Case study: Overloaded pump motor

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SYNOPSIS

The design of the cooling water system of the Seraya II petrochemical plant in Singapore includes a pump trip and restart scenario. This scenario is essential for the reliability of the plant, because temporary power dips may occur on the site. Due to lay-out changes on the plant during the system design, the steady state pump power exceeds the nominal motor power with 15%, which was considered acceptable by the design team. The pump motors could not be upgraded, because of limitations of the local electric grid. The trip and restart scenario revealed an overload of the pump motors. The overload event led to a redesign of the cooling water system such that the trip and restart scenario led to satisfying results.

This paper stresses the importance of the coupled transient behaviour of both the pump and motor for the evaluation of start-up and restart transients, particularly if a radial pump is to be restarted with an open discharge valve.

keywords: cooling water, pump model, pump motor, pressure transients.

1. INTRODUCTION

Large electric motors usually have relatively small pull-up torques compared to their full load operating torque. In some cases this may lead to start-up problems. This paper will focus on the hydraulic design of a cooling water system of a petrochemical plant.

The specified system performed well during the engineering stage of the hydraulic design, but during the transient analysis of the system shortcomings in the motor power became evident. This paper demonstrates the added value of a sophisticated pump model combined with the transient behaviour in the design of pumping stations.

The mathematical modeling and analyses are carried out with WANDA, version 3. WANDA 3 is used for the hydraulic analysis of steady and unsteady flow conditions in arbitrary configured pipe systems. WANDA 3 has been developed by the Industrial Flow Technology group of WL | Delft Hydraulics.

2. MODEL DESCRIPTION

The cooling water system for the Seraya II petrochemical plant consists of a pumping station, a pipeline system containing 10 unit groups including 49 heat exchangers and an outfall structure. The pumping station is designed for a discharge capacity of 30,750 m^3/h and consists of three operating pumps and one stand-by pump. The schematization of the plant lay-out is shown in Figure 1. The different cooling units are denoted by dark grey squares, while both the pumping station and the outfall facility are shown in a lighter shade of grey.

The pumping station is connected to the production plant area by means of a ∅1500 pipeline of approximately 800 m length. The actual piping inside the production plant consists of different diameters with an average length of 600 m to the outfall facility. The pipe material is GRP. The following pressure criteria were applicable for the pipelines:

- Design pressure is 10 barg.
- Minimum pressure is -0.3 barg for pipes with diameter greater than 750 mm
- Minimum pressure is full vacuum for pipes with diameter less than or equal than 750 mm. These pipes are full vacuum resistant for all sizes.
- The steady pressure downstream of the condensers with vacuum breakers must be above 0 barg, to prevent continuous air inflow via the vacuum breakers.

The flow distribution through the 49 heat exchangers is set during start-up of the system by manually operated valves in the cooling water loops and the Pressure Control Valve (PCV), with a parallel on/off valve, in the outfall structure. By throttling the PCV the pressure in the complete system increases. This pressure increase has to be compensated by the control valves until one control valve (in the most critical cooling water loop) is fully open. The minimum head loss across the fully open control valves is 1 m.

