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Use Of Advanced Composite Materials In The Construction Of Suspension Push-Rods For A Formula One Racing Car

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Summary
Advanced composite materials, particularly carbon fibre-reinforced epoxies, are used extensively in the construction of contemporary Formula One racing cars because of their high specific stiffness and strength properties. The present Chapter is concerned with one significant load-bearing component, namely the suspension push-rod. These push-rods exist at the four corners of a car and link the monocoque, engine and gearbox casing to a racing track and, as such, they are subject to severe inertial forces and dynamic compressive and flexure forces.

The design, manufacture and in-service behaviour of carbon/epoxy push-rods are discussed in the present Chapter. Both uniform and tapered layups of unidirectional and woven cross-ply prepreg have been used to manufacture push-rods of aerofoil cross-sections. These structural components have been loaded to catastrophic failure and the identified damage mechanisms were seen to include fibre micro buckling, fibre kinking and fibre fracture. A comparison has been made between the actual and theoretically predicted failure strains and it is seen that buckling was the ultimate cause of failure in all cases. The ultimate strength of the carbon/epoxy composite material was greater than the compressive strength of the push-rods and consequently, scope exists for improving the performance of the present design of composite push-rods.

Symbols

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<tr>
<td>a</td>
<td>half-major axis of ellipse</td>
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<tr>
<td>A</td>
<td>cross-sectional area of push-rod</td>
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<td>b</td>
<td>half-minor axis of ellipse</td>
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<td>E</td>
<td>Young’s modulus</td>
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<td>G</td>
<td>shear modulus</td>
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<tr>
<td>I</td>
<td>moment of inertia</td>
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<tr>
<td>K</td>
<td>buckling coefficient (= 0.7 for one end pinned and one end fixed, and 1.0 for both ends pinned)</td>
</tr>
<tr>
<td>L</td>
<td>length of push-rod (= 650 mm)</td>
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<tr>
<td>P</td>
<td>actuator load</td>
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1 Introduction
Approximately 80% of a modern Formula One racing car is made from polymer matrix composite materials. The majority of load-bearing components within a car rely on carbon fibre-reinforced epoxy composites, which have high specific stiffness and strength properties. Glass fibre and aramid fibre-reinforced composites are also used where a structural component needs to have good damage tolerance and impact energy absorption characteristics. A diverse variety of material architectures, including unidirectional and woven fabrics and continuous and chopped fibres, and various manufacturing processes such as autoclaving, thermoforming, resin transfer moulding, vacuum forming and filament winding are available when designing a high performance racing car. This diversity of materials and manufacturing processes permits design engineers to customise the performance of a composite structure to a degree that is not afforded solely by the selection of metals.

Many of the innovations that are first seen in Formula One racing cars eventually are transferred to niche or broad market commercial vehicles and the use of composite materials is no exception in this regard. Alternative fuel vehicles, including those powered by fuel cells, which are now manufactured by every major car manufacturer in the world, rely heavily on the use of advanced composite materials [1] despite the challenges of combining composites processing operations with the predominant steel stamping and metal forming manufacturing processes. General Motor’s 1992 gull-winged all-carbon fibre UltraLite and Peugeot’s Tulip (five segment glass polyester over foam core body module) are two such examples. The use of advanced composite materials in the construction of conventional internal combustion engine powered cars is continually increasing. The latest S-Type Jaguar uses a low density (550 kg/m³) 10% fibre reinforced reaction injection moulded (RRIM) polyurethane system in the production of the interior door panels to satisfy stringent side-impact protection standards [2]. The production of the moulded door panel in a single process makes it possible to produce a set of four doors in a mere five minutes.

Other examples of composite materials that are now commonly used in commercial production vehicles are readily found by considering different classes of composite materials, including carbon/carbon composites, metal/metal matrix composites, polypropylene composites, etc. The performance and efficiency of

$R$ smallest radius of gyration
$\nu$ Poisson’s ratio
$\sigma$ stress

Subscripts
$cr$ critical
$ext$ external
$int$ internal
$11$ longitudinal
$22$ transverse
both diesel- and petrol-powered internal combustion engines can be increased significantly if the moving parts of such engines are manufactured from materials that are lighter and have better high temperature resistance characteristics than conventional metals; this is even more apparent if active cooling is not required for such components. Consequently, pistons, cylinder liners, and valves and other engine components are being developed from carbon/carbon composites [3, 4] and early prototype test results suggest that these materials will eventually be used widely in the manufacture of production vehicles. Advantages that are particular to carbon/carbon composites include:

(i) their low coefficients of thermal expansion facilitate easy control of dimensional clearances, between various moving parts,
(ii) their low density leads directly to weight savings, and
(iii) their advantageous friction properties minimise the requirements for lubrication between moving parts.

In spite of these significant advantages, however, further development is required to reduce production costs, to develop suitable mass-production methods, and to improve the reliability of specialist coatings, including against corrosion and abrasion.

