SOLUTION TO CAVITATION INDUCED VIBRATION PROBLEMS IN CRUDE OIL PIPELINE PUMPS

by

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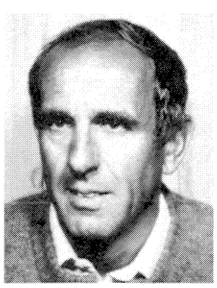

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ABSTRACT

As output was increased on the East-West crude oil pipeline in Saudi Arabia, increasing shaft seal failures were encountered. These limited the operation of the pipeline. Surveys on site and detailed model tests with flow visualization showed the cause of the seal failures to be pressure pulsations generated by collapsing cavitation zones. These cavitation zones occured on the impeller blades, and also in vortexes shed from the pump inlet features. Particular to this case was that the cavitation erosion was not severe, and was not the limiting factor for pipeline operation. In water, the same cavitation would have limited the impeller life due to cavitation erosion to a few hundred hours only. Through model testing an optimized impeller and casing modifications were developed which greatly extended the operating range of the pipeline. Rough guidelines were developed also to assess the danger of such pulsations in crude oil pumps

at the design stage, and to define, in critical cases, the model tests to be performed to ensure proper pump operation.

INTRODUCTION

The Aramco East-West crude oil pipeline stretches from the Eastern Province of Saudi Arabia to the Red Sea Coast, a distance of some 1200 km (750 miles). The pipeline has 11 pump stations, each with three pumps. A typical set is shown in Figure 1; the pump cross section is shown in Figure 2. It is a double suction, single stage pump. Its main data is given in Table 1. Extended operation at 120 percent of best efficiency flow is required. With a head of 2021 ft (616 m) in a single stage, this pump has quite a high power concentration.

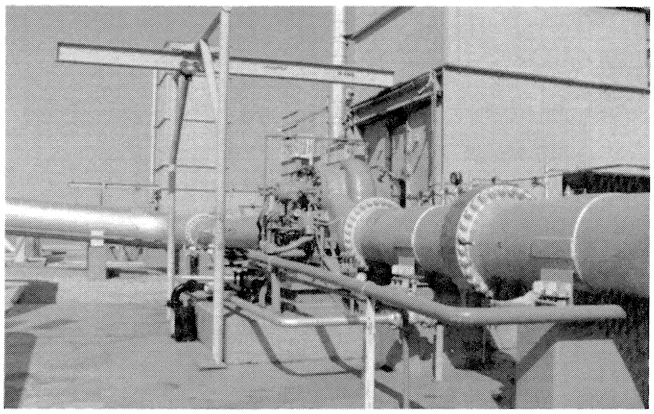

Figure 1. Main Oil Pipeline Pump on Site.

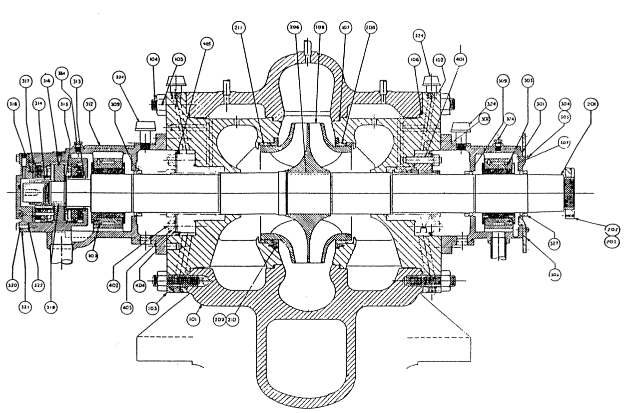

Figure 2. Cross Section of the Main Oil Pipeline Pump.

Table 1. Main Data of Pipeline Pump at Best Efficiency Point.

Quantity	1.0 Million Barrels per Day (MMBPD) = 29′300 GPM (1.85 m^3/s)
Speed	3,600 RPM
Head	2,021 ft (616 m)
Power, approx.	15,000 HP (11.2 MW)
NPSH available (minimum)	266 ft (81 m)
NPSHR (3%)	125 ft (38 m)
Impeller dia.	22½″ (0.572 m)

Commissioning was completed during mid 1981 with operation and maintenance being taken over by Aramco during the latter part of 1982. As output increased, a rising incidence of mechanical seal failures was observed and, by the end of 1984, failure rate reached chronic proportions with some seal faces lasting one day.

The radial and axial shaft displacement probes installed showed normal shaft vibration levels below 1.5 mil (38 μm) p-p, and did not indicate a dynamic problem. In spite of this, in 1986, detailed vibration measurements on the pumps were carried out.

