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DYNAMIC CHARACTERIZATION OF MOTORCYCLE HELMETS: MODELLING AND COUPLING WITH THE HUMAN HEAD

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Research into the protection of the human head calls for accurate modelling of both the protection system and the head. This study proposes a model incorporating both lumped parameters of the helmet and the head and their coupling during impact. The mechanical characteristics of the shell and of the helmet liner are determined by modal analysis and dynamic compression tests respectively. The coupling of these two components of the helmet is explored using numerical optimization methods based on impact tests which are also used to validate the model. A new dummy head, developed in a previous study and capable of simulating the relative brain–skull displacement was used in the parametric study of the helmet to optimize the density of the polystyrene liner. The ultimate purpose of the study is to devise methods of evaluating the protective aspects of the helmet and then to provide less-expensive methods for optimizing new products on the basis of biomechanical criteria. So far, the study has shown that the optimum density of the liner can be determined not only empirically but also theoretically. It has also shown that optimum helmet parameters depend on the mechanical properties of the dummy head used.

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1. INTRODUCTION

Motor cyclists have been wearing crash helmets intended to protect the head for many years. These helmets have changed considerably in recent decades mainly by the introduction of a polystyrene liner instead of inside straps, but also with a change in the shell material. Thermoplastics (acrylonitrile butadiene styrene copolymer or rubber-toughened polycarbonate) have been replaced by fibre-reinforced thermosetting plastics, such as polyester thermoset resin reinforced with glass fibre (GRP). Current standards are aimed at reducing the acceleration of a "dummy head" which is placed in the helmet for an experimental free fall. Research into head protection systems has investigated the possible use of mathematical models of the helmet for some years, with a view to optimizing the helmets. The general aim is to provide a model that is able to predict the acceleration of the dummy head during impact as a function of helmet characteristics. These studies are intended to evaluate the protective properties of the helmet, but also to provide a less-expensive way of designing new products.

The simplest models that are able to predict the behaviour of the helmet are lumped parameter models. Models of this type, based on experiments carried out using the full helmet or some of its components, have been described by Gilchrist and Mills [1] and Wilson and Carr [2]. In addition to these tests of helmet performance, some studies have

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concentrated on the dynamic response of the polystyrene lining, which ranges from 25 to 40 mm in thickness and has a specific mass of 40 to 70 g/l. Studies of quasi-static compression [3] and dynamic tests carried out at constant energy and variable velocity [4, 5] have shown that the stress–strain curve depends on the density of the polystyrene, but not on the loading velocity for velocities of up to 7 m/s.

In general, models using lumped parameters are not suitable for investigating the geometric aspects of helmets or the stress level in the continuum of materials of which it is made. This is why methods using finite elements techniques are increasingly being used [2, 3, 6]. However, these models encounter validation problems, particularly with regard to the energy absorbed by materials subject to complex constitutive laws and difficulties in modelling the liner–shell interaction. Lumped parameter models can be used to identify the parameters which may affect the performance of the helmets in a simple manner which does not call for high computing costs. Some authors [7] have included geometric properties based on reasonable approximations in their models, such as a locally constant radius of curvature of the shell.

It should also be noted that all the theoretical and experimental simulations of impact to a helmeted head reported in the literature use a rigid mass as a substitute for the head, whereas in reality the head is made up of components which are deformable. Only Enouen [8], who has investigated pedestrian accident reconstruction, changed the headform mass for each impact, so as to reproduce realistic damage on car hoods. The almost universal use of rigid headforms for motorcycle helmet testing is arbitrary and leads manufacturers to design helmets to meet the standards, rather than to minimize injuries.

The approach described in the present study envisages separate modelling of the two main components of the helmet: the shell and the liner. This model is based on vibration analysis of the materials and structures involved, followed by the theoretical and experimental simulation of impacts using a rigid head, as required by the guidelines of the current safety standards. This approach is based on the Gilchrist and Mills lumped-parameter helmet model [1]. Modal analysis is used to identify parameters, and numerical optimization techniques are used to study how the different components interact. Two helmets of very different design and conception are modelled in this study and validated for two levels of impact energies (30 and 60 J). Higher energy impact tests are not conducted, given that the authors hypothesize that minimizing head injury has to be considered at moderate energy. This is supported also by Gilchrist and Mills [1], who feel that “input levels authorized in standards are too high and may not be to optimize head protection”, and that “avoidable head injuries occur in moderate velocity impacts”.

The second part of the study is concerned with the problem of modelling the head itself. A new dummy head is described which had already been developed and which is capable of simulating brain–skull decoupling at 150 Hz. The coupling of this dummy head with the helmet is modelled and validated. Finally, a parametric study considering helmet mechanical properties and using this new substitute for the human head is presented. This study demonstrates the potential of the proposed approach in evaluating and optimizing helmets in terms of various brain injury mechanisms.

