The introduction of SOFC-GT systems as a standard power generation system finally depends on the system's cost. This cost is influenced by the process design and the hardware design as well. The heat exchanger cost can be influenced by a reduction of the excess air and a pressurisation of both sides of the air heater and the fuel heater. The SOFC-GT cycle with an external cooling gives the opportunity to realise these demands. The cost reduction potential of the air heater is estimated to be > 50% compared with other SOFC-GT designs. The design of the SOFC stack itself and its integration in the pressure vessel influence the power density of the pressure vessel. The connection between stack design and pressure vessel design can be described with a simple geometric design model. It can be shown that a small tube diameter and a hexagonal spacing deliver the highest power density. The power output of the unit influences directly the consumption of the insulation material and thus of the wall material of the pressure vessel as well. This influence can be easily measured by the power related cross-section of the insulation and the wall of the vessel. Small tube diameters and a hexagonal spacing show the lowest possible material consumption and thus the lowest possible cost. It seems to be very useful to analyse the design of small SOFC tubes more deeply to reach real cost benefits.

INTRODUCTION

The introduction of SOFC-GT systems as a standard power generation system finally depends on the system's cost. The actual cost studies of mature systems (1,2) lead to specific cost of about 1300 US$/kW for the market entrance. This cost will automatically decrease if the production quantity will increase. But it is important to analyse the influences on the cost efficiency of the design of SOFC-GT units for a necessary cost reduction to increase the value to the user and to strengthen the position of the SOFC-GT technology in the emerging markets of distributed generation. Fig.1 shows the main influences and their interactions.
The process design is mainly based on the thermodynamics but the cost is influenced by the demands of the hardware design and of the production as well and last but not least of the market that will guarantee the produced quantity. This paper will be concentrated on the effects of the process design and of the hardware design.

**INFLUENCES OF THE PROCESS DESIGN**

The reversible combined fuel cell - heat cycle is the exact thermodynamic reference cycle of any combined fuel cell - heat cycle (3). This cycle can be simplified for practical use as fig. 2 shows (4). The reversible fuel cell operates isothermally and is the heat source of any reversible heat cycle with a heat sink at the ambient state. Air and fuel are heated up by the cooling of the flue gas in an ideal heat exchanger system. This model can be extended to describe real cycles including the necessary fuel preparation of hydrocarbons (5). The results of calculations using the model indicate that combined cycles with SOFC can reach electric efficiencies of more than 80% related on LHV.

![Figure 2](image.png)

The simplified reference cycle and the main topics of the technical realisation of the model

The system cost is strongly influenced by the choice of the heat engine cycle and its connection with the heat exchanger system of the SOFC. An indication of a good integration is the value of the excess air $\lambda$ needed for the SOFC cooling. The influences of the excess air on the quality of the process design can be seen in fig. 3.
The excess air influences the size of the air heater, the quantity of the waste heat, that can be directly extracted from the cooling of the SOFC stack for further use, the flue gas loss of the system and the thermodynamic potential of work expressed by the Nernst voltage. The size and thus the cost of the air heater and the flue gas loss are directly influenced by the increasing excess air. Increasing values of the excess air greater than 2 don't really influence the Nernst voltage but they have an increasing negative influence on the other values. Thus an excess air greater than 2 should be avoided.

These influences of the excess air are a direct consequence of the quality of the approximation of the reference cycle. It shows that the excess air $\lambda$ should be only used to assure a good combustion in the SOFC and not to cool the SOFC. The isothermal SOFC operation with a direct use of the waste heat in a connected heat cycle can be approached e.g. by a gas turbine process. One possibility is to cool the SOFC stack by an integrated cooling loop and the coolant is the operating fluid of the connected heat cycle. Alternatively the SOFC stack can be divided in sub-stacks and the oxygen rich flue gas is cooled by a connected heat engine and is fed to the cathode of the next sub-stack again (6).
Fig. 4 shows an example of a SOFC - GT cycle with an external cooling (6,7). The incoming air and the incoming fuel are heated by the flue gas of the SOFC as shown by the reference cycle in fig. 2. The connected heat cycle is a GT cycle. The cooled flue gas is reheated by the integrated coolers of the SOFC and thereafter it is expanded in the gas turbine. The integrated reformer is used as an additional cell cooling. A combination with a steam turbine cycle is a further possible improvement for stationary applications. The depleted anode gas flow can be recycled as a steam source for the reforming. These process reaches an efficiency of more than 70 %. It can be combined with a steam turbine process to deliver an efficiency of about 76 %. An efficiency of about 80 % may be obtained in combination with a reheat gas turbine process.

