

## HIGH SUB-SYNCHRONOUS VIBRATION

### HOW TO ADDRESS VIBRATION IN A TURBOEXPANDER EQUIPPED WITH ACTIVE MAGNETIC BEARINGS

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**T**urboexpanders (TEX) are standard in the natural gas industry for liquefaction and dew point control. They are also used in the petrochemical, air separation, refrigeration and power generation industries.

The TEX expansion stage consists of a radial inflow turbine, often with variable-position inlet guide vanes. The compression stage is comprised of a centrifugal compressor stage with a vaneless diffuser.

Many traditional machines featured hydrodynamic oil bearings for supporting and controlling vibration. Active magnetic bearing (AMB) technology, offering operational and environmental advantages (Figure 1), was recently introduced in newer machines.

Forces can be induced due to cross-coupled stiffness by aerodynamic interactions between rotating and stationary components. If high enough, this

may induce unstable vibrations that impact the mechanical performance of the machine and even lead to failure.

The inlet of a typical expander is located downstream of the gas-liquid separator thus, the expander inlet gas is saturated. This impacts the thermodynamic state of the TEX inlet. The gas composition commonly exhibits a retrograde dew point. A retrograde dew point is a point on the vapor-liquid equilibrium line in a phase diagram in which a decrease in temperature or pressure results in condensing of the gas, ultimately leading to liquids entering the TEX (Figures 2 and 3).

The inlet gas is accelerated through the inlet guide vanes (IGVs) leading to high swirl or tangential velocity. This swirling gas enters the expander wheel, which is ideally spinning fast enough such that its blade tip speed matches the

gas swirl velocity.

The work performed by the gas is absorbed by the compressor, thereby losing angular momentum as it travels through the expander wheel. This power balance occurs at a speed based on the design and sizing of the expander and compressor wheels.

Most TEXs are designed with a 50% reaction turbine. Accordingly, the expander wheel usually has a high velocity, two-phase fluid surrounding its outer diameter. The aerodynamic forces on the rotor due to this complex flow are not well understood, particularly regarding their effects on lateral rotordynamics.

#### Troubled machine start-up

Take the case of a TEX that was unable to reach design speed because it suffered several trips on high sub-synchronous vibration (~40-50% of the rotation speed)

