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Trim and Floor Influence on Vibrational Response of an Aircraft Fuselage Model

Sylvain Nadeau* and Yvan Champoux†
Université de Sherbrooke, Sherbrooke, Quebec J1K 2R1, Canada
and
Luc Mongeau‡
Purdue University, West Lafayette, Indiana 47907

The influence of the interior trim and the floor partition on the vibratory response of the physical model of an aircraft fuselage model was investigated experimentally. The one-third scale model faithfully replicated the salient structural features of a rear-engine aircraft fuselage. Detailed modal analysis of the green fuselage, i.e., the bare fuselage skin without any floor, trim, or lining, was first performed. The effects of adding a honeycomb floor partition, a sound-absorbing blanket, and a thin plastic interior lining on the modal parameters of the structure were then determined. It was found that the addition of the sound-absorbing blanket significantly increased the structure's modal damping coefficients for its first five modes of resonance. Notable changes in the structural response at higher frequencies were also observed. These results strongly suggest, in agreement with recent studies reported in the literature, that sound-absorbing blankets do add significant damping to the vibration response of the structure (in addition to absorbing sound energy from the cabin interior). These damping properties may be used advantageously for the optimal design of skin-damping aircraft noise-control treatments for minimal added weight, and minimal cost.

Introduction

SIGNIFICANT aircraft interior noise is structure-borne, radiated by fuselage vibrations. Important sources include the engines, and the external pressure fluctuations caused by turbulent boundary layers over the fuselage. Structural interior noise-control methods involve the use of damping materials to attenuate the vibrations of the structure near resonance, e.g., damping tape applied directly onto the skin. Another acoustic treatment consists of installing sound-absorbing and sound-barrier materials, either on or in the vicinity of the sound-radiating panels. Many of these materials are incorporated within the interior trim. Blankets of materials, wrapped in bags for safety reasons, are installed between the fuselage and the interior decorative panel. The cost of these different treatments is significant, representing in some cases a sizeable fraction of the cost of the aircraft (primarily because of the need to tailor each bag to the varied cavity shapes). The acoustic treatments also add weight, which is always a primary concern in aircraft design.

In the past 20 years, numerous studies[1–6] have been reported on airplane interior noise radiation. Excellent reviews of the literature have been offered by Mixson and Wilby,[7] and, more recently, by Wilby.[6] Early in the 1970s, several experimental studies[8–11] clearly showed that the addition of damping materials to the skin, such as damping tapes, rubber wedges, or other stringer or frame-damping treatments, yield considerably reduced interior noise levels relative to those in unfurnished and untreated, bare (or green) aircraft cabins. Even considering added weight, which is at a high premium, such treatments often remain the solution of choice for airplane manufacturers to solve interior noise and vibration problems.

It is generally acknowledged that damping treatments effectively reduce both the vibration response and the associated radiated noise for resonant, lightly damped structures. Surprisingly, few studies have examined the contribution of various trim components to the overall damping characteristics of aircraft fuselages. A laboratory study by Mixson et al.[1] reported that the addition of acoustic treatments raises the sound transmission loss of treated panels measured in one-third octave bands over a wide frequency range. The primary benefit of the treatments (from a noise control point of view) was reported to be associated with the added mass. Grosveld and Mixson[2] investigated the vibrational response of a section of a propeller aircraft's sidewall. They suggested that damping tape did not appear to be effective in cases where other treatments already provided significant damping. Cooper[12] investigated the influence of structural damping treatments for an aircraft fuselage with interior trim panels installed. It was shown that the addition of the damping treatments did not affect the cabin interior noise levels, a result that could not be satisfactorily explained. Similar behavior was observed by Halvorsen and Emborg,[13] who reported, following a study of the interior noise characteristics of the Saab 340 propeller aircraft, that significant damping was provided by the sound-absorbing materials and the trim panel of the fully trimmed fuselage.

In parallel with these experimental studies, several theoretical models have been developed and used for the prediction of aircraft interior noise.[14–23] One of the most elaborate models,[19–21] based on the transfer matrix method, involves the calculation of the sound transmission, absorption, and reflection characteristics of the various layers of the trim. The theoretical models yield higher structural loss factors when the presence of the interior trim is accounted for.[19] The model, however, does not take into account the possible coupling between adjacent layers. For example, the transfer matrix for a layer of sound-barrier material does not depend on the skin characteristics and vice versa. Recently, evidence of coupling between the different treatments led Graham[22,23] to question the accuracy of current models. In particular, he suggested that the effectiveness of structural damping treatments for trimmed aircraft should be re-evaluated.

