IMPORTANCE OF LUBE REDUCTION

A REDESIGNED OIL RECOVERY SYSTEM AND AN ATTENTION TO SYSTEM-WIDE DETAILS, SLOAN TARGETS OVER-LUBRICATION IN GAS COMPRESSION

A Sloan Oil Recovery System With Custom Ceiling Mount Bracket

BY ERIC SLOAN

Sloan Lubrication Systems (Sloan) has made it a mission to solve the pervasive problem of over-lubrication in the gas compression industry. Its TriCip system, which enables up to a 90% reduction in lubricant consumption using an automated control system and a proprietary lubricant, builds on decades of success Sloan has had with its Watchman lubrication system. The Watchman enables up to a 50% lubricant reduction by optimizing the flow rates to each delivery point and providing fail-safe monitoring. Rounding out Sloan's lube reduction offerings is its recently redesigned oil recovery system (ORS).

ORS is designed to prevent compressor crankcase oil waste. In most slow-speed gas compressor engines, some of the crankcase oil makes it past the wipers and into the distance piece. Excess oil flowing through the rod packing may also contribute significantly to oil waste. This oil is normally piped into a waste oil tank and disposed of. The ORS collects and filters the oil and returns it to the crankcase.

"The original ORS system was designed for doghouse recovery for an old slow-speed compressor," said Matt McCarthy, senior sales representative at Sloan. The operating conditions were not particularly challenging. The redesign was to make the system more robust, reliable, and capable of meeting the requirements for a high-speed unit like an Ariel."

According to McCarthy, ORS has been redesigned from scratch. "It has a new powder-coated tank designed specifically for the application with a large access and clean out. It has a higher-quality pump, a new water-absorbing filter, level switch, and all stainless components. More importantly, the redesign increased the ORS operating pressure to 300 psi [20.6 bar]."

The daily amount of oil collected per compressor is typically between 2 and 4 gallons (8.8 and 17.6 L). The extent of the impact becomes apparent when considered over an entire pipeline footprint.

"If a compressor operator has 500 units that are all leaking an average of 3 gallons [13.2 L] per day, the cost of that wasted oil is equivalent to US$3,613,500 per year at US$11 per gallon, assuming a utilization rate of 60%," said McCarthy. "ORS is a cost-effective, off-the-shelf solution that pays for itself quickly. It's hard to imagine this type of expense being absorbed without question, but the cost of excess lubricant consumption can be hard to account for because it is spread over many locations. It is easy to overlook a few gallons per day at a single facility without thinking about the system-wide implications. Compressor operators are becoming aware of this, and there are currently several ORS pilot projects underway with great success."

This system-wide approach is commonplace at Sloan. The company presented a paper titled "The High Cost of Over-Lubrication" at the 2022 Gas Machinery Conference. The paper introduced an expense model for pipeline operators to calculate the hidden cost of over-lubrication from purchase to removal, and the costs of its negative impacts including valve failures, filter and metering station fouling, and gas stream contamination. The paper demonstrated how pipeline operators could boost the return on investment of future projects by utilizing lube reduction.

"Many compressor operators seem to think that if a little oil is good, then a lot of oil must be better. That is a mistake, and an expensive one at that," said CJ Sloan, chief technology officer at Sloan and co-author of the paper. "In projects where we reduced lube rates anywhere from 50% to 90%, we have never had a compressor failure. In fact, it is quite the opposite. Compressors operate better when they get the right amount of lubricant and no more."

One thing to consider before taking on lube reduction projects is proper design and system integrity. In most cases, operators cannot just reduce lube rates 50% without causing problems. Most systems would not deliver the proper ratios of lubricant to both compressor and packing and need to be modified ahead of time. Lube reduction also decreases the margin of error, making the importance of regular preventative maintenance paramount.

"We've determined that the best way to guarantee success with lube reduction is to offer maintenance contracts, eliminating the burden of yearly testing and maintenance for operators," said CJ Sloan. "This service has several advantages, including peace of mind and a reduced need for lube system training. In addition, if purchased with a new system, Sloan's maintenance contracts can extend the warranty up to five years.

