Experimental and numerical investigation on the sealing capability of a high pressure, dry running, hydrogen reciprocating compressor

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Abstract:
Green energy transition and decarbonization commitment require to design innovative machines, to successfully handle multiple fuels and accommodate an ever increasing quota of renewables. In the mid-long term, the hydrogen is expected to play a key role both as a fuel in the long range mobility context, as well as an energy vector within an innovative industry panorama, traditionally hard-toabate. Several technical applications require a high degree of gas purity, thus the development of dryrunning compressors, at high pressure levels, becomes a challenging objective that requires proper investigation and design tools. During compression and expansion strokes, large non-uniformity of the pressure distribution in the sealing system may lead to premature excessive wearing of the piston rings within the reciprocating compressors. Accurate experimental campaign and numerical computations are needed to obtain a reliable digital twin of the capacity of a newly designed high pressure, dry running, hydrogen compressor, during its lifetime. In this work, the sealing effectiveness of specialized piston rings, ad-hoc developed for a newly designed high pressure hydrogen compressor to avoid severe leakage and excessive wearing, is discussed by a proper test model, taking into account numerical predictions and experimental results. The proposed combined theoretical, experimental and CFD approach allowed to deepen the behavior of newly designed rings, showing their superior performance for high pressure, light gases applications, when compared with more conventional designs.

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1 Introduction

The challenging climate-neutral targets set by the EU for 2050 demand for active and innovative pathways to be explored and this entails that all parts of society and economic sectors must play a role. Ever increasing quota of energy from renewables, reduced pollutants, usage of "green" fuels are a bunch of opportunities that need to be pursued to carry on a global climate action. In this perspective, clean hydrogen shows a great potential in reducing direct emissions of pollutants avoiding greenhouse effect. Hydrogen[1] is expected as breakthrough in achieving a clean, secure and affordable energy future, mainly in strategic sectors such as transport and power generation[2].

Hydrogen is mainly required to be delivered with the highest degree of purity, and clearly handled and stored at very high-pressure levels. Oil-free compressors are needed in several processes when ensuring high-performance operation and high-quality products becomes essential. In this operating framework, dry-running compressors have to cope with very severe operating conditions and need to be suitably devised to ensure safety and reliability. Sealing capability of the piston rings pack becomes cornerstone along with the prediction of its expected lifespan which clearly is a relevant issue to deal with.

In this work a combined numerical modeling and experimental investigation approach is presented to evaluate the sealing capability of specialized piston rings (coded 262 NN) designed to minimize the flow leakages in piston ring packs of reciprocating compressors to be installed in a new HP H_2 compressor. The 1D numerical approach is first described and validated with an ad-hoc experimental campaign led at the Energy Systems and Turbomachinery laboratory of the University of Bergamo (UniBg Est Lab). Further study on the fluid dynamic phenomena occurring in the 262 NN rings is also presented by means of high-resolution CFD computations. The 1D H_2 dry code implemented has been developed to predict the compressor performance in terms of mass leakage, inter-ring pressure, wear quantification during operations. It has been realized as a complementing tool for compressor companies to promote the development of a reliable digital twin of a newly designed compressor, able to predict the sealing capability of the machine operating with hydrogen at high pressure loads. Description of a novel H_2 compressor design is also provided. Few concluding remarks are finally reported.

2 Dry running hydrogen reciprocating compressor: test bench

Reciprocating compressors in their multi-stage configuration can achieve high level of performance in terms of discharge pressure and offer also high degree of flexibility in terms of size and capacity. However, the recent requirement of high-pressurized hydrogen has pushed forth a number of technical challenges, *i.e.*, i) high pressure loads with, ii) H_2 embrittlement material selection robustness, iii) light weight gases, using iv) self-lubricated systems, that must be properly faced in order to ensure good operating conditions. Complex unsteady flow phenomena, relevant mass leakages, increased thermal heat transfer, occurring when the machine is self-lubricated, are expected to play a key role in the definition of the overall performance. Hydrogen compressors manufacturers have to extend their design experience through innovation and research. In the activity's framework required by the development of high-performance hydrogen compressors, a roadmap to define an innovative, digital-oriented, design of the machine has been drawn. In this perspective, the sealing topic is cornerstone and entails to select a high-level partner, developer of specialized solutions in sealing systems for high performance compressors.

