Context-Aware Gas Turbines for Adapting GT-PEMFC Hybrid Cycle to Vehicles Demands

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Abstract: This paper explains summary of a method to introduce context-awareness into gas turbine engines. One aspect of this concept is represented by a novel GT-PEMFC hybrid cycle, which potentially is able to operate in vehicles properly. Such hybrid cycle is modeled by computer in average-fidelity level and summary of results is presented.

Keyword: context-awareness, gas turbine, compressor, hybrid cycle, vehicle, computer modeling;

Nomenclatures

b	Axial blades width
C	Air Velocity
D	Diameter

m	Air Mass Flow Rate
n	Number of blades
N	Rotational Speed
P	Pressure
R	Universal gas constant
T	Temperature
U	Impeller tip speed
β	Back sweep angle at exit of impeller
γ	Specific heats ratio
η	Efficiency
Λ	Degree of reaction
ϕ	Flow coefficient
σ	Slip factor
ζ	Hub-tip ratio
Ψ	Temperature drop coefficient

Abbreviations

GT	Gas Turbine
HTS	High Temperature Superconductor
IT	Information Technology
PEMFC	Polymer Electrolyte Membrane Fuel Cell
s.f.c	Specific Fuel Consumption
SSL	Standard Sea Level
TIP	Turbine Inlet Pressure
TIT	Turbine Inlet Temperature

Units

C	Centigrade
Cm	Centimeter
K	Kelvin
Kg	Kilogram
kW	Kilowatt
mA	Milliampere
MW	Megawatt
Nm	Newton meter
s	second
V	Volt
W	Watt

Suffix

0	Stagnation State
1	Inlet to Compressor
2	Inlet to Combustor
3	Inlet to Turbine
4	Outlet from Turbine
a	Axial -and also- Ambient
C	Compressor
r	Passage average outlet velocity
R	Relative to design value
imp	Impeller
in	Inducer
T	Turbine

1. Introduction

Concept of “Context-Awareness” is result of direct interaction of IT and computer worlds with human. In fact an article posed this concept, which discusses the idea of orienting computer productions with different aspects of people behaviors **[1]**. Currently context-awareness has broad and profound meaning in the cyberspace. In this global network ability to adapt to user demands and different environmental elements is the crucial character without it this huge system is paralysis. In a word, context-awareness can be defined as adaptation ability to user demands and environmental elements.

Although entity of gas turbine engines completely differs from computers but concept of context-awareness has high potential to be utilized in these engines. Common operation of these systems urges adaptation to different environment and users demands, which shows this strong potential.

To tune each component function into environmental element and user demands, a context-aware system allows free access to all components. In fact, such a free access is the key to raise a device up to context-aware system. Accordingly, structure of computer technology has been developed to allow executive software to access freely and individually to all effective components from user opinion to hardware. In the contrary, conventional architecture of gas turbine engines creates limit area to access to components individually.
As a result, first step to apply context-awareness on these engines is to create suitable configuration for them allowing free access to all components individually. Such a configuration allows components functions to be adapted to user demands and environmental elements.

This paper briefly explains a method to heighten access ability in gas turbine engines carried out to raise them up to context-aware system. The main section of the paper deals with summary of a study of such an engine potentials to use in the vehicles.

2. Applying Context-Awareness on Gas Turbine Engines

Three major sections of gas turbine engines namely compressor, turbine and combustor are linked so strongly that independent access to their parts particularly in compressor and turbine is less than that should be. Although user demands are applied to the engine by throttle commands, yet adaptation ability to environmental elements does not exist practically. Such strong link allows control systems to prevent destructive operations rather adapt engine behaviors to environmental elements and user demands.

To heighten access ability, coherent consistency should replace this strong link. It means that all effective components of all sections should be capable to be regulated toward environmental elements and user demands independently and coherently. Individual operation of turbine in the framework of free power unit as well as combustor, which reflects user demands, are able to adapt to context-awareness, however, approaching gas turbine engine to desirable characters needs essential changes in power transmission and compressor operation.

2.1. Essential Changes

Traditionally, shaft has been known the most efficient way to transmit power from turbine to compressor. Although this device transmits power efficiently nevertheless strongly limits individual access to engine components, and then, narrows individual manageability of them.

As a result, shaft is not an efficient part for a context-aware system and accordingly a proper alternative should be introduced. This alternative should permit free access and management of sections individually.

This leads mind to use electric energy, which is capable to be transmitted from turbine to compressor sections under higher handling and controlling. This work also allows transmitting needful energy to each compressor stage individually, which needs converting ability in the stages independently.

As a result, each rotor in compressor should be an electric motor that seems infeasible especially in the multi-stage compressors. Electric motors are heavy and inefficient beside a simple shaft. In addition, fitting an electric motor capable to provide needful power into a stage disk seems impossible.

Yet, this view can be changed if available technologies to be considered pragmatically. Two major technologies promise producing such an electrical motor, one High Temperature Superconductors (HTS) and the other Carbon Nano Tubes (CNT). HTS rotating machines are currently being implemented, but different potentials of CNT such as very little and light electric motors are under development **[2]**.

HTS rotating machinery offers several advantages over conventional designs. HTS designs may be particularly compact and lightweight. For instance, American Superconductor’s 36.5-MW HTS ship propulsion motor has been designed to have 20% of the weight and volume of a corresponding conventional motor **[3]**.

In this study, such a gas turbine, in which compressor stages receive turbine power independently, is simulated by computer. In the modeled cycle an integrated management system that has access to all effective components individually is considered, so that engine components can be adapted to environmental elements and user demands. Such gas turbines potentially are able to create suitable engine for vehicles when they are used in the hybrid cycle under integrated management beside an independent backup energy supplier for compressor.

This work is another aspect of context-aware compressors potentials that author similarly showed for axial flow compressors. Summary of that study was published in **[4]**.

3. Gas Turbine in the Vehicles

To observe background of gas turbines in the vehicles, experiences of Chrysler can be helpful. This car producer has had long experiences to adapt this type of engines to the cars from 1940s to early 1980s. An article published by Chrysler in 1964 notices several advantages of gas turbine over conventional reciprocating engines **[5].** However, they faced two major problems to produce gas turbine engines capable to competition with reciprocating engines: first unconventional operation and second high fuel consumption.

As the article mentions they could overcome unconventional operation of gas turbine to use in the cars. In the last 1970s, even they could reduce fuel consumption, yet at the end of the way cost of gas turbines seems much higher than conventional reciprocating engines. Hence they closed the program in the early 1980s, in the spite of the fact that technical achievements were considerable.

Turbine hot sections create most of the cost of production, in the contrary, to gain enough power using high TIT is necessary. Therefore, producing enough power by lower TIT can help the problem.

This point adjoins cycle entity of these engines. In addition, relatively high fuel consumption of gas turbine engines is also linked to this entity in which, the configuration does not allow intense control of fuel dispersal.

When load is applied on a gas turbine engine, its response would be different depending on its configuration. Single shaft gas turbine confronts to reduction airflow causing fall of output torque. This behavior makes this type inappropriate for traction purposes. The free power turbine unit does not suffer airflow reduction and its output torque is determined by fuel flow. Their output torque remains relatively constant over a wide load range for a fixed compressor speed. As a result, the lower operational speed, the higher output torques. Such a behavior is more favorable than reciprocating engines for vehicles.

