Newsletter

WANDA Technical Newsletter 2005

WANDA 3

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Introduction

With this technical newsletter the WANDA development team of WL|Delft Hydraulics intends to start a tradition of focussing on a specific component or system. Although human knowledge about many hydraulic aspects of pipeline systems is large, it is not complete. Also the translation of physical properties of a specific component or system into a correct numerical relation or model has limitations. Including very detailed physical properties to a component often means that a lot of parameters have to be known of the component.

Measuring of those properties is not always possible, or carried out, or even described in standards, and often expensive. Inter- and extrapolating of data is frequently required. The art is to design components in such a way that simple questions require limited input data, and more complex questions need in addition more detailed input data. In such a way the same program can provide answers to simple questions on pipeline dynamics as well as more detailed answers of complex operating conditions by adding more information and specifications to the model and its components. We assume that the reader has sufficient background of the subject, because of many components extended handbooks are written, which we can not summarize here.

This technical newsletter can be seen as an addition to the manual, because it combines theory and component specification. We suggest that you collect them in your WANDA manual.
This technical newsletter presents the component ‘Pump’.

The Pump Model

The pump(s) of a pipeline system are designed to transport a specified flow rate from location A to B. The total pump head at the operation point of a pump consists of the static head (the difference in height between the suction and discharge reservoir), and the friction head (figure 1). The friction head is not constant, because it depends on the flow velocity, as well as time depending parameters such as a possible increase of the wall roughness.

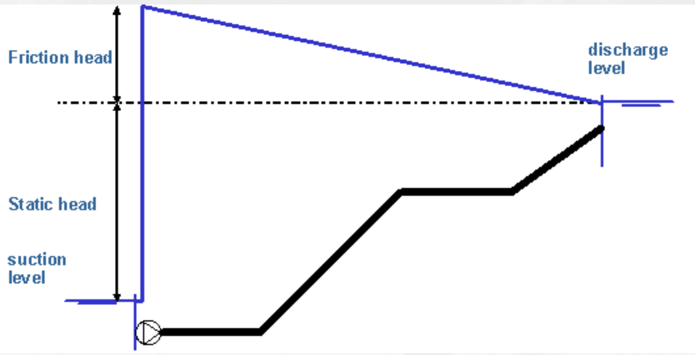


Figure 1: simple pipeline scheme with static and friction head loss

The original mathematical model for pumps in WANDA was designed to simulate the behaviour of starting, running and tripping pumps. The main limitation was that pump speeds had to remain greater than or equal to zero during the time integration. In most simulation models this was sufficient functionality, since the presence of check valves prevented the pumps from running backwards.

In current practice, however, WANDA is used more and more to simulate complicated what-if scenario’s in which pumps can be controlled, tripped, restarted and run with reverse speed in case for instance the check valve is failing or is not present at all. This behaviour can not be described by the standard QHE-tables, which are only valid for positive pump speeds.
To characterise the hydraulic properties of the pump for positive and negative speeds and discharges, the so-called Suter curves are used. In these dimensionless four quadrant curves head (W_H) and Torque (W_B) are plotted against an x-factor (figure 1). This x-factor, which ranges from 0 to 2π, reflects all speed/discharge combinations. The discharge Q is positive for $x > \pi$ and negative for $x < \pi$. The pump speed is positive for $\pi/2 < x < 3\pi/2$ (the standard QHE-tables), and negative for the complementary range.
Depending on the signs of the head difference, the discharge, the torque and the speed, different zones can be distinguished of normal and reverse pumping and turbining, as well as zones of energy dissipation (figure 2).

