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Steam turbines start-ups

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Abstract

The article presents a procedure and the chosen results of numerical computations of liquid flow parameters, which use the data from their measurements realized on the test stand, for the model of formed suction intake with a rib, supplied by the screened open wet well. Comparative analysis of numerical computations and these determined by measurements, did concern the standard flow acceptance criteria.

1 Principles of steam turbines start-ups

Steam turbines are being started based on operating instructions developed by turbine manufacturers and constituting an integral part of start-up instructions of a steam unit. The operating instructions describe the order of executing start-up actions and provide the values of operating parameters that should be maintained during start-ups. Rigorous following the order of individual start-up phases and maintaining the limit values of operating parameters is crucial from the point of view of safety and reliability of turbine operation. The start-up phase is a particularly dangerous and complicated phase of steam turbine operation as it consists of starting numerous equipment and auxiliary systems, and mechanical and thermal processes taking place have nonstationary nature (i.e., transient heating, varying steam flows, acceleration of rotors, vibrations, etc.). The principles of correct conducting start-ups have been developed over the long-term operation of steam turbines and contain experience of turbine designers, constructors and operators.

Three basic phases can be distinguished in the process of steam turbine unit start:

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- turbine preparation to start-up,
- running-up to idle run and synchronization,
- loading.

Turbine preparation to start-up begins with barring the rotor by the turning gear. The turning gear operates as teeth gear, driven by an electrical (electrical turning gear, see Fig. 1) or hydraulic (hydraulic turning gear) motor. Prior to start-ups turbine rotors must be thermally stabilised, and their deflection line must coincide with the gravity deflection line. Rotor cooling down during standstill would be leading to its thermal bent due to faster cooling from the bottom, which would ultimately lead to rubbing and damage of labyrinth seals.

Prior to cold start, the turbo-set rotating system should be rotated using turning gear as long as possible. Particular care should be paid to the start-up with barring time shorter than 72 h. In such a case the turbo-set will need prolonged heating-up at intermediate rotational speed. The heating time is limited by the achievement of proper dynamic state during continued run-up.

The turning gear can be switched off only when the highest temperature of an high-pressure (HP) and intermediate pressure (IP) inner casing is lower than 100 °C. If needed, the turning gear can be switched off for 15 min after a day of barring, and after 2 days of barring – for about 30 min on the condition that vacuum is broken and gland steam supply is switched off. The maximum time of turning gear standstill depends of the type of turbine.

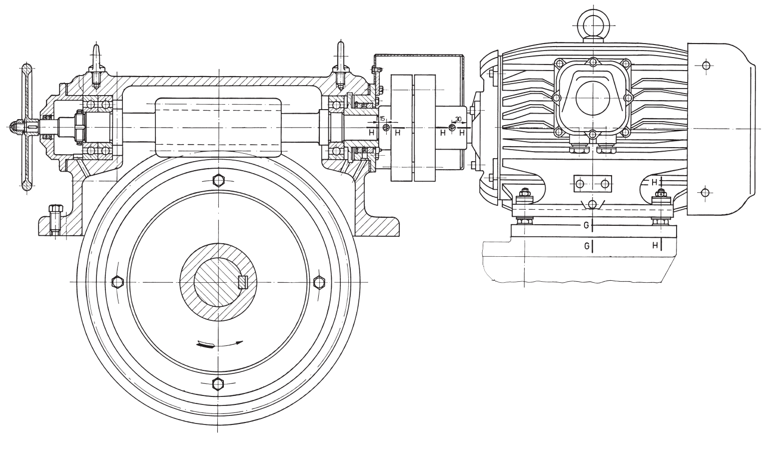

Figure 1. Electrical turning gear.

Prior to repeated switch on of the turning gear, turbine rotor must be manually turned by one revolution in order to check if rubbing has not taken place. If rubbing is found it is necessary to wait until the rotor rotates freely and only then the turning gear can be repeatedly switched on.

Prior to switching on the turning gear, oil system is started up. It is always necessary to check whether lube oil pumps has started and whether all bearings are supplied with oil. Control of lube oil pressure and temperature, as well as rotor axial displacement and eccentricity is necessary. The turning gear can be switched on only when all jacking oil pumps are in operation.

After starting the oil system and turning gear, the condensation system can be started, that is cooling water, air ejection and condensate pumping systems. Simultaneously, gland steam system is started to prevent air in-leakage. Gland steam must have appropriate temperature and pressure which must be controlled during start-up. Prior to supplying steam to the turbine, the steam supply pipelines have to be drained and heated up.

In case of turbines with cooling water preheating and low-pressure (LP) by-pass, prior to supplying steam the regeneration and dumping systems have to be started.

1.1 Run-up to idle run and synchronisation

One can distinguish three major types of start-ups:

- from a cold state – when HP and IP inner casing temperature is lower or equal 170 °C,
- from a warm start – when HP and IP inner casing temperature is lower or equal 430 °C,
- from a hot state – when HP and IP inner casing temperature is greater than 430 °C.

Cold state of a turbo-set usually occurs after a 6-day standstill. The difference between the above types of start-up consists in the fact that in the former case steam supplied to the turbine has greater than casing metal temperature, while in the latter two cases, depending on the boiler characteristics, steam temperature can be lower or higher than the casing temperature.

Since cold starts usually take place after long standstills, hence in such cases inspection of turbine and equipment according to their instructions is necessary.

1.1.1 Steam parameters required for cold start

Live and reheat steam temperature must be at least 50 °C higher than the saturation temperature and for instance for steam pressure 5 MPa live steam temperature should be min. 315 °C.

Turbine start-up for a cold state should be conducted with stabilised live steam parameters which for a high power output unit with steam parameters 535 °C/18 MPa typically assume:

- live steam: $p = 5$ MPa, $T = 350$ °C÷425 °C,
- reheat steam: $p = 0.8$ MPa, $T = 350$ °C÷400 °C.

To open HP and IP stop valves, condenser vacuum should be at least 0.03 MPa abs. Typical recommended condenser vacuum is on the level of 0.01 MPa abs, however it strongly depends on the cooling conditions. It is necessary to check if differential expansion and shaft eccentricity are within the allowable limits. When eccentricity exceeds the allowable value, it is necessary to wait turning the shaft on the turning gear and with open by-pass until shaft eccentricity is lower than the limit value.

Next HP and IP stop valves can be opened, with control valves closed. In the turbine governor the target rotational speed 3000 rpm is set as well as run-up rate (for instance 75 rev/min^2). In case the target speed is set on the value between the turning gear speed and 3000 rpm, one should avoid settings in the range of critical speeds. Prior to automatic run-up it is always necessary to check on low speed the rotor dynamic behaviour, temperature distribution in oil system and bearings, as well as auxiliary systems operation. When the proper condition of the turbine is confirmed, the automatic run-up can be started which in turn can be terminated any time. In case an attempt is made to stop run-up in the range of critical speeds, turbine run-up will be stopped only when the critical range has been passed, as it is forbidden to stop run-up in the range of critical speeds of turbine and generator rotors.

Turbine start-up can be done through all its cylinders simultaneously (i.e., HP, IP and LP) or with by-passing some of them (i.e, HP) in order to ensure better start-up conditions. For instance, for 360 MW turbines, depending on the HP inner casing temperature, one can distinguish two modes HP and IP valves control:

- metal temperature < 200 °C – idle run load is mainly covered by HP and LP turbines,
- metal temperature > 200 °C – idle run load is mainly covered by HP turbine.

During run-up it is necessary to monitor vibrations and indications of HP and IP temperature probes. When excessive vibrations occur, the turbine speed should be lowered and kept constant until vibrations are within the permissible limits and continue with running-up. To reduce excessive vibrations the turbine speed must be kept constant for a longer period of time. Typically, such a stop is done at 1000 rpm. If during run-up no excessive vibrations occur it is not necessary to heat-up the machine on intermediate speed. During running-up it is important to pass through the critical speeds with high rate (600–800 rev/min^2). The permissible value of rotor eccentricity below 600 rpm is typically equal to

200 $\mu\mathrm{m}_{p-p}$. One the threshold is exceeded, automatic trip of turbine takes place. Above 600 rpm rotor relative vibrations should be monitored. The real values of critical speeds should be measured at first start-up, included in turbine operating instructions and programmed in governor.

Within the range of rotational speed 0–3000 rpm high power output turbo-sets have a couple of critical speeds. Theoretical values of critical speeds can be as follows:

- I frequency:
  - generator rotor 940 rpm,
  - LP rotor 1490 rpm,
  - HP rotor 1900 rpm,
  - IP rotor 2230 rpm,
- II frequency:
  - generator rotor 2580 rpm.

