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INFLUENCE OF FRAHM'S DAMPING ON THE VIBRO-ACOUSTIC CHARACTERISTICS OF A ROTOR STRUCTURE

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Abstract

Influence of Frahm's damping on the vibro-acoustic characteristics of a rotor structure The present Work consists to evaluate by the method of the opposites and retiming the influence of damping of Frahm in its borderline case, on the vibroacoustic of a rotor structure. The structure is dynamically balanced by small masses fixed on the discs of the rotor. The rotor coupled with the engine is rigidly fixed at a platform. The platform is provided with a device of calibration of range dynamic absorbers. The variable exitation of the rotor is obtained through a variable speed transmission which makes it possible to have the parameters of the critical modes of the system. The external coupling is a condensation of the acoustic pressure on the structural equation of balance. It is the load which depends directly on the displacement and the elastic modes of the structure. This makes it possible to establish the matrix structure between the displacement and the acoustic excitation on one hand and on the other hand, the transfer function between the constraints and the acoustic excitation. A study and a temporal and frequential representation of the vibratory behaviour of the platform with and without the dynamic shock absorber with an experimental validation on the Eigen frequencies obtained by the finite element method. Also, the determination of the shock absorber of Frahm, starting from a given range of dampers of vibration and the evaluation of its first pulsation.

INTRODUCTION

The Frahm shock absorber is a device which is generally used to attenuate, on a determined band frequency, the vibrations of a mechanical system. It consists of a system oscillating, known as auxiliary, dissipative or not, which is associated with the main system, making it possible to increase its number of degree of freedom and then the number of resonances of the unit. The evaluation of the vibratory level provides

indications on its vibratory state, either by signals, or by descriptors whose analyses give information which takes part in a synthesis by which the expert puts a diagnosis and proposes maintenance actions [1-2]. In the case of significant unbalance, the vibratory level becomes too high.
Among of the various shock absorbers there is an elastic damper: natural rubber, synthetic rubber (chloropicrin, butane, silicone), polyurethane and polyamide. The internal dissipation of rubbers strongly depends on the temperature and the distribution of the constraints. The shock absorber with viscous coupling (Lanchester shock absorber of r)**:** inertia of the shock absorber is then coupled with the principal system only by one viscous element (dashpot) whose construction is safer but whose effectiveness evolves with the temperature, this coupling is called viscous or Coulomb coupling. The pendular shock absorber: The disturbing inertias of the reciprocating machines grow as the square of the number of revolutions, so that the amplitude of the vibrations remains constant. Therefore this shock absorber creates a restoring torque with the disturbing couple, also proportional to the square speed. As regards the Shock absorber, it consists of a mass of a free system of mass-spring, but with an amplitude within a limited distance from the vibrating structure.

FRAHM ABSORBER

When an exiting pulsation is close to resonance of the machine working at a permanent regime, it may cause a strong vibration which can be sometimes dangerous and disturbs the correct operation. It is often very difficult, and sometimes impossible, to modify the mass or the rigidity of an existing system in order to move its own pulsation. Furthermore the speed is almost always imposed and cannot be modified in order to move away the frequency of the excitation from the resonance frequency. There exit machines that run within a large range of speed and whose resonances are unacceptable. One can avoid these delicate situations when installing dynamic shock absorber or Frahm absorber, which is the aim of this study. The shock absorber associates to the system mass-arises (m_1, k_1) which schematizes the rotating machine, a second system mass-arises (m_2, k_2) [3].

CONSERVATIVE MODEL WITH TWO DEGREES OF FREEDOM

This model is a system with two degrees of freedom without damping, constituted of a spring of rigidity (k_1) and a mass (m_1), named vibrant system, and an auxiliary system of a mass $(m_2)$and a spring of rigidity (k_2).
F cosωt : harmonic force excitation of the structure.
$x_1(t)$: system displacement (m_1, k_1).
$x_2(t)$: system displacement (m_2, k_2).
when applying the Lagrangian formulation, we obtain the system (9.1) :

$$\begin{cases} m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2 x_2 = F\cos\omega t \\ m_2 x_2 + k_2(x_2 - x_1) = 0 \end{cases} \tag{9.1}$$