In a fixed rotational speed free power turbine needs raising airflow to increase output torque, but this increment needs more power for compressor, which urges more fuel flow. To clear this issue, order to generate output torque is considered:

Fuel flow $\mapsto$ TIT $\mapsto$ compressor power $\mapsto$ compressor speed $\mapsto$ airflow $\mapsto$ output torque

This order requires increase of fuel flow to reach higher torque when airflow is not under management practically. Therefore, consistency between fuel flow and airflow is not enough, which weakens fuel consumption control.

3.1. Novel Hybrid Cycle for Vehicles

Gas turbine engines has perfect power/weight ratio in addition to adaptation ability to different fuels. These capabilities make them suitable for today demands, if their cost as well as fuel consumption to be improved.

Applying context-awareness on gas turbine, compatible with configuration suggested in section 2, allows free access to all components. This free access creates proper condition to apply intense control over fuel flow, so dynamically that its value is regulated by regard to available airflow and load. Moreover, using a backup energy supplier allows operating in lower TIT, which influences cost of hot sections.

Using aforementioned configuration, conventional order to generate torque changes as follow:

Compressor power $\mapsto$ compressor speed $\mapsto$ airflow $\mapsto$ fuel flow $\mapsto$ TIT $\mapsto$ turbine torque

The cycle proposed in this article follows this order. It consists of a small context-aware turboshaft cycle, in addition to a PEMFC as independent backup power supplier for compressor. Fuel cell supplies initial power of compressor to start, and during steady operation, fuel cell provides compressor power partially. This decreases turbine load and thus increases power to drive when TIT reduces in compare with conventional turbines.

In this hybrid cycle, turbine drives gears besides an electric generator. Generator and PEMFC together supply energy of electric motor, which derives compressor, so that independent rotational speed would be provided for compressor stage. Furthermore, compressor stage rotational speed is capable to regulate toward user demands and environmental elements by changing power provided by PEMFC.

An integrated engine management system tunes power allocated to compressor and drive, so that power can be accurately balanced.

In fact, this system has control over all the cycle components, hence, each component operation is regulated to optimum on the basis of environmental elements and user demands. This system also provides coherent and consistent functions for the cycle components. Diagram of such a hybrid cycle is schematically indicated in **Fig.1**.

To evaluate the concept, author developed exclusive software capable to design context-aware gas turbine engine and model behavior of the design for arbitrary off-design condition in average-fidelity level **[6]**. Accordingly a sample GT-PEMFC hybrid cycle was modeled by this software.

4. Designing Cycle Components

Using the exclusive software, foregoing cycle was modeled. The process was aimed at designing an independent context-aware gas turbine firstly and then a suitable model of PEMFC unit was added to the cycle.

As a result, first a turboshaft cycle was optimized to generate about 150 *kW* power by using hydrogen **Fig.2**. Following specifications were considered for the cycle:

Working condition: SSL at static condition

Air Mass Flow Rate: 1.0 $^{kg}/_{s}$

Compressor Pressure Ratio: 1:5

Turbine Inlet Temperature: 600 K

Turboshaft cycle consist of a single stage centrifugal compressor beside one stage axial flow turbine. As it is seen, because of partial provision of power for compressor by fuel cell, there is no need to high TIT, reducing necessity of expensive parts for hot sections. PEMFC selected in this study uses hydrogen as fuel directly and then hydrogen is considered the cycle fuel, moreover, low compressor pressure ratio considered for the cycle removes need to big fuel cell.

3.1. Compressor

To design compressor, method of **[7]** is used, however some changes are made to gain proper framework to design and model behavior of compressor.

Usually rotor-passage-average-outlet velocity is considered about inlet axial velocity to impeller. This assumption results in acceptable outcome in the design process, however, it is not useful to model compressor behavior at off-design conditions. As a result, it is assumed that stage slip factor remains almost constant over a wide range of operation and by regard to empirical relation suggested by **[8]** for this factor:

$$\sigma = \left(1 - \frac{2}{n}\sqrt{\cos\beta}\right)\left(1 - \frac{C_r}{U}\tan\beta\right) \quad \textbf{(1)}$$

C_r can be calculated as follow:

$$C_r = \frac{U\left(-n + n\sigma \quad + 2\sqrt{\cos\beta}\right)\cot\beta}{n - 2\sqrt{\cos\beta}} \quad \textbf{(2)}$$

Method to determine efficiency also is not compatible with modeling process requirements. Obviously selecting a desirable efficiency and determining related geometrical properties of the stage is not suitable to model stage behavior. Therefore, relation suggested by **[7]** is considered:

$$\frac{b}{D_{imp}} = \frac{\left(\frac{\gamma+1}{2}\right)^{\frac{1}{\gamma-1}}\left(1-\zeta^2\right)\left(\frac{D_{in}}{D_{imp}}\right)^2}{4\left[1+\left(\frac{1+\eta_C}{2}\right)\left(\frac{\gamma-1}{2}\right)\left(\frac{U}{\sqrt{R\,T_{01}}}\right)^2\sigma\right]^{\frac{1}{\gamma-1}}} \quad \textbf{(3)}$$

By defining desirable value for axial blades width (*b*) and impeller diameter, stage efficiency is calculated by the deriving relation as follow:

$$\eta_C = \frac{4RT_{01}\left[-1+\left(-\frac{2^{-2-\frac{1}{\gamma-1}}\left(1+\gamma\right)^{\frac{1}{\gamma-1}}\left(\zeta^2-1\right)D_{in}^2}{b\,D_{imp}}\right)^{\gamma-1}\right]+U^2\sigma\left(1-\gamma\right)}{U^2\sigma\left(\gamma-1\right)} \quad \textbf{(4)}$$

During design process, all geometrical properties are determined besides qualitative parameters such as slip and work done factors. These compressor properties are saved and used to simulate its behavior at arbitrary off-design condition. Output of exclusive software for single stage centrifugal compressor designed in this study is indicated in **Table.1**.

3.2. Turbine

Turbine design process is carried out by using method suggested by **[9]**. Turbine in this cycle is a free power unit designed to generate needful power to drive besides compressor power. **Table.2** indicates output of exclusive software for single stage axial flow turbine designed in this study.

3.3. Fuel Cell

Fuel cell selected for the cycle is a free-breathing PEMFC, which needs no additional equipment to breathe, consumes hydrogen fuel directly and works under ambient condition outside the GT. These specifications satisfy such a hybrid cycle requirements.

In a traditional free-breathing PEMFC, the cathode channels are open to ambient air from both ends. Term free breathing refers to the natural convection by which the oxygen needed by the cell reaction is transferred into the cell. Natural convection of air in the cathode channels is driven by buoyancy, which is caused by temperature and gas composition gradients. Temperature gradient is caused by the heat generated in cell reactions and losses and with possible external heating of the cell. Changes in gas composition are caused by the cathode reactions, which consume oxygen and produce water.

[8] has tested such a fuel cell in ambient, 45 °C, 60 °C, and 75 °C at 1 atmosphere. Note that in this reference the ambient cell temperature refers to a measurement in which the cell was not externally heated. The cell temperature in this measurement was 31 – 34 °C, and thus it was a few degrees higher than the temperature of the surrounding air.

