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Computational analysis and design of components of protective helmets

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Abstract

Computer methods can assist in understanding the behaviour of the individual components of a helmet, beyond merely the headform output as is usually done in a laboratory environment or for test-house certification purposes. This design study uses a method that we have previously used to analyse the effects of helmet liner material properties. While the helmet liner is of vital importance for energy absorption, other design modifications can also serve to improve its performance. The equestrian helmet model previously developed and analysed by the authors was used in this study. The helmet shell and geometric factors such as a gap between the liner and shell, ventilation holes and ridges on the helmet liner were studied to observe their influence on helmet performance. By studying helmet design variations in terms of different variables other than headform linear acceleration, it is possible to determine which helmet configurations perform better, why they perform the way they do and how efficiently they perform. This can assist the product design and optimization process by suggesting models which would optimise cost, weight and helmet size.

Keywords

Equestrian helmet, helmet, finite element method, FEM, CAD, CAE.

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Introduction

The European equestrian helmet has varied considerably throughout the years. Initially, equestrian headgear provided little or no protection, but with the introduction of standards and their enforcement, headgear has continued to evolve¹. In an attempt to further improve the safety of horse riders, a new standard was released in 2005². This new standard was not intended to replace the previous standard, the last edition of which is 1997³, on the basis that the new standard was meant for high-performance activities. In addition to the 2005 equestrian helmet standard, there is also a British Product Assessment Specification (PAS)⁴ from 1998 which intends to improve upon the official 1997 standard. To date, no helmet manufacturer has released a helmet that conforms to the 2005 standard. This has led to the anomalous situation in which there are two European equestrian

helmet standards and a UK PAS for equestrian helmets, which has stimulated ongoing debate to improve the current equestrian helmet legislation in Europe.

The considerable head injury risks that horse riders face combined with the problems seen in recent standard equestrian helmet test development have led the authors to conduct research on equestrian headgear. While there have been a number of technical studies on protective headgear, those focussing on equestrian helmets have been few and have had limited technical scope.

While many studies have used finite element analysis to simulate protective headgear⁵⁻¹⁴, there are very few that analyse helmet performance based on more than the usual outputs that can also be obtained from laboratory testing¹⁵. The use of computer modelling methods allow helmet designs to be investigated in terms of the performance of individual helmet materials, sections or components in a manner that is not possible in laboratory testing. In laboratory testing, it is only possible to observe the performance of an entire helmet after the impact has occurred. In the best case, high-speed video footage of an impact can give a general idea of the performance of a helmet during impact. In computer modelling, it is possible to complement laboratory tests by analysing the performance of any portion of the helmet during and after the impact. This can be done with a high degree of detail in order to determine the most efficient design configurations for a given purpose.

Usually, the helmet liner is made of expanded crushable polymeric foam. The authors have conducted research concerning the energy absorption capabilities of crushable foam¹⁶ and the influence of crushable foam properties on equestrian headgear performance¹⁷. This material has adequate characteristics for helmets in terms of its price, weight, capacity to form a large variety of geometries, and, of course, impact energy absorption capability. As in many types of protective headgear, the foam liner is the part of the equestrian helmet that absorbs most of the impact energy. It is also the part which is easiest to modify from the manufacturer's point of view. Foam density is directly related to its impact energy absorption capabilities, and foam density can be accurately modified and controlled while manufacturing the helmet liner.

While the foam liner is the helmet component that can be readily modified to change the impact absorption performance of a helmet, it is not the only helmet component that can be modified significantly to alter helmet performance. This study shows how computational methods can be used to study the optimisation process of a helmet design to obtain the best possible combination of helmet components for a given range of impact scenarios.

Materials and Methods

Helmet Model Validation

A comparison with actual laboratory tests on 15 different equestrian helmets¹⁸ was done by simulating flat anvil impacts as specified in the certification tests. Impact speed was 6.26 m/s, as in physical comparison tests¹⁸. The laboratory tests were done on three impact locations on the helmet and the impact simulations were done to mimic these impact locations. The helmet model used is the same one as the one used in previous studied by the authors^{17, 19, 20}.