The displacements x_1 and x_2 in permanent regime are harmonic functions with same pulsation ω as the excitation force. We look for displacements with complex form $\underline{x_1}$ and $\underline{x_2}$, where X_1 and X_2 are the real parts.

$$\begin{cases} \underline{x_1} = X_1 e^{j(\omega t-\varphi_1)} = X_1 e^{-j\varphi_1}.e^{j\omega t} = \underline{A_1} e^{j\omega t} \\ \underline{x_2} = X_2 e^{j(\omega t-\varphi_1)} = X_2 e^{-j\varphi_1}.e^{j\omega t} = \underline{A_2} e^{j\omega t} \end{cases} \tag{9.2}$$

In the same way the external force is the real part of the complex force $\underline{f} = Fe^{j\omega t}$. Equations (9.1) become :

$$\begin{cases} \underline{A_1}(-\omega^2 m_1 + k_1 + k_2) - \underline{A_2} k_2 = F \\ -A_1 k_2 + \underline{A_2}(-\omega^2 m_2 + k_2) = 0 \end{cases}$$

From these equations, we evaluate complex numbers $\underline{A_1}$ and $\underline{A_2}$, and hence amplitudes X_1 et X_2.

From equation (9.2), complex modulus $\underline{A_1}$ is equal to the amplitude X_1. We deduce the force F:

$$F = \sqrt{\frac{X_1^2((k_1-\omega^2 m_1)(k_2-\omega_2)-\omega_2)-\omega^2 m_2 k_2)^2)}{(k_2-\omega^2 m_2)^2}}$$

Let

$\omega_{0p} = \sqrt{k_1/m_1}$ be a natural pulsation of the structure,

$\omega_{0f} = \sqrt{k_2/m_2}$ a natural pulsation of Frahm absorber,

$\beta_f = \dfrac{\omega_{1f}}{\omega_{0f}}$ a forced pulsation on natural pulsation structure ratio,

$\beta_p = \dfrac{\omega_{1p}}{\omega_{0p}}$ a forced pulsation on natural pulsation Frahm absorber ratio

$a = \dfrac{\omega_{0f}}{\omega_{0p}}$ a natural pulsations ratio, and

$\varepsilon = \dfrac{m_2}{m_1}$ a ratio of the Frahm absorber mass on the structure mass.

With the above notations, we determine the dynamic amplification factor

$$\mu = \frac{(\beta^2 - a^2)^2}{(\varepsilon a^2\beta^2 - (\beta^2-1)(\beta^2-a^2))^2}$$

$\mu = \dfrac{x_P}{x_0}$ dynamic amplification factor of the motion of structure.

$x_0 = \dfrac{F}{k_P}$ static displacement of the mass structure.

PRESENTATION OF THE FINITE ELEMENT METHOD

The determination of the mass and stiffness of the vibration absorber is obtained with a conservative model : embedded beam at one end, linked to a concentrated mass at the other end. The discretization to a finite number of elements, the search of numbers of degrees of freedom of the corresponding system, the set of equations governing the movement of each element, written in a matrix form are carried out in order to determine the natural modes [4]. In most cases, relative to vibrations of simple bending of the beams, distortions due to the normal effort are negligible compared to those due to bending (1/100). By neglecting the longitudinal displacement (due to the cutting effort) u(x, t), we study only the transverse displacement v(x, t) (due to bending).

- Considering infinitesimal transformations, we study the small oscillations about the stable equilibrium position. this allows to linearize the equations.
- The effect of shearing is negligible.
- Navier Bernoulli hypothesis: The material has a linear elastic behavior, the field of displacement is kinematically admissible.

This system is constituted of one beam (mass and rigidity), embedded at one end and a concentrated mass is attached to the other end.
We discretize the system in four elements of length L, width h and thickness e.

