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JACEK SZYMCZYK[1] and KRZYSZTOF KARAŚKIEWICZ

The modeling of centrifugal pump transients

Institute of Heat Engineering, Faculty of Power and Aeronautical Engineering, Warsaw University of Technology, pl. Politechniki 1, 00-661 Warsaw, Poland

Abstract

Centrifugal pumps are part of larger hydraulic systems. These hydraulic machines can work not only in the pumping mode but in a number of other modes whose job description is possible using complete characteristics. Some of these modes, such as the turbine mode and the reversed turbine mode, may have practical significance. Knowledge of the transition from one steady state to another state allows to estimate instantaneous values such as, e.g., flow current and torque, which may be even a few times higher than nominal. The principal issues in these transients are amplitudes of parameters of these pumps and the time constants associated with the characteristics of the system.

Keywords: Rotodynamic pumps; Complete pump characteristics; Transients

1 Problem description

Complete pump characteristics in their third form (Suter characteristics) include all pump working modes (areas). There are 8 physically possible modes. These characteristics relate flow flux, Q, head, H, torque, M and rotational speed, n, using two curves. On the basis of these curves parameters in each mode can be calculated. Possible pump working modes are:

[1]Corresponding Author. E-mail address: jacek.szymczyk@itc.pw.edu.pl

(a) mode A ($+Q$, $+H$, $+M$, $+n$)[1] – working as typical pump,

(b) mode C ($-Q$, $+H$, $+M$, $-n$) – pump working as turbine,

(c) mode E ($+Q$, $+H$, $+M$, $-n$) – 'reversed' pump – working as pump with reversed impeller direction,

(d) mode G ($+Q$, $-H$, $-M$, $+n$) – 'reversed turbine' – pump working as turbine with reversed than turbine direction of the flow,

(e) modes B ($-Q$, $+H$, $+M$, $+n$) , D ($-Q$, $+H$, $+M$, $-n$), F ($+Q$, $-H$, $-M$, $-n$), H ($+Q$, $-H$, $+M$, $+n$) – power dissipation.

Typical pump mode is mode A. Untypical operating conditions, apart from energy dissipation modes, include:

(a) mode C – turbine mode, for example as turbine operation due to power failure at preserved pump head (no valve or defective valve), the pump as a turbine in a hydroelectric power station or as energy recovery instead of a throttle valve;

(b) mode E – reversed pump, for example as a result of incorrect connection of the asynchronous motor to the power supply network and replacing two phases;

(c) mode G – reverse flow turbine, for example in series with other pumps due to lack of supply, or if low pressure is needed (enabled only when the required pressure is higher).

In case of pump start-up in typical conditions, the pump always remains in mode A and only changes its rotational speed, ω, from $\omega = 0$ to $\omega = \omega_N$[2] but flow changes from $Q = 0$ to $Q = Q_N > 0$. Sometimes, where the discharge tank is significantly higher than the suction tank and the one-direction valve is missing or broken, during the first time there is negative flow through the pump (inversed in comparison to standard pump work.) Then, if the pump impeller is fixed by any blockage, the pump starts in mode B, and during the whole start-up it changes from B to A. In a case when the impeller is not blocked and can rotate, the start point is in zone C and then changes from C to B to A.

[1] Sign '+' means positive value and '–' means negative value.
[2] Index N means nominal values.

2 Mathematical model

Section 1 describes different possible pump modes. Using basic pump characteristics, i.e., $H(Q)$ and $P(Q)$ (where P denotes pump power consumption), it is relatively easy to clearly describe pump work in transients but only when the pump is in one working mode. The situation is more complex, when pump changes its working modes. Then pump basic characteristics are already not sufficient and work representation using $H(Q)$ characteristics for these other modes becomes unclear. The solution of this problem are three representations of the pump characteristics, where the most clear in this case seems to be their third form, where all working zones of the pumps are shown by the solid line on a graph. A sample characteristic (complete pump characteristic) in the third form is shown in Fig. 1, but its shape differs depending on the pump specific speed [1–5].

