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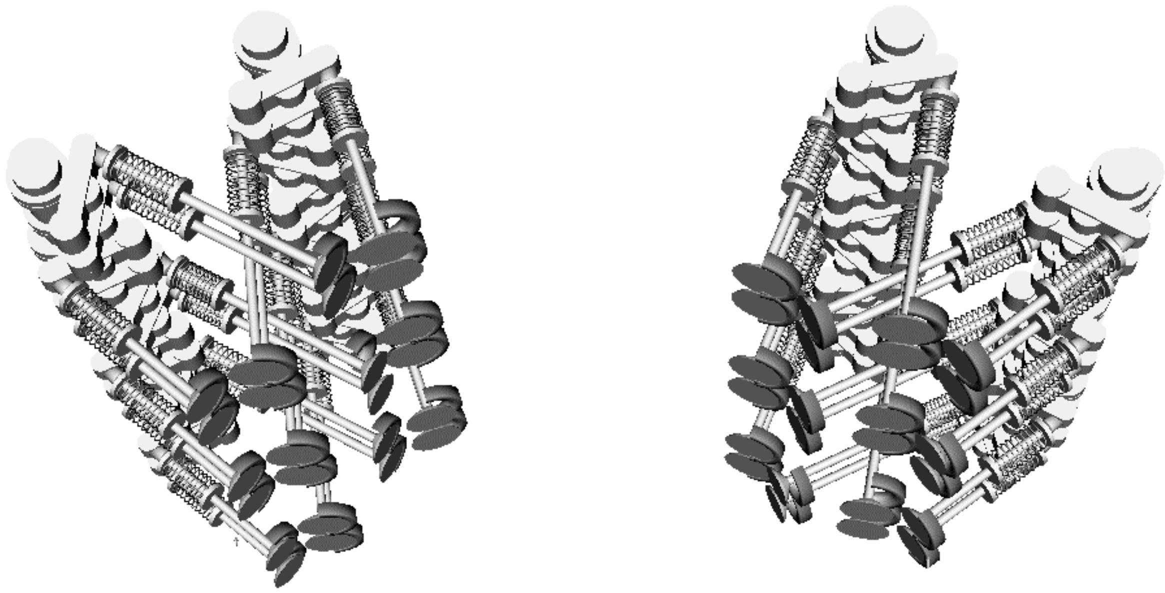
Calculation of Effects of Free Inertial Forces and Torques of the Valve Train to the Engine Vibration of a W12 from VOLKSWAGEN



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1. Abstract

Free inertia forces and torques of the crank train could be evaluated in a comparatively simple way by using analytical equations (Fourier series). The results principally find consideration in the design process of the engine and its mountings.
The analytical prediction of the inertia effects resulting from the moving parts of the valve train is extremely challenging due to the complex cam profiles in conjunction with the rocker arm valve train design and its position and orientation and the resulting valve timing of a VOLKSWAGEN W12 engine.
Especially with respect to this kind of comfort engine the inertia force excitations of the valve train have to be considered.

Therefore a model of the complete valve train is built in ADAMS/Engine. The valve train is attached to an inertia mass, which represents the rigid engine block. The engine block is connected to an inertia mass, which represents the rigid gear box. Both engine block and gearbox are mounted to ground with compliant bushings at the real engine mount positions.
The model is build almost completely parametrically. The reason is to enable the simple variation of many design variables of the engine and especially the valve train to study their effects.
The complete valve train model also allows to analyze the influence of various cam profiles on the engine vibrations.
Knowing the movement of the engine in its mountings induced by the valve train free forces and torques allows you to design more adapted mounts and its positions.

2. Introduction

Engines with a high number of cylinders are known for their low vibration. This means even sources of vibration with a low level of excitation can fall within the zone of perception.
The valve train is a source of vibration of this type. Valves, spring caps, parts of the valve springs and bucket tappets or (in a valve train with finger rocker arms) the levers are accelerated and decelerated by the interaction of cams and valve springs.
The W12 engine from Volkswagen under discussion here has 48 valves which are accelerated by a valve lift function at different times and in different spatial directions.

