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Experimental investigation of a pumping station from CET power plant Timisoara

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Abstract. In this paper the results of experimental investigations carried out on four high power centrifugal pumps from CET-Centre power plant Timisoara are presented. The aim of this investigation is to determine the operating curves of the pumps after more than 10 years of operation and to identify the optimum configuration of pumps to operate in parallel. Because of the differences between the pumps, problems occur in exploitation when try to make them operate in parallel configuration. One problem consists in not being able to operate the pumps in parallel in any configuration because of different pumping heads, and another problem is the high consumption of electric power because the pumps are not operated at best efficiency point. Using a measuring equipment based on thermodynamic method and a portable ultrasonic flow rate meter, experimental measurements were performed on each centrifugal pump to obtain the operating characteristics curves (pumping head vs. flow rate and efficiency vs. flow rate). Then analysis was performed to determine the best arrangement of pumps operating in parallel so that the necessary flow rate to be obtained and the parallel operation of the pumps to be without problems and with a minimum electric power consumption.

1. Introduction

The entire network for supplying heat and hot water for the citizens of the city Timisoara consists of 118 distribution stations and 5 district heating stations. The necessary heat and hot water for the cold season is supplied by the CET-Centre power plant together with another power plant, CET-South. This last power plant operates only during cold season, while CET-Centre operates all year long.

CET-Centre power plant is been operating since 1884 and was the first power plant from Europe to produce and supply the energy for the street illumination, Timisoara being the first city of Europe with streets illuminated by electric light. Nowadays this power plant is producing only heat and hot water. It contains five large boilers and in order to heat the water, natural gas and oil fuel is used. The water fed to the five boilers is supplied with the help of a pumping station that was refurbished first in the year 2004 and again in the year 2015.

2. Investigated pumping station

The pumping station from CET-Centre is equipped with three pumps (EPT1, EPT2 and EPT3) supplied by Grundfos in the year 2004 and one pump (EPT4) supplied by Pentair Fairbanks Nijhuis in the year 2015. These pumps have to operate in parallel, in different configuration depending on the demand of the network and to supply a prescribed flow rate and pressure for the consumers.

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From the three pumps manufactured by Grundfos, two of them have constant speed (EPT1 and EPT2) and one has variable speed (EPT3). From the two pumps with constant speed, EPT1 has the diameter of the impeller equal with 585 mm and EPT2 has the diameter of the impeller equal with 590 mm. The diameter of the impeller for the pump EPT3 is also 590 mm. The nominal operating point of this three pumps is characterized by a flowrate of 1300 m^3/h and a pumping head of 125 m at a rotational speed of 1485 rpm. This type of pumps has the inlet and outlet section on the same axis and the impeller is with double suction section, figure 1.

Figure 1. Centrifugal pump HS 300x350x590 manufactured by Grundfos.

The fourth pump, manufactured by Pentair Fairbanks Nijhuis, Venus1-2510.650 (EPT4), has an impeller with a diameter of 626 mm, the inlet section diameter is 300 mm and the outlet section diameter of 250 mm. The nominal operating point of this pump is characterized by a flowrate of 1200 m^3/h and a pumping head of 144 m at a rotational speed of 1490 rpm, figure 2. This pump also has variable speed.

Figure 2. Centrifugal pump Venus1-250.650 manufactured by Pentair Fairbanks Nijhuis.

The position of the four pumps inside the pumping station is presented in figure 3. During the spring, summer and autumn, when the CET has to deliver only hot water, the pumping station operates with maximum two pumps in parallel, usually one pump with variable speed and one pump with constant speed. During the winter, situations may occur when all the pumps have to operate in parallel configuration. The operating regimes cover a wide range for the flow rate and pumping head, the main concern being to supply optimal operating conditions for the most far distribution station.

