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DEMONSTRATION OF THE ROTORDYNAMIC EFFECTS OF CENTRIFUGAL LIQUID SEPARATION AND GAS COMPRESSION IN AN OIL-FREE INTEGRATED MOTOR-COMPRESSOR

Tim Griffin
Staff Rotor Dynamics Engineer
Dresser-Rand
Olean, NY, USA

William Maier
Principal Development Engineer
Dresser-Rand
Olean, NY, USA

ABSTRACT

The application of oil-free-integrated motor-compressors has become increasingly popular in recent years. One of the significant features of this class of machinery is compactness, providing space-savings compared to traditional-oil-lubricated compressors with associated gearboxes and lubrication systems.

The integration of a turbo separator with such a compressor has resulted in the creation of a new class of turbomachinery promising even greater system compactness. This new machine type provides further size reduction benefits through the elimination of large static separation vessels often required on traditional compressor trains.

A compressor manufacturer has successfully developed a centrifugal compressor with integrated turbo separator from design, development, and prototype testing on a demonstration rig through to manufacture, testing, and shipment of a production unit. This paper focuses on the details and results of the testing performed at the manufacturer's factory that confirmed the soundness and acceptability of the design. Rigorous testing of the demonstration rig has confirmed acceptable rotordynamic performance including stable operation over a wide range of operating pressures and liquid injection rates. The rotordynamic performance of this machinery type has been demonstrated to be virtually insensitive to liquid injection.

INTRODUCTION

Compact compression systems are playing an increasingly important role in the oil and gas industry as the compactness and relatively low weight can not only provide a significant cost savings for such applications as offshore oil and gas production but can also be an enabling feature for many projects where the compression system for an application must fit within a specific envelope which defines the remaining available space in a brownfield development. Oil-free integrated motor-compressors are a class of turbomachinery offering reliable operation with a compact footprint. The use of active magnetic bearings and a high speed motor close-coupled to a centrifugal compressor provides the opportunity to do away with traditional dry gas seals, the lube oil system (and all oil), and a speed increaser. This design approach results in a very compact, power-dense compression solution.

A turbo separator is another design feature that can be closely integrated with the centrifugal compressor to further reduce the traditional footprint required for a compression system. The application of centrifugal compression systems traditionally requires a large static separator vessel to be located upstream of the compressor to remove liquids upstream of the compressor. The weight and footprint of such traditional static separators can be quite large (especially for high pressure applications). Integrating a turbo separator into the compressor enables a further significant reduction in size and weight for the overall compression system by eliminating the need for the upstream scrubber.

Turbo separation is a relatively new technology. Like cyclonic separation, a turbo separator uses centrifugal force to augment density-based gas-liquid separation, but here the flow passages are rotating along with the fluids. This approach has the potential to greatly reduce near wall shear forces and thus minimize liquid re-entrainment, even at much higher fluid velocities. Commercial development of rotating centrifugal separator turbines for oil and gas applications has been pioneered by this OEM. Ross et al. (2001) and Rawlins and Ross (2001) published descriptions of earlier designs and field test results. Figure 1 is a representative drawing of a rotating centrifugal separator turbine.

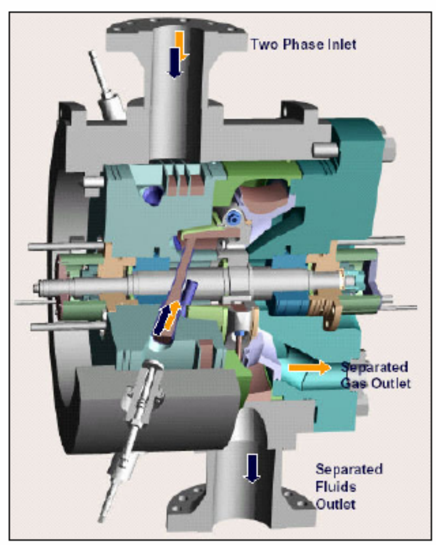

Figure 1. Two Phase Rotary Separator Turbine Layout after Oxley et al. (2003)

The development and testing of a self-powered version, the so-called IRIS®, is reported by Rawlins and Ting (2002). Figure 2 shows a typical field installation of an IRIS® rotating centrifugal separator.

