REVERSE WATER HAMMER IN KAPLAN TURBINES

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Summary

A serious Kaplan turbine accident due to the reverse water hammer is briefly described here. The later investigations were conducted on a scaled model and on identical unit.

Two different methods for the evaluation of the possibility of the reverse water hammer occurrence have been developed. The first method is to compute the pressure distribution on the turbine head cover by the use of model tests data (which has to be provided). The second one is somewhat simpler and is based on the axial thrust data. Both methods are used to analyze the situation which has led to the accident mentioned above.

The results of both computations explain what has happen in the power plant. Moreover, the described protection measures against reverse water hammer have been analyzed by the same methods and offer good results.

Résumé

Dans ce travail on a démontré brièvement une avarie de turbine Kaplan par suite d'un coup de bélier réversible et ensuite l'on a fait des recherches sur un modèle réduit et sur une machine naturelle.

Deux méthodes différentes de constatation de possibilité du phénomène de coup de bélier réversible y sont décrites. La première est basée sur la détermination de répartition de pressions sur le couvercle de la turbine à l'aide des résultats d'examens de modèles (qui doivent être faits). La deuxième est quelque peu plus simple et se base sur les données concernant la force hydraulique axiale.

Les deux méthodes sont utilisées pour l'analyse des phénomènes qui ont amené à l'avarie. Les résultats de calculs ont montrés ce qui s'est passé dans l'usine hydro-électrique. L'efficacité des mesures suggérées pour la préservation contre le coup de bélier réversible est analysée avec succès par tous les deux méthodes.
Nomenclature

\( A_R \) - runner sectional area
\( A_R, e_0 \) - guide vane opening
\( c \) - flow velocity
\( c_w \) - mean axial velocity of water in the runner
\( c_{lim} \) - limited velocity
\( D \) - runner diameter
\( d_g \) - diameter of runner hub
\( F_A \) - axial hydraulic thrust
\( g \) - weight of the rotor
\( g \) - acceleration due to gravity
\( H \) - head
\( H_{at} \) - atmospheric pressure
\( H_{lim} \) - limited pressure
\( H_{steam} \) - saturated vapor pressure
\( H_{s} \) - suction head
\( \Delta H_{s} \) - pressure rise behind the runner due to water hammer
\( H_{cover} \) - mean pressure on turbine cover
\( H_{turbine} \) - local pressure on the turbine cover
\( \Delta H_t \) - mean pressure drop through the runner
\( H_{turbine} \) - pressure on turbine head cover
\( H_{static} \) - static pressure
\( H_{sp} \) - pressure in spiral casing
\( K_{Q} \) - constant in flow coefficient equation
\( K_{Q} \) - constant in head coefficient equation
\( K_{Q} \) - constant in axial thrust coefficient equation
\( \Omega \) - speed of revolutions
\( P \) - power
\( Q \) - discharge
\( u \) - peripheral velocity of the runner
\( \eta_d = d_g/D \) - dimensionless diameter of runner hub
\( u \) - runner blade inclination
\( \phi \) - flow coefficient
\( \eta_{m} \) - head coefficient
\( \eta_{s} = \frac{h_{s} + H_{s}}{h_{c}} \) - pressure coefficient
\( \eta_{s} = \frac{h_{s} + H_{s}}{h_{c}} \) - pressure coefficient
\( \kappa \) - relative pressure
\( \kappa = \frac{K_{Q} F_{a}}{H} \) - axial thrust coefficient
\( \rho \) - density
\( \eta \) - draft tube efficiency
Introduction

A few years ago, a serious Kaplan turbine accident took place directly due to a reverse water hammer. The Mechanical Engineering Faculty, University of Belgrade, and the ZEP - Association of Power Industry, Belgrade, collaborate on a study of reverse water hammer in hydraulic turbines, the first stage of which has been completed. This paper considers, in brief, some results of the investigations, including two different methods of pressure calculation for examining the possibility of the reverse water hammer phenomenon.

1. DESCRIPTION OF AN ACCIDENT AND FIELD TESTS

The accident happened during the night, while the generating unit was operating with a low power output, and due to the governor failure, the guide vanes opened suddenly and the turbine began operating at a high speed in an unsteady working region with noticeable cavitation.

