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A New High-Efficiency Steam Power Cycle for Supercritical Steam Conditions*

The paper describes the trend toward increasing the efficiency of steam power cycles, and on this line a novel steam power cycle for high ,evidently supercritical initial steam conditions, with consistent application of feed water heating beyond the saturation temperature corresponding with the steam pressure at the first bleed from the turbine. Owing to this cycle, efficiencies higher than those of conventional cycles working at the same temperature limits can be achieved. Subsequently some characteristic features of the new cycle are briefly discussed.

1. Introduction

The rapid growth of electric energy demands brings about the growing trend towards larger unit ratings, and at the same time towards higher thermal efficiencies as well as towards the application of nuclear fuels, respectively [1].

In order to increase the thermal efficiency attainable in a steam power cycle, the basic Rankine cycle composed of two isobars (p_1, p_2) marking conditions of heat reception and extraction and of two adiabates determining the positive work of expansion and the negative work of compression (pumping) has been continuously developed towards:

— increasing the initial steam parameters (p_1, t_1) along with decreasing the exhaust pressure (p_2) to the level determined by the temperature of the cooling water,

— full adoption of the regeneration principle by means of bleeding condensing steam for feed heating in course of expansion in the turbine,

— application of single or even double interstage reheat of the expanding steam,

— consistent increasing the efficiencies of all components of the cycle.

Despite of making use simultaneously of all the methods mentioned above, the resulting thermal efficiency of contemporary modern steam power plants, even those designed for supercritical steam parameters at turbine intake, being dependent, among others, upon the cycle efficiency, the steam generator efficiency and the thermodynamic efficiency of the turbine, barely approaches the level of 42%. A further increase of efficiency would require the adoption of higher initial temperatures (t_1) what in turn is a material problem associated with mastering of high-temperature designs without loss in reliability. Thus

* Praca wykonana w ramach problemu resortowego PAN-19, grupa tematyczna 1.

the lack of proper heat resistant materials does not permit to increase the efficiency above the stated limit which may be indicated as the actual "efficiency threshold", in fact much lower than the ideal Carnot efficiency for given temperature limits.

Some slight improvements are still possible regarding the thermodynamic efficiency of the turbine, first of all by means of better understanding the problems of wet steam flow and by consistent application of design optimization procedures.

With increase of unit capacity what for a specified set of initial parameters (p_1, t_1) and for a p_2-value as determined by the temperature of cooling water means also the increase of mass- and volume flow of steam at turbine inlet, the thermal efficiency increases, as a result of lower flow losses as blade heights become longer. This relationship clearly comes out from the graph (Fig. 1) published by Harris [2] on the basis of statistical data for turbines in service. From here advantages resulting from the adoption of supercritical steam conditions for any ratings exceeding the level of about 500 MW become obvious, particularly when accompanied by a double interstage reheat in order to avoid excessive steam wetness in course of expansion. More detailed thermodynamic data for a 600 MW-unit were given by E. Scheurer [3].

It is exactly for these supercritical parameters, about 240 bar and 540 deg C, with double interstage reheat to the same temperature, that power units of an output in the

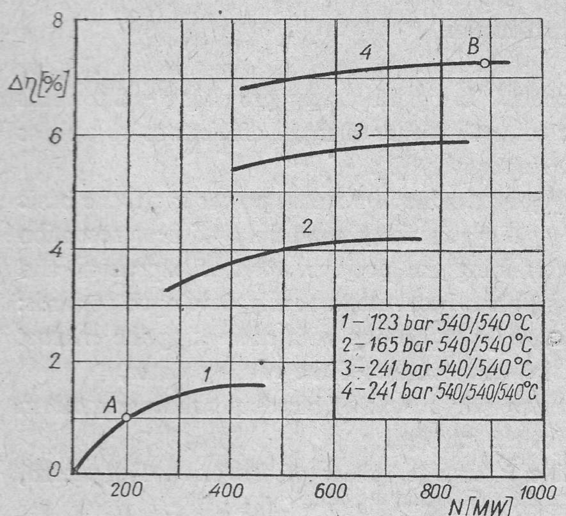


Fig. 1. Increase of thermal efficiency with unit turbine capacity and initial steam parameters

range of 500 to 900 MW make plant efficiencies of about 41 - 42% attainable! The gain in efficiency marking the improvement from a 200 MW-unit with initial steam conditions of about 120 bar and 540/540 deg C (point A on Fig. 1) to a 900 MW-unit, in fact actually a limiting capacity for supercritical steam conditions with double interstage reheat — about 240 bar and 540/540/540 deg C (point B) — is of the order of slightly above 6% what has to justify economically the trend towards supercritical steam conditions to be adopted in large power units.

With initial steam temperatures being unchanged even higher steam pressures could be adopted for turbine ratings exceeding the level of about 900 MW, with prospects for further rise in efficiency. However as this rating marks the actually limiting capacity of normal-speed one-shaft turbines, the all-over plant efficiency of the order of 41 - 42% has become a limiting value, to be indicated the actual "efficiency-threshold".

It should be noted, however, that by modification of the cycle resulting for the same turbine rating in a higher mass flow of the working fluid at turbine inlet, even higher inlet pressures could be admitted, without decreasing the volume-flow required for high efficiency conditions. More specifically, it can be stated that by increasing the inlet pressure, say from 240 to 380 or 450 bar, an additional gain in efficiency of 2.8 to 3.5%, respectively, can be achieved, and perhaps another gain of about 1.2 to 1.5% as a result of the 3-rd interstage reheat.

As no such modification of the Rankine cycle has been known, neither any other suggestion aimed to increase the cycle efficiency, the opinion originated that in turbine engineering a threshold of efficiency has been reached to overcome which the development of suitable heat-resistant materials seems to be undispensable.

Hence very much attention has been paid during the last decade to new concepts of increasing the cycle efficiency by means of combining the steam cycle with another high-temperature cycle of a higher temperature of heat reception. This refers as well to the combined gas/steam cycle with a high-temperature gas turbine designed for initial temperatures even as high as 1200 to 1500 deg C as particularly to the m-h-d-generator combined in a subposition cycle with a modern steam power plant of high efficiency. While the m-h-d-generator is not yet applicable to nuclear fuels in a closed cycle, the higher thermal efficiency of the combined cycle, from the economical point of view, is not necessarily superior to even lower cycle-efficiencies if they can be achieved with nuclear fuel.