"Another key benefit is an immediate and impactful reduction to the carbon footprint of compressor operations," added CJ Sloan. "Refining lubricant is extremely energy-intensive, and the feedstock itself is oil. Each gallon of lubricant saved has a carbon footprint of up to 29 lb. [13.2 kg]. At a time when pipeline operators are doing whatever they can to reduce their impacts, any effort that has a positive effect on both the environment and the bottom line is a no-brainer. And lube reduction does both."

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What Went Wrong?

BY W. NORM SHADE

No company, product, or individual is immune from experiencing failures or shortcomings. The solution to these problems can often result in new "best practices." By relating some of these real case histories, it is hoped that the lessons learned will be educational to the readers. In each case, a failure or incident is described and the question is asked, "*What went wrong?*" In each case, the answer (or at least the best speculation) and the solution that was applied to the problem will be explained.

CASE №27

Why were compressor cylinders cracking after many decades of reliable service?

Gas was discovered leaking from the wall of a high-pressure compressor cylinder in a natural gas storage and withdrawal facility. The cylinder, which had accumulated 94,000 operating hours spanning 60 years, was one of 19 identical cylinders on multiple integral engine-compressors in the facility. The unexplained failure after decades of reliable service left the operator concerned about the other 18 cylinders, asking the question, "*What went wrong?*"!

The failed cylinder was one of two 2nd-stage cylinders of the type shown in Figure 1 on a 330-rpm integral engine-compressor. The 15-in. (381-mm), 8.5-in.- (216-mm-) bore diameter, 5000-psig (344.8-bar) maximum allowable working pressure (MAWP) cylinder normally operated with a maximum discharge pressure of 4200 psig (289.6 bar). An extensive metallurgical analysis revealed that the cylinder had several cracks that had originated in the seating ledge radii in both the head and crank end discharge valves. Over time, the cracks had propagated, finally reaching the other wall, creating a leak near an indicator valve connection.

Inspection of the other 18 cylinders at the facility revealed that five had internal cracks near the discharge valve seating ledges. Although this find was alarming, it was somewhat of a conundrum that 13 of the 19 cylinders that operated under identical conditions were crack-free. With six decades of operation within design operating conditions and nearly 100,000 hours of operation at design speeds of 330 rpm, the cylinders had accumulated approximately 2×10^9 stress cycles. This long successful history essentially dispelled any notion of high-cycle fatigue as the cause of crack initiation. However, once cracks are initiated, fatigue could certainly drive crack propagation under continuous operation. Accordingly, the investigation of potential causes of crack initiation turned toward the forces on the cylinder valve seats caused by the assembly process. This ultimately revealed the smoking gun that answered the question, "*What went wrong?*"!

Figure 1. Second-stage cylinders similar to the type that failed on a 330-rpm integral engine-compressor are shown.

As shown in Figures 1 and 2, the cylinder valve caps were retained by nuts tightened onto ten 1.5-in.- (38.1-mm-) diameter studs. A 3-in.- (76.2-mm-) diameter jack bolt in the cap was used to hold the valve against the seat in the valve pocket. A large cap on the jack bolt was tightened against a metal gasket to prevent leakage around the jack bolt threads to atmosphere.

Figure 3 shows a cross section of the valve and cap assembly. The valve cap bolting provided the force needed to crush a round-wire steel gasket between the cap and the outer edge of the valve pocket bore in the cylinder to accomplish primary gas sealing between the

Figure 2. A 3-in. (76.2-mm) jack bolt in the valve cap was used to hold the valve against the seat in the valve pocket. A large cap on the jack bolt was tightened against a metal gasket to prevent leakage around the jack bolt threads to atmosphere.

cap and cylinder. The jack bolt was intended to provide the force necessary to crush the flat steel gasket under the valve at the valve pocket seating ledge. Both the valve cap bolting and the jack bolt had to provide the necessary preload to crush the metal gaskets for sealing. In addition, the valve cap bolting had to resist the hydrostatic pressure load on the underside of the valve cap.

The correct assembly practice for this type of assembly (that crushed two steel gaskets) required sufficient tightening of the nuts on the valve cap studs to resist all the hydrostatic and gasket seating forces. After tightening the nuts on the valve cap studs, the jack bolt would then be tightened to provide the necessary preload force on the valve seat gasket. A serious concern with this type of design is that if the cap bolting is tightened further after the jack bolt has already been tightened, then the additional cap bolting force transfers directly to the valve seat, loading the valve seat more than intended by the designers. Designs with jack bolts for preloading the valve seat gaskets can be very problematic and were phased out many years ago for new cylinders.

