

Cascade pressure exchange, thermal compressor, working cycle, temperature, power efficiency, performance, load characteristic

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THERMAL COMPRESSOR OF CASCADE EXCHANGE BY PRESSURE

Summary. A new method of organization of working cycle of device of direct transformation of heat in the located work of compression of air, based on principle of cascade exchange by pressure, is exposed; the results of pre-selection of basic dimensional and structural parameters of thermal compressor of cascade exchange by pressure are adduced; some special features of its working process are considered; main directions of perfection of working cycle of thermal compressors of cascade exchange of pressure are shown.

ТЕПЛОВОЙ КОМПРЕССОР КАСКАДНОГО ОБМЕНА ДАВЛЕНИЕМ

Аннотация. Раскрыт новый способ организации рабочего цикла устройства прямого преобразования теплоты в располагаемую работу сжатия воздуха, основанный на принципе каскадного обмена давлением, приведены результаты предварительного выбора основных размерных и конструктивных параметров теплового компрессора каскадного обмена давлением, рассмотрены некоторые особенности его рабочего процесса, показаны основные направления совершенствования рабочего цикла тепловых компрессоров каскадного обмена давления.

1. INTRODUCTION

Development of pneumatic transport of granular freights and technological pneumatic sets of road-building and special machinery supposes the reduction of manufacturing cost of the compressed air. Air injection by expensive and complex on design piston or vane-type compressors particularly with self-contained, for example, diesel drive is characterized by high power expenditures. The generalised efficiency of compressor with a diesel engine taking into account the losses of transformation of initial thermal energy into mechanical one in a diesel and mechanical energy in the located work of compression of air does not exceed the values of 0,2 0,21 in the best specimens.

2. THE FEATURES OF THE CASCADE PRESSURE EXCHANGERS

Considerable reserve of reduction of power expenditures of production of the compressed air and simplification of design of supercharger associates with the use of devices of direct transformation of thermal energy in the compressed air on the base of cascade pressure exchange (CPE). The sets realizing the cascade compression are a new variety of pressure exchangers, in particular applied in the systems of supercharging of internal combustion engines. Compression of air in CPE as well as in wave pressure exchanger of the known supercharging system of «Comprex» is carried out as a result

of direct contact with compressing gases, however with the substantial distinction of organization of working process. Principle of the action of CPE is described in works [1, 2, 3].

Typical construction of exchanger (fig. 1.) is the rotor with longitudinal head-exchanging cells, revolved in stator. Mass-changing channels as well as the window of admission and rejection of compressing gas are located in one of end face covers of stator (A), but the window of admission and rejection of compressed gas are located in the other end of face covers (B). The rotor is driven to the rotation with rotational speed of $2000 \dots 3000 \text{ min}^{-1}$ by means of an electric motor or other drive of small power. The still pictures of basic units of one of the CPE construction are shown on the fig. 2.

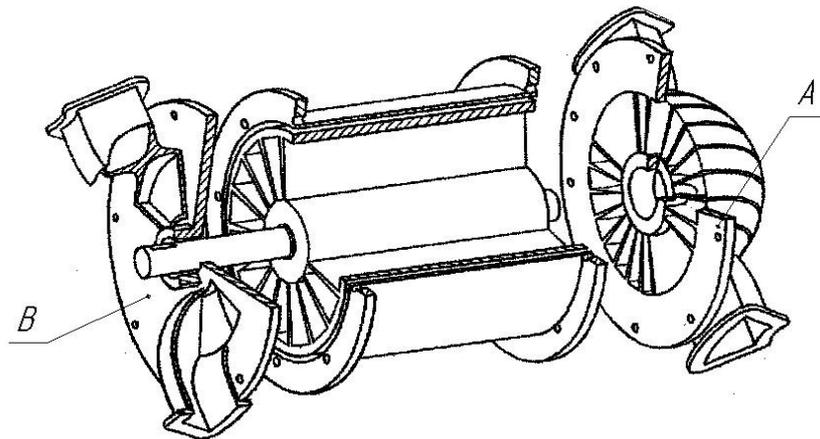


Fig.1. Principal view of cascade pressure exchanger (CPE)

