Application of an Up- and Downscaling-method to estimate the capability of micro gas turbines for local energy supply of residential buildings

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Abstract

The Swiss national research project *MicroPolygen* **[1]** is focussed on cogeneration applications for buildings, for which combined heat and power generation with an electrical power output less than 15 kW are of interest. Micro gas turbines (MGT) were analysed as a possibility for local energy supply in this power range. However, nowadays MGTs are only available down to 30 kW_{el}. To estimate the capability of MGTs with lower power output a theoretic model with the ability to simulate part load behaviour was developed. In a first step the model was calibrated based on performance data of a commercial state of the art MGT with a power output of 30 kW_{el}. In the second step the model was up scaled and compared to an existing 60 kW_{el} MGT. Finally based on these results the extrapolation to power less than 15 kW_{el} was performed.

As results of the upscale-model the manufacturer's performance data of

- the electrical efficiency as well as the generated electrical power output were reached with an average deviation maximum of 11.7 % and
- the waste heat temperatures were simulated with a relative deviation maximum of 12.5 %.

For the theoretical downscale-model it was postulated to get similar deviations as with the upscale-model. Therefore the part load behaviour of the downscale-model is assumed to be similar to that of the analysed commercial 30 and 60 kW micro gas turbines.

The downscale-model at full load was defined with

- an electrical power output of 15.2 kW,
- an electrical efficiency of 25 % and
- a waste heat temperature maximum of 252 °C.

The developed model for the estimation of the capability of micro gas turbines with electrical power output less than 15 kW should be understood as benchmark model for prospective developments in the MGT market.

1 Introduction

The Swiss national research project *MicroPolygen* **[1]** is focussed on cogeneration applications for buildings, for which combined heat and power generation with an electrical power output less than 15 kW are of interest. Micro gas turbines (MGT) were analysed as a possibility for local energy supply in this power range. However, nowadays MGTs are only available down to 30 kW_{el}. To estimate the capability of MGTs with lower power output a theoretic model with the ability to simulate part load behaviour had to be developed.

In this sense, the idea was to build a simple thermodynamic model, which reproduces the performance data of a commercial available 30 kW MGT and to use the manufacturer's given performance data to evaluate appropriate physical approaches.

Then, a self-defined upscale-model of 60 kW electrical power output had to be developed to validate the quality of the approaches made, according to the manufacturer's existing performance data for a commercial MGT in the same power range. The postulate was made, that, if the existing performance data is reached within an acceptable tolerance by using an upscale-model, a theoretical downscale-model – using the same approaches and state of the art technology – with 15 kW electrical power output could be developed with a similar accuracy to prospective MGT in this range. This benchmark model produces data, which have to

be understood as best-case performance data and conditions for possible future realistic developments on the MGT market.
Finally, the developed simulation models may be used to estimate the capability of MGT in different local energy supply or building simulations (IDA/ICE, TRNSYS) or the presented thermodynamic approaches may be used to build more specific models to simulate part load performance.

2 Micro gas turbine model

For the development of the thermodynamic approaches, a commercial MGT with 30 kW power output was taken as state of the art. Furthermore, the fuel was defined as natural gas and the given performance data were based on ISO conditions (15 °C ambient temperature, 1 atm ambient pressure, 60 % relative humidity). These conditions were also taken for the simulation model.
The common micro gas turbine process is drawn in **Figure 1**. At full load the process is described as follows:
First, the intake air at 15 °C and 1 atm is used for the cooling of the electronic devices and the generator. This cooling/heating effect was neglected in the simulation model, because of missing data for validation. After the generator, the air passes through a single stage centrifugal compressor, where the air with a mass flow of 0.31 kg/s is compressed to 3.8 bar (g) and reaches a temperature of about 205 °C. Then the air passes the annular-recuperator, heats up to 510 °C and enters the combustion chamber at a pressure of 4.8 bar (a). The air-fuel ratio λ is about 7.5.
The turbine inlet temperature of the waste gas is about 816 °C and expands in the single stage turbine to 594 °C and approximately 1 atm **[2]**. The produced work is used to drive the compressor as well as the permanent magnet generator. The rotation speed at full load is 96‘000 rpm. Electricity is produced at a frequency of 1’600 Hz. To transform the electric current to 50 Hz for the national grid a power converter is needed.
Finally, the expanded waste gas passes the recuperator, where it preheats the incoming air from the compressor outlet. The waste heat leaves the recuperator at a temperature of about 276 °C. Due to low emissions the waste heat may be used for drying processes or heating of domestic hot water.

