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Important Parameters for Small, Twin-Screw Refrigeration Compressors

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IMPORTANT PARAMETERS FOR SMALL, TWIN-SCREW REFRIGERATION COMPRESSORS

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ABSTRACT

This paper deals with open-shaft twin-screw refrigeration compressors in the 10 - 175 m^3/h (10 - 100 CFM) range.

Applications for such compressors are, for example, external combustion engine heat-pumps and transport refrigeration.

The twin-screw compressor concept is discussed from following view-points:

- operational mode and rotor housing design
- drive arrangement
- bearings
- lubricants
- cooling
- volume ratio
- capacity control
- economizer
- performance

INTRODUCTION

The twin-screw compressor is well established in the industrial refrigeration and air-conditioning field. To some extent it is also applied for bus air-conditioning and larger commercial air-conditioning. However, for smaller capacities the use of twin-screw compressors has been limited. Up till now the small twin-screw compressor has often been a scaled-down industrial refrigeration compressor or a converted air compressor. Also "the manufacturing state-of-the-art" has not been ready for large scale production of small high precision rotors. To-day, however, the situation is different.

OPERATIONAL MODE AND ROTOR HOUSING DESIGN

The compressor can be designed for oil-flooded or oil-reduced operation (reference 1). Oil-flooded operation is preferred at high pressure ratios and low rotor tip speed. Oil-reduced operation is preferred at low pressure ratios and high rotor tip speed.

The compressor housing or at least the discharge end plate should have similar thermal characteristics as the rotors. If this is not the case, a less favourable operational mode must be selected, viz. liquid refrigerant flooding (compare reference 1).

DRIVE ARRANGEMENT

Since the compressor is quite small (40 - 100 mm rotor dia) it is important to keep the rotor tip speed at a reasonably high level. This can be done with belts, gears or female rotor drive.

ROTORS

The rotors can be designed for oil-reduced as well as for oil-flooded operation (reference 1). The rotor production method/material should be selected for low cost and mass production. Since the rotors are so small the interlobe clearances are critical for the performance.

The rotor shaping methods available to-day are:

- single index milling
- hobbing
- grinding
- extrusion (for aluminium, close to final shape)
- injection moulding (for polymers, to final shape)

The polymer rotors have not yet been fully evaluated for high pressure refrigeration duty but results so far look promising.

BEARINGS

Antifriction bearings are preferred for this compressor size. The bearings should be able to operate with oil-mist lubrication as well as forced oil lubrication. Suitable types are cylindrical, tapered roller and ball bearings.

LUBRICANTS

Besides the normal requirements for a refrigeration compressor lubricant there is also the importance of good miscibility with the refrigerant so adequate oil circulation will be assured in the system (reference 1).

COOLING

When necessary, the discharge temperature is limited with external cooling of the oil (only oil-flooded operation) or liquid refrigerant injection. Regarding the performance loss with liquid refrigerant injection for a modern screw compressor, see fig. 1.

VOLUME RATIO

For applications like heat pumps and transport refrigeration, the compressor will operate over a wide range of pressure ratios. This calls for a variable built-in volume ratio (V_i).

In view of the small size, the most suitable type of V_i control would be that using a lift valve. The V_i slide valve may also be taken into consideration, but as a more costly alternative. A more detailed discussion about variable volume ratio can be found in reference 2.

CAPACITY CONTROL

When the compressor is driven by an engine, it is natural to use variable speed for capacity control. To increase the capacity range for a given speed range, the compressor could be equipped with a "dynamic suction port" (reference 2).

If the variable speed does not give sufficient capacity reduction, a capacity lift valve (reference 2) or an economizer by-pass valve can be used. The principle for the economizer by-pass arrangement is the following: the economizer port takes care of flash gas in one mode (full load) and the same port works as a bleed-off port in another mode (part load), in which case the economizer gas is by-passed to suction. The economizer by-pass valve can be incorporated in the compressor housing (figure 2).

ECONOMIZER

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 throttled in two steps and the flash gas created at the intermediate level is supplied to the secondary suction port, i.e. the economizer port. This arrangement gives increased refrigeration capacity, as well as improved compressor efficiency (see figure 3).