The unit was closed by pressing the turbine emergency shutoff button, while the generator was disconnected from the electric-power-supply network. The unit overspeed control device went into operation. When the machine came to a stop, a banging noise was heard from the turbine and the water was leaking out through the turbine head cover. A serious accident took place. One of the runner blades was broken in the root, while some others were damaged. Damages were also found in the lower and upper guide-vane rings, runner head, runner pit inner cone, inner head cover, etc.

The generating unit accident and the foregoing conditions are described in more detail in the paper [1].

After the accident a series of tests were made with another identical unit. An interesting strip chart, recorded during the turbine's emergency shutoff, is shown in Fig.1.1. It is important to notice that the pressure on the turbine head cover (h₁ and h₂) reaches an absolute vacuum at the end of the servomotor stroke. It means that an air-vapour cavity is formed above the runner, resulting in the water column separation. Pressure on the turbine head cover begins to rise only twenty seconds after the beginning of the transient process.

2. MODEL TESTS

The model tests [2], [5] were recorded in a number of stationary regimes presenting the pressure field in front of and behind the runner, in the draft tube and on the turbine head cover.
Fig. 2.1b gives the pressure distribution in front of the runner, and 2.2 shows the pressure distribution on the turbine head cover. The runner inclination $\theta = 23^\circ$ for stationary regimes shown in Fig. 2.1a. The relative $p$
sure on the ordinate is,

\[ \frac{h_m + H_s}{H} \]

where: \( h_m \) - static pressure; \( H_s \) - suction head; and \( H \) - effective turbine head.

The broken lines represent extrapolation according to literature [3]. It should be noted that the pressure distribution changes appreciably when turbine operating conditions are changed.

For further analysis, the following pressure coefficient is introduced:

\[ \psi = 2g \frac{h_m + H_s}{\rho \nu^2} \]

where: \( \nu \) - peripheral velocity of the runner; and \( g \) - gravity acceleration.

The model test results are used to determine the dependence of pressure coefficient \( \psi \) at the turbine head cover (locations A1, A2, A3 and A4, schematically given in Fig. 2.1) upon the discharge coefficient \( \phi = K/Q/Dn \) and head coefficient \( \psi = KH/(3Dn)^2 \) where: \( Q \) - discharge; \( H \) - head; \( D \) - largest diameter of runner; and \( n \) - speed of revolutions.

The pressure coefficient \( \psi \) on the turbine head cover near the guide vanes \( \psi_{00} \) is determined by extrapolating the measured values and on the basis of data taken from literature [3]. Figures 2.3 and 2.4 show this relationship between local pressure coefficients at the turbine head cover, locations 00 and 01, \( \psi_{00} \) and \( \psi_{01} \) and coefficients for discharge \( \phi \) and head \( \psi \) at the runner blade inclination 8=23°.

**Fig. 2.3 - The relationship of the local pressure coefficient \( \psi_{00} \) and the discharge and head coefficients (\( \phi \) and \( \psi \)) (location 00)**

**Fig. 2.4 - The relationship of the local pressure coefficient \( \psi_{01} \) and the discharge and head coefficients (\( \phi \) and \( \psi \)) (location 01)**
On the basis of these diagrams it is possible to determine the pressure distribution on the turbine head cover in a transient operating conditions.

If no data about the pressure distribution on the turbine head cover, such as the diagrams $Y_1(\phi, Y)$, is available, the possibility of the reverse water hammer occurrence can be evaluated by the use of the averaged pressure values under the turbine head cover $H_k$. In such a case it is convenient to introduce the axial thrust coefficient

$$
\sigma = \frac{F_a}{K_d \cdot \rho D^3 \cdot n^2}
$$

where: $F_a$ - axial force; $K_d$ - constant; and $\rho$ - density of water.

The dependence of the axial thrust coefficient $\sigma$ on the coefficients for discharge $\phi$ and head $Y$, determined on the basis of model tests, it is possible to find the mean pressure value under the turbine head cover in different states through which the turbine passes under transient operating conditions. Fig. 2.5 shows the relationship between the axial thrust coefficient and the coefficients for discharge $\phi$ and head $Y$ at the runner blade inclination $\beta=23^\circ$.

