ANTI-BACKLASH GEAR TRAIN INVESTIGATION

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1 INTRODUCTION

This paper is a continuation of study on Wärtsilä large diesel engine gear train noise and dynamic behaviour. In previous studies, it has been claimed that the gear train noise, as one of the most important large diesel engine noise sources, is caused by shafts torsional vibration generated gear teeth impact during gear meshing because of the existence of backlash. Two anti-backlash scissors gear wheels were designed for W32 engine gear train system, the camshaft gear and the large intermediate shaft gear, Figure 1. They were tested on W6L32DF laboratory engine in Vaasa.

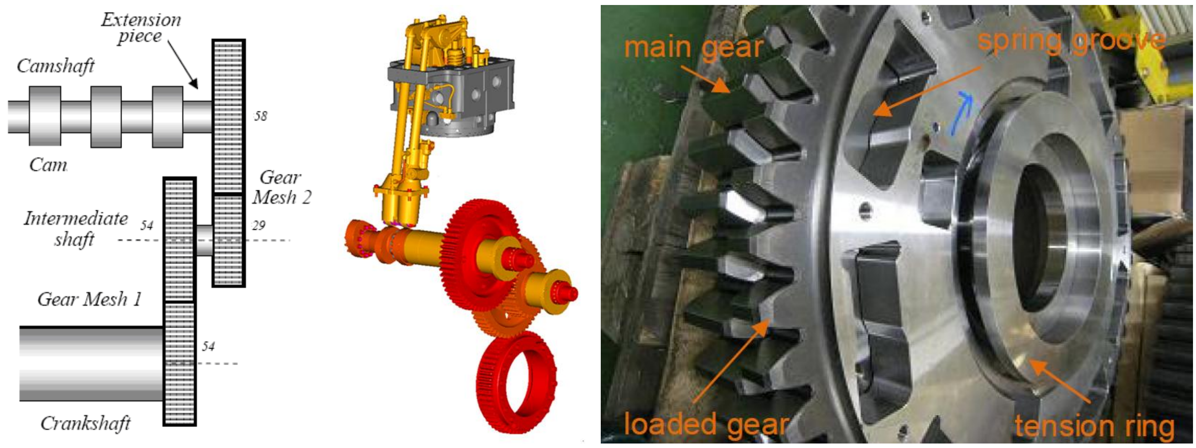

Figure 1. Left, schematic picture of the gear train system; Middle, W32 gear train model; Right, two parts of the anti-backlash large intermediate shaft gear.

Basically, the anti-backlash cam-gear and large intermediate shaft gear have quite similar design. The gear wheel splits into two parts: the main gear and the loaded gear. There are springs keeping the two gear wheels apart so that in principle the gear teeth should be in contact with both the working flank and the non-working flank of the meshing gear. In other words, the function of the spring is to resist the main gear teeth transferring back towards the non-working flank through the backlash during high torque variation. The two parts are kept together by a tension ring, which is attached to the main gear by shrink connection. There are ten spring grooves on the loaded large intermediate shaft gear, and eight on the loaded cam-gear. The springs for the cam-gear are stiffer in total than the ones for the large intermediate shaft gear. The anti-backlash gear wheels are heavier and have more moment of inertia than original designs. The anti-backlash large intermediate shaft gear has the same width with original design, while the anti-backlash cam-gear is 5mm wider than the original one. Both main gear wheels are 2/3 in width of the original design [1]. Reference measurements were carried out before hand for results comparison.

The main properties under investigation in the measurements are: camshaft and intermediate shaft torque, cam-gear and large intermediate shaft gear peripheral acceleration, calculated gear teeth impact force, and sound pressure level inside the cam-gear cover. Figure 2 shows some of the instrumentation setups. Camshaft and intermediate shaft torques were measured by strain gauges. In the reference measurement, two accelerometers were mounted 180 degree out of phase on both the cam-gear and the large intermediate shaft gear, so that by taking the average of the two acceleration signals, the peripheral acceleration for each gear wheel can be attained. In the anti-backlash gear wheel measurement, besides the two out-of-phase accelerometers on each main anti-backlash gear wheel, there is also one accelerometer mounted on each of the loaded gear wheel. Hydrophone was used for measuring the sound pressure level inside the cam-gear cover.

