Analysis of unsteady flow forces acting on the thermowell in a steam turbine control stage

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Abstract. In the present paper the phenomenon of unsteady flow forces acting on the thermowell for measuring steam temperature in a steam turbine control stage has been presented. The non-stationarity of fluid acting on the thermowell such as: Strouhal frequency, pressure amplitude, pressure peaks, pressure field, velocity field etc. have been studied analytically and numerically. There have been examined two cases of flow with changing mass flow rate, pressure and temperature in the control stage chamber of a turbine high-pressure cylinder. The problem of entry into resonance by thermowell has been described in the ASME standard PTC19.3 TW-2010 with providing detailed guidelines for thermowell designs.

1. Introduction

Generally, from the scientific and engineering point of view, unsteady flow forces acting on a solid structure are a very important issue. Therefore, in the paper, in order to specify the similarity between periodical phenomena and to take into account the case of non-stationary flows forces acting on the thermowell for measuring steam temperature in a steam turbine control stage, a novel CFD+CSD approach has been used.

However, it should be underlined that the presence of a cylindrical element in the steam flow is a reason of occurrence of the vortex shedding phenomenon, where the flowing steam detaches from the cylinder surface creating downstream a vortex structure [1]. The accompanying pressure pulsations generate flow-induced vibrations which may lead to thermowell damage due to high-cycle fatigue [2,3]. The problem is described in the ASME standard PTC19.3 TW-2010 [4] providing detailed guidelines for thermowell calculations.

1.1. Resonance and Strouhal number $\text{Sh}$

In order to specify the similarity between periodical phenomena and non-stationary flows, the Strouhal number $\text{Sh}$ is introduced, which is the similarity number defined as [5]:

$$\text{Sh} = \frac{a}{vt} = \frac{af}{v},$$

(1)

where: $a$ - characteristic dimension, $v$ - steady ambient velocity of the uniform flow, $t$ - period of tested phenomenon, $f$ - frequency of the phenomenon.
It should be added, that the Strouhal number \( Sh \) is used in the engineering technology to analysis of e.g. flow around the vibrating body (aerofoil, rod, thermowell), flow through a blade row mounted in the rotor disc, flow around a rotating propeller or screw propeller, etc.

There is broad literature discussing the problem of the formation of vortices, changes in their structure, the impact of vortex structures on objects on which they form. The research on the vorticity and the formation of structures at flow has been carried out for hundreds years, what is evidenced by sketches of Leonardo da Vinci [6]. Great importance in the development of knowledge of periodical phenomena had works of von Kármán about vortices generation and works of Strouhal about frequency of their formation. Detailed description of these issues can be found in the works of the pioneers [7,8] and of contemporary authors [5,9]. Most of the published articles related to the non-stationary structures in flow, concern the experimental work and analysis [10-13], but the importance of numerical works is growing [14-16].

The fundamental work in this regard is the analysis of the formation and development of vortex structures in a flow around obstacles. A significant number of studies in this field concerns laminar-turbulent transition in the area, where unrecognized and not well recognized phenomena in single theory occurs. There are known works, in which the obstacle has the form of sphere [13], rotating sphere [12], trapeze [10] or cube [11].

Especially important is the problem of vortices forcing vibrations, and vice versa - the vibration of devices affecting the vortex structure. In the last decade, a number of works was published [15] concerning interaction of fluid and structures arising in it, which resulted in creation of a new norm for ASME PTC 19.3 TW-2010. However, the main aim of this paper is analysis of interaction between steam and thermowell. Steam temperature measurements in thermal turbines are conducted using thermocouples placed in thermowell which is inserted into the steam flow. The thermowell protects the temperature sensor against detrimental action of steam, simultaneously introducing to the measurement system a thermal inertia resulting from the solid design of thermowell. The presence of a cylindrical element in the steam flow is a reason of occurrence of the vortex shedding phenomenon, where the flowing steam detaches from the cylinder surface creating downstream a vortex structure [17]. The accompanying pressure pulsations generate flow-induced vibrations which may lead to thermowell damage due to high-cycle fatigue [1,18-21].

The problem is well described in the ASME standard PTC19.3 TW-2010 [4] providing detailed guidelines for thermowell calculations. However, thermal and strength analyses of elements working in high-pressure were presented in [18-21]. Other studies [22,23] have modelled and predicted the stress rupture of alloys and the residual life of the most critical components while considering the effects of creep, thermo-mechanical fatigue, corrosion, and oxidation.

