Enhancing Our World's Energy Supply

SwRI engineers are developing new technology for subsea natural gas production

Magnetic bearings provide bearing support to the vertical rotor, eliminating the need for lubricants.

By J. Jeffrey Moore, Ph.D. and David L. Ransom, P.E.

With rising energy prices, new technologies are needed to produce natural gas from increasingly extreme environments. Many gas fields have remained untapped because of poor economic returns considering the construction costs of offshore platforms. However, developing a subsea natural gas compressor would permit harvesting these marginal gas fields without expensive platforms. Centrifugal compressors are most commonly used for this application. They must be designed to operate in a high-pressure, high-power environment (more than 1,000 psi and greater than 10,000 horsepower). Traditionally these compressors are driven by gas turbines or large electric motors. To minimize the subsea footprint, the compressor would be directly driven by a high-speed electric motor in a hermetically sealed casing suitable for the subsea environment.

In many applications, magnetic bearings support the rotor, eliminating the need for lubricants. They use powerful electromagnets coupled to a feedback control system and are well suited for a subsea application because they can operate in the pressurized hydrocarbon environment. The proposed design, developed by an SwRI client, uses a vertically oriented rotor that has several advantages in packaging and by minimizing contamination of the motor. If an upset of the magnetic bearing system occurs, the rotor will drop, or delevitate, onto auxiliary bearings, allowing the unit to be shut down without damaging the rotor or stator components. However, this transient delevitation has never been tested for a vertical rotor with three or more bearings.

A research program was jointly set up between the commercial client and Southwest Research Institute (SwRI) to investigate the rotordynamic behavior of a vertical rotor supported on magnetic bearings when the rotor is dropped onto auxiliary rolling element bearings. SwRI engineers designed and built a sub-scale test rig capable of delevitating the rotor from a speed of 30,000 rpm and performed experimental tests for a variety of operating conditions. As part of this study, SwRI researchers also developed a numerical tool to model the flexible rotor and stator dynamics as well as the auxiliary bearing characteristics.

Auxiliary bearings are critical components because they must endure multiple impacts and frictional contact forces associated with rapid acceleration, high rotational speed and radial load. They must operate in harsh temperature conditions with no lubrication. The lateral motion of the rotor is controlled by the auxiliary bearings, but a radial clearance, or deadband, exists between the rotor and the auxiliary bearings and complicates the dynamics. During the drop phase, the rotor may experience a lateral, self-excited vibration regime (backward or forward whirl), which can produce high dynamic loads on the bearings. In the case of a vertical-axis machine, the backward or forward whirl regime is even more likely to occur, because gravity loads will not contribute to stabilizing rotor position and prevent rotor whirl.

When the program started, auxiliary bearing design and overall machine behavior during the drop phase were identified as critical issues for robust machine operation, even in the case of active magnetic bearing failure. A second goal was to restart the machine after multiple delevitations without retrieving the entire unit from the seabed. Most delevitation technical knowledge has been developed for horizontal turbo-

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machinery. Only a few comparisons between predictions and measurements were performed for vertical units because they are less commonly used.

The SwRI team put in place a complete numerical and experimental approach to yield a predictive tool capable of analyzing prototype performance and drive-suitable design solutions. The numerical part of the task consisted of a MatLab® based code developed to predict rotor drop behavior through a nonlinear transient simulation, taking into account both rotor and housing flexibility and the auxiliary bearing's nonlinear dynamics. SwRI engineers validated this predictive tool by comparing its performance with experimental data obtained from the dedicated test rig at SwRI.

Experimental activity

SwRI engineers designed and built a test rig that represents a one-third scale motor/compressor. The rig consists of a three-bearing rotor suspended within a vertical casing. The primary bearings, which are commercially available active magnetic bearings (AMB), provide control of seven axes (six radial and one axial) at three bearing locations. The auxiliary bearings, also from a commercial supplier, are pre-loaded pairs of angular contact bearings and are supported in the radial direction by a damper ribbon, a wavy strip of spring material. To minimize cost, the test rig was built to reduced scale, but the rig exhibits behavior according to rotordynamic similitude, offering similar vibration modes and natural frequencies compared to the running speed. However, to accomplish this the test rig rotor must spin at up to three times the speed of the full-scale rotor, or about 30,000 rpm.

