ASPECTS ABOUT THE IDENTIFIES THE MAJOR SOURCES OF GEAR WHINE NOISE

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Abstract: *The excitation occurs in the gear mesh and the path includes the transmission of forces and motions through the shafts and bearings to the gear housing, which in turn radiates the noise. This paper focuses on the source aspect of the problem and identifies the major sources of gear whine noise and develops a metric that incorporates these sources into a single equation.*

Keywords: *gear noises, tooth impacts, mesh stiffness, transmission error, friction, air and lubricant entrapment, helical gear.*

1. INTRODUCTION

When gears rotate, forces are transmitted across the interfacing region between the meshing teeth. As the transmitted forces move from one tooth pair to the next, the time varying changes in these forces due to gear contact mechanics, imperfect teeth, tooth deflections, and tooth sliding result in time varying reaction forces that transmit through the bearings and excite the housing.

Identifying these bearing forces is often treated as a dynamics problem; however gear designers desire static excitation metrics so that an elaborate dynamic analysis need not be done for each prospective gear design.

Therefore, the excitations that are developed in this paper will be developed using a static approach.

2. THE SOURCES OF GEAR WHINE NOISE

The following phenomena have been identified as possible gear noise excitation factors [1]:

- mesh stiffness variation;
- tooth impacts;
- force axial shuttling;
- friction;
- transmission error;
- air and lubricant entrapment.

a. Mesh stiffness variations

In heavily loaded gears, as the number of tooth pairs in contact changes, abrupt changes in the gear pair stiffness occur (the mesh of spur gears with two tooth pairs in contact is roughly twice as stiff as when one tooth pair is in contact).

This variation, which is greatly reduced by using helical gearing, results in accelerations and decelerations of the gear pair that result in dynamic mesh forces.

However, the resulting motion error due to these stiffness changes may be treated as a transmission error excitation. Again, tip and root relief provide means of reducing the effect of stiffness variations.

b. Tooth impacts

Impacts have the longest history as an explanation of gear noise. The main form of this argument is that as the teeth enter contact there will be kinematics mismatches that result tooth tip impact forces that are transmitted through the bearings.

These impacts, which happen when teeth enter and leave contact, occur due to tooth deflections and/or tooth spacing errors. However, they are of such short duration that they possess little energy at gear mesh frequency and its lower harmonics.

Also, practitioners have learned that by providing adequate removal of material from the tooth tips (tip relief), these entering and leaving impacts may be totally avoided.

c. Force axial shuttling

This factor, which was investigated by Borner and Houser [2], is due to the slight axial shifting of the centroid of the mesh force and in the case of right angle gears the changing in the direction of force application, that occurs as the lines of contact shift when the gears rotate through mesh. Since this quantity is the result of elementary static analysis, it is easily justified as a real gear noise excitation. Houser and Harianto [3] have created a means of expressing this excitation in terms of a time varying bearing force that is created due to the shifting of the transmitted mesh force centroid.

d. Friction

Virtually all gear types have relative sliding between meshing tooth pairs. As the contact lines proceed through mesh, the force due to friction varies at the mesh frequency and is another logical gear noise excitation.

The main issues with regard to this force are the nature of the friction, whether it is Coulomb or viscous, and if Coulomb, the value of friction coefficient that should be used. The coefficient of friction is most certainly a function of lubricant viscosity [4] but is also a function of other factors such as surface roughness, sliding velocity, etc. Values of friction coefficient between 0.03 and 0.07 have been typically used for lubricated gear contacts [5,6].

The viscous force is a function of the relative sliding velocity and will be greatest near the tooth tips and will decrease as contact approaches the pitch point, where true rolling motion occurs.

e. Transmission error

The simplest definition, transmission error is the deviation from perfect motion transfer of a rotating gear pair.

A gear pair that has constant input rotation speed and constant output rotation speed has zero motion dependent transmission error and is said to have perfect or conjugate motion transfer.

The time variation of transmission error at mesh frequency and its harmonics is predominantly due to two factors, the first being a deviation in the tooth shapes from conjugate shapes (involute shape for parallel axis gears) and mesh deflections due to transmitted load (effect of mesh stiffness variation).

Minimizing transmission error has long been seen as the most important factor in minimizing gear noise. It should be noted that one may cancel the effects of mesh stiffness variation by intentionally providing tooth shapes with deviations from perfectly conjugate shapes.

In doing so, it is important to minimize transmission error in the torque range at which the gear noise is a problem. Since stiffness variation changes with torque loading, the optimum profile modification at one is likely not to be an optimum at another load.

f. Air and lubricant entrapment

Both of these excitation possibilities are related to the pumping action of the gear teeth in mesh, namely that air and lubricant that are trapped in the clearance region of the meshing teeth need some avenue of escape.

In high speed gearing, the forces generated in moving the fluids out of the mesh may be substantial and hence can be excitations of mesh frequency noise.

When considering the source-path-receiver approach to gear noise analysis, the major path of noise in gears enclosed in housings has always been felt to be a structural force path in which the forces generated at the mesh are transmitted through the bearings to the housing.

In this case we shall follow that assumption and create an approach for obtaining bearing forces that are the result of mesh forces created by each of the excitations.

