Design and Implementation of a Reciprocating Friction
Force Measurement System for the Investigation of Dry
Contact Bearings in a Controlled Atmosphere

A Thesis Presented to
The Faculty of the College of Engineering and Technology
Ohio University

In Partial Fulfillment
of the Requirements for the Degree
Masters of Science

by
Robert K. Baker
June, 1993
TABLE OF CONTENTS

TABLE OF CONTENTS ............................................................................................ i
LIST OF TABLES .................................................................................................. v
LIST OF FIGURES ................................................................................................. vi
LIST OF SYMBOLS ............................................................................................... xviii

CHAPTER 1: INTRODUCTION

1.1 PROBLEM DEFINITION ................................................................................... 1
1.2 ORGANIZATION OF THESIS ......................................................................... 3

CHAPTER 2: LITERATURE REVIEW

2.1 FRICTION THEORIES .................................................................................... 4
2.2 FRICTION BEHAVIOR OF POLYMERS ....................................................... 7
   MECHANICAL PROPERTIES .............................................................................. 8
   COUNTERFACE SURFACE ROUGHNESS ...................................................... 13
   TEMPERATURE ............................................................................................... 15
   GEOMETRICAL ARRANGEMENT - SINGLE OR MULTIPLE CONTACTS .... 19
   LOAD ............................................................................................................ 20
   SPEED ........................................................................................................... 22
   ENVIRONMENT ............................................................................................. 26
2.3 EARLIER STUDIES ....................................................................................... 27
CHAPTER 3: DESIGN OF FRICTION MEASUREMENT SYSTEM

3.1 INTRODUCTION ........................................................................................................ 29
3.2 METHODS OF MEASURING FRICTION ................................................................... 30
3.3 FRICTION MEASUREMENT SYSTEM ........................................................................ 31
    DESIGN PROCESS ......................................................................................................... 32
    FRICTION MEASUREMENT DEVICE ........................................................................ 34
    MEASUREMENT SYSTEM .......................................................................................... 38
    DEVELOPMENT OF THE TEST BEARING ASSEMBLY ........................................... 40
        BEARING HOLDER .................................................................................................... 41
        INERTIA SPRING SUBASSEMBLY .......................................................................... 52
        CONNECTOR SUBASSEMBLY .................................................................................. 53
        BEARING CASE SUBASSEMBLY ............................................................................. 54
        THRUST BEARING SUBASSEMBLY ...................................................................... 55
    INSTRUMENTATION ...................................................................................................... 57
        FRICTION AND LOAD MEASUREMENT ................................................................. 58
        STRAIN GAGE .......................................................................................................... 58
        WHEATSTONE BRIDGE .......................................................................................... 61
        AMPLIFIER ................................................................................................................ 62
        LOW-PASS FILTER ................................................................................................... 64
        TEMPERATURE MEASUREMENT .......................................................................... 65
    DATA ACQUISITION ....................................................................................................... 66
CHAPTER 4: DESCRIPTION OF TEST MATERIALS

4.1 RULON J .......................................................... 69
4.2 IGLIDE T500 ....................................................... 70
4.3 VESPEL SP-3 ..................................................... 71
4.4 COUNTERFACE .................................................. 73

CHAPTER 5: RESULTS

5.1 TEST MATRIX ..................................................... 75
5.2 PREPARATION PROCEDURES .................................. 76
5.3 CONTRAST TESTS ................................................ 78
5.4 LONG-TERM TESTS .............................................. 79

CHAPTER 6: DISCUSSION AND CONCLUSIONS

6.1 DISCUSSION OF TEST RESULTS .................................. 108
   RULON J .......................................................... 108
   IGLIDE T500 ..................................................... 110
   VESPEL SP3 ...................................................... 110
6.2 CONCLUSIONS ....................................................... 111
6.3 DISCUSSION OF MEASUREMENT SYSTEM ..................... 112
6.4 AREAS OF IMPROVEMENT ...................................... 113
6.5 RECOMMENDATIONS FOR FURTHER WORK .................. 116

APPENDIX A: FILTERED AND UNFILTERED TEST MEASUREMENTS.... 117
APPENDIX B: INFLUENCE OF TEMPERATURE ON THE FRICTION COEFFICIENTS ......................................................... 130

APPENDIX C: DESIGN CALCULATIONS

C.1 SPOKE DESIGN ........................................................................................................................................... 155

  ORIGINAL DESIGN ................................................................................................................................. 155

  REDESIGN .................................................................................................................................................. 163

  MODIFICATIONS ......................................................................................................................................... 166

C.2 INERTIA SPRINGS ....................................................................................................................................... 167

C.3 LOAD ........................................................................................................................................................ 169

C.4 SYSTEM MASS ......................................................................................................................................... 170

APPENDIX D: COMPONENT DRAWINGS OF TEST ASSEMBLY .......... 172
LIST OF TABLES

Table 2.1. Properties of Typical Reinforced "Teflon" TFE Resins .................................................. 13
Table 2.2. How Fillers Alter Properties of "Teflon" Resins .......................................................... 13
Table 2.3. Experimental Values of n ............................................................................................. 22
Table 4.1. Design Criteria for Rulon J .......................................................................................... 70
Table 4.2. Iglide T500 Properties ................................................................................................. 72
Table 4.3. Summary of Typical Properties for SP-3 ..................................................................... 73
Table 5.1. Contrast Test Matrix ..................................................................................................... 75
Table 5.2. Identification of Factor Levels ...................................................................................... 76
Table 5.3. Composition of Helium ................................................................................................. 77
Table 5.4. Figure Associations ...................................................................................................... 80
Table 5.5. Contrast Test Data for Iglide T500 .............................................................................. 81
Table 5.6. Contrast Test Data for Rulon J ................................................................................... 82
Table 5.7. Contrast Test Data for Vespel SP-3 ........................................................................... 83
Table 5.8. Long Term Test Data for Rulon J and Iglide T500 ..................................................... 84
LIST OF FIGURES

Figure 2.1. Difference Between Real ($A_r$) and Apparent ($A_a$) Area of Contact .... 6
Figure 2.2. Factors Affecting Polymer Friction.......................................................... 7
Figure 2.3. The Smooth Molecular Profile of PTFE (left) Compared to Polyethylene (right) ................................................................. 9
Figure 2.4. Monomers of Polyethylene (left) and PTFE (right) ......................... 10
Figure 2.5. Molecular Structures of Linear Polymers (a) Thermal and deformation treatments (b) Stress-strain relationships .......................... 11
Figure 2.6. Friction of Polymers as a Function of Counterface Roughness: $O = PA66; \; U = PA6; \; \Delta = POM; \; x = PETP$ ........................................... 14
Figure 2.7. Effects of Temperature on the Mechanical Properties of Polymers (a) Effect of temperature on the stress strain curve of cellulose acetate (b) Effects of temperature on the modulus of various resins ..................... 16
Figure 2.8. Variation of Friction with Temperature for Steel Sliding on Various Polymers at Low Speeds ................................................................. 17
Figure 2.9. The Effect of Temperature on Dry Friction of Some Plastics .......... 18
Figure 2.10. Variations of Modulus or Tensile Strength as a Function of Temperature for a Thermoplastic ................................................................. 19
Figure 2.11. Six Different Test Geometries: four-ball, block-on-ring, pin-on-disk, counter-rotating disks with adjustable slip, pin-in-V-block, and crossed cylinders ................................................................. 21
Figure 2.12. Variation of Friction Force and Coefficient of Friction with Load ..... 23
Figure 2.13. Variation of Friction with Load and Speed for Nylon 6 .................. 24
Figure 2.14. Variations of Friction with Speed for Various Polymers ............... 25
Figure 3.1. Method of Measuring Static Friction ....................................................... 31
Figure 3.2. The Phases of Design .............................................................................. 32
Figure 3.3. Test Apparatus ......................................................................................... 36
Figure 3.4. Part Components Required to Retrofit Wear Test Rig .................... 37
Figure 3.5. Friction Measurement System ................................................................. 39
Figure 3.6. Test Bearing Assembly .............................................................................. 40
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.7</td>
<td>The Original Bearing Holder Design</td>
</tr>
<tr>
<td>3.8</td>
<td>Friction Force Calibration</td>
</tr>
<tr>
<td>3.9</td>
<td>Load Force Calibration</td>
</tr>
<tr>
<td>3.10</td>
<td>Misalignment of System (a) An exaggerated view of the misalignment problems (b) With the implementation of alignment washers and the reorientation of the resonant springs, misalignment problems were reduced</td>
</tr>
<tr>
<td>3.11</td>
<td>Natural Frequency of Forced System</td>
</tr>
<tr>
<td>3.12</td>
<td>The Latest Version of the Test Bearing Holder</td>
</tr>
<tr>
<td>3.13</td>
<td>Inertia Spring Subassembly</td>
</tr>
<tr>
<td>3.14</td>
<td>Bearing Case Subassembly (a) original bearing holder (b) redesigned bearing case subassembly</td>
</tr>
<tr>
<td>3.15</td>
<td>Bearing Subassembly</td>
</tr>
<tr>
<td>3.16</td>
<td>Instrumentation Required for Friction and Load Force Measurements</td>
</tr>
<tr>
<td>3.17</td>
<td>Strain Gage Arrangements (a) Friction (b) Load</td>
</tr>
<tr>
<td>3.18</td>
<td>Filter Amplitude Response Versus Frequency</td>
</tr>
<tr>
<td>3.19</td>
<td>Data Acquisition System - Front Panel</td>
</tr>
<tr>
<td>5.1</td>
<td>Calculation of Friction Coefficient</td>
</tr>
<tr>
<td>5.2</td>
<td>Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm</td>
</tr>
<tr>
<td>5.3</td>
<td>Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm</td>
</tr>
<tr>
<td>5.4</td>
<td>Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm</td>
</tr>
<tr>
<td>5.5</td>
<td>Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm</td>
</tr>
<tr>
<td>5.6</td>
<td>Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm</td>
</tr>
</tbody>
</table>
Figure 5.7. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................... 87

Figure 5.8. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................... 88

Figure 5.9. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................... 88

Figure 5.10. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................... 89

Figure 5.11. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................... 89

Figure 5.12. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................... 90

Figure 5.13. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium,
surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................... 90

Figure 5.14. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................................................... 91

Figure 5.15. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................................................... 91

Figure 5.16. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................................................... 92

Figure 5.17. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................................................... 92

Figure 5.18. Influence of Time on the Friction Coefficient of Rulon J:
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ...................................................... 93

Figure 5.19. Influence of Time on the Friction Coefficient of Rulon J:
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ...................................................... 93
Figure 5.20. Influence of Time on the Friction Coefficient of Rulon J:
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ................................................... 94

Figure 5.21. Influence of Time on the Friction Coefficient of Rulon J:
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ................................................... 94

Figure 5.22. Influence of Time on the Friction Coefficient of Vespel SP-3:
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm .................................................. 95

Figure 5.23. Influence of Time on the Friction Coefficient of Vespel SP-3:
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm .................................................. 95

Figure 5.24. Influence of Time on the Friction Coefficient of Vespel SP-3:
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ................................................ 96

Figure 5.25. Influence of Time on the Friction Coefficient of Vespel SP-3:
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ................................................ 96

Figure 5.26. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium .................................................................... 97

Figure 5.27. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium .................................................................... 97

Figure 5.28. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium .................................................................... 98

Figure 5.29. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium .................................................................... 98

Figure 5.30. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium .................................................................... 99

Figure 5.31. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium .................................................................... 99

Figure 5.32. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium ..................................................................... 100
Figure 5.33. Influence of Time on the Friction Coefficient of Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium ......................................................... 100

Figure 5.34. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium ............................................................. 101

Figure 5.35. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium ............................................................. 101

Figure 5.36. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium ............................................................. 102

Figure 5.37. Influence of Time on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium ............................................................. 102

Figure 5.38. Influence of Time on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................. 103

Figure 5.39. Influence of Time on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 103

Figure 5.40. Influence of Time on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ......................................................... 104

Figure 5.41. Influence of Time on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ......................................................... 104

Figure 5.42. Influence of Time on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ......................................................... 105

Figure 5.43. Influence of Time on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ......................................................... 105

Figure 5.44. Influence of Time on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ......................................................... 106
Figure 5.45. Influence of Time on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 106

Figure 5.46. Influence of Time on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 107

Figure 6.1. Friction Force Signal Prior to Alignment Techniques .................. 113

Figure 6.2. Variation in the Coefficient of Friction as a Function
of Shaft Alignment ....................................................................... 114

Figure 6.3. Reproducibility of Results for Rulon J ..................................... 115

Figure A.1. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................. 118

Figure A.2. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................. 118

Figure A.3. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................. 119

Figure A.4. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................. 119

Figure A.5. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 120

Figure A.6. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 120

Figure A.7. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 121
Figure A.8. Reciprocating Coefficient of Friction for Iglide T500 (PEEK):
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ........................................... 121

Figure A.9. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 122

Figure A.10. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 122

Figure A.11. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 123

Figure A.12. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 123

Figure A.13. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ........................................... 124

Figure A.14. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ........................................... 124

Figure A.15. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ........................................... 125

Figure A.16. Reciprocating Coefficient of Friction for Rulon J:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ........................................... 125

Figure A.17. Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 126

Figure A.18. Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ........................................ 126
Figure A.19. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................. 127

Figure A.20. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................. 127

Figure A.21. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 128

Figure A.22. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 128

Figure A.23. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 129

Figure A.24. 
Reciprocating Coefficient of Friction for Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 129

Figure B.1. 
Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude),
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................. 131

Figure B.2. 
Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude),
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 131

Figure B.3. 
Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude),
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................. 132

Figure B.4. 
Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude),
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................. 132

Figure B.5. 
Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude),
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................. 133
Figure B.6. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 133

Figure B.7. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm .............................................................................. 134

Figure B.8. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 134

Figure B.9. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm .................................................................................. 135

Figure B.10. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 135

Figure B.11. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm .................................................................................. 136

Figure B.12. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 136

Figure B.13. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm ................................................................. 137

Figure B.14. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 137

Figure B.15. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm ................................................................. 138

Figure B.16. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ................................................................. 138
Figure B.17. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ........................................ 139

Figure B.18. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ............................... 139

Figure B.19. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ........................................ 140

Figure B.20. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ............................... 140

Figure B.21. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ........................................ 141

Figure B.22. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ........................................ 141

Figure B.23. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm ........................................ 142

Figure B.24. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm ........................................ 142

Figure B.25. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................ 143

Figure B.26. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ........................................ 143

Figure B.27. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................ 144

Figure B.28. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ........................................ 144

Figure B.29. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................ 145
Figure B.30. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ................................................................. 145

Figure B.31. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................................................ 146

Figure B.32. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ........................................................................ 146

Figure B.33. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................................................ 147

Figure B.34. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ........................................................................ 147

Figure B.35. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium ........................................................................ 148

Figure B.36. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium ........................................................................ 148

Figure B.37. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm ................................................................. 149

Figure B.38. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ........................................................................ 149

Figure B.39. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm ................................................................. 150

Figure B.40. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm ........................................................................ 150

Figure B.41. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm ................................................................. 151
Figure B.42. Influence of Interface Temperature on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................. 151

Figure B.43. Influence of Interface Temperature on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm
and Rtm = 0.25-0.35 μm ................................................................ 152

Figure B.44. Influence of Interface Temperature on the Friction Coefficient:
velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................ 152

Figure B.45. Influence of Interface Temperature on the Friction Coefficient:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N,
atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm
and Rtm = 0.9-1.20 μm ................................................................ 153

Figure C.1. Model of Bearing Holder Spokes .................................................. 155
Figure C.2. Modeling the Test Bearing Holder as a Spring Mass System .......... 159
Figure C.3. Magnification Factor as a Function of the Frequency Ratio .......... 160
Figure D.1. Specifications of Bearing Holder Design - Front View ................. 173
Figure D.2. Specifications of Bearing Holder Design - Rear View .................... 174
Figure D.3. Specifications of Bearing Case .................................................... 175
Figure D.4 Specifications of Bearing Holder Redesign - Front View ............... 176
Figure D.5. Specifications of Bearing Holder Redesign - Rear View ................. 177
Figure D.6. Specifications of Bearing Case Redesign .................................... 178
Figure D.7. Modifications Made to Bearing Holder Redesign ......................... 179
Figure D.8. Specifications of Inertia Spring Spacer ....................................... 180
Figure D.9. Specifications of Inertia Spring Holders .................................... 181
LIST OF SYMBOLS

A  Area

a  Acceleration

b  Width of spoke

c  Distance from the neutral axis (beams in bending)

E  Modulus of elasticity

E_i Strain gage excitation voltage

E_o Wheatstone bridge output

ε  Strain

F  Gage factor of strain gage

F_f Friction force

F_{fs} Friction force contribution to a single spoke

F_i Inertia force

F_{ib} Inertia force of the bearing assembly

F_n Normal load

f_n Natural frequency (Hertz)

G  Amplifier gain

h  Thickness of spoke

I  Moment of inertia

K  Spring constant - each spoke

K_t Spring constant - total for all 3 spokes

L  Length of spoke

L_L Mechanical load of springs (low)

L_H Mechanical load of springs (high)

M  Moment
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>Mass</td>
</tr>
<tr>
<td>n</td>
<td>Factor of safety</td>
</tr>
<tr>
<td>ρ</td>
<td>Mass density</td>
</tr>
<tr>
<td>P</td>
<td>Force</td>
</tr>
<tr>
<td>R</td>
<td>Resistance</td>
</tr>
<tr>
<td>R_F</td>
<td>Gain resistor - fine span</td>
</tr>
<tr>
<td>R_G</td>
<td>Gain resistor</td>
</tr>
<tr>
<td>S_e</td>
<td>Fatigue strength</td>
</tr>
<tr>
<td>S_ut</td>
<td>Ultimate tensile strength</td>
</tr>
<tr>
<td>σ</td>
<td>Stress</td>
</tr>
<tr>
<td>σ_a</td>
<td>Stress amplitude</td>
</tr>
<tr>
<td>σ_m</td>
<td>Midrange stress</td>
</tr>
<tr>
<td>μ</td>
<td>Coefficient of friction</td>
</tr>
<tr>
<td>V</td>
<td>Volume of material</td>
</tr>
<tr>
<td>ω_n</td>
<td>Natural frequency (radians/second)</td>
</tr>
<tr>
<td>X</td>
<td>Amplitude of reciprocating shaft</td>
</tr>
</tbody>
</table>
1.1 PROBLEM DEFINITION

The last two decades have seen a substantial increase in the use of polymer friction bearings in machine construction. Because of their special antifrictional properties, polymers make excellent bearings for limited lubrication applications. In addition, polymer bearings improve reliability, extend service life, and simplify machinery design by reducing or eliminating lubrication requirements. Typical examples of polymer bearing use are in motor vehicles, food machinery and medical equipment.

Today, new families of high performance polymer compounds are being continually researched and developed to solve diverse, often difficult, bearing applications worldwide. One of these applications is in the development of long-life bearings and seals for the free-piston Stirling thermal machines.

Free-piston Stirling thermal machines are currently being developed by Sunpower Inc. of Athens, Ohio for numerous applications, including the conversion of solar power or biomass to electricity and domestic refrigeration. If successful, the possible market would
be tremendous — estimated at over eight million domestic refrigerators alone [19]; however, the thermal machines do have their problems — mainly, reliability.

