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Calculating Torsional Vibrations in Drives with Hydrodynamic Couplings

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1. Introduction

In order to ensure high reliability and safety for a drive system, it is essential that resonance ranges and component loads are calculated in advance. This is achieved by simulation calculations which are carried out with more or less drastically simplified versions of the actual systems behaviour. In this case, the simplified systems behaviour is represented by mathematical models. Such models exist for many drive line elements which mostly originate from physical basics, for example Hooke's law.
Hydrodynamic couplings, however, are highly complex in their operation. Modelling across larger operating areas is therefore a very challenging task [1, 2, 3]. But in most cases, it is sufficient to limit the mathematical description to one operating range, the nominal operating point. Under these conditions, a rather simple description of hydrodynamic couplings was developed in the form of a Kelvin model [4], which can be easily determined and also reproduced with any standard simulation software program.
The validity of this Kelvin model is to be proven once more in this paper, as well as the fundamental characteristics of hydrodynamic couplings derived from it. Moreover, two basic practical problems for the design and simulation of drives with hydrodynamic couplings, especially marine drives, are explained.

- Simulation calculations in resonant ranges (second mode or higher) repeatedly show wide fluctuations between theory (simulation) and actual measurements, so that there are insecurities regarding the validity of coupling models in this area.

- In the past it was standard procedure that drives with hydrodynamic couplings were designed separately. Primary and secondary driveline were regarded independently, eigenfrequencies and torsional loads were determined separately. Over the years, doubts were, however, raised, whether hydrodynamic couplings might not have a much stronger influence on the eigenfrequencies and the torsional loads of primary and secondary drivelines after all, which would mean that separate designs could lead to major failures. For this reason, such systems are now no longer separated but designed as complete units.

2. Hydrodynamic coupling as Kelvin model

A stationary design of a hydrodynamic couplings is carried out as described in Equ. 1.

$$T = \lambda \cdot \rho \cdot D_P^{\,5} \cdot \omega_P^2 \qquad \text{Equ. 1}$$

The non-dimensional performance figure λ is in this case dependent on slip, the profile parameters and the degree of filling, and is obtained by experiment. It flows, like the density of the operating medium ρ, linearly into the calculation of the transmittable coupling torque.
With geometrically similar couplings, the course of the performance figure λ does not change, so that this torque, with the diameter of the pump impeller D_P and the pump speed ω can also be theoretically determined for other models and input speeds.

However, non-stationary procedures cannot be calculated with Equ. 1. For this, a different description is required, which has been developed in [4] in the form of a Kelvin model.

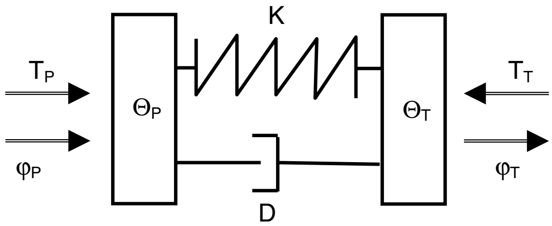

Fig 1: Kelvin Model (Indices: P = Pump; T = Turbine)

As is generally known, this model consists of a parallel arrangement of a Hooke spring K and a viscous damper D (Fig. 1) and is analogue to the description of highly flexible couplings [5]. For this coupling type, the spring capacity and the damping value are assumed as constant values.