Figure 2: IRIS® Rotary Separator

More recent developments of this technology are reported in Chochua and Maier (2007), Chochua et al. (2008), and Maier et al (2010).

The culmination of this effort is an advanced turbo separator suitable for integration with compact centrifugal compressors. Figure 3 shows a schematic of the turbo separator incorporated into a multistage centrifugal compressor.

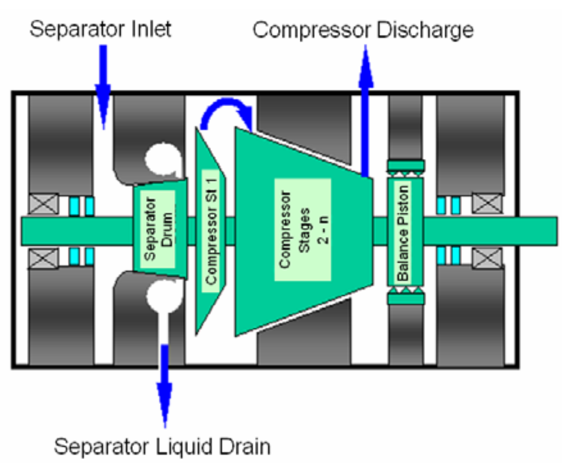

Figure 3: Schematic of Integrated Rotating Centrifugal Separator-Compressor.

The advantage in terms of compactness is clearly shown in Figure 4, a graph of the separation efficiency versus Souders-Brown factor for various separation technologies.

As shown in equation (1), this factor k is proportional to the characteristic through velocity of the separator U_G.

$$k = \frac{U_G}{\sqrt{\frac{(\rho_L - \rho_G)}{\rho_G}}} \qquad \textbf{(1)}$$

It can be seen that the turbo separator maintains good separation efficiency at velocities that are orders of magnitude greater than traditional static separation technologies. This higher through velocity translates directly into much more compact sizes for the same application.

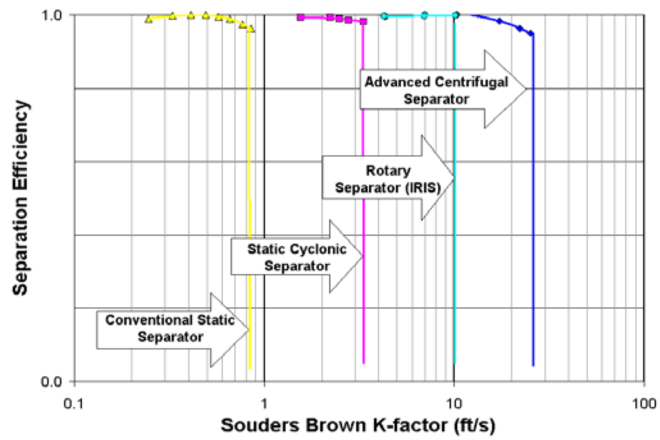

Figure 4. Comparison of Separation Technologies

An extensive amount of research and field troubleshooting has been directed at understanding, predicting, and eliminating self-excited vibrations associated with fluids in turbomachinery. Muszyńska (2005) provides a historical background for research on oil whirl and whip phenomena in journal bearings going back to Haag (1946) and Hull (1958). Ehrich (1967) and Wolf (1968) provided some of the early work investigating the effect of trapped fluid in turbomachinery rotors. Previous work by Brenne et al (2005) also documented the rotordynamic effect of operating a centrifugal compressor under wet gas conditions. It was noted that at high liquid flow rates some subsynchronous vibration components appeared in the vibration spectrum that were attributed to potential instability.