2. MODELLING THE HELMET

2.1. GENERAL ASPECTS

The first helmet investigated in this study is a full-face helmet with a composite shell constructed by placing layers of random-in-plane fibre mats (glass (70%)/carbon

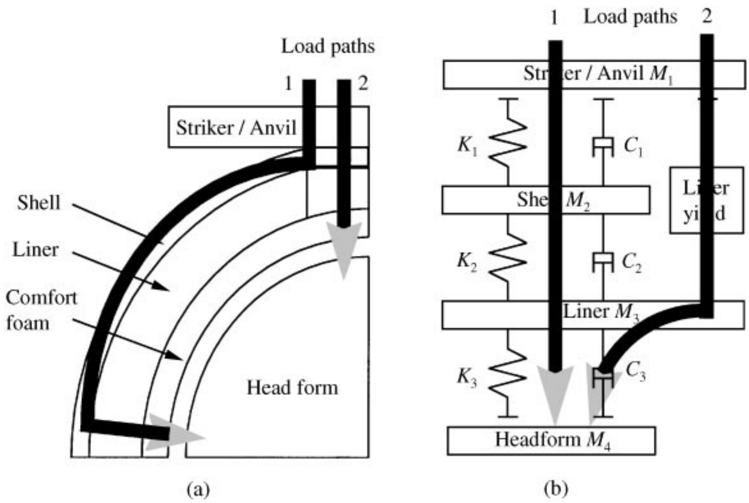

Figure 1. The two load paths between a rigid flat surface and the head (a), and the equivalent lumped model (b), (from Gilchrist and Mills [1]).

(15%)/polyethylene (15%)) before injecting vinylester resin into the mould. The shell thickness ranges from 2·5 to 3 mm. The liner consists of expanded polystyrene with a mean thickness of 20 mm and a mean density of 60 g/l. A layer of highly deformable comfort foam improves the stability of the helmet on the head and allows it to fit more head sizes. The total mass of the helmet is 1·320 Kg.

The load paths between the object impacted and the head are shown in Figure 1(a) and the corresponding model proposed by Gilchrist and Mills [1] is shown in Figure 1(b). Four masses are involved: M_1 the mass of the steel striker or anvil, M_2 the mass of the helmet shell, M_3 the mass of the helmet liner foam and M_4 the mass of the headform.

Load path 1 involves bending of the shell (parameters K_1 and C_1) and elastic deformation of the polystyrene foam (parameters K_2 and C_2), whereas load path 2 represents the direct force–deflection relationship of the crushed polystyrene foam. Both paths involve the deformation of the comfort foam (parameters K_3 and C_3). Movement of all the masses was restricted to the vertical axis, so this is a one-dimensional model.

Our study differs from that of Gilchrist and Mills [1] in the methods used for parameter identification, the helmet impact energies (30 and 60 J), the impact site and, above all, in finally combining the helmet model with a more biofaithful human head model. The next section deals with the experimental characterization and mathematical modelling of the dynamic response, firstly of the different components of the helmet separately, and secondly of the full helmet. The analysis focuses on a description of the occipital impact and various mechanical tests will be proposed to determine the parameters of the lumped model.

2.2. HELMET SHELL CHARACTERIZATION

Modal analysis was used to describe the dynamic behaviour of the shell. This method describes the structure successively in the time, frequency and modal domain. The impact was produced by an impulse hammer fitted with a force transducer at a point i of the shell and the acceleration of point j was recorded simultaneously. Figure 2 shows an example of such a response. The sampling rate was 10 kHz. The ratio of the Fourier transforms of these

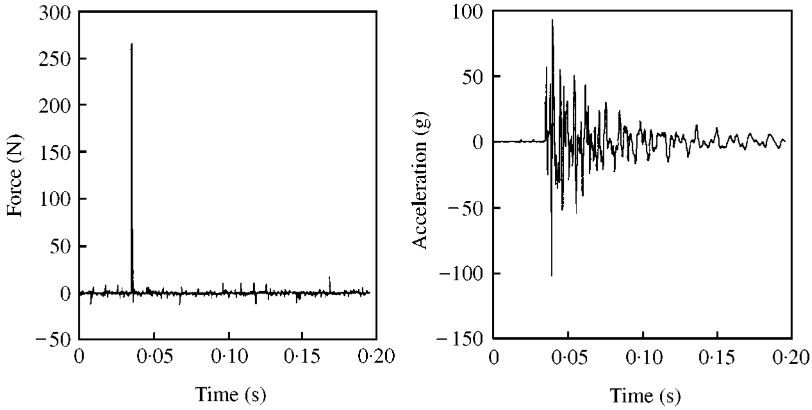

Figure 2. Occipital impulse force and resulting acceleration of helmet shell in the context of a modal analysis.

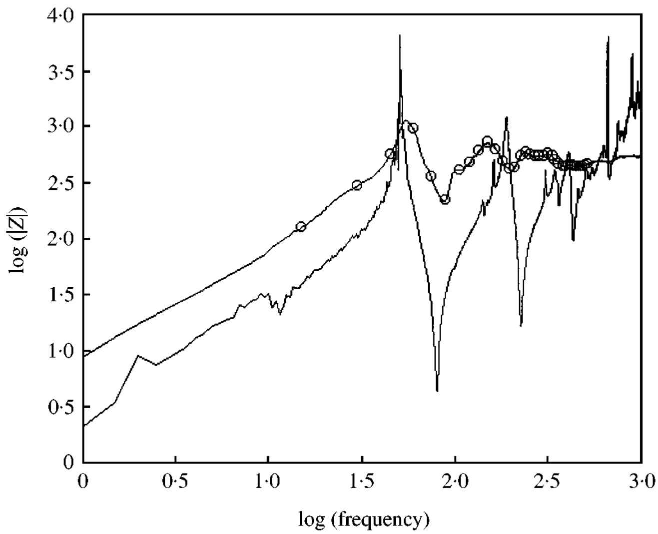

Figure 3. Driving-point mechanical impedance [N/m/s], of the helmet shell and the full helmet as a function of the frequency (Hz): ○, full helmet; —— helmet shell.