2. The basic concept

Against this background the question arises if the steam power cycle derived from the classical Rankine cycle by means of steam super- und resuperheating, even more than once in course of expansion, and by full application of regenerative feed heating, is still open — under given temperature limits — to further modifications and improvements. To answer this question it is necessary to go back to basic laws of thermodynamics which determine the conversion of heat into mechanical energy.

The thermal efficiency of such a process is given by the ratio of the amount of heat converted into mechanical energy to the amount of heat supplied:

$$\eta=\frac{Q_1-Q_2}{Q_1}=1-\frac{Q_2}{Q_1}. \qquad (1)$$

For the ideal Carnot cycle this expression marking the highest efficiency attainable within given temperature limits becomes transformed to:

$$\eta_C=1-\frac{T_2}{T_1}, \qquad (2)$$

where T_1 and T_2 indicate the upper and lower temperature limits of the cycle as expressed in absolute temperatures, deg K.

The improvement of a heat power cycle, it's carnotization, demands the decrease of the Q_2/Q_1-ratio to the value of the ratio of the limiting temperatures T_2/T_1.

In the conventional steam power cycle the lower terminal temperature T_2 is practically constant, determined by the temperature of cooling water in the condenser. On the contrary the higher terminal temperature T_1 varies depending on the phase of heat supply,

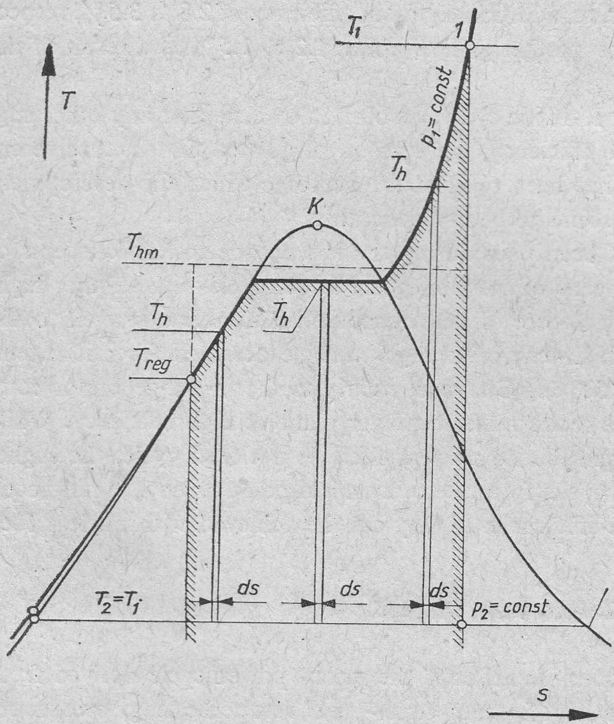


Fig. 2. Temperature-entropy diagram illustrating heat addition in the conventional steam power cycle

which consists consecutively in feed water heating, it's evaporation and finally the superheating of steam up to the assumed high temperature limit T_1.

Under these circumstances the expression for the ideal Carnot cycle efficiency can be applied only to an elementary cycle denoted by the limiting temperatures T_h and T_1:

$$\eta_{\text{elem}} = \eta_{\text{C}} = 1 - \frac{T_1}{T_h}, \tag{3}$$

where $T_1 = T_2$, while T_h varies from the feed water temperature at the exit of the regeneration system T_{reg} up to T_1 (Fig. 2).

For the cycle as a whole we arrive at the expression:

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \frac{1}{Q_1} \int_{\text{Treg}}^{T_1} \eta_{\text{C}} \left(\frac{T_2}{T_h} \right) dQ_1 \tag{4}$$

or substituting the area under the process curve from T_{reg} to T_1 by a rectangular area with the mean upper temperature T_{hm}:

$$\eta = 1 - \frac{T_2}{T_{hm}} < 1 - \frac{T_2}{T_1}, \tag{5}$$

where

$$T_{hm} = \frac{i_1 - i_{reg}}{s_1 - s_{reg}}. \tag{6}$$

Obviously, efficiency increases with T_{hm} approaching T_1. Therewith the old principle of Carnot in that increasing thermal efficiency depends on higher average temperatures of heat supply becomes confirmed. Consequently, we arrive at the conclusion that feed water heating which is generally applied to conventional steam power cycles may be advantageously conducted far beyond the level given by the saturation temperature of the first bleed point. This in turn may be accomplished by exchanging heat between highly superheated steam extracted in course of expansion at the turbine and the feed water stream heated up already by means of bled condensing steam in conventional regenerative heat exchangers. Such an exchange may prove particularly effective in the range of supercritical pressures [4].

3. Description of the cycle

According to this basic concept, steam of high initial parameters (p_1, t_1) is being expanded to a pressure $p_{2'}$ which, in principle, is also supercritical. At the state represented by point $2'$ (Fig. 3 and 4) the steam flow G_1 is being divided into two parts. One of them, $G_2 = G_1 (G_2/G_1)$ flows to the first interstage reheater and its temperature rises to $t_{1'}$ corresponding with point $1'$. The rest, $G_3 = G_1 - G_2 = G_1 (1 - G_2/G_1)$, exchanges heat in the regenerator with feed-water (G_1) which initially has been partly (G_2) heated by condensing steam extracted from the turbine in several bleeding points up to temperature t_5 and than finally to temperature t_6 as a result of mixing with the steam flux $G_1 (1 - G_2/G_1)$. In the regenerator the temperature of the feed-water (G_1) is being raised from t_6 to t_7 while the heating steam flux $G_1(1 - G_2/G_1)$ is simultaneously cooled down from temperature $t_{2'}$ to t_8, and finally heated again to t_9 in the process of pressure rising in the high-pressure feed pump P_2. The temperature difference $T_{HE} = t_8 - t_6$ is to be assumed on the ground of economic considerations.

Owing to the high supercritical steam conditions at point $1'$, the expansion of the steam flux G_2 requires at least one interstage reheat, performed in a conventional system in parallel with regenerative feed heating by extracted condensing steam.

Thanks to the cycle layout and particularly to more effective feed-water heating up to the temperature t_7 which is much higher than that (t_5) attainable solely by conventional regenerative heating, the mean upper temperature of heat supply T_{hm} increases, and accordingly the efficiency of the cycle, becoming definitely higher than that of a conventional one ($G_2/G_1 = 1$).

