Paper 211

©Civil-Comp Press, 2012
Proceedings of the Eleventh International Conference on Computational Structures Technology, B.H.V. Topping, (Editor), Civil-Comp Press, Stirlingshire, Scotland

Transient Fluid-Structure Interaction Analysis of Exhaust Gas Recirculation Cooler Pipes

T. Grätsch
Hamburg University of Applied Sciences, Germany

Abstract

The present paper shows how vibrations of EGR cooler tubes can be analyzed using transient fluid-structure interaction (FSI) analyses. The basis of the analysis is to use real engine test data which describe the vibrations of the cooler inlet and outlet arising from natural engine vibrations. These excitations are applied in the time domain to a three-dimensional FSI model of an EGR cooler. The response of the cooler tubes is also obtained in the time domain and can be transformed into the frequency domain using the Fourier transform. It is also shown how the error in the numerical analysis can be controlled using the goal-oriented concept of error estimation.

Keywords: EGR cooler, fluid-structure interaction, error estimation, ADINA

1 Introduction

Exhaust gas recirculation (EGR) is a key technology in the development of efficient and low emission combustion engines. The main idea of this technology is that a portion of the exhaust gas is recirculated to the engine cylinders in order to reduce the total amount of nitrogen oxides (NOx) in the exhaust gas.

An important part of the EGR circuit is the so-called EGR cooler which cools the hot exhaust gas prior to being mixed with the fresh intake air and to being reintroduced into the cylinders, see Figure 1. By cooling the gas the combustion temperature in the engine cylinder is reduced and hence NOx emissions are reduced, because NOx results from high temperatures in the combustion process.

Manufacturers of EGR coolers use various cooler types, where tube and shell are the most common types used currently. The name of the cooler type refers to the geometrical form of the part of the cooler in which the exhaust gas is transported.

This paper focuses on a tube-type cooler, which consists of a cooler housing filled with coolant and where long gas tubes are embedded in the coolant transporting the gas from the inlet to the outlet of the cooler.

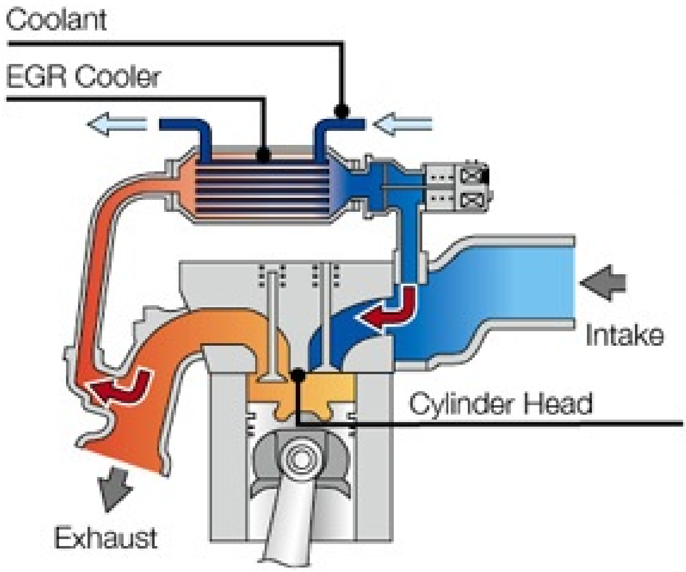

Figure 1: EGR cooling system.

One major issue in the development of tube-type EGR coolers is that due to natural engine vibrations single tubes in the cooler are forced to vibrate. Vibrations of tubes can be as severe that large amplitudes of vibrations cause tube clashes against the cooler housing which can lead to break of tubes and hence to a failure of the whole engine.

State-of-the-art analysis of tube vibrations is restricted to either classical modal analysis of the tube or to frequency domain analysis where ground excitations are applied to the tube supports which are obtained from engine measurements. The drawback of this approach is that the influence of the liquid in the cooler is completely neglected so that the amplitudes of vibrations are heavily overestimated.

One possibility to improve the analysis results by considering the mass influence of the coolant is to use the classical added mass concept, see Fritz [1]. However, it will be shown that this approach leads to inaccurate results for this kind of problem and there is only a slight improvement in the results obtained with frequency domain analysis. Hence a more sophisticated analysis procedure seems to be appropriate.

