Improvements on Refrigerating Screw Compressors of Large Capacity

D. Mosemann
VEB Kuhlautomat Berlin

S. Krienki
VEB Kuhlautomat Berlin

S. Nowotny
VEB Kombinat ILKA Luft- und Kaltetechnik

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D. MOSEMANN, S. KRIENKE, VEB Kühlautomat Berlin (GDR)
S. NOWOTNY, VEB Kombinat ILKA Luft- und Kältetechnik, SFT, Dresden
(German Democratic Republic)

1. INTRODUCTION

Large screw compressors which are to be used in refrigeration installations especially on board of fishing vessels as well as in stationary refrigerating equipment are becoming increasingly important.

Different details in design which show improvements to known solutions are demonstrated. The two-stage cycle of a screw compressor carried out in a single machine, the use of an economizer cycle, the protection against gas pulsation at high pressure ratios are explained.

A possibility to control the internal volume ratio is discussed which enables energy conversion to be improved at varying ambient conditions.

Finally the restrictions will be discussed which result from the lubrication conditions of a refrigerating screw compressor working under heat pump condition.

2. TECHNICAL DESCRIPTION

2.1 DETAILS OF THE DESIGN OF SCREW COMPRESSORS

In the GDR we have gained experiences for more than 15 years with respect to development, manufacture, and long term operation of refrigerating screw compressors having a theoretical volume flow rate ranging from 300 to 2500 m3/h.

Screw compressors are superior to conventional reciprocating compressors regarding life time, servicing, reliability and requirements resulting from the local conditions. Screw compressors have proved their advantages especially on board of ships where these parameters are needed in combination with a service-free running.

The small screw compressors up to a rotor diameter of 153 mm are both axially and radially embedded in ball bearings whereas the larger rotors are radially embedded in slide box bearings and axially in ball bearings. With the exception of the compressor having a volume flow rate of 450 m3/h the main rotor is driven. See Fig. 1.

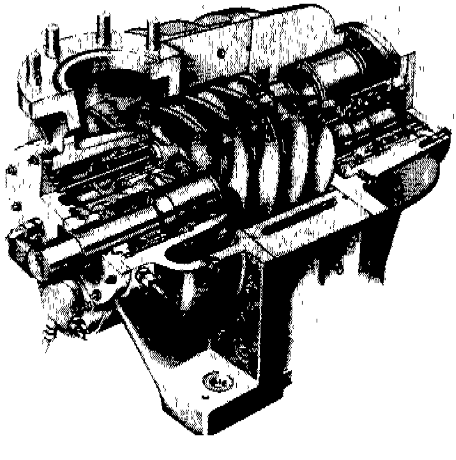


Fig. 1
Cross-sectional view of an large open-type refrigerating screw compressor

A pressure-assisted balancing disk reduces the axial forces of the main rotor which is driven by an electric motor. When using the propulsion of the lateral rotor, the gas forces influencing the main rotor are compensated by means of the gearing of teeth, and the axial reactive force will affect the lateral rotor and it has to be compensated by means of a pressure-assisted balancing disk. Below the rotor pair a control gear is installed in axial position which by means of hydraulic forces infinitely controls the refrigerant flow rate between 100 and approximately 0 to 15 per cent. The appropriate signal box which belongs to the control gear is positioned within the suction chamber resulting in a compact design. The position of the control slide is shown by means of a hermetically sealed position indicator which is used for part load control as well.

2.2 MEASURES TO IMPROVE ENERGY EFFICIENCY

An important measure to improve the energy efficiency was the optimization of the discharge port which resulted in improvements of the part load conditions and a reduction in the power input rating of 60 per cent at 50 per cent flow rate.

In order to increase the refrigeration capacity and to improve the coefficient of performance (COP) each screw compressor has an additional suction inlet, which is connected with the working chamber when the suction process has finished. This supercharge in combination with an economizer makes possible a two-stage working process by which the COP of the screw compressor is increased in the order of 30 to 40 per cent /3,4/. See Fig. 2

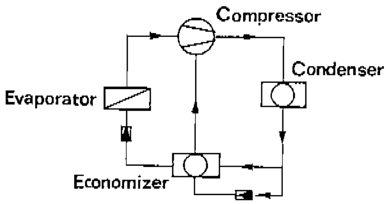


Fig. 2
Economizer cycle

By injecting liquid refrigerant into the working chamber the screw compressor can work without or with a reduced oil cooling facility depending on its application.

The screw compressors are equipped with a novel system /5/ which protects them against pressure pulsation when working at high pressure ratios. This system operates in such a manner that a channel which connects the discharge chamber and the working chamber reduces the free energy which initiates the gas pulsation at part load conditions near the zero flow rate so that resonance pulses will not be stimulated. This process is controlled through the control gear in the upper region of part load.