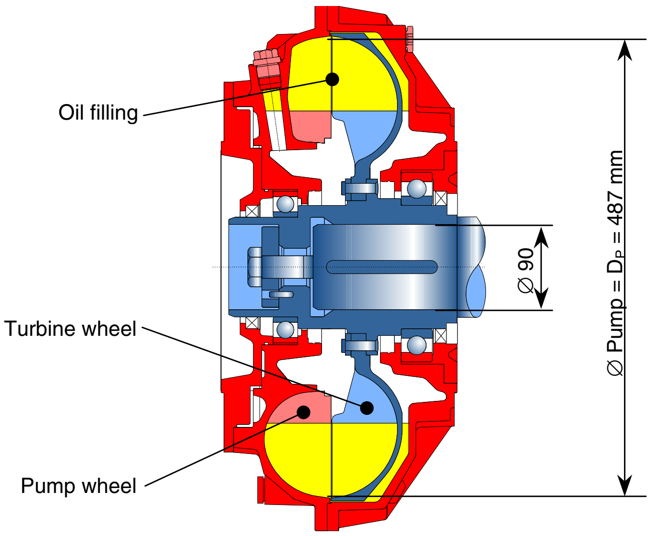

Fig. 2: Cross section of VTC 487 T coupling

The hydrodynamic coupling, however, distinguishes itself by non-linear behaviour even at nominal operating point, which results in frequency-dependent values for K and D. These are exemplarily illustrated for coupling size VTC 487 T (Fig. 2) at a pump speed of 1500 rpm and a nominal torque of 800 Nm in Fig. 3 and Fig. 4. At low excitation frequencies, the stiffness is low and the damping effect high. With increasing frequency, this effect reverses. In both cases, stiffness and damping effect are striving against a boundary value. Compared to other drive elements, the boundary value of the stiffness is very low. In this case, its value is at 3500 Nm/rad and corresponds, for example, to that of a steel shaft with a diameter of 30 mm and a length of 1863 mm. In a drive system, the stiffness of the other drive elements is at least one power of ten higher than that of the hydrodynamic coupling. Even highly flexible couplings are higher by a factor >3 at an identical nominal torque. This leads to the conclusion that the initial eigenfrequency of a drive is determined by the hydrodynamic coupling.

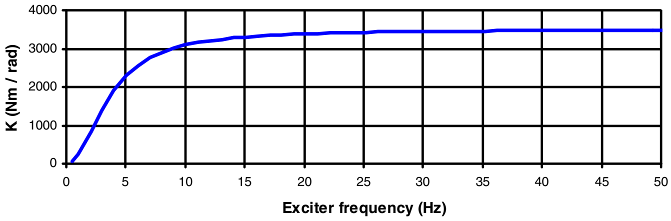

Fig. 3: Stiffness K

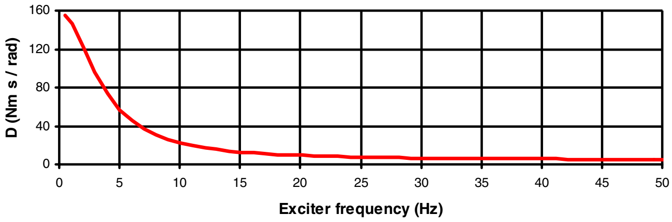

Fig. 4: Damping D

Outside the nominal slip area (from 0% to approx. 10%), the model according to [4] loses its validity and can therefore not be used for calculations of start-up and run-out procedures. Here, other modelling approaches need to be chosen [1, 2, 3], some of which are, however, highly complex. In most cases, it is sufficient to calculate start-up and run-out procedures such as these, which are often quasi-stationary, with the stationary characteristic curve of the coupling.

3. Characteristics of hydrodynamic couplings

3.1 Low-pass behaviour

From the stiffness K and the damping effect D, the transmission behaviour of hydrodynamic couplings can be deduced in the form of an enlargement factor V. It is defined as the quotient of coupling T_K to exciter torque T_E (Equ. 2) [5].

$$V = \frac{T_K}{T_E} \qquad \text{Equ. 2}$$

with $T_K = D \cdot \Delta\dot{\varphi} + K \cdot \Delta\varphi$

and $T_E = T_P \cdot \frac{\Theta_T}{\Theta_P + \Theta_T}$ or $T_E = T_T \cdot \frac{\Theta_P}{\Theta_P + \Theta_T}$

In Fig. 5, such a function is illustrated for coupling type VTC 487 T that has already been looked at. In this example, the eigenfrequency has been pre-set with 5 Hz and is meant to correspond with the first mode of the entire drive system. Due to the very low stiffness and the mass distribution in the drive system, this mode is determined by the hydrodynamic coupling.

