Development of a MEMS Turbocharger and Gas Turbine Engine
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Abstract

As portable electronic devices proliferate (laptops, GPS, radios etc.), the demand for compact energy sources to power them increases. Primary (non-rechargeable) batteries now provide energy densities upwards of 180 W-hr/kg, secondary (rechargeable) batteries offer about \( \frac{1}{2} \) that level. Hydrocarbon fuels have a chemical energy density of 13,000-14,000 W-hr/kg. A power source using hydrocarbon fuels with an electric power conversion efficiency of order 10% would be revolutionary. This promise has driven the development of the MIT micro gas turbine generator concept. The first engine design measures 23 x 23 x 0.3 mm and is fabricated from single crystal silicon using MEMS micro-fabrication techniques so as to offer the promise of low cost in large production.

This thesis describes the development and testing of a MEMS turbocharger. This is a version of a simple cycle, single spool gas turbine engine with compressor and turbine flow paths separated for diagnostic purposes, intended for turbomachinery and rotordynamic development. The turbocharger design described herein was evolved from an earlier, unsuccessful design (Protz 2000) to satisfy rotordynamic and fabrication constraints. The turbochargers consist of a back-to-back centrifugal compressor and radial inflow turbine supported on gas bearings with a design rotating speed of 1.2 Mrpm. This design speed is many times the natural frequency of the radial bearing system. Primarily due to the exacting requirements of the micron scale bearings, these devices have proven very difficult to manufacture to design, with only six near specification units produced over the course of three years. Six proved to be a small number for this development program since these silicon devices are brittle and do not survive bearing crashes at speeds much above a few tens of thousands of rpm. The primary focus of this thesis has been the theoretical and empirical determination of strategies for the starting and acceleration of the turbocharger and engine and evolution of the design to that end.

Experiments identified phenomena governing rotordynamics, which were compared to model predictions. During these tests, the turbocharger reached 40% design speed (480,000 rpm). Rotordynamics were the limiting factor. The turbomachinery performance was characterized during these experiments. At 40% design speed, the compressor developed a pressure ratio of 1.21 at a flow rate of 0.13 g/s, values in agreement with CFD predictions. At this operating point the turbine pressure ratio was 1.7 with a flow rate of 0.26 g/s resulting in an overall spool efficiency of 19%.

To assess ignition strategies for the gas turbine, a lumped parameter model was developed to examine the transient behavior of the engine as dictated by the turbomachinery fluid mechanics, heat transfer, structural deformations from centrifugal and thermal loading and rotordynamics. The model shows that transients are dominated by three time constants – rotor inertial (10^4 sec), rotor thermal (1 sec), and static structure thermal (10 sec). The model suggests that the engine requires modified bearing dimensions relative to the turbocharger and that it might be necessary to pre-heat the structure prior to ignition. Finally, the model suggests that the relatively large thermal mass of the combustor reduces the risk of compressor surge during the ignition transient.

Suggestions for future work include a re-design of the thrust bearing system, the addition of a functioning thrust balancing system, and improvement of the design used to introduce anisotropy to the journal bearing so as to reduce static torques acting on the rotor.

Thesis Supervisor: Alan H. Epstein
Title: R.C. MacLaurin Professor of Aeronautics and Astronautics
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I dedicate this work to my beloved grandfather, Theofilos Savoulides (1999).

He will always live in my memory as a paradigm...
CONTENTS

CHAPTER 1 .......................................................................................................21
Introduction .......................................................................................................21
1.1 Background ................................................................................................21
1.2 Demo Engine Research ............................................................................22
1.3 Objectives and Approach ..........................................................................27
1.4 Applications of Research ..........................................................................28
1.5 Previous Work ..........................................................................................29
1.5.1 Systems Integration .............................................................................29
1.5.2 Thermal, Structural, and Materials Design .........................................30
1.5.3 Combustion ..........................................................................................31
1.5.4 Turbomachinery and Fluid Dynamics ..................................................31
1.5.5 Bearing Design and Rotor Dynamics ....................................................32
1.5.6 Electrical Machinery ..........................................................................33
1.5.7 Fabrication and Packaging ...................................................................33
1.6 Literature Review .....................................................................................34
1.6.1 MEMS Gas Turbine Engine – Tohoku University (Japan) .................34
1.6.2 Micromachined Gas Turbine Generator – IHI and Tohoku University ....34
1.6.3 Micromachined Gas Turbine – Stanford CA and Honda Japan ........35
1.6.4 Ultra Micro Gas Turbines – University of Tokyo ..................................36
1.6.5 MEMS Rotary Engine – UC, Berkeley ...............................................36
1.6.6 MEMS Free-Piston Knock Engine – Honeywell ..................................38
1.6.7 Other Projects ......................................................................................39
1.6.8 Advantages / Disadvantages of Other Approaches ...............................39
1.7 Organization of Thesis ............................................................................39
1.8 References ...............................................................................................41

CHAPTER 2 .......................................................................................................51
Engine Design Evolution and Fabrication .......................................................51
2.1 Fabricated Devices ....................................................................................51
2.2 Turbocharger Layout ................................................................................52
2.3 Fabrication ................................................................................................55
2.3.1 Constraints ..........................................................................................56
2.3.2 Special Fabrication Processes ...............................................................57
2.3.3 Manufacturing Yield ............................................................................63
2.4 Design Evolution .......................................................................................65
2.4.1 Initial Design ........................................................................................65
2.4.2 1st Re-Design (Builds A, B and C) .......................................................67
2.4.3 2nd Re-Design (Build D1, D3, D4) .......................................................69
2.4.4 3rd Re-Design (Locos 4) ......................................................................83
## CHAPTER 5

- 5.3.2 Axial Stiffness ................................................................. 175
- 5.4 Assessment of Rotor Moments ........................................... 177
- 5.4.1 1st Moments ................................................................ 177
- 5.4.2 Axial-Radial Coupling - 2nd Moments ......................... 178
- 5.5 Rotordynamic Stability Phenomena ................................. 179
- 5.5.1 Radial Instability ............................................................ 179
- 5.5.2 Axial Instability ............................................................... 179
- 5.5.3 Summary of Crashes ....................................................... 181
- 5.6 Summary ........................................................................ 182
- 5.7 References ...................................................................... 182

## CHAPTER 6

- Testing - Turbomachinery ....................................................... 183
- 6.1 High Speed Challenges ...................................................... 183
- 6.2 Test Procedure ................................................................ 185
- 6.3 High Speed Data ............................................................... 189
- 6.4 High Speed Data Analysis ................................................ 194
- 6.5 Summary ........................................................................ 198

## CHAPTER 7

- Demo Engine Startup Transient Modeling ............................. 201
- 7.1 Ignition Setup and Operational Overview ......................... 201
- 7.2 Assumptions and Boundary Conditions ............................ 202
- 7.3 Cold Flow ....................................................................... 203
- 7.3.1 Compressor Map .......................................................... 204
- 7.3.2 Turbine Map ................................................................. 205
- 7.3.3 Cold Flow Equilibrium Running Point ......................... 206
- 7.4 Ignition Transient ............................................................. 207
- 7.4.1 Thermal Analysis .......................................................... 208
- 7.4.2 Rotor Inertia ................................................................. 218
- 7.4.3 Structural Deformations ............................................... 219
- 7.4.4 Rotordynamics ............................................................. 220
- 7.5 Model Validation ............................................................ 221
- 7.5.1 Angular Acceleration .................................................... 221
- 7.5.2 Combustion ................................................................. 224
- 7.6 Results and Ignition Acceleration Strategies...................... 226
- 7.6.1 Simulation 1 - Baseline ............................................... 226
- 7.6.2 Simulation 2 - Baseline + Clearance Variation .............. 235
- 7.6.3 Simulation 3 - Baseline + Device Pre-Heating ................ 239
- 7.6.4 Simulation 4 - Baseline + Thermally Isolated Compressor 243
- 7.6.5 Other Strategies .......................................................... 247
- 7.7 Summary ....................................................................... 247
- 7.8 References .................................................................... 249

## CHAPTER 8

- Operating Speed Limitations and Improvements ................. 251
APPENDIX E ................................................................. 297
Parasitic Power Loss Model............................................... 297

APPENDIX F ................................................................. 301
Compressor Map Interpolation ........................................... 301

CHAPTER G ................................................................. 305
Component Matching....................................................... 305
LIST OF FIGURES

Figure 1-1: a) Baseline Engine including generator, with compressor and turbine........... 24
Figure 1-2: 3-D Schematic of Demo Engine (courtesy Diana Park)............................ 24
Figure 1-3: The turbocharger is built by etching two dimensional features.................. 26
Figure 1-4: The turbocharger is built by bonding six wafers.................................... 26
Figure 1-5: (a) Top view of real engine and compressor............................................ 27
Figure 1-6: Steel 13mm diameter rotor rotary engine. (Courtesy Fernandez-Pello)........ 37
Figure 1-7: Components of the 1mm MEMS rotary engine........................................ 37
Figure 1-8: Honeywell knock engine............................................................................. 38

Figure 2-1: Turbocharger cross section indicating main components........................... 53
Figure 2-2: Schematic of the bearing system of the turbocharger............................... 54
Figure 2-3: Demo Engine: Identical to the turbocharger with the exception............... 54
Figure 2-4: Turbocharger pressure tap locations....................................................... 55
Figure 2-5: Etch non-uniformity.................................................................................. 57
Figure 2-6: Narrow and large feature etching............................................................. 58
Figure 2-7: Two masking materials can be used to etch overlaid features.................... 59
Figure 2-8: Stand alone bearing etch process............................................................. 60
Figure 2-9: Compressor bearing fabrication process................................................... 62
Figure 2-10: Oxide pad rotor release scheme............................................................... 63
Figure 2-11: Timeline of devices delivered by fabrication team...................................... 64
Figure 2-12: Initial design of the turbocharger........................................................... 66
Figure 2-13: 1st Re-Design......................................................................................... 67
Figure 2-14: 2nd Re-Design....................................................................................... 70
Figure 2-15: Hydrostatic journal supply resistor network model................................... 73
Figure 2-16: Pressure drops in hydrostatic supply system........................................... 75
Figure 2-17: O-Ring force per unit length for varying cross sectional area................. 77
Figure 2-18: Deflection at center of device due to O-Ring compressive force............. 78
Figure 2-19: Effect of center O-Ring on thrust bearing flow rate.................................. 79
Figure 2-20: Bond alignment feature.......................................................................... 81
Figure 2-21: 1st Turbine blade design change............................................................... 82
Figure 2-22: Build Locos 4......................................................................................... 83
Figure 2-23: Compressor blade and fiber optic tab re-design........................................ 85
Figure 2-24: Rotor deformation schematic due to centrifugal loading.......................... 86
Figure 2-25: (a) Variation of seal clearance with speed................................................ 88
Figure 2-26: 2nd Turbine blade design change............................................................. 89

Figure 3-1: 3D schematic of turbocharger rotor......................................................... 94
Figure 3-2: Hydrostatic thrust bearing operation......................................................... 96
Figure 3-3: Hydrodynamic journal bearing operation................................................. 98
Figure 5-5: Measurement of journal natural based on fiber data .......................................................... 171
Figure 5-6: Summary of Build C turbine bearing natural frequency .................................................. 172
Figure 5-7: Single compressor journal bearing natural frequencies .................................................. 173
Figure 5-8: Fiber data taken showing that the anisotropic compressor bearing .................................. 174
Figure 5-9: Extended single turbine journal bearing natural frequency .............................................. 175
Figure 5-10: Thrust bearing natural frequency .................................................................................. 176
Figure 5-11: Thrust bearing flow rates decreasing with increasing speed .......................................... 178
Figure 5-12: Thrust bearing instability .............................................................................................. 180

Figure 6-1: Rotordynamic acceleration map ...................................................................................... 187
Figure 6-2: Thrust bearing flow rate changes during an increase in .................................................. 188
Figure 6-3: Summary of high speed data ......................................................................................... 192
Figure 6-4: Spool overall efficiency ................................................................................................. 195
Figure 6-5: Turbine/compressor efficiency relation .......................................................................... 197
Figure 6-6: Power distribution in turbocharger ................................................................................ 198

Figure 7-1: Interpolation scheme example results .............................................................................. 205
Figure 7-2: Turbine map based on experimental results ..................................................................... 206
Figure 7-3: Engine Station definitions ............................................................................................... 208
Figure 7-4: Static structure thermal model ......................................................................................... 209
Figure 7-5: Rotor 1-D finite element model ....................................................................................... 215
Figure 7-6: Spindown test .................................................................................................................. 223
Figure 7-7: Combustor experimental results overlaid with the model predictions.............................. 225
Figure 7-8: Simulation 1 - Compressor map(s) .................................................................................. 228
Figure 7-9: Simulation 1 - Turbine flow rate ...................................................................................... 228
Figure 7-10: Simulation 1 - Temperatures ......................................................................................... 229
Figure 7-11: Simulation 1 - Rotor heat flux ....................................................................................... 229
Figure 7-12: Simulation 1 - Static structure heat flux ......................................................................... 230
Figure 7-13: Simulation 1 - Bearing clearance .................................................................................. 230
Figure 7-14: Simulation 1 - Speed, stability boundary, and natural frequency ................................. 231
Figure 7-15: Simulation 2 - Nominal Bearing Clearance 16 µm ......................................................... 236
Figure 7-16: Simulation 2 - Nominal Bearing Clearance 17 µm .......................................................... 237
Figure 7-17: Simulation 2 - Nominal Bearing Clearance 18 µm ......................................................... 238
Figure 7-18: Simulation 2 - Nominal Bearing Clearance 17.5 µm ..................................................... 239
Figure 7-19: Simulation 3 (C₀ = 17 µm) - Compressor map(s) ............................................................. 241
Figure 7-20: Simulation 3 (C₀ = 17 µm) - Temperatures ................................................................. 241
Figure 7-21: Simulation 3 (C₀ = 17 µm) – Bearing Clearance ............................................................. 242
Figure 7-22: Simulation 3 (C₀ = 17 µm) – Speed/Stability ................................................................. 242
Figure 7-23: Simulation 4 - Compressor map operating line ............................................................. 244
Figure 7-24: Simulation 4 - Temperatures ......................................................................................... 245
Figure 7-25: Simulation 4 – Bearing Clearance ................................................................................ 245
Figure 7-26: Simulation 4 – Speed/Stability ..................................................................................... 246

Figure 8-1: Turbocharger constraints imposed by fabrication tolerances .......................................... 252
Figure 8-2: Effect of thrust bearing clearance reduction on the axial natural ................................... 257
Figure A-1: Schematic of isothermal compressible flow between two flat plates ........ 266
Figure A-2: Mass flow rate comparison ...................................................................................... 271
Figure A-3: Numerical procedure followed to compute pressure along the duct ........ 273
Figure A-4: Pressure profile along a bearing with no entrance loss ................ 273
Figure A-5: Force acting on bearing wall with no entrance loss ........................................ 274

Figure B-1: Total flow rate through each node in the circumferential direction ........ 276
Figure B-2: Pressure at each node ........................................................................................ 277

Figure C-1: Turbocharger layout ............................................................................................. 279
Figure C-2: Turbocharger cold package .................................................................................... 281
Figure C-3: Turbocharger hot package (external electrically heated air) ........ 283
Figure C-4: Thermocouple plug for hot package ................................................................. 284
Figure C-5: Glass bead package schematic ........................................................................... 285
Figure C-6: Glass bead packaged turbocharger .................................................................... 286
Figure C-7: Combustion package ......................................................................................... 288
Figure C-8: Vacuum control system for combustion package ............................................ 289

Figure E-1: Thrust bearing power loss modeled as Couette flow ......................................... 297
Figure E-2: Parasitic power loss breakdown ............................................................................ 300

Figure F-1: CFD compressor map for 220 µm blade span (Sirakov) ............................... 301
Figure F-2: CFD compressor stage efficiency for 220 µm blade span (Sirakov) .......... 302

Figure G-1: Engine station definitions ................................................................................... 305
LIST OF TABLES

Table 2-1: 1st Re-Design Changes ........................................................................ 68
Table 2-2: 2nd Re-Design Changes .................................................................... 71
Table 2-3: Hydrostatic supply line pressure drop model (at 0.039 g/s - 2000sccm) ... 74
Table 2-4: Extrapolation of force per unit length for 0.406mm (0.016inch) O-Ring ... 77
Table 2-5: Calculated performance of original and improved turbine designs ....... 82
Table 2-6: 3rd Re-Design Changes .................................................................... 84
Table 2-7: Performance of original and improved compressor designs ............. 86
Table 2-8: Rotor deformations based on both compressor blade designs .......... 86

Table 3-1: Thrust bearing design ..................................................................... 96
Table 3-2: Typical values of parameters (single compressor bearing design) ....... 103
Table 3-3: Optimal bearing geometries (with no deformations) ......................... 110
Table 3-4: Journal Bearing Design (including deformations) ............................ 119
Table 3-5: Imbalance introduce by plane-like etch non-uniformity ..................... 142
Table 3-6: Imbalance model for turbine bearing device .................................... 143
Table 3-7: Predicted levels of imbalance based on fabrication capability estimates .. 144
Table 3-8: Imbalance model for compressor bearing device ............................. 145
Table 3-9: Predicted levels of imbalance based on fabrication capability estimates .. 145

Table 5-1: Thrust bearing geometry example - Build L4, Die 2 ....................... 164
Table 5-2: Journal bearing geometry example - Build C, Die 8 ....................... 167
Table 5-3: Seal Geometry ................................................................................. 168
Table 5-4: Measured Imbalance of devices ...................................................... 177
Table 5-5: Stability boundary predictions for turbine bearing devices .............. 181

Table 6-1: Summary of high speed devices ...................................................... 189

Table 7-1: Natural convection parameters .......................................................... 212
Table 7-2: Conduction parameters .................................................................... 213
Table 7-3: Silicon properties .............................................................................. 215
Table 7-4: Spindown Test Comparison with Model .......................................... 223
Table 7-5: Baseline simulation parameters ....................................................... 226
Table 7-6: Simulation 2 parameters .................................................................. 235
Table 7-7: Simulation 3 parameters .................................................................. 240
Table 7-8: Simulation 4 parameters .................................................................. 244

Table 8-1: Tilt Angle Due to Etch Variations of Hydrostatic Plenum (@ 6.9 kPa) ... 255
Table 8-2: Tilt Angle Due to No Pressure on One Side of Rim ....................... 255
Table C-1: Turbocharger layout connection ports .......................................................... 280
Table D-1: Pressure Transducer Uncertainty ................................................................. 292
Table D-2: Mass Flow Meter Uncertainty ..................................................................... 292
Table D-3: Journal Bearing Clearance Uncertainty .................................................. 294
Table D-4: Thrust Bearing Nozzle Diameter Uncertainty ....................................... 294
Table D-5: Turbine Pressure Ratio Uncertainty ......................................................... 295
Table D-6: Compressor Fluidic Power Uncertainty ................................................... 296
Table D-7: Turbine Fluidic Power Uncertainty ............................................................ 296

Table E-1: Parasitic power losses at design speed (1,200,000 rpm) ......................... 299
**NOMENCLATURE**

**Roman**

A  
area

Bi  
biot number

c  
damping, specific heat capacity of silicon

C  
journal bearing clearance

C_p  
specific heat at constant pressure

Dh  
hydraulic diameter

E  
imbalance (mg-um)

e  
imbalance (um)

f  
mooody friction factor (darcy friction factor)

f_{fan}  
fanning friction factor (f / 4)

Fo  
fourier number

h  
clearance

I  
rotor moment of inertia

Imb  
imbalance divided by rotor mass (um)

imb  
imbalance (mg-um)

k  
stiffness, thermal conductivity

L  
length

m  
rotor mass

m_s  
static structure mass

m  
mass flow rate

M  
mach number

P  
perimeter, power

p  
static pressure

p_t  
total pressure
\( \dot{Q} \)  
heat flux (watts)

\( r \)  
radius

\( R \)  
gas constant \((C_p - C_v)\)

\( \text{Re} \)  
reynolds number

\( t \)  
time

\( T \)  
static temperature

\( T_t \)  
total temperature

\( U \)  
velocity

**Greek**

\( \alpha \)  
principal axis angle w.r.t. geometric axial axis, thermal diffusivity

\( \gamma \)  
thermal expansion coefficient

\( \epsilon \)  
eccentricity ratio

\( \zeta \)  
journal bearing damping ratio

\( \eta_{\text{spool}} \)  
spool efficiency

\( \mu \)  
viscosity

\( \pi \)  
pressure ratio \( \left( \frac{P_{n+1}}{P_n} \right) \)

\( \rho \)  
density

\( \sigma \)  
steffan boltzmann constant

\( \tau \)  
torque, shear stress

\( \omega \)  
angular speed

\( \omega_n \)  
journal bearing natural frequency
CHAPTER 1

INTRODUCTION

As electrical devices proliferate and become a part of everyday life for personal and military use (cell phones, laptops, GPS, radios etc.) the capability of conventional batteries to supply the requisite long duration energy and power to drive these devices is being sorely pressed. Primary (non-rechargeable) batteries can provide energy densities upwards of 180 W-hr/kg at reasonable power levels, but these batteries are expensive and contain hazardous waste. Secondary (rechargeable) batteries provide about half the energy density of primary batteries. Imagine the potential of a portable power source which has well over five times the energy density of today’s best primary batteries. Hydrocarbon fuels have a chemical energy density of 13,000-14,000 W-hr/kg. A power source using hydrocarbon fuels with an electric power conversion efficiency of 10% would be revolutionary. This concept has driven the development of Power MEMS and the MIT microengine program.

This chapter will first introduce the microengine concept, giving a background discussion followed by an overview of the project. The objectives of this research as well as possible applications of microengines shall then be discussed. As this project has been ongoing for almost ten years some time will be spent presenting previous work done along with an extensive list of references. Finally, a literature review of similar research in Power MEMS will be presented.

1.1 Background

In 1994, Epstein et al. [1] proposed the concept of a MEMS (micro electro-mechanical systems) centimeter-scale gas turbine engine that would be manufactured using micro-fabrication techniques. This idea led to the creation of a multidisciplinary research program that currently employs about forty students, staff, and faculty. The device was to be made of silicon/silicon-carbide and was designed to operate based on a
Brayton cycle, just like conventional gas turbine engines. Successful creation of the device would rely on the development of high precision micro-fabrication techniques, high-speed turbomachinery, high-speed gas bearings, compact combustion systems, high-power micro electrical generators, high-temperature micro-scale packaging, high performance structures, and advanced materials. Possible applications considered at the inception of the program included propulsion, electrical power generation, motor-driven compressors for fluidic pumping and pressurization, micro-coolers for microchip cooling, and micro-rockets for small scale space propulsion [2,3,4,5,6,7]. Later the project focus shifted to electrical power generation and micro-rockets. This thesis will focus entirely on the gas turbine engine suitable for the electrical power generation application.

Even at a conventional scale, aircraft gas turbine engine design is challenging due to the fact that it consists of highly stressed high speed rotating components which are exposed to high temperature gases, must operate reliably and must be very light so as to be of any value for an aircraft. Fluid dynamic, thermodynamic, and structural concerns must be satisfied simultaneously for every component. In addition, all the secondary components such as the ignition system, secondary flow systems, starter motor, and electrical generator must be able to reliably operate at elevated temperatures.

Micro electro-mechanical systems are also highly interdisciplinary. Most MEMS devices rely on structural analysis, packaging design, circuit design, controls design, and fabrication process design. This makes a MEMS device challenging to build, usually requiring the effort of many people with different skills.

The microengine relies on both gas turbine engine design theory as well as MEMS design. This has led to the creation of a large team which occupies students, staff, and faculty from a large array of fields, which is split into several groups, each of which deals with a separate problem while retaining compatibility with each others work.

1.2 Demo Engine Research

The original idea conceived by Epstein was to build a micro gas turbine generator. The baseline design consisted of a single stage rotating spool with an integrated combustor and electrical generator. Efforts initially focused on building and testing a demo gas turbine engine and an electrical generator separately. Furthermore, the demo
engine development focused on two separate devices: a combustor and a turbocharger. The combustor retains all the features of the demo engine but does not have a rotating spool. The objective of this device is to characterize and determine the feasibility of combustion at the micro level and assess the thermal integrity of the design. The turbocharger is nearly identical to the desired demo engine, including a rotating spool and a combustor, it only differs in that the flow path of the compressor is separated from that of the turbine. The objective of this device is to characterize the performance of the turbomachinery, and test the rotordynamics. To further test the rotordynamics of the device a bearing device was also built and tested. This device consisted of just a turbine supported by a bearing system that was similar to the one in the engine. This thesis will focus entirely on the turbocharger and gas turbine engine. Figure 1-1 depicts the original baseline design, the demo engine design and the turbocharger design. Figure 1-2 is a 3-D schematic of the demo-engine.
Figure 1-1: a) Baseline Engine including generator, with compressor and turbine separated by a shaft. b) Demo Engine (removed generator and shaft). c) Turbocharger (separated compressor and turbine flow paths, but retains capability to run as an engine by externally connecting flow paths of compressor and turbine).

Figure 1-2: 3-D Schematic of Demo Engine (courtesy Diana Park)
The current turbocharger measures 23x23x2.9 mm and is made entirely of silicon. It consists of an 8 mm diameter compressor and a 6 mm diameter turbine that spin at 1.2 million rpm. Integrated in the device is a combustor which encircles the turbine. The airfoil spans for both the turbine and compressor are 210 μm. Once testing of the turbocharger has been completed, engines with the compressor flow path connected to the turbine flow path will be assembled. Design operating conditions for these devices call for a combustor exit temperature of 1600 K, a compressor pressure ratio of 2 with an adiabatic efficiency of 65%, a mass flow rate of 0.55g/s and are expected to have a marginally closed cycle. The blade spans would also have to be increased to 400 μm. With some additional improvements to the design, an overall (chemical to electrical) conversion efficiency of order 5-10% is expected.

The turbocharger is fabricated using MEMS semiconductor microfabrication techniques. Each engine is assembled from six 100 mm diameter single-crystal wafers that are etched to define features and then bonded together to form complete engines. Each wafer corresponds to one or more engine components. The bonded set of wafers is then cut into four individual engines (called ‘dies’). Figure 1-3 and Figure 1-4 depict the assembly process and Figure 1-5 includes several views of a completed real engine.
Figure 1-3: The turbocharger is built by etching two dimensional features on six separate levels which are then bonded together (Courtesy of Diana Park)

Figure 1-4: The turbocharger is built by bonding six wafers which gives rise to 10 dies - which was later reduced to the 4 corner dies due to etch uniformity concerns. (Courtesy of Diana Park)
1.3 Objectives and Approach

This research has two primary objectives. The first objective is to demonstrate high speed operation of an all-silicon micro turbocharger and characterization of the turbomachinery performance. The second objective involves an analytical assessment of the transient aero-thermo-rotordynamic coupling during the ignition transient of the micro-gas turbine engine.

The approach that was followed to achieve the first objective involved the experimental identification of governing rotordynamic phenomena, the development of tools to assess and evolve the design, and finally the development of an operational strategy to achieve high speeds based on experimental results, modeling and analysis.

The approach that was followed to achieve the second objective was to develop a discrete time simulation which captures the fluid mechanics, the thermal dynamics, and the rotordynamics which govern the ignition transient of the device. The model employs results from CFD and FEA models, as well as other analytical tools in addition to
experimental results. It accounts for geometric changes of the device during the transient due to structural loadings and thermal expansions. It also captures the effect of the non-adiabatic compressor during the transient and examines how the transient would be affected if the level of heat addition in the compressor was reduced. The model includes the effect of the highly thermally conductive combustor on the combustor exit temperature during the transient. Finally, the modeling concludes by assessing ways of igniting the engine.

1.4 Applications of Research

Microengines have potential to revolutionize portable power generation. If successful, microengines will have superior performance over traditional compact power sources such as batteries and perhaps fuel cells.

All else being equal, the power of a gas turbine engine scales with the square of its size (inlet area), whereas its weight scales with the cube of its size (volume). As a result, in shrinking a gas turbine engine the power to weight ratio is expected to increase. In concept a microengine could produce on the order of 100 times the power to weight ratio of a conventional gas turbine [1]. It should be noted though that such ratios are not currently achievable due to factors that will be discussed later. With an appropriate electrical generator, calculations indicate that the microengine could produce between 10 and 25 Watts of electrical power. This power range is very well suited for portable electronic devices.

When compared to batteries, microengines have a very fundamental and crucial difference. Microengines convert chemical energy of hydrocarbon fuels (∼43.3 MJ/kg) to mechanical and then electrical energy whereas batteries convert chemical energy to electrical energy by reaction process which delivers much less energy per unit weight of the chemical (∼0.5 MJ/kg). Due to the small size of microengines the weight of a full electrical power system is dominated by the weight of the fuel. As a result, for even very low thermal efficiencies such as 5%, the microengine would offer a power density that was 5-25 times better than today’s batteries [1].
1.5 Previous Work

The development of the demo microengine is part of a larger multi-disciplinary effort which is based on seven major areas of research, each of which address a specific aspect of microengine technology. These areas are: (1) systems integration (2) thermal, structural, and material design, (3) combustion, (4) turbomachinery and fluid dynamics, (5) bearing design and rotordynamics, (6) electrical machinery, and (7) fabrication and packaging. The work done in each area is summarized below.

1.5.1 Systems Integration

The concept of the microengine was initially proposed by Prof. A. H. Epstein [1,2]. Groshenry studied the overall system integration and laid the foundation of the demo engine, defining the initial geometry with its gas bearings, radial turbomachinery, integrated combustor, and electrical generator [8]. Esteve studied the role of the secondary flow system on the design and Liu studied the fluid-thermo-structural dynamic response of microengines [9,10].

The baseline design was for a single-spool, single stage engine with a shrouded compressor and an integrated electrical generator/starter. The design had a nominal thickness of 3mm, a die size of 10mm, compressor and turbine diameters of 4mm, and a journal bearing shaft length of 900 μm. There were eight wafers varying in thickness from 200 μm to 800 μm; one each for the fuel manifold, electrical generator/starter, compressor, gas bearing system, turbine, and exhaust nozzle. The combustor was made up of a combination of these wafers. The engine was designed to deliver a core mass flow of 0.15g/sec at a pressure ratio of 4:1, with a compressor tip speed of 500m/sec, a rotor speed of 2.4 million Rpm, and a turbine inlet temperature of 1600K. The material to be used was silicon. The rotor was to be supported radially by an eccentric hydrodynamic journal bearing and axially by hydrodynamic thrust bearings.

Due to its high fabrication complexity the baseline design was simplified by Protz who led the effort of design and initial testing of the first demo engine [11]. The simplified design was built from six 100 mm diameter single crystal silicon wafers and
measured 21x21x3.3 mm. The engine had an 8 mm diameter compressor and a 6 mm
diameter turbine designed to spin at 1.2 million rpm. The compressor and turbine blades
were designed to be 400 μm tall – but were built to 200 μm to ease fabrication
complexity. The compressor design pressure ratio and efficiency were 1.8 and 0.5
respectively. The combustor was to operate at 1600 K. The journal bearing was initially
designed to operate in a hydrodynamic mode but with minor modification was changed to
operate hydrostatically. The engine was designed to ingest 0.36 g/sec of air and have a
thermal efficiency of 2%. The device was spun up to 30,000 rpm.

In an attempt to increase the performance of the engine, Evans and Phillipon
[12,13] did cooling studies where they examined possible schemes that could be used to
cool the turbine and reduce heat transfer between the turbine and the compressor.
Sirakov is currently working on designing an advanced engine that will have substantially
improved performance over the current design [14]. His designs involve further
optimizing the dimensions of each component and possibly including a second spool to
increase the pressure ratio.

1.5.2 Thermal, Structural, and Materials Design

As with conventional gas turbine engines, to achieve high performance levels the
microengine has to operate at high temperatures (1400-2000 K gas and 800-1600 K
structural), high tip speeds (around 500 m/s), and thus high stress levels (100s of MPa).
In addition though, the microengine faces the challenge of being built using
microfabrication techniques. As a result structural and advanced material models had to
be created. Chen did an initial comprehensive study of the materials and structural issues
of the baseline microengine design focusing on silicon as the material of fabrication
[15,16,17,18,19]. Huang and Ye continued Chen’s analysis for the demo engine [20,21].
Lohner studied silicon carbide and other advanced materials and showed that carbide
could be incorporated into a silicon structure [22]. Miller extend Chen’s and Lohner’s
analyses in developing the concept of a hybrid carbide/silicon structure [23]. Moon and
Choi further extended the work on a silicon carbide engine [24, 25]. Finally, Berry has
helped with several specific structural topics in designing the current engine [26].
1.5.3 Combustion

Microengines rely on combustion which requires ingestion of air, fuel injection, fuel-air mixing, and chemical reaction of the fuel-air mixture. One challenge in the combustion process is to have enough time to mix and react the flow before it exits the combustor. While mixing times do decrease with scale, chemical reaction times are independent of scale and as a result micro-scale combustion is a challenging problem. The increase in surface area to volume resulting from reduced scale makes controlling heat loss through the walls another key challenge. This heat loss causes the combustor overall efficiency to be lower than the combustion efficiency itself.

Tzeng and Guaba first demonstrated pre-mixed combustion of hydrogen at micro-engine scales in a centimeter-scale steel and quartz combustor [27,28,29]. Mehra extended these results to a microfabricated silicon combustor that resembled the one used for the demo engine [30,31,32,33,34]. Lee did CFD studies of Mehra’s experiments [35]. Cadou and Spaddacinni extended Mehra’s results to hydrocarbon combustion with and without catalysts [36,37]. Spaddacinni is currently completing research that will help understand the characteristics of combustion with an array of different catalysts. Peck will be continuing Spaddacinni’s work [38].

1.5.4 Turbomachinery and Fluid Dynamics

Being less than one hundredth the size of conventional gas turbine engines, the micro-engine turbomachinery operates in laminar and transitional flow regimes with high heat transfer coefficients. The chord-based Reynolds number for the compressor is approximately 20,000 and for the turbine approximately 5,000, and secondary flow systems are well into the laminar regime (Re<2300). Heat transfer coefficients range from 1000 W/m²/K to 20,000 W/m²/K. In addition, microfabrication constraints only allow two-dimensional features. The turbomachinery was initially studied by Jacobson [39]. Mehra extended Jacobson’s 2D CFD calculations to 3D [40,41]. Gong extended Mehra’s compressor calculations to include the effects of heat transfer [42]. Phillipon
extended Mehra's turbine calculations to include the effects of heat transfer and possible blade cooling [13]. Phillipon also re-designed the turbine for better performance. Jacobson, Shirley, Khan, and Cadou studied micro turbomachinery in a dynamically scaled macro-compressor rig experimental facility [39,43,44,36]. Sirakov re-designed the compressor for better performance and is examining the effects of variable span blades [45]. Such blades were recently shown to be fabricationally feasible by a collaborative research team at the University of Maryland [46].

1.5.5 Bearing Design and Rotor Dynamics

The high angular speeds required for appropriate thermodynamic performance make supporting the rotor a challenging task. The smallest available ball bearings are about the size of the engine, maybe a bit smaller, the issue is that since they are of comparable size, the ball speeds required are vary large resulting in overwhelming friction losses. As a result, it was decided to support the rotor with air bearings. Orr, Piekos, Savoulides, Jacobson, Breuer, Ehrich, Wong, Teo, Liu, Paduano, Brisson, Spakovsky, et al have studied gas bearings. Piekos studied hydrodynamic journal bearings and showed that the rotordynamic stability requires that the rotor be spinning at a distance of approximately 1 μm from the static structure [47,48,49]. Orr demonstrated both hydrostatic and hydrodynamic gas bearings in a dynamically-scaled macro bearing rig [50,51]. Savoulides did analytical studies on hybrid gas bearings [52,53]. Lin, Wong, Frechette, Jacobson, et al. demonstrated high-speed operation of microfabricated bearing test rigs [52,53,54,55]. The highest speed achieved was 1.4 million rpm with a 4 mm diameter rotor. Wong demonstrated operation of hydrodynamic thrust bearings [56]. Teo is doing experimental work that is helping to better understand the governing rotordynamics of the bearing rig [57]. Liu is doing analytical work on hydrostatic gas bearings [58].
1.5.6 Electrical Machinery

Producing electricity from the demo-engine requires an integrated electrical generator. The generator could also be run as a motor to either start the engine or make a compressor run like a pump. Nagle designed and tested a non-rotating electrostatic ‘tethered motor’ to demonstrate microfabricated electrical machinery [59]. Frechette designed and tested a microfabricated motor driven compressor based on the microfabricated bearing device of Lin [52,53,54,55]. Livermore et al. in collaboration with Lincoln Labs developed a metal process, microfabricated electrical generator motor and Steyn is currently testing this device [60]. Koser, Lang, Allen et al. worked on developing a microfabricated electromagnetic motor [61]. Das is further developing the electromagnetic motor design [62].

1.5.7 Fabrication and Packaging

Microengines are MEMS devices built using semiconductor fabrication technology for precision fabrication and batch processing. A number of researchers including Ayon, Zhang, Khanna, Ghodssi, undertook extensive efforts to develop deep etching and bonding processes required to build the first set of demo engines [63-72]. Their work was continued by Li, Ho, Miki, Wang, Ward, who focused on refining some procedures such as the journal bearing etches as well as developing new processes required for the always evolving design.

London developed glass fritt micro packaging for fluidic interconnects that were required in order to operate a hot device [73]. Harrison did further studies of this packaging scheme [74]. Shim is further extending Harrison’s work [75]. Mehra, Lin, Frechette, London, Nagle, Spadaccinni, and Protz also performed considerable process developments for their respective microengine-related devices [76].
1.6 Literature Review

Since the inception of the microengine project several other research institutions have become involved in building micro heat engines that produce electricity in the range of 10mW to 100 Watts. This section provides an overview of some of these projects.

1.6.1 MEMS Gas Turbine Engine – Tohoku University (Japan)

Similar to the MIT micro-engine project, this team is also working on building a MEMS micro gas turbine generator [77]. The efforts of the team are currently focused on designing, building, and testing a turbocharger to characterize the performance of the turbomachinery. Their design also employs a 2-D centrifugal compressor and turbine but contrary to the MIT design has the compressor and turbine placed on the same side of the rotor – with the compressor on the outside. This was done to alleviate some of the challenges associated with the alignment of the two high speed rotors.

Their design is also fabricated from six wafers, two of which are Pyrex glass (the second and fifth levels). The fabrication process relies on deep RIE, anodic bonding, and precision mechanical drilling and wet etching for the glass. For rotordynamic support, the rotor relies on hydrostatic thrust bearings and a hydrostatic journal bearing. This device has been designed and fabricated and testing is underway.