Early machine design relying upon the well-established know-how of an experienced compressor company is performed employing ad-hoc devices and materials selected for their enhanced capabilities and properties when applied to hydrogen compression. Figure 1 shows the new compressor, also developed taking advantage from the results of this research. It is four stages, single effect V-type compressor. First and second stages are crank-end (CE) effect while third and fourth stages are head-end (HE) effect. Maximum discharge pressure is about 450 barG. Compressor valves and sealings (piston/packing) are dedicated design. At present, the compressor is successfully running under performance condition at a company workshop. In the upcoming months it will be placed on duty at a hydrogen production plant.

As already indicated, a deeper investigation is necessary to carry on the compressor design and testing phases when self-lubricated systems are considered. Proper tools and procedures must be defined and, owing to the challenging task, a multi-disciplinary approach is envisaged, and herein pursued by a combined experimental and numerical approach as described in the following sections.

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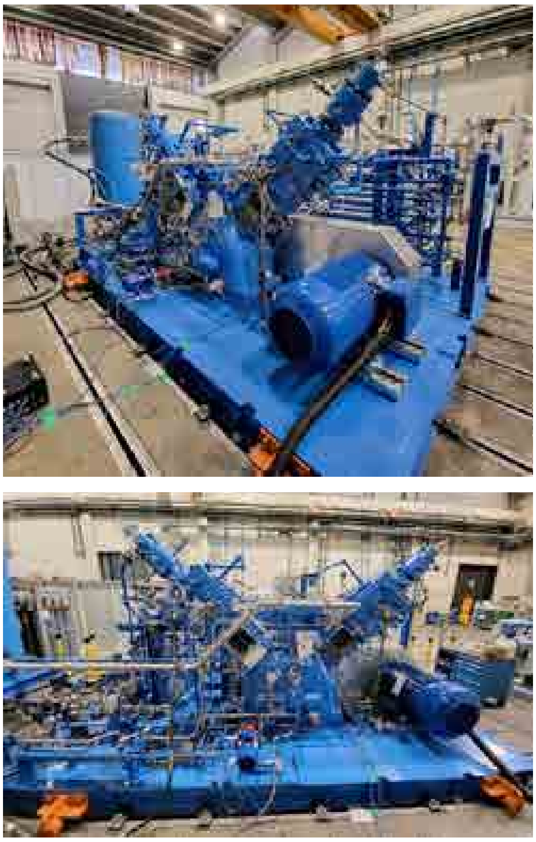

Figure1:H_2 HP reciprocating compressor.

3 Sealing performance 1D modeling

Any modeling approach adopted to predict the performance of a compressor requires to reliably define the mass leakage amount and the acting pressure on the sealing system.

The description of the fluid dynamic behavior within a sealing pack system requires to define hypothesis both from the geometric and physical point of view. First, we identify possible leakage passages within the cylinder-piston-rings pack, in particular, the gas may flow within gaps located between: i) rings and internal surface of the cylinder; ii) rings and lateral face of the piston ring groove; through iii) cuts of the rings.