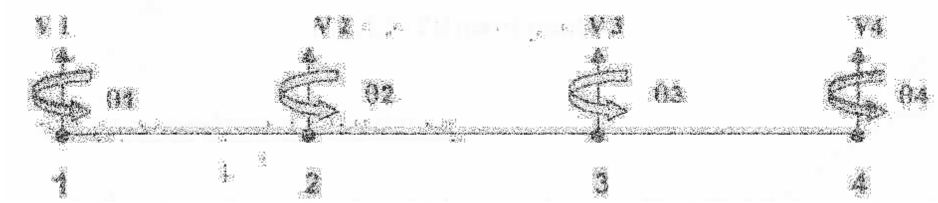


Figure-1 Discretization with the finite element method.

The kinematic parameters : $\{q\}^t = \langle v, L\theta \rangle$ generalised displacement vector of the element. The tensor of displacements of a point of the element beam is written as follows :

$$\{\Omega\}^e = \begin{cases} v(x,t) = \sum_{i=1}^{n} \theta_i(t) qj(t) \\ \theta(x,t) = \sum_{i=1}^{n} \left[\dfrac{\partial \theta_i}{\partial x} \right] qj(t) \end{cases}$$

N : number of degree of freedom

$$\phi(t)=\begin{cases}\phi_1(t)=1-3(x/L)^2+2(x/L)^3\\ \phi_2(t)=x-2x^2/L+x^3L^2\\ \phi_3(t)=3(x/L)^2-2(x/L)^3\\ \phi_4(t)=-x^2/L+x^3L^2\end{cases}\qquad \text{(Interpolation functions)}$$

Mass matrix formula

$$m_{ij}=\int_0^L pS\varphi_i(t)\varphi_j(t)dx$$

The mass matrix is symmetrical

Stiffness matrix formula

$$K_{IJ}=\int_0^L\left(El\frac{\partial^2\varphi_i}{\partial x^2}\frac{\partial^2\varphi j}{\partial x^2}\right)dx$$

These relations make it possible to establish the equations of Lagrange in forced vibrations under matric form:

$[M]_S\{\ddot{q}\}+[K]_S\{q\}=\{F(t)\}$, $[M_e]$:elementary mass matrix, $[K_e]$:elementary stiffness matrix.

Determination of the eigen values:

$$[M]_S=\frac{pSi}{420}[M]^*,[M]_S=\sum[M_e]\ ;\ [K]_S=\frac{EI}{L^3}[K]^*,\ [K]_S=\sum[K_e]$$

The geometric characteristics of the damper of vibrations are : the cross section of the beam : S=e.h, E=5mm, H=40mm, the length of the beam 3L=350mm, quadratic moment of the section I=1,7.$10^{-9}$$m^4$, $E=2.10^{11}N/m^2$, density $\rho=7800Kg/m^3$, the rigidity of the beam, K=5,84.10^3N/m, the elementary mass $m_e\approx183g$, the concentrated mass $m_0\approx800g$. Dynamic characteristics (impulses, amplification factors, transfert function..) are given by experimental (Tab.1-2), Fig.3-7, and theoretical results [5-6].

MEASURES ON THE FRAHAM ABSORBER

Signals :output in amplitude 0.1 mm, low frequency : 10 hz. 2 accelerometers type 4370 : on bearing and shock absorber. Selected range of the amplifier 2635 mv/in out : 3.16 sampling : 2048 points , sampling time: 1 ms. average on 2 acquisitions. The first test is done without the Frahm shock absorber under reasonable resonance frequency (fig. 3), then concentric masses are gradually added from 0,2 to 2kg.

