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LETTERS TO THE EDITOR

DYNAMIC CHARACTERIZATION OF AN ECONOMIC, ADJUSTABLE MACHINE MOUNT

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1. INTRODUCTION

The present study deals with the analysis of the vibrational characteristics of the machine mount shown in Figure 1. An inherent advantage of the system is the fact that it can be easily constructed in a machine shop by simply forming a rather thin steel bar into the desired shape.

Due to installation requirements two slots were machined in the bar in order to fasten the motor to the mount using standard bolts. Accordingly the lower natural frequencies of the mount were determined first considering the mount as a solid structure and then taking into account the presence of the slots. Experimental and finite element results were determined.

Finally the effect of small diameter holes placed in zones of maximum stress resultants when the mount vibrates close to the fundamental mode of vibration is investigated.

The present development was carried out in view of the necessity of lowering considerably the noise and vibration levels of air conditioning systems in newly constructed buildings in the Bahía Blanca area. While considerable research effort is invested in highly developed countries in lowering noise and vibration pollution parameters [1], only moderate effort is invested in developing countries [2].

2. FINITE ELEMENT SOLUTION

The system was divided into 96 elements of different lengths (longest element 4·5 mm; shortest (curved portion) 3·875 mm). Timoshenko's theory of vibrating beams was used employing the element developed by Przemieniecki [3]. The model consisted, for each particular determination, of a system of 288 degrees of freedom.

3. EXPERIMENTAL MODEL AND SET-UP

Figure 1 depicts the elastic mount and its corresponding dimensions. The material is steel, SAE 1010 ($\sqrt{E/\rho} = 510{\cdot}000$ cm/s).

Figure 2 shows, partially, the experimental set-up. A spectral analyzer (Hewlett-Packard Model 35670A) together with an infrared non-contacting transducer [4, 5] was used for the experimental determination of the natural frequencies.

4. FINITE ELEMENT EIGENVALUES AND EXPERIMENTAL RESULTS

Table 1 shows a comparison of natural frequencies obtained by means of the finite element method with values determined experimentally. Solid and slotted configurations have been studied. In the case of the slotted mount one observes a slight increment of the fundamental frequency with respect to the situation where a solid bar is considered.

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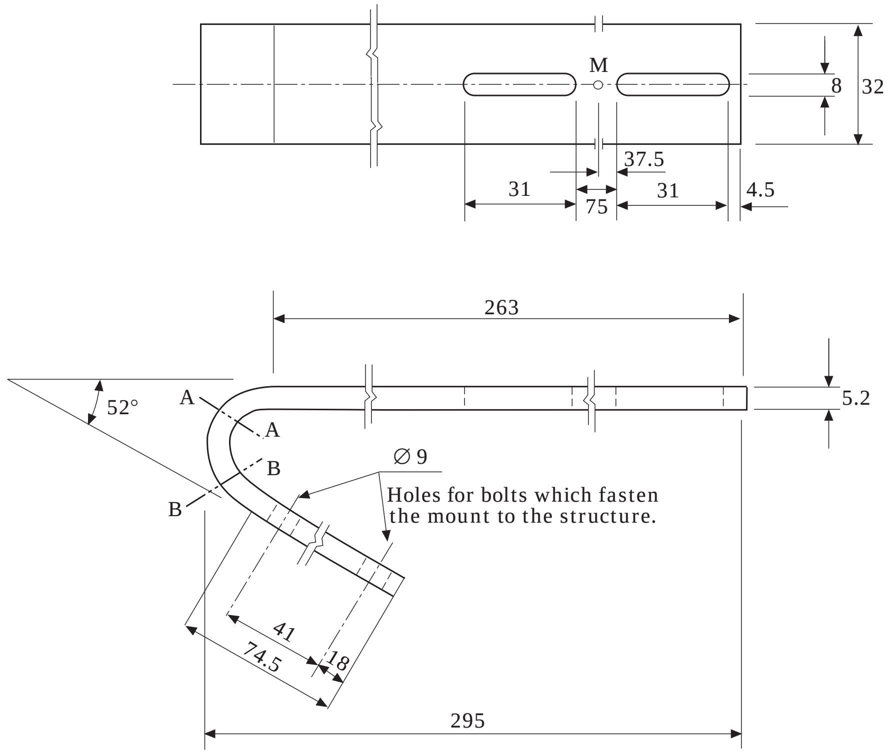

Figure 1. Elastic mount under investigation (dimensions in mm); material SAE 1010; total length 400 mm; M position of the concentrated mass.

Accordingly, dynamic stiffening is achieved. The agreement between experimental results and finite element predictions is reasonably good, from an engineering viewpoint.