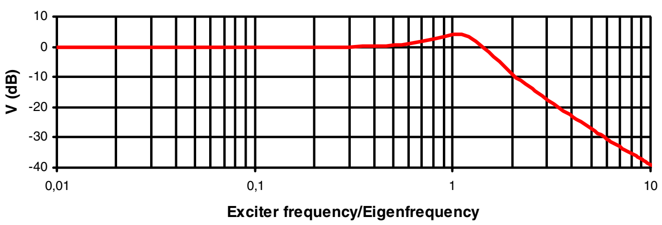

Fig. 5: Enlargement function

From Fig. 5, a low-pass behaviour with a low rise in resonance can be deduced. Up to the first eigenfrequency, the exciter torques are transmitted by the hydrodynamic coupling, but they are strongly damped. However, the coupling torque T_K decreases rapidly with increasing frequency. At six times the eigenfrequency (30 Hz) it is a mere -30,5 dB (V = 3%) of the exciter torque T_E. This means that there is a wide-reaching decoupling and/or separation of torque fluctuations. The resonance rise is uncritical, because, at +4,2 dB (V = 160%), it is in this case very low.

Of particular interest is the height of the maximum first eigenfrequency (first mode). Owing to the low stiffness of all standard coupling types, sizes and systems, it is <20 Hz, often even below 10 Hz. Hence, with speeds of < 600 rpm, the angular frequency of the low-pass behaviour of hydrodynamic couplings is very low. For vibration excitations of the second order or higher, as they are, for example, generated by combustion engines, it presents a genuine obstacle.

This situation is made clearer when looking at the measuring values in Fig. 6. They were obtained from test stand measurements at the VTC 487 T coupling. At a pump speed of 1500 rpm, a nominal operating point of 790 Nm was set, and the pump side was harmonically excited with increasing frequency and amplitude. The in- and output system each consisted of an inverter-controlled asynchronous machine. The primary and secondary torques and speeds were measured before and after the hydrodynamic coupling.

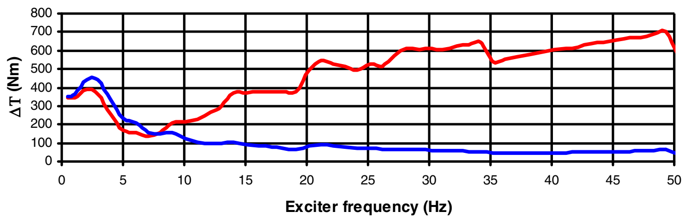

Fig. 6: Measuring values – Torque amplitudes of pump side (red) and turbine side (blue)

In Fig. 6, the torque amplitudes of the exciting pump side T_P and the adapting turbine side T_T are projected over the exciter frequency. Up to a frequency of 7 Hz, the turbine torque is above that of the pump impeller. This is the range of the resonance rise. The maximum quotient (T_T / T_P) of this rise is at 5 Hz, the first eigenfrequency of the system. Above 7Hz, the torque amplitudes of the turbine wheel continually decline, in spite of rising exciter amplitudes on the pump side. At the maximum exciter amplitude of 700 Nm (49 Hz), the torque amplitude measured at the secondary side is a mere 62 Nm.

3.2 First eigenfrequency in proportion to speed

For drives with hydrodynamic couplings, it is often the case that the first eigenfrequency is in proportion to the input speed. This is initially puzzling, but can be relatively easily explained. At a constant frequency ratio Ω (Equ. 3),

$$\Omega = \frac{\text{Exciter frequency}}{\text{Input speed}} = \text{const.} \qquad \text{Equ. 3}$$

modelling according to [4] results in a coupling stiffness proportional to the square of the input speed. (Equ. 4) As it is a well-known fact that the eigenfrequency changes with the square root from the stiffness (Equ. 5), it is also proportional to the input speed (Equ. 6).

$$K \sim (\text{Input speed})^2 \qquad \text{Equ. 4}$$

$$\text{eigenfrequency} \sim \sqrt{K} \qquad \text{Equ. 5}$$

$$\text{eigenfrequency} \sim \text{Input speed} \qquad \text{Equ. 6}$$

The frequency ratio Ω in Equ. 3 is constant in most drive systems, for example the exciter spectrum of a combustion engine, a propeller or the rotor of a wind power station. Here, the exciter frequencies are always in proportion to the input speed, and the frequency ratio Ω is hence constant. Normally, torsional eigenfrequencies are not a function of speed, but constant. (Fig. 7). The fact that they can still take on this role, is not alarming, but rather advantageous. As can be easily seen, no further intersections of the exciters with this eigenfrequency alignment exist above this input speed (except at 0 rpm).
A system as described in Fig. 7 is therefore in the overcritical range at any drive speed. But even a scenario, where the first eigenfrequency is close to an exciter line, can be quite permissible. This is to be illustrated later by the example of a marine drive.