The performance of the cell improves as the cell temperature rises from ambient to 60 °C. Further temperature increment causes significant performance loss.

Using empirical data of **[10]**, performance of the cell was drawn **Fig.3**. Three temperatures: ambient, 45 °C and 60 °C were set for cell internal temperatures in which by increasing temperature cell power also increases.

Accordingly, number of cells was selected to produce about 50 *kW* in the ambient temperature, and hence 291 cells by dimensions equal to $50 \times 50\, cm$ were considered. This arrangement generates 50098.19625 *W* in ambient temperature. Consequently, fuel cell generates 96903 *W* at 45 °C and 132909.885 *W* at 60 °C.
In fact, aforementioned powers show minimum available power of compressor regardless its operating condition.

4. Modeling Cycle Operation

Cycle modeling encompasses imposing arbitrary off-design condition on the cycle components and modeling their behaviors. Arbitrary off-design condition defines new environmental elements causing specific performance in the cycle known as its behavior. Each component is identified by specifications defined during design process and saved as data. Theses specifications are all the calculated during design process except performance, which is a function of input condition.
The procedure is started from compressor and then its output is imposed on the turbine when combustor effects are considered as pressure decrease and temperature increase.
Cycle operation is divided to two sections one acceleration/deceleration process, which models response to throttle command, and the other steady operation.

4.1. Steady Operation

Off-design condition is formed by changing $\frac{\sqrt{N_C}}{T_{01}}$ and C_a relative to their values at design point. These parameters are compressor input condition recognized as environmental elements. To model cycle, steady operation $\frac{\sqrt{N_C}}{T_{01}}$ is varied from 10% to 120% of its design value by increment of 1% and inlet axial velocity is varied from

5% to 100% of its design value by increment of 5%. Note that inlet pressure to compressor is constantly considered equal to SSL to agree with available data of fuel cell performance.

Environmental elements are detected and compressor rotational speed is regulated toward them to gain optimum value outside the surge margin. This regulation is carried out based on parameters effective to move stage into surge margin or other losses **[11, 12]**. The pattern used by software to adapt compressor rotational speed to environmental elements is tabulated in **Table.3**. In the regulation process, structural limitations are constantly checked to maintain maximum rotational speed in the tolerable range.

Finally, minimum available power of compressor is equal to fuel cell power, and hence, operating points in which compressor power is less than available power of fuel cell are not included to its working envelope.

Output power of fuel cell is determined by regard to available empirical curves **Fig.3**. Accordingly, three values mentioned in section 3.3 are considered for fuel cell power based on its size.

Turbine environmental elements are its input condition, namely TIT, TIP and airflow. Ambient condition similarly influences turbine operation and then it is considered as effective parameter to environmental elements.

Simulated management system collects points of cycle working envelope based on turbine outputs. Note that in the steady operation any specific pattern is not considered for power allocation to drive and compressor. Therefore, collected points encompass optimum net output power of cycle (total power output of turbine and fuel cell minus needful power of compressor) generated by minimum fuel flow in gas turbine; in addition, these points narrow working envelope of compressor.

Hence, all cycle components are in access to manage individually for adapting to environmental elements and user demands when their operations are capable to be set toward coherent and consistent functions.

4.2. Acceleration/Deceleration Process

Simulation of acceleration/deceleration process is divided to three categories depending on the fuel cell internal temperature. Ambient temperature means initial start up without engine warming up and other temperatures indicate acceleration after warming up when internal temperature of fuel cell rises.

In this process, the aim is to reach maximum available engine torque as soon as possible. This torque is defined as the target torque after that increasing power of compressor does not raise driving torque. As a result, available power of compressor and fuel flow (TIT in the simulation) rise concurrently before turbine running, which shows throttle command to reach target torque.

In the initial step, maximum available power of fuel cell, compatible with its internal temperature, is applied on compressor. Subsequently, $\frac{\sqrt{N_C}}{T_{01}}$ is started to increase by increment of 1% of its design value. In each step, by regard to available power of compressor, rotational speed is calculated and then inlet air axial velocity and air flow are measured.

Compressor outputs are applied on turbine after including combustor effects. In this process, turbine inlet temperature is increased from 85% to 115% of its design value by increment of 5%. For TIT in which turbine can produce power concerning TIP and airflow, simulated engine management system assesses total generated power by turbine and fuel cell. Unless maximum available torque is reached, this system allocates 50% of power produced by turbine to the compressor and remainder to drive.

Deceleration process is reverse of acceleration in which available power of compressor is decreased compatible with fuel flow and then airflow and TIT decrease synchronously. Such behavior results in swift declining of output torque.

5. Results

Fig.4 to **8** indicate summary of results. All the maps in these figures are categorized based on $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$ to show coherent functions of components. Effects of rotational speed regulation are clear in compressor characteristic map **Fig.4**. Compressor confronted unacceptable Mach number at diffuser entrance when $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$ varies between 1.21 and 1.3, and regulation of stage speed removed this problem **Fig.8**. According to pattern described in **Tabel.3**, to decrease Mach number at diffuser entrance software decreased the rotational speed. Although other banned conditions did not appear in this modeling procedure, however effects of rotational speed changes are clear in their values. As the **Fig.8** shows, the value of C_r ratio and flow direction at impeller outlet decline because of reduction of speed.

Form of changes for efficiency and pressure ratio versus input condition (stagnation pressure and temperature and airflow) is completely different. This different operation influences turbine operation similarly **Fig.5**. At $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$ less than 0.99, $\left.\frac{\sqrt{N_T}}{T_{03}}\right|_R$ changes between maximum and minimum when turbine pressure ratio shows minor changes. Nevertheless, by increasing $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$, $\left.\frac{\sqrt{N_T}}{T_{03}}\right|_R$ approaches a specific value when turbine pressure ratio changes between a maximum and minimum. In fact, compressor pressure ratio decreases by increasing $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$, and then the lower TIP, the lower turbine pressure ratio.

For each category variation of cycle, net power output versus turbine pressure ratio is relatively high showing ability of the cycle to confront to different loads in each value of $\left.\frac{\sqrt{N_C}}{T_{01}}\right|_R$ **Fig.6**. Input condition of turbine also indicates such a behavior **Fig.5**. Capability of compressor to produce a specific pressure ratio for a variety of input conditions creates these specifications **Fig.4**.

To calculate s.f.c fuel consumption of fuel cell was ignored, however Power-s.f.c curve represents points collected based on minimum fuel flow in gas turbine for optimum net output power of the cycle **Fig.6**.

Acceleration/deceleration process also shows rapid response to throttle command in just three steps **Fig.7**.

6. Conclusion

A method to introduce context-awareness into gas turbine engines was briefly presented in which coherent consistency replaces linkage among components. Such configuration allows free access to components individually toward adaptation to environmental elements and user demands.

To evaluate the concept preliminarily, a GT-PEMFC hybrid cycle was modeled by computer in average-fidelity level. This cycle potentially is able to adapt gas turbine engines to vehicles demands.

Summary of results indicates that this mechanism allows collecting optimum performance of cycle. Free access to each component allows adaptation of that to environmental elements and user demands whose outcome is such an adaptation for cycle entirely. This creates suitable condition to control cycle fuel consumption and adapt its behaviors to vehicle requirements.

