Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1986

# Use of Rotary Blade Compressor and Expander in the Refrigeration Cycles

Z. Gnutek

E. Kalinowski

# USE OF ROTARY BLADE COMPRESSOR AND EXPANDER IN THE REFRIGERATION CYCLES

Zbigniew Gnutek and Eugeniusz Kalinowski

Institute of Power Engineering and Fluid Mechanics
Technical University of Wrocław. 27 Wybrzeże Wyspiańskiego str. 50-370 Wrocław. Poland

## ABSTRACT

A special refrigerating system is built in laboratory of the Institute. In this system a rotary blade compressor and a rotary blade expander are used as elements for lowering the gas temperature. The compressor is a two stage rotary machine with capacity $\dot{m}$ = 150 kg/h and pressure 0,7 MPa. The expander gives at this pressure a possibility of arriving at - 100°C in the cooling room. The diagram of closed refrigerating system with the use of the rotary compressor and expander is presented in Figure 1. The advantage of this system is its small size, several times smaller than in the system which uses a piston compressor and expander of the same capacity. The first experiments gave the temperature 175 K in the cooling room and further inwestigations are made.

## INTRODUCTION

This paper presents a trial to obtain a gas refrigeration system arriving at the temperature of 150 K in a cooling room, with small sizes and not too high pressure. For these purposes the rotary blade compressor and expander are chosen. The expander was tested as the first one. The results gave the possibility of building the closed refrigerating system presented in Figure 1.

## CLOSED REFRIGERATION SYSTEM WITH BLADE ROTARY COMPRESSOR AND EXPANDER

Fig.1 presents a scheme of the closed refrigerating system. The inert gas was chosen as a refrigerant. The ambient air was used in the experiment. The air with 0,1 MPa pressure is sucked into the compressor and pressed

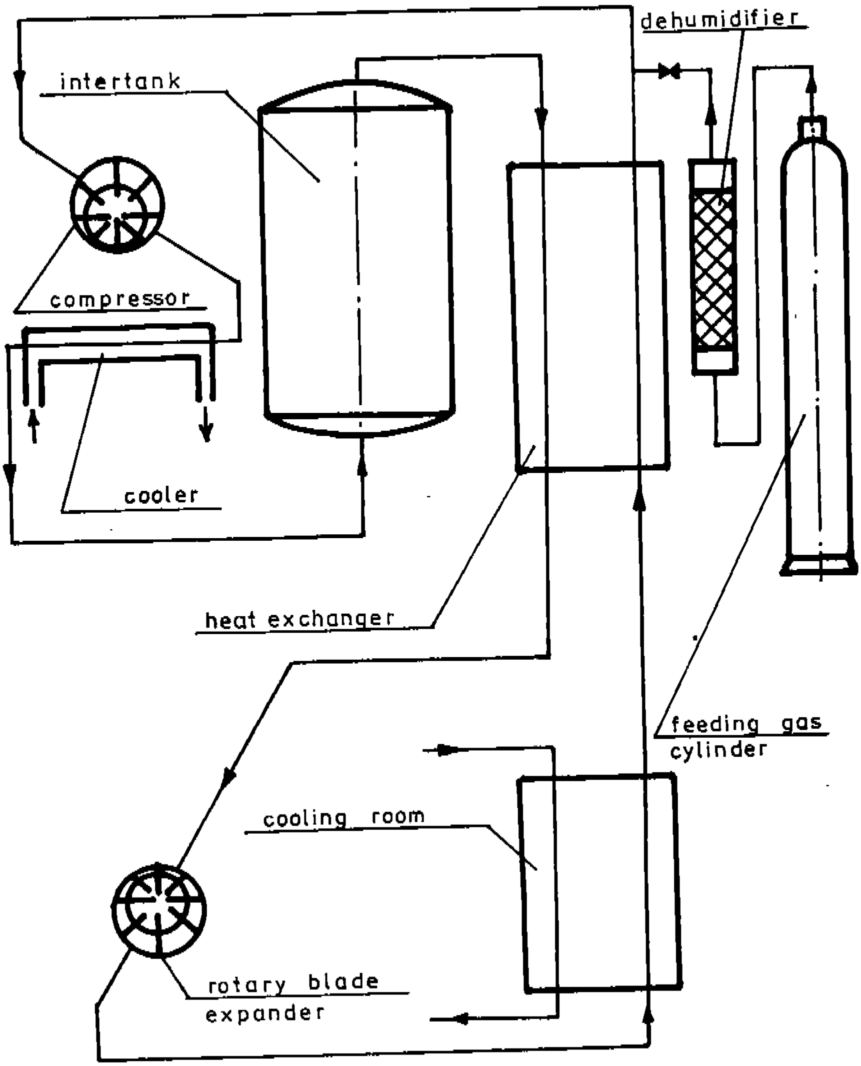


Fig. 1. Scheme of closed refrigerating system with rotary blade compressor and expander.

