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INSTITUTE OF FLUID-FLOW MACHINERY
POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

113

Selected papers from the International Conference on *Turbines of Large Output* devoted to 100th Anniversary of Prof. Robert Szewalski Birthday, Gdańsk, September 22-24, 2003

WYDAWNICTWO IMP PAN

GDAŃSK 2003

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Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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ISSN 0079-3205

No. 113, 2003, 3-4

Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100^{th} Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery*. At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 113, 2003, 5-24

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Steam turbines for supercritical power plants – design and operation problems

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Abstract

This paper regards design issues of large power output steam turbines for supercritical power plants. A special focus is set on features of turbine cycles, their parameters as well as on aerodynamical, material and strength problems. The second part of the paper presents an operation diagnostics system, which regards monitoring of stresses.

Keywords: Steam turbines; Supercritical stem parameters; Design; Operation

1 Introduction

Experience gathered so far, as well as new designs indicate that the classical condensation unit fueled with coal may still be considered to a technological potential. Its further development may be restricted by the progress of other technologies connected with coal and due to the availability of natural gas and its relative process, as well as by the development and wider application of gas-and-steam systems.

The fundamental condition for further development of coal-fired power stations is the considerable increase of the effective use of chemical energy of coal, meeting at the same time the growing requirements concerning the emission of noxious substances (dust, SO_2, NO_x and CO_2) into the environment.

The application of supercritical pressures, the increase of live steam parameters, endeavours to improve the structure of turbines and steam boilers as well as other devices, have made it possible to construct classical condensation units

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with an efficiency of more than 45 % (net, taking into account the power consumed by the emission control systems). Installations, the construction of which is under way, as well as intended new projects assume a further increase of the effective use of the chemical energy of fuels. The main sources of additional effects are: a more effective regeneration, the improvement of the cooling system of the condenser and the reduction of the own consumption of energy for the purpose of emission control. Investigations and designs indicate additionally, that in near future systems will be developed, the efficiency of which will exceed considerably the values attained until now. Simultaneously studies are carried on concerning the construction of machines and other installations for new power units.

2 Characteristic features of modern thermodynamic cycles of the steam turbines

The fundamental parameters of steam-water cycle and characteristics of the devices are determined by a multistage optimization, comprising, among others, the search for:

- the parameters of live steam,
- the number of reheats and their parameters,
- the optimal "cold-end" solution resulting from discussions concerning the outlet cross-sections of the turbines (number of outlets), the way of connecting the condenser, the characteristics and design of the condenser cooling system,
- the parameters of the steam-water cycle (the structure of the regeneration system, the kinds of the regenerative exchangers, the way of driving the feed-water pump etc.),
- the value of the power rate of a unit.

Finally, the economical and ecological criteria must be taken into account. The main features of the realized and designed (analyzed in research studies) condensation units for supercritical pressures have been presented in Fig. 1. Analyses of various solutions indicate that there is no disagreement as to the number of regenerative exchangers. In the low-pressure part mostly 5 or 6 exchangers are installed, whereas the high-pressure part contains 2 or 3 exchangers and a steam cooler. The solution with this latter one is generally applied.

The application of a regenerative bleeding from the casing of the high-pressure part complicates rather its construction, but has been accepted in the case of designed supercritical values of the pressure and temperature of live steam.

In order to achieve high unit efficiency a special exhaust heat regeneration

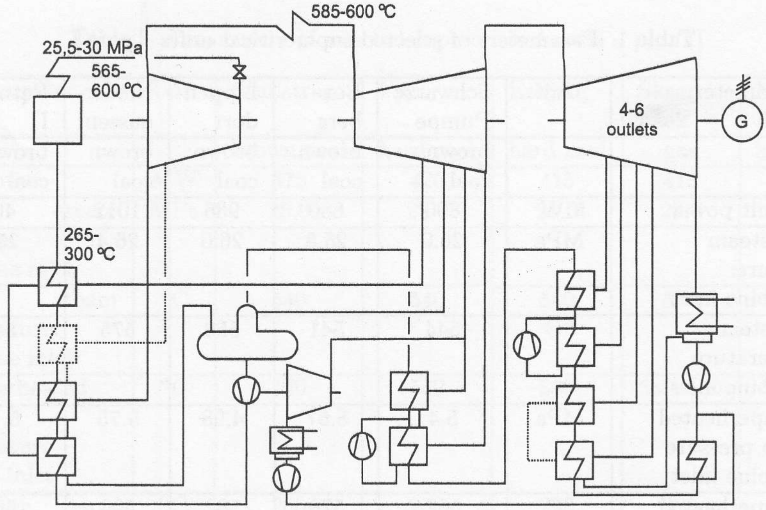

Figure 1. General thermodynamic diagram of present projects regarding supercritical pressure condensational units.

system is applied [1]. Its main components include:

- heat regeneration exchangers set of HP and LP turbine part located in a separated exhaust duct between the economizer and electrofilter,
- exhaust gas – water heat exchanger located in front of FGD (Flue Gas Desulphurisation),
- water – air heat exchanger located downstream of downcast fan.