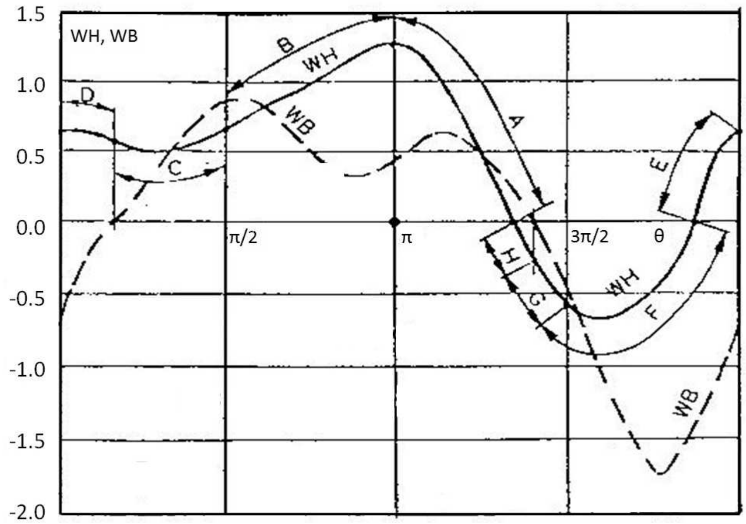

Figure 1: Areas of pump operation using $WH(\theta)$ and $WB(\theta)$ coordinates, where WH (specific pump head) and WB (specific pump torque) are defined in Eq. (1).

Figure 1 shows a chart of the $WH(\theta)$ and $WB(\theta)$ coordinates, where values are defined as follows:

$$\begin{cases} WH(\theta) = \frac{h}{\alpha^2+q^2} \\ WB(\theta) = \frac{m}{\alpha^2+q^2} \\ \theta = arctg\,\frac{q}{\alpha} \end{cases} , \tag{1}$$

where:

$$\theta = \pi + arctg\frac{q}{\alpha}, \quad h = \frac{H}{H_{BEP}}, \quad q = \frac{Q}{Q_{BEP}}, \quad m = \frac{M}{M_{BEP}}, \quad \alpha = \frac{\omega}{\omega_N}.^{3} \tag{2}$$

Pump start-up is characterized by increasing impeller rotational velocity. During gradual speed changes without going to the other working modes and with small mass of water in the piping system, when water velocity inside the pipes changes rapidly, there following formulas are correct [6–8]:

$$\begin{cases} \frac{Q}{Q_N} = \frac{n}{n_N} \\ \frac{M}{M_N} = \left(\frac{n}{n_N}\right)^2 \end{cases} . \tag{3}$$

If speed change is rapid or mass of water inside the piping system is large (high inertia), then the change of water flow inside in the system will not be equal to the value calculated from Eq. (3) and it will be lower (having regard to the sign – in the case of reverse flow during start-up). In case of pump start-up with initial negative flow $Q < 0$ but with blocked impeller the whole starting process can be divided into two specific parts:

(a) work with flow $Q < 0$, where pump rotational velocity gradually increases and water gradually stops until $Q = 0$;

(b) flow at $Q > 0$, where rotational speed and flux continue to increase until they reach their nominal values.

When water inertia is high and/or pump rotational speed increases fast enough the rotational speed may reach the nominal constant value before the flow reaches positive values.

The equation describing parameters (Q, H) during pump startup (in the transient state) looks as follows:

$$H(Q,\, t) = H_z + aQ|Q| + \frac{l}{gA}\frac{dQ}{dt} \tag{4}$$

and motor torque can be calculated as

$$M_s(Q,\, \omega) = I\frac{d\omega}{dt} + M_t + M_p, \tag{5}$$

[3] Index 'BEP' means optimal parameters where $\eta = \eta_{MAX}\ and\, n = n_N$

where:

H_z	–	hydrostatic head,
a	–	constant dependent on the the characteristics of the pump system;
g	–	gravitational acceleration,
l	–	length of the piping system,
A	–	cross-sectional area,
I	–	inertia of rotating parts,
M_t	–	sum of the static friction in the seals and the pump and motor bearings torques,
M_p	–	torque passed on to the liquid,
t	–	time,
ω	–	rotational speed.