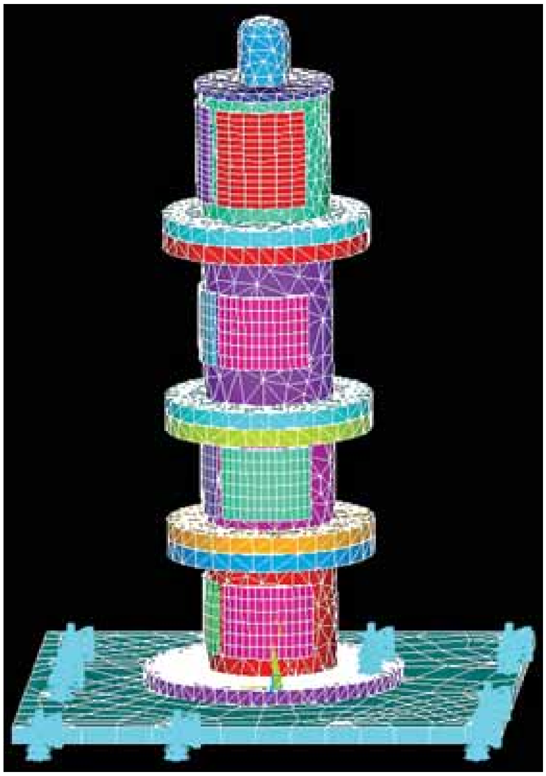
Because the scale model is designed for rotordynamic study, several of the full-scale components must be simulated by additional rotor mass. The actual motor core is simulated by a large dummy mass between the top and middle bearings. The compressor impellers are simulated with integral steel disks that have the appropriate scaled inertia properties.

The test rotor is driven by a 5.5-kilowatt induction electric motor using a variable-frequency drive connected via a flexible coupling to the top end of the motor-compressor assembly. This drive motor accelerates and decelerates the rotor in a controlled fashion, matching as closely as possible the anticipated rate of deceleration of the full-scale unit.

Instrumentation includes the AMB sensors at all three bearing locations (radial and axial), as well as two additional pairs of probes placed at rotor mid-span planes to define rotor orbits in these planes. Four velocity transducers, located on the casing at the upper and middle bearing planes, are used to detect housing vibration. The transient data is captured using a 24-bit data acquisition system that acquires all 16 channels in a continuous waveform at 12,000 samples per second. The test rig was placed in a concrete pit located in an SwRI laboratory and securely attached to the floor in the event of a catastrophic bearing failure. The motor, AMB and data acquisition system are controlled from a separate room, with two concrete walls providing additional protection.