Aircraft manufacturers are now using various types of foams as sound-absorbing and sound-barrier materials, in place of the traditional fiberglass blankets, at least for sections of the trim package. Foams are known to provide damping when placed in direct contact with steel structures.[24,25] New techniques are currently developed to measure these properties.[26] Recently, Atalla and Panneton[27] investigated the damping properties of multilayered sound-absorbing treatments for locally excited flat rectangular panels. Guérin[28] studied the influence of the trim on the vibration response of a shell.

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*Research Assistant, Mechanical Engineering Department.
†Professor, Mechanical Engineering Department.
‡Assistant Professor, School of Mechanical Engineering. Member AIAA.

Both studies indicated that the presence of porous sound-absorbing materials in close proximity to extended resonant structures may provide significant damping, even when the material is not in direct contact with the shell.

These previous studies therefore suggest that the use of constrained layer damping in the trim design of modern aircraft fuselage may be redundant in some cases because significant damping may already be provided by the sound-absorbing or sound-barrier treatments. The purpose of this study was to investigate this further by performing detailed measurements of the vibration response of a one-third scale model of an aircraft fuselage with and without interior trim. The influence of the floor partition was also considered. The results revealed that the damping provided by the sound-absorbing layer and the trim panel was significant. It is therefore concluded that this coupling effect may be taken into account to optimize trim design for minimal added weight, and minimal cost. Moreover, it was also found that the installation of the sound-absorbing materials plays a significant role in their effectiveness for damping.

Aircraft Fuselage Model

The test structure was a one-third scale physical model of a twin rear-engine jet aircraft fuselage. The fuselage model is shown in Figs. 1 and 2. The structure consisted of a ribbed cylindrical shell made of 0.51-mm-thick aluminum panels. It had an o.d. of 0.9 m and a length of 4.84 m. Other dimensions and characteristics of the model are summarized next:

1) Bare fuselage: material is aluminum, 0.90 m o.d., 4.84 m length; 52.7 kg; 0.51 mm skin thickness; 52 longitudinal stiffeners; 39 circumferential stiffeners; and 28,150 rivets (for practical reasons, the rivets' scale factor is $\frac{1}{1}$ instead of $\frac{1}{3}$).

2) Floor: material is Honeycomb (Nomex), 6.35 mm thick, and 5.9 kg.

3) Absorbing material (the scale factor does not apply to these materials because they are different than those used in aircraft): material is Conaflex F-100, 2.5 cm thick, and 6.4 kg.

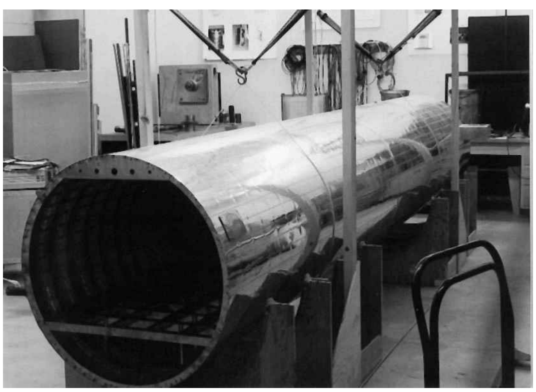

Fig. 1 Photograph of the physical model.

Fig. 2 Interior of the bare structure.

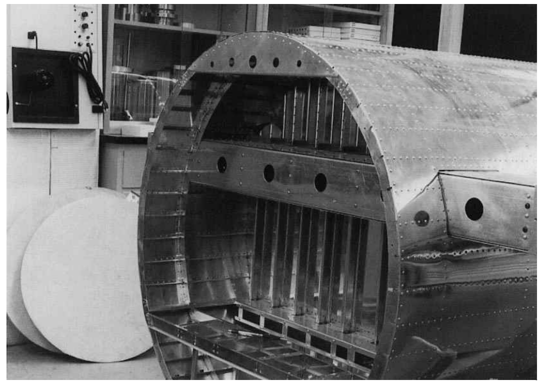

Fig. 3 Bulkhead and simplified torque beam.