two signals gives the transfer function, a complex function of frequency, which can be expressed in terms of apparent mass, mechanical impedance or dynamic stiffness [9]. These frequency responses provide a mechanical characterization of the structure related to points i and j. Figure 3 shows an example of an occipital driving-point mechanical impedance ($i = j$). The linear part of the curve for low frequencies (up to 60 Hz) is typical of mass behaviour. Above 60 Hz, a succession of resonances ($f_1 = 83$ Hz, $f_2 = 200$ Hz, $f_3 = 450$ Hz) were observed which exhibited varying degrees of damping. Modal analysis of the shell was then carried out by successive recordings of several mechanical impedances between points i and j. The accuracy of the description of a continuous structure depends on the number of discrete points analyzed. In this study, 21 points were used and their locations are shown in Figure 4. The switch from the continuous structure of the shell to its simulation by a lumped parameter model involves taking into account a limited number of vibration modes, by attempting to model the modal behaviour of the most prominent

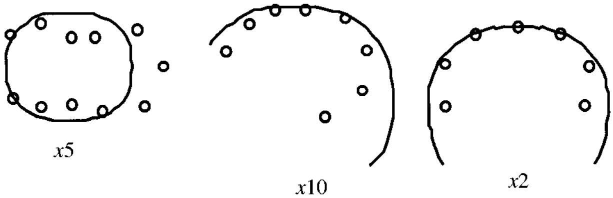

Figure 4. Mode shapes of the shell for the first vibration mode. The multiplication factor refers to the amplification required for good visualization of the mode shape.

modes which occur during occipital impact. The model of the shell is a mass attached in parallel to a spring and a dashpot. The first resonance frequency at 83 Hz, (the first dip in the impedance curve) was examined in detail.

All transfer functions were then analyzed to extract the mode shape for the first vibration mode. The amplitude of this mode shape at a given point was expressed by the imaginary part of the transfer function concerned, at the resonance frequency. This is justified by the fact that the mechanical impedance has a zero real part at the switch to the resonance frequency. In this way, the mode shapes shown in Figure 4 were constructed point by point.

The value of the modal mass is given by the behaviour of the shell at low frequency and is reported as M_2. The impedance in Figure 3 gives a modal mass at low frequencies of 0·52 kg. Given the structure of the model adopted for the shell, its stiffness is represented by

$$K_1 = M_2(2\pi f_1)^2, \tag{1}$$

where $M_2 = 0{\cdot}52$ kg is the mass of the shell, $f_1 = 83$ Hz, the frequency of the first vibration mode.

The stiffness K_1 of the spring is therefore equal to 136 710 N/m, which is about one-tenth of the value reported by Gilchrist and Mills [1] for a similar GRP shell. This can be explained by the fact that for Gilchrist and Mills' shell, rigidity was determined by a helmet compression test in the vertical direction, whereas modal stiffness relative to the first natural frequency for an occipital impact was determined here. This stiffness is not only related to material properties, but is also highly influenced by the structural geometry.

If the damping is assumed to be structural, its value can be determined from the attenuation at -3 dB of the admittance for the mode under consideration. This gives [9]

$$C_1 = \frac{dK_1}{\omega_1}, \tag{2}$$

where "d" is the loss factor defined by

$$d = \frac{\Delta f_{(-3\ \mathrm{dB})}}{f_1} \tag{3}$$

and f_1 is the 1st mode frequency.

This then gives a damping factor $C_1 = 13\ \mathrm{N\,s^2/m}$.

2.3. CRUSHING OF THE POLYSTYRENE FOAM

The behaviour of the liner differs from that of the shell essentially by the fact that it is subjected to local impact and has much less chance to undergo global deformation (see load path 2, "liner yield" in Figure 1). The stress–strain curve of the polystyrene was determined

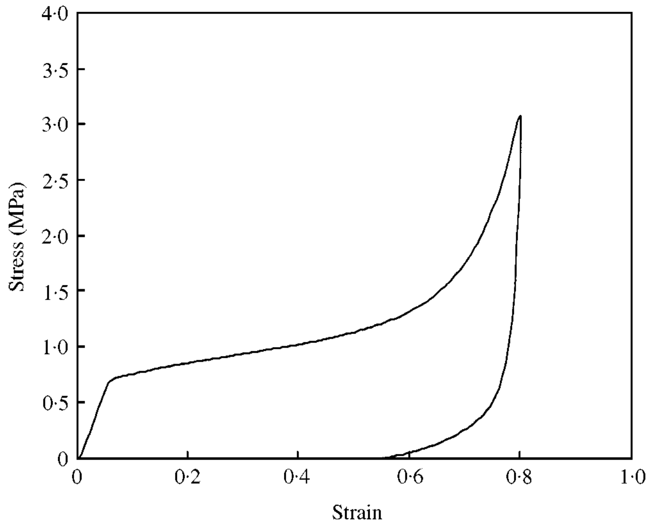

Figure 5. Stress–strain curve determined experimentally using a sample of polystyrene with a specific mass of 66 kg/m^3, at a compression rate of 0·66 s^{-1}.

by quasi-static compression tests of parallelepiped samples cut out of the helmet liner material. These compression tests were then carried out at strain rates of between 0·55 and 3·70 s^{-1}. One example, shown in Figure 5, reveals considerable non-linear aspects of the behaviour of this material. The results obtained are analogous with those reported in the literature [3, 4 10, 11].