The new approach in this paper is to analyze EGR cooler tube vibrations using transient fluid-structure interaction (FSI) analyses. In this approach, measured engine vibrations from frequency domain are transformed into the time domain using the inverse Fourier transform. This data will be applied to the tube supports in the domain where the analysis model consists of a full three-dimensional FSI-model including the tubes and the cooler. The analysis in the time domain is done using the industrial finite element software ADINA, see Bathe and Zhang [2]. The response of the cooler tube is obtained in the time domain and can be transformed back into the frequency domain using the Fourier transform. Numerical results of three-dimensional fluid-structure interaction analyses of EGR cooler tubes are presented at different engine configurations and engine speeds. The maximum tube vibrations are observed to be in good agreement with engine measurements. In addition, the results are also compared with the results obtained from the classical added mass concept. It will be shown that this approach leads to inaccurate results for this kind of problem.

Further this paper presents results regarding error control for the error in the numerical analysis which is here based on the goal-oriented concept of error estimation as presented by Grätsch and Bathe in [3] for steady-state fluid-structure interactions. Using the goal-oriented concept, the error in the numerical analysis can be generally controlled for arbitrary quantities of interest of both the structural and the fluid part of the EGR cooler system. Applying these techniques to the maximum amplitude of vibrations of the cooler tubes, the error in this quantity can be efficiently reduced using different types of fluid elements of standard finite element software, see Bathe and Zhang [4].

2 Structural dynamic analysis of a single tube

In order to validate and to assess the results of FSI analysis to be presented in Section 3, results of structural dynamic analysis in this Section are obtained using a structural model of the tube only. Further, it is shown in detail how the test data are transformed from the frequency to the time domain and, also, results based on the classical added mass concept are discussed.

2.1 Modal analysis

For a better understanding of the dynamical behaviour of a single tube a classical modal analysis is performed. The structural FE model consists of a slender beam-like single tube which is fixed at both ends. The length of the tube is 600 mm, the tube cross-section is a rectangular box of 14.0 x 5.0 mm and wall thickness of 0.4 mm. The tube is made of steel with Young's modulus E = 194000 N/mm^2, Poisson's ratio ν = 0.3, and density ρ = 7.98e-06 kg/mm^3. In Figure 2 the first eigenmode is shown at a natural frequency of 91.4 Hz.

2.2 Frequency domain analysis

In the frequency domain analysis of a single tube a modal damping ratio of ξ = 2% is assumed. A ground excitation at 91.4 Hz with some amplitude from engine measurements at that frequency is applied to both ends of the tube in the direction of main vibration. In Figure 3 the maximum response at tube center is shown for a frequency range of 0 to 600 Hz. The maximum amplitudes are 1.98 mm at 91.4 Hz and 0.77 mm at 490.2 Hz which refers to the second natural frequency.

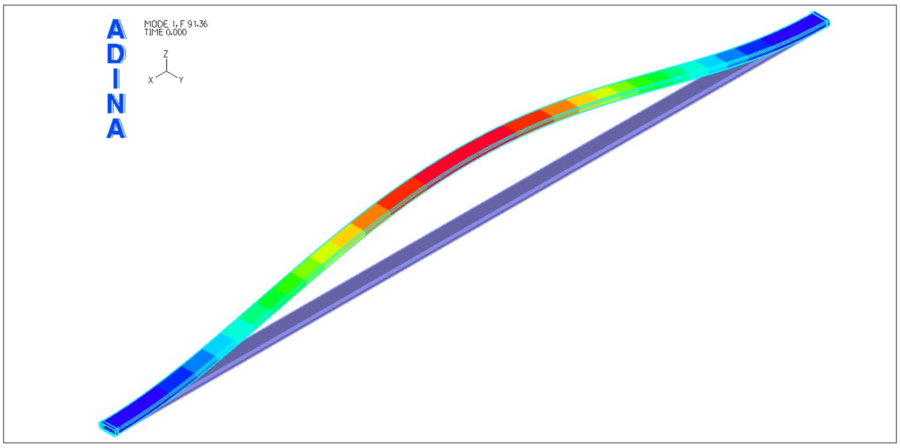

Figure 2: First eigenmode at 91.4 Hz of single tube.

