The Fourth International Workshop on
Micro and Nanotechnology for Power Generation
and Energy Conversion Applications
PowerMEMS 2004
November 28 - 30, 2004
Kyoto, Japan

Heat and Power Balance in Micro Gasturbine Rotors

R.A. Van den Braembussche (*), Z. Alsalihi and T. Verstraete
Von Karman Institute for Fluid Dynamics
Waterloose steenweg, 72, 1640, Sint-Genesius-Rode, BELGIUM
(*) Tel +32 2 359 96 09, Fax +32 2 359 96 00, Email vdb@vki.ac.be

Abstract

The paper provides a detailed analysis of the impact of internal heat transfer between the turbine and compressor of a micro gasturbine rotor on component efficiency, power output and gasturbine cycle efficiency. A coherent model to estimate the performance deterioration of compressor and turbine is presented. Input is the heat flux from the turbine to the compressor and the adiabatic performance of the components. The influence of impeller size and fluid temperature is evaluated by means of heat flux values obtained from a series of diabatic Navier Stokes calculations on different geometries with different wall temperatures. The impact on power output and efficiency of recuperative and non-recuperative cycles is presented.

Keywords: micro-gasturbine, Heat transfer, diabatic flow

1 INTRODUCTION

Internal heat transfer has an important impact on the performance of the very small turbomachines, used in micro and nano gasturbines. The heat flux from the hot turbine to the colder compressor results in a cooling of the flow in the turbine and a heating of the flow in the compressor. The performance of the components changes and can no longer be evaluated by measuring the flow conditions at inlet and outlet of the components. This problem has first been recognized and studied for small turbochargers where it was shown that the distance between the hot turbine and the cold compressor may have a considerable impact on the flow conditions [1] and procedures to correct for this internal heat transfer have been proposed [2]. The impact on micro-turbomachinery performance is discussed by Gong et al. [3].

Following describes a model to evaluate the impact of the internal heat transfer from the turbine to the compressor on the performance and outlet flow condition of each component. Input to this model is obtained from diabatic flow calculations on compressors of different size at different impeller surface temperatures. This allows the estimation of the impact of internal heat transfer on component and cycle efficiency and is discussed in the last section.

2 PERFORMANCE PREDICTION MODEL

The performance prediction model is based on the assumption that the friction losses in the impellers are not changed by the heat transfer.
However, heating the flow during the compression process results in a higher outlet temperature, hence lower density. The main consequences are less diffusion than with adiabatic flow and as can easily be evaluated from the outlet velocity triangles lower work input and pressure rise. The latter one results in a further decrease of the density at impeller outlet. In what follows one will assume that the corresponding velocity increase at the outlet is compensated by a proportional increase of the passage width at impeller exit. As a consequence the velocity is unchanged along a flow path. The original velocity triangles are reestablished and friction losses can be evaluated from the polytropic efficiency of an adiabatic flow.

Heating the flow during compression has a negative effect on the efficiency because the enthalpy *dh* needed for an elementary isentropic compression *dP* is increasing with the temperature.

$$dh = \frac{dP}{\rho} = \frac{dP}{P} R_G . T \tag{1}$$

Cooling the flow during the expansion in a turbine has also a negative effect on the efficiency because the energy *dh* obtained from an isentropic pressure drop *dP* decreases with decreasing temperature.

The first law of thermodynamics provides following relation for non isentropic and diabatic compression:

$$dh = \frac{dP}{\rho} + TdS = \frac{dP}{\rho} + dL + dQ \tag{2}$$

where *dL* is the heat produced by the internal friction losses and *dQ* is the amount of heat transmitted trough the walls (Fig. 1)

Expressing the losses and heat addition as a linear function of the enthalpy rise

$$dL = \frac{L_{1,2}}{h_2 - h_1} dh \qquad dQ = \frac{Q_{1,2}}{h_2 - h_1} dh \tag{3}$$

and substituting it into (2) provides following relation.

$$\left(1 - \frac{L_{12}}{h_2 - h_1} - \frac{Q_{12}}{h_2 - h_1} \right) dh = \frac{dP}{\rho} \tag{4}$$

