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Turbocharger Structural Integrity

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Keywords

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Abstract

Since the introduction of Euro VI in January 2014, all new diesel powered commercial vehicles have been equipped with turbocharged engines. It is virtually impossible to meet these emission regulations without using a turbocharger. Similarly, in the passenger car sector both on diesel and petrol (gasoline) powered vehicles, legislative pressure to reduce emissions of carbon dioxide are seeing the introduction of turbochargers across almost all new power units. Future legislation will continue this trend with engine manufacturers becoming increasingly reliant on turbocharging. As well as increasing the requirement for turbochargers, these external factors are also demanding that turbochargers become more responsive with reduced rotor inertia and lower thermal inertias. This in turn makes the task of ensuring that turbocharger components remain fit for purpose for the life of the turbocharger that much more difficult. In this paper some of the recent developments in turbocharger technology will be identified and the demands that these place on the structural components will be explored. The limitations of current methods of structural integrity assessment for some of these components will be discussed. Future developments of these methods will then be proposed.

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Introduction

Since 1st January 2014 all new commercial vehicles sold in the European Union have had to comply with the Euro 6 emissions regulations laid out in EU Regulation595/2009 [1]. This latest version of the emissions regulations has placed even more stringent requirements on vehicle manufacturers with regard to nitrogen oxides of nitrogen and particulate matter measured at the exhaust. The dramatic reduction in emissions since Euro 3 was introduced in 2006 is illustrated in Fig. 1. In order to comply with these regulations, engine manufacturers have used a dual approach of exhaust gas after treatment and optimization of the air-fuel mixture with the use of turbocharging, as shown in Fig. 2.

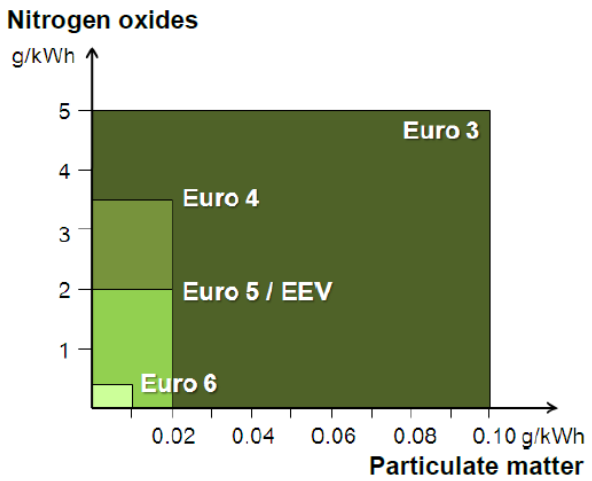

Figure 1 Heavy goods vehicle diesel engine emissions regulations (taken from [2])

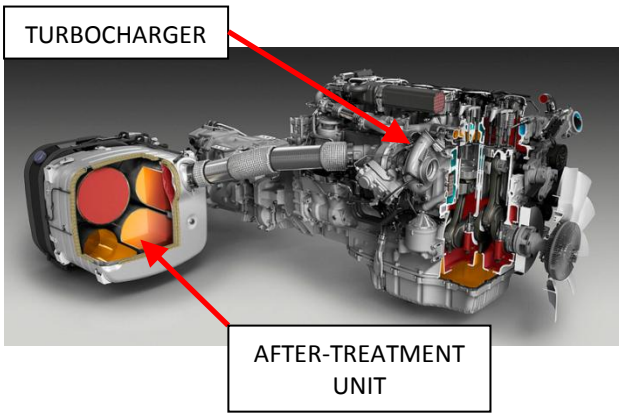

Figure 2 Scania Euro 6 engine and after-treatment system (taken form [2])

An additional demand for turbochargers also arises from emissions of carbon dioxide. In a number of countries and states, taxation of passenger cars is directly linked to the amount of carbon dioxide generated in a standard duty cycle. For example, in the UK there is zero taxation for vehicles generating up to 100 g/km and a tax of £500/year (≈€630/year) for CO_2 emissions over 225 g/km. In response to this, manufacturers are reducing the size of their engines either permanently or for parts of the duty cycle. For example, the 1 litre Ford EcoBoost engine [3]with emissions of 109 g/km is being used for the latest range of people carriers whilst VW have introduced active cylinder technology [4] to reduce the effective engine capacity when engine torque demands are low. In both engine types, a turbocharger is essential to allow the engine to deliver peak power demands. Similar downsizing of engines with turbocharging is being implemented in the larger

engine, commercial vehicle sector to improve fuel efficiency and reduce operator costs.

