VIBRATION DIAGNOSTICS OF PLANETARY GEARBOX OF ENGINEERING AND TRANSPORT MACHINES BASED ON SPATIAL MODEL APPLICATION

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Abstract. Planetary gearboxes are used widely and sudden failure of gearbox may cause large economic losses. Although vibration diagnostics is widely applied to detect failures in gears, planetary stages still create a problem, because of multiple modulated signals causing its vibration the signal of the faulted gear is masked by other well operating gears. The technique is required that would overcome the above problem and provide detection of any potential failures of planetary/differential gears, including planet driver cracking, assembling errors, etc.The paper considers the advanced technique that is capable to solve the problem. The authors discuss spatial-time approach to gear interactions in a planetary gearbox that are described by the impulse mesh model. As the tool for the above approach the Spatial Time Domain Distribution (STDD) is proposed. The paper discusses potential application cases for the spatial mesh model as estimation of dynamic loads distribution between any components of the planetary gear. Based on the results of analytical works the authors make the conclusion about STDD capability to become the basis of vibration diagnostic techniques for technical condition monitoring of planetary gearboxes in engineering and transport machines.

Key words: vibration, modelling, data transformation, diagnostics, planetary gears.

1. Introduction

Planetary gearboxes are used in different engineering and transport machines, including those applied in rural development. There is a wide range of such machines starting from widely used heavy excavators and trucks up to advanced automated guided vehicles (AGV) that distribute picked goods in robotized industrial greenhouses. Sudden failure of gearbox components, like ring sun, planets or its carrier may cause large economic losses as the above mentioned machines act as a link of united economic and industrial chain. For condition monitoring of such machines vibration diagnostic techniques have found the widest application. In regard to the gearbox, the principal task of vibration diagnostics is to detect in time fatigue cracks or any other failures caused by heavy loads or occasional factors. Planetary stages of gearboxes became the biggest issue because of its complicated vibration signature that contains multiple modulated signals. Different vibration analysis techniques developed over the last decades provide breakthrough in planetary gearboxes diagnostics. However, their wide application met some limitations.

Many works consider diagnostic indicators of gear problems in frequency domain using spectral analysis [1], spectral characteristics of vibration signal [2], energy ratio based on difference spectra [3] and so called non-iterative deconvolution approach [4]. The Harmonic Index feature in [5] is defined as the amplitude sum of all apparent sidebands of a specific gear meshing harmonic of the raw data.To separate modulations in a planetary gearbox, a modulation signal bi-spectrum based sideband estimator was developed and used to achieve a sparse representation for the complicated signal contents, which allows effective enhancement of various sidebands for accurate diagnostic information [6]. Frequency analysis of experimental data in frequency domain is also used for diagnostic indicators search [7] that combines the techniques of enveloping, Welch's spectral averaging and data mining-based fault classifiers. The technique using time windows for initial vibration separation [8-10] demonstrated more effective approach and found the widest application for the helicopter planetary gearbox. Time Signal Averaging (TSA) acknowledged earlier for vibration of ordinary gears is applied for planetary ones after preliminary separation by the time window. The width of the vibration window is equal to a single tooth mesh period. This window opens each time, when the planet gear passes to capture the vibration signal section. This captured signal length conforms to the time window width. The phase position of the window opens at the moment, when the selected tooth of specific planet passes the transducer attached to the annulus (ring) gear. As the window width is small in relation to the carrier rotation period, the transfer function between the zone of the tooth contact and the transducer can be considered constant. Later followers [11] offer to apply multiple time windows according to fixed ring gear teeth number. This way transfer features of vibration paths between each tooth and a transducer are taken into account.

Cumulative effect from the developed different vibration analysis techniques allows a breakthrough in planetary gearboxes diagnostics. However, their wide application discovers some limitations:

- TSA and similar algorithms require high resolution phase signal. Unfortunately, gearboxes of industrial machines have no phase indicators, but tachometers only.
- Most techniques consider using only part of measured signals separated by windows. Therefore, for evaluation of all planetary gears of a gearbox TSA requires hundreds to thousands rotations of the gearbox for vibration data collection. For some failures even few minutes could be critical and may generate risks of heavy losses.
- Well known techniques do not consider more than one measurement direction of vibration and no one uses spatial (3D) vector vibration measurements.
- TSA is well developed for planetary gears with a fixed ring gear, but there are many other types of planetary and differential gearboxes where all gears rotate.