In the framework of root cause analysis, steam flow, dynamic and mechanical integrity calculations were performed aiming at determining steam excitations acting on the thermowell, its natural frequencies and safety margins. For this purpose, CFD+CSD (Computational Fluid-Solid Dynamic) approach has been adopted.

2. Model description
In analyses case, the similarity criterion is the frequency of the force which can be directly compared with the free vibration frequency of the device, so that it can be determined whether the resonance occurs. An important tool in the analysis of this phenomenon is the combination of CFD + CSD, often called FSI - Fluid Solid Interaction or Fluid Structure Interaction.

2.1. CFD
Traditionally, the five balance equations (that consist of one mass balance equation, three momentum balance equations and one energy balance equation) and two evolution equation for parameters defining turbulence (equation for turbulent kinetic energy evolution \( \kappa \) and equation for turbulence dissipation evolution \( \epsilon \) ) have been described in [6,22]. To maximize the use of the easily implementable matrix calculus, the starting point for CFD computation is formulation of universal set
of mass, momentum and energy balance equations for the fluid, supplemented with equations for
turbulence evolution $k$-$\varepsilon$ in the form of:

$$\begin{align*}
\frac{\partial}{\partial t} & \begin{bmatrix} \rho \\ \rho v \\ \rho e \\ \rho k \\ \rho \varepsilon \end{bmatrix} + \text{div} \begin{bmatrix} \rho v \\ \rho v \otimes v \\ \rho e + p \mathbf{I} \\ \rho \varepsilon \\ \rho \varepsilon \mathbf{I} \end{bmatrix} = \text{div} \begin{bmatrix} \rho S_{\varepsilon} \\ \rho S_{\varepsilon} \\ \rho S_{\varepsilon} \\ \rho S_{\varepsilon} \\ \rho S_{\varepsilon} \end{bmatrix} \\
& + \begin{bmatrix} 0 \\ \sigma \varepsilon + q \\ 0 \\ 0 \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \rho b \end{bmatrix} + \begin{bmatrix} \rho \mathbf{b} \end{bmatrix} + \begin{bmatrix} \mathbf{J}_k \\ \mathbf{J}_\varepsilon \end{bmatrix}
\end{align*}$$

(2)

where: $\rho = \rho(x,t)$ - fluid density, generally dependent on time $t$ and location $x$, $v = v_i e_i$ - fluid velocity, $p$ - thermodynamic pressure, $\mathbf{I} = \delta_{ij} e_i \otimes e_j$ - unit tensor, $\mathbf{t}^{\text{ext}} = \mathbf{t}^{\text{in}} + \mathbf{t}^{\text{in}}$ - stress flux components, laminar and turbulent respectively, $\mathbf{b}$ - mass force of earth gravity, $e = u + \frac{1}{2} v^2$ - sum of internal and kinetic energy, $q$ - heat flux, $\mathbf{J}_k$, $\mathbf{J}_\varepsilon$ - diffusive flux of $k$ and diffusive flux of $\varepsilon$ with sources $S_k$, $S_\varepsilon$ (various definitions of different authors exist in literature). For each finite volume of the computational grid, seven equations are solved (one for mass, energy, $k$ and $\varepsilon$ transport balance equation and three momentum balance equations) [6].

2.2. CSD

The problem of natural vibration has been discussed in [2,6]. The problem of determination of non-stationary load coming from steam flow, amplitudes of excitation from non-stationary flow and partial load have recently been discussed in [2,6,21].

In analogy to CFD, the appropriate set of CSD governing equations is determined as follows [6]:

$$\begin{align*}
\frac{\partial}{\partial t} & \begin{bmatrix} 1 \\ \rho v \\ \rho e \\ \rho \varepsilon \end{bmatrix} + \text{div} \begin{bmatrix} 0 \\ \rho v \otimes v \\ \rho e + p \mathbf{I} \\ \rho \varepsilon \mathbf{I} \end{bmatrix} = \text{div} \begin{bmatrix} \sigma \\ \sigma v + q \end{bmatrix} + \begin{bmatrix} 0 \\ \rho b \end{bmatrix} + \begin{bmatrix} \rho \mathbf{b} \end{bmatrix} + \begin{bmatrix} \mathbf{J}_k \\ \mathbf{J}_\varepsilon \end{bmatrix}
\end{align*}$$

