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III НАУЧНАЯ КОНФЕРЕНЦИЯ

на тему

ПАРОВЫЕ ТУРБИНЫ БОЛЬШОЙ МОЩНОСТИ

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One of the Possible Alternatives in Indirect Operational Evaluation of Stresses in a Turbine Rotor

The economical starting up and flexible changes in the load of thermal power blocks, while maintaining their longs-term life expectancy unchanged, cannot be made without some information about the stress level present in the most unfavourably stressed parts of the technological equipment. The acquisition of such an information is preceded by a number of activities which are presented in this paper together with two different realized procedures and their incorporation into the electronic control system for ŠKODA turbo-sets.

Tensomarg, Tensogard

The ŠKODA Corporation has at its disposal under the trade mark TENSOMARG a device for indirect evaluation of the thermal component of turbine rotor stresses. Stress level is indicated to the operator by means of conventional recording instruments or by a special indicator.

The device combined with two so-called temperature limiting PI-controllers acting on turbine control valves makes a new unit, which is known as the TENSOGARD and is designed to automatically keep under control any attempt to exceed the allowable stress. Its introduction was inspired by an attempt to get reliable data for verification of a hypothetical rule about the long-term service life of ŠKODA turbines, as this very device can also be applied even to the manually controlled turbo-sets. Its task is to restrict the work of the operating personnel whenever the working conditions are changed to a mere pre-selection of the required opening of the turbine, the change itself being executed by the TENSOGARD in an optimum way with respect to the properties of the turbine metal and to the needs of the electric power network.

Fig. 1 presents the two devices and the way they are incorporated into the TELCON control system, whose concept is based upon the conventionally adopted hydraulic control system. The parameters of the system are, however, adjusted more precisely with the help of a superimposed electronic PI-controller of the generated power with an input signal of

* ŠKODA, National Corporation, Turbine Works, Plzeň.

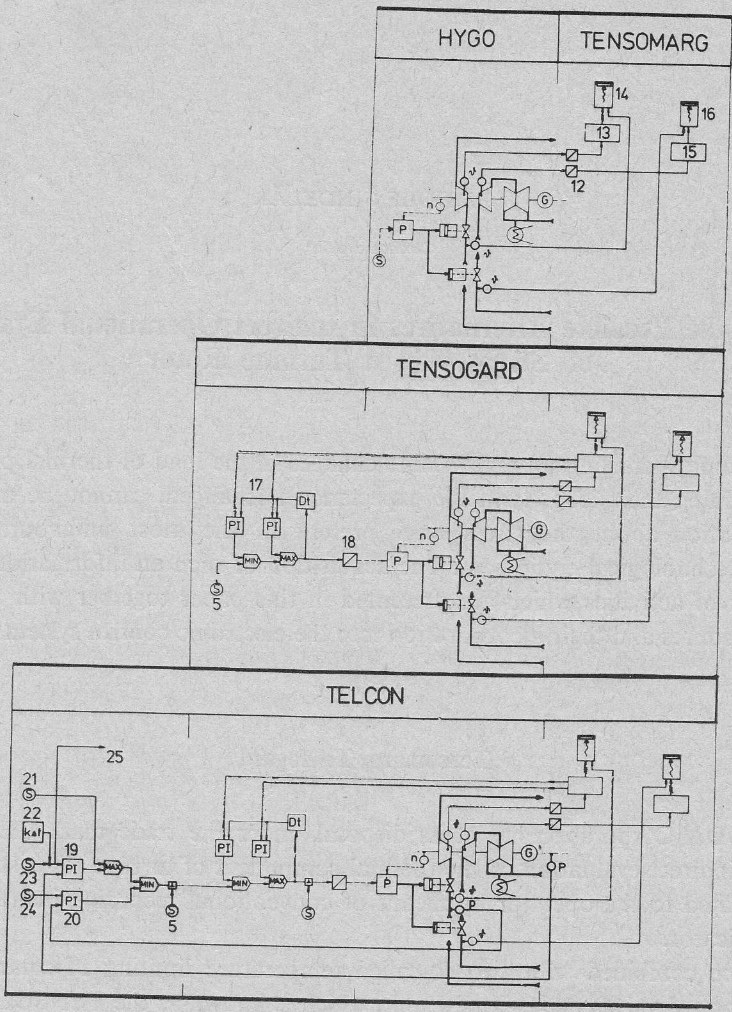

Fig. 1. Unit-assembly arrangement of TELCON (Turbine ELectronic CONTROL)

frequency deviation measured with a sensitivity of ± 5 mHz. The connection between the electronic and hydraulic parts is provided by an electromechanical converter equipped with a hydraulic output amplifier.

