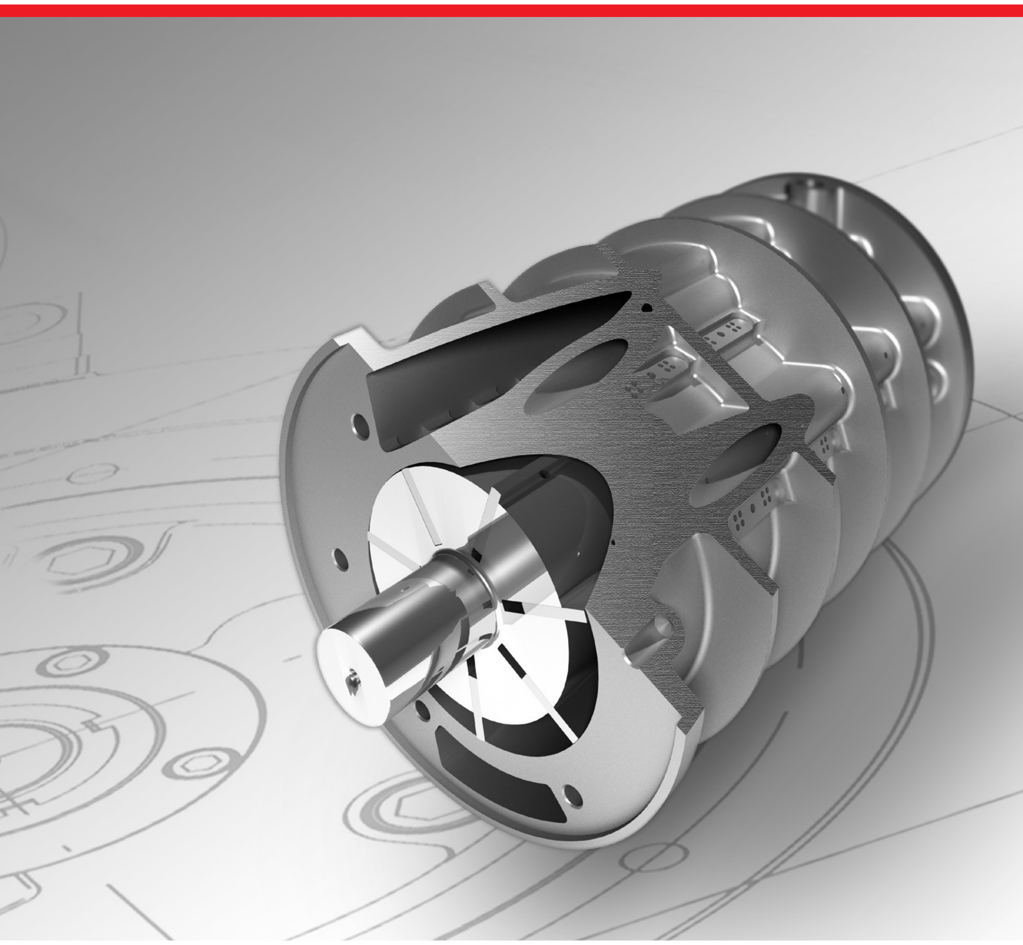

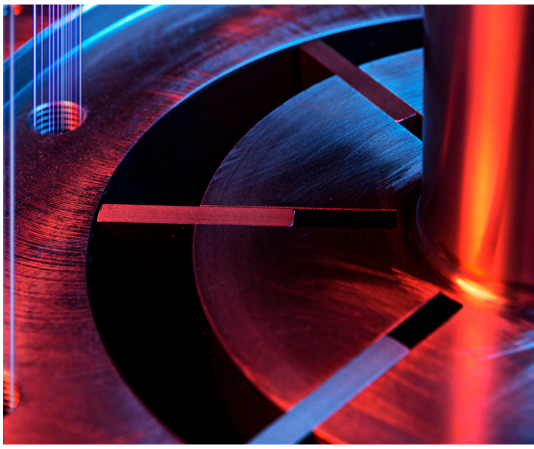

AIR COMPRESSORS

THE REAL ECONOMIC AND ENVIRONMENTAL IMPACT OF USING
THE CURRENT INDUSTRY STANDARD LIFE CYCLE COST ANALYSIS

ABSTRACT

Today's commonly accepted method of calculating Life Cycle Cost (LCC) for industrial Air Compressors is challenged due to the fact that assuming constant compressor efficiencies through the life of the compressor is incorrect. It is shown how, in a screw compressor, very small interlobe and axial clearance difference results in important performance losses and how these changes are well within the acceptable wear of the bearings used in the same compressors. It is also shown how in the Mattei Rotary Vane Compressor, there is a short running-in period, lasting roughly 1000 hrs, in which improvements of up to 5% in performance have been measured both in house and by the third party tester INTERTEK USA. With this data in hand, more realistic Life Cycle Cost calculations for Screw and Vane compressors are presented, highlighting how relying on Zero Hour CAGI verified performance sheets can be misleading for the end user. In the example provided, over a 10-year period with compressor overhaul, identical Zero Hour Specific Energy Compressor could end up having a difference in LCC of €168,000. It also demonstrates that adopting Zero Hour performance as a full life indicator of performance, when drafting new compressor industry legislations aimed at curbing the current global warming crisis, will put the expected energy consumption reduction targets at risk in this industry.

LIFE CYCLE COST AND INDUSTRIAL COMPRESSED AIR

Throughout industrial manufacturing, LCC is a well-recognised method to simulate the full cost of ownership for capital equipment. The calculation of the LCC for an industrial machine will vary from industry to industry, and in the Air Compressor Industry it is typically calculated by taking into consideration three major factors.

Capital Equipment Expenditure (Capex) – What is the cost of acquiring the equipment? If the LCC exercise is being run to compare two Air Compressors from competing brands, this will include only the compressor cost (as in this example). If the LCC exercise is being run to calculate full return on investment then installation and ancillary costs will also be take into account here.

Ordinary Maintenance Costs – What is the cost of maintaining the equipment? The manufacturer declared costs of maintaining the equipment regularly with the use of consumables, including the labour cost involved in the maintenance.

Energetic Consumption Costs – What is the cost of running the equipment? A simulation of how much the Air Compressor will cost to run. This depends first and foremost on the performance of the compressor and is typically measured by the number of kW needed to compress 1 m^3/min of air. This is known as the compressors Specific Energy. The Specific Energy can then be multiplied by the Free Air Delivery, the Operating Hours and the local cost of electricity, to have a complete cost of running the compressor.