Because the four pumps are of different types and also because over the years the operating characteristics of them may be altered by the operating cycles, the challenge is to find the operating regimes for each pump which allow smooth parallel operation.
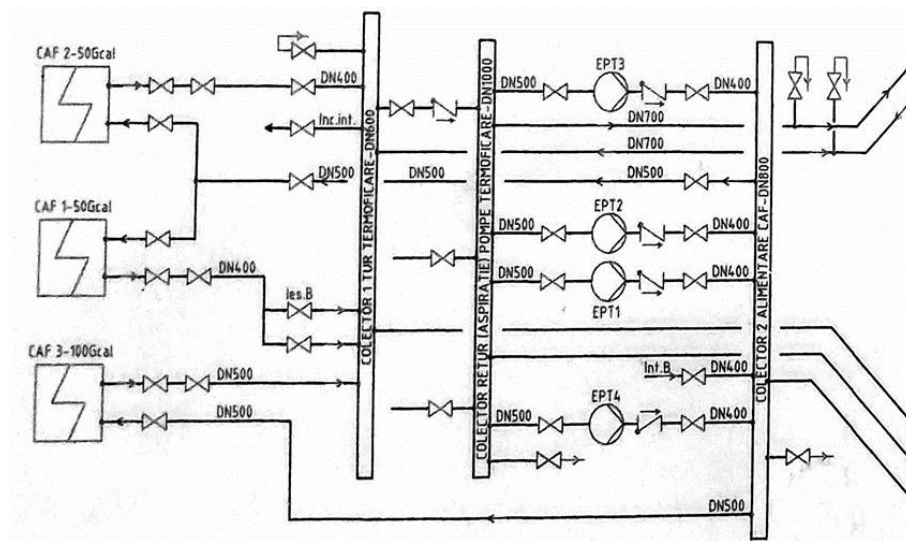
Figure 3. The displacement of the four pumps in the pumping station.

3. Measuring principle, equipment and setup

In order to determine the current operating curves of the four pumps, the measurement of the pressure, flow rate and the efficiency is needed. For the measuring of these parameters a portable ultrasonic flow rate meter is used together with an equipment based on thermodynamic method for measuring the pumping head and the hydraulic efficiency of the pumps.

The Fluxus ultrasonic flow rate meter is produced by Flexim and allows the measurement for pipes with the diameter up to 1.2 m. The measurement system consists of a transmitter and the ultrasonic transducers with the transducers cables. The ultrasonic transducers are mounted on the outside of the pipe. Ultrasonic signals are sent through the fluid and received by the transducers. The transmitter controls the measuring cycle, eliminates the disturbance signals and analyses the useful signals. The flow velocity of the fluid is measured in the TransitTime mode using the ultrasonic-transit time difference correlation principle. If the proportion of gas or solid particles is high, the transmitter can toggle to the NoiseTrek mode.
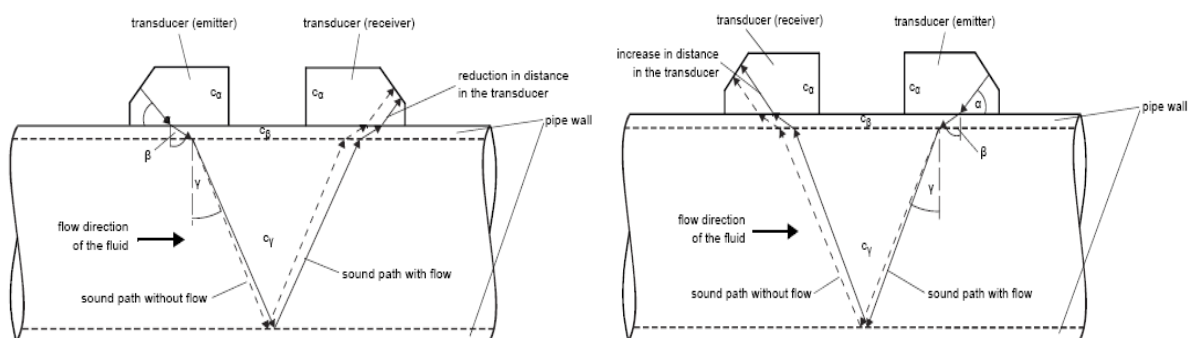

Figure 4. Measuring of the flow velocity in the TransitTime mode.

In the TransitTime mode the signals are emitted and received by a transducer pair alternatively in and against the flow direction. If the fluid moves, the signals propagating in the fluid are displaced with the flow. The displacement causes a reduction in distance for the signal in the flow direction and an increase in distance for the signal against the flow direction in the section of the receiving transducer. This causes a change in the transit times. The transit time difference is proportional to the average flow velocity. The average flow velocity of the fluid is calculated as follows for the TransitTime mode:

$$v = k_{\text{Re}} \cdot k_a \cdot \frac{\Delta t}{2 \cdot t_{fl}} \qquad (1)$$

where k_{re} is fluid mechanics correction factor, k_a is the acoustic calibration factor, Δt is the transit time difference and t_{fl} is the transit time in fluid.