This equation takes the form:

$$F = F_i \cdot \alpha_i + F_{as} \cdot \alpha_{as} + F_f \cdot \alpha_f + F_{et} \cdot \alpha_{et} + F_{al} \cdot \alpha_{al} \quad (1)$$

where:

- F is the bearing force;

- $\alpha_i, \alpha_{sh}, \alpha_f, \alpha_{et}, \alpha_{al}$ - coefficients for each term that depends on the gear and bearing configuration and the location of the bearings

- F_i - the impact force that contains two factors, one related to entering impacts (acts slightly off line of action) and the second being a viscous force caused by sliding impacts (acts at a right angle to the plane of action);

- F_{as} - the shuttling force that is calculated from static analysis of the mesh force (acts in the direction of the plane of action);

- F_f - the friction force computed by static analysis at the mesh (acts at a right angle to the plane of action);

- F_{et} - the effective transmission error force that also accounts for mesh stiffness variations (acts in direction of the plane of action and also in axial direction);

- F_{al} - the force due to the entrapment of air and lubricants (direction not clearly defined).

Because the goal is to develop an excitation measure that may be applied by gear designer, the above equation will be posed as a static excitation in this paper.

An alternative would be to perform a dynamic analysis of the gears and their supporting shafting in order to calculate a dynamic bearing force, in which case one would need to use the mesh force as the excitation.

The simplest reasonable model for performing gear dynamics modeling is shown in figure 1. The transmitted forces lie on the line of action, the line that is tangent to the respective base circles of the two meshing gears.

The transmission error is modeled as a displacement input where,

$$\Delta E = X_a - X_b \tag{2}$$

The impact force that is due to corner contact is approximated to be along the line of action, and a second impact force due to the viscous sliding impact is taken at right angles to the line of action, as is the friction force. It neglects the entrapment force, since it is felt that it becomes important only for extremely high-speed gearing.

It essentially have a pretty good feel for the evaluation of the shuttling force and the friction force (other than the approximation of the coefficient of friction), but in terms of obtaining an equivalent transmission error force, one has to look very closely at the dynamics of the situation.

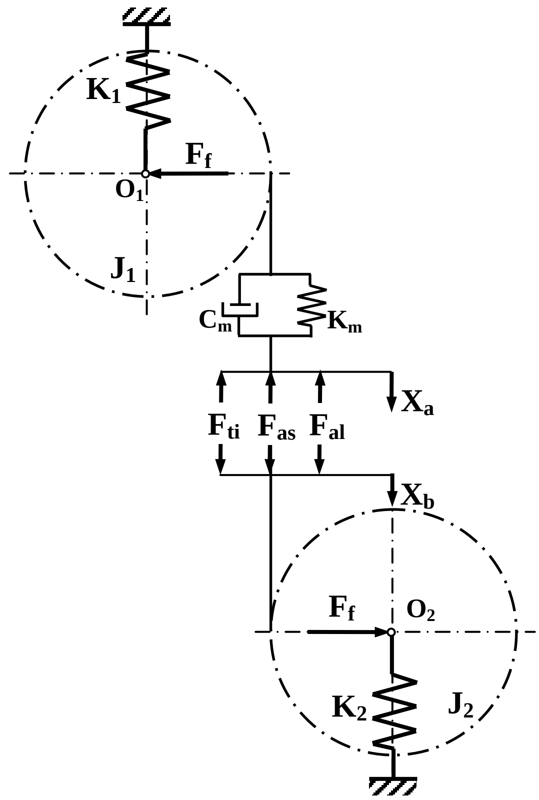

Fig. 1 The gear dynamics model

In order to put a rational value into the static excitation model, we are going to use the relationship:

$$F_{et} = k_{ams} \cdot \Delta E \tag{3}$$

where: k_{ams} is the average mesh stiffness coefficient.

This approximation would provide a valid mesh force if we are operating above the lowest torsional natural frequency of the simple system of fig.1.

Another way of looking at this force is that it is the bearing force that would result if the inertias were very large so that they remain vibrationally stationary when running at speed.

For helical gears, this force F_{eth} has two components, one along the line of action and a second axial force that is evaluated by multiplying the line of action force by the tangent of the helix angle, as shown below:

$$F_{eth} = k_{ams} \cdot \Delta E \cdot tg\gamma \tag{4}$$

where: γ is the helical angle.

The key to performing an appropriate analysis of the mesh excitation is the evaluation of the distribution of forces along the contact lines. A requirement of the analysis is the ability to account for either the design tooth topographies or the actual measured topographies of the tooth surfaces. This analysis requires a good contact algorithm as well as an evaluation of tooth compliances. The analysis is certainly feasible using finite element analysis.

With helical gears, an axial force must also be included in the force analysis. This axial mesh force not only creates an axial bearing force, but also affects the bearing forces along the line or plane of action. This latter component could be substantial when we are analyzing narrow face width gears that are common in manual transmissions and in transfer sets. It should also be pointed out that the á coefficients for each force component allow one to compensate for the relative housing force “coupling efficiency” of each force component. For instance, it is likely that axial forces at bearings are more efficient in creating radiated housing noise than are radial forces.

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