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Modelling of Active Vibration Isolation System for Air Compressor

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***Abstract*—A subject of this article is to model a vibroisolation system for a sterile air compressor. The goal is to investigate the main parameters involved in analysing the system, which consists of mounting structure, modelled as absolutely rigid body, the compressor itself, modelled by inertial mass, with disturbances in operational conditions. In the course of the study several vibroisolation techniques are investigated and a proposed strategy of using active damping mounts for the structure is given.**

Keywords—active control, modelling of vibration isolators, closed-loop feedback, air compressor.

I. Introduction

Vibrations in industrial machinery are mainly seen as problematic and not wanted. Disturbances such as oscillatory motion and repetitive shocks occurring in operational conditions in the day to day life of plant machinery are considered dangerous and harmful. It is not uncommon such phenomena to cause significant down time of industrial equipment and even injuries [1], [2], [3].

A essential engineering task is to provide different solution to reduce vibrations as much as possible.

Different types of vibro absorbers have been applied, but essentially the ones with passive control are the most commonly used [4]. The passive vibro absorbers accommodate the use of materials and physical properties to dissipate the energy that is being transferred in the system via vibrations. A simple but effective absorber is the tuned mass absorber—commonly used in different types of industrial applications [5]. The device consists of a mass, connected by a spring and/or a damper, to the primary object serving as host. The goal is that the secondary mass system is tuned specifically that the frequency that it oscillates is similar to the resonant frequency of the object it is coupled with, reducing its amplitude, shown in [6].

Most of the tuned mass damper applications are linear in nature, but heavily dependent of the object host type, some nonlinear applications are also commonly used [7].

The need of active vibro control comes from the fact that the tuned mass damper is only used in very narrow working conditions, depending on how it is set [8]. For general operations of machinery this creates a problem, since different vibrations arise from nearby equipment, or by shocks, or disturbances introduced to the system coupled with other equipment downstream. The active control damper uses different types of actuators or positioning systems together with sensors to form a closed loop system [9]. The feedback commonly used is displacement, velocity or acceleration.

A considerable amount of investigation and research has been accomplished in the field of active control, especially for these types of absorbers, since they have become wide spread due to rapid development and implementation of programmable logical controllers in industry.

Unlike the passive dampers, the active vibration systems can alter its response, depending on the input disturbance. The goal in every case of course is to keep the systems stable and stationary.

For air compressors, active systems are used in very special cases. Mostly when pulsating or heavy shaking occurs, mainly due to long pipe lines or shocks, due to sudden stops or high consumption of air, from other equipment in the process of operation.

In such cases active control finds substantial application, to make fine tuning when an installation is complete [10]. Because the coupling effect with different structures and surroundings in most cases is unknown or cannot be predicted adequately before project completion and equipment start-up.

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In this paper a sterile air compressor will be associated with a machine that uses the energy that is supplied to the main electric motor to produce an air pressure rise, via a compensation type of element, in this case a piston.

The pistons motion is back and forth, reciprocating, insider a cylinder Fig. 1.

Fig. 1. Air compressor for sterile air.

Key concept for this type of compressors is the oil-free part, associated with the series ISO 8573 international standard.

The air treatment is crucial especially for the fast-moving consumer's goods (FMCG) sector, where such compressors find wide application, especially coupled with blow molding machines for different containers and packages.

The polytetrafluoroethylene PTFE coated piston rings provide self-lubrication, without the need of oil. The expansion of PTFE is greater than metal rings, so precision is keen to prevent unwanted damage or seizing in operation.

Generally piston compressors are more affordable, have less maintenance compared to screw compressors, and can handle many on-off cycles [11].

Piston Compressors have negligible upfront maintenance, but in long-term costs are some times bigger. As parts wear, performance goes down and pressure boost cannot be achieved for prolonged periods, leading to a rebuild eventually.

Rotary Screw Compressors have higher routine maintenance costs. More service points which mean more expensive annual bills, but they avoid the gradual performance decay seen in piston models over a big operation period.

The final decision to make a good cost comparison must include the piston's eventual overhaul vs. the screw's higher yearly service [11].