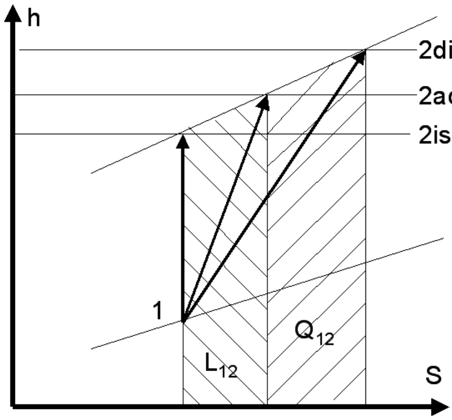

Fig. 1 H,S diagram for a diabatic compression

Using the relations for perfect gas with constant specific heat coefficient *Cp* to calculate the density and expressing the enthalpy in function of the temperature, one obtains:

$$\left(1 - \frac{L_{12}}{Cp.(T_2 - T_1)} - \frac{Q_{12}}{Cp.(T_2 - T_1)} \right) dT = \frac{\gamma - 1}{\gamma} T \frac{dP}{P} \tag{5}$$

Integration from T_1 to T_2 results in

$$\ln \frac{T_2}{T_1} \left(1 - \frac{L_{12} + Q_{12}}{Cp.(T_2 - T_1)} \right) = \frac{\gamma - 1}{\gamma} \ln \frac{P_2}{P_1} \tag{6}$$

or

$$\ln \frac{T_2}{T_1} = \ln \frac{P_2}{P_1}^{\mu} \tag{7}$$

where

$$\mu = \frac{\frac{\gamma - 1}{\gamma}}{1 - \frac{L_{12} + Q_{12}}{C_p.(T_2 - T_1)}} \tag{8}$$

Equation (7) is similar to the definition of polytropic efficiency of an adiabatic compression where μ stands for

$$\mu = \frac{\frac{\gamma - 1}{\gamma}}{\eta_p} \tag{9}$$

This means that for an adiabatic compression (Q_{12}=0) the value of L_{12} can be derived from the polytropic efficiency.

$$L_{12} = (1 - \eta_p) C_p.(T_2 - T_1) \tag{10}$$

The impeller exit temperature for an adiabatic flow T_{2adi} is then defined by (7) after substitution of the appropriate value of L_{12} and $Q_{12}=0$ in (8).

If $Q_{12} \neq 0$ the same equations provide the non adiabatic outlet temperature T_{2dia} .

3 HEAT FLUX CALCULATIONS

The amount of heat flux through the compressor walls ($Q_{12}>0$) depends on the compressor wall temperature which in turn is a function of the turbine inlet gas temperature, the blade and disk surface, the distance between turbine and compressor impeller, the conductivity of the material, the heat transfer coefficient, etc. A complete calculation requires a combination of a heat transfer calculation in the whole rotor with a diabatic flow calculation in the compressor and turbine.

In what follows one will estimate typical values of the heat flux in typical compressor impellers with different wall temperatures (Table 1).

Table 1. Navier Stokes test cases

Outlet diameter (mm)	8	20.
RPM	1050000	420000
Mass flow (gr/sec)	~1.	~6.7
$P^0{}_2/P^0{}_1$	2.1	2.1
T_{wall} (°K)	600	500/600/700

The flow and heat flux is calculated by the TRAF3D Navier Stokes solver developed by Arnone [4] on a grid with 400 000 cells using the Baldwin Lomax turbulence model. This may not be the most appropriate model but an experimental study of low Reynolds number flow in a rotating channel with wall roughness is presently under way to verify/improve it.

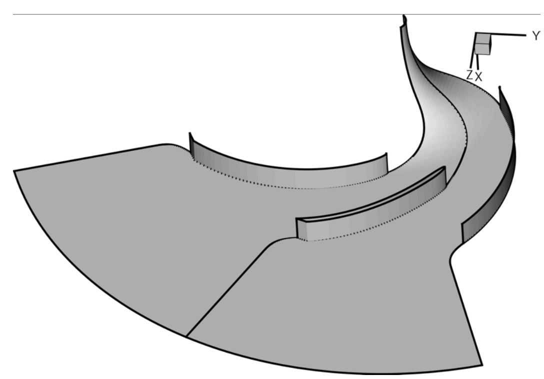

Fig. 2 View on 2D impeller section

The 20 mm diameter 2D compressor impeller (Fig.2.) is a geometrically scaled version of the 8 mm diameter one. It has four blades and four splitters.

The model explained in section 2 assumes the same velocity distribution inside the impeller for adiabatic and diabatic flows. Hence the inlet and outlet velocity triangles should be conserved during the diabatic calculations. This is achieved by:

* adjusting the pressure ratio to obtain the same mass flow and hence the same inlet velocity triangles,
* increasing the impeller outlet width *b2* to compensate for the decrease in outlet flow density.

