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A Balanced View of Reciprocating and Screw Compressor Efficiencies

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A BALANCED VIEW OF RECIPROCATING AND SCREW COMPRESSOR EFFICIENCIES

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ABSTRACT

The company has been engaged in reciprocating compressor design since the turn of the century and in double rotor screw compressor design since 1969.
During the last five years a thorough test programme has been carried out in co-operation with Norges Tekniske Høgskole, University of Trondheim, on our reciprocating compressor design to determine the exact influence of the various design parameters on compressor efficiency.
In particular the optimal design of the ring plate type suction and discharge valves has been studied.
The paper will report on the variations in volumetric - and isentropic efficiencies both for R22 and R717, in relation to speed, length of stroke, valve spring design, compression ratio, etc.
The results obtained with the reciprocating compressor are compared with those for the double rotor screw compressor, based on experience from more than 1000 units supplied, and on various results obtained on the company's behalf by the licenser's laboratory.
The differences are explained, and the conclusion is substantiated that both types of positive displacement compressors have their own merits, and that they complement each other to the extent that they may often be combined in one plant to obtain the most energy efficient installation under variable operating conditions.

INTRODUCTION

The company was founded in 1897, and designed and manufactured its first reciprocating compressor for refrigeration that same year. Since 1969, besides reciprocating compressors, we have also been engaged in design of SRM-type screw compressors and screw packages.
In 1954 we introduced the first models of the high speed SMC compressor programme, comprising three series of compressors with 65, 100, and 180 mm bore, respectively, and with from 4 to 16 cylinders arranged in V, W and 2 V designs.
In connection with a planned redesign in 1975, of the SMC 100 type, with 100 mm bore, we established a co-operation with Norges Tekniske Høgskole, University of Trondheim, comprising an extensive research programme to optimize the suction and discharge valves of our reciprocating compressors.
Examination and testing of compressor valves, and more specifically valves of the single ring plate type, is a field in which NTH has specialized since many years.
The research programme has a practical part including a great number of capacity tests with different refrigerants, and varied operating conditions, and a theoretical part which employs a computer simulation technique to optimize the suction and discharge valve designs.

DESCRIPTION OF BASICS

The valves in a reciprocating compressor control the gas flow to and from the cylinders, and the valve behaviour has a decisive influence on the compressor efficiency. More specifically the suction valve dynamics is of prime importance for the actual efficiency obtainable, and it is therefore necessary to be able to optimize the valve design if higher efficiencies are to be reached.
To optimize a valve design requires a thorough study of the effects obtained by changing all the parameters influencing the valve behaviour, and such a study can only be carried out, within a reasonable time span, by means of a computer simulation programme.
Fig. 1 shows an ideal compression cycle, and this is what we would like to achieve in the actual compressor.

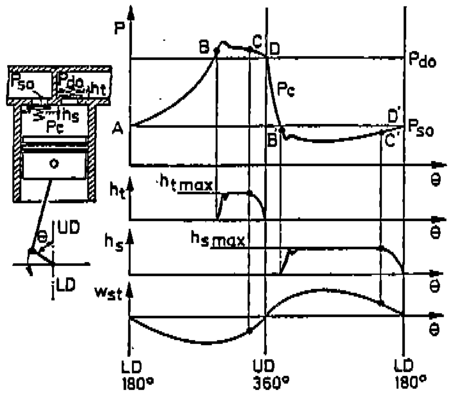

Fig. 1 - Ideal compression cycle

P_c = cylinder pressure

h_s = lifting height of suction valve

h_t = lifting height of discharge valve

W_{st} = piston speed

In order to follow the pressure variations in the cylinder, and the movements of the ring plate valves, a number of electronic pick-ups have been mounted in a test compressor as shown in Fig. 2.

