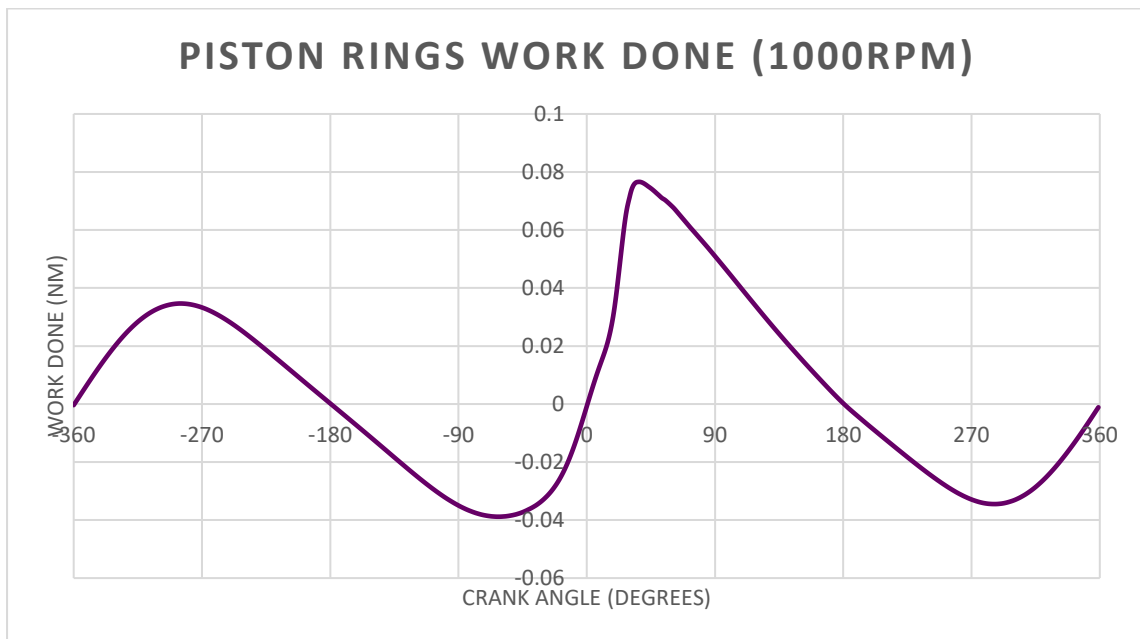


Figure 16-196: Work Done by the Piston Rings against Combustion at 1000RPM

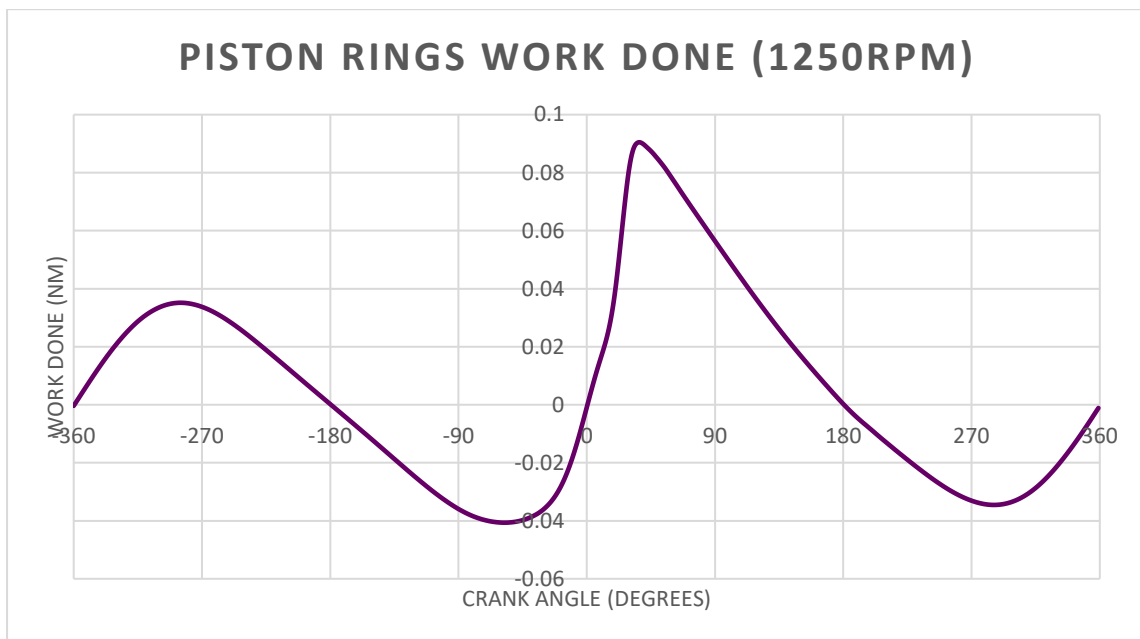


Figure 16-197: Work Done by the Piston Rings against Combustion at 1250RPM

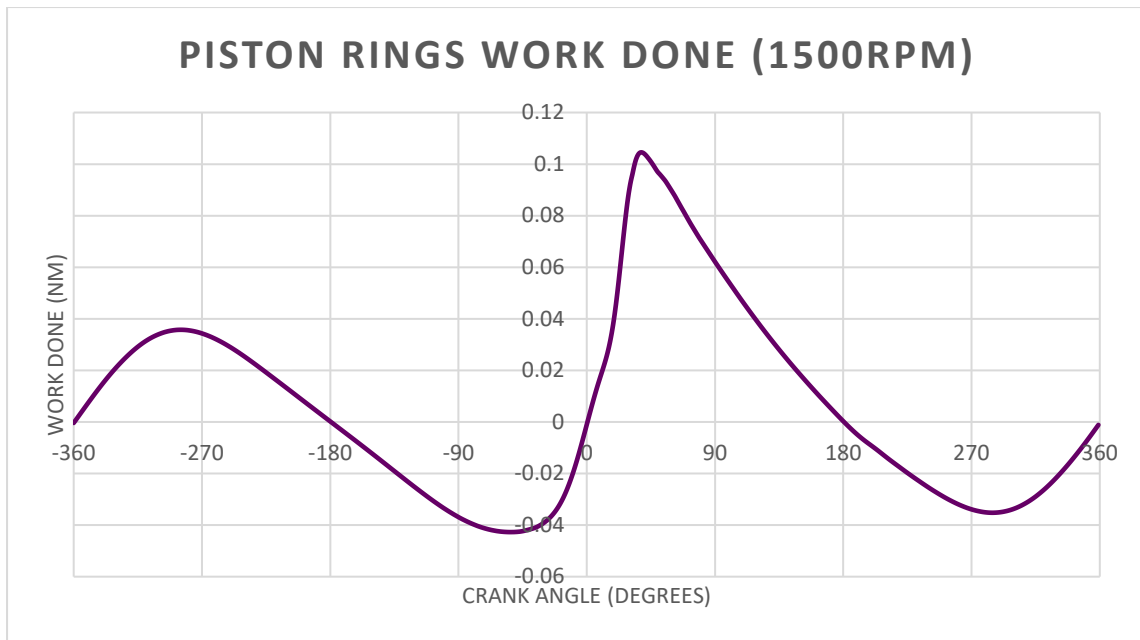


Figure 16-198: Work Done by the Piston Rings against Combustion at 1500RPM

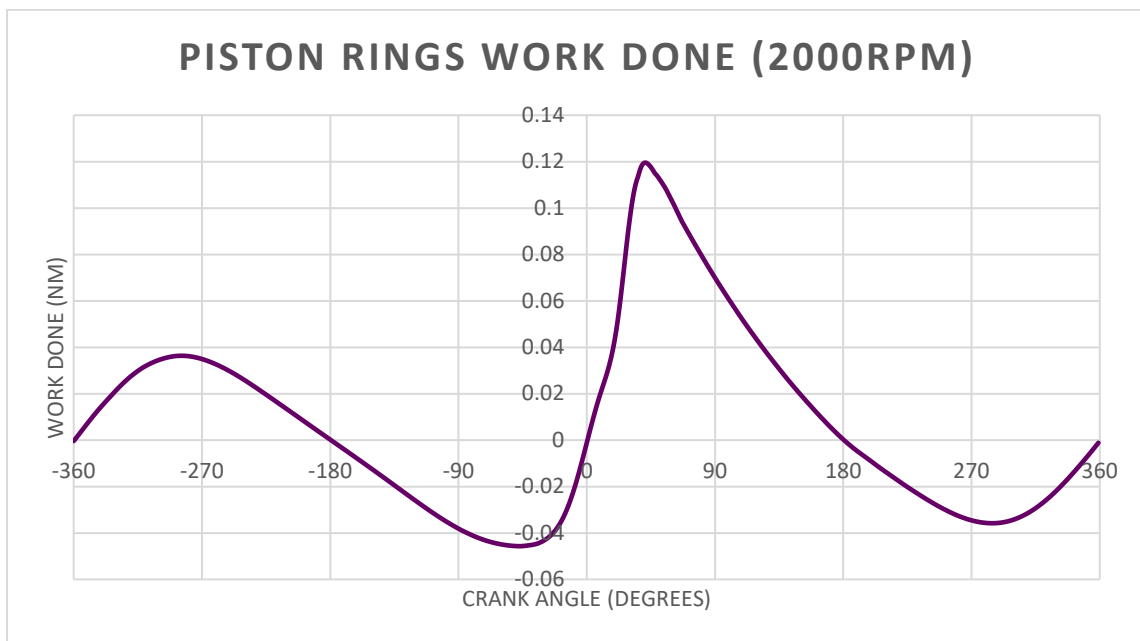


Figure 16-199: Work Done by the Piston Rings against Combustion at 2000RPM

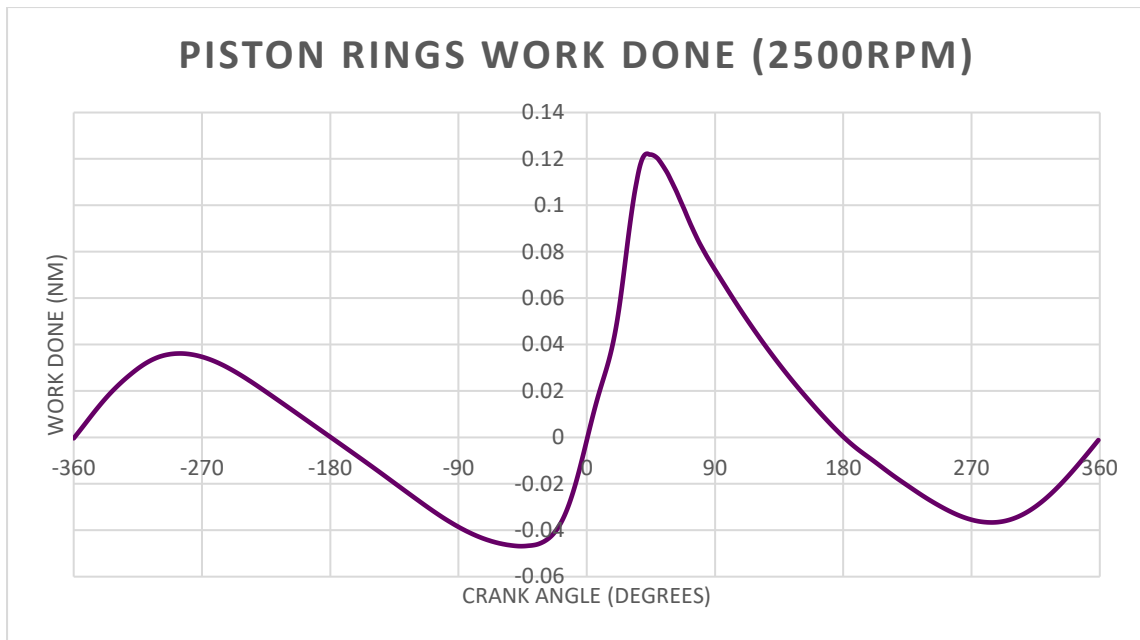


Figure 16-200: Work Done by the Piston Rings against Combustion at 2500RPM

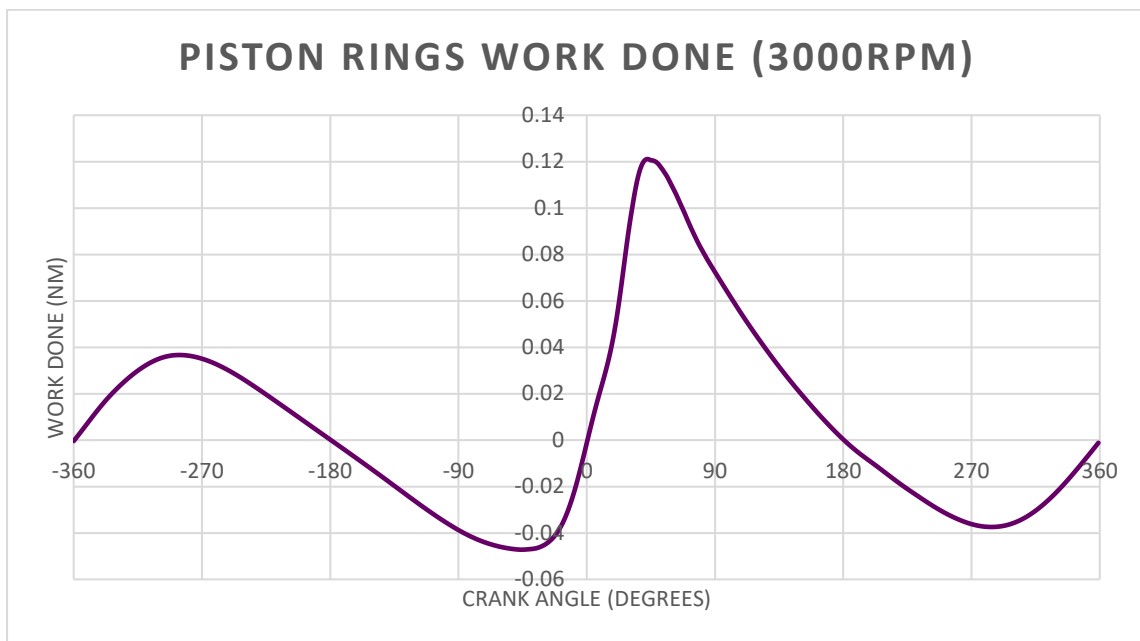


Figure 16-201: Work Done by the Piston Rings against Combustion at 3000RPM

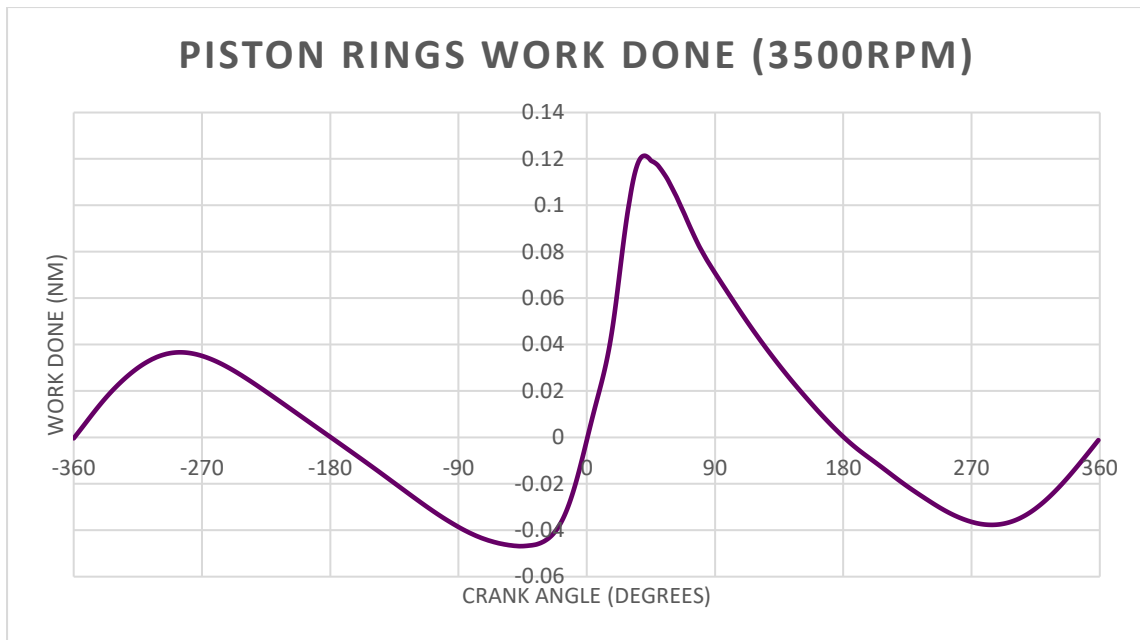


Figure 16-202: Work Done by the Piston Rings against Combustion at 3500RPM

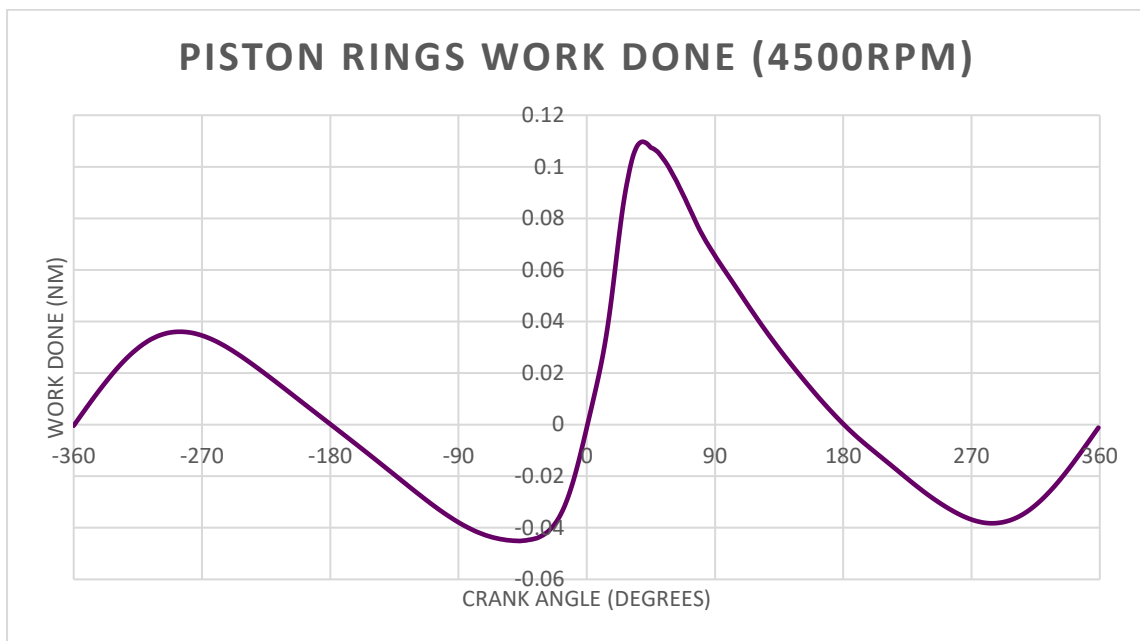


