FAILURE OF A CONTROLLABLE PITCH PROPELLER SYSTEM AT NO LOAD OPERATION

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It is often assumed that failure of a mechanical system can occur only under loaded or overload conditions. This assumption is correct if one only considers a non-dynamic condition. In addition to the static loading most mechanical systems are also subjected to dynamic loading due to the fact that they are considered to be finite stiff structures and not infinite stiff structures.

In regards to the marine propulsion systems, one is mainly dealing with two principal sources of excitation, the main engine and the propeller and they are linked by an elastic shaft system. In addition, the whole propulsion system is supported in a flexible hull structure. The forms of vibration possible in these propulsion systems are very diverse but similar techniques are used to analyze the system behavior for each respective form of vibration. This paper shows that if certain operating parameters have not been considered and analyzed at the design stage of the vessel, it can lead to expensive failures when the vessel is in service.

1. Introduction
In order to predict the behavior of any physical system, a model suitable for mathematical analysis is essential. The model should be able to predict the behavior of the system with sufficient accuracy. In addition, with the continuing development of higher output diesel engines and increased complexity in design and operation of propulsion plants, the need for mathematical models capable of predicting behavior at non-resonant conditions is now greater than ever. The analysis must involve all expected operating modes of the propulsion system to be analyzed.

In the case of a propulsion plant with a slow speed diesel engine and a direct driven propeller, only the first two modes of vibration are significant from a torsional point of view. In such a case the torsional vibration analysis becomes rather simple. First, the mass-elastic system is established and the equation of the motion is set up for each mass in the system. The result is a set of linear differential equations which are transformed into a characteristic system of linear algebraic equations and are solved as Eigenvalue problem to get the natural frequencies. Forced damped calculations are then performed using the system of simultaneous equations. In order to determine the response of a complex system with several sources of damping and to obtain a more realistic picture of the non-resonant conditions, the system of equations are solved for each order using the harmonic components of the tangential efforts as the external torque, including the first two propeller orders. The calculated results for each order and mode shape is added as a synthesis, taking into account the phase relations in order to find the overall vibration amplitudes or power dissipation in the shafting system.

The paper describes how a failure occurred when the torsional vibration analysis failed to consider all of the variations of the actual service profile of the vessels in question.
2. Vessel and propulsion plant description

The vessel in question is a 1400 TEU container vessel operating between the west coast of the USA and Alaska in a regular service route. In the winter time this vessel performs quite a few maneuvers in Alaska due to the presence of drift ice at the docks. During these maneuvers the engine is operating at the set idling speed of 60 rpm with the use of the variable pitch propeller and bow and stern thrusters until the vessel can be safely docked.

The propulsion plant in question is as follows:
- Main engine: MAN B&W 7L70MC
  MCR: 22540 BHP at 98 rpm
- CPP system: Kawasaki-Esher Wyss 2000 CB/570RS
  - Number of blades: 4
  - Diameter: 7.1 m
  - Main pitch (0.7R): 7.763 m
  - Blade area ratio: 0.525
  - Hub diameter: 2.0 m

3. Failure of CPP system

After many years of service, failures started to occur in the pitch control head in the propeller hub. A total of two failure on one vessel, one failure occurred on one sister vessel, and on the other sister vessel, crack initiation had already started. Since the manufacturer had quite a few of these systems in service including similar systems, there was no explanation to why these failures occurred. The failures occurred within a very short time frame, compared to the total service life of the vessels at the time of the failures.

Eventually through several meetings with the owner and propeller manufacturer it was determined that these failures did not relate to the design of the propeller equipment and to the normal service profile of the vessels. Consequently this led to the speculation that these failures could be related to the maneuvers in Alaska with the engine operating at idling speed.

4. Initial torsional vibration analysis

The initial torsional vibration analysis carried out by the engine manufacturer was not available for our review; consequently a mass-elastic system was constructed (Fig. 1 & Fig. 2) using the relevant data from each manufacturer involved. Two Plants were considered in this initial analysis to establish a baseline for our further analysis.

Fig. 1 shows the mass elastic system at full pitch and Fig. 2 shows the mass elastic system at zero pitch. Both these two Plants were submitted to the classification society for their approval.
As stated earlier, with a slow speed two stroke engine only the first two modes of vibration are being considered. The first mode of vibration is the propeller mode where the engine and propeller oscillate out of phase in relation to each other. The second mode of vibration is the crankshaft mode, usually with one node located in the crankshaft, and the second node located in the shafting system (Fig. 3 & Fig. 4).