Whilst Capex is fixed, both Maintenance costs and running costs will vary depending on a couple of factors, such as yearly running hours and local energy costs. LCC simulations become more common the larger the installed power of the compressor, and the larger the installed power, the longer the running hours in a year.

As an example that we will be using throughout this paper, let's consider an industry standard:

INSTALLATION PARAMETERS	DATA	UNITS
Compressor FAD	15	m^3/min
Specific Energy	**6,0**	**kW/m^3/min**
Acquisition Cost	50.000	€
Maintenance Cost	**4.000**	**€/Yr**
Operating Hours	8000	hrs/Yr
Energy Cost	**0,2**	**€/kWh**

Table 1 – Example of a medium sized Air Compressor installation parameters for LCC simulation

In this example the LCC for the 5-year life of the compressor is calculated in the following way:

LCC COST CENTER	METHOD	COST	% COST
Capex	Cost of acquiring the equipment	€ 50.000	6%
Maintenance @5yrs	**Yearly maintenance cost x5**	**€ 20.000**	**3%**
Energy Costs	Spec. En. per total hours per FAD per energy cost	€ 720.000	91%
	Total LCC	**€ 790.000**	**100%**

Table 2 – Typical LCC of a medium sized Air Compressor installation

EXAMPLE COMPRESSOR LCC

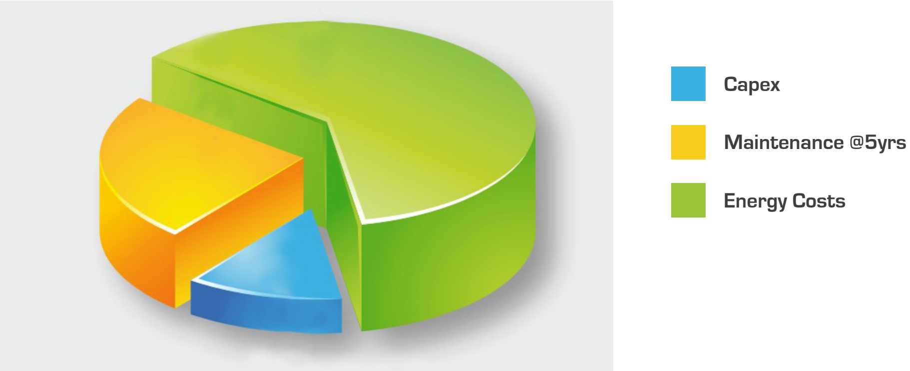

Fig 1 – Typical LCC of a medium sized Air Compressor installation

In this example the cost of running the compressor far outweighs the sum of the Capex and Maintenance costs, and makes up more than 90% of the overall LCC of this installation. The importance of the Specific Energy of a compressor has taken centre stage over the last decades due to these very high running costs. For this reason, many manufacturers have invested their R&D budgets to continually improve the performance of their products. Although the race to reduce the power consumption of air compressors has been typically driven by commercial aspects, nowadays, manufacturers of air compressors also have to face the fact that their products can directly affect the environment by means of their energy efficiency, and they must be held accountable for these performances.

AIR COMPRESSORS AND GLOBAL WARMING

Today, it is a widely accepted fact that global warming is currently the greatest threat to our Planet and the continued existence of Humanity. One independent study performed by the Intergovernmental Panel on Climate Change confirms that at the present greenhouse gas emission rate, a warming exceeding 4°C of the average global temperature by the end of the century is going to be unavoidable [1]. To the layman, this may not sound like much, until one considers that experts unanimously agree that a 2°C temperature increase is the limit to avoid irreversible damage to climate systems and to prevent the global socio-economic models from collapsing. On the 12th of December 2015, 195 nations approved a landmark climate accord in Paris, committing to addressing, with aggressive measures, the global warming crisis. Although the Paris Agreement is certainly a historic one, it will not, on its own, solve global warming. The best case scenario is that it will cut global greenhouse gas emissions by about half of what is necessary to avoid an increase in global temperatures of 2°C [2].
A major contributor to the greenhouse gas emissions scenario, and therefore the global warming emergency, is global electricity consumption. A detailed representation of the electricity consumption by sector is shown in [Fig 2].

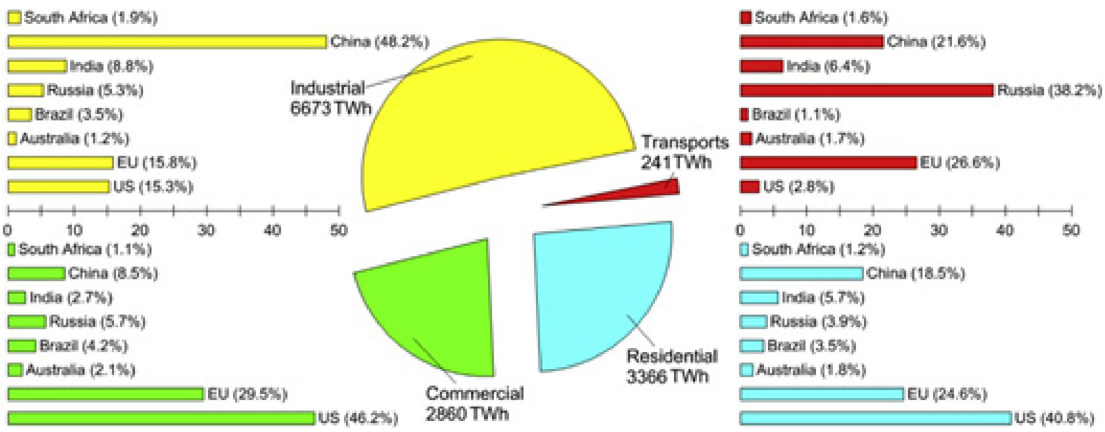

Fig. 2 – Electricity by sectors (year: 2013) [3]

The industrial sector accounts for more than 50% of the global electricity consumption and of this, up to 20% (i.e. 1335 TWh/y) is due to air compression and delivery to final uses [4]. Considering that the current global Rotary Air Compressor market is estimated to be worth USD 16 Billion and is expected to grow at a CAGR of 3.6% over the next 7 years, it is clear why energy saving or energy recovery in industrial Compressed Air Systems is considered an important issue when developing a plan to reduce greenhouse gas emissions and curb global temperature rises.

