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Dynamic Analysis of Hydrodynamic Gear Pump Performance Using Designof Experimentsand Operational Parameters

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Abstract: *A few researches have been carried out in dynamic analysis of gear pump. This work concerns external gear pumps for automotive applications, which operate at high speed, low pressure and presents a sensitivity analysis about the influence of design and operational parameters on the pump dynamic behavior. In previous work, a mathematical modeling was developed for eccentricity of shaft and pump, with the aim of including all the important effects, as well as to get a rather simple model. There are three main sources of noise, vibration and friction considered: Angle of pressure variation and gear meshing. The model has been taken in to account for comparison with experimental vibration data, in a wide range of operational parameters conditions for several profile errors. This paper is concentrated on dynamic analysis of the influence of the main design and operational parameters of the pump dynamic behavior. In particular, the effect of operational pressure, speed and friction, the influence of the clearance in the hydrodynamic bearing and between tooth tip and pump case, and the effect of the operational parameters like stiffness, damping, coefficient of friction, film thickness, speed, friction, load, and torque of the gear will be thoroughly discussed in the paper. Finally, the mathematical model and design could be a very useful and powerful tool in order to evaluate the dynamic improvements for noise, vibration and friction reduction.*

Keywords: *Gear pump-modeling, control volumes, pressure distribution, radial force on shaft*

I. Introduction

The gear pumps are among the oldest and most commonly used pumps in the industry. It has become the main choice for fuel system designers due to long life, minimum maintenance, and high reliability, capability to operate with low lubricating fuel, low heat input to fuel, and low weight. Their main applications can be found as lubrication pumps in machine tools, in fluid power transfer units or as oil pumps in engines. These features make them an interesting component in aerospace, industry, agricultural and automotive applications. On the other side, their main drawbacks are related with high noise, vibration and friction levels and unavoidable output pressure ripple.

This context increases the interest on modeling the dynamic behavior of this mechanical system as a way to improve the initial design reducing testing efforts. Therefore, a good dynamic model could be a useful and powerful tool for the identification of noise and vibration sources and design improvement[12]. For these reasons, the authors have applied eddy current cap sensor with lumped-parameter on hydro dynamic model in order to study the dynamic behavior of an external gear pumps for automotive applications. Fluid pressure distribution around the gears, which is time-varying, is instantaneously computed and included as a resultant external force and torque acting on each gear [1], [2], [3]. Gear meshing phenomena have received particular attention [1], [4]: in particular the time-varying meshing stiffness, damping effect and the tooth profile errors like module, addendum, dedundum with respective pressure angle and arc of path of contact, the effects of the backlash between meshing teeth, the lubricant squeeze and the possibility of tooth contact on both contact lines have been included in the model [1][4] [11]. One of the particular features of gear pump design is the use of hydrodynamic journal bearings for gear shaft support. The non-linear dynamics of this kind of bearings has been modelled using the theory of Childs called “finite impedance formulation and mathematical modeling”[5]. So, the model is highly non-linear. The dynamic equations of motion for each degree of freedom are obtained in [1], [4] and are numerically integrated in Simulink environment [6] [12] . With the aim of reducing the integration time, the average positions of the shaft axes inside the journal bearings are estimated before the integration of the dynamic model. This estimation is carried out setting the periodically variable pressure and meshing forces of the model to a constant value equal to their mean values. Then the ‘stationary’ axis positions (also called orbit centroid) are computed as the solution of a non-linear system of algebraic equations obtained from the force balance of each gear. A non-conventional validation procedure [1], [7] [11] has been developed in order to assess the dynamic model. The validation results can be considered rather satisfactory. Once the model is validated and its effectiveness is satisfactorily assessed, it can be used both in design optimization and diagnostics. In particular, since the model is able to predict the effects of variations of some parameters, a sensitivity analysis has been performed in this work: it will be shown hereafter the effect that a modification in the clearance into the journal bearing and between case and tooth tip has on the dynamic forces and on gear accelerations. Moreover, the effect of operational pressure and speed, and the effect of design parameter modifications (such as dimension of the relief grooves) will be thoroughly discussed.