Designing a turbomachine to separate liquids from gases using a turbo separator must be undertaken with consideration to these potential concerns regarding self-excited vibrations. This turbomachine has a unique design that has been employed with features that help to mitigate the potential for destabilizing liquid-rotor interaction. This paper addresses aspects of the design employed to address rotordynamic concerns associated with handling fluids in a separator / compressor and documents the successful testing of the demonstration rig that helped validate the successful design.

FLUID-INDUCED INSTABILITIES – KEY PARAMETERS

Fluid-induced instabilities occur in turbomachines as a result of the forces associated with the fluid adjacent to the rotor having an average velocity in the circumferential direction that is different from the speed of the rotor surface. For a fluid film journal bearing, the rotating shaft imparts energy to the lubricating oil which can lead to a strong circumferential flow as described by Muszyńska (2005). The strength of this circumferential flow directly impacts the tendency of the flow to contribute to instability. Thus by minimizing the strength of the circumferential flow by using non-cylindrical geometry (such as a multi-lobe sleeve bearing or a tilt-pad bearing) it is possible to help prevent instability.

This idea of reducing the strength of the circumferential flow also translates to fluid acting on the inside of a rotor. Research by Preussner and Kollmann (1988) studied the effect of circumferential segmentation of a liquid filled cavity. Figure 5 highlights the improvement in rotor stability afforded by circumferential sectoring of the liquid filled space on a rotor system. In the figure β is the non-dimensional rotational rotor speed given in equation (2).

$$\beta = \frac{\omega}{\sqrt{c/(\rho_L \cdot a^3)}} \quad \textbf{(2)}$$

The abscissa, b, of Fig. 5 is the ratio of the free liquid surface radius to the cavity outer radius. As can be seen in this figure the lower unstable region associated with the non-sectored cylindrical cavity results in potential instability across the entire range of fluid film thicknesses on the inside of the cylinder. By simply segmenting the cylinder (represented by the upper unstable regions) this significantly reduces the range of fluid film thicknesses that can result in instability. This segmenting of the fluid containing rotor limits the ability of the fluid to whirl with a strong circumferential velocity.

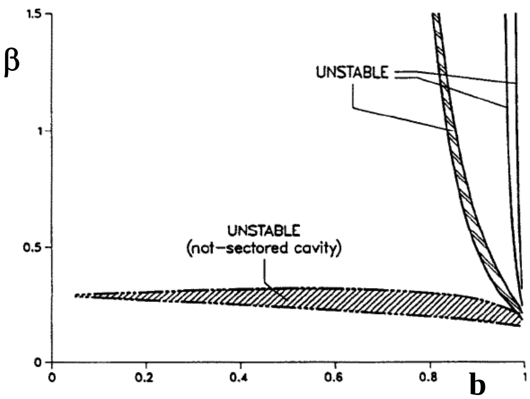

Figure 5: Theoretical Effect of Segmentation of Liquid Filled Cavity on Rotor Stability Zones (after Preussner and Kollmann (1988))

Ehrich (1992) noted that adding drain holes to a cavity that might otherwise tend to trap fluids is one way to prevent classic trapped fluid instability. He also noted that operating a rotor below its first bending mode is an additional means to prevent such instabilities. Ehrich (1967) also derived an equation for the rotor asynchronous whirl frequency for a cylinder with trapped liquid shown in Equation (3) from which it can be seen that as the liquid mass 'm' approaches zero the whirl frequency equals the dry rotor's natural frequency. As the mass of the liquid increases, the asynchronous whirl frequency 'Ω' (the natural frequency of the combined rotor-fluid system) becomes a fraction of the dry rotor's natural frequency. Thus the liquid in the cylinder has a mass effect that affects the system natural frequencies.

$$\Omega = \frac{\omega_n}{\sqrt{1+(m/2M)}} \quad \textbf{(3)}$$

SEPARATOR DESIGN

In line with the findings of these studies on trapped liquids, several features were incorporated into the turbo separator design specifically to minimize rotordynamic effects. These include monotonic sloping of the separator drum inner surface to ensure that liquid is driven to the liquid exit at the trailing edge of the drum, significant circumferential partitioning of the separator rotor to minimize the strength of circumferential liquid flow that could otherwise be destabilizing, limiting the rotor to subcritical operation, and designing the liquid collector to prevent pooling of liquids around the OD of the separator drum. Figure 6 schematically shows the associated geometric features of the separator drum.

