EXPERIMENTALLY VALIDATED IMPROVEMENTS OF A CATHODE AIR COMPRESSOR FOR PEM FUEL CELLS BY FIXED DIFFUSER VANES

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ABSTRACT
This study presents first-time experimental investigations of an electric turbocharger for the fuel cell air supply with different range-extending features in the centrifugal compressor. Experiments show the potential of pivoting the diffuser vanes: The surge margin of the compressor increases by up to 44.7 percentage points. This can be used to adapt the compressor map to the operating range requirements of an automotive fuel cell. However, due to efficiency losses, the compressor power consumption increases when the fuel cell is operated at full-load.

Further experiments with different leading edge angles of the diffuser vanes demonstrate that the map of the centrifugal compressor can be shifted towards lower mass flow rates even by a fixed-geometry diffuser. While the original compressor can only cover 68 percent of the total fuel cell’s operating range with a surge margin of 20 percent, a compressor with adapted leading edge angles of the diffuser vanes covers over 75.7 percent. By linearly reducing the leading edge angle over the vane height, the compressor power consumption decreases over the entire operating range of the fuel cell. A reduced power consumption of up to 6.3 percent is achieved at part-load.

KEYWORDS
FUEL CELL AIR SUPPLY, CENTRIFUGAL COMPRESSOR, OPERATING RANGE EXTENSION, VANED DIFFUSER

NOMENCLATURE
FC Fuel cell GCI Grid convergence index
LE Leading edge OP Operating point
PEM Proton Exchange Membrane PV Pivoting vanes
SL Surge line A Volute area
\( a_{OH^-} \) Activity of \( OH^- \) \( c_{rad,2} \) Radial flow velocity at diffuser inlet
\( E \) Electrode potential \( F \) Faraday constant
\( h_{rel} \) Relative vane height \( i_D \) Diffuser incidence
\( m_{corr} \) Corrected mass flow rate \( n_{corr} \) Corrected speed
\( p_{O_2} \) Oxygen partial pressure \( p^0 \) Standard pressure
\( P_{DC} \) Direct current power \( R \) Universal gas constant
\( SM \) Surge margin \( T \) Temperature
\( \beta_D \) Diffuser vane angle \( \theta \) Circumferential angle
\( \Pi_t \) Total pressure ratio \( \rho_2 \) Density at diffuser inlet
INTRODUCTION

Proton Exchange Membrane (PEM, also called Polymer Electrolyte Membrane) fuel cells are usually used for fuel cell vehicles in the automotive sector. The Nernst equation for the cathode side

\[ E = E^0 - \frac{RT}{F} \ln \left( \frac{a_{OH}}{p_{O_2}/p^0} \right), \]

which estimates the power density of a PEM fuel cell shows that the electrode potential \( E \) and, thus, the electrical output power of the fuel cell increases with the oxygen partial pressure \( p_{O_2} \) (Kurzweil, 2013). Berning and Djilali (2003) explained that the improvement in power density with the oxygen partial pressure is less pronounced above operating pressures of 3 bar. For this reason, typical operating pressures in the range between 2 ... 3 bar are used for PEM fuel cells in the automotive sector. Therefore, fuel cell vehicles are equipped with a centrifugal compressor to achieve these pressure levels. The compressor is the largest parasitic power consumer in the entire fuel cell system. Nevertheless, the efficiency increase of the fuel cell overcompensates the power demand of the compressor. Using a fuel cell system model, Hoeflinger and Hofmann (2020) showed that the highest system efficiency can be achieved with high operating pressures at full-load and lower operating pressures at part-load. The reason for this behaviour is found in the operating characteristics of the fuel cell. The fuel cell’s polarization curves show the dependence of the stack voltage on the current density. At high loads (and high current densities), the voltage can be increased by 20 percent by raising the operating pressure up to 3 bar. In comparison, the voltage increase at part-load is significantly lower, so that the high operating pressure is not rewarding.

At part-load the fuel cell requires minimal air mass flow rates together with relatively small pressure ratios (Hoeflinger and Hofmann, 2020). Due to these low air mass flow rates and the limited operating range of centrifugal compressors, the operating line of the specific fuel cell investigated in this project cannot be completely covered (Schoedel et al., 2021a, 2021b).

