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GAS TURBINE TOPPING STAGE BASED ON ENERGY EXCHANGERS: PROCESS AND PERFORMANCE

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ABSTRACT

Power generation in gas turbines is facing three main challenges today:

- Low pollution prescribed by legal requirements.
- High efficiency to obtain low operating cost and low CO₂ emissions.
- High specific power output to obtain low product and installation cost.

Unfortunately, some of these requirements are contradictory: high efficiency and specific power force the development towards higher temperatures and pressures which increase NO_x emissions and intensify the cooling and material strength problems. A breakthrough can be achieved by applying an energy exchanger as a topping stage. Inherent advantages are the self-cooled cell-rotor which can be exposed to much higher gas temperature than a steady-flow turbine and a very short residence time at peak temperature which keeps NO_x emissions under control.

The basic idea has been proposed long time ago. Fundamental research has now led to a new energy exchanger concept. Key issues include symmetric pressure-wave processes, partial suppression of flow separation and fluid mixing, as well as quick afterburning in premixed mode. The concept has been proven in a laboratory-scale engine with very promising results. The application of an energy exchanger as a topping stage onto existing gas turbines would increase the efficiency by 17% (relative) and the power by 25%. Since the temperature level in the turbine remains unchanged, the performance improvement can also be fully utilized in combined cycle applications. This process indicates great potentials for developing advanced gas turbine systems as well as for retrofitting existing ones.

INTRODUCTION

Gas turbine power plants are the cleanest and, if equipped with combined cycle, also the most efficient way of power generation based on fossil fuels. We have seen an impressive reduction of exhaust gas emissions during the last decade and there is still a strong legislative pressure on further reduction. There has also been a remarkable improvement in efficiency and specific power output, i.e., the power per kilogram of air flowing through the engine. The

increasing importance of CO₂-control, the requirements of reducing size and cost, both operational and first cost, will be the driver for further improvements. In this paper we focus on efficiency and specific power but being aware of the important role of exhaust gas emissions.

At present, the development of gas turbine performance is pushed forward in two areas: i) aerodynamic refinements of compressors and turbines to bring up component efficiencies; ii) higher pressure ratios and turbine inlet temperatures to improve the thermodynamic process. The aerodynamics of turbo-machines have already reached a very high level with component efficiencies around 90%. There is still room for improvement but *quantum jumps* seem to be not very likely. For simple cycle gas turbines, pressure ratio mainly enhances efficiency and turbine inlet temperature enhances specific power. In combined cycle, however, temperature is the optimal parameter to improve both efficiency and specific power, while high pressure ratios reduce specific power but being almost insensitive to efficiency. Hence a major interest exists in increasing turbine inlet temperature. Indeed, new high temperature materials and advanced cooling techniques developed during the last years have given temperature a boost. However, besides the cooling and material strength problems the emission requirements raise a new limitation for temperature. Unfortunately, NO_x-emissions are strongly increasing with combustion temperature above 1500°C in steady-state combustion chambers.

A qualitative jump in gas turbine performance can only be achieved by applying advanced thermodynamic processes which are not subjected to the above mentioned limitations. The gas turbine with topping stage based on an energy exchanger is such a challenging concept. An energy exchanger is essentially a pressure-wave machine with a rotor consisting of individual cells in which a direct fluid-to-fluid energy exchange between air and combustion gases takes place like in a periodic shock tube. The energy exchanger process is equivalent to a combination of a compressor and a turbine. Since the cell-rotor is exposed to both hot and cool gases at a very high frequency, the material temperature can be kept at a moderate level even for peak temperatures which are significantly higher than current turbine inlet temperatures. The very short residence time due to the non-stationary operating principle allows for NO_x-control at high temperatures.

The basic idea of building an energy exchanger topping stage for gas turbines has been first proposed by Seippel (1942). In the early 1950's Brown Boveri tested such an engine which was intended to power locomotives. The engine was brought into operation successfully, but the achieved results were not sufficient for commercialization. Problems originated from the asymmetric wave process and insufficient material properties. Rolls-Royce and General Electric conducted wave rotor programs in the 1960's (Moritz, 1985 and Mathur, 1985). General Electric's program seems to have been promising but it was stopped for non-technical reasons. Rolls-Royce planned to realize a helicopter engine. They reported aerodynamic and mechanical problems. A wave process designed for minimal losses has been suggested by Weatherston and Hertzberg (1967). Although the process is well-suited for high pressure ratios, its application is limited by the requirement of corresponding sound speeds for both driving and driven gases to achieve optimal performance. Kentfield and O'Blenes (1988) compare several methods to achieve a combustion driven pressure gain in gas turbines. Again energy exchangers turn out to be a favorable concept. Weber (1992) also predicts high cycle efficiencies for wave engines with low pre-compression ratios.

The only energy exchanger which is commercially available so far is the Compresx[®] which is used for supercharging internal combustion engines (Zehnder and Mayer, 1984; Keller, 1984). The Compresx[®] has been developed by Brown Boveri and is now installed in Mazda diesel cars. There is much more work on energy exchangers which is not cited here explicitly. An excellent overview covering the work until 1985 is given in the proceedings of a workshop on wave rotors (Shreeve and Mathur ed., 1985).

Looking on the statements addressed by some of the authors and on the results in the past, it turns out that the following issues are critical for making the energy exchanger a successful device:

- a) Avoid unbalanced secondary waves which are generated from stator walls or from pressure waves crossing fluid interfaces.
- b) Reduce mixing of hot and cold gases in the cell-rotor.
- c) Reduce heat exchange between hot and cold gases in the rotor.
- d) Reduce flow losses in the cells as well as during opening and closing.
- e) Aim at a uniform axial temperature distribution of the rotor.

An advanced wave process for energy exchangers which pays attention to the aforementioned problems has been proposed by Keller (1988). An experimental energy exchanger based on that process has been realized and tested in the ABB Corporate Research Center.