The modelling of load path 2, as proposed by Gilchrist and Mills [1], was applied and is described below. The area of impact corresponding to each displacement step must be known in order to determine the force–displacement ratio introduced in the model from the stress–strain curve. This is found by assuming that local curvature of the headform and the helmet are essentially spherical. In the case of an impact onto a rigid flat surface, the important parameter is the outside radius of curvature of the helmet shell, *R*. Impact tests have demonstrated that the deformation is uniform across the entire thickness of the crushed polystyrene [12] and hence that the stress depends only on the radial distance, *r*, to the centre of the impact area. The impact force can then be expressed as the sum of the forces applied to the concentric rings no *i*, for each of which the stress σ_i is different and given by the strain-stress curve (equation (4) and Figure 6(a)). Successive tests have demonstrated that 18 rings are sufficient to provide a good approximation of the integrated force [1].

$$F = \sum_{i=1}^{18} \sigma_i 2\pi r_i \Delta r \tag{4}$$

During impact simulation, the deformation was first calculated at the centre of the impact area for each successive time step in the following way. The total area of impact was deduced from geometrical characteristics, divided into 18 concentric rings, and the corresponding deformation for each of them was calculated. Finally, the stress was determined for each ring from the curve in Figure 5 and equation (4) gives the total force applied to the liner shown in Figure 6b. The parametric study on liner density used the polystyrene stress–strain expression determined by Mills [13] as a function of density given by:

$$\sigma = \sigma_0 + \frac{p_0 \varepsilon}{1 - \varepsilon - d} \tag{5}$$

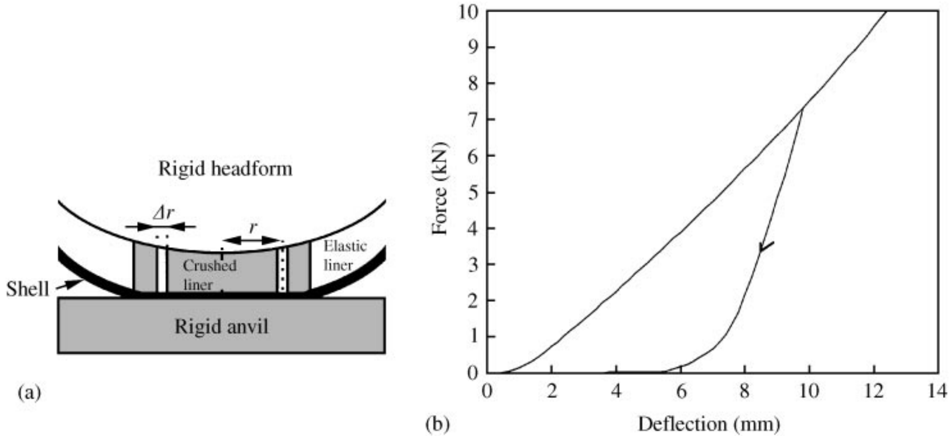

Figure 6. (a) Deformation mode proposed by Gilchrist and Mills [1] for the liner during impact. r is the radial distance to the centre of the impact area and Δr is the width of the ring for which the stress is supposed to be constant. (b) Impact force predicted between a rigid flat surface and a polystyrene liner with a radius of 160 mm, a thickness of 20 mm and a density of 66 kg/m^3, as a function of the central crushing.

where σ and ε are stress and strain, respectively, σ_0 the yield stress at density d, and p_0 the pressure of the gas in the cells in the undeformed foam (0·1 MN/m^2).

2.4. COUPLING OF THE SHELL AND THE POLYSTYRENE FOAM LINEAR

The model of the whole helmet is shown in Figure 1 and it amounts to an assembly of the model of the shell and that of the liner. With the aim of illustrating how the assembled shell–liner works (load path 1), the occipital mechanical impedance was recorded for the whole helmet, and then superimposed on the impedance recorded for the shell alone. This response, which is shown in Figure 3, shows that in addition to the additional mass effect at low frequency, the liner modifies the vibration of the shell considerably due to its damping properties and its stiffness. The shell's first resonance frequency at 83 Hz is still present, but is markedly damped which clearly demonstrates the effectiveness of this coupling. A similar result is observed at 200 and 450 Hz, but these higher modes are not taken into account in the present study.

It is difficult to determine directly the coupling parameters K_2 and C_2 of the shell to the liner because their mutual interaction is highly complex and non-linear. In their paper, Gilchrist and Mills [1] determined these parameters means a vibration mode (at 400 Hz) observed after a crown impact to the helmet–headform structure. This methodology is doubted here because it combines paths 1 and 2 and also because the 400 Hz natural frequency value is probably due to the shell vibration, as shown by the vibration analyses reported here, and is not relevant to the coupling under study.

To evaluate these two coupling parameters, the optimization algorithms provided in the optimization archive of the Matlab software were used. An initial value was assigned to each parameter, and mass M_4 (headform) acceleration was computed for a given impact. This result was then compared with the headform acceleration obtained experimentally. The optimization process then consisted of adjusting the values K_2 and C_2 until the two curves matched closely. This gave the values $K_2 = 1072$ kN/m and $C_2 = 2410$ N s^2 m^{-1}, with a liner foam mass "M_3" of 0·185 kg.

