The cocoon type microturbine

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Abstract
Based on previous studies on MEMS ultra microturbine concept, a new ultra microturbine architecture, of the cocoon type, in the power range of 10 to 100 Watts is described. The energetic study of a one stage or two stage turbomachine microturbine, with the purpose to verify the good fitting with a one disc magnetic type electric generator is then undertaken. Finally, the evaluation of the compressor efficiency with downscaling is approached.

1. Introduction

The designation "ultra microturbine" was first proposed by the Professor Toshio Nagashima [1] from the University of Tokyo, in order to make distinction between the decaWatt microturbine concept and the well known microturbine denomination for the range between 10 and 100 KW. Thanks to Professor Alan H. Epstein [2] the MEMS ultra microturbine concept was launched at the Gas Turbine Laboratory of the M.I.T. in 1994. Since this date, a tremendous amount of work on this subject has been performed by this Laboratory.
The potential applications of this micro energy device, in the 10 W to 1 KW range, are numerous and in particular microdrone propulsion, robots actuation and light portable electric sources with large self operation and high specific power are anticipated. Even if important researches have been performed for more than ten years in many engineering disciplines such as thermodynamics, internal aerodynamics, combustion, MEMS technology, gas bearings, electric generator, regulation, packaging, the practical feasibility of such a project, especially on the energetic point of view is not still proven.
Concerning the choice of the well known Brayton Joule thermodynamic cycle, we can sort the concepts underway in two classes.
The first class corresponds to a pure MEMS concept in which we find besides the M.I.T. work, a Singapore-Japan project [3] with the particularity that a piezoelectric converter is chosen.
In Japan, a couple of years ago, a consortium of University Laboratories [4], under an International NEDO contract, has been working both on palm top and finger top microturbines, the last one being topped with a wave rotor [5]. The Michigan State University College of Mechanical Engineering suggests [6] also a new architecture comprising a wave rotor to boost the pressure ratio. The Tohoku University in Sendaï has realized interesting MEMS fabrications and microturbine component tests [7] but now is mainly involved in the non MEMS IHI project described hereafter.
The second class concerns the projects that depart from the MEMS concept. More conventional microfabrications allow us to design 3D shaped turbomachines for the benefit of their efficiency and to choose the best materials for the different components. This choice is essential for the internal heat management. The Tohoku University-IHI project [8] aims to a 100W electric output microturbine in which the rotor group is obtained by machining. Stanford University and its industrial partners [9] are developing a miniature silicon nitride ceramic gas turbine designed to spin at 800000 rpm to generate also 100 Watts. A Belgian project [10] is also underway. The microturbine will be in the dm^3 range and will produce a power output of about 100 Watts. The Swiss Federal Institute of Technology from Zürich suggests [11] a non conventional solution in that two compressor-turbine spools are present, which gives the capability to design enhanced thermodynamic cycles. Actually most of these designs are slipping towards the one KW range.

At ONERA a watch over on this MEMS concept began in 1997. It was followed by prospective studies on different types of micro engines. In parallel we chose to deepen our understanding on the energetic behaviour of the ultra microturbine concept proposed by the MIT [12] and then to design, build and experiment three technological bricks [13], [14]. Taking advantage of our past experience, we have designed a new architecture that may be also placed in the non MEMS class of ultra microturbines.

2. Towards a new microturbine type

In order to describe with some accuracy the thermodynamic behavior of the M.I.T. microturbine concept and to determine its performance, we built and used an energetic model named "hot button" which deals with the well known fluid mechanics laws, both in the main fluid but also in the rotor/stator and bearings cavities and which performs a thermal balance in the rotor and stator materials.
From this energetic study we could not conclude if this MEMS concept can give a positive power ouput because many negative effects arise :
- first the low Reynolds numbers encountered lead to low turbomachine efficiencies, high losses in the rotor-stator cavities and in the gas bearings;
- the MEMS 2D turbomachine shapes lower one time more the compressor and turbine efficiencies;
- with the high heat conductivity of Si and SiC, the internal heat mixing lowers a lot the compressor pressure ratio and the turbine power;
- we must add the presence of high relative tip clearance of the rotors, the difficulty to burn all the fuel injected in a tiny volume, the difficulty to design a high efficient electric generator, with a good match between its rotation speed and that of the turbomachines.