This subject has become of great importance since the Ecodesign Directive 2009/125/EC, having identified the product group "Compressors driven by electric motors" as a priority group in the first iteration of the Ecodesign Working Plan (period 2009-2011), required the European Commission to present a study on Air Compressors and possible measures to improve their impact on the environment (Lot 31). The study is on-going and will most probably result in new legislation removing a large swathe of air compressors with poor Specific Energies from the global markets.

Although most of the major players in the industry have been successful in significantly reducing their Specific Energy over the last few decades, the typical calculation and comparison method that is used today as an industry standard to simulate lifetime energy consumption (and therefore Life Cycle Cost) of an industrial air compressor is intrinsically flawed, and this will be the focus of this article.

TYPICAL COMPRESSOR LIFE CYCLE COST EVALUATION

Currently a potential client, who is interested in buying an industrial air compressor, will typically ask the manufacturer for the compressor's technical data sheet to estimate the Life Cycle Cost of his investment over the next 5-10 year period. In the USA, the Compressed Air and Gases Institute (CAGI) has set up a consumer friendly portal from which one can download technical data sheets regarding air compressors of various global manufacturers. Any data published on the CAGI website has been independently verified and approved by a third party tester, INTERTEK USA, which will have tested the compressor under the current guidelines regulating air compressor performance evaluation defined in the International Standard ISO 1217.

As there tends to be scepticism around much of the published performance data of many global compressor manufacturers, the CAGI Datasheets have become recognised as the most precise and impartial way to calculate Life Cycle Costs in the industry, and in doing this CAGI has provided a valuable tool to protect end users.

Using the same principles as shown in Table 1 and extending the example to compare two Air Compressors with slightly different Capex and Maintenance costs but identical CAGI verified Specific Energies, would result in the following LCC simulations:

Installation Parameters	Data	Units
Compressor FAD	15	m^3/min
Operating hours	8000	hrs/Yr
Energy cost	0,2	€/kWh

Compressor 1

Specific energy - CAGI	6,0	kW/m^3/min
Acquisition cost	50.000	€
Maintenance cost	4.000	€/Yr

LCC Cost center	Cost
Capex	€ 50.000
Maintenance @5yrs	€ 20.000
Energy costs	€ 720.000
Total LCC	€ 790.000

Compressor 2

Specific energy - CAGI	6,0	kW/m^3/min
Acquisition cost	55.000	€
Maintenance cost	4.800	€/Yr

LCC Cost center	Cost
Capex	€ 55.000
Maintenance @5yrs	€ 24.000
Energy costs	€ 720.000
Total LCC	€ 799.000

Table 3 – Using typical LCC Simulation on two Compressors with identical CAGI verified Specific Energies but different Capex and Maintenance Costs.

In the above example, Compressor 2 costs 10% more in Capex and 20% more in Maintenance Costs than Compressor 1, but at identical CAGI verified Specific Energies, the resulting difference in LCC is only € 9,000 out of total of € 790,000 or 1,1%. The potential end user now believes that he has all the necessary data to make an informed decision on which Compressor to buy and install in his plant. Unfortunately this method relies on one key assumption that is fundamentally incorrect and can therefore result in potentially misleading end users in their decision making process.
The assumption is the following:

Air Compressor Specific Energy is constant over time.

This assumption cannot be applied either to Screw Compressors or Vane Compressors, for two very different reasons.

SCREW COMPRESSOR

To understand why the above assumption does not apply to screw compressors one has to examine the engineering principle. Rotary screw compressor design consists of a pair of meshing helical lobed rotors. The rotor shafts are supported by roller and thrust bearings and generally one rotor drives the other by means of the helical profiles.

During rotation the screw profiles uncover an intake orifice at one end of the stator, through which the air enters and fills the volume between the profiles. On the opposite side the profiles penetrate each other, thereby reducing the volume, which compresses the air until the delivery ports are uncovered. Lubricant is injected to seal, lubricate and cool the compressed air. The lubricant is subsequently removed in the reclaimer tank followed by a final coalescing element. The compressor is started and stopped through the system pressure switch set to the maximum and minimum settings [Fig 3], [Fig 4].

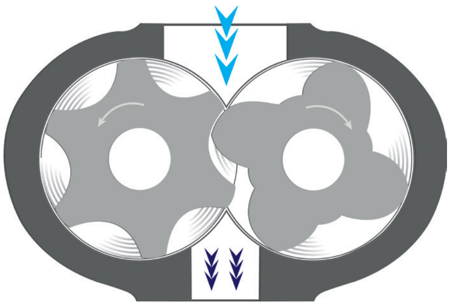

Fig. 3 – The rotors are fitted in a stator made from two cylinders that intersect longitudinally and in which the rotors turn with a minimum clearance.

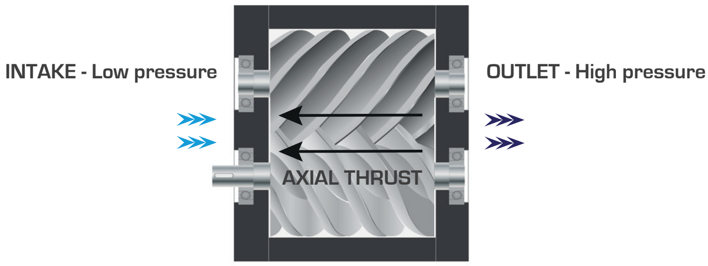

Fig. 4 – The intake and outlet ports are set at opposite ends of the compressor in the axial direction, giving rise to an unbalanced pressure profile along the compressor length

LEAKAGE PATHS AND CLEARANCES

To understand the important key role of clearances in a Screw compressor one must first have a clear picture of all the possible leakage paths. A cross section of a typical screw compressor, in which the leakage flow paths through the clearances are indicated, is shown in [Fig 5].

Of great importance for machine performance are both the clearance gap between the rotors (interlobe clearance), and the end clearance on the high pressure side (axial clearance). These leakage paths are connecting the high and low pressure working chambers, therefore the potential leakage is very high. The remaining leakage paths shown in [Fig 5]. involve smaller pressure differentials, and therefore are of lesser importance. The size of the radial and interlobe clearances is determined by the size and tolerances of the main compressor parts. The axial clearance is, however, set during the machine assembly [5].