The heat flux is defined by integrating the flux on all surface cells (i.e. on blades, hub and shroud). Fig. 3 shows the dependence of heat flux on compressor size and wall temperature. It varies between 27% of the large compressor power at 500 ^oK wall temperature to 62% at and 700 ^oK wall temperature.

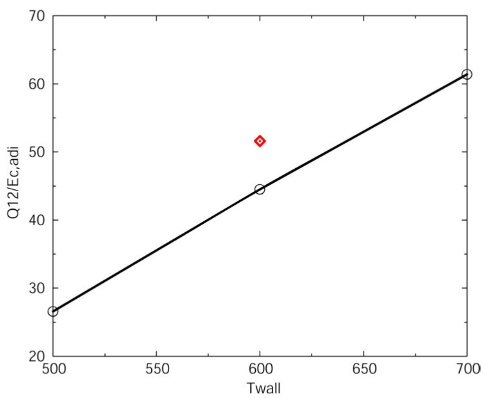

Fig 3. Variation of heat flux with wall temperature for 8 mm. Ø (◊) and 20 mm Ø (o) impeller.

The mechanical- or shaft power is obtained by subtracting the heat flux from the total energy addition defined by inlet and outlet temperature. This value is in good agreement with the one defined from the Euler momentum equation. The adiabatic impeller efficiency, based on shaft power and pressure ratio, reduces from 69.6% to 63.5% for the diabatic compression. It means that the power input, to reach the same pressure ratio with a diabatic compression is higher by a factor is 69.6/63.5 = 1.09 . This is in good agreement with the model predictions.

4 CYCLE EFFICIENCY

The total energy needed for an adiabatic compression of a unit of gas from P_1 to P_2 is

$$E_{adi} = C_p.(T_{2adi} - T_1) \tag{11}$$

Introducing a positive value of Q_{12} to (8) lowers the equivalent polytropic exponent μ. It allows defining the new outlet temperature T_{2dia} (7).

The mechanical energy required to compress the gas in a diabatic process is obtained by subtracting the value of Q_{12} from the total energy input.

$$E_{dia} = C_p.(T_{2dia} - T_1) - Q_{12} \tag{12}$$

Although both compression processes take place with the same friction losses, E_{dia} is larger than E_{adi} because the diabatic one takes place at a higher temperature. For negative values of Q_{12} (cooling of the compressor) the required energy would be lower than the one specified in (11) because of the lower fluid temperature during compression. The process is then closer to the more efficient isothermal compression.

The opposite phenomenon occurs in the turbine. Expanding the flow in the turbine at a lower than adiabatic temperature, reduces the power output because the available energy corresponding to a given pressure drop, decreases with decreasing temperature (1). Inverting station 1 and 2 in equations (11) and (12), one obtains the relation between the adiabatic and diabatic turbine shaft power. $E_{T,adi} > E_{T,dia}$

Heating the turbine would result in an increased power output in a way similar to what is expressed by the reheat factor for turbines.

Less power output from the turbine in combination with more power required by the compressor will result in a lower gasturbine power output.

$$E_{GT,dia} = E_{T,dia} - E_{C,dia} < E_{T,adi} - E_{C,adi} = E_{GT,adi}$$

and lower cycle efficiency

Lets evaluate the consequences for a typical small gasturbine rotor with following cycle characteristics:

$P_2/P_1 = 3.$	T_{1C}=293. ^oK	$\eta_{pC} = .7$
	T_{iT}= 1600. ^oK	$\eta_{pT} = .8$

The amount of Q_{12} depends on the impeller size (see section 2). It is expressed as a % of the adiabatic compressor power input (52% in a micro gasturbine with rotor diameter 8 mm at 600 ^oK wall temperature).

Fig. 4 shows the ratio of the diabatic over adiabatic power output as a function of the compressor polytropic

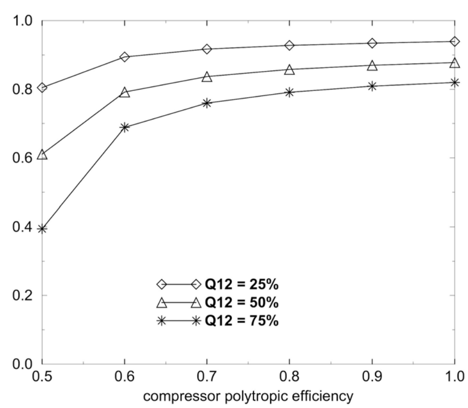

Fig. 4 Ratio between diabatic over adiabatic gasturbine power output as function of compressor polytropic efficiency.

efficiency for three typical cases. Heat transfer Q_{12} is 25.% 50.% or 75. % of the compressor adiabatic input power at polytropic efficiency of .7 .