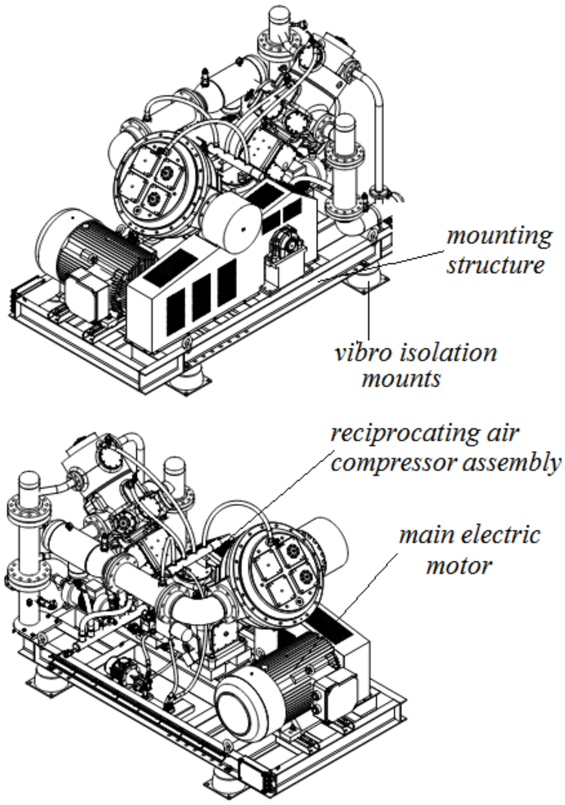

Fig. 2. Side view of SIAD "Vito" type vertical compressor.

A side view is shown in Fig. 2, only some components are highlighted. They will serve as the main points that the model will build upon.

II. Materials and Methods

A. Dynamic models of system

The modelled dynamic system is subjected to outside disturbance, mainly due to structural coupling in operational conditions.

Periodic motion is observed by the structure due to restoring forces. The forces are time dependant and tend to return the system to its original state.

The equations of motion of the free vibrations are derived using the main assumptions that the system undergoes free vibrations only under initial disturbances. The system is viscously damped and no other forces act on it, beside the spring restoring force.

Two models will be investigated, one is a single degree of freedom model (SDOF), and the the other is a two degrees of freedom model (2-DOF).

Initial parameters of the system are listed in Table 1.

TABLE 1 MODEL PARAMETERS TABLE

Initial Value Parameters			
Symbol	***Description***	***Value***	***Unit***
m_s	*Mass of structure*	*2700*	*kg*
m_a	*Mass of compressor assembly*	*2800*	*kg*
m_e	*Mass of electric motor*	*450*	*kg*
k_m	*Stiffness of vibro isolation mounts*	*1150*	*kN/m*
ζ	*Damping ratio of system*	*35*	*%*

Next, a derivation of the equation of motion of a SDOF system will be considered. The model is shown in Fig. 3.

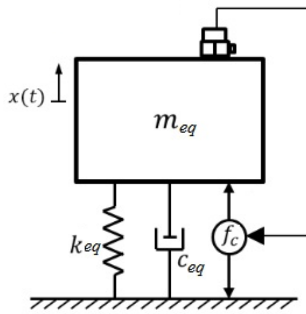

Fig. 3. Schematic of the spring-mass-damper dynamic system.

In the model shown above m_{eq} — is the equivalent mass of the compressor, structure and motor, f_c — the control force based on the controllers (Cr) feedback from the sensing element.

The stiffness k_{eq} represent the equivalent part of the spring force for all four mounts situated on each side of the structures frame. The constant of proportionality c_{eq} represent the dissipative part of the model, where over the course of time a force, proportional to velocity, forces the systems periodic motion to die down.

The equation of motion is as follows

$$m_{eq}\ddot{x} + c_{eq}\dot{x} + k_{eq}x = f_c \tag{1}$$

Rearranging gives the result

$$\ddot{x} + \frac{c_{eq}}{m_{eq}}\dot{x} + \frac{k_{eq}}{m_{eq}}x = \frac{f_c}{m_{eq}} \tag{2}$$

Where

$$\omega_n = \sqrt{\frac{k_{eq}}{m_{eq}}}, \frac{c_{eq}}{m_{eq}} = 2\zeta\omega_n, k_{eq} = k_m \tag{3}$$

Taking into account (3), the equation of motion becomes

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n{}^2x = \frac{f_c}{m_{eq}} \tag{4}$$

If the control force equates to zero (f_c=0), then the system become an ordinary passive vibro isolation system of single degree of freedom.

