Punch Press Vibration Isolation : a Case History

J.C. Golinval, M. Géradin

UNIVERSITE DE LIEGE
Laboratoire de Techniques Aéronautiques et Spatiales
21, rue E. Solvay (C3) B-4000 Liège (BELGIUM)
E-mail : JC.Golinval@ulg.ac.be

Abstract

This paper reports the case history of the vibration isolation of a heavy punch press used to stamp metal sheets. The design of the isolation system was based on the requirement that the press should operate between 0 and 500 counts per minute (cpm) with a maximum displacement amplitude of 2 mm. Initially, a high natural frequency isolation system was proposed by the isolator supplier. It followed a nuisance due to high harmonic components produced by the punching operation of the press. The solution to this engineering problem results from an unusual application of classical isolation techniques.

1. Introduction

This paper considers the design history of a heavy punch press isolation system. The press has a mass of 34 000 kg and develops a maximum dynamic force of 137 500 N. The punching force is produced by the reciprocating motion of the tool-rod assembly connected to a counterbalanced crankshaft. The speed of operation of the press ranges from 50 counts per minute (cpm) for tool adjustments to a maximum of 500 cpm. The punching pressure capacity depends on the thickness and hardness of the metal and on the hole pattern of the sheet. Because of the mode of operation, vibration generated in punch presses presents a frequency spectrum with higher harmonics than the operational frequency. If the press is not mounted on isolators, this undesired vibration is transmitted through the foundation to the environment and may cause vibration and noise nuisances. The solution consists in the mounting of the press on a spring and damper system which can be deactivated at low operation speed.

2. Theoretical background

The function of a vibration isolation system is to reduce the magnitude of force transmitted from the machine to its foundation. The performance of an isolator is usually evaluated in terms of transmissibility of the machine-isolator system submitted to steady-state sinusoidal vibration. In the following, the absolute transmissibility of the machine-isolator system is defined as the ratio of the force amplitude transmitted to the foundation to the amplitude of the exciting force.

For a viscously damped single degree of freedom system (figure 1), the absolute transmissibility (T) is given by

$$T = \sqrt{\frac{1+\left(2\,\varepsilon\,\Omega/\omega_0\right)^2}{\left(1-\Omega^2/\omega_0^2\right)^2+\left(2\,\varepsilon\,\Omega/\omega_0\right)^2}}$$

where

T = absolute transmissibility,

ω_0 = natural frequency,

Ω = frequency of the exciting force,

ε = fraction of critical damping.

The evolution of the absolute transmissibility as a function of the frequency ratio (Ω/ω_0) is shown in figure 1 for different values of the fraction of critical damping (ε). It results from this figure that :

- the transmissibility decreases for frequency ratios higher than 1;
- when the damping increases, the transmissibility decreases at resonance but increases at higher values of the forcing frequency.

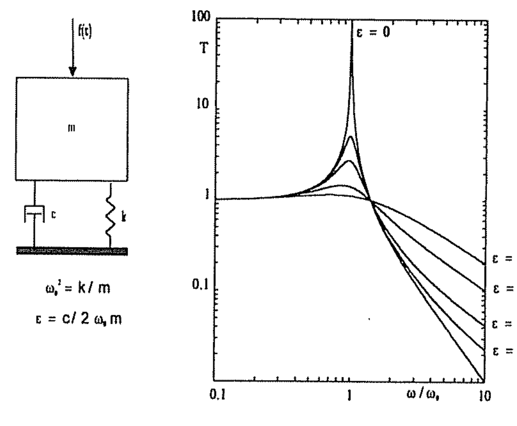


Figure 1 - Evolution of the absolute transmissibility as a function of the frequency ratio

2.1. Simple dynamic model for the press-foundation system

The press-foundation system can be modelled by the two degrees of freedom (DOF) spring-mass system represented in figure 2. The punch press is rigidly attached to a concrete inertia block which is supported by vibration isolators (k_2, c_2). The dynamic behaviour of the soil is represented by an equivalent mass (m_1), stiffness (k_1) and damping coefficient (c_1). The movement of the system can be described in terms of two independent co-ordinates q_1, q_2 such that :

- q_1 is the absolute displacement of the centre of gravity of the soil equivalent mass,
- q_2 is the absolute displacement of the centre of gravity of the press-foundation system.

