Comparison of a turbocharger model based on isentropic efficiency maps with a parametric approach based on Euler’s turbo-machinery equation

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Abstract: Faults in the intake and exhaust path of turbocharged common rail Diesel engines can lead to an increase of emissions and performance losses. Application of turbocharger models can help to detect and diagnose more faults as standard fault detection methods. The modeling of the turbocharger for onboard fault diagnosis can be obtained by different models. The differences between an approach based on the isentropic efficiencies and an approach based on Euler’s turbo-machinery equation are investigated in this paper. The two models for a GT1749MV turbocharger are parameterized with data from the engine testbed. The comparison is applied by issues of measured model inputs, number of internal parameters, parameterization effort and model accuracy. Both models are compared regarding the application for onboard diagnosis.

Keywords: Turbocharger modeling, heat transfer, isentropic efficiency, Euler’s turbo-machinery equation

1. INTRODUCTION

Dynamic models of turbocharged Diesel engines are needed for purposes of simulation assisted development of engine parts, control and diagnosis functions. The intake and exhaust path is an important part of the engine model, due to its growing complexity and nonlinear dynamics. The turbocharger model has a significant influence on the performance of the complete air and exhaust path model.

Mrosek and Isermann (2010) and Shaaban (2004) show that consideration of the heat transfer in the turbocharger housing leads to better model accuracy. Turbocharger efficiency maps usually delivered by the manufacturer are gained from the turbocharger hot gas test bench. They contain measurement points at medium to high turbocharger speeds with negligible effects of heat transfer and don’t reproduce the influence of pulsations occurring at the engine. Parameterization of the turbocharger using extrapolation from these measurement points, Guzzella (2009), leads nonetheless to insufficient modeling results. The alternative parameterization way is to use measurements from engine test bench in the operation region with low to high turbocharger speeds in order to take the heat transfer as well as engine pulsations into account.

In Mrosek and Isermann (2010) a fluidodynamic semi-physical model based on Euler’s turbo machinery equation with consideration of the heat flows in the turbocharger is presented. In Sidorow et al. (2011) the separation of aerodynamic compression respectively expansion from the heat flow is applied on a thermodynamic model based on isentropic efficiencies. The modeling approaches are both verified on measurements from the the dynamic engine test bench at Institute of Automatic Control. In this contribution both methods are compared.

2. TURBOCHARGER MODELS

The model of turbocharger with speed $n_{tc}$ as output is obtained from models of the compressor power $P_c$, turbine power $P_t$ and friction power $P_f$ combined in (1) taking the inertia $J_{tc}$ of compressor and turbine wheels and shaft into account, Merker et al. (2005).

$$\dot{n}_{tc} = \frac{P_t - P_c - P_f}{(2\pi)^2 n_{tc} J_{tc}}$$ (1)

Friction power $P_f$ is modeled using viscous friction equation

$$P_f = (2\pi)^2 n_{tc}^2 K_f$$ (2)

with friction coefficient $K_f$. The power models of the compressor and turbine compose of massflow and adiabatic enthalpy difference submodels

$$P_c = \dot{n}_{tc, model} \Delta h_{c,adi}$$ (3)

$$P_t = \dot{n}_{tc, model} \Delta h_{t,adi}$$ (4)
Fig. 1. Compression process a) and expansion process b) in a schematic h-s diagramm

The compressor- and turbine massflow models are given according to Zahn (2007), Zahn and Isermann (2008) by

\[
\dot{m}_{c,\text{model}} = f\left(\frac{p_{2c}}{p_1}, n_{tc}, T_1^*\right) \quad (5)
\]

\[
\dot{m}_{t,\text{model}} = f\left(\frac{p_3}{p_4}, \nu_{gt}\right) \quad (6)
\]

and are not the focus of this paper.