When MMCs are used as replacement materials for steel and cast iron in production engine applications, they are generally used because of their improved wear resistance, their increased specific stiffness and their superior resistance to high cycle fatigue. It is almost twenty years since aluminium based MMC was used to manufacture pistons for commercially produced Toyota diesel engines [5]. This particular composite piston used a chopped fibre preform in the ring groove of the aluminium alloy in order to provide local resistance against the wear and thermal fatigue of the piston. Other composite pistons used SiC particulate reinforced aluminium, which has a lower coefficient of thermal expansion than conventional Al-Si alloys. Consequently, this permits the use of smaller clearances between a piston and the walls of its cylinder, and so leads directly to greater engine efficiencies, power output and fuel consumption. The use of matrix reinforcements also serves to increase the specific stiffness of a piston over one that is manufactured without any reinforcement. More recently, the cylinder liners of the 2.3 litre engine in the Honda Prelude were selectively reinforced: hybrid performs of carbon and alumina fibres were infiltrated by molten aluminium [6]. Within the last decade other drivetrain components, such as the connecting rod, have been candidate components for manufacturing from MMCs [7]. The use of lighter components when manufacturing the piston/connecting rod subassembly can directly reduce the secondary dynamic inertial forces that can occur, thus prolonging the useful service life of such critical engine components. This is particularly true for smaller capacity engines. Reduced reciprocating forces should also serve to reduce the loads that are applied to the crankshaft and to reduce friction losses: consequently, this should enable increased fuel economy or engine performance to be achieved. [8]. However, despite these performance benefits, cost remains an obstacle to the more widespread use of MMCs in production vehicles.

Metal matrix composites, and in particular those based on aluminum matrices, are suitable candidate materials for manufacturing gears, clutch parts, brake callipers,
turbocharger and supercharger compressors, pump housings, valves, brackets, pulleys, and suspension pushrods and rockers. The physical characteristics for cast iron brake systems are such that the cast iron can be replaced by aluminium-based MMCs. In particular, the high thermal conductivity and the good wear resistant properties of aluminium MMCs mean that brake drums and disk brake rotors can be made from these lighter materials instead of heavier traditional cast iron. The static weight saving is approximately 50% and this provides added dynamic benefits by reducing inertial forces, increasing acceleration forces and reducing braking distances. Allison and Cole [8] reported results of brake dynamometer tests, which indicated that MMC rotors are quieter than cast iron rotors, they wear less, and they provide more uniform braking friction throughout the braking cycle. Specific examples of commercially produced automotive vehicles that use aluminium MMC disk brake rotors and brake drums would include General Motor’s EV-I, the Plymouth Prowler and the Lotus Elise. Within automotive racing applications, where improved performance takes priority over lower priced materials, Hurley [9] has reported the use of aluminium MMCs: brake callipers for Formula One racing cars that are produced from an aluminium 2124/SiC/25p MMC are stiffer than conventional cast iron and thus provide greater leverage and shorter stopping distances. The use of aluminium MMCs in the driveshaft of a vehicle capitalises on the greater specific stiffness of these materials as compared to more commonly used aluminium and steel. All driveshafts, irrespective of the material from which they are manufactured, are constrained by the rotational speed at which they become dynamically unstable. The critical speed depends directly on the inner and outer diameters and specific stiffness of the driveshaft and inversely on the length of the driveshaft. In space restricted areas, where it is inappropriate to increase the diameter of a driveshaft, the greater specific stiffness of MMCs offers attractive design alternatives to conventional metals: they permit longer lengths of driveshaft at given diameters or reduced diameters at given driveshaft lengths. For this reason, aluminium 6061/Al2O3 MMCs have been used to manufacture composite driveshafts for automotive vehicles.

Despite the steady increase over recent years and the projected future growth in using composites in automotive applications, which are largely due to the specific physical characteristics of the materials, a major breakthrough in manufacturing primary structural parts in high volume automotive applications has not occurred. Most successful uses of composites have been limited to non-structural and semi-structural applications, and this is particularly true in producing commercial vehicles. In addition to material and manufacturing costs, there are other factors, including the specific material or material combination being considered and the function of the part, that also must be considered in the design of automotive components and structures. Three separate categories of polymer matrix composites that are used in automotive applications can be identified, namely non-structural materials, semi-structural materials, and structural materials.

(i) Non-structural materials. These materials would broadly include short glass fibre-reinforced polymers with reinforcements accounting for 10-50% by weight. (ii) Semi structural materials. These materials would typically be either glass mat thermoplastics (GMT) and sheet moulding compound (SMC). (iii) Structural materials. Structural thermoplastic composites, structural reaction
injection moulded (SRIM)/foam core parts, resin transfer moulded (RTM) parts, and filament wound parts.