The control force equation is denoted by

$$f_c = -k_p x - k_d\dot{x} \tag{5}$$

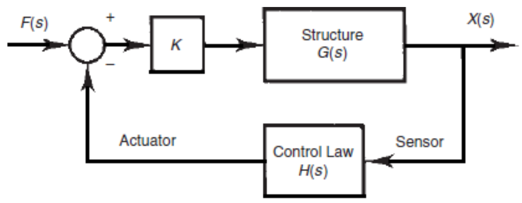

Fig. 4. Closed-loop feedback system.

When the systems is taken out of its equilibrium state from an outside force *F*, mainly due to shocks or pulses coming from other equipment, the active vibro control (AVC) reacts based on the closed-loop control illustrated in Fig.4.

An equivalent frequency domain representation is given by

$$X(s)/F(s) = KG(s)/(1 + KG(s)H(s)) \tag{6}$$

The control law measures velocity and position, also multiplies the measured values by a constant gain, respectively k_p and k_d, and the result is given by *H(s)* as

$$H(s) = k_d s + k_p \tag{7}$$

The role of the constant *K* is to multiply the steady state response of the structure *G*.

The quantity *G* relates to the ratio of the Laplace transform of the output to the Laplace transform of the input to the system. This ratio is defined as the transfer function or structure, in the vibration analysis field.

For the full state feedback of the system, considering the case shown on Fig.3, but with an additional outside disturbance, equation (6) becomes

$$\frac{X(s)}{F(s)} = \frac{K}{m_{eq}s^2 + (Kk_d + c_{eq})s + (Kk_p + k_{eq})} \tag{8}$$

The model is based on the transient vibration response, this response occurs when a system is unexpectedly brought out of rest by an unpredicted force.

The time domain equivalent of this equation is

$$m_{eq}\ddot{x} + \left(Kk_d + c_{eq}\right)\dot{x} + \left(Kk_p + k_{eq}\right)x = F \tag{9}$$

or

$$m_{eq}\ddot{x} + c_{eq}\dot{x} + k_{eq}x = F + f_c \tag{10}$$

Where

$$\omega_n = \sqrt{\frac{k_{eq} + Kk_p}{m_{eq}}}, \qquad \frac{c_{eq} + Kk_d}{m_{eq}} = 2\zeta\omega_n \tag{11}$$

The new effective natural frequency and the new effective damping ratio, depend on the gain constants, and on the passive vibro isolator properties. The force F corresponds to a brief outside action, common for such systems, mainly modelled as a unit impulse function [12].

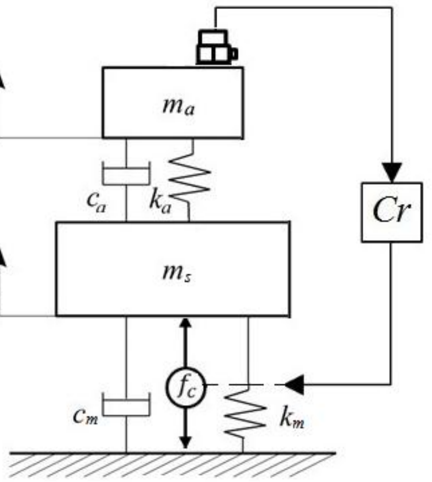

Fig. 5. Schematic of 2-DOF vibro absorber.

Considering the viscously damped 2-DOF system in Fig. 5, this describes the motion of the two masses by keeping track of the position x_a and x_s dependant on time.

The equations of motion are as follows:

$$m_a\ddot{x}_a = -c_a(\dot{x}_a - \dot{x}_s) - k_a(x_a - x_s) \tag{12}$$

$$m_s\ddot{x}_s = f_c + c_a(\dot{x}_a - \dot{x}_s) + k_a(x_a - x_s) - c_m\dot{x}_s - k_m x_s \tag{13}$$

Again the system is modelled as a classic vibro absorber when ($f_c=0$). Only when the structure is excited by some periodic disturbance force F, the controller forces the motion to degrade in time, acting as an additional dissipative element.

