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PROBLEMS OF CARNOTISATION OF STEAM TURBINE CIRCUITS

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Abstract: The paper presents new concepts related to improvement of the efficiency of thermal circuits of steam turbines. Starting from a traditional C-R system, the efficiency losses of the circuit depending on the number of heat recovery exchangers in the system were indicated. Three different types of heat recovery regeneration exchangers that can be used in thermal circuits and their impact on the efficiency of the steam turbine system are presented. Further in the paper, the concept of the internal superheating and the possibilities of increasing efficiency in its use are presented.

Keywords: efficiency of steam turbine, internal overheating, regenerative heat exchangers.

The problem of increasing the efficiency of thermal circuits in steam turbines is very old. There is a huge number of publications and studies on this subject whose aim is to deal with the issue of the increase in circuit efficiency. The analyses were carried out in a very diverse form, taking into account the unique characteristics of real or theoretical systems. This paper focuses on an analysis of the theoretical C-R circuit, which is a comparative circuit for actual steam turbine systems. This applies to systems used in both shipbuilding and land-based power generation. Against the background of the previous methods used in thermal circuits of steam turbines, new techniques and new equipment were proposed to improve the efficiency of the systems. While there is a great deal of information in the literature on the impact of the initial parameters of steam and pressure in the condenser on circulation efficiency, structural changes are presented much less extensively. Also, the types of exchangers used are limited to two: surface and contact.

Figure 1 shows a theoretical C-R circuit in a T-s system, which is a comparative circuit for steam turbine systems. Next to the figure, on the right there is a schematic representation of the system where K stands for boiler (steam generator), T stands for turbine, S stands for condenser, and P stands for feed water pump. In the circuit, 1-2 is an isentropic process of pumping the working medium, 2-3 represents an isobaric production of superheated steam, 3-4 is an isentropic expansion process in the turbine, and 4-1 is an isobaric steam condensation process in the condenser.

The assumption of thermodynamic transformations without entropy changes for processes 1-2 and 3-4, and for isobaric processes 2-3 and 4-1 without pressure losses is an idealisation that results in a theoretical C-R circuit.

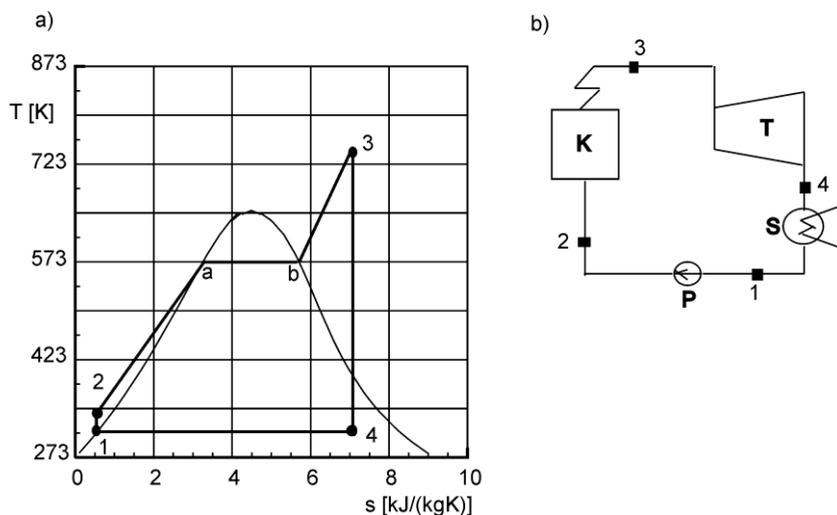


Fig. 1. The theoretical C-R circuit in a T-s system (a) and a schematic diagram of the system (b)

The circuit presented in this way shows three parameters that influence its efficiency. These are:

- 1 – pressure of the live steam feeding the steam turbine system,
- 2 – temperature of the live steam feeding the system,
- 3 – pressure of the steam at the turbine outlet.

For a circuit defined in this way, the impact of the parameters on its efficiency can be determined. In this case, there are three values that affect the resulting efficiency. These are the live steam pressure from the boiler p_0 , the live steam temperature t_0 , and the condenser pressure p_k . If the following values are assumed: $p_0 = 120$ bar, $t_0 = 500^\circ\text{C}$, and $p_k = 0.05$ bar, then by assuming that two of the three parameters are constants, the effect of the third on the efficiency of the C-R circuit can be determined.

Figure 2 shows the effect of changes in the initial pressure p_0 on the efficiency, Figure 3 shows the effect of temperature t_0 , and Figure 4 shows the effect of pressure p_k on the efficiency of the C-R circuit. The pressure in the condenser depends on the ambient conditions, mainly the temperature of the lower heat source. Therefore, a constant tendency to raise the initial parameters of the steam that is fed into the turbine can be observed in new designs of thermal circuits. However, in such a case,

the basic limitation is the materials, which have limited strength at very high temperatures and pressures.

Another way to increase the efficiency of steam turbine circuits is to change the structure of the system. The first common technique is to use internal superheating, the idea of which is presented in Figure 5. Introduction of superheating makes it possible to increase the average temperature of the heat fed to the circuit and, consequently, to increase the efficiency of the C-R circuit. In this case an important element is selection of pressure at which the steam superheating takes place in the interstage superheater. The turbine is divided into two parts: T1 and T2. The division pressure is the pressure at which heat is fed in the interstage superheater. The value of this pressure must be optimised.

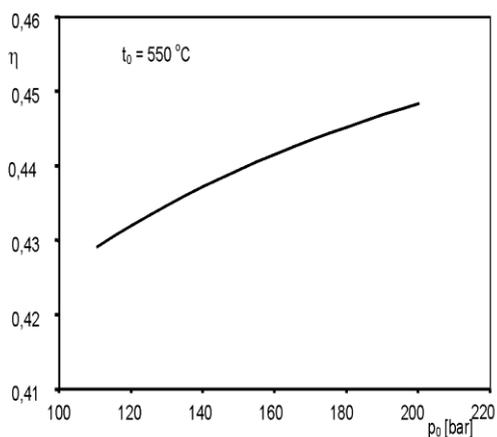


Fig. 2. The impact of pressure p_0

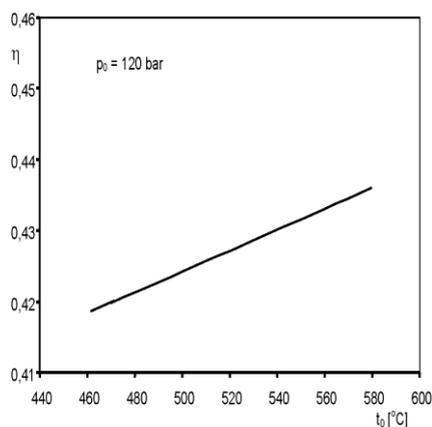


Fig. 3. The impact of temperature t_0

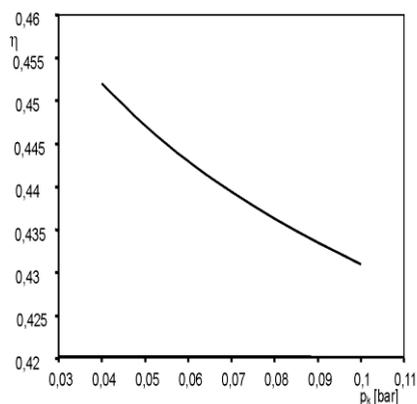


Fig. 4. The impact of pressure p_k

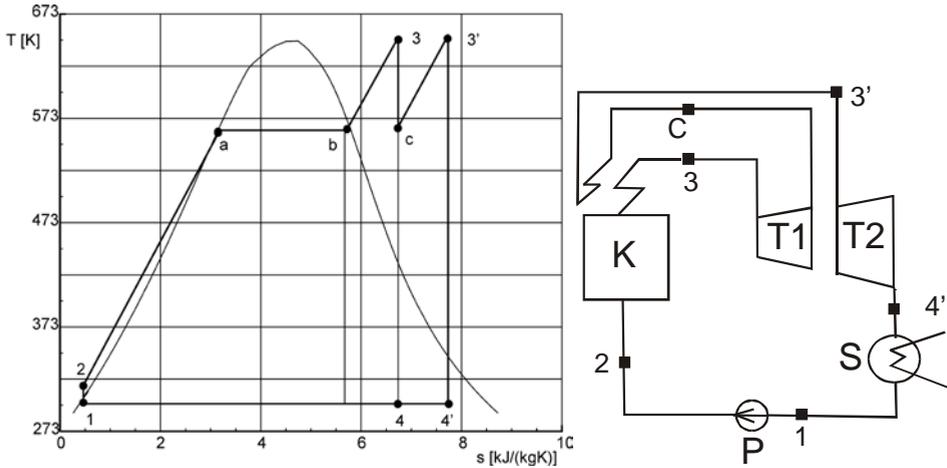


Fig. 5. A circuit with interstage superheating

If a variable pressure is assumed at point C, as shown in Figure 6, a variable circuit efficiency will result. There is an optimal point that makes it possible to achieve the highest value of efficiency.

