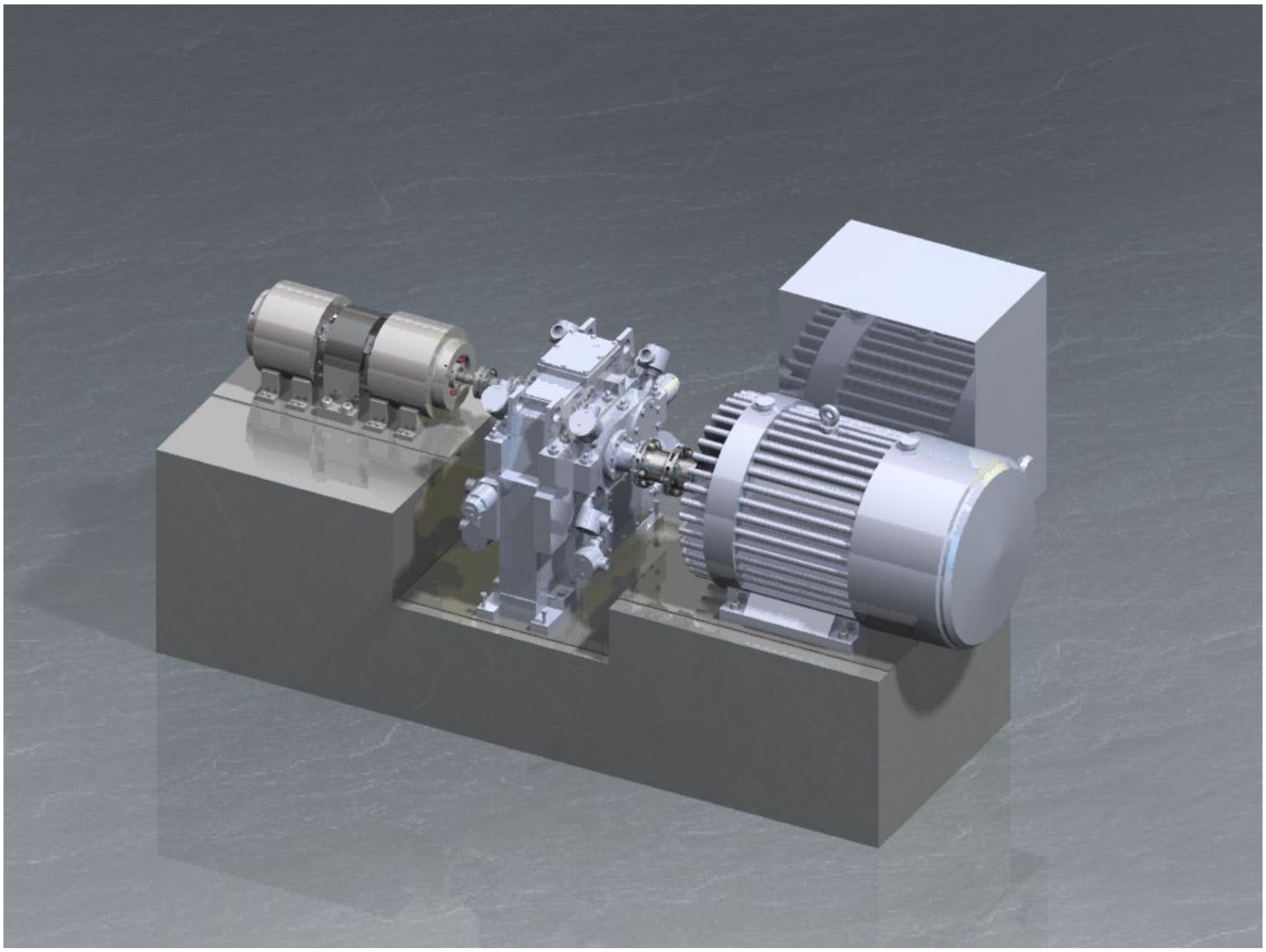
ROMAC NEWSLETTER

2007/2008

ROTATING MACHINERY & CONTROLS LABORATORY



Inside this issue:

Special points of interest:

- More improvements to MAXBRG
- New Test Rigs
- New Rotor Codes
- New Bearing Codes
- 2008 Annual Meeting in Charlottesville, VA

Paul Allaire, Wade Professor of Mechanical and Aerospace Engineering and Director of ROMAC

To ROMAC Industrial Members:

Our research efforts continue to expand to better serve our ROMAC member companies. ROMAC continues to grow rapidly. Over the past few years we have had many new members from marine applications of rotating equipment such as Bechtel, General Dynamics, Curtiss Wright, Knolls Atomic Power Lab, etc. Now we are also seeing a major expansion in the aerospace field with Boeing, GE Aviation, Pratt & Whitney, and Rolls Royce. This area couples with our NASTRAN efforts, squeeze film dampers, rolling element bearings, instability, and gears. We also continue to grow in the international area with new companies over the past couple of years such as Zollern and WEG (both in Brazil), Renk, ODS and Statoil (all three in Europe)

The many research projects outlined in this Newsletter cover most of the topics in dynamics of rotating machinery, from rotor modeling and computer codes to fluid film bearings, seals, gears, rolling element bearings, fluid flows, magnetic bearings, and optimization. A matter of significant interest is the new Graphical User Interface under development – the first version will be made available to companies in June, 2008. This goes along with a major improvement in our rotor dynamics codes – Comborotor and Matlabrotor that will be released in June, 2008 as well. Our association with NASTRAN through MSC Software continues with the integration of ROMAC codes. Our test rigs are coming along extremely well with the compressor surge test rig, the fluid film bearing test, and flexible rotor magnetic bearing/controls test rig leading the way. Several other test rigs on squeeze film dampers, seals, and magnetic bearings are underway.

Paul Allaire, ROMAC Director and Mac Wade Professor

2008 ROMAC Members

1. Aerojet
2. Bechtel Plant Machinery
3. Bechtel Bettis
4. Boeing
5. Cameron Compression
6. Curtiss Wright
7. Siemens DEMAG Delaval
8. Dow Chemical
9. Dresser Rand
10. Electric Boat
11. Elliott
12. ExxonMobil
13. Flowserve
14. GE Aviation
15. GE Oil & Gas
16. Innovative Power Solutions
17. Kingsbury
18. Knolls Atomic Power
19. Kobe Steel
20. Lufkin Industries
21. MSC Software
22. Mitsubishi
23. ODS
24. Petrobras
25. Pratt & Whitney
26. Renk
27. RMT
28. Rolls Royce Energy Systems
29. Rolls Royce Aviation
30. Shell
31. Solar Turbines
32. STATOIL
33. Hamilton Sundstrand
34. TurboCare
35. TCE
36. Waukesha Bearing
37. WEG
38. Zollern

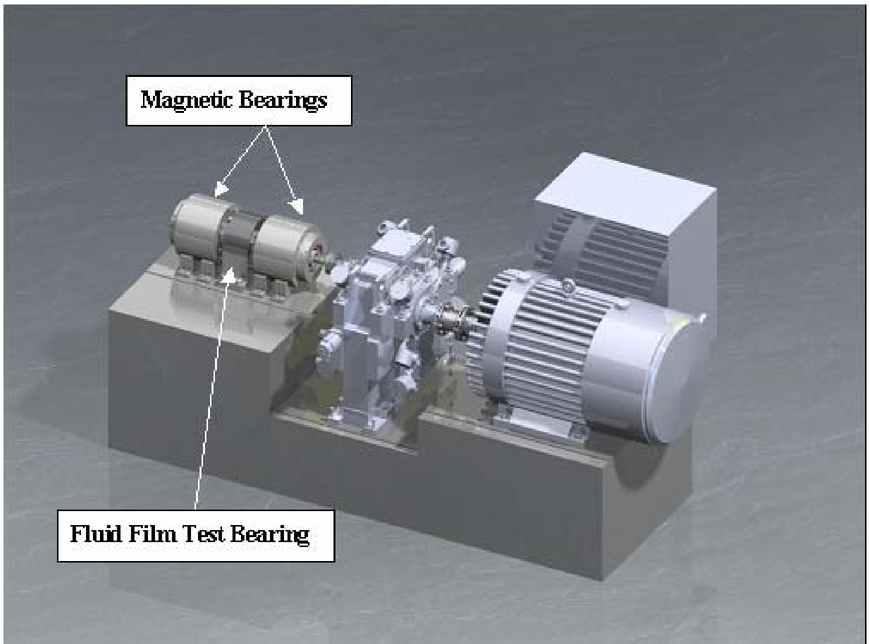
1.1 Fluid Film Bearing/Squeeze Film Damper Test Rig

Student: Tim Dimond, Tanner Hall, Mohit Chhabra

Faculty: Prof. Paul Allaire, Prof. Pradip Sheth, Prof. Zongli Lin

Research Professionals: Wei Jiang, Robert Rockwell

Funding: ROMAC, Fluid Film Bearing Test Rig Group



The ROMAC FFBTR

Figure 1

"We now have six sponsors, current or expected, of the FFBTR Group"

The fluid film bearing test rig (FFBTR) continues to be the highest rated project in ROMAC based upon last year's research objectives. As we all know, fluid film bearings are now operating at surface speeds that clearly place them in the turbulent flow range for the lubricant film, both for oil and water lubricant films. This means that both turbulent effects and inertia effects are present in the lubricant films. Extremely little measured data on dynamic properties of bearings in this range have been obtained.

The FFBTR is moving along rapidly. We now have six sponsors, current or expected, of the FFBTR Group: 1) Rotating Machinery Technology Inc., 2) Turbo Components and Engineering, 3) Kingsbury, 4) Zollern, 5) Bechtel/General Dynamics Electric Boat, and 6) Statoil. All of them are interested in testing bearings. The FFBTR Group is also extensively supported by several companies, providing materials, components, technical assistance, etc. The rig has been designed by Tim Dimond, several other graduate students, research professionals, and faculty. The 350 HP motor and drive have been received from Emerson/US Electric Motors, the gear box is on the way from Lufkin, the magnetic bearings are on the way from Innovative Power Solutions and RMT is constructing many of the test rig components. Other ROMAC companies are kind enough to supply technical expertise as needed. The FFBTR is expected to be operational in fall of 2008.

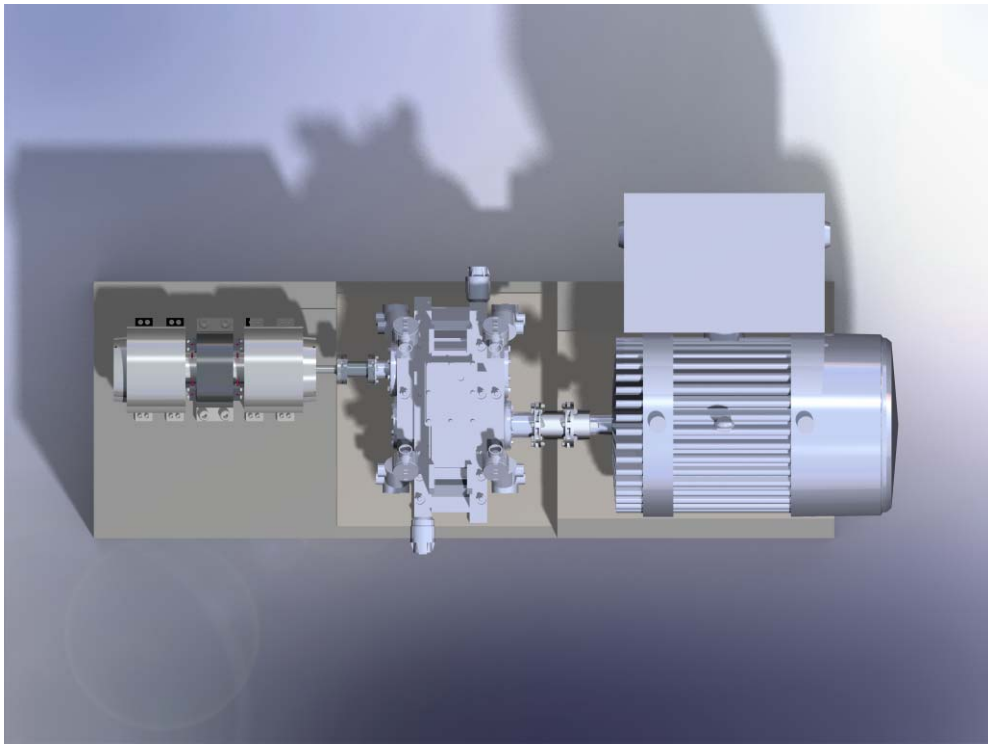


Figure 2: The ROMAC FFBTR—Top View

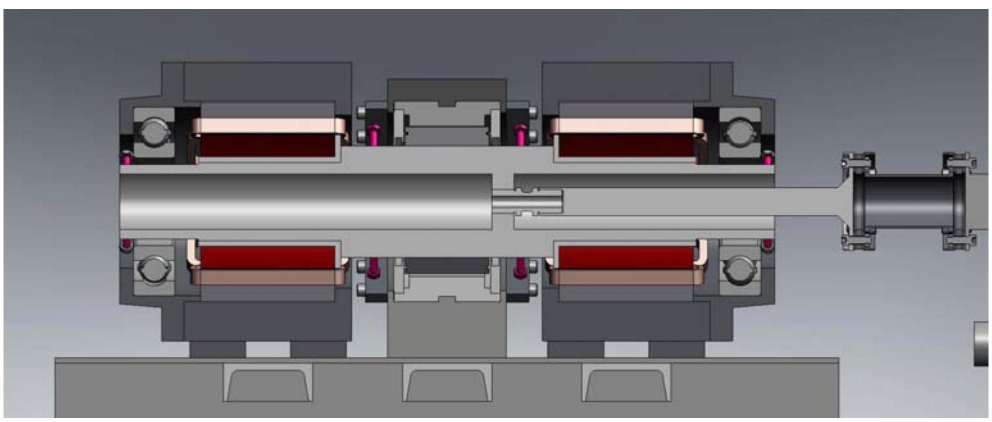


Figure 3: FFBTR Test Section Cross-Section

Major Equipment Estimated Delivery Dates:

Motor Drive: Delivered

Motor: Delivered

Gearbox: Estimated June 2008

"The FFBTR is expected to be operational in fall of 2008"

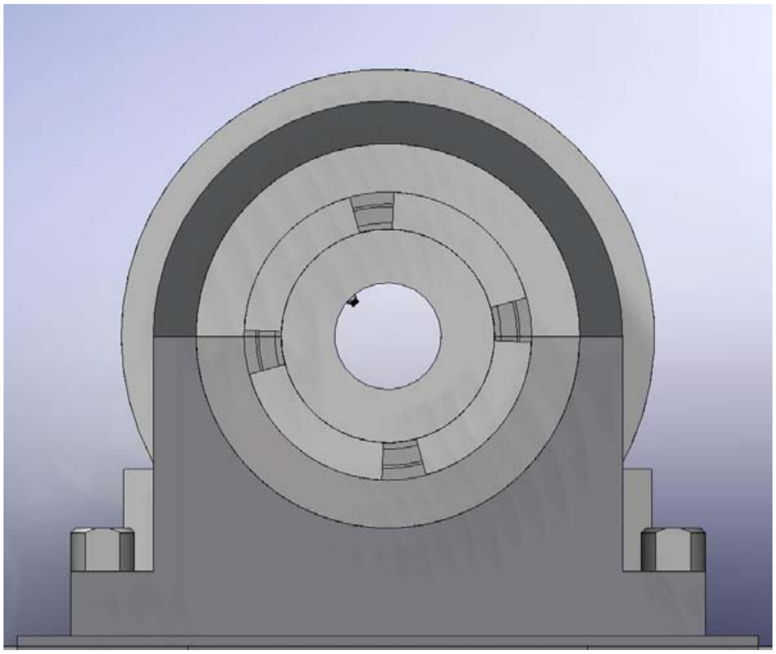


Figure 4: Fluid Film Bearing Cross Section, LBP configuration

Fluid Film Bearing Specifications

Test Section

The purpose of the ROMAC fluid film bearing test rig is to measure the load capacity, thermal effects, stiffness and damping of oil-lubricated and water lubricated bearings under high speed application conditions. A summary of the fluid film bearing technical specifications for the FFBTR follows in Table 1:

Table 1. FFBTR Characteristics

Test Rig Characteristic	*Values, SI*	*Values, BG*
Test Bearing Diameter	127 mm	5 in
L/D Ratios	0.5-0.75	
Pad Pivot Offsets	0.5-0.6	
Orientations	LBP, LOP	
Rotational Speed Range	9000-22000 rpm	
Surface Speed Range	60-149 m/s	196-480 ft/s
Oil lubricated, typical	136-149 m/s	445-480 ft/s
Water lubricated, typical	60-73 m/s	196-240 ft/s
Lubricants	ISO VG 32 ISO VG 46 Water	
Maximum Bearing Unit Load	3.3 MPa	480 psi
Dynamic perturbation displacement, 0-p	12 μm	0.5 mil
Excitation Frequency Range	60 Hz – 560 Hz	

Table 1

Notes:

1. Radial linearity of the lubricating film is assumed. The rotor motion of 0.5 mil 0- p results in shaft orbits that are approximately 6 percent of the total bearing clearance.