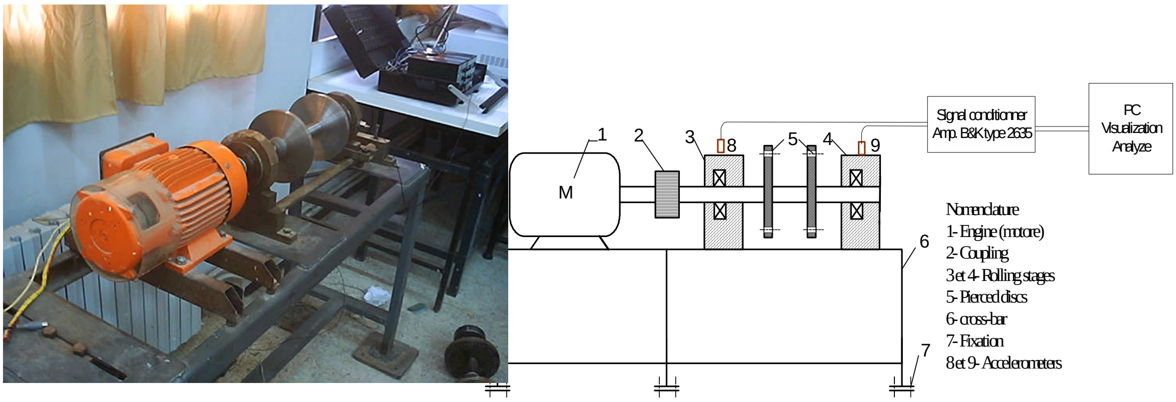


Figure 2-. Photo and Diagram of the test bench.

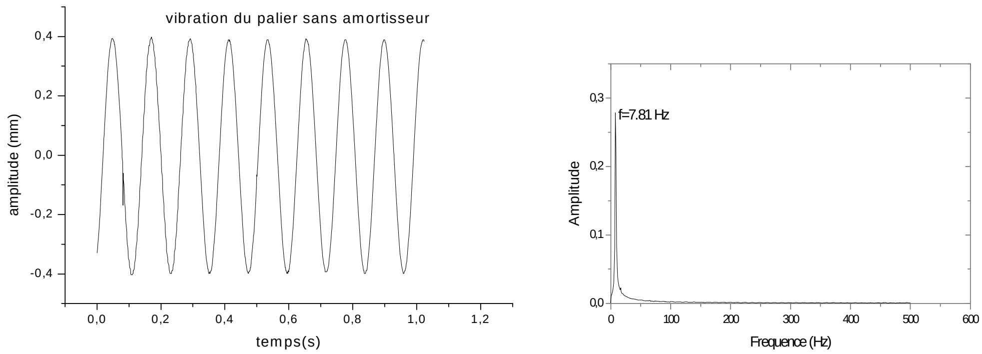


Figure 3- Vibratory signal without Frahm absorber

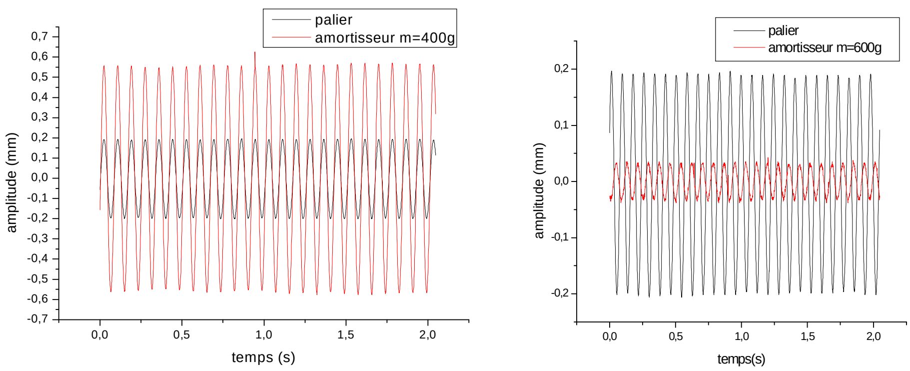


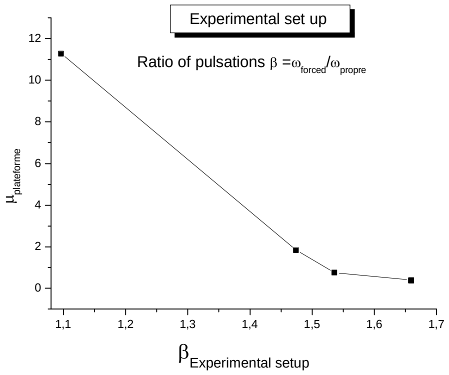
Figure 4- Vibratory signal with Frahm absorber

