ENGINEERING THE SOUND OF THE JAGUAR F-TYPE

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1 INTRODUCTION

Jaguar has a rich history of producing sports cars, the influence of which is evident in the DNA of the brand. The F-Type is the most focused driver product in the Jaguar product portfolio, demanding the dynamic characteristics of the brand DNA to be clearly evident. The sound of the engine during spirited driving is a key element of the experience.

Before the launch of the F-type, the range of vehicles available from Jaguar included Saloons (XJ, XF), Sports Saloons (XFR) and a Sports GT (XK). The high-performance derivative of these products utilise V8 engines which represent the latest generation of a line developed over 15 years. As these product lines have matured, so have the characteristics engineered into the sound of the V8 engine to communicate the brand DNA.

The industry trend for efficiency improvement through engine down-sizing is reflected by Jaguar with the development of a new V6 gasoline engine which is used in a high performance derivative of F-Type. With recent experienced focused of delivering V8 based engine sound quality, significant work was required to develop an understanding of how the Jaguar DNA may manifest in the context of a V6 engine sound.

2 PROCESS FOR ENGINEERING ENGINE SOUND QUALITY

The steps for engineering the sound of a product like the F-Type are as follows:

1) Define the product positioning in the Jaguar portfolio and in comparison with competitor vehicles.
2) Digitally develop a target sound to support the product position and communication of the brand DNA.
3) Determine the vehicle hardware strategy for the delivery of the target sound character.
4) Verify, using acoustic modeling and simulation techniques, that the vehicle hardware assumptions are capable of delivering the target sound within a range of tuning capability.
5) Demonstrate that the hardware can deliver the desired sound on early prototypes within a range of tuning capability.
6) Tune the sound of the car using fully representative prototypes.

It is important that the first four steps are as robust as possible in order to deliver early prototypes which are close to the desired sound characteristic. It is during these early phases of that the scope for engineering change can contain modifications to the designs to significantly influence the character of the sound. Once prototypes have been built, much of the hardware tooling costs has been committed, making it very expensive to make significant design changes.

3 MECHANISMS USED TO ENGINEER V6 SOUND QUALITY

In order to maximise the robustness of the engineering of the vehicle sound quality, it is necessary to understand the mechanisms responsible for the generation and the propagation of the noise into the cabin.

The mechanical noise generation mechanisms for most configurations of internal combustion engine are well understood. The propagation mechanisms are either structure-borne i.e. transmitted as vibration energy via physical connections between the powertrain and body structure, or air-borne i.e. transmitted acoustically from radiating noise sources.

3.1 Structural Noise Generation and Transmission

Mechanical engine excitation mechanisms can be categorised into those that are inertia related and those that are combustion related. The complex nature of internal engine dynamics gives rise to a large number of mechanisms, of which the most significant are summarised here.

3.1.1 Inertial Excitation Mechanisms

The inertial excitation mechanisms generally increase with the square of engine rotation speed. They are generated as a consequence of the acceleration of the moving parts within the engine, which include the crank-slider mechanism and valve-train.

The crank slider mechanism creates a torque fluctuation around the axis of the crank (vehicle for-aft or x-axis) due to offset internal forces being reacted by the cylinder housing and crank bearings. The frequency of excitation depends on the number of cylinders. For a V6 engine running a 4-stroke cycle this is 3 times the engine speed, or 3rd engine order (3e). This manifests itself as vertical vibration at the engine mounts and torque fluctuation in the driveline.