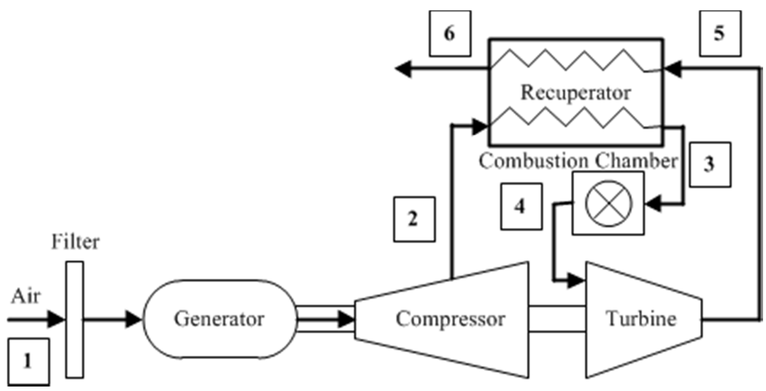

Figure 1: Micro gas turbine process

The simulation model of the micro gas turbine process bases on an open thermodynamic cycle. Furthermore, the manufacturer data **[3,4]** were used to determine the isentropic efficiency of the components.
For the model the following assumptions are applied:

- The preheating of the intake air due to the electronic device cooling is neglected.
- The efficiency of the isentropic compression keeps constant over the whole operation range.
- Heat and pressure losses are neglected.
- The combustion is ideal, adiabatic and isobar.

- The efficiency of the isentropic expansion keeps constant.
- Friction loss is neglected.
- The efficiency of the power generator is equal one and keeps constant over the whole operation range. In reality the efficiency is about 0.95 at full load and decreases with lower power.
- Mass flow changes due to fuel addition, air bearings etc. are neglected.

In the following a rough description of the component models is given.

Compression: 1 → 2

Based on manufacturer data an isentropic compressor efficiency of 84 % was determined. In literature efficiencies in the range from 60 – 86 % can be found for industrial compressors used in gas turbine processes **[5]**. The efficiency range of turbo compressors of the manufacturer Garret, which may be used for smaller applications, is between 71 – 80 % **[6]**. Therefore the determined efficiency is assumed to be realistic thought a bit optimistic. However, small variations of the temperature difference of 10 K may change the determined efficiency for turbine and compressor with about 4.5 – 8 %. These changes are factors of uncertainty in known measurements **[7]** and would also affect the simulation model, if e.g. the preheating of the intake air wouldn't be neglected.
The pressure ratio and mass flow is assumed to be the same as in the turbine.

$$T_{2s} = T_1\, pr^{\frac{\kappa-1}{\kappa}},\; T_2 = T_1 + \frac{(T_{2s}-T_1)}{\eta_{i,V}},\; P_V = c_{pm}\, \dot{m}\, (T_2 - T_1),\; \eta_{i,V} = \frac{T_{2s}-T_1}{T_2-T_1} \qquad \{2.1\}$$

Ambient temperature T_1, compressor's exit temperature T_2, compression pressure ratio $pr = p_2/p_1$, isentropic exponent κ, isentropic efficiency $\eta_{i,V}$, average specific heat capacity c_{pm}, power P_V, mass flow $\dot{m}$

Combustion: 3 → 4

The *fuel flow energy*, i.e. the energy, which is added in the combustion chamber, is known and related to the lower heating value (LHV).

$$T_4 = \frac{\dot{Q}_{BS}}{\dot{m}\, c_{pm}} + T_3 \qquad \{2.2\}$$

Turbine inlet temperature T_4, fuel flow energy $\dot{Q}_F$, mass flow $\dot{m}$, average specific heat capacity c_{pm}, recuperator exit temperature air side T_3

