EXPERIMENTAL INVESTIGATIONS IN RESPECT TO THE RELEVANCE OF SUCTION SPECIFIC SPEED FOR THE PERFORMANCE AND RELIABILITY OF CENTRIFUGAL PUMPS

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ABSTRACT

Recently, several recommendations have been proclaimed concerning limitation of the maximum suction specific speeds (n_{ss}) of centrifugal pumps. These recommendations result from statistical evaluations of pump failures that seem to indicate a correlation between n_{ss} and the failure probability. However, this correlation has not ben proven by systematic tests applying scientific research methods. Therefore, a corresponding research project was initiated by the Association of German Pump Manufacturers and carried out at a university laboratory with the capability for research on centrifugal pumps.

The aim of the project was to investigate possible effects of high values of nss on the operational behavior of centrifugal pumps, especially on those measurable dynamic quantities that can serve as indicators for the risk of failures. For the investigations, three standard pumps of different specific speed and size were selected that were designed and manufactured by three different German manufacturers. All these pumps were alternatively equipped with two impellers. One of them was designed for high n_{ss} -values, and one for n_{ss} values below the recommended limit. Two of the pumps were run at 3480 rpm, and one pump was run at 3100 rpm.

INTRODUCTION

For many applications of centrifugal pumps, the plant characteristics imply low values of $NPSH_A$. Increasing these $NPSH_A$-values (by, e.g., higher elevation or higher pressurization of suction vessels, lower elevation of pumps and/or reduced pressure losses in the suction piping) would lead to increased installation costs. To avoid the necessity to increase $NPSH_A$, pumps having sufficiently low values of $NPSH_R$ are needed for those applications. $NPSH_R$-values of centrifugal pumps are usually based on a cavitation condition that causes a defined head drop. Three percent head drop is a widely used criterion that defines the values of $NPSH_{R3\ percent}$. Depending on the individual circumstances (pump type, pump size and pump speed, liquid properties, impeller material), safety margins between $NPSH_A$ and $NPSH_{R3\ percent}$ are recommended to avoid damage of pump parts (especially of the impeller) by cavitation erosion. For an equal amount of safety margins, the $NPSH_{R3\ percent}$-values of pumps which can be operated without risk of cavitation erosion are dictated by the $NPSH_A$-values of the respective plant.

At a given flowrate and speed, the $NPSH_{R3\ percent}$-value depends on geometrical details of the pump, especially on the diameter and blade geometry at impeller inlet. Pumps which were designed for different values of flowrate (at bep) and speed can be compared (in respect to their $NPSH_{R3\ percent}$-values) by their individual values of the suction specific speed n_{ss}. The suction specific speed is calculated from the speed n, the flowrate Q (at bep) and the $NPSH_{R3\ percent}$-value (at bep) according to the equation

$$n_{ss} = n \,.\, Q^{0.5} / NPSH_{R3\ percent}^{\ 0.75} \quad (1)$$

For double suction pumps, half the pump flowrate has to be inserted as Q in Equation (1). The numerical value of n_{ss} depends on the units used for Q and NPSH. While in Europe the metric units m^3/s and m are commonly used, gpm and ft are the usual units in USA. (For better distinguishing the values, the denomination n_{qs} instead of n_{ss} is to be preferred in combination with the use of metric units.)

For pumps of conventional impeller design, the n_{ss}-values range from 8,000 to 10,000, approximately. By special impeller design, e.g., by enlarged suction eye diameter and/or blade inlet angles corresponding to $Q_{shockless} > Q_{bep}$, remarkably higher values of n_{ss} are attainable. On the other hand, especially in the USA, it is often recommended by pump users, consultants, and institutions to restrict the range of n_{ss} to values below 11,000 (in spite of the better cavitation performance of high n_{ss} pumps). The reasoning is that the risk of mechanical failures (of bearings, shaft sealing, or impeller) seems to increase distinctly for pumps having n_{ss}-values of 11,000 or above. Hallam [1] presented results of a statistical evaluation of pump failures that indicated an increased pump failure probability for $n_{ss} \geq 11{,}000$. As an explanation, he argued that for pumps of such impeller design, excessive suction recirculation may exist even for moderate part load operation (referred to Q_{bep}), when this operation bears in fact a strong part load condition in respect to $Q_{shockless}$. This argument implies direct relations between high n_{ss}-values and an enlarged ratio of $Q_{shockless} / Q_{bep}$ and of the latter with an increased risk of mechanical failures.

In Germany, no statistical study leading to results comparable to those reported by Hallam [1] has been reported. However, the questions concerning the role of n_{ss} as an indicator or evaluation criterion in respect to the operational safety and reliability of pumps was extensively discussed among German (and also other European) pump manufacturers. From these discussions at German pump conferences and within working groups of the German pump manufacturers, it resulted that the statistical correlation of n_{ss} and failure probability reported by Hallam [1] was not confirmed by the experience of German pump manufacturers. They delivered a large number of pumps having values of $n_{ss} > 11{,}000$, but they did not get feedback from their purchasers/users indicating an increased rate of failures of these pumps. Stoffel and Hergt [2] indeed showed by several examples that the suction specific speed cannot be taken as the only indicator for the reliability of pumps. They demonstrated that high values of n_{ss} can be reached by various design features (especially concerning the impeller inlet geometry) that affect, in a different way, the necessary NPSH-margins and the onset and extent of inlet recirculation. From this study, it can be concluded that the reliability of a high n_{ss} pump is determined by its individual design features and not simply by its numerical value of n_{ss}. To give support to their general experience with high n_{ss} pumps and to the argumentation presented by Stoffel and Hergt [2], a group of German pump manufacturers who are organized within the VDMA (German Association of Machinery and Plant Manufacturers) gave the order to the authors' laboratory to perform a special experimental investigation on a systematic and scientific basis. With this investigation, it was not intended to run long duration tests to create results on the life duration of pumps or their parts. Rather, it was the objective of the project to make comparisons between pumps having medium and high n_{ss}-values, respectively, in respect to measurable quantities of vibrations and fluctuations that give indication on the mechanical behavior of pumps and, thereby, on the risk of failures of pump parts (as bearings, shaft seals, etc.) by dynamic loading. For the measurements, three end-suction process pumps of different German manufactures and of different specific speed were chosen and were equipped alternatively with impellers for medium or high (> 11,000) values of n_{ss}. The selection of the individual pumps was decided by a working group according to various considerations. The type (single-stage, single-suction) and the range of performance data (including power input) of the pumps selected for the investigation can be taken as representative for a majority of process pump applications. The hydraulic and mechanical design features were considered to be sufficiently different among the selected pumps to cover the range found for this type of pumps made by the various German manufacturers. In view of the purpose of the investigation (i.e., direct comparison of vibration and pulsation quantities for high *vs* medium n_{ss} pump variants having identical casing, shaft, sealing, and bearing), a number of three different test pumps of the same type (each of them existing in two impeller variants) was thought by the working group to be sufficient. From the methodological approach of the study, it can be expected that the results found are neither strongly restricted to the pump type chosen for the investigation nor to the range of power input covered by the selected test pumps. Rather, the comparative character of the results (in respect to the effects of high *vs* medium n_{ss}) will allow to take them also as clear indications concerning the role of n_{ss} in respect to the dynamic pump behavior for pumps having higher power input then the test pumps and/or for other pump types (e.g. multistage or double-suction).

Tested Pumps and Measured Quantities

Cross sections of the three test pumps are given in the APPENDIX. In Tables 1, 2, 3, 4, 5, and 6 a survey of the tested pumps is given with some geometrical and hydraulic data. Pumps 1 and 3 were run at a rotational speed of 3480 rpm, while pump 2 was run at 3100 rpm (because of the maximum power output of 80 kW of the electrical driver). Also included in the tables are data for the different pumps and variant values of $Q_{shockless}$ (flowrate at shockless impeller inflow) and of $Q_{onset\ suc.rec.}$ (flowrate at onset of suction recirculation) related to Q_{bep} (flowrate at best efficiency). $Q_{shockless}$ is calculated from the impeller inlet geometry, assuming uniform meridional velocity at the blade leading edge, and taking into account the blockage affected by the blade thickness. It was possible to determine $Q_{onset\ suc.rec.}$ experimentally only for pump 1 (for the reasons, see following text). For information, also values of $Q_{onset\ suc.rec.}$ are presented in the tables that were estimated by Fraser [3].