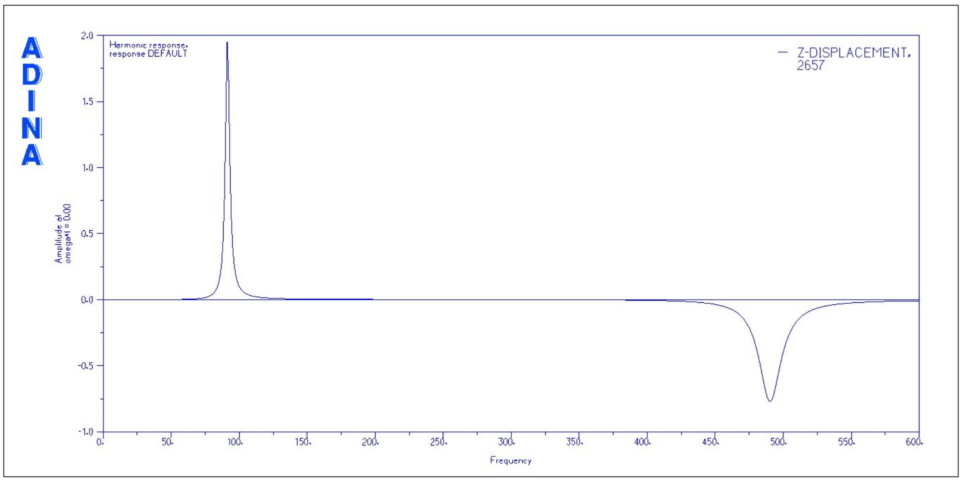

Figure 3: Maximum response at tube center of frequency domain analysis for frequency range of 0 to 600 Hz.

2.3 Time domain analysis

Since frequency domain analysis is always a linear analysis, the FSI analysis cannot be applied in the frequency domain, since FSI analysis is usually nonlinear due to the coupling of the fluid and the structure. Hence, we must find an alternative procedure, where time domain analysis is the method of choice here. For better understanding of time domain analysis, in the following the 'dry analysis' of the tube is conducted in the time domain.

2.3.1 Preparing the test data

From engine measurements a full frequency spectrum of amplitude characteristic $A(\omega)$ and phase response $\phi(\omega)$ were provided in the frequency range 0 to 500 Hz, where the data refers to the measured displacement amplitudes at both tube ends. Using

$$F(\omega) = \mathrm{Re}\,(A(\omega)\cdot e^{i\phi(\omega)}) \tag{1}$$

the frequency response $F(\omega)$ is calculated at every frequency point which is shown in Figure 4.

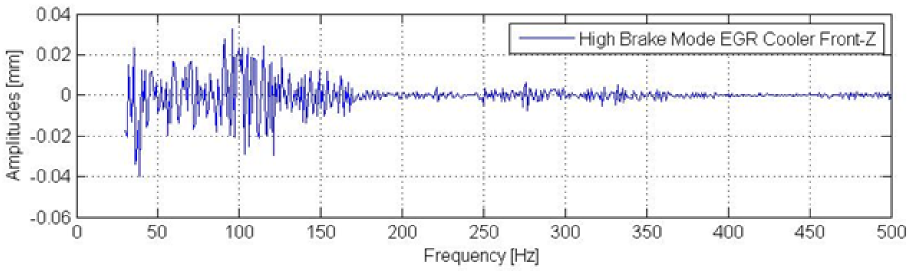

Figure 4: Displacement amplitudes at tube ends from engine test data in frequency domain.

The frequency response $F(\omega)$ in Equation (1) is then being transformed into the time domain using the Fourier transform which yields displacement amplitudes $u(t)$ in the time domain, see Figure 5.

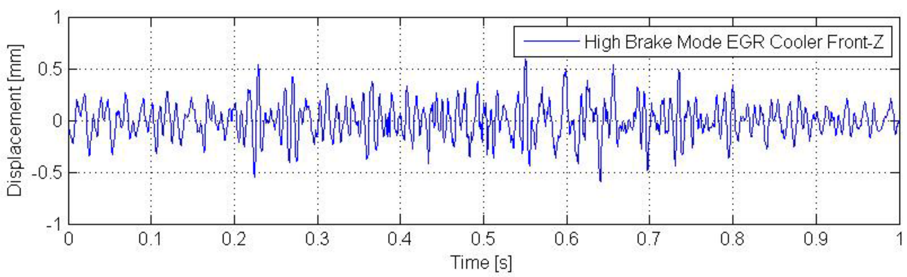

Figure 5: Displacement amplitudes at tube ends from engine test data in time domain.

2.3.2 Results of analysis

The ground excitations in the time domain shown in Figure 5 are applied to the structural dynamic model of a single tube. The analysis is based on the Newmark method with 2000 time steps and time step size of $\Delta t = 0.0005$ which spans a total analysis time of 1s. For the damping matrix **C** Raleigh damping is assumed

$$\mathbf{C} = \alpha\,\mathbf{M} + \beta\,\mathbf{K} \tag{2}$$

where **M** is the mass matrix and **K** denotes the stiffness matrix. The two Rayleigh parameters α and β are calculated from

$$\xi_i = \alpha / 2\,\omega_i + \beta\,\omega_i / 2 \tag{3}$$

with modal damping $\xi_1 = \xi_2 = 0.02$ for the first two frequencies and ω_i is calculated from the first two frequencies given in Section 2.2. From Equation (3) the two Rayleigh parameters $\alpha = 19.36132007$ and $\beta = 0.000010946$ are obtained. The analysis results are shown in Figure 6, where the maximum displacement amplitude yields 4.39 mm at time t = 0.81 s.