Рис.1. Общий вид каскадного обменника давления

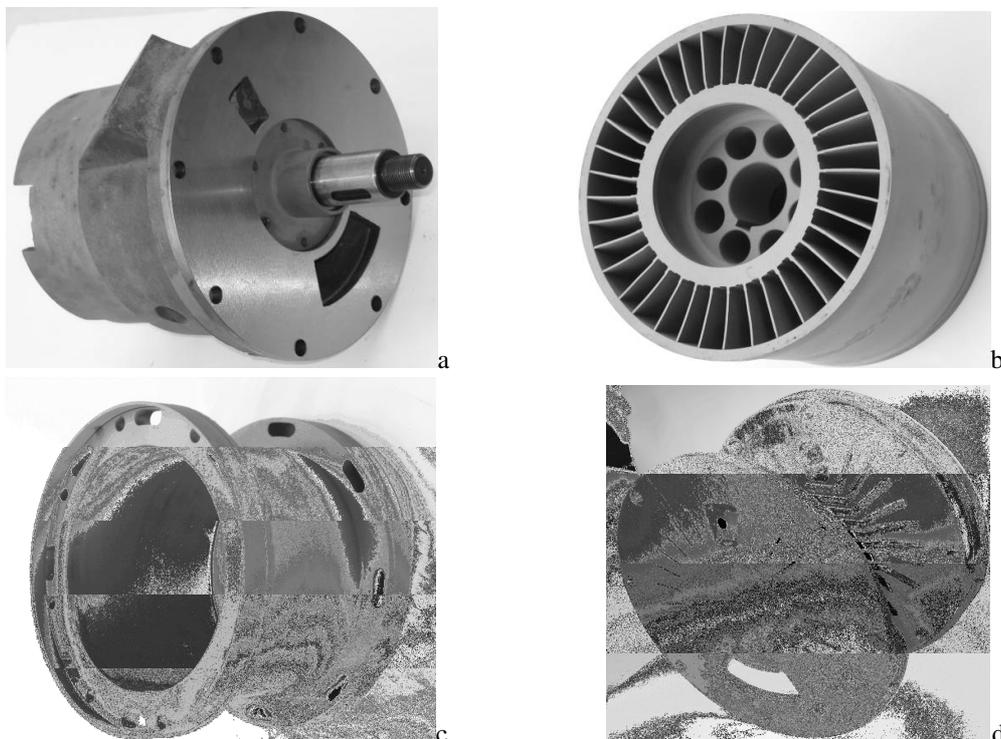


Fig. 2. Basic elements of construction of the experimental model of the CPE: a – stator; b – rotor; c – body; d – end face cover from the side of admission and rejection of compressing gas

Рис. 2. Основные элементы конструкции опытного образца КОД: а – статор; б – ротор; с – корпус; d – торцевая крышка со стороны подвода и отвода сжимающего газа

The high effectiveness of the unit of CPE is confirmed by the stand tests on the base of the engine 6FSch12/14. So, at the parameters of the compressing gas $T_{g1}=800$ K, $P_{g1}=0,25$ MPa the efficiency of pilot plant of the CPE achieves 0,84 and on the mode of $T_{g1}=850$ K, $P_{g1}=0,25$ MPa – 0,86.

Power perfection of working process of cascade exchanger is realized in the considerable exceeding of expense of the compressed air of relatively compressing medium, and the temperature of the medium is higher, to a greater degree.

