Vehicle door panel reinforcement design to reduce wind noise

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ABSTRACT

The door panel of a car is acted on by oscillatory pressures generated aerodynamically under driving conditions. The resulting structural panel vibrations are thought to be a significant cause of wind noise, since the deformation of the door panel relative to the structural frame may allow an airborne acoustic path for wind noise into the passenger compartment. This paper describes an investigation into door panel vibration amplitudes under typical driving conditions using the finite element method. Conclusions are drawn relating to the cost effective use of panel reinforcement to reduce this 'aspiration' wind noise.

1. Introduction

The design of a modern passenger vehicle door involves more than the basic structural durability, and extends to the consideration of passenger comfort and quality of ride. In this respect the reduction of the noise experienced by the driver and passengers has recently become a major factor in the comparative assessment of different vehicles, and is moreover subjected to regulations and industry standards[1,2]. Engineers involved in Noise, Vibration and Harshness (NVH) studies on automobiles have been able to make use of numerical analysis methods, based on various discrete techniques such as the finite element method[3] (FEM), boundary element method[4] (BEM), as well as ray tracing and statistical energy analysis (SEA). Examples of such application may be found in Trevelyan et al[5] and Zhang et al[6].

The problem of wind noise is generally more difficult to treat using a discrete analysis method than are, for example, the acoustic fields generated by panel vibrations excited by the automotive powertrain. The frequency spectrum is wide, extending to frequencies too high to be analysed practically using classical FEM and BEM approaches. Furthermore, analysis of body panel vibrational modes becomes difficult at frequencies above around 80Hz because of the increasing modal density.

However, it is possible to study the problem as one of 'aspiration noise'. This may be defined as the opening of a less resistive acoustic path between the exterior and the passenger compartment under vibration of the door panels. Relaxation of the fit of the door seal is viewed as significant in the generation of wind noise. In the worst case, if the vibration amplitude is greater than the compression of the door seal, an airborne acoustic path may be created. The problem may then be analysed as a structural one instead of a problem in acoustic wave propagation. The door panel is considered to be under steady state forced vibration, the forcing arising from the various aerodynamic pressure variations which act on the vehicle panels. The design of the door panel and its reinforcement is clearly influenced by criteria other than aspiration wind

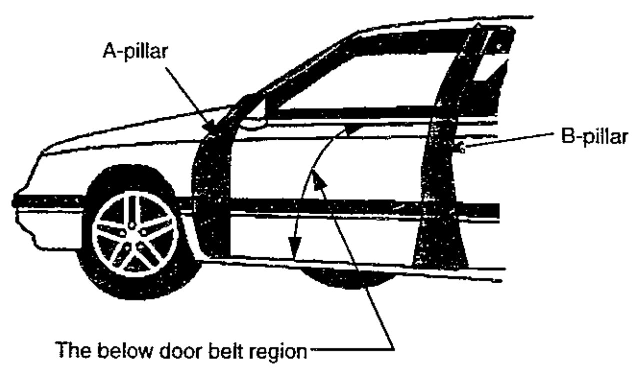

Figure 1. Door panel definitions

noise[7]. For example, in a design to resist side impact, the side impact safety bar and pad located in the below door belt region, and the reinforcement of the B-pillar lower part are the traditional solutions. But the most important concept of the side impact safety in the body panel part is to make the door panel and the side structure more absorptive against the impact. Other aesthetic considerations should also apply.

The designer aiming to satisfy these requirements is also faced with the added complexity of the window operation, door locks, hinges and other devices which need to be included within and on the panels.

The stiffening of door panels to reduce vibrational deformation carries penalties in terms of weight and cost, and therefore any noise reduction measures need to be carefully optimised in relation to these parameters.

Yang[8] describes the application of the finite element method in the optimisation of door panel reinforcement to reduce vibrational deformation due to oscillatory pressure differences between the air inside and immediately outside the door.

2. Finite element modelling of the door panel

The door panel of interest in this study has been modelled mathematically using the finite element method, using the commercial ABAQUS software[9] with its preprocessor ABAPRE. Geometric data was imported in the IGES standard format from a geometric CAD model, and meshed using quadratic thin shell elements throughout. Figure 2 shows the essential features of the door panel under investigation. This was described by an initial model (as shown in figure 3), containing 8448 thin shell elements. This was denoted model 1. Based on this model a set of variations on the model were built, each having different reinforcement designs as shown in figures 4 to 13.