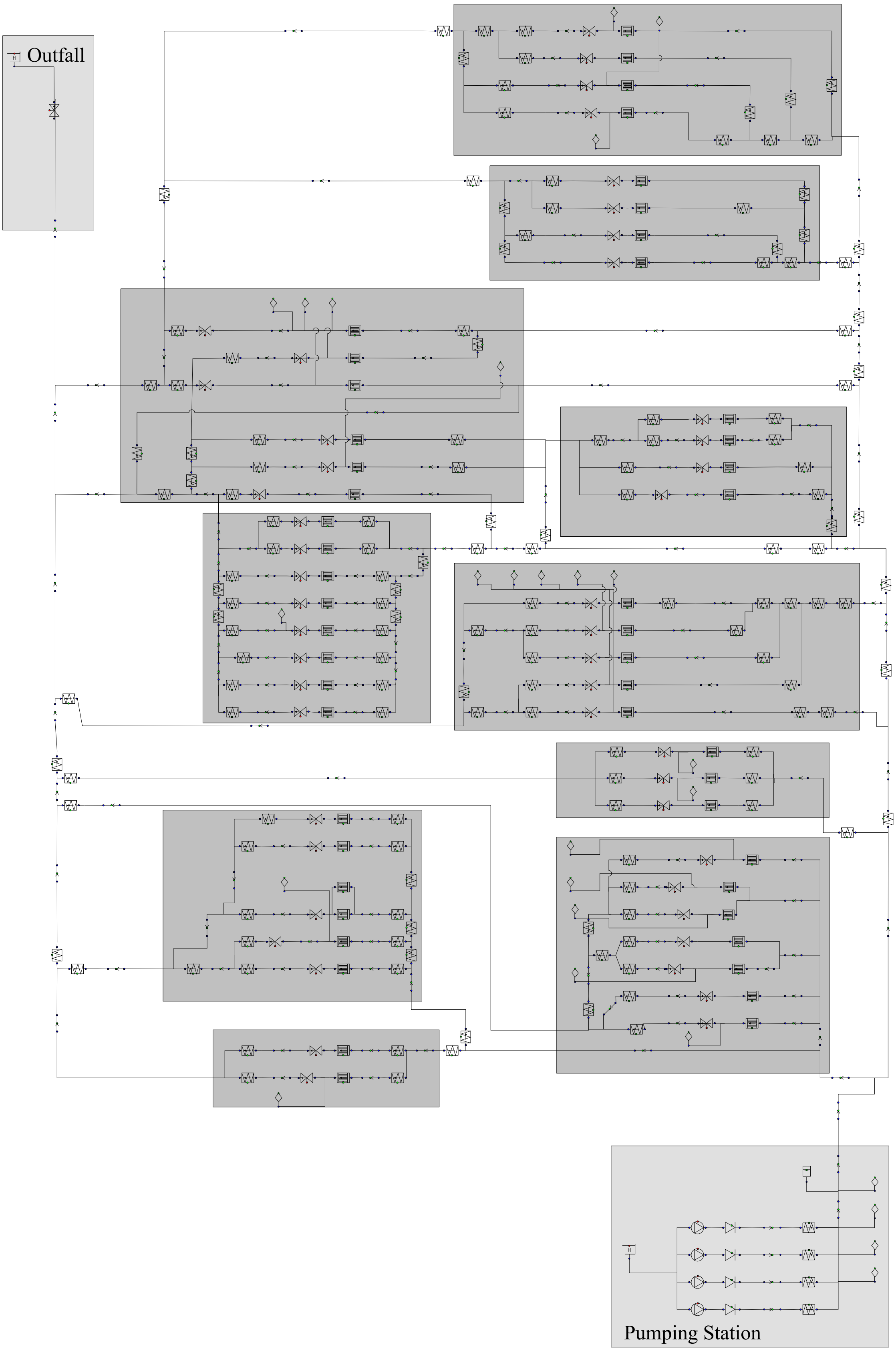

Figure 1: Overview of system schematization

The basic design calculations led to a required pump head of 57 m and motor power of 1900 kW per pump. The nominal motor power was 2100 kW. The speed torque curve of the motor is depicted in Figure 3. Before the hydraulic calculations of the detailed design of the cooling water system were carried out, the plant lay-out was modified and the elevation of several heat exchangers was increased. The results of these modifications on the cooling water system were the following:

- The local losses, due to T-pieces and elbows increased;
- Another loop became the most critical loop;
- The required pump head increased to 73 m, with a required motor power of 2400 kW (see Figure 2). The pump impeller was increased to 1174 mm, while the 2100 kW motor was not upgraded, because of the limited capacity of the local electric power grid.

The 2100 kW motor on the 2400 kW system was considered acceptable for start-up of the cooling water system. The total polar moment of inertia of the new pump motor combination is 367.5 kgm^2. The full load operating torque of 46.4 kNm at the specified maximum discharge is much larger than the starting torque of 13.8 kNm. The remainder of the paper discusses a complete pump trip and restart scenario with the new pump impeller and original motor.

