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USING METAL FOAMS IN GEAR-DRIVES TO REDUCE THE EMMITED NOISE

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This paper shows a new approach of gear body design, with the use of a new generation technological materials, the metal foams. This paper introduces analytically the possibilities of use. The basis of this paper is the paper with the title 'Analysis of gear drives and searching of noise reduction possibilities with the help of graphs'.

1. Introduction

In the literature there are detailed descriptions about important gear body shape modifications that are important in acoustical point of view. The acoustical behaviour of gear can achieve not only by gear profile shape modification, but such changes of the gear body that modifying the emitted noise of the gear.

The generated vibration in coupling of the drive's gears, get to the walls of the drive on the primary transmission route. From this viewpoint the vibration is emitted to the environment as air noise or solid noise. If we can establish barrier against the vibration spreading on this primary transmission route, then we can reach result in environmental view.

Examples to this type of barrier can be found at KOVÁTS [1], respectively in [4] and [10]. With the use and improve of the examples in [1] and the technological development since then, a new construction solution is shown in this paper. Technological development allows the industrial usage of the metallic foams in quantity production [2], [3].

2. About metallic foams

The name metallic foam indicates such a solid metallic material, which has more than 90% porosity (some manufacturer produces 'metal foams' less than 90% porosity). The density of this kind of materials is less by one order of magnitude. Metallic foam has more properties that make its use desirable in engineering. These properties are energy-absorbing, heat conduction, damping, sound-absorbing and filtering abilities. Metallic foams have two types; the open-cell and closed-cell. Most of the cases the material of the metallic foam is aluminium-alloy but metallic foam also can be created from other materials (steel, copper, silver and titan) [2], [3].

Many researches were done to determine physical, chemical and mechanical properties of metallic foams. The properties of the metallic foams depend on the size of the cells, the thickness of the walls (bridges) between the cells and the shape of the cells if the material is the same. With the use of the modified ratio between the solid metal density and the foam metal density, we can determine approximately the foam material properties. The computational equation is the equation (1).

$$\frac{P}{P_0} = k \cdot \left(\frac{\rho}{\rho_0} \right) \tag{1}$$

where,

P: a kind of property,

ρ: density,

0: index for metals, without index 0 is for metal foams,

n, k: can be chosen according to Table 1, parameters from measurements.

Table 1

k and n factors to determine the parameters of metal foams

Property	**k**	**n**
R (Ωm)	1	–1,6 … –1,85
λ (W/mK)	1	1,6 … 1,85
E (GPa)	0,1 … 4	1,8 … 2,2
σ (MPa)	0, … 1,0	1,5 … 2

3. Designing the gear body

Figure 1 shows gear body for general application. In the figure the parts of the gear were signed for the later references.

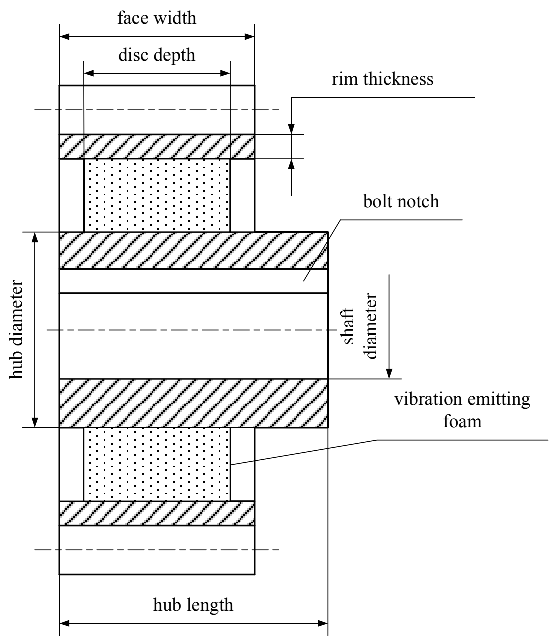

Figure 1: General shape of a gear with denominations

Radial position of the emitting material in the gear body is influenced by two factors. From the teeth of the gear to the direction of the shaft-line the first factor is the rim thickness. According to the strength calculations of gears a modify factor appears labelled with Y_B that takes the size of the gear ring into account. Y_B can be determined according to Figure 2 for external gears.