1.6.2 Micromachined Gas Turbine Generator– IHI and Tohoku University (Japan)

This team is working on building a palm sized micro gas turbine engine. Their engine is not fabricated using MEMS techniques; instead it is made by 5-axis micro-milling which produces fully 3-dimensional geometries [78]. The engine consists of a 10mm diameter centrifugal compressor, a combustor and a 10 mm diameter radial inflow turbine. The target cycle calls for a mass flow rate of 2 g/s, a compressor pressure ratio of 3:1, a combustion exit temperature of 1050 °C and is expected to have a thermal efficiency of 6% outputting 100 watts of electric power. The effort has been split into three separate categories. The first is the development of a micro turbocharger with a design rotational speed of 870,000 rpm. The second is the development of a micro combustor which can achieve stable self-sustaining combustion with appropriate thermal
isolation. The third is the development of fabrication techniques for the turbines, which are required to operate at high temperatures.

The turbocharger consists of a compressor and turbine which are separated by a shaft. The entire rotor is made of titanium and both the compressor and turbine are fully 3-dimensional. The shaft relies on herring bone grooves for the journal bearing and a thrust disk with spiral grooves for thrust bearings. This hydrodynamic setup is accompanied by hydrostatic gas bearings that are only used during startup and stopping procedures.

For the combustor, two separate designs were examined: a doughnut shape combustor and a canister shape combustor. The doughnut shape design exhibited a higher heat loss and as a result the canister shape was selected for the baseline design. A 2 cc canister shape combustor was tested with hydrogen and achieved 99.9% efficiency. In addition a larger, 15 cc, version of the combustor was tested with methane and also achieved 99.9% efficiency.

Fabrication advances have also been made. A 3-dimensional silicon nitride impeller has been made by 5-axis micro-milling. Also a method called silicon lost molding has been developed to create 2-dimensional impellers from materials such as silicon carbide which allow for high temperatures. Finally, MEMS fabrication techniques were used to create a 2-dimensional silicon rotor.

1.6.3 Micromachined Gas Turbine – Stanford CA and Honda Japan

This team is also developing the necessary technologies to produce a 100 watt fist sized gas turbine engine [79]. At this stage the team has built and tested a micro-turbocharger with a centrifugal compressor and turbine. The two rotors measure 12mm in diameter and are designed to spin at 800,000 rpm (500 m/s tip speed). They are mounted back-to-back on a shaft which is overhung from a conventional ball bearing.

The rotors are fully 3-dimensional and are made from silicon nitride by employing a Mold SDM process which was developed at Stanford University. The compressor is designed to have a flow rate of 2.38 g/s, a pressure ratio of 3:1 and an adiabatic efficiency of 65%. This device was successfully tested up to 420,000 rpm. Experimental
compressor maps were created and compared positively with CFD predictions. Due to its larger size it was possible to also include thermocouples to deduce efficiency which also compared favorably with CFD predictions. The device was not spun to higher speeds due to rotordynamic instabilities.

1.6.4 Ultra Micro Gas Turbines – University of Tokyo

This team is working on the development of a button size gas turbine engine [80]. Their final engine design would consist of a wave rotor with 8mm diameter centrifugal turbomachinery, a recuperator, and an ultra thin-type generator. As part of the development process the team has initially focused on building and testing of a 10 times-size engine as well as the development of the components of the original size engine.

The team has successfully demonstrated combustion in a canister type combustor for both the original size and 10 times-size combustors using hydrogen as the fuel, achieving 1500K combustor exit temperatures. The team has built and tested 10 times size turbines (3-dimensional and 2-dimensional) and compressors (quasi 2-dimensional). Finally, the team tested the entire 10 times size engine design under ignition conditions but did not achieve self-sustained operation.

1.6.5 MEMS Rotary Engine – UC, Berkeley

This group is working on building a MEMS micro-Wankel heat engine [81]. To prove the basic rotary engine design, the team initially fabricated an engine from steel using EDM techniques at ten times the scale of the MEMS design. This engine had a rotor diameter of 13 mm, an engine depth of 9 mm, a displacement of 348 mm$^3$ and achieved a power output of 4 watts at 9,300 rpm using a H$_2$-air mixture.

The MEMS rotary engine is fabricated from silicon, silicon carbide, and silicon dioxide. Two separate engines are being developed at the UC Berkeley Microfabrication Laboratory: one has a displacement of 0.08 mm$^3$ with a 1mm rotor diameter and the other has a displacement of 1.2 mm$^3$ with a 2.4 mm rotor diameter. The smaller of the two engines was used to develop the basic fabrication processes and determine what the
fabrication tolerances for the larger device would be. The 1 mm rotary engine’s rotor is 300 μm thick and is estimated to operate at a maximum speed of 40,000 rpm producing 10-100 mW. Figure 1-6 and Figure 1-7 depict the steel and the 1 mm diameter MEMS rotary engines respectively.

![Image of steel rotary engine](image)

**Figure 1-6:** Steel 13mm diameter rotor rotary engine. (Courtesy Fernandez-Pello)

![Image of MEMS rotary engine components](image)

**Figure 1-7:** Components of the 1mm MEMS rotary engine. On the left is the rotor and on the right is the rotor housing. (Courtesy Fernandez-Pello)

Fabrication of the engines involves multiple masks, deep reactive ion etching and wafer-to-wafer bonding. All the parts of the engine that are exposed to high temperatures and stresses are built using molded SiC or a Si substrate with a thin SiC coating. An
important aspect of the process is that for final assembly it is necessary to manually assemble the engine.

In order to produce electricity, an electric generator has been designed to be integrated into the engine. The generator uses a 40% - 60% nickel-iron alloy which is electroplated onto the rotor poles so that the rotor itself also acts like the generator rotor. The stator is external to the engine and is comprised of discrete components including a coil and a permanent magnet. This stator design eases the fabrication, improves the efficiency due to the larger volume of copper in the coil and relieves the issue of degrading magnetic properties at high temperatures due to the distance that separates it from the combustion chamber.

1.6.6 MEMS Free-Piston Knock Engine – Honeywell

This group is working on a free piston knock engine generator [82]. It is built using MEMS fabrication techniques and its design power output is 10-50W. The piston return system relies on compressed air which acts like a restoring spring. In order to produce electricity, the piston is magnetic, and is placed within a coil. As the piston oscillates during the thermodynamic cycle it excites the coil producing electricity. This electricity generation scheme is also used to start the engine. Figure 1-8 depicts this engine.

![Figure 1-8: Honeywell knock engine.](image-url)
1.6.7 Other Projects

Several other teams are also working on building devices to produce portable power. Among the teams that are relying on standard thermodynamic cycles combined with electrical generators are Georgia Tech, which is working on a MEMS ceramic laminate free piston generator, the University of Michigan which is developing a swing engine and UCLA which is developing a pulse-jet engine. Other approaches such as thermoelectric microgeneration are also being pursued. USC is working on a project called MicroFire aimed at producing 10s of mW. The University of Michigan is working on the Integrated Micro Combustor - TE Power Generator aimed at producing 1-10µW per thermopile. MIT – Jensen is working on microreactors/reformers.

1.6.8 Advantages / Disadvantages of Other Approaches

All the approaches which rely on conventional micromachining techniques result in superior performance turbomachinery (due to the 3 – dimensionality of the blading), the ability to easily thermally isolate the compressor from the turbine, and the option of including more sophisticated bearing designs. However, the one disadvantage they share is that the cost of building such devices is expected to be much larger than that of building a MEMS device since they cannot be batch processed.

Some design approaches including the single wafer rotor gas turbine might find it challenging to thermally isolate the turbine from the compressor. This could result in reduced performance due to the heat addition during the compression process. A similar situation may also be present for the Wankel engine design. This design however has the advantage that it is manually assembled, something which simplifies the fabrication process but could result in larger costs if the device was mass produced.

1.7 Organization of Thesis

Chapter Two presents the design of the engine and its evolution during the course of this research effort. To list the most important ones: Hydrostatic journal bearing instead of hydrodynamic bearing, compressor and turbine blade re-designs to improve
performance and/or alleviate fabrication problems, exhaust enlargement to accommodate larger flow through turbine, and correction and optimization of several other components. Fabrication challenges and process developments required to incorporate these design changes are also discussed.

Chapter Three describes the rotordynamics of the engine. The discussion is mostly focused on the journal bearing design and fluid dynamic and rotordynamic models that have been created to better understand its operation and aid in its improvement. The thrust bearings are also briefly discussed. A 1st moment rotor imbalance model is also presented as it turns out that unbalance is a crucial parameter for high speed stable operation. Furthermore a 2nd moment rotor imbalance model is presented.

Chapter Four describes the experimental setup including the gas handling system and the data acquisition system.

Chapter Five discusses the rotordynamic experimental results. The journal bearing and thrust bearing natural frequencies are identified and compared to models. Rotor imbalance measurements are also presented. Finally, rotordynamic instabilities are discussed.

Chapter Six discusses the turbomachinery performance measurements. Compressor maps and turbine maps are presented and the spool overall efficiency is deduced from the measurements.

Chapter Seven presents a transient fluid-thermo-rotordynamic model that is used to assess the feasibility of ignition of the gas turbine engine. The model accounts for geometric changes that occur during the ignition process, the effect of the static structure mass on the combustor performance during the ignition transient, the effect of heat addition in the compressor – as well as the effect of reducing it, and finally an assessment of the rotordynamic characteristics during the transient. The model concludes by suggesting feasible strategies for successful ignition.

Chapter Eight discusses operating speed limitations and suggests methods by which to improve the design so as to achieve the design speed.

Chapter Nine concludes the thesis, summarizing the work done to date and suggests area of future focus.
1.8 References


CHAPTER 2

ENGINE DESIGN EVOLUTION AND FABRICATION

This chapter begins by introducing all the fabricated devices relevant to the micro gas turbine engine. Attention then focuses on the turbocharger beginning with a description of its layout and its components. Following that is a discussion of the fabrication challenges and special fabrication techniques that were developed for this device. The chapter goes on to explain the evolution of the design and the key drivers for each design change.

2.1 Fabricated Devices

The current micro gas turbine generator is designed to operate using a Brayton cycle with a pressure ratio of 2 and turbine inlet temperature of 1600 K. It is to have a imbedded micro-generator and is to produce power on the order of 10 watts. Due to its complexity, as discussed in Chapter 1, four separate devices each representing a key engine technological challenge have been under development.

The first device is the turbocharger which is identical to the demo-engine with the exception that the flow path of the compressor and turbine are separate. The objective of this device is to demonstrate high speed operation and characterize the overall spool performance of the demo-engine without having to rely on closed cycle operation. Further tests of the turbocharger will demonstrate the effects of heating the turbine air and eventually the effects of combustion while spinning.

The second device is the micro-combustor which is also identical to the demo-engine with the exception that it has no rotating components. Its objective is to demonstrate combustion. Initial tests focused on burning hydrogen and subsequent tests
used hydrocarbons and hydrocarbons in combination with catalysts. All tests have concluded with successful combustion but all also suggest that a larger combustor volume is required [1].

The third device is the micro-bearing rig which is composed of a single 4mm diameter turbine rotor encased by all the required plumbing for its bearings. The objective of this device is to help understand the rotordynamics which dictate the stability of the rotor. It is also used to test advanced self sustaining bearing systems. Due to its smaller radius, the design speed for this device is 2.4 million rpm and the fastest it has been spun to is 1.4 million rpm (60% of design) [2].

The final device is the turbine-generator which is the same as the bearing rig but also includes a generator. The objective of this device is to produce electricity at high power density, driven by a high speed rotating turbine. This device has been shown to function as a motor. That is, applying electricity to it can change the angular speed of the rotor. However, electrical power has not yet been produced by the device.

Recently, an additional device called the magnetic generator has entered the project. It is currently in the design phase and its objective is to replace the electric generator in an effort to increase the efficiency with which mechanical energy is converted into electrical energy.

### 2.2 Turbocharger Layout

As was shown in Chapter 1, the turbocharger is built from six wafers. One wafer is used for the compressor and another wafer is used for the turbine. Two additional wafers placed on each side of the rotating structure are used for thrust bearings, journal bearing air supply lines and pressure taps. The combustor is located on the turbine wafer. It is an annular volume which encases the turbine. Figure 2-1 shows the cross-section of the most recent design of the turbocharger and labels the most important components.
The rotor has no shaft, so that the compressor and turbine are in direct contact with one another. The entire rotor floats on a cushion of air on the front and aft hydrostatic thrust bearings and the hydrostatic journal bearing, all of which are externally pressurized. The bearing system is shown schematically in Figure 2-2, and will be discussed in more detail in Chapter 3. The compressor diameter is 8 mm and the turbine diameter is 6 mm. The original design blade height for both the compressor and turbine was 400 μm but all devices fabricated to date have 200-250 μm blades. This was done to ease the fabrication process. The tip clearances on both the turbine and compressor blades are 20 μm. For the engine, the turbine is cooled by conducting heat to the compressor. The merit of this cooling scheme is its great simplicity when compared to other macro-scale conventional cooling schemes but one of the major drawbacks is that the heat transfer into the compressor flow deteriorates the compressor’s performance. The spool design temperature is 950K with an estimated heat transfer of 70W.

The annular combustor that encircles the turbine is cooled by a “cooling jacket” that guides the compressor flow to first go around the combustor and then enter it from its backside. The objective of the scheme is to reduce heat loss to the surroundings, and thus increase the efficiency of the device. In the case of the turbocharger, however, the compressor flow exits the device and does not loop around into the combustor. The entrance into the combustor consists of an array of rectangular inlet ports that hold the flame.
To start the device, external air is supplied to the turbine rotor to start spinning. This motion initiates flow through the compressor. In the case of the demo engine this air would be fed to the combustor and then the turbine and under closed cycle operation no additional external turbine air would be required. However, in the case of the turbocharger this air is dumped overboard to measuring devices. Figure 2-3 depicts the difference between the two devices.

![Diagram of turbocharger bearing system](image)

**Figure 2-2:** Schematic of the bearing system of the turbocharger. The rotor floats axially on hydrostatic air thrust bearings and radially on a hydrostatic air journal bearing. All of the bearings are pressurized externally.

![Diagram of demo engine](image)

**Figure 2-3:** Demo Engine: Identical to the turbocharger with the exception that a partition in the fifth wafer which separates the compressor flow from the turbine flow is not present.

In order to characterize the performance of the device several pressure taps are located throughout the turbocharger. The locations of all the taps are indicated in Figure 2-4. On the compressor side, there is a pressure tap located between the compressor blades and the compressor vanes. This tap is referred to as the compressor inter-row tap. Further downstream, there is a tap located after the vanes whose original intent was to add fuel in the demo engine but in the turbocharger is used to measure the pressure of the
flow after it has diffused through the vanes. Finally, there is another pressure tap located at the outmost radial position to account for any additional static pressure increase as the flow experiences vaneless diffusion due to the increasing flow area.

On the turbine side there is a pressure tap that is located at the outmost radial position of the die, where the turbine flow speed is the lowest. This tap measures the pressure of the turbine supply air. Further downstream, there is a turbine interrow pressure tap which measures the pressure between the turbine vanes and the turbine rotor blades. Finally, there is tap located in the exhaust that measures the pressure of the flow exiting the die.

The secondary flow system consists of the thrust bearing supply flows and the journal bearing supply flow. The pressure of the flow entering each of the thrust bearings is measured outside the die. This is feasible because the thrust bearing flow rates are very small and pressure drops occurring between the rig and the thrust bearings themselves are minimal. However, this is not the case for the journal bearing flow. The amount of air required for the journal bearing is substantial and thus there are large pressure drops between the rig and the die. As a result, a pressure tap inside the die is used to measure the pressure of the journal bearing supply flow.

![Turbocharger pressure tap locations](image)

**Figure 2-4:** Turbocharger pressure tap locations

### 2.3 Fabrication

MEMS microfabrication imposed several constraints on the design of the turbocharger. The following sections outline the most major of these constraints as well as some of the special techniques that had to be developed in order to fabricate an engine.
2.3.1 Constraints

Until recently, one of the most important constraints was that only two
dimensional shapes could be etched into the silicon and the only way that three
dimensional features could be created was by stacking multiple silicon wafers onto each
other. This limited what could be built to 3D right-prismatic shapes, which took a toll on
the performance of the turbomachinery and also limited the design. However, this
constraint is no longer present, as a team at the University of Maryland, which is working
in collaboration with MIT, developed a process that allows for tapered features [3]. This
technology has already been used to fabricate tapered compressor blades. Nonetheless,
this technology has not yet been incorporated into the engine.

Bond and mask alignment is another constraint. Masks are aligned to the silicon
to define patterns on both the front and aft side of wafers, which are then bonded to each
other. The alignment involved in each of these steps can be critical, especially in the
steps involved in fabricating the wafers containing the rotating spool. More specifically,
misalignments introduce imbalance, which affects the rotordynamic performance of the
device.

Etch nonuniformity, which refers to the etch depth at different locations on the
wafer, also leads to problems. Generally speaking large areas etch faster than narrow
trenches. The ratio of the etch depths can be as large as a factor of two or three. This
results in the need to avoid etching small and large features together. Furthermore, even
when identical patterns are etched on different locations of the wafer, the etch depth of
each pattern will not be identical due to variations in the plasma strength throughout the
etch chamber. This second type of etch variation is on the order of only a few microns
across the wafer but is critical in determining the imbalance of a rotor. Experiments have
shown that across the diameter of each rotor (8mm) this etch variation can be treated as a
tilted flat plane whose tilt is of order 5 μm from one side of the rotor to the other side.
The result of this is that the rotor's center of mass is moved away from its geometric
center, which has a direct impact on its rotodynamic performance. Figure 2-5 depicts both types of etch non-uniformities.

Figure 2-5: Etch non-uniformity. The top figure depicts the etch depth variation caused by feature size variation. The lower figure depicts the etch depth variation across an entire wafer.

The aspect ratio of deep etches (>100 µm) is also limited by technology. More specifically, the highest aspect ratio etch that has been routinely achieved is roughly 20:1 for a 350 µm deep trench. High aspect ratio etches are critical to the engine as they define the journal bearing which supports the rotor, as will be described in Chapter 3.

2.3.2 Special Fabrication Processes

This section will list several techniques which have been developed to specifically address some of the fabrication challenges that were faced in building the engine.

2.3.2.1 Wide and Narrow Feature Etches – “Halo Etches”

There are some features which require that both large areas as well as small narrow areas be etched at the same time and go to the same depth. As was mentioned in the previous section this is not possible. However, a technique was developed where instead of etching the entire large feature, a cut around narrow etch would be performed

57
[4]. The depth of this etch would be the same as the narrow feature and would cause the large feature to drop out as soon as another etch was performed on the other side of the wafer which would meet with the cut around etch. Figure 2-6 is a schematic of the process which came to be called “halo etching”.

![Figure 2-6: Narrow and large feature etching on the same side of a wafer employ a cut around etching technique which overcomes the problem of size dependent etching rates.](image)

2.3.2.2 Multiple Depth Features – “Nested Masks”

It is possible to etch overlaid features to two different depths on the same side of a wafer through the use of two masking materials which are usually photoresist and silicon dioxide. To achieve this, the first masking material is applied to the wafer and patterned to the shape of the larger feature. The second masking material is then applied to the wafer and patterned to the shape of the feature that is to lie within the larger feature. With the masks in place, an initial etch is performed to some pre-determined depth. The masking material which was used for this etch is then removed and a second etch is performed using the first mask which is still in place. During this etch both the inner and larger outer feature get etched. The result is that the inner feature gets etched deeper than the outer feature. Figure 2-7 depicts this process which came to be known as “nested masking”.
2.3.2.3 Bearing Etches

The bearing etch is possibly the most demanding feature of the engine. It is a 330x16 μm trench which defines the compressor rotor and the radial bearing which supports the spool. Numerical models indicate that this etch has to be done precisely so that stable high rotational speed operation is achievable. More specifically, the etch tolerances are: $330 \pm 5 \mu m$ and $16 \pm 1 \mu m$. The reason for these tolerances will be explained in Chapter 3.

The aspect ratio and tolerances required for the journal bearing etch are well beyond the specification for any available etch tool. An STS Deep Reactive Ion Etcher is used, and processes have been developed that push this tool to the limits of its capabilities. The etch is achieved through a combination of etch recipe and proper masking material. A 2 μm oxide layer under photoresist was found to be necessary for this high aspect ratio deep etch to maintain a sharp inlet profile as well as parallel sides [5]. The mask profile is shown in Figure 2-8.
As can be seen, the oxide is limited to just the area where the bearing is etched instead of being present over the entire wafer. This is done as it was found that 2 μm of oxide present over the entire area of a wafer bowed the wafer substantially (0(20 μm)) due to the mismatch between the thermal expansion coefficient of the silicon and the silicon dioxide. This bow would result in bonding problems later in the process. As a
result, the bearing etch itself requires two masks: one mask to define the oxide ring over the bearing area and another to define the bearing itself.

In building the turbocharger a further complication arises, as another etch has to be done on the same side of the bearing etch but to a smaller depth. This results in the need for two overlaid masks which requires an additional masking material. The bearing itself requires both oxide and photoresist and to achieve the second shallower etch nitride is used as the third masking material. Figure 2-9 illustrates the fabrication process. The oxide mask, which is 0.5 μm thick, is used to define the shallow feature, and the nitride mask is used to define the bearing oxide ring. Once the silicon inside the oxide bearing ring has been exposed 2 μm of thermal oxide are grown. A photoresist mask is then used, as in Figure 2-8, to pattern the bearing.

Once the bearing has been etched 1 μm of thermal oxide is grown in the exposed bearing area to protect it during further processing. The nitride is then removed and the nested mask is etched. The 0.5 μm of oxide on the wafer surface is then removed through a timed BOE etch so that a thin layer of the thicker protective oxide inside the bearing still remains. This protective oxide protects the bearing from damage during the blade etch which occurs later on the other side of the wafer and is removed only upon completion of the build.
2.3.2.4 Rotor Oxide Release Scheme

Upon completion of a build the spool has to be able to freely rotate without making contact with any of the surrounding casing. There is a processing challenge associated with this, however. As the engine is being built it is not possible to completely free the rotor without it falling out of the device. On the other hand if the rotor is physically attached to some point of its casing it will not be possible for it spin when the device is completed.
A solution to this problem was to hold the rotor in place during the fabrication process by bonding it to its casing through a material which could be chemically removed after completion of the device.

As a result, a design which involved oxide spokes on the front thrust bearing pad was developed, as shown in Figure 2-10 [6]. The rotor would be bonded to the casing through the spokes which could be removed upon completion of the device by placing the entire die in a BOE mixture that would consume the oxide spokes, thus releasing the rotor from the casing.

![Figure 2-10: Oxide pad rotor release scheme](image)

### 2.3.3 Manufacturing Yield

The yield of turbocharger devices from the fabrication team has been rather low. This can be attributed to the very challenging design and tight tolerances that are required to achieve high speed operation. More specifically, each device requires many process steps that are complex, and while each step may have a reasonable yield of 80-90%, when there are 20+ complex steps, the yield is quite low. In an effort to minimize possible catastrophic losses, each layer is kept independent as long as possible so that any errors affect only one wafer instead of an entire stack. Despite precautionary actions
such as this one, and many others, the yield has remained quite low and the implications of this have been very serious.

Due to the small size of the turbocharger, instrumentation is rather limited and rotordynamic stability is hard to monitor. When combined with the fact that rotor crashes at speeds over 3% of the design speed are usually catastrophic, it is easy to see how sensitive the success of this research is to the number of devices available to test.

Figure 2-11 depicts the history of the devices that were fabricated and delivered for testing. Above the timeline are the builds that had large fabricational imperfections which caused the devices to be either inoperable or limited their operation to very low speeds. The two most typical fabrication imperfections involved devices that leaked air due to poor bonding and devices that had poorly etched bearings. It should be noted that multiple other builds, not shown in the figure, were also lost due to an array of challenges that were encountered during their fabrication.

Below the timeline are the only two successful builds: Build C with a total of four devices and Build Locos 4 with a total of two devices. It should be noted that even though these builds were the best builds delivered they too did not meet the design specifications as will be discussed in Chapter 5. All of the high speed data that will be presented in this thesis comes from the testing of these six devices.¹

![Figure 2-11: Timeline of devices delivered by fabrication team. Dates indicate delivery of devices. Testing for each build varied from days to several months.](image)

¹ Preliminary tests were also performed on a dual bearing anisotropic design build which was delivered in December of 2003.
2.4 Design Evolution

The engine design has evolved substantially over the course of this research. Some of the changes were driven by fabrication constraints and others by engineering optimization of the device. The next few sections will walk through the most critical design changes that were included in each of the eight builds that were delivered by the fabrication team and include a total of 68 mask changes. Some of the builds are identical to each other as there only four major designs.

2.4.1 Initial Design

This was the final design that Protz [5] tested. The overall architecture of this design was similar to the most recent design, however several crucial features were different. The rotor was supported axially by hydrostatic thrust bearings that were identical to the ones in the current design. However, the journal bearing was on the turbine rim instead of the compressor rim and was operated in a hydrostatic mode which relied on grossly undersized supply lines. This was due to the fact that the device was originally designed to operate on a hydrodynamic journal bearing which would require much lower flow rates. Furthermore, the journal supply flow was not prevented from entering the much larger compressor gap (see Figure 2-12). This made it impossible to create large pressure differentials across the bearing and also did not allow for precise measurement of the bearing supply pressure.

From the fabrication perspective, the process flow was similar to the one currently employed, yet there were some key differences that will be pointed out as each major design is discussed. The original process flow is listed below and reference should be made to Figure 2-12 to understand the terminology used.

1. Bond compressor wafer with etched compressor blades to FTB wafer
2. Etch the backside of the compressor wafer, including compressor tip gap
3. Bond turbine wafer with etched bearing to compressor wafer
4. Etch the backside of the turbine wafer, (i.e. the turbine blades)
5. Bond the ATB wafer to the turbine wafer backside
6. Bond the two end plates to the four wafer stack

Figure 2-12: Initial design of the turbocharger. The device relied on a hydrostatic turbine journal bearing and there was no seal present to prevent the flow from entering the compressor gap.

This design was spun by Protz to 30,000 rpm and an identical build of engines (Build 9) with no changes from the initial design was delivered and tested during this research but with limited success due to the inadequate hydrostatic journal air supply system and also due to poorly fabricated bearings which were less than one half of the design length.
2.4.2 1st Re-Design (Builds A, B and C)

Several modifications were made in this design but the fabrication process flow was not altered. Figure 2-13 depicts the cross section of the design and Table 2-1 lists the changes that are described in the following sub-sections.

![Diagram](image)

**Figure 2-13:** 1st Re-Design. Changes included a new hydrostatic air delivery system, the addition of a seal to prevent journal flow from entering the compressor gap, and several other minor changes.
Table 2-1: 1st Re-Design Changes

<table>
<thead>
<tr>
<th>Modification # (Figure 2-13)</th>
<th>Modification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Addition of Hydrostatic Seal</td>
</tr>
<tr>
<td>2</td>
<td>Hydrostatic Piping Enlargement and Pressure Tap Addition</td>
</tr>
<tr>
<td>3</td>
<td>Other (leak path removed, compressor hub diameter reduced...)</td>
</tr>
</tbody>
</table>

2.4.2.1 Hydrostatic Seal

In order to reduce the mass flow rate of hydrostatic air that was supplied to the journal bearing system, and thus the pressure needed to deliver this flow, a seal was added to divert most of the flow that was originally going into the compressor gap to the turbine bearing. The seal was placed at a radius of 3.95mm and its clearance was set to 14 µm during fabrication and would become 11 µm when the rotor was centered axially. This clearance was picked so that when the rotor tipped to the side it would first touch the thrust bearing pads rather than the seal. In this way the operation of the thrust bearings would not be affected by the presence of the seal. The width of the seal was set to 100 µm. A wider seal would have been more effective in reducing the leakage flow rate but would also have had a higher power loss associated with it. The drag associated with the seal was computed under the assumption of simple Couette flow, and was found to be approximately 1 watt for cold operation and 2 watt for hot operation at the design speed. The leakage flow for this seal was also computed with a flow model that will be presented in Chapter 3 and was found to be slightly larger than the bearing flow with approximately 45% the total journal bearing supply flow going to the bearing and 55% going to the seal. This is a large improvement over the previous design in which roughly 5 times the journal bearing flow was supplied due to the leakage through the compressor tip gap.
2.4.2.2 Hydrostatic Piping Enlargement and Pressure Tap Addition

As was mentioned earlier, the hydrostatic system did not deliver the flow required by the journal bearing and the addition of the seal improved the problem by only a small fraction. To remedy this, the supply system was re-designed so that the pressures used to drive the flow stayed within the structural limits of the device. As an aide in this redesign, a resistor network model was created to assess the pressure drop through the hydrostatic supply line. In this re-design, changes were limited to only the aft thrust bearing wafer but in a later design, the piping in all the wafers was changed. Both the model and results of the changes are discussed in the next re-design section. Despite the decreased pressure loss in the supply system, a hydrostatic pressure tap was also added within the die to precisely measure the pressure of the flow being supplied to bearing as it was a critical parameter in better understanding its operation.

2.4.2.3 Other Modifications

There were several other minor modifications to the design such as the removal of a leak path which shorted the turbine supply air to the exhaust pressure tap. Another change was that the compressor hub diameter was decreased as in the previous design it slightly protruded over the turbine rim and could potentially cause the journal bearing to crash.

2.4.3 2nd Re-Design (Build D1, D3, D4)

This design was focused on realizing a device with two journal bearings, one on the compressor rim and one on the turbine rim. The reason for this was that two bearings would offer a higher stiffness and under the assumption of no misalignment between the compressor and turbine wafers, would also offer a higher stability boundary (for more details see Chapter 3). The fabrication process flow had to also be modified in this design.
**Figure 2-14:** 2nd Re-Design. Changes include going to a dual bearing design which required a modification in the fabrication process flow, further enlargement of the hydrostatic supply line, removal of the compressor vane tip clearance, the addition of an igniter port, and the addition of a feature which could be used to estimate the bond alignment. Furthermore, the turbine blade design was revised, and the exhaust was enlarged. Finally, the die was also enlarged.
### Table 2-2: 2nd Re-Design Changes

<table>
<thead>
<tr>
<th>Modification # (Figure 2-14)</th>
<th>Modification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Dual bearing (addition of compressor bearing)</td>
</tr>
<tr>
<td>2</td>
<td>Further Hydrostatic Piping Enlargement</td>
</tr>
<tr>
<td>3</td>
<td>Compressor Vane Tip Clearance Removal</td>
</tr>
<tr>
<td>4</td>
<td>Igniter Port Addition</td>
</tr>
<tr>
<td>5</td>
<td>Revised Turbine &amp; Enlarged Exhaust</td>
</tr>
<tr>
<td>6</td>
<td>Die Enlargement</td>
</tr>
<tr>
<td>-</td>
<td>Addition of Bond Alignment Feature</td>
</tr>
</tbody>
</table>

#### 2.4.3.1 Process Flow Modification

In the previous designs, the compressor rotor was first bonded to the front thrust bearing wafer and was released from the surrounding casing by performing the etch which defined the hydrostatic plenum (the 80 \( \mu \)m etch between the compressor rotor and the turbine static structure). This etch would cause the compressor gap etch to go deeper into the wafer until it met with the compressor blade etch, thus releasing the rotor.

In the new design however, the compressor gap was narrowed from 25 \( \mu \)m to 15 \( \mu \)m so as to become a bearing. As was discussed in previous sections, narrow features etch much slower than wider features. As a result, the hydrostatic plenum etch would not be able to cause the compressor bearing etch to go much deeper into the wafer and thus it would be hard to control the release of the rotor while maintaining the depth of the hydrostatic plenum close 80 \( \mu \)m.

To solve this problem, the process flow had to be modified so that the turbine and compressor wafers were first bonded to each other and the compressor rotor was released during the compressor blade etch. This two stack was then bonded to the FTB wafer and following this step, the turbine rotor would be released by performing the turbine blade etch. The following list outlines the procedure:
1. Bond compressor wafer with etched backside to turbine wafer with etched bearing
2. Etch compressor blades and bond to FTB wafer
3. Etch turbine blades
4. Bond the remaining wafers

There was one challenge with this process however, and that was that it involved bonding wafers with enclosed cavities. This is an undesirable situation as the bonding process involves placing the contacted wafers in a high temperature oven in order to anneal them. If gas is entrapped in any cavity, increasing temperature in the oven results in an increase in the pressure in the constant volume cavity, which can lead to delimitation during the anneal process. The solution to this problem was to perform the bond in a very low pressure environment so that very little gas would be entrapped inside the cavities. Several experiments were performed to confirm this approach and a single turbine bearing build (Build D1) was first built to demonstrate that it was feasible.

2.4.3.2 Further Hydrostatic Piping Enlargement

The hydrostatic supply lines were further enlarged and the diameter of the connection port on the front side of the device was also increased. This was done to further reduce pressure drops in the hydrostatic supply line. The same model which was discussed earlier was employed to perform these changes and is discussed in detail in the following section.

2.4.3.2.1 Hydrostatic Piping Pressure Loss Model

The objective of the model was to determine where most of the pressure drop was occurring and what design changes would have the greatest impact in reducing the pressure loss while requiring the least number of mask changes. The model was based on the assumption of incompressible isothermal flow. The hydrostatic supply line was then broken into multiple pieces as shown in Figure 2-15 representing each section of the system.
Figure 2-15: Hydrostatic journal supply resistor network model. Numbers refer to stations used in the model representative of the device geometry. 1-2 are in the front end plate, 3-4 are in the FTB wafer, 5-6 are in the compressor and turbine wafers, 7-12 are in the ATB wafer and 12-15 are in the hollowed turbine vanes.

To determine the pressure loss along each section, the Darcy-Wiechab relation was used:

\[ P_1 - P_2 = \zeta \left( \frac{1}{2} \rho U^2 \right) \]  
Eq. 2-1

where \( \zeta \) is the pressure loss coefficient specific to the geometry, \( \rho \) is the density of the fluid and \( U \) is the velocity of the fluid. For duct flow \( \zeta \) is determined using the following relation:

\[ \zeta = \frac{fL}{D_h} \]  
Eq. 2-2

where \( f \) is the Moody friction factor, \( L \) is the length of the duct and \( D_h \) is the hydraulic diameter of the duct. For entrance and corner losses the value of \( \zeta \) is constant and depends on the specific geometry. For more details refer to Idelchik [7].
The ducts in the initial design were so small that delivery of the required flow would choke the system. However, once the regions with the largest area constrictions were identified they were enlarged until velocities were small enough so that compressibility effects were negligible, and the model was developed assuming incompressible flow. At that point, the model was used to identify the locations of the largest pressure drops and the design was optimized so as to minimize the pressure drops in the entire hydrostatic supply system.

The following table summarizes the losses along the network for the modified designs for the case that the flow is going into a typical journal bearing at 34 kPa (5 psig), at a flow rate of 0.039 g/s (2000 sccm). It should be noted that prior to these changes, pressures greater than 350 kPa were required to deliver the same flow to the bearing.

Table 2-3: Hydrostatic supply line pressure drop model (at 0.039 g/s - 2000sccm)

<table>
<thead>
<tr>
<th>Location</th>
<th>1st Re-Design</th>
<th>2nd Re-Design</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>% Total Drop</td>
<td>% Total Drop</td>
</tr>
<tr>
<td>14-15</td>
<td>3.5%</td>
<td>16.9%</td>
</tr>
<tr>
<td>13-14</td>
<td>1.1%</td>
<td>5.3%</td>
</tr>
<tr>
<td>12-13</td>
<td>3.5%</td>
<td>16.9%</td>
</tr>
<tr>
<td>11-12</td>
<td>2.9%</td>
<td>8.0%</td>
</tr>
<tr>
<td>10-11</td>
<td>12.0%</td>
<td>4.3%</td>
</tr>
<tr>
<td>9-10</td>
<td>3.9%</td>
<td>8.1%</td>
</tr>
<tr>
<td>8-9</td>
<td>13.0%</td>
<td>18.6%</td>
</tr>
<tr>
<td>7-8</td>
<td>1.2%</td>
<td>1.6%</td>
</tr>
<tr>
<td>6-7</td>
<td>4.3%</td>
<td>3.7%</td>
</tr>
<tr>
<td>5-6</td>
<td>0.2%</td>
<td>0.3%</td>
</tr>
<tr>
<td>4-5</td>
<td>42.7%</td>
<td>12.0%</td>
</tr>
<tr>
<td>3-4</td>
<td>4.0%</td>
<td>2.6%</td>
</tr>
<tr>
<td>2-3</td>
<td>7.7%</td>
<td>1.5%</td>
</tr>
<tr>
<td>1-2</td>
<td>0.0%</td>
<td>0.0%</td>
</tr>
<tr>
<td>Package (experimental)</td>
<td>+ 31%</td>
<td>~ 0%</td>
</tr>
<tr>
<td>Total Press. Drop (psi)</td>
<td>50.0 kPa</td>
<td>7.93 kPa</td>
</tr>
</tbody>
</table>
The first re-design involved only changes on the ATB wafer whereas the second re-design involved changes on all of the wafers. A region of very large change from the initial design to the first re-design was the 9-10 duct whose area was increased from 0.10mm$^2$ to 0.36mm$^2$ and whose length was reduced from 35mm to 5.3mm. This resulted in much lower velocities through the duct as well as a large reduction in zeta. In the second re-design, the largest change was the enlargement of ducts 3-6. This involved changing the masks for multiple layers in the device and that is why it was not done in the initial re-design. Figure 2-16 depicts experimental measurements of the pressure drop in the hydrostatic supply lines for the original design, the first re-design, the second re-design and the model predictions.

**Figure 2-16:** Pressure drops in hydrostatic supply system, measured experimentally. The black line is based on the original design, the blue line is the result of the first re-design and the red line is the result of the second re-design.
Examining the experimental results it can be seen that the model under predicted the pressure losses. One reason for this is that the model has some assumptions which are not strictly true. This is especially true for some of the corners which are very close to each other causing the assumption of fully developed flow prior to the corner to not hold any more. In any case, the principal objective of the model was to determine how to improve the design by pinpointing to where the highest pressure drops were occurring and not to actually predict the exact pressure drop of the flow path.

2.4.3.3 Compressor Vane Tip Clearance Removal

Initially, the compressor vanes had a tip clearance, for thermal isolation, so as to avoid heat conduction from the static structure surrounding the combustor to the compressor tip casing. However, packaging included a 2.4mm O-Ring on the compressor inlet. This O-Ring applied a force on the forward die surface which, due to the vane tip clearance, was supported at a radius of 10mm. The result of this was that the die front surface would deflect, closing down on the thrust bearing clearance. This was an unwanted effect as the thrust bearing operation is sensitive to the total axial clearance, 6 μm as designed.