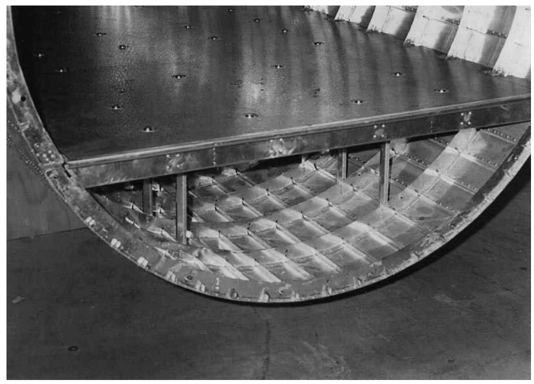

Fig. 4 Details of the floor construction.

4) Decorative panels (the scale factor does not apply to these materials because they are different than those used in aircraft): material is polypropylene, 1 mm thick, and 7.7 kg.

The structure frame was stiffened using 52 longitudinal stingers and 39 circumferential ring frames. The stingers were 90-deg angles, having dimensions of 9×9 mm, riveted to the inside of the skin. The ring frames were made of 90-deg angles, having dimensions of 9×25 mm, with the smaller area riveted to the inside of the skin. The test structure was opened at one extremity, and closed by a bulkhead at the other. Details of the bulkhead and the simplified torque beam are shown in Fig. 3.

The floor supporting structure (see Fig. 2) was made of I-beams (18×34 mm) in the longitudinal direction, and C-beams (9×34 mm) in the lateral direction. Each C-beam was vertically supported by two C-beams (6×14 mm), as shown in Fig. 4.

Trim Configurations

Four different configurations were investigated to evaluate the relative influence of different trim components and other structural elements on the damping characteristics of the structure:

Configuration 1: No trim and no floor (Figs. 1 and 2). This configuration will be referred to as the bare or green fuselage configuration.

Configuration 2: No trim, with a floor bolted on the green fuselage's floor support structure (Fig. 4). Honeycomb (Nomex 6.35 mm thick with 33 cells per 6.45 cm^2) material that was lined with fiberglass layers on each side was used. The floor supports are shown in Fig. 2, together with an interior view of the fuselage model.

Configuration 3: Same as configuration 2, with an absorbing material in the upper part of the test structure. The 2.5-cm-thick Blachford Conaflex F-100 absorbing layer was simply forced into the space between the circumferential stiffeners, with no bonding (Fig. 5). All interior walls above the floor were lined, including the bulkhead.

Configuration 4: Same as configuration 3, with the addition of interior decorative panels installed over the absorbing material. The interior panels, made of 1-mm-thick polypropylene (Fig. 6), were forced in place with no bonding and were in contact with the sound-absorbing material.

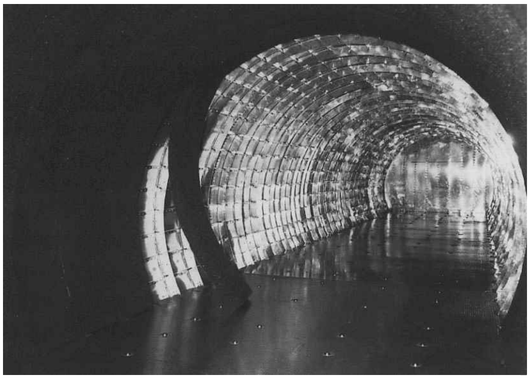

Fig. 5 Installation of the sound-absorbing blankets.

Fig. 6 Installation of the interior trim panel.

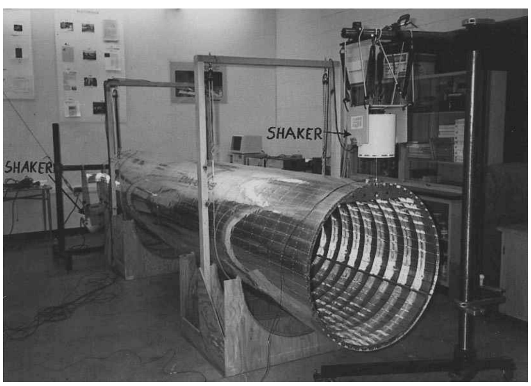

Fig. 7 Photograph of the experimental apparatus.

