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Influence of wheel bearing performance on In-wheel motor advanced applications

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Abstract

In-Wheel motor torque transmission concepts enable the realization of innovative solutions for the electric or hybrid new vehicle platforms. Protean Drive system integrates in the unused space behind the wheels and provides unique performances in terms of generated power, torque and compactness. The performance gain with respect to other more conventional arrangements is due to the full integration and synergies created with the mechanical components and in particular the wheel bearing. The final performance is connected to the ability of the wheel bearing to provide the required stiffness that controls the reduction of the air gap between the motor rotor and stator. This paper presents an overview of how the In-Wheel motor is connected to the chassis of the vehicle, through the wheel bearing, with special attention to the relationship between bearing stiffness and motor air-gap.

Keywords: electric drive, HEV (Hybrid Electric Drive), motor design, permanent magnet motor, simulation, wheel hub motor.

1 Introduction

The design activity of a bearing, conceived in order to be integrated into an In-Wheel motor targeted to be a ‘bolt-on’ application to a large variety of vehicle topologies (from city cars to commercial vehicles), is characterised by demanding technology challenges:

- Definition of bearing performances, fulfilling requirement of the “worst case” configuration;
- Definition of a common space envelope dedicated to the wheel bearing, dealing with different chassis;
- Ensuring In-Wheel motor performances dependant on bearing, to be granted into all operating conditions

In-Wheel motor performance is strongly affected by the motor air gap, as shown in Figure 1.

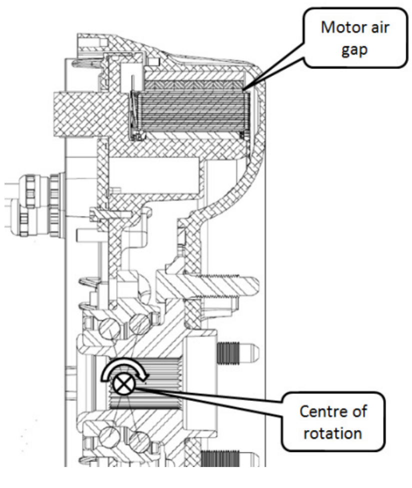

Figure 1

There are three geometric factors that make up the motor air gap: the diameter Z, the motor length X and air gap length Y. For motor performance, it is desirable to maximise the diameter and motor length and minimise the air gap length. See Figure 2

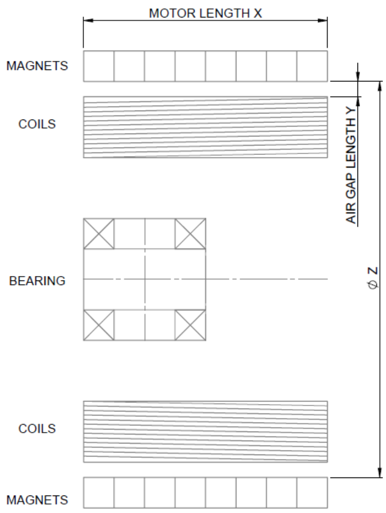

Figure 2

The control of this air gap under different operating conditions is vital to ensuring efficiency and high levels of performance targeted by the In-Wheel motor. Road loads in their worst cases however, represent a significant challenge in maintaining the optimum level of air gap, hence the need to tightly control its variation. Furthermore, there is a risk of magnets touching the wound teeth if too much variation is allowed, with serious consequences for mechanical damage, performance and durability of the In-Wheel motor. The wheel bearing design and in particular the tilting stiffness, influence the design of the motor length and airgap length, which in turn influence the motor performance.

2 Wheel bearing requirements

The motor length and diameter are ultimately limited by the space inside the wheel and suspension envelope as shown in Figure 3.

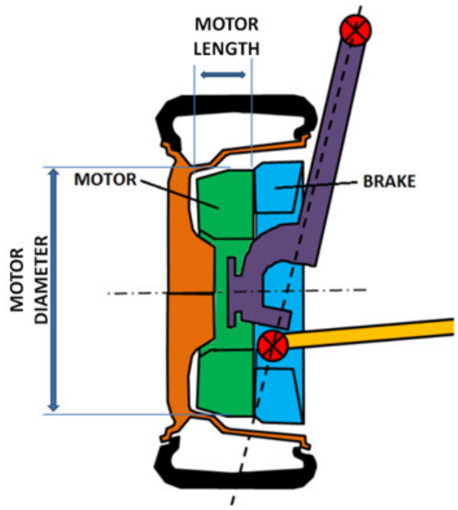

Figure 3

The motor diameter Z is primarily limited by the wheel profile. As it is not influenced by the bearing design, it will not be discussed further in this paper.