To assess the impact of package clamping on the internal clearances, a simple structural model was put together. The overhung silicon area was modeled as a flat disk that was clamped at a radius of 10mm and exposed to a ring pressure load at a radius of 2.4mm. The overhung area consisted of two wafers one of which had a large fraction of its area etched away. As a result, the model examined two cases. In one case the deflection was based on the stiffness of just one wafer and in the other case the deflection was based on the stiffness of both wafers. Also, the effect of varying the first wafer thickness was examined to see the benefit of using a thicker wafer – its nominal thickness is 450 μm.

To determine the pressure load that the O-Ring would exert, data from the O-Ring manufacturer was collected and is depicted in Figure 2-17:
By means of linear extrapolation the above data was used to estimate the force per unit length of O-Ring for the O-Ring used for the compressor inlet, which had a cross-sectional thickness of 0.406mm (0.016inches) as do all O-Rings used to make the fluidic connections to the turbocharger ports. Table 2-4 depicts the results.

Table 2-4: Extrapolation of force per unit length for 0.406mm (0.016inch) O-Ring

<table>
<thead>
<tr>
<th>Compression</th>
<th>Force per Unit Length (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10 %</td>
<td>0.1321</td>
</tr>
<tr>
<td>20 %</td>
<td>0.3697</td>
</tr>
<tr>
<td>30 %</td>
<td>0.7944</td>
</tr>
<tr>
<td>40 %</td>
<td>1.5379</td>
</tr>
</tbody>
</table>

By employing the above numbers and the structural relations governing this problem (Roark’s Formulas for Stress and Strain), Figure 2-18 was created.
Figure 2-18: Deflection at center of device due to O-Ring compressive force. (a) Assumes that the second wafer does not contribute to the stiffness. (b) Assumes that second wafer fully contributes to the stiffness.

As can be seen the models predict that the deflections are substantial when compared to the thrust bearing clearance of 6 μm. In the best case scenario, with the second wafer fully contributing to the stiffness, as if it were not etched, and an O-Ring
compression ratio of 20%, the predicted deflection is about 1 μm. This level is unacceptable given that the tolerance of the thrust bearing clearance is ±0.25 μm, as will be discussed in Chapter 3. Even thickening the first wafer is clearly not enough when considering that design O-Ring compression levels are close to 30% and the second wafer has large areas of silicon removed.

This analysis was further confirmed through experimental evidence of the thrust bearing flow rates which appeared to be different depending on whether the O-Ring was present or not. Figure 2-19 depicts the results of such a test. A thrust bearing flow model showed that the changes in the flow were consistent with the model that assumed that the second wafer was fully contributing to the stiffness.

![Figure 2-19: Effect of center O-Ring on thrust bearing flow rate. The O-Ring appears to decrease the flow through the thrust bearing.](image)

Based on all of the above information it was decided that the tip clearance over the vanes would be removed.

---

2 The nozzles for the thrust bearings of this device were smaller than the nominal size exhibiting flow rates that were less than half of the nominal value.
2.4.3.4 Igniter Port

An igniter port was added to the device so as to give us the capability of burning. The igniter port was designed to match a glass frit that was used for the igniter in the micro-combustor device. The igniter itself is a thin wire to which voltage is applied causing it to heat up thus causing the combustor flow to ignite.

2.4.3.5 Bond Alignment Feature

Based on an imbalance model it was determined that the largest source of imbalance that was not measured during the fabrication process was the bond alignment. As a result, a bond alignment feature was introduced which could be used to measure the bond alignment after the compressor and turbine wafers had been bonded. This was achieved by etching a feature in the compressor using the compressor gap mask and by etching a negative mold of that feature in the turbine bearing mask. With an additional “window” etch during the turbine blade etch it was possible to determine the bond alignment by examining how centered the compressor feature was with the respect to the turbine feature. Figure 2-20 depicts this feature. Unfortunately this scheme was not successful as the feature that was etched during the compressor bearing/gap etch was completely etched away.
2.4.3.6 Revised Turbine and Enlarged Exhaust

A revised turbine airfoil geometry was introduced by Philippon to increase turbine power and efficiency. The largest part of the modification was to move the turbine blades to a larger radius. Unfortunately, the blades were placed too close to the vanes and this resulted in poor local etch uniformities which affected the release of the rotor and thus was a cause for shorter bearings than designed. This problem was remedied in a later design by slightly altering the shape of the turbine vanes, and was assumed to not affect the performance. Table 2-5 summarizes the performance of both the original and revised designs for 400 μm blades.

The exhaust size was also enlarged so as to match the larger turbine. The new exhaust inner diameter remained unchanged at 2mm and its outer diameter was enlarged from 3.1mm to 4mm.
Figure 2-21: 1st Turbine blade design change. Left picture is original blade design and picture on right is the improved design.

Table 2-5: Calculated performance of original and improved turbine designs

<table>
<thead>
<tr>
<th>Metrics</th>
<th>Baseline Stage</th>
<th>Improved Stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stage pressure ratio</td>
<td>1.65</td>
<td>2.1</td>
</tr>
<tr>
<td>Mass flow (kg/s)</td>
<td>2.28E-4</td>
<td>2.93E-4</td>
</tr>
<tr>
<td>Shaft work (W)</td>
<td>15</td>
<td>49</td>
</tr>
<tr>
<td>Stage efficiency</td>
<td>0.3</td>
<td>0.54</td>
</tr>
<tr>
<td>Reaction</td>
<td>0.27</td>
<td>0.13</td>
</tr>
</tbody>
</table>

2.4.3.7 Enlarged Die Size

The die size was increased from 21x21mm to 23x23mm because several dies had been damaged during the die saw process by cutting into plena which were within 100 μm of the die saw line. Since the process had moved to 4 dies per wafer, enlarging the dies had no negative impacts. In conjunction with the modification of the hydrostatic connection port, this change made it necessary to build a new package for the device.
2.4.4 3rd Re-Design (Locos 4)

The major change in this design was the optimization of the journal bearing design. This design removed the bearing from the turbine and optimized the journal on the compressor to obtain the highest possible stability boundary. This was done based on the results from the analysis performed by Liu that will be discussed in detail in Chapter 3. The reason why the design reverted back to a single bearing was to simplify the fabrication process and also because it was believed that it would be easier to characterize the rotordynamic performance of a single bearing device due to its simpler geometry. Figure 2-22 depicts the cross section of this design. Achieving this design also required some modifications in the process flow.

Figure 2-22: Build Locos 4
Table 2-6: 3rd Re-Design Changes

<table>
<thead>
<tr>
<th>Modification # (Figure 2-22)</th>
<th>Modification</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Optimized Single Compressor Bearing (&amp; Removed Turb. Bearing)</td>
</tr>
<tr>
<td>2</td>
<td>Hydrostatic Plenum Moved to Turbine Wafer</td>
</tr>
<tr>
<td>3</td>
<td>Revised Compressor Blades and Fiber Optic Tabs</td>
</tr>
<tr>
<td>4</td>
<td>Hydrostatic Seal Addition</td>
</tr>
<tr>
<td>5</td>
<td>Turbine Blade Modification</td>
</tr>
<tr>
<td>-</td>
<td>Journal Bearing Anisotropy Addition</td>
</tr>
</tbody>
</table>

2.4.4.1 Process Flow Modification and Movement of Hydrostatic Plenum

Based on the journal bearing modeling efforts it was concluded that the bearing(s) had to be built to within very tight tolerances. This requirement led to the development of the bearing etching procedure that was outlined in an earlier section which requires a total of three masking materials [Li]. Furthermore, the hydrostatic plenum etch was moved from the compressor rotor to the turbine static structure as can be seen in Figure 2-22. This was done to avoid damaging the compressor bearing entrance region while performing this etch. An additional benefit of this change was that it removed a possible source of imbalance from the rotor.

2.4.4.2 Anistoropic Journal Bearings

Further analysis from the bearing modeling effort [Liu] suggested that anisotropy has favorable effects on the stability of the bearing (see Chapter 3 for more details). More specifically, the model suggests that a different direct stiffness in each principal direction can move the stability boundary to higher speeds. In this design this was achieved by creating an asymmetry in the hydrostatic supply system by blocking four of the eight feed lines.
2.4.4.3 Revised Compressor Blades and Fiber Optic Tabs

The compressor blades were re-designed so as to increase performance. The new design would offer a higher pressure ratio, higher efficiency, and higher flow rates for the same blade height. The aerodynamic design was done by Sirakov.

In addition, this opportunity was used to also improve the geometry of the tabs used to measure the speed of the rotor, as they are defined by the same mask which is used to pattern the blades. The edges of the tabs were made radial lines so as to improve the precision of the fiber optic measurements. The new tabs also contained a feature which could be used to extract phase information from the fiber data. The feature itself introduced an imbalance to the rotor but it was positioned on each die so as to offset a portion of the imbalance that was introduced by the radial non-uniformity of the STS DRIE (this will be better explained in Chapter 3). Figure 2-23 depicts the changes and Table 2-7 summarizes the performance of both designs [Sirakov].

\textbf{Figure 2-23:} Compressor blade and fiber optic tab re-design. Original design is picture on left and new design is picture on right. Compressor performance improved and tab design made asymmetric for possible extraction of phase information by fiber optic.
Table 2-7: Performance of original and improved compressor designs

<table>
<thead>
<tr>
<th>Metrics</th>
<th>Baseline Stage (200 μm blades)</th>
<th>Improved Stage (220 μm blades)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stage pressure ratio</td>
<td>2.1 (1.54)</td>
<td>3.0 (2.5)</td>
</tr>
<tr>
<td>Mass flow (g/s)</td>
<td>0.16 (0.95)</td>
<td>0.39 (0.31)</td>
</tr>
<tr>
<td>Stage efficiency</td>
<td>0.35 (0.17)</td>
<td>0.50 (0.38)</td>
</tr>
</tbody>
</table>

( ) Includes effect of heat addition at a wall temperature of 800K

This blade re-design not only changed the performance of the compressor but also changed the deformations the rotor would experience due to centrifugal loading. More specifically, as the rotor speed increases, the compressor and turbine expand radially and also curve downwards toward the aft thrust bearing wafer. Figure 2-24 depicts this schematically and Table 2-8 lists the predicted values of the deformations based on an FEA analysis [Berry].

Figure 2-24: Rotor deformation schematic due to centrifugal loading

Table 2-8: Rotor deformations based on both compressor blade designs

<table>
<thead>
<tr>
<th></th>
<th>Old Compressor</th>
<th>New Compressor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp. Tip Radial Expansion (μm)</td>
<td>3.4</td>
<td>3</td>
</tr>
<tr>
<td>Comp. Tip Axial Deflection (μm)</td>
<td>-13.4</td>
<td>-7.8</td>
</tr>
<tr>
<td>Turb. Tip Radial Expansion (μm)</td>
<td>1.2</td>
<td>1.5</td>
</tr>
<tr>
<td>Turb. Tip Axial Deflection (μm)</td>
<td>-7.2</td>
<td>-4.0</td>
</tr>
</tbody>
</table>

Courtesy Shawn Berry.
As can be seen the rotor appears to deform much less with the new compressor blade design. This is attributed to the reduced mass of the new blades which results in a reduced bending torque on the rotor while the rotor spins. This is another advantage of the new design as smaller rotor deformations simplify the bearing design and also allow for smaller seal clearances.

2.4.4.4 Hydrostatic Seal Addition

With the new design relying on a single compressor journal bearing, a seal had to be inserted to prevent the majority of the bearing supply flow from going into the turbine gap – similar to the seal that was present in the single turbine bearing designs. The radial location of this seal was a trade however. Placing the seal at the smallest possible radius – right next to the turbine – would have multiple advantages. First, for a given amount of leakage, it would be consuming less power due to the lower local speed of the compressor. Second, it could have a smaller clearance as for the same rotor angular motion the local geometric motion would be smaller – this could further reduce leakages. The final advantage is that the leakage would also be reduced by the fact that the circumference of the seal exposed to the high pressure would be smaller.

Placing the seal at small radius however, increased the area over which the hydrostatic supply plenum was forcing the rotor forward and thus more adjustments would have to be made to maintain an axially centered rotor. It was decided that having a lower thrust balance sensitivity to hydrostatic supply pressure changes was more important than all of the advantages discussed and as a result the seal was placed at the largest possible radius (3.39mm).

As the seal radius was smaller than the one used for the single turbine bearing device, and would thus be exposed to a lower local compressor speed, its width was enlarged to 200 µm. To determine the seal clearance, the geometric deformations of the rotor due to centrifugal loading were accounted for this time. With the new compressor blades it was estimated that the rotor would deflect downwards by approximately 5.3 µm at the seal radius when spinning at the design speed. As a result, the fabrication clearance was set to 11 µm which would become 8 µm when the rotor was centered axially. At
design speed, this would allow for all the rotor structural deformations as well as rotor tilting by approximately half of the available thrust bearing angular clearance. At low speeds where the rotor was not structurally deformed the rotor thrust bearing’s would touch before the seal touched. Figure 2-25 summarizes the variation of seal clearance and leakage flow with speed. A model for the leakage flow will be discussed in Chapter 3.

![Graph of Seal Clearance Variation with Speed](image)

![Graph of Leakage Flow Variation with Speed](image)

**Figure 2-25:** (a) Variation of seal clearance with speed. (b) Variation of seal leakage flow with speed (at 34.4 kPa (5psi) supply pressure - journal bearing flow rate at this pressure is 52mg/s (2600sccm)).

88
As can be seen the clearance of the seal varies quadratically with the speed and as a result most of the change in the clearance occurs at the higher speeds. For the seal leakage plot it was assumed that the bearing supply pressure was 34.4 kPa (5psi). For this supply pressure the designed journal bearing is predicted to have a flow rate of 52 mg/s (2600 sccm). This means that the leakage flow varies from roughly 18% to 1% of the total journal bearing flow as the speed is increased from zero to the design value.

2.4.4.5 Revised Turbine

In response to the problems encountered in the previous design, the vanes of the turbine were moved out radially to increase the vane-blade gap and thus improve etch uniformity. This was a necessary change in order to guarantee a controlled release of the rotor and thus a correct bearing length, and was not expected to have a large impact on the turbomachinery performance.

Figure 2-26: 2\textsuperscript{nd} Turbine blade design change. Only the vane design was altered so as to increase the space between the rotor blade leading edges and the vane trailing edges for fabrication purposes.
2.5 Summary

This chapter first discussed the four different types of devices that have been fabricated during the engine development process – the micro turbocharger, the micro combustor, the micro bearing rig, and the micro electrical generator. The turbocharger layout was then presented followed by a discussion of the fabrication innovations and challenges associated with building it.

A total of four separate designs were made built and tested. Each design had multiple variations from the previous design and only two of the fabricated designs were successful in achieving high speed operation. The first design was adopted from Protz and relied on a single turbine journal bearing with an undersized hydrostatic supply system. The second design differed in that the hydrostatic supply system was enlarged to allow for the required flow to be fed to the bearing. The third design added a second journal bearing that was on the compressor. Finally, the fourth design had a single anisotropic compressor bearing.

Other changes in each design included the addition of an igniter port, a bond alignment feature, enlargement of the die size from 21x21mm to 23x23mm, and the removal of the vane tip clearance. Furthermore, both the compressor and turbine blades were modified, along with the exhaust in an effort to improve the turbomachinery performance of the device. Finally, accompanying these changes were several modifications to the fabrication process in an effort to achieve the required design tolerances.

2.6 References


Rotordynamics have proven to be one of the most challenging issues in the design and testing of the microengine. This chapter will introduce the unique geometry of the turbocharger rotor and will then give a quick overview of rotordynamics. Then the discussion will focus on the journal bearing design and the key issues that affect its performance. Following that, models used to evaluate both the static and dynamic geometry of a given device will be presented. Then, a model used to design an anisotropic bearing flow system will be discussed. Finally, an analysis which sets the fabrication requirements needed to meet the rotordynamic performance objectives will be presented.

3.1 Engine Rotordynamics Overview

The performance of a Brayton cycle is largely dictated by the compressor pressure ratio and the turbine inlet temperature. The turbine inlet temperature is limited by the thermal capabilities of silicon and the cooling scheme employed. The compressor pressure ratio per stage scales with the compressor tip speed squared, and the maximum tip speed is set by structural limitations. Multi-staging is possible in theory but was not pursued due to complexity. The turbocharger was designed with a compressor rotating at a tip speed of 500 m/s, which translates to 1.2 million rpm for a 4 mm radius compressor.

Supporting a 4 mm radius rotor rotating at 1.2 million rpm is challenging. Conventional ball bearings that approach such angular speeds exist, however the losses associated with them (due to the large surface area of the balls moving so fast) are
unacceptably high for this device. As a result, the rotor was designed to operate on air bearings.

Several other high speed devices using air bearings have been designed and tested at MIT over the past few years, however the turbocharger is unique because its rotating components are comprised of two instead of one wafer. These wafers are bonded together with a random misalignment of order a few microns, set by the alignment tool. Another difference the turbocharger has from all other devices is that it includes both a turbine and a compressor. Previous devices had one or the other. Figure 3-1 is a depiction of the turbocharger rotor.

Figure 3-1: 3D schematic of turbocharger rotor
3.1.1 Thrust Bearings

For axial support the turbocharger is supported by two hydrostatic thrust bearings. The thrust bearings are comprised of multiple restrictors (twenty for the current design) which are symmetrically positioned at a fixed radius on the non-rotating side. They are externally pressurized and all feed a thin film of air which supports the rotor against the static pad. The gap separating the two acts as a variable resistance flow path while the restrictors act as fixed resistances. When the gap narrows its flow resistance increases, reducing the mass flow rate. As a result, for a fixed supply pressure to the restrictors, the pressure at the restrictor exit increases as the gap narrows, giving rise to axial stiffness. Figure 3-2 is a schematic indicating this effect. The stiffness in the figure can be estimated by integrating the pressure change over the surface area of the pad and dividing by the axial motion that caused the change.

A major design requirement for the thrust bearings is that the axial natural frequency be above the design speed. Furthermore, the thrust bearings must be able to support axial loads that arise from thrust balance mismatches during device operation\(^1\). A thrust bearing design that maximizes the natural frequency does not necessarily maximize its load carrying capability. The thrust bearings were thus designed as a compromise between maximum stiffness and maximum load carrying capability [1]. Table 3-1 lists the design parameters of the thrust bearings.

---

\(^1\) Note that the original design intent for the axial support system was that at 100% design operating speed, the thrust bearings not support any axial load, which would be the job of the thrust balance piston. As the design evolved however, the thrust balance piston became part of the hydrostatic journal supply system and without any further design changes the two are currently coupled. This makes the load carrying capability of the thrust bearings an important characteristic of the system.
Figure 3-2: Hydrostatic thrust bearing operation

Table 3-1: Thrust bearing design

<table>
<thead>
<tr>
<th>Design Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Nozzles</td>
</tr>
<tr>
<td>Nozzle Location Radius</td>
</tr>
<tr>
<td>Pad Radius</td>
</tr>
<tr>
<td>Nozzle Diameter</td>
</tr>
<tr>
<td>Nozzle Length</td>
</tr>
<tr>
<td>Total Gap</td>
</tr>
<tr>
<td>Total Flow Rate (@ 827 kPa - 120psi)</td>
</tr>
<tr>
<td>Natural Frequency (@ 827 kPa - 120psi)</td>
</tr>
<tr>
<td>Load Capacity (@ 827 kPa – 120 psi &amp;</td>
</tr>
<tr>
<td>Eccentricity ratio = 0.4)</td>
</tr>
<tr>
<td>Pressure at which nozzles choke (ecc. ratio = 0)</td>
</tr>
</tbody>
</table>
In the above table, the total gap refers to the clearance the thrust bearing pad has from the static structure when the rotor is bottomed out in one direction. The thrust bearing eccentricity ratio is a number between -1 and 1 and represents the axial offset of the rotor from a centered position normalized by half of the total thrust bearing gap – 3 μm as designed.

3.1.2 Journal Bearings

For radial support, the turbocharger relies on an air journal bearing. There are two basic types of bearings that could be used – hydrostatic or hydrodynamic - and both will be explained. In both designs, the bearing consists of a thin film of air supporting the rotating structure against the static structure, just like the thrust bearing. However, as the rotor rotates the film of air starts to exhibit some interesting effects attributable to the rotation. More specifically, if the rotor is perturbed from the center of the bearing, the film will respond by providing a restoring force which can be broken into two components. One component is in the opposite direction of the perturbation, giving rise to direct stiffness, and the second component is perpendicular to the original perturbation. This second component gives rise to a cross stiffness which is the most fundamental problem of journal bearing stability.

A simple hydrodynamic bearing relies on the properties of the fluid film to provide stable operation. This requires that the rotor be operating eccentrically (the rotor center offset from the stator center) so that there is adequate direct stiffness. Figure 3-3 depicts the characteristics of a hydrodynamic bearing. Analytical and experimental work done by Piekos [2] and Orr [3] has indicated that the rotors such as those in this engine must operate at eccentricity ratios in the range of 0.9 to 0.99, where eccentricity ratio (ε) is defined as the distance between the rotor’s center and the stator’s center normalized by the initial bearing gap (c). Positioning the rotor at such eccentricities requires that a side force be imposed. In a traditional bearing, this force is applied by the rotor weight when the axis of rotation is horizontal. However, in the case of the micro-turbocharger the
weight of the rotor is not large enough since as the scale is decreased the mass (volume) decreases faster than bearing forces (area). As a result, the rotor requires to be side loaded by, for example, some pressurization mechanism where one side of the bearing is fed with higher pressure air than the other side, thus creating a net force on the rotor.

Figure 3-3: Hydrodynamic journal bearing operation

The engine was initially designed to operate on such a hydrodynamic bearing, but initial tests proved that such operation was not viable, given current microfabrication precision, as it required that the rotor spin at less than 1 \( \mu \text{m} \) from the stator wall. The challenge associated with this is apparent when it is realized that the surface roughness of the bearing walls is itself on the order of 1 \( \mu \text{m} \). As a result, attention shifted to hydrostatic operation.

During hydrostatic operation, the rotor is kept centered and direct stiffness is provided from an external pressurization source, just like the thrust bearings. All the hydrodynamic effects are still present, but they are very weak so long as the rotor is operating far from the wall. Traditional hydrostatic journal bearings have center injection
feeds to the bearing, as shown in Figure 3-4, and operate on the same principals as the thrust bearings. However, due to fabrication constraints such a design could not be built without increasing the complexity of the device. Instead, end injection hydrostatics were introduced. In this case pressurized air is supplied to one end of the bearing and encounters a pressure loss due to the entrance effect and an additional viscous pressure loss along the length of the bearing. Even though the entrance loss does not behave like a fixed resistance it has an effect similar to that of the restrictors and introduces stiffness to the bearing. [2, 3, 4]

**Figure 3-4:** Hydrostatic journal bearing operation

Unlike the thrust bearings, the natural frequency of the journal bearing is substantially less than the design speed as will be shown in Section 3.4.1. As a result, it is necessary to cross the journal natural frequency without crashing and then accelerate to the design speed in a stable fashion. To better understand the underlying physics
associated with crossing the natural, it is helpful to make reference to the Jeffcott rotor [5].

The Jeffcott rotor models a rotor as a single mass, concentrated at a point on a massless flexible shaft that is supported at each end by a bearing. An imbalance force associated with some initial eccentricity, \( e \), of the center of mass from the rotational centerline of the rotor is also added to introduce a forcing term and damping is also included. For the turbocharger, the imbalance arises from three major sources: bond misalignment of the turbine rotor to the compressor rotor, mask misalignments of the blades to the rotor disks, and blade etch non-uniformities – all of which will be discussed in more detail towards the end of this Chapter. Figure 3-5 summarizes the setup.

![Figure 3-5: Jeffcott rotor](image)

Writing the governing equations for the motion of the Jeffcott rotor, the following solutions are obtained [5]:

\[ \text{Equations} \]
Amplitude = \frac{e^{\left(\frac{\omega}{\omega_n}\right)^2}}{\sqrt{1 - \left(\frac{\omega}{\omega_n}\right)^2}^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2}} \quad \text{Eq. 3-1}

Phase = \tan^{-1}\left[\frac{2\zeta\left(\frac{\omega}{\omega_n}\right)}{1 - \left(\frac{\omega}{\omega_n}\right)^2}\right] \quad \text{Eq. 3-2}

where \(\omega\) is the angular velocity of the rotor, \(\omega_n\) is the natural frequency defined as follows:

\[
\omega_n = \sqrt{\frac{k}{m}} \quad \text{Eq. 3-3}
\]

and \(\zeta\) is the damping ratio:

\[
\zeta = \frac{c}{2m\omega_n} \quad \text{Eq. 3-4}
\]

where \(k\) and \(c\) are the direct stiffness and damping of the system, respectively, and \(m\) is the rotor’s mass, which is estimated to be 64 mg.

Making the analogy to the journal bearing of the turbocharger, Eq. 3-1 shows that at speeds well below the natural frequency the journal center is aligned with the bearing’s center. As the speed increases, the journal’s center starts to synchronously precess around the bearing’s center. At the natural frequency the precession reaches its maximum amplitude\(^2\). If the amplitude equals the bearing clearance, the rotor will strike

\(^2\) This is true only if \(\zeta\) is much smaller than 1. For larger \(\zeta\) the peak amplitude occurs at higher frequencies.
the wall. If this speed is successfully exceeded, the amplitude of the journal’s precession will start decreasing again and eventually, the center of mass of the rotor will align itself with the center of the bearing. Figure 3-6 graphically depicts this procedure and also examines the effect of changing the imbalance and damping ratio. As can be seen, increasing the imbalance and decreasing the damping ratio both increase the peak amplitude, making it harder for the rotor to cross the natural frequency.

![Graph](image)

**Figure 3-6**: Example of amplitude of journal’s center precession from bearing center.

So in order to help cross the natural frequency it is desired that the device have a small imbalance and a large damping ratio. For a given device, the imbalance level is fixed. However, the natural frequency and hence the damping ratio can be varied by changing the bearing supply pressure. More specifically, the natural frequency, which is directly dependent on the bearing supply pressure, can in theory be reduced to any desired value by decreasing the supply pressure.

Figure 3-7 examines the effect of varying the natural frequency, by changing the bearing supply pressure, while maintaining a constant level of imbalance and a constant
level of damping. As can be seen, as the natural frequency is decreased, the maximum amplitude of precession also decreases due to the fact that the damping ratio is increasing. This suggests that it is easier to cross lower natural frequencies than higher ones and is exploited experimentally.

\[ \zeta = 0.1, \; e = 2\mu m \]

\[ \omega_n = 2500 \text{ Rpm} \]
\[ \omega_n = 5000 \text{ Rpm} \]
\[ \omega_n = 10000 \text{ Rpm} \]

Figure 3-7: Effect of natural frequency on precession amplitude

Table 3-2 indicates typical values of the natural frequency, damping ratio and imbalance for a turbocharger compressor journal bearing at two different bearing supply pressures. The results depicted are based on models that will be discussed later in this Chapter.

Table 3-2: Typical values of parameters (single compressor bearing design)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value at 689 Pa Supply Pressure (0.1 psi)</th>
<th>Value at 34,474 Pa Supply Pressure (5 psi)</th>
</tr>
</thead>
</table>
| 103
The Jeffcott rotor examines the effect of any imbalance in the radial direction. However, the rotor can also have a 2\textsuperscript{nd} moment imbalance where the products of inertia in its inertia tensor are not equal to zero. In the absence of external restoring torques, the presence of such an imbalance would cause the rotor to precess spinning about its principal axis instead of its geometric axial axis, where the principal axis is defined as the axis about which the products of inertia are zero.

To better understand this, it is helpful to model the rotor as massless rod with two point masses located on either end of it, one slightly above the rod and the other slightly below the rod. Figure 3-8 depicts the setup and also indicates the principal axis of the system for the stationary rod. Once the rod starts spinning it can be easily seen that in the lack of external torques its principal axis will align itself with the rotation axis. More specifically, due to centrifugal forces the two masses will be forced to lie on a plane which is perpendicular to the axis of rotation.

---

3 Damping ratios were computed using a numerical simulation created by Liu. The number in parentheses accounts for non-linear effects which increase damping due to increased whirling amplitude.
Figure 3-8: Objects want to spin about principal axes if no external torques are applied.

In the turbocharger, this means that if the rotor has any 2nd moment imbalance about the rotation axis, it will want to synchronously precess (essential spinning about it’s principal axis). The thrust bearings will oppose such a precession but the torque driving the precession will increase quadratically with the speed. One important question is in the case that the thrust bearings cannot support the torque, will the rotor be able to still sustain operation once the principal axis becomes aligned with the rotating axis. More specifically is the angle between the principal axis and an axis perpendicular to the rotor...
greater than the tilt that is required for the thrust bearing edges to touch the static surface. A model will later be presented to assess this.

3.2 Journal Bearing Design Methodology

All of the builds discussed in this thesis operated on hydrostatic journal bearings. Some of the builds had a single turbine journal bearing, others a single compressor bearing and some had dual bearings – that is both a compressor and a turbine journal bearing. In this section discussion will focus on the methodology that was followed to design the bearings.

3.2.1 Single Bearing Design

The design of each individual journal bearing was done by Liu [4]. The objective was to develop a bearing geometry which would allow for design speed operation. The output of the analysis would be a bearing length and clearance accompanied with the required tolerances. This output relied on two inputs. One of the inputs was the bearing radius, which was either 4.1mm or 3mm, depending on whether a compressor or turbine bearing was being examined. And the other input was the amount of imbalance, which was assumed to be fixed for any given device and was experimentally found to range between 1 and 6 μm for all the devices tested.

As was discussed earlier, one of the inherent problems of gas bearings is the presence of a de-stabilizing cross stiffness. The approach Liu followed was to design a bearing such that this cross-stiffness was eliminated or reduced enough so that full speed operation could be achieved. His analysis indicated that for a given imbalance, there is a functional relationship between the clearance and the length of the bearing that drives the de-stabilizing cross stiffness to zero. The implications of this are that such a bearing has a theoretically infinite stability boundary. Eq. 3-5 depicts this relationship for the case of no imbalance and Figure 3-9 schematically depicts the results for several different imbalances for a compressor bearing:
\[ \frac{2rc}{L^2} = 1 \]  

Eq. 3-5

Figure 3-9: Relation between bearing clearance and length that result in a theoretically infinite stability boundary. Results depend on the level of imbalance and the bearing radius \[4\].

The above results have the profound implication that if the fabrication process was perfect it would be possible to build devices which in theory should have infinite stability boundaries. Unfortunately though, the fabrication process yields rotor’s with stochastic variations in imbalance levels and stochastic variations in bearing dimensions. Figure 3-10 examines the effect of varying the clearance of a 330 \(\mu\)m long, bearing with an imbalance of 3 \(\mu\)m on the bearing’s stability boundary.
Figure 3-10: Stability boundary for a 330 µm long compressor bearing with an imbalance of 3 µm. The radius of the bearing is 4.1mm, the mass of the rotor is 64 mg and the pressure drop across the bearing is 6.9 kPa (5 psi) [4].

Figure 3-10 suggests that as the clearance gets either larger or smaller than the optimal clearance suggested by Figure 3-9, the stability boundary for the bearing drops sharply. If the deviation is large enough, the stability boundary falls below the design speed of 1.2 million rpm. So if the bearing imbalance was known it would be possible to use the above chart to set a tolerance for the bearing clearance that would guarantee stable operation up to the design speed.

Unfortunately, the imbalance itself also varies. This complicates matters and to solve the problem we revert back to using Figure 3-9 where the clearance tolerance, required for the design speed, for each condition is superimposed on the plot as an error bar. The result of this is depicted in Figure 3-11. As can be seen the error bars for each of the fixed imbalance lines appear to overlap in some regions. At this point an assumption is made about the possible range of imbalance the fabricated rotors are expected to have based on both analytical and experimental results which will be
described later. Once this is done, and the bearing length is chosen, the clearance tolerance can be set by examining the clearances over which relevant error bars overlap. As an example, if the bearing length was 400 μm and the imbalance was assumed to range from 1-3 μm, using Figure 3-11, the clearance would be set at 20 μm +/- 2 μm – the region over which the blue and black error bars overlap.

![Figure 3-11: Compressor bearing design chart. Error bars indicate clearance tolerance for each imbalance level which will guarantee a stability boundary which is over 1.2 million rpm. The radius of this bearing is 4.1mm, the mass of the rotor is 64 mg and the pressure drop across the bearing is 6.9 kPa (5psi) [4]](image)

As the length of the bearing gets longer, the optimal clearance also gets larger and the tolerances on the bearing get looser (the overlaps are larger). As a result, with regards to tolerances, longer bearings are better. However, the larger clearances associated with the longer bearings result in journal bearing flow rates that can be larger than the main flow rates of the turbine and the compressor. In an effort to keep the journal flow rates low, the bearing was designed to be 15 μm wide by 330 μm long. Based on
measurements and models explained later in this chapter, the imbalance was assumed to be between 3-5 μm. This led to the following tolerances on the bearing: +/- 0.5 μm on the clearance and +/- 5 μm on the length. As will be discussed later, as the rotor spins up it deforms so that the optimal starting compressor bearing geometry changes. A similar analysis for an optimal turbine bearing gave the following results: 15 μm wide by 285 μm long with the same tolerances as the compressor bearing. The deformations of the rotor have a smaller impact on the turbine and as a result its design is not as affected by the rotation. Table 3-3 summarizes the results of the analysis.

Table 3-3: Optimal bearing geometries (with no deformations)

<table>
<thead>
<tr>
<th></th>
<th>Compressor Bearing</th>
<th>Turbine Bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Imbalance (assumed)</td>
<td>3-5 μm</td>
<td>3-5 μm</td>
</tr>
<tr>
<td>Length</td>
<td>330 +/- 5 μm</td>
<td>285 +/- 5 μm</td>
</tr>
<tr>
<td>Clearance</td>
<td>15 +/- 0.5 μm</td>
<td>15 +/- 0.5 μm</td>
</tr>
</tbody>
</table>

Figure 3-12 examines the effect the level of imbalance has on the tolerance of the bearing for a single compressor bearing device. First, as the imbalance increases, the tolerance on the bearing clearance appears to slightly decrease. Second, as the imbalance range increases, the tolerance on the bearing clearance decreases. Finally, for a given range of imbalance, as the imbalance increases, the bearing clearance tolerance decreases.
Compressor Bearing Tolerances

Imbalance: 0-2um
Clearance: 12.6um - 14.6um

Imbalance: 2-4um
Clearance: 13.7um - 15.0um

Max Clearance

Figure 3-12: Effect of imbalance on the compressor bearing clearance tolerance.4

---

4 Plots created by using Liu's journal stability model.
On a systems level, given that the objective is to maximize the tolerance of bearing clearance it is concluded that it is best for the rotor to have the lowest possible imbalance and the lowest possible imbalance range (assuming the bearing is appropriately designed for the expected imbalance levels). A similar figure constructed for the single turbine bearing device yields similar results with slightly tighter tolerances on the bearing clearance for all the imbalance levels. The turbine bearing design is expected to have a larger imbalance and also a larger imbalance range since the compressor is almost twice as massive as the turbine and bond mis-alignment will have a larger impact. This would result in tighter tolerances on the turbine bearing clearance making it a less preferable design. As an example, if the turbine bearing imbalance was assumed to range between 4-7 \( \mu m \) instead of 3-5 \( \mu m \) there is no clearance tolerance that will guarantee stable operation at the design speed.

### 3.2.2 Anisotropy

In an attempt to relax the tolerances that were required, Liu has shown that if the rotor had a different stiffness in each of two principal directions - as shown in Figure 3-13, the stability boundary would be substantially improved [6]

![Figure 3-13: Anisotropic journal bearing](image-url)
Figure 3-14 portrays how the stability boundary would be affected by the anisotropy for a specific case [4]. The method by which anisotropy was introduced to the system will be discussed later in this chapter.

![Diagram showing the effect of anisotropy on stability boundary](image)

**Figure 3-14**: Effect of anisotropy on stability boundary [4]

### 3.2.3 Dual Bearings

In attempt to further lower the impact of imbalance and rotor misalignment, a dual bearing rotor was designed on the suggestion of Ehrich [7]. The design procedure that was followed was one based on the assumption of superposition. More specifically, an optimal single compressor bearing was designed and then and optimal turbine bearing was designed. The stability models were then used to calculate the stability boundary for such a bearing with the assumption that the two wafers are perfectly aligned. The models concluded that a dual bearing allows looser tolerances than those required by the single bearing rotors. Also, the rotor is slightly more resistant to torques and the journal damping ratio is also increased.
3.2.4 Design Implementation and Rotor Deformations

So far the discussion has focused on the bearings themselves. However, on a systems level, there is also the question of how to supply hydrostatic air to the bearing(s). This is achieved in this design by hollowing the turbine vanes and using them as feed holes to the hydrostatic supply plenum which is in contact with both running gaps. In the case of a single bearing device a seal is inserted so as to cause most of the flow to go into the operational bearing. In the case of a dual bearing device such a seal is not required. Figure 3-15 is a schematic showing all possible bearing designs and the flow paths in each.
Figure 3-15: Cross section of (a) single compressor bearing design requiring the use of a seal to prevent flow from leaking to the turbine, (b) is a cross section of a dual bearing device which does not require such a seal, (c) is a cross section of a single turbine bearing design, which also requires a seal like the compressor bearing design. In all three designs the journal bearing flow comes from a supply plenum which feeds through the eight hollow turbine vanes. All the dimensions are based on design values and not on fabricated devices.
In designing the seal, the objectives were to reduce leakage while minimizing drag. There was also the constraint of the seal not touching, before the thrust bearings touched when the rotor tilted so as to not interfere with the thrust bearing operation. Another reason for this constraint was that while spinning, a rub on the seal would possibly be more detrimental than a rub on the thrust bearings since the radius of the seal is more than three times the radius of the thrust bearing pad and will thus be moving at a higher speeds.

Another issue that was considered during the design process was the deformation of the rotor as the speed increased. An FEA analysis was performed by Shawn Berry at Lincoln Labs and the results are shown in Figure 3-16, and Figure 3-17.