In a similar way the TENSOGARD is incorporated into the CONTURS, which as an electro-hydraulic system does not require the conventionally hydraulic governor to be used [1].

Critical point in a turbine

It was revealed by an analysis of stresses acting upon the individual turbine parts in starting up that it is the high-pressure rotor and its inner gland (Fig. 2) that can be regarded most frequently as the limiting parts in a turbine. In turbines with steam reheating, and

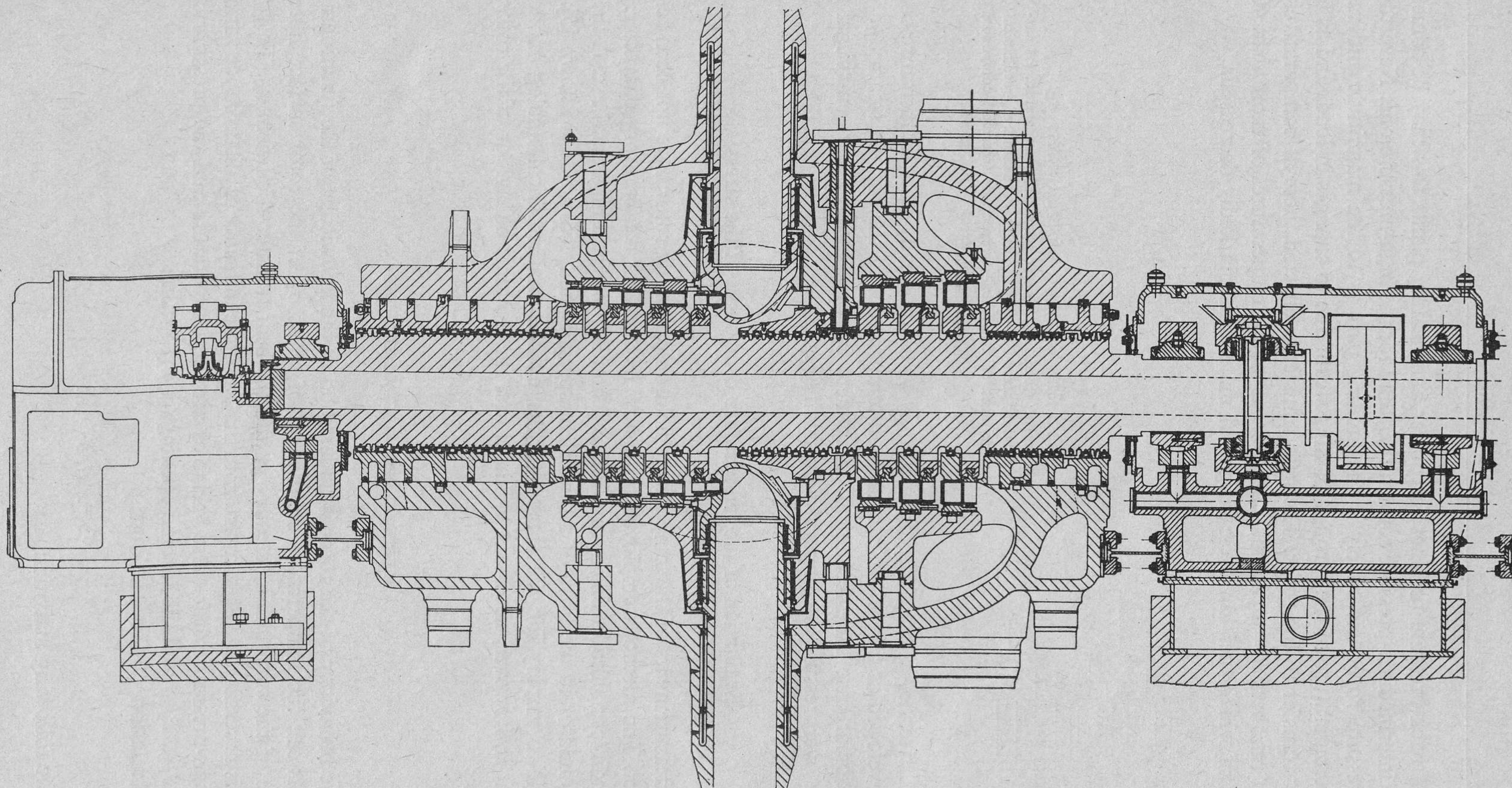

Fig. 2. Sectional view of the high-pressure section of the 500 MW turbine. The probe in the inner gland body

especially in back-pressure turbines for power and heating plants this may be sometimes true for the intermediate-pressure rotor rather than for the high-pressure one. Nevertheless, the high-pressure rotor exclusively is the limiting part as to the magnitude and rate of changes in temperature of the through passing steam due to changes in load, and as to the rate of heat transfer to the rotor. The heat transfer is effected there at comparatively the highest Biot numbers $Bi=\alpha r/\lambda$ and, moreover, at metal temperatures of more than 400°C, on account of which the low-cycle fatigue of material is markedly decreased.