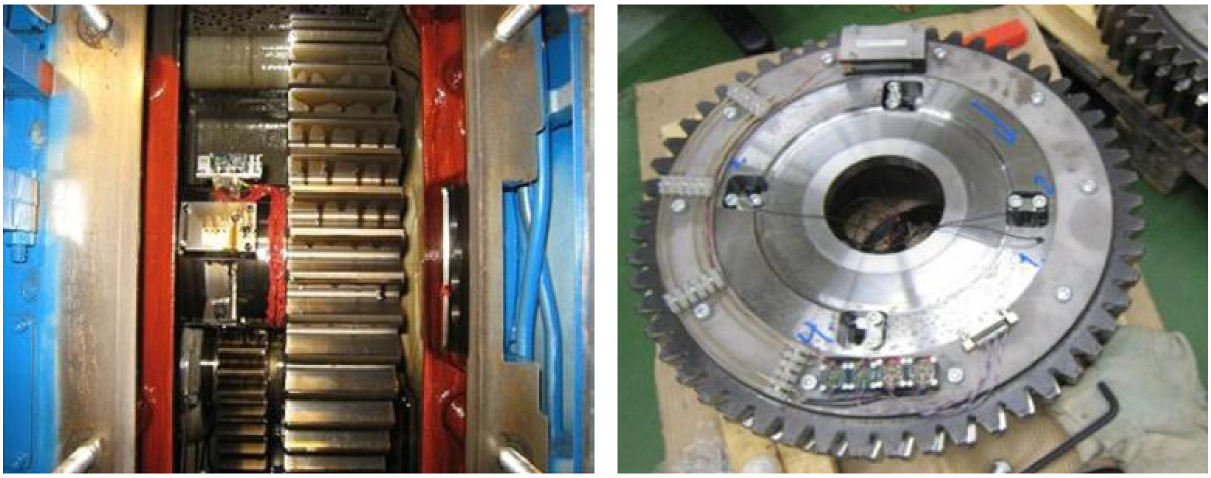

Figure 2. Left, strain gauge on the extension piece of the camshaft next to the cam gear; Right, acceleration sensors setup for the anti-backlash large intermediate shaft gear.

2 REFERENCE MEASUREMENT – GEAR HAMMERING DETECTION

The gear teeth impact force can be calculated from the measured shaft torque and gear wheel peripheral acceleration by Equation (1) based on mathematic model for driven gears in a gear mesh, i.e. the cam-gear and the large intermediate shaft gear.

$$F_{driving} \qquad \frac{T_{shaft} \quad I_{wheel} \quad \theta''_{wheel}}{r_{wheel}} \tag{1}$$

where $F_{driving}$ is the driving force applied on the driven gear teeth, T_{shaft} is the torque of the shaft that the driven gear locates on, I_{wheel} is the moment of inertia of the driven gear, $\ddot{\theta}_{wheel}$ is the peripheral acceleration of the driven gear wheel, and r_{wheel} is the radius of the driven gear.

The shaft torque, gear peripheral acceleration and calculated gear teeth impact force are plotted in Figure 3 for diesel running 100% load, nominal speed 750rpm.