The interest in using textile thermoplastic composite materials in the manufacture of automotive components has increased substantially within the past decade. This is due to a number of reasons [10-13]: (i) it is much easier to recycle thermoplastic matrix composites than those based on thermoset resins, (ii) the shelf life of a thermoplastic matrix prepreg is almost unlimited and there are no requirements for storage at sub-zero temperatures, (iii) there is large potential for fast automated manufacturing using processes similar to compression moulding and diaphragm moulding of glass mat thermoplastic (GMT) materials, (iv) thermoplastic composites may be post-shaped and welded, (v) a thermoplastic matrix has a longer elongation at fracture and a higher resistance to crack propagation than ordinary thermoset matrices and (vi) textile processes, such as weaving, braiding and knitting, enable faster fabrication and tailoring of the fibre architecture of preforms than currently offered by conventional and prepreg techniques. Commingled yarns, i.e., yarns in which the reinforcing fibres are intimately mixed with spun thermoplastic matrix fibres, are a novel form of intermediate materials suitable for manufacturing load carrying thermoplastic composites. As the yarns are flexible, they can be used in any textile process. Consequently, it is possible for drapable and highly conformable fabrics and structures to be produced. A wide variety of fibre/matrix combinations are available in commingled form. Production cycle times range from below one minute to several hours depending on the thermoplastic matrix material as well as on the required pressure and processing temperature.

In commercial automotive vehicles, the use of polypropylene (PP) matrix composites has become a ubiquitous success due to the excellent processing and mechanical characteristics of PP, its good chemical and weather resistance, and its relative cost. High performance PP composites with good physical properties have been developed in the last two decades by chemically combining elastomeric (rubber) or inorganic (talc) fillers with PP to provide a useful material with an extremely wide range of physical properties. The use of rubber fillers provides a material that is both impact resistant and damage tolerant and has lead to one of today’s major applications of PP composites, namely, the automobile bumper. The incorporation of talc fillers served to increase the rigidity of this material and subsequently facilitated the development of large, wraparound PP bumpers.

Prior to the mid-1970s, advanced composite materials had only been used to manufacture secondary and non-structural components in Formula One racing cars. However, the failure of a rear wing at the 1975 Spanish Grand Prix, and the consequent death of four spectators, dampened the budding enthusiasm of designers for composites. The current extensive use of advanced composite materials in Formula One World Championship motor racing can be traced back to the latter half of the 1970s when Colin Chapman of Lotus designed the ‘wing-car’ to have large downward thrust forces. This was achieved by using large wing shaped under bodies on either side of the car. While this successfully allowed greater speeds, the chassis width had to be reduced in order to
incorporate these large under wings. However, the reduced chassis width served to reduce the torsional rigidity of the car and this was exacerbated by the increased cornering speeds afforded by the under wings. The ‘wing-car’ design proved so successful that within five years after it was introduced, lateral acceleration forces of up to 3g were being recorded; these severe forces and the inadequacies of the suspension system were physically demanding on drivers. The late 1970s saw the emergence of turbochargers that were capable of providing thrust forces in excess of 1400 BHP. This forced design engineers to consider using the specific stiffness advantages and design flexibility of advanced composite materials as primary structural load-bearing components instead of the high-performance aluminium alloys that had previously been used.

The first major breakthrough in the use of composites in Formula One came with the 1981 McLaren MP4 car, the monocoque of which was moulded from a carbon fibre-reinforced epoxy polymer. The monocoque, which was manufactured by Hercules in America, was moulded over a machined aluminium tool that was subsequently removed in pieces through the cockpit opening. Unidirectional carbon/epoxy was used as skins that surrounded an aluminium honeycomb core. This one-piece construction proved so successful that it continued to be used by McLaren virtually unchanged for six seasons. Gustav Brunner, in 1983, developed his ATS Formula One car by moulding the monocoque in top and bottom halves, both parts being built in a female mould. Not having to remove the moulding tool meant that the monocoque could have a more aerodynamic and rounded shape and the cockpit opening could be smaller than the McLaren car. However, this two-piece construction, which had a circumferential joint horizontally around the monocoque, did not have the same structural integrity as did McLaren’s one-piece.

Since the early 1980s the issue of joining and joints has become less problematic and designers have been able to focus their attention on using composites to manufacture components other than primary structural parts but which are equally important to the optimum performance of a car. These have included the gearbox, where torsional rigidity is critical, and suspension components, where push-rods require high stiffness when setting the suspension. Traditional metal suspension components have begun to be replaced by composite materials that have greater specific stiffness and thereby provide designers with greater control over the localised and overall stiffness of a complete suspension system. The push-rod is the single component that most determines the stiffness of the suspension system. However, it is to be noted that the change from metal to composites has not been without problems for many Formula One teams. Williams, for example, replaced their metal push-rods with composites but had to revert to metal after a series of rear suspension failures during testing. Strength and energy absorption have also become important design issues and these are made more complex with the use of increasingly sophisticated geometric shapes and constructions.