Along with the increase of the cycle-efficiency there is also an increase of steam flow in the high-pressure turbine, very favourable from the point of view of applying much higher steam pressures at turbine inlet and thus again increasing the attainable cycle-efficiency. Accordingly, the overall gain in efficiency is composed of two parts — one referred to the application of much higher steam pressures at turbine inlet and one more

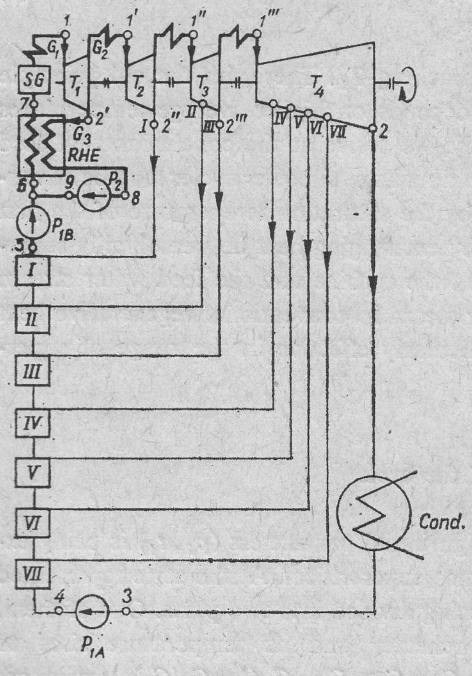


Fig. 3. Basic concept of the supercritical steam power plant with high-temperature regeneration

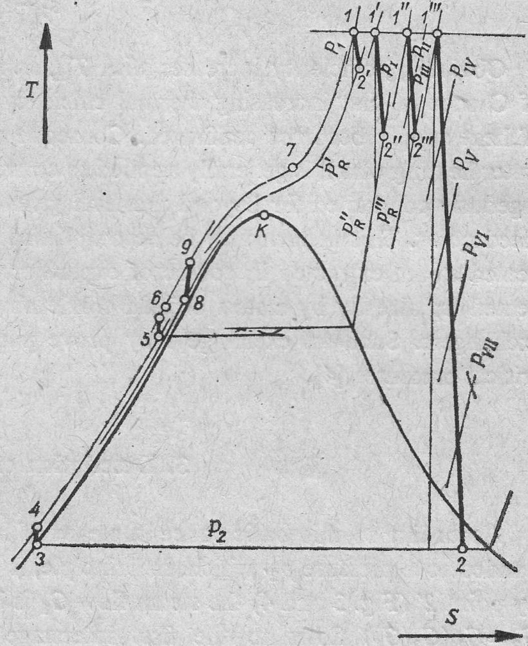


Fig. 4. The novel supercritical steam power cycle in the $(T-s)$-diagram

interstage reheat than in the conventional cycle, and another bound to more effective feed water heating, beyond the saturation temperature corresponding with the steam pressure at the first conventional bleed from the turbine. In other words the new cycle is being accomplished by superposition of two component cycles — the first a rather conventional one *1—2′—1′—2″—1″—2‴—1‴—2—3—4—5—7—1* with supercritical steam inlet conditions, and the second one with the same steam parameters at inlet but contained entirely within the range of supercritical pressures: *1—2′—8—9—6—7—1*. Obviously, this second cycle which may be regarded as a "closed gas turbine cycle", while supplied with heat from the external source in the steam generator, after work being done in turbine T_1, discharges all the remaining heat, behind point *2′*, to the stream of feed-water. The thermal efficiency of this closed cycle approaches therefore 100%, even if work is being done not without losses.

4. Cycle performance analysis

Proceeding from the general description and qualitative opinion to a more quantitative analysis of the novel cycle obtained by superposition of two component cycles with the same supercritical steam inlet conditions, as mentioned above, we have to discuss before all the problem of efficiency.

Denoting:

L_T — internal work of the turbine per unit mass flow at inlet,

L_{0T} — the same quantity of a turbine working on a conventional cycle, without high-temperature regeneration,

L_P, L_{0P} — the works of feed pumps in the compared, regenerative and conventional (reference) cycles, respectively,

Q, Q_0 — heat supply in both cycles, per unit mass flow at inlet,

η, η_0 — "net" internal efficiencies of the two cycles in question,

Δi_{T_1} — enthalpy drop at the high-pressure turbine T_1,

Δi_{P_2} — enthalpy rise at the feed pump P_2,

$g=G_2/G_1$ — the ratio of the steam flow in the condensing cycle to the steam flow at turbine inlet,

we arrive at the relations:

$$\eta=\frac{L_T-L_P}{Q}, \quad \eta_0=\frac{L_{0T}-L_{0P}}{Q_0} \tag{7}$$

and

$$L_T=gL_{0T}+(1-g)\,\Delta i_{T_1}, \tag{8}$$

$$Q=g\cdot Q_0+(1-g)(i_1-i_5)-(i_7-i_5). \tag{9}$$

The value of specific enthalpy "i_7" is determined from energy balance of the high-temperature heat exchanger:

$$i_7=i_6+(1-g)(i_{2'}-i_8), \tag{10}$$

$$i_6=g\cdot i_5+(1-g)\,i_9, \tag{11}$$

$$i_9=i_8+\Delta i_{P_2}. \tag{12}$$

Hence it follows that:

$$i_7=g\cdot i_5+(1-g)(i_{2'}+\Delta i_{P_2}) \tag{13}$$

and also

$$Q=gQ_0+(1-g)(\Delta i_{T_1}-\Delta i_{P_2}). \tag{14}$$

Similarly, we arrive at expressions for work of the feed pumps:

$$L_{0P}=\Delta i_{P_1};$$

$$L_P=g\cdot\Delta i_{P_1}+(1-g)\,\Delta i_{P_2}=gL_{0P}+(1-g)\,\Delta i_{P_2} \tag{15}$$

and hence the expression of turbine efficiency:

$$\frac{\eta}{\eta_0}=\frac{1+\frac{1}{\eta_0}\cdot\frac{1-g}{g}\cdot\frac{\Delta i_{T_1}}{Q_0}\left(1-\frac{\Delta i_{P_2}}{\Delta i_{T_1}}\right)}{1+\frac{1-g}{g}\frac{\Delta i_{T_1}}{Q_0}\left(1-\frac{\Delta i_{P_2}}{\Delta i_{T_1}}\right)}. \tag{16}$$

The form of the efficiency equation:

$$\frac{\eta}{\eta_0}=\frac{1+\frac{1}{\eta_0}\cdot a}{1+a}$$

indicates the efficiency of the novel cycle being higher than that of the conventional one, the more the higher the value of a, what means the smaller g, the larger the ratio $\Delta i_{T_1}/Q_0$, and the smaller $\Delta i_{P_2}/\Delta i_{T_1}$. The factors $\Delta i_{T_1}/Q_0$ and $\Delta i_{P_2}/\Delta i_{T_1}$ result from the assumed cycle parameters $(p_1, t_1, p_{2'}, p_2)$ and considerably affect the attainable increase of efficiency. The influence of g is even more pronounced. Basically, the smaller the value of g the greater the attainable increase of efficiency.