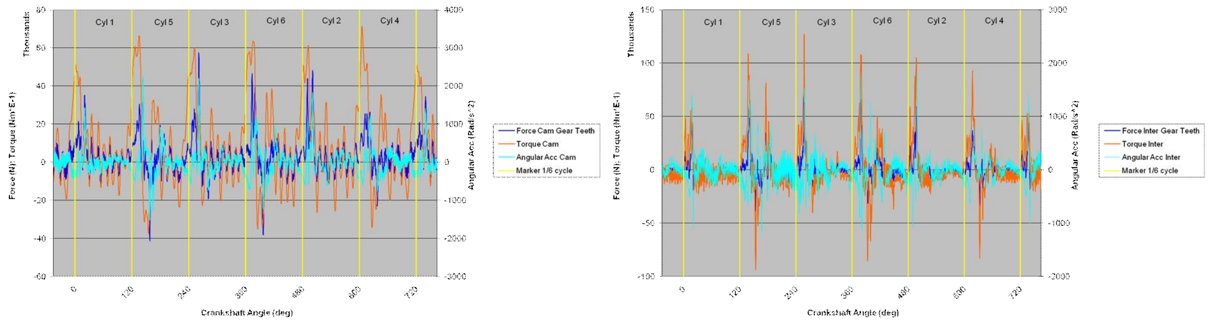

Figure 3. W6L34DF shafts torque, gear wheel peripheral acceleration, and calculated force applied on the gear teeth at diesel running 100% load, nominal speed 750 rpm. Left, camshaft; Right, large intermediate shaft.

Each peak of the camshaft torque represents the fuel injection period of the corresponding cylinder. The driving force rises up at the meanwhile. After fuel injection, the torque of camshaft decreases fast down to below zero because of the potential energy stored during fuel injection period. The sharp negative peaks of the force indicate that the gear teeth have impacted on the non-working flank in the gear mesh. Gear teeth impact is more severe during the fuel injection period of a cylinder that locates further from the flywheel end. For example the absolute amplitudes of the negative force peaks of cylinder 6, 5, and 4 are larger than those of cylinder 1, 2 and 3. For the large intermediate shaft gear, even the second impact on the working flank can be observed after the non-working flank impact during the injection periods of cylinder 5 and 6. The torque and acceleration after fuel injection look more like free vibration, i.e. gear teeth are out of contact most of the time. The intermediate shaft torque is larger than the camshaft torque in amplitude in general.

Figure 4 shows the applied gear teeth forces in different load conditions.

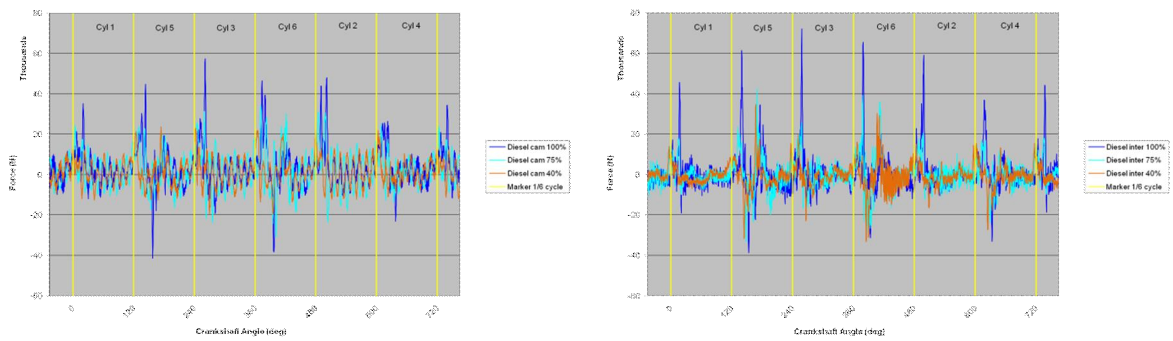

Figure 4. W6L34DF calculated gear teeth force at 750 rpm, 100%, 75% and 40% load diesel running condition. Left, cam-gear; Right, large intermediate shaft gear.

The gear teeth force amplitudes decreases with the load decrease. The gear teeth non-working flank impact force is more severe for higher load conditions.

3 ANTI-BACKLASH GEAR WHEELS MEASUREMENT – COMPARISON

Figure 5 shows the comparison of shaft torque between standard and anti-backlash gear trains.

