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THERMODYNAMIC CALCULATION OF A DUAL SCREW COMPRESSOR BASED ON EXPERIMENTALLY MEASURED VALUES TAKING SUPERCHARGE INTO ACCOUNT

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1. INTRODUCTION

In the sphere of compressor engineering the screw compressor is characterised by an ever growing importance. The screw compressor has been used to compress air and gas, but it is increasingly being used as refrigerating compressor. Its advantages are a good controlability, high ride quality, and a high reliability and duration of life.

In refrigeration technology the screw compressor has been applied in a wide temperature range. At high pressure ratios, e.g. in the low temperature region the leakage pathes are increasingly responsible for the reduced efficiency. By means of the super charge process, i.e. the economiser cycle, the energetic efficiency may substantielly be increased.

2. COMPRESSION PROCESS WITHOUT SUPERCHARGE

Contrary to the reciprocating compressor the compression chamber of which is relatively tight the compression chamber of the screw compressor is characterised by a number of leakage flows. See Fig. 1.

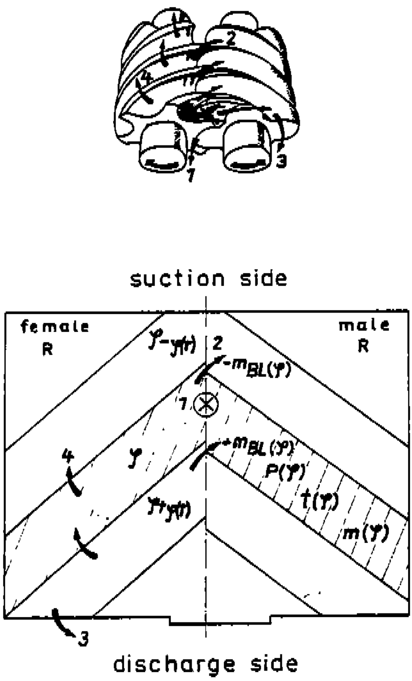

Fig. 1 - Overview of the different types of gap flow at a screw compressor

1 - gap flow through the gear path, 2 - gap flow through the blow hole, 3 - gap flow through the face path, 4 - gap flow through the pheriferal path, φ - rotary angle at the main rotor, $\varphi(\tau)$ - partition angle at the main rotor

Losses resulting from leakage must be taken into consideration when the thermodynamic process is being simulated. See Fig. 2.

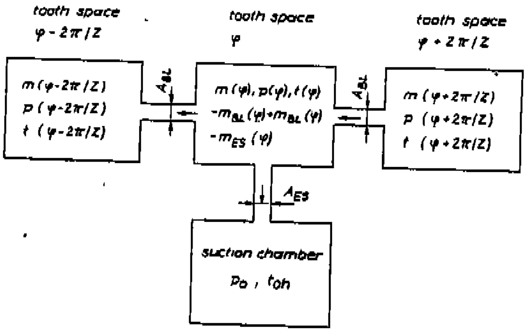

Fig. 2 - Flow model vallid for gap flow at a screw compressor

The calculation of the compression process including the discharge phase is solved by taking small rotary steps $\Delta\varphi$, i.e. a certain number of action angles for each tooth into account. The calculation starts when the suction process is finished, the tooth gap volume had reached its maximum and begins to decrease. For each angle of action the balance of mass for the tooth gap φ is established (fig. 2). From the foregoing tooth gap $\varphi+2\pi/z$ the mass of gas $+m_{Bl}$ is flowing back through the blow and periferal hole resp. into the tooth gap considered. The mass of gass $-m_{Bl}$ is leaving the tooth gap φ and flows into the tooth gap $\varphi-2\pi/z$ that is following. The quantity $-m_{ES}$ is leaving the tooth gap φ as well through the gearing path and flows directly into the suction chamber.

All types of gap flow are considered to take place under real gas conditions and are calculated by using the theory of gas dynamics.