Fatal accidents of the helicopter caused by destruction of planet carrier had outlined the problem of the existing diagnostic techniques and great needs in an advanced one. The industry also needs such technique capable to estimate unevenness of dynamic loads distribution between planet gears. The problem is caused by frequent faults of both planet gears and their bearings that some technological mistakes in assembling of the gearbox might cause.

Taking the above considered limitations into account the set of requirements is developed to the advanced approach for vibration diagnostic of planetary gears, which has to support the following aims:

- expansion of vibration diagnostic area application, including differential and multistage gearboxes;
- diagnosis of gearboxes without phase indicators, provided by tachometers only;
- monitoring of the helicopter gearbox in real time.

To achieve above targets the advance technique must provide ability to monitor the planetary gearbox in real time; 3D vibration signal optimal utilization; detection of any potential failures of planetary/differential gears, including planet carrier cracking, assembling errors, etc.

2. Spatial-time model of gear interactions

2.1. Data separation

Further consideration is based on the sample of a planetary gearbox consisting of N fixed planet gears (z_g teeth each) and both rotating ring (z_r) and sun (z_s) gears. Vibration paths of N planets mesh differ a little bit. Typical TSA for vibration signal separation applies the time window that is equal to the interval of single tooth mesh. Based on the number of teeth z_r and speed ω_r this interval is

$$\Delta T_r \frac{2\pi}{z_r \omega_r}, \tag{1}$$

Such approach works well when the transducer is fixed to the immovable ring gear next to the mesh zone. In this case the closest mesh energy reaches the transducer almost undamped but the opposite ring tooth energy is damped multiple times. However, for common cases, where vibration paths of different teeth vary slightly, effectiveness of such windowing is limited. The problem is that actually within the time interval ΔT_r all N planets interact simultaneously with both the ring and sun gears. It means $2N$ meshes occur during typical TSA window ΔT_r and by applying such time interval, one sums up responses of $2N$ meshes.

To separate responses of $2N$ meshes the width of the window must be $2N$ times less than in (1)

$$\Delta T_w = \frac{2\pi}{2N z_r \omega_r} = \frac{T_r}{2N z_r}, \tag{2}$$

where T_r – rotation period.

So, separation of vibration data by the single mesh window ΔT_w provides data for each elementary interaction of gears.

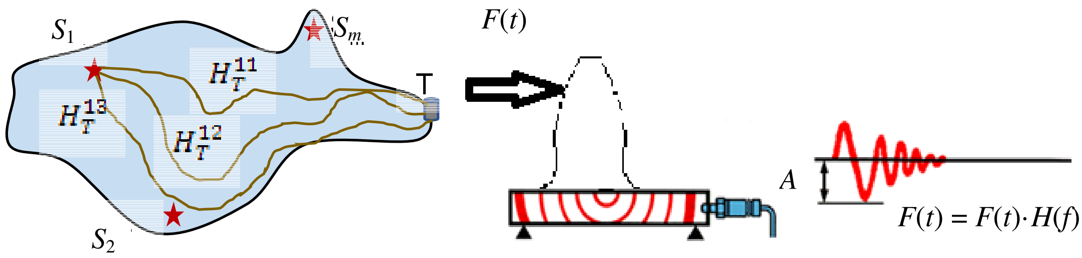
2.2. Gear mesh modelling

Impulse meshing

The shock force acting on a tooth grows from the touchdown moment until the moment of its maximum. Then the force is falling down until teeth disconnection. In the considered gear model there are more than $2N$ mesh points as each satellite is in mesh with the sun and ring gears by more than one tooth simultaneously. There is a problem to separate summarized response based on smooth phase variation from $2N$ mesh points, if its vibration paths differ a little. However, impulse phases of the above meshes follow each other with average time shift ΔT_w (2) that allows separation in time responses of a transducer to the above impulses. Fig. 1, b illustrates formation of the force $F(t)$ impulse from each mesh. Thus, a transducer on a gearbox housing may sense signals from each mesh separated in time. Therefore, the impulse interactions of planetary gears teeth allow its separation in time and based on gear kinematics allows identification of the above signal sources.

