P O L S K A   A K A D E M I A   N A U K
INSTYTUT MASZYN PRZEPŁYWOWYCH

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

TRANSACTIONS
OF THE INSTITUTE OF FLUID-FLOW MACHINERY

70-72

W A R S Z A W A – P O Z N A Ń 1 9 7 6
P A Ń S T W O W E   W Y D A W N I C T W O   N A U K O W E

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

*

THE TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

REDAKCJA – EDITORIAL OFFICE
Instytut Maszyn Przepływowych PAN
ul. Gen. Józefa Fiszera 14, 80-952 Gdańsk, skr. pocztowa 621, tel. 41-12-71

Printed in Poland

PAŃSTWOWE WYDAWNICTWO NAUKOWE – ODDZIAŁ W POZNANIU

Wydanie I. Nakład 630+90 egz. Ark. wyd. 63,25 Ark. druk. 49,25 Papier druk. sat. kl. V. 65 g., 70×100. Oddano do składania 4 VII 1975 r. Podpisano do druku w czerwcu 1976 r. Druk ukończono w czerwcu 1976 r. Zam. 528/118. H-16/245.
Cena zł 190,–

DRUKARNIA UNIWERSYTETU IM. ADAMA MICKIEWICZA W POZNANIU

III KONFERENCJA NAUKOWA

na temat

TURBINY PAROWE WIELKIEJ MOCY

Gdańsk, 24 - 27 września 1974 r.

*

IIIrd SCIENTIFIC CONFERENCE

on

STEAM TURBINES OF GREAT OUTPUT

Gdańsk, September 24 - 27, 1974

*

III НАУЧНАЯ КОНФЕРЕНЦИЯ

на тему

ПАРОВЫЕ ТУРБИНЫ БОЛЬШОЙ МОЩНОСТИ

Гданьск, 24 - 27 сентября 1974 г.

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH
1976 Zeszyt 70 - 72

LESZEK PARYS

Poland*

A Thermodynamic-Constructional Analysis of the ZAMECH Heating-Plant Turbines

1. Introduction

Economical advantages of generating electrical and heat energy in a system of inter-connected economy are well known today. There exists only a problem of optimal selection of design and operational parameters of a thermo-electric power plant at which the lowest costs of generation of the both types of energy are attained. It should be borne in mind that the selling price of a unit of electrical energy is several times higher than the price of a unit thermal energy. Hence, the tendency to obtain the highest economically substantiated electrical power (from a given amount of supplied energy) is justified.

One of the methods of increasing the electrical power output of thermo-electric turbo-sets is the application of the multistage thermal heating in place of the single-stage heating. As a results of lowering the pressure of the exhaust steam in the case of the multistage heating, the internal drop of enthalpy in a turbine rises by the value which corresponds to the difference of enthalpy of the exhaust steam in a system of the single-stage heating and the enthalpy of exhaust steam in a system with the multistage heating. In consequence the power rating of the turbine rises.

Thermal diagrams and steam expansion curves for turbines with the single-stage and multistage heating are presented in Fig. 1.

In order to accentuate the differences between the two types of turbines backpressure heating plant turbines will be taken into consideration.

The equipment and the expansion curve common for the both turbines are drawn with thick line while the additional equipment and the additional drop of enthalpy in the turbine with the multistage heating are drawn with a thin line.

The investment costs of the turbine, heat exchangers and of the power station as a whole rise simultaneously with the application of the multistage heating. The problem is therefore reduced to the techno-economical optimization of the turbine in a range of heat plant tappings and heat exchangers.

---

* ZAMECH – Mechanical Works, Elbląg.

The results of an analysis of the influence of the number of heating stages on the turbine rating, distribution of heating of mains water to the separate heat exchangers and the heat exchange area, and the influence of these parameters on the shaping of the turbine costs together with their optimization, will be discussed in this paper.

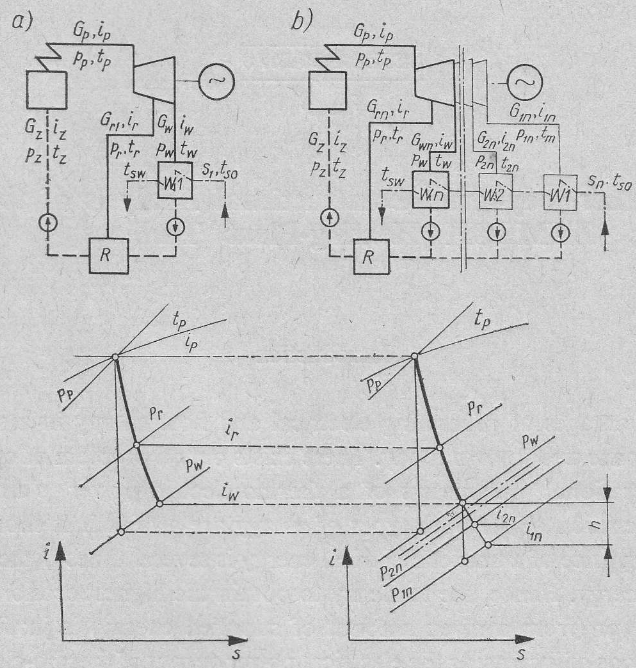


Fig. 1. Heat diagrams and steam expansion curves in heat plant backpressure turbines for the following systems: a) with single-stage heat plant heating, b) with two-stage heat plant heating

2. Thermodynamical analysis

The thermodynamical analysis was based on the comparison of power systems with the multistage heating with a system with the single-stage heating under the following assumptions.

2.1. Assumptions

– Equivalent energy rates are supplied to each of the systems in accordance with the equation:

$$E_p = N + Q = \text{const}, \tag{1}$$

where E_p – energy rate supplied to the turbine, N – internal power rating of the turbine, Q – heat flow rate.

– Equal initial live steam conditions (parameters): p_p – pressure, t_p – temperature.

– Equal internal efficiency of the turbines being compared.

– For the sake of simplicity, a single-stage feedheating which possesses the same parameters of blade steam: p_r – pressure, i_r – enthalpy.

– Identical parameters of feedwater: p_z – pressure, t_z – temperature, $G_z = G_p$ – mass flow rate of feedwater equal to the mass flow rate of live steam.

– Identical temperatures of the heating network water: t_{s0} – temperature of the heating network return water, t_{sw} – temperature of the heating network outlet water.

– Individual draining of the condensate from each heat plant heat exchanger was assumed and, due to its insignificant value, the power consumption of the condensate pumps was neglected in the power balance.

– A constant value assumed for the final temperature difference between the heating steam saturation temperature in the heat plant heat exchanger and the temperature of the mains water on the outlet from that heat exchanger, $\delta_t = \text{const}$.

– A turbine with the single-stage heating (a reference turbine) was assumed as a reference system with which the other heating systems were compared. Numerical examples were calculated on the basis of the reference turbine, which possesses the following technical characteristics:

$N_i = 55$ MW internal rating of the turbine,
$p_p/t_p = 130$ ata/535°C live steam conditions,
$G_p = 238$ t/h mass flow rate of the live steam,
$Q = 97.8$ Gcal/h heat plant energy flow rate,
$t_{s0} = 50$°C temperature of the network return water,
$t_{sw} = 86$°C temperature of the network outlet water.

