SPEED-UP METHODS FOR THE MODELING OF TRANSIENT TEMPERATURES WITH REGARD TO THERMAL AND THERMOMECHANICAL FATIGUE

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ABSTRACT

The accurate prediction of the life cycle in turbomachinery design is one of the most challenging issues. Traditionally, life cycle calculations for radial turbine wheels of turbochargers focus on mechanical loads such as centrifugal and vibrational forces. Due to steadily increasing exhaust gas temperatures of automotive and commercial engines in the last years, thermo-mechanical fatigue in the turbine wheel is a major topic of current investigations.

In order to account for the thermally induced stresses in the turbine wheel and the turbine housing as a part of the standard design process, a fast method is required for predicting metal temperatures.

In the present paper, a fast method to calculate the transient temperatures in a radial turbine is presented. In this method the specific heat capacity of the solid state is reduced by a "speed up factor" in order to shorten the duration of a transient heating or cooling process. With the shortened processes, the computing times can be reduced significantly. After the calculations, the resulting times are transferred into realistic heating or cooling times by multiplying them with the speed up factor. The method is evaluated against experimental data and against the results of a numerical method known from literature. The method shows a good agreement with those data.

KEYWORDS

transient, conjugate heat transfer, thermo-mechanical fatigue, radial turbine

NOMENCLATURE

A	Area
Bi	Biot-Number
c_p	Specific heat capacity
Fo	Fourier-Number
D_W	Characteristic length
$\dot{m}$	Mass flow rate
SF	Speed up factor
T	Temperature
t	Time
V	Volume
y^+	Dimensionless wall distance
α	Heat transfer coefficient
ΔT	Temperature difference
λ	Thermal conductivity
ρ	Density
σ	Stress
τ	Time scale

Indices

0	Time point zero
$-\infty$	Time point before thermal shock
∞	Time point after thermal shock
av	Averaged
C,I	Compressor inlet
C,O	Compressor outlet
FL	Fluid
H	Housing
max	Maximum
MH	Measuring point turbine housing
MW	Measuring point turbine wheel
O	Outlet
OP	Operating Point
SL	Solid
t,I	Total inlet turbine

INTRODUCTION

The efficiency of turbocharged engines is highly dependent on the inlet temperature of the turbocharger turbines (Gabriel et al., 2006). In consequence, a trend towards higher turbine inlet temperatures can be observed in the last years. By 2006, gas temperatures of up to 1050°C were achieved in gasoline fired engines (Dornhöfer et al., 2006). Due to these high temperatures, thermal and thermo-mechanical fatigue drew more and more attention in the design process of turbochargers. According to Scharf et al. (2009) most of the damages in turbochargers occur in the turbine housing and in the turbine wheel. In the turbine wheel, high temperature gradients in combination with local mechanical strain hindrances cause thermal stresses. These thermal stresses superimpose the centrifugal stresses. The thermal stresses are especially high in transient operation. Thus, a transient investigation of the turbocharger is obligatory. To this end a fast method is required which enables a calculation of the transient temperatures within the time limits of a standard design process of a turbocharger and which ensures sufficient accuracy.

First transient investigations regarding thermal fatigue in the turbine of a turbocharger were published in the last few years. In 2005 Heuer et al. (2005) performed transient numerical conjugate heat transfer simulations of a turbine housing and validated the results by means of steady state measurements. The authors proved that the highest thermal stresses in the turbine housing occur in transient operation.

Oberste-Brandenburg et al. (2012) presented a simplified approach for the design of a turbocharger housing regarding thermal fatigue and compared it with numerical simulations. In their approach, the time consuming CFD simulations (Computational Fluid Dynamics) were replaced by a simple 1D model and the geometry for the FEA model (Finite Element Analysis) was reduced to the main components. This approach delivers a good agreement with CFD calculations by comparing averaged heat transfer coefficients. However, the predicted plastifications differ significantly between the simplified and the detailed model.

An alternative approach was presented by Ahdad et al. [2010]. The authors used a semi-empirical approach to calculate the thermal stresses in the turbine housing. The model was developed and validated on basis of FEA calculations. Due to the empirical correlations used in this model it has to be calibrated for each kind of turbine housing geometry.

Unlike the turbine housing the turbine wheel has only rarely been subject of thermomechanical research. In the work of Ibaraki et al. (2008) two turbine wheels were compared which differ in the respective materials. Long term tests and numerical calculations were performed at inlet temperatures of about 1050°C. It was shown that the durability of a MarM material turbine wheel is higher than of a common Inconel turbine wheel.

In 2006 Heuer et al. (2006) extended their research to the turbine wheel of a turbocharger. The authors identified critical zones regarding thermo-mechanical stresses. Furthermore, they showed that the centrifugal loads are dominant in the turbine wheel. However, the thermal loads are not negligible.

Diefenthal et al. (2015) used the method of Heuer et al. (2006) and validated it by transient test data. They described that the method shows inaccuracies in the transient calculation of the turbine wheel temperatures and extended it to improve the accuracy. Based on this result they showed that three dimensional flow structures and secondary flows are the dominant heat transfer effects causing thermal stresses in the turbine components. They concluded that one dimensional approaches as used in Oberste-Brandenburg et al. (2012) are not sufficient to describe the transient temperature fields with regard to thermal or thermo-mechanical fatigue. Thus, three dimensional or empirical methods have to be used.

In the present work, a fast three dimensional method is presented which enables the modelling of transient fluid/solid heat transfer considering three dimensional flow structures. This method is validated against test data and against the results of the extended numerical method described in Diefenthal et al. (2015).

TEST RIG

In the present work a commercial vehicle turbocharger with a scalloped turbine wheel with a diameter of about 90mm is investigated. For the validation tests it was assumed that the maximum thermal stresses in a real driving cycle occur when the turbine inlet temperature is changed abruptly. On the basis of this assumption a test rig was designed which is able to impose rapid changes in turbine inlet temperature and which allows to record material temperatures in the rotating turbine wheel (later referred to as "thermal shock test"). A detailed description of the test rig can be found in Tadesse et al. (2015). The principal behavior of the fluid and solid temperatures in a thermal shock test is shown in Figure 1.