Yet another concern with the use of metal gaskets is that many gaskets don't conform to the requirements for the material to be very soft (i.e., very low yield strength).[1] A very low carbon content in a fully annealed condition is required for steel metal gaskets. Harder gasket material requires more bolting force to provide adequate surface pressure and crush (i.e., gasket yielding) for sealing.

In this case, the historical bolt torque specifications were not well documented. Most likely, the torquing varied over time to prevent external leaks past

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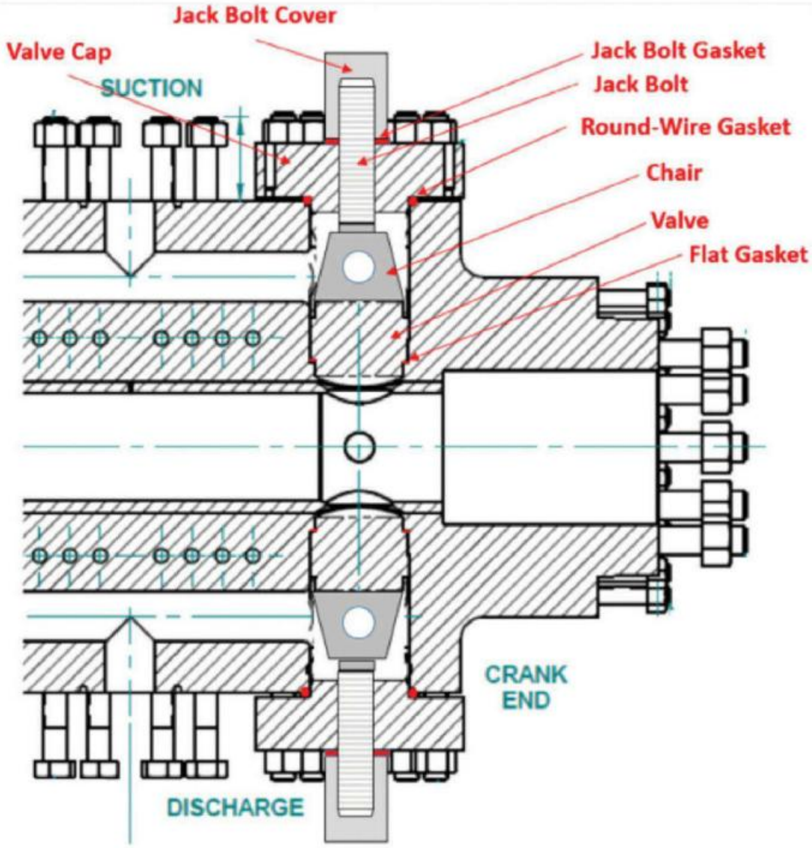

Figure 3. Cross section of the valve and cap assembly on the original cylinders. The valve cap bolting provided the force needed to crush a round-wire steel gasket between the cap and the outer edge of the valve pocket bore in the cylinder to accomplish primary gas sealing between the cap and the cylinder. The jack bolt was intended to provide the force necessary to crush the flat steel gasket under the valve at the valve pocket seating ledge. Both the valve cap bolting and the jack bolt had to provide the necessary preload to crush the metal gaskets for sealing; in addition, the valve cap bolting had to resist the hydrostatic pressure load on the underside of the valve cap.

the round wire gaskets or to avoid internal knocking from loose valve assemblies. In any event, there was a lot of uncertainty about the magnitude of the assembly load applied to the valve caps and the valve seats over 60 years of service. The required bolt loads were calculated, and historical records indicated that the maximum cylinder discharge pressures were 4230 psig (291.7 bar), about 85% of the cylinder maximum pressure rating. The rectangular-shaped cylinder was carefully measured, and a 3D computer-aided design (CAD) model was generated. The CAD model was used to conduct extensive finite element analyses (FEA) that considered assembly loads, thermal loads, and static and cyclic pressure loads. FEA studies confirmed that the peak stresses coincided at the exact location of the crack origin as determined from the metallurgical investigation.

