Rotating Machinery and Controls

2006-2007

ROMAC NEWSLETTER

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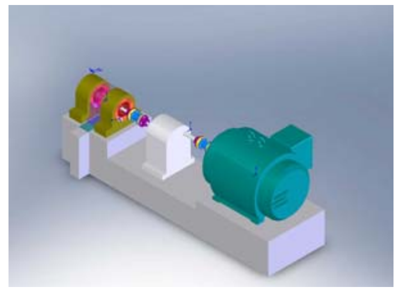
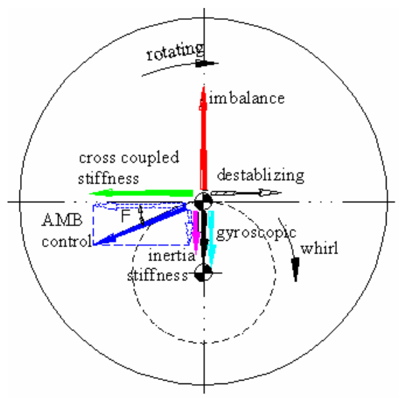
Special points of interest:

- More improvements to MAXBRG
- New Companies join RO-MAC
- 2007 Annual Meeting in Las Vegas

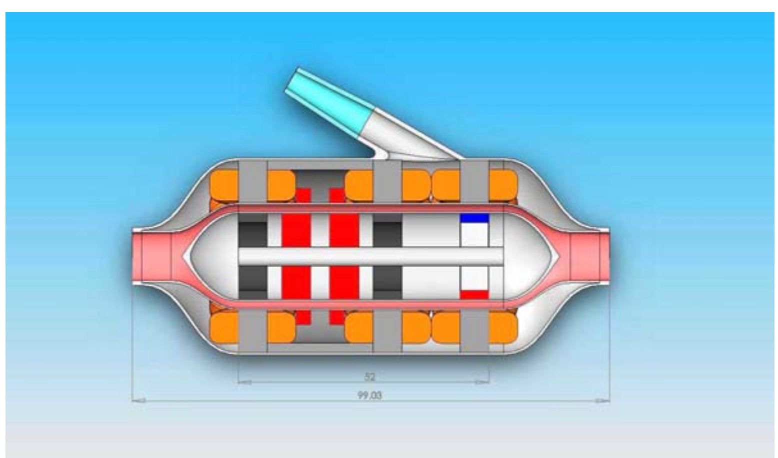
Rotor Stability Test Rig Shaft



Fluid Film Bearings Test Rig Drawing



Forces in Gyroscopic Rotor Systems



Artificial Heart Pump Design

MESSAGE FROM DIRECTOR

ROMAC Updates

This newsletter brings you a summary of the research topics in ROMAC this year and for the future. The general areas of research are 1) Fluid Film Bearings, 2) Rotor Dynamics, 3) Seals, 4) Flows, 5) Rolling Element Bearings, and 6) Magnetic Bearings. The number of research projects has increased significantly over the past few years due to the energy of the faculty, research professionals and students. Many companies attended the annual meeting: Areva, Boeing Commercial Aircraft, Cooper, EMD Curtiss-Wright, Dresser-Rand, Dow Chemical, ExxonMobil, Hamilton Sunstrand, Knowles Atomic Power, Kobe Steel, Kingsbury, Petrobras, RMT, Rolls Royce Energy Systems, Shell, Siemens Demag Delaval, Solar Turbines, TurboCare, Turbo Components Engineering, Waukeshau Bearings, General Electric, Re-Genco, ODS, Bechtel Plant Machinery, and Pratt and Whitney. However, several companies were not able to attend so this newletter will help to inform them of our ativities.

New Industrial Members

The industrial membership in ROMAC is increasing. This year we have added seven new companies: Statoil (oil and gas producer in Norway) , Bettis Atomic Power/Bechtel, MSC Software (developers of NASTRAN finite element Code), Innovative Power Solutions (producer of high performance motor and generators), Renk (fluid film bearing manufacturer in Germany), ODS and General Dynamics/Electric Boat Company. In addition, Duke Power has rejoined. Several more companies have shown strong interest in joining ROMAC and are likely to join in the next year or so.

ROMAC Faculty

The ROMAC faculty are: Paul Allaire, Harsha Celliah, Chris Goyne, Zongli Lin, James McDaniel, Walter Pilkey, Bob Ribando, and Pradip Sheth. This gives us eight ROMAC faculty members in the Mechanical and Aerospace Engineering Department and Electrical and Computer Engineering Department. Our latest addition to the ROMAC faculty is Robert Ribando who an expert on heat transfer, user friendly software development, and graphical user interface (GUI) software. Some of you may recall that he was a ROMAC faculty member some years ago. Their role in particular projects is detailed in the newsletter.

Research Professional Staff

Our professional staff consists of five highly qualified people: Hunter Cloud, Minhui He, Wei Jiang, Robert Rockwell and Alex Untaroiu. Of special note is Minhui He, who is the ROMAC Lab Engineer. All have their Ph. D.s earned in ROMAC or similar areas in recent years. Some are part time in ROMAC and others are full time. These excellent Research Professionals contribute a great deal to the success of ROMAC research projects. We also have one or two Visiting Professors or Industrial Visitors in ROMAC every year. Again, their specific roles are detailed in the specific projects in this Newsletter.

Office Manager

Karen Marshall is our Office Manager. She takes care of industrial company contacts, assigns passwords to the software, provides purchasing and other services to ROMAC faculty, research professionals and students.

ROMAC Graduate Students

There are quite a few ROMAC students either graduated or current. Over the past year, the five students who have graduated are: Hunter Cloud (Ph. D.), Bin Huang, (Ph. D.), Guoxin Li (Ph. D.), Dorsa Sanadgol (Ph. D.), and Jack Zhou (Ph. D.). Currently we have 13 graduate students: Amer Al-Dhafiri, Jawad Chaudhry, Tim Dimond, Gavin Garner, Stephen Evans, Jason Feely, Nina Mohleji, Simon Mushi, Paeblo Yoon, Marion Reid, Blake Stringer, Bo Thieu, and Amir Younan. These students are funded through ROMAC industrial funds, US Army, NASA, Saudi Aramco, Fellowships and Teaching Assistantships.

2007 Annual Fee

The annual membership fee is $19,000 for 2007. This is a small increase over last year's fee of $17,500 or about 8%.

2007 Magnetic Bearing Short Course

With the increase in the use of active magnetic bearings in the rotating machinery industry in recent years and for the future, there is a need for a short course on these devices. There will be a ROMAC Short Course on Magnetic Bearings. The lecturers will be drawn from ROMAC, magnetic bearing manufacturers, OEMs and end users. The current plan is have it in July, probably in Washington DC. The specific dates and hotel site are not yet set.

2006 Company Research Ratings

At last year's meeting, the research ratings were given by the ROMAC members. The values were:

Fluid film bearing test rig – 12 companies

Seal codes – 12 companies

MAXBRG – 11 companies

Thrust – 11 companies

Seal test rig – 8 companies

Stability test rig – 8 companies

Rolling element bearings – 6 companies

Magnetic bearings – 6 companies

Compressor surge rig – 6 companies

ROMAC Graphical User Interface /RotorLab – 4 companies

Impeller/turbine tip clearance effects – 4 or more

Compressor stage flow analysis-2 companies

We are actively working at this time on all of the topics that received at least 3 votes.

Fluid Film Bearing Test Rig

Student: Tim Dimond

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

Research Professionals: Minhui He, Wei Jiang, Robert Rockwell

Funding: ROMAC, Fluid Film Bearing Test Rig Group

The fluid film bearing test rig (FFBTR) is 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.

Fluid Film Bearing Specifications

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. Based on feedback from the 2006 ROMAC annual meeting, and due to design development, there has been a significant revision in the technical specifications for the FFBTR. However, the redesign preserves the turbulence/inertia levels and journal surface speeds originally presented at the 2005 annual meeting.

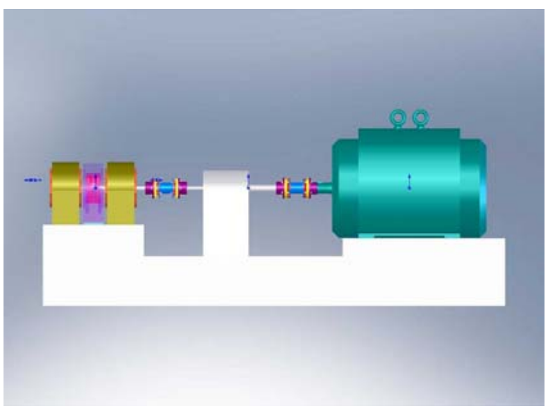


Figure 1: Side View of Fluid Film Bearing Test Rig

A summary of the fluid film bearing technical specifications for the FFBTR follows:

Nominal rotational speed:

20,400 RPM, oil lubricated bearings

10,000 RPM, water lubricated bearings

Maximum rotational speed:

22,440 RPM, oil lubricated bearing

11,000 RPM, water lubricated bearings

Total Speed Range:

9000 RPM-22440 RPM

Surface Speeds:

445 fps (nominal)

490 fps (max)

Total Surface Speed Range:

196 fps-490 fps

Fluid Film Bearing Size:

5''

Bearing Orientations:

LOP, LBP

L/D Ratios:

0.5-0.75

BEARINGS AND SEALS

Fluid Film Bearing Test Rig Cont.

Preload Ranges:

0-0.3

Pad Pivot Offset Ranges:

0.5-0.65

Lubricants:

Water, Oil - ISO VG 32 & ISO VG 46

Bearing Static Load, no dynamics:

400 psi, L/D 0.5

275 psi, L/D 0.75

Bearing Static Load, with dynamics:

200 psi, L/D 0.5

150 psi, L/D 0.75

Frequency Sweep Range:

0.5x – 1.5 x

Rotor 0-p Displacement:

0.5 mil

A parameter optimization study is underway to finalize design values for the test rig. ROMAC will be working directly with bearing manufacturers to test the tilting pad bearings of primary interest. Many of the other comments received on temperature and pressure measurements, measurement of pad deflection, and pad motion are under development.

Notes:

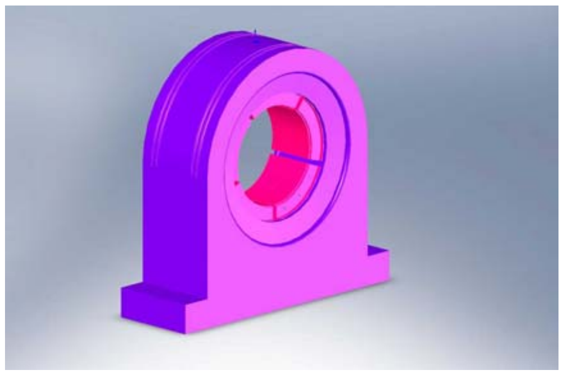


Figure 2: Magnetic Bearing For FFBTR

1. Rotational test speeds of up to 22,440 rpm possible regardless of bearing lubricant.
2. 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.
3. 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: $m = 0.3$
- Offset: $\alpha = 0.65$
- 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 coficients

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

Magnetic Bearings and Force Measurement Capability

Magnetic bearings will be used as exciters to support the shaft and perturb it with small displacement and velocity motions.

AMB Actuator Size:

6" – composed of 4 E cores

AMB Actuator capacity: 2700 lbf/bearing

AMB Manufacturer:

Innovative Power Solutions, LLC

The major effort in instrumentation has been development of a primary force measurement method. Strain

Fluid Film Bearing Test Rig

"The purpose of this prospectus is to set up a cooperative industrial group called the Fluid Film Bearing Test Group (FFBTG)"

gauges on the magnetic bearing poles and other method will be used to measure the forces to determine the load capacity, stiffness and damping. The two sensing methods under consideration are fiber-optic strain gages (FOSGs) and Hall sensors. The FOSG method for measuring magnetic bearing forces was originally developed at Texas A&M.

Both methods will be tested using a Dynamic Force Measurement Validator (DFMV). Although total force levels will not be replicated, the design intent is to produce strains in the legs of the e-core linear magnetic actuator that are comparable to predicted strains in the radial magnetic actuator predicted for the FFBTR. The difference between the DFMV and the FFBTR will then be the calibration constant. The magnetic flux levels will also be comparable to peak flux levels for the FFBTR. Static and dynamic forces will be applied to a guide piston attached to the magnetic actuator target. Dynamic loading will be applied up to 560 Hz, which corresponds to 1.5x excitation at 22,440 rpm.

One goal of this testing is to determine which force measurement method produces force data with the least uncertainty. Another goal is to identify and resolve any force measurement calibration issues that can be resolved prior to implementation in the FFBTR.

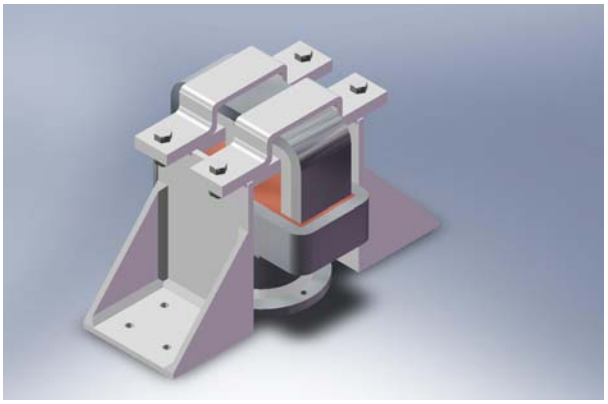


Figure 3: Dynamic Force Measurement Validator

Motor and Speed Increaser Specifications:

Motor Horsepower Rating:
350 HP

Motor Rotational Speed:
1800 RPM at 60 Hz
4400 RPM at 146.7 Hz

Motor Manufacturer:
Reuland Electric

Speed Increaser Horsepower Rating:
350 HP

Speed Increaser Gear Ratio: 1:5

Outline of fluid film Bearing Test Group (FFBTG)

The purpose of this prospectus is to set up a cooperative industrial group, called the Fluid Film Bearing Test Group (FFBTG) within ROMAC to sponsor a fluid film bearing test rig to measure static and dynamic properties of tilting pad fluid film bearings at high surface speeds and high loads, typical of modern industrial use. There has never been a fluid film bearing test rig constructed of this advanced capability. The test rig shaft will be mounted in magnetic bearings to provide the static and dynamic load forces necessary to measure these properties. This prospectus provides many design details of the test rig.

BEARINGS AND SEALS

Fluid Film Bearing Test Rig Cont.

Due to the limited funding within ROMAC from annual membership fees and the high cost of this test rig, we need to have additional funding for the rig, as discussed in annual meetings and in many other discussions with companies. The ROMAC group is asking member companies of the FFBTG to commit $10,000 each for the first year (2007), $10,000 for the second year (2008) , and $10,000 for the third year (2009) to this project. This is a commitment of $30,000 over the three year period. This commitment may be either a totally monetary contribution or up to one half in appropriate equipment. Each company that is a FFBTG member will have the right to have two bearings tested in the rig. One bearing test will have the results open for access by all ROMAC members and may have the results published in the open literature. The second bearing test may be proprietary to the company, if the company desires.

Another option is that two ROMAC member companies may partner together to join the FFBTG, at $15,000 per year, for the three year program. This commitment may be either a totally monetary contribution or up to one half in appropriate equipment. In this case, they will have testing capability of three bearings. One of these tests will be open. The other two tests will have the option of being specified as proprietary to the two companies.