To face this situation we decided:
- to design turbomachines of optimal specific speed;
- to use two independent compressor-turbine spools. This allows to reduce the peripheral velocity of the rotors and to arrange enhanced thermodynamic cycles;
- to place the combustor(s) in the centre part and the cold parts at the periphery;
- to use several materials suited to the heat management.

3. Description of the cocoon type ultra microturbine

The volume shape of the microturbine is cylindric (fig.1) [15]. In the centre part two overlapped burners are placed, the first burner being at the periphery and the second at the inner part. The two independent spool turbomachines are situated on the bases of the cylinder, the compressors facing outside. Three cylindric envelopes determine two concentric channels. The external envelope is heat conductive, the middle one is heat resistant, the internal one is conductive. The cylindric external channel connects the LP compressor exhaust to the HP compressor inlet and acts as a heat cooler by heat exchange with outside. The internal channel connects the HP compressor exhaust to the first burner and acts as a preheater. After expanding through the HP turbine, a post heat is organized in the second burner to offer more power to the LP turbine. Finally, after expanding through the LP turbine, the gases exhaust through discrete radial pipes.
An opportunity is offered to build a combined thermodynamic cycle by using water (fig.2). For this purpose, the initial microturbine cylinder is placed in a water tank in order to cool the outside walls.
This tank acts like a pressure cooker and the overpressure vapor is injected in the main part through a valve, towards the LP turbine. The exhaust discrete pipes are also immersed to heat the water, so the exhaust gases flow outside with a moderate temperature.
In the figure 2 and as an example, the ignition of a low temperature propergol cartridge delivers a pressurized gas that starts the LP spool. As for the delivery of the ouput power, the figure shows, as an example, a pneumatic coupling of the main part with a radial turbogenerator that may be of the MEMS type.

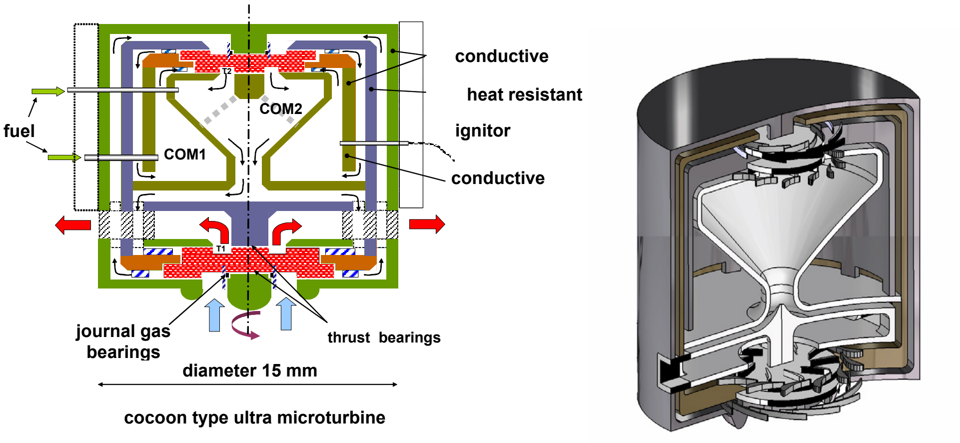

Fig.1 Basic cocoon ultra microturbine

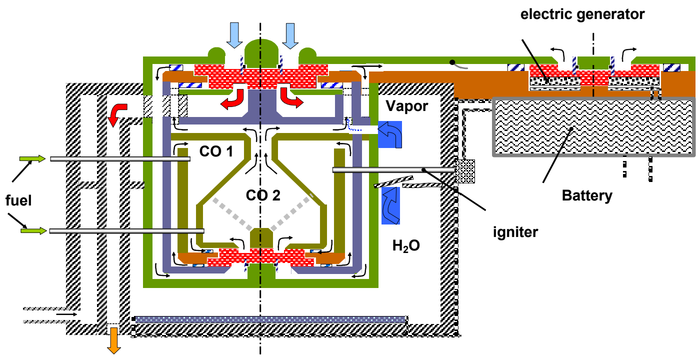

Fig.2 Combined cycle cocoon ultra microturbine

4. Electric generator Integration

The thermal efficiency of a two stage turbomachine arrangement with two independent shafts was studied in order to determine the size of the ultra microturbine and of its components, mainly the combustor and the turbomachines [16]. To simplify the analysis, a simple Brayton Joule thermodynamic cycle is kept, with an adiabatic operation for the turbomachines. On a qualitative point of view, we can consider that the real non adiabatic operation is counterbalanced by that of the enhanced thermodynamic cycle, which gives some sense to our analysis. A more realistic but more complex calculation is underway.