EXPERIMENTS

Identification of structural problem areas: In use, a turbocharger is subjected to a number of harsh loading conditions which can potentially result in structural failure. These can be broadly categorized as:

External loads due to equipment attached to the turbocharger: The turbine side of a turbocharger is an integral part of the engine exhaust system. As such, it is subjected to loads due to thermal expansion of components and vibration of other components due to engine running vibrations, engine movement during acceleration and deceleration and whole vehicle movement. On the compressor side, components such as air filter housings and charge air coolers place similar vibratory loads on the turbocharger although the thermal loads are much less significant.

Internal operating loads: The turbocharger rotor will typically rotate at speeds in excess of 100,000 rpm during engine acceleration. This places considerable loading on the rotor components. In addition, the turbine and the turbine housing are subjected to temperatures which in a gasoline engine, can be in excess of 950°C. Finally, the purpose of the turbocharger is to extract energy from the pressurised exhaust gas and pump this back into the engine as pressurised induction gas. Whilst these pressures are not large, they must still be contained and they will generate bending, axial and torsional loads on the turbocharger.

Loads due to failure of the turbocharger: It has long been recognised that turbocharger rotor failure is a risk that is extremely difficult to eliminate and must therefore be guarded against. Whilst rotor assemblies in smaller goods vehicles and passenger cars are not large, their high speed gives them a considerable amount of energy. This energy must be contained within the turbocharger should the rotor fail.

The major structural components of a typical turbocharger are identified in Fig. 3. It is worthwhile considering how these components interact with the various loading mechanisms.

Turbine housing: As outlined above, this component will be subjected to substantial external loads as it forms an integral part of the exhaust system load path. It will also be subjected to very high thermal loading that is cyclical in nature. These loads will combine to generate fatigue failures. However, the highest loads that the turbine housing will be subjected to are those resulting from the failure of the turbine wheel. Whilst it is accepted that the housing will almost invariably be damaged beyond repair by such an incident, it is essential that the housing contains all the turbine rotor fragments.

Compressor housing: Again, there are significant external loads on the compressor housing due to vibration of engine components. Internal pressure must also be contained and any resultant loads accommodated. However, the largest loads are again those due to rotor failure. The damage that can result from a compressor impeller burst is shown in Fig. 4 which shows the result of a deliberately induced compressor impellerburst. Whilst the impellerfragments have been successfully contained, the damage to the housing is considerable.

Bearing housing: In some applications the bearing housing may be bolted directly to the engine. It will therefore be in the path of external vibratory loads applied to both the turbine and compressor housings. In applications where the attachment to the engine system is achieved by the turbine housing being attached to, or being integral with the exhaust manifold, the bearing housing is still subjected to loads applied to the compressor housing. In addition, during either turbine wheel or compressor impellerburst, considerable axial and torsional loads may be applied to the relevant housing which must in turn be reacted by the bearing housing.

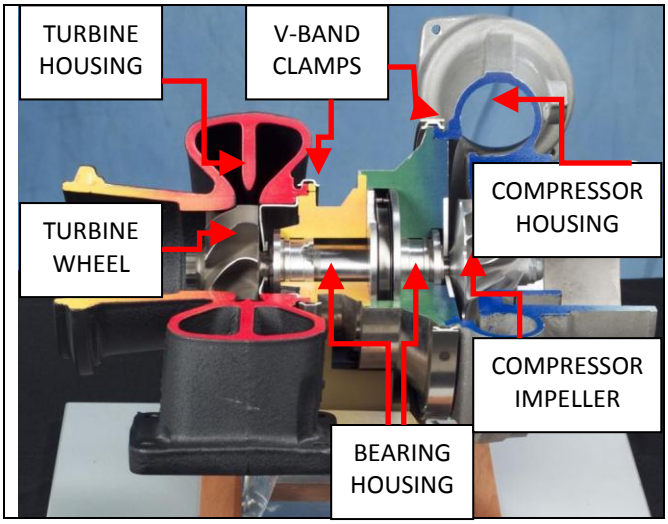

Figure 3 Turbocharger structural problems areas

Figure 4 Compressor impeller failure (taken from [5])

V-band clamps:These apparently simple components are now widely used on modern turbochargers to attach the three housings together. As such they are an essential part of the same load path as the bearing housing both for external loads and burst and containment loads. An example of the failure of a V-band clamp during a burst and containment test is shown in Fig. 5.

