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

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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 contructive, 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 $t' = 30 \div 40\,^{o}C$. The thermal transformation of this heat in the heat pump on the grade $t_{wl} = (60 - 90)\,^{o}C$ makes it usable in the heating systems or in the technological processes.

The main attention is paid on the temperature boundery effect of the heat pump cycle (t_o and t_k) on the pressure ratio Π, conditional Mach number $M_u = u_2/a'_o$, and through it, on the characteristics of the one-stage turbocompressors.

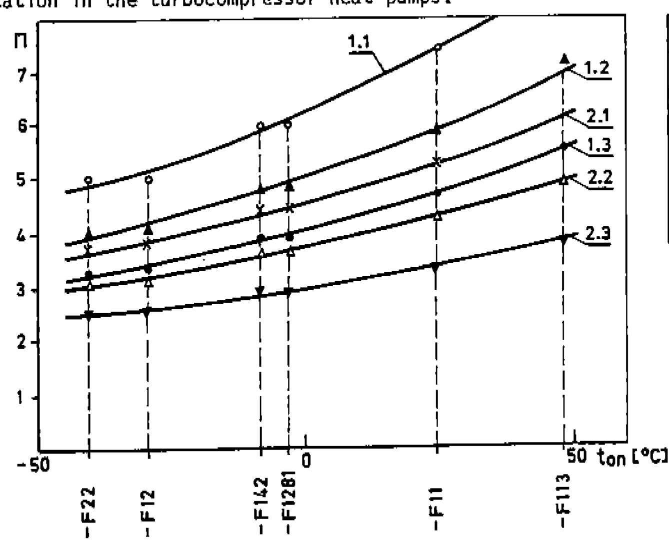
The main characteristic of these compressors is work in conditions with high Mach numbers. Optimal design of the flow space and the regulating sistem of the turbocompressor is possible if the modelling of the expected characteristics of the turbocompressor is done, and the work in conditions with high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

The basis for optimal design of the flow space and the regulating sistem 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_o and t_k) the needed pressure ration Π is the function of the refrigerant characteristics. The determined dependence of Π from the normal evaporation temperature t_{on} and from t_o and t_k (Fig.1), shows that the refrigerants with lower t_{on} are more satisfactory for the aplication in the turbocompressor heat pumps.



N°	t_o °C	t_K °C
1.1	20	90
1.2	20	80
1.3	20	70
2.1	30	90
2.2	30	80
2.3	30	70

Fig.1. Dependence of Π on the refrigerant properties

Fig.2 shows the boundary periferical velocity u_2 and the boundary pressure ratio Π 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 ψ= 0,5, and polytropic coefficient of performance η_p=0,85) and for particular conditions (t_0= 30°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 reletively small speed.

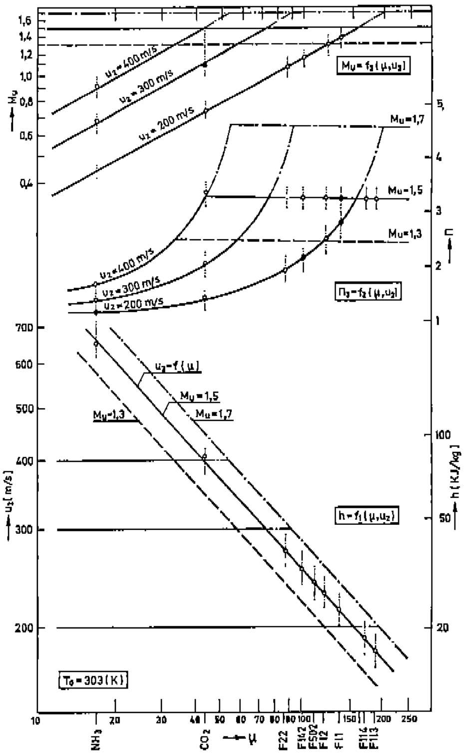


Fig.2. Dependence of u_2, M_u, Π, and h on the refrigerant

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_u, and the kind of the refrigerant, Fig.3.

