



Final report

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Compressor Degradation Assessment and Wear Mitigation Strategy

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Executive Summary

Wear occurring in industrial screw compressor is a long-ignored subject which can potentially result in significant avoidable energy consumption and reduce refrigeration plant operating efficiency. This study illustrates the principles of compressor wear and its causes and lack of recoverability by compressor rebuild.

An on-site compressor performance test procedure has been developed and used to examine 54 compressors within 7 different sites to obtain an understanding of typical wear conditions of industrial compressors. The results have shown:

- Wear widely exists amongst aged industrial screw compressors, leading to some of them being degraded up to 55%. A 23% averaged degradation corresponding to a 22 years averaged compressor age has been observed.
- Most compressors started to wear during the initial 10-15 years although the level of degradation varied dramatically and could only be determined from on-site compressor performance tests.
- Mycom, Frick and Stal compressors appeared to show very similar characteristics in terms of wear. More data was required to investigate the development of wear for compressors from other manufacturers.
- Compressors on different stages of the refrigeration cycle showed no apparent difference on compressor wear.

The following suggestions have been proposed to achieve an accurate and cost-effective measurement to determine compressor degradation due to wear for future plants:

- Implement complete documentation of compressor commissioning, maintenance history, site incidents and run hours on a sufficiently regular basis.
- Install mass flowmeter on the common suction line of the refrigeration plant to allow a continuous monitoring of development of compressor wear.

Considering the actual state of typical refrigeration plants in the Red Meat Industry, replacement of worn compressors should be considered urgently, although the financial payback can only rely on indicative prediction of compressor degradation, which could potentially lead to contradictory conclusions.

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1. Overall project objectives

Screw compressors operating in the two refrigeration plants at JBS Dinmore are responsible for the bulk of the electric power consumption, yet many of the compressors are old and have been operating for significant total hours. Replacing rather than rebuilding such compressors may be a cost effective energy savings opportunity. The engineering consultant firm, Minus40, has developed a range of compressor test methodologies, generally requiring careful site specific planning, test preparation and after-hours execution, to conclusively quantify screw compressor wear. Compressor testing as part of routine service and maintenance program with a specific focus on efficiency and wear of equipment has not previously been carried out at a red meat abattoir.

The objective of the project is to apply these methodologies to the JBS Dinmore site to determine the level of compressor wear on each of the compressors, monitor plant operation to understand current plant operating profile, conduct energy modelling on compressor replacement scenarios, and consequently estimate the savings that would result from the replacement of these compressors. The outcome of this proposed work will assist JBS and the wider industry in quantifying the energy, carbon and cost savings and developing the required internal company capability to develop improved plant operating, maintenance and record-keeping programs to manage the effects of compressor wear.

The results of similar work conducted by Minus40 on other industrial refrigeration plants in Australia are included in the report.

2. Background

In the Red Meat Industry (RMI), compressors operating during refrigeration processes are responsible for the bulk of the site electricity power consumption. At nearly all the RMI refrigeration plants, either twin-screw compressors or reciprocating compressors have been installed. Generally, twin-screw compressors have replaced reciprocating compressors due to their lower maintenance costs and minimal requirement for regular refurbishment.

The screw compressor was first introduced in the 1970's and is now widely used for large industrial refrigeration applications and many different compressor suppliers, including Mycom, Howden, Frick and Stal etc. have brought their products to the Australian market over the years.

In current RMI plants, a large number of the early compressors are still in use and many of them are over 30 years old. Significant reduction in energy efficiency is expected on these aged compressors due to accumulated wear between the compressor rotors and bore, which could result in the refrigeration plant operating inefficiently. Replacing these with new compressors will generate significant energy savings, although corresponding capital costs should also be considered. Therefore, determining the level of compressor degradation within sufficient confidence becomes very crucial, this will allow an accurate estimate of the financial gain for a given compressor replacement project.

Unlike reciprocating compressors, for which it is possible to conduct a simple static compression test to determine the effectiveness of the compression process (Wilcox and Brun, 2009), there is no cost-effective strategy developed in the past that can be conducted to determine the condition of a screw compressor. This has added difficulties in determining the compressor efficiency reduction.

Apart from this, strategy to overcome existing compressor wear also needs to be stressed accordingly for efficient operation of industrial plants.

3. Screw compressor and its wear

To gain a better understanding of compressor wear, the principle of screw compressor operation is described in this section as well as various factors which potentially result in worn compressors.

3.1 Screw compressor principle

Rotary screw compressors are available in two basic designs: single rotor or twin rotors. Whilst some single rotor machines are in service in Australia, twin-screw compressors are by far the most common and hence this report will focus on twin-screw compressors.

The modern day twin-screw compressor technology was first patented by Alf Lysholm in the 1930's and commercialized by the Swedish company, SRM (Svenska Rotor Maskiner). In the time period from the first patent issued on screw compressor technologies through to today, screw compressors have undergone considerable advancement. Many of the advancements and success in commercial applications of technology were the result of progress in computer-controlled machining equipment. Screw compressors and applications of the technology continue to be fertile ground for the issuance of patents. In the period from 1976-2009, over 509 patents on screw compressors and associated applications were issued by the U.S. patent office (Reindl and Jekel, 2010).

Today, screw compressors are available in sizes ranging from 50-2,000 kW for application in refrigeration systems (commercial and industrial), gas compression and air compressors. Screw compressors are the fastest growing compression technology in the current industrial refrigeration marketplace.

