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DOI:
10.1243/09544070JAUTO1556

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The manuscript was received on 15 March 2010 and was accepted after revision for publication on 9 August 2010.

DOI: 10.1243/09544070JAUTO1556

Abstract: Requirement for low emissions and better vehicle performance has led to the demand for lightweight vehicle structures. Lighter gauge panels are being used to construct the body-in-white (BIW) monocoque structure, which is the basic component of the vehicle body. Since lighter gauge panels tend to generate more vibration and interior noise, it is necessary to optimize the dynamic performance of lightweight vehicle structures in order to achieve acceptable levels of vibro-acoustic performance. The design of a light commercial van structure has evolved over the years and through a lightweighting exercise the current BIW is about 10 per cent lighter than the previous BIW even though the volume capacity was increased by 15 per cent and the load-carrying capacity by 18 per cent. In this study, the dynamic performance of the current production light van BIW structure is investigated. Its performance is assessed against the structural dynamic performance standards which have been established for this class of structures. While the input mobility performance was found to exceed the standards easily, the modal mobility performance was found to be unsatisfactory owing to the occurrence of local panel resonant modes in the two side panels. A finite element model of the structure was developed to study the effect of adding stringers to the roof and side panels to eliminate some of the local panel modes and thus to improve the dynamic performance of the structure.

Keywords: dynamic performance, body-in-white, vibro-acoustic, refinement, lightweighting, finite element model

1 INTRODUCTION

There is increasing demand for vehicles to be lighter in order to reduce fuel consumption. The vehicle body structure, being one of the heaviest components of the vehicle system, is therefore a prime target for weight reduction. The vehicle body structure is the receptor of vibrational energy inputs from the road and the powertrain, some of which is then radiated into the passenger compartment as acoustic energy. The body-in-white (BIW) is the basic structure to which the trim pack is added to form the vehicle body structure and to a large extent controls the dynamic behaviour of the vehicle body from the noise, harshness, and vibration point of view. A reduction in the weight of the BIW is often achieved by using lighter gauge panels, but this can result in higher levels of vibration and interior noise, hence negatively impacting on passenger comfort.

This is quite often due to the occurrence of local resonant modes in the vehicle body panels. The overall vibro-acoustic performance of the whole structure is greatly influenced by the performance of each panel. It is therefore important that the individual panels are properly integrated together to form the whole structure.

In this study the dynamic performance of a light commercial vehicle BIW structure is analysed to assess the effect of a lightweighting process. The BIW structure is 10 per cent lighter than the structure that

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it replaces, has 15 per cent more volume, and has 18 per cent more load-carrying capacity. In addition, no damping pads have been applied to control local panel resonant modes. The aim of the study is to assess, empirically, the dynamic performance of the structure on the basis of structural dynamic performance standards and to determine the effect of suggested modifications to the structure to eliminate local panel modes using finite element analysis (FEA), as depicted in Fig. 1.

2 STRUCTURAL DYNAMIC PERFORMANCE STANDARDS

The concept of using vehicle structural dynamics performance standards to assess the performance of vehicle body structures was developed in the late 1970s [**1**, **2**]. The concept is based on the basic mechanism of vehicle interior noise due to the vibrating cabin walls [**3**]. The techniques based on the concept were applied for the first time in 1977–1978 [**4**] in the development of a prototype vehicle structure and subsequently in the development of a whole series of prototype vehicle structures. The original standards proposed were based on tests carried out on a wide variety of vehicle body structures [**2**] and are shown in Table 1. These techniques have since been adopted by many vehicle manufacturers. The application of structural dynamic performance standards facilitates the assessment of vibro-acoustic performance of vehicle structures over both narrow and wide frequency bandwidths against specified dynamic standards and, with this, specified design acceptability criteria [**5**]. It also provides diagnostic information for identifying discrete structural problem areas so that corrective measures can be taken to improve the structure.

It is generally recognized that vehicle interior noise under 500 Hz is predominantly structure borne [**6**]. The structural performance standards are based on the transmission of vibration energy to the structure and are normally assessed over the frequency range 10–200 Hz, highlighting the vibro-acoustic performance of the structure. This is the frequency range where road-induced booming noise occurs in the vehicle interior [**7**]. The input point mobility (the input point velocity normalized by the driving force) is used to assess the performance of mounting points on the structure for dynamic subsystems such as the engine, suspension, gearbox, and exhaust system, as it physically describes the ability of a point in

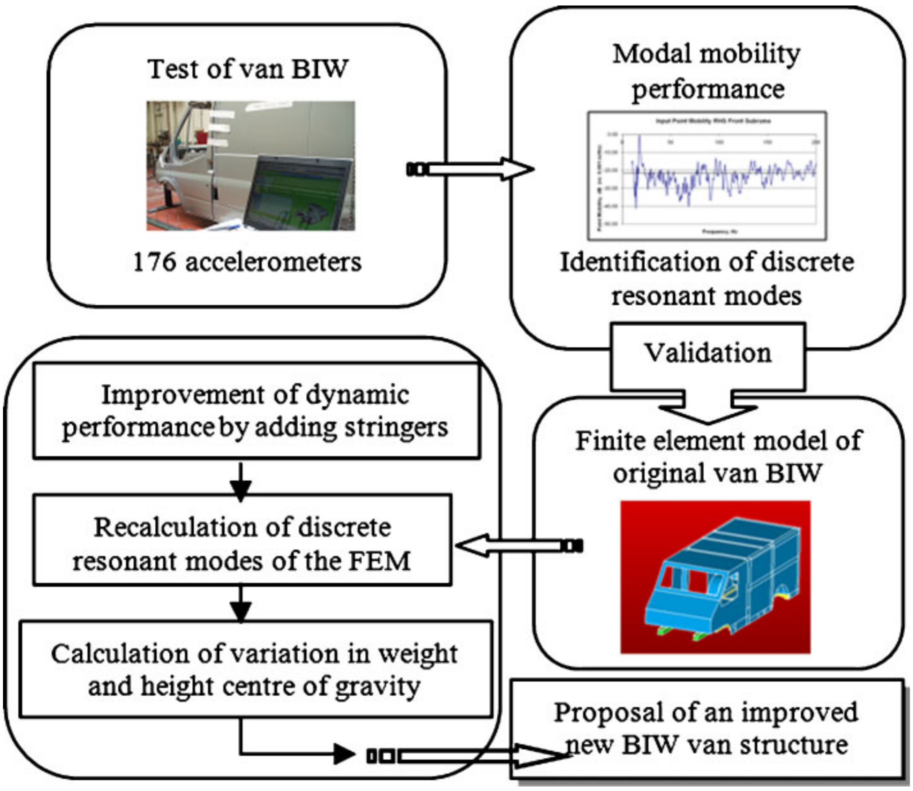

Fig. 1 Assessment of and improvement in the van BIW performance (FEM, finite element method)