Vibration frequency range

Transfer properties of vibration paths between the mesh points and the transducer determine the response of the last one. Transfer properties for each of K vibration paths from the source S to the transducer T (Fig. 1, a) are described by the transfer function K_s^k.



a) b)

Fig.1. **Vibration paths and signal propagation**

Vibration paths differ from each other by length, materials, joints, etc. Integral transfer function H_Σ accounts for all paths between the source and transducer.

$$H_\Sigma^1(f) = \sum_{k=1}^{K} H_s^k(f), \tag{3}$$

Each $H_1^k(f)$ describes specific transfer features of the path, through which the wideband frequency (impulse) signal from the mesh point S_1 passes. The impulse signal propagates in material as stress waves and achieving the transducer actuates the electric signal (Fig. 1, b). The wave shape and velocity vary along the path, so $H_s^k(f)$ are essentially nonlinear and have many resonances.

Amplitude A of the response (Fig.1, b) may be used for estimation of excitation by measuring peak-peak or total energy of the response within the time window ΔT_w. Keeping this in mind, the sampling frequency F_d must be at least twice the frequency of dominating the vibration components. That is why the structure of actual gearbox vibration has to be considered preliminary to determine dominants that depend both on actuation and on vibration path features.

For the considered planetary gear the number of meshes in one period is $2Nz_r$. So the excitation frequency is estimated as $F^{imp} = 2Nz_r\omega_r$ that for typical gearboxes varies within 3-10 kHz.

Elastic (stress) waves carrying energy of impulse excitations have combined structure containing both longitudinal and transverse waves. These waves act as a carrier of impulse excitations from teeth mesh to a transducer. This wave carrier frequency has relatively wide frequency range as vibration paths include different materials and joints. According to the experimental research conducted by SIA ”D un D centrs” the main vibration energy of vibration paths for typical gearboxes was observed in 3-7 kHz band.

Thus, depending on the gearbox type both excitation and its carrier frequency can be the maximal frequency to be sampled. To describe the waveform of a response 10 samples are considered as minimal so, for a gearbox with F^{imp} = 7 kHz the sampling frequency must be at least 70 kHz.

Identification of gear tooth mesh

Vibration data measured can be utilized for 100 %, if the mesh window ΔT_w of equation (2) to be applied $2Nz_r$ times for one period T_r of ring gear rotation. The time shift between multiple windows is the same as the window width $\Delta t = \Delta T_w$. So, starting to apply mesh windows to vibration from initial moment of period one will have the data of Nz_r mesh windows shifted equally for the whole period. The vertical line on Fig.2a schematically reflects the time vector of one period of ring gear revolution and the data ensemble of $2Nz_r$ mesh windows. Based on known step width one can calculate position

$$p_j = j\frac{T_r}{2Nz_r} \tag{4}$$

of j^{th} mesh within the period.