NOMENCLATURE

c	sound speed
c _p	specific heat capacity
L	length of the cell-rotor
m	mass flow
N _{HPT}	relative HP-turbine mass flow
P	power of turbines or compressors
q _{cool}	cooling air rate of base turbine
R	median radius of the cell rotor
S _t	total scavenging (flushing) ratio
s	width of partition between hot gas inlet and outlet
T	temperature
U	velocity ratio: real machine compared to ideal process
u	velocity
V	rotation speed of the cells
Greek symbols:	
φ	flow rate in the real machine compared to ideal process
γ	ratio of specific heats for hot gas
κ	ratio of specific heats for air
η	efficiency
Π	pressure ratio

GAS TURBINE WITH TOPPING STAGE

The concept of a gas turbine with topping stage is outlined in Fig. 1. In principle, the conventional combustion chamber is replaced by the topping stage. The air from the compressor enters the energy exchanger where it undergoes a wave compression by typically a factor of 2. Further on, the air is heated up in a high pressure combustor. The combustion gases are then split into two streams which drive the high pressure turbine and the air compression in the energy exchanger, respectively. The temperature of the stream to the energy exchanger is further increased in the afterburner which allows for very short combustion times. The hot gas (driver gas) directly transfers energy to the air (driven gas) via a compression wave and undergoes an expansion wave afterwards. The exhaust gases of the energy exchanger and the high pressure turbine are mixed and further expanded in the base turbine. Optimum performance of the topping stage requires independent control of the inlet temperatures to the high pressure turbine, the base turbine and the energy exchanger. This can be accomplished by inserting a heat exchanger between exhaust gas and high pressure air which is not included in Fig. 1.

The thermodynamic cycle of the engine is shown in Fig. 2. The process of the base gas turbine and the topping stage are marked by different patterns. For the topping stage the pattern indicates an expansion process which is thermodynamically equivalent to the combined processes of the HP-turbine and the expansion wave. The base turbine is operated at the same temperature and pressure as a conventional gas turbine. The pressure drop across the energy exchanger is comparable to that of a combustor. Hence, the total gain in power output is equal to the power output of the high pressure turbine since the energy exchanger is designed for zero shaft power. Furthermore, assuming the topping stage to be adiabatic, the power output of the high pressure turbine has to be equal to the increase in fuel consumption compared to that of the base machine:

$$P_{HPT} = \Delta P_F \quad (1)$$

Going along with these arguments, a very simple expression for the thermal efficiency of the gas turbine with topping stage can be given:

$$\eta_{GTT} = \frac{P_{GT} + P_{HPT}}{P_F + P_{HPT}} = \eta_{GT} \frac{1 + \frac{P_{HPT}}{P_{GT}}}{1 + \eta_{GT} \frac{P_{HPT}}{P_{GT}}} \quad (2)$$

θ temperature ratio

ζ pressure drop

Abbreviations:

HP high pressure

LP low pressure

Subscripts:

a ambient conditions

C base compressor

F fuel

GT base gas turbine (consisting of base turbine and base compressor)

GTT gas turbine with topping stage

HPT HP-turbine

S shock wave

T base turbine

0 stagnation conditions

1 scavenging zone (LP-zone)

1e hot gas (driver gas) outlet

1i air (driven gas) inlet

2 charging zone (HP-zone)

2e air (driven gas) outlet

2i hot gas (driver gas) inlet

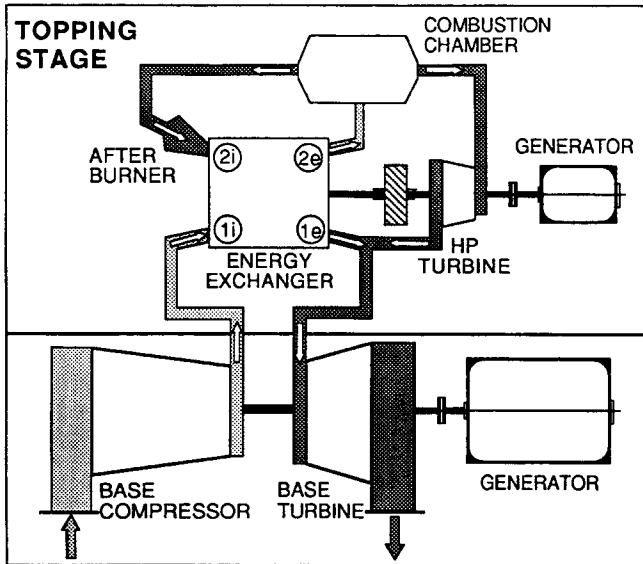


Fig. 1 Sketch of a gas turbine with energy exchanger topping stage (duct labels 1i, 1e, 2i, 2e according to Fig. 3).

where η_{GT} is the thermal efficiency of the conventional base turbine:

$$\eta_{GT} = \frac{P_{GT}}{P_F} = \frac{P_T - P_C}{P_F} \quad (3)$$

The relative gain in power output, P_{HPT}/P_{GT} , turns out to be a measure for the increase in thermal efficiency. Finally, the power output of the high pressure turbine can be written as:

$$P_{HPT} = \eta_{HPT} N_{HPT} (1 - q_{cool}) \dot{m}_C c_p T_{HPT} \left(1 - \Pi_{HPT}^{\frac{\gamma-1}{\gamma}} \right) \quad (4)$$

The relative mass flux, N_{HPT} , which is the ratio of the flow rate through the high pressure turbine to the air flow entering the energy exchanger, $(1 - q_{cool}) \dot{m}_C$, is the most important parameter characterizing the performance of the energy exchanger. The pressure ratio of the HP-turbine is assumed to be equal to that of the energy exchanger.

