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Vibration Mount Optimization of Two-Stage Screw Compressors with Harmonic Analysis

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ABSTRACT

Compressor manufacturers need analytical tools to develop reliable, highly efficient and profitable new compressors. Researching the mathematical model to predict the dynamic behavior of the machine saves considerable time in machine design and reduces the cost and costs of prototypes. In this study, a nonlinear model has been developed to examine the dynamic behavior of a two-stage screw compressor with vibration mounts. MSUP (Mode Superposition Harmonic Analysis) analysis was performed with Ansys Workbench software to solve the vibration model. The resulting vibration and displacement were experimentally compared with the vibrations of the relevant machine and a good fit was observed. These observations are presented in the study.

1. INTRODUCTION

Reducing dynamic loads and speed fluctuations in screw compressors will also reduce compressor vibration levels. Reducing vibration levels, on the other hand, reduces possible damage caused by vibration and affects compressor life in a positive way. In addition, lowering vibration levels means lowering the noise level of the machine. Many studies on compressors offer a useful approach to solving this problem. In this study, MSUP (Mode Superposition Harmonic Analysis) analysis of the double-stage screw compressor was performed with Ansys Workbench.

In order to make Ansys analysis, first of all, harmonic analyzes of compressor bodies and compressor sub-parts were examined in the literature. In addition to these processes, the body damping of the structural parts and the mathematical models that give the vibration behavior of the vibration mounts under load are examined.

Screw and motor masses, positions of vibration mounts and hardness values have a direct effect on the vibration values of the compressor system. Thanks to the harmonic analysis of the modal analysis values of the centers of mass under the natural frequencies, the vibration response speed, vibration deformation, vibration stress values that will occur during the natural frequency on the mounts have been finalized and the vibration mounts have been locally optimized. Within the framework of this optimization, it was determined with which mount hardness the amount of displacement on the mount should be met according to the position of the mount. (Bloch, Heinz P., 1933)

The noise values specified according to the ISO 2151 standard are around 80 db. For this reason, compliance with these standards is sought in Dalgakiran Compressors. One of the most important factors affecting the noise level in compressors negatively is vibration. The biggest source of this vibration in compressors is the incomplete vibration damping of the motor and screw group under dynamic operating conditions. In order to eliminate this problem, vibration analyzes and optimization studies on the mounts were made. It is aimed that this study will shed light on the design of more efficient, quieter and more reliable screw compressors in the future.

2. MATHEMATICAL MODEL

The motor, bellhousing and screw connections can be seen in figure 1. In addition, the positioning of the mounts can also be observed on this figure.

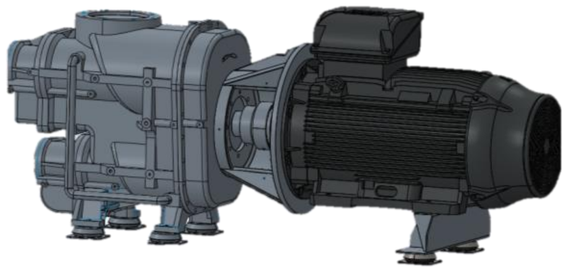

Figure 1. General view of the double-stage screw compressor

Mathematical models used in finite element programs of rotary drive systems are derived from the base vibration absorption body such as a double-stage screw compressor. The natural frequency value is the F_0 force rotating around an axis parallel to the Y axis in the XZ plane with the angular velocity value and the oscillation moment moving around the Y axis is $M_o = \sin(\omega t)$ (Kathuria et al. Modern Dictionary of Electronics, 1999).

$$F_x = F_0 \cos \omega t \tag{1}$$

$$M_x = 0 \tag{2}$$

$$F_y = 0 \tag{3}$$

$$M_y = F_0 (d_z\ cos\omega t - d_x sin\omega t) \tag{4}$$

$$F_z = F_0 \sin \omega t0 \tag{5}$$

$$M_z = 0 \tag{6}$$

The definition of moments of inertia in the X, Y, Z axis is as follows.

$$I_{xy} = I_{xz} = I_{yz} = 0 \tag{7}$$

If it is evaluated in a single plane, it can be put into the equation of motion of the supports welded from the mounts and can be edited (Blake,1996).

Equation 8 shows that the mount forces are sinusoidally balanced at x, y and z. In this equation, the symbol k denotes the damping coefficient of the mount, and a denotes the displacement.

$$\sum k_{yy} a_x = \sum k_{zz} a_x = \sum k_{yy} a_x a_z = \sum k_{zz} a_x a_y = 0 \tag{8}$$

If the sinusoidal vibration structure originating from the motor is re-evaluated over the angles of the rotational movements, the linear u, v and w values for the x, y and z axis and the angular alpha, beta and gamma values should be equal to zero. The equilibrium of these values is shown in equation 9.