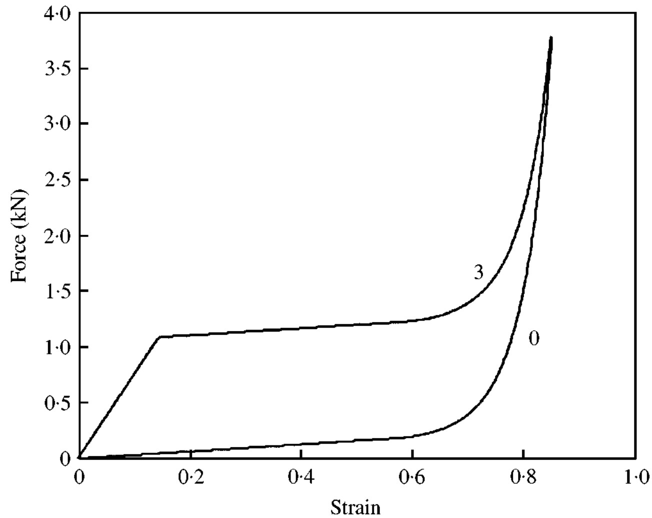

Figure 7. Force- strain curve of the comfort foam for a constant impact area (50 mm in radius) fitted using the equation having $K_3 = 30$ N/m and $C_3 = 400$ Ns^2/m^1 (from [1]). The curves are labelled with the compression rates in m/s ($v = 0$ means quasi-static compression).

2.5. COMPRESSION OF THE COMFORT FOAM

The comfort foam usually consists of polyurethane or PVC foam, and is often covered by a layer of fabric. It is intended to improve the fit of the helmet on the head. It is difficult to measure the stress induced in the foam during impact, because it is of a much lower order of magnitude than that in the polystyrene. Quasi-static compression tests carried out by Gilchrist and Mills [1] showed that the force increased rapidly when the deformation exceeded 80%. The experimental curve obtained by this author has been modelled using the equations

$$F_{34} = K_3(x_4 - x_3) + C_3(V_4 - V_3) \quad \text{if } \varepsilon < 0{\cdot}6, \tag{6}$$

$$F_{34} = (K_3 \exp(K_3 \times 10^{-3}(\varepsilon - 0{\cdot}6)^2))(x_4 - x_3) + C_3(V_4 - V_3) \quad \text{if } \varepsilon > 0{\cdot}6. \tag{7}$$

A dashpot placed in parallel with the spring provided a considerable viscoelastic effect as soon as the velocity increases. The numerical values of $K_3 = 30$ N/m and $C_3 = 400$ N s^2 m^{-1} were obtained from the values found in the literature [7], which led to the law shown in Figure 7 as proposed by Gilchrist and Mills [1]. The space δ, between the skull and the liner is difficult to estimate once the helmet is placed on the head, and it was therefore left as a variable and determined using the optimization algorithm. The value of δ obtained for this helmet is $1{\cdot}5 \times 10^{-3}$ m. The mechanical parameters related to this first helmet are summarized in Table 1.

2.6. VALIDATING THE HELMET MODEL

In this section, the validation of the helmet model by a theoretical simulation of the experimental impact is described. This impact test consisted of fitting a standard dummy head (monomasse type) to the helmet, and then projecting the assembled system against a rigid wall. The headform mass, connected to a rigid neck and the linkage to the pendulum

TABLE 1

Helmet and head model parqameters

Components	Symbol	Helmet 1	Helmet 2	Head	Unit
Shell stiffness	K_1	136 710	85 900		(N/m)
Shell damping	C_1	13	21		(Ns^2/m)
Shell mass	M_2	0·52	0·67		(kg)
Liner stiffness (path 1)	K_2	1 072 000	1 072 000		(N/m)
Liner damping (path 1)	C_2	2410	2410		(Ns^2/m)
Liner mass	M_3	0·185	0·14		(kg)
Comfort foam stiffness	K_3	30	30		(N/m)
Comfort foam damping	C_3	400	400		(Ns^2/m)
Comfort foam thickness	δ	0·0015	0·0014		(m)
Liner thickness	e	0·02	0·02		(m)
External liner radius	R	0·1	0·1		(m)
Monomasse head	M_4			7·8	(kg)
Bimass 150 head					
Skull mass	M_5			6·5	(kg)
Brain mass	M_6			1·28	(kg)
Skull–brain stiffness	K_4			1100	(kN/m)
Skull–brain dumping	C_4			200	(Ns^2/m)

had a total mass of $M_4 = 7{\cdot}8$ kg, which is slightly higher than in standard tests (5 kg in BS6658 and up to 6·1 kg in prEN398). This experiment was carried out using a pendulum apparatus and the velocity prior to impact was 3·9 m s^{-1} for the test at 60 J and 2·8 m s^{-1} for the 30 J impact energy. The tests were carried out in an occipital impact configuration and the acceleration of the centre of gravity of the dummy head was recorded.

Theoretical simulation of this impact was carried out using Matlab software, with a third order Runge–Kutta method in which the acceleration of the head is given by calculating the acceleration of mass M_4. The theoretical and experimental results are shown superimposed in Figures 8(a) and 8(b) for both 30 and 60 J impact energy. The experimental results can be said to be satisfactorily described by the model for the energy levels investigated, even if a delay of about 5 ms is introduced by the model. Some differences in the acceleration shape also subsist, probably due to the complex behaviour of the composite shell, where delamination can occur. This was why Gilchrist and Mills [1] decided that it would be better to present results for the more consistent thermoplastic shells only. The consistency of the model and the proposed methodology for parameter identification was therefore checked by modelling a second helmet design.