Helmet Shell Stiffness

The helmet performance was analysed in light of changes to shell stiffness properties. The helmet shell stiffness was varied to assess the influence of this component's stiffness on helmet impact attenuation performance. A representative impact position for the standard flat anvil impact tests was used (Figure 1) and impact simulations were done at 4.4, 5.4 and 7.7 m/s according to the equestrian helmet standards EN 1384:1997 and EN 14572:2005. Five helmet shell stiffnesses were used: 2, 7.25 (baseline), 20, 35 and 50 GPa. These shell stiffnesses represent a range of possible helmet shell materials that are available in the market. The 2 GPa stiffness corresponds to PET, which is common for bicycle helmets. The 7.25 GPa stiffness corresponds to a fibreglass reinforced polymer shell commonly used in equestrian helmets. This is the helmet shell stiffness used for the baseline helmet used throughout this study. The 50 GPa stiffness corresponds to carbon fibre reinforced polymer. Intermediate shell stiffnesses were used to cover the possibility of intermediate helmet shell stiffnesses that could be available. The headform peak linear acceleration, contact area of the shell with the helmet liner and the headform with the helmet liner, Von Mises stress distribution on the helmet liner and average Dissipated Plastic Energy Density (DPED) of the helmet liner are shown and discussed.

Shell Fracture Model

Since ABAQUS/Explicit 6.7 was introduced, it is possible to implement a damage and failure model for fibre-reinforced composites²¹. Even though the helmet liner is the main energy absorbing component on a helmet, an equestrian helmet during impact suffers damage on the helmet shell which assists in the energy absorption process. The fibre reinforced composite model was tested with the helmet model to evaluate its capability of modelling an equestrian helmet shell and to reach a response even closer to that of an actual equestrian helmet during impact.

Comparison of helmet models with and without shell fracture

The performance of the baseline helmet simulated in ABAQUS/Explicit version 6.5 with no shell damage model was compared with the same baseline model simulated in ABAQUS/Explicit version 6.7 with a fibre reinforced shell composite model.

The linear acceleration of both helmet simulations was recorded for the three impact speeds specified by the EN 1384:1997 and EN 14572:2005 standards (4.4, 5.4 and 7.7 m/s). The Von Mises stress distribution was also calculated for the helmet liner for both helmet models and the three impact speeds. A representative (Figure 1) impact position was chosen.

Figure 1: Impact position and direction for helmeted headform drop test simulation.

Four compressive and four tensile tests were done based on the EN 2747:1998²² standard for testing fibre-reinforced polymers. Custom made flat sheets for the shell material were obtained from an equestrian helmet manufacturer who was able to manufacture fibreglass sheets in the same configuration as in their own helmets. The summary results of these tests are shown in Table 1.

	Minimum	Maximum	Mean
Tensile strength (MPa)	31.6	83.9	63.8
Compressive Strength (MPa)	22.4	76.9	44.7
Tensile fracture energy (kJ/m ²)	42.7	216.4	102.8
Compressive Fracture Energy (kJ/m ²)	10.6	105.3	58.7

Table 1: Summary of fibreglass shell material laboratory test results for four compressive and four tensile tests.

Actual helmet tests and simulations with shell fracture model

Two nominally identical equestrian helmets were tested using a drop tower and a custom headform made from cast aluminium. The comfort foam was stripped off the helmets to facilitate the simulations and to ensure that the foam liner, foam block and shell were the only components on the helmet. The headform was bolted at the bottom of the drop tower and the helmets were placed on top of the headform. A flat striker mass of 7.764 kg was dropped onto the helmet. The impact force was recorded through a force cell located within the striker assembly. The drop parameters were chosen to keep the same energy levels as those of the drop test specified in EN 1384:1997. With this in mind, the corresponding drop height was set to 1 m, giving a drop speed of 4.4 m/s. Simulations of the same setup were done by modifying the geometry of the baseline helmet to match the geometry of the tested helmets as closely as possible. Impact location and angle of impact was the same as for the drop tower tests (Figure 2). The foam liner density used was 64 kg/m³. The helmet shell fracture model was used, and the parameters were varied within the observed limits (See Tables 1 and 2) to obtain a realistic force-displacement behaviour.