Although the FEA confirmed the crack initiation point at the edge of the 1/16-in. (1.6-mm) fillet radius at the discharge valve seating ledge, the high magnitude of the stress predicted from the assembly loads was very troubling. The FEA analysis did not consider nonlinear effects (i.e., yielding), and the calculated peak stress from the valve cap and jack bolt assembly was more than twice the yield strength of the material. So, these FEA results were not realistic, because once the local stress reached the material yield strength, yielding (i.e., plastic deformation) would occur. A small amount of local yielding is not necessarily a concern, but the wide area of predicted high stresses in this case indicated that there would be substantial material yielding in the valve seat fillets caused by the normal assembly loads on the bolting. This very high local plastic strain throughout the valve seat fillet would be repeated every time the valve caps and/or the valve cap jack bolts were tightened. Although the stress magnitude might be limited by plastic deformation, the high plastic strain magnitude over several cycles led to the theory that low-cycle fatigue was the likely source of crack initiation and failure.

Low-cycle fatigue is characterized by a high strain amplitude and a low number of cycles, as shown in Figure 4.[2] The low-cycle fatigue regime is characterized, very arbitrarily, as the range from one to 10,000 cycles. The actual range depends on the material's properties. It is the point where most of the cyclic strain is plastic rather than elastic. As depicted in Figure 4, low-cycle fatigue failure can occur in just a few cycles if the cyclic strain is high enough. In the case of low-cycle fatigue, stress produces both elastic and plastic strains, but the bulk of the strain is plastic.[3] This is particularly important in areas with geometric discontinuities or stress risers. As a result of the plastic deformation in low-cycle fatigue, failure occurs in a reduced number of cycles compared to high-cycle fatigue failure.

In the low-cycle range of metal life, the primary parameter governing life is the plastic strain per cycle.[4] A practical analogy can be drawn by bending a piece of copper or soft, low-carbon steel wire. The wire is very ductile and can accommodate a significant amount of plastic strain and deformation. However, if the plastic strain produced by each bend cycle is high enough, it does not take more than a dozen or so cycles before the wire breaks.

Figure 5 is useful for explaining the differences in failure mechanisms. In the left image, static failure can be produced by one cycle of high loading that causes large global deformation and a very large plastic strain. Since only six of 19 cylinders had cracks in the valve seats, especially considering their long period of service, static failure was improbable as the cause of crack initiation. In the right image in Figure 5, high-cycle fatigue results in small local elastic strains, which, if high enough, can initiate a surface crack that is then propagated over time. As explained earlier, the fact that these cylinders had about 2×10^9 stress cycles (two orders of magnitude above the typical material endurance limit) before any cracks became evident made high-cycle fatigue improbable as the cause of crack initiation. In the middle image of Figure 5, the case of low-cycle fatigue is characterized by local plastic strains that will eventually initiate a crack. Loads that increase the mean stress above the yield stress