The objective of the calculation is to examine the effect of the free inertial forces and moments of the valve train on the vibrational displacements at the engine/gearbox bearing points and to indicate potential for reducing the vibration.

To this end, ADAMS/Engine was used to set up a model of the W12 valve train that has a high level parameter component. The complete valve train is connected to a part which has the mass and inertial properties of the entire engine/gearbox unit. In turn, the mass of the unit is supported by flexible bearings at the original engine/gearbox mounting points.

3. Model setup

In the concept phase, the requirement profile for the calculation model of the entire valve train was defined as follows:

- Variability of the model: This means it should be easy to alter the basic dimensions of the crankshaft drive and valve train. Parameters such as cylinder gaps, opening angles and valve angles should be adjustable.
- Simple modification of component properties: It should be possible to modify the masses of moved parts such as valves, finger rocker arms and camshafts as well as the cam profiles in a straightforward fashion.
- The vibrational displacements actually occurring on the mounting points of the unit should be ascertained.

Initially, the permitted degree of simplification was defined for the given task. The calculation model was not to examine the following:

- The dynamics of the single valve trains.
  This means the elements of the single valve train can be rigid bodies and the valve springs can be represented by a simple characteristic curve.
- The interactions between the individual valve trains and the dynamics of the camshafts.
  This means the camshafts can be rigid bodies which are mounted in the cylinder head with a rotational degree of freedom.
- The dynamics of the timing mechanism and the cranktrain.
  It is sufficient to assume a fixed connection between the rotary movements of the camshafts and the drive shaft. The details of the camshaft do not have to be known.
- Excitation of the bending and torsional resonances of the unit.

However, the model was to be conceived in such a way that it could be used subsequently for analysing the interactions between the valve trains and the valve timing gear. In addition, it should offer the option of implementing a timing mechanism and a cranktrain.

This catalog of requirements can be used for deriving the number of templates and subsystems required for the complete valve train model of a W12. The 48 subsystems of the single valve trains are based on 8 templates with different structures. The subsystems of the 4 camshafts, both cylinder heads as well as the engine and the gearbox are also based on their own templates.

The individual subsystems are combined into a complete mechanism in the assembly process. They find their positions and dependencies via what are referred to as roles and communicators. The high level of parameterisation means that a range of information is exchanged between the individual modules and processed. For example, the initial angle of every single cam is calculated from the parameters of the bank angle, row angle, angle of the particular single valve train and the ignition point of the cylinder in question. Its exact position is calculated using the parameters of the bank angle, row angle, relative camshaft position, cylinder gap and the gap between the valve trains.
The complexity of the model demands a high level of documentation. This is especially important if modifications are to be incorporated later on.
As a result, in parallel to the modelling process, each variable, construction frame, communicator and every single element was represented with its interdependencies using coloured flowchart elements in order to document the information and for the purposes of understanding the entire model. Parameter variables are shown in green, input communicators in orange, output communicators in blue, matching names in white and construction frames in yellow.

Fig. 1 shows the structure of the template for the intake camshaft of the right-hand cylinder head. The cams of the valve trains are modelled in the templates of the single valve trains. The cams are given their initial angle position from the camshaft during the assembly process. This is indicated by the red arrow on the right-hand side. The cam angles are calculated on the basis of the angles listed above, which are in part a feature of the camshaft, and are defined there using parameter variables.
The orientation of the camshaft is taken from the cylinder head by means of a communicator. It in turn gets its orientation from the engine block.
The angles of the single valve trains are also taken from the cylinder head.
The angular offset between the intake and exhaust also has to be included in the case of the exhaust camshaft, while the ignition interval of the first cylinder on the left must also be considered for the shafts on the left-hand cylinder head.