The isothermal combustion can be approximated by a multi-stage reheat gas turbine cycle. A generalised design of the SOFC-GT cycle with an intermediate expansion is shown in fig. 5. The first sub-stack is behind the compressor and the air heater in the flow direction. The burner of the depleted fuel is placed behind the sub-stack and before the first gas turbine expanding the flue gas to the inlet temperature of the next sub-stack. The oxygen rich flue gas of the first gas turbine is used as a cathode gas of the next sub-stack and the depleted cathode gas and the depleted fuel is burnt in the second burner and expanded in the second gas turbine etc.. These steps will be repeated until the designed oxygen content at the end of the last stage is reached. Than the flue gas is expanded to the designed exhaust temperature of the final gas turbine. The expanded hot flue gas is used to heat the air flow behind the compressor and the fuel flows. The sub-stacks can be cooled additionally by the integrated reformer. The depleted anode gas flow can be recycled as a steam source for the reforming. A cycle with two sub-stacks reaches an efficiency of about 70 %, but a steam cycle cannot be added anymore.

The SOFC-GT cycle with the external cooling has roughly the same pressure between the compressor outlet and the gas turbine inlet. Thus both sides of the heat exchangers operate roughly at the pressure of the compressor outlet. The occurring small pressure difference is caused by the pressure losses in the stack and the heat exchangers only. The SOFC - GT cycle with the intermediate expansion has the maximum possible
pressure difference (the pressure difference of the compressor) between the both sides of the heat exchanger surfaces of the air heater. Thus one side of the heat exchanger operates with the maximum pressure and the other side operates with the ambient pressure.

The influence of the pressure distribution in a heat exchanger system on the heat transfer coefficient shows an example in fig. 6. The ratio $U(p)/U(1)$ of the heat transfer coefficient of the pressurised system $U(p)$ and the heat transfer coefficient of an ambient system $U(1)$ as a reference is plotted against the (maximum) system pressure.

![Figure 6. An example of the pressure influence on the heat exchanger design](image)

The system pressure and its distribution influence the Reynolds numbers and therefore the Nusselt numbers on both sides and the heat transfer coefficients $U$ of the heat exchangers - at a certain geometry and at certain thermodynamic and flow conditions. A heat exchanger system pressurised at one side only shows an increase of the ratio $U(p)/U(1)$ to about 2.5 at a pressure of 16 bar. A system pressurised at both sides with the same pressure shows an increase of the ratio $U(p)/U(1)$ to more than 7 at 16 bar. The choice of the material of the heat exchanger of the SOFC-GT cycle with an external cooling is facilitated by a disappearing pressure difference over the walls. The SOFC-GT cycle with an external cooling needs an extra internal cooler but the SOFC module can be located in only one pressure vessel to reach efficiencies over 70%. Efficiencies of about 80% can be reached by a reheat version and the excess air can be chosen as low as possible for the SOFC. Thus we can carefully estimate that the pressurisation of the heat exchanger system on both sides will deliver a reduction of its cost of more than 50% (SOFC-GT cycle with an external cooling).

INFLUENCES OF THE HARDWARE DESIGN

The hardware design follows the process design and we see that the geometry of the SOFC stack strongly influences the power density and thus the cost of the insulation and the pressure vessel. The geometric relations are comparable simple but their influence is important and should be considered. The tubular design has some benefits compared with a planar design as already shown (8). Thus only tubular designs are considered. For the simplicity of the description let us assume that all SOFC tubes are parallel and the insulation...
and the pressure vessel can be assumed as a big surrounding tube. We can discuss this topic as a pure geometric problem, if we assume that the SOFC surface related power density \( U_i \) is constant and independent of the geometry. With these assumptions we can reduce the description of the power density and the material consumption of the insulation and of the walls of the pressure vessel to a pure geometric problem. The main parameters are:

- the diameter \( d \) of the SOFC tube,
- the distance \( s \) between two tubes and
- the type of tube spacing (hexagonal, quadratic).

Fig. 7 shows the hexagonal and the quadratic spacing of the SOFC tubes and their influence on the distribution of the modules in the pressure vessel.

The cell voltage \( U \) was assumed to be 0.7 V and the current density \( i \) was assumed to be 200 mA/cm\(^2\) for the calculations. The inner diameter of the insulation was assumed to be 1.05 \( D_i \) (\( \Delta \)). It was assumed in all cases that the accepted heat loss is 1 % of the produced power of the SOFCs in the control volume. The resulting outer diameter of the insulation was assumed to be the inner diameter of the wall of the vessel. The diameter \( D_i \) (\( \Delta \)) was varied between 1.0 m and 2.4 m. The tube diameter \( d \) was varied from 2 mm to 22 mm and the distance \( s \) was assumed to be 1 mm in all cases. An increase of \( s \) up to 2 mm did not change the quality of the results. The results may change in the case of integrating reformers or heat exchangers in the SOFC module but the tendencies will remain because this equipment will be integrated in the similar geometry. The target of this analysis is to get relations between different geometry's and not the exact value of any actual design. Thus the pressure vessel and the insulation as well are assumed to be a tube element surrounding the SOFC module.
Figure 8. The influence of the SOFC tube diameter $d$ and the tube spacing on the generated power of the SOFC module

Fig. 8 shows the influence of the SOFC tube diameter $d$ and the tube spacing on the generated power of a SOFC module with a defined geometry in a tube element with a length of 1 m. An increase of the diameter $D_i$ from 1,0 to 2,4 increases the power output by the factor 6 for both types of spacing. The quadratic spacing will deliver an output of about 2/3 of the hexagonal spacing in the same volume. An increase of the SOFC tube diameter $d$ from 2 mm to 22 mm decreases the output by about 5. The minimum value in the figure is 142 kW/m ($D_i = 1m$) and the maximum value is 4777 kW/m ($D_i = 2,4 m$).

