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Rating Technique for Refrigeration Twin-Screw Compressors

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ABSTRACT

The twin-screw compressor is constantly attaining higher performance due to new rotor profiles, new rotor manufacturing technology and new designs. Besides this, new refrigerants as well as new lubricants can be expected on the market. Therefore there is always a need to update performance data. To minimize the number of test points and still receive representative data, there is a need of an accurate and compact rating equation system. Certain relationships have become clear for representation of volumetric efficiency and power consumption as a result of analysing the behaviour of the twin-screw compressor. The relationships include fixed and variable volume ratio, as well as economizer operation. Basic rating equations are described and comparisons between rating equation data and test data are made.

INTRODUCTION

The curve fitting technique described in this paper is mainly developed for the oil-flooded twin-screw refrigeration compressor (ref. 1). The twin-screw compressor is a positive displacement compressor with a built-in volume ratio, no suction or discharge valves, no clearance volume, but with a certain amount of internal leakage. This fact makes it to behave differently from other positive displacement compressors, like the reciprocating compressor. Rating technique for reciprocating compressors is described in reference 2. The performance rating data has to be based on test data and/or performance simulation data (ref. 3, 4, 5). Such performance simulation programs should be used for analysing the compressor itself and they are normally too complicated and the executing time is too long for refrigeration system analysis. The rating equations described in this paper are more suited for refrigeration system design, and analysis, centered around the twin-screw compressor.

CAPACITY CALCULATIONS

The refrigeration capacity Q (kW) is:

$$Q = NVOL \times VS \times \frac{NM}{60} \times \frac{DH}{V} \qquad (1)$$

where NVOL = volumetric efficency,
VS = theoretical compressor displacement (m^3/rev),
NM = input male rotor speed (rpm),
V = specific volume at suction (m^3/kg),
DH = enthalpy difference over the evaporator (kJ/kg).

Seeing this equation it is natural to curve fit the volumetric efficiency instead of the refrigeration capacity. The volumetric efficiency is normally a straight line when plotted against pressure ratio (for constant discharge pressure).

The volumetric efficiency drops when the pressure ratio is increased and the drop is magnified by:

- low built-in volume ratio,
- low rotor speed,
- low viscosity of lubricant-refrigerant mixture.

Figure 1 shows the volumetric efficiency for a given refrigerant, oil, discharge pressure and rotor speed.

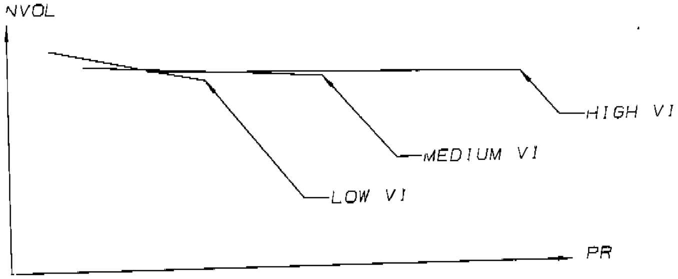

FIGURE 1. VOLUMETRIC EFFICIENCY (NVOL) VS. PRESSURE RATIO (PR)

The relative level of the volumetric efficiency drops at higher discharge pressures. For a given screw compressor, refrigerant and lubricant, following equation is created for the volumetric efficiency NVOL:

$$NVOL = VO(1) + VO(2) \times PD + VO(3) \times (NO/NC)^{R} \times PRC \times (3.5/VI)^{VR} \qquad (2)$$

where VO(1), VO(2), VO(3), R, VR are empirical determined coefficients for the volumetric efficiency equation,
PD = discharge pressure (MPa),
NO = nominal male rotor speed for the coefficients (rpm),
NC = calculation male rotor speed (rpm),
PRC = calculation pressure ratio,
VI = built-in volume ratio.

VR, the coefficient for the VI dependence is normally quite small and can for approximative calculations be set to 0.
R, the coefficient for the rotor speed dependence is normally 0.5 to 1.
The VO(1) coefficient gives the relative level of NVOL.
The VO(2) coefficient (negative) gives the discharge pressure dependence.
The VO(2) coefficient (negative) gives the pressure ratio dependence.

There are limitations to how much the volumetric efficiency can increase with rotor speed. When comparing speed it is common to relate to male rotor tip speed. The male rotor tip speed UM (m/s) is:

$$UM = \frac{PI \times DM \times NM}{60} \qquad (3)$$

where DM = male rotor diameter (m),
NM = input male rotor speed (rpm),
PI = 3.1416.

The nominal male rotor tip speed (the tip speed for which the coefficients where fitted for) NO (m/s):

$$UO = \frac{PI \times DM \times NO}{60} \qquad (4)$$

Common limitations for the NVOL increase with high male rotor tip speed are:
If UM is larger than 40 m/s, let $NC = \frac{40 \times 60}{DM \times PI}$ in equation (2).

If UO is larger than 40 m/s and UM is larger than 40 m/s let NC = NO in equation (2). Else NC = NM in equation (2).

