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Matching of a Gas Turbine and an Upgraded Supercritical Steam Turbine in Off-Design Operation

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Abstract

The results of the research presented in this paper refer to an upgrade of a power generating unit with a supercritical steam turbine of a moderate live steam temperature. The upgrade project involves replacing the high pressure part with a turbine driven by a high live steam temperature and adding a gas turbine. The aim of the upgrade is to increase power generation efficiency. The specific problem analyzed in the paper corresponds to the matching between the gas and steam turbine especially in off-design conditions. The analysis is based on the numerical modeling of the thermodynamic cycles. Various conditions of the operation were simulated. The results obtained enable efficiency to be assessed in the analyzed variants.

Keywords: supercritical steam turbine, gas turbine, upgrade, combined cycle

1. Introduction

The working history of steam turbines indicates that at some stage they typically undergo an upgrade which involves the application of new technologies. The upgrade project described in this paper focuses on power generating units that are relatively new or are currently under construction. The reason for the upgrade is that the best available technology is not fully applied to the units. The turbines are to operate at levels of steam pressure and temperature that are lower than the allowable values specified for the materials [1].

Modern turbines operate with steam at supercritical levels. The live steam temperature reaches 560°C at pressure of 27 MPa, while the reheat steam is delivered at 580°C. However, turbines operating with the live steam temperature at 600°C and reheat steam at 610°C are becoming more common [2]. Research conducted in several R&D centers seeks to raise the live steam temperature up to 650°C or even 700°C [3, 4]. Hence future upgrades to this end are not only technically available but also quite likely to happen.

The history of subcritical turbines indicates that new designs use previous designs as a basis. This approach allows one to match new components to the already existing parts of a turbine. The aim of the upgrade project described here is to replace a rotor, an inner casing and blades with new components that are able to operate at higher temperatures. The result of the upgrade is a higher live steam temperature and higher efficiency power generation.

To increase the live steam temperature, more heat must be delivered to the steam cycle. Any increase in steam temperature in existing fossil boilers is usually restricted. To increase the heat transfer surface area would require significant changes to the piping and the support system for the pipes. Since the upgrade project presented here assumes the minimum possible change to the existing steam cycle, a full-scale refit of the boiler is excluded.

Additional heat may be provided in an additional gas turbine cycle. The existing steam cycle can be combined with a new gas cycle. The project assumes a relatively

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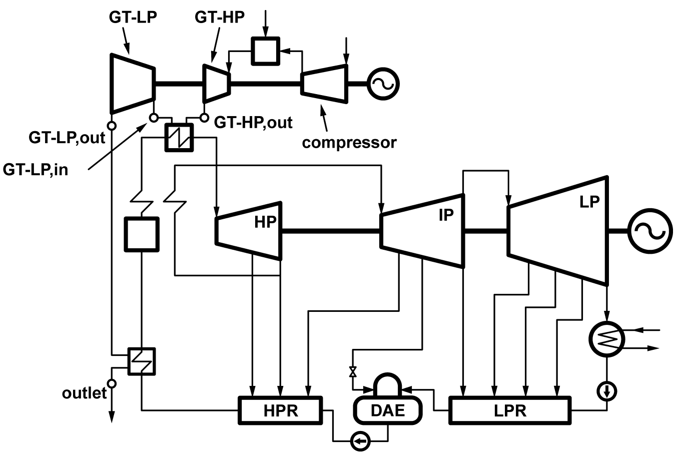

Figure 1: Upgraded power generation unit

high live steam temperature: 650°C or 700°C. These values are significantly higher than in typical combinations of gas and steam cycles. Therefore the gas path is divided into two expanders: a high-pressure and a low-pressure part. The additional gas-steam heat exchanger that raises the live steam temperature to the desired level is located between the expanders. The exchanger is fed with steam from the fossil boiler, which remains unaltered after the upgrade.

The main problem is the size of the gas turbine, which must match the existing steam cycle [5]. Choice of the new machines is based on the efficiency criterion [6]. The matching of the gas and steam cycle is analyzed for various operating conditions, including off-design operation.

2. The existing steam cycle

The cycle under investigation is a 460 MW supercritical steam turbine. In the design conditions the live steam flow rate is 1300 t/h at 560°C and 27.5 MPa. The reheat steam temperature is 560°C.