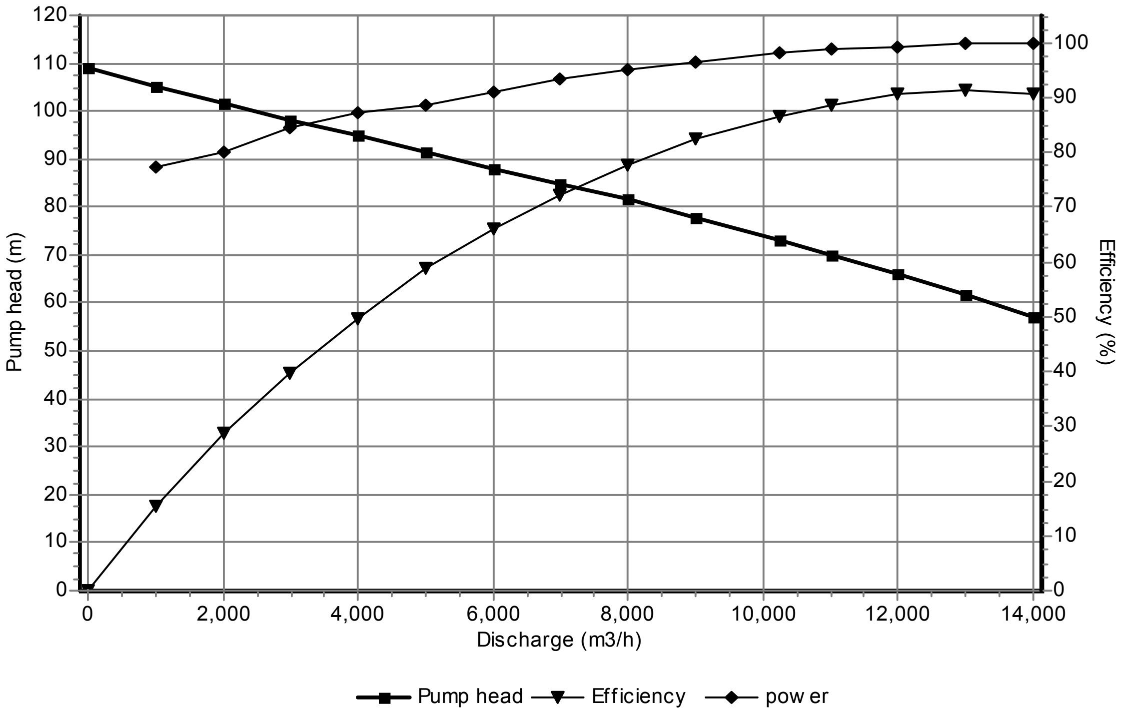

Figure 2: Pump head, efficiency and power curves

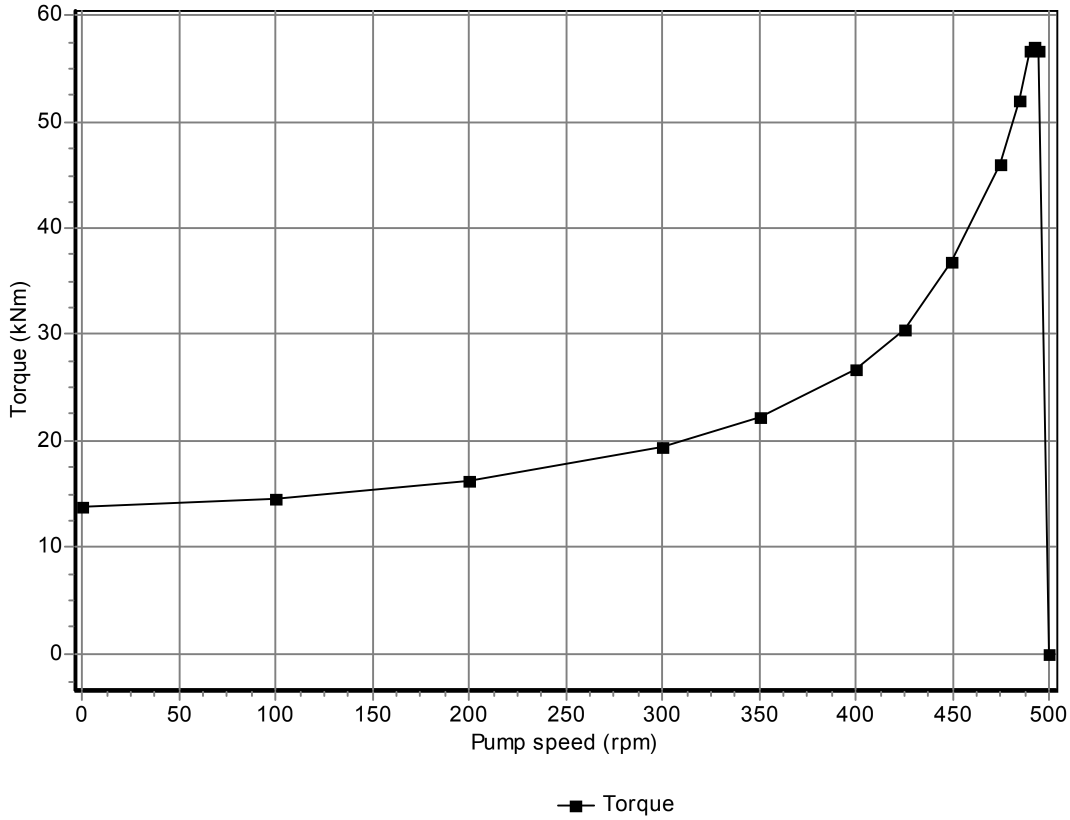

Figure 3: Pump torque - speed Curve

3. CASE DESCRIPTION

The steady state head and the pipeline profile along the most critical condenser are shown in Figure 4. The pressure downstream of the most critical condenser is around 0.1 barg.

The cooling water system should be capable of handling a trip and restart scenario without violating the pressure criteria. The trip and restart scenario is essential for the reliability of the cooling water system and the whole petrochemical plant. Temporary dips in the electricity supply, which may occur on this site, will trigger pump trips. Immediate restart of the pumps prevents a complete plant shut down. The control logic tries to restart pumps during 4 seconds following a pump trip. If the first pump can be restarted within 4 seconds, the loss of cooling capacity is acceptable and the cooling water system returns back to its normal operating point, after the restart of a second and third pump.

The pumps trip after 1 second of simulation. If the first pump can not be restarted within 4 s, the pumps are not restarted and a complete trip scenario occurs. This scenario focuses on a pump restart after 4 s, with consecutive restarts of the second and third pump with 5 s intervals. According to the control logic the PCV just upstream of the outfall facility and on/off valve will start closing and open again, however this operation has not been accounted for in this scenario (conservative approach).

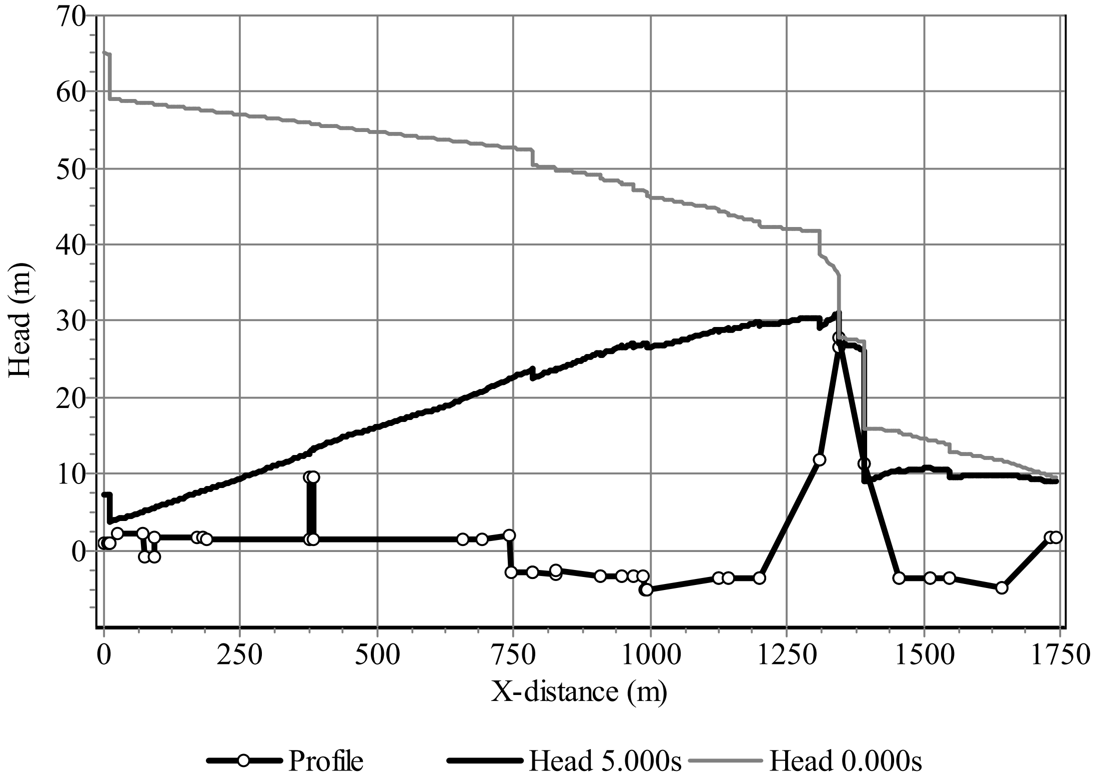

Figure 4: Head and Profile- Route Pumping station to outfall facility at simulation start and in just before restart of the first pump

Such a pump trip-and-restart scenario is dominated by the way the pump speed ramps down and more important how the pump speed ramps up again. Therefore this scenario requires an advanced numerical model of the coupled behaviour of the pump impeller and pump motor. Section 4 briefly outlines this model, as it is implemented in WANDA. Section 5 then continues with the scenario results.

