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Temperature Boundary Effect of the Large Heat Pump Cycle on the Characteristics of the Turbocompressor

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1. INTRODUCTION

The application of one-stage turbocompressors in the heat pumps has constructive, technological and operation-energy justification. This paper is a contribution to the study of the characteristics of the one-stage turbocompressors, designed for the utilization of the waste heat gained from the industrial water cooling systems. The temperature level of the waste heat flux is of the grade \( t^1 = 30 - 40 \degree C \). The thermal transformation of this heat in the heat pump on the grade \( t^1 = (60 - 90) \degree C \) makes it usable in the heating systems or in the technological processes.

The main attention is paid on the temperature boundary effect of the heat pump cycle \( (t^1 \) and \( t^k) \) on the pressure ratio \( \Pi \), conditional Mach number \( \mathcal{M} = \frac{u_2}{a_1} \)

and through \( \mathcal{M} \), on the characteristics of the one-stage turbocompressors.

The main characteristic of these compressors is work in conditions with high Mach numbers. Optimal design of the flow space and the regulating system of the turbocompressor is possible if the modelling of the expected characteristics of the turbocompressor is done, and the work in conditions with high Mach number is taken into consideration.

The basis for optimal design of the flow space and the regulating system of the turbocompressor from the aspect of maximal economy during the exploitation is the modelling of the heat pump operation with joining the characteristics of its essential elements (turbocompressor, condenser, evaporator).

2. COMPARATIVE ANALYSIS OF THE REFREGERANTS

Besides the essential demands for the characteristics of the refrigerant, at an optimal choice of the refrigerant in the turbocompressor heat pumps, some additional criteria are imposed.

At given temperature boundaries of the heat pump cycle \( (t_0 \) and \( t_k) \) the needed pressure ratio \( \Pi \) is the function of the refrigerant characteristics. The determined dependence of \( \Pi \) from the normal evaporation temperature \( t_{on} \) and from \( t_0 \) and \( t_k \) (Fig.1), shows that the refrigerants with lower \( t_{on} \) are more satisfactory for the application in the turbocompressor heat pumps.

![Fig.1. Dependence of \( \Pi \) on the refrigerant properties](image-url)
Fig. 2 shows the boundary peripheric velocity $u_2$ and the boundary pressure ratio $\Pi$ connected with the stress properties of impeller and gas-dynamic conditions in the flow space (Mach number), for different refrigerants. The data from Fig. 2 refer to a particular type of impeller (the coefficient of pressure ratio $\psi = 0.5$, and polytropic coefficient of performance $\eta_p = 0.85$) and for particular conditions ($t_0 = 30^\circ C$). Freons are characterised with large molecular mass and for them, in most cases, Mach number is a limiting criterion. In the one-stage freon turbocompressors high Mach numbers are achieved at relatively small speed.

![Graph showing dependence of $u_2$, $M_0$, $\Pi$, and $h$ on the refrigerant](image)

**Fig. 2.** Dependence of $u_2$, $M_0$, $\Pi$, and $h$ on the refrigerant

The influence of the heat pump capacity upon the dimensions of the turbocompressor can be seen through the dependence of the impeller diameter $D_2$ upon the heat capacity of the evaporator $Q_0$, $M_0$, and the kind of the refrigerant, Fig. 3.

![Graph showing dependence of $D_2$ upon $Q_0$, $M_0$, and the refrigerant](image)

**Fig. 3.** Dependence of $D_2$ upon $Q_0$, $M_0$, and the refrigerant
3. MODELLING OF WORK CHARACTERISTICS OF THE ONE-STAGE TURBOCOMPRESSORS DESIGNED FOR HEAT PUMPS

The operation of the turbocompressor in conditions with high $M_2$ leads to the sheer characteristics of the turbocompressor caused by the influence of the high pressure ratio and the critical phenomena in the flow space of the turbocompressor.

3.1. The influence of the high pressure ratio on the turbocompressor

The influence of the high pressure ratio on the work characteristics of the turbocompressor results from the intensive change of the specific volume at the outlet of the impeller with the change of work condition. The geometry of the impeller and the losses at the flow space determine the dependence of $\Pi$ upon the coefficient of flow $\xi_2$ [{eq}\Pi=f(\xi_2)\], Fig.5c.

$$\Pi=[1+(\beta_{tr}+\beta_p)(k_1-k_2\phi_2\text{ctg}\beta_2(\xi_2-1))\phi_2^2(\xi_2)/(\xi_2-1)]\eta_p$$ (1)