When fluids with a high proportion of gas bubbles or solid particles are measured, the attenuation of the ultrasonic signal increases and can inhibit the propagation of the signal in the fluid. A measurement in the TransitTime mode is not possible anymore and the NoiseTrek mode is used. This mode uses the presence of gas bubbles and solid particles in the fluid. The measurement setup used in the TransitTime mode does not need to be changed. Ultrasonic signals are sent into the fluid at short intervals, reflected by the gas bubbles or the solids particles and again received by the transducer. The transit time difference between two consecutive measuring signals that are reflected by the same particle is determined. The transit time difference is proportional to the distance covered by the particle in the time between two consecutive measuring signals and therefore to the velocity at which the particle moves through the pipe. The average value of all measured velocities of gas bubbles and/or particles corresponds to the flow velocity of the fluid:

$$v = k_{\text{Re}} \cdot k_a \cdot \frac{\Delta t}{2 \cdot t_s} \qquad (2)$$

where t_s is the time interval between the measuring signals.

Depending on the signal attenuation, the error of measurement in the NoiseTrek mode can be greater than in the TransitTime mode.
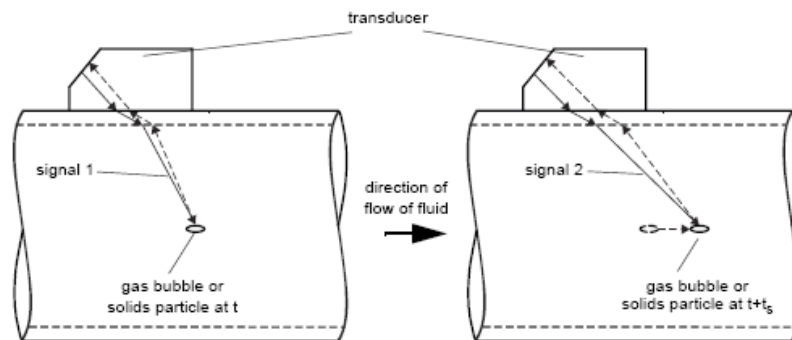

Figure 5. Measuring of the flow velocity in the NoiseTrek mode.

There is also a HybridTrek mode that combines the TransitTime mode and NoiseTrek mode. During a measurement in the HybridMode, the transmitter automatically toggles between the TransitTime mode and the NoiseTrek mode depending on the gaseous or solid content of the liquid.

With the flow velocity measured, the volume flow rate, Q, and the mass flow rate, $\dot{m}$, is calculated using the following equations:

$$Q = v \cdot A \qquad (3)$$
$$\dot{m} = \rho \cdot Q \qquad (4)$$

where A is the cross-sectional pipe area and ρ is the density of the liquid.

The flow rate meter is able to measure flow velocities up to 25 m/s and the error limit of the measured flow rate is 0.2%.

The P22F equipment based on the thermodynamic method is produced by Robertson Technology and allows the measurement of the pumping head and the efficiency of the pump. In the thermodynamic method, pump efficiency is measured by means of temperature and pressure probes fitted to tapping points on the inlet and outlet section of the pump. The critical parameter is the differential temperature across the pump, which must be measured to an accuracy of typically 1 mK. This is achieved with Robertson Technology's CoolTip[TM] technology incorporated into the P22F to provide accurate and

stable measurement of pump efficiency. On-site constraints make it difficult to accurately measure pump efficiency under installed conditions by the same method that pump manufacturers traditionally use for work tests. The advent of the thermodynamic method has provided a solution to this problem. Now accurate measurements can be made on installed pumps. That is because the thermodynamic technique requires measurement of only two parameters, temperature and pressure, to determine pump efficiency and energy difference is effectively being measured. A 5% error in the measurement of energy difference (typically 20%) leads to a corresponding error in the pump efficiency measurement of 1%, for a pump operating at 80% efficiency. However, with the traditional technique, and 5% instrumentation accuracy, the error in the pump efficiency measurement would also be 5%.