4. PUMP MODEL

In the most general case a pump can operate in four QH-quadrants for N>0 and in four QH-quadrants for N<0. In WANDA, a complete pump model for all N (<0, =0 and >0) is available. The running pump is characterized by the following functions in the first quadrant for N>0:

$$H = H(Q) \quad , N = N_r \qquad (1)$$

$$E = E(Q) \quad , N = N_r \qquad (2)$$

in which:

H	=	pump head	[m]
Q	=	pump discharge	[m^3/s]
E	=	pump efficiency	[-]

N	=	pump speed	[rpm]
N_r	=	rated pump speed at which the functions H(Q) and E(Q) are specified	[rpm]

The head and efficiency curves ((1) and (2)) are normally specified by the pump manufacturer in the first quadrant only (H, Q > 0). To describe a tripping or starting pump the torque function must be known as well. The torque function at the rated speed is derived from the head and efficiency curve according to:

$$T = \frac{\rho_f \cdot g \cdot Q \cdot H(Q)}{E(Q) \cdot \frac{2 \cdot \pi \cdot N_r}{60}} \tag{3}$$

in which:

ρ_f	= fluid density	[kg/m^3]
g	= grav. acceleration	[m/s^2]

To derive the functions (1), (2) and (3) for other pump speeds, N_i , than the rated pump speed, N_r , the affinity rules are used:

$$\begin{aligned} \frac{H(N_i)}{H(N_r)} &= \left(\frac{N_i}{N_r}\right)^2 \\ \frac{Q(N_i)}{Q(N_r)} &= \frac{N_i}{N_r} \\ \frac{T(N_i)}{T(N_r)} &= \left(\frac{N_i}{N_r}\right)^2 \end{aligned} \tag{4}$$

Now, the pump head, efficiency and fluid torque can be computed for all positive pump speeds in the first quadrant, where both the pump head and flow are positive. However, the simulation of a pump trip or start requires the pump head and torque to be known in the second and fourth quadrant as well. To get the values outside the table range, the tables are extended to the second and fourth quadrant with so-called extrapolation parabolas, which have been derived from a set of completely measured pump curves at different specific speeds.

In the case of a starting pump, the driving torque, T_d , is determined by the speed-torque relation (in tabular form) of the driving motor-gearbox combination (See Figure 3). When the motor is powered on, the pump speed increases as long as the driving torque exceeds the fluid torque, according to equation (5):

$$T_d - T = I_p \frac{dN}{dt}\frac{2\pi}{60} \quad \Leftrightarrow$$
$$dN = \frac{60}{2\pi}\frac{T_d - T}{I_p} dt \qquad (5)$$

where the total polar moment of inertia, I_p , includes the polar moments of inertia of the impeller, including liquid, and the pump motor.

The speed-torque curve is characterized by a sharp drop (cut-off) around the operating speed. With an increase in load the speed will not drop much. This phenomenon is denoted as elastic behaviour. If the maximum torque just before the drop is sufficiently large the steady state value of the pump speed is well defined. In the steady state Wanda will compute two equilibriums: 1) pump (QH) curve versus system resistance (QH relation) and 2) fluid torque versus motor torque. Both are varying at each iteration step and to reach convergence a dedicated iteration strategy has been employed.

In normal running operation $T_d = T$ and the pump runs at a steady speed.

Due to e.g. power failure, the driving torque suddenly drops to 0 and the decaying pump speed can be calculated from:

$$dN = -\frac{60}{2\pi}\frac{T}{I_p} dt \qquad (6)$$

The fluid torque of a starting or tripping pump does not only depend on the transient pump heads and flows but also on the instantaneous pump speed via the affinity rules. Therefore the numerical computation of the new pump speed is carried out with a sub-time-step to accurately assess the average fluid torque during each time step.

5. NUMERICAL SIMULATION RESULTS

Due to pump trip the pumps will ramp down after 1 second of simulation (See Figure 6). The pumps decelerate slowly due to their high moment of inertia (See equation 6). The air vessel depressurises to a head of 5 m and dampens the pressure transient in the condensers. The check valves located in the pump discharge lines close after 0.8 s due to the presence of the air vessel. The steady state head in the pipeline and the instantaneous head profile after 5 s are shown in Figure 4.

After 5 seconds of simulation the first pump is restarted by the control system. At this moment, the pump speed has dropped to about 100 rpm. The discharge of pump P1 temporarily increases to around 13,000 m^3/h after 10 s, which is just sufficient to prevent negative pressures in the air vessel. Consequently, the second pump restarts against a similarly low back pressure. The speed of the second pump has reduced to about 60 rpm, when it restarts. The discharge of pump P2 temporarily increases to around 12,000 m^3/h after 15 s. Only after restart of the second pump, the air vessel partially refills, illustrated by the negative air vessel discharge in Figure 5. The speed of the third pump drops to only 40 rpm, when it restarts after 15 s. The discharge of the third pump rises to 6000 m^3/h, while the pump speed has not even reached 300 rpm (see Figure 6 and Figure 8). The large pump flow rates at low back pressures impose large hydraulic torques on the pump impellers, because the pumps are temporarily operating at a very bad efficiency on the right-hand side of the (scaled) best efficiency point.