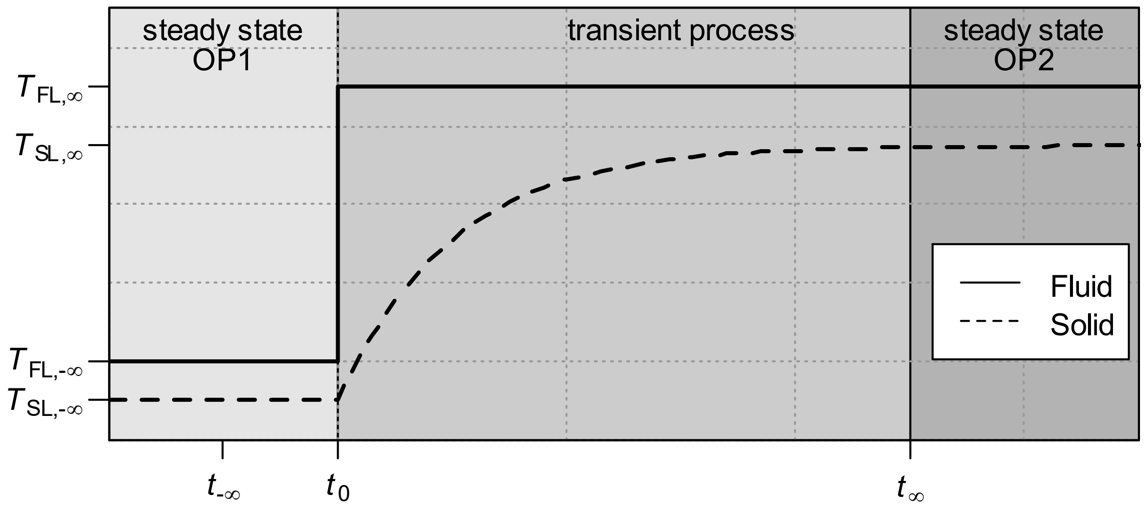

Figure 1: Behavior of fluid and solid temperatures in a heating thermal shock test

To measure the temperatures on the turbine wheel during the thermal shock tests, four K-Type thermocouples are mounted in the turbine wheel. The individual positions of the thermocouples are shown in Figure 2. Additional K-Type thermocouples are attached on the outer surface of the turbine housing to record the temperature distribution. A detailed description of the measurement setup is given in Tadesse et al. (2015).

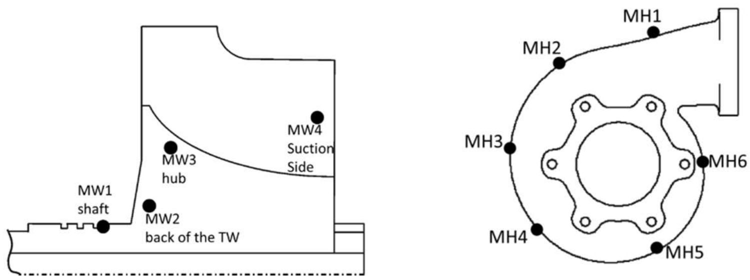

Figure 2: Position of the thermocouples at the turbine wheel and the turbine housing

CALCULATION METHOD FOR TRANSIENT FLUID/SOLID HEAT TRANSFER

Two calculation methods of transient fluid/solid heat transfer can be distinguished in literature. In the coupled method or the conjugate heat transfer method (CHT), a CFD calculation is iteratively coupled with a conductive FEA calculation. The continuity of the temperatures and the heat fluxes at the fluid/solid interfaces is ensured by solving the energy equation at the coupled boundaries. Examples of this method can be found in Bohn et al. (2003, 2005). The uncoupled method is based on the assumption of a linear relationship between the convective heat fluxes across the boundary layer and the driving temperature differences (constant heat transfer coefficients). Based on this assumption the heat transfer coefficients are determined by means of a CFD simulation and used as boundary conditions for a subsequent FEA calculation. Examples of this approach are the heat transfer calculations of rotor stator cavities published by Alizadeh et al. (2007) and Lewis et al. (2004). In comparison to the coupled method, the uncoupled method can save calculation time, since no iterative solution is necessary at the fluid/solid boundary. Owing to the high computational costs, the coupled method is mainly used for steady state calculations. A time accurate calculation on basis of the coupled method requires an extreme computational effort, since the timescales of the convective and the conductive heat transfer differ vastly (He and Oldfield, 2011). As a consequence simplifying assumptions have to be made to reach acceptable calculation times for a coupled method simulation of a transient heating or cooling process.

Frozen Flow method

In the present paper two numerical methods are used to calculate transient fluid/solid heat transfer. The first method is described and validated in Diefenthal et al. (2015). With the assumption of a partly constant pressure and velocity field of the fluid state, mass, momentum and turbulence equations are not solved and consequently larger time steps can be applied. This method is named Frozen Flow method (FF). The results of this method serve as reference for the validation of the newly developed method presented in this paper. The model for the Frozen Flow calculations comprises the turbine housing, the turbine wheel and the inlet and outlet pipes.

