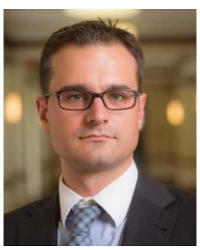
47TH TURBOMACHINERY & 34TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 17-20, 2018
GEORGE R. BROWN CONVENTION CENTER

HIGH RELIABILITY PISTONS FOR RECIPROCATING COMPRESSORS WITH VALIDATED PERFORMANCE MODELLING

Andreas Brandl
Engineering Manager
HOERBIGER Service Inc.
Houston, TX, USA

John Ladd
Solutions REE Engineer / Compressor analyst
HOERBIGER Service Inc.
Houston, TX, USA

Bruce Hermonat
Principal Project Engineer
HOERBIGER Service In.c
Houston, TX, USA



Andreas Brandl is the Engineering Manager at HOERBIGER Service Inc. in Houston TX. His work focuses on Reciprocating Compressors for the Oil & Gas and Chemical/Petrochemical industry.
Before coming to Texas he was working in the corporate R&D department for HOERBIGER in Austria. Andreas earned his Master's degree in mechanical engineering at the Vienna University of Technology and his MBA at the Jones Graduate School of Business at Rice University.

John Ladd is a Solutions REE Engineer and Compressor Analyst at HOERBIGER Service In. in Houston, TX. His primary role is conducting comprehensive technical evaluations of reciprocating compressors to identify and quantify unit improvements in reliability, efficiency, and environmental soundness. Prior to his current role John earned his Master's in mechanical engineering at Colorado State University with a focus on legacy integral pipeline compressors.

Bruce Hermonat is a principal Project Engineer at HOERBIGER Service Inc. in Houston TX. His work focus is on reciprocating compression system design and project management in the process gas industry. He has worked for HOERBIGER since May of 2007. Prior to working for HOERBIGER he worked as an application Engineer and a product manager for Turbine Fuel Treatment Systems.

ABSTRACT

Piston slippage (blow-by) on reciprocating compressors is highly predictable due to defined leakage paths at the end gaps of piston rings. A new engineering approach quantifies the slippage through the piston rings and determines the dynamic pressure difference on each ring on the piston. With this approach the expected discharge gas temperature increase, expected capacity losses and the risks of rider bands activation due to piston ring slippage can be quantified. The piston design and ring styles can be iterated to find an optimized piston layout for a given application.

Within the last two years piston performance has been evaluated and tracked on all compressors that have been subject to technical surveys (Reliability, Efficiency and Environmental Soundness – REE – Audits). The result of this study suggests that on 30% of the reciprocating compressors in the process gas industry there is at least one cylinder with a piston that shows high sensitivity to piston ring leakage and subsequent performance related issues.

This paper suggests quantifying piston performance as a standard when evaluating compressor reliability and efficiency. The industry managed to reduce compressor valve related problems due to more sophisticated modelling tools and smart design changes on valves. It is time to go that next steps on pistons.

INTRODUCTION

Most industrial reciprocating compressors in the Oil & Gas industry have double acting cylinders. That means that gas is compressed during both, the inward and the outward stroke. The piston rings seal the gas between the two cylinder chambers (head end – HE, and crank end – CE), the rider band carries the weight of the piston. Figure 1: Piston with rings and riders in a cylinder of a reciprocating compressorFigure 1 shows a piston with piston rings and rider bands in the cylinder. Both functions (Sealing the gas and carrying the piston weight) are fundamental for reliable compressor operation. And both – rider bands and piston rings – are a frequent cause of unplanned outages. The study from Goebel, 2014 suggests that 9% of compressor damages are due to piston ring related problems and 9% due to rider ring related problems. Both together constitute with 18% the 2nd most frequent failure cause of reciprocating compressors after valve failures.

Most cylinders are lubricated which reduces the amount of frictional wear. Non-lube application that are not uncommon (special materials for non-lube service are required) are especially critical with respect to life time and blow by. Very often problems on piston rings or rider bands are not identified early enough and the impacts on compressor performance (e.g. increased discharge temperatures) are attributed to the wrong cause and are thus not addressed. When valves fail, the valve covers get hot, if the rod packing fails, the packing leakage rates increase and operators can identify the cause of the observed performance losses. Cylinder ring related problems such as piston ring leakage and rider band wear are harder to identify and can only be confirmed with data acquisition equipment. The MTBF of the components are expected to be, depending on the application and industry between one and five years.

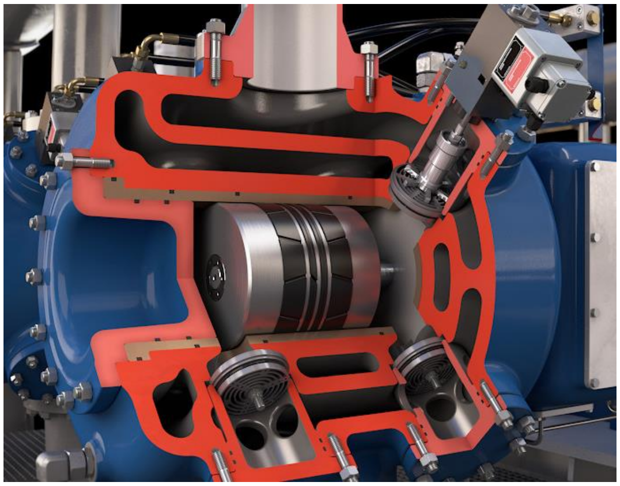


Figure 1: Piston with rings and riders in a cylinder of a reciprocating compressor

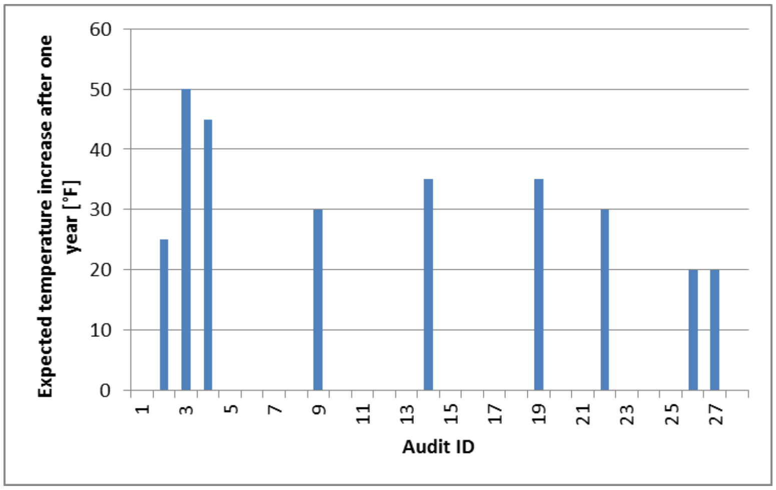


Figure 2: Compressor audits with substantial piston performance concerns.