(3)

where: $\sigma$ - tensor of solid stresses; $\mathbf{e}^{pl}$ - tensor of plastic strain; $\mathbf{a}$ - kinematic hardening; $\mathbf{r}$ - isotropic hardening, $\mathbf{J}_\mathbf{r}$ - diffusive flux of $r$; $S_{pl}, S_a, S_r$ sources of plasticity, kinematic and isotropic hardening typical for the Chaboche model. Our analysis and calculations in solid body are based on the 3D modal and harmonic analysis typical for the CSD [2]. CSD is point-blank analogy of CFD (Computational Fluid Dynamics). Both methods use the same balance equations (mass, momentum and energy). Discretization method for the CSD and CFD is arbitrary (FEM, FVM, etc) but the governing equations are identical. This architecture of solving equations greatly simplifies FSI (Fluid Solid Interaction) [2,6] and thermal-FSI [22] analyses.

3. Analysis

3.1. Geometry and boundary conditions

The thermowell analyzed here operated in the control stage chamber of a turbine high-pressure cylinder (figure 1). Figure 2 presents the geometry model discretized with finite volume mesh which was used to simulate the general steam flow conditions within the control stage. It shows four inlet pipelines, nozzle sectors, control stage rotating wheel and chamber, and first full-arc admission stage.
For appropriate description of the flow nature within the control stage chamber, the numerical model precisely describes dimensions of the control stage (64 blades), nozzle rings whose shape exactly describes the geometry of real stage. In order to better describe the flow conditions and obtain a real picture of the control stage flow, the computational domain was extended to the first not-regulated stage. The whole model was discretized using structured mesh providing the most reliable results with optimum density. The areas of expected big gradients were additionally refined, which resulted in the number of elements reaching 6 million.

Boundary conditions for CFD calculations are presented in table 1. There were considered two cases of work of the control stage at reduced pressure and mass flow rate of steam - night work with 180 MWe and daily work with 380 MWe at nominal parameters. Previously described model was used. However the ideal gas was considered so viscosity effect was omitted.

| Parameter | Unit | Daily work – full load | Night work – partial load |
|-----------|------|------------------------|--------------------------|
| \(\dot{m}\) | kg/s | 101 | 76 |
| \(p\) | bar | 153.7 | 119.6 |
| \(T\) | °C | 535 | 520.7 |
| \(N\) | MWe | 380 | 180 |

3.2. Calibration of the CFD model

In order to confirm the correctness of the results, some resultant points from [2] were marked on the diagram presenting dependence of Reynolds number on Strouhal number (figure 3), taken from the ASME norm [4]. The position of points in figure 3 should be considered as satisfactory because they are within the area defined experimentally. It is worth mentioning that the diagram from refers to the obstacles, immersed in the whole height of the channel with a uniform cross-section, and in the case of analyses thermowell the flow was only partially disturbed, and additionally the thermowell has the shape of a truncated cone.
3.3. Boundary condition of the CSD model

Fig. 4a) shows the geometry of the thermowell after making discretization which was used to calculate the frequency of free vibrations of the housing of steam temperature sensor. To mount a structure, 6 degrees of freedom was taken away from the area marked by bold lines. As the material of the thermowell was taken steel ST12. In the Fig. 4a), there are marked places (p) of application the aerodynamic loads, which values were determined on the basis of pressure field on the surface presented on Fig.4 b).

![Figure 4.a) The housing of the thermowell after making discretization and with marked place (p) of application of the boundary conditions. b) surface used to determine pressure field acting on solid.](image)

4. Results and discussion

Numerical simulations were carried out with boundary conditions describing turbine operation with nominal and part load. Figure 5a) presents spatial fields of static pressure computed for one full rotation of the rotor at nominal load. The results were presented for selected time step mainly $t=0.0036$ s. The time step was equal to $\Delta t=0.000312$ [s] which corresponds to 10 steps per each inter-blade channel for 64 rotor blades. Local pressure variations do not exhibit significant fluctuations. At this phase, after examining distribution of the flow parameters and its character, it can be stated that the flow at the control stage chamber is stable and does not exhibit symptoms of big oscillations.