Both the air gap length Y and motor length X require clearances between stator and rotor components. The clearances are affected by manufacturing and assembly tolerances, as well as operating conditions such as thermal expansion and deflection under braking. Additional clearance is also required to account for the movement of the bearing arising from its tilting stiffness.

The motor length is primarily limited by the suspension and wheel geometry. The bearing tilting stiffness also has a contribution as a result of the axial clearance required at the ends of the wound coils and magnets to allow for the axial movement of the rotor in relation to the stator resulting from the tilting of the bearing.

Like the motor length, the air gap length is also influenced by the bearing stiffness and its required clearances, but to a much greater degree.

Imagine designing a motor within a fixed cylindrical design envelope. If a bearing could be created with infinite tilting stiffness, then the motor could be designed with only clearances necessary to take account of part and assembly tolerances and operating conditions. This would result in a motor with airgap dimensions X, Y & Z, as shown in Figure 2.

In reality, a bearing has a finite stiffness which allows the magnets to rotate about the bearing centre. The need to take account of the clearances in the motor resulting from this movement drives a reduced motor length V and an increased design of air gap U.

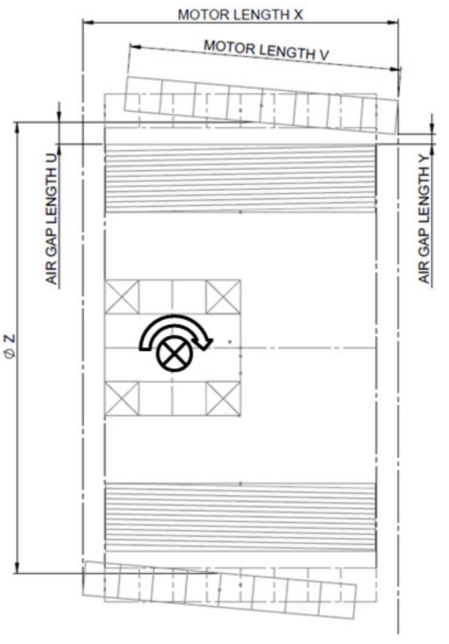

Figure 4

Figure 5 shows the clearance in millimetres that is required for a given bearing stiffness.
The airgap length clearance = airgap length U – airgap length Y.
The motor length clearance = motor length X – motor length V.
In both cases, the clearances should be minimised.

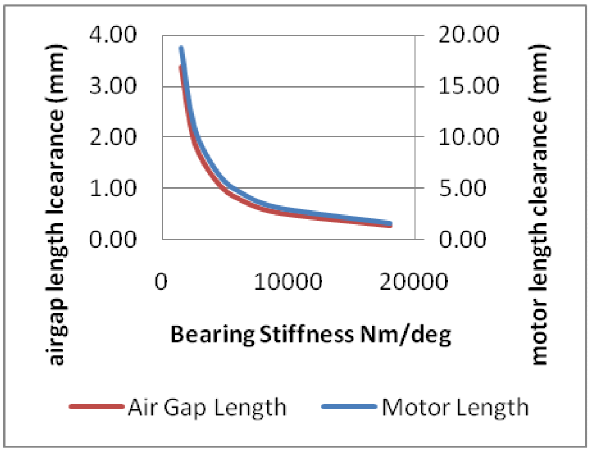

Figure 5

The shape of this curve demonstrates that both the air gap length and motor length exhibit a similar benefit from a stiff bearing. Beyond a specific value of stiffness that benefit diminishes.
Figure 6 shows how the bearing stiffness might influence the motor length V and airgap length U of a given in wheel motor configuration, designed with a length and diameter constraint typical of a common vehicle package.