A look on the T-s diagram indicates that a gas turbine with topping stage is superior to a conventional gas turbine having a high pressure ratio like aero-derivatives. Although such an engine achieves comparable efficiencies in simple cycle, the specific power is less and the performance in combined cycle is worse because of the low exhaust gas temperature. In contrast, the topping stage can fully utilize the performance increase in the combined cycle due to the unchanged exhaust gas temperature.

ENERGY EXCHANGER

Idealized Wave Process

As mentioned above, the energy exchanger in the topping stage consists of a cell-rotor and two casings (stators) for different gas ports. The cell-rotor is the central core of pressure-wave machines where pressure-wave processes take place.

The selection of an appropriate wave process is the most critical issue in energy exchanger design. The functional requirements can be fulfilled by different types. It is possible to arrange the inlet ports and the exit ports on one side each. Such a parallel-flow machine was suggested by Seippel (1942). It is also possible to have the hot gas ports and the air ports on one side each as it is realized in the

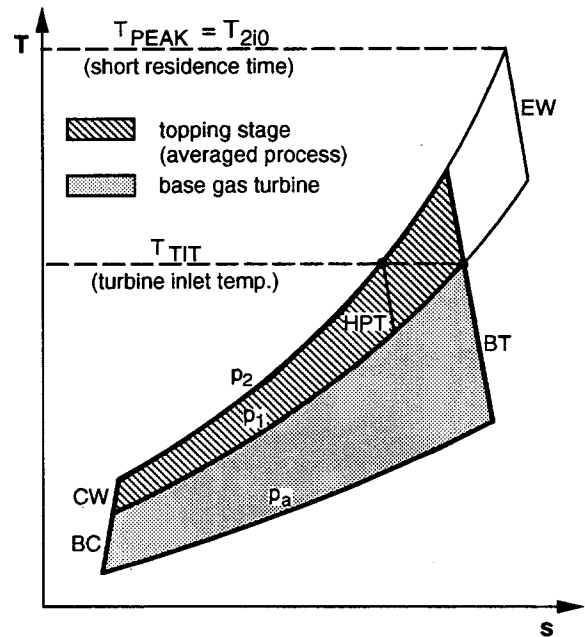


Fig. 2 Thermodynamic cycle of a gas turbine with topping stage. BC: base compressor, BT: base turbine, CW: compression wave, EW: expansion wave, HPT: high pressure turbine.

Comprex[®]. Such configurations are advantageous for special applications because of the simpler ductwork.

From a gasdynamic point of view which is crucial for efficiency, however, a symmetric counter-flow wave process is preferable. Fig. 3 shows the wound-off middle section of the cell-rotor and the casings schematically (Keller, 1988). The counter-flow wave process has two symmetric cycles. Each cycle consists of four in- or outlet ports: air inlet 1i (from base compressor), wave-compressed air outlet 2e (to combustion chamber), high pressure hot gas inlet 2i (from combustion chamber) and exhaust gas outlet 1e (to base turbine), respectively. The air and the combustion gas are often called driven gas and driver gas, respectively. Hereafter both terminologies are used. Further details of the energy exchanger will be described in a later section (laboratory-scale experimental machine).

The sketched process is idealized by applying simple compact waves which would correspond to a situation with infinitely narrow cells and small wave amplitudes. The cells move at velocity V . After a cell is filled with air flowing through port 1i, the driven gas is compressed by a shock wave induced by the driver gas (dotted pattern) entering at port 2i under high pressure and temperature. The compressed driven gas leaves at port 2e. After the cell is completely filled, the driver gas undergoes a two-step expansion wave: the first stops the flow and the second reverses the flow. The driver gas is flushed out through the outlet port 1e by the incoming LP driven gas from the opposite side. This is the beginning of the next cycle. The contact of hot gas and air in each cell illustrates very clearly the direct fluid-to-fluid energy exchange.

The partitions in the casings are only necessary to separate the different ports. This means the partitions can be made as thin as possible. Thin partitions reduce the residence time of the hot gas in the machine which is essential for low NO_x emissions.

The counter-flow process starting from both sides of the rotor maintains a symmetric and almost uniform temperature distribution of the rotor to optimally utilize the material properties at high temperatures. Due to the changing flow direction, the unsteady boundary layers developed in the cells are typically thinner than steady counterparts. The waves are purely controlled by the pressure levels of the

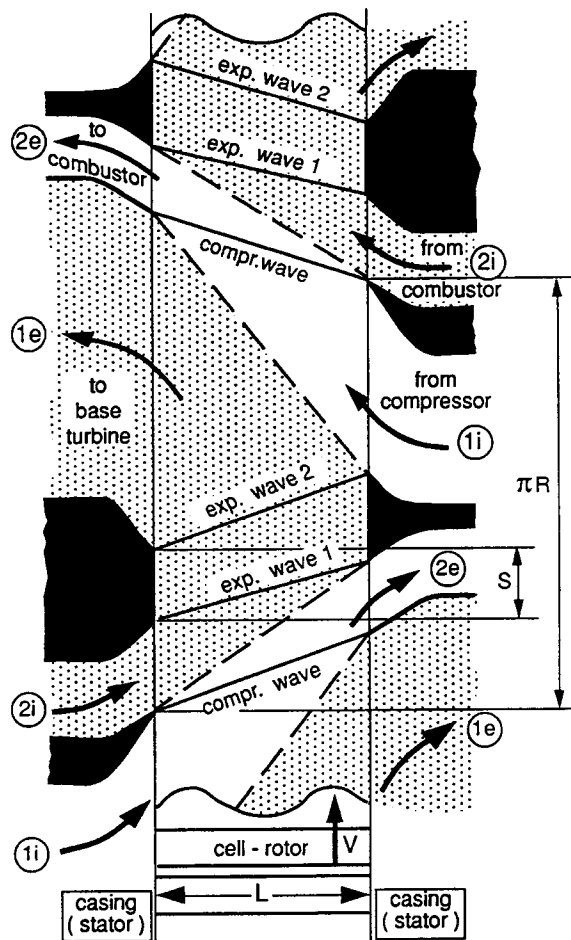


Fig. 3 Wound-off middle section of the symmetric counter-flow wave process consisting of two cycles (— waves, - - - - interfaces).

gas streams. By properly adjusting the pressure levels and the positions of the partitions, it is always possible to avoid reflected and secondary waves. This results in uniform velocity and pressure distributions in the charging zone 2 (region 2*i* and 2*e*) and the scavenging zone 1 (region 1*i* and 1*e*).