Table 1 Original dynamic performance standards for monocoque saloon car and van structures [2]

	Point mobility performance* (dB)	Modal mobility performance* (dB)
10 Hz bandwidth average	−10	−22
Broadband average	−15	−30

*Ref. 10^{-3} N m/s.

the structure to admit vibration energy into the structure. The transfer mobility describes the response of other points in a structure to the vibration input at a vehicle structure attachment point. It is therefore a measure of the transfer of vibration energy from an input point to other points in the structure. The modal mobility is defined as the average transfer mobility vector for a whole structure or a definitive part of it with respect to a specific input point [1]. It therefore expresses the total transfer mobility response as a single parameter. It is thus a measure of the dynamic performance of the whole structure or definitive part of it with respect to the vibration input at a specific input point, taking into account elemental phase cancellation due to structural damping. It can therefore be extended to provide a measure of the vibro-acoustic performance of the structure in respect of structure-borne radiation. Formal definitions of the structural dynamic parameters can be found in reference [1].

The measurement of structural dynamic parameters is accomplished by the use of a custom-built computer-controlled automatic vibration excitation and structural-response-measuring system. The vehicle structure is suspended from rigid A-frames as shown in Fig. 2 on four soft springs such that none of its six rigid body natural frequencies is higher than 5 Hz so as not to influence the structural response in the test frequency range (10–200 Hz). The vibration input is applied to the structure with an electromagnetic actuator which is attached to the structure at the selected attachment point by a shaft fixed to an adaptor at the actuator and carries an impedance head for measuring the input point acceleration and input force. A self-centring ball-and-socket joint is situated between the impedance head and the structure for correct alignment of the axis of the actuator, which is rigidly clamped when alignment is made. The actuator axis is aligned in the direction that vibration inputs will normally feed into the structure at the attachment point, e.g. the axis of a McPherson strut. The test is carried out using a swept-sine input while, in response to an input force of 20 N, the input acceleration response and acceleration responses at the grid points indicated in Fig. 3 are measured at each test frequency. The structural dynamic response parameters are computed from the measured acceleration data.

The critical advantage of applying structural dynamic performance standards is that a vehicle structure can be assessed in terms of actual performance figures over a narrow or broad frequency bandwidth, taking into account the integrated effects of resonant and non-resonant modes in that bandwidth [8]. To this end, structural dynamic performance standards have been set for different classes of vehicle body structure based on independent tests on a large number of BIW structures. This allows new vehicle body structures to be judged against the best-performing structures in their class.

A fundamental requirement of the use of structural dynamic standards is that they must be under constant review and the current standards (Table 2) reflect the performance levels of some of the most refined structures tested [9].