From the energy balance at the high-temperature heat exchanger inlet (11) with help of eq. (12) follows:

$$g=\frac{i_8-i_6+\Delta i_{P_2}}{i_8-i_5+\Delta i_{P_2}}. \tag{17}$$

With the assistance of the $i(t)$ — diagram (Fig. 5) it is easy to determine both Δi_{P_2} and g as functions of t_8, with the terminal temperature difference $\Delta t_{HE}=t_8-t_6$ and the pump efficiency η_P as parameters.

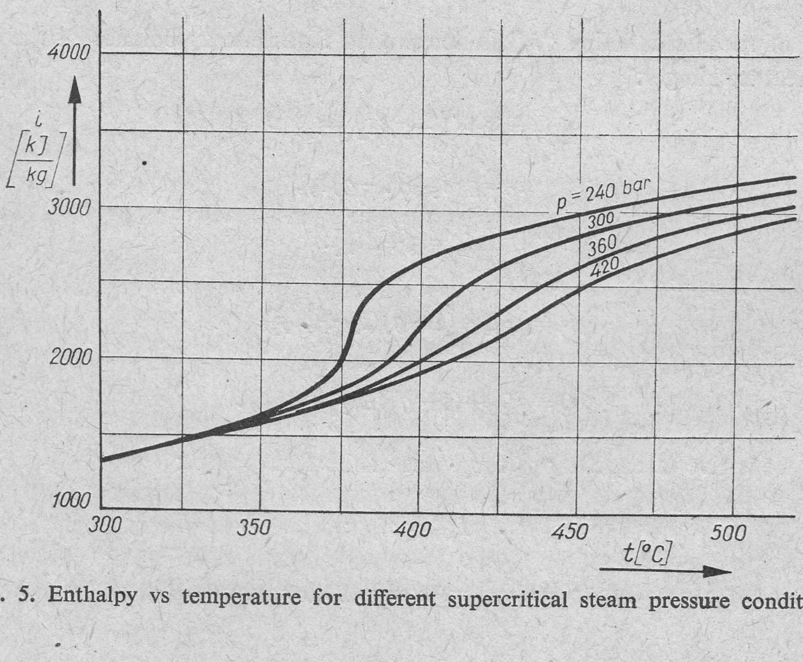


Fig. 5. Enthalpy vs temperature for different supercritical steam pressure conditions

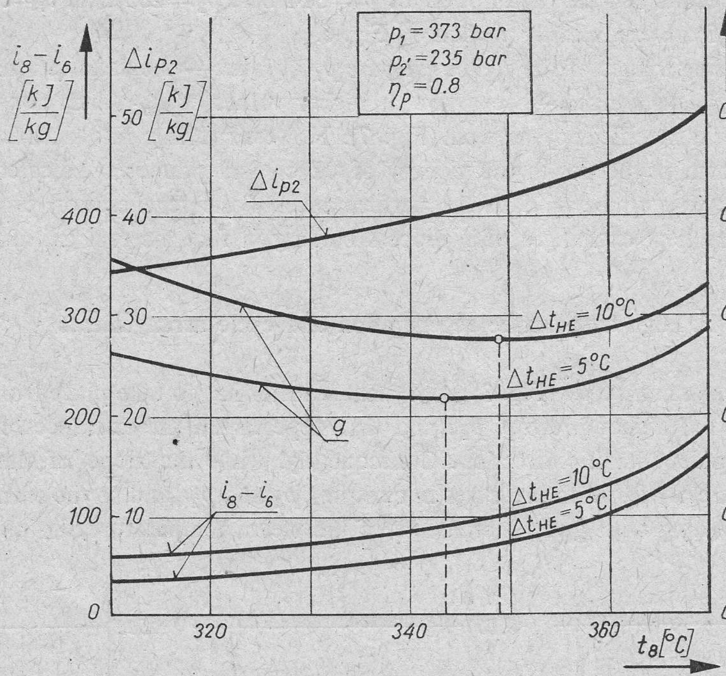


Fig. 6. Calculation diagram for pump work Δi_{P2} and g-ratio in function of t_8, for a set of assume conditions

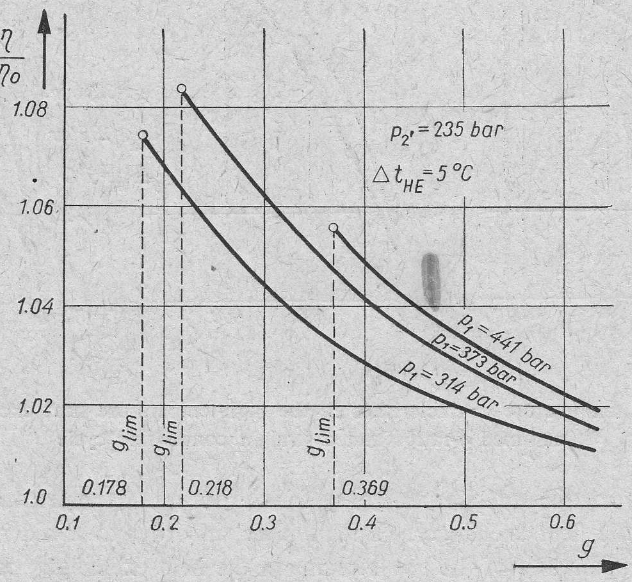


Fig. 7. Increase of efficiency as function of g-ratio with initial pressure as parameter

Numerical results for the assumption of $p_1=314$ bar, $p_{2'}=235$ bar, $t_1=565°C$, $t_5=$ $=265°C$, $\eta_P=0{,}8$ are exemplarily plotted on Fig. 6.