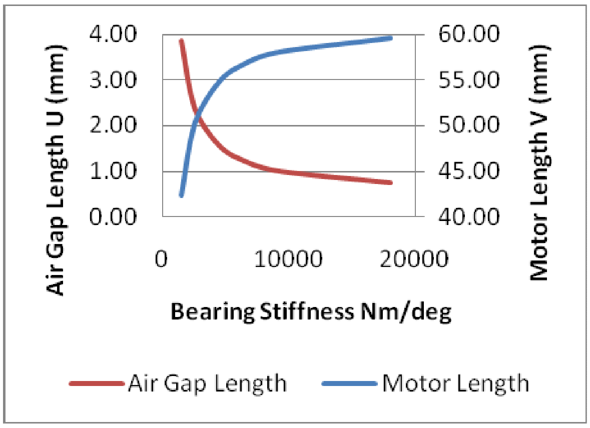

Figure 6

Figure 7 shows the % contribution the bearing stiffness makes to the total length of the motor and air gap. It can be seen that the bearing stiffness makes up the larger portion of the total air gap length and for lower stiffness bearings it can also have a significant contribution to the motor length.

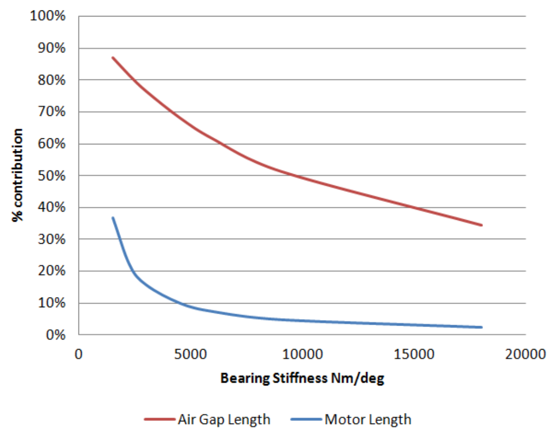

Figure 7

The nominal value of the air gap has been designed through optimising the performance of the In-wheel motor. Assembly tolerances and variations due to thermal cycles account for approximately 50% of the overall permissible variations. The remaining 50% of the permitted air gap variation has been designed to be controlled by the wheel bearing structural stiffness. Reviewing curves in Figures 6 & 7 it can be shown that this 50%/50% split represents an optimum value for bearing stiffness. For this purpose, the stiffness of the wheel bearing under worst case cornering conditions plays a significant role. Worst case cornering conditions for different classes of vehicles are identified in terms of lateral acceleration of the axle mass, expressed as G-force. For this study, a value of 1G for the full axle mass has been adopted for passenger car vehicles and 0.9G for commercial vehicles. Three values of axle masses and rolling radii were identified, for three different categories of vehicles. Loads due to extreme cornering are therefore expressed in terms of bending moment according to Table 1. From this table, the worst cornering load can be identified with the rear axle mass of the commercial vehicle.

Table1: Extreme cornering load conditions for different classes of vehicles

Platform	Axle mass [kg] x rolling radius [m]	G-Force Extreme cornering	Loads at Extreme cornering [Nm]
Commercial vehicles	1650 x 0.34	0.9g	5000
Sport Saloon	1600 x 0.32	1g	5000
Medium size Saloon	1200 x 0.31	1g	3700

Table2: Misuse conditions for different classes of vehicles

Platform	G-Force at Misuse Conditions	Loads at Misuse Conditions [Nm]
Commercial vehicles	2 g	10000
Sport Saloon	2 g	10000
Medium size Saloon	2 g	7400

The main requirement for the wheel bearing under extreme cornering conditions is to allow a maximum deflection angle of 0.45°, which corresponds to the remaining 50% of the permitted air gap variation, apart from the variation due to tolerances.
Furthermore, structural strength performance of the bearing at which the bearing must grant no yield, occurs as a result of extreme cornering loads.

Table 2 shows the loads assumed when considering misuse conditions.

The main requirement, in this case, is to allow no structural failure of the bearing and In-Wheel-Motor.

3 Integration on different classes of vehicles

Another key requirement for the wheel bearing is the ability to fit into different categories of vehicles with minimal re-design of suspension components and wheel rims. In addition, impact on the vehicle ride and handling should be minimised. Modifications of the body-work are not permissible, nor any change to the original track width.

Given these constraints, the permissible envelope for the wheel bearing on the classes of vehicles specified in Table 1 is defined as a cylinder with a diameter of Ø184mm and an axial width between outboard flanges to inboard outer ring side face of 66.7mm. This cylindrical envelope is shown in Figure 8.

