TORSIONAL ROTORDYNAMIC EXCITATION, NATURAL FREQUENCY AND DAMPING CHARACTERISTICS IN A VISCOUS CENTRIFUGAL PUMP APPLICATION

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ABSTRACT

This lecture is concerned with the characteristics of torsional vibration on a single-stage, double-suction centrifugal pump designed to handle a highly viscous product. This test case serves to verify engineering assumptions about torsional excitation and damping that are commonly applied when performing torsional analyses, but on which the margins of uncertainty are known to be significant. The uncertainty, along with the scarcity of information available, originates from the difficulty to measure torsional vibrations, especially in the field, and constitutes the most apparent difference to lateral vibrations.

Although several researches have been devoted to this topic through the years, even the most advanced industry standards are still quite vague on the analysis side (API 610 [1] does not define typical values of torsional excitation, unlike it does for instance for
residual mass unbalance), and incomplete on the measurement side (API 670 [2] contains no section on torsional vibration measurement). API Recommended Practice (RP) 684 [3] represents a significant improvement, covering torsional analysis and field testing in Chapter 4. However the bulk of the information contained therein pertains mainly to gas handling turbomachines.

The experimental characterization presented in this study is conducted in the pump test laboratory using mineral oil at varying viscosities and measuring static and dynamic torque with a torque meter. The main steps of extracting information about the relevant sources of torsional excitation and about the modal parameters (torsional natural frequency and damping), starting from the frequency spectra of the dynamic torque, are illustrated in detail. In addition, the influences of the relative flow and of the oil viscosity, changed according to the test matrix, are investigated.

A torsional model is developed to correlate the measured torsional vibration response to the excitation source, while highlighting the impact of critical parameters, such as the impeller wet inertia and the coupling stiffness, on the torsional natural frequency (TNF).

The damped torsional model is used to back-calculate the torsional excitations existing at the impeller, based on the dynamic torque measured on the coupling. Finally, the estimated magnitudes of the hydraulic torque components at the blade passing frequency (BPF) and at two-times the blade passing frequency (2xBPF) are compared to the transient CFD prediction.

The authors review several options and make recommendations on how to enhance the numerical predictions by properly including the added fluid inertia, and by estimating torsional damping in a less conservative, more realistic way, in case experimental data are missing.

INTRODUCTION

Torsional vibration is described as a “silent machine’s killer”, because it can be in place since the early phase of the machine installation and give no detectable signs of its existence until a shaft or coupling failure event. In practice, torsional rotor vibration measurements are still relatively unusual on industrial equipment compared to translational (axial and radial) vibration measurements.

One of the main causes of failures is torsional resonance, namely the coincidence of a torsional natural frequency with a source of excitation. A torsional analysis performed in advance can be greatly effective in preventing such scenario. As a general rule, the need for a torsional analysis calculation and report is determined during the design phase based on the application risks, on the design, manufacturing and operational experience, and on the machinery power level.

The Oil & Gas pump industry standard API 610 11th Edition [1] offers good guidance through many aspects of a torsional analysis procedure, including: the cases where the analysis is mandatory (§6.9.2.2 – train configuration); the recommended kind of analysis (§6.9.2.1 – undamped natural frequency, steady-state response, transient response); the typical frequencies of torsional excitation (§6.9.2.5); and the minimum required separation margins (§6.9.2.6). Also across other industries besides Oil & Gas, e.g. Power and Water, customers often specify a “torsional critical speed analysis” report in the contractual deliverables to prove the integrity of the full train against torsional resonance.

Unfortunately, the actual levels of alternating torsional stresses on pump rotors are difficult to anticipate, especially at a resonance, due to the lack of reliable input data necessary for the response analyses. It is claimed that as long as sufficient separation margins between exciting frequencies and natural frequencies are fulfilled in the analysis and maintained in reality, torsional stress levels will be “conservatively low”, thus ensuring infinite fatigue life of the component with no further need for a torsional response analysis. However this claim only relies on theoretical arguments: above all a high accuracy of the torsional natural frequency prediction in the design phase; and second, the confidence in achieving the required separation margins, even at cost of many design iterations.

Situations arise in pumping applications where the actual frequency separation margins are smaller than the minimum acceptable values (10% according to API 610 [1]) due to manufacturing and assembly tolerances, without necessarily posing a concern for the rotor integrity. Moreover, if the design process has come to an advanced stage, the only viable option to shift a torsional natural frequency is the change of coupling stiffness and inertia, which might not be enough for reaching a separation of 10%. Finally in variable-frequency drive (VFD) pumping applications, a large operating speed range does not allow a completely resonance-free design. In all aforesaid situations, the machinery has to be qualified via torsional response and subsequent shaft stress analysis, for which realistic estimates of the exciting torques and the degree of torsional damping are required.

Large uncertainties affect both the steady-state excitation (acting mechanism, frequency and magnitude) and the damping (sources and amount). Rules of thumb used for many years, and recently echoed by the API RP 684 [3] for compressors, prescribe the magnitude of a torsional steady-state excitation to be a small fraction (typically 1%) of the static torque acting on the pump shaft, with no distinction between electro-magnetic, mechanical and hydraulic sources. In a similar way, torsional damping is roughly quantified as a small fraction (typically 1%) of the so-called critical damping factor; in other words, a damping factor of 1% is equally assigned to all the modes of concern, irrespective of any geometric or operating parameters.

A further grey area is the influence of flow and fluid parameters on the torsional vibration characteristics, namely torsional natural frequency and damping. This topic is relevant to pumps designed to handle fluids other than water (e.g. crude oil), for which the question arises, as to what extent the water factory-acceptance test truly reflects the field conditions.
LITERATURE REVIEW

Many sources of information are publicly available on the general subject of torsional vibration, with a great majority of them dedicated to reciprocating machinery (engines and compressors), which are inherently prone to be excited by the large alternating pressure and inertia loads, and for which design and analysis practices seem consolidated for many years.

The number of relevant studies on rotating machinery becomes progressively smaller, and the contained information more uncertain, once approaching centrifugal pumps. Finally, the authors have found that only few references deal with the experimental characterization of torsional vibration, comprising natural frequencies, excitation and damping.

This review includes a critical evaluation of the state-of-the-art on torsional analysis methodology.

Torsional Fatigue Failures (informative)

The current work does not originate from failures suffered on centrifugal pumps of the authors’ Companies in recent times.

As correctly observed about centrifugal compressors [4], these events are destructive but rare, despite the fact that little emphasis is placed in the machinery standards on the torsional measurement side. API 610 11th Edition [1] kept the provision for torsional vibration measurement during complete unit tests (§8.3.4.4) as non-compulsory. Moreover the standard API 670 [2] about Machinery Protection is still silent on torsional vibration measurements as a permanent monitoring and diagnostic technique, supposedly due to the low grade of standardization and robustness as well as the high complexity of the instrumentation used nowadays for that purpose.

By some accounts (Brown [5]), shaft torsional fatigue, owing to a machine variable loads or to resonance, is experienced more frequently than commonly realized, being a serious contributor in more than 10% of the industrial (reciprocating and rotating) equipment failures. Corcoran et al. [6] document several case histories of undetected torsional vibration on rotating and reciprocating machinery which had been running smoothly when at some point the coupling failed “suddenly and without warning”. The successive post-incident surface examination revealed rotating fatigue as the failure mode, in most cases resulting from a dominant torsional component and some residual bending. Other reports of torsional breakages exist mainly from gear-driven machinery (Wang et al [4]), affecting the driven and the driver shaft sides alike. As torsional fatigue is the underlying failure mechanism, the cracks tend to initiate from surface regions of high torque load and high stress concentration (grooves, keyways, shrink fits, shoulders etc.) and later propagate into the core material. The most recurrent damages on gearboxes themselves are tooth breakage (Wachel and Szenasi [8]) and accelerated wear rate (Wachel and Szenasi [8]), bringing a deterioration of the running quality and an increased noise level.

