FLOW MARGIN IMPROVEMENT OF EXISTING COOLING WATER PUMPS WITH RESPECT TO EXTREME PART LOAD OPERATION CONDITIONS

by

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ABSTRACT

In the present paper, the adaptation of existing cooling water pumps to extreme part load conditions by cutting down the impeller in the meridional plane is described. The influence of different impeller shapes on the head curve characteristic and the development of the part load recirculation zone are investigated theoretically. The theoretical investigations are accompanied by model pump tests. Besides the head curve and efficiency characteristics, the noise and vibration levels were measured and assessed for the different impeller shapes. The vibrations especially show a distinct characteristic that allows a very precise definition of limiting flowrate. Comparing the theoretical and experimental results, good agreement can be claimed. In particular, the onset of the part load recirculation is predicted very well. The results obtained for the model pump are successfully validated by prototype pumps.

INTRODUCTION

In many facilities in the fields of energy or process technology, tubular casing pumps are installed to ensure the fresh water or cooling water supply. Normally, high flowrates and relatively low discharge heads are required, so that axial or semiaxial impeller pumps with comparatively high specific speeds ($n_q \approx 90$ to 150 min^{-1}; n_s = 4650 to 7750 rpm in US units) are in use. In most cases, the water is taken out of inlet chambers that are connected to cooling tower systems or to the river (or sea) side. Depending on the plant conception, either pumps running in parallel feeding the whole system or multiple stand alone solutions are implemented. Because of the high installed power and the long-term running periods, these pumps contribute significantly to the overall energy consumption of the whole plant. Therefore, high pump efficiency is demanded, and further high pump reliability is required to ensure a continuous production process.

Pumps are seldom operated only at the design point (Q_{BEP}) during the whole running period. Depending on the production process, operation at part load ($Q < Q_{BEP}$) and overload ($Q > Q_{BEP}$) conditions are frequently required. Thereby it has to be ensured that the guaranteed operation range in the form of limiting flowrates for long-term operation (LTO) $Q_{LTO\ max}$ and $Q_{LTO\ min}$ given by the pump manufacturer will not be exceeded. Exceeding these limits for long-term operation leads to failures and may, at worst, damage the pump. This is especially critical for flowrates below the minimum flow limit $Q_{LTO\ min}$.

It is well known that for flow conditions $Q_{min} \approx 0.6$ to $0.7 \cdot Q_{BEP}$, a distinct reverse flow zone at the pump impeller inlet (part load recirculation) is formed. For axial and semiaxial flow pumps, the sudden development of the part load recirculation leads to a drop or instability in the head curve characteristic and to an increase of noise and vibration level, which is described by Stepanoff (1965), Tanaka (1980), or Sakar (1992). The increase of vibration damages the bearings very rapidly and causes total pump failure after a short time. This is why the minimum flowrate for long term operation $Q_{LTO\ min}$ is defined with a distinct distance to the onset of part load recirculation Q_{min}.

For certain reasons, the boundary conditions in an existing plant may be changed in a way that the installed pumps would operate partially below the part load operation limit. This was the case in a plant where nine mixed flow tubular casing pumps were installed. A schematic representation of the pumps is given in Figure 1. The main design parameters are listed in Table 1. The corresponding pump characteristic is shown in Figure 2. The pumps run at a speed of n = 495 min^{-1}, reaching a peak efficiency of η_{BEP} = 91 percent at Q_{BEP} = 23.600 m^3/h (103,840 gpm). The operating range is limited to $Q_{LTO\ min}$ = 16,000 m^3/h (70,400 gpm). Due to some changes in plant conditions, long-term operation was additionally required for a flowrate of Q = 13,500 m^3/h (59,400 gpm) after pump installation (open triangle in Figure 2). Moreover a new main duty point at Q = 19,200 m^3/h (84,480 gpm) and H = 36 m (118 ft) (closed circle in Figure 2) was also required. This target operation point nearly matches the original head curve characteristic ($\Delta H/H \approx 3$ percent), so that two main hydraulic requirements appeared:

- Reduction of part load operation limit $Q_{LTO\ min}$
- Maintenance of head curve characteristics at target operation point

Normally the pumps would have to be replaced with smaller ones to adjust for the new conditions. To avoid the pump replacement, investigation was undertaken to see if the minimum operation limit could be shifted by only modifying the existing pump hydraulics. In order to keep the cost as low as possible, the existing pump impellers and diffusers had to be used too.

From the literature (Goto, 1993; and Cumpsty, 1989), it is known that the onset of part load recirculation can be shifted to lower capacities using active or passive control mechanisms in

