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Large screw compressors in the refrigeration industry

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1. SYNOPSIS/ABSTRACT

The author has spent some 40 years in the business of designing, specifying and operating refrigeration systems for large food factories throughout the world. He has seen the industry develop to the point where screw compressors now dominate the industry. This paper deals with refrigeration systems – not the detailed design of the compressors themselves – and how the compressors chosen can influence the efficiency of such systems. The author is not a compressor designer and has selected over the years machinery on the basis of equipment supplier data and his own operational experience. The paper consequently deals with the choice of units now available, the significance of large industrial refrigeration screw compressors and their advantages and limitations over other machinery and in particular how the end user views their application.

2. HISTORICAL DEVELOPMENT

Originally, some 100 years ago, large refrigeration systems were serviced by steam driven horizontal large bore single cylinder reciprocating compressors. In modern terms, the cost of such plant was enormous and occupied a large building area. By the 1930s, the compressor manufacturers were offering multi cylinder vertical reciprocating units with either direct or belt electrical drives. The reduction in cost and space requirements was dramatic. The 1950s and 60s saw further economising on capital cost. Single cylinder booster compressors were replaced with rotary centrifugal compressors which could deal with large gas volumes over a low pressure ratio while the second stage machinery was now serviced by V block reciprocating compressors. In the 1960s the twin screw compressor arrived, providing high capacity with reduced size and cost together with an option to operate with high compression ratios allowing single stage systems for low temperature refrigeration requirements. Good volumetric efficiency and long periods between major overhauls were claimed. Large industrial plants are now mainly serviced by the screw compressor for both high and low stage systems.

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3. COMPRESSOR CHARACTERISTICS

Comparisons between the presently available range of compressors will now be addressed.

3.1 Capacity

Figure (1) shows an approximate range of volumetric capacities for compressors presently supplied by the major manufacturers to the refrigeration industry. The modern screw units can deal with very large capacities per compressor compared with the available rotary booster or reciprocator range – this provides a most significant reduction in capital cost.

