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TORSIONAL VIBRATION MEASUREMENT AND GEAR TRANSMISSION ERROR

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INTRODUCTION

With the ever-increasing demand for compact lightweight designs as well as high performance and speed from rotating machinery such as engines, turbines, and motors as well as transmission systems such as gear trains, transmission joints, and belts, the measurement of load capacity, rotation accuracy, transmission accuracy, vibration, noise, and efficiency has become extremely important. Rotation and transmission accuracy are extremely important factors effecting vibration and noise. The authors have developed a measurement system capable of measuring torsional vibration and transmission errors in rotating machinery with both high accuracy and response. The measurement method consists of mounting a measurement gear on the rotating shaft and detecting the rotation using a non-contact electromagnetic detector. This article will describe the principle involved and discuss measurement examples.

MEASUREMENT PRINCIPLE

The measurement principle relies on deriving pulses which is proportional to the rotational speed of the body being measured. The detector used in this method consists of, as shown in Fig. 1, a gear mounted to the rotating shaft and a non-contact electromagnetic pick up. This electromagnetic pickup creates a sinewave signal of one cycle for each tooth on the gear. The frequency of the sinewave signal is proportional to the rotation speed. As shown in Fig. 2, detectors are mounted to both ends of the shaft of the rotating machinery. In this measurement setup, as the shaft rotates, the detectors produce individual sinewave signals. The phase difference between these two signals is proportional, therefore, to the torsional angle between the two detectors. Therefore, by measuring the phase difference (T_{mi}/T_i) for each period (T_i) of the sinewave, it is possible to measure the instantaneous torsional angle.

The authors used a high-speed quartz oscillator to control the calculation of the phase difference of (T_{mi}/T_i) and developed a digital method for determining T_{mi}/T_i in real time, this information being converted finally to an analog signal for output using a D/A converter. While the above example is that of measuring the relative torsional angle between the two ends of a shaft, the method to be described below is that which measures the torsional vibration using only one end of the shaft. The method uses the above described detector mounted on only one end of the shaft of the rotating body. A PLL (phase locked loop) circuit is used to determine the average angular velocity from the sinewave signal derived from the detector. The previously described phase converter is used to determine the torsional vibration by using the phase difference between the signal corresponding to the average angular velocity and the directly measured signal.

MEASUREMENT SPECIFICATIONS

The technique discussed above enables measurements on actual machines at normal running speeds with high speed and accuracy not achievable previously.

Frequency range for phase calculation	20Hz ∿ 20kHz
60P/R detector rpm measurement range	20 ∿ 20,000rpm
" " measurement resolution	6/1000 (deg)
360P/R detector rpm measurement range	3.3 ∿ 3333rpm
" " measurement resolution	1/1000 (deg) (3.6 seconds)

MEASUREMENT EXAMPLES

Measurement of gear transmission error

The measurement of torsional vibration can be applied to enable the measurement of gear transmission error. Fig. 3 shows the measurement system setup. This example uses two optical rotary encoders as the rotating angle detectors. Fig. 4 shows the data derived from this setup.

Measurement of engine torsional vibration

This measurement example uses gears mounted to both ends of the engine crankshaft to measure the crankshaft torsional angle. Fig. 5 shows the block diagram of the measurement setup, with Fig. 6 showing actual measurement results. Fig. 7 shows a three-dimensional representation of the spectrum resulting from varying the rpm of the engine.

Transmission error of constant velocity universal joints

While it is often assumed that constant velocity universal joints exhibit no transmission errors, actual measurements will reveal such errors. The setup for the measurement of such errors is shown in Fig. 8, with actual measured results given in Fig. 9. In this example, the in-phase error refers to the condition of $\alpha_1 = \alpha_2 = 20$ (deg), while the reverse-phase error refers to the condition of $\alpha_1 = -20$ (deg) and $\alpha_2 = 20$.