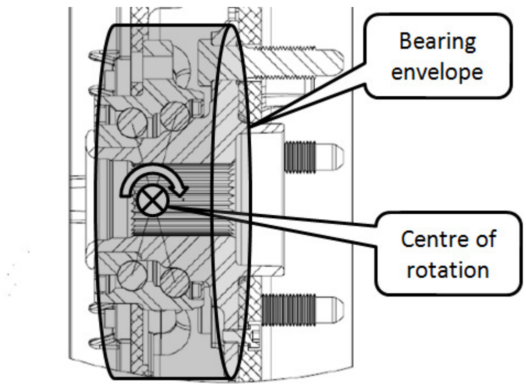

Figure 8

The ideal position for the bearing centre of rotation is to be midway between the ends of the airgap, as this minimises the contribution of the bearing stiffness to the airgap length. Suspension and wheel constraints, however, may prevent this.

The wheel bearing needs to be configurable with all the required vehicle interfaces and for this purpose specific adaptor plates are required for some platforms. In addition, depending on the vehicle configuration (hybrid or fully electric), spline mating interface with an axle driveshaft is also required.

4 In Wheel Motor performance

The performance of the in wheel motor is a function of the air gap length and motor length. Furthermore the torque is proportional to both airgap length and motor length.

Therefore, if we plot the change in torque due to the change in airgap length as a result of possible bearing stiffness, we see that the curve exhibits a similar shape to that seen in Figure 6.

Figure 9 shows the percentage of torque versus bearing stiffness resulting from the bearing stiffness influence on the air gap length for different designs of bearing.

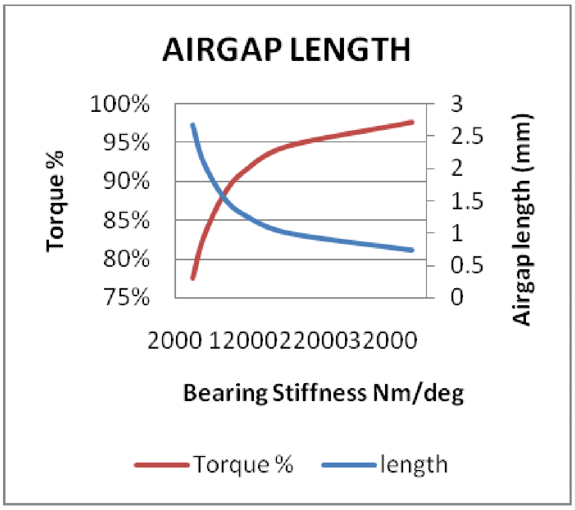

Figure 9

Figure 10 shows the percentage of torque versus bearing stiffness resulting from the bearing stiffness influence on the motor length for different designs of bearing. It can be seen that both the air gap length and motor length have similar contributions to the motor torque despite the fact that the bearings contribution to the airgap length is much greater, as demonstrated in Figure 7.

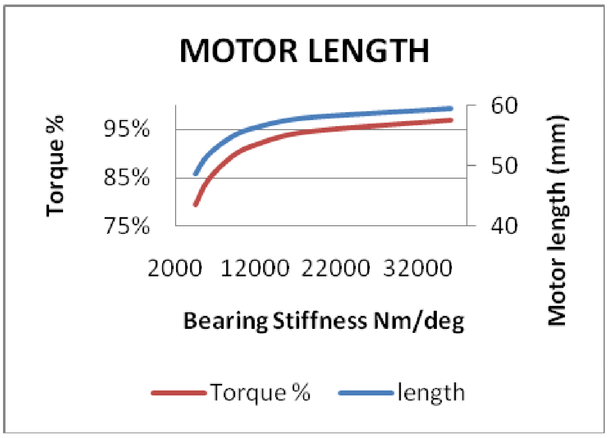

Figure 10

Figure 11 sums the effects of Figures 9 & 10 to show the percentage of torque versus bearing stiffness resulting from the bearing stiffness influence on both the motor length and airgap length combined for different designs of bearing. These three graphs demonstrate that the bearing stiffness has the potential to greatly influence the performance of the motor when producing designs within a given envelope.