Expansion: 4 → 5

The determined turbine efficiency at full load is 70.5 %. In literature efficiencies for industrial turbines up to 88 % are documented **[5]**. In the considered range, e.g. turbochargers made by Garrett, show efficiencies between 65 – 78 % **[6]**, i.e. the calculated turbine's efficiency is realistic.

$$T_{5s} = T_4 \left(\frac{1}{pr}\right)^{\frac{\kappa-1}{\kappa}},\; T_5 = T_4 - \eta_{i,T}(T_4 - T_{5s}),\; P_T = c_{pm}\, \dot{m}\, (T_5 - T_4),\; \eta_{i,T} = \frac{T_5-T_4}{T_{5s}-T_4} \qquad \{2.3\}$$

Turbine inlet temperature T_4, turbine exit temperature T_5, pressure ratio $pr = p_4/p_5$, isentropic exponent κ, isentropic efficiency η_{Ts}, average specific isobaric heat capacity c_{pm}, power P_T, mass flow $\dot{m}$

To simulate the turbine part load performance, the volumetric flow rate function for nozzles and orifices (outflow from a tank, filled with a compressible fluid **[7]**) is applied. With the turbine inlet temperature and the mass flow rate the corresponding pressure ratio of each part load state can be determined.

$$\dot{m} = \mu\, A_a\, \Psi_s \sqrt{2\, p_{t4}\, \varrho_{t4}} \qquad \{2.4\}$$

$$\Psi_s = \sqrt{\frac{\kappa}{\kappa-1}\left[\left(\frac{p_a}{p_{t4}}\right)^{\frac{2}{\kappa}} - \left(\frac{p_a}{p_{t4}}\right)^{\frac{\kappa+1}{\kappa}}\right]} \qquad \{2.5\}$$

Mass flow $\dot{m}$, outflow factor μ, exit area A_a, outflow function Ψ_s, isentropic exponent κ, turbine inlet pressure $p_{t4} = f(T_4)$, fluid density $\varrho_{t1} = f(T_4, p_{t4})$ and ambient pressure at turbine exit $p_a = 101'325\ Pa = constant$

An alternative to the outflow function would be the Stodola equation for the turbine model **[7,9]**. This model was not applied in this study.

Recuperator (Air-Air, counter flow): 2 → 3 / 5 → 6

The common recuperator, which is integrated in the considered MGT, is a counter flow annular-recuperator. However, data about geometry, flow speed and construction were missing. Therefore a simple model of a counter flow heat exchanger was used.

$$\dot{Q} = k\ A\ \Delta T_m = \dot{m}\ c_{p56}\ (T_5 - T_6) = \dot{m}\ c_{p23}\ (T_3 - T_2) \quad with\ R_{23} \neq R_{56} \tag{2.6}$$

Heat flux $\dot{Q}$, heat transfer coefficient k, heat transfer surface area A, log mean temperature difference ΔT_m, mass flow $\dot{m}$, specific isobaric heat capacity (for air) c_p, waste gas inlet temperature T_5, waste gas outlet temperature T_6, air inlet temperature T_2, air outlet temperature T_3, heat flux capacity ratio R

The part load behaviour of the recuperator was modelled as follows **[9,10]**:

$$Nu_m = \frac{\propto d_i}{\lambda}; Re = \frac{w\ d_i}{\nu}; 2300 \leq Re \leq 10^4; 0.6 \leq Pr \leq 1.5; d_i/l \leq 1 \tag{2.7}$$

$$Nu_m = 0.0214(Re^{0.8} - 100)\mathrm{Pr}^{0.4}[1 + \left({d_i}/{l}\right)^{\frac{2}{3}}] \cong 0.0214 Re^{0.8} \tag{2.8}$$

$$with\ Re = \frac{\dot{m}\ d_i}{A\ \nu}, k = \frac{1}{\frac{1}{\alpha_1}+\frac{s}{\lambda}+\frac{1}{\alpha_2}}\ and\ the\ simplified\ relation$$

$$k \sim \alpha \sim Re^{0.8}\ or\ \dot{m}^{0.8}\ results\ as\ first\ approximation$$

$$kA \cong (kA)_0 \left(\frac{\dot{m}}{\dot{m}_0}\right)^{0.8} \tag{2.9}$$

Nusselt number Nu_m, characteristic length d_i, length l, thermal conductivity λ, Reynolds number Re, fluid's velocity w, kinematic viscosity ν, Prandtl number Pr, heat transfer coefficient k, heat transfer surface area A, wall thickness s, mass flow $\dot{m}$, index 0: full load (operating point)