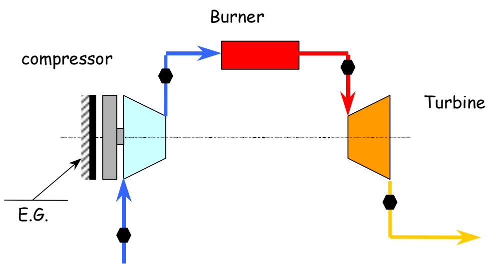

Fig. 3 Mechanical coupling of the electric generator (E.G.)

For comparisons, several configurations were considered : a one stage main turbomachines with a mechanical coupling with the electric generator (fig.3) or an electric turbogenerator driven by a part of the L.P. compressed air or an electric turbogenerator driven by a free downstream turbine. The same configurations but with two stage main turbomachines are also analyzed (fig. 4 and 5).

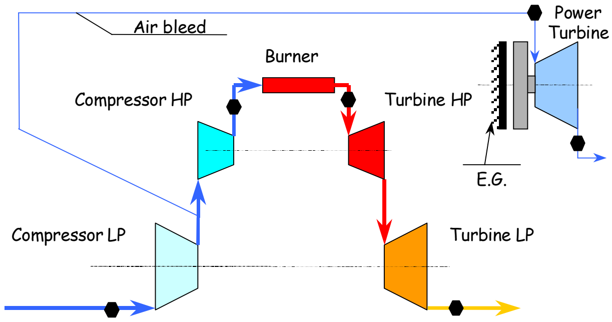

Fig. 4 Two stage turbomachines and compressed air coupling with the E. G.

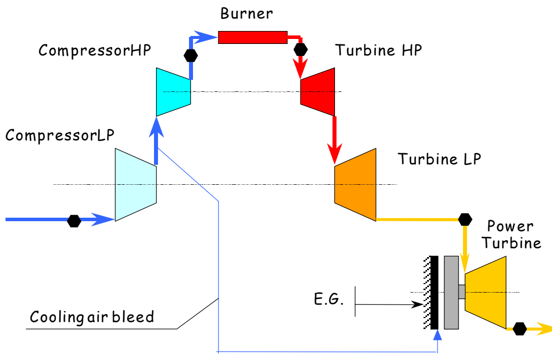

Fig. 5 Two stage turbomachines with a free turbine coupled with the E.G.

The calculations are performed to obtain an electric power output of 20 Watts, and the radial turbomachine efficiencies are optimized on a specific speed basis : $N_S = 0.6$.
The hypotheses chosen for the thermodynamic calculations are as follows :
the flow in the turbomachines is adiabatic;
For the reference case A the input are :
the maximum temperature in the burner is $T_{max} = 1600°K$;
the compressors polytropic efficiency is $\eta_{pc} = 0.6$ and for the turbines $\eta_{pt} = 0.7$;
the combustion efficiency of the burner is $\eta_b = 0.9$;
the pressure loss in the combustor is 5%;
the efficiency of the E.G. is $\eta_{EG} = 0.65$;
the cooling air bleed to the E.G. is 5% of the main mass flow rate (free turbine configuration);
the maximum peripheral speed of the turbomachines and of the E.G. is $U_{max} = 500m/s$.
Three different cases are studied that differ by the input given in table 1.

Table 1

	A	B	C
Tmax	1600K	1300K	1300K
η_{pc}	0.6	0.6	0.55

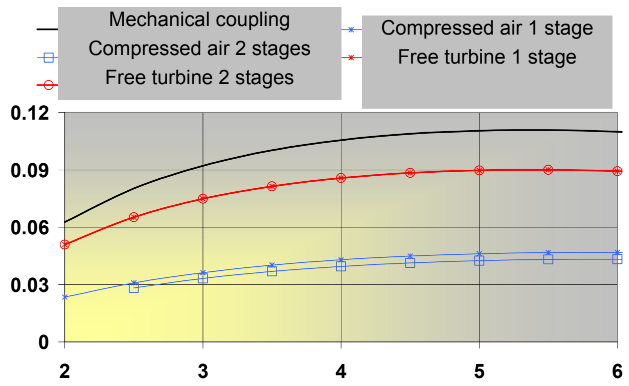

Fig.6 Thermal efficiency versus pressure ratio (case A)