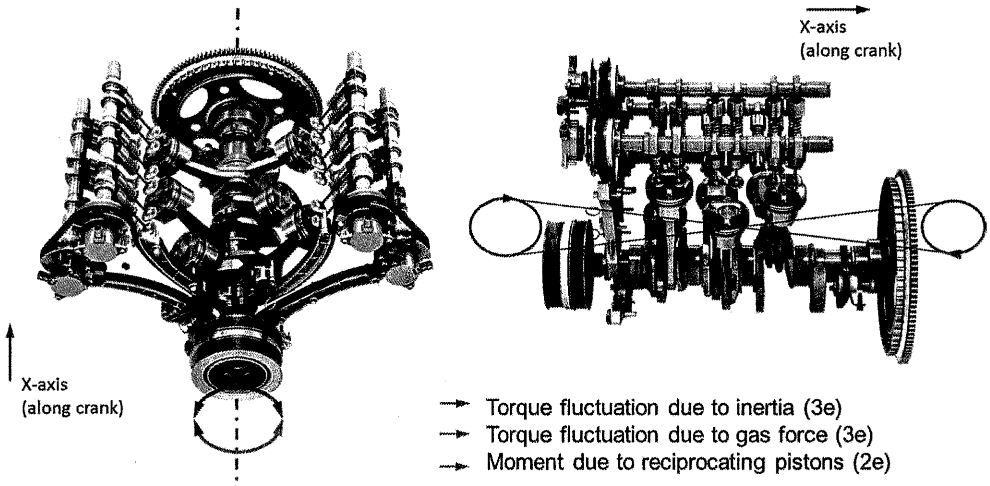

Figure 1, Moments and torque fluctuations applied due to resolved internal forces.

In addition, the pistons apply an axial force due to their reciprocating motion in the cylinder bores. The odd number of cylinders per bank in a V6 causes these forces to resolve along the length of the crank shaft to rolling moments around the vehicle lateral (Y) and pitching moments about the vertical (Z) axes. The frequencies generated are 1st and 2nd engine order. The V6 fitted to F-Type uses balance shafts to cancel the 1st order moment, leaving the 2nd order to manifest itself as lateral and vertical vibration at the mount attached to the gearbox.

Further inertial excitation is applied by the valve-train, which generates a harmonic series based on 1.5th engine order.

3.1.2 Combustion Excitation Mechanisms

Combustion excitation of the engine structure occurs as a consequence of the firing and rapid pressure rise in the cylinder. This can be considered as a series of impacts applied to the pistons in turn during the firing cycle. For a 4-stroke V6, the firing cycle is two complete revolutions, so the rate of impact is 3 per revolution. This gives rise to 3rd engine order excitation with higher harmonics due to the shape of the pressure pulse in each firing event.

The fundamental firing excitation manifests itself as a torque fluctuation around the crank-shaft axis, the magnitude of which is dependent on the force applied to the piston as a consequence of combustion. Therefore the torque fluctuations vary with the load demanded from the engine with maximum excitation observed at full open throttle.

The 3rd engine-order torque fluctuations due to combustion oppose those that are generated due to inertia. Therefore, depending on the relative amplitudes, cancellation can occur. Figure 2 shows 3rd engine order acceleration measured on the left-hand engine mount bracket on the vertical (Z) axis with the engine running at full load. In this case, the torque fluctuations due to combustion are dominant at low speeds with the inertia-related fluctuations causing the reduction at higher engine speeds due to the cancellation effect.

The responses measured at the engine mounting brackets are further modified by the modal behaviour of the power unit. The major bend and torsion modes are in the 150-200Hz region and become more numerous as frequency increases. These can cause significant peaks and dips in the vibration at the mounts as seen in the curves in Figure 2.

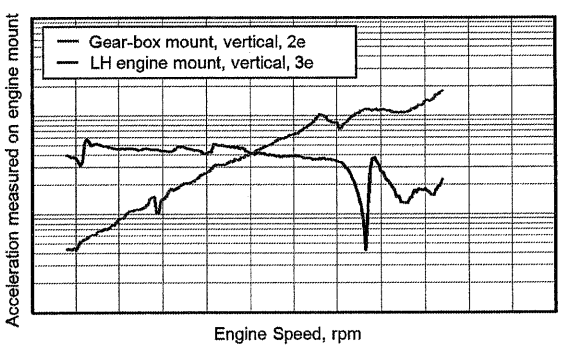

Figure 2, acceleration measured on the left-hand engine mount and the gear-box mount, full load.

Other significant features include the bending modes of the engine mount brackets which typically occur in the 600-1000Hz region. These can introduce increased forcing into the vehicle body structure.

The modal behavior of the power-unit at higher frequencies can manifest itself as inter-cylinder variability when observed as vibration at the mounts. This gives rise to intermediate ½ engine order content with a corresponding increased perceived "rough" character.

In summary, the numerous excitation and transmission mechanisms present forcing signals at the power-unit mounts which exhibit complex harmonic patterns which vary with engine speed and load. Furthermore, these combine with complex vehicle transfer functions to generate a unique mix of characteristics which define the sound experienced by the occupants. The fundamental character is defined by the layout of the power unit. However, the balance of the sound can be changed by modifying the engine, mount and body characteristics to remove or introduce resonance features as desired. All of these items are controlled on the F-type to support the delivery of the desired engine sound in-vehicle.