With increasing values of the steam pressure p_1 at inlet — the assumptions according $p_{2'}$ and η_P being unchanged — we arrive at higher values of g_{min} and, to some extent, the expected gain in efficiency as well (Fig. 7). However taking into account the large deviation of steam properties in the vicinity of the critical point, as indicated in Fig. 5, special attention has to be paid to the performance of the regeneration process in the high-temperature heat exchanger, this process barring in fact the real gain in efficiency.

5. High-temperature regeneration and cycle arrangements

According to assumptions (Fig. 3 and 4) heat transfer has to take place from the lower pressure $p_{2'}$ to the higher pressure p_1, and that depends on the existance of a positive temperature gradient in this direction. Consequently, while the slope of the $i(t)$-curve (Fig. 5) differs greatly for both pressures in question, p_1 and $p_{2'}$, and so the enthalpy increments corresponding with given increments of the steam temperature, the mass-flows of

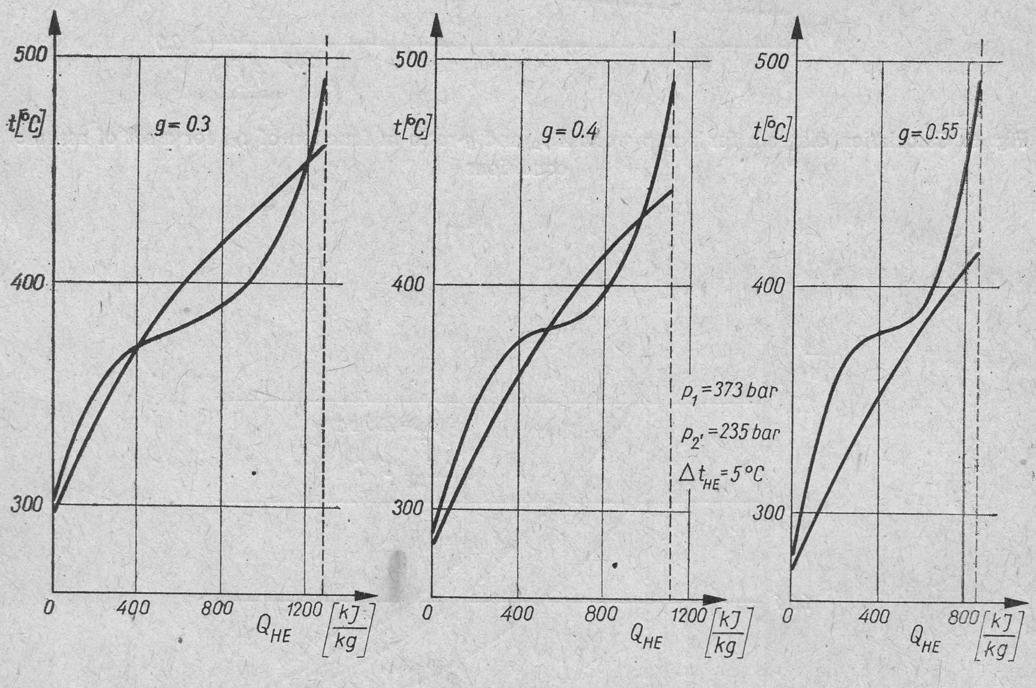


Fig. 8. Calculated temperature characteristics in the high-temperature heat exchanger for different assumed g-ratios and a straight condensing cycle

both streams have to be matched not only for the terminal temperature differences but at any point of the heat exchanger. On this basis, in order to avoid intersection of the temperature characteristics of the two streams in the heat exchanger, as shown spectacularly on Fig. 8, we are compelled to apply much higher values of g as predicted from Fig.

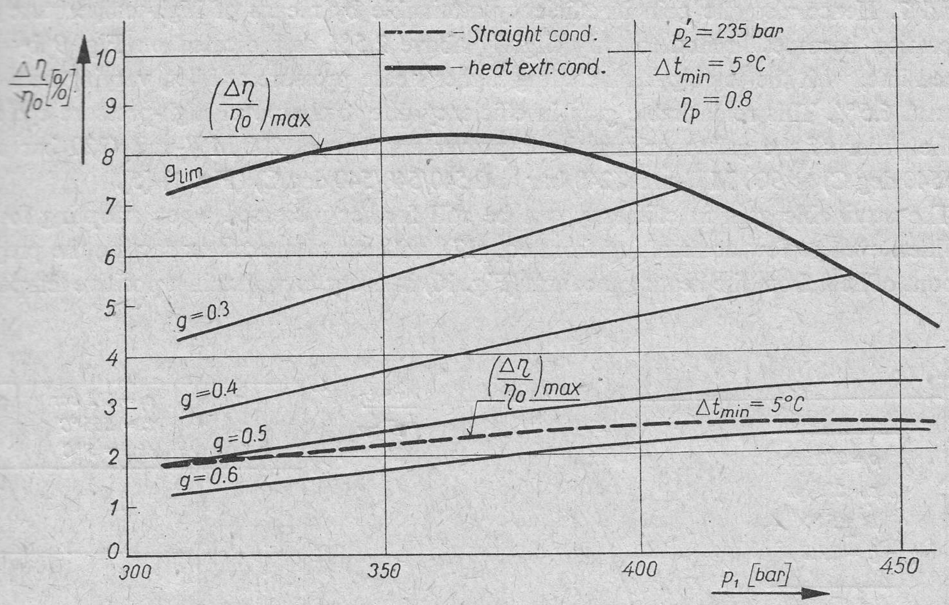


Fig. 9. Increase of efficiency as function of initial steam pressure with g-ratio as parameter