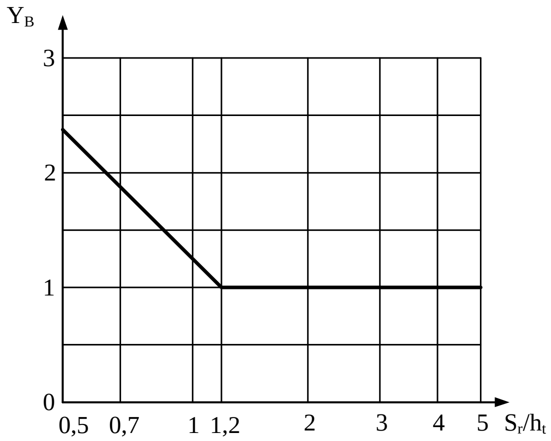

Figure 2: Rim thickness factor

For the determination the ratio between the tooth thickness and the rim thickness is needed. As a function of this ratio the value of Y_B can be chosen. In the diagram it can be very well seen when the ration reaches the value 1.2 Y_B is equal to 1 and has no effect on the root stress. Under the ratio value 1.2 Y_B has increasing effect on the root strength.

It can be determined that for the size of the vibration emitting foam in case of the input (small) gear is quite limited. So if it is possible it is strongly recommended to fill in the whole space between the hub and the rim with the metallic foam.

The lubrication of gear-drives is realised by oil in many cases. As the foam has a percolative structure it should be supervised no oil can get into the cells of the foam. In case of closed cell structure it does not mean a problem, because the surface cells are opened according to the producing. In case of open cell foams those cells that are on the outer surface should be sealed to avoid the oil getting into the gear body.

In case of the output gear a little bit different situation is given. The structure of the output gear is usually the same as the input gear. The only difference is in the radial sizes, because output gears generally have larger sizes. In case of output gear designer has the possibility to determine free the radial size of the emitting foam. As for the recommendations of the foam width a simple mechanical model and calculation is introduced in the following.

4. Mechanical model

The most important task of the vibration emitting foam to stop the vibration is realised during the connection of the gears. The cause of the vibration is the deflection that occurs under loading the tooth of the gear and later flexion that happens when the tooth quits the connection and unloaded again. As in case of the strength calculation the tooth can be modelled as a one end walled-up rod with constant intersection in this case too. At the end of the rod (top land) the deflecting tangential force is working (Figure 3).

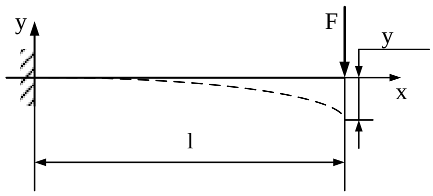

Figure 3: The model of a tooth to calculate the deflection

Deflection can be determined according to Figure 3 and equation (2).

$$y(x = l) = \frac{F \cdot l^3}{3 \cdot I \cdot E} \tag{2}$$

where:

F: deflecting force working at the end of the rod,
l: length of the rod,
I: second moment of the rod,
E: modulus of elasticity of the rod material.

Vibration arises during tooth connection is a damped vibration. As the starter amplitude of this damped vibration displacement counted by the equation (2) can be used. The frequency of the arisen vibration is the connecting vibration that can be determined according to the equation (3) when the number of teeth and revolution is known.

$$f_k = z \cdot \left(\frac{n}{60}\right). \tag{3}$$

Vibration springs while tooth connection moves on the gear body and damps, its amplitude slightly decreases. In case of solid material the decrease of the amplitude is not considerable, because of the relatively low damping factor of the average used steel material. The damping factor of the metal foam that is between the hub and the rim is much higher than the damping factor of the solid metal, so a more significant damping characterises this section. Unfortunately literature is quite reticent due to the damping factor of materials. MAKHULT [5] gives LEHR damping factors and logarithmical decrements for traditional structure materials. Literature [6] gives for damping factor of metal foams about 10 times higher values than for solid metals. On the basis of the literature [3] introduced conversion equation and the vales of the Table 1, an approximate value for the damping factor of the metal foam can be counted. Of course it cannot be forget reliable data can only be derived from measurements. Measuring of damping factors for different materials is not only complicated but expensive as well. It might be the reason for few data for the damping factor of metal foams. If the necessary data of the damped vibration managed to collect equation (1) can be written, and counted that describes the vibration [7]. Nominations of equation (4) are explained by Figure 4.