A Jet style helmet with a Lexan ML 3459 polycarbonate shell ($e = 3$ mm) and a 20 mm thickness polystyrene foam with a 58 g/l density, was subjected to the same tests and modelled with the above-presented methodology. Details of the model's parameters relative to this second helmet are given in Table 1. Obviously, the main differences observed were those affecting the behaviour of the shell structure, where resonance frequencies at 57, 160 and 251 Hz were recorded with a similar mode shape for the first vibration mode. This shows that the polycarbonate shell is less stiff than the composite shell. This homogeneous, isotropic elastic material is easier to model and as expected, agreement between the numerical and experimental accelerations were found and reported in Figures 8(c) and 8(d) for both energy levels. The comparisons of the fit between theoretical and experimental curves are equivalent to those observed previously, i.e., reasonable acceleration amplitude and duration with a slight delay of the numerical peak value. In addition, the calculated

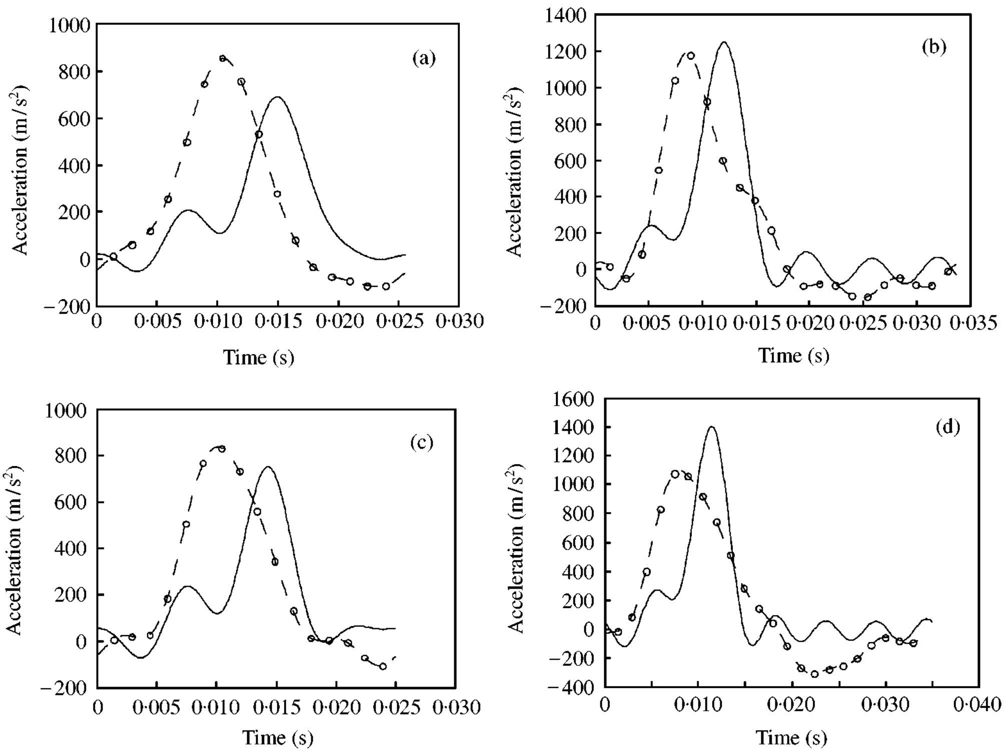

Figure 8. Theoretical and experimental helmeted head acceleration during an occipital impact against a flat anvil, firstly with helmet no. 1 at, respectively, 30 J (a) and 60 J (b) impact energy, then with helmet no. 2, at, respectively, 30 J (c) and 60 J (d) impact energy: ——, model; ○, experimental.

accelerations present a low-amplitude peak at the beginning of the impact, which is not observed in the experimental curve. This artefact may be explained by the difficulty in modelling the head–helmet interaction at early time steps or by a poor understanding of the helmet shell dampening when coupled to the liner. It is hereafter assumed that this first acceleration peak of less then 20 Gs does not fundamentally affect the head response.

The proposed helmet model predicts the level of acceleration of a "monomasse" head during impact with a reasonable accuracy. It can therefore be used to carry out parametric studies and to compare helmets even before the final prototype has been made. In the next section, it will be coupled to a more biofaithful human head model.

3. HELMET – HUMAN HEAD COUPLING

3.1. HEAD MODEL AND COUPLING WITH THE HELMET

This section describes the simulation of impacts using a more biofaithful dummy head, known as the "Bimass 150", to go beyond helmet modelling and evaluate the characteristics of the helmets with regard to the involved specific cranio-encephalic lesion mechanisms. The previously developed new dummy head prototype [14] has the advantage of distinguishing between the mass of the brain and that of the rest of the head, as shown in Figure 9 and Table 1. The monomasse head M_4 is replaced by two masses M_5 and M_6 which represent the mass of the skull and that of the brain respectively. Brain mass

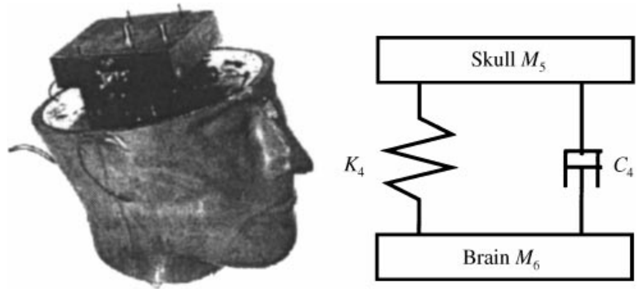

Figure 9. "Bimass150" dummy head and its equivalent lumped parameter model.