$$u = v = w = \alpha = \beta = \gamma = 0 \tag{9}$$

If the vibration in one axis is evaluated, the formulation of the displacement value in the x axis is as follows;

$$\frac{x_0}{u_0} = T = \frac{1}{1 - \frac{\omega^2}{{\omega_n}^2}} \tag{10}$$

When the value of ω is equal to the operating frequency of the system, the system enters resonance and the displacement in one axis reaches infinite orders. If it is evaluated as $\omega_N = \sqrt{\left(\frac{k}{m}\right)}$ and replaced in equation 10, the frequency of the system will be as follows; (Lee, F.Y., Huang Y.C., 2012 , Paul C. Hanlon, Compressor Book)

$$f_z = \frac{1}{2\pi}\sqrt{\frac{4k_z}{m}} \tag{11}$$

3. ANSYS MODEL PREPERATION

It is very important to prepare for the harmonic analysis process with Ansys Workbench. In terms of duration, accuracy and editable quality of the analysis, it is aimed that Mesh elements can be applied more easily, especially on the geometry, without changing the center of gravity of the system. For these reasons, the very complex structures on the motor and screw were deleted and the model size was reduced.

The connections of the assembly parts with each other (called Joint and Connection) are arranged according to whichever of the connection elements required in the bolt-nut and welding connections used. These connection processes and boundaries are very important in terms of providing the natural vibration behavior of the system as close to reality and making the vibration damping at the connection points smoother.

For the sequential modeling of Static Structural>Modal Analysis>Harmonic Response analyzes called MSUP (Mode Superposition Harmonic Analysis), this assembled part was simplified in the Space Claim environment and then subjected to weight, moment loads by performing static analysis. Static analysis was carried out to define the weight pre-stress applied to the mounts by the engine and screw parts in the Ansys program. With the mode structures and vibration results arising from the geometry of the system, the static analysis has the highest effect on the mount. The loads acting on the compressor are given with the values given in Table 1. The weight of the motor and screw parts is on the Y axis (vertical axis), the motor torque acts sinusoidally due to the number of revolutions and is on the X axis. The spring coefficients of the mounts are also given as the spring coefficient of a conical mount in the X, Y and Z axes. These values are the design values of DPR-D 200 series compressors within the body of Dalgakıran compressor.

Table 1. Static analysis loads and other inputs

Engine Weight	1527 kg
Screw Weight	1365kg
Other Parts Avg. weight	200kg
Max. moment	1285 Nm
Engine operating frequency	25Hz(1500RPM)
Vib. Pads Stiffness Coeff.	3074 N/mm

Obtained static analysis results were related to modal analysis from the result section to perform pre-stressed modal analysis in Workbench environment.

With this analysis, a total of 6 frequency points were calculated, the participation factor of the system was calculated, and the frequency range and 6 entered mode numbers were confirmed. Participation Factor value is expected to be close to 100%. When this value is high, the mode number specified for Ansys is appropriate. If the participation factor value is low, more mode numbers should be calculated in the Ansys program. In table 2, participation factor values of the geometry obtained in Ansys software are seen and since these values are close to 100%, 6 mode numbers are left.

Table 2. Modal Analysis Participation Factor

X axis	%99.455
Y axis	%97.830
Z axis	%96.734
X Rotation	%99.033
Y Rotation	%96.262
Y Rotation	%98.125

Results are gathered for 4.74 Hz, 8.10 Hz, 10.65 Hz, 11.98 Hz, 16.72 Hz and 30.06 Hz natural frequencies. There are several visualized results in this case, only just one frequency visual is shown in the below for natural frequency of 4.74 Hz.

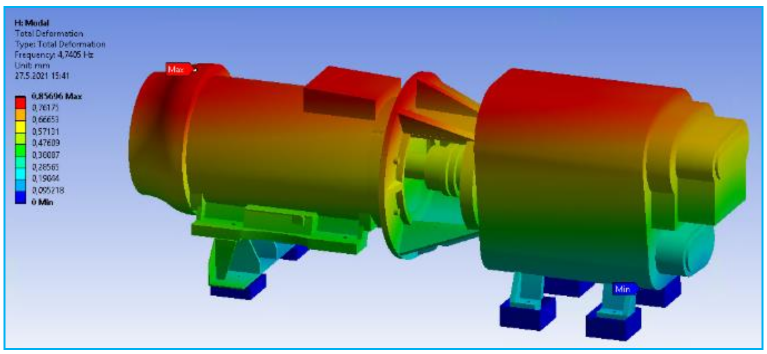

Figure 2. Mode Structure for 4.74 Hz

According to these natural frequency values, no natural frequency value causes continuous resonance for 25 Hz, which is the operating frequency of the machine. However, it will be expected that instantaneous resonance amplitudes will occur at the natural frequency values given above, between 0-1500 rpm, until the machine switches to the regime state. For this reason, the results of the modal analysis were transferred to the harmonic response analysis and the vibration amplitudes that would emerge at the natural frequency points were observed.

Geometric movements obtained as a result of modal analysis were added to harmonic response analysis and MSUP analysis was completed (Zhang, X., Gao, K., Shen, Z., & Xie, Y. 2015)

After calculation, it was concluded that the spring coefficients of the chocks on the engine side should be increased. For this reason, the analysis was repeated with the mounts with a spring coefficient of 4140 N/mm instead of the mounts with a spring coefficient of 3074 N/mm used on the engine side.