The thermodynamic cycle after the upgrade is shown in Fig. 1. The new part is the gas cycle with high-pressure (GT-HP) and low-pressure (GT-LP) expanders. The peak temperature at the combustor outlet is 1350°C, which is assumed in light of recent developments in the field of gas turbines. The power output of the gas turbine is a value under analysis.

The exhaust gas from the high-pressure expander feeds the additional (new) heat exchanger that enables the live steam temperature to be raised above its original nominal value. The analysis presented here includes two cases with a concurrent-flow and counter-flow heat exchanger.

The exhaust gas flows from the exchanger into the low-pressure expander. After the expansion process the gas temperature is still high and it may feed yet another heat exchanger: gas - feed water. The feed water temperature may be increased by no more than 10K in reference to its original design value. This limits the amount of heat that may be transferred from the exhaust gas into the feed water.

The appropriate matching of the gas and the steam cycle is an optimization task. It requires a definition of restrictions that define the possible range of the solution. The restrictions are mostly the values of the temperatures of the working fluids (gas and steam) in the characteristic nodes of the combined cycle. They are set out in Table 1 below.

3. Numerical model

The analysis investigated several cases of the upgrade. The research involves the creation of a numerical model of the cycle after the upgrade enabling each case to be compared. The model included the machines and devices that make up the thermodynamic cycle: the turbines, the steam boiler, the heat recovery system and the auxiliary equipment (feed water pump, deaerator, condenser etc.). The numerical model also employs available information about the existing cycle: performance maps for the expanders, the heat exchangers and the condenser.

The input data for the numerical model includes parameters that may be altered by the operator (live steam temperature, pressure and flow) and values that depend on the environmental conditions (pressure in the condenser).

A very important issue in the research is the modeling of the off-design operation. Usually for gas turbines the off-design conditions are modeled according to the performance maps provided by the manufacturers. Due to the narrow range of the operating conditions covered in these maps, the analysis is restricted to the states close to the design state. Therefore the numerical model described here employs a different approach: it uses the available performance maps together with the relations that describe the thermodynamic processes in the gas cycle.

The operation of a gas turbine depends on the cooperation of its main components. The analysis of the off-design operation focuses on the cooperation between the compressor, the expander and the combustor. The following list sets out the quantities that combine the models of a gas turbine sections:

- **Rotational velocity.** The compressor and the expander are on a single shaft or on two shafts with a transmission gear.

Table 1: Restrictions for the upgrade

Description	Value
combustor outlet temperature	1350°C
temperature difference between the steam outlet and gas outlet in the additional exchanger:	
- concurrent-flow	25 K
- counter-flow	0
feed water temperature	295.5°C
temperature difference between the exhaust gas outlet and feed water inlet	40 K

- **Working fluid flow.** The amount of the exhaust gas in the expander depends on the amount of air delivered to the compressor and the fuel in the combustor. In addition one must factor in the flow losses, the cooling flows and additional flows such as for example the anti-icing system.
- **Pressure and temperature.** The parameters of the flow at the inlets are equal to the parameters at the outlets of the upstream parts of the cycle.

The cooperation of the components of a gas turbine is usually presented in the compressor performance graph. This graph describes the relation between the pressure ratio and the flow through the compressor for various inlet guide vane angles and/or rotational velocities. An example of such a performance graph is shown in Fig. 2. All the parameters are non-dimensional and given in reference to the nominal (design) values (marked N):

- non-dimensional pressure ratio

$$\beta_{bw} = \frac{\beta}{\beta_N} = \frac{\bar{p}_{out}/\bar{p}_{in}}{\bar{p}_{out,N}/\bar{p}_{in,N}} \tag{1}$$

- non-dimensional corrected flow

$$m_{Z-bw} = \frac{m_{in}\sqrt{\bar{T}_{in}}}{\bar{p}_{in}} / \frac{m_{in,N}\sqrt{\bar{T}_{in,N}}}{\bar{p}_{in,N}} \tag{2}$$

- non-dimensional isentropic efficiency

$$\eta_{bw} = \eta/\eta_N \tag{3}$$

- non-dimensional inlet guide vane angle

$$IGVA_{bw} = IGVA/IGVA_N \tag{4}$$

Since the rotational speed for the gas turbine is assumed constant in this upgrade, control of the compressor operation involves adjustment of the inlet guide vane angle. The coordinates of (1,1) represent the design conditions of operation.

