Excitation of the Modal Vibrations in Gear Housing Walls

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Modal analysis, realized by application of finite elements method, results as a rule in a large number of modal shapes (frequencies) of natural oscillation. They are possible shapes and frequencies of oscillation. In real conditions, only some of them are excited. Determination of ways (mechanisms) of excitation and conditions under which a modal shape will be excited is the central question of this paper. For the purpose of elaboration of procedure and conditions of excitement, i.e. for the purpose of defining rules of excitation, we have used the results of modal analysis by applying FEM using the method of direct integration within FEM and and results of modal testing. The subject of analysis and examination is a gear housing. Modal activity of housing walls is in direct relation with the structure and intensity of noise emitted by the gearbox into the surrounding. Therefore, research of modal activities is of general importance for modelling the process of generation of noise in mechanical systems.

Keywords*: *modal analysis, gearbox noise , modal testing

1. INTRODUCTION

Gearbox noise is sound waves emitted from the housing walls into the surrounding. The housing surface emits the sound which penetrates from the inner space through the walls, as well as the sound generated by the housing with its natural oscillation. From this aspect, the housing walls have a double role: to be an obstacle to penetration of sound waves from the inside, i.e. to be the insulator of inner (internal) sound sources and the generator of tertiary sound waves due to natural oscillation. Insulation properties of sound of acoustic partitions, i.e. the housing walls are also in direct relation with modal properties (natural frequencies and shapes of oscillation). Therefore, it is very important to find an answer to the question what are possible modal shapes of oscillation of the housing walls and under which conditions each of them can be excited. The way and conditions of excitation of natural oscillation are, in this sense, of special interest. They are important, before all, because they will offer answers to many questions relating to penetration of sound waves through the housing walls. On the other hand, the "mechanism" of excitation of natural oscillation represents an unexplored field, which is the main aim of this paper.

For the time being, the available literature does not provide a direct answer to the question in which way and under which conditions certain modal shapes are excited. The author's experiences so far clearly point to the necessity of applying modal analysis for the purpose of solving that important question. However, in many practical cases, when harmonic excitations are close to structural frequencies, standard identification techniques fail. Therefore, the Mohanty and Prasenjit [8 and 9] suggest, on the example of a steel plate, a technique based on the Ibrahim Time Domain method which explicitly includes the harmonic frequencies known a priori. Therefore, the modified technique allows proper identification of eigenfrequencies and modal damping even when harmonic excitation frequencies are close to the natural frequencies of the structures. By applying the Finite Elements Method, modal analysis is used in a computer motherboard in [7], gears [10], and a gearbox [4] for the purpose of defining modal parameters, determination of amplitude-frequency characteristics, main shapes of oscillation, etc. In the field of low frequencies, the FEM application in modal analysis gives excellent results. However, modeling and vibration analysis by FEM and modal analysis become more difficult as the frequency becomes higher, and some approximations and hypotheses that have not yet been proven to be accurate are used in SEA (statistical energy analysis). In [6] a vibration analysis method based on modal analysis and the statistical method which enables analysis of the high-frequency vibration is presented. Example results are shown for a single-plate structure and an L-shaped structure. Gardanio, Ferguson and Fahy deal with the examination of plane wave transmission characteristics of circular cylindrical sandwich shell of the type used in aerospace industry in [2]. They have developed a model for prediction of the structural response and transmitted noise when a number of discrete masses are applied to cylindrical plate structures. Simulations show the effect of the number of structural and acousic modes included on the

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calculated frequency response, and indicate the number necessary for an accurate prediction of the resonant and non-resonant sound transmission through the structure. It is shown that restricting the acoustic and structural modes to those having natural frequencies within an interval of ±40 Hz and ±60 Hz, respectively, of the excitation frequency produces acceptably small errors in transmission estimate.

2. MODAL ANALYSIS

The research was realized on the example of a housing shown in Figure 1. It is a cast housing of a two-degree gearbox, reinforced with ribs and rings for increasing stiffness. A complex shape was thus obtained having small sensitivity to excitation. As such, it offers a possibility for detection of all important details in clarification of excitation mechanisms and nature of natural oscillation of each structure. Modal analysis of the given housing was performed by applying the finite elements method. The linear 3D-brick finite element with 12 degrees of freedom (three translations per each node) was used. The finite elements mesh shown in Figure 1 contains a total of 6385 finite elements, 12950 nodes with 38850 degrees of freedom.