in it to 0,7 MPa. Then it flows through the water cooler into the gas container goes into the heat exchanger in which it is cooled and then it flows into the expander. In the expander both pressure and temperature are lowered. The cold refrigerant receives heat from the cooling

room and the heat exchanger. Then it's temperature arrives approximately to the ambient temperature and is sucked by the compressor.
Fig. 2 shows a photo of the cooling set and Fig. 3-a photo of the high pressure part of the refrigerating system.

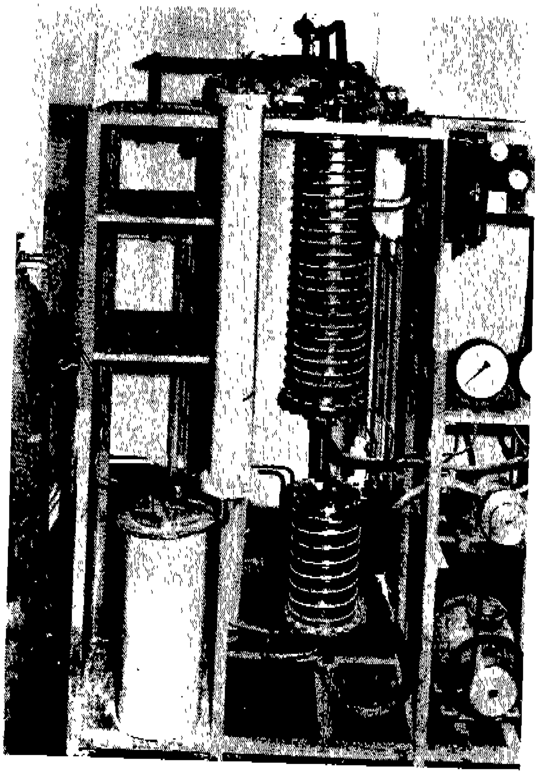

Fig. 2. The cooling set.

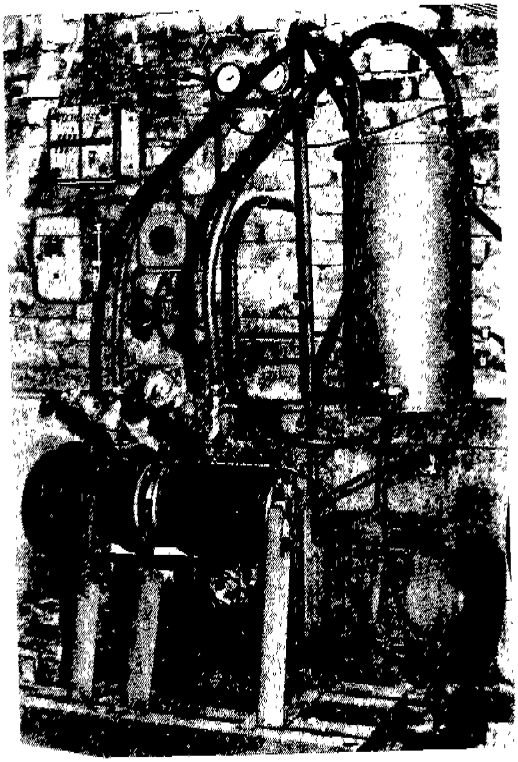

Fig.3. The high pressure part of refrigerating system.

EXPANDER

Scheme of the rotary blade expander, used in the refrigerating system is presented in Fig.4.

The work chamber of the expander is the space between a cylinder and a rotor limited by two succesing blades and two head surfaces. In the time of rotation gas volume is changed. At the inlet of the expander the gas parameters are $p_1$ and $T_1$, and after the expansion and performing of work $L_r$, they obtain the values of $p_2$, $T_2$ and $v_2$. Temperature drop $\Delta T = T_1 - T_2$ of the gas depends on the expander design. Analysis of processes in the used expander gives the possibility of defining the following analytic equation(1).