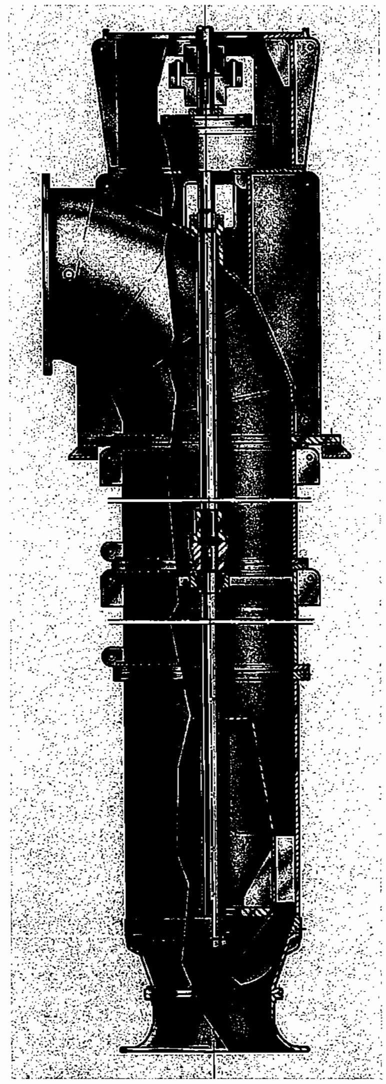

Figure 1. Tubular Casing Cooling Water Pump.

Table 1. Design Parameters of the Prototype Pump.

Design flow rate	Q	= 22,500 m^3/h
Design head	H	= 35 m
Peak efficiency	η_{BEP}	= 91%
Power	P	= 2.4 MW
Speed	n	= 495 min^{-1}
Specific speed	n_q	= 86 min^{-1}, m^3/s, m
	n_s	= 4441 rpm, gpm, ft
Discharge diameter	DN	= 1400 mm
Impeller inlet diameter	D_1	= 1000 mm
Impeller outlet diameter	D_2	= 1320 mm
Number of blades	z	= 4
Impeller type	Semi-axial open impeller	

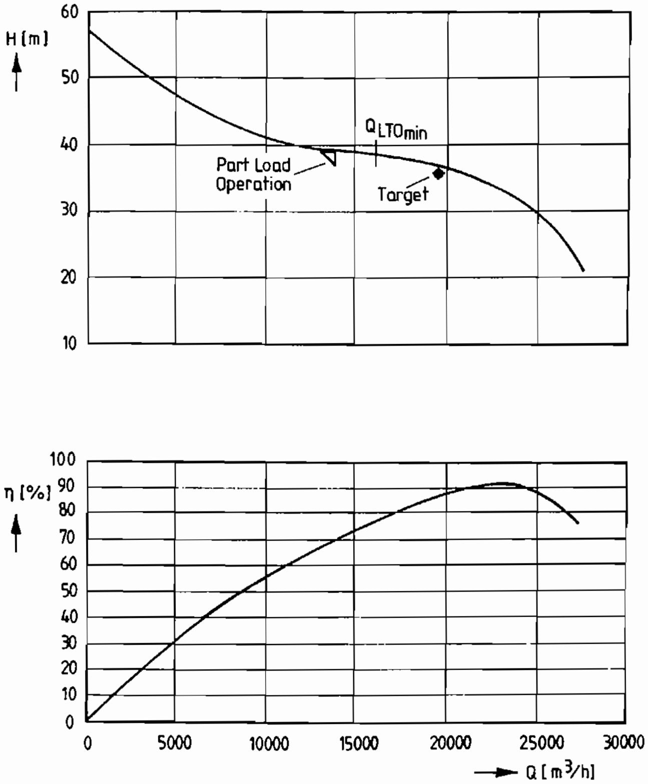

Figure 2. Head Curve Characteristic of Prototype Pump.

terms of flow injection or casing treatment, respectively. While active control requires additional peripheral equipment, passive control often implies loss in overall efficiency. Therefore investigation was undertaken to see whether the target described above could only be reached by a special cutting down of the open impeller in the meridional plane and adaptation of new wear rings.

THEORETICAL INVESTIGATIONS

Reducing the flowrate of an existing impeller can be achieved by cutting down a complete streamtube in the impeller meridional plane, as illustrated in Figure 3. In case of radial impellers, the outlet diameter (and, most of all, the blade outlet angle) is constant for all streamtubes. Therefore the impeller head remains unchanged by cutting down a complete streamtube. This is not the case for mixed flow impellers, where outlet diameter and blade outlet angle vary along the impeller exit. Simply assuming that the flowrate depends mainly on the impeller inlet while the head is affected mainly by the impeller outlet, a conical cutback was considered to achieve the desired flow reduction without decreasing head. Thereby the impeller inlet diameter is reduced by a certain amount, while the outlet diameter remains unchanged (Figure 3). In order to keep the adaptation as simple as possible, linear variation of radius between inlet and outlet was desired.

The effect of different meridional impeller shapes on the theoretical pump characteristic is shown in Figure 4. A simple but very fast 2D-singularity method was used to calculate the theoretical head produced at each streamtube. From these, the overall head was evaluated using the relation of radial equilibrium. Even if viscous and 3D-effects are neglected here, the effect of the different impeller shapes on the theoretical head can be obtained qualitatively. For the theoretical head curve of the original impeller shape (D_1 = 1000 mm/D_2 = 1320 mm), which will be referred to as *Configuration 0* in the following, the design point at Q = 22,500 m³/h (99,000 gpm) and the operation limit $Q_{LTO\ min}$ = 16,000 m³/h (70,400 gpm) are added in Figure 4 as derived from field tests, indicated by the solid triangle and the dashed line, respectively. For those points, the calculated flow pattern at the impeller inlet