2. The actuator sizing and bearing static load limits are based on the following:

- Nominal bearing clearance of 1.5 mil/in
- Four pad bearing, LBP
- Preload: *m* = 0.3
- L/D =0.75
- Offset: = 0.6
- ISO VG 46 lubricant
- Rotor 0-p displacement at FFB of 0.5 mils
- Maximum excitation frequency of 1.5x
- Rotational speed of 22,440 rpm
- Bearing coefficients calculated with MAXBRG
- Rigid Rotor
- Full Bearing coefficients

Bearing configurations with lower stiffness require lower dynamic force for rotor displacement and can be tested at higher static load.

<u>Motor</u>

The motor was sized to overcome the parasitic losses predicted by MAXBRG for a fluid film bearing with preload of 0.3, L/D ratio of 0.75, ISO VG 46 lubricant and maximum bearing static unit load of 480 psi. Additional parasitic losses from magnetic bearing eddy current and the gearbox were also considered. A factor of safety of 1.5 was also applied to allow robust speed control during a bearing test.

A summary of the motor specifications follows:

TABLE 2. Motor Specifications

Test Rig Characteristic	*Values, SI*	*Values, BG*
Output Powers Rating	261kW	350 HP
Base Speed (60 Hz)	3600 rpm	
Maximum Speed (73 Hz)	4400 rpm	

Figure 5. U.S. Electric Motor

Figure 6. Motor Drive

Speed Increaser

The preliminary design for the speed increaser has been provided by Lufkin Industries, Inc. The gearbox has been designed to deliver the rotational speed required for high-speed fluid film bearing tests and to deliver the input power required to overcome the parasitic losses from the fluid film bearing and the magnetic bearings. The gearbox will also be instrumented with radial probes, accelerometers, and RTDs to measure radial and axial vibrations and operating temperatures. The data from these probes will be used to validate combined torsional-lateral-axial finite elements developed by ROMAC by Blake Stringer, Jawad Chaudhry, and Amir Younan, and for other experiments involving geared systems.

Speed Increaser Horsepower Rating:	350 HP
Speed Increaser Gear Ratio:	1:5.0263

Active Magnetic Bearings

The FFBTR is being constructed with two magnetic bearings employed as the means of applying the static and dynamic loads to the single fluid film bearing. The magnetic bearings will be used as exciters to support the shaft and perturb it with small displacement and velocity motions. The bearings will be manufactured by Innovative Power Solutions, LLC. Rotating Machinery Technology is constructing the shaft and bearing housings.

The active magnetic bearings employ four e-cores separated into quadrants. The design characteristics of the active magnetic bearings are summarized in Table 3.

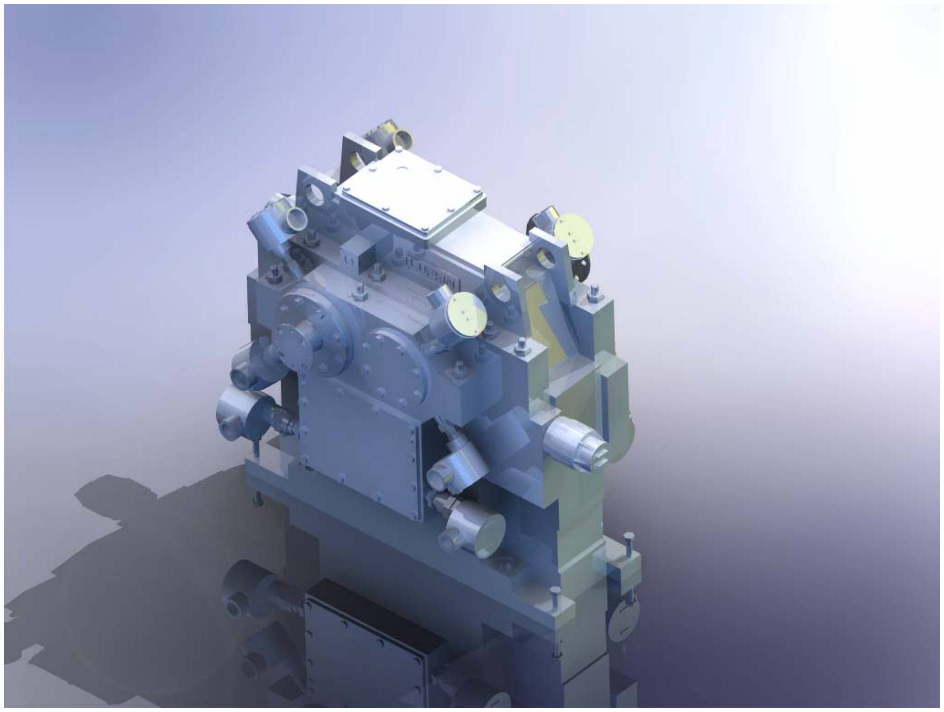
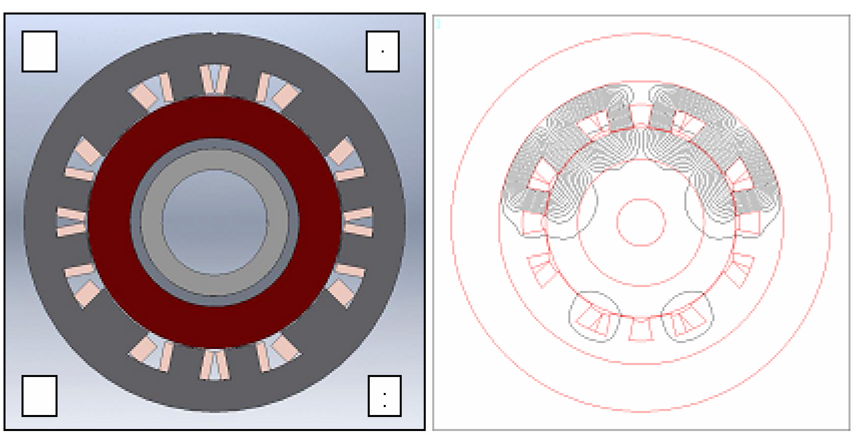


Figure 7. Lufkin Industries Gearbox

Table 3. Active Magnetic Force Exciter Characteristics

Force Exciter Characteristic	*Values, SI*	*Values, BG*
Magnetic Bearing Diameter	152.4 mm	6 in
Stator Back Iron Material	M-19 Magnetic Steel	
Rotor Target Material	M-19 Magnetic Steel	
Total Load Capacity, Per Bearing	11 kN, linear operation 13 kN, nonlinear operation	2500 lb_f, linear operation 3000 lb_f, non-linear operation
L/D Ratio	1	
Main Pole Width	38.1 mm	1.5 in
Auxiliary Pole Width	19.1 mm	.75 in



Active Magnetic Bearing Cross-Section **Magnetic Flux Line Plot**

Figure 8.

Static force design curves for the active magnetic bearings were developed for varying levels of bias current to determine when non-linear back-iron saturation effects become important. Based on the analysis, the bearing force profile is essentially linear. Based on this analysis, the bearing force profile is essentially linear for bias currents up to 27 A. For higher bias currents, the non-linear effects become increasingly important. Operation in the non-linear region will be necessary to perform some experiments on heavily loaded fluid film bearings. The maximum static unit load that can be applied while operating with bias current of 27 A is 2.2 MPa (320 psi). The API limit for bearing unit load in gearboxes is 3.4 MPa (500 psi) [13]. For a bias current of 36 A, a bearing unit load of 3.3 MPa (480 psi) can be applied, but the range of linear operation and the available slew are reduced.

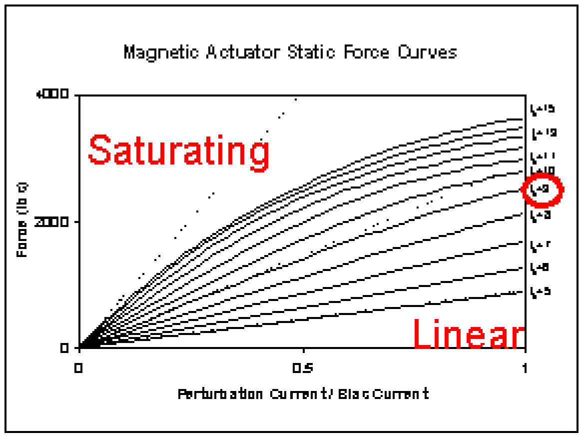


Figure 9. Static Force Design Curves, FFBTR Active Magnetic Bearings

The active magnetic bearings will act both as force application devices and force measurement devices. The theoretical correlation between magnetic flux and input force and magnetostructural strain and input force were developed by Bob Rockwell. This information is used to select either Hall sensors or fiber optic strain gages as the primary input force measurement method. A side-by-side comparison of the two sensors will be done with a separate experiment, which is described in the Force Tester section of the newsletter.

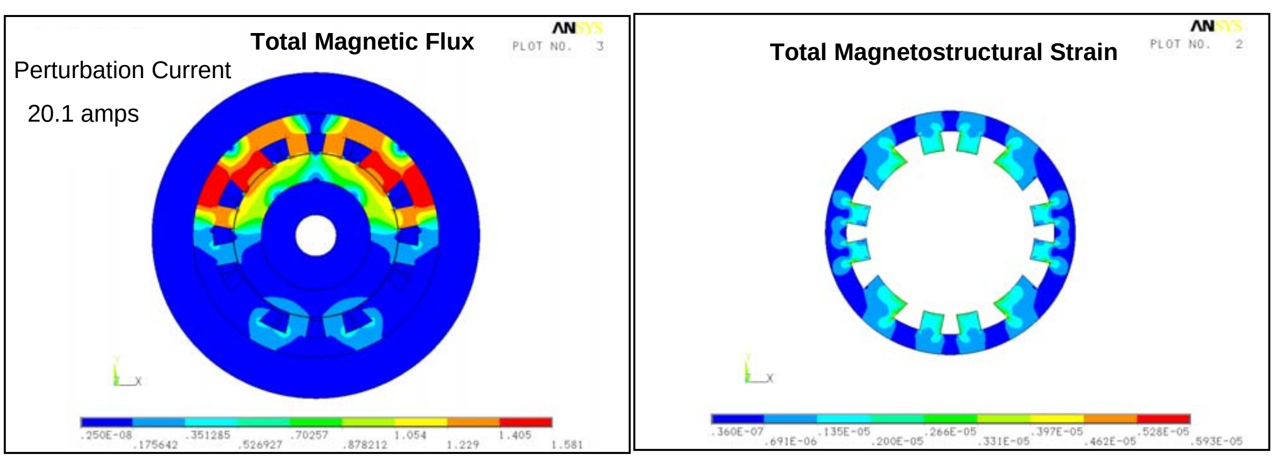


Figure 10. Magnetic Flux And Magnetostructural Strain Contour Plots

Rotordynamics and Control

Rotor modeling of the entire FFBTR train is ongoing. We are carrying out detailed critical speeds, unbalance response and stability analyses. The goal is to identify any undesirable modes that could adversely impact FFBTR dynamic measurements and mitigate or eliminate them. The analysis will include the use of the combined torsional-lateral-axial finite element developed in ROMAC by Blake Stringer, et all. The controllers for the active magnetic bearings are under development by Wei Jiang and Mohit Chhabra.

Static and Dynamic Measurements

In order to measure bearing stiffness and damping, we have to have a method of determining the force exerted by the magnetic bearings. We will be using either fiber optic sensors attached to the magnetic bearing radial poles, or Hall sensors placed in the magnetic bearing pole ends, in addition to measuring the currents in the magnetic bearing coils. A special force test rig with a single sided e-core bearing is being constructed to evaluate and establish the best methods for this. The force test rig is described in detail in a separate section of the newsletter.

This project includes three additional topics: identification of experimental data for fluid film bearings, full coefficient dynamic analysis of tilting pad bearings and measurement of squeeze film damper properties. Several papers on these topics have been written already and are available as ROMAC reports.

Shaft and pad motions will be measured using non-contact sensors and/or accelerometers for evaluation of full dynamic stiffness and damping, including pad properties. Pressure measurements will be taken in the rotating shaft to verify the computer calculations. Temperature measurements will be taken in the fluid film bearing and on the journal. Squeeze film dampers can be placed around rolling element bearings.

One of the challenges in obtaining on-journal data is obtaining telemetry at the rotational speeds required to produce full turbulence in the bearing lubricant. Conventional slip rings introduce noise into the measurements and have a top rotational speed of 10,000 rpm. Oil-lubricated bearings will be tested at twice that speed. Wei Jiang and Mohit Chhabra are developing data acquisition methods that can store the data on the rotor during a test for download after a test is completed.

Logistics

The FFBTR will share space with the ROSTR in the Aerospace Research Laboratory. New lab space has recently become available, which is a significant upgrade over the originally planned Quonset hut lab. High power electrical supply is being upgraded by the University of Virginia Facilities Management group. A concrete base for the FFBTR test rig will be poured soon. Remote operation will be carried out in a separate room.

Figure 11: Aerospace Research Laboratory-Location of FFBTR

1.2 Dynamic Force Measurement Validator

Students: Tanner Hall, Tim Dimond (ROMAC Test Lab Engineer)

Faculty: Paul Allaire, Pradip Sheth

Research Professionals: Wei Jiang, Robert Rockwell

The development of a technique to accurately measure forces exerted by the magnetic bearings on the Fluid Film Bearing Test Rig (FFBTR) is necessary to determine the applied unit load, stiffness, and damping in the rig's test bearing. The motivation behind the Dynamic Force Measurement Validator (DFMV) is to provide data showing which force measurement method is better for the FFBTR (with the least amount of uncertainty.) The DFMV will compare the force measuring capability of Hall sensors to fiber optic strain gages (FOSG). The FOSG method was originally developed by Texas A&M while the Hall sensor method has been used by Aenis & Nordmann and Knopf & Nordmann. The FOSG method has the benefit of being immune to electromagnetic interference, but could prove to be less accurate due to the expected low strain levels compared to the full scale of the sensors. The FOSG we will be using has a range of 0 – 500 microstrain, but we expect actual strains to be only 50 microstrain. While Hall sensors are more effective over the full range of magnetic flux, they are susceptible to noise and will require an alteration to the e-core geometry, either by increasing the air gap or countersinking within the e-core material.

The current design of the DFMV, shown in Figures 1 and 2, features a single-sided e-core linear magnetic actuator bearing suspended under a welded structure by an array of three dynamic load cells, which provide stability and a calibration standard for the rig. The e-core will levitate a target plate and experience strain and magnetic flux from the plate's variable set of weights attached underneath. The static forces caused by the hanging weights and the dynamic forces, introduced through perturbation frequencies, applied to the e-core will produce comparable strains and magnetic fluxes to those predicted in the radial magnetic bearings for the FFBTR.

Progress has been made in designing the surrounding welded structure with careful consideration to avoid any natural modes of vibration near our perturbation frequency range of up to 560 Hz. This corresponds to an excitation of 1.5x at a rotational speed of 22,400 rpm in the FFBTR. We have also acquired the fiber optic strain gages and Hall sensors and are currently working with Innovative Power Solutions to build the e-core with these sensors embedded.