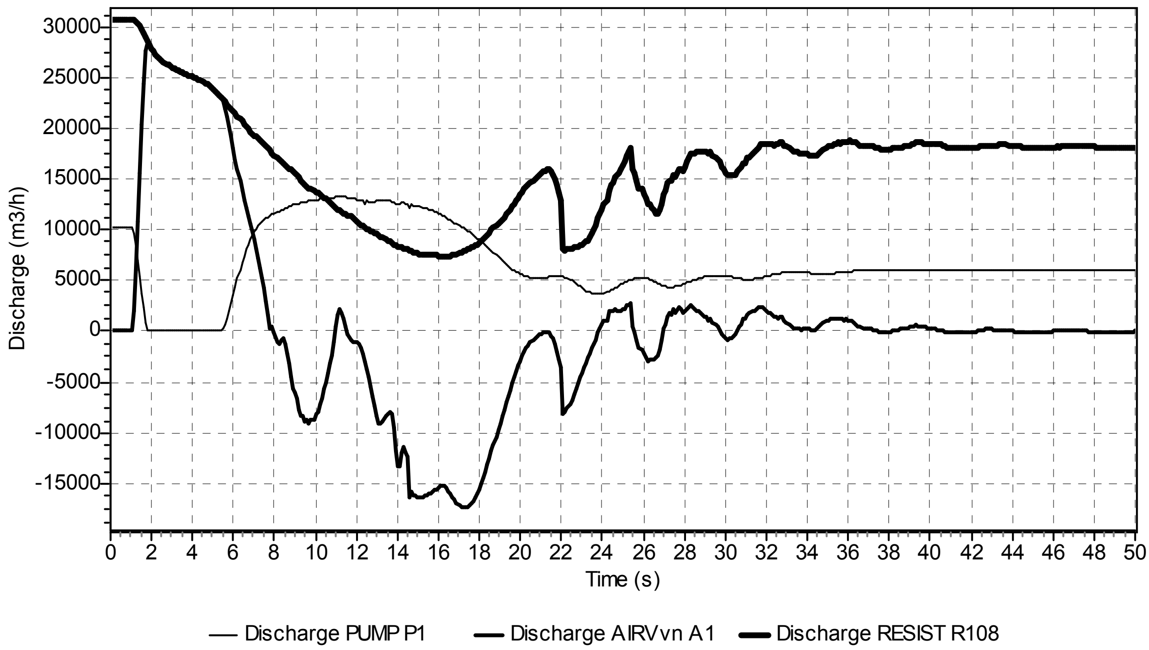

Figure 5: Flow rates from pump P1, from the air vessel (+ into the system) discharge and into the main line (RESIST R108).

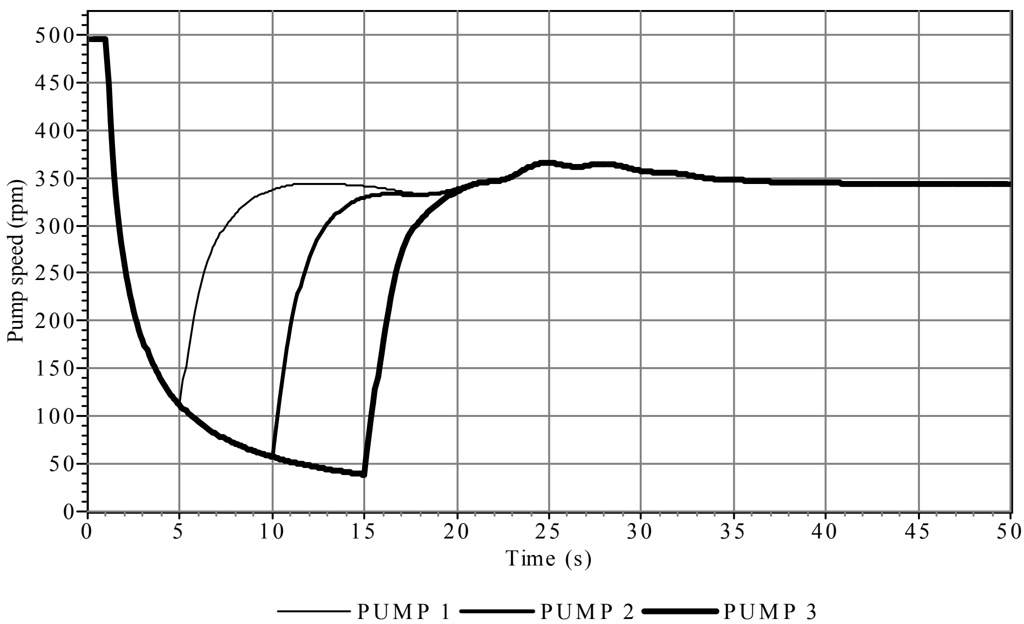

Figure 6: Time series - Pump speeds

The pumps speed up until the motor torque equals the torque exerted by the fluid on the impeller (See Figure 7). Hence the motor torque, specified by the speed-torque curve of the motor, must exceed the fluid torque in order to increase the pump speed. Due to the large discharge, the fluid torque equals the motor torque of pump P1 before the nominal speed has been reached (See Figure 7). The motor and fluid torque are now in equilibrium and there is no excess motor torque available to accelerate the impeller. The pump will not reach its nominal speed and stabilizes around 340 rpm (See Figure 6).