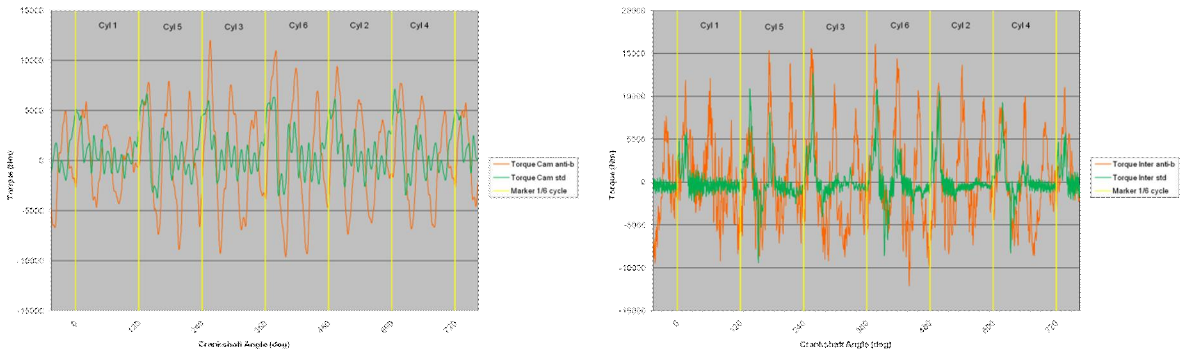

Figure 5. W6L34DF shaft torque comparison between standard and anti-backlash gear trains, Diesel 100% load, and 750 rpm. Left, camshaft; Right, intermediate shaft.

The high frequency components in the standard gear train torque curve have been reduced while the low frequency components have increased by the application of anti-backlash gear train. The torque variation peak to peak value has increased significantly. The third order of firing frequency is dominating, i.e. 112.5 Hz.

Figure 6 shows the shaft torque in the speed sweep measurements for anti-backlash gear train.

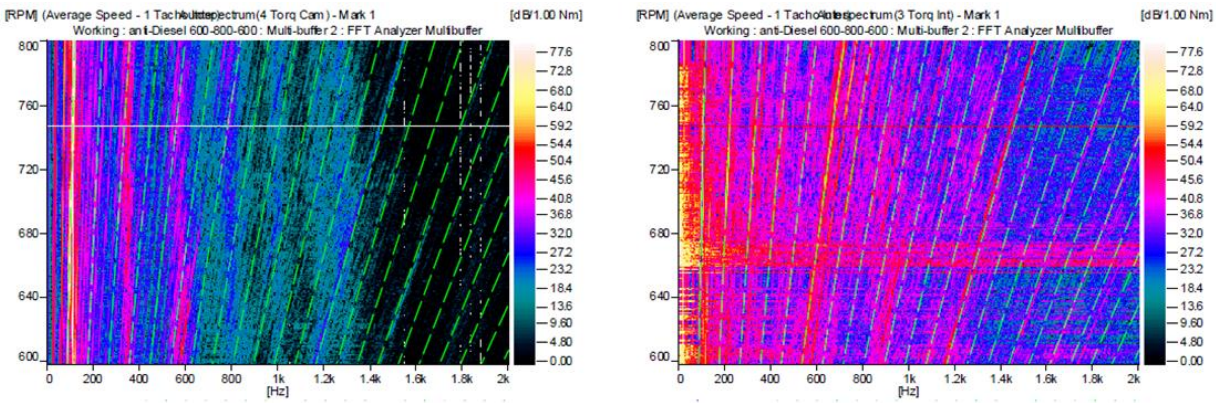

Figure 6. W6L34DF shaft torque with anti-backlash gear train application, diesel 40% load condition, 600-800-600 rpm. Left, camshaft; Right, intermediate shaft.

As indicated by the green dashed harmonic cursor, the highest peaks are from the fundamental frequency of 112.5 Hz, i.e. the 3rd order of firing frequency, which might be attributed to injection and valve train functioning.