3.2 Airborne Noise Generation and Transmission

The key airborne noise sources for sound quality tuning are the air intake and exhaust systems. Other airborne noise sources include the power-unit itself which contributes mainly high-frequency noise. This is normally minimised by the implementation of a vehicle acoustic pack designed to maximise noise insulation.

3.2.1 Air Intake System Noise

The action of the engine air intake valves creates pressure pulses in the intake manifold. The resulting engine harmonics signature is determined by the rate at which the pulses occur and the configuration of the manifold. The resultant waveform propagates towards the intake orifice, opposite to the direction of air flow.

The Jaguar V6 engine incorporates a twin-scroll supercharger to provide forced air induction. This has 2 major effects on the intake noise. Firstly it provides high levels of attenuation of the acoustic pressure wave propagating from the intake manifold. Secondly, it introduces its own series of harmonics associated with the passing of the lobes on the scrolls through the outlet port. This mechanism generates a characteristic "whine" commonly associated with superchargers.

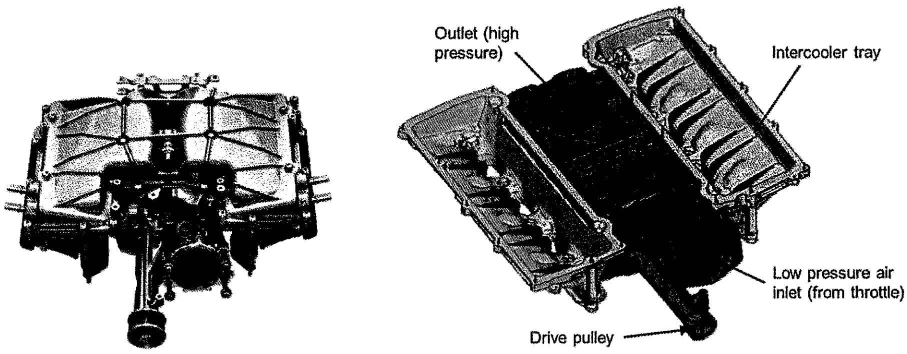

Figure 3, layout of supercharger system.

For the F-type, the supercharger noise propagating through the intake is treated to attenuate it to a level so as not to detract from the fundamental engine sound character. In order to use the intake system to enhance the sporting character of the engine sound, an additional noise path has been engineered in the form of an intake feedback system. This consists of a pressure tapping from the intake manifold, a static pressure isolation device (Mann & Hummel Symposer), a valve and a sealed orifice in the vehicle cabin.

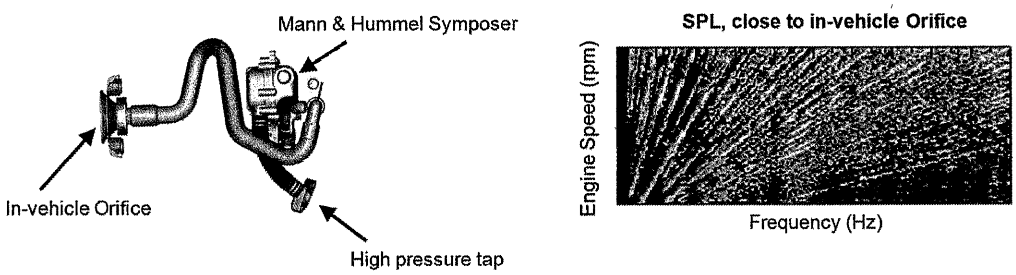

Figure 4, layout and orifice noise of the intake noise feedback system.

The intake feedback system provides content at high engine speeds to enhance performance driving. The system is tuned to propagate a sound which is harmonious with the other noise sources in the vehicle and is consistent with the Jaguar DNA. The valve is used to disable the feedback at lower engine speeds and loads where it is not required and to avoid the transmission of supercharger noise.

3.2.2 Exhaust Orifice Noise

The source of acoustic excitation of the exhaust system is the release of pressure into the exhaust manifold after combustion. For the V6 engine layout in the F-Type, each bank fires alternately, creating in effect two separate 3-cylinder engines. Typical pressure pulses measured in the manifold close to each cylinder are shown in figure 5, in which the red and blue traces represent events measured in each bank. The chart covers two complete revolutions of the crank.