The technical regulations governing the design of a Formula One racing car are defined at the beginning of each racing season by the Fédération Internationale de l'Automobile [14], the FIA, which is the ultimate governing authority for motor
sport throughout the world. These comprehensively address the bodywork, the dimensions, the weight, the engine, the fuel, oil and coolant systems, the electrical systems, the transmission system, the suspension and steering systems, the brake system, wheels and tyres, the cockpit, safety equipment and safety structures, impact testing, roll structure testing, and static load testing. The key parameters affecting structural design are those of material, geometry and weight. Constraints are placed on the dimensions of the bodywork of a car (overall width and height must not exceed 180 cm and 95 cm, respectively), the weight of a car (must not be less than 600 kg), the maximum engine capacity (four stroke engines of not more than 3000 cc, twelve cylinders, and five valves per cylinder) and the size of the driver envelop within the cockpit. It is estimated that 10 kg of excess weight can lead to a time difference of at least 0.1 sec around the length of a typical racing circuit; such a time penalty is critical in Formula One. Static and impact design loads are specified for the safe operation of critical components of a car, the roll bars and the driver’s survival cell. All tests are required to be carried out in accordance with FIA Test Procedures in the presence of an FIA technical delegate and by using suitably calibrated measuring equipment.

A modern Formula One racing car has a central load-bearing structure that connects the front and rear suspension systems. This structure consists of the monocoque, the engine, and the gearbox casing. The driver, fuel tank and front suspension dampers are contained inside the monocoque, while the engine is jointed to the back of the monocoque on four separate studs. The gearbox casing is attached to the rear face of the engine. It is this three-piece box-beam structure of monocoque, engine and gearbox that carries the inertial loads to the four corners of the car. Various wing structures, under bodies, cooler ducting and bodywork are attached to and arranged around this box-beam. The dynamic behaviour of this box-beam structure exerts a major influence on the performance of a car during motion. While ‘setting-up’ a racing car at a circuit, subtle changes are made to the suspension elements in order to modify its handling characteristics. Ideally, any small changes in the stiffness of a component should be noticed in the balance of the car. However, this will not be the case if the structural members transmitting the loads are insufficiently stiff. Therefore, the chassis members must have good stiffness characteristics if the handling and speed around a circuit are to be optimal.

The present Chapter describes the design, manufacture and structural response to compressive forces of a set of carbon fibre-reinforced epoxy suspension push-rods. The structural behaviour and specific stiffness of the push-rods is compared to that of equivalent weight steel and aluminium push-rod designs and it is shown that advanced composite materials can provide superior performance advantages when compared to conventional engineering metallic materials. The ultimate catastrophic failure of the composite push-rods, which, in normal design practice would occur at loads far in excess of operating loads, was examined and the particular mechanisms of fracture and damage are detailed.

2 Design And Manufacture Of Suspension Push-rods
2.1 Design
In order to minimise the effects of wind-drag around the push-rods, it was decided to utilise an aerofoil cross-section instead of a circular cross-section. Uniform and tapered layups were used, the purpose of the taper being to increase the equivalent modulus along the critical section of the push-rod and consequently, to increase the load at which buckling would occur. Since the end sections of both the tapered and uniform layups were identical, it was anticipated that the load at which compression failure should occur would be identical for both types of push-rod. The push-rods were 650 mm long, whilst the nominal wall thickness was 1.825 mm for the uniform push-rod, and varied between 1.825 and 2.450 mm for the tapered push-rods. The major and minor external dimensions of the airfoil axes were nominally specified at 38 x 18 mm.

One objective of this investigation was to establish the influence of the layup on the possible buckling response of the push-rod. Since the principal in-service mechanical load on the push-rod was uniaxial compression, it was necessary to maximise the number of 0° plies within the stacking sequence in order to provide maximum uniaxial stiffness. A number of cross-plies were necessary, however, to prevent longitudinal splitting of the push-rod. The first stacking sequence that was considered was a uniform layup of (0°/90°; 0°/90°; 0°/90°), i.e., two external 0°/90° cross-plies of woven prepreg surrounding nine unidirectional 0° plies. This layup differs from that which is typically used in current Formula One design only in that there are no tapered plies within the stacking sequence. As such, it was anticipated that the ultimate mechanical response of this push-rod design would be a lower bound limit and failure would be due to buckling.

The remaining two push-rods were identically tapered along their midlengths and the particular stacking sequence that was used was 0°/90°; 0°/90°; 0°/90°; 0°/90°; 90°; 0°/90°), i.e., two outer 0°/90° cross-plies of woven prepreg surrounding seven 0° plies, one 90° ply and six 0° plies. The taper was obtained by only placing some of the 0° and 90° plies along part of the 650 mm length of the push-rods.

Commerially available laminate analysis software was used [15] to estimate equivalent laminate properties and strengths from the precise ply properties of Table 1. The values of these equivalent laminate properties are detailed in Table 2.