Figure 2: Schematic of drop tower test arrangement.

The parameters for the shell material used for the simulation (taken from datasheet values and laboratory tests) are shown in Table 2. The shell material of this particular helmet manufacturer consisted of an epoxy (Synolite 2020-P-1) resin matrix with a fibreglass bi-directional weaving and chopped short fibreglass fibres.

Property	Value
Density (kg/m^3)	1200
Young's Modulus (GPa)	7.25
Shear Modulus (GPa)	2.79
Poisson's Ratio	0.3
Tensile Strength (MPa)	52.7
Compressive Strength (MPa)	45.7
Shear Strength (MPa)	26.4
Tensile Fracture Energy (kJ/m^2)	129
Compressive Fracture Energy (kJ/m^2)	59.7

Table 2: Material properties defined for ABAQUS/Explicit Hashin composite shell fracture model.

Air Gap Influence

An equestrian helmet can have a space between the foam liner and the outer helmet shell. This air gap is formed by separating the foam liner and shell by using an elastic foam block and binding the helmet liner and the helmet shell to this foam block. The foam block therefore acts as the element that binds the components of the helmet together. This particular helmet design is characteristic of racing helmet designs having a glass fibre reinforced shell, with varying foam block thicknesses and densities. Nevertheless, it is acknowledged that many other helmet designs do not follow this design (especially leisure racing helmets), opting instead to attach the liner directly to the shell at the crown, or to leave an air gap at the crown area without any foam block and to attach the liner to the shell at the lower sections of the helmet. The influence of the air gap on helmet performance was studied by comparing the baseline helmet to a helmet with all characteristics being the same as the baseline but without the air gap.

Features on Helmet Liner

Two helmet liner types were generated based on the baseline helmet model (Figure 3). 20 mm diameter holes were created on the intended area of impact of the helmet liner for one case, and a ridge of 7 mm thickness, 15 mm wide and 61 mm in length was created for the second case. The features were added on the impact site corresponding to the impact position (Figure 1). The helmet shell was kept the same as for the baseline helmet. Simulations were done using the 45 Side impact positions for three impact speeds (4.4, 5.4 and 7.7 m/s). Linear acceleration, contact area, Von Mises stress volume proportion and average DPED were calculated for both cases and compared to the baseline helmet.

Figure 3: Helmet liner with ridge (left) and ventilation holes (right).

Combination of Design Modifications

Combinations of helmet modifications were studied to see the possibilities to improve helmet impact attenuation performance to achieve a helmet configuration that reaches closer to the minimum EN 14572:2005 standard requirements. Each design change contributes to the helmet's performance. There is a relationship between all helmet components where one component change will affect its own behaviour as well as the behaviour of other unchanged elements. Combinations of more than one helmet modification will be tested against the baseline helmet model.

Analysis of volume stressed within plateau stress range

The foam liner material used in commercially available equestrian helmets is typically expanded polystyrene (EPS) foam. EPS foam is popularly utilised in energy absorbing structures. The typical stress-strain behaviour of EPS is shown in Figure 4. The curve is tri-linear in form, corresponding to elastic (I), plateau (II), and densification (III) stages. It is more efficient that the foam liner absorbs energy within the plateau stage as the stress remains nearly constant over a large strain. One of the unique aspects of this study is that the quantity of foam stressed within the plateau range (or above the plateau range) was calculated to study how efficiently the foam was used in the helmet. The foam is considered to “bottom-out” when it has exceeded the plateau stress range.

Figure 4: Typical stress-strain behaviour of EPS.

Results and Discussion

Helmet Model Validation

It is seen from Table 3 that the simulation results lie within the range of linear accelerations obtained from the laboratory impact tests. Some actual test curves and the simulation results curves are shown in Appendix A.