Torsional Natural Frequencies

Wang et al. [4] analyzed both the sources and the effects of uncertainties in the calculation of TNF, leading in some cases to actual separation margins lower than 10%. They identified the flexible coupling as the main source of uncertainty, and advised to pay particular attention to the value of torsional stiffness and to the modeling of the connections with the shaft hubs.

They also pinpointed the effects of added fluid on the torsional properties, namely the wet impeller inertia and the damping. Using the best available information to build a torsional model, the torsional natural frequency can be predicted with an overall accuracy of 5%.

In light of the API standard requirement of 10% separation margin (for prediction), they proposed a separation margin of 5% for a field-verified TNF.

Torsional Excitations

The review concerns only the steady-state torsional excitations of mechanical and hydraulic nature. Electrical steady-state and transient excitations are not considered in this study.

Mechanical Excitations

Although API 610 [1] and API RP 684 [3] advise to always consider 1X and 2X as potential sources of torsional excitation in the analyses, measurements show significant amplitudes at these frequencies only in geared trains. In directly-driven trains the measured 1X and 2X are negligible, as experimentally verified by Kaiser et al. [9], while their generating mechanisms are still unclear.

The habit of classifying these excitation frequencies as “mechanical” stems from the analogy with lateral rotordynamics where 1X is often the “telltale symptom” of unbalance and 2X of misalignment. This parallel makes sense when considering geared trains, which exhibit a strong coupling between lateral and torsional vibration.

The rules of thumb given by Wachel and Szenasi [8] to use 1% of the static torque of 1X and 0.5% of the static torque for 2X for both directly-driven and gear-driven trains have been largely accepted, see for example Corbo and Malanoski [10] and API RP 684 [3] on compressors. These excitation magnitudes, if used in torsional response analyses of directly-driven trains, together with worst-case assumptions about damping (e.g. 1%) and acting location, may lead to significantly over-predicting the alternating stress (especially in the coupling); and in case of potential resonance, they may result in numerical values beyond endurance limits of the shaft or coupling materials. This could cause unjustified concerns, as well as costly and unnecessary design modifications.
Hydraulic Excitations

Theoretically, the generating mechanism of the hydraulic torsional excitation is the same as for the dynamic forces causing structural vibrations, and has been known for long time (e.g. Robinett et al. [11]): pressure pulsations produced by the wake flow from the rotating impeller as it passes a stationary diffuser or a volute lip. Yet the literature about the influence of said excitation on torsional vibrations is in disagreement on the typical magnitudes to be assigned to the blade passing frequency and to its integer multiples.

- Güllich [12] states that pump impeller torque fluctuations are usually between 1 and 5% of the rated torque, with the higher values applying to volute pumps, blade numbers less than 5, and at partload. The portion of the signal at blade passing frequency amounts to 70-90% of the overall dynamic torque.
- Blanco et al. [13] conducted transient parametric CFD analyses on a single-stage, single-volute pump at different impeller diameters and different flow rates. The blade passing torque component increases while operating far from the best efficiency point (BEP), reaching up to 4% of the static torque at a flow rate of 20% BEP.
- Kaupert [14] published dynamic torque values for a multi-stage diffuser pump derived from transient CFD, which are remarkably higher, probably due to the tighter radial gap between impeller and diffuser: 5% at BEP, 13% at 65% BEP, and 8% at 125% BEP.
- Wachel and Szenasi [8] claim, based on their experience, that torsional excitation at blade passing frequency is so low that can be disregarded in practice, except in case of acoustic resonance. They give a rule of thumb, based on the number of blades \( z_b \), to estimate the magnitude of the impeller exciting torque, that is to consider \( (1/z_b)^2 \) % of the static torque value.
- Corbo and Malanoski [10], expressing reserves for the rule of thumb made in [8], suggest the ubiquitous 1% of the static torque, on the base of its (supposed) conservatism.

Torsional Damping

While the source and the nature of torsional damping are still debated within the academic world, Original Equipment Manufacturers (OEMs) generally strove to quantify its overall effect on the resulting torsional stress, particularly at resonant conditions.

A direct approach would require rigorous experiments on several items, such as impellers, couplings and gears, through a “representative” load, speed and frequency test matrix, as well as the development of reliable and fast computational programs capable of estimating damping coefficients starting from geometrical and working input parameters. This was basically the way the so-called bulk-flow codes, used in lateral rotordynamics, historically evolved. Nevertheless, very few cases of torsional rotordynamic instability are known in the Turbomachinery industry (Adachi and Murphy [15] presented one in 2016), a sign that torsional damping, regardless of its exact amount, is generally small but positive. This observation might explain the low interest of the industry to develop codes for the prediction of torsional damping coefficients, and the widespread tendency to favor the dimensionless modal damping form over the dimensional coefficient form.

- Material damping provides the lowest contribution to torsional damping (0.16% for steel according to Ehrich [16]), followed by the internal, i.e. structural and material, damping (0.5% according to Frei et al. [17]).
- For directly-driven machines using standard laterally flexible couplings with membrane or diaphragm constructions, Wachel and Szenasi [8] suggest values of damping ratio in the range 1.67-2.5%.
- For directly-driven machines using special torsionally flexible couplings with resilient elements, Frei et al. [17] mention damping ratio varying from 2.5% up to 10%, depending on the coupling design.
- For gear-driven systems the quoted damping ratios go from a minimum of 1.5-2% for tightly fit gears with no backlash (Vance [18]), to a maximum of 2.5-5% for geared systems having significant torsional-lateral coupling and equipped with fluid-film bearings (Pradetto and Baumann [19]).
- Corbo and Malanoski [10] present probably the most thorough literature overview of the torsional damping sources and of the values of the associated damping factors for directly-driven and gear-drive rotating machines, not hiding their “somewhat nebulous” derivation. They recommend using a damping ratio of 1% for directly-driven systems and 2% for gear-driven systems. These recommendations became widespread in many following publications due to their conservatism and simplicity.
- Several references ([10] [16] [17] [18] [20]) about torsional vibration in turbomachines report, with few detail differences, another source of damping, called “load damping” or “quasi-static damping”, arising from rotating bladed wheels, i.e. fans, impellers and propellers. Corbo and Malanoski [10] referring to earlier studies mention typical values: 5-10% for marine propellers; 0.15-0.50% for turbines; 0.2% for centrifugal compressors.

The authors of the present study are convinced, based on their experience, that this form of damping cannot be neglected in centrifugal pumps and is actually the dominant source in directly-driven trains. Hence the fundamentals of the torsional load damping theory, as applied to the pump investigated in this lecture, are summarized in APPENDIX.

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TESTING

This study represents an attempt to bridge the gap between theoretical knowledge and experimental results concerning centrifugal pumps torsional rotordynamics. As a part of a large research and development program for the Pipeline Research Council International (PRCI), several tests were conducted on a single-stage (between-bearing) model pump in the test laboratory. The possibility to execute a large number of accurate measurements of physical variables in a controlled R&D environment was the essential prerequisite for verifying the correctness of commonly applied engineering assumptions about torsional excitation and torsional damping. In addition, design calculations carried out with preliminary, partial data showed the 1st torsional natural frequency and the blade passing frequency were in sufficient proximity, to make this pump a very interesting test case.

Model Pump

The pump is made of high-precision milled aluminum to reduce the manufacturing tolerances and allow accurate and repeatable hydraulic performance tests. The waterways consist of a double-suction centrifugal impeller fit into a double-volute casing. The main hydraulic design parameters, which are reported in the table below, are similar to many crude oil pipeline pumping applications.

<table>
<thead>
<tr>
<th>Impeller blade number</th>
<th>Impeller specific speed</th>
<th>Volute-impeller distance gap B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$z_b$</td>
<td>$n_q$</td>
<td>12%</td>
</tr>
<tr>
<td>7</td>
<td>33</td>
<td></td>
</tr>
</tbody>
</table>

The blades on the two impeller halves are not circumferentially staggered and no center rib is present on the hub. Figure 1 shows the cross-sectional view of a typical pump used for water and oil transport with the same hydraulic design as the one under test.