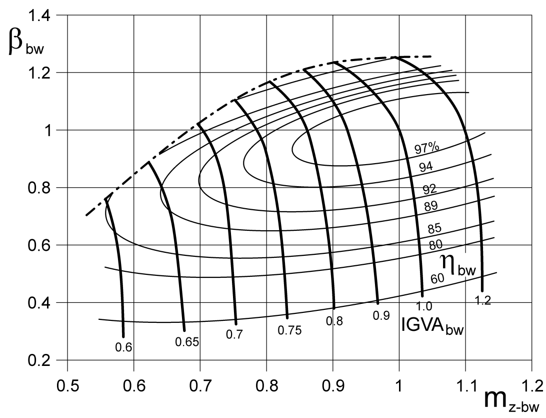

Figure 2: Performance map for a gas turbine compressor

The performance of the gas cycle in off-design conditions is determined in the following numerical procedure: the simultaneous solution of the balance equations for each component makes it possible to calculate the performance of the cycle for given operating conditions.

An example of a partial solution is shown in Fig. 3. The dotted line in the graph connects the points that represent the performance of the particular combustor and expander for a fixed amount of fuel delivered to the combustor. The combustor outlet temperature is however different in each point. The set of the balance equations for the combustor and the expander without the compressor is incomplete; hence its solution is a straight line. For a fixed amount of fuel the exhaust gas temperature depends on the amount of air delivered to the combustor.

The solid lines in Fig. 3 correspond to the solutions of the combustor (CB) and the expander (TB) balance equations. The balance equations for the combustor are solved for a given exhaust gas temperature. This temperature depends on the air temperature at the combustor inlet, which in turn depends on the air pressure. The balance equations for the combustor allow one to calculate the amount of air (m_{z-bw} - horizontal axis in Fig. 3) compressed with the pressure ratio of b_{bw} (vertical axis in Fig. 3) required to obtain the given exhaust temperature. Then the CB lines