$$x = A \cdot e^{-\beta \cdot t} \sin(\omega \cdot t + \alpha)\,, \tag{4}$$

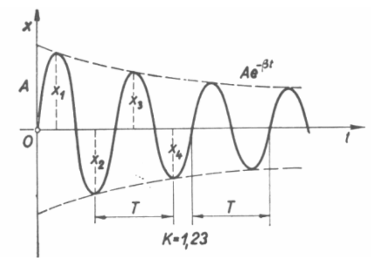

Figure 4: Figure of a damped vibration in case of K = 1.23 damping ratio [7]

where:

- A: starting amplitude,
- β: damping factor,
- e: 2.718 the basic number of natural logarithm,
- ω: pulsatance. ω = 2·π· f = (2·π)/T, where, f is the frequency, T is period time,
- α: starting phase.

As all parameters are known in equation (4) amplitude belonging to an optional t can be counted. Damping ratio signed by K (5) and its natural basic logarithm, the logarithmical decrement signed by Λ (6) that characterises damping are introduced.

$$K = \frac{x_1}{x_3} = \frac{e^{-\beta t_1}}{e^{-\beta(t_1+T)}} = e^{\beta T}\,, \tag{5}$$

$$\Lambda = \ln K = \beta T\,. \tag{6}$$

Knowing the logarithmical decrement and the frequency of the vibration damping factor (8) can be counted, using equation (6):

$$\omega = 2 \cdot \Pi \cdot f = \frac{2 \cdot \Pi}{T}\,, \tag{7}$$

$$\beta = \frac{\Lambda}{T} = \frac{\Lambda}{\frac{1}{f}} = \Lambda \cdot f\,. \tag{8}$$

Counting equation (8) the frequency and the starting amplitude of the damped vibration and the size of damping is known. After these it should be determined what kind of effect the thickness of the selected metal foam has for the amplitude of the vibration. Finding the answer easier the damping in solid material and the reflection in the border of the two media are eliminated only the amplitude reduction happens in emitting metal foam is noted.

In solid metals (c) spreading speed of longitudinal and transversal waves is different. In Table 2 wave spreading speeds for solid steel and aluminium can be seen in m/s. Spreading speed of the longitudinal wave in solid body can be counted according to equation (9).

$$c = \sqrt{\frac{E}{\rho}} \tag{9}$$

where:

E: modulus of elasticity of the material,

ρ: density.

For transversal waves there is no equation like equation (9). So the approximate spreading speed in metal foam material is determined on the basis of the longitudinal and transversal spreading speed of solid metals.

Table 2

Spreading speed of vibration in different materials

	Speed of the longitudinal wave in m/s	**Speed of the transversal wave in m/s**
steel	5100	3100
aluminium	5200	3100

Literature [8] gives a relative density range of 0.04÷0.65 for the density of metal foams that corresponds to 315÷5100 kg/m^3 'real' density. According to the introduced table the 0.3÷0.4 relative density range is the most frequent, so the average of this range (0.35) will be used in the following. In the calculations the density of the metal foam is 2750 kg/m^3. Estimating the spreading speed of the wave spreading in the metal foam elasticity modulus of the foam is also necessary. Literature [8] also shows data for modulus of elasticity with similarly wide ranges as in case of density. In case of 0.35 relative density, modulus of elasticity should be E = 5600 MPa. The spreading speed (for longitudinal waves) in metal foams can be calculated according to equation (10) following from the above mentioned.

$$c_{metalfoam,long.} = \sqrt{\frac{E}{\rho}} = \sqrt{\frac{5600 \cdot 10^6\,Pa}{2750\,\frac{kg}{m^3}}} = 1427\,\frac{m}{s}\,. \tag{10}$$

Assuming that the ratio between spreading speed of longitudinal and transversal waves for metal foams is the same as for solid metals the spreading speed of transversal wave in metal foam is 60% (856 m/s) of the value determined in equation (10). In view of spreading speed and frequency the wavelength can be determined according to equation (11).

$$\mathrm{c} = \lambda \cdot \mathrm{f} \Rightarrow \lambda = \frac{\mathrm{c}}{\mathrm{f}} \tag{11}$$