The heat transfer from the turbine to the compressor as well as the lower compressor efficiency increase the compressor exit temperature and hence combustion chamber inlet temperature. The effect is comparable to a regenerator and the corresponding increase of the combustion chamber inlet temperature means that less fuel will be needed to reach a given turbine inlet temperature. This partially compensates the decreased compressor and turbine efficiency and explains the relatively small impact of heat transfer on the cycle efficiency (Fig.5).

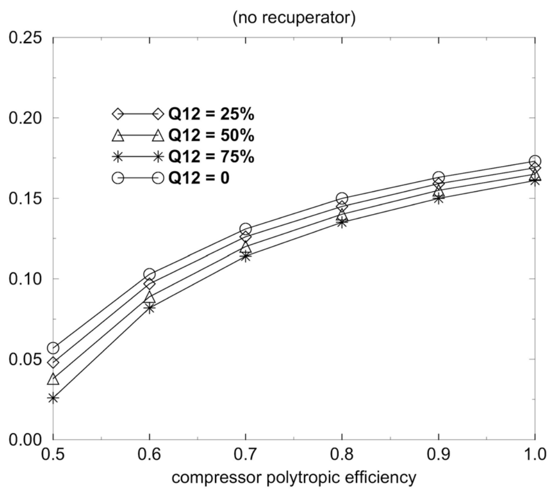

Fig. 5 Impact of heat transfer on cycle efficiency in (without regenerator)

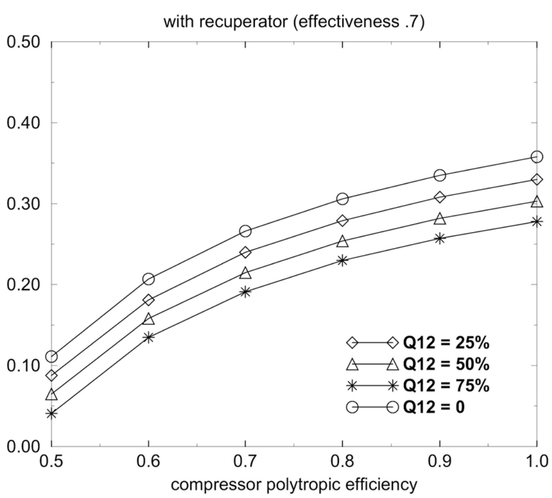

Fig. 6 Impact of heat transfer on cycle efficiency (with regenerator).

This compensation does not occur if a regenerator is used because the smaller difference between the turbine- and compressor exit temperature results in a smaller heat recuperation by the regenerator. Results shown on Fig. 6 assume a constant (.70) effectiveness of the regenerator

CONCLUSIONS

The heat transfer from the turbine to the compressor lowers the compressor and turbine efficiency. This change in performance cannot be evaluated from the compressor and turbine inlet and outlet flow conditions (temperatures) but requires a direct torque measurement.

The diabatic flow model presented in this paper allows the calculation of the performance deterioration. However it requires a correct estimation of the heat flux.

A parametric study provides an estimation of the decrease of gasturbine power output and cycle efficiency as a function of heat flux. Estimations of the heat flux by means of 3D Navier stokes solver allows the evaluation of performance deterioration in relation to the impeller size.

It is shown that the decrease in cycle efficiency is almost independent of compressor efficiency but increases for a regenerative cycle.

AKNOWLEDGEMENT

Research presented in this paper is financed by IWT (Institute for Promotion and Innovation by Science and Technology in Flanders) under the project SBO 030288 “PowerMEMS”

REFERENCES

[1] Rautenberg M., Mobarak a. and Malobabic M., “Influence of Heat Transfer between Turbine and Compressor on the Performance of small Turbochargers”, GTSJ Intl. Gasturbine Congress, Tokyo, 23-29 November, 1983.

[2] Rautenberg M and Kammer N., “On the thermodynamics of non-adiabatic compression and Expansion Porcesses in Turbomachines”, 5th Intl. Conference for Mechanical Power Engineering, October 13-15, 1984.

[3] Gong Y., Sirakov B.T., Epstein A.H.and Tan C.S., “Aerothermodynamics of micro-turbomachinery”, ASME-GT2004-53877.

[4] Arnone A., “Viscous analysis of Three-Dimensional Rotor Flow Using a Multigrid Method” ASME Journal of Turbomachinery, Vol. 116, pp 435-445.