The fig. 6 shows that the maximum thermal efficiency is obtained for a low compressor pressure ratio, between 5 and 6. This result is a direct outcome of the low values of the turbomachine efficiencies. The mechanical coupling of the E.G. gives the best efficiency, the free turbine coupling suffers from the free turbine efficiency and the compressed air coupling suffers both from the compressor and the output turbine efficiencies. The table 2 gives the maximum thermal efficiency that can be obtained for the cases A, B, C. The combined influence of the maximum temperature and of the compressor efficiency appears to be very important. The size of the combustors will be also very dependent on these parameters. In particular between the cases A and C the volume of the combustors is multiplied by a factor of about 5.

Table 2 Maximum thermal efficiency for cases A, B, C.
(mechanical coupling)

	A	B	C
$\eta_{th\ \%}$	11.5	7	4.7
Π_{opt}	5.5	3.5	3
Air mass flow rate m gr/s	0.21	0.42	.61

For a two stage turbomachines it seems advantageous to keep a total pressure ratio Π=3 in order to keep the thermomechanical constraints in the main turbines at a low value and to reduce the friction losses in the gas bearings. Now we can take a look at the compressor rotor diameters given in table 3. Except for the L.P. stage with compressed air coupling, we must underline that the rotor diameters are very small and much less than 10 mm.

Table 3 Compressor rotor diameter (Π = 3)

ϕ compressor diameter (mm)	A	B	C
Mechanical coupling, 1 stage	1.8	2.4	3
Compressed air coupling, 1stage	3.3	4.1	4.8
Compressed air coupling, LP stage	7.6	8.6	9.6
Compressed air coupling, HP stage	2.9	4	5
Free turbine coupling, 1 stage	1.9	2.7	3.5
Free turbine coupling, LP stage	3	4.1	5
Free turbine coupling, HP stage	2	2.7	3.3

Table 4 Electric generator diameter (ϕ_{EG} , for $\Pi = 3$), case A

U_{max} = 500m/s $\Rightarrow \phi_{EGmax}$	ϕ_{PT} mm	N_{PT}	ϕ_{EGmax} mm	P_{EG} Watts	P_{EGmax} Watts
Mechanical coupling	1.8	$5.4E^6$	1.8	0.6	0.6
Compressed air coupling, 1stage	3.9	1.710^6	5.6	1.5	5.6
Compressed air coupling, 2 stages	9.1	3.9 10^5	25	2.3	125
Free turbine coupling, 1 stage	6.2	1.2 E^6	8	4.6	12.8
Free turbine coupling, 2 stages	5.9	1.3 E^6	7.4	4.4	11

The diameter ϕ_{PT} and the rotational speed N_{PT} of the power turbine are given in table 4. If $\phi_{EG} = \phi_{PT}$ then we obtain the electric power given by the magnetic generator, by assuming that $P_{EG} = K\ \phi^2_{EG}{*}U^2$, the constant K being taken from recent experiments on a radial magnetic type E.G. (Raisigel, 2006). As the E.G. power is generally less than 20 Watts, we calculate the new value of this electric power P_{EGmax} by increasing the E.G. diameter up to $U = U_{max}$. Only the compressed air coupling configuration with two stages allows to deliver an electric power higher than 20 Watts, this last value being obtained for ϕ_{EG} =15.7 mm.

5. Evaluation of the compressor efficiency

As the thermodynamic calculations show us the important influence of the compressor efficiency on the overall microturbine performance, the accurate prediction of this component efficiency in function of its parameters is mandatory.