Assuming one-dimensional unsteady flow without losses, the wave process can be described by the following equations.

The velocity change of the air across the shock wave considered in a fixed frame of reference is:

$$\frac{u_2 - u_1}{c_{1i}} = \frac{1}{\kappa} \left(\frac{p_2}{p_1} - 1 \right) \left[1 + \frac{\kappa + 1}{2\kappa} \left(\frac{p_2}{p_1} - 1 \right) \right]^{-\frac{1}{2}} \quad (5)$$

The velocity change generated by the expansion waves 1 and 2 in the hot gas is:

$$\frac{u_2 + u_1}{c_{1i}} = \frac{2}{\gamma - 1} \frac{c_{2i}}{c_{1i}} \left[1 - \left(\frac{p_2}{p_1} \right)^{-\frac{\gamma - 1}{2\gamma}} \right] \quad (6)$$

These two equations already determine the axial flow in the rotor. The geometrical parameters like position and width of the ports are obtained by formulating the travelling times of the waves and the fluid interfaces along the rotor. Requiring the wave process to be periodic, the rotation speed of the cells which depends on the speed of the shock wave can be determined as follows:

$$\frac{V}{c_{1i}} = \frac{\pi R - s}{L} \left(\frac{c_{1i} + c_{1i}}{u_1} + \frac{c_{1i}}{u_2} - \frac{c_{1i}}{u_s} \right)^{-1}, \quad (7)$$

$$\frac{u_s}{c_{1i}} = \frac{u_1}{c_{1i}} + \left[1 + \frac{\kappa + 1}{2\kappa} \left(\frac{p_2}{p_1} - 1 \right) \right]^{\frac{1}{2}} \quad (8)$$

The result is based on one wave cycle starting from each casing at an offset of 180°. Typical values for the rotational Mach number do not exceed 0.3 which is significantly lower than those in conventional turbo machines. Equations (5) - (8) indicate that the idealized counter-flow wave process only depends on the dimensionless parameters such as the pressure ratio, p_2/p_1 , the sound speed ratio, c_{2i}/c_{1i} , the specific heat ratios of air and hot gas, κ and γ , and the geometric parameters, R/L and s/L . For any combination of pressure ratio and sound speed ratio (temperature ratio), a perfectly tuned counter-flow wave process can be determined.

Losses in Real Wave Process

The investigation of real wave processes requires the consideration of losses. Main sources are flow losses, leakage losses and losses due to mixing and heat exchange between the fluids. These items have already been addressed in previous papers (Weatherston and Hertzberg, 1967; Shreeve and Mathur, 1985). We mainly focus on the aspects related to the new energy exchanger design.

Flow losses are caused by boundary layer friction in the cells and ports and by throttling the flow during the finite opening and closing time of the cells. Friction depends on the Mach number, the Reynolds number, the length of the cells and the presence of secondary waves which lead to non-uniform velocity distributions in the ports. Fortunately, Mach numbers are usually fairly low. Only the compressed air reaches high subsonic values. Secondary waves are largely avoided by the counter-flow process. The throttling losses strongly depend on the contour of the partitions. Since the flow at the edges of the partitions is unsteady, separation can be suppressed by properly shaped edges. Radii of curvature which lead to Strouhal numbers above 0.3 satisfy the unsteady-nonseparation criterion (Sturtevant and Keller, 1978).

Heat transfer in the rotor is almost exclusively due to convection since regions with stagnant fluid in the cells are not present in the proposed process and radiation becomes important only for large dimensions. Hence, all steps which reduce flow losses will also reduce the heat exchange between air and hot gas.

Mixing of air and hot gas occurs during cell opening, due to centrifugal forces on the fluid interface and the fluid, which is trapped in the boundary layer at the cell walls and not expelled by the fluid interface. Mixing during the finite cell opening time is closely related to the corresponding loss aspect discussed in the previous paragraph. In fact, the idea of suppressing separation by applying curved edges originates from the necessity to reduce mixing. The instability of the interface separating fluids of different density, which are exposed to the strong centrifugal field in the rotor, is related to the hydraulic lock-exchange problem (Keller and Chyou, 1991). The imbalance of the hydrostatic pressure across the interface generates gravity or centrifugal currents which lead to an overlapping region of high and low density fluids. The length of the overlapping region depends on the density ratio and on the travelling time of the interface through the rotor compared to the time for one revolution. The mixing can be avoided by compensating the centrifugal forces in a waisted rotor. For typical operating parameters the overlap in the charging zone, which is assumed to be most critical, would occupy 15-20% of the cell length. In this case a waist with 10% reduction in radius can compensate the centrifugal field. Nevertheless, Coriolis forces exist. However, they change sign in the middle of the rotor which makes them less critical. Compensation would require S-shaped cells which do not fit into a symmetric wave process. Since

boundary layers are thin in the counter-flow process, the third contribution to mixing caused by the boundary layer fluid being flooded by a different bulk fluid is small.