2.2. System with two-stage heating

As a result of energy comparisons carried out the following relation for the turbine rating increment was obtained after changing over from a single-stage to a two-stage heat plant heating:

$$\Delta N_2 = G_p \alpha (1 - ab) \frac{h_0 \Delta i_{12} - \Delta i_{12}}{h_0 + a h_0 \Delta i_{12} - a \Delta i_{12}^2}, \tag{2}$$

where

$$a = \frac{1}{i_r - i'_w}, \tag{3}$$

$$b = i_r - i_w, \tag{4}$$

$$\alpha = \frac{G_w}{G_p}; \tag{5}$$

ΔN_2 – power increment for system with 2-stages of heating, G_p – mass flow rate of the live steam, G_w – mass flow rate of the exhaust steam for a turbine with the single-stage heating, i_w – enthalpy of the exhaust steam for a turbine with the single-stage heating, i'_w – enthalpy of the condensate for a turbine with the single-stage heating, h_0 – the maximum additional enthalpy drop in comparison with the single-stage heating, which is theoretically available, corresponding to the drop from a pressure p_w, resulting from the end temperature of the network water t_{sw}, to a pressure p_0 corresponding to the saturation temperature of the heating steam at zero heating of the network water in the first heat plant heat exchanger (see Fig. 2), Δi_{12} – drop of the steam enthalpy between the higher and the lower (exhaust steam) heat plant tapping.

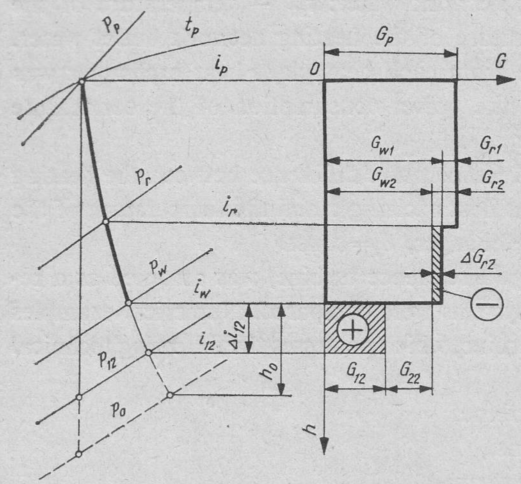


Fig. 2. Steam expansion curve in a turbine and graphic illustration of turbine ratings with single- and two-stage heating

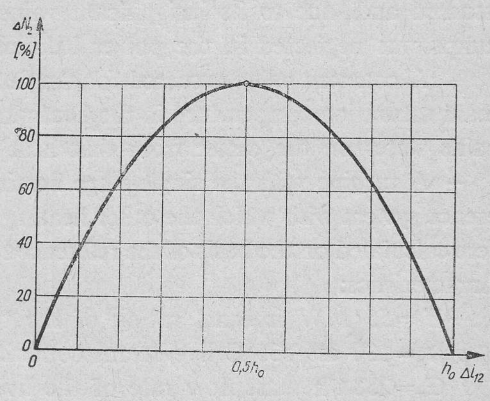


Fig. 3. Diagram of $\Delta N_2 = f(\Delta i_{12})$

The turbine steam expansion curve and the graphic presentation of the turbine rating with the single- and two-stage heating are shown in Fig. 2.

The diagram of the function $\Delta N_2 = f(\Delta i_{12})$ is shown in Fig. 3. From the analysis of the equation (2) it follows that the maximum increment of the power rating ΔN_2 takes place for a drop of the steam enthalpy between the heat plant tappings equal to

$$(\Delta i_{12})_{\text{opt}} = \tfrac{1}{2} h_0 .$$

This is turn causes the appearance of:

– equal heating of the network water in the both heat exchangers,

– equal steam mass flow rate to these heat exchangers.

The magnitude of the maximum power rating increment is:

$$(\Delta N_2)_{\max} = G_p \alpha h_0 \frac{1 - ab}{4 + a h_0} . \tag{6}$$

2.3. Three-stage heat plant heating system

For a turbine with a three-stage heating the following relationship of the power rating increment in respect to a turbine with the single-stage heating is obtained

$$\Delta N_3 = G_p \alpha (1 - ab) \frac{h_0 \Delta i_{13} + h_0 \Delta i_{23} - \Delta i_{13}^2 - \Delta i_{23}^2 - \Delta i_{13} \Delta i_{23}}{h_0 + a h_0 \Delta i_{13} + a h_0 \Delta i_{23} - a \Delta i_{13}^2 - a \Delta i_{23}^2 - a \Delta i_{13} \Delta i_{23}} , \tag{7}$$

where additionally: Δi_{13} – drop of the heating steam enthalpy between the higher and intermediate pressure heat plant tapping, Δi_{23} – drop of the heating steam enthalpy between the intermediate and the lowest pressure (outlet steam) heat plant tapping.

The turbine steam expansion curve and the graphic comparison of power ratings of turbines with the single- and three-stage heating are shown in Fig. 4.

In equation (7) two independent variables appear Δi_{13}, Δi_{23}, each of which can take on values from zero to h_0. The diagram of this function is a quadric which is presented graphically in Fig. 5. It follows from equation (7) that the maximum increment of the power

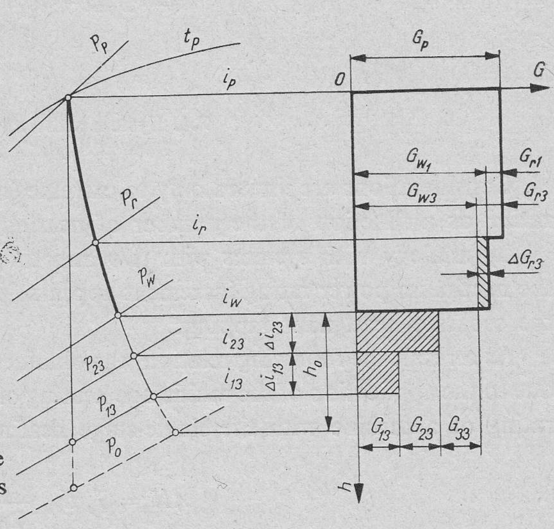


Fig. 4. Steam expansion curve in a turbine and graphic illustration of turbine ratings with single- and three-stage heating

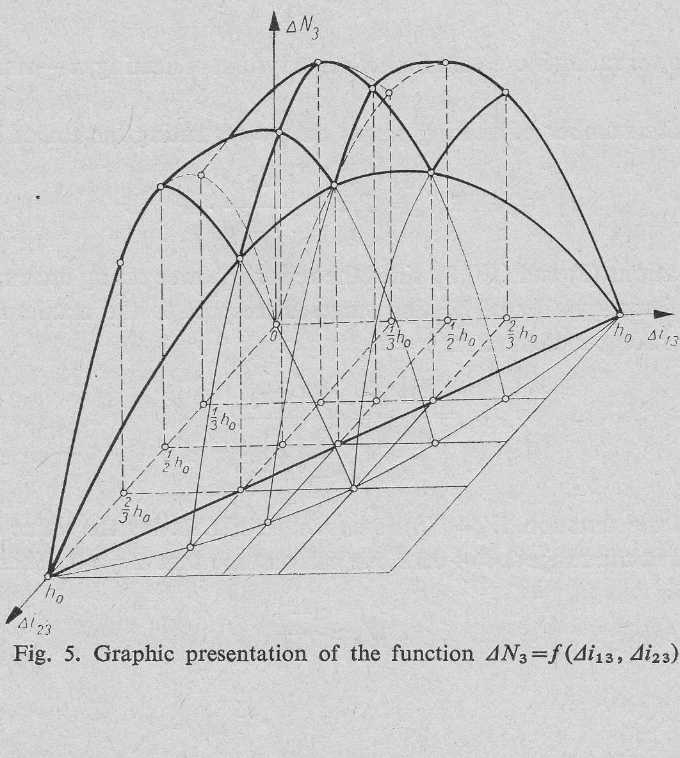


Fig. 5. Graphic presentation of the function $\Delta N_3 = f(\Delta i_{13}, \Delta i_{23})$

rating $(\Delta N_3)_{max}$ takes place for equal drops of the enthalpy

$$(\Delta i_{13})_{opt}=(\Delta i_{23})_{opt}=\tfrac{1}{3}h_0$$

which in turn causes the following:

- equal heating of the network water in all the heat plant heat exchangers,
- equal steam mass flow rate to each of the three heat exchangers.