The compression process is illustrated in Fig. 1 a) using h-s-diagramm with lines of constant pressure \(p_1\) and \(p_{2c}\) and measured temperatures \(T_1\) and \(T_{2c}\). The diabatic enthalpy difference \(\Delta h_{c,\text{dia}}\), obtained from the measured temperatures represents both compression process and heat transfer. In order to separate the adiabatic compression from the heat inflow, the diabatic process can be divided into three parts. Thereby heat transfer into the compressor is assumed to occur on two lumped locations on the flow path (Shaaban (2004)). The specific heat inflow before compression \(q_{c,b}\) leads to temperature increase from \(T_1\) to \(T_1^*\). The adiabatic compression of the intake air from pressure \(p_1\) to \(p_{2c}\) is attended with temperature increase from \(T_1^*\) to \(T_{2c}\). The specific heat inflow after compression \(q_{c,a}\) leads to temperature increase from \(T_{2c}\) to \(T_{2c}^*\). The introduced parts of diabatic compression can be expressed by means of thermodynamic enthalpy definition, compare Zahn and Isermann (2008)

\[
\Delta h_{c,\text{dia}} = c_{p,a} (T_{2c} - T_1) \quad (7)
\]

\[
= q_{c,b} + \Delta h_{c,\text{adi}} + q_{c,a} \quad (8)
\]

\[
= c_{p,a} [(T_1^* - T_1) + (T_{2c} - T_1^*) + (T_{2c} - T_{2c}^*)] \quad (9)
\]

with \(c_{p,a} = 1080 \frac{J}{kgK}\) the specific heat capacity at constant pressure for the fresh air. Similar considerations are applied for the diabatic expansion process in the turbine illustrated in Fig. 1 b). The specific heat outflow before the expansion \(q_{t,b}\), adiabatic expansion \(\Delta h_{t,\text{adi}}\) and specific heat outflow after the expansion \(q_{t,a}\) can be expressed in the equation

\[
\Delta h_{t,\text{dia}} = c_{p,e} (T_3 - T_4) \quad (10)
\]

\[
= q_{t,b} + \Delta h_{t,\text{adi}} + q_{t,a} \quad (11)
\]

\[
= c_{p,e} [(T_3 - T_3^*) + (T_4^* - T_4)] \quad (12)
\]

with \(c_{p,e} = 1050 \frac{J}{kgK}\) the specific heat capacity at constant pressure for the exhaust gas and \(T_3^*\), \(T_4^*\) not measured temperatures.

Previous works of Shaaban (2004), Mrosek and Isermann (2010) show that turbocharger is part of a complex thermal system which contains different heat flows between compressor, turbine, engine housing, lubrication oil and environment. The most significant heat input before and after the adiabatic compression is assumed to come from the turbine over the housing, see Mrosek and Isermann (2010). According to Newton’s law of heat transfer the temperatures before and after adiabatic compression can be obtained from equations

\[
T_1^* = T_1 + \frac{\alpha_{q,c,b} A_{c,b}}{c_{p,a} \dot{m}_c} (T_3 - T_1) \quad (13)
\]

\[
T_{2c}^* = T_{2c} - \frac{\alpha_{q,c,a} A_{c,a}}{c_{p,a} \dot{m}_c} (T_3 - T_2) \quad (14)
\]

and the temperatures before and after adiabatic expansion concerning the heat outflow from the turbine are given by

\[
T_3 = T_3 - \frac{\alpha_{q,t,b} A_{t,b}}{c_{p,e} \dot{m}_t} (T_3 - T_1) \quad (15)
\]

\[
T_4 = T_4 + \frac{\alpha_{q,t,a} A_{t,a}}{c_{p,e} \dot{m}_t} (T_4 - T_1) \quad (16)
\]

with heat transfer coefficients \(\alpha_{q,c,b}\), \(\alpha_{q,c,a}\), \(\alpha_{q,t,b}\), \(\alpha_{q,t,a}\) and heat transfer areas \(A_{c,b}\), \(A_{c,a}\), \(A_{t,b}\) and \(A_{t,a}\).

The parameterization of the diabatic enthalpy differences using measured temperatures is applied together with the heat transfer models. Thereby the nonmeasured temperatures \(T_1^*, T_{2c}^*, T_3\) and \(T_4\) are estimated which can be interpreted as a shift of the line describing adiabatic compression and adiabatic expansion in Fig. 1 as is implied by the grey arrows. The thermodynamic and fluid-dynamic approach for adiabatic enthalpy differences \(\Delta h_{c,\text{adi}}, \Delta h_{t,\text{adi}}\) is introduced in the following sections.

2.1 Thermodynamic approach

The turbocharger modeling with thermodynamic approach is based on isentropic efficiencies of the compressor and turbine according to Sidorow et al. (2011) and considers heat transfer in turbocharger housing. In contrary to Merker et al. (2005), Guzzella (2009) model is parameterized from the engine test bench measurements instead of extrapolation of the OEM’s turbocharger maps.