In summary, there are two main requirements for the centrifugal compressor of a fuel cell air supply: The highest possible compressor efficiency in the range of the fuel cell operating line and a sufficient operating range to cover all operating points. Usually centrifugal compressors with a vaned diffuser are used for the fuel cell air supply since they achieve higher efficiencies than vaneless diffusers. However, their operating range is smaller (Galloway et al., 2018), reducing the operational flexibility of the fuel cell (Lueck et al., 2020). In order to supply the fuel cell with sufficient air at any operating point and keep the compressor in a stable range even during transient operations, it is necessary to bypass some fraction of the compressor air at part-load. This bypassed air mass flow does not contribute to the power generation of the fuel cell, increasing parasitic power consumption and lowering the system’s efficiency. Consequently, the compressor map must be adjusted to the fuel cell operating line of a specific application to optimize the entire system’s efficiency.

This study aims to experimentally investigate the influence of different fixed-geometry modifications of an existing centrifugal compressor in order to improve the performance of the compressor considering a specific fuel cell operating line. Furthermore, the potential of a variable-geometry diffuser is analyzed. Schoedel et al. (2021a, 2021b) have investigated the mentioned geometry modifications numerically.
Investigated Compressor and Geometries

As mentioned above, the investigations are based on an existing electric turbocharger, provided by FISCHER Fuel Cell Compressor AG. Table 1 summarizes some basic information about the centrifugal compressor of the original FISCHER turbocharger.

Table 1: Basic information about the original FISCHER compressor (FISCHER FCC AG, 2022)

<p>| | | | |</p>
<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>Maximum compressor speed</td>
<td>120,000 min⁻¹</td>
<td>Maximum mass flow rate</td>
<td>158 g/s</td>
</tr>
<tr>
<td>Maximum total pressure ratio</td>
<td>3.1</td>
<td>Diffuser geometry</td>
<td>Vaned</td>
</tr>
</tbody>
</table>

Figure 1 shows an overview of the modified geometries investigated in this paper’s scope: A variable-geometry diffuser with a volute redesign and a modified fixed-geometry diffuser. The variable-geometry diffuser describes a diffuser with pivoting vanes. To become rotatable, the vanes are shortened compared to the baseline design. In future applications the pivot angle $\Delta \beta_{D, PV}$ can be adapted according to the fuel cell operating point. However, since the implementation of a movable system is not possible for the experiments of this study, the investigations are conducted with static inserts. Those have different fixed pivot angles between $-6^\circ$ and $+2^\circ$ (see Figure 1a). Gaps between the diffuser vanes and end walls are neglected. Negative pivot angles, which achieve a shift of the compressor map towards lower mass flow rates, should improve the compressor performance at least in the part-load range of the fuel cell operating line. In order to consider the smaller diffuser outflow angles caused by negative pivot angles at part-load, a volute with a linear area progression and a smaller outlet area compared to the baseline volute is designed (see Figure 1c). All the following investigations with the variable-geometry diffuser are carried out with the redesigned volute.

For further experiments, only the leading edge angle of the diffuser vanes is modified, maintaining the trailing edge angle (see Figure 1b). This case is referred to as a ”modified fixed-geometry diffuser” (Schoedel et al., 2021a, 2021b).
**CFD Setup**

This chapter describes the numerical setup used for the investigations that is shown schematically in Figure 2. By modelling the domains diffuser, volute, and outlet pipe (not shown in Figure 2) over the entire circumference, the circumferential asymmetry of the volute is considered. All other domains are modelled as single passages with periodic boundaries. Mixing plane interfaces are used between the rotating impeller domain and the inlet pipe as well as the diffuser. At the compressor inlet, averaged values of total pressure, and total temperature are specified for each speed line to match the experimental boundary conditions. The mass flow at the compressor outlet is varied. For the meshing of the fluid spaces, the software ICEM CFD 19.2 (inlet pipe and outlet pipe structured, volute unstructured) and ANSYS Turbogrid 2020 R1 (impeller and diffuser) are used. The cell closest to the wall is dimensioned such that maximum $y^+$ values smaller than 2 result at the compressor operating point with the maximal investigated mass flow rate. An exception are the impeller blades, at whose trailing edge only significantly higher $y^+$ values of up to 60 can be achieved due to the meshing topology.