If we exclude the critical phenomena from the analysis and suppose that the compressor has the same dimensionless work characteristic $\Pi=f(\xi_2)$, Fig.5a, for two different $M_2$, then the transformation of this dependence and the dependence $\Pi=f(\xi_2)$, Fig.5c, in functional dependence $\Pi=f(\xi)$, Fig.5b, and $\Pi=f(\bar{M})$, Fig.5d, that in fact represents work characteristic of the turbocompressor, is done by the following relations:

$$\psi_2 = \frac{b_2}{b_1} \frac{D_2}{D_1} \frac{\tau_2}{\tau_1} \frac{k_2}{k_1} \phi_2$$ (2)

$$k_{v2} = \frac{v_0'}{v_2} = \left[1+\phi(\xi_2-1)(k_1-k_2\phi_2\text{ctg}\beta_2(1+\beta_{tr}+\beta_p\eta_p)\phi_2^2(\xi_2)/(\xi_2-1)]\eta_p^{-1}ight]^{-1}$$ (3)

$$k_{v1} = \frac{v_0'}{v_1} = 1 - \frac{1}{2} \frac{\phi_2}{\phi_2}$$ (4)

Fig.4. Dependences between the dimensionless and work characteristics of the turbocompressor.
From the equations 1, 2, 3, 4 and from Fig 5, it can be seen that at the high pressure ratio turbocompressors (high $M_w$), the change of the coefficient of the flow $\Phi_1$ does not follow the change of $\Phi_2$, due to the influence of the fluid compressibility $\kappa$ which is noticeable at the high pressure ratio freon turbocompressors. In the equations (1) - (4):

- $\Phi_1$ and $\Phi_2$ - the coefficients of the dimensionless criterion of the fluid head $\Phi$, which depend upon the geometric and constructive characteristics of the impeller and upon the work condition
- $F_u$ - conditional adiabatic exponent
- $\eta_p$ - polytropic coefficient of performance
- $b_2, b_3, a_2, a_3$ - width, that is, the outlet diameter, that one in the inlet of the cascade of the impeller
- $\tau_2, \tau_1$ - the reduction coefficient of outlet or inlet crosssection at the cascade of the impeller
- $\rho$ - degree of reactivity
- $M_0 = c_0/a_0$ - Mach number of the gas flow at the inlet of the impeller (0).

3.2. The influence of the critical phenomena in the flow space of the turbocompressor

The increase of $M_u$ always leads to the increase of Mach numbers in the gas flow $M_1$ and $M_2$. The value of these Mach numbers, associated with their critical phenomena, depend upon the geometric and constructive characteristics in the flow space of the turbocompressor, and influence the gasdynamic losses, and the work characteristics of the turbocompressor.

The dependence of the gasdynamic losses in the impeller upon the geometric and constructive characteristics of the impeller, and upon the condition of work may be expressed with the equation:

$$\Delta \Phi_1 = \left| \frac{\Phi_0^2}{2 \Phi_{2u}} + \frac{\Phi_1^2 + \Phi_2^2}{2 \Phi_{2u}} \right|$$

$\Phi_0$ - coefficient of gasdynamic losses at the inlet of the impeller;
$\Phi_1 = c_1/c_0$ - coefficient of acceleration of the gas flow at the inlet of the impeller $(c_0)$, just before the inlet of the cascade of the impeller $(c_1)$;
$\Phi_2 = c_0/u_2$ - coefficient of the flow at the inlet of the impeller
$\kappa = (\kappa_1 + \kappa_2 + \kappa_3)_m$ - coefficient of the gasdynamic losses at the cascade of impeller;
$\kappa_1 = f(Re, \delta)$ - coefficient of gasdynamic losses in the impeller due to friction, which is the function of the Reynolds number $Re$ and the relative roughness $\delta$ of the channels between blades;
$\kappa_2 = f(\psi_u, \psi_1)$ - coefficient of gasdynamic losses in the cascade of the impeller due to diffusivity of the channels between the blades, which is the function of the equivalent angle of the diffusivity condition $\psi_u$, and the angle of groove of the channel between blades at the inlet edge of the cascade $\psi_1$;
$\kappa_3 = f(1)$ - coefficient of gasdynamic losses in the cascade of the impeller due to the deviation from the shockless flow in the cascade, and is the function of the angle of attack of the gas flow on the blades of the cascade $\psi_1$;

