The Parameters influencing the GWN

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Abstract – This study describes, from the experimental point of view, the parameters influencing the phenomenon called "gear whine noise", which involves the gear-box. A particular attention is devoted to the main causes of such an acoustic emission (*e.g.*, micro-geometry of the tooth and tooth surface texture) and to the most appropriate modifications to be carried out already during the design stage, in order to reduce it. In particular, a comparison was carried out in the fourth gear, from vibro-acoustic point of view, on gear-boxes, showing micro-geometric differences, made by two different makers.

Key-Words: - gear whine noise, PPTE, swell, transmission error, vibration.

1 Introduction

The constant increase of quality standards on the motor vehicles, due to the development of advanced technologies as well as to the process, design, production, and often imposed by specific rules, brings the car manufacturers to compete mainly on the quality standard of vehicles.

Some studies converge on the common thesis that the customer considers to be of "good quality" car which has a low level of noise inside the vehicle during the running. This led designers to consider closely the development of products oriented to the reduction of noise. Major attention has been devoted primarily to the reduction of the noise of the engine, in the past considered as the main and sometimes the unique source of considerable noise. Reduced the contribution of the engine, became predominant the noise from transmission and especially from gear-box.

Therefore, the reduction of noise of motor vehicles today is one of the most important quality factors. In particular, major attention is given to that part of the acoustic noise produced by the gears. Inside the vehicle this noise can mainly be referred to the mechanical transmission, to the transfer of torque or is produced by the differential gear.

The gear noise occupies an important role in the automotive field of growing interest; consequently it is considered to be one of the major engineering problems nowadays [1]. Designers and engineers try often to reduce the noise transmission paths by modifying the external structure of the gearbox and the inside soundproofing, although the optimal solution is to reduce, or even to eliminate the sources of noise. In order to design less noisy mechanical systems it is, therefore, necessary to know the origins and the characteristics of noise, by starting from the gear-box [3,15].

The main purpose of the gear is to transmit power as uniform and regular as possible between a pair of meshing wheels. In theory, such conditions can be satisfied by means of an ideal mechanical system as should be the gear-box: a) perfect tooth geometry, b) infinite stiffness and c) softness of meshing. In practice there are many factors which can generate deviations from such an ideal situation: in particular, the shape of the flanks of tooth as accurately realized might not be ideal from a micro-geometrical point of view.

Besides, due to the limited hardness of the material, we can not think that the tooth and all transmission gears involved in the transport of torque are free from elastic tension, and therefore from deformation.

As consequence, the gear-box generates a noise composed of several tones of discrete frequencies, due to the number of teeth involved per revolution (order of meshing), and its harmonics. These frequency components, affecting the wheels, because the load is transmitted from a pair of teeth to another one, generate a noise known as Gear Whine (or GWN for short). It manifests as a "whistle" or a "siren" and its influence on the overall vehicle noise is significant and must be strictly limited [4,5,13,16].

The parameters which mainly influence this phenomenon are: the transmission error, the friction forces and the peak to peak transmission error.

2 Main Parameters influencing the GWN

2.1 – Transmission Error

The Transmission Error (or TE for short) is defined as the difference between the actual position occupied by the driven gear and the ideal position that it should occupy if the gear wheels (*i.e.*, driven and driving) were perfectly conjugated [6,18]. Really, however, the tooth can not be produced geometrically perfect (ideal), as well as a tooth can not be made too rigid in order to avoid the bending under load. When a heavy load is applied it tends to break down: the tooth must be composed of a very hard surface, able to withstand wear and of a core rather than elastic, able to well absorb the dynamic load variations.

For the aforesaid two reasons, the teeth tend to move from their ideal position during the meshing, and, therefore, a load step acting on the gear is the source of shock and vibration.

In addition, the actual position of teeth generates a dynamic that leads, during the meshing, to a strong sliding and to a specific pressure higher than those designed.

In Fig. 1 it is shown the bending of teeth A and B due to the torque applied by the engine to the gear 1 (driving), transmitted to the gear 2 (driven). Due to the bending of teeth A and B, the driven wheel has a delay which generates an early contact between the teeth C and D, creating an impact, called *access* or *feeding impact* and generating specific high pressure on the areas where these pressures should not be high. Similarly the tooth next to the one immediately receives the full force and this instability may lead to an impact on the output, called *recessing impact*, generally of less entity if compared to the previous one [7].

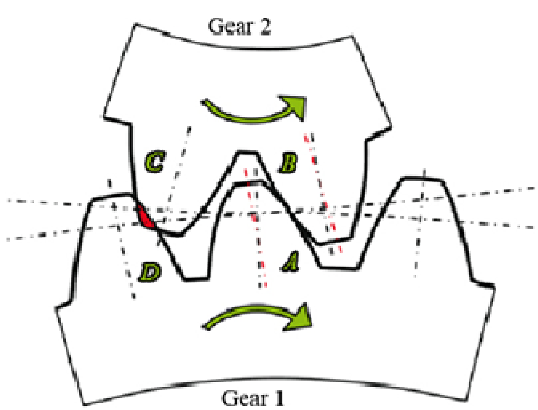

Fig. 1 - Meshing gear: red colour highlights the early contact

These anomalous contacts, therefore, in addition to shocks, generate a series of wrong loads distributions on the teeth. Because such dynamical loads occur on all pairs of meshing wheels, they are the source of vibration and noise [19].