Figure 16-203: Work Done by the Piston Rings against Combustion at 4500RPM

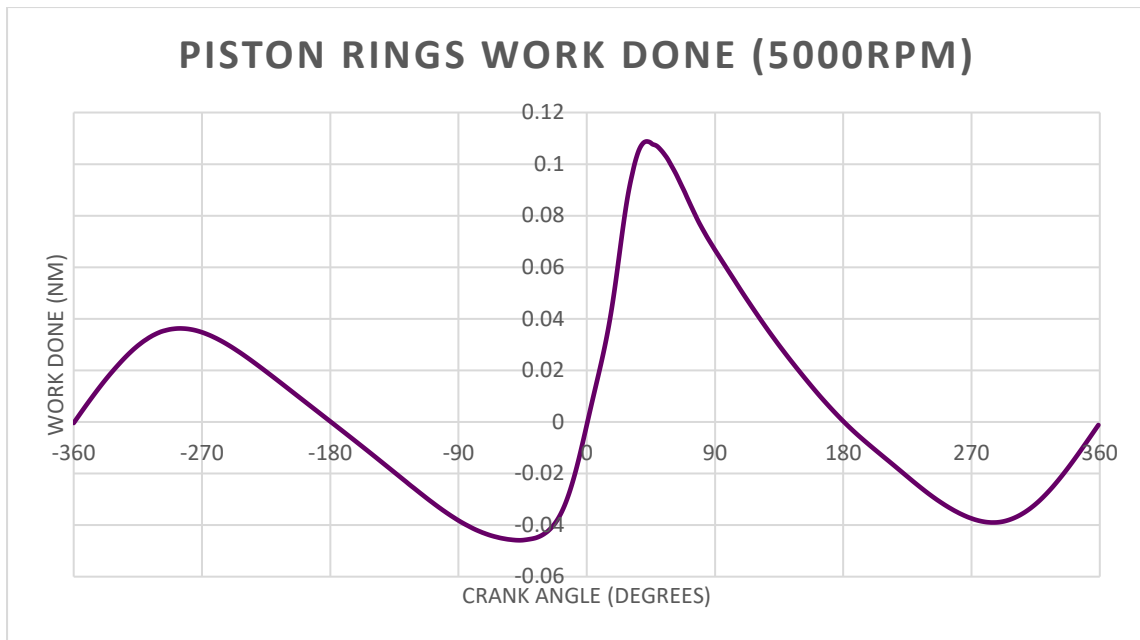


Figure 16-204: Work Done by the Piston Rings against Combustion at 5000RPM

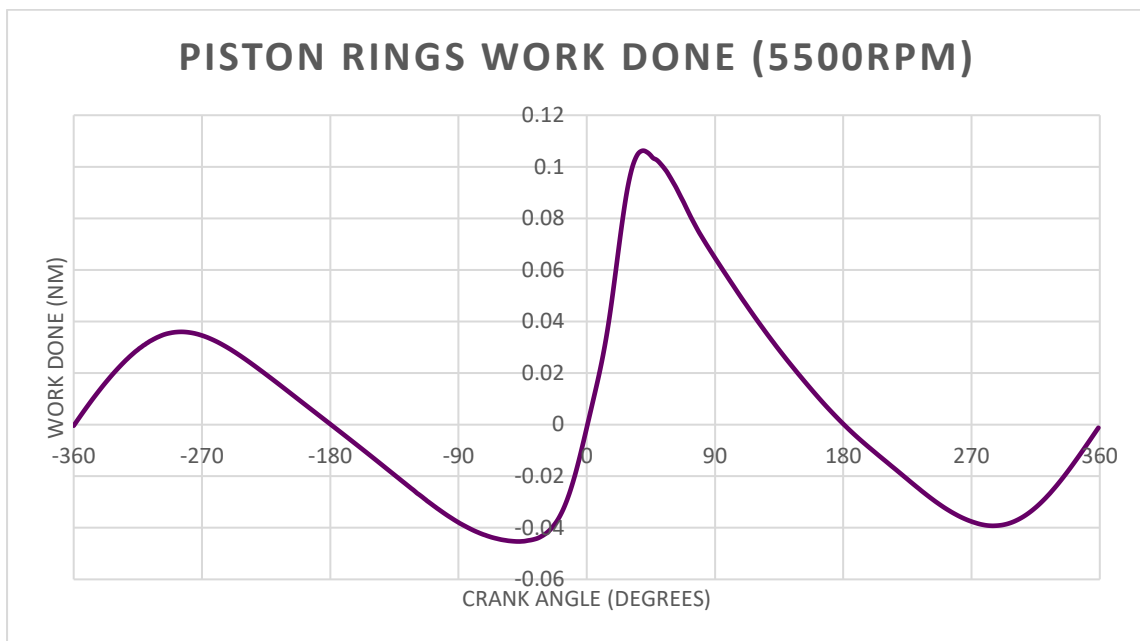


Figure 16-205: Work Done by the Piston Rings against Combustion at 5500RPM

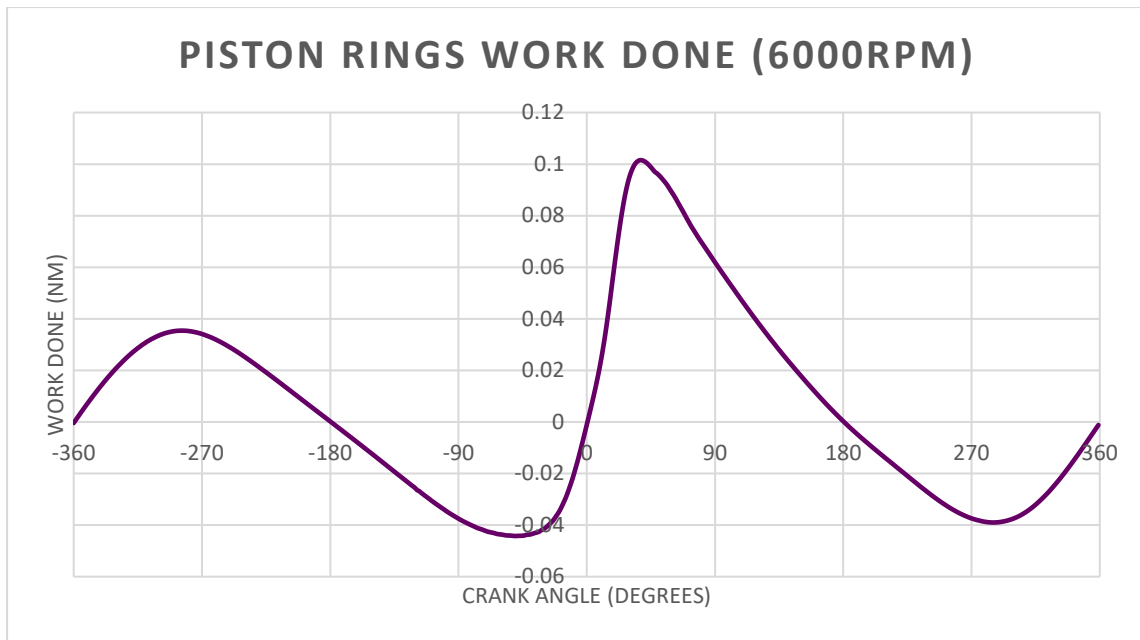


Figure 16-206: Work Done by the Piston Rings against Combustion at 6000RPM

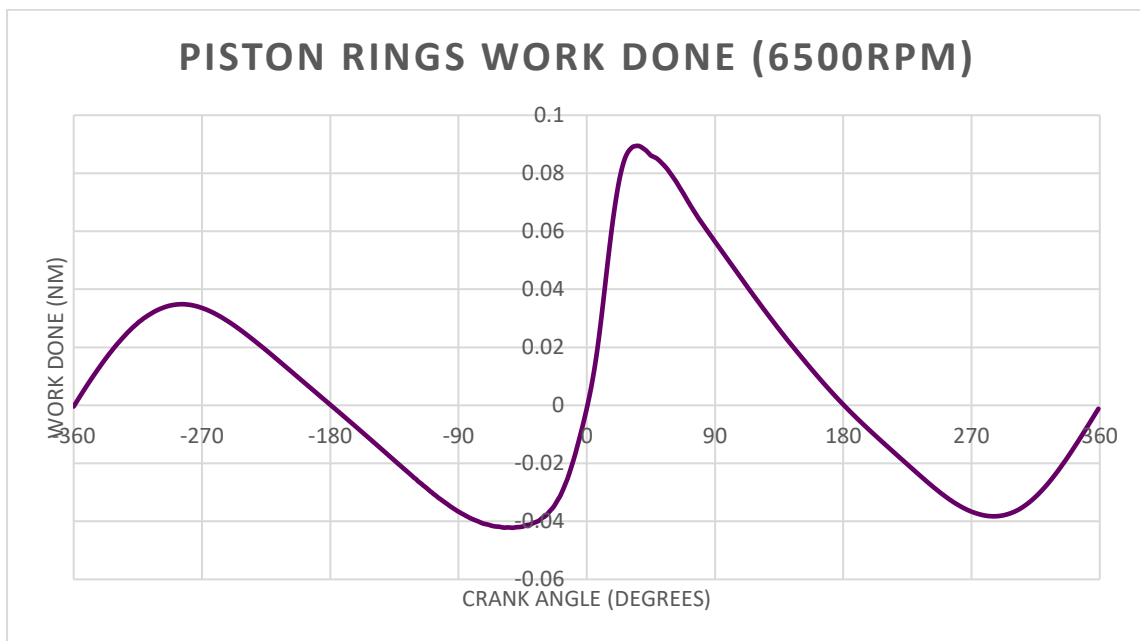
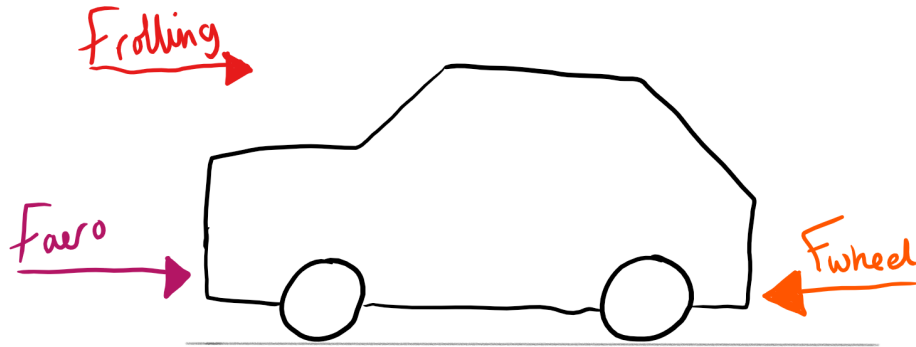


Figure 16-207: Work Done by the Piston Rings against Combustion at 6500RPM

16.5 Appendix E – Supervisor Notes

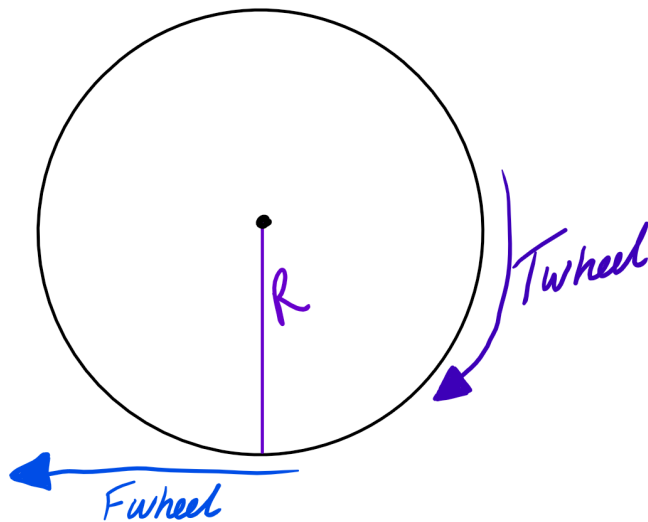


$$\text{Net Forward Force} = F_{Wheel} - F_{Aero} - F_{Rolling}$$

$$F_{Wheel} = ma$$

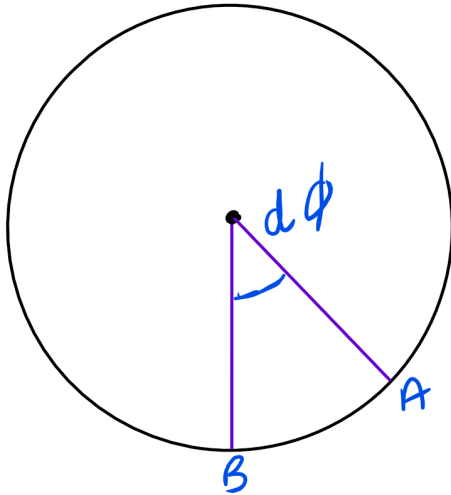
$$F_{Aero} = \frac{1}{2} \times \rho \times CD \times A_F \times v^2$$

$$F_{Rolling} = \mu_{Rr} \times m \times g$$



$$T_{Wheel} = F_{Wheel} \times R_W$$