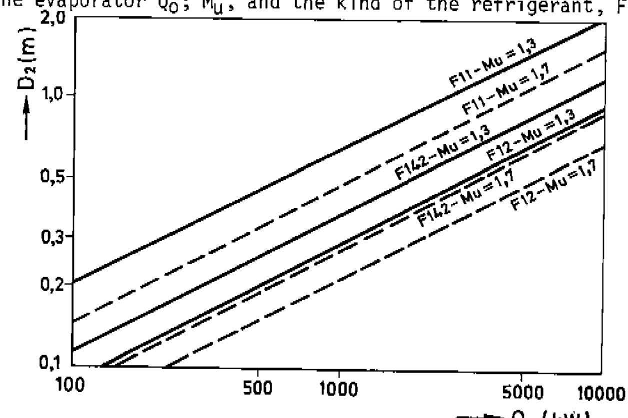


Fig.3. Dependence of D_2 upon Q_0,M_u and the refrigerant

3. MODELLING OF WORK CHARACTERISTICS OF THE ONE-STAGE TURBOCOMPRESSORS DESIGNED FOR HEAT PUMPS

The operation of the turbocompresor in conditions with high M_u 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 Π upon the coefficient of flow $\varphi_{2r}[\Pi=f(\varphi_{2r})]$, Fig.5c.

$$\Pi=[1+(\beta_{tr}+\beta_p)(k_1-k_2\varphi_{2r}\mathrm{ctg}\beta_2(\bar{k}_u-1)M_u^2]^{(\bar{k}_u)/(k_u-1).\eta_p} \tag{1}$$

If we exclude the critical phenomena from the analysis and suppose that the compressor has the same dimensionaless work characteristic $\Pi=f(\varphi_{2r})$, Fig.5a, for two different M_u, then the transformation of this dependence and the dependence $\Pi=f(\varphi_{2r})$, Fig.5c, in functional dependence $\Pi= f(\varphi_{1r})$, Fig.5b, and $\Pi=f(\dot{M})$, Fig.5d, that in fact represents work characteristic of the turbocompressor, is done by the following relations:

$$\varphi_{1r}=\frac{b_2}{b_1}\,\frac{D_2}{D_1}\,\frac{\tau_2}{\tau_1}\,\frac{k_{v2}}{k_{v1}}\,\varphi_{2r} \tag{2}$$

$$k_{v2}=\frac{v_0'}{v_2}=[1+\rho(\bar{k}_u-1)(k_1-k_2\varphi_{2r}\mathrm{ctg}\beta_2)(1+\beta_{tr}+\beta_{pr})M_u^2]^{(\bar{k}_u)/\bar{k}_u-1)\eta_p-1} \tag{3}$$

$$k_{v1}=\frac{v_0'}{v_1}=1-\frac{1}{2}M_0^2 \tag{4}$$

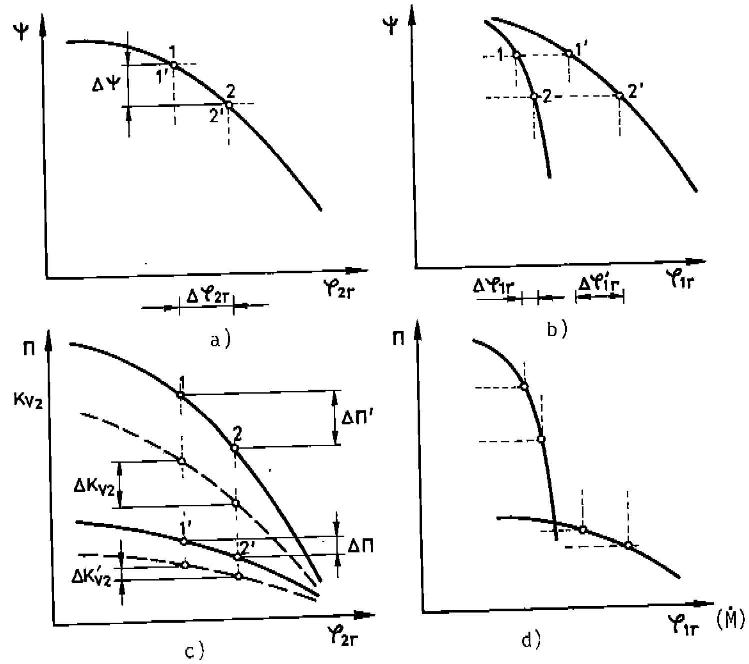


Fig.4. Dependences between the dimensionaless and work characteristics of the turbocompressor

From the equations 1, 2, 3, 4 and from Fig.5, it can been seen that at the high pressure ratio turbocompressors (high M_u), the change of the coefficient of the flow φ_{1r} does not follow the change of φ_{2r}, due to the influence of the fluid compressibility (k_{v2}) which is noticeable at the high pressure ratio freon turbocompressors. In the equations (1) - (4):