![Fig.2.5 - Hydraulic thrust characteristic for runner blade inclination $\beta=23^\circ$](image)

3. CALCULATION OF TRANSIENT PROCESSES

Firstly, one of the known methods [7] is used to calculate the water hammer, which gives the changes in pressures at inlet and outlet of the turbine, and the changes in discharge, head and speed of the unit. These results are shown in Figg. 3.1, 3.2 and 3.3 for different closure modes of guide vanes.

Now, for chosen moments, an analysis is made of the hazard of reverse water hammer in one of the two methods explained below.
3.1 Determination of the pressure distribution on the turbine head cover (first method)

For the chosen intervals of time, and from the water hammer calculation results, coefficients for discharge $q$ and head $H$ are determined, and using such diagrams as those in Fig.2.3 and 2.4, pressure coefficients $v_s$ are determined for the given points at the turbine head cover.

The absolute pressure on the turbine head cover can now be found by using the expression:

$$H_k = H_{at} + \Delta H_s - H_s + \frac{u_e^2}{2g} v_s$$

where: $H_{at}$ - atmospheric pressure; $\Delta H_s$ - pressure rise behind runner due to water hammer; $H_s$ - suction head.

Figures 3.1, 3.2 and 3.3 show changes in absolute pressures $H_{k00}, H_{k01}, H_{k02}$ and $H_{k04}$ at the turbine head cover (the locations are schematically represented in Fig.2.1a) for different closure modes of guide vanes. On this basis, Fig.3.1a, 3.2a and 3.3a show the pressure distribution on the turbine head cover at given moments, as measured from the beginning of the nonstationary phenomenon.

Fig.3.1 - Calculation of prolonged closure time ($T_c = 3.5$ s)

Fig.3.1a - Calculated pressure distribution on the turbine cover ($T_c = 3.5$ s)

Calculation of unit quick closure ($T_c = 3.5$ s): $A$ - guide-vane opening; $H$ - head; $Q$ - discharge; $n$ - speed; $H_k$ - mean pressure on turbine cover; $H_{kxx}$ - local pressure on turbine cover; $F_a$ - hydraulic thrust; $c_m$ - mean meridional velocity of water in the runner.
Fig. 3.2 - Calculation of the prolonged closure time (Tc = 9.5 s)

Fig. 3.2a - Calculated pressure distribution on the turbine cover (Tc = 9.5 s)

Fig. 3.3 - Calculation of the prolonged closure time (Tc = 20.0 s)

Fig. 3.3a - Calculated pressure distribution on the turbine cover (Tc = 20.0 s)
The condition for the reverse water hammer occurrence is that the runner is dewatered when the water starts to flow back from the draft tube. This will happen if in the space above the runner there is pressure equal to that of the saturated vapour, provided that the water flows out at meridional speed \( u_m \) higher than an allowable value \( c_{lim} \). Since the calculation is only an approximation which disregards pressure pulsations \([8]\), a certain pressure reserve \( H_{lim} \) should be introduced, which means that the risk of reverse water hammer is present if a pressure lower than the allowable value \( H_{lim} \) occurs in a sufficiently large zone above the runner, and the mean value of the meridional component of water outflow speed from the runner is higher than the allowable speed \( c_{lim} \).

The allowable speed value should be determined by another analysis. In a first approximation, we can consider that \( c_{lim} = 0.5 \text{ m/s} \); consequently, the condition under which there will be no reverse water hammer is: \( H + H_{lim} > H_{vp} \) \( (3.2) \)

In case the condition is not satisfied, then:

\[
\frac{c}{u} = \frac{D_{g}^{2}}{2} (1-r_{g}^{2}) \quad (3.3)
\]

where: \( D_{g} \) - largest diameter of runner; \( r_{g} = d_{g}/D \) - dimensionless diameter of hub; \( d_{g} \) - hub diameter.

Naturally, during the entire process the axial hydraulic force must not raise the unit rotor, and therefore we must have:

\[
g > 1 \quad (3.4)
\]

where: \( g \) - weight of generating-unit rotating parts.

It should be noted that the pressure distribution measured on the model, as well as Eq. (1.1), apply only when there is no absolute vacuum in the zone above the runner. If that happens, the pressure distribution must be corrected. In this, however, there is no data available, and the values \( c_{lim} \) and \( H_{lim} \) are still subject to checking.