Experimental Methodology

Because of the size of the structure, more than one shaker was used to distribute the energy evenly and to excite all of the modes of interest. Hence, a multiple input/multiple output approach was followed for the modal analysis of the test structure in all configurations. The free–free cylindrical structure was excited using two Modal 50 shakers, shown in Fig. 7. The two shakers were fed with random bursts, having 25% of random signal from 20 to 420 Hz, thus providing two uncorrelated force signals with minimal leakage errors. The data acquisition was made using a Bruel & Kjaer Model 3550 multichannel frequency analyzer, and GenRad's Star Modal System software package. The frequency response functions were then imported into another computer program (SDRC's I-DEAS software) for the calculation of the modal parameters. The frequency response functions were fitted using polyreference in the time domain to obtain the poles (frequencies and damping), whereas the residues were obtained using polyreference in the frequency domain to include residual terms for modes outside the frequency range of interest. A polyreference complex exponential method was chosen to evaluate the poles to use the frequency response data from both sources in a global least-squares sense. This algorithm allows the properties of damped complex structures with high modal density and repeated roots to be accurately evaluated.

Results

A typical frequency response function (FRF) for the green structure is shown in Fig. 8 (configuration 1). The modal density is very high, as demonstrated by the uniformity of the FRFs at frequencies above ~270 Hz. This result is qualitatively similar to the frequency response of a bare full-size aircraft fuselage reported by Halvorsen and Emborg.[13] A typical FRF for the structure with floor panels and interior trim (configuration 4) is also shown in Fig. 8. The addition of the floor partition and the trim elements clearly modifies the structural behavior.

The natural frequencies of the first five modes are shown in Table 1. The modal assurance criteria (MAC) for configurations 1 and 4 for the first five modes are presented in Fig. 9. These results show clearly that the modes selected were not significantly

Table 1 Natural frequencies for the first five modes for all configurations tested

Test configuration	Mode frequency, Hz 1	2	3	4	5
1	29.3	53.7	58.1	74.5	137.1
2	30.4	61.5	58.0	73.4	141.2
3	29.1	57.8	53.2	67.0	120.0
4	27.8	55.8	50.3	64.0	112.2

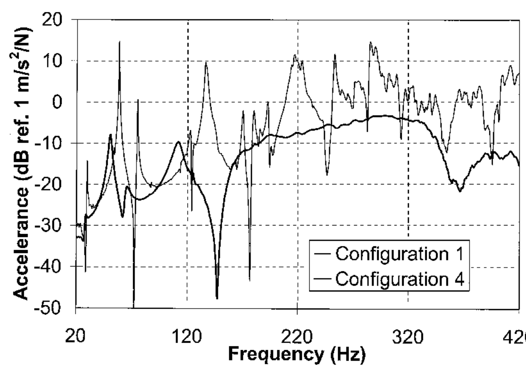

Fig. 8 FRF of the green structure (configuration 1) and the structure with the trim (configuration 4).

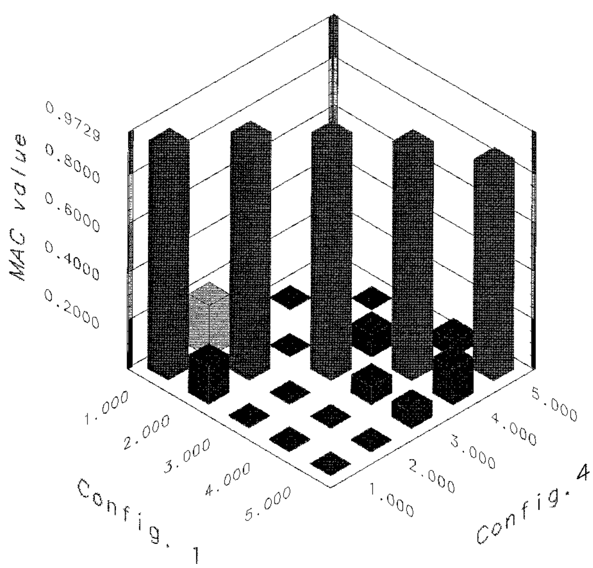

Fig. 9 MAC for the first five modes, configurations 1 and 4.