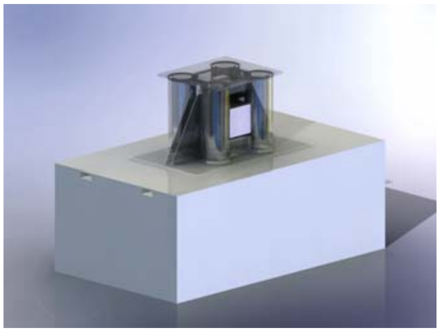


Figure 1: Perspective View of Dynamic Force Measurement Validator on Concrete Base

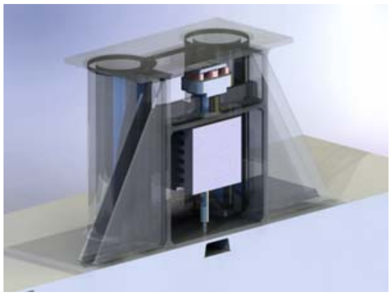


Figure 2: Cutaway View of Dynamic Force Measurement Validator

Figure 3: Force Tester E-core Magnetic Bearing

1.3 TEHD Analysis of Fluid Film Journal Bearings (Computer Code MAXBRG (5.0)

Student: Amir Younan

Faculty: Paul Allaire

Project Start Date: September 1996

Report Number: 458

Project Overview

Fluid film journal bearings are widely used in turbomachinery. Accurate prediction of the bearing properties is critical to a machine's design and analysis. The objective of this project was to develop a state of the art thermoelastohydrodynamic (TEHD) algorithm for industrial journal bearing analysis.

Many advanced models are utilized in this finite element based algorithm. The pressure is calculated from the generalized Reynolds equation. The pad and film temperatures are obtained from a unique energy equation that combines computational efficiency and broad capability. Turbulence is automatically handled. Deformations of the pad, journal, shell, and pivots are all taken into account. A coupled film-pad approach is employed to achieve good numerical robustness.

This algorithm can be used to analyze directly lubricated bearings as well as conventional fixed geometry and tilting pad bearings. Modeling of pressure dam bearings is extended to include adiabatic thermal effects. In addition to the normal flooded condition, this algorithm handles several special operating conditions, including starvation, high ambient pressure and axial flow. Moreover, the computer code is flexible in that the users can select and combine a variety of modeling options according to their specific needs.

Future work

MAXBRG will continue to be maintained and upgraded in the future. As in the past, upgrades will largely be based on feedback from ROMAC members. Your input is very important and will be highly appreciated. In the past year, pivot rotational stiffness was added. This new feature allows the modeling of tilting pad bearings whose pivots are not free to rotating. The next plan is to improve MAXBRG so that it is suitable for water bearing analysis. Water lubrication is a rather new subject and has high interest among ROMAC members. More details about water lubrication will be discussed later.

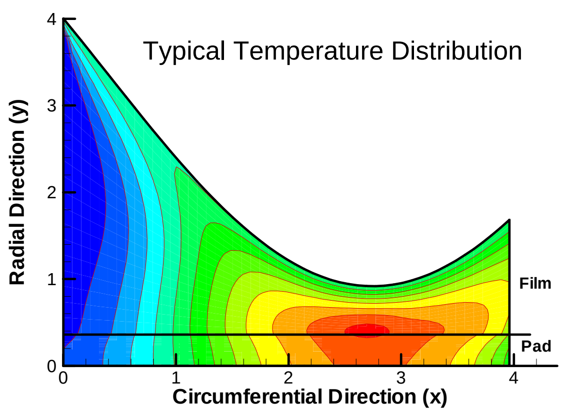


Figure 1

MAXBRG Structure

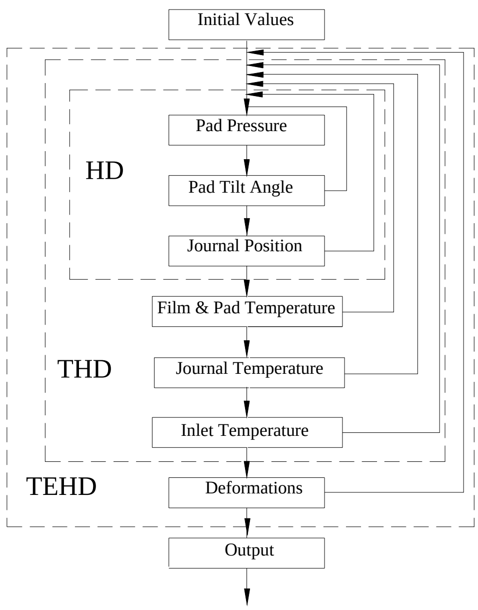


Figure 2

Progress of Starvation

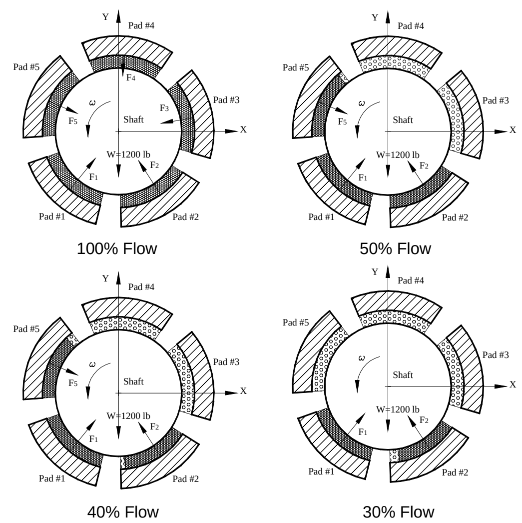


Figure 3

Mechanical Deformation of a Tilting Pad

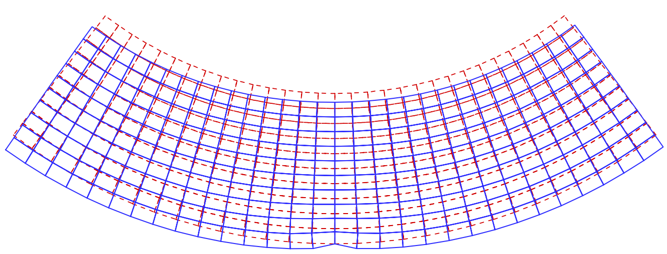


Figure 4

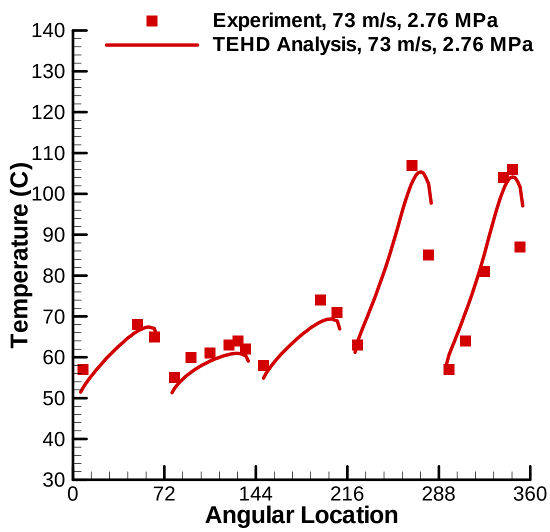


Figure 5

1.4 Water Lubrication

Student: Amir Younan

Faculty: Paul Allaire

Project Start Date: August 2006

The majority of fluid film bearings are lubricated by oil that has relatively high viscosity. Consequently, the classic lubrication theories were developed based on oil. However in recent years, more and more ROMAC members are interested in modeling bearings that use water as their lubricant. Compared to oil, water has much lower viscosity. Naturally, some questions have been raised. How accurate are the existing codes when used to predict water bearings? And how to improve the predictions if necessary?

To answer these questions, ROMAC has teamed up with some members to investigate water lubrication. The first subject is turbulence. Due to much lower viscosity, water-lubricated bearings are almost always highly turbulent. And turbulence is an important source of load carrying capacity. In a case we examined last year, turbulence accounted for up to 50% of the load capacity. Thus, it is necessary to have a suitable turbulence model in the bearing codes. The turbulence model used in MAXBRG was assessed through comparisons with CFD simulations. For the cases investigated, MAXBRG predicted load capacity that was about 9% lower than that from the commercial software ANSYS-CFX. The turbulence model used in the existing ROMAC code MAXBRG (and THRUST) seems to be reasonably accurate.

The second issue is fluid inertia. The classic lubrication theory employs the Reynolds equation that neglects the fluid inertia. One basic assumption for the Reynolds equation is that the viscous force is dominant compared to the inertia force. This assumption is valid for oil that is fairly viscous. But for water, dimensional analysis can prove that inertia force is not negligible. Thus, it ought to be included in the governing equations. The 9% static load deficit in MAXBRG could largely due to the absence of inertia effect in the pressure calculation. Dynamically, the inertia force has additional temporal component that makes it even more significant. Our preliminary study indicated that, in addition to the conventional stiffness and damping coefficients, mass coefficients could be necessary to represent the linearized bearing force.

The inclusion of fluid inertia requires a major upgrade for the Reynolds equation. This has been implemented. It is clear that the fluid inertia effects can be successfully included without solving the full Navier-Stokes equation.

There are some other issues related to water lubrication. For example, after entering the tight clearance, lubricant slows down due to the adverse pressure gradient. Per Bernoulli, pressure is expected to rise. Because of large inertia, such local pressure surge is more likely to be observed in water bearings. Moreover, water bearings are usually working in submerged condition (water is also the process fluid). Some of them have many more pads than typical oil bearings. All these may require special treatments for accurate modeling. The new development for water lubrication will be applied to the journal bearing code MAXBRG and thrust bearing code THRUST.

1.5 Time Transient Squeeze Damper Analysis (New Computer Code)

Student: Amir Younan

Faculty: Paul Allaire

Project Start Date: January 2008

Project Overview

The existing ROMAC computer code was written over 20 years ago. Since then, significant progresses have been made in squeeze film damper's design and analysis. There is a wide range of designs involving various retaining springs, end seals and oil supply mechanisms. Recent studies also indicate that some physical phenomena must be included to accurately predict a damper's performance, for instance, cavitation and fluid inertia. Our objective is to write a state of the art transient computer program that accurately predicts the film force, flow rate and power loss. When combined with the rolling element bearing code that is currently under development, it will serve as an even more valuable tool for the gas turbine industry. In this early stage, work is focused on theoretical development and collection of design and experimental data. The actual programming is expected to begin in early 2008.

1.6 Rolling Element Bearings

Student: Amir Younan

Faculty: Paul Allaire

Rolling element bearing analysis and computer code development continues at a relatively rapid pace. This includes both cylindrical rolling element and ball bearings. An innovative finite element solution method has been developed for the elastohydrodynamic (EHL) solution of Reynolds equation for the pressure and the elasticity equation for the solid surface deformation. The new solver is based on solving those two equations simultaneously which capture the interaction between these two fields. A major advantage of this finite element method over the previous multi-grid finite difference methods is that the finite element solution process is implemented automatically for a wide range of geometries and loads in contrast to the previous multi-grid method which requires much fine tuning.

This analysis is now being extended to the full rolling element bearings geometries. The balls or rollers, the races, and the cages for these bearings will be included. Generally rolling element bearing analysis computer modeling codes are developed by manufacturers. However, these codes focus on steady state operation and life predictions. They are very good at that. However, they do not predict the dynamic properties of rolling element bearings very well. The focus of the ROMAC analysis methods and computer code is on dynamic analysis such as stiffness and damping as well as nonlinear dynamics. Rolling element bearing manufacturers are also working to develop EHL codes.

One objective is to couple this rolling element bearing dynamic analysis with the squeeze film damper analysis for use with rotordynamics of aircraft gas turbine engines and ground based gas turbines. It is expected that this will be integrated with NASTRAN for gas turbine applications.

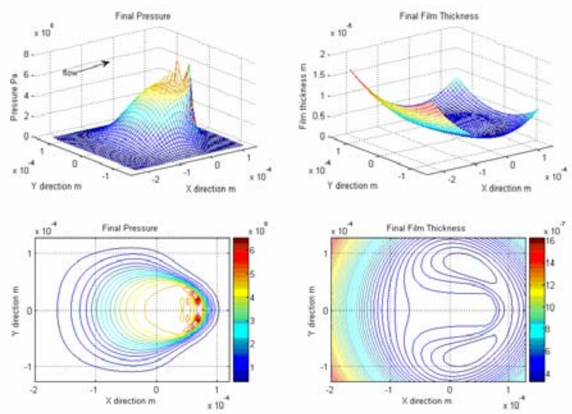


Figure 1 Point Contact Results

The pressure distribution topology (upper left) and contour plot (lower left)

The film thickness profile topology (upper right) and contour plot (lower right)

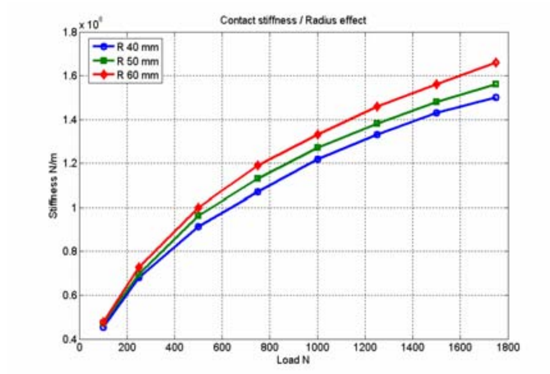
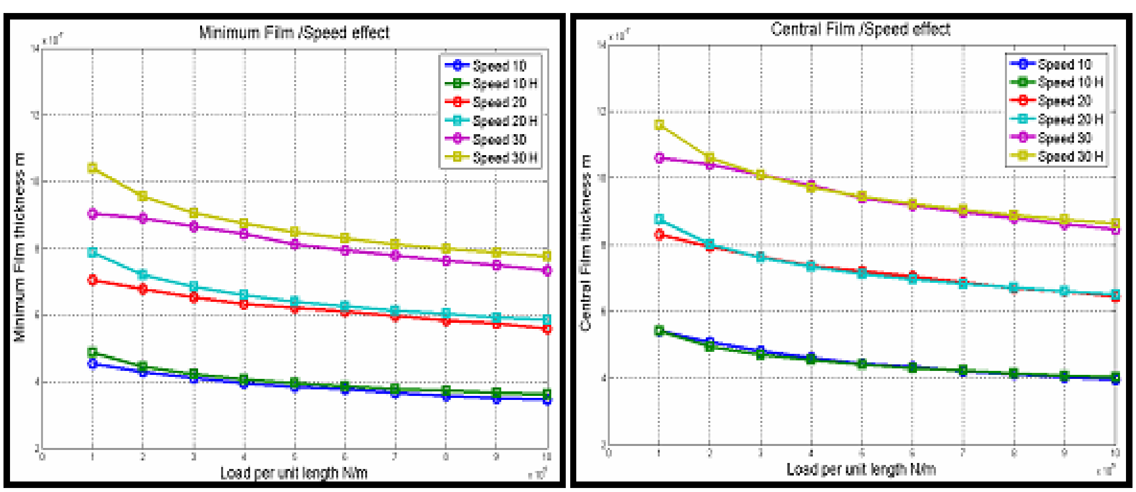


Figure 2 Point Contact Stiffness

The contact stiffness calculated by the perturbation method applied to the steady state solution of the pressure and film thickness. (Variation with the ball radius)



Code Result Verification
The film thickness profile was compared with the Hamrock Dowson equations.
The minimum film thickness (left figure) and the central film thickness (right figure)

Figure 3

2.1 Industrial Annular Seals Analysis

Student: Patrick Migliorini

Faculty: Houston Wood, Paul Allaire, Chris Goyne

Research Professional: Alex Untaroiu

Annular seals in compressors, turbines, and other rotating machines are commonly used to limit leakage between different pressure regions and are extremely important to rotordynamics. They are also some of the least well known components in any advanced rotordynamic analysis of high performance machines. ROMAC is developing detailed seal analysis for industrial annular seals using computational fluid dynamics (CFD) as the best method for obtaining the flow rates, velocity and pressure distributions, stiffness and damping coefficients. This analysis applies to labyrinth seals and hole-pattern seals (Figure 1), but can also be extended to more complex seal geometries.