3. APPLICATION OF THE CASCADE PRESSURE EXCHANGERS IN COMPRESSORS OF HEAT COMPRESSION

The noted property of cascade exchangers is fixed in the basis of creation of compressor of direct transformation of thermal energy in the located work of the compressed air. A part of the forced air is discharged to the user in the simplest chart of the thermal compressor, and other part of air is heated in the source of admission of heat and is directed in the window of admission of high pressure of exchanger, where is used as a compressing medium. Unique simplicity and rather high efficiency of single-stage thermal compressor stipulate attractiveness of its application in the systems of air supply with maximal pressure of supercharging to 0,4...0,5 MPa depending on the maximal temperature of the working cycle T_z .

Substantially high pressure head and power efficiency has been realized by the double-stage compressor of cascade exchange by pressure. The chart of the modernized compressor of the CPE thermal compressor with the intermediate cooling and preliminary heating of the compressed air and compressing medium is shown on the fig. 3.

Pumping of air in the given device is carried out as a result of successive compression of cold air and thermal expansion of hot air in aggregates of the CPE of the first and second stages. Thus, the relations of degrees of compression of air in the first and second stages π_1 , π_{II} are not arbitrary, as, from one side, it is subordinate to the condition of balance of charges of working mediums in the lines of high pressure of the first stage and low pressure of the second stage, on the other side – to the condition of providing of blowing out and ousting of working medium in the cells of both CPE rotors in the indicated lines.

In such a case the relations π_1 and π_{II} depend on total pressure head of the thermal compressor π_k and maximal temperature of the working cycle T_z .

Gas dynamic analysis of double-stage thermal compressor of the cascade change of pressure is enough labor-consuming in spite of the structural simplicity of device of the CPE thermal compressor. The analysis includes the search of the modes of joint operations of CPE aggregates of the first and second stages. Therefore the pre-selection of basic dimensional parameters of elements of the system, meeting the condition of realization of the productivity of compressor, must precede to the more precise computation of working cycle of the CPE thermal compressor.

The results of calculations of dimensional relations of frontal sizes of rotors of exchangers of the first and second stages are below brought on the example of compressor by the productivity $G_b=0,1$ kg/sec with different total pressure head π_k , with the maximal temperature T_z of the cycle and differential pressures created by the scavenger fan ΔP_B .