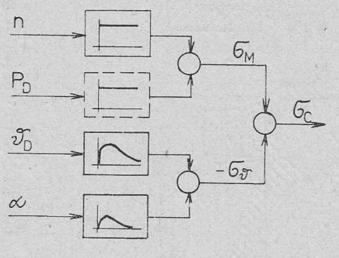

Fig. 3a. Variation of total stress σ_C: n – rotor speed, P_D, ϑ_D – pressure and temperature of surrounding steam, α – heat transfer coefficient, σ_M-mechanical component of stress, σ_ϑ – temperature (thermal) component of stress

The total stress σ_C is practically formed only by a non-steady-state temperature component $\sigma_{\vartheta N}$ developed as a result of a change in temperature of the passing steam ϑ_D, since in this particular case the mechanical component σ_M produced by the centrifugal force and outside overpressure is comparatively small (Fig. 3a).

Evaluation of the temperature component of the stress

Direct measurements of the temperature field on operating rotor have not been reliably mastered yet. The methods employed until now have rather a laboratory character. This is why an indirect method using approximating simulators is still in general use. A thermo-physical simulator in the form of an inserted, specially adapted probe is known [5]. It is also possible to employ for the same purpose even a mathematical analog model. In both cases it is, however, necessary to know first the dynamic behaviour of the rotor part in question.

Dynamic behaviour of the rotor

Dynamic behaviour of the rotor is understood as the knowledge of the functional relationship between the temperature component of stress σ_ϑ of the rotor surface fibre, the temperature of the surrounding steam ϑ_D, and the heat transfer coefficient α_D. To control the stress it is necessary to know, apart from that, also the influence of disturbance variables. With the high-pressure rotor these variables are the temperature and the pressure of the live steam (ϑ_{FD}, P_{FD}) and the turbine opening (p_2) (Fig. 3b).

The formulation of the relationship

$$\sigma_\vartheta = f(\vartheta_{FD}, P_{FD}, p_2)$$

represents the following activities:

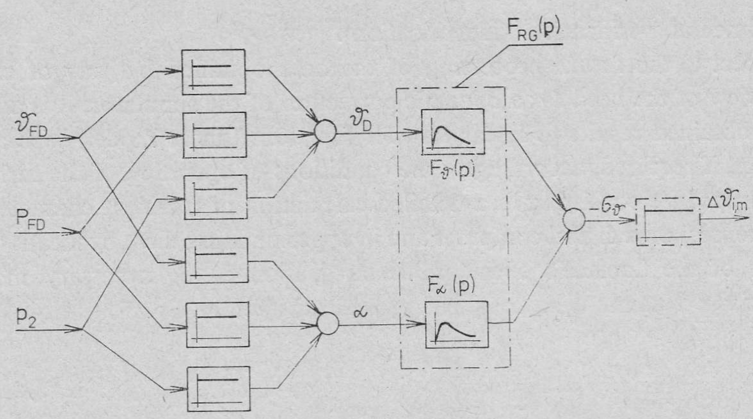

Fig. 3b. Block-type representation of variation of the temperature component of stress

– investigation of the nonstationary rotor temperature field $\vartheta(r, t)$ (as a consequence of solving the Fourier's equation),

– transformation of the temperature field into the stress field (as a consequence of solving the Euler's distortion equation),

– investigation of relationships

$$\vartheta_D = f_\vartheta(\vartheta_{FD}, P_{FD}, p_2), \quad \alpha = f_\alpha(\vartheta_{FD}, P_{FD}, p_2)$$

for the region of starting up and for load changes (a static characteristic of a turbine or taking account of thermal capacities between the turbine valves and the rotor),