3 Experiment results and calculations

Calculations are based on experimental results for a double-suction centrifugal pump with the specific speed equal to $n_q = 36.5$. Its parameters and the parameters of the pumping system, also shown in Fig. 2, are:

- nominal flow Q_N=3.194 m^3/s,
- nominal head H_N=30 m,
- nominal rotation speed n_N=370 min^{-1},
- nominal motor power P_m =1.1 MW,
- moment of inertia of the rotating parts I=1050 kg m^2,
- characteristic of pumping system $H_{sys} = H_z + 1.96Q|Q|$
- hydrostatic head H_z=10 m,
- total length of the piping system l=305 m,
- inner diameter of the pipe D=1 m.

The water was pumped from the lower reservoir to the upper one at the height of 10 m with the possibility of reverse flow. During the experimental flow, rotational speed, torque and motor power was measured. In this case in the first moment $t = 0$ all valves were opened to let water flow down and the pump impeller was blocked so its rotational initial speed $n = 0$. So after starting the pump in these first seconds it was working in mode B.

There are no characteristics for pumps with $n_q = 36.5$, so the closest values in literature are equal $n_q = 33$, 35, and 55. If need arises to obtain characteristics for different values interpolation should be carried out, but in this case the characteristics for $n_q = 35$ can be used with quite good accuracy.

Figure 3 illustrates the process of starting the pump in the conditions de-

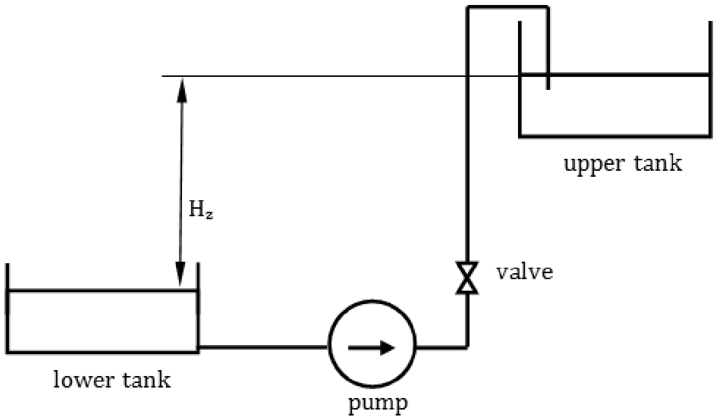

Figure 2: Scheme of testing stand.

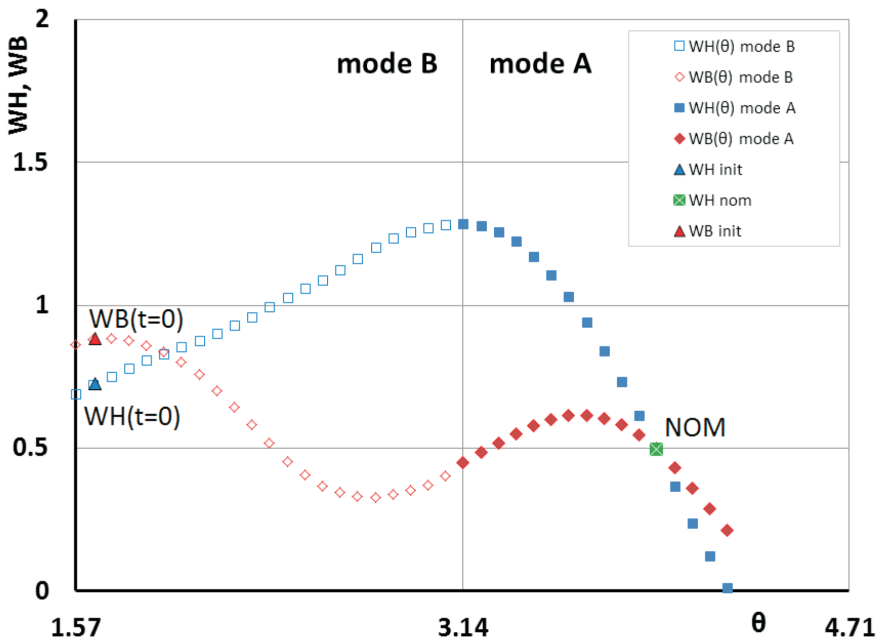

Figure 3: Starting the pump on the $WH(\theta)$ and $WB(\theta)$ coordinate system with initial backflow and initial rotational speed $n(t = 0) = 0$.