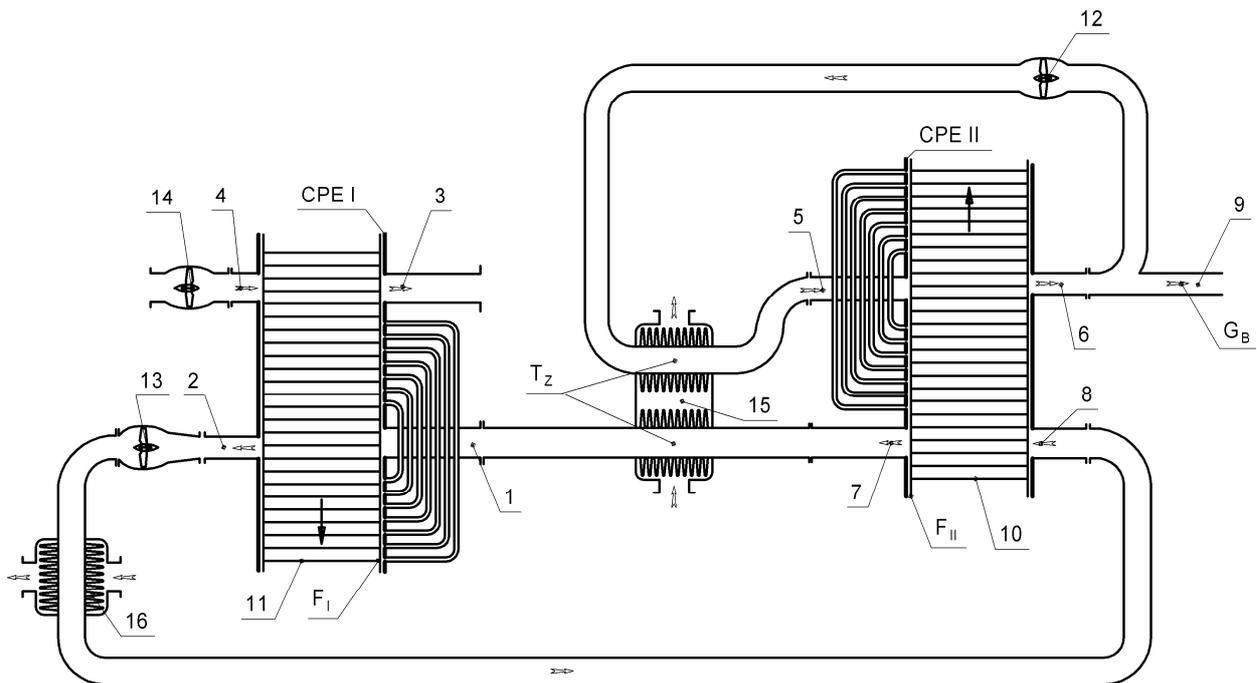


Fig. 3. Flow diagram of the double-stage thermal compressor of the CPE

1, 2, 5, 6 – windows of admission and removal of high pressure (AHP and RHP) according to CPEI and CPEII; 3, 4, 7, 8 – windows of removal and admission of low pressure (RLP and ALP) accordingly CPE I and the CPE II; 9 – branch pipe of the compressed air bleed-off; 10 – rotor of the CPE II; 11 – rotor of the CPE I; 12 – positive-displacement fan of the second stage; 13 – positive-displacement fan of the first stage; 14 – scavenger fan; 15 – heater; 16 – cooler

Рис. 3. Принципиальная схема двухступенчатого теплового компрессора КОД

1, 2, 5, 6 – окна подвода и отвода высокого давления (ПВД и ОВД) соответственно КОД1 и КОД2; 3, 4, 7, 8 – окна отвода и подвода отвода низкого давления (ОНД и ПНД) соответственно КОД I и КОД II; 9 – патрубок отбора сжатого воздуха; 10 – ротор КОД II; 11 – ротор КОД I; 12 – вытеснительный вентилятор второй ступени; 13 – вытеснительный вентилятор первой ступени; 14 – продувочный вентилятор; 15 – нагреватель; 16 – охладитель