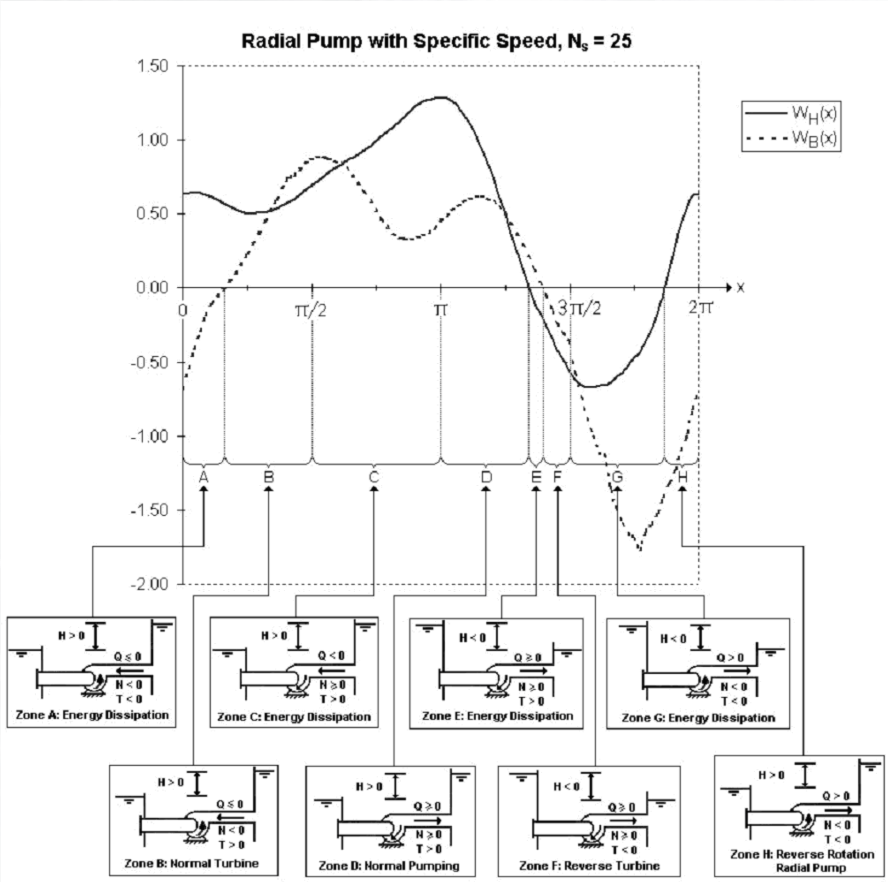


Figure 2: dimensionless Head (W_H) and Torque (W_B) relation for $0<x<2\pi$ (specific speed N_s = 25)

An example of the input and output property sheet for the pump component is shown in figure 3. Either a QHE table or a Suter table must be specified in the characteristic type field.

Suter curves for different specific speed pumps can be selected from the tools+ menu. The specific speeds range from Ns=25 (radial flow, figure 4) up to Ns=261 (axial flow, figure 5).

Since the Suter curves are dimensionless, the discharge Q, head H and torque T in the best efficiency point at the rated pump speed must be specified by the user.

For the steady state calculation the initial condition of the pump must be specified. The five possibilities are: speed, upstream head (H1), downstream head (H2), discharge (Q), or motor frequency. If the initial speed is chosen to be different from the rated speed, the affinity law is applied to modify the QH-curve to be taken into account in the hydraulic computation. If one of H1, H2 or Q is specified the hydraulic computation will compute the other two, after which the pump speed is computed to match the (Q,H2-H1) point with the QH-curve. If the initial motor frequency is specified, the hydraulic computation will also take into account the equilibrium between the motor torque and the hydraulic torque. The actual efficiency and shaft power are computed as a result of the steady state computation as well.

An example of the motor speed-torque curve is shown in figure 6. This is a typical case of a two coil (four pole) electric motor for 50 Hz AC current. The so called synchronous speed is the net frequency divided by the number of coils, in this case 1500 rpm. This is the rotational speed of the magnetic field in the motor. If the rotor would have the same speed, no electric current would flow and the torque will be zero. Torque on the shaft can only exist mutually with some amount of slip. The more the slip the more torque will be exerted on the shaft. A slip percentage of e.g. 4 % of 1500 rpm corresponds to 60 rpm. In the example the torque at the resulting speed of 1440 rpm is 490.7 Nm. If the hydraulic torque increases due to changing discharge and head the pump speed will decrease slightly until the new balance is achieved.