The results of the simulation model, based on the defined boundary conditions and the used simplified approaches are shown in **Figure 2**, **Figure 3** and **Figure 4**. The abbreviation *HPNG* mentioned in the figures stands for High-Pressure-Natural-Gas and assumes, that the used gas is provided at the required pressure by an existing source, i.e. no additional power for compression is needed.

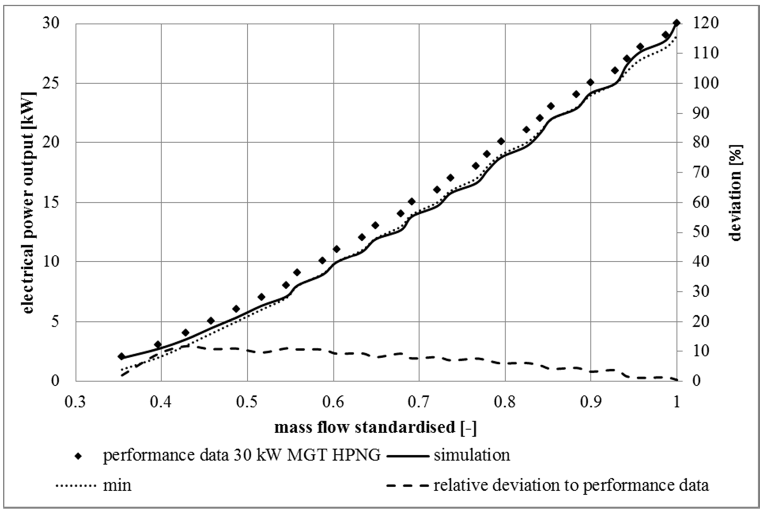

Figure 2: Electrical power output to mass flow

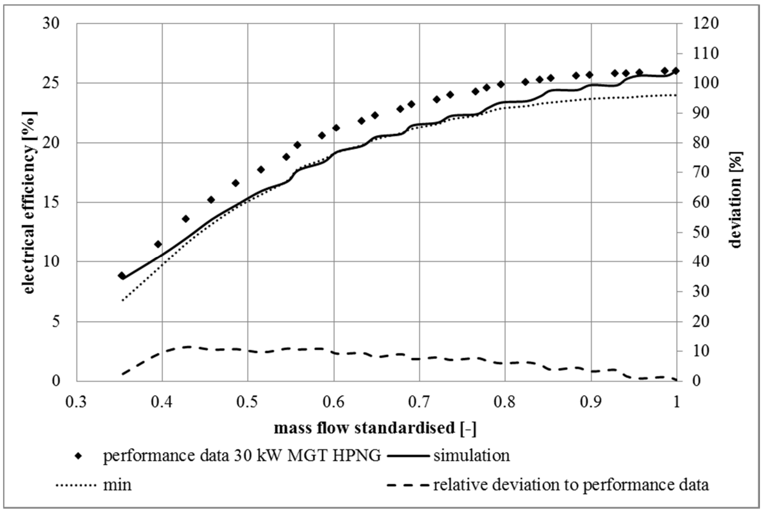

Figure 3: Electrical efficiency to mass flow

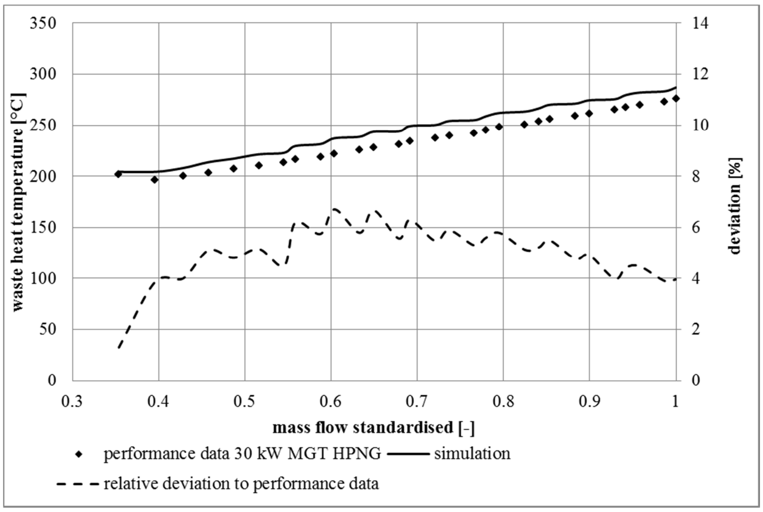

Figure 4: Waste heat temperature to mass flow

The produced electrical power output (**Figure 2**) shows a relative deviation maximum of 12 % (average 7 %) between the existing performance data and the simulation. The manufacturer states a possible deviation of -1 kW_{el} for the power output and -2 % points for the electrical efficiency at full load. Due to the lack of data this tolerance is assumed to be valid over the whole operation range (min values). Doing so, the simulated power output mostly stays within this range of tolerance. The same applies to the electrical efficiency in **Figure 3.** The waste heat temperature of the simulation in **Figure 4** shows a relative deviation maximum of 6.7 % (average 5 %) compared to the performance data. Some of the deviation may be explained by the isentropic efficiencies of the turbine and compressor. In reality these efficiency will vary over the operation range. Since no such data for the investigated MGT is available and to hold the simulation model as simple as possible these efficiencies are kept constant. Nevertheless, the defined approaches simulate the performance of the MGT system with less than 7 % relative average deviation and show correct characteristics. Therefore with the developed model a good approximation can be achieved. Improvements could be accomplished if more data about the components and measurements would be available.

3 Upscale-model

The upscale-model for 60 kW electrical power output was designed similar to the 30 kW MGT model.

The idea of up scaling was, to evaluate the introduced approaches with performance data of a commercial 60 kW MGT of the same manufacturer.