From the point of view of physical modeling, several reduced-order discretization approaches have been presented in the open literature which can be roughly divided into two groups: a) labyrinth-like structure with fluid modeled as an isentropic flow through convergent nozzle, see *e.g.* the work by Furuhama and Tada[5] and b) labyrinth passages like orifice for end-gap regions and laminar compressible flow within ring-side clearance. Most of the models are related to internal combustion engine applications and therefore they are typically applied to lubricated sealing systems. The choice between a) and b) is a function of the flow regime assumptions and overall operating conditions. A mixed viscous and inviscid modeling is proposed by Flade[4] where a calculation of sealing performance is also presented to investigate the unsteady flow in reciprocating compressors. It is however suggested that the laminar viscous assumption in small gaps can be applied only when the ring is sitting at either the top or bottom surface of the groove[6]. Yu et.al.[8] presented a method to simulate the unsteady flow behavior within end-gap piston rings in a high-pressure non-lube hydrogen compressor, by means of 1D compressible isentropic flow modeling, considering only gas passing in the end-gap region. Braga et. al.[3] proposed a numerical analysis on the effect of the clearance geometry and piston velocity on leakages of an oil-free reciprocating compressor efficiency. The discretization is based on a simplified form of the Reynolds equation to approximate the pressure field and the mass flow rate is computed as a Couette-Poiseuille flow. More complex modeling approaches have also been presented in the open literature, by combination of the inviscid flow assumption in the end-gap region with a laminar viscous approximation in the ring-side clearance, see, *e.g.*, Kuo et.al. [7]

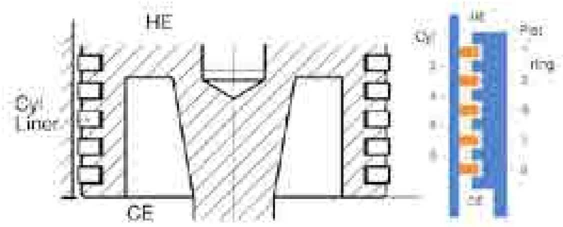

Figure 2: Conceptualization of the sealing system adopted in the 1D mode.

In the 1D sealing modeling code, herein presented, the discretization relies upon inviscid flow assumption through labyrinth-like structure. The flow field is discretized as 1D along the stroke direction and it is subject to the following assumptions: the fluid is in a single gaseous phase, the ideal gas equation of state is applied, the flow is considered as iso-thermal and the temperature of the gas assumed equal to the piston temperature; finally, the flow behaves as an isentropic flux through an orifice. Figure 2 displays the conceptualization of the sealing system through chambers and orifices from head-end (HE) to crank-end (CE).

The governing equation for each chamber reads

$$\frac{dP_i}{d\varphi}=\frac{R_g T_i}{\omega V_i}\left(\dot{m}_{i-1}-\dot{m}_i\right), \tag{1}$$

where omega defines the speed of rotation, V the volume, P and T the pressure and the temperature values at chamber i.

The governing system describes the unsteady pressure evolution within the operating cycle. At each angular position φ, an implicit system arises, since the mass flow rate is a function itself of the

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overall pressure distribution $\dot{m}_i = f(P_i)$. Therefore, a proper linearization procedure must be performed before computing the pressure values. Mass flow is computed as a function of the regime: at total fixed upstream condition (pressure and temperature), when the critical (sonic) condition is achieved,

$$\beta > \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)} = \beta_{cr}\,, \qquad (2)$$

the flow is choked and the mass flow is constant and computed as

$$\dot{m}_i = K_C A_i \left[\frac{2\gamma}{R_g(\gamma-1)}\right]^{(1/2)} \frac{P_i}{T_i^{1/2}} (\beta_{cr})^{(1/\gamma)} [1 \,. \qquad (3)$$

Otherwise the mass flow rate is obtained as:

$$\dot{m}_i = K_C A_i \left[\frac{2\gamma}{R_g(\gamma-1)}\right]^{(1/2)} \frac{P_i}{T_i^{1/2}} \left(\frac{P_{i+1}}{P_i}\right)^{(1/\gamma)} \Bigg[\qquad (4)$$

A key issue in the mass leakages computation is the discrepancy between the maximum mass flow rate (Eqs. (3) and (4)) and the actual one, which is also influenced by viscous phenomena. In order to account for this effect, the discharge coefficient K_C is typically employed. In the simplest definition it can be assumed equal to a constant value as for example shown by Furuhama and Tada[5] which indicates that K_C can be safely fixed in the range 0.8-0.9. Another, widespread K_C approximation is provided by a quadratic function of the local pressure ratio acting on a selected chamber $\beta_i = \frac{P_i}{P_{i+1}}$, namely $K_C = 0.85 - 0.25\,\beta_i^2$. This coefficient is an important parameter to characterize the sealing capability of piston-rings, that allows to compare different geometries, flow of different gases operating at different temperatures and pressures. This topic will be addressed in the following sections.