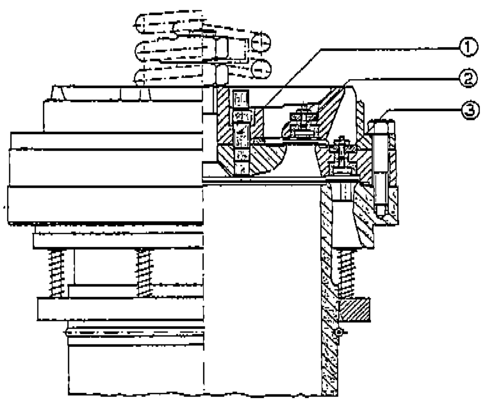

Fig. 2 - Cylinder liner with electronic pick-ups

1 - piezo-electric pressure pick-up
2 - capacitive electrode measuring discharge valve movement (total of 3)
3 - capacitive electrode measuring suction valve movement (total of 3)

Actual tests are used to correct the constants in the simulation programme, and to check the results from the programme.
Fig. 3 shows an example of the excellent fit obtained by the simulated programme describing the movement of the valve when subjected to different spring forces.

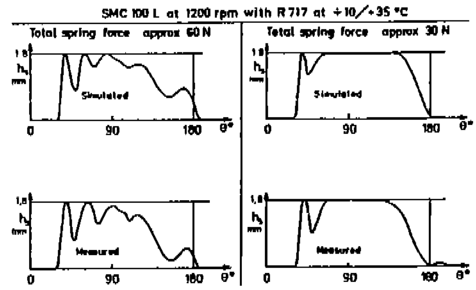

Fig. 3 - Suction valve movement with different spring forces

The lowest spring force results in the most stable valve behaviour at these operating conditions, the variation in volumetric and isentropic efficiencies, however, is not significant. At low evaporating temperature, the spring force would show a considerable influence on compressor efficiency, with the weak spring being the better choice.
Once the simulation programme has been perfected, as demonstrated above, the selection of valve springs can be based on theoretical investigations of valve behaviour over the whole range.

COMPRESSOR EFFICIENCIES

The efficiencies referred to in the following are:

Volumetric efficiency

$$\eta_{vol} = \frac{V_{suct}}{V_{theo}} = \frac{G_{cirk} \times v_{suct}}{V_{theo}}$$

Isentropic efficiency

$$\eta_{is} = \frac{N_{theo}}{N_{shaft}} = \frac{G_{cirk} \times \Delta h_{is}}{N_{shaft}}$$

where

V_{suct} = actual volume of suction gas per unit of time

V_{theo} = compressor swept volume per unit of time

G_{cirk} = mass of refrigerant circulated by the compressor per unit of time

v_{suct} = specific volume of suction gas (v_1) in Fig. 4

N_{theo} = theoretical power consumption for isentropic compression

N_{shaft} = actual power input to shaft

Δh_{is} = enthalpy gain from isentropic compression (h_2-h_1) in Fig. 4

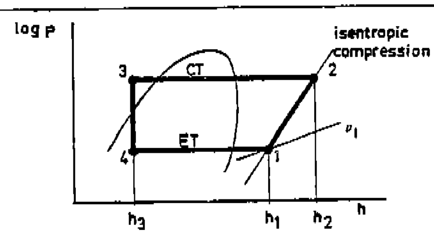

Fig. 4 - h - log p diagram

The volumetric efficiency is an indication of the refrigerating capacity of the compressor per unit of swept volume, and says nothing about the power consumption.
The isentropic efficiency indicates the power consumption, and is related to the C.O.P. (coefficient of performance) as follows:

$$COP = \frac{Q_o}{N_{shaft}} = \eta_{is} \times \frac{Q_o}{N_{theo}}$$

where Q_o is the compressor refrigerating capacity.

INFLUENCE OF COMPRESSION RATIO

Fig. 5 shows volumetric and isentropic efficiencies in relation to the compression ratio at various condensing temperatures.