5.1 internal heat losses and aerodynamic losses split

Due to the small size of the turbomachines, the Reynolds number is low, which means that the convective heat transfer in the compressor is high. In this situation, the flow evolution in the compressor is diabatic.
On other hand it is advantageous to separate the heat transfer effect from the aerodynamic losses effect, because the first one is dependant both on the materials thermal properties and on the overall architecture, whereas the other effect is more intrinsic of the compressor parameters. However the heat transfer must be predicted with accuracy to control and optimize the flow evolution in the rotor and so that the inlet diffuser flow angle match the suction side leading edge stagger angle of the blades.
A good split between the heat mixing and the aerodynamic losses is proposed by using the following formula that link the pressure ratio to the total temperature ratio of the turbomachine :

$$\frac{T_{i2}}{T_{i1}} = \left(\frac{P_{i2}}{P_{i1}}\right)^{\left(\frac{\gamma-1}{\gamma K}\right)}$$

where $K = \eta_{pol\ comp}/(1+\lambda)$ for compressors

and $K = 1/(\eta_{pol\ turb} \times (1-\lambda))$ for turbines;

η_{pol} represents the aerodynamic polytropic efficiency for adiabatic flows and λ the fraction of heat transferred to the flow when compared to the mechanical power (compressor or turbine). This approach, assumes, as a first approximation , that the aerodynamic losses do not change with heat transfer.

5.2 Optimum specific speed choice

Because of the negative influence of both the internal heat transfer and the low Reynolds numbers effects on the compressor efficiency, it is mandatory to design the compressor with an optimum specific speed even if it leads to a mismatch between the turbomachines rotational speed and the electric generator one. The following figure shows the strong influence of the specific speed on the impeller polytropic efficiency, for a classical turbulent Reynolds number.

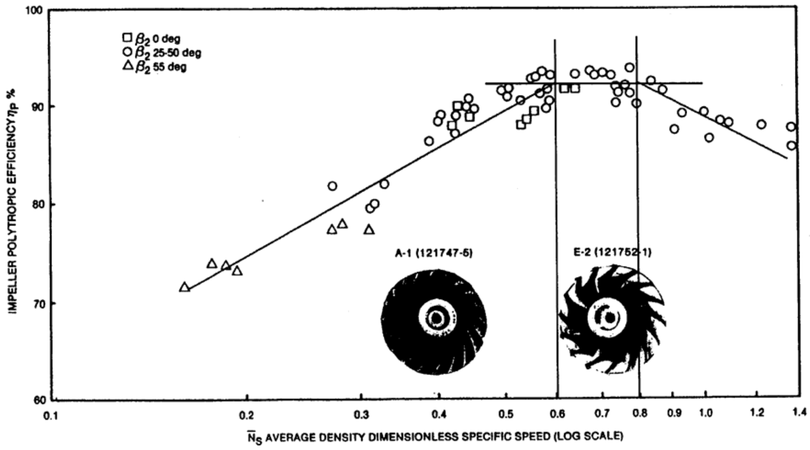

Fig. 7 rotor polytropic efficiency versus specific speed [18]

5.3 Reynolds number effect on aerodynamic losses

From the knowledge of the aerodynamic polytropic efficiency ηp of a state of the art compressor in the high Reynolds number range, characterized by Re, we can determine the performance of the small scaled compressor associated with Re' [19], [20]. Many formula are proposed, we chose the following correlation :

$\eta p' = 1 - (1-\eta p)*(Re/Re')^k$ where $k = A - B* Ln(Re_{mean})$ and $Re_{mean} = (Re* Re')^{1/2}$ valid for $Re_{mean} > 4 .10^5$

and where A and B are taken from experimental data.

The table 5 shows an example of the evolution of the aerodynamic polytropic efficiency (adiabatic) given by such a formula (optimum specific speed and 3 to one pressure ratio).

Table 5

ηp	Φ_{mm}	Re
0.85	.29	8.2E6
0.75	8E-3	2.2E5
0.69	4E-3	1.1E5
0.62	2E-3	5.6E4

These numbers constitute a guide, it shows that downsizing the rotor to 2 millimeter is not impossible from an energetic point of view (more difficult to manufacture) but we must also account for the tip clearance effect in the rotor, for the relative thickness of the blades (thickness to passage), for the relative roughness of the surfaces, these effects being negative with downscaling. So for a 2 millimeter diameter rotor, a polytopic efficiency value of about 0.55 seems more realistic. From this fact, it should be perhaps more appropriate to try to design a compressor of the Tesla type [19], with a larger diameter. It should be then possible to integrate directly the electric generator rotor on the disc close to the stator.