The ring dynamics is modeled by assuming that the ring side and groove are flat and axisymmetric, and therefore only the ring axial movements are considered. By using a balance of acting forces, provided by the simple Newton's law of motion, the ring position is established.

The discretization herein proposed allows to account for unsteady phenomena, such as the time-lag occurring in terms of chamber pressurization. In Figure 3 the 1D model is applied to a three-rings test case with imposed HE and CE pressures. The computed inter-ring pressure distribution in chambers labeled as 2 and 4 is reported as a function of the crank angle. Notice a ring has changed its position within its groove when the pressure distribution highlights a curvature discontinuity. Furthermore, a time-lag establishes since, in each chamber, the maximum P value is reached at different positions.

Figure 4 shows the results obtained for a validation test case selected in Furuhama and Tada[5] investigating the mass leakages in a two rooms system toward the crank case. Under imposed HE pressure load, the inter-ring pressure distribution within rooms herein named 2 and 4 (according to the nomenclature of the present discretization) are computed for a shaft revolution. The results highlight good agreement between the 1D predicted profiles and the reference levels, both in terms of functional distribution and maximum pressure occurring.

Further validation of the model capability against experimental data and high-resolution simulations is presented in the next section.

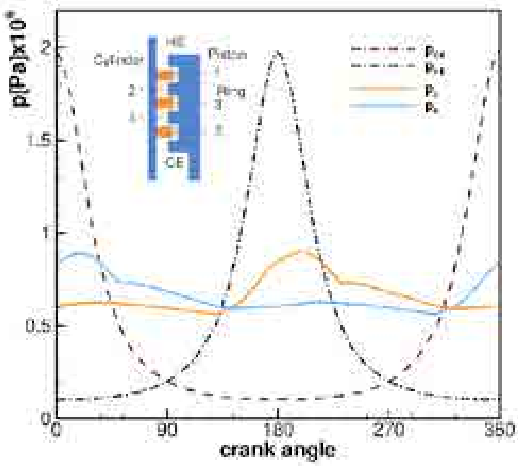

Figure 3: 1D modeling of a three rings system - double effect compression: inter-ring pressure distribution of chamber labelled 2 and 4 (end gap regions) due to imposed HE and CE pressure distributions.

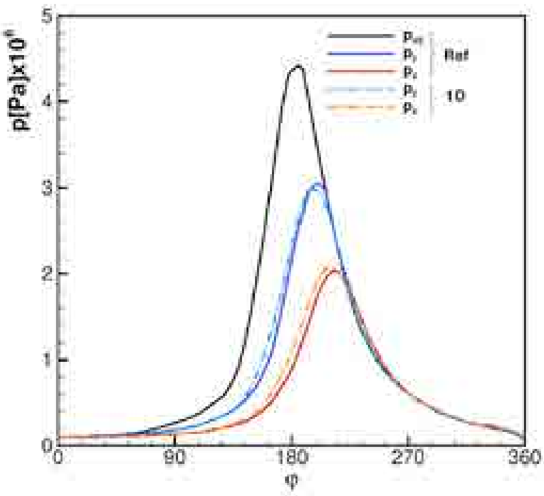

Figure 4: 1D Modeling validation: inter-ring pressure distribution computed in comparison with reference data.

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4 Modeling validation: test rig measurements over model cylinder and CFD computations

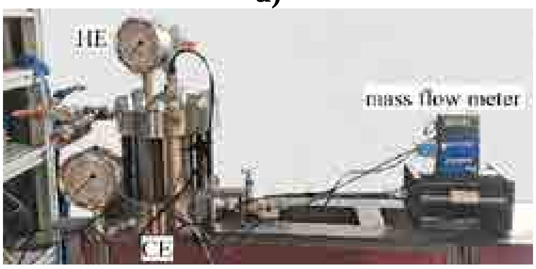
This section is dedicated to the validation and discussion on the 1D code capability in predicting the sealing performance of the selected piston-rings.