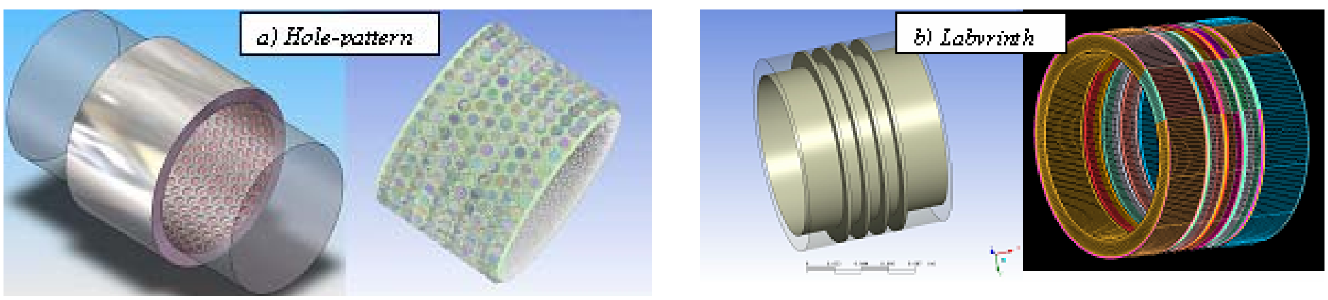


Figure 1: Seal Geometry and corresponding Computational Domain

Fluid forces arising from the flow in these seals exert a strong influence on the dynamic characteristics of the turbomachinery and can cause a rotor to become unstable. Therefore, as related to rotor dynamics, seal analysis has the objective of determining the reaction forces acting on the rotor arising from shaft motion within the seal. Prediction of dynamic coefficients using a code based on bulk flow theory is relatively simple and computationally efficient. On the other hand, this solution technique, either a one-control volume or three-control volume, provides only a rough estimate of a seal's performance because it requires characterization of interfaces between control volumes, and these empirical parameters require calibration with experimental data.

As an alternative to bulk-flow codes, CFD has the potential of providing a better accuracy of the solution by solving the full Navier-Stokes (N-S) equations as opposed to solving only a simplified version of continuity and momentum equations as derived in the bulk flow theory. There are a nuthe one on which Ansys-CFX is based, is known as the finite volume technique. In this technique, the region of interest is divided into small sub-regions, called control volumes or grid elements. The N-S equations are discretised and solved iteratively for each control volume, and as a result, an approximation of the value of each variable at specified points throughout the domain is obtained. In this way one can derive a full picture of the behavior of the flow. However, the major advantage of CFD methods is that CFD analysis provides details of the local flow dynamics, consequently one can get a better understanding of the flow pattern; and all these can lead to a seal design optimization.

A method to calculate leakage and rotor dynamic coefficients has been developed considering steady-state flow and known temperatures and pressure drops using a 360 degree seal geometry. Even though the seal geometry is axisymmetric and only a small angular section can be modeled to accelerate convergence and reduce computational time, a full 360 degree analysis of the seal geometry is carried out to determine the stiffness and damping coefficients (Figure 2). A full model is required in order to capture the eccentricity generated by the off-centered position of the impeller, so that its subsequent impact on the magnitude of radial fluid forces is incorporated.

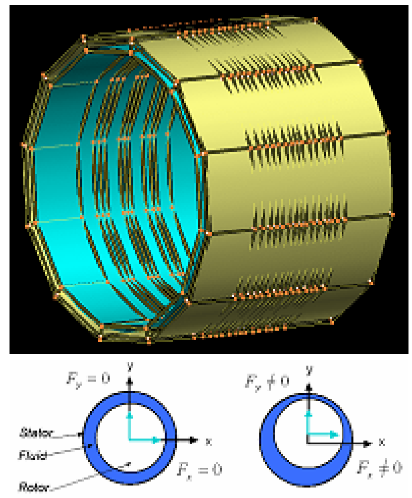


Figure 2: Topology constructed to generate a full 360 deg computational model. Same topology can be used for both centered and off-center rotor positions

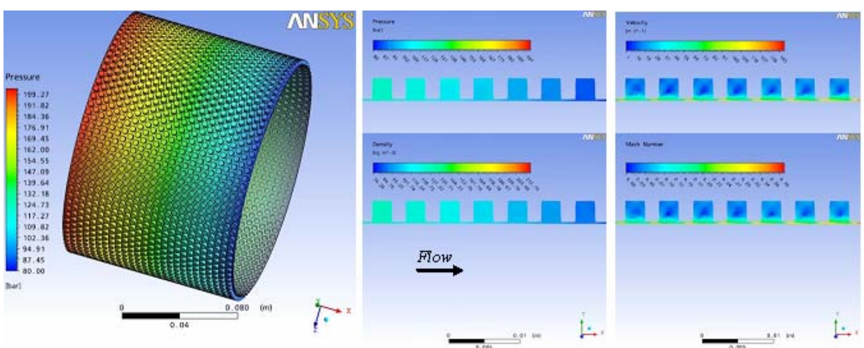


Figure 3: CFD Results for Hole-Pattern Seal Model

The largest difficulty is the development of accurate meshing of the seal geometry since the number and the size of grid elements has a direct impact on the CFD solution. Usually, the number of elements is decided based on a grid convergence study performed to determine the trade off between solution accuracy and the computational cost involved. To obtain a good quality computational model, the mesh is more refined along the walls to capture the details of the boundary layer. Also, more elements are needed in regions where large gradients are expected to occur; for instance, in the radial clearance region along the tip of the teeth and also in the direction of the jet flow. The disadvantage of the CFD approach to calculate leakage and rotor dynamic coefficients is that this is a time consuming and computationally intensive method. The more intricate the seal geometry is, the larger the number of grid elements, and the longer the convergence time is. However, this disadvantage is greatly diminished by taking advantage of the parallel-processing capability of the Ansys-CFX solver and the use of research computer clusters available at UVA. The mesh for the hole-pattern seal illustrated in Fig.1 has approximately 3 million elements and is the largest computational seal model developed and run so far at ROMAC.

In the following months, the CFD calculations for pressure, temperature, and corresponding velocity field distributions will be compared against PIV experimental data taken on a mock labyrinth seal as illustrated in Figure 2. This mock seal simulates an angular section of the shaft. More details about the mock seal and the PIV method employed are given in the next section. This comparison will be used to validate the CFD estimations and also identify the most suitable turbulence model to be used in simulating this type of flow (a turbulence model capable to reproduce as close as possible the velocity field distribution as given by the PIV measurements).

"To obtain a good quality computational model, the mesh is more refined along the walls to capture the details of the boundary layer.

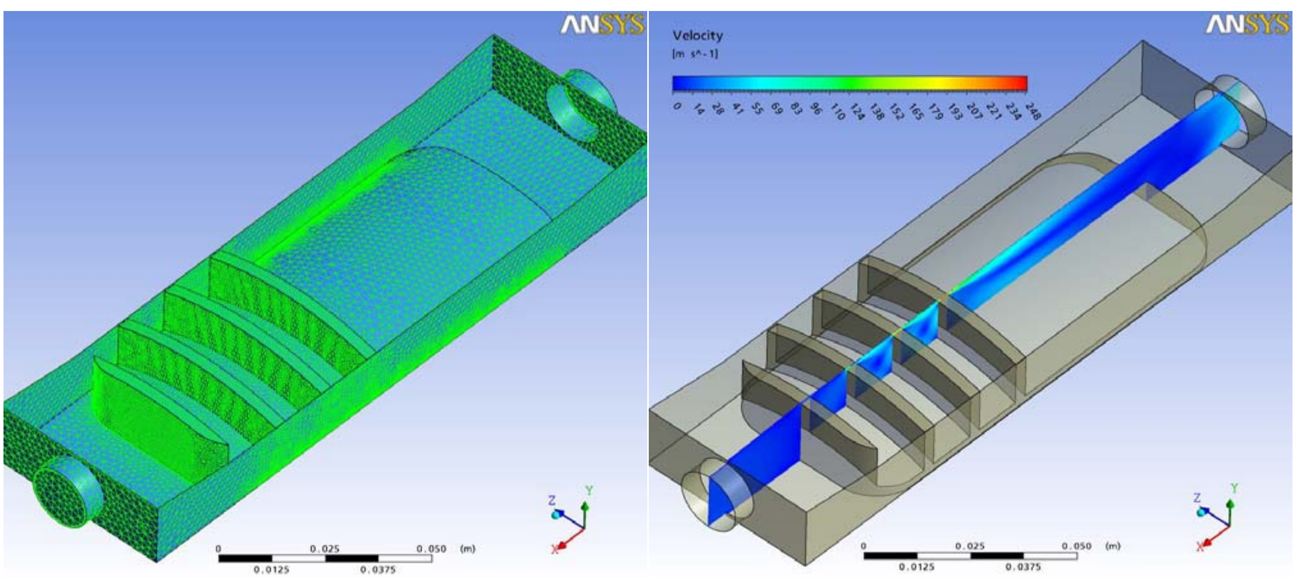


Figure 4 CFD Model of Mock Labyrinth Seal

2.2 Seal Design Optimization

Student: Patrick Migliorini

Faculty: Houston Wood, Paul Allaire, Chris Goyne

Research Professional: Alex Untaroiu

A new approach to seal design will employ advanced optimization approaches to improved seal geometry (Fig. 1). The generic geometry of seal will be chosen and its design process will be cast as an automatic optimization problem in the following standard form: minimize $f(x)$, with $x=\{x_1, x_2 \ldots x_n\}$, subjected to the constraints of the form $a_i \leq x_i \leq b_i$; $i=1,n$; where $f(x)$ is the leakage rate selected as the objective function and x is a vector whose components are the main seal dimensions x_i, selected as design variables (e.g. clearance ,tooth width, cavity width, ect.) The ranges of design variables (a_i, b_i) will be chosen based on the manufacturing constraints. The optimum design will be determined using advanced optimization algorithms, such as: Adaptive Response Surface Methodology, Genetic Algorithm, etc.

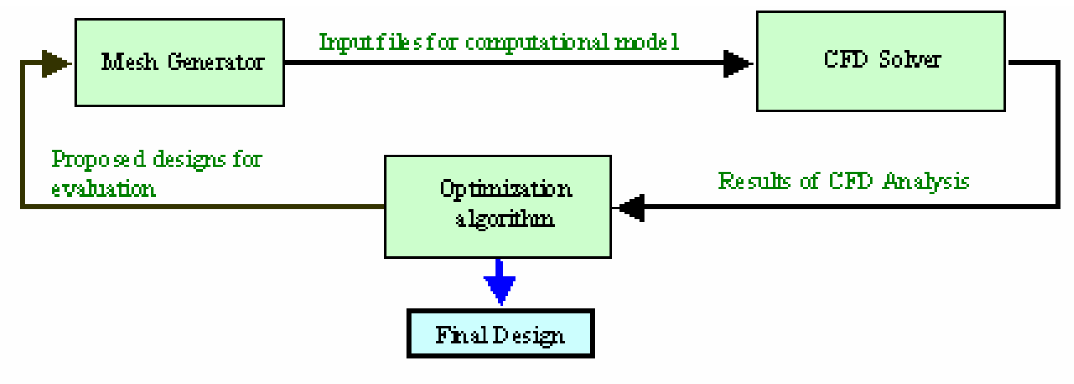


Figure 1 Schematic of automatic geometry optimization for seal design

2.3 Annular Seal Test Rig

Students: Marion Reid, Amir Younan, Tim Dimond, Josh Keely, Tommy Meriwether

Faculty: Chris Goyne, Paul Allaire, Houston Wood, Roland Krauss, George Gillies

The ROMAC Seal Test Rig (ROSTR) again received a high interest rating after the Annual Meeting. Continued improvements to compressors and turbines require accurate predictions of leakage rates and dynamic coefficients. There is a great deal of room for improvement in the current modeling/prediction techniques, including bulk flow codes and CFD. However, in order to drive these improvements, the amount of experimental data related to seals must be increased. Few rigs exist that are capable of measuring dynamic coefficients, and no 3D, full-field velocity measurements for gas flows in seals could be found in the literature.

The purpose of the ROSTR is to provide a library of seal measurements, including pressures, flow rates, temperatures, and dynamic properties (stiffness and damping). Additionally, the rig will provide velocity field measurements in the seal. From these velocity fields, other flow characteristics like vorticity and turbulence intensity can be determined. The design will allow various seal geometries to be tested with minimal disassembly, including tooth-on-stator, tooth-on-rotor, and interlocking labyrinths, as well as gas damper seals and possibly brush seals.

The design of the seal test rig is progressing steadily, and initial steps in the construction will begin shortly. The test rig consists of a seal test chamber, two magnetic bearings to excite the shaft, a high-speed coupling, a gear box, a motor and DC drive, and a pressurized housing, all mounted on a concrete base. Figure 1 presents the conceptual arrangement of these components, except for the DC drive. The visualization of the seal test section will be presented later. The seal test section has a shaft diameter of five inches. The exact dimensions of the seal inserts have yet to be determined, but seals with a L/D ratio up to 0.75 should be accommodated. Applied pressure drops of up to 3,000 psi will be obtained with gas supplied from a series of high pressure cylinders. Cylinders were determined to be the most cost-effective and efficient way of obtaining the desired pressure drops. Two identical seals will be placed in a back-to-back configuration within the test section in order to balance the thrust loads as much as possible. Interchangeable swirl vanes will be located before each of the seals so various preswirl ratios can be obtained.

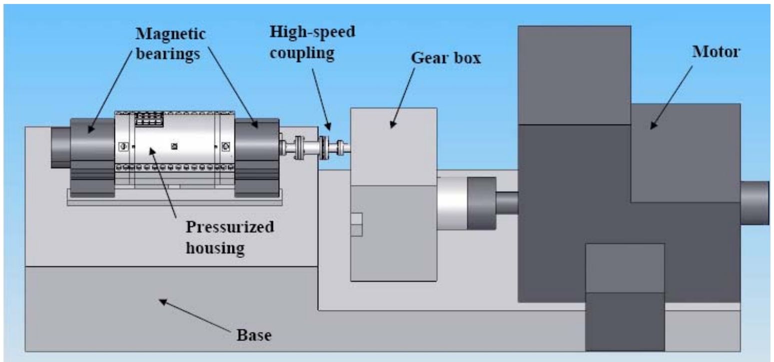


Figure 1: ROSTR Component Layout

Several components for the rig have already been obtained, and will soon be ready for assembly. The motor, DC drive, and gear box were all donated to ROMAC by Solar Turbines some years ago for use with another project. They allow shaft speeds in excess of 22,000 RPM. The motor has a maximum speed of approximately 1750 RPM, and the gear box increases the speed at a ratio of approximately 1:12.75. The motor, shown in Figure 2, and drive have been inspected by the motor manufacturer and were deemed as working properly after some minor refurbishment. The gearbox, as seen in Figure 3, was inspected by a Philadelphia Gear representative and found to be fully operational. Rotordynamic analysis of the test rig design is currently underway. An initial free-free analysis, which did not include the stiffness of the magnetic bearings, has the first critical speed at approximately 44,000 RPM, or twice the maximum running speed.

Figure 2: ROSTR Motor

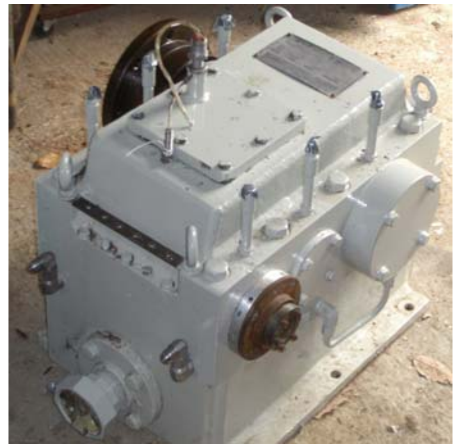


Figure 3: ROSTR Gear Box

The design of the pressurized housing is currently a major focus of attention. It must be capable of safely handling initial cylinder pressures of up to 6,000 psi in accordance with the ASME Boiler & Pressure Vessel Code. Professional assistance is required in order to ensure a proper design. Figure 4 shows an exterior view of the ROSTR casing, with initial diameter and thickness estimates at 12 in. and 2.25 in. respectively. The bearing housings are separated from the casing to limit the effects of the high pressure flow impinging on the bearings. Additionally, this also allows the seal test sections to be interchanged without disassembly of the bearing housings. The bearings are identical to those used in the Fluid Film Bearing Test Rig (FFBTR), and a detailed description of the bearing system can be found in the FFBTR article. A separate, backup bearing system consisting of a rolling element bearing on each side of the casing will be installed to avoid backward whirl and prevent a rotor drop onto the magnetic bearings.