Impact Position	Linear acceleration (g)	
	Laboratory test (95% CI)	Simulation
Front	243.7 $\pm$ 73.2	232.3
Side	287.1 $\pm$ 71.5	231.2
Crown	231.7 $\pm$ 81.6	252.7

Table 3: Summary results of simulation and laboratory helmeted headform test results.

Helmet Shell Stiffness

Linear Acceleration

Table 4 shows that for the lower two impact velocities, the lowest shell stiffness of 2 GPa led to the lowest linear acceleration values. For the high energy impact case, the lowest accelerations corresponds to the stiffer 20 GPa shell, but the acceleration values were very similar to those of the 7.25, 35, and 50 GPa stiffness shells. The higher accelerations for the two lower impact speed cases were given by the highest shell stiffness, while for the high impact energy case the highest accelerations were given by the lowest shell stiffness.

Shell Young's Modulus (GPa)	4.4 m/s		5.4 m/s		7.7 m/s	
	Peak Accel. (g)	Peak Accel. (g)	Time Acc > 150g	Time Acc > 150g	Peak Accel. (g)	Time Acc > 150g

2	134.4	175.1	1.7	1995.7	1.6
7.25	166.7	199.2	2.3	316.7	3.5
20	191.4	228.2	2.6	315.9	3.6
35	192.8	233.3	2.6	317.9	3.3
50	199.6	243	2.7	320.7	3.4

Table 4: Linear acceleration for helmets of different liner stiffnesses at three impact speeds.

Contact Area

The inner and outer contact areas of the foam liner are shown for three impact speeds in Table 5. The inner and outer contact areas increased as the shell stiffness increased. Stiff and compliant shells perform favourably for respectively high and low impact energies.

	4.4 m/s		5.4 m/s		7.7 m/s	
Shell Young's Modulus (GPa)	Inner	Outer	Inner	Outer	Inner	Outer
2	208	286	234	328	279	421
7.25	232	319	263	370	305	439
20	237	344	275	397	339	471
35	244	351	282	414	335	485
50	249	376	287	413	333	471

Table 5: Contact area on the helmet liner (cm²) inner and outer surface for different impact speeds. Liner density is 64 kg/m³.

Figure 5 shows how the Von Mises stress is distributed on the helmet liner surface for two very different shell stiffnesses at the same impact speed at the time of peak acceleration. The stiff shell spreads the load on a larger area, which reduces peak stress values, while the compliant shell concentrates the load on a very small section of the helmet. This is reflected by the results seen in the contact area measurements and the following results in Von Mises stress distribution and average DPED results.

Figure 5: Contour plots at peak headform acceleration for Von Mises stress for 50 GPa stiffness shell (left) and 2 GPa stiffness shell (Right)

Von Mises Stress Distribution

The Von Mises stress distribution of the helmet liner 64 kg/m³ density foam stressed within the plateau range is shown in Figure 6(a) - 6(c). The low shell stiffness allows the foam liner to be stressed at higher levels within the plateau range for the low and middle impact speed cases. For the high energy impact, the higher shell stiffness allows the liner to be stressed at higher levels within the plateau range.

(a)

(b)

(c)

Figure 6: Von Mises stress distribution for the liner foam for stresses within the plateau stress range (450-970 kPa) for different shell stiffnesses (Young's Modulus in GPa) for 4.4(a), 5.4(b) and 7.7(c) m/s impact speeds. Percentage is over total form volume stressed within plateau stress range.

DPED

The average DPED for the whole helmet liner is summarised for the three impact speeds and shell stiffnesses in Table 6. For the low speed impact case the highest DPED was for the 50 GPa shell and the second largest was for the 2 GPa shell. For the medium speed impact the highest DPED corresponded to the 2 GPa shell and the second highest was that of the 50 GPa shell. For the high speed impact case the highest DPED corresponded to the 2 GPa shell and the second highest was for the 7.25 GPa shell.

Shell Young's Modulus (GPa)	Average DPED (J/m ³)		
	4.4 m/s	5.4 m/s	7.7 m/s
2	23261	58773	173200
7.25	20172	38804	142450
20	16702	39708	121520
35	21403	39834	117550
50	25160	42798	124680

Table 6: Average DPED for the helmet liner for different shell stiffnesses and impact speeds.