It must also be understood that with a controllable pitch propeller the amount of entrained water will increase with higher pitch settings, consequently, the effective propeller inertia will increase. This is the reason that the frequency of the first mode of vibration will decrease with the increase in the propeller pitch. The table below lists the calculated natural frequencies of the two plants evaluated in the Vibrations Per Minute VPM for the first two modes of vibration.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Plant 1</th>
<th>Plant 2</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>235</td>
<td>250</td>
</tr>
<tr>
<td>2</td>
<td>1239</td>
<td>1239</td>
</tr>
</tbody>
</table>

Table 1. Natural Frequencies

A resonant speed diagram was constructed for both plants (Fig. 5 & Fig. 6).
Based on the results from the natural frequency calculations, it can be seen that the seventh engine order excites the first mode of vibration around 30 rpm, which is well below the idling speed of the engine. It should therefore not be of any concern. Judging by the natural frequency calculations and the constructed resonant speed diagram, the initial concern is the fourth engine order – first mode of vibration and the first propeller order excitation which both have a resonance condition around 60 rpm, this being the idling speed of the engine.

5. Forced damped calculation for the two plants evaluated

It must be stated at this point that we have not included the calculated crankshaft stress amplitudes as these have no influence on the problem discussed.

Forced damped calculations were performed for each of the two plants at different engine speeds. In the case of plant 1, the load was assumed to be in accordance with the nominal propeller curve based on the propeller law up to the MCR rating at 16590 kW at 98 rpm. For part 2, two load conditions were used for the forced damped calculations at zero pitch, and they are based on information from the propeller manufacturer, which specifies the load at zero pitch to be 2870 kW at 98 rpm. The first case assumes that the load on the engine increases with the engine speed according to the propeller law up to the maximum load of 2870 kW, and a constant load for higher engine speeds. The second case assumes a constant torque based on 2870 kW at 98 rpm.

In all cases, the harmonic excitations are calculated for each order based on the gas pressure and the reciprocating inertia force in each cylinder, including the propeller excitation. Two excitation conditions were considered in this initial analysis.
1. Normal combustion
The harmonic excitation from each cylinder is equal. This is an idealized condition not likely to occur in service. The significant orders of excitation for a 7 cylinder two stroke inline engines are 1st, 7th and 14th engine orders.

2. 15% Misfire in one cylinder
This is a condition attempting to simulate the normal irregularities that will occur in the cylinders during operating over time.

6. Calculated results from the initial analysis
We have presented the calculated results in the form of graphs for each section of the line shaft system. Based on these results (Fig. 7, Fig. 8, & Fig. 9) with the engine operating in accordance with the nominal propeller curve up to the MCR rating of 16590 kW at 98 rpm, the stress amplitudes are well within acceptable limits, as can be seen from the graphs below. This includes the simulation with one cylinder 15% misfiring (Fig. 10, Fig. 11 & Fig. 12).
Calculations with a zero pitch condition and an engine output of 2870 kW at 98 rpm reveals a higher level of stress amplitudes. The reason for this increase is mainly due to the reduction of propeller damping, which is reduced by $2/3$ from the full pitch values. These stress amplitudes are still within acceptable limits except for the propeller shaft where the first mode of vibration 7th engine order resonance peak exceeds ABS continuous operation limit. The actual resonant peak will be comparatively smaller than the calculated peak and therefore should be of no concern since the engine has a quick pass through during start, through this area of the resonant peak (Fig. 13 through Fig. 18).
Fig. 13 Vibratory shear stress in intermediate shaft – Zero pitch

Fig. 14 Vibratory shear stress in control shaft – Zero pitch

Fig. 15 Vibratory shear stress in propeller shaft – Zero pitch

Fig. 16 Vibratory shear stress in intermediate shaft – Zero pitch - Misfiring
With constant torque and zero pitch operation we see a similar picture (Fig. 19, Fig. 20 & Fig. 21) as above and the same consideration as discussed are valid.
Based on this initial analysis it is clearly seen that the propulsion system is safe from a torsional point of view and consequently approved by the classification society.

7. Further analysis leading to the problem that caused the failure

In the initial analysis we were, as is common, only concerned with the vibratory stress amplitudes, and based on these calculated stress amplitudes the system was seen as acceptable from a torsional point of view. Consequently the next step in the analysis was to evaluate the vibratory torque in the system. Normally with direct driven two-stroke engine installations the only parameter evaluated from a torsional point of view is the vibratory stress amplitudes. Vibratory torque in the system is generally of no concern.

In this analysis, the engine load is assumed to follow the propeller law to a maximum load proportional to the pitch setting. Only the normal firing condition is evaluated here and we did not simulate the condition with one cylinder with a 15% misfiring.

Four plants were considered in this analysis

**Plant 3**
Idling speed with zero pitch: This assumes that the load on the engine increases with the engine speed according to the propeller law up to maximum load of 2870 kW, and at a constant load for higher engine speeds (Fig. 22, Fig. 23 & Fig. 24).

