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Evaluation of gear noise behaviour with application force level

Conference proceedings

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Abstract Reducing excessive vibration and undesired noise is one of the most important goals throughout the whole development phases of gearboxes. Among different development stages, different technical measurements are available to estimate the excitation behaviour of gear meshes.

However, the comparison of the result of these measurement types among the development stages may be inconsistent due to different dimensions. In order to ensure the best comparability as well as comprehensive interpretation of the vibration characteristics of gear meshes, an application force level was developed. This characteristic value allows noise behaviour evaluation by means of different measurement types. Moreover, the evaluation boundary of this characteristic value is customizable to encourage the flexibility for further specific evaluations.

In this article, the formulation of application force level is derived and its usage is shown. Although the application force level was intentionally developed for the gear design, it can also be applied to other kinds of application fields. The application force level is the result of the research project funded by the German research association for drive technology (FVA).

Zusammenfassung Bei der Entwicklung von Getrieben ist ein günstiges Anregungsverhalten von Zahnrädern einer der wichtigsten Aspekte. In den Entwicklungsphasen können unterschiedliche Größen zur Beurteilung des Anregungsverhaltens verwendet werden, deren Vergleichbarkeit nicht immer gegeben ist aufgrund unterschiedlicher physikalischer Größen.

Um die durchgängige Auswertung des Anregungsverhalten in allen Entwicklungsphasen zu ermöglichen, wurde der Anwendungskraftpegel entwickelt. Mit Hilfe des Anwendungskraftpegels kann das Anregungsverhalten von Zahnrädern auf Basis unterschiedlicher Daten unter Berücksichtigung eines Auswertebereichs ausgewertet werden. Der Auswertebereich ist anwendungsspezifisch definierbar. Dadurch ist dieser Kennwert flexibel für verschiedenste Fälle einsetzbar.

Nachfolgend werden die Formulierung des Anwendungskraftpegels und Anwendungsbeispiele gezeigt. Obwohl der Kennwert ursprünglich zur Anwendung in Zahnradgetrieben entwickelt wurde, ist die Anwendung des Kennwerts in anderen Anwendungsbereichen möglich.

Der Anwendungskraftpegel ist Ergebnis von Forschungsvorhaben der Forschungsvereinigung Antriebstechnik e. V. (FVA).

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1 Introduction

Noise and vibration harshness is one of major aspects for gear applications. In order to ensure proper acoustic exposure, the reduction of gear noise begins in the very early development stage and is a perpetual procedure throughout

Table 1 Common measurements for the evaluation of gear noise behaviour

Measurement	Source	Dimension
Quasi-static Loaded Transmission Error	Calculation/ Assessment	Displacement
Tooth Force Excitation	Calculation	Force
Dynamic Tooth Force	Calculation	Force
Dynamic Loaded Transmission Error	Calculation/ Assessment	Displacement
Torsional Acceleration	Assessment	Acceleration
Structure-borne noise	Assessment	Velocity/ Acceleration
Airborne noise	Assessment	Pressure

the whole development stages. Proper natural frequencies and excitation frequencies of gearboxes are the result of considerate design of gear macro geometries, for example number of stages, topology of gearboxes, number of teeth, etc. In the latter stages, load distribution on the gear face can be determined. This can directly be influenced by the micro geometries. Therefore, appropriate design of micro geometries reduces the mesh excitation and improves the gear noise behaviour.

Different measurements can be used to resolve the gear noise behaviour. Common measurements, which are used for the evaluation of gear noise behaviour throughout the development stages of gearboxes are shown in Table 1. Calculation measurements to estimate the noise behaviour of gear mesh can be found, for example, in [2, 10]. These technical measurements range from a quasi-static loaded transmission error during the design phase, where only rough details of the gearbox may be provided, to the airborne noise measurement of a test rig or an in-situ gearbox, where the noise behaviour of the real gearbox can be evaluated under the real condition. The comparison of noise behaviour by means of these measurement types can be troublesome due to their different characteristics and orders.

In order to ensure the best comparability as well as comprehensive interpretation of the vibration characteristics of gear meshes, an application force level was developed. This characteristic value allows different measurement types as input. Moreover, the evaluation boundary is customizable, which enables the specific evaluation of noise behaviour.