The mechanical system under study is an external gear pump for vehicle steering. The most usual configuration has two twin gears (see Figure 1a), which are assembled by a couple of lateral floating bushes that act as seals for the lateral ends. Gear and floating bushes are jointly packed inside a case that encloses both the components and defines the isolated spaces that carry the fluid from the low to the high pressure chamber. This lateral floating bushes act as supports for gear shafts by means of two hydrodynamic bearings, which are hydraulically balanced in order to avoid misalignments between gear shaft and journal bearing. Power is applied to the shaft of one gear (gear 1), which transmits it to the driven gear (gear 2) through their meshing. Gear 1 is connected by an Oldham coupling with an electric drive. In the gear meshing area, when two tooth pairs come in contact, a trapped volume could arise and could undergo a sudden volume reduction and consequently a violent change in its pressure. To avoid this, the trapped volume is put in communication with the high or low pressure chambers. That is the role of the relief grooves milled in the internal face of the lateral bushes whose shape and dimension are therefore very important in the resulting dynamic behaviour.

In more detail, we can refer as seal line the segment $\overline{CD}$ of the line of action (see Figure 1b),limited by the intersection with the relief groove edges. Three different situations can occur: the seal line can be equal, larger or shorter than the base pitch. If the seal line is larger than the base pitch, w~~hen~~ the second tooth pair in contact enters into the seal line (CD in Figure 1b), the other meshing contact is still into the seal line and therefore during the gear rotation, the trapped volume will have a reduction and a consequent rise in pressure that can cause high vibration levels[12].

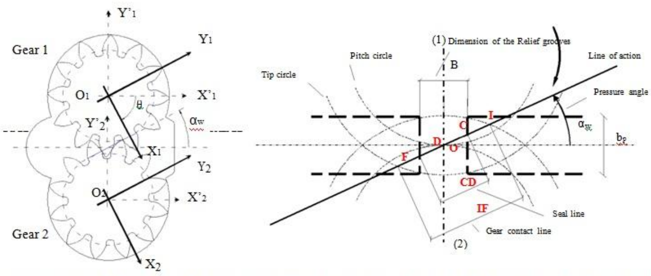


Figure 1. (a) Schematic of the gear pump and reference frames and (b) gear contact line and dimension of the relief grooves.

In the pump understudy, namely GENB, the dimension of the seal line is shorter than the base pitch and therefore when the second tooth pair enters into the seal line, the other tooth pair in contact is already beyond the D point (Figure 1b). In such a situation, there is contemporaneous communication between the inlet and outlet volume and consequently the volumetric efficiency of the pump decreases but the pressure rise in the trapped volume is reduced. In the GENB pump, the relief groove dimension is B=2.9 [mm] and the seal line length is :

$$\overline{CD} = \frac{B}{\cos(\alpha_w)} = \frac{2.9}{\cos(27.727)} = 3.276\ [mm] \qquad (1)$$

where α_w is the pressure angle in working conditions. On the other hand the base pitch is :

$$P_b = \frac{2\cdot\pi\cdot r_b}{z} = \frac{2\cdot\pi\cdot 6.484}{12} = 3.395\ [mm] \qquad (2)$$

where r_b is the base radius and z the teeth number. Therefore the difference between $\overline{CD}$ and P_b is 0.119 [mm] and in percentage of the base pitch becomes :

$$\%diff = \frac{(B/\cos(\alpha_w)) - P_b}{P_b}\cdot 100 = -3.5\% \qquad (3)$$

So, the inlet and outlet chamber are in contemporaneous communication for 3.5/100 of the base pitch.

Two difference reference frames for each gear are used (see Figure 1a), both having their origins coincident with the centres of the gears; in reference frames O_1 X_1Y_1 and $O_2X_2Y_2$, the X-axis is perpendicular to one of the lines of action and the Y-axis is parallel. On the other hand, in the reference frames O_1 $X'_1Y'_1$ and $O_2X'_2Y'_2$, the Y'- axis is along the line connecting the centres of the gears and the X'-axis is orthogonal. In addition, Gear 1 is the driving gear and Gear 2 is the driven one.