The TE can be expressed through simple relations, which take into account the number of teeth (Z_1 and Z_2) and the angular position (ϑ_1 and ϑ_2) of the meshing wheels (driving and driven respectively) (angular TE), and the base radius of the driven wheel (r_{b2}) (TE along the line of action):

$$TE_\vartheta = \vartheta_2 - \frac{Z_1}{Z_2}\vartheta_1 \quad [\text{rad}]$$

$$TE_r = r_{b2}\left(\vartheta_2 - \frac{Z_1}{Z_2}\vartheta_1\right) \quad [\mu\text{m}].$$

The first equation represents the angular error of the gear wheel, measured in radians. The second equation represents the TE detected on the line of action, measured in μm.

To graphically obtain the curves of TE we can use the map of Harris [20].

Finally, we remark an objective difficulty to measure TE during the production of gearboxes [8,9].

2.2 – Forces of Friction

These kinds of forces are due to friction between the gear teeth during their meshing. They are characterized by a combination of rolling and of sliding (therefore not only rolling). Such a condition generates a vibration at meshing frequency.

The change of the direction of sliding at the primitive point suddenly generates a reverse of the forces of friction. These forces can be large enough to cause an excitation which can influence the noise of the gear. This effect is most pronounced on the spur gears, because the point of contact of the primitive is limited within a well-defined angular position. Because of relative motions of the teeth during their contact it is essential the finish of teeth surface in order to reduce the vibrations.

The manufacture of a gear goes through several steps which tend to improve more and more the mechanical, geometrical and topological properties of the surface of the teeth, with the main goal of achieving the maximum of strength and the minimum meshing noise.

The methods of surface finishing are manifold and each is characterized by more or less advantageous aspects depending on the type of wheels to be produced. The most commonly finishing methods are:

• Shaving, is performed before the heat treatment, and this is its major limitation. In fact, the geometrical distortions caused by heat treatment affect, sometimes seriously, the good quality of surface finishing achieved with this method;

• Grinding, this is performed after the heat treatment and thus provides an excellent geometrical accuracy on the finished surface. We can distinguish at least two fundamental grinding of gears: the threaded wheel grinding and the profile grinding wheel, profiled like the flank of the tooth;

• Honing or surface polishing, is performed after heat treatment. Again we distinguish:

a) the pure and simple honing consisting on polishing surface after earlier finishing operations which aims to improve the state of the tooth surface (lower R_a, defined in the following);
b) power honing, the most employed today, *i.e.*, a finishing operation performed after the tooth generation, which upgrades both the geometry and surface condition.

Each finishing method leaves on the surface marks of irregularities due to machining features. One of the parameters used for judging the condition of the surface is the roughness R_a, measured in micrometers. Irregularities can occur randomly, such as on the pieces obtained by casting or subjected to sanding. Sometimes the irregularities show a regular pattern characteristic of machining with chip removal. The roughness is actually composed of a series of grooves more or less ordered and regular, showing variable depth and placed on the surface.

Such a parameter of roughness is defined and measured by means of a plane orthogonal to the surface. This plan, called *relief plan*, crossing the surface, defines the *actual profile* as the intersection of two surfaces (Fig. 2).

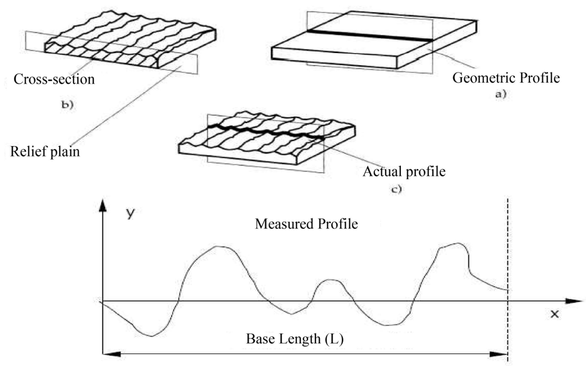

Fig. 2 – The definition of the roughness

The surface roughness is measured by means of an instrument called *profilometer*. It inspects the surface with a thin probe and records the irregularities on a diagram (or on display) after appropriate amplification. Normally, the roughness is denoted by the symbol R_a (measured in micrometer) as we shall see shortly, however it does not give a complete *state of the surface*.

The detection of roughness is performed on a length L_n called *length of evaluation*. It is 5 times the base length L which depends on the expected value of the roughness according to the values reported in the table below (Table 1).

Expected roughness R_e (μm)		Base Length L (mm)	Evaluation Length L_n (mm)
Beyond	Up to		
0.006	0.02	0.08	0.4
0.02	0.1	0.25	1.25
0.1	2.0	0.8	4.0
2.0	10.0	2.5	12.5
10.0	80.0	8.0	40.0

Table 1 **-** Evaluation of length for detecting the roughness

For the determination of the roughness R_a the average of the line profile is taken as reference. It is the line which minimizes the squared sum of distances of the points of profile from the line itself (Fig. 3).

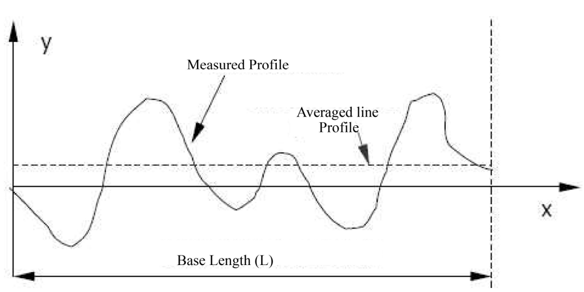

Fig. 3 – The determination of the average line reference

Once selected the mean line as *x*-axis, the roughness R_a is defined as the average value of the ordinates *y* (the absolute value) of the profile. So R_a is the arithmetic average of the distances of contour points from the mean reference line (measured in micrometers).