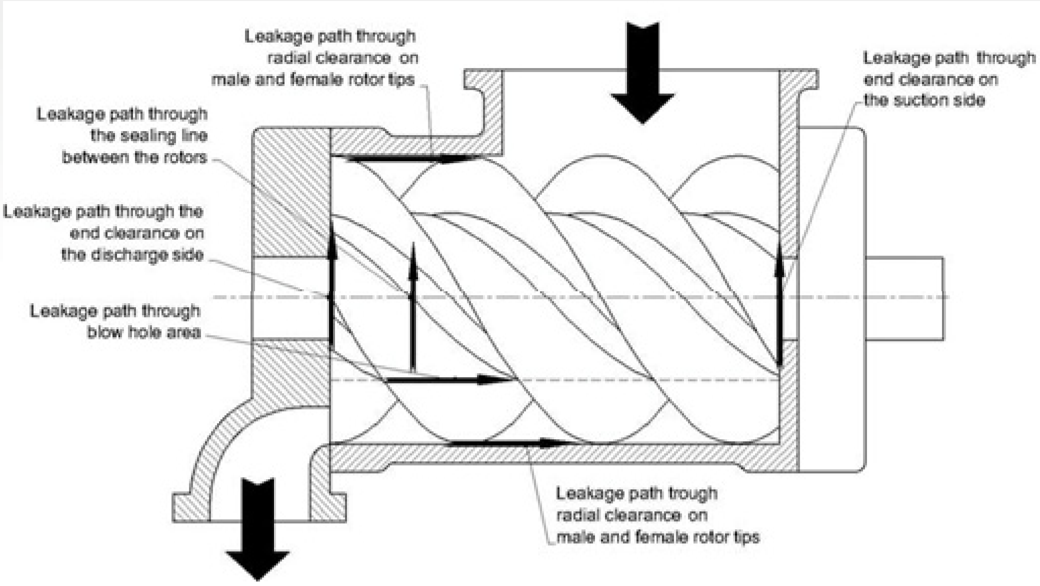

Fig. 5 – Leakage pathways in a screw compressor

Therefore, since the performance of screw compressors is highly affected by leakage, any modifications of the clearances within them must have an important effect on their efficiency [6].

INTERLOBE CLEARANCE AND ITS EFFECT ON SCREW COMPRESSOR PERFORMANCE

Today modern rotor machining centres have been shown to maintain extreme tolerances of up to 3μm. This means that, as far as rotor production alone is concerned, clearances between the rotors can be as small as 12μ [7] (as a frame of reference the average width of a human hair is 70μm).
Although this allows the reduction of the interlobe clearance and, as a consequence, improves the volumetric efficiency of the compressor, the clearances are now so small and actually comparable to the rolling element bearing clearances that they can in fact interfere in the reliable and efficient performance of the compressors.
The effect of clearance size and distribution has been studied thoroughly, highlighting the importance of these very small clearances. **It was shown that displacing the discharge bearings by only 50μm resulted in an important change in the specific energy of the compressor being examined of 2.5% (@1500rpm and 9 bar discharge pressure) [6]. It was also shown that increasing interlobe clearance by 31.5%, for example from 15μm to 20μm, led to a measured loss of volumetric flow of 1.7% [8].**

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AXIAL CLEARANCE AND ITS EFFECT ON SCREW COMPRESSOR PERFORMANCE

Whilst the dimensions of the radial and interlobe clearances are determined by the size and tolerances of the main compressor parts and by the positioning of these relative to the roller bearing clearances, the axial clearance is set during the machine assembly.

Due to its geometry, the pressurised air in a screw compressor produces an axial thrust making the rotors reduce the side clearance at the intake side and increase the clearance at the delivery side, where sealing is most critical. The manufacturers take this into account and offset their low pressure and high pressure axial clearances accordingly. These values may vary depending on size and manufacturer but can be considered in the 25µm to 50µm High Pressure discharge clearance, to 100µm to 150µm, low pressure suction clearance.

The side thrust is borne by thrust bearings, preventing the rotors from touching the surface of the end cover. Correct sealing is therefore achieved due to the quality and resistance of the thrust bearings.

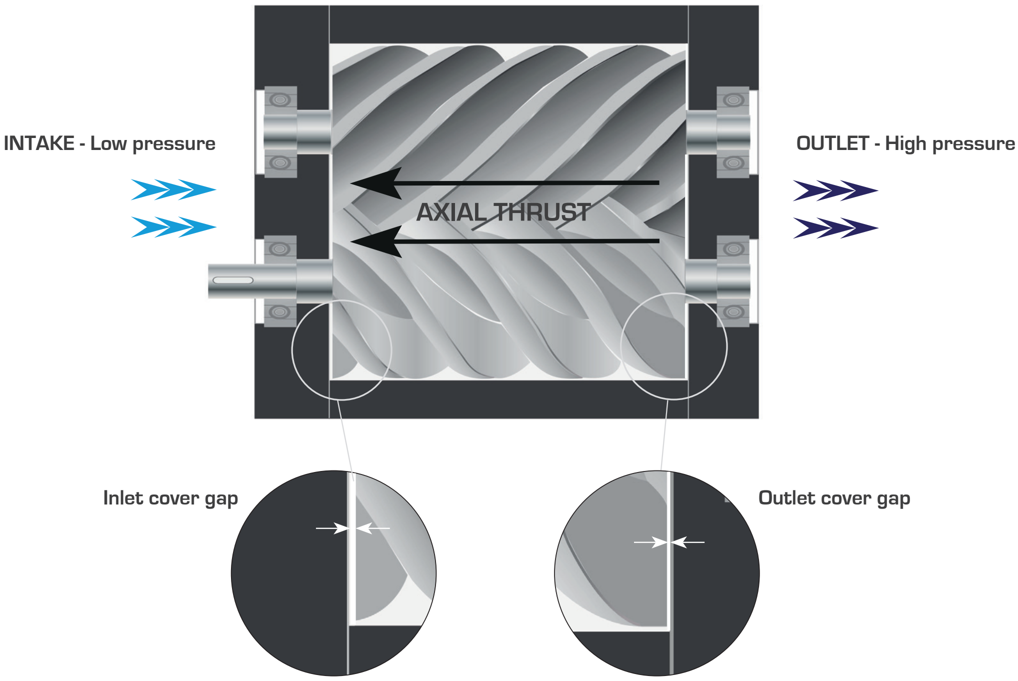

Fig. 6 – The pressurised air produces axial thrust, which reduces the clearance at the intake side and increases the clearance at the delivery side where sealing is most critical

In an oil-free screw compressor a fluctuation of 35µm on the discharge end clearance gave rise to a 22% increase in specific energy [5]. In the case of an oil-flooded screw compressor one expects the resulting effect on the specific energy to be less significant but it is nonetheless obvious there is a very strong correlation between these clearance values and the overall compressor performance.

