TURBOMACHINERY & PUMP SYMPOSIA | HOUSTON, TX
DECEMBER 14-16, 2021
SHORT COURSES: DECEMBER 13, 2021

L120: THERMO-STRUCTURAL TRANSIENT ANALYSIS OF A BOILER FEEDWATER PUMP

Elicio Giulio
R&D Principal System Engineer
Baker Hughes
Bari, Italy

Annese Francesco
Engineering Manager
Baker Hughes
Bari, Italy

Giulio Elicio is principal engineer with large experience in centrifugal pumps, working since 2010 in the R&D department as system engineer for Baker Hughes. He joined Baker Hughes in 2007 after 3 years' experience in other engineering companies and worked for the R&D and standardization of the centrifugal compressors in Florence. Mr. Elicio holds a M.S. degree (mechanical engineering) from the Politecnico di Bari.

Francesco Annese is team manager of a group dedicated to Centrifugal Pumps and Valves, design and technical support to commercial operation for Baker Hughes. He joined GE in 2007 after 2 years' experience as design engineer for gas metering systems. Mr. Annese holds a M.S degree (mechanical engineering) from the Politecnico di Bari.

ABSTRACT

In the last few years, the installed capacity and production of electricity from all the renewable energies have significantly risen, supported by an ever-increasing spread of new energy policies all around the world. Because of that, thermoelectric power plants suffer of high variability: the continuous starts & stops per day of the steam circuit, instead of the reliability during continuous operation, become the main requirement for typical boiler feedwater pump. The most critical item of the steam circuit is the Boiler Feedwater Pump (BFWP): it is typically a multistage, between-bearing, single-casing ring section or double-casing barrel pump, designed for power rating up to tens of megawatts and water temperatures in the range of 150 °C (302 °F) and 210 °C (410 °F). The most critical condition that the pump can face during its life is the so-called "Cold Start-up": there can be extreme and exceptional conditions where the pump is required to promptly start-up from a cold state (site temperature, say 20/30 °C, 68 °F/86 °F) with the water at its maximum temperature (210 °C, 410 °F). Those conditions are extremely unfavorable for the pump: transient and alternating thermo-structural stress can lead, on one side, to the complete closure of the running clearances of the rotor and, on the other side, to fatigue stress of the tie-rods.

The running clearances of the pump, as for all turbomachines, are required to be as tighter as possible to reduce the power consumption, and this is of primary importance for a high-power pump. But, on the contrary, the tighter the running clearance, the more the risk of rotor seizure during the cold start-up. When the water at 210 °C (410 °F) runs over the pump at stand-still, the thin-walled components, such as the impellers and the rotating wear rings, expand more rapidly than the thicker-walled components, such as the stator that holds the statoric wear ring. The numerical approach of the finite element analysis (FEA) is used to find an optimal compromise between two conflicting needs: having a high efficiency pump (with the running clearances as tighter as possible) without the risk of pump seizure during the cold start-up, through the fulfillment of a defined criterion.

The continuous starts & stops cause also the pump tie-rods to be subjected to high-cycle fatigue, increased by the transitory thermal

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

load. Thus, the finite element analysis also verifies that the minimum design life of 30 years for the tie-rods is guaranteed.

The subjects of this paper are a barrel pump (BB5, as per API 610 designation), 6 stages with maximum power 15 MW and a ring section pump (BB4, as per API 610 designation), 8 stages with maximum power 4.5 MW running with the maximum water temperature of 210 °C (410 °F). The former is the typical Boiler Feedwater Pump used for coal-to-fire power plants in configuration 3x50%, the second for a combined cycle power plant in configuration 2x100%. Outcomes of the simulation of the cold start-up are the axial deformation and displacement of the rotor, the radial deformation in close proximity to the running clearances and the stress of the tie-rods.

INTRODUCTION

The typical pump selected for boiler feedwater application is dependent on the power plant type. Double-casing, radially split, multistage, between bearing pumps (barrel pumps, BB5 as per API 610 designation) for coal-to-fire power plants; the output power can be of tens of megawatts. Single-casing, radially split, between bearing pumps (ring section, BB4 as per API 610 designation) for combined cycle power plant; the output power is up to 7 MW. For both, the temperature of the superheated water is in the range of 150 °C (302 °F) and 250 °C (482 °F), and the liquid phase is guaranteed by high suction pressures.

Due to such high power involved, the efficiency of the pump is a key point in the selection and the design: the annular seal clearances are as tight as possible to increase the volumetric efficiency. Also, from a rotor-dynamic point of view, it is very important to have tight clearances, because the operating fluid (hot water) has lower density and viscosity, resulting in lower stiffening effect.

Unluckily, tight tolerances are unfavorable during one of the particular operating conditions of the boiler feedwater pump: the cold start-up. According to the load required by the power plant, there can be exceptional cases when the pump is started-up from stand-still at ambient temperature with the operating fluid entering the pump at its maximum temperature. In these cases, all the thin-walled components, such as the shaft and the impellers, experience a thermal expansion more rapid than the thicker-walled components, such as the stator. In unfavorable conditions, this can lead to complete blockage of the rotor when at least 2 opposite points of the annular seal result to have zero gap. The severity of this failure mode is furthermore increased for unspared configurations; but in any case, since the impact of such an undesirable event for the plant is significantly high, actions to mitigate the risk shall be put in place.

A way to prevent the failure mode, without being forced to increase the annular seal clearances, is by warming-up the pump. This is the typical solution deployed for spared configurations: a small flowrate exiting from the operating pump is diverted to the stand-by pump from suction drain with the rotor slowly rotating. This solution has the advantage to keep the pump at a temperature almost equal to the operating one so that it can be instantaneously operated without any risk of rotor blockage; the drawbacks are the additional costs due to the controls and the loop, along with the steam consumption.

Another way to mitigate the risk, when the warm-up cannot be an option (for cost impact or for unspared configurations where the steam for the warm-up is not available), is a detailed finite element analysis of the thermo-structural transient of the cold start-up, considering the non-steady-state heat transfer and conduction. The model used in the analysis can be an axisymmetric 2D geometry or a complete 3D geometry. Whatever the geometry, the most important phase of the analysis is the set-up of the FEA model, since the thermal characterization of the system results to be quite complex, due to the stagnation regions present in the pump, the contact regions, the main passages of the heat load not easily determinable without the help of CFD calculations. That's why a complete and accurate methodology should be made of defined steps: contact regions and stagnation regions characterization, CFD characterization of heat and fluid flow within the pump; a thermal transient analysis of the whole system; a mechanical transient analysis. The pumps under investigation are presented in the FIGURE 1 and FIGURE 2. The first is a BB5 diffuser pump, 6 stages, 15 MW at 5850 RPM, rated capacity 1340 m^3/h (5900 US GPM); the impeller size is 16 inches with front wear ring diameter of 230 mm and rear wear diameter of 180 mm, radial gaps according API 610 prescriptions (0.53 and 0.48 mm respectively); total length of the rotor is 3150 mm and journal bearing of 140 mm. The second is a BB4 diffuser pump, 8 stages, 4.5 MW at 3480 RPM, rated capacity 630 m^3/h (2773 US GPM), the impeller size is 15 inches with front wear ring diameter of 220 mm and rear wear diameter of 155 mm, radial gaps according API 610 prescriptions (0.5 and 0.45 mm respectively); total length of the rotor is 2550 mm and journal bearing of 121 mm. They both have an intermediate take-off to the heat recovery steam generator. The analysis has been performed on the 3D geometry; the scope of the analysis is the estimation of the axial deformation of the shaft, the radial deformations of the annular seals to monitor the gap in the transient, the transient stresses on all the pump components and finally the fatigue of the tie-rods for the ring section pump.