By means of the shaving, surface with R_a = 0.8μm can be obtained. By means of grinding and honing it can be achieved 0.1μm as roughness. It depends on the working conditions but mainly by the grain size of the wheels. But in addition to the depth of the marks of manufacturing we have to consider their direction and distribution onto the surface of interest. We are speaking here about the *surface structure* (or *Texture*).

Because the type of texture affects the noise generated by the teeth coupling, it is clear that many efforts, in order to improve the state of the surface and to understand the relationship between texture and noise with reference to various couplings, are currently making [10,16].

Fig. 4 shows three typical textures obtained through the aforesaid finishing operations. In particular, in the Fig.4a are shown marks of shaving after the heat treatment; the Fig.4b shows the typical form of marks left by the grinding and the Fig.4c shows the surface texture after the honing treatment.

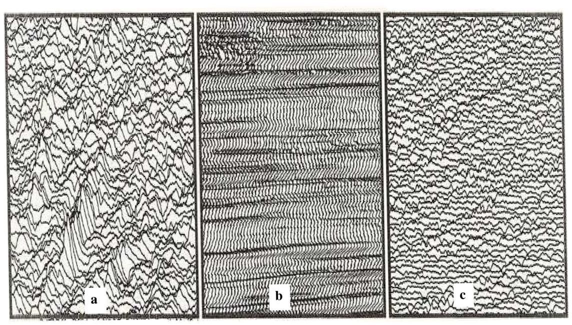

Fig. 4 – The texture of surface generated by three different finishing methods

We can immediately see that, the marks produced by means of grinding have a longitudinal direction. By coupling two gears obtained with this manufacturing system, there is a kind of extra meshing between the marks that are in contact, plus a rubbing, mainly along a radial direction, more pronounced when the contact area is far from the primitive: this causes a component of noise like that a hissing. Many studies suggest, when it is possible, the coupling a pair of gear wheels with different treatments of surface finish [3,15,16].

From the noise point of view, the gear shaving leaves the diagonal lines which improve the GWN but the surface texture is better if finished by honing. This justifies the current growth of such a method of gear finishing in terms of mass production. However, because the operation of honing is quite expensive, several methods have been proposed for producing a less expensive texture. One of these is the so-called Low Noise System (LNS)[1] in which the grinding wheel, in addition to the rotation and advancement, has a particular vibration in a radial direction which allows to modify the texture of the surface, without affecting the geometrical accuracy and the manufacturing time. Another method, proposed by Gleason[2] is the Variable Speed Method (VRM) that, during the operation, changes the feed and the speed of rotation in order to obtain a change of the distribution of the marks of manufacturing. Gleason's VRM grinding process makes delicate kinematic modifications in the grinding process to make less noisy gears.

2.3 – Dynamic meshing forces and PPTE

The dynamic meshing forces are the result of the combination of stiffness variations and of transmission errors; they are transmitted through the bearings and from their seats to the gearbox.

It is well known that the sliding reaches the maximum value when the teeth come into contact, while during the meshing it decreases down to zero. That value is reached when the point of contact between the wheels reaches the point of the primitive, where there is pure rolling.

The fluctuations of these friction forces, due to "passage" from rolling to sliding, are often so high to determine an excitation capable of influencing the noise affecting the gear.

As said above, the TE is generated by the load acting on the teeth during their meshing which, in part, depends on their bending [7].

Therefore, when the load varies also varies the TE. A useful parameter for measuring the TE is called Pick to Pick Transmission Error (or PPTE for short).

It represents the difference between the minimum and maximum of the values shown by the TE during the meshing. It is useful, as argued below, for a preliminary evaluation of the noise of the gear-box (Fig.5).

[1] Reishauer, Gear Grinding Technology, Industriasse 36, 8304 Wallisellen, Switzerland.
[2] Gleason Corporation, 1000 University Avenue Rochester (NY).

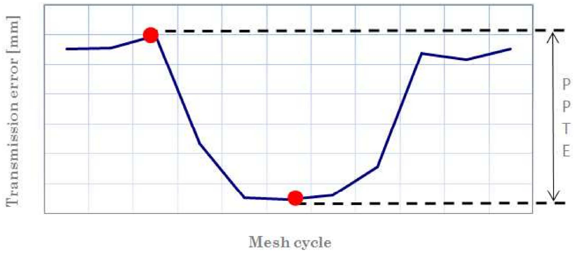

Fig. 5 - Peak to Peak Transmission Error

To reduce the noise of the gears, one of the main actions is to minimize the variation of TE, and this means to reduce to an absolute minimum the PPTE. To achieve this goal it is necessary "to restore" the tooth profile to the ideal values by operating through the modifications of the micro-geometry, in order to compensate the TE, due to the bending of the teeth [17]. Such improvements have as main goal the reduction of unwanted impacts during the meshing of pairs of teeth and to get less impulsive tendencies for the distributions of loads, as shown in Fig. 6, where the position is normalized.

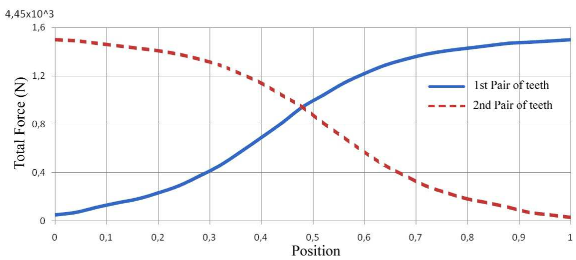

Fig. 6 – Distribution of forces between two pairs of meshing teeth

Micro-geometric modifications, made on the profiles of the teeth, for the considerations mentioned above, are optimized for specific torque values (*e.g.*, see Fig. 7). On the gear-boxes, especially in the automotive market, because they work at variable torque values, often, such modifications are made by trial and error.