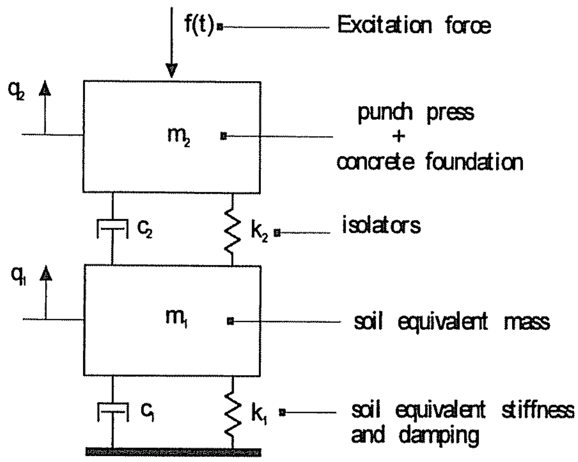


Figure 2 - Two degrees of freedom model of the punch press-foundation system

The movement equations of the system are written in the form :

$$M\,\ddot{q} + C\,\dot{q} + K\,q = F(t)$$

where

$q^T = \{q_1 \quad q_2\}$ is the displacement vector and $F^T = \{0 \quad f(t)\}$ the force vector. The mass, stiffness and damping matrices are written respectively :

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix};$$

$$K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix};$$

$$C = \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}$$

3. Initial isolation selection

The isolator selection has to be based on the following requirements :

- the press should be able to operate between 0 cpm and 500 cpm, the minimum operation speed of actually 60 cpm corresponding to the press setting speed;
- the maximum vibration amplitude of the machine should be limited to 2 mm.

As shown in figure 1, the basic principle of vibration isolation consists in selecting isolator characteristics (stiffness and damping) so that the natural frequency of the system is much smaller than the lowest forcing frequency. Figure 1 also suggests that the damping of the isolator should be made as small as possible in order to reduce the force transmitted to the foundation. The role of damping is to limit displacement amplitude at resonance. In this case however, the requirement that the press should be able to operate at a very low speed had induced the designer to make the natural frequency of the system much higher than the maximum operation speed and to choose highly damped material. For this reason, the punch press was initially mounted on isolators fabricated of elastomers. Elastomers have good characteristics regarding deflection capacity, energy storage, dissipation and space requirements. The size of the inertia block was calculated regarding the location and the space available for the press installation in order to meet the requirement on the maximum dynamic deflection allowed for the punch press.

3.1. Problem diagnosis

Measurement data were collected when the press is working at different operation speeds (240, 300, 400 and 500 cpm). These data are summarised in figure 3. They show the presence of a lot of higher harmonics than the operational frequency. The occurrence of these harmonics can be attributed to a large extent to the punch press operation at speeds over 300 cpm. The acceleration movement of the head and of the body of the machine were measured and compared in figure 4 along with the predicted values. It follows from this figure that the exciting force is sinusoidal at low operating speeds (up to 300 cpm). At higher speeds, the periodic motion of the punching head becomes non-harmonic and presents high peak values (up to $\sim 100\,m/s^2$).

As shown in figure 3, harmonics in the frequency spectrum measured in the surrounding of the press are not at all filtered but at the contrary are reinforced by the non-linear behaviour of the foundation stiffness and of the soil. The design procedure used in this case for the selection of isolators overlooked completely the basic idea of vibration isolation concept. It also appeared afterwards from the press operation that the inertia block and the elastomer isolators were installed improperly. Removable polyethylene pads were used to serve as a form for the concrete foundation. Some of these pads could not be removed from under the inertia block so that they seriously hindered the isolation system. It resulted in tremendous ambient vibrations in the vicinity of the machine. These vibrations and more particularly the large amplitude of the high frequency components also caused major acoustic nuisances in offices located outside the factory at distances up to a hundred meters.