TRANSIENT THERMAL SIMULATION

CFD calculations of the Heat Transfer Coefficients

The hot water entering the pump is the responsible of the heat flow and generates a heat gradient; the flow heats up the wet surfaces of the pump through heat transfer. The thin-walled parts are heated up by the flow very rapidly, after few seconds, and this leads to very fast thermal expansion; the statoric components, on the contrary, take more time to reach the same temperature of the rotor and the fluid; furthermore, due to the big mass of the statoric component and the natural convection outwards, the expansion of the stator is slow and restricted by the external region of the stator.

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

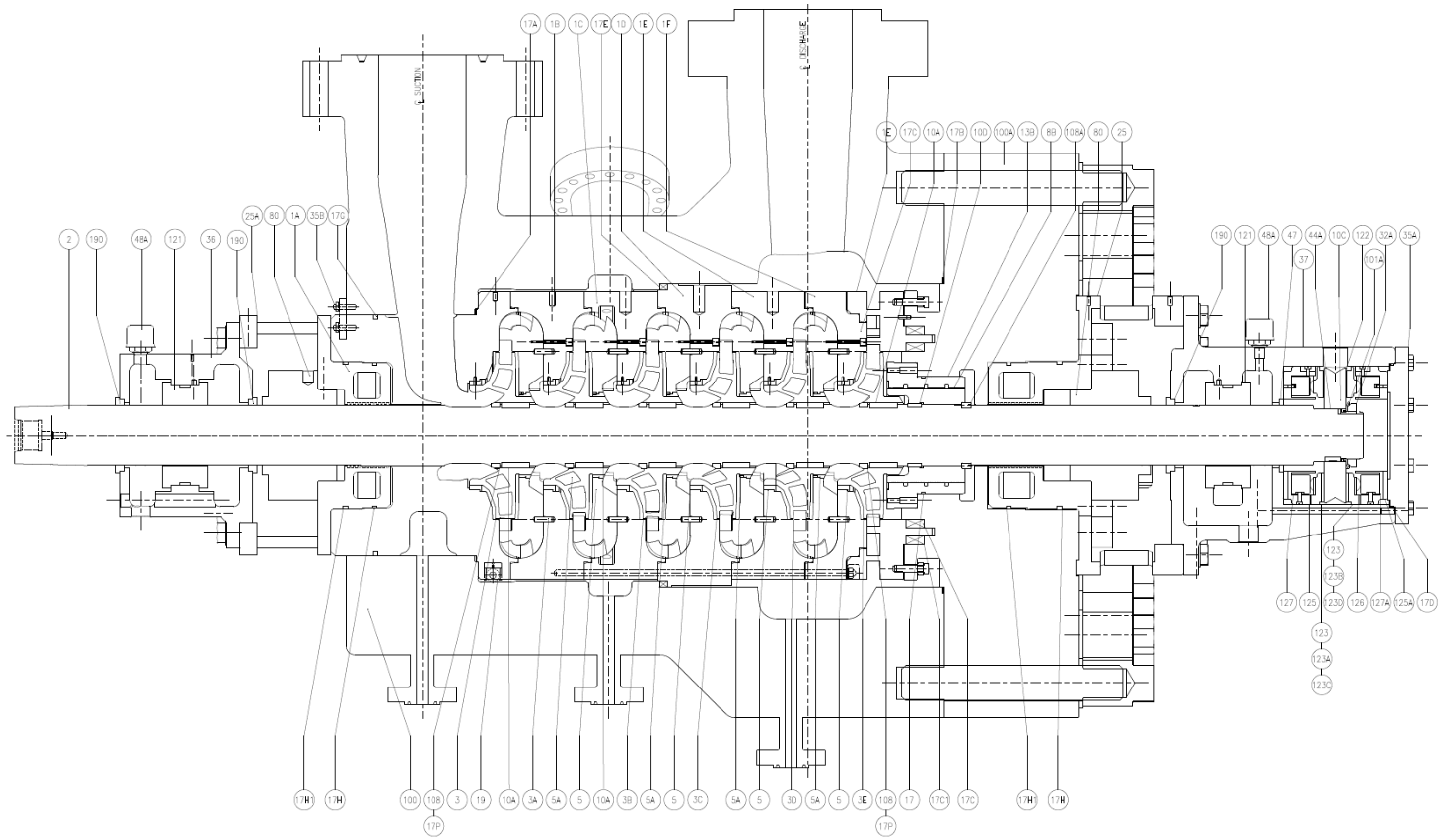

FIGURE 1: BB5 boiler feedwater pump for coal-to-fire power plant

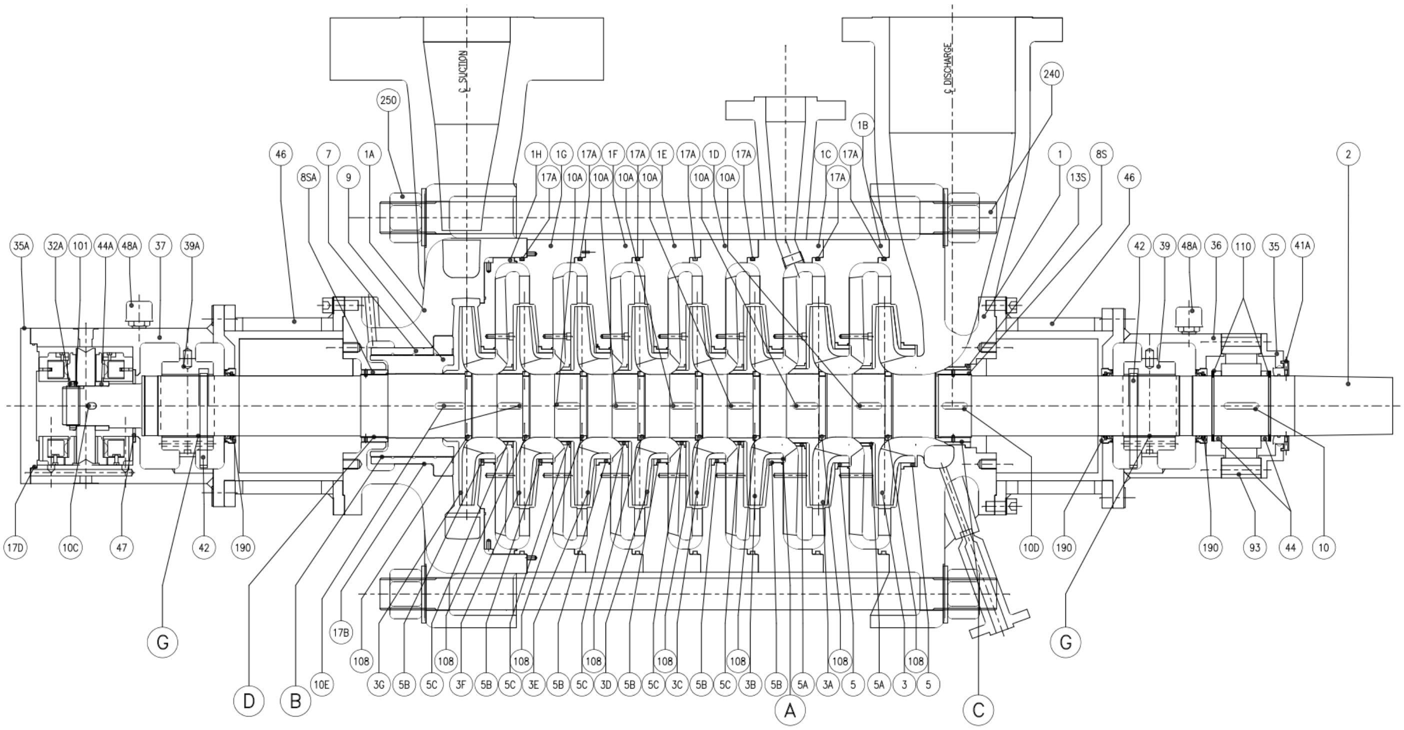

FIGURE 2: BB4 boiler feedwater pump for combined cycle power plant

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

Computational Fluid Dynamics (CFD) simulations of convective and conductive heat transfer have been considered. The surface coefficients for heat transfer are generally not easily calculated by mean of analytical calculation. The values of surface Heat Transfer Coefficients (HTC, from now on) depend on defined variables: material properties, boundary conditions, and so on. To get reliable results from the analysis, it is very important to correctly set-up the model. The first step of the analysis is to get the mapping