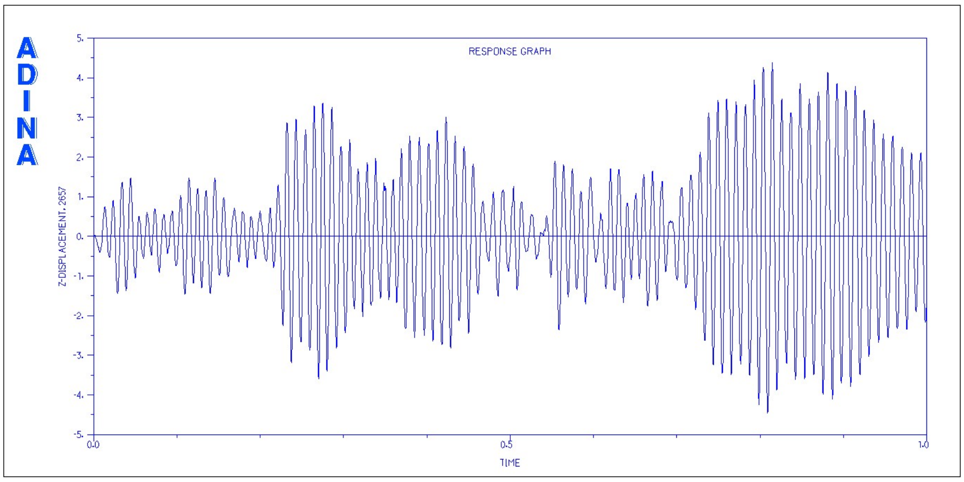

Figure 6: Maximum response at tube center in the time domain.

Applying the Fourier transformation to the time response shown in Figure 6, the response in the frequency domain is obtained with two peaks at the two natural frequencies f_1= 91.4 Hz and f_2 = 490.2 Hz, of course.

2.4 Time domain analysis using added mass concept

In order to account for the inertia effects due to the coolant in the EGR cooler, the analysis shown in Section 2.3.2 is repeated using the added mass concept [1]. In fluid mechanics, added mass or virtual mass is the inertia added to a system because an accelerating or decelerating body must move some volume of surrounding fluid as it moves through it. For simplicity this can be modeled as some volume of fluid moving with the object, calculated by fluid density times the volume of the body.

In the literature there are several corrections factors available to account for the geometrical form of the cross-section of the considered beam-like structure. For a perfect circular cross-section the factor is 1.0, where it is slightly different for rectangular cross-section. Since the tube cross-section is slightly rounded at the four corners there is no such value available, hence in the following a value of 1.0 is used.

The analysis is done with same parameters as in Section 2.3.2 where the added fluid mass is based on coolant density of 1.0e-06 kg/mm^3. The analysis results are shown in Figure 7, where the maximum displacement amplitude yields 3.53 mm at time t = 0.55 s.

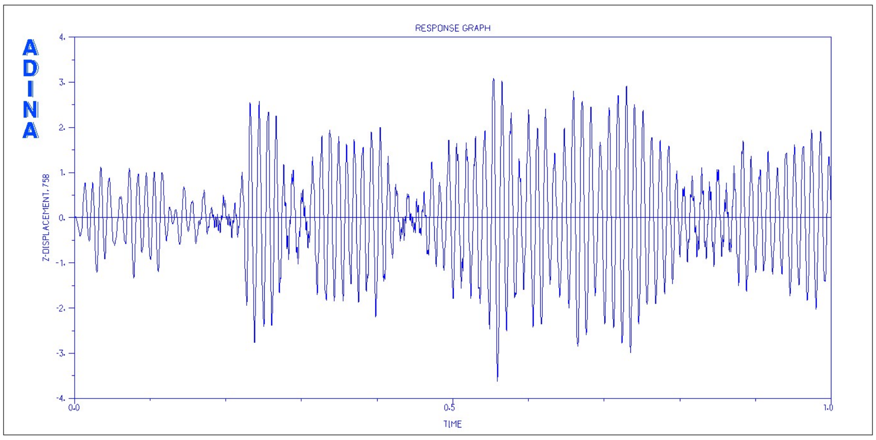

Figure 7: Maximum response at tube center in the time domain using the added mass concept.

2.5 Comparison of results

The structural dynamic analysis yields 3 different results depending on the analysis assumptions. The pure frequency domain analysis with ground excitation at the first natural frequency yields a maximum amplitude of 1.98 mm. However, this value seems not to be a realistic number for design purpose of the tubes, because the excitation due to frequencies in the neighbourhood of the natural frequency is not considered in this approach.