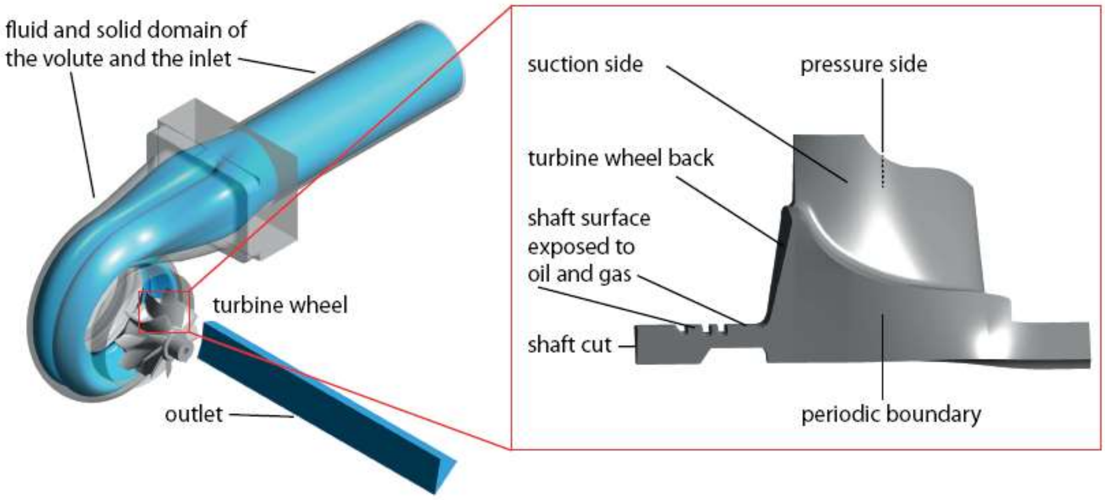

Figure 3: Numerical model of Turbine Housing (TH) and Turbine Wheel (TW)

As depicted in Figure 3 the turbine wheel is modeled as a single rotor passage by using peripheral averaging (mixing plane) at the rotor stator interfaces. The bearing housing and the compressor are not included in the model. At the boundary plane at the shaft, boundary conditions are defined to describe the heat fluxes to the lubricant oil and to the compressor side. These boundary conditions are estimated by using experimental data. The heat shield at the turbine wheel back is assumed to be adiabatic as it is expected that the convective and radiative heat fluxes to the bearing housing are small. The mesh of the whole turbine model consists of approximately 6.2 million nodes. 3.7 million nodes are located in the fluid state and 2.5 million nodes discretize the solid body. In the fluid boundary layer the dimensionless wall distance y^+ is lower than one. A mesh study has been conducted to ensure that the results are unaffected by the mesh. To model the turbulence in the thermal sublayer, the low Reynolds kω-SST turbulence model is applied. The numerical time step changes during the time. All calculations are conducted with ANSYS CFX 15.0. A more detailed description and a validation of the Frozen Flow method can be found in Diefenthal et al. (2015).

Equalized Timescales method

For the Equalized Timescales method (ET), the same model is used as in the FF calculation. The new ET method is a time saving approach for the calculation of the transient temperature fields with regard to thermo-mechanical fatigue. In this method the fluid and the solid state are calculated coupled with the whole transient process. However, the fluid calculation is not simplified like in the first method. Thus, a method characterized by high accuracy and small user effort is expected. A coupled calculation of fluid and solid state leads to extremely high calculation times due to the different timescales of fluid and solid. These different timescales cause long conductional heat transfer processes in the solid state that have to be calculated with small time steps to achieve numerical stability of the fluid calculation (He and Oldfield, 2011). In order to reduce the calculation times in this method, the timescales of fluid and solid are equalized by modifying the material properties of the solid state. Thus, this method is named Equalized Timescales method (ET).

To describe the approach of the ET method in more detail, a simple analytical approach is considered in the following assumptions. For a solid body with a homogenous temperature distribution, the timescale of a convectional heating or cooling process can be determined from a simple analytical equation:

$$\rho_{\mathrm{Sl}}\, c_{\mathrm{p,Sl}}\, V_{\mathrm{Sl}}\, \frac{\partial T_{\mathrm{Sl}}}{\partial t} = -A_{\mathrm{Sl}}\, \alpha\, (T_{\mathrm{Sl}} - T_{\mathrm{Fl}}) \qquad \textbf{Equation 1}$$

With the assumptions of constant material properties, a constant heat transfer coefficient and a constant fluid temperature over the time, the solution of Equation 1 leads to Equation 2.

$$T_{Sl} = \left(T_{\mathrm{Sl,0}} - T_{\mathrm{Fl}}\right) e^{-\frac{\alpha\, A_{\mathrm{Sl}}}{c_{\mathrm{p,Sl}}\, \rho_{\mathrm{Sl}}\, V_{\mathrm{Sl}}} t} + T_{\mathrm{Fl}} = \left(T_{\mathrm{Sl,0}} - T_{\mathrm{Fl}}\right) e^{-\frac{t}{\tau}} + T_{\mathrm{Fl}} \qquad \textbf{Equation 2}$$

The exponent of the Euler's Number represents the timescale τ of the heating or the cooling process. It depends on the heat transfer coefficient α, the density ρ_{Sl} and the specific heat capacity $c_{\mathrm{p,Sl}}$. In the ET method, the timescales of the fluid and solid are equalized by reducing the specific heat capacity of the solid state by the speed up factor *SF*. The modified specific heat capacity is named c_{p}* and is defined by the following equation.

$$c_{\mathrm{p,Sl}}{}^{*} = \frac{c_{\mathrm{p,Sl}}}{SF} \qquad \textbf{Equation 3}$$

From Equation 3 it can be derived that the reduction of the specific heat capacity results in a reduced heating or cooling time of the solid body. The modified time is named *t** in the present work and is connected with the real time by the speed up factor *SF* for this simplified case. The definition of the modified time is given in Equation 4.

$$t^{*} = \frac{t}{SF} \qquad \textbf{Equation 4}$$

Implementing this approach to a coupled fluid/solid heat transfer simulation results in significantly reduced calculation times as shorter conductional heating or cooling processes have to be calculated and the time step of the fluid calculation can remain constant. To determine the real heating or cooling time of a transient process from these simulations the resulting time *t** is multiplied by the speed up factor SF according to Equation 4.