M_6 was estimated to be 1·28 kg and skull mass 2·68 kg [14]. However, due to the attachment of this "skull" to a rigid neck and the pendulum linkage device, a value of 6·5 kg was attributed to mass M_5. An elastic dampened linkage ($K4 = 1100\,\mathrm{kN/m^1}$, $C4 = 200\,\mathrm{N\,s^2/m}$) has been developed [14], so that the assembled system has a decoupling frequency between the brain and the rest of the head at about 150 Hz, which matches the values obtained in *in vivo* vibration analyse*s* of the human head [15].

Industrial helmet standardization tests are all conducted using a surrogate head consisting of a single rigid mass (or monomasse) as used in the validation process of the helmet model. Measurement of the linear acceleration of the centre of gravity of this "headform" is used to calculate the head injury criteria (HIC) defined as follows:

$$HIC = (t_2 - t_1)\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2} a_{res}(t)\,\mathrm{d}t\right]^{2\cdot 5},$$

where a_{res} is the resultant linear acceleration expressed in g and the time interval $(t_2 - t_1)$ is determined so as to maximize HIC.

The advantage of the new dummy head used in this study is that it can be used to determine various theoretical and experimental "intracranial parameters" at the moment of impact, such as brain acceleration or the relative brain–skull displacement, parameters which influence the mechanisms by which brain injuries occur. Superimposition of predicted and recorded skull and brain acceleration is shown in Figure 10 for the full-face helmet with a composite shell at 30 and 60 J impact energy. Skull acceleration is estimated with a similar accuracy to the monomasse head acceleration in the previous part of this study. Brain acceleration is much more difficult to simulate mainly because the damping of the brain–skull interface is only known approximately. Nevertheless, trends are respected and the impacts show that brain–skull decoupling occurs and that brain acceleration is significantly higher than skull acceleration.

3.2. PARAMETRIC STUDY AND OPTIMIZATION

In order to consider how the helmet can be optimized, a parametric study is planned of the density of the polystyrene liner, a crucial parameter which is difficult for helmet manufacturers to determine. Impacts were simulated at 60 J, first with the "monomasse" head and then using the "Bimass150" head prototype, with polystyrene densities of between 5 and 60 g/l. The low level of energy compared to the standard test is justified by the fact that several authors [1, 16], including ourselves, question the validity of using high-energy impacts in the standard test. It is often suggested that the limit is too high and that

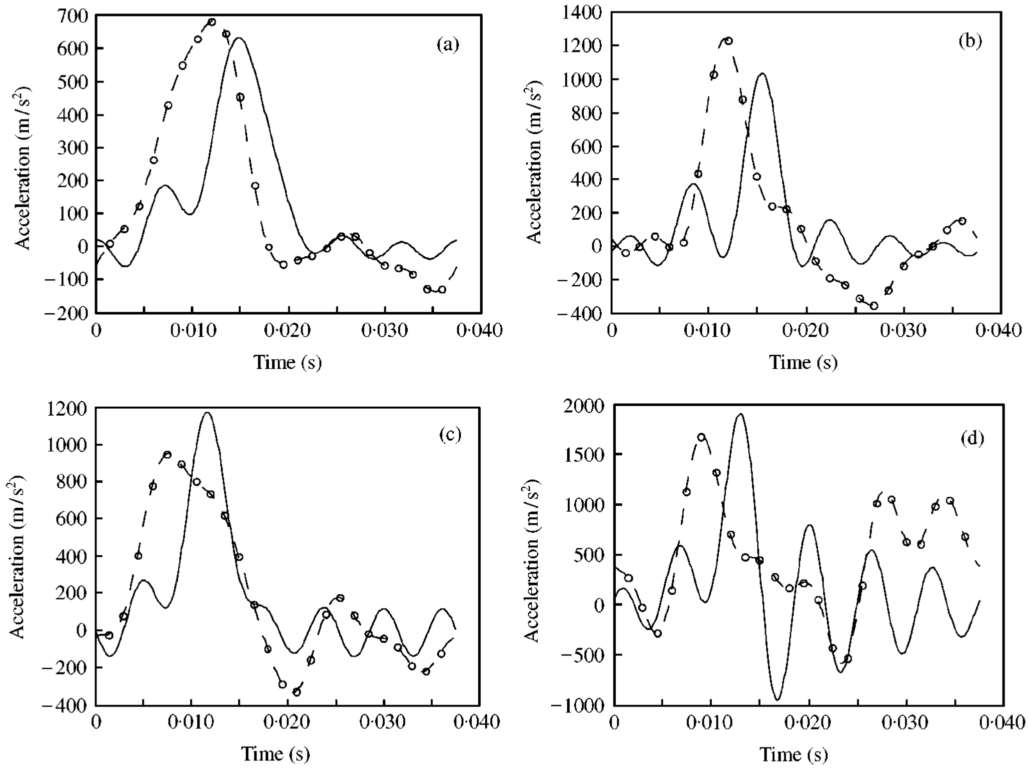

Figure 10. Theoretical and experimental helmeted "Bimass 150" head response under occipital impact (helmet no 1) at 30 J impact energy (skull (a), brain (b)) and at 60 J impact energy (skull (c), brain (d)): ——, model; ○, experimental.

avoidable head injuries could occur in moderate energy impacts. Apart from the problem of human tolerance thresholds, the main objective of this section is to study the headform influence on the optimization process rather than to suggest an optimum helmet. Such a study was also suggested by Gilchrist and Mills [16], when it was stated, in the conclusion, that the use of rigid headforms has led to helmet designs that do not minimize the injuries caused to motorcyclists in crashes.