Figure 3-16: Finite element analysis of the rotor structural deformation at design speed. The deformations are exaggerated in the vertical directions. [Courtesy Shawn Berry]
Figure 3-17: Rotor tip axial and radial deflections as a function of angular speed for both the compressor and turbine
As can be seen, the rotor deforms substantially compared to the bearing geometric features as the speed increases. The most prominent effect is that the rotor exhibits a cupping effect which is due to the fact that the compressor disk overhangs the turbine. The compressor blades contribute only slightly to this effect because of their small size, unlike the previous compressor where the compressor blades were very massive and further enlarged the vertical deformations (see Chapter 2 for more details on the two compressor blade designs).

To make matters more complicated, the fact that the rotor deforms as it spins has to be included in the bearing design. Figure 3-18 depicts a stability boundary overlayed with the clearance trajectories as a function of speed. The specific design is for a 330 μm long bearing with an imbalance of 3 μm. As can be seen, the trajectory of the 15 μm initial clearance enters the unstable region at higher speeds. As a result in order to maintain stable operation, the trajectory is shifted to the right by increasing the design clearance from 15 μm to 16 μm. This way, the bearing is able to operate stably at all speeds.

![Figure 3-18: Stability boundary for a single compressor bearing with overlayed bearing clearance variation due to rotor radial expansion caused by rotation [stability boundary courtesy of Liu]](image-url)
Table 3-4 summarizes the bearing design requirements for non-anisotropic bearings based on the assumption of 3-5 μm imbalance. This imbalance range was chosen based on experimental and analytical results and will be explained in more detail later in the chapter.

Table 3-4: Journal Bearing Design (including deformations)

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Length (μm)</th>
<th>Width (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>330 +/- 5</td>
<td>16 +/- 0.5</td>
</tr>
<tr>
<td>Turbine</td>
<td>285 +/- 5</td>
<td>15 +/- 0.5</td>
</tr>
</tbody>
</table>

3.3 Assessment of Static/Dynamic Journal and Seal Geometries

As discussed in the previous sections, the rotordynamic performance of the device is highly sensitive to its geometry. Specifically, the dimensions of the journal bearing(s) are critical. Some of these critical dimensions can be measured during the fabrication process but others can only be estimated. For example, the bearing length can be easily determined by measuring the wafer thickness and then subtracting the blade span from it. However, the bearing clearance cannot be measured so easily, as there is some surface roughness which makes it hard to determine where the measurements should be taken at. Furthermore, only the entrance to the bearing is visible, with the profile of the rest of the bearing hidden from microscope lens. As a result, static air flow models were created and were employed to determine the geometry of crucial features of the device by measuring flow as a function of the pressure drop. Both incompressible and compressible models were developed so as to evaluate the importance of compressibility effects. Both models assume a pressure driven flow between flat plates with laminar fully developed viscous flow.

5 "Witness" journals are also used in assessing the journal geometry but they are exposed to slightly different boundary conditions than the real journal bearings and thus cannot be entirely relied upon.

6 Liu and others initially implemented incompressible flow models.
In addition to determining the static geometry of the device, the flow models were also employed to determine the geometry of the device during its operation. One such model is a flow resistor network model that was created to determine the axial position of the rotor in a single bearing device by measuring the variation in the journal bearing flow rate caused by changes in the seal clearance. The axial location of the rotor is a crucial variable as the rotor is constantly being exposed to varying pressures on the turbine and compressor side, which could overcome the load carrying capabilities of the thrust bearings and thus induce a crash.

3.3.1 Evaluating Effects of Compressibility

As was mentioned above, both incompressible as well as compressible flow models were created. These models are presented in detail in Appendix A, and below is a comparison of some of the results. For all the comparisons the bearing entrance loss has not been included in the calculations and the bearing exit pressure is set to standard atmospheric conditions.

![Flow Model Comparisons](image)

**Figure 3-19:** Mass flow rate comparison between incompressible and compressible models
Figure 3-20: Pressure profile along a bearing with no entrance loss based on incompressible and isothermal compressible models for supply pressures of 51.7 kPa (7.5 psi) and 17.2 kPa (2.5 psi).

Figure 3-21: Force acting on bearing wall with no entrance loss based on incompressible and compressible models
In Figure 3-19 the predicted flow rate is plotted as a function of the supply pressure under the assumption of incompressible flow, isothermal compressible flow and adiabatic flow. As can be seen, the model predictions lie very close to each other at pressures below 35 kPa. However, at higher pressures, the compressibility effects become more prominent leading to a discrepancy of almost 20% at a supply pressure of 103 kPa (15 psi).

In Figure 3-20 the pressure profile of the flow along the bearing has been plotted for a supply pressure of 17.2 kPa (2.5 psi) and a supply pressure of 51.7 kPa (7.5 psi). As can be seen, for the incompressible model the pressure distribution is linear whereas for the compressible model it is not. The difference again becomes more apparent as the supply pressure is increased.

In order to compute the stiffness of the bearing later it is necessary to compute the integral of the pressure force acting on its surface. Figure 3-21 depicts the force acting on each unit length of bearing wall. This force comes from integrating the pressure distribution shown in Figure 3-20 over the entire bearing length. As a result, the compressible model predicts a larger force than the incompressible model at the larger pressures.

Based on the above comparisons, it is concluded that at pressures below 35 kPa (5 psi) the error associated with the compressibility effects is minimal. However, at higher pressures only the compressible models should be used if high accuracy is desired. With both models in place, this thesis will rely only on the compressible flow model regardless of the importance of compressibility effects.

3.3.2 Static Seal and Journal Flow Model Results

Upon delivery of a device one of the first tests done is a fluidic characterization of the journal bearing using the following procedure. Pressure versus flow data from the bearing is taken and is compared to the model predictions for an array of clearances. The model predictions rely on the length of the bearing, which is measured during the fabrication process. Figure 3-22 depicts the results for a compressor bearing which is
330 μm long (the design value). By examining which clearance causes the best fit between the collected data and the model, the bearing clearance is estimated.

![Compressor bearing flow rate graph](image)

**Figure 3-22**: Compressor bearing flow rate. The compressor bearing radius is 4.1mm and its length is 330 μm. The design clearance is 16 μm.

Figure 3-23 is the equivalent of Figure 3-22 for a 285 μm long turbine bearing.
Figure 3-23: Turbine bearing flow rate. The turbine bearing radius is 3.0 mm and its length is 285 μm. The design clearance (if a turbine bearing is present) is 15 μm.

In the case of single bearing devices, a similar procedure is followed to characterize the seal clearance. Figure 3-24 depicts the flow rate across the seal in a compressor bearing device for an array of clearance.
3.3.3 Dynamic Axial Eccentricity Model

In trying to better understand the rotordynamic behavior of the turbocharger during a test, it is useful to know the rotor’s axial position at all times. One way to tell the position is by examining the flow rate through the thrust bearings. More specifically, for thrust bearings operating at a fixed pressure, if the rotor moves closer to the front thrust bearing, the front thrust bearing flow decreases and the aft thrust bearing flow increases. However, during some of the tests the thrust bearing pressures were changed, which complicates interpretation of the thrust bearing flow rate data. Furthermore, at high pressure settings, the thrust bearing nozzles choke and only respond to axial motions when the rotor get very close to one of the pads (which unchokes the nozzles).

Fortunately, in the case of single bearing devices, there is always a seal present on the back of the compressor whose leakage flow depends on its clearance (as was shown in Figure 3-24). As a result, by measuring or computing the flow through the seal it is possible to determine the seal clearance and thus the rotor’s axial eccentricity. A model
was therefore created to help understand a test with one of the dies from the single
turbine bearing build. In this design, the seal was located on the rotor and was preventing
most of the flow from entering the compressor gap. During the experiment, only $P_{tap}$,
$P_{t\_int}$, $P_{c\_int}$ and the total journal supply flow rate are recorded.

![Flow resistance network for computation of rotor axial position](image)

**Figure 3-25:** Flow resistance network for computation of rotor axial position

The model is a flow resistor network, as shown in Figure 3-25. It includes a flow
resistance associated with the turbine bearing in parallel with the resistance of the seal
and the compressor gap which are in series with respect to each other. Upstream of the
entire network is a resistance associated with the pressure drop across the vane feed lines.
The flow resistances are computed using the isothermal compressible model discussed
earlier and the seal clearance is iteratively varied until all the boundary conditions are
satisfied and the flow rate is equal to the known total journal supply flow rate. This
procedure is repeated through time and the history of the seal clearance is thus
determined. The output of this model is the rotor’s axial eccentricity as a function of
time and the results for a specific run are shown in Figure 3-26.
As can be seen the model suggests that the rotor moved axially during the test but not more than 30% of the available clearance. This suggests that the rotor did not crash due to a possible rub on the thrust bearings something that is consistent with the fact that neither of the two thrust bearing flow rates showed any large flow rate drops.

### 3.4 Isotropic/Anisotropic Stiffness Models

As was discussed earlier, Liu suggested that an anisotropic bearing would have a higher stability boundary than a non-anisotropic one. As a result, an anisotropic bearing was designed for the turbocharger. To begin the analysis, the stiffness of an ordinary isotropic journal bearing will be derived. Then several alternate approaches introducing anisotropy will be presented and analyzed.
3.4.1 Isotropic Stiffness Model

The natural frequency of an isotropic journal bearing stiffness was modeled as follows. The rotor is statically displaced from the journal’s center and the bearing clearance is broken into multiple rectangular ducts whose width varies sinusoidally with respect to their angular position. The flow models, which are presented in detail in Appendix A, are then used to compute the pressure distribution within each duct. The resulting pressure fields are then integrated along the duct lengths and the net force acting on the rotor is computed by vector addition of the force from each duct. To compute the stiffness, the net force is divided by the original displacement of the rotor. The stiffness turns out to be non-linear with respect to the displacement and as a result the smaller the displacement, the closer the stiffness prediction is to the stiffness of a perfectly centered rotor. Figure 3-27 is plot of the stiffness for a 16x330 μm compressor bearing as a function of the bearing supply pressure.

![Stiffness of a 16x330 μm compressor bearing](image)

**Figure 3-27:** Stiffness of a 16x330 μm compressor bearing

Experimentally, the stiffness itself is not observed, rather the natural frequency of the bearing is seen and as a result it is more typical to plot the bearing’s natural frequency. The two are simply tied by the following definition:
\[ \omega_n = \sqrt{\frac{k}{m}} \]  

Eq. 3-6

For the case of a 64 mg rotor, the natural frequency plot is derived from Figure 3-27 and is shown in Figure 3-28.

![Graph showing natural frequency vs. journal bearing supply pressure](image)

**Figure 3-28:** Natural frequency of a 16x330 μm compressor bearing

As can be seen the journal natural frequency increases with increasing bearing supply pressure. However, it is increasing at a decreasing rate and clearly lies below the design speed. At higher pressures, the bearing flow is eventually expected to choke and experimental results which will be presented in Chapter 5 indicate that the journal bearing natural frequency starts decreasing after a pressure of around 103 kPa (15 psi). In addition, large bearing pressures create significant axial forces on the rotor (due to the bearing supply system design) which make it hard to keep the rotor centered. As a result, for most experiments the bearing supply pressure has been kept below 48 kPa (7 psi).
3.4.2 Anisotropic Stiffness Models

Having outlined the procedure for determining the stiffness of an isotropic journal bearing, alternate anisotropic designs shall be presented. As reminder, an anisotropic bearing is one which has a different stiffness in each of two principal directions as was shown in Figure 3-13. According to Liu even a small percentage difference (~5%) in the each of the principal stiffnesses can have a significant effect. A simple way of introducing anisotropy is to block the bearing supply flow from entering the bearing in two diametrically opposed regions of the bearing. Complete blockage is not possible since some clearances are required for the rotor to spin. Figure 3-29 depicts four separate geometries which were examined.
Figure 3-29: Anisotropy Designs A, B, C, and D. Figures on the left are a top view and figures on the right are cross-sections of the designs.

As can be seen, designs A, B, and C relied on inserting two diametrically opposed physical obstructions to the flow entering the bearing whereas design D relied on blocking four of the eight supply feed lines and creating the asymmetry from the flow resistance of the hydrostatic supply plenum itself. In all of the designs, a 200 μm seal was present at a radius of 3.39 mm to reduce the flow leakage to the turbine gap.

In design A a seal was placed over a 60° sector of the bearing entrance. The seal clearance was set to 11 μm. A flow resistor network model was then constructed to determine the effectiveness of the design and also used to examine the effect of varying the seal length. The results for a supply pressure of 5 psi are depicted in Figure 3-30. As can be seen the stiffness is higher in the direction of the seal. This is an unintuitive result and is attributed to the fact that the seal causes the bearing entrance flow resistance to be better matched to the resistance associated with the bearing length itself. However, as the seal is enlarged eventually, the stiffness in the direction of the seal will be lower. The highest anisotropy was achieved with the shortest seal and is approximately 75%. Of course the anisotropy of this design could be increased by having the seal occupy a larger sector of the bearing area.
Figure 3-30: Stiffness anisotropy for design A.

In design B a sealed cavity was introduced underneath a sector of the bearing. The seals used to define the cavity were 200 µm in length, 750 µm in width, and had a clearance of 11 µm. The supply pressure was once again set to 5 psi and a similar resistor network model was created and used to assess the viability of the design. Figure 3-31 depicts the stiffness in the two directions as a function of the angular extent of the sealed region. As can be seen, at small plenum angles the stiffness is again higher in the direction of the sealed plenum but as the sealed plenum angle increases that trend reverses. Again, the reason for this has to do with bearing resistance matching. The maximum anisotropy achieved in this design was on the order of 80 % but could be further increased if the angular extent of the cavity was further enlarged.
For design C, a seal similar to that of design A is placed underneath a sector of the bearing entrance but this time the seal extends all the way to the seal for the turbine gap. It was not possible to construct a simple analytical model for this design and as a result, a CFD calculation was performed by Gong [8] which indicated that the pressure after the seal would be very low and as a result, large levels of anisotropy should be expected.

Even though designs A, B, and C clearly can introduce anisotropy to the system, they have two disadvantages over design D. The first disadvantage involves the fact that they require a shallow etch over the compressor bearing which can damage the bearing’s entrance. The second and most important disadvantage is that they all are highly sensitive to the seal clearance which is bound to change during the operation of the device by as much as 80% due to structural deformation of the rotor caused by centrifugal forces (as was explained earlier in this Chapter). As a result, attention shifted to design D, due to its simplicity and lack of first order dependence on the rotor’s deformations.

To assess design D an analytical model was put together. Similar to the other models, the bearing supply configuration was modeled as a resistance network as shown

---

**Figure 3-31:** Stiffness anisotropy for design B
in Figure 3-32. A numerical procedure was followed to compute the two stiffnesses where the geometry was broken into small angular sections.

Figure 3-32: Resistance network for anisotropic bearing

Nodes A, B, C, and D represent the location directly after the four supply feed lines. A known supply pressure, $P_{\text{supply}}$, is fed to each of the four lines. In going through the feeds, the flow experiences a pressure drop which is depicted as a resistance in the figure. Focusing attention to node A, the flow then splits up into four separate flow paths – along the plenum towards node B, along the plenum towards node D, into the bearing gap, and into the seal. Examining the node that is directly to the right of node A, the flow
now splits into three paths – towards node B, into the bearing gap, and into the seal. This pattern persists for all the nodes. In moving along the hydrostatic plenum the tangential flow rate keeps decreasing as more flow keeps going into the seal clearance and the compressor bearing. Eventually, the flow originating from Node A meets the flow originating from node B and at that point the tangential flow rate in the plenum is zero and the pressure of the two flow paths has to be the same. Convergence of the numerical model requires that these two conditions along with all the other boundary conditions are satisfied. Appendix B depicts the numerical method used to solve the problem in more detail.

The results of the model for the case of a single compressor bearing device with bearing dimensions of 330 μm by 16 μm, seal clearance of 8 μm, and plenum height of 50 μm are depicted in Figure 3-33. As can be seen there is limited anisotropy present in this design and is a maximum at around 10 kPa.

![Figure 3-33: Natural frequency for anisotropic single compressor bearing.](image)

Figure 3-34 depicts the effect of decreasing the plenum height on the level of anisotropy. As can be seen, the direction of the anisotropy is reversed as the plenum
height is reduced. This is similar to what was seen in the other designs. The 40 μm plenum appears to not be introducing any level of anisotropy at the low pressures and a small amount at higher pressures. The 30 μm plenum introduces a level of anisotropy which is over 30% and is persistent over the entire range of bearing supply pressures.

![Graph showing the effect of plenum height on single compressor bearing device anisotropy.](image)

**Figure 3-34:** Effect of plenum height on single compressor bearing device anisotropy.

The model was run for several other cases and was also modified to allow for the characterization of dual bearing anisotropy levels. Sensitivity studies on the bearing clearance and the seal clearance indicated that the above results were only slightly sensitive to changes in those parameters. Based on the fact that the 50 μm and 40 μm plena gave only marginal levels of anisotropy, it was decided that the anisotropic bearing design would have a 30 μm hydrostatic supply plenum.
3.5 Rotor Moments

As was discussed in section 3.2, the imbalance of the rotor is a critical parameter that must be controlled to achieve high speed operation. In the earlier stages of the program it was believed that lower imbalance was better. However, as was discussed earlier this is not necessarily true. In fact, for any imbalance level, if the bearing is designed correctly the design speed should be achievable [Liu]. So the focus shifted from desiring devices which had zero imbalance to devices whose imbalance was more tightly controlled. For the 2nd moment imbalance, however, it is maintained that the lower the level the better it is.

There are three sources of imbalance on the turbocharger – mask alignment, etch uniformity, and bond alignment. First however, let us re-define what imbalance is. For a single bearing device, the imbalance is the radial distance between the geometric center of the disk which has the journal bearing, and the center of mass of the entire rotor. At supercritical speeds, the rotor spins about its center of mass so a natural limit to the imbalance level is the bearing’s clearance – which is around 15 μm.

3.5.1 Mask Alignment

Mask alignment refers to how well each mask pattern is aligned to some reference mark on a wafer. Any misalignments in the masks used to pattern the rotor introduce imbalance. Masks can be either aligned to a reference mark on the same side of the wafer that they will be patterned on or they can be aligned to a reference mark on the other side of the wafer. In the first case the alignment is called front to front alignment, and in the second case it is called front to back alignment. Discussions with the fabrication team indicate that with the alignment tool used (an EV), the front to front alignment can be within 0.5 μm and front to back alignment can be within 1 μm.

3.5.2 Bond Alignment

Bond alignment refers to the alignment of two wafers when they are bonded. The bonder that was used in building the turbocharger relied on a front to front aligner. More specifically, the only way to bond the compressor wafer to the turbine wafer was by aligning a reference mark on the front of the compressor wafer to the front side of the
turbine wafer or by aligning a reference mark on the back side of the compressor wafer to
the back side of the turbine wafer. This is unfortunate because most of the rotor weight is
on the disks of the compressor wafer (back side of the compressor wafer) and the turbine
wafer (front side of the turbine wafer) and as a result a front to back bonder would have
been better. Recent bond alignment data indicates that it is possible to bond with a
misalignment of less than 2 μm.

3.5.3 Etch Uniformity

In etching the blades of the compressor and the turbine, a large fraction of the
silicon is removed from the wafers. The depth uniformity of this etch over the rotor
introduces imbalance. An experiment was conducted in which the blade depth was
measured around a rotor to quantify this effect. Depth measurements were taken for each
blade at four different radial locations. Locations A, B, and D were determined by
constructing lines connecting blade leading and trailing edges and measuring the depth of
the regions shown in Figure 3-35. The deepest point in each of those regions was
recorded and plotted. Location C was determined by constructing a bisector to the
trailing edge of a blade and then measuring at a distance of 230 μm from the blade’s
trailing edge.

Figure 3-35: Blade depth measurement locations for old compressor design
The measurements are plotted in Figure 3-36 with sinusoidal functions fit to the data. The most important feature to notice is that the data fits a sinusoid well indicating that the etch variation is planar. In order to confirm that the etch variation observed over the rotor was planar, Figure 3-37 was constructed, where the amplitude of the sinusoidal function fitted to the data has been plotted as a function of the radial position at which it was measured. As would be expected for a planar etch variation, the amplitude increases linearly as a function of radius.

Another thing to notice is that the mean depth of the etch is different at each location. More specifically, as the region being measured was more narrow the shallower the etch was. This is evident in Figure 3-36 where it can be seen that the average depth measurements at location A, which is the most confined area, are almost 6% less than the measurements at locations C and D, which are more exposed. This is a known effect of deep reactive ion etching (discussed in greater detail in Chapter 2, referred to as "loading") but due to symmetry arguments it does not affect the imbalance of the rotor.

![Figure 3-36: Circumferential blade depth measurements for a single compressor.](image)
Similar experiments for the turbine blades confirmed planar etch variations as well. AutoCad was then used to compute the imbalance introduced from the blade etches as a function of the amplitude of the best fit sine wave by creating 3-D representations of the rotor. The results are listed for all blade designs (as defined in Chapter 2) in Table 3-5. One important thing to notice is the large increase in the imbalance introduced by the new compressor blades vs the old ones. This is due to the fact that the new compressor blades are much thinner demanding that more silicon be etched away.

---

7 The new compressor and turbine blades were designed to improve performance. Their geometry is different from the old blades.
<table>
<thead>
<tr>
<th>Blade Design</th>
<th>Imbalance (mg-μm) per 1 μm Amplitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>Old Compressor Blades</td>
<td>61.0</td>
</tr>
<tr>
<td>New Compressor Blades</td>
<td>114.4</td>
</tr>
<tr>
<td>Old Turbine Blades</td>
<td>42.9</td>
</tr>
<tr>
<td>New Turbine Blades</td>
<td>43.4</td>
</tr>
</tbody>
</table>

Furthermore, it was experimentally found that the directionality of the etch variation was also consistent. More specifically, the etch depth of the blades was deeper towards the edge of the wafers. As a result, for each die, the amplitude of the etch variation in combination with the die location on the wafer could be used to determine the imbalance vector due to the blade etch non-uniformity.

### 3.5.4 Imbalance Model

Given the importance of the imbalance level, a model whose original framework was constructed by Jacobson, was extended and refined to compute the imbalance level for a rotor given some assumptions about each of the three sources of imbalance. The model initially focused on a single turbine bearing device (Build C). As the directionality of each source of imbalance with respect to each other was independent, two models were used. One model computed the worst case imbalance. That is, the imbalance that would result if every source of imbalance lined up in the same direction. The second model computed the root mean square imbalance and is more representative of the expected level of imbalance.

The approach followed for both models was to first compute the imbalance level of the turbine, then compute the imbalance of the compressor, and finally compute the imbalance introduced by bonding the compressor to the turbine in addition to the imbalance introduced by the etch non-uniformity. Table 3-6 lists the type of alignments involved for each component of the rotor and their respective mass.
Table 3-6: Imbalance model for turbine bearing device

<table>
<thead>
<tr>
<th>Turbine Bearing</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Blades</td>
<td>b-b, b-f, f-f</td>
</tr>
<tr>
<td>w.r.t. Bearing;</td>
<td></td>
</tr>
<tr>
<td>$m_{imb , blades} = 3.0 \text{mg}$</td>
<td></td>
</tr>
<tr>
<td>Comp. Disk</td>
<td>b-b</td>
</tr>
<tr>
<td>w.r.t. Alignment Mark used for Bond;</td>
<td></td>
</tr>
<tr>
<td>$m_{comp , disk} = 26.8 \text{mg}$</td>
<td></td>
</tr>
<tr>
<td>Comp. Blades</td>
<td>f-f, f-b</td>
</tr>
<tr>
<td>w.r.t. Alignment Mark used for Bond;</td>
<td></td>
</tr>
<tr>
<td>$m_{comp , blades} = 13.7 \text{mg}$</td>
<td></td>
</tr>
<tr>
<td>Entire Compressor</td>
<td>BOND, b-b, f-f</td>
</tr>
<tr>
<td>w.r.t Turbine Bearing;</td>
<td></td>
</tr>
<tr>
<td>$m_{comp} = m_c , imb + m_c , disk + m_c , blades = 46.1 \text{mg}$</td>
<td></td>
</tr>
<tr>
<td>Turbine Blade Etch Uniformity</td>
<td>ETCH-T</td>
</tr>
<tr>
<td>Compressor Blade Etch Uniformity</td>
<td>ETCH-C</td>
</tr>
</tbody>
</table>

The following equations were used to compute the worst case and rms imbalance levels:

$$imb_{worst} = m_n (ff + fb + ff) + m_d (ff) + m_c (ff + fb) + m_C (BOND + bf + ff) + imb_{a} + imb_{e}$$  \hspace{1cm} Eq. 3-7

$$imb_{rms} = \sqrt{m_n \cdot (ff^2 + fb^2 + ff^2) + m_d \cdot (ff^2) + m_c \cdot (ff^2 + fb^2) + m_C \cdot (BOND^2 + bf^2 + ff^2) + imb_{a}^2 + imb_{e}^2}$$  \hspace{1cm} Eq. 3-8

Eq. 3-7 and Eq. 3-8 give the imbalance in mg-μm. In order to convert this to the distance between the bearing center and the center of mass of the rotor, the following equation is used:

$$e (\mu m) = \frac{imb (mg-\mu m)}{m_{rotor}}$$  \hspace{1cm} Eq. 3-9

where $m_{rotor}$ is 68.9mg

As can be seen in Eq. 3-7 and Eq. 3-8 the unknown parameters are the front to front alignment, the front to back alignment, the bond alignment, and the blade etch uniformity. Building the turbocharger pushes all the tools involved past the machine
manufacturer’s specifications. As a result the tolerance of each parameter is derived from the experience of the fabrication team. Table 3-7 has been created based on three separate sets of tolerances: A “Bad” case which is possible but rather unlikely. A “Medium” case which is what is typically achieved. And a “Good” case which is possible if the fabricators are very careful and willing to repeat processes (such as the bond alignment). The imbalance models where then used to compute the worst case and rms imbalance for each of the scenarios.

Table 3-7: Predicted levels of imbalance based on fabrication capability estimates for turbine bearing device. The turbine etch uniformity was assumed to be the compressor etch uniformity linearly scaled by the ratio of their radius (3/4.1).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Bad (μm)</th>
<th>Medium (μm)</th>
<th>Good (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alignment ff (μm)</td>
<td>2</td>
<td>1</td>
<td>0.5</td>
</tr>
<tr>
<td>Alignment fb (μm)</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>Alignment BOND (μm)</td>
<td>4</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>Comp. Etch Amplitude (μm)</td>
<td>5</td>
<td>3</td>
<td>1.5</td>
</tr>
<tr>
<td><strong>Imbalance Worst (μm)</strong></td>
<td>14.1</td>
<td>8.5</td>
<td>4.6</td>
</tr>
<tr>
<td><strong>Imbalance Rms (μm)</strong></td>
<td>6.1</td>
<td>3.8</td>
<td>2.1</td>
</tr>
</tbody>
</table>

As can be seen depending on the case and the model used, the predicted imbalance ranges from 2.1 μm to 14.1 μm. An ultimate maximum for the imbalance is the bearing clearance which is 15 μm so it is good to see that this number is not exceeded. Overall, the expectation for this device would be to have an imbalance level between 2.1 μm and 6.1 μm.

The procedure that was outlined for the turbine bearing design was then followed for the compressor bearing design (Build Locos 4). Table 3-8 lists all the parameters that were input to the compressor imbalance model based on the fabrication processes followed.
Table 3-8: Imbalance model for compressor bearing device

<table>
<thead>
<tr>
<th>Component</th>
<th>Alignments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor Blades</td>
<td>f-f, f-b, b-b</td>
</tr>
<tr>
<td>w.r.t. Bearing; ( m_{\text{comp blades}} = 4.6\text{mg} )</td>
<td></td>
</tr>
<tr>
<td>Turbine Disk</td>
<td>f-f, f-b</td>
</tr>
<tr>
<td>w.r.t. Alignment Mark used for Bond; ( m_{\text{turb disk}} = 22.0\text{mg} )</td>
<td></td>
</tr>
<tr>
<td>Turbine Blades</td>
<td>b-b</td>
</tr>
<tr>
<td>w.r.t. Alignment Mark used for Bond; ( m_{\text{turb blades}} = 3.0\text{mg} )</td>
<td></td>
</tr>
<tr>
<td>Entire Turbine</td>
<td>BOND, b-b</td>
</tr>
<tr>
<td>w.r.t. Compressor Bearing; ( m_{\text{turb}} = m_{\text{turb disk}} + m_{\text{turb blades}} = 25.1\text{mg} )</td>
<td></td>
</tr>
<tr>
<td>Turbine Blade Etch Uniformity</td>
<td>ETCH-T</td>
</tr>
<tr>
<td>Compressor Blade Etch Uniformity</td>
<td>ETCH-C</td>
</tr>
</tbody>
</table>

In this design, the total number of front to back and front to front alignments have been reduced and the bond alignment has a smaller significance as it is weighted by the turbine rotor mass instead of the larger compressor rotor mass. The blade etch non-uniformities are also different as new blade designs were used. The rotor mass in this design is 69.9 mg. The results of the models are depicted in Table 3-9.

Table 3-9: Predicted levels of imbalance based on fabrication capability estimates for compressor bearing device including new blade designs. The turbine etch uniformity was assumed to be the compressor etch uniformity linearly scaled by the ratio of their radius \((3/4.1)\).

<table>
<thead>
<tr>
<th></th>
<th>Bad</th>
<th>Medium</th>
<th>Good</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alignment ff (( \mu \text{m} ))</td>
<td>2</td>
<td>1</td>
<td>0.5</td>
</tr>
<tr>
<td>Alignment fb (( \mu \text{m} ))</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>Alignment BOND (( \mu \text{m} ))</td>
<td>4</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>Comp. Etch Amplitude (( \mu \text{m} ))</td>
<td>5</td>
<td>3</td>
<td>1.5</td>
</tr>
<tr>
<td>Imbalance Worst (( \mu \text{m} ))</td>
<td><strong>14.7</strong></td>
<td><strong>9.0</strong></td>
<td><strong>4.7</strong></td>
</tr>
<tr>
<td>Imbalance Rms (( \mu \text{m} ))</td>
<td><strong>8.7</strong></td>
<td><strong>5.3</strong></td>
<td><strong>2.7</strong></td>
</tr>
</tbody>
</table>
As can be seen despite the multiple changes that occurred, the levels of imbalance for this design remained very similar to the old design. More specifically the expectation of imbalance ranges from 2.7 μm to 8.7 μm. It should be pointed out though that this design was fabricated a couple of years after the turbine design and the fabrication experience was greater, making the “Bad” scenario less likely.

To better understand the contribution of each source of imbalance on each of the two designs the following chart was constructed using the worst case model and the “Good” tolerances listed in Table 3-9. As was indicated in Table 3-7 and Table 3-9 the total imbalance for both designs appears to be very similar, however the distribution of the imbalance among the four sources is quite different.

![Imbalance Composition (Worst Case Scenario)](image)

**Figure 3-38:** Imbalance contributions from each of the four sources for the turbine bearing design and the compressor bearing design.

In the compressor bearing design the imbalance introduced by the bond alignment appears to have been substantially reduced. This was expected since in this design the
bearing is on the compressor which is roughly twice as massive as the turbine. Also the
imbalance introduced by the front to front alignment has been reduced. This is partly due
to the elimination of one front to front alignment and also due to the thinner and thus
lighter compressor blades. Finally, the one thing that has gotten much worse in this
design is the imbalance introduced by the blade etch non-uniformity. In the new design,
the blade etch non-uniformity contributes 67% of the imbalance as opposed to 44% in the
turbine bearing design because the thinner compressor blades require that a larger area be
etched.

3.5.5 Principle Axis Model

Similar to the imbalance model a model was put together to estimate the angle
between the principal axis and the perpendicular to the rotor wafer, based on the
fabrication tolerances of the rotor. The only effects that were examined were the etch
uniformity and the bond alignment. The asymmetry of the speed bumps was also taken
into account in this model. The following steps list the procedure that was followed in
this parametric study:

1. Construct a solid 3D Rotor in AutoCAD
2. Introduce some fabrication imperfection (bond alignment or etch non-uniformity)
3. Use AutoCAD to compute center of mass of rotor
4. Use AutoCAD to compute moments (6 total) of rotor about its C.O.M

\[
I = \begin{bmatrix}
I_{xx} & I_{xy} & I_{xz} \\
I_{yx} & I_{yy} & I_{yz} \\
I_{zx} & I_{zy} & I_{zz}
\end{bmatrix}
\]

(symmetric matrix) \hspace{1cm} \text{Eq. 3-10}

5. Use Matlab to compute eigenvectors of moments of inertia matrix
\[ V_1 = \begin{bmatrix} V_{x1} \\ V_{y1} \\ V_{z1} \end{bmatrix}, \quad V_2 = \begin{bmatrix} V_{x2} \\ V_{y2} \\ V_{z2} \end{bmatrix}, \quad V_3 = \begin{bmatrix} V_{x3} \\ V_{y3} \\ V_{z3} \end{bmatrix} \]

Eq. 3-11

6. Determine the angle the principal axis, whose largest component is in the z-direction, makes w.r.t. the z-axis.

\[ \alpha = \tan^{-1} \left( \frac{\sqrt{V_x^2 + V_y^2}}{V_z} \right) \]

Eq. 3-12

7. Compare \( \alpha \) to max of 0.14° allowed before T.B. touch

In doing the parametric study it was assumed that the etch uniformity of the compressor and the turbine were in the same direction and specifically in the opposite direction of the asymmetric speed bump. The two etch non-uniformities tend to offset their effects on the 2\textsuperscript{nd} moment of the rotor and since the objective of the analysis was to get an estimate of how large \( \alpha \) could be, only one etch non-uniformity at a time was examined. Furthermore, the effects of any bond misalignment were superimposed on the effects of the etch non-uniformities under the assumption that the bond misalignment was in the direction which maximized its effect on the value of \( \alpha \). Figure 3-39 depicts the results of the analysis. The lines that slope upward indicate the effects of compressor etch non-uniformity (with no etch non-uniformity for the turbine) for three separate conditions: no bond misalignment, a bond misalignment of 2 \( \mu \)m, and a bond misalignment of 5 \( \mu \)m. The lines that slope downward indicate the effects of turbine etch non-uniformity for the same three conditions.
Figure 3-39: Angle between rotor principal axis and rotor perpendicular. Upward sloping lines examine effect of compressor etch non-uniformity with varying levels of bond misalignment and downward sloping lines examine the effect of turbine etch non-uniformity.

As can be seen, the tilt introduced over the entire range of bond misalignments and etch non-uniformities is always less than the maximum allowable (0.14°). This is fortunate and leads to the conclusion that even though the rotor is precessing, its precession angle is not large enough to make it bottom out angularly. It is also interesting to note that the etch non-uniformities can have a much bigger impact on the precession angle than the bond alignment can.

3.6 Assessment of Rotor Moments

In order to assess the rotor moments there are two separate procedures that are followed. The first procedure involves precise measurements of the device while it is
being fabricated and the other procedure involves actively measuring the rotor’s precession orbits while it is spinning. The first procedure can be used to estimate both the first and second moments of the rotor whereas the second procedure can only be used to measure the rotor’s first moment (imbalance).

3.6.1 Fabrication Measurements

Estimating the rotor’s moments while it is still in fabrication has an important advantage in that it saves wafers from being bonded to a poor compressor/turbine stack. There are two types of measurements that are taken in the clean room: alignment measurements and blade etch depth measurements. These measurements are taken for both the turbine and compressor wafers. With these measurements the rotors moments can be computed as outlined in the previous section and if they are found to be larger than prescribed limits the wafers are not pushed further in the fabrication process.

3.6.2 Fiber Data Measurements

To measure the rotor’s angular speed, a fiber optic displacement sensor (from Philtec, model D6) is used. The fiber optic outputs a voltage that is proportional to the distance between its tip and the silicon directly beneath it. It is focused above the speeds bumps, and as each one goes by the output voltage varies in a step-like fashion. It was shown earlier that at sub-critical speeds the rotor spins about its geometric center and as the speed is increased it starts spinning about a point that is off of its center and finally, after crossing the natural frequency spins around its center of mass. This movement can be captured by the fiber optic as is portrayed in the following schematic:
The speed bump edges lie on radial lines from the rotor's center. As a result, at a given angular speed, the speed bumps will create a longer or shorter time trace on the fiber optic output if the rotor is not spinning about its center. This fact was exploited under the lead of Dr. Paduano [9] who created a signal processing algorithm that computes how far the rotor’s geometric center is from the point about which it is spinning. The procedure always involves taking a very low speed subcritical reference measurement which is then compared to the point of interest. When the reference measurement is compared to a supercritical measurement point, the imbalance is determined.

3.7 Summary

This chapter discussed the unique geometry of the turbocharger rotor and the bearing system that supports it. The rotor relies on hydrostatic thrust bearings for axial support and a hydrostatic journal bearing for radial support. Both bearings are externally pressurized.

The methodology followed to design the journal bearing(s) was then described. This invokes the analysis done by Liu as well as structural deformations the rotor experiences due to centrifugal loading when it is spinning. The conclusion is that in order to achieve high speed operation the imbalance levels have to be tightly controlled.
(without necessarily being zero) and the bearing dimensions have to have very stringent tolerances. Liu’s work further suggests that these tolerances can be relaxed if anisotropy is introduced to the bearing system.

Fluidic models used to characterize the geometry of the devices are then discussed. Both incompressible and compressible models are presented and it is shown that the incompressible model results are consistent with the compressible ones up to bearing supply pressures of approximately 35 kPa (5psi). These models are further extended to predict the journal natural frequency. Furthermore, flow resistor network models are created to assess possible ways to introduce anisotropy to the bearing as well as to assess the rotor axial position during high speed operation.

Finally, models used to assess the rotor’s moments are presented. One model computes the expected radial imbalance and another model computes the expected angle between the rotor axis of rotation and the relevant principal axis. Both of these models rely on tolerances that the fabrication team believes to be achievable. Methods to assess these moment’s experimentally are finally described.

3.8 References


CHAPTER 4

EXPERIMENTAL SETUP

This chapter describes the experimental setup including the gas handling system, the instrumentation and the data acquisition system.

4.1 Gas Handling System and Instrumentation

The gas handling system is used to provide flow to the turbocharger, measure pressures and mass flow rates coming into and out of the device and monitor the rotor speed. The interface between the gas handling system and the turbocharger is achieved through a mechanical package, which is discussed in detail in Appendix C.

Figure 4-1 depicts the flow paths for the compressor and turbine. As can be seen, the compressor inlet is exposed to atmosphere and discharges to a throttle valve which is maintained at a fixed setting during each experiment. After exiting the throttle the flow passes through a mass flow meter which records the flow exiting the compressor. For the turbine, air is supplied via a mass flow controller. After the flow leaves the turbine, it passes through a valve which is followed by another flow meter which is connected to a vacuum pump.\(^1\) The vacuum is used to give the operator the ability to maintain thrust balance by lowering the turbine exhaust pressure.