Furthermore, a mesh convergence study is carried out with three mesh densities, according to the American Society of Mechanical Engineers (2009). For the used medium mesh density with a total of approximately 17.3 million cells, a grid convergence index ($GCI$) of 0.030% results for the total pressure ratio at a selected operating point at a speed of $0.91 \cdot n_{cor,max}$ (see Figure 4). The $GCI$ of the polytropic efficiency at the same operating point is 0.089%. Since the Menter $k-\omega$ SST model, frequently used in the field of turbomachinery, usually overestimates flow separations, the simulations are carried out with the Menter baseline model (Menter, 1993). Finally, the following criteria are used to monitor the stability and convergence of individual operating points: RMS residuals smaller than $10^{-4}$, imbalances between inlet and outlet of the CFD domains smaller than 0.1%, and a negative slope of the total pressure ratio plotted against the mass flow rate at a constant speed line.

![Numerical setup of the compressor simulations](ARIEL, 2022)

**Experimental Setup**

A special test setup is developed for the safe operation of the electric turbocharger and to record all relevant thermodynamic values, illustrated in Figure 3. The turbocharger and the other main test components are controlled via the control PC and the test bench automation, which also collects the pressure, temperature, volume flow, and speed data. At the inlet of the air path (1), the volume flow, the static inlet temperature, and pressure of the ambient air are measured. This is followed by measuring the static temperature and pressure at the compressor (2) outlet. Furthermore, a compressor back pressure unit (3), consisting of three different throttle valves, is used to control the compressor mass flow rate. As a next step, the pressurized air enters a
Table 2: Measurement technology for the experimental investigations of the compressor

<table>
<thead>
<tr>
<th>Measured variable</th>
<th>Sensor</th>
<th>Measurement range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Static pressure $\Delta p$ (relative to environment)</td>
<td>DMT Pressure Scanner 9116</td>
<td>Inlet: 0 ... 103.4 kPa Outlet: 0 ... 310.3 kPa</td>
<td>+/- 0.05 kPa +/- 0.16 kPa</td>
</tr>
<tr>
<td>Static temperature $T$</td>
<td>Pt100</td>
<td>-100 ... +450$^\circ$C</td>
<td>+/- (0.04$^\circ$C + 0.2% $\cdot T$)</td>
</tr>
<tr>
<td>Compressor speed $n$</td>
<td>turboSpeed DZ135</td>
<td>200 ... 200,000 min$^{-1}$</td>
<td>+/- 400 min$^{-1}$</td>
</tr>
<tr>
<td>Volume flow $\dot{V}$</td>
<td>FlowSic600</td>
<td>15 ... 840 m$^3$/h</td>
<td>+/- 1.0% $\cdot \dot{V}$</td>
</tr>
<tr>
<td>Voltage $U$</td>
<td>LMG671</td>
<td>0 ... 600 V</td>
<td>+/- (0.48 V + 0.02% $\cdot U$)</td>
</tr>
<tr>
<td>Current $I$</td>
<td>Power Analyzer</td>
<td>0 ... 75 A</td>
<td>+/- (0.06 A + 0.03% $\cdot I$)</td>
</tr>
</tbody>
</table>

water-cooled heat exchanger (4), to adjust the turbine inlet temperature to approximately 80$^\circ$C, which is a typical value in fuel cell applications. At the inlet and the outlet of the turbine (5), static temperatures and pressures are measured again before the air exits the test bench to the environment (6). The measuring tubes containing the static temperature and pressure probes upstream and downstream of the compressor and turbine are manufactured very precisely, so that the flow area is known very exactly. This means that the flow velocity and thus the total pressure and temperature can be calculated from the measured static values and the volume flow.
The electric turbocharger is directly connected to the power electronics, which is connected to a DC power source. In order to record the electric performance data of the turbocharger very precisely, current and voltage are tapped (between the DC source and the power electronics as well as between the power electronics and the compressor) via breakout boxes and measured with the help of a ZES Zimmer LMG671 power analyzer. Table 2 gives an overview of the different measurement technologies.