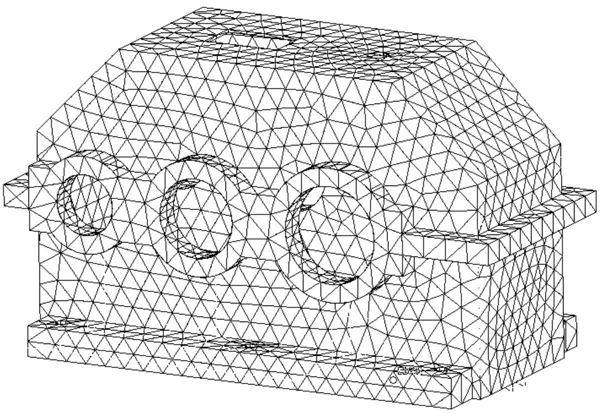


Figure 1. Discretized model of the chosen gearbox housing

The model from Figure 1 was used for calculation of 88 natural frequencies and determination of as many modal shapes of oscillation for the frequency range of 0-3000 Hz. Figures 2,3,4 and 5 show only some of the chosen shapes of oscillation for different frequencies. With increase in natural frequency, the shape of oscillation, i.e. deformations during oscillation become more complex. Distribution of stresses and strains in the housing walls at modal oscillation was determined by using the program for static in the finite elements method and by introducing displacements during oscillation. These distributions are shown in Figures 2-5. Several conclusions that characterize modal oscillation of this housing and other structures can be deduced from these presentations. The first characteristic of modal oscillation is that, at a certain modal frequency. The structure is divided into a certain number of zones which oscillate separately from one another with the same frequency. Waves propagate from

a)

b)

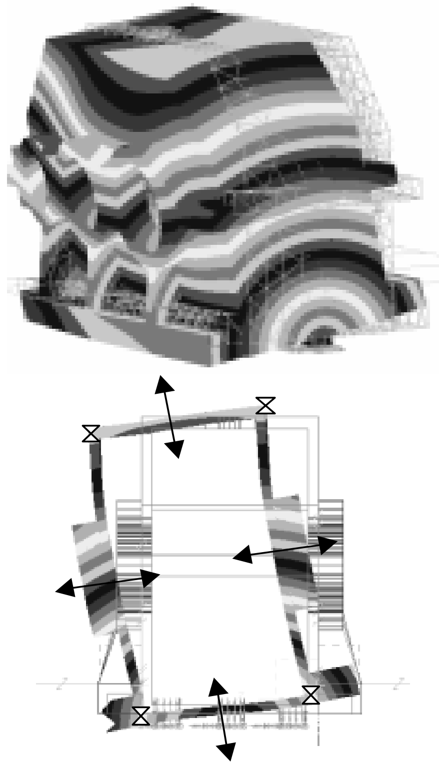


Figure. 2. Distribution of deformations in modal shape of oscillation with frequency *f* =155 Hz:
a) axonometric presentation,
b) chosen section

a)

b)

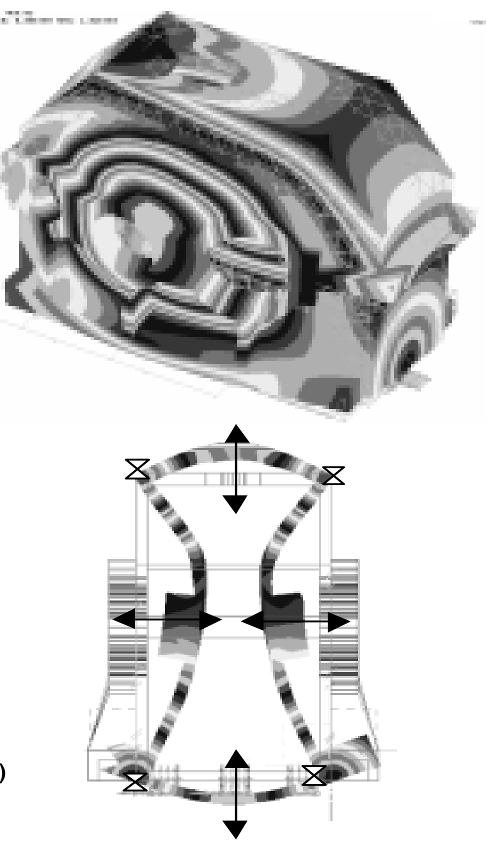


Figure 3. Distribution of deformations in modal shape of oscillation with frequency *f* =359 Hz:
a) axonometric presentation,
b) chosen section