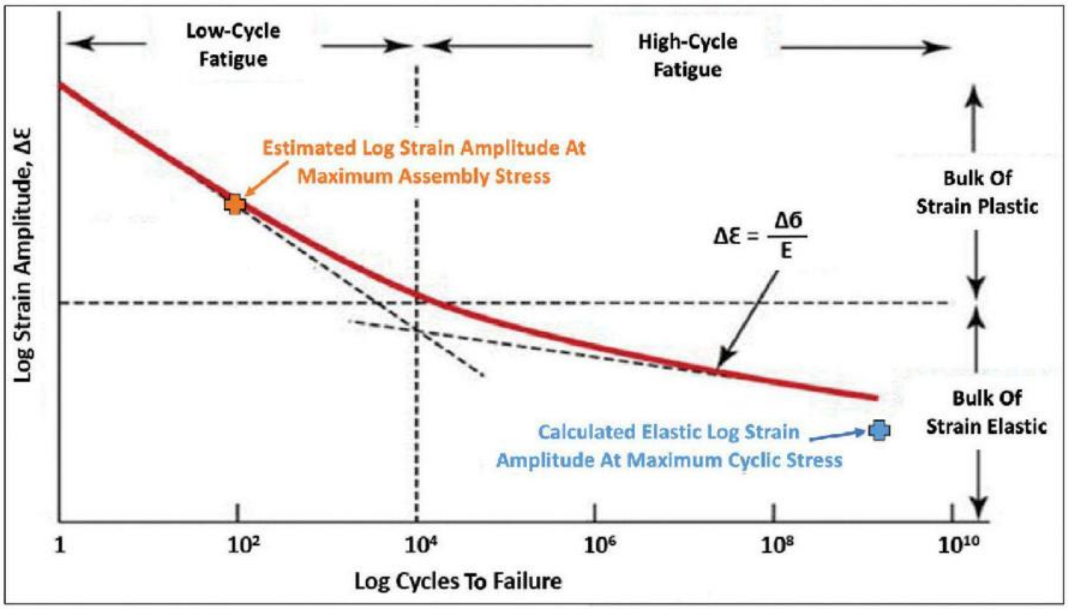

Figure 4. Representation of strain amplitude vs. cycles to failure showing low-cycle and high-cycle fatigue regimes. The strain amplitude that resulted from the calculated cyclic stress amplitude during the compressor operation in this case is shown symbolically by the blue cross at 2×10^9 cycles, well below the failure line. The loading during the valve cap assembly process resulted in a strain shown symbolically by the orange cross at 1×10^2 cycles, reaching or exceeding the low-cycle fatigue limit.

produce failure by yielding rather than by fatigue, and classical fatigue analysis becomes inapplicable.[5]

In Figure 4, the blue cross at 2×10^9 cycles represents the calculated strain amplitude that resulted from the calculated cyclic stress amplitude during the compressor operation. The cyclic strain during normal compressor operation is substantially less than the yield strength and therefore entirely elastic. Since it is also well below the failure line, high-cycle fatigue failure is not likely. However, the loading during the valve cap assembly process resulted in a strain shown symbolically by the orange cross at 1×10^2 cycles in Figure 4. The actual strain (which is mostly plastic) and the number of cycles was not known. However, each time the valve cap was assembled, the high plastic strain was repeated, eventually reaching the low-cycle fatigue limit that would initiate a crack. Over many years and many dozens of cylinder valve cap assembly events, even if detailed assembly specifications had been provided and adhered to, the loading on these legacy cylinders was such that low-cycle fatigue became a serious concern.

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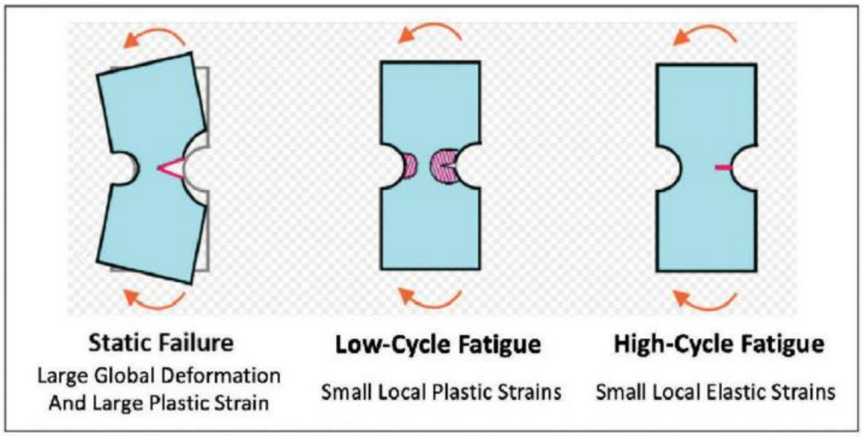

Figure 5. Differences in common failure mechanisms. (Left) Static failure can be produced by one cycle of high loading that causes large global deformation and a very large plastic strain. (Right) High-cycle fatigue results in small local elastic strains, which, if high enough, can initiate a surface crack that is then propagated over time. (Middle) The case of low-cycle fatigue is characterized by local plastic strains that will eventually initiate a crack.