**Plant 4**
Idling speed with 5 degree ahead-pitch: The engine load is assumed to follow the propeller law to a maximum load proportional to the pitch setting (Fig. 25, Fig. 26 & Fig. 27).

**Plant 5**
Idling speed with 10 degree ahead-pitch: The engine load is assumed to follow the propeller law to a maximum load proportional to the pitch setting (Fig. 28, Fig. 29 & Fig. 30).

**Plant 6**
Idling speed with 15 degree ahead-pitch: The engine load is assumed to follow the propeller law to a maximum load proportional to the pitch setting (Fig. 31, Fig. 32 & Fig. 33).

All of the above operating conditions simulate the operating modes that take place while docking the vessel in the winter time in Alaska. Astern pitch condition is not a part of this analysis, since the parameters are very similar to the ahead-pitch conditions.
8. Calculated results from the second analysis

In regards to Plant 3 results, we can see that our initial concern with the 4th engine order and first propeller order was valid; they both have a resonant peak at 62 rpm. Based on the synthesis calculations we reach a vibratory torque of approximately 1100 kNm in all three shaft segments. This is 2.6 times higher than the mean transmitted torque at that rpm (Fig. 22, Fig. 23, & Fig. 24). As the pitch increases up to 15 degrees, this vibratory torque is reduced to 2.2 times the mean transmitted torque at the same rpm (Fig. 25 through Fig. 33). But it must be understood at this point that as the pitch increases so does the engine rpm. The point setting where the engine rpm increases is set by the load program of the engine. As can be seen from these calculated results, the worst condition that the system is exposed to is when the engine operates at idling speed with zero pitch setting.
Fig. 25 Vibratory torque in intermediate shaft – 5 degree pitch

Fig. 26 Vibratory torque in control shaft – 5 degree pitch

Fig. 27 Vibratory torque in propeller shaft – 5 degree pitch

Fig. 28 Vibratory torque in intermediate shaft – 10 degree pitch
Fig. 29 Vibratory torque in control shaft – 10 degree pitch

Fig. 30 Vibratory torque in propeller shaft – 10 degree pitch

Fig. 31 Vibratory torque in intermediate shaft – 15 degree pitch

Fig. 32 Vibratory torque in control shaft – 15 degree pitch
9. Governor oscillations
An additional problem noticed on all three vessels was the fact that at idling speed with zero pitch setting, the engine was hunting; the engine speed varied from around 51 rpm to 63 rpm, a speed variation of 12 rpm in the worst case was observed. Consequently it was decided to commence governor stability calculations. Based on the calculated results we could see that at the idling speed with zero pitch setting, the speed variation was calculated to be 11 rpm, caused by governor oscillation. At the same time we commenced the same calculations, but the idling speed was moved up to 65 rpm. As can be seen the governor oscillation and consequently speed variation was reduced quite substantially (Fig. 34). This was a very important point, since in some cases; the governor system can behave as a tuned magnifier and cause a considerable increase in the vibratory loading of components in the system.

Based on these calculations it was decided to measure these speed variations on all three vessels, and the results are presented in Fig. 35. The measured results showed a significant difference between the three vessels. This became a very important parameter to evaluate since these speed variations will increase the excitation (vibratory loading) of the system. This relates to the fact that larger the speed variation the larger the excitation becomes. Based on these parameters we can clearly see why one vessel had two failures, one vessel had one failure and the third vessel had crack initiation starting.
Another question arose as to why there was such a difference between the three vessels. It was later confirmed that the compensating spring in each governor was set at different spring loads.

10. Conclusion and recommendations
It was very unfortunate that the chosen idling speed of 60 rpm coincided with the resonance condition of the first propeller order and 4th engine order – first mode of vibration. This condition would not have been a problem from a torsional point of view under normal operating conditions. The problem arose when the engine was allowed to operate at the idling speed with zero pitch or relative low pitch settings for a long time. The high vibratory torque amplitude in the shaft system increased the linear acceleration of the propeller to such an extent it eventually caused the failures.

Our recommended solution was to increase the idling speed of the engine from 60 rpm to 67 rpm, and to correct the governor oscillation problem to an acceptable level. These recommended solutions reduced the vibratory torque amplitudes in the shaft system from approximately 1100 kNm to 150 kNm. As a result of moving the idling speed away from the existing resonant peak and using the correct spring load for the compensating spring, the speed variation of the engine was reduced from 12 rpm to 4 rpm.

As we have clearly demonstrated in this paper, it is no longer sufficient to carry out the standard calculations covering the normal operating modes. In consideration to the increased complexity of today’s propulsion plants it has become even more important that calculations are performed for the modes of operation not normally envisioned but which could possibly be experienced. Failure to do so could result in very expensive failures such as occurred in this case.