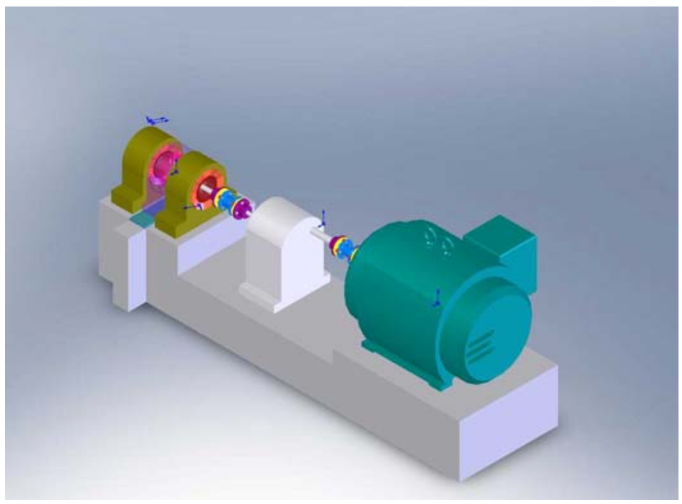


Figure 4: Perspective View of Fluid Film Bearing Test Rig

If we get a significant number of companies to commit funding by January 2007, as expected based upon conversations with a number of member companies, we plan to have the rig operational by 2008. The primary research directions for the Fluid Film Bearing Test Group will be set by the industrial members of that group and periodic special meetings, either in person or by conference call, will be held with them to keep their input in the loop. Payments are due in February of each year.

After the test rig is operational in 2008, we plan to undertake two programs of research. In the first research program, we will accommodate the industrial members of the FFBGT that wish to have two particular bearings tested. A standard bearing test protocol will be developed by consultation between the FFBGT member companies and ROMAC faculty and staff. The order of testing will be determined by the order of company commitments to joining the FFBTG group.

In the second research program, we consider the case of a ROMAC member company that is not an original member of the FFBTG yet wishes to have one or more particular bearings tested. After the test rig has been finished and the FFBTG members have their

Fluid Film Bearing Test Rig Cont.

bearings tested, the cost will be $15,000 per fluid film bearing, with a standard

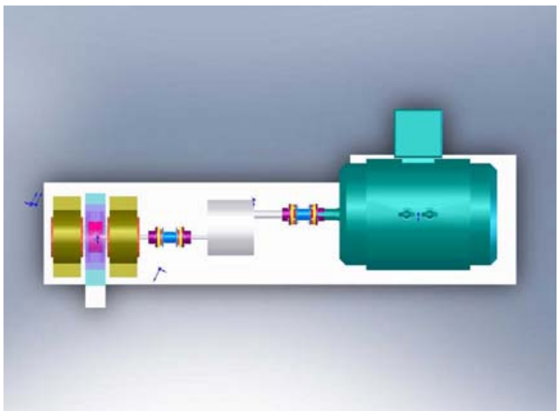


Figure 5: Side View of Fluid Film Bearing Test Rig

testing protocol, for such a company and more for additional specialized testing. These test results will also be either open or kept proprietary, as desired by the funding company.

BENEFITS OF MEMBERSHIP IN FLUD FILM BEARING TEST GROUP

The advantages of creating and the advantages to industrial members of the fluid film bearing test group are several. First, the additional funding will allow a much more rapid test rig development program than if only normal ROMAC membership funds are employed. This will allow ROMAC to meet the industrial firm's needs in a timely manner.

Second, the group's members will be directly involved in the fluid film bearing rig specifications as the rig evolves to make sure that their needs are met.

Third, each member company will receive the rights to have ROMAC test two bearings at no additional cost (or three in the case of a partnership between two companies). The order of testing will be determined by the order in which individual companies commit to join the FFBTG.

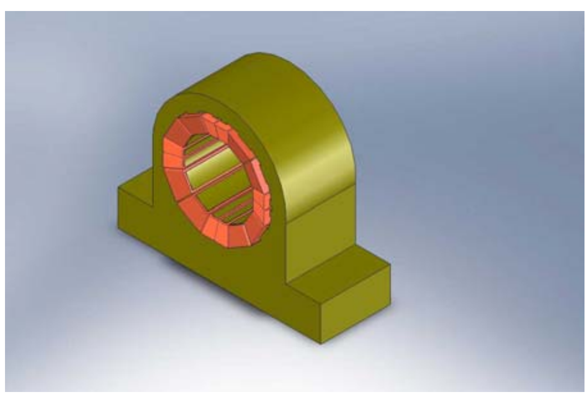


Figure 6: Magnetic Bearing for Fluid Film Bearing Test Rig

Fourth, there will be a number of teleconferences and special meetings to determine the test rig design and capabilities.

"The group's members will be directly involved in the fluid film bearing rig specifications as the rig evolves to make sure that their needs are met.."

BEARINGS AND SEALS

Rolling Element Bearing Analysis and Code

Student: Amir Younan
Faculty: Paul Allaire
Funding: ROMAC

Rolling element bearings are employed in many rotating machines such as pumps, compressors and jet engines. A number of modeling codes are available to evaluate static operating

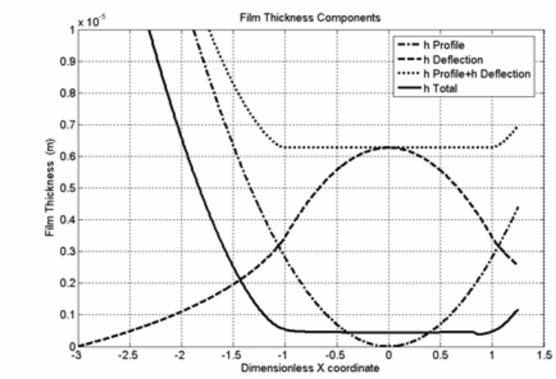
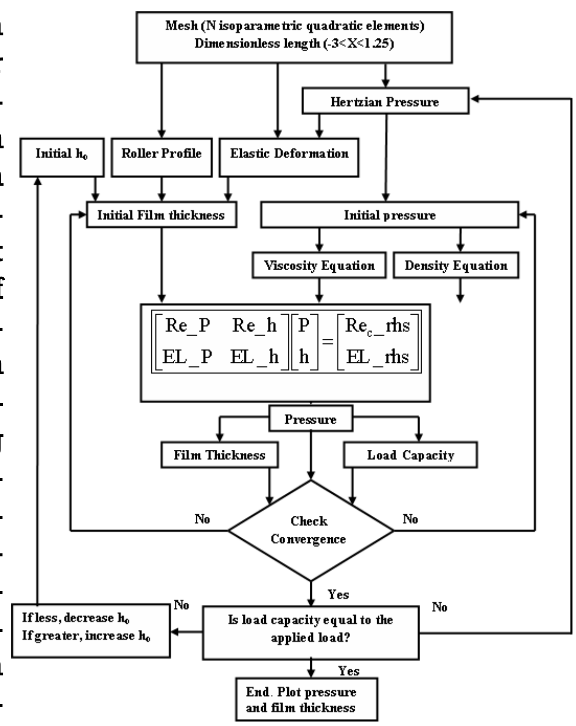


Figure 1. Film Thickness Components

properties but not very many are capable of evaluating dynamic properties. In rolling element bearings, the applied load is supported by a small contact area between the balls (or rollers) and the races that is called an elastohydrodynamic (EHD) lubrication regime. In EHL, there are two primary deformations of interest: the elastic contact surface and the lubricant film.

A new general approach to finite element analysis of rolling element bearings has been developed and verified for this code. Reynolds equation governs the fluid lubrication regime while the elasticity equation determines the pressures and deformations in the film. A mixed problem involving a second order form of Reynolds equation for large film regions is coupled to the first order form of Reynolds equation in the very thin film lubrication region. The coupling parameter is the lubricant viscosity. The viscosity-pressure relation employs Roelands power law viscosity relation and a density pressure relation. Figure 1 shows the film thickness profiles for the undeformed film thickness, the deformed film thickness and the total film thickness.

The flow chart for the rolling element bearing solution is given in Fig. 2. Figure 3 shows a plot of the pressure profiles for various loads on a single rolling contact example. Figure 4 shows a schematic of stiffness for an EHD film illustrating the Hertzian contact stiffness and the lubricant film thick-



$$\begin{bmatrix} Re\_P & Re\_h \\ EL\_P & EL\_h \end{bmatrix} \begin{bmatrix} P \\ h \end{bmatrix} = \begin{bmatrix} Re_c\_rhs \\ EL\_rhs \end{bmatrix}$$

Figure 2. Flow chart of solution method for rolling element bearings

Rolling Element Bearing Analysis Code

ness. The next steps include assembly into a full bearing system with multiple rollers, inner and outer races and other rolling element bearing components. Small perturbations on the pressures in the lubricant film and deformations of the solid component deformations will yield the bearing stiffness and damping coefficients for rolling element bearings.

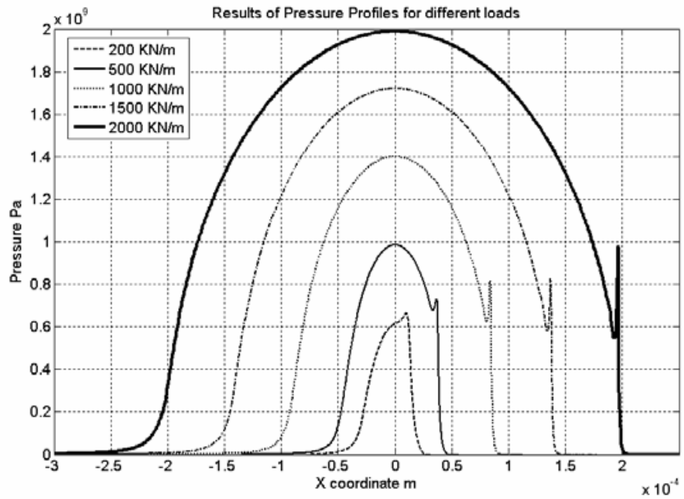


Figure 3. Pressure profiles for different loading values

"A new general approach to finite element analysis of rolling element bearings has been developed and verified for this code"

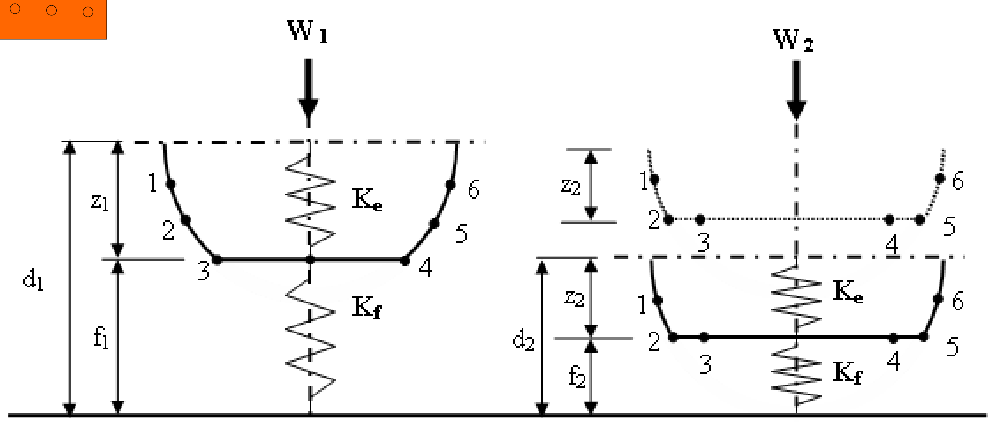


Figure 4. Diagram of Hertzian stiffness and lubricant film stiffness

BEARINGS AND SEALS

TEHD Analysis of Fluid Film Journal Bearings(Computer Code MAXBRG 4.4)

Research Professional: Minhui He
Faculty: Paul Allaire
Project Start Date: 1996
Report Number: 458

Project Overview:

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

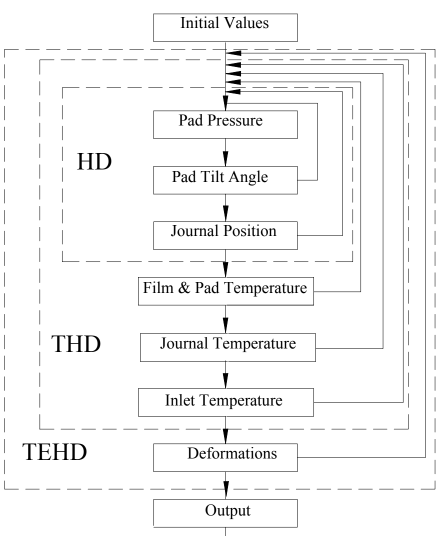


Figure 1: Flow Chart for MAXBRG Code

Many advanced models are utilized in this finite element based algorithm as shown in Figure 1. 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 as shown in Figure 2. A coupled film-pad approach is employed to achieve good numerical robustness.

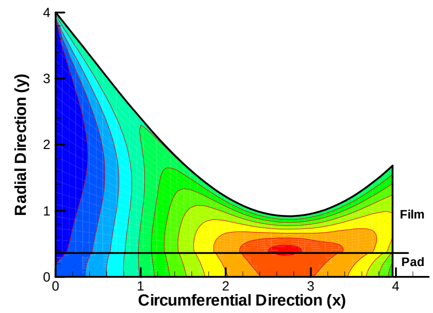


Figure 3: Pad & Film Temperature Contours

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 needs and engineering judgment.

Progress in the past year

Several upgrades were made in the past year. The most noticeable one is the improved modeling of turbulence. In cooperation with EMD, the turbulence model was fine-tuned based on detailed CFD simulations, which leads to significantly improved predictions for high Reynolds number cases. Other upgrades include:

- A trailing edge taper is added to the pad geometry. The bearing pads can have tapers of different geometries at both leading and trailing edges.
- Temperatures at specified pad locations are added to the output file. This can be used to simulate thermocouple output.
- For a specified pad and load case, the pressure, temperature and pad deformation can be graphically presented using TECPLOT.
- The convergence properties of various iteration loops are improved.

Minor improvement of the output format.

Future work

MAXBRG will continue to be maintained and upgraded in the future. As in the past, future upgrades will largely be based on feedback from ROMAC members. Please tell us your questions, suggestions and comments. Your input is very important and highly appreciated.

Thrust Bearing Code Improvements (Computer Code THRUST)

Research Professional: Minhui He, Ted Brockett

Faculty: Paul Allaire

Project Start Date: May 2005

THRUST is a three-dimensional finite element code that performs TEHD analysis for thrust bearings. It was originally developed by Dr. Ted Brockett and is considered one of the most advanced and powerful codes for thrust bearing analysis. Unfortunately, it experienced a serious convergence problem when the flow was turbulent *and* heat conduction was included. After one year of work, we are happy to report that this problem has been successfully resolved. By changing the formulation of the energy section of the code, the separate iterations between the film and pad temperatures, as illustrated in Figures 1 and 2, are now integrated into a single formulation. The new formulation has been combined with the other solvers in the code and works very well. Potential future upgrades include dynamically allocated arrays, improved search of the pad tilt angles, new pad geometries, improved turbulence model, and the capability for direct lubrication.