Besides the characteristic curves, the NPSH-values corresponding to the conditions of cavitation inception as well as to a head drop of one and three percent, respectively, were determined. At flowrates of 40, 50, 75, 100, and 125 percent of the nominal flowrate, four dynamic quantities were measured at four cavitation conditions (no cavitation, cavitation inception, one and three percent head drop):

- the shaft deflection at locations near to the shaft sealing
- the pressure pulsations at inlet and outlet of the pump
- the vibrations at the bearing casing
- the dynamic (AC-) component of the axial thrust

Table 1. Data of the Tested Pump 1 (SI-units).

pump		#1, medium nss	#1, high nss
suction specific speed	[min^{-1}]	190	237
nominal flow rate	[m^3/h]	42	42
nominal head	[m]	113	114
rotational speed	[min^{-1}]	3480	3480
specific speed	[min^{-1}]	11	11
suction branch diameter	[m]	0.080	0.080
outlet branch diameter	[m]	0.040	0.040
diameter of impeller eye	[m]	0.076	0.082
impeller outlet diameter	[m]	0.259	0.259
blade angle at inlet β_{s1a}	[°]	24	17
No. of blades		3	3
Velocity v_{eye} at bep	[m/s]	2.57	2.21
$Q_{shockless}/Q_{bep}$	[-]	1.12*	0.98*
meas. onset of recirculation	[m^3/h]	47	44
calc. onset of recirculation	[m^3/h]	38.0	41.5
meas. $Q_{onset\ suc.\ rec.}/Q_{bep}$	[-]	1.12	1.05
calc. $Q_{onset\ suc.\ rec.}/Q_{bep}$	[-]	0.905	0.987
shaft seal		mechanical seal	mechanical seal
balancing of axial thrust		balancing holes and 2nd sealing gap	balancing holes and 2nd sealing gap

* Values of $Q_{shockless}/Q_{bep}$ given by manufacturer.

Table 2. Data of the Tested Pump 1 (US-units).

pump		#1, medium nss	#1, high nss
suction specific speed	[-]	9814	12241
nominal flow rate	[gpm]	185	185
nominal head	[ft]	371	371
rotational speed	[rpm]	3480	3480
specific speed	[rpm]	568	568
suction branch diameter	[in]	3.15	3.15
outlet branch diameter	[in]	1.57	1.57
diameter of impeller eye	[in]	2.99	3.23
impeller outlet diameter	[in]	10.2	10.2
blade angle at inlet β_{s1a}	[°]	24	17
No. of blades		3	3
Velocity v_{eye} at bep	[ft/s]	8.43	7.25
$Q_{shockless}/Q_{bep}$	[-]	1.12*	0.98*
meas. onset of recirculation	[gpm]	207	194
calc. onset of recirculation	[gpm]	167.3	182.7
meas. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	1.12	1.05
calc. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	0.905	0.987
shaft seal		mechanical seal	mechanical seal
balancing of axial thrust		balancing holes and 2nd sealing gap	balancing holes and 2nd sealing gap

* Values of $Q_{shockless}/Q_{bep}$ given by manufacturer.

Table 3. Data of the Tested Pump 2 (SI-units).

pump		#2, medium nss	#2, high nss
suction specific speed	[min⁻¹]	196	268
nominal flow rate	[m³/h]	157	157
nominal head	[m]	93	91
rotational speed	[min⁻¹]	3100	3100
specific speed	[min⁻¹]	22	22
suction branch diameter	[m]	0.100	0.100
outlet branch diameter	[m]	0.080	0.080
diameter of impeller eye	[m]	0.100	0.120
impeller outlet diameter	[m]	0.260	0.260
blade angle at inlet β_{s1a}	[°]	26	17.5
No. of blades		6	6
Velocity v_{eye} at bep	[m/s]	5.52	3.86
$Q_{shockless}/Q_{bep}$	[-]	1.31	1.16
meas. onset of recirculation	[m³/h]	not measured	not measured
calc. onset of recirculation	[m³/h]	78.9	118.2
meas. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	-	-
calc. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	0.503	0.753
shaft seal		stuffing box	stuffing box
balancing of axial thrust		balancing holes and 2nd sealing gap	balancing holes and 2nd sealing gap

Table 4. Data of the Tested Pump 2 (US-units).

pump		#2, medium nss	#2, high nss
suction specific speed	[-]	10123	13842
nominal flow rate	[gpm]	691	691
nominal head	[ft]	302	302
rotational speed	[rpm]	3100	3100
specific speed	[rpm]	1136	1136
suction branch diameter	[in]	3.94	3.94
outlet branch diameter	[in]	3.15	3.15
diameter of impeller eye	[in]	3.94	4.72
impeller outlet diameter	[in]	10.24	10.24
blade angle at inlet β_{s1a}	[°]	26	17.5
No. of blades		6	6
Velocity v_{eye} at bep	[ft/s]	18.11	12.66
$Q_{shockless}/Q_{bep}$	[-]	1.31	1.16
meas. onset of recirculation	[gpm]	not measured	not measured
calc. onset of recirculation	[gpm]	347.4	520.4
meas. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	-	-
calc. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	0.503	0.753
shaft seal		stuffing box	stuffing box
balancing of axial thrust		balancing holes and 2nd sealing gap	balancing holes and 2nd sealing gap

Table 5. Data of the Tested Pump 3 (SI-units).

pump		#3, medium nss	#3, high nss
suction specific speed	[min⁻¹]	173	232
nominal flow rate	[m³/h]	192	192
nominal head	[m]	50.4	50.4
rotational speed	[min⁻¹]	3480	3480
specific speed	[min⁻¹]	42.5	42.5
suction branch diameter	[m]	0.100	0.100
outlet branch diameter	[m]	0.100	0.100
diameter of impeller eye	[m]	0.110	0.117
impeller outlet diameter	[m]	0.188	0.188
blade angle at inlet β_{s1a}	[°]	19	17
No. of blades		6	6
Velocity v_{eye} at bep	[m/s]	5.61	4.96
$Q_{shockless}/Q_{bep}$	[-]	0.94	1.12
meas. onset of recirculation	[m³/h]	not measured	not measured
calc. onset of recirculation	[m³/h]	106.3	120.5
meas. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	-	-
calc. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	0.554	0.627
shaft seal		double mechanical seal	double mechanical seal
balancing of axial thrust		none	none

Table 6. Data of the Tested Pump 3 (US-units).

pump		#3, medium nss	#3, high nss
suction specific speed	[-]	8935	11983
nominal flow rate	[gpm]	845	845
nominal head	[ft]	164	164
rotational speed	[rpm]	3480	3480
specific speed	[rpm]	2195	2195
suction branch diameter	[in]	3.94	3.94
outlet branch diameter	[in]	3.94	3.94
diameter of impeller eye	[in]	4.33	4.61
impeller outlet diameter	[in]	7.40	7.40
blade angle at inlet β_{s1a}	[°]	19	17
No. of blades		6	6
Velocity v_{eye} at bep	[ft/s]	18.41	16.27
$Q_{shockless}/Q_{bep}$	[-]	0.94	1.12
meas. onset of recirculation	[gpm]	not measured	not measured
calc. onset of recirculation	[gpm]	468.0	530.5
meas. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	-	-
calc. $Q_{onset\ suc.\ rec}/Q_{bep}$	[-]	0.554	0.627
shaft seal		double mechanical seal	double mechanical seal
balancing of axial thrust		none	none

MEASURING EQUIPMENT

Each pump was mounted in a closed circuit test-section filled with water of appr. 20°C (68°F). For the determination of the characteristic curves served a magnetic flowmeter (for the flowrate), static pressure transducers at inlet and outlet of the pump (for determining NPSH and the pump head H), a temperature transducer (for the water temperature), a sensor for the rotational speed of the pump and a torque-dynamometer installed between the pump and driver shafts (for the pump torque).