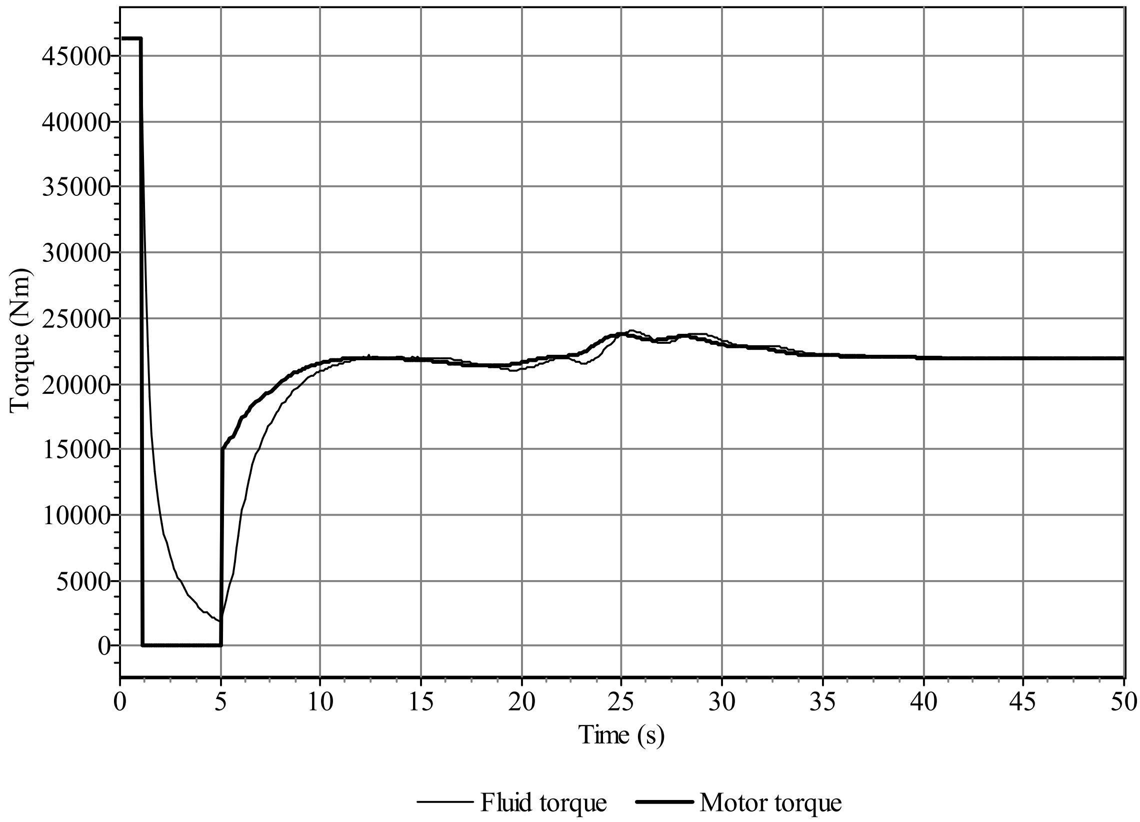

Figure 7: Time series - Fluid torque and motor torque for pump 1

The motor and fluid torque on the impeller of the second pump are similar with the first pump. The lower flow rate through the third pump allows a slightly larger pump speed increase, but the third pump's speed had dropped to 40 rpm only. Consequently the excess torque of the motor is still insufficient to reach the nominal speed of 495 rpm.

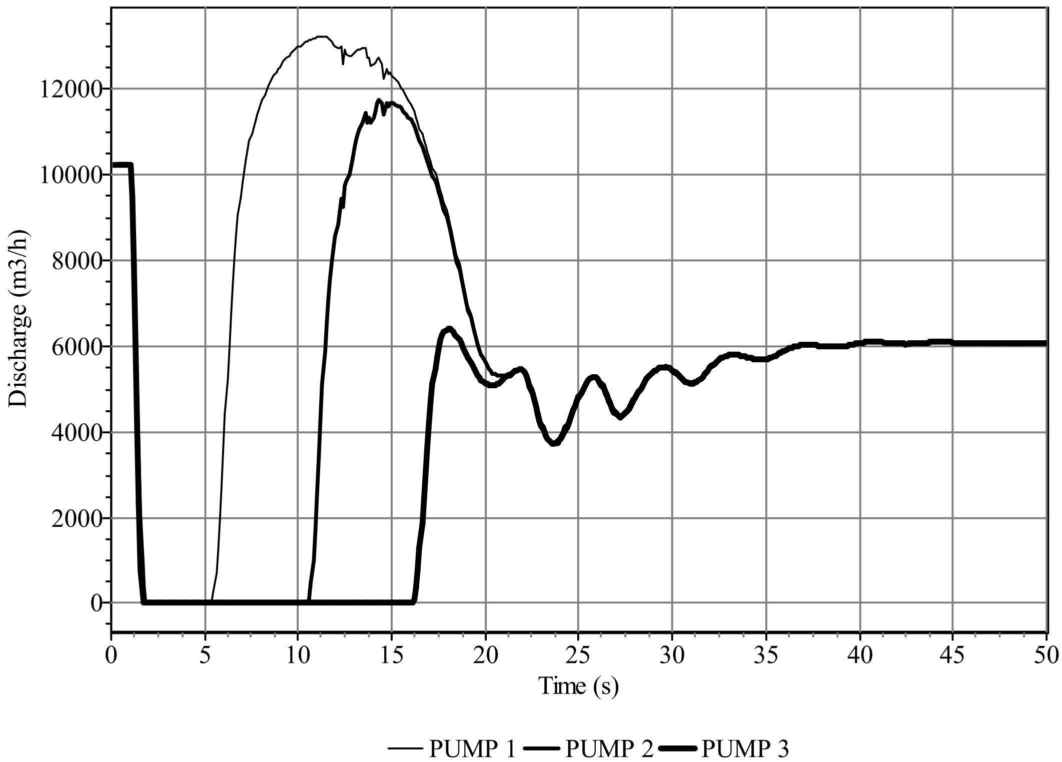

Figure 8: Time series - Pump discharge

To show the behaviour of the pump along the speed-torque curve Figure 9 gives an overview of the fluid and motor torques. The motor torque drops quickly (One time step) from the full load operating torque to zero. The fluid torque drops more slowly and therefore decelerates the impeller. Pump 2 is restarted with an impeller speed of 60 rpm. The motor torque will increase along the specified torque speed curve. The fluid torque will increase depending on the pumps characteristic curve, efficiency and the system response. Figure 10 shows the duty points over time. The pump will establish a duty point along a scaled characteristics point for each pump speed. The lines in Figure 10 are moving around the equilibrium point due to pressure waves in the system.