Applying the ground excitation in the time domain to the cooler yields a maximum amplitude of 4.39 mm, which shows the influence of considering excitations at all frequency points. This value can be reduced to 3.53 mm if the inertia of the surrounding fluid is considered using the classical added mass concept. Unfortunately both values do not match experimental observations, where tube clashing against the housing was observed when the distance between outer tube and housing was less than 1.0 mm. The very possible reason is that all the approaches in this Section neglect the influence of the interaction between the fluid and the tubes. Hence, in the next Section a full three-dimensional fluid-structure interaction analysis is performed.

3 Fluid-structure-interaction analysis of EGR cooler

3.1 FSI analysis model

The structural parameters of the FSI analysis model are described in Section 2.1. For the coolant a density of 1.0e-06 kg/mm^3 and a dynamic viscosity of 2.822e-06 Ns/mm^2 is assumed. The maximum coolant flow in the EGR cooler is 1.4 l/s and the absolute coolant pressure is 2.1 bar.

The fluid part of the FSI model consists of 54000 fluid elements with 8 nodes per element, see Figure 8, and incompressible flow is assumed. The structural part consists of 5200 hexaeder elements with 8 nodes per element. The total number of degrees of freedom of the FSI model is 230000 and iterative coupling is employed at the interface. The FSI analysis is based on 2000 time steps and time step size of $\Delta t = 0.0005$ with spans a total analysis time of 1s.

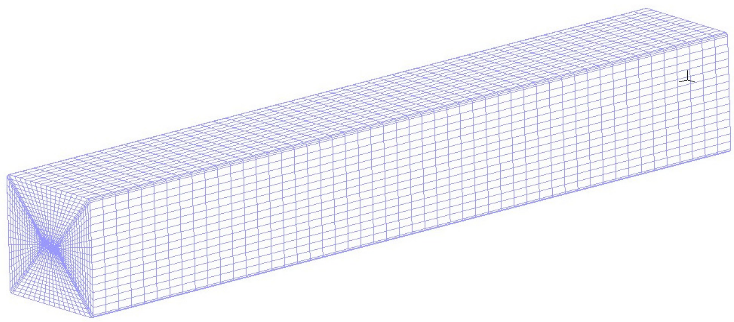

Figure 8: Fluid mesh of FSI model.

The boundary conditions is wall (no slip) at the fluid boundary and equal displacements and tractions at FSI boundary. The ground excitations shown in Figure 5 are applied to the two tube endings. At the coolant inlet of the EGR cooler prescribed velocity is applied and at the outlet prescribed pressure is applied.

3.2 Results of FSI analysis

The FSI analysis results are shown in Figure 9, where the maximum displacement amplitude yields 0.95 mm at time t = 0.60 s. Figure 10 shows the then present velocities and the pressure field. Here it is interesting to note that the pressure is almost not affected by the vibrations of the tube (value of 0.21 in the scale refer to absolute pressure of 2.1 bar). The most important observation is that the maximum amplitude of tube displacement is smaller of a factor up to 4 compared to the amplitudes discussed in Section 4, which now perfectly matches experimental observations.

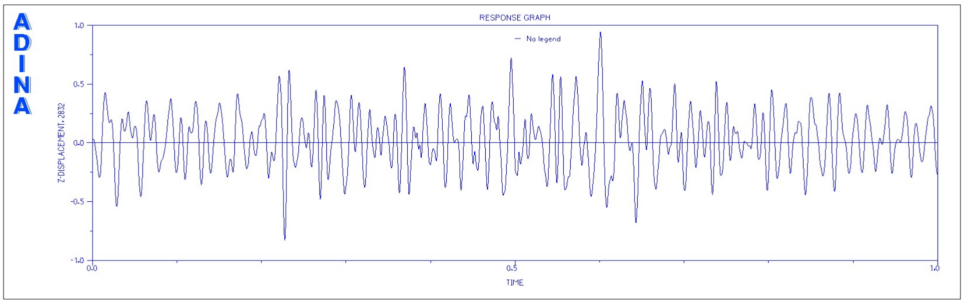

Figure 9: Maximum response at tube center from FSI analysis.