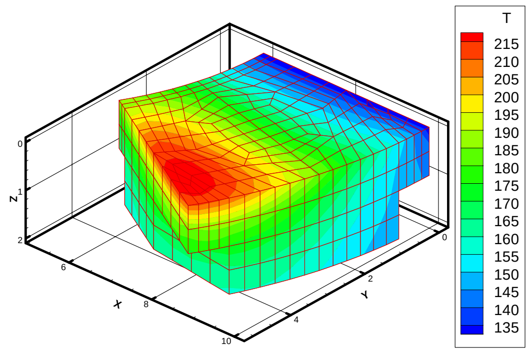


Figure 1: Temperature Plot for Thrust Bearing Pad

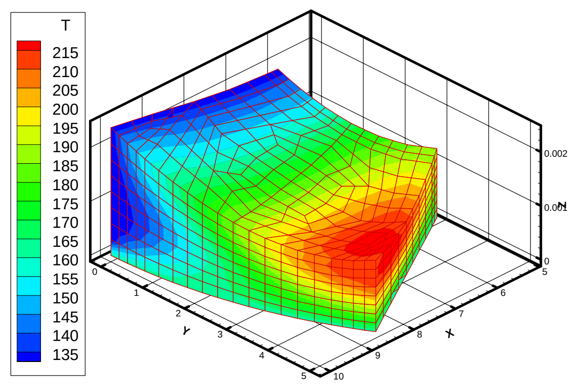


Figure 2: Temperature Plot for Thrust Bearing Film

Labyrinth Gas Seal Code/Program Update

Student: Jie Zhou

Faculty: Paul Allaire, Houston Wood

Research Professional: Minhui He

Project Start Date: Sept. 2000

Funding: ROMAC

Laby4 is a new gas labyrinth seal analysis program under development. It will replace the current code LABY3. LABY3 gives out reasonably good results for cross coupled damping dynamic coefficients, but not as good for principal dynamic coefficients of stiffness and damping. The new key approach is to use three control volumes following a method by Prof. Nordmann from Germany, to better model the fluid stresses, forces and velocity of the fluid filed in the seals. Tascflow, a 3-D computational fluid flow code from AEA, is being used to determine the flow properties, such as the control volume velocity and pressures, as well as suitable boundary conditions to obtain a better flow description for the three control volumes in each industrial labyrinth seal configuration.

Progress of the past year

Validation of the computer code Laby4 was carried out by comparing results obtained

BEARINGS AND SEALS

Labyrinth Gas Sal Code/Program Development

from CFD simulation, experimental measurement, Laby4 prediction and Laby3 analysis.

Prasad's three teeth case and Stocker's four teeth case were selected for validation of leakage flow prediction as shown in Figure 1. For Prasad's case, the maximum error of CFX simulations to experimental is 2.47%, Laby3, 18.61%, and Laby4, 7.72%. For Stockert's case, two clearance cases are compared. The maximum relative error of CFX simulations to the experiment was 3.88% for the smaller clearance,

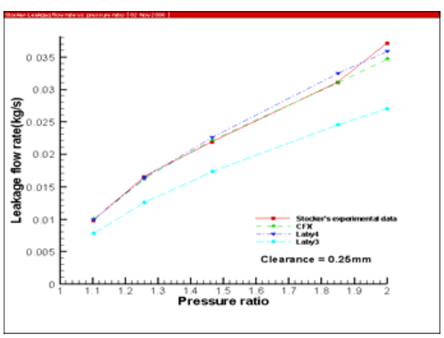


Figure 1: Leakage Rate in Laby Seal

Laby3,27.04%, and Laby4, 4.54%. CFD simulation results are quite near the experimental results. Laby4 gain much improvement than Laby3 and the results are comparable to CFX prediction. In some cases, Laby3 fails to arrive at a solution due to singularity of the matrices.

Benckert's four teeth case was selected for validation of rotordynamic coefficients prediction of different inlet swirl velocity so the comparison could be made between experimental data, CFX simulation, Laby4 and Laby3. Figure 2 shows the results. Except for Laby3, all other tools predicted rotordynamic coefficients for all inlet swirls. The maximum relative error of CFX simulations to the experiment is up to 33%, Laby3, 33%, and for Laby4, 70% for one case and slightly over-predicted the pre-swirl 230 m/s case only about 5%.

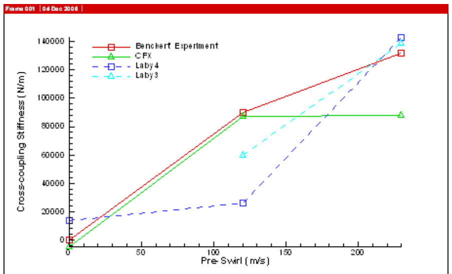


Figure 2: Cross Coupled Stiffness in Laby Seals

Comparison to Prasad, Stocker and Benckert's experiments have shown that CFX is a good tool to predict leakage and rotordynamic coefficients and might work as an experiments replacement. Figure 4 shows swirls in Laby Seals.

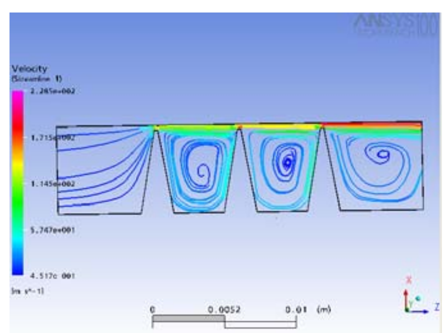


Figure 4: Swirls in Laby Seals

Further validation of Laby4 for rotordynamic coefficients was carried out for Childs 16 teeth case as shown in Figure 3. Laby4 yields better results than Laby3 with a properly supplied parameter set. Laby3 results are usually an order of magnitude smaller than the experiment results. For Laby4, stiffness prediction is about 188% over estimated, cross-coupling stiffness 88%, and direct coupling, 80%.

Laby4 will be released this January.

Future work.

There will be no more future work for bulk flow type codes in ROMAC. Future work will focus on CFD simulation on labyrinth seals directly, such as developing a simulation guide on labyrinths seal leakage and rotordynamic coefficients. Problems that need to be solved may include geometry set up, mesh size optimization and boundary conditions determination,

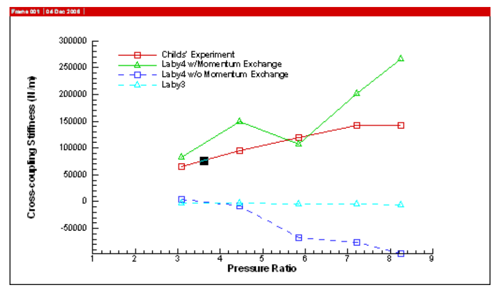


Figure 3: Cross Coupled Stiffness in Laby Seal

Seal Test Rig and Seal CFD Analysis

Student: Josh Keely, Marion Reid, Amir Younan

Faculty: Chris Goyne, Paul Allaire, Jim McDanial, Houston Wood

Funding: ROMAC

Start Date: September 2005

Of particular interest to the air compressor industry is the coupling between the fluid mechanics and rotor dynamics of high pressure aerodynamic seals. Unfortunately the seal coupling is often poorly understood and poorly modeled and this results in poor prediction of compressor performance and failure limits. There is thus a need for experimental investigation of the phenomenon. In order to fully understand the physics involved, not only are surface measurements needed, such as wall pressure and temperature, but also instream measurements of the flow properties. Of particular interest to fluid modelers are the velocity fields within the seal, both temporally averaged and instantaneous. With velocity databases in hand for various seal geometries at typical compressor operating conditions, seal modelers will be in a position to significantly improve the prediction accuracy of seal performance and fluid-rotor coupling.

This project will involve the development of a test rig to perform these measurements on honeycomb, hole pattern, and labyrinth type seals. Current existing rigs of this type are limited to pressures of approximately 1000 psi, while industrial pressures are often found near 3000 psi. Of primary importance for this new rig is the ability to achieve actual seal pressures for better comparison to industrial conditions. Data will then be collected and used to validate and improve upon current seal CFD codes.

Professor Houston Wood of Mechanical and Aerospace Engineering will lead the fluid modeling and validation effort that began in the summer of 2006. Professor Wood has experience in modeling and CFD analysis of fluids in rotating machinery, and he has most recently been responsible for designing the blood flow path in artificial heart pumps. He was a ROMAC faculty member and Director of ROMAC about 15 years ago.

Due to the aerodynamic diagnostics involved with the experiment, two faculty members with aerospace backgrounds have become involved. Prof. Chris Goyne and Prof. Jim McDaniel from the UVa Aerospace Research Laboratory will monitor the project, with Paul Allaire advising in the more conventional ROMAC areas. A M.S. student, Joshua Keely, began work on the project in 2005, and Marion Read is doing the mechanical design work to this test rig.

The conceptual design of the test rig is very similar to that of conventional fluid bearing rigs with roller bearings located on either end of a shaft that goes through the seal. The rig will be driven by a high speed electric motor with variable frequency drive. The rig is to be designed so that the bearings are interchangeable, with initial tests being done using roller bearings before it

BEARINGS AND SEALS

Seal Test Rig and CFD Analysis

is fitted for magnetic bearings. The use of magnetic bearings will enable the rig to be used for the measurement of fluid-rotor coupling. The aerodynamic seal itself will also be interchangeable, so that various geometries can be tested. A high pressure window will provide optical access to the seal geometry, and a Laser Doppler Velocimetry (LDV) system will be utilized to measure the flow velocities.

Progress in the past year*:*

Optical efforts since the ROMAC Annual Meeting in June have focused on developing the small-scale optical test experiment discussed at the meeting, while providing support in the fluid dynamics field to the development of the full-scale seal test rig at the University.

The goal of a small-scale optical test experiment was to provide confirmation that particle image velocimetry (PIV) measurements were capable within the seal clearances to be studied by the full-size rig (C = 0.05"). The initial design presented at the Annual Meeting, in which an acrylic tube surrounding a non-rotating aluminum cylinder formed the simulated stator and rotor respectively, was refined to include proper hardware and sealing components. However, the design grew steadily more complex, and eventually failed to resemble the original idea of a simple and cost-effective feasibility experiment. The concept was revisited, and a simpler, smaller design, utilizing symmetry in the original model was developed. An *AutoDesk Inventor* rendering of the design is shown in Fig. 1.

The issue of imaging of the flow field was also thoroughly investigated. Optical issues including magnification, working distances, field of view, and camera angles were examined, and appropriate lenses were chosen. Efforts are currently underway to rent the lenses for an initial experiment to validate calculations and ensure they are a viable solution. Other PIV experiments being conducted at the Aerospace Research Laboratory (ARL) were observed in order to become familiar with the PIV technique. A calibration target needed for the cameras was designed and produced within the University via direct laser etching, providing substantial savings ($3000) over commercial options.

Concerning the overall rig design, choices of the first geometries to be tested were narrowed down to simple seals that could be easily manufactured and modeled via Computational Fluid Dynamics (CFD) or analytical methods. They are similar to those tested in the water rig at Texas A&M, so data may be compared between the experiments. These simpler designs, while not of the greatest concern to industry, will allow the functionality of the rig to be improved, if necessary, before testing of the more complex honeycomb and hole-pattern geometries.

Seal Test Rig and Seal CFD Analysis

Finally, a thorough analysis of the costs associated with either renting pre-filled pressurized cylinders to provide the air flow for the rig (low initial cost, high cost-per run), or purchasing a compressor and cylinders so filling could be done in-house (high initial cost, low cost-per-run) was undertaken. The number of runs necessary to for the latter option to be optimal was calculated for various scenarios, but it was determined that the former option would be used for the first few runs, and then the problem would be reexamined.

Josh Keeley apologizes for the initial "young graduate student who has no idea what he's talking about" schedules presented at the Annual Meeting. The goals now are to begin fabrication on the apparatus for the small-scale experiment by the first of the year, to have the flow and seeding system designed by the end of January, and data from the experiment ready for presentation for the 2007 Annual Meeting in June.

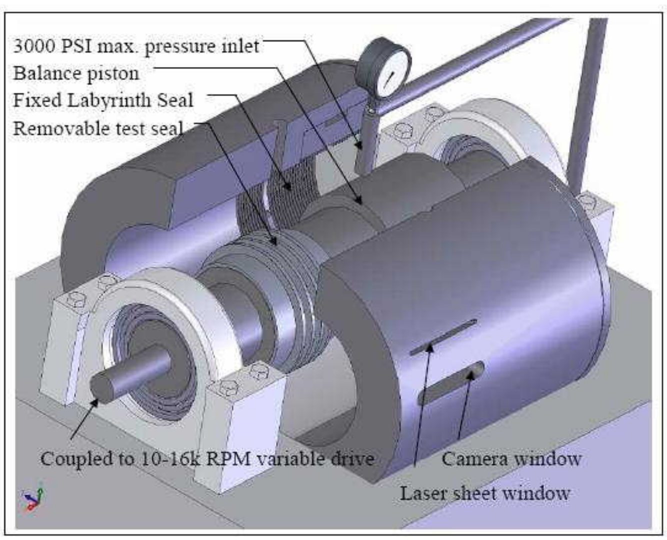


Figure 1: Seal Test Rig

Seal Analysis by Computational Fluid Dynamics

Students: Nina Mohleji

Faculty: Houston Wood

Research Professional: Alex Untaroiu

With modern computers and software, the new *fluid dynamics laboratory* consists of *numerical experiments* conducted by computational fluid dynamics (CFD) methods. For example, today airplanes can be designed, built and flown with little or no wind tunnel testing, and Romac intends to bring this state of the art methodology to fluid seals.

The effort has been started using CFD to examine labyrinth seals, and the CFD code will be validated against known solutions and verified against experimental measurements. In the future, other seal geometries, such as hole pattern seals and others, will be modeled by CFD. This numerical laboratory will provide fundamental understanding of the fluid flows in seals and will allow improved designs to reduce leakage and to provide dynamic behavior of seals. There are a number of ways in which this type of analysis for seals can be made available to the Romac community. Over the next few months, the Romac faculty will develop some plans for discussion at the annual meeting in June 2007 at which time the member companies can offer suggestions and recommendations for this effort.

Centrifugal Compressor Surge Control Test Rig with Active Magnetic Bearings

Graduate Student: Dorsa Sanadgol

Faculty: Eric Maslen, Zongli Lin

Funding: ROMAC, Kobe Steel

Project Start Date: 1999

Project Objectives:

The primary goal of this test rig is to determine the effectiveness of active magnetic bearings (AMBs) to suppress surge in high speed centrifugal compressors. The magnetic bearing assembly, consisting of thrust and radial magnetic bearings, has been constructed and tested for the surge control compressor test rig. There are two specific areas of interest as a part of the overall goal: 1) experimentally evaluate the sensitivity of the compressor characteristic near surge to determine the feasibility of surge suppression and 2) measurement of the AMB forces required to accomplish surge control. If a suitable AMB thrust bearing is present in the compressor, either using an existing AMB thrust bearing or including one in a new compressor design, the surge control can be implemented with AMB software.

Experimental Testing in Past Year

A University of Virginia test facility has been renovated, power supplied and the compressor base installed. Figure 1 shows the current assembled test rig base, com-

Figure 1: Assembled Test Rig

pressor spindle and control rack. We have a single stage air compressor from Kobe Steel with a rating of 75 hp (55kW) with flow rate of 1475 cfm (2500 cu-m/hr) with a pressure ratio of 1.72 and impeller of 9.7 inches (250 mm)diameter. Figure 2 shows more detail of the existing experimental components of this rig. Figure 3 shows the compressor geometry. Miniature pressure transducers have been placed in the compressor housings to measure dynamic and static pressures during operation.

The rotor, AMB amplifiers and sensors have been characterized. The non-rotating AMB rotor/impeller spindle was levitated with an H_{inf} thrust bearing controller and PID controller for the radial bearings using RT Linux. After initial levitation, the AMB open loop stiffness and actuator gain were measured. Figure 4 shows some of the test results and figure 5 shows the control panel. Following these test results, the test rig was hit by lightening and the sensors severely damaged.