Shell Fracture Model

The linear acceleration was noticeably lower for the helmet for the helmet with the shell failure model for the low and medium speed impacts (Table 7). The damage model implemented with the helmet shell was seen to considerably increase linear acceleration for the high speed impact.

	4.4 m/s	5.4 m/s	7.7 m/s
Shell model type	Peak Accel.	Peak Accel.	Peak Accel.
Elastic (Baseline)	166.7	199.2	316.7
With Hashin failure	147.7	177	1190

Table 7: Linear acceleration (g) for helmets with different shell models at three impact speeds.

The Von Mises stress distribution plots for the foam liner for the helmet with and without a shell fracture model are shown in Figure 7. For all impact speeds, it is evident that the amount of foam stressed within the plateau stress has shifted towards higher stress levels. For the low and medium impact energy levels, the shift towards higher stress values is maintained within the plateau stress range. For the high speed impact case (Figure 7(c)), there is also a shift of the amount of material involved in energy absorption to higher levels.

(a)

(b)

(c)

Figure 7: Von Mises stress distribution for the helmet liner foam within the plateau stress range (450-970 kPa) for helmets with and without fracture model for 4.4(a), 5.4(b), and 7.7(c) m/s impact speeds. Percentage is over total foam volume stressed in plateau range.

When the impact energy dissipated by the liner is calculated for the helmets with and without damage model it is seen that the energy dissipated by the helmet liner for both the low and medium impact energies is practically the same (Table 8).

	4.4 m/s	5.4 m/s	7.7 m/s
Shell w. fracture	19.15	33.19	67.12
Baseline w/o fracture	19.25	33.4	79.5

Table 8: Plastic energy dissipated (J) by the helmet liner for different impact speeds for helmet models with and without damage model.

For the high speed impact case, the energy absorption of the helmet liner is lower than that of the helmet with no damage model.

When the contact areas are calculated (Table 9), the influence of the shell stiffness degradation due to the damage model becomes clearer. The lower stiffness of the shell with damage generates smaller contact areas than the shell with constant stiffness with no damage.

	4.4 m/s		5.4 m/s		7.7 m/s	
	Inner	Outer	Inner	Outer	Inner	Outer
Shell with fracture	212	287	244	333	267	394
Baseline w/o fracture	232	319	263	370	305	439

Table 9: Contact area (cm²) of the helmet liner for different impact speeds for helmet with shell with and without damage model.

The Hashin tensile matrix initiation criterion contour plot is compared to the actual shell matrix damage in Figure 8 showing good agreement between the simulated damage behaviour and actual damage.

Figure 8: Actual test shell damage see from the shell interior (top) and simulation damage initiation contour plot (bottom). Dimensions of the foam block are 62x56 mm.

The force-displacement results show a noticeable variability between the two helmets, despite being tested in identical fashion (Figure 9). Despite this, the simulation force-displacement curve shows good agreement when compared with the two helmet tests.

Figure 9: Force-Displacement curves obtained from helmet drop test and curve from simulation.

Air Gap Influence

The presence of the air gap in the baseline helmet seems to contribute significantly to the linear acceleration reduction. The accelerations are lower than those without the air gap by 13, 15 and 21% for the low, medium and high speed impact cases respectively. The time that the acceleration exceeded 150 g for the medium and high impact speeds was also less. The headform acceleration time history is shown for the three impact speeds in Figure 10. The plots clearly show that the duration of the acceleration curve has been expanded across a longer time; this in turn means that the energy was dissipated more slowly which decreased the peak accelerations. The acceleration-time pulses were prolonged by 2 ms. As the impact time increases for the same impact energy, the peak acceleration decreases.

Figure 10: Linear acceleration time history for helmets with and without air gap for 5.4 m/s impact speed.

The time-history of the contact area was calculated (Figure 11) to show how the foam liner interacted with the internal shell surface during the impact. The contact area on the helmet liner without the air gap is greater than that of the baseline helmet. This does not really come as a surprise, as the helmet shell in the helmet with no air gap is practically touching the complete outer surface of the helmet liner even before the impact. As the load is spread more evenly on the helmet liner on its outside surface, the contact area on the inside of the helmet liner is greater as well.

