INŽENÝRSKÁ MECHANIKA 2005
NÁRODNÍ KONFERENCE
s mezinárodní účastí
Svratka, Česká republika, 9. - 12. května 2005

ANALYSIS OF VIBRATION TRANSFER FROM MOTOR TO BOTTOM GROUP OF ROTARY DRIER

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Summary: *In this paper transfer of vibration from motor to the bottom group of rotary drier is analyzed in the ADAMS/Vibration module environment. Excitation from unbalanced motor shaft is transferred through bearings mounted in end shields to transmission device and bottom plate. Output results were time-domain courses of displacement, velocity and acceleration and transfer functions, frequency response functions and modal coordinates corresponding to excitation frequency.*

1. Description of goals

The goal of this analysis was the evaluation of vibration transfer from motor to the bottom group of rotary drier. Obtained results should be used as comparative values from point of view of allowed level of vibration. To obtain physically relevant results for basic insight of its functional and vibrational properties. 3D model of drier bottom group (Fig. 1) includes bearings, transmission-device with belt and bottom plate with attachment elements.

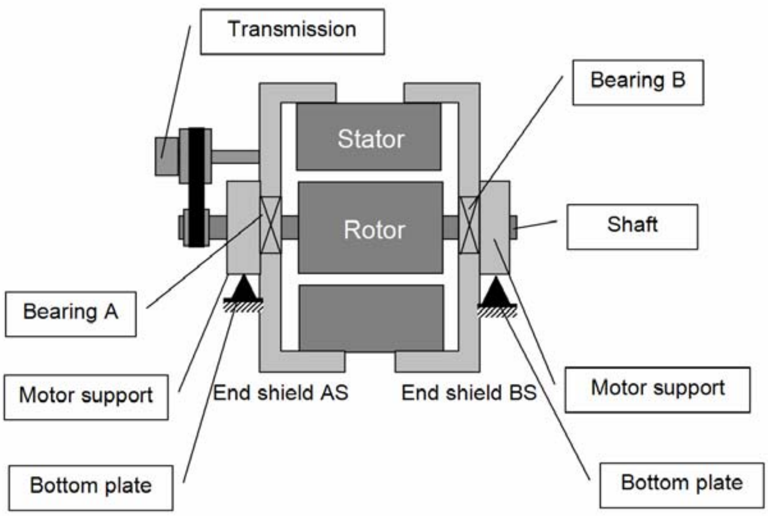

Fig. 1 Scheme of drier bottom group

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2. Task steps

1. Dynamic simulation with 3D-Submodel of drier motor including bearing, transmission-device and belt (without bottom plate). An unbalance of 200mg on rotor (rotation 2730rpm) and torsion of 100Hz, 200Hz and 300Hz was used as dynamic load. The results were time dependent quantities on the outer ring of the bearing, end shields and stator (rigid bodies).
2. Vibration of the motor was investigated using MSC.ADAMS/Vibration. Obtained results contain frequency response functions from excitation point on motor to the outer ring, end shields and stator.
3. Based on the results form the first and second step a virtual numerical 3D-Model of drier bottom group was developed. Time-domain dynamic analyses were used to verify functionality and proper behavior of model. The time dependent displacements, velocities and accelerations of the bearings outer rings, end shields and stator (rigid bodies) were obtained for simulations based on the same loading conditions as in step 1.
4. Final step was to analyze transfer of vibration form motor to defined points on bottom plate (Fig. 7). Excitation was in range from 0.1 Hz to 10000 Hz and it was caused by rotating of unbalanced mass.