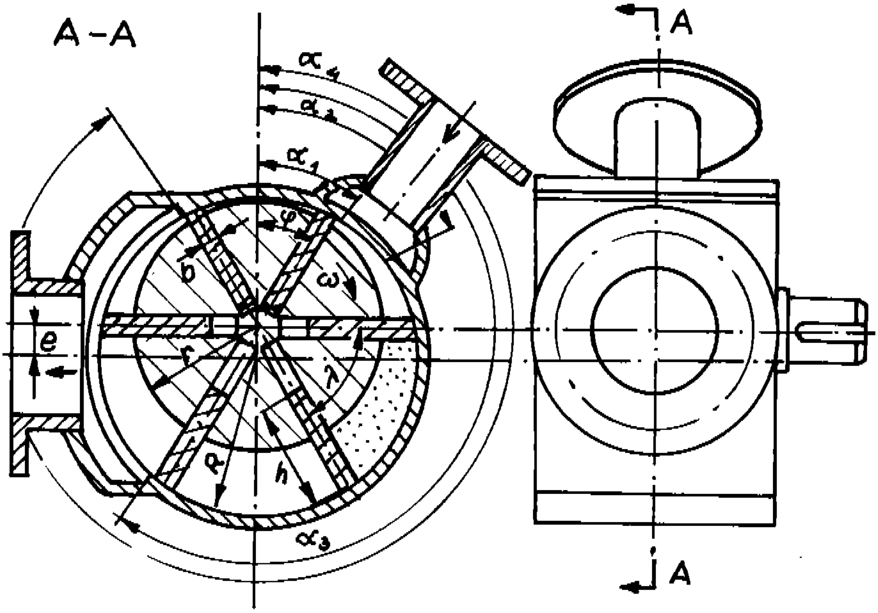


Fig.4. Scheme of rotary blade expander

$$T_1-T_2 = T_1 - \frac{1}{2}\left\{\left[1+(X_1-X_2)\right]T_k + \left[1-(X_1-X_2)\right]T_1\right\} +$$

$$+\ B \int_{\alpha_3-\lambda}^{2\pi+\alpha_2} F_1(\varphi)\,d\varphi \qquad (1)$$

where:

$T_1$ - refrigerant's temperature at the inlet of the expander

$T_k$ - refrigerant's temperature in the chamber at the end of expansion

$$F_1 = \frac{k_1+k_2\sin^2\varphi - k_3\cos\varphi}{1-k_4\sin^2\varphi - \mu_c k_5 \sin\varphi}$$ function of the angle

describing the chamber position in cylinder and constants $k_1 - k_5$ ; The constants depend on the designed parameters of the expander

$$B = \frac{4\pi^2 n^3 z\, R^4 L \left(\frac{b}{R}\right)\left(\frac{h}{R}\right)\varrho \left(1+\frac{2}{\pi}\ \frac{e}{R}\right)\left(\mu_{cu}+ |g|\ T_m\right)}{\dot{m}\ c_p}$$

is a constant dependend on the designed parameters and the total flow rate

$X_1$ - ratio of waste flow rate which isn't converted to work, to the total flow rate

$X_2$ - ratio of waste flow rate which is converted to work in the chamber, to the total flow rate

## COMPRESSOR

In the closed refrigerating system two stage rotary blade compressor is used. Scheme of the high pressure part of the refrigerating system is presented in Fig.5.

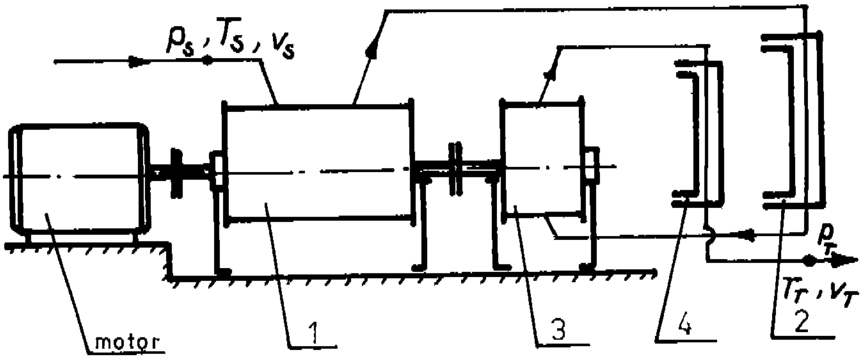


Fig. 5. Scheme of high pressure part of the refrigerating system.

The sizes of the cross-sections of both stages are the same. They differr only in the length of the chamber. The scheme of cross-section of the compressor is presented in Fig.6.