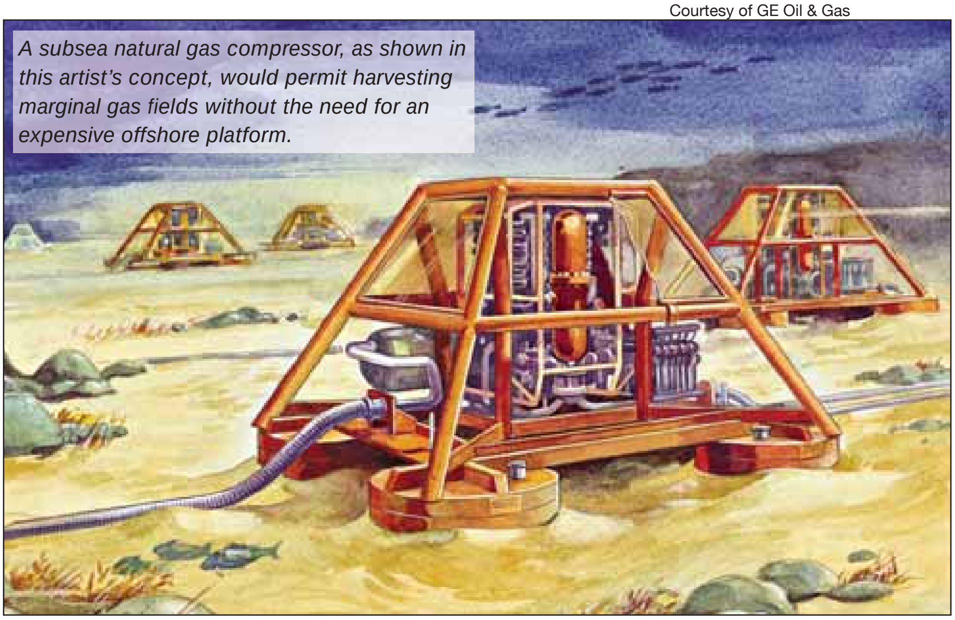
Numerical modeling

Several key characteristics determine the nature of this simulation. First, the landing event is time-transient because both speed and vibration are changing in time. The model must include nonlinear bearing supports because of the com-



This fully meshed finite element model of the test rig casing includes all non-rotating components, with more than 500,000 degrees of freedom.

Courtesy of GE Oil & Gas



A subsea natural gas compressor, as shown in this artist's concept, would permit harvesting marginal gas fields without the need for an expensive offshore platform.

bination of the dead-band clearance in the auxiliary bearings and the nonlinear stiffness of the combined angular contact bearing pair and the damper ribbon. Second, the geometry includes a flexible rotor and a casing with structural modes in the operating speed range. Typically, rotating machinery is designed to avoid structural modes within the operating speed range, but the vertical orientation makes this impractical. These features combine to require a fully flexible dynamic model of both the rotor and casing.

The finite element models are quite large, especially for the solution of a fully transient problem. This model includes all of the non-rotating components of the test rig, with more than 500,000 degrees of freedom (DOF). Although the finite element model of the rotating components is much smaller, less than 400 DOF, it is still much larger than those typically used in similar transient simulations. Most models have only 10 or 20 degrees of freedom, often without flexible rotor or structure models.

Using the process of component mode synthesis, which is an order reduction technique used for large finite element models, the rotor and casing models are reduced to only a few modes, while critical dynamic information, such as casing vibration modes, are maintained. These reduced models are then coupled by interface force equations instead of the traditional direct stiffness approach. The reduced-order model is simulated using an integration method in Matlab.

Central to the success of the simulation is the calculation of the interface forces between the various simulation components. The pair of preloaded angular contact bearings comes in direct contact with the rotating shaft. This bearing pair is mounted within a preloaded ribbon damper, mounted in a bearing housing that is bolted to the machine case.

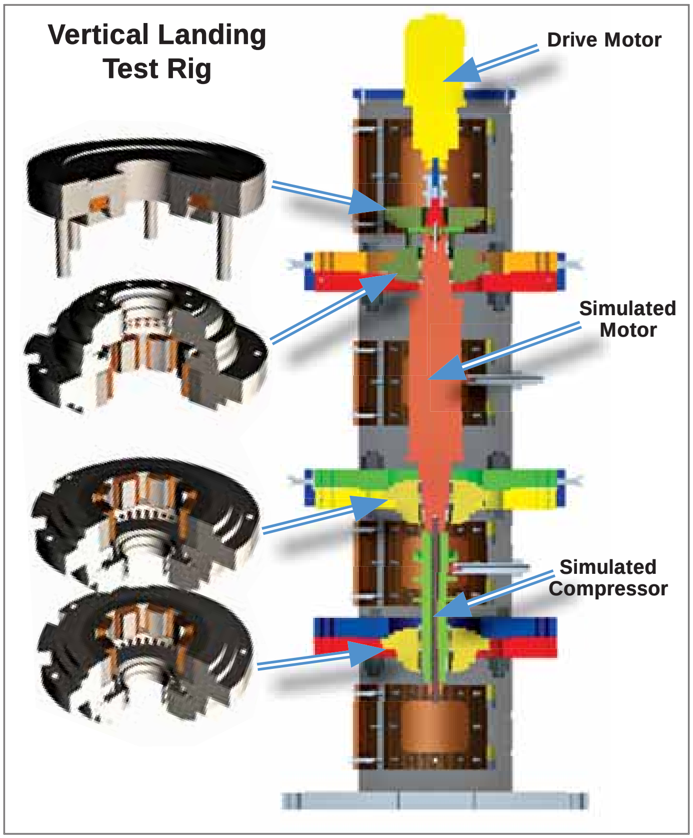
Three interface force locations are considered in this simulation. At position one there is direct contact between the rotor and the bearing inner race. This contact can result in both radial and tangential force transmission, and is represented with a Hertzian contact model. The contact results in the sudden acceleration of bearing inner race.