$$\text{Angular Velocity} = \omega = \frac{d\phi}{dt}$$



$AB = R_W \times d\phi$ When $d\phi$ is in Radians

If $\therefore AB = 2\pi R_W$

Velocity from A-B

$$= \frac{AB}{dt} = \frac{R_W \times d\phi}{dt}$$

$$v = \frac{R_W \times d\phi}{dt}$$

$$\omega = \frac{d\phi}{dt}$$

$$v = \omega R_W$$

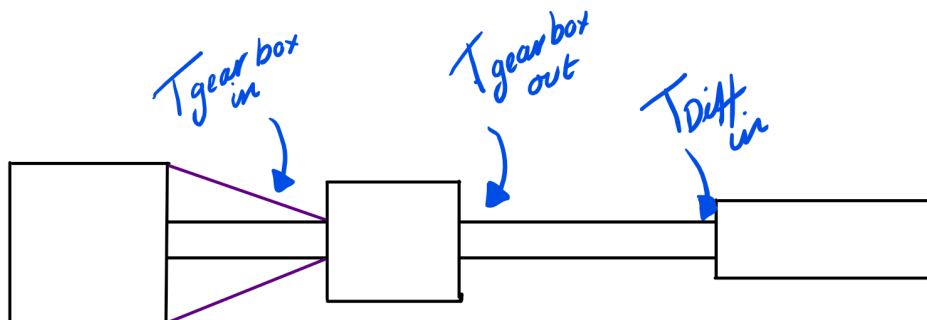
All masses have the same angular velocity and the same angular acceleration

$$F = m \times \frac{R_W d^2 \phi}{dt^2}$$

$$T = I\alpha$$

$$\text{Torque Used Accelerating the Wheel} = I_{\text{WHEEL}} \times \alpha_{\text{DRIVESHAFT}}$$

Diff out torque only equals driveline torque if moment of inertia of the driveshaft is small.



$$\text{Work Done (Nm)} = \text{Force (N)} \times \text{Distance (m)}$$

$$Work\ Done/Rev = F \times 2 \times \pi \times R$$

Assume the $T_{GEARBOX\ OUT} = T_{DIFF\ IN}$

$$T_{GEARBOX\ IN} = \frac{T_{GEARBOX\ OUT}}{Gear\ Ratio \times \eta}$$

$$Energy/Sec = Torque \times Radians/Sec$$

In first gear $\eta=0.94$ (efficiency)

Engine – Crankshaft – flywheel ~ Moment of inertia I_E

$$Torque\ Needed\ To\ Accelerate\ Engine = I_{ENGINE} \times \frac{d^2\phi}{dt^2}$$

$$Engine\ Acceleration \left(\frac{d^2\phi}{dt^2} \right) = Gear\ Ratio \times Final\ Drive\ Ratio \times Wheel\ Acceleration$$

N	O	P	Q	R
Diff in Torque	Gearbox Out Torque	Gearbox In Torque	Engine Angular Acceleration	Engine Dyno Torque
		$= \frac{O}{Gear\ Ratio \times \eta}$	$= Wheel\ Angular\ Acc \times Gear\ Ratio \times Final\ Drive\ Ratio$	$= T_{GEARBOX\ IN} + I_E \times \frac{d^2\phi}{dt^2}$

$$Engine\ RPM = \frac{(Wheel\ Velocity \times Gear\ Ratio \times Final\ Drive\ Ratio) \times 60}{2\pi}$$

Mean Effective Pressure

Mean effective pressure which acting on the power down stroke. Does the same work as the varying pressure over two revolutions.