There are several different designs of piston rings and rider rings. The most common piston ring is an angle cut ring. Figure 3 shows different kind of piston rings including a straight cut ring, an angle cut ring, a step cut ring and a pressure balanced ring. One of the most common rider bands is the solid shrunk on rider band without relief grooves. Figure 3 gives an overview of the most common ring styles for cylinder rings. The application engineering of these designs is often based on rule of thumb methods. The failure statistics prove (Goebel 2014) that a more comprehensive approach in designing pistons and cylinder rings is required.

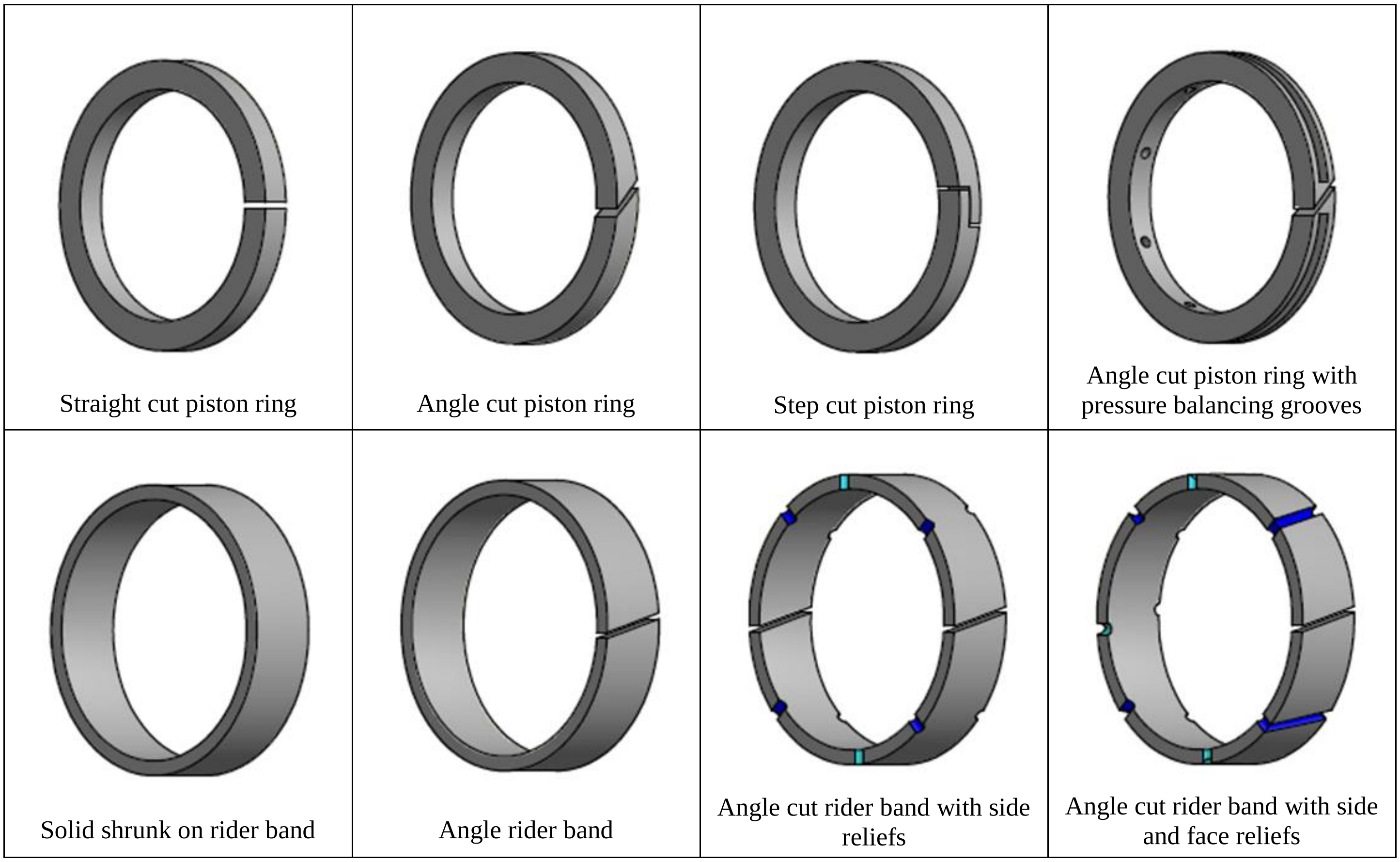


Figure 3: Different styles of piston rings and rider bands as provided by all major suppliers of compressor components.

MODELLING

The mass flow model

Every cylinder ring (piston and rider ring) is represented in the model by one orifice. The flow model used to describe the flow through these orifices is based on the steady state isenthalpic throttle process through an effective leakage area. At a given wear state of the rings the effective flow area of the piston ring can be calculated.

$$\dot{m} = \emptyset\sqrt{2p_0\rho_0}\sqrt{\frac{\kappa-1}{\kappa}\left(1-\left(\frac{p_1}{p_0}\right)^{\frac{\kappa+1}{\kappa}}\right)} \tag{1}$$

The compression chamber model

The cylinder pressures and temperatures in both chambers (HE and CE) are calculated using the standard equations for pressure and density changes in reciprocating compressors. The volume change is given by the compressor geometry, the mass flow through suction and discharge valves are calculated using equation (1). The effective flow area of the valve is a valve specific parameter that can be provided by the valve manufacturer. There are several correlations available to model the heat transfer within the cylinder (e.g. Disconzi 2012) in case adiabatic change of state cannot be assumed. For details and equations on standard compressor simulation see

Brandl, 2010.

The piston ring leakage model

In order to determine the effective leakage area of each piston ring the following assumptions are made:

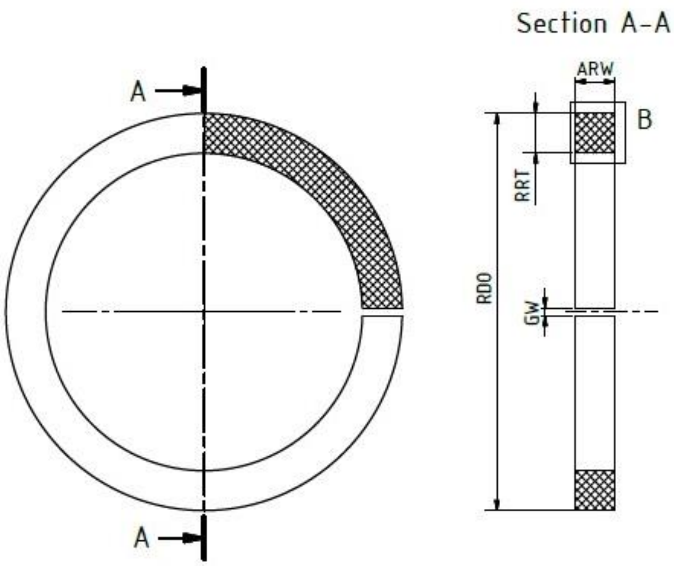


Figure 4: End gap on straight cut piston ring.