2 The application force level

In order to enable the evaluation of the gear noise behaviour by means of different data types, the application force level was designed to handle these data in a common way, as similar to the evaluation of gear noise from dynamic measurements [9, 15]. Therefore, the data basis from the Campbell-diagram is used for the characteristic value formulation. The Campbell-diagram contains complete information of gear noise behaviour along the range of operational speed. An example of a Campbell-diagram from a structure borne noise measurement of a gearbox is depicted in Fig. 1. The spectra are represented as dimensionless orders with respect to the pinion speed (pinion shaft order). Therefore, this type of diagram is known as an order diagram.

By means of the representation, the teeth excitation can be seen as the distinct vertical lines in the diagram at a specific order and its multiples. Each order indicates how often the excitation is taking place within a complete shaft rotation. The base order represents therefore the number of teeth of the pinion, as this also represents the recurrences of each mesh excitation within one shaft rotation. In the diagram, the distinctive excitation at the 43rd order suggests that the number of the pinion teeth is 43. This order is also called first mesh order.

From the diagram, intrinsic influences, i. e. natural frequencies, are also visible as hyperbolical lines. Therefore, resonances can be identified, when these hyperbolical lines cross the vertical lines.

Despite the complete information of noise behaviour, the usage of the order-diagram can be inconvenient due to the bulky 3-dimensional representation. A comparison between different gear variants as well as different operational areas can be almost impossible. Therefore, a compression of the diagram into a scalar characteristic value is a good compromise, which still contains the noise behaviour and ensures comparability.

The compression begins with the definition of the investigated area. In Fig. 1 the area is described by 3 parameters, the speed range (horizontal lines), frequency range (hyperbolical lines) and the investigated orders (vertical lines). These ranges are customizable depending on the purpose of the investigation.

The definition of frequency range affects the length of the evaluable order range at each speed. Subsequently, at a specific order, the length of evaluable speed range is restricted

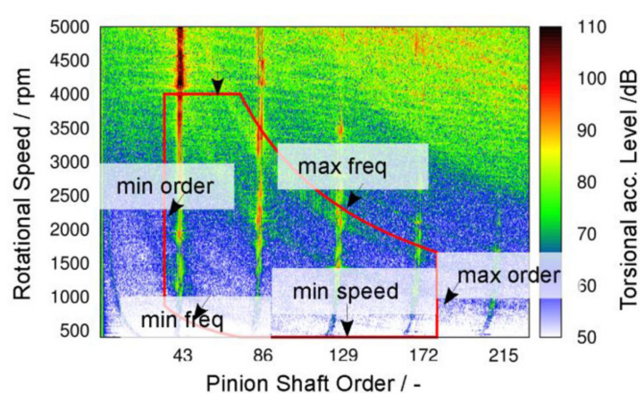

Fig. 1 Order diagram of a torsional acceleration measurement with a predefined investigated area for application force level

due to the frequency range. Thus, the averaged force amplitude along each order can be calculated under consideration of this effect as:

$$\overline{F}_{\mathrm{ord}} = \frac{\int_{n_l(ord)}^{n_u(ord)} F_{\mathrm{ord}}(n)\cdot \mathrm{d}n}{n_u(ord) - n_l(ord)} \cdot \frac{n_u(ord) - n_l(ord)}{n_u - n_l} = \frac{\int_{n_l(ord)}^{n_u(ord)} F_{\mathrm{ord}}(n)\cdot \mathrm{d}n}{n_u - n_l} \qquad (1)$$

$\overline{F}_{\mathrm{ord}}$ [N] Averaged force amplitude at order *ord*
$F_{\mathrm{ord}}(n)$ [N] Force amplitude at order *ord* and speed *n*
ord [–] Order (mesh or shaft order are both possible)
$n_u(ord)$ [rpm] Upper speed boundary at order *ord*
$n_l(ord)$ [rpm] Lower speed boundary at order *ord*
n_u [rpm] Global upper speed boundary
n_l [rpm] Global lower speed boundary

It should also be noted that in (1), the averaged force amplitude resolves the significance of each specific order, where, for example, resonance may be identified.