The thrust bearings are supplied with air via pressure regulators to maintain a fixed supply pressure. The pressure and flow to each thrust bearing is monitored and recorded at all times. The journal bearing air requires more fine control of the pressure.

\(^1\) The two flow meters on the turbine can be used to determine how much of the journal supply flow is going out of the turbine versus how much is going out of the compressor.
As a result a pressure regulator followed by a metering valve is used. The two are set so that the pressure drop through the valve is much higher than the pressure drop through the bearing which makes the system maintain constant flow when perturbed and not constant pressure. The bearing mass flow rate, supply pressure and a bearing pressure tap within the die are all monitored and recorded.

Multiple pressure taps are also used to monitor the compressor inter-row pressure, the compressor pressure after the stators (which was designed in the die as the fuel supply for the engine), the compressor pressure at the outmost radial position of the die, the turbine supply pressure within the die, the turbine inter-row pressure and the turbine exhaust pressure.

Figure 4-1: Turbocharger flow path setup

The following figure is a schematic of the gas handling system that was used to implement the required fluidic setup:2

2 The details of the instrumentation used as well as an uncertainty analysis can be found in Appendix D.
Finally, the angular speed of the rotor is measured by a Philtec D6-H2TV high frequency fiber-optic position sensor. The sensor outputs a voltage that is proportional to the distance from the surface it is facing. This voltage is then fed to an HP87410A DC-10MHz spectrum analyzer, which converts the voltage signal into a rotational speed by identifying the maximum amplitude frequency. The sensor is positioned in the fiber optic access port (see Appendix C) overlooking the compressor and is held in place with a Newport 461 high-precision XYZ stage. The compressor is equipped with six speed bumps, one of which has a notch so as to act as a reference mark when processing data. Figure 4-3 portrays the operation of the sensor schematically.
In the future, hot flow tests and combustion tests are to be performed with a similar setup with the exception of the turbine supply flow. In the hot tests, the turbine supply flow would be heated by an electric heater upstream of the die. In the combustion tests, the turbine supply flow would be pre-mixed with hydrogen by employing an additional mass flow controller that would set the fuel flow rate to a fixed percentage of the main mass flow. All other connection in both testing schemes would be identical. Appendix C depicts the alternate packages required for such modes of operation.

**Figure 4-3:** Fiber optic operation schematic
4.2 Data Acquisition System

The turbocharger used a modified version of the data acquisition system that was originally setup by Frechette. A 300 MHz Dell Optiplex GX1 PC acquired and recorded the output of the pressure transducers, mass flow sensors, and a speed sensor (fiber optic). The pressures, mass flows, were sampled at 1000 scans per second per channel on a National Instruments PCI-6071E A/D board multiplexed to 64 channels. Every recorded data point is the average of 200 scans. All pressure transducers were connected to the A/D board as differential inputs. All mass flow meters were connected as single-ended inputs with common ground. The A/D system was originally programmed in LabVIEW by Frechette [1] but was modified substantially during the course of this research. Speed data was collected from the spectrum analyzer and was downloaded through a GPIB connection.

During rotordynamic experiments it was necessary to collect high frequency data from the fiber optic so as to better understand the orbits the rotor was following. As a result, a second Dell computer was used. Here the fiber signal was directly fed into a high frequency data acquisition card which could acquire data up to 3MHz. This system was programmed in LabVIEW by Jacobson [2]. The acquisition would begin whenever the user instructed the computer to begin acquiring data and would end after a pre-specified number of data points were collected. Typically, the number of data points that were to be acquired was set so as to allow for at least twenty rotor revolutions.

4.3 References


CHAPTER 5

TESTING - ROTORDYNAMICS

The biggest challenge to date in operating the turbocharger is stable operation at high rotating speed. As a result, a large fraction of the testing performed during this research focused on the rotordynamic performance of the devices. This chapter will discuss the rotordynamic experiments that were performed.

First discussion will focus on the geometric characterization of the devices through flow tests. Then the radial and axial stiffnesses of the devices are measured and compared to model predictions. Following this are experimental measurements of the rotor imbalance as well as experimental evidence that the rotor spins tilted at low thrust bearing pressures. Finally, a summary of all the high speed crashes is presented.

5.1 Assessment of Device Geometries

Assessing the geometry of test devices is crucial in establishing a connection between design, rotordynamic performance, and theory. The geometry was assessed in two ways. The first was based on measurements taken during the fabrication process. The second was based on static (non-spinning) flow tests, in which pressurized air was flowed through the device and the relation between the supply pressure and measured flow rate were used to assess the geometry being examined.

The second form of testing is a necessity as some dimensions of the device cannot be nondestructively measured. For example, the journal bearing length can be easily and precisely measured during the fabrication process but the journal width cannot. Four static flow tests were performed on all devices tested. One test on each of the thrust
bearings, one on the journal bearing and one for either the hydrostatic seal or the second journal bearing depending on whether the device being tested was a dual bearing or a single bearing device.

5.1.1 Thrust Bearing Geometry

For the thrust bearings, the objective of the testing was to estimate the effective diameter of the nozzles since only the length of the nozzles and the thrust bearing clearance could be precisely measured during the fabrication process. This measurement was performed by separately pressurizing each thrust bearing over a range of pressures from 0 – 830 kPa (120 psi) and measuring the corresponding flow rate. By employing a 1-D Compressible Flow Model [1], it was then possible to determine what the effective diameter of the nozzles was under the assumption that their diameter was constant along their length and that the rotor was bottomed out in the opposite direction from the pressurized thrust bearing. Figure 5-1 is an example of the results of such a test.
Figure 5-1: Static flow test results example for the front thrust bearing of die 2 from build locos 4. The fabrication measurements indicated that the length of the nozzles was 110 μm so the thrust bearing model [1] was run for that length for multiple nozzle diameters. The analysis concludes that this thrust bearing’s nozzles have an average diameter of 9.1 μm +/- 0.21 μm.

Once the model thrust bearing diameter has been adjusted so as to fit the experimental data, the data is consistent with the fit to within 4% over the entire pressure range. Furthermore, for this device, the model estimates that the nozzle effective diameter was 9.1 μm +/- 0.21 μm while the design value was 11 μm.

Table 5-1 lists the dimensions of the thrust bearing systems for the devices which spun to high speed. Also included is a prediction of the axial natural frequency and load carrying capability. A comparison between the predictions and experimental measurements will be presented in a later section for one of these devices.
Table 5-1: Thrust bearing geometry example - Build L4, Die 2

<table>
<thead>
<tr>
<th></th>
<th>C-2</th>
<th>C-3</th>
<th>C-8</th>
<th>L4-2</th>
<th>L4-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>FTB Length (µm)</td>
<td>123</td>
<td>123</td>
<td>123</td>
<td>108</td>
<td>108</td>
</tr>
<tr>
<td>FTB Diameter (µm)</td>
<td>10.7</td>
<td>12.3</td>
<td>12.1</td>
<td>9.0</td>
<td>9.6</td>
</tr>
<tr>
<td>ATB Length (µm)</td>
<td>87</td>
<td>87</td>
<td>87</td>
<td>111</td>
<td>111</td>
</tr>
<tr>
<td>ATB Diameter (µm)</td>
<td>10.2</td>
<td>10.5</td>
<td>10.8</td>
<td>9.3</td>
<td>9.1</td>
</tr>
<tr>
<td>Clearance (µm)</td>
<td>6.0*</td>
<td>6.0*</td>
<td>6.0*</td>
<td>6.1</td>
<td>6.0</td>
</tr>
<tr>
<td>Natural (rpm) @413.7 kPa / 827.4 kPa (@ 60psi/120psi)</td>
<td>865,300/1,118,900</td>
<td>872,700/1,147,300</td>
<td>853,100/1,116,500</td>
<td>811,700/1,044,300</td>
<td>831,500/1,072,900</td>
</tr>
<tr>
<td>Load Capacity (N) @ 0.25 eccentricity</td>
<td>0.42/0.79</td>
<td>0.49/0.94</td>
<td>0.46/0.90</td>
<td>0.37/0.64</td>
<td>0.43/0.75</td>
</tr>
</tbody>
</table>

*Checklist confirmed to be within specification +/- 0.25 µm but exact measurement is not available

In order to confirm that the rotor moved freely in the axial direction, an additional set of flow tests was devised [1,2]. In these tests one thrust bearing pressure was set to a fixed value and its flow rate was monitored while the pressure of the opposite thrust bearing was varied. By plotting the flow rate of the fixed pressure thrust bearing as a function of the pressure in the opposing thrust bearing, what came to be called “S-Curves” were created. Figure 5-2 is an example of such a test.
Figure 5-2: Thrust Bearing “S-Curves” for the entire build C (single turbine bearing design). The upper set of curves correspond to tests performed at a front thrust bearing pressure of approximately 145 kPa (21 psi) and the lower curves at a front thrust bearing pressure of 283 kPa (41 psi).

These curves were used in a qualitative way. More specifically, they were expected to be smooth curves as an indication that the rotor was freely moving and not cocking. Also, the flow rate should asymptote to zero as the pressure of the opposing thrust bearing is increased as an indication that the surface of the thrust bearing pad was smooth and flat and so could seal when the rotor was pressed upon it.

5.1.2 Journal Bearing and Seal Geometry

As was discussed in Chapter 2 the journal bearing design was initially isotropic and was later modified to introduce anisotropy by blocking four of the eight hydrostatic supply lines (hollow turbine vanes). Static flow testing procedures for the two designs differed. For the isotropic design, flow was supplied to the bearing through the eight supply lines. The flow was prevented from flowing through either the second bearing or the seal (depending on whether it was a single or dual bearing device) by blocking that
side of the die. The supply pressure was measured using the hydrostatic pressure tap inside the die. The pressure versus flow relation was then used in combination with the measured length of the bearing to estimate the bearing's width. This was done by employing a 1-D compressible flow with friction model which was discussed in Chapter 3. Figure 5-3 is an example of the output of the procedure.

![Figure 5-3: Static flow test results example for the turbine bearing of die 8 from build C. The fabrication measurements indicated that the length of the bearing was 280 μm so the journal bearing flow model was run for that length for multiple bearing width's. The analysis concludes that this journal bearing has a fluidic width of 14.7 +/- 0.66 μm.](image)

Again, the data functionally agrees with the model. As can be seen for this turbine bearing, which measured 280 μm long, the model indicates that the bearing width was 14.7 μm +/- 0.66 μm. Table 5-2 summarizes the journal geometries for the devices which were spun to high speed.
Table 5-2: Journal bearing geometry example - Build C, Die 8

<table>
<thead>
<tr>
<th></th>
<th>C-2</th>
<th>C-3</th>
<th>C-8</th>
<th>L4-2</th>
<th>L4-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Journal Length (µm)</td>
<td>280</td>
<td>280</td>
<td>280</td>
<td>291</td>
<td>291</td>
</tr>
<tr>
<td>Journal Clearance (µm)</td>
<td>15.3</td>
<td>15.2</td>
<td>14.7</td>
<td>17.4</td>
<td>16.3</td>
</tr>
</tbody>
</table>

Clearance +/-0.66 µm

For the anisotropic bearing design, the feed system was such that the bearing entrance pressure varied circumferentially. However, in order to assess the bearing clearance it was necessary for the bearing entrance pressure distribution to be uniform. This was achieved by altering the bearing feed system so that the feed line acted as a pressure tap and the bearing was supplied with air that came from the turbine and sequentially went through the turbine gap, the seal and into the compressor bearing. The pressure vs. flow data was then used as before to determine the bearing clearance.

The seal clearance (in the case that a seal existed) was measured during the fabrication process and static flow tests were performed only as means of confirmation. As the rotor was free to move axially, tests were conducted to measure the seal clearance with the rotor touching the front thrust bearing (max seal clearance) or the rotor touching the aft thrust bearing (min seal clearance). The axial position of the rotor was dictated by the pressurization scheme that was used.

---

1 Measured prior to releasing rotor when possible (so as to guarantee rotor was centered). If rotor was released it would be spun and then locked into position by employing the thrust bearings.
Table 5-3: Seal Geometry

<table>
<thead>
<tr>
<th></th>
<th>C-2</th>
<th>C-3</th>
<th>C-8</th>
<th>L4-2</th>
<th>L4-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Seal Length (µm)</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>200</td>
<td>200</td>
</tr>
<tr>
<td>Max Clearance - Metrology (µm)</td>
<td>12.5</td>
<td>12.5</td>
<td>12.5</td>
<td>10.4</td>
<td>10.4</td>
</tr>
<tr>
<td>Max Clearance – Fluidic (µm)</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>9.9</td>
<td>11</td>
</tr>
<tr>
<td>Min Clearance – Fluidic (µm)</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>5.2</td>
<td>5.5</td>
</tr>
</tbody>
</table>

5.2 Testing Procedure – Spinning for the 1st Time

Once the evaluation of the geometry of a device was completed with static flow tests, efforts shifted to spinning the rotor. Spinning a specific rotor for the first time was usually a lengthy process. The procedure followed was to pressurize both thrust bearing (to approximately 400 kPa) and the journal bearing (to approximately 1.5 kPa) and then supply air to the turbine at a low flow rate (approximately 0.04 g/s). At this point, the rotor would sometimes begin spinning.

In the more frequent case were the rotor would not spin, small bursts of air would then be supplied to the turbine so as to impulsively increase the torque acting on it and unlock it. At this point it was imperative that the rotor – which was under constant view from the microscope – showed some rotational motion. If this did not work bursts of air would then be supplied to the compressor exhaust to torque the rotor in the opposite direction. Finally, quick sequential bursts in the turbine and compressor exhaust would be tried as well as changing the journal supply pressure.

If the rotor still did not rotate, the thrust bearing pressures were altered so as to move the rotor axially in an effort to help unlock it and also to confirm that the thrust bearing flow rates still exhibited the “S-Curve” behavior which was measured during the static flow tests. The previous procedure would then be repeated until the rotor would turn for a few revolutions during one of the bursts of air. At that point the rotor was usually free to rotate and would be connected to the turbine supply air and spun at a low speed of approximately 3000 rpm for some time.
5.3 Assessment of Rotordynamic Stiffnesses

Both the radial and axial natural frequencies were experimentally observed and compared to model predictions. The next two sections discuss the methods developed to measure these frequencies and their comparison to model predictions.

5.3.1 Radial Stiffness

As was discussed in Chapter 3 the journal natural frequency is of order 10,000 rpm and is a function of the journal bearing supply pressure. Going to high speeds (order of $10^6$ rpm) involved crossing this natural and its experimental identification was deemed as a useful step in better understanding the operation of the device and as a means of tying the experiments to the modeling efforts.

During the course of this research, two separate methods were developed to experimentally determine the journal natural frequency. Both of these methods require that the rotor be spinning and data being collected while the rotor speed crosses the natural frequency. The first method relies on pressure vs flow relations for the bearing whereas the second method involves processing of the fiber optic signal\(^2\).

Based on the Jeffcott rotor analogy, at subcritical speeds, the rotor is expected to be spinning about its geometric center. As the speed is increased, the rotor is expected to start whirling, reaching a maximum whirl amplitude at the natural frequency (see Chapter 3). It is this idea that is exploited by both methods. The first method relies on the observation of a bearing supply pressure drop as the speed crosses the natural frequency. Given that rig’s bearing supply system is such that the flow rate to the bearing is constant, such a pressure drop is indicative of a reduction in the fluidic resistance of the bearing which in turn is consistent with an increasing whirling amplitude. Figure 5-4 depicts this pressure drop as the speed is increased for multiple journal bearing supply pressures.

\(^2\) This method was developed by Dr. Paduano
Figure 5-4: Measurement of journal natural based on pressure dip method. Data shown for single compressor bearing device build L4, die 2.

The second method relies on post processing data collected by the fiber optic as was outlined in section 3.6.2. This way, the physical whirling amplitude of the rotor can be directly determined. Such data is presented in Figure 5-5. Also overlaid on the figure is a Jeffcot rotor model fit to the data as well as scaled pressure data.
Figure 5-5: Measurement of journal natural based on fiber data. Also overlaid is the pressure drop data indicating that the two methods give the same result. Data shown for single turbine bearing device build C, die 8.

As can be seen in Figure 5-5, the behavior observed in the optical data agrees well with the simple Jeffcott rotor model. The coincidence of the peak in the optical data and the pressure data also lend credence to the hypothesis that the pressure dip is indicative of the rotor radial position. Using the pressure dip method the journal natural frequency for the single turbine bearing devices and the single compressor bearing devices was determined and is depicted in Figure 5-6 and Figure 5-7 respectively.
**Figure 5-6:** Summary of Build C turbine bearing natural frequency. Model prediction for a turbine bearing of dimensions 280x15 including the effects of an 11 µm seal followed by a 250x25 µm compressor gap is also plotted. The journal natural is the higher of the two lines. The lower line is believed to be a superharmonic response of the system.

For this design, the measured natural frequency appears to lie within 6% of the model predictions. Furthermore, it also interesting to note that these devices exhibited a second (superharmonic) resonance below the natural frequency. This resonance has been attributed to non-linear effects in the bearing system.
Figure 5-7: Single compressor journal bearing natural frequencies. This device includes a geometry designed to introduce anisotropy.

The single compressor design die measured for Figure 5-7 was fabricated with some journal bearing anisotropy present. This design was intended to have some anisotropy present. The model predictions and the data appear to functionally agree to within a few percent. Furthermore, it can be seen that both the predicted and measured levels of anisotropy are minimal (approximately 5%).

For this device, optical data was taken as well. In the first test, one fiberoptic located at 45° with respect to the two principal axes collected data, Figure 5-8a. The fiber detected two peaks in the whirl amplitude. Data was also taken with two fibers positioned in the direction of each of the two principal axes, Figure 5-8b. The data shows that each fiber senses only the whirl amplitude corresponding to its direction. The location of the peak amplitudes was identical to the previous test. Furthermore, the peak amplitudes were roughly 1.45 times that of the single fiber test. Based on geometry, they should have differed by a factor equal to the square root of two (1.41), providing further
confirmation that the fibers were observing the two natural frequencies. These tests were done at a journal supply pressure of 35 kPa (0.5psi).

Figure 5-8: Fiber data taken showing that the anisotropic compressor bearing build exhibited anisotropy. (a) Single fiber placed at 45° to each of the stiffness axes. (b) Two fibers placed in each of the stiffness axes directions.

3 Fiber data processed by C. J. Teo
Journal natural frequency data at higher journal supply pressures was generally not collected as it would require higher rotational speeds which would jeopardize the devices. The only set of such data is depicted in Figure 5-9 and as can be seen, the natural frequency appears to have a peak at a journal supply pressure of roughly 110 kPa. This is believed to be associated with the inception of choking in the bearing although an extensive study of the phenomenon was not undertaken.

![Figure 5-9: Extended single turbine journal bearing natural frequency. At pressures exceeding 110 kPa the natural frequency appears to start decreasing.](image)

5.3.2 Axial Stiffness

The engine was designed so that the bearing natural frequency is above the maximum rotor speed. The model predicts only a small speed margin, so experimental verification of the thrust bearing natural frequency is important.
During some of the single compressor bearing device tests an audible sound would be heard at thrust bearing pressures greater than 530 kPa. Spectrum analyzer data showed a frequency peak at about 15,000 Hz (900,000 rpm) when the sound was heard. This frequency was on the order of the thrust bearing natural frequency and appeared to increase as the thrust bearing pressures were increased and disappeared when the pressures were below 530 KPa – the pressure at which the model predicts that they should choke. Figure 5-10 presents peak frequency as measured by the spectrum analyzer data at different thrust bearing pressures while the rotor was stationary and while it was rotating with predictions of the thrust bearing model for the geometry of the specific device. The measured frequencies agree with those predicted by the thrust bearing model [1] to within 5 % and show a similar trend. An increase in the thrust bearing clearance from 6.1 μm to 6.3 μm in the model would be enough for the data to essentially fit the model predictions perfectly. It is also confirmed that the current thrust bearing design is only marginal for design speed operation.

Figure 5-10: Thrust bearing natural frequency
Modeling performed by Teo [3] indicated that at choked thrust bearing pressures there exists an axial instability which could excite the thrust bearing natural frequency. It is believed that this instability is what made the thrust bearing natural frequency appear on the spectrum analyzer.

5.4 Assessment of Rotor Moments

5.4.1 1st Moments

The expected rotor imbalance was determined in the previous chapter and was shown to be a function of the design, the fabrication sequence, and the fabrication tolerances. The fiber optic was then used to determine the actual imbalance of each device experimentally as was discussed in Section 3.6.2. The following table lists the imbalance measurement results and compares them to the model expectations

<table>
<thead>
<tr>
<th>Device</th>
<th>Imbalance Measurement</th>
<th>Bad Case (Rms)*</th>
<th>Medium Case (Rms)*</th>
<th>Good Case (Rms)*</th>
<th>Bearing Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Build C Die 2</td>
<td>5.5 μm</td>
<td>6.8</td>
<td>3.8</td>
<td>2.1</td>
<td>Turbine</td>
</tr>
<tr>
<td>Build C Die 3</td>
<td>6.5 μm</td>
<td>6.8</td>
<td>3.8</td>
<td>2.1</td>
<td>Turbine</td>
</tr>
<tr>
<td>Build C Die 8</td>
<td>5.0 μm</td>
<td>6.8</td>
<td>3.8</td>
<td>2.1</td>
<td>Turbine</td>
</tr>
<tr>
<td>Build L4 Die 2</td>
<td>0.8 μm</td>
<td>8.7</td>
<td>5.3</td>
<td>2.7</td>
<td>Compressor</td>
</tr>
<tr>
<td>Build L4 Die 3</td>
<td>3.8 μm</td>
<td>8.7</td>
<td>5.3</td>
<td>2.7</td>
<td>Compressor</td>
</tr>
</tbody>
</table>

* Bad, Medium and Good Case model expectations as defined in Chapter 3.

As can be seen, the measured imbalance levels of the single compressor bearing devices are lower than those of the single turbine bearing devices. Also it can be seen that the model expectations for the imbalance of each design under each of three possible scenarios, which were outlined in Chapter 3, indicate that the imbalance level for the compressor bearing design was expected to be higher than that of the turbine bearing
design. This discrepancy can be attributed to either pure chance (as there are very few samples) or an increase in the precision of the fabrication process.

5.4.2 Axial-Radial Coupling - 2\textsuperscript{nd} Moments

As will be discussed in Section 5.5.2 the thrust bearings experience an instability when the nozzles are choked. In order to avoid this, for some of the tests, the thrust bearing pressures were reduced to approximately 400 kPa. It is believed that two devices fatally crashed due to this operational change and Figure 5-11 depicts a possible reason. As can be seen, both the thrust bearing flow rates appear to be decreasing as the speed is increased. Ultimately the thrust bearing flow rates are reduced by almost 8\% whereas the thrust bearing supply pressure remained constant within 0.3\%. Such behavior suggests that the rotor is tilting either in a static or rotating frame.

![Figure 5-11: Thrust bearing flow rates decreasing with increasing speed due to 2\textsuperscript{nd} moment imbalance or a static tilt caused by the low thrust bearing supply pressure (\sim 410 kPa).](image)

There are two possible sources for such tilt. One involves the angle between the rotor principal axis and the rotor geometric axis which would cause the rotor to start a synchronous precession, whose angle increase as the speed increased. The second
possibility is that non-uniformities internal to the device are creating a static torque on the rotor which is causing it to spin tilted in a static frame.

Careful examination of the data reveals that the thrust bearing flow rate decrease is directly connected with increases in the journal bearing supply pressure (which was increased as the speed was increased). This points to the idea that the tilt is a static one and is induced by the hydrostatic air supply system (possibly by etch non-uniformities in the plenum itself). Flow calculation performed by Teo [3] indicate that the rotor tilted by as much as 0.07°. Such a tilt is not enough for the thrust bearing pads to touch the static structure but it is possible that this tilt could be introducing some instability by coupling the thrust bearings to the journal bearing.

5.5 Rotordynamic Stability Phenomena

5.5.1 Radial Instability

Radial instability of the rotor is hard to monitor and in general is only observed when a rotor crashes. This is unfortunate as crashes over 2% of the design speed have historically proven to be fatal. Sometimes prior to a crash there is some indication in the spectrum analyzer in the form of increased frequency content surrounding the speed frequency but this sometimes occurs at speeds which are well below the speed the device crashes at.

5.5.2 Axial Instability

During the testing of both of the anisotropic compressor bearing devices, it was observed that at thrust bearing pressures above 530 kPa (pressures at which the thrust bearing nozzles choke) the rotor would oscillate axially at its natural frequency, something which was analytically predicted by Teo [3]. Furthermore, at random times, the thrust bearing flow rates would start to increase and decrease in phase with each other at a relatively low frequency. This is suggestive of the rotor oscillating in an angular sense. The following figure shows such data.
Thrust Bearing Instability

Figure 5-12: Thrust bearing instability. (a) Thrust bearing flow rates seen to fluctuate before the rotor crashes. (b) Enlargement of flow oscillation region indicates that the flow rates are oscillating in phase. Data is taken every 0.2 seconds. The principal frequency is about 1 second.
At the inception of these oscillations an audible sound could also be heard. The reason why the frequency for such a light device is so low has not yet been understood and needs to be further studied. At this point however, it has been decided to avoid performing experiments at thrust bearing pressures over 530 kPa.

5.5.3 Summary of Crashes

A total of five devices have been spun to high speeds during the course of this research. The reason why higher speeds were not achieved is attributed to rotordynamic instabilities even though it is hard to say with certainty exactly how each device crashed. Table 5-5 summarizes the crash points for each of these five devices along with the model predictions of Liu [4].

Table 5-5: Stability boundary predictions for turbine bearing devices [4]

<table>
<thead>
<tr>
<th></th>
<th>C-2</th>
<th>C-3</th>
<th>C-8</th>
<th>L4-2</th>
<th>L4-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crash Point</td>
<td>250,000 rpm</td>
<td>310,000 rpm</td>
<td>390,000 rpm</td>
<td>480,000 rpm</td>
<td>210,000 rpm</td>
</tr>
<tr>
<td>Pred. Journal Stab. Boundary</td>
<td>431,000 rpm</td>
<td>882,000 rpm</td>
<td>622,000 rpm</td>
<td>&gt; design*</td>
<td>&gt; design*</td>
</tr>
</tbody>
</table>

*Assuming 5% anisotropy

All of the build C devices are believed to have crashed in the journal as there was no evidence that the thrust bearings had any problems. For build L4 the improved bearing design and fabrication combined with the presence of anisotropy is believed to have increased the stability boundary of the journal bearing and this time the crashes were caused from thrust bearings problems. For device L4-2 this is evidenced as erratic fluctuations in the thrust bearing flow rates prior to the crash point. In device L4-3 the thrust bearing pressures were lowered so as to avoid this instability but the result was that the rotor started tilting until it finally crashed.
5.6 Summary

The geometry of several turbochargers was characterized and used to predict their rotordynamic performance. Two separate methods were developed to determine the radial natural frequency. The first method relies on bearing pressures vs. mass flow variations as the speed of the rotor crosses the natural frequency and the other method\(^4\) relies on measurements taken by the fiber optic probe. Both methods give the same results and were used in experimentally identifying the radial natural frequency of the devices. The measurements compared favorably with model predictions. Furthermore, the rotor axial natural frequency was experimentally identified and also compared favorably with the models.

Measurements of rotor imbalance were made using the fiber optic and found to be close to the expected levels based on the fabrication capabilities. Furthermore, some experiments suggested that the rotor spins tilted at low thrust bearing pressures. The tilt level is consistent with expectations of the rotor’s misalignment between its principal axis and its geometric spinning axis. Finally, a thrust bearing instability was experimentally identified and associated with high thrust bearing pressures – pressures at which the thrust bearing nozzles choke.

5.7 References


\(^4\) Fiber optic method developed by Dr. Paduan
CHAPTER 6

TESTING - TURBOMACHINERY

This chapter details the turbocharger turbomachinery testing procedure, the test results, and analysis of the aerodynamic data.

6.1 High Speed Challenges

The objective of the turbocharger was to demonstrate high speed rotor dynamic operation, measure the aerodynamic performance of the components and thereby provide data to validate the turbomachinery performance prediction methodology. High speed rotordynamic operation proved challenging. These challenges can be classified into two categories: fabrication and operations. The following is a list of the pacing fabrication challenges that led to low yields of devices, none of which to date have met the design specifications:

- Etching precise high aspect ratio trenches, which define the bearing(s). These etches have an aspect ratio that is greater than 20 with a width of 15 +/- 0.5 μm.
- Bonding wafers with a misalignment of less than 2 μm so as to avoid introducing unacceptably high levels of imbalance on the two wafer rotor.
- Etching deep features (O(300 μm)) with a high level of etch uniformity (<3 μm across the 8 mm diameter rotor) so as to avoid introducing unacceptably high levels of imbalance.
- Aligning masks within less than 1 μm so as to avoid introducing unacceptably high levels of imbalance.
- Etching precise shallow etches (5.5 μm +/- 0.25 μm) to ensure correct thrust bearing performance.
- Etching precise nozzles (11 μm +/- 1 μm x 100 μm) to ensure correct thrust bearing performance.
- Etching multiple end point etches where all features being etched have to fall out at the same time so as to avoid damaging the underlying wafers.
- Hermetic bonding of six high fidelity wafers to produce non-leaking devices.

The result of these challenges was that the yield was a small yield. Specifically, a total of only six devices have been produced with a geometry that allowed them to spin to speeds above 200,000 rpm. An additional challenge associated with the fabrication was that techniques were not available to nondestructively characterize the geometry of critical features to the precision required. For example, it was possible to know the bearing length and the average equivalent fluidic width of a bearing but not the exact profile of a bearing (which typically varied from die to die) without sectioning the device. And even then, if the device had experienced a high speed crash the bearing was destroyed on impact.

From the operational side there were also several key challenges:
- Due to the small size of the device, instrumentation was limited and measurements of the axial and radial eccentricity of the rotor had to be inferred indirectly through flow measurements and/or fiber optic spectrum analyzer data.
- The dies as currently designed did not include an independent control for the thrust balance of the rotor which therefore required either pressurizing the inlet or applying a vacuum to the exhaust. This coupled the thrust balance with all the other settings.

Despite these challenges a turbocharger was spun to 480,000rpm. This was achieved by meticulously following a well structured strategy.
6.2 Test Procedure

The first part of the strategy involved the geometric characterization of the device being tested based on measurements taken during its fabrication and during static flow tests (as was described in Chapter 5). A rotordynamic characterization of the device would then follow where the imbalance and the journal natural frequency of the device were mapped out. At this point the device was ready to be spun to high speeds.

The first step for a high speed run was to set the compressor throttle to the desired setting so as to avoid having to change it while the rotor was spinning and possibly upsetting the stability of the journal. This was done by employing the CFD predictions linearly corrected for the as fabricated compressor blade height so as to ensure that the compressor data would lie in a useful part of the compressor map.

At this point, all the fluidic connections to the device were made with the exception of the vacuum to the exhaust. The thrust bearings were first pressurized to the desired pressure (825 kPa - 120psi) and ideally would not be changed throughout the entire test. Following this, the journal was supplied with air at a low pressure (350 kPa - 0.05psi). At that point the rotor was surrounded by a cushion of air in all directions and was free to rotate. The turbine main air was then turned on to a low flow setting (1mg/s - 500sccm) to initiate spinning at a subcritical speed (~2000 Rpm).

The next step was to take the device supercritical. Doing this required that the damping ratio was above a certain threshold that was dictated primarily by the imbalance of the rotor and the geometry of the bearing(s). If the damping ratio was below this threshold the rotor would crash while trying to cross the natural frequency due to high amplitude whirling. As a result, the journal bearing pressure differential was lowered to the lowest level that would allow stable subcritical operation so as to maximize the damping ratio (see Chapter 3). At this point the turbine main flow was abruptly increased so as to dash through the natural. The objective was to bring the rotor speed to several times (about 3-5x) the natural frequency.

With the device spinning supercritically, at a speed that would be limited to below 20,000 Rpm, the throttled vacuum was attached to the turbine exhaust. To achieve this, the throttle valve was first adjusted to match the flow rate of the turbine so that upon
connection a sudden thrust balance mismatch would not occur. After a successful connection the rotor would be spinning supercritically with an exhaust pressure that was very close to atmospheric. By closing off the vacuum throttle valve the turbine exhaust pressure would increase and push the rotor towards the front thrust bearing and by opening the vacuum throttle valve the turbine exhaust pressure would decrease and pull the rotor towards the aft thrust bearing.

At this point the rotordynamic map was used to follow a pre-specified speed-bearing pressure trajectory to reach higher speeds. The map had the bearing pressure differential on one axis and the speed on the other axis. On it were plotted the natural frequency, all integer multiples of the natural frequency and the predicted stability boundary. The strategy that was usually followed was to stay between the third and fourth integer multiples of the natural frequency until a bearing pressure differential of approximately 415kPa (6psi) when the speed would be increased with no further increases in the bearing pressure differential. This path was chosen to stay away from the stability boundary and was not placed any lower as it was empirically found that the rotor would not be stable at speeds that were less than two times the natural frequency. Figure 6-1 is an example of the rotordynamic map and the schedule that was followed during a high speed test.
Figure 6-1: Rotordynamic acceleration map. Operating schedule chosen to initially stay between the third and fourth multiple of the natural frequency.

Moving along the rotordynamic acceleration map involved changes in the speed, that were achieved by adjusting the turbine flow rate, and changes in the bearing pressure differential, that were achieved by adjusting the journal flow rate. The rotor axial position exhibited high sensitivity with respect to these parameters so all changes had to be small. For example, each of the steps that is seen in Figure 6-1 is composed of multiple small steps where the varying parameter of that step was changed slightly and the vacuum throttle was immediately changed to compensate for any thrust imbalance. The rotor’s axial position was estimated from the flow rate through the thrust bearings. Figure 6-2 depicts how the thrust bearing flow rates varied as changes were made.
As can be seen, the thrust bearing flow rates were constant between changes. When the journal bearing pressure was increased, the flow rate in the aft thrust bearing increased while the flow rate in the front thrust bearing decreased. This indicates that the rotor was moving forward due to increased forward force from the hydrostatic journal air supply plenum under the rotor. The vacuum throttle at the turbine exit was then adjusted to re-center the rotor. This procedure was repeated multiple times until the desired bearing pressure differential was achieved. Then the same thing was done while the speed was increased (which is depicted as a scaled turbine supply pressure rise in the figure). Figure 6-2 corresponds to a single step in Figure 6-1 and lasts more than 30 minutes. Because of the large number of interaction among manual settings, a typical high speed run required several hours.
6.3 High Speed Data

The results from the most successful tests are shown in Figure 6-3. The figure depicts results from the testing of two single turbine bearing devices (both from the same build) and two single anisotropic compressor bearing devices (both also from the same build). The two separate builds have different blade designs and blade heights as discussed in Chapter 2. A summary of the geometry of the devices tested is shown in Table 6-1.

Table 6-1: Summary of high speed devices

<table>
<thead>
<tr>
<th></th>
<th>Turbine Bearing Device</th>
<th>Comp. Bearing Device</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comp. Blade Height</td>
<td>225 μm</td>
<td>260 μm</td>
</tr>
<tr>
<td>Turbine Blade Height</td>
<td>245 μm</td>
<td>255 μm</td>
</tr>
<tr>
<td>Comp. Bearing Width (@R=4.1mm)</td>
<td>27 μm</td>
<td>16.3 – 17.4 μm</td>
</tr>
<tr>
<td>Comp. Bearing Length (@R=4.1mm)</td>
<td>250 μm</td>
<td>290 μm</td>
</tr>
<tr>
<td>Turb. Bearing Width (@R=3.0mm)</td>
<td>14.3-15.5 μm</td>
<td>50 μm</td>
</tr>
<tr>
<td>Turb. Bearing Length (@R=3.0mm)</td>
<td>280 μm</td>
<td>245 μm</td>
</tr>
<tr>
<td>Seal Mean Radius</td>
<td>3950 μm</td>
<td>3390 μm</td>
</tr>
<tr>
<td>Seal Clearance (centered rotor)</td>
<td>9-10 μm</td>
<td>7 – 8 μm</td>
</tr>
<tr>
<td>Seal Width</td>
<td>100 μm</td>
<td>200 μm</td>
</tr>
<tr>
<td>Thrust Bearings (DxLxg)</td>
<td>10.2-12.3 x 87-123 x 6.0</td>
<td>9.1-9.6 x 108-111 x 6.0-6.1</td>
</tr>
<tr>
<td>Imbalance (microns)</td>
<td>5.0 - 6.5</td>
<td>0.8 – 3.8</td>
</tr>
</tbody>
</table>
Compressor and Turbine Pressure Ratio

1.7
1.6
1.5
1.3
1.1

Turbine C-2 comp.

Comp. C-2

0.9 0.5 1 1.5 2

Speed (Rpm)

(a)

Turbine and Compressor Rotor Inlet Flow Rates

Flow Error +/- 0.0039 g/s

Turbine C-8 (Argon)

0.3 0.25 0.2 0.15 0.1

Flow Rate (g/s)

(b)
Figure 6-3: Summary of high speed data. Build C devices have a single turbine bearing. Build L4 devices have a single anisotropic compressor bearing. The two designs also have different turbomachinery blade shapes for both compressor and turbine.

Figure 6-1(a) depicts the pressure ratio variation with speed for the turbines (upper set of lines) and the compressors (lower set of lines) which, as expected, are functionally quadratic. It should be noted that the single turbine bearing devices were tested first and as can be seen the compressor pressure ratio appears to vary at some specific speeds. This is because during the early tests the compressor throttle was varied at predetermined speeds so as to create compressor maps. This procedure was not repeated for the later compressor bearing device tests because it was believed to increase the risk of fatally crashing the devices, since in this geometry adjusting the compressor throttle would indirectly change the bearing conditions. Instead, it was decided to run these devices to high speed at constant compressor throttle setting and then repeat the test at a different compressor throttle setting. This would result in two point compressor speed lines. Overall, the pressure ratio across the turbine appears to be substantially higher than that across the compressor but this is expected as the turbine air is cold and thus a higher turbine pressure ratio is required to achieve power balance.
Figure 6-3(b) depicts the mass flow variation with speed for the turbines (upper set of lines) and the compressors (lower set of lines) which as expected is functionally linear. It should be noted that for the single turbine bearing device C-8 which went to 390,000Rpm argon was used instead of nitrogen. In this specific test however the compressor was still pumping ambient air.

