B. Degueurce and F. Banquet of Electricité de France and J. Denisant and
D. Favrat of Institute CERAC Switzerland discuss how energy savings can be
achieved by using twin screw compressors for steam compression.

Use of a twin screw compressor for steam compression
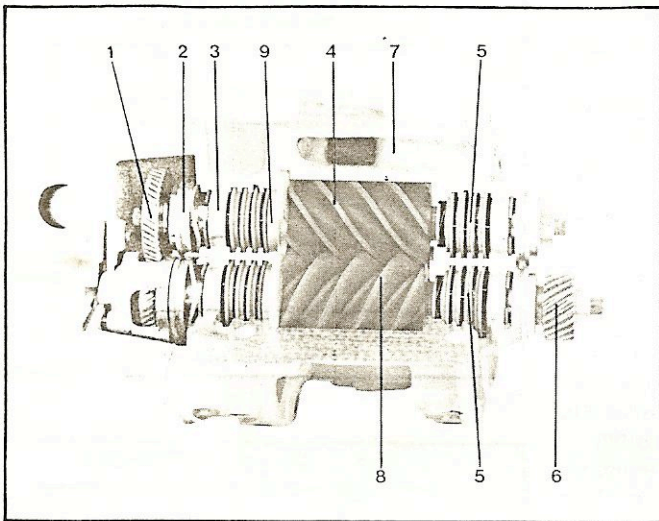
Introduction

Figure 1
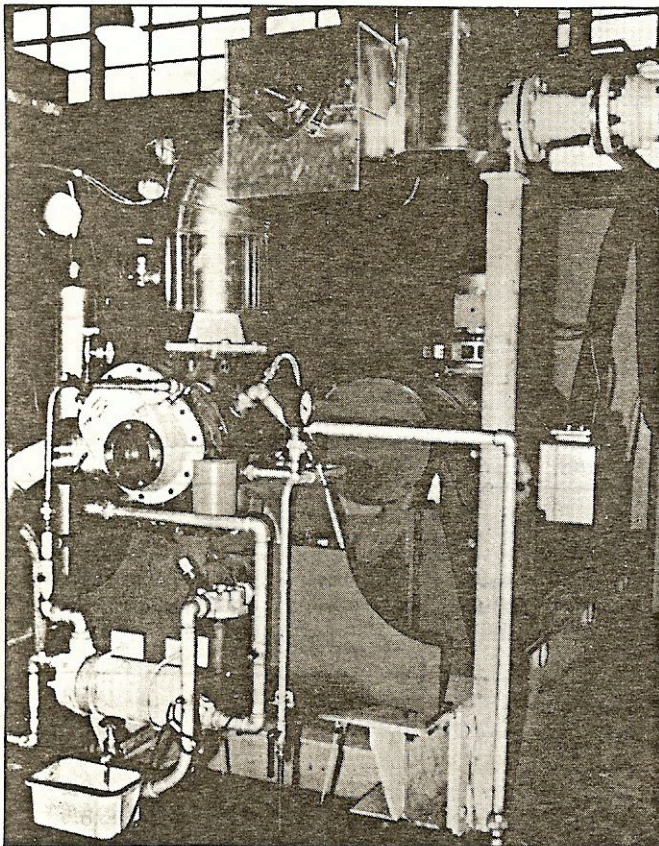Open view of an Atlas Copco twin screw compressor.

Figure 2
Installation of the twin screw compressor tested at Electricité de France.

BECAUSE OF its non-toxicity and its availability steam is frequently used as an energy carrier in industrial processes. Steam is also the by-product of energy intensive processes such as drying, concentration, evaporation and distillation. Although the upgrading concept of open cycle heat pump, commonly called mechanical vapour recompression (MVR), has been known and applied from the beginning of the century, low pressure steam from many industrial processes has traditionally been vented away with its valuable latent heat content.

As part of its efforts to promote a rational use of electricity and the substitution of imported fossil fuels, Electricité de France has been involved for many years in a significant research programme to stimulate the development of mechanical vapour recompression. Part of this research has been focussed on the key component, the steam compressor, and several prototypes of different technologies have been tested on EDF steam facilities described in previous papers (Reference 1,2).