The Fourier- and the Biot-Number are assessed in order to prove the suitability of the described approach for the determination of the thermal behavior of the solid state in a transient process. As described in Baehr and Stephan (2010), transient temperature fields of the same geometry and the same boundary conditions are similar if the Biot-Number and the Fourier-Number are equal. Thus, the Biot- and the Fourier-Number must remain constant when the specific heat capacity is modified. In Equation 5 the modified specific heat capacity c_{p}* and the modified time *t** are introduced in the definition of the Fourier-Number.

$$Fo = \frac{\lambda_{\mathrm{Sl}}\, t}{c_{\mathrm{p,Sl}}\, \rho_{\mathrm{Sl}}\, {D_{\mathrm{W}}}^{2}} = \frac{\lambda_{\mathrm{Sl}}\, t^{*}\, SF}{c_{\mathrm{p,Sl}}{}^{*}\, SF\, \rho_{\mathrm{Sl}}\, {D_{\mathrm{W}}}^{2}} = \frac{\lambda_{\mathrm{Sl}}\, t^{*}}{c_{\mathrm{p,Sl}}{}^{*}\, \rho_{\mathrm{Sl}}\, {D_{\mathrm{W}}}^{2}} = Fo^{*} \qquad \textbf{Equation 5}$$

As shown, the unmodified and the modified Fourier-Numbers are identical for the described simplified approach. Unlike the Fourier-Number, the Biot-Number is independent of the specific heat capacity and of the time. This can be seen in Equation 6. However, the Biot-Number depends on the heat transfer coefficient.

$$Bi = \frac{\alpha L}{\lambda_{\mathrm{Sl}}} = Bi^{*} \qquad \textbf{Equation 6}$$

With the assumption of a constant heat transfer coefficient, the Biot-Number remains constant respectively. To investigate the assumption of a constant heat transfer coefficient in more detail and to validate/verify the ET method, the method is applied to a turbocharger and validated against experimental data and again a common numerical method.

STRUCTURE MECHANICAL MODELS

In order to investigate the turbine housing and the turbine wheel with regard to the thermal and the thermo-mechanical stresses, simple structure mechanical models of turbine wheel and turbine housing are set up. These models do not aim to calculate realistic stresses quantitatively. Instead, they enable an indication of the critical zones and an assessment of the impact of the different temperature fields of the different aerothermal calculation methods on the total stresses. The models of turbine wheel and turbine housing are described in the following two sections.

Turbine wheel

Owing to the slow solid temperature changes in a heating or a cooling process, a steady state model is used for the structure mechanical simulations. In this model the same mesh is applied for the turbine wheel as in the thermal calculations. Referring to Heuer et al. (2006), Ibaraki et al. (2008) and Ohri and Shoghi (2012), the deformations in the turbine wheel are assumed to be linear elastic and the expansion coefficient of the material assumed to be isotropic. The thermal expansion coefficient, the Young's modulus and the density are modeled depending on temperature. Analogous to the thermal calculations a periodical behavior of the turbine wheel is assumed in the structure mechanical simulations. Thus, only a single blade segment is modelled. The rotational speed and the three dimensional temperature fields of the aerothermal calculations are used as boundary conditions. The pressure forces of the fluid state and the gravitational forces are neglected.

Turbine housing

In contrast to the turbine wheel, non-linear deformations are considered in the turbine housing (Oberste-Brandenburg et al., 2012). To model the plastic material behavior a multilinear approach is used which is described in Schicke et al. (2010). Stiffening of the material is taken into account as well. An isotropic and temperature dependent expansion coefficient is used to describe the thermal expansion. The density and the Young's modulus are also temperature dependent. To reduce the calculation times and the hard drive storage capacity, a mesh is used for the turbine housing which differs from the one of the thermal calculations. The mesh of the turbine housing consists of about 0.3 mio. nodes. Analog to the turbine wheel approach, the three dimensional temperature distribution of the aerothermal calculations are used as boundary condition and are mapped to the structure mechanical mesh. A fixed support is assumed at the inlet flange of the turbine housing. In accordance with the turbine wheel approach, the pressure forces of the fluid state and the gravitational forces are not considered in the model. With the two structure mechanical models, steady state calculations are conducted in each discrete time point of the thermal or aerothermal calculations.

VALIDATION

To validate the numerical model, a heating up thermal shock process is calculated by the ET method. In order to allow a comparison of the results for different speed up factors, two calculations of this thermal shock are conducted with different speed up factors of 10000 and 1000. For both simulations a numerical time step of 10^{-5}s is applied. In addition, two calculations with a time step of 10^{-4}s and speed up factors of 10000 and 20000 are conducted. The different time steps allow an investigation of the influence of the numerical time step by a constant speed up factor of 10000. Using a speed up factor of 20000 allows the calculation accuracy to be investigated by interchanging the ratio between the timescales of fluid and solid. In this case the timescales of the fluid state can be higher than the modified one of the solid state. Due to the high calculation times for a speed up factor of 1000, only a period of 200s is calculated for this case. For all ET simulations of the turbocharger, the fluid state is initialized by a quasi steady CFD. In this CFD calculation the wall temperatures imported from the steady state CHT calculation at OP1 (Figure 1) are used as a boundary conditions. The fluid inlet temperature is set however to $T_{F,\infty}$ and therefore this calculation represents the fluid state immediately after switching the fluid inlet temperatures from $T_{F,-\infty}$ to $T_{F,\infty}$ (Figure 1). The solid state of ET calculations was initialized with the temperatures taken from the CHT simulation at OP1.

For the validation of the ET method the heat transfer coefficients and the total von Mises stresses of the FF method are used as reference and are compared with the results of the ET method in order to assess the accuracy of this method. In addition, the transient temperatures at the measuring points are used to validate the results of the ET method with experimental data and with the FF method.