Figure 6-3(c) plots the data in terms of conventional compressor maps. For the single turbine bearing devices, the compressor throttle was varied while the rotor was spinning to create constant speed line data. For the single compressor bearing devices however, the constant speed line data consisted of only two points as the throttle was not varied during operation of the devices. It should be noted that the two designs had different compressor blades shapes as discussed in Chapter 2, which in combination with tighter throttle settings resulted in the substantially different flow rate capabilities evidenced in this figure.

At this point it is important to clarify how the compressor flow rate and pressure ratio were measured. The compressor flow rate depicted in the above figures is the flow rate that entered the compressor rotor inlet. The flow rate exiting the compressor was larger than this since it included the bearing air. The compressor rotor inlet flow rate was estimated using the following approach. First, the journal seal flow rate was determined by subtracting the turbine main flow rate from the turbine exhaust flow rate. This way the compressor journal flow rate could be determined from the total journal flow rate. The compressor journal flow rate would then be subtracted from the compressor exhaust flow rate and the rotor inlet flow rate would therefore be determined.

The pressure was measured by a tap at the compressor vane exit as illustrated in Figure 2-4. So, this measurement included the effects of the vane diffusion as well as any further pressure recovery which occurred due to the radially increasing flow area. The assumption in this is that the journal bearing flow rate did not have a significant impact on the pressure recovery of the vanes as it was only a fraction of the compressor rotor flow rate (23% at 480,000 rpm). Figure 6-3(d) compares the compressor pressure ratio across the impeller only to CFD predictions and it appears that results agree to within a few percent.
Figure 6-3(e) presents the turbine maps for each of the tested devices. The map is plotted using the turbine corrected flow rate based on the pressure prior to the turbine vanes normalized by standard reference pressure conditions (101300Pa) and the assumption of standard reference temperature conditions (288K). As can be seen in all tests the turbine has not yet choked. The turbine flow rates in the compressor bearing dies appear to be higher than those of the turbine bearing dies. This can be attributed partially to slightly larger blade heights and also the turbine blade re-design.

To summarize the data that was collected at the speed of 480,000 rpm, the compressor achieved a pressure ratio of 1.21 while the turbine was operating at a pressure ratio of 1.7. The turbine flow rate was 0.26g/s and the compressor rotor inlet flow rate was 0.14g/s. The journal flow rate was split between the compressor journal bearing and the seal at 0.036g/s and 0.008g/s respectively. The combined thrust bearing flow rates were only 0.005g/s.

### 6.4 High Speed Data Analysis

In order to assess the overall performance of the spool, the following definition of spool efficiency is introduced:

\[
\eta_{spool} = \frac{\text{Compressor Isentropic Power Out}}{\text{Turbine Isentropic Power In}}
\]

Eq. 6-1

Eq. 6-1 can be re-written in terms of pressure ratios and mass flow rates as follows:

\[
\eta_{spool} = \frac{m_c C_{pc} T_{12} \left( \pi_c^{(y-1)/\gamma} - 1\right)}{m_t C_{pt} T_{14} \left(1 - \pi_t^{(y-1)/\gamma}\right)}
\]

Eq. 6-2

This is simply a measure of the overall efficiency of the spool, defined as the ratio of fluidic power being produced by the compressor to the fluidic power being supplied to the
turbine. During all of the tests the temperature of the flow entering the compressor and the turbine was assumed to be at standard atmospheric conditions. Figure 6-4 depicts the spool overall efficiency for all the high speed tests.

Figure 6-4: Spool overall efficiency defined as fluidic power out of the compressor divided by fluidic power into turbine.

As would be expected, the spool efficiency increases with speed. This can be attributed to component efficiencies increasing with speed. The highest overall efficiency achieved was approximately 19%. The "kinks" in the curves are caused by the changing journal bearing flow rate which when increased would have the same effect as closing the compressor throttle. This would decrease the compressor flow rate, causing the compressor to operate further off design and thus reduced the compressor efficiency.
Due to the small size of the turbocharger it was not feasible to include thermocouples of sufficient accuracy to measure the exit air temperature of either the compressor or the turbine. This made it impossible to measure the efficiency of each component separately. Nonetheless, by modeling the bearing parasitic losses it was possible to determine a relation between the two efficiencies at each operating point. This procedure is outlined as follows: the first step is the power balance, the power delivered by the turbine must equal the power consumed by the compressor and all bearing and parasitic losses:

\[ \eta_t \left[ m_i C_p T_{1a} \left( 1 - \frac{\pi_{i}}{\gamma} \right) \right] = \frac{m_c C_p T_{12} \left( \frac{\pi_{c}}{\gamma} - 1 \right)}{\eta_c} + P_{\text{losses}} \]  

Eq. 6-3

Second, the parasitic power losses to the bearings, seal, and plena are computed as outlined in the parasitic loss model developed and presented in Appendix E. Finally, the relation of compressor and turbine efficiency that satisfy Eq. 6-3 is derived. This relation is plotted for four different speeds during the 480,000 rpm run in Figure 6-5 where each line corresponds to a single operating point in Figure 6-4.
Figure 6-5: Turbine/compressor efficiency relation based on parasitic bearing loss model at standard temperature.

Under the assumptions of the parasitic power loss model, Figure 6-5 depicts the relation between the turbine and compressor efficiencies. More specifically, for a given speed, in order for the power balance requirement to be satisfied the compressor and turbine efficiencies must lie on a constant speed line. It is interesting to note that at the highest speed achieved, if the efficiency of the two components was assumed to be the same the analysis indicates that the two efficiencies are approximately 48%. Considering that the efficiencies appear to be increasing with increasing speed, this result is encouraging.

Another way of examining the data is by plotting the power of each component as a function of speed. The following figure depicts the fluidic power being supplied to the turbine, the fluidic power being produced by the compressor, and the predicted parasitic power loss.
Figure 6-6: Power distribution in turbocharger. Fluidic power to turbine, power left over after parasitic losses, and fluidic power out of compressor.

As can be seen, at 480,000Rpm the turbine is being supplied with approximately 11 watts of fluidic power, the compressor is outputting approximately 2.2 watts and the losses are predicted to be approximately 1 watt.

6.5 Summary

This chapter presented the turbomachinery data for all the high speed runs. The data comes from a total of four devices from two separate designs – a single turbine bearing design and a single anisotropic compressor bearing design. In addition to rotordynamic differences, the designs had different turbomachinery blading, which is evidenced in the compressor and turbine maps.

The highest speed achieved was 480,000 rpm (205m/s tip speed) which is 40% of the design speed. At that speed, the compressor pressure ratio was approximately 1.21
with a mass flow rate of 0.13 g/s while the turbine pressure ratio was approximately 1.7 with a mass flow rate of 0.26 g/s. A parametric analysis of the component efficiencies shows that if both the compressor and turbine had the same efficiencies the experiments suggest it would be 48%. If there were no heat transfer to the compressor, such efficiencies are high enough for closed cycle operation of a gas turbine engine.
CHAPTER 7

DEMO ENGINE STARTUP TRANSIENT MODELING

This chapter presents a model designed to explore the behavior of the demo engine (not the turbocharger) during the startup transient. One use of the model is to develop specific ignition strategies that capture the changes in bearing and rotordynamic behavior as the combustor is ignited and the engine accelerated.

The model as implemented is a discrete time simulation which uses turbomachinery performance maps, the 1-D thermal dynamics of both the rotor and the static structure, structural deformations due to thermal and centrifugal loads, and the governing rotordynamics.

7.1 Ignition Setup and Operational Overview

For an ignition experiment, the engine will be setup as was discussed in Chapter 4. The bearing air will be supplied from external, independently controllable sources. To simplify the model, the engine is assumed to have a single compressor bearing with a seal preventing any journal flow from entering the turbine gap. The compressor flow path is internally connected to the turbine with the turbine receiving the sum of the compressor flow and the journal bearing flow.

The starting mechanism will rely on external pressurization of the compressor inlet and a turbine exhaust which will be exposed to the atmosphere. An external nozzle was not included in the setup as it would make it harder for the engine cycle to close (or
approach closing) since it would increase the exhaust total pressure to higher levels than atmospheric.¹

With the rotor stationary, the inlet is pressurized causing the rotor to accelerate. As soon as there is a power balance between the turbine and the compressor and all parasitic power losses the rotor stops accelerating. This is the cold flow equilibrium speed for the chosen inlet pressure. In the simulations that will be presented herein, this point will be chosen so that the rotor is supercritical with respect to its journal natural frequency. The transient which leads to this point will not be examined.

With the rotor spinning at its “cold flow” equilibrium speed, a fixed flow rate of hydrogen fuel is introduced to the engine flow path through the fuel injectors which are located after the compressor vanes. The igniter is then heated and the air/fuel mixture starts burning in the combustor. The ignition causes the pressures, the main flow rate and the rotor speed to change. The rotordynamics can also be affected by the ignition. It is this transient that the model assesses.

7.2 Assumptions and Boundary Conditions

There are several assumptions made in the model. The following is a list of all the assumptions:

1. Thrust bearing performance and flows are not included in the analysis.
2. Journal bearing flow goes entirely into compressor bearing
3. Fuel mass flow rate is not accounted for in the cycle analysis
4. Static structure is isothermal
5. Rotor heat conduction is modeled as 1-dimensional and non-isothermal
6. Heat transfer coefficients scale with the square root of the Reynolds number (assumes laminar flow)
7. Effects of heat addition on compressor are modeled as separate heat addition followed by compression [1]

¹ The inherent nozzle due to the right angle turn the turbine air makes in exiting the device is included as part of the turbine map.
8. Journal bearing flow assumes the static structure temperature by the time it reaches the bearing

9. Compressor map is based on CFD predictions [2] and turbine map is based on experimental results.

10. Rotordynamics are governed by the bearing pressure differential and geometry
    a. The pressure differential is computed using a compressible flow model
    b. The geometry is computed based on linear thermal expansions and centrifugal expansions computed by FEA [3]
    c. The stability boundary is computed by Liu's model [4]

The boundary conditions to the problem:
   1. Fixed inlet pressure
   2. Exhaust pressure fixed at a standard atmosphere, 101325 Pa
   3. Fixed journal bearing flow rate
   4. Fixed fuel flow rate
   5. Fixed ambient temperature for heat losses due to radiation, convection and conduction to standard temperature of 288 K.

The validity of the assumptions listed above will become more evident later in this chapter.

7.3 Cold Flow

The following section will describe the procedure followed to determine the cold flow equilibrium running point for a given inlet pressure. First, the compressor and turbine maps will be presented. Then the compatibility requirements along with some assumptions will be discussed followed by the determination of the equilibrium running point.
7.3.1 Compressor Map

The compressor map was constructed from 3-D steady CFD predictions [2] which were made for the current compressor blade design with blade spans of 220 µm for four different speed lines. An interpolation scheme used to compute other points on the map is outlined in Appendix F. An example of the results of the interpolation scheme for a speed of 955,000 rpm is depicted in Figure 7-1.

---

2 Experimental data was not used as the maximum pressure ratio achieved was only 1.21 and extrapolating to higher speeds could have possibly introduced large errors. As soon as compressor data at higher speeds becomes available, the experimental maps should be used instead of the CFD predictions.
Figure 7-1: Interpolation scheme example results. For a speed of 955,000 Rpm and resulting pressure ratio and efficiency for a corrected mass flow rate of 0.4g/s marked with a "**". The dotted lines are beta lines which are used for the interpolation procedure.

7.3.2 Turbine Map

For the turbine map a simpler approach was followed. Since experimental results were available up to a pressure ratio of 1.7 they were used instead of CFD predictions. When pressure ratios greater than 1.7 were examined a fifth order polynomial fit to the data was used to extrapolate. However, most simulations did not require such an extrapolation. Furthermore, the turbine map was assumed to be the same for all speeds. This is a typical assumption for turbine maps as the constant speed lines tend to lie very close to each other unlike the constant speed lines of the compressor map. This assumption is confirmed by the data, taken at two different compressor throttle settings, in Figure 7-2.
7.3.3 Cold Flow Equilibrium Running Point

Since the engine will be spun to an initial speed by pressurizing the inlet, it is necessary to compute the operating point for a given inlet pressure. The objective of this analysis is to determine the steady state cold flow equilibrium running point and not the transient that will get the engine there. This involves matching of the components which requires that the following three conditions be satisfied:

1. **Speed**: Compressor angular speed is equal to turbine angular speed
2. **Continuity**: Turbine flow rate is equal to sum of compressor and journal flow rate
3. **Power**: Turbine power is equal to sum of compressor power and parasitic losses

Some additional assumptions include that the flow temperature entering the turbine is equal to the flow temperature exiting the compressor and that there is a 3% pressure drop in the combustor [5]. The details of the matching procedure are outlined in Appendix G.
The results of the match include the pressures and flows through the engine as well as the angular speed of the rotor.

7.4 Ignition Transient

With the rotor spinning at some equilibrium speed, it is now possible to ignite the combustor. To do this, an externally controlled, fixed hydrogen fuel flow rate is supplied to the fuel injectors located after the compressor. The mass flow rate of the fuel is on the order of 1% of the total flow rate and its impact on the speed is not accounted for. With the fuel/air mixture flowing into the combustor, the combustor is ignited by applying a voltage to a thin wire that is packaged into the combustor.

As soon as ignition takes place, the pressures and flows instantaneously change whereas the speed starts to gradually increase. As the speed increases, further changes in the temperatures and pressures take place and continue to do so until the device has reached its new equilibrium state. It is the objective of this analysis to examine the details of this transient.

The problem is solved by discretizing time and computing how all the variables change during each of the finite time steps. The procedure that is followed during each time step is similar to the one outlined in the previous section with a few exceptions. First, $T_{4}$ is no longer the same as $T_{3}$. The value of $T_{4}$ is set by the fuel flow rate as well as the thermal boundary conditions of the combustor. Second, the power balance includes a term for the change in the angular momentum of the rotor. Third, $T_{21}$ is no longer at standard atmospheric temperature as heat is conducted to the compressor which heats the compressor flow. Fourth, the parasitic losses are now a function of the speed and temperature since the viscosity of the gases and the clearances are changing. Finally, the rotordynamics change since the bearing clearance and fluid temperatures change. The station numbering is indicated in Figure 7-3.
Figure 7-3: Engine Station definitions.

7.4.1 Thermal Analysis

Thermal analysis includes the heat transfer between the internal flow paths, the structures, and the ambient environment. The rotating and static structures are treated as separate problems with their own thermal boundary conditions. For the static structure the combustion process releases energy which is assumed to go into increasing the fluid temperature, increasing the static structure temperature, and losses to the ambient. For the rotor, heat is transferred from the turbine to the compressor via conduction through the rotor. The next two sub-sections will explain how each of the two models were set up.

7.4.1.1 $T_{t4}$ and Static Structure Temperature

The static structure was modeled as an isothermal mass which received heat from the combustor and rejected heat to the compressor flow, to the journal flow, and to the surroundings. The following figure depicts the basis of the model:
The flow enters the static structure after having been compressed by the compressor and is thus already at an elevated temperature $T_{13}$. As the flow travels around the combustor it receives heat from the static structure, $Q_{cold}$, reaching a temperature of $T_{135}$ before entering the combustor. As soon as the flow enters the combustor it is assumed that the temperature rises to $T_{14}$ and remains at that temperature until it reaches the turbine. The combustor is the source of heat to the static structure, $Q_{hot}$. Finally, the static structure loses heat to the surroundings, $Q_{losses}$ and to the journal flow. For the journal flow it is assumed that by the time it reaches the journal bearing it has assumed the static structure temperature.

A control volume analysis for the combustor is:

$$Q_{comb} = mC_p(T_{14} - T_{135}) + Q_{hot}$$  \hspace{1cm} Eq. 7-1

where $Q_{comb}$ is the energy released by the combustion process.
\[ Q_{\text{comb}} = \dot{m}_f \cdot h \]  
Eq. 7-2

where \( h \) is the lower heating value of hydrogen: 120,000 kJ/kg.

A similar control volume analysis is done for the flow path preceding the combustor:

\[ 0 = \dot{m} C_p (T_{135} - T_{13}) - \dot{Q}_{\text{cold}} \]  
Eq. 7-3

For the journal flow rate the following equation is used:

\[ 0 = \dot{m} C_p (T_s - T_{\text{amb}}) - \dot{Q}_{\text{journal}} \]  
Eq. 7-4

For the static structure, a control volume analysis yields the following result:

\[ m_s c_s \frac{dT_s}{dt} = \dot{Q}_{\text{hot}} - \dot{Q}_{\text{cold}} - \dot{Q}_{\text{losses}} - \dot{Q}_{\text{journal}} \]  
Eq. 7-5

which can also be written in discretized form as:

\[ T_{s,i+1} = T_{s,i} + \Delta t \left[ \frac{\dot{Q}_{\text{hot}} - \dot{Q}_{\text{cold}} - \dot{Q}_{\text{losses}} - \dot{Q}_{\text{journal}}}{m_s c_s} \right] \]  
Eq. 7-6

Where \( m_s \) is the static structure mass (computed to be 2.6 g for the current engine design), \( c_s \) is the specific heat capacity of silicon (700 J/kg/K), \( T_s \) is the static structure temperature, \( \Delta t \) is the time step, and \( i \) is the discretization index. With three equations and six unknowns, three more equations are needed to solve the system.
By employing heat transfer coefficient estimates, the values of each of the three heating terms can be computed in terms of temperatures. First, for $Q_{\text{hot}}$, the following relation is introduced:

$$Q_{\text{hot}} = (hA)_{\text{hot}} [T_{i4} - T_s]$$  \hspace{1cm} \text{Eq. 7-7}

where $(hA)_{\text{hot}}$ is extracted from an analysis done by Jacobson [6].

For $Q_{\text{cold}}$ the problem is a little more involved because the bulk fluid temperature is increasing along the flow path. Under the assumption that the static structure is isothermal, a control volume analysis yields the following result.

$$\dot{Q}_{\text{cold}} = (hA)_{\text{cold}} \left[ T_{i3} - T_{i35} \right] \left[ \ln \left( \frac{T_s - T_{i35}}{T_s - T_{i3}} \right) \right]$$  \hspace{1cm} \text{Eq. 7-8}

Finally, for the heat loss the device experiences to its surroundings, there are three separate sources: natural convection, conduction along the packaging tubes, and radiation [7].

$$Q_{\text{loss}} = Q_{\text{conv}} + Q_{\text{cond}} + Q_{\text{rad}}$$  \hspace{1cm} \text{Eq. 7-9}

Natural convection occurs on the top side of the device, the bottom side of the device and along its sides. The following equation states that the total natural convection is the sum of each of these:

$$Q_{\text{conv}} = (h_{\text{up}} A_{\text{up}} + h_{\text{dn}} A_{\text{dn}} + h_{\text{side}} A_{\text{side}}) [T_s - T_{\text{amb}}]$$  \hspace{1cm} \text{Eq. 7-10}
where $T_{\text{amb}}$ is the ambient temperature, assumed to be 300 K. The following equations list estimates for the heat transfer coefficients for each of the natural convection sources:

$$h_{\text{up}} = 1.32 \left[ \frac{(T_s - T_{\text{amb}})}{l_{\text{up}}} \right]^{0.25} \quad \text{Eq. 7-11}$$

$$h_{\text{dn}} = 0.49 \left[ \frac{(T_s - T_{\text{amb}})}{l_{\text{dn}}} \right]^{0.25} \quad \text{Eq. 7-12}$$

$$h_{\text{side}} = 1.42 \left[ \frac{(T_s - T_{\text{amb}})}{l_{\text{side}}} \right]^{0.25} \quad \text{Eq. 7-13}$$

Table 7-1 lists the parameters that were used:

**Table 7-1: Natural convection parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$l_{\text{up}}$, $l_{\text{dn}}$</td>
<td>23mm</td>
</tr>
<tr>
<td>$l_{\text{side}}$</td>
<td>2.9mm</td>
</tr>
<tr>
<td>$A_{\text{up}}$, $A_{\text{dn}}$</td>
<td>529mm$^2$</td>
</tr>
<tr>
<td>$A_{\text{side}}$</td>
<td>267mm$^2$</td>
</tr>
</tbody>
</table>

For conduction, the following equation was used:

$$Q_{\text{cond}} = \frac{kA_{\text{tubes}}}{l_{\text{tubes}}} [T_s - T_{\text{amb}}] \quad \text{Eq. 7-14}$$

Where $k$ is the thermal conductivity of the tubes, $l_{\text{tubes}}$ is the length of the tubes and $A_{\text{tubes}}$ is the sum of the cross-sectional areas of the tubes. Table 7-2 lists the parameters that were used:
Table 7-2: Conduction parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
<td>15.6 W/m/K</td>
</tr>
<tr>
<td>$A_{tubes}$</td>
<td>17 mm$^2$</td>
</tr>
<tr>
<td>$l_{tubes}$</td>
<td>10 mm</td>
</tr>
</tbody>
</table>

The tube area was computed based on the packaging scheme which requires a total of 16 tube connections.

Finally, the heat loss due to radiation was computed as follows:

$$ Q_{rad} = \varepsilon \sigma (A_{up} + A_{dn} + A_{side}) \left[ T_s^4 - T_{amb}^4 \right] $$

Eq. 7-15

Where $\varepsilon$ is the emissivity and was assumed to be 0.3 for silicon [7] and $\sigma$ is the Steffan Boltzmann constant ($5.67 \times 10^{-8}$ W/m$^2$/K$^4$). For the gas turbine engine, based on the expected wall temperatures, the heat loss is split almost evenly between the three heat loss mechanisms.\(^3\)

Once the mass flow rates of air and fuel are known along with an initial temperature of the static structure, a compressor exit gas temperature and an ambient temperature, the combustor exit gas temperature and the rise, during the discrete time step, in the static structure temperature can be computed.

### 7.4.1.2 $T_{\Omega 1}$ and Rotor Temperature

In the rotor, heat is conducted from the turbine to the compressor. This heat transfer is detrimental to the performance of the compressor as increasing the pressure of

\(^3\) In the combustor (tested by Spadaccini), the conduction losses are smaller as there are fewer packaging fluidic interconnects.
a hotter gas requires more power than that required for a colder gas (assuming same flow rate and pressure ratio).

As was suggested by Gong [1], the heat transfer to the compressor is modeled separately from the compression process. Specifically, it is assumed that heat addition increases the temperature of the inlet air from $T_{i2}$ to $T_{i2f}$ and the compressor performs as if it were simply ingesting the hotter air. The problem is reduced to simply computing $T_{i2f}$.

The rotor is currently made of silicon and the turbine is in direct contact with the compressor. Calculation of the Biot number based on CFD estimates of the heat transfer coefficients yield Biot numbers on the O(0.1) suggesting that the rotor can be accurately modeled as isothermal. However, in order to improve the performance of the cycle it has been suggested that in the future the turbine and compressor be thermally isolated from each other via a hollow shaft or some other means that would introduce a high thermal resistance between the two [8]. As a result, the rotor model was constructed so as to allow for the user to input the thermal resistance between the compressor and turbine.

The model involves solving the unsteady 1-D heat diffusion equation (Fourier’s Equation):

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}$$

Eq. 7-16

Where $\alpha$ is called the thermal diffusivity of the material and is defined as follows:

$$\alpha = \frac{k}{\rho c}$$

Eq. 7-17

In Eq. 7-17 $k$ is thermal conductivity of silicon, $\rho$ is the density, and $c$ is the specific heat capacity. The values of those parameters (used in solving this problem) are listed in Table 7-3:
Table 7-3: Silicon properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2330 kg/m³</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>700 J/kg/K</td>
</tr>
<tr>
<td>Conductivity</td>
<td>141.2 W/m/K</td>
</tr>
</tbody>
</table>

While solving Eq. 7-16 is simple in the case of symmetric boundary conditions by the method of separation of variables, the boundary conditions for the rotor problem are not symmetric and as a result, an explicit finite-difference method was used.

The rotor is treated as an infinite plate of finite thickness which is sliced into multiple layers. Figure 7-5 depicts the geometry:

Figure 7-5: Rotor 1-D finite element model.⁴

The plate’s thickness is that of the rotor excluding the blade thickness (namely 330+285+50 = 635 μm). The cross sectional area through which conduction occurs is

⁴ Heat transfer coefficients from the rotor were obtained from Sirakov [2]
computed as the area of a cylinder which has the same volume as the combined volume of the compressor and turbine disks while in the meantime has the same thickness:

\[
A_{\text{cond}} = \frac{\left(\pi R_c^2 h_c + \pi R_t^2 h_t\right)}{h_c + h_t}
\]

Eq. 7-18

In order to proceed, the heat transfer coefficients for the hot and cold sides are multiplied by the appropriate areas and then normalized by the conduction area. All of this is done so as to maintain the simplicity of a 1-D model without disregarding important 2D effects pertinent to the problem. Eq. 7-19 and Eq. 7-20 depicts the normalization:

\[
h_{\text{hot,1D}} = h_{\text{hot}} \left(\frac{A_{\text{hot}}}{A_{\text{cond}}}\right)
\]

Eq. 7-19

\[
h_{\text{cold,1D}} = h_{\text{cold}} \left(\frac{A_{\text{cold}}}{A_{\text{cond}}}\right)
\]

Eq. 7-20

The use of an explicit first order finite forward difference scheme implies that in solving the problem a time stepping procedure is followed where the temperature at each node can be computed using the temperatures of itself and other nodes at a previous point in time [9. As a result, no iterations are required. One drawback of this method is that the maximum allowable time step is limited by a numerical stability requirement:

\[
\frac{\alpha \Delta t}{(\Delta x)^2} \leq \frac{1}{2(1 + Bi)}
\]

Eq. 7-21

In the above equation, the Biot number is based on \(\Delta x\), or the thickness of each slice:
Bi = \frac{h\Delta x}{k} \quad \text{Eq. 7-22}

Also, the left hand side of the equation is also referred to as the Fourier number:

\[ Fo = \frac{\alpha\Delta t}{(\Delta x)^2} \quad \text{Eq. 7-23} \]

So as to not limit the size of the time steps of the entire transient analysis, for each time step of the discrete time simulation, multiple time steps are performed in the rotor temperature model.

The basis of the model is a simple energy balance for each slice:

\[ \text{Change in Internal Energy} = \text{Net Conduction} + \text{Net Convection} \]

For the interior nodes, that is nodes 1-5 which are not exposed to convective heat transfer, the following discretization is used:

\[ T^{i+1}_m = Fo(T'_{m-1} + T'_{m+1}) + (1 - 2Fo)T'_m \quad \text{Eq. 7-24} \]

where \( m \) refers to the spatial node location and \( i \) refers to the time step.

For the edge nodes, that is nodes 0 and 6, there is a convective boundary condition and their temperature is computed as follows:

\[ T^{i+1}_o = 2Fo(T'_{i} + BiT'^i_{\text{fluid}}) + (1 - 2Fo - 2FoBi)T'_o \quad \text{Eq. 7-25} \]

where the Biot number is based on the 1-D corrected heat transfer coefficients.
In order to accommodate variable internal resistance for the rotor, one of the internal slices is altered to have a lower $k$. This way the temperature gradient from one side of the rotor to the other side increases while the heat flux through the rotor decreases. Overall, this model requires an input of the time step, the initial temperature distribution in the rotor, the thermal resistance of the rotor, and the heat transfer coefficients on each side of the rotor. The output of the model is the temperature profile of the rotor after the time step as well as the heat flux that is flowing into the compressor flow path. The heat flux is then converted into an increase of $T_{12}$ to $T_{121}$ as follows:

\[ \dot{m}_c c_p (T_{121} - T_{12}) = \dot{Q}_{c,\text{rotor}} \]  

Eq. 7-26

### 7.4.2 Rotor Inertia

As was mentioned earlier, during the ignition transient, the net power between the turbine, compressor, and parasitic losses goes into increasing the angular momentum of the rotor. The net power itself can be computed as follows:

\[
P_{\text{net}} = m_t C_{p_t} T_{t4} \eta_t \left(1 - \left(\frac{P_{13}}{P_{14}}\right)^{\frac{\gamma_re}{\gamma_r}}\right) - \dot{m}_c C_{p_c} T_{121} \left(\frac{P_{13}}{P_{12}}\right)^{\frac{\gamma_c-1}{\gamma_c}} - k\mu\omega^2
\]  

Eq. 7-27

The net power is then converted into a torque as follows:

\[
\tau = \frac{P_{\text{net}}}{\omega}
\]  

Eq. 7-28

The torque is then used to compute the angular acceleration:

\[
\tau = I \frac{d\omega}{dt}
\]  

Eq. 7-29
where the moment of inertia is computed to be $4.4 \times 10^{-10} \text{ kg-m}^2$

Finally, the angular acceleration is converted into a change in the angular speed during the finite time step:

$$\omega^{i+1} = \omega^{i} + \Delta t \left( \frac{d\omega}{dt} \right)$$

Eq. 7-30

### 7.4.3 Structural Deformations

The only structural deformations which are accounted for in this model are those involved with the journal bearing. The reason for this is that the journal bearing rotordynamics are highly sensitive to the bearing geometry which is a 15 μm gap between two separate surfaces at a radius of 4100 μm.

There are two sources of change for the bearing geometry. The first is centrifugal loading of the rotor which varies with the square of the speed. At the design speed, FEA [3] indicated that the rotor would expand by 3 μm. As a result, the following relation was used to determine the bearing clearance change due to centrifugal loading as a function of speed:

$$\Delta C_{load} = -\left(3 \times 10^{-6} \right) \left( \frac{\omega}{\omega_{des}} \right)^2$$

Eq. 7-31

The second source of change for the bearing geometry is thermal expansions. The rotor and static structure do not necessarily have the same temperature at all times. As a matter of fact, the rotor which is lighter than the static structure heats up much faster. This thermal mismatch causes the bearing gap to initially decrease and then increase as the stator heats up. By employing the thermal expansion coefficient of silicon the following relation is developed to determine the bearing clearance change due to differing temperatures between the rotor and the static structure:
\[
\Delta C_{\text{temp}} = \alpha (4.1 \times 10^{-3} \mu m) (T_{\text{static}} - T_{\text{rotor}})
\]  
Eq. 7-32

where \( \alpha \) is the thermal expansion coefficient of silicon – \( 2.6 \times 10^{-6} / \text{K} \).

The overall bearing width change is the sum of the two effects and is computed as follows:

\[
\Delta C = \Delta C_{\text{load}} + \Delta C_{\text{temp}}
\]  
Eq. 7-33

The bearing gap is re-computed during each time step and is used in the rotordynamic analysis in order to determine whether stable operation can be maintained at all times.

### 7.4.4 Rotordynamics

As was discussed earlier in this chapter, with regards to the rotordynamics, only the journal bearing performance was modeled. The model assumes that the journal bearing flow rate, which is set externally, is constant throughout the ignition transient.\(^5\) It also assumes that there is no leakage flow through the seal and thus all of the flow supplied to the bearing system goes to the journal bearing.

During the transient several variables which affect the journal bearing’s performance change. First, the temperature of the bearing flow changes as the static structure heats up. More specifically, it is assumed that the journal flow temperature follows the static structure temperature. Second, the journal bearing clearance is changing due to speed changes and temperature differences between the rotor and the static structure. Finally, the journal bearing exit pressure is also changing as the rotor speed changes since the exit point is at the compressor rotor exit.

The above parameters are tracked during the transient, and as a result, given the assumption of constant journal bearing flow rate, it is possible to determine the evolution of the bearing supply pressure with respect to time. This is done by employing the 1-D

\(^5\) This could be achieved be employing a mass flow controller or a high pressure drop valve.
compressible flow with friction model developed in Chapter 3 which accounts for the effects of changing viscosity and changing geometric parameters.

With the bearing pressures, flow temperatures, and geometric parameters now known, the natural frequency is determined as was described in Chapter 3. Finally, an analysis developed by Liu [4] is employed to determine the journal stability boundary at all points in time. One of the major objectives of this analysis is to determine whether two conditions are met during the startup transient. Those are whether the speed ever surpasses the stability boundary and whether the speed ever drops below the natural frequency – both of which would have catastrophic results.

7.5 Model Validation

Without hot engine tests, a full validation of the model is not possible. However, a couple of comparisons to experimental data are shown to confirm that the model captures the governing physics. Furthermore, the model satisfies energy conservation on the entire cycle and, as will be shown later on the heat flux plots, on a component by component basis.

7.5.1 Angular Acceleration

In order to assess the rotor inertia analysis an experiment was performed where the rotor was spun to a fixed speed (~20,000rpm) and suddenly the turbine air, which was driving it, was removed. The rotor decelerated coming to a full stop within a couple of seconds. The speed trace was recorded and the following analysis was conducted to determine whether the modeling approach that was followed was appropriate.

The net torque acting on the rotor causes the rotor to decelerate. The magnitude of the deceleration is directly proportional to the ratio of the net torque acting on the rotor and inversely proportional to its angular inertia as was shown in Eq. 7-29. In this experiment, there are three sources of torque which are decelerating the rotor: parasitic losses, compressor windage losses and turbine windage losses. The model that was used in this analysis does not have the capability to include any windage losses and as a result the comparison between the two is only to see if there is functional agreement and also to
confirm that the model underpredicts the deceleration since it is not accounting for the additional decelerating terms.

The parasitic torque, was assumed to be of the form discussed in Appendix E resulting in the following equation for the rotor angular speed:

\[ k\omega = -I \frac{d\omega}{dt} \]  \hspace{1cm} \text{Eq. 7-34}

where \( k \) is constant derived from the geometry of the device and the fluid viscosity.

In the case that the rotor is initially spinning at a certain speed \( \omega_0 \), and the driving torque is suddenly removed, the solution to the above equation is:

\[ \omega = \omega_0 e^{\frac{kt}{I}} \]  \hspace{1cm} \text{Eq. 7-35}

Figure 7-6 depicts the experimental evolution of speed with respect to time. Also included on the plot is the model's prediction based on estimates of \( k \) and \( I \). Furthermore, a curve fit was performed using the same functional form as what the model suggests so as to determine whether functionally the model is correct and the equivalent value of \( k/I \) required to cause such a deceleration under the assumption of no windage losses.
**Figure 7-6:** Spindown test. Rotor was spun up to 20,025 rpm using turbine air which was then removed. Data depicts the speed decay due to parasitic losses and was obtained using a very simple filter. An exponential curve fit shows good functional agreement with the data.

As expected, the model overpredicts the time required for the rotor to come to a stop. Furthermore, functionally, the model appears to agree with the deceleration curve. The following table summarizes the parameters used for each of the two non-experimental curves:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Curve Fit Result</th>
<th>Actual / Expected Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega_0$</td>
<td>20,025</td>
<td>20,025</td>
</tr>
<tr>
<td>$k / I$</td>
<td>1.48</td>
<td>0.86</td>
</tr>
</tbody>
</table>
7.5.2 Combustion

During the simulation runs, the model tracks multiple temperatures, flows, and pressures in the device as a function of time. Among these results are predictions of the static structure temperature as well as the combustor exit gas temperature \(T_{\text{e}}\). The model suggests that at small flow rates it takes on the order of one minute for the static structure to reach its steady state temperature. For the small size of the device this appears unintuitive. As a result, transient startup data from the combustor experiments were examined and compared to the predictions of the model.

Furthermore, another interesting physical effect the model points to is that during the transient the static structure will be consuming a large fraction of the combustion energy making the combustor perform as if it had a lower thermal efficiency. The effect of this is that the combustor exit gas temperature is predicted to initially rapidly rise to a certain value and then gradually continue increasing as the static structure heats up. The exit gas temperature will stop increasing only when the static structure temperature has reached its equilibrium level.

Figure 7-7 depicts the comparison between the model predictions and transient ignition data from the combustor experiments:
Figure 7-7: Combustor experimental results overlaid with the model predictions for those settings. Experimental data from combustor experiments performed by Spadaccini [5]. Parameters: $m_{\text{air}} = 0.0196 \text{ g/s}$, $m_{\text{fuel}} = 0.000334 \text{ g/s}$.

As can be seen the data confirms the model predictions. The data indicates that it takes almost 2 minutes for the static temperature to equilibrate and more importantly confirms the prediction that the exit gas temperature initially jumps to a certain value and then continues to increase until the static temperature reaches its steady state value.

These predictions are critical to this model as they both have a great impact on the transient. The fact that $T_{14}$ experiences an initial jump followed by a more gradual increase is very beneficial as it makes the speed increase more gradual and also helps avoid surge (as will be explained later). Furthermore, the large time constant associated with the heating up of the static structure is directly related to the evolution of the bearing geometry.
7.6 Results and Ignition Acceleration Strategies

This section presents the results of several simulation runs. In all the simulations the rotor inlet is pressurized and the journal is being supplied with air from an external source. A fixed hydrogen fuel flow rate is also assumed. The first simulation examined serves as a baseline and will be used as an example to discuss some of the key phenomena which take place during the ignition process. Several other simulations will then be presented so as to analyze the effects of varying parameters and to suggest a feasible ignition strategy.

7.6.1 Simulation 1 - Baseline

For the baseline simulation, the following parameters were used:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure</td>
<td>34.5 kPa (5.0 psi)</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.003 g/s</td>
</tr>
<tr>
<td>Journal Flow Rate</td>
<td>0.040 g/s</td>
</tr>
<tr>
<td>Bearing Nominal Clearance</td>
<td>15.0 μm</td>
</tr>
</tbody>
</table>

The following figures depict the results of the simulation. The first figure depicts the transient operating line on the compressor map and the rest of the figures depict the evolution of the most important variables with respect to time. In all cases, the “cold operating point” is also included. For some of the variables as soon as ignition takes place, an instantaneous change occurs (i.e. combustion temperature, engine flow rate ...), whereas other variables respond more gradually to the ignition (i.e. rotor temperature, static structure temperature ...). The problem is dominated by three time constants, namely a time constant associated with the angular inertia of the rotor (O(0.2 s)), a time constant associated with the thermal inertia of the rotor (O(1 s)), and time constant
associated with the thermal inertia of the static structure ($O(30s)$). Since there is such a large variation in these time constants, two figures are presented for some of the characteristics – one in the 1 second time scale and another in the 1 minute time scale.
Figure 7-8: Simulation 1 - Compressor map(s). (a) Entire operating line. (b) Operating line zoom into a region where the corrected flow rate appears to decrease with time. (c) Operating line efficiency points.