The present study shows (Figure 11) that the HIC value (obtained with the monomasse headform) is the lowest for polystyrene with a density of 13 g/l. This "HIC-related" optimum density varies from 35 g/l at 115 J to 63 g/l, for a 174 J impact energy. The latter theoretical result is in good agreement with the manufacturer's empirical value obtained at high energy, which is between 50 and 60 g/l. It also fits in with Gilchrist [7], who proposed an optimum polystyrene density of 32 g/l for moderate impact velocities. Figure 11 also shows that the optimum density varies depending on the injury mechanism considered. In particular, for the new head model, the optimum density for lesions caused by the relative brain–skull displacement, or linked to the acceleration of the brain mass, is greater than that obtained when only the acceleration of a rigid monomasse head is considered, for the same 60 J impact energy. This difference of about 30% in the optimum density when the new criteria are considered is also observed in theory when high-energy (174 J) impacts are simulated.

These findings show that the mechanical parameters of a helmet can be considered theoretically and not just empirically, as has generally been the case. This study has

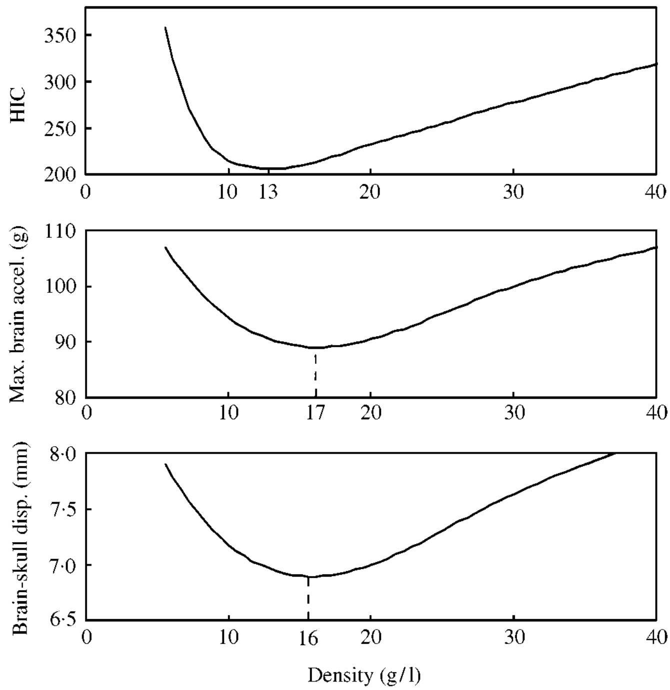

Figure 11. HIC for the "monomasse" head, maximum "brain" acceleration and relative "skull–brain" displacement for the "Bimass150" head, versus polystyrene liner density for 60 J impact energy. The optimum density depends on the headform used, and therefore on the injury mechanism considered.

illustrated how helmet optimization depends on the mechanical characteristics of the dummy head used. In the future, helmet optimization based on biomechanical criteria will require a more "biofaithful" dummy head than the monomasse heads currently in use.

4. CONCLUSIONS

This study describes a new method of characterizing the helmet and its component parts. The dynamic mechanical behaviour of the shell is analyzed on the basis of detailed modal analysis, whereas the liner is investigated more from the point-of-view of the intrinsic material properties of polystyrene. The liner–shell coupling parameters are determined using numerical optimization methods. After identification, these parameters are then introduced in Gilchrist's lumped parameter helmet model. The coupling of the helmet model with a standard "monomasse" dummy head makes it possible to validate reasonably the model for impact simulations at 30 and 60 J which have been subjected to theoretical and experimental analyses. This approach, which is very similar to that used for normative testing, has shown that the acceleration of the head during impact can be predicted with reasonable accuracy. The proposed approach was used on a second, very different, helmet design. The helmet model was then coupled with a more biofaithful physical and mathematical model of the human head developed in a previous study and which enables it to distinguish between the brain and the skull mass, so that intracranial parameters during impact could be calculated. After validation at 30 and 60 J, a parametric study of the helmet liner polystyrene density has identified an optimum density for the various lesion mechanisms. The "HIC-related" optimum, calculated from the acceleration of a monomasse headform, changes as a function of impact energy from 13 g/l at 60 J to 63 g/l at 174 J. If "intracranial" parameters, recorded using the more biofaithful dummy head are taken as optimization criteria, the optimum polystyrene density values are 30% higher then previously.

This study has demonstrated that a simplified lumped model can provide a quicker and cheaper prediction than that provided by an empirical approach to optimizing helmet parameters. Investigations of the new dummy head prototype have shown that optimization of the helmet depends on the mechanical properties of the human head surrogate and that more biofaithful heads should be used in the development and evaluation of protective helmets.

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