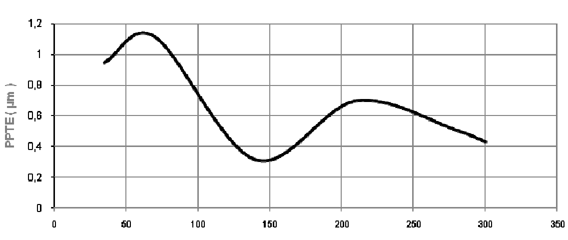

Fig. 7 – Variation of PPTE *vs* Torque

3 Methods

For the characterization of the phenomenon and for the vibro-acoustic comparison, in optical Gear Whine, we employed the test bench showed schematically in the Fig. 8 [10,13,21].

The test bench, from the mechanical point of view, can therefore be considered a "hybrid" machine, because it combines the "dynamometric" technology (for reproducing the torque/average speed to the primary and the resistances to the drive-shafts) with the "servo-hydraulic" technology (for reproducing the complex irregular motion characteristics of internal combustion engines).

However, regarding the operation, we can say that the primary shaft of the gearbox is put in rotation by an electric motor connected in series, via a flexible coupling, to a torsional hydraulic pulsator; at the same time, the two drive-shafts are braked by means of two additional electric brakes (Fig. 8).

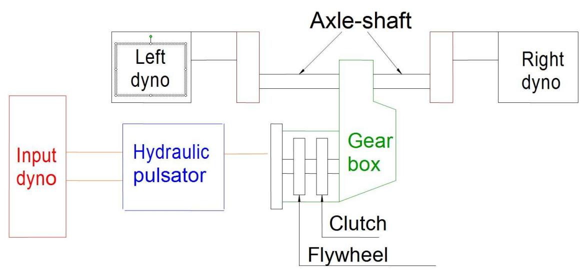

Fig. 8 – The scheme of the test bench

In the following the main components:

- 2 electric motors, 200-220kW power, torque of 3000-3200Nm and 3000rpm, reproducing, by means of appropriate coefficients, the load path and the simulation of the behavior of the car;

- 1 electric motor, 370kW power, 880Nm torque and rotation speed of 7000rpm, which simulates the working car engine;

- 1 hydraulic actuator, capable of reproducing the irregularities of the combustion process with a dynamic rotation of $\mp 35°$, static torque of 2260Nm and rotation speed of 7000rpm.

Such a configuration allows us to disengage completely from the vehicle and from the engine applications. To allow a detailed study of the phenomenon, the bench is installed in a semi-anechoic room, where the electric motors are shielded and the floor is acoustically covered with soundproofing material. Thus the gear-box, in practice, represents the unique source of noise. Such a set-up minimizes the background noise and the reflections of the room.

The detection procedure, performed by following the new standard of Noise Vibration Harshness (NVH), requires, for the acquisition of the signals, the following sensors:

• 1 magnetic pick-up, used for signal acquisition of the rotation speed of the primary. It is screwed into a threaded hole, drilled in the gearbox bell near the flywheel;

• 1 thermocouple (Thermo Engineering, Ni-Cr), used for monitoring the temperature of lubricant inside the gearbox. It was placed in a hole of the outer box, directly in contact with the lubricant;

• 1 single-axis accelerometer (Brüel & Kjær, type 4384) used for vibration measurements. It was placed at the bottom of the gearbox (Fig. 9);

• 3 microphones (Brüel & Kjær, type 4190) were employed for acoustic measurements. They were placed in a semicircle close to the gear-box at a distance of one meter from it.

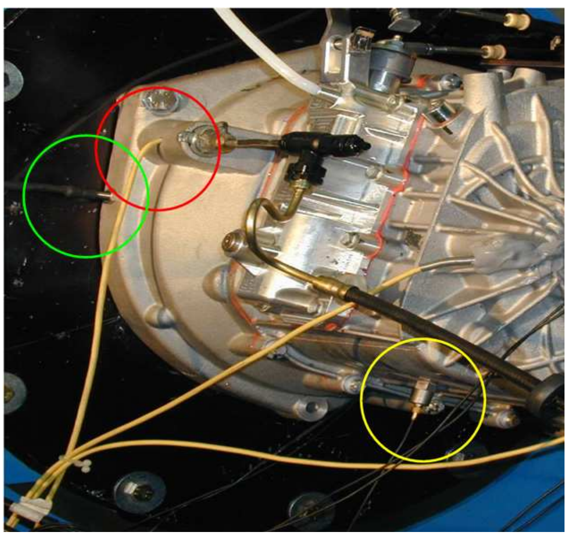

Fig. 9 – The set-up of the gearbox: the accelerometer (yellow), the pick-up (green) and the thermocouple (red)

For a more detailed analysis, with reference to the number of revolutions, the measurement range has been divided into three sub-ranges: low speed (1000÷2000rpm), middle range (2000÷3000rpm), high speed (3000÷4000rpm).

The tests, related to the rotation speed, were performed as below:

• Slow acceleration in the 4th gear from 1000rpm to 4000rpm within 60s with constant torque applied to the primary, ranging between 30÷300Nm, with the increment of 30Nm for the whole acceleration ramp;

• Steady speed in the 4th gear at 1000, 2000, 3000, 4000rpm with constant torque applied to the primary, ranging between 30÷300Nm, with a step of 30Nm for each test.

Following the acquisition of the acoustical and acceleration signals as well as the tachometer signal (performed by means of the LMS SCADAS MOBILE, Test.Lab, data-logger), we proceed to the data processing by extracting, from the recorded signals, the meshing orders of most interest for evaluating the phenomenon of gear whine. In particular it was extracted, for each gear, the order of meshing and the order of the last drive ratio (LDR for short).