In addition to guarding against containment failure, V-band clamps must also accurately locate the housings to maintain the very small clearances between the rotor wheels and the housings. If the clearance between the wheel and the housing increases, turbocharger efficiency decreases significantly.

Conversely, closure of this gap will result in a wheel rub and frequently a destructive event.

Rotor assembly: This assembly comprising the turbine wheel, compressorimpellerand the connecting shaft is subjected to very substantial centrifugal loads. As such, the components must be carefully designed and manufactured to prevent early life failure. Failure of the assembly can also occur if it is operated above the design speed, a situation that can occur due to a fault in the turbocharger control system. Also, as identified by [5], ingestion of foreign bodies into the compressor will very often result in a wheel failure event. Recent developments in engine technology, particularly engine downsizing have placed an increasing demand on turbochargers to respond rapidly to changing operating conditions. This requires the rotor to accelerate rapidly which can only be achieved by reducing rotor inertia.

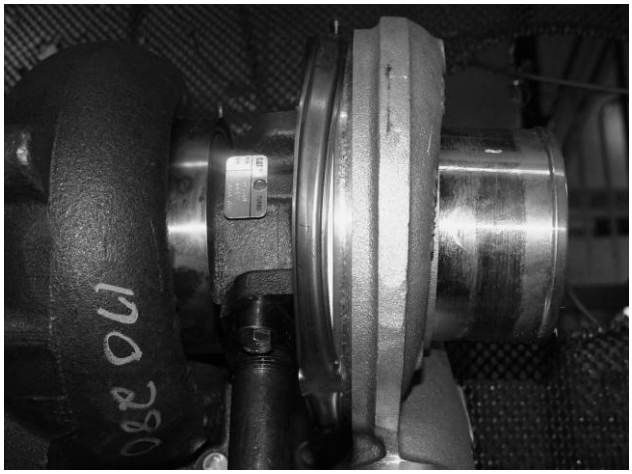

Figure 5 Compressor end V-band failure during burst and containment test (taken from [5])

Mitigating against structural failure: Low cycle fatigue (FCL): As Christmann et al [7] identify, low cycle fatigue is characterized by the material being loaded beyond its elastic limit such that plastic deformation takes place. All the structural components within the turbocharger can fail due to LCF if not correctly designed and used but three are particularly susceptible to this type of failure: the compressor and turbine wheels and the turbine housing.

For both compressor and turbine wheels, low cycle fatigue can occur due to the demand to minimise the mass and hence inertia of these components. They are therefore designed to meet the demands of normal operation within the material's elastic limit but with only a small safety margin. Hence, under extreme operating conditions the elastic limit can be exceeded. Whilst a single event of this nature is unlikely to lead to failure, repeated events could result in LCF failure. This was recognised in [7] and it was proposed that on-board monitoring of the rotor speed could be used to estimate the cumulative fatigue damage. This could then be coupled with the turbocharger management system to either limit future rotor maximum speed (and hence reduce peak engine performance), or prompt the operator to have the turbocharger replaced. Whilst this technology is feasible, the authors concluded that a preferable route would be to mill compressor wheels from stronger billet aluminium rather than casting them. However, the implementation of this type of technology should not be dismissed.

For the turbine wheel the estimation of cumulative damage would have to take into account variations in temperature of the wheel both to estimate the thermal stresses in the wheel and also to account for the effect of temperature on the rate of fatigue damage. A further barrier is that itwould be more difficult to monitor wheel temperature than speed.

As described by Nagode et al [8], the prediction of low cycle fatigue failure of the turbine housing has to take into account the effects of creep as well as plastic deformation. This significantly complicates the analysis and as the authors have pointed out, further material data from biaxial testing at elevated temperature is required to validate the multi-axial behaviour of the material model.