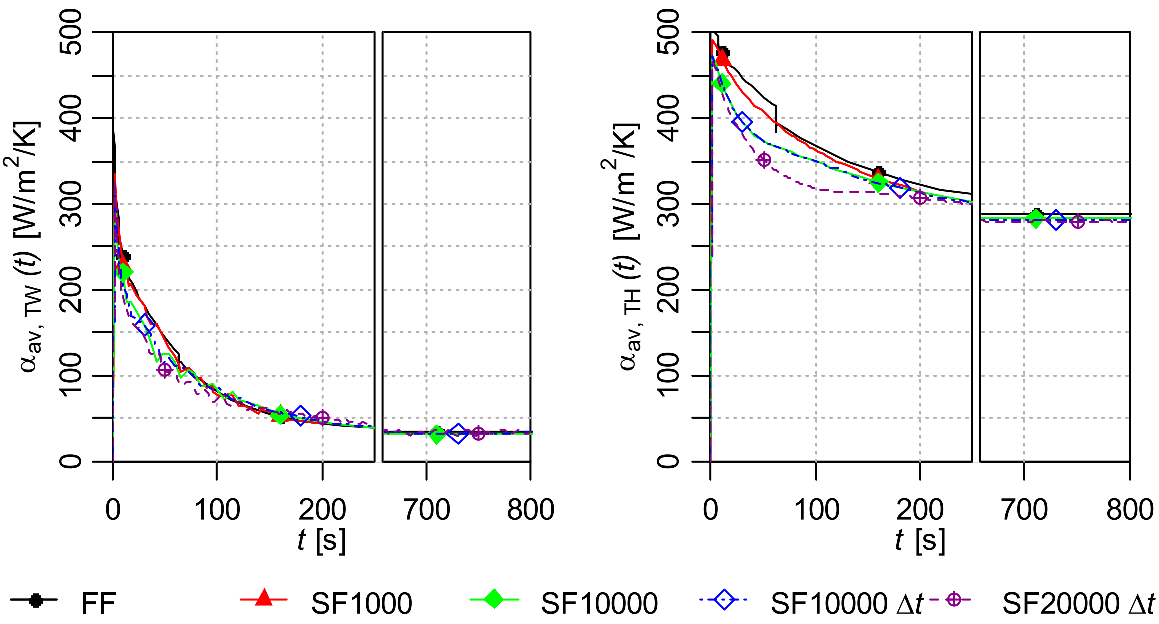

Figure 4: Averaged heat transfer coefficients at the turbine wheel (left) and at the turbine housing (right)

Figure 4 shows the evolution of the area averaged heat transfer coefficients at the turbine wheel (left) and the turbine housing (right) for all speed up factors and for the FF method. The high numerical time steps of 10^{-4}s are labelled with *Δt* here and in the following figures. The time step of 10^{-5}s is not labeled. The heat transfer coefficients of the turbine wheel reveal a good agreement for all speed up factors. For a speed up factor of 10000 and 20000 small oscillations occur in the transient behavior of the heat transfer coefficients of the turbine wheel. These oscillations result from unsteady flow effects which can have a high influence on the solid temperatures for high speed up factors due to the similar timescales of fluid and solid state. Compared to the turbine wheel, higher relative deviations up to maximal 12.5% at about 50s of thermal shock process occur in the turbine housing for a speed up factor of 10000 in reference to FF method. It has to be mentioned, that direct after the first FF update (detailed description in Diefenthal et al. (2015)) the difference decreases rapidly and remains small. Therefore, instead of the FF approach the ET method with speed up factor of 1000 could be used in the first 60s as the reference. High relative differences between the FF method and the ET method are shown at the turbine housing for a speed up factor of 20000 (up to 17.8% at 50s). Here, a monotonous behavior of the heat transfer coefficients is expected for a thermal shock process. However, the evolution of the heat transfer coefficients shows a small increase at about 150s after a strong decreasing period at the beginning. As shown in the turbine wheel this increase probably results from unsteady flow effects that have a high influence on the solid temperatures for high speed up factors.

Nearly no differences occur when comparing the calculations with the different numerical time steps. This shows that the chosen time steps have almost no influence on the solution. As the heat transfer coefficient is the dominant influence parameter on the Biot-Number, similar trends like in the heat transfer coefficient emerge in the distribution of the Biot-Number as well. The distribution of the Biot-Number over the Fourier-Number is depicted in Figure 5.

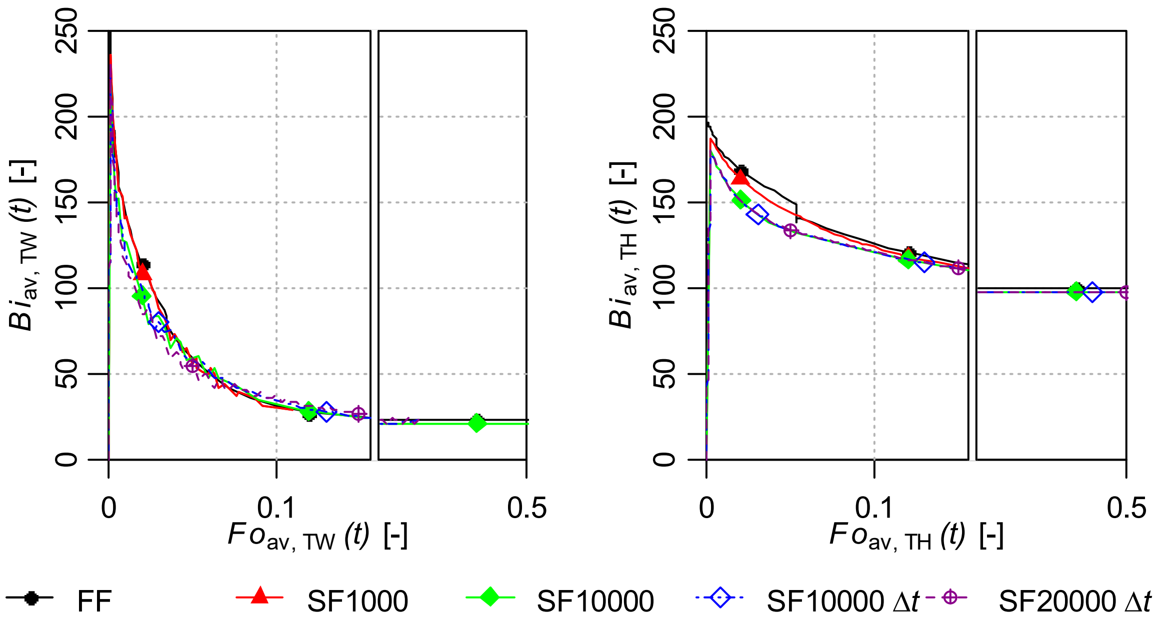

Figure 5: Averaged Biot-Number at the turbine wheel (left) and at the turbine housing (right) over the Fourier-Number