The maximum magnitude of power rating increments is

$$(\Delta N_3)_{max}=G_p\alpha h_0\,\frac{1-ab}{3+ah_0}\,. \tag{8}$$

2.4. System with n stages of heating

Considerations are limited only to extreme (maximum) increment of the turbine power rating as a function of the number of heating stages.

By analogy with systems with two and three stages of heating it was assumed that the maximum power rating increment appears for enthalpy drops divided equally between the separate heat plant tappings.

As a result the following relation was obtained for the turbine power rating increment due to the application of the multistage heating of the network water in respect to the power rating of a turbine with the single-stage heatings:

$$\Delta N_n=G_p\alpha h_0\,\frac{1-ab}{\dfrac{2n}{n-1}+ah_0}\,, \tag{9}$$

where ΔN_n – power rating increment for a n-stage system of heating, n – number of heating stages.

For an infinite number of heating stages the power rating increment is

$$\Delta N_\infty=G_p\alpha h_0\,\frac{1-ab}{2+ah_0}\,. \tag{10}$$

By dividing equations (9) and (10) by sides the relative power rating increment is obtained for a turbine of any number of heating stages in respect to the maximum theoretically possible power rating increment at $n=\infty$

$$W_n=\frac{\Delta N_n}{\Delta N_\infty}=\frac{2+ah_0}{\dfrac{2n}{n-1}+ah_0}\,. \tag{11}$$

The diagram of the function $W_n=f(n)$ is shown in Fig. 6. As the value of the product ah_0 is contained in the range $0.06\div0.12$ one can assume that $ah_0\approx0$. Then:

$$W_n\approx\frac{n-1}{n}$$

and for

$$n=2 \qquad n=3 \qquad n=4 \qquad n=5$$

$$W_2 \approx 0.5 \qquad W_3 \approx 0.66 \qquad W_4 \approx 0.75 \qquad W_5 \approx 0.80$$

As it follows from Fig. 6 the greatest power rating increment of the turbine is obtained on changing over from the single-stage to a two-stage heat plant heating. Further increase of the number of stages gives decreasing increments of power rating.

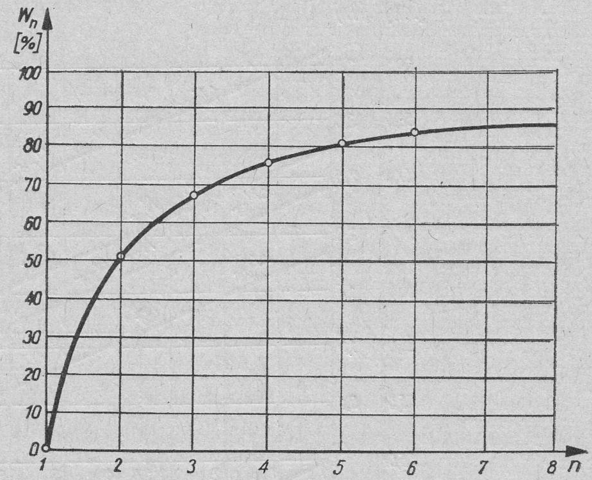


Fig. 6. Specific increments of turbine rating depending on the number of stages of heat plant heating

2.5. Thermodynamic analysis with variability δ_t considered

The relation for the power rating increment $\Delta N_n = f(n)$ was derived on the assumption of a constant value of δ_t in all the heat plant heat exchangers of the systems taken into consideration. This allows for indication of the differences of the power ratings of turbines operating in systems with different degrees of heat plant heating.

For a definite heat plant turbine a change of the value δ_t in heat exchangers causes in principle only a change of turbine power rating of the magnitude of:

$$\Delta N_\delta = \sum G_{\text{in}} (\delta_t - \delta_t') c \eta_i , \tag{13}$$

where ΣG_{in} – sum of steam mass flow rates to heat plant heat exchangers, c – mean specific heat of steam, η_i – turbine efficiency within the range of heat plant tappings.

Besides, the value δ_t has a basic influence on the heat exchange area in heat plant heat exchangers, and therefore on the cost of the heat exchangers.

The function of investment costs variation depending on n and δ_t is in turn associated with the type of the turbine and with its size, and for that reason also a further analysis was carried out on a numerical example for backpressure heat plant turbines derived from the reference turbine, the performance data of which are given in the assumptions (see section 2.1).

The value δ_t was therefore introduced into the thermodynamic analysis as an independent variable varying in the range of $\delta_t = 1 \div 10°C$ and the variability of n was limited to the values of $n = 1 \div 4$ in consideration of the technical reality.

The results of this analysis and therefore the turbine power rating, its power rating increment and the value of the heat plant energy flow rate as a function of the number n of heating stages and also the final value of the temperature difference δ_t are presented in Fig. 7.

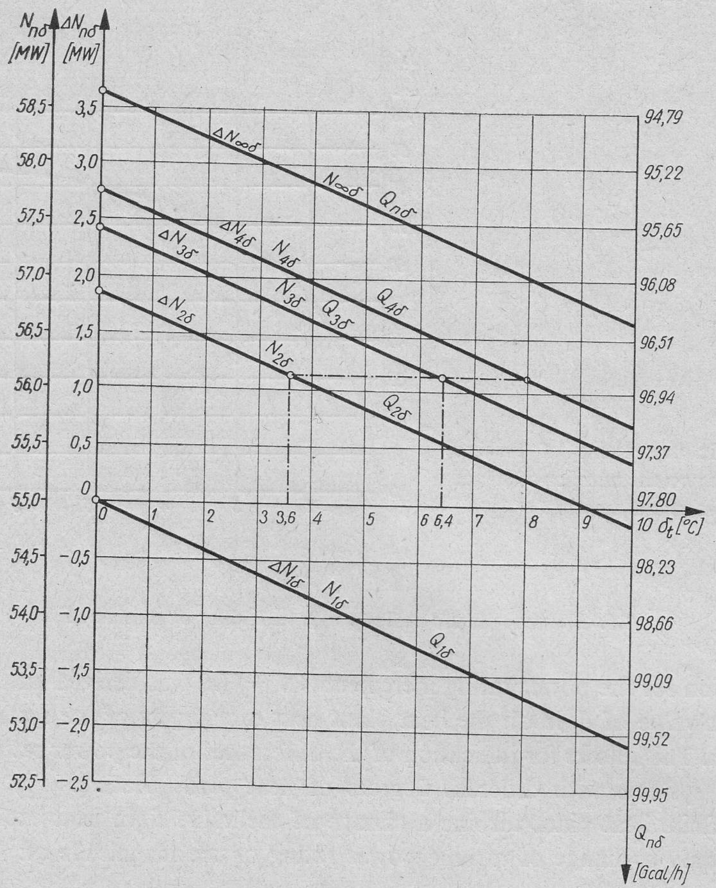


Fig. 7. Diagrams $\Delta N_{n\delta} = f(n, \delta_t)$, $N_{n\delta} = f(n, \delta_t)$, $Q_{n\delta} = f(n, \delta_t)$

It is evident from this figure that:

– the turbine power rating drops with the rise of the final temperature difference δ_t in the heat plant heat exchangers for a system of a determined number of heating stages, while the heat plant flow rate of the energy rises,

– the turbine power rating increment on changing over from a single-stage to a multi-stage system is basically constant, independent of the value of δ_t,

– equal end effects – equal pairs of two parameters: $N_{n\delta}$ – power rating of turbine, $Q_{n\delta}$ – heat plant energy flow rate can be achieved in different design solutions (heat systems).

For instance, the same turbine power ratings and heat plant energy flow rates are achieved in the following systems (at the same value of energy supplied to the turbine in accordance with the assumptions given above):

- with the two-stage heating at $\delta_t=3.6°C$,
- with the three-stage heating at $\delta_t=6.4°C$,
- with the four-stage heating at $\delta_t=8°C$.

Selection of the appropriate design parameters n, δ_t requires carrying out techno-economical calculations.