A pneumatically activated throttling valve was used to adjust the various values of the flowrate. To adjust the NPSH-values, the pressure of the air volume above the water level in a large vessel was varied by pressurizing or evacuating the air.

For the measurement of the fluctuating part of the pressure at inlet and outlet, the pumps were equipped with piezoelectric pressure transducers. The deflection of the shaft was measured by two noncontacting distance sensors (located at an angular distance of 90 degrees). A piezoelectric acceleration sensor served for the determination of the casing vibrations. The device for measuring the axial thrust was specially designed and manufactured for the investigation. The normal axial roller bearing of the test pumps was replaced by a radial roller bearing when the axial thrust was measured, and the outer ring casing of a special roller bearing was connected to a special support that is elastically deformed under the axial bearing load. The support element was carefully dimensioned and manufactured. The deformation itself is measured by strain gauges.

The device was calibrated by applying known weight forces. The measuring device for the axial thrust is shown in Figure 1.

(During the measurement of all other quantities, this measuring device was inactive, and the pump shaft was supported in its normal bearings.) The location of the different sensors is shown schematically in Figure 2.

All measured data were digitized and stored by the means of a PC based data acquisition system. Especially for the dynamic quantities, a high-frequency transient recorder that was controlled by the PC served for data acquisition.

EVALUATION OF THE MEASURED QUANTITIES

The characteristic curves (Pump head H and pump efficiency η *vs* relative flowrate) were directly computed from the measured hydraulic quantities. The NPSH-values corresponding to defined head drop conditions were determined by special head drop test runs. The condition of cavitation inception was acoustically determined with the aid of the piezoelectric pressure transducer located at the pump inlet. The output signal was high-pass filtered and monitored on the screen of a digital oscilloscope. Cavitation inception was detectable by the appearance of characteristic high-frequency peaks of the pressure signal.

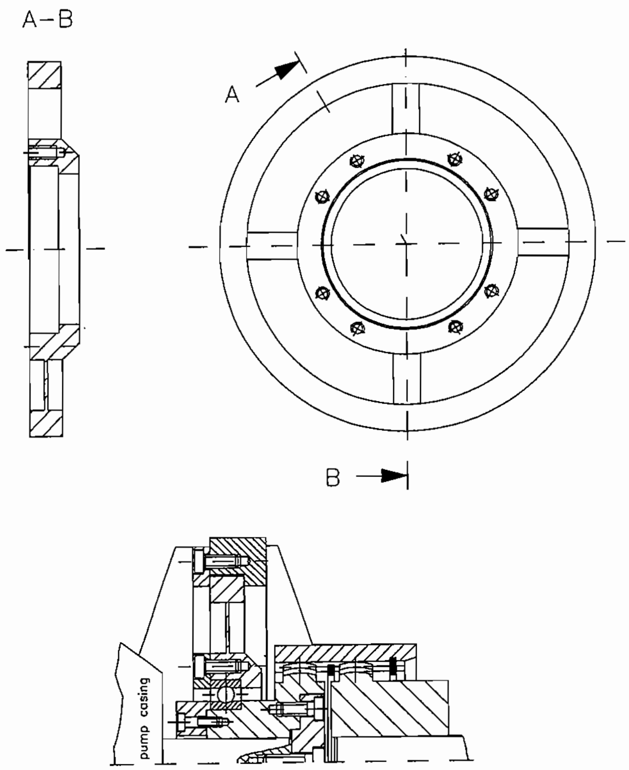


Figure 1. Measuring Device of the Axial Thrust

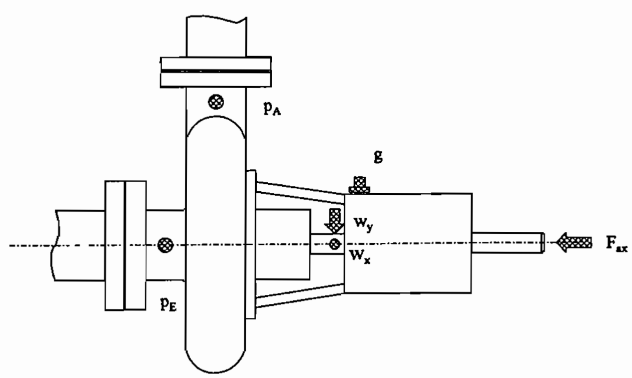


Figure 2. Location of the Dynamic Sensors.

For each pump variant, the predetermined NPSH-values for the four different cavitation conditions combined with the five values of the relative flowrate defined a set of twenty operating conditions for which the dynamic quantities were determined individually.

After recording the dynamic signals in the time domain, they were transformed by FFT and analyzed in the frequency domain to identify characteristic frequencies and corresponding amplitudes for the individual quantities. For several test runs, enlarged amplitudes were observed at twice the frequency of rotation ($2*f_n = 116$ Hz corresponding to 3480 rpm), which were strongly dependent on how precise the pump and driver shafts had been aligned when assembling the pump and coupling. Small deviations from the correct alignment caused a high amplitude at $2*f_n$. Because there is no hydraulic effect responsible for this frequency, its contribution to mean values calculated from the measured signals was suppressed. The amplitude spectra in the frequency domain (after elimination of the amplitude at $2*f_n$) were then taken to calculate a mean value for comparisons. For the vibration velocity of the casing, the resulting value was calculated according to the formula given in ISO 2372.

$$v_{RMS} = \sqrt{\frac{1}{2}\sum_{i=1}^{n}\left(\frac{a_i}{\omega_i}\right)^2} \tag{2}$$

where v_{RMS} is the RMS-value of the vibration velocity, a_i is the amplitude of the vibration acceleration corresponding to the frequency ω_i.

For the other quantities, RMS-values were calculated in a similar way according to the equation

$$x_{RMS} = \sqrt{\frac{1}{n}\cdot\sum_{i=1}^{n} x_i^2} \tag{3}$$

where x_{RMS} is the RMS-value of the quantity x and x_i are its amplitudes at the individual frequencies.

The output signals of the axial thrust measuring device were also influenced by the alignment conditions and by small deviations of the shaft geometry from exact concentricity. Regarding the frequency spectra, it was obvious that a peak at the frequency of rotation, f_n was responsible for differing RMS-values after mounting the pump in different ways. Therefore, the amplitude at f_n was also suppressed when calculating the RMS-values of the axial thrust to eliminate these influences that are not related to hydraulic effects and not connected with the individual impellers.

EXPERIMENTAL RESULTS

Characteristic curves

One of the basic requirements for all comparisons was that the characteristic curves of both impellers should be nearly identical for both variants of each pump . As an example for the characteristic curves, Figures 3 and 4 show for both variants of pump 1 the head H and the efficiency η *vs* the relative flowrate q, respectively. It is obvious that the difference between both variants is very small in this respect. The same holds true for pumps 2 and 3.

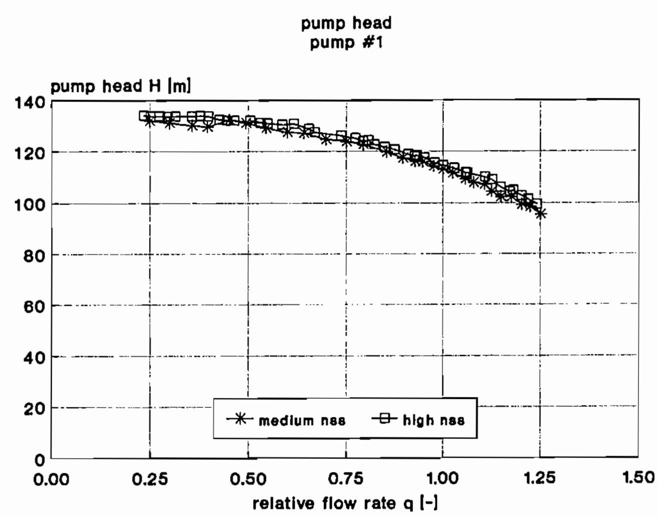


Figure 3. Pump Head vs *Relative Flowrate, Pump 1.*

Cavitation Performance

Since the $NPSH_R$-values for defined head-drop are most important in respect to the actual n_{ss}-values of the tested pump variants and to the relevant test conditions as well, they are presented for all pumps and variants in the Figures 5, 6, and 7. Curves corresponding to one and three percent head-drop, respectively, are plotted *vs* the relative flowrate q.