Since twin-screw compressors first appeared on the market in the late 1970's, many manufacturers and models have come in and some cases gone. However, due to the very robust mechanical design of most industrial screw compressors, many if not most of the screw compressors ever sold in Australia are still in service today, in some cases over 30 years after installation. Table 2-1 provides an overview of common industrial twin-screw compressor manufacturers known in Australia:

Table 2-1 Industrial twin-screw compressor manufacturers in Australian market

Manufacturer	Country or Origin	Comment
Stal	Sweden	Early market entrant, ceased production in 1998, large number of old units still in operation, most 20-35 year old
Sabroe	Denmark	Available, but uncommon, a few very old units around
Frick	USA	Available, becomes very common since early 1990's
Dunham Bush	USA	Early market entrant, available in USA, not sold in Australia since 1990's, some old units still in operation
Hitachi	Japan	Available in Japan, not currently sold in Australia, most units are over 15 years old
Mycom	Japan	Available, very common, wide range of vintages in use
Grasso	Germany	Late market entrant, available, less common, most units are less than 15 years old
Howden	UK	Available, very common, wide range of vintages in use
Sullair	USA	No longer manufacture refrigeration compressors, some old units still in service

The current twin-screw compressor market is dominated by Mycom, with Grasso, Frick, Howden and Sabroe with lesser share.

Figure 1 shows a sectional view of a twin-screw compressor, which operates on the positive displacement principle with oil injection operating for a wide range of refrigerants. Two meshing helical screws, known as rotors, are used to compress the gas. The rotors are male or female and in most cases the male rotors are driven. The working cavity of the compressor is enclosed by the housing bores, housing end plates and the helical surfaces of the male and female rotors. Various bearings are installed on the rotors to support radial and angular load.

As the rotors rotate, the volume of the working cavity varies from zero to its maximum and from its maximum to zero periodically in a manner determined uniquely by the geometry of the compressor. As a consequence of this periodic variation, the compressor completes its suction, compression and discharge processes.

The compressor's capacity and internal volume ratio, V_i , can be set by two adjustable slides, i.e. the primary and secondary slides. Adjustment and setting are by means of pressurised oil which is supplied to the compressor by an oil pump.

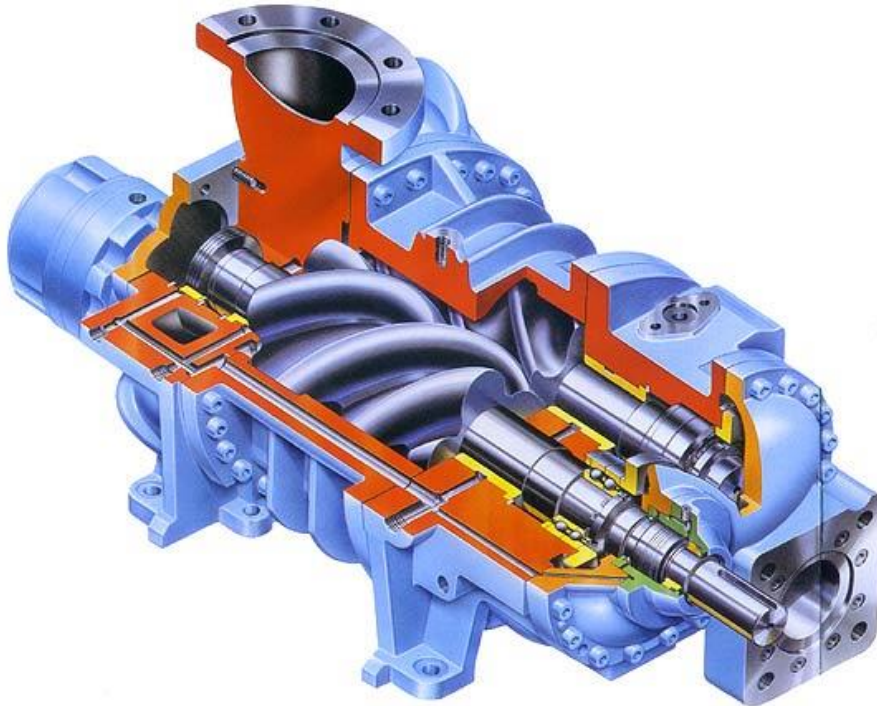


Figure 1 Sectional view of a twin-screw compressor

Compression

The compression process of screw compressors is a continuous sweeping motion, so there is very little pulsation or surging of flow, as occurs with piston compressors. While the rotors are turning, the meshing shifts from the suction side to the discharge side. A V-shaped space is formed between two male and female teeth in each case which increases to a maximum size (Figure 2(1-3)).

As the rotor continues to turn, the V-shaped space is closed by the new meshing on the suction side. This space gradually becomes smaller as the meshing continues (Figure 2(4-5)).

The compression process ends when the tooth space reaches the control edges which are incorporated in the casting and control slide. The control edge has an axial and a radial part.

The position of the control edges determines the size of the compressed volume and thus the level of compression. As the rotors continue to turn, the compressed volume is pushed out of the tooth space which diminishes to zero into the discharge area of the compressor. The process described above is repeated for every subsequent tooth space, thus achieving an almost continuous delivery (GEA, 2009).