of the HTC of the operating fluid. To do that, for both the models, CFD calculations have been performed to map the heat transfer coefficients of the operating incompressible fluid (water) at the rated flowrates (1340 m3/h - 5900 US GPM for the BB5 pump, 630 m3/h - 2773 US GPM for the BB4 pump) entering the pump with the rotor at standstill: the wall temperature is superimposed to the solid fluid to calculate the heat transfer coefficients. The temperature considered is 210 °C (410 °F), the total simulated time 20 h, the natural convection conditions are applied at the outer surfaces.

The material of the pump components is belonging to the C6 class as per API 610 designation (12% chrome steel), except for the barrel of the BB5 pump that is in carbon steel. The characterization of the stagnation regions, like for example the sidewall gaps between the impellers and the statoric parts and the tight clearances of the sleeves (FIGURE 3), is calculated by thin material (water) regions and correlations by literature to calculate the HTCs and natural convection is applied at the outer surfaces.

The grids of the models are made of hybrid meshes (tetra and prisms): the final output gives 24 million elements for the BB5 pump, 47.8 million for the BB4 pump.

The mapping of the HTC as function of the pump flowrate (velocity) for the BB5 pump is reported in the FIGURE 4.

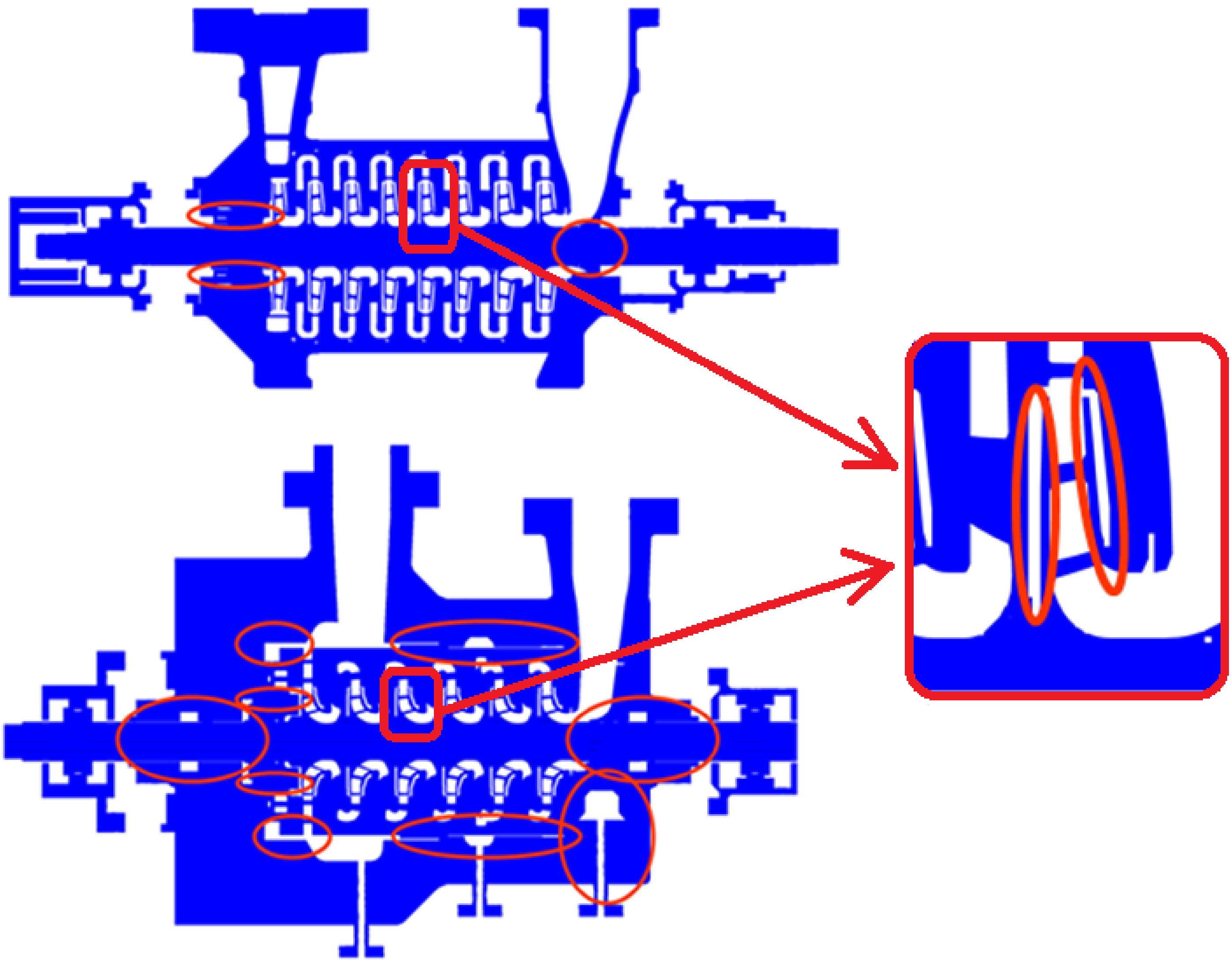

FIGURE 3: Stagnation regions for the 2 pump models: impeller sidewall gaps, annular seals and sleeves