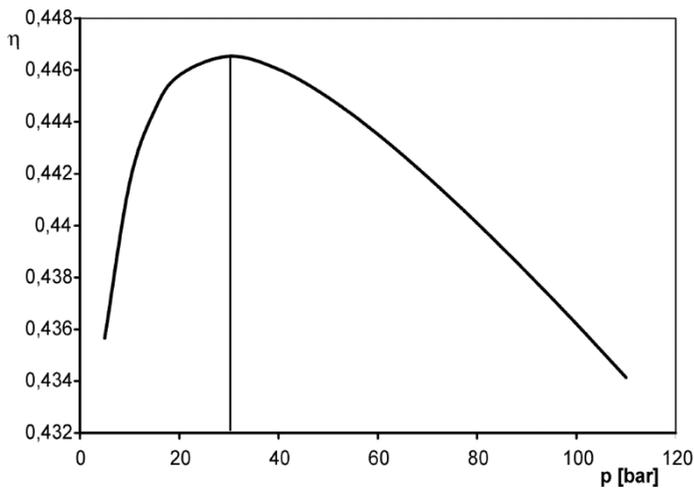


Fig. 6. Dependence of circuit efficiency on the pressure of the interstage superheater

Nowadays, single interstage superheating is commonly used in large steam turbine systems, as well as in new ship turbine circuits. There are circuit designs where double interstage superheating is used. There are two important elements that should be noted which are a consequence of the use of superheating. The first one is to reduce the humidity in the steam at the outlet of the turbine; this is a very positive

element. Another element is to move the entire line of expansion in the turbine towards higher specific entropy. As a result, more heat is released to the environment in the condenser.

Figure 7 shows the change of the degree of steam dryness for the theoretical C-R circuit for data identical to the data in Figure 6.

A change in the division pressure has a very strong impact on the dryness of the steam coming out of the turbine (point 4'). Since from the design point of view the permissible degrees of steam dryness should not be less than 0.85, for a given structure of the circuit and its component devices, the selection of the point of optimum turbine division must always be controlled with the permissible value for steam dryness level X.

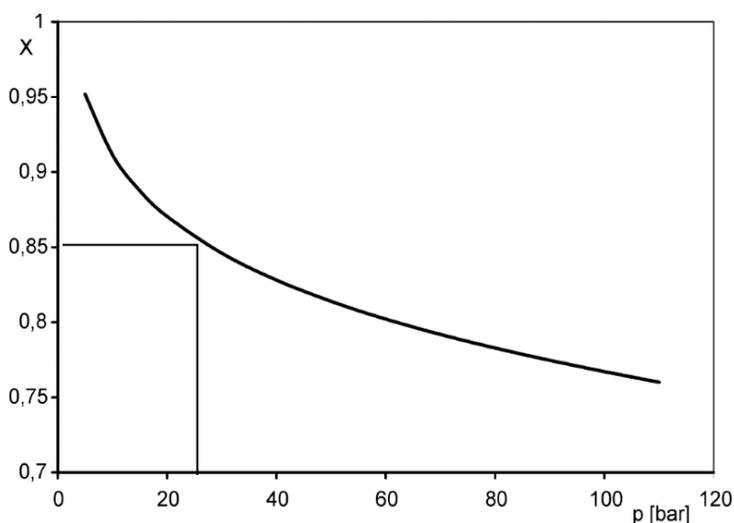


Fig. 7. Dryness level of steam X at the turbine outlet as a function of steam superheating pressure p

Other changes in the structure of the theoretical C-R circuit that are used to increase the efficiency of the circuit are the regenerative heating of the boiler feed water. Figure 8 shows a diagram of a system in which one regenerative heat exchanger is used.

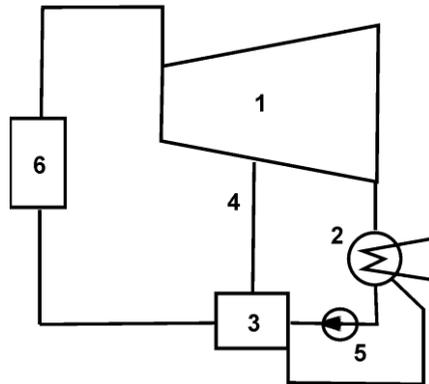


Fig. 8. A system with one regenerative heat exchanger

In the systems currently under construction, a large number of regenerative heat exchangers are used in order to increase the temperature of the boiler feed water. The number of heat exchangers in a large steam turbine system can be as many as 7 to 8. In ship systems, the number of heat exchangers is usually smaller and does not exceed 5. The temperature of the condensate at the condenser outlet is about 30°C and the temperature of the water at the boiler inlet can be up to 300°C . The theoretical C-R circuit can be analyzed as a circuit working in the wet steam area; in such a case, the beginning of expansion starts from the $X = 1$ line.

Figure 9 shows the C-R circuit operating in the wet steam area. While for the theoretical circuit the expansion process in the turbine (3-4) does not increase entropy, heating the feed water for isobaric heating (2-a) does result in a change of entropy marked in the figure (Δs). This results in the C-R circuit being much less efficient than the Carnot cycle.

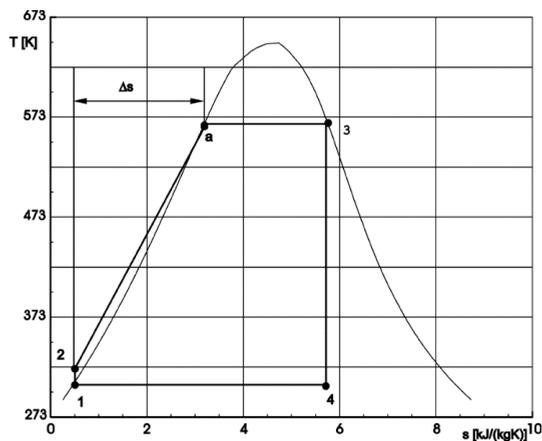


Fig. 9. The C-R circuit in the wet steam area

An analysis of such a theoretical circuit can demonstrate, for operation in the wet steam area, that there is a possibility of full carnotisation of the C-R circuit. The prerequisite for this is the number of regenerative exchangers in the recovery system going to infinity.

The analysis of the operation of the recovery system in the wet steam area is shown in Figure 10.

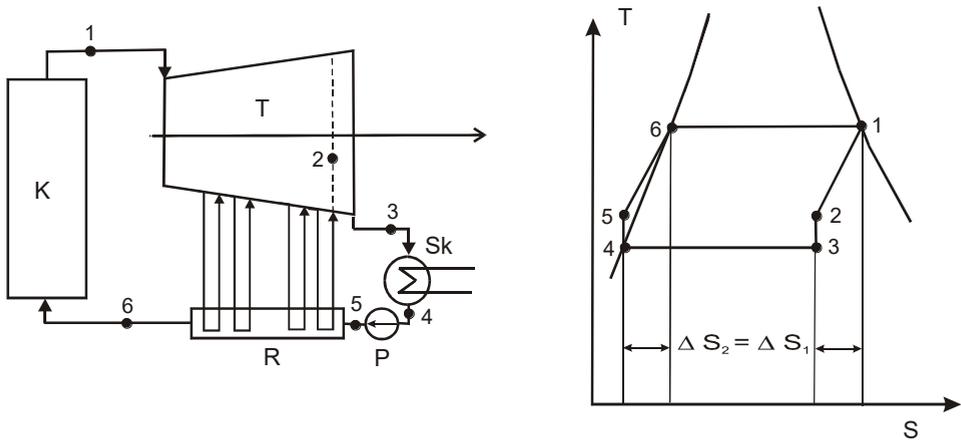


Fig. 10. Operation of the recovery system for an infinite number of exchangers in the wet steam area

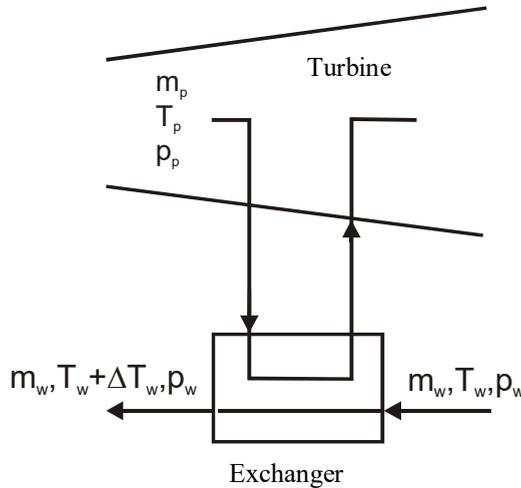


Fig. 11. Part of the thermal circuit

A part of such a system is shown in Figure 11. Steam taken from the turbine flows into the regenerative exchanger and gives off heat by heating up the feed water flowing into the boiler. The steam is then reintroduced into the turbine for further expansion. An infinite number of such regenerative exchangers leads to a situation in which the value Δs_1 , which describes the change of entropy of the heating steam, is identical to the value Δs_2 , which describes the change of entropy of the heated water (Fig. 10).

In a real system, one has to take into account the impossibility of using an infinite number of regenerative exchangers. In this situation, it is important to determine the impact of a finite number of exchangers on the process of carnotisation of the whole system. Introduction of a finite number of exchangers makes it possible to determine the amount of losses that will occur if the number of exchangers is less than infinity.

Figure 12 shows the transformations in the T-s system for one regenerative exchanger.