Table 2 shows a comparison between experimental and finite element values of the natural frequencies in the case of a slotted mount for three different values of the

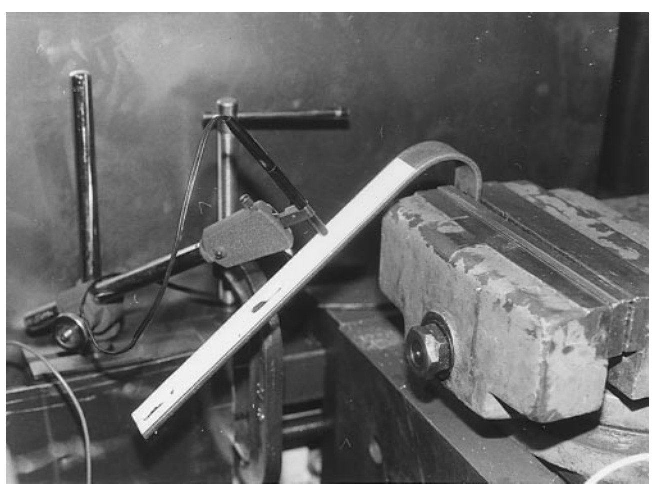

Figure 2. Elastic mount (without concentrated mass) and infrared transducer used in the experimental phase of the investigation.

TABLE 1

Comparison of experimental and finite element results corresponding to the elastic mount shown in Figure 1

Frequency (Hz)	Solid bar		Bar with slots	
	Exp.	F.E.	Exp.	F.E.
f_1	36	36·66	38	38·57
f_2	228	208·20	220	212·98
f_3	396	333·38	382	340·28
f_4	832	815·51	826	824·16
f_5	1548	1524·42	1528	1532·90

Note: Results obtained with no concentrated mass acting on the system.

TABLE 2

Comparison of experimental and finite element results corresponding to the slotted, *elastic mount shown in Figure* 1 *when a concentrated mass is acting on the system*

Frequency (Hz)	M (g)					
	100		400		1000	
	Exp.	F.E.	Exp.	F.E.	Exp.	F.E.
f_1	30	31·48	20	22·25	13·5	15·73
f_2	208	195·39	192	158·78	—	121·57
f_3	346	328·11	332	314·83	—	308·20
f_4	756	751·95	720	689·90	—	662·96
f_5	—	1395·80	—	1324·20	—	1301·10

concentrated mass ($M = 100$, 400 and 1000 g). In all cases the mass is applied concentrically between the slots; see Figure 1. The agreement between experimental values and finite element results is again satisfactory.

Table 3 illustrates the effect of placing holes of different diameters in section A–A, Figure 1, upon the fundamental frequency of the bare, slotted bar. One can see that the presence of the hole has little influence upon the value of f_1. Nevertheless one is able, in principle, to lower the fundamental frequency of the mount *in situ* and, accordingly, increase the efficiency of the system since one reduces the transmissibility factor of the mass–elastic mount system.

TABLE 3

Effect of a circular hole, *placed at* 283 mm *from the free edge*, *upon the fundamental frequency of the slotted bar* (*without concentrated mass*)

Hole diameter (mm)	Fundamental freq. (f_1, Hz) Experimental
0	38·00
6	37·50
8	36·75
10	36·25
12	36·00
14	35·50

TABLE 4

Effect of an additional circular hole of varying diameter placed at 317 mm *from the free edge of the slotted bar*, *when a* 14 mm *diameter hole is present at* 283 mm *from the free edge* (*concentrated mass* 400 g)

Hole diameter (mm)	Fundamental freq. (f_1, Hz) Experimental
6	20·0
10	18·5
14	17·5

This is also illustrated in Table 4 where the value of f_1 has been tabulated when a hole of 14 mm diameter is present in section A–A and one places another hole in section B–B (close to the clamped end of the bar). A fixed value of M has been taken now ($M = 400$ g). As the hole diameter in section B–B varies from 6–14 mm one observes that f_1 diminishes in value, as expected. Again, one is able in this manner to increase the efficiency of the mount. The agreement between experimental values of the fundamental frequency and finite element results is satisfactory for both mechanical configurations (Tables 3 and 4).

A practical application of the present device was to mount (on a couple of these systems) the electric motor which drives, by means of a V-belt, the fan of an air conditioning system. The final result was a substantial reduction of 66% of the overall vibration level of the system (90% of the 1 × RPM component) and a final sound pressure level of 48 dB(A) from an original one of 55 dB(A) measured in the office adjacent to the control room where the electromechanical system operates.

In conclusion: it is hoped that the present study will be useful to mechanical designers since the elastic mount described here is quite versatile and economic.

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