A technical tolerable conformity of the results of the upscale-model would confirm the plausibility of the approaches made. Furthermore, this generates the postulate, that for the opposite case of a theoretical down-scale model with similar efficiencies for the components, the simulation would deliver similar technical tolerable results. However, since no part load data for the down-scale model are available they are extrapolated from the 30 kW MGT data. This had been made for both of the models with 15 and 60 kW_{el}. Therefore, an air-fuel ratio had to be defined, with consideration of technical aspects. In this section the results of two different 60 kW MGT simulation models are shown in **Figure 5**, **Figure 6** and **Figure 7** and compared to the manufacturer's performance data.

The simulation model with extrapolated data shows a better conformity with the manufacturer's data for electrical power output and electrical efficiency, than the simulation model based on the given performance data for part load. Only the simulated waste heat temperatures in the range below 16 % (10 kW) of full load are worse in the simulation based on extrapolated data. However, the simulated waste heat temperatures are generally too high, because of the defined low turbine and compressor efficiency for MGT in this power range (60 kW). The reason therefore can be found in the defined upscale conditions: Usually large system components (higher power) are more efficient than smaller components. However, the efficiencies for turbine and compressor in the upscale-model are the same as in the 30 kW MGT model. Furthermore, these conditions change the recuperator's dimensioning and at the end the systems temperature characteristics as well as the other performance values. In this case, the resulting waste heat temperatures are too high. In real applications a MGT will never be operating below 50 % load in steady-state, because of economic reasons. With that kept in mind, the simulation results are in a technical reasonable range of the same quality. The average relative deviation of the simulation of a 60 kW MGT system compared to the manufacturer's performance data was 11.7 % for the electrical efficiency and electrical power output and 12.5 % for the waste heat temperature.

Finally, considering that in the upper 50 % of load, the relative deviation maximum never exceeds about 10 % for every significant value, the developed thermodynamic approaches and the data extrapolation are confirmed in their usefulness for upscale and downscale applications.