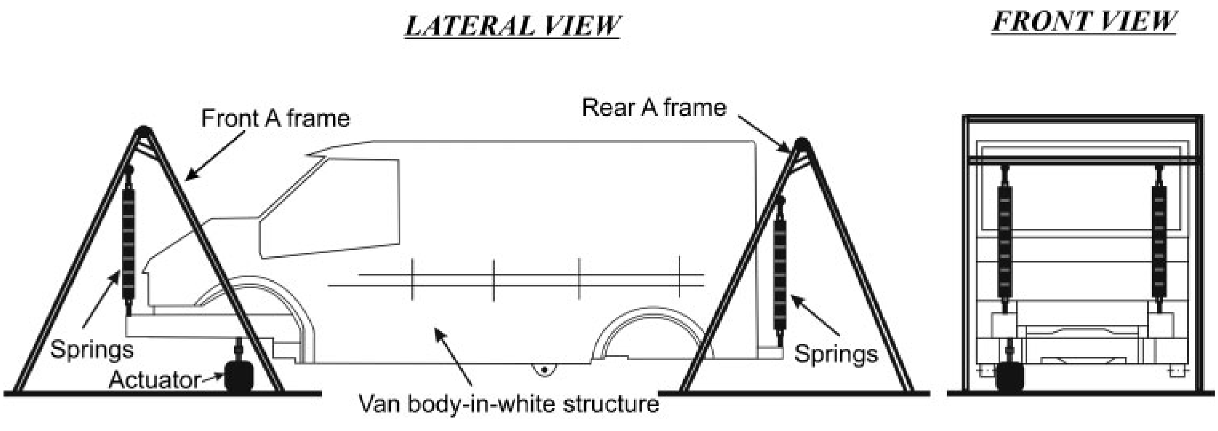

Fig. 2 Boundary conditions of the tested van BIW structure

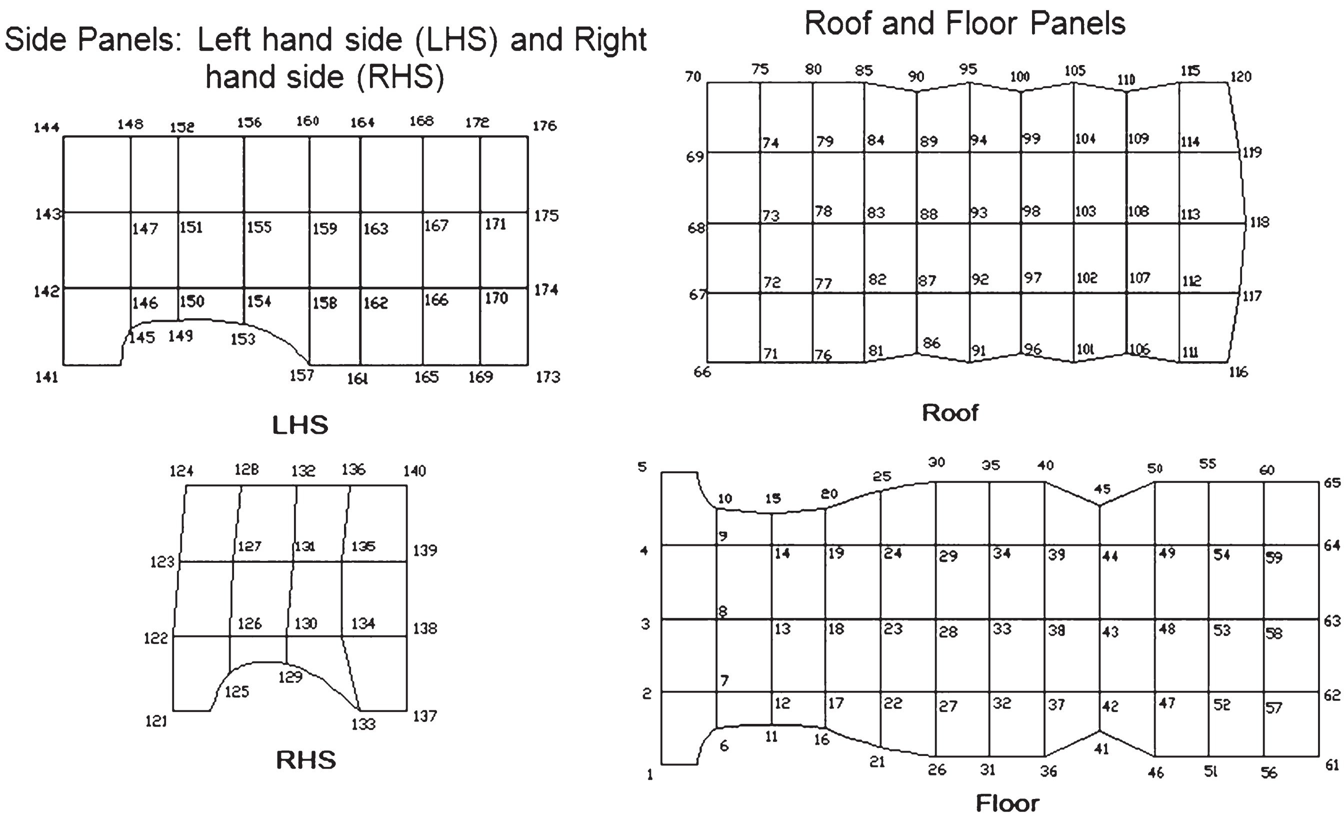

Fig. 3 Layout of transducers on body panels

3 STRUCTURAL DYNAMIC PERFORMANCE OF THE LIGHT VAN BIW STRUCTURE

3.1 Input point mobility

Figure 3 shows the layout of transducers on the body panels for the structural dynamic tests on the vehicle structure. A total of 176 measuring points were used on the structure. Figures 4 and 5 show typical input mobility responses at the front and rear respectively of the structure. The broadband average mobility performances of −21 dB (front) and −23 dB (rear) far exceed the current standards for BIW car structures, namely −15 dB [**8**]. This is an indication of the great improvement in the performance of attachment points for suspension and other vibration input points on the structure. Nevertheless, for the front subframe attachment point there is a discrete resonant mode at 17.5 Hz where the peak input mobility is −0.40 dB which is far higher than the average level and therefore could be the cause of serious refinement problems in terms of ride. Similarly, for the rear spring attachment point, there is a discrete mode at 32 Hz (−6 dB) and, to a lesser extent, another at 23.5 Hz (−11.6 dB), which are relatively higher than the average level. However, any problems caused by these discrete modes can be treated relatively easily.

Table 2 Current dynamic performance standards for monocoque saloon car and van structures [**9**]

	Point mobility performance* (dB)	Modal mobility performance* (dB)
10 Hz bandwidth average	−10 dB	−28 dB
Broadband average	−15 dB	−36 dB

*Ref. 10^{-3} N m/s.

3.2 Modal mobility

Figure 6 shows a typical modal mobility response for the complete structure with respect to the front subframe attachment point. The broadband average modal mobility performance of −29.5 dB falls short of the current standard for BIW car structures, namely −36 dB [**8**]. However, this satisfies the 'old' standard in reference [**9**]. The indication is that improvement in the modal mobility performance has not matched the improvement in the input mobility performance. The discrete resonant mode at 17.5 Hz observed in the input mobility response for this attachment point is clearly reflected in the overall structure modal mobility response with respect to the same attachment

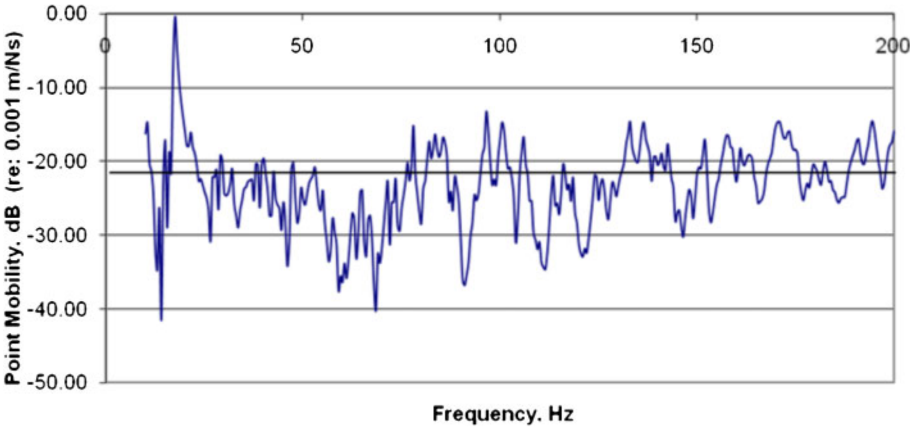

Fig. 4 Input mobility response of the front subframe attachment point on the RHS of the van BIW

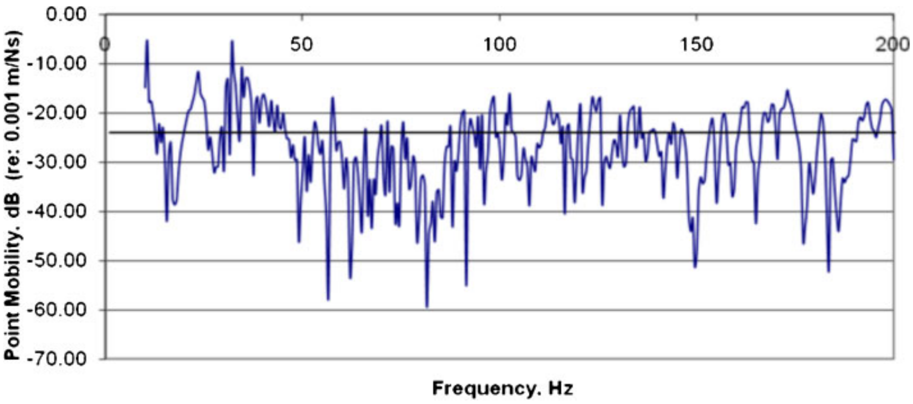

Fig. 5 Input mobility response of the rear leaf spring attachment point (RHS)