Figure 7-9: Simulation 1 - Turbine flow rate.
Figure 7-10: Simulation 1 - Temperatures

Figure 7-11: Simulation 1 - Rotor heat flux
Figure 7-12: Simulation 1 - Static structure heat flux

Figure 7-13: Simulation 1 - Bearing clearance
The simulation reveals several interesting phenomena. First, examining the compressor map, Figure 7-8a, it can be seen that as soon as ignition takes place, the compressor corrected flow rate abruptly drops. This is due to the sudden temperature change in the combustor which causes the engine flow rate to drop as the turbine exhibits higher resistance to the lower density flow. This instantaneous drop in flow rate is depicted more clearly in Figure 7-9. During this change the rotor speed remains the same as the rotor has inertia and has not had time to accelerate. Furthermore, all of the temperatures except for the combustion temperature remain the same as they are dependent on the structure temperature which also has thermal inertia.

As soon as ignition occurs, the higher enthalpy flow into the turbine causes the rotor to accelerate. The specific acceleration path is depicted on the compressor map. As time proceeds, an interesting phenomenon is observed on the compressor map where both the corrected flow rate and the pressure ratio start decreasing and then continue increasing. The explanation for this is the difference between the rotor thermal time constant and the rotor inertial time constant. Specifically, as the rotor accelerates the compressor pressure ratio and corrected flow rate increase. The rotor stops rapidly accelerating after approximately 0.2 seconds at which point the compressor corrected flow rate starts decreasing since $T_{21}$ is still increasing because the rotor has not reached
its steady state temperature. It is thus evident, that the “backward loop” is caused by the rotor heat transfer effects. The reason why it can be seen on the compressor map is that the time constant for the rotor acceleration is smaller than the time constant for the rotor temperature increase. This difference in the time constants was also noted in the work of Chunmei Liu [10].

The next question that needs to be answered is why the trend of decreasing corrected flow rate and decreasing pressure ratio reverse again. The answer to this lies in yet another time constant – the static structure temperature time constant. The static structure surrounding the combustor is 30x more massive than the rotor and as a result it takes much longer for its temperature to equilibriate. As the static structure temperature is increasing, it is absorbing energy from the flow in the combustor and is thus lowering the combustor exit temperature. As the rate of increase in temperature of the static temperature decreases, the amount of energy being consumed by it decreases and thus the combustor exit temperature increases. Increasing combustor exit temperature has the effect of further accelerating the rotor and thus increasing both the pressure ratio and flow rate.

The ignition transient operating line of this device is fortunately shaped such that it avoids the surge line. In conventional engines, fuel supply control laws must be designed to avoid surge during acceleration but in this engine the heat capacity of the static structure achieves the same objective: a gradual increase of $T_{st}$ matched with increase in rotor speed.

Figure 7-10 depicts all of the relevant temperatures in the analysis. $T_{st}$ appear to have a small overshoot initially. This is caused by the fact that as the speed increases, the engine flow rate increases whereas the fuel flow rate is held constant and thus the combustion temperature decreases. After 0.2 seconds however the combustion temperature starts to gradually increase again. The driving force for this temperature increase, is the static structure temperature which can be seen to increase with $T_{st}$. In this
simulation \( T_{r4} \) ranges from roughly 800 K to 1300 K\(^6\). The static structure temperature begins at 300 K and asymptotes to roughly 800 K.

The rotor temperature is seen to start at 300 K reaching an initial steady state of 650 K within 1 second and continues to increase to almost 900 K - as \( T_{r4} \) is increasing. It should be noted that the cold side of the rotor is essentially at the same temperature as the hot side of the rotor - due to the high thermal conductivity of silicon. Accompanying its temperature increase, is an increase in \( T_{21} \) which almost reaches 500 K during steady state operation. The compressor exit temperature, \( T_{13} \) is also depicted. Its difference from \( T_{21} \) is the work done by the compressor which increases as the speed increases.

The driving force for the structure temperature changes are the heat fluxes which are depicted in Figure 7-11, and Figure 7-12. For the rotor, it can be seen how initially, the heat transfer on the hot side reaches a level of close to 80 Watts and then decreases, as the rotor initially heats up, to about 40 Watts. However, as \( T_{r4} \) starts rising again, the hot side heat transfer increases to almost 60 Watts. On the cold side of the rotor, the heat flux is initially zero since the rotor and compressor inlet flow are at the same temperature. As time proceeds, this heat flux gradually increases to match the hot side heat flux at which time the rotor temperature stops changing. The area enclosed between these two curves represents the energy that is absorbed by the rotor.

For the static structure, the case is similar to that of the rotor with the exception that the static structure has essentially three cold sides – the internal compressor cold side flow, the journal flow, and the external cooling due to convection, conduction, and radiation. An interesting fact to note is that the “cold side” of the static structure is initially actually cooling the flow instead of heating it. This is due to the fact that the rotor heats up much faster than the static structure and convectively transfers heat to the compressor flow path whose temperature is higher than the static structure for the first couple of seconds. The steady state heat flux is on the order of 100 watts of which only roughly 30 watts are being lost to the environment which are evenly distributed among the three heat loss mechanisms.

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\(^6\) Higher combustion temperatures could be achieved with higher fuel flow rates, however this could cause the rotor temperature to exceed its limit (≈1000K).
Figure 7-13 depicts the journal bearing clearance evolution with time. As can be seen, the bearing clearance starts off at 15 μm (the nominal turbocharger design) and asymptotes to 12-13 μm. However, during the first couple of seconds, the clearance drops to 10 μm. These are important changes in clearance. To better understand the source of the bearing clearance variation, overlaid on the plot is the projected bearing clearance in the absence of centrifugal loading. As can be seen, during the first few seconds, the largest fraction of the bearing shrinkage can be attributed to the thermal expansion differences between rotor and static structure. However, during the later part of the transient, most of the shrinkage can be attributed to centrifugal loading since the rotor speed further increases and the centrifugal expansion goes with the square of the speed.

Finally, the most important conclusion from this analysis is that this ignition strategy would fail due to rotordynamic instability. This can be seen in Figure 7-14 where the device is initially spinning at around 400,000 rpm with an infinite stability boundary. As ignition occurs, the stability boundary drops below the operating speed line at 0.1 seconds implying that the rotor would crash at this point. Nonetheless, it should be noted that as time further progresses, the stability boundary starts rising again but never reaches the rotor speed (~1,000,000 rpm).

The implications of this are profound. A good bearing for a cold flow low speed device is not a good bearing for a hot flow high speed device. With this in mind, a strategy will be proposed which will allow for successful ignition by changing externally controlled variables and/or the nominal bearing clearance.

For the assumed imbalance level (4 μm) and bearing length (330 μm), the optimal bearing clearance, i.e. that which gives the highest stability boundary, is 15 μm. As a result a design which keeps the bearing clearance as close to 15 μm at all times, but especially at the higher speeds, has to be developed. In this first simulation the bearing appears to have a clearance which is less than 15 μm at all times; so one approach would be to start with a larger bearing. This approach would not eliminate the root cause of the bearing clearance variations but could remove the symptoms and results.

Another approach is to reduce or eliminate the source of the bearing clearance variations. As was discussed earlier there are two sources which are responsible for the
changes. The first one is centrifugal loading which cannot be eliminated. However, if the objective is to only sustain an ignition transient, the effects of the loading could be removed almost entirely if the ignition was to take place at a lower speed. The second source is the differential thermal expansion between the rotor and the static structure caused by temperature differences. This problem is most prominent during the first few seconds since the rotor heats faster than the static structure. Removing this effect is challenging as the rotor is less massive than the static structure. However, two possible solutions which alleviate the problem are to ignite at a lower temperature or to externally preheat the device so that the temperature changes during the ignition transient are smaller. Finally, a third approach is to thermally isolate the compressor from the turbine so that its temperature does not change as the device ignites, and thus reduce the overall variation in the bearing clearance. The rotordynamic advantages of this scenario however are not as obvious.

To summarize the following solutions are suggested:

1. Change bearing initial clearance
2. Reduce $T_{4}$ during the transient
3. Reduce speed at ignition (only alleviates ignition problem)
4. Pre-heat device
5. Thermally isolate rotor (implications not as obvious)
6. Engineer thermal time constant of static structure

7.6.2 Simulation 2 – Baseline + Clearance Variation

The simplest approach, enlarging the nominal bearing clearance, is examined first. Specifically, everything is kept the same as the baseline simulation with the exception of the bearing clearance which is varied from 16 to 18 μm.

Table 7-6: Simulation 2 parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure</td>
<td>5.0 psi</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.003 g/s</td>
</tr>
<tr>
<td>Journal Flow Rate</td>
<td>0.040 g/s</td>
</tr>
</tbody>
</table>
**Figure 7-15:** Simulation 2 - Nominal Bearing Clearance 16 μm. (a) Bearing clearance. (b) Speed/ Stability
Figure 7-16: Simulation 2 - Nominal Bearing Clearance 17 µm. (a) Bearing Clearance. (b) Speed/Stability.
As can be seen from the above figures, increasing the initial bearing clearance makes the stability boundary go higher at the longer times but makes things worse at the
shorter times. Specifically, both the 16 and 17 μm initial clearance simulations indicate that the bearing will go unstable during the first second of the transient but will later become stable again. The 18 μm initial bearing clearance however, starts off unstable and becomes stable with 0.1 seconds, remaining stable until the end of the transient. Moreover, the approach of only varying the initial bearing clearance so as to maintain rotordynamic stability through the entire transient does not seem to be a viable option. This is further confirmed in the simulation for an initial clearance of 17.5 μm which is depicted in Figure 7-18.

**Figure 7-18:** Simulation 2 - Nominal Bearing Clearance 17.5 μm. Speed and Stability indicate that the journal is unstable at time zero and at higher times.

7.6.3 Simulation 3 – Baseline + Device Pre-Heating

Once the device has reached steady state high temperature operation, the temperature difference between the rotor and the static structure is small. The implication of this is that the steady state bearing clearance differs from the initial clearance mostly due to centrifugal loading. Furthermore, the thermal effects on the bearing dimensions mostly arise during the first few seconds of the transient when the rotor heats up prior to the static structure heating up. The effect of this temperature
mismatch is to cause a dip in the bearing clearance as well as the stability boundary (depending on the initial settings).

If the static structure was pre-heated to a temperature close to the steady state temperature, the transient temperature mismatches would be substantially reduced as will be shown in the following simulation. The initial temperature of the rotor and static structure was set to 600 K. To further optimize the simulation the bearing nominal clearance was set to 17 μm to allow for the expansion caused by centrifugal loading.

Table 7-7: Simulation 3 parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure</td>
<td>5.0 psi</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.003 g/s</td>
</tr>
<tr>
<td>Journal Flow Rate</td>
<td>0.040 g/s</td>
</tr>
<tr>
<td>Bearing Nominal Clearance</td>
<td>17 μm</td>
</tr>
<tr>
<td>Rotor/Static Struct. Initial Temp.</td>
<td>600 K</td>
</tr>
</tbody>
</table>
**Figure 7-19**: Simulation 3 ($C_o = 17$ μm) - Compressor map(s). (a) Entire operating line. (b) Operating line zoom into a region where the corrected flow rate appears to decrease with time but phenomenon is less pronounced than baseline case.

**Figure 7-20**: Simulation 3 ($C_o = 17$ μm) - Temperatures
As can be seen in Figure 7-20 the initial temperature of the rotor and the static structure are at 600 K. Furthermore it should be noted that the initial temperature overshoot of $T_{14}$ is still present but the secondary rise in its value, which is attributed to the increasing static structure temperature is much lower than the baseline design. This is because in this case the static structure which is warmer consumes less energy from the
fuel during the startup transient, thus increasing the initial value of $T_t$ and making the secondary rise in its value smaller. It should be noted however that the steady state value of $T_t$ is the same as the baseline case.

The original objective of this simulation was to remove the dip in the bearing clearance associated with the different time constants for the heating up of the rotor and static structure. Figure 7-21 clearly portrays how the objective has been achieved. There is still a small dip which could in theory be entirely removed if the structure was pre-heated up to its steady state value. The implications of having removed the bearing clearance dip can be seen in Figure 7-22 where it is evident that stable operation is maintained throughout the entire transient.

Examining Figure 7-19, another interesting aspect of the simulation arises. The operating line still exhibits a kink but the kink is located much closer to the final operating point. This is due to the fact that $T_t$ experiences a much smaller secondary rise since the static structure needs to be heated up much less than the baseline case. One drawback of this is the fact that the operating line now lies closer to the surge line throughout the transient. Whether this is acceptable depends on the surge margin requirements.

### 7.6.4 Simulation 4 - Baseline + Thermally Isolated Compressor

A final simulation was run to examine the effects of thermally isolating the compressor from the turbine. It is assumed that a thermal resistance exists between the turbine and the compressor of 100 times more than that of the current device. Furthermore, it is assumed that the expansion of the compressor is not affected by the turbine and is solely determined by the compressor temperature and centrifugal loading. The rest of the settings were kept the same as the baseline design with the exception of the fuel flow rate which was increased to 45 mg/s so as to give the same steady state $T_t$. 
Table 7-8: Simulation 4 parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Pressure</td>
<td>5.0 psi</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.0045 g/s</td>
</tr>
<tr>
<td>Journal Flow Rate</td>
<td>0.040 g/s</td>
</tr>
<tr>
<td>Bearing Nominal Clearance</td>
<td>15 µm</td>
</tr>
<tr>
<td>Rotor/Static Struct. Initial Temp.</td>
<td>288 K</td>
</tr>
<tr>
<td>Compressor Thermally Isolated</td>
<td>Yes (0.99)</td>
</tr>
</tbody>
</table>

Figure 7-23: Simulation 4 – Compressor map operating line
Figure 7-24: Simulation 4 - Temperatures

Figure 7-25: Simulation 4 – Bearing Clearance
In this simulation, it can be seen that the steady state speed of the rotor is higher than what it was in the baseline design, and as a matter of fact is equal to the design speed. Since the combustion temperature is roughly the same, this is attributed entirely to the improved performance of the compressor which is pre-heating the flow to a lesser extent. This is depicted in Figure 7-24 where the compressor cold side temperature is roughly 600 K instead of 900 K that it was in the baseline simulation. The result of this is a much lower $T_{c21}$, below 400 K instead of close to 500 K. The price that is paid for this is a hotter turbine as less heat is being extracted from it, requiring it to operate at a higher structural temperature, namely 1100 K instead of 900 K.\footnote{1100K is above the limits of silicon. As a result for such an ignition silicon carbide would have to be introduced in the rotor or a cooling scheme should be used.}

It should also be noted that the operating line enters the surge region. This is due to the high values of $T_{t4}$ during the first second. The reason why $T_{t4}$ goes so high is that there is more fuel energy due to the increased fuel flow rate used in this simulation. The high value of $T_{t4}$ at these transient low speeds pushes the operating line into surge as imposed by the matching of the compressor to the turbine. This implies that this ignition strategy would fail.
The next important thing to notice is that the steady state bearing clearance is roughly the same as the initial clearance with a large dip during the first second. This time the dip is due almost entirely to the rapid increase of speed which is later overcome by the thermal expansion of the static structure. The implications of this are seen in Figure 7-26 where the bearing appears to go unstable for a certain region of the transient. Physically, no unstable region can be sustained for any period of time, so this simulation concludes without success.

Nonetheless, successful ignition of device with a thermally isolated compressor is possible. A possible solution to alleviate the rotordynamic constraints would be to preheat the device as was shown in the third simulation. With regards to the compressor surge it would be possible to alleviate the problem by igniting with a lower fuel flow rate.

7.6.5 Other Strategies

The above sections presented several simulations and outlined some of the key phenomena governing the ignition transient. It should be noted however that a fully comprehensive study of all possible strategies with all possible settings could not be presented in this document. Nonetheless, it should be pointed out that an important additional lever to a successful ignition is the initial speed, prior to ignition, which is set by the inlet supply pressure. By lowering this initial speed the variation of the journal bearing clearance can be reduced as the angular speed through the entire transient will be smaller and thus the impact of centrifugal expansions will be less damaging to the rotordynamic stability of the system.

Additional strategies could involve changing the journal supply flow rate and/or the fuel supply flow rate. It is even possible to consider introducing a fuel control system as the time constants associated with the transient are relatively large.

7.7 Summary

This chapter examined the engine startup problem and presented a model developed to simulate the engine startup transient. The model relied on a CFD compressor map and an experimental turbine map. Thermally, the rotor was modeled as
separate from the static structure with both of their temperatures governed by the heat flux from the fluids and conduction, convection, and radiation to the environment. The effect of heat transfer to the compressor was modeled as separate heat addition prior to the compression process [1]. For rotordynamics, the bearing clearance variation with time due to thermal expansions and centrifugal loading was included. A stability boundary analysis was also performed by employing a 1-D compressible flow with friction model in combination with a stability model developed by Liu.

The results of the simulation revealed several interesting aspects of the ignition transient which can be attributed to the fact that the problem is dominated by three different time constants: a rotor inertial time constant, a rotor thermal time constant, and a static structure thermal time constant. For the simulations examined, the rotor inertial time constant was on the order of 0.3 seconds, the rotor thermal time constant was on the order of 1 second, and the static structure thermal time constant was on the order of 30 seconds. These different time constants substantially increased the complexity of the ignition problem.

First, the mass of the static structure, which contains the combustor, is a heat sink during the transient, effectively reducing the combustor exit gas temperature. The effect of this is that for a given fuel flow rate, $T_{14}$ initially experiences a sudden increase and then continues to gradually increase until its steady state value is reached. Since the rotor inertial time constant is much smaller than the time constant associated with the temperature rise of the static structure, a kink in the operating line on the compressor map is created which helps avoid surge during the transient. This eliminates the need for a fuel control strategy which is typically used to avoid surge due to an abrupt increase in $T_{14}$.

Second, the different time constants associated with the rotor inertia and the rotor-temperature increase cause the operating line to reverse directions on the compressor map for approximately one second. This is due to the performance deterioration associated with the rotor’s heating after its speed has increased. The effect of this however, is quickly overcome by the increase of $T_{14}$. The operating line changes direction again and continues on an upward trend.
The third and most important conclusion of the analysis is that the rotordynamics are greatly affected by the transient. More specifically, bearing stability is highly dependent on the journal bearing clearance which varies substantially during the ignition process. The bearing clearance is driven by two parameters: the rotor speed and the temperature difference between the rotor and static structure. Clearly, all three time constants are involved in this. The implications of this were that a bearing designed for cold operation could not maintain stability through the ignition process. The model concludes by suggesting modifications to the bearing design and a possible ignition strategy which would allow for a successful ignition. More specifically, the bearing clearance is to be enlarged slightly and the entire device is to be pre-heated prior to ignition.

7.8 References


CHAPTER 8

OPERATING SPEED LIMITATIONS AND IMPROVEMENTS

This chapter outlines some of the constraints in turbocharger design and construction that have impacted test behavior to date and suggests design approaches to achieve full speed operation.

8.1 Rotordynamic Design Space

The turbocharger’s success is contingent on high speed rotation and high speed rotation relies on good rotordynamic performance. The rotordynamic performance in turn is dictated by the design of the bearing system and the fabrication realization of that design, especially the manufacturing tolerances. Figure 8-1 depicts constraints that make high speed operation challenging and demonstrates the effect fabrication tolerances have on these constraints.
Figure 8-1: Turbocharger constraints imposed by fabrication tolerances. Each constraint requires that the speed be to the left of the line.

Several limitation can be seen in the figure. At low speeds (around 200,000 rpm) there are two boundaries associated with the thrust bearing load carrying capabilities without thrust balance. These boundaries represent the maximum achievable speed, using the standard operational protocols described in Chapter 6, if the thrust bearings were the only source of thrust balance\(^1\). The calculation was done under the assumption that the thrust bearings can operate at a maximum eccentricity ratio of 0.2\(^2\). The two curves separately indicate the effect of fabrication tolerances in the thrust bearing clearance and in the thrust bearing nozzle diameters. These boundaries are clearly well below the design speed and also depict the negative impact of the fabrication tolerances. Nonetheless, this constraint has been overcome through operational changes by applying

\(^1\) Thrust load comes from the turbomachinery (turbine pressures are higher than compressor pressures because the turbine flow is cold and does not have the design enthalpy) as well as the journal bearing pressurization.

\(^2\) Typically, thrust bearing eccentricity ratios of up to 0.4 are used however a tighter tolerance was imposed due to possible rotor tilting and/or principal axis concerns.
a throttled vacuum to the exhaust of the turbine so as to relieve the thrust bearings from carrying any thrust loads.

The thrust bearing critical frequency lies close to 900,000 rpm. The design intent has been to always operate below the natural frequency but, the current design does not achieve this unless the thrust bearing nozzles are choked. There are several solutions to this. One solution is to increase the operating pressure of the thrust bearings, to levels that would choke the nozzles. The disadvantage of this scheme is that there is an instability associated with choked operation and as a result it is not advised [1]. Another solution involves the use of different gases which would provide adequate levels of stiffness prior to choking the nozzles but could have a negative impact on combustion. Finally, the thrust bearing system could be re-designed to achieve the necessary stiffness at pressure levels which do not choke the nozzles. This is the recommended option and will be discussed further in the thrust bearing section. It should also be noted that the fabrication tolerance have a clear effect on these boundaries as well.

The journal bearing stability boundaries are then plotted for three separate cases so as to determine the effect of journal bearing clearance tolerances. One case is for a presumed rotor imbalance of 4um, and the other two cases for an imbalance of 4 +/- 1 um, and an imbalance of 4 +/- 2 um. The most important thing to note is the high sensitivity the stability boundaries have to the tolerance of the journal bearing clearance. For the case of 4um +/- 1um imbalance, the clearance tolerance for design speed operation is +/-0.5um. For the same levels of imbalance if the clearance tolerance was to be +/- 1um the maximum achievable speed would be below 800,000 rpm. It is evident that the journal bearing stability boundaries are very sensitive to the fabrication tolerances unless anisotropy is introduced to the design as has been done.

Finally, close to 730,000 rpm is a boundary associated with the rotordynamic impact of the structural deformations of the rotor due to centrifugal loading. As the rotor speed increases the bearing clearance shrinks. The implications of this are that a bearing fabricated with the required clearance for high speed operation would only go to 730,000
rpm. This problem however has been overcome by fabricating a non-optimal bearing that becomes optimal as the speed increases.\textsuperscript{3}.

To summarize, fabrication tolerances are a key driving parameter for design speed operation. Achieving the design speed involves maintaining very tight fabrication tolerances and a change in the thrust bearing system design or operational procedure. Further, thought should be given to a thrust balancing scheme so as to avoid the need of applying a vacuum to the exhaust.

### 8.2 Radial-Axial Coupling

The rotordynamic design of the turbocharger to date has not included the effect of possible rotor tilting or rotor precessions. Rotor synchronous precessions could be caused by a misalignment of the rotor principle axis from the rotor axial geometric axis. Models indicate that the angle between these two axes is expected be below 0.04\(^{\circ}\), based on estimates of fabrication tolerances. Such angles are smaller than the maximum tilting angle available in the thrust bearings (0.14\(^{\circ}\)) and as a result it is suggested that the rotor should be allowed to spin about its principal axis instead of its geometric axial axis.

However, the rotor is also exposed to moments in the stationary frame. Such moments could arise from geometrical asymmetries such as a circumferential variation in the hydrostatic plenum depth or a variation in the turbine feed plenum. The only restoring torque to such moments currently comes from the thrust bearings. A dual journal bearing system can also provide a restoring torque but it is almost 100 times smaller than what the thrust bearings can provide (due to the small moment arm between the two journal bearings).

If such moments exceed the maximum angular load capability of the thrust bearings a crash is unavoidable. To assess this possibility, a simple analysis was pursued which compared possible static torques to the thrust bearing angular stiffness.

\textsuperscript{3} As the speed increases the rotor not only expands radially but it also starts deforming axially ("umbrella mode"). This tapers the bearing and the rotordynamics implications of this have not been rigorously examined.
Experimental evidence indicated that the rotor tilting was directly coupled to the journal bearing supply pressure. The journal bearing plenum located underneath the compressor rim is fed by a symmetric set of feed lines. As a result, it was assumed that the primary source of any static torque would be due to variations in this plenum’s depth.

The anisotropic stiffness model was then employed to compute the torque that would result if half the plenum was the design depth less 1, 2, or 3 μm and the other half of the plenum was the design depth plus 1, 2, or 3 μm deeper with a supply pressure of 6.9 kPa (1 psi). The resulting torque was then compared to the thrust bearing angular stiffness – 0.095 N-m/radian [1] – so as to estimate the angular tilt that it would be induced. The results are depicted in Table 8-1.

Table 8-1: Tilt Angle Due to Etch Variations of Hydrostatic Plenum (@ 6.9 kPa)

<table>
<thead>
<tr>
<th>Plenum Depth Variation</th>
<th>Torque (N-m)</th>
<th>Angle (degrees)</th>
<th>% Max (0.14°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>+/- 1 μm</td>
<td>5.8 E - 6</td>
<td>0.0035</td>
<td>2.5%</td>
</tr>
<tr>
<td>+/- 2 μm</td>
<td>1.2 E - 5</td>
<td>0.0072</td>
<td>5.1%</td>
</tr>
<tr>
<td>+/- 3 μm</td>
<td>1.7 E - 5</td>
<td>0.011</td>
<td>7.8%</td>
</tr>
</tbody>
</table>

As can be seen, the torque introduced by possible etch variations appears to be too small to bottom out the thrust bearing in an angular sense. To confirm this result, the following model is put together where it is assumed that half of the compressor rim overhanging the turbine is exposed to a certain pressure and the other half is not exposed to any pressure (a physical impossibility but a check of the limits).

Table 8-2: Tilt Angle Due to No Pressure on One Side of Rim

<table>
<thead>
<tr>
<th>Pressure</th>
<th>Torque (N-m)</th>
<th>Angle (degrees)</th>
<th>% Max (0.14°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>689 Pa (0.1 psi)</td>
<td>2.0 E - 5</td>
<td>0.012</td>
<td>8.6%</td>
</tr>
<tr>
<td>6894 Pa (1 psi)</td>
<td>2.0 E - 4</td>
<td>0.12</td>
<td>86%</td>
</tr>
</tbody>
</table>

The above results indicate that angular stiffness of the thrust bearings is very large compared to possible internal static torques and that it is very unlikely that the thrust
bearings could bottom out angularly. However, further work should be conducted to assess if the tilt of the rotor can indirectly introduce any instabilities. For example, if the thrust bearing are not perfectly symmetric, rotor tilting is bound to introduce a radial side load on the rotor as suggested by Jacobson [2]. Clearly, if this side load is large enough it would be possible to radially crash the device.

8.3 Suggested Rotordynamic Design Changes

The following section suggests design changes in each of the rotordynamic components to help achieve design speed operation.

8.3.1 Thrust Bearings

The thrust bearings currently face two challenges. The first is to be able to provide an axial natural frequency (at unchoked nozzle conditions) which lies above the design speed and the second is to provide adequate angular stiffness.

Figure 8-2 depicts the effect of reducing the thrust bearing clearance on the axial natural frequency\(^4\). In the current design the clearance is 6 \(\mu\)m and the choke point occurs at approximately 530 kPa. At this supply pressure it can be seen that the maximum natural frequency occurs at a clearance of approximately 4.75 \(\mu\)m and its value is still below the design speed. However, as the thrust bearing clearance is reduced the pressure required to choke the nozzles also increases. If this effect is taken advantage of the axial natural frequency can be easily taken to levels above the design speed by reducing the clearance to below 5 \(\mu\)m combined with increased thrust bearing supply pressure.

\(^4\) Employed thrust bearing model developed by Jacobson.
With regards to the angular stiffness of the thrust bearings preliminary calculations indicate that it is very large when compared to possible internal static torques and further work as indicated in Section 8.2 should be pursued to decide whether it is necessary to increase it. If it is found that it needs to be increased, the modifications suggested for increasing the axial natural frequency should introduce some improvement. However, if further increases are required it may be necessary to increase the thrust bearing pad diameter or even move to a doughnut design which places the thrust bearings at a larger radius. Such a design would require that the blades be shrouded, a substantial design change, which would also increase the thrust bearing drag, but nonetheless necessary for the later inclusion of a generator.

8.3.2 Journal Bearing

The current journal bearing design has very tight tolerances on the 16um clearance (+/- 0.5um) which limits the yield and thus the rate at which devices can be fabricated. Relaxing these tolerances would be very beneficial. Possible schemes involve the introduction of anisotropy (which has already been pursued), but there are some challenges associated with this scheme as will be discussed in the next section.
Another approach would be to actively change the imbalance level after the bearing has been fabricated. Such an approach would involve dynamic balancing which could be achieved by employing a laser to remove material from the rotor. The benefits of such a scheme would be to increase the clearance tolerance up to +/- 2-3 um (even without the inclusion of any anisotropy).

Consideration should also be given to whether the design should involve a single journal bearing or a dual journal bearing. The dual journal bearing design has several advantages over the single bearing design as well as a few disadvantages. Among the advantages are higher levels of stiffness and damping as well as a higher stability boundary under the assumption of no bond misalignment. On the other hand there are also some disadvantages. First, it is harder to fabricate a device which has two journal bearings since the journal bearing is one of the most challenging fabrication steps. Second, connecting the experimental results to the modeling efforts will require the measurement of more features including the bond misalignment in vector form, the rotor imbalance level in vector form, and the characterization of a second journal bearing. Metrology for such measurements is limited and as a result this characterization process might prove challenging.

8.3.3 Anisotropy System

Journal bearing anisotropy was introduced to the device to relax the journal bearing clearance tolerance. Several schemes were examined where a seal would block a large fraction of the flow from entering the bearing. These schemes were not pursued however as the level of anisotropy was found to be very sensitive to the seal clearance which was bound to change (potentially closing up completely) as the rotor increased speed and began deforming. The final design employed the resistance of the hydrostatic supply plenum itself to introduce the anisotropy. This design relies on shrinking the depth of the plenum and having flow travel circumferentially underneath the compressor rim. However, in this design, plenum depth variations are bound to introduce static torques to the rotor as was shown in Section 8.2. As a result, consideration of alternate schemes of introducing anisotropy should be made.
One suggestion involves moving to a center injection hydrostatic scheme where the turbine and compressor radii are equal. In this scheme the injection nozzles would be connected to two separate plena so that anisotropy could be introduced in a very controlled way. One further advantage of this scheme is that it would decouple the journal bearing operation from the thrust bearing operation. The disadvantage however is that it involves substantial modification to the fabrication process flow as well as elimination of a region which could be used for thrust balance.

8.3.4 Thrust Balance System

The current turbocharger does not include a thrust balance piston. As a result, as the speed is increased the rotor experiences a net axial force which, as shown in Figure 8-1, cannot be sustained by the thrust bearing pads alone. To overcome this problem the inlet may be pressurized or the exhaust may be exposed to vacuum (the currently employed scheme). Such schemes however, are not viable options for the final engine and as a result, it is imperative that a thrust balance system be designed.

The net axial force in the devices tested was estimated to be on the order of a Newton in the direction of the front thrust bearing. If the turbine air was heated however, this force would be much smaller as the turbine would require a smaller pressure ratio to produce the same amount of work. The rotor thrust balance is therefore not only a function of speed but also a function of temperatures, something which further complicates the problem.

One suggestion is to have hybrid thrust bearings which derive their stiffness from hydrostatics but have increased load carrying capabilities through hydrodynamic effects. Another possibility involves the addition of a second plenum beneath the compressor rim.

8.4 Summary

High speed operation of the turbocharger has been limited by rotordynamics. Some of the limitations are imposed by the design itself whereas others are imposed by the required tight fabrication tolerances. To reduce the fabrication tolerances anisotropy
has been introduced to the design, however the means by which it was introduced appears to have created undesirable side effects linked to rotor tilting. Future work should focus on better understanding of the tilting mechanism and its implications as well as a re-design of the anisotropy system so as to possibly remove the source of tilting.

Furthermore, the thrust bearings are to be re-designed so as to have an axial stiffness which is higher than the design speed, without requiring that the nozzles be choked as it has been shown that choked operation of the nozzles is linked to an axial instability. Preliminary calculations indicate that a small reduction of the thrust bearing clearance should be enough to achieve this.

With these changes it is believed that design speed operation of the turbocharger should be achievable.

8.5 References


CHAPTER 9

CONCLUSION

This chapter summarizes the work presented in this thesis, outlines the main contributions, and suggests possible future work to be done.

9.1 Summary

In an effort to create a MEMS gas turbine generator, this research has focused on the development of a MEMS turbocharger. The objective of the turbocharger has been the demonstration of high speed operation and the characterization of the turbomachinery performance. To achieve this, the design had to be modified as dictated by experimental and modeling results.

A total of three separate generations of the turbocharger have been designed, fabricated and tested. The key differences between each design lie in the rotordynamics, specifically in the journal bearing design. All designs relied on a hydrostatic journal bearing which was placed either on the turbine rim (original design), on the turbine rim as well as the compressor rim (dual bearing), or on just the compressor rim. The dual bearing was designed in an effort to maximize hydrostatic stiffness and alleviate misalignment concerns between the two rotors. The single compressor bearing was designed due to its lower imbalance levels relative to the single turbine bearing design and its fabricational simplicity and ease of characterization as compared to the dual bearing design.

Rotordynamic testing of the devices included the experimental discovery of the journal natural frequency(ies) and thrust bearing natural frequency which compared
favorably with model predictions. Rotor imbalance levels were also dynamically measured and compared to model expectations. In addition, other rotordynamic phenomena, including a thrust bearing instability associated with the choking of the thrust bearing nozzles, were observed.

In order to achieve high speed, a strategy was devised which relied on crossing the journal natural frequency at a very low speed and then following an operating line which lay between the third and fourth multiples of the natural frequency. In order to provide thrust balance, a throttled vacuum was connected to the exhaust of the device and was adjusted by monitoring the flow rate of the thrust bearings. Following such a strategy a speed of 480,000 rpm was ultimately achieved. Higher speed operation was limited by journal and thrust bearing instabilities.

Turbomachinery performance measurements were taken with a total of four devices (two single turbine bearing, and two single compressor bearing). Based on these measurements, turbine maps as well as compressor maps were created. Furthermore the spool efficiency was characterized. At 480,000 rpm, the compressor pressure ratio was 1.21 with a flow rate of 0.13 g/s and the turbine pressure ratio was 1.7 with a flow rate of 0.26 g/s. By employing a parasitic power loss model, it was concluded that if the efficiencies of both the compressor and the turbine were the same during this test, they would have been 48%.

Finally, a demo engine startup model was developed to assess possible ways to ignite the engine. The model employed turbomachinery performance maps, the rotor inertia, the thermal dynamics of the system, structural deformations of the rotor, as well as rotordynamics. It was determined that with the current engine design successful ignition was not feasible. Simulation results indicated that altering the journal bearing geometry combined with either low pre-ignition speed or pre-heating of the device could lead to a feasible ignition.

### 9.2 Contributions

The contributions of this work can be classified into two separate parts:
1. High speed operation of a MEMS turbocharger
   - Identification of governing rotordynamic phenomena and comparison to analytical and numerical predictions.
   - Development of tools to assess and evolve the design subject to the constraints imposed by the fabrication capabilities.
   - Development of operational strategy to achieve high speeds based on experimental results, modeling and analysis
   - Characterization of the turbomachinery performance.

2. Elucidation of the aero-thermo-rotordynamic coupling in the micro gas turbine engine during the ignition transient
   - The transient is dominated by three time constants – a rotor inertial time constant $O(0.2 \text{ seconds})$, a rotor thermal time constant $O(1 \text{ second})$, and a static structure thermal time constant $O(30 \text{ seconds})$
   - Rotordynamic stability is highly sensitive to the ignition transient. Successful ignition requires that the bearing dimensions be modified relative to the turbocharger and that the device is either operated at a low pre-ignition speed or be pre-heated.
   - The relatively large thermal mass of the static structure alleviates the potential for surge in the compressor during the ignition transient as well as reducing the rotor angular acceleration.

9.3 Future Work

Future efforts should focus on achieving higher speed. To do this, a thrust bearing re-design is suggested where the axial natural frequency will lie above the design speed without requiring that the thrust bearing nozzles are choked, as choking of the nozzles has been shown to introduce an axial instability. Furthermore, effort should be placed into understanding the effects of rotor tilt on rotordynamic stability. It is believed that the primary source of tilt is related to the scheme employed to introduce anisotropy.
and as a result it is suggested that alternate anisotropy designs which do not introduce any tilting torques be developed.

Once higher speed has been achieved, the effect of heat addition on the compressor performance should be characterized. The setup for such an experiment is already available and involves pre-heating the turbine air with an electric heater and monitoring the temperature of the flow within the device through a fine thermocouple which has been engineered and built for this specific application.

Following such a test, it is suggested that combustion tests are performed. In these tests the combustor is to be ignited following the strategy suggested by the ignition transient model. The setup for such tests, as well as the necessary packaging are also already in place.
Two separate flow models were developed to analyze the flow through the journal bearing: an incompressible flow model, and an isothermal compressible flow. Both models assumed pressure driven fully developed viscous flow between two parallel plates.

A.1 Incompressible

In computing the pressure loss along the bearing, the incompressible model treats the flow through the bearing as flow through infinite parallel plates. The Darcy-Weisbach equation, which was discussed in Chapter 2, is used to determine the pressure drop for this special case and is written below in terms of the mass flow rate:

\[
\Delta P = \left( \frac{48L\mu}{\rho AD_h^2} \right) m
\]

Eq. A-1

where \( A \) in this case is equal to the circumference of the bearing multiplied by its clearance, \( D_h \) is twice the bearing clearance and \( L \) is the bearing length.

Eq. A-1 indicates that without the inclusion of entrance loss effects, for a fixed geometry the pressure drop through infinite parallel plates varies linearly with the mass flow rate. Furthermore, it can also be seen that for a fixed pressure differential, the flow rate decreases linearly with increasing length and increases with the cube of the hydraulic diameter or the bearing clearance.
In addition to the viscous pressure loss along the bearing length there is also a pressure drop associated with the flow entering the bearing. Based on Idelchik (and confirmed by CFD – Gong) the pressure drop associated with the flow entering the bearing is given as follows:

\[ \Delta P_{\text{ent}} = 1.5 \left( \frac{1}{2} \rho U^2 \right) \quad \text{Eq. A-2} \]

where \( U \) is the velocity of the flow in the bearing.