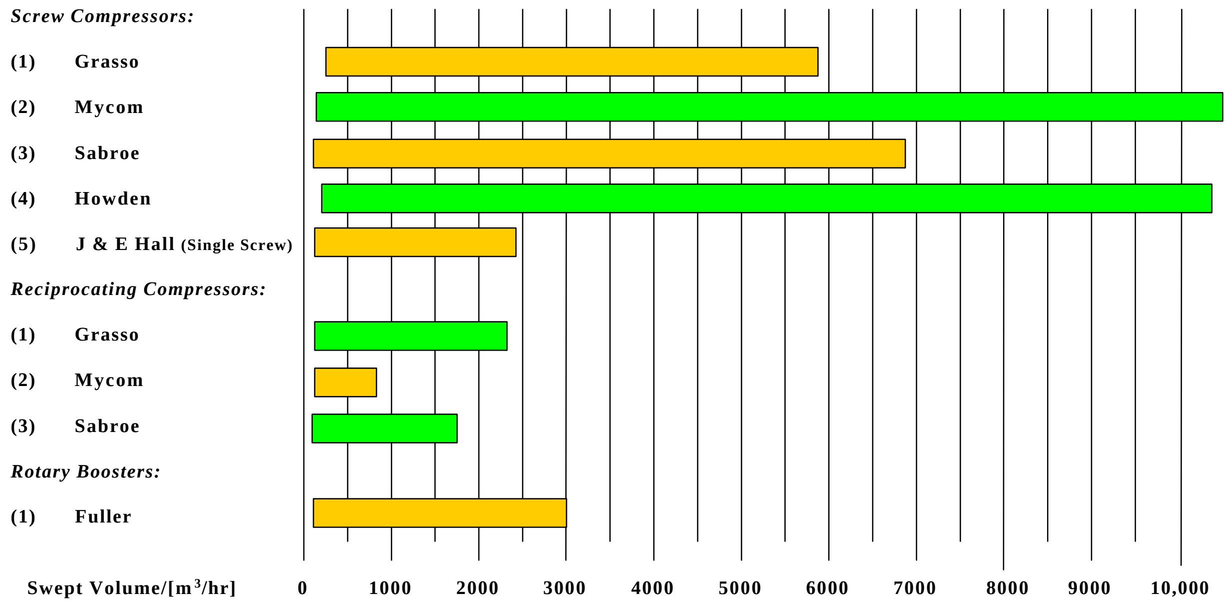

Figure 1

3.2 Compression Ratios and Power Consumption

The higher the compression ratio of a system, the higher the discharge temperature of the gases leaving the compressor become. Discharge temperature limitations on reciprocating machines provides a particular advantage for the screw by eliminating two stage systems through oil injection.

Figure (2) shows electrical consumptions per kW of refrigeration for the various different refrigerants now available for different refrigeration requirements.

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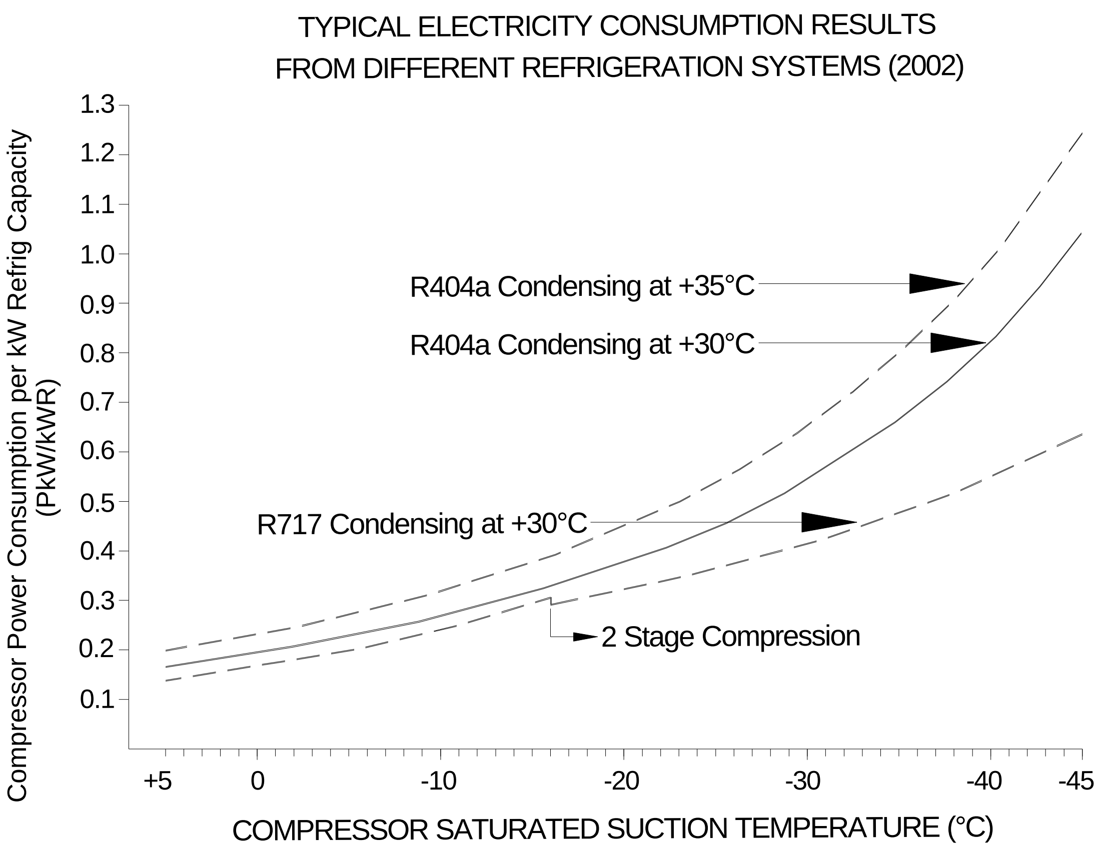