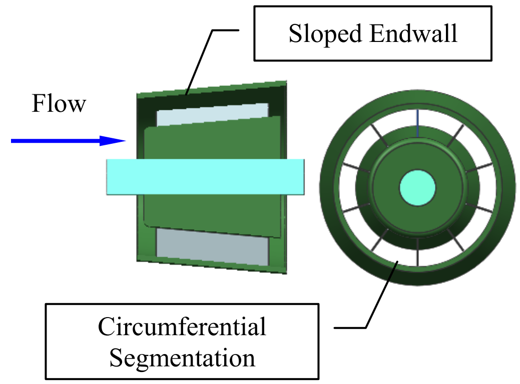

Figure 6: Schematic of Separator Drum Features Intended to Limit Rotordynamic Impact.

DEMONSTRATION RIG

The demonstration rig hardware shown in Figure 7 utilizes a beam-style rotor configuration with the casing designed to incorporate a liquid drain. The rotor includes the gas/liquid separator incorporated within a single stage of compression. The rotor is supported on tilt-pad bearings and has a critical speed associated with the first forward bending mode of approximately 18,000 rpm (above the maximum test operating speed of 13,500 rpm).

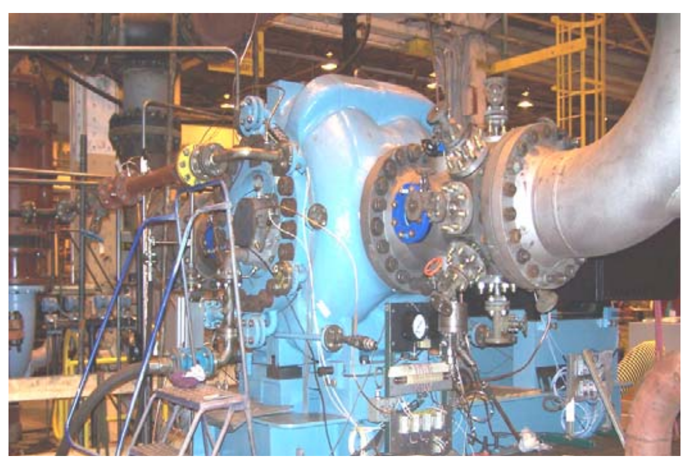

Figure 7: Demonstration Rig

The demonstration rig setup also included a Magnetic Bearing Exciter (MBE) to provide asynchronous perturbations of the rotor at various operating conditions. Exciter testing was conducted to determine the logarithmic decrement (log dec) of the rotor-bearing system. The MBE was used to apply forces to the rotor to allow response measurement for the purpose of identifying the log dec of the rotor during operation, using the same concept as previously done at the authors' company (Moore, et al. (2002) , Moore and Soulas (2003)) and as further described in the next section. The MBE is mounted on one end of the compressor shaft (see Figures 8 and 9). The exciter is driven by an open-loop control system to introduce forces of varying frequency (asynchronous excitation). The outcome is an assessment of the rotordynamic stability at various operating conditions with increasing rates of liquid flow to the separator/compressor. This is accomplished via the log dec measurement of the first forward whirling mode.