influenced by the trim and the floor partition. The same trend was reported by Halvorsen and Emborg.[13] Similar results were also obtained for the other configurations tested. The mode shapes for configuration 2 associated with the first five natural frequencies are shown in Figs. 10–14. The addition of the floor partition (configuration 2) only influenced the second mode, whose natural frequency increased by 15%. Comparing the mode shapes for modes 1 and 2 (Figs. 10 and 11) reveals similar circumferential mode orders, but different longitudinal ones. Therefore, the increase in frequency appears to be a result of the longitudinal deflection of the floor. For

Table 2 Viscous damping ratios for the first five modes for all configurations tested

Test configuration	Mode damping, %				
	1	2	3	4	5
1	0.3	0.2	0.2	0.3	0.9
2	0.8	1.0	0.3	0.4	1.0
3	1.2	1.8	1.8	2.6	2.7
4	3.4	2.9	2.9	3.3	4.0

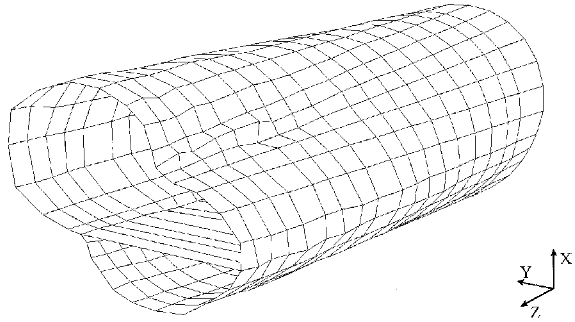

Fig. 10 Mode shape for the first mode, configuration 2.

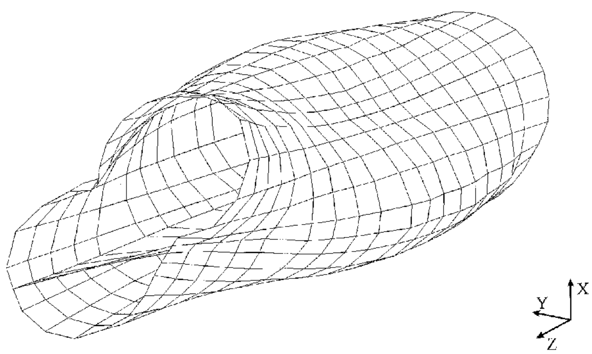

Fig. 11 Mode shape for the second mode, configuration 2.

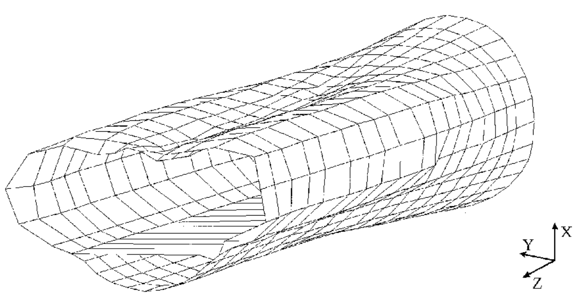

Fig. 12 Mode shape for the third mode, configuration 2.

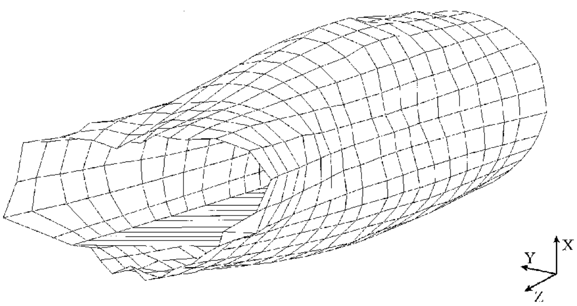

Fig. 13 Mode shape for the fourth mode, configuration 2.

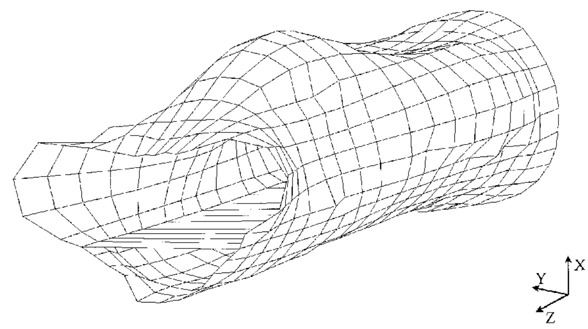

Fig. 14 Mode shape for the fifth mode, configuration 2.