Figure 2 - Electricity Consumptions

The curves generally represent both reciprocating or screw compressors as presented by manufacturers' data. Evaporating conditions under approximately –25^{0}C for R717 (Ammonia) require a two stage system to ensure economical running conditions. Furthermore, the pressure ratio becomes too high for reciprocating machines under –25^{0}C evaporation due to high discharge temperatures. For instance at –40^{0}C evaporation such a system will save approximately 50% of electricity consumption compared with a single stage plant where reciprocating compressors could be used due to the lower gas discharge temperatures with R404a. For a large industrial plant consuming much electricity, R717 is the refrigerant that is most economical to use. For small installations with low overall electricity usage, the lower capital cost single staging with R404a is often chosen. The screw compressor, because of its high pressure ratio capabilities due to oil injection, can significantly cut down the capital cost of the large ammonia refrigeration plant.

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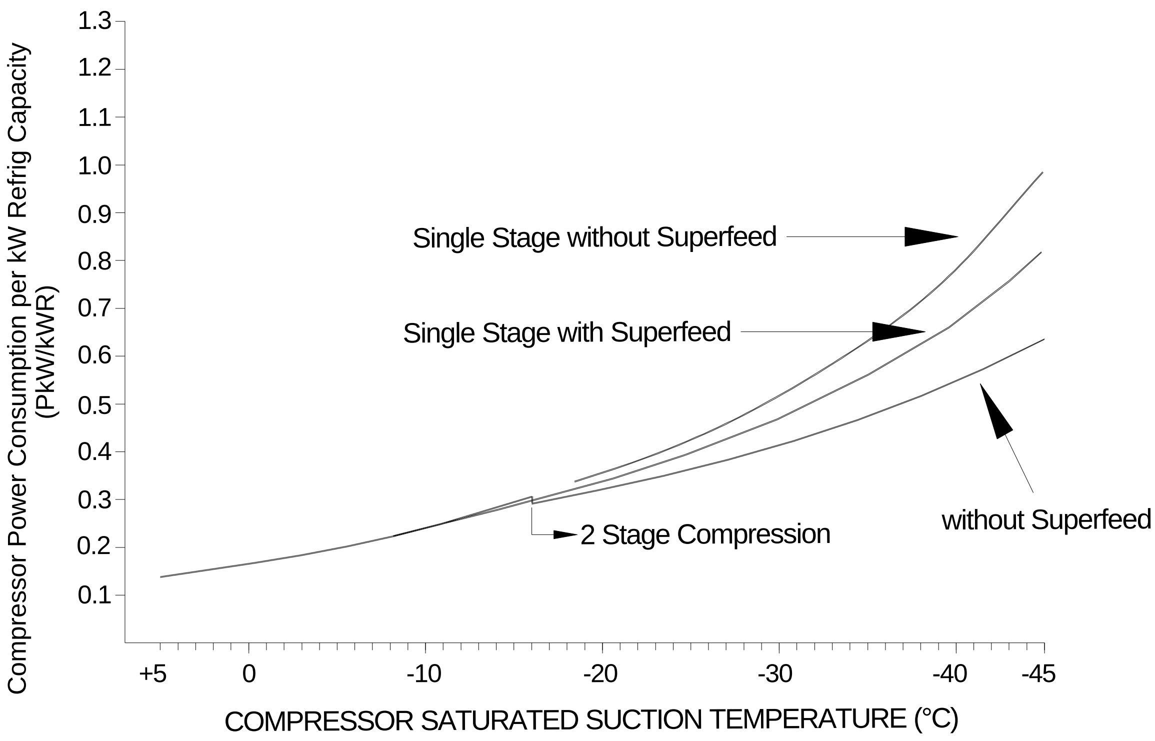