SCREW COMPRESSOR AT ZERO HOUR

Off the manufacturing assembly line the Screw compressor interlobe and axial clearances will be precise and within the manufacturers guidelines to allow the specified compressor performance: this is clear from the fact that, at Zero Hours, there are a few major Screw compressor manufacturers whose CAGI-verified data is among the best on the market.

But what happens to these important clearances when the Screw Compressor starts running?

It is a well-known fact that both roller bearings and thrust bearings are subject to wear, and their wear rate is subject to both speed and load [9], and although manufacturers may decide to use different types and sizes of bearing, they will all advise a full air-end overhaul at a specific number of hours. This overhaul consists of substituting all major rolling and thrust elements, returning the compressor to a “safe” running condition, and avoiding any catastrophic air-end failure. Most manufacturers advise a major overhaul between 40 and 50 thousand hours of operation.

To put this bearing wear into context: in the axial direction it is acceptable to consider a 50μm wear on a thrust bearing as a point of no return, with catastrophic failure occurring at any point between 50μm and 200μm.
In the radial direction, bearing wear anywhere above the original manufacturers rotor clearance (12μm -25μm) will lead to catastrophic failure. In both of these cases it has been shown that the change in these clearances leads to very significant losses in performance, therefore is it essential to understand that Screw compressor performance cannot be considered constant in time.

Although, interestingly, it is hard to find academic literature that studies this phenomenon, the real-world scenarios in which energy audit companies measure flow and power drawn on old, pre-overhaul, screw compressors are many and well documented. In one case [10] out of 27 refrigeration screw compressors tested of varying ages up to 10 years old, the average performance degradation level measured was 30%, with the worst compressors performing at 55% degradation level.
Obviously one can only assume that these are extreme cases and bearing choices in today’s Screw compressors have improved drastically. **Nonetheless, it is impossible to claim that there is no performance degradation in a screw compressor due to the basic nature of bearings and of the engineering principles involved in Screw Compressors.**

ROTARY VANE COMPRESSOR PRINCIPLE

Also when considering the Rotary Vane compressors, the Zero Hour Specific Energy does not remain constant over time. Once again to explain this one has to examine the engineering principles.

The assembly consists of a single offset rotor rotating within a cylindrical stator. The compression element is sealed with two end covers that house two white metal bushings. The rotor has machined longitudinal slots, into which fit free sliding blades or vanes. The rotor is generally directly driven usually between 1000 and 1500 rpm (50Hz) causing the blades to make sealed contact with the stator wall thereby forming compression pockets. Air is drawn in, along the length of the stator at the point of greatest volume, becomes trapped in the pocket and the volume reduced (pressure increased) through one rotation. At the point of smallest volume air is discharged from the compression element (maximum pressure setting). Internally generated air pressure is used as the lubricant pump.

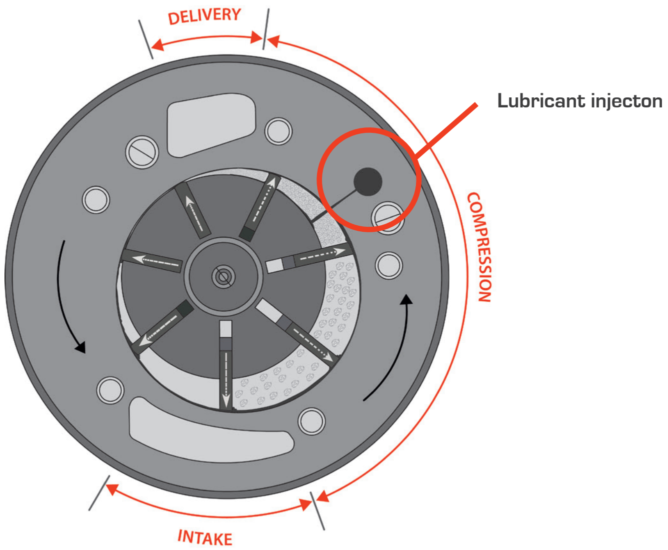

Fig. 7 – The lubricant injected into the stator lubricates the moving parts and absorbs the heat of compression. All operational clearances (ends of rotor and blade tips) are completely sealed with the lubricant preventing high to low pressure leakage

LEAKAGE PATHS AND CLEARANCES

In vane compressors the vanes are always in contact with the lubricant film on the internal surface of the stator. This keeps the two metal surfaces apart and seals between adjacent cells. A lubricant wedge exists at the leading edge of the sliding vane. The precisely machined vane tip radius, adhesion of lubricant to the sliding element and the supporting surface (the stator) increases the lubricant pressure and creates a hydrodynamic lubrication film between the two surfaces. The viscosity increase that occurs in the lubricant, when extremely high pressure is applied, allows the lubricant to avoid being squeezed out from in between the surfaces, maintaining a constant film in time. The lubricant also behaves as a perfect seal [Fig 8].