It is possible to change the frequency of the AC current by means of an electronic frequency transformer. This leads to an proportional shift of the synchronous speed

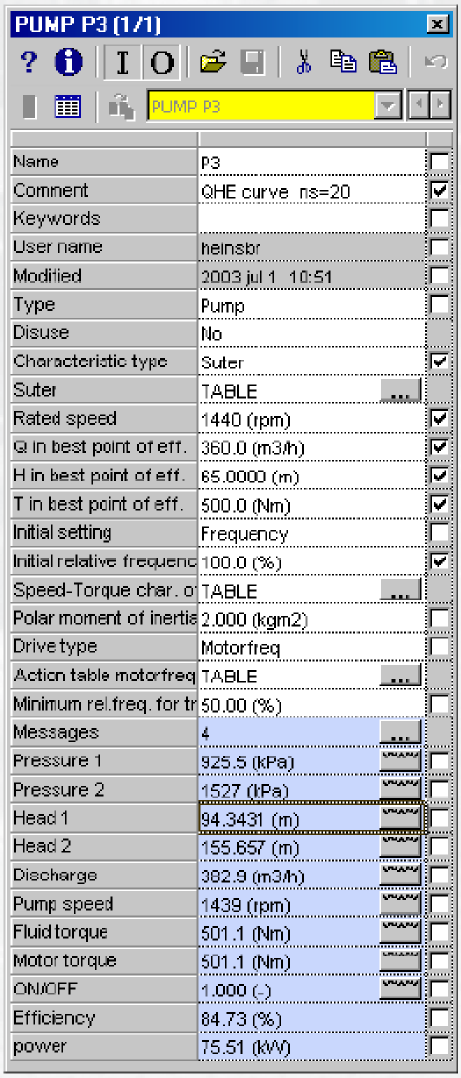
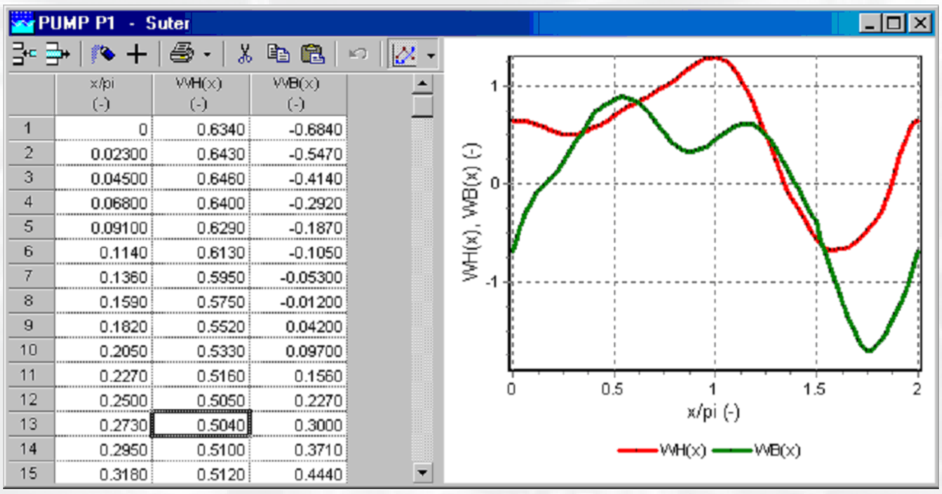
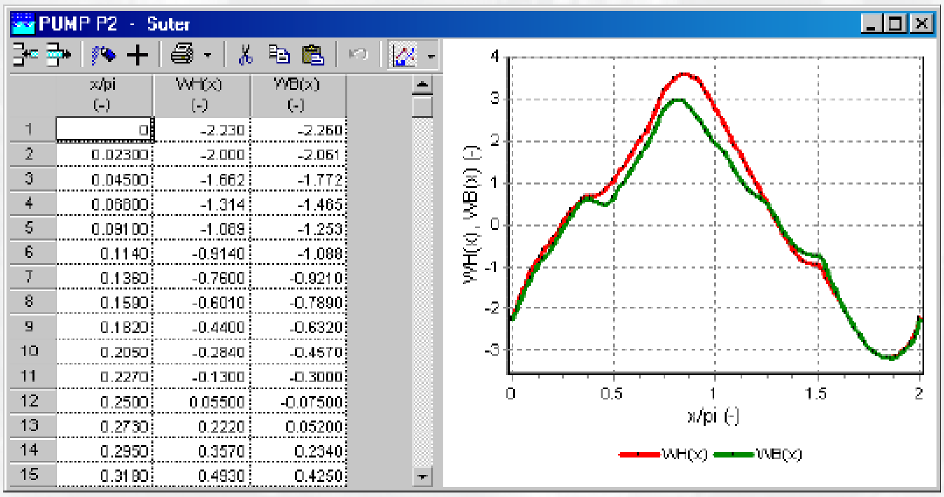