Figure 3 - Screw Compressors with Superfeed

Figure (3) again shows a typical power consumption graph for a two stage R717 system, as also a screw compressor working single stage down to -40^0C evaporation. Rather than consuming twice the electrical power as with R404a, the increase in power consumption is now only some 50%. By supplying the compressor with an additional suction port further along the screw, this allows gases at a higher suction pressure to be drawn into the compressor. Some manufacturers have called this additional feature a “superfeed port”, and this arrangement allows a two stage expansion of the refrigerant to the -40^0C condition. Figure (4) diagrammatically represents the pressure enthalpy diagram for R717 under these conditions and it can be seen that an approximate 20% refrigeration duty improvement can be provided without any substantial additional electrical input by expanding sub cooled refrigerant so created by the superfeed port.

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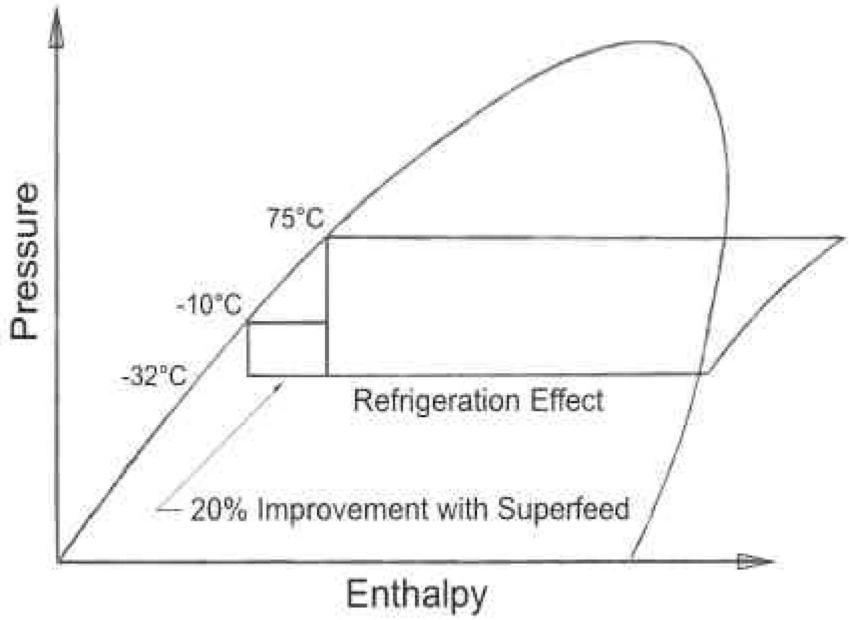

Figure 4 - Screw Compressor Superfeed Economy

There can therefore be a commercial case for using a screw compressor installation single stage evaporating at -40^{0}C because the complication and cost of a two stage plant is eliminated against a 20% increase in electricity usage.

-40^{0}C suction conditions are usually required for quick freezing of food products. However if a cold store has no quick freezing requirements, this allows the refrigeration plant to be built for an evaporation condition of -32^{0}C. Furthermore, if a two day freezing cycle is acceptable, cartons of at least 150 mm thickness can be frozen from a -32^{0}C system which is often a cold storage required service. Under these conditions (-32^{0}C suction), the superfeed screw compressor will save the installation of a reciprocating compressor high cost two stage installation, while theoretically sacrificing only 10% electricity consumption.

3.3 Maintenance and Running Costs
The screw compressor manufacturers claim significantly reduced maintenance costs. Their reasoning is twofold. Firstly, an installation could be serviced by fewer compressors which means less overhauls. Major overhauls are recommended between 35-50,000 running hours, and the only significant components needing replacement are oil seals, the rotor and thrust bearings and possibly the capacity control slide mechanism. This compares with overhauls on reciprocating compressors for valves at 10,000 hours, piston rings and bottom end bearings at 20,000 hours and main bearings and liner replacements at 40,000 hours on compressors of 8 to 16 cylinders each.