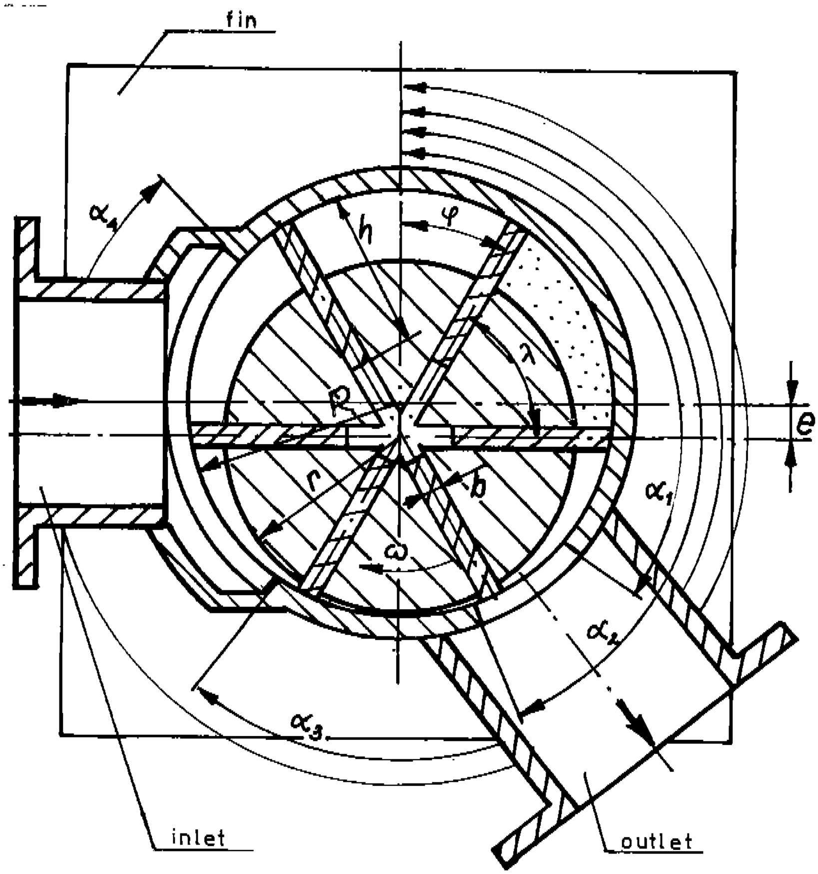


Fig.6. The cross-section of the compressor.

The compressor sucks the gas flux with $p_s$, $T_s$ parameters, measured at the inlet, and presses it to $p_I$, $T_I$.

At outlet from stage I there is an intercooler 2 (look at Fig.6.) where water isobarically lowers the temperature aproximately to the ambient temperature. From the intercooler the gas goes into the second stage 3. Gas parameters at the outlet from the second stage are $p_T$ and $T_T$. From that place gas goes through the final cooler 4 into the intertank.

For given compressor and expander the flow rate $\dot{m}$ of the gas measured on the suction pipe is described by an analytical function

$$\dot{m} = \frac{z\ n\ R^2\ L\ p_S}{\bar{R}\ \ T_S}\ K\left(2\pi - \frac{\lambda}{2}\right)\left[1 - \frac{K\left(\pi - \frac{\lambda}{2}\right)}{K\left(\alpha_2 - \lambda\right)}\right] \quad (2)$$

where:

$z$ - number of blades

$n$ - amount the revolutions

$R, L$ - radius and length of chamber

$p_S, T_S, \bar{R}$ - gas parameters at the inlet

$K(\varphi)$ - quantity proportional to the chamber volume, dependend on angle $\varphi$.

For the described refrigerator, the flow rate is $\dot{m}$ = 150 kg/h. Refrigerant is compressed to the pressure $p_T$ at the outlet which can be determined as follows

$$p_T = \frac{\dot{m}\ \bar{R}\ \ T(\varphi)}{V(\varphi)} - p_{CH} \quad (3)$$

where:

$T(\varphi)$ - temperature at the and of the compression in the second stage of the compressor

$V(\varphi)$ - volume at compression end

$p_{CH}$ - hydraulic resistance of final cooler.

Temperature at outlet of the refrigerant $T_T$ is aproximately equael to the ambient temperature.

## OTHER DATA AND CONCLUSION

The heat exchanger was built from pipes rolled on the cylinder.

In the experiments the expander was braked by the electrical generator loaded with light. It is planned to couple the expander with the compressor in the further investigations for lowering the power consumption.

Gas cooled in the expander goes into the cooling-room, where the temperature $T = 175$ K was obtained.

The further experiments are carried out.

## REFERENCES


1) Z.Gnutek. Mathematical model of rotary expander with movable rotor blades. XV International Congress of Refrigeration, Venezia 1979.

2) Z.Gnutek, E.Kalinowski, B.Malik. Low Pressure Condensing Unit for Gases of Group A with Use of Rotary Blade Expander. Report I.T.C. i M.P. Series SPR 55/84. Technical University of Wrocław. 1984(in polish)