In fact, to analyze the acquired signals one of the fundamental data that we must take into account is the number of teeth of each meshing gear as well as the final drive ratio (pinion - differential crown).

By means of such data it is possible to calculate the meshing order and the order of LDR for each meshing gear. This is essential to extract the related diagrams from the processed signals.

Taking as reference the rotation of the primary as the same rotational speed of the engine, we can define the concept of *order of meshing gear* as the number of events occurring during a complete revolution of the crankshaft. For a complete visualization of the data a special color map was created showing on the abscissa the number of revolutions of the primary, on the ordinates the values of torque applied to the primary and a color scale representing the values of acoustic pressure due to the noise. By evaluating the color changes it is possible to distinguish the areas where the noise is higher (red) from the area showing lower noise (blue) (*e.g.*, see Figg.10 and 11).

For the specific pairing of motor-gearbox under test, the meshing order of each gear corresponds to the number of teeth of the respective driving gear.

Instead, the order of LDR for each gear can be easily calculated with the following formula:

$$\mathrm{LDR} = \frac{Z_{pinion}}{\tau}$$

where LDR, as said above, is the order of last drive ratio, Z_{pinion} is the number of teeth of the pinion of the secondary which meshes with the crown of the differential and τ is the transmission ratio of the gear engaged.

From the aforesaid diagrams we can make a comparison between two gear-boxes tested during the trials.

In particular, by comparing such diagrams for several speed and torque applied to the primary, it is possible to evaluate the acoustic level (noise) of the two gear-boxes showing just a little geometrical difference on the profile of teeth.

4 Results and Discussion

The main results arising from the experimental test-bed can be summarized as follows.

Data reported on the ColorMap (Figg. 10 and 11) show the acoustic results in the 4^{th} meshing gear.

They concern two different gear-boxes made by the maker A (say) and the maker B (say).

- Low speed (1000÷2000rpm)

For low values of the number of revolutions, the gear-box A globally presents a rather low noise level, which follows an increasing trend, directly proportional to the number of revolutions. In particular, the values of acoustic pressure vary between 40dB(A) and 65dB(A). Only in some limited range (1650÷1800rpm and 1900÷2000rpm) the level of the acoustic pressure reaches 70dB(A).

The comparison of the acoustic noise arising from these diagrams shows that the gear-box made by the maker A, at the lower speeds, is less noisy than the one made by the maker B. The middle range (*i.e.*, between 2000÷3000rpm, red area) shows a sharp deterioration of the gear-box A if compared to the B one. A substantial improvement of the gear-box A at higher revolutions is quite evident. The two gear-boxes tested have quite different behaviors. It must be emphasized that, due to the low masking effect of the engine at medium engine speeds, the gear-box B is globally less noisy if compared to the A one.

For that reason we proceeded to the study of micro-geometry in the fourth gear by making specific geometrical modifications to the profile of the teeth in order to reduce the acoustic noise of the gear-box A (denoising improvement) [11,12,17].

The gear-box B, made by the competitor, at low engine speeds is noisier. In particular, at very low speed (1100rpm), the gear-box under test reaches the value of roughly 70 dB(A).

The higher noise level for this gear-box also occurs at 1800÷2000rpm, where the value of acoustic pressure reaches 80dB(A). Furthermore, the trend, almost sinusoidal, of the values of the acoustic pressure, as the speed increases, produces an unpleasant effect siren (during the ramp), which at low speed, due to a limited masking of the engine, is quite annoying.

- Middle range (2000÷3000rpm)

In the middle range of rotation the worst operating conditions for the gear-box A are found. In fact, from 2000 to 3000rpm, only in some intervals (2450 and 2950rpm) the values of acoustic pressure fall below 65dB(A), while in the rest of operating points, the value exceeds in many cases 70dB(A). It also identifies a particularly critical area of running (2600÷2750rpm), where the values go over 75dB(A).

Vice-versa, the diagram of the gear-box B shows a sinusoidal trend, but shows lower values of acoustic pressure if compared to the gear-box A, reaching the average values of less than 65dB(A).

In addition, the gear-box of the competitor B never attains the critical values of 75-80dB(A), achieved on the contrary by the gear-box A.

- High speed (3000÷4000rpm)

From 3000 to 4000rpm it is found that the gear-box A shows a considerable noise reduction. The recorded average level was of 65dB(A) in almost all the area of the high speed range.

The gear-box made by the competitor B abandons the sinusoidal trend of the values of acoustic pressure, which, in this operating range, are close to the value of 70dB(A).

- Analysis of the applied torque

For the gear-box A the trends of the acoustic pressure values as function of applied torque are almost similar for all the speeds. In particular, we can observe a trend at first decreasing and then increasing with the increase of the applied torque. The diagrams, for almost all the engine speeds, show a minimum value of noise emissions, approximately around 120Nm, which is the theoretical value optimized by the engineers. Only at low engine speed (1000÷1200rpm), the trend reverses and the diagram shows the maximum value of 120Nm.

For the gear-box made by the competitor B the noise emission values as a function of torque applied show a trend similar to that previously described for the gear-box A, but the curves show in this case two minimum points, one at 30Nm and the other one at 180Nm.

The data analysis shows that the correlation between the trend of the values of acoustic pressure and number of revolutions is not always directly proportional to the applied torque [2].

To verify the above statement, in the Fig.12 are reported diagrams on the acoustic pressure of the gearbox A obtained with stationary engine revolutions in the 4th gear at 1000, 2000, 3000 and 4000rpm and torque ranging from 30 to 300Nm.