Turbine running-up with maximum rate can be done when temperature probes signals are $> -40\%$ or $< +80\%$. Once these threshold values are exceeded, the running-up rate should be lowered so as to at values -50% and $+100\%$ it is equal 0 (turbine run-up terminated).

At 3000 rpm auxiliary oil pump can be switched-off. After the pump has been stopped, the jacking oil pumps are switched-off automatically. During start-up the operation of LP turbine should be supervised with special emphasis on:

- steam temperature downstream the penultimate stage,
- partial vacuum break at turbine shutdown,
- protection against condenser pressure raise.

Steam parameters before turbine depend on its thermal state which is determined by standstill duration. In turbines with live steam parameters 535 °C/18 MPa and reheat steam parameters 535 °C/4 MPa steam should have the following parameters after:

a) 2-hour standstill

live steam	$p = 12$ MPa	$T = 480$–500 °C
reheat steam	$p = 0.8$ MPa	$T = 480$–500 °C

b) 8-hour standstill

live steam	$p = 6$ MPa	$T = 450$–480 °C
reheat steam	$p = 0.8$ MPa	$T = 430$–480 °C

c) 36-hour standstill

reheat steam	$p = 5$ MPa	$T = 400$–430 °C
reheat steam	$p = 0.8$ MPa	$T = 390$–420 °C

Additionally, the following conditions should be fulfilled prior to supplying steam to the turbine:

- live steam temperature can be maximum 50 °C lower than instant temperature of HP inner casing,
- reheat steam temperature can be maximum 50 °C lower than instant temperature of IP inner casing.

1.1.2 Start-up from a hot state

Turbine running-up and loading must take place faster than during a cold start so as to avoid inadmissible cooling down of the turbine. The rate of running-up should be on the level 250–300 rev/min^2. During run-up the indications of temperature probes should be monitored as well as differential expansion, and permissible values should not be exceeded. During running-up fast (600–800 rev/min^2) transition through the critical speeds should take place. Typically, time required to run-up the rotor to 3000 rpm is approx. 10 min.

Once the sub-synchronous speed approx. 3000 rpm has been achieved, the turbine governor is ready to synchronise the turbo-set with power grid. When the generator has been synchronised with grid, the governor automatically loads the turbo-set with initial load equal to some percents of nominal load. After synchronizing the turbine its drains can be closed and auxiliary systems should be monitored.

Independently from the type of start-up bearing temperatures must be monitored. The idle run duration with nominal live steam parameters should not usually exceed 30 min, and the HP exhaust temperature should not be higher than 430–450 °C.

Turbine loading is conducted according to the operating instructions of turbine governor. Load increase is usually realised with increasing live steam pressure and gradual opening of turbine control valves. Nominal load is achieved with the last valve partly open which enables power output control below and above the nominal value. Nominal load is typically achieved by opening fourth HP control valve. With regeneration system disconnected, load achieved with nominal steam parameters is some percent higher than nominal load, however its generation take place with decreased thermodynamic efficiency of thermal cycle.

Also during load control, the start-up probes records should be observed and in case the permissible limits are exceeded, the loading rate should be decreased, and if necessary, further control can be done manually. Maximum loading rate is achieved when relative thermal stresses in HP and IP turbine are equal +80%. After this value has been exceeded, the loading rate must be decreased so as at 100% thermal stress the loading is completely stopped. In case of temperature probe failure, the gradient of inner casing metal temperature is to be used for start-up control and the value of 2–3 °C/min should not be exceeded. If the temperature gradient indicates that the limit value can be exceeded, the loading rate should be decreased.

During loading also differential expansion of rotors should be monitored. It should kept within the limits defined by signalisation being set individually for each turbine on the basis of axial clearances measurement. After every overhaul or any modifications to the steam path, the differential expansion sensors must be set again. A consequence of excessive thermal expansion is rotor rubbing. Turbine loading leads to positive expansion, while unloading results in negative differential expansion.

During whole loading phase the indications of special measuring apparatuses must be checked. If any indication reaches its limit value, loading should be interrupted, and if necessary turbine load can be decreased.

Turbine shutdown takes place after steady-state operation period with stable parameters of steam. Shutdown consists in decreasing the turbine power output to minimum load by closing the control valves with possible simultaneous decreasing the inlet steam parameters. Usually, load decrease starts with constant pressure of the live steam and only when a certain level of part load is reached, continued load decrease takes place with sliding pressure of the live steam. When the minimum load is reached, the valves are closed completely which cuts off the steam flow to the turbine. During de-loading forced cooling of turbine components takes place caused by decreasing the live and reheat steam temperature and additionally by throttling in the control valves. Once the valves have been completely closed, natural cooling starts and this process runs significantly slower with typical cooling rates on the level of 0.1 °C/min. Cooling time depends on the mass of HP and IP rotors and casings and turbine thermal insulation. An example of cooling curves of HP and IP cylinders of a condensing turbine of high power output is shown in Fig. 2.

Once the steam supply to the turbine has been cut off, its rotors does not stop immediately but their rotational speed is decreasing slowly according to the so-called run down curve from nominal speed 3000 rpm to turning gear speed (for example 40 rpm). An example of cooling curve is shown in Fig. 3. When the rotor is operating on turning gear, the process of cooling down can be significantly accelerated by the use of accelerated cooling system. In such systems for forced cooling of the turbine compressed air of ambient temperature is used, which is supplied to the turbine through drain or extraction pipelines. The flow of cooling

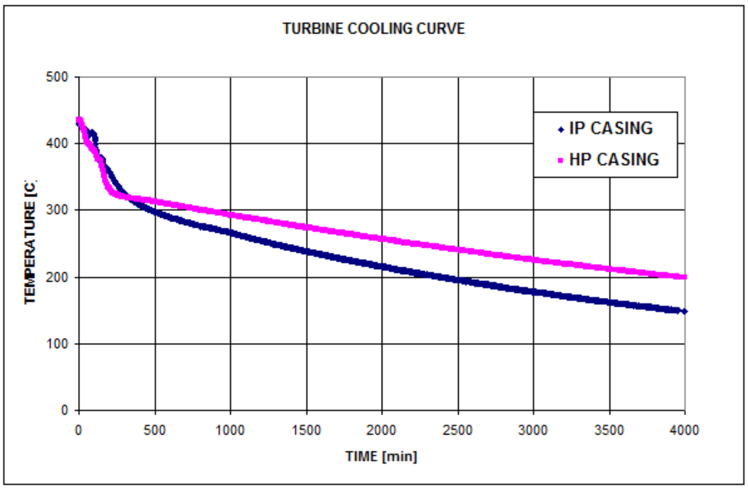

Figure 2. Turbine cooling curve.

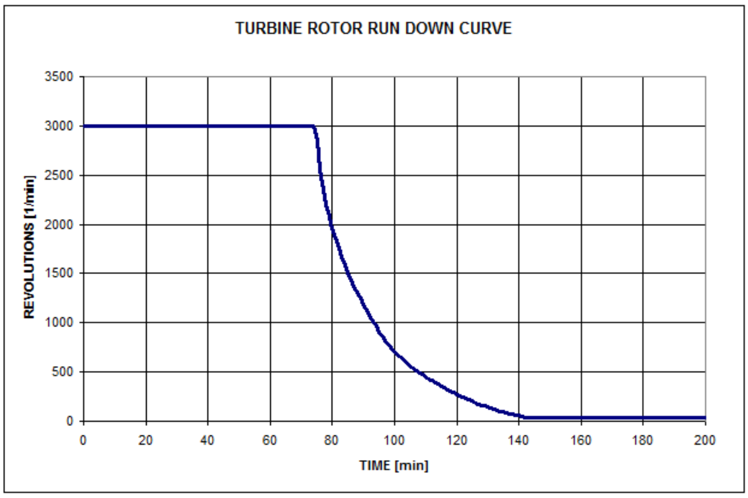

Figure 3. Turbine rotor run down curve.

air must be properly organized (Fig. 4) and controlled, so as not to damage the turbine. Depending on local conditions, the natural cooling time typically in the range 80–170 hours, can be reduced by forced cooling by some 40% (Fig. 5).