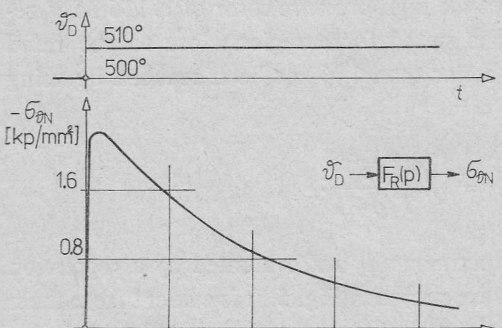

Fig. 4. Unit function response $\sigma_\vartheta/\vartheta_D$

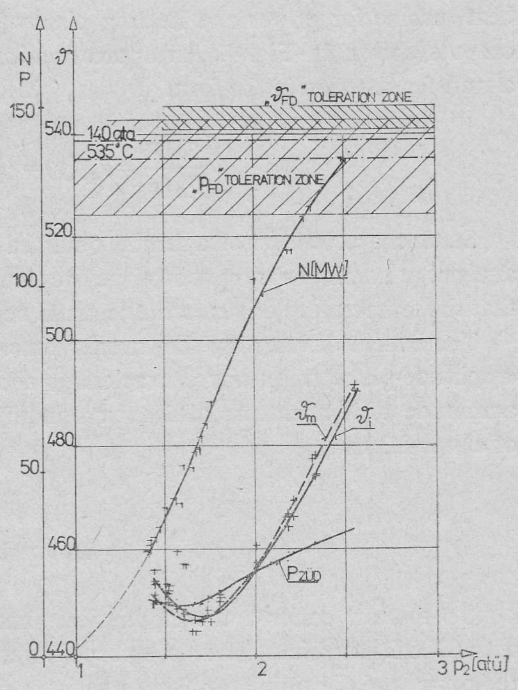

Fig. 5. Steady-state characteristic: ϑ_i – surface temperature of probe, ϑ_m – mean integral temperature of probe, p_2 – turbine opening (oil pressure), N – electrical output of turbo-set, $P_{ZÜD}$ – pressure in reheater

– experimental verification (identification).

With respect to the future processing of the data obtained by way of experimentation and in view of the need for a dynamic correction of the simulator, it is advantageous to form the obtained results to linear transfer functions (linear model) with a possibility to adapt them, if required, according to the conditions of operation of the turbine. In this respect the procedure consisting in a detailed calculation of transient characteristics (unit function responses) on a digital computer and in approximating them by means of the transfer functions of the Laplace's operator, which guarantees the necessary more general knowledge of the dynamic behaviour, has proved satisfactory.

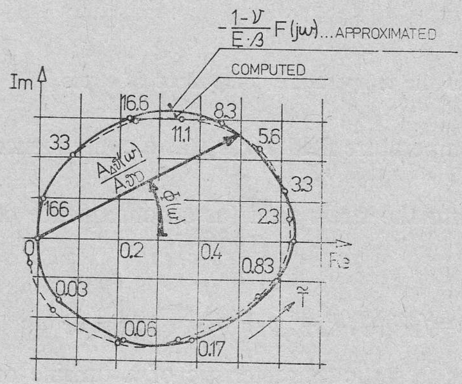

Fig. 6. Frequency characteristic of the high-pressure rotor

The so investigated unit function response of the high-pressure rotor of a 135 MW reheat turbine for power and heating plants [4] is suited for a region close to the rated outputs shown in Fig. 4. An approximation better than 1% has been reached by using a transfer function in the form of

$$\frac{\sigma_\vartheta(p)}{\vartheta_D(p)} = F_R(p) = \sum_{i=1}^{3} (-1)^i \frac{a_i}{T_i p + 1}\,.$$

Static characteristics of a turbine obtained experimentally and evaluated by a regression analysis at nominal parameters of the live steam are shown in Fig. 5. It is evident from them that the temperature of steam in the gland depends rather strongly on the turbine opening.

The benefit of knowing the transfer function $F_R(p)$ [2] is demonstrated on the derived amplitude-phase frequency characteristic shown in Fig. 6. It is clear therefrom that the amplitude of temperature of the surrounding steam ϑ_D is minimum damped at a period of about 2.3 min.

Rotor simulator

Up to now either a thermophysical simulator, or an electrical simulator were used with ŠKODA turbines. The main purpose was to reach the best possible agreement between the rotor and the simulator. That was more difficult to reach in the first case.

Thermophysical simulator

This type of a simulator, in the form of a cylindrical body (a probe) inserted radially into the space of the inner gland body (Fig. 2), was employed for a total of 6 machines. The probe itself is provided on its face with an annular labyrinth with a central steam supply and a radial outlet of steam withdrawn from the gland in the critical point of the rotor. The labyrinth serves to simulate the heat transfer coefficient for the inner gland under various working conditions. Conclusions in respect of the stresses acting upon the surface rotor fibre are drawn according to the measured difference in surface temperature ϑ_i and mean temperature of the probe ϑ_m, measured by means of fixed built-in and opposite interconnected shell-type thermo-couples.