3. Techno-economical optimization

3.1. The principle of optimization

Technical-economical optimization was carried out by means of the so called heat plant investment effectiveness indicator in accordance with the principles of the Provisional Departmental Instructions for Investigation of Effectiveness of Power Plant Investments with the difference that the common factor for all the versions taken into consideration was the equal value of the energy supplied and not the final effect (electrical power and heat plant energy) as it is assumed for thermal-electric power station calculations.

The indicator of the effectiveness of a thermal plant investment, expressing in principle conventional unit cost of thermal energy generation is defined as follows:

$$E_c=\frac{A-B}{C}, \tag{14}$$

where E_c [zł/Gcal] – indicator of thermal effectiveness, A [zł/year] – sum of annual fixed costs and variable costs in which the following are contained: amortization of investment capital during the period of construction, fuel costs, costs of labour, current repairs and general overhauls together with the general shop overhaul costs, B [zł/year] – product of the annual value of electric power produced and the indicator of the effectiveness of the substitution power plant investments, which in principle corresponds to the value of the electrical power produced during the year, C (Gcal/year) – value of the annual production of electrical energy.

3.2. Change of fixed and variable costs of the thermal-electric power station depending on n and δ_t

The shaping of the investment and operational costs was considered next as a function of n and δ_t regarded as the basic factors having a decisive influence on the value of the term A in formula (14).

The investment costs were treated as a sum of costs of the turbine itself, the heat exchangers, the engine room excluding the turbine and the heat exchangers, and the rest of the thermal-electric power station costs.

It may be said when comparing the components of the thermal-electric power station investment costs, that the greatest influence on the variability of these costs, depending on the number of heating stages and the final difference of temperatures in the heat plant heat exchangers, is that of the costs of the heat plant heat exchangers which, in this case, amount from $3 \cdot 10^6$ zł at $n=1$ and $\delta_t=10°C$ to $18 \cdot 10^6$ zł at $n=4$ and $\delta_t=1°C$ (see diagram Fig. 8). That means that they undergo a sixfold change, constituting from 15 to 70% of the cost of the turbine itself.

The cost of the heat exchangers is comparatively easy to estimate. This considerable range of costs (equivalent to the area) results from the fact that with the increase of the number of heating stages and the decrease of the value δ_t, the value of the mean logarithmic difference of temperatures falls and the total heat exchange surface of the heat plant heat exchangers rises

$$\Delta t_{\ln} = \frac{\Delta t_s}{2.3 \lg \left(1 + \dfrac{\Delta t_s}{n \cdot \delta_t}\right) n}, \tag{15}$$

where $\Delta t_{\ln}$ – mean logarithmic difference of temperatures for heat plant heat exchangers with n-stages of heating, $\Delta t_s = t_{sw} - t_{s0}$ – total heating of the network water in the heat plant heat exchangers.

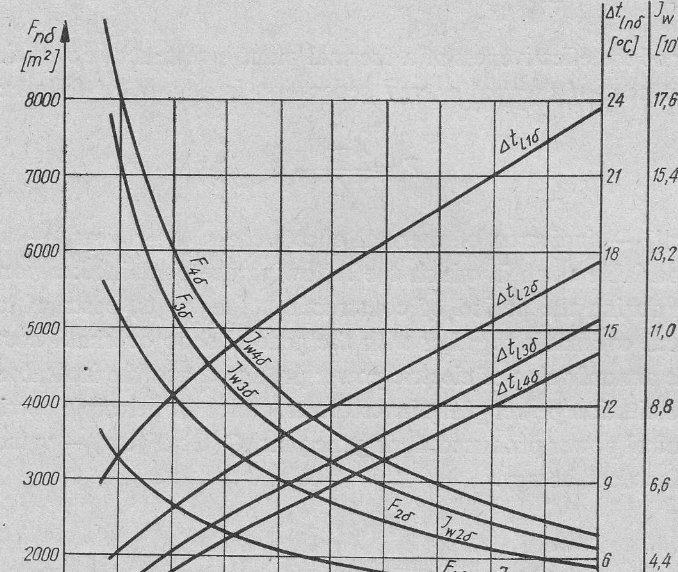


Fig. 8. The exchange area $F_{n\delta}$, the logaritmic temperature difference $\Delta t_{\ln\delta}$ and the cost of the heat exchangers J_w as a function of the number of heating stages and the value of the final temperature difference

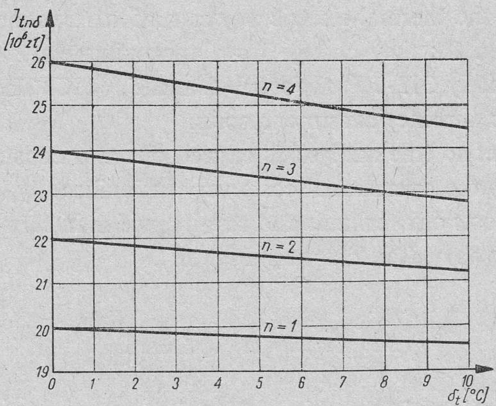
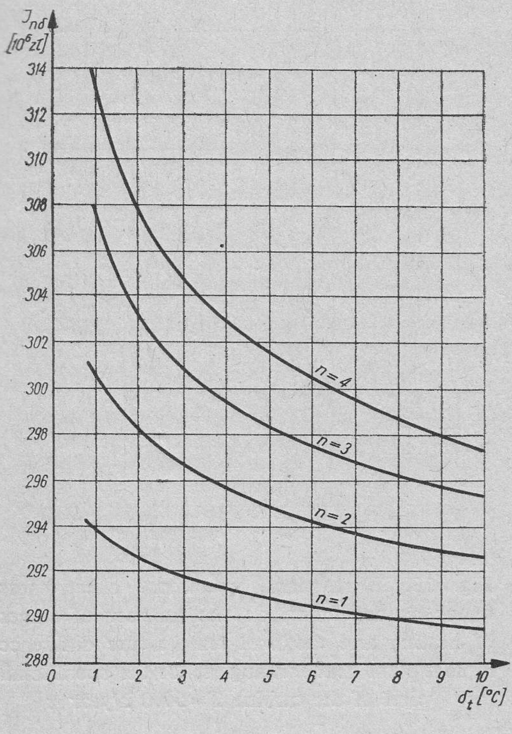


Fig. 9. Investment costs of the turbine as depending on the number of stages of heating and the final temperature difference

The other factor, the variability of which is considerably depending on n and δ_t, are the turbine investment costs (see diagram Fig. 9). In this example the magnitude of the turbine costs fluctuates from $20 \cdot 10^6$ zł for $n=1$ and $\delta_t=10$°C to $26 \cdot 10^6$ zł for $n=4$ and $\delta_t=1$°C. Therefore the variation of the costs of the turbine is about 30% of the basic turbine costs.

The remaining investment costs either vary insignificantly (engine room, excluding the turbine and the heat plant heat exchangers) and may be ignored, or take on a constant value (boiler room).



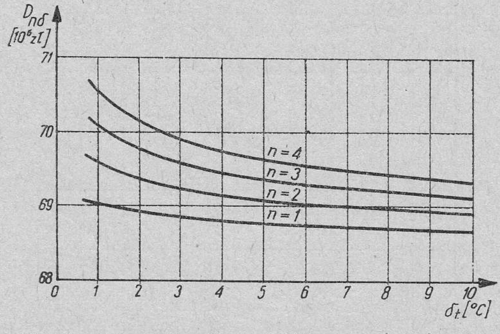


Fig. 11. Operating costs of thermo-electric power station as depending on the number of stages of heating and the final temperature difference

Fig. 10. Thermo-electric power station investment costs as depending on the number of stages of heating and the final temperature difference

The total values of the thermal-electric power station investment costs are illustrated diagrammatically in Fig. 10; they change from approximately $290 \cdot 10^6$ zł, for $n=1$ and $\delta_t=10°C$ to approximately $313 \cdot 10^6$ zł for $n=4$ and $\delta_t=1°C$. They change therefore by about 8% of the basic turbine investment costs.