The main feature of a twin-screw compressor with economizer and variable built-in volume ratio is that the discharge port can be corrected for the gas coming from the intermediate pressure level as well as for the gas coming from suction. Figure 4 indicates the relationship between the optimum built-in volume ratios for compressors with and without economizers.

To get a better appreciation of the economizer performance there is a need for representation also in absolute values. Since the most common efficiency for compressor performance is the isentropic efficiency, there should be an isentropic efficiency also for a compressor operating with economizer. Figure 5 and 6 show two isentropic efficiencies for the economizer operation. One is proportional to COP and one is more strict from thermodynamic stand-point.

Figure 7 shows the performance with and without economizer and with the different isentropic efficiency definitions. The strict isentropic efficiency with economizer ($\eta_{is,ECO,S}$) is somewhat lower (at low pressure ratios) than the isentropic efficiency without economizer (η_{is}). The corresponding loss comes mainly from leakage from the supercharged thread to suction. This can also be seen in the volumetric efficiency.

At high pressure ratios, however, the strict isentropic efficiency with economizer is higher than the isentropic efficiency without economizer because the economizer-equipped compressor can operate without undercompression at a higher pressure ratio than the compressor without economizer. The isentropic efficiency proportional to COP and valid for economizer operation ($\eta_{is,ECO,C}$) is of course always larger than the isentropic efficiency without economizer.

PERFORMANCE EXAMPLES

<u>Larger Compressor</u>

Compressor: twin-screw*

Operation: oil-flooded (cooled oil)
with and without economizer

* SRM K 318

Rotors: 5-6 (male-female) lobe combination
Male rotor diameter 113.4 mm (4.465 inch)
Male rotor material: steel
Female rotor diameter 95.8 mm (3.772 inch)
Female rotor material: nodular iron
Rotor length: 150 mm
Male rotor drive

Displacement: 175.3 m^3/h (103.2 CFM) at 3550 RPM

Built-in volume ratio: optional within 2.0 - 5.7

Lubricant: PAO ISO 200

Oil temperature to compressor: 45^{o}C (113^{o}F)
Performance for R22, see figure 7.
Further details, see reference 1.

Smaller Compressor

Compressor: twin-screw*

Operation: oil-flooded (uncooled oil)

Rotors: 4-6 (male-female) lobe combination
Male Rotor diameter: 47 mm (1.85 inch)
Female rotor diameter: 44.5 mm (1.75 inch)
Rotor material: steel
Rotor length: 80 mm (3.15 inch)
Female rotor drive

Displacement: 22.3 m^3/h (13.1 CFM)
at 3000 RPM (female rotor)

Built-in volume ratio: 2.7 and 3.5

Lubricant PAO ISO 400

Performance for R22 see figure 8.
Performance for R12 see figure 9.
The variation in efficiencies represent different rotor clearancies and oil-draining arrangements.

* SRM K 257

Small Booster Compressor

Compressor: twin-screw*

Operation: oil-flooded

Rotors: 3-5 (male-female) lobe combination
Male rotor diameter: 69.0 mm (2.717 inch)
Male rotor material: Aluminium
Female rotor diameter: 60.8 mm (2.394 inch)
Female rotor material: Injection moulded polymer
Rotor length: 138 mm (5.43 inch)
Male rotor drive

Displacement: 59.5 m^3/h (35 CFM) at 3000 RPM

Built-in volume ratio: 2.5
Lubricant: PAO ISO 68
Oil temperature: 30°C (86°F)
Performance for R22, see figure 10

CONCLUSION

Important parameters for small open-shaft refrigeration twin-screw compressors have been discussed. For a given application, there might be a number of design possibilities, but normally only a few design that can adequately fulfill the most basic requirements of reliability, high performance and low cost. It appears that the twin-screw compressor can meet also the demands given by small, open-shaft compressor applications.

REFERENCES

1) Sjöholm, L.

Different Operational Modes for Refrigeration Twin-Screw Compressors

Purdue Compressor Technology Conference (1986).

