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NON-STATIONARY TEMPERATURE STRESSES IN THE INDUSTRIAL STEAM TURBINE ROTOR

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ABSTRACT
The usage of industrial steam turbines in different industrial branches (chemistry, petrochemistry, refineries, sugar and ethanol plants, etc.) for a generator drive for electricity generation or a mechanical drive for compressors, blowers and pumps, is characterized by the need for high flexibility of operation. High flexibility includes numerous start-ups, shut-downs and power changes during the useful life. Changes in power and steam mass flow lead to changes of the working fluid state in the single turbine stages, and thus their aerodynamic and thermodynamic characteristics. During these transient working regimes in steam turbine rotors, large space and time-dependent temperature gradients appear, which can result in high non-stationary temperature stresses, i.e. increased local stress concentrations, what has a negative impact on the useful life of the rotor. In the worst case they can cause fracture of the turbine rotor. Today, for the determination of thermal stressed state of the steam turbine parts the user softwares based on numerical methods are used. In this paper the results of numerical modelling and calculations of non-stationary temperature fields and related stresses in the rotor of industrial steam turbine of 35 MW power during transient operating regime (a cold start-up) will be presented. The results of the calculations serve for estimation of the transient regime impact on the stresses of the rotor, as well as on its entire useful life.
Key words: industrial steam turbine, transient regimes, temperature stresses, numerical modelling

1. INTRODUCTION

Industrial steam turbines convert thermal energy of steam into mechanical work necessary for mechanical drive of pumps, fans and turbo-compressors or for electric generators drive in cogeneration power plants. They are different from the turbines in classical power plants by power and design. Usually, these are back-pressure and condensing turbines with one or two automatic extractions.

During their useful life industrial steam turbines are subjected to high values of thermal, static and dynamic loads, which in stationary and non-stationary (transient) conditions lead to compound stresses, which are composed mainly of stresses due to pressure differences of working fluid and the environment, centrifugal forces, temperature stresses, dynamic stresses and interference tension of shrunk-on elements. Compound stresses in stationary and non-stationary working regimes, through different mechanisms of material degradation, lead to the reduction in useful life of vital components.

From industrial steam turbines a long and reliable operation is required with simultaneous high flexibility: numerous start-ups, shut-downs and power changes during the useful life. Such conditions lead quickly to low-cycle fatigue of critical components such as turbine rotors and casings.

Today's standards for industrial steam turbines (API 611, API 612) prescribe a useful life for turbine of at least 20 years (minimum of about 150,000 working hours) [1].

The aim of the useful life calculation of parts operating at high temperatures (higher than 400 °C) is to determine material exhaustion of turbine components during non-stationary and stationary working

regimes. In calculation, the total exhaustion of useful life is divided into exhaustion because of low-cycle fatigue in non-stationary regimes and creep in the stationary regime [2]. Steam turbine parts which operate at high temperatures are designed for the finite useful life at the stresses that occur in stationary working regime. Repeated thermal cycles can lead to complete exhaustion of the available useful life on fatigue, causing the appearance of surface cracks. Further cycles tend to increase the depth of cracks in the element, leading to problems such as leaks through steam turbine casing or steam turbine rotor vibration problems. After the expiration of two thirds of the design useful life of industrial turbines, there is need to estimate the remaining useful life considering the known working regimes in the previous period [2].
The results of impact analysis of non-stationary regimes on temperature fields and stresses will give guidance and basis for the calculation of the remaining useful life, and making technical and economic analysis for the replacement and reconstruction of turbine parts (revitalization), change of the operation regimes, determination of new interval between the specific material testing and overhaul, what achieves higher reliability and safety of the turbine operation and useful life extension of its vital components.
This paper presents the results of the analysis of temperature fields and associated stresses in industrial steam turbine rotor at a cold start-up turbine (non-stationary regime) due to a change in the steam flow, which are obtained by numerical simulation based on the defined thermal and mechanical boundary conditions. The results show the impact of transient regime on temperature fields and stresses in the rotor and provide possibilities for useful life i.e. remaining useful life assessment of turbine components.