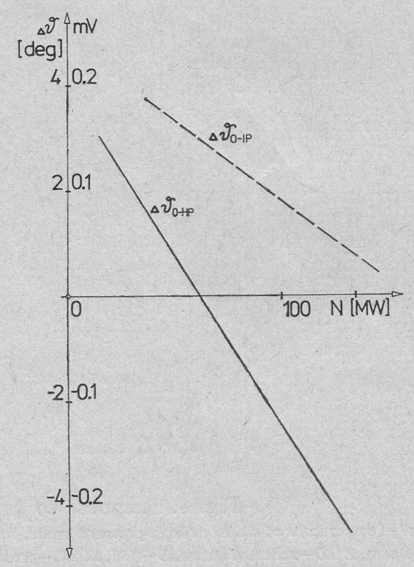

Fig. 7. Steady-state difference in temperatures on probe $\Delta\vartheta_0$:
indices: HP – high-pressure, IP – intermediate-pressure

The probe was dimensioned in accordance with results of calculation and according to the basic assumptions verified by means of an experiment. Unfortunately the first trials with the probe on a standard machine showed rather a large discrepancy in the dynamic behaviour. Neither the steady state of the probe has confirmed the assumption of a negligible radial heat flow as it should be to correspond to the rotor. The measured non-zero steady-state temperature difference was not constant at the varying output, and with the high-pressure rotor it even changed its polarity (Fig. 7). It depended on the position of the probe relative to the hottest point of the body and on the perfectness of insulation. A certain portion of the discrepancy accounted to the inaccuracy of the thermo-couple, as the manufacturer allows an error of ± 3°C at a temperature of 400°C. The probe required not only a dynamic, but also a statistical correction. Any change of the probe design was too laborious and time consuming under the given circumstances, and therefore the correction was made first to the electric signal proportional to the difference in temperatures $\Delta\vartheta = \vartheta_i - \vartheta_m$. For that purpose a reliable measurement was done with the help of a SOLARTRON data logger, with the output onto a punch tape which was then processed on a digital computer.

Responses of significant quantities to special testing signals (step turbine opening, linear steam temperature rise) were measured. They served then as a basis for computation of dynamic transfer functions and investigation of correction transfer for high- and intermediate-pressure turbine rotors relating to them. There were reviewed various shapes of correction transfers. It can be stated from experience with them that the correction placed into the difference signal results in a complex shape of the correction transfer. (In this respect it is more advantageous to measure the absolute temperatures ϑ_i and ϑ_m). Finally, a compromise was chosen in resorting to a more simple type of the correction transfer.

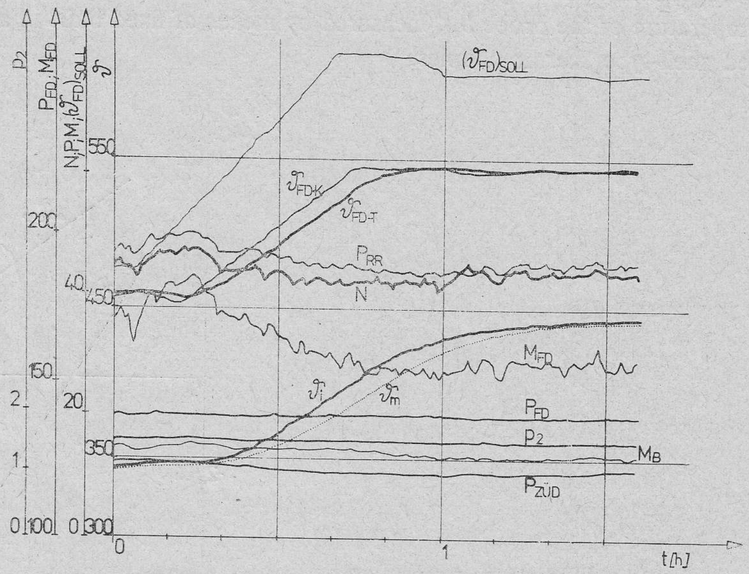

Fig. 8. Response to generated temperature rise rate of live steam
ϑ – temperature of steam or high-pressure probe, M – steam quantity, P – steam pressure, p_2 – turbine opening. Indices: FD – live steam, $ZÜD$ – reheated steam, B – fuel, $SOLL$ – preset value, RR – space of control stage, K – "behind boiler", T – "before turbine"