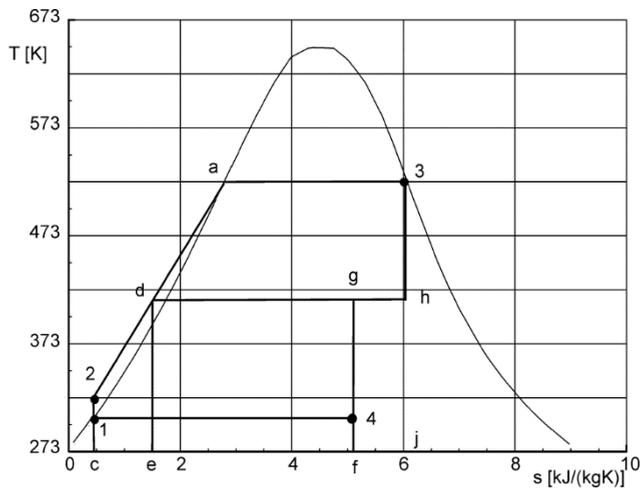


Fig. 12. Transformations in the circuit with one regenerative exchanger

In Figure 13, T_{pi} indicates the temperature of the steam in the next exchanger “i”. Size Δs_{1i} indicates the change of specific entropy of steam and Δs_{2i} indicates the change of specific entropy of water. If the quantity of heat released by the steam is Δq_{pi} , then the quantity of heat absorbed by the water Δq_{wi} assuming that the heat loss to the environment is ignored. The amount of heat released by the steam can be written as follows (for 1 kg of the working medium):

$$\Delta q_{pi} = T_{pi} \Delta s_{1i} \quad (1)$$

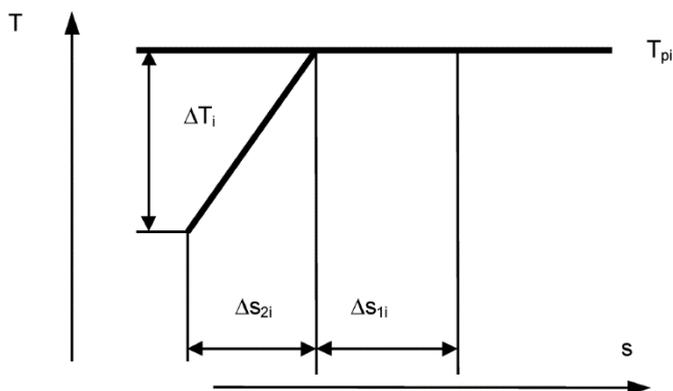


Fig. 13. Comparison of the processes of isothermal steam condensation and isobaric water heating

The amount of heat taken for heating water to increase its temperature by ΔT_i can be written with the following formula ($m_w = 1$ kg):

$$\Delta q_{wi} = T_{pi} \Delta s_{2i} - \frac{1}{2} \Delta T_i \Delta s_{2i} \quad (2)$$

We assume that the heat flow resistance in the heat exchanger is ignored and the heated water can reach a temperature equal to the temperature of the heating steam T_{pi} . For the sake of simplicity, the pumping operation is omitted and it is assumed that the limit line $X = 0$ can be locally interpreted as a segment of a straight line. The equations presented give the following results:

$$\left. \begin{aligned} T_{pi} \Delta s_{1i} &= \Delta s_{2i} \left(T_{pi} - \frac{1}{2} \Delta T_i \right) \\ \frac{\Delta s_{2i} - \Delta s_{1i}}{\Delta s_{2i}} &= \frac{1}{2} \frac{\Delta T_i}{T_{pi}} \end{aligned} \right\} \quad (3)$$

Locally, for the next “i” exchanger, the assumption of equal increments of Δs_{1i} i Δs_{2i} cannot be made if the value of ΔT_i is greater than zero. The water in the recovery system is heated in n exchangers connected in series within temperature range of T_4 to T_3 (the pumping operation is omitted). If the total change of entropy (between temperatures T_4 and T_3) of the water being heated is marked as Δs_2 and an identical change of Δs_{2i} in all n exchangers is assumed:

$$\Delta s_{2i} = \frac{\Delta s_2}{n} \quad (4)$$

then for the number n going to infinity, the limit can be written as:

$$\lim_{n \rightarrow \infty} \sum_{i=1}^n \frac{\Delta s_{2i} - \Delta s_{1i}}{\Delta s_{2i}} = \lim_{n \rightarrow \infty} \sum_{i=1}^n \frac{1}{2} \frac{\Delta T_i}{T_{pi}} \quad (5)$$

The sequence of ΔT_i divisions of temperature increase of heated water in successive n exchangers from T_4 to T_3 meets the conditions of normal division and the exchangers are connected in series. The right side of the equation (1.5) can then be recorded as an integral within T_4-T_3 as

$$\lim_{n \rightarrow \infty} \frac{1}{2} \sum_{i=1}^n \frac{\Delta T_i}{T_{pi}} = \frac{1}{2} \int_{T_4}^{T_3} \frac{dT}{T_{pi}} = \frac{1}{2} \ln T \Big|_{T_4}^{T_3} \quad (6)$$

The right-hand side of the equation (1.5) is defined as an integral (1.6). The left-hand side of the equation (1.5) still needs to be linked to the changes in the specific entropy of the water and steam being heated. The total change of specific entropy of the heating steam between temperatures T_4 and T_3 can be written as the sum of local entropy changes in subsequent exchangers:

$$\Delta s_1 = \sum_{i=1}^n \Delta s_{1i} \quad (7)$$

The total change in the specific entropy of the water being heated in n heat exchangers connected in heaters can be presented as a sum:

$$\Delta s_2 = n \Delta s_{2i} = \sum_{i=1}^n \Delta s_{2i} \quad (8)$$

Calculation of the ratio of the difference between the total entropy changes of the water being heated and the total entropy changes of the heating steam to the total entropy changes of the water being heated results in:

$$\frac{\Delta s_2 - \Delta s_1}{\Delta s_2} = \frac{\sum_{i=1}^n \Delta s_{2i} - \sum_{i=1}^n \Delta s_{1i}}{\sum_{i=1}^n \Delta s_{2i}} \quad (9)$$

The left-hand side of this equation determines the ratio of the global changes in the circuit for n exchangers connected in series between the temperatures T_4 and T_3 .

Due to the assumption that the local changes in the specific entropy of the heated water in the subsequent exchangers connected in series are the same:

$$\Delta s_{21} = \Delta s_{22} = \dots = \Delta s_{2n} = \text{const} \quad (10)$$

the right-hand side of equation (1.9) can be transformed as follows:

$$\frac{\sum_{i=1}^n \Delta s_{2i} - \sum_{i=1}^n \Delta s_{1i}}{\sum_{i=1}^n \Delta s_{2i}} = \frac{\sum_{i=1}^n (\Delta s_{2i} - \Delta s_{1i})}{n \Delta s_{2i}} = \frac{1}{n} \sum_{i=1}^n \frac{\Delta s_{2i} - \Delta s_{1i}}{\Delta s_{2i}} \quad (11)$$

Equations (1.9) and (1.11) result in:

$$n \frac{\Delta s_2 - \Delta s_1}{\Delta s_2} = \sum_{i=1}^n \frac{\Delta s_{2i} - \Delta s_{1i}}{\Delta s_{2i}} \quad (12)$$

The final form of the relationship that takes into account n exchangers connected in series is the following:

$$\left. \begin{aligned} \frac{\Delta s_2 - \Delta s_1}{\Delta s_2} &= \frac{1}{2n} \ln \frac{T_3}{T_4} \\ \frac{\Delta s_1}{\Delta s_2} &= 1 - \frac{1}{2n} \ln \frac{T_3}{T_4} \end{aligned} \right\} \quad (13)$$

The relationship given here makes it possible to evaluate the amount of losses in relation to the general Carnot cycle when the number of exchangers is less than infinity. It allows the determination of losses in relation to the efficiency of the Carnot cycle that result from the use of a finite number of regenerative exchangers.

For a set temperature of the water flowing from the condenser to the recovery system equal to approx. 300 K.

Figure 14 shows the values of the $\Delta s_1/\Delta s_2$ ratio for a finite number of exchangers in the recovery system as a function of temperature T_3 . The size of this ratio depends on the initial temperature T_3 and the final temperature T_4 , and on the number of exchangers n . The ratio between the changes of the entropy of the heating steam Δs_1 and the changes of the entropy the heated water Δs_2 determines the perfection of the water heating process in the recovery system. When this ratio reaches 1, full carnotisation in the recovery system is achieved. Values less than 1 make it possible to evaluate the losses in relation to the efficiency of the Carnot cycle.

The impossibility to use an infinite number of regenerative exchangers in the circuit is, as one can see, always associated with losses. As can be seen from the graph in Figure 14, when the initial steam temperature is 500 K (226.85°C) for the number of heat exchangers equal to 5, the losses compared to the Carnot cycle are equal to about 5%.

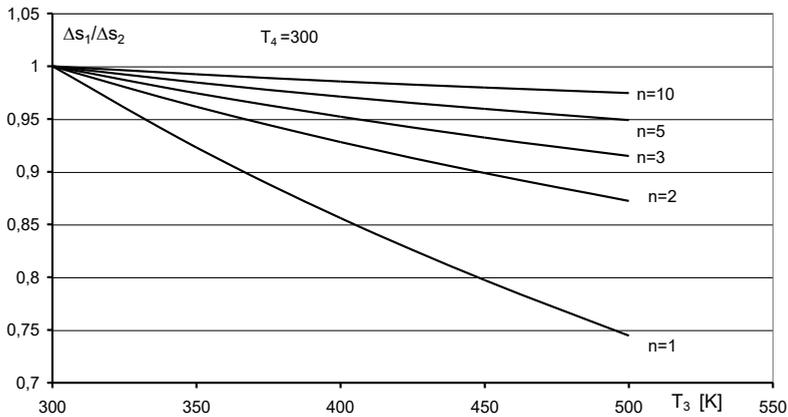


Fig. 14. The ratio of the changes in the entropy of the heating steam and of the heated water

The relationship specified herein, which describes the changes in the entropies of the heating steam and the heated water, concerns the theoretical circuit. If the efficiency of the expansion process in the turbine is assumed to be 0.9, a comparison of the derived equation for the circuit with an isentropic expansion process with balance calculations for a circuit where the efficiency of the turbine is assumed to be 0.9 can be presented in Table 1.