OBSERVED PUMP VIBRATIONS

Besides recordings of the shaft vibrations at various speeds and flows, extensive surveys of bearing housings, pump and pipe vibrations were carried out, using accelerometers. Typical results are shown in Figure 3. The vibration spectra show practically no synchronous vibrations, quite small (< 0.1 in/s) vibrations at the blade passage frequency, but considerable broad band vibrations in the frequency range between about 1000 and 2000 Hz. The pipes vibrated strongly and the vibration spectra contain many individual peaks. This is a typical response of the various pipe radial natural frequencies to broad band pressure pulsations in the liquid. On the bearings, rms levels of vibration velocity of more than 0.5 in/s (12.7 mm/s) were observed, while levels in a "peak hold" mode reached 1.2 in/s (30 mm/s). This large difference indicates an unsteady excitation. Peak-to-peak values of the vibration velocity were at least 3.4 in/s (86 mm/s). Increasing the flow increased the vibration levels, and at that time it was suspected that flow turbulence was the cause of the excitations. Later investigations showed that besides the flow, the suction pressure had a major influence on the vibration levels.

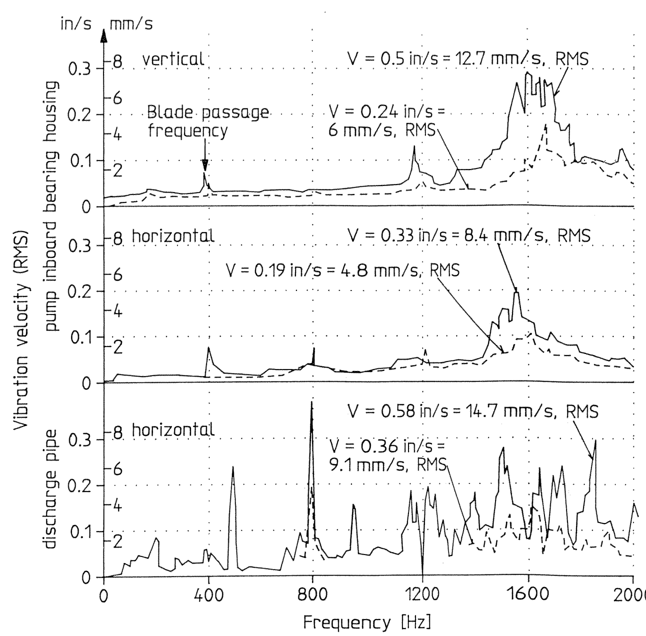

Figure 3. Vibration Velocities Measured at Site in 1986. n = 3320 rpm, Q = 1.35 MMBPD (2.48 m^3/s), solid lines. n = 3400 rpm; Q = 1.1 MMBPD (2.0 m^3/s), dashed lines.

Inspection of impellers revealed some cavitation damage on the suction side of the vanes and on the hub, but not very severe. Quite possibly, a 40,000 hr erosion life could have been reached, although this was not a specified requirement. A correlation of this damage to operating parameters was not possible, as flow, speed, and suction pressure varied widely in operation. It seemed that the pumps were operated at times at very high flowrates, exceeding 135 percent BEP-flow. Also observed were some fatigue cracks on the vane inlet tips. The major concern, however, remained with the low life of the mechanical seals. In order to reduce these problems, the operating window of the pumps was limited to maintain vibration levels at the bearing housings to maximum 0.27 in/s (6.9 mm/s) rms. Detailed measurements at various speeds, flows and suction pressures yielded a set of curves defining the operating window (Figure 4). Here, the dominating effect of the suction pressure, especially at high flows, is very obvious. Based on this, and the observed cavitation damage, it was concluded that the broad band vibrations observed were caused by pressure pulsations originating from the bubble collapse of severe cavitation. It is highly likely that these pressure pulsations caused the seal failures. The shaft seals are well proven heavy duty cartridge seals on a shaft of six inches diameter. The rotating seal ring is floating and made of tungsten carbide, the stationary seal ring is a balanced carbon ring, also floating. In case of seal failure, a backup floating carbon bush limits the leakage. The exact damage mechanism could not be ascertained, but the damage appeared to be caused by axial movement of the seal sleeve and carbon ring and shattering of the carbon faces.