Λ value of metal foams can be forecast using equation (6) and values introduced in literature [8] of 'n' and 'k' presented in Table 1. During forecast k can be chosen from the range and for 'n' 2 is suggested by literature [8]. Knowing Λ logarithmic decrement and T period time β can be determined. Contexture introduced above could be right for the emitting factor, but unfortunately it is not. Because this way the emitting factor of the metal foam is less than the emitting factor of the solid metal. This contradicts the uncountable amount of literature that celebrates the excellent vibration emitting feature of metal foams. Modifying equation (1) equation (12) arisen.

$$\beta_{foam} = k \cdot \left(\frac{\rho_{foam}}{\rho_{solid}} \right)^n \cdot \beta_{solid} \cdot \tag{12}$$

In the characteristic curve k multiply factor changes in range of 0.1÷4 till other factors are constant. Counting the value of emitting factor for metal foams with equation (12) the

$$k \cdot \left(\frac{\rho_{foam}}{\rho_{solid}} \right)^n \tag{13}$$

multiplier is still 0.5 at the highest value (that is equal to 4) of k, so enlargement does not happen. Conclusion can be defined: for the emitting factor of metal foams equation (1) is not suitable.

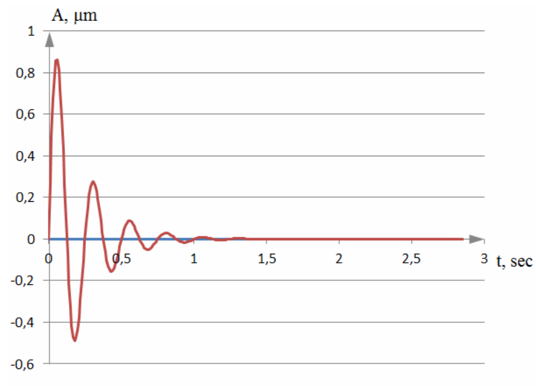

Figure 5: Damped vibration in case of high strength steel

Literature is quite reticent referring to the emitting factor of metal foams. The only source [9] for logarithmical decrement suggested a range 0.22÷0.62 for it. In Figure 5 an image of a damped vibration in case of high stress steel shows the changes of the vibration amplitude.

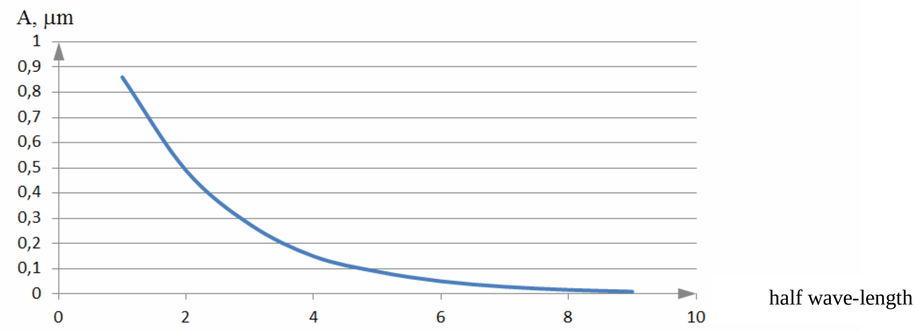

Figure 6: Characteristic of amplitude reduction depending on half wave-length

It can be seen quite well that after four wave-lengths there is almost no amplitude. In case of solid metals depending on the frequency of the vibration wave-length is in the range of 10 m [according to equation (11) and Table 2]. On the basis of the four wave-lengths the total disappear of the vibration amplitude can be learned in distance of 40 m. In Figure 6 reduction of positive-negative vibration amplitudes coming subalternating can be seen depending on the half wave-length that shows the above written amplitude reduction in a little bit more expressive way.

In case of steel metal foams taking the average (that is 0.4) of the logarithmical decrement range given by literature [9] Figure 7 introduces the image of the damped vibration.