Another major effort is being focused on the flow path through the seal test section. Appropriate methods for back pressure control and safe outlet paths for the high pressure air are being determined. Presently, the pressurized gas enters the rig through an annulus in the center of the test section and flows axially through the rig to exhausts on either side. It initially passes through the swirl vanes, then through the test seals. Solenoid valves downstream of the test seals will allow for back pressure control. In order to isolate the stiffness and damping coefficients of the test seals from other flow control seals in the rig, the other seals will be oriented perpendicular to the test seals. These secondary seals will serve to regulate the main exhaust flows which exit

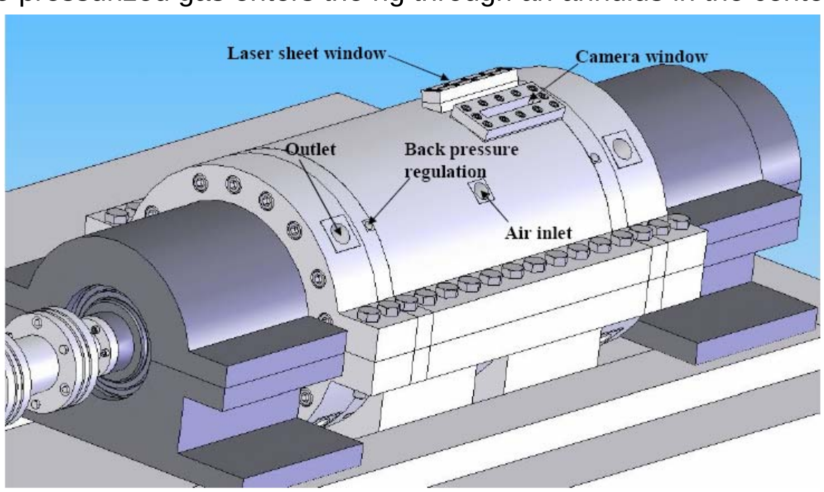


Figure 4: ROSTR Casing Design

prior to the magnetic bearings. Figure 5 shows a cross-section through the side of the casing, with the inlet annulus on the far left. The alternate side of the test section is symmetric about this inlet.

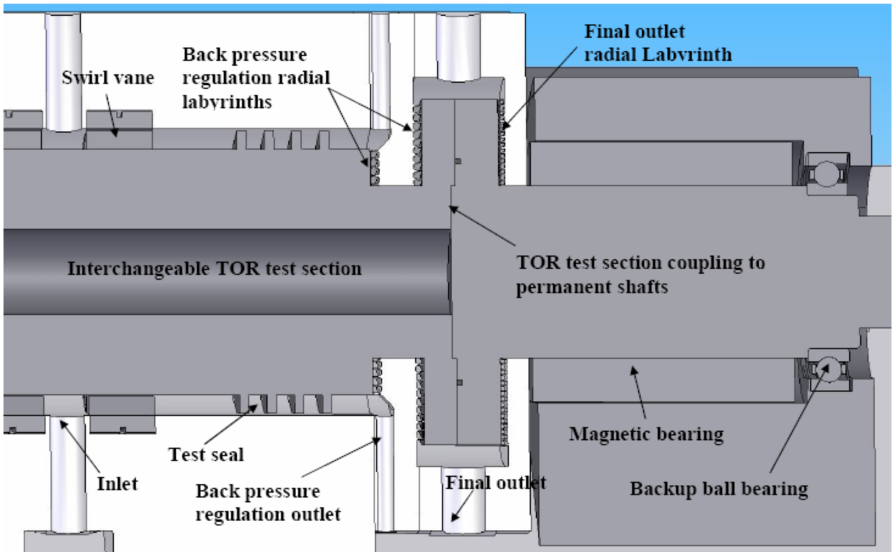


Figure 5: ROSTR Cross-Section

Concerning instrumentation and measurements, the pressures, flow rates, and temperatures will be measured by conventional instrumentation. The seal dynamics will be measured using the magnetic bearings, in a manner similar to that used in the FFBTR, as previously described. The velocity fields will be measured using Particle Image Velocimetry (PIV), a full-field, laser-based measurement technique. Optical windows in the pressurized housing will allow laser access, and enable measurements with the seal. Tooth-on-rotor labyrinth seals offer the most attractive configuration for velocity measurements, but the technique can also be applied to the seal inlet area for preswirl measurements.

The feasibility experiment to test out the PIV technique is nearing completion. A mock labyrinth seal has been constructed, simulating an angular section of the shaft. It is surrounded by a pressurized housing, and an acrylic window allows laser access. The experiment does not consider rotation, and allows for a maximum inlet pressure of 75 psi. It has allowed ROMAC to learn much of what is necessary to implement the PIV technique into the ROSTR, including discovery of the paint that is necessary to coat the ROSTR rotor to allow for near-wall measurements. It is anticipated that data from the mock seal will be available shortly. Figure 6 shows the setup for the feasibility experiment, and Figure 7 shows the PIV laser in operation.

The schedule for completion of the ROSTR itself is also taking shape. Both the ROSTR and the FFBTR will be located in the Aerospace Research Laboratory, shown in Figure 8. The available lab space is already equipped to handle most of the loads necessary for the two rigs. The PIV feasibility experiment will determine the optical window requirements for the pressurized housing, and will allow its design to be finalized, and manufacturing to commence. If everything proceeds as planned, initial results should be available by early 2009.

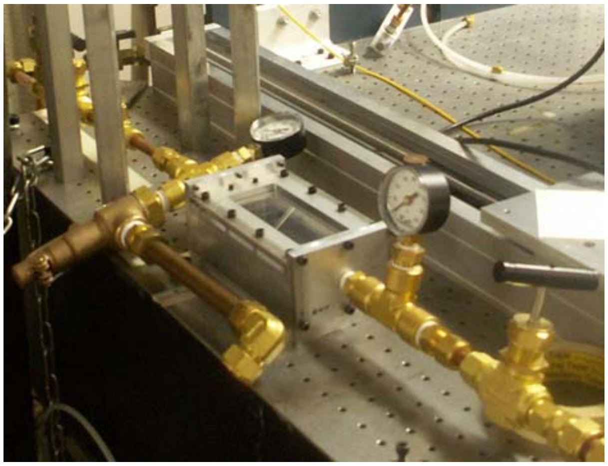


Figure 6: Mock Labyrinth Seal Feasibility Experimental Setup

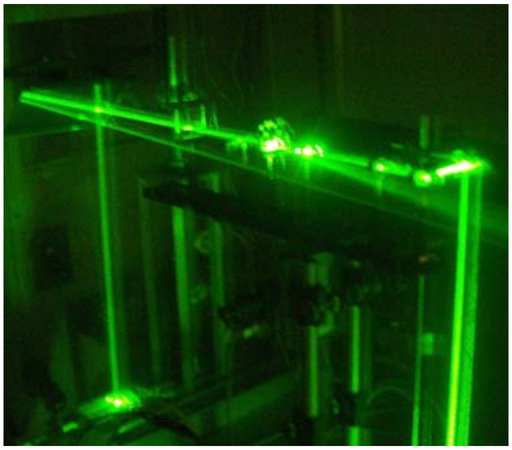


Figure 7: PIV Laser Path to Seal Feasibility Experiment

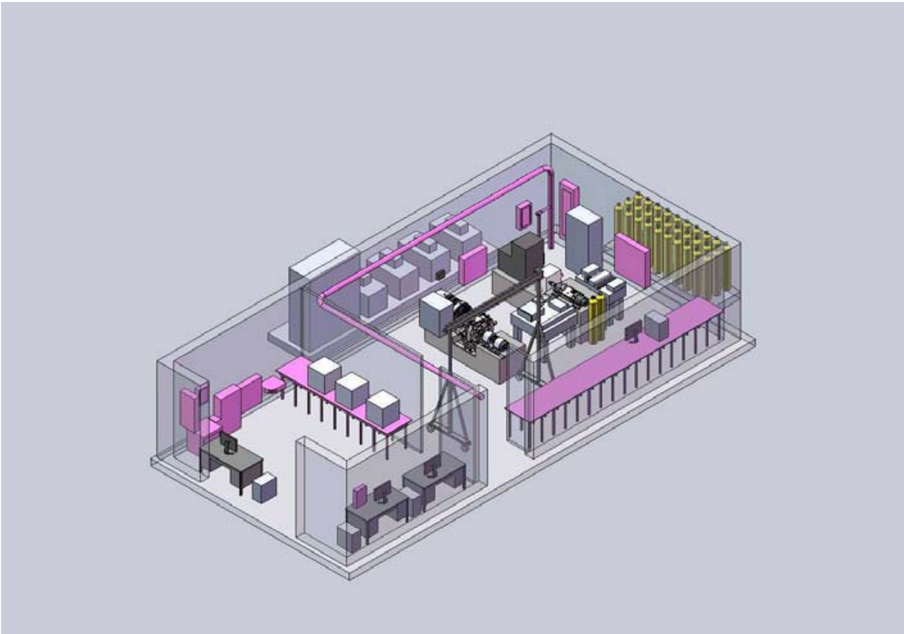


Figure 8 Aerospace Research Laboratory

3.1 Compressor Surge Test Rig

Students: Kin Tien Lim, Pablo Yoon, Vincent Chenal, Matt Goodhart

Faculty: Zongli Lin, Chris Goyne, Paul Allaire, Eric Maslen

Research Professionals: Wei Jiang, Dorsa Sanadgol

Compressor surge is a major problem in industrial compressors. This project seeks to use an innovative approach with magnetic thrust bearings to suppress surge. This new method for active surge control in the centrifugal compressor is to use the magnetic thrust bearings to modulate the unshrouded impeller's tip clearance during surge. If the position of the shaft can be actuated with sufficient authority and speed, the induced pressure from the modulation could suppress the surge.

A theoretical model of the compression system was developed by former PHD student Dorsa Sanadgol. The model simulates the flow dynamics of the compressor with the axial modulation of the impeller, and it gives a good basis to develop the surge controller. Initial simulation results has been promising, showing compressor surge suppressed within acceptable impeller clearance, typically within 20% or less of the available clearance, during these axial excursions of the impeller.

Figure 1: Compressor front view

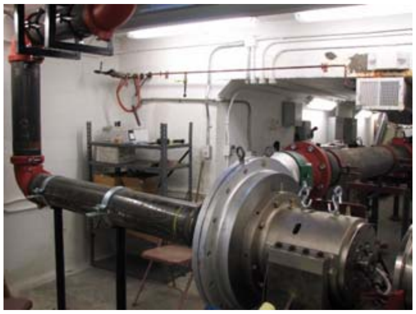


Figure 2: Discharge piping and supports

Figure 3: Discharge piping and supports

Figure 4: Discharge piping and supports

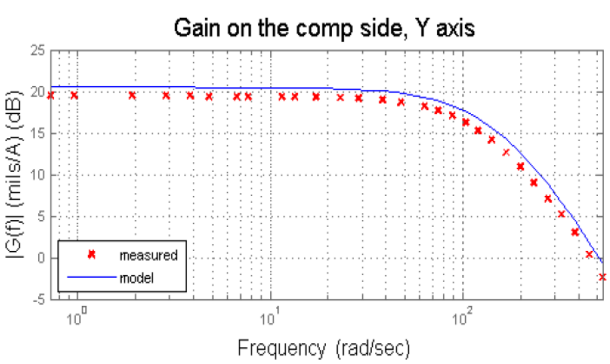
Further effort on improving the surge controller is being continued. Current work addresses the control of surge when the mass flow rate measurement is not available for feedback to the controller, and when the magnetic bearing dynamics are included to the analysis of the control problem. Recent theoretical and simulation results confirm that this flow instability can be suppressed using solely information of the pressure rise. A robust surge control scheme will be necessary in order to guarantee stability of the compressor during implementation, where the uncertainty of the thrust bearing dynamics will affect the stability of the overall system.

The next phase of this project aims to implement the surge control on the high speed centrifugal compressor test facility. The results obtained from the test facility will be used to validate the model and refine it as required. The test rig can also be used for assessing the impeller and bearings loads at off design conditions. A fluid flow model of the impeller is also planned to be developed jointly with Kobe after the compressor is commissioned.

The centrifugal test facility comprises of a single stage centrifugal compressor supplied by Kobe Steel and a variable frequency drive induction motor used for direct drive of the compressor. Compressor design speed is 23000rpm. Kobe Steel continues to work with ROMAC to get this project up and running.

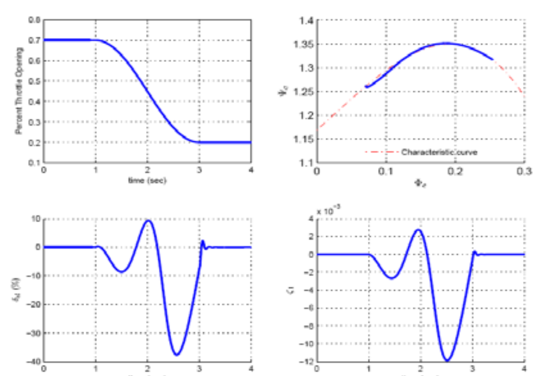
Building and assembling of the test facility is ongoing. The motor is back on site after a shaft repair and modifications to improve its bearings. Its cooling system has been set up. Magnetic bearing sensor calibration and model validation tests are being carried out. Recently the compressor has been reassembled and the rotor magnetically levitated and sensors calibrated. The piping system construction is progressing with the supports being fabricated and flow measurements equipment to be purchased. The motor has also been recently solo run at part speed to assess its health and conditions.

Figure 4



Compressor model validation test

Figure 5



Robust surge control with bearing dynamics

4.1 Rotordynamics

Students: Amir Younan, Jawad Chaudhry, Andy Edwards, Blake Stringer

Faculty: Pradip Sheth, Paul Allaire

The RotorDynamics activity in ROMAC includes a number of interrelated efforts. The following schematic illustrates the capabilities of the available RotorDynamics software tools from ROMAC—these are the programs which have been developed, refined, and matured through many years of industrial and academic usage:

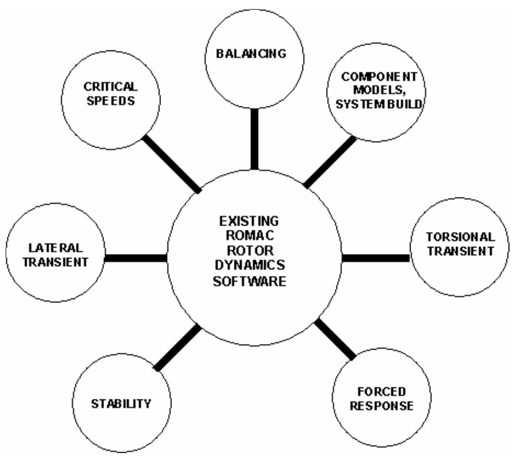


Figure 1

The ROMAC RotorDynamics research and development activities can be described by the three pronged approach illustrated in the following diagram:

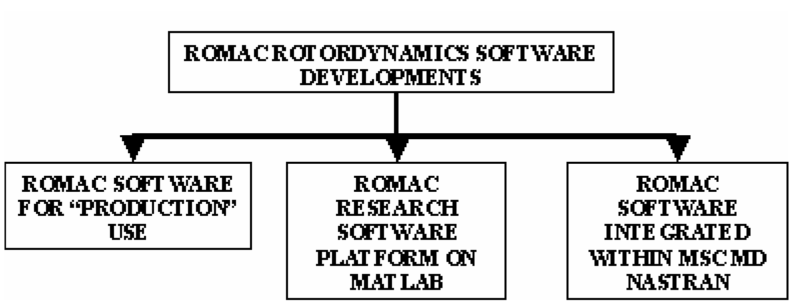


Figure 2

The "production" use software is the software utilized by the ROMAC members, and includes such existing ROMAC software tools as:

- Lateral Stability FORSTAB, ROTSTB
- Lateral Forced Response FRESP2, RESP2V3
- Lateral Critical Speeds MODFR2, CRTSP2
- Torsional Transients TORTRAN3
- Torsional Critical Speeds TWIST2
- Lateral Transients COTRAN

- Balancing BALOPT
- Component Modeling such as THPAD, THBRG, MAXBRG, and SEAL3 for creating bearing and seal coefficients for rotodynamics

These are just a few examples of the "production" software available to ROMAC members. Detailed User manuals and theoretical background are documented in the ROMAC library available on the web and on a CD. These "production" programs are the primary tools which are continuously updated and maintained by ROMAC. Elsewhere in this newsletter, Amir Younan and Jawad Chaudry describes the continuing developments of these "production" tools. In addition, a fully integrated system called COMBOROTOR is under development, with Beta versions already released, which will combine the capabilities of all of the different codes indicated above.