Figure 11: Helmet liner inner and outer contact area time history for helmets with and without air gap for 5.4 m/s impact speed.

Geometric Features

The linear acceleration is slightly reduced when the ventilation holes or the ridge were included on the helmet liner for the low and medium impact speed cases (Table 10). It seems that adding ventilation holes increases the impact dissipation capacity for lower impact energies, while for the high impact case it actually slightly decreases it. The ridge was seen to be effective at low impact energies as well, but it did not change performance for the high energy impact.

	4.4 m/s	5.4 m/s	7.7 m/s
	Peak Accel.	Peak Accel.	Peak Accel.
Helmet Type	(g)	(g)	(g)
Baseline	166.7	199.2	316.7
Vent. Holes	159.8	190.8	327.5
Ridge	156	192.1	316.5

Table 10: Linear acceleration results for helmets with different features.

The contact area is slightly decreased for both ventilation holes and ridge cases, but more so for the ventilation holes (Table 11).

	4.4 m/s	5.4 m/s	7.7 m/s
Helmet Type	Outer Area	Outer Area	Outer Area

Baseline	319	370	439
Vent. Holes	309	350	431
Ridge	313	367	433

Table 11: Contact area (cm²) results from helmet with different geometrical features.

The ventilation holes and the ridge effectively decreased the contact area and in turn increased the stresses on the helmet liner. For the low and medium speed impact cases, the amount of energy absorbing foam which was stressed at higher levels within the useful plateau range increased, which explains the slight improvement in impact attenuation performance (Table 12). For the high energy impact case, the amount of bottomed out foam was increased for the ventilation holes and ridge cases.

	V.M. Stress Upper 1/2	V.M. Stress Upper 1/3	Bottomed out foam
Helmet Type	4.4 m/s	5.4 m/s	7.7 m/s
Baseline	0.44	0.61	23.4
Vent. Holes	0.77	2.02	25.8
Ridge	6.48	7.84	30.1

Table 12: Quantity of foam (cm³) deformed within upper half and upper third of the plateau stress and bottomed out foam.

The average DPED results show that the baseline helmet liner has a higher average energy dissipation density than the ventilation hole and ridge cases (Table 13).

	Average DPED (J/m ³)		
Helmet Type	4.4 m/s	5.4 m/s	7.7 m/s
Baseline	20172	38804	142450
Vent. Holes	18723	33033	133440
Ridge	19934	36774	123190

Table 13: Average DPED results for helmets with different features on the helmet liner.

Discussion

This study has shown how helmet design changes can be analysed through FE modelling beyond just replicating the methods used in laboratory helmet testing. FE modelling methods and computer models in general, allow the study of many variables in a controlled manner. In previous research on helmets, FE modelling has been applied already to study the influence of helmet design variables, such as the shell¹⁵ and liner^{7, 10}. In these studies the output variables observed are from a head model, such as the kinematics of a headform model, or the loads on the brain tissue model from a human head model.

It is possible to understand the reason why the design changes on a helmet actually affect the outputs from a head model through the use of FE modelling. If these mechanisms are thoroughly understood, it would be possible to use this knowledge to optimise helmet designs in a much more focused and methodical manner.

Helmet Shell Stiffness

Linear Acceleration

It seems that optimizing a helmet for high and low energy impact cases leads to contradictory solutions as the best configuration for the lower impact speeds is the worst performing for the high impact speed. The best performing shell stiffness overall is the baseline stiffness of 7.25 GPa, since it gives the lowest acceleration values for the lower impact speeds and it still gives a relatively low value for the high speed impact. Nevertheless, the acceleration values are still too high for the EN 14572:2005 linear acceleration requirements. The best performing shell configuration gives an acceleration twice the required amount for the low energy impact, while for the high impact speed case the acceleration is 66 g higher than the allowed upper acceleration limit.

