Our measurements have proved that the theoretic supposition of average yearly noise growth of 1dB in great towns is true, but it is not only the noise level that has increased, but the time of the exposure as well. The noise during the daily hours appears to be of an continuous character, the phenomenon of the so called traffic peak has not been observed anymore.

5 Conclusion

The theoretical hypothesis claiming that acoustic milieu in the cities may represent an environmental risk factor to the health of the population can be proved also by results of this work. Further increase of noise energies is undesirable because of its influence on origin and development of "civilisation diseases". Improvement of general conditions should take into account environmental noise and feasible strategies to avoid excessive levels should be developed whenever possible.

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ACTIVE LOWERING OF NOISE OF TRANSMISSIONS OF MOBILE MACHINES

Ján Haško

The paper deals with the simulation method of the course of total meshing stiffness of gears with a slant gearing by means of finite elements method that was applied to searching of the optimal choice of contact ratios εα and εβ from the point of view of the compensation of the fluctuation of the course of total meshing stiffness.The result is optimizing criterion for the choice of basic parameters of the gearing that was also verified by an experiment.By utilization of proposed optimizing criterion at the design of transmissions is possible to lower additional dynamic loads of teeth what is connected also with increasing of durability,reliability and mechanical efficiency of gears and also with decreasing of their noise.

1. Introduction

It is practically impossible to ensure the fulfilment of more and more strict noise limits by passive insulation methods. It is inevitable systematically to investigate physical properties of individual machine parts from the point of view of further noise lowering and in this way to create presumptions for the effective solution of this matter. This activity, named sometimes also the active noise lowering, besides the noise lowering of machines, is connected also with increasing of their mechanical effectiveness, durability and reliability, and in this way practically contributes to the quality increasing of the technical level of machines. The analysis of particular noise sources of mobile machines, according to requirements of our manufacturers, resulted in the need to aim mainly at the solution of noise matters of transmissions of mobile machines.

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2. Analysis of causes of transmissions noise

The main cause of the noise generation in transmissions are impulses arising in gears, caused by the torsional vibration of individual parts of transmissions. The torsional vibration of these parts is caused by the moments of inertia of individual parts of the driving mechanism, torsional stiffness of shafts, and also by the sources of inner dynamical phenomena. The theory of torsional vibration of linear systems, caused by the moments of inertia of individual parts of driving mechanism and by the torsional stiffness of shafts, belongs between the best managed fields of the machinery dynamics. There are mainly the matters of inner dynamics of gearings that became significant from the point of view of noise lowering and transmissions vibration.

The following sources of inner dynamical phenomena belong between the most important: periodical change of meshing stiffness, production deviations in toothing, meshing impulse, change of specific length load of teeth as the result of total length change of teeth contact, and step change of the frictional force direction in the rolling point caused by the change of direction of slip speed.

There are mainly matters concerning the torsional vibration of toothed wheels, as the result of periodical change of meshing stiffness, that become significant from the point of view of transmissions noise of mobile machines, in connection with expanded technological possibilities and increased production accuracy [1]. By many professionals in this field (Kesan, Özgüven, Houser), this influence to the noise is decisive mainly in precise, high-speed, modified gears.

From the point of view of lowering of noise and vibration of gearing is advantageous such "smooth" course of meshing stiffness as possible [2]. It was also one of reasons of extended utilization of toothing with slant teeth. But, in connection with it, there is not till now a satisfactory answer for the question of optimal values choice of contact ratios of profile ε_α and step ε_β [3],[4],[5]. This question is important mainly for high-speed gears, in operational range of which several parameter resonance fields are located, resulted from the meshing stiffness that is periodically changing. The gears of transmissions of mobile machines with the wide spectrum of operational modes are the typical example.

