

GEARBOX DYNAMIC SIMULATION MODEL OF DRIVEN GROUP IN TERMO POWER PLANT REK BITOLA

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Abstract: In this paper is given the results of the permanent measuring the vibrations on the gearbox as a part's of the driven group of the plants (T5.1' and T5.2') in Thermo Plant REK Bitola, R. Macedonia, biggest energetic institution for manufacturing the electrical power in the country. Also, we compare the results for our mathematical model with measured vibration.

Those results will help us in early diagnostic and monitoring the vibration of gearbox which used of the plants in Thermo Plant REK Bitola.

Keywords: Vibration, gearbox dynamic model, FFT analysis

1. Introduction

Simulation is an effective tool for understanding the complex interaction of transmission components in dynamic environment. Vibro-dynamics simulation of faulty gears and rolling element bearings allows the analyst to study the effect of damaged components in controlled manners and gather the data without bearing the cost of actual failures or the expenses associated with an experiment that requires a large number of seeded fault specimens. Gears, by their inherent nature, cause vibrations due to the large pressure which occurs between the meshing teeth when gears transmit power. Meshing of gears involves changes in the magnitude, the position and the direction of large concentrated loads acting on the contacting gear teeth, which as a result causes vibrations. Extended period of exposure to noise and vibration are the common causes of operational fatigue, communication difficulties and health hazards. Reduction in noise and vibration of operating machines has been an important concern for safer and more efficient machine operations.

Design and development of quieter, more reliable and more efficient gears have been a popular research area for decades in the automotive and aerospace industries. Vibration of gears, which directly relates to noise and vibration of the geared machines, is typically dominated by the effects of the tooth meshing and shaft revolution frequencies, their harmonics and sidebands, caused by low (shaft) frequency modulation of the higher toothmesh frequency components. Typically, the contribution from the gear meshing components dominates the overall contents of the measured gearbox vibration spectrum. Transmission Error (TE) is one of the most important and fundamental concepts that forms the basis of understanding vibrations in gears.

What made the TE so interesting for gear engineers and researchers was its strong correlation to the gear noise and the vibrations. TE can be measured by different types of instruments. Some commonly used methods are: Magnetic signal methods, strain gauge on the drive shaft, torsional vibration transducers, tachometers, tangential accelerometers and rotary encoders systems. According to Smith [1], TE results from three main sources: 1) Gear geometrical errors, 2) Elastic deformation of the gears and associated components and 3) Errors in mounting. Figure 1. illustrates the relationship between TE and its sources. Transmission Error exists in three forms: 1) Geometric, 2) Static and 3) Dynamic. Geometric TE (GTE) is measured at low speeds and in the unloaded state. It is often used to examine the effect of manufacturing errors. Static TE (STE) is also measured under low speed conditions, but in a loaded state. STE includes the effect of elastic deflection of the gears as well as the geometrical errors. Dynamic TE (DTE) includes the effects of inertia on top of all the effects of the errors considered in GTE and in STE. The understanding of the TE and the behaviour of the machine elements in the geared transmission system leads to the development of realistic gear rotor dynamics models.

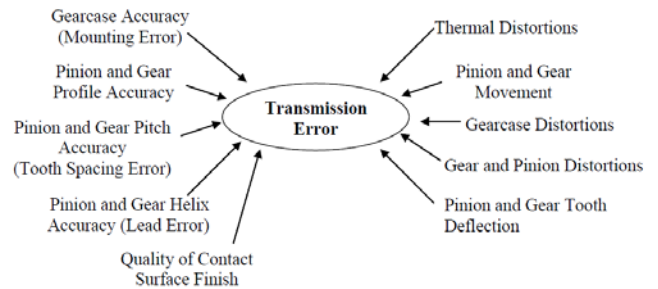


Fig.1 Sources of transmission error

2. Driven group of the plants

The drive is shown in Fig.2. The plant consists of a motor, hydro clutch, brake and gearbox. Disc brakes stop the system (eg. In an emergency). Clutch transmits power from the drive drum gearbox. Hydraulic (fluid) clutch allows softer start. The strength of the gearbox drive drum is transmitted by the clutch flange. The complete plant is mounted on the gearbox frame that is supported by a steel structure through backbone of torque.