2) Sjöholm, L.

Variable Volume-Ratio and Capacity Control in Twin-Screw Compressors.

Purdue Compressor Technology Conference (1986).

* SRM K 319

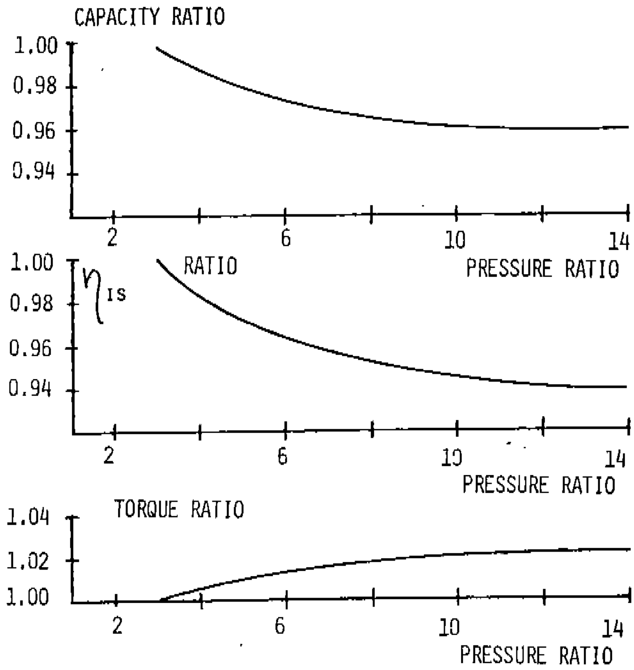
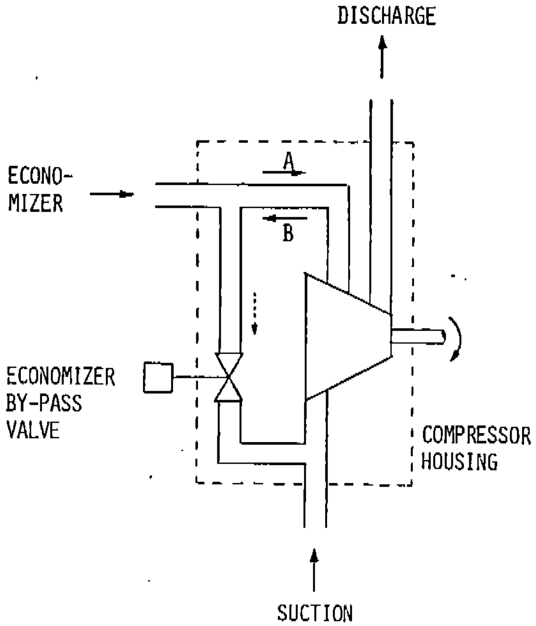


FIG. 1: PERFORMANCE LOSS WITH LIQ. REF. INJECTION
R 22, COND. TEMP. = 54.4°C (130°F)
V_I = 2-5.7
INTERNAL COOLING (LIQ. INJ.) COMPARED
WITH EXTERNAL COOLING (WATER COOLED OIL)