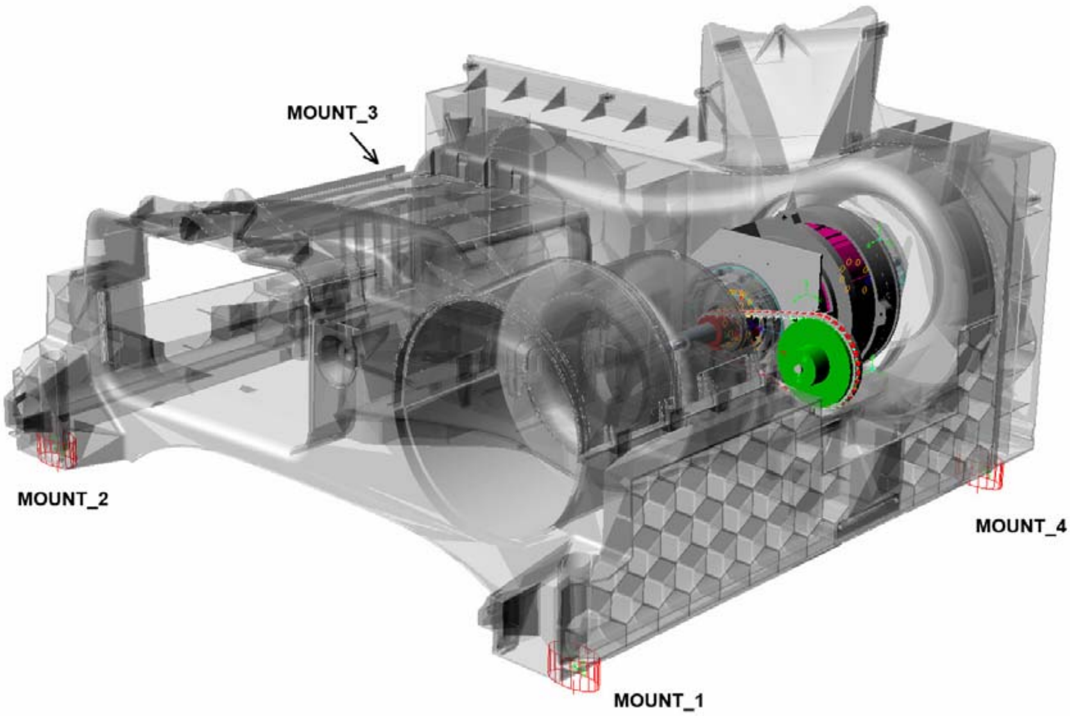

Fig. 2 3D model of drier bottom group

3. Specification of used computational technology

Vibration analysis is a frequency domain simulation of MSC.ADAMS models. This simulation can be a normal modes analysis in which the eigenvalues and mode shapes for the model are computed. The frequency domain simulation can also be a forced response analysis using the input and output channels along with the vibration actuators.

Input channels provide a port into your system so you can plot the frequency response or drive your system with an input force using a vibration actuator. You must create an input function to vibrate your system. A vibration actuator applies an input force to vibrate the system. A vibration actuator can contain expressions that let you use both time and frequency inputs. Each input channel must reference only one vibration actuator. Each vibration actuator, however, can be associated with multiple input channels.

Swept sine defines a constant amplitude sine function being applied to the model.

$$f(\omega) = F \cdot (\cos(\theta) + j \cdot \sin(\theta)) \tag{1}$$

where f is forcing function, F is the magnitude of the force and θ is the phase angle.

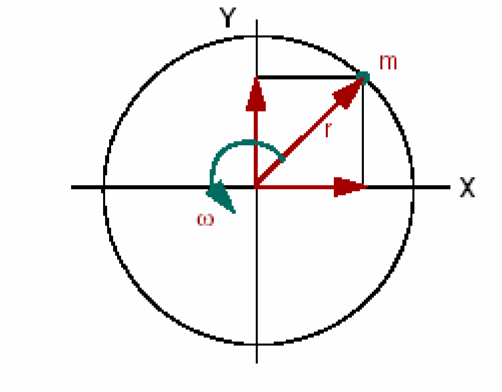

Fig. 3 Principle of leading and lagging excitations

Transfer function is the magnitude and phase response produced by a given input channel, at a given omega, for a given output channel.

For frequency response computation, the linearized model is represented as

$$\begin{aligned} x(s) &= Ax(s) + Bu(s) \\ y(s) &= Cx(s) + Du(s) \end{aligned} \tag{2}$$

where s is the Laplace variable A, B, C and D are state matrices for the linearized model.