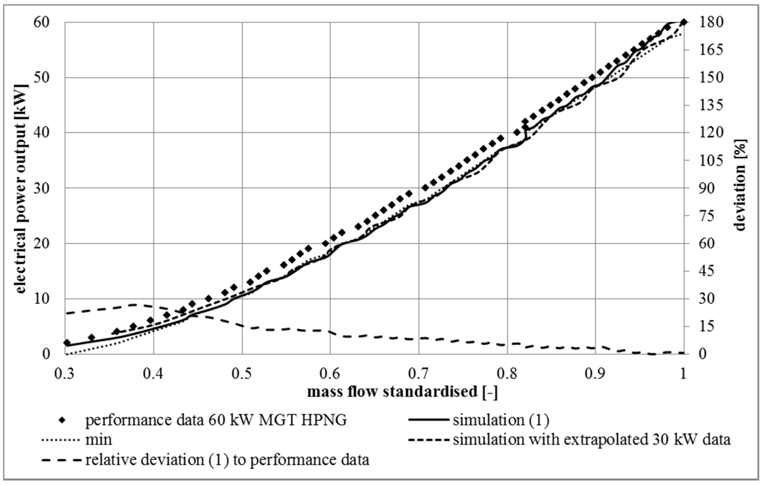

Figure 5: Electrical power output to mass flow

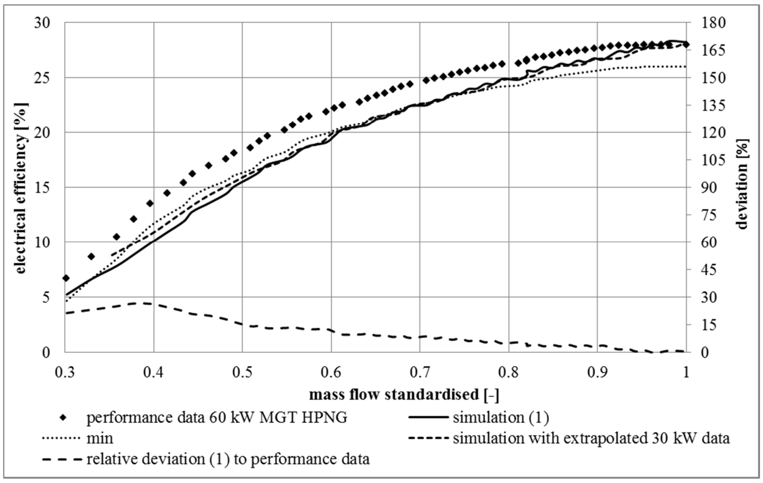

Figure 6: Electrical efficiency to mass flow

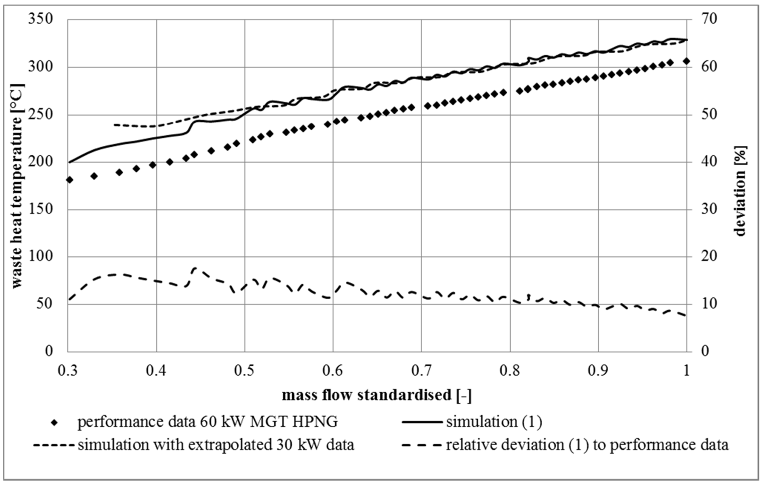

Figure 7: Waste heat temperature to mass flow

4 Downscale-model

The dimensioning of the downscale-model has been done as for the 30 kW and 60 kW MGT models. Furthermore, technical aspects such as the properties of used materials (e.g. temperature range) and technical restrictions for smaller components (e.g. technical possible pressure ratio) were considered.