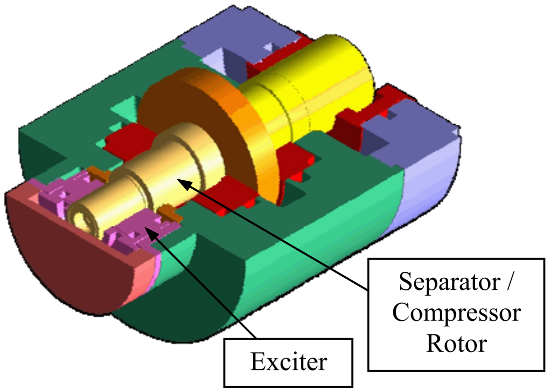

Figure 8: Solid Model of the Magnetic Bearing Exciter

Figure 9: Magnetic Bearing Used for the Excitation

CALCULATING THE LOGARITHMIC DECREMENT

A rotor can be modeled using techniques resulting in the general linear system of differential equations (Equation (4)),

$$[M]\ddot{X} + [C]\dot{X} + [K]X = F(t) \quad \textbf{(4)}$$

For the homogeneous solution (free vibration), a harmonic solution is assumed as Equation (5),

$$X(t) = \overline{X}\, e^{\lambda t} \quad \textbf{(5)}$$

The eigenvalue may be solved for and takes the form of Equation (6),

$$\lambda = -\zeta\,\omega_n + \omega_d\, i \quad \textbf{(6)}$$

The real part of the eigenvalue determines the level of damping or stability, where ζ is the damping ratio. The logarithmic decrement (δ) is another common way to state the level of damping in a system and is related to the damping ratio by Equation (7),

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \quad \textbf{(7)}$$

Notice the log dec is not defined for damping ratios (ζ) equal to and greater than one.

The log dec can be measured using the phase slope method to identify the point of maximum phase slope '$P'_{\max}$'. The amplification factor 'AF' is calculated using the maximum phase slope and the frequency 'ω_n' associated of the maximum phase slope as shown in Equation (8). The log dec 'δ' is related to the amplification factor as shown in Equation (9).

$$AF = \frac{\pi}{360}\omega_n P'_{\max} \quad \textbf{(8)}$$

$$\delta = \frac{\pi}{AF\sqrt{1-\dfrac{1}{4AF^2}}} \quad \textbf{(9)}$$

DEMONSTRATION RIG TEST RESULTS

Rig testing included a series of tests with varying inlet gas pressures, operating speeds, gas flow rates, and liquid flow rates. The measured vibration of the compressor was stable during all testing. Figure 10 shows a spectrum plot of the shaft displacement in a dry operating condition. Figure 11 shows a corresponding spectrum plot while a liquid flow rate of 214.2 gpm (liquid/gas mass ratio (LGMR) of 0.39) was being supplied to the inlet of the machine. Note that the overall vibration amplitudes are very low and that spectrum shows only a small 1X component (suggesting stable operation with a very small amount of residual unbalance). No other significant frequency content is seen.

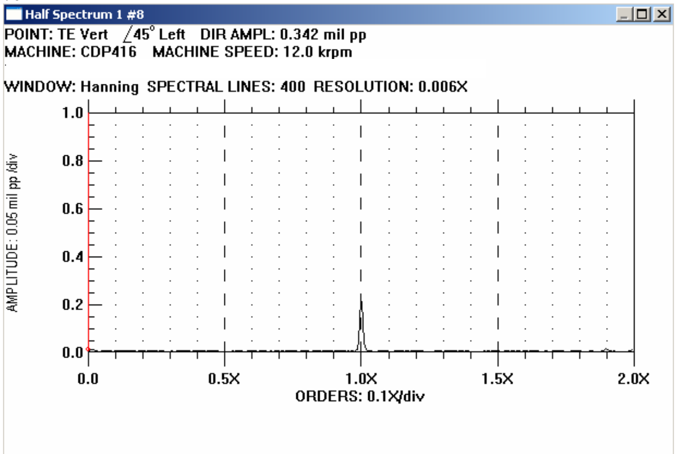

Figure 10: Vibration Spectrum of Demonstration Rig with No Liquid Injection, Inlet Pressure of 550 psi, and Operating Speed of 12,000 rpm