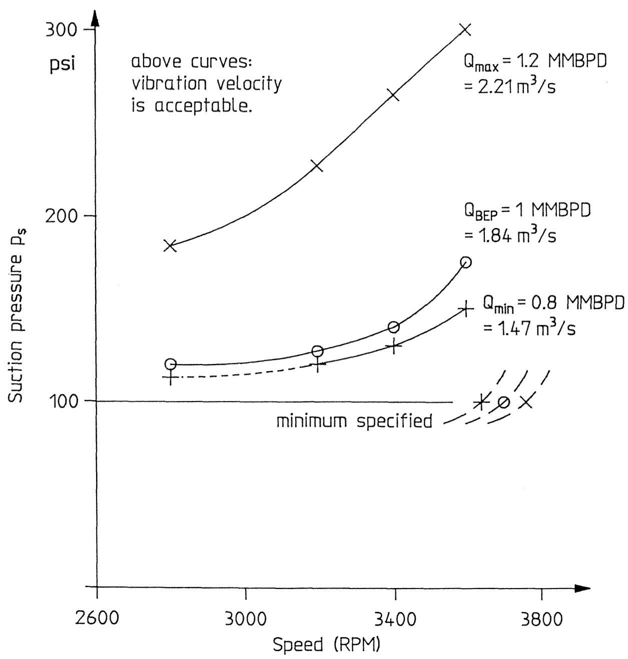

Figure 4. Lines Denoting Suction Pressure as a Function of Speed for which the Vibration Velocity at the Inboard Bearing is 0.27 in/s (6.9 mm/s) rms. Operation is acceptable above the lines. Solid lines: pump in original condition, dashed lines: modified pump.

SHORT TERM MEASURES AND FURTHER REQUIREMENTS

Besides the limitation of the operating window as discussed, ARAMCO replaced some impellers with modified ones having a smaller eye diameter. This increased the operating window considerably, but was still insufficient to fulfill longterm operational needs, as especially at the first four pumping stations the available suction pressure was limited. To maximize the capacity of the pipeline the manufacturer was commissioned to construct a model pump and conduct flow visualization testing to design a replacement impeller, which would allow the pumps to be operated at 3600 rpm, 1.2 MMBPD (2.21 m^3/s) at a suction pressure of 100 psig with vibration levels below 0.27 in/s (6.9 mm/s) rms.

MODEL TESTS

Tests on a model pump [1] were scheduled to confirm the cause of the high-frequency vibrations and to develop an improved impeller with reduced excitation forces when operating under cavitation conditions. In order to achieve this goal a scaled-down baseline impeller was tested along with the new impeller in order to get a truly meaningful comparison between the baseline and the new impeller. The following criteria were used to quantify the improvement:

- cavity length
- cavitation noise
- pressure pulsations
- radial excitation forces acting on the pump shaft
- casing vibrations
- paint erosion tests

The tests were carried out on a model pump with all waterways being exactly geometrically similar to the full-scale pump. A cross section of the model pump is shown in Figure 5 and the main data are given in Table 2.

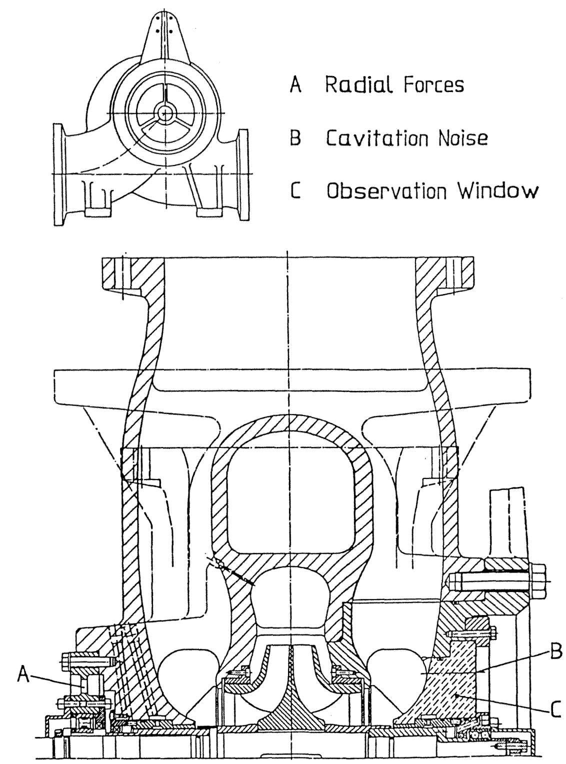

Figure 5. Cross Section of Model Pump.

Table 2. Model Pump Data.

Scale ratio	1:1.65
Impeller outer diameter	13.8 in (350 mm)
Test speed	2000 RPM
BEP: flow rate	3'610 GPM (0.228 m^3/s)
head	229 ft (68.8 m)

The double suction model pump was equipped with a transparent suction cover for flow visualization on the non-drive end and with strain-gauge equipped bearing brackets on the drive end for radial force measurements. The pump was tested in a closed loop with partially deaerated water.