The model was dimensioned for an electrical power output of 15 kW. For the best-case the same isentropic efficiencies for the turbine and the compressor had been taken. Moreover, the recuperator was dimensioned for the defined operating point. The mass flow is 0.19 kg/s and in the combustion chamber about 61 kW (LHV) natural gas is burnt. For the part load performance the air-fuel ratio data of the 30 kW MGT were extrapolated and applied to the downscale-model, as it had been done for the 60 kW MGT upscale-model with extrapolated data.

Other boundary conditions are the turbine inlet temperature of about 800 °C and the pressure ratio maximum of $\pi = 3$ **[6].** Furthermore, the turbine exit temperature must be kept below 650 °C. Otherwise the recuperator (made of stainless steel) would be damaged (e.g. by corrosion and/or deposit), which would strongly reduce its life time **[11]**.

The simulation results of the 15 kW MGT are shown in **Figure 8**, **Figure 9** and **Figure 10.** They should be understood as best-case scenario (same component efficiencies as 30 kW_{el} MGT).

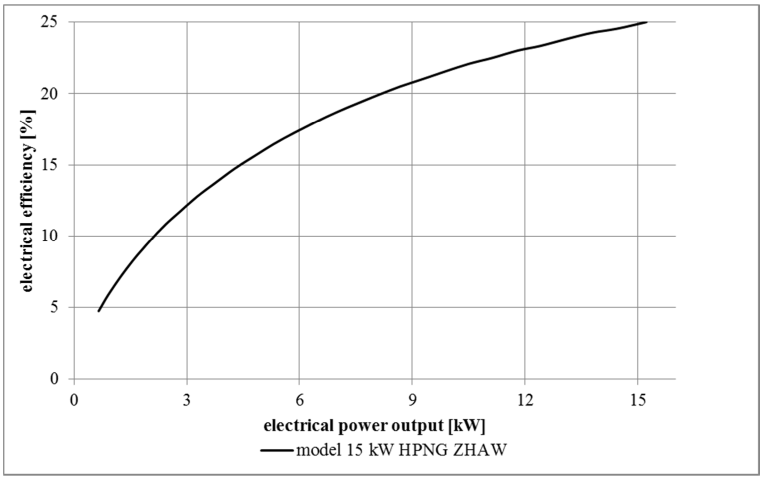

Figure 8: Electrical efficiency to electrical power output

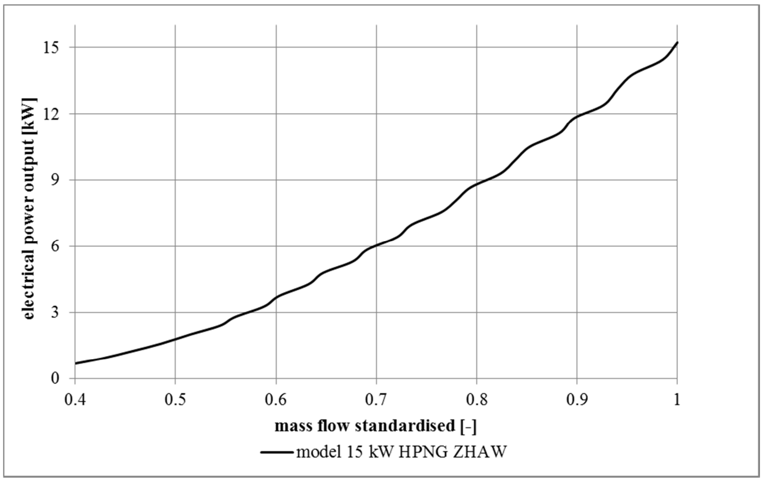

Figure 9: Electrical power output to mass flow

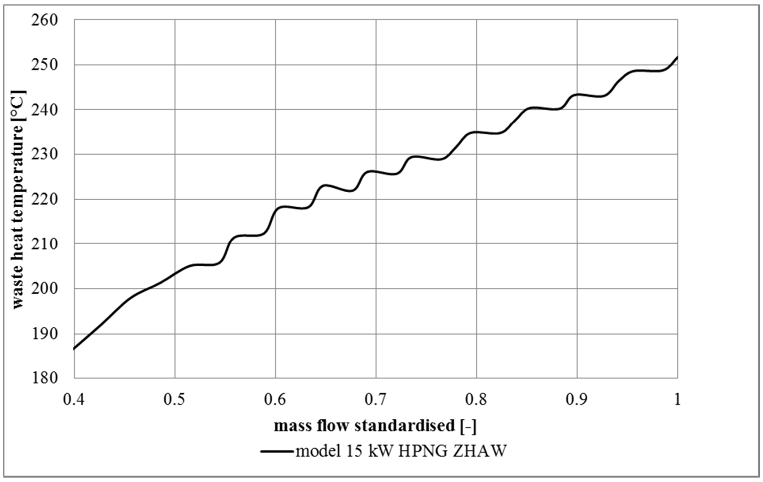

Figure 10: Waste heat temperature to mass flow

5 Conclusion

Based on existing performance data of a commercial 30 kW MGT, a thermodynamic model was developed. This model simulates the part load performance of a MGT system and its components by use of physical approaches. With these approaches a relative deviation maximum of 12 % (average 7 %) to the performance data was achieved.

The analysis of the existing performance data, the calculated efficiencies of the components, the scientific findings of their behaviour and their influence on part load operation describe the benchmark for state of the art systems. This best-case or benchmark – based on the market leader's know-how – can be found in the component's efficiency and has to be reached in prospective developments for smaller systems.

The turbine and compressor isentropic efficiencies as well as the pressure ratio maximum, mass flow and data of a dimensioned recuperator at the defined operating point are required input values for the simulation model. Furthermore, the air-fuel ratio has to be defined to create the part load behaviour. So, the developed simplified model may simulate the part load performance of a possible (future) MGT system. The generated simulation models can be used for the estimation of capabilities of this technology in various applications.