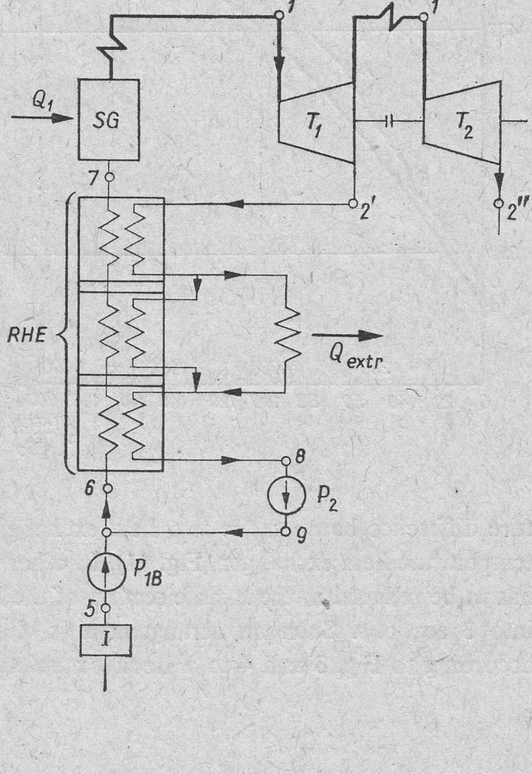
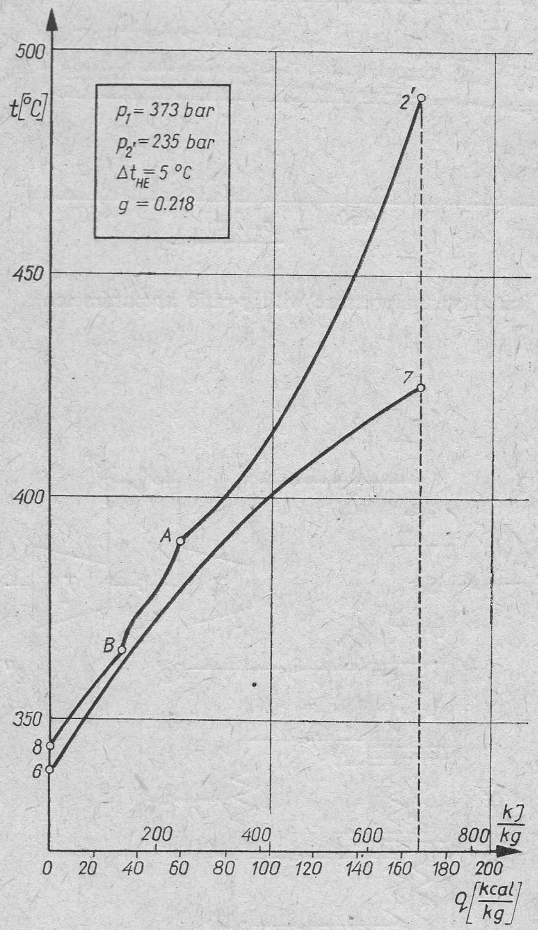


Fig. 10. The high-temperature heat exchanger in typical sectional arrangement for the power cycle with heat extraction

6 and 7. Hence also the gain in efficiency attainable by means of high-temperature regeneration becomes limited to only slightly above 2.5% as indicated on Fig. 9 by the dotted line, with the total gain, including higher steam pressure at inlet, varying between 5.5 and 6.5%. This remarkable gain in efficiency due to the novel high-pressure cycle is of the order of the gain marking the improvement from a 200 MW-unit (120 bar and 540/540 deg C) to 900 MW-unit (240 bar and 540/540/540 deg C) (Fig. 1) [5].

The attainable gain in efficiency can be still further increased when adopting lower g-values, even as low as resulting from eq. (17), with simultaneous extracting the proper amount of heat from the heating medium (Fig. 10) in order to maintain a positive tempera-



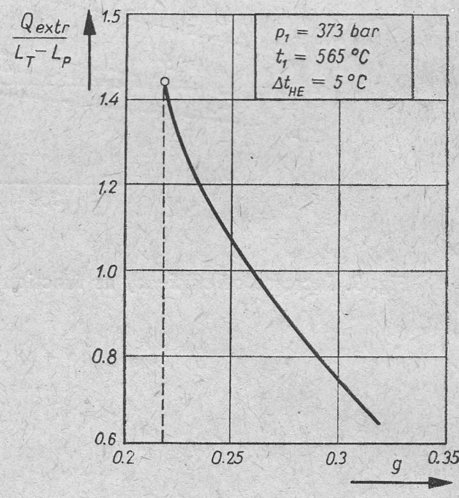


Fig. 12. The extracted heat to power ratio as function of g-ratio for a set of assumed conditions

Fig. 11. Calculated temperature characteristics in the high-temperature heat exchanger for a set of assumed conditions of a heat-extraction cycle

ture difference between the two heat exchanging mass streams at any point of the high-temperature heat exchanger (Fig. 11). In other words the mass flow of the heating medium has to be reduced in the middle section of the heat exchanger which becomes now divided into 3 sections. Such an arrangement is changing the straight condensing cycle according to Fig. 3 and 4 to a heat extraction cycle (Fig. 10).