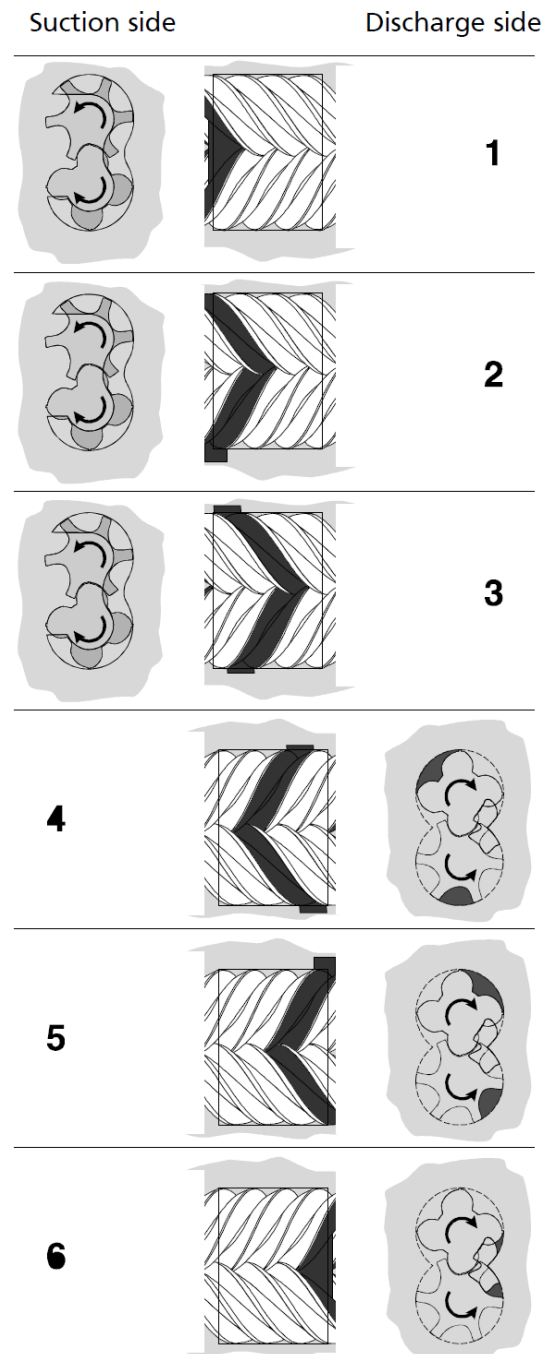


Figure 2 Compression process of a screw compressor (1-3: upper side view of the rotors; 4-6: lower side view of the rotors)

Oil system

The oil system is crucial for screw compressor operation, injecting lubricating oil into the compression chamber and bearings etc. The oil is then conveyed together with the compressed refrigerant from the compressor discharge into an oil separator where the oil is separated from the refrigerant.

Oil supplied to the compressor mainly serves the following purposes:

- Provide lubrication and sealing between the meshing positions between the rotors, as well as the surrounding housing. Without oil, the refrigerant gas trapped in the gulley will tend to leak backwards from a region of higher pressure to areas of lower pressure.
- Provide lubrication to bearings (journal and thrust bearings) in the compressor.
- Help keep the compressor clean by suspending and carrying particulates for removal by filters in the lubricant circulation system.
- Absorb and dissipate the heat generated during the compression process, resulting in discharge temperatures considerably lower than common for reciprocating compressors.
- Reduce noise by minimizing frictional effects.

3.2 Compressor wear

As described above, during normal operation the screw compressor relies on the sealing within the compression chambers. A brand new compressor is expected to provide good sealing, which will allow the compressor to operate at the compression efficiency specified by the manufacturer. However, after experiencing long service period, compressor wear will gradually accumulate and result in internal gas leakages and hence decreased compression efficiency.

The formation of compressor wear is due to many factors, such as poor maintenance, excessive bearing wear, capacity slide damage and erosion etc. Significant gas leakage will occur in the compressor chamber where wear develops to a certain level.

Fleming and Tang (1995) have indicated six leakage paths through which the working fluid leaks into, or out of, a working cavity. A worn compressor tends to show increased gas leakage through the following two paths:

- Contact lines between the male and female rotors. The refrigerant gas leaks to suction pressure from the considered enclosed cavity across this contact lines.
- Sealing lines between the rotor tips and the housing bores. The refrigerant gas leaks to the following cavity or suction pressure from the considered cavity across the following sealing lines.

In a screw compressor, both male and female rotors have small ridges on their tips, as shown in Figure 3. These ridges are designed in such a way to allow the friction between the housing and rotors to be reduced. There is no contact between these ridges and the housing or the opposite rotor, the clearance between them is very close and bridged by lubricating oil to minimise the gas leakage.

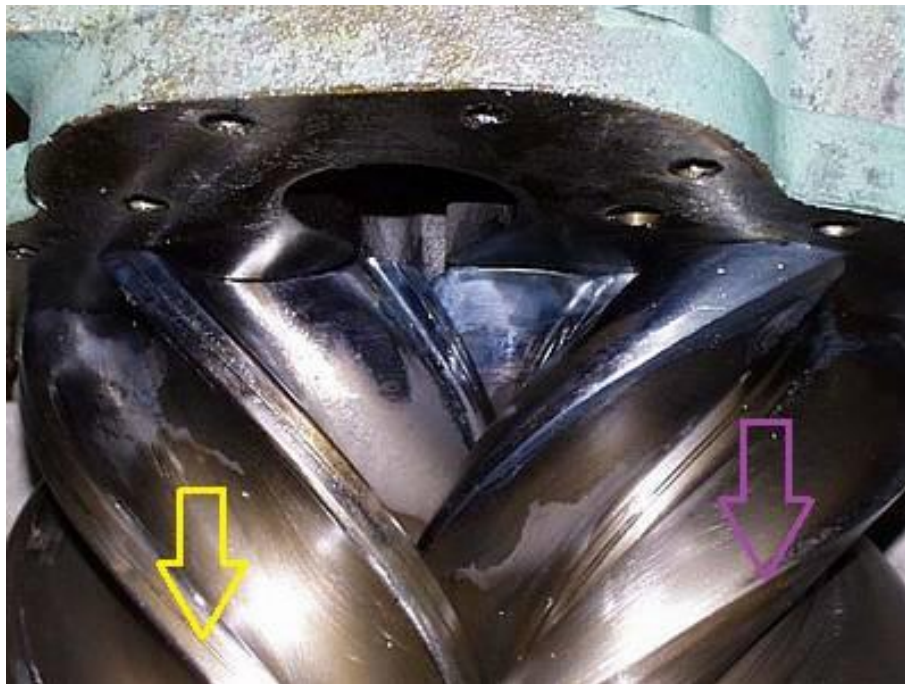


Figure 3 Screw compressor male and female rotors with small ridges (indicated by purple arrow for the male rotor and yellow arrow for the female rotor)

Due to this fine structure of the ridges, they are the components tending to get worn over time the easiest. Wear is formed via:

- Molecular erosion after long-term operation

- Excessive bearing wear causing rotors to move laterally and touch the housing (a higher discharge pressure may tend to accelerate wear due to a greater force imposed onto the bearings)
- Impurities in oil or ingress of foreign particles into the system, which is usually caused by poor maintenance
- Contact between rotor tips and capacity control slides which are located underneath the rotors.