3. Modelling of the course of total meshing stiffness of wheels with slant teeth

The periodical change of meshing stiffness is caused by the change of tooth deformation during the mesh and by the change of number of teeth pairs that are in mesh in the same time. The typical course of deformations of teeth that are together in mesh $\delta_{1,2}$ on the meshing way (d.z.) is stated on figure 1. There is also the course of one-pair meshing stiffness c_p and the course of total meshing stiffness c, that is the superposition of one-pair meshing stiffnesses. Illustrated courses of meshing stiffness of wheels with slant teeth result from the presumption of continuous and linear load distribution along the all contact length, defined as the sum of contact lengths of all teeth pairs that are in the meshing field in the same time. This presumption can be ensured approximately only by well placed precise toothed wheels.

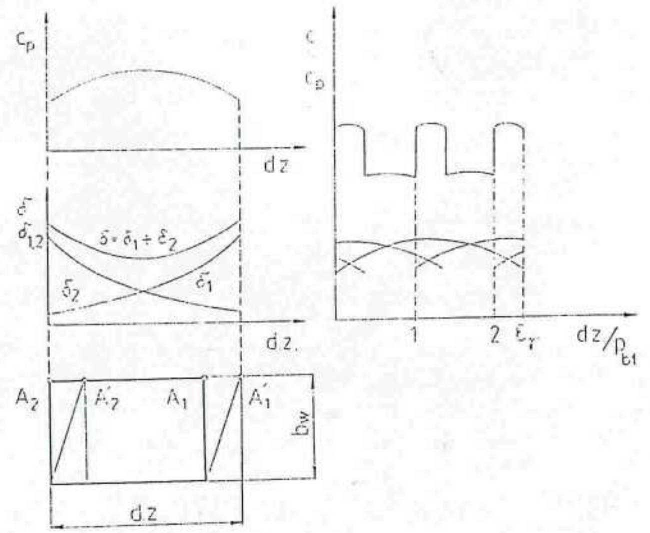

Fig. 1 Course of meshing stiffness of gears with slant teeth

An influence of contact ratios of profile ε_α and step ε_β ,while the value $\varepsilon\gamma$ on "fluctuation" of the course of total meshing stiffness was a non-integer constant value, was verifying by the calculation of tooth deformation on the meshing way by the method of finite elements, from which than corresponding courses of total meshing stiffness were calculated. Individual courses of total meshing stiffness were comparing according to their root-mean-square deviations and according to coefficients of their Fourier series,with an objective to find an optimal choice of ε_α and ε_β from the point of view of compensation of "fluctuation" of the total meshing stiffness course.

Courses of total meshing stiffness of gears were calculated with the following parameters:

- number of teeth of wheel and pinion: $z_1/z_2 = 34/25$
- standard module: $m_n = 2.55$ mm
- angle of teeth slant on the distance circle: $\beta = 28°21'$
- sum of unit shiftings: $x_1 + x_2 = -0,975$

Individual possibilities of the choice of contact ratios of profile and step, at approximately constant value $\varepsilon\gamma$, were ensured by the mutual axial shifting of toothed wheels, that means by the change of operational width of toothing b_w. The following possibilities of choice of ratio ε_β, when $\varepsilon_\alpha = 1.5$, were investigated:

1. $\varepsilon_\beta = 1$, then $\varepsilon_\alpha/\varepsilon_\beta = 1.5$
2. $\varepsilon_\beta = 0.9$, then $\varepsilon_\alpha/\varepsilon_\beta = 1.67$
3. $\varepsilon_\beta = 1.1$, then $\varepsilon_\alpha/\varepsilon_\beta = 1.36$

On the basis of the comparison of root-mean-square deviations and Fourier series coefficients of individual courses of total meshing stiffness (table 1), can be stated, that by the suitable choice of the contact ratios ε_α and ε_β, by such way the ratio $\varepsilon_\alpha/\varepsilon_\beta$ was maximal at given ε_γ, "fluctuation" of theoretical course of total meshing stiffness can be decreased in a great deal. By increasing of the ratio value $\varepsilon_\alpha/\varepsilon_\beta$ at given ε_γ, the height of teeth is increasing and their width is decreasing in the same time. As a result from found courses of total meshing stiffness of gears, the reduction of teeth width will be showed by decreasing of one-pair meshing stiffness mainly in the middle of the mesh, when the contact line passes through the all tooth width. Then the curve of one-pair meshing stiffness at the constant ratio value ε_γ becomes more flat, and the magnitude of "fluctuation" of the course of total meshing stiffness will decrease.