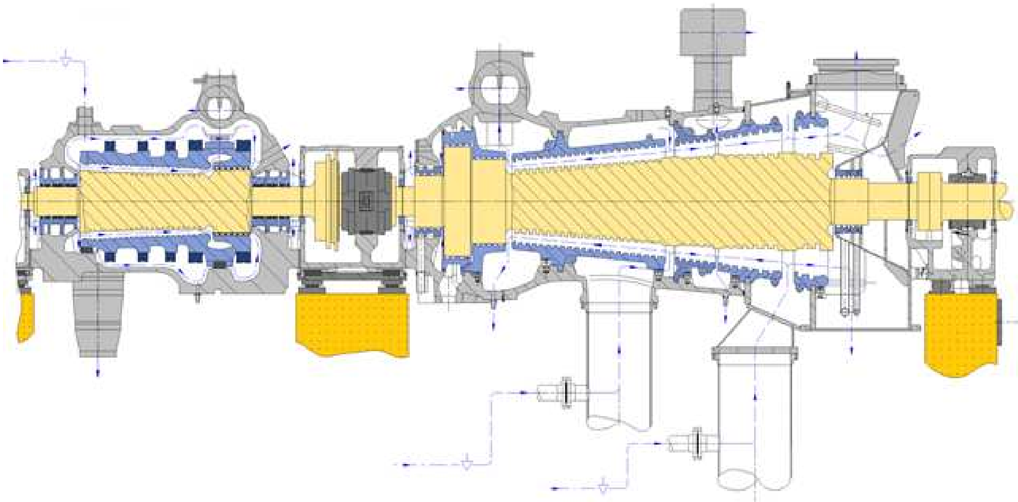

Figure 4. Turbine accelerated cooling system.

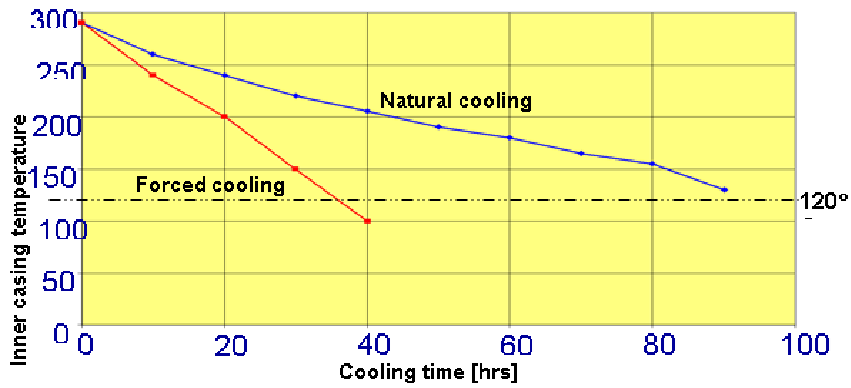

Figure 5. Effect of application of turbine accelerated cooling system.

2 Phenomena occurring during start-ups

Turbine start-up consists in running up the rotor to nominal speed, typically 3000 rpm, and loading the turbine to part or full load. The increase of rotational speed or load is accompanied by the increase of live and reheat steam parameters (temperature, pressure) and the increase of steam mass flow through the turbine. A consequence of these changes is heating-up of turbine components, mainly in high and intermediate pressure sections. Temperature variations are largest in the inlet sections of HP and IP turbine, as temperature changes from an initial value, which for a cold start is the same for whole turbine, to a value in steady-state operation, which is highest in the inlet sections of the turbine. For example, during a cold start of a 360 MW machine, the inlet steam temperature at the beginning of start-up is 350–400 °C and increases to 535 °C in steady-state. The steady-state temperature in the control state reaches 510 °C, while at the HP exhaust is only 340 °C. For this reason, the inlet areas of rotors and casings are heating-up at a fastest rate, and the accompanying stresses and deformations are highest.

Thermal stresses are generated due to nonuniform heating of component sections (for example casings wall or rotor shaft) and restrained thermal expansion. Heat is transferred to the component and is propagated within it with finite speed, and due to this the surface being in contact with hot steam has higher temperature than regions located deeper which remain cool. Material layers of higher temperature tend to expand due to thermal expansion but this expansion is restrained by the neighboring layers of lower temperature, which is the reason of generation of internal thermal stresses.

Heating up of a turbine rotor proceeds in such a way that its external surface if intensively heated by the steam flowing through interstage and end glands, and heat flows from the rotor surfaces towards its centerline. During transients, the rotor surface has significantly higher temperature than its axis, and surface is in compression, while the rotor center is in tension. There is such a point in a rotor section where thermal stresses disappear and it is located closer to the rotor surface (heated region). Temperature distribution in a rotor section is shown schematically in Fig. 6. Stress attained maximum values at the heated surface and are proportional to the difference between the surface temperature and rotor mean temperature (temperature at the point where thermal stresses disappear). This relationship can be written in the form:

$$\sigma^T = -\frac{E\beta}{1-\nu}(T_p - T_{sr}) \;, \tag{1}$$

where: E – Young modulus, β – thermal expansion coefficient, ν – Poisson number, T_p – surface temperature, T_{sr} – mean temperature.

For simulating the characteristic temperature difference of rotor a temperature probe has been used, whose surface temperature is equal to the rotor surface temperature, while its average temperature is equal to the rotor average temperature. Such a philosophy of temperature measurement and rotor stress control was introduced by BBC company in sixties. Newer designs of temperature probe employ only one metal temperature measurement (rotor surface) or steam temperature in critical region. Determination of mean temperature and thermal stress takes place using a mathematical model of rotor heating.

Exemplary temperature field in a reaction turbine rotor at the late stage of

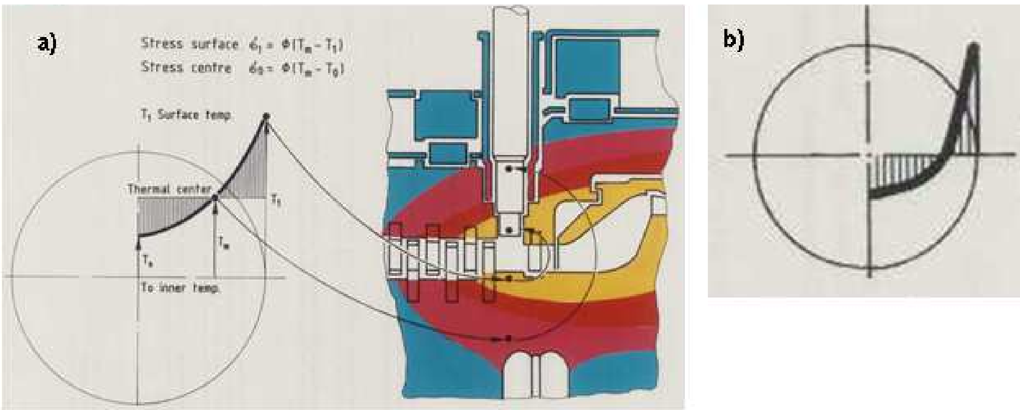

Figure 6. Distribution of temperature (a) and stress (b) within rotor during heating-up.

start-up is presented in Fig. 7. The highest absolute values of temperature and its radial gradients occur in the inlet sections of the rotor – in the control wheel and balance piston. Temperature fields become more mild when moving towards the exhaust and end glands. Stress field corresponding to such a temperature field is shown in Fig. 8. As it is seen from the diagram, the highest stress is generated on the shaft surface (balance piston, blade grooves) in the regions of big temperature gradients, and due to the above these regions of the rotor are particularly endangered by thermal fatigue and cracking.

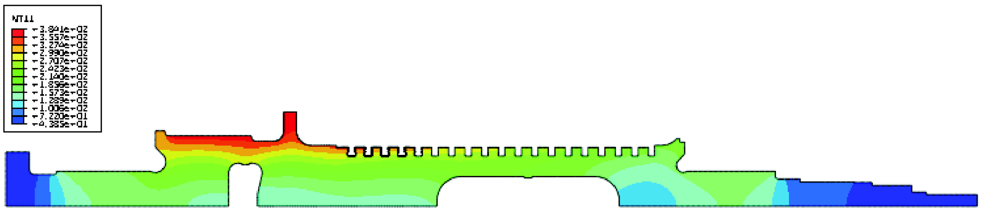

Figure 7. Transient temperature field within rotor during heating-up.

Temperature differences, and thus thermal stresses, are the bigger the thicker component sections are (casing wall thickness, rotor shaft diameter) and the faster temperature changes are. Consequently, the highest casing or rotor stresses occur in the inlet sections of these components where heating rate is highest.

An additional factor influencing the magnitude of thermal stresses is steam-metal temperature differential occurring at the beginning of start-ups. The higher the temperature differential, the higher the stress produced at a later phase of start-up.

For the above mentioned reasons, the operating instructions impose limita-

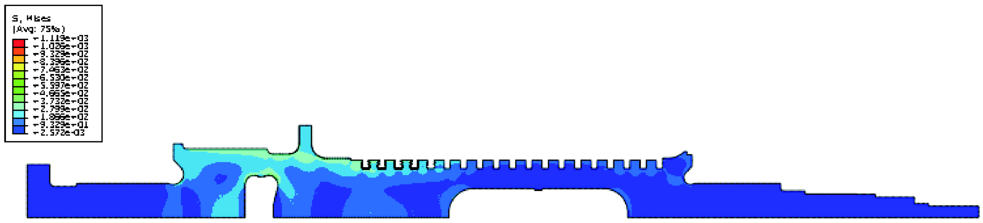

Figure 8. Transient stress field within rotor during heating-up.

tions on the initial temperature differential and steam temperature growth rate. These are so-called temperature criteria developed in order to control thermal stresses indirectly, and thus to ensure safe operation and turbine design life.