Figure 3: input and output property field



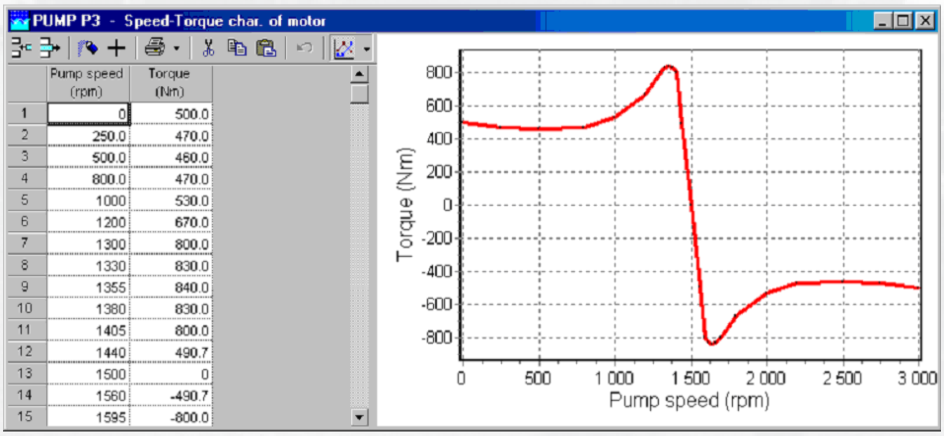
	x/pi (-)	WH(x) (-)	WB(x) (-)
1	0	0.6340	-0.6840
2	0.02300	0.6430	-0.5470
3	0.04500	0.6460	-0.4140
4	0.06800	0.6400	-0.2920
5	0.09100	0.6290	-0.1870
6	0.1140	0.6130	-0.1050
7	0.1360	0.5950	-0.05300
8	0.1590	0.5750	-0.01200
9	0.1820	0.5520	0.04200
10	0.2050	0.5330	0.09700
11	0.2270	0.5160	0.1560
12	0.2500	0.5050	0.2270
13	0.2730	0.5040	0.3000
14	0.2950	0.5100	0.3710
15	0.3180	0.5120	0.4440

Figure 4: Suter input field for N_s = 25 (radial flow pump)



	x/pi (-)	WH(x) (-)	WB(x) (-)
1	0	-2.230	-2.260
2	0.02300	-2.000	-2.061
3	0.04500	-1.662	-1.772
4	0.06800	-1.314	-1.465
5	0.09100	-1.069	-1.253
6	0.1140	-0.9140	-1.088
7	0.1360	-0.7600	-0.9210
8	0.1590	-0.6010	-0.7890
9	0.1820	-0.4400	-0.6320
10	0.2050	-0.2840	-0.4570
11	0.2270	-0.1300	-0.3000
12	0.2500	0.05500	-0.07500
13	0.2730	0.2220	0.05200
14	0.2950	0.3570	0.2340
15	0.3180	0.4930	0.4250