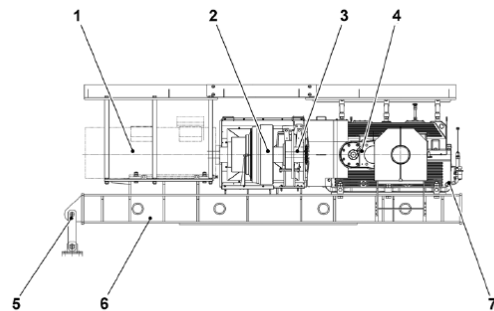


Fig. 2 Drive group of the plant's

Schematic representation of the gearbox with his measuring points is shown in Figure 3. In order to define the optimal area frequently, few measurements have been made in the area frequently (0 to 1000 or 2000) Hz. Since the dynamic behavior of the transmitter but not stationary character level of vibration is changing over time without any time dependence, three complete measurements, as each measurement is recorded with eight results. Each driveline has ten measuring points - probes for temperature monitoring and measuring ten places - probes for monitoring the vibration of bearings electric motor bearings and shaft. This system is completely automated. Managing and monitoring the performance of this system is carried out from the control room. In the control room through SKF software have visually monitoring the vibration of bearings all propulsion groups of transporters. Selected is the displacement amplitude of vibration (PEAK TO PEAK) to be the parameter to be monitored and obtained directly from the instrument, and it is up to the appropriate software that is delivered with equipment used for diagnosis and monitoring of driveline. Also, the vibration acceleration parameter is necessary in certain calculations and it is registered.

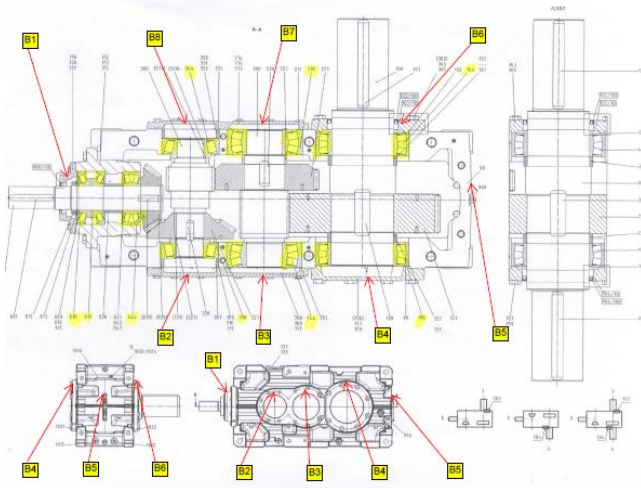


Fig.3 Schematic representation and photos of the gearbox with his measuring points

Also shown below are photos from inside of gearbox figure 4.

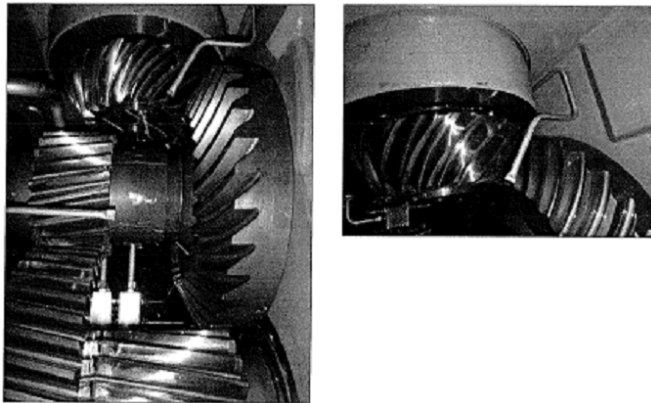


Fig.4 inside of the gearbox

Reducers used with the following data are an integral part of the transport driving groups of bands:

Gearbox: SBN500 type "A"; $i=28$ with irreversible brake

Coupling: FLENDER FVO655

Electric: AO Y-500 4-315 [kW]

Initially there were measurements vibration plants (T5.1 and T5.2) in order to determine the vibration condition of the plants. The results of the measurements of vibration of the plants (T5.1 and T5.2) are shown in the tables below which gives the instrument (Fig.5).

Last Measurement Report

16-JUN-11

Report Description Last Measurements

ID	Units	Date	Last Value	Previous Value	Percent Change	Alarm Status
Machine: T-5/1' Desc: DEPONJA						
1H	mm/sec	10-MAR-10	1.941	1.603	21.1	
1V	mm/sec	10-MAR-10	2.394	4.47	-46.5	
1A	mm/sec	10-MAR-10	2.64	5.626	-53.1	
2H	mm/sec	10-MAR-10	2.023	2.184	-7.4	
2V	mm/sec	10-MAR-10	2.898	3.593	-19.4	
2A	mm/sec	10-MAR-10	2.718	3.834	-29.1	
3H	mm/sec	10-MAR-10	1.877	4.408	-57.4	
3V	mm/sec	10-MAR-10	3.616	4.623	-21.8	
3A	mm/sec	10-MAR-10	2.001	4.121	-51.5	
4H	mm/sec	10-MAR-10	1.489	3.78	-60.5	
4V	mm/sec	10-MAR-10	3.778	5.451	-30.7	
4A	mm/sec	10-MAR-10	2.606	3.149	-17.2	
Machine: T-5/2' Desc: DEPONJA						
1H	mm/sec	28-OCT-09	1.684	1.798	-6.3	
1V	mm/sec	28-OCT-09	3.773	3.835	-1.6	
1A	mm/sec	28-OCT-09	2.821	1.57	79.6	
2H	mm/sec	28-OCT-09	1.762	1.816	-2.9	
2V	mm/sec	28-OCT-09	4.011	3.577	12.1	
2A	mm/sec	28-OCT-09	4.451	2.058	116.3	
3H	mm/sec	28-OCT-09	2.901	2.741	5.8	
3V	mm/sec	28-OCT-09	4.649	3.363	38.2	
3A	mm/sec	28-OCT-09	2.523	2.724	-7.4	
4H	mm/sec	28-OCT-09	3.193	2.604	22.6	
4V	mm/sec	28-OCT-09	2.343	3.732	-37.2	
4A	mm/sec	28-OCT-09	2.646	3.086	-14.3	