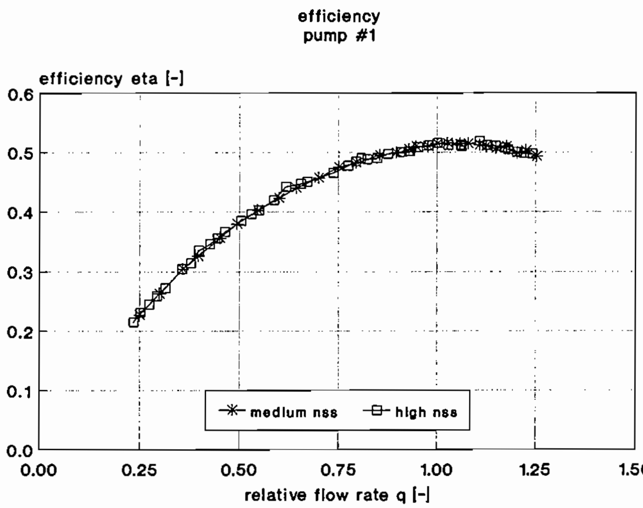


Figure 4. Efficiency vs *Relative Flowrate, Pump 1.*

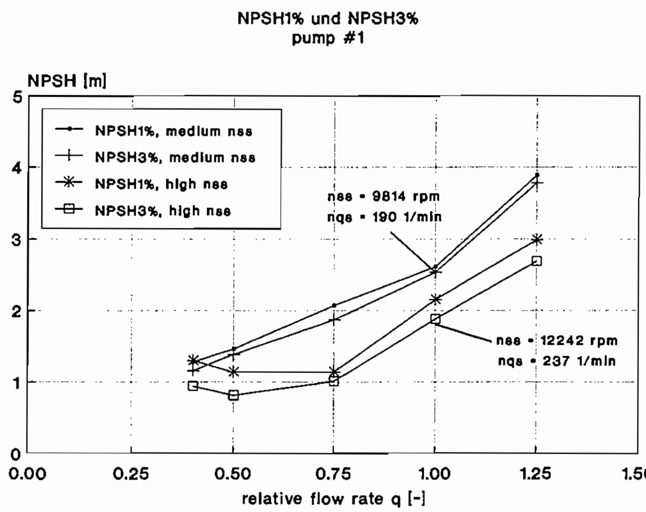


Figure 5. $NPSH_{1\ percent}$ and $NPSH_{3\ percent}$ vs *Relative Flowrate, Pump 1.*

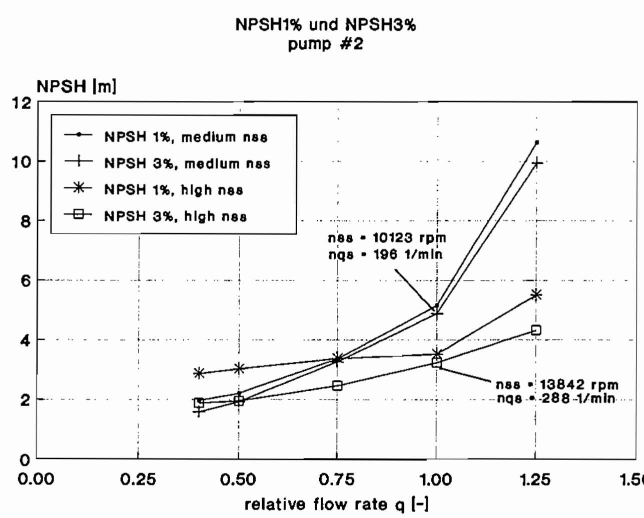


Figure 6. $NPSH_{1\ percent}$ and $NPSH_{3\ percent}$ vs *Relative Flowrate, Pump 2.*

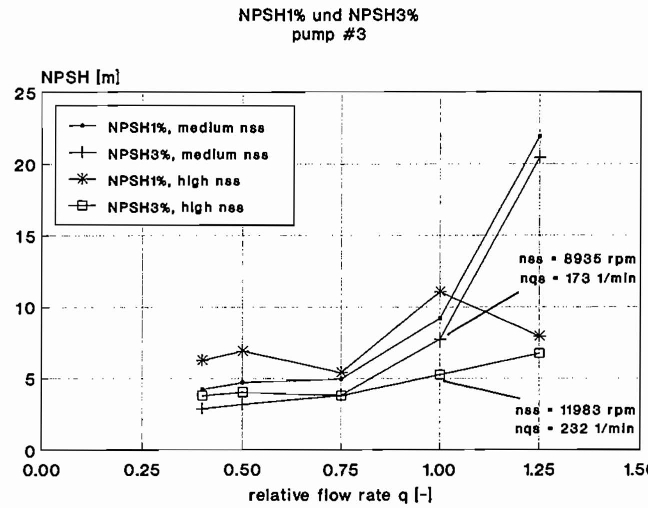


Figure 7. $NPSH_{1\ percent}$ and $NPSH_{3\ percent}$ vs *Relative Flowrate, Pump 3.*

Onset of Suction Recirculation

Only for pump 1 it was possible to detect the onset of suction recirculation and to determine, therefrom, the corresponding value of the flowrate. For this pump, the onset of suction recirculation was detected from the typical rise of the static wall pressure at a location shortly upstream of the inlet eye (which results from the swirl of the recirculating flow). The onset of suction recirculation occurs at a flowrate of 1.1 times nominal flow rate for the medium n_{ss} variant of pump 1, while it occurs at 1.05 times nominal flowrate (i.e., at an even smaller flowrate) for the high n_{ss} variant of pump 1. For the medium n_{ss} variant, the measured value is nearly 20 percent higher than the calculated one (according to Fraser [3]), while the difference between the measured and calculated values is even smaller in the case of the high n_{ss} variant (see Tables 1 and 2). The detection of suction recirculation was not possible for pumps 2 and 3 since they were equipped with flow straighteners in the suction nozzle.

Dynamic Quantities

Because of the large amount of tests, only a few characteristic examples will be given herein of the experimental results for each dynamic quantity measured at one of the tested pumps. A survey is presented to give an overview of the range of values found for the individual pump variants.

Shaft Deflection

The shaft deflection of pump 3 is presented in Figure 8 and 9. The medium n_{ss} variant (Figure 8) shows dynamic amplitudes of the shaft deflection in the range of two to five microns, depending on the relative flowrate and the cavitation condition. The highest values were measured at NPSH = $NPSH_{R_{3\ percent}}$. For the high n_{ss} variant (Figure 9), values were found within nearly the same range. Only the results at a relative flowrate $q > 1.0$ show slightly higher values at less strong cavitation conditions.

Pressure Pulsations

To demonstrate effects concerning the pressure pulsations at pump inlet and outlet, pump 2 is chosen as the example. All values of the pressure pulsation are related to the corresponding pump head at the same relative flowrate.

At pump inlet, for the cavitation conditions of one and three percent head drop so many vapor bubbles exist in the impeller eye, that the pressure fluctuations generated in the impeller are strongly attenuated along the way of transmission to the pressure sensor. This effect can be seen from Figures 10 and 11. The curves for no cavitation seem to be almost identical for both pump variants. For incipient cavitation in the high n_{ss} variant, the values drop down in

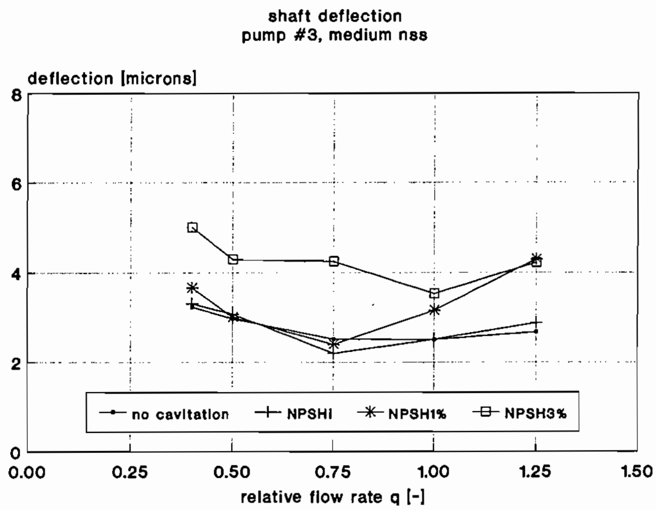


Figure 8. Shaft Deflection of Pump 3, Medium n_{ss} Variant.