To establish the analytical functional dependences of the coefficients of gasdynamic losses, we use experimental and empirical data for turbocompressors.
Characteristic value which determines the flow in the cascade of the impeller is Mach number $M_{kr}$ in the critical crosssection of the cascade. It is very difficult and practically impossible to find the critical crosssection of the turning channel between the blades, using the theoretical methods of aerodynamic of the cascade, if we examine a real flow in the cascade connected with the detachment of the boundary layer, reversible flow phenomenon, jet-wave effect etc. That is why Mach number $M_{w1}=w_1/a_1$ is taken as characteristic value which defines the flow in the channel between the blades of the impeller. Critical phenomena in the cascade of the impeller are included with defining $M^r_{kr}$ and $M^\text{max}_{w1}$, which represent the characteristics of the cascade. Mach number $M_{w1}$ is called critical $M^r_{kr}$ if at any place on the profile of the blade, a critical speed appears. The critical $M^r_{kr}$ depends on the angle of attack in the cascade of the impeller $i$ (Fig.5) and that dependence in general may be expressed by the function /3/:

$$M^r_{kr} = A_0 a_1^2$$

(6)

The Mach number $M_{w1}$, is called maximal if critical flows appear at the critical crosssection of the cascade, and for given impeller it may be considered constant.

$$M^\text{max}_{w1} = A_1$$

(7)

The coefficients $A_0$, $a$ and $A_1$ depend on the geometric and constructive characteristics of the impeller /3/.

![Fig.5. Critical phenomena in the impeller](image)

![Fig.6. Dependence of the coefficient of gasdynamic losses in the impeller upon $M_w$ and $\theta_1$](image)

The appearance of critical flow at any place in the channel between the blades leads to the appearance of a densed blow wave, which is a typical thermo-dynamic irreversible process. Additional energy losses from this phenomenon result from the influence of sudden change of speed, pressure, density, and temperature on the cascade flow, which is followed by the detachment of the boundary layer, appearance of the reversible flow etc. The influence of critical phenomena on the coefficient of performance is included in $k_M$. The coefficient $k_{M}=1$ in the conditions when $M_w < M^r_{kr}$ (from B to C, Fig.5), and $k_{M} > 1$ in the conditions when $M_w > M^r_{kr}$ (left from B and from C to D, Fig.5). The coefficient $k_{M}$ may be expressed as a polynomial /3/:

$$k_{M} = \frac{M_{w1} - M^r_{kr}}{M^\text{max}_{w1} - M_{w1}}$$

(8)

In the condition D, it comes to the stop in the increase of flow in the channel between the blades. The results from the experimental studies for the coefficient of losses in the impeller /1/, Fig.6 determine these observations.

3.3. Some specialties in the geometric and constructive characteristics of the impeller with high Mach numbers

The optimization of the inlet of impeller from the condition for minimal losses in the impeller, at high $M_{w0}$, is connected with finding the minimum of the relati-
ve speed at the inlet of the cascade of the impeller $w_1$. For defined dependence $u_1 = \frac{u_1}{u_2} + \frac{c_1}{u_2}$, from the condition $3 |(w_1/u_2)^2|/3D_0 = 0$, we get the diameter at the inlet of the impeller $D_0 = D_0 / D_2$ at minimal $w_1$:

$$D_{omin} = d + \frac{4 \cdot k_v b_2 x_2 r c(1-\frac{2}{1-2M_0})^{1/2}}{1-2M_0}$$

(9)

And the corresponding angle of the blade:

$$\tan \theta_{lmin} = \frac{(D_0 - d)^{1/2} - 2M_0}{D_0 / 1 - M_0}$$

(10)

The results from the mathematical models for work characteristics of the one-stage turbocompressors /3/ show that optimal inlet in the impeller is established at:
- diameter of the inlet of the impeller $D_0 = D_{omin}$
- angle of the blade at the inlet $\theta_{lmin}$
- coefficient of acceleration $k' = 0.95 \sqrt{w_1}$
- diameter of the hub $d = 0$.

The impellers with small relative width ($b_2 / D_2 = 0.015 - 0.025$) enable work with higher $h_1$, and with that, establishing higher $n_1$; with lower $M_1$ and from that aspect they are more satisfactory in the turbocompressor heat pumps.

3.4. Adjustment of the diffuser and impeller work

The conception for the application of diffuser without blades in the freon turbocompressors appears as unsatisfactory, because its usual advantages cannot be utilized from the following reasons:

- to achieve work characteristics with wider operating zone, with application of a diffuser without blades, conceptually is unacceptable because of the sheer work characteristic due to the above mentioned reasons;
- to change the compressor capacity, i.e. to widen the work range is possible with the use of regulating systems (inlet guide, turning of the blades of the diffuser, changing the number of revolution);

![Fig.7. Dependence of the coefficients of performance of the impeller, diffuser and turbocompressor upon $\tau_{2r}$](image)
The work conditions are not coordinated with the best performances of the impeller and the diffuser. The diffuser without blades has bad performances in the conditions with small coefficients of flow $\varphi_2$. To achieve high $N$, it is necessary to have a high coefficient of fluid head $\varphi_2$, and that means that the calculated work condition of the impeller is with small $\varphi_2$. The impeller in that work condition has the best performances, while the diffuser without blades works with low coefficient of performances (Fig. 7).