A tip clearance mathematical model of the blade tips and the adjacent stationary shroud

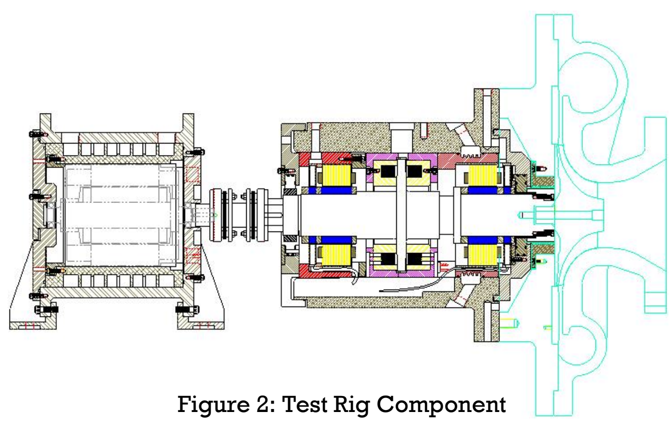


Figure 2: Test Rig Component

has been developed. This is based upon Greitzer's nondimensional mass flow rate and

Centrifugal Compressor Surge Control Test Rig with Active Magnetic Bearings

Figure 3: Compressor Geometry

compressor characteristic equations. They yield a highly nonlinear mathematical system. Also, Senoo's efficiency loss model is included in the model for a quasi-steady tip clearance model. The clearance model was linearized for use in the AMB thrust bearing controller implementation.

A theoretical study of compressor flow system stabilization with mass flow feedback was carried out. The compressor surge suppression is fundamentally a transient response problem that must be activated by the AMB control system. The theoretical study of the compressor tip clearance/AMB control system was evaluated to determine the effects of physical limitations such as the actuator/amplifier bandwidth on the performance. The transient response theory for a backstepping control method was developed for this system. It was determined that a high bandwidth thrust bearing/amplifier was not necessary, based upon the tip clearance mathematical model. Also, an uncertainty analysis was performed for this tip clearance model which showed that the thrust bearing dynamic properties needed to be included for proper system position control performance.

All of these results were completed in August of 2006. The details are reported in the associated ROMAC report – Dorsa Sanadgol's Ph. D. thesis.

Future Research

There is much yet to be done on this test rig. The new sensors must be installed and made operational. The piping system has been designed, as shown in Figures 5 and 6, and piping is available for the test rig. The test rig must be assembled and flow established. The motor must be operated and the AMB system re-characterized at operating speed and with flow. The motor is a prototype induction motor without manufacturer's data for the heat generation. There is a water jacket with a forced convection radiator. It needs to have a cooling system designed

The specific compressor characteristic flow parameters must be evaluated and placed in Greitzer's model as well as in Senoo's efficiency formula. The tip clearance static and dynamic parameters must be measured and the model verified. The mass flow

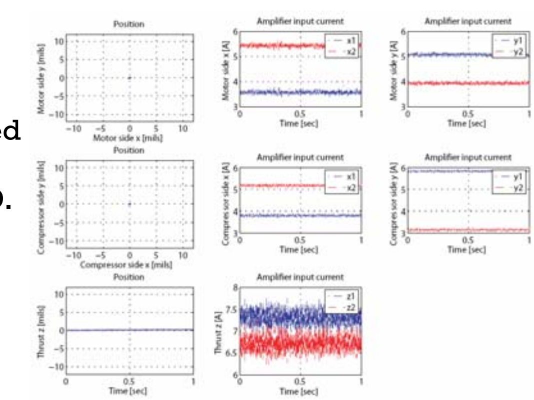


Figure 4: Test results for compressor surge rig

must be estimated from the pressure measurements. A specific backstepping control

FLUID FLOWS

Centrifugal Compressor Surge Control Test Rig with Active Magnetic Bearings Cont.

method based in these parameters must then be developed. The AMB surge control effectiveness must be measured and compared to baseline compressor performance. Finally, the AMB controller must be made robust to system uncertainties.

There are a number of issues that need to be resolved concerning this test rig. First, some specific ROMAC company needs and interests in this rig need to be identified. Approximately 6 companies expressed interest in continuing this research at the Annual ROMAC meeting in June. This is a rather expensive test rig for ROMAC to fund as compared to many of our theoretical (software) projects. Second, several companies have commented during the Annual Meeting and later, that this compressor may not have the optimal configuration to yield useful test results. Third, Dorsa has graduated with her Ph. D. ROMAC has very well qualified personnel to continue work on the test rig as well as students that wish to work on it. We are in the process of identifying interested companies. Fourth, a new Ph.D. student, Paeblo Yoou is starting work on this rig in January 2007.

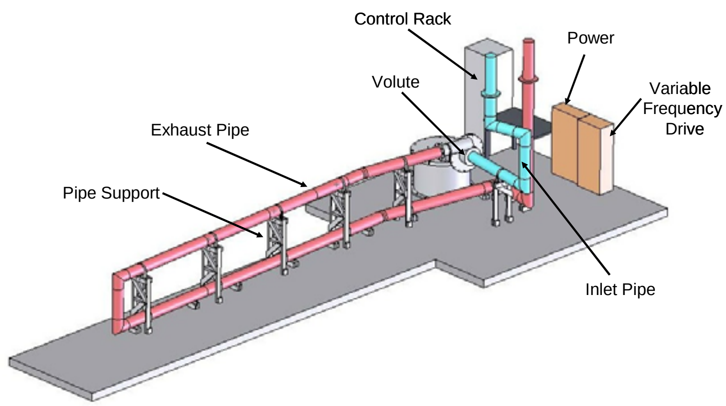


Figure 6: Piping System Design

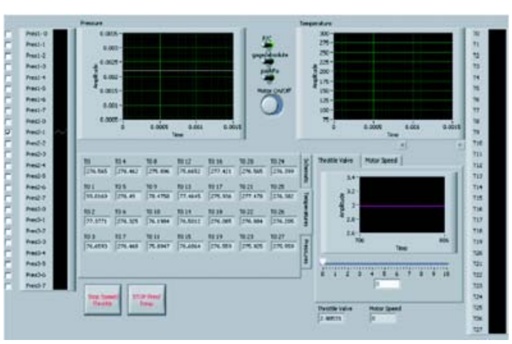


Figure 5: Control Panel

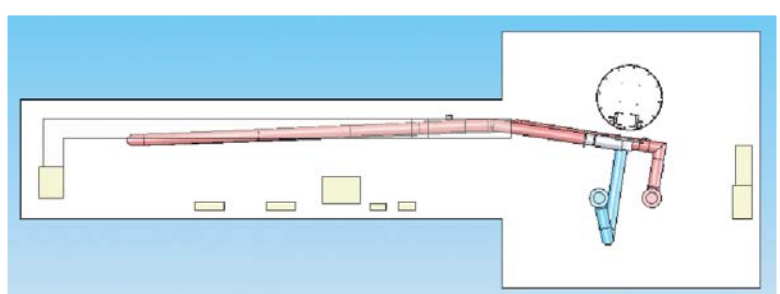


Figure 7: Top view of piping for surge test rig

Rotating Machinery Stability Test Rig

Student: C. Hunter Cloud

Faculty: Lloyd Barrett and Eric Maslen

Project Start Date: January 1999

Primary Funding: ExxonMobil, ROMAC, Petrobras

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 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*:*
For two bearing preload designs (0.3 and 0.1), measurements of pad temperatures, shaft centerlines, unbalance response and stability have been compared to modeling predictions. Base and threshold stability measurements were performed at speeds above and below the first critical speed. The first forward and backward modes' stability were also measured as a function of cross-coupled stiffness.

Predictions based on frequency dependent tilting pad bearing dynamics exhibited significantly better correlation with the stability measurements than those assuming frequency independent dynamics. However, unbalance response and stability correlations indicated that bearing damping and support anisotropy are overpredicted. These deviations emphasize the value of conducting such stability measurements to avoid rotor instability problems. They also highlight the need for continued improvements with respect to fluid film bearing modeling. Work is continuing on this project by Hunter Cloud.

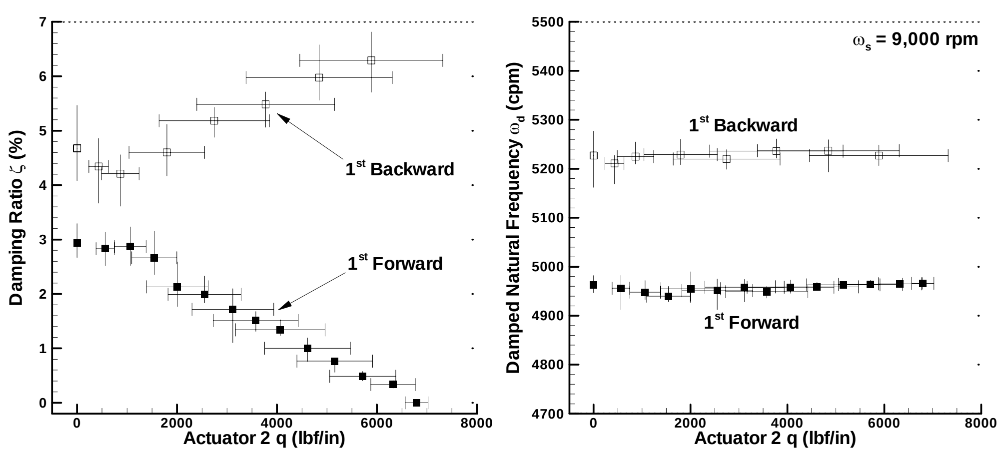


Figure 1: First forward and backward mode rotor stability sensitivity curve measurements (m_p = 0.1)

ROTORDYNAMICS

Rotordynamics

Introduction

The Rotor Dynamics activity in ROMAC includes a number of interrelated efforts. Figure 1 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:

The ROMAC RotorDynamics

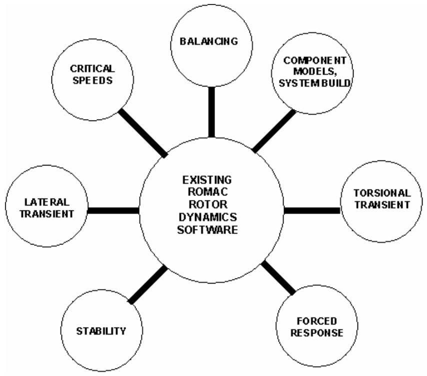


Figure 1: Dynamic Software Research

research and development activities can be described by the three pronged approach illustrated in Figure 2.

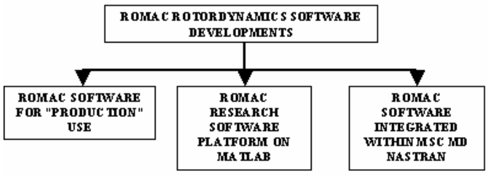


Figure 2: Rotor Dynamic Software Tools

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, RESP2V#
- 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 rotorDynamics*

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. Dr.Min Hui He in this Newsletter elsewhere describes the continuing developments of these "production" tools.

New Rotordynamics Codes and Rotor Modeling

New and emerging issues in RotorDynamics have motivated three 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
- Develoments 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

COMBOROTOR - Coupled Lateral –Torsional-Axial Rotordynamic Analysis

(New Computer Code)

Professional Staff: Minhui He

Faculty: Paul Allaire, Pradip Sheth

Student: Jawad Chaudhry, Amir Younan

Rotordynamics

Project Start Date: October 2006

ROMAC has had the same basic steady state rotor dynamics codes for critical speed, stability and forced response for many years. A lot of ROMAC companies like them and we will continue to support these codes. However, one must move forward to better codes over time – just as ROMAC members must improve their products, add new technology, etc. This project aims to advance the ROMAC "production use" rotor dynamic software in several areas.

First, it will combine the features of the most popular rotor dynamic codes: critical speeds, stability and forced response (CRTSP2, ROTSTB and FORSTAB - a separate codes at this time) into one unified code (COMBOROTOR) for multiple rotors that does not require any file transfers during a full rotor dynamic analysis. Thus, it will be much more convenient for ROMAC member companies to conduct a comprehensive industrial rotor dynamic analysis using this new code.

Second, it will employ the finite element method that is more capable and reliable than the transfer matrix method used by some of our existing codes. A full six degree of freedom coupled beam element including four lateral displacements (two translations in x,y directions and two angular displacements), one torsional angular displacement, and one axial displacement. It will allow for lumped mass elements to model impellers, turbines, and other rigid wheel type structures. Fluid film and rolling element dynamic properties are taken into account as linearized stiffness, damping and mass matrices placed in suitable locations along the shaft. Seals are also taken into account as linearized stiffness, damping and mass matrices placed in suitable locations along the shaft. It will achieve the new capability to analyze coupled lateral-torsional-axial vibration that exist in some geared systems and other systems.

Third, the eigenvalue and other solvers in the new rotor dynamics code are taken from open source Lapack routines that we are incorporating directly into the code instead of older ROMAC internally generated solvers. The Lapack solvers are highly successful, extremely well verified and very well documented routines. A version of them is used in Matlab. In rare cases, the older ROMAC iterative solvers have some errors in frequency or modes. These errors will be eliminated in the new code. Also, tilting pad bearings and flexible supports will be modeled in the time domain, which eliminates an internal search that can potentially lead to missing modes.

This new steady state COMBOROTOR computer program is currently under development. The equations have been completely formulated and the software is currently being written by Minhui He, the developer of MAXBRG. The code will be verified in detail by the MATLAB ROMAC internal rotor dynamics code described next. It will use ROTORLAB or a new user interface as the user interface in its final release.

MATLAB ROMAC Internal Rotor Dynamic Code

Student: Jawad Chaudhry, Blake Stringer

ROMAC Engineer: MinHui He

Faculty: Pradip Sheth, Paul Allaire

To address many of the above rotor dynamic issues, during 2006 ROMAC has developed a RotorDynamic system

Rotordynamics Cont.

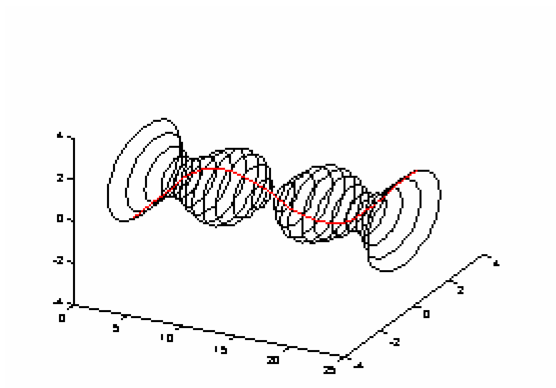
in the MATLAB m-file script. This system is being referred to as a research software platform for ROMAC to provide an in-house research platform for RotorDynamics to experiment with new rotor dynamic ideas that have not been investigated before. MATLAB is chosen for its extremely efficient and easy to 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.

The first part of the project was to replicate the current ROMAC codes, namely CRTSP2, ROTSTB and FORSTAB. The MATLAB code currently has the following capabilities:

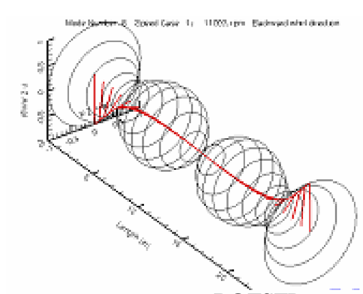
- Uses a finite element formulation to calculate the stability, damped, or undamped natural frequencies and mode shapes for multiple level rotors.
- Includes gyroscopic, rotary inertia and shear deformation effects.
- Fixed or Flexible constraints, including linearized fixed geometry bearing coefficients.
- 4 DOF or 6 DOF beam element with axial, torsion, and bending capabilities.