The reliability problems of the free-piston Stirling thermal machine stem from the short operation life of its bearings and seals. The Stirling machine has very demanding operating conditions. The bearings and seals must undergo high reciprocating frequencies (0 ... 115 Hz) and surface velocities (velocity amplitudes ≤ 6.5 m/s). In addition, they must operate in a completely dry environment — no lubrication and little to no water vapor. Finally, the bearings and seals must withstand high temperatures and temperature swings.

From a bearing and seals standpoint, the operating conditions of the Stirling machine are very unique. In fact, few tests have been performed to evaluate the performance (friction and wear) of bearings and seals under such circumstances. Sunpower does have limited experience with some available materials in their prototype operations, but none have performed to their manufacturers’ expectations. There are several reasons for the discrepancies. First, material performance is not predictable or even consistent from one machine to another. Manufacturers commonly use the unidirectional pin-on-disk testing device which typically produces adhesion wear; however, the Stirling engine operates in a reciprocating (linear) oscillating motion. Sunpower’s experience indicates the presence of other wear mechanisms, including abrasion and fatigue. Second, manufacturers generally do not have application experience within the operating conditions of the Stirling machines. As a result, bearing and seal performance is simply not available.

An assessment of Sunpower’s problem revealed operating conditions that far exceed the applications discussed in the literature and in bearing material specifications. Given the potential differences in wear processes and operating conditions, extrapolation of existing friction and wear data for the design of a bearing or seal is simply not possible — at least not with confidence.
The objective of the EMTEC Bearing and Seal Development Program at the Center for Stirling Technology Research (CSTR) is to identify operating conditions and materials for improving the friction and wear performance of dry contact bearings and seals in the free-piston Stirling thermal machine. The purpose of this study is to investigate the frictional behavior of promising polymer wear pairs. Parameters that affect the frictional behavior of materials are common in tribology literature and have been reviewed. Results of the contrast tests will be obtained under contrasting (high and low) values of influential operating conditions such as load, velocity, time and counterface roughness.

To evaluate the effects of the parameters on friction, a special friction measurement system has been designed and built specifically for use in the CSTR wear test rig.

1.2 ORGANIZATION OF THESIS

The thesis begins with a brief introduction and problem assessment in Chapter 1. Chapter 2 presents the review of existing literature, including a basic review of friction, polymers and the parameters that influence polymer friction. Chapter 3 begins with a very brief review of existing friction measurement systems and the design process. The remainder of the chapter delves into the design of the friction apparatus and its associated components. Candidate materials and testing procedures are described in Chapter 4. Chapter 5 provides information about the test matrix, procedure and contrast testing. In addition, tests are presented graphically as well as in tabular form. Discussion of results, conclusion and performance appraisal is covered in Chapter 6. Appendices A, B, C and D provide details on instantaneous friction measurement, friction versus temperature, design calculations and component drawings of the test assembly.
2.1 FRICTION THEORIES

The friction coefficient has been used since the sixteenth century with the pioneering activity of Leonardo da Vinci, who noticed a coefficient of proportionality between frictional force and applied load (0.25) [24]. Two hundred years later, Amontons, a French glass lens polisher, rediscovered the work of da Vinci and assigned a proportionality value $\mu$ of 1/3 [24], which was invariable for the materials tested, provided the pressure remained constant. At a later time, Coulomb also studied friction; in 1785 he was awarded the Prize of the Academy of Sciences for his three memoranda which have come to be known as Coulomb's law or the Coulomb friction factor $(F_f = \mu_n F_n)$ [24].

The works of Amontons and Coulomb came to be known as the "roughness hypothesis." They hypothesized that friction is due to the deformation or displacement of material resulting from interlocking of the surface irregularities (or surface asperities) [4]. As such, they were able to explain why friction was proportional to load and independent of the contact area.
The roughness concept was maintained through the nineteenth century and into the twentieth century. Around 1920, the works of Hardy and Tomlinson -- theories into elastic and plastic deformation -- began to revive the adhesion theory attributed to Amontons, but discussed by Coulomb and even as far back as Leonardo da Vinci. The adhesion hypothesis is the concept that friction force is caused by the rupture of small junctions produced by adhesive forces over the "real" areas of contact [25]. Although the adhesion theory was discussed, it implies that friction force is proportional to the contact area, which was contrary to experimental evidence at the time.

It was not until the 1940s that progress was made in the molecular nature of friction. Holm [26], who was studying the properties of electrical contacts, investigated differences between real and apparent area (Figure 2.1). He expressed the friction coefficient as

$$\mu_s = \frac{\psi_s}{P}$$

E[2.1]

where $\psi_s$ is the "specific friction force" ($\psi_s = F_d/A_s$, where $A_s$ is the loaded contact area), and $P$ is the sliding contact pressure.

In a study published in 1940, Ernst and Merchant [28], who were investigating the metal cutting process, derived a relation for friction based on a geometric model. They found that for smooth surfaces, the friction coefficient could be approximated by $S/H$, where $S$ is the shear strength and $H$ is the hardness of the softer material.

Bowden and Tabor (1942) [25], who had experience in surface chemistry, also investigated the differences between apparent and real areas of contact by determining the electrical resistance between surfaces of metal cylinders of varying load. In this study, the contact area was found to be proportional to the applied load.
Figure 2.1. Difference Between Real \( A_r \) and Apparent \( A_a \) Area of Contact [27]

The findings by the three different groups of researchers (Holm, Ernst & Merchant, and Bowden & Tabor) pointed out that the real area alone determines the magnitude of the friction force. Their work led to a relatively simple form of the friction coefficient

\[
\mu = \frac{\tau_f}{p_c}
\]

where \( \tau_f \) is the shear strength of the microjunctons and \( p_c \) is the yield pressure of the softer material. Despite criticism, the adhesion hypothesis became the second main component in determining the friction force between sliding surfaces.

Although, Saka et al. (1984) [30] has shown that \( \mu_a \) varies as a function of three components: adhesion, deformation and ploughing. It is generally accepted, however, that friction force involves only two main components: the friction force due to adhesion \( F_a \) and the friction force due to displacement \( F_d \). Thus, \( F_{total} = F_a + F_d \).
2.2 FRICITION BEHAVIOR OF POLYMERS

Although the adhesion and displacement forces define the friction force between sliding surfaces, each is governed by a host of underlying factors. As such, there exists no such thing as a unique friction coefficient, only a value for a particular sliding system, so caution should be used in the interpretation of such data. The factors affecting the friction of polymers is summarized in Figure 2.2.

![Figure 2.2 Factors Affecting Polymer Friction [4]](image-url)
MECHANICAL PROPERTIES

Mechanical properties (Young’s modulus, hardness, shear strength, yield strength, creep, etc.) play an important role in the magnitude of friction. For instance, Bowden and Tabor [25] showed that the coefficient of friction of plastics could be interpreted as the ratio of the shear strength to the flow pressure of the softer material, \( \mu = \tau_f / P_c \), providing that deformation was plastic. For elastic deformation, Lancaster [4] points out that it is not possible to calculate \( A_e \) without making assumptions in the areas of real contact. As a result, \( \mu = K S W^{x-1} / E^x \), where \( E \) is Young’s modulus and \( K \) and \( x \) depend on the postulated surface. Also, Ernst and Merchant [28], as mentioned earlier, approximated the friction coefficient by \( S / H \), where \( S \) is the shear strength and \( H \) is the hardness of the softer material.

Although the predominant friction force for most sliding systems is adhesion (90% [32]), the deformation or displacement term can often be significant. In these cases, polymeric rigidity plays a significant role in friction.

Mechanical properties vary greatly for various polymeric materials because of their different molecular structure, chemical structure, reinforcing agents and many other factors. The basic plastic molecule is made up of a backbone of carbon atoms, to which other elements attach. Different backbone structures produce different properties. In thermoplastics, for example, the carbon atoms are arranged in a single chain, often with branches. In the case of thermosetting materials, the side chains interconnect forming two or three-dimensional networks. The difference in structure between the two main groups of plastics -- thermoplastic and thermosetting -- results in profound difference in their properties. The relative movements of the chains in thoroughly cross-linked polymers are far more restrictive than in linear polymers. Because of this restriction, thermosetting
materials are far more rigid and generally of higher strength than thermoplastics, especially at elevated temperatures. On this basis alone, one would expect that the performance of plastics in friction depends on whether a thermoplastic or thermosetting material is used. The work of Booser et al. [33] with bearings of both thermosetting and thermoplastic polymers have shown this experimentally. Furthermore, Pooley and Tarbor [36] showed that crosslinking PTFE, a thermoplastic, by neutron irradiation increases friction.

Molecular profile is also believed to affect friction [34, 37]. It has been suggested that the smooth profile of the rodlike molecules of PTFE permit easy slippage across planes parallel to the carbon axis (Figure 2.3); thus, its very low coefficient of friction. In contrast, polymers with bulky sidegroups or amorphous materials, such as polyethylene, become "hung-up", which leads to an increase in friction.

Figure 2.3 The Smooth Molecular Profile of PTFE (left) Compared to Polyethylene (right) [34]
In addition, chemical structure (those elements that attach to the backbone structures) has been suggested by Steijn [34] to influence friction. In an article on PTFE (polytetrafluoroethylene), Hanford and Joyce [35] suggested that the shielding of the charge of the carbon atom by the large fluorine ions minimizes interchain bonding forces, which results in low molecular cohesion. The much smaller hydrogen atoms found on the polyethylene molecule, on the other hand, is not as effective in screening the carbon atom; thus, the cohesion in this polymer is higher. The molecular structures of both PTFE and polyethylene are shown in Figure 2.4.

![Diagram of monomers of polyethylene and TFE](image)

Figure 2.4. Monomers of Polyethylene (left) and TFE (right) [34]

Pooley and Tabor [36] investigated the low friction of PTFE. In their study, modifications were made to the molecular profile by replacing some of the fluorine atoms by larger chlorine atoms or by CF3 groups. Consequently, the coefficient of friction increased, presumably, by interfering with interchain slip.

Another important aspect of polymer structure and friction is structural order. Buckley [38] points out that thermoplastics behave like metals. As such, many of the properties associated with metals can be carried over to the field of polymer science. In metallurgy, when one quenches or cools low iron steel from the molten state, different mechanical properties are obtained (phase diagrams): pearlite, bainite and martensite. Linear polymers
behave in a similar fashion. If a polymer is in a molten state, the molecules in the polymer have a random orientation (Figure 2.5). As the specimen is quenched, it transforms into a glassy state. By slowly cooling the molten material rather than quenching, the structure becomes crystalline and ordered. Once the crystalline form of the polymer has been formed, the polymer characteristics can undergo change. For instance, the polymer can be cold worked to orient or texture the structure, or it can be annealed to regenerate a random orientation. Also, if the glassy state is cold worked, it undergoes recrystallization, which develops orientation (Figure 2.5). The process -- crystallization -- can be controlled in some polymers so that their structure can be considered to have an order varying from amorphous to highly crystalline. The different structures result in differences in mechanical properties (Figure 2.5). As can be seen, the molten state is the weakest structure; it has the lowest mechanical strength. The mechanical properties of the crystalline and glassy states fall between metals and rubbers [38].

![Figure 2.5. Molecular Structures of Linear Polymers (a) Thermal and deformation treatments (b) Stress-strain relationships](image)
Furthermore, Bowden and Tabor [39] have performed considerable research with polymer structures and have indicated that adhesion (friction) is influenced by the orientation of the molecular chains in polymers. In their experiments, a steel hemisphere was slid on PTFE across two different directions relative to the polymer chain. They found a distinct difference in the friction properties for the two orientations of the polymer; sliding along the direction of the molecular chain produced a lower friction than sliding across it.

The excellent friction performance of common fillers such as graphite and molybdenum disulfide is directly related to the orientation of the layer-lattice structure. Within their crystal structures exist sheets (layers). Each sheet is held together by strong attractive forces, but the forces holding various sheets together are much weaker. As a result, the friction obtain from layer-lattice solids are affected by orientation. For graphite, Rabinowicz [54] obtained low friction coefficients (= 0.1) when the sliding took place on a face parallel to the sheet direction and much higher friction coefficients (= 0.3) when sliding took place perpendicular to the face. Similar results were obtained for molybdenum disulfide [55].

Moreover, the mechanical properties of polymeric resins can be enhanced by the direct addition of reinforcing agents. Such additives include graphite, molybdenum disulfide, glass, copper, bronze, aluminum oxide, etc. The purpose of these reinforcements is to improve strength, stiffness and resistance to deformation. How fillers alter the properties of PTFE is illustrated in Tables 2.1 and 2.2.

The list goes on, but it is not the intent of this work to present every effect, only to make the reader aware of some of the more important mechanical factors affecting polymer friction.
Table 2.1 - Properties of Typical* Reinforced "Teflon" TFE Resins [40]

<table>
<thead>
<tr>
<th>property</th>
<th>Ratio of improvement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resistance to wear by rotating shafts</td>
<td>Over 1000 times</td>
</tr>
<tr>
<td>Resistance to initial deformation under load</td>
<td>30% to 60%</td>
</tr>
<tr>
<td>Stiffness</td>
<td>2 to 3 times</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>3 times</td>
</tr>
<tr>
<td>Resistance to creep</td>
<td>2 to 3 times</td>
</tr>
<tr>
<td>Thermal dimensional stability</td>
<td>2 times</td>
</tr>
<tr>
<td>Hardness</td>
<td>10% to 15%</td>
</tr>
</tbody>
</table>

*TFE can be compounded with Fiberglass, graphite, bronze or other fillers to give these improvements in mechanical properties.

Table 2.2 - How Fillers Alter Properties of "Teflon" Resins [40]

<table>
<thead>
<tr>
<th></th>
<th>Unmodified TFE Resins</th>
<th>TFE with 25% of Typical Filler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deformation, %</td>
<td>25</td>
<td>1.7</td>
</tr>
<tr>
<td>(load of 2,000 psi, applied for 24 hr.)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wear, mg./hr.</td>
<td>0.74</td>
<td>0.0015</td>
</tr>
<tr>
<td>(Weight loss in sleeve-bearing test; 35 psi., 150 rpm., 1/4-in. S/S shaft)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient of linear thermal expansion, in./in.-°F</td>
<td>6.9x10^-5</td>
<td>3.6x10^-5</td>
</tr>
<tr>
<td>Coefficient of friction (For low speeds with 2,000-lb. load at room temperature)</td>
<td>0.016</td>
<td>0.031</td>
</tr>
</tbody>
</table>

COUNTERFACE SURFACE ROUGHNESS

Santner and Czichos [5] have investigated the friction coefficients of four thermoplastic polymers: PA66 (polyamide filled with 20% carbon short fibers), PA6 (polyamide - nylon composition not given), POM (polyoxymethylene) and PETP (polyethyleneterephthalate) and have found that friction coefficients depend upon counterface material. The friction coefficients of the four thermoplastic polymers first decreased with increasing surface roughness (Figure 2.6) and reached minimum values at about 0.2-0.3 μm Rz (mean peak-to-valley roughness). Once the minimum value was reached, further increases in surface roughness resulted in higher friction. According to
Rabinowicz, the friction tends to be high with very smooth surfaces, because the real area of contact (Ar) grows excessively; whereas with rough surfaces, the friction is high because one surface has to be lifted over the asperities of the other. In other words, adhesion forces dominate for small values of surface roughness, and abrasive processes prevail for higher surface roughnesses [6,7]. Somewhere in the intermediate range of roughnesses, friction is at its minimum value. Since the coefficient of friction is in many instances the driving factor (efficiency rather than wear), bearing manufacturers often provide ideal shaft roughnesses.

![Graph](image)

Figure 2.6. Friction of Polymers as a Function of Counterface Roughness: ○ = PA66; [] = PA6; Δ = POM; x = PETP [5]
TEMPERATURE

The effect of temperature on the friction of plastics has been studied by a number of investigators. Flinn and Trojan conducted tests on specimens of cellulose acetate at different temperatures and found that high strength and brittle behavior are found at low temperatures, while lower strength and ductile performance are found at higher temperatures (Figure 2.7a). In addition, they noted a change in the modulus (E) with temperature for a group of polymers (Figure 2.7b). As mentioned previously, mechanical properties influence friction; thus, a change in these properties will be reflected in the magnitude of friction.

Lancaster [4] has examine the frictional properties of various polymers at various ambient temperatures and has concluded that thermal effects play an important role in the friction of polymers. He attributes the decrease or increase in friction due to adhesion, \( \mu \sim \tau_f/p_c \), with the change in the magnitudes of shear strength of the junctions and hardness (or elastic modulus) brought about by the change in temperature. Both properties -- shear strength and hardness -- decrease with temperature, but not at the same rate; consequently, friction may increase or decrease depending on the polymer used. An example of the variations of friction obtained during the sliding of a steel ball over various polymer disks at various ambient temperatures is shown in Figure 2.8.

Fort [42] determined the friction of several plastics -- cellulose acetate, nylon, PAN (polyacrylonitrile), PET (polyethylene terephthalate), and PTFE -- against themselves in the temperature range of 25 to 150 °C; the results are shown in Figure 2.9. It can be seen that the coefficient of friction for all the plastics vary (increase or decrease) with temperature. Like Lancaster, Fort attributes this behavior to the formation and subsequent breakage of adhesion bonds expressed as \( F = \tau_f A_p \). Friction, he asserts, is the result of the changes in
Figure 2.7. Effects of Temperature on the Mechanical Properties of Polymers (a) Effect of temperature on the stress strain curve of cellulose acetate (b) Effects of temperature on the modulus of various resins [41]
Figure 2.8. Variation of Friction with Temperature for Steel Sliding on Various Polymers at Low Speeds [4]

At low temperatures, the increase in $A_r$ outweighs the reduction in shear strength. However, at higher temperatures, the bulk begins to soften; as a result, the reduction in shear strength far exceeds the increases in contact area. PEEK (polyetheretherketone) exhibited similar characteristics [43].

Temperature can also change the molecular structure of polymers, thereby changing the mechanical properties of the material. Amorphous polymers exists at a certain temperature range (Figure 2.10). Below this range, the amorphous structure (wriggling earthworms [34]) transforms into a glassy state characterized by fixed or frozen chain motion (high degree of stiffness) -- thus, the term glass transition temperature.
Above the transition temperature, the polymer is in the rubbery state (stiffness decreases with temperature). Again, one can easily visualize how the transition from a glassy to a rubbery state can affect friction.

A semicrystalline polymer, by the way, will have a much smaller decrease in stiffness with increased temperature. Only the amorphous portion of the polymer will undergo the transition to the glassy state. Consequently, the reduced stiffness results in an overall increase in toughness.