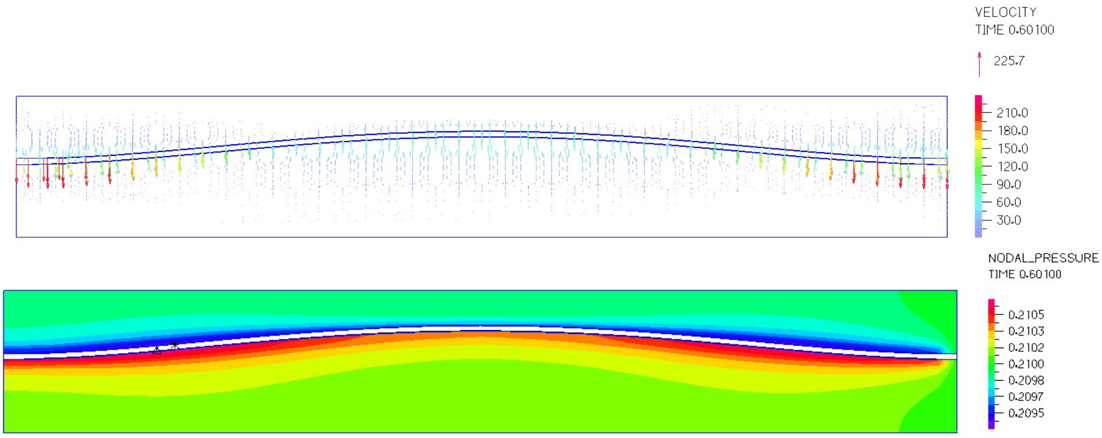

Figure 10: Velocity and pressure at time t = 0.60 s from FSI analysis (scale factor for deformation is 25).

Although results in frequency domain cannot be obtained directly using FSI analysis it is possible to calculate the Fourier transform of the maximum displacement response as given in Figure 9 in order to obtain the corresponding response in the frequency domain. The result of this transformation is shown in Figure 11.

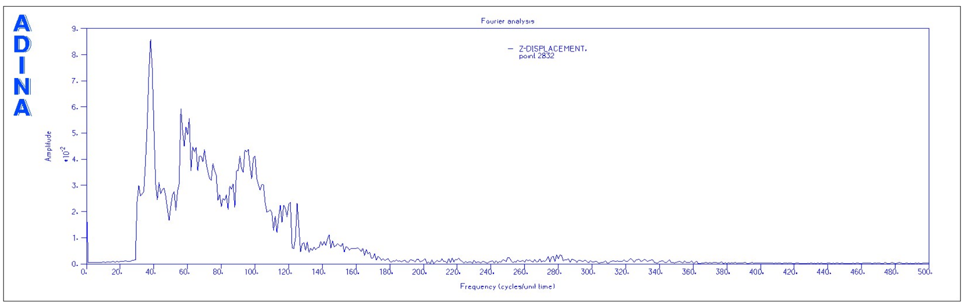

Figure 11: Maximum response at tube center in frequency domain obtained from Fourier transformation of FSI time response given in Figure 9.

In Figure 11 a number of peaks at different frequencies are visible which are due to coupling of the fluid and the structural part and which are not seen in the frequency response using standard (linear) structural dynamic analysis.

It is also interesting to investigate the influence of gas pressure inside the tubes. While the analysis results in Figure 10 do not include the gas pressure in the next step a gas pressure of 5 bar is assumed. Figure 12 shows the stresses in the tube for both load configurations (gas pressure = 0 bar / 5 bar). The deformation due to the gas pressure is clearly visible where the stresses are larger when considering the gas pressure.

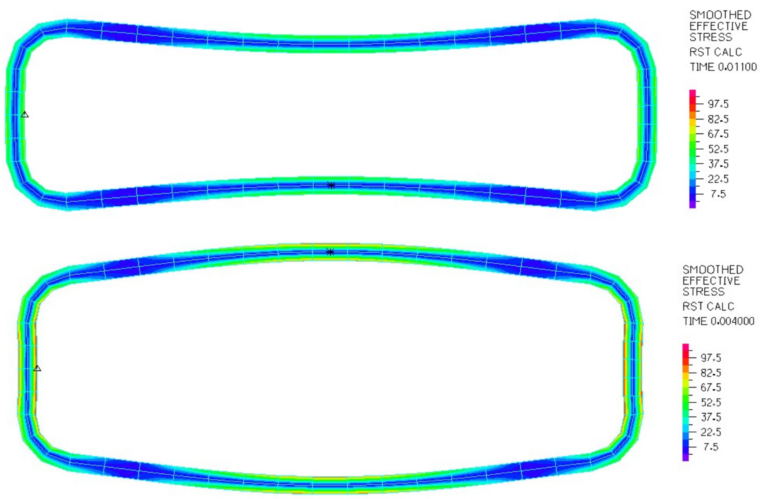

Figure 12: Von Mises stresses and tube deformations for tube gas pressure of 0 bar (top) and 5 bar (below).