Visualization tests on the baseline impeller revealed, depending on flow and suction pressure, some sheet cavitation on the impeller vane suction side, but also vortexes emanating from the inlet rib and from relatively sharp corners at the casing and impeller inlet, as indicated in Figure 6. Based on these observations, a number of modifications were made, as shown in Figure 7:

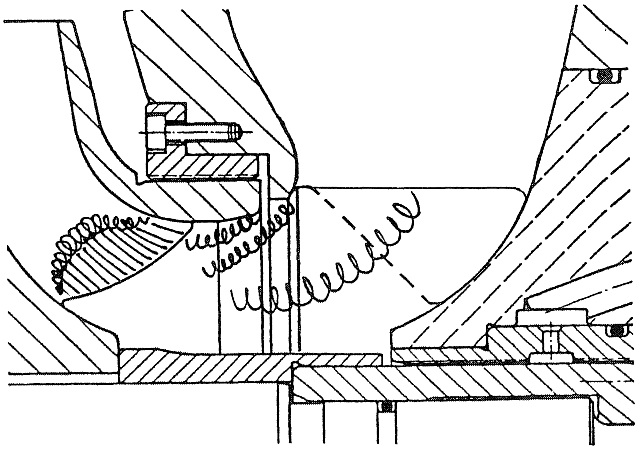

Figure 6. Vortexes and Bubble Extension Observed on Baseline Hydraulic.

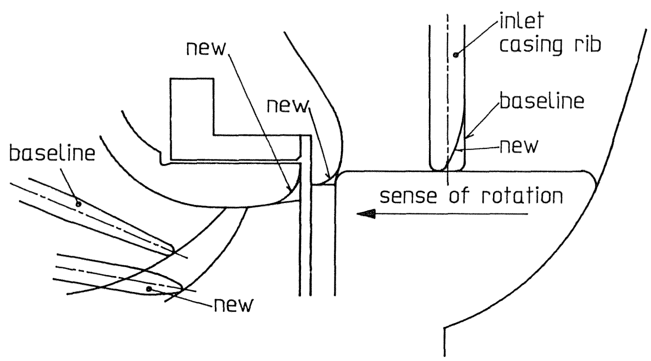

Figure 7. Casing and Impeller Modifications Compared with Baseline Hydraulic.

- The inlet rib was modified to reduce vortex shedding
- The casing and impeller contours at the inlet were slightly rounded
- The inlet vane shapes and the leading edge profile was optimized to reduce cavitation inception and cavity volume. The inlet angle was slightly reduced. The new impeller was required to give exactly the same performance as the baseline impeller. Therefore, the meridional contour of both impellers, the outlet angle of the vanes, the number of vanes (seven) and the vane shape near the outlet, were kept identical in both impellers.

Both impellers, and the effect of the casing modifications were thoroughly tested. As an example of the cavitation bubbles observed, the baseline and the new impellers are shown in Figure 8 at best efficiency point and at a suction pressure (referred to the fullscale pump at 3600 rpm) of 130 psig (9 bar). While the baseline impeller shows a considerable cavitation bubble extended over the entire blade span, the new impeller shows only a very slight bubble near the hub. The drastic improvement in cavity length is demonstrated in Figure 9. Between 80 and 120 percent of BEP-flow, the cavity length follows a pattern similar to that in Figure 9. At 60 percent flow no sheet cavitation occured, and the flow shows clear recirculation for both impellers. The drastic improvement in visual cavitation inception (first bubbles recognizable) is shown in Figure 10. The three percent head drop is little affected, it even increases slightly for the new impeller. This figure makes clear, that cavitation inception, being mainly influenced by the leading edge geometry, and three percent head drop, which is influenced by cavitation bubbles reaching the throat area, are not directly related.