- The end gap (see dimension GW in Figure 4) of the piston ring is the only leakage gap. Feistel shows that this is a viable assumption.
- The effective flow area is derived from the geometric flow area with the help of a discharge coefficient. Radcliffe 2001 performed measurements for different rings styles. The ring style (e.g. angle cut or step-cut) is accounted for in the model by applying a discharge coefficient corresponding to the ring design. Radcliffe suggests a discharge coefficient of 0.65 for straight cut rings.
- According to Liu (1986) cylinder lubrication reduces the blow by 30% to 35% compared to the non-lubricated case. The effect of lubrication on the piston leakage is accounted for by reducing the effective leakage area of the piston rings accordingly.
- The pressure loss in tangential direction (in case the ring gaps of two neighboring piston rings are not aligned) is neglected. According to Ruddy 1981 this is an acceptable assumption for bores smaller than 500mm (20in). As the results show, piston ring leakage is unlikely to have a significant impact on bores larger than 15in. This assumption is thus justified as it covers the diameter range of interest (<15in).
- Heat transfer is accounted for by a constant head transfer coefficient and by a heat sink with the temperature of the cylinder coolant.

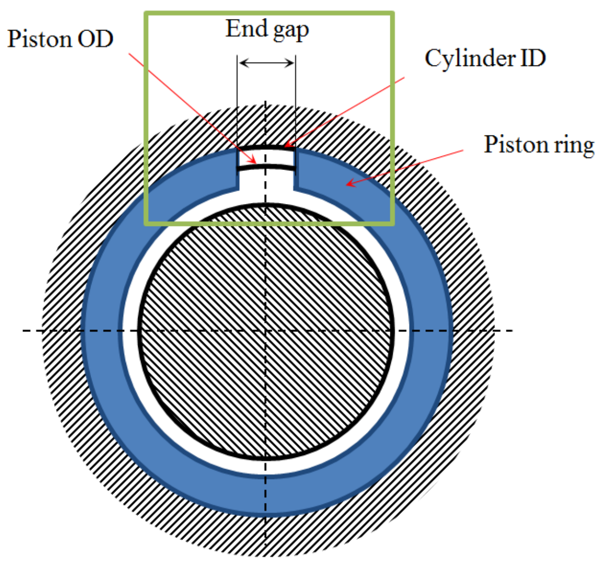


Figure 5: Piston ring in piston ring groove.

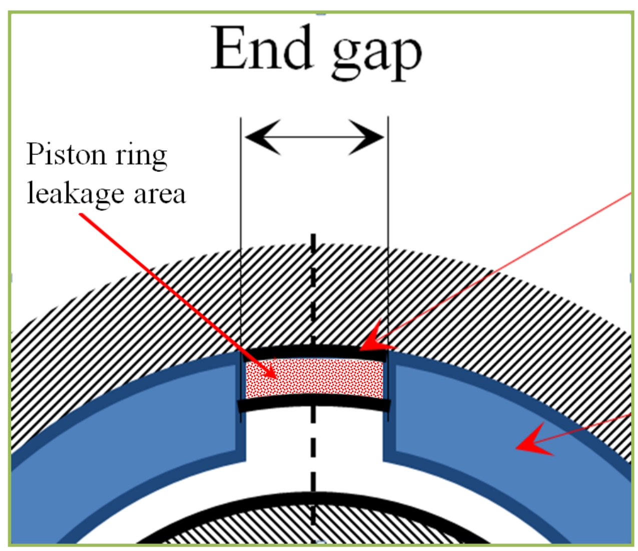


Figure 6: Leakage gap on a straight cut piston ring.

The effective leakage area of each piston ring is given by cylinder and piston diameter, end gap and discharge coefficient.

$$\emptyset = \frac{d_{cyl} - d_{piston}}{2} g C_d \qquad (2)$$

The calculation of pressures, temperatures and gas densities in between the piston rings is similar to the approach in the compression chamber with the exception that the change in volume is zero. The governing equations are derived from mass balance, energy balance and the ideal gas law.

$$d\rho_j = \frac{1}{V_j}\left(d_e m_{ij} - d_e m_{jk}\right) \tag{3}$$

$$dp_j = \kappa \frac{p_i}{\rho_i V_j} d_e m_{ij} - \kappa \frac{p_j}{\rho_j V_j} d_e m_{jk} + \frac{\kappa - 1}{V_j} d_e Q_j \tag{4}$$

The tribological model

Radial wear of the piston rings has a direct impact on the leakage area. Wear on the OD of the piston ring reduces the radial thickness and opens the end gap by 2π times the radial wear rate.

$$g = g_{ini} + 2\pi\, w_{radial} \tag{5}$$

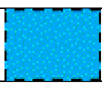
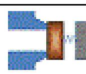
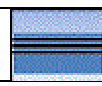
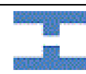
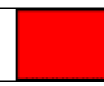
This linear correlation between the wear rate and the leakage area is one reason for the decreasing performance on reciprocating compressors over time.

The wear rate is determined by a wear coefficient that depends on multiple variables and normally has to be found via experiments. Typical wear coefficients from wear tests (see Radcliffe, 2001) are in the range of 1e(-16) m^3/Nm. The wear rate that determines the end gap is given by:

$$w_{radial} = k p_{contact}\, v_{rel} t \tag{6}$$

Example of an implementation for piston performance modelling

The piston that is given in Figure 7 has two angle cut rider rings with face and side relief grooves and four angle cut pressure balanced piston rings (see Figure 3). Figure 8 shows the corresponding model to determine the piston ring leakage for given operating conditions.

				
Ambiance e.g. Suction and discharge conditions	Suction or discharge valve	Compression chamber (HE or CE)	Piston ring or rider ring	Volume in between the piston rings

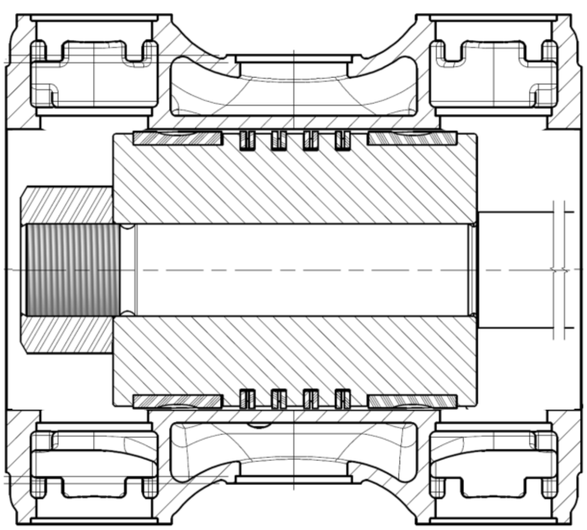


Figure 7: Piston with two rider bands and four pressure balanced piston rings in cylinder