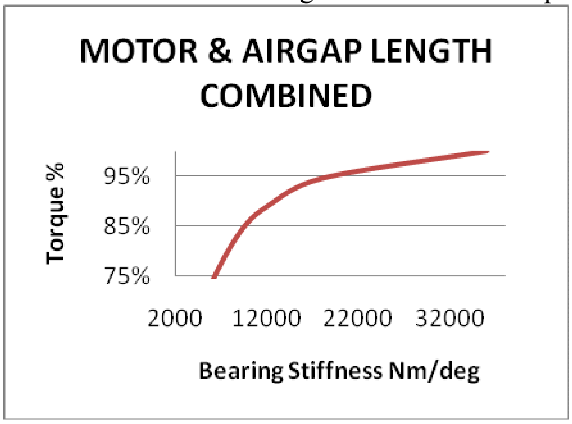

Figure 11

5 Wheel Bearing Performance

The air gap between magnets and wound teeth under different operating conditions is mainly controlled by the performance of the wheel bearing. Figure 12 shows the In-Wheel motor assembly, highlighting the flange interface between In-Wheel motor and bearing (red rectangle). Figures 13 and 14 show the bearing flange deflection under low and extreme cornering loads. The contour line represents the undeformed configuration.

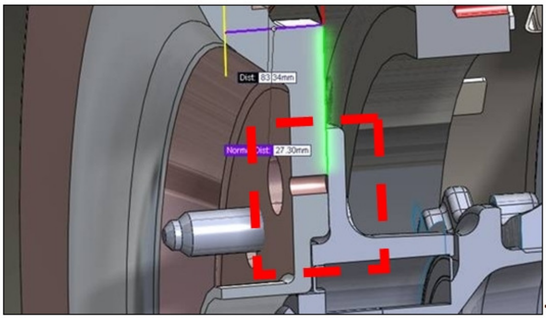

Figure 12 - IWM assembly

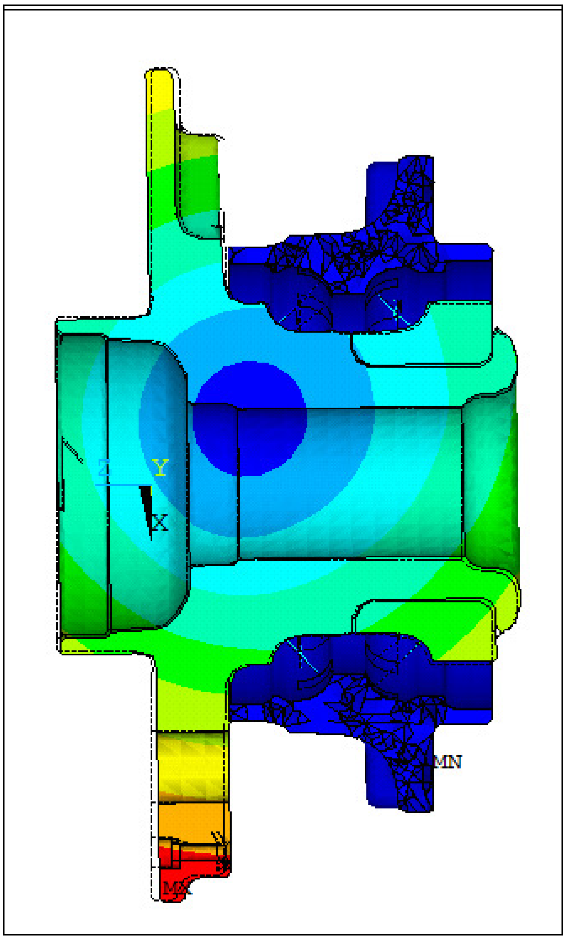

Figure13: bearing flange deflection under low cornering external loads

From Figure 14, it is possible to appreciate that bearing flange deflection under extreme cornering occurs without shape distortion to the bearing flange. Such a deformation (similar to rigid rotation) has beneficial impacts on air-gap length variation, which is also important for In Wheel Motor performance as well as radial air-gap variation.