Rotor temperature field is axisymmetric due to rotor axisymmetric geometry and its rotation around own axis, which ensures symmetric heating-up. A completely different situation is in casings, which are stationary components of complicated and irregular geometry. In the inlet section of high-pressure casings the nozzle boxes are located, through which steam is supplied. During run-up or loading only a part of circumference is supplied with inlet steam, and consequently only a part of casing is heated-up which results in nonuniform temperature field within casing. This leads to generation of thermal stresses and deformations, which at big number of start-ups can be a reason of crack initiation and casing deformations. Thermal stresses are concentrated in the areas of geometrical notches, which in casings are wall thickness changes and transition radii.

Casing heating from inside by steam brings about deformations in several planes. Internal surfaces, which air heated-up faster, expand and cause casing flanges detachment towards casing outside and the resulting compression of internal edges of flange parting planes connected with bolts. Once the casing temperature field has become more uniform in steady-state, the edges detach and cause leakages (Fig. 9).

Thick flanges are also a reason of casing deformation in the parting plane. During start-up the casing shell of significantly thinner wall than flange, is heated-up faster and brings about bending of the parting plane edges towards axis in the central sections, and towards exterior at the casing ends. This phenomenon is more clearly seen the bigger the difference between casing shell and flange thickness is (Fig. 10).

An unfavourable phenomenon of thermal nature, the so-called cat's back is caused by different temperatures of casing top and bottom half. Casing top has higher temperature than its bottom, and due to this casing tends to bow towards top (Fig. 11). This phenomenon is caused by lower temperatures of condensation equipment halls and heat transfer through drain and extraction pipelines, which are situated in the bottom half of casing. Through the pipelines casing can be flood by water during a start-up and suffer from complete seizure. At lower top-bottom temperature differentials, the labyrinth clearances can be closed and rubs can occur locally.

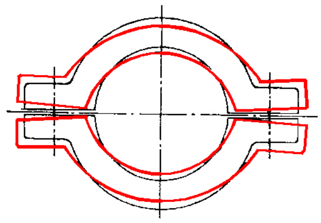

Figure 9. Transverse deformation of casing.

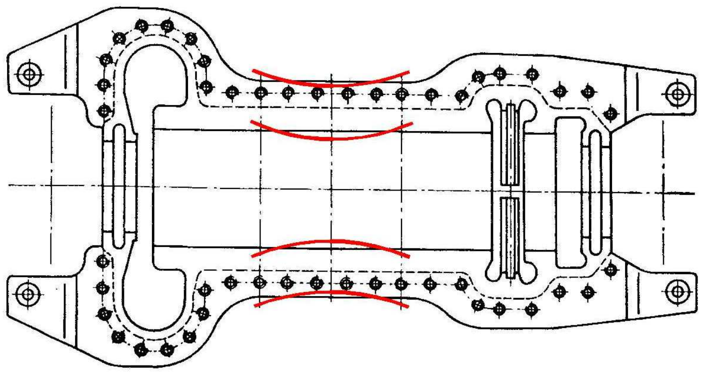

Figure 10. Transverse deformation of casing.

Both during start-up and steady-state operation, temperature difference between rotor and casing is observed. It results from differences in masses of both components – casing is heavier than rotor, and different heat transfer conditions – heat transfer coefficients for casings are lower than those for rotors. Temperature differences are largest during start-ups and it is always the case that rotor has higher temperature casing. Due to this rotor expands radially more than casing and clearances between rotor and stator blades also change. Relative thermal displacement of rotors is particularly visible in impulse turbines, where rotor is lighter than casing. Reaction turbines, although longer, have big but more equal masses of casings and drum rotors. Heating-up lasts longer, but relative axial displacement is smaller.

Besides the unfavourable phenomena related with the interaction of hot steam

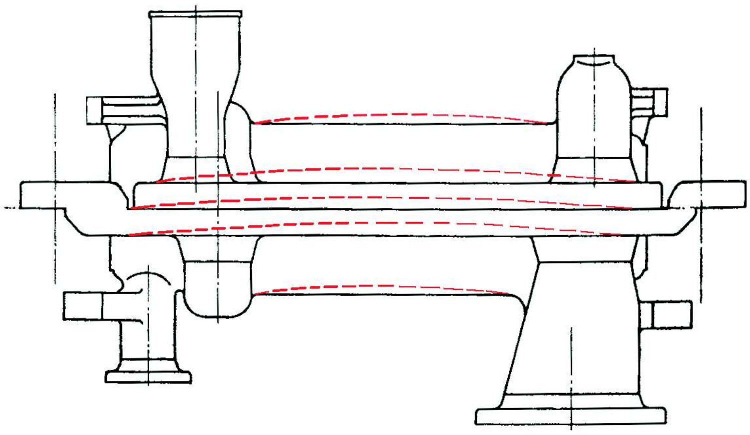

Figure 11. Bow of casing towards top, so-called cat's back.

and high-temperature components during starts, also negative influences occur in low-pressure (low-temperature) turbine. They are related with ventilation losses during idle run and concern the longest blades of last stages of LP turbine. Ventilation work causes heating-up of steam to some 200–250 °C and consequently the raise in temperature of casing, rotor discs and blades. Such a temperature increase is dangerous for rotor as it may cause weakening of disc shrunk fit on the rotor, LP casing heating, thermal stresses and change of clearances in the steam path. In order to reduce the consequences of ventilation, the LP exhausts are cooled by means of condensate injection. This cooling systems has however unfavourable effects. In case of flow separation or reverse flows in the last and penultimate stages, the injected condensate droplets are transported to the region of blade root and cause erosion of rotor blades trailing edges. This phenomenon can lead to the last stage blades cracking. Moreover, the wet steam cools the outlet edges of blades, while the remaining parts of them remains in the ventilation flow of higher temperature. Nonuniform temperature field in blade leads to thermal stresses, particularly high in trailing edges. Thermal stresses summing up with kinetostatic and dynamic stresses may lead to blade damage in the region of erosion notches weakening blade integrity.

Increase of steam temperature due to ventilation can also occur in high-pressure turbine. When start-up is conducted through IP turbine, HP turbine is under vacuum and its final stages may work in ventilation conditions. It leads to heating-up of components and increase of temperature to dangerous values. A means o reducing this phenomenon is the increase of steam flow through HP turbine, in which case steam expansion reduces the HP exhaust temperature.

Besides unfavorable phenomena of thermal nature, also non desired dynamic processes take place during start-ups, what additionally reduces turbine lifetime. These are resonant vibrations of rotating components of steam turbines. The design of steam turbines rotors is such that during run-up or loading increased vibrations of blades and shafts occur making difficult the turbine start-up or leading to damages to steam path components.

When increasing rotor rotational speed, the frequency of excitations resulting from multiples ($n = 1, 2...$) of rotational speed (k): $k \cdot n$ and multiples of the number of stationary blades(z): $z \cdot n$ also change proportionally. Natural frequencies of HP and IP rotor blades are on the level of thousands hertz and when increasing rotor rotational speed pass through the resonance with the multiples of the number of stationary blades. Long blades of LP last stages have lower natural frequencies, on the level of hundreds hertz, and when running-up the rotor pass through the resonance with frequencies being multiples of rotational speed. Passing through the resonance region is accompanied by increased vibration amplitude causing increase of blade dynamic stresses. This leads to fatigue of blade material, additional to fatigue resulting from kineto-static stresses. At a big number of turbine starts, the fatigue processes can cause blade cracking and such examples are known in steam turbine operation.

Also steam turbines shafts have critical speeds in the range 0–3000 rpm and at rotor run-up pass through the resonance region. In multi-cylinder condensing turbines of high power output, the critical speeds of HP, IP and LP turbine rotors and generator rotor are within the start-up speeds and during run-up several resonances of turbo-set rotors occur. In the resonance regions the increased vibration amplitudes can lead to decrease of clearances in labyrinth seals, rotor rubbing, and even blade failures and rotor bent. In order to reduce the consequences of resonant vibrations during start-ups it is necessary to pass the resonance regions as fast as possible, for example with acceleration 600–800 rev/min^2.

Besides synchronous rotor vibration, also low-frequency self-exciting vibrations can occur. Amplitudes of these vibrations are very high, and their frequency corresponds approximately to one of rotor critical frequencies. A reason of rotor self-exciting vibrations can be oil forces in bearings or aerodynamic forces in shaft seals. Low-frequency vibrations caused by aerodynamic forces appear rapidly after exceeding some load (so-called threshold load) and disappear after unloading the turbine below this power output. If in a turbine the phenomenon of threshold load occurs, then its start-up is impossible due to high vibrations and it's necessary to introduce design modifications.