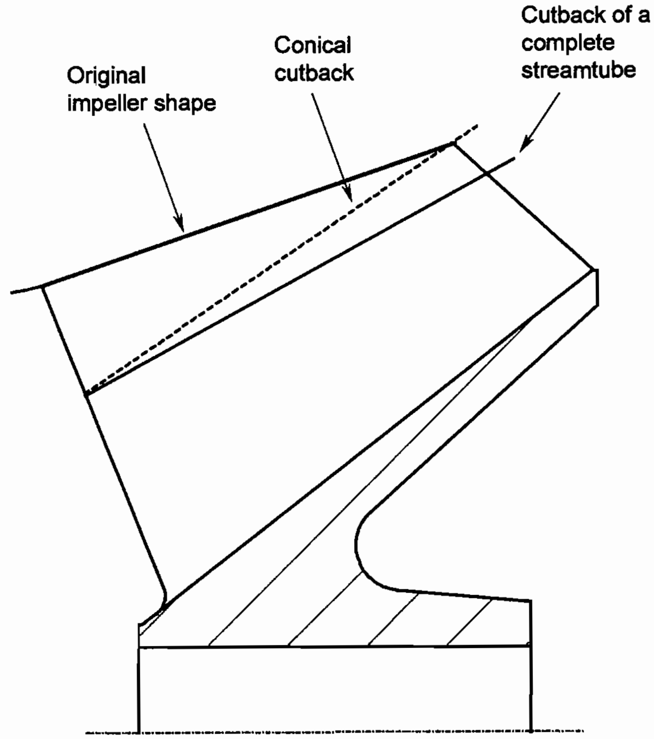

Figure 3. Cutting Down of Impeller.

(velocity components and flow angle) was assumed to be representative. Turning down the impeller by a full streamtube (D_1 = 935 mm/D_2 = 1284 mm) shifts the theoretical head curve to lower flows and lower heads. By comparing the calculated flow pattern with that of the full impeller it was found that the part load operation limit was reduced to $Q_{LTO\ min}$ ≈ 13,500 m³/h (59,400 gpm). Moreover, design point conditions appeared at Q ≈ 19,500 m³/h (85,800 gpm) (open triangle), which nearly represents the new target flow.

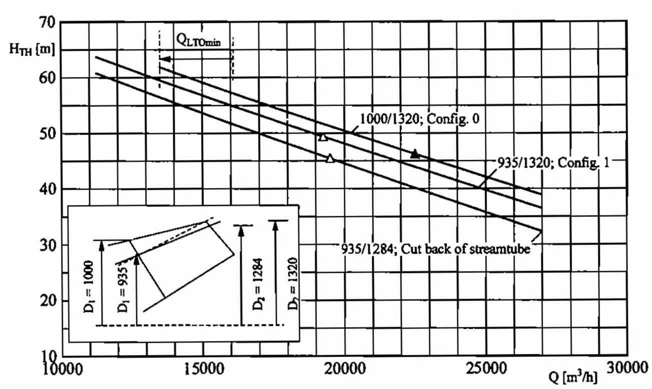

Figure 4. Influence of Impeller Shape on Theoretical Head Curve.

Comparing the theoretical head at this point, a reduction of $\Delta H_{th}/H_{th}$ ≈ 12.5 percent is seen, which exceeds the allowable limit. By conically turning down the impeller (D_1 = 935 mm/D_2 = 1320 mm), nearly the same flow reduction is achieved whereby the loss in impeller head is only $\Delta H_{th}/H_{th}$ ≈ 4.5 percent, which nearly meets the requirements. In the following, this configuration will be referred to as *Configuration 1*.

In order to confirm these results, the effect on head curve characteristic and especially the development of the part load recirculation zone was investigated for the given configurations by means of a 3D-Euler code. Even this method does not include viscous effects, but takes into account the influence of energy distribution on the resulting 3D-flow field. Assuming that the change of energy distribution inside the impeller initiates the part load recirculation onset, it should be predictable by means of this tool. A comparison of theoretical head predicted by both calculation methods is given in Figure 5. In comparison to the results of the 2D-method, the theoretical head at design flow conditions given by the 3D-code (Q = 22,500 m^3/h (99,000 gpm), H_{th} = 36.9 m (120.95 ft)) corresponds fairly well to the measured head (H = 35 m (114.7 ft)) presented in Figure 2. The reason for overpredicting the head in the case of 2D-singularity methods is well known and is related to the method of modeling the trailing edge. Nevertheless, the ratio of head reduction is equal for both methods ($\Delta H_{th}/H_{th} \approx 4.5$ percent). While, for the 2D-singularity method, the shift of minimum flow was correlated to the calculated flow velocity triangles and the documented onset of part load recirculation, the 3D-calculation is able to predict this onset. Looking at the theoretical head curve evaluated for the original configuration (*Configuration 0*, D_1 = 1000 mm/D_2 = 1320 mm) a distinct change of slope can be observed at Q = 16,000 m^3/h (70,400 gpm), which corresponds to the documented $Q_{LTO\ min}$. A similar change of head curve appears for *Configuration 1* (D_1 = 935 mm/D_2 = 1320 mm) at Q = 13,500 m^3/h (59,400 gpm). These discontinuities are connected to the onset of a recirculating flow pattern inside the impeller as shown below.