$$Force\ on\ Piston = MEP \times A_B$$

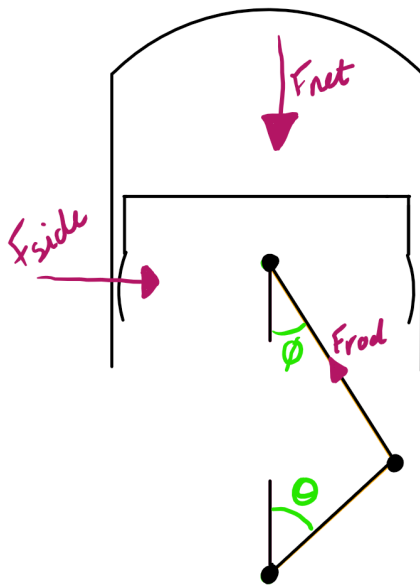
$$Work\ Done\ on\ Power\ Stroke = MEP \times A_B \times L_S$$

$$\therefore Work\ Done\ on\ Power\ Stroke = MEP \times Engine\ Displacement$$

$$Work\ Done\ in\ 2\ Revolutions = Torque \times 4\pi$$

$$Work\ Done\ in\ 2\ Revolutions = BMEP \times Engine\ Displacement$$

16.5.1 18th October 2017



$$F_{Net} = F_{Gas} - F_{Inertia}$$

$$F_{Gas} = P_{Piston} \times A_{Bore}$$

$$F_{Inertia} = M_{Rec} \times \left\{ (R_c \omega^2 \times \cos(\theta)) + \left(\frac{R_c}{L_c} \times R_c \times \omega^2 \times \cos(2\theta) \right) \right\}$$

$$M_{reciprocating} = M_{Piston} + M_{Gudgeon} + M_{PistonRing} + \frac{1}{3} M_{Conrod}$$

Resolve vertically

$$F_{Net} = F_{Rod} \times \cos(\varphi)$$

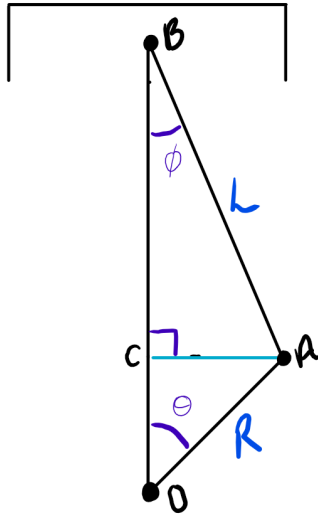
Resolve horizontally

$$F_{Side} = F_{Rod} \times \sin(\varphi)$$

$$\frac{F_{Side}}{F_{Net}} = \frac{F_{Rod} \times \sin(\varphi)}{F_{Rod} \times \cos(\varphi)}$$

$$\frac{F_{Side}}{F_{Net}} = \frac{\cancel{F_{Rod}} \times \sin(\varphi)}{\cancel{F_{Rod}} \times \cos(\varphi)}$$

$$F_{Side} = F_{Net} \times \tan(\varphi)$$



$$AC = L_C \times \sin(\varphi) = R_C \times \sin(\theta)$$

$$\sin(\varphi) = \frac{R_C}{L_C} \times \sin(\theta)$$

$$\varphi = \sin^{-1} \left\{ \frac{R_C}{L_C} \times \sin(\theta) \right\}$$

θ	ϕ	F_{Gas}	$F_{Inertia}$	F_{Net}	F_{Side}
----------	--------	-----------	---------------	-----------	------------

$$S = (R_C + L_C) - (R_C \times \cos(\theta)) - (L_C \times \cos(\varphi))$$

Down is positive

Φ to the right is positive

F_{Side} to the right is positive

$$F_{Friction} = \mu_P \times F_{Side}$$

$$Work\ done = F_{Friction} \times (S_2 - S_1)$$

16.5.2 25th October 2017

$$Wheel\ RPM = \frac{Rads/sec}{2\pi} \times 60$$

$$Engine\ Acceleration = Wheel\ Acceleration \times Gear\ Ratio \times Final\ Drive$$

$$Torque\ Lost\ Accelerating\ Engine = Engine\ Inertia \times \frac{d^2\varphi}{dt^2}$$

$$\therefore = 0.15 \times 43 = 6Nm$$

$$Engine\ Torque = Torque\ Out + Torque\ Lost$$

$$BMEP \times Engine\ Displacement = Work\ done\ in\ 2\ Revs$$

$$T \times 4\pi$$

The total work done in two revolutions at 1000RPM/cylinder is 189J.

For the engine at 1000 RPM, the total work done is 4x189 J.

$$FMEP \times Engine\ Displacement = Work\ Done\ in\ 2\ Revs$$

$$FMEP = \frac{Work\ in\ 2\ Revs}{Engine\ Disp}$$

$$S = R + L - R\cos\theta - L\cos\phi$$

$$\sum \Delta w = Indicated\ Work\ in\ 2 \frac{Revs}{Cylinder}$$

$$Total\ Indicated\ work\ in\ 2\ Revs\ for\ Engine = 4 \times \sum \Delta w$$

$$IMEP = \frac{4 \times \sum \Delta w}{Engine\ Disp}$$

$$BMEP \times Engine\ Disp = Work\ Out\ of\ Engine\ in\ 2\ Revs$$

$$\therefore Torque \times 4\pi$$

$$BMEP = \frac{Torque \times 4\pi}{Engine\ Disp}$$

16.5.3 8th November 2017

Drive Cycle

$$Engine\ RPM = 2 \times Prop\ RPM$$

$$\frac{Engine\ RPM}{Wheel\ RPM} = 2 \times 3 = 6$$

$$\frac{\text{Engine RPM}}{\text{Wheel RPM}} = \text{Gear Ratio} \times \text{Final Drive Ratio}$$

$$\frac{\text{Engine RPM}}{\text{Prop Shaft RPM}} = \text{Gear Ratio}$$

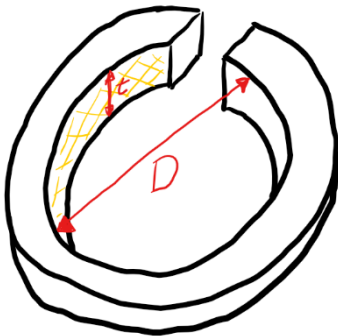
Find the FMEP for the whole engine

$$FMEP = \sum_{\text{All Cylinders}} \frac{\text{Work Done in 2 Revs}}{\text{Engine Displacement}}$$

$$FMEP = 4 \times \frac{\text{Work Done in 2 Revs}}{\text{Engine Displacement}}$$

$$FMEP = \frac{\text{Work done in 2 Revs in Cylinder}}{\text{Cylinder Displacement}}$$

Friction Model



$$\text{Inner Area} = \pi Dt$$

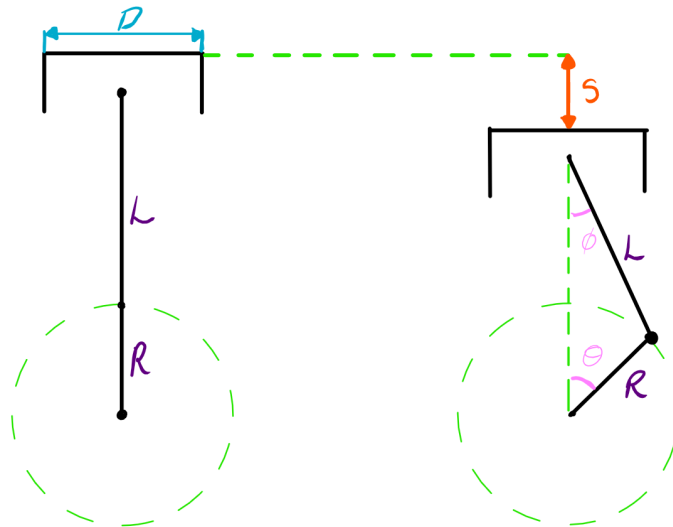
$$\text{Radial Force} = P\pi Dt$$

Friction Force Vertically

$$= \mu P\pi Dt$$

+ Friction Force From Other two Rings

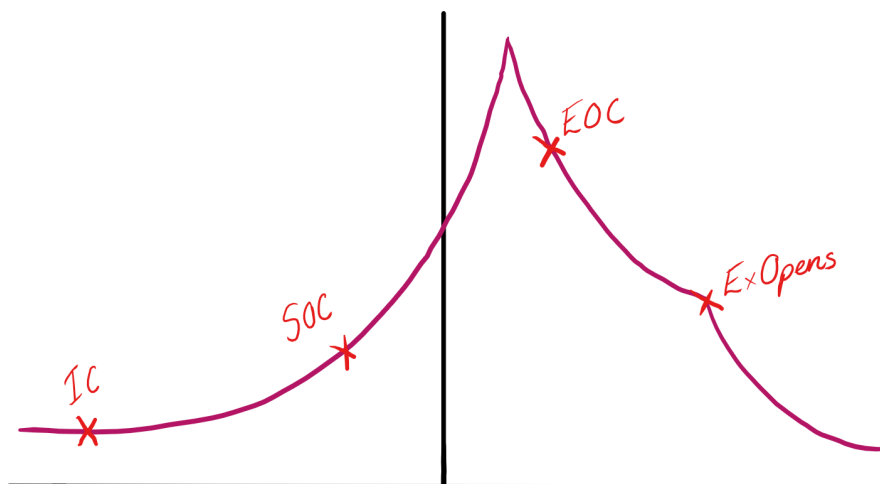
Combustion Model



$$S = R + L - (R \cos \theta + L \cos \phi)$$

$$V_{\text{cylinder}} = \text{Clearance Volume} + S \times \frac{\pi D^2}{4}$$

A	B	C
Crank Angle	Piston Average Pressure	Volume



1-2 Ideal Otto cycle $PV^\gamma = \text{Constant}$

3-4 ideal $PV^\gamma = \text{Constant}$

Adiabatic changes?? In P & V (without Combustion)