Conclusion

The experience gained by ONERA, as well on technology work as on energetics, has led to design a new architecture of microturbine, of the cocoon type, in that two overlapped burners are placed in the center part, close to two turbines whereas the two compressors, driven by these turbines, face outside. The circuits that wrap the burners with appropriate materials allow us to design an enhanced thermodynamic cycle and to manage at best the heat transfer both inside and outside the micromachine.
The thermodynamic study of a one or two main stage turbomachines for a 20 Watts class ultra microturbine, shows that the compressor and turbine diameters must be smaller than 5 millimeters, if optimum specific speed is retained, except for the L.P. compressor with pneumatic coupling to the power turbine. It enlightens also the difficulty of actuating a disc type magnetic generator. In order to obtain a better thermal efficiency, the turbomachine discs must be small, on the other hand to obtain a sufficient electric power one needs a large disc size rotating at high peripheral speed. This conflict is overcome when using a 2 stage turbomachines by coupling the L.P. compressed air with the electric generator but in this configuration the thermal efficiency suffers from the product of 2 turbomachine efficiencies. A good solution, which pushes the difficulty towards technology, should be either to use a reductor or by coupling the L.P. main compressor directly to a cylindrical magnetic generator. An alternative solution should be to replace the conventional compressor by a Tesla compressor, well suited to the electric generator integration. Finally this study shows that it will be difficult to design a microturbine prototype of the class of 20 Watts and that a step on a class of 100W to 1 KW will be certainly necessary.

References

[1] Nagashima, T. “et al.”. Cycles and Thermal Integration Issues of Ultra-Micro Gas Turbines, Micro Gas Turbines, *RTO/AVT/VKI LECTURE SERIES*. 2005.
[2] Epstein, A. H.. Millimeter-Scale, MEMS Gas Turbine Engines. *ASME GT-2003-38866*.
[3] Shan, X. C. ”et al.”. A Silicon-Based Micro Gas Turbine Engine for Power Generation. *DTIP of MEMS and MOEMS, TIMA Editions/DTIP*.2006.
[4] Matsuo, T. “et al.”. Towards the Development of Finger-Top Gas Turbines. *IGTC 2003 TOKYO OS-103.*
[5] Okamoto, K, Nagashima, T.. Simple Numerical Modeling for Gasdynamics Design of Wave Rotors. *Journal of Propulsion and Power, Vol. 23, No. 1.* 2007.
[6] Iancu, F. “et al.”. Ultra-Micro Gas Wave Rotor Investigations. *PowerMEMS 2005, Tokyo.*
[7] Hara, M., “et al.”. Rotational infrared polarization modulator using a MEMS based air turbine with different types of journal bearing*. Journal of Micromech. Microeng., 13 (2003) 223-228.*
[8] Isomura, K., “et al.”. Effects of Reynolds Number and Tip Clearance on the Performance of a Centrifugal Compressor at Microscale*. ASME GT2006-90637.*
[9] Kang S., “et al.”. Micro scale radial flow compressor impeller made of silicon nitride. Manufacturing and performance. *ASME GT2003 38993.*
[10] Verstraete, T., “et al.”. Numerical study of the heat transfer in micro gas turbines, *ASME GT2006 90161*.
[11] Schneider, B. “et al.”. Ultra high energy density converter for portable power. *PowerMEMS2005, Tokyo.*
[12] Ribaud, Y., and Dessornes, O.. Energetic behavior of a MEMS microturbine concept. *CIEPLNE MASZYNY PRZEPLYWOWE (2005), No.128.*
[13] Guidez, J., “et al.”. Microcombustor for micro gas turbine engines”. *18th ISABE conference, Beijing, to be published.* 2007.
[14] Kozanecki, “et al”. Tilting pad bearings for microturbine*. PowerMEMS2006, Berkeley.*
[15] Ribaud, Y., “et al.”. High efficiency thermal engine. *Patent deposit n° 05.11426, INPI, France.* 2005.
[16] Rouzaud, S. Sizing of a turbo electric generator. *CNAM internal report, Paris.* 2007.
[17] Raisigel, H.. Planar magnetic micro turbogenerator. *PHD thesis, INPG LEG Grenoble.* 2006.
[18] Rodgers, C.. Specific Speed and Efficiency of Centrifugal Impellers. *Proc. ASME 25th Gas TurbineConference.* 1980
[19] Casey, M. V.. The effects of Reynolds number on the efficiency of centrifugal compressor stages. *ASME 84-GT-247.*
[20] Ribaud, Y.. Disc flow transmitters. *5th ISAIF Gdansk*. 2001.

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