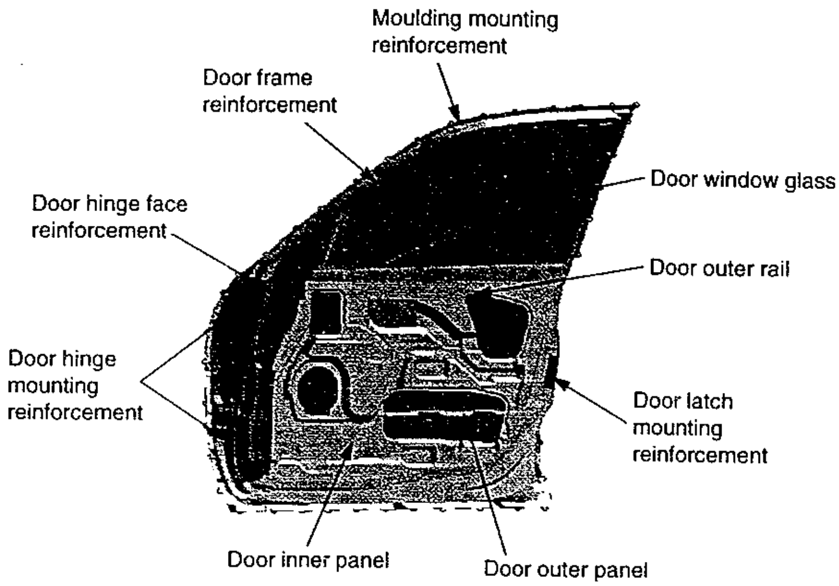

Figure 2. Door panel under investigation, showing various features

Notice that model 2 is not shown, since this has no reinforcement. The design optimisation for reinforcement is basedon these finite element models, which contained similar numbers of elements.

Some initial tests showed that, as expected, the presence of the rubber door seal provided negligible stiffness in comparison with that of the door

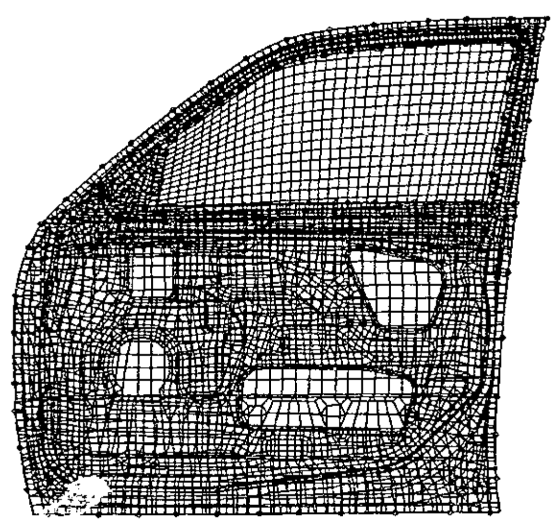

Figure 3. Initial f.e. model of the panel (model 1)

panel, and so this feature was omitted from the models. Further tests showed that the connectivity between the window glass and the frame made little difference to the results, and it was decided to make the connection ideal in this area.

The displacement constraints for all models were provided by fixed conditions at the two door hinges on the A-pillar and the latch on the B-pillar.

The panels were analysed under steady state forced vibration, being forced by sinusoidal pressures applied to the elements modelling the outer panel of the door and the glass. The pressure was assumed to be constant over the surface of the panel, which is clearly a simplification which could be addressed by future work. However, it was assumed for the purpose of this study comparing different reinforcement designs that the comparative displacement response would yield a similar result to a comparative study considering a more realistic pressure distribution. The value of pressure coefficient C_p is taken from Howell[10], whose work on a Rover vehicle (of similar dimension to the Hyundai vehicle under consideration in this study) determined a value of $C_p = 0.5$ in the region of the driver's window.