In order to validate the calculations of the ET method, the transient temperatures are compared with experimental data at the measuring points of the turbine wheel and of the turbine housing. Figure 6 illustrates the temperature evolutions at measuring point MW3 of the turbine wheel (left) and at measuring point MH3 of the turbine housing (right).

As expected from the accuracy of the heat transfer coefficients, the temperatures at the measuring points show a very good agreement with the experimental data for all measuring points and all speed up factors. Even for the speed up factor of 20000, the differences seem to be tolerable. The differences of the calculations of the different numerical time steps are very small. In order to quantify the differences, the experimental and the numerical data are normalized to a time step of 1.0s by a cubical interpolation. With the resulting temperature distributions, a relative root mean square (RRMS) deviation is determined over a period of 0s to 200s by the following equation:

$$\Delta T_{\mathrm{RRMS}} = \sqrt{\frac{1}{n}\sum_{i=1}^{n}\left(\frac{T_{\mathrm{exp,i}} - T_{\mathrm{sim,i}}}{T_{\mathrm{exp,i}}}\right)^2} \qquad \textbf{Equation 7}$$

The averaged deviations are summarized in Figure 7. The deviations of the FF method and the ET method with a speed up factor of 1000 are very similar and it seems that both methods result in a comparable accuracy. A speed up factor of 10000 tends to slightly higher deviations (maximal 9.5K at messing point MH1 referred to FF approach). Again, the influence of the numerical time step on the results of the speed up factors of 10000 is negligible. For a speed up factor of 20000, higher differences even up to 17.7K at messing point MH2 (compared with FF method) occur. This shows that the speed up factors cannot be increased without limitation. Thus, for the investigated turbocharger no higher speed up factors are investigated.

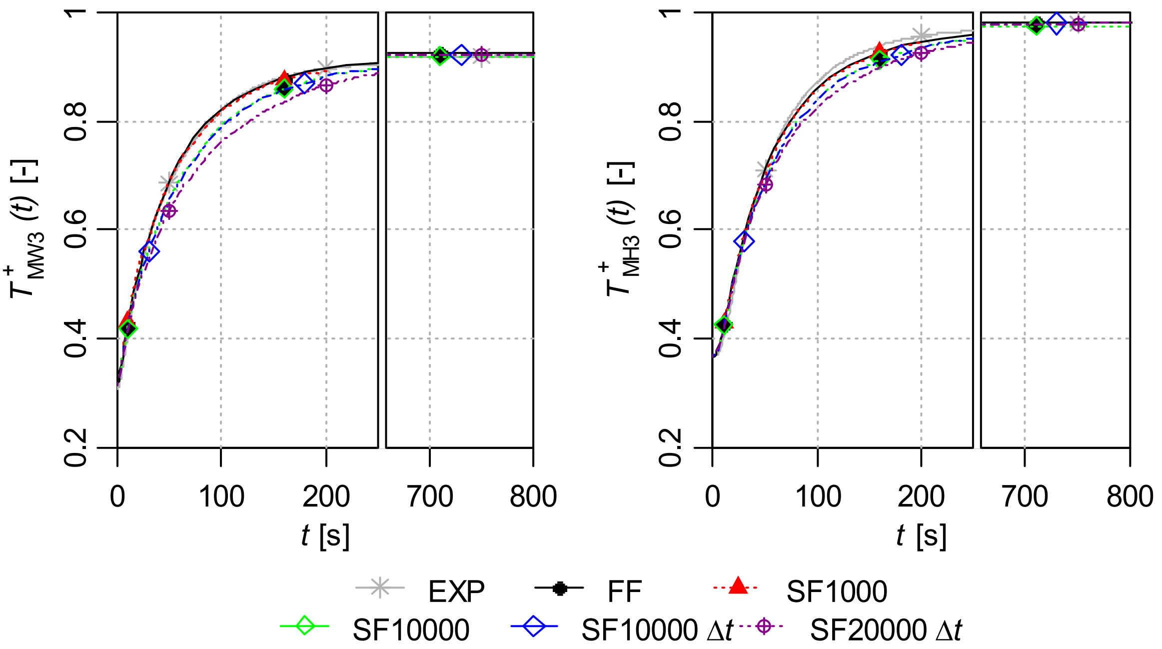

Figure 6: Measured and calculated temperatures of measuring point MW3 at the turbine wheel (left) and of measuring point MH3 at the turbine housing (right)