With the problem set so (the analysis is carried out at a constant value of the energy supplied to the turbine) the operational costs vary insignificantly, as the costs of fuel are constant and only the costs of overhauls and repairs, which in turn are proportional to the total investment costs, vary (see diagram Fig. 11).

3.3. Results of optimization

The duration of operation of the turbine during the whole year, designated by the symbol T was introduced as an independent variable in order to carry out the optimization of n and δ_t with the use of formula (14).

The results of calculations for two values of T are presented in Fig. 12, for $T=3000$ h/year and in Fig. 13 for $T=5400$ h/year. Taking the results of optimization into consideration it may be said that the selection of the optimum number of heating stages should be carried out simultaneously with the selection of the optimum value of the final temperature

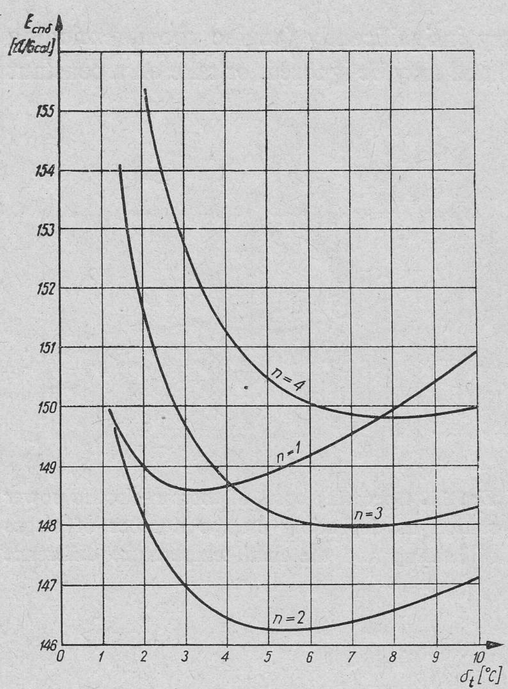


Fig. 12. Heat plant investment effectiveness factor as depending on the number of stages of heating and the final temperature difference in heat plant heat exchangers, for the operational time of the turbine $T=3000$ h/year

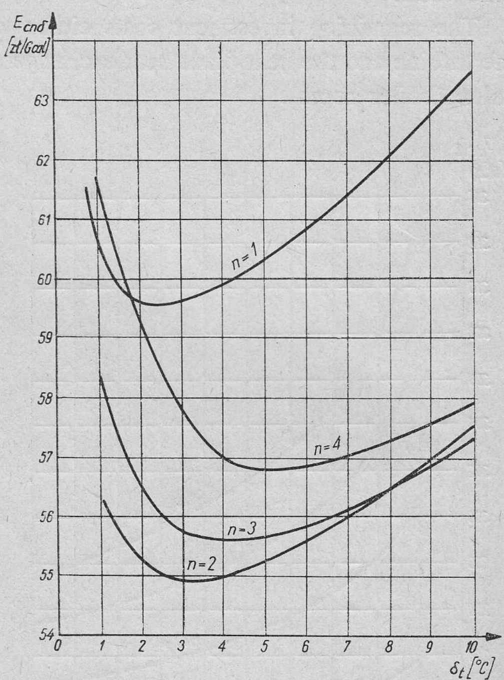


Fig. 13. Heat plant investment effectiveness factor as depending on the number of stages of heating and the final temperature difference in heat plant heat exchangers, for the operational time of the turbine $T=5400$ h/year

difference in the heat exchangers. For a small time period T there are no definite advantages from use of the multistage heating. Minimum values of E_c are attained at higher values of δ_t, with the influence of the value of δ_t on E_c being relatively small. It is different for large periods of time T, where the advantages from using the multistage heating are considerable, minimum E_c values are attained at lower values of δ_t and the influence of the value of δ_t on E_c is signicant.

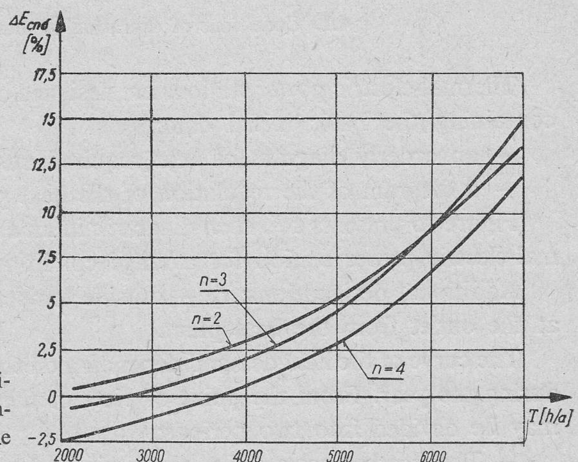


Fig. 14. Savings on the indicator of the effectiveness of the thermal plant investment as depending on the duration of operation of the turbine during the year

The selection of an optimum number of heating stages may be carried out on the basis of the diagram (Fig. 14), which presents savings on the indicator of the effectiveness of the thermal plant investment depending on the operation time of the turbine during the year i.e. the percentage decrease of the unit cost of generation of thermal energy expressed by the formula:

$$\Delta E_{cn} = \frac{E_{c1} - E_{cn}}{E_{c1}}, \tag{16}$$

where E_{cn} – savings of the thermal investment indicator, E_{c1} – minimum value of E_c for the single-stage heating, E_{cn} – minimum value of E_c for n-stages of heating.

Analyzing the diagram $\Delta E_{cn} = f(T)$ and bearing in mind a certain margin of uncertainty as to the economic calculations it may be stated that for short-term operation of turbines up to about 3000 h/year, the use of the single-stage heating is appropriate. For operational times of about 5000 h/year, which corresponds to the duration of the heating season, usage of a two-stage heating system is deemed indisputable, all the more so that values of ΔE_{cn} reach up to about $\sim 6\%$. Application of three-stage heating seems to be purposeful only upon year-round operation of the turbine.

These considerations were carried out for the case when during the period of T the turbine operates on constant loading.

The turbine load changes actually due to variations of the network return water temperature and variations of thermal and electric energy requirements; this having negative influence on the turbine efficiency in the area of heat plant tappings. Detailed economic calculations should be carried out taking into consideration the actual yearly loading curve. It may, however, be estimated that such a more accurate optimization must lead rather to a smaller number of heating stages.

4. Thermal-electric power station turbines of ZAMECH production

4.1. Conditions of operation of thermal-electric power stations

Thermal-electric power stations are designed on the basis of the two-following diagrams containing the basic initial data:

– an orderly diagram of heat required annually,

– a diagram of the regulation of the heat plant network.

Figure 15 presents two such diagrams in an arrangement characteristic for a large thermal-electric power station for average conditions in Poland.

Additional designations: t_z – outside temperature, t_{se} – network water temperature at the outlet from the heat plant.

The curve of the heat demand depending on the outside temperature $Q=f(t_z)$ is variable during the year. From the point of view of the thermal requirements the calendar year may be devided into two periods:

– The heating season, i.e. the period when the outside temperature is lower than +12°C. The heat demand in this period is inversely proportional to the outside temperatures, which is clearly shown on the diagram of regulation of the thermal network.