New and emerging issues in RotorDynamics have motivated two new developments at ROMAC RotorDynamics this year. These motivating issues include:

- Need to continuously correlate and calibrate the “production” tools
- Larger, more detailed models including foundation support structure
- Geared drives leading to coupled torsional/lateral vibrations
- Developments for Model reconciliation procedures
- Nonlinearities, including ability to analyze large orbits and limit cycles
- Model based diagnostic systems
- Modeling tools for repair/replacement of subsystems in existing machinery
- Need to investigate and implement Model reduction schemes for specific applications

To address these issues, a research software platform in MATLAB is established to provide an in-house research tool for RotorDynamics to experiment with new ideas that have not been investigated before. MATLAB is chosen for its extremely efficient and easy-to- use built in functions for numerical procedures for large and complex matrices, a very large selection of numerical integration algorithms for differential equations, and excellent visualization tools. The entire system is written in a compact and general form with explicit sub functions that are called during the modeling of the system. This platform, for example, is being utilized for integrating gear models described elsewhere in this newsletter.

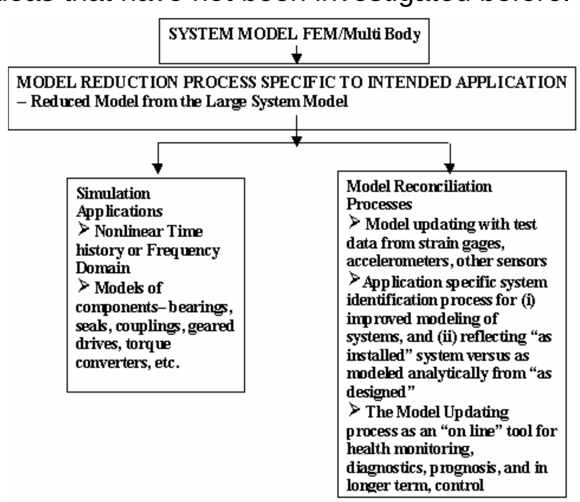


Figure 3

The ability to perform diagnostics, and more importantly prognosis of rotating machinery, is tied to modeling research in model reduction schemes and component models as illustrated.

The efforts in this area include the work on Helicopter power train modeling and the integration of MSC NASTRAN with ROMAC codes.

4.2 Coupled Later-Torsional-Axial Rotordynamic Analysis (Combo rotor/Matlab rotor)

Students: Jawad Chaudry, Tim Dimond

Faculty: Pradip Sheth, Paul Allaire

Project Start Date: October 2006

Project Overview

This project aims to advance the ROMAC rotordynamic software in several areas. First, it combines the capabilities of the most popular rotordynamic codes, including CRTSP2, TWIST2, ROTSTB and FORSTAB, into one code. Thus, it will be much easier for the user to conduct a comprehensive rotordynamic analysis. Second, it employs the finite element method that is more capable and reliable than the transfer matrix method used by some existing codes. Third, the tilting pad bearings and flexible support are modeled in the time domain, which eliminates a search that can potentially leads to missing modes. In addition, it achieves the new capability to analyze coupled lateral-torsional-axial vibration that exists in some geared systems. This new steady state computer program is currently under development and will be the main solver behind the new ROMAC GUI. A Fortran version (comborotor) and a Matlab version (Matlab rotor) are under development.

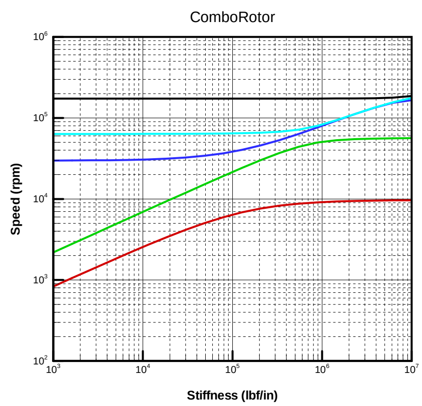


Figure 1

Progress in the Past Year

Major progress has been made in the past year. A beta version that performs lateral analysis was released in June 2006.   Since the new GUI is still under development, Comborotor is currently incorporated in the old GUI (shell) and uses TECPLOT for its post-processing graphics. Shear deformation and rotary inertia are included in the program.

Future Work

The final code will be released at the annual meeting in June 2008. It can be used to conduct coupled analysis as well as analysis of a separate type, for example, lateral only or torsional only. We will add the capabilities of undamped torsional analysis and stability analysis for coupled lateral-torsional systems.

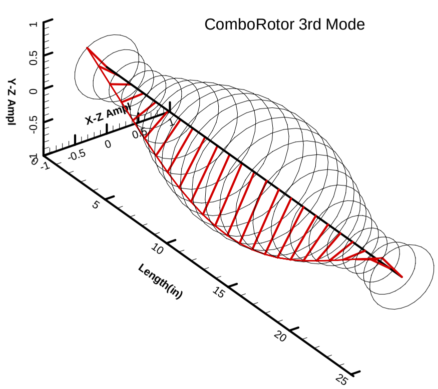


Figure 2

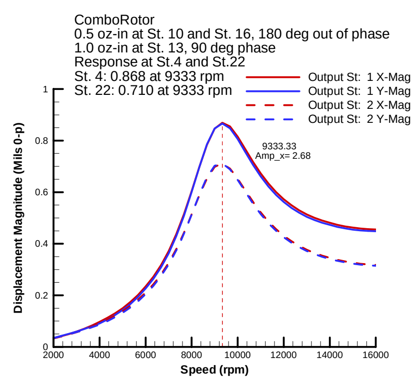


Figure 3

4.3 MATLAB ROTOR

Student: Jawad Chaudhry

Faculty: Pradip Sheth

MATLAB ROTOR has been created as a Research platform to develop, test and incorporate new tools for rotor dynamics within ROMAC research as dictated by the industrial members of ROMAC. The code is also released to ROMAC members who may be interested in utilizing the MATLAB environment for rotor bearing dynamics. New developments are prototyped, tested and validated by this code before migrating to the Production Codes. The present version of MATLAB ROTOR code combines most of the capabilities of current production ROMAC codes such as CRTSP2, ROTSTB, RESP2V5, TWIST2, and FORSTAB. Since the MATLAB ROTOR code is developed entirely within MATLAB, the numerical methods and visualization tools embedded in MATLAB functions are utilized. MATLAB ROTOR code is written in an open source format such that the interested ROMAC members can utilize the current version and do their own customization, modifications, and additions that may be specific to their rotordynamic problems and requirements. Continuous feedback from ROMAC users on the use of MATLAB ROTOR is used to further advance the codes' capabilities. The first version of MATLAB ROTOR will be relased at the Annual Meeting in June 2008. The first version of MATLAB ROTOR will include all the capabilities listed below and will be compatible with the Comborotor version such that the same models generated by the GUI can be used in both codes. Additional exclusive features that are only available in MATLAB ROTOR (i.e. mode tracking) are included in the internal MATLAB ROTOR GUI.

Current Capabilities of MATLAB Rotor

- Full 12 DOF beam element with axial, torsion and bending capabilities; Includes gyroscopic, rotary and shear deformation effects; each node thus has 6 DOF.
- Reduced bearing stiffness and damping coefficients can be added-e.g. , from THPAD or MAXBRG
- Steady State Synchronous Response
- Campbell Diagrams
- Seals and Squeeze Film Damper Coefficients
- Allows Coupled or Reduced Uncoupled analysis
- 2D/3D elliptical whirl mode shapes and animations
- Multiple-level Rotors
- Mode Tracking
- Component Mode Synthesis method to handle large systems
- Generalized 6x6 matrix constraint to any degree of freedom including bearing support locations

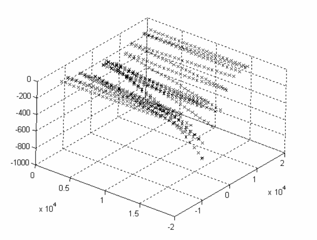


Figure 1: 3D Visualization of the Campbell Diagram

Description of Mode Tracking Method in MATLAB ROTOR

The Tracking of modes as a function of machine speed is an issue in Rotordynamics due to a variety of factors, including (i) Gyroscopics, (ii) interaction with structural/foundation dynamics, and (iii) the effects of speed dependent bearings and annular seals. The need for more precise tracking of modes is further evidenced in the recent machinery where rotor bearing systems have strong interactions with their support structures, and as additional dynamic elements such as gears and multiple shafts are modeled. The MATLAB ROTOR algorithm for mode tracking considers both the real and imaginary parts of the eigenvalues by focusing on the relative magnitude and trend of those values across the speed range. The traditional Campbell diagram is thus converted to a 3-Dimensional diagram by adding the real part of the eigenvalues for each speed that was calculated for the system. The 3-D Campbell plot of the eigenvalues is then generated. In the following figure, the imaginary part is plotted versus the real part with the shaft speed as the third dimension. This helps in distinguishing the modes that have the same damped natural frequency at a certain speed but still have different real parts.

By applying the method, The Campbell diagram can then be reconstructed with each mode now tracked. The following Figure shows the result of the algorithm applied to this example problem.

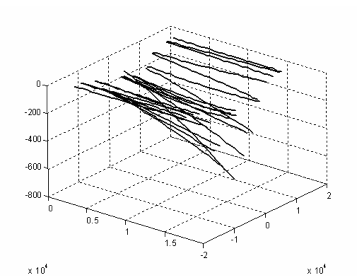


Figure 2: Tracked modes for the 3-D Campbell

From this figure, it is seen that each mode is now considered as one set of data set in the MATLAB ROTOR code. The algorithm correctly tracks each mode without any crossings even at the starting speeds. With the modes now tracked, the original 2-D Campbell diagram can then be generated with each mode now tracked and there is no misinterpretation of the modes.

Future Work:

MATLAB ROTOR will Continue to be used as a research platform and new methods will be tested and implemented in the code. Preliminary work has been started on the Optimization of Rotor/Foundation support structures.

4.4 Rotordynamics Integration with NASTRAN

Students: Andy Edwards, Jawad Chaudry

Faculty: Pradip Sheth, Paul Allaire, Costin Untaroiu

Research Professional: Bob Rockwell

This is a collaborative effort with MSC Software, Boeing, GE Avaiation and other companies to use NASTRAN and other existing codes combined with the ROMAC codes to analyze very complex rotors such as gas turbines. We are currently working on the non-proprietar gas engine model originally developed by Pratt & Whiteny/Boeing for research use. The squeeze film damper and rolling element bearing codes will be integrated into the NASTRAN framework. Integration of ROMAC codes in the NASTRAN framework has begun with THPAD, MAXBRG, and the upcoming squeeze film damper codes.

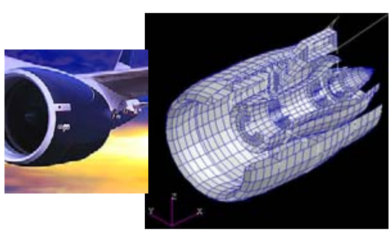
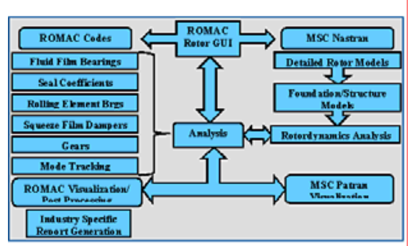


Figure 1: Prototype Engine Rotor Dynamics Model

NASTRAN/ROMAC Integration



4.5 Optimum Balancing of Rotors with Uncertainity

Student: Bin Huang

Faculty: Zongli Lin, Paul Allaire

Some additional work with Bin Huang, who graduated with his Ph. D., continues on optimum balancing of rotors with uncertainty. The primary purpose is the application of this method to industrial rotors. This work is carried on jointly with Boeing**.**

4.6 Rotating Machinery Stability Test Rig

Experimenter: C. Hunter Cloud

Project Start Date: January 1999

Primary Funding: ExxonMobil, ROMAC

Project Objective:

The stability of turbomachinery is a major concern. This project is focused on avoiding these problems in the future by conducting research in the following two areas:

A. Determining test techniques which are suitable for accurately measuring the stability of a rotor/bearing system.

B. Examining how tilting pad bearing characteristics and common phenomena such as unbalance influence the actual stability levels and thresholds versus modeling predictions.

To investigate these issues, a test rig is being constructed which will simulate the dynamic behavior of many types of turbomachinery such as pumps, compressors, and steam turbines. Magnetic actuators will supply excitations in the form of destabilizing cross-coupled stiffness and non-synchronous forcing (sine sweep, impulse, etc). Several bearing designs will be tested with the base design being

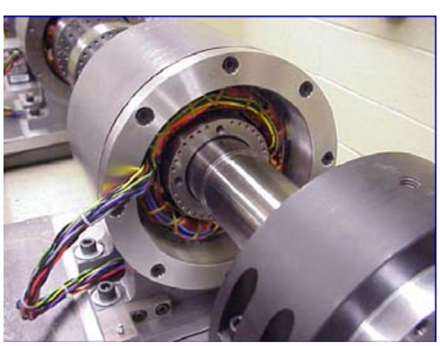


Figure 1: Rotating Machinery Stability Test Rig

a 5 pad, load between pad bearing with L/D = 0.75, 0.3 preload, center offset rocker back pivots.

Progress in the Past Year:

Work on the rotating machinery stability test rig continues. It is expected that some new bearing results related to different tilting pad fluid film bearings will be carried out. One objective is to confirm that a full tilting pad coefficient matrix including pad degrees of freedom is necessary to accurately predict the rotor-dynamic stability of industrial machines.

5. ROTOR/BEARING OPTIMIZATION

5.1 Optimization of Structures and Rotor Bearing Foundations

Student: Jawad Chaudhry

Faculty: Pradip Sheth, Paul Allaire, Randy Cogill (Systems and Information Engineering)

Research Professional: Costin D. Untaroiu

The continuing and ever increasing demands of lighter weight, higher speeds, and greater reliability of rotating machinery and their structural systems demand a rigorous optimization process for their design. The basic analysis tools of rotor-bearing-seal dynamics, and modeling of structural systems are at a mature stage and the opportunity to further apply optimization strategies for the design of these systems is now practical and realistic. This research has been stimulated from discussions with the Boeing Company (a ROMAC member). This research addresses the applications of numerical optimization algorithms for the optimization of static and dynamic performance of structures to meet specific criteria such as vibration and stress levels, and transient dynamics under design constraints. Topology, shape, and size characteristics of structures are addressed in the optimization process. The use of a variety of search techniques in conjunction with finite element models is being explored. Further, to make these tools practical, it is important to establish algorithms which can quickly evaluate the structural and the rotor dynamic systems, and therefore we are evaluating model reduction schemes for applicability to rotating machinery and structural systems.

We have initiated the development of an inverse eigenvalue solution process for a rotor bearing dynamic system mounted on a simple foundation to demonstrate the utility of this approach. In this process, the question to be answered is: given a rotor dynamic system with its dynamic matrices, can we design a mounting foundation structure such that the entire system has desired eigenvalues, and how can we physically realize such a structure assuming we cannot change the rotor bearing system? We are utilizing an inverse component mode synthesis approach for this problem.

As this research further develops, we anticipate applications to the entire rotor dynamic train including complex foundation supports, integrating optimization with rotor dynamics and finite elements.

6.1 Geared Systems

Students: Blake Stringer, Amir Younan, Jawad Chaudhry

Faculty: Pradip Sheth, Paul Allaire

We have developed a full twelve degree of freedom stiffness and damping model for geared systems. This is in cooperation with Major (US Army) Blake Stringer, as part of the Army's Uniformed Army Scientist and Engineer Program, and the US Army's Vehicle Technology Support Directorate at NASA Glenn to further rotorcraft transmission technology. The power transmission system in helicopters is one of the primary sources of noise, vibration and excess weight in helicopters. The objective is to develop a condition based approach to maintenance procedures.

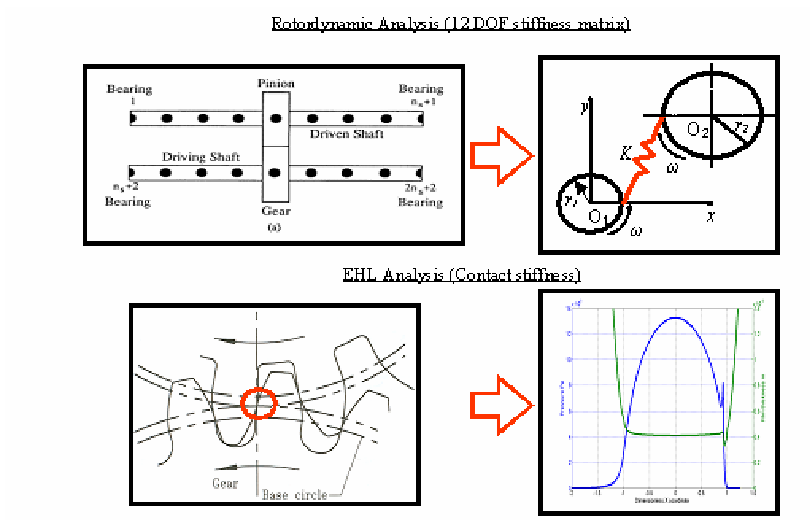


Figure 1

Such geared systems are also often found in industrial geared systems. The analysis of these geared systems requires a rotordynamic analysis where the lateral/torsional/axial vibration problem is solved. The primary generalized rotordynamics code for this is described above in Section 2.1. The combined vibration analysis captures the coupled vibration modes generated from the gear forces. The gear forces in spur gear generate radial and torsional forces, however, in the helical gear, axial forces are added to the system of forces and the system becomes coupled (see figure 2).