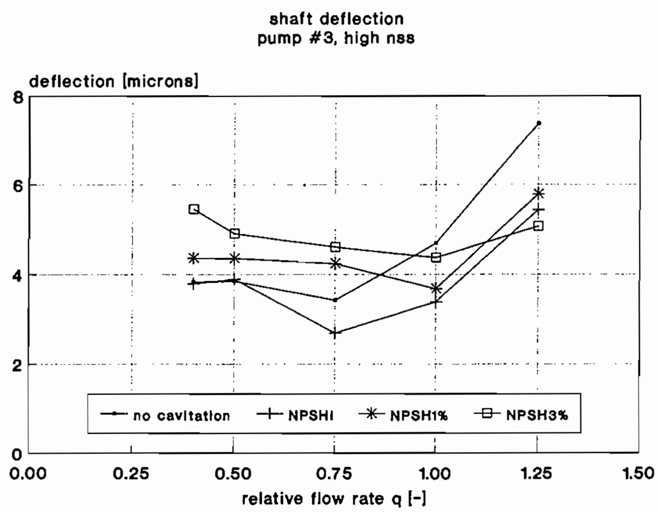


Figure 9. Shaft Deflection of Pump 3, High n_{ss} Variant.

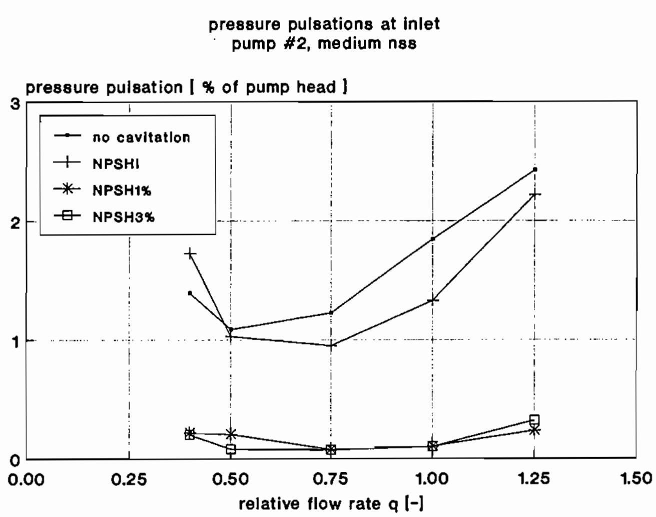


Figure 10. Pressure Pulsations at Pump Inlet of Pump 2, Medium n_{ss} Variant.

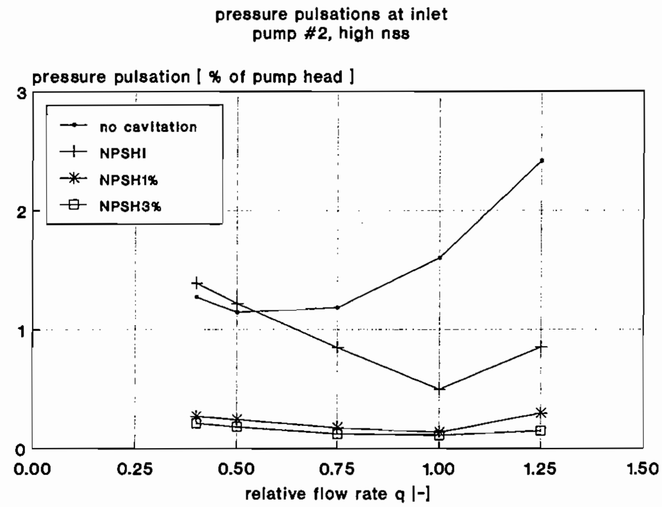


Figure 11. Pressure Pulsations at Pump Inlet of Pump 2, High n_{ss} Variant.

the range $q \geq 0.75$ towards the curves that correspond to more developed cavitation conditions. This drop might be the effect of only a small variation in NPSH, so that at these test points, the actual NPSH-value may have been slightly below $NPSH_{R_i}$ (the value for acoustical cavitation inception) and a small amount of vapor bubbles were already existing and attenuating the transmission of pressure fluctuations from the impeller to the sensor.

At the pump outlet, the pressure pulsations of pump 2 chosen as an example were found to be even slightly higher for the medium n_{ss} variant (Figure 12) than for the higher n_{ss} variant (Figure 13), especially at three percent head drop. The latter variant shows a maximum of pressure pulsations at extreme part load while the former shows the highest values at $q > 1.0$.

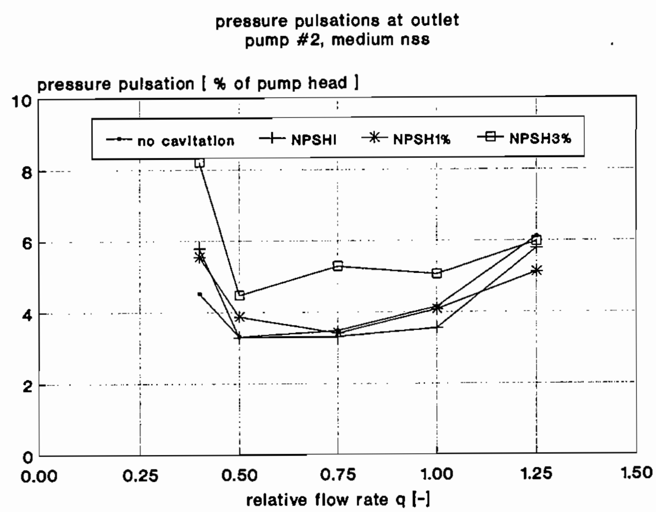


Figure 12. Pressure Pulsations at Pump Outlet of Pump 2, Medium n_{ss} Variant.

Casing Vibration

Casing vibration velocities are shown for pump 1 in Figures 14 and 15. The curves for both variants show no strong dependence on the flowrate and only a slight dependence on the cavitation condition. The condition of no cavitation causes the lowest values of about 2.0 mm/s, the highest value is obtained for the high n_{ss} variant when operated at $NPSH_{R_{3\ percent}}$ and at nominal flowrate. All other values are within the range from 2.0 to 3.0 mm/s.

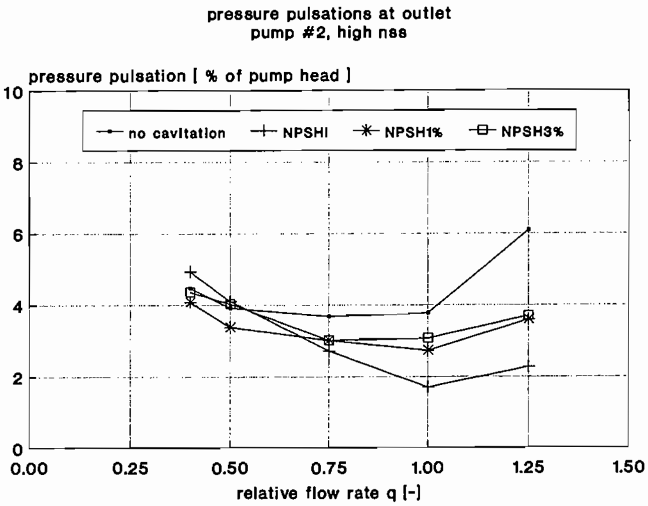


Figure 13. Pressure Pulsations at Pump Outlet of Pump 2, High n_{ss} Variant.