Tests have been concentrated on the medium capacity range of volume flow up to 0.85 m^3/s, where very few compressor alternatives, if any, were available at the beginning of our research programme. In this capacity range and according to the potential need for compression ratios of three and above, depending on the application, rotary positive displacement compressors have early been recognized as a promising alternative.

The main part of this paper describes the results obtained with an oil-free twin-screw compressor of the standard air compressor series of Atlas Copco.

The twin screw compressor

The compressor (Figures 1 and 2) designed by Atlas Copco is based on an asymmetric 4.6 twin screw arrangement with both the male (8) and the female rotor (4) mounted on ball and roller bearings (2,3) within a close tolerance casing. The male rotor is driven by the motor through a speed multiplier gear box (6) normally conceived for driving two compressors in parallel. The female rotor is driven by a set of timing gears (1) which maintain the optimum play between the rotors while avoiding direct contact and potential wear. The diameters of the male and female rotor respectively, are 189 mm and 164 mm with a length of 284 mm.

The rotors were coated with a special coating in order to reduce the internal clearances and after a few hours of lapping, obtain a good internal sealing. The shaft seals are made of several self-adjusting rings (5) held in position by spring tension and forming three sealing chambers, of which two are opened to atmosphere with separate drains. The shaft seal arrangement is completed by an helical groove which pumps the oil back to the bearing chambers. A possibility exists to connect the drain from the rotor side to a vacuum pump in order to further reduce risks of oil contamination by the steam or to allow subatmospheric working conditions. The casing has a cooling jacket (7) to water cool during compression. The fixed built-in volume ratio of the compressor was equal to 2.1.

Test facilities

The facilities which are described in Reference 2, allow conditions of testing very similar to the operating conditions of an industrial heat pump. They include a desuperheater, a condenser, expansion valves, an evaporator and a superheater. The main difference being that the heat is supplied to the evaporator by direct contact of part of the superheated steam throttled at the outlet of the compressor. Whenever possible, critical measurement data, such as mass flow, shaft torque, pressures and temperatures were duplicated.

Mass flow and temperatures of the cooling water have also been measured.

The data acquisition and processing was computerized.

Tests

The objectives of the tests were to demonstrate that the compressor could operate with steam and to meausre the performance within a range of parameters compatible with Mechanical Vapour Recompression processes. With the exception of the connection of some seal drains to vacuum, the compressor had not been modified to adapt it to steam duties. Although the thermodynamic properties of the two fluids are different, the specific isentropic compression power is of the same order of magnitude at equivalent suction pressure. The main difference is the average level of gas temperature which is higher for steam at equivalent pressures and the effects of condensate which could result from a high rate of cooling of the casing.

The tests made at EDF were without direct water injection but with water cooling of the casing and permitted the study of temperature related problems (regulation of the temperature of cooling water, limitation of the peak temperature achievable, etc) and measurement of the detailed efficiencies of the compressor for different duty points and varying pressure ratios, pressure levels and speed of rotation.

The following definition for the efficiencies have been used:

— Discharge volumetric efficiency:
The volumetric efficiency of the compressor is the ratio of the volumetric flow of the steam measured at compressor discharge, reduced to the intake conditions, over the volume swept by the compressor.
— Global isentropic efficiency:
The global isentropic efficiency is the ratio of the theoretical isentropic compression power calculated for the discharge flow of the compressor, over the shaft power.
— Coefficient of performance (COP) of the heat pump associated with the compressor:
The practical coefficient of performance is defined as the ratio of the power that would be available at the condenser of the heat pump associated with the compressor over the

compressor shaft power. The CARNOT theoretical coefficient of performance is the ratio of the absolute saturation temperature at discharge over the difference between saturation temperature at discharge and intake.