However, some points remain open:

- To calculate s.f.c, fuel consumption of fuel cell was ignored, as a result, it does not represent fuel consumption of the cycle. Collecting more precise results requires high-fidelity level modeling in which both ion transportations between fuel cell electrodes and flow condition in compressor and turbine are numerically simulated.
- At steady operation variation of load and throttle command form another aspect of user demands and environmental elements. More accurate model of cycle behaviors needs a certain pattern to divide power between drive and compressor based on different operating situations. Control and management of such a cycle also needs significant model, which urges an independent study.

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Figures Captions

Figure 1: Output of exclusive software for turboshaft cycle optimization

Figure 2: Cell performance

Figure 3: Compressor characteristic map (Fuel cell temperature 60 °C)

Figure 4: Turbine characteristic map (Fuel cell temperature 60 °C)

Figure 5: Cycle outputs indicating coherent functions of components (Fuel cell temperature 60 °C)

Figure 6: Acceleration/deceleration process

Figure 7: Effect of speed regulation on compressor stage operation (Fuel cell temperature 60 °C)

Figure 8: Schematic of hybrid cycle diagram

Figure(s)
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Figures

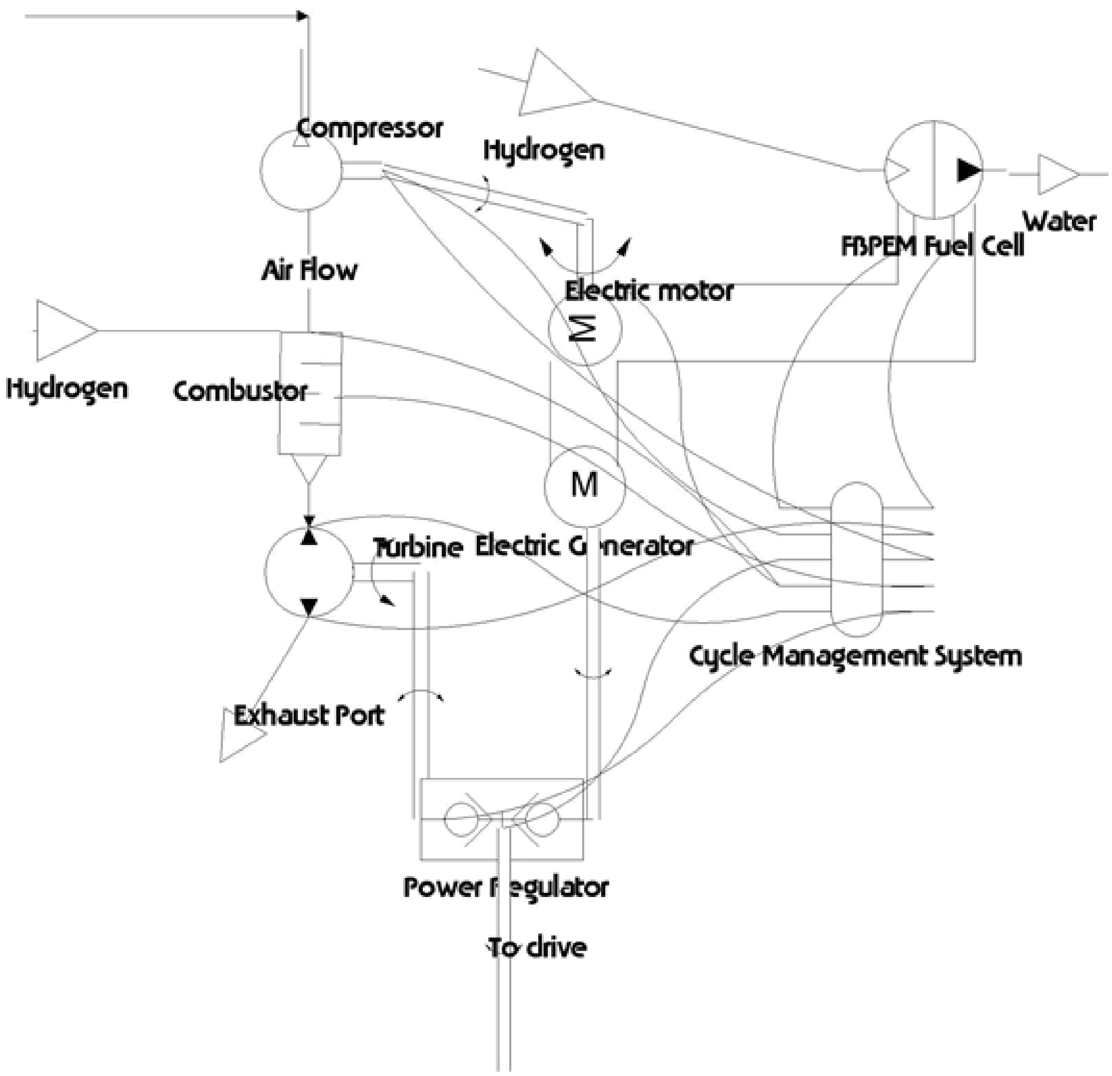
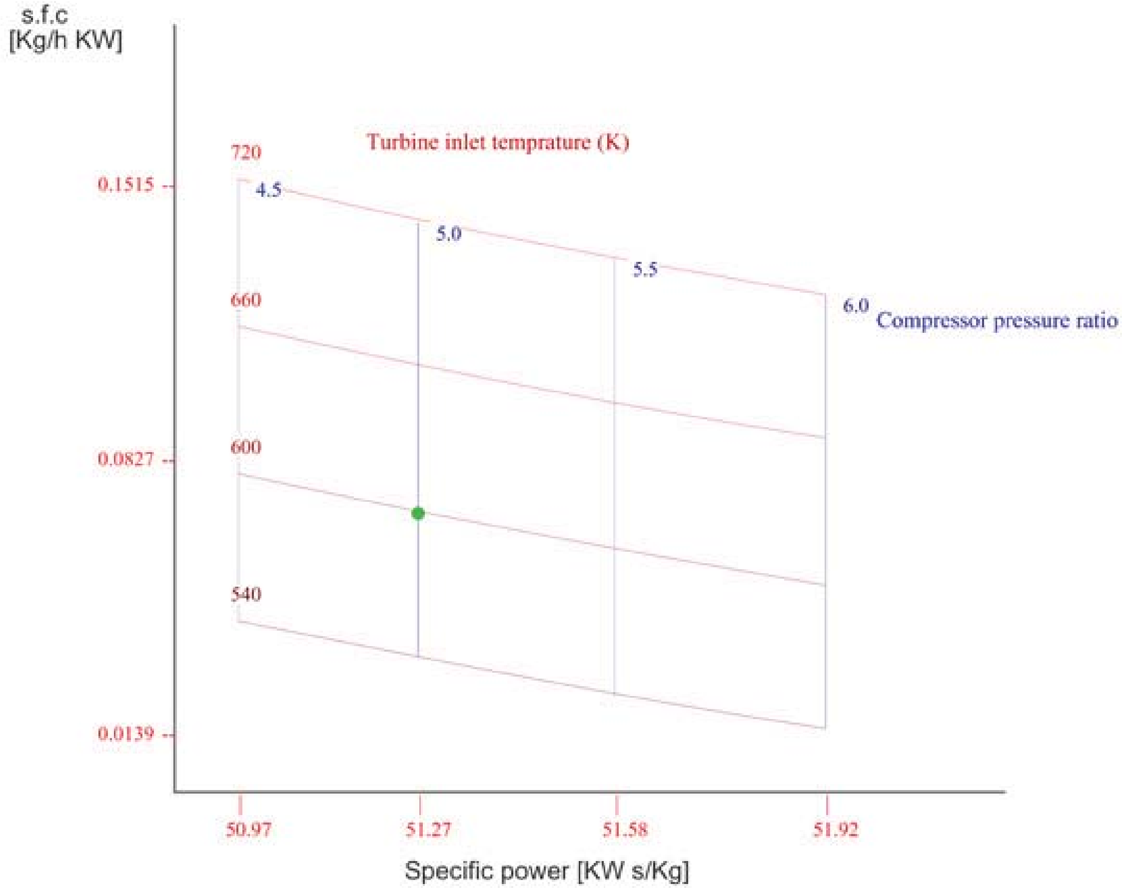