High cycle fatigue (HCF): High cycle fatigue is characterized by the material being subjected to cyclic loading at levels within the elastic limit of the material. This can occur within turbochargers in both the turbine and compressor wheels due to the varying gas pressure around the circumference of the volume swept by the wheel. Hence, as an individual blade on the wheel rotates around housing, the load on it varies. Since the wheel can be rotating at speeds in excess of 100,000 rpm, the total number of load cycles the blade is subjected to will exceed 10^9in less than seven days. This situation is made worse when a wheel natural frequency being excited is a multiple of the load cycle. For example, a blade could be excited at frequencies in the kHz range resulting in a wheel that would self-destruct in seconds. To avoid this problem, wheels are designed to avoid critical resonances where possible or to incorporate sufficient damping to ensure that stresses at resonance are low. This can be achieved by 'de-tuning' the wheel such that resonance in one blade does not match resonance in another. To date this detuning of the wheel is largely fortuitous: there is sufficient variability in the manufacturing process to ensure that blades are not matched.

In an attempt to better control blade resonance and hence avoid high cycle fatigue failure, finite element analysis (FEA) can be used to determine natural frequencies and the impact of material damping. As with any FEA, some form of validation is then required. The commonest technique is to attach a small number of strain gauges either all to one blade or one each on a number of blades and then run the rotor through its speed range. However, as Allport et al [9] point out, this gives very limited information about the response of the wheel and also, the addition of the strain gauge to the wheel changes its dynamic response. To determine the simultaneous behaviour of all the blades on a wheel and to provide more information regarding the behaviour of individual wheels, an optical system has been developed as described in [9].

Now that a robust technique is available to validate the FEA modal analysis, more refined work can be carried out to develop wheels where HCF is eliminated by design rather than by chance.

Burst and containment: To guard against the consequences of a turbine or compressor wheel burst, manufacturers design turbocharger assemblies to contain the wheel fragments if the burst occurs at a given percentage above the maximum design speed. A recent paper [10] described an FEA procedure that can be used to determine the containment capability of compressor and turbinehousings. However, the correct wheel failure speed could only be obtained by tuning the material failure criteria rather than referring back to material test data. To be fully effective as a design tool, this technique needs to use true material data and typical flaws within the material. A stochastic analysis involving many simulations with a range of initial burst states would also be required.

Manufacturers also validate designs by carrying out burst and containment tests. Turbochargers are run up to speeds a little above their maximum design speed and induced to burst. Typically two tests can be carried out: a hub burst test where the wheel is induced to split from the centre into two or three large sections and a 'blade off' test where only a portion of the wheel containing a number of blades detaches. This second situation can subsequently result in a hub burst as the detached section collides with the remaining wheel but typically the wheel will have lost a portion of its kinetic energy before this occurs. The hub burst test is therefore generally regarded as the most aggressive.

A hub burst can be induced by enlarging the bore of a compressor wheel or drilling a hole into the centre of a turbine wheel. A 'blade off' event is often created by cutting a shallow slot on the back of the wheel. Recent work by Clark [11] found that it was very difficult to predict the speed at which a 'blade off' event would occur for a given slot length, depth and position. Further work is required in this area to correctly relate the actual material properties to the prediction of failure within the FEA model. In addition, whilst the slot method does induce a blade off failure, it may not be representative of an actual blade off event which is more likely to be initiated either by a crack propagating from the root of a blade or a foreign body hitting the blade side of the wheel.

V-band as a structural component: As identified above, as an integral part of the load path between the turbocharger housings, the V-bands are critical components in terms of both external loads and also for containment purposes. In 2003 Shoghi [12] noted that there was very little theoretical understanding of V-band behaviour. Since then progress has been made with Shoghi et al [13] providing a theory for the stresses generated in a clamp when it is assembled onto flangesand also the axial clamping load generated by a clamp[6]. More recently, a theory to predict the axial stiffness of this type of connector has been generated [14] and whilst this agrees well with FEA simulations, it requires experimental validation. Further work is also required to determine how this type of clamp interacts with flanges separated by a flexible gasket. The torsional capacity of a joint formed using this type of connector has also been analysed [15] although refinement of the theoretical model is required for low aspect ratio bands.