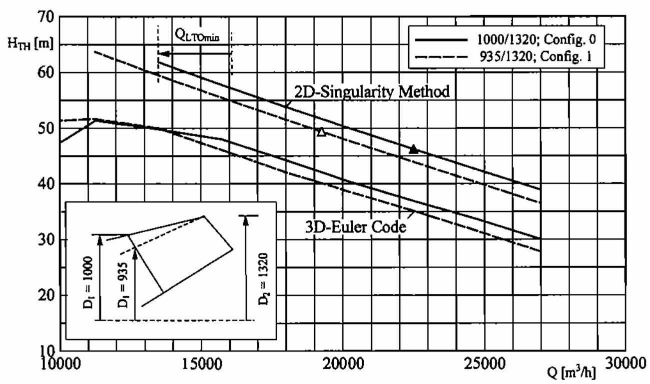

Figure 5. Comparison of Predicted Theoretical Head Curves; 2D-Singularity Method Versus 3D-Euler.

For the original configuration (*Configuration 0*), the calculated (3D-Euler code, pitchwise averaged) meridional velocity components at the impeller inlet are plotted in Figure 6 for different flowrates. At design flow (Q = 22,500 m^3/h (99,000 gpm)), the meridional velocity is slightly curved with an average magnitude of about $c_m \approx 8$m/s. Even at $Q_{LTO\ min}$ = 16,000 m^3/h (70,400 gpm) a homogeneous velocity profile is predicted. At a flowrate of $Q_{min\ th.}$ = 13,500 m^3/h (59,400 gpm), the development of part load recirculation is indicated by negative velocity components near the casing. The evolution of part load recirculation is well illustrated by the calculated velocity vector field in the meridional plane, plotted for the blade-to-blade center section in Figure 7. At design flow (Q = 22,500 m^3/h (99,000 gpm)), a homogenous vector field exists. At Q = 16,000 m^3/h (70,400 gpm), the first flow disturbances appear inside the impeller near the casing, forming a recirculating flow pattern that increases with further flow reduction. At a flowrate of Q = 13,500 m^3/h (59,400 gpm), it leaves the impeller entrance in the upstream direction, forming the well known part load recirculation (Q = 10,980 m^3/h (48,310 gpm)).

The calculated meridional velocity profiles for the conical impeller shape (*Configuration 1*) are presented in Figure 8. From the figure, it clearly appears that even for the desired part load conditions (Q ≈ 13,500 m^3/h (59,400 gpm)), a homogeneous velocity profile exists. The onset of part load recirculation is predicted at $Q_{min\ th.}$ = 10,980 m^3/h (48,310 gpm) which contains a sufficient safety margin with respect to the desired minimum flowrate. Analogous to Figure 7, the predicted velocity vectors are presented in Figure 9. In this case, the first recirculating flow pattern can be observed at Q = 13,500 m^3/h (59,400 gpm), increasing with decreasing flowrate and leaving the impeller inlet at Q = 10,980 m^3/h (48,312 gpm).

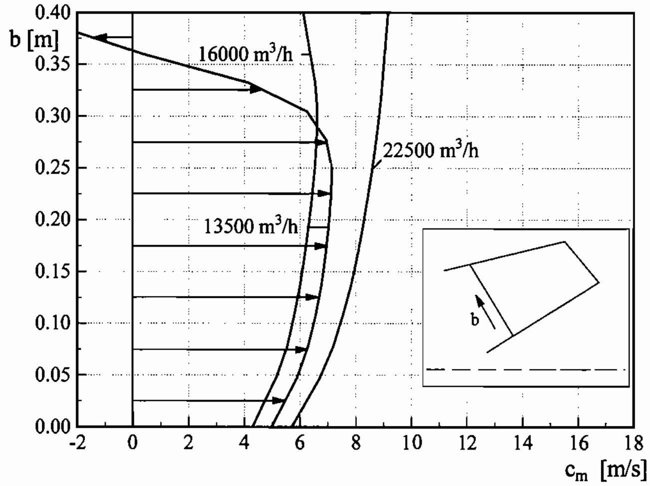

Figure 6. Calculated Meridional Velocity Profiles at Impeller Inlet, Configuration 0.