The subsequent figures show the local variations of flow parameters. Figure 5b),c) presents the static pressure fields calculated at initial phase of rotor rotation at the control stage chamber: Fig.5b) – after the control stage at the inlet to control stage chamber and; Fig.5c) – outlet of the control stage chamber before the first not-regulated stage. The results from Figure 5b) and 5c) correspond to the turbine operating with partial load. This operation mode differs from the nominal load by opening of two nozzle boxes only. The remaining two are fully closed by the stop valves and steam is flowing only through the boxes with a smaller number of nozzles (compare with Figure 5a - nominal load). Consequently, higher non-uniformity of pressure Fig.5b) and velocity distribution Fig.6 is observed for the steam leaving the rotor blades. Figure 5c) presents static pressure variations at the exit plane of
the control stage chamber. The exit is located close to the inlet to stationary blades of the first non-regulated stage. Clear difference between open and closed nozzle sectors is seen from the pressure distributions. This distinction is visible even though the steams flow through the entire control stage chamber.

**Figure 5.** The static pressure fields at initial phase of rotor rotation: a) spatial distributions; b) cross section after the control stage; c) cross section before the first not-regulated stage.

**Figure 6.** Velocity distribution at plane cutting the thermowell at subsequent time steps: a) 0.0036 [s]; b) 0.007 [s].

Figure 6 presents the velocity fields at selected time steps. It can be noticed that the flow forms the main core of higher velocity beginning at the exit from the control stage and ending at the inlet to the stator blades of the first non-regulated stage. At the bottom part of the control stage chamber, a stagnation region persists due to high divergence of the chamber and reversed flow can be expected there. This flow can generate vortices and continuous circulation of steam inside the chamber.

**Figure 7.** Plots of total pressure distribution around thermowell: left–180MW; right–380 MW.

Figure 7 presents distributions of the total pressure around the thermowell at 180MW (partial load) and 380 MW (full load). The pressure distribution changes both its values and shapes with changing load, which means that both pressure amplitude and resultant force direction vary with turbine load. It can be thus concluded that direction of the pressure force acting on the thermowell depends on the operation regime (figure 7). In the case of analyses thermowell the flow has different character of
turbulence and pressure deviation in comparison to the steam main flow direction which is behaviour as axial.

These loads expressed as difference of total pressure for the thermowell section inserted into the flow are presented in figure 8. It is seen that the aerodynamic loads are one-sided which means that steam pushes the thermowell in line with the steam flow direction but with different time-dependent force. The key role in variable loading of the thermowell is played by the total pressure. The difference in steam pressure varies between 0.5 - 4.5 bar for part load and 0 - 0.9 bar for full load.

![Figure 8](image)

**Figure 8.** Plot of total pressure difference variation between upstream and downstream side of the thermowell section inserted into the flow (part load - solid line, full load – dotted line).

4.1. CSD Analysis

Figure 9 presents a variation of natural frequencies with temperature and comparison of the excitation frequencies with forbidden ranges of natural frequencies. As the main excitation the trailing edge wakes from 64 rotating blades were assumed and based on this the excitation frequency $n_{zw} = 3200$ Hz was computed. The excitations due to non-uniform steam supply to nozzle boxes were not analyzed here, as the frequencies equal to 50 Hz and 100 Hz for part and full load conditions are very small comparing with the thermowell natural frequencies.

![Figure 9](image)

**Figure 9.** Variation of natural frequencies with temperature.

According to the ASME standard PTC 19.3 TW-2010, the required safety margin for natural frequency is ±20%. As it is seen from figure 9 this condition is satisfied with a considerable margin for the analyzed natural frequencies of the thermowell. Taking into account the above results it can be concluded that the thermowell natural frequencies are within a safe zone as compared with the excitation frequencies.

5. Conclusions

Summarizing the calculation results it can be said that, operation with partial load is characterized by larger amplitudes of pressure acting on the thermowell. The total pressure amplitude for, part load reaches 4.5 bar, while at full load is equal maximum 0.9 bar. Pressure distribution curve around the thermowell changes both its shape and value bringing about a variation of not only the pressure amplitude but also the load direction. Two major excitations from the steam flow in the control stage chamber can be distinguished, namely due to non-uniform steam supply to the nozzle boxes and due to
aerodynamic wakes of rotor blades trailing edges. The excitation frequency due to non-uniform steam supply to the nozzle boxes is 100 Hz for 180 MW load and 50 Hz for 380 MW load. The excitation frequency due to aerodynamic wakes is 3200 Hz in the entire load range. At steady state operations the criteria of tuning the natural frequencies against steam forces basic frequencies are met and the thermowell is free of resonance.

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