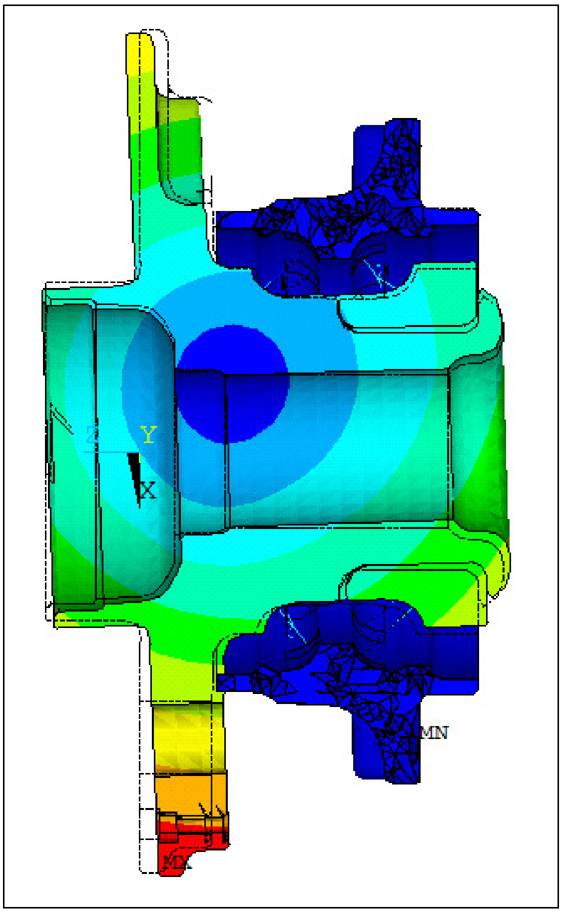

Figure14: bearing flange deflection under extreme cornering external loads

Figure 15 and Figure 16 shows the relationship between bearing performance in terms of stiffness and In Wheel motor air-gap close-up control.

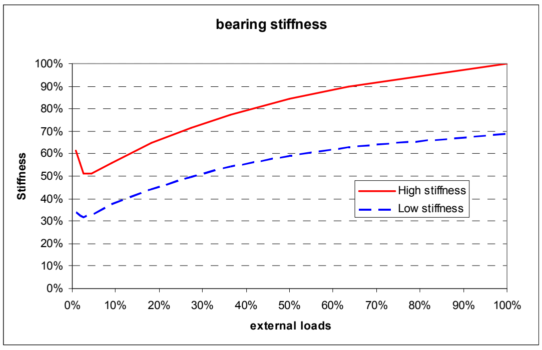

Figure15: Bearing stiffness

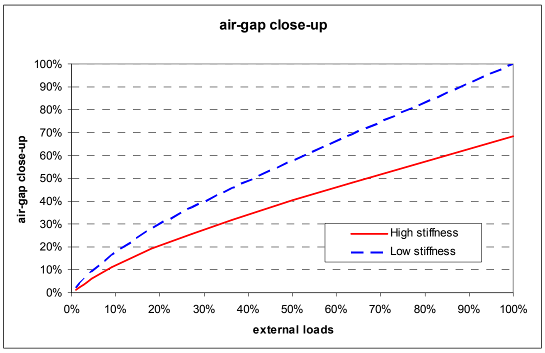

Figure16: In-Wheel motor air-gap close-up

Increasing bearing stiffness in a given design results in an increase of bearing size (and consequently weight). For this reason, bearing design activity must be optimized, in order to reach exactly the required stiffness target, without exceeding the space envelope available.

6 Design Process

Bearing design development makes use of numerical simulations in order to ensure all mechanical requirements are fulfilled. Bearing stiffness must ensure a maximum flange deflection corresponding to 50% of air-gap variation. Structural strength of bearing flanges must cope with structural fatigue requirements, targeting product life. Rolling elements must cope with rolling fatigue endurance requirements.
Once design is developed and released, bearing performance will be verified through a complete testing campaign performed on prototype parts.

7 Conclusion

It has been shown that bearing stiffness has a large influence on optimising performance. Great care must be taken to optimise the stiffness characteristics, package dimensions and weight for a given application.

Acknowledgments

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Authors

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Gareth Roberts graduated from Brighton Polytechnic with a Bachelor of Mechanical Engineering in 1990. After graduation, he worked for Ricardo Consulting Engineers as a CFD & FEA analyst and as a design engineer. While at Ricardo he gained a Master of Science in Automotive Engineering from the University of Hertfordshire in 1997. Gareth has also worked for Ford Motor company and the Engine design and development group of Southwest Research Institute. He is currently the Manager of Mechanical Systems at Protean Electric Limited.