A: ECONOMIZER BY-PASS VALVE CLOSED

B: ECONOMIZER BY-PASS VALVE OPEN

FIG. 2: PRINCIPLE OF ECONOMIZER BY-PASS VALVE

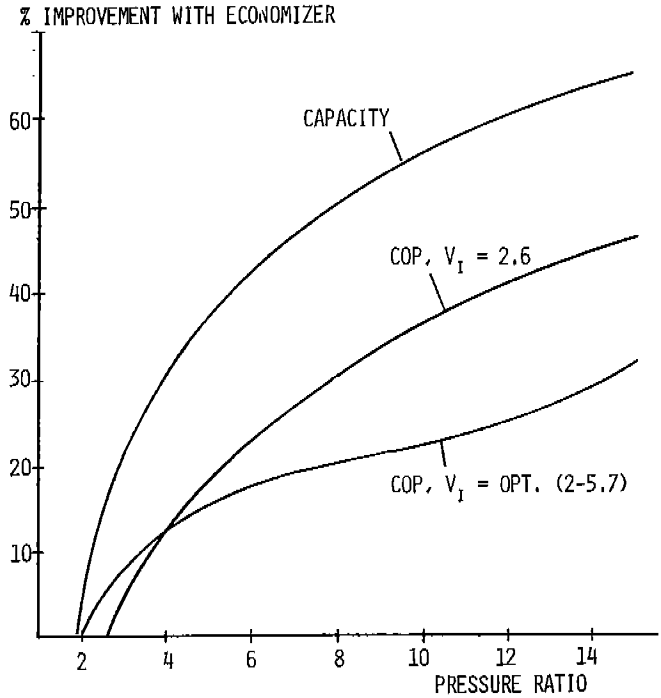


FIG. 3: PERFORMANCE COMPARISON, WITH AND WITHOUT ECONOMIZER
R 22, COND. TEMP. = 54.4°C (130°F)

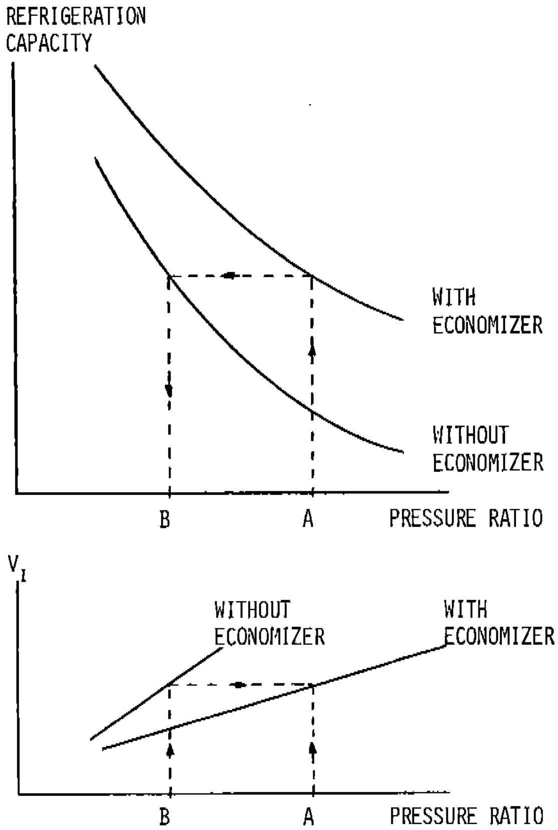


FIG. 4: PRINCIPLE OF V_I-CORRECTION FOR ECONOMIZER OPERATION

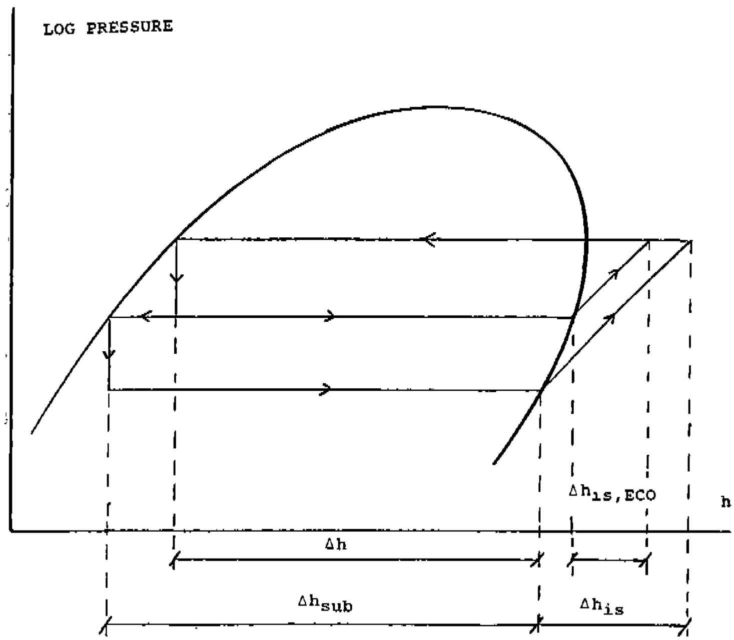


FIG. 5: DEFINITIONS OF ENTHALPIES

WITHOUT ECONOMIZER

$$\eta_{is} = \dot{m}_{in} \times \Delta h_{is} \times \frac{1}{\underline{P}}$$

WITH ECONOMIZER

1) ISENTROPIC EFFICIENCY PROPORTIONAL TO COP

$$\eta_{is,ECO,C} = \dot{m}_{in} \times \Delta h_{is} \times \frac{1}{\underline{P}_{ECO}} \times \frac{\Delta h_{sub}}{\Delta h}$$