The interface forces at position two are determined from the angular contact bearing load/deflection curve. This curve includes the compliance of the angular contact bearing as well as additional compliance caused by the kinematics of the bearing preload technique. The third interface, position three, involves a somewhat bilinear stiffness, with radial and tangential components.

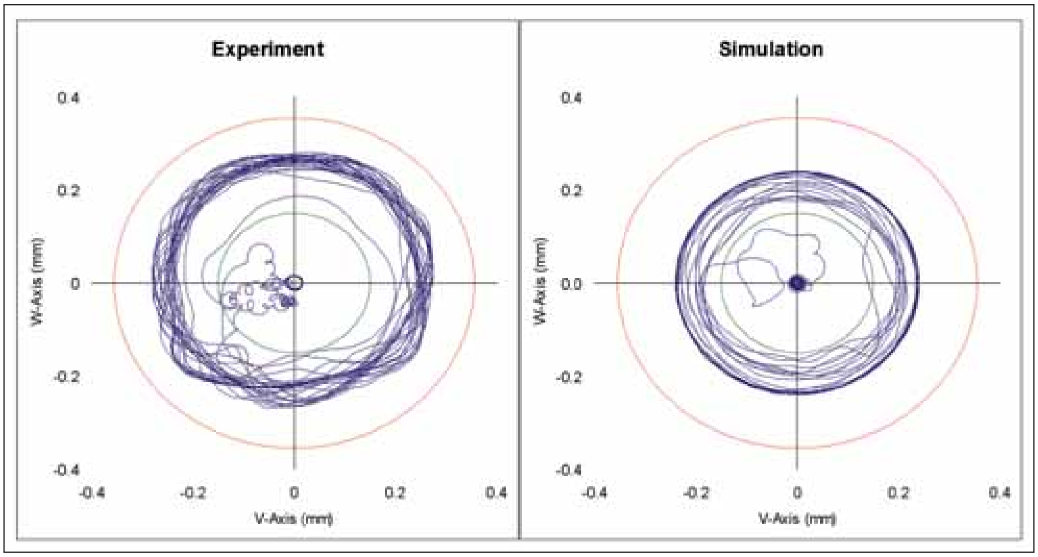
For the range of damper ribbon radial motion, the radial and tangential forces are calculated from a complex stiffness developed by the bearing manufacturer. A separate test program is being pursued to validate this critical model.

Significant findings

At the beginning of the program, the greatest unknowns revolved around the characteristics of rotor whirl in the vertical orientation. Prior to this work, the engineering community generally considered the whirl frequency, which is the rate at which the rotor precesses (not its rotational speed), to be determined by the mass of the rotor and the stiffness of the catcher-bearing support system. This is a fairly common approach in rotordynamics and has been verified repeatedly in more typical industrial machines such as high-power compressors supported on oil film bearings. For this test rig, the predicted whirl frequency using this simplified linear model is around 150 Hz. However, the rotor very clearly whirls at no more than 90 Hz and in most cases has shown whirl at around 68 Hz.



The SwRI-designed test rig represents a one-third scale of the compressor, and consists of a three-bearing rotor suspended within a vertical casing.