Table 1. Natural frequencies and relative amplitude of maximal displacements for 88 modal shapes obtained by FEM

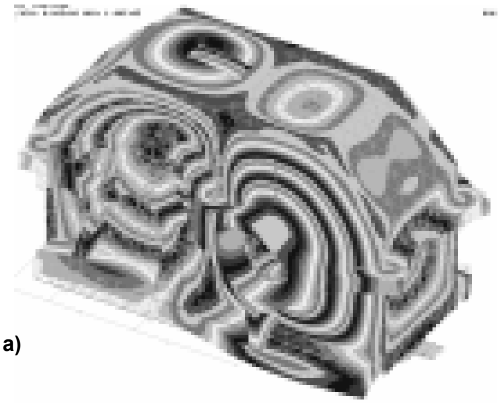
i	f_{n_i}, Hz	a_i / a_{max}	i	f_{n_i}, Hz	a_i / a_{max}	i	f_{n_i}, Hz	a_i / a_{max}	i	f_{n_i}, Hz	a_i / a_{max}
1	155	0.7464	23	1246	0,7246	45	1934	0.7246	67	2492	0.8551
2	350	0.8478	24	1287	0,7246	46	1962	0.7609	68	2504	0.7246
3	359	0.7246	25	1310	0,7536	47	2005	0.7536	69	2538	0.7391
4	460	0.7391	26	1399	0,9420	48	2009	0.7246	70	2576	0.9782
5	503	0.7464	27	1407	0.7536	49	2031	0.8768	71	2583	0.8261
6	523	0.7391	28	1412	0.7971	50	2084	0.7608	72	2599	0.8985
7	674	0.8333	29	1429	0.7246	51	2108	0.7246	73	2651	0.7246
8	693	0.7246	30	1448	0.949	52	2128	0.7681	74	2663	0.7971
9	753	0.7246	31	1512	0.7246	53	2138	0.7391	75	2666	0.8188
10	767	0.9565	32	1546	0.7391	54	2161	0.7898	76	2718	0.9275
11	819	0.7246	33	1615	0.7319	55	2170	0.9130	77	2741	0.7319
12	842	0.7246	34	1617	0.7246	56	2199	0.7319	78	2751	0.7246
13	844	0.7609	35	1639	0.7536	57	2243	0.7319	79	2780	0.7753
14	945	0.7246	36	1661	0.7319	58	2250	0.7681	80	2811	1
15	977	0.7464	37	1695	0.7319	59	2256	0.9130	81	2818	0.8696
16	1042	0.7246	38	1735	0.7319	60	2302	0.7753	82	2851	0.9783
17	1076	0.7246	39	1743	0.8188	61	2360	0.8261	83	2882	0.7391
18	1106	0.7609	40	1769	0.7391	62	2379	0.7464	84	2890	0.8768
19	1125	0.7391	41	1788	0.7319	63	2417	0.8116	85	2905	0.7319
20	1162	0.7246	42	1867	0.7536	64	2446	0.9782	86	2932	0.7681
21	1178	0.7319	43	1927	0.7971	65	2448	0.7319	87	2961	0.8841
22	1212	0.8188	44	1930	0.7246	66	2490	0.7246	88	2995	0.8623

every zone and get into "collision" at points which represent "partitions" between these zones. By analogy with the laws of physics, this oscillation represents a "stationary wave", and the partitions represent nodes of the stationary (standing) wave. The second characteristic is that the number of modal zones increases with increase of natural frequency. Their number does not depend only on frequency but on the complexity of shapes, arrangement of ribs and other reinforcements, their thickness of walls, total dimensions of the housing, shape and size of the opening, etc. The third characteristic is that the sources of waves of natural oscillation are at points of the greatest displacements, and that at points of nodal partitions (nodes) displacements are close to zero. It is opposite to stresses. Stresses are greatest at points of nodal partitions which act as clamping (constraining), and they are smallest at points of wave sources, where displacements are greatest. This analysis leads to the following conclusion. If any disturbance, for example an impact, introduces potential energy into a structure, it is released from the structure by natural oscillation. The points from which that energy is released are the points from which these waves propagate. The subject of this paper is the question which of the oscillation shapes will be excited to release that energy.