2D/3D elliptical whirl mode shapes plots.

The results from the code were compared with the current ROMAC codes, CRTSP2, ROTSTB and FORSTAB. A single rotor system with three disks and two bearings was used. The following figure shows correlation between the MATLAB and ROTSTB at a specific modeshape.



MATLAB



ROTSTB

Figure 3: Rotor Mode Shape Using Matlab and ROTSTB

The current ROMAC codes incorporate specific numerical algorithms for the eigenvalues/eigenvectors solutions. The MATLAB platform now provides an alternate method to check the efficacy of the current ROMAC codes and the new coupled lateral-torsional-axial code. Another independent validation correlation of these codes is being done by comparing with the published data in the DyRoBeS User Manual. The next step in the MATLAB software platform development is to make this platform self-contained for obtaining the bearing and seal coefficients from the ROMAC codes by executing them through the MATLAB platform. Our plan is to make the MATLAB platform available to ROMAC members if they wish to independently experiment with the code. This will be done once a brief User Document and Manual is developed. Also, over time, the developments from the MATLAB platform may migrate to the "production" software as well.

Rotordynamics Cont.

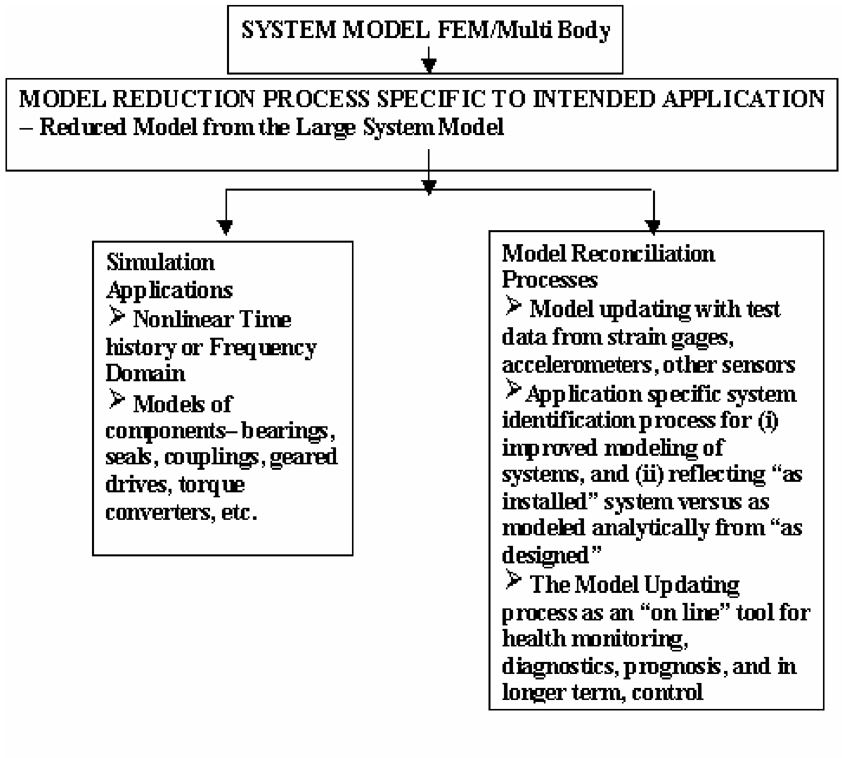


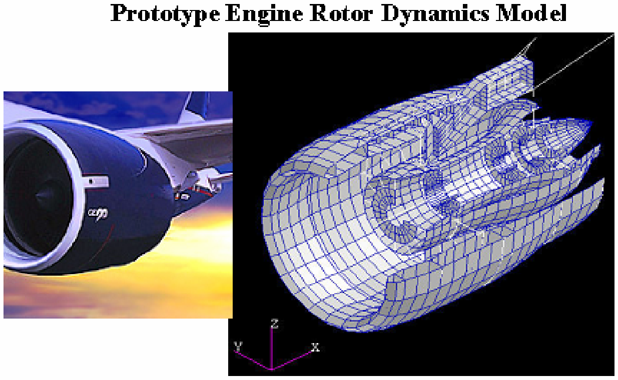
Figure 4: Rotor Dynamics Modeling with NASTRAN

Link with NASTRAN

Another key development in October, 2006 is the implementation of MSC Software MD NASTRAN, ADAMS, PATRAN, and others in ROMAC. MSC Software has joined ROMAC to participate in the integration of RotorDynamics within the NASTRAN framework as shown in Figure 4. Our plan is to utilize the DMAP facility within NASTRAN to create a system which will utilize the full finite element capabilities of NASTRAN. This is expected to allow a much detailed modeling environment for foundations, casings, other support structures as they interact with the rotor bearing system. This integrated environment is expected to eventually incorporate the ROMAC software for bearings and seals. This development on MSC NASTRAN is initially motivated applications to jet engine rotor dynamics, as illustrated in Figure 5, but the development is expected to benefit the ground based turbo machinery RotorDynamic analysis as well.

Figure 5:

Prototype Engine Rotor Dynamics Model



- Fictitious engine model developed at Pratt & Whitney for testing analysis techniques
- Nacelle, strut, and wing models developed at Boeing are also included in the model
- All parts of this model are fictitious, and any resemblance to actual designs past or present is entirely coincidental.

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 below.

In August, US Army Major Blake Stringer began his PhD research in ROMAC to develop a foundation for model based diagnostic/prognosis system for RotorDynamic systems. Blake is pursuing his PhD through the Army's Uniformed Army Scientist and Engineer Program. As a fully Army funded student, Blake brings the opportunity for conducting research in support of the Army's Vehicle Technology Directorate at the NASA Glenn Research Center to further rotorcraft transmission technology.

The transmission is the primary source of noise, vibration, and weight in rotorcraft, which leads to excessive wear

Rotordynamics Cont.

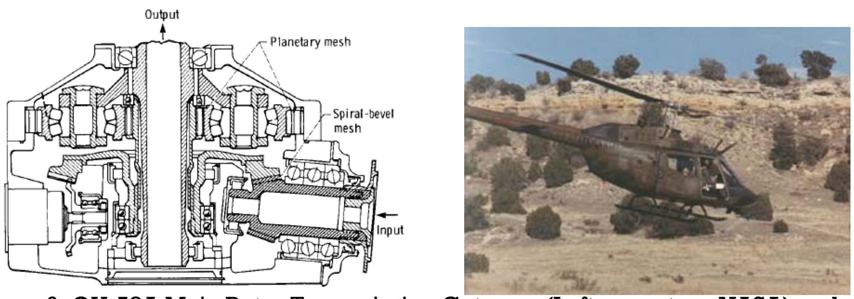


Figure 6: OH-58A Main Rotor Transmission Cutaway (Left—courtesy NASA) and Helicopter (right)

and fatigue of parts as shown in Figure 6. To ensure the safety and reliability of the aircraft while at the same time lowering maintenance costs, the Army is pursuing a condition-based approach to maintenance procedures, or simply, to replace components when they need replacing. A primary requirement for achieving this successfully is the development of prognostic health management systems, which can monitor system components and predict how much useful life remains. One of the criteria for successful implementation of prognostic capabilities is a mathematical physics-based model of the helicopter transmission, which has yet to be fully achieved in the literature.

The dynamics of the gear meshes within the power train are a significant source of excitationas shown in Figure 7. Our research is to capture the effect of these excitations on the transmission gearbox and its effect on the rest of the system. While several examples of geared rotor analysis exist in the literature, the dynamic impact on transmission casings is not well documented, particularly in rotorcraft applications. These modeling developments are expected to assist RotorDynamic modeling for other applications as well.

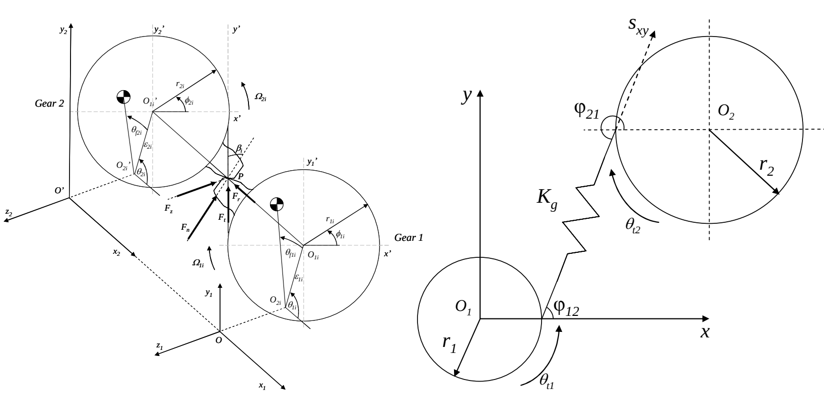


Figure 7: Gear Mesh Modeling Techniques (courtesy Neriya (left) and Fawcett(right)

Optimum Balancing of High Speed Rotors with Uncertainty

Students: Bin Huang, Guoxin Li

Faculty: Zongli Lin, Paul Allaire

Research Professional: Wei Jiang, MHI: Daiki Fujimura, Simens Delaval:Thomas Shoup

Balancing Methods and BALOPT Code

Industrial rotating machines are subject to mass unbalance that is unknown. The only available measurements of this unbalance are obtained indirectly from displacement or velocity probes at limited locations along the rotor. Usually, the rotor unbalance is identified and corrected using the influence coefficient method with a set of trial weights used to predict the correction weights. Figure 1 illustrates the general procedure. The least squares method is employed to minimize the square root of the sum of the squares of the predicted vibration after the application of the correction weights. In this method, it is assumed that the influence coefficients are exact – that is the in-

Rotordynamics Cont.

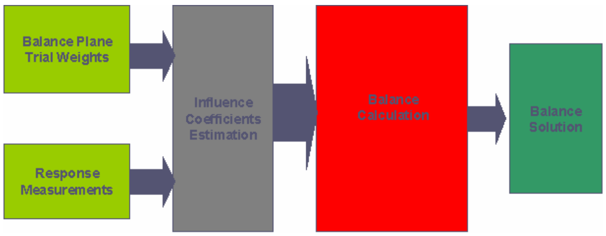


Figure 1: Balancing Procedures for Industrial Rotors

fluence coefficients do not have any uncertainty.

However, many rotors produce influence coefficients which have uncertainty. That is, if more than one set of trial weights is placed on the rotor and a set of influence coefficients evaluated for each set of trial weights, significant differences occur between the different sets of influence coefficients – indicating that there are uncertainties in the influence coefficients.

Four balancing methods were developed in this work and are now available to ROMAC industrial members through the code BALOPT. The first two balancing methods are 1) the standard least squares method, 2) a min-max method where the largest vibration amplitude is minimized, 3) a stochastic optimization method where the uncertainty distribution of the influence coefficients is taken into account in the predicted correction weight calculation, and 4) a worst case min-max method where the maximum uncertainty value is taken into account in the predicted correction weigh calculation but the distribution is unknown. A formal open source mathematical optimization algorithm, called second order cone programming, is employed in BALOPT to obtain the solutions. The full formal equations for the influence coefficient calculation with uncertainty perturbations in complex form are presented in the thesis.

Also, BALOPT takes into account such practical balancing constraints including correction weight limits, weight splitting constraints, response limits at specific rotor locations, and weightings which allow different speeds balancing speeds to be rated differently. Weight splitting is used in cases where only specific balance weight hole locations are available for balance weights.

MHI 1150 MW Turbine Generator – Min-Max Balancing

A Mitsubushi Heavy Industries large industrial turbine generator weighing 727,000 kg and 61.65 m long underwent balancing. It is shown in Fig. 2. The operation speed is 1800 rpm. Vibration measurements were taken at 11 bearings. The objective was to minimize the residual vibration using only the planes 4&5 for balance

Figure 2: Mitsubushi Heavy Industry Large Turbine Generator

weights. The predicted reduction in balancing maximum value using the least squares method was a predicted reduction from a maximum value of 21 micro-m p-p with least squares balancing to 17 micro-m pp in BALOPT with the min-max method with constraints.

MHI Industrial Nuclear Steam Turbine

A MHI large steam turbine case history was also provided to ROMAC for analysis with BALOPT. It is an extremely large unit with one high pressure turbine and three low pressure turbines, as shown in Fig. 3. The operating speed is 1800 rpm, the HP turbine has the first critical speed of 1240 rpm and the LP turbines have the first critical speed of 1340 rpm. The available balancing planes are 4, 5, and 8. BALOPT was employed to predict the residual vibration after

Rotordynamics Cont.

min-max vibration with constraints of maximum balance weights, maximum allowed amplitude and higher weighting at the rated speed (as compared to the critical speed). The results are shown in Fig. 4

Figure 3: Nuclear Steam Turbine Balancing Case History

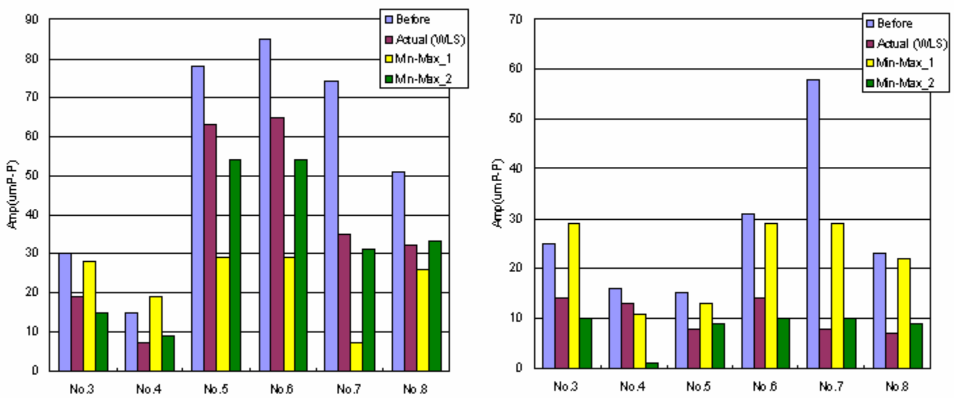


Figure 4: Results on Balancing Nuclear Steam Turbine with BALOPT with Min-Max Method

which show that the maximum residual vibration was 14 micro-m p-p in the actual case with a predicted value of 10 micro-m p-p with the BALOPT min-max method.

Uncertainty Balancing with Experimental ROMAC Three Mass Flexible Rotor

As a part of the research, the influence coefficient uncertainties were measured for a ROMAC three mass 26 inch long flexible laboratory rotor on fluid film bearings as shown below in Fig. 5. Six eddy current sensors were used to measure the vibrations. The resulting influence coefficients had a maximum amplitude difference of 20% and a maximum difference in phase angle of 10% at an operational speed of 2240 rpm, below the critical speed of 2650 rpm. When the ROMAC three mass rotor was balanced with two different methods, the optimum stochastic method reduced the maximum final vibration by 43% as compared to the least squares method. When the min-max approach was employed, maximum final vibration after balancing using the optimum stochastic method was 30% less than the min-max result.