With wear accumulated over time, the compressor efficiency will be reduced significantly: the absorbed power turning the rotors remains the same but the compressor capacity drops.

3.3 Challenges

Once the extent of compressor degradation resulting from wear is established, the unnecessary energy cost can be estimated and the plant operator/owner will be able to decide whether to carry out a compressor replacement or modify the compressor staging sequence to gain financial benefit reduced running of worn compressors.

However, there are apparent challenges to estimate the amount of gas leaking, as well as quantify in terms of performance loss (Hanlon, 2001) and this complicates the estimation of unnecessary energy cost. A well-arranged on-site compressor test might be able to achieve relatively accurate measurements although the cost for this becomes a drawback. Additionally, to conduct these tests, a period without production is required, which will disturb normal operation for certain plants.

In the Red Meat Industry, the screw compressor wear problem so far has been ignored, partly due to the difficulties in identifying the compressor degradation and partly due to a long-standing fallacy that after a rebuild, screw compressors will achieve an efficiency which is the same as a new unit. For a reciprocating compressor, a full efficiency recovery can be gained (Bloch and Hoefner, 1996). However, during the rebuild of a screw compressor, the compressor bore, rotors and slide mechanisms are not replaced or repaired, so the existing wear remains.

3.4 Project objectives

Due to the above gaps identified in the Red Meat Industry, investigations should be carried out to acquire more understanding of screw compressor wear and develop feasible and cost-efficient solutions to minimise the effect of this during plant operation. This project therefore aims to achieve the following objectives:

- Develop and conduct a screw compressor test for RMI to determine typical levels of compressor degradation.
- Estimate energy and cost savings that would result from the replacement of degraded screw compressors.
- Provide a comprehensive assessment for the RMI that will enable design and implementation of plants to be able to conveniently overcome the wear problem.

4. Methodology

To obtain an understanding of the level of degradation due to compressor wear, on-site compressor performance test procedures have been developed and described below.

4.1 Compressor performance test

Conducting on-site compressor performance test is a challenging task which requires test operators having thorough understanding of the refrigeration system of the tested plants. The exact testing procedure will vary from plant to plant due to their different system arrangements and operating conditions. However, tests can be generally classified into three groups:

- Type I: Plant operates at steady base load conditions, with measurable or calculable loads (e.g. fan power, heat transmission etc.). This can be regarded as a reference load. Apparent compressor load, which is the calculated theoretical load from monitored operating parameters (suction and discharge gas pressures, compressor speed and compressor load position) based on manufacturer's performance data, is then compared to reference load. A higher apparent load is an indicator of wear.
- Type II: Plant operates against at least one compressor which is known to be in good condition (no degradation). This compressor will be selected as a reference compressor and reference load can be calculated based on its operating parameters. For any tested compressor, if its apparent load is higher than reference load, this is a wear indicator.
- Type III: Plant operates with an accurately measured additional heat load generated by electrical heaters placed within refrigeration spaces. This will be considered as a reference load increase, which will then compared to apparent load increase and a higher apparent load increase is an indicator of wear.

For the three above types, a stable base load condition is required, which usually occurs at night or during the weekend when there is no production activity. During the test, compressors are sequenced to allow only one single compressor to operate

each time, per compression stage. Compressor operating parameters are monitored for a sufficient period to allow all compressor parameters to be stabilized, which usually takes around 30 mins. Then the compressor is switched off and the next one is started and the same procedure repeated.

If the base load is generated by fan motors or electric heaters, it becomes type I method. The base load should be large enough to allow the compressor to operate at a relatively high part load level. Within the compressor capacity, a higher base load will result in more accurate test results.

For certain plants, there are always loads within the refrigeration spaces to maintain the plants operating normally even without production, e.g. storage of carcass. These parts of loads are mostly unable to be quantified accurately. Therefore, extra accurately measured heat load needs to be added to provide reference load increase and this becomes type III method.

The most convenient method is type II, which does not require heaters etc. to provide external heat load, so is expected to be at lower cost. However, it is only applicable to those plants which at least have one compressor in known good condition (= new).

The selection of test types will depend on characteristics of the plants. For the purpose of enhancing reliability, it is favourable to select multiple methods to achieve triangulation.

The tested compressor degradation can be expressed as:

$$Degradation = \left(1 - \frac{Reference\ load\ (increase)}{Apparent\ load\ (increase)}\right) \times 100\% \quad (3.1)$$

where the reference load (increase) is the external heat load placed in the refrigeration space for types I and III, and can be calculated based on reference compressor operating parameters for type II; the apparent load (increase) can be calculated based on the tested compressor operating parameters.

It should be noted that the accuracy of the test highly relies on the accuracy of the pressure sensors and compressor load state on-site. In most industrial refrigeration plants, these are calibrated on a regular basis, although large errors are still

commonly experienced. Therefore, it is important to perform calibration before the tests.

5. Results and discussions

5.1 Compressor test results

Applying the above methodology, 54 screw compressors in total from 7 different industrial plants have been tested and the details are listed in Table 4-1.