Frictional heat at the sliding surface may be of some influence. Vinogradov and Bezborodko [44] investigated flat rings of plastic in facewise contact with flat rings of steel at speeds of 45 cm/sec. They found that the coefficient of friction depended upon how fast the frictional heat could be removed. Friction was lower when heat removal was less efficient; thus, friction is influenced by the thermal conductivity of polymers. This aspect was further demonstrated by Vinogradov et al. [45].
Figure 2.10. Variations of Modulus or Tensile Strength as a Function of Temperature for a Thermoplastic [41]

GEOMETRICAL ARRANGEMENT - SINGLE OR MULTIPLE CONTACTS

Tribological studies in the behavior of materials tend to use simple test geometries -- partly because of simplicity, ease of construction, level of funds, degree of control, as well as a host of other factors not mentioned. However, different geometries can produce different friction and wear processes (adhesion or displacement) [46]. For example, the most common method of measuring friction is the pin-on-disk geometry; the block-on-ring is also prevalent. But, the friction and wear results from different geometries are not always in agreement. Dickens et al. [3] have shown in their experiments that the friction
coefficients in a rotating line contact were appreciably higher than those in a pin-on-disk arrangement. It has also been found that the tribological behavior of polymers in oscillating contacts can be quite different from that in unidirectional motion (continuous friction measured on a pin-on-disk, for instance) [47]. In fact, Blau [46] cites that different processes can dominate within the same basic geometry just by the orientation of the experiment. For instance, if the pin-on-disk is oriented such that the disk is horizontal, loose debris can remain in the wear track and be mechanically processed. On the other hand, if the disk was turned vertical, loose debris would fall clear of the contact zone. In addition, Blau notes that the high Hertz stress produced by the hemispherical pin-on-disk geometry widens the distance between contact spots or decreases the number of larger contacts as the track width increases. Several different geometries are illustrated in Figure 2.11.

LOAD

Amonton's first law \((F_f = \mu L)\), which claims that friction force \(F_f\) is proportional to the normal force (load) \(L\), is generally well obeyed, but not by plastics. Because of its importance on the friction of plastics, load has been well researched. Research performed in lieu of the problems associated with textile friction has found that the friction of polymers may be written as

\[ F = kL^n \]  

where \(F\) is the force of friction, \(k\) is a proportionality constant, \(L\) is the normal load and
n < 1. From equation 2.3, one can easily visualize why the coefficient of friction decreases as the load increases [34].

Figure 2.11. Six Different Test Geometries: four-ball, block-on-ring, pin-on-disk, counter-rotating disks with adjustable slip, pin-in-V-block, and crossed cylinders [46]
Stein [34] points out that the underlying cause for the variation of friction with load is thought to be the variation of the real contact area $A_r$. In addition, there are indications that the shear strength also increases slightly with load causing an increase in friction ($F = \tau_l A_r$) [48].

The value of $n$ in equation 2.3 has been found experimentally by numerous researchers (Stein [34] cites references) to lie between 0.66 and 0.96. A list of $n$ values for various polymers is given in Table 2.3. In one of the more profound works, Allen [49] measured the variation of friction over eleven decades of speed for PTFE (Figure 2.12) and found $n$ to be 0.86.

<table>
<thead>
<tr>
<th>Plastic</th>
<th>$n$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cellulose acetate</td>
<td>0.96</td>
</tr>
<tr>
<td>Nylon (undrawn)</td>
<td>0.90</td>
</tr>
<tr>
<td>Nylon (drawn)</td>
<td>0.80</td>
</tr>
<tr>
<td>Viscose rayon</td>
<td>0.91</td>
</tr>
<tr>
<td>Nylon 66</td>
<td>0.74</td>
</tr>
<tr>
<td>Polyvinylidene chloride</td>
<td>0.87</td>
</tr>
<tr>
<td>Polyethylene</td>
<td>0.70</td>
</tr>
<tr>
<td>PTFE</td>
<td>0.73</td>
</tr>
<tr>
<td>Nylon 610</td>
<td>0.78</td>
</tr>
<tr>
<td>PTFE</td>
<td>0.86</td>
</tr>
<tr>
<td>PTFE</td>
<td>0.85</td>
</tr>
</tbody>
</table>

SPEED

The third empirical law of friction, Coulomb's law, states that friction force is independent of the sliding velocity $v$ [27]. However, recent work has shown this as a
A gross oversimplification. Wanatabe et al. [50] conducted tests on the friction of nylon 6 (counterface-steel) with various speeds and loads and found that the coefficient of friction varies with speed (Figure 2.13). In addition, McLaren and Tabor [51] have published reports illustrating the influences of speed on the coefficient of friction for various polymers, including PTFE, polyethylene, polypropylene, nylon 66, polystyrene, PMMA and silicone resin; results of their study are provided in Figure 2.14. Similar trends have been reported by other researchers [3, 8, 52, 53].

Figure 2.12. Variation of Friction Force and Coefficient of Friction with Load [49]
Lancaster [4] attributes the rise in friction to temperature. He explains that variations in friction are not directly related to variations in load or speed, but indirectly through their changes in temperature. Frictional heating and thermal softening lead to an increase in the real area of contact, which relates to a rise in the coefficient of friction. At higher PV (pressure-velocity) values, surface melting occurs (molten material acts as a lubricant) and friction is reduced. However, Mclaren and Tabor [51] measured the friction of several polymers at engineering speeds where friction heating could hardly be of any consequence and concluded that friction rise with speed was not due to thermal softening. They did
agree, however, that the drop in friction at high speeds was very likely the result of surface melting.

Figure 2.14. Variations of Friction with Speed for Various Polymers [51]
ENVIRONMENT

Environment is an important factor in the friction of polymers. For example, Fusaro [56] examined the wear and friction behavior of polyimide films in dry air, dry argon and moist air at different temperatures and found that polyimides are affected by the type of atmospheric gas molecules. In the experiments of Lancaster [4], the friction of linear crystalline thermoplastics, such as polyethylene or polyamide, showed only small differences in friction in air, inert gas or vacuum. On the other hand, friction for amorphous thermoplastics, such as polystyrene or polymethyl methacrylate, and thermosetting resins tended to be 50% greater in vacuum than in air. The changes in friction were attributed by Lancaster and Rabinowicz [4, 54] to the change in environment, which ultimately influenced the shear strength of the junctions during sliding.

If a metal counterface is used, one must also consider the effects of metal contamination. In general, an unlubricated metal is contaminated with a number of films, including an oxide layer (produced by the reaction of air with metal), except noble metals, an absorbed gas layer (molecules of water and gas atmosphere) and a contaminant layer (grease or oil). The presence of these films, Rabinowicz [54] points out, greatly affects friction by reducing surface interaction of contacting materials.

Furthermore, bearing additives (use of fillers) can be environmentally dependent. The low friction of graphite is made possible only by the presence of moisture or some other volatile material. The function of which, according to Rabinowicz [54], is to help split off the layer-lattice sheets. Molybdenum disulfide, on the other hand, requires no auxiliary material for low friction. Subsequently, molybdenum disulfide would be the filler of choice in an inert atmosphere.
2.3 EARLIER STUDIES

All the topics discussed in the previous section -- mechanical properties, surface roughness, temperature, geometrical arrangement, load, speed, environment -- illustrate one point: there exists no such thing as a unique friction coefficient, only a value for a particular sliding system. As such, comparison between the findings published by other investigators is fraught with difficulties -- either the speed was different, atmosphere was not controlled, load was larger and so on. Nevertheless, the frictional performance of several bearings will be mentioned along with a brief description of the experimental method.

Bushan and Wilcox [1] reported a friction coefficient of 0.23 for Rulon J. Data was obtained using a unidirectional sliding mode (pin against cylinder) with a load of 0.69 MPa and a speed of 1.2 m/s. In addition, the counterface material was Nitrided SAE 7140 Steel (Rc 62). Santner and Czichos [5] published a friction coefficient of 0.20 for PTFE. The measurement was obtained with a pin-on-disk apparatus under the following conditions: sliding velocity 0.1 m/s; normal force 20 N; temperature of the surrounding air 23 °C; relative humidity 50%. Similar results for unfilled PTFE (0.20) were recorded by Blanchet and Kennedy [53], who performed their test in air on an oscillatory pin-on-disk rig. Experimental details were as follows: load 52.4 N; speed 0.1 m/s; counterface material 316 stainless steel with a roughness of Ra = 0.02 μm.

Friedrich et al. [12] investigated the friction of PEEK composites on a self-designed pin-on-disk-type testing machine (allowed pins of material to be run against steel rings of 100 CR, HRC 62 and surface roughness of Ra = 0.06 μm). At a temperature of about 100 °C and a PV value of 1 MPa•m/s, the coefficient of friction was approximately 0.13 for PEEK + 10 wt.% CF + 10 wt.% PTFE + 10 wt.% graphite. A friction coefficient of 0.05
was recorded by Briscoe et al. [9] for a PEEK composite (90% PEEK, 10% PTFE). In their experiments, a "scuffing" device with a 10 N load and a velocity of 0.45 mm/s was used; temperature was 100 °C. Schelling et al. [8], on the other hand, reported a coefficient of friction of 0.42 for pure PEEK using a pin-on-disk apparatus. Conditions were as follows: temperature 100 °C; load 0.5 MPa; counterface material X 20 Cr 13 stainless steel with hardness of 273.9 HV 10; speed 0.25 m/s.

Bushan and Wilcox [1] also performed tests on various polyimides including SP-21 (polyimide with 15% graphite powder), Dixon-PI-HL-800-2 (polyimide with 15% CdO-graphite-Ag), and FMI (polyimide with 50% graphite weave). Their coefficients of friction were 0.47, 0.37 and 0.46 respectively. Testing conditions were the same as those listed under Bushan and Wilcox - Rulon J above.
DESIGN OF FRICTION

MEASUREMENT SYSTEM

3

3.1 INTRODUCTION

To fulfill the capability of determining bearing performance as outlined in subtasks 2 and 3 of the CT-30 project, the test rig developed by Bjorn Carlsson and Sunpower Inc. of Athens, Ohio was to be retrofitted with a device capable of measuring reciprocating friction. To determine the nature of this device, an intensive literature review of previous test machines was conducted from existing literature. Unfortunately, all of the known friction-measurement designs were systematically eliminated by fit, form or function.

There is little likelihood that an existing design would fulfill all the necessary requirements of a "special" purpose machine; to do so usually requires one's recognition in the earliest stages of design. Consequently, the limitations associated with the test rig geometry prompted the final decision to develop a new friction-measurement system. The resulting friction-measurement device has the capacity to investigate the effects of load, speed, atmosphere, temperature, counterface material and surface roughness on reciprocating friction and wear.
The friction measuring device is not equipped with a heating element to control or elevate operating temperatures, but it does allow the bearing temperature to be monitored through the use of a thermocouple. In the future, a heating element could be installed on the device; however, the test rig motor and bearing holders must be redesigned due, in part, to their temperature sensitivity.

3.2 METHODS OF MEASURING FRICTION

Hundreds of friction force measuring devices are described in the tribology literature. These methods can range from simple to complex, depending on one’s accuracy and the ease with which friction can be measured. Perhaps the simplest method of measuring static friction is the pulley set-up shown in Figure 3.1. A known load N is placed on material A. Weights are then added to the cup until material A begins to slide. The coefficient of friction, \( \mu \), between materials A and B can be determined by dividing F, the summation of the cup weight, applied weights and string or wire weight by the normal load N.

More sophisticated friction-measurement devices are found in tribometers equipped for sliding wear tests. In these devices, the friction force is obtained by measuring strains on the surface of an elastic member. Of the numerous experimental methods for measuring strain, the most widely used device is the electrical-resistance strain gage. The basic principle behind the use of strain gages is simple: when a force is applied to the gage, its electrical resistance changes. This change in resistance can be measured by a Wheatstone bridge or a commercial strain sensing device and converted into a force. Thus, a calibrational relationship exits between force and voltage or force and strain.
3.3 FRICTION MEASUREMENT SYSTEM

The purpose of the friction-measurement system was to complement the existing wear test rig developed by Bjorn Carlsson and Sunpower. With the advent of a friction apparatus, the capability of measuring bearing performance will be fulfilled. As a result, the influences of load, speed and surface roughness on the dry-sliding friction and wear of polymeric journal bearings on steel can be measured and evaluated for reciprocating applications.

Based on the review of earlier measurement systems, it was decided that a new means of friction measurement must be constructed, rather than an alteration of an existing design. This new device will be designed specifically for this project. Although no existing designs will be incorporated, it will employ the basic laboratory principle for measuring friction force -- the force transducer.

Friction Force, \( F = \text{cup weight} + \text{applied weight} + \text{string/wire weight} \)

Static Friction Coefficient, \( \mu = \frac{F}{N} \)

Figure 3.1. Method of Measuring Static Friction
DESIGN PROCESS

Shigley describes the complete design process in Figure 3.2. The process begins with a recognition of need, and following many trials, it ends with a solution to satisfy that need. This project (CT-30) also began with a need – the need to measure bearing performance under dry reciprocating conditions. What follows is a brief description of those trials undertaken during the design of the friction measurement device.

Figure 3.2. The Phases of Design [18]
To begin, the extent of the problem must be defined. As such, all operating quantities and their associated limitations should be specified at the onset. The general design guidelines for the retrofit are as follows:

- **Input quantities**
  - The input quantities are load force, friction force and bearing temperature.

- **Dimensions and limitations of operation**
  - To fit within the current test rig constraints, the part must not exceed 118 mm in diameter and 32 mm in length. In addition, an internal diametrical opening of at least 24.5 mm is required for movement of the piston shaft. Length limitations may be exceeded if alterations are made to the alignment bearing holders.
  - The minimum life expectancy of the part should be $5 \times 10^8$ cycles, 2300 hours, at 60 Hertz for a reversed load of approximately fifty (50) Newtons in magnitude.
  - Operating temperatures will range from 22 to 100°C.
  - Operating environment will depend upon the atmosphere of choice. Primarily, the atmosphere will be either air, helium or nitrogen.

- **Output quantities**
  - In order to link up with Lab-SE, the data acquisition board, and present data acquisition programs, the load and friction output signals should be conditioned for a response range of -10 to +10 Volts. The bearing temperature will be monitored through the use of a thermocouple and a calibrated digital thermometer.
FRICTION MEASUREMENT DEVICE

Originally, the wear bearing holder was to be fitted with three load cells. These load cells would be connected to a feedthrough, and their signals increased by an amplifier, before being read by a data acquisition system. However, concerns about bearing load accuracy and possible replacement costs prompted the switch from load cells to the more cost-effective strain gages.

A new bearing holder was designed and manufactured with more promising features. Mainly, it offered the advantage of measuring the loads applied to the bearing rather than the outer structure of the bearing holder. This design resembled a spoked wheel. On one of the three elastic spokes, four strain gages were mounted and wired to a feedthrough connector. The connector was linked to a toggle switch that reconfigured key strain gage positions within a Wheatstone bridge circuit. By flipping the switch, the user could select either load or friction measurement. The output of the Wheatstone bridge -- a small voltage differential -- was increased by an instrumentation amplifier before it was read by the data acquisition system. Also, the bearing temperature was monitored by a thermocouple placed within the bearing case and linked to a digital thermometer.

Results from preliminary friction measurements revealed a number of complications: friction response degraded at high amplitudes, friction magnitude depended on shaft alignment and higher harmonics appeared in the square wave signal. The first two problems were reduced by the reorientation of the resonance springs in the spring cartridge and the implementation of alignment washers in the spring seat to shaft connection. The elimination of the higher harmonics, however, meant a redesign of the bearing holder with an additional criteria in mind -- system rigidity.
In the second design, an attempt was made to optimize system rigidity yet maintain the current levels of load and force resolution. To perform this task, the spokes were shortened in length, thinned in width and thickened in depth. Also, two strain gages, instead of the initial four, were trimmed and placed as close as possible to the outer bearing-holder ring. In addition, the bearing case and inner bearing holder ring were redesigned to reduce system mass.

The preliminary results were disappointing. While the friction resolution increased, the load resolution was drastically reduced. Even though a load-resolution loss was expected, its severity was not.

Rather than designing a third device, it was decided that the design would undergo some changes. Special thrust bearings were mounted on both sides of the bearing holder. Unlike earlier versions, this design only measured friction force. Previous designs used inertia springs which secured the outer ring of the bearing assembly to the alignment bearing holder, but the contact between the two parts reduced the load applied to the bearing. Because the reduction in load could not be determined, the load to the bearing had to be measured. With the advent of special thrust bearings, the load applied to the bearing holder via the two mechanical springs is essentially the load applied to the bearing since each thrust bearing has a very low coefficient of friction, \( \mu = 0.0015 \); thus, the need for load measurement was unnecessary. The inertia springs, however, were not eliminated from the system. They were simply removed from the bearing assembly and located in the spacer.

It was believed that this latest version, shown in Figure 3.3, would adequately fulfill the friction-measurement capabilities that were set forth at the start of this project. As such, it was used in the collection of data for this thesis.

All the part changes required to retrofit the wear test rig with the friction measurement
Design of Friction Measurement System
device are described in Figure 3.4. A description of the original test rig is given in [17].

Figure 3.4. Part Components Required to Retrofit Wear Test Rig
MEASUREMENT SYSTEM

The system begins at the signal conditioning box (3) (see Figure 3.5). The strain gage excitation voltage is sent via the housing feedthrough to each strain gage. If the test bearing assembly (2) is correctly installed in the 60 Hz wear test machine (1), the interstitial contact between the pressed test bearing and the reciprocating shaft results in a sliding friction force. This dynamic force is sensed by strain gages and passed back to the signal conditioning box as an electrical signal.

Nestled inside the conditioning box is a device for detecting small voltages. Essentially, the conditioning box wires the two active strain gages and two precision resistors in a Wheatstone bridge configuration and amplifies the potential difference between the output terminals of the bridge circuit. After amplification, the output passes through an AC coupling switch. If AC coupled, only dynamic voltages pass.

A shielded coaxial cable links the output of the signal conditioning box to the input of Lab-SE**, a data acquisition card installed in a Macintosh SE* (5). Although the Lab-SE card does provide the means for signal detection, it does not provide the method. Rather, methods for signal retrieval, manipulation and storage are performed by data acquisition programs written by the user in LabVIEW 2.0.6**.

Since the speed performance of the Macintosh SE was sluggish at best, its service was limited to the retrieval of signals from Lab-SE, and their subsequent storage to floppy disks (6). Data manipulation and analysis, on the other hand, were left to the speed and performance of a Macintosh IIci* (7).

* Macintosh SE and Macintosh IIci are trademarks of the Apple Computer, Inc., 20525 Mariana Avenue, Cupertino California 95014.
** Lab-SE and LabVIEW are trademarks of the National Instruments Corp., 6504 Bridge Point Parkway, Austin, Texas 78730-5039.
To measure bearing performance, a type T thermocouple was placed against the bearing surface in the test bearing assembly and connected to an Omega CL 23 digital thermometer.

Figure 3.5. Friction Measurement System
DEVELOPMENT OF THE TEST BEARING ASSEMBLY

The test bearing assembly, Figure 3.6, is comprised of the bearing holder (1), inertia springs (2), connector subassembly (3 - 4), bearing case subassembly (5 - 10) and thrust bearings (11). This assembly drawing is just a general description of the type of components within the bearing assembly. Throughout the design process, many of these parts are either eliminated, redesigned or relocated.