Rotating machines always require a minimal clearance between the rotor and the casing through which high pressure gas can escape and finally enter the low pressure ports without performing useful work. This classical problem is well-known in turbomachines. The situation in the energy exchanger, however, is more difficult from a design point of view since rotor and casings do not have the same temperature and its distribution in the casings is non-uniform. Clearance control is necessary to achieve high performance. Different techniques have been suggested. Automatic devices apply materials with different thermal expansion like in the Compress[®]. Active devices use shiftable casings or rotors. The leakage losses are characterized by the ratio of clearance to cell height and by the shape of the clearance gap. Small relative clearance is easier to achieve in large machines.

LABORATORY-SCALE EXPERIMENTAL MACHINE

In the following section the layout and setup of the hot gas driven experimental pressure-wave machine will be described. The transfer of the operating conditions in a gas turbine power plant to the experimental model is carried out through fluid dynamic similarities. The model is operated under same pressure and temperature ratios as well as the same Mach number. However, the Reynolds number can not be matched since the model is reduced in size and the low pressure side is discharged to atmospheric air. The consequence is that the friction losses and convective heat transfer in the experimental machine become substantially noticeable.

Fig. 4 provides an impression of the build-up of the experimental machine. The aforementioned counter-flow pressure-wave process is realized in the experimental machine, where two cycles are generated during one rotation of the cell-rotor. The channels of the opposite sides are off-set by 180° with mirror-image geometry.

The inlet air is supplied by a compressor with integrated cooler. The high pressure air, compressed in the energy exchanger, is directed towards a collecting container where an equal access of both cycles is ensured. The amount of air to drive the high pressure turbine in the topping stage is simply blown out in the experiment. The remaining part flows into a second container for supplying the combustion air, which is mixed with the fuel (propane) in swirl combustors. The expanded hot gas together with the over-scavenged air is cooled down in a heat exchanger. The exhaust gas pressure can be regulated by a valve in the exhaust pipe.

The rotor of the energy exchanger is coupled with an electric motor, so that arbitrary number of revolutions can be set. This is essential for optimizing the wave processes.

The casings are water-cooled to avoid material problems and difficulties due to thermal expansion. The connecting part between the casings is also water-cooled in the first version, so that the distance between the casings can be held constant. It is possible to match the axial gap between the rotor and the casings, through an axial adjustment of the rotor. In addition, the axial seal surfaces of the high pressure (HP) channels are coated with graphite layers. These layers are continuously eroded due to the thermal expansion of the rotor, so that the HP-leakage can be kept small throughout the temperature range.

As a basis of the layout of the experimental machine, the idealized wave process is implemented and supplemented through a one-dimensional nonstationary numerical simulation. However, the viscous effect and the opening and closing processes of the cells are not yet taken into account. The design point of the machine is chosen according to the temperature and pressure ratios that are realizable in the topping stage. The stagnation pressure ratio in the topping stage is $\Pi_0 = 1.9$ (cf. Eq. 9). The stagnation temperature ratio becomes $\vartheta_0 = 3.74$ (cf. Eq. 9) based on an intake temperature of 15°C, a

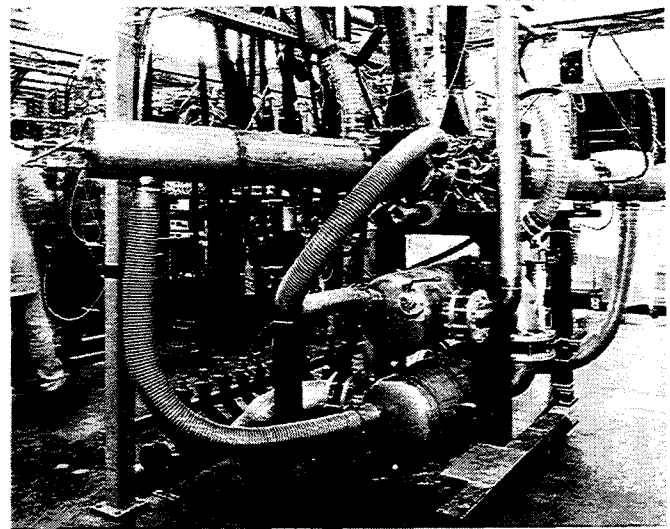
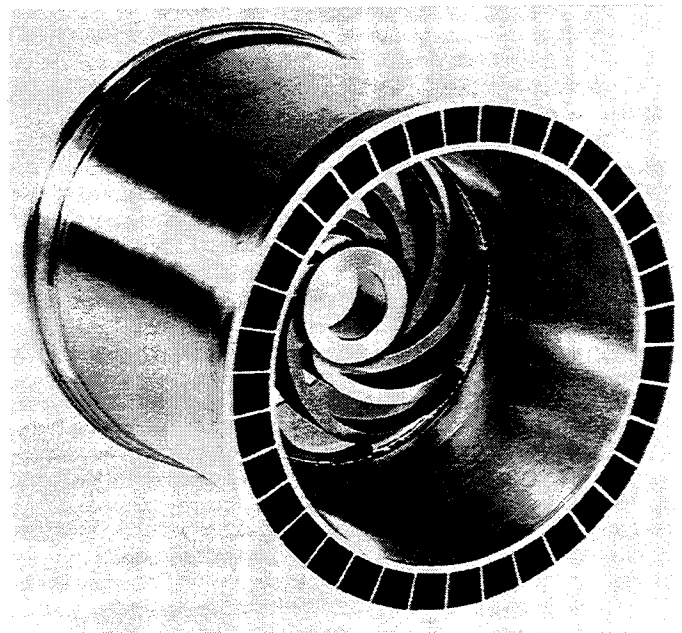


Fig. 4 Energy exchanger test rig including combustion chambers and air collectors.

compressor efficiency of 90%, a pressure ratio of 10 (compressor outlet temperature: 308°C) and a peak temperature in the energy exchanger of 1900°C. In the experimental case with atmospheric inlet air at $T_{110} = 25^\circ\text{C}$, $P_{110} = 1\text{bar}$, the corresponding hot gas temperature is $T_{210} = 840^\circ\text{C}$.