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3.3 Unloading Characteristics

Another claimed benefit of the screw compressor is its variable output. The reciprocating compressor usually unloads in banks of 2 or 3 cylinders progressively, which results in uneven capacity regulation. The screw compressor contains a slide valve which alters the position of the suction port in relation to capacity required from some 15% to 100% output.

4. OPERATIONAL EXPERIENCE

The above describes the major advantages claimed for the screw compressor. After some 30 years of being involved with operational plants with a variety of machinery, I would challenge various of these claims. The following considerations need reassessing:-

4.1 Capacity And Power Claims

The output and electrical consumption figures for the screw compressor as shown in the graphs of figures (2) and (3) come directly from the manufacturers' graphs and tables. Our experience has shown however that although output and efficiency obviously declines as wear takes place in any compressor, the actual output of the screw compressor can in some instances drop by some 30 – 45% of the manufacturers' figures.

This output reduction first became apparent some years ago to our organisation in an old meatworks plant room installation which was modernised on the second stage system with two screw compressors. One 1930s three cylinder reciprocating compressor was retained as a standby unit. After the screw units had reached some 30,000 hours of operation, it was noticed that when the old reciprocating unit was occasionally run, the system refrigeration back pressure would drop significantly, meaning that this old worn-out reciprocating compressor, which had an almost equivalent capacity, was pumping more gas than the new and latest screw compressor. The rotor sleeve bearings had worn down to such a degree that the tip seals on the rotors had worn away on the body of the machine causing the discharge gases to bypass back through the rotors. Although no exact performance deterioration figures were recorded (as it was impossible to know the efficiency of the old reciprocator), there was every indication that the machine performance had fallen by some 20 – 30%.

Around the same time period, tests were also being carried out in our New Zealand operations on refrigeration system efficiencies. It was during these tests that it became apparent that the screw compressor installations were not achieving expected outputs. In several installations the conclusion was that some screw compressors had dropped in efficiency by some 35% and average coefficient of performance ratings by around 40%. These tests never fully concluded why these drops had occurred.

A more recent experience in 2001 in the UK indicated that things had not changed. In 1996 two large screw compressors had been installed for blast freezing product at –40^{0}C. The makers recommended overhauls to take place at 40,000 hours. It was agreed that new units would replace the operating screws on an exchange basis. It had been noted for some 2 years that with freezing at full capacity, both screws continually operated together at between 90 and 100% output. After replacement, it was only necessary to run one unit at 100% to meet the same freezing loads. During the 40,000 hour running period, the unit output of these two compressors had fallen to approximately 55% of the originally manufacturer's stated output. The drop in performance could only be attributed to worn rotor tip seals from bearing wear and/or tip seal wear due to wear from the slide valves. I understand that small screw compressors are usually designed with roller (anti-fraction) bearings. I am not a specialist in bearing applications or characteristics, but no such bearings are offered by the large screw manufacturers. If such bearings were able to be used, this major problem of tip seal wear from bearing wear-down should be overcome.

What all this means is that the power curves for the screw compressors in figures (2) and (3) should in practice be downgraded by this loss in efficiency. Furthermore, the effective capacities of the units shown in figure (1) should also be similarly reassessed.

4.2 Maintenance

The screw compressor manufacturers usually recommend a full overhaul between 35 and 40,000 hours of operation. However, it is clear that if the machines run this period of time they could become exceptionally inefficient and in some instances may not be of sufficient capacity to deal with the original refrigeration loads of the factory. I have found that the manufacturers are reluctant to provide initial data with respect to tip seal clearances. The end user could find out this information by measuring the clearances on a totally new machine, but this is invariably impracticable as the machine arrives new, ready for installation and disassembly costs money and the down time is unacceptable. Specifying a precise tip seal clearance at the time of manufacture is critical to performance and each manufacture must obviously define what is required to its machine shop. For an end user to obtain this simple information is difficult as no such information is provided in sales literature or workshop manuals.