Figure 5: Suter input field for N_s = 261 (axial flow pump)



	Pump speed (rpm)	Torque (Nm)
1	0	500.0
2	250.0	470.0
3	500.0	460.0
4	800.0	470.0
5	1000	530.0
6	1200	670.0
7	1300	800.0
8	1330	830.0
9	1355	840.0
10	1380	830.0
11	1405	800.0
12	1440	490.7
13	1500	0
14	1560	-490.7
15	1595	-800.0

Figure 6: motor speed-torque curve

and the speed-torque curve will shift accordingly. If the frequency is reduced to 60 % (50 Hz -> 30 Hz) the synchronous speed will become 900 rpm. The voltage will be reduced to obtain the same torque as for 100 % frequency. In case the frequency is increased the curve shifts to the right. The voltage should in that case be increased to maintain the same torque, but this is not possible. In that case (running the pump "over synchronously") the magnetic field will weaken and the torque will decrease accordingly. Care must then be taken that the maximum motor torque will not drop below the increasing hydraulic torque, causing an overloaded motor. Figure 7 shows the speed-torque curves for different frequency % (100 % is the nominal frequency).

The field 'action table' of figure 2 can show following realistic pump action (figure 8), starting from the steady state example above.

The following results will be computed (figure 9) when no check valve is installed. The initial gradual decrease in motor frequency is followed by the pump speed (in red) and the discharge (in green). At T=6 s the motor is switched off and the pump will trip. A negative speed of nearly 1500 rpm is reached. It is interesting to see that the discharge reaches a minimum value of -1200 m^3/h at about T=7.2 s, at which time the pump speed has become -300 rpm. The further increase of the negative pump speed causes increasing centrifugal "forces" thereby reducing the negative pump flow from about -1200 m^3/h to about -700 m^3/h. This behaviour is comparable to that of a runaway situation for a radial type turbine (Francis turbines).

The time histories of the fluid torque (red) and motor torque (green) are given in figure 10. The difference between both corresponds to the deceleration and acceleration stages of the scenario. During starting of the pump after 10 s the reverse flow through the accelerating impeller causes relatively large hydraulic torques, thereby slowing down the starting up of the pump. Between 13 and 14 seconds the maximum torque of the motor is encountered and the pump speed increases progressively.