Figure 1: Schematic of hybrid cycle diagram



At desing point:
copressor pressure ratio is 5.0
turbine inlet temprature is 600.0 K
Specific fuel consumption is 0.06816 Kg/h KW
Specific power is 51.265 KW s/Kg

Figure 2: Output of exclusive software for turboshaft cycle optimization

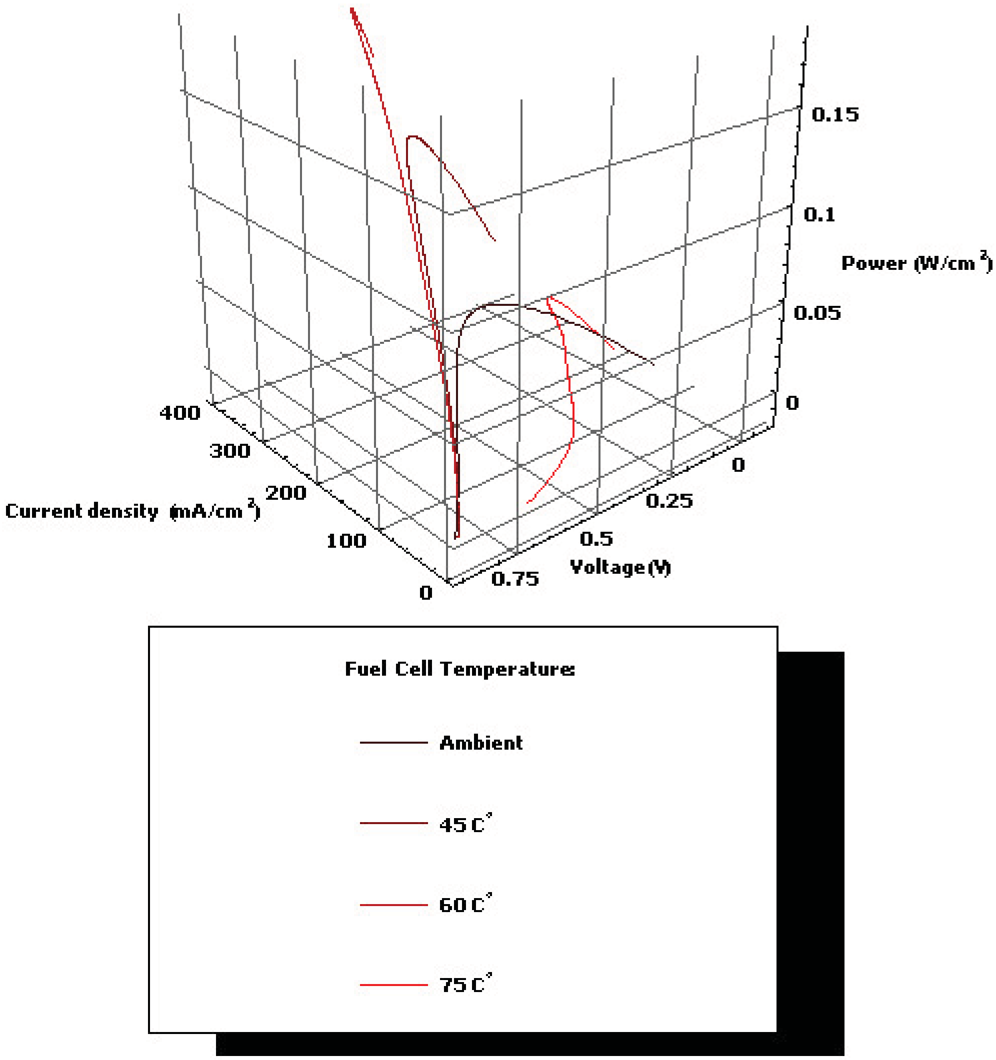


Figure 3: Cell performance

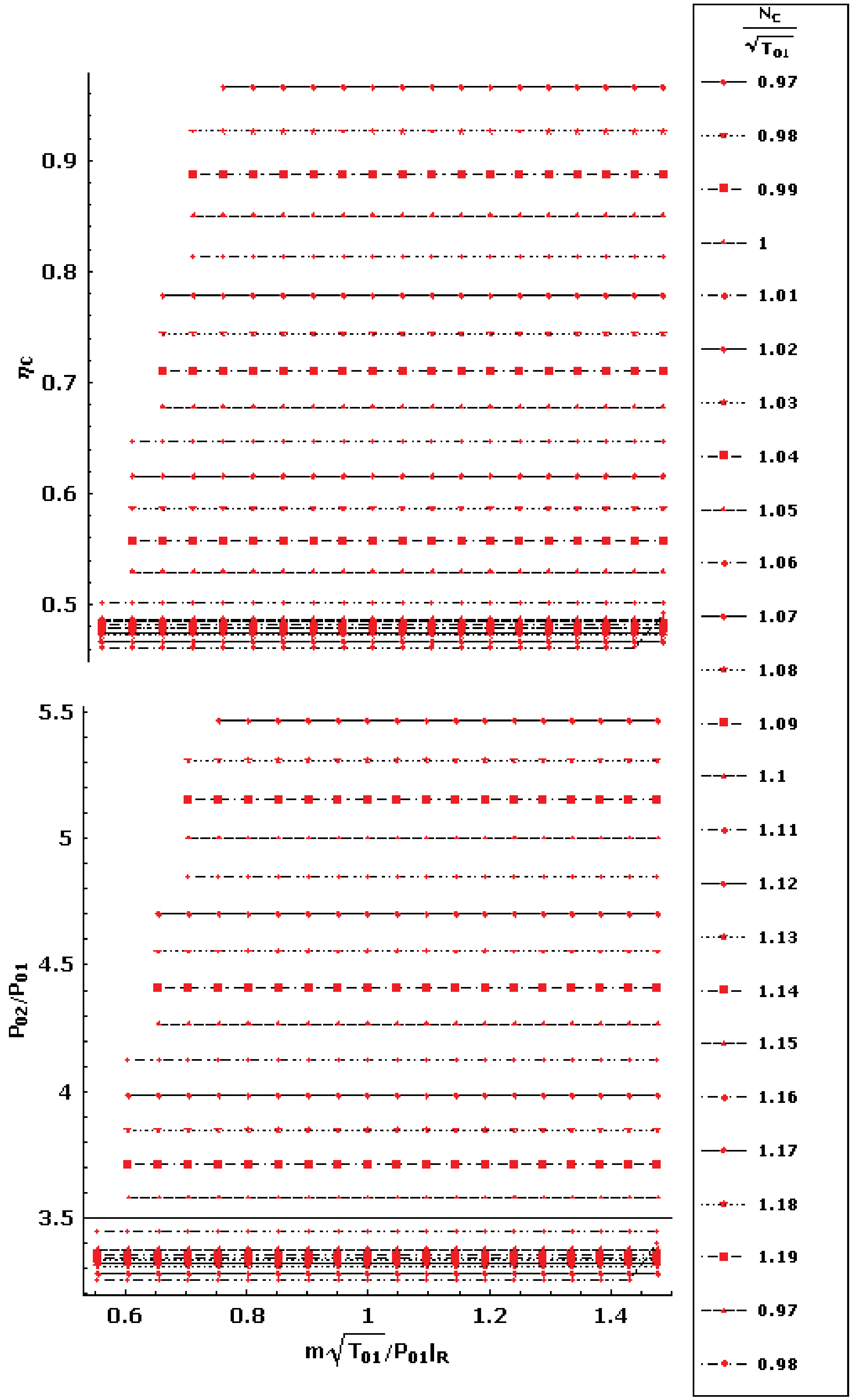


Figure 4: Compressor characteristic map (Fuel cell temperature 60 °C)

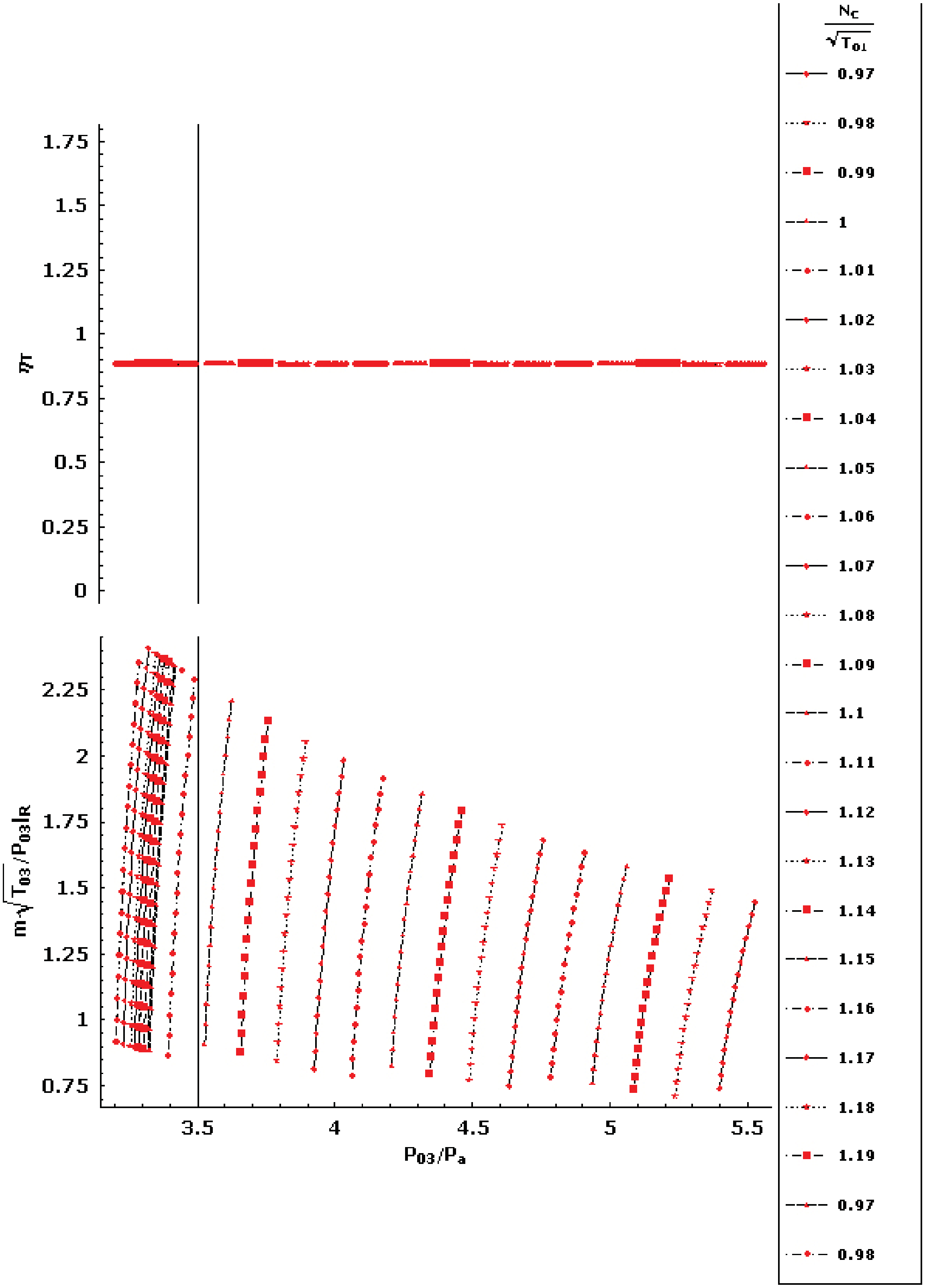


Figure 5: Turbine characteristic map (Fuel cell temperature 60 °C)