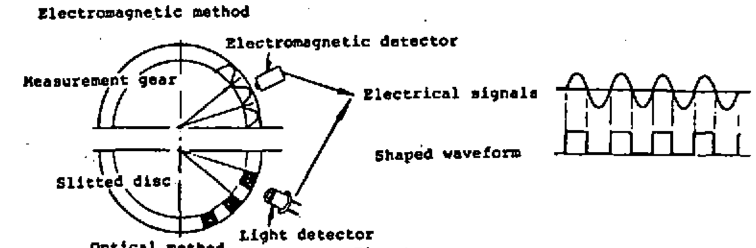


Fig. 1

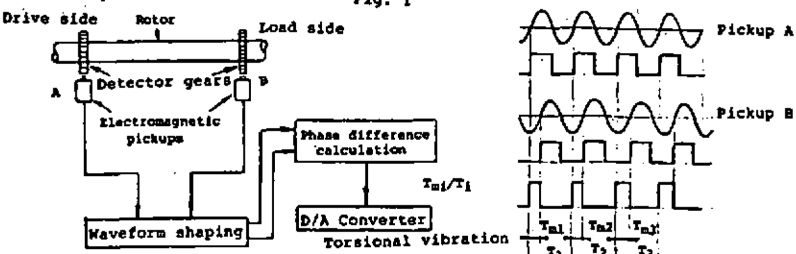
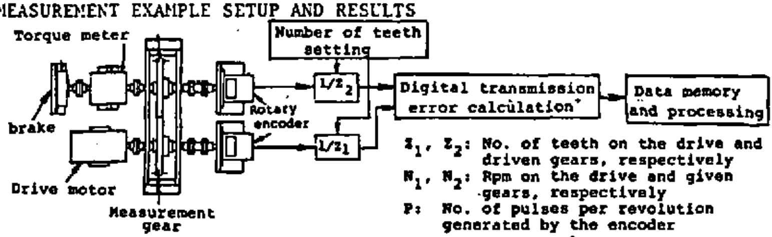


Fig. 2

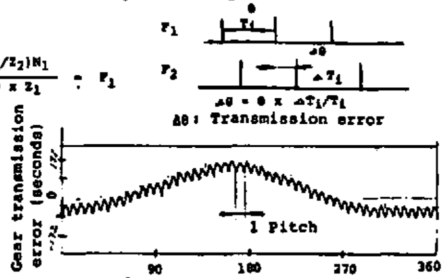
MEASUREMENT EXAMPLE SETUP AND RESULTS



Z_1, Z_2: No. of teeth on the drive and driven gears, respectively
N_1, N_2: Rpm on the drive and given gears, respectively
P: No. of pulses per revolution generated by the encoder

$$F_1 = P \times \frac{N_1}{60} \times \frac{1}{Z_2}$$

$$F_2 = P \times \frac{N_2}{60} \times \frac{1}{Z_1} = P \times \frac{(Z_1/Z_2)N_1}{60 \times Z_1} \doteq F_1$$



$\Delta\theta = \theta \times \Delta T_1/T_1$

$\Delta\theta$: Transmission error

Z_1 Gear rotational angle (deg)

Transmission error data

Measurement gear specifications

No. of teeth	$Z_1 = Z_2 = 50$
Modulus	$m = 3.5$
Pressure angle	20 (deg)
Gear width	20mm
Type	Flat tooth