II. Pump Description

Figure 2 shows a cross-sectional view taken through the gears of a typical gear pump

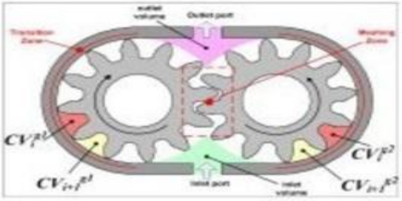


Figure2 Control Volumes Defined in the Fluid DynamicModel

Note: like most actual gear pumps, these pump hastwo identical gears to displace fluid. The superscripts g1 and g2 denote the driving and driven gears re- spectively. The tooth number of gear 1 is the same as that of gear 2. The Control Volumes (CVs) within the tooth gaps are variables dependent on the gear angle. The subscripts i is the index of these Control Volumes. The Outlet and Inlet volumes are two fixed volumes at the outlet and inlet ports respectively.

To produce a flow within a gear pump, fluid is carried by the CVs from the intake side of the pump to the discharge side of the pump through the transi-tion zone. As the gears rotate, these CVs increase their pressure to when reach the high-pressure chamber. As the gear teeth mesh in the meshing zone, flu-id is squeezed out of each tooth gap by the mating tooth. When two tooth pairs contact, a trapped volume is generated. This may cause positive pressure peaks and the onset of cavitations. (To avoid this, the Trapped volume must be connected to the high or low pressure chambers, which is the role of the relief grooves in the lateral bushes.) On the intake side, the gear teeth are coming out of the mesh. The volumes of CVs increase so that fluid is inhaled into the tooth gaps. This process repeats itself for each revolution of the pump and therefore displaces fluid at a rate proportional to the pump speed.

III. Model Description

3.1 Overview

In this section, we first give an overview of the pump model. Some components come from a free library HyLibLight [9] [11]based on Modelica.

The aim of this paper is to construct a hybrid model of the external gear pumps. It considers the leakages, compressibility of the oil, flow ripple, pressure distribution, etc. Specifically, the pump is modeled under the following assumptions: the simulation. Proportioned.

1) The gears and the housing case are rigid; only the oil is compressible
.2) The position of the shaft is known before the simulation and is fixed during
3) The pressure at every single isolated region and fixed volume is well-
4) The temperatures in all CVs are the same and constant.
5) The tooth numbers of the two gears are the same.

In Figure2, a pump is divided into 2n+2 control volumes, where n is the tooth number of a gear. Figure 3 shows he flows between those control volumes. This method is similar to but not the same as that in [10]. There is no variation in the number of the control volumes which causes variation in the number of differential equations.

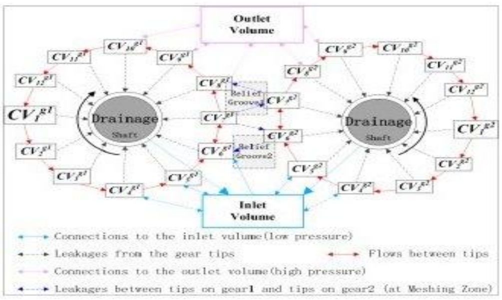


Figure 3: Shows the flows between those control volumes.

IV. Influence of Operational Parameters: Pressure and Rotational Speed, Load, Friction & Torque.

Figure 4 and Figure 5a illustrate the influence of the output pressure on the stationary centre locations and on pressure distribution around the driving gear. The simulation results are obtained at rotational speed of 2000 rpm and increasing the output pressure from 20 to 90 bar; the circle in Figure 4 represents the maximum displacement of gear axes, allowed by the nominal clearance (h_{rn}) of 0.0245 mm between gears and pump case (see also Figure 1a).

The results depicted in Figure 4 show that as the output pressure increases, the eccentricity modulus of both gears increases, in fact the markers in Figure 4 move closer to the maximum displacement circle. This behaviour can be interpreted as a consequence of the bearing reaction increase due to the enhancing of the gear pump output pressure. In fact, as the pressure forces increase, the new axis centre locations move reducing the actual clearance in the journal bearings (i.e. increasing the eccentricity modulus) and therefore the bearing reaction increases; this way, the new bearing reaction can balance the increased pressure forces.

On the other hand, the pressure distribution in the tooth spaces, normalized to the output pressure value (Figure 5a), exhibits a sudden variation in the first tooth spaces; as depicted in Figure 2, due to the eccentricity direction, the minimum meatus height is located in the first tooth spaces. Moreover, the higher the output pressure, the more anticipated in the first isolated spaces this sudden variation is. This behaviour can be explained referring to the volumetric flow rate. In fact, it is worth noting that the volumetric flow rate directly depends on the pressure drop Δp_i Between two consecutive vanes and on the meatus height h_i ($Q_i \propto h_i^3 \cdot \Delta p_i$, subscript i refers to the generic meatus) while the meatus height strictly depends on gear eccentricity.