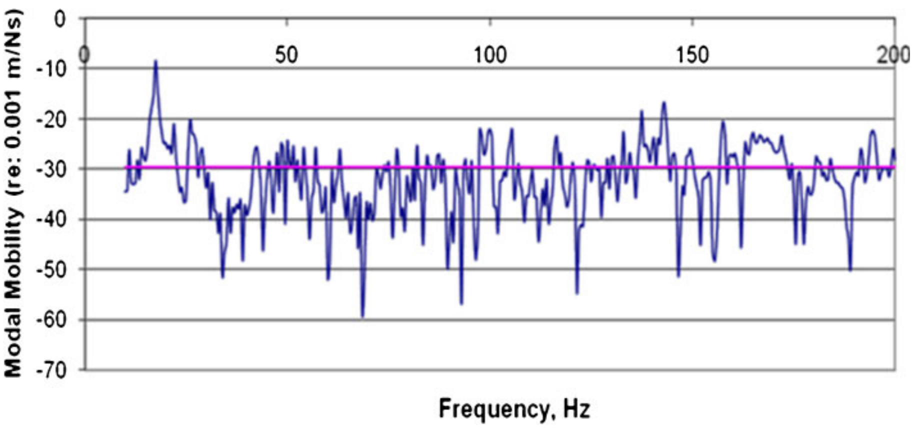

Fig. 6 Complete structure modal mobility (RHS front subframe)

point. This indicates a whole structure mode rather than a local mode as it shows as a high peak in the responses of the roof panel and the two side panels as reflected in the modal mobility responses of these panels (Fig. 7) and in the 10–20 Hz bandwidth average levels (Table 3). To a lesser extent there is another prominent resonant mode at 143 Hz which can be attributed to a local resonance in the roof panel. This mode does not feature prominently in the input mobility response or any of the other panels. Table 3 shows that the two side panels, i.e. the RHS panel and the LHS panel, exhibit quite poor modal mobility performances compared with those of the roof and floor panels.

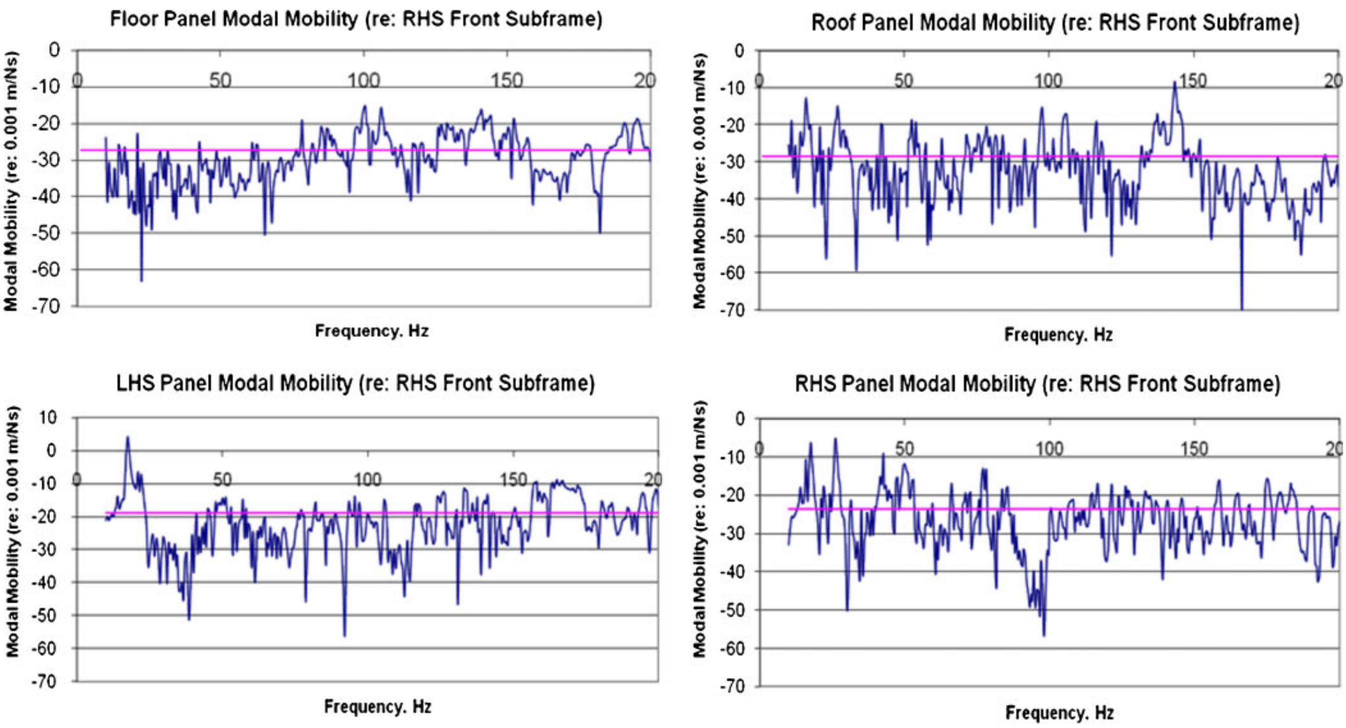

Fig. 7 Individual panel modal mobility responses (RHS front subframe)

Table 3 Significant bandwidth average modal mobilities (RHS front subframe mount)

	Bandwidth average modal mobility level (dB)				
Bandwidth (Hz)	Floor	Roof	RHS	LHS	Complete structure
10–20	−33.4	−23.9	−18.0	−9.2	−21.7
20–30	−34.7	−24.3	−18.6	−17.8	−26.8
130–140	−22.7	−24.3	−24.1	−18.5	−27.3
140–150	−21.4	−19.3	−27.2	−21.3	−26.1
160–170	−34.5	−37.6	−25.6	−11.8	−25.8
190–200	−22.6	−34.4	−28.9	−17.3	−28.1
10–200	**−27.4**	**−28.5**	**−23.7**	**−18.8**	**−29.5**