The normalised amplitude of the steady state response of the structure X_s/F in the time domain is given as

$$\sqrt{\frac{\left(1-\frac{r^2}{\beta^2}\right)^2+4\left(\zeta_a\frac{r}{\beta}\right)^2}{\left\{\frac{r^4}{\beta^2}-\left[\frac{4\zeta_a\zeta_s}{\beta}+\frac{1}{\beta^2}+(\mu+1)\right]r^2+1\right\}^2+4\left[r\left(\zeta_s+\frac{\zeta_a}{\beta}\right)-\frac{r^3}{\beta}\left(\zeta_a+\frac{\zeta_s}{\beta}\right)-\frac{r^3}{\beta}q\right]^2}} \tag{14}$$

Adopting the notations

$$\omega_s = \sqrt{\frac{k_s}{m_s}}, \zeta_s = \frac{c_s}{2m_s\omega_s}, \omega_a = \sqrt{\frac{k_a}{m_a}}, \zeta_a = \frac{c_a}{2m_a\omega_a}, \\ \beta = \frac{\omega_a}{\omega_s}, \mu = \frac{m_a}{m_s}, r = \frac{\omega}{\omega_s}, q = \zeta_a\mu, \tag{15}$$

The system is modelled as equally distributed because of the weight ratio, neglecting the electric motor's mass.

The model of a force actuated absorber with feedback from the main compressor assembly, with adjustable damping and/or stiffness is given in the diagram below Fig. 6.

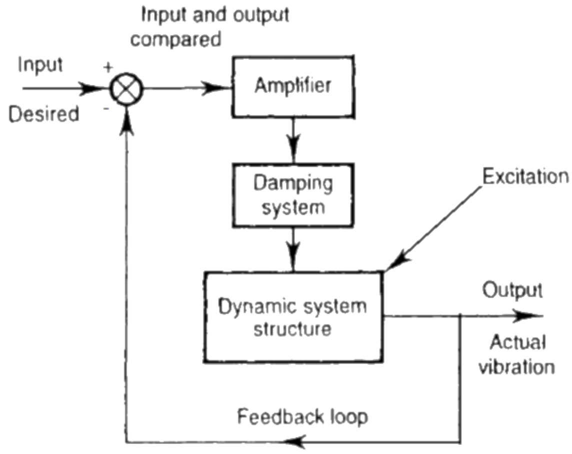

Fig. 6. Closed-loop feedback for adjustable damping in active system.

In an active system the measurements are fed back so that the parameters are continuously adjusted for optimum vibration and motion control.

In the modelled response (14) c_a and k_a can also be functions of the gains k_d and k_p. The values of these functions may change and take positive or negative form depending on the control strategy.

B. Methodology

Both models, their equations of motion and steady state response, will be investigated.

The solutions of the equation are given by specific software using a differential equation solver. Each step will be at a 10% rise from the one before, varying the gain constants.

The results will be plotted via the same software that provided the differential equation solutions — Wolfram Mathematica.

III. Results and discussion

The plots below are generated with the initial data given in Table 1. The SDOF model is presented in Fig. 7 to Fig. 11.

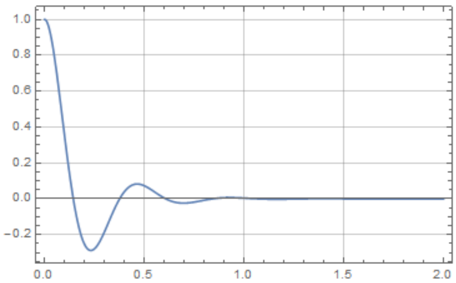

Fig. 7. System response k_p=*10%*k_{eq}, k_d=*10%*c_{eq} and $K=1$.

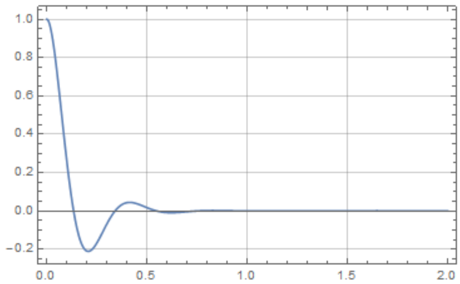

Fig. 8. System response k_p=*50%*k_{eq}, k_d=*50%*c_{eq} and $K=1$.