A proper stagger angle between the blades causes a phase shift between the two sides of the impeller, thus lowering the magnitude of the hydraulic pressure pulsations and of the blade passing forces [11].

Test Loop

The pump operates in an oil-filled closed loop, consisting of steel pipe, an ultrasonic flow meter, a water cooler heat exchanger and a control valve, see Figure 2. A 3 phase, 50 Hz, 4 pole electric induction motor drives the pump at almost constant speed, about 1495 rpm. The tests were performed with mineral oil at four kinematic viscosities ($\nu$) between 90 and 500 cSt and flow rates ranging from 10% to 105% of the best efficiency point (BEP) flow.

The oil type and its standard viscosity grade were carefully chosen to reflect most common pipeline conditions. Starting from a suction temperature of 20 °C, the oil was progressively heated up to achieve the desired viscosity grade by operating the pump in the closed loop at minimum cooling capacity. The matrix of the four viscous tests, with the relevant temperature-dependent oil properties is shown below. The reported values are nominal, so they do not account for the instrument accuracy and for the inevitable fluctuations experienced during the tests.

<table>
<thead>
<tr>
<th>Test ID</th>
<th>Speed - n [rpm]</th>
<th>Fluid</th>
<th>Temperature – T [°C]</th>
<th>Density – $\rho$ [kg/m$^3$]</th>
<th>Kinematic Viscosity – $\nu$ [cSt]</th>
</tr>
</thead>
<tbody>
<tr>
<td>VT-1</td>
<td>1495</td>
<td>Oil</td>
<td>20</td>
<td>887</td>
<td>500</td>
</tr>
<tr>
<td>VT-2</td>
<td>1495</td>
<td>Oil</td>
<td>25</td>
<td>884</td>
<td>350</td>
</tr>
<tr>
<td>VT-3</td>
<td>1495</td>
<td>Oil</td>
<td>35</td>
<td>878</td>
<td>200</td>
</tr>
<tr>
<td>VT-4</td>
<td>1495</td>
<td>Oil</td>
<td>50</td>
<td>868</td>
<td>90</td>
</tr>
</tbody>
</table>

Figure 1: Pump sectional drawing (typical)
Instrumentation

All instruments directly measuring or used to determine pump hydraulic performances were calibrated per ISO 9906 Grade 1. These included flow, power, suction and discharge pressure, pump speed and fluid temperature sensors.

Oil flowrate, viscosity and density were measured with an inline ultrasonic flow meter equipped with high-resolution temperature and pressure sensors. Resistance temperature detectors (RTDs) measured the fluid temperatures at the pump suction side, at the discharge side and at the flow meter locations.

A slip ring torque meter, mounted in a “floating” configuration between the pump and the motor (Figure 3-left), measured torque and speed (through a digital encoder), namely the shaft power. In this mounting arrangement the two shaft ends of the torque meter are connected to the motor and pump shafts with flange couplings. For brevity, in the following pages the entire torque meter rotating assembly will be referred to as “coupling”. The torque meter accuracy, declared by the manufacturer as a percentage of the nominal full scale reading, is ±0.45% for the speed and ±0.90% for the torque, resulting into a combined accuracy of ±1.0% for the power.

An optical sensor and a keyphasor (redundantly) measured the shaft speed on the motor end side of the torque meter (Figure 3-right).
Performance Curves

Pump performance as well as several static and dynamic measurements were taken, including: static and dynamic pressures at the pump suction and discharge sides; axial static (thrust) and dynamic forces; and radial static (thrust) and dynamic forces.

The head coefficient $\psi$ and the efficiency $\eta$ are plotted against the flow coefficient $\varphi_2$ in Figure 4. The definitions of these and other non-dimensional coefficients are given in the NOMENCLATURE. It can be observed that decreasing viscosity and for the same flow coefficient, the pump head coefficient increases, and efficiency increases, since the absorbed power (and torque) decreases.

As expected, the best efficiency point shifts to higher flow rates as the fluid viscosity decreases and the corresponding peak efficiency $\eta$ goes up.

![Figure 4: Pump performance curves - head coefficient ($\psi$) and efficiency ($\eta$) vs. flow coefficient ($\varphi_2$)](image)

Two important concepts for the pump operation within its hydraulic coverage are based on relative flow $q^*$:

$$ q^* = Q/Q_{\text{BEP}} = \varphi_2/\varphi_{2,\text{BEP}} $$

- The Preferred Operating Range (POR) is the region around the best efficiency point where long-term satisfactory operation is ensured. API 610 [1] defines it as the flow range from 70% to 120% of the BEP. Due to motor power limitation, relative flows above 105% could not be achieved, hence the POR for this pump is $q^* = [0.7 – 1.05]$.

- The Allowable Operating Range (AOR) is the region where pump operation is still permitted but sub-optimal due to technical reasons, most of all increased level of vibration. Generally, the shaft or the bearing housing radial vibrations measured during the performance test are used to define it. The AOR for this pump is $q^* = [0.5 – 0.7]$. 
DATA PROCESSING

All dynamic signals were sampled at 5120 Hz (yielding a 2000 Hz bandwidth) using a Data-Acquisition System with default anti-aliasing filter settings. Data were collected for each flow rate lasting a four minute (240 second) time period.

The current work focuses mainly on the dynamic torque signal. Its correlation with other dynamic signals, such as forces and pressure pulsations, although interesting, is beyond the scope of this lecture.

Static and Dynamic Torque

The torque meter shown in Figure 3-left measured the complete static and dynamic torque. This signal went through an acquisition and a digital post-processing stage, which decoupled the AC component (dynamic alternating torque) from the DC component (static torque), used for performance calculation, namely power and efficiency.

The influence of both the flow rate and of the viscosity on the torque signals is investigated. Figure 5 reports the non-dimensional static and dynamic torque measured at the coupling, in function of the relative flow at the four fluid viscosities. No uncertainty information is shown in Figure 5 and in the following figures. An approximate assessment of the overall experimental uncertainty, including measurement and data processing sources, is presented at the end of this Chapter.

The static torque is graphed in Figure 5-a in a normalized form with respect to the corresponding value at the best efficiency point. The static torque at BEP changes slightly with the oil viscosity, amounting on an average to 520 Nm. The curves show the increase of absorbed torque and power with increasing flow rate. The decrease in oil viscosity has also an effect, in that it decreases the fluid losses and the absorbed torque, leading to an improved efficiency (Figure 4).

The overall alternating torque in Figure 5-b is in a normalized form with respect to the static torque [Nm], to enable a comparison between different tests as well as with typical values found in literature, expressed in [% 0-peak]. The effect of the flow on the overall torsional vibration is qualitatively the same as observed for the measured pressure pulsations and for the radial and axial forces (not discussed in this document): as the flow reduces, the vibration increases due to hydraulic phenomena related to partload recirculation, which increases the excitation magnitude. The overall torque fluctuation measured at the coupling grows from roughly 4% at BEP, to 6% at q* 0.7 (POR), and 8% at q* 0.5 (AOR).

Oil viscosity has only a marginal impact on the measured torsional vibration, showing a sort of “turning point” at q* 0.5, whose interpretation would require an in-depth hydraulic analysis, which is beyond the current scope.

From the overall dynamic torque signal, it is not entirely possible to get insight on the steady-state sources of excitations, in particular on the incidence of hydraulic phenomena. Therefore, analysis in the frequency domain is necessary.

![Graphs](a) Static Torque Ratio \( M_{c,s} = f(q^*) \)

![Graphs](b) Dynamic Torque as % of the static torque \( M_{c,o}[overall] = f(q^*) \)

Figure 5: Non-dimensional static (a) and dynamic (b) overall torque vs. relative flow
Dynamic Torque Frequency Spectra

The frequency content of the dynamic torque signals is obtained with the Fast Fourier Transform (FFT). Figure 6 shows the dynamic torque spectra for various flow ratios ($q^* = 0, 0.2, 0.4, 0.6, 0.8, 1.0$) and oil viscosity equal to 500 cSt and 90 cSt. Similar results are found for 350 and 200 cSt, but are not reported for space limitations.