3.2. Determination of the model parameters

On-site measurements were performed in order to determine the parameters of the 2 DOF model of the press-foundation system shown in figure 2. The parameters of the resilient members (elastomer isolators) were given by the isolator supplier. The equivalent parameters for mass, stiffness and damping of the soil were calculated using reference [2] and from speed measurement of wave propagation in the soil.

The numerical values of the model parameters are summarised in table 1.

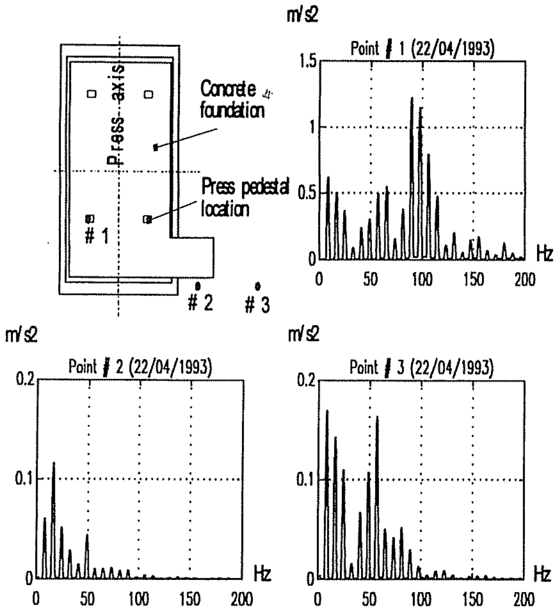


Figure 3 - Initial mounting : vibration measurements in the surrounding of the press (operation frequency = 8.33 Hz i.e. 500 cpm)

In order to validate the model, an experimental identification of the press-foundation system was performed partially using a reaction-type vibration machine as excitation. In this type of machine, the excitation force is generated by a rotating-mass unbalance. Vibration amplitudes were recorded by accelerometers placed on the concrete inertia block and on the soil. The amplitude of the frequency response function (FRF) of the press-foundation system

$$H_{22}(\omega)=\left|\omega^2 Q_2(\omega)/F(\omega)\right|$$

and the soil

$$H_{12}(\omega)=\left|\omega^2 Q_1(\omega)/F(\omega)\right|$$

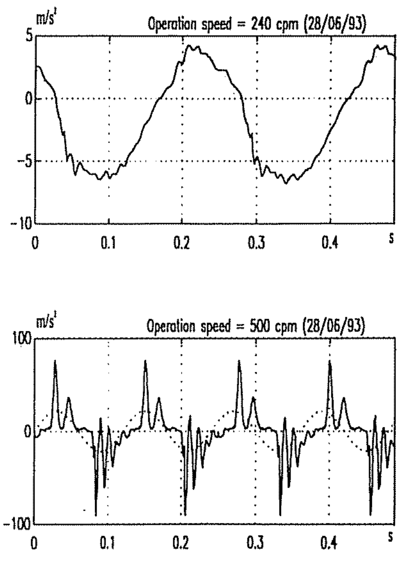
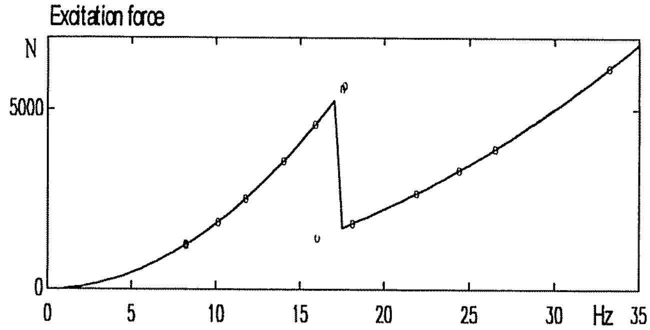


Figure 4 - Acceleration movement of the press head at 240 cpm and 500 cpm

are plotted in figure 5. However, the phase of the FRFs is not available as the force could not be directly recorded. As shown in figure 5, the comparison between calculated and measured FRFs gives satisfactory results. The 2 DOF model of figure 2 was also used to compare the predicted response of the press-foundation system with measured accelerations during press operation (figure 6).