Waterfall Spectrum Plot DEPONJA T-5/1' 1H Page 1

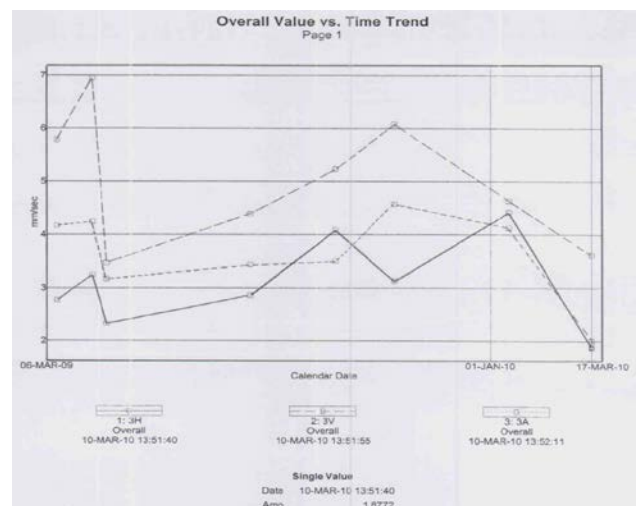
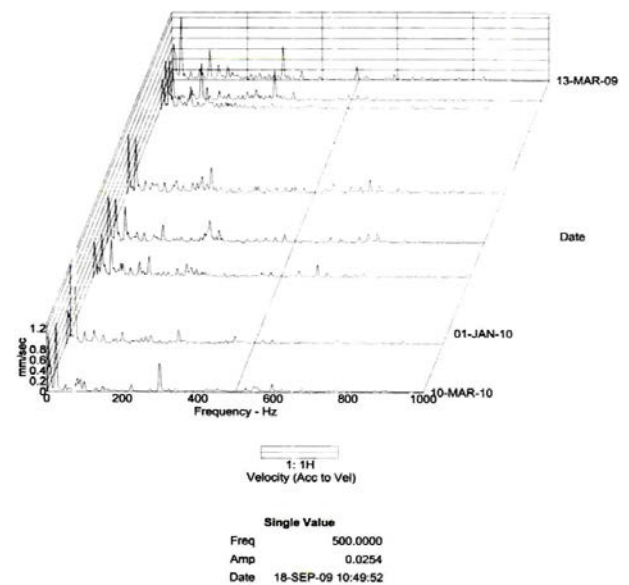


Fig.5 Tables and spectrum which gives the instrument

3. Gear dynamics models

It is a standardized design procedure to perform STE analysis to ensure smoothly meshing gears in the loaded condition. However, a more realistic picture of the gear's dynamic properties can not be

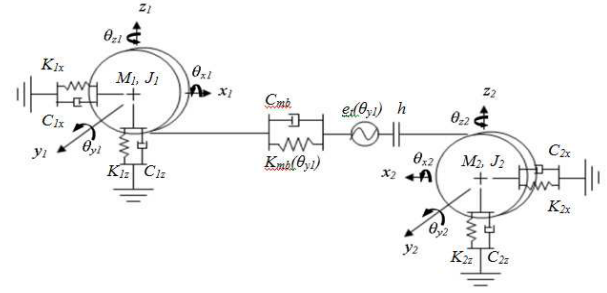
captured without modelling the dynamics of the assembled gear drive system. Solution of engineering problems often requires mathematical modelling of a physical system. A well validated model facilitates a better understanding of the problem and provides useful information for engineers to make intelligent and well informed decisions. A comprehensive summary of the history of gear dynamic model development is given by Ozguven and Houser [2]. They have reviewed 188 items of literature related to gear dynamic simulation existing up to 1988. In Table 1, different types of gear dynamics models were classified into five groups according to their objectives and functionality.

Traditionally lumped parameter modelling (LPM) has been a common technique that has been used to study the dynamics of gears. Wang [3] introduced a simple LPM to rationalize the dynamic factor calculation by the laws of mechanics. He proposed a model that relates the GTE and the resulting dynamic loading. A large number of gear dynamic models that are being used widely today are based on this work. The result of an additional literature survey on more recently published materials by Bartelmus [4], Lin & Parker [5], Gao & Randall [6], Amabili & Rivola [7], Howard et al [8], Velex & Maatar [9], Blankenship & Singh [10], Kahraman & Blankenship [11] show that the fundamentals of the modelling technique in gear simulations have not changed and the LPM method still serves as an efficient technique to model the wide range of gear dynamics behaviour. More advanced LPM models incorporate extra functions to simulate specialized phenomena. For example, the model presented by P. Velex and M. Maatar [12] uses the individual gear tooth profiles as input and calculates the GTE directly from the gear tooth profile. Using this method they simulated how the change in contact behaviour of meshing gears due to misalignment affects the resulting TE.

Table 1.