3 Supervision and monitoring of start-ups

In the process of steam turbo-set start-up it is necessary to control thermal and dynamic processes by measuring appropriate parameters in the turbine flow path. Basic principles of start-ups are given in the turbine operating instructions and variation of rotational speed, power output and steam parameters over time is described in the so-called start-up curves. The start-up curves are generated independently for each individual turbine type and their run depends, among others, on boiler start-up curves, type of start-up, turbine design, design turbine lifetime and grid requirements. An example of start-up curves of condensing steam turbine with reheat is shown in Fig. 12. These are start-up curves for a cold start after 72 h of unit standstill. The start-up curves provide the recommended variations over time of such parameters like rotor rotational speed, turbine power output, live and reheat steam temperature and pressure. Similar character have start-up curves for a warm and hot start, but with correspondingly shorter times to synchronization and nominal load. Nowadays' turbine designs for conventional coal fired plants enable fast start-ups of duration typically between one and three hours.

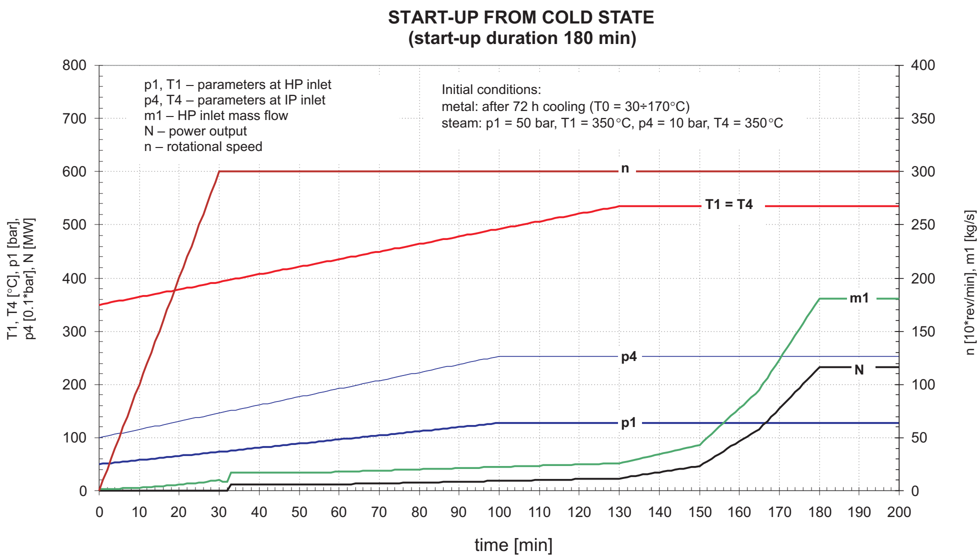

Figure 12. Start-up curves of steam turbine.

In turbine engineering practice, distinguishment among different start-ups is done basing on the following criteria:

- cold start: $T_0 < 170$ °C or standstill time $t_{standstill} > 60$ h,
- warm start: 170 °C $< T_0 < 430$ (steady-state casing temperature, -100 °C) or standstill time $8 \leq t_{standstill} \leq 60$ h,
- hot start: $T_0 > 430$ (steady-state casing temperature minus, 100 °C) or standstill time $t_{standstill} < 8$ h,where T_0 is the initial temperature.

For designing start-up curves turbine manufacturers employ special engineering tools dedicated for machines manufactured by them. Currently designed start-up curves are optimal ones and ensure minimal start-up times while maintaining components' stresses within the permissible limits.

The control of turbine components heating-up is realized on the basis of steam and metal temperature measurements as well as the measurements of relative expansion. In order not to generate excessive thermal stresses and deformations, the measured steam-metal temperature differences and steam and metal temperature rates must be kept within the permissible limits.

The values of criterion temperature differentials and rates are evaluated by calculations and subsequently verified experimentally on real turbines. Typical values of these criterion quantities recommended for steam turbines components are listed in Tab. 1.

Table 1. Permissible temperature rates in selected steam turbines components [K/min].

Component	Temperature range [°C]		
	<200	200–400	>400
Live steam pipeline	5	4	3
HP stop valve casing	3	2	2
IP stop valve casing	4	3	3
HP and IP control valves casing	6	5	3
HP and IP inner casing	3	2	2
HP and IP outer casing	4	3	3

Additionally, on some turbines the temperature differentials in inner and outer casings are supervised. Permissible values of temperature differentials depend on the casing design and radial clearances in the turbine steam path. It is usually recommended that the differences not exceed 50 °C. In order to reduce the consequences of non-uniform heating-up of top and bottom halves of outer casings and to maintain the temperature difference within permissible limits, special heating mats are used. They switch on when the temperature difference in the casing is approaching the permissible limit and cause heating-up of the bottom valve. They are in operation mainly during a start-up phase when the temperature differences are biggest. When steady-state is reached casings get heated-up and temperatures of both halves practically equalize. A heating mats installation on the turbine outer casing is shown in Fig. 13. This figure presents a schematic arrangement of heating mates (marked red) and casing metal temperature measurement points dedicated to the control of heating-up.

Heating installations are also used for bolts and casing flanges. Bolts are heated-up more slowly than flanges due to thermal resistance of screw joint. Excessive temperature difference would lead to bolt permanent deformation and leakages on the parting plane. This phenomenon is particularly clearly visible on casings with thick and heavy flanges. Bolts heating-up aims at limiting unfavourable temperature difference flange-bolt and enabling turbine start-up with normal rate.

Flanges are also the reason of casing nonuniform heating and thermal deformations. Nonuniform temperature field exists also within the flange which results in significant thermal stresses during turbine start-up or shutdown. Flange heating using steam efficiently reduces casing temperature differences and prevents from generation of dangerous stresses and deformations.

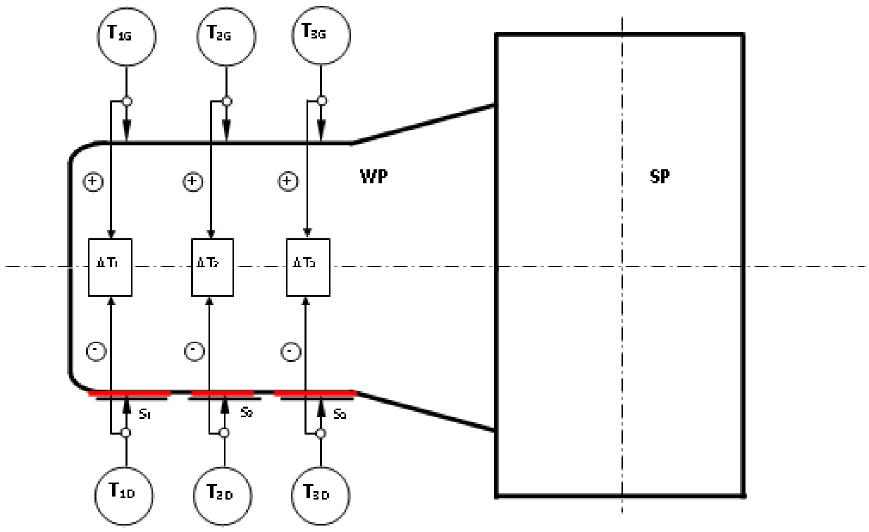

Figure 13. Installation of heating mats on turbine outer casing.

At fast start-ups in thick-walled turbine components like rotors or casings, high thermal stresses are produced, which in the areas of stress concentration can exceed the yield stress. Cyclic repetition of transient thermal stresses during start-ups and shutdowns leads to thermal fatigue of material and crack initiation provided that the frequency of starts is so high that exceeds fatigue life of the material.

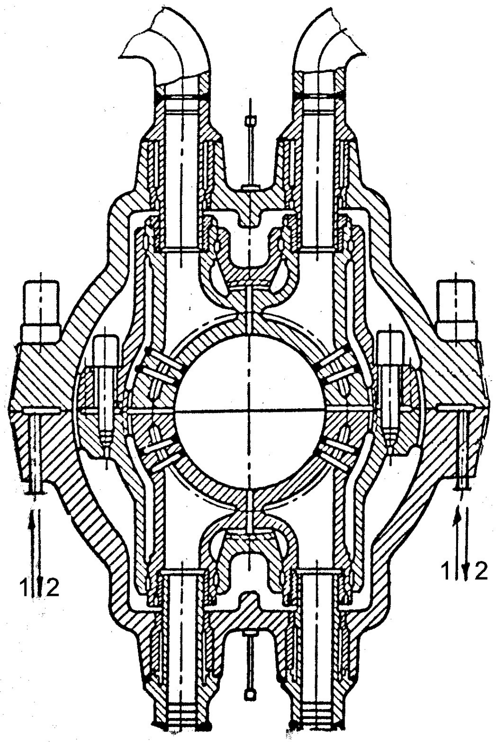

Figure 14. Heating of casing flanges.