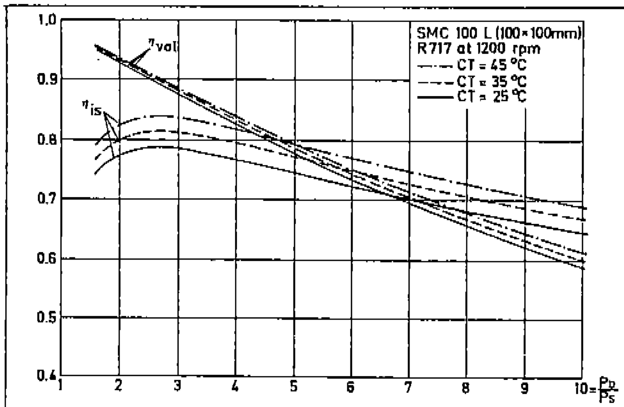

Fig. 5 - Volumetric and isentropic efficiency in relation to compression ratio

Note that the volumetric efficiency varies with the compression ratio, but is only slightly influenced by the pressure level as determined by the condensing temperature.
The isentropic efficiency also varies with the compression ratio, but the pressure level has much more influence than in the case of volumetric efficiency.

COMPRESSOR SPEED

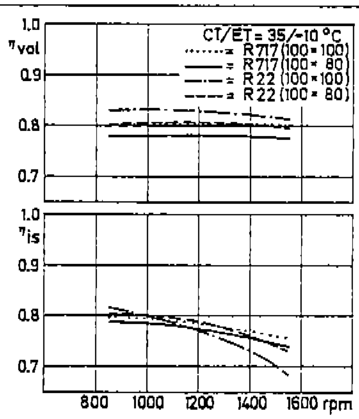

Fig. 6 - Compressor efficiencies in relation to compressor speed

The volumetric efficiency for both R717 and R22 is practically constant within the whole operating range from 1000 to 1500 rpm.
The isentropic efficiency for R717 decreases slightly with increasing speed, while the isentropic efficiency for R22 decreases relatively more with increasing speed.
The isentropic efficiency for R22 is reduced approx. 10%, when the speed is increased from 1000 to 1450 rpm at the operating conditions noted in Fig. 6, while the corresponding reduction for R717 is approx. 3.5%.
A compressor with 80 mm stroke and R22 would show a corresponding reduction of approx.6%.

LENGTH OF STROKE

Fig. 7 shows efficiencies with R22 and R717 for compressor type SMC 100 in L-version (100 mm stroke) and in S-version (80 mm stroke).

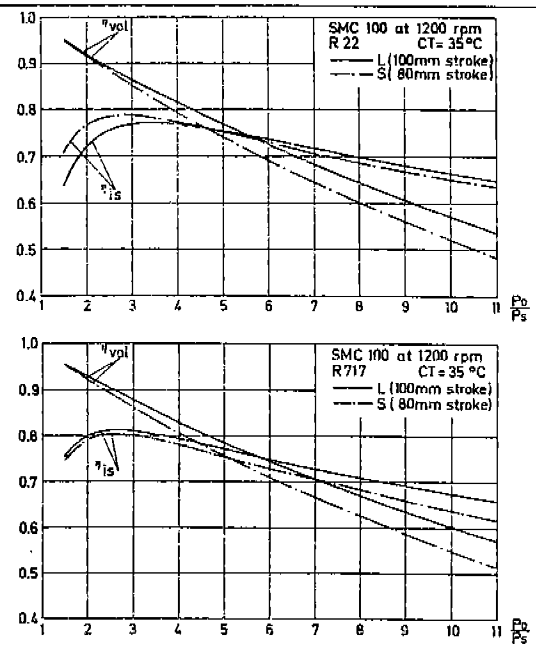

Fig. 7 - Influence of piston stroke on compressor efficiencies

An increase in stroke will result in higher volumetric efficiency for R22 as well as for R717, due to a relatively smaller clearance volume.
The isentropic efficiency for R22 will be reduced at small compression ratios, but increased at the high compression ratios, while the isentropic efficiency for R717 will increase overall when the stroke is increased.
Tests run at 1450 rpm still show a marked improvement in volumetric efficiencies for R22 and R717 with increased stroke. However, the reduction in isentropic efficiency with R22 at low compression ratios corresponding to relatively high evaporating temperaturs is considerable. This is due to the larger loss of energy in the increased mass flow of heavy R22 gas through the valves, resulting from the increase in piston stroke.