2) STRICT ISENTROPIC EFFICIENCY

$$\eta_{is,ECO,S} = \frac{\dot{m}_{in} \times \Delta h_{is} + \dot{m}_{ECO} \times \Delta h_{is,ECO}}{\underline{P}_{ECO}}$$

where $\dot{m}_{in}$ = mass flow into compressor at suction (kg/s)

$\dot{m}_{ECO}$ = mass flow into economizer port (kg/s)

h = spec. enthalpy (KJ/kg)

$\underline{P}$ = compressor input power (kW)

$\underline{P}_{ECO}$ = compressor input power with economizer (kW)

FIG. 6: DEFINITIONS OF ISENTROPIC EFFICIENCIES

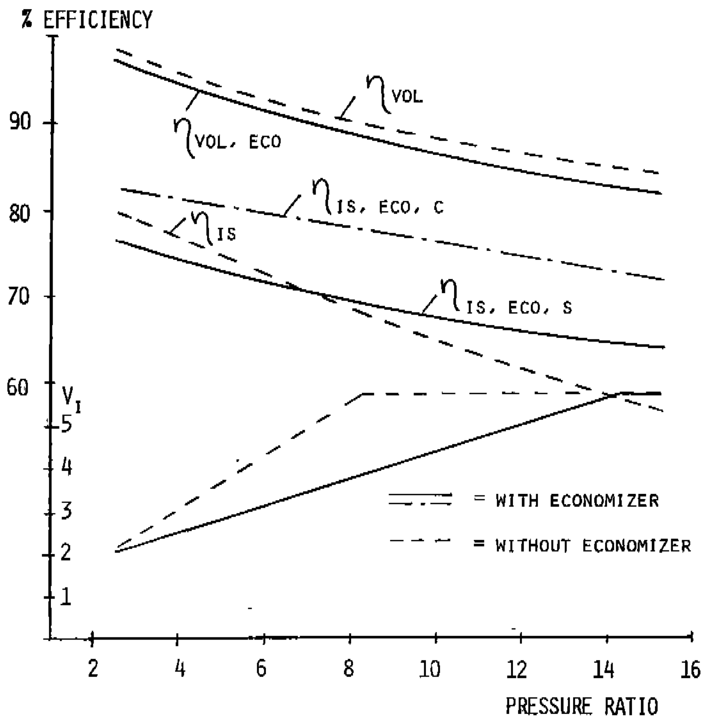


FIG. 7: PERFORMANCE WITH ECONOMIZER
DISPLACEMENT = 175.3 M^3/H (103.2 CFM)
AT 3550 RPM
R 22, COND. TEMP. = 40.6°C (105°F)