At low pressure ratios the volumetric efficiency curve has a tendency to flatten out, especially for low rotor speeds. This is handled by letting the volumetric efficiency curve be independent of pressure ratio at low pressure ratios.

Cut-off point for NVOL increase at low pressure ratios:

PRCUT = constant - constant x UM
(Common is: PRCUT = 4.667 - 0.0667 x UM)

If PR is smaller than PRCUT let PRC = PRCUT else PRC = PR in equation (2)
where PR = actual pressure ratio.

POWER CALCULATIONS

The three basic alternatives for curve fitting are:

- power or torque,
- isentropic efficiency,
- power or torque loss.

Figure 2 shows the isentropic efficiency NIS and torque M versus pressure ratio for a screw compressor with 3 different volume ratios and optimal volume ratio.

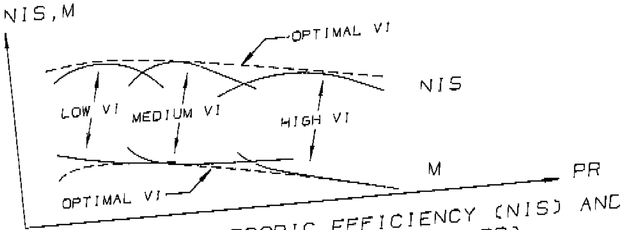

FIGURE 2. ISENTROPIC EFFICIENCY (NIS) AND TORQUE (M) VS. PRESSURE RATIO (PR)

The relative level of the isentropic efficiency drops at low discharge pressures (booster duty) due to the larger relative power losses within the machine. With this knowledge in mind a torque loss approach is appropriate. The theoretical process to compare with should be a process, which includes the fixed volume ratio dependence, since the screw compressor is a machine with fixed inlet and outlet ports.

The technical work per mass unit ETR (J/kg) for isentropic compression is:

$$ETR = \frac{n}{n-1} \times PS \times 10^6 \times V \times \left(PR^{\frac{n-1}{n}} - 1\right) \qquad (5)$$

where n = isentropic exponent,
V = specific volume at suction (m^3/kg),
PS = suction pressure (MPa).

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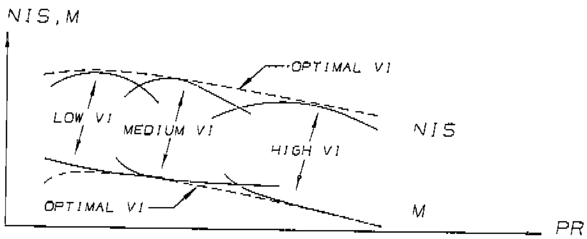

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ECONOMIZER CALCULATIONS

The twin-screw compressor can be equipped with a secondary suction port situated between the primary suction port and the discharge port. The refrigerant is expanded in two steps and the flash gas created at the intermediate pressure level (in a flash tank), is supplied to the secondary suction port, often referred to as the economizer port (ref. 5). The high pressure liquid can also be subcooled with a heat exchanger where the economizer port is handling the evaporated refrigerant coming from the heat exchanger. These arrangements give increased refrigeration capacity, as well as improved COP (the Coefficient of Performance).

The volumetric efficiency is reduced due to the injected vapour. This reduction is mainly dependent on male rotor tip speed UM (m/s).

$$\frac{NVOLE}{NVOL} = A_V + B_V \times UM \tag{12}$$

where NVOLE = volumetric efficiency with economizer (to be used in equation (1),
NVOL = volumetric efficiency without economizer,
A_V, B_V = empirical determined coefficients.

The economizer pressure PE (MPa) drops, normally with increased pressure ratio (constant discharge pressure), see figure 3.

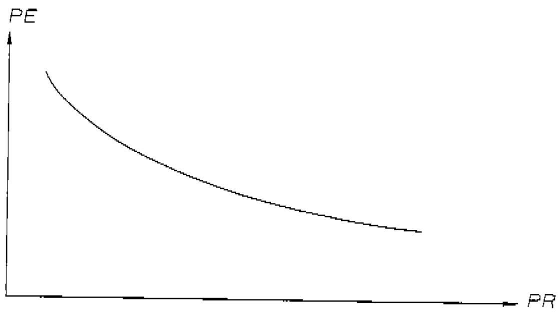

FIGURE 3. ECONOMIZER PRESSURE (PE) VS. PRESSURE RATIO (PR)

An equation of second order will fit the economizer pressure.

$$PEO = A_{PE} + B_{PE} \times PR + C_{PE} \times PR^2 \tag{13}$$

where PEO = nominal economizer pressure (MPa), the economizer pressure corresponding to the nominal discharge pressure PDO (MPa),
A_{PE}, B_{PE} C_{PE} = empirical determined coefficients.

The economizer pressure, at a constant pressure ratio, is basically proportional to the discharge pressure:

$$PE = \frac{PEO}{PDO} \times PD \tag{14}$$

The economizer pressure is, of course, dependent of port size, port location and refrigerant etc.,however, an approximate economizer pressure can be estimated:

$$PE = \frac{(PD \times PS \times 100)^{1/2} - \text{constant}}{10} \quad (15)$$

where the constant is about 1 for a compressor with the economizer port close to the suction port. This equation is more suited for extrapolations then equation (13) and (14).