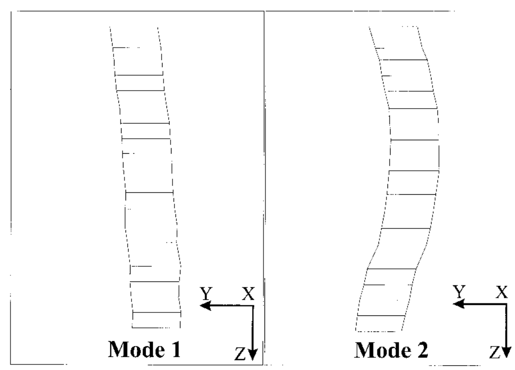

Fig. 15 Mode shapes for the first and second modes of the floor partition, configuration 2 (top view).

these modes, the floor displacement is important along the X and Y directions only. The displacement in the X direction because of torsion around the Z axis is similar for both modes. The floor deformation along the Y direction was not significant for mode 1, but was large for mode 2, as shown in Fig. 15. Therefore, the frequency increase for mode 2 was likely caused by a significant increase of the in-plane floor stiffness.

One can also note in Table 1 that all natural frequencies decreased with the addition of absorbing materials and interior trim (configurations 3 and 4). Hence, the added mass from the absorbing material and the interior trim panels was more important than the added stiffness. Atalla and Panneton[27] reported the same trend for unbonded urethane foam in close proximity of rectangular compliant panels. Note that they observed an increase in the resonance frequencies for the case of bonded foams (none of the trim panels nor the sound absorbing blankets were bonded in this study).

The damping ratios for the first five modes are shown in Table 2. The damping ratio is significantly higher in the presence of the floor partition, the sound-absorbing blanket, and the interior trim panel, respectively. The addition of the floor partition (configuration 2) increased damping for modes 1 and 2. The addition of absorbing material (configuration 3) and the addition of the interior decorative trim panel (configuration 4) significantly increased the damping values for all five modes. Figure 16 shows the damping ratios for all four test configurations. These results are again in good qualitative agreement with those obtained by Halvorsen and Emborg[13] for an actual twin-propeller aircraft.

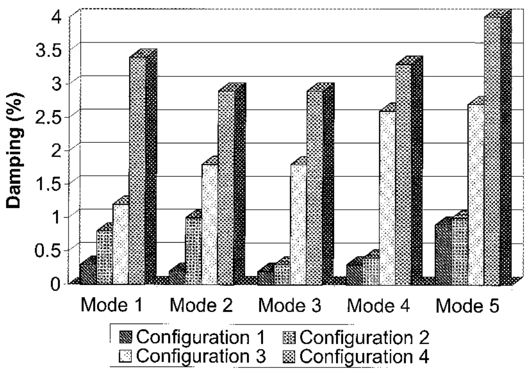

Fig. 16 Viscous damping ratios of all structures tested for the first five modes.

Discussion

Three dissipation mechanisms may have caused the observed damping coefficients increase.[27] Firstly, the relative movement between the sound-absorbing material and the skin panels may have caused coulomb damping. Secondly, energy dissipation of near-field acoustic waves in the absorbing material may have reduced structural loading. Finally, energy dissipation caused by structural deformation of the sound-absorbing material skeleton, driven by the structural deformations, may have occurred. While further work is needed to quantify the relative importance of the contributions, it may be inferred from this short analysis that the installation of the trim elements may have a significant impact on their effectiveness as damping treatments.

Conclusions

The structural response of a one-third scale model of an aircraft fuselage was measured with and without sound-absorbing blankets lining the interior skin panels, trim panels, and floor panels. It was observed that each of these elements (in particular the trim panels) provided significant damping to the structure in the absence of other damping treatments. Based on these results, it is suggested that the damping provided by the sound-absorbing blanket in a multilayer interior acoustic treatment for structure-borne noise may, in some cases, make the use of additional skin-damping materials redundant. These properties could be exploited to design-optimized trim configurations with maximal damping/weight and damping/cost ratios.

Acknowledgments

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