I believe maintenance should take place at approximately 10,000 hour intervals. To maintain tip seals and performance, the four rotor bearings should be replaced at 10,000 hour running periods or at least be measured for wear down at these inspections.

My experience on reciprocating compressors is that if the wearing parts are automatically exchanged at each overhaul, most can extend valve overhauls to 25,000 hours and piston overhauls to 50,000 hours. Thus in effect the reciprocating machine may in certain circumstances be less expensive to maintain contrary to the screw compressor manufacturers' claims.

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4.3 Capacity Control

With a screw compressor capacity control is variable from some 15% to 100% progressively. However, extensive inefficiencies can result by running the unit at lower outputs than 100%.

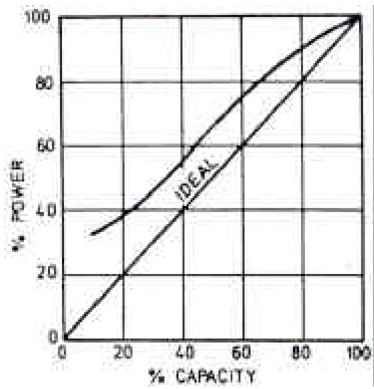

Figure 5 - Manufacturer's Screw Compressor Capacity Control Data

Figure (5) gives a typical power consumption versus refrigeration output curve for a screw compressor. As the slide valve reduces capacity, the reduced power consumption is not proportional to the new output of the machine. Thus this manufacturer's machine operating at 50% refrigeration output would consume some 65% of the electrical power of 100% capacity, not 50% as would be expected. As the slide valve reduces capacity further, the power consumption per kW of refrigeration gets progressively worse. This graph does demonstrate the importance of 100% capacity operation but it does not provide the whole truth. Figure (6) provides a more accurate representation of the true picture.

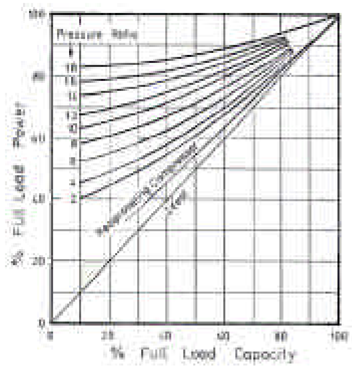

Figure 6 - Unloading Inefficiencies Against Compression Ratios

The graph in figure (5) would be fair if the system compression ratio was some 2.5 as it may be for a first stage booster operation. However, for a high stage system with a typical pressure ratio of 4, the power consumption would be not 65% but 70% as shown on the graph. If on the other hand a low temperature cold store installation was installed with the superfeed port, this would represent a pressure ratio of around 12 and the graph figure then rises to 82%.

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Figure (6) also shows the equivalent reduction in efficiency in unloading reciprocating compressors, and the electrical saving by using a reciprocating compressor at part load becomes evident. The straight line on the graph interconnects the various unloading points as stepless capacity control is obviously not possible on a reciprocating compressor.

Power consumption for a screw compressor at outputs below 50% become excessive particularly with an installation with only one machine where loads cannot be shared. Even on 50% capacities, the inefficiency of the screw compressor, particularly on high pressure ratios, is in my opinion unacceptable and the machines should, whenever possible, be used as base load units with only reciprocators providing the capacity control function. Consideration should be given to fitting variable speed controls on screw compressor motors. This would allow the machines to run without slide valves and at acceptable efficiency. Although, tip seal speed is also critical to efficiency if the motor was designed to run the compressor from say 4500 rpm down to 2500 rpm, we feel sure that efficiency drops as shown on figure (6) would be significantly improved. I understand that the optimum tip seal speed is in the order of 35 metres/second and so the most suitable motor speed range will obviously vary with the particular screw rotor diameter. However, to convince non technical end users to pay extra for these variable speed motor controls is difficult. A large screw compressor motor of some 600 kW or higher with an inverter drive would raise the capital cost of each compressor by some £10,000.