Tab. 1 Coefficients of the Fourier series c_n and root-mean-square deviations s of courses of total meshing stiffness

n	c_n [N.mm^{-1}]		
	1.	2.	3.
1	- 9 796.1	1 989.7	- 13 773.9
2	10 480.8	10 431.1	5 768.9
3	- 1 606.1	- 3 645.8	3 788.9
4	-1 998.4	-822.7	- 1 443.4
5	- 133.3	1 466.7	- 1 266.7
s	12 297.3	9 729.1	13 381.9

4. Experimental verification of the proposed optimizing criterion

An influence of the choice of ε_α and ε_β to the magnitude of "fluctuation" of the course of total meshing stiffness was verified by the indirect, comparative method, by comparing of level of power spectral density (hereafter L_{PSD}) components of sound pressure on the meshing frequency, and of its harmonic components with individual options of ε_α and ε_β, on the gears of the 5th transmission step of transmission of the mobile machine. Its parameters are stated in chapter 3. It was the precise modified gears, the noise of its then depended from "fluctuation" of the course of total meshing stiffness in a great deal. Because the total meshing stiffness is periodically changing on the meshing way section, that is equal to the front pitch on the basic circle p_{bt}, the torsional vibration on the meshing frequency will cause proportional magnitudes of "fluctuation" of the course of total meshing stiffness. Individual possibilities of the choice of ratios ε_α and ε_β were ensured by the small axis shifting of toothed wheels of measured gears, by the same way as courses of total meshing stiffness were modelling, for the same values of ratios ε_α and ε_β. By such way, the value of ε_β was changing and then also the value of ratio $\varepsilon_\alpha/\varepsilon_\beta$ at approximately of constant value ε_γ. Between the examined options ε_α and ε_β, the constant value ε_β was also included, and by this way the suitability of such choice was verified. Individual options ε_α and ε_β were verified at various revolutions and loads on the open testing state according to figure 2.

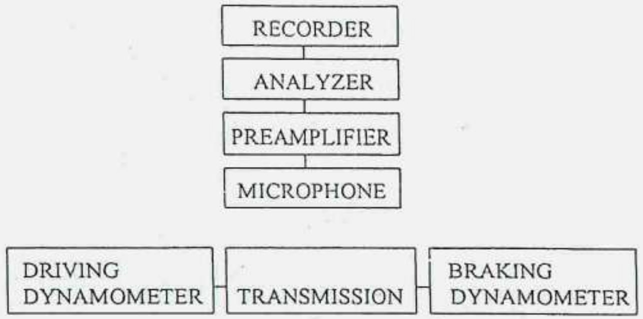

Fig. 2 Block diagram of testing state

Measured values of L_{PSD} components of sound pressure are with the combined uncertainty u = 0.4 dB at individual operational modes and at individual choices of ratios ε_α and ε_β processed clearly in table 2. On the basis of the comparing of these values is possible to say, that by the suitable choice of ε_α and ε_β by such way the ratio $\varepsilon_\alpha/\varepsilon_\beta$ was maximal at given ε_γ, the noise of gearing can be lowered in a great deal, as the result of decreasing of the magnitude of "fluctuation" of the course of total meshing stiffness. In our case, the value of step contact ratio 0.9 corresponded to the maximal ratio $\varepsilon_\alpha/\varepsilon_\beta$ and that is in conflict with recommendations of some directives prescribing ε_β as the integral. It results from the comparison of L_{PSD} values of sound pressure at individual operational modes, at individual ratios $\varepsilon_\alpha/\varepsilon_\beta$, that from the point of view of noise lowering of gears, the maximal ratio $\varepsilon_\alpha/\varepsilon_\beta$ is more significant than the constant value ε_β on the condition that the value ε_β or ε_α is still close to the integral.