**A.2 Isothermal Compressible**

The compressible flow model includes compressibility effects but assumes that the flow is isothermal. The following figure depicts an infinitesimal section of a constant area duct that will form the basis of the model:

![Figure A-1: Schematic of isothermal compressible flow between two flat plates](image)

The analysis of the configuration shown in Figure A-1 will be based on the conservation of mass, momentum, and the perfect gas law. Note that the energy equation is not required since the flow is assumed to be isothermal.
Conservation of mass (const. area):

$$\frac{dp}{\rho} + \frac{dU}{U} = 0$$  \hspace{1cm} \text{Eq. A-3}

Conservation of momentum:

\textit{Net Pressure Force – Wall Shear Force} = (Mass Flow Rate)x(Velocity Out – Velocity In)

$$pA - (p + dp)A - \tau_w(P)dx = \rho UA(U + dU - U)$$  \hspace{1cm} \text{Eq. A-4}

Where $P$ is the perimeter of the duct and $\tau_w$ is the shear stress on the wall. The shear stress is then replaced by the Darcy friction factor by employing the following definition:

$$\tau_w = f_{fan} \frac{1}{2} \rho U^2$$  \hspace{1cm} \text{Eq. A-5}

Where $f_{fan}$ is the Fanning friction factor which is related to the Darcy friction factor $f$ according to the following relationship:

$$f_{fan} = \frac{f}{4}$$  \hspace{1cm} \text{Eq. A-6}

By introducing the hydraulic diameter, employing a few compressible flow relations and some simplifying, Eq. A-4 is converted to the following relation:

$$\frac{dp}{p} + \frac{2f_{fan}}{D_H} dx\gamma M^2 + \gamma M^2 \frac{dU}{U} = 0$$  \hspace{1cm} \text{Eq. A-7}
Perfect gas law (isothermal):

For the case of isothermal flow, the perfect gas law can be written as:

\[ \frac{dp}{p} = \frac{d\rho}{\rho} \]  

Eq. A-8

Now, combining Eq. A-8 with Eq. A-3, the following relation is obtained:

\[ \frac{dU}{U} = -\frac{dp}{p} \]  

Eq. A-9

Furthermore, the following relation is also introduced, based on the definition of the Mach number and the assumption of constant temperature:

\[ \frac{dM}{M} = \frac{dU}{U} \]  

Eq. A-10

Combining Eq. A-8 - Eq. A-10 and with some manipulation, Eq. A-7 can be written as follows:

\[ \frac{dM}{\gamma M^3} \frac{dM}{M} = \frac{2f_{\text{fan}}}{D_n} dx \]  

Eq. A-11

Integration of Eq. A-11 between points 1 and 2 with a distance \( L \) between them gives the following:

\[ \left[ \frac{1}{2\gamma M_1^2} - \frac{1}{2\gamma M_2^2} \right] - \ln \left[ \frac{M_2}{M_1} \right] = \frac{2f_{\text{fan}} L}{D_n} \]  

Eq. A-12
Based on Eq. A-12, for a given duct, when either the exit Mach number or the entrance Mach number is known, the other can be computed (using numerical techniques). Furthermore, integration of Eq. A-8, Eq. A-9, and Eq. A-10 gives:

\[
\frac{U_2}{U_1} = \frac{\rho_1}{\rho_2} = \frac{\rho_1}{\rho_2} = \frac{M_2}{M_1}
\]

Eq. A-13

As a result, once the Mach numbers at the entrance and exit of the duct have been computed, using Eq. A-12, the relative changes in all the other variables can then be computed by using Eq. A-13.

For the models that will follow, it turn out that there are two cases which are of special interest. The first case involves knowing the entrance and exit pressures in a duct and computing the mass flow that will result. The second case involves knowing the mass flow through a duct along with the entrance pressure and computing the resulting exit pressure.

For the first case, Eq. A-12 is converted to a second order polynomial in \( M_i \) by replacing \( M_2 \) with \( P_1, P_2 \) and \( M_i \) by employing Eq. A-13:

\[
\left[ \ln \left( \frac{P_1}{P_2} \right) + \frac{2 \frac{f_{fan} L}{D_h}}{M_i} \right] + \left[ \frac{1}{2 \gamma} \left( \frac{P_2}{P_1} \right)^2 - \frac{1}{2 \gamma} \right] - 0
\]  

Eq. A-14

Using the relations developed in Chapter 2 and Eq. A-6, the fanning friction is removed and Eq. A-14 becomes:

\[
\ln \left( \frac{P_1}{P_2} \right) M_i^2 + \frac{48 \mu L \sqrt{RT}}{D_h^2 P_1 \sqrt{\gamma}} M_i + \frac{1}{2 \gamma \left( \frac{P_1}{P_2} \right)^2} - \frac{1}{2 \gamma} = 0
\]

Eq. A-15

So once \( P_1 \) and \( P_2 \) are set, \( M_i \) can easily be computed as the solution of a quadratic equation. \( M_i \) is then converted to a velocity since the temperature is known. Finally, the
velocity can be converted to a mass flow rate since the density can be computed from the equation of state.

If the entrance loss of the duct needs to be included in the calculation, an iterative procedure is followed where Eq. A-15 is solved and the calculated entrance velocity is used to compute the real entrance pressure, after the entrance loss, as follows:

\[ p_{1, \text{NEW}} = p_{1, \text{OLD}} - \zeta_{\text{E.L.}} \frac{1}{2} \rho \left( \frac{U_1}{L} \right)^2 \]  

Eq. A-16

where \( \zeta_{\text{E.L.}} \) is the entrance loss coefficient.

The new entrance pressure which includes the pressure drop due to the viscous entrance loss is input into Eq. A-15 and the iteration proceeds until the entrance pressure converges to its correct value.

For the second case where the mass flow and entrance pressure are known, Eq. A-13 is used to convert Eq. A-15 into the following relation between \( M_1 \) and \( M_2 \):

\[
\frac{1}{2\gamma M_1^2} - \frac{1}{2\gamma M_2^2} - \ln \frac{M_2}{M_1} - \frac{48 \mu L \sqrt{RT}}{p_1 M_1 \sqrt{\gamma D_n^2}} = 0
\]

Eq. A-17

where \( M_1 \) is easily computed from the mass flow and the entrance pressure by employing the equation of state. An iterative procedure which employed Newton’s method was used to solve this equation for \( M_2 \) and thus \( p_2 \). If an entrance loss needs to be included that is done by correcting \( p_1 \) for entrance pressure loss prior to its insertion into Eq. A-17 – this time no iteration is required.

**A.3 Model Comparison**

The incompressible and compressible models were then compared in two ways. First, the flow versus pressure drop relation was examined. Second, the pressure force seen by the duct walls was compared. The reason for comparing the force will become more evident in a later section where the stiffness of the journal bearing is computed.
A.3.1 Flow Comparison

For the flow versus pressure drop comparison, a compressor bearing (radius 4.1 mm) with a clearance of 16 µm and a length of 330 µm was examined. In the entrance viscous loss has been omitted so as to only compare the differences between the two models. The results of an adiabatic flow model are also included in the figure (even though the journal flow is more closely resembled by isothermal flow due to high heat transfer coefficients).

![Flow Model Comparisons](image)

**Figure A-2:** Mass flow rate comparison between incompressible and compressible models

As can be seen, at low pressure the incompressible model gives very similar mass flow results to the compressible models. However, above 40 kPa the compressibility effects start having a noticeable effect and the incompressible models should no longer be used.
A.3.2 Force Comparison

Both models will now be used to compute the force acting on each side of the bearing walls. Once again, the entrance loss will be ignored. The force acting on each unit width of wall (the width being the circumference of the bearing) can be computed by integrating the pressure field along the length of the wall as shown in Eq. A-18:

\[
\frac{F}{b} = \int p(x)dx \quad \text{Eq. A-18}
\]

Starting with the incompressible case, the force per unit width is given by the following relation:

\[
\frac{F}{b} = p_iL - \frac{24\mu}{\rho AD^2} mL \quad \text{Eq. A-19}
\]

In the case of compressible flow, the force per unit width is harder to compute since the pressure along the length of the pipe is a more complicated function of the position along the pipe. As a result, a numerical procedure is followed where once the inlet and exit Mach numbers are found the pipe is divided into a finite number of sections of varying length but of known inlet and exit Mach numbers as shown in Figure A-3. Eq. A-17 is used to compute the length of each of those sections. Using Eq. A-13 the pressure at each section is computed based on the section’s entrance pressure and Mach number and the section’s exit Mach number. Then, the force per unit width of each section is computed by multiplying the length of that section by the average pressure along that section (i.e. the sum of the section’s entrance pressure and exit pressure divided by two). Finally, the forces calculated from each section are summed up to compute the total force per unit width.
Figure A-3: Numerical procedure followed to compute pressure along the duct. The duct is cut into N sections with a linear progression Mach number from the first to the last section. The length of each section is then computed so that the flow relations are satisfied.

The following figure depicts the pressure profile along the bearing based on both the incompressible and the isothermal compressible model at a bearing entrance pressure of 17.2 kPa (2.5 psi) and a bearing entrance pressure of 51.7 kPa (5 psi):

Figure A-4: Pressure profile along a bearing with no entrance loss based on incompressible and compressible models
Since the force per unit width is the area under the curves depicted in Figure A-4 it can be seen that the incompressible model will tend to underpredict the force acting on the walls of the pipe. Figure A-5 depicts the force per unit width prediction for each of the two models as a function of the supply pressure.

![Figure A-5: Force acting on bearing wall with no entrance loss based on incompressible and compressible models](image)

Figure A-5: Force acting on bearing wall with no entrance loss based on incompressible and compressible models

In summary, at bearing supply pressures that are below 40 kPa, the error in the force is less than 6%. However, at higher pressure the errors can become considerable.
The algorithm used to compute the flow characteristics and therefore the stiffnesses of the anisotropic Design D is outlined below. Reference should be made to Figure 3-31 in Chapter 3.

Initially, it is assumed that the pressure after the feed lines is set to the supply pressure. To understand the algorithm we will examine the flow characteristics between supply node A and supply node B. An initial guess is made for the flow going through supply node A in the direction of supply node B. Based on compressible flow relations, the pressure drop in going to the adjacent node is computed. Knowing the pressure at the next node, along with the pressure at the exit of the seal and the compressor bearing (which are boundary condition in this problem), allows for the calculation of the flow that will depart from the tangential flow path. Based on this new flow, the pressure drop to the next node is computed and the procedure continues until the center between supply node A and supply node B. If the initial flow guess was too small there will not be enough flow to reach the center. If the initial flow guess was too large, the pressure will have dropped to below the seal and compressor bearing exit pressures before the center. Using Newton’s method and a numerical formulation which is pertinent only to this problem, the initial flow required to assure no flow at the center node is found and the pressure at that location is recorded. The same procedure is followed for the flow coming from supply node B and the pressure at the center node is compared to the pressure from the flow coming from supply node A. For a perfectly centered rotor the two pressures match. However, if the rotor is not centered, the pressure of the two flow paths at the center node will be different. As a result an iteration is performed where the location of
the center node is moved closer or further from supply node A so that the pressures from the two flow paths match. This procedure is repeated for each of the four sections.

Once all the pressures and flows have been found, the total flow through each of the nodes is computed and the numerical procedure is repeated with revised pressures for each node that account for the pressure drop that occurs in going through the feed lines. This changes the original flow values so the procedure has to be repeated several times until the numbers converge. With all the pressures known, the net force acting on the rotor is easily computed. To obtain the stiffness in the two principal directions the rotor is first offset in the x-direction and then in the y-direction.

Figure B-1 depicts the tangential flow rate through the hydrostatic plenum for a centered and offset position. As can be seen once the rotor is offset in one direction, the flow increases in the direction of the offset and decreases in the other direction. It is also important to notice that the node of zero flow rate shifts from the center point between the supply nodes. Figure B-2 depicts the pressure at each node for a centered and offset position in both of the principal directions.

![Effect of 0.1 Ecc on Tangential Flow Rate](image)

**Figure B-1:** Total flow rate through each node in the circumferential direction. Blue line corresponds to centered rotor and red line corresponds to offset rotor. Angular position defined increasing clockwise starting at Node A.
**Figure B-2:** Pressure at each node. Blue line is the centered case. Red line is offset in x-direction and green line is offset in y-direction. Angular position defined increasing clockwise starting at Node A.
APPENDIX C

PACKAGING

The turbocharger measures 23x23x3 mm and has 15 gas flow connection ports. There are two sizes of connection ports, 2 mm and 0.9 mm diameter circular holes. Figure C-1a depicts the original layout of the ports. The layout later changed and is depicted in Figure C-1b. This layout was present from Build D1 onwards and reasons for the changes are described in Chapter 2. Table C-1 lists the function of each port.

Figure C-1: Turbocharger layout. (a) Prior to build D1. (b) Build D1 and all later builds.
To test the turbocharger, fluidic connections from all of the ports to the external environment are required. For room temperature tests of the turbomachinery, a cold package was developed. For tests with an operational combustor, a combustion package was developed. Finally, for tests which involved externally heating the turbine air a third hot package was developed. All experiments performed during the course of this research used the cold package.
C.1 Cold Flow Packaging

In the cold flow tests the turbocharger is supplied with room temperature gases. The test objective is to better understand the rotordynamics of the device and characterize the performance of the turbomachinery. All fluidic connections to the gas handling system are done through the use of a 4 layer stainless steel package with miniature BN70 O-Rings which are in direct contact with the device. Figure C-2 is a schematic of the package.1

1 This cold flow package is an evolved version of the package that was originally designed by Protz
In designing the inlet pressurization system, an O-ring was not placed on the inlet of the die as experiments had shown that the force of the O-ring substantially deflected the die and affected the thrust bearing performance (as was discussed in Chapter 2). Instead O-rings were placed on the front and rear outer edges of the second plate (from the top) and the original objective of sealing the inlet was successfully achieved without impacting the dies structural integrity. Nonetheless, with the exception of static flow tests the inlet pressurization scheme was never used during high speed operation.

For the exhaust system, two O-Rings, are used. One O-Ring directly seals the exhaust to the package and another O-Ring, lying outside of it, seals the ignitor port (see Figure C-2).

C.2 Hot Flow Packaging

In order to understand the effect of heat addition on the performance of the turbomachinery without getting involved in the complications associated with combustion, an experiment was setup were the turbine supply air would be heated externally through the use of an electric heater.\(^2\) The turbine supply temperature would be measured by exploiting the ignitor port and placing a fine thermocouple inside the combustor. Since the temperature of the device would be elevated during these experiments, the BN O-Rings were replaced by Silicon O-Rings. Figure C-3 depicts the hot flow package.

\(^2\) The electric heating tube was originally created by Spadaccini and was modified for this application.
Figure C-3: Turbocharger hot package (external electrically heated air)

The package involved a modification of the original cold package to allow for the thermocouple to be placed within the ignitor port. The thermocouple consisted of fine, 0.0254 mm (0.001 inch) diameter thermocouple wire which was run through ceramic insulating tubing which was epoxied inside a stainless steel hollow plug. Figure C-4 is a schematic of the thermocouple design as well as a picture of an assembled thermocouple.
Due to time constraints, spinning experiments with this package were never pursued by the author but tests with a non-rotating device indicated that temperatures of over 450 K could be achieved within the combustor.

**C.3 Combustion Flow Packaging**

When the combustor is ignited, the die wall temperatures may approach 1000K (according to the combustor rig experiments). O-Rings cannot withstand such temperatures and as a result the die must be packaged by some other means. A procedure introduced by London for the MIT microrocket program was used [x,y] It involves connecting the die to a Kovar plate via Kovar tubes that are brazed onto the plate and sealed onto the die by glass frits. An ignitor is also mounted onto the die using this technique. The ignitor is simply a thin wire that protrudes into the combustor and is resistively heated to initiate combustion. An exhaust tube is also mounted to keep the hot exhaust gases from attaching to the die’s rear surface.³

---

³ This effect was originally observed by Mehra
Figure C-5 depicts the setup schematically and Figure C-6 is an example of a dummy test. In Figure C-6 an additional block of material is seen between the die and the plate. The block is a carbon fixture that is used to help locate the tubes into the die and plate during the packaging procedure. This fixture is not removed after the die has been packaged, but nonetheless does not interfere with the operation of the device.

**Figure C-5:** Glass bead package schematic
In the original packaging scheme developed by London, and also currently used by the combustor group, the connections between the Kovar tubes, the Kovar plate and the die are done in one single pass through a furnace. After this, the Kovar tubes that are not flush with the Kovar plate are sanded down to make a smooth surface for O-Rings to seal on. During this step, small particles could fall into the Kovar tubes and once the ports are pressurized these particles could flow into the engine. Unlike the combustor, the turbocharger has some very fine clearances (order of a few microns) that could be seriously impacted by the ingestion of any such particle.

As a result, the packaging scheme was slightly modified. Instead of connecting the Kovar tubes to the Kovar plate and the die all at once, the process is broken into two steps. In the first step, the Kovar tubes are only mounted onto the Kovar plate but not to the die. To do this, a carbon replica of the die is used to hold the tubes in place. The Kovar plate is then sanded down and cleaned without the fear of any particles falling into the device. After this step, the die is mounted to the entire assembly of the tubes and the plate and passed through the furnace. Once it exits the furnace no further processing needs to be done.

The Kovar plate is then mounted, and sealed by O-Rings, onto the uppermost plate of the cold flow package shown in Figure C-2. However, it is still necessary to be able to lower the pressure of the exhaust for thrust balance reasons.
this, the entire packaged die is placed within a large sealed container whose pressure can be lowered to sub-atmospheric levels. In order to provide visual access to the die the tank is fitted with three viewports. Figure C-7 is a schematic of the setup.
A hermetic seal feedthrough is used to pass thermocouple wires and a power supply for the ignitor into the chamber. The thermocouples measure the exhaust temperature, the die wall temperature, and the chamber ambient temperature.

Controlling the pressure in the chamber as the turbine exhaust flow rate changes can be rather challenging due to the large enclosed volume since the time constant associated with pressure changes due to flow changes is on the order of 10's of second’s. As a result, instead of a manually controlled metering valve, a vacuum pressure regulation control system was installed. Figure C-8 depicts how this instrument operates. The operator can at any point reset the desired chamber pressure. However, it is not necessary to constantly compensate for increasing chamber pressures whenever the turbine exhaust flow rate increases. The only time that action is required is whenever the thrust balance needs to be adjusted.
Figure C-8: Vacuum control system for combustion package
APPENDIX D

UNCERTAINTY ANALYSIS

This appendix presents an uncertainty analysis for the measurements taken during the experiments. First, the uncertainty in the independent measurements is presented and then the uncertainty in derived quantities is discussed.

D.1 Uncertainty of the Independent Measurements

D.1.1 Pressure

All the pressures were measured using Honeywell 240PC pressure transducers. For the thrust bearings, pressure transducers rated to 1,724 kPa (250 psi) were used. For the journal bearing hydrostatic pressure tap, a 103 kPa (15 psi) pressure transducer was used and for all other pressure measurements, transducers rated to 689 kPa (100 psi) were used. For all of these transducers the manufacturer reports a linearity error of +/- 0.50 % of span and a repeatability / hysteresis error of +/- 0.25 % of span. These two errors combined in a root mean square sense give an overall error which is 0.56 % of span. The following table lists the error for each of the three pressure transducers:
Table D-1: Pressure Transducer Uncertainty

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>Span (psi)</th>
<th>Absolute Error (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>242PC15M</td>
<td>103,422 Pa (15psi)</td>
<td>+/- 579 Pa (+/- 0.084 psi)</td>
</tr>
<tr>
<td>242PC100G</td>
<td>689,480 Pa (100 psi)</td>
<td>+/- 3,861 Pa (+/- 0.56 psi)</td>
</tr>
<tr>
<td>242PC250G</td>
<td>1,723,700 Pa (250 psi)</td>
<td>+/- 9,653 Pa (+/- 1.40 psi)</td>
</tr>
</tbody>
</table>

D.1.2 Mass Flow Rate

All the mass flow rates were measured using MKS Instruments 1179 flow meters/flow controllers. For the thrust bearings, the flow meters were rated to 500 sccm and for all other flow rate measurements, the meters were rated to 20,000 sccm. For both of these instruments the manufacturer reports a total error of +/- 1 % of span. The following table lists the error for both of the flow meters used:

Table D-2: Mass Flow Meter Uncertainty

<table>
<thead>
<tr>
<th>Mass Flow Meter / Controller</th>
<th>Span (sccm)</th>
<th>Absolute Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1179A</td>
<td>10.42 mg/s (500 sccm)</td>
<td>0.1042 mg/s (+/- 5 sccm)</td>
</tr>
<tr>
<td>1179A</td>
<td>0.4166 g/s (20,000 sccm)</td>
<td>4.166 mg/s (+/- 200 sccm)</td>
</tr>
</tbody>
</table>

D.1.3 Speed

The speed is determined by the spectrum analyzer that processes the data acquired from the fiber optic. The error estimated by Jamonet was 0.035% and that corresponds to +/- 200 rpm at 500,000 rpm.

---

1 Values for Nitrogen (for air multiply by 1.035)
D.2 Uncertainty of Derived Quantities

Uncertainty in derived quantities is determined from the uncertainty in the independent measured quantities. If \( y \) is the quantity in question, and is a function, \( f \), of the independent quantities \( x_1, x_2, \ldots x_n \), a set of influence coefficients \( C_{xi} \) is defined and computed as follows:

\[
C_{xi} = \left| \frac{\partial f}{\partial x_i} \right| \left| \frac{x_i}{y_o} \right|
\]

Eq. D-1

The overall fractional uncertainty in \( y \) is then determined as follows:

\[
S_y = \sqrt{\sum_{i=1}^{n} (C_{xi}S_{xi})^2}
\]

Eq. D-2

Where \( S_{xi} \) is the fractional uncertainty in each of the measured quantities defined as the uncertainty in the value \( x_i \) divided by the indicated value of \( x_i \).

D.2.1 Bearing Clearance

The uncertainty in the bearing clearance is determined using influence coefficients derived from the flow model. The two error sources considered are the mass flow rate and the supply pressure. The following table lists the calculation for a typical case.

293
Table D-3: Journal Bearing Clearance Uncertainty

<table>
<thead>
<tr>
<th>$x_i$</th>
<th>Value</th>
<th>$C_{xi}$</th>
<th>$S_{xi}$</th>
<th>$S_c$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m$</td>
<td>48.391 mg/s</td>
<td>0.475</td>
<td>8.61 %</td>
<td>-</td>
</tr>
<tr>
<td>$P_{\text{gauge}}$</td>
<td>34,474 Pa</td>
<td>0.400</td>
<td>1.68 %</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>-</td>
<td>4.14 %</td>
<td>(+/- 0.66 μm)</td>
</tr>
</tbody>
</table>

Based on compressor bearing with reference clearance 16 μm, length 330 μm and supply pressure of 34,474 Pa (5psi)

D.2.2 Thrust Bearing Nozzle Diameters

The thrust bearing diameter is determined by matching the model results with the experimental results. The following table summarizes the uncertainty in the thrust bearing nozzle diameter.

Table D-4: Thrust Bearing Nozzle Diameter Uncertainty

<table>
<thead>
<tr>
<th>$x_i$</th>
<th>Value</th>
<th>$C_{xi}$</th>
<th>$S_{xi}$</th>
<th>$S_{\text{th,d}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m$</td>
<td>3.18 mg/s</td>
<td>0.55</td>
<td>3.28 %</td>
<td>-</td>
</tr>
<tr>
<td>$P_{\text{gauge}}$</td>
<td>827 kPa</td>
<td>0.55</td>
<td>1.17 %</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>-</td>
<td>1.92 %</td>
<td>(+/- 0.21 μm)</td>
</tr>
</tbody>
</table>

Based on thrust bearing with reference clearance 6 μm, length 100 μm, diameter 11 μm, and supply pressure of 823 kPa (120 psi)

D.2.3 Compressor Rotor Inlet Flow Rate

The compressor rotor inlet flow rate is computed by employing the turbine supply and exhaust flow rate, the compressor exhaust flow rate and the journal flow rate. The influence coefficient for each of these terms is one and thus the uncertainty in the compressor flow rate is found to be +/- 8.332 mg/s irrespective of the flow rate. At the
maximum compressor flow rate of 0.138 g/s this uncertainty corresponds to 6.0 % of the measured value.

**D.2.4 Compressor Pressure Ratio**

The uncertainty in the compressor pressure ratio originates only from the pressure transducer downstream of the compressor. This results in a pressure ratio error of +/- 0.038 for all pressures.

**D.2.5 Turbine Pressure Ratio (Reciprocal)**

The uncertainty in the turbine pressure ratio originates from the turbine supply pressure transducer and the exhaust pressure transducer. The uncertainty depends on the pressure settings. The following table is an example of the uncertainty of the reciprocal of the turbine pressure ratio at the maximum speed.

<table>
<thead>
<tr>
<th>xi</th>
<th>Value</th>
<th>C_{xi}</th>
<th>S_{xi}</th>
<th>S_{1/\text{rt}}</th>
</tr>
</thead>
<tbody>
<tr>
<td>pt4</td>
<td>138,800</td>
<td>1</td>
<td>2.78 %</td>
<td>-</td>
</tr>
<tr>
<td>pt5</td>
<td>81,700</td>
<td>1</td>
<td>4.73 %</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td></td>
<td></td>
<td>5.5 % ( +/- 0.094)</td>
</tr>
</tbody>
</table>

Based on 480,000 rpm, turbine pressure ratio of 1.7

The uncertainty in the turbine corrected flow rate at 480,000 rpm (~0.19 g/s) is computed similarly and is found to be 3.2 % ( +/- 6.1 mg/s).

**D.2.6 Compressor and Turbine Fluidic Power**

The fluidic power produced by the compressor is computed as follows:

\[ P_c = mC_pT_{r2}\left(\frac{\gamma}{\gamma - 1}\right) \]

Eq. D-3
Table D-6: Compressor Fluidic Power Uncertainty

<table>
<thead>
<tr>
<th>xi</th>
<th>Value</th>
<th>C_{xi}</th>
<th>S_{xi}</th>
<th>S_{pc}</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>0.13 g/s</td>
<td>1</td>
<td>6.41 %</td>
<td>-</td>
</tr>
<tr>
<td>\pi</td>
<td>1.21</td>
<td>5.37</td>
<td>3.14 %</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>-</td>
<td>18.0 %</td>
<td>(+/- 0.40 W)</td>
</tr>
</tbody>
</table>

Based on 480,000 rpm, compressor pressure ratio 1.21 with a flow rate of 0.13 g/s. Nominal fluidic power was 2.19 W.

The fluidic power supplied to the turbine is computed as follows:

\[ P_t = m C_p T_{t4} \left( 1 - \pi^{\gamma / T} \right) \]  
Eq. D-4

Table D-7: Turbine Fluidic Power Uncertainty

<table>
<thead>
<tr>
<th>xi</th>
<th>Value</th>
<th>C_{xi}</th>
<th>S_{xi}</th>
<th>S_{pt}</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>0.263 g/s</td>
<td>1</td>
<td>1.58 %</td>
<td>-</td>
</tr>
<tr>
<td>\pi</td>
<td>1.7</td>
<td>1.73</td>
<td>5.5 %</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>-</td>
<td>9.6 %</td>
<td>(+/- 1.1 W)</td>
</tr>
</tbody>
</table>

Based on 480,000 rpm, turbine pressure ratio 0.588 with a flow rate of 0.263 g/s. Nominal fluidic power was 11.1 W.
The power consumed by each thrust bearing is modeled as Couette flow between two disks separated by half of the total thrust bearing clearance as shown in Figure E-1.

\[ h = \text{gap}/2 \]

**Figure E-1:** Thrust bearing power loss modeled as Couette flow.

By definition, the local shear stress, \( \tau \), acting at any point on each disk surface is given by the following relation where \( u \) is the local velocity, \( \mu \) is the viscosity and \( y \) is an axis as shown in Figure E-1:

\[ \tau = \mu \frac{\partial u}{\partial y} \]

Eq.E-1
In the special case of Couette flow the velocity profile between the two plates is linear and as a result, the partial derivative in Eq.E-1 can be replaced by the local velocity of the rotating plate divided by the gap separating it from the stationary plate:

\[ \tau_{\text{local}} = \mu \frac{\omega r}{h} \quad \text{Eq. E-2} \]

Where \( h \) is the gap separating the stationary plate from the rotating plate, \( \omega \) is the angular velocity and \( r \) is the local radius. The local shear force is then converted to a torque by multiplying by the local radius. The power is then computed by integrating the local torque multiplied by the angular velocity over the entire area of the disk:

\[ P_{T,B} = \int_0^R (\omega) r \left( \mu \frac{\omega r}{h} \right) 2\pi r \, dr = \frac{2\pi \mu \omega}{4h} R_{TB}^4 \omega^2 \quad \text{Eq. E-3} \]

where \( R_{TB} \) is the radius of the thrust bearing pad.

Similarly, in the case of a single bearing device, the power consumed by the seal can be computed as follows:

\[ P_{\text{Seal}} = \frac{2\pi \mu \omega^2}{4h} \left( R_{\text{out}}^4 - R_{\text{in}}^4 \right) \quad \text{Eq. E-4} \]

where \( R_{\text{out}} \) is the outer radius of the seal and \( R_{\text{in}} \) is the inner radius of the seal.

The power consumed by the hydrostatic plenum is given by:

\[ P_{\text{Plenum}} = \frac{2\pi \mu \omega^2}{4h} \left[ \left( R_{\text{comp}}^4 - R_{\text{out}}^4 \right) + \left( R_{\text{in}}^4 - R_{\text{turb}}^4 \right) \right] \quad \text{Eq. F-5} \]

where \( R_{\text{turb}} \) is the radius of the turbine and \( R_{\text{comp}} \) is the radius of the compressor.
Finally, the power consumed by the journal bearings does not require any integration and is given by the following equation, where $R_{\text{bearing}}$ is the radius of the bearing being examined:

$$P_{\text{bearing}} = \frac{2\pi\mu L \omega^2}{h} R_{\text{bearing}}^3 \quad \text{Eq. E-6}$$

Table E-1 summarizes the total parasitic power losses for each of the two designs tested at standard atmospheric temperature, at elevated temperature (assuming the geometry is unchanged), and also the generalized expression for all speeds and viscosities.

<table>
<thead>
<tr>
<th></th>
<th>Turbine Bearing Dies</th>
<th>Compressor Bearing Dies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parasitic Power at 288K</td>
<td>4.99W</td>
<td>4.91W</td>
</tr>
<tr>
<td>Parasitic Power at 1000K</td>
<td>11.6W</td>
<td>13.7W</td>
</tr>
<tr>
<td>Generalized Parasitic Power</td>
<td>1.786E-5 * ($\mu\omega^2$)</td>
<td>2.115E-5 * ($\mu\omega^2$)</td>
</tr>
</tbody>
</table>

(based on matlab model D:/parasitic.m)

As can be seen the parasitic power losses for both devices are on the order of 5 Watts when operating cold and almost triple at a temperature of 1000 K. This is due to the increase in air viscosity with temperature. It is clearly a significant loss source compared to the turbine power which is on the order of 100 watts.

Figure E-2 provides a breakdown of the overall parasitic loss. For both designs, the thrust bearings appear to be making the smallest contribution. In the case of the compressor bearing design, the effect of the second non-operational bearing is seen to be substantially reduced as its clearance was larger (50um instead of 27um) than that of the turbine’s non-operational bearing. It should also be noted that the hydrostatic plenum itself is a large fraction of the power loss and that this could turn out to be a problem if higher levels of anisotropy are introduced in future designs by reducing that plenum’s height.
Figure E-2: Parasitic power loss breakdown for (a) the turbine bearing device and (b) the compressor bearing device. This distribution is the same at all speeds as each loss term scales with the square of the angular speed.
This Appendix outlines an interpolation scheme used to determine operating points on the compressor map other than the ones predicted by the CFD.

The 3-D steady CFD predictions for four separate speed lines are shown in the following figure:

![CFD 10 Blade Constant Span 220μm - Sirakov](image)

**Figure F-1:** CFD compressor map for 220 μm blade span (Sirakov)
As a first step, for each speed line, MATLAB was used to fit a piecewise cubic hermite interpolating polynomial ("pchip") to the CFD points in order to interpolate the relation between the pressure ratio and corrected mass flow rate for each given speed. Next, beta lines were constructed and overlaid on the compressor maps to allow interpolation among speeds. Beta lines are second order polynomials with no linear term whose coefficients are chosen so that they span all the relevant locations of the compressor map:

$$\pi_\beta = \alpha m_{corr}^2 + 1$$  \hspace{2cm} Eq. F-1

The intersection of each of the beta lines with each of the constant speed lines was then determined analytically. It was then assumed that constant speed lines on the map follow a geometric progression between each of the calculated speed lines. The geometric progression was assumed to be in the length of the arc of the beta lines between each speed line. So it was necessary to first compute the length of each of the arcs.
The length of a curve for any polynomial $y$ in $x$ is given by:

$$l = \int_a^b \sqrt{1 + \left(\frac{dy}{dx}\right)^2} \, dx$$  \hspace{1cm} \text{Eq. F-2}$$

Employing Eq. F-2, the length of the beta line polynomial between two points is given by the following relation:

$$2a \left[ \frac{m}{2} \sqrt{\left(\frac{1}{2\alpha}\right)^2 + m^2} + \frac{1}{2(2\alpha)^2} \ln \left( \frac{m + \sqrt{m^2 + \left(\frac{1}{2\alpha}\right)^2}}{m} \right) \right]^{m_2}$$  \hspace{1cm} \text{Eq. F-3}$$

Two separate geometric progressions are then determined. One is for the regions between 0 to the 48% speed line and the other region is between the 48% and 100% speed lines. For each of these two progressions, an initial step size, $x_0$, is determined as well as the stepping ratio, $k$. The summation of a geometric series is given by the following expression:

$$S_n = x_0 \frac{1 - k^n}{1 - k}$$  \hspace{1cm} \text{Eq. F-4}$$

This relation is applied to each of the two separate regions and two equations are developed to solve for the two unknowns. Examining the 48% to 100% speed line regions, the following two equations are developed:

$$\frac{1 - k^{32}}{1 - k^{24}} = \frac{l_{48\%-100\%}}{l_{48\%-72\%}}$$  \hspace{1cm} \text{Eq. F-5}$$
These two equations are solved in $x_o$ and $k$ for each of the beta lines. With this information, it is now possible to determine how far along each beta line a given speed will intersect it. With a numerical procedure this length is converted to points on the beta lines. These points are fitted with a piecewise cubic hermite interpolating polynomial as were the original CFD points. The same procedure is followed for the region between 0 to 48% speed if the speed of interest falls in that region.

Now it is necessary to determine the efficiency. This is done by first fitting curves to the efficiency data that was supplied by the CFD analysis for each of the given speed lines. Then, the beta lines from the compressor map are transported to the efficiency plot since the flow corresponding to each of the intersections they had with the speed lines was determined in the previous analysis. The beta line intersection points are themselves then curve fitted. Once the speed is known, the intersections of efficiency constant speed line with each of the beta lines can be determined and plotted. This is the efficiency curve for the speed line being examined.

An example of the results for a speed of 955,000 rpm is depicted in Chapter 7.
This Appendix outlines the component matching procedure followed to determine the cold flow equilibrium running point for the transient startup model. Figure G-1 defines the engine station numbers.

**Figure G-1:** Engine station definitions

There are two compatibility conditions that have to be met at any operating point. The first is that the angular speed of the compressor equals that of the turbine:

\[
\omega_t = \omega_c
\]  
Eq. G-1

This statement is then written in terms of the non-dimensional speeds as that is the form in which speed is presented on the component maps:
The second condition is continuity, that the flow through the turbine is equal to the flow through the compressor with the addition of the flow from the journal bearing:

\[ m_i = m_c + m_j \]  

Eq. G-3

This statement is then written in terms of the non-dimensional mass flow rates:

\[
\frac{m_i \sqrt{T_{i4}/T_{\text{ref}}}}{P_{i4}/P_{\text{ref}}} = \frac{m_c \sqrt{T_{i21}/T_{\text{ref}}}}{P_{i2}/P_{\text{ref}}} \left( \frac{T_{i4}}{P_{i4}} \right) + m_j \left( \frac{\sqrt{T_{i4}}}{P_{i4}} \right) \]  

Eq. G-4

In addition to Eq. G-2 and Eq. G-4, a power balance requirement has to be satisfied. Specifically, the power being produced by the turbine has to equal the power being consumed by the compressor plus the power being consumed by all other parasitic losses. Writing this in equation form we have:

\[ P_t = P_c + P_{\text{losses}} \]  

Eq. G-5

Eq. G-5 can then be expressed in terms of the pressure ratios and efficiencies of the compressor and turbine as follows:

\[
m_i C_p T_{i4} \frac{1}{\eta_t} \left( 1 - \frac{P_{i5}}{P_{i4}} \left( \frac{r_e}{r_i} \right) \right) = m_c C_p T_{i21} \left( \frac{P_{i3}}{P_{i2}} \left( \frac{r_e}{r_i} \right)^{-1} - 1 \right) + k\mu \omega^2 \]  

Eq. G-6

As there is no external nozzle, \( P_{i5} \) is set to be atmospheric and the turbine map used to compute the ratio of \( P_{i5} \) to \( P_{i4} \) includes the effect of the inherent right angle turn nozzle.
$P_{i2}$ is set to a known fixed value, as it is externally controlled, and $T_{i2}$ is set to be at standard atmospheric temperature (only in the cold flow case). Furthermore, based on experimental results it is assumed that the pressure drop between the compressor and turbine is 3% [Spaddaccini]:

$$\frac{P_{i4}}{P_{i3}} = 0.97 \quad \text{Eq. G-7}$$

Finally, as the combustor is not lit and there is no heat addition, the total temperature of the flow entering the combustor is assumed to be the same as the total temperature of the flow leaving the compressor:

$$T_{i4} = T_{i3} \quad \text{Eq. G-8}$$

With all of the above equations and the use of the compressor and turbine maps, it is now possible to determine what the cold flow operating point will be when given the inlet pressure and journal bearing flow rate. The solution procedure is iterative and is outlined in the following paragraphs:

First a non-dimensional compressor speed is assumed and for this speed an iteration, over the non-dimensional mass flow rates, is performed until the flow compatibility equation is satisfied. More specifically, an initial guess is made for the compressor non-dimensional flow rate and is used in combination with the assumed non-dimensional compressor speed to compute the compressor pressure ratio and efficiency from the compressor map. The pressure loss in the combustor is then computed and the turbine pressure ratio is determined since the exhaust pressure is known to be atmospheric. By employing the turbine map, the turbine non-dimensional flow rate is found. The flow compatibility equation is then solved for $T_{i4}$. The resulting value of $T_{i4}$ is then compared to $T_{i3}$ and the assumed compressor non-dimensional mass flow rate is iteratively adjusted until the two are equal.

Once the mass flow compatibility equation has been satisfied, the power balance equation is used to iteratively adjust the assumed non-dimensional compressor speed.
until the net power of the system is zero. It should be noted that for each speed the mass flow compatibility has to be re-evaluated. Once the iterations have converged, the cold flow operating point has been determined.