4.2 Other Considerations

A most important consideration with a refrigeration screw compressor installation concerns lubricating oils.

The oil injected screw compressor used in the refrigeration industry relies upon large quantities of lubricating oil being injected into the rotors. This significantly reduces gas discharge temperatures and allows high pressure ratio operation. This means that a large amount of lubricating oil is continually heated to the gas discharge temperature. The oil will however pass into the refrigeration system which will produce poor heat transfer. This results in higher electrical power consumption and less refrigeration available. A record of oil consumption must be maintained on a continuous basis for each compressor together with that recovered from the system. Oil separation systems have improved over the years but probably in an attempt to keep first cost to a minimum, oil passing into the refrigeration system remains considerable.

Many of the original installations had inadequate coolers resulting in oil delivery temperatures of over 60^0C. With conventional mineral oils this can result in a chemical breakdown of the oil resulting in the development of black sludgy deposits which penetrated the whole refrigeration system. In my experience, oil coolers should be sized to deliver oil to the screw bearings and injection points at temperatures not exceeding 50^0C. This usually results in having to fit coolers of double the surface area originally designed by the manufacturers. Synthetic oils have been introduced which allow high temperatures without chemical breakdown. However, the oil costs are virtually tripled for the working life of the plant.

High oil temperatures can also influence the compressor performance. As the temperatures rise, the sealing effect the oil had on the rotors becomes increasingly ineffective resulting in less volumetric efficiency and higher electrical consumptions in excess of the makers' performance data.

5. OVERALL ECONOMICS

In order to emphasise the dilemma that an operator has with respect to making a decision on what type of machinery he should install and how tempting a low capital cost with screw compressors can be, I will take a typical medium sized New Zealand freezer works processing beef and sheep and compare the options available. For such an operation, a low stage refrigeration load (evaporation at -32^0C) of approximately 3000 kW for cold storage and freezing requirements and a high stage load of 900 kW (-10^0C evaporation) for beef chilling would be normal. This high stage load rises to 4500 kW with intercooling for a two stage system.

Assuming tip seal performance of the screw compressor is maintained by regular replacement of the main bearings, then the screw installation would be most economically served by two machines running from -32^0C to condensing (pressure ratio 12:1) of approximately 6000 m^3/hr each and the second stage load of 900 KW refrigeration being serviced by two small reciprocating machines of 700 m^2/h each.

To provide a compressor combination of reciprocating machines would need six of the largest units of some 2000 m^3/h each for the booster load and four units of the same capacity for second staging (4500 kW). The reciprocating compressors would be unable to single stage through a pressure ratio of 12:1. Assuming the rest of the installation is common to both schemes, then the difference in first cost between the two systems would be the price of the compressor packages and extra vessels and piping for the two stage scheme.

We estimate the installed cost of each of the two screw (6000 m^3/hr) units would be approximately £60,000 each or £120,000. The two small reciprocators for the beef chilling would add a further £30,000, making a total installation cost of £150,000. The estimated cost of the ten reciprocating packages would be in the order of £35,000 each with extra pipework and vessels adding another £50,000, making a total of £400,000.

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The electrical cost at 4 pence per kWh for running the two screw compressors, assuming an average –32^{0}C refrigeration load throughout the year of 1500 kW, as meat works can be very seasonable, would theoretically be approximately 750 kW electricity or approximately 8.5 million kWh per year, or almost £180,000 per annum. However, with the two screw compressors for the one stage dealing with variable loading throughout the year, we would have to assume an average slide valve condition of 50%. Viewing figure (6) and a compression ratio of 12:1, the inefficiencies would rise on the graph figures from 50% to 80% or an increase of 60%. Thus the electricity consumption on these two screw compressors only would rise to approximately £290,000 per annum.