The continuation of research was marked by the excitation of natural oscillation of the structure and the analysis of response to excitation. Numerical and experimental approaches were used. In the numerical approach, the method of direct integration of the finite elements mesh with impulse excitation was used. Modal frequencies and modal shapes of oscillation obtained by means of the FEM are only the possibilities for realization of natural oscillation. Whether and how those possibilities will happen depends on a large number of conditions and influences which should be examined. Starting from the fact that modal (natural) oscillation is determined by the modal shape of deformations, modal; frequency and modal damping, the elaboration of mechanism of modal shapes excitation will be realized by the analysis of influences of exciting deformations, exciting frequencies and modal damping.

3. INFLUENCE OF EXCITING DISPLACEMENT

Deformations at modal oscillation can be in the form of pressure-tension when they are spread as longitudinal waves, then in the form of bending when they are spread by deflecting waves, in the form of shear when they are spread by deflecting waves and, at the end, torsional deformations which generate torsional waves (Fig.6). Each type of waves, i.e. deformations has its corresponding speed of propagation c_w, wave length during wave motion λ_w, waving frequency $f_w = c_w / \lambda_w$ and period of waves oscillation $T_w = \lambda_w / c_w$. In a real structure of complex shape of a machine part such as the gearbox housing and its modal shapes of oscillation, all shapes of deformations are present depending on which of them is predominant, which space for forming of the modal zone for oscillation is available, etc. These waves combine with one another. The shape of oscillation and its final frequency depend on the combination. The number of possible combinations is great and hence the number of possible shapes of oscillation is also great.



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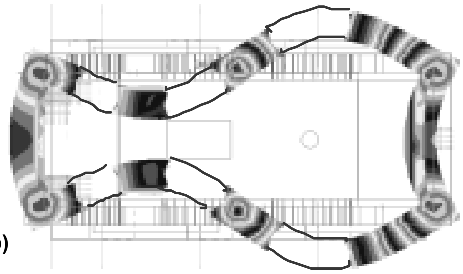
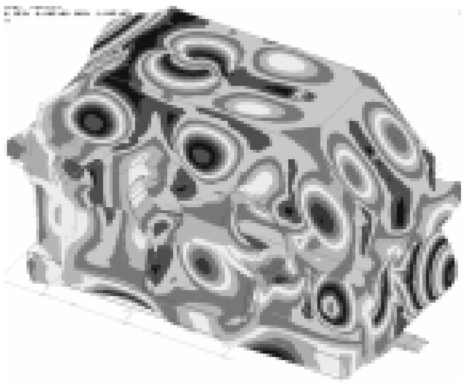


b)

Figure 4. Distribution of deformations in modal shape of oscillation with frequency *f* =693 Hz:
a) axonometric presentation,
b) chosen section



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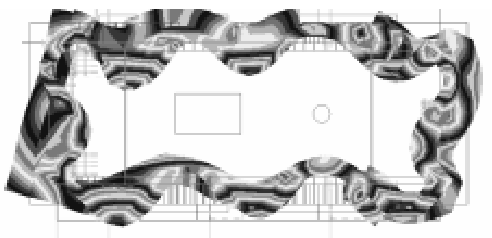


b)

Figure 5. Distribution of deformations in modal shape of oscillation with frequency *f* =2504 Hz:
a) axonometric presentation,
b) chosen section

Real modal shapes are combinations of elementary shapes of wave motion shown in Figure 6. However, in each of them the participation of some elementary waves prevails and some deformations are more prominent than the others. In order to excite a certain mode, it is necessary to realize deformations at the point and in the direction where it is greatest in that mode. The phenomenon is similar to that in a music instrument when a wire is radially pulled and let to oscillate. If Figures 2b and 3b are seen, it can be noticed that these two modal shapes can be excited by the action of impact in horizontal direction in the zone of support for bearings. In the first case (for 155 Hz), this force (displacement) acts in the sense of slanting bending (turning over of the housing). Deformations are deflecting combined with compressive and shear ones. The same force in the second case (Fig. 3b- f = 359 Hz) creates deformations of predominantly shear character combined with deflecting. In the first case, the housing volume makes one modal zone, and in the second case (359 Hz), each of the walls forms one of nodal zones. Nodal partitions are at joints of the housing walls. Frequencies are different by 2.3 times mainly only because in both cases the second type of deformations is realized, oscillation zones are reduced, wave paths are shorter, which probably influenced the increase of modal frequency. These two shapes of oscillation could be excited by the action of force at the other points where deformations are clearly seen. In Figure 2b, it would be possible for a horizontal force to act on the upper housing edge. In such a way, deformations corresponding to the frequency of 155 Hz could be removed from the equilibrium position. In the modal shape in Figure 3b, it is possible to apply a force (impact) vertically in the middle of horizontal upper plate. This wall would be thus removed from the equilibrium state and let to oscillate with the frequency of 359 Hz.