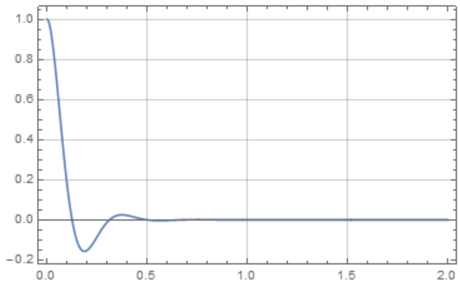

Fig. 9. System response k_p=*100%*k_{eq}, k_d=*100%*c_{eq} and $K=1$.

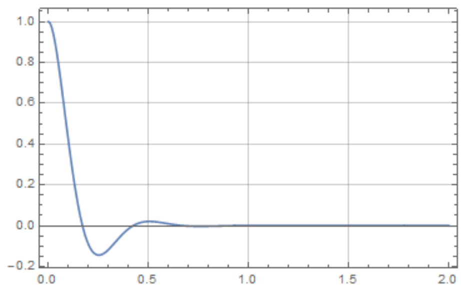

Fig. 10. System response k_p=*10%*k_{eq}, k_d=*50%*c_{eq} and $K=1$.

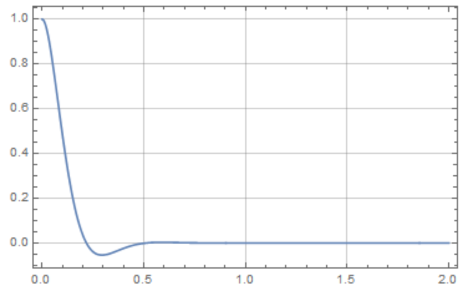

Fig. 11. System response k_p=*10%*k_{eq}, k_d=*100%*c_{eq} and $K=1$.

As seen from the plots, k_p does not present such significant difference compared to k_d.

The initial conditions of the state of the system presented above are $x(0) = 1$, and $\dot{x}(0) = 0$.

The systems is left to oscillate until the periodic motion stops.

The 2-DOF system is presented from plot in Fig. 12 to plot in Fig.14, given below.

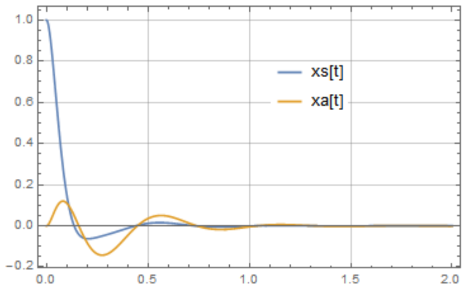

Fig. 12. System response k_p=*10%*k_m, k_d=*10%*c_m,c_a=*50%*c_m and $K=1$.

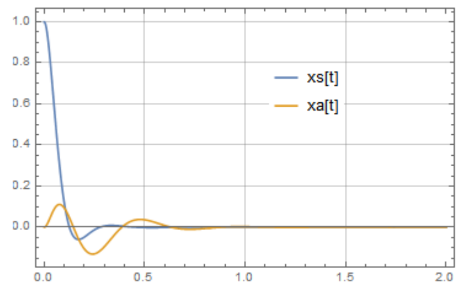

Fig. 13. System response k_p=*50%*k_m, k_d=*50%*c_m,c_a=*50%*c_m and $K=1$.

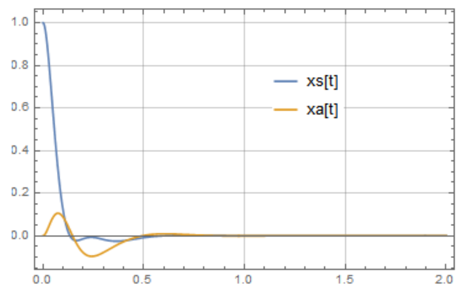

Fig. 14. System response k_p=*10%*k_m, k_d=*100%*c_m,c_a=*50%*c_m and $K=1$.

The initial conditions of the state of the system presented above are $x_s(0) = 1$, and $\dot{x}_s(0) = 0$.

IV. CONCLUSION

In conclusion the paper presents different strategies for vibration suppression using active control. Two models were investigated, one is a single degree of freedom model, the other is a two degrees of freedom model.

Constant gains were varied to see how the system reacts to changes. Based on the results the constant k_d, which in this case can be considered as virtual damping, has greater influence compared to k_p, when investigating the systems response.

The relationships established in the paper can be used for theoretical and practical analysis of such problems, providing a step by step strategy to eliminate unwanted vibrations from outside disturbances.

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