scribed above. The initial points are: for dimensionless head – $WH(t = 0)$, and dimensionless torque – $WB(t = 0)$. During the starting process working points move along their specific line (WH or WB) toward point 'NOM'. If the pump is well chosen for the pipeline and the final rotational velocity is nominal, the ending values of WH and WB will be equal to 0.5 and $\theta = 2.356$, because then $q = 1$ and $\alpha = 1$ and $h = 1$.

If the pump had no blocked impeller so it could rotate at the beginning, the starting point would be in mode C, where the pump works as a turbine. Even if the initial (turbine) flow would be greater than $-|Q_N|$ the starting point would not exceed $\theta = 1.57$ toward less values, because the tangent function asymptote is at $\theta = \pi/2$.

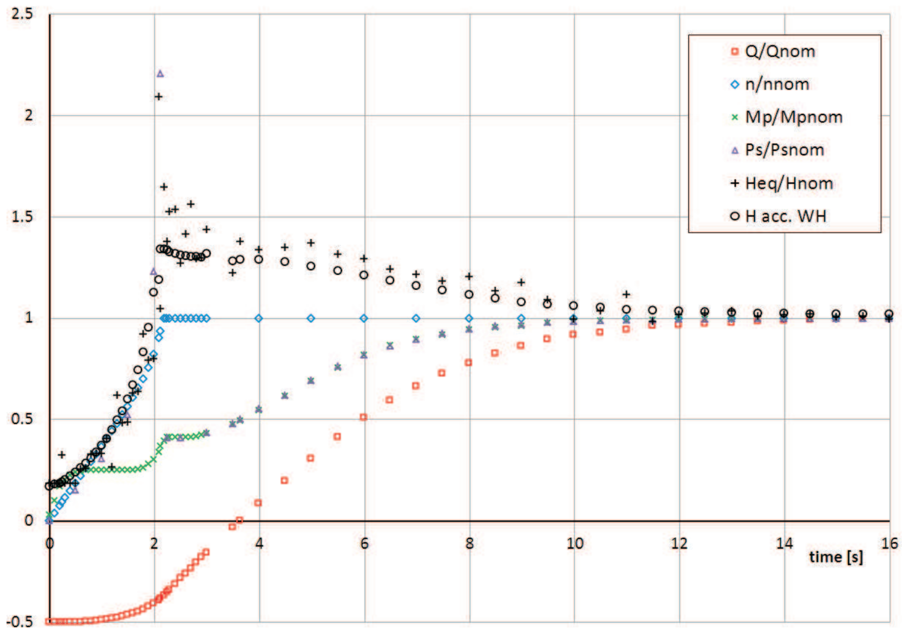

Figure 4: Main parameters during pump startup. Q – flow, n – rotational speed, M_p – pump torque, P_s – motor power, H_{eq} – head calculated using (4) formula, H acc. WH – head calculated according to literature characteristics at $n_q = 35$, 'nom' denotes nominal parameters.

Figure 4 shows the results of measurements and calculations. As it can be seen, the rotational speed n stabilizes just after about $t = 2.2$ s, but flow Q, torque M_p and power P_s are stabilized after about 14 seconds. This is caused by large amount of water and the resulting inertia of the water system.

The head of the hump was calculated from Eq. (4) and it is also shown in the Fig. 4. Its changes are not smooth, because of the accuracy of reading values of Q. At around $t = 2.2$ s, where rotational speed stabilizes and its gradient is the largest, there is local maximum of power P_s (measured) and the head h. From the other hand values o head were calculated using WH published in literature. These points are lower than calculated from Eq. (4) and have no sharp peaks. It is probably because all changes of Q are also smooth.