Contact Area

If compliant and stiff shells were to be used for high and low energies respectively, a stiff shell at a low impact energy would spread the load too much on the helmet liner and would not allow the liner to absorb energy. This in turn leads to high acceleration values. A compliant shell at a high impact energy concentrates the load on a small area which would yield excessively if the impact energy is sufficiently high. This would also lead to undesirable high levels of acceleration.

Von Mises Stress Distribution

The results shown in Figure 6 confirm the previous observations when linear acceleration and contact area were analysed. The low stiffness shell improves the liner performance for the lower impact energies while it worsens performance for the high impact energy. The Von Mises stress distributions show a clear difference between the 2 GPa stiffness shell and the higher stiffness shells. This difference is not as evident between the shells of stiffnesses between 7.25 and 50 GPa.

DPED

The high DPED value for the 2 GPa shell in the low impact case together with the linear acceleration results shows that the shell contributed to a more efficient use of the foam material available. The high acceleration combined with the high DPED value for the 50 GPa shell shows that a considerable amount of foam was deformed, hence the high average DPED value, but the foam did not deform sufficiently to absorb as much energy as it did for the 2 GPa shell case. The energy was spread over a larger proportion of helmet liner volume, hence the large average DPED value as it is the average value for the entire helmet liner, but this did not contribute to reduce acceleration as the liner was not allowed to deform sufficiently to absorb enough energy. A similar situation was seen for the medium speed impact, where the high DPED values analysed along linear acceleration results show that the 2 GPa shell improved the efficiency of the helmet liner usage, while the high DPED value for the 50 GPa shell was due to the spreading of the energy absorption over a larger proportional volume. In the case of the high speed impact the high DPED value alongside the high acceleration value for the 2 GPa shell shows that the foam material suffered excessive deformation to the point of “bottoming out”. In the meantime, the high DPED value for the 7.25 GPa shell helmet shows that

this configuration exhibited a high efficiency of foam usage in terms of energy absorption and deformation.

Shell Fracture Model

Results shown could suggest that the foam liner material has been used more efficiently as in the low and medium speed impact cases. However, from Figure 7(c) alone it is not possible to see how much foam material has bottomed out. When this amount of material is calculated, it is seen that the amount of energy absorbing foam liner which has bottomed out is higher than that of the helmet with no shell fracture model (33 cm³ vs 23 cm³). This explains the considerably higher acceleration seen in Table 7 for the high speed impact.

The shell damage model allows for the helmet liner to be engaged at higher stress levels due to the stiffness degradation of the shell during the damage process. Stiffness degradation adversely affects the impact attenuation capability of the helmet liner at higher impact speeds. For the helmet to sustain impact energies effectively, a stiffer shell should be utilized to compensate for damage degradation. The shift of the proportion of liner material to higher stress levels is similar to the one seen in the helmet liner when a low shell stiffness with no damage model.

The Von Mises stress distribution plots indicate that the foam has been used more efficiently for the helmet model with the shell damage model, but this alone does not explain the lower linear accelerations, since the shell in this case also absorbs impact energy.

The difference in energy absorption lies with the absorbed energy due to damage of the helmet shell. This additional absorbed energy is what ultimately contributes to the reduced acceleration. This indicates that the Von Mises stress distribution change should be interpreted in the sense that a larger quantity of foam liner has been stressed at higher stress levels, but since the energy absorption is the same, this indicates that the helmet liner was used more effectively in some regions, but less energy has been absorbed in other liner sections. The stiffness degradation effect of the fibre reinforced composite damage model allowed some regions of the liner to deform more, but as it also absorbed energy, other regions on the helmet liner did not absorb as much energy as before.

The degraded stiffness of the shell did not allow the load to spread more evenly on the helmet liner, therefore there was more bottomed out material and the total energy absorbed by the helmet was higher. The fact that the helmet liner foam was stressed to a higher degree due to the degraded shell stiffness led to excessively high accelerations. A very small area of the liner that completely densifies is sufficient to generate undesirably high accelerations.

Results shown in Table 9 show that the smaller contact areas led to higher contact stresses which in turn lead to larger deformations and regions of the liner stressed at higher levels.