Muller [16] recognized that fatigue failure of V-band clamps, as noted in the field would be impacted by the band manufacturing process. Work was therefore undertaken to determine the residual stress and plastic strain generated in the band cross section as it was created by the 6 pass rolling process using an explicit FEA. The end of the first stage of the analysis is shown in Fig. 6(a) and the final, hoop forming stage shown in Fig. 6(b). However, this work was not progressed to the stage where fatigue failure of bands in service could be analysed or predicted.

The ultimate axial load capacity of V-bands has also been investigated [17]. This analysis showed that a theoretical prediction was possible and showed good agreement with FEA simulations. However, it was clear from the experimental work that a number of factors not least the geometric tolerance of the bands, were strongly influencing ultimate failure.

No work appears to have been undertaken to understand the bending moment capacity of V-band joints, the effect of temperature on V-band joints or the observed drop off in V-band clamping load over time.

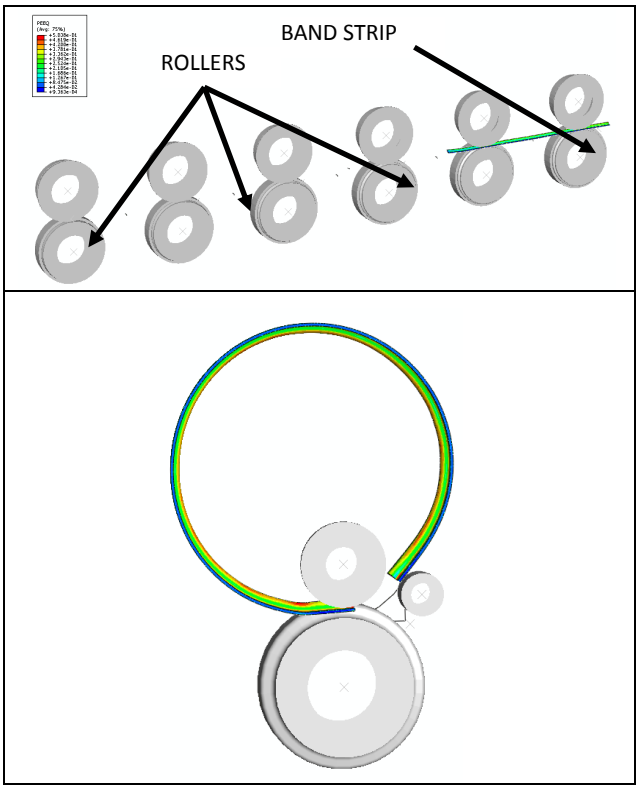

Figure 6 a, b V-band rolling process showing plastic strain (a – 6 pass first stage, b – hoop forming second stage)

CONCLUSION AND FUTURE DIRECTION OF RESEARCH

It is clear that there will be an increasing demand for turbochargers as engine manufacturers seek to reduce emissions. Future turbochargers will have to be more responsive with lower rotor inertias, will have to tolerate increasing exhaust gas temperatures and will need lower thermal inertias. This is going to require on-going work to develop turbocharger technology, particularly in the area of structural integrity. Particular areas of development will include:

- Development of modelling techniques to simulate Wheel burst events coupled with more reliable and robust test methods to reduce the time and cost or validation.
- Methods of initiating blade off events during testing need to be developed to better represent actual events. Associated with this will be improved methods of simulating this type of event during the design stage.
- Substantial development of techniques to analyse blade vibration leading toHCFfailure are required to make the design of wheels and blades more robust. This will include development of the current state of the art optical techniques for measuring blade vibration.
- The understanding of the behaviour of V-band clamps need to be improved. Particular areas for development include:
  - Improving the pressure sealing capability by more evenly distributing the axial load around the flange.
  - Determining the performance of clamps when subjected to impact loading during a burst event.
  - Thermal effects on clamp performance including differential thermal expansion and changes in material properties.
  - Understanding and resolving the issue of clamping load drop off.

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- The development of housings from alternative materials to lower mass, reduce thermal inertia and allow for higher exhaust gas temperatures will require development of more refined techniques for analysing the consequences of wheel burst events and also external loads.

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