Table 1. A comparison of equation (1.13) and balances of circuits $\Delta s_1 / \Delta s_2$

Number of exchangers	Equation (1.13)	Circulation
n = 20	0.987229	0.987124
n = 10	0.974587	0.974497
n = 7	0.963484	0.963842
n = 5	0.948917	0.949979

The relationship for the theoretical thermal circuit makes it possible to determine with high accuracy the changes in the efficiency of the theoretical steam turbine circuit taking into account the number of exchangers. For the theoretical circuit, the impact of the number of exchangers can be presented as a change of Δs_1 and Δs_2 . It can be presented as shown in Figure 15. The efficiency of the circuit can be determined as shown in the relationships below. The change in efficiency depends on the number n . The appropriate relationship takes into account the losses that will occur if the number of exchangers in the recovery system is less than infinity.

The heat released from the circuit per 1 kg of the working medium can be expressed by the following formula:

$$q_w = T_4(\Delta s + \Delta s_n) \quad (14)$$

while the heat supplied to the circuit can be expressed as:

$$q_d = T_3 \Delta s \quad (15)$$

The value Δs can be written as:

$$\Delta s = \frac{r(p_3)}{T_3} \quad (16)$$

where $r(p_3)$ is the evaporation heat. Based on Figure 15, one can write the following relationship:

$$\Delta s_n = \Delta s_2 - \Delta s_1 \quad (17)$$

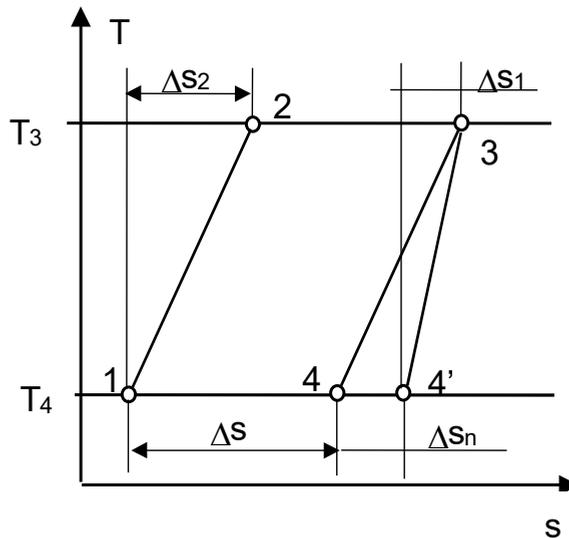


Fig. 15. A circuit including losses in the recovery system

Taking into account the previous equation:

$$\frac{\Delta s_2 - \Delta s_1}{\Delta s_2} = \frac{1}{2n} \ln \frac{T_3}{T_4} \quad (18)$$

and assuming that the formula for the change in the entropy of the heated water is the following:

$$\Delta s_2 = s_2 - s_1 = c_p(p_3) \ln \frac{T_3}{T_4} \quad (19)$$

the value Δs_n can be defined as follows:

$$\Delta s_n = \Delta s_2 \frac{1}{2n} \ln \frac{T_3}{T_4} = \frac{c_p}{2n} \left(\ln \frac{T_3}{T_4} \right)^2 \quad (20)$$

Ultimately, the efficiency with a finite number of exchangers is expressed by the following formula:

$$\eta = 1 - \frac{T_4(\Delta s + \Delta s_n)}{T_3 \Delta s} = 1 - \frac{T_4}{T_3} - \frac{c_p T_4}{2nr} \left(\ln \frac{T_3}{T_4} \right)^2 \quad (21)$$

which can be written as follows:

$$\eta = \eta_c - \Delta \eta = \eta_c - \frac{c_p T_4}{2nr} \left(\ln \frac{T_3}{T_4} \right)^2 \quad (22)$$

where the value η_c indicates the efficiency of the Carnot cycle.

Figure 16 shows a comparison of the efficiency of Carnot cycles and the efficiency of theoretical C-R circuits for a finite number of exchangers n . The temperature assumed for the lower heat source is $T = 300$ K and for the upper source it is 500 K and 600 K.

As in previous analyses, all the exchangers used are surface exchangers. The graph in Figure 16 shows a very strong impact of the number of exchangers on the process of increase of the efficiency of the steam turbine system.

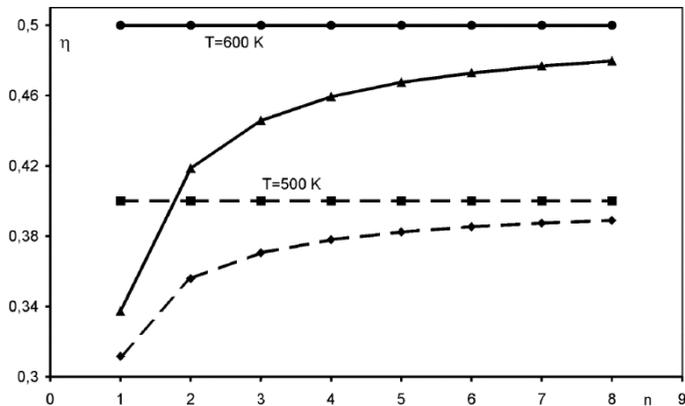


Fig. 16. Efficiency of the theoretical C-R circuit as a function of the number of exchangers n

Of note are the different types of exchangers that can be used in the recovery system. The recovery system can be equipped with contact exchangers.

The formulas and diagrams presented so far refer to a system in which exchangers of the contact type are used.

Figure 17 shows three types of exchangers that can be used in a recovery system.

The relationship and the graphs were created for a recovery system where only surface exchangers are present. As has been mentioned, a recovery system can use mixer type exchangers. However, in addition to these two types of exchangers, a recovery system can be made using separation exchangers. These three types of regenerating exchangers are shown in Figure 17 in the following order from the left side: a surface exchanger, a mixer exchanger, and a separation exchanger.

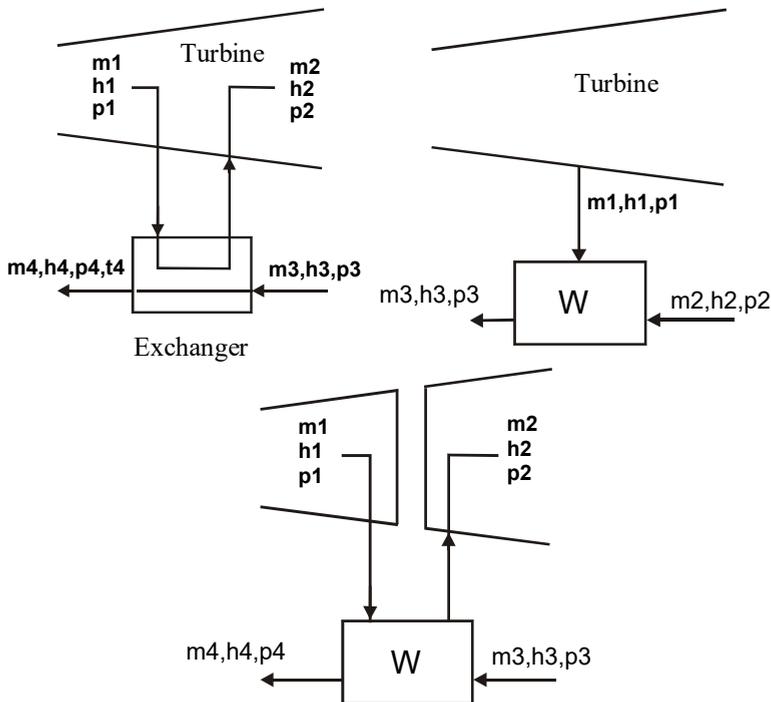


Fig. 17. Three types of regenerative exchangers

These three types of exchangers make it possible to build three different thermal circuits. The diagram and the transformations in the first circuit (OB1) are shown in Figure 11.

The diagram and the transformations in T-s system for the second circuit (OB2) are shown in Figure 18. For a system with regenerative and separating exchangers (OB3), they are shown in the next figure.

It is necessary to explain the principle of operation of the regenerative and separating exchanger. In this exchanger, the feed water is heated and moisture is separated from the steam coming out of the exchanger to the turbine flow system simultaneously. After expansion in the turbine, the steam flows into the exchanger where it releases heat back to the feed water and, additionally, the condensed moisture is separated.

Figure 20 shows in a schematic form the concept of a regenerative and separating exchanger.