The pressure traces show that the banks have alternate events with a rate of 1.5 times the engine speed, generating a fundamental firing order of 1.5e. In order to investigate how the layout of the exhaust could be used to engineer the character of the sound radiated from the orifice, a range of acoustic models were constructed using Ricardo WAVE software.

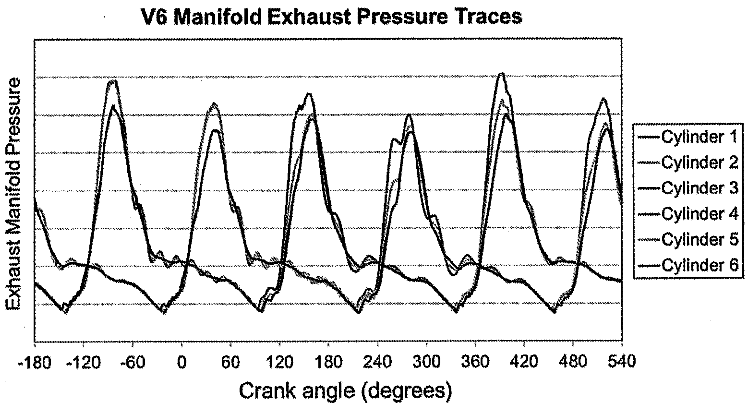

Figure 5, pressure measured in the exhaust manifolds.

Figure 6a shows a system which has separate exhaust runs for each cylinder bank with equal length pipe runs in the manifolds. This generates a harmonic series based on a fundamental of 1.5^{th} engine order at each tailpipe. When the banks are mixed with equal length downpipes, the resultant tailpipe signal has a harmonic series based on 3^{rd} engine order, as shown in figure 6b. Subjectively, this has a pitch that is double that of the unmixed case and is smoother in character due to reduced modulation of the higher harmonics. It is possible to generate a combination of these signatures by introducing asymmetry in the exhaust downpipes before mixing.

In addition to variation in downpipe layout it is also possible to significantly modify the sound quality by changing the relative lengths of the pipes (runners) in the exhaust manifolds. Variations in propagation time between the individual cylinders and the point at which they mix introduce harmonics based on a ½ engine order series. This is because each cylinder fires once every two cycles. Figure 6c shows the output of an acoustic model with unequal length manifold runners with equal length downpipes in a mixed exhaust system.

The layout shown in figure 6c has strong 3^{rd}, 6^{th} and 9^{th} engine orders present, with additional ½ engine order harmonics becoming more dominant at higher frequencies. These modulate to create an impulsive character which is most obvious ay high engine speeds.