Such diagrams, in fact, allow an immediate evaluation of trends of acoustic pressure versus the torque and speed variation. The noise, therefore, as shown in the previous paragraphs, is dependent on the geometric parameters of the gears, confirming that the "microgeometry" of gear plays a key role in terms of noise.

From the diagrams of Fig.12 it is also possible to highlight that the profiles of the gears in the 4th gear, as said before, have been acoustically optimized for a torque value close to 120Nm.

In the following are superimposed the diagrams in the 4th gear at 2000rpm *vs* the acoustic pressure (Fig.13), and *vs* the acceleration (m/s^2) (Fig.14).

By comparing such diagrams, we observe a strong correlation of trends, proof of the fact that the transmission error is one of the major factors to take into account when we design the gears in order to contain the global noise level of the gear-box.

Based on such considerations, we proceeded to study the gear from a micro-geometrical point of view by making specific modifications to the profile of the teeth in order to improve acoustically the gearbox A (denoising improvement).

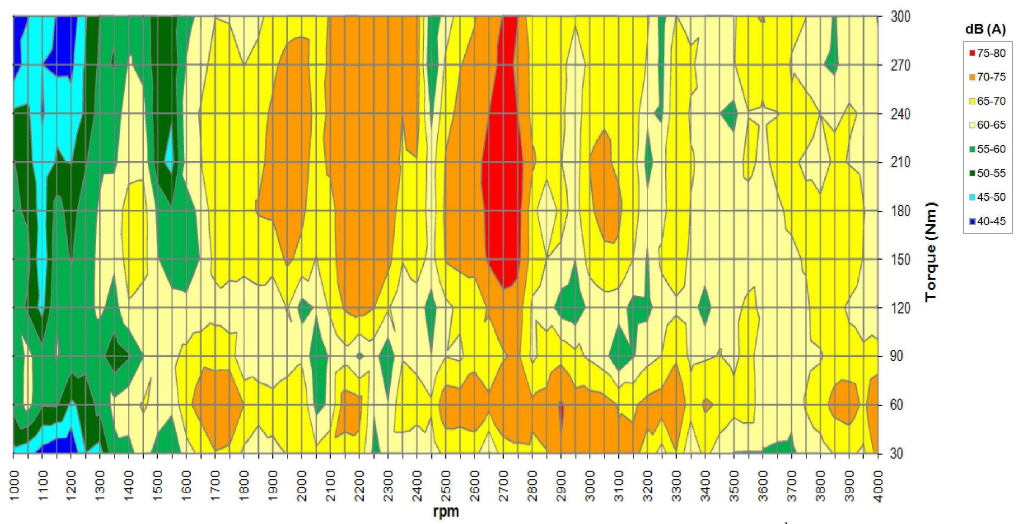

Fig. 10 - ColorMap - Microphone - Order of meshing gear - 4th gear, maker A

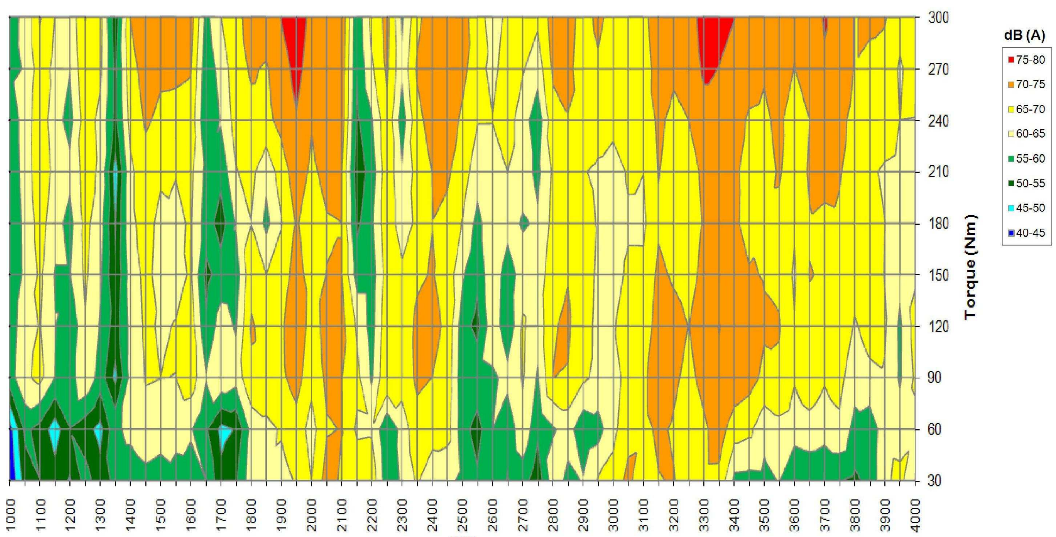

Fig.11 - ColorMap - Microphone - Order of meshing gear - 4th gear, maker B

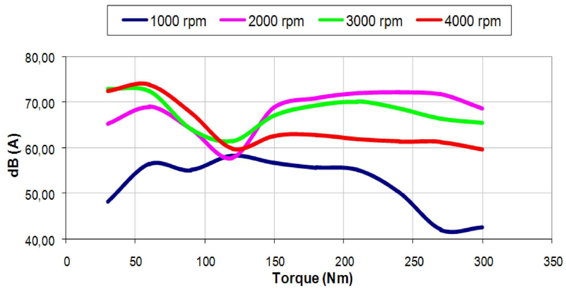

Fig.12 – Acoustical pressure *vs* applied torque, in 4th gear, for several rpm

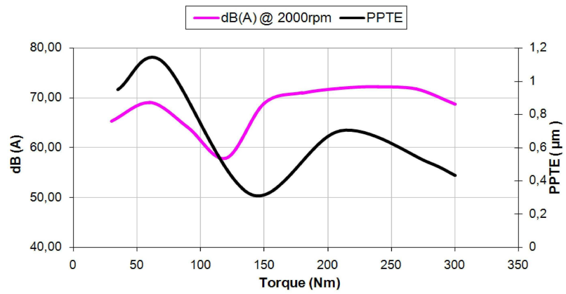

Fig.13 – Comparison of acoustical pressure and PPTE *vs* applied torque, in 4th gear, at 2000rpm