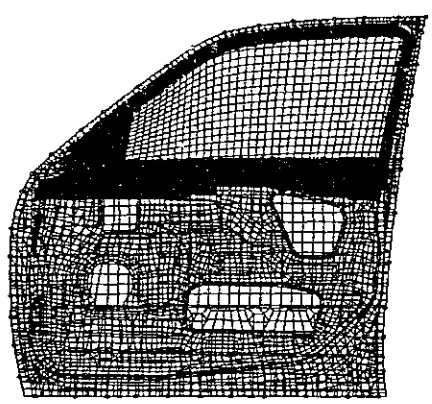

Figure 4. Model 1 reinforcement

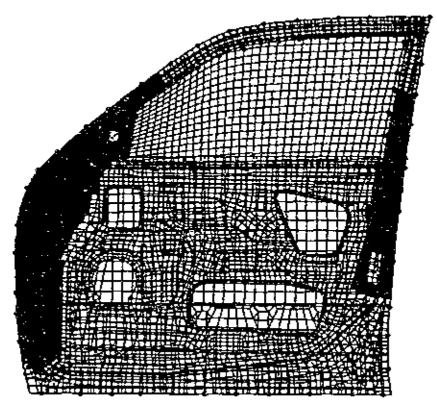

Figure 5. Model 3 reinforcement

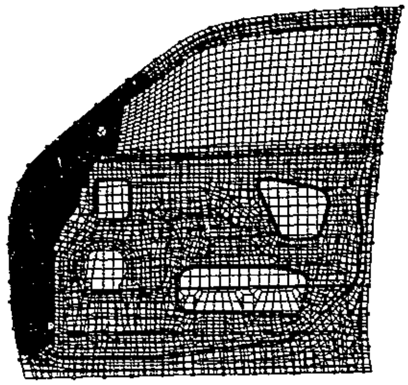

Figure 6. Model 4 reinforcement

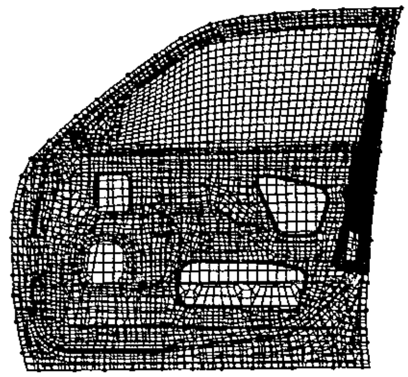

Figure 7. Model 5 reinforcement

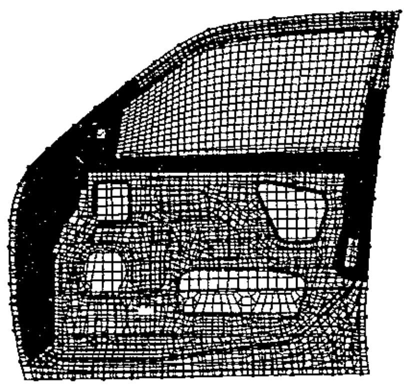

Figure 8. Model 6 reinforcement

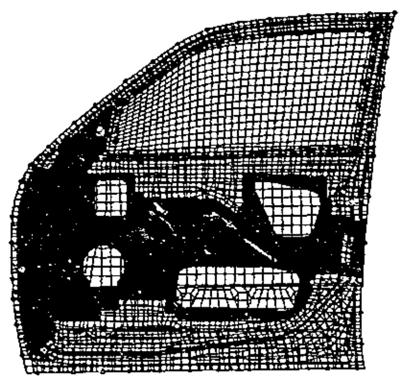

Figure 9. Model 7 reinforcement

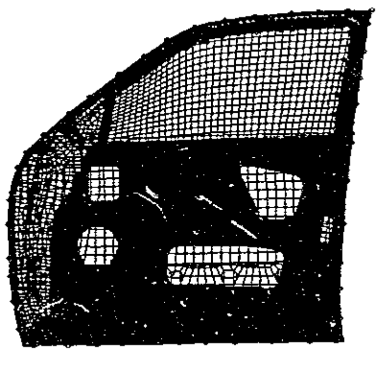

Figure 10. Model 8 reinforcement

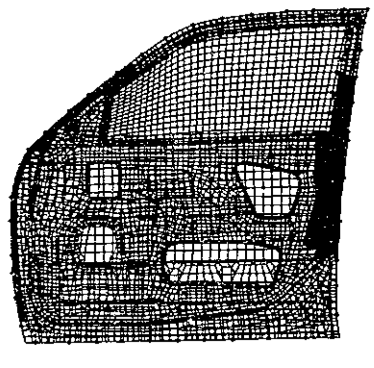

Figure 11. Model 9 reinforcement

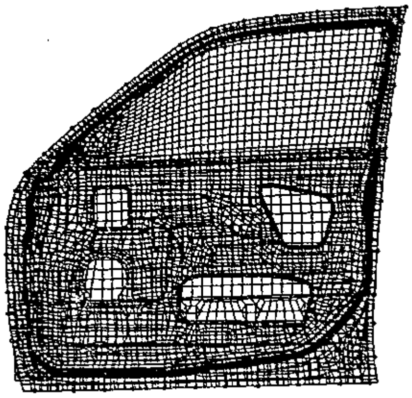

Figure 12. Model 10 reinforcement

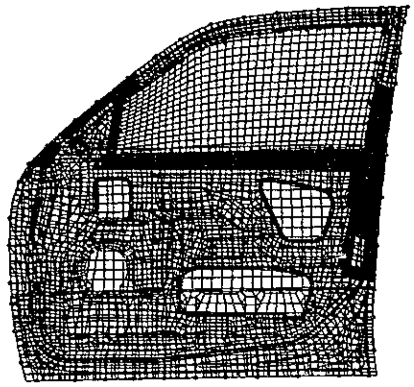

Figure 13. Model 11 reinforcement

Howell demonstrated different C_p values in the below door belt region, but for the analysis of door panel vibration it was decided in this study that the pressures acting over the glass would dominate. The mode shapes found from the ABAQUS solutions confirmed this behaviour. These data are supported by Docton[11] who studied vehicle behaviour under cross wind conditions. The pressure coefficient of 0.5 gives rise to a pressure amplitude used in this study of 0.2363 Pa, and this was applied to all exterior panel surfaces.

Material properties for the steel and glass are given in Table 1. The particular units shown result from the standard use of the millimetre as the unit of length in Hyundai Motor Company.

	Steel	Glass
Young's modulus (kg/mm·sec^2)	2.068×10^6	6.5×10^7
Density (kg/mm^3)	7.827×10^{-6}	2.334×10^{-6}
Poisson ratio	0.3	0.183

Table 1. Material properties

Finally, the damping factor ζ was determined using a simple experimental test on a similar vehicle. The door was given an impact, resulting in a decaying damped free oscillation. The acceleration response was recorded on a storage oscilloscope and the value of $\zeta = 0.08$ obtained from the logarithmic decrement method.

3. Analysis methodology

The comparison of different design options was based on steady state forced vibration analysis. The methodology for assessing the performance of each of the designs was based on a worst case vibration. This was deemed to be for the case in which the pressure oscillations are applied at one of the natural frequencies of the system. For each model, the first three modes of vibration were determined first in an eigenmode extraction analysis. A subsequent analysis step then applied the aerodynamic pressure with constant amplitude to the outer door panel and glass at each of the three natural frequencies in turn. It is assumed that at no other forcing frequency will the displacement response amplitude be so great.