4.1 Test rig measurements

A test rig at the UniBG Est Lab consisting of a single piston of characteristic dimension D=125 mm equipped with two rings has been used to characterize the behavior of sealing devices featuring different design, subject to varying operating conditions and different gases. The cylinder test cases were monitored by standard control and measurement devices, two pressure transducers and a mass flow meter as shown in Figure 5(a). The cylinder model may be equipped with up to two 125 mm diameter rings. Detail of the inner core of the model is provided in Figure 5(b).

The mass flow meter can deal with clean gases up to $1000\, Nl/min$ and it is characterized by an accuracy of $\pm 1.5\,\%$ full scale.

a)



b)

Figure 5: Test bench at UniBG EST Lab (a); detail of the sealing apparatus (b).

A thorough measurement campaign has been carried out to evaluate the sealing capability of the piston rings operating at high pressure. The test rig corresponds to a static configuration of the real machine. Tests have been carried out by imposing the total inlet conditions and subsequently modifying the static outlet pressure. Data acquisition has been done for the steady configuration for each pressure ratio considered.

In particular, in this work we considered standard sealing devices, such as the 45deg inclined cut and the step cut, along with the specialized rings specifically developed do deal with HP H_2 compressor, called for the sake of brevity 262 NN, see Figures 6 and 7.

Figure 6: Piston rings tested: 45deg inclined cut, step cut, 262 NN.

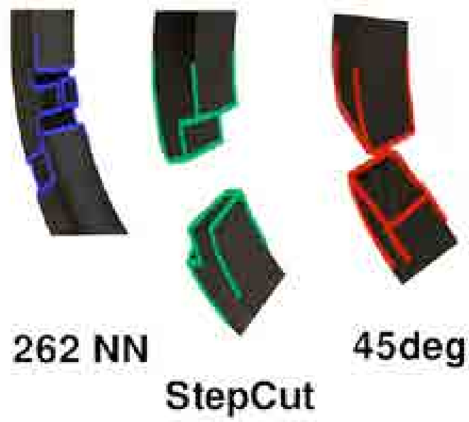


Figure 7: Piston rings tested end gap detail.

The 262 NN ring has been designed to handle light gases compression at very high-pressure levels. They have been conceptualized to enhance the flow blockage within the ring packs by devising complicated flow passages and using high-quality dedicated materials. Effective sealing at low and high pressure loads is ensured by means of a mixed radial and axial design of narrow flow passages using a multi-pieces assembly. The 262 NN piston ring consists of paired two-piece outer rings and a single one-piece inner ring. The two outer rings include an anti-rotation tongue that is milled into the ID of the rings and fits into the clearance gap of the inner ring. This keeps the clearance gaps of the outer rings at opposite orientations to prevent any direct flow path through the assembled piston ring.

This ring was developed for small-bore high pressure applications, where the milled feature provides a more robust anti-rotation method than a dowel pin. Also, the segmented assembly can be installed on a solid piston.

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The test cases have been performed at fixed inlet total conditions, by varying the static outlet pressure value. At each pressure ratio β, the HE and CE pressure values and the mass flow rate data acquisition has been performed at 1kHz, with measurements run as soon as the steady state condition was achieved in every case. The data have been averaged over one thousand samples and then analyzed to obtain inlet and outlet pressures and corresponding mass flow leakage rates.

Figure 8 displays the non-dimensional mass flow rate for the three different geometries, scaled with respect to the maximum value recorded. Helium was the working gas. Such measurements highlight how the sealing capability scales with varying pressure loads and how it varies with different installed rings. Very good sealing capability is ensured by the 262 NN rings.