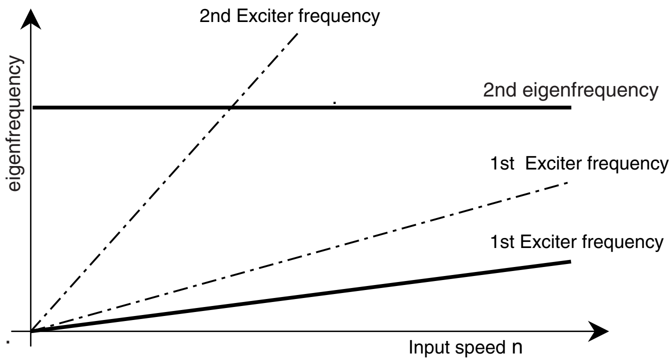

Fig. 7: Example of a resonance graph

4. Verification of Kelvin models for hydrodynamic couplings

The most influential parameters on the torsional vibration behaviour of hydrodynamic couplings are the coupling size and the coupling design, the filling level, the operating medium used, as well as the input speed, the nominal operating point and the exciter frequency.
In view of such a high number of influential parameters, a model verification can only be carried out selectively. A complete experimental investigation would be immensely costly and labor-intensive. The coupling behaviour therefore needs to be extrapolated to other conditions.
With the description per [4] and the above–mentioned influential parameters, the two parameters 'stiffness' and 'damping' of the Kelvin model can be relatively easily determined, so that this extrapolation can be carried out without problems. In this way, the influence of individual parameters and their effect on the drive system can also be examined.
The model verifications presented here were carried out at a marine drive with a nominal output of 1800 kW and a hydrodynamic double coupling size 1150 (VTC 1150 DTM), as well as by test stand measurements at a coupling size 487 (VTC 487 T) at an output of 125 kW.

4.1 Model verification on the basis of test stand measurements

The torque amplitudes in Fig. 6 were obtained in test stand measurements at a VTC 487 T coupling. As described above, an operating point of 790 Nm (slip = 3.3%) was set at a pump speed of 1500 rpm, and the pump side was harmonically excited with increasing frequency and torque amplitude. In Fig. 8 the measured speed amplitudes are illustrated.

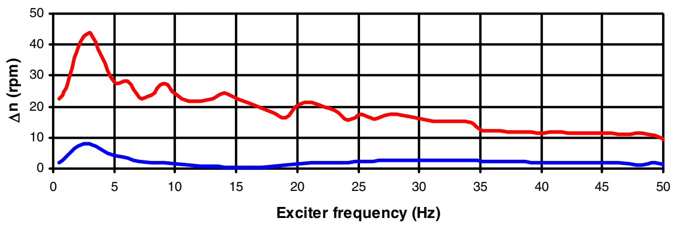

Fig. 8: Measuring values – speed amplitudes of pump side (red) and turbine side (blue)

From the four measured variables in Fig. 6 und Fig. 8, the stiffness and damping values for the individual exciter frequencies can be determined with the phase information not shown here. This is shown in Fig. 9 und Fig. 10, and, for direct comparison, also the theoretically determined values per [4]. The conformity is very good and is hence another confirmation of the theory of coupling modelling.