It is appropriate at this stage to examine this methodology and assess its applicability. This has been done in two ways. Firstly, studies have been carried out on one model, in which the analysis was extended to include the first ten eigenmodes. From the displacement response results shown in Table 2, it seems reasonable to conclude that the first three modes dominate in the response calculations. Notice that the displacements are presented for comparative purposes in normalised fashion such that the original design (model 1) is assumed to have unit displacement response.

Eigenmode	Natural frequency (Hz)	Normalised displacement response amplitude
1	35.6	0.407
2	38.3	0.368
3	46.1	0.538
4	54.3	0.114
5	56.2	0.101
6	64.1	0.105
7	67.0	0.156
8	77.3	0.040
9	82.7	0.027
10	83.8	0.025

Table 2. Displacement response amplitudes for first ten eigenmodes for model

Secondly, the frequency of the pressure oscillation may be estimated analytically. Massey[12] derives an expression for the frequency of vortex shedding from a bluff body, the frequency f being given by

$$\frac{f \cdot d}{v} = 0.198\left(1 - \frac{19.7}{Re}\right) \tag{1}$$

where d is a characteristic problem dimension, v is the relative velocity between the fluid and the body, and Re is the Reynolds number describing the flow. For all practical driving speeds, the Reynolds number will be much greater than 19.7, and so the frequency of vortex shedding will be given by

$$f = \frac{0.198v}{d} \tag{2}$$

For the case of a vehicle travelling at 100km/h, this expression predicts a vortex shedding frequency of around 40Hz from a wing mirror. Since this is within the range of the first three modes for every design model, it was further assumed that the analysis methodology was applicable.

4. Results and discussion

The maximum displacement amplitudes found for each of the models are presented in normalised fashion in Table 3, which also shows the eigenmode at which this amplitude occurs. Displacement amplitudes are shown normalised with respect to the amplitude of the initial design, model 1. In addition, the maximum stress has been determined from the finite element results for each model, and this is indicated in the table also, similarly normalised with respect to model 1.