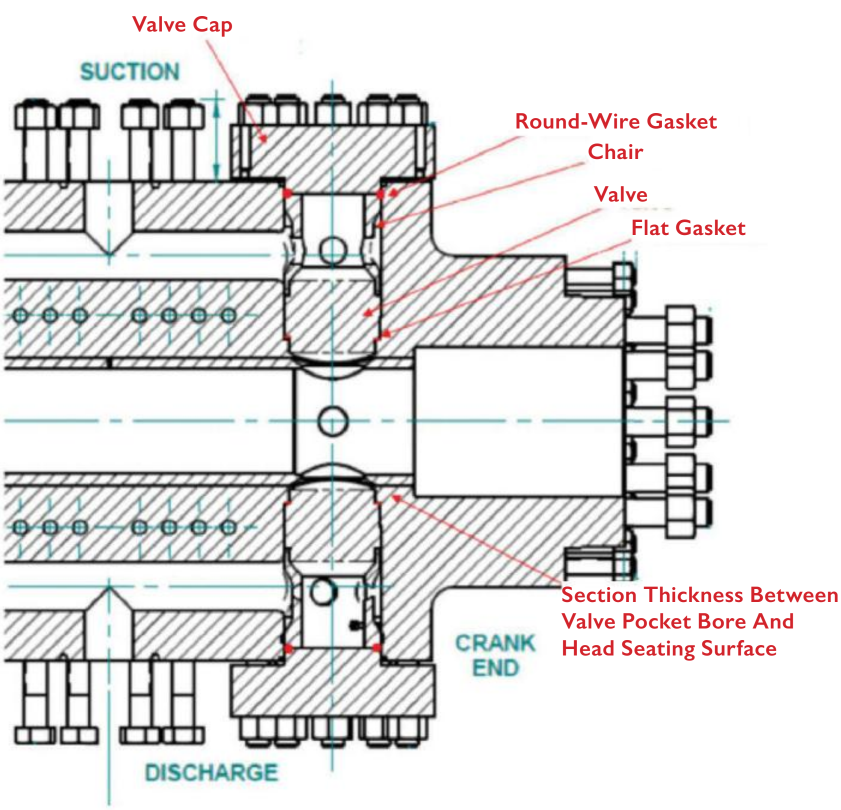

Figure 6. To resolve the problem, new cylinders were designed to eliminate the valve cap jack bolts. Because of the high operating pressure and concern about the reliability of elastomeric O-ring seals in this case, it was necessary to retain metal seals for the valve caps as well as the valve seats. Accordingly, the cylinder design had to be significantly improved to eliminate its susceptibility to low-cycle fatigue, including tolerance for uncertainties in metal gasket hardness and assembly procedures over time. The most important improvement was upgrading the cylinder body material to increase the yield strength. Other significant changes included increasing the valve seating ledge fillet radius and increasing the cylinder width to reduce the secondary effects of the cylinder head assembly loads on the valve pockets.

Since the high valve cap and jack bolt assembly loads caused very high plastic strains in the valve seat fillets, by inference from Figure 4, it would take a relatively small number of cycles to initiate a crack. Once a crack started, additional assembly events over time would propagate the crack, as would normal high-cycle fatigue from the cyclic internal pressure that occurs with each piston stroke. As the crack propagated, the section would become weaker and the rate of propagation would increase. Discussions with the station operator indicated that individual valve caps could have been installed and the bolting tightened somewhere in the range of 60 to 120 times over the 60-year operating span before the gas leak was detected. Some valve caps would have been assembled more than others, and with the uncertainty about the assembly procedures and bolt torquing that was applied, the low-cycle fatigue theory reasonably explained why the cylinders had performed reliably for so many years, and why some were now developing cracks.

To resolve the problem, new cylinders were designed to eliminate the jack bolts as shown in Figure 6. Because of the high operating pressure and concern about the reliability of elastomeric O-ring seals in this case, it was necessary to retain metal seals for the valve caps as well as the valve seats. Accordingly, the cylinder design had to be significantly improved to eliminate its susceptibility to low-cycle fatigue, including tolerance for uncertainties in metal gasket hardness and assembly procedures over time. The most important improvement was upgrading the cylinder body material to increase the yield strength by 89%, from 55,600 to 105,000 psi (3833 to 7239 bar). In addition, the valve seating ledge fillet radius was increased to ¼ in. (6.4 mm) to reduce the stress concentration, and the cylinder width was increased by 20% to reduce the secondary effects of the cylinder head assembly loads on the valve pockets. FEA studies of the new cylinder design showed that the valve cap assembly stress was 63% of the new material's yield strength. It was still higher than the original material yield strength, which dictated the upgrade to the stronger material. The new cylinders were designed to replace the original cylinders without changing pipe connections or mounting.