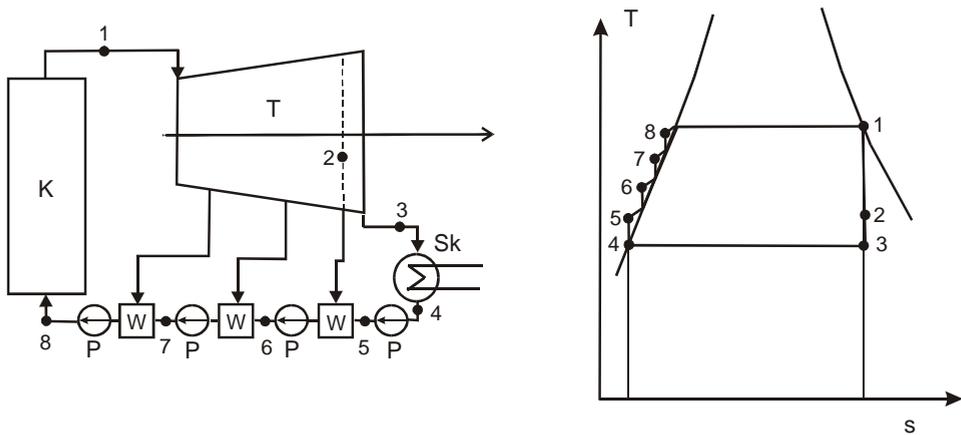


Fig. 18. A diagram and transformations for a circuit with mixer exchangers

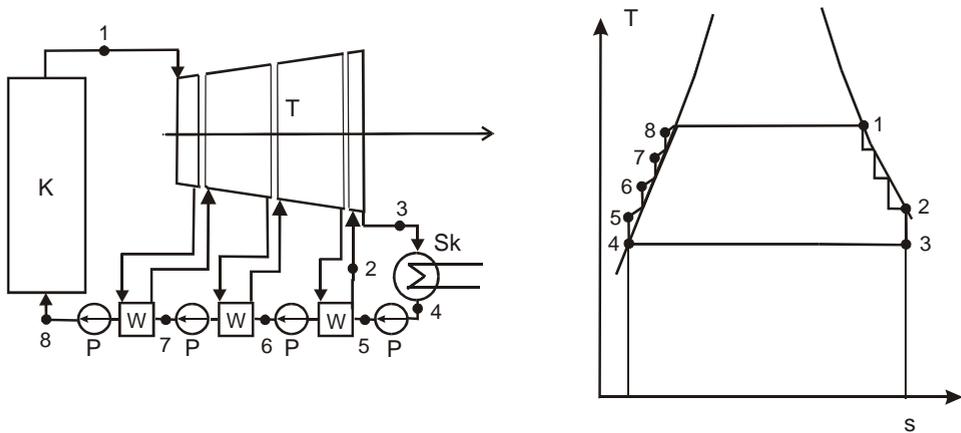


Fig. 19. A diagram and transformations for a circuit with separating exchangers

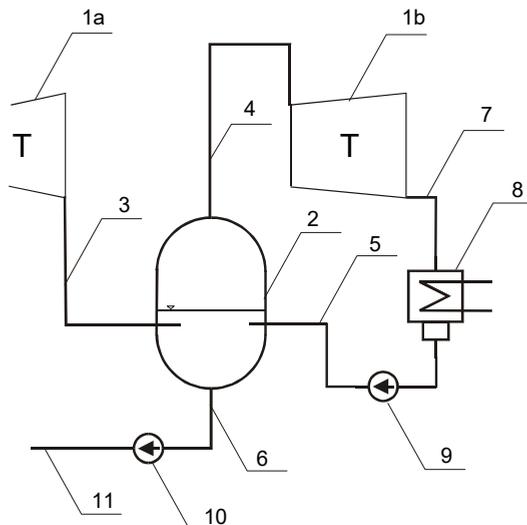


Fig. 20. A diagram of the regenerative and separating exchanger; 1a – steam turbine body, 1b – another steam turbine body, 2 – regenerative and separating exchanger, 3 – steam output from the turbine body, 4 – saturated steam intake from the regenerative and separating exchanger to the next turbine body, 5 – water supply to the regenerative and separating exchanger, 6 – condensate intake from the regenerative and separating exchanger, 7 – steam outlet to the condenser, 8 – steam condenser, 9 – condensate pump, 10 – condensate pump for the condensate from the regenerative and separating exchanger, 11 – pipeline feeding water to the boiler or the regenerative exchangers

Figure 21 shows a comparison of the efficiency of thermal circuits for the aforementioned three exchangers used in recovery systems. The balance calculations made it possible to determine the efficiency of the thermal circuits in which three different heat exchangers were used in the recovery systems shown in Figure 17. OB1 refers to a circuit in which only surface exchangers are used. OB2 refers to a recovery system that uses mixer exchangers. OB3, on the other hand, concerns a circuit in which the regenerative and separating type exchangers are used. The horizontal axis indicates the number of exchangers used. For the number of heat exchangers going to infinity, all three analyzed systems have one limit, namely the efficiency of the Carnot cycle for the set temperature of the upper and lower heat sources. However, it is important to note the differences in the efficiency for a finite number of exchangers. When discussing carnotisation of a thermal circuit, it is important to take into account not only the number of heat exchangers but also their type. A circuit with regenerative and separating heat exchangers with a small number of exchangers allows for high efficiency of the thermal circuit.

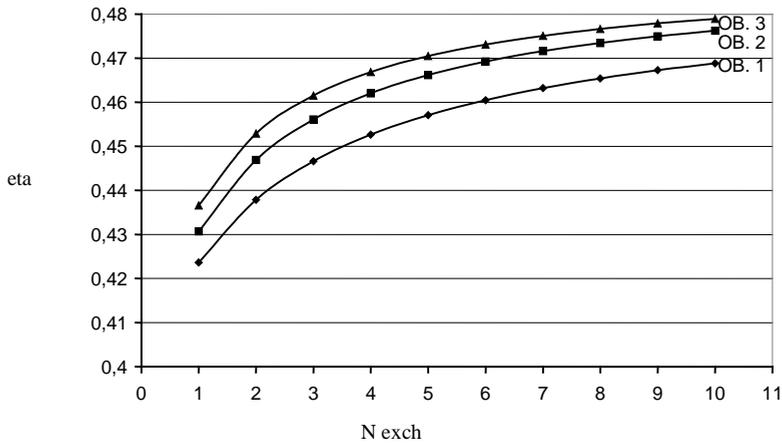


Fig. 21. A comparison of efficiency for three different types of exchangers in the recovery system

For OB3, an efficiency of 46% can be achieved with three regenerating and separating exchangers, while for OB1, up to six surface exchangers are required. The differences in efficiency between the circuits are shown in Figure 22.

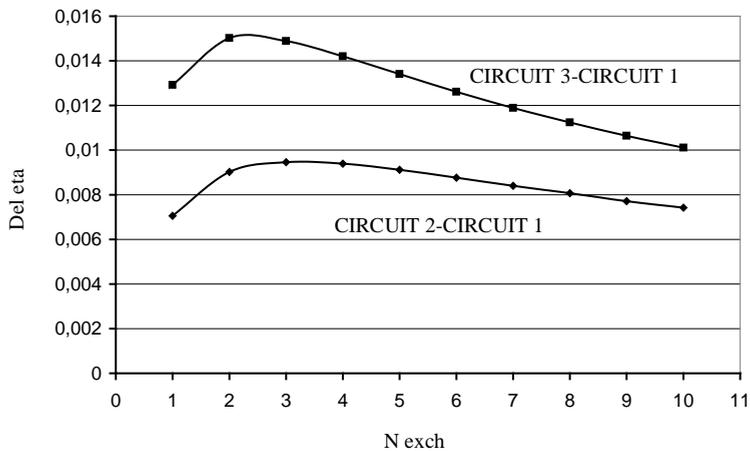


Fig. 22. The differences in efficiency for different recovery systems

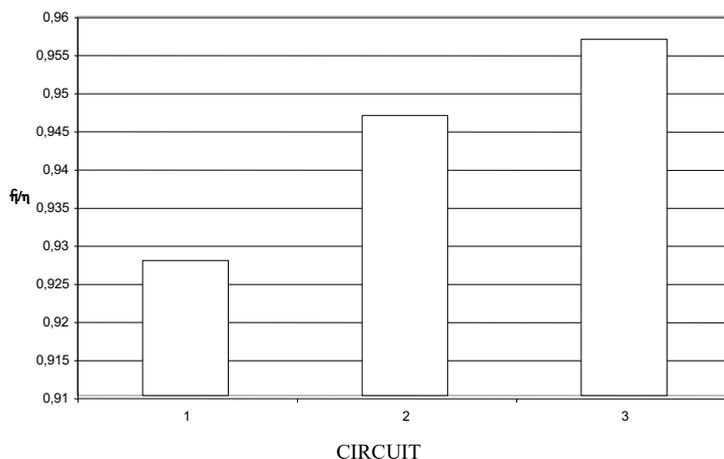


Fig. 23. The ratio of efficiency of three circuits to the efficiency of the Carnot cycle

Figure 23 shows the efficiency of the circuits in relation to the efficiency of the Carnot cycle for 5 regenerative exchangers. As shown in Figure 22, the efficiency differences can be as high as 1.5%. This is a large difference. New publications describing the design and performance of power units under construction indicate efficiency increases of 0.5% for entire systems. From this point of view, new types of regenerative exchangers can provide a significant increase in the efficiency of thermal circuits.

The biggest differences occur for a small number of exchangers. The graphs and analyses presented herein are to show the complexity of the process of selection of the complete thermal circuit. The techniques used so far have included mainly the use of surface exchangers in recovery systems. Regenerative and separating heat exchangers are particularly suitable for nuclear systems with low live steam superheating temperatures. This is similar for ship systems with a small number of regenerative exchangers.

So far, the analysis has been carried out for the theoretical thermal circuit working in the wet steam area. In the case of modern high capacity power units, there are very high parameters of the live steam flowing from the boiler to the turbine. Changes in the structure of the thermal circuit can then be implemented differently.

Figure 24 shows the changes in the C-R circuit for superheated live steam. Let us assume that the analysis will concern a circuit with surface heat exchangers. If expansion starts in the highly superheated steam area, the steam taken from the outlets is highly superheated. This applies to heat exchangers fed with high-pressure steam. If the concept of average superheating temperature defined by the following relationship is introduced:

$$\bar{T}_{przi} = \frac{1}{s_3 - s'_i} \int_{s'_i}^{s_3} T(s) ds \quad (23)$$

and the impact of the number of exchangers on the operation of the recovery system is determined, the following relationship will result

$$\frac{\Delta S_1}{\Delta S_2} = \frac{1}{n} \sum_{i=1}^n \left(1 - \frac{1}{2} \frac{\Delta T_{wi}}{T_{pi}} \right) \frac{T_{pi}}{\bar{T}_{przi}} \quad (24)$$

where s_i is the entropy of boiling water at the steam pressure in the regenerative exchanger.

As one can see, the ratio of the saturation temperature in the regenerative exchanger T_{pi} to the average superheating temperature \bar{T}_{przi} of the steam significantly affects the ratio of the changes in the entropy of the heating steam ΔS_1 to the changes in the entropy of the heated water ΔS_2 .