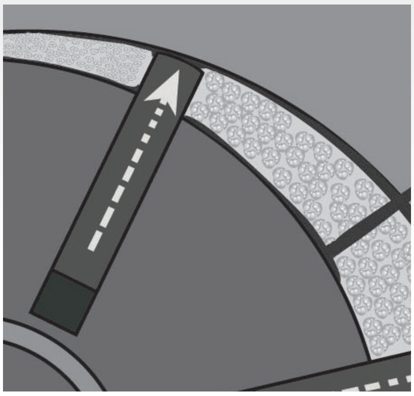

Fig. 8 – The vanes move freely in the rotor slots and always seal against the stator wall. Performance does not deteriorate even after many tens of thousands of operating hours

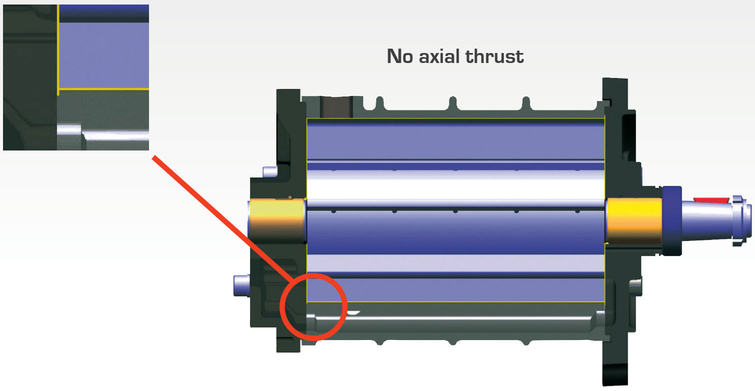

Fig. 9 – There is no axial thrust in a rotary vane compressor. The rotor is free to move axially and is kept equally spaced from the end covers by means of the lubricant, which is injected under pressure. The injected lubricant prevents the air from escaping along the side planes

Another potential leakage path is via the compressor end covers. The vane compressor has no axial thrust pushing the rotor against either end cover. It is, therefore, unnecessary to control its axial position by means of thrust bearings. The axial clearance is set during the machine assembly. As the rotor is free to move axially, it is kept equally spaced from both the end covers by means of the lubricant which is injected, under pressure, through dedicated injection ports in the end covers, thus preventing contact and providing efficient sealing [Fig 9].

As there are no wearing roller and thrust bearings inside a Rotary Vane compressor, the manufacturing set clearances are constant throughout the lifetime of the compressor. The benefit is two-fold, first there will never be any loss in volumetric efficiency over time, and second the compressor will never require an overhaul to substitute the worn bearings allowing Mattei to extend their compression unit (airend) warranty to 10 years with unlimited hours.

Having shown that volumetric efficiency does not change over time, why then is the aforementioned assumption incorrect also for Rotary Vane compressors?

BLADE POLISHING AND THE FIRST 1000 HOURS

From the moment you turn on a rotary vane compressor to about the 1000-hour mark, the blades undergo a polishing process on their sides. Although Mattei finishes the blade sides to a very precise tolerance, the first 1000 hours of running allow a complete and unique polishing between the slot and the blade sides. In tribological terms, the polishing removes the asperities on both contacting surfaces, and since these are made of complementary materials this initiates a microscopic material transfer that will last for the full lifetime of the compressor.

This is not to be confused with wear, in which one of the two rubbing surfaces loses material at a constant rate whilst the other is not affected, as this would result in catastrophic failure after very few running hours. **Instead, the special materials used in the manufacturing of the Mattei Rotary Vane compressor ensure that the original blades never need to be substituted and last well over the 10-year warranty mark.**

This polishing effect has a significant positive impact on the power lost to friction and, consequentially, on the power drawn from the compressor. This effect had been known to operators in the rotary vane industry for many years but it had not been independently and scientifically tested...until now!

In 2016, Mattei ran two parallel, long-term, tests. The first on a 50Hz Maxima 75 Xtreme model in the recently completed, state of the art R&D testing facility in Mattei HQ, and a second on a 60Hz Maxima 55 model at INTERTEK HQ in the USA. INTERTEK is the same institute that runs all the compressor verification tests for CAGI in the USA.

In both cases, compressor performance was taken at Zero Hours and data was then collected every 100hrs of operation. The results were outstanding to say the least. In both cases a significant and measurable decrease in power drawn at a constant Free Air Delivery was achieved, resulting in significant improvements in Specific Energy level.

Model	Frequency	Tester	Specific Energy Change
Maxima 75 Xtreme	50 Hz	Mattei R&D	-5%
Maxima 55	60 Hz	Intertek	-4%

Table 4 – Two separate long tests performed under controlled conditions, proving Specific Energy improvement over time for Mattei Rotary Vane Compressors

Therefore it is clear that, when considering a Rotary Vane Compressor, the CAGI verified Zero Hour Specific Energy is not to be mistaken for the Lifetime Specific Energy.

NEW LIFECYCLE COST ANALYSIS

With the data discussed above one can proceed to paint a more realistic scenario when simulating a compressor Life Cycle Cost.

Returning to the Example examined earlier, although the Vane Compressor and the Screw Compressor have identical CAGI Zero Hour Specific Energy, we can now proceed to include the performance deterioration for the screw and the performance improvement for the Vane in the LCC calculations.

It is important to underline that different Screw Compressor manufacturers choose to use a variety of different bearings in their machines. For this reason one cannot arbitrarily choose one deterioration rate for all Screw Compressors. **On the other hand it has been shown, through scientific research, that all it takes is bearing wear equivalent to one fourteenth of the width of a human hair (5μm) to lose nearly 2% in volumetric performance in a Screw Compressor.** Therefore we will run the New LCC calculations with -2%, -5% and -10% screw performance deterioration over the 5-year lifetime of the compressor (before the required overhaul) in an attempt to encompass most Screw Compressor manufacturers bearing choices.

Considering identical Capex and Maintenance costs, and selecting two 75kW compressors of identical CAGI data sheet Zero Hours Specific Energy, one could represent the specific energy over time in the following manner.