The application of blade diffusers enables:
- adjustment of the work conditions with the best performances of the impeller and the diffuser and achievement of the maximal coefficients of performance of the turbocompressor;
- regulation of the turbocompressor with turning of diffuser blades.

4. JOINING OF THE TURBOCOMPRESSOR CHARACTERISTICS WITH THE HEAT PUMP ELEMENTS

Modelling of the work of the heat pump by joining the characteristics of its elements represents the basis for optimal design of the flow space and the regulating system of the turbocompressor.

Working point of the heat pump is achieved with joining the characteristics of its elements (evaporator, turbocompressor, condenser). The characteristic of the evaporator is represented with functional dependence $t_\ell=f(Q_\ell, t')$, where the temperature of the waste water $t'$ appears as a parameter. With joining the characteristics of the turbocompressor we gain the characteristic of compressor-evaporator sub-system. In modelling the characteristics of the compressor-evaporator we take into consideration the influence of the change of evaporation temperature upon the non-dimension work characteristic of the turbocompressor $\psi=f(\varphi_2)$. This results from the influence of the $M_{\varphi_2}$ change with the temperature level. The condenser characteristics are given with the functional dependence $t_k=f(Q_k, t_w)$, where the temperature of the hot water $t_w$ is taken as a parameter. Working point of the heat pump is determined at the point where the characteristic of compressor-evaporator sub-system cuts the characteristic of the condenser.

5. CONCLUSION

The application of one-stage turbocompressors for utilization of heat the waste water with heat pump for heating purposes is suitable in the analysed temperature range $t'=(30 - 40) ^\circ C$ and $t_w=(60 - 90) ^\circ C$. The essential specialty which characterizes the one-stage turbocompressors for heat pumps is the operation in the conditions with high Mach numbers. Increasing the difference in temperature levels of the heat pump cycle causes the increasing the necessary pressure ratio $N$, and the necessary $M_{\varphi_2}$. The coefficient of performance of the compressor designed for operation with high Mach numbers is insignificantly smaller than the coefficient of performance of the turbocompressor estimated with low Mach numbers. Work characteristics of the turbocompressor with high $M_{\varphi_2}$ are sheer which results from:
- the influence of the compressor high pressure ratio
- the influence of the critical phenomena in the flow space of the compressor.

In designing the flow space, special attention should be paid to the adjustment of the work conditions with the most satisfactory performances of the impeller and the diffuser. The application of the diffuser without blades is shown to be unsuitable. With correct designing of the regulating system (inlet guide, turning of diffuser blades, change of numbers of revolution), it is possible to change efficiency of the heat pump capacity in function of the heat consumer demands. Optimal design of the flow space and regulating system is impossible without previous knowledge of the turbocompressor work conditions in the heat pump. This is achieved by joining the characteristics of the turbocompressor with the characteristics of the condenser and the evaporator by using the model for their prediction.

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/2/ Sheets H.E., Nondimensional Compressor Performance for a Range of Mach Numbers and Molecular Weights, Akrom, Ohio
SUMMARY: An analysis has been made of the industrial water cooling systems which can be used as a heat source. For this analysis, the practical range of the temperature boundaries of the heat pump cycles are considered. The author has analysed the application of F12 as a working body. Some research results are given of the losses in the impeller and diffuser, their interactions, expected efficiency and expected gasdynamic characteristics. The criteria are defined for the optimal design of the one-stage centrifugal compressors applied in heat pumps at given temperature boundaries of the cycle.

RÉSUMÉ: On a élaboré une analyse des systèmes industriels de refroidissement qui peuvent être utilisés comme source de chaleur. On a établi l’aire pratique des températures frontières du cycle des pompes à chaleur en employant F12 en tant que corps de travail. On a présenté les résultats des recherches faites sur les pertes dans le circuit et le diffuseur, leurs actions mutuelles et les caractéristiques dynamiques de gaz respectives ainsi que les rendements. On a défini les critères pour projet optimum des compresseurs centrifuges à un étage appliqués dans les pompes à chaleur ayant des températures frontières du cycle.