In order to control start-up thermal stresses special supervision systems are used, namely the thermal stress controllers (TSC). Thermal stress controller is a system monitoring and controlling turbine operation. It supervises unsteady thermomechanical states of the machine and directly influences its safe operation. The main task of TSC is control of turbine start-up/shutdown in order to safely use its loading capabilities depending on the material condition. It enables so fast changes of thermal loading so as to protect the turbine against exceeding the permissible stresses. Control takes place by correcting the program of speed and load increase implemented in the turbine governor. The degree of limiting the rate of speed and load variation is a function of maximum effort of the most highly loaded location of turbine major components.

Thermal stress control is done simultaneously in several turbine components. Most frequently these are HP and IP rotors and inner casings. Component stresses are calculated in the most highly loaded locations and as a criterion value is selected the maximum stress. The maximum stresses are compared with permissible

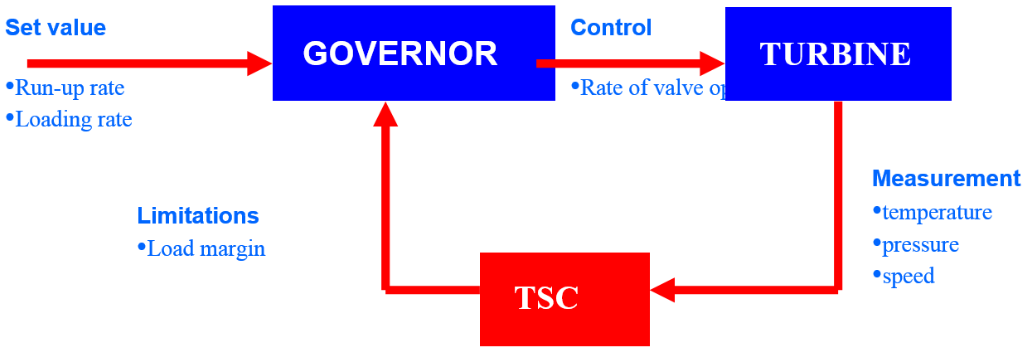

Figure 15. Structure of start-up supervision system with TSC.

stresses derived from material fatigue characteristics and on this basis relative load of components is determined and expressed as load fraction. During start-up the load fraction of individual components significantly changes but cannot exceed 100%. At different phases of start-up also the leading (i.e., mostly loaded) components change and stress maxima can occur several times during one start. An example of such a start from a warm state is shown in Figs. 16 and 17. At the early phases of start-up most highly loaded are IP turbine components and IP inner casing is a leading one. After some 3 hrs HP turbine starts dominating and its rotor attains 95% of permissible load. The load fractions of all components are characteristic of several peaks causing material fatigue damage.

3.1 Differential expansion

During rotational speed and load variations, relative changes of dimensions of rotating (rotor) and stationary (casing) components occur. The reason for this is double:

- temperature difference between rotor and casing, resulting from different heating conditions,
- rotor axial contraction due to centrifugal forces.

Differential expansion exists also at steady state but largest values are assumed during transients (run-up, loading). During start-up rotor expands radially due to both rotational speed and higher than casing temperature. Relative change of dimensions leads to decreased clearances in the turbine steam path, both in radial (radial clearances) and axial (axial clearances) direction. Excessive expansion would lead to zero clearances and rubs in the steam path resulting in damages to blading and seals. Particularly dangerous are axial expansions whose

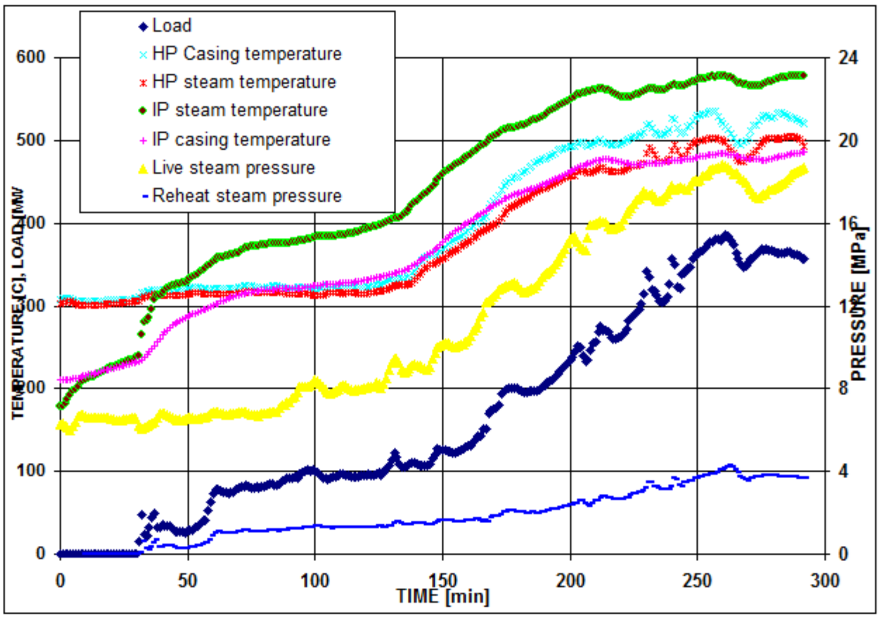

Figure 16. Variation of steam and metal parameters during start-up.

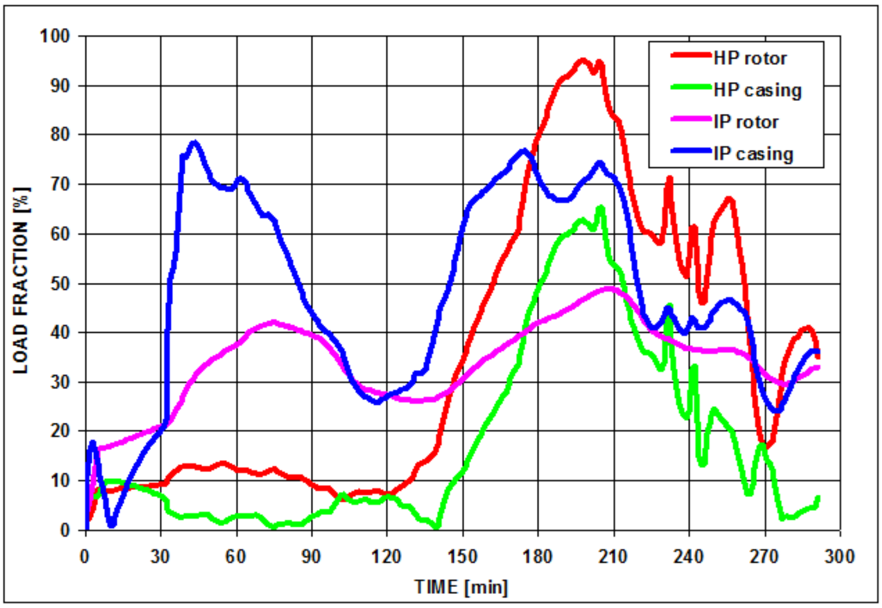

Figure 17. Variation of components load fraction during start-up.

direction may be different in various sections of the turbine and different for individual components. Exemplary schematic of rotor and casing thermal expansion of multicylinder condensing turbine of high power output consisting of high pressure turbine, double-flow intermediate pressure turbine and two double-flow low pressure turbines is shown in Fig. 18. Particular colours indicate:

- blue – inner casing expansion and relative fix-point of inner and outer casing,
- green – outer casing expansion and absolute fix-points of outer casing/bearing pedestal and foundation,
- red – rotor expansion and relative fix-point of rotor in combined axial-radial bearing.

For supervising differential expansion during turbine operation position sensors are used and connected with expansion displays. An example of LP turbine differential expansion control is shown in Fig. 19.