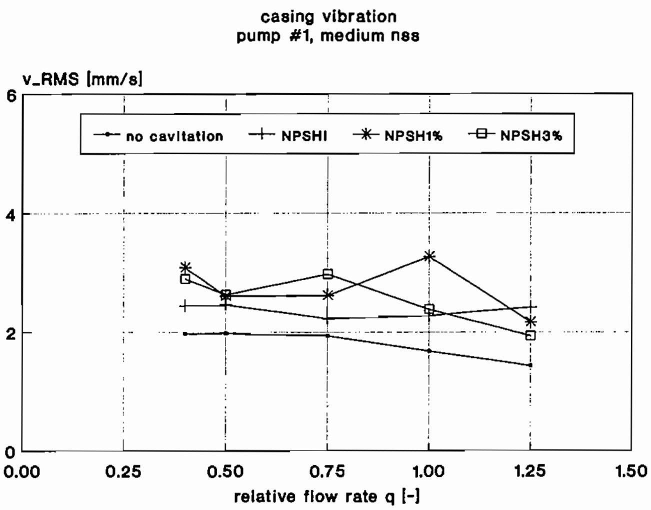


Figure 14. Casing Vibrations of Pump 1, Medium n_{ss} Variant.

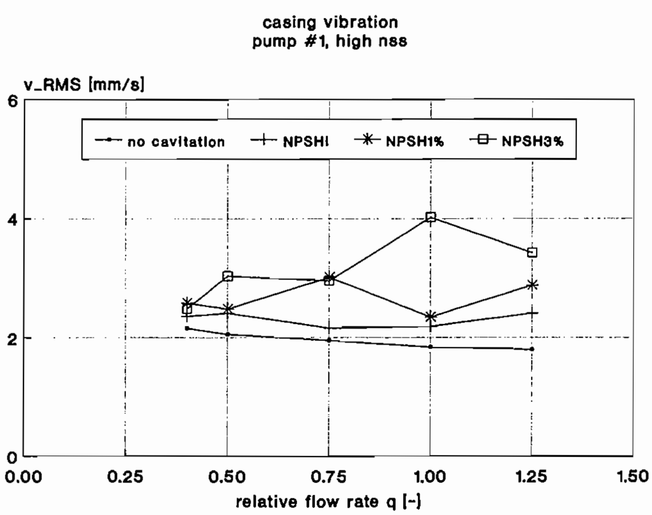


Figure 15. Casing Vibrations of Pump 1, High n_{ss} Variant.

Axial Thrust Fluctuations

Results for this dynamic quantity are given for two of the tested pumps because of their different type of axial thrust balancing. While pump 1 was balanced by the means of a sealing gap and holes on the back shroud, pump 3 was unbalanced.

For the pump with balanced impeller, the medium n_{ss} variant (Figure 16) shows a decrease of the axial thrust fluctuations with increasing flowrate. The values measured for all cavitation conditions are within a very narrow range from 20 to 60 N. For the high n_{ss} variant (Figure 17), almost the same tendency appears, only the fluctuations at $q > 1.0$ increase to slightly higher values.

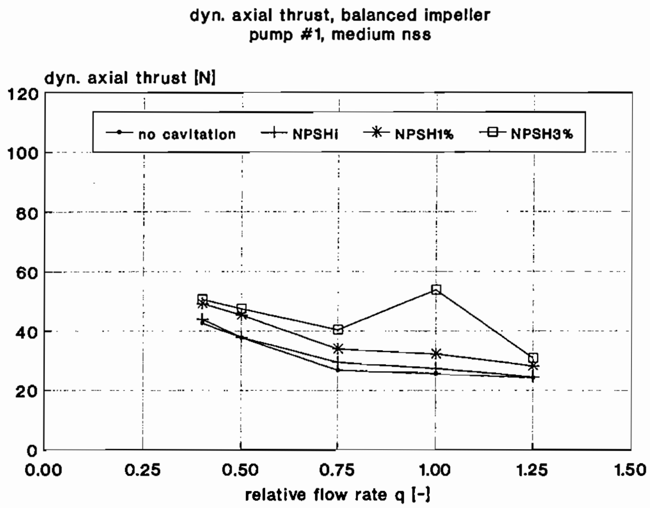


Figure 16. Axial Thrust Fluctuations of Pump 1 (Balanced Impeller), Medium n_{ss} Variant.

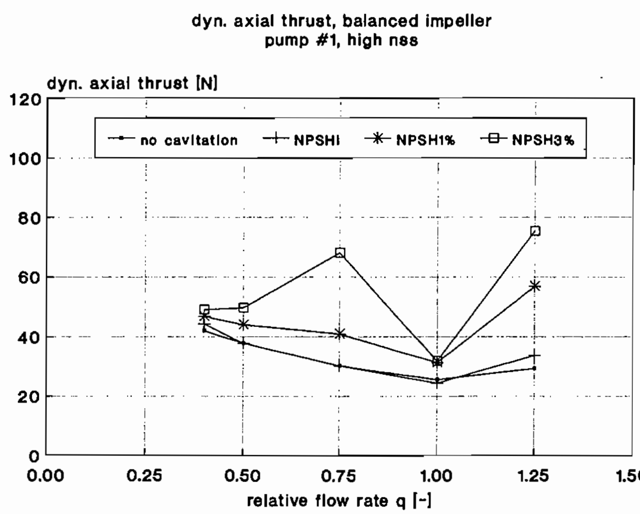


Figure 17. Axial Thrust Fluctuations of Pump 1 (Balanced Impeller), High n_{ss} Variant.

For the pump with unbalanced impeller, the differences in the axial thrust fluctuations between the medium n_{ss} variant (Figure 18) and the high n_{ss} variant (Figure 19) are even smaller compared to the case of the pump with balanced impeller.

Survey of All Results

Because of the large number of results it was only possible to show some examples here that, nevertheless, are representative in respect to the general findings. The lower and upper limits of the range of the measured values are given in Tables 7 and 8 for each quantity and for each tested pump variant. In summarizing these tables, no distinction is made in respect to the specific influences of the relative flowrate or of the cavitation condition. Rather, the table shall serve for a general comparison of the two different variants of each test pump.

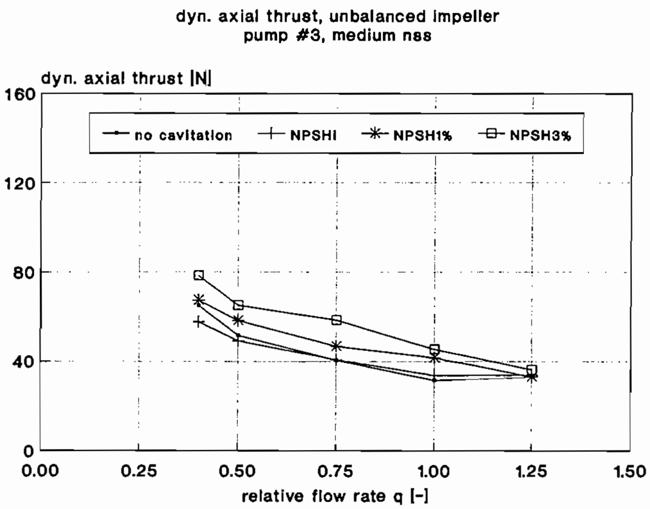


Figure 18. Axial Thrust Fluctuations of Pump 3 (Unbalanced Impeller), Medium n_{ss} Variant.

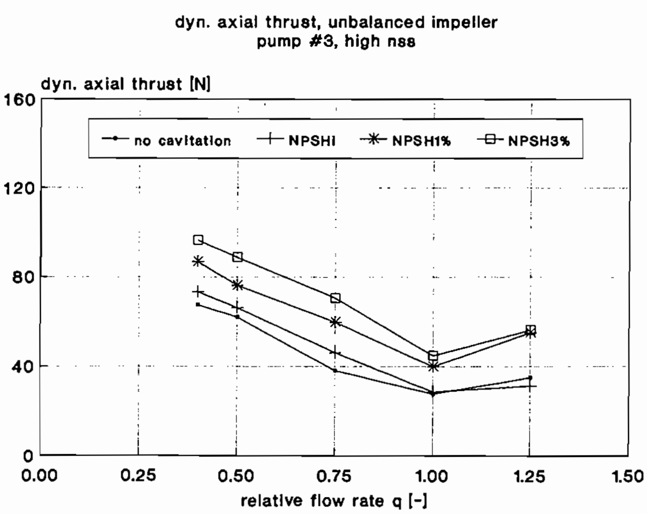


Figure 19. Axial Thrust Fluctuations of Pump 3 (Unbalanced Impeller), High n_{ss} Variant.