The quantity of mass which flows during a time interval τ expressed by the rotary angle of action $\Delta\varphi$ through the gap having a gap area A is determined by the following equation

$$m_{\Delta\varphi} = A\cdot\tau\sqrt{2(h(\varphi) - h(\varphi \pm 2\pi/z))}/v$$

The enthalpy h is calculated according to the thermodynamic condition of the tooth gap

$$h = h(t, v)$$

In order to calculate the thermodynamic and caloric properties functions can be made use of taking real gas conditions into account which are prepared as subroutines that can be linked into the computer program.

The actual quantity of mass for a rotary angle of action is determined by means of

$$m(\varphi) = m(\varphi-\Delta\varphi) + m_{Bl}(\varphi) - m_{Bl}(\varphi) - m_{ES}(\varphi)$$

The quantity of mass and the tooth gap volume can be used to calculate the specific volume and the thermodynamic properties like pressure and temperature.

Resulting from the calculation of the thermodynamic process without supercharge we obtain

- the lapse of pressure
- the lapse of temperature
- the alteration of mass.

By using the mass flow rate - reduced by the sum of gap losses -

leaving the tooth gap, the refrigerating capacity can be determined.

$$Q_o = \dot{m} \cdot \Delta h$$

When the lapse of pressure is integrated the effective work may be calculated.

The computer-assisted solution of the simple thermodynamic process is described in /1/.

2. TAKING SUPER CHARGE INTO CONSIDERATION

Fig. 3 shows the refrigerating cycle of a screw compressor with supercharge and the appropriate lg p,h-diagram.

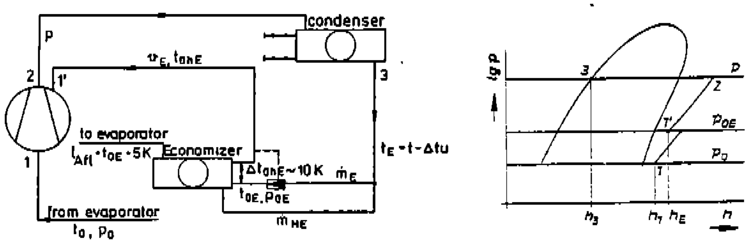

Fig. 3 - Refrigerating cycle of a screw compressor with supercharge and the appropriate lg p,h-diagram

The main mass flow rate $\dot{m}_{HE}$ which comes from the condenser is cooled at the economiser from the temperature t_E to that of t_{Afl}. This will result in a higher refrigerating capacity achieved at the evaporator.

The subcooling of the main mass flow rate results from the evaporator of the supercharge mass flow rate $\dot{m}_E$ that apears at the economiser.

The improvement of the refrigerating capacity resulting from the process of supercharge is all the better the lower the evaporating temperature of the process of supercharge is, and the greater the ratio of the supercharge mass flow rate divided by the main mass flow rate resp. becomes.

$$\frac{Q_{o\lambda}}{Q_o} \sim \frac{\dot{m}_E}{\dot{m}_{HE}}$$

Compared with /2/ all the calculations and experimental results obtained were carried out for a supercharge opening which allows to achieve a maximum effect by the supercharge (the opening is situated at the transport phase of the female rotor /3/.

From a higher supercharge capacity, i.e. from a higher supercharge mass flow rate an additional growth of the driving power will result. In order to deside upon an effective use of an economiser it would be desirable to estimate in advance the yields in the effectifity of the supercharge process.

The greater the difference in pressure between the evaporating pressure at the economiser p_{oE} and the suction pressure p_o at the screw compressor becomes, all the better the ability of the screw compressor will be to process the evaporating supercharge mass flow rate of the refrigerant.