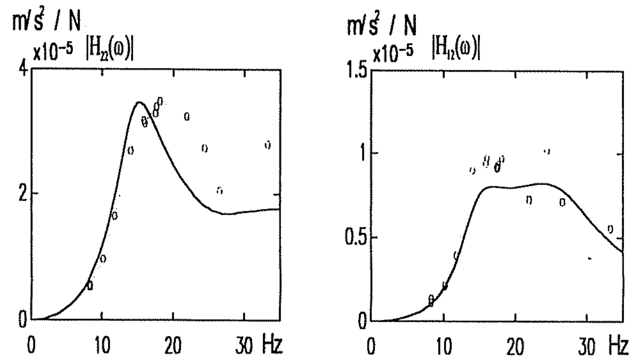


Figure 5 - Comparison of predicted and measured frequency response functions

4. Proposed solution

A new design of the isolation system has been realised so that the natural frequency of the system is much smaller than the lowest forcing frequency, which is then in contradiction with the requirement that the press should be able to operate at very low speeds. In order to apply the basic principle of vibration isolation at usual speeds and to allow the press operation at very low speeds, it was decided to select an isolation system with the following features :

- the natural frequency of the press-foundation system should be very low (typically around 1 Hz);
- the isolators would be active only for operation speeds above 200 cpm; this threshold speed was chosen regarding to the acceptable vibration level transmitted from the press to the foundation;
- the isolators could be neutralised at low operation speeds (typically under 200 cpm).

These requirements were met by selecting an air suspension system with additional rubber springs in the transverse direction in order to prevent instability. The punch press was attached on a massive inertia block designed to lower the centre of gravity of the system.

The position of the press-foundation system can be controlled by the air pressure in the air springs. For low operation speeds, the inertia block is lying on a rubber-like carpet. For operation speeds above 200 cpm, the inertia block is raised by increasing pressure in the air springs.

The numerical values of the 2 DOF model parameters corresponding to this solution are summarised in table 2. Figure 7 shows that the vibration force transmitted to the foundation above 200 cpm is drastically reduced. Frequency spectra shown in figure 8 can be compared with the corresponding frequency spectra of figure 3. In particular, the frequency spectrum measured in the surrounding of the press shows that harmonics of the operation frequency are effectively filtered by the air suspension system. In the same way, acoustic problems have been eliminated.

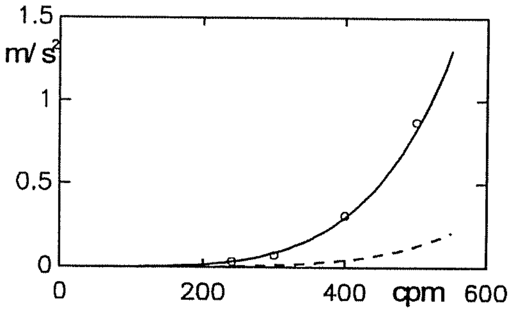


Figure 6 - Comparison of measured and predicted results during press operation

5. Conclusion

The design of a vibration isolation system should be seriously considered as early as possible in the installation of machines. In particular, the knowledge of the type of dynamic disturbance (shock or vibration) is very important for the selection of an isolator. In the present case, the selection of isolator characteristics such that the natural frequency of the system is much smaller than the lowest forcing frequency, was incompatible with the range of operation speeds of the punch press. The solution to this cumbersome problem was found in an unusual application of classical isolation techniques. A low frequency isolation system was chosen with the feature that it would be active above a threshold operational speed and that it could be neutralised at very low speed.

References

[1] Harris C. M., *Shock & Vibration Handbook* (Third Edition), Mac Graw Hill, 1988.

[2] Arya S., O'Neill M., Pincus G., *Design of Structures and Foundations for Vibrating Machines*, Gulf Company, 1978.