The system transfer function can be represented as

$$H(s) = \frac{y(s)}{u(s)} = C(sI - A)^{-1}B + D \tag{3}$$

where $H(s)$ is the transfer function for the model and I is the identity matrix of dimension equal to the number of system states.

For a given vibration analysis, the system frequency response is given as

$$y(s) = H(s)u(s) \tag{4}$$

Modal coordinates are states in the frequency domain solution associated with a specific mode. Modes most active in a frequency response can be identified from the modal coordinates. The modal coordinates are computed as

$$x(s) = (sI - A)^{-1} Bu(s) \tag{5}$$

PSD of output channels for given input PSDs is given as

$$p(s) = H^*(s)U(s)H(s) \tag{6}$$

where $p(s)$ is the matrix of power spectral density, $H^*(s)$ is the complex conjugate transpose of $H(s)$ and $U(s)$ is the matrix of input spectral density.

Contact forces

To modeling contacts in our model were used 2D contacts, which include the interaction between planar geometric elements (circle and point).

$$F_n = k*(g^e) + STEP(g,0,0,d_{\max},c_{\max})*dg/dt$$

Where g represents the penetration of one geometry into another, dg/dt is the penetration velocity at the contact point, e is a positive real value denoting the force exponent, $d_{\max}$ is a positive real value specifying the boundary penetration to apply the maximum damping coefficient $c_{\max}$.

Idealized geometric constraints

To preserve guidance of belt against pulley and balls against rings in bearings there were used planar joints.

4. Results

On the Fig.4 is frequency response function and transfer function corresponding to the excitation by unbalanced mass of 200mg on rotor.

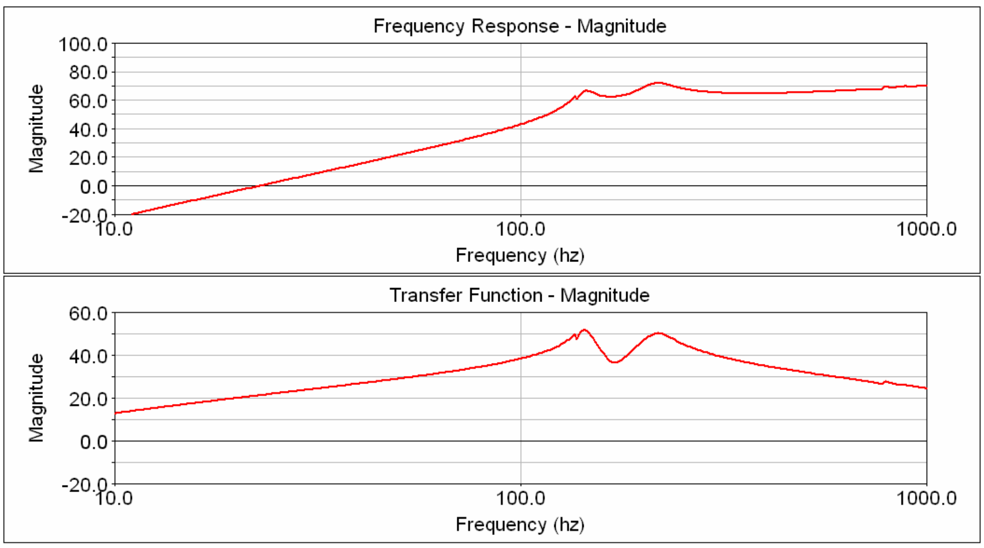

Fig. 4 Frequency response function of AS shield acceleration and transfer function of AS shield acceleration