The application of components described above decreases the amount of bleeding steam feeding the HP and LP heat regeneration exchangers, which in turn leads to increase of unit efficiency.

Tables 1 and 2 present a comparison of steam parameters of selected supercritical pressure condensation units. Table 1 shows units operating in German power plants. There are also the parameters of new supercritical power unit in Pątnów Power Plant in the last column. Table 2 lists supercritical units operating in Denmark.

3 Design features of steam turbines

The design of any machine ought to be an optimal one, i.e. it should be the best in the given conditions due to the assumption of the criterion (or criteria) of optimization. Analyzing various criteria of optimization we can specify the requirements which a modern turbine must satisfy. These are:

Table 1. Parameters of selected supercritical units – part I

Parameter	Unit	Schwarze Pumpe	Box-berg	Lippen-dorf	Nieder-aussem	Pątnów II
Fuel		brown coal	brown coal	brown coal	brown coal	brown coal
Unit power	MW	800	880	936	1012	460
Live steam pressure: – turbine inlet	MPa	26.0	25.8	26.0	26.4	25.8
Live steam temperature – turbine inlet	°C	544	541	550	575	540
Re-superheated steam pressure – turbine inlet	MPa	5.4	5.67	4.98	5.75	5.15
Re-superheated steam temperature – turbine inlet	°C	562	578	582	599	565
Feedwater temperature	°C	274	274	270	295	275
Steam pressure – turbine outlet	kPa	3.5/4.6	4.1	3.8	3.6	ca 4.5-5.0
Unit effectiveness	%	41.0	41.7	42.3	45.2	ca 41

- high efficiency,
- good availability,
- high thermal flexibility,
- slow creep and fatigue wear of turbines components,
- long time between overhauls.

In order to achieve these aims, special attention should be paid, while designing a turbine, to achieve the following tasks:

- improvement of aerodynamics of rotor blades,
- improvement of aerodynamics of other elements of the flow part, including valves, steam ducts to regenerative exchangers, outlet diffusers etc.,
- detailed analysis of the states of stresses and displacements, as well as the proper choice of materials,
- development and application of cooling methods, particularly of the rotors in the high-pressure (HP) and intermediate-pressure part (IP),
- screening details which are exposed to high temperatures.

Table 2. Parameters of selected supercritical units – part II

Parameter	Unit	Studstrup	Fyn	Esbjerg	Skaer-baek	Nordjy-lland
Fuel		brown coal	hard coal	hard coal	gas	hard coal
Unit power	MW	375	420	415	413	411
Live steam pressure: – turbine inlet	MPa	25.0	25.0	25.0	29.0	29.0
Live steam temperature: – turbine inlet	°C	540	540	560	580	580
Re-superheated steam temperature – turbine inlet	°C	540	540	560	580/580	580/580
Feedwater temperature	°C	260	280	275	300	300
Steam pressure: – turbine outlet	kPa	2.7	2.7	2.3	2.3	2.3
Unit effectiveness	%	42	43.5	45	49	47

3.1 Aerodynamic problems

The development of numerical methods in fluid mechanics and the continuous extension of information concerning the dissipation of energy in flow systems constitutes the basis for more and more reliable methods of calculations and algorithms of aerodynamic optimization. This development is achieved thanks to discarding simplifications in the mathematical description of phenomena, the possibility of taking into consideration complex geometries, a more complete description of the properties of the working media and also the application of advanced procedures in the analysis and processing of the results of calculations. And yet the question remains which kind of problems in the analysis and synthesis of fluid-flow machines can be solved at present with a sufficient reliability permitting formulation of practical design guidelines? Another question concerns the criteria of choosing the most adequate methods and numerical algorithms, increasing our knowledge about the physics of fluid-flow processes. These questions cannot be answered without experimental investigations. In spite of that our confidence and accessibility to numerical codes have doubtlessly extended their application in investigations of all kinds of fluid-flow machines. Thanks to this:

- efficiency of the conversion of energy in the fluid flow has been increased,
- reliability of investigated machines could be improved,

- process of designing has been essentially activated, and
- identification of the sources of generating dissipative effects has become more accurate.