2. MODELLING OF NON-STATIONARY TEMPERATURE STRESSES

Start-up is the most critical procedure in turbine operation. Improper start-up may have consequences that can be divided into two groups. The first group includes some problems, such as touching contact, which occurs immediately at the start; the second group includes certain phenomena which are noticeable much later, buckling or cracking. To minimize the negative effects of start-up, the start-up of industrial steam turbines which operate with high temperature steam requires following of start-up graphs, according to which the increase in rotor speed and load are determined by time period, Figure 1 [3].
The time required to start-up of the turbine depends on a number of factors, of which the basic are the temperature of the turbine casing and the relative elongation of the rotor. It is in turn closely related to the duration of downtime since the last operation and to the manner in which the turbine is stopped. Basically, there are two extreme cases of the turbine start-up: cold start-up and warm start-up. Other cases of the turbine start-up are located between these two extremes and are engage in the operation of the turbine in warm conditions.

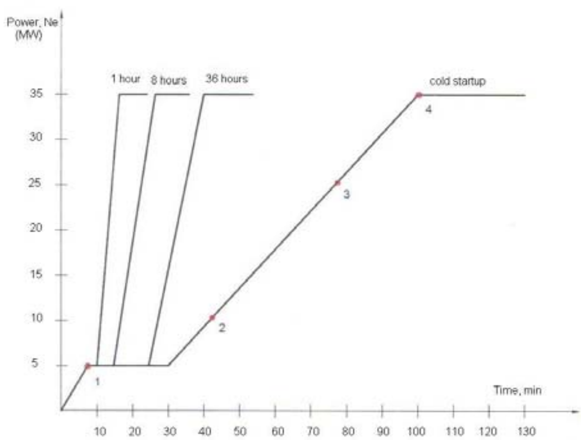

Figure 1. Start-up graphs of the industrial steam turbine of 35 MW power [3]

The object of analysis is the single-casing, condensing industrial steam turbine of the impulse type and it is the prime mover of the electric generator. The effective power of the turbine is 35 MW. It consists of 21 stages. On the entrance into the high-pressure part of the turbine there is the impulse-type controlling stage with two rows of moving blades (the Curtis turbine), which is followed by 3 high pressure stages. On the entrance into the low pressure part the impulse-type controlling stage is placed with a single row of moving blades followed by 16 low-pressure stages. Between the high and the low pressure part the turbine features automatic extraction of steam at 4.2 MPa for heat production for technological requirements. The moving blades are located on discs which have been forged together with the shaft, except the disc of the last turbine stage which is set with shrunk on a turbine shaft. Therefore, the rotor of the analysed steam turbine has been designed as a mono-block (of a single piece) with the central bore with radius 0.045 m.
For efficient estimation of turbine operation reliability in non-stationary working conditions, in this case caused by change in the steam flow rate, it is necessary to know as accurately as possible the non-stationary temperature field in steam turbine rotor, as well as the resulting thermal deformations and stresses.
Today, for the calculation of temperature fields, thermal deformations and stresses in various exploitation regimes, the methods of numerical modelling by means of sophisticated user software programs (software packages) are used, based on the method of finite elements, as described in [4, 5].
The analysed problem of non-stationary thermal and mechanical phenomena in the steam turbine rotor is in principle a three-dimensional one.
The thermal boundary conditions are set for the temperature field calculations on the surfaces of moving blades, in the labyrinth glands, and on disc surfaces by means of convective heat transfer coefficients, which have been calculated by means of original generalized statistical correlations from [6]. On the surfaces of the rotor sleeves which are under the influence of oil temperature in bearings, the constant temperature boundary conditions are set.
The mechanical boundary conditions are determined from the design characteristics of the rotor: zero axial shift of the rotor on the disc of thrust (axial) bearing, while the radial shifts are determined by the condition of axis-symmetric characteristic of the rotor model.