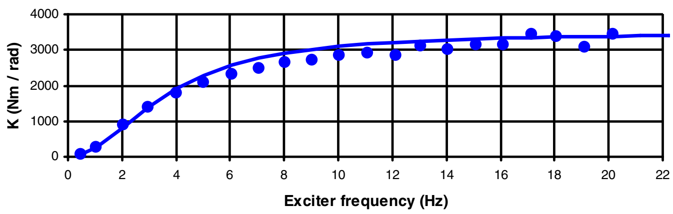

Fig. 9: Stiffness K – Measurements (dots); theory (continued)

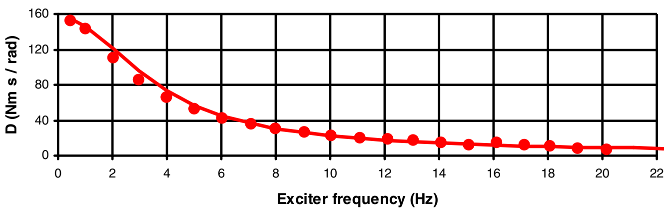

Fig. 10: Damping D – Measurements (dots); Theory (continued)

An experimental determination of the stiffness and damping values at an exciter frequency above 20 Hz is very difficult, because the difference angles and speeds, from which the Kelvin parameters are determined, are declining with increasing frequency. Inaccuracies in the measuring value recordings would have a strong influence on the result, so that such measurements were omitted.
On the other hand, model verifications for exciter frequencies above 20 Hz are not strictly necessary. As mentioned earlier, the first eigenfrequency of a drive system is determined by the

hydrodynamic coupling and is at < 20 Hz for most standard coupling designs, sizes and systems, sometimes even below 10 Hz. Higher eigenfrequencies occur in the primary and secondary drive line and are hardly influenced by the hydrodynamic coupling. This subject will be referred to in greater detail at a later stage of this paper. Verifying the coupling model for exciter frequencies above 20 Hz would therefore be irrelevant

4.2 Model verification at a marine drive

The marine drive shown in Fig. 11 originates from the water tractor "M.V. Taurus". This boat is equipped with two eight-cylinder diesel engines rated at 1812 kW, both of which drive a Voith Schneider Propeller through a hydrodynamic Voith coupling. Contrary to the test stand measurements, it is in this case not possible to generate targeted individual exciter frequencies and hence determine stiffness and damping. The examination of the coupling model is carried out by means of the calculated and measured eigenfrequencies.
Torque measurements were carried out by strain gauge at the connecting shaft of the gear coupling. At this point, the first and the third eigenfrequency could be determined, both of which are in the secondary-side system. Owing to the low-pass behaviour of the hydrodynamic coupling, primary-side located modes could not be measured.
In the marine drive in Fig. 11, the main excitation occurs through the propeller, i. e. primarily at 5 and 10 times the propeller speed (number of blades = 5). Under consideration of the gear stage, the first propeller exciter frequency equals the 0.39^{th} order of the input speed, i. e. it is proportional to it.

Exciter frequency propeller = 0.39 order of engine speed

The requirements of Equ. 3 are therefore fulfilled. The calculation [4] shows that the first eigenfrequency of the drive is equal to the 0.37^{th} order of the engine speed.

Calculated first eigenfrequency= 0.37 order of engine speed

Exciter frequency and eigenfrequency are therefore close to each other. In the frequency spectrum, a rise occurs precisely in this area (0.37^{th} to 0.39^{th} order), which increases with the engine speed and the pitch adjustment of the propeller. Alternating torques of variable heights were measured in the connecting shaft of the gear coupling during various applications of the water tractor. In unfavourable cases, the value was +/- 3.86 kNm. German Lloyd prescribe a value of +/- 30% of the medium nominal torque in the speed range of 90% to 105% as maximum permissible alternating torque. With a maximum of +/- 22.3% of the nominal torque, we are therefore within the permissible range with this drive.
As shown by this example, operation that is close to the first eigenfrequency caused by the hydrodynamic coupling, can certainly be permissible. This statement can, however, not be generalized and needs to be examined from case to case.
The third eigenfrequency was determined with 31.8 Hz on the basis of the measurements and is found in the secondary-side drive system. With a frequency of 30.4 Hz, the simulation calculation is a mere 4.4% below the measuring value, which is a good result. It must be remarked here, that the stiffness of the hydrodynamic coupling has no significant influence on this eigenfrequency. The third mode to 29.8 Hz was also calculated without it (see following chapter 5).