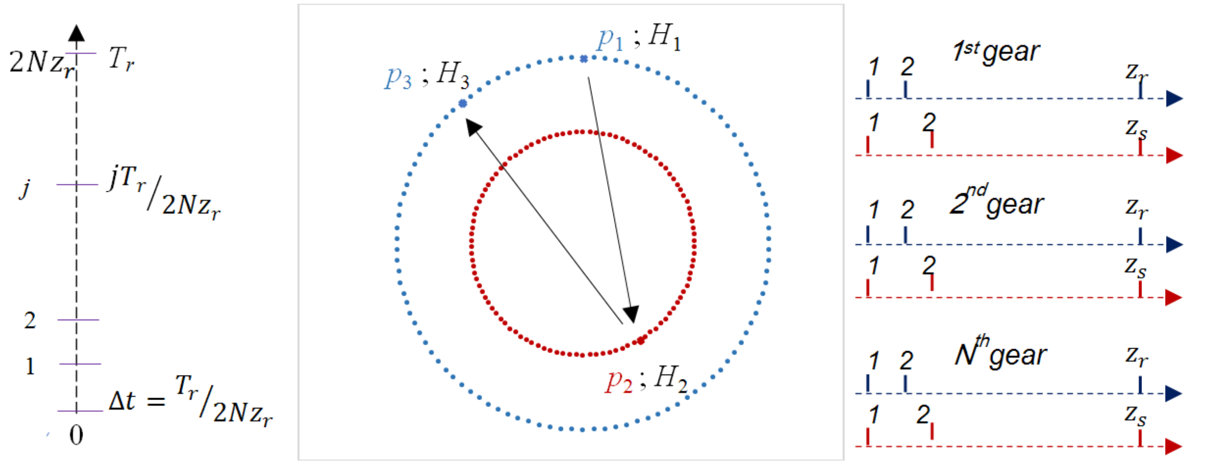


Fig. 2. **Schematic illustration of teeth mesh sequence:** a – time sequence of mesh data; b – spatial model presentation; c – mesh data redistributed to structural order

Sequential number j of the mesh on a scale $(0\ldots2Nz_r)$ means p_j moment of the period and corresponds to spatial position of the mesh (n_{il}, r_m, s_p). This mesh is related to n_i planet and its l tooth as well as m tooth of the ring and p tooth of the sun. Such positioning of each mesh is determined by kinematics data.

Based on kinematics of the planetary gear the next interacting tooth number of the ring and sun can be calculated using mesh step

$$\Delta z = \mathrm{int}\left(\frac{z}{N}\right).$$

The next tooth number for the ring and sun:

$$z^r_{j+1} = z^r_j - \mathrm{int}\left(\frac{z_r}{N}\right), \tag{4a}$$

$$z^s_{j+1} = z^s_j - \mathrm{int}\left(\frac{z_s}{N}\right), \tag{4b}$$

where the negative index means mesh sequence following contrary to the rotation direction.

Planet gears interact with the ring and sun with single step

$$z^{gr}_{j+1} = z^{gr}_j - 1, \tag{5a}$$

$$z_{j+1}^{gs} = z_j^{gs} - 1. \tag{5b}$$

Number of the next interacting planet with the ring and sun:

$$n_{j+1}^{r} = n_j^{r} - 1, \tag{6a}$$

$$n_{j+1}^{s} = n_j^{s} + 1, \tag{6b}$$

Fig. 2, b illustrates the example of initial three teeth mesh according to equations (4-6). The blue dotted circle shows allocation of ring-gear mesh points, the red one – points of sun-gear meshes. Taking the ring-planet gear as the first mesh (p_1), the second one (p_2) will happen between another planet-sun and then the third mesh (p_3) will be between the third planet and ring. Being allocated along above circles each mesh p_j is related to the teeth pair: one of the planet gear and another with the ring or sun gear. Such allocation could be described using polar coordinates and its centre coinciding with the ring and sun gear rotation axis. Polar coordinates of the spatial mesh model are determined by the gear construction and kinematics. Equations (4-6) describe allocation of mesh points along polar coordinates according to the order of interactions.

Using the gearbox dimensions the polar coordinates of each mesh can be converted to coordinates X_j, Y_j, Z_j of 3D system with a 3-axial transducer at the system zero point.

By this way p_j mesh located in specific position (n_{ij}, r_j, s_j) of the planetary gear has also geometrical coordinates as the point of solid body the having transfer function $H_j = H(n_{il}, r_m, s_p)$.

Therefore, based on kinematics rules and known data of the planetary gear each single mesh window can be definitely related to a specific teeth pair and the identified mesh point can be related to a transducer by the predetermined transfer function.