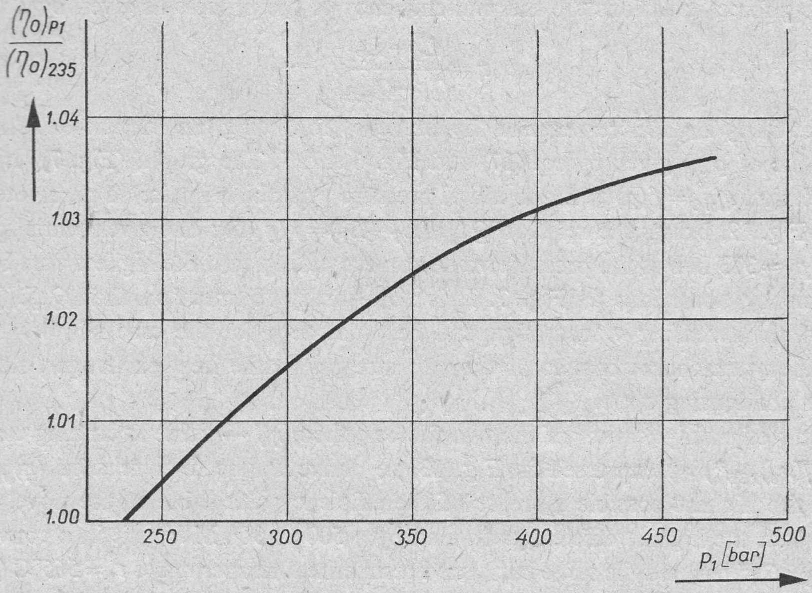


Fig. 13. Increase of basic cycle efficiency as function of initial pressure

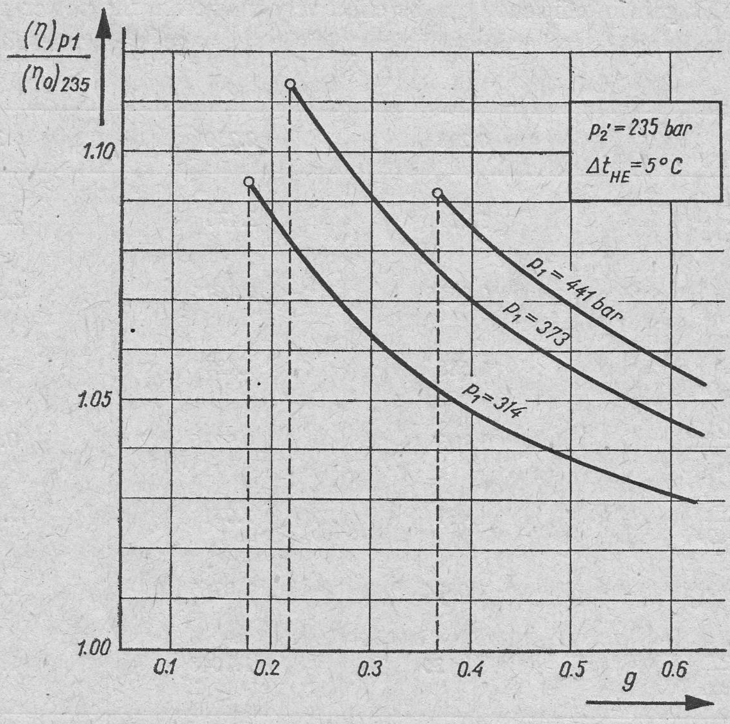


Fig. 14. All-over gain in efficiency attainable in the novel heat extraction cycle as function of *g*-ratio with initial pressure as parameter

Assuming for calculations the thermal efficiency of the process as given by

$$\eta = \frac{L_T - L_P}{Q_1 - Q_{extr}} \tag{18}$$

we arrive at fairly high increments of the efficiency following the adoption of the lowest possible value of g as given by eq. (17) and Fig. 6 and 7. The graph, (Fig. 7), is illustrating the relation $\eta/\eta_0 = f(g)$ with the initial pressure at turbine intake as parameter, while the graph, Fig. 12, brings the relation $Q_{extr}/(L_T - L_P) = f(g)$ for one given initial pressure, in fact for $p_1 = 373$ bar as the pressure value corresponding with the highest possible increase of efficiency according to Fig. 7. It is noteworthy to mention that $Q_{extr}/(L_T - L_P)$ decreases with increasing values of g, and becomes zero for the case of no interaction of the temperature-characteristics of the two streams in the heat exchanger, that is for the straight condensing cycle.

By means of multiplication of the two factors, the η/η_0 — value according to eq. (16) and $(\eta_0)_{p1}/(\eta_0)_{235}$*) as expressed in Fig. 13 we arrive at the final relation $(\eta)_{p1}/(\eta_0)_{235}$ describing the highest possible increase of the efficiency attainable in the novel cycle as compared with the highest efficiency attained by a 500 to 900 MW unit in a conventional cycle for $p_1 = 235$ bar and the same initial and resuperheat temperatures $t_1 = 565°C$ (Fig. 14).

Thus, for $p_1 = 373$ bar and $t_1 = 565/565/565$ deg C an all-over gain in efficiency of 11.35% can be achieved. Compared with the gain attainable in the straight condensing cycle-about 6% or slightly more – that means an essential step forward.

Of course, this gain in efficiency can be effectively utilized in the power plant but in the case of maintaining the rather high efficiency of the steam generator, despite the much

Table 1

g	0.218	0.250	0.300
$\left(\frac{G_1}{G_{01}}\right)_N$	3.929	3.470	2.970
$\left(\frac{G_2}{G_{02}}\right)_N$	0.8565	0.8675	0.8910
$\left(\frac{Q_1}{Q_{01}}\right)_N$	1.588	1.424	1.288
$\left(\frac{Q_1 - Q_{extr}}{Q_{01}}\right)_N$	0.9166	0.9235	0.9380
$\frac{Q_{extr}}{L_T - L_P}$	1.437	1.071	0.7499
$\left(\frac{\eta}{\eta_0}\right)_{p_1}$	1.0834	1.0765	1.0620
$\frac{(\eta)_{p_1}}{(\eta_0)_{235}}$	1.1135	1.1048	1.0918

*) The indices 235 and p_1 denote the initial steam pressures at turbine inlet: 235 bar and $p_1 > 235$ bar, respectively.

higher temperature of the feed water. Calculations undoubtedly confirm this possibility while simultaneously heat exchanging areas are shifted to the air preheater.

Along with increasing efficiency the steam rate at inlet — as compared with a conventional steam turbine of the same rating — considerably increases, enabling the efficient application of very high pressure supercritical steam. On the other hand, the steam rate in the "conventional" part of the cycle, in the last turbine stage and in the exhaust to the condenser decreases, as well as the amount of heat rejected in the condenser to the cooling water. That leads either to a reduction of the dimensions of final stages, particularly of the last stage itself, designed to operate at extremely severe conditions, or to a very desirable increase of the power rating attainable at given dimensions of the last stage. Finally, and that is a consequence of the decreased steam flow at turbine outlet, the amount of heat rejected to cooling water in the condenser diminishes, thus reducing size and cost of the condenser as well as the required amount of cooling water.

As a result of calculations made for the assumptions: $p_1=373$ bar, $t_1=565/565/565$ deg C, $\Delta T_{HE}=5$ deg C and heat extraction according to Fig. 10, following numerical factors have been achieved (Table 1). Herein index N denotes the value of the respective factor for identical power output.

6. Some economic considerations

It is understandable that while increasing the efficiency on the other hand a certain increase in power plant investment expenditures is to be taken into account. The turbine T_1 working in the range of supercritical steam pressures is rather small in dimensions, mostly single or double staged. On the contrary the high-temperature heat exchanger represents a new bulky element of the power plant, with an estimated area of 0.01 to 0.04 m²/kW, depending on cycle parameters. More severe working conditions refer to the feed pumps, particularly to pump P_2, because of high temperature of the working fluid at inlet, high volume-flow and high pressure-ratio. And finally, the steam generator of the through-flow type bears the features of a superheater, and in comparison with a conventional design for the same rating is characterized by a larger steam flow while a lower heat consumption as expressed by the factor $[Q_1-Q_{\mathrm{extr}})/Q_{01}]_N$.

When discussing this problem in more details, against the fuel savings related to increased efficiency some increments of capital costs have to be considered. The rule established in power generation practice balances 1 per cent in heat rate roughly against 10 per cent of the turbo-alternator price. This relation refers to a high load factor corresponding with about 7000 hours pro year full load while for lower load factors this relation may change even to 1 : 7.