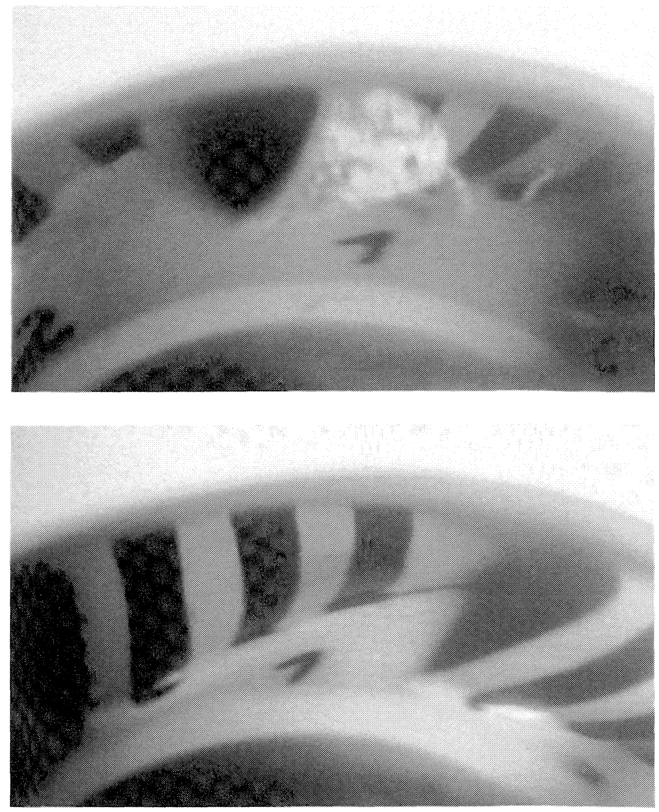

Figure 8. Comparison of Cavitation Bubble Extension Observed on Impeller Vane, Suction Side for BEP. The bubble length corresponds to a suction pressure of p_s = 9 bar (130 psig). Original casing.

The observations made so far all pertain to a reduction of the cavitation bubble extension on the impeller blades, based on the knowledge that smaller cavitation bubbles create less vibration excitation. A more direct measure of excitation forces is the pressure fluctuation in the suction chamber, as indicated in Figure 5. A piezoelectric pressure transducer with a resonance frequency of about 80 kHz is used for this purpose. For the analysis, two frequency bands were chosen, one from 800 to 2000 Hz, representing the frequency range of strongest vibrations, and the other from 550 Hz to 180 kHz, in line with a method developed for cavitation damage predictions [2, 3]. Results from this latter frequency range are called "cavitation noise." The cavitation noise is shown in Figure 11 for the baseline impeller, the new impeller, and the new impeller combined with the casing modifications shown in Figure 7. It is clear that the casing modifications contribute significantly to the reduction of the cavitation noise, particularly towards lower suction pressures. The pressure pulsations in the frequeny band from 800 to 2000 Hz (Figure 12), also measured at the pump discharge, again clearly show the influence of the casing modifications at lower suction pressures, particularly at high flows. Based on these successful results, it was decided to realize

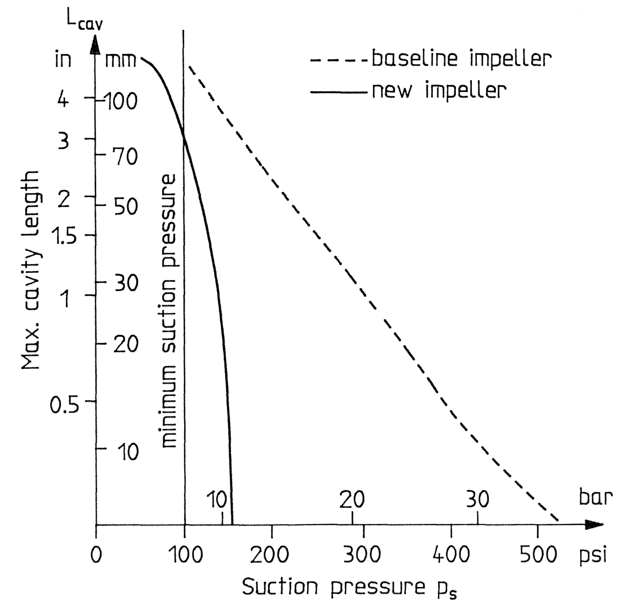

Figure 9. Maximum Cavity Length on Impeller Vane, Suction Side at BEP, Observed on Model Pump Impeller and Converted for Full Size Pump at 3600 rpm. Original casing.

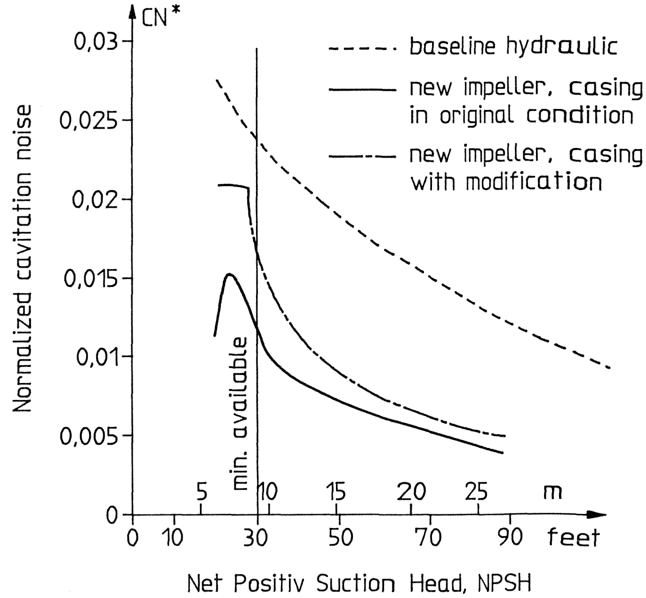

Figure 11. Cavitation Noise at BEP Measured in Model Pump at 2000 rpm. (Min. available NPSH on model pump corresponds to 100 psig suction pressure on full size pump).