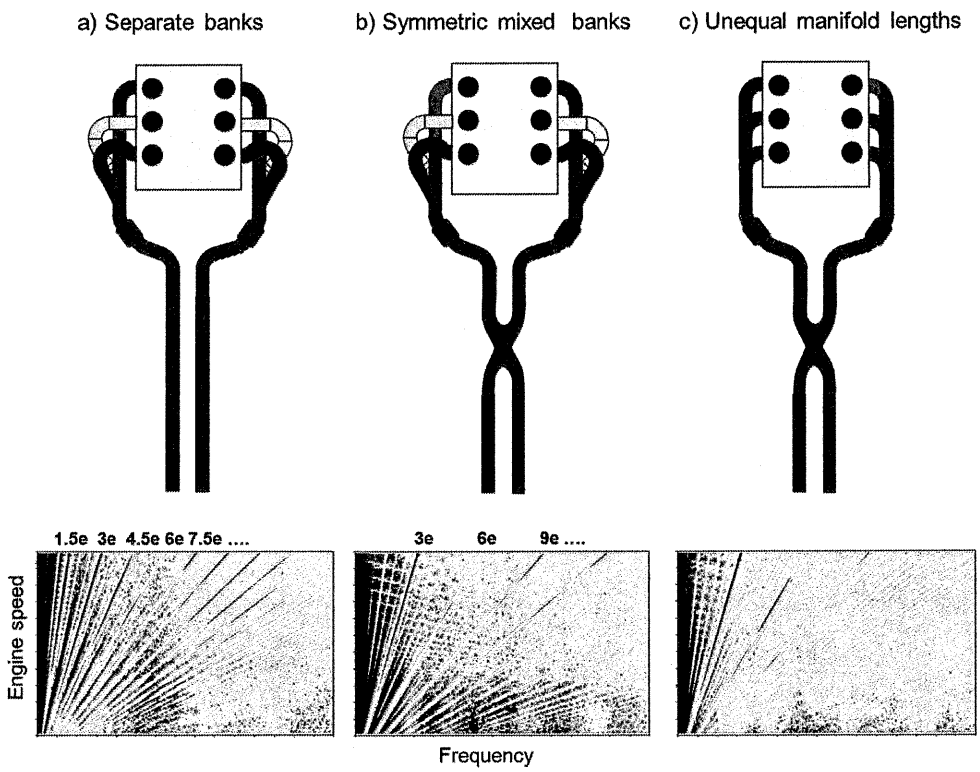

Figure 6, effect of exhaust configurations on individual tailpipe signature, full engine load.

a) Separate symmetrical exhaust banks, equal length manifold runners.
b) Mixed exhaust banks with equal downpipes, equal length manifold runners.
c) Mixed exhaust banks, unequal length manifold runners.

In-vehicle the exhaust layout is limited by the packaging of the power-unit in the engine bay which can accommodate only the unequal length manifold configuration. The assessments of the acoustic model confirm that the impulsive character generated at higher engine speeds is consistent with the Jaguar brand DNA. This character is comparable to the V8-engined vehicles and works in harmony with the contribution from the Symposer intake noise feedback device.

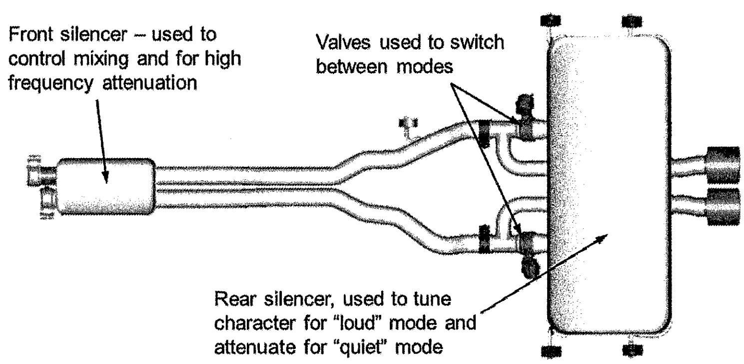

Figure 7, exhaust layout after downpipes

The downstream layout of the exhaust system was developed to maximize the tuning range for the containment of noise levels and the delivery of sound quality within the constraints of the under-floor package. The configuration of the system is illustrated in figure 7.

The small silencer is used after the downpipes to control the mixing between the banks and to provide high-frequency attenuation using expansion.

The under-floor pipes split into two paths per side prior to entering the rear silencer, in which a central solid baffle maintains the separation of the banks. This layout creates two possible paths for gas flow and acoustic propagation through the rear silencer before combining at a common tailpipe per side.

Valves are used prior to the rear silencer to disable propagation through one of the paths, which is referred to as the "loud" route. This path has minimal attenuation, consisting of high frequency absorption and discontinuities to introduce resonance features in the sound radiated at the tailpipe.

The second path is always enabled and follows a much more restrictive route through the rear silencer, using the expansion volume to control noise levels to create a "quiet" route. This path provides gas flow restriction which would inhibit the performance of the engine at higher speeds. Therefore, the loud route is always enabled above 3000rpm under high engine load.