The geometry modifications investigated in this paper influence the axial load of the compressor. For safety reasons, however, the maximum axial load of the original Fischer compressor (see Table 1) should not be exceeded. Therefore, the experimental investigations are carried out with a maximum compressor speed, which is slightly below the maximum speed of the original compressor. During the experiments, the compressor is throttled at a constant corrected speed until the fluctuations of outlet pressure (surge detection of the test bench automation) or the speed reach predefined threshold values. The operating points with the minimal achievable mass flows represent the experimental surge line.

Results

The following section, which is based on the final report of the project ARIEL (ARIEL project consortium, 2022; in German), describes the results of the experimental investigations of the baseline compressor, the compressor with a variable-geometry diffuser, and the compressor with a modified fixed-geometry diffuser. For the baseline compressor experimental results are also compared with CFD results. The total pressure ratios and mass flow rates in the compressor maps and other diagrams below are normalized relative to the maximum operating point of the fuel cell. Corrected mass flow rates and speeds are calculated according to the turbocharger standard of the SAE J922 (1995). Furthermore, the surge margin

\[
SM(n_{corr} = const) = \frac{\Pi_{t,SL} \cdot m_{corr,OP}}{\Pi_{t,OP} \cdot m_{corr,SL}} - 1
\]

is used to evaluate the distance between an operating point and the surge line at the same corrected speed. For safety reasons and to ensure that transient speed changes are possible, a minimum surge margin of 20% is required over the entire operating range of the fuel cell.

Baseline Compressor

As shown in Figure 4, there are slight deviations between the experimental and CFD compressor map of the baseline compressor. In the high-speed range, the CFD overestimates the total pressure ratio by up to 1.4%. The deviations can be explained by some simplifications of the CFD model: Firstly, the impeller-diffuser interaction is not fully resolved due to the mixing plane interfaces. In addition, the fillet radii at the blade root are removed, and the leakage into the turbocharger motor section is neglected. Finally, there is uncertainty regarding the axial impeller tip gap of the CFD simulations due to the axial bearing clearance and measurement uncertainties during the gap adjustment for the experiments.

The comparison of the experimental and numerical map in Figure 4 clearly shows that the numerical stability limit is usually at higher mass flow rates than the experimental surge line. This is a typical behaviour, as steady-state CFD simulations often cannot achieve convergence near the surge line due to unsteady flow phenomena. An exception is the upper speed line, where the experimental surge line is at higher mass flow rates. It is assumed that the implemented surge detection mechanism of the test bench detects a pre-surge event due flow instabilities in...
the diffuser and the corresponding strong pressure fluctuations. Also the transition to transonic impeller inlet conditions is a reasonable explanation, as shown by Lou et al. (2022).

Figure 4 illustrates that the experimental and numerical compressor map, cannot completely cover the required fuel cell operating line in the low-speed range. The required minimum surge margin of 20% is only reached for normalized corrected mass flow rates (below referred to as mass flow rates) larger than 32% in the experiment, and 59% in the CFD. Thus, some fraction of the compressor air must bypass the fuel cell at the operating points with lower mass flow rates. Consequently, the compressor can maintain a sufficient surge margin and still supply the required mass flow rate or total pressure to the fuel cell. As mentioned previously, the bypassed air at part-load does not contribute to the power generation of the fuel cell, thus, increasing the parasitic power consumption of the compressor.

The CFD simulations allow an analysis of the diffuser inflow near the numerical stability limit and, thus, also close to the experimental surge line. Using Figure 5, both the radial mass flow density at the diffuser inlet and the incidence of the diffuser are evaluated near the numerical stability limit at a speed of $0.55 \cdot n_{corr,max}$. Obviously, a backflow region exists upstream of the diffuser above a relative vane height of 80%. This region is also responsible for the positive diffuser incidence above about 71% of the relative vane height. Below 71% of the relative vane height, there is a negative incidence. A positive diffuser incidence leads to a pressure-sided inflow, which is considered particularly critical concerning the stability of the compressor and should be avoided. In contrast, a negative diffuser incidence results in a reduced compressor efficiency.
Variable-Geometry Diffuser

The experimental investigations show that the compressor with a redesigned volute achieves only minor changes in the operating range of the compressor (not illustrated here). Although the volute redesign leads to a minimal improvement of the polytropic efficiency almost over the entire map, there are also regions with a slight efficiency decrease, for example at a speed of $0.55 \cdot n_{corr,\text{max}}$ near the surge line. Evaluating the polytropic efficiency along the fuel cell operating line, one obtains a maximum increase of up to 0.7 percentage points at a mass flow of about 59.6% and a decrease in efficiency of up to 0.4 percentage points at a mass flow of about 30.0%.