$\beta_{tr}+\beta_{pr}$ - means the friction and ventilation losses at the impeller disc and the losses at slots

k_1 and k_2 - the coefficients of the dimensionaless criterion of the fluid head φ_{2u} which depend upon the geometric and constructive characteristics of the impeller and upon the work condition

$\bar{k}_u$ - conditional adiabatic exponent

η_p - polytropic coefficient of performance

b_2, D_2, b_1, D_1 - width, that is, the outlet diameter, that one in the inlet of the cascade of the impeller

τ_2, τ_1 - the reduction coefficient of outlet or inlet crossection at the casade of the impeller

v_0', v_1, v_2 - specific volume at the inlet of the compressor (0), the inlet (1) and outlet of the cascade of the impeller (2)

ρ - 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_w and M_c. 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 characterists of the impeller, and upon the condition of work may be expressed with the equation:

$$\Delta\eta_k=\zeta_0 \frac{\varphi_0^2}{2\varphi_{2u}} + \zeta_k \frac{\varphi_{1r}^2 + \bar{D}_1^2}{2\varphi_{2u}} \tag{5}$$

$\zeta_0=f(k_c)$ - coefficient of gasdynamic losses at the inlet of the impeller;

$k_c'=c_1/c_0$ - 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);

$\varphi_0=c_0/u_2$ - coefficient of the flow at the inlet of the impeller

$\zeta_k=(\zeta_{k1}+\zeta_{k2}+\zeta_{k3})k_m$ - coefficient of the gasdynamic losses at the cascade of impeller;

$\zeta_{k1}=f(Re,\delta)$ - coefficient of gasdynamic losses in the impeller due to friction, which is the function of the Reynold's number Re and the relative roughness δ of the channels between blades;

$\zeta_{k2}=f(\nu_u,\nu_1)$ - 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 ν_u, and the angle of groove of the channel between blades at the inlet edge of the cascade ν_1;

$\zeta_{k3}=f(i)$ - 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 i;

$k_M=f(M_{w1}^{kr},M_{w1}^{max})$ - coefficient of gasdynamic losses due to the critical phenomena in the impeller;

$\bar{D}_1 = D_1/D_2$ - relative diameter at the inlet edge of the cascade of the impeller.

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_{wkr} in the critical crossection of the cascade. It is very difficult and practically impossible to find the critical crossection 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_{w1}^{kr} and M_{w1}^{max}, which represent the characteristics of the cascade. Mach number M_{w1} is called critical M_{w1}^{kr} if at any place on the profile of the blade, a critical speed appears. The critical M_{w1}^{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/:

$$M_{w1}^{kr} = A_0 e^{a i^2} \tag{6}$$

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

$$M_{w1}^{max} = A_1 \tag{7}$$

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

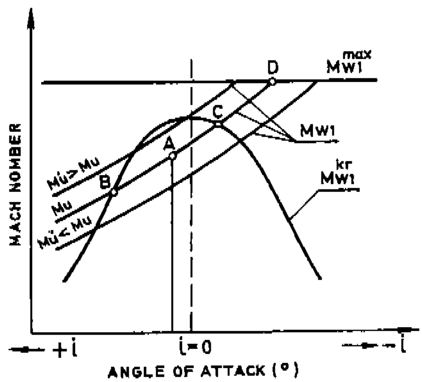


Fig.5. Critical phenomena in the impeller

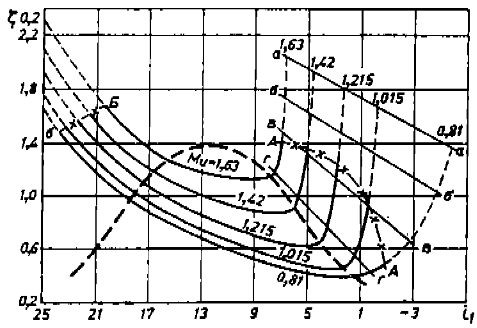


Fig.6. Dependence of the coefficient of gasdynamic losses in the impeller upon M_u and i_1

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 $M_w < M_{w1}^{kr}$ (from B to C, Fig.5), and $k_M > 1$ in the conditions when $M_{w1} > M_{w1}^{kr}$ (left from B and from C to D, Fig.5). The coefficient k_M may be expressed as a polynomial /3/:

$$k_M = P\left(\frac{M_{w1} - M_{w1}^{kr}}{M_{w1}^{max} - M_{w1}}\right) \tag{8}$$