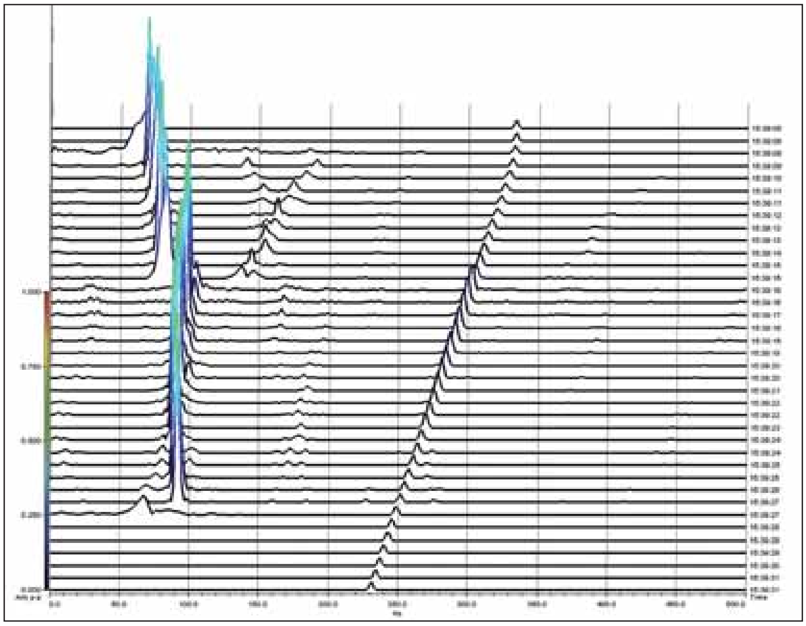


Measured and predicted vibration orbits showing forward whirling of the rotor during delevitation.

The synchronous response to residual shaft unbalance is shown as starting at 333 Hz (20,000 rpm). At about 300 Hz, the rotor passes through its own natural bending frequency, resulting in peak vibrations for this drop test.

From a machine-design perspective, the lower than predicted whirl frequency is good news as bearing loads decrease proportional to whirl frequency squared, and angular contact bearing life is strongly dependent on load. From a design-analysis perspective, the question then became whether the team could predict the real whirl frequency.

Another important experimental finding involves the direction of the whirl. Initial simulation results indicated the possibility of backward whirl. In such an event, the rotor would tend to roll inside the inner race, in a direction opposite the direction of spin. Typically, this is an undesirable condition that can lead to catastrophic damage if contact occurs between the rotor and stator outside the catcher bearings. However, in the complete series of drop tests performed (18 in all, most above 20,000 rpm), the rotor has proceeded directly into forward whirl. This finding led the team to search for the mysterious force that drives the rotor into forward whirl. This phenomenon has been observed by other researchers but has yet to be satisfactorily explained.



Waterfall vibration plot showing whirl frequency measured during delevitation of the rotor onto auxiliary bearings.

Future work

This forward whirl phenomenon is now a focal point of the research program, and researchers are investigating several explanations. These include correctly modeling the influence of residual unbalance, internal friction of the rotating components, and asymmetric contact at the axial bearing, driving the rotor into forward whirl. Understanding the source of this forward whirling force is important to the successful design and analysis of a full-scale machine.

In addition, the experimental program is continuing, with emphasis on establishing endurance limits for the angular contact bearings, thereby determining how many delevitations the bearings can accommodate. This program includes two series of identical drop tests in which the health of the angular contact bearings is monitored by a combination of vibration, sound and drag measurements. This information will be important in validating the design tools the bearing manufacturer used in predicting bearing life.

This research program is enabling SwRI engineers to use skills in mechanical design, rotordynamics, solid mechanics, linear control theory, nonlinear transient simulation and fabrication of a high-speed test rig. Its results closed many technology gaps, enabling the client to accelerate its product development while minimizing risk, all without consuming the client's in-house resources. The success of this program highlights the value that SwRI brings to clients to streamline their product development process. v

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