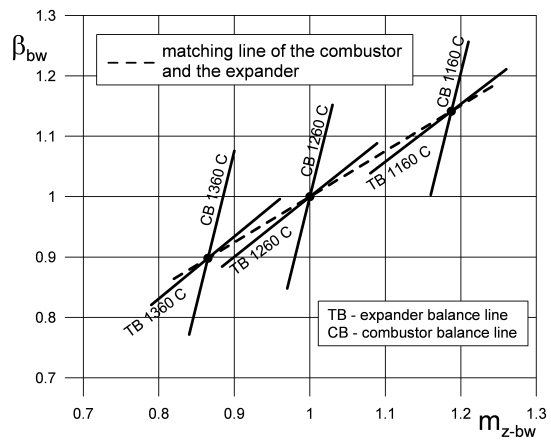

Figure 3: Performance map for a gas turbine compressor

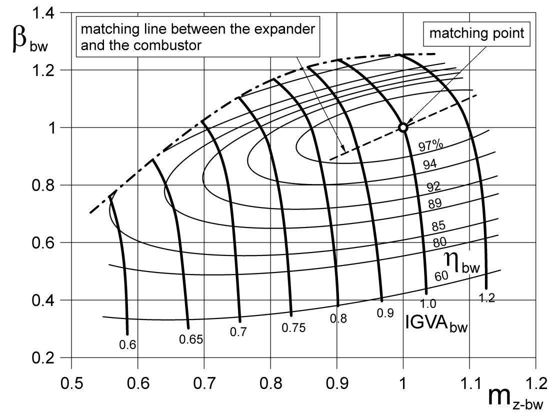

Figure 4: Design point of operation

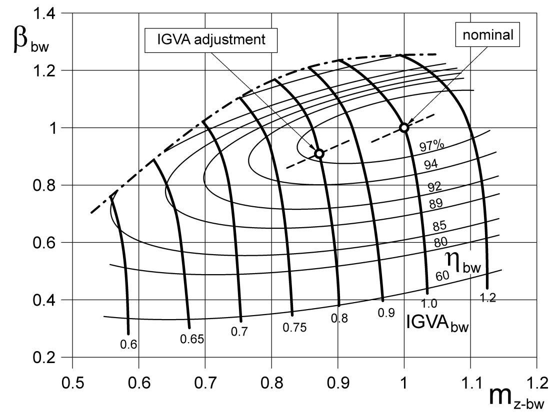

Figure 5: IGVA adjustment

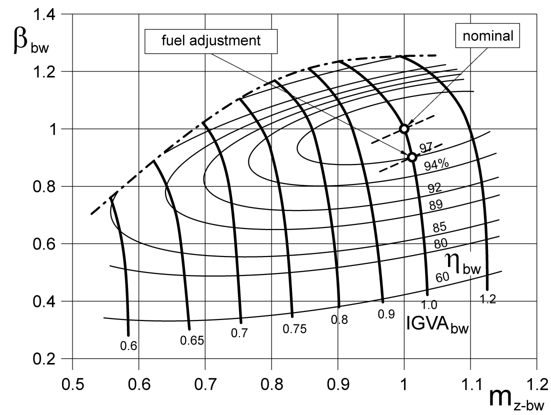

Figure 6: Adjustment of the fuel amount

in Fig. 3 show the relation between the flow and pressure ratio established for the combustor.

The lines for the expander (TB) represent the solution of the balance equations for the expander. The off-design model [7, 8] relates the pressure at the inlet to the expander to the amount of the expanding gas. The inlet pressure and the gas flow are related respectively to the compressor pressure ratio and the flow in the compressor. Therefore the solution of the balance equations for the expander may also be plotted in the b_{bw} - m_{z-bw} coordinate system. The relation between the pressures involves pressure losses in the combustor, while the relation between the flows involves the cooling flows.

The set of balance equations mentioned above is complete when the compressor is also taken into the consideration. The solution reduces from a line to a single point that is located at the intersection of the combustor-expander line (from Fig. 3) and the appropriate line for the compressor. This is shown in Fig. 4.

The change in operating conditions causes the gas cycle to operate in off-design states. The balance equations change their solution and the operating point shifts to a different location. Figs. 5 and 6 compare the operation in off-design and nominal conditions for two different cases.

In the first case the gas turbine operation is adjusted through the change in inlet guide vane angle in the compressor. In the latter case the amount of fuel delivered to the combustion chamber is lowered. The matching points of the operation are located on different lines in the graphs.

The approach presented in this section is applied to the numerical modeling of the whole cycle after the upgrade.

4. Operation after the upgrade

According to the assumptions made earlier the upgrade is evaluated with respect to the efficiency of power generation. Efficiency is expressed in the following values:

total efficiency

$$\eta = \frac{N_{el,SC} + N_{el,GT}}{Q_k \frac{1}{\eta_k} + m_g LHV_g} \tag{5}$$

- steam cycle efficiency

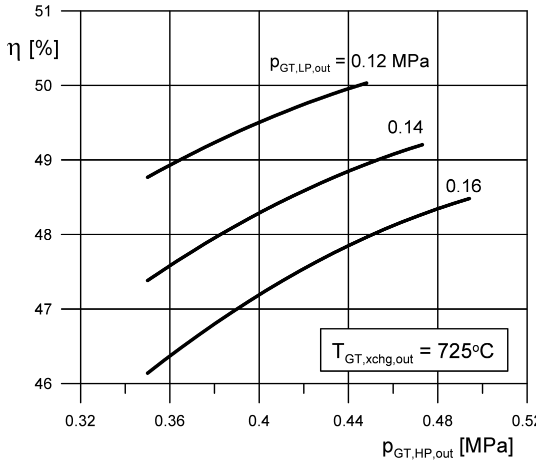

Figure 7: . Total efficiency after the upgrade (concurrent-flow exchanger)