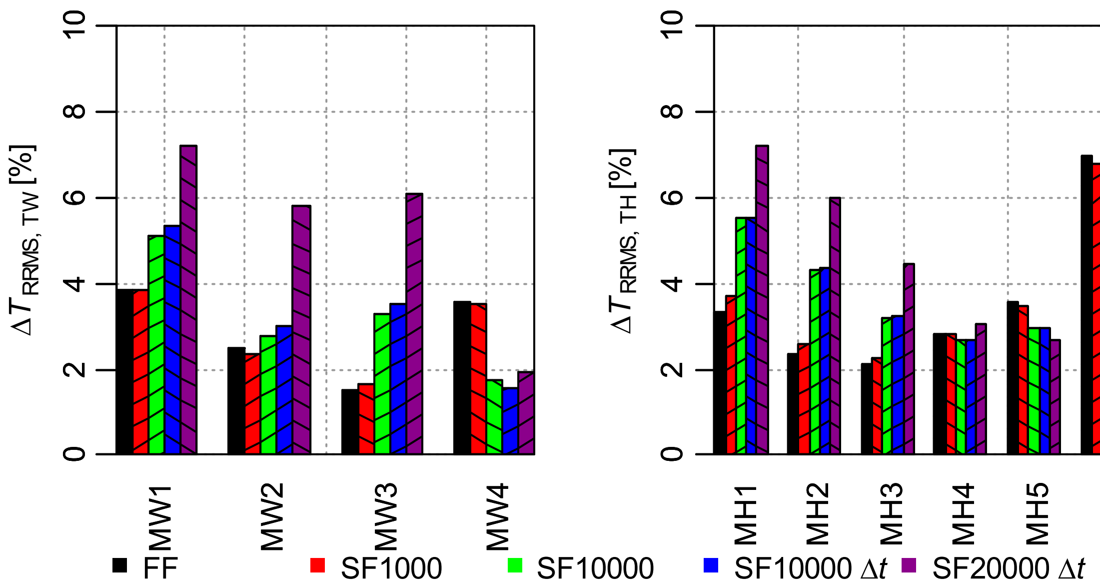

Figure 7: RRMS deviation between measured and calculated temperatures at the measuring points on the turbine wheel (left) and the turbine housing (right)

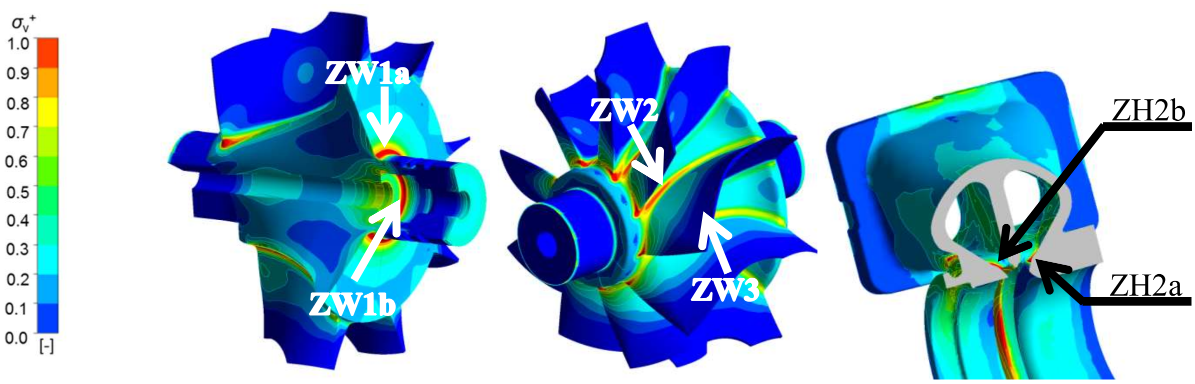

Figure 8: Von Mises stresses on the surface of the turbine wheel and the turbine housing

The critical zones are labeled in the Figure 8 with Z for zones, with W for turbine wheel and with H for turbine housing. They are numbered in descending order by the magnitude of the corresponding von Mises stress. The identified zones are in agreement with investigations of Heuer et al. (2006) and Ibaraki et al. (2008). In Figure 9 the transient evolutions of the von Mises stresses for the FF method and for the ET method with all investigated speed up factors is depicted for the critical zones ZW3 of the turbine wheel and ZH2a for the turbine housing.

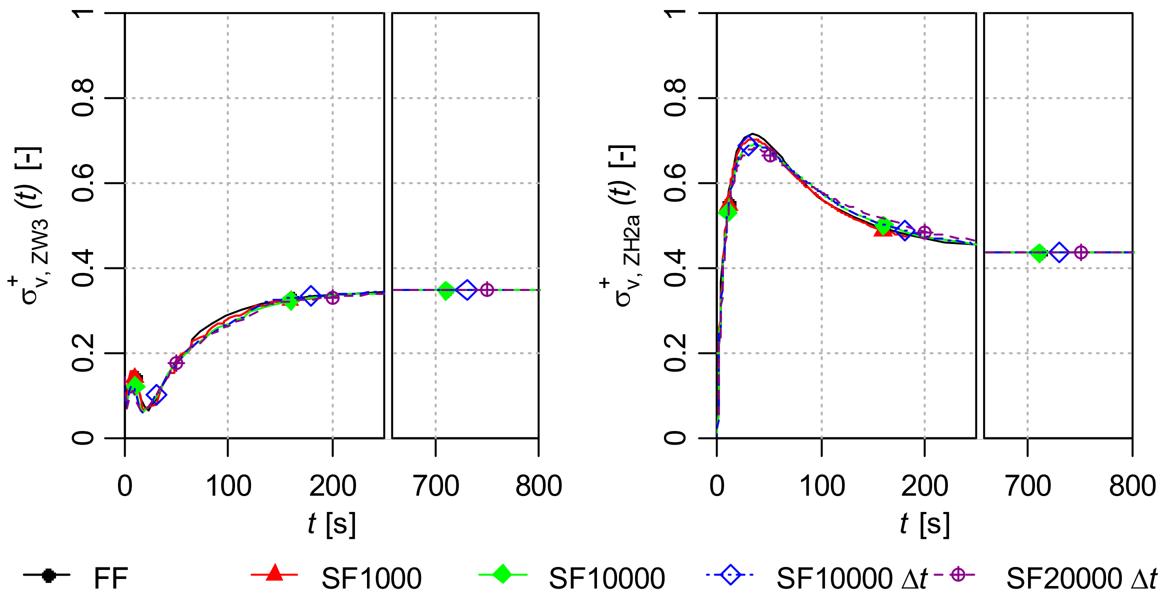

Figure 9: Transient evolution of the von Mises stresses at the critical zones ZW3 of the turbine wheel (left) and ZH2a of the turbine housing (right)

As can be seen, the results of the different methods of the different speed up factors and time steps are in good agreement. In the turbine housing the deviation of the maximum von Mises stress is about 2% for a speed up factor of 1000, about 4% for a speed up factor of 10000 and about 4% for a speed up factor of 20000. In the turbine wheel the deviations are similar. The deviations of the maximum von Mises stresses in the other critical zones are summarized in Figure 10 for all speed up factors. The higher deviations for speed up factors of 1000 are not shown for all critical zones as the maximum deviations occur at the end of the thermal shock but for a speed up factor of 1000 only the first 200s were calculated.