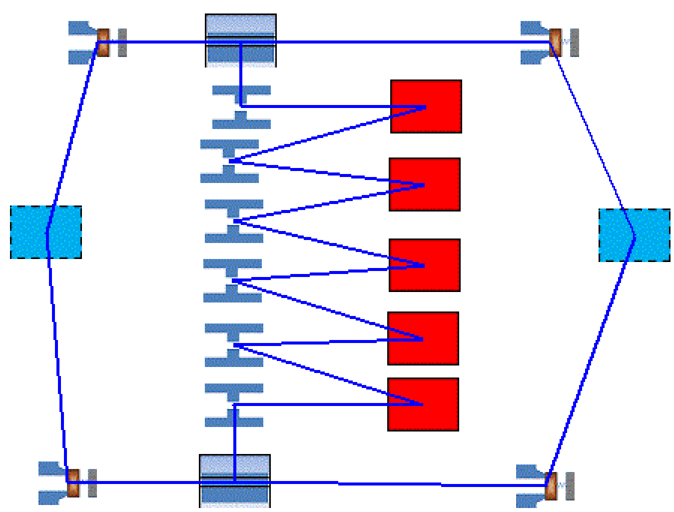


Figure 8: Corresponding piston leakage model

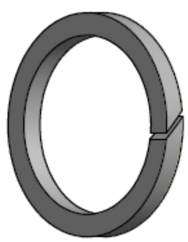
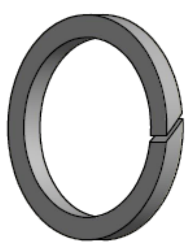
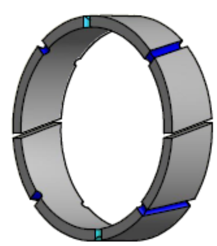
CASE STUDY 1

Case description

This described piston performance modelling procedure was applied on a compressor that came down every six months due to rider band failures on the 1st stage.

- Speed: 507 rpm
- Rated power: 500hp
- 1st stage cylinder bore: 17.5in
- 2nd stage cylinder bore: 9.5in
- Stroke: 9in
- Rod dia: 2.25in
- Isentropic exponent: 1.167
- Molar mass: 45 kg/kmol
- psuc 1 = 10psig
- pdis 1 = 75psig

Two different ring arrangements are compared:

	Piston ring design	# of piston rings	Rider ring	# of rider rings	Location of rider ring
Existing		2		2	Outboard
Proposed		2		2	Outboard
Changes	No changes	No changes	Face and side relief grooves	No changes	No changes

Results

Figure 9 shows the pressure trends for the original ring layout. Each of the cylinder rings (rider ring – piston ring – piston ring – rider ring) causes a pressure drop. The rider rings do not have relief grooves and the leakage area of the rider bands is comparable to the leakage area of the piston rings. The result is that the pressure drop on the rider bands is very similar to the pressure drop on the piston rings. Rider bands are typically designed for a contact pressure of 5psi (non-lube). Any pressure difference across the rider band will increase the contact pressure and will accelerate the wear. The radial wear length of the riders is limited with ~0.120in. Rider bands that are exposed to a pressure difference are likely to fail prematurely.

Figure 10 shows the pressure trends with the rider band with relief grooves. Due to the wide relief grooves on the rider bands the piston ring leakage does not cause any significant pressure drop across the rider bands. Looking at Figure 10, there are only two distinct pressure drops, piston ring 1 and piston ring 2. The pressure difference (and thus the wear rate) on the piston rings increases which is acceptable as the available wear thickness is an order of magnitude larger compared to rider rings.

Figure 11 and Figure 12 show the average pressure difference on each of the rings with the original set up and the recommended ring design. The recommendation was implemented and the run time increased by a factor of three.

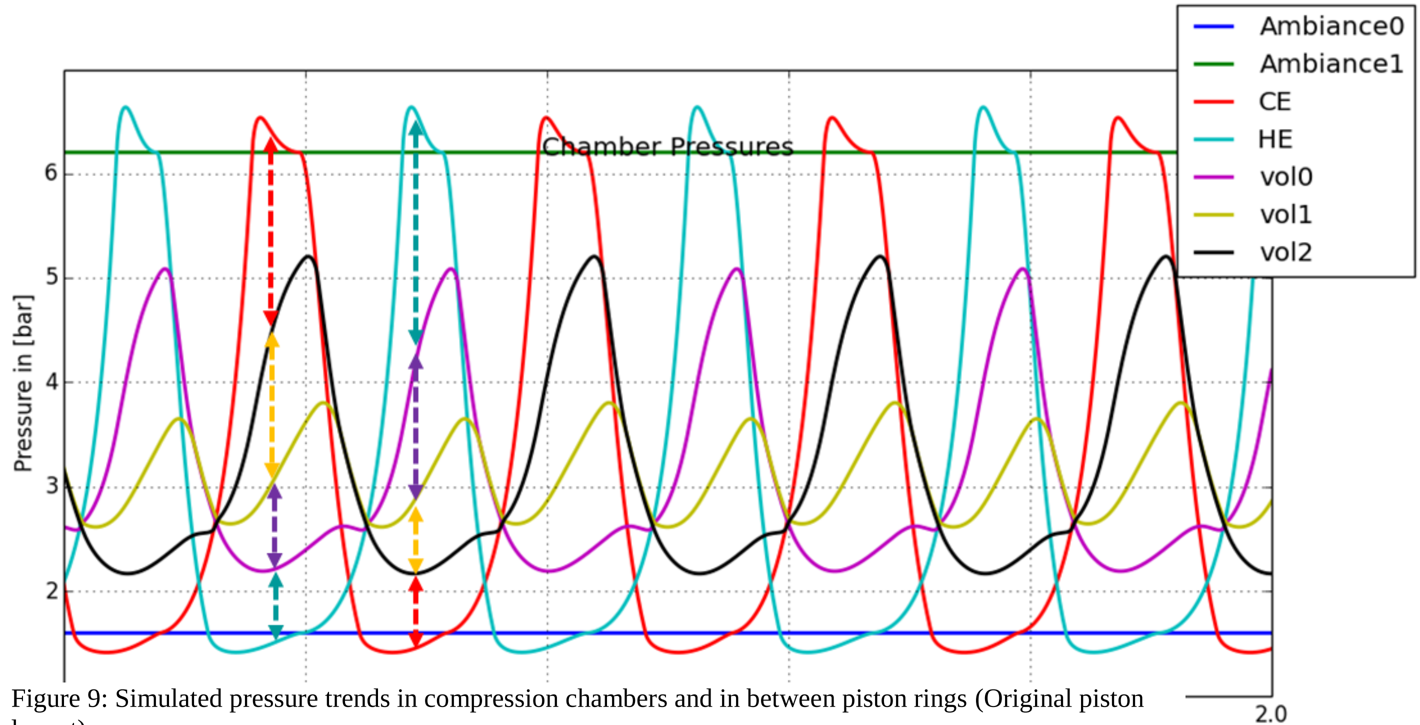


Figure 9: Simulated pressure trends in compression chambers and in between piston rings (Original piston layout)