COMPRESSOR COMPARISONS

The comparisons in the following are based on twin-rotor screws with oil injection of the SRM type.
First a word of warning on comparisons between different types of machines.
Some years ago a European manufacturer of screw compressors published Fig. 8 (excluding the upper heavily traced curve). The curves marked 2.6, 3.5 and 4.5 represented the isentropic efficiency of his screw compressor, and the dotted line was said to represent reciprocating compressors.

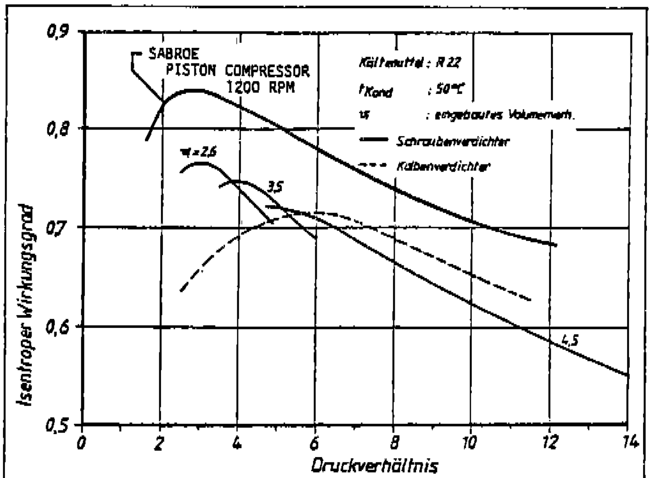

Fig. 8 - Example of comparison between screw compressor and reciprocating compressor

While we will readily admit that it is possible to find reciprocating compressors with very poor efficiencies, we state, that a modern, well designed reciprocating compressor should offer isentropic efficiency somewhat like the heavily traced curve, which happens to be a SABROE compressor at 1200 rpm. It should be mentioned, that there are 2 or 3 competing makes which are approximately on the same level.

SCREW COMPRESSORS

One of the characteristics of screw compressors, is the built-in volume ratio, which necessitates selection from a range of standard, built-in volume ratios.
The actual obtained energy efficiency depends on how well the selected volume ratio corresponds with the operating conditions.
Fig. 9 shows screw compressor efficiencies for two of the standard range of built-in volume ratios.

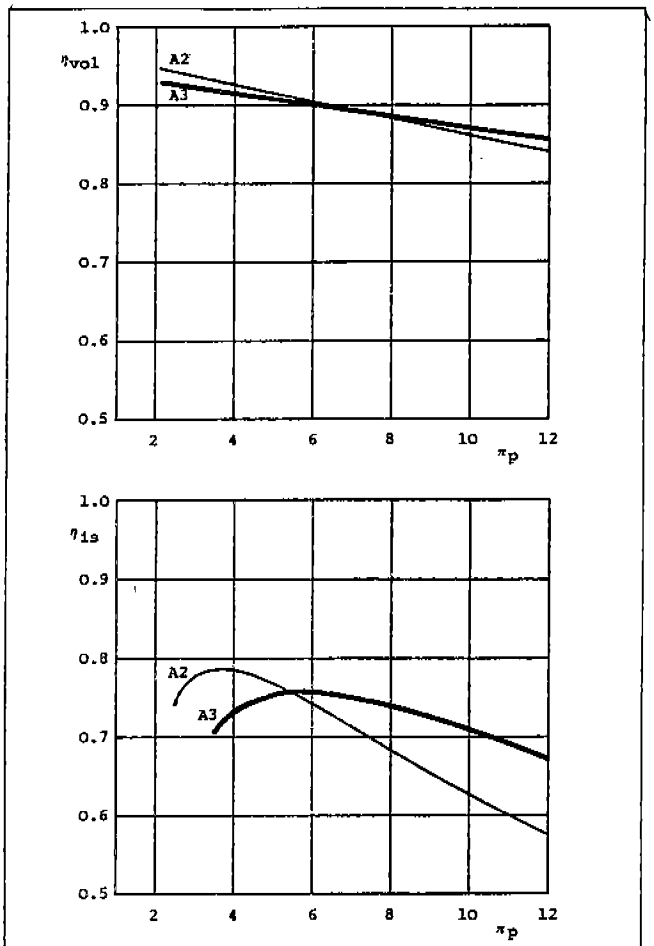

Fig. 9 - R717 screw compressor efficiencies

Type VMY 336, rotor dia. = 215 mm (8.46")
L/D = 1.6, n = 2950 rpm, CT = 30°C (86°F)
Port A2 w. volume ratio = 3.0
Port A3 w. volume ratio = 4.8