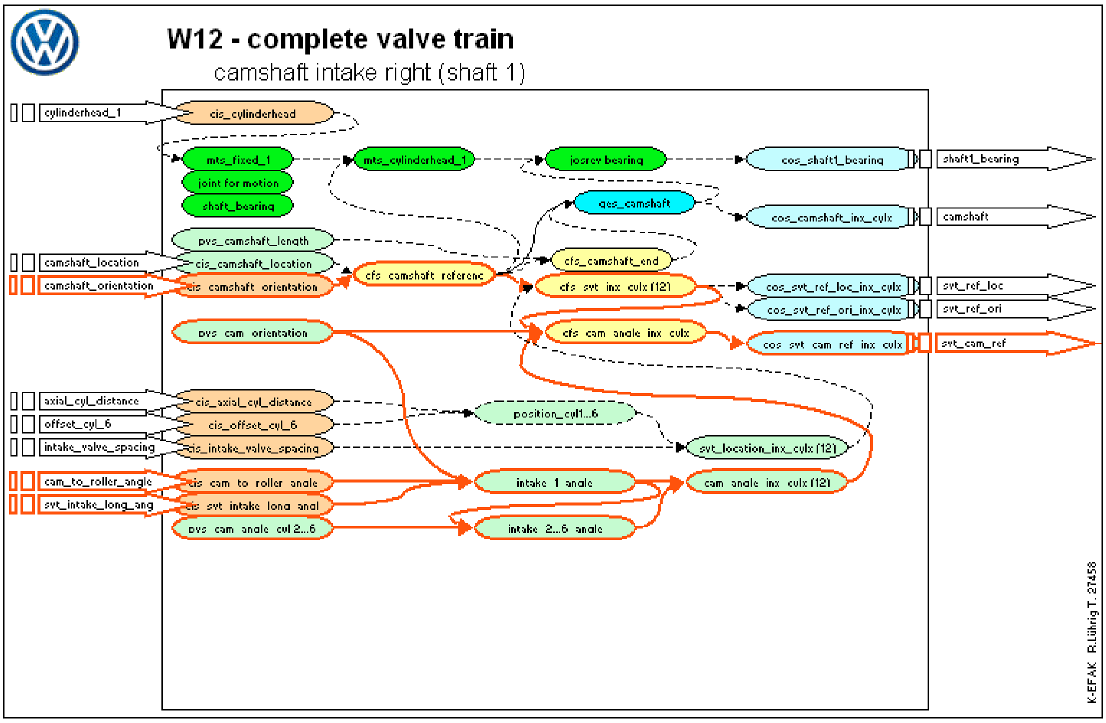

Fig. 1 Documentation for the intake camshaft of the right-hand cylinder head

Fig. 2 shows the complete model of the W12 valve train. It indicates the engine/gearbox mountings as well as the parts representing the mass of the engine and the gearbox.