Table N°i: Dynamic characteristic of experimental set up

Masse [kg]	$\omega_0 f$	$\omega_1 f$	[M]	[K]	βf	a= $\omega_0 f / \omega_0 p$	$\varepsilon = M_p / M_f$	μ
0.2	94,48	360,62	$1,81.10^{10}$		3,82	1,89	0,04	1,55
0,4	84,64	323,06	$2,25.10^{10}$		3,82	1,69	0,04	9,43
0,6	77,35	285,24	$2,69.10^{10}$		3,68	1,54	0,05	0,41
0,8	71,65	273,56	$3,14.10^{10}$	$0,33.10^7$	3,82	1,43	0,06	10,59
1,0	67,07	256,04	$3,58.10^{10}$		3,82	1,34	0,07	3,16
1,2	63,27	241,51	$4,03.10^{10}$		3,82	1,26	0,08	1,93
1,4	60,04	229,20	$4,47.10^{10}$		3,82	1,20	0,09	1,40
1,6	57,27	218,60	$4,92.10^{10}$		3,82	1,14	0,10	1,11
1,8	54,84	209,35	$5,36.10^{10}$		3,82	1,09	0,11	0,78
2,0	52,71	201,17	$5,80.10^{10}$		3,82	1,05	0.11	0,60

Table N°2: Dynamic characteristic of Frahm absorber

Masse [kg]	$\omega_1 p$	[k]	$\omega_0 p$	βp	ε	μ	F_P (N)	H Fp/Amp.(10^7)
	7,81			0.12				
0.2	73,57			1,47	0,04	11,27	1,9	0.047
0,4	76,65			1,53	0,04	1,83	9,02	0,020
0,6	82,81			1,66	0,05	0,74	14,83	0,024
0,8	82,81	$5,52.10^4$	49,91	1,66	0,06	0,38	27,17	0,679
1,0	82,81			1,66	0,07	0,38	26,14	0,333
1,2	82,81			1,66	0,08	0,37	19,07	0,950
1,4	82,81			1,66	0,09	0,37	22,13	2,210
1,6	82,81			1,66	0,10	0,37	22,24	2,220
1,8	82,81			1,66	0,11	0,37	18,59	1,230
2,0	82,81			1,66	0,11	0,36	16,57	1,650



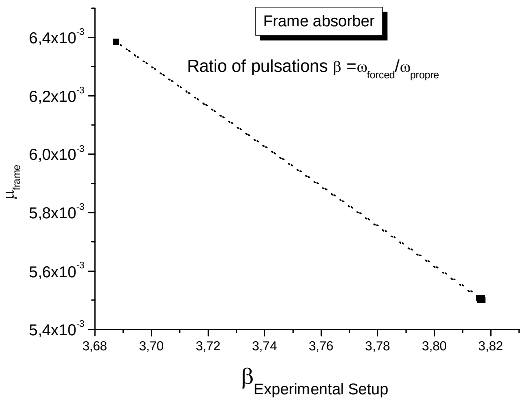


Figure 5,6- Pulsation ration β of experiment set up and Frahm absorber in relation whith there dynamic amplifier.

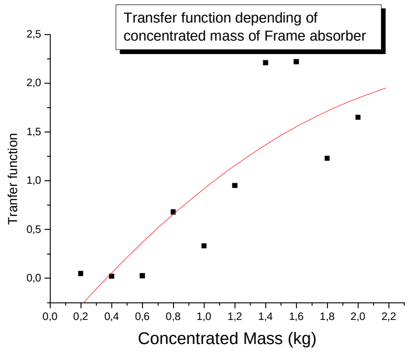


Figure 7- The transfer function in relation to the concentrated mass of Frahm sorber

CONCLUSIONS

The Frahm absorber is a system which is adapted to reduce vibrations. Factors of mass and stiffness influence on resonance frequencies. From the obtained results, we note that optimization doesn't depend only on the ratio of masses and the pulsations of Frahm absorber and the structure, but also on the mechanical features and constructions. When changing the concentrated masses of the Frahm absorber, we noted a stability of vibrations by the reduction of impulses and a growth of the transfer function [6], which is related to the stocked energy until a certain limit.

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