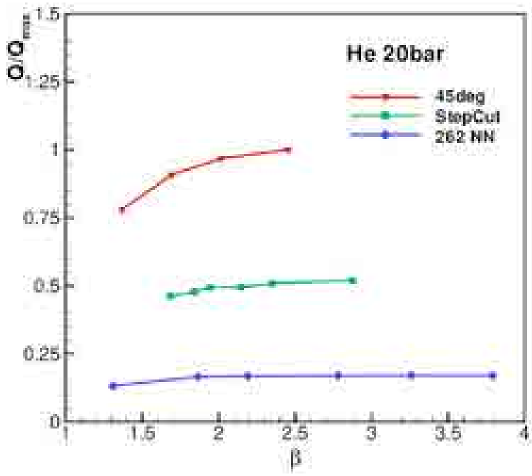

Figure 8: Experiments: non-dimensional mass flow rates measured as a function of the compression ratio, helium at 20 barG inlet total pressure imposed.

The definition of the discharge coefficient is a scaling of the mass flow passing within the rings, a general approach that requires an accurate evaluation of the maximum flow rate admitted by each configuration, *i.e.* accurate definition of the isentropic, adiabatic flow rate as given by Eqs (3) and (4). This target is not straightforward at all, the actual flow is only approximately adiabatic, real fluid effects take place, compressibility also plays a role, furthermore, an accurate measurement of the passage area is also required. This last parameter is not trivial since during operations different flow passages may establish. To draw considerations in terms of a general discharge coefficient we used a "comparative approach" useful both to estimate the passage area A and to validate the 1D prediction against measurements.

The comparison between the 1D mass prediction and experimental data stems into the results highlighted in Figure 9 which displays, as an example, the non-dimensional mass flow rate values obtained for a sealing system equipped with a 262 NN ring operating with helium at 20 barG. The numerical calculations have been performed by prescribing the experimental boundary data, *i.e.*, pressure loads and using as passage area the value derived by targeting the measured choked mass flow. The measured and numerical mass flow rate trends appear in very good agreement with one another.

Using the choke condition as target state, an estimation of the passage area is thus obtained for all tested geometries and then adopted to define the K_c distribution shown in Figure 10. Also, these results indicate the good sealing capability of the 262 NN device.

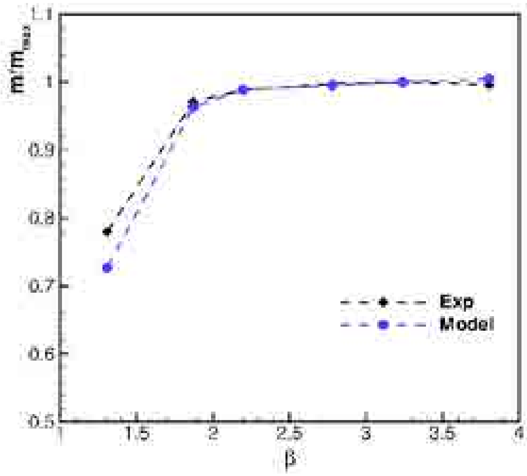

Figure 9: 1D model vs experiments, 262 NN ring operated at 20 barG with helium: non-dimensional mass flow rate as a function of β.

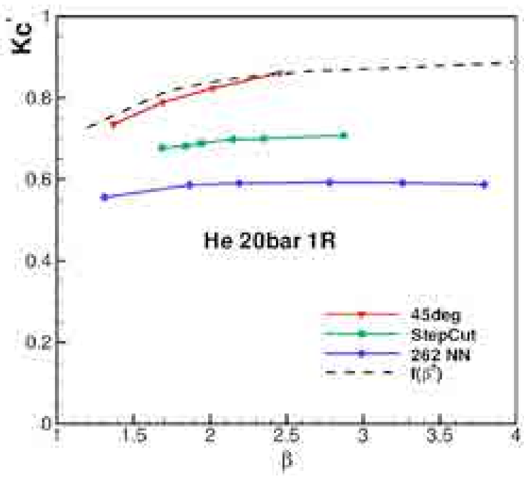

Figure 10: Experiments: K_C values of 45deg, step cut and 262 NN rings operating at high pressure with helium.