Though the theoretical bandwidth is 2000 Hz, the spectra are shown up to a maximum frequency of 500 Hz, which contains more than 95% of the sampled signal frequency content.

The graphs are dominated by a “hump” at about 143 Hz, the measured torsional natural frequency (TNF), and by a component at 175 Hz equal to 7 times running speed (7X), associated to the impeller blade passing frequency (BPF). The measured separation margin between these two frequencies is:

$$ SM_m = \left| f_n / f_{ex} - 1 \right| = \left| \text{TNF} / \text{BPF} - 1 \right| = \left| 143 / 175 - 1 \right| = 18.3\% $$ (2)

This separation margin value is significantly in excess of the initial estimate based on preliminary information, confirming the observations of Wang et al. [4] about the impact of model uncertainties on the torsional natural frequency prediction.

Another prominent peak is located at 350 Hz, coinciding with two-times the blade passing frequency. According to some references (e.g. [10]), this might be explained by the interaction of the impeller with a double-volute casing.

Both the region around the TNF and the low frequency region rise at partload in consequence of the broadband hydraulic excitation.

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It is difficult to identify other components at discrete frequencies from the linear scale plots (Figure 6). Using semi-logarithmic plots and replacing the frequency with orders of the shaft speed (Figure 7) has many advantages:

- It enables a better visualization of other discrete and broad-band components. Non-negligible peaks are visible about the BPF and 2xBPF frequencies, which suggest possible modulation phenomena (1X sidebands), see for instance the 8th and 15th harmonics of the running speed.

- It displays more clearly the modal information, namely the torsional natural frequency (position and peak) and its amplification factor, which is inversely proportional to the damping factor. The spectra exhibit a noteworthy resemblance to the frequency response function of single degree of freedom (SDOF) system with low damping.

- It also makes evident the role of the flow rate in lifting the “noise floor” across the whole frequency range at partload. Remarkably sufficient energy is contained in the broad-band hydraulic excitation to wake up the 1st torsional mode of vibration without the need of an external or internal forcing frequency tuning with it and causing resonance. Even if lightly damped, this is normally not the case for the structural vibration modes, which have to be directly excited on their frequency to “express themselves”, e.g. with Experimental Modal Analysis (EMA) methods based on narrow-band techniques (e.g. shakers) or broad-band techniques (e.g. impact hammers).
Torque Amplitude at Discrete Frequencies

Order-tracking filters are a special form of narrow-band pass filters used to extract the amplitudes of the frequency components which are integer multiples of the shaft running speed (1X). In this case the filters captured the amplitudes of all discrete harmonics of the shaft speed up to the 20th (equal to 500 Hz/25 Hz).

The results at varying flow and at fluid viscosities of 500 and 90 cSt are reported only for 1X and 2X, blade passing frequency and twice blade passing frequency (Figure 8). They are explicitly mentioned in the API 610 standard [1] as excitation orders to be considered both in the natural frequency analysis and in the steady-state response analysis.

- Figure 8 shows the amplitudes of these discrete components and of the overall dynamic torque overlaid in the same semi-log graph for comparison. Except for the blade passing, the other components are generally below 1%.

The 2xBPF component is small but not negligible and slightly exceeds 1% of the static torque at flow rates far below the allowable operating range (AOR) for an oil viscosity of 90 cSt. Near the BEP the blade passing is by far dominating, in agreement with Güllich [12]. Outside the POR ($q^*<0.7$), the spread between the overall torque and the blade passing component is due to the broad-band “noise” of hydraulic origin, most evident in the low frequency region and in the vicinity of the TNF.
Figure 9 shows essentially the same information but groups the results relative to each discrete frequency component for all the four oil viscosity tests in the same (linear scale) graph. In this way the differences due the viscosity can be better appreciated. The 1X and 2X components follow qualitatively the same trend as the overall vibration (Figure 5-b), although they are quantitatively much smaller (<0.5% inside the AOR). Since these frequencies are well below the torsional natural frequency, their amplification factor is almost 1, hence the numbers graphed should provide a good approximation of the corresponding excitation magnitudes. The BPF component does not show the same monotonically decreasing behavior with respect to flow observed for the overall, 1X and 2X. The 2xBPF may provide some insight into the interaction of the impeller with the double-volute casing. For high viscosities, the 2xBPF trend monotonically decreases with the flow, while for low viscosities, local minima appear.

Compared to flow, fluid viscosity has a less clear effect on the measured dynamic torque. This might be interpreted as the result of two opposite tendencies: the lower the viscosity, the lower the static torque load, and (possibly) the lower the excitation magnitude; but also the lower the damping.

Figure 9: Dynamic torque components at 1X (a), 2X (b), BPF (c) and 2xBPF (d) vs. relative flow
Modal Parameter Extraction

The peak observed in all the spectra at 143 Hz looks substantially different than the sharp peaks at the discrete orders (Figure 7), which is the reason why some literature refers to it as “hump”.

The analysis of the phase relation between the torque signal and the keyphasor signal proves that this is the rotor fundamental (1\textsuperscript{st}) torsional natural frequency. The cross-correlation frequency function in Figure 10 shows not only an evident amplification of the magnitude across the frequency of 143.4 Hz, but also a steep, nearly 180° phase change.

Extraction of the modal parameters is not a straightforward process for multi-degree of freedom (MDOF) vibrating systems, requiring the usage of advanced numerical tools, well beyond the scope of this study. In case of torsional rotor vibration, a difficulty is that the system’s excitation, e.g. mechanical or hydraulic, is unknown. Hence modal extraction cannot be performed on the Frequency Response Function (FRF, i.e. Bode plot) but has to be performed directly on the response spectrum, which contains the broadband and the discrete sources of excitations.

For the broadband source, typically of hydraulic nature, a reasonable engineering assumption is that the magnitude does not change significantly in the frequency range of interest for the modal parameter extraction. This source constitutes the so-called noise floor of the analyzed signal and is essentially the hydraulic equivalent of the “white noise” obtained with an impact hammer.

For the discrete sources, the hypothesis is that they are integer harmonics of the rotational speed, so can be captured by means of order-tracking filters and then rejected. The problem of using multiple band-reject filters is however that, for the 5\textsuperscript{th}, 6\textsuperscript{th} and 7\textsuperscript{th} harmonics, they would not only remove the orders of the excitation but also a big part of the underlying modal peak. Alternatively, a data smoothing algorithm based on moving average has provided satisfactory results. Subsequently the spectral data are curve-fitted to the classical SDOF amplitude function:

\[
M(f) = M_0 \left[ \left( 1 - \left( \frac{f}{f_n} \right)^2 \right)^2 + \left( 2 \cdot D \cdot \frac{f}{f_n} \right)^2 \right]^{-1/2} \tag{3}
\]

The torsional natural frequency is indicated with \(f_n\), while \(D\) is the damping factor. The parameter \(M_0\) denotes the asymptotic response at frequencies well below the TNF. In order to minimize the uncertainty carried by this parameter when evaluated from the noisy low-frequency region (as per its definition), the above expression is replaced using the amplitude of the natural frequency peak \(M_R\), much easier to identify from the spectra of Figure 6 and Figure 7:

\[
M(f_n) = \frac{M_0}{2 \cdot D} = M_R \rightarrow M(f) = 2 \cdot D \cdot M_R \cdot \left[ \left( 1 - \left( \frac{f}{f_n} \right)^2 \right)^2 + \left( 2 \cdot D \cdot \frac{f}{f_n} \right)^2 \right]^{-1/2} \tag{4}
\]

The equation (4) is curve-fit to the full data set, to extract the modal parameters. In Figure 11 is just an illustrative example from the test VT-3 - 200 cSt and relative flows \(q^*\approx 1, 0.8\) and 0.2.