With real engine and heat losses

$$PV^n = \text{Constant}$$

$$n \neq \gamma$$

If

$$PV^n = \text{Constant}$$

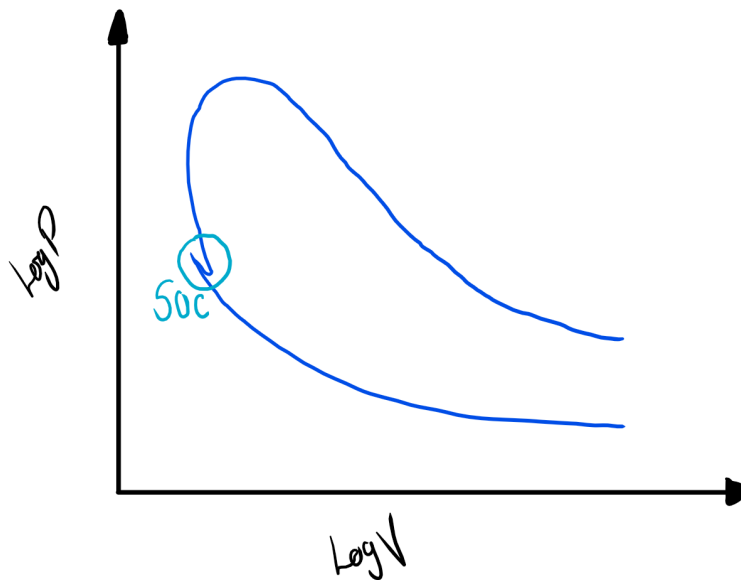
$$\text{Log}(PV^n) = \text{Constant}$$

$$\text{Log}P + \text{Log}V^n = \text{Constant}$$

$$\text{Log}P + n\text{Log}V = \text{Constant}$$

$$\text{Log}P = -n\text{Log}V + \text{Constant}$$

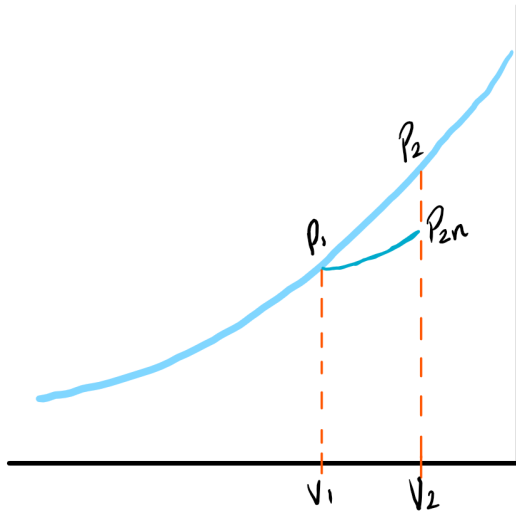
$$\uparrow y = mx + c$$



$$\text{Slope } M = -n$$

$$M = \frac{y}{x}$$

$$n = -\frac{y}{x} = \text{Polytropic Index}$$



P_{2n} = Pressure it would have got to from piston and Friction alone

ΔP = Extra Pressure from Combustion

$$\Delta P = P_2 - P_{2n}$$

$$P_1 V_1^n = P_{2n} V_2^n$$

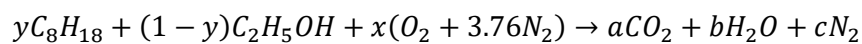
$$P_{2n} = \frac{P_1 V_1^n}{V_2^n}$$

$$P_{2n} = P_1 \left(\frac{V_1}{V_2} \right)^n$$

16.5.4 15th November 2017

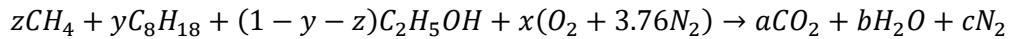
Fuel Model

Stoichiometric



C	8y+2(1-y) 6y+2	a a
H	18y + 6(1-y) 18y +6 -6y 12y+6 6y +3	2b 2b 2b b
O	(1-y)+2x	2a+b

y	A=6y+2	B=6y+3	$x = a + \frac{b}{2} + \frac{(1-y)}{2}$	C=3.76x
0.05				
0.1....				



Change the percentage of each fuel

$$z + y + (1 - y - z) = 1$$

16.5.5 22nd November 2017

- Recheck the polynomials
- Going over of the pressure increase 21.6.3 8th November 2017

θ	P	V	Log(P)	Log(V)	$n = \frac{\log P_2 - \log P_1}{\log V_2 - \log V_1}$	$P_{2n} = P_1 \times \left(\frac{V_1}{V_2}\right)^n$	$\Delta P = P_2 - P_{2n}$
----------	---	---	--------	--------	---	--	---------------------------

At higher RPM the polytropic index needs to change.

16.5.6 29th November 2017

Assume 1L engine has same moment of inertia and gear ratios as 2L engine car, engine internal torque is same for both engines because cars have same acceleration.

$$BMEP \times Engine Displacement = Work Done in 2 Revs$$

$$BMEP \times Engine Displacement = Torque \times 4\pi$$

$$BMEP = \frac{Torque \times 4\pi}{Engine Displacement}$$

- Input the Fox fuel mass percentage table creating fuel usage charts and polynomials for fuel flow, to input in as if everything is the same except fuel
- What is the advantage of smaller engines on the drive cycle?

E.g. at 500RPM 10Bar BMEP 157Nm Torque fuel flow 0.606gm/sec

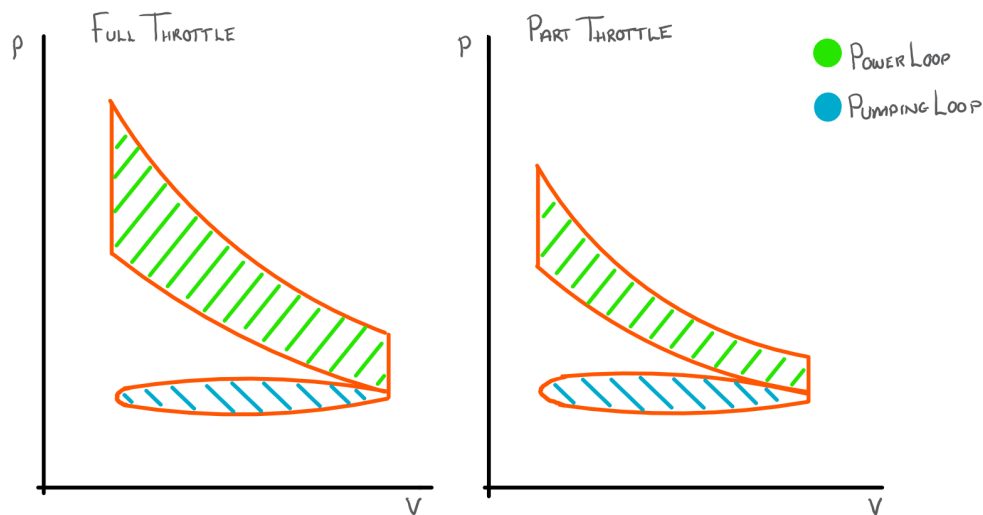
$$Power(kW) = \frac{2\pi \times Torque \times RPM}{60000} = 8.22kW$$

$$Fuel = 0.606 \times 3600 = 2181.6 \text{ gms/hr}$$

$$\frac{0.606 \times 3600}{8.22} = 265.4 \frac{\text{gms}}{\text{kW}} \cdot \text{hr}$$

16.5.7 6th December 2017

- Turn VW graph (table) upside down
- Rotate 3D graph



At full power pumping loop area is a small proportion of the power loop area ~ less of a problem.