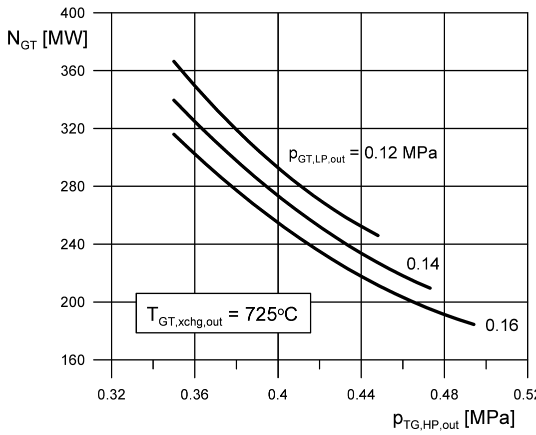

Figure 8: Power output of the required gas turbine (concurrent-flow exchanger)

$$\eta_{SC} = \frac{N_{el,SC}}{Q_k} \tag{6}$$

- gas cycle efficiency

$$\eta_{GT} = \frac{N_{el,GT}}{m_g LHV_g} \tag{7}$$

Fig. 7 presents the total efficiency of the cycle after the upgrade plotted against the pressure at the outlet of the HP gas expander ($p_{GT,HP,out}$) and the outlet of the LP gas expander ($p_{GT,LP,out}$). Fig. 8 shows the required power output of the gas turbine that must be selected for the existing steam cycle.

The two pressure levels are among the most important parameters to select in the upgrade project. They affect the temperature distribution in the gas cycle and hence the heat transfer conditions in the exchangers, which impacts the overall performance of the whole combined cycle.

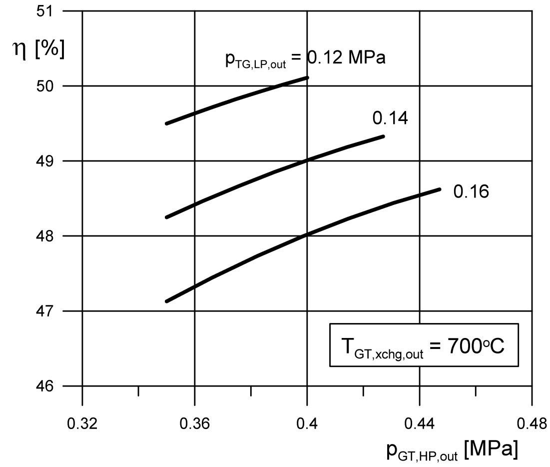

Figure 9: Total efficiency after the upgrade (concurrent-flow exchanger)

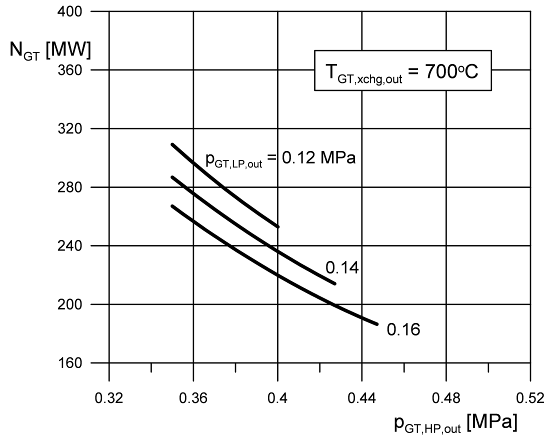

Figure 10: Power output of the required gas turbine (concurrent-flow exchanger)

Similar graphs are presented in Figs. 9 and 10 for the counter-flow heat exchanger.

Maximum total efficiency is restricted by several factors. The one encountered first in the calculations is the temperature of the exhaust gas at the outlet of the gas - feed water heat exchanger. Too low a temperature would cause corrosion problems. In this research the minimum temperature was assumed at 240°C.

The restrictions on the upgrade are assumed with great caution and may seem overly preventive. For example, the gas temperature at the outlet of the counter-flow exchanger is 700°C. This could be several degrees lower, which would of course result in a larger surface area of the exchanger, but would significantly increase total efficiency.

There is a similar issue with the steam boiler. One of the main goals of the upgrade is to limit the number of modifications to the existing steam cycle. If however the design