Table 4-1 Compressor performance test results

No.	Make	Model	Age	Conditions	Capacity (kW)	Degradation (%)	Comments
Site A							
1	Sabroe	VMY436B	30	-45 SST, -11 SCT	345	45%	LS
2	Mycom	320SULX	22	-45 SST, -20 SCT	486	15%	LS
3	Mycom	250LGLX	22	-45 SST, -11 SCT	360	30%	LS
4	Mycom	250LGLX	22	-45 SST, -11 SCT	360	30%	LS
5	Mycom	250SG-MX	22	-11 SST, 25 SCT	1066	20%	HS
6	Mycom	250SG-MX	22	-20SST, 25 SCT	728	35%	HS
7	Mycom	250 VLD	12	-11 SST, 25 SCT	1550	0%	HS
8	Frick	TDSL283L	22	-45 SST, -11 SCT	380	40%	LS
9	Frick	TDSL283XL	22	-45 SST, -11 SCT	380	40%	LS
10	Frick	TDSH233L	2	-11 SST, 25 SCT	1250	0%	HS
Site B							
11	Stal	S57	31	-42 SST, -12 SCT	215	20%	LS
12	Stal	S57	31	-42 SST, -12 SCT	215	35%	LS
13	Stal	S73	31	-42 SST, -12 SCT	425	30%	LS
14	Stal	S57	31	-36 SST, 25 SCT	275	35%	LS/HS/SS
15	Stal	S71	31	-12 SST, 25 SCT	1000	25%	HS
16	Mycom	2520SSC	29	-40 SST, +30 SCT	320	35%	SS
17	Mycom	2520LSC	29	-40 SST, +30 SCT	465	40%	SS
18	Mycom	2520SSC	29	-40 SST, +30 SCT	320	10%	SS
Site C							
19	Frick	RWB399	20	-40 SST, -12 SCT	678	15%	LS
20	Frick	RWB399	20	-40 SST, -12 SCT	678	10%	LS

				SCT			
21	Grasso	XDR-30B	10	-40 SST, -12 SCT	1209	45%	LS
22	Stal	S75	20	-40 SST, -12 SCT	807	20%	LS
23	Frick	RWB316	20	-12 SST, 25 SCT	1695	40%	HS
24	Frick	RWB177	20	-12 SST, 25 SCT	913	45%	HS
25	Stal	S71	20	-12 SST, 25 SCT	1048	55%	HS
26	Grasso	XAR26S	10	-12 SST, 25 SCT	2118	0%	HS
Site D							
27	Mycom	FM160L-M	16	-35 SST, 35 SCT	152	30%	SS
28	Mycom	FM160L-M	14	-35 SST, 35 SCT	152	40%	SS
29	Mycom	FM160M-M	13	-35 SST, 35 SCT	126	20%	SS
Site E							
30	Mycom	2520 LSC- MBM	25	-42 SST, 27 SCT	382	10%	Compound
31	Mycom	250SG-MX	23	-42 SST, -9 SCT	198	35%	LS
32	Mycom	250LG-LX	32	-42SST, -9 SCT	424	25%	LS
33	Stal	S75EB-26A	32	-42SST, -9 SCT	593	20%	LS
34	Mycom	250SG-MX	32	-42SST, -9 SCT	265	25%	LS
35	Mycom	320LUD-LBX	4	-42SST, -9 SCT	750	15%	LS
36	Stal	S71E-34A	32	-9 SST, 27 SCT	1165	25%	HS
37	Stal	S71E-34A	32	-9 SST, 27 SCT	1165	5%	HS
38	Stal	S57E-26A	36	-9 SST, 27 SCT	905	30%	HS
39	Stal	S73E-34A	32	-9 SST, 27 SCT	1782	5%	HS
Site F							
40	Mycom	200lg-mx	19	-11 SST, 26 SCT	795	55%	HS
41	Stal	S65-2D	28	-39 SST, -11 SCT	318	45%	HS
42	Stal	S65-2D	27	-39 SST, -11 SCT	318	40%	LS
Site G							
43	Mycom	250LG-MX	18	-11 SST, 27 SCT	1567	15%	HS
44	Frick	TDSH283S	3	-11 SST, 27 SCT	1689	10%	HS
45	Frick	TDSH283S	15	-11 SST, 27 SCT	1689	5%	HS
46	Frick	TDSH233XL	15	-11 SST, 27 SCT	1453	0%	HS
47	Mycom	320LU-LBX	18	-43 SST, -11	800	15%	LS

				SCT			
48	Howden	WRV255/165 36	20	-43 SST, -11 SCT	408	15%	LS
49	Howden	WRV255/165 36	20	-43 SST, -11 SCT	408	10%	LS
50	Frick	SGC2821	6	-43 SST, -11 SCT	673	0%	LS
51	Dunham -bush	2516	20	-11 SST, 27 SCT	1586	5%	SS
52	Frick	TDSH283S	16	-11 SST, 27 SCT	1698	15%	SS
53	Frick	TDSH283S	16	-11 SST, 27 SCT	1698	0%	SS
54	Frick	TDSH283S	14	-11 SST, 27 SCT	1698	5%	SS

* SST: saturated suction temperature, SCT: saturated condensing temperature, SS: single stage compressor, HS: high stage compressor, LS: low stage compressor.

The above data indicate that compressor wear is a common issue for aged industrial screw compressors and some of them have been degraded up to 55% due to this. An averaged 23% degradation corresponding to an averaged compressor age of 22 years has been observed.

5.2 Effect of compressor age

To study the effect of compressor age on level of degradation, data in Table 4-1 have been summarized and shown in Figure 4. It indicates that compressor wear may appear at almost any stage, even within 5 years for some compressors, which is possibly due to inappropriate installation and commissioning etc. However, a majority of them start to show degradation during the initial 10-15 years (100,000 hours). After 15 years, nearly all screw compressors show more or less degradation, although there is no apparent correlation between compressor age and the level of degradation. For compressor over 15 years old, their degradation levels vary widely, ranging from 0 up to 55% with an averaged degradation of 26%.