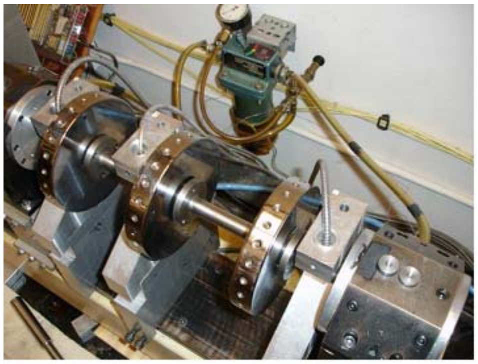


Figure 5: ROMAC Three Mass Flexible Rotor Balancing Test Rig

Ten Stage Industrial Steam Turbine

Balancing data for a 10 stage industrial steam turbine, as illustrated in Fig. 6, balancing case was provided by Siemens Demag Delaval for analysis purposes. It has 4 balance planes, 2 sensor locations, and 5 speeds. Measured influence coefficients were provided for this rotor. Also, the balance pit data results were provided for this rotor using the normal least squares method. The BALOPT min-max method yields a predicted reduction in maximum final vibration response of 7% compared to the least squares maximum vibration response. The differences (uncertainties) in the influence coefficients were simulated and further unbalancing methods examined. The optimum stochastic balanc-

Rotordynamics Cont.

ing method with BALOPT and correction weights produced a predicted reduction of 39% compared to the least squares result. The worst case min-max balancing method and correction weights obtained a predicted reduction of 23% compared to the least squares method. Figure 7 shows a comparison of the predicted results with the stochastic balancing predicting the lowest residual vibration after the placement of balance weights calculated by the four methods.

BALOPT is now available for use by ROMAC companies. Verification has been carried out on a three mass laboratory rotor. It has been tried by several companies and found to be relatively successful as compared to conventional least squares balancing methods.

Sample of uncertainty 1	X1	0.335	0.345	0.015	-0.315	-0.345	-0.035
	Y1	-0.128	-0.128	-0.028	-0.158	0.012	0.432
Sample of uncertainty 2	X2	-0.102	0.068	-0.042	-0.002	0.048	0.028
	Y2	-0.065	-0.145	0.015	0.275	-0.045	-0.035

Figure 6: Industrial Steam Turbine Balancing

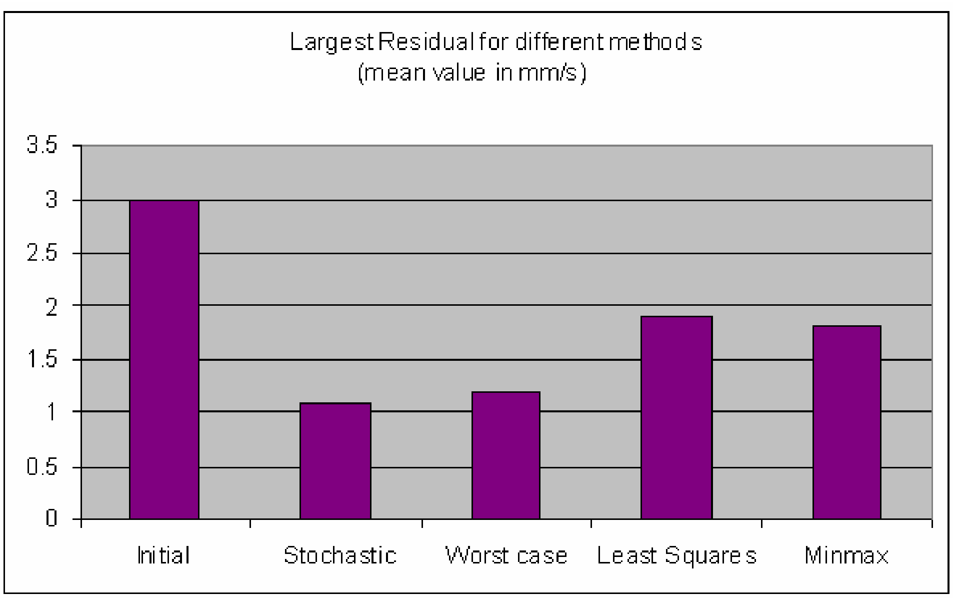


Figure 7: Predicted Residual Vibration Levels After Computation of Balance Weights Using Four Methods

MAGNETIC BEARINGS

Robust Stabilization of Rotor-Active Magnetic Bearing-Substructure Systems

Student: Guoxin Li

Faculty: Paul Allaire, Zongli Lin

Start Date: 1999

Funding: ROMAC, American Flywheel Systems

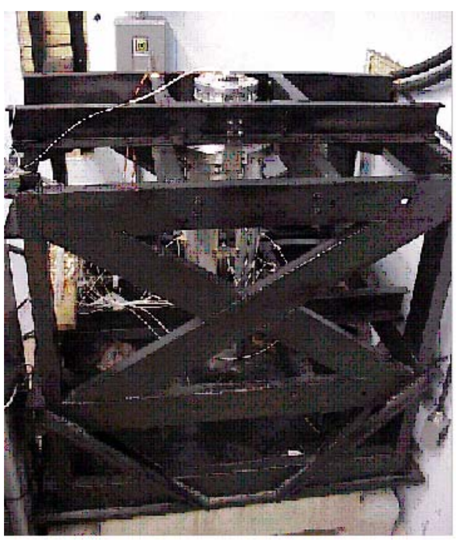
Introduction and Objectives

Active magnetic bearings (AMBs) are increasingly found in industrial rotating machinery such as compressors. They offer the advantages of elimination of oil supply, control of rotor vibrations, increased machine efficiency and other features. The advent of subsea oil and gas production needs will further increase the number of machines on AMBs. Many new machines will be developed over the next decade. However, there are issues that need to be addressed in order to make this happen successfully. As rotors become more flexible, machine speeds increase, seal cross coupling stiffness increases, advanced AMB controllers must be employed. In order to accomplish this, system uncertainties need to be better understood. Also, industrial standards for AMB systems need to be established.

The objective of this work is to help advance the state of the art in AMBs for flexible rotors on substructures as employed in industrial compressors, turbines, flywheels, and other devices. AMBs introduce feedback control into the machine through the bearings. Of significant importance is the consideration of uncertainty in rotor-AMB-substructure systems.

Active Magnetic Bearing Systems

This work developed a systematic modeling procedure for the rotor-AMB-substructure system using a state space formulation as well as AMB actuators. Also included are modes for the sensor, amplifier, anti-aliasing filter and digital controller with time delay. An uncertainty model for each component of the system is introduced into the system model – a feature not often considered in industrial AMB systems.



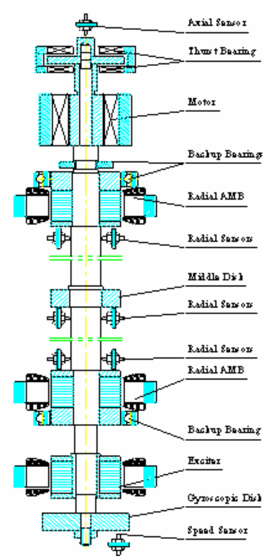


Figure 1. Flexible Rotor and AMB Control Test Rig

An energy storage flywheel vertical test rig with two radial AMBs and one thrust AMB, as shown in Fig. 1, was constructed at UVA to verify these concepts experimentally. The motor and simulated flywheel disk are shown in Fig. 2. Shown in Fig. 3 is the Campbell diagram of critical speeds and operating speeds for this flexible rotor is shown in figure 4. This test rig was originally funded by American Flywheel Systems to use for the development of an energy storage flywheel intended for communication satellites.

The modeling methods developed in this work are verified by comparison with experimental results on the flexible rotor - flexible substructure test rig. The free-free rotor was characterized using singular value decomposition and a gap metric approach employed for uncertainty to be used later to determine the importance of different types of uncertainty.

A good example of a system uncertainty is the substruc-

Robust Stabilization of Rotor-Active Magnetic Bearing-Substructure Systems

ture in this test rig as shown in Fig. 3. In 2000, the substructure had the first mode at 113.8 Hz while in 2004 the frequency had changed to 116.5 Hz for a change of 2.4%. Similarly, the second mode shifted 1.6% as also shown in Fig. 5. The high performance controller for this rotor-AMB-substructure became unstable. Another magnetic bearing system for an energy storage flywheel in a NASA project is shown in Fig. 6. The energy momentum wheel stores 500 W at 75,000 rpm. It is supported in a low loss magnetic bearing. One of the primary issues in this device is the gyroscopic effects.

Uncoupled Simplified (Single Input/Single Output) Rotor Analysis

Often in the AMB field, the effect of various control parameters and their effect on system stability is not easily understood. To better understand their effects, a simplified decoupled single input/single output (SISO) rotor-AMB system was formulated and analyzed. It contained 1) a controller gain, 2) a rigid rotor-AMB component, and 3) equivalent single mass flexible rotor component with seal cross coupling and gyroscopics, and 4) an equivalent single mass flexible substructure component as well as 5) a digital controller time delay model. The system equations are shown in Fig. 7 with a system nominal dynamic system and a system dynamic model perturbed with uncertainty.

It is important to know what system parameter uncertainties are easily suppressed by controllers and those that are not easily controlled. In

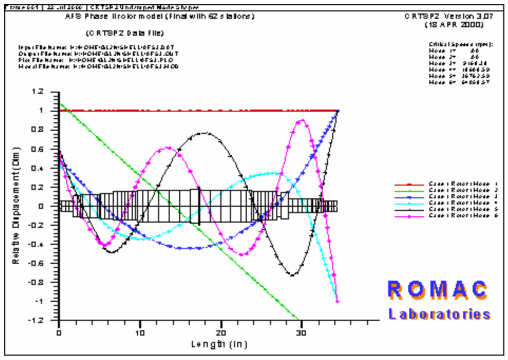


Figure 3: Rotor Free-Free Modes

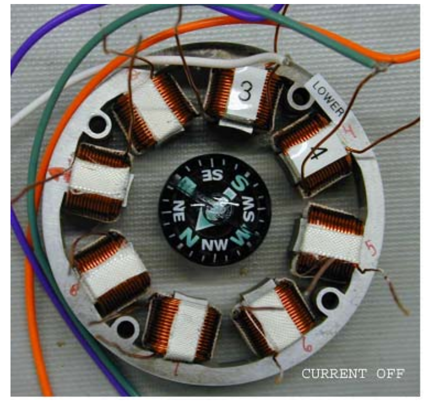


Figure 6: NASA Energy Storage Flywheel With Gyroscopic Effects

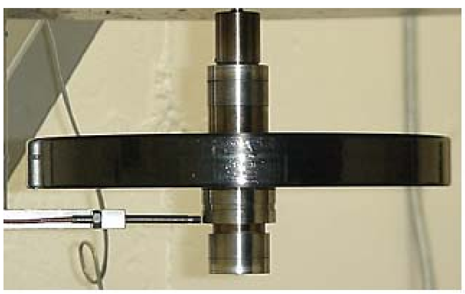


Figure 2: Simulated Flywheel Disk in AMB Control Test Rig

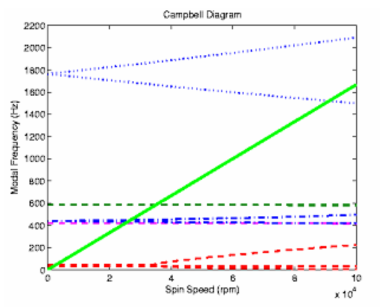


Figure 4: Campbell Diagram

- 1st Mode: 113.8 Hz—116.5 Hz or 2.4% shifting
-
- 2nd mode: 136.6 Hz—138.8 Hz or 1.6%

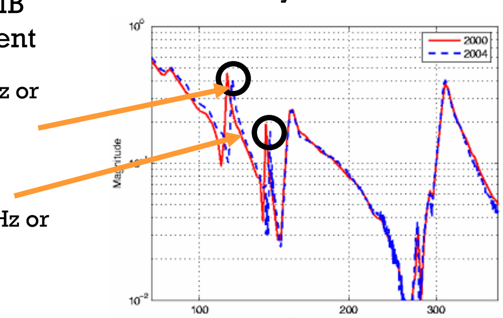


Figure 5: Change in Substructure Frequency for Test Rig Lead to Unstable Controller

Robust Stabilization of Rotor-Active Magnetic Bearing-Substructure Systems

$$G(s) = k_i \left[\underbrace{\frac{\varphi}{s^2 - i\Omega g_0 s - \omega_0^2}} + \underbrace{\sum_{r=1}^{m} \frac{\eta_r}{s^2 + (c_r - i\Omega g_r)s + k_r - \tau_r i}} + \underbrace{\sum_{k=1}^{n} \frac{\phi_k}{s^2 + d_k(s) + \omega_k^2}} \right] e^{-\theta s}$$

Figure 7: Simplified, Uncoupled, Rotor-AMB-Substructure Model for Understanding Control Effects

the analysis of this simplified system listed above, the following results were obtained:

Uncertainties Easily Suppressed by Stabilizing Controllers:

1. Rotor substructure internal damping
2. AMB open loop stiffness and actuator gain
3. Digital controller time delays
4. Sensor gains and amplifier gains

Critical Uncertainties:

1. Seal and aerodynamic cross coupling
2. Rotor or substructure natural frequencies
3. Gyroscopic forces

Then, we know which rotor-AMB-substructure component uncertainties to focus attention on. Many details of this analysis are given in the thesis.

Uncertainty Characterization

The gap metric employed in this work is defined as the "distance" between the open loop nominal and perturbed dynamic system by a scalar parameter between 0 and 1. The gap metric is closely related to the stability of the system. The nominal system control design with adequate general stability margins is robust to uncertainties that generate small gap metrics. It can be evaluated in terms of frequency as the distance between two system transfer functions, as described in the thesis.

Performance Measures of Rotor-AMB-Substructure Systems

Generally it is desired that performance measures be established for these systems. There are four performance functions that are important:

Performance Functions:

1. Output sensitivity function – sensivity to noise, load and command tracking
2. Input sensitivity function – sensitivity to load disturbance
3. Output complemetary sensitivity function – sensitivity to noise in plant output
4. Input complementary sensitivity function – sensitivity to load disturbance on control input

These four functions are discussed in detail in the thesis. It is further shown that there are only four independent transfer functions which determine system internal stability. The control loop is illustrated in Fig. 8. Here G represents the plant transfer function and C represents the controller transfer function. It should be noted that the recently adopted ISO standard for rotor-AMB-substructure only specifies the output sensitivity function, a necessary but not sufficient condition for real system robust performance.

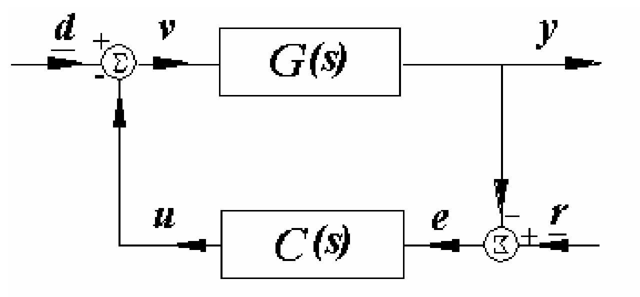


Figure 8: Feedback Control

Robust Stabilization of Rotor-Active Magnetic Bearing-Substructure Systems

Advanced Mu Synthesis Controller

Several advanced controllers were obtained for this rotor-AMB-substructure test rig. Some advanced controllers, such as H-infinity controller, may utilize pole/zero cancellation, have unstable poles, create clusters of system modes and have other features that are not acceptable for industrial rotating machines where the critical speeds/natural frequencies may change over time or with changes in operating conditions. These controllers are not robust to system uncertainties. Generally, the mu synthesis approach controller design was found to be more robust for industrial rotating machines. Again, many more details are given in the thesis.