### 3.2 Determination of the mean value of the pressure in the space above the runner (Second method)

If no data about the pressure distribution on the turbine head cover is available, the calculation can be carried out by applying an integrated procedure \([1], [4]\), utilizing the axial hydraulic thrust data obtained by model tests. For the known moments of time \( t \), on the basis of quantities \( \theta_{a}, \Delta \theta_{a}, \rho \), the coefficient of axial thrust \( \eta_{a} \), or the axial force \( F_{a} \), can be obtained.

The mean pressure drop \( \Delta H_{k} \) through the runner can be determined as:

\[
\Delta H_{k} = \frac{F_{a} - \frac{K_{a} \alpha_{a} - u_{a}}{D_{a} \rho}}{A - (1-r_{g}^{2})} \quad (3.5)
\]

and the mean value of absolute pressure in front of the runner will be:

\[
H_{k} = H_{at} - H_{s} + \Delta H_{s} - \frac{F_{a} \rho}{2g} + \Delta H_{k} \quad (3.6)
\]

where: \( \eta_{a} \) - draft tube efficiency.

Now, the conditions that prevent a reverse water hammer are:

\[
H_{k} > (H_{vp} + H_{lim}) = \eta_{a} + 3 \text{ mWh} \quad (3.7)
\]
and in case the condition is not satisfied, then
\[ c_m < c_{lim} = 0.5 \text{ m/s} \] (at \( H < H_{yp} + H_{lim} \)) \hspace{1cm} (3.8)

4. DISCUSSION ON THE RESULTS

To analyze the possibility of reverse water hammer it is necessary to calculate, according to the methods described here, the pressure distribution on the turbine head cover, or the mean pressure on the head cover, for various closure modes of guide-vanes as shown in Fig.3.1 through 3.3.

The time-dependent change of local pressures on the turbine head cover \( H_{ko0}, H_{ko1}, H_{ko3} \) and \( H_{ko4} \) is given by broken lines, and that of the mean pressure \( H_k \) by the full line. Fig.3.1 shows that the mean pressure drops below the allowable value at the moment 3.2s from the beginning of nonstationary operating conditions, the mean meridional velocity at the runner outlet being higher than the allowable velocity, \( c_m = 1.9 \text{ m/s} > 0.5 \text{ m/s} \), which means that there is a danger of the reverse water hammer phenomenon. The same conclusion is reached if the pressure distribution on the cover shown in Fig.3.1a is considered. As early as the moment \( t = 2.75 \text{s} \) the vacuum spreads over about 25% of the cross-sectional area, and at the moment \( t = 3.25 \text{s} \) the vacuum occupies the entire flow space.

During the guide-vane closure shown in Fig.3.2 there is also a hazard of reverse water hammer, because \( H_k \) drops below the allowable value at the moment \( t = 8 \text{s} \), the speed being \( c_m = 1.6 \text{ m/s} > 0.5 \text{ m/s} \), and the vacuum spreads over a large zone in the space above the runner between the 7th and 9th seconds from the beginning of the nonstationary phenomenon. It should be, however, noted that at the moment of water column separation the meridional velocity \( c_m \) is lower than in the previous case, which means that the reverse flow speed is lower and, consequently, the shock will be weaker.

For the guide-vane closure mode given in Fig.3.3 one can say that there is no danger of reverse water hammer, since the mean pressure on the turbine head cover \( H_k \) never drops below the allowable value \( H_{lim} \), that is, as shown in Fig.3.3a, the vacuum in the space above the runner never spreads over a large zone.

5. CONCLUSION

The hydraulic water hammer hazard may be estimated by either of the methods described here. The method based on calculating the pressure distribution on the turbine head cover by means of pressure coefficients is somewhat more reliable but requires additional measurements during the model testing. The method based on calculating the mean pressure on the turbine head cover by means of the axial force is simpler and requires no additional measurement since the axial force is measured during the model tests, but we cannot yet estimate, with sufficient certainty, the values of the allowable pressure \( H_{lim} \) and the allowable mean meridional velocities \( c_m \).

The differences between these two methods, or rather, between the values of allowable pressures and velocities, are the subject of present investigations at the Mechanical Engineering Faculty, University of Belgrade.

References


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