In some cases one compressor is specified to operate alternatively with a high compression ratio corresponding to a low temperature of evaporation, and with a low compression ratio corresponding to a higher evaporating temperature. In these cases it is advisable to select a compressor with a built-in volume ratio corresponding to the high evaporating temperature, in order to avoid internal over-compression resulting from the other alternative. Either way the end result will be lower efficiency when the compressor is working at the set of conditions for which it was not originally selected, but that, as always, is the price of compromises.

SCREW VERSUS RECIPS.

Objective comparisons between two different design concepts are always difficult to make in general terms. The various losses inherent in the two compressor types in question, are of different nature, and comparisons should refer to equal operating conditions for actual projects, and further be based on compressor types and sizes, which are relevant.

Fig. 10 shows volumetric and isentropic efficiencies for one reciprocating compressor and two screw compressors of different make and size, with the following basic data:

	SMC	VMY	125 LU
Make	own	own	compet.
Type	recip.	screw	screw
	SMC 100 L	VMY 336 H	125 LU
Dim. mm	100 x 100		
Rotor dia.		215 mm	125 mm
L/D		1.6	1.65
Vol.ratio		3.0	3.6
Speed rpm	1200	2950	3550
Swept vol.			
m^3/h	226	1381	353
CFM	133	812	208
Refrigerant	R717	R717	R717
Subcooling	0	0	0
Superheat	0	0	0
CT ^oC	30/50	30/50	30/50
^oF	86/122	86/122	86/122

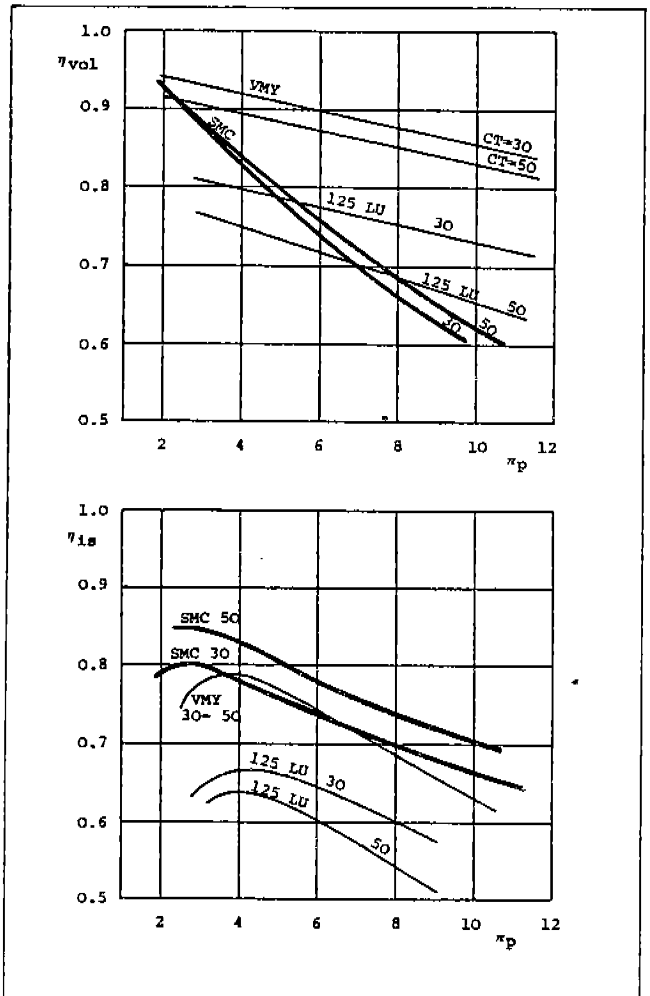

Fig.10 - R717 efficiencies for various compressor types

The comparison diagrams in Fig. 10 can be commented as follows:

- the volumetric efficiency of the reciprocating compressor (SMC) varies primarily with variations in the compression ratio (π_p), while a change in condensing temperature (pressure range) has little effect
- the volumetric efficiency of a large screw compressor (VMY) is depending on the compression ratio to a much lesser degree, than in the case of the reciprocating compressor, while change in pressure range has a larger effect, and in the opposite direction than in the case of the reciprocating compressor
- the volumetric efficiency of the VMY compressor is larger than that of the reciprocating compressor for all compression ratios larger than 2 - 2.5
- the smaller screw compressor (125 LU) has much lower volumetric efficiency than the large one, and it only becomes better than the reciprocating compressor for compression ratios above 5 - 8.

When we consider the isentropic efficiency the picture changes:

- the isentropic efficiency of the reciprocating compressor primarily depends on the compression ratio, and shows a maximum at about 2.5 -3. The isentropic efficiency is still to some extent depending on the pressure range, and it is noted that the isentropic efficiency increases with increasing condensing pressure
- the isentropic efficiency for screw compressors is much more influenced by the compression ratio than is the case for reciprocating compressors. The reason being, that the screw compressor has a built-in compression ratio (closely related to the built-in volume ratio), around which point the isentropic efficiency reaches its maximum. If the actual operating conditions deviate from the built-in compression ratio, the losses from either overcompression or undercompression will reduce the isentropic efficiency
- the isentropic efficiency of a smaller screw compressor (125 LU) is considerably lower than for a large screw compressor (VMY)
- a large screw compressor will show better isentropic efficiency at low condensing temperature, and operating conditions corresponding to its built-in volume ratio, than a reciprocating compressor; this a.o. makes the screw compressor very suitable for booster operation; at high condensing temperatures, however, the isentropic efficiency of the reciprocating compressor is better than that of the screw compressor
- the smaller screw compressor has considerably lower isentropic efficiency, and COP at all operating conditions, than the reciprocating compressor.

PART LOAD OPERATION

Another area where reciprocating and screw compressors have different characteristics is part loading.
Fig. 11 shows the relatively poor part loading performance of screw compressors. An average curve representing the stepwise cylinder unloading of a reciprocating compressor would show better performance than the lines representing the screw compressor at the same operating pressure ratio.

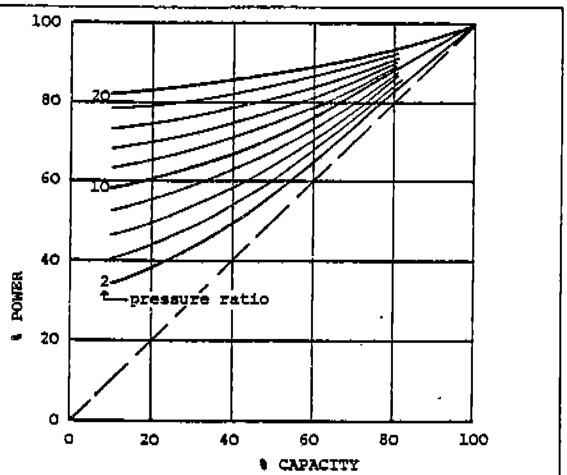

Fig.11 - Relation between power consumption and capacity for screw compressors at part load

This underlines the recommendation that screw compressors should not be used much below 50 - 60% of full capacity, at least not without realizing the resulting very low energy efficiency.

CONCLUSIONS

In many cases the final choice of compressor type will greatly influence the operating economy of the plant. The present paper has underlined some of the most important characteristics of two different compressor types. It is not intended to show that one is better than the other, nor, indeed that the two types mentioned are better than any other type of compressor available.
The comments offered on compressor efficiencies, part load performances, etc., in relation to operating conditions, lead to our conclusion, that selection of compressor sizes and compressor types should take all relevant characteristics into consideration, and that combinations, e.g. screw compressors as booster or base load machines with reciprocating compressors as high stage or as load adaptors would lead to very energy efficient installations.
Besides this issue, when mentioning energy efficiency, we should not overlook the many possibilities of saving of energy in existing plants or in new projects, simply by being more careful with design details such as temperature and pressure levels, pressure losses, heat influx, etc.