According to basic assumptions of the rotor geometry, such as $L/2R \approx 1$ (square rotor) and $h/2R \approx 0.1$ (cell height around 10% of the rotor diameter), as well as estimations of the friction loss and the heat exchange between air and hot gas in the rotor, the following rotor dimensions are set: 150mm median diameter in the inlet and outlet plane, 165 mm rotor length, 12mm cell height and 36 cells. To compensate the centrifugal force in the HP-process, a concave profile down to 90% of the nominal diameter in the middle of the rotor is imposed (Fig. 5).

With these data the wave process and the casing geometry can be determined. A typical result of the numerical simulation is shown in



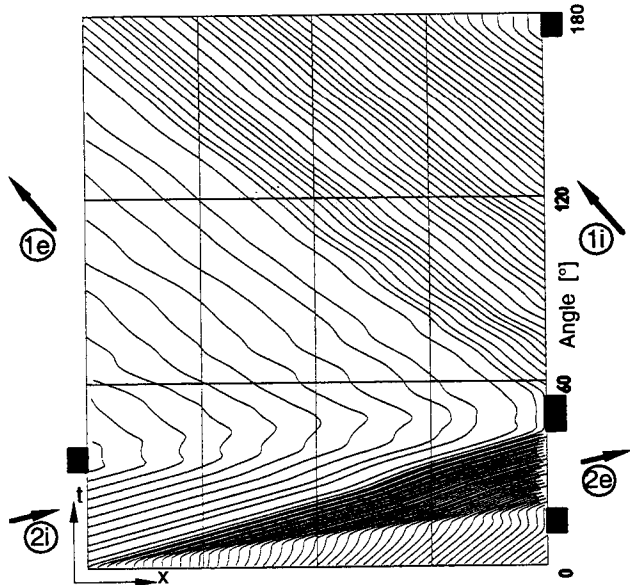


Fig. 6 Numerical simulation of the particle trajectories for one cycle: $\Pi_0 = 2$, $\vartheta_0 = 4.3$.

Fig. 6. The particle trajectories of the wave cycle coincide surprisingly well with the ideal case.

The closing edge of the air outlet channel is exchangeable, so that a good tuning at different operating conditions can be achieved. To ensure good inlet and outlet flow conditions in the scavenging cycle, the low pressure (LP) channels are built with guide vanes. For achieving good efficiency it is essential to recover the high kinetic energy of the air at the HP-outlet.

An interesting technical aspect concerning the operation of the energy exchanger is the behavior at rapid load changes. They often cause difficulties in turbomachinery, because they are usually accompanied by rapid temperature changes. The rotor of the pressure-wave machine responds relatively insensitively to this aspect, because the heat transfer through the thin rotor walls is substantially better than the convective counterparts to and from the walls, so that only small temperature differences in the material appear. For larger machines and high pressure these relationships become somewhat less favorable (also due to elevated radiation). It is important that a sufficient radial elastic connection for compensating the thermal expansion be installed between rotor casing and hub.

In the first build-up stage the machine is equipped with measuring instruments for mass flow rates, pressure and temperature of air and gas in certain locations, rotor speed and torque. The axial gap of the rotor during operation can be observed through a video camera with an endoscope. In a further step the arrangements for the measurements of non-stationary pressure in the rotor, the velocity and temperature profiles in air the outlet are installed.

EXPERIMENTAL RESULTS

Energy Exchanger

Similar to turbomachines, an energy exchanger is expected to be mainly operated in the design point. However, it has to be able to cover a wide range of load conditions in a stable manner so that it can be connected to a gas turbine. An important question is whether a machine with fixed geometry can satisfy these requirements. To prove this aspect, the results are reported for pressure ratios of 1.4-2.1 and temperature ratios of 2.9-4.2. In contrast to Equ. (5) and (6) the pressure and temperature ratios are based on stagnation conditions which are more practical for application:

$$\Pi_0 = p_{2e0}/p_{1i0}; \quad \vartheta_0 = T_{2i0}/T_{1i0}. \quad (9)$$

During this test series the rotor speed has been adapted to achieve optimal wave tuning in the charging zone. The speed had to be increased by about 22% from the lowest to the highest pressure ratios investigated. The total scavenging ratio, the ratio of low pressure air to high pressure air mass flow, has been kept at a value of 1.15:

$$S_t = \dot{m}_{1i}/\dot{m}_{2e}. \quad (10)$$

Over-scavenging helps to reduce the amount of hot gas contained in the high pressure air which is due to unavoidable mixing processes and incomplete cell filling at off-design conditions.

The first parameter which quantifies the real process compared to the ideal process is the driver gas consumption rate. It is defined as the ratio of the driver gas mass flow in the real machine to the counterpart in an ideal process with the same inlet conditions for driver and driven gases as in the experiment:

$$\Phi_{2i} = \dot{m}_{2i,exp}/\dot{m}_{2i,id}. \quad (11)$$

The same ideal process will be used as a reference for the following parameter definitions. Fig. 7 indicates that the mass flow in the real machine is always slightly higher than that under ideal conditions based on a uniform inlet velocity distribution. This is due to an increased flow velocity into partly admitted cells during opening and closing. The parameter decreases with increasing pressure ratio since the velocity overshoot becomes weaker at higher Mach numbers. The temperature ratio has only little effect on the driver gas consumption.

The second parameter characterizing the charging process is the driven gas delivery rate. It is defined as the ratio of compressed air mass flow in the real machine to that in an ideal process:

$$\Phi_{2e} = \dot{m}_{2e,exp}/\dot{m}_{2e,id}. \quad (12)$$

The high pressure driven gas flow in the machine is generally lower than that in an ideal process (Fig. 8). One of the reasons is the reduced density due to heat exchange and mixing in the rotor. The aerodynamic loss in the machine is another reason. Higher Mach numbers which are a consequence of higher pressure ratios increase losses and reduce the mass flow. At a pressure ratio of 2.1 the high pressure air flow reaches roughly 70% of that in the ideal case.