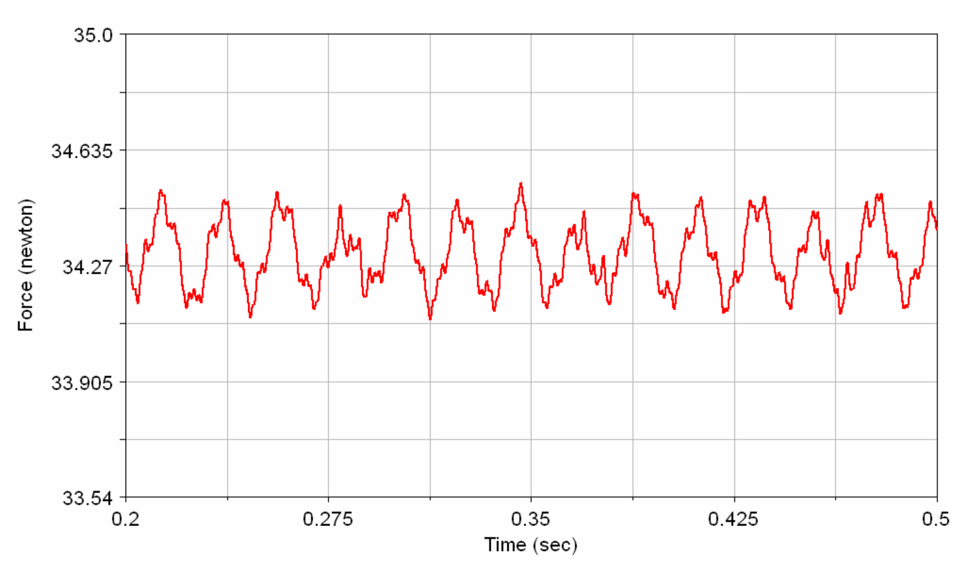

Fig. 5 Force response in attachment mount (MOUNT_1 on Fig.2) to the excitation from motor.

On Fig.5 is steady state portion of force response in attachment mount (MOUNT_1) after low pass filtering wit cut off frequency 200Hz because working range of drier is about excitation frequency 45.5Hz.

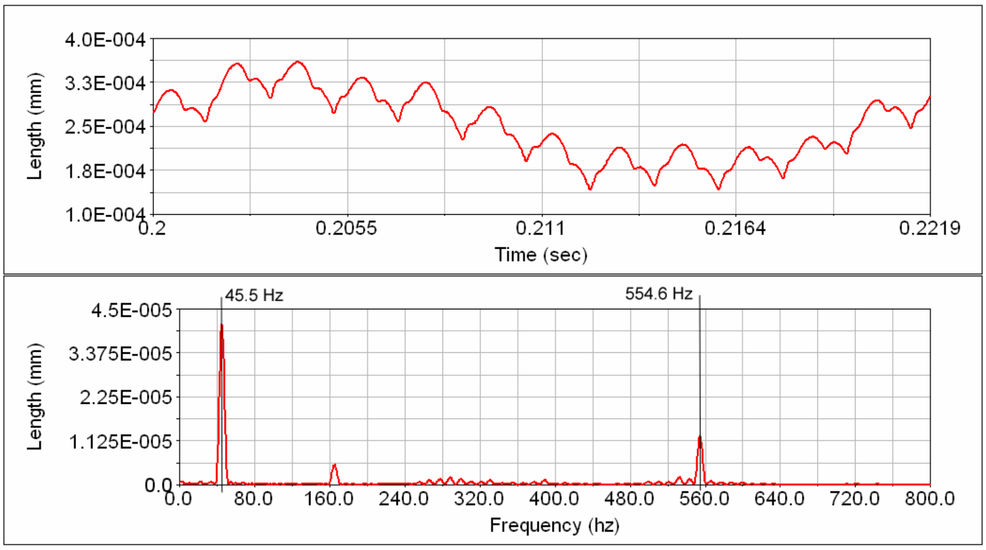

Fig. 6 FFT analysis of displacement response of bottom plate in attachment mount (MOUNT_1)

On Fig.6 time range between 0.2 and 0.2219 corresponding to one revolution of motor shaft. From results yields that lower value (45.5 Hz) corresponding to rotation of unbalance mass and higher value (554.6 Hz) is caused by excitation due to contact of belt segments with pulley.

In this section we deals with obtained frequency response functions (FRF from excitation point to the center of gravity of the outer ring, end shields, stator and bottom plate). Input point of excitation is located on the motor shaft.