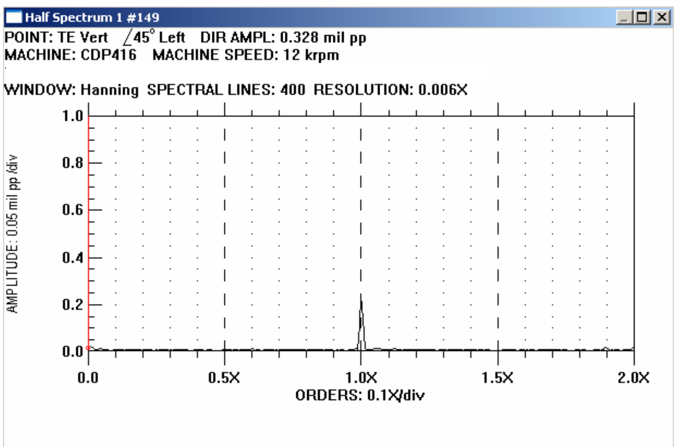

Figure 11: Vibration Spectrum of Demonstration Rig with 214.2 gpm (LGMR = 0.39) of Liquid Injection, Inlet Pressure of 550 psi, and Operating Speed of 12,000 rpm

The magnetic bearing exciter was used to collect the vibration response data associated with an asynchronous excitation imposed on the non-drive end of the compressor shaft via the magnetic bearing exciter. A sample plot showing the asynchronous response of the drive end of the separator/compressor rig shaft with an excitation imposed using the exciter located on the non-drive end is shown in Figure 12. The log decrement of the first forward bending mode was measured using the phase slope technique for a variety of test conditions with the goal of determining the effect of increasing rates of liquid flow on the measured log dec values providing an indication of the effect of the liquid injection on the stability of the unit.

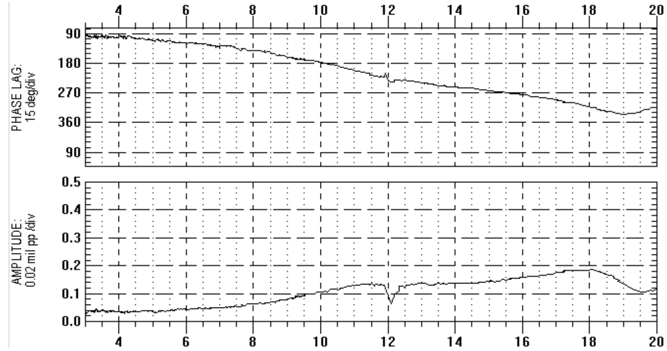

Figure 12: MBE Asynchronous Sweep at P_{inlet} = 550 psi: Vibration Amplitude & Phase Angle vs. Excitation Frequency (kcpm)

Figure 13 shows the variation of log dec values for a variety of test points at a compressor running speed of 12,000 rpm with the liquid/gas mass ratio plotted on the x axis. The trends for measured log dec variation with increasing liquid/gas mass ratio showed little change or a slight increase suggesting a neutral or slightly positive impact on stability associated with increasing liquid injection.

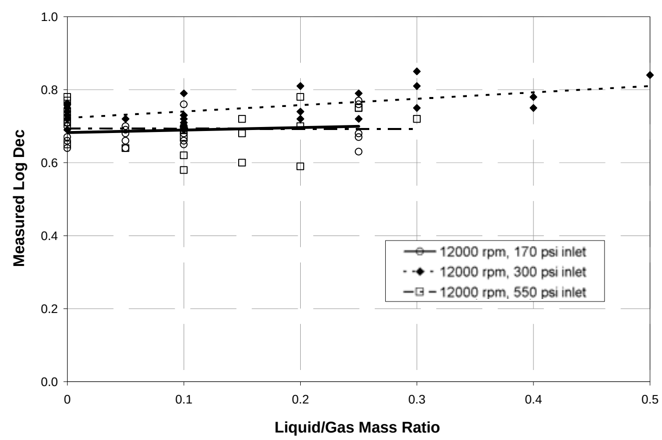

Figure 13: Measured Log Decrement Values for the Demonstration Rig Operating at 12,000 rpm

Figure 14 shows a similar trend with a log dec values that change little or increase slightly with increasing liquid/gas mass ratio for tests with the demonstration rig operating at 550 psi inlet pressure and 12,000 rpm and 13,500 rpm operating speeds.