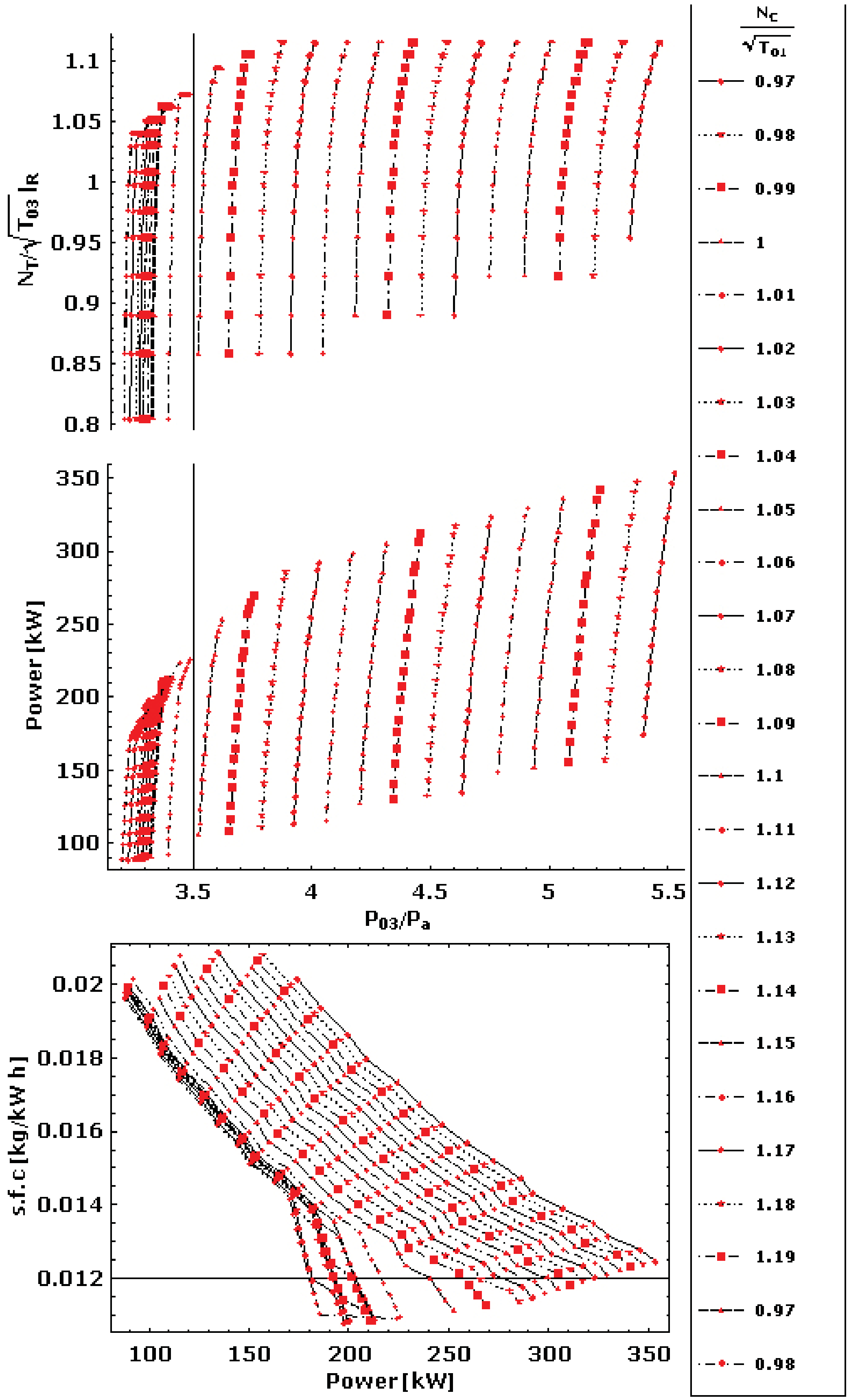


Figure 6: Cycle outputs indicating coherent functions of components (Fuel cell temperature 60 °C)