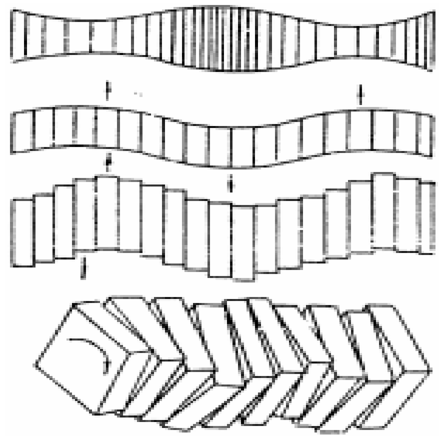


Figure 6. Main shapes of deformations and waves in wave motion in the elastic environment: a) longitudinal (compression), b) deflect, c) shearing, d) torsional

By analyzing the shapes of deformations for the frequency of 693 Hz (Fig.4) which is by further 1.9 times higher than the one in Figure 3, the following conclusions can be made. The housing mass is distributed in several modal zones which oscillate with this frequency. Almost each housing wall has two modal zones of oscillation. Modal zones are approximately twice smaller, and shear deformations are dominant even in this shape of oscillation. The points where deformations (displacements) are greatest are not in the middle of the housing any more in this shape of oscillation. It cannot be excited by the impact in the middle of the wall. Impact should be moved to a zone which is to the left or the right of the middle where displacements are greatest (Fig. 4) By this analogy, the modal shape in Figure 4 should be excited by the action of force at the other points where relatively great deformations are present. This also holds for the modal shape in Figure 5 for 2504 Hz. In this case, the division into modal zones is very complex. The number of these zones is great and they are mostly distributed among the ribs and other zones of increased rigidity. Deformations of pressure and shear deformations to some extent are prevailing. These deformations and narrowed zones of oscillation contributed to the strong increase of the frequency of this type of oscillation (3.6 times in relation to the previous one). The values of displacement are extremely reduced. Besides, it is not possible to apply the excitation force in the zone of the greatest deformations in order to incite compressive displacements. The point where shear deformations are increased should be found and then a force vertical to the housing wall should be applied at that point. This modal shape is extremely difficult to excite. On the other hand, it is very favorable circumstance for this structure. The aim is not to excite modal shapes easily but, on the contrary, not to make them appear in operation.

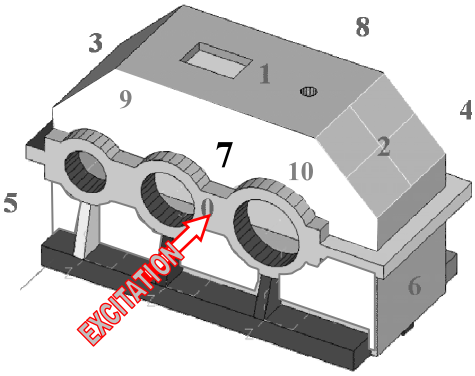
The previous analysis referred to the point and direction of action of excitation. Another question concerning the intensity of excitation follows. This energy should be at the level of energy which can be released by natural oscillation of a certain modal shape. The orientation value of the potential energy which is absorbed is

$$E_p = \frac{F\,\delta}{2} = \frac{c\,\delta^2}{2}\,. \qquad (1)$$

where F is intensity of the force applied, δ is the displacement (deformation) in the direction of the force, and c is the stiffness at the point of force action. The absorbed potential energy can also be determined as the integral of stresses within the volume covered by those stresses which have different values at each point. A more suitable approach for determination of potential energy for this purpose is by means of stiffness and displacement, which is given through the mentioned formula. The problem of the level of disturbing, i.e. potential energy, which should be introduced into the system in order to excite a certain modal shape of oscillation, is complex and must be considered together with the other excitatation parameters, such as, before all, frequency and damping.

4. INFLUENCE OF EXCITATION FREQUENCY

In order to excite modal oscillation, it is not always enough to move the system out of the equilibrium state at the point and in the direction of the greatest deformations, as it is shown in the previous section. Sometimes it is necessary to realize excitation with a frequency which is equal to the natural frequency of the modal shape that should be excited. So, the deformation or the force should be changed with the same frequency. As modal testing is usually performed in order to identify modal frequencies, they are not known in advance. Therefore, excitation is achieved with all frequencies, and the system response is realized with natural frequencies. In that way, the response of all natural frequencies corresponding to the direction of deformation in which excitation is realized is obtained. The spectrum of the force F of all frequencies is applied on the system. If the response of all possible modal shapes were the same, the spectrum of vibrations obtained would contain equal responses for each frequency. As this is not the case, the intensities of responses are very different. It is necessary to find an answer to the question why some modal shapes do not react to the incitation caused by the same frequency.