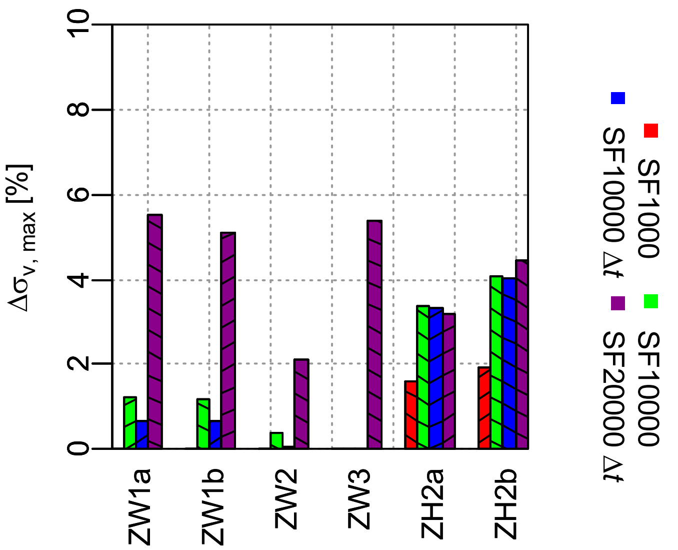

Figure 10: Deviations of the maximum total von Mises stresses between the FF method and the ET method for all critical points

The maximum deviation of the maximum total stresses is about 6% between the different methods and the different time steps. Based on the comparison of the methods and the validation of the experimental data, it can be clearly seen that the ET method and the FF method lead to very similar results with a similar accuracy. Thus, both the ET method and the FF method are suitable to calculate transient component temperatures with regard to thermal and thermo-mechanical fatigue. The choice of the optimal calculation method for the considered numerical problem is the matter of requirements for calculation times, user input and accuracy. Though the accuracy of the FF and the ET method is almost the same, the calculation times differ strongly. For a process period of 200s they are summarized in Table 1 using the unit core-h.

Table 1: Summary of the calculation times of the different numerical methods for a calculation period of 200s

Comparison of the calculation times of the different methods [core-h]					
Method	**Steady state CHT**	**Steady State CFD**	**Transient temperature**	**Total**	**Total [%]**
FF	≈ 60+60	-	≈ 500	≈ 620	≈ 100
ET SF1000	≈ 60+60	≈ 40	≈ 23800	≈ 23960	≈ 3870
ET SF10000	≈ 60+60	≈ 40	≈ 2300	≈ 2460	≈ 402
ET SF10000 Δt	≈ 60+60	≈ 40	≈ 215	≈ 375	≈ 61
ET SF20000 Δt	≈ 60+60	≈ 40	≈ 160	≈ 320	≈ 51

All calculations are conducted on the JARA HPC partition of the RWTH Aachen University and the Forschungszentrum Jülich on Intel Westmere-EP processors each with 6 Cores with 3 GHz. In the right column of the table the total calculation time of the FF method is used as a reference and is set to 100%. The calculation times of the ET method are given in relation to the FF method in this column. The calculation times of the ET method simulations with a numerical time step of 10^{-5}s and speed up factors of 1000 and 10000 are considerably higher than the times of the FF method. They are about a factor of 4 and 39 higher. For a time step of 10^{-4}s the calculation time is smaller than the FF method. For a speed up factor of 10000 the calculation time can be reduced by about 40% and by about 50% for a speed up factor of 20000. Higher speed up factors are not feasible as the calculation times of the stationary calculations become dominant over the total time. Furthermore, the accuracy of higher speed up factors is expected to be undesirable.

CONCLUSION

Due to steadily increasing inlet temperatures of turbocharger turbines, thermal and thermo-mechanical fatigue is a major topic of current investigations in the turbocharger industry in the last years. In the turbine components inhomogeneous temperature fields or temperature gradients cause thermal stresses in the material that can superimpose the mechanical stresses. Thus, the thermal stresses have to be considered in the design process of a turbocharger. To consider the thermal stresses in the design processes, a method is required that enables a calculation of the transient temperature fields within the timescales of a standard design process.

In the present paper a fast method is presented that is based on a full coupled transient calculation of fluid and solid heat transfer. Such calculations typically cause an high calculation effort since the timescales of fluid and solid differ significantly. Due to these different timescales, long conductional processes have to be calculated with small time steps to achieve numerical stability of the fluid calculation. In the presented method, the timescales of fluid and solid are equalized by reducing the specific heat capacity of the solid state by a constant factor called the speed up factor. With the modified heat capacity, the duration of the heat transfer process becomes shorter and the calculation time can be reduced significantly. To transfer the resulting times of this method into realistic times, the calculated times are multiplied by the speed up factor after the calculations are completed. In the present work it is shown that the Biot- and the Fourier-Number remain constant for a simple analytical approach of a heated or cooled mass point by introducing a speed up factor. This indicates a similarity of the temperature fields with and without using a speed up factor. Furthermore, the described method is applied to a turbocharger for speed up factors of up to 20000 in order to validate the results. The method shows a good agreement with reference calculations and with experimental data for small speed up factors. For higher speed up factors the deviations become higher. Nevertheless, with the ET method the calculation times can be reduced by about 40% compared to the reference calculations achieving a sufficient accuracy.

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