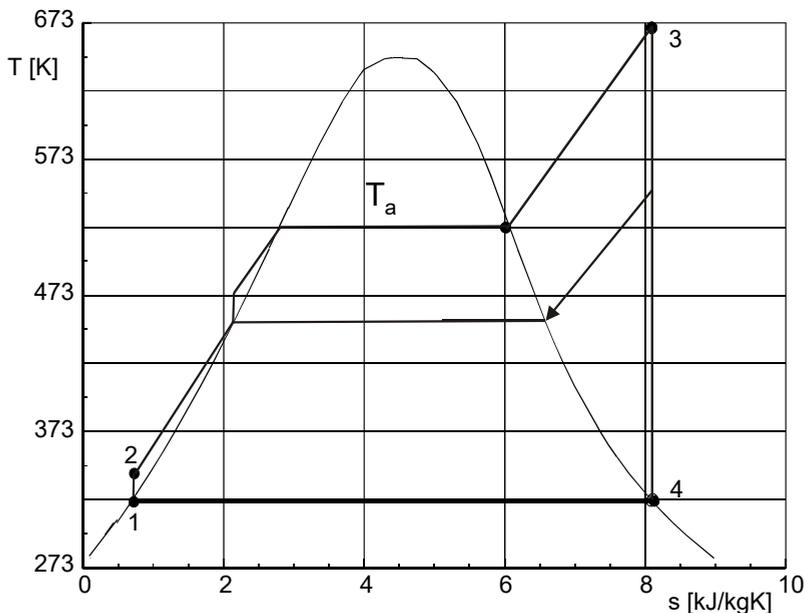


Fig. 24. The C-R circuit for superheated steam

Figure 25 shows the ratio $\Delta S_1/\Delta S_2$ for the circuit in the wet steam area and for a circuit with superheated steam as a function of the number of regenerative exchangers n . Introduction of a highly superheated steam into the regenerative exchanger causes significant deterioration of the recovery system. A lower $\Delta S_1/\Delta S_2$ ratio worsens the carnotisation of the entire thermal circuit.

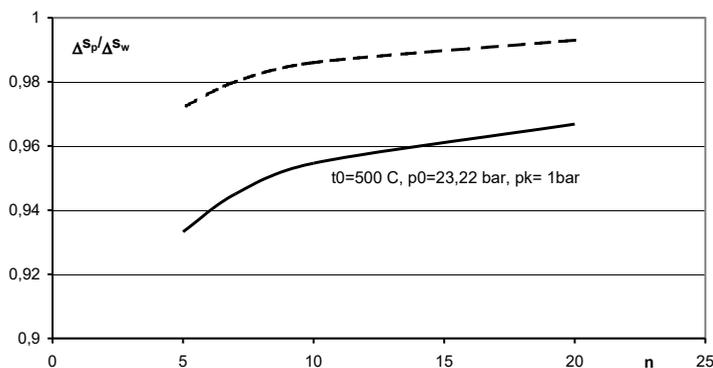


Fig. 25. The $\Delta S_1/\Delta S_2$ ratio for expansion in the wet steam area (dashed line) and for expansion of superheated steam (continuous line) as a function of the number of exchangers

Examples of the superheating values of the steam flowing into regenerative exchangers for a 460 MW supercritical power unit are shown in Figure 26. The value of superheating of the steam flowing into the regenerative exchangers can be very high. It can be as high as 250°C. The value of this superheating depends on the position of the outlet in the turbine. According to the definition given above, the average superheating temperature of the steam flowing into the recovery system can be determined.

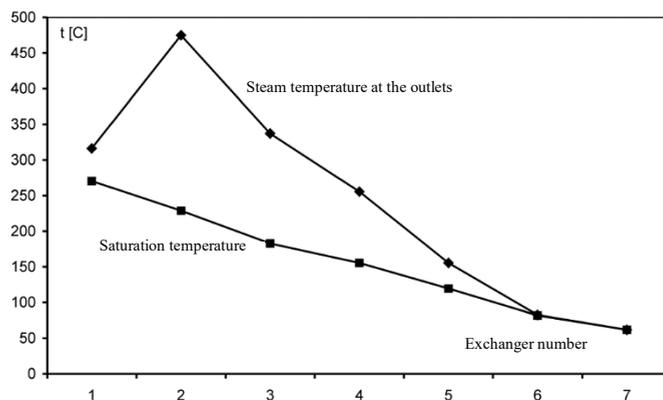


Fig. 26. Steam temperatures in outlets of a 460 MW turbine

Figure 27 shows the steam temperature at the outlet, the average superheating temperature, and the saturation temperature in the regenerative heat exchanger.

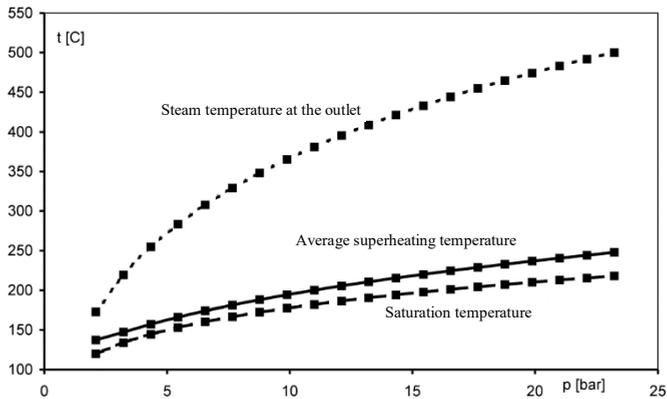


Fig. 27. A comparison of the steam temperature at the turbine outlet, the average superheating temperature, and the saturation temperature in the regenerative exchanger

The graphs presented here clearly show the strong impact of steam superheating on the recovery system. If superheated steam flows into the regenerative exchangers, one should expect additional losses of efficiency of the thermal circuit.

If the carnotisation process is treated as a reduction to a minimum of losses occurring in the thermal circuit, another element can be pointed out which strongly affects the efficiency of the recovery system. This element is superheating of the steam flowing into the regenerative exchangers.

In this case, losses can be counteracted by introducing the concept of internal superheating.

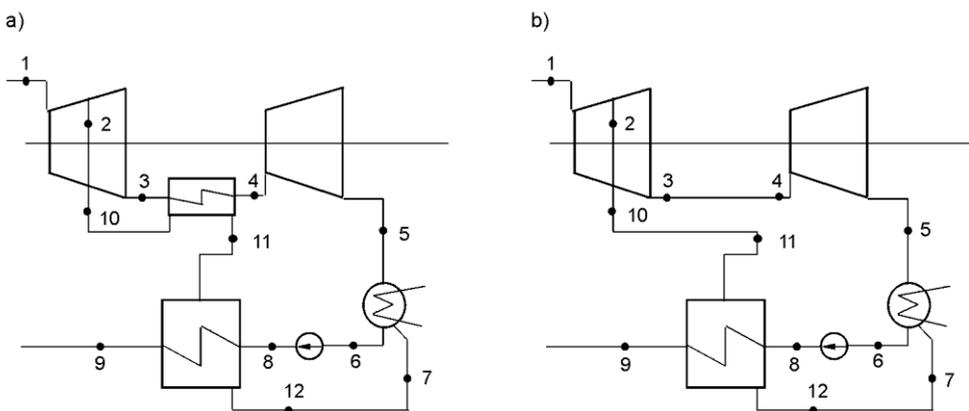


Fig. 28. Comparison of a system with internal superheating (a) and a traditional system (b)

Figure 28 shows a schematic representation of a system with internal superheating (a) and a traditional steam turbine system (b). The essence of internal superheating is to cool down the steam before it enters the recovery system. In system (a), between points 10 and 11, the steam flowing into the recovery system is cooled down. The heat is absorbed from the discharged steam and transferred between points 3 and 4 to the steam flowing into the low-pressure part of the turbine. This reduces or eliminates superheating of the steam flowing into the recovery system. The result is a slight increase in circuit efficiency. This applies to a system with one regenerative exchanger.

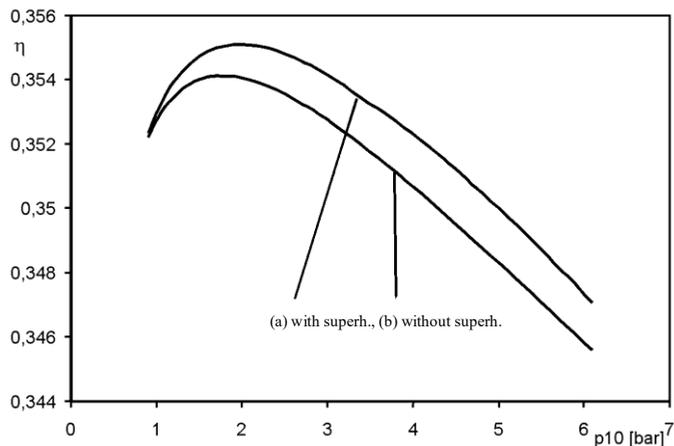


Fig. 29. A comparison of the efficiency of circuits with internal superheating and without superheating

The measure of the improvement of the recovery system is the $\Delta s_p/\Delta s_w$ ratio where Δs_p indicates the change of the entropy of the heating steam and Δs_w indicates the change of the entropy of the heated water.

Figure 30 shows a comparison of the ratios of the changes of the entropy of the heating steam to those of the heated water for one regenerative exchanger as a function of the pressure of the steam taken from the turbine outlet.

It can be noted that the maximum value of efficiency in a system with superheating is obtained at a higher pressure at the outlet than in a system without internal superheating. The search for efficiency reserves in conventional heat circuits makes it possible to determine efficiency gains using the internal superheating concept for large systems such as supercritical power units.