The equations for the thermodynamic method are:

$$\eta = \frac{E_h}{E_m} \tag{5}$$

$$E_h = \frac{p_2 - p_1}{\rho} \tag{6}$$

$$E_m = a \cdot (p_2 - p_1) + c_p \cdot (T_2 - T_1) \tag{7}$$

where E_h is the hydraulic energy per unit mass of fluid, E_m is the mechanical energy per unit mass of fluid, p_1, p_2 are the pressure of the fluid at the inlet and outlet section of the pump, T_1, T_2 are the temperature of the fluid at the inlet and outlet section of the pump, ρ is the density of the fluid, a is the isothermal coefficient of the fluid, c_p is the specific heat capacity of the fluid.

The pumping head is calculated with the following equation:

$$H = \frac{p_2 - p_1}{\rho \cdot g} + z_2 - z_1 + \frac{v_2^2 - v_1^2}{2 \cdot g} \tag{8}$$

where z_1, z_2 are the position of the tapping points at the inlet and outlet section of the pump and v_1, v_2 are the velocity of the fluid at the inlet and outlet section of the pump.

The theoretical background to the thermodynamic method for pumps is documented in ISO 5198 and other standards, [9]. The performance of an instrument employing this method is determined by the design, accuracy and stability of the temperature and pressure probes. The pump efficiency measured with the P22F equipment is measured by accurate and innovative temperature and pressure probes. The electronic circuits are contained in the probe handles, thus eliminating potential errors from connector and cable resistances. The design allows the self-contained calibration of each temperature and pressure probe, without reference to any external electronic components. This greatly simplifies operational use and maintenance. The probes are connected to each other and to the control computer by an RS485 serial interface (Modbus[TM] protocol), which provides high immunity to electromagnetic interference over long distances. The connecting cables only carry the digital signals and the power to the probes.

The standard setup for the components of the P22F equipment is presented in figure 6. The temperature probes employ a novel design to ensure high sensitivity and long-term stability. Each temperature probe has dual sensors, to detect drift in an individual sensor. The sensor and signal conditioning electronics are stable over long time period. Experience over several years shows no observable drift, less than 0.25mK. the sensors give a high electrical signal, which minimises electronic noise. With the standard temperature probes (0-60° C), the precision of each temperature point due to electronic noise is 0.11 mK. A set of 25 readings results in a standard error of 0.025 mK for the average temperature. The standard error in the average differential temperature due to electronic noise will then be 0.035 mK. The signal conditioning electronics minimises self-heating effects in the sensors. The probes are also designed to minimise the stem effects, which can otherwise occur due to differences between the fluid and ambient temperatures. The pressures probes allows measurement of pressure in the range of -1 to 25 bar. These probes have built-in temperature sensors and active temperature correction. The accuracy of 0.1% is maintained over a wide temperature range.
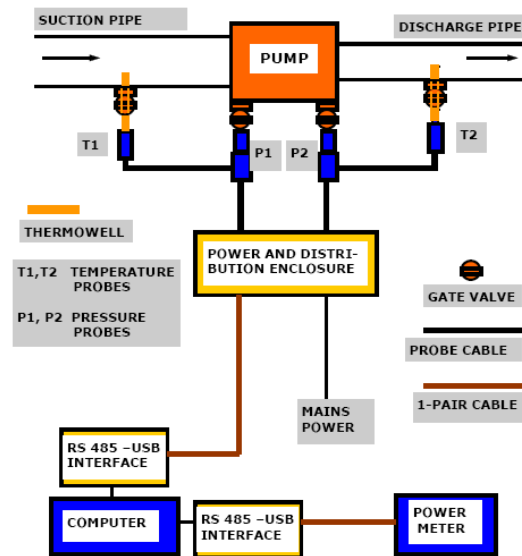
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Figure 6. Standard configuration for P22F equipment.

Tapping points are required on the inlet and outlet of pump, ideally about two pipe diameters from the pump flanges, but one pipe diameter is sufficient if space is tight.

With the P22F, the accuracy of the pump efficiency measurement is typically ±1%. The temperature rise across the pump increases with head, so the higher the head, the more accurate the efficiency measurements. Also, the temperature rise is higher for less efficient pumps, as more energy is being lost in the pump, so the lower the pump efficiency, the more accurate are the measurements. The accuracy for this equipment, following international standard, is defined in terms of uncertainty, at the 95% confidence level. Thus, if the efficiency is 70%, with an uncertainty of 1%, there is a 95% probability that the pump efficiency lies between 69 and 71%.