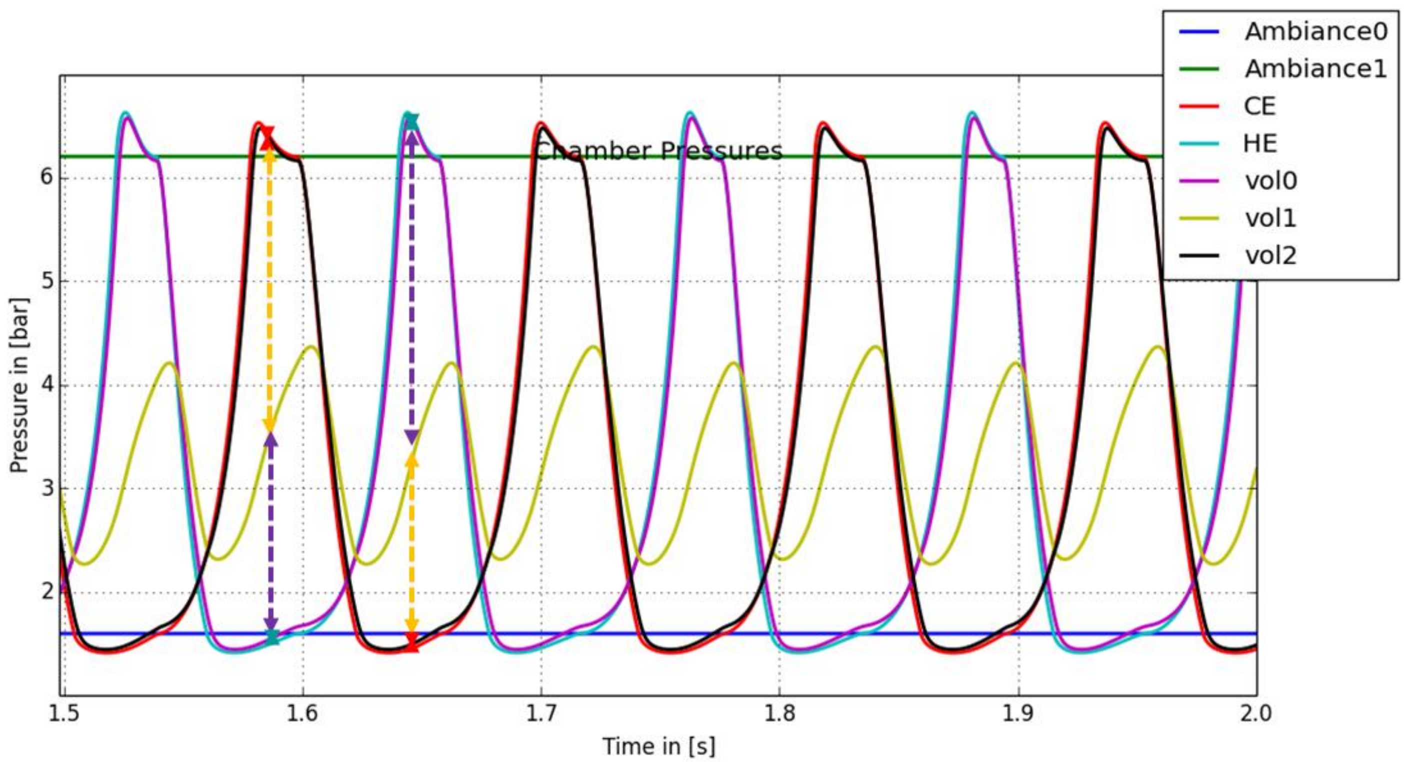


Figure 10: Simulated pressure trends in compression chambers and in between piston rings (Recommended piston layout)

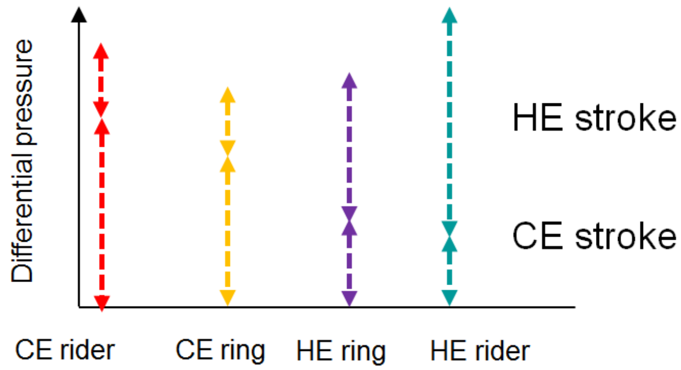


Figure 11: Average pressure difference across piston and rider rings

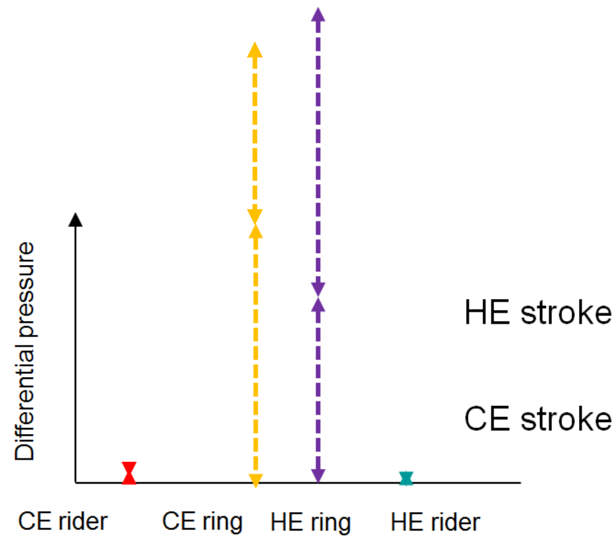


Figure 12: Average pressure difference across piston and rider rings

CASE STUDY 2

Case description

High discharge temperatures on both stages on a two throw non-lube Hydrogen compressor. The compression ratios on both stages were moderate (2.25 on stage 1 and 1.65 on stage 2) but the discharge temperatures became critically high within one year of operation. Measured discharge temperatures were 50°F higher than the isentropic discharge temperatures. The valve losses were low so most of the irreversibility that caused the temperature increase could be attributed to piston blow-by.