Table 1 and Figures 3 and 4 summarize the main results obtained by varying the pressure ratio at nearly constant speed of rotation (1470 r/min) with a constant flow of cooling water. The volumetric efficiency is very stable over the range of pressure ratios measured indicating good internal clearances. The global isentropic efficiency is also stable for pressure ratios superior to 2 and gradually decline with diminishing pressure ratios as a result of the increased relative importance of the external losses and the increasing offset compared to the built-in pressure ratio.
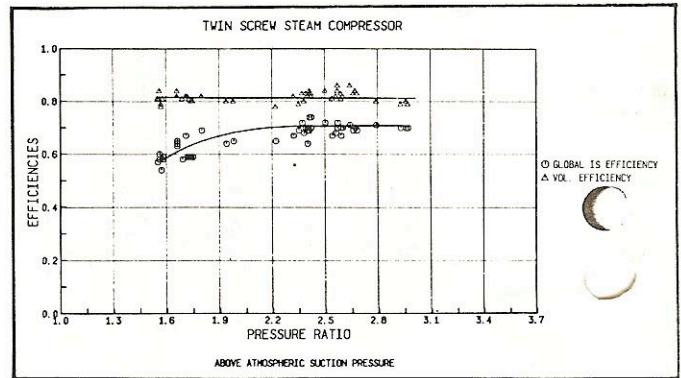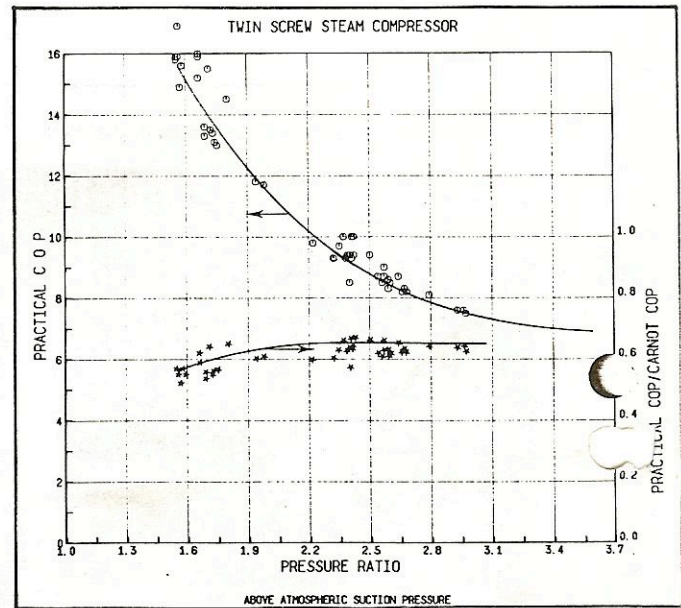

Figure 3

Figure 4

Suction		Discharge								
Temp. (0° C)	Pressure (kNm⁻²)	Temp. (0° C)	Pressure (kNm⁻²)	Pressure ratio	Suction flow (m³/h)	Shaft power (W)	Volumetric efficiency	Global isentropic efficiency	Practical COP	Practical COP CARNOT COP
114	157	162	245	1.55	2085	74097	0.81	0.57	15.8	0.55
110	108	159	188	1.73	2093	62378	0.81	0.59	13.3	0.56
109	122	178	242	1.98	2079	80731	0.80	0.65	11.6	0.61
124	115	217	276	2.40	2133	94324	0.82	0.71	9.2	0.63
108	112	209	285	2.54	2095	101509	0.81	0.68	8.5	0.62
118	123	246	365	2.96	2067	124501	0.79	0.70	7.6	0.64

Table 1 — Test Results versus Pressure Ratio

The practical COP values are excellent, in the range of 62 to 65% of the ideal Carnot COP, for most of the measured data. Values of practical COP higher than 7.5 are achieved even at a pressure ratio of 3 corresponding to a particularly high temperature lift.

Table 2 as well as Figures 5 and 6 show that a reduced suction pressure down to 64 kNm^{-2} does not affect the performance of the compressor. By lowering the suction pressure, pressure ratios up to 3.7 have been achieved within the temperature limit of the compressor (250° C), confirming the stability of the global isentropic efficiency at higher pressure ratio than built-in.