In comparison to [9], where pump propeller startup is described, the startup time here is shorter by about 1 s for n and about 3 s for the remaining parameters. This is due to the 4 times higher inertia of water system in [9].

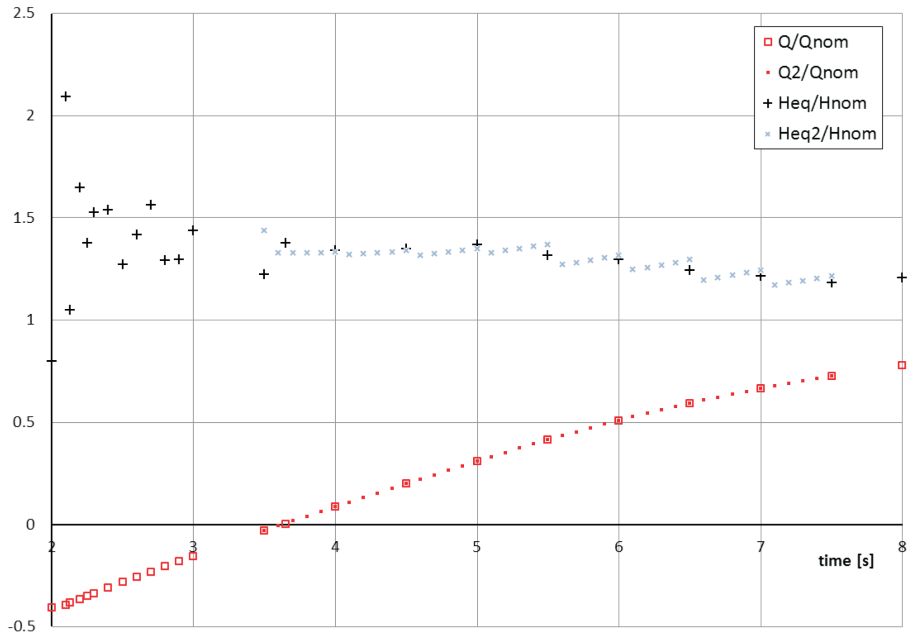

Figure 5: The effect of numerical compaction of the measured values.

The influence of flow Q accuracy is noticeable. Figure 5 shows values q and h calculated from Eq. (4) using pure numerical compaction of time (increase 0.1 instead of 0.5 s) and linear compaction for flow between two known points (Q_2/Q_{nom} points in the picture). Values of flow Q are measured on the laboratory and read with time increase 0.5 s but head H_{eq} is calculated based on Q values using Eq. (4). However values of Q_2 are calculated (for time $t > 3.5$ s) with time increase 0.1 s using linear approximation between two known values Q – closest lower and greater, and head H_{eq2} is calculated based of flow Q_2 using also Eq. (4). Such operation generates sharp stair-shaped curve of h (H_{eq2}/H_{nom} points in the picture) and then highest inaccuracy in this case is equal about 4.2%. Such operation could be useful if these points are used in interpolation process in longer time, but is inaccurate for calculating directly exact values.

4 Summary

1. Simplified method of pump startup calculation may be used only for systems with closed discharge valve. In this case, where water inertia is high, there is a large discrepancy between the values calculated using flow similarity formulas and the measured ones, but the stabilization time of the flow, head and power is even more than six times longer than the stabilization of the rotational speed.

2. Head calculation for pump in startup using formula (4) generates a large local maximum occurring at the moment of the largest speed gradient. The same phenomenon applies to the motor power. The accuracy of the results calculated using formula (4) would be higher and values would be less dispersed if the calculation time step was lower and the accuracy of flow values was higher.

3. For the calculation of unsteady flow, the third form of characteristic complete can be successfully used. However, in the case of pumps with the not mentioned in the literature specific speed interpolation should be made based on known characteristics and accuracy of the interpolation can have a significant impact on the quality of the obtained results.

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