**Compressor** The adiabatic enthalpy difference of the compressor according to equations (8), (9) depends on the unmeasured temperatures \(T_1^*\) and \(T_{2c}^*\). The temperature \(T_1^*\) is obtained from (13). The temperature after the compression \(T_{2c}^*\) can be calculated using of isentropic efficiency of the compressor \(\eta_c\) which shows the entropy increase
during the adiabatic compressing process in relation to isentropic compression process, Watson and Janota (1982):
\[
\eta_c = \frac{\Delta h_{c,s}}{\Delta h_{c,adi}}
\]  
(17)
The isentropic compression is shown as vertical line in the h-s diagram (Fig. 1 a)) from point \( T_1^* \) to \( T_{2c,s} \) and can be expressed as:
\[
\Delta h_{c,s} = c_{p,a} \cdot (T_{2c,s}^* - T_1^*)
\]  
(18)
The temperature \( T_{2c,s} \) is obtained from Merker et al. (2005)
\[
T_{2c,s}^* = T_1^* \left( \frac{p_{2c}}{p_1} \right)^{\frac{\kappa_a - 1}{\kappa_a}}
\]  
(19)
with isentropic exponent \( \kappa_a = 1.399 \) of air. Substitution of (19) in (18) and (18) in (17) yields
\[
\eta_c = \frac{c_{p,a} T_1^* \left( \frac{p_{2c}}{p_1} \right)^{\frac{\kappa_a - 1}{\kappa_a}} - 1}{\Delta h_{c,adi}}
\]  
(20)
In order to avoid the calculation in (20) the isentropic efficiency is modeled as neural net of type LOLIMOT (see Nelles (97)) with inputs turbocharger speed \( n_{tc} \) and compressor massflow \( \dot{m}_{tc} \), which are corrected by reference conditions \( T_{ref} \) and \( p_{ref} \) according to Merker et al. (2005), Guzzella (2009):
\[
\eta_c = \text{f}_{LOLIMOT}(n_{tc,corr}, \dot{m}_{c,corr})
\]  
(21)
\[
n_{tc,corr} = n_T \sqrt{\frac{T_{ref}}{T_1}}
\]  
(22)
\[
\dot{m}_{c,corr} = \dot{m}_c \frac{p_{ref}}{p_1} \frac{T_1}{T_{ref}}
\]  
(23)
Solving (20) for \( \Delta h_{c,adi} \) yields
\[
\Delta h_{c,adi} = \frac{c_{p,a} T_1^* \left( \frac{p_{2c}}{p_1} \right)^{\frac{\kappa_a - 1}{\kappa_a}} - 1}{\eta_c}
\]  
(24)
\textbf{Turbine}  
The adiabatic enthalpy difference of the turbine is introduced in (10) concerning Fig. 1 b)). The temperature \( T_3^* \) is calculated using (14). The temperature after the adiabatic expansion process \( T_3^* \) can be calculated with help of isentropic efficiency \( \eta_t \), see Watson and Janota (1982)
\[
\eta_t = \frac{\Delta h_{t,adi}}{\Delta h_{t,s}}
\]  
(25)
with isentropic expansion (see Fig 1 b)).
\[
\Delta h_{t,s} = c_{p,e} \cdot (T_{4,s}^* - T_3^*)
\]  
(26)
Applying equation
\[
T_{4,s}^* = T_3^* \left( \frac{p_4}{p_3} \right)^{\frac{\kappa_a - 1}{\kappa_a}}
\]  
(27)
with isentropic exponent \( \kappa_e = 1.361 \) of exhaust gas, compare Merker et al. (2005), and substitution analog to the compressor yields:
\[
\eta_t = \frac{\Delta h_{t,adi}}{c_{p,e} T_3^* \left( 1 - \left( \frac{p_4}{p_3} \right)^{\frac{\kappa_a - 1}{\kappa_a}} \right)}
\]  
(28)
According to Guzzella (2009) \( \eta_t \) depends on turbine blade speed ratio \( c_u \)