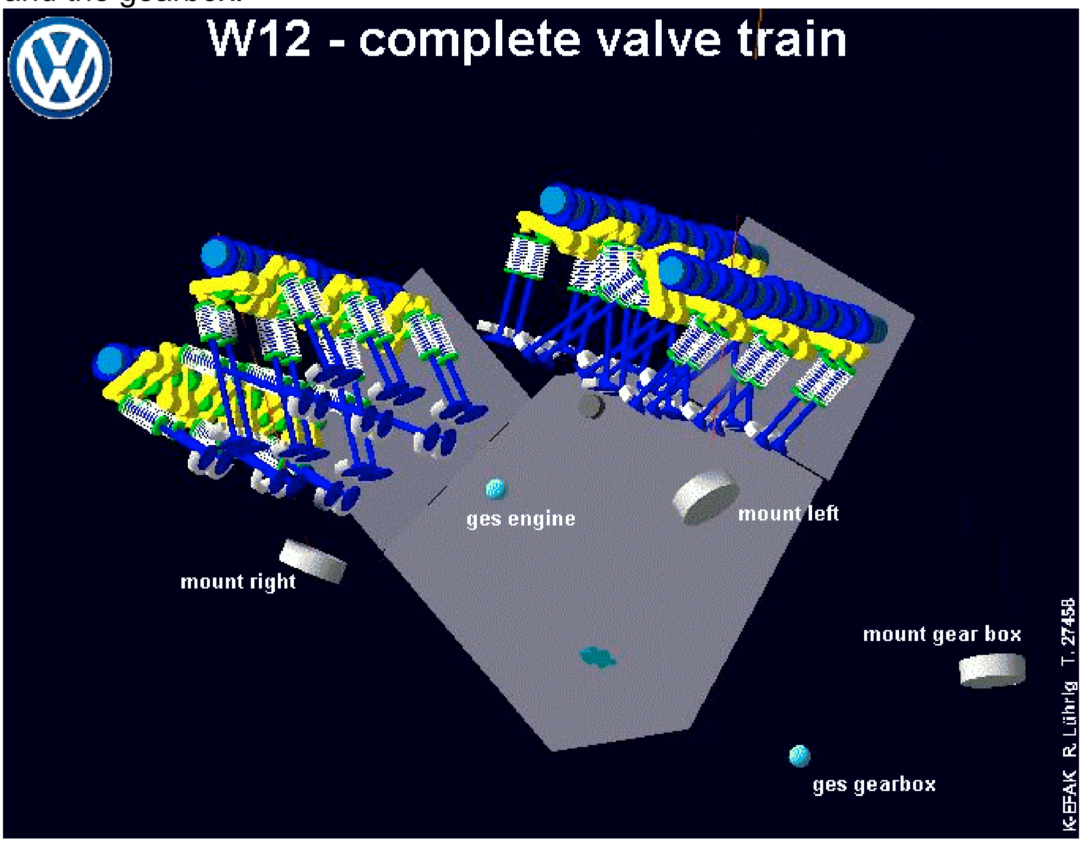

Fig. 2 Model of the complete valve train

4. Calculation and evaluation

The complete valve train model is intended to be used for calculating the vibration amplitudes at the engine/gearbox mountings at to assign orders of vibration.
To do this, a calculation was performed over several rotations of the crankshaft and then the time curves of the vibrational displacements on the engine/gearbox mountings were transformed in the frequency range using FFT.
Vibration amplitudes of rotating systems are normally represented by the order rather than the frequency. This means with reference to the multiple of the rotation frequency of a reference speed. In combustion engines, the crankshaft speed is normally used as the reference speed.
This means vibration such as is caused by an imbalance on the camshaft appears as the 0.5th engine order in the order spectrum. This is because the camshaft turns at half the speed of the crankshaft.

The advantage of considering vibrational displacements instead of forces or accelerations is that vibrational displacement amplitudes excited by mass are independent of speed as long as the frequency is not in a resonance range. This means it is sufficient to make the calculation with a single speed.
The amplitudes occurring can be considered to be the fundamental excitation of the system and they offer a very good basis for comparing the calculations of variants.

5. Results

A calculation was performed over 20 crankshaft rotations at an engine speed of 2000 rpm.
The build-up process of the engine/gearbox unit in its bearings was complete after 2 – 3 rotations.
The result of the calculation is then shown in the time curves for the movements of the unit in its mounting points in three spatial directions. Fig. 3 shows, by way of example, the movement of the gearbox bearing point in the Y-axis direction (transverse) over 10 rotations of the crankshaft.

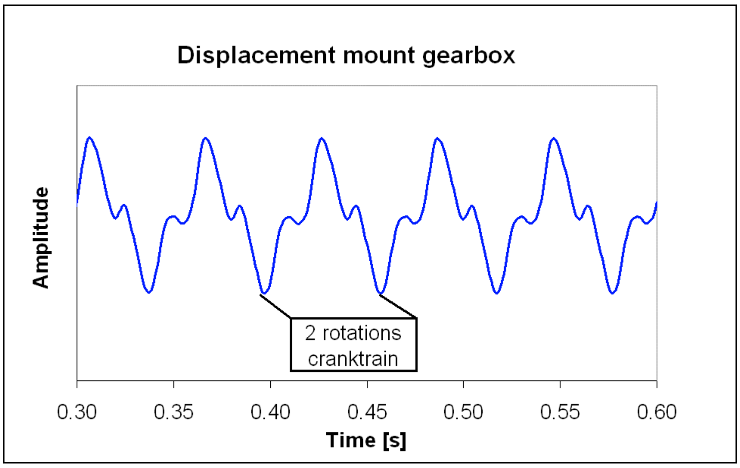

Fig. 3

Fig. 4 shows the result of the Fourier transformation (FFT) of this time signal.
As well as the 0.5th engine order, which can be avoided by balancing the camshafts, the other typical engine orders are the 1.5th and, at a lower amplitude, the 3rd. They result from the oscillating masses accelerated by the valve lift function.