Based on experimental studies the following correlations were obtained:
The supercharge volume flow rate $\dot{V}_E$ is a function of the difference in evaporating pressure within the economiser and the suction pressure

of the compressor expressed by means of the saturation temperature

$$\dot{V}_E = F_E \, \dot{V}_{th}$$

$$F_E = f\,(\Delta t), \quad \Delta t = t_{oE} - t_o$$

The reduction in the volumetric efficiency by means of the supercharge may as well be expressed as a function of the difference in temperature Δt

$$\lambda = f(\Delta t)$$

The difference in temperature Δt is itself a function of the difference in pressure between the condenser pressure and the evaporator pressure

$$\Delta p = p - p_o$$

at the compressor and the evaporator resp.

$$\Delta t = f(\Delta p,\ t_o)$$

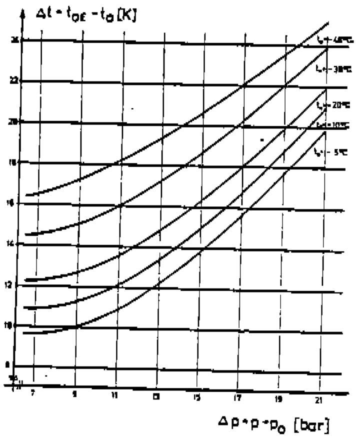

Fig. 4 - Difference in temperature Δt versus Δp and t_o

From Fig. 4 we learn that the difference in temperature is increasing with the growing pressure difference of the system and with the decrease in evaporating temperature.

The influence of the temperature difference Δt on the factor influencing the supercharge mass flow rate is demonstrated in Fig. 5.

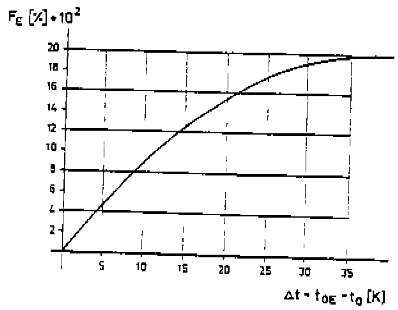

Fig. 5 - Factor influencing the mass flow rate of supercharge, $F_E = F(\Delta t)$

$$F_E = -1.369 \cdot 10^{-3} \Delta t^2 + 1.055 \cdot 10^{-2} \Delta t$$

$$\lambda = 1.00286 - 0.005429 \cdot \Delta t$$

These data make it possible to calculate the supercharge performance.

The mass flow rate $\dot{m}$ results from the calculation carried out without taking supercharge into account. Taking reduction of the volumetric efficiency into consideration we get

$$\dot{m}_{HE} = \dot{m} \cdot \lambda$$

the main mass flow rate under supercharge conditions.

The supercharge performance is the refrigerating capacity resulting from the evaporation of the supercharge mass flow rate within the economiser.

$$\dot{Q}_{oE} = \dot{m}_E (h_E - h_3)$$

$$h_E = h(T_{ohE}, v_E)$$

$$t_{ohE} = \Delta t + t_o + \Delta t_{ohE}$$

The total refrigerating capacity under supercharge conditions results from

$$\dot{Q}_{o\lambda} = \dot{m}_{HE}(h_1 - h_3) + \dot{Q}_{oE}$$

The increase in the driving power under supercharge can approximately be calculated by taking the supercharge mass flow rate into consideration

$$P_E = (\dot{V} + \dot{V}_E)\ P/\dot{V}$$

Figures 6 and 7 show the factors increasing refrigerating capacity and COP by means of supercharge.