- Speed: 412 rpm
- Rated power: 250hp
- 1st stage cylinder bore: 5.75in
- 2nd stage cylinder bore: 4.25in
- Stroke: 9in
- Rod dia: 2in
- Gas: Hydrogen
- Isentropic exponent: 1.4
- Molar mass: 2 kg/kmol
- psuc 1 = 405psig
- pdis 1 = 929psig
- pdis 2 = 1534psig

Table 1 – Relevant operating conditions – discharge temperatures are 54°F higher (stage 1) and 35°F higher (stage 2) than the isentropic discharge temperature for reversible, adiabatic processes.

Π_1	$T_{dis1\ Theor.}$ [°F]	$T_{dis1\ measured}$ [°F]	ΔT_1	Π_2	$T_{dis2\ Theor.}$ [°F]	$T_{dis2\ measured}$ [°F]	ΔT_2
2,3	226	280	54	1,69	190	225	35

Pressure data was analyzed and severe piston blow-by was confirmed on both stages, see Figure 13 and Figure 14. Comparison of the measured pressure trends (solid curves) and the ideal pressure trends (dotted curves) reveals a typical piston leakage pattern (cylinder pressure rises faster (compared to the theoretical curve) at the beginning of the compression stroke and slower at the end of the compression phase).

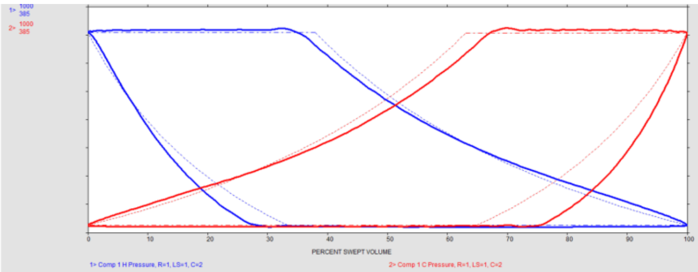


Figure 13: Measured pV diagram on stage 1 (Head end chamber in blue, crank end chamber in red)

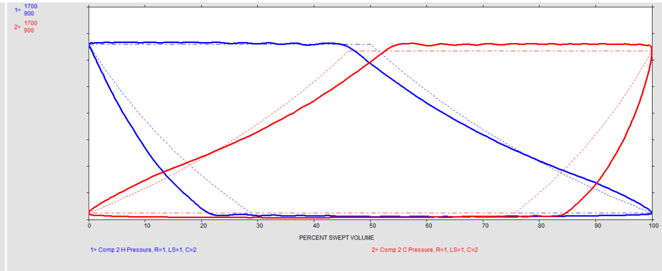


Figure 14: Measured pV diagram on stage 2 (Head end chamber in blue, crank end chamber in red)

The piston performance modelling technique described above was applied for the existing pistons (see Figure 15 and Figure 16). The results with regards to temperature trends can be seen in Figure 18 and Figure 17. The expected temperature increase on stage one after 4000hrs of operation is about 60°F on stage 1 and about 35°F on stage 2. A comparison with Table **1** – Relevant operating conditionsTable 1 shows that the simulation result corresponds well with the measured discharge temperatures (The 1st stage discharge temperature increase due to blow-by is almost twice as high as the 2nd stage discharge temperature increase).

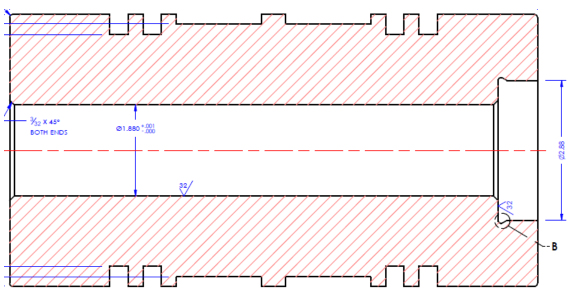


Figure 15: Existing piston design, stage 1

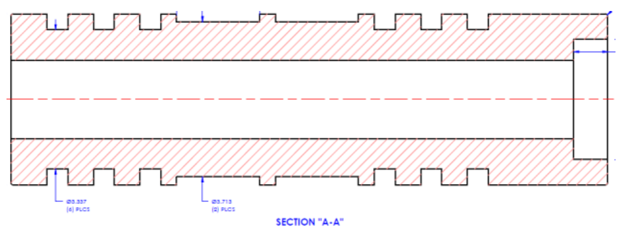


Figure 16: Existing piston design, stage 2

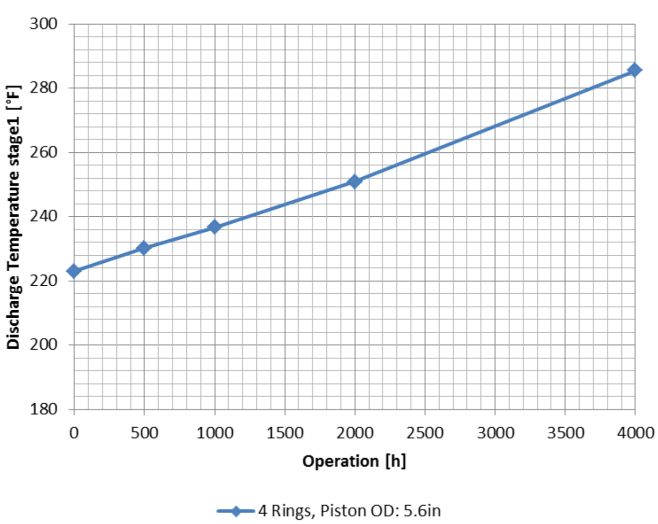


Figure 18: Simulation results - Discharge temperature as a function of time on stage 1.

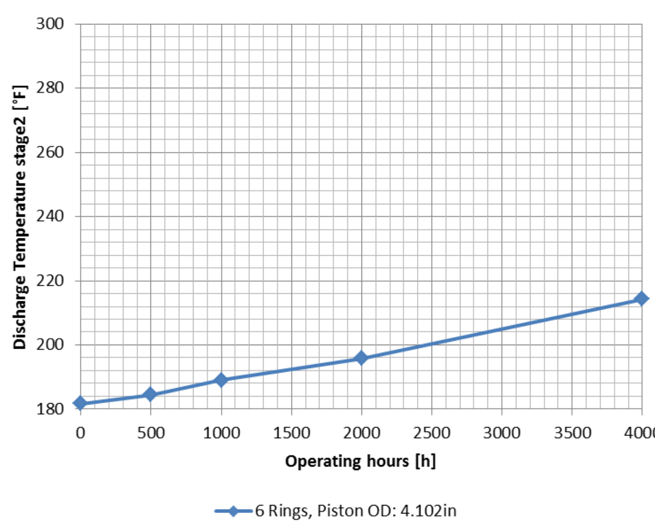


Figure 17: Simulation result - Discharge temperature as a function of time on stage 2

The model was then used to evaluate different piston designs. Figure 20 and Figure 19 show the results of the performance simulation for both stages. The solid blue line gives the discharge temperature for different piston ring end-gap flow areas. The remaining the straight lines represent different piston designs. These graphs allow determining the piston design based on a required run time and maximum allowed discharge temperature. On the 1st stage a piston design with 6 piston rings, a liner to piston gap of 0.065in and pressure balanced piston rings was selected to give a maximum discharge temperature of 250°F after one year of operation. On the 2nd stage a eight ring piston design with 0.06in liner to piston gap was chosen (maximum expected discharge temperature after one year: 210°F).