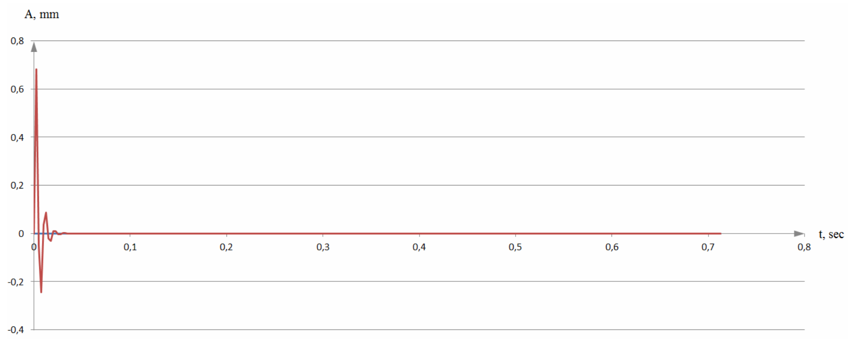

Figure 7: The image of the damped vibration in steel foam

In case of vibration can be seen in Figure 7 it can be determined that after two wave-lengths amplitude can hardly be measured. Estimated spreading speed in metal foam can be calculated according to equation (11). In this case quite smaller wave-length appears. Wave-length is in the range of metre. This is favourable in the point of view of vibration decreasing. If it is possible to use metal foam that is twice thick as the wave-length of the emerged emitted vibration in a gear-body, the amplitude of the vibration eliminates. It means still at least a 2 meter range. Changes of the amplitudes depending on the half wave-length can be seen in Figure 8. In practice there is no need for such a drastic vibration decreasing in many cases of gear-drives. There is only need for a couple of per cents, maximum 10% vibration amplitude reduction. With this kind of reduction expectations usually can be hold. On the basis of the characteristics of Figure 8 it can be seen that during one wave-length about 70% of amplitude reduction realises. Of course in the output gears of gear-drives there is not enough space for emitting material to reach the above mentioned 70% amplitude reduction.

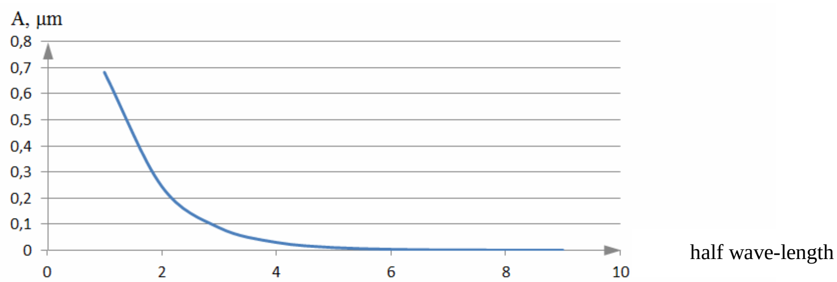

Figure 8: Changes of the amplitudes for metal foams depending on the half wave-length

However it is possible to build in metal foam with the thickness of 100 mm (that is the tenth of the wave-length from the meter range), we can calculate with a 7% reduction in the meaning of the vibration amplitude. Generally there is a space about 100 mm in output gears of gear-drives. According to the above mentioned equations suggestions can be defined for vibration emitting effect of steel metal foam as an inset, and for the foam thickness for given amplitude reduction. Of course it cannot be forget that the above mentioned are conclusions on the basis of clearly theoretical correlation that had the aim to improve if it is useful to apply this kind of material in the gear-body, in the primary transmission route. The result is absolutely positive.

For locating real processes it is necessary to experimentally determine vibration emitting values.

5. Strength analysis of gear with metal foam in it

For the strength analysis of gear body and metal foam built in it is indispensable to know the stresses work on them and strength indexes for the metal foam. In case of spur gears basic stress of gear-body is torsion that derives from the torque on the shaft. According to Figure 1 the volume between hub diameter and the rim of the gear ring can be replaced by metal foam. This volume is cylinder with a ring intersection (tube). Applying equation (14) for metal foams the emerging shearing stress can be calculated.

$$\tau = \frac{T}{K_P}, \tag{14}$$

where:

τ: shearing stress deriving from clean torque,

T: torque in the given intersection,

K_P: polar section modulus.

Polar section modulus in case of ring intersection can be calculated according to (15).

$$K_P = \frac{D^4 - d^4}{16 \cdot D}, \tag{15}$$

where:

D: the larger diameter of the ring intersection,

d: the smaller diameter of the ring intersection.