Tab. 2 Measured values of L_{PSD} components of sound pressure in dB

Seq. of har.	Possib. of choice of ε_α ,ε_β	Revolutions [s^{-1}]/Load [Nm]									
		8.3/ 90	8.3/ 110	16.6/ 90	16.6/ 110	25/ 90	25/ 110	33.3/ 90	33.3/ 110	41.6/ 90	41.6/ 110
1.	1.	53.7	55.1	57.6	58.9	69.3	68.5	71.3	74.4	79.5	81.0
	2.	51.5	53.0	55.5	56.2	68.2	66.6	70.9	72.9	76.1	76.5
	3.	55.0	55.9	61.7	59.0	73.1	70.4	73.8	73.8	77.5	79.1
2.	1.	48.0	48.0	58.0	57.0	70.0	70.0	67.1	67.0	63.0	63.0
	2.	47.0	48.0	56.0	56.0	66.9	70.0	63.0	64.0	63.9	65.0
	3.	52.0	52.0	56.1	57.1	68.0	72.0	64.0	62.1	62.1	66.0
3.	1.	58.0	58.0	65.0	67.0	63.0	63.0	70.0	67.0	68.0	68.0
	2.	56.0	57.0	63.0	63.0	63.0	62.0	63.0	62.0	68.0	68.0
	3.	57.0	57.2	63.1	65.0	63.0	63.0	62.0	65.0	66.0	62.0

5. Conclusion

Results of theoretical and experimental investigation indicate in coincidence the possibility to decrease the magnitude of "fluctuation" of the course of total meshing stiffness of gears and by such way also the noise of gearings by the suitable choice of basic parameters of toothing. By utilization of proposed optimizing criterion at the design of transmissions of mobile machines, the further noise lowering of mobile machines can be reached. Its utilization is supposed mainly at the design of precise, high-speed, modified gears of transmissions of mobile machines, that are tuned at correct frequency from the point of view of outer and inner dynamics, and at which the total length of teeth contact is approximately constant. Besides the noise lowering, by the proposed optimalization of basic parameters of toothing, they will also lower additional dynamic loads of teeth that is connected also with increasing of durability and reliability of gearings and also with increasing of their mechanical effectiveness. By this way the proposed optimizing criterion contributes to increase technical level of quality of our machines.

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NOISE IN RAILWAY TRANSPORT

Tomislav Gruber - Peter Zvolenský

The negative effects of railway transport on environment. The locomotive as a system with certain quality characteristics. The quality parameters of locomotive the share of noise. Microclimate in a locomotive driver's cabin and influence of infrasound on the driver. Experimental methods of investigation noise and infrasound.

1. Introduction

The development of railways in Slovakia is based on the effort for integration into European transports routes. One of the prior fields of development is quality of transport, quality of transport vehicle. Quality of transport vehicle is define by more parameters. It is determine by safety, power, speed, reliability, maintenance, energy, quality of construction and by ecology parameters. Gas and liquid emission have big influence on the pollution of air, water, earth. Among the detrimental emissions loading the environment belongs the noise of railway vehicles too. Noise and vibrations what are not removable elements of each dynamic system influent the whole quality markedly.

The consequences of noise in railway transport has been systematically observing since the 60[th] [5]. Although in the past was achieved respectful success in the region of eliminate the noise the continually increase of transport involve additional reduction the acceptable limits. The sources of noise, transmissions ways, the effect at environment but also at the engine driver are searched on purpose increase the quality of railway vehicles. Many of quality parameters the engine like dustiness, warm, humidity, color adjustment, visibility and another ergonomic components are able to influent in positive way in benefit of operating staff. Noise and vibration at the stand of engine driver are not removable. Therefore is them spend uncommon attention by looking for quality of railway vehicles.

One of the noise components that hasn't been spent much room in research is the infrasound. It is known that effects of the noise are registered after a long time. Influence of the infrasound on humans is not direct and often is its effect debited to other sources.

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