A similar behavior is observed for the velocity ratio of the driven gas at the high pressure outlet. The aerodynamic losses reduce the pressure of the driven gas which gives rise to a counter-running post compression wave to match the static pressure at the outlet, thereby decreasing the flow velocity.

The ideal process assumes isobaric conditions in the charging and the scavenging zones. The static pressure drop across the scavenging zone is in the order of 2-3% which is comparable to the pressure drop in conventional combustors.

The static pressure difference across the charging zone is displayed in Fig. 9:

$$\zeta_2 = (p_{2i} - p_{2e})/p_{2i}. \quad (13)$$

The pressure difference shows positive and negative values which is close to the ideal situation. Negative values indicate that the kinetic energy recovered in the outlet diffuser is not sufficient to meet the pressure losses in the high pressure loop including the dynamic head at the hot gas inlet. Thus, the losses outside the pressure wave machine also amplify the post compression induced in the cell-rotor and reduce performance.

The most decisive parameter towards application is the relative HP-turbine mass flow as pointed out earlier. The definition given in Eq. (4) is evaluated in the following manner:

$$N_{HPT} = [\dot{m}_{2e} - (\dot{m}_{2i} - \dot{m}_F)]/\dot{m}_{1i}. \quad (14)$$

The fuel consumed in the test rig, \dot{m}_F , is taken into account, but the fuel which would have to be added in the engine to heat up the air

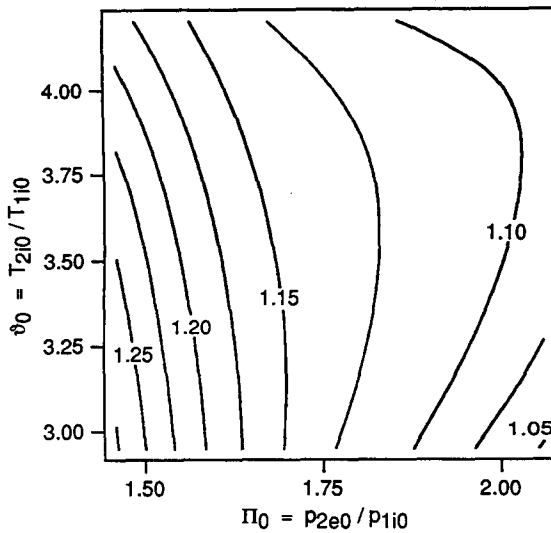


Fig. 7 Driver gas consumption rate ϕ_{2i} .

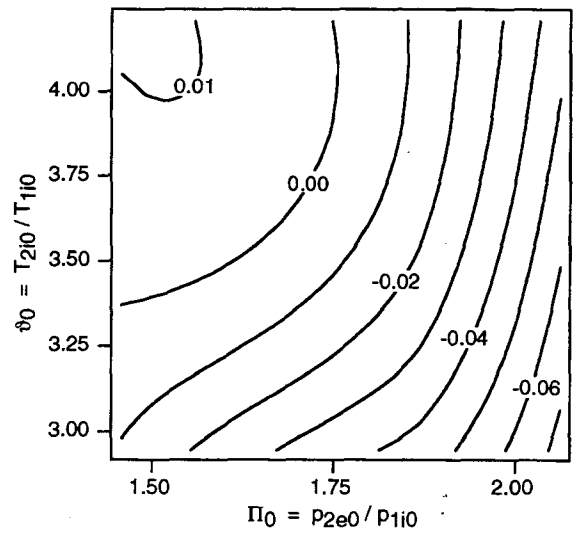


Fig. 9 Static pressure difference ζ_2 across the charging zone.

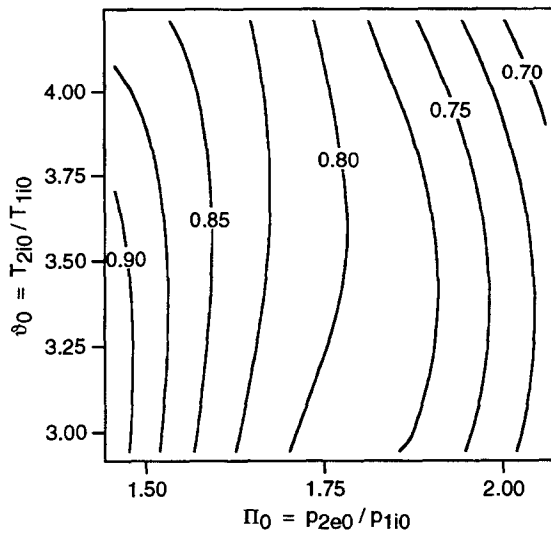


Fig. 8 Driven gas delivery rate ϕ_{2e} .

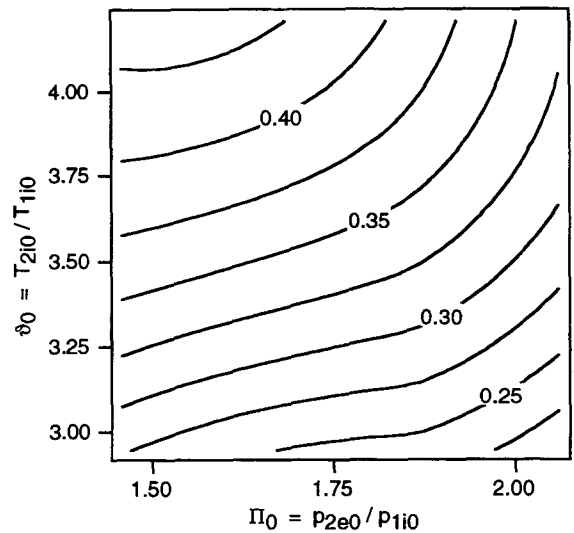


Fig. 10 Relative HP-turbine mass flux $N_{HP,T}$.

to turbine inlet temperature is excluded. The relative mass flux (Fig. 10) is most sensitive to the temperature ratio. High driver gas temperatures reduce the density and consequently the consumption of driver gas. Increasing pressure ratio reduces the relative mass flux. The effect is weak for the ideal process but it is amplified by the losses in the real machine. For the present configuration up to more than 40% of the air entering the energy exchanger can be used in the HP-turbine.