Model	Eigenmode	Normalised displacement response amplitude	Normalised maximum stress
1	1	1.000	1.000
2	1	1.215	1.126
3	2	0.534	0.449
4	1	1.158	1.075
5	2	0.550	0.345
6	2	0.485	0.412
7	1	1.112	1.085
8	1	0.533	0.521
9	2	0.523	0.390
10	1	0.773	0.980
11	1	0.409	0.289

Table 3. Maximum displacement amplitudes and stresses under forced vibration

As expected, the maximum displacements occur primarily in the first mode of vibration, whose mode shape is typically of the form shown in isometric view in Figure 14. It can be seen that the mode shape is dominated by bending deformation in the upper door panel and window glass region. It may then be expected that models with reinforcement around the A- and B-pillar regions around the latch and hinges will be effective in reducing displacement amplitudes.

This behaviour is confirmed by the results for the various models (Table 3), in which it can be seen that the displacement is smallest in models 11 and 6, both of which may be characterised by reinforcement in these regions. Other designs, such as model 3 and model 9, show good stiffness characteristics with a comparatively low level of reinforcement, both these designs concentrating the reinforcement in the pillar sections.

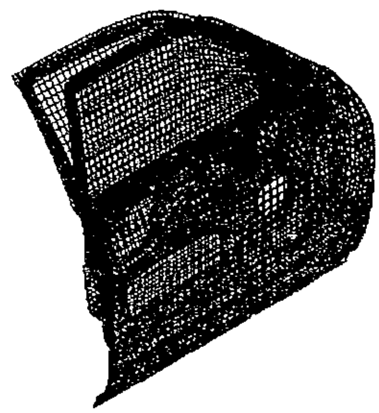

Figure 14. Isometric view of mode shape 1

The results show that models with more reinforcement in total (e.g. model 8) did not always improve the dynamic stiffness performance of the door panel, and that stiffness provided by reinforcement in the A- and B- pillar areas was the most important factor in reducing vibrational deformation. It is also noted that the models having the lowest deformation also exhibit low peak stress values, giving further benefits in durability and fatigue resistance. Closer examination of the results shows also that increasing the reinforcement of the B-pillar region is more effective in reducing response amplitude than increasing the A-pillar reinforcement.

The engineering benefit of increasing dynamic stiffness and thereby reducing 'aspiration' wind noise must also be considered in combination with the weight of reinforcement material required to bring about the increase. Extra weight will detract from the overall vehicle performance, and there is moreover a broad relationship between vehicle weight and cost of production. Table 4 shows the displacement amplitudes of the models considered in relation to the overall weight of the door panel.

The choice of an optimum configuration may be made with the information presented in Table 4. However, the optimum reinforcement can only be designed for a vehicle in relation to its market position. For a luxury car a very quiet, but possibly heavier, panel design may be the optimum, whereas for a high volume, low end, vehicle the engineer may prefer to choose a more lightweight design at the possible cost of more noise at certain driving speeds. In general terms, of the models analysed in this study, it may be concluded that model 11 (figure 13) represents a good solution, having the highest dynamic stiffness with a comparatively low weight (a weight saving of almost 1kg over the original design shown as model 1).

Model	Normalised displacement response amplitude	Eigenmode	Natural frequency (Hz)	Weight (kg)
1	1.000	1	24.3	18.495
2	1.215	1	25.6	15.901
3	0.534	2	39.3	18.488
4	1.158	1	28.6	17.591
5	0.550	2	33.6	16.761
6	0.485	2	40.1	18.884
7	1.112	1	28.2	18.162
8	0.533	1	32.1	21.062
9	0.523	2	38.3	17.777
10	0.773	1	28.8	17.686
11	0.409	1	35.6	17.532

Table 4. Dynamic response results shown in relation to panel weight

5. Conclusions

A finite element investigation has been carried out to study the vibrational stiffness of a car door panel. Various reinforcement configurations have been tested, showing the use of computational analysis to be a very powerful tool in the rapid assessment of a variety of design options to draw conclusions about a sensible engineering optimum. It has been found that by replacing the original reinforcement by a reduced level of reinforcement, but concentrated around the B-pillar door latch region as shown in figure 13, substantial noise reduction may be achieved, while simultaneously reducing weight and also improving durability and fatigue resistance.

6. References

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