2.2 Theoretical Predictions Of Structural Behaviour
Using simple Euler buckling theory it was possible to predict the loads and strains associated with buckling by approximating the airfoil cross-section with that of an ellipse:

\[
\sigma_{cr} = \frac{\pi^2 E}{(KL/r)^2}
\]

where \(r\), the smallest radius of gyration, is defined by

\[
r = \sqrt{I/A}
\]

For push-rod 1 (PR1), classical laminate theory predicts a longitudinal modulus of 211GPa (cf Table 2). The moment of inertia is calculated from the half-major
and half-minor dimensions of the cross-section of the push-rods. These external dimensions are 18.87mm and 8.94mm, whilst the corresponding internal dimensions are 17.05mm and 7.12mm, respectively. Consequently, the moment of inertia of PR1 is given as:

\[ I_{PR1} = \frac{\pi}{4} (a_{ext} b_{ext}^3 - a_{int} b_{int}^3) = 5756 \text{mm}^4 \]

Similarly, the cross-sectional area of PR1 is calculated as 148.6mm$^2$. In order to calculate both the critical buckling load and the surface strains within the push-rods at this corresponding level of load, it is necessary to determine the correct buckling coefficient, $K$, for the particular boundary conditions that are applied to the push-rods. The nominal boundary conditions, selected to represent what would typically occur in service, define one end as pinned and allowed to rotate about this point and the other end as fixed and prevented from any displacements or rotations about this point. For such a case, the buckling coefficient $K = 0.7$. However, the physical conditions within the constrained end of the push-rods may not actually be quite as severe in reality and the design could reasonably be considered insufficient to fully prevent rotation because of the fact that the push-rods are hollow with a relatively thin wall section and the interior walls of the push-rods are not prevented from rotating in towards the middle of the push-rods. The particular loading arrangement used for the experimental tests (described in section 17.3) did not permit the fixed boundary condition to transfer bending moments into the push-rods. Consequently, it is more realistic to consider that the actual boundary conditions were closer to both ends being pinned ($K = 1$) than to one end being fixed and the other being pinned ($K = 0.7$). On this basis, the critical buckling load, $P_{cr} = 57.9 \text{kN}$ if $K = 0.7$ and $P_{cr} = 28.4 \text{kN}$ if $K = 1$. The corresponding levels of surface strain would be 0.19% if $K = 0.7$ and 0.09% if $K = 1$. It will be shown later that these values compare quite well with the average failure strain of 0.10% and the failure load of 31.0kN for push-rod 1.

The half-major and half-minor external and internal dimensions of tapered push-rods PR2 – PR3 were 18.87mm, 8.94mm, 16.42mm and 6.49mm, respectively. The moment of inertia and cross-sectional area were therefore calculated as 7064mm$^4$ and 195.2mm$^2$, respectively. The critical buckling load $P_{cr} = 74.4 \text{kN}$ if $K = 0.7$ and $P_{cr} = 36.5 \text{kN}$ if $K = 1$. The corresponding levels of surface strain are 0.17% and 0.08%. These values will be shown later to compare well with the actual failure loads of the push-rods.

2.3 Manufacture

Three separate carbon/epoxy push-rods were manufactured by wrapping the various plies of prepreg around a hollow elliptical silicone mandrel. This was then placed within an elliptical two-part cavity mould and cured in an autoclave as shown in Fig. 1. The hollow mandrel acted as an expandable bladder during the curing cycle, thereby pressing the prepreg firmly against the walls of the mould and ensuring that a uniform wall thickness was produced along the length of the push-rods.

The autoclave curing cycle for the woven and unidirectional carbon/epoxy prepreg involved a 1.5 hour cure at 125°C and 700kPa with a heat-up and cool-down rate of 3°C per minute. When the temperature reached 125°C the vacuum
was vented to atmosphere. Pressure was then introduced and ramped at 100kPa per minute to 700kPa. When the pressure cycle was completed the pressure was ramped down at 50kPa per minute to 0kPa, at which stage the vacuum was reintroduced.

Both the mould and the silicone mandrel were reused when manufacturing all three push-rods and these were cleaned and degreased before being coated with release agent (Freekote) prior to the plies of carbon/epoxy prepreg being wrapped around the mandrel and placed within the mould. Figure 1 illustrates how the complete assembly was vacuum bagged to evacuate air, solvents and entrapped volatiles from the laminate and to allow the positive autoclave pressure to consolidate the laminate against the mould surface. A breather cloth bagging assembly was used to absorb any excess resin flow and also to smooth out the sharp corners of the mould, which could cause the vacuum bag to rupture under the high autoclave pressures. A solid release film was placed against the mould walls to prevent the breather cloth from sticking to the mould surfaces. Upon completion of the curing cycle, the vacuum bag assembly was removed from the autoclave. The bag and breather were discarded and the end plates were removed prior to the mould being opened. The composite push-rod was then taken from the mould and the silicone mandrel removed from the centre of the push-rod.

Before the actual carbon/epoxy push-rods could be manufactured, it was necessary to manufacture a suitable elliptical mould and elliptical silicone mandrel so that the finished push-rods would be of the required thickness and cross-section. The mould was machined from aluminium whilst the silicone mandrel was manufactured using GFRP slips, an elliptical copper pipe and the mould, as shown schematically in Fig. 2. The copper pipe was located centrally within the mould cavity through an aluminium end-plate. The end-plate was subsequently bolted to the mould and the mould was inverted. The GFRP slips were placed against the mould walls and de-aerated liquid silicone rubber was poured into the space between the GFRP slips and the copper pipe in the mould. This assembly was left under room conditions for fourteen hours to allow the rubber compound to cure and was then placed in an oven at 120°C for 1.5 hours to complete the curing process. The hollow silicone mandrel was then removed from the mould and the copper pipe was extracted from the mandrel. No significant air bubbles or voids, which would have made the mandrel unsuitable for manufacturing the push-rods, were detected visually.