---

1. Test bearing holder
2. Inertia springs, five (5) total
3. Multipin connector
4. Socket head cap screw, two (2) total
5. Test bearing case
6. Test bearing
7. Spacer
8. Inner retaining ring
9. Outer retaining ring
10. Washer
11. Thrust bearings, six (6) total

Figure 3.6. Test Bearing Assembly
BEARING HOLDER

As stated earlier, the original wear bearing holder was to be fitted with load cells; however, there were some initial concerns about bearing load accuracy and complete installation costs. To circumvent these problems, a simple yet inexpensive strain gage technique would be used. As a result, a new test bearing holder had to be constructed.

Keeping in mind the quantities required for retrofit and the principle behind strain gage operation, the new design would consist of two concentric cylinders joined by three rectangular spokes. The outer diameter of the outer cylinder and the length of the inner and outer cylinders will remain the same, 117.5 and 32 mm respectively. In addition, the inner diameter of the inner cylinder will be 45 mm, hence, the bearing holder can incorporate the existing bearing case subassemblies. All other dimensions will be determined by maximizing bridge output.

Bridge output is governed by the following functions: gage factor, bridge excitation, strain and number of active strain gages. For example, if two active strain gages are subject to equal and opposite strains (beam in bending), the bridge output would be

$$E_0 = (F \varepsilon \times 10^{-3}) E_i / 2$$  \hspace{1cm} (Meas. Grp,TN-507)  \hspace{1cm} E[3.1]

On the other hand, if four strain gages are active (full bridge) with pairs subjected to equal and opposite strains, the bridge output would be

$$E_0 = (F \varepsilon \times 10^{-3}) E_i$$  \hspace{1cm} (Meas. Grp,TN-507)  \hspace{1cm} E[3.2]
or twice the response of the half-bridge circuit.

Ideally, all four of the factors should be maximized; however, some are partially beyond control. The two that have the most impact on design are strain and the number of strain gages. To minimize potential noise and maximize bridge output, it was decided that four strain gages were to be used. Furthermore, these gages -- two pairs front and rear -- were to be mounted as close as possible to the outer cylinder fillet radius; the fillet radius was expected to provide the largest strain gradient.

The next step is to select the appropriate strain gage and terminals for the task. This will provide the minimal space necessary for installation and a general idea of the constraints or requirements that have to be satisfied. In this case, the general purpose WK-13-062AP-350 strain gage and CPF-50C bondable terminal were chosen from the Vishay Measurements catalog. The bondable surfaces measures 7 mm in length and 4 mm in width for the strain gage and 4.5 mm in length and 4 mm in width for the bondable terminal. In short, sensor installation would require a surface of 13 mm in length and 9 mm in width. But, for ease of installation, it was decided that the spoke should be 19 mm in length and 21 mm in width. The inner ring thickness was arbitrarily set at 6 mm.

Now, the only dimension not specified is the thickness of the spoke. Again, several important factors play a crucial role in its determination. First, the strength of the spoke should be designed to meet the minimum fatigue life of 2300 hours. Second, the strain gage has performance limits. As a result, strain and cyclic endurance should not exceed the performance characteristics of the strain gage or its adhesive. Next, one must consider the calibration resolution. If the thickness of the spoke is too large, small strains will occur. Consequently, the difference between a friction force of twenty-two (22) Newtons and twenty-three (23) Newtons might be indiscernible. Finally, there was the decision of whether to use two, three or even four spokes.
Taking into consideration the factors mentioned above, the thickness was determined to be two (2) mm if a fifty-six (56) Newton load was equally dispersed over three spokes. Methodology and calculations can be seen in Appendix C. The new bearing holder design is shown in Figure 3.7. For the detailed design, see Appendix D.

![Figure 3.7. The Original Bearing Holder Design](image)

Upon arrival of the fully instrumented bearing holder from the shop, the test bearing assembly was ready for calibration. The friction force calibration was performed outside the test rig. To calibrate for the friction force, known loads from a force gage were placed on the bearing case, and the resulting voltage responses were recorded. The load calibration, on the other hand, was performed while the bearing assembly was in the test rig and the signal conditioner box was set to load. Again, the force gage was used to supply the load, but this time it was to the bearing holder. Varying loads were applied to
both sides of the bearing and their corresponding voltage responses were recorded. The results obtained from the calibration tests are presented in Figures 3.8 and 3.9.

After the bearing assembly was calibrated, it was time to rid the system of any bugs. Preliminary testing showed that this design was successful in measuring both friction and load, but the results also revealed a number of complications. The complications were as previously stated: friction response degraded at high amplitudes, friction magnitude depended on shaft alignment, and a higher harmonic was present in the friction response. If accurate bearing analysis was to take place, the problems had to be rectified or severely reduced. Further testing of the test rig resulted in a number of explanations. First, the shaft was not aligned properly; an ideal positioning of the shaft required several hours, at best,

\[
y = 0.14725 + 0.11931x \quad R^2 = 1.000
\]
but it could be done. Rubbing of the shaft on the spring cartridge and uneven wear of the piston wear band quickly became evident. It was thought that the resonant springs were not compressing straight; thereby, additional loads and excessive vibrations were transferred to the shaft. One way to correct the problem was to grind the springs in their compressed working height rather than in the relaxed position. This was a plausible solution; however, it was impractical because the ends of the spring would have to be clamped for grinding. The only expedient solution was to reorient the springs in the spring cartridge until the effects were minimized. Also, alignment washers were installed to allow the spring seat to pivot (see Figure 3.10 for details). After the shaft was aligned using the above methods, the first two complications were reduced significantly.

Now, the only unresolved issue was the presence of a higher harmonic in the friction response signal. It was thought that the harmonic was the result of either bearing assembly
movement (inertia springs not strong enough to keep the bearing assembly in a firm position) or the web flanges and inner bearing holder mass (inner ring & bearing case subassembly) behaving as a separate vibratory system. One, both or none of the above problems could be contributing to the harmonic, therefore several more tests had to be conducted. At first, the inertia springs were switched with others (twice their spring constant), yet the harmonic remained the same. Then, the springs were replaced by a

Figure 3.10. Misalignment of System (a) An exaggerated view of the misalignment problems (b) With the implementation of alignment washers and reorientation of the resonant springs, misalignment problems were reduced.
rubber washer with a very high spring constant. Again, the harmonic remained the same. In conclusion, the harmonic was not due to bearing assembly movement. The only possible explanation was the concept of a resonance within the bearing assembly.

Essentially, the case bearing holder, inner ring of the bearing holder and elastic spokes act as a spring mass system. The amplitude of an externally excited spring mass system generally depends upon the natural frequency of the system and its excitation frequency. When a component of the excitation frequency approaches one of the natural frequencies of the system, the amplitude can become very large. This condition is often referred to as resonance; it has a very disastrous effect of increasing stresses and strains which have the potential for causing failure. It was believed that the harmonic present in the square wave was the result of an excitation frequency too close to the natural frequency of the system. Generally, the frequency of a designed system should be at least one hundred times greater than its excitation frequency to maximize system rigidity and minimize vibratory noise.

The analysis began by comparing the estimated natural frequency of the spring mass system with the higher harmonic from the friction tests. Theoretically, the natural frequency of the system was 730 Hz (see Appendix C for calculations). Experimentally, the higher harmonic varied in response from 660 to 680 Hz (Figure 3.1). Certainly, these numbers are close enough to warrant further investigation. After all, the theoretical frequency was based on several assumptions, mainly, that the spoke was an exact parallelepiped.

To determine if the vibratory system was creating the harmonic, the inner bearing system mass had to be changed. A change in the system mass would change the frequency of the harmonic if it was indeed caused by the vibration of the spring mass system. In order to change the system mass, a new bearing case was machined out of brass. Theoretically, the natural frequency of the new system was 508 Hz (Appendix C). The
experimental frequency of the harmonic changed to 425 Hz. Consequently, the vibratory system was deemed responsible for the harmonic.

Upon completion of the tests, an immediate design error was recognized. Originally, the inner bearing system was designed on the assumption that it would be subjected to a sinusoidal excitation frequency of 60 Hz; however, this assumption was wrong. In actuality, the system is excited by a 60 Hz square wave. Instead of having a system that was over one hundred times the excitation frequency (600 Hz), the design was much lower since the system actually sees odd harmonics of sixty hertz (60, 180, 300, 420, 540, ...).

In lieu of the testing, it became apparent that the design simply would not perform to specifications. Both the friction measurement capability and the life expectancy of the bearing assembly were compromised by the presence of the harmonic. Something had to
be done. Either a new design must be built emphasizing system rigidity or modifications must be made to the existing assembly.

At first, all efforts were focused on the modification of the current assembly, but the only change that could be performed was a reduction in system mass. As a result, the inner bearing case and inner holder ring were machined. Their effects, although promising, were minimal. Only minimal mass could be eliminated from the system. Another route must be taken.

What if the system mass was altered such that the natural frequency of the system would be at an even harmonic? Would the effects of the forcing excitation, odd harmonics of 60 Hz, be eliminated or severely reduced? Again, testing was done, but the results were inconclusive. The only other course of action was to redesign the bearing assembly.

In the redesign, an attempt was made to optimize system rigidity yet maintain the current levels of load and force resolution. By maximizing key variables within the basic design equations, improvements could be made. The first important design equation is beam stress. The maximum stress for a beam fixed at one end and free but guided at the other end with a concentrated load at the guided end is commonly known.

$$\sigma = \frac{Mc}{I} = \frac{(PL)(h/2)}{bh^3/12} = \frac{6PL}{bh^2}$$ \hspace{1cm} E[3.3]

The spring constant for the same beam is

$$K = \frac{12EI}{L^3} = \frac{12E(h^3/12)}{L^3} = \frac{Ebh^3}{L^3}$$ \hspace{1cm} E[3.4]

As can be seen, a change in some of the variables in one equation would have an adverse effect on the other. As always, the design involved compromise.
Since the spring constant was the driving factor, it weighed more heavily in this design. To increase rigidity yet maintain current measurement resolution, the spoke was increased from 2.0 to 5.8 mm in depth, shortened from 19.0 to 12.7 mm in length, and thinned from 20.6 to 4.7 mm in width. Also, two strain gages, instead of the earlier four, were trimmed and placed as close as possible to the outer ring of the bearing holder. These design changes were expected to increase system rigidity by 1828% and reduce friction force resolution by 48% (see Appendix C for details).

Again, this was not enough. To effectively reduce the appearance of the harmonic, the natural frequency of the system must be increased to at least 6000 Hz -- one hundred times the excitation frequency of 60 Hz. Currently, the system stood at 3124 Hz.

The remaining increase must come from a separate sector of the system -- mass. The natural frequency of a spring mass system is

\[ f_n = \frac{\omega_n}{2\pi} = \frac{\sqrt{K/m}}{2\pi} \quad \text{E}[3.5] \]

If the mass of the system was decreased by a factor of four, then the natural frequency of the system would double. To eliminate excessive system mass, the inner ring of the bearing holder and the bearing case subassembly were redesigned. As a result, system mass was reduced from 244 to 82.6 grams. Consequently, the natural frequency of the system was projected to be at 7716 Hz.

Again, the bearing assembly had to be calibrated. During the early stages of calibration, another problem developed; load resolution was drastically reduced. In fact, there was no discernible difference between a load application of nine and thirty Newtons. When the bearing holder was redesigned, the losses in load resolution were to be offset by a reduction in cross sectional area of the spoke. A load resolution loss of 23% was
expected, but this was much greater. The friction force response, on the other hand, was as expected, no harmonics.

Quick calculations followed. If the spoke depth was reduced from 5.8 to 4.1 mm, system rigidity would not be compromised (natural frequency of system was estimated to be 4585 Hz) and load resolution would increase by 9%. The bearing holder went back into the shop for modifications.

Upon completion, the part was recalibrated and tested — problems remained. Load resolution remained unsatisfactory, and the harmonic was present in the friction force signal. However, both problems were corrected. To eliminate the need for load measurement, special thrust bearings were placed on both sides of the bearing. These bearings assured that the load applied to the bearing holder was equal to the load applied to the test bearing because of their low losses due to friction. With the advent of thrust bearings, load measurement was unnecessary. The inertia springs on the bearing holder were relocated to the test rig spacer ring (The latest version of the bearing holder is shown in Figure 3.12). The harmonic was not eliminated at the design level. Rather, its presence would be reduced (attenuated) from the friction force signal by a low-pass filter. Although preliminary estimates of the natural frequency of the system were found incorrect — \( \omega_n = 3191 \) Hz rather than the previously mentioned 4585 Hz — testing proved successful; this version will be used to evaluate the friction contrast tests.
The bearing holder is just one of the many parts in the test bearing assembly. The inertia springs -- initially a total of five -- provide a means for securing the bearing holder. Otherwise, the test bearing assembly would be free to move with the reciprocating shaft or the motion of the housing.

The spring force required to hold the test piece against the alignment bearing holder is, in the worst case, the sum of the bearing friction force and the inertia force of the test bearing assembly. This force -- determined to be 72.2 N -- must be less than the combined
force of the inertia springs in their compressed state. Calculations are provided in Appendix C.

Initially, the five inertia springs were located in the bearing holder. However, when the bearing holder was revamped to accommodate the thrust bearings, they were relocated to the test rig spacer.

The original inertia springs were 22.0 mm in length and 4.5 mm in diameter. As such, they required a thin deep chamber in which to fit, or they would easily buckle when compressed. The test rig spacer was very thin, only 9.5 mm. If the springs were placed in the spacer, buckling problems might arise. Rather, new springs were ordered and a new spacer was machined.

The new inertia spring assembly (Figure 3.13) was placed between the alignment bearing holder and the spring cassette (refer to Figure 3.4). Although its placement was different, its function remained the same (Detailed drawings are provided in Appendix D).

**CONNECTOR SUBASSEMBLY**

The connector subassembly consists of a 24-pin plug and two socket head cap screws. At one end of the plug, ten of its pins are soldered to shielded wires. These wires are also soldered to the bondable terminals of the strain gages. At the other end, the plug connects with its receptacle. The receptacle is connected to a shielded strand of wires that pass through a housing feedthrough and on to the signal conditioner box. The two screws, as implied, fasten the plug to the bearing holder.
After the redesign of the bearing holder, only four of the pins were required for strain gage interfacing.

![Figure 3.13. Inertia Spring Subassembly](image)

**BEARING CASE SUBASSEMBLY**

The bearing case subassembly consists of a bearing case, test bearing, spacer, inner retaining ring, outer retaining ring and a washer. Originally, the bearing case subassembly was identical to those used in the alignment bearings. This was no accident. To minimize time as well as costs, the original bearing holder was designed to accommodate the bearing
case subassembly; however, when the bearing holder was redesigned for increased rigidity and reduced system mass, the bearing case subassembly had to be redesigned.

The focus of the redesign was the elimination of system mass. As such, the bearing holder was made of polycarbonate and reduced in length. In addition, the washer was eliminated, and the test bearing was reduced in length. Overall, the mass of the bearing case subassembly was reduced from 156.7 to 82.6 grams (see Figure 3.14).

![Diagram of bearing case subassembly](image)

**Figure 3.14.** Bearing Case Subassembly (a) original bearing holder (b) redesigned bearing case subassembly

**THRUSt BEARING SUBASSEMBLY**

When the redesigned bearing holder was tested, load resolution was drastically reduced. In an attempt to boost load resolution by 25%, the bearing holder was placed back in the shop to decrease the thickness of the spokes from 5.8 to 4.1 mm. The load
resolution did increase, but the results were still unacceptable. Furthermore, the system harmonic -- reduced in amplitude -- was back; however, further machining was out of the question.

There were two solutions: eliminate the need for load measurement or increase the load resolution. Load resolution could only be increased by increasing the excitation voltage, the active number of strain gages or signal amplification. The amplification of the amplifier in the signal conditioning box was increased, but the results were unsatisfactory. The other two methods for increasing load resolution involved redesigning the bearing holder. Rather, a method of load transferal was devised; thus, load measurement was unnecessary.

From its initial concept, load transferal meant the use of bearings. The ball thrust bearing seemed like an excellent candidate because of its low coefficient of friction and range of motion. But, there was a drawback. The thrust bearing consisted of three separate pieces. Somehow, the thrust bearing must be installed on the bearing holder and placed within the test rig without separating the pieces or disturbing their alignment.

The rubber retaining band was the answer to all of the problems. It would maintain part alignment and freedom of movement. In addition, its losses due to friction were essentially negligible.

The bearing assembly has six (6) components: two thrust washers, a roller and three retaining bands (Figure 3.15). Essentially, the bearing assembly is a modified INA ball thrust bearing (part # FT01). The only modification was the addition of the three retaining bands.

The bearing assembly was fixed to the bearing holder by epoxying one of the washers to the surface of the machined groove. The remainder of the assembly was then free to deflect by small perturbations when a load was applied. This created a seemingly
frictionless surface. A total of six bearing assemblies were used -- three on each end of the bearing holder.

![Figure 3.15 Bearing Subassembly](image)

**INSTRUMENTATION**

It is not the purpose of this thesis to be concerned with the electronics or electronic theory beyond that which is required to make intelligent use of the equipment for measurement purposes. Rather, the following is a brief discussion directed towards their application.

In order to adequately examine bearing behavior, the bearing assembly was instrumented with two systems. One system measures friction and load whereby the coefficient of friction could be determined. The other -- a thermocouple placed in the bearing case -- measures temperature rise in the test bearing.
FRICITION AND LOAD MEASUREMENT

Friction and load measurements were the results of a four-step process that began with the strain gage. When the strain gage is deformed, it changes resistance. The changes in gage resistance from the imposed strains, although small, but they can be detected by a Wheatstone bridge.

The resistance change that occurs when the strain gage is deformed is very small. As a result, the Wheatstone bridge unbalance and associated voltage change is also very small -- too small to be detected by the data acquisition system. Consequently, the output signal from the bridge circuit has to be amplified to an appropriate level. After amplification, the signal must pass through a filter to improve its signal to noise ratio. (The entire circuit set-up for friction and load measurement is shown in Figure 3.16.)

STRAIN GAGE

The initial step in preparing for friction and load measurement was the selection of an appropriate strain gage to be bonded to the vertical spoke of the bearing holder. This might sound like a simple exercise, but it is not. The installation and operating characteristics of a strain gage are affected by the following parameters: strain-sensitive alloy, backing material, gage length, gage pattern, self-temperature-compensation number, grid resistance, and other available options [16].

Like design, strain gage selection involves compromise. The gage selection process consists of determining which parameters best satisfy one's operating conditions, constraints and environment. Some of the limitations associated with strain gage use are as
Figure 3.16. Instrumentation Required for Friction and Load Force Measurements
of installation and environment. Despite the large number of variables involved, the Vishay Measurements Group reduced the strain gage selection process to only a few basic steps. These steps are: the gage length and pattern, the gage series, the resistance, and the S-T-C number.

The gage length and pattern are normally the first two selections to be made. These are usually based on the space available and the nature of the strain gradient. The space available for mounting the strain gages on the spoke was very limited. In addition, the strain gage was to be placed in the vicinity of a stress concentration — the spoke to outer ring fillet radius. Anticipating problems with installation, a medium gage length of 1.57 mm (0.062 in) was selected.

In selecting the gage pattern, the first consideration was whether a single-gage or rosette was required. Since the directions of the principle axes were known for the bearing holder, a single-grid gage was employed. Also, for ease of installation and limitation of space, a compact, small, general purpose pattern was preferred. In light of these considerations, the 062AP pattern was chosen.