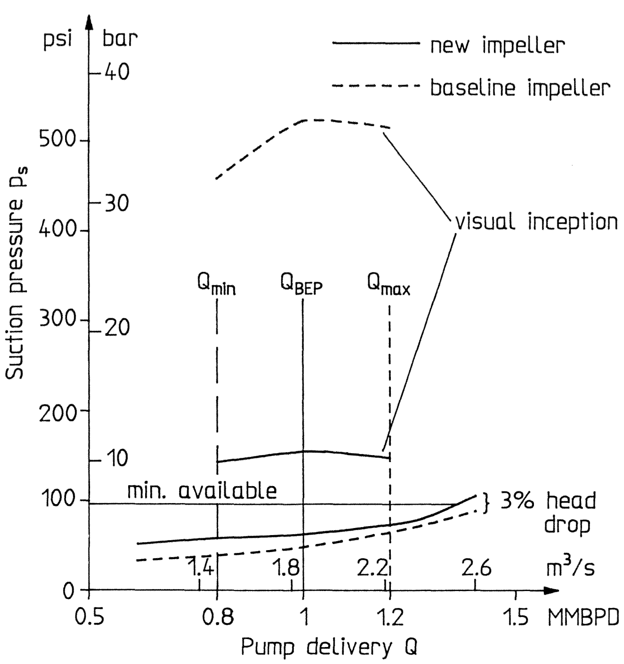

Figure 10. Cavitation Inception on Impeller Vane, Suction Side and Three Percent Head Drop, Observed on Model Pump, Converted to Full Size Pump at n = 3600 rpm. Original casing.

both the impeller and the casing modifications in one pump in the field in order to confirm the model tests.

FIELD MEASUREMENTS

One pump in the field was fitted with a new impeller based on the geometry of the successful new impeller of the model tests. The impeller geometry was precisely duplicated using the lost wax casting technology (precision casting). Vibration measurements were then carried out at the lowest available suction pressure of 100 psig for various speeds and flows. In a second step, the casing modifications were carried out based on the model tests, and the vibration measurements were repeated. The results at 3600 rpm are shown in Figure 13. Only with both the new impeller and the casing modifications, as predicted by the model tests, could the required vibration levels be reached over the entire flow range. As expected, at lower speeds the vibration levels were even lower. The modifications were thus successful and are now being implemented step by step on all other pumps. The tremendous extension of the operating field is also indicatd on Figure 4, where the dashed curves were extrapolated from the measurements at various speeds at 100 psig suction pressure.

DISCUSSION OF THE RESULTS

The tests confirmed that the broad band vibrations observed on bearing housings and on the pipes, and the short lifetime of the mechanical seals, were caused by pressure fluctuations originating in the collapse of cavitation bubbles. It is significant to note that the vibration problems arose in crude oil pumps. If the pumps had been pumping water, cavitation erosion at the high flowrates and low suction pressures would have been so severe that the impeller life would have been only a few hundred hours. Thus, while in water the limiting factor for cavitation usually is given by impeller erosion, in crude oil, and most probably generally in hydrocarbons, the limiting factor for cavitation may be given by pressure pulsations causing vibrations and seal damage. As this has not been generally recognized, and does not seem to apply to all pumps, further criteria are needed to assess the risk of such pulsation problems in hydrocarbons.

It can be can reasonably assumed that the amplitude of the pressure pulsations are a direct indicator for seal malfunction. For a given liquid, the pressure pulsations are a function mainly of the suction pressure (minus vapor pressure) and the size of the cavitation bubble. A relationship can be deduced from the work of Gülich [2, 3]. By a semi empirical method, he has developed relationships for the erosion rate vs cavity length and the erosion rate versus cavitation noise:

$$E_R \sim (L_{cav})^{2.8} \cdot (p_s - p_{sat})^3$$
$$E_R \sim (CN)^{2.9} \qquad (1)$$