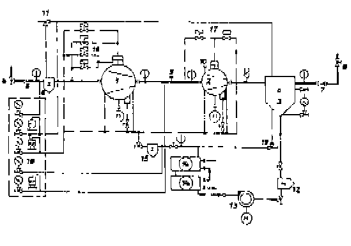


Fig. 3
1- low pressure, 2- high pressure compressor, 3- oil separator, 4- suction shut-off valve, 5,7- back pressure valve, 6,12,15- filter, 8- shut-off valve, 9- injection of liquid refrigerant, 10- additional suction inlet, 11- oil return, 13- oil pump, 14- oil cooler, 16,17- hydraulic controls, 18- safety appliances

We have gained special experiences with screw compressors working at two-stage conditions in an apparatus /6,7/. See Fig. 3. In this apparatus the discharge side of the low pressure compressor is directly connected with the suction side of the high pressure compressor. A common oil separator and a common oil circuit are installed together with the whole equipment on a joint frame, and are factory assembled and tested. Such a unit can work at evaporating temperatures of -55 °C and condensing temperatures of 36 °C using R22 as refrigerant and cooling water of 30 °C.

The quantity of 100 ppm oil which is to be found in the refrigerant flow crossing the two-stage screw compressor unit is compensated by means of a special devise which makes the refrigerant oil free. See the simplified cycle in Fig. 4. With a special facility 10 per cent of the refrigerant flow rate of the high pressure line are completely evaporated in a heat exchanger for oil return and are then again conducted to the supercharge inlet of the low pressure compressor. The heat needed to evaporize the part flow rate is then used for cooling the condensate before being expanded, which enables this cycle to increase the refrigerating capacity in the order of 10 per cent without an increase in power input rating. It should be mentioned that this cycle cannot be implemented when applying a conventional piston compressor.

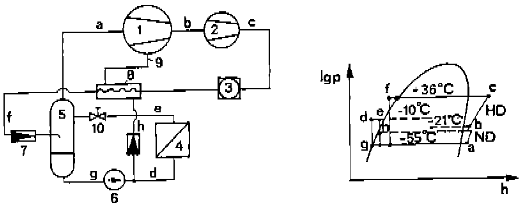


Fig. 4 - Schematic view of the cycle for oil return

When using oil return this installation can use oil having a low solubility within the refrigerant at low evaporating temperatures (concentration of oil within the refrigerant at 0.1 per cent).

2.3 SAFETY APPLIANCES

The compressure unit is equipped with a back pressure valve and safety appliances which protect it against

- -short circuit of the electric motor
- -low suction pressure
- -high discharge pressure
- -high oil temperature
- -superheat
- -high pressure loss in the oil filter
- -low oil pressure or missing oil flow on the flow control sensor

and enable the unit to work around the clock without supervision, infinitely working part load control as well as unloaded start and stop.

2.4 FURTHER POSSIBILITIES TO IMPROVE ENERGY EFFICIENCY

As a result of R and D there has been a general change in the number of teeth on the main and lateral rotor from 4/6 to 5/6 resp. simultaneously a controllable internal volume ratio has been built into the screw compressor which allows the ratio to be infinitely varied from Vi = 2 to 5. This ratio control is combined with part load control which makes possible an infinite variation of the internal pressure ratio at full load as well as the regular part load control. See Fig. 5. Two different solutions have been studied:

1. Vi-control by means of variation of the radial discharge outlet opening ratio. See Fig. 6.

In this case the fixed stroke, the so-called "lift limiter", is shif-

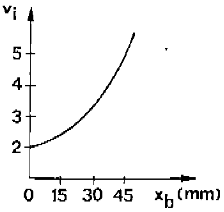


Fig. 5
Controllable internal volume ratio

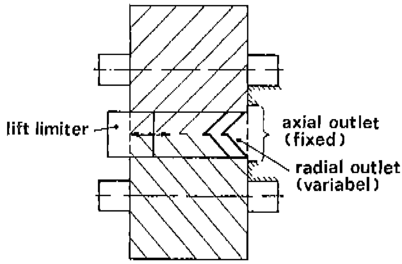


Fig. 6
Vi-control by means of the radial discharge outlet opening ratio

ted at full load position so as to change the internal compression ratio at full load only. The variation of the Vi-value or the part load control is carried out in such a manner that the spring-assisted lift limiter is locked (part load control) or unlocked (Vi-variation). In the first case the control slide is shifted alone whereas in the second case the control slide and lift limiter are shifted together. See Fig. 7.