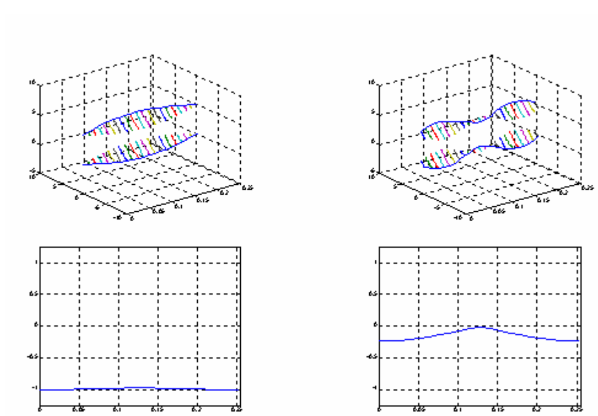


Figure 2: First (left column) and Second (right column) coupled modes in a rotordynamic system for two shafts coupled together with gear pair. The top figure represents the radial degrees of freedom and the bottom represents the torsional degrees of freedom

6. GEARS

The twelve degrees of freedom stiffness matrix is the relation between the six degrees of freedom of the pinion and gear. This generalized stiffness matrix is able to describe any gear mesh orientation which is required in a rotorcraft transmission. The matrix is developed using the direct method to capture the effect of each force on the different degrees of freedom. The matrix is function of the pressure angle and the helix angle of the gears in mesh. The stiffness matrix is also function of the gear mesh stiffness between the two teeth in contact. The mesh stiffness can be expressed using three descriptions: Time-varying stiffness, spatially-varying stiffness, time-averaged stiffness. The later is the most common applied method and it is adopted in our code.

The eigenvalue analysis for the geared system depends on the magnitude of the mesh stiffness. As seen in the figure, the natural frequency increased from (1000Hz) to (5000Hz) for two order of magnitude change in the mesh stiffness. In the literature, this stiffness value is not accurately determined and usually chosen to be between $1\text{x}10^8$ and $3\text{x}10^8$ N/m.

The mesh stiffness value is calculated from two major parts. The first is bending stiffness of the gear teeth which must be included in the analysis through a finite element codes such as NASTRAN or ANSYS. The second major part is the EHL contact stiffness which is calculated using the EHL method of analysis, described above in Section 1.2. The method was originally developed for rolling element bearings which can use the same software code. In the EHL analysis, the film thickness is constructed from the teeth profile and the elastic deflection. The analysis is successfully carried by solving the Reynolds equation and the deflection equation simultaneously. The EHL stiffness is calculated using perturbation method of the steady state pressure and film thickness profile. As shown in figure, the mesh stiffness is a load dependent parameter and depends on the tooth profile. Currently, ROMAC has the ability to conduct lateral/torsional/axial vibration problem for the geared system using EHL stiffness properties.

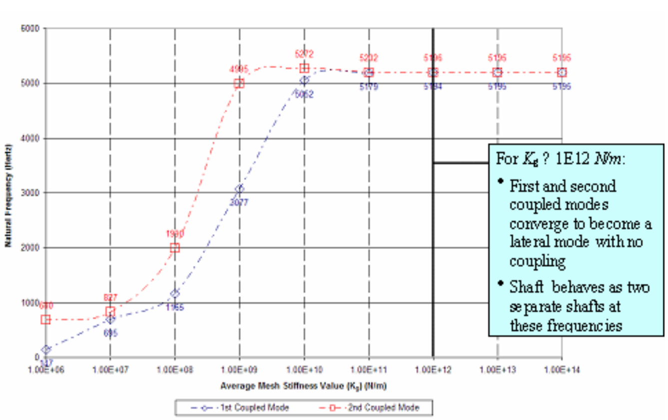


Figure 3: Effect of the contact stiffness coefficient value on the Rotor natural frequencies

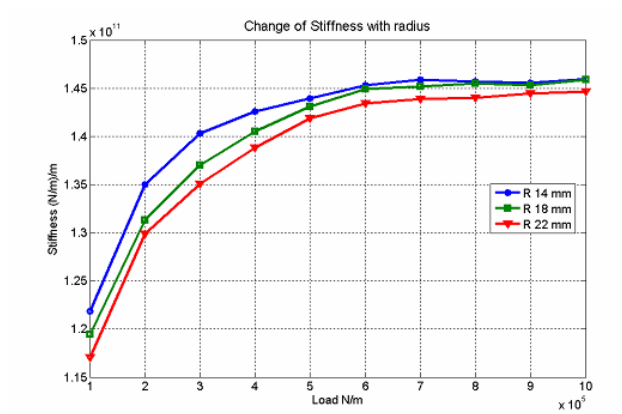


Figure 4: The contact stiffness variation with the load as calculated from the EHL analysis

7.1 Magnetic Bearing Controls/Flexible Rotor Rig

Students: Simon Mushi, Stephen Evans, Amir Younan

Faculty: Paul Allaire, Zongli Lin

Research Professional: Wei Jiang

A magnetic bearing flexible rotor test rig is currently under construction. The 1 meter long shaft, approximately 2.5 inches in diameter, has two support magnetic bearings, one at each end. It will operate at speeds of approximately 11,000 rpm, well above several modes. The rig also has two interior magnetic bearings that will be used to impose cross-coupled stiffness effects to simulate interior seal destabilizing effects. Auxiliary bearings will be placed at both ends. The motor drives the test rig through a speed increasing belt drive through a flexible coupling.

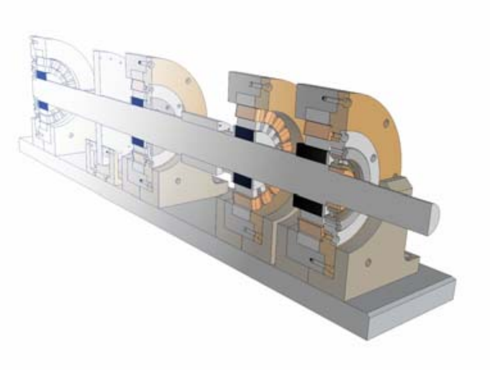


Figure 1: Flexible Rotor Rig

There are several overall purposes of this test rig. First, this rig can be used to design and test various new control algorithms for industrial machines such as compressors. Second, the rig can be employed to model existing or new compressor designs as a design audit for magnetic bearing performance. Third, the test rig can be employed for rotor drop tests on various auxiliary bearing designs. Fourth, this rig can be employed as a learning tool for new company engineers to better understand magnetic bearing supported flexible rotors. With the advent of sub-sea oil and gas production, expected to employ many magnetic bearing supported rotors, and many other uses of magnetic bearing supported rotors, significant research needs to be performed.

Controls associated with magnetic bearing supported rotors are usually developed by magnetic bearing vendors who may or may not have a full understanding of the rotordynamics of complex flexible compressors including seal destabilizing effects. This test rig will contribute to the joint understanding of the effects of different control algorithms between OEMs, End Users, and Magnetic Bearing Vendors. For example, industrial control systems normally are linear in spite of the fact that magnetic bearings are nonlinear devices due to the magnetic saturation properties of the magnetic material. Nonlinear controllers, developed at ROMAC, have advantages when the loads on the bearings become large and the magnetic bearings approach the saturation level. This test rig provides the opportunity to test this kind of controller, and others, in a cost effective way that can closely simulate the industrial environment.

For many operators of magnetic bearings, the highest area of concern is the auxiliary bearings. Questions include: how many drops can a set of bearings take, what is the best auxiliary bearing to use for a given machine, and can the auxiliary bearings take some of the load when the magnetic bearings cannot take transient loads. This test rig provides the ability to test bearings under rotor drop conditions in a situation with many common features to an industrial machine, but at a much lower cost, and without risking damage to a high cost, critical rotor. The objective is to calibrate rotor drops in the test rig with industrial auxiliary bearings under various magnetic bearing and seal force conditions, model the rotor drop events with a rotordynamics computer code until they match, and then scale the rotordynamics analysis up to the full rotor size.

Test Rig Details:

Drive System

The updated drive system consists of a shorter shaft attached using a belt drive to a 5-hp Toshiba AC motor. This shaft is suspended on roller bearings and attached to our main shaft using a flexible coupling. This setup will provide isolation of the belt disturbances and improve the character of the rotor drop experiment. The combined use of the a variable frequency drive (VFD) and the belt system allow us to achieve shaft speeds in the range of 14,000 rpm.

Magnetic Suspension

Two 16-pole heteropolar AMBs provide radial shaft support. The bearings have a RMS load capacity of 110 lbf (490N) and peak load capacity 180lbf (800N). Eight Copley 413 series power amplifiers have been chosen to drive the support AMBs using a quadrant control scheme. Eddy current sensors are initially being used to provide rotor displacement feedback at each control axis. Sensor signal conditioning, amplifier status and control boards have been fabricated and are presently being tested.

Control System

A digital controller implementation based on the modular M6713 digital signal processing (DSP) platform from Innovative Integration was selected for its speed, expandability and developer support. The board contains a floating point Texas Instruments DSP chip with a host of features ideal for industrial high-speed control. Our addition of a servo card with up to 100kHz sample rate and 16-bit resolution will allow us to work with 16 actuators and 16 differential sensors. Additional data-capture functionality will be available for the rotor drop experiment. AMB controller algorithms can be directly translated from Matlab m-code or Simulink models and implemented on the M6713. This feature is useful and will allow a wider audience of engineers see the link between control design and implementation.

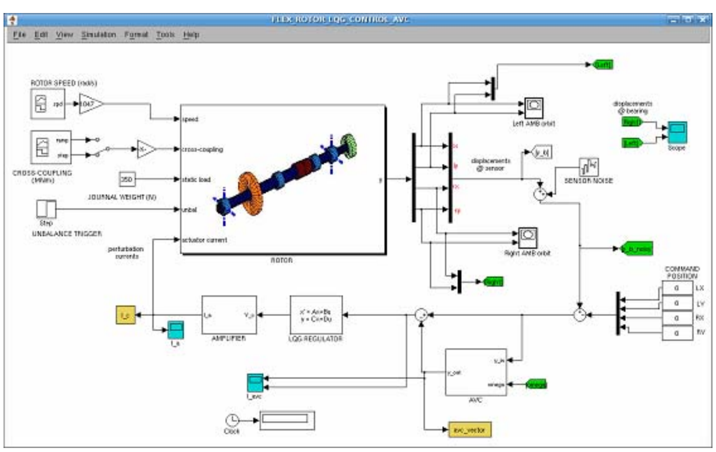


Figure 2: Simulink model of rotor, magnetic bearings, power amplifiers and LQG-based controller with adaptive control of rotor vibration (AVC) as a result of unbalance

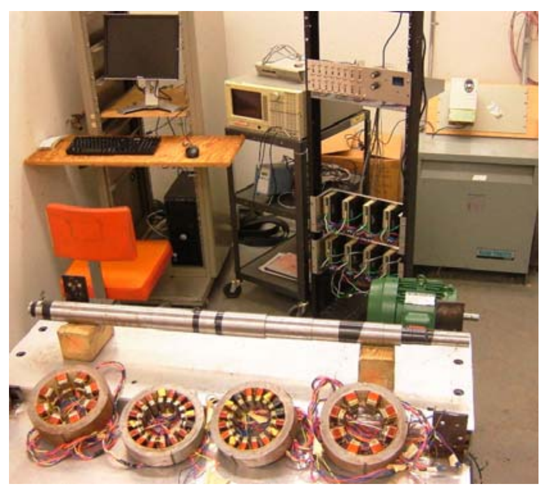


Figure 3: Photograph of test rotor, AC motor, magnetic bearings and control rack

7.2 Rotor Drop Analysis on Auxiliary Bearings

Students: Amir Younan, Tim Dimond

Faculty: Paul Allaire

The purpose of this research project is to carry out a detailed analysis of rotor drops on auxiliary bearings for magnetically supported rotors. The overall desired effect is to determine the vibration of the rotor patterns during the rotor drops and the forces associated with those actions. It is desired to evaluate the performance of various types of auxiliary bearings and the forces resulting under various possible operating conditions in presence of such effects as magnetic bearing forces, seal forces and unbalance forces.

At this point, a nonlinear transient analysis of a three mass rotor has been developed in the form of a MATLAB code. The rotor drop of the fluid film bearing test rig was studied. The three masses were selected to capture the possible asymmetric motion of the two ends of the shaft. In this particular case, it was assumed that the fluid film bearing continued to have the static load as well as stiffness and damping properties. The results indicated the rotor vibrations were not huge but that the auxiliary bearings would experience a force in the range of 18,000 lbf.

This three mass rotor code and more general multi-mass rotor transient analysis will be used to study various rotor drops for magnetic bearing supported rotors. The rotor drop analysis will be correlated with measured results in the magnetic bearing/flexible rotor test rig discussed above.

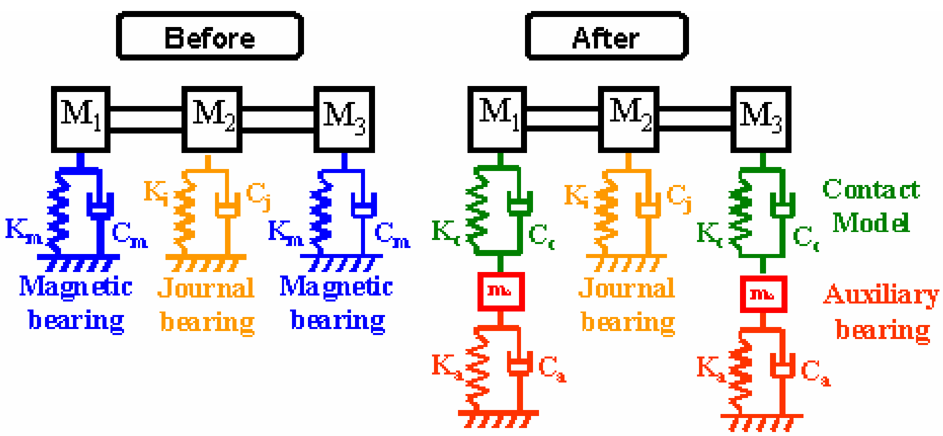


Figure 1: Three mass model before and after the drop—In this example, the rotor drop for the fluid film bearing test rig is evaluated. Before the drop, the rotor is supported on the magnetic bearing and the Journal tested bearing. After the drop, the rotor is supported by the auxiliary bearing with the help of the journal bearing. The Hertzian contact model is used to describe the contact between the rotor and the auxiliary bearing.

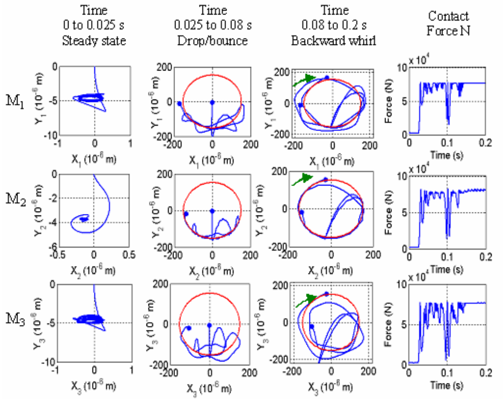


Figure 2: Transient analysis of the rotor drop—The response of the rotor is calculated inside the clearance of the auxiliary bearing and the journal bearing. The steady state analysis (first column) represents the initial conditions for the transient analysis (second column). A backward whirl is observed in the transient response (third column). The contact force on the auxiliary bearing (Fourth column) is calculated during the rotor drop

7.3 Self-Sensing of Magnetic Bearings

Research Professional: Wei Jiang

Faculty: Paul Allaire

Magnetic bearings require rotor position or other feedback signals for control purposes. Usually the position signals are provided by either eddy current sensors or inductive sensors. In ROMAC, we have developed a highly innovative self sensing method using a pair of coils supplied with high frequency voltage signal. This is combined with a magnetic circuit model and a detection circuit to sense the position of the rotor. Basically, this method uses the magnetic bearing circuit itself as the sensor.