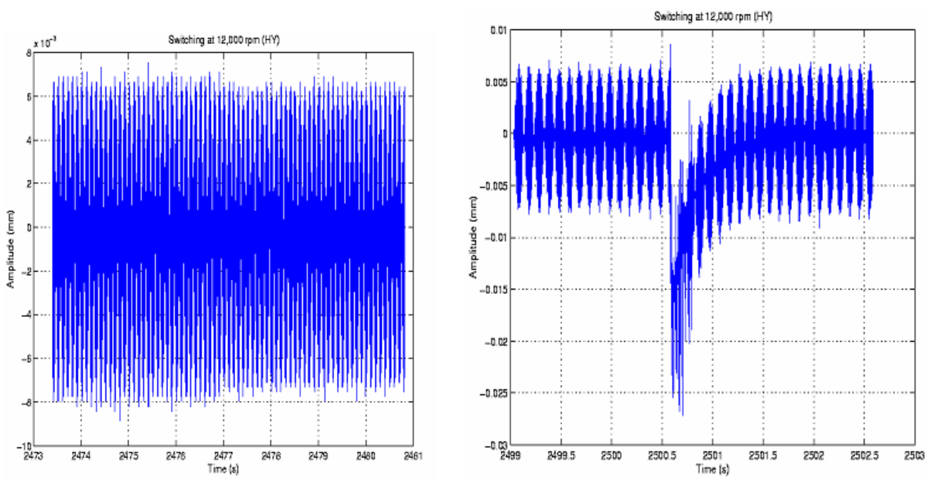


Figure 10: Measured Bumpless Transfer in UVA Test Rig at 12,000 rpm

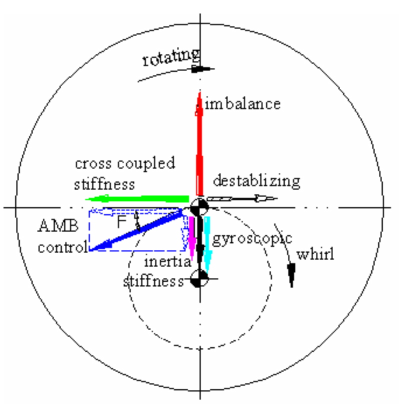


Figure 9: Forces in Gyroscopic Rotor Systems with Inertia Forces, Seal Forces, Unbalance Forces

Control Design for Gyroscopic Systems

Control design for gyroscopic AMB rotors, with cross coupled stiffness such as in industrial seals, poses special problems as illustrated in Fig. 9. Gyroscopic forces are very speed dependent that are best compensated for by gain scheduling. A piecewise mu synthesis method was found to work well. These methods were applied to the UVA test rig shown in Fig. 1 and the NASA energy storage flywheel shown in Fig. 6.

As transfer from one mu synthesis controller to the next occurs, a bumpless transfer was developed by using the system existing operating conditions as initial conditions for the next controller. This was successfully implemented at 12,000 rpm as shown in Fig. 10. The plot on the right shows a major change in rotor displacement with bumpless transfer while the plot on the left shows the use of bumpless transfer without a major shift in rotor displacement.

MAGNETIC BEARINGS

Self Sensing of Magnetic Bearings with Cryogenic Applications

Research Professional: Wei Jiang

Faculty: Paul Allaire, Pradip Sheth

Students: Bo Wang, Gavin Garner

Funding: NASA

Start Date: 2005

Active magnetic bearings (AMBs) usually require one (thrust bearing) or two (radial bearing) position sensors to determine the rotor position and provide the feedback signal to allow the controller to operate. There are several types used currently: 1) eddy current, 2) inductive, 3) optical, 4) capacitive, etc. These position sensors are relatively expensive, increase the wire count of the AMB, and reduce the reliability of the AMB in the long run due to potential wire or sensor failures. In applications where the sensor has to operate through one or two conductive metal cans, such as in a canned pump, these types of sensors do not perform well. Various self sensing methods have been published in the literature: 1) observer methods, 2) switching frequency modulation method, and others. In many cases, the developers of these methods use the coil currents in the AMB which are prone to noise instead of the voltage.

We have developed a self sensing method that eliminates the need for a separate sensor and employs the AMB magnetic and coil structure to accurately determine the rotor position. It uses the voltage between opposing bearing coils to evaluate the difference in air gaps on each side of the rotor. Part of this system involves a permanent magnet bias that provides a constant bias to the bearing at no power consumption cost. It uses controllable microchips on a small circuit board to analyze the voltage signals and determine the rotor position. A digital voltage signal combines controlling the currents in the AMB with self sensing. Figure 1 shows the control board 4" xy 5". Fig. 2 shows the example measure rotor position giving a very linear signal.

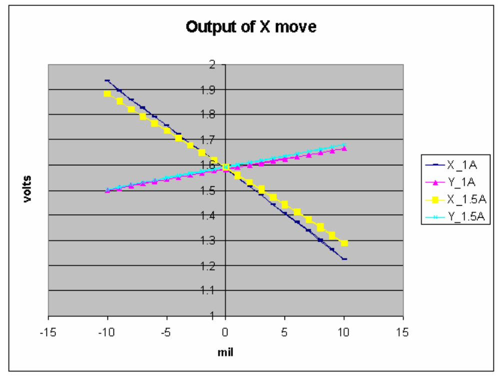


Figure 2: Example Measured Displacement in Self Sensing AMB

The first application of this system, funded by NASA, is for an AMB system in a cryogenic atmosphere found on the moon. This project is directed toward drilling holes about 2 meters deep in the moon surface to determine if there is any evidence of frozen water. If water could be found and melted for astronaut drinking (think what it costs to send a liter of earth water to the moon in the Space Shuttle for the astronauts to drink), this would be a very useful thing.

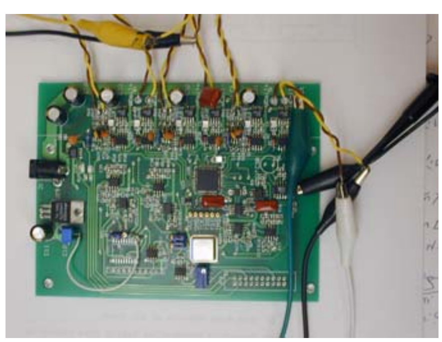


Figure 1: Prototype Self Sensing Magnetic Bearing Control Board

We feel that this self sensing bearing will have many other applications in industrial machines where accurate position sensing is required but it is simpler to not use a separate sensor. The reliability will be higher and the cost lower. Also, for any applications with a can, this is an optimal approach. Current research is optimizing the AMB self sensing bearing.

Horizontal AMB Controls and Rotor Drop/Retainer Bearing Test Rig/Analysis

Students: Sephen Evans, Simon Mushi, Amir Younan

Faculty: Paul Allaire, Zongli Lin

Research Professionals: Bingxin Tang, Wei Jiang

Start Date: 2006

Introduction and Objectives

There is a significant need for more research on active magnetic bearings (AMBs) supporting flexible rotor, the controllers associated with these AMBs, and rotor drops on retainer bearings for these systems. Industrial machines with flexible rotors on AMBs are subject to instabilities due to several factors: seal cross coupling, magnetic bearing open loop stiffness, and other factors. Also, there is a significant need for the development of API style standards for AMB supported rotors – there is no standard at this time. This current situation makes it very difficult for OEMs and end users to establish a performance standard for use with the AMB manufacturing firms.

ROMAC is developing a new horizontal AMB controls test rig for studying advanced controls effects in flexible rotor applications. We already have a vertical rotor-AMB-substructure controls rig, as described in the previous article on the Ph. D. thesis work by Guoxin Li. The objectives for this test rig are to study the interaction between the AMB controller and instability mechanisms including seal cross coupled stiffness effects at the center of the rotor, advanced controls development, system uncertainty effects and improved controls development, and rotor drop effects on the retainer bearings as well as on the rotor.

Test Rig Design

The new AMB controls horizontal test rig is under design at this time. Figure 1 shows the rotor geometry. It has two AMBs, one at each end, two masses perhaps representing the two sets of compressor impellers in a back to back compressor, and a third AMB at the center providing cross coupled stiffness representing destabilizing seal cross coupling effects such as found in a balance piston seal in a back to back compressor. We have several AMBs available from previous ROMAC projects (by Mary Kasarda on the NASA sponsored AMB power loss test rig in her Ph. D. thesis) to use in this test rig. Also, two rolling element retainer bearings are placed just outside of the two support AMBs. An existing motor and speed increaser (formerly used by Ron Flack for his fluid film bearing test rig) will drive the rotor to approximately 10,000 rpm. The entire unit will be placed upon an existing large concrete base in the ROMAC lab.

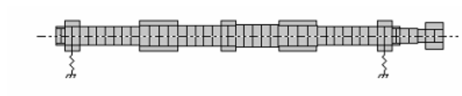


Figure 1 Rotor Geometry for AMB Horizontal Controls Test Rig

The rotor is approximately 36 inches long and an average of approximately 2.25 inches in diameter. It is a flexible with two bending critical speeds below the maximum operating of 10,000 rpm, as illustrated in Fig. 2. The AMB bearing closed loop stiffness is in the range of 15,000 lbf/in to 20,000 lbf/in.

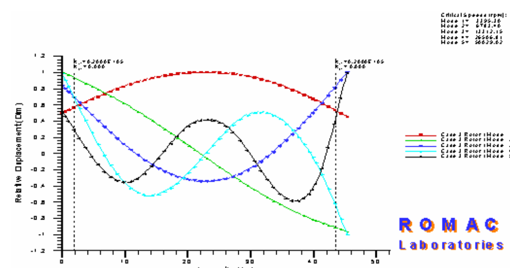


Figure 2: Example Critical Speed Mode Shapes for Hoirzonal Controls Test Rig

The rotor has been designed and will be machined this winter. The AMBs, motor, speed increaser, concrete base, amplifiers and much other required test rig material is al-

MAGNETIC BEARINGS

Horizontal AMB Controls and Rotor Drop/Retainer Bearing Test Rig/Analysis

ready available. The rest is easily purchased.

Rotor Drop on Retainer Bearings

Another major issue in AMB supported rotors is the effects of dropping the rotor on retainer (auxiliary or back up) bearings. The rotor is not in contact with the retainer bearings during normal operation. When the power supply to the AMBs is cut off, the rotor drops rapidly on the retainer bearings. Rolling element retainer bearings are definitely not designed to take such abuse. Often, the can only take between 5 and 15 drops before they have to be replaced.

Our studies on rotor drop will be both theoretical and experimental on the AMB controls horizontal test rig. The theoretical transient flexible rotor drop analysis is needed because there are only so many experimental rotor drops one can practically do while the theory can illustrate many problems and potential solutions to retainer bearing life issues. Coupling this rotor drop model with our modeling of rolling element bearings (as described previously in this newsletter relating to the Ph. D. work of Amir Younan), will provide much information on rotor drops.

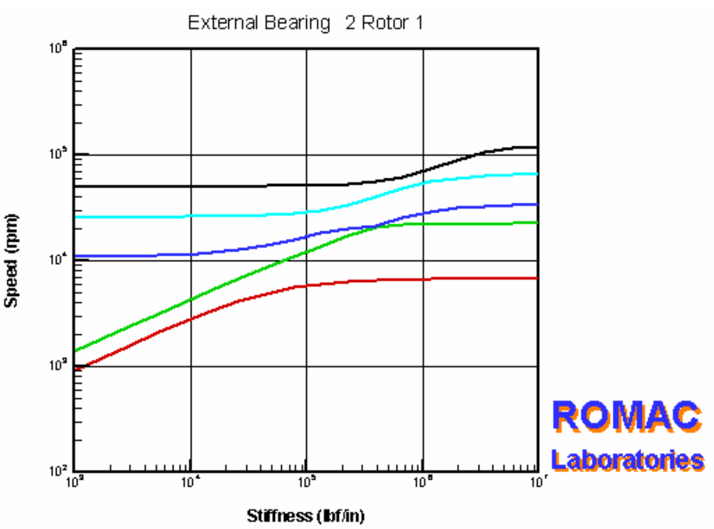


Figure 3: Critical Speed Map

A full set of nonlinear transient response equations has been developed for the rotor drop analysis including the effects of rotor unbalance. It includes 1) initial free fall of the rotor downward, 2) impact on the retainer bearing, 3) rotor sliding along the retainer bearing inner surface, 4) rotor rolling of the rotor on the retainer bearing surface, and 5) rotor whirling along the retainer bearing inner surface. The equations have been implemented in a Matlab code and several case examined.

This work will be coupled to the rolling element bearing analysis, also under development. One cannot really understand what is happening with the rolling element retainer bearing without a simultaneous analysis of the rolling element bearing components.

Summary

Both the experimental work on the test rig and the theoretical modeling will continue. This is a quite low cost effort by ROMAC as most of the test rig components are available from former projects. We are very interested in industrial company input and partners in this research effort. It is expected that various industrial firms interested in AMB supported rotors may want to have specific issues evaluated for their machine configurations on this test rig or using these theoretical models.