Which problem concerning fluid-flow machines is nowadays most dealt with making use of advanced numerical codes? Basing on analyses of publications and conferences devoted to fluid-flow machines, the following problems may be pointed out:

- interaction of the blade rims (dissipative effects, transport of vortices, effects on the boundary layer, wave effects),
- effectivity of geometrical effects (influence of geometry of restricting the flow ducts, geometry of the inlet and outlet edges, the different geometrical aspects of the blades, etc.),
- effect of fluid flows in the seals on the structure of the flow,
- designing the channels, considering the criterion of aerodynamic effectiveness,
- aerodynamic aspects of blade system cooling (including the coupled problem of heat transfer),
- transonic flow with steam condensation,
- influence of flow parameters on dissipation (the effect of Reynolds number, Mach's number, the level of turbulence).

In spite of the essential development of the DNS and LES algorithms, it is still of much importance to search for adequate models of turbulence, particularly in the analysis of non-stationary effects, fluid flows with large flow separations and with a laminar turbulent transition.

Due to the complex character of the fluid flow in the channels of fluid-flow machines, no turbulence model elaborated so far seems to be universal. Linear two-dimensional models of the group $k - \varepsilon(\omega)$, modified with various functions of the wall provide good results in many cases of fluid-flows without separation and as well as stationary flows. In order to enlarge their range of applicability these models, requiring less time for calculations than potentially more accurate methods applying Reynolds stress-transport equations, are still modified.

An important way of modifying models seems to be the introduction of non-linear terms into the equation determining Reynolds stress components:

$$-\rho\overline{u_i u_j} = -\frac{2}{3}\rho k \delta_{ij} + F_{ij}\left(S_{ij}^2, S_{ij}^3, ...\right) \tag{1}$$

where S_{ij} is the strain rate tensor.

The non-linear term F_{ij} can comprise only a square term (1) or also the term

S^3_{ij} [2]. This latter solution usually warrants a better numerical stability than the former one. Another approach, proved to be correct in calculations of non-steady flows, are combined models with characteristics of non-linear models (in the region of the boundary layer) as well as of LES in the region of large eddies.

From among various ways of widening the range of the application of linear two-equation models attention should be drawn to attempts of combining them with the Reynolds algebraic model of stresses, suggested by Rodie [3] and further developed by the authors of [4] and [5]. Such a model expresses exactly the effect of curvature and acceleration in a relative fluid-flow. The application of Reynolds stress transport equations is, due to their complex form, less popular in investigations of fluid-flow machines. Available results indicate to be promising, indeed [3].

Lately, there has been an increased focus on the analysis of wet steam flow through the last stages of turbines [6-8]. The analysis regards homogeneous as well as heterogeneous condensation. Developed calculation models include also procedures for determination of two-phase flow losses [9].

3.2 General structure of the unit

Organization of the steam flow in a turbine is a function of many factors, the main ones being: power rating of the unit, number of reheats, structures of regeneration, number of shafts, number of outlets to the condenser. The unit power of the units with supercritical steam pressure ranges at present between 400 and 1000 MW. In Europe new projects usually deal with a power rating of 700-800 MW. In the USA and Japan (Hitachi project – 1000 MW) units with a higher power rating are being considered.

A double reheating complicates the construction of the high-pressure part. In the case of the Hitachi solution an asymmetric double-flow high-pressure part was applied. In other solutions, e.g. in the construction of MAN/GEC ALSTOM/SIEMENS usually a single-flow part is isolated (without a regulation stage), called the super high-pressure part, as well as two double-flow parts (both of them asymmetrical), functioning as high-pressure and intermediate-pressure parts.

An example of a four-pressure turbine, composed of two separated high-pressure parts HP1 and HP2, a intermediate-pressure part IP and a low-pressure part LP with four outlets has been illustrated in Fig. 2.