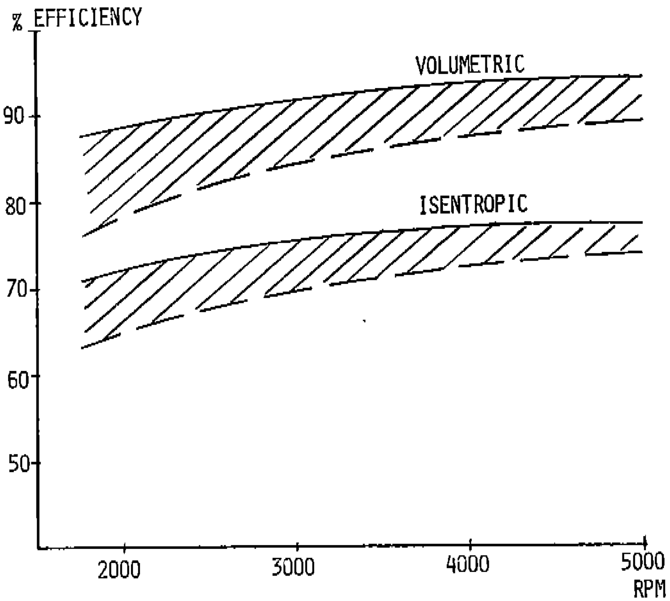


FIG. 8: PERFORMANCE OF TWIN-SCREW COMPRESSOR
DISPLACEMENT = 22.3 M^3/H (13.1 CFM)
AT 3000 RPM
R 22, COND. TEMP. = $50^{o}C$ ($122^{o}F$)
PRESSURE RATIO = 3.0
V_I = 2.7

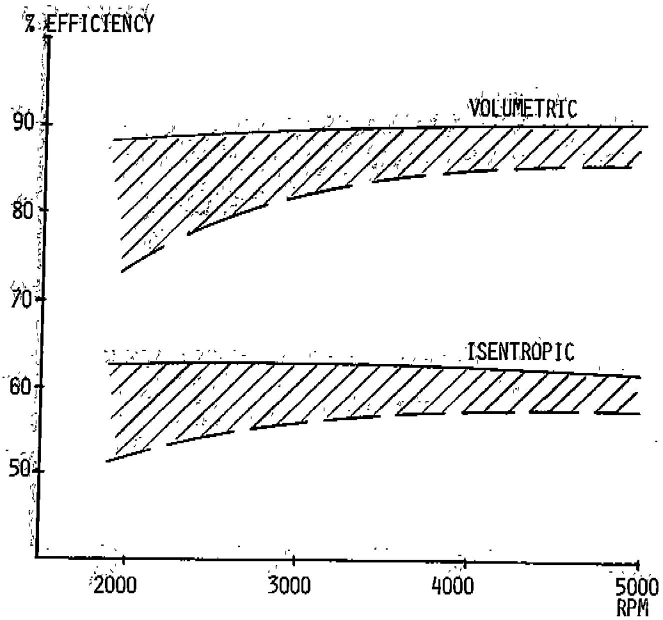


FIG. 9: PERFORMANCE OF TWIN-SCREW COMPRESSOR
DISPLACEMENT = 22.3 M^3/H (13.1 CFM)
AT 3000 RPM
R 12, COND. TEMP. = 32°C (89.6°F)
EVAP. TEMP. = -23.5°C (-10.3°F)
PRESSURE RATIO = 6
V_I = 3.5

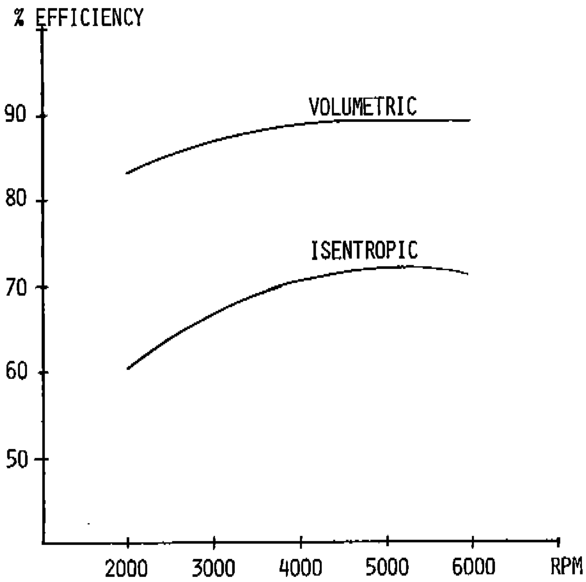


FIG. 10: PERFORMANCE FOR SMALL BOOSTER COMPRESSOR
DISPLACEMENT = 59.5 M^3/H (35 CFM)
AT 3000 RPM
R 22, COND. TEMP. = -6.67°C (20°F)
EVAP. TEMP. = -34.4°C (-30°F)
$V_I = 2.5$