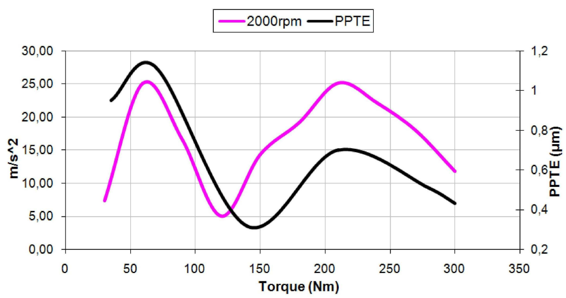

Fig.14 – Comparison of vibration and PPTE *vs* applied torque, in 4th gear, at 2000rpm

With this aim, several modifications were made by acting firstly on the curvature of the involutes profile through the grinding and secondary by reducing the tolerances of the machines tools. It was concentrated on the flank of the tooth and the task is to compensate not only the deformation of the body wheel, but also the shaft and the support structures (Fig.15).

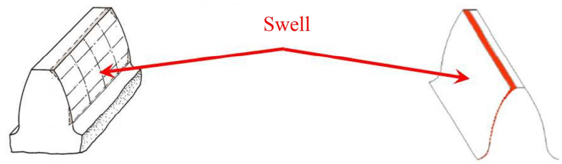

Fig.15 – The swell of the involutes profile

By examining the results of the acoustic pressure after such modifications of the curvature of flanks, we obtained the following ColorMap (Fig. 16).

The modification of the gear-box A seems to be a better solution in terms of noise, as well as after being improved the critical area of middle range. The gear-box shows lower values even at low and at high rotation speeds (Fig. 16).

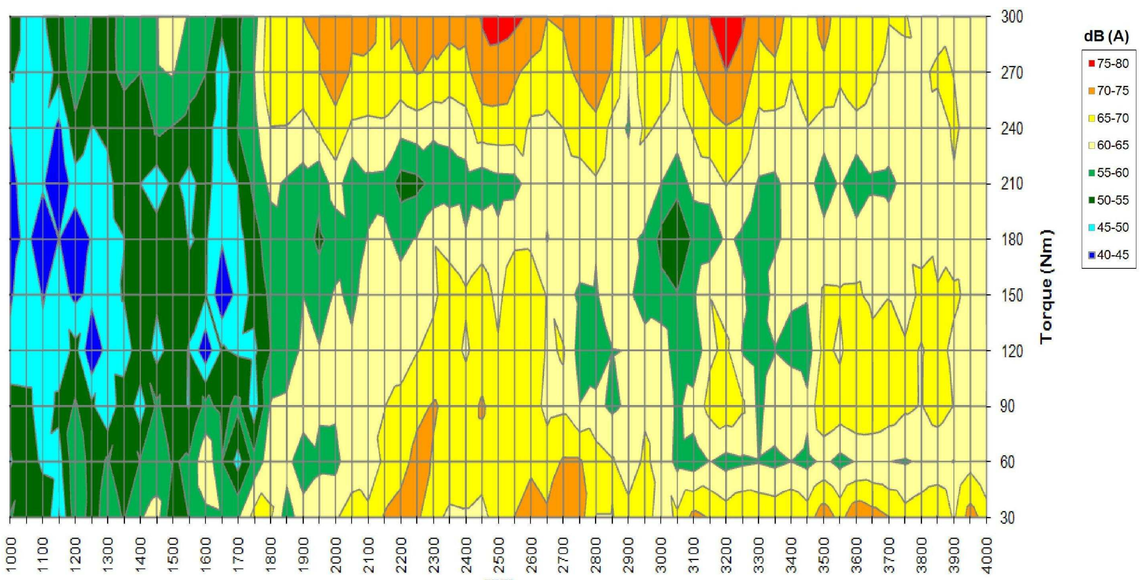

Fig.16 - ColorMap - Microphone - Order of meshing gear - 4th gear, gear-box A modified

The excessive reduction of the value of curvature increases the contact of the surfaces. This leads, as collateral effect, high specific pressure localized on the top of tooth, generating critical areas with subsequent wear of the gear tooth due to the fatigue (Fig.17) [11,12].

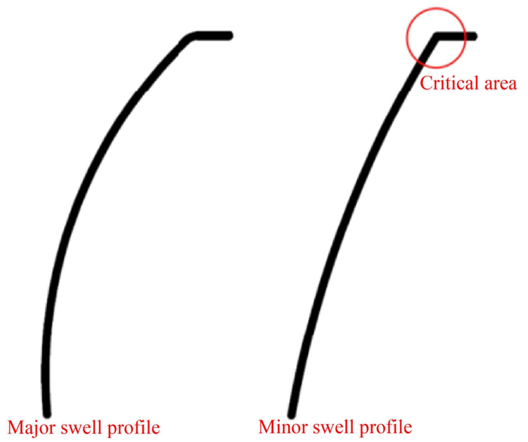

Fig.17 - Comparison of swell profile

5 Conclusions

To limit the phenomenon of the Gear Whine Noise it has been shown that we can act on several parameters:

- reduction of friction forces, by improving the finish of the teeth surface. Sometimes the cost of this intervention is much expensive;

- modification of the curvature of the profile (swell), designed to minimize the specific pressures on the tooth;

- machining tolerances, by reducing them it allows to limit the variation of transmission error.