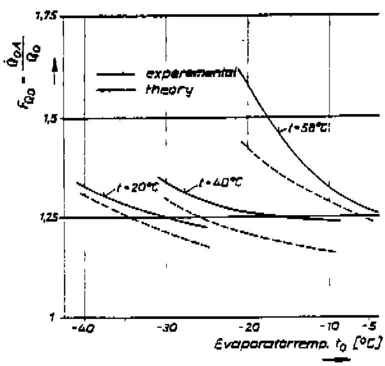

Fig. 6 - Factor $F\ Q_{o\lambda}$ increasing refrigerating capacity

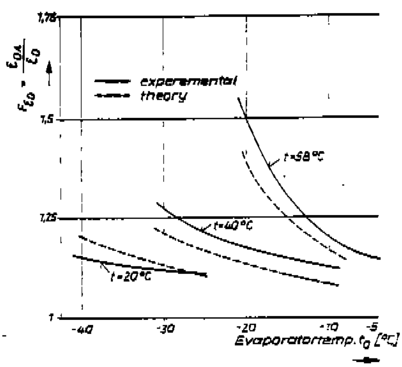

Fig. 7 - Factor $F\ \varepsilon_{o\lambda}$ increasing COP

The supercharge process results in an increase of the gap pressure that leads to increased forces in bearings at idle motion.

By means of a simulation program /1/ and by using the supercharge pressure using equation $p_E = p(t_{oE})$ the lapse in pressure and the increased bearing forces may be calculated.

Fig. 8 shows with respect to the male rotor the influence of the supercharge on the bearing forces.

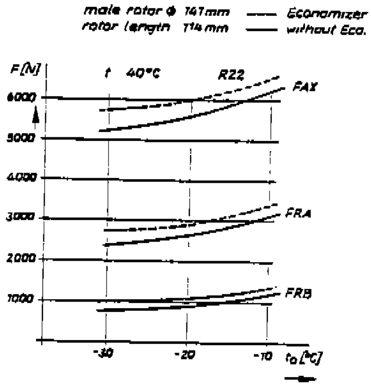

Fig. 8 - Bearing forces of the male rotor with and without supercharge
FAX - axial force
FRA - radial force of the bearing situated on the pressure side
FRB - radial force of the bearing situated on the suction side

3. CONCLUSION

The process of the thermodynamic calculation of a screw compressor has been improved by the determination of the refrigerating capacity resulting from supercharge and the driving power taking generalised experimental results into consideration. This method has made it possible to determine in advance for any individual refrigerating screw compressor and any working condition the refrigerating capacity resulting from supercharge, and the driving power that may occur within the limitations shown in Fig. 4.

The deviations that may occur between calculation and measurements will be in the order of 5 to 10 per cent.

The process of supercharge during compression results in an increase of the pressure gap during compression. This again leads to higher bearing forces. According to Fig. 8 the bearing forces increase in the order of 5 to 15 per cent.

Nomenclature

A	Gap area
$\dot{m}$	Mass flow rate without supercharge
$\dot{m}_{HE}$	Main mass flow rate taking supercharge into account
$\dot{m}_E$	Supercharge mass flow rate
m_{Bl}	Loss resulting from blowing hole
m_{ES}	Loss through gearing blowing hole
$\dot{V}_E$	Supercharge volume flow rate
$\dot{V}_{th}$	Theoretical value of the swept volume
F_E	Factor expressing supercharge volume flow rate
λ	Reduction of the volumetric efficiency resulting from supercharge
φ	Motion angle of the rotor
p	Condenser pressure
p_o	Suction pressure
p_E	Supercharge pressure
t_o	Evaporating temperature
t_{oE}	Evaporating temperature within the economiser
t	Condensing temperature
t_{ohE}	Super heat temperature at the economiser
P	Driving power without using an economiser
P_E	Driving power taking the economiser process into account
Q_{oA}	Refrigerating capacity with supercharge
Q_{oE}	Refrigerating capacity at the economiser

REFERENCES

1. A. PIETSCH. Einfluß der Spaltverluste auf Förderstrom und Antriebsleistung beim Schraubenverdichter (Influence of the gap losses on the swept volume and on the driving power of a screw compressor. LKT (1982)2.
2. D. XIAO, Y. YU. A new mathematical model of screw refrigeration system with an economiser. XVIIth International Congress of Refrigeration, Vienna 1987.
3. L. KINNE. Untersuchung an Kältemittelschraubenverdichtern zur Aufladung, Economiserbetrieb (Investigation on a refrigerating screw compressor working with supercharge and economiser). LKT (1987)2.