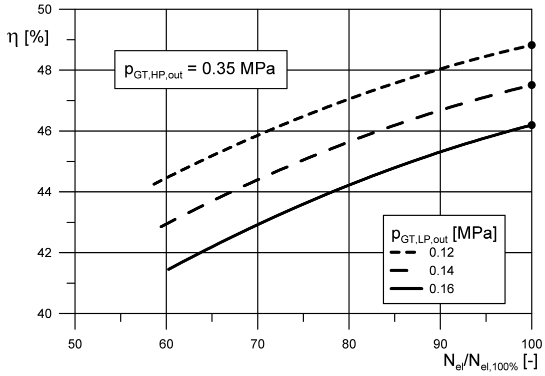

Figure 11: Total efficiency after upgrade in off-design operation

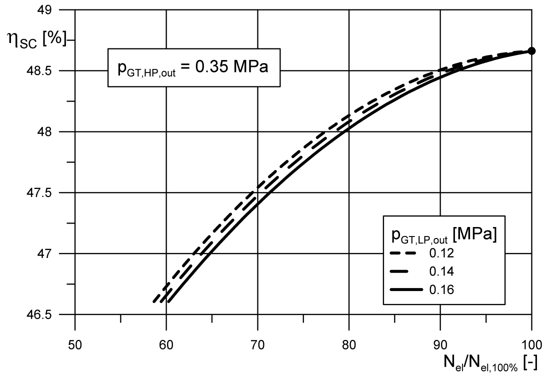

Figure 12: Steam cycle efficiency in off-design operation

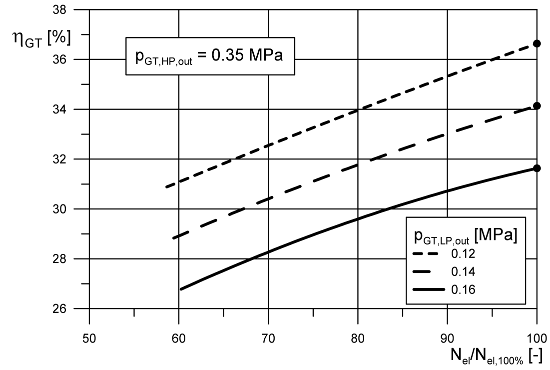

Figure 13: Gas turbine efficiency in the off-design

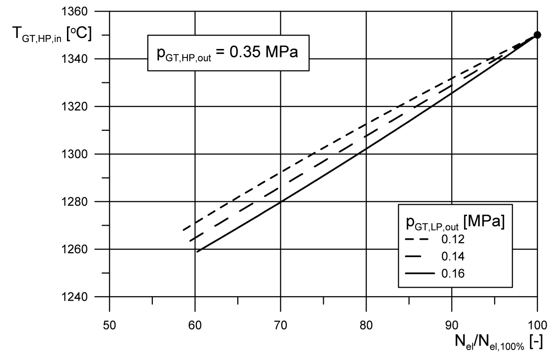

Figure 14: Combustor outlet temperature in off-design operation

of the boiler made it possible to increase the feed water temperature by more than 10K in reference to the original design conditions, then the overall performance would be much improved. Smaller load in the steam boiler leads to better total efficiency. However, a problem occurs if the smaller load in the economizer results in a higher flue gas temperature in the boiler which is unacceptable.

5. Off-design operation after the upgrade

The results of off-design operation after the upgrade are analyzed for the whole possible range of operation. It is worthwhile underlining that several parts of the original system must operate in off-design conditions, even at full load. The increase in total efficiency is much higher than the losses caused by off-design operation of some parts. The calculated total efficiency in off-design conditions is shown in Fig. 11.

The changes in efficiency of the steam and the gas turbines are shown in Figs. 12 and 13 respectively.

It is assumed that the adjustment of the gas cycle involves the change of the amount of fuel delivered to the combustor. If this type of control is not enough to decrease the power below the required level then the IGVA is also changed. The off-design conditions correspond to lower temperature in the outlet of the gas turbine combustor. This is shown in Fig. 14.

In each case the turbines were adjusted in order to achieve maximum total efficiency for a given part-load. The flows in the turbines were changed in view of the efficiency criterion.

6. Summary

The working history of existing steam turbines strongly suggests that newly-constructed supercritical steam turbines will undergo upgrades in the future. The difference between the technology applied to the turbines and the actual state-of-the-art technology is already quite large. One may expect that in the near future the difference will also become economically appealing, which means that an upgrade will be economically justified.

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Nomenclature

η_k steam boiler efficiency

LHV_g lower heating value of the fuel

m_g amount of gas fuel

$N_{el,GT}$ gas turbine power output

$N_{el,st}$ steam turbine power output

Q_k heat transferred in the steam boiler