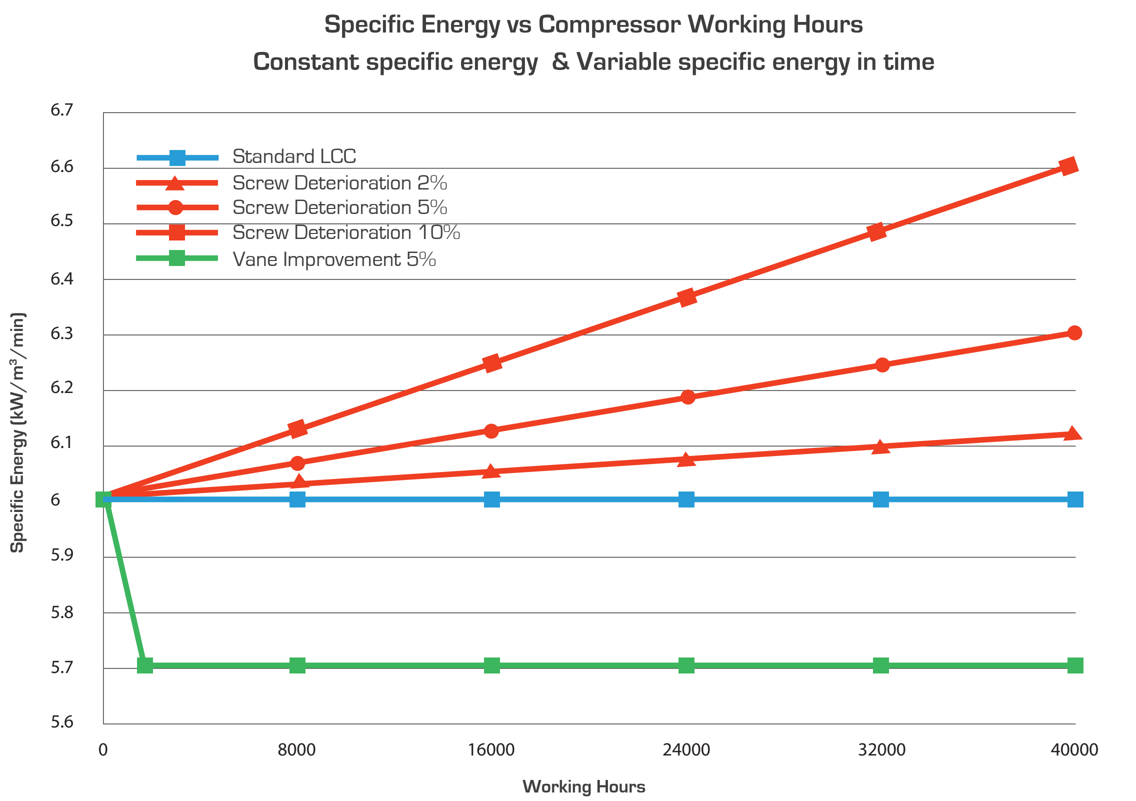

Fig. 10 – Changes in Specific energy over Compressor Working hours for various screw performance degradations over 5 years

Applying the New LCC considerations to the sample compressor installation from Table 1, the following results are obtained:

	Vane New LCC	Vane Standard LCC	Screw Standard LCC	Screw New LCC -2%	Screw New LCC -5%	Screw New LCC -10%
	Improvement +5% @1000 hrs	Constant Spec. En. in time	Constant Spec. En. in time	Deterioration -2% @4000 hrs	Deterioration -5% @40.000 hrs	Deterioration -10% @40.000 hrs
Capex	€ 50.000	€ 50.000	€ 50.000	€ 50.000	€ 50.000	€ 50.000
Maintenance @5yrs	€ 20.000	€ 20.000	€ 20.000	€ 20.000	€ 20.000	€ 20.000
Energy costs	€ 684.450	€ 720.000	€ 720.000	€ 727.258	€ 738.00	€ 756.000
Total LCC	€ 754.450	€ 790.000	€ 790.000	€ 797.258	€ 808.000	€ 826.000
Difference on Vane New LCC	€ -	€ 35.550	€ 35.550	€ 42.808	€ 53.550	€ 71.550

Table 5 – 5 year Life Cycle Cost comparison between the Standard calculations, with Constant Specific Energy and the New calculations with Variable specific Energy

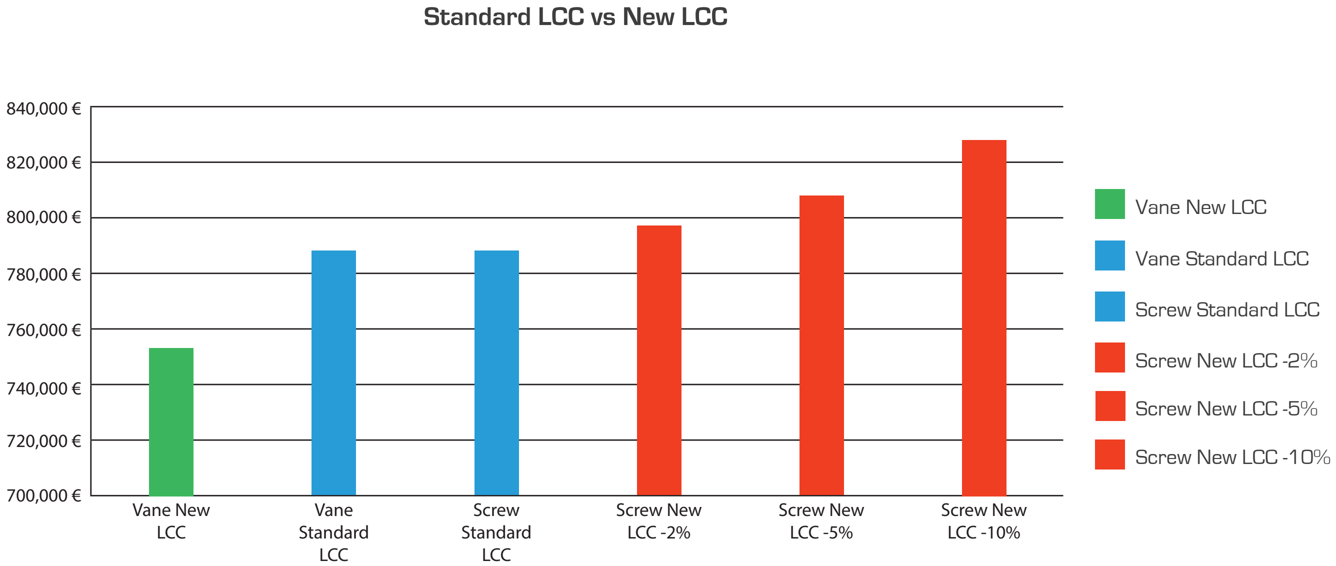

Fig. 11 – Graphical representation of data in Table 5

Extending the LCC calculation to a 10-year period, one has to consider the costs of the Screw Compressor complete overhaul to return the air end to its original condition and avoid bearing failure. In this example the industry standard cost of the original manufacturer overhaul of 50% of Capex is used. **For the Rotary Vane there is no overhaul cost over the lifetime of the compressor and this is evident by the 10 year air end warranty that Mattei applies to its compressors.** In this example the overhaul cost for the Vane compressor is equivalent to zero.