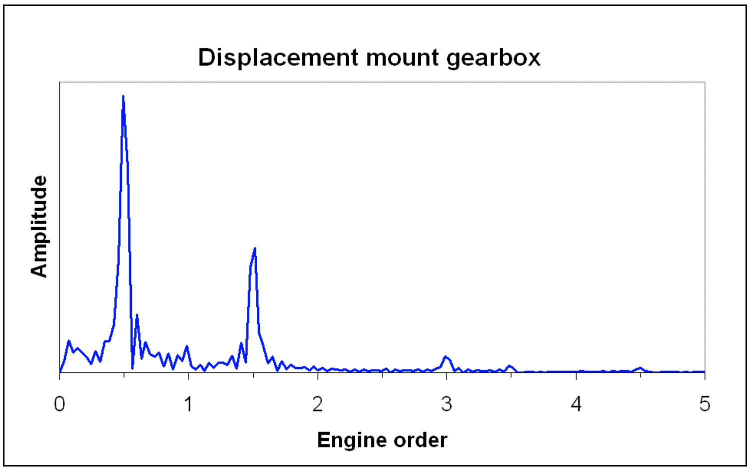

Fig. 4

Fig. 5 finally shows an overview of the vibrational displacements on the 3 engine/gearbox unit mountings.
The unit is rotating about its longitudinal axis in the 0.5th order. As a result, the amplitudes are greatest in the Z-axis direction at the engine mountings projecting sideways.
The 1.5th order is a rotational movement of the unit about its vertical and transverse axes. This means the end of the gearbox tries to move in a circle. Due to its distance from the pivot point, the greatest

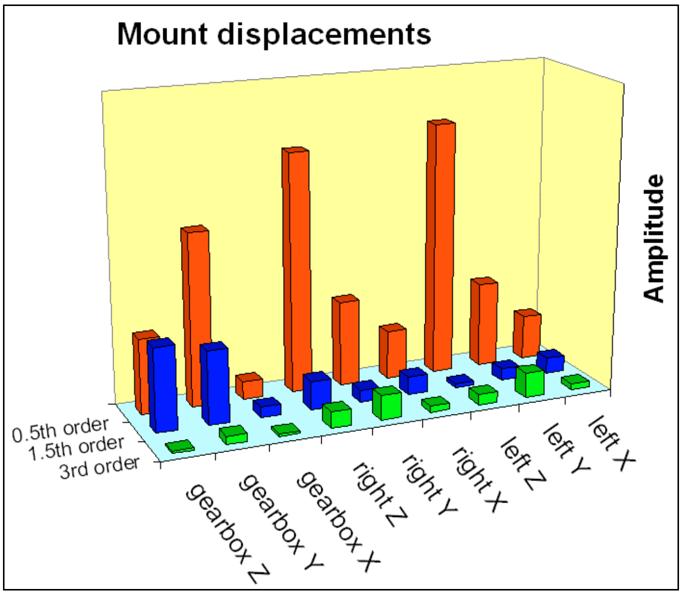

Fig. 5

amplitudes are also located there.
The 1.5th order is acoustically relevant for two reasons.
The frequency of this order is 175 Hz at 7000 rpm. As a result, this may mean that the first natural flexural frequency could be reached in the case of very long engine/gearbox units. Furthermore, the cardan shaft connected to the end of the gearbox is excited, in particular with all-wheel drive vehicles.

6. Concluding remarks

The calculation model presented here for the complete valve train of a W12 makes it possible to observe in isolation the engine/gearbox unit vibrations which are excited by the valve train. The effects on the vibration of the entire engine/gearbox unit can be investigated and parameter studies can be performed.
These results are enormously complicated to achieve by measurement, since excitation factors from charge changing processes and combustion also occur in the engine orders in question.