At part throttle (low power & big pumping loop) the pumping loop area is almost as large as power loop area ~ ineffective

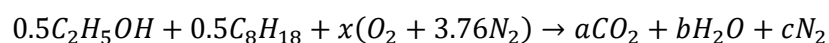
$$\text{Net Work Done/s} = \text{Power Loop Area} - \text{Pumping Loop Area}$$

- Run Passat drive cycle with no changes from fox ford data
- Suppose answer is lower fuel usage
- Analyse – explain why hopefully you can show it is spending most of its time in a lower specific fuel consumption region
- Sort mass fraction burn

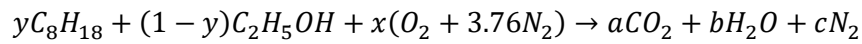
16.5.8 17th January 2018

Resultant turbulent air speed

$$S_T = S_L + k \times \text{RPM}$$



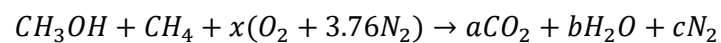
C	$(0.5 \times 2) + (8 \times 0.5) = 5$	a	a=5
H	$(0.5 \times 6) + (0.5 \times 18) = 12$	2b	b=6
O	$0.5 + 2x$	$2a + b = 16$	$x = 7.75$



C	$8y + 2(1-y)$ $6y + 2$	a a
H	$18y + 6(1-y)$ $18y + 6 - 6y$ $12y + 6$ $6y + 3$	2b 2b 2b b
O	$(1-y) + 2x$ $(1-y) + 2x$ 2x 2x x	$2a + b$ $2(6y + 2) + 6y + 3$ $12y + 4 + 6y + 3 + y - 1$ $19y + 6$ $9.5y + 3$

- Re check fuel percentage spreadsheet

16.5.9 24th January 2018



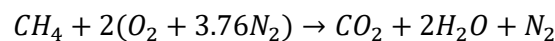
C	1+1	a	a=2
H	4+4	2b	2b=8 b=4
O	1+2x	2a+b	2a+b=8 1+2x=8 2x=7 x=3.5

Below 800RPM (idle) in the drive cycle, the clutch is slipping.

Multiply first gear by 1.2 to get new ratio. New engine RPM = $800 \times 1.2 = 960$ RPM and required engine torque will also drop because wheel torque needed stays constant, but this can be achieved with a lower engine torque with a numerically higher 1st gear ratio.

GDI calculate net enthalpy released/kmol air

$$\frac{\text{Net Enthalpy Released}}{\text{kmols Air}}$$



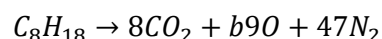
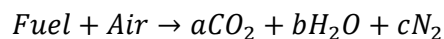
Enthalpy needed for reactants is 74.9MJ. enthalpy needed for products 877MJ. Results in 802.1MJ of enthalpy. The number of Kmols in the mixture of air and fuel stays constant.

$$PV = nRuT$$

Where P= 1 bar, V= cylinder Displacement, T=293K, Ru is constant

$$n = \frac{PV}{RuT} \quad \text{which is fixed for all mixtures.}$$

16.5.10 31st January 2018



The enthalpy released by the products 5322MJ. The enthalpy needed by the reactants to get to zero level = 224MJ. GDI engines air comes in valves closed & fuel is injected. Changing fuels but keeping same engine and turbo charger, the volume of air sucked in stays constant. The enthalpy released for every fuel/kmol of air is important to establish which fuel delivers the most power from the same air.

E.g.

$$\begin{aligned} \text{Enthalpy Released / kmol air} &= \frac{5100}{12.5(1 + 3.76)} \\ &= 90.7 \text{ MJ/kmol. air} \end{aligned}$$

Air fuel ratio by mass = 15

$$\text{Air: Fuel ratio by mass} = \frac{x(32 + 3.76 \times 28)}{(\text{fuel mass})}$$

New fuel released = 85 MJ/kmol.air

To produce same power, we need $\frac{90.7}{85}$ times more air & fuel.

From the Air:Fuel ratio, if the new fuel produced the same amount of power as the old fuel/kmol.air

$$\text{New Fuel Amount Needed} = \frac{1}{\left(\frac{\text{New Air: Fuel Ratio by Mass}}{\text{Old Air: Fuel Ratio by Mass}} \right)}$$

EXAMPLE

If gasoline = 15:1 and Ethanol = 9:1

$$\text{Ethanol Amount Needed} = \frac{15}{9} = 1.67$$

New amount of fuel needed if they produce the same power

$$\frac{\text{Old Air Fuel Ratio}}{\text{New Air Fuel Ratio}} \times \text{Old Fuel Mass}$$

$$\frac{15}{9} \times \text{Old Mass (gms/sec)}$$

But if the new fuel isn't as good at developing power for same amount of air power is down by a factor of $\frac{90.7}{85}$

To get the power back you need to use more fuel. New gms/sec to get the same torque (or Power or energy release)

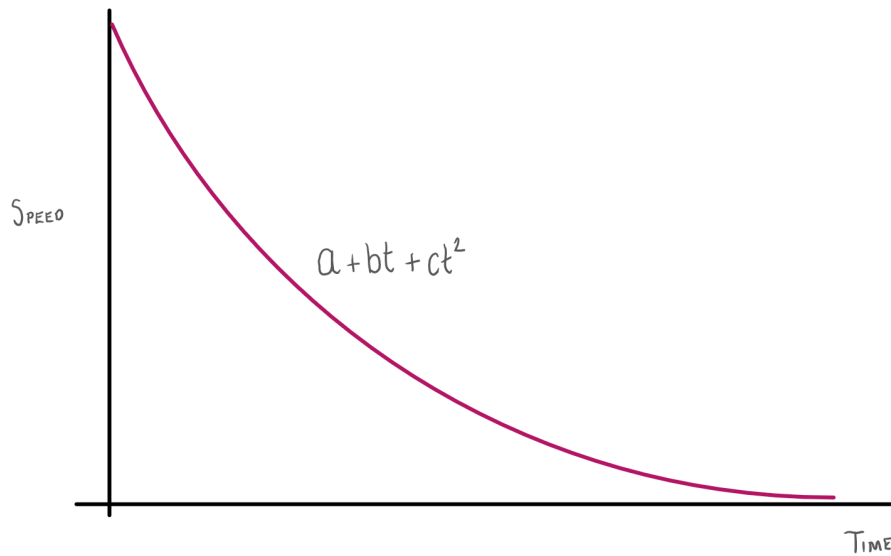
$$\frac{90.7}{85} \times \frac{\text{old Air: Fuel Ratio}}{\text{New Air: Fuel Ratio}}$$

$$\frac{90.7}{85} \times 1.67$$

16.5.11 14th and 21st February 2018

$$F_{\text{Wheel}} = F_{\text{Aero}} + F_{\text{Rolling}}$$

$$F_{\text{Car}} = \frac{1}{2} \rho C D A v^2 + f_0 + f_1 v_1$$



$$v = \text{Speed} = a + bt + ct^2$$

differentiate

$$acc = \frac{dv}{dt} = 0 + b + 2ct$$

$$F_{Car} = a + bv + cv^2$$

Differentiate with respect to v

$$\frac{dF_{Car}}{dv} = b + 2cv$$

Differentiate with respect to v

$$\frac{d^2F_{Car}}{dv^2} = 2c$$

These constants are then used to find the forces in a coast down test, the aerodynamic coefficients can then be used to find the drag of the vehicle.

16.5.12 28th February 2018

At 2000RPM Full Load the products

$$= \alpha \{7.83CO_2 + 9.09H_2O + 0.25CO + 47N_2\}$$

$$\text{Total Kmols in Exhaust} = \alpha \{7.83 + 0.25 + 9.09 + 47\}$$

$$NO \text{ ppm in exhaust} = 1553$$

$$Kmols \text{ in Exhaust} = \frac{1553}{10^6} \times 64\alpha$$

$$\text{Mass of NO in exhaust at that second} = \frac{1553}{10^6} \times 64\alpha \times (14 + 16)$$

$$CO_2 \% \text{ by volume} = 13.5\%$$

$$= \alpha \{aCO_2 + cH_2O + bCO + dN_2\}$$

The total number of Kmols in the exhaust $\sim 64\alpha$

$$\frac{a}{a + b + c + d} \times 100 = 13.5\%$$

We now know actual CO_2

$$\therefore \text{Exhaust } CO_2 = \alpha \times a \times 44 \quad \text{in that second}$$

Of actual fuel consumption

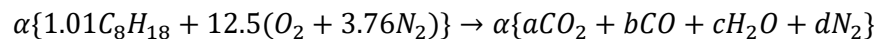
$$\text{actual fuel used} \times \frac{Le}{(Le + 25)}$$