Considering the volumetric flow rate as approximately constant in all the meatus, the global pressure drop from outlet to inlet volumes is distributed among the meata as inversely proportional to h_i^3. Therefore, increasing the output pressure value, the actual clearance between tooth tip and pump case globally decreases, but this alters the ratios between two consecutives meatus heights h^3 in such a way that the sudden variation of the Normalized pressure distribution is more anticipated in the first isolated spaces.

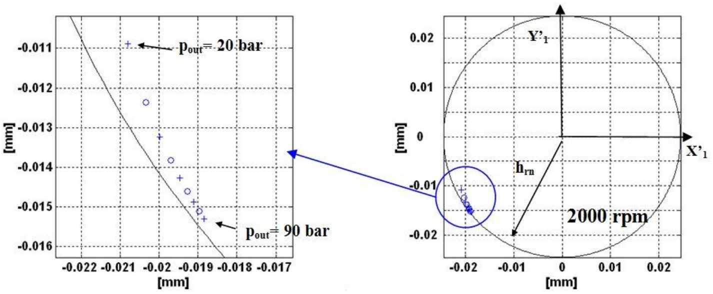


Figure 4. Stationary centre position for the driving gear with different output pressure values; the representation is in the reference frame of Figure 1; the circle represents the maximum displacement of gear axes, allowed by the nominal clearance h_{rn} = 0.0245 mm between gears and pump case.

Furthermore, Figure 5b shows the influence of the pressure variation on the pressure force concerning the driving gear in Y-direction, for one meshing period T, starting at the instant when the second tooth pair comes in contact. It can be noted that the pressure force is noticeably affected by an increase of output pressure due to their direct dependence on Sure distribution; the same behavior occurs for the other directions and for the driven gear, not shown hereafter. On the other hand, for the gear accelerations, depicted in Figure 6a, the output pressure dependence is not as pronounced as for the pressure forces because the gear accelerations are the balance between all the dynamic forces (meshing forces, bearing reactions, pressure forces); anyway, considering the peaks in the accelerations at about 25% of the meshing period, the higher peaks are relative to the accelerations obtained at higher output pressure.

(a) (b)

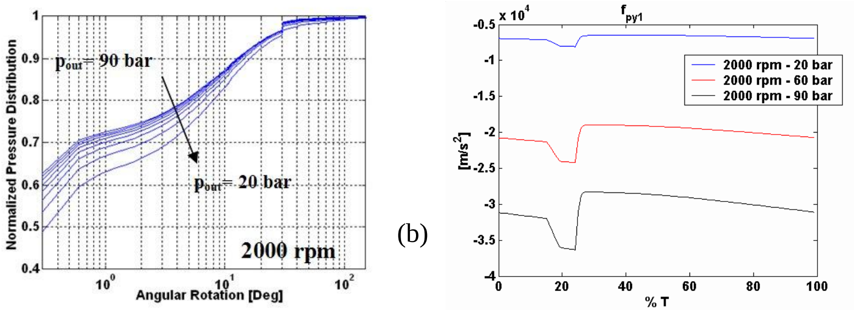


Figure 5. Normalized pressure distribution (a) and pressure force (divided by the gear mass) in Y_1-direction (b) in Gear 1 for different output pressure values.

(a) (b)

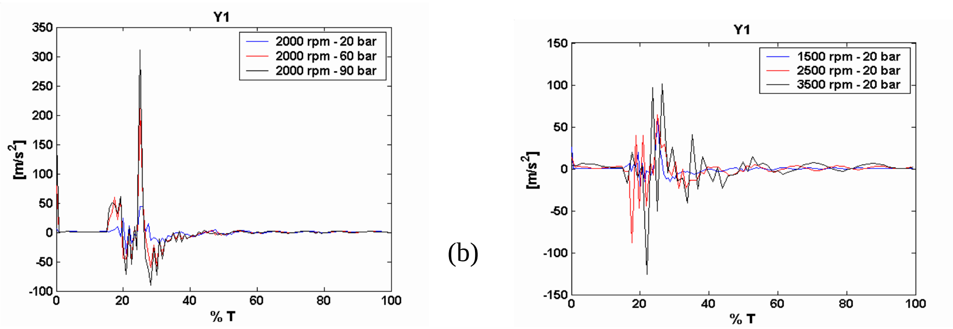


Figure 6. Acceleration for the driving gear in Y_1-direction for different (a) output pressure