Group-1: Simple Dynamic Factor Models
Most early models belong to this group. The model was used to study gear dynamic load and to determine the value of dynamic factor that can be used in gear root stress formulae. Empirical, semi-empirical models and dynamic models constructed specifically for determination of dynamic factor are included in this group.
Group-2: Models with Tooth Compliance
Models that consider tooth stiffness as the only potential energy storing element in the system. Flexibility of shafts, bearings etc is neglected. Typically, these models are single DOF spring-mass systems. Some of the models from this group are classified in group-1 if they are designed solely for determining the dynamic factor.
Group-3: Models for Gear Dynamics
A model that considers tooth compliance and the flexibility of the relevant components. Typically these models include torsional flexibility of shafts and lateral flexibility of bearings and shafts along the line of action.
Group-4: Models for Geared Rotor Dynamics
This group of models consider transverse vibrations of gear carrying shafts as well as the lateral component (NOTE: Transverse: along the Plane of Action, Lateral: Normal to the Plane of Action). Movement of the gears is considered in two mutually perpendicular directions to simulate, for example, whirling.
Group-5: Models for Torsional Vibrations
The models in the third and fourth groups consider the flexibility of the gear teeth by including a constant or time varying mesh stiffness. The models belonging to this group differentiate themselves from the third and fourth groups by having rigid gears mounted on flexible shafts. The flexibility at the gearmesh is neglected. These models are used in studying pure (low frequency) torsional vibration problems.

FEA has become one of the most powerful simulation techniques applied to broad range of modern Engineering practices today. There have been several groups of researchers who attempted to develop detailed FEA based gear models, but they were troubled by the challenges in efficiently modelling the rolling Hertzian contact on the meshing surfaces of gear teeth. Hertzian contact occurs between the meshing gear teeth which causes large concentrated forces to act in very small area. It requires very fine FE mesh to accurately model this load distribution over the contact area. In a conventional finite element method, a fully representative dynamic model of a gear requires this fine mesh over each gear tooth flank and this makes the size of the FE model prohibitively large. In our research we use Lumped parameter model (LPM) is shown in figure 6.



- x_i, y_i, z_i Translation at i th Degrees of Freedom.
 $\theta_{xi}, \theta_{yi}, \theta_{zi}$ Rotation about a translational axis at i th Degrees of Freedom.
 C, C_{mb} Damping matrices. The subscript 'mb' refers to damping at the gearmesh. Typically for C_{mb} , $\zeta = 3 \sim 7\%$.
 K, K_{mb} Linear stiffness elements. The subscript 'mb' refers to stiffness at the gearmesh.
 h An 'on/off' switch governing the contact state of the meshing gear teeth.
 e_t A vector representing the combined effect of tooth topography deviations and misalignment of the gear pair.
 i Index: $i=1, 2, 3 \dots$ etc.

Fig.6 A Typical Lumped Parameter model of meshing gears

Vibration of the gears is simulated in the model as a system responding to the excitation caused by a varying TE, 'et' and mesh stiffness 'Kmb'. The dominant force exciting the gears is assumed to act in a direction along the plane of action (PoA). The angular position dependent variables 'et' and 'Kmb' are expressed as functions of the pinion pitch angle ($\theta y1$) and their values are estimated by using static simulation.

Equations of motion derived from the LPM are written in matrix format as shown in Equation-1. The equation is rearranged to the form shown in the Equation-2; the effect of TE is expressed as a time varying excitation in the equation source. The dynamic response of the system is simulated by numerically solving the second order term (accelerations) for each step of incremented time. The effect of the mesh stiffness variation is implemented in the model by updating its value for each time increment.