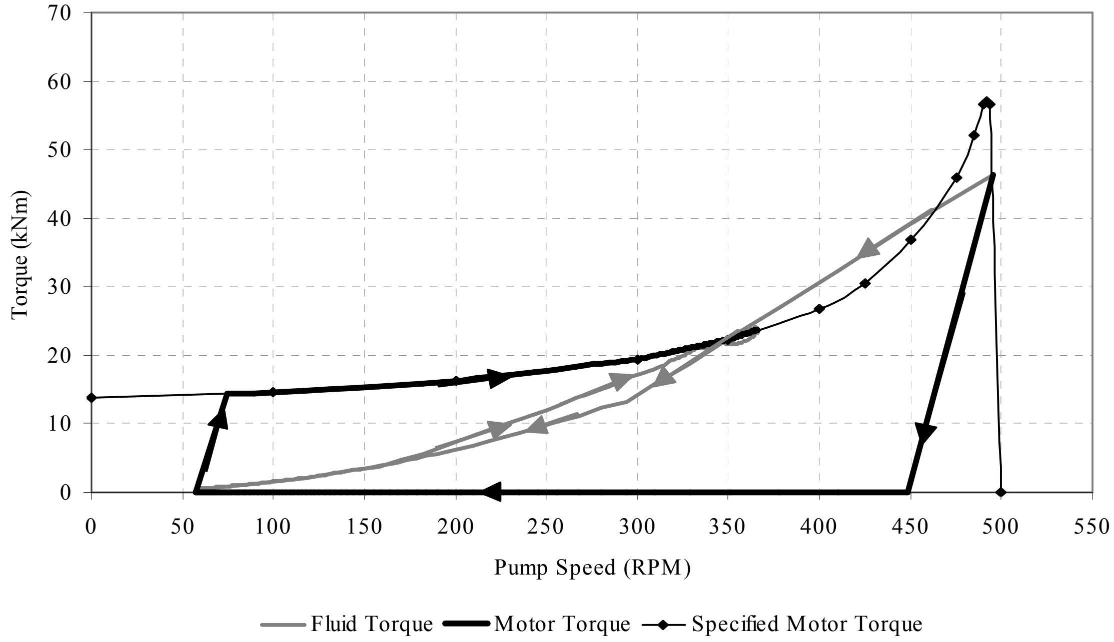

Figure 9: Fluid torque and motor torque for pump 2

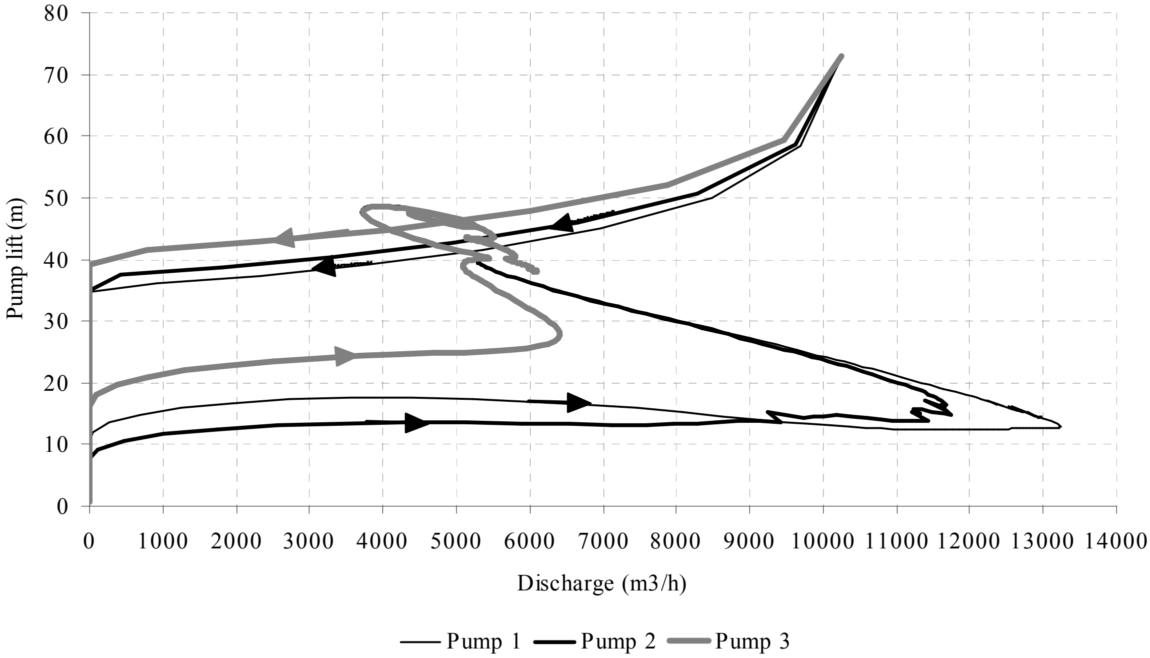

Figure 10: Pump Lift - Discharge

If an individual pump is started against a closed valve (See Figure 11) the motor torque is just sufficient to exceed the fluid torque until the nominal speed is reached. The available acceleration torque is small however and the pump will start in 7.3 s. During the normal start-up scenario the pump will be started against an isolation valve and will be able to reach the nominal speed.