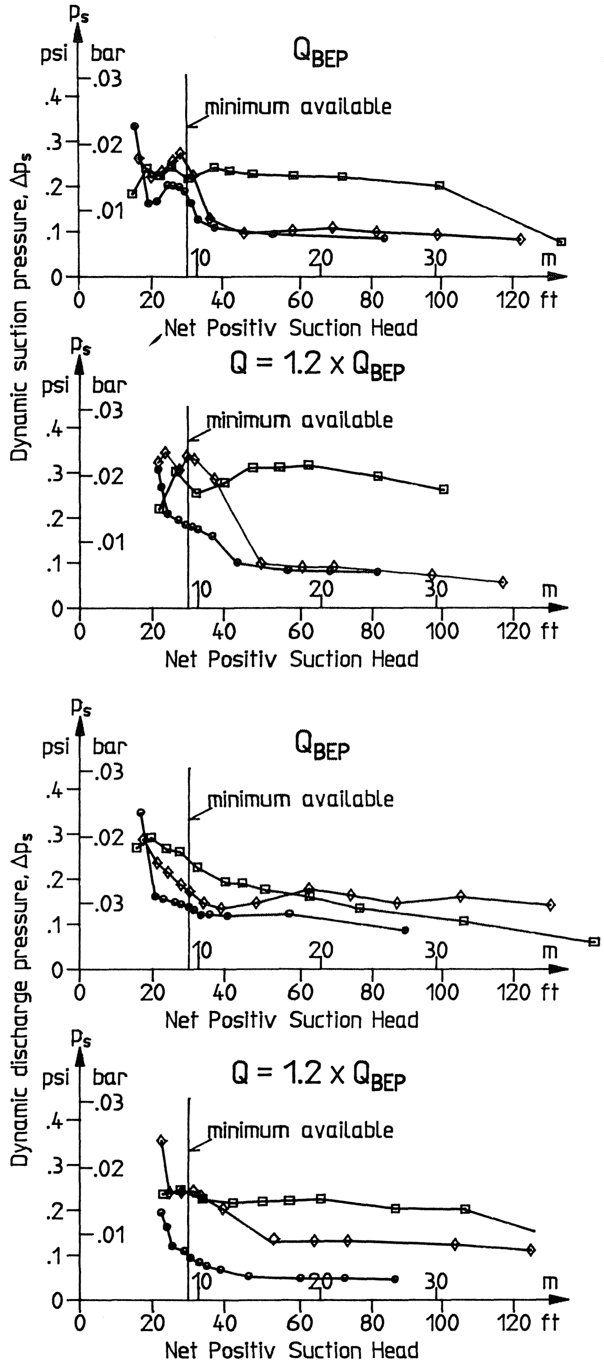

Figure 12. Pressure Pulsations at BEP and 1.2 · Q_{BEP} Measured on Model Pump at n = 2000 rpm. □—□—□ = baseline hydraulic, ◇—◇—◇ = new impeller, casing original conditions, ○—○—○ = new impeller, modified casing.

Eliminating the erosion rate E_R, for given liquid properties the following is found:

$$(CN)^{2.9} \sim (L_{cav})^{2.8} \cdot (p_s - p_{sat})^3$$

As the cavitation noise is physically the same as pressure pulsations induced by cavitation, the quantity of interest, it can be written approximately (neglecting small differences of exponents)

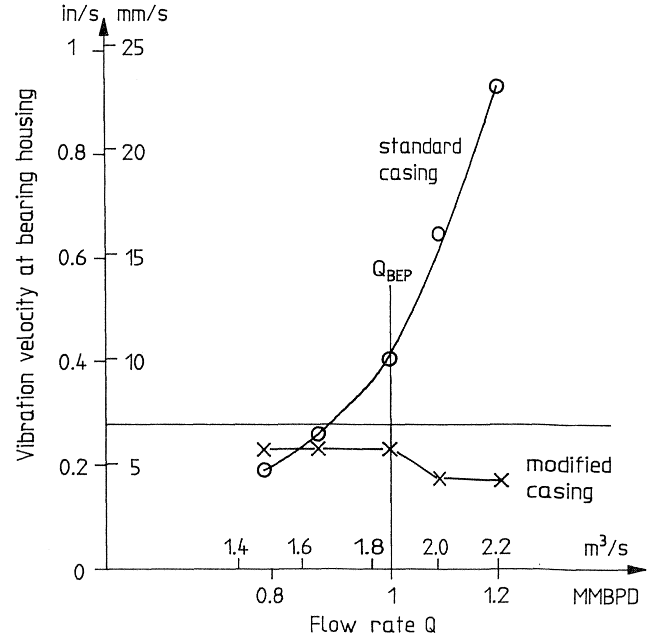

Figure 13. Influence of Casing Modifications, Measured on Site with New Impeller. n = 3600 rpm, suction pressure = 100 psig (6.9 bar).

$$\Delta p_s \sim L_{cav} \cdot NPSHA \qquad (2)$$

Here, use was made of the fact that NPSH is proportional to $p_s - p_{sat}$.