Le =electric range in km

16.5.13 7th March 2018

NO_x at 2000RPM full load there is 1553 ppm. $\phi = 1.01$

In the engine



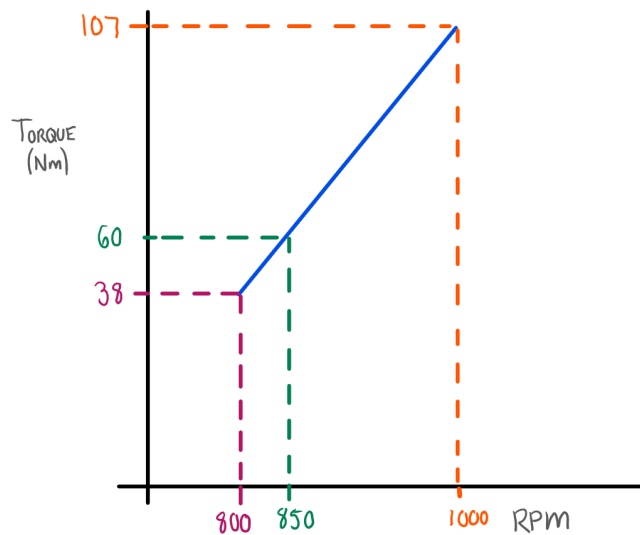
$$\alpha \times 1.01 \times 114 = \text{Fuel Per Cycle}$$

$$\alpha \times 1.01 \times 114 = \frac{11.35}{60 \times 2000 \times 1.5}$$

$$\alpha = \frac{11.35}{60 \times 2000 \times 1.5 \times 1.01 \times 114}$$

- The friction can be linked to the drive cycle with the work done, how much power is from the drive cycle & how much is used up with friction

- Get a different gear ratio that allows the level of Torque as Fox cannot have 60Nm of torque at 800RPM



Suppose you need 850RPM to get 60Nm change the gear ratios

$$= \text{Original First Gear} \times \frac{850}{800}$$

$$\alpha = \frac{\text{kg/hr}}{60 \times \text{RPM} \times 1.5 \times \lambda \times 114}$$

$$\text{gms/sec NOx} = \frac{\text{PPM}}{10^6} \times 64\alpha \times 30 \times 10^3$$

1.5 is the cylinder displacement.

$$\text{gms/sec NOx} = \text{actual kw} \times \frac{\text{SFC NOx (g/kWhr)}}{3600}$$

NEDC ~ 733grams, 1180 seconds, 11 km

WLTP ~ 1317grams, 1800 seconds, 23 km

$$\text{NEDC gms/km fuel} = \frac{733}{11} = 66.6 \text{ gms/sec}$$

$$\text{WLTP gms/km fuel} = \frac{1317}{23} = 57 \text{ gms/sec}$$

16.5.14 14th March 2018

- 1) Turn off fuel during deceleration i.e. if BMEP < 1 make gms/sec 0
- 2) Hybrid system ~ energy recovery.
 - a. Wheel to battery = 80% efficient
 - b. Battery to wheel = 80% efficient
When slowing from 30m/s to 10m/s, the kinetic energy you can recover =
 $\left(\frac{1}{2} \times m \times 30^2\right) - \left(\frac{1}{2} \times m \times 10^2\right)$ this then is multiplied by the efficiency to find the actual recovery in to the battery.
 - c. Total battery extra energy regenerated when slowing down is the sum of the actual recovery in to the battery.
- 3) In tests you can start with the battery charge (could be empty) but you must end the test with the same or less battery charge.

For example, if the sum of the braking recovery is 25000J, this could be used at any time

Try to use it in the urban cycle until its run out then switch to engine or save it for the Extra Urban cycle (Extra Urban is 75mph & uses more fuel and produces more NOx).

The amount available to replace the power at the wheels is

$$0.8 \times 25000 = 20000J$$

20000J of available wheel energy to use when you want

- 4) You need battery size capable of storing the recovered energy of 25000J. you could factor in the extra mass coming from the battery pack and motor and controller.

Motor size depends on where you use it, e.g.

Extra Urban might need 20kW

Urban might need 10kW

- 5) If you want to 'cheat' the drive cycle you need a big enough battery pack to do 11km NEDC and then official fuel consumption:

$$= actual \times \frac{25}{(25 + 11)}$$

- 6) Aim is ~ 40km NEDC
~ calculate the battery pack and motor and add to the Passat, to calculate the official fuel consumption.

16.5.15 28th March 2018

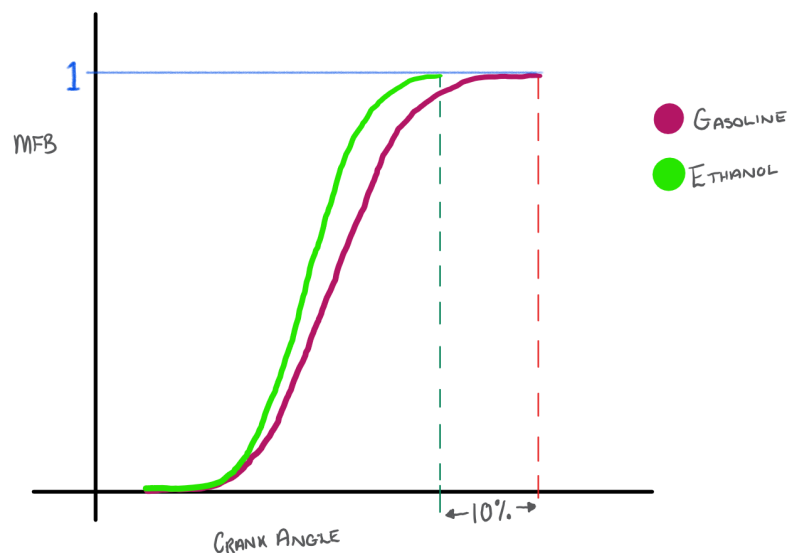
Change the fuels in the combustion, calculate the new stoichiometric Air:Fuel ratio & adjust all the fuel numbers proportionally.

E.g.

Ethanol Air:Fuel ratio = 9:1

$$\text{New Fuel Used} = \frac{14.7}{9} \times \text{the old fuel mass}$$

If ethanol burns 10% faster, shorten all the MFB numbers to have a burn duration 10% less.



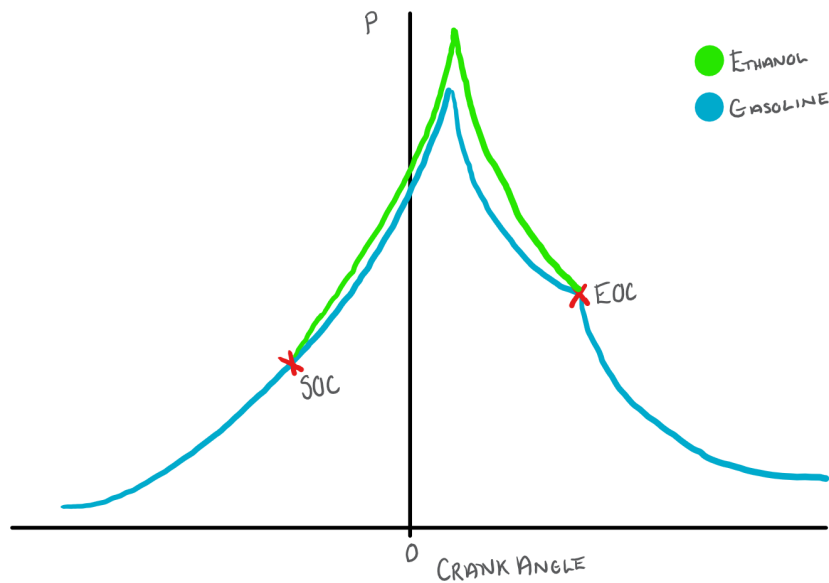
Make $\Delta P_{\text{New}} = 1.1 \times \Delta P_{\text{Old}}$

$$\Delta P = P_2 - P_1 \left(\frac{V_1}{V_2} \right)^n$$

$$P_2 = \Delta P + P_1 \left(\frac{V_1}{V_2} \right)^n$$

once combustion starts ΔP is the previous pressure

make new ΔP 10% bigger & start of combustion at the same angle and same original pressure.
because ΔP are bigger, peak pressure will be higher. end combustion 4° earlier & start the expansion



$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)^{n_{exp}}$$

you can plot a new curve from P_1 to end of combustion.

New work done from P_1 to exhaust open

θ	V	P_{old}	P_{New}	Δw_{Old}	Δw_{New}
-				$= P_{Old}(V_2 - V_1)$	$= P_{New}(V_2 - V_1)$
SOC					
-					
EOC				Σ	Σ

The difference in the sums is the extra work done by the faster burning fuel

$$\frac{\text{Extra Work With Faster Combustion}}{\text{Original Total Work per Cycle}}$$

= factor of increase of output over original combustion. I.e. if 10% more output you would use 10% less fuel (approximately).

So the new ethanol consumption

$$= \frac{14.7}{9} \times \frac{1}{1.1} \times \text{Old Fuel Mass}$$

From the pressure you calculate the indicated work in 2 revolutions

$$IMEP = \frac{\sum_{-360 \rightarrow +360} \Delta w}{Engine\ Displacement \times 10^5}$$

At each RPM.

You need to find the FMEP for the piston on the cylinder wall, typically piston friction is 30% of total friction.

Longer connecting rods or something to reduce piston friction, suppose you reduce it 20%

$$Total\ Friction\ Reduction = 20\% \times 30\% = 6\%$$

New FMEP is 6% lower. IMEP stays fixed

$$New\ BMEP = IMEP - 0.94 Old\ FMEP$$

New work out is 6% better than old for same fuel, fuel consumption drops 6%.