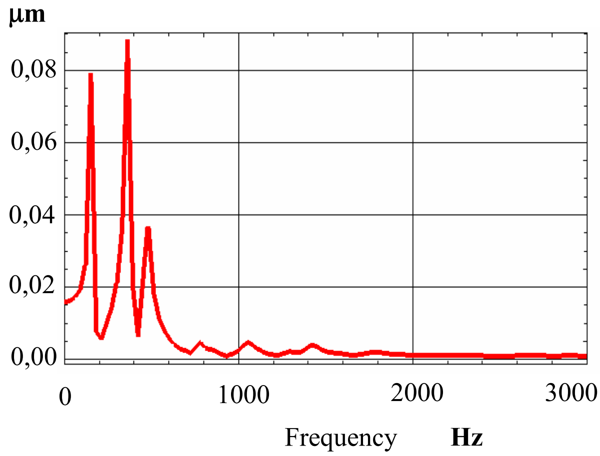


Fig. 7. Frequency response in the point No 7 (at the white side of the housing) obtained by application of the FEM and direct integration

For determination of response by direct integration of the FE structure, it is suitable to use sine excitation functions whose frequencies are equal to the frequencies of modal shapes whose response is determined. The frequency of excitation force should be varied so that it coincides with modal frequencies. Thus the response of the system for corresponding frequencies depending on the point of force action will be obtained (Fig. 7, 8). They show that there is a response to excitation only for some frequencies, i.e. only several of 88 shapes of oscillation are excited. Besides, intensities of responses are extremely different so that the effect of those extremely weak can be neglected. This phenomenom is common in comparasion with results of modal analysis performed by using the FEM and modal testing.

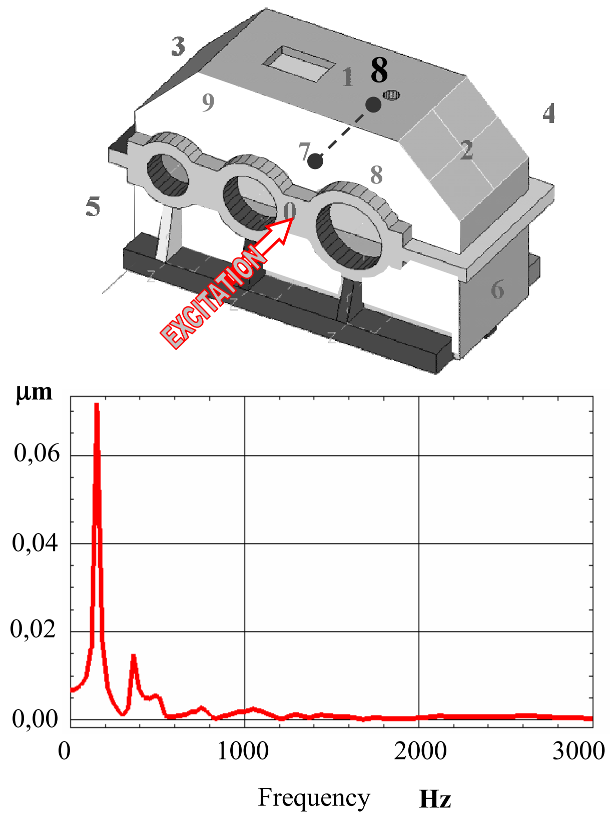


Fig. 8. Frequency response in the point No 8 (at the lateral side of the housing) obtained by application of the FEM and direct integration

If the excitation force acts at the point and in the direction of the greatest deformation and if it changes with the frequency equal to the corresponding natural frequency, it does not mean that only that modal shape will be excited. The excitation energy can be transferred to the other modal frequencies. It means that if excitation acts with a frequency which is not equal to any of natural frequencies some of them can be excited. The potential energy of excitation absorbed in the system structure is released by modal oscillation with the frequency which is most suitable for those conditions. This refers to transmitting energy from one frequency to another. During this "transferring", a considerable part of energy is "lost" so that the intensity of the response is considerably smaller in comparison with the state when the excitation frequency and natural frequency coincide. If a deformation of the value δ is realized by deformation of the housing wall with the stiffness c, and if the excitation frequency is f, the absorbed energy in unit time, i.e. the excitation power W_p.