$$M\ddot{\underline{x}} + C(\dot{\underline{x}} - \dot{\underline{e}}_t) + K(\underline{x} - \underline{e}_t(\theta)) = F_s \quad (1)$$

$$M\ddot{\underline{x}} + C\dot{\underline{x}} + K\underline{x} = F_s + hC_{mb}(\theta)\dot{\underline{e}}_t + hK_{mb}(\theta)\underline{e}_t \quad (2)$$

where,

- $\underline{x}, \dot{\underline{x}}, \ddot{\underline{x}}$ Vectors of translational and rotational displacement, velocity and acceleration.
 θ Angular position of pinion.
 K, K_{mb} Stiffness matrices (where K includes the contribution from K_{mb}). The subscript 'mb' refers to stiffness at the gearmesh.
 C, C_{mb} Damping matrices (C including contribution from C_{mb}). The subscript 'mb' refers to damping at the gearmesh.
 h An 'on/off' switch governing the contact state of the meshing gear teeth.
 F Static force vector.
 $\underline{e}_t, \dot{\underline{e}}_t$ A vector representing the combined effect of tooth topography deviations.

Simulated gearbox vibration signal for 840rpm $T=15\text{Nm}$; $T=67\text{Nm}$ and $T=120\text{Nm}$ is shown in fig.7

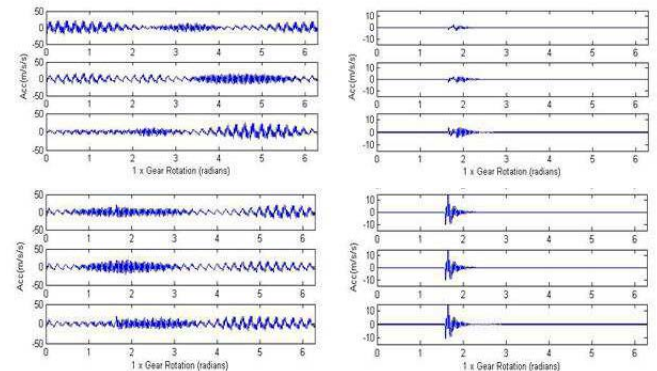


Fig 7. Simulated vibration for 840rpm

In machine condition monitoring signal processing techniques are often developed to detect and quantify the symptoms of a damage buried in a background noise. By being able to see the "clear" effect of a fault, the most effective signal processing technique can be applied to target and monitor the symptoms of the damage. In actual applications the signal is complex and must be carefully analysis. (Fig.8)

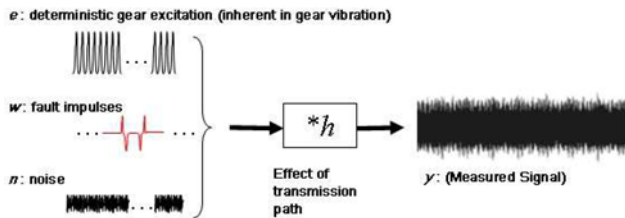


Fig.8 Vibration of a Gearbox

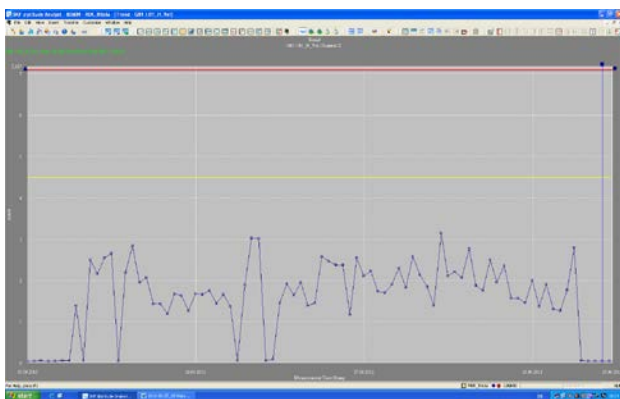
3. RESULTS AND DISCUSSION

The fault detection and diagnostic techniques based on vibration signal analysis are the ideal non-destructive machine health monitoring method, that can be applied in a minimally intrusive manner; i.e. by attaching an accelerometer on a gearbox casing.