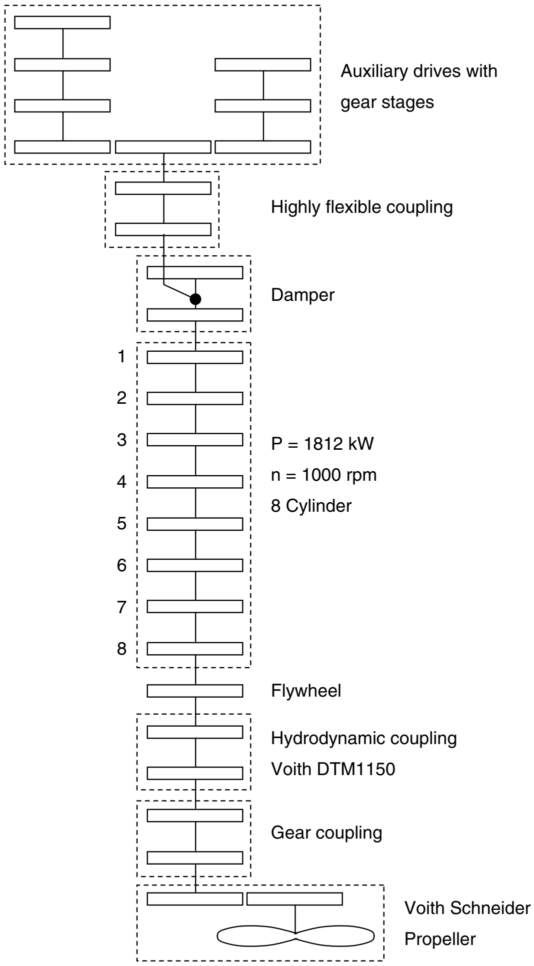

Fig. 11: Mass scheme of marine drive

5. Separate design of primary and secondary-side drivelines

A separate design of primary and secondary-side drivelines can be highly advantageous, if, at an early project stage, not all drive elements are known. In this case, the issue is not so much the stationary design, but the determination of eigenfrequencies and the question of possible resonance excitations. In the past, separate designs were carried out, for example, for marine drives. Here, it can be the case that drive motor or propeller systems are not yet known during the project phase. In order to be able to determine the relevant other, primary or secondary-side drive system, the entire system used to be hypothetically separated by the hydrodynamic coupling. An influence of the primary side on the secondary side and vice versa was hence excluded. Over the years, this method was, however, put in doubt, so that separate designs are no longer practised today. For the above reasons, this makes the work of the project engineer more difficult, so that he will always raise the question of the validity of the previous design approach. For drives with wide differences in mass and/or stiffness distribution, eigenfrequencies can also be allocated directly to individual components, such as drive shafts, highly flexible couplings or gearboxes. Drive systems with hydrodynamic couplings often show large differences when it comes to the distribution of their stiffness; in this case, the coupling stiffness is usually by at least one power of ten lower than the stiffness of the other elements.

This results in the first eigenfrequency being determined by the turbo coupling, amounting to less than 20 Hz for all standard drive systems, often even to less than 10 Hz. For the determination of the first eigenfrequency, it is therefore sufficient to reduce the entire drive system to a two-mass flywheel and calculate it with the coupling stiffness [4]. The frequency thus calculated is higher than the actual frequency. The minimum first eigenfrequency can be determined by applying Neuber's limit value theory [6].
Higher eigenfrequencies can initially be determined by treating the primary and secondary side of drivelines separately. In this case, the stiffness of the hydrodynamic coupling is neglected. Modes determined in this way coincide quite well with actual values. Table 1 shows a comparison of eigenfrequencies that have been calculated with and without coupling stiffness. The example refers to a torsional vibration chain with 6 masses and the values stated in Table 2.