After selecting an initial gage size and pattern, the next step was to select the gage series. The Karma (K) alloy is preferred because of its excellent stability and good fatigue life. In addition, the K alloy can be temperature compensated over a wider range of temperatures. Either the SK or WK Series could be used, but the WK gages were preferred, because they have integral leadwires.

In this instance, two gage patterns in the WK-Series were available from Micro-Measurements; the only difference was their grid resistance -- 120 or 350 ohms. The 350 ohm resistance was preferred, because it has the advantage of decreasing leadwire effects and improving signal-to-noise ratio. In addition, a higher gage resistance reduces heat generation by a factor of three [16].
Finally, the last step was choosing a self-temperature-compensated number to match the structural material on which the gage was to be used. The bearing holder was machined from 6061 aluminum with an appropriate S-T-C number of 13. Combining all the results of the selection procedure, the WK-13-062-350 / Option LE gage was selected.

**WHEATSTONE BRIDGE**

The Wheatstone bridge arrangement has been used for many years to measure an unknown resistance. In this instance, the output voltage from an unbalanced bridge is a function of the amount of friction force or load applied to the bearing.

Originally, the strain gages were to be wired to separate bridge circuits and amplified individually. As a result, friction and load force could be monitored continuously. However, in an attempt to minimize installation costs, only one amplifier was ordered. To account for the discrepancy, a toggle switch was used to rearrange the bridge circuit prior to amplification. The toggle switch has two settings: friction and load. The bridge arrangements for these settings are shown in Figures 3.17.

Unlike most commercial bridges, the bridge circuits described in Figure 3.17 do not employ variable gage resistors to balance the bridge circuit before measurement. Rather, the bridge offset becomes the reference base-line for the force calibration curves.

One of the most serious sources of error in measuring strain with strain gages is thermal output. Once the strain gage is installed, a change in temperature of the gage environment will produce a resistance change in the gage. This temperature induced resistance change is independent of the stress induced strain in the test part. Again, this error source warrants careful consideration. If left unchecked, the error due to thermal output can become much greater than the strain to be measured.
In theory, the error due to thermal output can be completely eliminated by employing two identical "active" strain gages in an adjacent arm of the Wheatstone bridge. This works well for friction measurements (Figure 3.17 [a]); however, the load bridge arrangement offers no such advantages. In fact, the errors due to thermal output in the load arrangement are doubled. To correct for thermally induced resistance changes in the load configuration, load measurements were taken at the very start of the test when the errors due to thermal output are minimal – 0.0 με at 24° C.

Nonlinearity errors in the bridge circuits, if present, were ignored because of the small magnitude of strain being measured.

AMPLIFIER

Since the output voltage level from the Wheatstone bridge is very low, the 2B31J strain gage conditioner and its AC1213 mounting card from Analog Devices were used to
increase the voltage level for subsequent processing. The 2B31J is a high performance, low cost, compact signal conditioning module designed for high accuracy interface to strain gage transducers. The 2B31J consists of three basic sections: instrumentation amplifier, three-pole low pass Bessel filter and adjustable transducer excitation. The AC1213 is an edge connector card with pin receptacles for plugging in the 2B31J. The mounting card has provisions for installing the gain resistors and filter programming components. In addition, the AC1213 has adjustable pots for bridge excitation, input offset, filter offset, fine span and output offset. The 2B31J/AC1213 combination is powered by a 952 Dual Power Supply -- +/- 15 VDC -- from Analog Devices. Specifications on the 2B31J and AC1213 are provided in Appendix C.

Of all the design features, the two most important operations of the 2B31J/AC1213 are the adjustability of gain and bridge excitation voltage. The differential gain, \( G \), of the amplifier was determined according to the equation:

\[
G = \left(1 + \frac{94k\Omega}{R_G}\right)\left[\frac{20k\Omega}{(R_F + 16.2k\Omega)}\right]
\]

Theoretically, the gain is 1166 \((R_G = 99.3\Omega \text{ and } R_F = 57\Omega)\). Experimentally, the gain was 1150; however, the experimental gain was determined by a static test. Since amplifiers are frequency sensitive, a friction or load measurement made under a static condition (calibration curves) will be different for the same measurement given a dynamic condition (60 Hz testing). Fortunately, the gain nonlinearity of the 2B31J signal conditioning module is very low \(-\pm 0.01\%\) maximum; therefore, signal compensation was not required.

Bridge excitation voltage is very important in strain gage performance. As such, the 2B31J employs a bridge voltage adjustment pot. When voltage is applied to a strain gage,
some power is dissipated in the form of heat. The heat generated by the strain gage must be transferred from the strain gage by either conduction or convection. Usually, the heat flow is conducted to the mounting surface. As a result, the sensing grid and substrate operate at a higher temperature than ambient. When the temperature rise is excessive, a loss in self-temperature-compensation and zero (no load) stability is experienced.

To be certain that the excitation level was not excessive, performance tests were run at the maximum environmental temperature. In these tests, the bridge excitation under zero-load conditions was steadily increased until a definite zero instability was observed. Then, the excitation was reduced until it was stable again. The bridge excitation voltage was 2.528V.

LOW-PASS FILTER

Measurements must often be made in the presence of electrical and/or magnetic fields which can superimpose unwanted inputs (electrical "noise") on the measurements. If not controlled, noise can mask the true signal, yielding inaccurate results and incorrect interpretation of the measurements. Through the use of some circuitry, some or all of the noise can be filtered out. To reduce noise bandwidth and aliasing errors, the 2B31J conditioning module provides a three-pole, Bessel-type active filter.

Initially, the cutoff frequency of the three-pole Bessel filter was set to 60 Hz; however, the square wave friction signal was being transformed to a sinusoidal signal during preliminary testing. As a result, it was determined that the friction signal is actually a combination of many frequency components or harmonics (Fourier expansion) that begins
at 60 Hz and continues with each odd harmonic (60, 180, 300, 420, 540, ...). In fact, this discovery led to the redesign of the bearing holder.

After the bearing holder was redesigned, the 2B31J filter was dropped for a more versatile interactive filter program written in LabVIEW by Mahyar Esmaili. The program enables the user to tailor a low pass Bessel filter by specifying the number of poles and cutoff frequency.

In this case, the brunt of the unwanted signal was not the result of electrical or magnetic field disturbances. Rather, the noise was an induced harmonic of the system's natural frequency. The most effective filter setting used to attenuate the unwanted high frequency system response had twenty poles and a cutoff frequency of 1100 Hz. (Figure 3.18 shows the filter response.)

**TEMPERATURE MEASUREMENT**

Temperature rise in the test bearing was monitored by a thermocouple placed at the bearing-to-bearing case interface. During assembly, the thermocouple was located on top of the test bearing and in the middle of the expected contact zone. A housing feedthrough connected the type T thermocouple to an Omega digital thermometer, model CL 23. The digital thermometer was used to record bearing temperature throughout the tests.
Signal retrieval and storage of the friction force measurements were the function of the data acquisition program (One Channel DAQ) written by Esmaili in LabVIEW 2.0.6 for the bearing project. The One Channel DAQ is very versatile; it lets the user tailor the data acquisition system to his/her needs by selecting from a variety of options located on the front panel (Figure 3.19). The front panel options are gain, counts per channel, sample interval, time base, minutes between samples, seconds between samples, DAQ mode and polarity.

Within the program structure is an electronic amplifier or gain. The gain is the amplification ratio (if greater than unity) or attenuation (if less than unity) of the output to input of the data acquisition system. If for some reason the user chooses to change the signal voltage, he/she could do so by changing this selection.
The counts per channel is the number of data points to be sampled at the sampling rate. There are no restrictions on the number of data points; however, large amounts of data do require a significant amount of time for disk storage. If the user is in the semiautomatic mode (described below), he/she must make sure that the minutes and seconds between trials allow the data acquisition system sufficient time for signal retrieval and subsequent storage to disk. Otherwise, a file error will occur and processing will stop.

The sample interval and time base options control the sampling rate (time between data points). The sampling rate is determined by the following equation:

\[
\text{Sampling Rate} = \frac{\text{Time Base}}{\text{Sample Interval}}
\]

where a time base of 1 is 1E6 clock cycles.
The LabSE card allows only two forms of input - single ended (unipolar) or differential (bipolar). As such, the user must differentiate between the two forms, thus, the polarity switch. The polarity should be switched to the unipolar mode if the incoming signals range between 0-10 Volts. If, on the other hand, the incoming signals range from -5 to 5 Volts, the bipolar mode should be selected.

If chosen, the data acquisition system can be placed in a manual (one set at a time), semiautomatic (continuous with prompt) or fully automatic (continuous) mode by switching the DAQ mode lever. As its name implies, the manual mode leaves the timing of the sets at the discretion of the user. The second mode, continuous with prompt, allows the user to select the time reference frame between sets (minutes and seconds between samples). The final DAQ mode option, continuous, continuously cycles - no time between sets - the data acquisition system from signal retrieval to storage. The front panel in Figure 3.19 shows the exact data acquisition settings used throughout the bearing tests.
4.1 RULON J

Rulon J, manufactured by Dixon - Division of Fluorocarbon, is an all-polymeric reinforced PTFE compound with polyimide powder fillers. Rulon J was chosen for the bearing because of its attributes: low friction and wear, self-lubrication and long life. In fact, the low friction coefficients (static and dynamic) of Rulon J makes it ideally suited for start/stop applications. Design criteria for the Rulon J bearing (dry running conditions - ambient temperature) is given in Table 4.1.

The Rulon J bearings were injection molded to the specifications listed below; however, several modifications were made to the bearings prior to use. Like most plastics, the Rulon J compound has a higher coefficient of thermal expansion than most metals. When the temperature of the test rig rose, the bearing, confined axially in the bearing case housing, grew radially. As a result, the ID of the bearing was reduced. This resulted in seizing of the shaft. To accommodate for the thermal expansion, the bearings were bored
out 1/10 mm diametrically. Also, the length of the bearing was machined into 1/4 inch lengths to fit within the confines of the redesigned bearing case.

**Table 4.1. Design Criteria for Rulon J [2]**

<table>
<thead>
<tr>
<th>RECOMMENDED OPERATING LIMITS</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature - Typical Range</td>
<td>-240/+287 °C</td>
</tr>
<tr>
<td>Maximum PV (continuous)</td>
<td>.26 N/mm²*m/s</td>
</tr>
<tr>
<td>Maximum P - (static)</td>
<td>5.17 MPa</td>
</tr>
<tr>
<td>Maximum V - (no load)</td>
<td>2.03 m/s</td>
</tr>
<tr>
<td>Shaft Hardness - Minimum</td>
<td>RB 25</td>
</tr>
<tr>
<td>Recommended Shaft Roughness</td>
<td>.2-.41 μm</td>
</tr>
<tr>
<td>Shaft Material</td>
<td>316 Stainless</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ENGINEERING INFORMATION</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Fiction - Static &amp; Dynamic</td>
<td>.12-.20</td>
</tr>
<tr>
<td>Water Absorption ASTM D570</td>
<td>0%</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>.24 W/m*K</td>
</tr>
<tr>
<td>Thermal Expansion 25.5-149°C</td>
<td>9.36x10⁻⁵ Bearing Diameter</td>
</tr>
<tr>
<td></td>
<td>12.24x10⁻⁵ Bearing Length</td>
</tr>
<tr>
<td>Dimensions</td>
<td>1&quot; ID, 1.250&quot; OD, 1.5&quot; L</td>
</tr>
</tbody>
</table>

4.2 IGLIDE T500

Iglide T500, manufactured by Igus, Inc. of East Providence, Rhode Island, is a composite material with internal fillers and fiber reinforcements. Unfortunately, the exact composition was not disclosed by Igus because of the proprietary nature of the information. However, closer investigation has shown that the Iglide T500 bearing
consists of a PEEK structure, manufactured by ICI, Inc., that is reinforced with carbon fillers and dispersed with PTFE and graphite powder lubricants.

Iglide T500 was chosen because of its high PV limit, excellent speed performance and high operating temperatures. In addition, Iglide T500 has a very low dynamic coefficient of friction against steel -- between 0.11 and 0.17. A list of SP-3 bearing properties is given in Table 4.2.

The bearings were injection molded to the specifications in Table 4.2. However, the length of the bearing was machined into 1/4 inch lengths to fit within the confines of the redesigned bearing case.

4.3. VESPEL SP-3

Vespel SP-3, manufactured by Du Pont, is a polyimide resin with a solid filler. The internal filler, molybdenum disulfide (MoS₂), comprises 15% of the weight of the bearing. The addition of MoS₂ provides lubrication for seals and bearings in vacuum or dry environments.

Vespel SP-3 is Du Pont's latest development in its family of polyimide bearings. As a result, bearing performance could not be accessed from the literature, and PV limits could not be obtained from the manufacturer. Vespel SP-3 was chosen because of its high operating temperatures, low coefficient of friction and unlubricated performance. Furthermore, its recent appearance made it an interesting candidate material. A summary of the typical properties for SP-3 is provided in Table 4.3.

The bearing test samples were not injection molded to the specifications below. Rather, they were machined from roundstock by the manufacturer to the specifications
listed in Table 4.3. In addition, the length of the bearing was machined into 1/4 inch lengths to fit within the confines of the redesigned bearing case.

Table 4.2. Iglide T500 Properties [13]

<table>
<thead>
<tr>
<th>Category</th>
<th>Property</th>
</tr>
</thead>
<tbody>
<tr>
<td>GENERAL</td>
<td></td>
</tr>
<tr>
<td>Specific Gravity</td>
<td>1.48</td>
</tr>
<tr>
<td>Coefficient of Friction,</td>
<td></td>
</tr>
<tr>
<td>Dynamic, against steel</td>
<td>.11-.17</td>
</tr>
<tr>
<td>PV Limit (no lubrication)</td>
<td>1.8 N/mm²·m/s</td>
</tr>
<tr>
<td>Shaft Material</td>
<td>Steel, hardened steel, or carbon steel</td>
</tr>
<tr>
<td>Recommended Shaft Roughness</td>
<td>.46-.81 μm</td>
</tr>
<tr>
<td>Shaft Hardness - Minimum</td>
<td>RC 50</td>
</tr>
<tr>
<td>MECHANICAL</td>
<td></td>
</tr>
<tr>
<td>Tensile Strength</td>
<td>118 N/mm²</td>
</tr>
<tr>
<td>Flexural Strength</td>
<td>230 N/mm²</td>
</tr>
<tr>
<td>Flexural Modulus</td>
<td>6600 N/mm²</td>
</tr>
<tr>
<td>Izod Impact Strength, notched</td>
<td>86 J/m</td>
</tr>
<tr>
<td>Unit Pressure Limit</td>
<td>150 N/mm²</td>
</tr>
<tr>
<td>Hardness, Rockwell</td>
<td>R125</td>
</tr>
<tr>
<td>PHYSICAL &amp; THERMAL</td>
<td></td>
</tr>
<tr>
<td>End Use Temperature, long term</td>
<td>-100/+250 °C</td>
</tr>
<tr>
<td>End Use Temperature, short term</td>
<td>+315 °C</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>.6 W/m²·K</td>
</tr>
<tr>
<td>Water Absorption at 50% humidity</td>
<td>0.1 % (72°F)</td>
</tr>
<tr>
<td>Dimensions</td>
<td>1&quot; ID, 1.125&quot; OD, 1.5&quot; L</td>
</tr>
</tbody>
</table>
Table 4.3. Summary of Typical Properties for SP-3 [14]

**MECHANICAL (22.7°C)**
- Tensile Strength: 58.5 MPa
- Elongation: 4.0%
- FlexuralStrength: 75.8 MPa
- Flexural Modulus: 3275 MPa
- Izod Impact Strength, notched: 21.3 J/m

**WEAR AND FRICTION**
- Wear (unlubricated in air @ PV=0.875 MPa m/s): 17-23 m/s x 10^-10
- Friction Coefficient (unlubricated in air @ PV=0.875 MPa m/s): .25
  - PV=3.5 MPa m/s: .17

**THERMAL**
- Thermal Conductivity: .47 W/m°C

**OTHER PROPERTIES**
- Water Absorption: 0.23 %
  - 24 hrs @ 22.7°C
  - 48 hrs @ 49.9°C: 0.65 %
- Specific Gravity: 1.60
- Hardness, Rockwell: RE 40-55
- Dimensions: 1" ID, 1.125" OD, 1.5" L

4.4. COUNTERFACE

Experimentally, data has shown that the shaft hardness and finish of the mating surface effect the wear rates and friction coefficients of polymer bearings. In the contrast tests, the polymer bearings were mated against a 52100 bearing steel of tolerance class "S" and surface hardness of 60-65 Rockwell C. The friction behavior of the candidate
materials, on the other hand, was tested at two metal roughness values -- 0.05-0.08 μm Ra (0.25-0.35 μm Rtm) and 0.15-0.18 μm Ra (0.90-1.20 μm Rtm). To achieve the desired surface roughness, the steel shaft was sanded perpendicular to the sliding direction with 1500 and 220 grade sandpaper. All surface roughness measurements were made by a Surtronic 3P profilometer with a 112/1503 pick up and a stylus tip radius of 10 μm.
5.1 TEST MATRIX

To evaluate the influences of load, speed, and counterface roughness on the friction coefficients of the candidate bearing materials, a contrast test matrix (Table 5.1) was structured in a one-factor-at-a-time manipulation. A total of eight tests -- three factors (load, speed and counterface roughness) at two levels of each (high and low) -- were performed for each bearing material. All tests were conducted at 60 Hz.

Table 5.1. Contrast Test Matrix

1. 1111  5. 2111
2. 1112  6. 2112
3. 1121  7. 2121
4. 1122  8. 2122
Each test listed in Table 5.1 consists of an identity code which identifies the test conditions. Each factor was identified by a sequence of numbers. The order of the sequence was speed, atmosphere, load and counterface roughness. The magnitude of the number, on the other hand, reflects the level of the individual factor. Factor levels are given in Table 5.2. For example, an identity number of 2121 is characterized by the following: velocity = 0.48 m/s (2 mm amplitude), atmosphere = helium, load = 112.3 N, and a surface roughness = 0.05-0.08 μm Ra.

Table 5.2. Identification of Factor Levels

<table>
<thead>
<tr>
<th>Factors</th>
<th>Level 1</th>
<th>Level 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>0.24 m/s</td>
<td>0.48 m/s</td>
</tr>
<tr>
<td>Atmosphere</td>
<td>dry helium @ 400 KPa</td>
<td>Air @ 400 KPa</td>
</tr>
<tr>
<td>Load</td>
<td>82.7 N</td>
<td>112.3 N</td>
</tr>
<tr>
<td>Counterface Roughness</td>
<td>Ra = 0.05-0.08 μm</td>
<td>Ra = 0.15-0.18 μm</td>
</tr>
<tr>
<td></td>
<td>Rtm = 0.25-0.35 μm</td>
<td>Rtm = 0.90-1.20 μm</td>
</tr>
</tbody>
</table>

5.2. PREPARATION PROCEDURES

In order to ensure reproducibility of the results, a standard test method was used. First and foremost were the cleaning procedures. Prior to each test, the test rig and all internal components (includes test bearing) were cleaned with alcohol and a cloth to remove any debris. Then, the shaft was sanded to the desired surface roughness (removal of transfer layers, surface scale or oxides) and cleaned with alcohol and a cloth; this was followed by thoroughly wetting the test bearing surface with alcohol and drying it with a
gauze sponge. Finally, the shaft was scrubbed by cottonballs wetted with alcohol until no discoloration was evident.