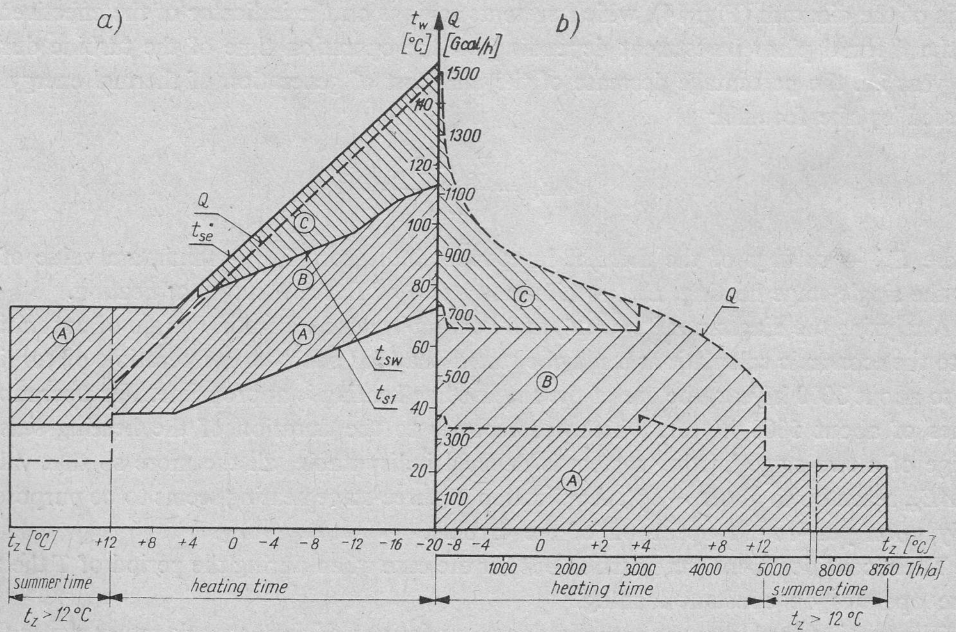


Fig. 15. a) An orderly diagram of annual heat requeirement. b) Diagram of heat plant network regulation

– The summer season, i.e. the period when the outside temperature is higher than +12°C. During this period the heat demand is practically constant and amounts to 10÷15% of the maximum demand of the winter period.

The annual heat demand may be divided into three areas:

– the basic demand, almost constant all year round,

– the seasonal demand, only during the heating season, variable, depending on the outside temperature,

– the peak demand, short-lived during the year, extremely variable, depending on the outside temperature.

For economically optimal fulfilment of the heat demand, each of these areas requires a different type of heat plant turbine.

4.2. The types and basic properties of heat-plant turbines of ZAMECH production

Basic heat-plant turbines

They are designed for constant operation at almost constant load, for delivery of thermal energy corresponding to the area *A* in the diagram Fig. 15. By design they are backpressure turbines with uncontrolled backpressure, with two- or three-stage exhaust to heat plant heat exchangers.

The thermal system of these turbines functions on, the principle of autoregulation, that means it possesses no organs regulating the distribution of steam.

Zamech produces a whole range of this type of turbines, beginning with ratings of 25 MW way up to 110 MW.

Two basic sizes of these turbines are being mainly installed in Poland.

1. Thermal-electric power station turbines of ratings 50÷60 MW produced in two versions as backpressure and as extraction-backpressure turbines with controlled steam extraction from receivers between the HP and IP parts at the pressure levels of 10÷12 ata. These are impulse turbines of the disc (wheel) type, two-cylinder of double-shell design in the HP part, live steam conditions 130 ata/535°C.

Zamech produced the first of this type of turbines, of the symbol 13P55-0, in 1969. It is running at present in the thermal-electric power station Łódź III. The longitudinal section of this turbine is shown in Fig. 16. The first units of these turbines were made in a three-stage heating arrangement with $\delta_t = 6.4$°C, assumed in the heat plant heat exchangers. At present these turbines, due to dubious advantages resulting from three-stage heating, are being produced with two-stage heating, with $\delta_t = 3.6$°C assumed, and power rating and heat flow rate remaining unchanged.

2. Backpressure heat-plant turbines of power rating 100÷115 MW, symbol 13P110-0, are also built in two versions as backpressure and as extraction-backpressure ones with a controlled steam bleed from HP and IP receiver at the pressure level of 10÷12 ata. The longitudinal section of this turbine in the backpressure version is shown in Fig. 17. This is a two-cylinder turbine of a double shell design in the HP part with a single flow inlet to the low

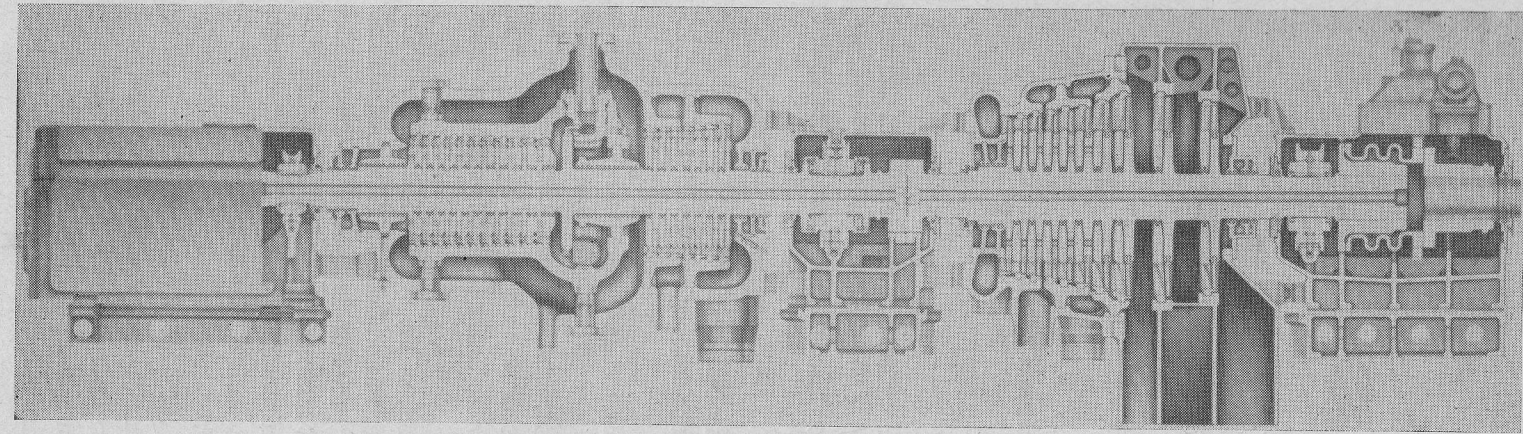


Fig. 16. The longitudinal section of the 13 P55-0 turbine

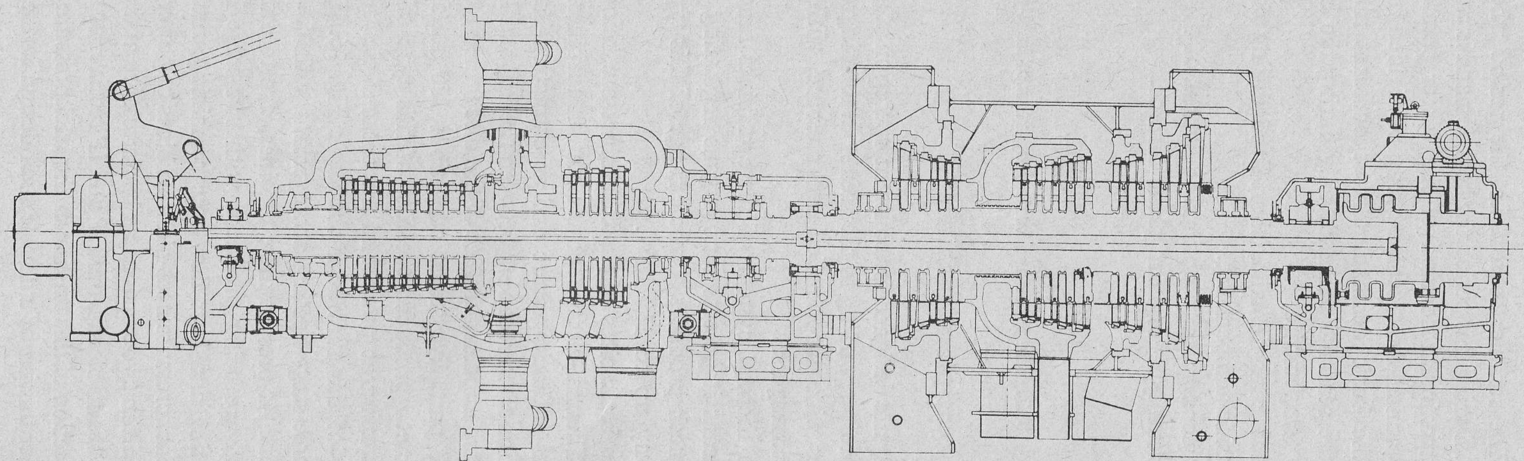


Fig. 17. The longitudinal section of the 13 P110-0 turbine

pressure part and a double flow outlet to two heat plant heat exchangers with two-stage heating. Live steam conditions 130 ata/535°C.