Following are several lessons learned from this experience:

First, be aware that legacy cylinders with bolted joints having soft steel or iron gaskets can be susceptible to low-cycle fatigue failure after application of assembly loads from many maintenance cycles. Assembly torque specifications for legacy cylinders are often lacking entirely or missing. The specifications may have evolved away from the manufacturer's recommendations because operators found that more assembly torque was necessary to avoid leaks past metal valve cap gaskets. Accurate application of fastener torque is often a difficult (if not impossible) process on large equipment, especially on discharge valve cap bolting that is hard to access. In addition, units with jack bolt valve retention are susceptible to severe overstressing of cylinder valve seats if proper assembly procedures are not adhered to. These factors increase the risk of crack initiation from low-cycle fatigue as the number of maintenance cycles increase over time.

Use metal gaskets with the compressor manufacturer's specified hardness and ductility. These specifications may not be available on legacy assemblies. Several sources supply soft steel and iron gaskets, and there are significant variations in hardness. Round wire gaskets have a weld that is often harder and larger in diameter than the base material. Even flat metal gaskets, which are stamped or laser cut from thin sheets, may be harder than specified. ASME gasket calculations assume that soft steel or soft iron gaskets have a yield strength of less than 18,000 psi (1241 bar).[6] If they are harder than that, sealing will require more assembly load, and that will increase stresses on all the components in the assembly. The steel gasket material must be very low-carbon content and fully annealed, often in a hydrogen-rich atmosphere, to achieve the required very low yield strength. Unfortunately, many replacement gaskets are harder than assumed in the design calculations, and most operators are unaware of the non-conformance and its potential adverse implication. Metal gaskets take a permanent set when assembled, which changes their shape and work hardens them. Accordingly, metal gaskets should never be used more than once.

Classical methods of stress analysis (which were all that were available to most manufacturers before the early 1970s) fell short of predicting valve seat stresses accurately enough to avoid the potential for low-cycle fatigue failure from multiple assembly cycles. Fortunately, use of today's FEA tools with carefully chosen boundary conditions allow stresses in valve seats and other critical areas to be evaluated under all loading conditions. This avoids stresses and excessive strains that would make the cylinders susceptible to failure from both low-cycle and high-cycle fatigue as long as specified assembly procedures and bolt tightening specifications are consistently followed and the cylinders are operated within rated conditions throughout their lives.

Cylinders with an unknown history or that have not been maintained consistently using manufacturer's procedures and bolt tightening specifications may be at risk of low-cycle fatigue failure as they accumulate maintenance cycles. This can be true for legacy cylinders on slow-speed compressors, but it can also be an issue for comparatively newer, high-speed compressor cylinders that have had unusually high numbers of valve maintenance events and/or have not been consistently serviced in accordance with the manufacturer's procedures and bolt tightening specifications.

REFERENCES

[1] N. Shade, "What Went Wrong? Case No. 24: What Caused the Cylinder Gas Leak?" *Gas Compression Magazine* 7, no. 12 (2022); p. 42.

[2] T. Clark, K. Edwards, G. Jedrosko, and N. Shade, "Cumulative Lifetime Assembly-Induced Low-Cycle Fatigue in Legacy Compressor Cylinders," GMRC Gas Machinery Conference (Fort Worth, TX: GMRC, 2022).

[3] A.F. Madayag, ed. (1969), *Metal Fatigue: Theory and Design* (New York, NY: Wiley & Sons, Inc.).

[4] S.S. Manson, *Thermal Stress and Low-Cycle Fatigue* (New York, NY: McGraw Hill, 1966).

[5] C.C. Osgood, *Fatigue Design* (New York, NY: Wiley & Sons, Inc., 1970).

[6] J.H. Bickford, *An Introduction To The Design And Behavior of Bolted Joints* (New York, NY: Marcel Dekker, Inc., 1981).