The HP1 part, having a very compact structure, comprises an inlet and fluid-flow system made of nickel alloy and the external casing made of martensitic steel. From the HP1 part the steam flows directly into the part WP2, which has been made entirely of martensitic steel, already applied for supercritical parameters. The medium-pressure part is constructed similarly as the HP1 part, i.e. the fluid-

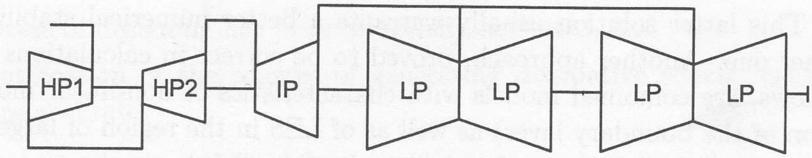

Figure 2. Turbine four-pressure system with supercritical parameters.

flow system is made of nickel alloys and the casing of martensitic steel.

In units without double reheating a single-flow high-pressure part is applied as well as a symmetrical intermediate-pressure part; this latter solution has not become a rule, though. Much is to be said for the single-flow intermediate-pressure part because of lesser secondary losses in the first stages and the elongated expansion line. Its application is limited, however, by the static and dynamic loads of the last stage. According to [12] the single-flow intermediate-pressure part may be adapted for a power rating of 1000 MW.

3.3 The number of outlets to the condenser

The decision concerning the number of outlets to the condenser is of essential importance for the economic management of the unit. The choice depends mainly on the cross-section of one single outlet proposed by the producer, as well as on the characteristics of the condenser and the conditions of its cooling. If the angular velocity is 50 1/s, the application of alloy steel generally permits to obtain a cross-section of the outlet amounting to about 9-11 m^2, and about 12-15 m^2 when titanium alloys are applied. At a given pressure in the condenser a larger number of outlets results in a reduction of outlet losses; the capital costs, however, increase. Making use of the optimization process it must be kept in mind that the conditions of cooling the condenser vary in the course of the year.

3.4 Single reheat power generation application

The available configurations for single-reheat applications are shown in Fig. 3 [11]. For most applications, an opposed flow HP/IP section in a single casing can be utilized. This section would be combined with either one or two double-flow LP sections depending on the actual rating and design exhaust pressure. The use of the combined HP/IP section makes possible a smaller overall power island with resultant savings in turbine building, foundation and maintenance costs. To meet the requirements of specialized application and customer preferences, single-flow HP and IP sections in separate casings are also available. As unit rating increases, stability requirements and last IP bucket length make a configuration utilizing a single flow HP section and double flow IP section in separate casings the appropriate selection. These two high temperature sections can be combined with one, two, or three double-flow LP sections depending on the design exhaust pressure. Tandem compound configurations of this type with three IP sections are capable of the highest unit ratings currently contemplated for ultrasupercritical power plants.

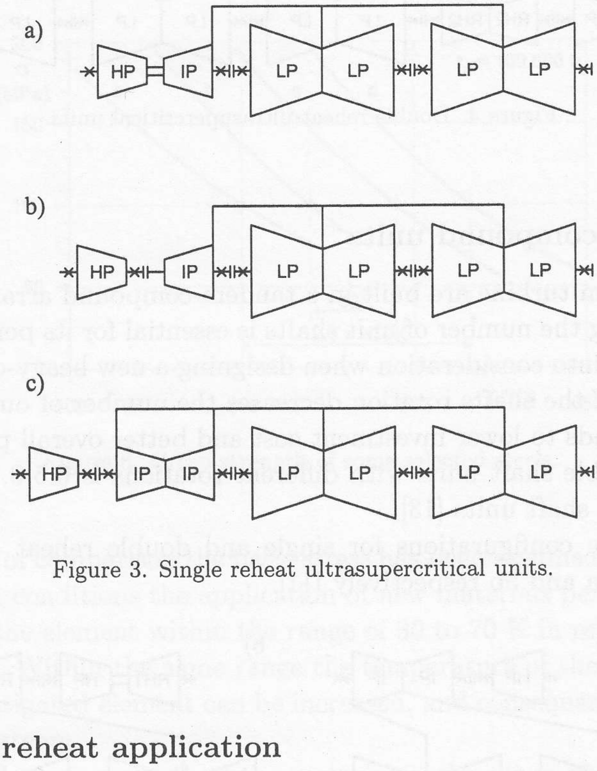

Figure 3. Single reheat ultrasupercritical units.

3.5 Double reheat application

The available configurations for double reheat applications are shown in Fig. 4 [11]. For many applications, a single-flow HP section in its own casing can be combined with a second casing having the two reheat sections in an opposed flow arrangement. The high pressure and reheat sections are directly coupled to one two, or three double-flow LP sections depending on the application rating and design exhaust pressure.