Figure 6 compares the compressor maps for different pivot angles of the diffuser vanes (without considering the gaps between diffuser vanes and end walls). Large pivot angles, such as $\Delta \beta_{D,PV} = -6^\circ$, lead to a significant shift of the surge line towards lower mass flow rates. The surge margin increases by up to 44.7%. Consequently, the operating points of the fuel cell can be covered for mass flow rates larger than 18.4% (with a minimum surge margin of 20%). However, it is observed that for all investigated pivot angles (including the $0^\circ$-position, which is not shown here), there is a decrease in polytropic compressor efficiency almost over the entire fuel cell operating line. It should be mentioned again that the diffuser gaps are not considered in the experiment. According to the CFD predictions of Schoedel et al. (2021b), the diffuser gaps cause an additional efficiency deficit.

A final evaluation of the potential of the variable-geometry diffuser is possible by comparing the change in DC power of the entire turbocharger relative to the DC power of the baseline turbocharger for different pivot angles (see Figure 7). The DC power in the speed range smaller than $0.55 \cdot n_{corr,\text{max}}$ is interpolated from the map data for this comparison. Furthermore, the use of the bypass is assumed for fuel cell operating points with a nominal surge margin smaller than 20%. Thus, the compressor operating point is adjusted so that the required surge margin of 20% is maintained over the entire operating range of the fuel cell. In order to improve readability,
Figure 6: Experimental compressor maps of the variable-geometry diffuser and fuel cell operating line: (a) overview of the maps for different pivot angles of the diffuser vanes and baseline, (b) comparison between baseline and $\Delta \beta_{D,PV} = +2^\circ$, (c) comparison between baseline and $\Delta \beta_{D,PV} = -2^\circ$, (d) comparison between baseline and $\Delta \beta_{D,PV} = -6^\circ$. (b-d) show the difference in polytropic efficiency $\Delta \eta_{pol} = \Delta \eta_{pol,Basis} - \Delta \eta_{pol,\text{Baseline}}$ relative to the baseline compressor, 95% confidence interval (ARIEL, 2022).
the error bars of the first five working points are hidden.

Figure 7 shows that the volute redesign achieves only a slight power reduction over the entire fuel cell operating line. In contrast, a positive or zero pivot angle leads to an increase in the compressor power demand over the entire operating range (e.g., 0.6% for $\Delta \beta_{D, PV} = 0^\circ$ at full-load). With negative pivot angles, the power demand can be decreased for small mass flow rates by up to 6.6% for $\Delta \beta_{D, PV} = -6^\circ$. However, for $\Delta \beta_{D, PV} = -6^\circ$, the power demand increases significantly by up to 11.8% at full-load. Since the variable-geometry diffuser leads to an increase in power demand over a large operating range and diffuser gaps have not even been considered within the experiments of this study, the overall potential of this system to improve the compressor performance is rated as rather low. In addition, a variable-geometry diffuser has a very complex design with many individual parts in real applications.

![Figure 7: Experimental results: Change of the DC power of the entire turbocharger relative to the DC power of the baseline turbocharger along the fuel cell operating line for the variable-geometry diffuser as a function of the fuel cell/compressor mass flow, 95% confidence interval (ARIEL, 2022)](image)

**Modified Fixed-Geometry Diffuser**

As shown by the map comparison in Figure 8, the modified fixed-geometry diffuser achieves only small shifts of the surge line compared to the variable-geometry diffuser (see above): The fuel cell operating line is covered for mass flow rates larger than 24.3% in the case of a linear reduction of the leading edge angle $\Delta \beta_{D, LE} = 0^\circ \ldots -4^\circ$ and for mass flow rates larger than 20.7% in the case of a constant reduction of the leading edge angle $\Delta \beta_{D, LE} = -4^\circ$ (with a minimum surge margin of 20%).