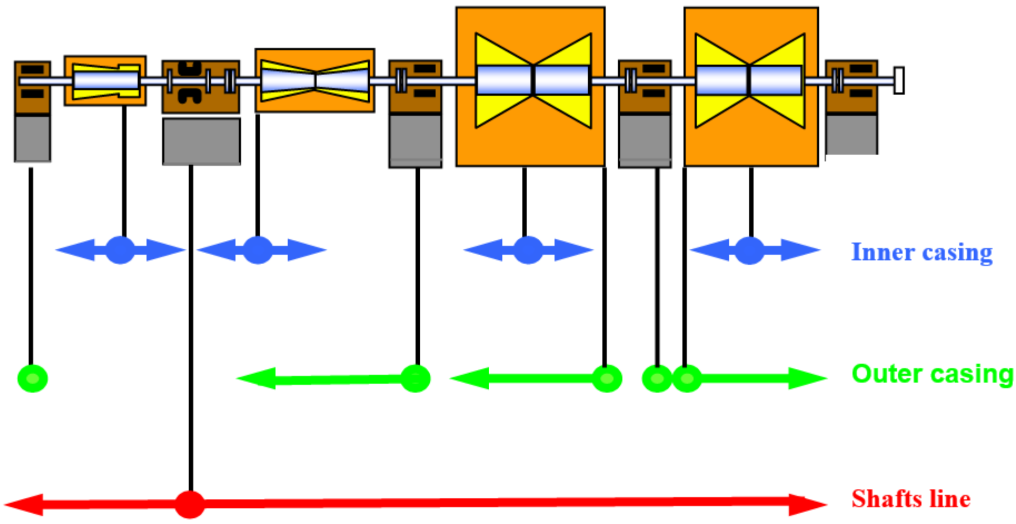

Figure 18. Schematic of multicylinder condensing turbine expansion.

3.2 Axial displacement

Axial displacement measurement is used to protect primarily rotor and blading, not the bearings. For bearings protection thermocouples are used. Typical values of axial displacement trigging alarm are ±0.4 (0.6) mm while turbine is typically tripped at ±0.8 mm. An example of axial displacement measurement is shown in Fig. 20.

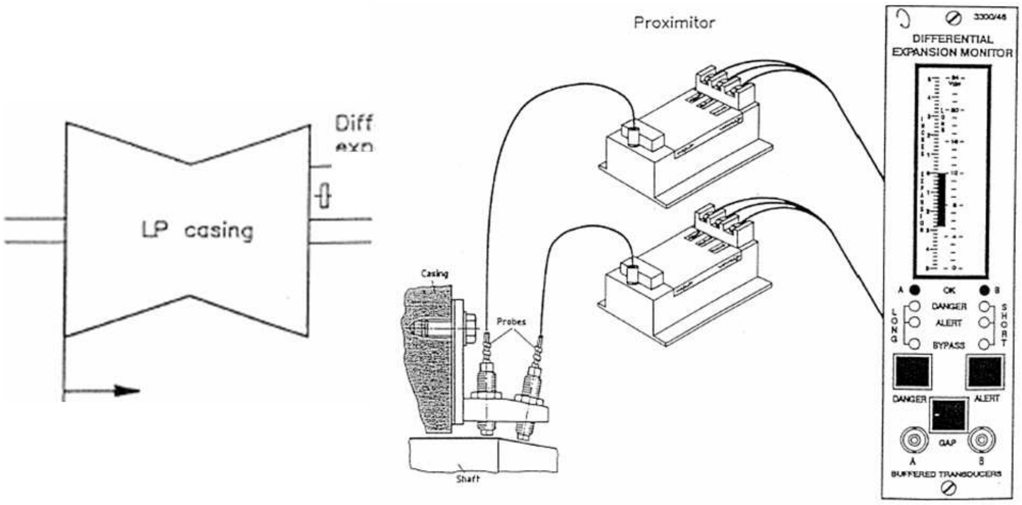

Figure 19. Measurement system of LP turbine differential expansion.

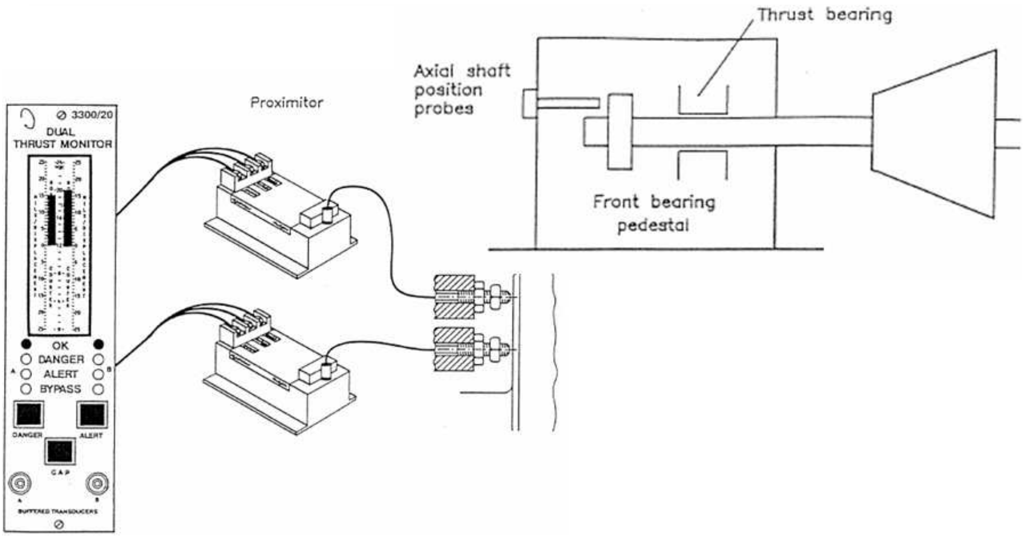

Figure 20. Measurement system of shaft axial displacement.

3.3 Vibrations

It is essential for the health of rotating equipment that it is operated with rotor and bearing vibration levels that are within the recommended limits. Excessive vibration can cause damage to the seals within the plant, leading to increased leakage flows, reduced efficiency and other possible problems as well as causing increased dynamic loading on the bearings and rotor support structure. One of the most common faults that can prevent or restrict the operation of rotating machinery in power plants is excessive rotor or bearing vibration. Examples of malfunctions that can give rise to increased vibrations are:

- Unbalance
- Bearing misalignment
- Coupling eccentricity
- Oil and steam instabilities
- Mechanical looseness
- Rotor/stationary component interference
- Resonance
- Foundation anomalies
- Thermal bow
- Rotor cracks
- Blades deposits

Typically, for vibration control two relative vibration sensors are used for each bearing, located at an angle of 90$^{\mathrm{o}}$ relative to each other. Alarm is generated when vibration levelreaches the first threshold, for example 165 μm p-p, and the turbine is tripped when the trip threshold is exceeded, for example 260 μm p-p.
Traditionally, the supervision of turbine vibrations has been done on the basis of bearing pedestal vibration measurements (Fig. 21a). Machines are now often being operated at increasingly severe operating conditions such as frequent load changes, shift operation, extended periods between overhaul, extended operation lifetime. Consequently more restrictive requirements are being specified for operating vibration values in order to ensure continued safe and reliable operation. In such conditions, vibration measurements done on bearing pedestals may not adequately describe vibrational behaviour of steam turbines. Measurements of shaft vibration (Fig. 21a) enable more accurate and sensitive detecting of changes in turbine vibrational behaviour. Measurement is done using contactless transmitters in two directions at an angle of 90$^{\mathrm{o}}$. Shaft vibration measurement is particularly recommended for machines with casings that are relatively stiff and/or heavy compared to the rotor mass. Furthermore, measurements on the non-rotating parts may not be totally adequate for steam turbines, which have several modes of vibration in the service speed range. Examples of vibration sensors fixation on impulse and reaction machines are shown in Fig. 22.

4 Means of starting versus turbine design

Turbine run-up and loading take place by means of increasing the steam flow fed to the turbine through the control valves. The relationship between steam flow

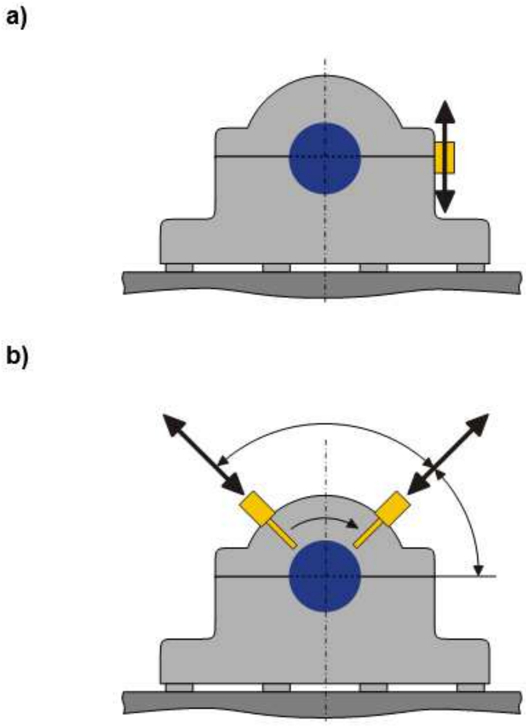

Figure 21. Vibration measurement of bearing pedestals (a) and turbine shaft (b).

rate and inlet steam pressure (downstream control valves) is linear for condensing turbines, thus the power output is determined by steam pressure before the blading. This pressure depends on the control valves opening and the live steam pressure. The above relationship gives an opportunity to control power output either by changing the live steam pressure or changing the cross section area of control valves. In practice, various combinations of the above means of power control are used, which affects the design of steam distribution system (number of control valves) as well as the design of steam turbine inlet sections (nozzle boxes, inlet scroll, control stage). Operation with variable inlet pressure is called sliding pressure operation, while operation with constant inlet pressure – constant pressure operation. The former mode of operation can only be used in unit systems, where one boiler cooperates with one turbine. From the point of view of pressure drop realization in the steam distribution system, the following types of control can be distinguished:

- throttle,
- nozzle,
- by-pass.