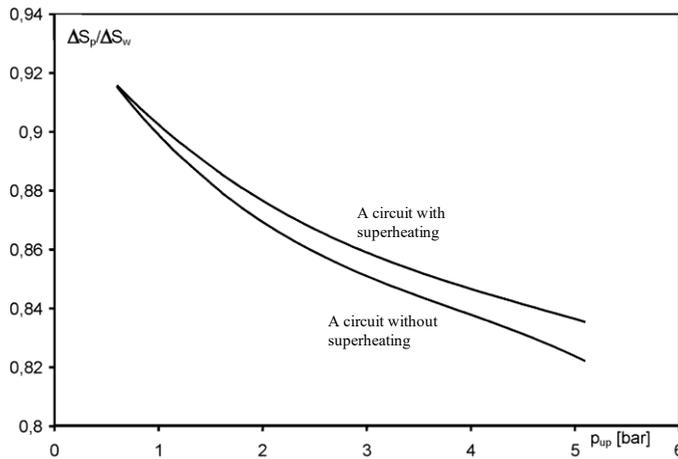


Fig. 30. The $\Delta S_p / \Delta S_w$ ratio as a function of pressure p_{10}

A solution used to date for typical thermal circuits of condensing turbines is shown in Figure 31A. The new solution is shown in Figure 31B. Its most important element is introduction of a new heat exchanger between the recovery system and the turbine outlets. In this exchanger, the steam taken from the turbine is cooled down and the absorbed heat is transferred to the steam which expands in the low-pressure part. In the standard solution, the steam taken from the outlets is directly introduced into the regenerative exchangers. In the new solution, the steam is first cooled before it is fed into the exchanger. The steam coming from the outlets of turbine (1) heats the steam coming from the outlet of the medium-pressure part, which after being heated in the internal superheater (3) is fed into the low-pressure part of turbine (1) where expansion takes place. The result is a greater decrease of enthalpy in the low-pressure part. An additional effect is a change of the final dryness of the steam at the outlet of the low-pressure part. For example, if originally the degree of dryness at the turbine outlet is equal to approx. $X = 0.9$ for the standard system A, then after introduction of internal superheating (3) the degree of dryness at the outlet increases to approx. $X = 0.99$ for system B. The losses of the damp steam flow through turbine (1) are reduced. In addition, the flow system is not at risk of erosion. Another effect of introduction of internal superheating is a reduction in the mass flow rate of steam at the outlet of the low-pressure part and a reduction in the amount of heat released from the circuit in the condenser. The diagrams shown in Figures 31A and 31B apply to 200 MW subcritical power units. Then the difference in efficiency between a power unit with internal superheating and a power unit without internal superheating is 1.5%.

The expansion lines for both compared systems are shown in the Figure 32.

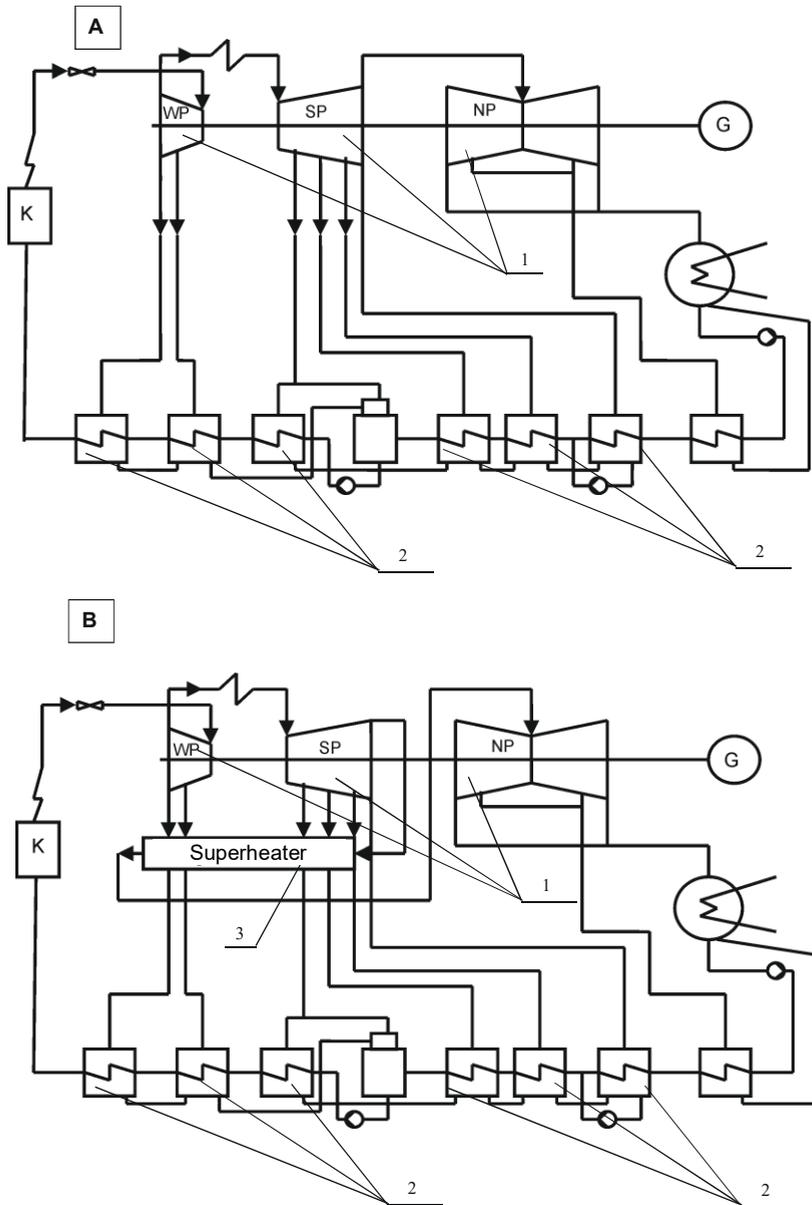


Fig. 31. Systems without internal superheating (A) and with internal heating (B)

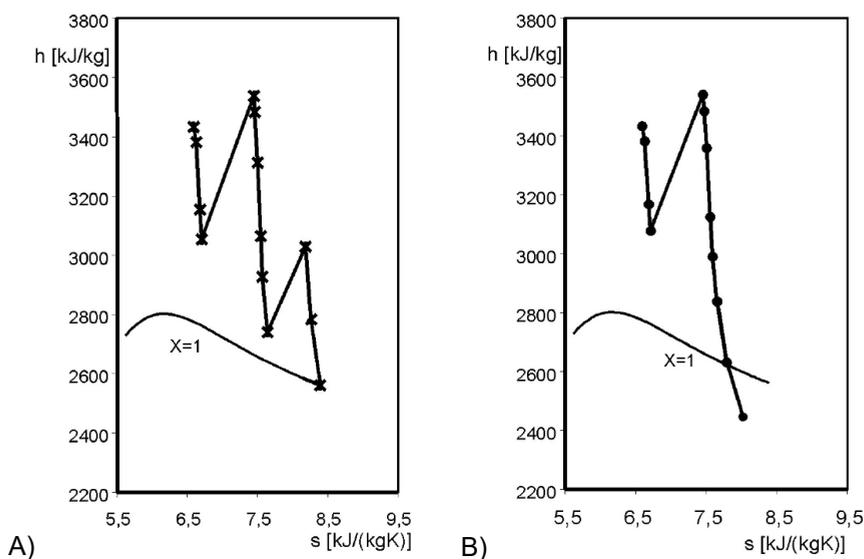


Fig. 32. A comparison of the expansion lines with and without superheating:
 A) a system with internal superheating, B) system without superheating

The basic operating parameters assumed for the supercritical power unit are:

Pressure of live steam from the boiler	285 bar;
Temperature of live steam from the boiler	600°C;
Pressure of steam upstream of the superheater	55.442 bar;
Pressure of steam upstream of the medium-pressure part of the turbine	49.987 bar;
Temperature of steam downstream of the superheater	620° C;
Pressure in the condenser	0.05 bar.

For the components of this circuit, the boiler efficiency of 0.95 was assumed. The internal efficiency of all groups of stages was assumed to be constant and equal to 0.9. Loss of mechanical energy in the turbine – 600 kW. Generator efficiency – 0.988. There is an auxiliary turbine driving the feedwater pump in the system and the internal efficiency of the auxiliary turbine is 0.85. Efficiency of the boiler feed water pump – 0.85.

Efficiency of the regenerative heaters – 0.995.

Capacity achieved in full load conditions – 600 MW.