The GFRP slips were fabricated using the two halves of the mould. After spraying release agent on both halves of the mould, six plies of GFRP were stacked in each half of the mould. The two halves of the mould were covered with a release ply, covered with a bleeder cloth and placed in a vacuum bag, which was then sealed. The assembly was placed in the autoclave and cured using an appropriate curing cycle.

3 In-service Behaviour Of Push-rods
Three push-rods have been tested statically to failure under compression using a displacement mode of control on a 100kN uniaxial servohydraulic fatigue machine (Series 8501 Instron). The loading was introduced at both ends of a
push-rod using female end-fixtures which had been designed to provide boundary conditions that were pin-jointed at the bottom and cantilevered at the top in order to simulate in-service support conditions. This is shown schematically in Fig. 3.

Surface strains, from gauges at three different positions on the push-rods, were recorded using a data acquisition system which operated on a keypress sequence. Strain gauges were aligned longitudinally and transversely along the length of the push-rods to measure the performance under compressive load. Two gauges (SG1 and SG2 of Figs 4(a)-(c)) were aligned axially at the midlength and on opposite faces of the push-rods: these provided information on the presence of buckling, the deviation from linearity in the mechanical response of the push-rod, and the fracture strains. A third strain gauge (not included in Figs 4(a)-(c)) was aligned normal to the first two gauges, close to the mid-length position of the push-rod, and was used to calculate values of Poisson’s ratio.

4 Response Of Push-rods To Compressive Loads

4.1 Gross Structural Response
All push-rods were loaded statically to failure by means of a displacement mode of control. Load, displacement and strain values were collated at discrete increments of actuator load. As the applied load was increased from zero, the response of the push-rods was initially linear elastic. Figures 4(a)-(c) detail the variation of compressive surface strains at the mid-length position on opposite sides of the three push-rods with increasing actuator load. The strain responses deviated from linearity at approximately 90% of the final failure load although minor fracture events occurred before this deviation from linearity in the first push-rod test (a minor fracture event occurred at 10.8kN in Fig. 4(a)). This deviation of the strain difference (i.e., magnitude of strain difference between front and back faces of the push-rods = SG2-SG1) from linearity, which occurred at approximately 90% of the final failure load identified the onset of catastrophic fracture. Ultimate failure of the first uniformly laid-up push-rod occurred some 40mm from the centre of the specimen whilst failure of the remaining three push-rods was concentrated around the bottom ends of the specimens, close to the pin-jointed end-fixture of the testing machine. Table 3 details the loads and strains at which ultimate failure and the deviation from linearity occurred during the three push-rod tests.

The measured actuator loads and surface strains are presented in Figs 4(a)-(c) for the three push-rods. The average load-strain relationship for all the push-rods is essentially linear almost until fracture. However, the individual strain-gauge readings deviate from linearity immediately with the application of load and this deviation continues to increase directly with applied load up until failure.

Table 3 identifies the maximum direct compressive strains which were measured during each test and may be compared against the failure strains of the carbon-fibres of 1.5% [16]. While the maximum direct strain reading at failure of push-rod 1 (i.e., 0.18%) is greater than those recorded during the other three tests (this is to be expected since the strain gauge position of this push-rod was closer to the
failure site than in all other tests), this is considerably less than the fibre failure strain. Consequently, failure of these components is considered to be due to geometric and manufacturing factors rather than to material limitations.

4.2 Buckling Behaviour
A simple first mode of buckling was apparent along the length of the four test specimens, with maximum lateral deformation (i.e., crest of the buckle) occurring close to the mid-length of the push-rods. Buckling initiated with the application of load in all push-rod tests, as can be seen from the deviation of the two sets of surface strain gauge readings (Fig. 4(a)-(c)) from the average compressive strain. The amplitude of the buckle increased linearly in magnitude with actuator load until failure. No dial gauges were used during the tests to quantify the amplitude of the buckle although this could be estimated from the degree of bending that has been measured by the surface strain gauges. The average failure strain and failure load of push rod 1, PR1, can be seen from Figure 17.4(a) to be 0.10% and 31kN, respectively. These compare quite well with the previous (c.f. Section 17.2.1) theoretical predictions for buckling with $K = 1$, i.e., $P_{cr} = 28.4$ kN and strain = 0.09%. Similarly for push rod 2 and push rod 3, PR2 and PR3, the theoretically predicted load and strain with $K = 1$ (c.f. Section 17.2.1) are 36.5kN and 0.08%, respectively. These compare reasonably well with the experimental data shown in Figures 17.4(b) and (c) of average failure strain = 0.07% to 0.10% and failure load of 30kN.