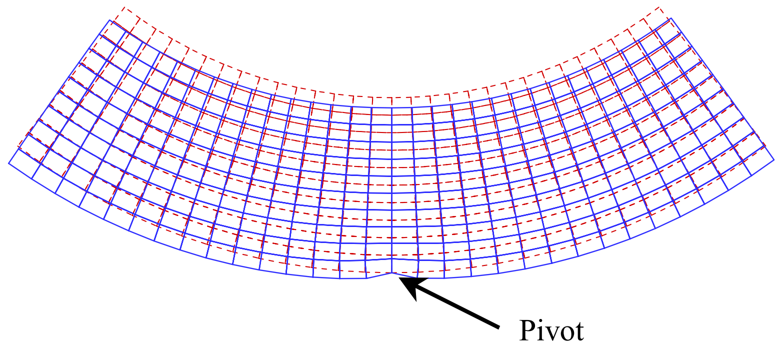
SOFTWARE HIGHLIGHTS

CRTSP2

*Increased maximum allowable stations to 400

ROTSTB

*Increased maximum allowable stations to 400



MAXBRG, Mechanical Deformation of a Tilting Pad

MAXBRG

*New Version 4.4

*Tapers at pads' trailing edges

*Temperature readings at specified pad locations

*Improved turbulence model

*Improved convergence

*Improved output format

*TECPLOT graphics for a pad's pressure, temperature and deformation

*Scripted editor version 4.4

*MAXPLT version 4.4

*Manual 4.4

THRUST

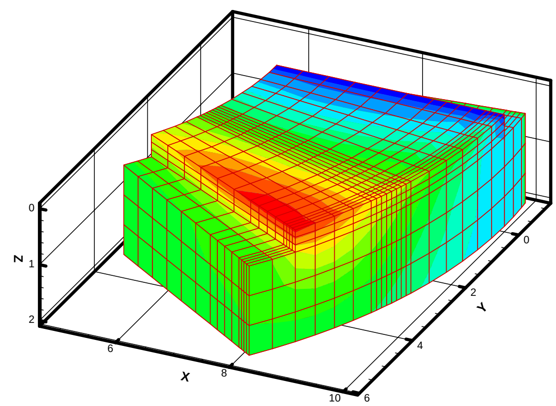
*Version 5.0

*Script editor 2.0

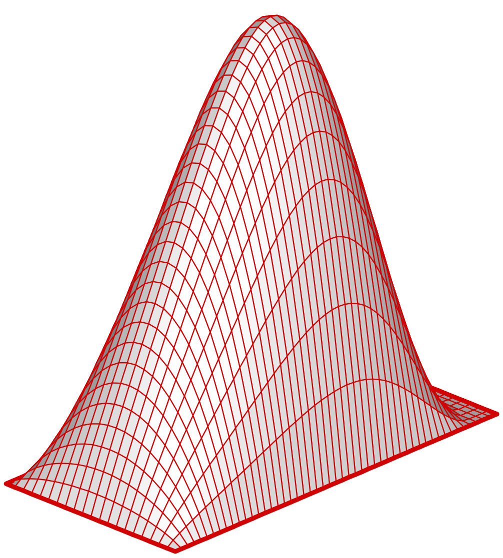
*Reformation of the thermal computation

*Improved convergence when conduction is included

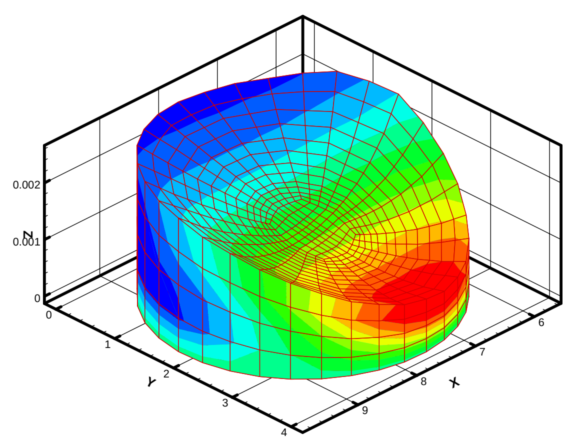
*New cylindrical crown



THRUST, Temperature of a Fixed Pad with OD Dam



MAXBRG, TECPLOT Pressure Graphics



THRUST, Temperature of a Circular Pad Film

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SOFTWARE HIGHLIGHTS

BALOPT

*Version 1.2

*New optimum stochastic method and the worst case min-max method

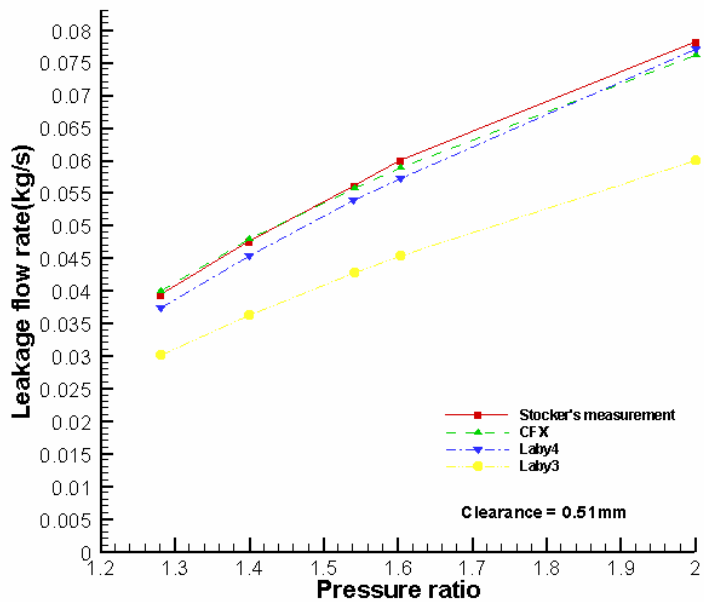
*New advanced discrete weight splitting function

*Solution of removing weight as well as adding weight

*Option to keep the previous trial weights

*Script editor 1.2

*Manual 1.2



LABY4, Comparisons of Leakage Predictions

<u>LABY4</u>

*Version 1.0

*For gas labyrinth seals

*Bulk flow theory based

*Three control volume method

*Teeth-on-Rotor and Teeth-on-Stator

*Extra terms for momentum exchange between control volumes

*Enhanced inlet, land and outlet boundary conditions obtained from CFD simulations

*User specified frequency

*Leakage and rotordynamic coefficients prediction

MAXBRG—New Version 4.4

Programs in Progress-

Rotordynamics Code and

Rolling Element Bearing Code

Programs in progress

<u>Rotordynamic code</u>

*Multiple rotors

*Coupled lateral-torsional-axial analysis

*Eigenvalue and steady state forced response analysis

*Finite element method based

*Time domain method for tilting pad bearings and flexible support

<u>Rolling element bearing code</u>

*Code calculates the Elastohydrodynamic lubrication in line contact problem

*The analysis is based on the coupled solution of the Reynolds equation for the lubricant behavior and the elastic deformation of the contact surfaces.

*Code inputs are: External load, surface speed, the surface geometry, material properties of the surface (Modulus of elasticity and Poisson's ratio) and the lubricant properties (Viscosity and density).

*Code outputs are: the pressure distribution over the contact length and the film thickness profile over the contact length.

HEARTPUMP

Heart Pump Activities

Students: Amer Azzam Al-Dhafiri, Isaac Cecil, Stephen Evans, Michael Sue-Ling,

Faculty: Houston Wood, Paul Allaire, Zongli Lin

Research Professionals: Wei Jiang, Alex Untariou

Funding: Mechanical and Aerospace Engineering, University of Florida

Objective:

As all of you know, the leading cause of death in the United States and some other countries is congestive heart failure. The University of Virginia has a long term effort to develop the next generation of compact, efficient, long life ventricular assist device. Basically it consists of an axial flow blood pump with magnetic suspension. The University of Virginia developed the world's first magnetically suspended ventricular assist pump in 1999. Now we are working with the Dr. Curt Tribble, Head of the Cardiovascular Surgery Dept. at the University of Florida to design, construct and test a new prototype. Below are some features of the new pump design.

Flow Path Redesign:

Because of new ideas the Romac team has implemented in magnetic suspension for the axial flow heart pump, the pump can be made smaller and will have fewer components. A render-

Figure 1

ing of the blood contact surfaces (excluding the diffuser) is shown below.

The activities pursued to develop the new pump are summarized here.

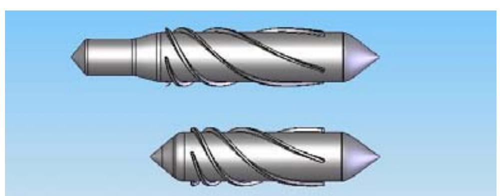


Figure 2– New impeller design which is shorter and has less number of blades then the previous design.

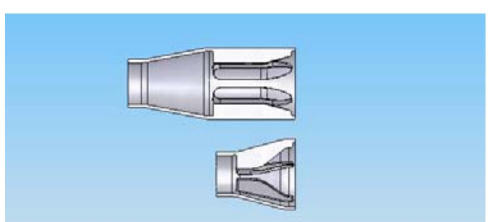


Figure 3: New inducer design

Design:

The new design flow rate was reduced from 6 lpm to about 4.5 lpm and the new flow range was also reduced form 2-10 lpm to 2 -8 lpm. Furthermore, based on the study conducted by Jiang Wei, the length of the 16.4 mm section of the impeller should be reduced to 55 - 52 mm. In addition, the need for the active magnetic bearing inside the inducer was eliminated. The new designs for the impeller and inducer are shown in Figures 1,2, and 3. Further evaluation will be done to see the effect of increasing the impeller diameter.

Features are:

77 mm tip to tip distance.

53 mm section with 16.4 mm diameter

72 degree leading edge curvature and 22 degree trailing edge curvature.

Magnetic Suspension and Motoring

The magnetic suspension involves the use of two radial active magnetic bearings (AMBs) with permanent magnet (PM) bias and self sensing, one axial PM thrust bearing, and one PM brushless DC motor. The radial AMBs with PM bias and self sensing are described in more detail under Self Sensing Magnetic Bearings.

MISCELLANEOUS

2007 ROMAC Annual Conference

Las Vegas, Nevada
June 10-13, 2007

The 2007 ROMAC Annual Conference will be held in Las Vegas, Nevada at the Rio All-Suite Hotel & Casino. They have offered us a room rate of $149.00 per single/double room. You can visit their website at:: www.riolasvegas.com

A Sunday afternoon reception will begin the conference on June 10. The meeting sessions will be Monday—Wednesday with breaks and lunches each day. There will also be a dinner banquet provided on Monday night. Members will be receiving their registration packet in March. If you should have any questions, contact Karen Marshall at (434)924-3292. Details are also available on our website at www.virginia.edu/romac/annualmeetings.htm

We hope to see you in Las Vegas!!

ROMAC Graphical User Interface

We were informed by Concepts in spring of 2006 that they were no longer interested in maintaining or improving the ROMAC RotorLab software in return for a ROMAC industrial membership. Mike Platt, who worked for Concepts and did most of the work on RotorLab, left and they stated that they had no one else to work on it. A new person would have to be trained and that would be expensive. They stated the RotorLab software had been developed by Concepts and that it belonged to Concept, not to ROMAC. They wanted to be paid separately by each user to continue to add new ROMAC codes to the RotorLab interface and continue to improve the package. They sent a letter to us proposing that a fee on the order of $2,000 per year from all of the ROMAC companies would be what they wanted. They also stated that the use of the current executable file we have, as of 2006, would continue to be available for use.

Many of you were at the ROMAC meeting last year when this subject was discussed. As it turned out, hardly any companies wanted to pay Concepts to continue to develop RotorLab. Only about 8 to 10 companies indicated that they actually use RotorLab. Most of the companies stated that they use the older ROMAC Shell. However, this is also copyrighted by Concepts. Thus, we have a significant problem – either pay Concepts quite a bit of money and still not own the ROMAC software (Concepts would still retain ownership) or move in a different direction.

Concepts offered to sell RotorLab in its existing form to ExxonMobil, who has invested significant funding in RotorLab in the past, for $75,000. Even if RotorLab were purchased by ExxonMobil, then someone would have pay for significant amounts of additional money for improvements over the years.

We at ROMAC and ExxonMobil discussed this issue with a world class expert on user friendly engineering software and graphical user interface materials, Professor Bob Ribando in Mechanical and Aerospace Engineering. He is writing a book on heat transfer for Prentice Hall using these methods and teaches courses with them. He states that when Concepts started to develop this software, it was a great deal of work. Now, with many advanced tools available such as Visual Basic, spreadsheets, and other materials, much of the hard work has been removed. He has agreed to develop a new ROMAC graphical input and results system for rotor dynamics analysis tied to ROMAC rotor dynamics codes as well as bearing and seal codes. The first version will appear by the end of 2007 and further advancements in coming years. This will require some ROMAC funding, some funding help from ExxonMobil that they have kindly offered contingent upon suitable progress, and help from other ROMAC personnel. It will be more advanced and easier to use than RotorLab. A major advantage is that this code will belong to us, not someone else, who can stop working on it at any time.

Software Key

As requested by ROMAC members, we are making progress in establishing a software key that will have to be employed to run ROMAC software. It will be put in place early in 2007 after consultation with a number of member companies.

MISCELLANEOUS

2007 ROMAC COMPANIES

1. Bechtel/BPMI
2. Bechtel/ Bettis*
3. Boeing Commercial Airplane Group
4. Concepts
5. Cooper
6. Curtiss-Wright/EMD
7. Siemens DEMAG Delaval
8. Dow Chemical Company
9. Dresser Rand
10. Duke Power Company*
11. General Dynamics/ Electric Boat*
12. ExxonMobil
13. AREVA
14. Flowserve
15. Innovative Power Solutions *
16. Kingsbury
17. Martin Marietta/KAPL
18. KOBE Steel
19. MSC Software*
20. Mitsubishi
21. ODS*
22. Petrobras
23. Renk*
24. Rotating Machinery Techonology
25. Rolls Royce Energy Systems
26. Shell
27. Solar Turbines
28. Statoil*
29. Sundyne
30. Turbocare
31. TCE
32. Waukesha Bearings

*Companies joining in 2006/7

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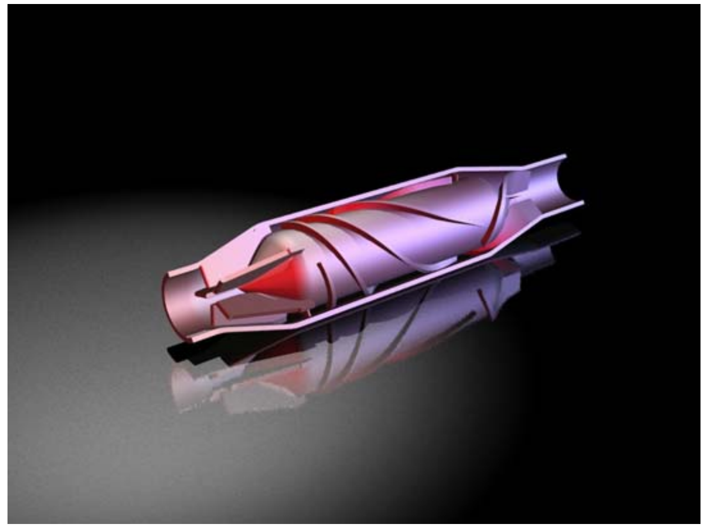
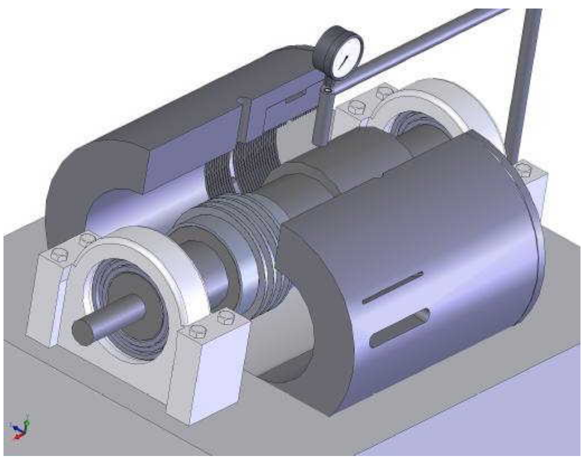
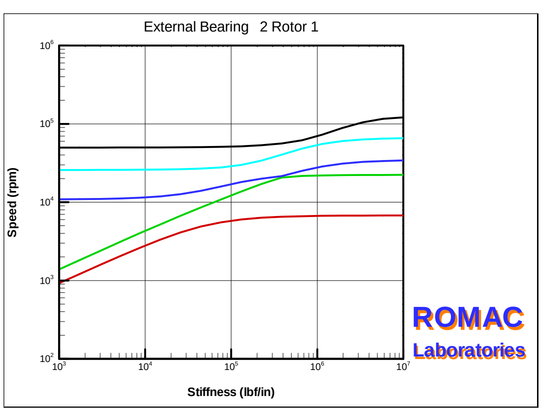


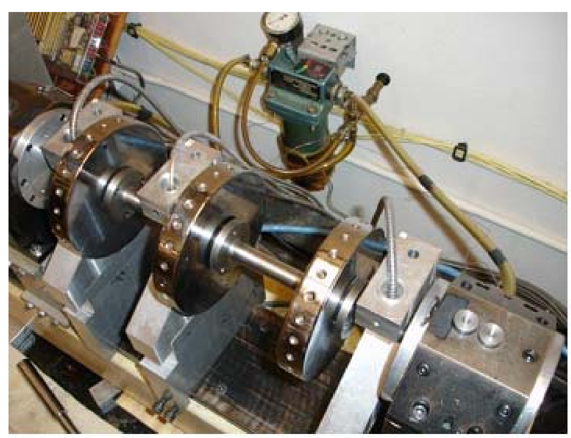
Artificial Heart Pump Design



Seal Test Rig



Rotor Critical Speeds vs. Bearing Stiffness



Three Mass Rotor for Balancing Experiments