Finally, Figure 11 reports the discharge capability of the case with two specialized HP rings in comparison with the most common 45deg rings when operated at HP with such a low molecular weight gas. As it readily appears, better sealing is still highlighted by the 262 NN elements when operated at 20 bar gauge. The measured leakage also suggests that the specialized piston rings can properly cope with increasing pressure loads imposed at head-end. Worth to be noticed, such a peculiar behavior when considering the 262 NN sealing rings, the flow factor in fact decreases whilst the prescribed pressure at HE is increased from 20 to 80 barG. This may even be a matter of the actual passage area, though fair comparison at fixed passage area, suggests the above commented characteristic. Figure 12 displays the sealing behavior measured when using the model cylinder equipped with two 262 NN rings operated at imposed 20 to 80 bar gauge at head-end, using helium or nitrogen. The comparison shows that the predicted K_C value obtained using the helium is equal or higher than using nitrogen. In this case, the lower molecular weight appears to play a major role as the pressure increases.

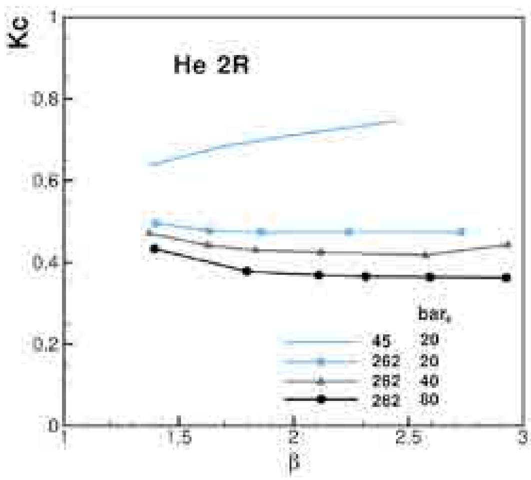

Figure 11: Experiments, 45deg and 262 NN rings operated at 20, 40, 80 bar$_g$ with helium using two rings.

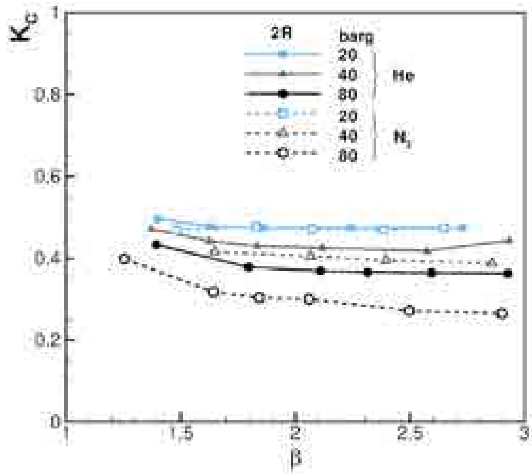

Figure 12: Experiments, 262 NN rings operated at 20, 40, 80 bar$_g$ with helium and nitrogen using two rings.

4.2 3D CFD simulations

Further study on the complex behavior developing within the considered D125 piston equipped by two 262 NN rings is provided by means of high-resolution CFD computations. In particular, 3D steady-state computations, based on a second-order accurate discretization, have been carried out by means of the numerical suite StarCCM+ by Siemens. Proper sensitivity analysis on the mesh coarseness has been performed (not reported here for the sake of brevity) and the final computational grid consists in 21 million polyhedral elements. Convergence of the computations was achieved with scaled residuals lower than 10^{-6}, also evaluated by monitoring the variation of relevant physical values, *e.g.* mass flow rate balance. The computation has been done imposing the inlet total conditions and the static outlet ones according to values prescribed during experiments. Static operational behavior has been investigated by varying the imposed static pressure at the outlet to match the selected pressure ratio.

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Figure 13: 3D CFD: streamlines path within inter-ring flow passages at β=2.2.