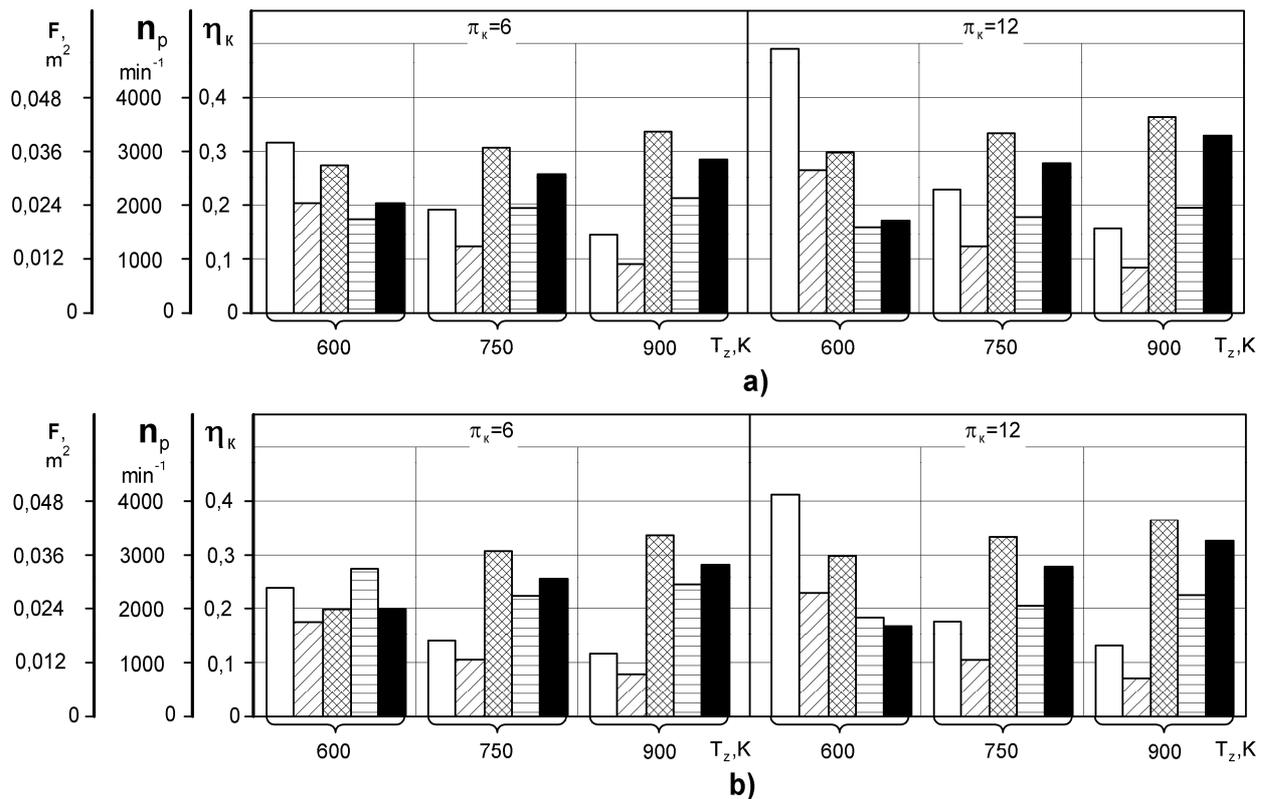


Fig. 4. Combination of structural and regime parameters of the CPE compressor by productivity $G_b=0,1$ kg/sec for different conditions of its work: a – at pressure head of scavenger fans of the first and second stages $\Delta P_{BI}=\Delta P_{BII}=15$ kPa; b – at pressure head of scavenger fan of the first and second stages $\Delta P_{BI}=\Delta P_{BII}=20$ kPa; \square - cross-section area of rotor flow part of the CPEI; F_{P1} ; \square - cross-section area of rotor flow part of the CPEII, F_{P2} ; \boxtimes - rotational speed of the CPEI rotor, n_1 ; \boxminus - rotational speed of the CPEII rotor, n_{II} ; \blacksquare - power efficiency of the compressor, η_c

Рис. 4. Сочетание конструктивных и режимных параметров компрессора производительностью $G_b=0,1$ кг/с КОД для различных условий его работы: а – при напорности вытеснительных вентиляторов первой и второй ступени $\Delta P_{BI}=\Delta P_{BII}=15$ кПа; б – при напорности вытеснительных вентиляторов первой и второй ступени $\Delta P_{BI}=\Delta P_{BII}=20$ кПа; \square - площадь сечения проточной части ротора КОД I, F_{P1} ; \square - площадь сечения проточной части ротора КОД II, F_{P2} ; \boxtimes - частота вращения ротора КОД I, n_1 ; \boxminus - частота вращения ротора КОД II, n_{II} ; \blacksquare - энергетический к. п. д. компрессора, η_c

The principal method of reduction of overall sizes of the CPE thermal compressor is an increase of the pressure head of scavenger and positive-displacement fans 14, 13, 12 (see the Fig. 3.). Main purpose of these fans is the- realization of circulation of working mediums in the contours of low and high pressure of both exchangers. As seen from the histograms resulted on a fig.4, even the insignificant rise of pressure head of scavenger fans allows considerably to decrease the frontal areas (F_{P1} and F_{P2}) of rotors of both CPE. At the same time, rise of ΔP_{BI} and ΔP_{BII} is accompanied by the increase of consumptions of mechanical energy on the drive of fans. Thus, the pressure differentials created by circulation fans, are essentially the key parameters of varying in search of the trade-off between sizes and efficiency of the thermal compressor.