It is speculated that the compressor run hours may have a more direct relation with its degradation. However, only in very few test cases have compressor run hours data been well documented to the extent that they are insufficient to conduct analysis.



Figure 4 Compressor degradation as a function of compressor age

5.3 Effect of manufacturer

Figure 5 shows the degradations for Mycom, Frick and Stal compressors. It indicates that compressors from different manufacturers all experience the wear problem and it is very hard to identify any dependence of compressor manufacturers to degradation.

However, based on currently available data, it appears that Mycom compressors tend to wear around 3-5 years earlier than Frick although after 20 years of operating, their behaviours become very similar.

Almost all Stal compressors in service are around 30 years old and their degradations are mostly within the range of 20-40%.

Data for other manufacturers are limited and unable to comment at this stage.

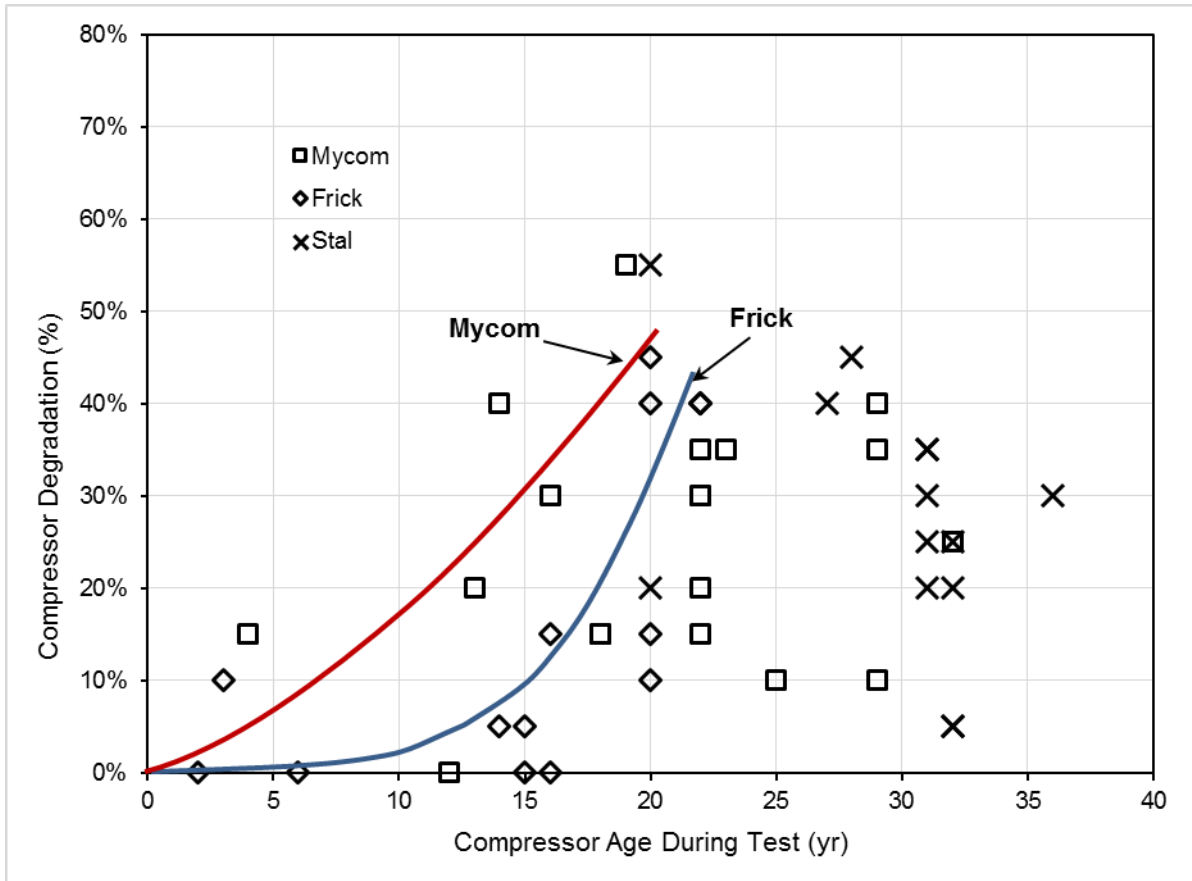


Figure 5 Compressor degradation versus compressor age for Mycom, Frick and Stal compressors

5.4 Effect of site conditions

Figure 6 compares degradation data at different sites, indicating that compressors on site G generally show low degradations, with the maximum value up to 15%. For site F, compressor wear is generally more severe with the degradation ranging from 40 to 55%.

However, a large number of sites give diversified degradations, e.g. site C with degradation ranging from 10 to 55%.

It is suspected that the maintenance strategy for different sites may play a role on the rate of compressor degradation, however, due to the absence of well-documented maintenance records for most of the sites, further discussion becomes impractical.