A difficult task is to verify the response of the simulator to a linear temperature rise of the passing steam, for reaching a quasisteady state on the turbine requires effecting a linear rise of the preset temperature of the live steam for a duration of 60 to 80 minutes. Owing to the availability of a drum boiler fired with crude oil several trials were carried out during which the linear rise of temperature of the live as well as reheated steam was generated at a fixed opening of the turbine. The variation of significant quantities in one of the trials is shown in Fig. 8. Under the given assumptions it was possible, when chosing a lower generated rate of temperature rise (below 1°C/min), to attain the quasisteady state on the turbine. Strictly spoken, to reach such a state on the probe was in principle impossible due to the non-zero steady-state difference in temperatures $\Delta\vartheta_0 = \vartheta_{i0} - \vartheta_{m0}$. Therefore, its automatic compensation was proposed and brought to effect, roughly in dependence on the electric output of the turbo-set, and finely by using a hybrid method consisting in establishing a symmetric zone ± 1.5°C wide. A comparatively slow control circuit is activated in this zone, and designed so as to equalize the remaining steady-state difference to zero. Outside of the zone the work of the circuit is interlocked.

In this way the adjustment of the probe, without its dismantling, to a state suited for operation was finally accomplished. If no regard is taken of the number of amplifiers needed for realization of the correction, the dynamic correction of the probe can be carried out with a great accuracy. What, however, still remains to be settled is to get true data about the behaviour of the probe in operation. It also turned out that transferring of the results obtained for one turbo-set into a similar one is questionable. By way of measurements it was even revealed that the behaviour of congruent probes is qualitatively different with the same types of turbines. The causes thereof have not been explained in any satisfactory manner as yet.

Besides, in consequence of the constant position of the thermocouple measuring temperature ϑ_m an error in the stress value of as much as 30% occurs at quick changes. In the author's opinion, backed by the experience, application of the probe is not a good solution.

Electric simulator

The experience gained with the preceding type of simulator made it possible to formulate the requirements for a new method of indirect evaluation of rotor stresses in the future:

- simplicity and satisfactory accuracy under both slow and quick changes in thermal conditions,
- possibility to verify the work without any expensive tests,
- easy tuning,
- accuracy of results, reproducibility,
- reliability of the equipment,
- easiness of location and failure removal.

In principle, the above requirements are best met by a mathematical analog model materialized with the aid of active electrical elements (Fig. 9). The measured temperature of steam in the turbine acts here as an input signal. The evaluating part is formed by two parallel-connected single-capacity members, their output signals forming, after subtraction, a signal proportional to the temperature component of stress $\sigma_{\vartheta N}$ of the rotor surface

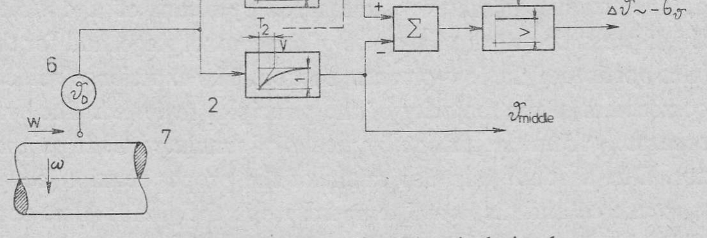

Fig. 9. Block-diagram of an electrical simulator

fibre. The output signal of the member with a prolonged time constant stands for the mean metal temperature.

The dynamic behaviour of the simulator is determined by two time constants T_1, T_2, and the amplification V. It should conform with the following conditions for consonance of the rotor and simulator:

a) coordinate of the extreme (peak) of the unit function response $\sigma_\vartheta/\vartheta_D$,

b) response to the linear steam temperature rise: $\dot{\vartheta}_D = d\vartheta_D/dt$ ($\dot{\vartheta}_D \sim -\sigma_{\vartheta q} \sim \Delta\vartheta_q$).

The same position of the peaks gives evidence of a good agreement under quick changes in temperature of the passing steam (sudden changes in load) and the same quasisteady state indicates a harmony under slow changes (in starting up). The following procedure proved satisfactory: from the unit function response $\Delta\vartheta/\vartheta_D$ computed, investigated and verified by an experiment the coordinates of extreme $\Delta\vartheta_E$, t_E are to be determined. In the same way the quasisteady state is obtained, making it possible to put stress $\sigma_{\vartheta Nq}$ (in terms of temperature $\Delta\vartheta_0$) to the temperature rise rate $\dot{\vartheta}_D$. The above described three parameters are then determined from the following equations:

$$t_E = \frac{T_2 T_1}{T_2 - T_1} \lg \frac{T_2}{T_1},$$

$$\Delta\vartheta_E = V\left[\exp\left(-\frac{t_E}{T_2}\right) - \exp\left(-\frac{t_E}{T_1}\right)\right],$$

$$\frac{\Delta\vartheta_q}{\dot{\vartheta}_D} = (T_2 - T_1)\,V.$$

The relationships stem from an analytical expression of the transient characteristic and the quasisteady state of the evaluating circuit of the simulator as shown in Fig. 9 (concerned is a derivative transfer function with two delayed-action capacities).