$$W_p = \frac{c\delta^2}{2} f \,, \quad W_n = \frac{c\delta_n^2}{2} f_n \,, \text{ za } W_p = W_n \,,$$

$$\delta_n = \delta \sqrt{f / f_n} \,. \tag{2}$$

If all excitation energy in the same unit time is released by oscillations with the natural frequency f_n, where the rigidity is the same, displacement during natural oscillation would be proportionally greater or smaller proportionally to the square root of the ratio between the excitation frequency and the natural frequency. These energies are not identical. A considerable part of energy is lost to internal damping, and these losses are certainly greater during transformation of energy from one frequency to another. Besides, the absorbed energy of excitation is dispersed to several modal shapes at the same time. It is not possible to determine the ratio in which energy is distributed into several modal shapes. These distributions and the quantity of energy absorbed by one modal shape of oscillation depends on the distribution of nodal zones, distribution of stresses and deformations, value of frequencies, etc. The potential energy of deformations of each modal shape of oscillation and their ratio can be the way of establishing the coefficient of transmission of excitation energy from one frequency to another. This is certainly one of the important details in the examination of mechanisms of modal shapes of oscillation of each elastic structure and hence of gearbox housing walls.

5. INFLUENCE OF DAMPING

The frequency equation is obtained by omitting the member which refers to damping forces and the member which refers to excitation forces from the equation of dynamic equilibrium (equation of oscillation). The rest is the frequency equation which enables determination of frequencies with which the system or the structure can oscillate. A great number of natural frequencies which would be noticeable if there were no damping and if excitation were appropriate is this obtained. In reality, dispersion of energy always exists, and excitation can be realized so that only some modal shapes are excited. Dispersion of excitation energy can be the consequence of damping (conversion of energy into heat) or the consequence of certain re-distributions whose effect is a reduced response of the system, i.e. similar effect as that of damping is realized.

The mechanism of damping natural oscillation is far more complex in comparison with the complexity of shapes of deformation and with creation of natural frequencies. Modal damping is the consequence of internal resistance in the material. However, this resistance acts very differently depending on shapes of oscillation, frequency, point and way of excitation action. That is why results of measurement are different depending on how the excitation is realized and how and where the response is measured. When diagrams of the system response at a certain point are obtained for the corresponding mode n, with the frequency f_n, the non-dimensional factor of modal damping is

$$\zeta_n = \frac{\sigma}{2\pi f_n} \,. \tag{3}$$

Figure 9 gives designations of double strip width of the frequency range σ at the distance 3 dB from the extreme value of modal output. The ordinate, i.e. the output must be expressed in logarithmic units, i.e.

decibels. Thus obtained non-dimensional factor of damping can be within the limits ζ_n=0...1. When this coefficient approaches one, damping is great enough to prevent response that is modal oscillation with that frequency.

The internal damping b in the material can, in some cases, be sufficient to prevent response of a certain modal shape, and in another case not. In discrete systems, these ratios are the following:

$$\omega^2 = \frac{c}{m}, \quad 2\zeta\omega = \frac{b}{m}, \quad \zeta = \frac{b}{2\sqrt{cm}}. \tag{4}$$

Depending on the rigidity c and the mass m, the damping b can be sufficient to create ζ close to one and prevent response of the modal shape, and in other case it can be insufficient.

If the state in elastic structures is similar, the following effects are possible. Firstly, by varying the ratio of points, directions and frequencies of excitation as well as the point and direction of determination of response, the diagram shown in Figure 8 can be very different. Due to this, the non-dimensional factor of damping ζ varies. By the rule, it is greater in higher modal frequencies. If these frequencies have arisen as a consequence of greater modal rigidity, then according to the given formulae, the coefficient ζ should be smaller, and the response should have greater intensity. But, this is not the case in reality. The explanation for this illogicality can lie in the formulae (2). If the same potential energy (power) W_n is always introduced during excitation, in higher frequencies that power can result in a considerably smaller response than in a lower frequency. Thus reduced value of response in the form of the displacement δ_n , according to the diagram in Figure 9 and according to the formula (3), gives a high value of the non-dimensional factor of damping ζ. The illusion of increased damping which alleriates the response of high frequencies of oscillation is thus created. It refers to common action of damping and the "lack" of potential energy of excitation to provide enough energy for increasing the amplitude of oscillation in higher frequencies of oscillation.