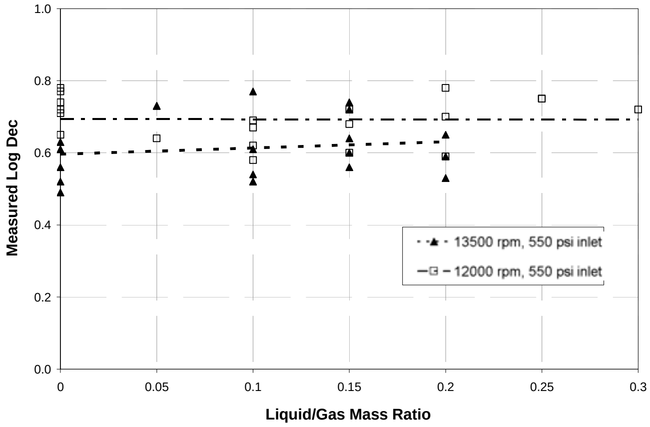

Figure 14: Measured Log Decrement Values for the Demonstration Rig Operating at 13,500 rpm

A test was also performed with a sudden increase in liquid flow rate to determine the sensitivity of the separator/compressor vibrations to a sudden change in the liquid flow rate. Figure 15 provides the results of this test demonstrating steady vibration as the liquid flow rate was increased rapidly from 0 gpm to 163 gpm and then suddenly decreased back to 0 gpm.

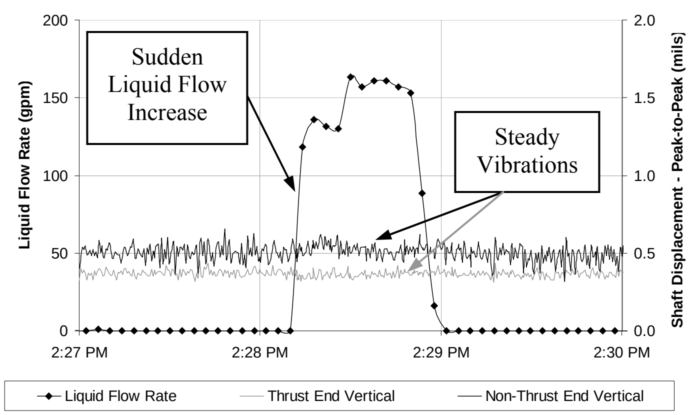

Figure 15: Vibration Trends for Test with Sudden Liquid Rate Increase

CONCLUSIONS

Testing has been successfully performed on a demonstration rig that validates the rotordynamic acceptability of the separator / compressor design operating in both dry and wet conditions.

NOMENCLATURE

a	= Liquid cavity outer radius
AF	= Amplification Factor
b	= Non-dimensional radius of liquid free surface
c	= Bending rigidity of shaft
[C]	= Damping matrix
i	= $\sqrt{(-1)}$
F(t)	= Force vector
K	=Souders-Brown factor
[K]	= Stiffness matrix
m	= Mass of trapped liquid inside a rotor
M	= Mass of rotor
[M]	= Mass matrix
P'_{max}	= Maximum phase slope
P_{inlet}	= Inlet pressure
t	= Time
U_G	=Gas superficial velocity
X	= Displacement vector
$\dot{X}$	= Velocity vector
$\ddot{X}$	= Acceleration vector
β	= Non-dimensional rotational rotor speed
δ	= Logarithmic decrement
λ	= Complex eigenvalue
ρ_L	= Liquid density
ρ_G	= Gas density
ω	= Angular velocity of rotor
ω_d	= Damped natural frequency (imaginary part of complex eigenvalue)
ω_n	= Undamped natural frequency
Ω	= Asynchronous whirl frequency

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