Thus, the pressure pulsations due to sheet cavitation are directly proportional to the cavity length times the available net positive suction head. If other sources of pressure pulsations exist, such as were found in the model in the form of vortexes with bubbles in their core, pressure pulsations are higher than indicated by Equation (2). In Equation (2), the NPSH value is known, and the cavity length can be reasonably well calculated from the model tests in the following way:

$$L_{cav,full} = L_{cav,mod.} \cdot \frac{D_2 \text{ full}}{D_2 \text{ model}} \qquad (3)$$

This relationship holds for operation of full scale pump and model at the same cavitation number and flow coefficient:

$$\sigma_{u1} = \frac{2g \cdot NPSHA}{u_1^2}, \quad \varphi = \frac{Q}{u_2 \cdot D_2 \cdot B_2 \cdot \pi} \qquad (4)$$

The pressure pulsation p_s is not known, however, we can assume that for broad band vibration excitation, and in a given frequency band, the pressure pulsations are proportional to vibrations. Knowing that a pulsation level corresponding to a vibration velocity of 0.27 in/s (6.9 mm/s) rms leads to acceptable seal life, we can deduce a critical value for the product $L_{cav} \cdot NPSH_A$, valid in crude oil:

$$VP_{crit} = L_{cav} \cdot NPSHA \qquad (5)$$

Using the cavity length measured in the model tests, and Equation (3) to determine cavity length for the full size pump, values of VP_{crit} were calculated for the baseline and new impeller, and the operating points of Figure 4. The values ranged from 836 to 1,226 in · ft (6,480 to 9,500 mm · m), except for the baseline impeller at 3,600 rpm, 1.2 × BEP, where a lower value of 722 in · ft (5,600 mm · m) was found. At that condition, strong vortexes from the casing add to the excitation from the cavity on the impeller, resulting in a lower VP_{crit} value calculated. Taking an average value, it can be written:

$$L_{cav} \cdot NPSHA \leq 1000 \text{ (in} \cdot \text{ft)} \quad (6)$$

This is only to be taken as a guideline and only for crude oil pumps, as it represents a simple attempt to describe a very complex phenomenon. For a particular application, Equation (6) allows one to determine the acceptable cavity length at given NPSHA conditions. These values can be converted to data for a model, using Equations (3) and (4). Model tests including flow visualization can then be done to ensure proper operation of the pumps in the field. Obviously, this procedure is reasonable only for critical pumps of large size and high head per stage (high NPSH required). Equation (6) in itself can give some guidance on potential problems, as on pump selection, NPSHA is known, and a maximum cavity length can be estimated from the distance of the blade leading edge to the throat area of the impeller inlet. Once the cavity reaches the throat area, head drop starts to occur, and a further increase in cavity length could lead to excessive head degradation.

CONCLUSIONS

High pressure pulsations due to the collapse of cavitation bubbles have led to severe failures of the mechanical seals in high head, large crude oil pumps. The investigations showed that in pumping crude oil, and quite possibly most hydrocarbons, the pressure pulsations may limit the acceptable degree of cavitation, and not the erosion on the impeller as is most likely for pumping water. As a rough guideline, Equation (6) can be used to assess the risk of such pressure pulsations, and if the situation is critical, to define model tests ensuring proper function of the full scale pump. In this particular case, model testing with flow visualization proved to be the key to understanding the field problems encountered and to optimizing the pump inlet geometry and the impeller for extended operating requirements of this pipeline.

NOMENCLATURE

BEP	Best efficiency point
B_2	Impeller exit width (m)
CN	Cavitation noise (pressure fluctation) bar, RMS values in frequency band 550 Hz. to 180 kHz.
CN*	$\frac{CN \cdot 10^5}{\frac{1}{2}\rho u_1^2}$, normalized cavitation noise
D_2	Impeller diameter in (m)
E_R	Erosion rate
g	Acceleration of gravity (9.81 m/s^2)
L_{cav}	Cavity length, in (mm)
NPSHA	Net positive suction head available, ft (m)
n	Speed, RPM
p_s	Suction pressure, psig (bar)
p_{sat}	Vapor pressure of liquid at suction conditions psi (bar)
Q	Flow, Million Barrels per Day, MMBPD (m^3/s)
Q_{BEP}	Flow rate at BEP
Q_{min}, Q_{max}	Minimum and maximum specified flow rates
u_1	Impeller eye peripheral velocity (m/s)
u_2	Impeller peripheral velocity (m/s)
v	Vibration velocity, in/s (mm/s) RMS
VP_{crit}	Critical product of bubble length times NPSHA to limit pressure pulsations in crude oil pumps, in · ft (mm · m)
ρ	Liquid density (kg/m^3)
Δp_s, Δp_D	Suction and discharge pressure fluctuations, psi (bar), RMS values in frequency band 800-2'000 Hz
σ_{u1}	Cavitation number, Equation (4)
φ	Flow coefficient

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