![Figure 10: Torque vs. Keyphasor signal cross-correlation](image1.png)

![Figure 11: Example of curve-fitting for modal parameter extraction](image2.png)
This procedure is applied for all test points. For the lowest flowrate however \( (q^*<0.5) \), the index of the fit goodness worsens, meaning higher uncertainty in the modal parameter extraction. For this reason it was not possible to confidently extract damping out of the test VT-4 – 90 cSt, while identifying TNF was still possible. The results have been plotted versus the relative flow and the kinematic viscosity, respectively in Figure 12 and Figure 13.

The dependency of the torsional natural frequency upon the flow is evident in Figure 12-a. It can be proven that the changes of the rotor material shear modulus \( (G) \) with the temperature are too modest to justify the observed TNF changes. A more plausible physical explanation is that, as the flow decreases, the amount of added fluid interacting with the rotor and participating in the vibratory motion increases due to flow separation (e.g. during recirculation), therefore the torsional natural frequency slightly drops.

In the damping plot (Figure 12-b) some oscillations with the flow appear, with local maxima located at \( q^* \) between 0.6 and 0.8 depending on the oil viscosity. No physical explanation can be provided about this behavior.

Both results do show a relationship with oil viscosity. For simplicity, the same modal parameters have been plotted against oil viscosities at the relative flows \( q^* \approx 1, 0.75, 0.5, 0.25 \) in Figure 13.

![Figure 12: Estimated torsional natural frequency (a) and estimated damping factor (b) vs. relative flow](image1)

![Figure 13: Estimated torsional natural frequency (a) and estimated damping factor (b) vs. oil viscosity](image2)
The natural frequency shows a decreasing trend with increasing viscosity. This is believed to be the combined effect of the increase of the added inertia (due to thicker boundary layers), and of the increase of damping. In fact, the damped natural frequency, corresponding to the maximum response, is linked to the undamped natural frequency by the following equation:

\[ f_n = f_{n,u} \cdot \sqrt{1 - D^2} \]  

(5)

The torsional damping factor shows an increase with viscosity at relative flows higher than 0.5. Near the BEP, damping drops from 3.7% to 3.1% as the oil viscosity goes from 380 cSt in VT-1 to 145 cSt in VT-4. This finding shows the importance of the fluid viscosity in the estimation of the damping factor: values of 1% for similar cases (e.g. crude oil pipeline pump) would be too conservative.

Uncertainty Assessment

The uncertainty inherent to the curve-fitting technique represents an important caveat, so an error assessment is discussed here below (for VT-3 -200 cSt only). Relative errors, i.e. normalized to their nominal parameters, are presented.

The curve-fitting method consists in finding the values for the three parameters (natural frequency, natural frequency peak and damping ratio) that minimize the least square error, providing the base for the total or combined standard deviation. This increases significantly (Figure 14) at low flow rates \(q^*<0.5\), meaning the spectra cannot be satisfactorily approximated with SDOF functions, hence the risk of inaccurate inferences on the modal parameters, especially damping.

The standard deviations on the three individual parameters can be also calculated through statistics. As can be seen from Figure 14, the uncertainty on the torsional natural frequency is very small, meaning this parameter might be even removed from the curve fitting and basically estimated “by eye” from the FFT. On the opposite side, damping uncertainty is the largest portion of the combined uncertainty, attaining almost 9% at minimum flow. For practical purposes however, a relative standard uncertainty of ±3% can be attributed to the damping factors estimated within the pump AOR.

This value does not take into account other sources of errors that, propagated, negatively impact the estimate of the modal parameters. These sources are mainly the instrument accuracy (e.g. torque meter ±0.9%), the repeatability of the test conditions, the quantization error during the signal acquisition, and the computational errors during the signal processing (e.g. FFT). A coverage factor of 2 for a 95% confidence interval is applied to cover these additional uncertainty sources. Hence the so-called “extended” uncertainty in the modal parameter extraction (most of all damping) is 2*4%=8%.

![Figure 14: Estimated relative errors due to the curve fitting](image-url)
TORSIONAL ANALYSIS

In order to derive the equivalent torsional excitation from the measured dynamic torque, a sufficiently accurate model needs to be developed for the entire shaft line, including pump, instrumented coupling and electric motor. The model allows to mathematically relate the dynamic torque measured at the coupling, properly analyzed in its frequency content through the FFT, with the excitation torques arising from other locations on the shaft line, e.g. pump impeller.

This process is relatively straightforward for the hydraulic and electric (not considered here) sources of excitations, for which the acting location is known. For other excitation sources carrying higher uncertainty, e.g. 1X and 2X (see LITERATURE REVIEW), the acting location is chosen with a worst-case approach, based on the maximum sensitivity of the model at that given frequency.

Model Tuning

Using the best available nominal information from the motor and coupling vendors (uncertainty figures were also provided) and from the pump impeller and shaft CAD model, an initial estimate of the torsional natural frequency is arrived at. The declared relative uncertainty on the coupling torsional stiffness is ± 10% of the nominal value. This reflects possible deviations in the floating mounting method of the torque meter assembly, which is in agreement with the outcome of a recent survey on coupling vendors [4].

The angular mode shape in Figure 15 is characterized by the largest angular deflection towards the pump non-drive end, near the impeller (critical location or mode anti-node), while the center of the coupling has almost zero angular deflection (sensitive location or mode node) but large relative twisting, making it the ideal place for dynamic torque measurement. Due to its comparatively large polar inertia, the motor rotor has a low degree of participation to this mode, aside for some angular deflection at its drive-end, near the cooling fan. The driving parameters of the model are therefore the pump polar inertia and the coupling torsional stiffness.

<table>
<thead>
<tr>
<th>Motor rotor inertia ( I_{m,c} ) [kg m(^2)]</th>
<th>Coupling torsional stiffness ( K_t,c ) [Nm/rad]</th>
<th>Pump impeller inertia (dry) ( I_{p,i} ) [kg m(^2)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.8 ± 2%</td>
<td>2.0 ( \cdot 10^5 ) ± 10%</td>
<td>0.1209</td>
</tr>
</tbody>
</table>

There are two important observations regarding the preliminary estimated value of the TNF at 163.3 Hz (Figure 15-top):

- The calculated torsional natural frequency, using an untuned dry model, has less than 10% separation to the blade passing frequency. This does not fulfill API 610 [1] and requires a steady-state response analysis:
  \[
  SM_{p_1} = \left| f_n/f_{ex} - 1 \right| = \left| TNF_1/BPF - 1 \right| = \left| 163.3/175 - 1 \right| = 6.7\%
  \] (6)

- The first predicted natural frequency deviates from the experimental value (143.5 Hz is assumed as an average value) by 13.8%. The discrepancy is even larger on the separation margins, with a measured value from equation (2) (18.3%) almost 3-times larger than the predicted one. This makes it necessary to refine and tune the model to higher inertia/lower stiffness.

![Figure 15: Torsional natural frequency rotordynamic predictions: untuned (top) and tuned (bottom) models](image-url)
**Equivalent Reduced Model**

Even though the initial prediction is quantitatively inaccurate, it is qualitatively right in highlighting that the pump torsional system behaves like a simple 2-disc system with respect to the 1st TNF. According to Vance et al. [21], this reduced model (or even the one with just 1 disc, due to the generally larger motor inertia) is found to be a valid approximation in many examples of rotating machinery.

![Diagram](image)

**Figure 16: Fundamental train torsional natural frequency (from [21]): “real” system (left) and equivalent models (right)**

With reference to this reduced model (Figure 16-right), equation (7) can be rearranged to determine the torsional stiffness that the equivalent 2-disc system would have for a given natural frequency. The untuned dry model has an equivalent torsional stiffness of:

\[
K_{eq} = (2\pi \cdot f_n)^2 \cdot I_{eq} = (2\pi \cdot TNF_1)^2 \cdot \frac{I_p \cdot I_M}{I_p + I_M} = (2\pi \cdot 163.3)^2 \cdot \frac{0.1209 \cdot 2.8}{0.1209 + 2.8} = 1.22 \cdot 10^5 \text{ Nm/rad}
\]

This stiffness is lower than the coupling one, meaning the pump and motor shaft cannot be considered torsionally rigid.