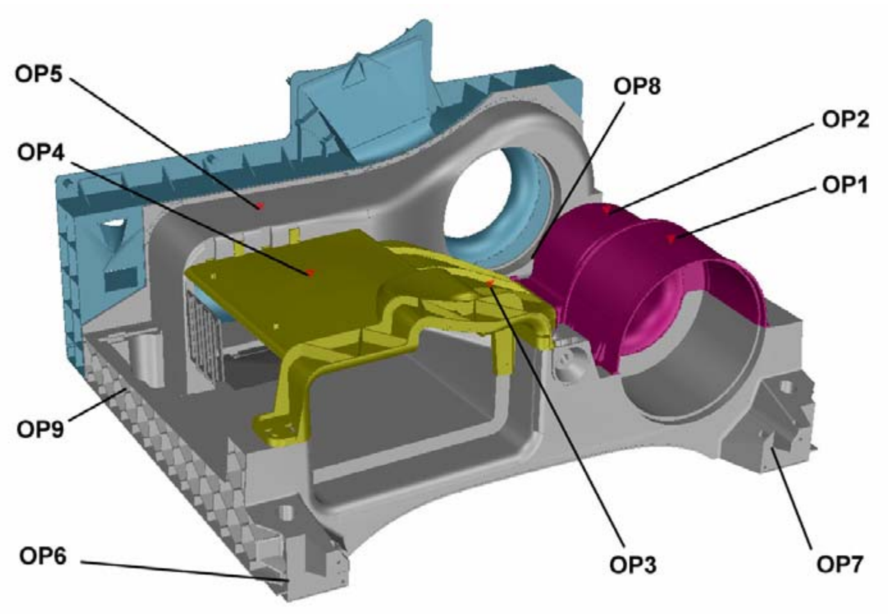

Fig. 7 Position of output points for requested vibration responses

Response outputs are denoted OP1-OP9 according to Fig.7 with defined positions of output points. Output points OP6, OP7, OP8 and OP9 are on attachment mounts of bottom plate to the ground.

On figures Fig.8 are results for acceleration of output points, because acceleration is often used as a measured quantity in real experiments. For comparison the frequency 357.73 Hz was selected, because in this value we can see amplification of responses.

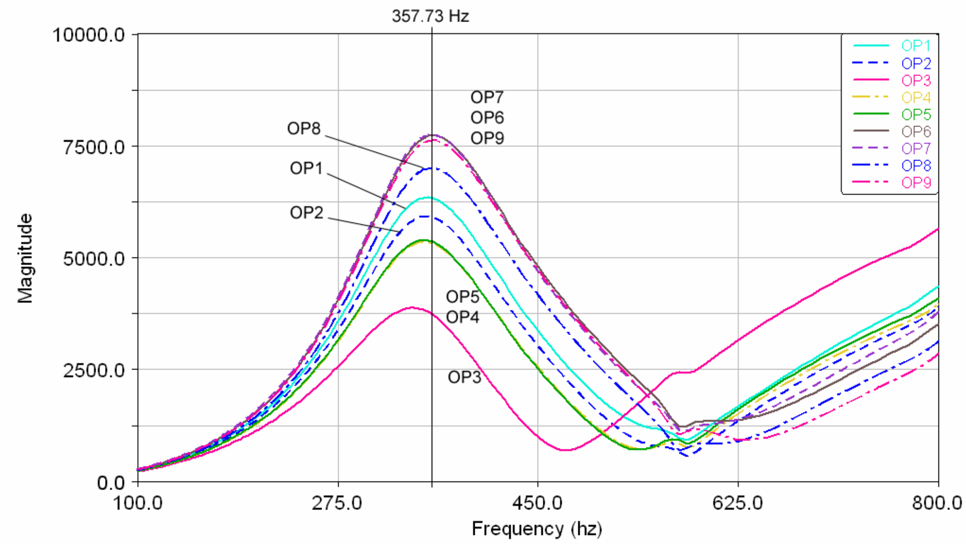

Fig. 8 Frequency response functions for accelerations of bottom plate output points (frequency and magnitude axis are in linear scale)