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_0, is connected with finding the minimum of the relative speed at the inlet of the cascade of the impeller w_1. For defined dependence $(\frac{w_1}{u_2})=(\frac{u_1}{u_2})^2+(\frac{c_1}{u_2})^2$, from the condition $\partial|(w_1/u_2)^2|/\partial\bar{D}_o=0$, we get the diameter at the inlet of the impeller $\bar{D}_o=D_o/D_2$ at minimal w_1:

$$D_{omin}=\bar{d}+\left[2\,\frac{4\tau_2 k_{v2}\bar{b}_2\varphi_{2r}k'_c}{\tau_1 k_D k_{vo}}\left(\frac{1-M_o^2}{1-2M_o^2}\right)^{1/2}\right]^{2/3} \tag{9}$$

And the corresponding angle of the blade:

$$tg\beta_{1wlmin}=\frac{(\bar{D}_o-\bar{d}^2)^{1/2}}{\bar{D}_o}\left(\frac{1-2M_o^2}{1-M_o^2}\right)^{1/2} \tag{10}$$

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 $D_o=D_{owlmin}$
- angle of the blade at the inlet $\beta_1=\beta_{1wlmin}$
- coefficient of acceleration $k'_c=(0,95\div 1)$
- diametar of the hub $\bar{d}=0$.

The impellers with small relative width $(\bar{b}_2=b_2/D_2 = (0,015-0,025))$ enable work with higher M_u, and with that, establishing higher Π_c with lower M_{w1} 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);

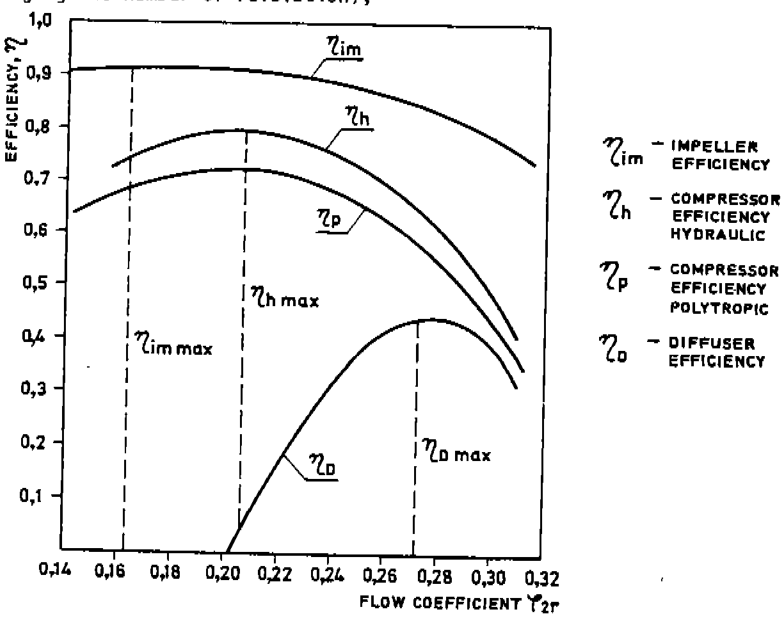


Fig.7. Dependence of the coefficients of performance of the impeller, diffuser and turbocompressor upon φ_{2r}

- 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 φ_{2r}. To achieve high Π, it is necessary to have a high coefficient of fluid head φ_{2u}, and that means that the calculated work condition of the impeller is with small φ_{2r}. 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. JOING 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 $t_o=f(Q_o,t')$, 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 $\psi=f(\varphi_{2r})$. This results from the influence of the M_{w1} change with the temperature level. The condenser characteristics are given with the functional dependence $t_k=f(Q_o,t_{w1})$, where the temperature of the hot water t_{w1} 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 $t'=(30 - 40)^oC$ and $t_{w1}=(60 - 90)^oC$. The essential specialty which charaterizes 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 Π, and the necessary M_u. 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_u 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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TEMPERATURE BOUNDARY EFFECT OF THE LARGE HEAT PUMP CYCLE ON THE CHARACTERISTICS OF THE TURBOCOMPRESSOR

SUMMARY: An analysis has been mode 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 impller and diffuser, their interactions, expected efficiency and expected gasdynamic characteristics. The criteria are défined for the optimal design of the one-stage centrifugal cómpressors applied in heat pumps at given temperature boundaries of the cicle.

INFLUENCE DES FROTIÈRES DE CHALEUR DU LE CYCLE DES GRANDES POMPES A CHALEUR SUR LES CARACTERISTIQUES DU TURBOCOMPRESSEUR

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