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

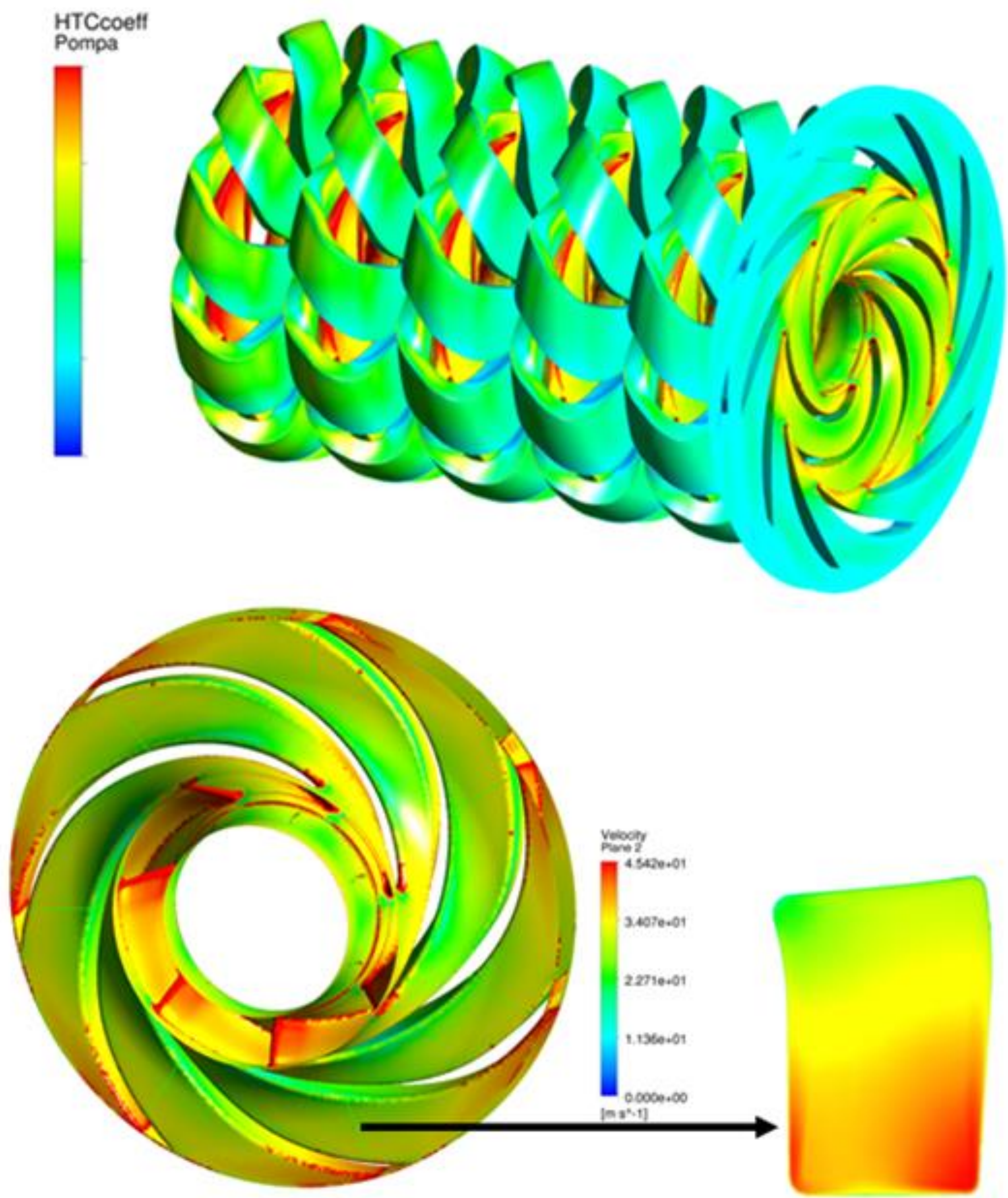

FIGURE 4: Heat Transfer Coefficient mapping for the BB5 pump as function of the fluid velocity

Thermal mechanical transient simulation

The HTC mapping has been used as a boundary condition for the mechanical simulation; the mapping is imported in Ansys Workbench v17® [2] by mean of an external Ansys routines. The energy balance has been checked: the heat transfer through all the internal surfaces has been prescribed and, to do that, area transfer functions are considered to assure a proper energy balance of the whole system. The mechanical simulation is conducted in a decoupled way, the rotor and the statoric assembly are simulated separately. After that, kinematic transfer function has been applied to the rotor domain during the post-process: axial rigid constraints are superimposed at the axial thrust bearing, vertical rigid constraints are superimposed at the journal bearings (FIGURE 5). The outer casings are fixed along the vertical and axial direction at the foot close to the suction, only along vertical direction at the foot close to the discharge; this is to allow the casings to move and compensate the thermal expansion.

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

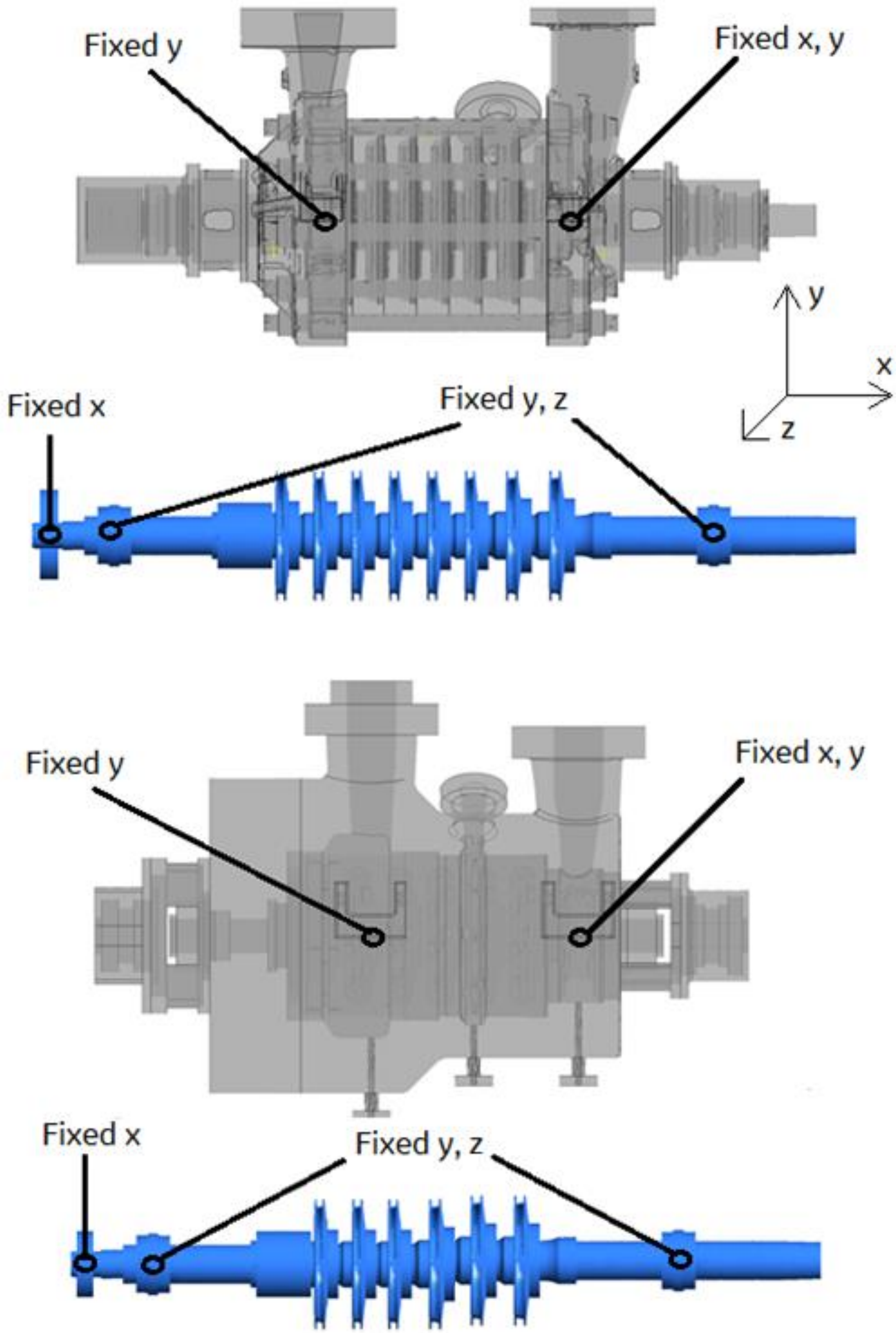

FIGURE 5: Constraints to the models: rotors and statoric assembly

Axial deformations

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

The axial deformations are first investigated and calculated to check the displacements superimposed by the thermal transients to the couplings: the maximum displacement shall be less than the admissible by the coupling. With reference to the FIGURE 6, the displacements of the thrust bearing supports have been reported keeping the foot close to the suction as the zero reference; the displacements of the couplings have been reported keeping the thrust wheel as the zero reference. Therefore, the overall rotor displacements at the coupling result to be the vectoral sum of the 2.