For units of higher rating, a configuration with a single-flow HP section and single-flow first reheat section located in a common casing and coupled to a double-flow second reheat section in a separate casing is utilized. As with the configuration described above, the high temperature sections are directly coupled to one, two, or three double-flow LP sections based on the rating and exhaust pressure.

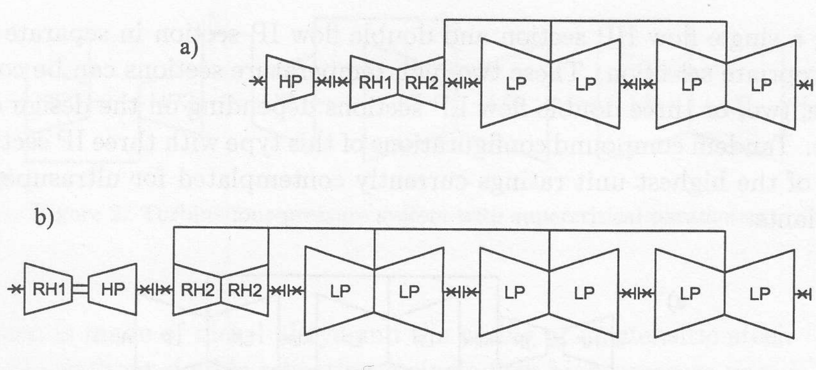

Figure 4. Double reheat ultrasupercritical units.

3.6 Cross – compound units

Usually steam turbine are built in a tandem-compound arrangements. However, determining the number of unit shafts is essential for its performance and it should be taken into consideration when designing a new heavy-duty power unit. Diversification of the shafts rotation decreases the number of outlets to the condenser, which leads to lower investment cost and better overall performance.The heat rate of double shaft units with different rotations is 0.5-0.7 % lower when compared to one shaft units [13].

The available configurations for single and double reheat applications are shown in Figs. 5a and 5b respectively [11].

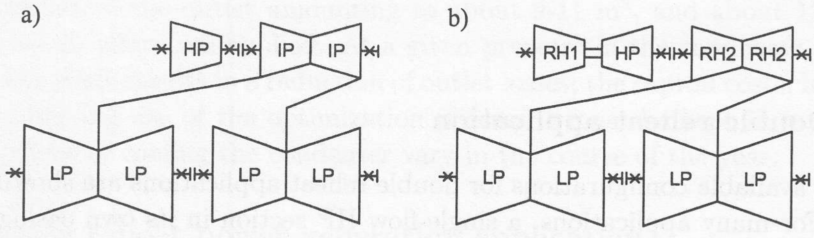

Figure 5. Cross-compound units.

4 Materials selection, turbine components cooling and stress analysing

The achievements of material engineering in the range of materials applied in energy engineering are considerable. Quite a number of materials has been developed which can successfully operate at elevated temperatures [14, 15]. Without going into details, let's pay our attention to the fundamental properties of these new materials, particularly their creep strength.

Fig. 6 shows the creep strength of materials applied so far (materials a and b) in comparison with the new ones (materials c and d) during the working time $tB = 100,000$ h. Chemical composition of the compared materials has been presented in Table 3 [14].

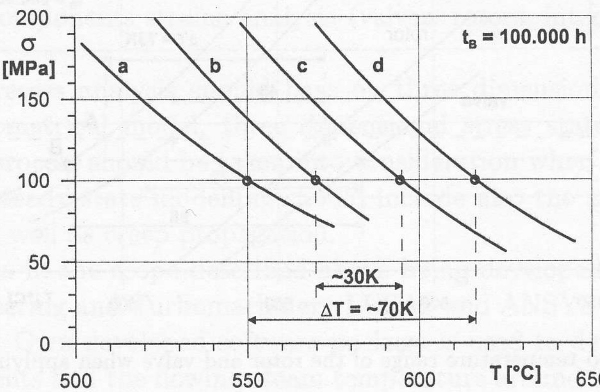

Figure 6. Creep strength of some selected steels.

For the sake of comparison the basic stress has been assumed as $\sigma = 100$ MPa (Fig. 6). In such conditions the application of new materials permits to raise the temperature of the element within the range of 30 to 70 K in relation to conventional materials. Within the same range the temperature of the medium flowing around the investigated element can be increased, and consequently also the temperature of live steam.

An increased creep strength of the material in the range of elevated temperatures results in a slow-down of its wear and prolongation of the life time of the given element.