Fig. 2. Impeller v elocity triangles according to Zahn and Isermann (2008)
\[
c_u = \frac{d_{t,b} \pi n_{tc}}{\sqrt{2 c_{p,T} T_3^* \left( 1 - \left( \frac{p_4}{p_3} \right)^{\frac{\kappa_a - 1}{\kappa_a}} \right)}}
\]  
(29)
which is calculated inter alia from \( n_{tc} \), and variable geometry actuator \( s_{agt} \). Thereby is \( d_{t,b} \) diameter of the turbine wheel. Due to measurement heat transfer influence, \( \eta_t \) is interpolated by polynomials, see Sidorenko et al. (2011) and further is modeled according to Zahn (2007) as LOLIMOT:
\[
\eta_t = \text{f}_{LOLIMOT}(c_u, s_{agt})
\]  
(30)
The dependency of \( \eta_t \) on \( n_{tc} \) is considered in (29). Solving (28) for \( \Delta h_{t,adi} \) yields:
\[
\Delta h_{t,adi} = c_{p,e} T_3^* \eta_t \left( 1 - \left( \frac{p_4}{p_3} \right)^{\frac{\kappa_a - 1}{\kappa_a}} \right)
\]  
(31)
\textbf{2.2 Fluid dynamic approach}  
The turbocharger modeling with fluiddynamic approach is based on Euler’s equation of turbo-machinery. The adiabatic enthalpy difference of compressor and turbine are calculated according to Watson and Janota (1982), Zahn and Isermann (2008), Mrosek and Isermann (2010) using the parametric approach derived from physical equations of fluiddynamic theory.

\textbf{Compressor}  
The adiabatic enthalpy difference of compressor is derived from the velocity triangles of the impeller in- and outlet shown in Fig. 2. From Euler’s equation of turbo-machinery yields:
\[
\Delta h_{c,adi} = u_2 c_{2c,u} - u_1 c_{1u}
\]  
(32)
with peripheral velocities \( u_2 \) and \( u_1 \) and peripheral components of the absolute velocity at the impeller inlet \( c_{1u} \) and outlet \( c_{2c,u} \), see Fig. 2. Neglecting impeller inlet component yields:
\[
\Delta h_{c,adi} = d_{2c,u} \pi n_{tc} c_{2c,u}
\]  
(33)
with diameter of the compressor wheel \( d_{2c} \). From Fig. 2b) yields
\[
c_{2c,u} = \mu \left( u_{2c} - c_{2c,m} \cot (\beta_{2c,b}) \right)
\]  
(34)
with the meridional component of the absolute velocity \(c_{2c,m}\) and slip factor \(\mu\), compare Stodola (1945)

\[
\mu = \frac{c_{2c,u}}{c_{2c,u,\infty}} \tag{35}
\]

where

\[
c_{2c,u} = c_{2c,u,\infty} - c_{\text{slip}} \tag{36}
\]

with simplified approach

\[
c_{\text{slip}} = k_{\text{slip}} u_{2c} \tag{37}
\]

Substitution of the (37), (36) and (35) yields:

\[
\mu = 1 - \frac{k_{\text{slip}} u_{2c}}{u_{2c} - c_{2c,m} \cot \beta_{2c,b}} \tag{38}
\]

Using the mass flow continuity equation and compressor blade width \(b_{2c}\) one gets:

\[
c_{2c,m} = \frac{\dot{m}_c}{\rho_{2c}^3 d_{2c} \pi b_{2c}} \tag{39}
\]

with compressor outlet air density

\[
\rho_{2c}^3 = \frac{p_{2c}}{RT_{2c}} \tag{40}
\]

By substitution of (34) and (39) in (33) and taking (38) into account, adiabatic enthalpy difference \(\Delta h_{t,\text{adi}}\) can be expressed as:

\[
\Delta h_{c,\text{adi}} = \mu \left( (d_{2c} \pi n_{tc})^2 - \frac{\dot{m}_c \dot{n}_c}{\rho_{2c}^3 b_{2c}} \cot \beta_{2c,b} \right) \tag{41}
\]

**Turbine** The expansion in the turbine can be modeled analogous to the previously described. Approach similar to (32) is assumed:

\[
\Delta h_{t,\text{adi}} = u_3 c_{3u} - u_4 c_{4u} \tag{42}
\]

Neglecting the swirl at turbine outlet yields:

\[
\Delta h_{t,\text{adi}} = \pi d_{3u} n_{tc} c_{3u} \tag{43}
\]

where \(d_{3u}\) is turbine diameter.