Such a solution of the low pressure part gives thermodynamic advantages in the variable conditions of turbine operations and advantageous possibilities of planning arrangements of heat plant exchanger systems and steam piping under the turbine.

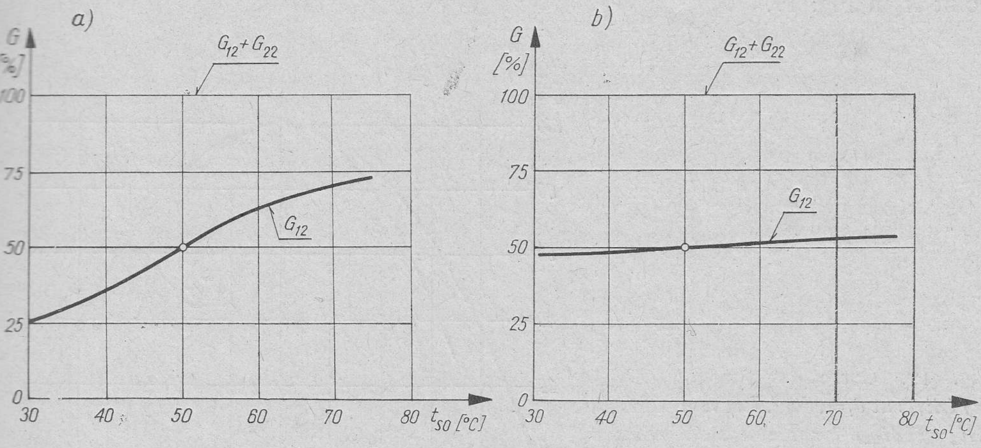


Fig. 18. Steam flow rate to heat plant heat exchangers as depending on the temperature of network return water

The thermodynamic advantages may be shown by comparing the operation of this turbine with a turbine to the design arrangement of the turbine 13P55-0 in variable working conditions.

Diagrams of the steam flow rate to the separate heat plant heat exchangers depending on the temperature of the network return water at the constant live steam flow rate for backpressure heat plant turbines in series arrangements of heat plant tappings (according to the design arrangement of the turbine 13P55-0) and shunt arrangements as in the turbine 13P110-0 are shown in Fig. 18.

It follows from Fig. 18 that the steam flow rate in the turbine 13P110-0 to the separate heat exchangers remains almost constant, independently of the variation of the return network water temperature t_{s0}. For a turbine with a series arrangement of heat plant tappings at low temperatures of the network water, the heat plant heat exchanger of lower pressure is relieved while the load on the higher pressure one is raised. At high network return water temperatures the situation is reversed; the load on the exchanger of higher pressure is raised and on the lower pressure exchanger it is decreased.

The thermodynamic analysis carried out in section 2 revealed that, for the simplifications assumed, the following relation exists between the enthalpy drop of steam in the turbine between the heat plant tappings, the branchings of steam to the separate exchangers and the network water heating in these exchangers.

$$\frac{\Delta i_{12}}{h_0} \approx \frac{G_{22}}{G_{12}+G_{22}} \approx \frac{t_{sw}-t_{s1}}{t_{sw}-t_{s0}}, \tag{17}$$

where G_{12} – steam flow rate to higher pressure heat exchanger, G_{22} – steam flow rate to lower pressure heat exchanger, t_{s1} – network water temperature beyond the lower pressure heat exchanger.

It follows from the analysis of the diagram of Fig. 18 and the relation (17) that in the turbine 13P110-0, independently of the temperature of the network return water, maximum increments of the power rating resulting from a two-stage heat plant heating are achieved, point A in Fig. 19.

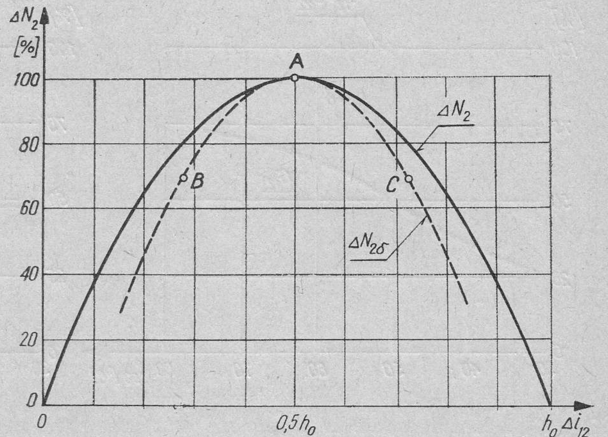


Fig. 19. Curve of function $\Delta N_2 = f(\Delta i_{12})$ at $\delta_t = \text{const}$. Curve of function $\Delta N_2 = f(\Delta i_{12})$ for real system when δ_t is variable

For a turbine with a series arrangement of heat plant tappings the maximum increment of the power rating (point A in Fig. 19) is attained only on equal branchings (distribution) of steam i.e. at the nominal temperature of the network water, while for low temperatures of the network water the increments of power rating are about 30% lower, point B, and similarly for high temperature, point C, of the diagram Fig. 19. Points A, B, C lie on the turbine power rating increment curve $\Delta N_{25} = f(\Delta i_{12})$ resulting from the introduction of the two-stage heating for a real system in which the variability of values in heat plant heat exchangers depending on the heat flow rate supplied to the exchanger is taken into consideration.

Seasonal heat plant turbines are designed mainly for satisfying the requirements for heat which correspond to the area B on the diagram in Fig. 15.

According to the current opinion in Poland on the subject the extraction-condensing heat plant turbines operating in the heating season with a variable consumption of the heat and during summer as condensing, electric power turbines, are deemed to be most appropriate.

In association with this high efficiency (corresponding to the efficiencies of electric power station turbines equivalent with regard to the power rating) is required from these turbines. They are, therefore, generally interstage reheat turbines. For these purposes, ZAMECH makes extraction-condensing heat-plant turbines with two-stage heating in two versions, the turbine designated by 180UK135-0, of power rating 135 MW for the live steam conditions of 180 ata/535/535°C and the turbine designated by 13UK125-0, of 125 MW power rating for the live steam conditions of 130 ata/535/535°C. Both versions possess the same low pressure part.