Speed (r/min)	Pressure ratio	Suction flow (m^3/h)	Shaft power (W)	Volumetric efficiency	Global isentropic efficiency	Practical COP CARNOT COP
1563	1.97	2255	80725	0.82	0.65	0.61
1468	1.98	2093	74884	0.81	0.65	0.60
1366	1.97	1907	68965	0.79	0.64	0.59

Table 2 — Test Results versus Speed of Rotation
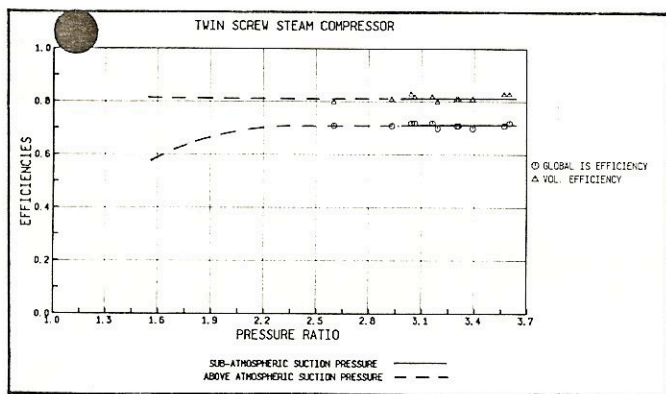
Figure 5
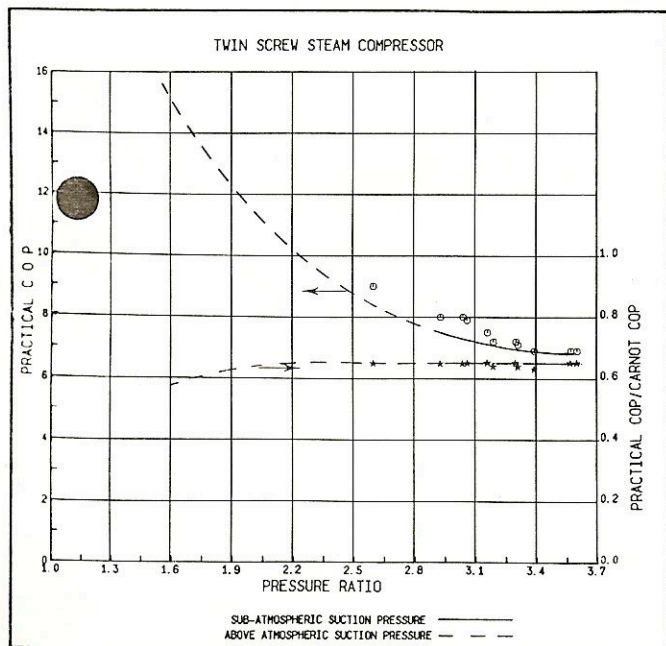
Figure 6

Suction		Discharge					
Temp. (0° C)	Pressure (kNm^{-2})	Temp. (0° C)	Pressure (kNm^{-2})	Pressure ratio	Volumetric efficiency	Global isentropic efficiency	Practical COP CARNOT COP
93	64	238	230	3.61	0.83	0.72	0.65
96	72	217	222	3.07	0.82	0.72	0.65
101	96	196	250	2.60	0.80	0.71	0.65

Table 3 — Test Results at Sub-Atmospheric Suction Pressures

Similar stable results have been obtained by varying the speed of rotation as shown in Table 3 which indicate a good ability of this type of compressor to adapt to variable speed drives whenever necessary for regulation purposes.

Test observations

The cumulated testing time for this compressor was 220 hours. Observations of the rotors at the beginning and at the completion of the tests indicated that the coating was in perfect condition, apart from a slight change in colour.

One possible problem was the noise level from this non-insulated prototype measured at 97 dB(A) at a distance of one metre. However, from experience on air compressors one knows, that a normal acoustic enclosure for the compressor will overcome this problem.

Developments

Based on these very encouraging results, a new version of this type of compressor has been developed specially adapted for steam, with stainless steel rotors, the possible use of buffer fluids in the shaft seals, and the optional water injection for superheat control during compression at higher pressure ratios. Commercial units are now operating in the range of inlet volume flows of 0.14 m^3/s to 6 m^3/s and with single stage pressure ratios up to 8. Electricité de France is planning to introduce a similar unit in a closed cycle heat pump prototype used for a drying installation.

Conclusion

These test results together with previously published data (References 1, 2, 4) are intended to give the process designer some of the previously missing key information required for the evaluation of mechanical vapour recompression installations. The good performance exhibited suggests that payback periods as short as two years or less can be realistically envisaged. Twin screw steam compressors will adequately complement other technologies commonly available such as centrifugal compressors in the low to medium range of capacities in particular. Present results with twin screw compressors are very encouraging and open the door for further energy savings by extending the application of the MVR concept to installations requiring increased temperature lifts.

References

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