According to velocity triangle at turbine inlet shown in Fig. 3 a) the equation

\[
c_{3u} = c_{3m} \cot \alpha_3 \tag{44}
\]

with meridian component of the absolute velocity at the turbine inlet \(c_{3m}\) is derived. Considering the variable turbine geometry actuator, the parameter \(\cot \alpha_3\) is modeled as polynomial function of 4th order dependent on the guide vanes actuator position \(s_{\text{vgt}}\):

\[
\cot \alpha_3 = f_{\text{poly}} (s_{\text{vgt}}) \tag{45}
\]

\(c_{3m}\) is derived from the mass flow continuity equation

\[
c_{3m} = \frac{\dot{m}_t}{\rho_3^3 d_3 \pi b_3} \tag{46}
\]

with turbine blade width \(b_3\) and the exhaust gas density

\[
\rho_3^3 = \frac{p_3}{RT_3} \tag{47}
\]

Finally the adiabatic enthalpy difference is expressed by

\[
\Delta h_{t,\text{adi}} = \frac{n_{tc} \dot{m}_t}{\rho_3^3 b_3} \cot \alpha_3 \tag{48}
\]

### Table 1. Compressor model parameters

<table>
<thead>
<tr>
<th>method</th>
<th>compressor parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>thermodynamic</td>
<td>(\alpha_{q,c,b} A_{c,b}), (\alpha_{q,c,a} A_{c,a}), (n_c ), (n_{tc,\text{cor}}, \dot{n}<em>c, \cot (\alpha_3, s</em>{\text{vgt}}))</td>
</tr>
<tr>
<td>fluiddynamic</td>
<td>(\alpha_{q,c,b} A_{c,b}), (\alpha_{q,c,a} A_{c,a}), (k_{\text{slip}}), (\frac{\cot (\alpha_3, s_{\text{vgt}})}{\beta_3})</td>
</tr>
</tbody>
</table>

### Table 2. Turboe model parameters

<table>
<thead>
<tr>
<th>method</th>
<th>turbine parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>thermodynamic</td>
<td>(\alpha_{q,t,b} A_{b}), (\alpha_{q,t,a} A_{a}), (n_t ), (e_c, s_{\text{vgt}})</td>
</tr>
<tr>
<td>fluiddynamic</td>
<td>(\alpha_{q,t,b} A_{b}), (\alpha_{q,t,a} A_{a}), (\frac{\cot (\alpha_3, s_{\text{vgt}})}{\beta_3})</td>
</tr>
</tbody>
</table>

Modeling approaches presented in last sections contain parameters which have to be determined applying identification methods. The effects of flow unsteadiness are considered in both models first by using the pressures before and after the compressor respectively turbine as inputs of massflow oscillations. Further engine test bench measurements containing gas pressure and massflow calculations are used for parameter estimation. In following the difference in the parameterization effort of both models is outlined. Furthermore the modeling approaches are compared concerning model inputs and model accuracy.

#### 3.1 Comparison concerning identification issues

Unknown parameters of compressor and turbine model have to be estimated iteratively by minimizing the model error which is given by mathematical norm between the calculated model outputs and measurements. The compressor parameters to be estimated are summarized in the table 1 for the thermodynamic and fluiddynamic approach. The model parameters of the turbine are summarized in the analogous manner in the table 2 for both methods.

The parameters of the compressor model can be identified from measured adiabatic enthalpy with corrected temperatures \(T_1^*\) and \(T_2^*\). (13), (14) and \(\Delta h_{c,\text{adi}}\) from the corresponding thermodynamic respectively fluiddynamic calculation. The parameters from table 1 have to be estimated minimizing compressor model error

\[
e_c = ||\Delta h_{c,\text{adi}} - \frac{k_{\text{slip}}}{\pi n_{tc} c_{3u}}||_2 \tag{49}
\]
Experiment is designed with as wide as possible covering of the engine operation region. The control of (49), (50) and (51), accepting increase of computational complexity due to selection of weights and rising amount of parameters. The turbocharger efficiency maps obtained over heat transfer estimation motivated by physics are expected to be more accurate at lower speed turbocharger operation region than the extrapolated OEM maps without measured data. However the reliability is limited by the heat transfer rate which increases with opening turbine geometry actuator and lower engine speed. Both thermodynamic and fluiddynamic turbocharger models contain additionally the compressor- and turbine massflow models, which are calculated by (5) and (6) and have to be parameterized separately minimizing the error between the measured and modeled massflow. Finally the turbocharger speed $n_{tc}$ can be calculated according to (1).