In summary, the novelty of the presented approach is to employ a four step procedure for three-dimensional FSI analysis of EGR coolers. First based on measurements in the frequency domain a time signal is obtained using the inverse Fourier transform for which a fully three-dimensional coupled FSI analysis is performed. The response of the structure in time domain is again transformed into the frequency domain using the Fourier transform which yields a frequency spectrum of the response based on three-dimensional FSI analysis, see Figure 13.

4 Goal-oriented error assessment

Goal-oriented error estimation is used for controlling and estimating the error with respect to an arbitrary quantity of the solution space, which can be the displacement at a point or the stress in a small region of the domain. While the procedure has been successfully employed for linear and other single field problems, see e.g. [5] – [10], the application to fluid-structure interaction problems is a rather new line of research, see [3], [11]-[14].

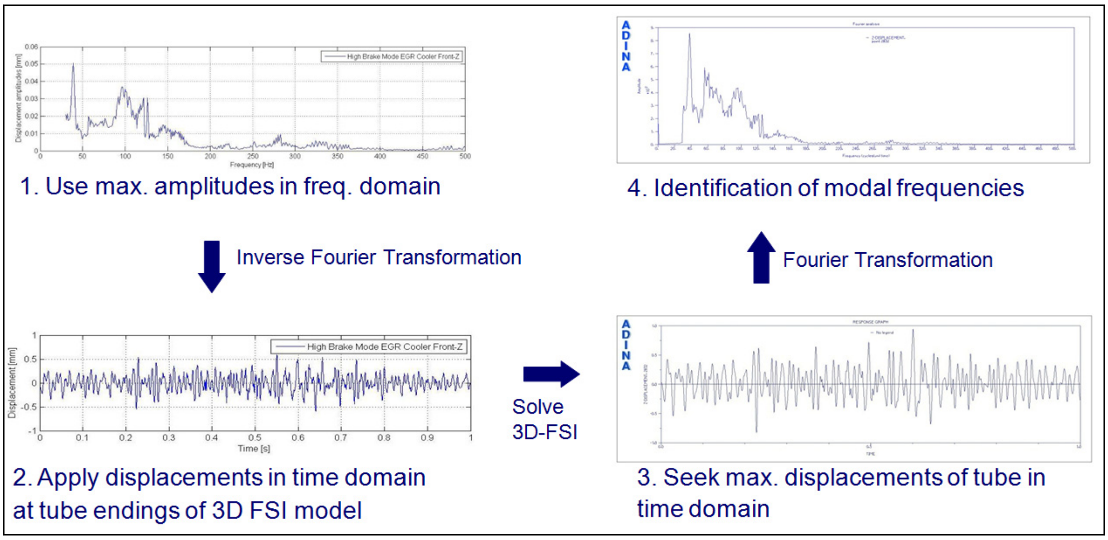

Figure 13: Four step approach as procedure for three-dimensional FSI analysis.

In goal-oriented error estimation a so-called dual problem needs to be solved, where the solution of this auxiliary problem filters out the necessary information for an accurate and effective estimate for the error in the quantity of interest. Following the approach of Grätsch and Bathe in [3], sensitivity based error estimation for fluid-structure interactions can be employed by approximating the linearized solution of the dual problem with help of finite difference approximations, for details see [3]. Applying this procedure at every time step of the solution leads to an error representation for arbitrary quantities of transient fluid-structure analysis, see [15].

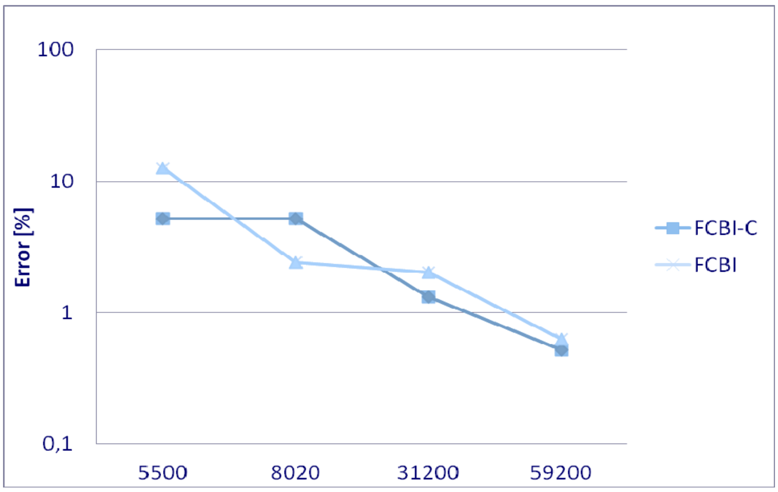

Figure 14: Error in goal quantity (maximum displacement at tube center) using the goal-oriented concept of error estimation based on the approach in [3] and [15].