The experimental setup for measuring one of the investigated pump is presented in figure 7. One can see the temperature and pressure probes of the P22F equipment on the inlet and outlet section of the pump and the transducers of the Fluxus flow rate meter.

Figure 7. P22F equipment and Fluxus flow rate meter setup for an investigated pump.

4. Results and analysis

The experimental investigation of the operation of the four large centrifugal pumps was carried out during 4 days, on each day a single pump being investigated. For each pump, 8 operating points were measured. For every investigated operating point, three sets of measurements were performed, each containing 20 samples. The results of all that measurements were averaged and so was obtained the parameters (flow rate, pumping head and efficiency) for each investigated operating point for every pump. Each pump was operating at a constant rotational speed of 1490 rpm.

Table 1. Operating points for the pumps EPT1 and EPT2.

Pump	EPT1			EPT2		
No.	Q $[m^3/h]$	H $[m]$	η $[\%]$	Q $[m^3/h]$	H $[m]$	η $[\%]$
1	1316.13	110.99	59.78	1599.23	126.28	86.37
2	1197.62	110.55	57.68	1443.58	129.57	85.09
3	1068.92	113.60	56.46	1315.58	132.40	83.77
4	916.34	115.41	53.41	1154.22	132.99	80.35
5	708.32	119.13	47.81	997.56	133.74	75.68
6	567.38	117.87	43.04	867.66	135.87	71.39
7	454.49	118.40	36.55	616.73	136.24	58.39
8	254.90	122.33	24.73	239.51	138.66	28.14

Table 2. Operating points for the pumps EPT3 and EPT4.

Pump	EPT3			EPT4		
No.	Q $[m^3/h]$	H $[m]$	η $[\%]$	Q $[m^3/h]$	H $[m]$	η $[\%]$
1	1605.47	118.24	81.79	1213.36	117.26	82.75
2	1512.26	120.36	81.55	1136.51	122.87	84.36
3	1375.54	123.04	80.58	1035.37	130.40	85.62
4	1262.56	125.88	80.01	931.56	137.62	86.57
5	1099.65	128.18	77.34	726.71	144.17	80.63
6	965.29	129.39	74.02	503.80	150.18	72.05
7	626.42	133.47	62.80	394.06	155.86	68.20
8	294.76	136.09	34.15	295.08	153.06	57.67

Analysing figure 8, one can observe that even though the pumps EPT1, EPT2 and EPT3 are similar and manufactured by the same producer, there is a big difference between pump EPT1 and the other two pumps, EPT2 and EPT3 regarding the pumping head characteristic curve. The significant difference may be caused by the more unfavourable flow conditions from the inlet section of the pump EPT1 determined by the different geometry of the inlet piping. That can lead to a flow with pre-swirl on the inlet section and this can cause a reduce value for the pumping head, [1]. Also, a cause for the low values of the pumping head, might be some alteration of the impeller caused by cavitation. The presence of cavitation may be determined by the shape of the inlet piping system of this pump which is different from the other two similar pumps. An inspection of the impeller of the pump EPT1 is required in order to determine the presence of the effects of cavitation.