3. Spatial mesh model

3.1. Mesh points allocation

The above consideration allows modelling of the planetary gear as spatial allocation of meshes playing role of dynamic sources for housing vibration. The spatial model of dynamic interactions in the planetary gear considers one period of gearbox operation that is a single revolution of a typical output shaft. Totally there are $2Nz_r$ meshes in one period, where each mesh generates the dynamic force impulse. Each mesh relates to its own position p_j in spatial domain and the impact from each position is transformed to vibration by the transfer function $H_j(f)$. Physically mesh points are allocated along two concentric circles with Nz_r mesh points on each.

Being allocated around any mesh p_j is related to the teeth pair: one of the planet gear and another of the ring or sun gear. Polar coordinates of the spatial mesh model are determined by the gear construction and kinematics. Equations (4-6) provide the number for each mesh point allocated in polar coordinates according to the order of interactions. To relate spatially allocated mesh points to the transducer the 3D coordinates of each mesh must be calculated. For this purpose, polar coordinates are converted into 3D coordinate system with a 3-axial transducer at the system zero point. Thus, the spatial model provides for each mesh point its:

- 3D coordinates X_j, Y_j, Z_j in relation to the transducer on the gearbox housing,
- individual transfer function to the transducer $H_j(f)$ based on above coordinates,
- sequential number of interaction from the period beginning.

3.2. Potential application cases for spatial mesh model

As described above the spatial model relates sequential numbering of mesh points to specific teeth numbers of each planetary gear. Each of $2Nz_r$ meshes within one period is related to mesh of two specific teeth of the planet and the ring or sun gear. To extract information about all teeth the raw data of a period must be rearranged, which means their replacement from time sequence to structural order of tooth numbering. The set of routines for data redistribution according to the spatial mesh model is combined into Spatial Time Domain Distribution (STDD).

The data developed by STDD routines may be accumulated, enhanced and averaged aiming to estimate dynamic loads distribution between any components of the planetary gear.

For example, using data of one period the set of $2kz_r$ samples for each of N planets will be obtained for interaction with both the ring and the sun. Grouped by k samples this data can serve for estimation of load distribution between z_r teeth ofring if one averages k samples for each tooth. The same way can be used for z_s teeth of the sun.

If one assembles data by kz_r values in N groups, then it is possible to calculate the load distribution between the planet gears. Hundreds of averages will be available, taking into account the kinematic data of a typical planetary gearbox and the sampling frequency of about 70 kHz. Such averaging could be considered as adequate to estimate dynamic load distribution between the planet gears. For estimation of teeth distribution at any gear more periods for data accumulation and averaging could be used.

Dynamic load distribution between planet gears is considered as the most sensitive diagnostic tool for early detection of planet carrier cracks. This parameter may also be used for quality control in industry for a gearbox tested after assembly.

4. Discussions and conclusions

Based on the above consideration we may say that using kinematics rules and known data of the planetary gear each single mesh window can be definitely related to a specific teeth pair and identified mesh point. Each mesh impulse then can be related to a transducer by a predetermined transfer function. That means all information contained in vibration signal may be used for diagnostics in contrary to already used versions of TSA techniques. The set of routines for data redistribution according to the spatial mesh model are combined into Spatial Time Domain Distribution (STDD). The STDD approach based on the impulse mesh model of gear interactions allows monitoring of the planetary gearbox in real time that is the basic requirement to the advanced diagnostic technique. This is an actual advantage in comparison to other known techniques that require much more time for data accumulation. Such advantage provides vibration signal optimal utilization, detection of any potential failures of planetary/differential gears, including planet carrier cracking, assembling errors, etc.

To provide advantages of the impulse mesh model and STDD technique the technical requirements are to be satisfied. To relate definitely each single mesh window to specific tooth high sampling frequency and wide dynamic range has to be provided. STDD technique requires clear understanding of kinematics rules and also analysis of the gearbox structure and experimentally defined transfer functions to the transducer.

The dynamic load distribution between the teeth is considered as an effective tool for specific tooth fault detection. The same distribution between the planet gears is considered as the most sensitive among different diagnostic parameters for early detection of planet carrier cracks in operating gearboxes.

Further experimental works are supposed for STDD effectiveness estimation and comparison with other vibration diagnostic techniques for planetary gears.

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