Table 3. Chemical composition of the investigated steels.

Steal	C	Cr	Mo	W	V	Nb	N	B
a	0.28	1.0	0.9	–	0.30	–	–	–
b	0.21	12.0	1.0	–	0.30	–	–	–
c1	0.12	10.0	1.5	–	0.20	0.05	0.05	–
c2	0.12	10.0	1.0	1.0	0.20	0.05	0.05	–
d	0.18	9.0	1.5	–	0.25	0.05	0.02	0.01

Fig. 6. provides a comparison of the creep strength of some selected ma-

terials in the course of $t_B = 100,000$ hours of operation. In this comparison $\sigma = 100$ MPa has been assumed as the basic stress. Fig. 7 presents a similar comparison concerning the rotor and valve of a large power output steam turbine. It has been assumed that the considered elements ought to operate for 300,000 hours, and that the creep stresses amount to $\sigma^c = 90$ MPa in the case of the rotor and $\sigma^c = 75$ MPa in the case of the valve. These results indicate in

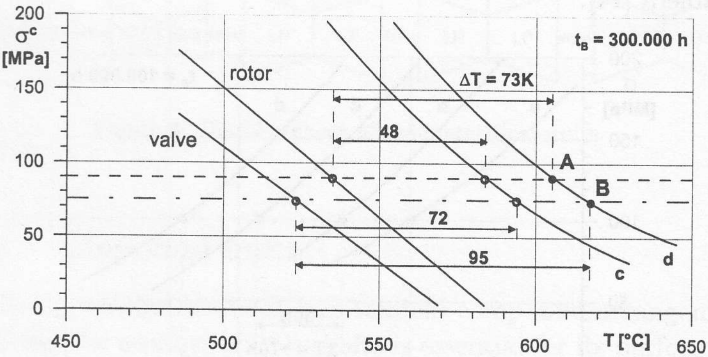

Figure 7. Increased temperature range of the rotor and valve when applying new materials.

fact that the temperature of the element can be still raised if new materials are employed. This is also confirmed by the characteristics shown in Fig. 8.

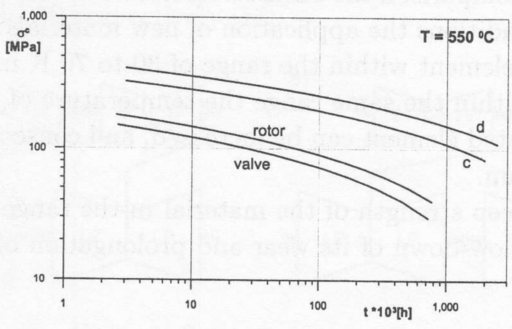

Figure 8. Creep characteristics comparison.

The range of the application of new materials may be restricted by cooling the structural elements in the region of highest temperatures. Analysis of thermal states of principal turbine elements has shown that cooling is indispensable in the following regions:

- control stage of the high-pressure part,

- first two stages of the intermediate-pressure part of turbines with single reheat [12-16].

The increase of initial steam parameters leads to higher turbine components loads. Rotor mass loads caused by centrifugal forces of rotor weight are the only ones that remain the same. Surface loads, which are caused by steam pressure, increase. Thermal loads also change. Regarding the above, in the case of supercritical turbines, there are higher standards required for stress state as well as HP and IP components strains analysis (valves, rotors, internal and external casings).

Thermal stresses analysis should base on three dimensional models (three dimensional geometrical model, three dimensional stress state). Components material creep process should be taken into consideration when analysing steady states. The unsteady state modelling should include also the analysis of elasto-plastic states as well as creep propagation.

In researches in the scope described above being developed at the Institute of Power Engineering and Turbomachinery LUSAS and ANSYS commercial software is applied. Own developed software package is used to determine the heat transfer coefficients and the flowing steam temperature around investigated components.

5 Optimisation of starting processes

A very important design and operation problem, when regarding new turbines, is a development of optimal start-up conditions. Start-up procedures are determined by start-up curves. An example of such curve is shown in Fig. 9.

In [17] a possibility of start-up time reduction was analysed. For reference, start-up complying with the guidelines of the manufacturer (Fig. 9) was assumed, for which the maximum intensity of stress occurring in the components was determined.