Finally, the averaged force amplitude can further be compressed into a dimensionless characteristic value, “Application Force Level” by means of level formulation as:

$$L_{A,F} = 10\log\left[\frac{1}{F_{\mathrm{bez}}^2}\sum_{\mathrm{ord}=\mathrm{ord}_l}^{\mathrm{ord}_u}\overline{F}_{\mathrm{ord}}^2\right] \qquad (2)$$

$L_{A,F}$ [–] Application Force Level
ord_u [–] Global upper order boundary
ord_l [–] Global lower order boundary
F_{bez} [N] Reference force according to DIN EN ISO 1683 (= 1×10^{-6} N) [3]

Depending on development stages, different types of force can be used in (2). Therefore, the application force level can be classified into 3 types (methods) according to the completeness of dynamic influences in the force term.

In the very early development stage, the mesh excitation is determined by quasi-static approaches. Different kinds of excitation, like loaded transmission error (path excitation – see also [7, 8]) as well as tooth force excitation (force excitation – see also [13]), can be calculated based on the assumption of the boundary conditions [16]. shows correlation between the loaded transmission error and the measured torsional acceleration of a gearbox. However, the quasi-static approaches neglect the dynamic influences of the system, so that the spectra of the excitation are assumed to be constant along the speed range (see Fig. 2c). In this case, the application force level yields similar result as the conventional tooth force level. In case of path excitation, the value can be roughly transformed into force excitation by means of multiplying with mesh stiffness. The estimation of gear noise behaviour with application force levels with these basic excitations is called the estimation according to method C ($L_{A,F,C}$).

In many cases, dynamic influences of the system may be approximated either numerically or experimentally. These dynamic influences are often described in term of transfer functions (also known as Frequency Response Function – FRF). In practice, the transfer functions can be derived either analytically or from measurements. For example, the analytical transfer function according to Geiser [5], which will be utilized in the later section of this paper, can be used to predict the dynamic force from the tooth force excitation approach. By considering this with the quasi-static excitation, the dynamic influences can be visualized in the order diagram (see Fig. 2b). The estimation of gear noise with application force level of quasi-static excitation under consideration of dynamic influences is called the estimation according to method B ($L_{A,F,B}$).

Finally, if the complete dynamic data of gear mesh is available, either from a dynamic simulation or from a dynamic measurement, the evaluation of gear noise behaviour can be done based on the evaluated order diagram (see Fig. 2a). This is called the evaluation according to method A ($L_{A,F,A}$).

The designation A, B and C is used in compliance with the DIN-Standard convention, which depends on the com-

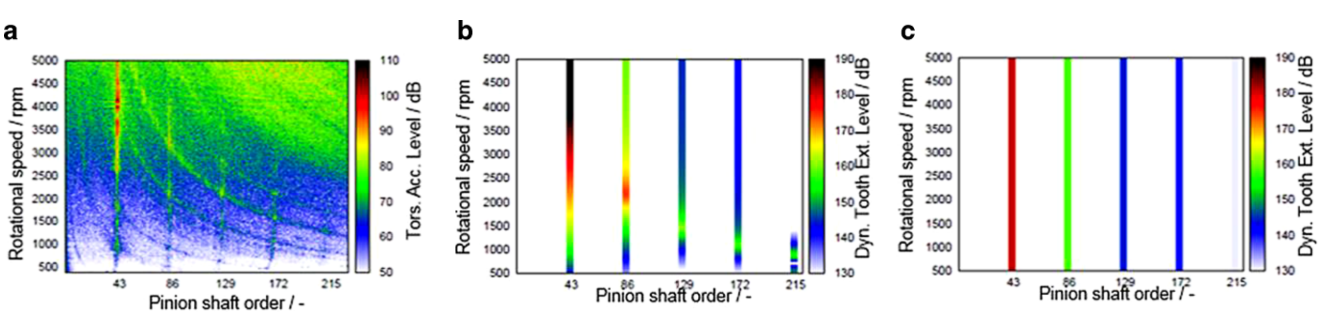

Fig. 2 Order diagrams of different data basis

plexity of data. The evaluation accuracy depends strongly on the utilized method. However, the levels of noise behaviour between different methods can be compared qualitatively due to the dimensionless formulation.

3 Evaluation example of reduction of gear noise by micro geometries

In this example, a usage of application force levels throughout different development stages of a gearbox and their comparability of gear noise evaluation will be shown. The gear macro geometries for the evaluation are shown in Table 2.