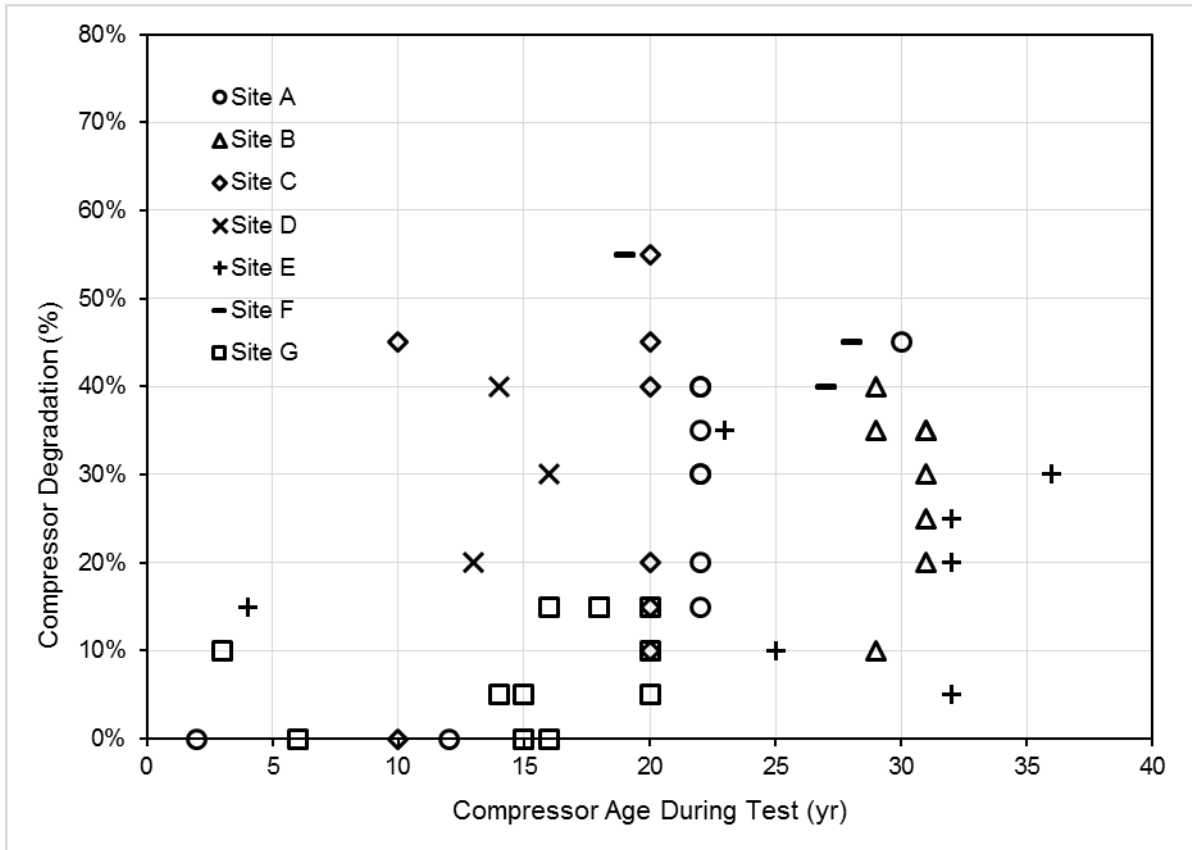


Figure 6 Compressor degradation versus compressor age for Site A-G

5.5 Effect of working pressures

Figure 7 compares the degradation of high stage and low stage compressors, the former operating at a much higher pressure boundary and expected that their bearings, rotors and housing will need to handle greater force than low stage counterparts during operation. However, this effect is not obvious, according to current data: there are two high stage compressors showing 55% degradation although the rests are comparable to low stage cases.



Figure 7 Compressor degradation versus compressor age for high stage and low stage units.

5.6 Recommendations

As shown above, current data appear to be highly random: except for those within 10 years old, the degradation of a certain compressor can range from 0 up to 55%. The degradation does not show dependence on any of the factors discussed above, indicating that there might be other factors which affect the development of compressor wear, however, due to insufficient site monitoring devices and limited site maintenance history documentation, accurate prediction of degradation resulted from compressor wear becomes impossible.

To allow compressor degradation to be quantified conveniently without conducting costly testing procedures in future, the following suggestions are proposed:

- **Documentation:** Compressor commissioning, maintenance history, site incidents and run hours to be well documented, which should include inspected and replaced parts for each time of the maintenance with run hours to be updated on a regular basis.

- Suction Gas Flowmeter: A mass flowmeter should be installed on the common suction line of the refrigeration plant and connected with site PLC/SCADA system, this will allow the mass flow, which can be easily converted to plant actual load, to be monitored constantly. A long period of monitoring will build a trend of degradation development, which will offer great help on further understanding of the compressor wear problem.

Currently, several energy management information systems are available on the market, such as SENSEI®, which is a powerful platform for collecting and analysing various plant data, plant actions documentation and energy saving management. These tools can provide comprehensive support to implement the above suggestions.

5.7 Compressor replacement strategy

Once compressor degradation is quantified, it will be possible to evaluate the effect on the plant energy consumption and corresponding cost. A replacement of the degraded compressor should be conducted if a required payback, e.g. within two years, can be achieved. In some cases it is possible to retain the compressor unit such that only compressor blocks are replaced, e.g. Mycom, which reduces the overall capital cost. For other manufacturers, eg Stal, their products are no longer supported, so the whole compressor unit has to be replaced, at greater cost.

For cases that replacement will result in long payback, a feasible alternative/interim solution is to place the degraded compressors to the back of the operating sequence so their running hours can be reduced.

Specifically, the following procedure is suggested:

- 1) Estimate unnecessary annual energy consumption due to compressor wear based on compressor degradation and annual power consumption data
- 2) Estimate unnecessary compressor operating cost based on an assumed electricity price
- 3) Obtain quotes from refrigeration contractors about compressor replacement projects
 - a. For Mycom, Howden, Grasso, Frick and Sabroe compressors, only block replacement is suggested

- b. For Stal, Hitachi, Dunham Bush and Sullair compressors, complete unit replacement including oil system is required.
- 4) Within required payback, projects will be processed, otherwise, compressors move to the back of the sequence as temporary solutions.

It should be noted that in addition to the fact that a compressor replacement does improve compressor performance, it also increases system reliability and reduces the risk of compressor bearing failures.

6. Business cases and case studies

This section provides two examples to demonstrate the compressor replacement strategy.

6.1 Business case – Site G

A large meat processing site has a two stage plant which consists of eleven industrial ammonia screw compressors (two Howden, two Mycom and seven Frick compressors) with four operating on high stage and others on low stage.

It is identified that one Mycom compressor operating on high stage shows signs of wear. Relevant information has been listed in Table 5-1.