In simulated investigations of the effect of turbine start-up acceleration on thermal and stress state of its components finite elements method was applied. Particularly the rotors and casings of the HP and IP parts were analysed in detail. Calculating the rotor the mass load caused by the centrifugal force of the rotating mass was taken into account, as well as the irregular distribution of temperature and the surface load due to the pressure of steam. In calculations of the casings the irregular preheating and load of the steam pressure had to be taken into account. For boundary thermal conditions the temperature distribution in the flow system was assumed. The heat transfer coefficients depend on the thermodynamic parameters of the medium, the kinetic parameters of the medium and the kind of the surface of analysed components [18].

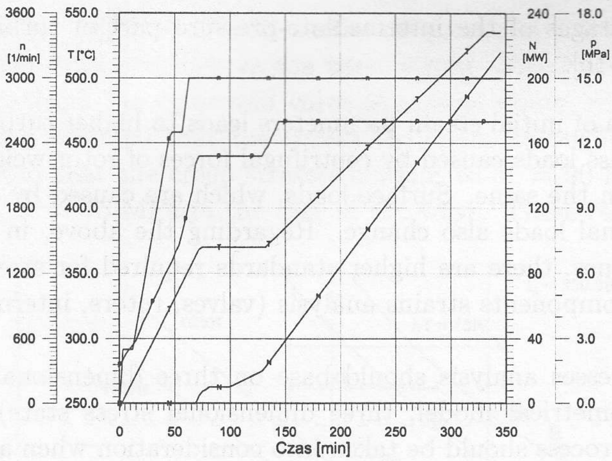

Figure 9. Manufacturer's start-up curve from the cold state of a large power turbine.

By increasing the rate of temperature rise and power output the starting time of the turbine is shortened, and this leads to higher stresses. Fig. 10 shows the relative change of maximum stresses (without taking into account the concentration of stresses in the thermal grooves) as a function of relative time of starting for the rotor and the HP casing. The time scale has been stated as the percentage of the time assigned to the start according to the characteristics provided by the producer. The determined dependence has been marked as areas, because the maximum stresses depend not only on the shortening of the starting time, but also on the way in which the parameters of steam and power output are being raised within the time that is at our disposal.

As can be seen from the diagrams, the range of possible stresses in a given starting time is rather wide. Thus, for instance, in the case of a starting time shortened by 40% – if compared with that suggested by the producer – a difference of stresses approaching 50 MPa was reached.

A slight reduction of the starting time does not lead to any radical changes in the state of the casings and rotors of the turbine. An intensive increase of stresses can be observed only when the starting time is considerably shortened. This effect might still be reduced to minimum. The process of starting the turbine can always be optimised in such a way that the increase of stresses in the respective components of the turbine is kept as low as possible in the given condition. Fig. 11 describes an adequate optimisation problem of this kind.

The optimised quantities comprise:

- the temperature characteristics of starting the turbine in the course of time,

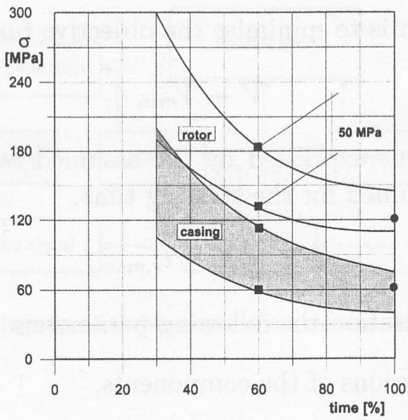

Figure 10. Stress changes in rotor and casing of HP part with start-up shortening.

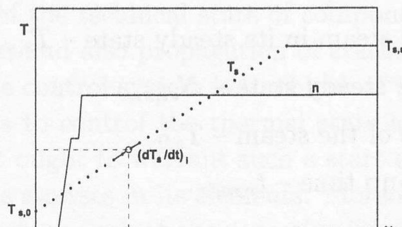

Figure 11. Description of start-up curves optimisation.

i.e. the dependence of the derivative of the temperature T_s of live steam:

$$x = \frac{dT_s(t)}{dt} \tag{2}$$

- changes of the power N of the turbine in time

$$x = \frac{dN_{el}(t)}{dt} \tag{3}$$

As the objective function the maximum stress σ_{max} in the components is assumed

$$V = max(\sigma_{max}) \tag{4}$$

The aim of optimisation is to minimise the objective function

$$V \rightarrow V_{min} \tag{5}$$

keeping to the conditions expressed by the assumed restriction. In our case a limitation has been assumed for the starting time.

$$\varphi_i = t_r \leq t_{r,max} \tag{6}$$

In the process of optimisation the following parameters have been assumed:

- geometrical dimensions of the components,
- properties of the material,
- initial temperature of the metal – $T_{m,0}$,
- rotations – $n(t)$,
- temperature of the steam in its steady state – $T_{s,s}$,
- power rating in the steady state – $N_{el,s}$,
- initial temperature of the steam – $T_{s,0}$,
- maximum starting-up time – $t_{r,max}$.