During the design stage, the gear noise behaviour can be numerically estimated. A good numerical approach for gear mesh excitation can be found in [4]. In this phase, gear micro geometries can be designed for the reduction of gear noise behaviour. Good examples of the design of gear micro geometries can be found in [11, 14]. In [12], waveform flank modifications were introduced as a potential modification considering the noise behaviour. In this example, the

Table 2 Gear macro geometries

Macro geometries	Symbol	Pinion	Wheel	Unit
Normal module	m_n	3		mm
Pressure angle	α_n	20		°
Helix angle	β_n	–21	21	°
Number of teeth	$z_{1,2}$	43	45	–
Addendum modificaition coefficient	$x_{1,2}$	–0.211	0.237	–
Width	$b_{1,2}$	39.5	39.5	mm
Profile contact ratio	ε_α	1.50		–
Axial contact ratio	ε_β	1.50		–

Table 3 Micro geometries on wheel

Type	Magnitude in µm	Length in % of transverse path of contact
Crowning	3	–
Tip relief	17	33
Root relief	14	33

standard micro geometries are to be applied to the wheel. The micro geometries are shown in Table 3.

Different approaches can be used for gear noise behaviour estimation depending on the provided level of complexity. However, the quasi-static approaches are often used thanks to the computing time benefit, which becomes very advantageous for example in the optimization calculation. By means of these approaches, the tooth force excitation is one of the obtainable quantities. However, as we have mentioned in the previous section, the tooth force excitation does not contain any dynamic influences and thus yields rather constant spectra throughout a whole speed range. Therefore, an analytical transfer function of gear force according to Geiser shall be used as an approximation of dynamic influences, which is defined as [5]:

$$V = \frac{\eta^2}{\sqrt{(1-\eta^2)^2 + 4 \cdot (D \cdot \varepsilon_\alpha \cdot \eta)^2}} \tag{3}$$

V [–] Force transfer function according to Geiser
η [–] Speed ratio
D [–] Lehr's damping factor (for gear mesh, see also [6])
ε_α [–] Profile contact ratio

Under this consideration, the application force levels of both gear mesh variants can be calculated according to method B.

In this example, the noise behaviour of the gear mesh at load stage T_I = 1000 Nm shall be estimated within following boundaries:

- Speed range: 1000–4500 rpm
- Frequency range up to 5000 Hz
- Mesh order range: 0.9–4.2

By means of the gear noise behaviour estimation with application force level according to method B, of which the results are shown in Fig. 3, the reduction of gear noise behaviour by the designed micro geometries around 6 dB of $L_{A,F,B}$ is possible.

Meanwhile, the torsional acceleration measurement was carried out. This kind of measurement is available in prac-

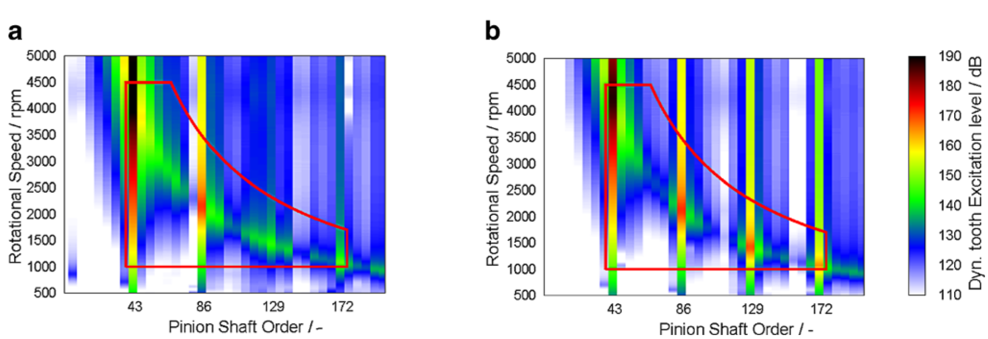

Fig. 3 Application force levels $L_{A,F,B}$ from tooth force excitation with Geiser's transfer function of unmodified and flank-modified gear meshes within the predefined area