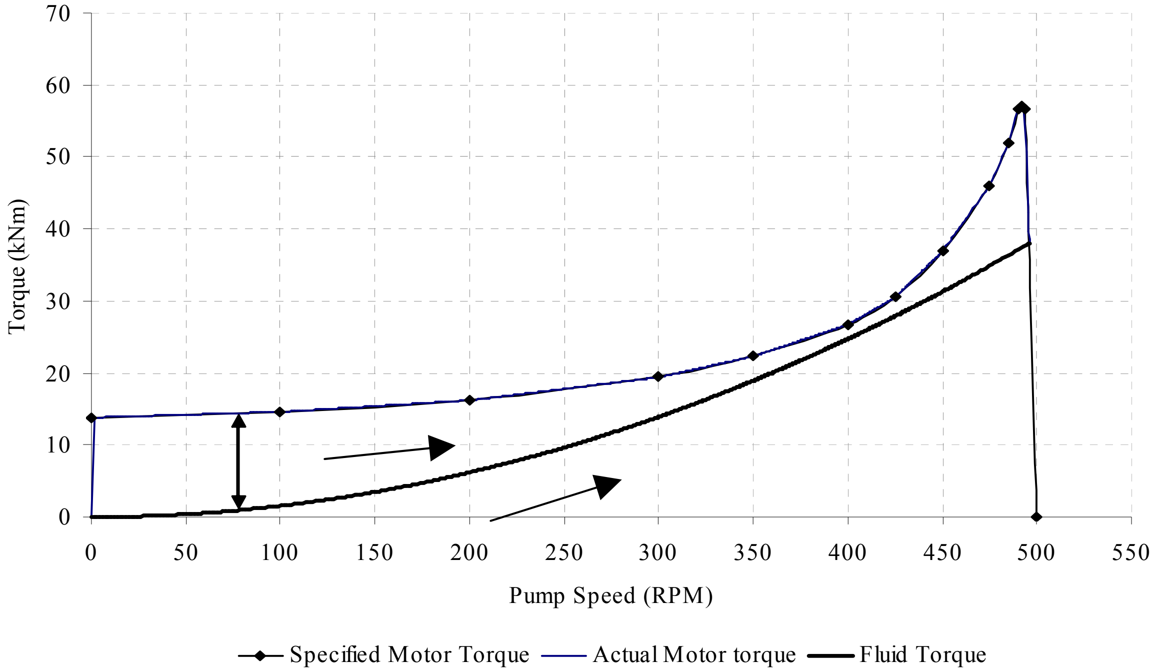

Figure 11: Pump start against closed valve

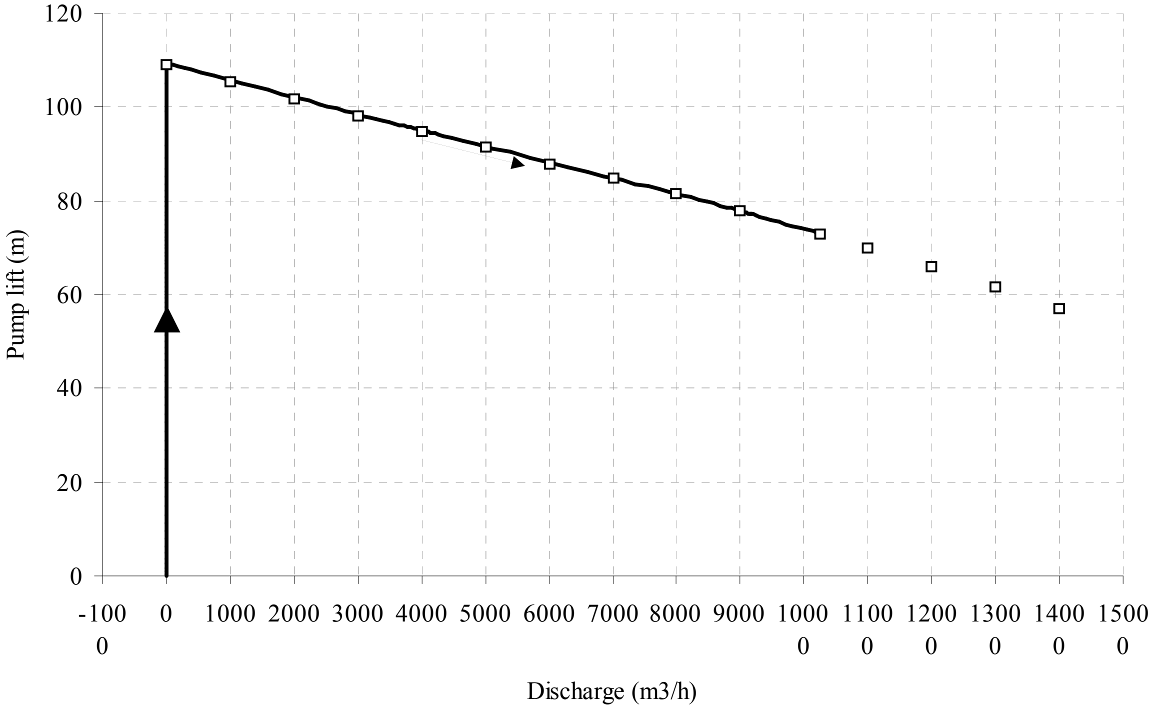

Figure 12: Pump Lift – Discharge

Since the pump motors could not be upgraded, because of limitations of the electricity network, a number of critical pipe diameters were increased in order to reduce the required pump head with at least 15 m.

6. DISCUSSION

The elaborated case of the Seraya II cooling water system discusses the pump trip and restart scenario, which is essential for the reliability of the plant. The restart event is in fact very similar to a pump start with an open discharge valve, which involves relatively large fluid torques for radial pumps. The WANDA computation shows that the pumps cannot return to their nominal speed during this scenario, because of overloaded pump motors; i.e. the transient fluid torque equals the motor torque before the nominal speed has been reached.

This paper demonstrates the added-value of a coupled numerical model of pump impeller and pump motor for the evaluation of start-up and restart transients, particularly if a radial pump is to be restarted with an open discharge valve.

Acknowledgements

The authors wish to thank ABB Lummus Global and Shell for permission to publish this paper.

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