The comparison between the required unit function response and that materialized by the simulator is shown for a high-pressure rotor in Fig. 10.

The dependence on the change in the heat transfer coefficient, if necessary at all to be taken into account, is included by adaptation of all the three parameters according to some of the characteristic independent quantities. In such a case the simulator is of hybrid construction.

An electrical simulator of two-channel construction was installed this year, without any adaptation of its parameters, in a 112 MW turbo-set for Hanasaari Power and Heating Plant, Finland. The steam temperature is measured by means of a double thermocouple placed for the high-pressure rotor in the space of the control stage, and for the low-pressure rotor in the inlet portion of the low-pressure section. The experience gained so far with it confirms the correctness of the majority of the principles reviewed. The evaluating circuit was adjusted separately from the turbine by means of simulated testing signals at the input of the simulator. Due account was also taken of the proper relation between the steam temperature as measured and the actual temperature in the gland. After connecting to the turbine the verification was restricted merely to covering the step change in opening of the

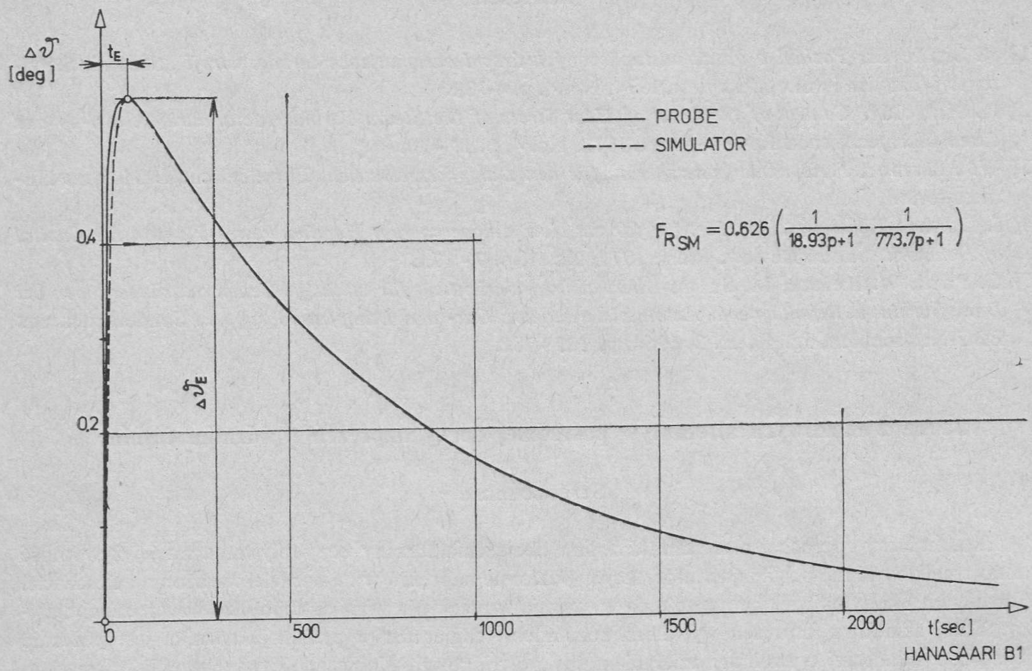

Fig. 10. Degree of approximation of the unit function response of a high-pressure rotor and a simulator

control valves. The assumption of elimination of great errors has been confirmed. Even additional adjustment was unnecessary.

Up to now two ways of getting an input signal have been investigated. The first way is using a probe, but adopted exclusively as a means for inserting a thermocouple into the steam withdrawn from the inner gland, and the second way is measurement of the steam temperature on the gland periphery by means of interconnected thermocouples. The aim of averaging the data obtained is to keep down the peripheral asymmetry due to the nozzle control. All the temperatures are measured by means of double shell-type thermocouples. Any damage done to some of the thermocouples is signalled.

Conclusion

A change in favour of an electrical simulator seems to be justified. The experience gained so far with it is promising for the future. The device is simple and sufficiently accurate at the same time for both slow and quick changes. The setting is convenient and is done independently of the turbine. Problems relating to the compensation of the non-zero heat flow in the steady-state do not exist. Abandoned also can be the complicated check tests. The results are transferable to a congruent type of turbine without running any risk of great errors. The electrical simulator can also be installed as an additional device. It does not require making any great adaptations to the turbine. It can be conveniently incorporated into the digital computer without any additional effort of the project designer.