The algorithm of solving the afore-said problem of optimising the turbine start-up as well as exemplary results of these investigations may be found in [19, 20].

6 Diagnostic supervision and control system

The operation of turbines for supercritical parameters, for sake of high thermal loads of the elements ought to be controlled by means of the most modern systems of diagnostic supervision and control [21-26]. A general structure of such system is to be seen in Fig. 12, in which the two basic modules of the systems MONITORING and BOTT have been distinguished. In each one of these modules an own solution of computer-aided monitoring and control of the operation of steam turbines has been suggested, taking into account the criteria of durability and effectiveness. The aim of this proposed system is to control the effectiveness of turbine operation, a continuous control of operation as well as running diagnostics of the main elements and control of transient processes connected with the starting of the operation and change of the power rating.

The starting point of monitoring the state of the turbine is a continuous (or quasi-continuous) measurement of adequately chosen characteristic quantities. These measurements are carried out within the frame of a central registration of

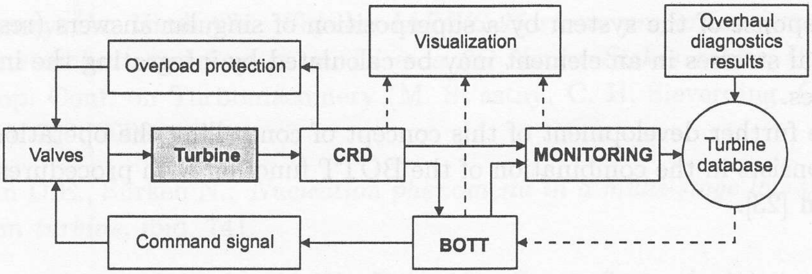

Figure 12. General diagram of turbine operation control and monitoring system.

data (CRD).

The range of MONITORING (operation diagnostics) comprises determination of the effective turbine operation, analysis of thermal states and strength, as well as the assessment of the technical state of components (thermal load, creep stability and fatigue life, and also propagation of cracking).

The chief part of the control system is the block of thermal stress limitations (BOTT), which permits to control the thermal state and strength of main elements of the turbine. It ought to warrant such a start of the turbine that would not cause any dangerous stresses in its elements. Moreover, it should protect the main elements of the turbine against the excessive intensification of wear.

Practically the control of thermal states and strength can be realized by means of various methods, to which correspond various versions of the block of thermal stress limitations of the turbine.

Previous versions of BOTT correlated the level of maximum stresses in some given element with derivatives of the temperature of the metal in some adequately selected point

$$\sigma_{max} = f(dTm/dt) \tag{7}$$

The accuracy of this chosen correlation depended, however, on the proper choice of the measurement point, and hence indications for some elements may have deviated from the real state [21].

In latest solutions of the block of thermal stress limitations at the Institute of Power Engineering and Turbomachinery Green's function and Duhamel's integral [22, 23] have been applied.

For any non-stationary thermal state, stresses were determined by functions of influence in the form of Green's function and Duhamel's integral.

The functions of influence determine the effort of the material per step function, i.e. of the temperature of medium flowing around the component. Discretizing the continuously changing input function into a sequence of small (within the range of infinitely small) stepwise changes of temperature, we replace a continuous response of the system by a superposition of singular answers (responses). The total stresses in an element may be calculated by integrating the individual responses.

The further development of this concept of controlling the operation of turbines consists in the combination of the BOTT function with procedures of optimization [23].

7 Conclusions and final remarks

The progress in steam turbines development bases on the utilization of various domains of science and technology. Numerical methods of fluid dynamics allow to design turbine blades with lower energy losses. The progress in material engineering allowed the construction of thermal cycles of ultrasupercritical steam parameters.

Increased parameters of live steam and re-superheated steam, as well as higher requirements concerning thermal elasticity have raised the importance of problems dealing with the strength and reliability in the design and construction of steam turbines. Of much importance is the continuous control of the state of stresses and its utilization of transient processes in the operation of the units. It should be stressed that this is an essential condition permitting a prolongation of the life of turbines (beyond the 250,000-300,000 h assumed so far).

Received 20 June 2003

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