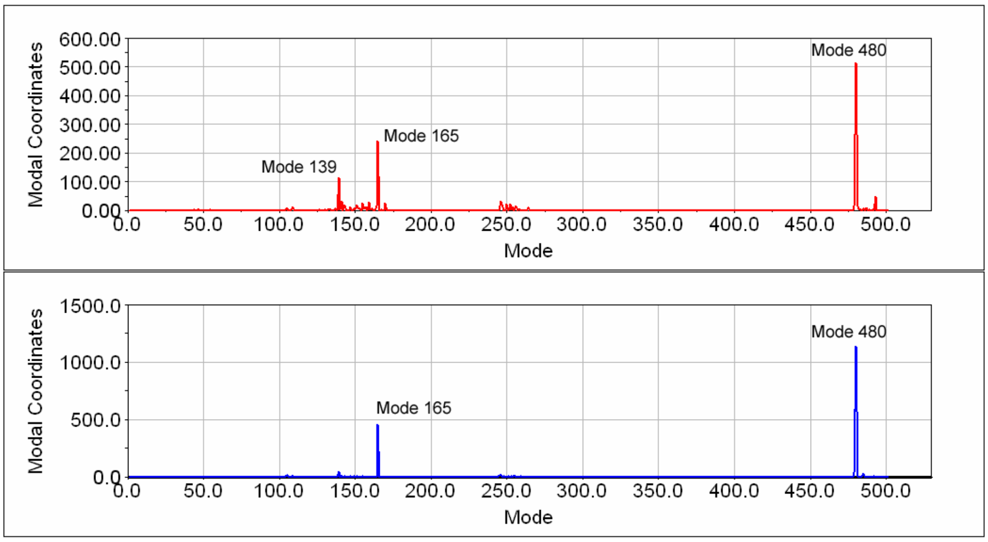

Fig. 9 Modal coordinates corresponding to excitation frequency 45.5 Hz red line for input channel in horizontal (x-direction), blue line for input channel in vertical (y-direction)

From whole spectrum we concentrate on responses for excitation frequency 45.5 Hz. On Fig.9 we see which normal modes (165, 480 in y-direction and 139, 165, 480 in x-direction) have highest modal coordinates.

5. Conclusion

All three types of obtained results (functional, time domain and frequency domain) confirmed that decision to start research of 3D model of drier bottom group was methodical right, because this access enabled better understanding of vibration transfer from motor to the bottom plate. From obtained results it can be concluded, that all results are acceptable from physical point of view.

The FFT analysis from dynamic simulation of the motor in time domain confirmed correctness of the excitation frequency 45.5 Hz corresponding to the rotation 2730rpm.

In the first dynamic time domain simulation we detected unwanted influence of belt, which is documented by FFT analysis on Fig. 6 frequency (554.6 Hz) corresponding to excitation of belt segments (for one revolution of shaft, 12.2 segments passes over shaft pulley and therefore 12.2*45.5 = 555.1).

From Fig.8 we can conclude that the highest values of acceleration are on attachments points of bottom plate to the ground (OP6, OP7, OP8 and OP9).

For documentation how is possible to obtain better insight into modal properties we prepared figures Fig.9with modal coordinates related to excitation frequency (45.5 Hz). The highest modal coordinates give us information which normal modes (165, 480 in y-direction and 139, 165, 480 in x-direction) contribute to unwanted frequency response.

As was stated, from methodical point of view the virtual model used for this research consist of rigid motor shaft and bottom plate with compliant attachments, which is initial phase of reality representation for study of vibration transfer to the bottom plate.

The main goal in next steps of research will be to achieve properties of virtual 3D-model of drier bottom group closer to the reality. Necessary refining of the virtual 3D-model of bottom group should be achieved using input data obtained by physical experiments (nonlinear characteristics of compliant attachment elements).

Further step of refining the rigid parts considered in the initial model (shaft, end shields, bottom plate) should be replaced by flexible bodies.

6. Acknowledgement

This work was supported by the Scientific Grant Agency of the Slovak Republic VEGA under the grant number 1/2092/05. This support is gratefully acknowledged.

7. References

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