By using Equation (2), the gear teeth driving force can be calculated for anti-backlash gear wheels.

$$F_{driving} \quad \frac{T_{shaft} \quad I_{main} \quad \theta''_{main} \quad I_{loaded} \quad \theta''_{loaded}}{r_{wheel}} \tag{2}$$

where $F_{driving}$ is the driving force applied on the entire piece of anti-backlash gear teeth, T_{shaft} is the torque of the shaft that an anti-backlash gear wheel locates on, I_{main} is the moment of inertia of the driven anti-backlash main gear wheel, I_{loaded} is the moment of inertia of the driven anti-backlash loaded gear wheel, $\ddot{\theta}_{main}$ is the angular acceleration of the driven anti-backlash main gear wheel, $\ddot{\theta}_{loaded}$ is the angular acceleration of the driven anti-backlash loaded gear wheel, and r_{wheel} is the radius of the driven gear wheel.

The results of the calculated driving force for anti-backlash gear wheels are compared with the ones from reference measurement with standard gear wheels, Figure 7.

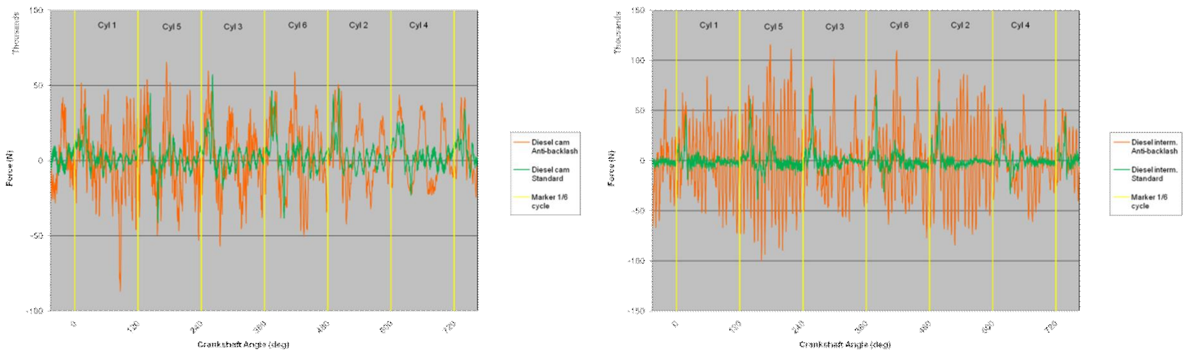

Figure 7. W6L34DF gear teeth driving force comparison between standard and anti-backlash gear trains, diesel 100% load. Left, cam-gear; Right, large intermediate shaft gear.

The deep negative peak, which appears for every camshaft rotation, in the anti-backlash cam-gear wheel force curve is due to a periodical error from the loaded cam-gear wheel acceleration sensor. The anti-backlash gear train peak to peak force value is larger than that of standard gear train. In the frequency spectra which are not presented in this paper, the crank-gear meshing frequency, 675 Hz, is dominating the anti-backlash intermediate shaft gear teeth driving force spectrum, and the 2^{nd} order of cam-gear meshing frequency, 725 Hz, is dominating the anti-backlash cam-gear teeth driving force spectrum.

By listening experience of laboratory workers, it can be concluded that the noise has increased considerably after changing the standard gear wheels to anti-backlash ones at all different load and fuel conditions. Figure 8 shows the comparison of the cam-gear cover inside noise spectra between the standard and anti-backlash gear train at diesel 100% load.