Another validation of the plausibility of the model results and its physical approaches has been done by simulating a 60 kW MGT system. Furthermore, a 60 kW upscale-model with extrapolated data of the air-fuel ratio of a 30 kW MGT was defined and reached similar quality in the results as the simulation of a 60 kW MGT with the known data for the air-fuel ratio which confirms the plausibility of the upscale-model. Furthermore it can be postulated that the simulation would generate similar technical reasonable results for the opposite case of a theoretical downscale model with the estimated benchmark efficiencies for the components.

Therefore a 15 kW MGT model was developed, which has to be named benchmark or best-case model, i.e. the part load performance is similar to that of the existing 30 and 60 kW MGT systems and the dimensioned components have the same efficiencies. This model meets the requirements to estimate the capabilities of MGT technology for local energy supply of residential buildings. Furthermore, future MGT with power output of 15 kW or less have to reach this benchmark performance to become competitive. Above all, the cost-efficient components like the compressor as well as the turbine need high efficiencies. Otherwise, to reach electrical system efficiency of about 25 % a higher temperature at the recuperator exit on the air side would be needed which leads to higher system temperatures in the combustion chamber and results in a higher turbine inlet temperature as well. Higher emissions (especially NO_x) would result. Furthermore the components would be negative affected by the high temperatures. To achieve a higher air temperature at the recuperator exit a larger recuperator would be necessary. Since the recuperator is very cost-intensive the price competitiveness is hardly given. A high electrical efficiency and low manufacturing cost of the components are essential for the competitiveness **[12]**. These facts have to be kept in mind, because energy efficiency can always be reached, but environmental impact and price competitiveness may not be forgotten. Because of these reasons, the described downscale-model is based on the introduced benchmark of the commercial 30 kW MGT system.

Finally, the estimated simulation model represents the prospective capability of MGT systems with power output less than 15 kW, which could be reached with the market leader's know-how and highly efficient components. The theoretical *15 kW MGT HPNG ZHAW* model was designed for an electrical power output of 15.2 kW and has an electrical efficiency of 25 %. The waste heat temperature maximum is 252 °C.

6 Acknowledgement

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