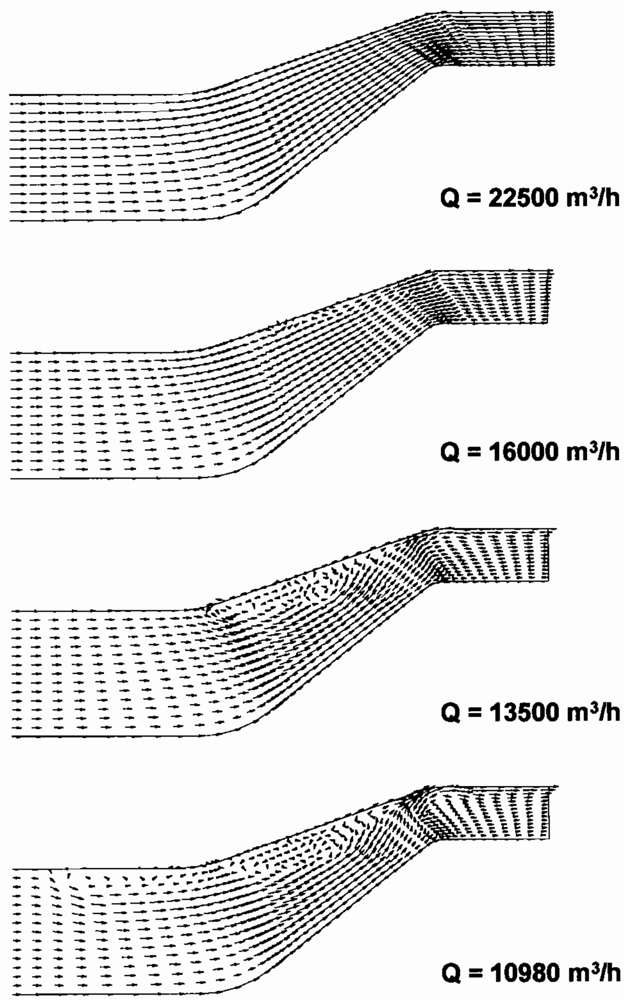

Figure 7. Calculated Meridional Velocity Vector Field for Configuration 0, Blade-to-Blade Center Section.

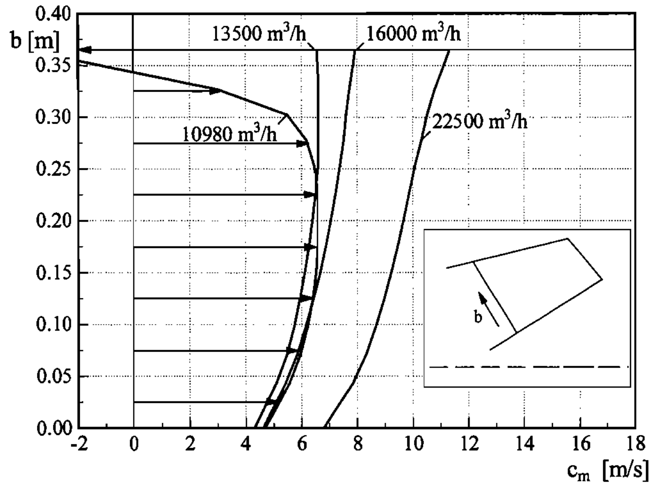

Figure 8. Calculated Meridional Velocity Profiles at Impeller Inlet, Configuration 1.

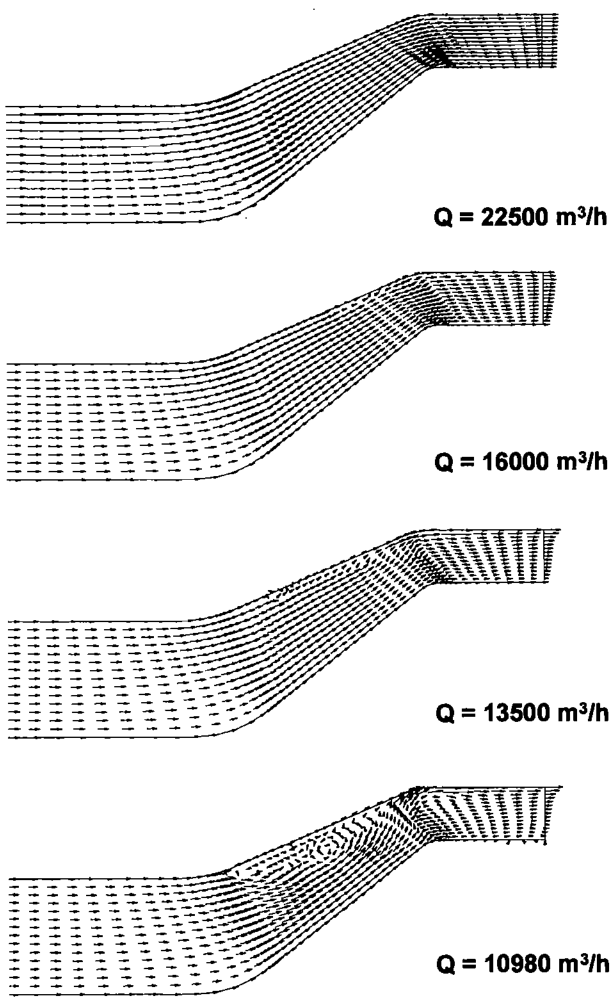

Figure 9. Calculated Meridional Velocity Vector Field for Configuration 1, Blade-to-Blade Center Section.

EXPERIMENTAL INVESTIGATIONS

The theoretical investigations were accompanied by model tests. For the tests, a model pump was manufactured at a scale of m = 0.314 and installed in a closed loop. A cross sectional view of the model pump is given in Figure 10. Besides the performance measurements, noise and vibration measurements were carried out. For these, the model pump was equipped with a vibration sensor fixed at the pump casing. Additionally, microphones were positioned outside the pump at a distance of one meter, located in the plane of the shaft axis. The pump casing is made out of Plexiglass, so that the resultant flow pattern could be observed from the outside. In order to realize constant tip clearance (s = 0.3 mm) the casing was individually adapted to each impeller shape. All measurements were carried out at a speed of $n = 1480\ \text{min}^{-1}$ and with suction pressures representing the plant conditions.