For an economizer with flash tank, use the corresponding saturated liquid enthalpy, at the economizer pressure PE, in equation (1).

For an economizer with heat exchanger, use the enthalpy of the subcooled liquid leaving the heat exchanger in equation (1).

The power consumption for a compressor with economizer can be handled by curve fitting the power increase with economizer.

$$\frac{EWE}{EW} = A_E + B_E \times PR + C_E \times PR^2 \quad (16)$$

where EWE = power consumption with economizer (kW),
EW = power consumption without economizer (kW),
A_E, B_E, C_E = empirical determined coefficients for the economizer power multiplier equation.

This equation (16) is practically independent of discharge pressure, but dependent of built-in volume ratio VI (different equations for different built-in volume ratios). C_E is normally very small and can at least for low built-in volume ratios be eliminated.

EXAMPLE

The SRM compressor K 608 is tested and the test results are fitted with the above described method, see figure 4 and 5.

The K 608 compressor:
Male rotor diameter = 60 mm (2.36 inch)
Built-in volume ratio = 3.0 (open VI lift valve)
Displacement = 0.1502 x 10^{-3} m^3/rev (5.30 x 10^{-3} ft^3/rev)
Compressor cooling = uncooled
The compressor is equipped with two capacity lift valves, one VI lift valve and an economizer port.

Conditions:
Refrigerant = R12
Lubricant = Polyalphaolefin ISO 400
Condensing temperature = 50°C (122°F)
Male rotor tip speed = 11.8 m/s (38.7 ft/s)

It should be mentioned that this compressor is small for a screw compressor and the male rotor tip speed is low. Most common tip speed is 20 to 60 m/s (65 to 197 ft/s). However, the small tip speed in this example was selected to make it little harder to curve fit the test results.

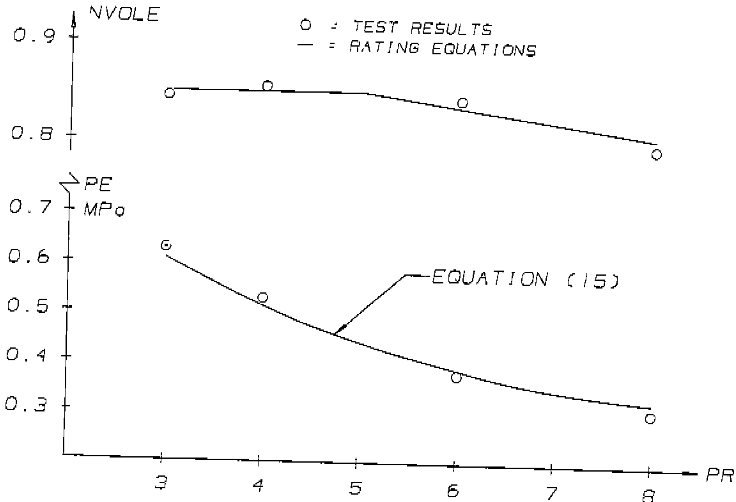

FIGURE 4: VOLUMETRIC EFFENCIENCY WITH ECONOMIZER (NVOLE) AND ECONOMIZER PRESSURE (PE) VS. PRESSURE RATIO (PR)

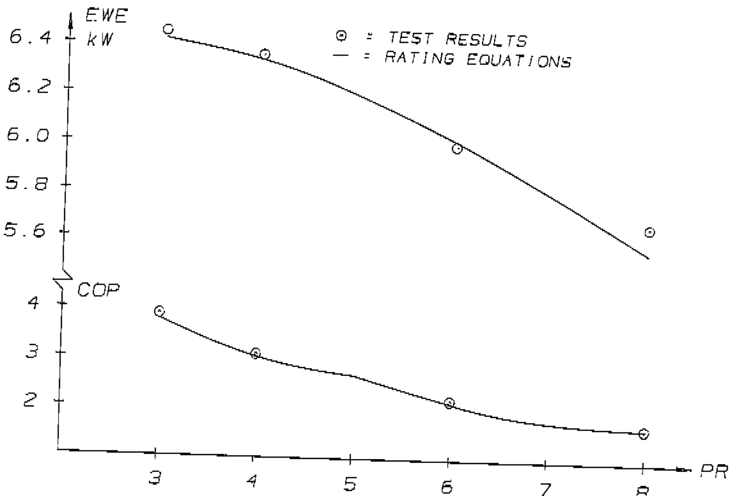

FIGURE 5: POWER CONSUMPTION (EWE) AND COP BOTH WITH ECONOMIZER VS. PRESSURE RATIO (PR)

CONCLUSION

A compact and accurate compressor rating equation system, for the refrigeration twin-screw compressor, has been described. This compressor rating procedure can be used both by compressor manufacturers and refrigeration system builders. Ultimately, it will help the refrigeration system designers to take full advantage of the screw compressor.

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