After the cleaning procedures were completed, the test rig was assembled, and the shaft/spring cartridge cassette was aligned. The alignment process continued until a suitable friction force signal for three millimeters of amplitude was obtained.

Once aligned, the outlet valve and FLDT (Fast Linear Displacement Transducer) cap were installed and purging began. The purging process took seventy-five minutes. For the first fifteen minutes, a continuous flow of helium (composition listed in Table 5.3) was directed through the test rig while it was operating at one millimeter of amplitude. Then, the outlet valve was closed and the helium pressure was raised to 400 KPa. Afterwards, the rig was purged for two minutes every fifteen minutes for the next hour. Testing followed.

In addition to purging, moisture absorption by the test rig and test bearings were reduced by the use of desiccant bags. Furthermore, all test bearings were stored until use in a chamber pressurized with dry helium.

Table 5.3. Composition of Helium [17]

<table>
<thead>
<tr>
<th>Gas</th>
<th>Composition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Helium</td>
<td>&gt; 99.995 %</td>
</tr>
<tr>
<td>Oxygen</td>
<td>&lt; 5 ppm</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>&lt; 25 ppm</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>&lt; 5 ppm</td>
</tr>
<tr>
<td>Methane</td>
<td>&lt; 5 ppm</td>
</tr>
<tr>
<td>Humidity</td>
<td>&lt; 34 ppm</td>
</tr>
</tbody>
</table>
5.3. CONTRAST TESTS

The friction measurement system was used to measure the friction force applied to the test bearing. Originally, the friction coefficient $\mu$ was to be determined according to Amontons-Coulomb as follows:

$$\mu = \frac{1}{n} \sum_{1}^{n} [F_f]$$  \hspace{1cm} E[5.1]

where $F_f$ is the friction force, $F_n$ is the force applied to the bearing, and the friction coefficient is calculated by measuring 2000 times ($n=2000$) $F_f$ during three periods of movement. However, equation E[5.1] was not used in this manner because of the complexity created by the introduction of the low-pass filter. While the low-pass filter attenuated "noise", it also degraded any higher order frequencies which defined the edges of the friction response. Thus, the evaluation of E[5.1] over three entire periods of movement would have resulted in lower friction coefficients. Rather, the friction coefficients were evaluated by measuring 100 points at the center of each high/low response (see Figure 5.1).

The results of the 60 Hz contrast tests are presented in Tables 5.5 - 5.7. In addition, the tabular data is displayed graphically in two separate sections. The first section, Figures 5.2 - 5.45, show the trends of the friction coefficient as a function of test duration. The second section, Figures B.1 - B.44, present the trends of the friction coefficient as a function of test bearing temperature. Each section is broken into four subsections which illustrate the variations in the coefficient of friction with load, speed and counterface roughness for all three polymers. A list of figure associations is provided in Table 5.4.
The variations in the friction coefficients during two cycles of measurement are presented in Appendix A.

Figure 5.1. Calculation of Friction Coefficient

5.4. LONG TERM TESTS

After the contrast test matrix was completed, questions arose as to the predictability of the friction coefficients. After as much as 105 minutes, many of the friction coefficients tended to increase or decrease in a somewhat unpredictable pattern (very few "steady states" were apparent). To provide insight, two materials -- Rulon J and Iglide T500 -- were selected to undergo long-term testing at a single matrix condition -- 1122. The test results are tabulated in Table 5.8 and presented in Figures 5.46 and B.45.
Table 5.4. Figure Associations

Coefficient of Friction as a Function of Time

<table>
<thead>
<tr>
<th>Figure</th>
<th>Material</th>
<th>Factor</th>
<th>Matrix Comparisons</th>
<th>Figure</th>
<th>Material</th>
<th>Factor</th>
<th>Matrix Comparisons</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.2</td>
<td>Iglide</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>5.24</td>
<td>Vespel</td>
<td>Speed</td>
<td>1121 versus 2121</td>
</tr>
<tr>
<td>5.3</td>
<td>Iglide</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>5.25</td>
<td>Vespel</td>
<td>Speed</td>
<td>1122 versus 2122</td>
</tr>
<tr>
<td>5.4</td>
<td>Iglide</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>5.26</td>
<td>Iglide</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>5.5</td>
<td>Iglide</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>5.27</td>
<td>Iglide</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>5.6</td>
<td>Rulon</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>5.28</td>
<td>Iglide</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>5.7</td>
<td>Rulon</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>5.29</td>
<td>Iglide</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>5.8</td>
<td>Rulon</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>5.30</td>
<td>Rulon</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>5.9</td>
<td>Rulon</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>5.31</td>
<td>Rulon</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>5.10</td>
<td>Vespel</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>5.32</td>
<td>Rulon</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>5.11</td>
<td>Vespel</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>5.33</td>
<td>Rulon</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>5.12</td>
<td>Vespel</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>5.34</td>
<td>Vespel</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>5.13</td>
<td>Vespel</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>5.35</td>
<td>Vespel</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>5.14</td>
<td>Iglide</td>
<td>Speed</td>
<td>1111 versus 2111</td>
<td>5.36</td>
<td>Vespel</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>5.15</td>
<td>Iglide</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>5.37</td>
<td>Vespel</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>5.16</td>
<td>Iglide</td>
<td>Speed</td>
<td>1121 versus 2121</td>
<td>5.38</td>
<td>All</td>
<td>-</td>
<td>1111</td>
</tr>
<tr>
<td>5.17</td>
<td>Iglide</td>
<td>Speed</td>
<td>1122 versus 2122</td>
<td>5.39</td>
<td>All</td>
<td>-</td>
<td>1112</td>
</tr>
<tr>
<td>5.18</td>
<td>Rulon</td>
<td>Speed</td>
<td>1111 versus 2111</td>
<td>5.40</td>
<td>All</td>
<td>-</td>
<td>1121</td>
</tr>
<tr>
<td>5.19</td>
<td>Rulon</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>5.41</td>
<td>All</td>
<td>-</td>
<td>1122</td>
</tr>
<tr>
<td>5.20</td>
<td>Rulon</td>
<td>Speed</td>
<td>1121 versus 2121</td>
<td>5.42</td>
<td>All</td>
<td>-</td>
<td>2111</td>
</tr>
<tr>
<td>5.21</td>
<td>Rulon</td>
<td>Speed</td>
<td>1122 versus 2122</td>
<td>5.43</td>
<td>All</td>
<td>-</td>
<td>2121</td>
</tr>
<tr>
<td>5.22</td>
<td>Vespel</td>
<td>Speed</td>
<td>1111 versus 2111</td>
<td>5.44</td>
<td>All</td>
<td>-</td>
<td>2121</td>
</tr>
<tr>
<td>5.23</td>
<td>Vespel</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>5.45</td>
<td>All</td>
<td>-</td>
<td>2122</td>
</tr>
</tbody>
</table>

Coefficient of Friction as a Function of Temperature

<table>
<thead>
<tr>
<th>Figure</th>
<th>Material</th>
<th>Factor</th>
<th>Matrix Comparisons</th>
<th>Figure</th>
<th>Material</th>
<th>Factor</th>
<th>Matrix Comparisons</th>
</tr>
</thead>
<tbody>
<tr>
<td>B.1</td>
<td>Iglide</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>B.23</td>
<td>Vespel</td>
<td>Speed</td>
<td>1121 versus 2121</td>
</tr>
<tr>
<td>B.2</td>
<td>Iglide</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>B.24</td>
<td>Vespel</td>
<td>Speed</td>
<td>1122 versus 2122</td>
</tr>
<tr>
<td>B.3</td>
<td>Iglide</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>B.25</td>
<td>Iglide</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>B.4</td>
<td>Iglide</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>B.26</td>
<td>Iglide</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>B.5</td>
<td>Rulon</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>B.27</td>
<td>Iglide</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>B.6</td>
<td>Rulon</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>B.28</td>
<td>Iglide</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>B.7</td>
<td>Rulon</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>B.29</td>
<td>Rulon</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>B.8</td>
<td>Rulon</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>B.30</td>
<td>Rulon</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>B.9</td>
<td>Vespel</td>
<td>Load</td>
<td>1111 versus 1121</td>
<td>B.31</td>
<td>Rulon</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>B.10</td>
<td>Vespel</td>
<td>Load</td>
<td>1112 versus 1122</td>
<td>B.32</td>
<td>Rulon</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>B.11</td>
<td>Vespel</td>
<td>Load</td>
<td>2111 versus 2121</td>
<td>B.33</td>
<td>Vespel</td>
<td>C. R.</td>
<td>1111 versus 1112</td>
</tr>
<tr>
<td>B.12</td>
<td>Vespel</td>
<td>Load</td>
<td>2112 versus 2122</td>
<td>B.34</td>
<td>Vespel</td>
<td>C. R.</td>
<td>1121 versus 1122</td>
</tr>
<tr>
<td>B.13</td>
<td>Iglide</td>
<td>Speed</td>
<td>1111 versus 1111</td>
<td>B.35</td>
<td>Vespel</td>
<td>C. R.</td>
<td>2111 versus 2112</td>
</tr>
<tr>
<td>B.14</td>
<td>Iglide</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>B.36</td>
<td>Vespel</td>
<td>C. R.</td>
<td>2121 versus 2122</td>
</tr>
<tr>
<td>B.15</td>
<td>Iglide</td>
<td>Speed</td>
<td>1121 versus 2121</td>
<td>B.37</td>
<td>All</td>
<td>-</td>
<td>1111</td>
</tr>
<tr>
<td>B.16</td>
<td>Iglide</td>
<td>Speed</td>
<td>1122 versus 2122</td>
<td>B.38</td>
<td>All</td>
<td>-</td>
<td>1112</td>
</tr>
<tr>
<td>B.17</td>
<td>Rulon</td>
<td>Speed</td>
<td>1111 versus 2111</td>
<td>B.39</td>
<td>All</td>
<td>-</td>
<td>1121</td>
</tr>
<tr>
<td>B.18</td>
<td>Rulon</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>B.40</td>
<td>All</td>
<td>-</td>
<td>1122</td>
</tr>
<tr>
<td>B.19</td>
<td>Rulon</td>
<td>Speed</td>
<td>1121 versus 2121</td>
<td>B.41</td>
<td>All</td>
<td>-</td>
<td>2111</td>
</tr>
<tr>
<td>B.20</td>
<td>Rulon</td>
<td>Speed</td>
<td>1122 versus 2122</td>
<td>B.42</td>
<td>All</td>
<td>-</td>
<td>2121</td>
</tr>
<tr>
<td>B.21</td>
<td>Vespel</td>
<td>Speed</td>
<td>1111 versus 2111</td>
<td>B.43</td>
<td>All</td>
<td>-</td>
<td>2121</td>
</tr>
<tr>
<td>B.22</td>
<td>Vespel</td>
<td>Speed</td>
<td>1112 versus 2112</td>
<td>B.44</td>
<td>All</td>
<td>-</td>
<td>2122</td>
</tr>
</tbody>
</table>
Table 5.5. Contrast Test Data for Iglide T500

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>1111</th>
<th>1112</th>
<th>1121</th>
<th>1122</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>μ</td>
<td>Temp (°C)</td>
<td>μ</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>0</td>
<td>0.170</td>
<td>23.4</td>
<td>0.178</td>
<td>23.3</td>
</tr>
<tr>
<td>15</td>
<td>0.172</td>
<td>29.5</td>
<td>0.150</td>
<td>29.6</td>
</tr>
<tr>
<td>30</td>
<td>0.172</td>
<td>30.2</td>
<td>0.156</td>
<td>30.2</td>
</tr>
<tr>
<td>45</td>
<td>0.170</td>
<td>30.7</td>
<td>0.153</td>
<td>30.7</td>
</tr>
<tr>
<td>60</td>
<td>0.169</td>
<td>31.1</td>
<td>0.149</td>
<td>31.0</td>
</tr>
<tr>
<td>75</td>
<td>0.166</td>
<td>31.2</td>
<td>0.147</td>
<td>31.3</td>
</tr>
<tr>
<td>90</td>
<td>0.167</td>
<td>31.4</td>
<td>0.146</td>
<td>31.4</td>
</tr>
<tr>
<td>105</td>
<td>0.161</td>
<td>31.5</td>
<td>0.142</td>
<td>31.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>2111</th>
<th>2112</th>
<th>2121</th>
<th>2122</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>μ</td>
<td>Temp (°C)</td>
<td>μ</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>0</td>
<td>0.167</td>
<td>26.3</td>
<td>0.272</td>
<td>22.3</td>
</tr>
<tr>
<td>15</td>
<td>0.147</td>
<td>36.6</td>
<td>0.223</td>
<td>32.1</td>
</tr>
<tr>
<td>30</td>
<td>0.125</td>
<td>38.1</td>
<td>0.194</td>
<td>32.1</td>
</tr>
<tr>
<td>45</td>
<td>0.129</td>
<td>38.7</td>
<td>0.198</td>
<td>32.8</td>
</tr>
<tr>
<td>60</td>
<td>0.115</td>
<td>39.1</td>
<td>0.192</td>
<td>33.5</td>
</tr>
<tr>
<td>75</td>
<td>0.113</td>
<td>39.4</td>
<td>0.171</td>
<td>33.8</td>
</tr>
<tr>
<td>90</td>
<td>0.118</td>
<td>39.8</td>
<td>0.159</td>
<td>34.4</td>
</tr>
<tr>
<td>105</td>
<td>0.125</td>
<td>39.9</td>
<td>0.159</td>
<td>34.8</td>
</tr>
</tbody>
</table>
Table 5.6. Contrast Test Data for Rulon J

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>1111 μ</th>
<th>Temp (°C)</th>
<th>1112 μ</th>
<th>Temp (°C)</th>
<th>1121 μ</th>
<th>Temp (°C)</th>
<th>1122 μ</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.149</td>
<td>25.3</td>
<td>0.144</td>
<td>23.8</td>
<td>0.181</td>
<td>23.5</td>
<td>0.182</td>
<td>23.2</td>
</tr>
<tr>
<td>15</td>
<td>0.174</td>
<td>29.2</td>
<td>0.134</td>
<td>27.6</td>
<td>0.210</td>
<td>29.1</td>
<td>0.182</td>
<td>27.6</td>
</tr>
<tr>
<td>30</td>
<td>0.162</td>
<td>30.4</td>
<td>0.135</td>
<td>28.4</td>
<td>0.200</td>
<td>30.3</td>
<td>0.177</td>
<td>29.1</td>
</tr>
<tr>
<td>45</td>
<td>0.177</td>
<td>30.9</td>
<td>0.139</td>
<td>28.7</td>
<td>0.191</td>
<td>30.8</td>
<td>0.181</td>
<td>29.5</td>
</tr>
<tr>
<td>60</td>
<td>0.193</td>
<td>31.3</td>
<td>0.142</td>
<td>29.0</td>
<td>0.185</td>
<td>31.2</td>
<td>0.182</td>
<td>29.9</td>
</tr>
<tr>
<td>75</td>
<td>0.212</td>
<td>31.9</td>
<td>0.140</td>
<td>29.3</td>
<td>0.182</td>
<td>31.2</td>
<td>0.180</td>
<td>30.2</td>
</tr>
<tr>
<td>90</td>
<td>0.211</td>
<td>32.2</td>
<td>0.143</td>
<td>29.6</td>
<td>0.181</td>
<td>31.8</td>
<td>0.181</td>
<td>30.3</td>
</tr>
<tr>
<td>105</td>
<td>0.216</td>
<td>32.6</td>
<td>0.146</td>
<td>29.9</td>
<td>0.178</td>
<td>32.0</td>
<td>0.179</td>
<td>30.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>2111 μ</th>
<th>Temp (°C)</th>
<th>2112 μ</th>
<th>Temp (°C)</th>
<th>2121 μ</th>
<th>Temp (°C)</th>
<th>2122 μ</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.140</td>
<td>24.4</td>
<td>0.221</td>
<td>24.6</td>
<td>0.201</td>
<td>22.9</td>
<td>0.126</td>
<td>21.5</td>
</tr>
<tr>
<td>15</td>
<td>0.140</td>
<td>25.1</td>
<td>0.229</td>
<td>34.6</td>
<td>0.227</td>
<td>35.3</td>
<td>0.128</td>
<td>31.9</td>
</tr>
<tr>
<td>30</td>
<td>0.145</td>
<td>25.6</td>
<td>0.227</td>
<td>36.4</td>
<td>0.256</td>
<td>37.8</td>
<td>0.131</td>
<td>35.1</td>
</tr>
<tr>
<td>45</td>
<td>0.150</td>
<td>26.0</td>
<td>0.224</td>
<td>37.1</td>
<td>0.257</td>
<td>39.2</td>
<td>0.127</td>
<td>36.3</td>
</tr>
<tr>
<td>60</td>
<td>0.158</td>
<td>26.4</td>
<td>0.227</td>
<td>37.8</td>
<td>0.273</td>
<td>40.5</td>
<td>0.129</td>
<td>37.1</td>
</tr>
<tr>
<td>75</td>
<td>0.155</td>
<td>26.8</td>
<td>0.222</td>
<td>38.2</td>
<td>0.274</td>
<td>41.7</td>
<td>0.130</td>
<td>38.1</td>
</tr>
<tr>
<td>90</td>
<td>0.154</td>
<td>27.2</td>
<td>0.219</td>
<td>38.8</td>
<td>0.281</td>
<td>43.1</td>
<td>0.131</td>
<td>38.9</td>
</tr>
<tr>
<td>105</td>
<td>0.170</td>
<td>27.6</td>
<td>0.224</td>
<td>39.3</td>
<td>0.301</td>
<td>43.8</td>
<td>0.129</td>
<td>40.0</td>
</tr>
</tbody>
</table>
Table 5.7. Contrast Test Data for Vespel SP-3