4.3 Damage And Catastrophic Failure Of Push-rods
The compressive failure mechanisms that occurred in all four push-rods were similar although failure of push-rod 1 occurred at a position close to the mid-length of the component whereas failure was close to the pin-jointed end for the other three push-rods. The reason for this different failure site is due to the fact that push-rod 1 was manufactured without any tapered region in its mid-section, unlike the other two push-rods. The general appearance of the fracture associated with the two tapered push-rods, PR2 and PR3, is shown in Fig. 5. The appearance of the fracture surface is different both around the perimeter of the push-rod and through the thickness of the push-rod. The fracture is not uniformly compressive around the perimeter and this is due to the different degrees of compressive strains that existed on opposite sides of the push-rod. The lack of similarity of the through-thickness fracture features is partly due to the variation of compressive strains and partly due to the different ply orientations through the thickness of the push-rod. Figures 6 to 8 detail the compressive failure sites that led to ultimate fracture of the push-rods as identified using scanning electron microscopy. Many buckled and broken fibres and crushed fibre ends are evident in Fig. 6 and the manner in which these fibres fractured is characteristic of compressive failure, i.e., fibre microbuckling and localised fibre fracture [17-20].

5 Discussion
A conventionally designed tubular steel cylindrical push-rod, of 22 mm external diameter and 2 mm wall thickness, would have a critical buckling load of $P_{cr} = 31.1$ kN and average surface strain of 0.12% for $K = 1$. The corresponding weight of such a push-rod would be 0.613 kg (density = 7500 kg/m$^3$). This compares against the composite push-rods of the present paper which would have
similar buckling strength and stiffness but which would be almost 70% lighter and would weight approximately 0.203 kg (density ≈ 1600 kg/m³). A tubular aluminium push-rod, on the other hand, would also be significantly heavier than the composite push-rod and would need to be 26 mm outer diameter and of 5 mm wall thickness to avoid buckling before approximately 30 kN.

While it is clear that composite suspension push-rods can be designed and manufactured to provide equal stiffness and strength characteristics as conventional steel or aluminium push-rods, and at the same time offer some degree of weight savings, it is necessary to realise that other issues are important in deciding whether current metallic materials could be replaced safely by high performance composites. The amount of energy absorption by the suspension system in the event of an accident to the corner of a car, for example, may well establish that plastic deformation occurring in wishbone tubes outperform a more brittle mode of fracture in a composite system. In other cases, the reasons for using a composite system might not be structural but could be due to aerodynamic factors where the flexibility of manufacturing from composites permits greater geometric freedom.

6 Conclusions
Unidirectional and woven cross-ply carbon/epoxy composites were used to manufacture suspension push-rods that could typically be used in a Formula 1 racing car. These were subsequently loaded to failure under compression using end boundary conditions which approximated those that would be used typically in service. Two push-rods had a tapered mid-section consisting of 0° and 90° plies whilst an initial trial specimen was of constant thickness along its length.

The results of this investigation can be summarised as follows:
1. Fracture of the trial specimen (push-rod 1) occurred close to the mid-length whilst fracture in both other cases was close to the pin-jointed support in the loading frame. Buckling occurred in all cases and this increased directly with the application of load.
2. The ultimate performance of these particular push-rods was limited by geometric, manufacturing and support parameters and fracture occurred at load levels that were far less than the ultimate fibre failure strains of the materials that were used. Consequently, it is possible that, with redesign, additional improvements in performance could be obtained.
3. While these composite push-rods offer valuable weight savings over conventional steel push-rods for Formula One motor racing without compromising strength and stiffness, it is important that issues such as manufacturing complexity and crashworthiness are evaluated thoroughly before conventional engineering materials are replaced by high-performance composite materials.

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University of Ireland – Dublin (President’s Research Award) is gratefully acknowledged.

References


FIGURE CAPTIONS

Figure 1: The autoclave moulding arrangement used to manufacture the push-rods.

Figure 2: Arrangement of the mould assembly for manufacturing the hollow silicone mandrel.

Figure 3: Compression test set-up used to establish the ultimate performance of the carbon/epoxy push-rods.

Figure 4: Variation of surface strains with actuator load during testing of push-rods 1-4 ((a)-(d), respectively). Buckling is identified by the difference between the values of the two surface strain readings and begins with the onset of actuator load. Incipient fracture is identified by the deviation of strain difference from linearity, which occurs at approximately 90\% of the ultimate failure load in all four tests. The solid and dashed lines represent average strains and strain differences, respectively.

Figure 5: Compressive fracture of push-rod 4 as identified visually. The outer 0°/90° woven ply is clearly visible. The damage mechanism that initiated failure was due to compressive stress (far side in photograph). The external width and depth of this push-rod are 38mm and 18mm, respectively.

Figure 6: Global view micrograph of compressive fracture of push-rod identifying crushed fibre ends and fibre kinking, fibre micro-buckling and fibre fracture.

Figure 7: Local view micrograph of compressively fractured fibres.