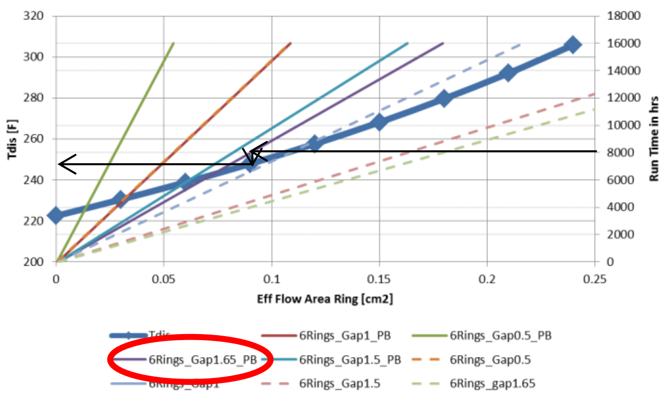


Figure 20: Temperature increase over time for different piston designs (stage 1)

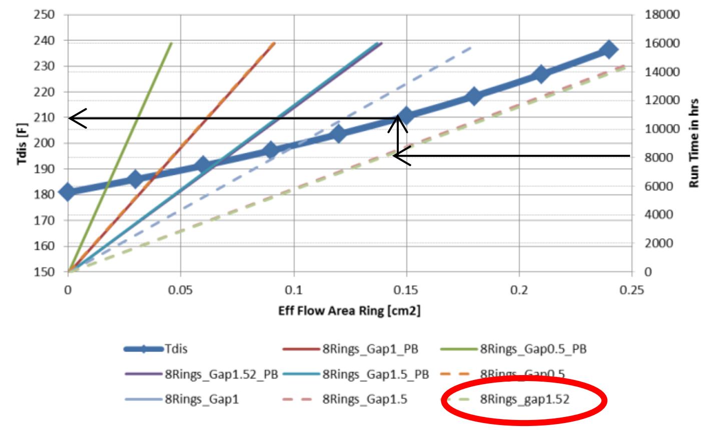


Figure 19: Temperature increase over time for different piston designs (stage 2)

Based on the piston performance analysis, the pistons were designed (Assembly drawings see Figure 21 and Figure 22). Figure 24 and Figure 23 show the manufactured pistons with installed rider bands for 1st and 2nd stage. The pistons were installed and the first compressor with upgrades was started on 8/21/2015. As expected the discharge temperatures were significantly lower and the piston and cylinder rings are no longer the bottle neck on this compressor.

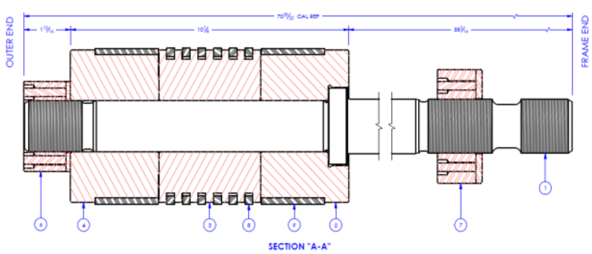


Figure 21: Drawing of new 1st stage piston

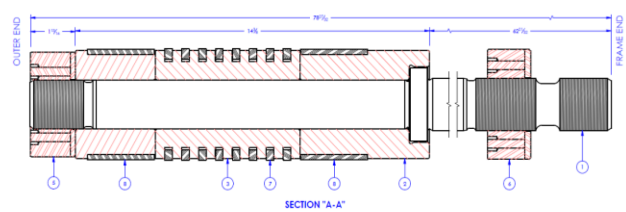


Figure 22: Drawing of new 2nd stage piston

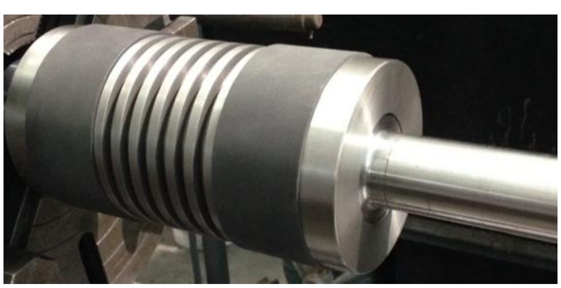


Figure 24: 1st stage piston

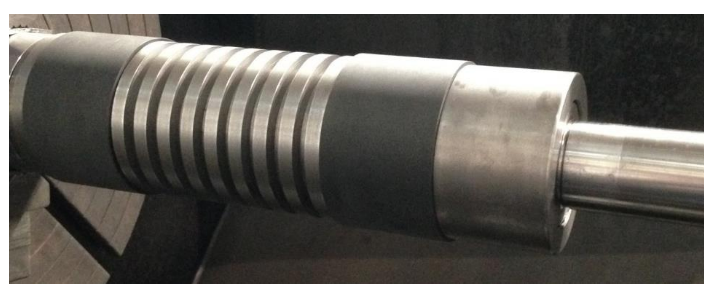


Figure 23: 2nd stage piston

CASE STUDY 3

Case description

The capacity on a single stage non-lube Hydrogen compressor decreased over time. The recycle valve (the primary capacity control system on this unit) had to be closed over time. At zero percent recycle valve opening the compressor had to be shut down in order to replace the wear components. A snapshot analysis revealed high piston blow-by and a piston performance analysis was conducted.

- Speed: 412 rpm
- Rated power: 170hp
- Cylinder bore: 6.5in
- Stroke: 9in
- Rod dia: 2in
- Gas: Hydrogen
- Isentropic exponent: 1.4
- Molar mass: 2 kg/kmol
- psuc = 395psig
- pdis = 810psig

Figure 25 shows the original piston design with four standard angle cut piston rings and two (two inch wide) rider bands. Figure 25 shows the performance result of the current piston. After 8000 hours of continuous operation the expected effective leakage area on the piston rings is 0.125cm^2 (0.02in^2) which corresponds to capacity losses of almost 15%.