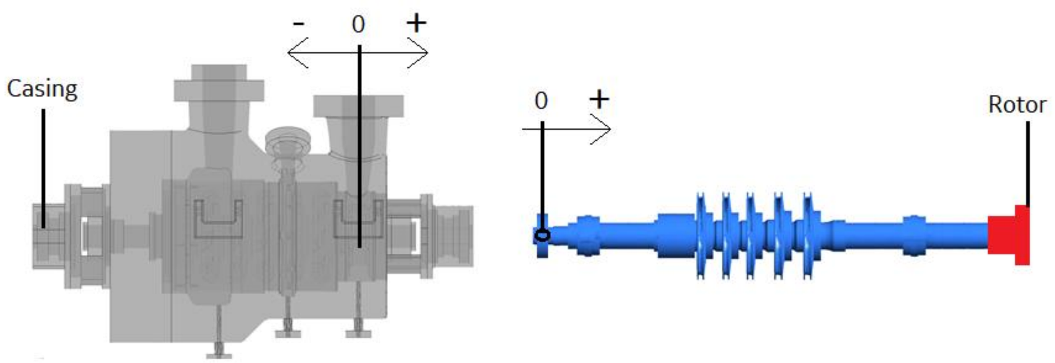

FIGURE 6: References for the axial displacements of the coupling (BB5 pump)

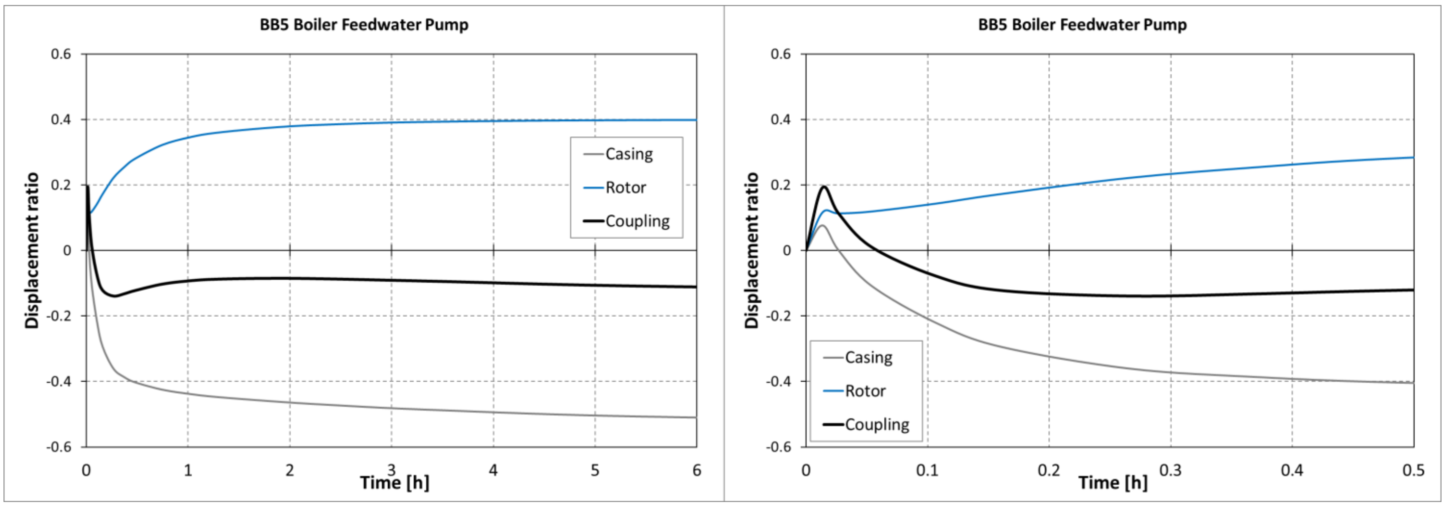

FIGURE 7: Transient axial displacement ratios of the BB5 pump at coupling as difference of the axial displacement of the casing and the shaft at different time scales (the right graph is the representation of the first half a hour)

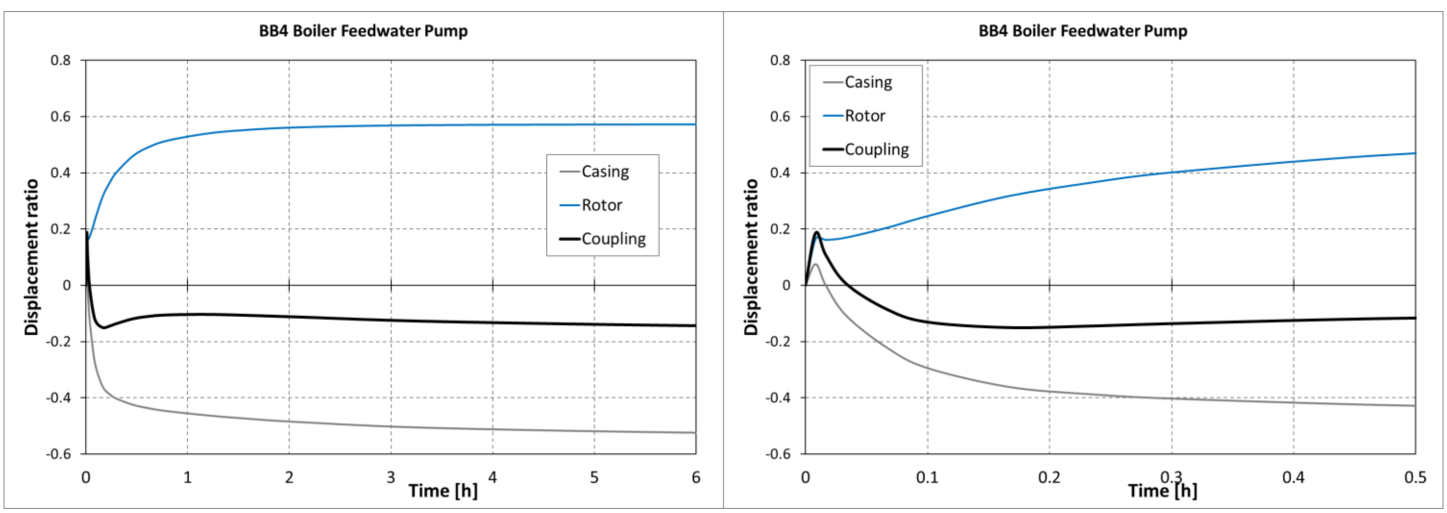

FIGURE 8: Transient axial displacement ratios of the BB4 pump at coupling as difference of the axial displacement of the casing and the shaft at different time scales (the right graph is the representation of the first half a hour)

The FIGURE 7 and FIGURE 8 report the overall transient displacements of the couplings (black curves) for the BB5 and the BB4 pumps respectively as difference of the rotor (blue curves) and casing (grey curves) displacements. The displacements are reported as

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

fractions of the maximum allowable coupling displacement. For both situations, the rotor displacements at the coupling are below the admissible value and this is true during all the transient time period. This means that the axial displacements meet the cold start-up requirements with an adequate safety margin (i.e., the remaining 25%).

Furthermore, from the graphs below, it can be also observed that the BB5 rotor has a thermal expansion lower than the BB4: this can be easily explained with the fact that the BB5 rotor has 3 fewer stages. The BB5 casing takes much more time than the BB4 to get to the steady-state temperature, 10 hours about against 3, due to the thermal inertia of the BB5 barrel higher than BB4.