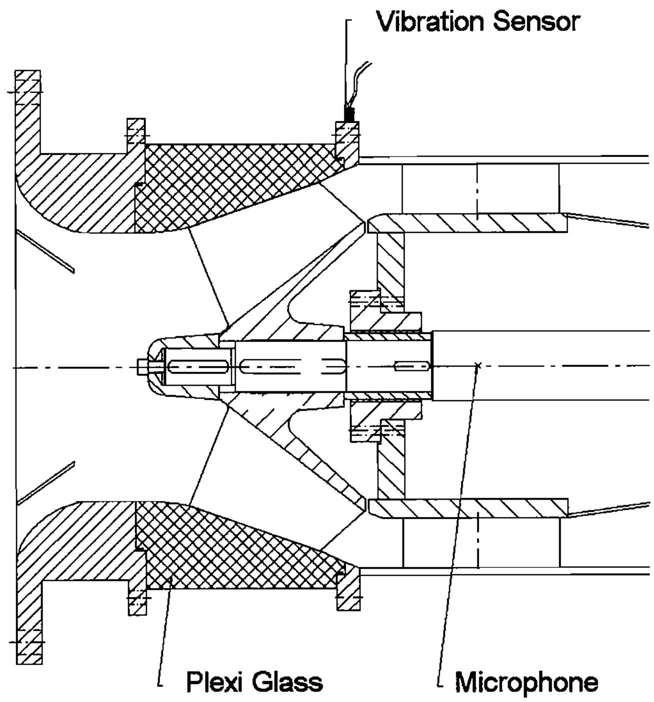

Figure 10. Model Pump, Cross Sectional View.

The experimental model pump results achieved for the original contour (*Configuration 0*) are plotted in Figure 11 in form of head (H), vibration (v_{eff}), and noise level (L_{pA}) characteristics. The flow capacity is given in two scales. The upper one represents the flowrate of the model pump, while the lower one represents the corresponding flowrate of the prototype. At design conditions (Q_{model} = 2084 m³/h (9170 gpm); Q = 22,500 m³/h (99,000 gpm)), vibration and noise levels are quite low. By decreasing the flowrate, noise and vibration increase slightly until the part load operation limit ($Q_{LTO\ min\ model}$ = 1482 m³/h (6520 gpm); $Q_{LTO\ min}$ = 16,000 m³/h (70,400 gpm)) is reached. Further flow reduction leads to progressive increase of noise and vibration levels until the part load recirculation is formed suddenly at $Q_{min\ model.}$ = 1200 m³/h (5250 gpm) (Q_{min} = 13,000 m³/h (57,200 gpm)). After development of part load recirculation, noise and vibration are abruptly reduced but then increase once again, reaching their maximum at shutoff. The sudden drop of noise and vibration corresponds to the development of part load recirculation for which the reverse flow zone could be observed clearly. Both the measured and the predicted onset of part load recirculation ($Q_{min\ th.}$ = 13,500 m³/h (59,400 gpm), refer to Figure 6) are added to Figure 11 and are in fairly good agreement. The experimental results for the modified impeller *Configuration 1* are compared with those of *Configuration 0* in Figure 12. For design flow conditions, the vibration level is slightly higher for *Configuration 1*, but is reduced at part load operation conditions. The onset of part load recirculation was observed at $Q_{min\ model}$ = 1010 m³/h (4444 gpm) (Q_{min} = 10,911 m³/h (48,000 gpm)), which is in good agreement with the prediction ($Q_{min\ th.}$ = 10,980 m³/h (48,310 gpm), refer to

Figure 8). Assessing the noise and vibration level for the desired part load operation point (Q = 13,500 m^3/h (59,400 gpm), Q_{model} = 1250 m^3/h (5500 gpm)), long-term operation is deemed admissible.