The same parameters of the geometry influence the material consumption of the insulation and of the wall of the pressure vessel. The material consumption must be related on the power output of the surrounded SOFC module element to get comparable figures. The material consumption of the insulation is proportional to the cross-section of the insulation and the material consumption of the wall of the vessel is proportional to its cross-section.

Fig. 9 shows the influence of the tube diameter $d$ and the tube spacing on the material consumption of the insulation described by the power related cross-section of the insulation. The inner temperature is 1000 °C and the outer temperature is 70 °C. An increase of the diameter $D_i$ from 1,0 m to 2,4 m decreases the power related cross-section by the factor 7 to 50 depending on the tube diameter $d$ and the type of spacing. The quadratic spacing delivers a clearly larger power related cross-section than the hexagonal spacing in the same volume. The values are about 2,5 times higher for a tube diameter of 2 mm and up to 6,5 times higher for a tube diameter of 22 mm. The influence of the spacing increases with the tube diameter. An increase of the SOFC diameter $d$ from 2 mm to 22 mm increases the power related cross-section of the insulation by a factor between about 30 and 230 depending on the diameter $D_i$ and the spacing. The minimum value in the figure is 0,94 cm²/kW ($D_i = 2,4m$) and the maximum value is 4282 cm²/kW ($D_i = 1,0 m$).

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The calculation of the insulation delivers the inner diameter of the pressure vessel. The necessary wall thickness can be calculated and we receive the power related cross-section of the wall as a measure for the material consumption.

Figure 9. The influence of the tube diameter and the tube spacing on the material consumption of the insulation

Figure 10. The influence of the tube diameter and the tube spacing on the material consumption of the wall of the pressure vessel
Fig. 10 shows the results of the chosen geometry. The calculation is based on the results presented in fig. 9. The increase of the diameter $D_i(\Delta)$ from 1.0 m to 2.4 m decreases the power related cross-section by the factor 1.5 to 24 depending on the tube diameter $d$ and the type of spacing. The quadratic spacing delivers a clearly larger power related cross-section than the hexagonal spacing in the same volume. The values are about 1.6 to 2 times higher for a tube diameter of 2 mm and up to 2 to 6 times higher for a tube diameter of 22 mm. Thus the influence of the spacing increases with the tube diameter again. An increase of the SOFC diameter $d$ from 2 mm to 22 mm increases the power related cross-section of the wall by a factor between about 7 and 120 depending on the diameter $D_i$ and the spacing. The minimum value in the figure is 0.38 cm$^2$/kW ($D_i = 2.4$ m) and the maximum value is 133 cm$^2$/kW ($D_i = 1.0$ m).

CONCLUSIONS

These results show that a small SOFC tube diameter and a hexagonal spacing deliver the highest power density. The benefit of a high power density is that the material consumption for the insulation and the wall of the pressure vessel is low. These influences are very clear and we see a very high potential of cost reduction by reducing the tube diameter as far as possible. Additionally there are some more benefits of the use of very small tube diameters in a SOFC stack. Fig. 11 shows it. The use of a small tube allows to increase the allowable velocity of temperature changes drastically, as shown in (9). Today we operate with an allowable velocity of about 200 K/h in stationary systems. In the case of a tube diameter of 2 mm we could reach an allowable velocity of 200 K/min, thus 60 times higher than today. The high velocity of temperature changes is important because it determines the start up time of any SOFC application. There may be a chance to introduce SOFC systems in mobile applications, if we use small SOFC tubes and an intermediate-temperature SOFC electrolyte. The needed power capacity per year of the automotive industry is much higher than the needed power capacity per year of the stationary applications. Thus there is an additional option of a cost reduction by increasing the production quantity.

![Figure 11](image)

**Figure 11.** Benefits and necessary R&D of small tubular SOFC stacks

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But there are good reasons, why the SOFC diameter of the most actual developments is usually much bigger than 2 mm. It is important to have a sufficient length for a sufficient mass transport within the SOFC. The design of a sufficient length demands a minimum diameter d to reach a sufficient mechanic stability. But there are indications that solutions are possible to realise a stack design with smaller tubes that fulfill these process requirements. Necessary steps for a realisation are e.g. a relevant theoretical model of the tubular design and flow experiments including a further design analysis. This work will be integrated in a further development of the process design.

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