The variability of the impact test results comes mainly from the helmet shell, as the foam manufacturing process is highly regulated while the shell manufacturing process is still done by hand and the composition of the shell material (amount of fibres, fibre direction and thickness of resin)

varies within each and between the helmet shells. This was also observed from the material test results shown in Table 2.

Air Gap Influence

A much larger contact area is reached more quickly for the helmet with no air gap. While the baseline helmet takes 6 to 7 ms to reach the maximum contact area it takes 3-4 ms for the helmet with no air gap. This means that the helmet liner surface for the helmet with no air gap is engaged more quickly, which in turn could not allow a sufficient amount of foam liner to yield more extensively or to absorb more energy. The air gap has the effect of delaying the impact acceleration duration not only because of the additional deformation and travelling distance it causes, but also because it allows the helmet liner to be involved in the energy absorption process in a more gradual manner. This has important implications in helmet design, as the helmet performance could be altered by changing the speed at which the liner is engaged for a given impact speed.

Geometric Features

In the case of the ventilation holes, the lower contact areas seen are explained due to the actual decrease of surface area of the liner. For the ridge case, even though there is no surface area subtracted like in the ventilation hole case, the ridge does not allow full contact between the area of the liner which is impacted and the helmet shell.

Results suggest that the stresses, while higher, did not go beyond the upper limit of the plateau range, for both medium and low impact speeds. This explains the reduced linear accelerations for these cases.

For the high speed impacts, the amount of bottomed out foam increased. This explains the increase in impact acceleration for the ventilation hole case. For the ridge case, the amount of bottomed out foam is higher since the ridge completely collapses, but the accelerations are kept practically the same, since there was still foam material available for energy absorption.

DPED results might seem contradictory with respect to previous results found in regards to Von Mises stress volumetric proportions, but this is not the case. The lower DPED values for both the ridges and ventilation holes in conjunction with the higher Von Mises stress values means that even though the helmet portion stressed at higher stress was larger, the stress was concentrated more, which indeed lowered the average value of the DPED for the whole helmet liner. The DPED revealed that even though the impact attenuation was improved for the low and medium speed impacts, a smaller portion of the whole helmet liner was actually engaged in absorbing energy. The ventilation holes and ridge on the helmet liner both concentrate the load on a smaller section of the helmet, while increasing the stress level on that particular area. From the current results it is seen that if the load is concentrated in a sufficiently small area, the DPED is lowered despite having larger stresses present on the helmet liner.

Conclusions

The helmet shell was seen to greatly assist helmet performance by distributing the load on the liner, and this was quantified through the analysis of contact areas and stress distributions. It was seen that the lowest shell stiffness tested (2 GPa) led to the lowest linear acceleration values. For the high speed impacts, the lowest accelerations corresponded to the 20 GPa stiffness shell. Fracture of the helmet shell adds another factor to consider in helmet optimization, as the induced shell fracture had the effect of degrading shell stiffness. This had the effect of reducing linear accelerations for lower impact speeds, but had the opposite effect for high speed impacts. This can be used to increase helmet performance if appropriate combinations of shell stiffness and liner density are chosen in light of the results presented in this thesis. The gap between the helmet shell and the liner assists in delaying the impact duration and allows a more gradual use of the helmet liner. Of a total impact duration of 8-9 ms, peak accelerations were delayed from 2 to 4 ms, while maximum contact area was delayed around 3 ms. Adding other features to the helmet such as ventilation holes or ridges on the liner can assist in increasing helmet performance. The lower contact areas due to small added features (around from 3 to 5% lower) force higher stresses on the liner. This effect could be used to improve liner performance locally where necessary and create a layered liner behaviour without actually changing liner density.

The power and potential of applying computer modelling for helmet research is just beginning to be utilized. Equestrian helmets have improved considerably throughout the years, but empirical methods are limited in their capacity to advance equestrian helmets. From that point on, analytical research techniques can be employed to analyse helmet performance in detail and assist the design process.

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Appendix A: Sample test results¹⁸ and simulation results (bottom right).