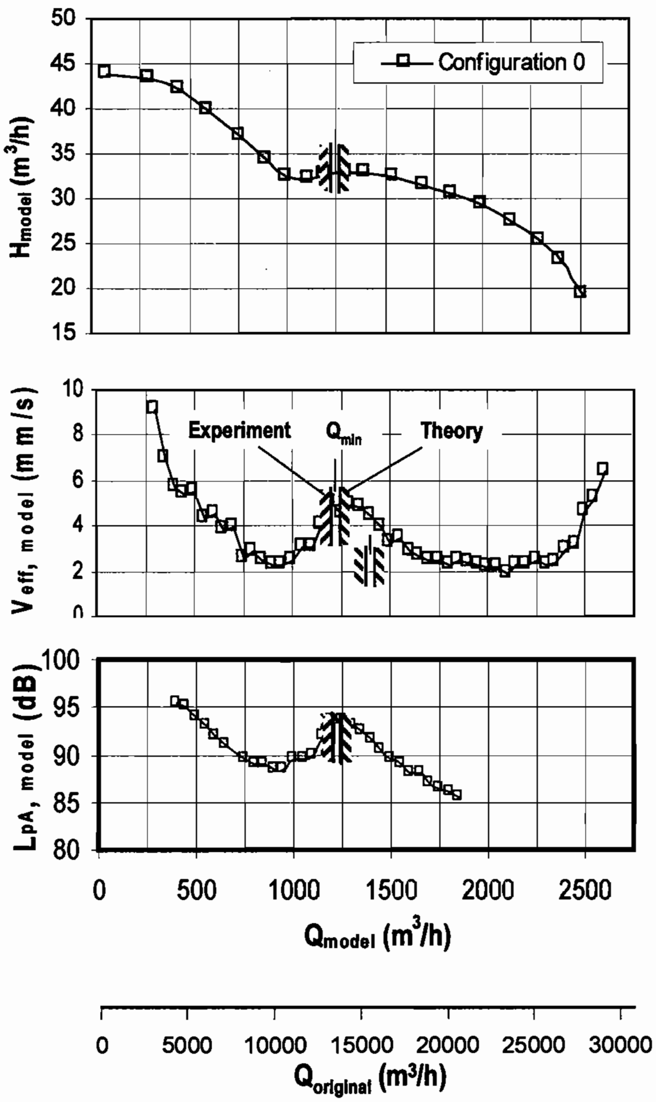

Figure 11. Results of Model Pump Test, Configuration 0.

Besides the reduction of the part load operation limit, the maintenance of the head curve characteristic at the target operation point was required. In Figure 13, the prototype pump performance curves, calculated from the model test results, are compared for both *Configurations 0* and *1*. It is obvious that the head curve of *Configuration 1* is shifted to lower flows, exactly reaching the target point without loss in peak efficiency. In comparison to the theoretical results, the calculated head loss is slightly overpredicted (Figure 5). Moreover, the slope of the measured head curve is slightly changed by cutting down the impeller (Figure 13).

The results of the model pump tests showed clearly that the above defined hydraulic goal is reached by cutting down the impeller in the meridional plane according to *Configuration 1*. Finally, all nine pumps in the plant were changed by cutting down the existing impellers and adapting new wear rings, and were tested afterwards. The results of vibration measurements at site are presented in Figure 14, compared with the model test results obtained for *Configuration 0* (Figure 11, v_{eff} versus Q). For the original design (*Configuration 0*) the characteristics of model and site test results are similar, even if the base level is higher onsite.

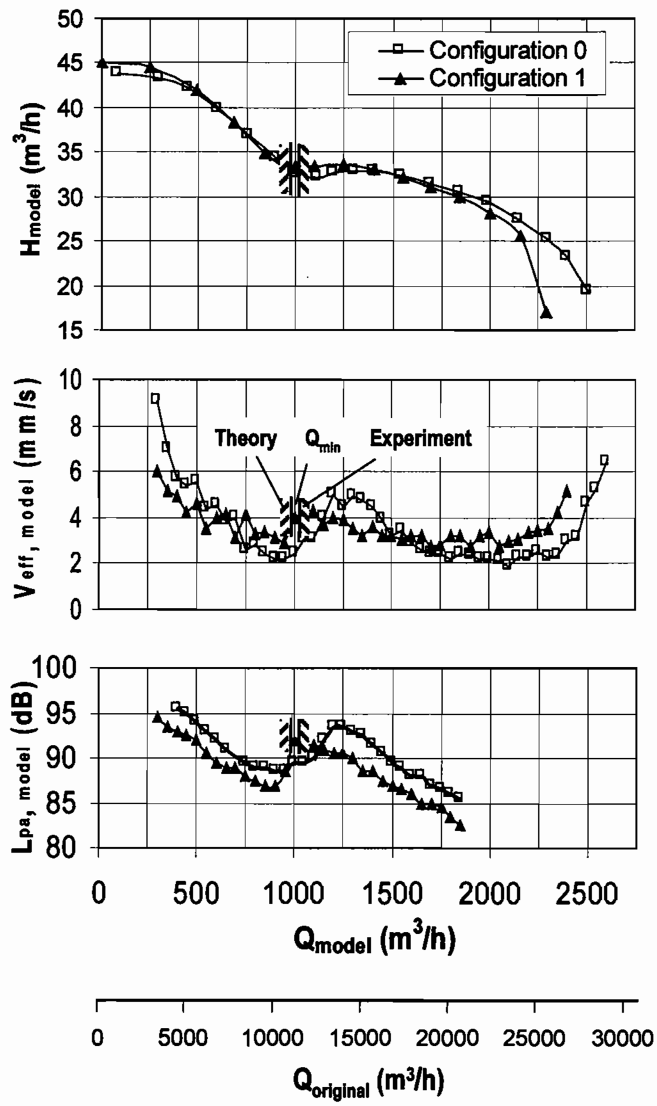

Figure 12. Results of Model Pump Test, Configuration 0 and Configuration 1.

By adapting the new design (*Configuration 1* – Site test) the onset of part load recirculation was successfully shifted to lower flows, forming low vibration velocities in the desired flow range.