4. EXPERIMENTAL SETUP

After the design and production process, engine and screw components were placed on the chassis to measure vibration. DEWE-43A data logger was used for measuring and combined with 3 axis accelerometers. In the below measurement points are showed.

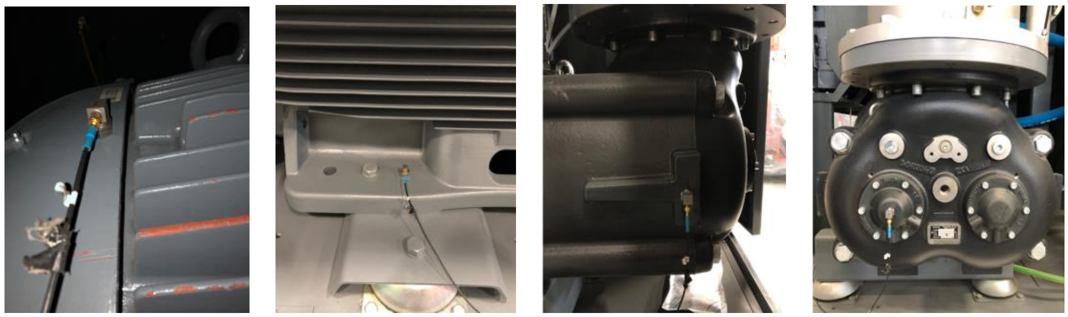

Figure 3. Measurement points

When measurements were completed, results were collected on a computer to compare with ansys simulations results.

4. RESULT AND DISCUSSIONS

2.1 ANSYS Results

When the spring coefficient of the mounts used on the engine side is used as 3074 N/mm, high vibration values are observed on the engine side. It was aimed to distribute the vibration equally to the mounts by increasing the spring coefficients of the engine-side mounts, and these mounts were reanalyzed with mounts with 4140 N/mm hardness.

According to the results obtained, the vibration intensity observed on the engine side was also distributed to other regions in the system and the vibration frequency speed values were greatly reduced.

Table 3. Frequency speeds for X, Y, Z axis with two different damping ratios with simulation.

	Initial Analysis Vibration Rate	Last Analysis Vibration Rate
X-axis vibration speed	5,78 mm/s	2,83 mm/s
Y-axis vibration speed	7,45 mm/s	3,21 mm/s
Z-axis vibration speed	6,21 mm/s	3,74 mm/s

It has been concluded that by increasing the stiffness of the motor mounts according to the given frequency speed values, the vibration intensity is spread throughout the system and vibration oscillations are provided at lower frequency speeds. At the same time, the machine was installed with the conditions given in the first analysis and the maximum vibration velocity values were recorded according to the engine speed up to 1500 rpm. With these values, the mount changes were made as indicated in the second analysis, and the maximum vibration velocity values observed according to the engine speed up to 1500 rpm are shown in Table 3.

In Table 4, damping displacements analysis results with different damping ratios are showed.

Table 4. Comparison of the damping of the two analyzes in the X, Y and Z axes

	Initial analysis damping values	Last analysis damping values
X-axis mount damping values	5,74 mm	3,79 mm
Y-axis mount damping values	0,36 mm	0,27 mm
Z-axis mount damping values	0,29 mm	0,22 mm

2.2 Test Results and Discussions

In Table 5, the test results on experimental vibration rates are shown. Comparison with three axis and two different damping ratios.

Table 5. Frequency speed for X, Y, Z axis with two different damping ratios with experiment.

	First Experimental Vibration Rate Result	Last Experimental Vibration Rate Result
X-axis vibration speed	5,51 mm/s	2,94 mm/s
Y-axis vibration speed	7,38 mm/s	3,36 mm/s
Z-axis vibration speed	6,40 mm/s	3,55 mm/s

In table 6, the test results on experimental vibration rates are shown. Comparison with three axis and two different damping ratios showed that reduces of the total displacements of the system.

Table 6. Comparison of the damping of the two experiments in the X, Y and Z axes.

	Initial experimental damping values	Last experimental damping values
X-axis mount damping values	5,62 mm	3,73 mm
Y-axis mount damping values	0,30 mm	0,19 mm
Z-axis mount damping values	0,23 mm	0,21 mm

5. CONCLUSIONS

In this study, experimental test results and Ansys simulations are compared for engine-screw couple to observe different damping ratios of the vibration mounts affects on the total vibration frequency speeds and displacements. The total displacements are reduced with vibration mounts that have higher stiffness ratios on the most vibrated side of the assembly. These analysis results were showed higher stiffness ratios on the motor side of assembly would be more effective vibration issues.

When higher damping ratios mounts applied to analysis system, analysis results have been changed to get more effective vibration results that under the vibration standard limits that a compressor manufacturers should be obey.

After the simulation when these results applied to real machine to get experimental results, same improvements were gained on the assembly system that were measured with DEWE-43A data-logger as discussed test results and discussions section.

NOMENCLATURE

M_o	Moment (Nm)
F_x	Force (N)
I_{xy}	Moment of Inertia (Nmm^4)
$k_{yy}a_x$	Stiffness Force (N)
ω_N	Natural frequency (Hz)

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