Figure 8 shows the modal mobility response for the complete structure with respect to the rear leaf spring attachment point. The broadband average modal mobility performance of −31 dB again falls short of the current standard for BIW car structures but satisfies the old standard of −30 dB. The discrete resonant mode at 17.5 Hz is again prominent in the modal mobility response in spite of not featuring in the input mobility response. Again this mode shows as a high peak in the responses of the roof panel and the two side panels, indicating that it is a whole structure mode. Two other high peaks in this modal mobility response are observed at 26.5 Hz and 31.5 Hz. The 26.5 Hz mode is again prominent in the responses of the roof and side panels while the 31.5 Hz mode is prominent in the responses of the roof and floor panels (Fig. 9). These are the resonant modes that have the most effect on the average broadband modal mobility as reflected in the bandwidth average modal mobility levels for the 20–30 Hz and 30–40 Hz bandwidths (Table 4). Again, the two side panels exhibit quite poor broadband modal mobility performances (−17.1 dB and −21.7 dB) compared with those of the roof and floor panels (−27.8 dB and −35.9 dB) and are responsible for the modest overall structure modal mobility performance.

4 FINITE ELEMENT MODELLING

Finite element (FE) models are very useful to study, inparticular, the response of a vehicle structure which has been modified so as to optimize it [**10**] or to improve its dynamic behaviour [**11, 12**]. Therefore, an FE model of the structure was analysed using ANSYS with a view to studying the effects of

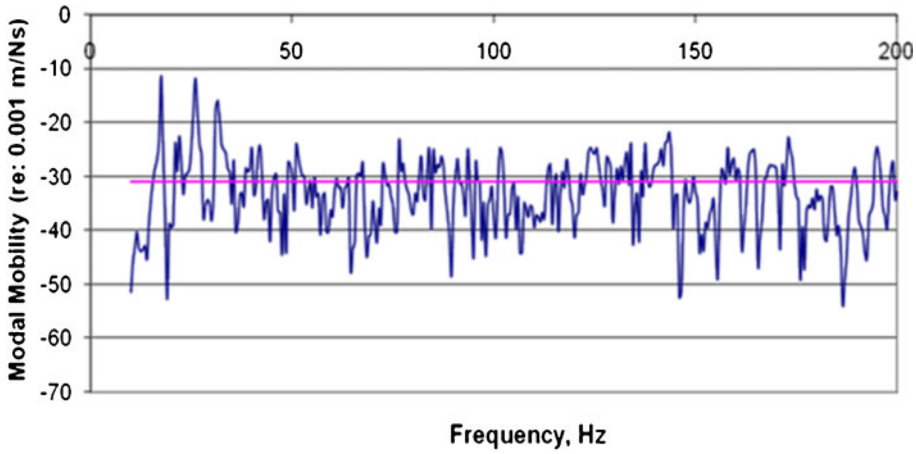

Fig. 8 Complete structure modal mobility (RHS rear leaf spring mount)

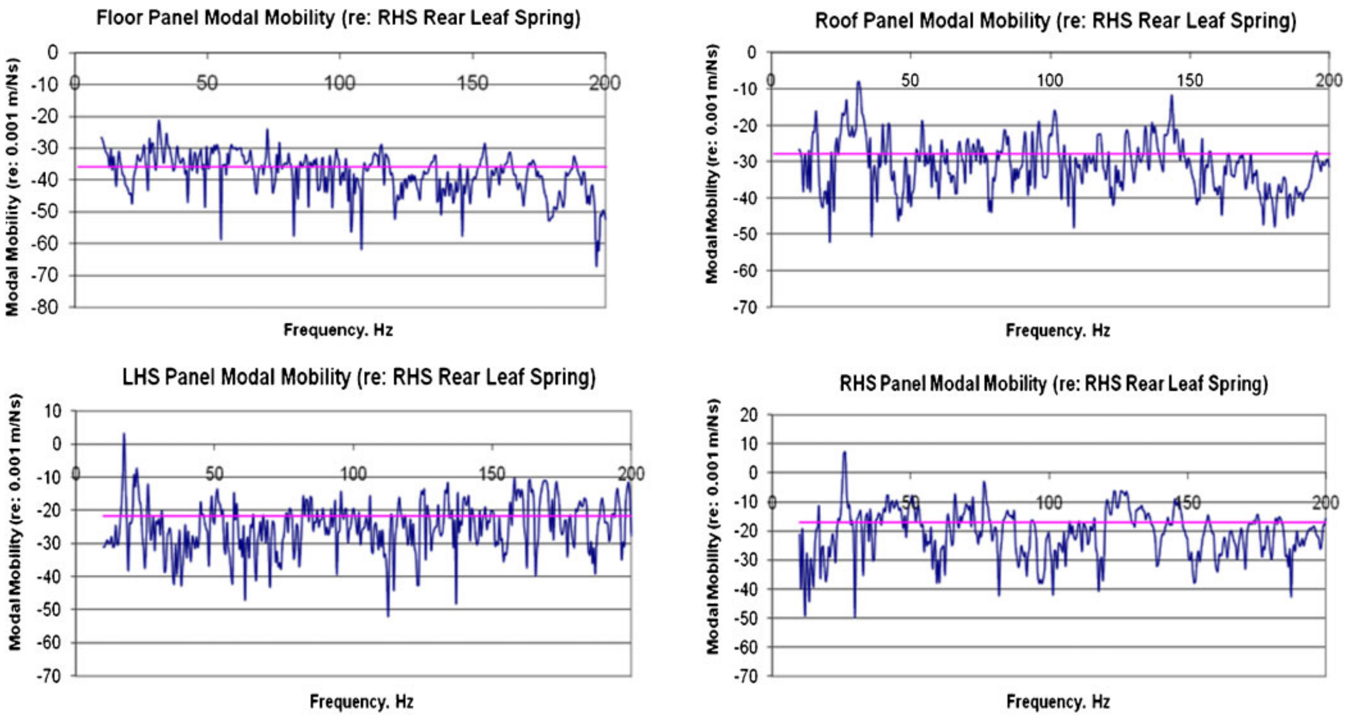

Fig. 9 Individual panel modal mobility responses (RHS rear leaf spring)