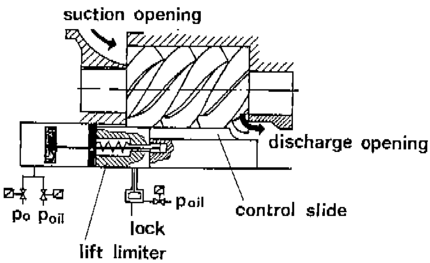


Fig. 7
Schematic view of the Vi-control using hydraulic actuation

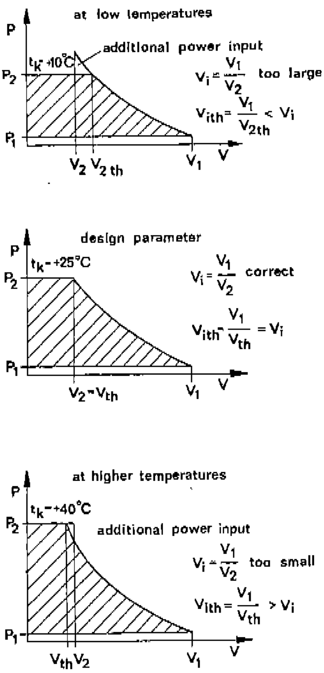
2. Vi is controlled by means of variation of the axial discharge outlet opening ratio.

In this case the front wall on the discharge consists of circular disks. By making these control disks of fixed and movable segments the control edge which fixes the beginning of the outlet can be changed by rotating the movable segments. This control strategy enables the Vi-value to be varied both at full and part load.

Vi-control does away with an important drawback of the screw compressor which normally appears when the compressor cannot be operated under conditions it was originally designed for. See Fig. 8. By completely adapting the screw compressor to the ambient conditions it can always be operated with highest efficiency. When operated on board of ships under varying climatic conditions the ship owner will save 20 per cent of the energy costs when using Vi-control. See Fig. 9.

2.5 SAFEGUARDING THE LUBRICATION

In order to safeguard the lubrication of large screw compressors which use slide box bearings a certain minimum of viscosity has to be guaranteed so as to the hydrodynamic lubrication conditions in the bearings. The upper limit of viscosity needed for the lubrication of the bearings may be in the range of 7 to 12 cST depending on how the operating limits, the bearings geometry and the number of revolutions were fixed. When the compressor is operated with refrigerant R12 at a condensing temperature of 77 °C, which corresponds to a pressure of 2.07 MPa, protection against evaporation of the refrigerant from the oil must be ensured by means of a pressure rise using an oil pump and



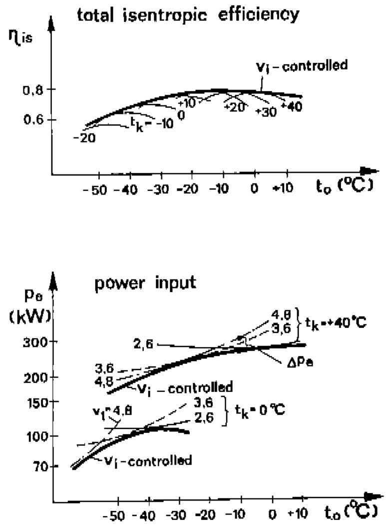


Fig. 9
Saving of energy resulting from the control of the internal volume ratio Vi

Fig. 8 - Functioning of the Vi-control when the screw compressor works under conditions it was originally not designed for

an additional oil cooler. For this oils with a generally high level of viscosity are necessary. As a result the gap within the slide box bearings of the main rotor which revolves at a higher speed must be carried out in the order of two pro mill to ensure higher hydraulic friction. As a result of the increased oil flow rate the increase in temperature will be reduced. The slide box bearing of the lateral rotor therefore has to be carried out with a relative gap of one pro mill in order to guarantee the capability of bearing.

Another possibility to solve the problem would be to control the equilibrium of the oil-refrigerant-mixture at an intermediate pressure level close to the suction pressure in order to ensure a sufficient oil viscosity within the bearings. This solution, however, requires additional equipment.

3. CONCLUSIONS

The improvements on screw compressor design as shown above have proved that there are further possibilities which should be taken into consideration. Improvements in tooth geometry and tooth shape accurscy will result in higher energy efficiency.

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AMELIORATIONS APPORTEES AUX COMPRESSEURS FRIGORIFIQUES A VIS DE GRANDE PUISSANCE

RESUME: Les grands compresseurs à vis destinés à l'utilisation dans des installations frigorifiques, notamment pour les bâtaux de pêche mais aussi pour les installations frigorifiques terrestres, s'imposent de plus en plus.

Les auters montrent différents détails de construction qui représentent des améliorations par rapport aux solutions connues jusqu'ici. On explique le fonctionnement bi-étagé de compresseurs à vis dans un agrégat, le service économiseur, la protection contre les oscillations du gaz dans les cas de taux de compression élevés.

On discute la solution pour le réglage du rapport du débit volume en vue d'améliorer l'economie d'énergie dans des conditions de fontionnement variables.

Finalement, on montre les limites d'utilisation des compresseurs frigorifiques à vis en fonctionnement de pompe à chaleur du point de vue de la tribologie.