Defining the quantity of interest as the displacement amplitude at the tube center and using the approach presented in [3] and [15] for different kinds of fluid elements the results shown in Figure 14 are obtained. From Figure 14 it can be seen that the error in the goal quantity decreases reasonably with increasing number of elements in the fluid-structure interaction model. The error estimation is robust with respect to the fluid elements used that are the FCBI element and the FCBI-C element presented in [4].

5 Conclusions

In the present paper it is shown how vibrations of EGR cooler tubes can be analyzed using transient fluid-structure interaction (FSI) analyses. In contrast to structural dynamic analysis the results are much more accurate because the interaction of the structural part and the fluid part is essential for the vibration behaviour of the tubes. It is also shown that using the classical added mass concept to account for the inertia of the fluid improves the structural dynamic results only slightly, whereas the results obtained with the fully coupled FSI model are much more accurate compared to experimental validation.

The novelty of the presented approach – from a technical point of view – is to use a four step procedure for three-dimensional FSI analysis of EGR coolers where the (inverse) Fourier transform is used twice to switch the data from the frequency domain to the time domain and vice versa as needed in the analysis procedure. Finally a frequency spectrum of the maximum tube response can be obtained based on fully coupled three-dimensional FSI analysis.

A practical approach for controlling the error in transient fluid-structure interaction analyses has been employed which is based on the concept of goal-oriented error estimation. The approach is practical and leads to accurate and robust results in the test problem considered.

Next steps should focus on the three-dimensional FSI analysis of a package of gas tubes where pulsating (time-dependent) gas pressure at inner tubes should be applied. Also, the analysis should cover different cooler types and should be conducted considering different engine conditions.

References

[1] R. J. Fritz, "The Effect of Liquids on Dynamic Motions of Immersed Solids", ASME Journal of Engineering for Industry, 94, 167-173, 1977.

[2] K.-J. Bathe, H. Zhang, "Finite element developments for general fluid flows with structural interactions", International Journal for Numerical Methods in Engineering, 60, 213–232, 2004.

[3] T. Grätsch, K.-J. Bathe, "Goal-oriented error estimation in the analysis of fluid flows with structural interactions", Computer Methods in Applied Mechanics and Engineering, 195(41-43), 5673-5684, 2006.
[4] K.-J. Bathe, H. Zhang, "A Flow-Condition-Based Interpolation Finite Element Procedure for Incompressible Fluid Flows", Computers & Structures, 80, 1267-1277, 2002.
[5] K. Eriksson, D. Estep, P. Hansbo, C. Johnson, "Introduction to adaptive methods for differential equations", Acta Numerica, 105–58, 1995.
[6] R. Rannacher, F.-T. Suttmeier, „A feed-back approach to error control in finite element methods: application to linear elasticity", Computational Mechanics, 19, 434–46, 1997.
[7] M. Paraschivoiu, J. Peraire, A. Patera, "A posteriori finite element bounds for linear-functional outputs of elliptic partial differential equations", Comput Methods Appl Mech Engng, 150, 289–312, 1997.
[8] F. Cirak, E. Ramm, "A posteriori error estimation and adaptivity for linear elasticity using the reciprocal theorem", Comput Methods Appl Mech Engng 156, 351–62, 1998.
[9] W. Bangerth, R. Rannacher, "Adaptive finite element methods for differential equations". Basel: Birkhäuser; 2003.
[10] T. Grätsch, K.-J. Bathe, "A posteriori error estimation techniques in practical finite element analysis", Computers & Structures, 83, 235–265, 2005.
[11] T. Dunne, "An Eulerian approach to fluid-structure interaction and goal-oriented mesh adaptation", International Journal for Numerical Methods in Fluids, 51 (9-10), 1017-1039, 2006.
[12] K.G. van der Zee, E.H. van Brummelen, and R. de Borst, „Goal-oriented error estimation for Stokes flow interacting with a flexible channel", International Journal for Numerical Methods in Fluids, 56(8), 1551-1557, 2008.
[13] F. Bengzon and M.G. Larson. "Adaptive finite element approximation of multiphysics problems: A fluid-structure interaction model problem", International Journal for Numerical Methods in Engineering, 84(12), 1451-1461, 2010.
[14] K. Selim, "Adaptive finite element methods for fluid-structure interaction and incompressible flow", Faculty of Mathematics and Natural Sciences, University of Oslo, PhD thesis 2011.
[15] T. Grätsch, "A practical approach to error control in time-dependent fluid-structure interaction analysis", in "Proceedings of Sixth M.I.T. Conference on Computational Fluid and Solid Mechanics", K.-J. Bathe, (Editor), Cambridge MA, 2011.