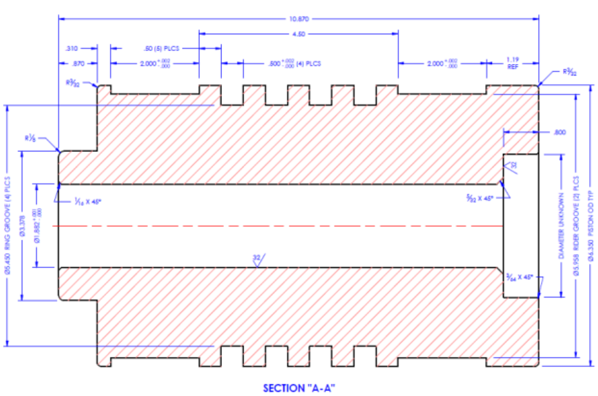


Figure 26: Original piston design

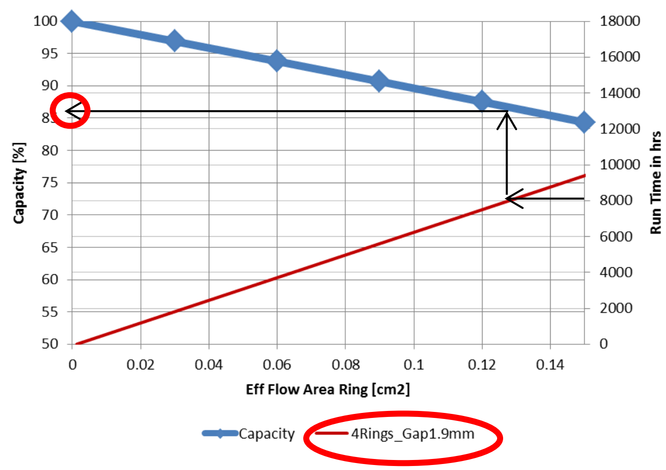


Figure 25: Performance graph original piston design

In order to reduce the capacity losses the performance of different piston designs was modelled and compared. Figure 28 shows the result for six piston ring (increase by two), pressure balanced rings and different liner to piston gaps. A liner to piston gap of 0.055 in was chosen resulting in an expected capacity loss after one year (continuous operation) of 3% (as compared to 15% before). Figure 27 shows an assembly drawing of the new piston including the six piston rings wider rider bands. The piston was installed in 2016 with the expected result of reduced capacity losses. The sister machine has since been upgraded.

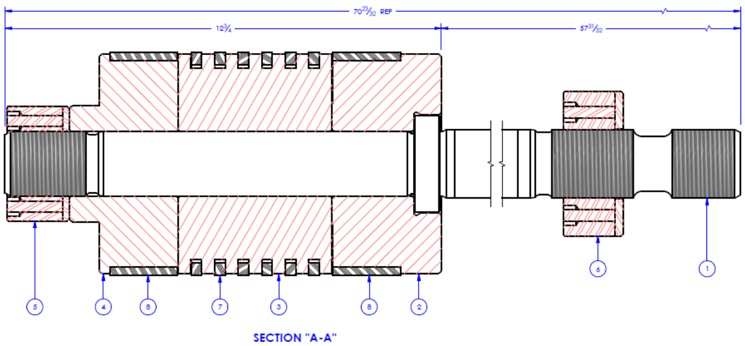


Figure 27: New Piston design

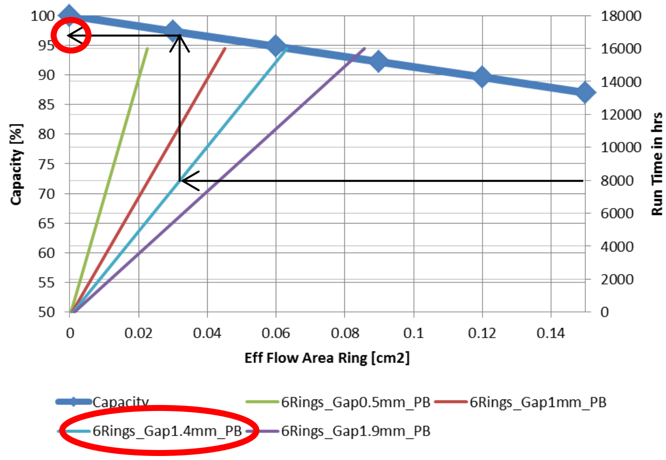


Figure 28: Performance graph for new piston design

CONCLUSIONS

Piston performance modelling has identified several deficiencies in piston design and cylinder ring selection. Utilizing the modelling capabilities optimizes piston design resulting in:

- Substantially decreased discharge temperatures.
- Increased unit capacity.
- Greater MTBF on wear components.

The complexity of piston performance modelling is moderate and can be included in everyday engineering work with a focus on problem cases and technical challenges.

The past years have shown that not all applications are equally prone to increased piston ring leakage. Cases with a high risk of high blow-by and subsequent risks of premature failure include:

- Non lube machines
- Cylinder sizes < 10in
- Suction pressures above 300psig
- Compression ratios > 2

NOMENCLATURE

h	= Enthalpy	(J/kgK)
$\dot{m}$	= Mass Flow	(kg/s)
$\emptyset$	= Effective flow area	(m^2)
p	= Pressure	(N/m^2)
ρ	= Density	(kg/m^3)
κ	= Isentropic exponent	
V	= Volume	(m^3)
dV	= Incremental change in volume	(m^3/s)
$d_e m$	= Incremental mass flow into or out of the considered volume (kg/s)	
i	= chamber upstream of the considered chamber	
j	= considered chamber	
k	= chamber downstream of the considered chamber	
w	= wear	(m)

g	= leakage gap	(m)
d	= diameter	(m)
k	= wear coefficient	(m^3/Nm)
$p_{contact}$	= contact pressure	(N/m^2)
v_{rel}	= sliding velocity of ring against liner	(m/s)
t	= time	(s)
C_D	= discharge coefficient	
CE	= Crank End	
HE	= Head End	

FIGURES

REFERENCES

Brandl, A, 2010, Efficient cooling in cylinder heads for air brake compressors. International compressor engineering conference at Purdue.

Radcliffe, C. D., 2001, Development of piston rings for reciprocating compressors. IMechE Paper Number C591/057/2001, IMechE 2001.

Feistel, N. Influence of piston ring design on the capacity of a dry-running Hydrogen compressor, Burckhardt publication.

Liu, Y., Yongzhang Y., 1986, Prediction for the Sealing Characteristics of Piston Rings of a Reciprocating Compressor. International Compressor Engineering Conference, Purdue University 1986.

Goebel, D., 2014, Reciprocating compressor suction and discharge valve monitoring. COMPRESSORTech2, June 2014.

Disconzi, F.P., Deschamps C.J., Pereira E.L., 2012, Development of an In-Cylinder Heat Transfer Correlation for Reciprocating Compressors. International Compressor Engineering Conference, Purdue University 2012.

Ruddy, B.L., Dowson, D., Economou, P. N., 1981, The prediction of pressures within the ring packs of large bore diesel engines. Proceedings of the IMechE Journal of Mechanical Engineering Science, 1981, 23(6), 295-304.