CONCLUSION

In the present paper, the adaptation of existing mixed flow impeller pumps to extreme part load conditions by turning down the impeller in the meridional plane is described. The part load operation limit QLTO min was to be reduced by about $\Delta Q/Q_{LTO\ min} \approx 15$ percent, while the head curve characteristic at target operation point was to be maintained. The influence of different impeller shapes on the head curve characteristic and the development of the part load recirculation zone are investigated theoretically. The flow field at the impeller inlet, calculated by means of a 3D-Euler code, is evaluated and assessed for different flowrates. The appearance of reverse flow in the inlet meridional velocity profile is taken as an indication of part load recirculation onset. It was found that a meridional impeller shape with reduced inlet and unchanged outlet diameter fulfils the requirements defined above. The theoretical investigations are accompanied by model pump tests. Besides the head curve and efficiency characteristics, the noise and vibration levels were measured and assessed for the different impeller shapes. In particular, the vibrations show a distinct characteristic that allows a very precise determination of the limiting flowrate.

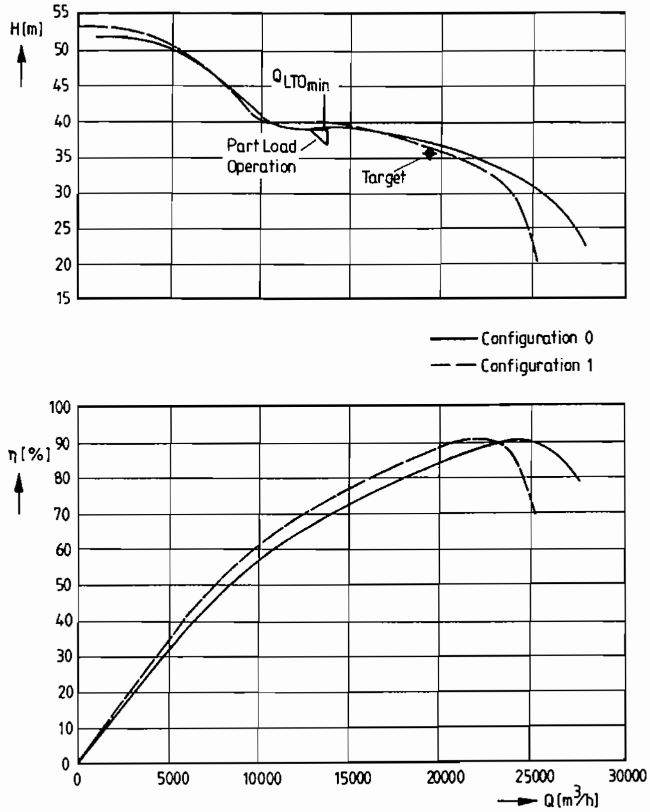

Figure 13. Comparison of Head Curve Characteristics.

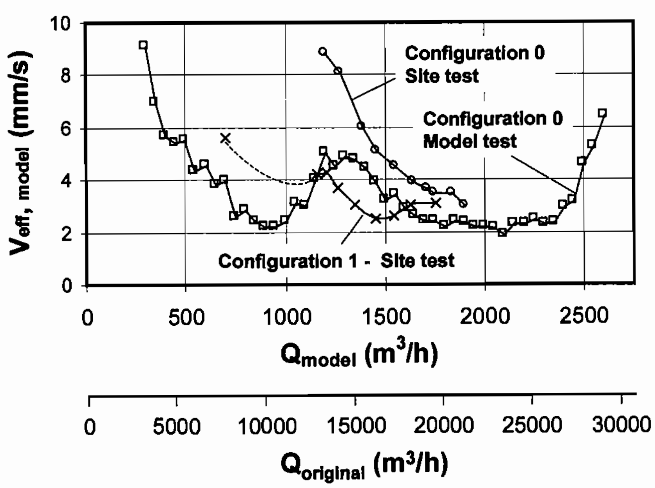

Figure 14. Vibration Measurements, Comparison of Site and Model Test Results.

Comparing the theoretical and experimental results, good agreement can be claimed. The onset of the part load recirculation is predicted especially well.

NOMENCLATURE

c_m	=	Meridional velocity
H	=	Head
DN	=	Discharge diameter
D_1	=	Impeller inlet diameter
D_2	=	Impeller outlet diameter
L_{pA}	=	Noise pressure
m	=	Model scale
n	=	Speed
n_q	=	Specific speed (SI units)
n_s	=	Specific speed (US units)
P	=	Power
Q	=	Flowrate
Q_{min}	=	Onset of part load recirculation
$Q_{LTO\ min}$	=	Minimum flowrate for long term operation
s	=	Tip clearance
v_{eff}	=	vibration velocity
η	=	Efficiency

Subscripts

BEP	=	Best efficiency point
DES	=	Design point
LTO	=	Long term operation
th.	=	Theory

REFERENCES

Cumpsty, N. A., 1989, "Compressor Aerodynamics," England: Longman Scientific & Technical.

Goto, A., 1993, "Suppression of Mixed-Flow Pump Instability and Surge by the Active Alteration of Impeller Secondary Flows," ASME Paper 93-GT-298.

Sarkar, S., 1992, "Performance Study of a Mixed Flow Impeller Covering an Unsteady Flow Field," Journal of Power and Energy, *206*, pp. 83-93.

Stepanoff, A. J, 1965, "Pumps and Blowers," New York, New York: John Wiley and Sons.

Tanaka, T., 1980, "An Experimental Study of Backflow Phenomena in a High Specific Speed Propeller Pump," ASME Conference, New Orleans, Louisiana.

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