3. RESULTS OF NUMERICAL MODELLING

In [3] the calculations of non-stationary temperature fields, and the respective thermal deformations and HMH stresses have been carried out for four working points during the cold start-up of the turbine, presented by points 1, 2, 3 and 4 in Figure 1. The power and the flow rates of the main steam and of automatic extraction in cited points are:

Working point	Power	Flow rate of main steam	Flow rate of automatic extraction
1	5 MW	40 t/h	20 t/h
2	10 MW	110 t/h	80 t/h
3	25 MW	200 t/h	140 t/h
4	35 MW	250 t/h	160 t/h

As example, the calculation results have been presented for point 1.
Figure 2 presents the temperature field [3]. The highest temperatures are on the entrance in to the turbine at controlling stage with two rows of moving blades (the Curtis turbine) and further to the 5th turbine stage and place of automatic extraction. Increased temperatures appear at the place of the front labyrinth gland which prevents steam losses from turbine in the environment.
The HMH stresses, Figure 3, which are established in the rotor due to temperature distribution and rotation of the rotor, have maximum value in the transition region from the disc body on the shaft (forged mono-block rotor) and on the surface of the central control bore [3].
On the surface of the central control bore these stresses are the highest, and this is known from the exploitation of this turbine, where cracks in the material surface are identified. The same have been successfully repaired during the overhaul.

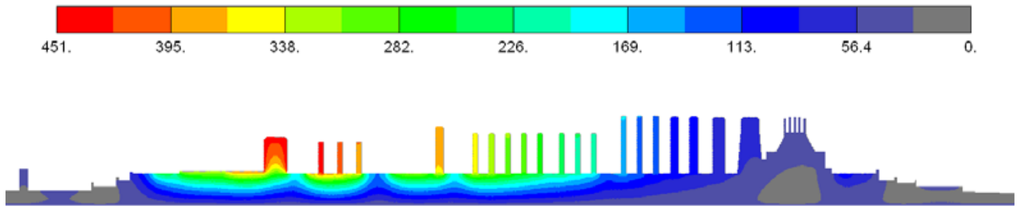

Figure 2. Temperature field (^0C) in the rotor during cold start-up [3]

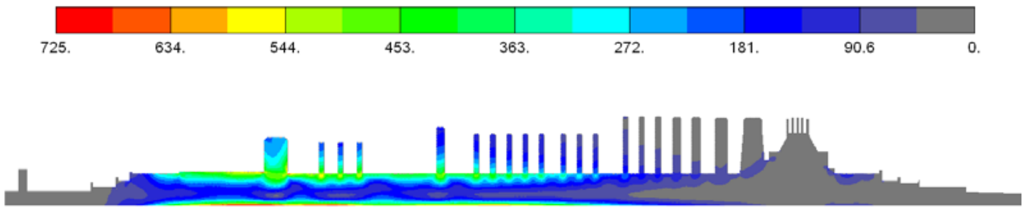

Figure 3. HMH stresses (N/mm^2) in the rotor during cold start-up [3]

4. CONCLUSION

This paper presents the results of analysis of thermal stressed state in the rotor of industrial steam turbine during transient regime - turbine cold start-up, using sophisticated user software based on the finite element method. The results of analysis of thermal stressed state of the rotor during this transient regime show that there are places with increased stress values: controlling stage with two rows of moving blades (the Curtis turbine), high pressure part of the rotor - to the 5th turbine stage, place of automatic extraction, place of the front labyrinth and the central control bore in the level of controlling stage. However, on the basis of the obtained values of the numerical calculations of the thermal stressed states in the non-stationary regimes that occur in the useful life of analyzed industrial steam turbine, the existing design of the rotor can be evaluated as satisfactory. On the other hand these places with increased stresses should be subjected to more convenient methods of nondestructive control of material over the activities during the turbine overhaul.

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