Radial deformations

As said, the hot water entering the impellers is the source of the heat flow; the wet surfaces are heated through the heat transfer and their temperature rises reaching the fluid temperature at steady-state. The impeller walls are components thinner than the statoric components (the casing components) and, as a consequence, they reach the steady-state temperature earlier than the casings. Therefore, the annular seal clearances experience a reduction and the risk of rotor blockage is then increased. The condition to avoid the blockage of the rotor is identified when the average clearance is reduced by no more than 75% of the nominal: this may mean that in some control points the clearance can be lower than 75% (internal standard) and, in extreme cases, the rotor can lean on one point of the seal. It is typical of pumps, in fact, to start-up leaning and crawling against the seals, because of the elastic deformation of the rotors, that are very slender compared to any other turbomachinery: the stabilizing forces of the gyroscopic effect (particularly high for low specific speed impellers) and the Lomakin effect (due to the friction losses of the flow entering the annular seals) tend to lift the rotor and to center it in the annular seal clearances. To avoid the rotor stick, the annular seals are coated with a hard coating.

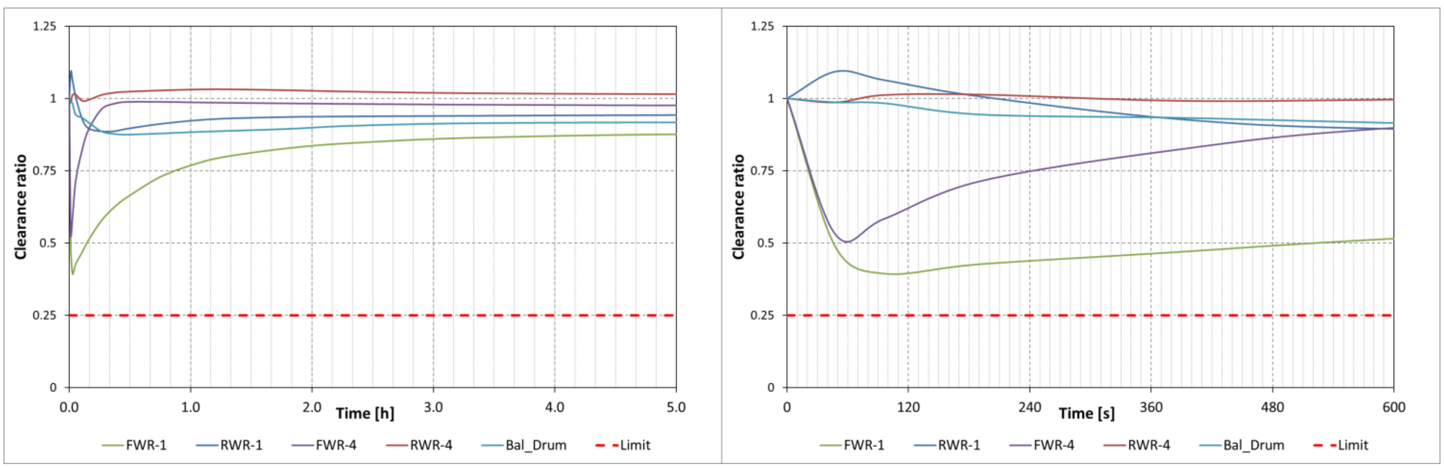

FIGURE 9: Average annular seal clearance of the first and forth seal, front and rear, and the balancing drum, BB5 pump at different time scales (the right graph is the representation of the first half a hour)

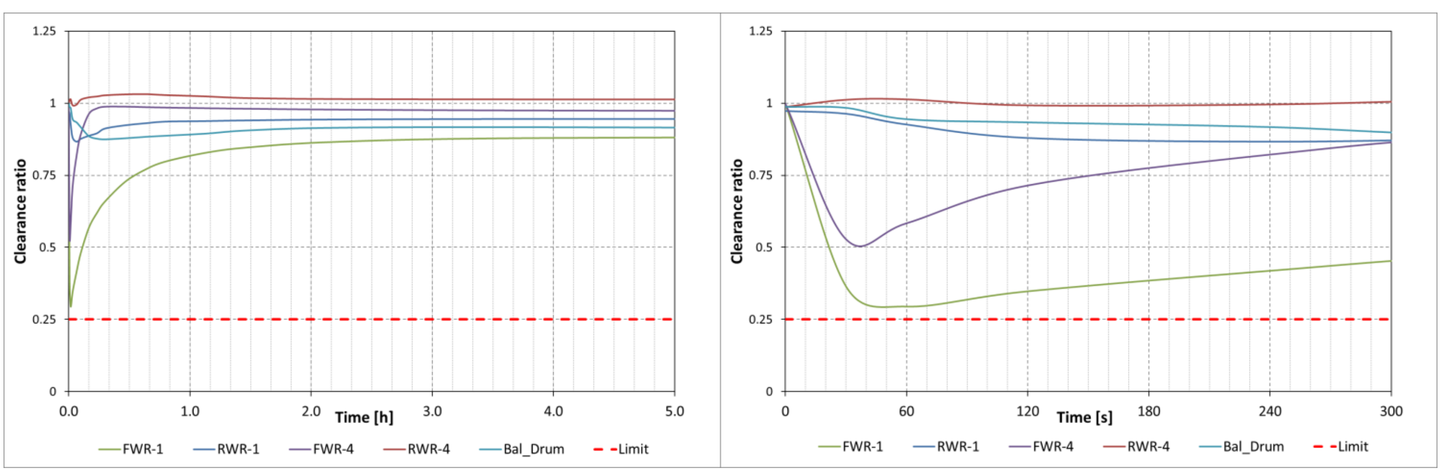

FIGURE 10: Average annular seal clearance of the first and forth seal, front and rear, and the balancing drum, BB4 pump at different time scales (the right graph is the representation of the first half a hour)

The thermal transient displacements have been checked considering a total of 4 control points, 2 along the vertical axis and 2 along the horizontal, equally spaced by 90°; the average clearances are calculated the basis of these 4 points (FIGURE 12). The seals under

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

investigation are the first stage, the balancing drum and one of the intermediate between the two; for all the seals considered, the front and rear seal displacements are calculated.

The FIGURE 9 and FIGURE 10 report the average clearances calculated for the BB5 and BB4 pumps; once again, due to the great thermal inertia of the BB5 casing, the BB5 reaches the steady-state condition later than the BB4. As can be seen, the first front annular seals for both pumps can be considered the most critical, even though the criteria result to be satisfied. The worst condition of the clearance happens after 100 seconds about for the BB5 pump, 45 seconds about for the BB4 pump. Although safely validated, the BB4 first seal clearance shows the closest condition to the safety transient limit (maximum allowable clearance reduction of 75%), due to the higher number of stages, and it deserves further investigations.

After the cold start-up, it can be seen that, although the average clearance is always positive and within safety margin, the vertical lower control point (the red curve "Vert_Down" of the FIGURE 12) has negative values from 0' 24" to 23' 30"; but the opposite control point (the green curve "Vert_Up" of the FIGURE 12), after 30 seconds about, begins to increase its clearance and tends to compensate the negative clearance of the lower control point. This can be only explained by analyzing the transient temperature distribution inside the pump and the consequent deformation (FIGURE 13). As can be seen, the geometry of the suction module has a thermal inertia different in the plane of the cross-section: in particular, the thin-walled suction nozzle is heated up faster than the lower thick-walled module, hence, the upper deformation of the module tends to rotate rigidly the bearing bracket support, causing a rigid downward movement of the rotor.