From here it follows that the increase of efficiency of 11.35% would justify for base-load power plants more that doubling of the turbo-alternator investment costs while the price of other elements of the power plant remaining unchanged.

In the case under consideration besides the turbine itself and the condenser, the steam generator, the feed pumps and the high-pressure ducting are to be taken into account. According to Schröder's fundamental work on steam power plants of great output the over-all investment costs can be divided approximately into three parts:

buildings	23%
mechanical equipment	64%
electrical equipment	13%
	100%

Out of the mechanical equipment we can assume approximately the relations:

the steam generator	40%
the turbo-alternator-condenser unit	34%
the h.p. ducting	10%
div.	16%
	100%

Similarly within the turbo-alternator-condenser unit:

the steam turbine	40%
the electric generator	40%
the condenser	20%
	100%

and finally within the steam turbine itself:

the HP-cylinder	25%
the MP-cylinder	25%
$2\times$LP-cylinders	50%
	100%

The cost of the VHP-part (T_1) of the turbine can be estimated as approximately 75% of the HP-cylinder, that means $0.75\cdot0.25\cdot0.4=0.075$ (7.5%) of the turbo-alternator price.

The cost of the LP-part of the turbine, because of 14.5% lower steam flow, is expected to diminish at least at $14.5/2=7.25\%$. The saving is than about $0.0725\cdot0.5\cdot0.4=0.0145$ (1.45%) of the turbo-alternator price.

The price of the condenser can be assumed lower nearly at 14.5%, say for instance at 14%, what corresponds with a deduction of $0.14\cdot0.2=0.028$ (2.8%).

The high-temperature heat exchanger has an estimated area of 0.01 to 0.04 m^2/kW. That means for instance for a 500 MW unit 5000 to 20,000 m^2 with an additional investment expenditure in relation to the turbo-alternator price of 3.5 to 14%.

The feed pumps become evidently more expensive and require additional expenditures estimated to 5% of the turbo-alternator price.

The steam generator does not require more heat transfer surfaces. At any case in order to minimize the risk of a novel design some 10% higher capital costs of the generator are to be taken into account. That gives additionally nearly 12% of the turbo-alternator.

Finally, the high pressure ducting may become more expensive, up to 20%, that is up to about 6% of the turbo-alternator.

Summing up both additional expenses and savings we have to provide increased capital costs of the power plant as related to the turbine-alternator of about 29.7 — 40.2%, what is a rough estimation.

This increase of capital costs is equivalent to fuel costs savings in the order of 2.97 - 4.02% for a base load plant, and 4.2 - 5.7% for the average load factor. Subsequently the straight condensing cycle will prove economically advantageous at least in the case of high load factors while the heat extraction cycle seems to offer always remarkable profits in comparison with the conventional steam power cycle, even that for supercritical initial steam conditions and the highest efficiency ever attained in practice.

Higher power plant efficiency of the novel cycle and lower fuel consumption are accompanied by lower demands on cooling water and lower imperiling of the environment.

Finally, it seems to be necessary to point out that the realization of the new cycle does not require the application of any new elements which basically are not proven already in long years practice.

Promising possibilities seem to offer to this cycle also other working fluids as, e.g. carbon dioxide, the critical parameters of which are definitely lower than those for steam. A more thorough analysis of such a cycle is however beyond the scope of this paper.

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Nowy obieg termodynamiczny wysokiej sprawności dla ponadkrytycznych parametrów pary

Streszczenie

W pracy przedstawiono metody i środki zmierzające do podwyższenia sprawności termicznej turbin i siłowni. Wskazano na możliwość dalszego wzrostu sprawności przez podwyższenie ciśnienia początkowego pary w przypadku odpowiedniego wzrostu natężenia przepływu pary na wlocie. Z kolei, nawiązując do rozważań Carnota, wskazano na dalszą możliwość wzrostu sprawności przez podwyższenie średniej temperatury doprowadzania ciepła do obiegu. Uwzględniając obydwie te możliwości, przedstawiono koncepcję nowego obiegu termodynamicznego jako wynik nałożenia na siebie dwóch obiegów — jednego raczej konwencjonalnego obiegu na parametry nadkrytyczne i drugiego na te same parametry początkowe pary (rys. 3 i 4), związanego z pierwszym obiegiem termicznie poprzez wysokotemperaturowy wymiennik ciepła, lecz przebiegającego całkowicie w obszarze parametrów nadkrytycznych. Nowy obieg zanalizowano pod względem sprawnościowym (wzór (16)) i konstrukcyjnym. Wskazano na szczególną, kluczową rolę wysokotemperaturowego wymiennika ciepła, który

w wykonaniu tzw. sekcyjnym, ze zmiennym stosunkiem natężeń przepływu obu czynników wymieniających ciepło w wymienniku, pozwala na realizację w obiegu z upustem ciepła sprawności o ponad 11% wyższej od najwyższej sprawności osiągniętej dotąd w obiegu konwencjonalnym w identycznych granicach temperatur. Koncepcję nowego obiegu uzupełniono rozważaniami natury techniczno-ekonomicznej.

Новый термодинамический цикл для сверхкритических параметров пара, характеризующийся высокими значениями к.п.д.

Резюме

В работе представлены методы и средства, служащие повышению термического к.п.д. турбин и силовых установок. Указывается на возможность дальнейшего возрастания к.п.д. путем повышения начального давления пара в случае соответственного увеличения расхода пара на входе. Затем, учитывая рассуждения Карно, указывается на дальнейшую возможность повышения к.п.д. путем повышения средней температуры подвода тепла к циклу. Принимая во внимание обе эти возможности, представлена идея нового термодинамического цикла, возникающего вследствие наложения на себя двух циклов — одного скорее конвенционального цикла на сверхкритические параметры и другого на те же самые начальные параметры пара (рис. 3 и 4), связанного термически с первым циклом через высокотемпературный теплообменник, но полностью пробегающего в области сверхкритических параметров. Новый цикл проанализирован с точки зрения к.п.д. (формула (16)) и конструкции. Указывается на особую, узловую роль высокотемпературного теплообменника, в т. наз. секционном исполнении с изменяемым отношением интенсивности потоков обеих сред, обменивающих тепло в обменнике, позволяет осуществлять в цикле с отбором тепла значения к.п.д. на свыше 11% больше, чем значения к.п.д., достигаемые до сих пор в конвенциональном цикле и при таких же пределах температур. Идея нового цикла пополняется технико-экономическими рассуждениями.