The fourth pump, EPT4, being a totally different centrifugal pump than the other three, has a completely different pumping head curve. Because of the different pumping head curve characteristics, a stable parallel operation of these pumps will be difficult to obtain.
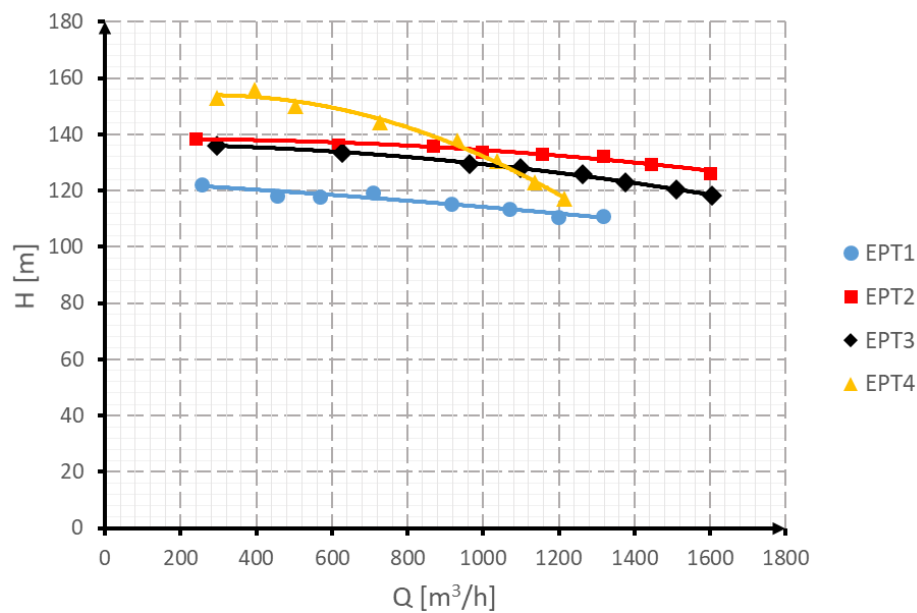
Figure 8. Pumping head vs. flow rate for the investigated centrifugal pumps.

From the efficiency point of view, figure 9, the best efficiency for lower values of the flow rate is achieved by the centrifugal pump EPT4 and for higher values of the flow rate the best efficiency is achieved by both pumps EPT2 and EPT3. The centrifugal pumps EPT2 and EPT3 have similar efficiency values for almost all the operating range. There is a slight difference in efficiency only between 1400 m^3/h and 1600 m^3/h.

Because of the poor values of the pumping head, pump EPT1 has also low values of the efficiency compared to the other three pumps. When operating, this pump will consume more electrical power than the other two similar pumps (EPT2 and EPT3) leading to higher costs. A refurbishment of this pump is recommended.
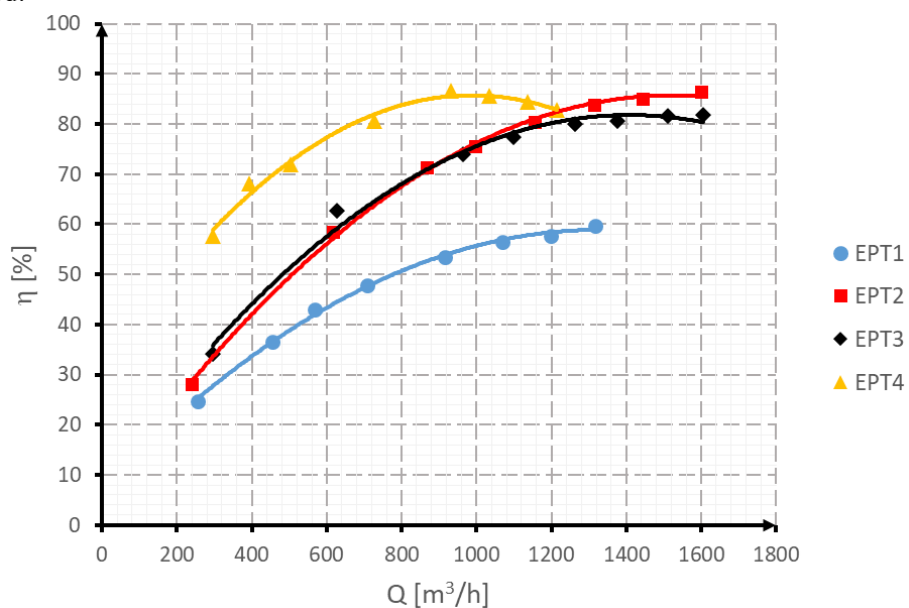