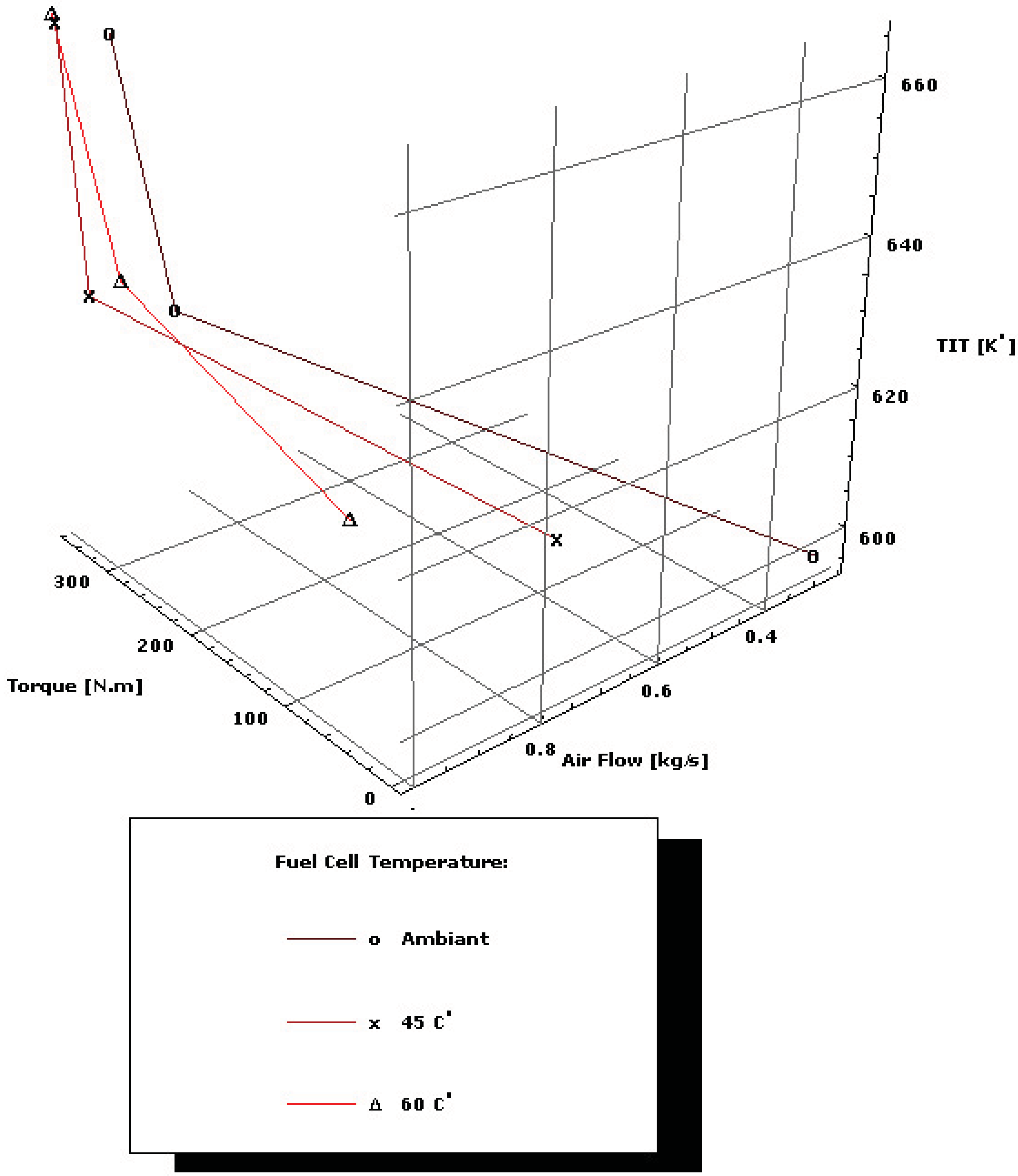


Figure 7: Acceleration/deceleration process

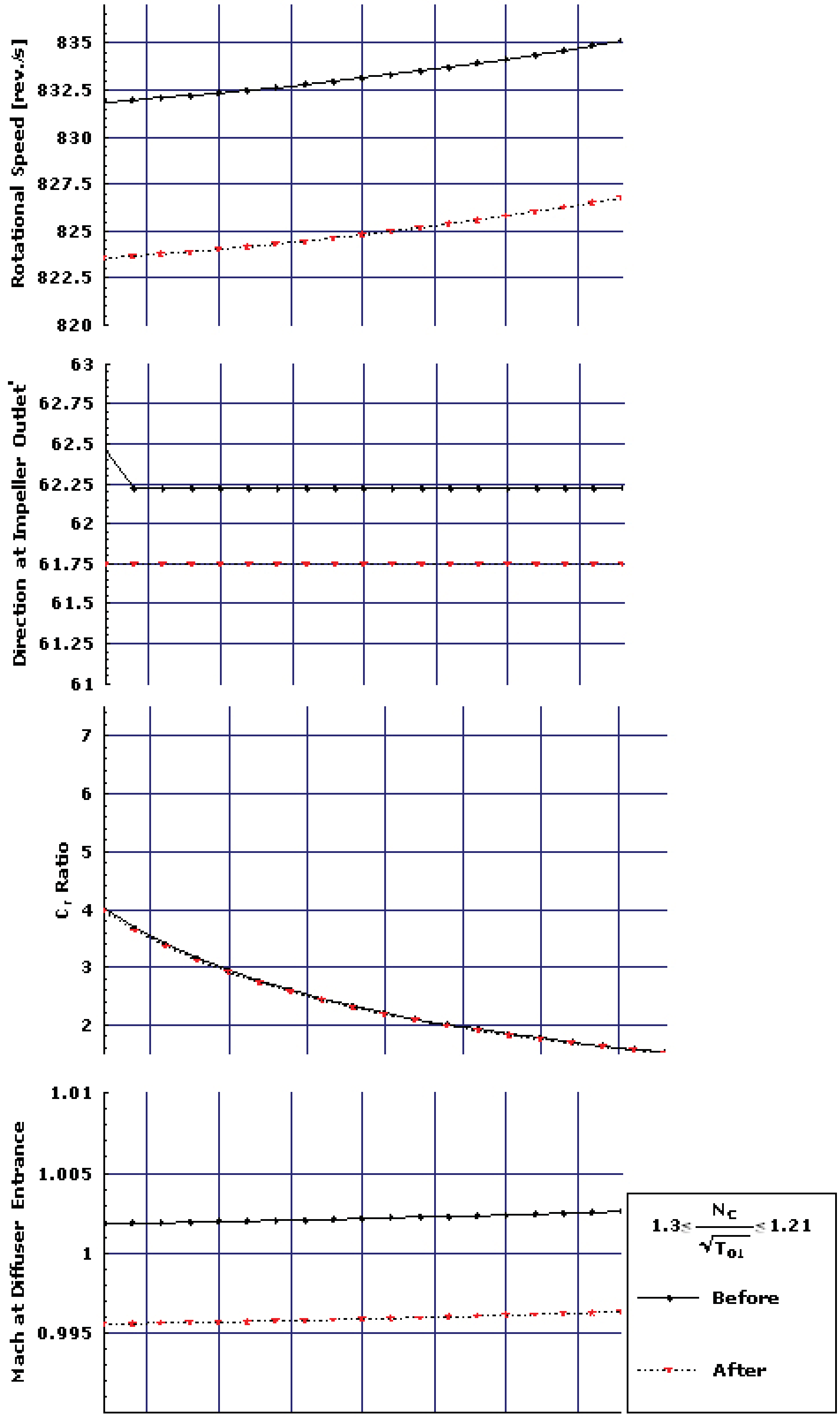


Figure 8: Effect of speed regulation on compressor stage operation (Fuel cell temperature 60 °C)

Geometrical Characteristics:

Overall diameter of impeller [m]	0.2
Eye tip diameter [m]	0.168187
Eye root diameter [m]	0.0840937
Depth of impeller channel [m]	0.0067152
Annulus area of impeller eye [m^2]	0.00555414
Number of impeller vans	15
Number of diffuser vanes	12
Radial width of vane less space [m]	0.01
Approximate mean radius of diffuser throat [m]	0.05
Width o diffuser throat in each passage [m]	0.0145564
Depth of diffuser passage [m]	0.05
Angle of diffuser vans' leading edges	88.396°

Impeller:

Entrance axial velocity [m/s]	50
Angle of prewhirl	20°
Entrance whirl velocity [m/s]	18.1985
Entrance Mach number	1.10126
Slip factor	0.9
Exit whirl velocity [m/s]	416.925
Exit radial velocity [m/s]	97.7365
Vane angle at eye root	14.397°
Vane angle at eye tip	7.314 °

Performance of Stage 1

Power input factor	1.03
Isentropic efficiency	0.85
Rotational speed [$rev./s$]	737.258
Impeller tip speed [m/s]	463.68
Stage pressure ratio	5.00067
Stage temperature rise [K]	198.794
Total pressure ratio	5.00067
Power require for this stage [MW]	0.203939
Total needful power [MW]	0.203939
Inlet stagnation pressure [atm.]	1.00702
Inlet stagnation temperature [K]	289.399

Diffuser

Whirl velocity at leading edge [m/s]	379.023
Radial velocity at leading edge [m/s]	10.6126
Mach number at leading edge	0.92508
Flow direction at leading edge	56.219 °
Whirl velocity at channels' throats [m/s]	833.85
Radial velocity at channels' throats [m/s]	557.804

Table 1: Exclusive software outputs for compressor design