It should be noted, however, that since some moment relying on a general dimensionality and feature of structural execution of compressor, further reduction of ΔP_{BI} and ΔP_{BII} is not accompanied by the rise of η_c because of strengthening of losses through the end face interfaces of rotors in view of the increase of sizes of the lasts.

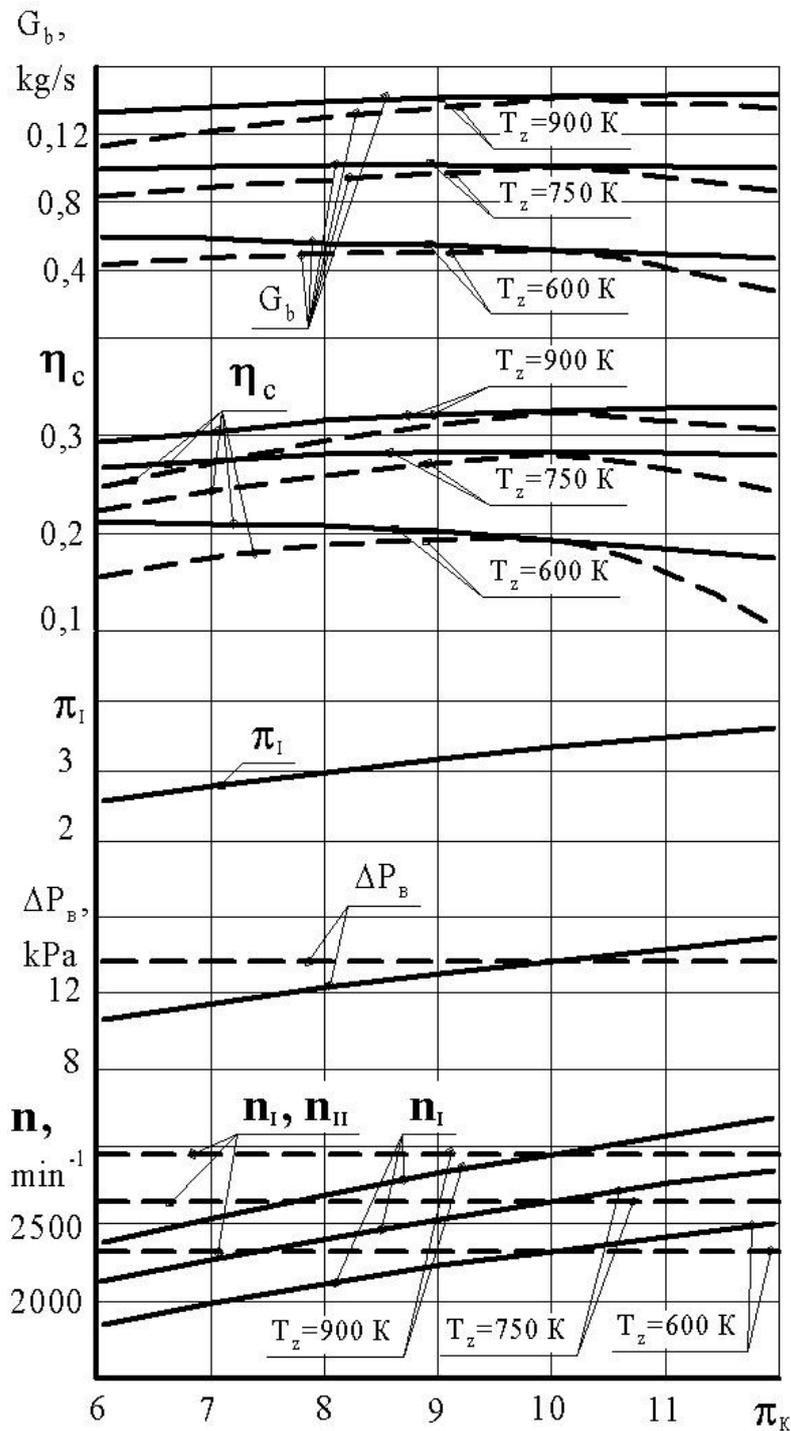


Fig. 5. Load characteristic of the CPE double-stage compressor:

— — — without regulation;

— — — with regulation of rotational speed n_{PI} of the CPE rotor and pressure head ΔP_{BI} of scavenger fan of the first stage

Рис. 5. Нагрузочная характеристика двухступенчатого компрессора КОД:

— — — без регулирования;

— — — с регулированием частоты вращения ротора n_{PI} КОД и напорности вытеснительного вентилятора первой ступени ΔP_{BI}

The maximal temperature T_z of the cycle exerts highly noticeable influence on the indexes of working process. The T_z increase is not only the instrumental in the substantial increase of efficiency but also provides the lowering of frontal dimensions of the exchangers of both stages. Thus the T_z influencing on η_c . does increase as far as the rise of general compressor pressure head π_k . In its turn, the lowering of sizes of the exchangers at the T_z rise is related to reduction of relative expenditure of compressing medium on the compression of the compressed air.

The effectiveness of heat compressor aeration is estimated by power efficiency according to expression:

$$\eta_c = \frac{G_b \cdot H_c}{Q + \frac{N_F}{\eta_e}} \quad (1)$$

where: G_b - a mass consumption of the compressed air; H_c - adiabatic heat drop in compressor; N_F - power, spent on drive of auxiliary fan; η_e - efficiency of conditional heat-power-plant, realizing drive of fan, Its taken $\eta_e = 0,31$.

At the temperature $T_z=900K$ on the mode $\pi_k=12$, the calculation value of efficiency η_c . makes 0,32. In the real cycle of the pilot plant of thermal compressor, created in the laboratory of the DVS of EUNU named after V.Dahl, an index of efficiency $\eta_c=0,28$ is attained already on the stage of preliminary tests. Possibility to carry out the working cycle of the CPE with $T_z=900K$ is conditioned by the cyclic cooling of the rotors of the CPE in the process of the periodic filling of pressure head-changing cells of the CPE rotor by an air charge. The level of power efficiency realized by the thermal compressor is unattainable on today, even in the theoretical cycle for the off-line units of transformation of thermal energy in the located work of compression of air, as far as the authors know.

The positive feature of working cycle of the double-stage thermal compressor with the intermediate cooling of the compressed air and intermediate heating of the compressing medium consists in the declivity of its hydraulic characteristic (fig. 5.).

Thus in the case of the simultaneous regulating of rotational speeds n_{PI} and n_{PII} of the CPE and pressure head ΔP_{BI} of scavenger fan depending on total pressure head π_k of the compressor, the expenditure G_k of the forced air and compressor efficiency η_c are subject to the insignificant influencing of hydraulic resistance of user in the range of the operating modes of work of the set, saving substantial dependence only on the maximal temperature T_z of the cycle.

And finally, it is impossible not to note the failure-free operation of the CPE set exposed in the process of tests of the pilot construction of compressor and row of pre-production models of the aggregates of the CPE. It is due to the absence of mechanical displacers, discretely controlled gas-distributing organs, as well as because of relatively low rotational speed of the CPE rotor.

4. CONCLUSIONS

High power efficiency and propitious hydraulic characteristic of discharge under pressure of the CPE double-stage thermal compressor in combination with reliability of its construction exposes prospects of its application as an aggregate of air supply of wide class of thermal power plants.

The principal reserve of further increase of the compressor efficiency consist in the rise of maximal temperature T_z of the cycle by means of application of new high-temperature materials (for example, carbon-fibrous polymers) for manufacture of channel running elements of device, and also due to reduction of losses of working medium through the end interfaces of co-rotors.

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