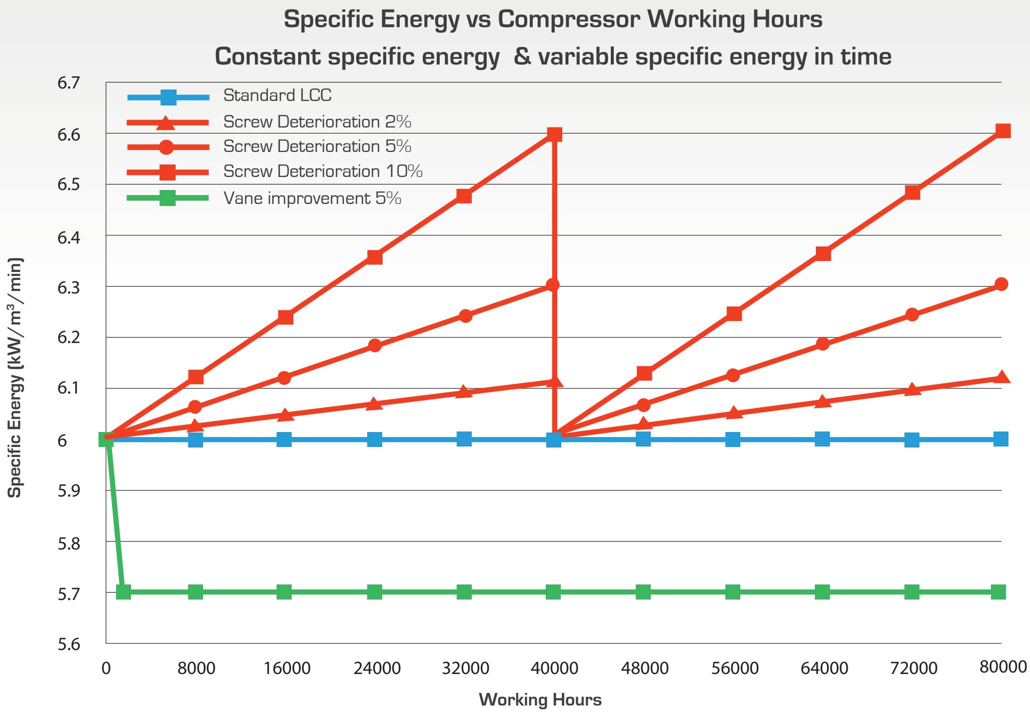

Fig. 12 – Changes in Specific energy over Compressor Working hours for various screw performance degradations over ten years with Compressor Overhaul at 40k hrs

	Vane New LCC	Vane Standard LCC	Screw Standard LCC	Screw New LCC -2%	Screw New LCC -5%	Screw New LCC -10%
	Improvement +5%@1000hrs	Constant Spec. En. in time	Constant Spec. En. in time	Deterioration -2%@40.000hrs	Deterioration -5%@40.000hrs	Deterioration -10%@40.000hrs
Capex	€ 50.000	€ 50.000	€ 50.000	€ 50.000	€ 50.000	€ 50.000
Maintenance @10yrs	€ 40.000	€ 40.000	€ 40.000	€ 40.000	€ 40.000	€ 40.000
40k hr Overhaul	€ -	€ -	€ 25.000	€ 25.000	€ 25.000	€ 25.000
Energy costs	€ 1.368.450	€ 1.440.000	€ 1.440.000	€ 1.454.516	€ 1.476.000	€ 1.512.000
Total LCC	€ 1.458.450	€ 1.530.000	€ 1.555.000	€ 1.569.516	€ 1.591.000	€ 1.627.000
Difference on vane new LCC	€ -	€ 71.550	€ 96.550	€ 111.066	€ 132.550	€ 168.550

Table 6 – 10 year Life Cycle Cost comparison between the Standard calculations, with Constant Specific Energy and the New calculations with Variable specific Energy.

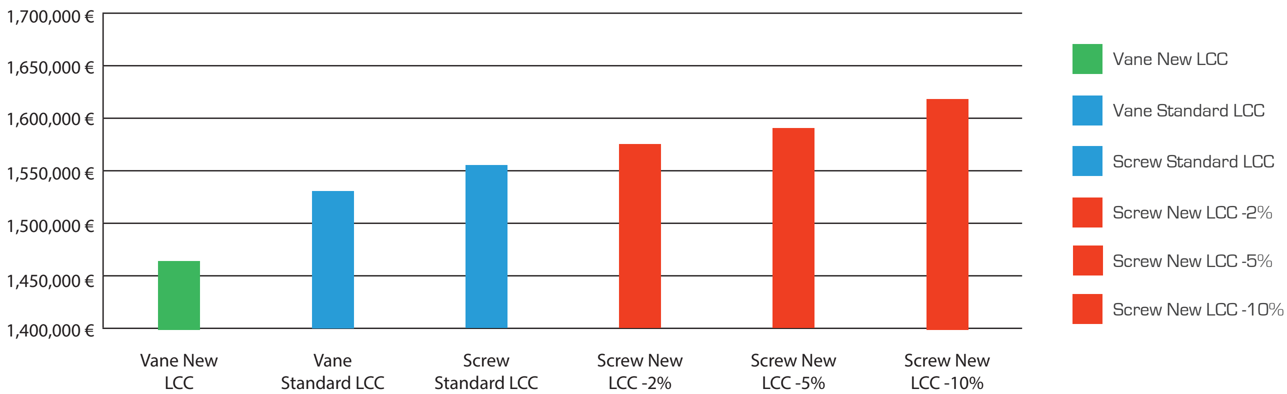

Fig. 13 – Graphical representation of data in Table 5

CONCLUSION

The New Life Cycle Costing method proposed takes into consideration both the improvement in the performance of the Rotary Vane compressor, and the loss of performance of the Screw Compressor over time. It is evident that there is an important difference in energy cost simulations when applying the New LCC method, as opposed to the industry recognised standard LCC method. **In the case of a 10% Screw performance degradation, over a ten-year period with overhaul, the client could spend €168,000 or 12% more (over 3 times the original Capex cost) by selecting a Screw over a Vane even though these have identical CAGI verified Zero Hours Compressor Performances.** This issue has to be addressed, especially in view of the fact that Zero Hour Compressor Performance data is now being used to draft new legislation to help curb the current global warming crisis. If such important information is not considered, the beneficial effect expected on lowering the energy consumed by Industrial Air Compression by the new legislation is at serious risk of not being reached.

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