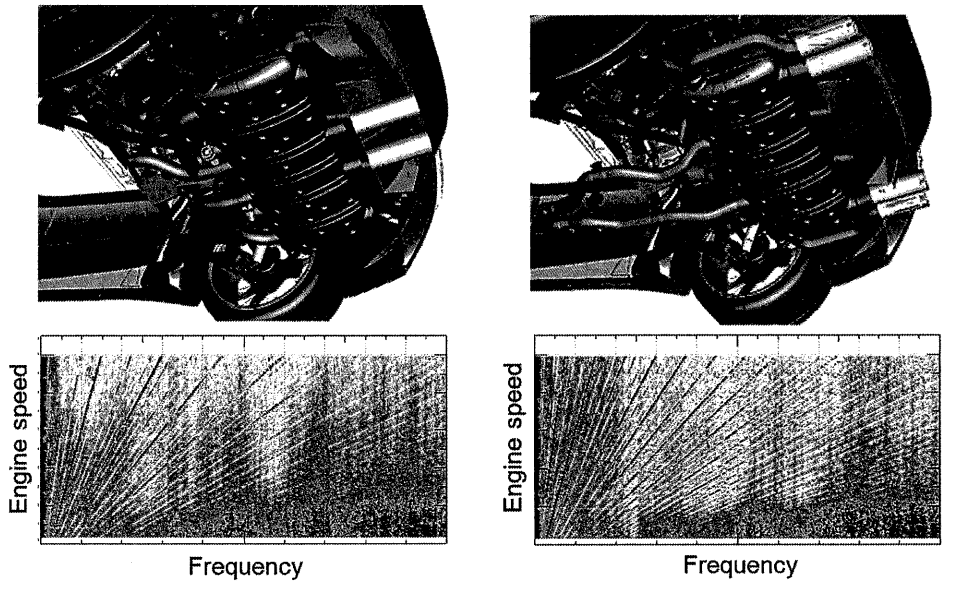

Figure 8, Effect of tailpipe configuration on interior noise (roof down)

At the rear of the exhaust are central inboard twin tailpipes. This configuration was introduced to provide a styling cue to distinguish the V6 powered derivative from the very high performance V8. The close spacing of the tailpipes has the effect of mixing the banks, limiting the opportunity to generate the 1.5^{th} engine order harmonic series.

Figure 8 shows interior noise measurements made on a vehicle with the inboard and outboard tailpipe configurations with notionally similar silencer internal layouts and identical under-floor configurations. The inter-bank mixing effect is clearly evident with the 1.5 engine order harmonics significantly reduced with the inboard tailpipes. Assessment of these conditions from both simulated and tested results confirmed that the inboard condition is preferred and has a more tonal and less "distorted" quality.

4 TUNING THE VEHICLE SOUND QUALITY

As a result of the simulations undertaken during the vehicle design phase and test work completed on surrogate vehicles, prototypes were delivered with exhaust and intake feedback systems which delivered noise content close to the desired character of the engine sound. In addition, a number of potential component modifications had been identified to enable fine tuning of the sound character.

The vehicle tuning exercise was required to match the sound delivered with the overall character of the vehicle. Key activities on the F-type included setting the valve switching points for both the intake feedback and exhaust systems, tuning the high-frequency response of the exhaust and modifying the level of mixing in the under floor chamber.

In all a large number of iterations were assessed to strike a balance between the driving characteristics of the vehicle under normal and performance driving in roof up and roof down conditions. Figure 9a shows the final status of the vehicle interior sound measured under full load on a Vehicle Semi-Anechoic Chamber (VSAC).

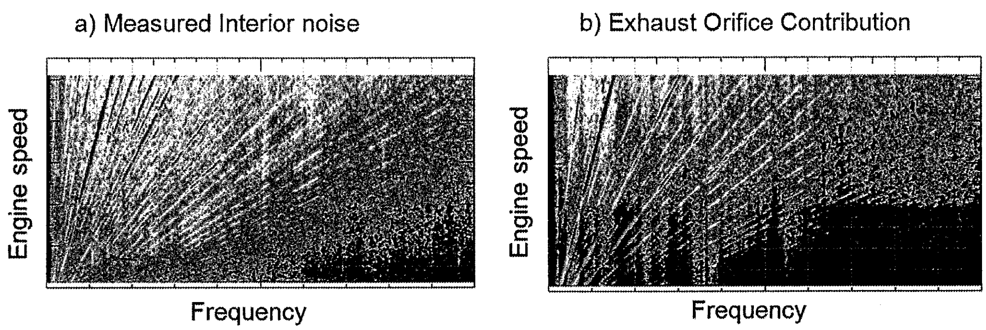

Figure 9, Final Interior Noise signature, roof closed position, full engine load, VSAC
a) Total measured interior noise and driver's outboard ear.
b) Contribution of exhaust tailpipe to interior noise, driver's outboard ear.

5 CONCLUSIONS

The use of a supercharged V6 power unit in the Jaguar F-type posed significant challenges to the project team responsible for delivering the sound character of the car. The application of advanced acoustic and structural modelling techniques enabled design opportunities in the exhaust and intake system to be identified and exploited before prototypes had been built.

The significant opportunity to fine-tune the vehicle at the prototype stages contributed to delivery of a focussed sports car with engine sound quality which communicates the character of the F-Type and that of Jaguar.

6 REFERENCES

1. Bosch. Automotive Handbook, 3rd edition, Robert Bosch GmbH, 376-389. (1999).