Table 7. Survey of the Measured Data (SI-units)

pump	n_{ss}	shaft deflection [μm]	pressure pulsations at pump inlet * [% of the actual head]	pressure pulsations at pump outlet [% of the actual head]	vibration velocity of the casing [mm/s]	dyn. axial thrust [N]
pump #1	medium	8-23	0.45-0.83	2.33-3.97	1.4-3.3	24-54
	high	11-25	0.60-1.97	2.77-6.54	1.8-4.0	24-75
pump #2	medium	38-44	0.95-2.43	3.28-8.22	0.9-1.7	29-126
	high	32-38	0.50-2.41	1.70-6.09	1.1-1.9	30-117
pump #3	medium	2-5	0.84-1.44	1.02-2.14	2.7-10.4	32-78
	high	3-7	0.90-1.59	0.88-2.44	3.1-8.8	28-97

*pressure pulsations at inlet excluding the values at $NPSH_{1\%}$ and $NPSH_{3\%}$ because of too much attenuation by vapour existing in the inlet region

Table 8. Survey of the Measured Data (US-units)

pump	n_{ss}	shaft deflection [mil]	pressure pulsations at pump inlet * [% of the actual head]	pressure pulsations at pump outlet [% of the actual head]	vibration velocity of the casing [in/sec]	dyn. axial thrust [lb(force)]
pump #1	medium	0.315-0.906	0.45-0.83	2.33-3.97	0.055-0.130	5.39-12.14
	high	0.433-0.984	0.60-1.97	2.77-6.54	0.071-0.157	5.40-16.86
pump #2	medium	1.496-1.732	0.95-2.43	3.28-8.22	0.035-0.067	6.52-28.33
	8high	1.260-1.496	0.50-2.41	1.70-6.09	0.043-0.075	6.74-26.30
pump #3	medium	0.078-0.197	0.84-1.44	1.02-2.14	0.106-0.409	7.19-17.53
	high	0.118-0.276	0.90-1.59	0.88-2.44	0.122-0.346	6.29-21.81

*pressure pulsations at inlet excluding the values at $NPSH_{1\%}$ and $NPSH_{3\%}$ because of too much attenuation by vapour existing in the inlet region

INTERPRETATION OF THE RESULTS

The amplitudes of the dynamic shaft deflection can be considered as to be an indicator for dynamic effects that can reduce the lifetime of the shaft sealing (especially in the case of mechanical seals) and for the dynamic loading of the bearings and of the shaft itself. The magnitude of the casing vibrations (measured near the bearings) gives indication on the dynamic loading of the bearings, but also of the pump structure and foundation. The amplitude of the fluctuating part of the axial thrust is directly responsible for the dynamic loading of the axial bearings; increased values are probably to be responsible for axial bearing failures. The amplitudes of the pressure pulsations can serve as an additional indicator for the dynamic loading of the pump casing, of the pump flanges and pipe connections, and of the connected piping installation as well.

For each of the three test pumps, the only constructional difference between the medium n_{ss} and the high n_{ss} variants consisted in assembling different impellers. All other mechanical details as bearings, shaft, shaft sealing, etc,. were identical for both variants of the same test pump. Also, the transducers and the transducer locations were not changed from one variant to the other. As the reference for the relative flowrate served the same value of Q (nominal value of the medium n_{ss} variant) for both variants of a test pump. Therefore, when comparing the measured quantities for the two variants of a test pump and for the same cavitation condition (cavitation inception or defined head drop), the only influence comes from the different impeller design. It has to be mentioned that the same magnitude of head drop and the corresponding cavitation condition occurs at remarkably lower NPSH-values for the high n_{ss} variants compared to the medium n_{ss} variants.

Due to the slightly different relative locations of the transducers in the different test pumps, it is not possible to compare directly the values of the dynamic quantities between the three different pumps. Only the device for measuring the axial thrust was located identically at the end of the shaft of each pump. These results are also influenced by the individual casing structure and by the kind of shaft seal (mechanical seal, stuffing box). The results concerning the shaft deflection include the influence of the individual distance of the sensor location to the bearings and to the seal. From constructional reasons, it was not possible to install the sensors for measuring the shaft deflection at the same distance to the shaft seal at the three different pump casings/shafts. The location that was chosen to measure the acceleration of the casing is very important in respect to the application of ISO 2372. These locations were fixed for each of the three pump casings.

The main result of the measurements is the fact that the investigations did not indicate for the test pumps any significant correlation between the measured vibration and fluctuation quantities and the individual values of the suction specific speed. Although, in some cases, slightly higher amplitudes were found for the high n_{ss} variants, there are also cases of higher amplitudes for the medium n_{ss} variants (all of them at a relatively low level). Therefore, it is very unlikely that a value of $n_{ss} > 11,000$ should generally by itself be the reason for broken seals, damaged bearings, or other types of failures.

In many cases, the measured quantities are strongly dependent on the actual cavitation condition and on the relative flowrate. Typically, the differences in the amplitudes according to the different values of n_{ss} were nearly negligible compared to the variations of the amplitudes caused by varying operating conditions.

The results of the experimental investigation that are presented here seem to be in an obvious contradiction to the findings of the statistical evaluation reported by Hallam [1]. Nevertheless, it is not the intention of the authors and of the German pump manufacturers to doubt the correctness of the statistical data underlying Hallam's conclusions [1]. Rather, it should be tried to combine both approaches and to find explanations for the differing outcomings. First of all, it must be taken into account that high n_{ss}-values can result from various design features the choice of which depends on the manufacturers design philosophy. The test pumps selected for the experimental investigation were designed by German pump manufacturers and, therefore, were only representative for their design philosophy. Especially, the high n_{ss}-values of the corresponding test pump variants are not attributed to extraordinarily large $Q_{shockless}/Q_{bep}$ ratios. Moreover, the data of the tested pumps presented in Tables 1, 2, 3, 4, 5, and 6 show that it is possible to design high n_{ss} pump impellers having values of $Q_{shockless}/Q_{bep}$ that are even smaller than those of the corresponding medium n_{ss} variant. The operation of these pumps at nominal flowrate or slight part load is, therefore, not necessarily connected with strong suction recirculations and corresponding detrimental effects (as it is when high n_{ss}-values follow from large $Q_{shockless}/Q_{bep}$ ratios).