Table 4 Significant bandwidth average modal mobilities (RHS rear leaf spring mount)

	Bandwidth average modal mobility level (dB)				
Bandwidth (Hz)	Floor	Roof	RHS	LHS	Complete structure
10–20	−33.7	−28.2	−26.1	−17.2	−30.1
20–30	−33.8	−22.5	−8.6	−18.3	−25.1
30–40	−30.0	−19.8	−17.4	−29.5	−26.5
120–130	−42.0	−30.3	−9.8	−22.9	−29.3
130–140	−38.2	−25.7	−16.0	−20.4	−29.9
140–150	−41.6	−22.7	−15.7	−21.8	−29.5
10–200	**−35.9**	**−27.8**	**−17.1**	**−21.7**	**−30.9**

various proposed modifications to improve the performance of the structure. The ANSYS FE model had 35 598 elements and 34 437 nodes. Initially, eigenvalue extraction was carried out to identify the natural frequencies of the structure. The mode shapes of identified modes were plotted for comparison with the measured structural response data. Of particular interest were the modes which were identified as being responsible for the poor structural dynamic performance especially those associated with local panel modes.

The panels of the van BIW FE model have been modelled by means of shell elements. Therefore, the midplane of all the surfaces has been drawn in ANSYS. The stiffness of the connection between panels is given by the dimensions of the connecting panels and no extra stiffness has been added because of welding. In addition, the block Lanczos method, which is used for large shell element models, has been employed. This method does not allow any damping. Finally, no boundary conditions have been applied to the FE model, as a modal analysis is being carried out in free–free conditions.

4.1 Original structure

The original van BIW structure has a length of 4900 mm, a height of 1550 mm, and a width of 1680 mm. The thickness of the body frame panels is 0.8 mm, the thickness of the chassis beams is 1.4 mm, and the thickness of the front rails is 2.4 mm. This original van BIW structure is first analysed in ANSYS, being modified afterwards in order to improve its structural dynamic performance.

After performing the FEA, the results are presented. Figure 10 shows a number of local roof panel modes which were identified from the FEA as being significant. Figures 10(a) and (b) show local roof panel modes at 21.75 Hz and 22.4 Hz respectively. Both modes are close to the 21 Hz mode highlighted in the structure test. Because of the closeness of the two modes, it is difficult to separate them in a structure test as pure modes are difficult to excite using a single shaker. Therefore the two modes become merged and are observed as a single mode in the structure test. A similar situation applies with the modes shown in Figs 11(a) and (b) where two roof modes occur at 33.7 Hz and 32.7 Hz respectively with local panel modes at the rear and middle sections of the roof panel. Again both modes are combined into the mode observed at 32 Hz in the structure test. A further local roof panel mode observed in the structural test at 98 Hz is associated with two natural roof modes occurring at 99.2 Hz and 99.1 Hz, as shown in Figs 12(a) and (b) respectively.

A discrete resonant mode was observed at 17.5 Hz in both the input mobility and the modal mobility responses. This mode features as significant peaks in the modal mobility responses of both side panels. This is borne out by the FEA results shown in Fig. 13 at 17.51 Hz. The individual panel modal mobility responses show that the LHS panel's response is significantly higher than those of the other panels at this frequency, indicating that the LHS panel's response dominates at this frequency. Two further significant local modes of the RHS panel are shown in Figs 14(a) and (b), highlighted in the panel's modal mobility responses at 26.5 Hz and 81 Hz respectively.

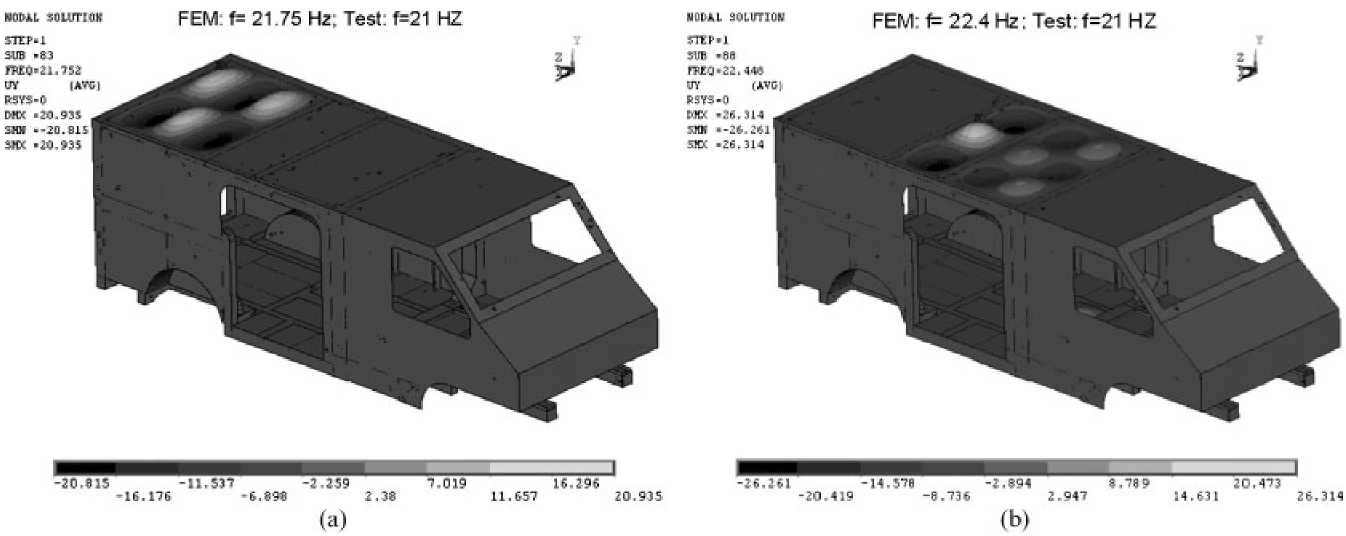

Fig. 10 Local natural modes of the roof panel around 21 Hz: (a) 21.75 Hz; (b) 22.4 Hz