Table 1: Inherent frequencies calculated with and without coupling stiffness

	with hydrodynamic coupling		without hydrodynamic coupling	
	primary	secondary	primary	secondary
Mode 1	4,030 Hz (4,135 Hz)[1]		0,000 Hz	
Mode 2		19,922 Hz		19,461 Hz
Mode 3	73,552 Hz		73,431 Hz	
Mode 4		90,241 Hz		90,179 Hz
Mode 5	1219,6 Hz		1219,6 Hz	

[1] Value in brackets obtained after reduction to two-mass flywheel

Separate designs must, however, be treated with some caution. It needs to be ensured that the coupling stiffness is at least one power of ten lower than that of the other drive elements. Experience has shown that deviations resulting from separate frequency determination can be assumed to amount to +/- 6%. Since the validity of the method described here is not theoretically proven and also dependant on the distribution of masses, it is always recommended to check the entire system. This applies in particular, if the "separate" calculation results in resonance-near operation.

Table 2: Values of torsional vibration chain

Mass inertias	Stiffness with hydrodynamic coupling	Stiffness without hydrodynamic coupling
Θ_1 = 6 kg m²	K_{12} = 5,0*10⁷ Nm / rad	K_{12} = 5,0*10⁷ Nm / rad
Θ_2 = 1 kg m²	K_{23} = 4,5*10⁵ Nm / rad	K_{23} = 4,5*10⁵ Nm / rad
Θ_3 = 3 kg m²	K_{34} = 3,0*10³ Nm / rad	K_{34} = 0,0 Nm / rad
Θ_4 = 2 kg m²	K_{45} = 2,0*10⁵ Nm / rad	K_{45} = 2,0*10⁵ Nm / rad
Θ_5 = 1 kg m²	K_{56} = 3,0*10⁴ Nm / rad	K_{56} = 3,0*10⁴ Nm / rad
Θ_6 = 5 kg m²		

The question, whether a secondary-side eigenfrequency is excited by the primary side (or vice versa) through the hydrodynamic coupling, and how high the amplitudes are, can only be answered by the system as a whole. Systems excitations with high frequencies may be drastically reduced by a hydrodynamic coupling, but they are still transmitted. However, this usually does not result in a noticeable increase of the total load. With modes that are only slightly damped, this may, nevertheless lead to problems. In this case, higher resonance rises are possible even with low exciter amplitudes.
It also needs to be observed that the total load on the system consists of the sum of all individual loads. It is therefore not sufficient to simply look at the amplitudes of each individual exciter frequency, in order to establish whether permissible values have been exceeded. Especially with resonances that are close to each other, the superposition of individual excitations can quickly result in a doubling of the load.

6. Simulation calculations within the resonance range

In theory, simulation calculations within the resonance range are indeed possible. The question here is to determine the amplitudes during excitation closely to or directly at the point of resonance. What may work well theoretically, often proves difficult in practice. There may be considerable differences between the simulated and the measured amplitudes which can amount to several 100%.
The reasons for this are primarily incorrect or only partly accurate data and/or assumptions regarding the drive system or the systems excitations. Even slight deviations between the calculated and the actual eigenfrequencies, or wrong assumptions regarding the systems excitation, might be sufficient to falsify the simulation results considerably. A similar scenario occurs, if the damping values, e. g. gears, shafts and connecting elements, are inaccurate.
In general, simulation calculations within the resonance range are to be carried out with utmost care and to be evaluated critically. The decisive question is here, by which component (shaft, gearbox, turbo coupling, etc.) the mode, i. e. the eigenfrequency, is primarily determined.
The biggest problems occur, if a resonance is determined by a component that is only slightly damped (e. g. cardan shaft or torsionally stiff connecting coupling). Slight shifts of the assumed exciter or the calculated eigenfrequency have a particularly noticeable effect in this instance. However, the exciter amplitude, too, has a major influence in connection with the system damping. As a guide value, simulation calculations in the +/- 20% frequency range should either be completely avoided for such a resonance point, or at least be looked at with a very critical eye.

Fig. 12 illustrates the enlargement functions of a slightly and a strongly damped system. As can be seen, within the range +/- 20%, the function increases drastically by the eigenfrequency (dots in the diagram). For example, a frequency shift η = 0,92 to η = 0,96, i. e. 4.4%, would result in a doubling of the amplitude. This example demonstrates clearly, how sensitive calculations in this area are.