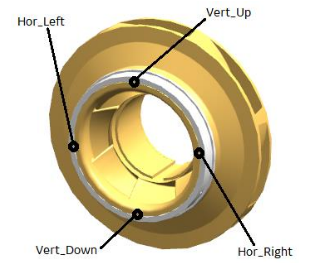

FIGURE 11: Control points of the annular seal clearances

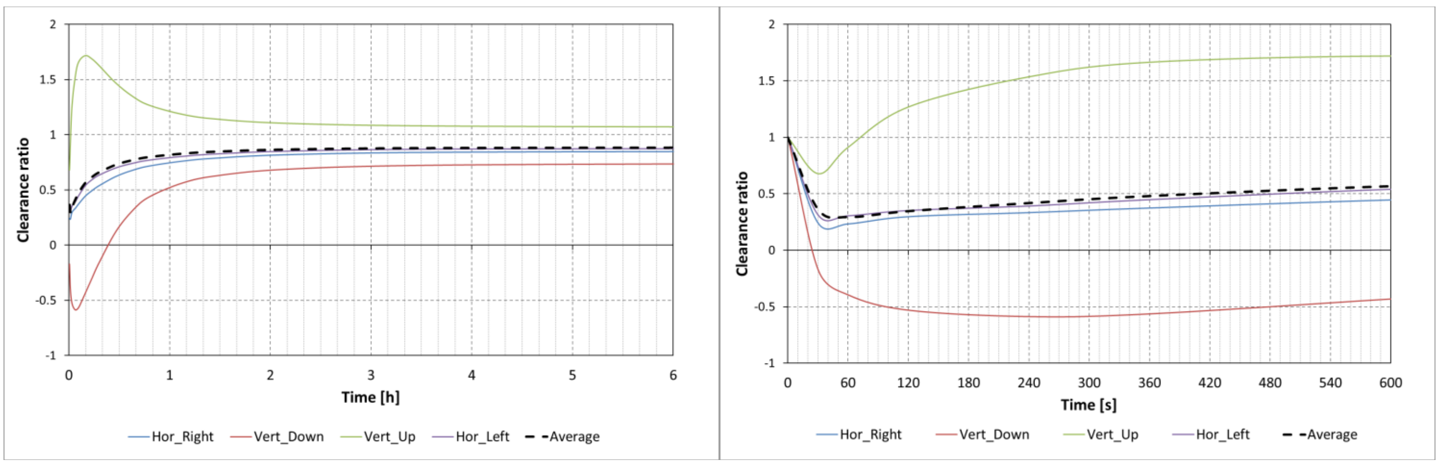

FIGURE 12: Front annular seal clearance of the first impeller, BB4 pump at different time scales (the right graph is the representation of the first half a hour)

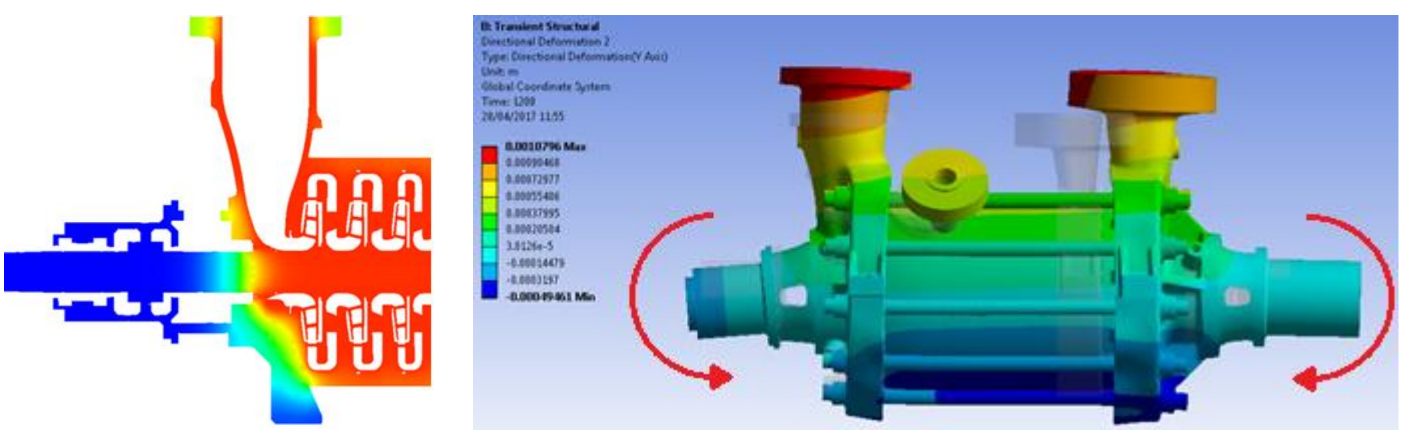

FIGURE 13: Transient thermal temperature distribution and deformations of the BB4 pump

With reference to the FIGURE 12, the rigid body motion of the rotor could be easily isolated considering the horizontal transient clearances as the product of the lone thermal transient expansion. Anyway, the following conclusions can be considered:

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

- The rigid body motion of the rotor does not affect the calculation of the average annular seal clearances, that is the main parameter to be checked to guarantee the safety cold start-up of the pump without any risk of rotor blockage due to the complete closure of the clearances;
- The negative value of the lower vertical control point is not a problem during the cold start-up, because it occurs after 24 seconds after the running: it is judged a sufficient time period for the pump to establish appropriate differential pressure and, thus, for the gyroscopic effect of the impellers and for the stiffening effect of the liquid annular seal to lift the rotor.
- Once reached the steady-state condition after more or less 7 hours (FIGURE 14), the rigid body motion, experienced during the transient, is not fully recovered and that's the reason why the vertical control points (FIGURE 12) do not fully converge towards the horizontal control points.

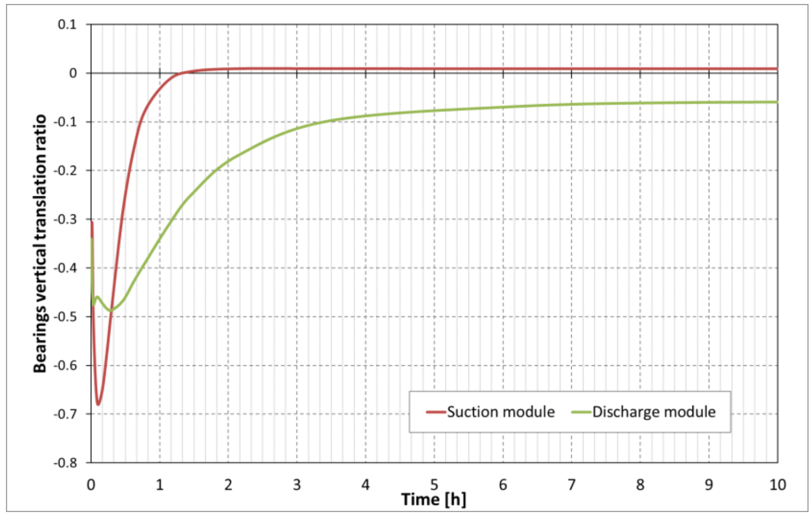

FIGURE 14: References for the axial displacements of the coupling (BB4 pump)

Stress and fatigue analysis

Along with the transient thermal deformations superimposed by the cold start-up to the annular seal clearances, it is also possible to define the thermal stresses of the components. The analysis of the stresses has been separated between the rotor and the stator.