Figure 8: Local view micrograph of fracture surfaces of fibre ends. Note the degree of poor fibre-matrix adhesion where the fibres are debonded from the epoxy resin.
TABLES

<table>
<thead>
<tr>
<th>Mechanical Property</th>
<th>$0^\circ/90^\circ$ Woven Ply</th>
<th>Unidirectional Ply</th>
</tr>
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<tbody>
<tr>
<td>Thickness</td>
<td>0.35mm</td>
<td>0.125mm</td>
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<tr>
<td>Longitudinal stiffness</td>
<td>53GPa</td>
<td>310GPa</td>
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<tr>
<td>Transverse stiffness</td>
<td>52GPa</td>
<td>5.9GPa</td>
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<tr>
<td>Shear modulus</td>
<td>0.011GPa</td>
<td>0.012GPa</td>
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<td>Poisson’s ratio, $\nu_{12}$</td>
<td>0.1</td>
<td>0.2</td>
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<tr>
<td>Longitudinal tensile strength</td>
<td>690MPa</td>
<td>1960MPa</td>
</tr>
<tr>
<td>Longitudinal compressive strength</td>
<td>59MPa</td>
<td>700MPa</td>
</tr>
<tr>
<td>Transverse tensile strength</td>
<td>690MPa</td>
<td>354MPa</td>
</tr>
<tr>
<td>Transverse compressive strength</td>
<td>59MPa</td>
<td>354MPa</td>
</tr>
<tr>
<td>Shear strength</td>
<td>80MPa</td>
<td>100MPa</td>
</tr>
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Table 1: Mechanical properties of the $0^\circ/90^\circ$ woven and unidirectional carbon/epoxy material systems used to manufacture the composite push-rods.

<table>
<thead>
<tr>
<th>Equivalent Laminate Property</th>
<th>Uniform Layup</th>
<th>Tapered Layup</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal stiffness, $E_{11}$</td>
<td>211GPa</td>
<td>221GPa</td>
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<tr>
<td>Transverse stiffness, $E_{22}$</td>
<td>23.7GPa</td>
<td>34.7GPa</td>
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<tr>
<td>Poisson’s ratio, $\nu_{12}$</td>
<td>0.115</td>
<td>0.068</td>
</tr>
<tr>
<td>Poisson’s ratio, $\nu_{21}$</td>
<td>0.013</td>
<td>0.011</td>
</tr>
<tr>
<td>Shear modulus, $G_{12}$</td>
<td>11.6MPa</td>
<td>11.7MPa</td>
</tr>
</tbody>
</table>

Table 2: Equivalent mechanical properties of uniform and tapered layups used to manufacture the different push-rods.

<table>
<thead>
<tr>
<th>Specimen / Push-rod Number</th>
<th>Actuator Load at Deviation from Linearity</th>
<th>Ultimate Failure Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actuator Load</td>
<td>Surface Strain</td>
</tr>
<tr>
<td>PR1</td>
<td>25.40kN</td>
<td>0.182%</td>
</tr>
<tr>
<td>PR2</td>
<td>28.50kN</td>
<td>0.160%</td>
</tr>
<tr>
<td>PR3</td>
<td>28.50kN</td>
<td>0.111%</td>
</tr>
</tbody>
</table>

Table 3: Summary of experimental test data (cf Figs. 4(a)-(c)).
Figure 1: The autoclave moulding arrangement used to manufacture the push-rods.

Figure 2: Arrangement of the mould assembly for manufacturing the hollow silicone mandrel.
Figure 3  Compression test setup used to establish the ultimate performance of the carbon/epoxy push-rods.
Figure 4a: Variation of surface strains with actuator load during testing of push-rod 1. Buckling is identified by the difference between the values of the two surface strain readings, and starts with the onset of actuator load. Incipient fracture is identified by the deviation of strain differences from linearity, which occurs at approximately 90% of the ultimate failure load. The solid and dashed lines represent average strains and strain differences, respectively.
Figure 4b: Variation of surface strains with actuator load during testing of push-rod 2. Buckling is identified by the difference between the values of the two surface strain readings, and starts with the onset of actuator load. Incipient fracture is identified by the deviation of strain differences from linearity, which occurs at approximately 90% of the ultimate failure load. The solid and dashed lines represent average strains and strain differences, respectively.
Figure 4c: Variation of surface strains with actuator load during testing of push-rod 3. Buckling is identified by the difference between the values of the two surface strain readings, and starts with the onset of actuator load. Incipient fracture is identified by the deviation of strain differences from linearity, which occurs at approximately 90% of the ultimate failure load. The solid and dashed lines represent average strains and strain differences, respectively.
Figure 5. Typical view of compressive fracture of tapered push-rod as identified visually. The outer $0^\circ/90^\circ$ woven ply is clearly visible. The damage mechanism that initiated failure was due to compressive stress (far side in photograph).

Figure 6. Global view micrograph of compressive fracture of push-rod identifying crushed fibre ends and fibre kinking, fibre micro-buckling and fibre fracture.
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