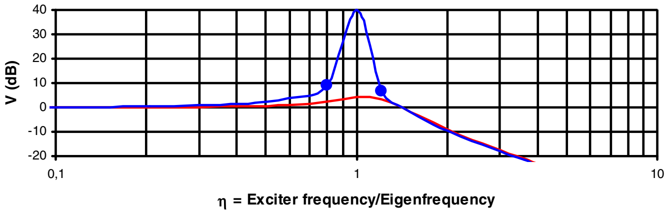

Fig. 12: Enlargement function red – strongly damped (D = 0,4); blue – slightly damped (D = 0,005)

On the other hand, there are fewer problems, if an eigenfrequency is determined by a strongly damped component (e.g. hydrodynamic or highly flexible coupling). With sufficient accuracy, amplitudes may even be determined directly at their

point of resonance. Inaccuracies at the assumed exciter or the calculated eigenfrequency bear hardly any effect. The same applies to variations of the exciter amplitudes.
The terms "slightly" or "strongly" damped system are to be quantified on the basis of Lehr's damping factor D. Strictly speaking, the latter has only been defined for a one-mass system. With the help of modal analysis, a transformation to several one-mass systems is possible. We are actually speaking of a modal damping measurement according to Lehr. In Table 3, the experience values for D are listed. As a general rule, the following can be assumed:

$D > 0{,}1$ strongly damped system
$D < 0{,}1$ slightly damped system

Table 3: Experience values for Lehr's Damping Factor

Transmitting component	Lehr's Damping Factor
Steel shaft d < 100 mm	0,005
Steel shaft d > 100 mm	0,01
Transmission gearing P < 100 kW	0,02
Transmission gearing 100 kW < P < 1000 kW	0,04
Transmission gearing P > 1000 kW	0,06
Highly flexible coupling	up to 0,13 (at 10 Hz)[1]
Hydrodynamic coupling at n = 1500 rpm	0,19 (at 10 Hz)[1] 3,58 (at 0,5 Hz)[1]

[1] Derived from relative damping with exciter frequency = eigenfrequency of non-damped system

As these statements show, the occasionally quite considerable differences between measurements and calculations in the resonance area are not automatically to be attributed to faulty coupling modelling. It is essentially in the nature of simulations which are always carried out on the basis of assumptions and simplifications that eigenfrequencies and systems excitations can never be determined precisely. This, in combination with high systems sensitivities, can quickly lead to false results and conclusions which in most cases carry considerable consequential costs.

7. Summary

For the torsional vibration behaviour of hydrodynamic couplings, a simplified description in the form of a Kelvin model with frequency-dependent stiffness and damping was presented, from which two fundamental characteristics were deduced. On the one hand, there is a low-pass behaviour with low resonance rise which can be attributed to high damping by the hydrodynamic coupling. The maximum eigenfrequency, i.e. the maximum angular frequency of this low-pass behaviour, is below 20 Hz for all standard drive systems, coupling sizes and designs, often even below 10 Hz. As a result, excitations of higher frequencies at a transmission through the coupling are strongly reduced. Yet caution needs to be applied. Especially with slightly damped modes, such excitations can still lead to higher resonance rises.
The second fundamental characteristic is the speed-proportional eigenfrequency of drives with hydrodynamic coupling. The prerequisite for this is, however, a constant ratio of exciter frequency to input speed. This is the case with most drives (e. g. combustion engines, propeller or rotor of a wind power station).
By means of test stand measurements, as well as theoretical and experimental examinations of a marine drive, the applicability of the Kelvin model could be verified. Based on the assumption that the stiffness of the hydrodynamic coupling is lower by a least one power of ten than the stiffness of the remaining drive elements, a separate design of primary and secondary-side drivelines is permissible. This design is, however, limited to the observation of stationary loads, as well as the determination of eigenfrequencies. The calculation of torsional loads as a result of systems excitation must always be carried out for the entire system. The same applies to the urgently recommended examination of "separately" determined modes. Simulation calculations in the resonance range or even directly at the point of resonance, are only permissible for strongly damped modes. With slightly damped eigenfrequencies, calculations in the +/-20% area around a point of resonance should either be avoided completely, or at least be looked at with a very critical eye. Errors of several 100% are possible.

8. Literature

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