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Jedna z możliwych alternatyw pośredniej oceny naprężeń w wirniku turbiny

Streszczenie

Ekonomiczny rozruch i elastyczność zmian obciążenia turbiny bez wpływania na jej żywotność trudno zrealizować nie posiadając informacji o poziomie naprężeń w najbardziej narażonym elemencie. W turbinach Škody największe naprężenia występują w przekroju wirnika w pobliżu wewnętrznej dławnicy. Tutaj składowa naprężeń wywołana rozkładem temperatur w przekroju wirnika jest znacznie większa od składowej wywołanej działaniem siły odśrodkowej. Konieczność wyznaczenia wirującego pola temperatur w warunkach roboczych lub też oceny wartości termicznej składowej naprężeń zmusza projektanta do zastosowania oceny pośredniej. Opiera się ona na symulowaniu warunków cieplnych w wirniku za pomocą wprowadzonego lub dołączonego równolegle modelu fizycznego lub metody analogowej. Ta ostatnia metoda wydaje się w chwili obecnej korzystniejsza. Jedną z najbardziej godnych uwagi korzyści, jakie daje metoda analogowa, jest możliwość niezawodnego ustalania stałych lub charakterystyk pożądanych przez dostawcę turbiny przed oddaniem jej do użytku. Podstawowym warunkiem możliwości stosowania metody pośredniej jest znajomość zmiennego w czasie pola temperatur w warunkach roboczych. Najbardziej użyteczne są tu charakterystyki dynamiczne wyznaczane teoretycznie lub doświadczalnie. Wykorzystując takie dynamiczne charakterystyki przejścia można dokonać syntezy modelu oraz jego sprawdzenia i ewentualnej korekcji. Weryfikację odpowiedzi przejściowych przeprowadza się bezpośrednio na turbinie.

Niepowodzenia w stosowaniu specjalnych sond doprowadziły nas do skonstruowania prostego, ale wystarczająco użytecznego modelu analogowego pracującego z jednym tylko sygnałem wejściowym. Sygnałem tym jest zmierzona temperatura przepływającej pary wodnej; może to być również temperatura stagnacji pary lub, nawet w przypadku krańcowym, temperatura na powierzchni kierownicy w stopniu regulacyjnym.

Opisana wyżej metoda pozwala na przejście od modelu analogowego do obliczeń numerycznych za pomocą komputera bez dalszego angażowania projektanta turbiny.

Одна из возможных альтернатив посредственной оценки напряжений в турбинном роторе

Резюме

Экономический пуск и эластичность изменений нагрузки турбины, без влияния на ее живучесть, трудно осуществить не располагая информацией об уровне напряжений в наиболее нагруженном элементе. В турбинах фирмы ШКОДА самые большие напряжения имеют место в поперечном

сечении ротора вблизи внутреннего уплотнения. Здесь температурная составляющая напряжений, вызванная распределением температур в поперечном сечении ротора, гораздо больше составляющей, вызванной центробежной силой. Необходимость определения вращающегося температурного поля в эксплуатационных условиях или оценки значения термической составляющей напряжений, заставляет проектанта применять посредственную оценку. Она основывается на симулировании термических условий в роторе при помощи введенной или параллельно присоединенной физической модели, или же аналоговым методом. Этот последний метод кажется теперь более полезным. Одной из наиболее ценных польз, которые приносит аналоговый метод, является возможность точного определения постоянных или же характеристик, в которых особенно нуждается поставщик турбины перед сдачей ее в эксплуатацию. Основным условием возможности применения посредственного метода является знание переменного во времени температурного поля в эксплуатационных условиях. Наиболее пригодными являются здесь динамические характеристики, определяемые теоретически или экспериментально. Используя такие переходные динамические характеристики можно произвести синтез модели, ее проверку и возможную корреляцию. Проверка переходных ответов проводится непосредственно на турбине.

Неудачи в применении специальных зондов привели к разработке конструкции простой, но в достаточной степени полезной, аналоговой модели, которая, в противоположность общепринятой практике, работает только с одним входным сигналом. Этим сигналом является замеряемая температура протекающего водяного пара, но может ним быть также температура заторможенного пара или же, в крайнем случае, температура на поверхности направляющего аппарата регулируемой ступени.

Вышеизложенный метод позволяет переходить от аналоговой модели до цифровых расчетов на ЭВМ без дальнейшего ангажирования проектанта турбины.