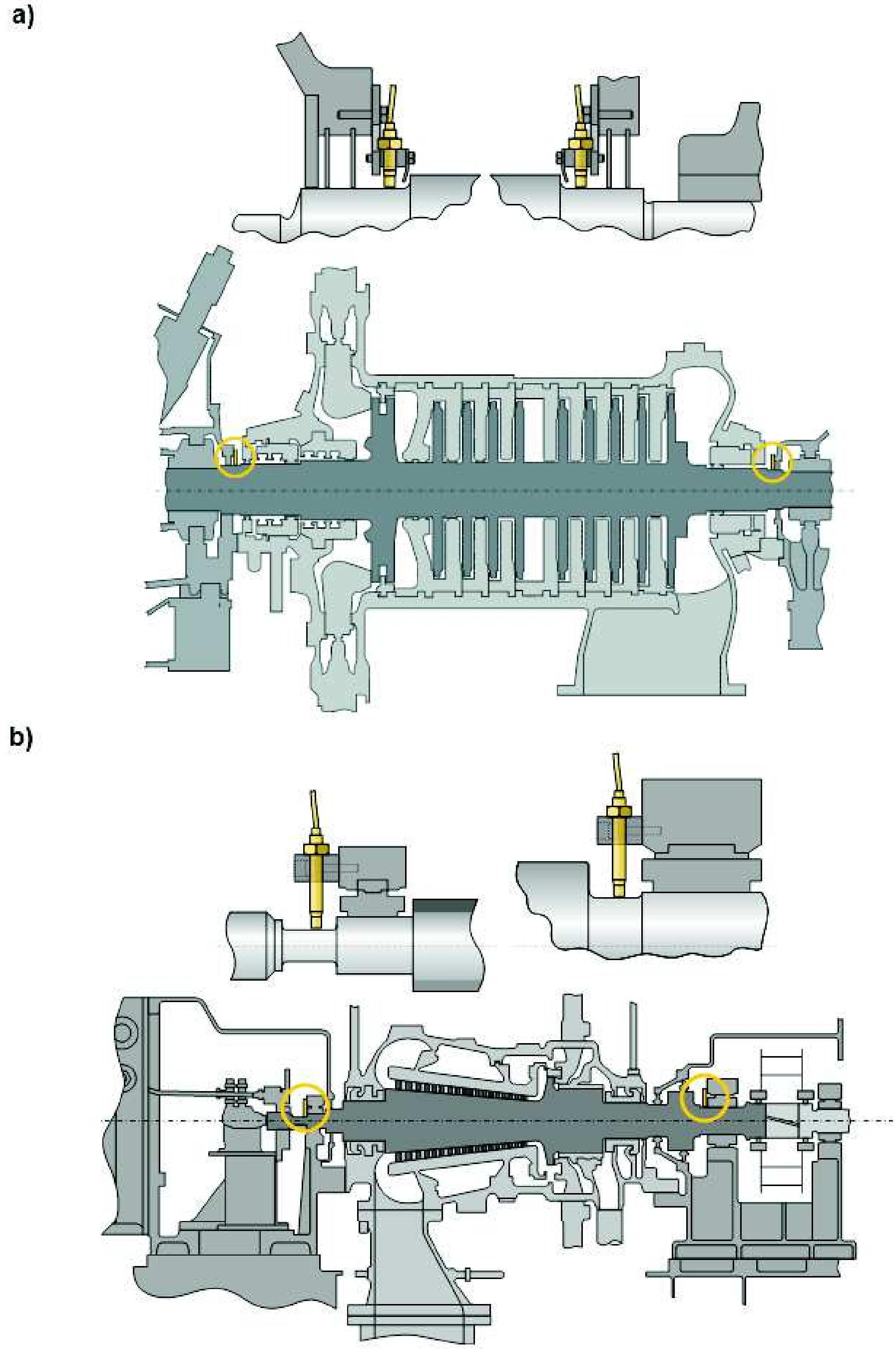

Figure 22. Fixation of relative vibration sensors in impulse (a) and reaction (b) turbine.

Throttle control is suited for turbines dedicated for base load operation, operated near the design point with control valves wide open. If turbine is to be frequently operated at low loads, then throttle control is not suitable from thermodynamic point of view as it leads to significant reduction of turbine efficiency at low loads, particularly in nonreheat machines. At this type of control there is no need to use the control stage and the turbine casing can be designed with inlet scroll (Fig. 23). The major advantage of such a control are favourable operation characteristics and the turbine can be quickly loaded and deloaded. In variable operating conditions, temperatures in turbine stages are only slightly changing and due to this during transients big temperature changes do not occur in turbine sections. Consequently, the resulting transient thermal stresses are not high, as well as differential and absolute expansions.

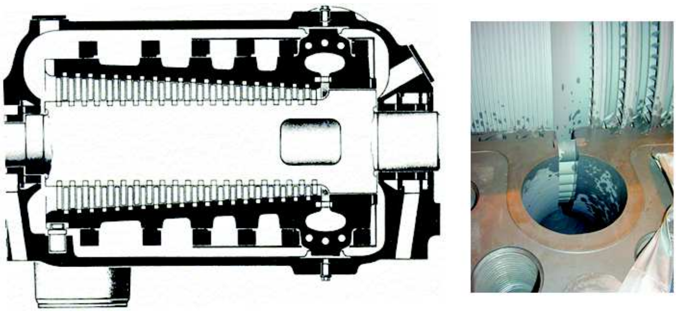

Figure 23. Turbine with inlet scroll.

Turbines dedicated to operation at variable operating conditions have nozzle-type control. In this case, turbine rotor is designed with a control stage (Fig. 24), and inner casing is equipped with nozzle boxes supplying steam to the rotor. Nozzle control is more favourable than throttle control as throttling losses occur only in a portion of steam flowing through not fully open control valves. During turbine loading big changes of enthalpy drop take place in the control stage, resulting in big variation of steam temperature in the control stage chamber. Steam temperature variation in the HP downstream stages is not so intensive. Big temperature changes over time cause fast heating of turbine components leading to the generation of high thermal stresses. Also rotor and casing expansion is bigger which worsens turbine operational flexibility as compared to throttle control.

The most rarely used is by-pass operation where steam is partially supplied direct to downstream stages by-passing the inlet stages. This type of control is used for overload control and typically is combined with throttle or nozzle group control.

The most frequently used means of turbine start-up together with their advantages and disadvantages are summarised in Tab. 2.

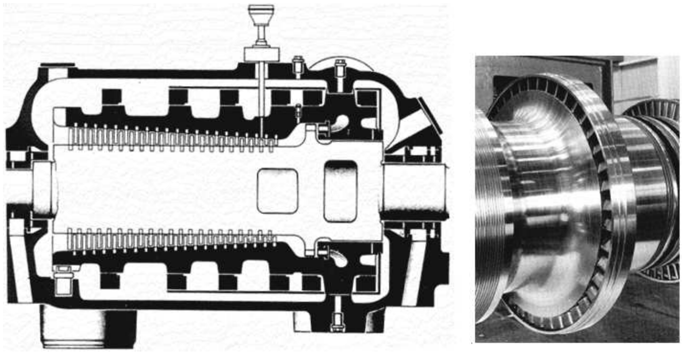

Figure 24. Turbine with control stage.

Table 2. Most frequently used means of starting-up steam turbines.

Means of start-up	Characteristics	Advantages	Drawbacks
Constant pressure operation with throttle control	Constant live steam pressure in the entire load range Throttle control Typically two control valves No control stage	Load step changes Low cost	High throttling losses in partial load operation
Constant pressure operation with nozzle group control	Constant live steam pressure in the entire load range Nozzle group control Typically four control valves Control stage	Load step changes High efficiency at partial load operation	Expensive control stage
Pure sliding pressure operation	Live steam pressure changes proportionally to live steam flow rate No throttling in control valves Typically two control valve No control stage	Low cost	Quick load step changes for frequency control not possible
Modified sliding pressure operation	Load changes according to grid requirements Valves not wide open Typically two control valves No control stage	Quick load step changes possible in the range of throttling	High throttling losses in the entire load range
Modified sliding pressure/constant pressure operation	Sliding pressure at low loads constant pressure at high loads Typically four control valves Control stage	Quick load step changes possible	Expensive control stage

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