**Impeller Added Fluid Inertia**

In the 2nd step the added fluid contribution to the pump impeller is taken into account. Several methods are available to calculate this effect, having quite different orders of accuracy and of engineering effort.

a. Using the dry (metal-only) impeller inertia value, routinely calculated with modern 3D-CAD models. Although this seems highly inaccurate, such assumption is in good agreement with recent experimental findings by Marscher et al. [22], and is suggested by the Hydraulic Institute 9.6.8 – 2014. It can be noticed that, by following this simplified approach, the torsional natural frequency (TNF₁) was substantially over-predicted.

b. Using a fraction of the entrained fluid on the base of non-dimensional impeller parameters, such as the number of blades \(z_0\), the specific speed \(n\), and other geometrical values \(d_1/d_2, b_2/d_2\) etc., see for instance Nordmann et al. [20] and van Wijk [23].

c. Using the full impeller enclosed fluid inertia, e.g. Wang et al. [4], which is equivalent to assume the fluid is solidly rotating and vibrating together with the impeller. This leads to an overestimate of the wet impeller inertia, which depends mainly on the impeller geometry.

The authors decided to not pursue any of these approaches. To obtain a precise estimate, a modal finite-element analysis (FEA) is carried out on the impeller geometry connected to a torsional spring of equivalent stiffness, such that the 1st torsional natural frequency from the untuned model is reproduced, see Figure 17-left. For simplicity the motor rotor, on the other side of the torsional spring, is modeled as a lumped point mass with polar inertia equal to 2.8 kg m². The numerical value of the torsional stiffness is extremely close (+0.3%) to \(1.22 \cdot 10^5 \text{ Nm/rad}\) per equation (6).
The natural frequency calculated by the finite-element program is representative of the in-vacuum or dry free torsional vibration. To simulate the presence of the fluid and its interaction with the vibrating structure (Fluid-Structure Interaction, FSI), a possible way is to use a class of fluid elements called acoustic elements which can be directly coupled to the structure for modal or response analyses. This method is less computationally expensive than a full coupling with the fluid dynamic solver and has received special attention in the field of hydraulic turbines, e.g. Hübner et al. [24]. The numerical code solves for the primary coupled quantities, displacement for the structural field and pressure for the acoustic field. The acoustic field is solved using the potential flow theory, which applies to non-viscous, non-dissipative fluid flows. Although “ideal”, this assumption is of practical significance for many FSI problems, where the target is a realistic estimate of the added fluid effects, e.g. stiffness (Lomakin’s effect in pumps), damping and mass.

Adding acoustic fluid elements to the volume inside and outside the impeller and performing the modal FEA-FSI analysis yields a wet natural frequency of 146.1 Hz (Figure 17-center), which is very close to the experimental torsional natural frequency. The value of the wet (impeller + added fluid) inertia can be estimated as follows:

$$ p_{\text{wet}} = p_{\text{dry}} \cdot \left( \frac{f_{\text{dry}}}{f_{\text{wet}}} \right)^2 = 0.1209 \cdot (163.37/146.11)^2 = 0.1511 \text{ kg m}^2 $$

The outcome of the FEA-FSI analysis is not very sensitive to the chosen fluid volumes (which in this example are extended from the impeller to the volute-casing) and to the mesh settings, but mainly depends on the fluid acoustic properties, density and speed of sound. The acoustic pressure in the fluid field is shown in Figure 17-right for completeness.

**Coupling Stiffness Adjustment**

The FEA-FSI still slightly over-predicts the torsional natural frequency by 1.8%, indicating an over-estimate of the total equivalent stiffness in the analysis model of 3.6%. To adjust for it, the coupling stiffness has to be reduced by 5.8% from its nominal value, which is well within the accepted tolerance of the supplier information.

Using the wet impeller polar inertia and the adjusted coupling stiffness leads to a prediction of 144.65 Hz, Figure 15 - bottom. This value is deemed close enough (i.e. ± 1%) to the experimental TNF to be able to rely on the model as a diagnostic tool relating the measured dynamic response with the magnitude of applied excitation.

**Torsional Damping**

As a final step, damping is taken into account to avoid the model predicting infinite response at resonance (not practically possible). The analyst has several options at hand:

a. Using the conservative rule of thumb of 1% damping factor. This is acceptable only if no other experimental or design information is available. As seen in the section on Modal Parameter Extraction, damping ratios above 3% are estimated due to the high oil viscosity (Figure 13-b).

b. When design information is available, a more realistic damping factor is obtained by the theory of the “impeller quasi-static damping” valid near the pump BEP. The basics of such theory, i.e. the theoretical derivation and the testing verification, are explained in previous pump literature, and are reported in APPENDIX. By rearranging the respective equation in a non-dimensional form, namely normalizing with respect to the 1st torsional mode critical damping, yields the following damping factor:

$$ D_{\text{th}} = \frac{15}{\pi^2} \cdot \frac{M_{P,S}}{I_p + I_M} \cdot \frac{I_p + I_M}{I_p} = \frac{15}{\pi^2} \cdot \frac{M_{P,S}}{I_p} \cdot \frac{I_{\text{wet}} + I_{m,c}}{I_{m,c}} = \frac{15}{\pi^2} \cdot \frac{520}{1495 \cdot 143.5} \cdot \frac{0.1511 + 2.8}{0.1511 \cdot 2.8} = 2.57% $$

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According to Nordmann et al. [20], the quasi-static impeller damping theory is valid within loosely-defined boundaries of frequency and flow rate, which are mentioned in the APPENDIX. However, the frequency boundaries possibly reflect more the limitations in the testing procedure and equipment than the nature of this source of damping.

c. Whenever possible, using experimentally derived damping values. For the current case, these are shown as a function of the relative flow and of the oil viscosity in Figure 12-b and Figure 13-b, respectively. As long as the excitation frequencies of interest are sufficiently separated from the natural frequency, a very precise quantification of the damping factor is not necessary. By averaging over the entire experimental data, a damping factor of 3.3% is calculated and used for the subsequent steady-state forced response analysis.

**Torsional Steady-State Response**

The damped tuned model is used to carry out the steady-state response analysis with the scope of reproducing the level of torsional vibration measured during the viscous tests. Because the excitation input magnitudes are unknown, this is the inverse problem of a typical response analysis. Since the model is linear, frequency-dependent transfer functions are determined using input “trial” excitations of 1 Nm 0-p. The excitation is applied on the pump impeller (worst case) and the response is read at the coupling:

\[
TF(f) = \frac{\text{Response torque}}{\text{Exciting torque}} = \frac{M_{C,d}(f)}{M_{P,d}(f)} = \frac{M_{C,d,analysis}(f)}{1}
\]  

(12)

The transfer functions for the discrete sources 1X, 2X, BPF (7X) and 2xBPF (14X) are graphed in Figure 18 for the entire train, while the values in the table refer to the dynamic torque response at the coupling. This is the information needed to back-calculate the equivalent torsional excitations, according to the equation (12). The same procedure can basically be extended to any frequency and any other measurement location.

The 1X and 2X have both transfer function values comparable to 1, meaning torsional excitations at these frequencies are transferred “statically” from their acting location to the measurement location.

The blade passing excitation from the impeller is considerably amplified by the proximity to the torsional natural frequency. Finally, the 2xBPF component is reduced to roughly 20% of its excitation value at the impeller, meaning the torsional excitation at this frequency is more than 5-times the value measured at the coupling, thus overcoming the blade passing magnitude itself.