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>1111 µ Temp (°C)</th>
<th>1112 µ Temp (°C)</th>
<th>1121 µ Temp (°C)</th>
<th>1122 µ Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.318 26.8</td>
<td>0.253 26.5</td>
<td>0.269 25.8</td>
<td>0.397 26.6</td>
</tr>
<tr>
<td>15</td>
<td>0.356 39.4</td>
<td>0.283 34.6</td>
<td>0.274 37.7</td>
<td>0.367 40.6</td>
</tr>
<tr>
<td>30</td>
<td>0.363 42.2</td>
<td>0.297 36.1</td>
<td>0.340 40.6</td>
<td>0.387 42.9</td>
</tr>
<tr>
<td>45</td>
<td>0.366 43.6</td>
<td>0.306 36.7</td>
<td>0.407 42.1</td>
<td>0.389 43.8</td>
</tr>
<tr>
<td>60</td>
<td>0.349 44.7</td>
<td>0.306 37.3</td>
<td>0.452 43.0</td>
<td>0.398 44.4</td>
</tr>
<tr>
<td>75</td>
<td>0.357 45.6</td>
<td>0.311 37.6</td>
<td>0.476 45.3</td>
<td>0.394 44.9</td>
</tr>
<tr>
<td>90</td>
<td>0.357 46.6</td>
<td>0.320 38.0</td>
<td>0.473 46.7</td>
<td>0.399 45.4</td>
</tr>
<tr>
<td>105</td>
<td>0.337 47.6</td>
<td>0.321 38.4</td>
<td>0.464 47.4</td>
<td>0.412 45.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Time (min)</th>
<th>2111 µ Temp (°C)</th>
<th>2112 µ Temp (°C)</th>
<th>2121 µ Temp (°C)</th>
<th>2122 µ Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.353 23.6</td>
<td>0.225 26.3</td>
<td>0.258 23.8</td>
<td>0.237 24.2</td>
</tr>
<tr>
<td>15</td>
<td>0.313 50.1</td>
<td>0.239 40.6</td>
<td>0.275 44.7</td>
<td>0.269 43.2</td>
</tr>
<tr>
<td>30</td>
<td>0.327 54.7</td>
<td>0.254 44.7</td>
<td>0.364 51.8</td>
<td>0.323 47.6</td>
</tr>
<tr>
<td>45</td>
<td>0.334 57.4</td>
<td>0.261 45.8</td>
<td>0.378 56.4</td>
<td>0.364 51.3</td>
</tr>
<tr>
<td>60</td>
<td>0.344 60.0</td>
<td>0.275 46.7</td>
<td>0.443 60.8</td>
<td>0.385 53.2</td>
</tr>
<tr>
<td>75</td>
<td>0.329 62.2</td>
<td>0.277 47.4</td>
<td>0.500 65.7</td>
<td>0.378 54.7</td>
</tr>
<tr>
<td>90</td>
<td>0.330 64.8</td>
<td>0.289 48.3</td>
<td>0.488 67.4</td>
<td>0.389 55.7</td>
</tr>
<tr>
<td>105</td>
<td>0.330 66.4</td>
<td>0.298 49.2</td>
<td>0.470 69.9</td>
<td>0.398 56.8</td>
</tr>
</tbody>
</table>
Table 5.8. Long-Term Test Data for Rulon J and Iglide T500

<table>
<thead>
<tr>
<th>Time (hours)</th>
<th>Iglide T500 - 1122 μ</th>
<th>Temp (°C)</th>
<th>Time (hours)</th>
<th>Rulon J - 1122 μ</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-</td>
<td>26.4</td>
<td>0</td>
<td>0.215</td>
<td>24.4</td>
</tr>
<tr>
<td>3</td>
<td>0.217</td>
<td>36.9</td>
<td>3</td>
<td>0.179</td>
<td>39.3</td>
</tr>
<tr>
<td>6</td>
<td>0.209</td>
<td>39.3</td>
<td>6</td>
<td>0.186</td>
<td>41.6</td>
</tr>
<tr>
<td>9</td>
<td>0.188</td>
<td>41.5</td>
<td>9</td>
<td>0.204</td>
<td>41.3</td>
</tr>
<tr>
<td>12</td>
<td>0.193</td>
<td>41.9</td>
<td>12</td>
<td>0.210</td>
<td>40.3</td>
</tr>
<tr>
<td>15</td>
<td>0.198</td>
<td>41.8</td>
<td>15</td>
<td>0.215</td>
<td>40.2</td>
</tr>
<tr>
<td>18</td>
<td>0.202</td>
<td>41.9</td>
<td>18</td>
<td>0.213</td>
<td>41.7</td>
</tr>
<tr>
<td>21</td>
<td>0.191</td>
<td>42.1</td>
<td>21</td>
<td>0.212</td>
<td>41.7</td>
</tr>
<tr>
<td>24</td>
<td>0.199</td>
<td>42.2</td>
<td>28.8</td>
<td>0.219</td>
<td>41.7</td>
</tr>
</tbody>
</table>
Figure 5.2. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.3. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.4. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.5. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.6. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - $Ra = 0.05-0.08 \mu m$ and $Rtm = 0.25-0.35 \mu m$

Figure 5.7. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - $Ra = 0.15-0.18 \mu m$ and $Rtm = 0.9-1.20 \mu m$
Figure 5.8. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.9. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.10. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.11. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.12. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - $Ra = 0.05-0.08 \mu m$ and $Rtm = 0.25-0.35 \mu m$.

Figure 5.13. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - $Ra = 0.15-0.18 \mu m$ and $Rtm = 0.9-1.20 \mu m$. 
Figure 5.14. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.15. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.16. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm

Figure 5.17. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm
Figure 5.18. Influence of Time on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - $R_a = 0.05-0.08 \mu m$ and $R_{tm} = 0.25-0.35 \mu m$

Figure 5.19. Influence of Time on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - $R_a = 0.15-0.18 \mu m$ and $R_{tm} = 0.9-1.20 \mu m$
Figure 5.20. Influence of Time on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.21. Influence of Time on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.22. Influence of Time on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.23. Influence of Time on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.24. Influence of Time on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm.

Figure 5.25. Influence of Time on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm.
Figure 5.26. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure 5.27. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure 5.28. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure 5.29. Influence of Time on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure 5.30. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure 5.31. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure 5.32. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure 5.33. Influence of Time on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure 5.34. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure 5.35. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure 5.36. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium.

Figure 5.37. Influence of Time on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium.
Figure 5.38. Influence of Time on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.39. Influence of Time on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.40. Influence of Time on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.41. Influence of Time on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Results

Figure 5.42. Influence of Time on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure 5.43. Influence of Time on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure 5.44. Influence of Time on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm.

Figure 5.45. Influence of Time on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm.
Figure 5.46. Influence of Time on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.90-1.20 µm
6.1 DISCUSSION OF TEST RESULTS

RULON J

Load, speed and counterface roughness had a significant influence on the reciprocating friction coefficient of Rulon J. Although variations in the friction coefficient were present as the result of load and speed, no clear trends were detected. This was surprising in that Dixon observed a rapid decrease in the friction coefficient with increased load [2]. Moreover, Dickens et al. [3] reported that friction increased continuously with speed for PTFE. This, however, does not substantiate a contradiction in results. According to Dr. Lancaster [4], the friction coefficient of polymers may increase, decrease or remain independent of load or speed if frictional heating is significant. As mentioned earlier, the
test bearing temperature was only monitored, not controlled; therefore, changes in the coefficient of friction may be the result of variations in temperature. The friction coefficient, on the other hand, clearly decreased with an increase in the roughness of the shaft. Identical trends have been reported by Santer and Czichos [5]. The explanation for this behavior in the literature [6,7] is that, for smooth surfaces, adhesion forces dominate, whereas for rough surfaces, abrasive processes prevail. For each material there exits an optimum surface roughness for which the effects of adhesion and abrasion are minimized. The highest surface roughness used in this work, $Ra = 0.15-0.18 \mu m$, coincides with the value suggested by the manufacturer.

Although friction coefficients were measured, comparison of the results in this work to the findings of other investigators is fraught with difficulties because of the complex factors affecting polymer friction. As mentioned earlier, variations in the test factors (summarized in Figure 2) will result in a change in the friction coefficient. As a result, correlations should be made under similar conditions.

The coefficient of friction for Rulon J ranged from 0.126 to 0.301. For test 2121, the coefficient of friction was found to be 0.201 at start-up. This result compares favorably to results published by other investigators. For example, Bhushan and Wilcox reported a friction coefficient of 0.23 [1]. However, the friction coefficient in their test was measured unidirectionally with a pin against cylinder apparatus subjected to slightly different test conditions. For instance, the average sliding speed was much higher -- 1.2 m/s -- and the applied force was much lower -- 12 N. On the other hand, the mating material and surface roughness were very similar to those used in test 2121. Dixon, the bearing manufacturer, found the friction coefficient to be 0.20 [2]. Unfortunately, no experimental specifics were given.
IGLIDE T500

Variations in the friction coefficient of Iglide T500 with load, speed and counterface roughness were measured. It can be seen from the results that the coefficient of friction decreased with increases in speed and counterface roughness. In addition, no clear trends of $\mu$ as a function of load could be detected.

The influences of speed and load in this work were rather suprising. Under reciprocating sliding, Schelling et al. [8] reported that pure PEEK and polyetheretherketone composite D450HF30 increased with sliding velocity. Furthermore, Briscoe et al. [9] observed a decrease in $\mu$ with increased load. Again, a possible explanation for the differences is temperature. If the dominate factor for the coefficient of friction is temperature, as suggested by Dr. Lancaster [10,11], then dissimilarities among test factors may have resulted in different interface temperatures. In the present work, $\mu$ decreased with increased temperature. Similar results are reported by [9,12].

The coefficient of friction for Iglide T500 ranged from 0.272 to 0.036; the majority of the measurements fell between 0.17 and 0.11. Ingus Incorporated, the bearing manufacturer, reports a dynamic coefficient of friction of 0.11 to 0.17 [13]. Identical results were obtained by Friedrich et al. [12].

VESPEL SP-3

Like Rulon J and Iglide T500, the coefficient of friction for Vespel SP-3 changed with variations in load, speed and counterface roughness. It can be seen from the results that the coefficient of friction decreased with increases in speed and roughness. Furthermore, increased load resulted in higher friction coefficients.
Unfortunately, no experimental data was found in the literature that describes the effects of load, speed or counterface roughness on the frictional behavior of Vespel SP-3. The only documentation available was a properties list provided by Du Pont describing the coefficient of friction for two PV (Pressure-Velocity) values. For PV values of 0.875 and 3.5 MPa m/s, the friction coefficients were 0.25 and .17 respectively [14]. In this work, the coefficient of friction varied from 0.500 to 0.225.

6.2 CONCLUSIONS

The effects of load, speed and counterface roughness on the frictional behavior of the three bearing materials -- Rulon J, Iglide T500, and Vespel SP-3 -- were measured and compared under identical conditions. Each material exhibited conditions of optimum performance; however, Iglide T500 was the most efficient (low frictional losses) -- test 2122, see Table 5.4 for figure associations.

Again, the intent of this work lied in the design of a reciprocating friction force measurement system for measuring bearing performance. It is the responsibility of the tribologist, on the other hand, to interpret the frictional behavior of the contact bearings.
6.3 DISCUSSION OF MEASUREMENT SYSTEM

The performance of the reciprocating friction force measurement system met the design requirements, namely to measure discrete differences in friction force of selected journal bearings due to changes in their operating conditions. In addition to its intended use, the friction force measurement system was used as a tool by which improvements were made to the test rig (illustrated shortcomings within the test rig design). For example, when the system was first retro-fitted, the square response was unrecognizable amid the friction force signal, (Figure 6.1). Further testing, however, determined that the "poor" friction force response was related to the tightened position of the shaft (Figure 6.2), which implied a misalignment within the test rig set-up. Alignment, stiffness and vibrational effects have been shown to alter both friction and wear [23]. To reduce these effects, the resonant springs were reoriented, and alignment washers were implemented (see Chapter 3). Afterwards, the square response improved considerably for amplitudes less than 3 mm.

The alignment problem could have been further reduced or eliminated if the squareness of the resonant springs was improved (did not compress straight), or the test rig was redesigned. Such improvements, however, would have required considerable time and additional costs — commodities unavailable at this stage in the project.

Because of the complexities of friction testing, one of the greatest assets a tribometer can possess is the ability to reproduce consistent results. As a result of taking great care in the testing procedure, reproducibility tests indicated a laboratory precision of 7% — a very admirable percentage, despite all the possible instrumental effects on friction. One of the three tests — Rulon 1122 — is provided in Figure 6.3.
6.4 AREAS OF IMPROVEMENT

While the results were encouraging, improvements can be made to not only the friction measurement system, but also the test rig. The friction force measurement device simply measured friction force — it did not measure load. Bearing load was determined by the force required to deflect the load springs a given distance. In the future, load measurement should be reincorporated into the friction system, since shaft alignment might affect bearing load. In addition, the coefficient of friction can be monitored over a range of loads by simply adjusting the height of the hex bolt to which it is attached. Currently, only one load can be applied per test, because the bearing atmosphere is compromised by the removal and subsequent replacement of the load springs.
One of the factors not adequately controlled was the relative humidity within the test rig chamber. Steps were taken to ensure a dry environment (dry helium, desiccant bags, and purging), but humidity was never measured within the chamber of the test rig during
either friction or wear measurements. As mentioned earlier, humidity influences the coefficient of friction. Since measurements were not made, it is possible that variations within the friction coefficients may have been the result of fluctuations in relative humidity. Therefore, the test rig should be equipped with a humidity probe. Also, an analyzer should be installed to determine the exact content of the gas within the test rig chamber prior to testing.

![Graph showing coefficient of friction over time](image)

**Figure 6.3. Reproducibility of Results for Rulon J -- Test 1122**

Furthermore, temperature was left unconstrained. To develop adequate relationships between load and friction or sliding velocity and friction, temperature must be controlled. In future tests, temperature could be regulated by placing an induction coil around the
circumferential surface of the test bearing or within the test rig itself. Both instances, however, require redesigning the test rig due to the temperature limitations of the motor.

6.5 RECOMMENDATIONS FOR FURTHER WORK

The effects of the contrast tests are in some respects suprising. It is believed that more testing should be performed to thoroughly investigate the effects of load, speed and counterface roughness on the oscillatory coefficient of friction for Iglide T500, Rulon J, and Vespel SP-3. Even though only a few parameters were studied, it may be in the best interest to restrict the scope of the tests to establish the effects of the most important factor -- load -- as a function of temperature and its effects on temperature.
Figure A.1. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure A.2. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm
Figure A.3. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure A.4. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm
Figure A.5. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm

Figure A.6. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure A.7. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm

Figure A.8. Reciprocating Coefficient of Friction for Iglide T500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm
Figure A.9. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm

Figure A.10. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm
Figure A.11. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure A.12. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm
Figure A.13. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - $Ra = 0.15-0.18 \mu m$ and $Rtm = 0.9-1.20 \mu m$.

Figure A.14. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - $Ra = 0.15-0.18 \mu m$ and $Rtm = 0.9-1.20 \mu m$. 
Figure A.15. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm

Figure A.16. Reciprocating Coefficient of Friction for Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure A.17. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm.

Figure A.18. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm.
Figure A.19. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure A.20. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm
Figure A.21. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm

Figure A.22. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm
Figure A.23. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm

Figure A.24. Reciprocating Coefficient of Friction for Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
APPENDIX B

INFLUENCE OF TEMPERATURE ON THE FRICTION COEFFICIENTS
Figure B.1. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): 
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - 
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.2. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): 
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - 
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.3. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and Rtm = 0.25-0.35 µm

Figure B.4. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and Rtm = 0.9-1.20 µm
Figure B.5. Influence of Interface Temperature on the Friction Coefficient of Rulon J:
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - 
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.6. Influence of Interface Temperature on the Friction Coefficient of Rulon J:
velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - 
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.7. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.8. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.9. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.10. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.24 m/s (1 mm amplitude), atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Appendix B

Figure B.11. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness -
Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.12. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3:
velocity = 0.48 m/s (2 mm amplitude), atmosphere = 400 KPa helium, surface roughness -
Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.13. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK):
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 µm and
Rtm = 0.25-0.35 µm

Figure B.14. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK):
load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 µm and
Rtm = 0.9-1.20 µm
Figure B.15. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK):
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and
Rtm = 0.25-0.35 μm

Figure B.16. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK):
load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and
Rtm = 0.9-1.20 μm
Figure B.17. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm.

Figure B.18. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm.
Appendix B

Figure B.19. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.20. Influence of Interface Temperature on the Friction Coefficient of Rulon J: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.21. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 \(\mu\)m and Rtm = 0.25-0.35 \(\mu\)m

Figure B.22. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 \(\mu\)m and Rtm = 0.9-1.20 \(\mu\)m
Figure B.23. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.24. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.25. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.26. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure B.27. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.28. Influence of Interface Temperature on the Friction Coefficient of Iglide T-500 (PEEK): velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure B.29. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.30. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Temperature (°C)

Figure B.31. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.32. Influence of Interface Temperature on the Friction Coefficient of Rulon J: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure B.33. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.34. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3:
velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure B.35. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium

Figure B.36. Influence of Interface Temperature on the Friction Coefficient of Vespel SP-3: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium
Figure B.37. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.38. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.39. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm.

Figure B.40. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm.
Figure B.41. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm.

Figure B.42. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 82.7 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm.
Figure B.43. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.05-0.08 μm and Rtm = 0.25-0.35 μm

Figure B.44. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.48 m/s (2 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
Figure B.45. Influence of Interface Temperature on the Friction Coefficient: velocity = 0.24 m/s (1 mm amplitude), load = 112.3 N, atmosphere = 400 KPa helium, surface roughness - Ra = 0.15-0.18 μm and Rtm = 0.9-1.20 μm
C.1 SPOKE DESIGN

ORIGINAL DESIGN

Initially, the width and length of the spoke were selected for ease of strain gage installation -- 21 and 19 mm respectively. The depth of the spoke was determined by taking into consideration the following: fatigue life of spokes, performance limits of strain gages, natural frequency of system and bridge resolution.

The design began by performing a force analysis. The spokes of the bearing holder were modeled as a beam fixed at one end and free at the other end (Figure C.1). The beam was subjected to concentrated loads (friction and normal forces) at the free end.

![Figure C.1. Model of Bearing Holder Spokes](image)

The maximum applied load, $F_n$, to the spokes by the mechanical load springs was 112.3 N. The maximum friction force, $F_f$, was determined by the equation

$$F_f = \mu F_n$$  \hspace{1cm} E[C.1]
By examination of the literature, a maximum coefficient of friction of 0.5 was chosen. As a result, the maximum friction force was 56.15 N.

Next, a lower bound approach was used to establish a minimum spoke thickness. The strength of the part must be designed to meet a design life of at least 2300 hours or 4.97E+8 cycles.

\[
2300 \text{ hrs } \times (60 \text{ cycles/sec } \times 3600 \text{ sec/hr}) = 4.97E+8 \text{ cycles}
\]

The fatigue strength \( \sigma_r \) for a typical aluminum alloy subjected to \( 10^9 \) cycles of reversed bending is 68.95 N/mm\(^2\) (Schaum's, p.73). If the fatigue strength was equated to the bending stress, an approximation for the depth of the spoke could be computed. The bending stress from the friction force is

\[
\sigma_b = \frac{Mc}{I}
\]

where the \( M \) is the moment, \( c \) is the distance from the neutral axis to the outermost fiber (\( h/2 \)), and \( I \) is the moment of inertia. The maximum moment is calculated by

\[
M = ((F_{fs}) \times L) \quad \text{E[C.2]}
\]

for a cantilever beam. \( F_{fs} \) is the friction force for each spoke -- 1/3 of total friction force \( F_f \); thus, \( F_{fs} \) was 18.72 N. The moment of inertia for a rectangular cross section is

\[
I = bh^3/12 \quad \text{[20]} \quad \text{E[C.3]}
\]
where \( b \) is the base (21 mm) and \( h \) is the spoke thickness. By setting \( \sigma_r = \sigma_b \),

\[
68.95 = 6\times F_{ta}L/(bh^2) = 6\times(18.72\times19)/(21\times h^2)
\]

\[
h = 1.21 \text{ mm}
\]

the spoke thickness was 1.21 mm. However, this was just an initial approximation using only bending stress. A more accurate approximation included axial stress (Equation C.5).

\[
\sigma_r = Mc/I + F_n/A \quad \text{E[C.4]}
\]

\[
68.95 = 6\times(18.72\times19)/(21\times h^2) + 112.3/(1.21\times21)
\]

\[
h = 1.25 \text{ mm}
\]

The thickness of the spoke from equation C.5 could be plugged back into the equation and an even more accurate value could be obtained. But, for all practical purposes, this measurement was refined enough.