Table 5-1 Operating details of Mycom compressor

Manufacturer	Mycom
Compressor age (yrs)	18
Current annual electricity consumption (kWh)	2,900,000
Electricity price (\$/kWh)	0.15
Compressor block replacement cost (\$)	\$156,000
Compressor unit replacement cost (\$)	\$270,000

Degradation tests on this compressor indicated a 15% degradation. Therefore, the excessive electricity consumption due to compressor wear is:

$$\text{Excessive electricity consumption} = 2,900,000 \times 20\% = 435,000 \text{ kWh}$$

The savings due to compressor replacement is

$$\text{Compressor replacement savings} = 435,000 \times 0.15 = \$65,250$$

Mycom compressors allow only the block to be replaced, so the payback is

$$\text{Payback} = \frac{156,000}{65,250} = 2.4 \text{ years}$$

Based on the above simple calculation, it is concluded that a compressor block replacement will result in a payback within 2.5 years, so it is financially feasible to implement.

6.2 Case study – Site C

A food process facility has four low stage compressors, including two Fricks, a Grasso and a Stal, and four high stage compressors, including two Fricks, a Grasso and a Stal. One of the Frick compressors on high stage and a Grasso low stage booster are found to be heavily degraded, as per information listed in Table 5-2.

Manufacturer	Grasso	Frick
Compressor age (yrs)	10	20
Current annual electricity consumption (kWh)	1,888,000	733,000
Electricity price (\$/kWh)	0.12	0.12
Compressor block replacement cost (\$)	\$148,160	\$92,960
Compressor unit replacement cost (\$)	\$352,930	\$356,075

The degradation of the Grasso and Frick compressors were determined at 45%, so that the payback can be calculated following a similar method as for Example 1. In case of the Grasso::

$$\text{Payback} = \frac{148,160}{1,888,000 \times 45\% \times 0.12} = 1.45 \text{ years}$$

where a 45% degradation and a compressor block replacement cost were used. For complete unit replacement the payback is calculated as:

$$\text{Payback} = \frac{352,930}{1,888,000 \times 45\% \times 0.12} = 3.45 \text{ years}$$

Which exceeds the acceptable payback. In this case the site chose to replace the compressor block only.

In case of the Frick:

$$\text{Payback} = \frac{92,960}{733,000 \times 45\% \times 0.12} = 2.34 \text{ years}$$

where a 45% degradation and a compressor block replacement cost were used. For complete unit replacement the payback is calculated as:

$$\text{Payback} = \frac{356,960}{1,888,000 \times 45\% \times 0.12} = 3.5 \text{ years}$$

Which exceeds the acceptable payback. In this case the site also chose to replace the compressor block only.

The total expected annual savings from the replacement of these two compressors blocks was anticipated to be $(1,888,000+733,000) \times 45\% = 1,179$ GWh/annum.

After compressor block replacement, an energy savings verification process was conducted in formal compliance with IPMVP (International Performance Measurement and Verification Protocol) processes subsequently demonstrated the actual savings at 953 GW, or 80% of the expected savings.

The effective verified project for the two compressor replacements was determined as:

$$\text{Payback} = \frac{241,125}{953,000 \times 0.12} = 2.1 \text{ years}$$

Considering the accuracy range of the site degradation tests, this is a fair result and demonstrates the capability of the degradation tests to identify effective energy savings projects.

7. Conclusions and Recommendations

This study highlights the long ignored screw compressor wear problem in industrial refrigeration plants, which result in unnecessary energy consumption and reduced plant operating efficiency. An on-site compressor performance test procedure has been proposed and applied to 7 industrial sites with 54 screw compressors being tested in total.

The compressor test results showed that wear may occur at any stage, however, most of the compressors appear to be worn during the initial 10-15 years. After around 15 years, compressor wear shows no obvious dependence on vintage. Mycom and Frick compressors generally show very similar behaviours due to wear although Mycom tend to experience developed wear 3-5 years earlier than Frick; degradation for Stal compressor in service is generally ranging from 20 to 40%. Different site condition may result in less or more serious degradation, which is possibly due to their various maintenance strategies, however, this cannot be confirmed with limited compressor maintenance history records.

To achieve accurate and convenient measurement of compressor degradation, the following suggestions have been proposed:

- 1) Document compressor commissioning, maintenance history and run hours data on a sufficiently regular basis;
- 2) Install mass flowmeters on common suction line of the refrigeration plant to achieve continuously monitoring of wear development. Various energy management information systems can provide comprehensive support to implement the above suggestions.

With given compressor degradation, the following is proposed:

- A) Implement compressor block or unit replacement if the calculated payback is within the required period, otherwise,
- B) Place the compressor to the back of the control sequence to minimize run hours and hence reduce energy wastage, as a temporary solution.

Bibliography

Bloch, H. P, Hoefner, J. J 1996, *Reciprocating Compressor Operation and Maintenance*, Gulf Publishing Company, Houston, pp 227-302.

Fleming J. S, Tang Y 1995, The analysis of leakage in a twin screw compressor and its application to performance improvement, *Proceedings of the Institution of Mechanical Engineers, Part E: Journal of Process Mechanical Engineering*, Vol. 209, pp. 125-136.

GEA, 2009, *Screw Compressor Grasso SC LT Series Installation Instruction*.

Hanlon, P 2001, *Compressor Handbook*, McGraw-Hill Inc., New York, pp 2.26.
http://www.apvgn.pt/documentacao/compressor_handbook_hanlon.pdf

Reindl, D. T, Jekel, T. B 2010, *Industrial Refrigeration Energy Efficiency Guidebook*, Industrial Refrigeration Consortium, pp 2-13.

Wilcox, M and Brun, K 2009, *Guideline for Field Testing of Reciprocating Compressor Performance*, Gas Machinery Research Council & Southwest Research Institute.