[3] Wolf J. P., *Foundation Vibration Analysis Using Simple Physical Models*, Prentice-Hall, Englewood Cliffs, 1978.

Soil equivalent parameters	
Mass	$m_1 = 170\,000\ kg$
Stiffness	$k_1 = 4\quad 10^9\ N/m$
Critical damping fraction	$\varepsilon_1 = 0.15$
Press-foundation system parameters	
Mass	$m_2 = 62\,000\ kg$
Isolation stiffness	$k_2 = 6.26\quad 10^8\ N/m$
Isolation critical damping fraction	$\varepsilon_2 = 0.3$

Table 1 – Identified parameters of the 2 DOF model

Soil equivalent parameters	
Mass	$m_1 = 352\,000\ kg$
Stiffness	$k_1 = 4.5 \quad 10^9\ N/m$
Critical damping fraction	$\varepsilon_1 = 0.15$
Press-foundation system parameters	
Mass	$m_2 = 100\,000\ kg$
Isolation stiffness	$k_2 = 4.974 \quad 10^6\ N/m$
Isolation critical damping fraction	$\varepsilon_2 = 0.016$

Table 2 – System parameters for the air suspension system

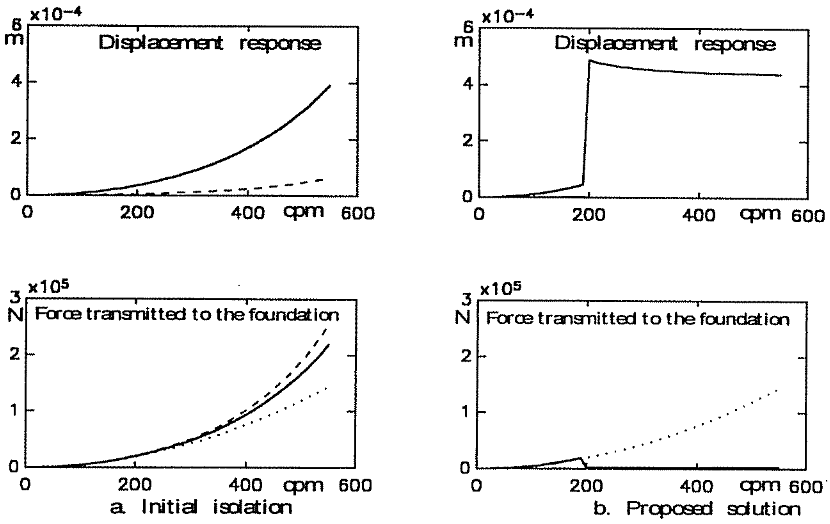


Figure 7 - Comparison of simulation results

——— *press-foundation system,* -------- *soil,* …… *excitation force*

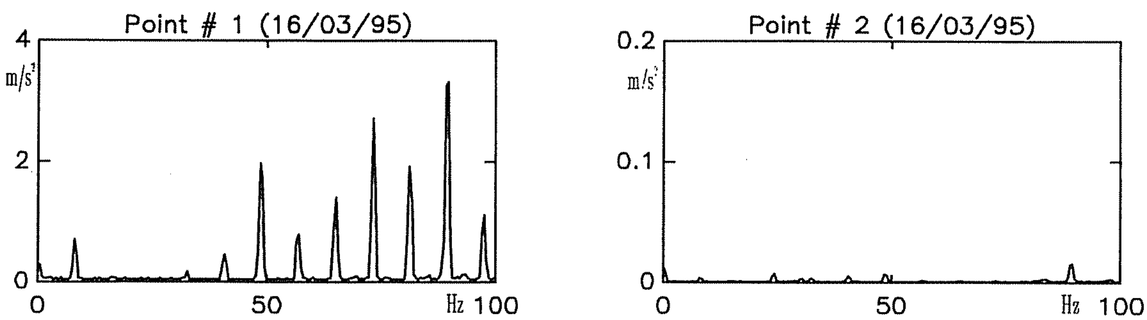


Figure 8 - Vibration measurements for the air suspension system