The smallest value of the inner diameter for the metal foam inset cylinder can be calculated from the stressing of the hub part of the gear. The hub part of the gear made from solid metal has also a ring intersection, so shearing stress generating here can be determined also by equation (14) and (15). Imagine that the diameter of the hub increases and the bore of the gear does not changes the stress emerging in the gear-body shows a sheer falling character on the basis of equation (16).

$$\tau_{hub} = \frac{T \cdot 16 \cdot D_{hub}}{D_{hub}^4 - d_{hub}^4}, \tag{16}$$

where:

D_{hub}: outer diameter of ring intersection hub,

d_{hub}: inner diameter of ring intersection hub.

In ring intersection that is given by the distance from the shaft, where stress emerges in the hub of the gear decreases a level where it can transmit torque, the size of the solid metal hub is determined. This value can be calculated by equation (17) solving for D_{hub}.

$$D_{hub}^{4} \cdot \tau_{hub} - T \cdot 16 \cdot D_{hub} - d_{hub}^{4} \cdot \tau_{hub} = 0 \cdot \qquad (17)$$

Determining the size of gear ring and rim Figure 2 should be followed, as the value of the tooth root stress should not increase because of the factor Y_B. During the determination of the size of gear ring and rim the larger diameter of metal foam cylinder is given. Knowing the allowed shearing stress for metal foam (τ_{foam}) the inner diameter of the metal foam cylinder can be calculated with the help of equation (18).

$$d_{foam} = \sqrt[4]{\frac{D_{foam}^{4} \cdot \tau_{foam} - T \cdot 16 \cdot D_{foam}}{\tau_{foam}}} \cdot \qquad (18)$$

If the value of d_{foam} coming from the equation (18) is greater than, or equal to D_{foam} coming from the equation (17) the metal foam can be used and can transmit the torque. In other case it cannot be applied.

When building the metal foam in the gear body an other condition also has to be realised ensuring the operation. On the border of the metal foam and solid metal a connection should be established that can transmit the drive.

6. Connecting the elements of the gear-body

Connecting the elements of the gear-body is a very important stress viewpoint. It seems to be obvious to use some kind of shaft-hub connection. There are several kinds of shaft-hub connections. Choosing the applied type it should be analysed what characters and effects the different types have for solid metal and for metal foam. In case of frictional connections friction force emerges between surfaces transmits torque. Friction force is generated by pressing the surfaces of the hub and the shaft to each other. Generating the necessary friction force generally quite a large pressing force is needed. This expects strength requirements that cannot be realised in case of metal foam.

Connecting the foam and solid metal parts an other opportunity is to create shape locking connections. These kind of shaft-hub connections are widely spread. From the existing solutions the one should be chosen that is suitable for metal foam. Keeping this viewpoint in front of the eye there are only two solutions for the task. One of them is the spline drive shaft, the other one is the polygon connection. There are pros and contras in both cases.

As spline drive shafts are quite frequent, it is easy and cheap to produce. The connecting surface sizes of the shaft splines can be determined only then when the allowed surface stress for metal foam is known. Its symmetrical shape does not cause unbalance. The edges of splines are stress concentration points is the only disadvantage of this connection.

Technical application of polygon connection is quite rare in proportion to spline drive shafts. The reason for this is the expensive production. It does not consist of sharp edges, so there are no stress concentration points. Despite of its small size it can transmit large moment.

The use of metallic glue is the third opportunity to fasten the parts of the gear-body, besides the above mentioned. The shear strength of metallic glues reaches the 175MPa, while they are stable until 80 °C. [11]. There are glues that are still stable at higher temperature, but these have shear strength that is lower one order of magnitude. Advantage of the bonding is that the glue material can easily take up between the parts of the gear. The parts of the gear-body have to be placed into a device during the hardening process of the glue. Glues are usually not expensive. Disadvantage is that many times additional operations have to be done during bonding, for example pressing the parts to each other, or using heating. The time of glue hardening changes in the function of the applied temperature and the type of the used glue. The engineer has to count with these parameters when choosing the technological process.

The combination of the above mentioned three types of connection between the parts of gear-body also can be adapted. For example frictional connection between the outer surface of inner part of gear-body (hub) and the inner surface of metallic foam and a bonded connection between metallic foam and gear rim can be used. The task of the engineer is to choose the proper connection type for the elements of the gear-body.

7. Summary

According to the above mentioned it can be determined that there is an opportunity to use metal foams in the gear-body to take advantage of the good vibration emitting properties of the metal foams. Specific constructional suggestions will be established in a further publication.

Acknowledgement

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