The two stage operation would be approximately 10% better on electricity consumption (see figure 3) or theoretically the cost would come to 90% of £180,000 or £160,000. Thus there would be a yearly energy saving of around £140,000 with reciprocating compressors.

It would be reasonable to assume an average yearly cost to maintain the two screw compressors of £20,000. On the basis of £7,500 per year per reciprocating machine, then yearly compressor maintenance would be £75,000.

Item	**Screw Installation**	**Reciprocating Compressors**	**Cost Difference**
Energy	£290,000	£160,000	£140,000
Maintenance	£ 20,000	£ 75,000	£ 55,000
	Reciprocator yearly savings		£ 85,000
Capital Cost	£150,000	£400,000	£250,000

Table I – Capital & Operating Cost Summary

From the summary of costs in table I, the reciprocating compressor choice can produce substantial yearly savings. However, the large capital cost savings ensures most large installations end up with screw compressors as most end users have little idea as to the screw compressor inefficiencies and the compressor manufacturers certainly make no effort to enlighten them. Variable speed motors fitted to the screw compressors could greatly improve the above power inefficiencies. On the above example the increased electricity costs should be recoverable within about three years by using reciprocating compressors. If tip seal deterioration is also taken into account, it could be significantly faster.

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6. SUMMARY/CONCLUSIONS

The industrial refrigeration industry over the years has continuously been serviced by compressor design changes. This has resulted in ever decreasing real capital cost rather than any great improvement in efficiency of the refrigeration systems. Volumetric efficiency and coefficients of performance have continually decreased from the slow speed 1930 compressors through to the faster speed V block compressors with a further dramatic reduction in efficiency with the screw compressor.

The screw compressor has very significantly contributed to capital cost reduction. Instead of needing five or six reciprocating compressors or more to meet a requirement, 2 screw compressors should easily meet the loads due to their very high outputs. Capital costs can be further reduced by the screw being able to deal with high compression ratios with oil cooling reducing high gas discharge temperatures. Furthermore with the introduction of the superfeed port, the screw can theoretically be economical to run single stage down to –32^{0}C evaporation which usually will service all cold storage requirements and most freezing arrangements. Single staging again reduces the number of compressors required by eliminating the second stage machinery and capital costs accordingly. Claims were also originally made concerning the reduction of maintenance costs. Overhauls were originally required at 50,000 intervals rather than a succession of maintenance requirements at shorter periods for the reciprocating compressor. The screw compressor also encompasses a slide valve system which allows infinitely variable capacity control compared with staged control on reciprocating units and virtually no control on centrifugal booster compressors.

Thus the screw compressor has undoubtedly been a significant asset to the refrigeration industry particularly with respect to lowering capital costs. However, my many years experience in system design and in the operation of refrigeration plants has identified a series of major drawbacks in using screws. The major problem centres around the drop in performance of the screws due to bearing wear down and rotor tip seal wear. In one instance a drop in output approaching 55% of original maker's claims was discovered. I suggest that this deficiency can be mainly overcome by rotor bearing replacement periods of 10,000 hours rather than the 50,000 hours originally recommended or, alternatively, the use of roller bearings. This drop in output proportionally increases electrical consumption per unit of refrigeration produced, and this also means that the original plant installed may not be able to provide the original service needed. Further problems are related to oil cooling and oil deterioration. These can be mainly solved by using synthetic oils but at an additional cost. Finally, capacity control, although variable, cannot economically be applied below 50% output as lower capacities are highly expensive in electricity consumption, particularly on high pressure ratios. Even at 75% loadings, the coefficient of performance of the plant can be reduced by some 20%.

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Our conclusions therefore are that the screw compressor is essential to reduce capital costs for large installations. But in order to ensure the plant operates with reasonable efficiency, the most important consideration concerns constant monitoring of rotor bearing wear which affects rotor tip seals and thus economical operation and also consideration should be given to fitting variable speed drives to reduce inefficient part load operation.

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ACKNOWLEDGEMENTS

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