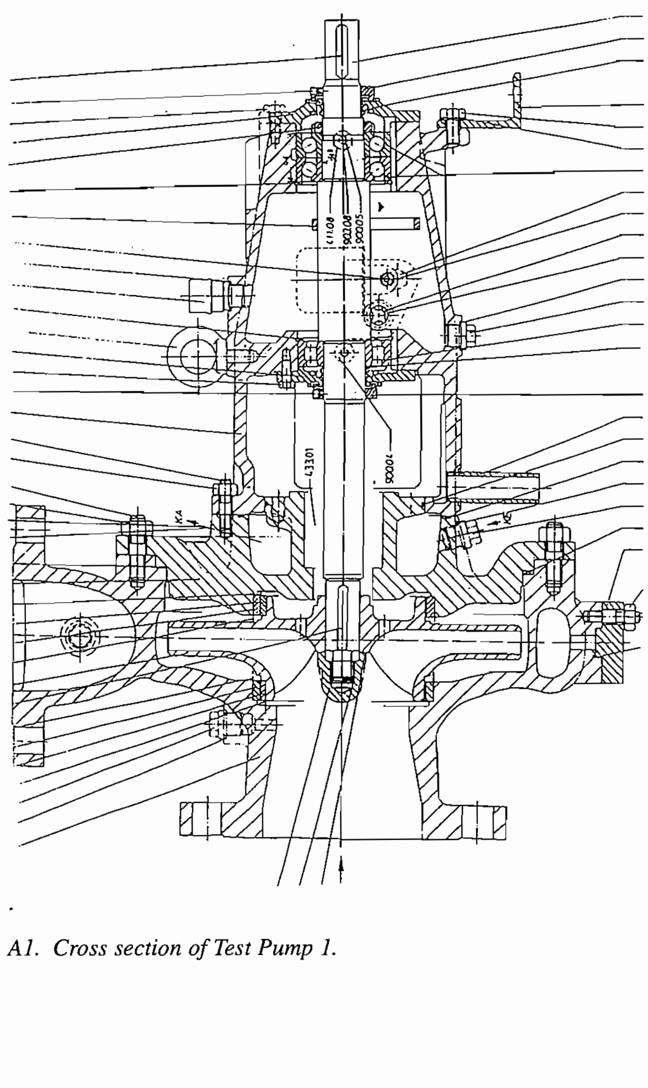
Another reason for the result of the statistical evaluation presented by Hallam [1] may be the fact that high n_{ss} pumps are preferably installed in plants with excessively low NPSHA-values (in many cases 1.0 to 2.0 m or even below). For such suction conditions, the suction-side piping installation must be designed and constructed with extreme care in respect to disturbances (e.g., pipe bends, valves) which can provoke cavitation. Otherwise, strong interactions of the suction piping and the cavitating pump can result and can produce considerably higher dynamic effects compared to an undisturbed (noncavitating) suction installation. This explanation for possible trouble with high n_{ss} pumps under unfavorable piping installation conditions was confirmed by some measurement results of our experimental investigation. In the case of the test pump variant showing the lowest values of $NPSH_{R3\,percent}$, the depressurization of the suction vessel by the means of the available vacuum pump was not sufficient to adjust the NPSHA at sufficiently low values. Therefore, throttling in the suction line became necessary. Unexpectedly high pressure fluctuations were found when a gate valve (installed in a pipe section of relatively large diameter and at a distance of several meters from the pump inlet) served for throttling. The fluctuations decreased to very moderate values (which were taken as the final measurement results) when the necessary throttling was affected by a pipe section filled with a porous package without generation of cavitation at the throttling location.

CONCLUSIONS

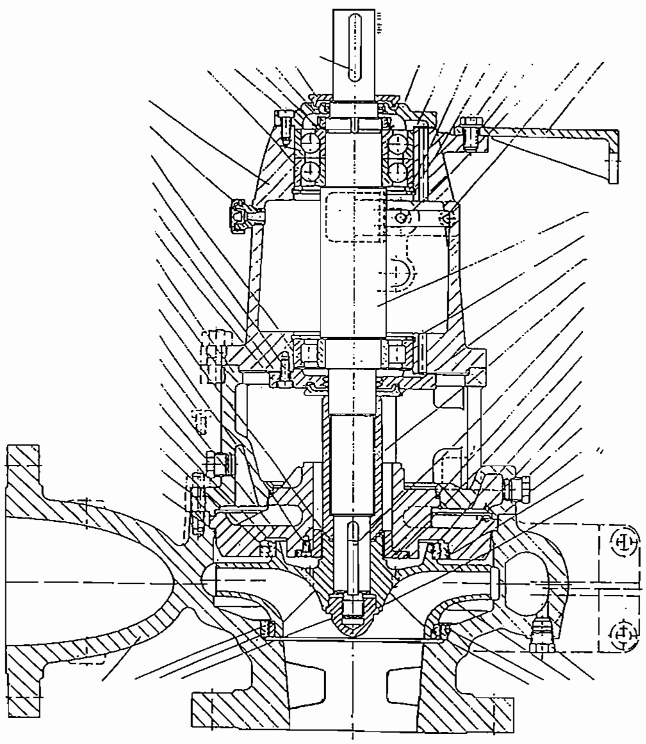
For each of the test pumps and variants, several dynamic quantities were measured that can serve as indicators for the dynamic loading and, thereby, for the probability of failures of pump parts. Some of these dynamic quantities as shaft deflection and fluctuating part of axial thrust require application of special sensors and/or measuring devices, which are normally not available and practicable for common pump tests on the manufacturers test fields or for pump testing and/or monitoring onsite. Nevertheless, they provided important information in the frame of the comparative tests that were presented here. Other dynamic quantities as casing vibration and pressure fluctuations that were also measured in our investigation do not need special preparation of the pump casing, bearing, etc., and can be easily determined for each pump installed in a test rig or onsite. When cavitation exists, pressure pulsations at pump discharge do obviously better reflect the dynamic excitations by unsteady flow phenoma inside the pump than suction pressure pulsations. The reduction of suction pressure pulsations in the case of cavitation is a consequence of the attenuation of pressure waves by the vapor volume existing at impeller inlet and is not an indication of really a reduced dynamic excitation. For practical purposes, discharge pressure pulsations can serve to indicate the magnitude of dynamic excitation by fluid forces acting on the pump and piping and, thereby, to set allowable operating limits which, e.g., may be assessed from inflection points on curve plots.

From the results of the experimental investigation, it can be concluded that the better cavitation performance of the high n_{ss}-variants of all three test pumps, which is attributable to the special shape of the corresponding impellers, does not correlate with a deterioration in respect to the measured dynamic quantities. Since the measured quantities can be taken as indicators for the dynamic loading of bearings, shaft seals, etc., the results of this investigation demonstrate clearly that a value of $n_{ss} > 11{,}000$ does not generally cause effects, which increase the failure probability and impair the reliability of a pump. This contradiction to results that came out from a statistical evaluation presented in [1] is evident, but must be explainable. As explanations can serve individual design features of high n_{ss} pumps designed by different manufacturers as well as the sensibility of low NPSHA applications to suction line installation effects.

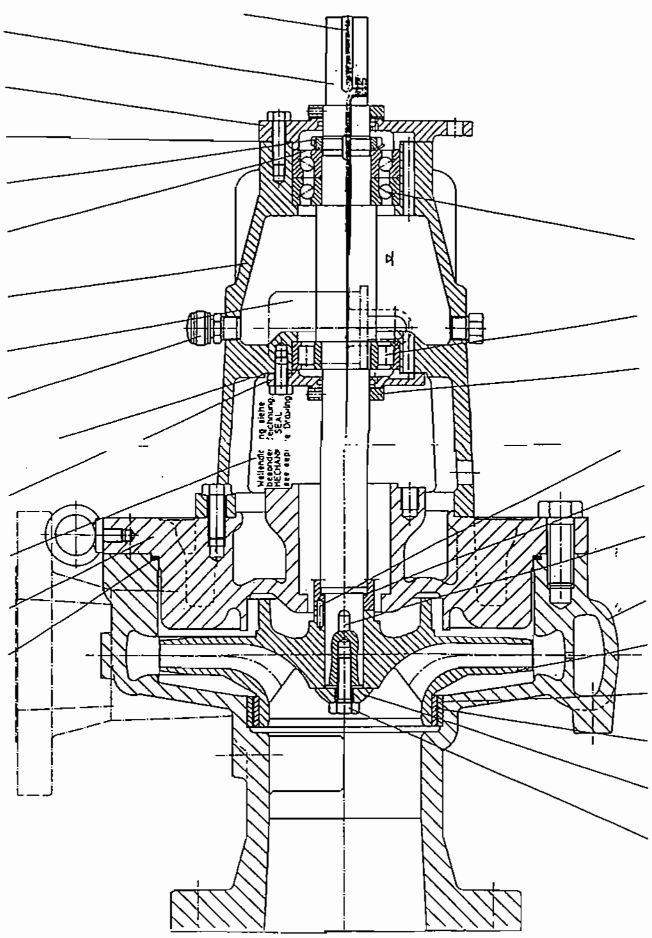
APPENDIX



A1. Cross section of Test Pump 1.



A2. Cross section of Test Pump 2.



A3. Cross section of Test Pump 3.

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2. Stoffel, B. and Hergt, P., "On Problems of the Specific Suction Speed as Evaluation Criterion for Operational Safety and Reliability of Centrifugal Pumps," (In German), Pump Congress Karlsruhe, Sess. B 8 (1988).
3. Fraser, W. H., "Recirculation in Centrifugal Pumps," World Pumps, pp. 227-235 (1982).

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