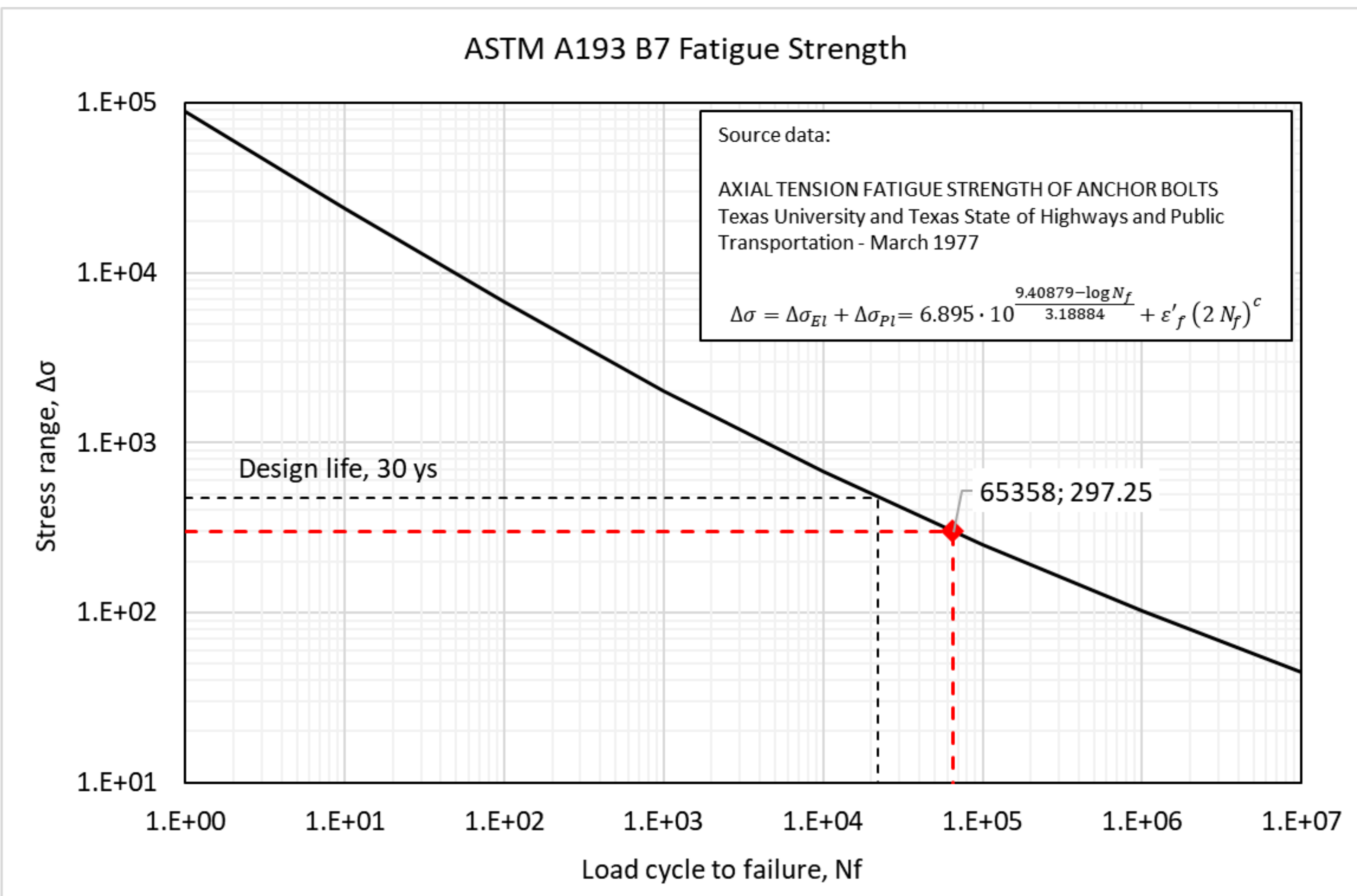

FIGURE 15: Fatigue stress of BB4 tie-rods

The stresses on the rotors aim to define the integrity of the shrink-fit of the impellers on the shaft: the impellers are, in fact, heated

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station

up faster than the shaft and, thus, there could be the risk that the interference between them could be compromised. The transient thermal calculation shows that, although the expansions are different, the interference is always guaranteed.

As far as the statoric components are concerned, to check their integrity, the thermal stresses are not considered in their transient condition, but steady-state: the only components analyzed in transient conditions are the tie-rods of the BB4 pump; the tie-rods material is ASTM A193 B7. Due to their peripheral position on the pressure boundary, they suffer very little the heat flow and the temperature increase is almost negligible, while the thermal expansion of the statoric components increase their tension. In the worst condition ever, the cold start-up can be repeated twice-a-day, exposing thus the tie-rods to fatigue stress. The analysis has considered the bolt tensioning and the maximum allowable working pressure, along with the transient thermal stress; the resultant alternating stress in the worst condition is 297.25 MPa [3]. The alternating stress corresponds to more than 65000 cycles for the tie-rods material (FIGURE 15), more than 89 years considering a maximum of 2 cold start-up per day. It is hence demonstrated that the design life of 30 years for the pump is guaranteed.

CONCLUSIONS

The cold start-up of a boiler feedwater pump is an extreme condition, which it can be subjected to, that needs to be verified to guarantee a higher variability of the loads without compromising the availability of the machine. The transient thermal stresses superimposed by the cold start-up on the 2 types of boiler feedwater pumps, barrel and ring section pumps, have been investigated through a fluid-dynamic and mechanical finite element analysis, based on 3D models. The CFD helps getting the mapping of the HTC of the heat flow generated by the hot water entering the flow paths; the CFD output has been used as an input of the mechanical finite element analysis. The performed verifications are below summarized:

- Axial displacement: during the transient, the thermal expansion of the rotor, made by thin-walled surfaces, is faster than the casing components, made, on the contrary, by thick-walled surfaces; it is important to verify that during this time period the resultant axial displacement is not greater than the maximum displacement allowable by the coupling. The calculation demonstrates that the requirement is fulfilled.
- Radial displacement: the close clearances of the annular seals are subjected to reduction due to the different thermal expansion of the statoric and rotating wear rings in the transient; 4 control points have been defined for first wear ring, the balancing drum and the intermediate seal between the two. The average clearances have been evaluated: the minimum average clearance shall not be reduced by 75% of as-new condition. Although the criterium is satisfied for all the clearances, further investigations are carried out on the first seal of the BB4 pump. It has been found that the non-symmetric geometries of the suction and discharge modules causes a rigid body motion of the bearing bracket support and, as a consequence, of the rotor; after 24 seconds from the start-up, the lower part of the clearance is zero: it is judged not compromising the rotor integrity, because the lifting forces of the gyroscopic effect and the rotor-dynamic stiffening effect are well established.
- Fatigue stress: due to the continuous starts-and-stops, the tie-rods of the BB4 pump are subjected to fatigue stress; although the cold start-up is an extreme condition, the worst case of 2 cold start-up per day has been considered; the transient finite element analysis takes into account the tightening torque, the maximum allowable working pressure and the transient thermal stress from CFD. The output alternating stress is the input in the fatigue strength graph of the tie-rods material: the design life verification is confirmed.

ACRONYMS

FWR	= Front wear ring
RWR	= Rear wear ring
Bal_Drum	= Balancing drum
HTC	= Heat transfer coefficient
Hor_Right	= Horizontal right control point
Hor_Left	= Horizontal left control point
Ver_Down	= Vertical lower control point
Ver_Up	= Vertical upper control point

REFERENCES

[1] Ansys CFX® v16, Turbomachinery CFD Software
[2] Ansys Workbench® v16, Turbomachinery Mechanical Software
[3] Axial tension fatigue strength of anchor bolts, March 1977, Texas University and Texas State of Highways and Public Transportation.

Copyright© 2021 by Turbomachinery Laboratory, Texas A&M Engineering Experiment Station