This approach has been tested in the ROMAC labs over the past year. A small magnetic bearing and rotor have demonstrated that this method works. Much of the work over the past year has concentrated on noise reduction in the position signal. A full scale self-sensing operation will be tested in the magnetic bearing/flexible rotor test rig described in Section 5.1 above.

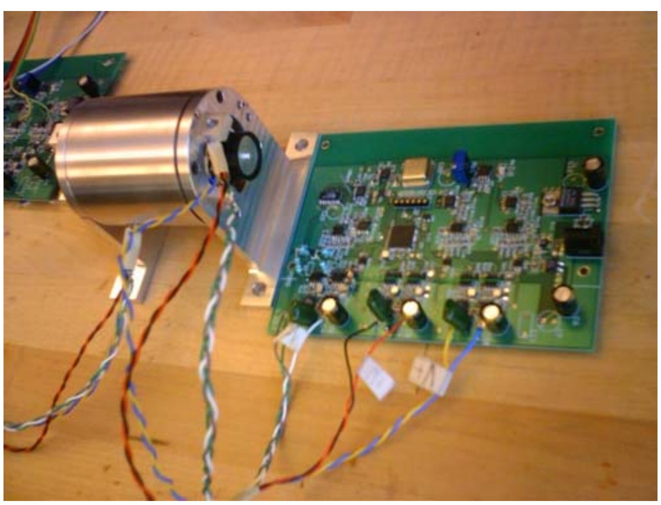


Figure 1

A fully suspended small self-sensing test rig shown in the following picture has been built. Experiments are in progress to compare the performance of self-sensing with the eddy current sensor. Self sensing is a very useful approach for magnetic bearing supported rotors. If the separate physical sensor can eliminated, rotors can be made shorter and thus less flexible. Also, the ROMAC self sensing approach using a small circuit board will reduce the cost of magnetic bearing systems. Finally, the use of self sensing allows for the provision of self sensing as a backup sensing unit that could potentially make magnetic bearing systems be much more reliable.

7.4 Preliminary Magnetic Bearing E-Core Design for High Load Aerospace Application

Students: Tim Dimond, Thomas Meriweather

Faculty: Paul Allaire, Zongli Lin

Research Professional: Bob Rockwell

Start Date: October 2007

Primary Funding: IPS, ROMAC

In order to improve generator efficiency, companies such as IPS wants to replace roller bearings with Active Magnetic Bearings. However, the magnetic bearings will need to be properly designed to replace the current roller bearings in a pulse power generator for the F35 Joint Strike Fighter.

This project is currently in the early stages of design. Preliminary calculations for the bearing sizing are fairly simple. We began by identifying the maximum force the magnetic bearings will need to provide in order to prevent any damage. Assuming a symmetric e-core the total magnetic force is given by:

$$F = \frac{A_p B^2}{\mu_o}$$

where A_p is the cross-sectional area of the central pole, B is the flux (in our case the saturation flux of Hyperco 50a), and μ_o is the permeability of a vacuum. If each auxiliary pole has a cross-sectional area $A_p/2$ the total pole face area is $2A_p$. Our design will utilize four e-cores with each central pole oriented at 45^o to the vertical axis. As an approximation, the force that each e-core will need to produce to keep the shaft stable under maximum loading can therefore be reduced by a factor of $\sqrt{2}$.

In the early stages of design, the width of the e-core can be approximated by the arc length of the shaft; keeping in mind the lamination thickness on the shaft should be no thinner than the auxiliary pole width. This approximation of thickness also works well for the back iron.

Finally, one must also consider the area of windings necessary to generate the necessary flux. This can be determined by dividing the total current NI (where N is the number of turns, and I is the current per turn) by the current density. The total current can be determined by:

$$NI = \frac{2Bg}{\mu_o}$$

Approximately 60% of the cross sectional area of copper will go on the main pole and 20% of this area on each auxiliary pole. The packing factor must also be taken into account when determining the actual cross sectional area of the windings.

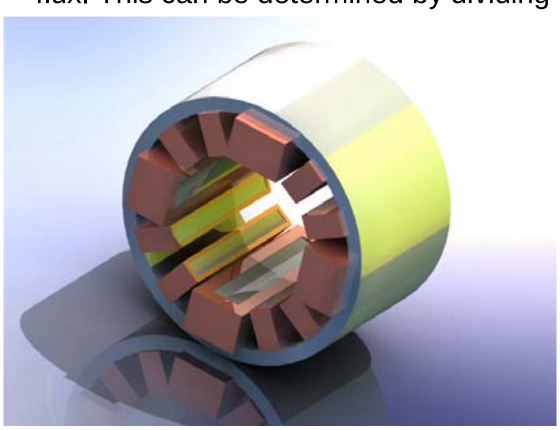


Figure 1

8.1 RotorLab+ Development

Faculty: Bob Ribando

Research Professional: Sean Travis

Work on the new RotorLab+ graphical user interface has been progressing well. The new interface is intended to replace the existing RotorGUI and RotorLab tools. Shortly after the 2007 summer annual meeting (and with extra financial support from ExxonMobil) we hired Sean Travis, an experienced programmer analyst to work full time on the project. Sean brings to us formal education in software engineering (UVa B.S. in Computer Science) and about six years of hands-on experience in large scale project development.

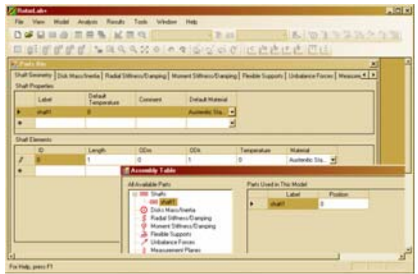
Our new RotorLab+ is being created in the Visual Studio.Net environment and provides a solid foundation upon which later deliverables will be based. Currently it replicates much of the same interface as the current RotorLab and illustrates many planned future improvements.

Features that have already been incorporated include:

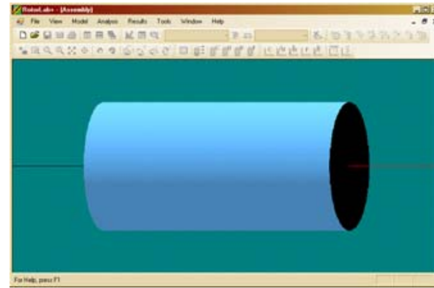
- Opening and saving of assemblies (projects).
- Parts Bin.
- The Bearing/Seal analysis interface is now integrated into RotorLab+ and no longer uses the separate SED utility.
- Assembly Table.
- Assembly View (3D graphic view) is partially implemented. Shading has been improved. We are using DirectX graphics primitives, which once successfully implemented, provide a very powerful and versatile rendering environment.
- Analysis Setup window (except for the meshing options).
- All three ROMAC analysis programs (FORSTAB, THPAD, and CRTSP2) used by RotorLab are accessible. Analysis output and error reporting has been improved.
- Viewing of analysis programs' input and output files.
- Window management.
- Several interface oddities, such as multiples of the same menu item, have been removed.

As work continues during the winter and spring, we will be soliciting additional input from selected ROMAC companies. We will have an early version ready to distribute more widely at next summer's meeting.

GUI Figure 1



GUI Figure 2



GUI Figure 3

9. HEARTPUMP

9.1 LifeFlow Ventricular Assist Device

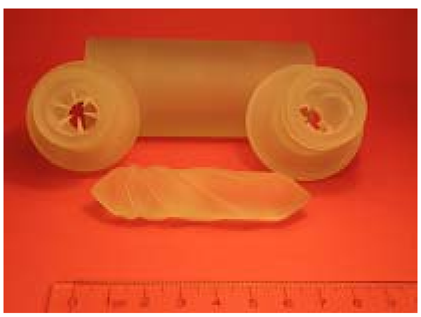
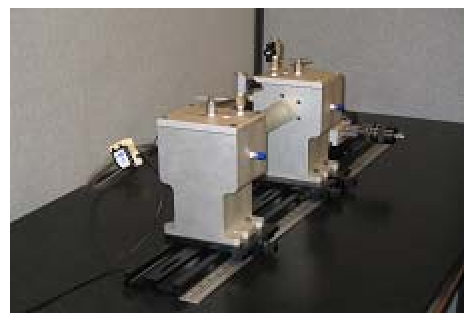


Figure 1: Plastic Pump Prototype

Figure 2: Fluid Test Rig



Students: Stephen Evans, Simon Mushi, Isaac Cecil, Patrick Graydon, Kinga Dobolyi, Devendra Rai

Faculty: Paul Allaire, Houston Wood, Zongli Lin, John Knight

Research Professionals: Alex Untaroiu, Wei Jiang, Kim Wasson

Collaborators: Curt Tribble, Kent Harmon, Charles Klodell, Charles Hobson (University of Florida-College of Medicine); Shashank Desai, Nelson Burton, Tonya Elliott, Lori Edwards (Inova Fairfax Hospital-Inova Heart and Vascular Institute)

Real progress has been made on this project lately. New collaborators from the Computer Science Department at UVA and surgeons from the University of Florida and Inova Heart and Vascular Institute joined our multidisciplinary research team.

As a direct result of including a self sensing magnetic bearing design, the flow path of the pump has been redesigned. The new design configuration of the pump is more compact and the overall length is reduced by 25% as compared to the previous design. Since this device is intended to be an implantable ventricular assist device, these two features create potential for a better anatomical fit while reducing the overall invasiveness of the implant.

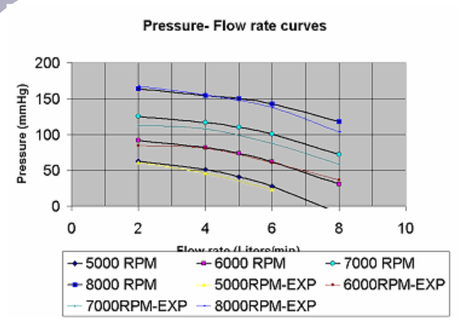


Figure 3

Figure 4

The LifeFlow's current design has the capability of providing flow rates from 2 to 8 lpm over physiologic pressures, for rotational speeds of the impeller varying from 5,000 to 9,000 rpm. A plastic pump prototype of the LifeFlow has been recently constructed (Fig. 1) in order to assess the flow performance of the pump and to validate the CFD prediction of main flow parameters. The plastic pump prototype does not include magnetic system components, thus, the impeller is shaft-driven. The fluid test rig illustrated in Fig.2 was constructed to facilitate flow measurements on the plastic prototype. Recent flow measurements taken by grad student Isaac Cecil demonstrated a very good agreement between the experimental data recorded for pressures and flow rates and the CFD estimations. Figure 3 is a plot comparing the pump-flow curves of the computational model to the pump-flow curves of the experimental test rig. The data taken from the experimental test rig corresponds closely to that of the computational results. The largest percent difference in the CFD and experimental data is approximately 10.4 percent and that is found on the plot where the pump speed is rotating at 7000 RPM.

In parallel with the flow path re-design and testing, a magnetic suspension test rig was constructed in order to test and validate current suspension system design (Fig.4,5). The magnetic suspension testing is currently underway. As soon as the experimental testing of both main components of the pump is concluded, the two will be combined and a magnetically levitated pump prototype will be build for in vivo testing.

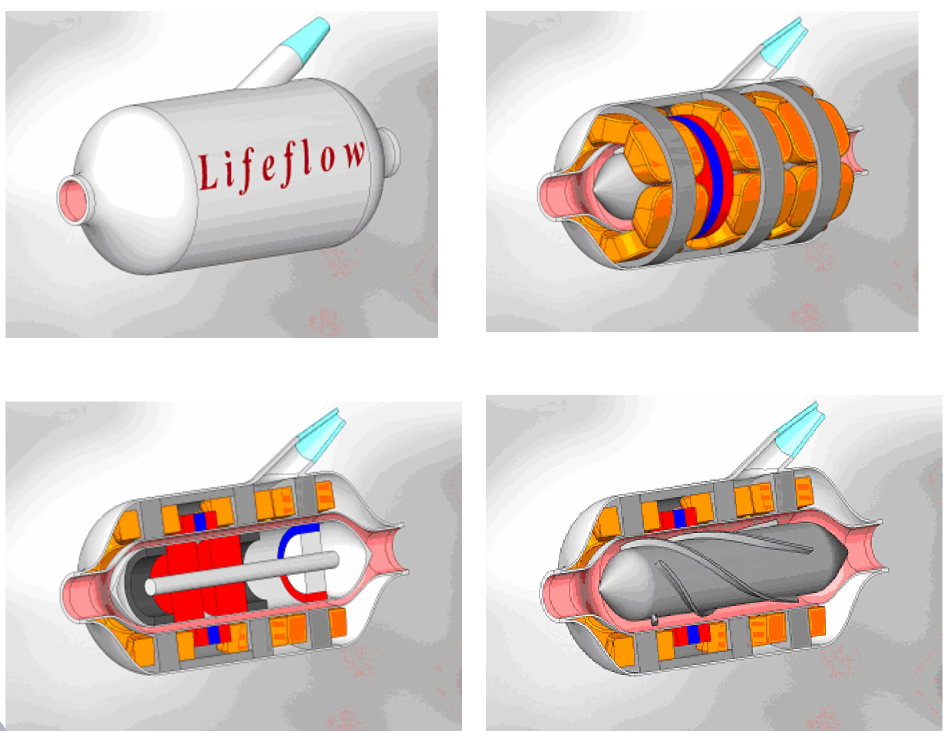


Figure 5: LifeFlow System—assembly and cut-away view of components

ROMAC ANNUAL CONFERENCE

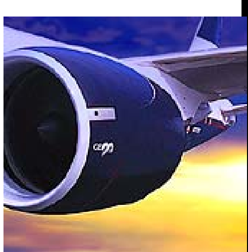
Charlottesville, VA

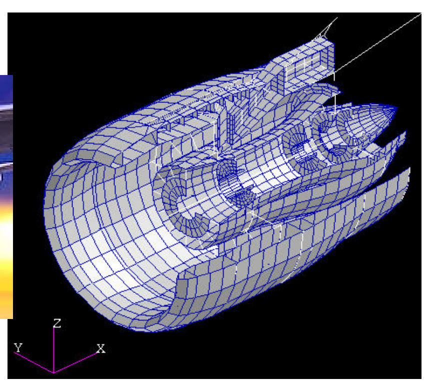
June 17-20, 2008

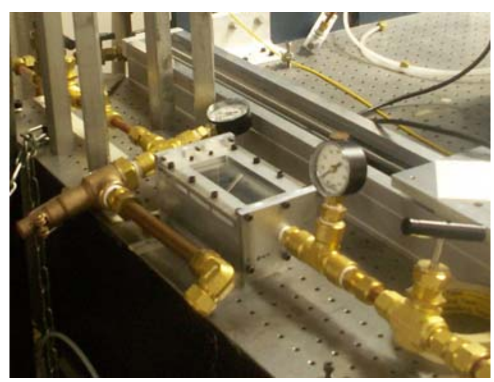
The 2008 ROMAC Annual Conference will be held in Charlottesville, Virginia at the Omni Hotel. They are giving us a room rate of $149 single/double per night. You can make your reservations by calling 1-800-843-6664. When calling the Omni, please let them know that you are part of the ROMAC Conference. Something new: You can now register with the Omni via the web. Visit their website to make your reservations directly online. We will begin the conference on June 17 with a welcome reception in the late afternoon. The actual meeting will begin on June 18. For more information or to download the registration form, visit our website at www.virginia.edu/romac/annual_meetings.

2008 Rotordynamics/Magnetic Bearings Short Course

The 2008 Rotordynamics/Magnetic Bearings Short Course will be held on August 4-8, 2008 at the Mechanical Engineering Building in Charlottesville, VA. This is a course in rotordynamics, bearing dynamics and applied dynamics, as well has topics relating to magnetic bearings. We have gotten a large number of high quality industrial speakers, in addition to the ROMAC faculty. **A nominal fee of $500 per attendee for employees of ROMAC member companies will be charged. The fee for non ROMAC members is $1500 for the full course.** For more information, visit our website at www.virginia.edu/romac/shortcourses.







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