As with most systems, a factor of safety was incorporated into the fatigue design. Besides, it is very unrealistic to expect a mechanical design to duplicate the fatigue strength values obtained in the laboratory. Afterall, the rotating beam specimen used in the laboratory is prepared very carefully and is tested under controlled conditions. A design factor could, in effect, negate any discrepancies. For this reason, the thickness of the spoke was chosen to be 2 mm. The design factor is defined by the modified Goodman relation as

\[
\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \quad \text{E[C.6]}
\]

[21]
where $\sigma_a$ is the alternating bending stress, $S_c$ is the fatigue strength of aluminum 6061, $\sigma_m$ is the midrange stress, $S_{ut}$ is the ultimate tensile strength of aluminum 6061, and $n$ is the factor of safety. The factors mentioned above were: $\sigma_a = +25.41$ N/mm$^2$, $S_c = +68.95$ N/mm$^2$, $\sigma_m = -2.67$ N/mm$^2$, and $S_{ut} = +310$ N/mm$^2$. According to the modified Goodman relation, the factor of safety was 2.78.

Strain gage performance was another limiting factor. Like the spokes, the strain gages were also subject to fatigue failure. The fatigue strength of the WK-13-062-350 strain gage is $\pm 2000 \mu \varepsilon$ for $10^8$ cycles [16]. In order for the gages to safely endure strain of this magnitude, the spoke thickness had to be properly sized. The sizing process began by converting the endurance strength from strain to stress. Stress was computed by the general equation:

$$\sigma = E \varepsilon$$

where $E$ is the modulus of elasticity, and $\varepsilon$ is the strain. The modulus of elasticity for aluminum 6061 is 70000 N/mm$^2$ [22]. The fatigue strength of the strain gage was

$$\sigma = (70000 \times 0.002) = 140 \text{ N/mm}$$

Plugging $\sigma$ in E[C.4] for $\sigma_f$ yielded the following quadratic equation:

$$h^2 - (5.35/140)h - 101.62/140 = 0$$

Its solution was a spoke thickness of $h = 0.871$ mm.
The third and final constraint was the natural frequency of the system. In the friction plane, the bearing case assembly, inner ring of the bearing holder and the three spokes act as a spring mass system (Figure C.2). For forced vibrations, the ratio of the forcing frequency to natural frequency, $\omega/\omega_n$, should be very small or very large. In either of the extreme cases, the magnification factor is minimal (Figure C.3). When $\omega$ approaches $\omega_n$, resonance occurs, and the amplitude of vibration is limited only by the damping present.

Unfortunately, the ratio of $\omega/\omega_n$ could not be designed above 0.3 because of the limitations imposed by the fatigue strength of the spoke ($h > 1.25$ mm). Consequently, the ratio was designed on the low side.

![Diagram](image)

Figure C.2. Modeling the Test Bearing Holder as a Spring Mass System

In order to achieve a low ratio of $\omega/\omega_n$, the thickness of the spokes was calculated so that the natural frequency of the system was at least ten times the forcing frequency of 60 Hz. The natural frequency of the system was related to the spring stiffness by the following equation:

$$\omega_n = \sqrt{\frac{K}{m}}$$

E[C.6]
where $K$ is the total spring stiffness and $m$ is the total system mass. Estimating the system mass from the component drawings in Appendix D to be 0.25 Kg, the spring stiffness (3 x's the spoke stiffness) was

$$K = \left(\omega_n\right)^2 \cdot m = \left(2\pi \cdot 10 \cdot 60\right)^2 \cdot 0.25 = 3553058 \text{ N/m}$$

![Figure C.3 Magnification Factor as a Function of Frequency Ratio [15]](image)

As a result, each spoke had a stiffness of 1184353 N/m. For a cantilever beam, the spring stiffness is defined by $E[3.4] \left( K = Ebh^3/L^3 \right)$. By substituting the total stiffness into equation 3.4, the minimum spoke thickness was calculated to be

$$1184353 = 70E9 \cdot 0.021 \cdot h^3/(0.019)^3$$
The thickness $h = 1.769 \times 10^{-3}$ m = 1.77 mm

Spoke fatigue, strain gage fatigue and the natural frequency of the system were all limiting constraints that defined the lower bound of the spoke thickness. In lieu of all the consequences, the most important constraint was the fatigue life of the spoke; it dictated the design of the member.

Finally, the spoke dimensions were checked against voltage resolution. If the resulting voltage signals were not within the resolution of an oscilloscope (5 mV), the 2 mm spoke thickness would have to be revamped. To check calibration output, a friction force of 22 and 23 Newtons was applied to a theoretical system setup (strain gage, Wheatstone bridge and an instrumentation amplifier).

A 22 N load results in a stress which induces a strain on the strain gage. The stress and strain were calculated as follows:

$$\sigma_s = \frac{Mc/I}{6Fr_sL/(bh^2)} = \frac{6(7.333333)(19)/(21 \times 2^2)}{9.952381 \text{ N/mm}}$$

$$\varepsilon = \frac{\sigma}{E} = \frac{9.952381}{70000} = 1.421769 \times 10^{-4}$$

As a result of the induced strain, the strain gage resistance changes. The change in resistance is defined as

$$\Delta R = \varepsilon FR$$

where $\Delta R$ is the change in resistance of the active strain gages, $\varepsilon$ is the strain, $F$ is the gage factor of the strain gage and $R$ is the strain gage resistance. For the WK-13-062-350 strain gage, the gage factor and resistance is 2.09 and 350 ohms, respectively.
The change in strain gage resistance is measurable by a Wheatstone bridge. The output of the Wheatstone bridge is given by the equation:

$$E_0 = \left( \frac{R_1 + \Delta R}{R_1 + R_2} - \frac{R_4 - \Delta R}{R_4 + R_3} \right) E_i$$

where $E_0$ is the voltage differential across the output terminals, $E_i$ is the strain gage excitation voltage and $R_1$ through $R_4$ are the four active strain gages. Because of their arrangement bridge arrangement, two of the strain gages increased in resistance (tension) while the other two decreased in resistance (compression). The excitation voltage for the strain gages was 2.5 VDC.

$$E_0 = \left( \frac{0.50014857 + 0.49985143}{0.50014857 + 0.49985143} - \frac{0.49985143 - 0.50014857}{0.50014857 + 0.49985143} \right) E_i = (7.429E-4) \times 2.5 = 7.429E-4 \text{ V}$$

The bridge output was calculated to be 7.429E-4 V. Since this output was very low, an amplifier was used to increase signal voltage. The amplifier gain $G$ was 1100. The signal output was

$$\text{Signal Output} = G \times E_0 = 1100 \times 7.429E-4 = 0.817 \text{ V}$$
Following the same procedure as above, a load of 23 N resulted in a signal output of 0.854 V. The difference between the two loads was 0.037 V. With this resolution, loads could accurately be predicted to within an eighth of a Newton. Taking into consideration the factors mentioned above, it was determined that the spoke thickness should be 2 mm.

REDESIGN

Although friction and load was successfully measured, preliminary testing revealed a design flaw. Originally, the spokes were designed on the assumption that the system was subjected to a sinusoidal forcing frequency of 60 Hz; however, this assumption was wrong. In actuality, the system is subjected to a square forcing frequency which, according to the Fourier expansion series, is comprised of odd harmonics of 60 Hz. So, the system is, in effect, excited by an infinite number of sinusoidal frequencies at every odd harmonic of 60 Hz -- 60, 180, 300, 420, 540, etc.

The harmonic that appeared in the friction force response varied from 660 to 680 Hz -- approximately the eleventh harmonic of 60 Hz. Theoretically, the natural frequency was determined by the equations 3.4 and 3.5. Substituting values into these equation resulted in a natural frequency of 730 Hz. The theoretical frequency was very close to the experimental value.

\[
K = \frac{Ebh^3}{L^3} = 70000 \times 21 \times 2^3 / (19)^3 = 1714.535 \text{ N/mm}
\]

\[
K_t = (1714.535 \times 1000) \times 3 \text{ spokes} = 5143606 \text{ N/m}
\]

\[
m = 0.244 \text{ Kg}
\]

\[
f_n = \frac{\sqrt{K_t/m}}{2\pi} = 730 \text{ Hz}
\]
To ensure that the natural frequency of the system was responsible for the harmonic, a simple frequency test was performed. The aluminum bearing case in the test assembly was replaced by a replica machined from brass. The natural frequency of the new system was 508 Hz.

\[ f_n = \frac{\sqrt{514360775027}}{2\pi} = 508 \text{ Hz} \]

The experimental frequency of the new system was 425 Hz. Consequently, the vibratory system was responsible for the harmonic.

To entirely eliminate the higher harmonic required a high \( \omega/\omega_n \) ratio, but, as previously mentioned, the highest attainable ratio was 0.3. The only other course of action was to increase the natural frequency of the system -- an increase in \( \omega_n \) would result in a decrease in the harmonic amplitude.

In the redesign an attempt was made to increase the natural frequency of the system to 6000 Hz, yet maintain the current levels of load and force resolution. This was accomplished by optimizing key variables in equations 3.3, 3.4, and C.6.

By examining equation 3.4 and C.6, the best approach for increasing the natural frequency of the system lay in maximizing the thickness of the spoke, while minimizing its length. The length was selected to be 12.7 mm -- the shortest possible length that would permit strain gage application.

The remaining variables were determined by a computer program. The program searched for acceptable dimensions within the following constraints:

1. Spoke length = 12.7 mm
2. Spoke width ≥ 4.7 mm
3. Maximum stress ≤ 68.95 N/mm -- the endurance strength
4. System mass = 0.04 Kg (theoretically)
5. The natural frequency of the system ≥ 6000 Hz
6. Friction response degradation ≤ 50%

Of all the acceptable possibilities, the following solution was selected:

1. Spoke length = 12.7 mm.
2. Spoke thickness = 5.8 mm.
3. Spoke width = 4.7 mm.
4. Two active strain gages

Theoretically, the redesign increased system rigidity by 1828% and reduced friction force resolution by 48%.

\[
\frac{K_{\text{after}}}{K_{\text{before}}} = \frac{12EI}{L^3} \frac{bh^3}{L^3} = \frac{4.7(5.8)^3}{12(2)^3} = 18.28 = 1828\%
\]

\[
\frac{\sigma_{\text{after}}}{\sigma_{\text{before}}} = \frac{6PL}{bh^2} \frac{(19-5)}{21(2)^2} = 0.482 = 48.2\%
\]

In addition, the natural frequency of the redesigned system was 7716 Hz.

\[
K = Ebh^3/L^3 = 70000\times4.7\times5.8^3/(12.7)^3 = 31337.815 \text{ N/mm}
\]

\[
Kt = (31337.815\times1000)\times3 \text{ spokes} = 94013446 \text{ N/m}
\]
m = 0.040 Kg

\[ f_n = \frac{\sqrt{K/m}}{2\pi} = 7716 \text{ Hz} \]

After the friction apparatus was returned from the shop, testing resumed. During the early stages of calibration, another problem became evident — load resolution had greatly diminished. Theoretically, a load resolution loss of 23% was expected, due to the reductions in spoke area and the number of active strain gages. However, the experimental reduction was far greater.

\[
\frac{\sigma_{\text{after}}}{\sigma_{\text{before}}} = \frac{\frac{P \times 1}{A \times 2}}{\frac{P}{A}} = \frac{\frac{1}{2bh}}{\frac{1}{bh}} = \frac{1}{2(4.7)(5.8)} = 0.77 = 77\%
\]

On the other hand, friction force was as expected -- no harmonics.

MODIFICATIONS

Load measurement was an absolute necessity, if the load at the bearing was unknown. As such, the load resolution was increased back to its former level by machining the thickness of the spokes to 4.1 mm.

\[
\frac{\sigma_{\text{after}}}{\sigma_{\text{before}}} = \frac{1}{\frac{2(4.7)(4.1)}{(21)(2)}} = 1.09 = 109\%
\]
The reduction in the thickness of the spoke changed the theoretical natural frequency to 4585 Hz.

\[
K = \frac{Ebh^3}{L^3} = 70000 \times 4.7 \times 4.1^3/(12.7)^3 = 11069.712 \text{ N/mm}
\]

\[
Kt = (11069.712 \times 1000) \times 3 \text{ spokes} = 33209135 \text{ N/m}
\]

\[
m = 0.040 \text{ Kg}
\]

\[
f_n = \frac{\sqrt{Kt/m}}{2\pi} = 4585 \text{ Hz}
\]

However, the estimate for the design mass was incorrect. After weighing the redesigned bearing case components, the mass of the system was recalculated to be 0.0826 Kg. As a result, the natural frequency of the system was actually 3191 Hz. This system was used for testing.

**C.2 INERTIA SPRINGS**

The function of the inertia springs is to secure the test bearing assembly against the alignment bearing holders. Otherwise, the bearing assembly would be free to move with the reciprocating motion of the shaft or the housing of the test rig.

The spring force required to hold the test piece against the alignment holders is, in the worst case, the sum of the friction and inertia forces imposed on test bearing assembly by the housing. Assuming a conservative coefficient of friction of 0.5 and a normal load of 112.3 N, the maximum frictional force was calculated to be 56.15 N.

\[
F_f = \mu F_n = 0.5 \times 112.3 = 56.15 \text{ N}
\]
The inertia force of the bearing assembly was calculated from the following equation:

\[ F_{ib} = ma \quad \text{E[C.9]} \]

where \( F_{ib} \) is the inertia force, \( m \) is the mass of the test bearing assembly, and \( a \) is the acceleration of the test bearing assembly. The mass of the test bearing assembly is 0.377 Kg. If the bearing assembly is in constant contact with the alignment bearings, the acceleration of the test bearing assembly and the test rig housing are equal, since the alignment bearings are fastened to the housing by set screws. The acceleration of the housing is equal to the inertia force of the housing divided by its mass (equation C.9 solved for \( a \)). The mass of the housing is 64 Kg. The inertia force is equal to the inertia force of the moving parts (piston, shaft and one-third of the resonant springs) since the center of mass for the test rig is constant, and there is no reactionary forces. The maximum inertia force of the moving parts was calculated by the equation:

\[ F_i = m\omega_r^2X \quad \text{E[C.10]} \]

where \( F_i \) is the inertia force, \( m \) is the mass of the moving parts, \( \omega_r \) is the forcing frequency of the system, and \( X \) is the amplitude of motion. For a moving mass of 2.35 Kg, an amplitude of 7 mm (maximum allowable) and a forcing frequency of 60 Hz, the inertia force was calculated to be 2.34 KN (see below).

\[ F_i = 2.35 \times (60 \times 2 \times \pi)^2 \times 0.007 = 2.34 \text{ KN} \]
By dividing $F_i$ by the mass of the housing, the acceleration was calculated to be 36.6 m/sec. Finally, the inertia force of the test bearing assembly was determined to be 16.03 N.

$$F_{ib} = ma = 0.438 	imes 36.6 = 16.03 \text{ N}$$

The total force required to secure the test bearing assembly was 72.18 N.

$$F = F_{ib} + F_f = 16.03 + 56.15 = 72.18 \text{ N}$$

The spring spacer and spring holder was designed so that each spring would be compressed 3.41 mm. As a result, each spring (6 total) exerts a force of 15 N.

### C.3 LOAD

Load was applied to the test bearing by two mechanical springs sitting at an angle of 45° with the vertical centerline of the machine. The springs were designated black and white. The black spring was used for all low load tests. Both springs were used during the high load tests. At their compressed lengths, the springs exerted forces of 80.07 N (black) and 73.40 N (white). Also, a load of 3.69 N was applied to the test bearing by the weight of the test bearing assembly. Calculations of the force vectors resulted in magnitudes of 82.7 N and 112.3 N for the load contrast tests.

low load (black spring):

$$L_L = (80.07 \times \cos 45^\circ)i - (80.07 \times \sin 45^\circ)j - 3.69j = 82.7 \text{ N @ } -46.8^\circ$$
high load (black & white springs): 

\[ L_H = L_L - (73.40 \times \cos 45°)i - (73.40 \times \sin 45°)j = 112.3 \text{ N @ -87.6°} \]

C.4 SYSTEM MASS

The effective system mass was found by summing the individual masses of the bearing case subassembly, inner ring of the bearing holder and one-third of each spoke. The mass of the bearing case subassembly was computed from its weight to be 0.0616 Kg. The mass contributions from the spokes and inner ring of the bearing holder were estimated from the equation

\[ m = \rho V \quad \text{E[C.11]} \]

where \( m \) is the mass, \( \rho \) is the mass density of aluminum 6061 (2700 Kg/m³ Gerre p.743), and \( V \) is the volume of material used. According to the calculations, the mass of the inner ring of the bearing was 0.0201 Kg, and the mass of each spoke was 0.007 Kg.

inner ring:

\[ m_1 = \rho V = \rho \pi (r_o^2 - r_i^2)h = 2700(0.0318^2 - 0.0286^2)7.9 = 0.01295 \text{ Kg} \]
\[ m_2 = \rho \pi (r_o^2 - r_i^2)h = 2700(0.0349^2 - 0.0318^2)4.1 = 0.00719 \text{ Kg} \]
\[ m_t = m_1 + m_2 = 0.02014 \text{ Kg} \]

spokes:

\[ m = \rho V = \rho \pi tL = 2700\times0.0047\times0.0041\times0.0127 = 0.00066 \text{ Kg} \]
\[ m_t = 3\times m = 0.00198 \text{ Kg} \]
The effective system mass was 0.0824 Kg.

\[ m_s = m_{\text{bearing case}} + m_{\text{inner ring}} + m_{\text{spokes/3}} \]

\[ m_s = 0.0616 + 0.02014 + 0.00066 = 0.0824 \text{ Kg} \]
APPENDIX D

COMPONENT DRAWINGS OF TEST ASSEMBLY
Figure D.1. Specifications of Bearing Holder Design - Front View
Figure D.2. Specifications of Bearing Holder Design - Rear View
Figure D.3. Specifications of Bearing Case
Figure D.4. Specifications of Bearing Holder Redesign - Front View
Figure D.5. Specifications of Bearing Holder Redesign - Rear View

- 6.0 DRILL x 12.7
- 5 Holes

SECTION A-A
Figure D.6. Specifications of Bearing Case Redesign
Figure D.7. Modifications Made to Bearing Holder Redesign

Appendix D
Figure D.9. Specifications of Inertia Spring Holders
BIBLIOGRAPHY


[21] Shigley and Mischke 299.


[40] *Bearings*. Pamphlet from Sparta Manufacturing Company.


[54] Rabinowicz 70-82.