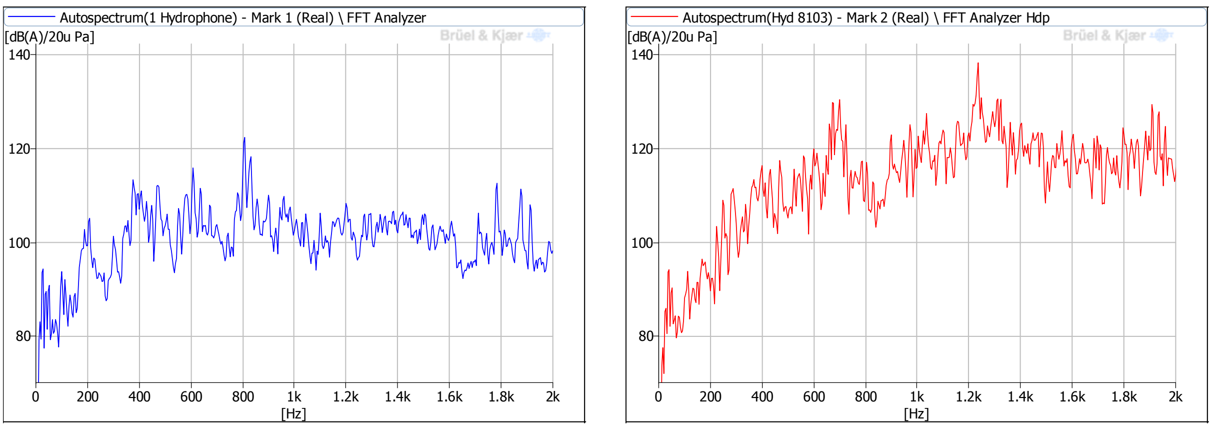

Figure 8. W6L34DF cam-gear cover inside A-weighted sound pressure level comparison, diesel 100% load, 750 rpm. Left, standard gear train; Right, anti-backlash gear train.

The dominant peak of the noise inside the cam-gear cover is 808 Hz for the standard gear train and 1236 Hz for the anti-backlash gear train, which seems to be a resonance from certain part of the anti-backlash gear train system. Besides, at some other load conditions, the crank-gear meshing frequency 675 Hz and the 2nd order of the cam-gear meshing frequency 725 Hz are dominating in the cam-gear cover inside A-weighted sound pressure level spectra.

Figure 9 shows the imprints on gear wheels after anti-backlash gear train test.

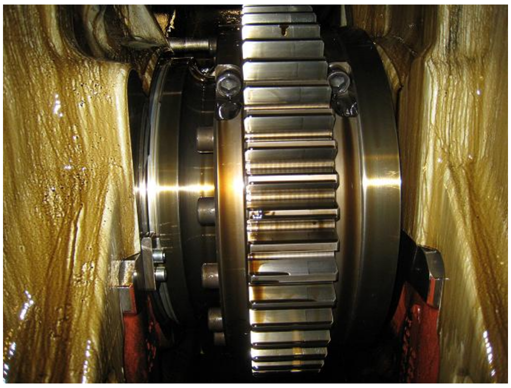

Figure 9. W6L34DF gear wheels after running for anti-backlash gear train test. Left, crank-gear; Right, small intermediate shaft gear.

The imprints indicate that the gear teeth have a risk of fast wearing when running with anti-backlash gear train. It might be because the springs used in the anti-backlash gear train are too stiff, which could also be a cause of friction noise inside the cam-gear cover.

4 CONCLUSIONS AND SUGGESTIONS

In this paper, the gear teeth impact phenomenon for a large diesel engine gear train system has been demonstrated and clarified. The impact is more severe at higher engine load, and during the fuel injection period of a cylinder that locates farther from the flywheel end. The anti-backlash gear train has introduced a frequency component of 112.5 Hz, i.e. the 3rd order of firing frequency, which might be attributed to injection and valve train functioning, to the camshaft and intermediate shaft dynamics. It has led to significantly high torque variation. Engine noise has increased considerably by the application of the anti-backlash gear train. At engine full load, a resonant noise at 1236 Hz is dominant. At other load conditions, the crank-gear meshing frequency 675 Hz and the 2nd order of cam-gear meshing frequency 725 Hz are also important for gear train noise. The springs used in the anti-backlash gear wheels are too stiff, which might cause the gear teeth fast wearing and large gear teeth friction noise.

For further study on anti-backlash gear train, one should make a modal analysis by measurement or calculation on each piece of the anti-backlash gear wheels and as a whole as well. It is encouraged to investigate the cause of shaft torque variation at 112.5 Hz. One should lower the stiffness of the springs for the next anti-backlash gear train test.

REFERENCE

1. JANI TÄHTINEN, "*Design of a Gear Train in a 4-Stroke Engine with Zero Backlash*", Master thesis project at Wärtsilä Finland Oy, May 2009.