The influence of the rotational speed on the stationary centre position is similar to the influence of the pressure already described above; in particular as the shaft rotational speed decreases from 3500 to 1500 rpm, the eccentricity modulus value increases from 0.02284 to 0.02372 mm for gear 1 and from 0.02308 to 0.02384 mm for gear 2. In fact, as the rotational speed decreases, the bearing reaction tends to decrease too, due to the direct relation with the operational speed [1][2] and the new axis centre location will change, decreasing the actual clearance in the journal bearing (i.e. larger eccentricity modulus); this way, a new equilibrium between the dynamic forces is established. As a consequence, when the rotational speed decreases, the pressure distribution in the tooth spaces is altered in a similar way as occurs when the output increases (i.e. its sudden variation is more anticipated in the first isolated spaces), for analogous reasons. In addition, Figure 4b shows the influence of the operational speed variation on the acceleration in Y_1-direction for gear 1: it can be noted that an increase of the operational speed determines an increase of the gear acceleration.

V. Effect Of Design Parameters

5.1 Effect of clearance modifications

In the pump under testing, the nominal clearance h_{rn} between pump case and tooth tip has the same value as the nominal clearance C_r in the journal bearing, equal to 0.0245 mm. These dimensions, as shown below, have an important influence on pressure distribution, on gear accelerations as well as on the stationary centre position.

In this section the influence of a modification of the nominal clearances C_r and h_{rn} will be shown. Reference [1] [11]includes all the details about the clearances involved in the pump working and several clarifying figures. The simulation results presented in this section are obtained at the operational condition of 3350 rpm and 20 bar. Table 1 collecting the normalized eccentricity modulus and the eccentricity direction angle in the stationary centre position, shows the influence of a clearance alteration on the stationary centre position of gear 1; it can be observed that as the nominal clearance increases, the stationary centre position becomes closer to the maximum allowed displacement, in fact the eccentricity modulus increases too. Moreover, Table 1 Collects the values of the actual minimum clearance $C_{r,\chi}$ in the journal bearings tha obviously depends not only on the nominal clearances C_r but also on the eccentricity ($C_{r,\chi} = (1-\chi)\cdot C_r$). It can be noted that as the nominal clearance C_r increases, the bearing reaction tends to decrease; so, clearance decreases. due to the dynamic force equilibrium, the actual minimum On the other hand, about the influence of these clearances on pressure distribution around the driving and driven gear, it can be noted that as the nominal clearance value increases, the pressure in the first tooth spaces reaches a higher pressure value earlier (as it occurs if the output pressure increases). This pressure distribution behavior can be explained as a consequence of the increase of the eccentricity modulus (Table 1) which alters the ratios between two consecutives meatus heights (h_i) as explained in the previoussection.

The pressure forces depend on clearance alteration too, since they are obtained using the pressure distribution. Figure 7a shows the influence of the clearance value on pressure force evaluated over one meshing period, starting at the instant when the second tooth paircomes in contact. It can be noted that, as the clearance value increases, thepressure force amplitude also increases due to the different trends on the pressure distribution. Finally, Figure 7b shows the clearance influence on the acceleration acting on gear 1 at operational condition of 3350 rpm and important differences20 bar expressed as a function of meshing period. No in accelerations between the three clearance conditions can be observed; in fact, even if pressure forces increase as the clearance value increases, the bearing reactions change in order to balance the pressure forces at best. For this reason, theaccelerations, that are obviously the consequence of the balance between the pressureforces, meshing forces and the bearing reaction forces, do not change in an important way.

Table 1. Clearance influence on stationary centre position (in modulus and direction angle) at 3350 rpm and 20 bar for gear 1 and influence on the actual minimum clearance.