Fig. 4 Measurement of gear transmission error

Measurement of engine torsional vibration

Relative torsional vibration between the two ends

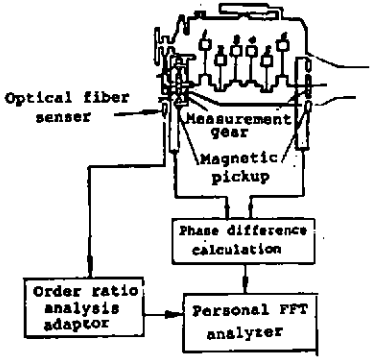


Fig. 5

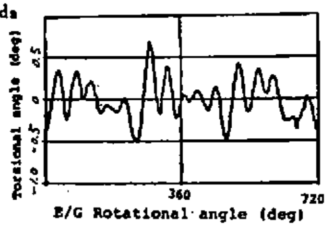


Fig. 6

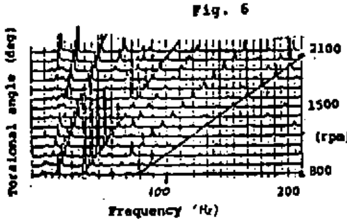


Fig. 7

Transmission error of constant velocity universal joint

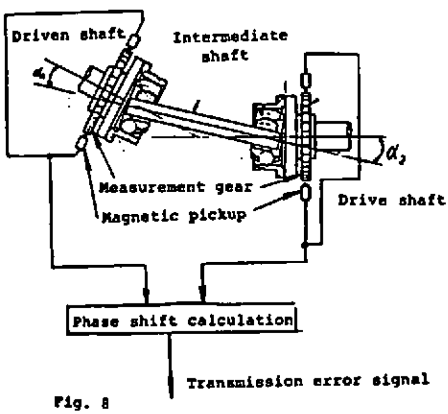


Fig. 8

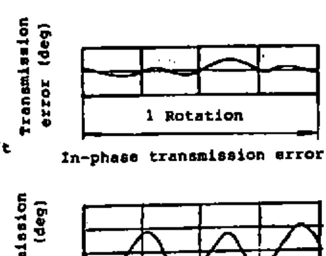


Fig. 9

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INVESTIGATION AND PREDICTION OF NOISE AND VIBRATION OF PNEUMATIC IMPACT MECHANISMS

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The problem of an improvement of the comforts when using the pneumatic impact machines requires not only a decrease of vibration and noise levels but also the objective evaluation of their effect on an operator's organism! The best result is achieved when decreasing the vibration and noise even in the source itself when the mechanisms of their origin are previously known and there is the possibility of the physico-mathematical description for the prediction of the vibroacoustical characteristics of a machine at the design stage. At the same time the mechanism of generation of oscillations both sound ones and vibration ones is inseparably linked with the dynamics of the work of the machines and mechanisms, therefore, it is necessary to jointly consider the mentioned matters, for example, by means of modelling of the work of the pneumatic impact mechanisms.
Modelling is carried out on the basis of calculation of changes of parameters of the pneumatic impact mechanism during the cycle time based on the solution of the system of differential equations. The given system includes the equations describing the changes of the thermodynamic parameters of the energy carrier in the chambers of the pneumatic impact mechanism.

$$\frac{dp}{dt}=\frac{K}{V}\left[\sqrt{\frac{2gKR}{K-1}}\sum_i\left(p_i\sqrt{T_i}\,\frac{1+\operatorname{sign}\varphi_i}{2}+p\sqrt{T}\,\frac{1-\operatorname{sign}\varphi_i}{2}\right)\varphi_i-p\frac{dV}{dt}\right] \quad (1)$$

$$\frac{dT}{dt}=\frac{T}{pV}\left\{\sqrt{\frac{2gKR}{K-1}}\sum\left[p_i\sqrt{T_i}\left(K-\frac{T}{T_i}\right)\frac{1+\operatorname{sign}\varphi_i}{2}+p\sqrt{T}(K-1)\frac{1-\operatorname{sign}\varphi_i}{2}\right]\varphi_i-(K-1)p\frac{dV}{dt}\right\} \quad (2)$$

and the equation of movement of its mechanical parts

$$\frac{d^2x}{dt^2} = \frac{1}{m}\left(\sum_{\ell} \overline{S_\ell p_\ell} + \sum F\right) \qquad (3)$$

By means of a numerical solution determined are the resultant of the forces acting on the stem of the pneumatic impact mechanism,

$$F_{st}(t) = \sum_{z} \overline{S_z p_z}(t) + \sum F(t)$$

and also the velocity of exhausted air streams

$$v^* = \frac{Q_\ell g R T}{f\, p}\left(\frac{p}{p_a}\right)^{\frac{1}{k}} \quad \text{, and} \quad v^* \leq C = \sqrt{g k R T \left(\frac{p_a}{p}\right)^{\frac{k-1}{k}}} \qquad (4)$$

and the pressure at the edge of the exhaust ports

$$p_n = p_a + \frac{Q_\ell^2\, g R T}{2 f^2 p}\left(\frac{p}{p_a}\right)^{\frac{1}{k}} \qquad \text{when } v^* < C$$

$$p_n = \frac{1}{p^{k-1}}\left(\frac{Q_\ell\, g R T}{f\, C}\right)^{k} + \frac{Q_\ell\, C}{2 f} \qquad \text{when } v^* = C \qquad (5)$$

The vibration parameters of the pneumatic impact mechanism are calculated by the resultant using the data on the dynamic characteristics of a human body and the mechanical system "stem-handle" of the pneumatic impact mechanism. Using the Fourier transform,which is obligatory when employing the dynamic characteristics as impedance curves, and the Parseval equality permits to determine the parameters of vibration of the pneumatic impact mechanism transmitting to operator's hands.