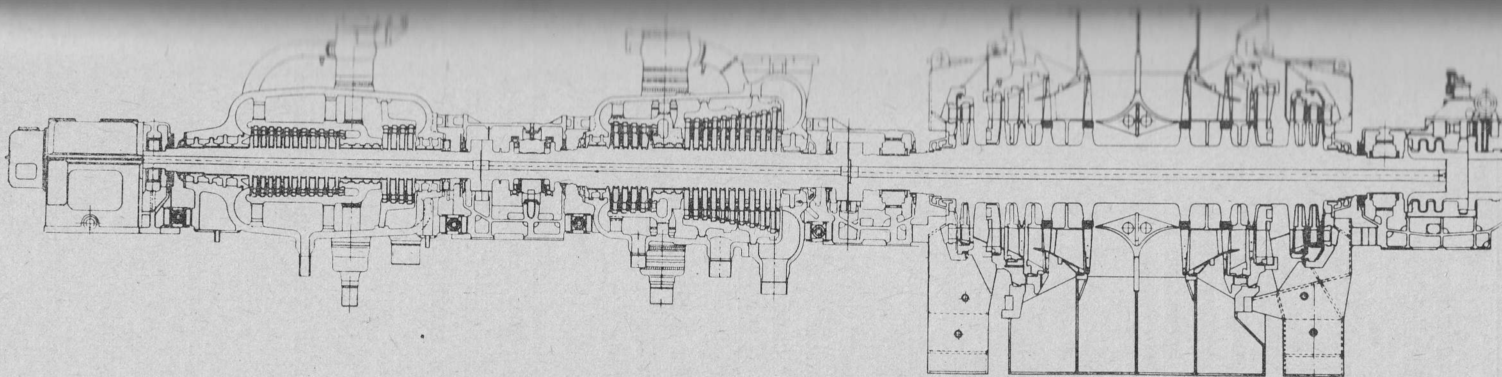


Fig. 20. The longitudinal section of the 18UK135-0 turbine

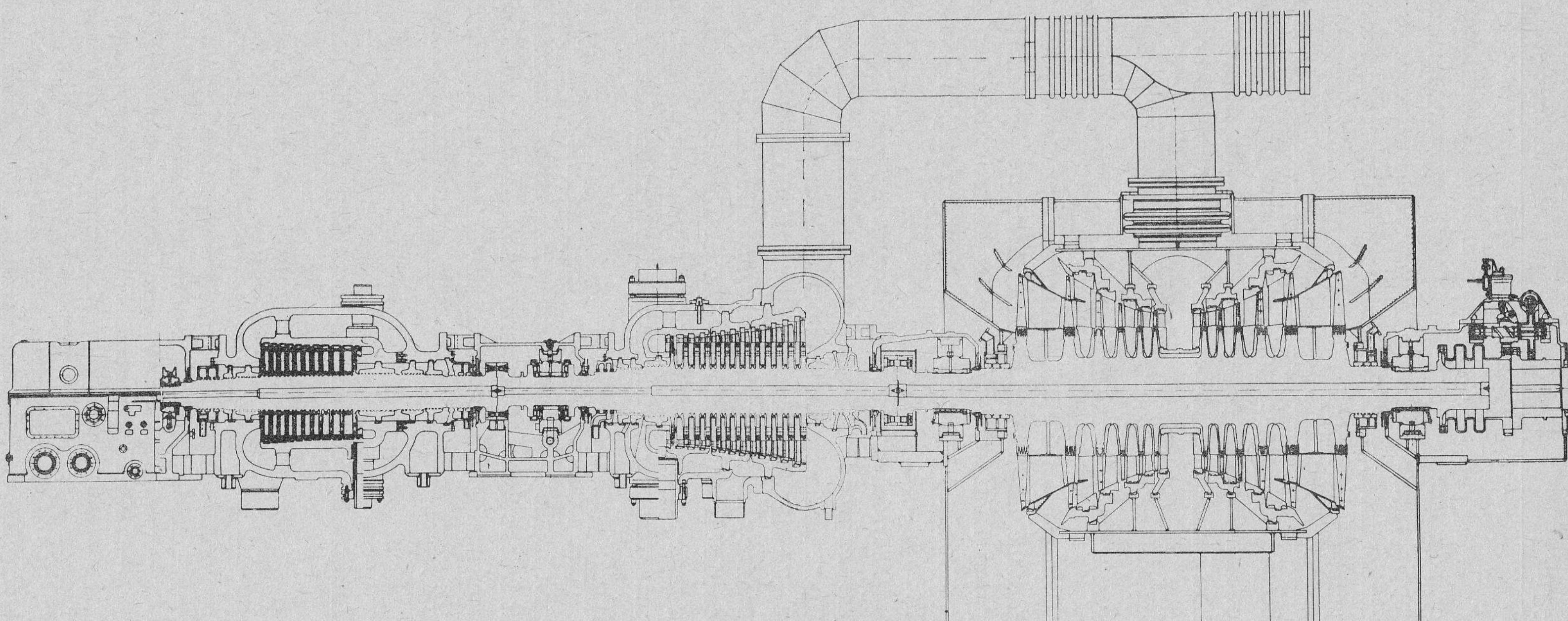


Fig. 21. The longitudinal section of the 13UK125-0 turbine

The longitudinal section of the 18UK135-0 turbine is shown in Fig. 20. This is a three-cylinder turbine with a two-stream low pressure part. Two parallel, controlled heat plant tappings are located in the LP part of the turbine, beyond the second stage of the first steam stream and beyond the third stage of the second stream. Steam extractions for heat plant purposes are controlled by means of rotating governing diaphgrams. The basic advantage of such a solution of the design of the LP part is attaining the same efficiencies in the range between the pressure of the both heat plant tappings on condensing operation as well as on heat plant operation and almost constant steam extraction to each of the heat plant heat exchangers, independently of the return network water temperature. This last property was indicated when discussing the turbine 13 P110-0.

Peak heat plant turbines

In most of the Polish thermal-electric power stations the peak demands for heat, area *C* in the diagram Fig. 15, are taken care of by steam generating boilers. In thermal electric power plants from which greater generation of electrical power is required, due to the power conditions peculiar to the given region peak-load type heat plant turbines are installed.

They deliver heat only during the period of peak heat plant demands. Such turbines are designed for co-operation with backpressure heat plant turbines of the type 13P110-0. These turbines operate through a greater part of the year as condensing power turbines.

ZAMECH produces one type of this kind of turbine. It is an extraction-condensing turbine designated with the symbol 13UK125-6 of 125 MW power rating for live steam conditions of 130 ata/535°/535°C. The section of this turbine is shown in Fig. 20. It is a three cylinder turbine with a two-stream LP part. A controlling flap valve is located in such a manner that a controlled extraction of steam for heat plant purposes is made possible.

The steam is directed to the peak heat exchanger in which the network water, heated first in the heat plant heat exchangers of a turbine 13P110-0, is additionally heated. As it follows from the network control diagrams and from the diagram of annual heat requirement, Fig. 15, the extraction for heat plant purposes is of greatly variable character; the steam pressure on the tapping fluctuates from 1 to 6 ata.

Analiza termodynamiczno-konstrukcyjna turbin ciepłowniczych produkcji ZAMECHU

Streszczenie

Przeprowadzono termodynamiczną analizę turbin ciepłowniczych pod kątem wpływu na moc turbiny i wartość energii ciepłowniczej następujących czynników:

- ilości stopni podgrzewu ciepłowniczego,
- rozdziału podgrzewu wody sieciowej na poszczególne wymienniki,
- wielkości powierzchni wymiany ciepła w wymiennikach.

Następnie zbadano wpływ zmiennych warunków ciepłowniczych, tzn. temperatury powrotnej i natężenia przepływu wody sieciowej na moc turbiny.

Na tle tej analizy przedstawiono rozwój turbin ciepłowniczych produkowanych w ZAMECHU, który zilustrowano licznymi przekrojami i zdjęciami maszyn ciepłowniczych.

Термодинамическо-конструкционный анализ теплофикационных турбин, изготовляемых заводом ЗАМЕХ

Резюме

Проведен термодинамический анализ теплофикационных турбин с целью определения влияния на мощность турбины и значение теплофикационной энергии таких факторов, как:

– количество ступеней теплофикационного подогрева,
– распределение подогрева сетевой воды на отдельные теплообменники,
– значения поверхности теплообмена в теплообменниках.

Очередно исследовано влияние переменных теплофикационных условий, т.е. поворотной температуры и расхода сетевой воды, на мощность турбины.

На фоне этого анализа представлено развитие теплофикационных турбин, выпускаемых заводом ЗАМЕХ, иллюстрируемое многими сечениями и снимками машин.