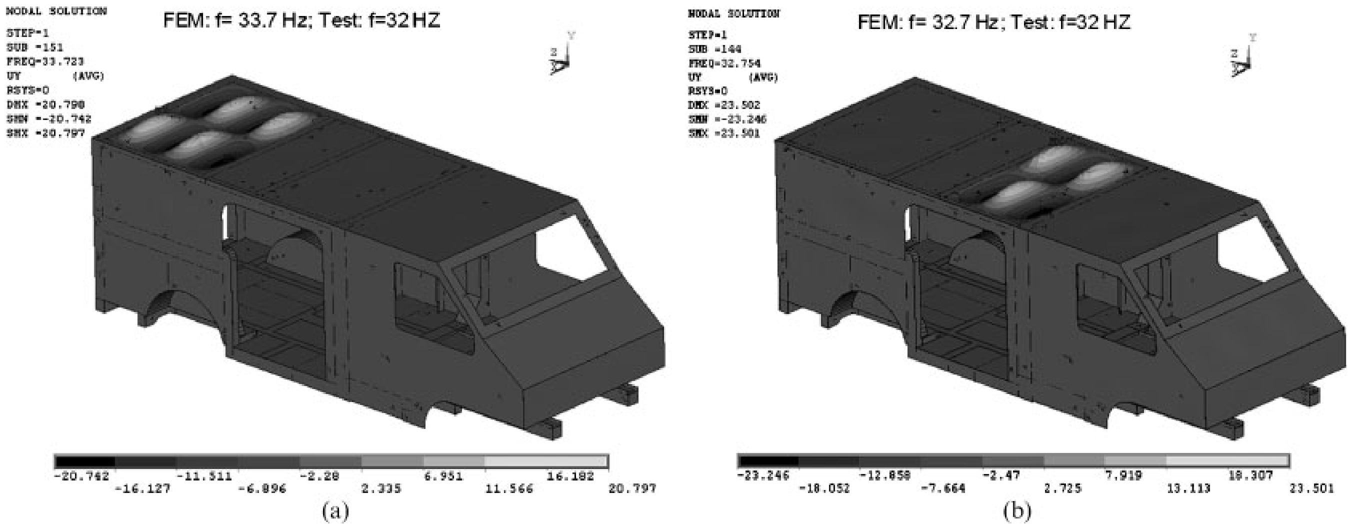

Fig. 11 Local natural modes of the roof panel around 32 Hz: (a) 33.7 Hz; (b) 32.7 Hz

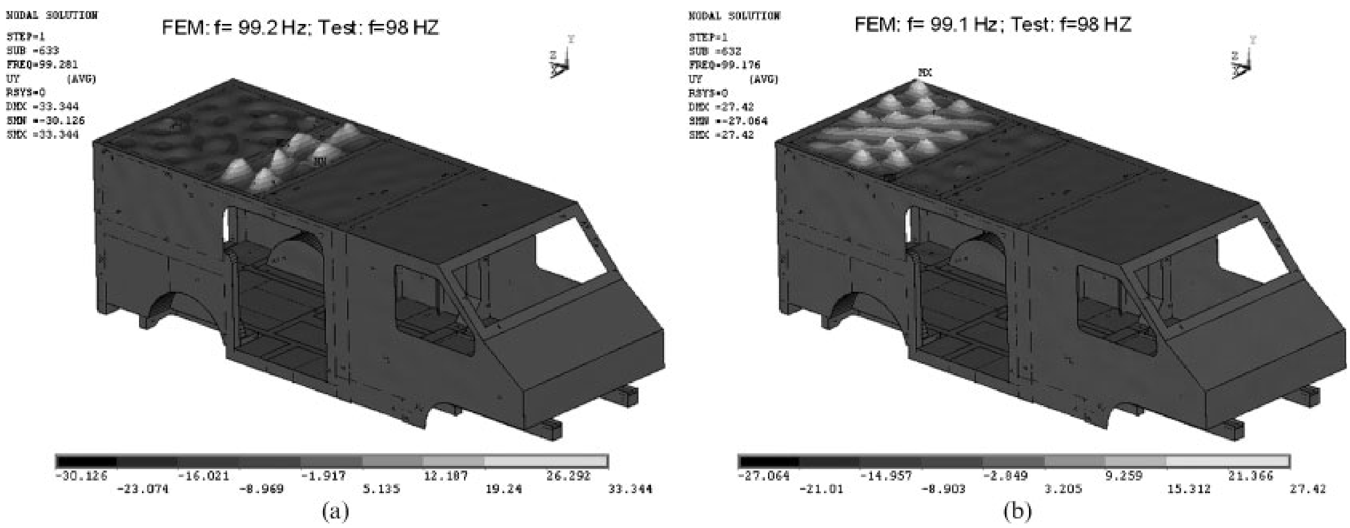

Fig. 12 Local natural modes of the roof panel around 98 Hz: (a) 99.2 Hz; (b) 99.1 Hz

4.2 Modified structure

In order to improve the dynamic performance of the structure, it was decided to use stringers to stiffen the side and roof panels in order to eliminate some of the low-frequency panel modes. The modified van has the same dimensions as the original van (4900 mm length, 1550 mm height, and 1680 mm width) but diagonal stringers of 1.4 mm thickness had been placed in the rear part of both sides of the van as well as in the roof, as depicted in Fig. 15. Of particular concern was the whole structure mode at 17.5 Hz which featured a particularly high modal response in the side panels, especially the LHS panel. It was recognized that the addition of the stringers would result in increased weight of the structure as well as the height of the centre of gravity. However, the elimination of these low-frequency modes was