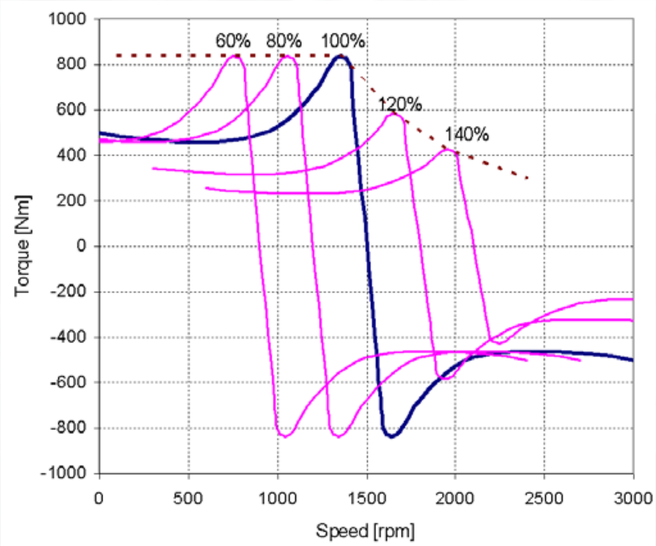


Figure 7: torque-speed relation for different frequency %

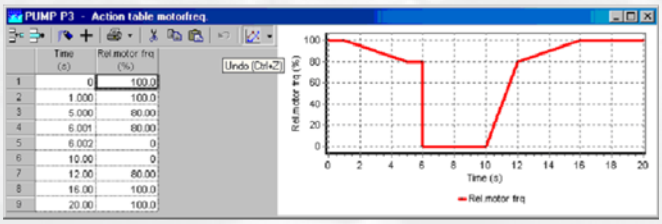


Figure 8: action table time-relative motor frequency relation

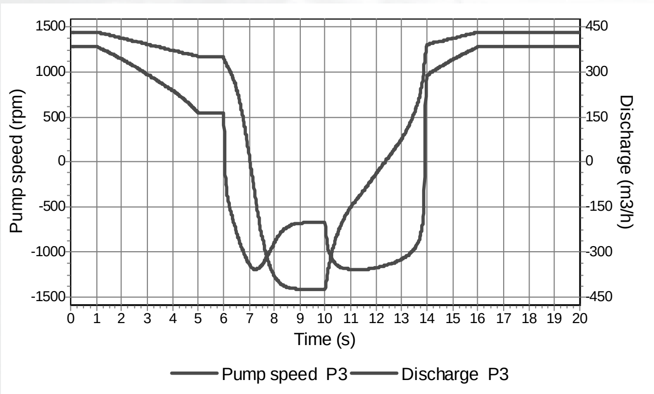


Figure 9: pump speed and discharge results from action table according to

To complete the “picture” of the pump behaviour the time histories of the heads upstream (orange) and downstream (blue) of the pump are given in figure 11. Due to the increasing centrifugal forces between 7 and 9 seconds, the head difference increases.

In case the motor data are not available it is also possible to specify the pump speed as a function (table) of the time. The program will than still compute the motor torque necessary to obtain the specified speed and speed variations.

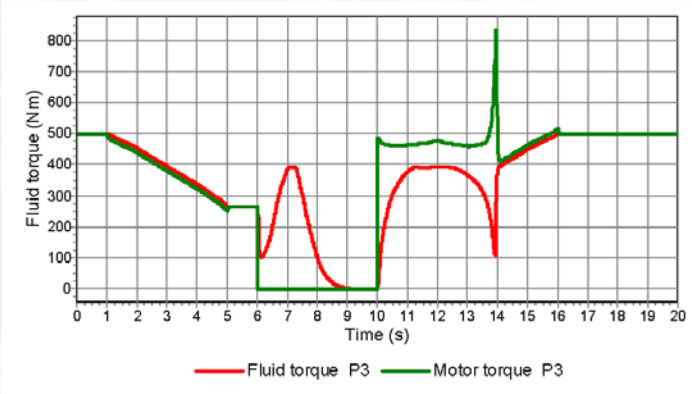


Figure 10: fluid and motor torque-time relation

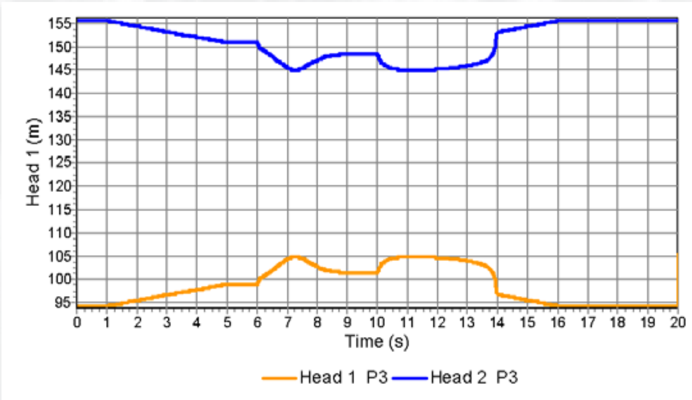


Figure 11: head upstream (orange) and downstream (blue) of the pump in the time

The unique combination to model the full pump characteristic together with the frequency controlled motor behaviour opens the way to all possible well balanced and optimized engineering solutions of pumps in any pipeline system. In combination with the CONTROL module of WANDA the complete operational control of pump stations can be analysed and optimized.

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WL | Delft Hydraulics

Decisive advice: from multidisciplinary policy studies to design and technical assistance on all water-related issues.

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