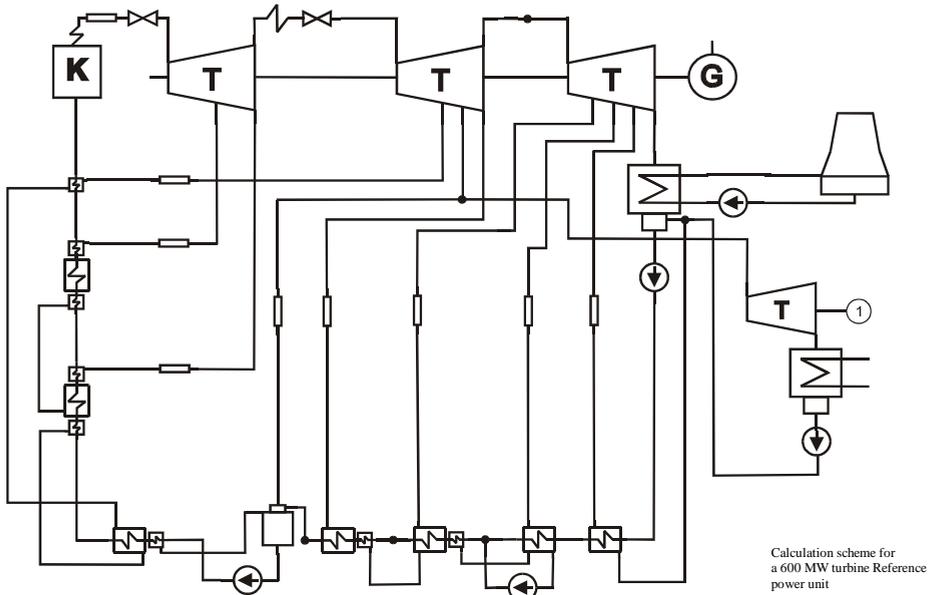


Fig. 33. A schematic diagram of the reference thermal circuit of a supercritical power unit without internal superheating

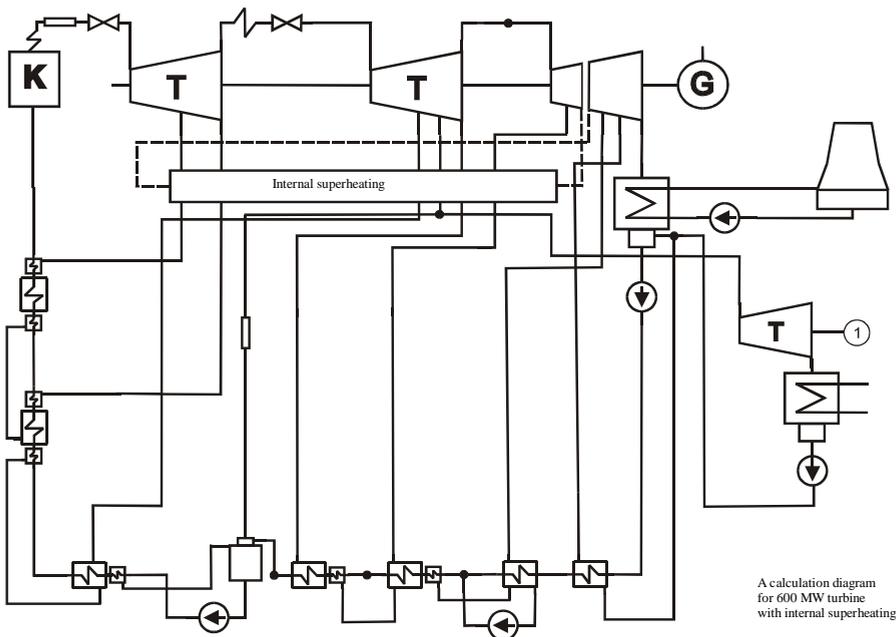


Fig. 34. A schematic diagram of the reference thermal circuit of a supercritical power unit with internal superheating

The calculations were carried out for a circuit without internal superheating at the set mass of the steam flowing from the boiler equal to 438.791 kg/s, taking into account the impact of steam humidity on turbine efficiency. Efficiency of the thermal circuit was 0.49394. Unit heat consumption was 7.288.39 kJ/kWh.

The calculations for the circuit with superheating were carried out in a similar manner at the set mass of the steam flowing from the boiler equal to 438.791 kg/s. Efficiency of the thermal circuit was 0.49734. Unit heat consumption was 7.238.51 kJ/kWh.

As previously noted, systems with superheating require much higher steam pressures in the outlets than systems without superheating. If the steam pressures in the outlets supplying regenerative exchangers are changed, the efficiency of the circuit with superheating will increase and the efficiency of the circuit without superheating will decrease. For increased pressures at the outlets for the circuit with superheating, the calculated efficiency is equal to 0.503 and the unit heat consumption is equal to 7.156 kJ/kWh. A significant change takes place in the quantity of live steam taken from the boiler for the capacity of 600 MW. The live steam mass flow rate is then equal to 467 kg/s. Additionally, the temperature of the water flowing into the boiler will change. For the new pressures, it is equal to 311°C, while for the original pressures the temperature of the water flowing into the boiler was 289.6°C.

As part of the Strategic Project Advanced Energy Acquisition Technologies, a number of analyses were carried out for a 900 MW power unit in task 1. As the calculations show, the use of internal superheating makes it possible to increase power unit efficiency by 0.7%. In the studies carried out, the pressure distributions at the turbine outlets were optimised and the pressure upstream of the low-pressure part of the turbine was analysed. The calculations show that the optimum pressure upstream of the low-pressure part is 0.8 bar. An important effect of internal superheating is the reduction of humidity of the steam at the turbine outlet. As a part of this task, calculations were performed for the exchanger that constitutes the internal superheater. The concept of a shell-and-tube heat exchanger was adopted, with the heating steam flowing inside the tubes and the steam being heated flowing outside the tubes. To intensify the heat exchange on the heated steam side, finned tubes are used. This design of the heat exchanger minimised the pressure losses on the heated steam side. As a part of the work, two parallel heat exchangers were spatially planned.

Figure 35 shows the foundation of the heat exchangers. The exchangers are set vertically, which limits the space required for their foundation. A lot of information about internal superheating can be found in the publication [Report from the Strategic Project... 2013, 2014, 2015]. The dimensions of the exchanger that constitutes an internal superheater are considerable, as its width is 5 m.

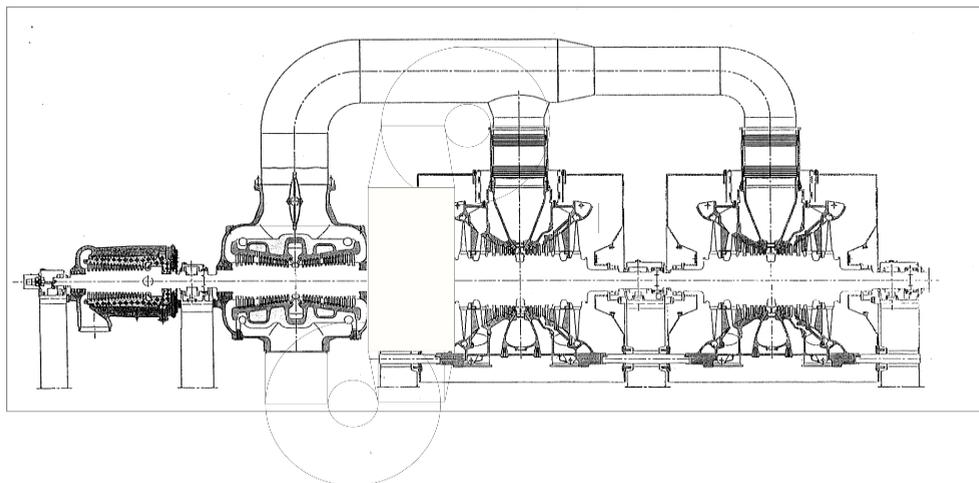


Fig. 35. A 900 MW turbine with internal superheating

CONCLUSIONS

In conclusion, the problems of carnotisation of thermal circuits involve several issues. The first is the impact of the number of exchangers on the efficiency of the circuit. The second concerns the different types of exchangers that can be used in the recovery system. Of note are the large differences in the efficiency of the circuits depending on the type of exchangers used in the recovery system for a finite number of them. Adding an internal superheater in the thermal circuit increases its efficiency, but on the other hand it leads to less efficient operation of the recovery system. With the ever-higher parameters of live steam, one has to account for significant superheating of steam taken to the regenerative exchangers. This results in a loss of exergy and thus a decrease in the efficiency of the circuit. In order to reduce these losses and increase the efficiency of the recovery system, it was proposed to use internal superheating. For a supercritical block, the balance calculations indicate a possible increase in the efficiency of the system with internal superheating by about 0.6%.

The topics presented herein make it possible to evaluate the impact of structural changes in the thermal circuit on its efficiency. The tendency to continuously increase the initial parameters of thermal circuits and the introduction of multiple interstage superheaters is justified by the need to increase the efficiency of steam turbine systems.

REFERENCES

- Callen, H., 1985, *Thermodynamics and an Introduction to Termostatistics*, J.Wiley, New York, USA.
- Chmielniak, T., 1988, *Obiegi termodynamiczne turbin ciepłych*, Ossolineum, Wrocław.
- Chmielniak, T., 2004, *Technologie energetyczne*, Politechnika Śląska, Gliwice.
- Cwilewicz, R., Perepeczko, A., 2014, *Okrętowe turbiny parowe*, Wydawnictwo Akademii Morskiej w Gdyni, Gdynia.
- Davies, J., 1991, *Modern Power Station Practice*, Pergamon Press, Oxford, UK.
- Krzyślak, P., 2006, *Nowe koncepcje wzrostu sprawności obiegów ciepłych z turbinami parowymi*, Rozprawy, no. 401, Politechnika Poznańska, Poznań.
- Krzyślak, P., Lippert, T., 1995, *Die Gradientenmethode zur Berechnung von Dampfkreislaufen*, Interner Bericht ITSM Stuttgart, Stuttgart, Germany.
- Marecki, J., 2000, *Podstawy przemian energetycznych*, Wydawnictwo Naukowo-Techniczne, Warszawa.
- Perycz, S., 1988, *Turbiny parowe i gazowe*, Politechnika Gdańska, Gdańsk.
- Report from the Strategic Project Advanced Energy Acquisition Technologies task 1, Collective work [Raport Projektu Strategicznego Zaawansowane Technologie Pozyskiwania Energii zad. 1, praca zbiorowa], Gliwice 2013, 2014, 2015.
- Staliński, J., 1955, *Siłownie okrętowe*, Wydawnictwo Komunikacyjne, Warszawa.
- Traupel, W., 1988, *Thermische Turbomaschinen*, Springer, Berlin, Germany.