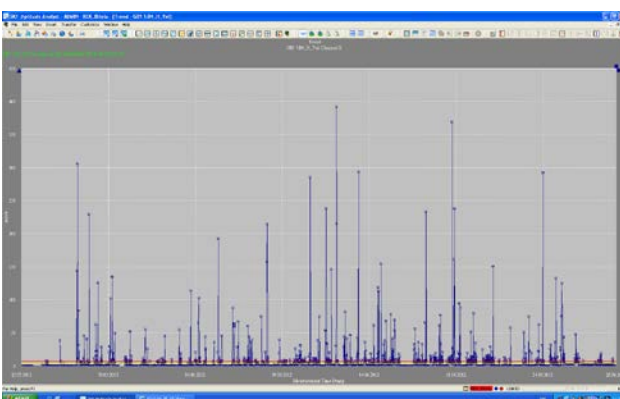
However, the dynamic interaction amongst the machine elements of a gearbox is often complex and the vibration signals measured from the gearbox is not easy to interpret. The diagnostic information that directly related to an emerging fault in a gear or a bearing is typically buried in the dominating signal components that are driven by the mechanisms of the transmission system themselves: For example, gear meshing signals.

The following presents the results of measurements of suitable measuring points of driveline.

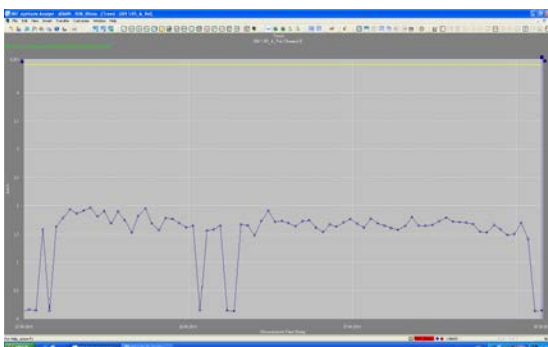
TL5 GB1 Sensor 1



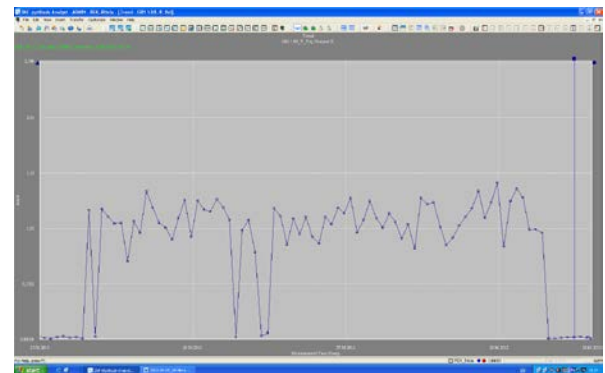
TL5 GB1 Sensor 4



TL5 GB1 Sensor 5



TL5 GB1 Sensor 8



The dynamic model (LPM) used in simulation of vibration gearbox preventive and allows us to quickly we can identify possible defects that would have occurred in the gearbox especially bearing vibration that would cause malfunction of drive. Therefore, the presented data from eight measuring points shown are the results of measuring the signals of bearings. This model can be modified but its experimental verification requires large financial resources.

4. CONCLUSION

Some general conclusions as a part of the researching:

- The main purpose of the damage modelling is to simulate the effect of the faults on the dynamics of a geared transmission system that can be used in improving the understanding of the diagnostic information that manifest in the vibration signal mix from a gearbox.
- The observation of the simulated residual signals has led to an improved understanding of the characteristics of impulses caused by the faults and the distorting effect of the transmission path (from the origin of the fault signal to the measurement location). The improved understanding of the fault signal obtained from the simulation studies led to the development of more effective signal processing techniques.
- In the near future, accurately simulated signals of a faulty gearbox can aid the machine learning process of fault diagnosis algorithms based on neural networks. Performing this task in experiments are time consuming and costly exercise; simulation model based approach appears much more desirable.

It is necessary to continue monitoring the behavior of the drive group and perform vibration measurements from time to time with the same assessment of the safety aspect of plants.

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