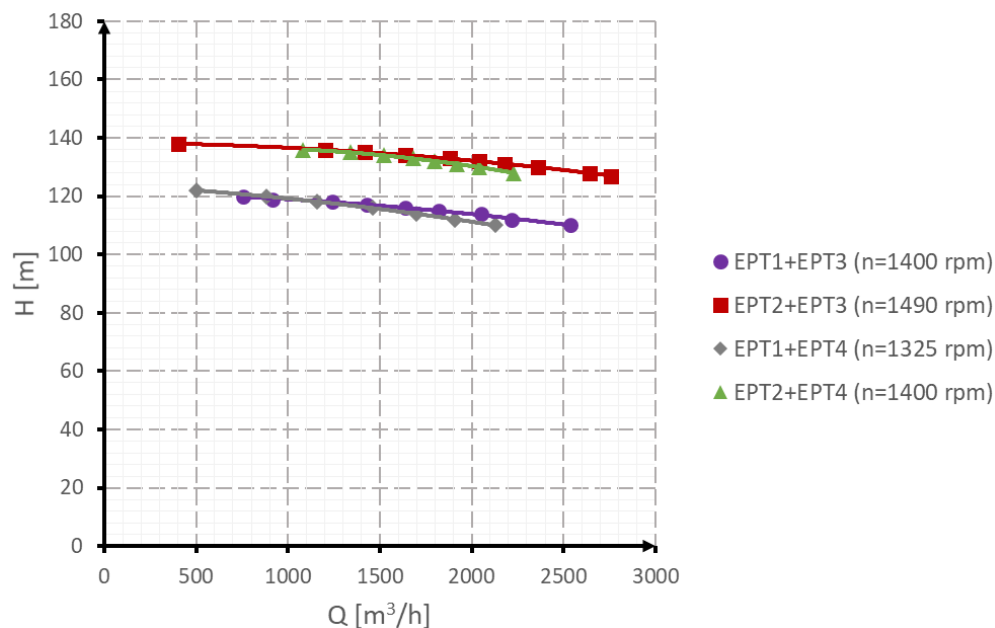**Figure 9**. Efficiency vs. flow rate for the investigated centrifugal pumps.

Figure 10. Pumping head vs. flow rate for the investigated centrifugal pumps operating in parallel.

Gulich, [1], shows that for parallel operation stable, steadily falling Q-H-curves are required, because unambiguous intersection points between the combined pump characteristics and the system curve must be obtained. If the pump Q-H-curve is flat in the part load range, one pump can displace another during parallel operation, since the Q-H-curves of the individual pumps are not exactly identical due to manufacturing tolerances and wear. In our case the Q-H-curves for the four pumps are very different which will lead to problems in parallel operating. Even pumps whose Q-H-curves are clearly different can be operated in parallel. In that case, the pump with the lower shut-off head should be started up only when the required head is lower than the shut-off pressure of the pump to be added. Similarly, the pump with the lower shut-off pressure has to be shut down before the head required by the system exceeds the shut-off pressure of the smaller pump.

We analysed the possibility of operating these four centrifugal pumps in parallel configuration. Only two pumps are required to operate in parallel at a time. From this analysis four possible scenarios were obtained, figure10, where one pump has constant speed and the other one has variable speed:

- pump EPT1 operating in parallel with pump EPT3 which has a rotational speed of 1400 rpm,
- pump EPT1 operating in parallel with pump EPT4 which has a rotational speed of 1325 rpm,
- pump EPT2 operating in parallel with pump EPT3 which has a rotational speed of 1490 rpm,
- pump EPT2 operating in parallel with pump EPT4 which has a rotational speed of 1400 rpm

The best cases for parallel operation of the pumps are the last two, because the values of the pumping head are higher than the requested value of 125m for all the flow rate domain. From those two cases the most favourable is that of pump EPT2 operating together with pump EPT3 because the range of the flow rate is the largest. Because the system curve is changing a lot due to the variable demand from the network, an operating of these pumps only in the range of high values of the efficiency is not possible.

5. Conclusions

In this paper are presented the results of the experimental investigations carried out on four large centrifugal pumps which operates in a pumping station from the CET-Centre power plant. The investigations were performed in order to identify the best parallel operation scenario of those pump. For that, the operating curves were necessary to be obtained. In order to obtain the operating curves, the operating parameters of the pumps (flow rate, pumping head and efficiency) were needed to be

measured. The operating parameters were measured using an ultrasonic flow rate meter and an equipment based on thermodynamic method.

A poor operation of the pump EPT1 was discovered. This can be caused by the wear of the impeller due to cavitation and also by the unfavourable flow condition on the inlet section of the pump. A refurbishment operation of this pump is needed.

From the analysis of the operating curves, the best scenario for operating these pumps in parallel configuration, appear to be the operation of the pump EPT2 with the pump EPT3 which has to operate at a rotational speed of 1490 rpm. Using this scenario of operating the pumps, higher values of pumping head is achieved over a large range of flow rates.

Acknowledgements

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