Finally, we want to remark the importance of such improvements in particular from the economical point of view in terms of cost/benefits and of company image.

The present experimental work pointed out the importance of the transmission error as the main factor that causes the GWN.

In fact, the analysis assessed on the test bench shows that the gear whine is proportional to the change of the transmission error during the meshing cycle. In order to understand deeply the described phenomenon a new series of tests are been planned with further instrumentation of test-rig.

However, at this stage of the present research, we can certainly make some valid considerations to reduce the phenomenon of GWN.

Good results can be primarily achieved through the reduction of machining tolerances. Generally, the lower are the tolerances the higher the costs of production.

Once the tolerances are defined, the modification of the teeth profile is the quickest and cheapest way to get good results in terms of acoustic noise.

Secondly, the adoption of modifications on profile of the teeth allows to limit the transmission error under load.

This action, as demonstrated by our experimental measurements, allows a good reduction of global acoustic emissions, although the acoustic noise level is optimized for only a narrow range of values of applied torque.

Therefore, the aim of designers is not easy: to find a balance between all the parameters involved by trying to minimize costs and to optimize the acoustic emissions.

If we consider all the variables involved, it is clear that, despite the known sources of gear whine, not always the precautions taken by the designers guarantee good results.

For that reason, downstream of production, in order to certify the quality, specific vibro-acoustic characterizations are performed on the gear-boxes.

References

[1] E. Zwicker, H. Fastl, *Psycho-acoustics: Facts and Models*, Springer-Verlag Berlin Heidelberg, 2007.

[2] A. Lorea, G. Morra, G. Ruspa, Advanced Statistical Methods for the Correlation Between Noise and Transmission Error in Gears, *Fiat Research Center*. (SAE Technical Paper 865144).

[3] R. J., Drago, How to Design Quiet Transmissions, *Machine Design*, Dec. 1980.

[4] V. Niola, V. Avagliano, G. Quaremba, The Gear Whine Noise, *4th WSEAS Int. Conf. in Engineering Mechanics, structures engineering Geology*, Corfù, July 14-16, 2011, pp. 445-450.

[5] V. Niola, V. Avagliano, G. Quaremba, The problem of GWN, *Mathematical Methods and Techniques in Engineering and Environmental Science, Catania*, Sicily, Italy, November 3-5, 2011.

[6] D. B. Welbourn, Fundamental Knowledge of Gear Noise – A Survey, *Proc. Noise & Vib. Of Eng. and Trans.*, I Mech E., Cranfield, UK, July 1979, pp 9-14.

[7] J. Borner, D. R. Houser, Friction and Bending Moments as Gear Noise Excitations, *SAE Technical Paper*, 961816.
[8] D. R. Houser, G. Wesley, Methods for Measuring Gear Transmission Error Under Load and at Operating Speeds, *SAE Technical paper* 891869.
[9] C. Park, The Load Transmission and Vibration Characteristics of Automobile Gear, *SAE Technical Paper* 932917.
[10] A. Forcelli, A New Approach for Objective Measurements of Transmission Gear Rattle Noise, *JSAE, Tokyo*, 2004.
[11] Y. Iwase, K. Miyasaka, Proposal of Modified Tooth Surface with Minimized Transmission Error of Helical Gears, *JSAE Review* 17 (1996) pp 191-193.
[12] M. S. Tavakoli, D. R. Houser, Optimum Profile Modifications for the Minimization of Static Transmission Errors of Spur Gears, *Journal of Mechanisms, Transmissions and Automation in Design* Vol. 108 (March1986) pp 86 – 95.
[13] V. Niola, G. Quaremba, The Gear Whine Noise and vibro-acustic emission of gearbox, *WSEAS Int. Conf.*, Venice, Italy, Mar 8-10, 2011, pp.138-143.
[14] S.N. Doan, J. Ryborz and B. Bertsche, Rattling and clattering noise in automotive transmissions — Simulation of drag torque and noise, *Tribology and Interface Engineering Series*, Vol. 43, pp. 109-121, 2003.
[15] S.K. Jha, Characteristics and sources of noise and vibration and their control in motor cars, *JSV*, Vol. 47, Issue 4, pp. 543-58, 22 august 1976.
[16] V. Niola, G. Quaremba, The Gear Whine Noise: the influence of manufacturing process on vibro-acoustic emission of gear-box, 10th *WSEAS Int. Conf. on Signal Processing, Robotics and Automation* (ISPRA '11), Cambridge Feb. 20-22, 2011.
[17] D.R. Houser, J. Harianto, Design Robustness and Its Effect on Transmission Error and Other Design Parameters, *The International Conference on Mechanical Transmission*, 2001.
[18] H. N. Ozguven and D. R. Houser, Mathematical models used in gear dynamics – A review, *Journal of Sound and Vibration*, 121, 383-411, 1988.
[19] J. Wang, R. Li and X. Peng, Survey of Nonlinear Vibration of Gear Transmission Systems, *Appl. Mech. Rev*, 56 (3), 309-329, 2003.
[20] S. L. Harris, Dynamics loads on the teeth of Spur Gears, *Proc. of the Institute of Mechanical Engineers*, Vol. 172, n. 2, 87-101, 1958.
[21] V. Niola, G. Quaremba, A. Forcelli, Gear noise detection. An integrated method using Fourier and Wavelet approach, *6th WSEAS Int. Conf. on System Science and Simulation in Engineering*, Venice, Italy, November 21-23, 2007.