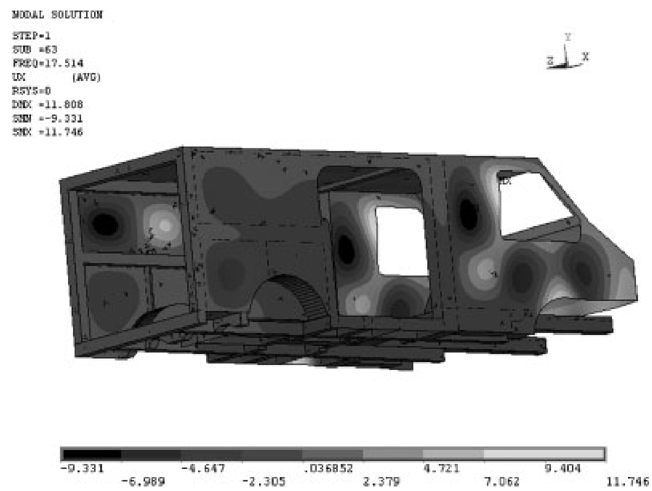

Fig. 13 Natural mode for 17.5 Hz

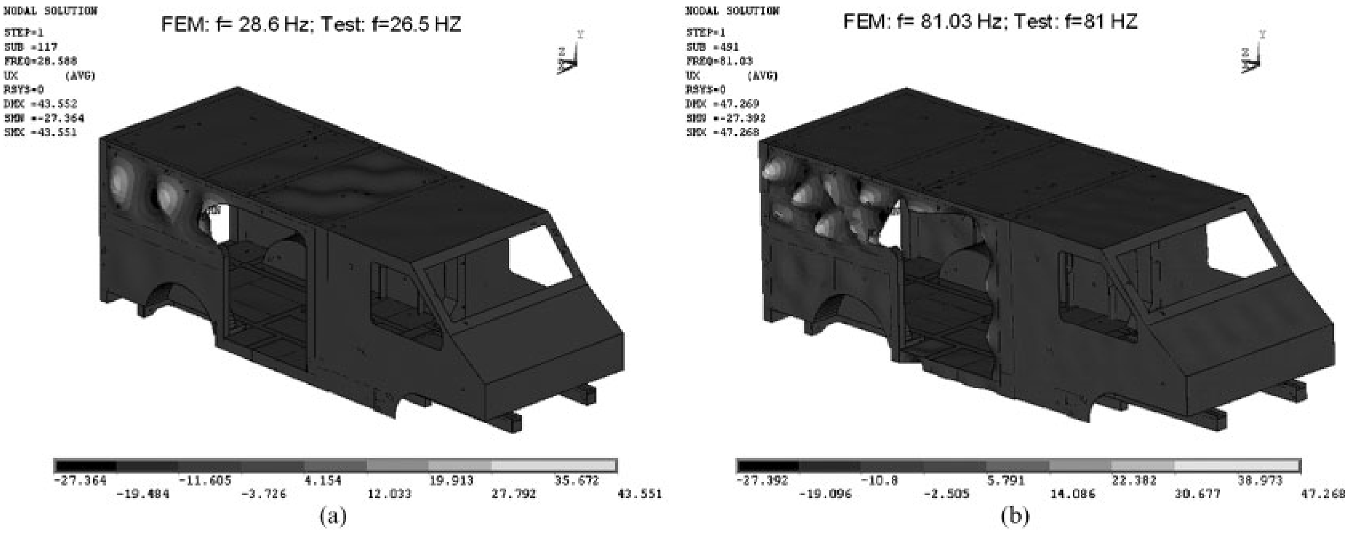

Fig. 14 Local natural modes of the RHS: (a) around 26.5 Hz; (b) around 81 Hz

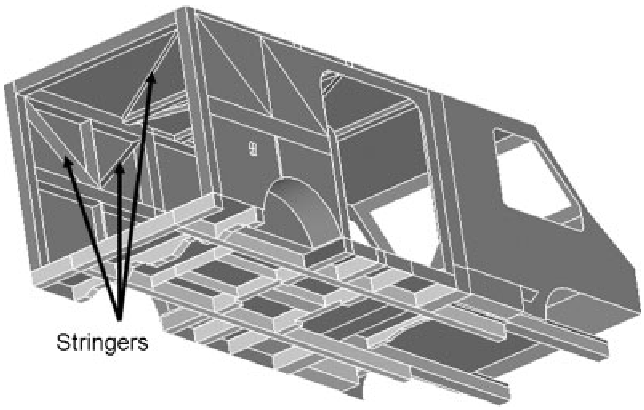

Fig. 15 FE model of the modified structure

deemed essential from the point of view of improving the ride performance of the structure. The original FE model was therefore modified to include the proposed stringers in order to predict their effect on the structure's dynamic performance.

As depicted in Fig. 16, the stringers have avoided local panel resonant modes at 17.4 Hz in both sides of the van BIW structure. The addition of the stringers resulted in an increase in weight of 1.8 per cent, which is just 5.3 kgf. Given that a weight reduction of 10 per cent over the previous structure had been achieved, this is a modest increase in weight. The height of the centre of gravity also increased by 3.3 per cent, which does not give any cause for concern.

5 CONCLUSION

Structural dynamic tests carried out on the light van BIW structure enabled the dynamic performance to

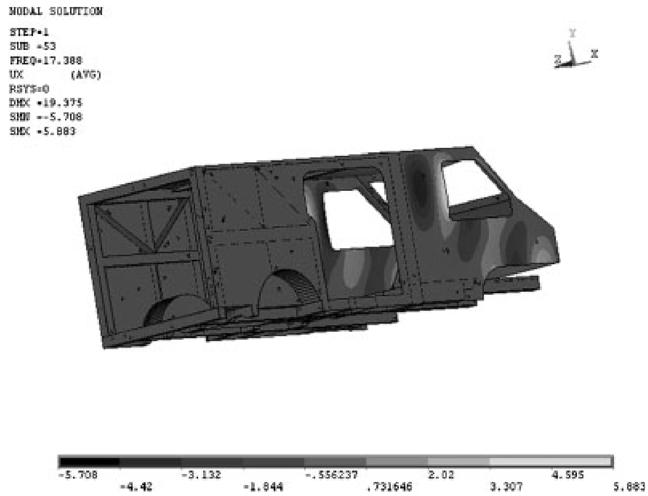

Fig. 16 Mode of the modified van BIW structure at 17.4 Hz

be assessed empirically on the basis of structural dynamic performance standards. The results indicated that, although the input mobility performance of attachment points for engine mount and suspension had improved considerably compared with the previous generation structure, there had been little improvement in the modal mobility performance which satisfied the old standards but failed to meet the more stringent current standards. This was attributed to the lightweighting exercise carried out on the structure which had reduced its weight by almost 10 per cent compared with the previous generation structure. The lightweighting exercise involved a reduction in the gauge thickness of some of the panels, liberal use of panel swaging, and redesign of the underframe structure. Since the interior noise is controlled by the vibrating cabin walls, the modal mobility of the cabin panels reflect the noise-generating capacity of the structure. Investigation of the modal mobility of individual panels reveals the occurrence of local panel modes particularly in the side panels, suggesting lack of proper integration into the whole structure. The empirical results are in agreement with the results of FEA of the structure. Simulation of suggested modifications using FEA shows that the dynamic performance of the structure can be greatly improved by eliminating certain local panel modes without significantly increasing the weight of the structure.

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