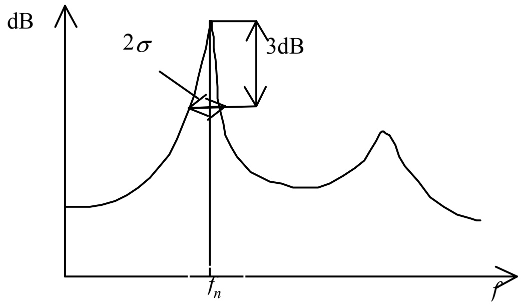


Fig. 9. Empiric determination of modal damping

In the continuation of these considerations, it should also be pointed out that, during noise generation at high frequencies, considerably smaller displacements of elementary surfaces of the housing walls are necessary, so that a certain quantity of acoustic energy could be transferred to the environment according to the same principle. From this point of view, the magnitude of amplitude of displacement during modal oscillation is relativised in relation to the frequency. The basic value for comparison must be the oscillation energy, which means the potential energy of excitation, the kinetic energy of vibrations and the acoustic energy of sound waves which are transmitted to the environment.

6. PROBABILITY OF MODAL RESPONSE

The previously performed analysis shows that modal oscillation is a form of dissipation of absorbed excitation energy. The excitation energy W_p is dissipated within the elastic structure. One part of this energy is converted into heat, and the other part is transformed into the energy of natural oscillation. Which part will be converted into heat, and which one into the energy of natural oscillation depends on a series of conditions. It is a relation between excitation and natural frequencies, a relation between natural and excitation displacements (intensity, place and direction of incitation). It also depends on damping whose effects vary depending on modal stiffness and modal masses. The number of possible combinations of these conditions is large. That fact shows that the probability of repeatability of responses at the same excitation is small. Each repeated excitation gives a response which is, in principle, different from the other responses. The response primarily differs with respect to the relation of intensities of natural oscillation for a certain natural frequency. Two main groups of reasons for stochasticity of modal responses can be mentioned. One of them covers stochasticity of combinations of the mentioned excitation conditions (intensity, place and direction of incitation, frequency, damping, etc.). The other group refers to stochasticity in distribution of excitation energy to modal shapes of oscillation (natural frequency). Excitation energy is spread by means of waves of various shapes and speeds (Picture 6). The response in the form of natural oscillation is also realized by different combinations of these waves. The combinations also have to be stochastic.

The mentioned analysis points to the fact that the most probable modal response at a certain excitation can be discussed. The total modal behaviour can be covered by the application of stochastic functions and parameters for each modal frequency and excitation.

7. CONCLUSION

Modal oscillation of gearbox housing walls and other elastic structures is very important for the noise emitted by systems into the surroundings. The mechanism of excitation of modal oscillation which gives answers to many questions that could be classified in the following groups is elaborated.

1. By modal analysis (e.g. by applying the FEM) only possible modal shapes can be obtained. In reality, conditions for these shapes excitation cannot be

fulfilled. Only several out of extremely large number of modes are usually active.
2. Excitation of the chosen modal shape consists of selection of point and direction of force action, selection of frequency of excitation and selection of damping. This mechanism of certain modal shape excitation is elaborated.
3. Modal oscillation can also be excited when not all conditions are fulfilled. Excitation energy can be transmitted from one frequency to another. Energy losses during this transmission are veritable.
4. In real conditions, the excited modal shapes are the result of combination: way of excitation action, excitation frequency, damping, transmission of exciting energy, etc. These combinations are random and modal responses are also random.

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ПОБУДА МОДАЛНИХ ВИБРАЦИЈА У ЗИДОВИМА КУЋИШТА РЕДУКТОРА

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Модалном анализом оствареном применом методе коначних елемената, по правилу се добија велики број модалних облика (фреквенција) сопственог осциловања. То су могући облици и фреквенције осциловања. У реалним условима побуђују се само неки од њих. Утврђивање начина (механизма) побуде и услова под којим ће неки модални облик бити побуђен је тежишно питање у овом раду. У циљу разраде процедуре и услова побуђивања, односно у циљу дефинисања правила побуђивања коришћени су резултати модалне анализе применом методе коначних елемената, коришћењем методе директне интеграције у оквиру МКЕ и резултата модалног испитивања. Објекат анализе и испитивања је кућиште зупчаног преносника. Модална активност зидова кућишта је у непосредној вези са структуром и интензитетом буке коју преносник емитује у околину. Стога су истраживања модалних активности од општег значаја за моделирање процеса генерисања буке машинских система.