The thermodynamic model contains, in addition to constant parameters, two neuronal nets $\eta_c(n_{tc,corr}, n_{tc,corr})$ and $\eta_t(e_{u,avg})$ of type LOLIMOT, which have to be parameterized in every iteration step. These models compose of several local linear models, which are interpolated by Gaussians, see Nelles (97). The amount of parameters $\theta$ for a LOLIMOT net with $\nu$ inputs and $\gamma$ local linear models is defined by $\theta = \gamma(\nu + 1) + 2\gamma \nu$. The number of parameters of isentropic efficiencies in thermodynamic model is shown in table 3. Contrary to the thermodynamic model, the fluiddynamic model contains only constant parameters which have to be estimated. Summing up, the 74 parameters of the thermodynamic approach and the circumstance of training the neuronal nets in every iteration step show a higher parameterizing effort then the 10 parameters of the fluiddynamic model.

<table>
<thead>
<tr>
<th>model</th>
<th>inputs</th>
<th>local models</th>
<th>parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_c$</td>
<td>2</td>
<td>7</td>
<td>49</td>
</tr>
<tr>
<td>$\eta_t$</td>
<td>2</td>
<td>3</td>
<td>21</td>
</tr>
</tbody>
</table>

3.2 Measured input analysis

The thermodynamic and fluiddynamic models of compressor power (3) are analyzed in the following. The massflow model according to (5) is the common for both approaches and requires, regarding the feedback of $n_{tc,corr}$, measured inputs $p_1$, $p_2$, and $T_1$. Applying the thermodynamic approach, the adiabatic enthalpy difference $\Delta h_{t,adi}$ is calculated using the temperature at compression $T_3^*$. In contrary to this, $\Delta h_{t,adi}$ calculated using fluiddynamic approach has the gas density after the compression (40) as input. Considering the heat transfers (13), (14) both modeling approaches need the measured temperature $T_3$. Fluiddynamic model requires an additionally measurement of $T_{2C}$. By analyzing of both approaches of turbine power (4) using equations (6), (16), (24) and (48) can be derived that the inputs for both models are $p_3$, $p_4$, $T_3$, $T_4$ and $s_{avg}$. The required measured inputs are shown in the Fig. 6. Summing up the fluiddynamic model needs one measured input, the temperature $T_{2C}$, more than the thermodynamic model.

3.3 Comparison concerning model accuracy

The practical application of introduced models for onboard diagnosis is verified on measurement from the test bench with a turbocharged 1.9 liter Opel Common Rail Diesel engine. Experiment is designed with as wide as possible covering of the engine operation region. The control
In this contribution two different turbocharger modeling approaches have been presented. Both approaches consider heat outflow from the turbine and heat inflow into the compressor and are parameterized from measurements accomplished on the dynamic engine test bench at Institute of Automatic Control. The introduced modeling approaches are compared concerning identification issues, parameterization effort, analysis of model inputs, accuracy. The fluiddynamic model shows better performance concerning the investigated criteria of comparison. Both models are applicable for onboard fault diagnosis.

5. CONCLUSION

The presented contribution is developed in cooperation between the Institute of Automatic Control at TU-Darmstadt and GM Powertrain Europe.

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The results of the comparison can be summarized in table 4. Due to manageable amount of parameters fluiddynamic model is more suitable for the onboard applications. The thermodynamic approach has a higher parameterization effort, due to application of neuronal networks for the

<table>
<thead>
<tr>
<th>approach</th>
<th>parameters</th>
<th>inputs</th>
<th>RMSE in 1/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>thermodynamic</td>
<td>74</td>
<td>10</td>
<td>60.57</td>
</tr>
<tr>
<td>fluiddynamic</td>
<td>10</td>
<td>11</td>
<td>55.53</td>
</tr>
</tbody>
</table>

isentropic efficiencies of the compressor and turbine. The fluiddynamic model contains one measured input more then the thermodynamic. Both models show good accuracy in the considered operation range.

Table 4. Comparison results

In this contribution two different turbocharger modeling approaches have been presented. Both approaches consider heat outflow from the turbine and heat inflow into the compressor and are parameterized from measurements accomplished on the dynamic engine test bench at Institute of Automatic Control. The introduced modeling approaches are compared concerning identification issues, parameterization effort, analysis of model inputs, accuracy. The fluiddynamic model shows better performance concerning the investigated criteria of comparison. Both models are applicable for onboard fault diagnosis.

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