However, considering the efficiency in Figure 8, the main advantage of the modified fixed-geometry diffuser is obvious: The efficiency decrease over the fuel cell operating line is significantly lower compared to the variable-geometry diffuser. From the middle of the map towards the low-speed range, an efficiency increase of up to 2.8 percentage points can be achieved compared to the baseline compressor. Furthermore, $\Delta \beta_{D, LE} = 0^\circ \ldots -4^\circ$ offers an improvement or at least an achievement of the baseline efficiency in all fuel cell operating points, which are in the stable compressor operating range. Linearly reducing the leading edge angle of the diffuser
Figure 8: Experimental compressor maps for the modified fixed-geometry diffuser and fuel cell operating line: (a) overview of the maps for different leading edge angles of the diffuser vanes and baseline, (b) comparison between baseline and $\Delta \beta_{D,LE} = -2^\circ$, (c) comparison between baseline and $\Delta \beta_{D,LE} = -4^\circ$, (d) comparison between baseline and $\Delta \beta_{D,LE} = 0^\circ - 4^\circ$. (b-d) show the difference in polytropic efficiency $\Delta \eta_{pol} = \Delta \eta_{pol,\beta_{D,LE}} - \Delta \eta_{pol,Basis}$ relative to the baseline compressor, 95% confidence interval (ARIEL, 2022).
vanes has one advantage: At the hub, where the incidence of the baseline diffuser is already negative, the vane angle and, thus, the incidence are not changed. At the shroud, where incidence of the baseline diffuser is critical positive, the vane angle and, thus, the incidence are reduced. In this way, a shift of the surge line is achieved, but the negative influence on the efficiency in the lower part of the vanes is reduced compared to a diffuser with a constant leading edge angle reduction.

Figure 9 is used to evaluate the change in DC power of the entire turbocharger relative to the DC power of the baseline turbocharger for different leading edge angles of the diffuser vanes. On the one hand, a constant reduction of the leading edge angle of the diffuser vanes $\Delta \beta_{D,LE} = -4^\circ$ achieves the highest decrease in the compressor power demand of up to 9.2% at part-load. On the other hand, the power demand increases by 2.9% for $\Delta \beta_{D,LE} = -4^\circ$ at full-load. In contrast, a linear reduction of the leading edge angle of the diffuser vanes improves the compressor performance along the entire fuel cell operating line: The DC power demand decreases by 1.4 to 6.3%. In particular, the diffuser with linearly reduced vane leading edge angles is a promising concept for improving the performance of centrifugal compressors for the fuel cell air supply.

![Figure 9: Experimental results: Change of the DC power of the entire turbocharger relative to the DC power of the baseline turbocharger along the fuel cell operating line for the modified fixed-geometry diffuser as a function of the fuel cell/compressor mass flow, 95% confidence interval (ARIEL, 2022)](image)

CONCLUSIONS

Centrifugal compressors with vaned diffusers, commonly used to supply PEM fuel cells in automotive applications, are the largest parasitic power consumers of the overall fuel cell system. In addition, these centrifugal compressors limit the operating range of the fuel cell at part-load and may require a bypass to cover the entire fuel cell operating line. This significantly increases the compressor’s power demand at part-load. In order to minimize the operating range with bypass, it is essential that the compressor map is matched to the specific fuel cell operating line, depending on the application. Based on the first experimental investigations with an electrically driven turbocharger and a specified fuel cell operating line, various range-
extending features for the compressor were investigated during this study.

While the baseline compressor covers only 68% of the fuel cell operating line with a surge margin of 20%, a variable-geometry diffuser with pivoting vanes can shift the compressor map towards smaller mass flows and thus cover up to 81.6% of the fuel cell operating line. Accordingly, the required bypass operating range at part-load is reduced by 13.6%, resulting in a reduced power demand of the compressor. However, the pivoting vanes increase the power demand at the full-load operating point of the fuel cell.

By simply adjusting the leading edge angle of the (non-variable) diffuser vanes and especially by reducing the leading edge angle linearly over the vane height, the operating range of the compressor is shifted towards smaller mass flows. At the same time, the compressor efficiency is shown to be increased along the entire operating line of the fuel cell. Overall, the power demand of this improved compressor is reduced along the entire operating line of the fuel cell by between 1.4% and 6.3%. With a linear reduction of the leading edge angle of the diffuser vanes 75.7% of the fuel cell’s operating line is covered.

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