In so doing the given calculation may be performed also with due ragard for the vibration protection means used in the given pneumatic mechanism thus making it possible to determine the degree of their effectiveness. As to the noise from the exhaust of the pneumatic impact mechanism, as it is known the noise is caused by pulsations of the pressure at the edge of the exhaust ports sound power of which is

$$W = \frac{1}{p_a c_a}\,\frac{f}{T_c}\int_0^{T_c}(p_n - p_a)^2\, dt \qquad (6)$$

and by vortex formation when outflowing the air current in the form of jets emitting the sound power at every instant:

$$W_{a.c.} = N\kappa_1 \frac{\rho^2 U^{*6}}{\rho_a C_a^3} D^2 \qquad \text{when } M < 0,5$$

$$W_{a.c.} = N\kappa_2 \frac{\rho^2 U^{*8}}{\rho_a C_a^5} D^2 \qquad \text{when } M > 0,5$$

Hereat the standardized parameters of noise along with its spectrum are also found using the Fourier transform for the pressure pulsations as well as the dependency of distribution of sound power of a jet upon the Strukhal number.

Therefore the modelling of the pneumatic impact mechanism operation permits to predict even at the design stage not only its energetic characteristics but also sound and vibration ones, and that determines the choice of a scheme and gives the opportunity to project measures of noise and vibration prevention in the source of their origin and evaluate their effectiveness in situ. However not only the prediction of the vibroacoustical characteristics but also the data of the in-situ investigations do not permit correctly enough to determine the degree of their harmful influence on man by means of a simple estimation. For example, it has been ascertained that in the mining conditions in restrictions of mine workings near power sources of noise which are some kinds of pneumatic mining equipment,air oscillations can give rise to vibration of objects being met along the path of their propagation including a human body. The parameters of such vibration induced by sound waves can be determined by a contactless method taking into account the dynamic characteristics(input mechanical impedance) of the human body (or its separate parts). However the estimation of the danger of not only such vibration but also ordinary one due to direct contact, for example, with the working pneumatic mechanism taking into account an existing approach to the standardization is the rather intricate problem. The modern standardization is based on the evaluation of octave or corrected levels of root-mean-square values of vibration velocity (vibration acceleration) as the investigation of the action of vibration on a human organism was carried out with sinusoidal vibrations. At the same time in real life,in many cases,presented is impact vibration whose effect evidently is to substantially depend on not only frequency composition but also on phase shifts between frequency components, that is, on impulsiveness of vibrational process. The evaluation of the proposed parameter of the vibration velocity can be one of the ways to take account of the above mentioned:

$$L_v = \frac{L_{v\,RMS} + L_{v\,PEAK}}{2} - 1{,}5$$

It is evident that the evaluation per the given parameter takes into account an impact character of vibration and for harmonic vibrations it simply coincides with the existing one at present.

In summary, the creation of safe and even comfortable conditions of labour during operation of the pulsed equipment and the pneumatic impact mechanisms is not possible without improving the methods of investigating the sources, ways of propagation and action of vibration and noise as well as standardization of the said harmful factors.

Notations

P, T, ρ-pressure, temperature and density of air in chambers; index i relates to parameters at other end i-channel; index a - to atmosphere;

$$\varphi_i = F_i \times \begin{cases} \varphi(p, p_i) \text{ when } p_i \geq p \\ -\varphi(p_i, p) \text{ when } p_i < p \end{cases} \qquad \varphi(p, p_i) = \begin{cases} \sqrt{\left(\frac{p}{p_i}\right)^{\frac{2}{K}} - \left(\frac{p}{p_i}\right)^{\frac{K+1}{K}}} \text{ when } \frac{p}{p_i} \geq 0{,}528 \\ 0{,}259 \quad \text{when } \frac{p}{p_i} \leq 0{,}528 \end{cases}$$

F_i-sectional area of i-channel; m-mass of moveable part; $\bar{V}$-chamber volume; t-run time; K-index of air adiabat; g-free fall acceleration; R-universal gas constant; x-coordinate of moveable element of pneumatic impact mechanism; S_ℓ area of moveable element which is affected by pressure P_ℓ taking into account directions; F -forces directly not connected with action of energy carrier (for example, frictional force, weight and so on); Q_δ-momentary consumption of exhaust air; f-sectional area of exhaust ports at edge; C-sound velocity; M-Mach number; D, N-diameter and quantity of jets; T_c-cycle time; K_1 and K_2-some experimental coefficients; $L_{v\,RMS}$ and $L_{v\,PEAK}$ - corrected levels of root-mean-square and peak values of vibration velocity.