Nominal clearance [mm]	0.019	0.0245	0.0275
Eccentricity modulus χ (normalized)	0.900	0.935	0.946
Actual minimum clearance $C_{r,\chi}$ [mm]	0,0019	0,0016	0,0015
Eccentricity direction angle [deg] in the reference frame $O_1X'_1Y'_1$	194.6	201.8	204.7

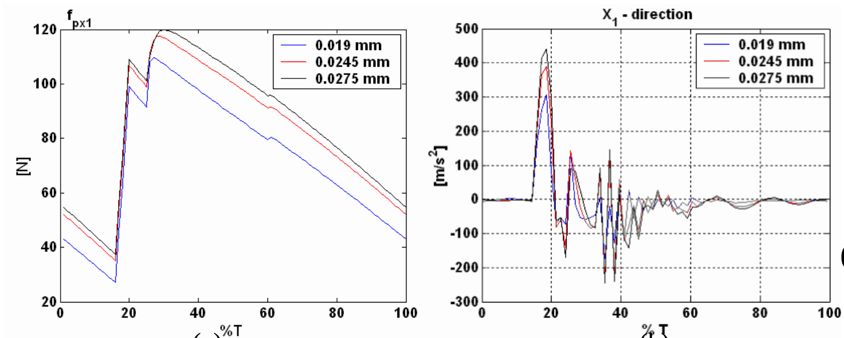


Figure 7. Clearance influence on (a) pressure force and (b) acceleration in X_1-direction for gear 1 over one meshing period T; operational conditions of 3350 rpm and 20 bar.

5.2 Effect of the relief groove dimension

Figure 8a shows the pressure force in X_1-direction for different kinds of relief grooves (length *B* of 2.4, 2.7 and 2.9 mm, see also Figure 1b); such a dimension has great influence on the performance of the gear pump, in fact it can be noted that with reduced length *B*, the discontinuities on the pressure forces is smoother due to the increase of the contemporaneous communication of the inlet and outlet chambers. In fact, remembering equation (3) the contemporaneous communication between the inlet and outlet volume is 3.5% of the meshing period with dimension *B* of 2.9 mm, while with *B*=2.7 mm it is 10% and finally with *B*=2.4 mm it is 20%.

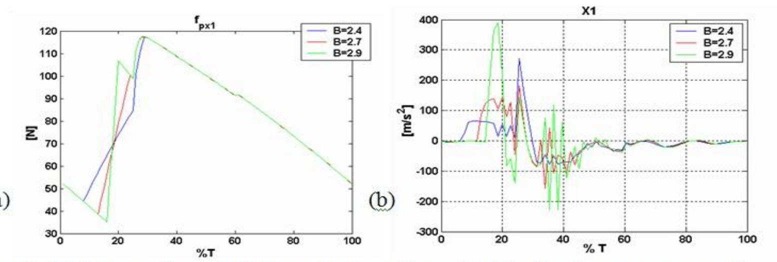
(a)


Figure 8. (a) Pressure force and (b) acceleration of gear 1 in X_1-direction over one meshing period with different dimensions of the relief grooves (length *B* of 2.4, 2.7 and 2.9 mm).

The results in terms of gear accelerations (Figure 8b) show that the relief grooves with length 2.9 mm determine larger oscillations than with lower relief groove lengths due to the decrease of the contemporaneous communication with the inlet-outlet volumes; moreover the smaller length *B*, the lower the pump efficiency. Therefore, the designer has to take into account that in order to reduce the gear vibrations, the relief groove length has to be reduced, but in order to increase the pump volumetric efficiency, the relief groove length has to be increased.

VI. Conclusions

In this work an extensive sensitivity analysis by using a design of experiments andope rational parameters of a gear pump has been performed. In particular, the model has been developed in the design phase in order to evaluate performance of analysis of hydraulic ge ar pump modifications (stiffness, damping effect, and friction, load &pressure distrib ution of the gear pump) and in order to evaluate the influence of operational parameters such as load, speed, noise and vibration.

The graphical representation re sults have concentrated that the various variation in the operational range of the rotational speed gives a lower effect on the gear accelerations than the output pressure variation.

Therefore, the designer has to take into account that in order to reduce gear noise, vibrations and friction the path of arc of contact has to be increased, but in order to increase the pump performance; the clearance has to be maintained. Finally, this work will be useful in order to evaluate design directions and in order to foresee the influence of working conditions and design modifications on friction, vibration and noise generation.

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