![Graphical and numerical values of the transfer functions](image-url)
ESTIMATED TORSIONAL EXCITATIONS

Back-Calculated Excitation Magnitudes

The values of the transfer functions included in Figure 18 are used to back calculate the expected levels of impeller torque excitation, starting from the amplitudes measured at the coupling (Figure 9). The estimated excitation amplitudes are plotted in Figure 19.

It can be noticed that the 1X and 2X torsional excitations are at the same level as their response counter-parts and are consistently below 1% of the static torque for most of the analyzed flow conditions. A realistic estimate for 1X and 2X excitation proposed in this work for the allowable operating range (AOR) is 0.5%. From their qualitatively similar trend, it can be argued that both have hydraulic origin, and are not associated to any of the mechanical root causes described in the literature.

The blade passing excitation is lower than its response counterpart, due to the large value of the transfer function, amplified by the closeness to the torsional natural frequency. Values as high as 2.3% of the static torque are estimated in the preferred operating range ($q^*: 0.7-1$). These values are in line with literature on hydraulic forces in volute pumps, see G ülich [12] and Blanco [13], while they are significantly above some “old school” approaches still used in the industry, e.g. $1/2z_b\%$ (0.14% for 7 blades) from [8]. The estimated twice blade passing component exceeds in magnitude the blade passing component for most of the analyzed flow range, becoming comparable with the BPF component for $q^*:0.8$. At $q^*:0.5$ the 2xBPF attains a magnitude up to 4.4% (for an oil viscosity of 200 cSt), decreasing sharply when approaching the best efficiency point. Both BPF and 2xBPF are indicative of the values expected when a double suction, non-staggered impeller design is used in combination with a double-volute casing in crude oil applications.

When blade stagger is implemented on a double-suction impeller, hydraulic dynamic torque is expected to have smaller magnitudes than the one estimated in this study. The exact amount, however, is difficult to quantify on the base of the sole geometrical parameters (e.g. stagger angle) and without resorting to testing or CFD. This is due to the still not completely clear mechanism whereby pressure pulsations generated by the rotor-stator interaction determine hydrodynamic forces and moments responsible for structural and shaft vibrations. Pump literature (e.g. Robinett et al. [11]) only reports typical figures for pressure pulsations and radial forces.

![Figure 19: Estimated dynamic excitation torque components at 1X (a), 2X (b), BPF (c) and 2xBPF (d) vs. relative flow](image-url)
Impeller Torque from Transient CFD

The outcome of the previous section triggered further investigation on the impeller torsional excitations at blade passing and twice blade passing frequencies.

A computational fluid dynamic (CFD) analysis was performed on the model pump geometry for an oil viscosity of 90 cSt (VT-4) and for three relative flows \( q^* = 1, 0.6 \) and 0.3. The full pump hydraulic domain comprising impeller, volute casing, side rooms, suction and discharge nozzles was simulated (Figure 20). Time-transient simulations with sufficiently small time steps are required to properly characterize hydrodynamic forces and moments due to the rotor-stator interactions. For \( q^* = 1.0 \) a total of 5 impeller revolutions with an angular increment of 1° per iteration were calculated, amounting to 1800 iterations. For \( q^* = 0.6 \) and 0.3 the number of impeller revolutions was 10 and 20 to take into account the unsteadiness of the flow and the resulting slower convergence.

![Figure 20: CFD fluid domain](image)

The torque signal was generated during post-processing based on the integration of pressure and shear forces on the domain boundaries. The static torque, used to predict the power and the hydraulic efficiency, was in excellent agreement with the measurements. The dynamic torque time waveforms were also calculated, and normalized to the static torque.

The frequency spectra of the impeller dynamic torque at \( q^* = 1 \) and 0.6, shown in Figure 21, exhibit a strong 2xBPF peak, which is dominating at partload and is comparable with the BPF component at the best efficiency point. Dynamic torque results for \( q^* = 0.3 \), not reported for brevity, also follow the qualitative trend just described.

From the quantitative side, a good agreement between the CFD-derived excitation magnitudes and those extracted from the measurement is evident in Figure 19-c and d. The tight manufacturing and assembly tolerances on the model pump are critical for performing the experimental validation of new hydraulic designs, as well as the verification of engineering tools, i.e. CFD and FEA.

![Figure 21: CFD calculated impeller dynamic torque spectra at q* 1.0 (a) and 0.6 (b)](image)
CONCLUSIONS

This paper describes the process of extracting the main torsional vibration parameters of a model centrifugal pump using test data. The API 610 standard is certainly useful when dealing with torsional analyses, providing the requirements that new pump designs must satisfy (e.g. separation margin), but it omits important pieces of information, such as typical torsional excitation magnitudes and typical damping factors. It leaves it entirely to the OEMs to define them, based on their knowledge and experience. Moreover, there are no generally accepted best practices within the technical literature on these subjects, but only rules of thumb, whose indiscriminate application may generate problems and unnecessary work. One of the well-known rules is to assume a torsional damping factor of 1% of the critical value) and dynamic excitation torque of 1% of the static value). In many cases, this approach is over-conservative and could result in high calculated levels of torsional stress, posing an integrity concern. About the hydraulic sources of excitation (blade passing), the opposite problem might be encountered when using a magnitude of 1%, as more recent experimental and CFD studies demonstrate that values up to 5% are possible.

For the present case, the good quality of the dynamic torque measurements allowed to thoroughly investigate the influence of flow and oil viscosity, and to make statistical inferences on the torsional damping and on the torsional excitation. Even though the methodology was applied on a model machine in a controlled R&D environment, the figures on the damping factor and hydraulic excitation (BPF and 2xBPF) are considered to be relevant for crude oil pipeline pumps with similar designs.

Torsional damping was extracted from a frequency domain analysis using data smoothing and curve-fit techniques. The extracted values, which show some dependency on the test conditions (flow and viscosity), range from 2.5% to 4.5%, significantly larger than 1%. As the coupling and the torque meter assembly are torsionally stiff (all-metal with no resilient elements), their contribution to damping is likely negligible. The authors believe the quasi-static load damping represents the main source of damping, followed by others which are more difficult to identify. One of these, directly dependent on the fluid viscosity, might be the damping generated by the disc friction on the impeller shroud (“windage damping” in [10]). The damping figures published herein are affected by a degree of uncertainty up to 8%, propagating from the measurement to the data processing and curve fitting stages.

Dynamic torque at blade passing frequency is prominent in the frequency spectra of the measured torque and is clearly a relevant source of excitation. The torsional analysis model provided the tool to link the measured values at the coupling with the unknown (unmeasured) excitations at the impeller location. This leads to estimates of excitation magnitudes for the 2xBPF which are larger, almost double, than the BPF ones. Values of 2.3% for the BPF component and 4.4% for the 2xBPF are estimated in the pump allowable operating range. While there is a possibility of benchmarking the estimated exciting torque at BPF against literature, no references are available on the 2xBPF component. A subsequent transient CFD analysis fully confirmed the estimated magnitudes at these two frequencies and their driving role. These values, substantially larger than 1%, are specific to the pump under study, first and foremost: no blade stagger between the two halves of the impeller and no center rib. However, neither are these values representative of different pump types (e.g. multi-stage), nor can a strong dependency on other parameters (which are not accounted for or could not be varied in this study) be excluded. In light of the encouraging results, CFD might be used in the future to predict the impeller dynamic torque and get a grasp on other design parameters, such as the number of blades, the stagger angle, the specific speed, the casing type (diffuser or volute), and the radial gap.

The findings relative to 1X and 2X torsional excitations cannot be automatically extended to full-scale machines and full-load scenarios, as these depend on the mechanical setup. From a quantitative point of view, this study corroborates the assumption that the excitation frequency magnitudes at these frequencies are very small (<1%) in directly-driven trains, thus it proposes to downgrade them to “negligible” or to use a still conservative 0.5% when performing torsional response analyses. Qualitatively speaking, the authors would tend to attribute the measured 1X and 2X to hydraulic origin and not to unclear explanations such as “unbalance” or “misalignment”, which have no meaning in the torsional world, except when lateral and torsional vibrations are strongly coupled, such as the case of gear-driven trains.