It should be mentioned that the maps shown here are not limited by any instability. The flow exhibits no steady-state adverse pressure gradient which is different from turbo-compressors. The high pressure blow-off duct which feeds the HP-turbine in a real configuration can even be closed without causing problems. The maximum pressure ratios achieved such way are much beyond the values presented here.

Application to Gas Turbine

Finally we transfer the results from the test engine to a gas turbine with topping stage. An interesting application arises from retrofit modules for existing gas turbines. The need for low emission combustion technology can be beneficially combined with an upgrade

in efficiency and power output. Hence, the thermodynamic analysis is based on a low performance gas turbine: 30% efficiency, 250kJ/kg specific power, 15% cooling air, pressure ratio of 10. For the HP-turbine 89% efficiency and 1100°C inlet temperature are assumed. To obtain a conservative estimate, no Reynolds number upscaling of the experimental results has been undertaken.

Figs. 11 and 12 show the relative performance gain of the gas turbine with topping stage by applying Eqs. (2) and (4). For the highest temperature ratios the efficiency improves by 18.5% (relative) and the power output by 28%. Optimum performance is achieved at pressure ratios between 1.9 and 2 which is close to the design point. Like in simple cycle gas turbines the optimum value increases slightly with temperature ratio. The temperature ratios investigated correspond to driver gas temperatures in the engine between 1410°C and 2160°C (dissociation not taken into account). Setting the upper limit at 1900°C which is equivalent to a temperature ratio of 3.74 which was used as the design value, the gains in power output and efficiency are still 25% and 17%, respectively. Hence, the specific power goes up to 315kJ/kg and the efficiency exceeds 35%. The topping stage converts the base gas turbine with low performance into a state-of-the-art engine.

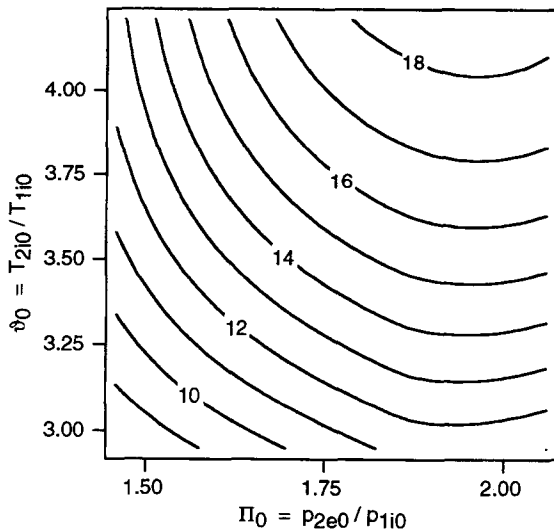


Fig. 11 Relative efficiency gain [%] of a gas turbine with topping stage.

Similarly, the topping stage can also be applied to future gas turbines. Starting from a new design offers the opportunity to integrate the energy exchanger into the turbomachine which makes the whole engine very compact and elegant. The relative performance gains would be somewhat smaller than for retrofit modules. It can be seen from Eq. (2) that improvements are more difficult to obtain when starting from already higher performance levels. It is expected that a new generation of gas turbines with topping stage comes up to 40% efficiency and 400kJ/kg specific power in simple cycle. Combined cycle efficiencies will be beyond 55%.

CONCLUSIONS

Topping stages based on energy exchangers has been proven to be a promising concept for significantly improving the gas turbine process:

- The proposed symmetric counter-flow wave process can be perfectly adapted to any values of pressure and temperature ratios. It inherently avoids the crossing of fluid interfaces and pressure waves.
- Partitions with contoured edges reduce flow losses. Combined with a waisted rotor, they also help to minimize the mixing of driver and driven gases.
- Shock wave compression as applied here induces no flow separation. Thus, the energy exchanger exhibits a wide and stable map.
- The self-cooled rotor allows operation at high combustion gas temperature. Short residence time compensates the effect of increased temperature on NO_x emissions.
- Transferring the experimental results from a laboratory-scale engine to an existing gas turbine, the efficiency can be increased from 30% to 35% and the specific power from 250kJ/kg to 315kJ/kg. Future advanced gas turbines, with a high performance level, are expected to be boosted to 40% efficiency and 400kJ/kg specific power by applying a topping stage.

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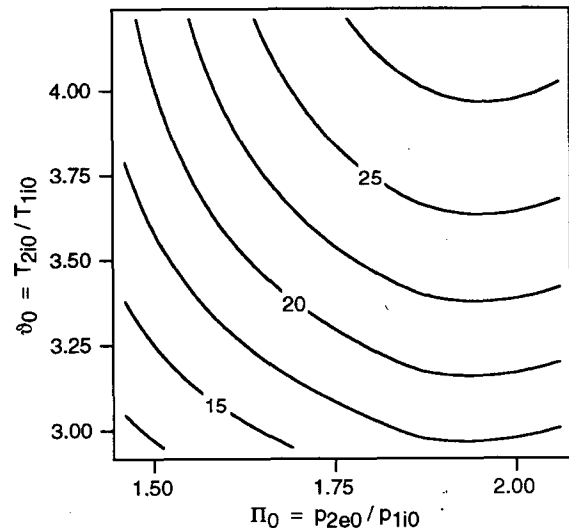


Fig. 12 Increase in specific power output [%] of a gas turbine with topping stage.

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