Figure 13 displays the simulation result at β=2.2 showing the complex streamlines paths originated at the narrow gaps and chamfers of the ring. The computed non-dimensional inter-ring pressure contours are displayed in Figure 14 showing how the first ring bears most of the pressure load. Notice that these contours may be helpful to evaluate the wear status and to introduce information needed to perform a lifetime prediction of the machine.

4.3 Discussion of results

Finally, comparison in terms of sealing capability is provided by Table 1 reporting the percentage deviation of discharge capability predicted by CFD against measurements assumed as a reference. The passage area adopted in the simulations corresponds to the end gap area (this quantity has been employed to computed the discharge coefficient). As shown, the results highlight a good mutual agreement. Possible reasoning behind small discrepancies may be found in the fact that steady-state simulations only accounting for the CAD "end-gap" region have been led, whilst possible alternative leakages may occur otherwise.

Final remarks are devoted to the comparison of the 1D prediction values against experiments and 3D CFD simulations. The former one (1D vs exp) gives rise to a perfect agreement, with a relative error of order lower than 1%. This is directly related to the ad-hoc tuning procedure used in the code against the available choked reference condition. Instead when comparing 1D vs 3D results, we had a different outcome. We first remind that in this case, the passage area considered is the one adopted in the computational domain, i.e., the CAD area. We observe that larger discrepancies arise, amounting to a relative error on K_C equal to about 10% (see Table 2). This difference is clearly related to the missing viscous contribution underlying to the 1D discretization employed. However, the trend of the discharge coefficient is perfectly captured. Furthermore, good agreement is also obtained in the evaluation of the inter-ring pressure level, for which few points per cent of relative difference have typically been observed (not reported here for the sake of brevity).

β	**CFD vs. Exp**
1.8	-2.56%
2.0	-2.69%
2.2	-0.61%

Table 1: Two rings 262 NN: Exp and CFD discharge coefficient for different pressure ratios.

β	**1D vs. CFD**
1.8	+7.26%
2.0	+9.89%
2.2	+8.61%

Table 2: Two rings 262 NN: 1D and CFD discharge coefficient for different pressure ratios.

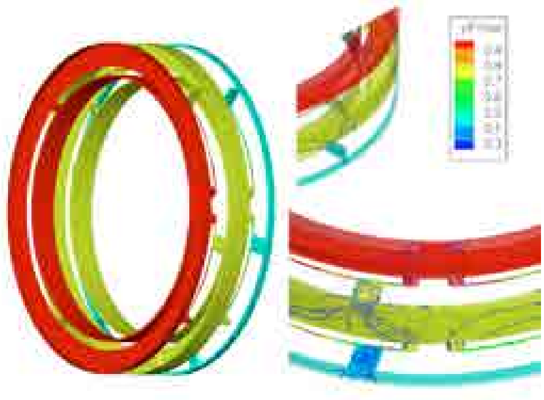

Figure 14: 3D CFD: streamlines path within inter-ring flow passages at β=2.2.

5 Conclusion

Theoretical, experimental and numerical activities have been adopted in the presented research in a synergistic way to reliably predict gas leakage when operating at very high pressure with light gases. The effect of gas blow-by on the ring's tribological behavior still needs to be investigated, as well as the ring's dynamic motion within its pertaining groove to also predict ring wear. Optimization of geometrical parameters and shapes of labyrinth seals to minimize the leakage is a further research topic for future investigation.

Looking at the H_2 HP dry compressors growing market, sealing rings of enhanced capability, like

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the 262 NN model, are requested. The efficiency of this kind of rings has been investigated and compared to most common solutions and the results validated by lab experiences and a CFD study.

A 1D numerical model that can perform a sealing capability estimation under imposed pressure load has been developed, herein presented and validated. The 1D model will be useful to predict pressure across each sealing ring, the leakage flow amount and, through data coming from ad-hoc wear tests, the numerical tool is expected to give an estimation of the working lifespan.

Activities are still ongoing to have a better modelling and to fit the model against real cases like the field data coming from the ongoing testing of the new H_2 HP dry compressor delivering hydrogen at 450barG.

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