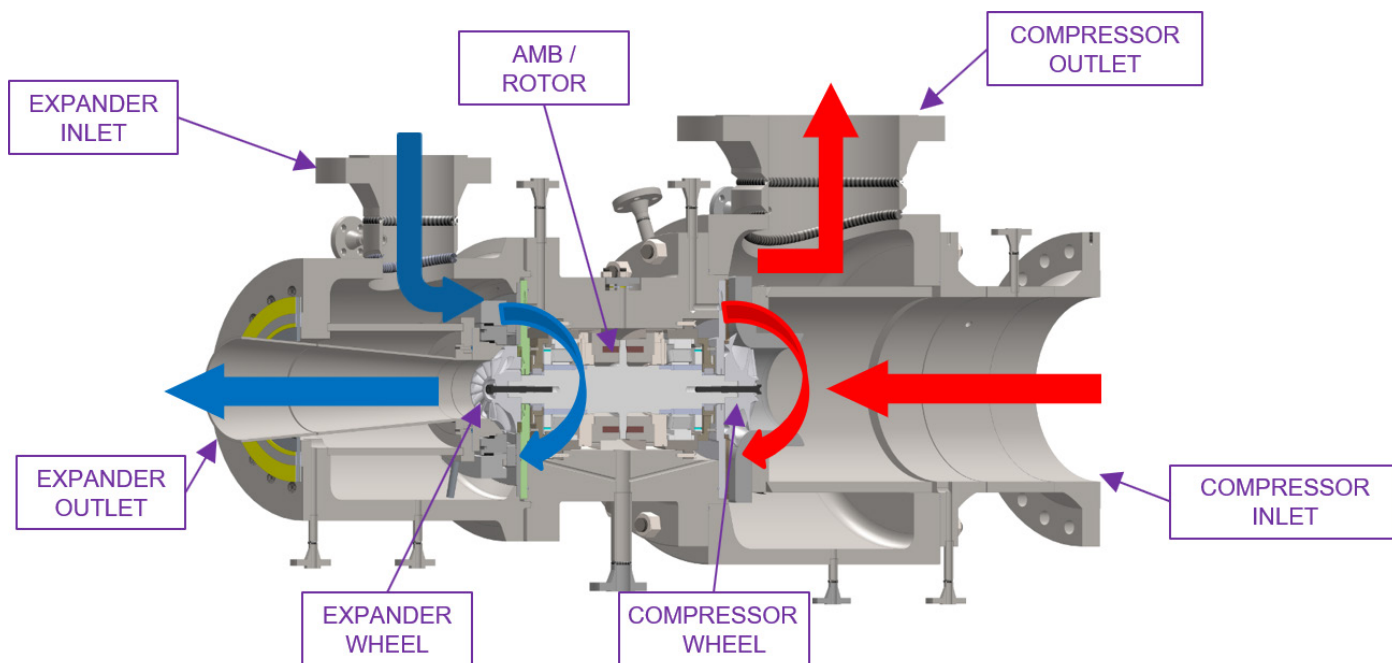
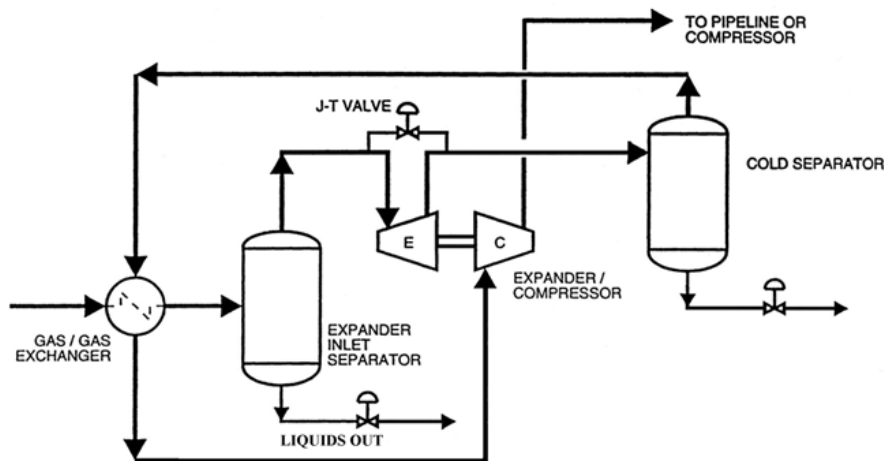


Figure 1: Typical turboexpander design equipped with active magnetic bearings



**Figure 2: Typical natural gas liquefaction process**

at a speed of around 15,000 rpm. The amplitude of vibration on the expander side bearing peaked to as high as 4 mils while the trip level was set at 3 mils.

At first, it was thought that reducing the amount of liquids present around the wheel outer diameter (OD), as well as the swirl velocity, would alleviate the problem. The theory was that liquid was being trapped in the wheel cavity causing the instability.

An attempt was made to operate the machine with the IGV in almost fully open position, using the expander inlet valve to attempt to regulate flow. However, the machine tripped again due to high sub-synchronous vibration.

It was taken offline and removed for teardown and inspection. The seal was cleaned and the clearances opened to help evacuate trapped liquids from the wheel cavity. The machine was then re-assembled and tested using different ramp rates. These attempts were unsuccessful as the machine was only stable for a short period of time before it would trip.

Various other unsuccessful remedies included ensuring dry gas was supplied to the expander inlet, adding heat tracing and insulation, and revising the design and sizing of the inlet separator. Operational improvements brought process conditions closer to design conditions.

However, the machine kept tripping due to high sub-synchronous vibration, now at ~21,500 rpm. Additionally, the AMB supplier maximized the stiffness and damping of the AMBs by modifying the controller. The unit could now reach ~25,000 rpm before tripping again (still falling short of design capacity).

The only recourse left was to reduce the cross-coupling effects by means of an invasive re-design to reduce liquid formation after the inlet guide vanes, reduce

swirl velocity and add damping at seal locations.

The first two goals are best achieved via an aerodynamic re-design of the expander wheel. A shroudless wheel was desirable due to its superior aerodynamic performance. Whatever advantages are afforded by a shroudless wheel, they could not outweigh the risks of a structural failure. In such a high visibility situation, a shrouded wheel was selected.

High density and high swirl velocity were thought to be the key contributors to cross-coupling effects, i.e., the amount of liquid leads to an increase in density. Increasing the pressure at the wheel outer diameter, then, would have the largest benefit (a reduction in cross-coupled stiffness), and a shrouded wheel would maximize this benefit.