Starting from the need of an accurate and reliable torsional model for estimating torsional excitation, this document offers some insight into the following topics:

- Impeller wet inertia due to the added fluid. This parameter heavily impacts the TNF prediction accuracy. A numerical method based upon FEA-FSI coupling is briefly discussed.
- Pump load damping. An analytical method valid near the BEP to predict the pump damping factor, on the base of main geometric and operating parameters, is discussed in APPENDIX.

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APPENDIX - LOAD DAMPING THEORY

Centrifugal pump impeller contribution to the overall torsional damping is considerable and should be correctly taken into account when performing a response analysis in a nearly resonant condition. Damping is one of the three sets of coefficients used to linearly model the dynamic interactions between a solid structure, in this case rotating, and the surrounding fluid. The other two are stiffness and mass. For an impeller the torsional added stiffness part is negligible, while the added inertia part is embedded in the wet inertia of the model. No general methods, suitable for rotordynamic analyses, have been devised so far for the calculation of the full impeller interaction coefficients. However approximate methods exist for the estimation of the added inertia and of the damping.

Pump shafts are primarily loaded by static torque. The amount of torque needed for the pump to operate depends primarily on the flow rate, and obviously on the machine design (number of stages, impeller and casing hydraulic efficiency etc.). For a given design and for practical scopes, the torque-speed characteristic curve can be well approximated with a parabola, see Figure 22.

\[ M_p = a \cdot \Omega^2 \]  

(13)

The quadratic coefficient \( a \) is an increasing function of the relative flow \( q^* \).

![Figure 22: Pump impeller quasi-static damping from the torque-speed curve](image)

Let \( \Omega(t) \) be the instantaneous angular velocity of the pump shaft. This signal can be split in its DC static part \( \Omega_s (=2\pi n \text{ [rpm]} / 60) \) and in its AC dynamic part \( \Omega_d(t) \) (angular vibration velocity [rad/s]), representing a small perturbation. Both quantities can be measured directly using non-contact optical techniques, for example the rotational laser vibrometer described in [4]. The angular vibration velocity is the first time-derivative of the oscillating angle \( \theta(t) \).

\[ \Omega(t) = \Omega_s + \Omega_d(t) \]  

(14)

The quasi-static load damping method is based on a simple but key assumption: that the impeller response to an angular velocity perturbation follows approximately the static torque-speed characteristic curve, as if it would occur in a quasi-static way, i.e. at low frequency. Substituting the instantaneous angular velocity equation (14) in the torque equation (13) leads to:

\[ M_p = a \cdot \Omega^2 = a \cdot (\Omega_s + \Omega_d(t))^2 \]  

(15)

Finally deriving the torque equation (15) with respect to the angular vibration velocity yields the torsional damping coefficient:

\[ C_p = \frac{\partial M_p}{\partial \Omega} = \frac{\partial M_p}{\partial \Omega_d} = 2 \cdot a \cdot (\Omega_s + \Omega_d(t)) \]  

(16)

The equation (16) highlights the non-linear and frequency-dependent nature of the pump impeller damping, and the difficulty to account for it through a fixed, i.e. speed and frequency independent, coefficient. Neglecting the vibrating term, a rough estimate of the quasi-static (i.e. truncated at the \( \theta^0 \) order) pump impeller damping is obtained:

\[ C_p \cong 2 \cdot a \cdot \Omega_s = 2 \cdot \frac{M_p}{\Omega_s} = \frac{60}{\pi} \cdot \frac{M_{ps}}{n} \]  

(17)

The equation (17) is reported as-is by Corbo and Malanoski [10], Frei et al. [17] and Vance [18] for impellers, while Ehrich [16] includes an empirical multiplier factor 1.5-2 for propellers. Nordmann et al. [20] state this equation can be used to make conservative torsional damping estimates, but warn about its range of validity. Based on their test findings, equation (17) would be valid for frequencies up to about 4-times
the rated running speed of the pump (4X), and for flow rates near the best efficiency point (Figure 22).

To enable a direct comparison with the dimensionless damping factors experimentally derived in the present study, the pump damping coefficient from equation (17) is divided by the critical damping coefficient.

$$C_{cr} = 4 \pi \cdot f_n \cdot I_{eq}$$  \hspace{1cm} (18)$$

The equivalent “modal” inertia, valid for the 1st torsional natural frequency, is derived from the 2-disc torsional system (Figure 16-right):

$$C_{cr} = 4 \pi \cdot f_n \cdot \frac{I_p + I_M}{I_p \cdot I_M}$$  \hspace{1cm} (19)$$

The theoretical damping factor \(D_{th}\), due to the pump torque load, is finally calculated dividing equation (17) by equation (19):

$$D_{th} = \frac{C_p}{C_{cr}} = \frac{15}{\pi^2} \cdot \frac{M_{p_s}}{n \cdot f_n \cdot \frac{I_p + I_m}{I_p \cdot I_M}}$$  \hspace{1cm} (20)$$

NOMENCLATURE

AC
Alternating Current (used with the general meaning of “alternating” or “dynamic”)

AOR
Allowable Operating Range

BEP
Best Efficiency Point

BPFE
Blade Passing Frequency

CFD
Computational Fluid Dynamics

DC
Direct Current (used with the general meaning of “mean” or “static”)

DOF, SDOF, MDOF
Degree of Freedom, Single Degree of Freedom, Multiple Degrees of Freedom

FEA
Finite Element Analysis

FFT
Fast Fourier Transform

FSI
Fluid-Structure Interaction

POR
Preferred Operating Range

TNF
Torsional Natural Frequency

VT
Viscous Test

\(b_2\) [m]
Impeller outlet width

\(C_p, C_v\) [Nm/s/rad]
Pump impeller damping coefficient, critical damping coefficient

\(d_2, d_3\) [m]
Impeller outlet diameter, volute cut-water diameter

\(D, D_h\) [-]
Torsional damping factor, theoretical (predicted) torsional damping factor

\(f, f_n, f_c\) [Hz]
Frequency, natural frequency, exciting frequency

\(K_c, K_{eq}\) [Nm/rad]
Coupling torsional stiffness, equivalent torsional stiffness (reduced model)

\(H\) [m]
Pump total differential head

\(I_{p_s}^{dry}, I_{p_s}^{wet}\) [kg m²]
Polar mass moment of inertia of: whole pump, pump impeller dry, pump impeller wet

\(I_{s_w}, I_{s_u}\) [kg m²]
Polar mass moment of inertia of: whole motor, motor core

\(M_c, M_{c_2}, M_{c_d}\) [Nm]
Coupling torque, static, dynamic

\(M_p, M_{p_s}, M_{p_d}\) [Nm]
Pump torque, static, dynamic

\(n\) [rpm]
Running speed

\(Q\) [m³/s]
Pump flow rate

\(s_b\) [-]
Number of impeller blades

\(v\) [cSt]
Fluid (oil) kinematic viscosity

\(\rho\) [kg/m³]
Fluid (oil) density

\(\Omega, \Omega', \Omega\) [rad/s]
Angular velocity, static, dynamic

\(\text{gapB} = \frac{d_3}{d_2} - 1\)
Radial distance between impeller outlet and volute

\(u_2 = \frac{\pi}{60} \cdot n \cdot d_2\)
Impeller outlet peripheral velocity

\(n_s = \frac{n}{\sqrt[3]{H/0.75}}\)
Impeller specific speed (Q/2 for a double-suction impeller)

\(\varphi_2 = \frac{Q}{(\pi \cdot d_2 \cdot u_2)}\)
Flow coefficient (Q/2 for a double-suction impeller)

\(\psi = 2 \cdot g \cdot H/\omega_2^2\)
Head coefficient

\(q^* = \frac{Q}{Q_{BEP}} = \varphi_2 / \varphi_{2BEP}\)
Relative flow

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Literature