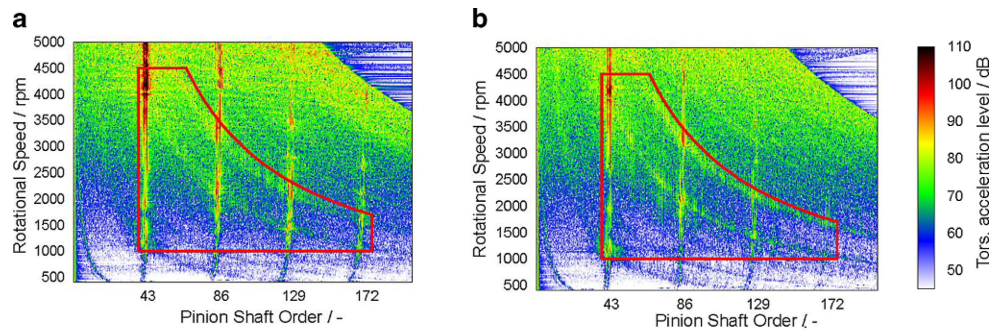

Fig. 4 Application force levels $L_{A,F,A}$ from dynamic measurement of unmodified and flank-modified gear meshes within the predefined area

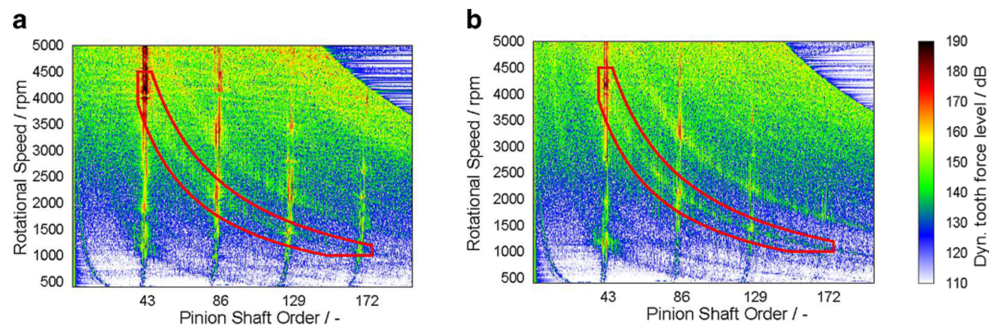

Fig. 5 Application force levels $L_{A,F,A}$ from dynamic measurement of unmodified and flank-modified gear meshes within the predefined, frequency-specific area

tice in the latter development stage, when the gear prototypes are produced. As mentioned earlier, the gear noise behaviour from such measurement can be evaluated according to method A.

Results of the evaluation are shown in Fig. 4 and confirm the similar gear noise reduction that was estimated in the previous calculation according to method B (Fig. 3) around 8 dB of $L_{A,F,A}$. A qualitative comparison between application force levels according to different methods is therefore possible.

Another notable feature of the application force level is the evaluation possibility of gear noise behaviour within a specific frequency range. Fig. 5 shows an investigated area, in which the frequency range is defined between 2500–3500 Hz. The noise behaviour evaluation under this boundary is also possible with the application force level.

4 Conclusion

In this article, the application force level and its usage for evaluation of gear noise behaviour are introduced. The main purpose of this new characteristic value is to enable a standardized gear noise evaluation procedure throughout the whole development stages, during which different data basis are available. The formulation of this characteristic value is derived in this paper. Besides handling ability for various data types, the application force level evaluates gear noise behaviour based on information from an order diagram under consideration of an investigated area. This area is specified by three boundaries, which are speed, frequency and order ranges. The investigated area is customizable. Therefore, the application force level can also be used as a tailor-made tool in some specific applications, in which for example noise behaviour within a narrow band of frequency range is to be investigated. With the application force level, the flank modification can accurately be designed not only for a specific load range, which is often realized by the loaded transmission error or force excitation approach, but also for an optimized operational speed range under consideration of the dynamic influential parameters.

Because different data bases contain different degrees of dynamic information, the application force level can be classified into three subtypes according to the dynamic information available at that stage. However, it is shown in the latter part, that the comparison of application force level between different data basis is qualitatively possible.

Although the application force level was intentionally developed for the gear design applications, the usage of this characteristic value is indeed applicable for other fields of NVH evaluation, for example in the prediction of dynamic behaviour in [1], where the dynamic torque of an electric drive was predicted and measured.

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