Velocities $[^{m}/_{s}]$	Root	Mean	Tip
U	226.5009	310.0000	393.4991
U	208.4640	310.0000	411.5360
V	678.0038	402.1257	267.4382
V	723.0498	673.8422	695.6871
C	857.5201	626.5455	493.5948
C	0.0000	316.5455	0.0000
C	532.0578	402.1257	344.0356
C	248.0000	248.0000	248.0000
C	248.0000	248.0000	248.0000
C	532.0578	402.1257	344.0356
C	892.6616	673.8422	552.3947
C	248.0000	402.1257	248.0000
Angles$^{\circ}$	**Root**	**Mean**	**Tip**
α	0.0000	0.0000	0.0000
α	73.8698	68.4053	63.3234
α	62.2176	51.9228	43.8748
β	68.5444	51.9228	21.9796
β	69.9408	68.4053	69.1157
Coefficients	**Root**	**Mean**	**Tip**
Λ	0.1063	0.5000	0.6987
ϕ	1.0949	0.8000	0.6302
Ψ	11.5691	6.0845	3.8123
Radiuses [m]	**Root**	**Mean**	**Tip**
r_1	0.0386	0.0411	0.0436
r_2	0.0300	0.0411	0.0522
r_3	0.0276	0.0411	0.0546

Blades	Rotors	Nozzles
Opinion [m]	0.0020	0.0020
Chord [m]	0.0090	0.0074
Pitch [m]	0.0054	0.0055
Blades Number	48.0000	47.0000
Y	0.0032	0.0189

Inlet pressure [atm.]	4.93507
Inlet temperature [K]	600
Stage pressure ratio	0.244352
Stage temperature drop	290.489
Needful power [MW]	0.259121
Available power [MW]	0.260997
Exit pressure [atm.]	1.20589
Exit temperature [K]	309.511
Gas bending stress [MN]	108.233
Centrifugal stress per density $^{N\,m^3}/_{kg}$	57360.9
Stage isentropic efficiency	0.851273

Table 2: Exclusive software outputs for turbine design

Flow status $\downarrow$	Regulation status $\rightarrow$	Rotational speed is increased	Rotational speed is decreased
Mach number at inducer entrance ≥ 1.4		$\otimes$	
Mach number at diffuser entrance ≥ 1.0			$\otimes$
Ratio of passage average velocity at impeller inlet per its outlet (C_r ratio) ≤ 0.7		$\otimes$	
Flow direction at impeller outlet $\geq 80^{\circ}$			$\otimes$

Table.3: Method to regulate impeller rotational speed.