The OEM predicted the redesigned expander wheel would increase wheel inlet pressure by ~20%, increase wheel

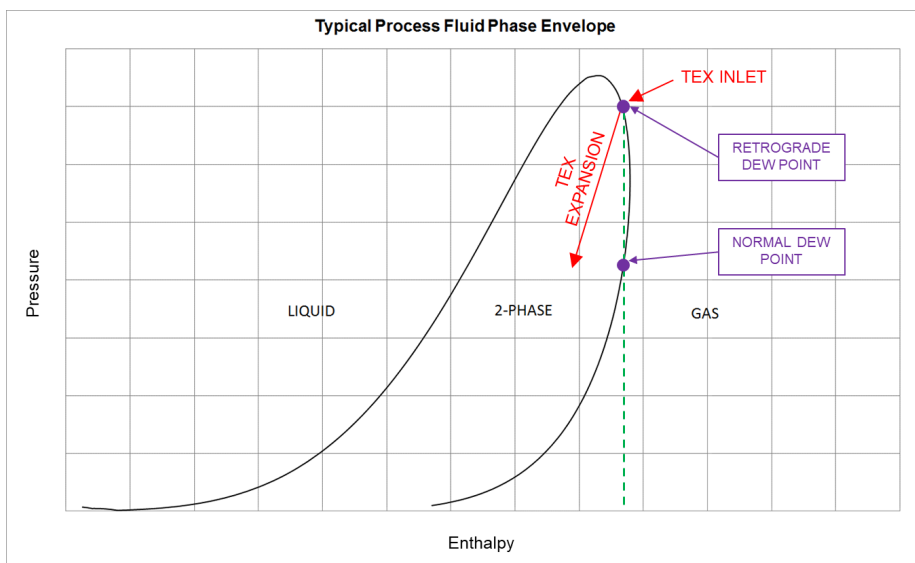
inlet density by ~15%, reduce liquid content by ~14%, and reduce swirl velocity by ~13%.

The third goal of the re-design was to add damping at the seals in the expander wheel. Shrouded wheels are usually equipped with two seal areas, one in the back and one in the front. The original labyrinth seals were replaced by pocket damper seals (PDS) (Figure 4).

They introduced a higher direct damping coefficient (which attenuates vibrations), as well as a reduction in cross-coupled stiffness (to eliminate sub-synchronous instabilities). The PDS features narrow-width rectangular cavities in which gas pockets form. The rotating shaft and pressure pulsations inside the pocket tend to oppose rotor vibration. Also, partition walls in the pockets obstruct gas swirl.

This type of seal operates at an increased clearance and thus suffers from increased seal leakage, leading to reduced wheel efficiency. However, the pocket-damper seal design offered more damping than a labyrinth seal. In addition, process engineers were confident that the expected reduction in expander efficiency would not jeopardize production requirements.

A rotordynamics analysis suggested PDS seals had the potential to alleviate sub-synchronous vibration. The redesign included: an expander wheel with reduced discharge flow area (maximized the pressure at the outer diameter of the expander wheel to reduce swirl velocity); pocket damper seals (reduced cross-coupled stiffness and added direct damping); and PDS equipped with integral swirl brakes. Due to these modifications, the TEX could operate at full capacity with negligible vibration.



**Figure 3: Typical natural gas phase diagram**

The root cause for the high sub-synchronous vibration problem stemmed from aerodynamic cross-coupled forces induced by the expander wheel and labyrinth seals. This exceeded the stabilizing capabilities of the AMB system.

These forces are poorly understood, and the lack of analytical tools and experimental data pose a challenge for

TEX designers. Currently designers must rely on empirical relationships and rules of thumb to predict these forces.

Recently, a qualitative approach was suggested to provide guidance for TEX design. That approach is covered in a paper presented at the 46th Turbomachinery Symposium, 2019, "Addressing High Sub-Synchronous Vibrations in a

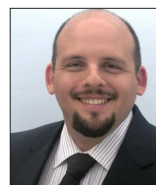
Turboexpander Equipped with Active Magnetic Bearings," by Avetian, T.; Rodriguez, L.E.; and Park, J. ■



Figure 4: Redesigned front wheel seals



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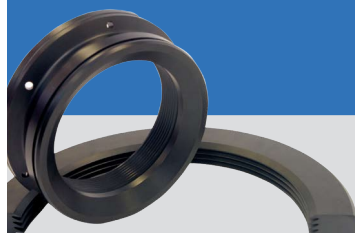
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