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AIR-COOLING OPEN CYCLE WITH PERIPHERAL MACHINES

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ABSTRACT

Air-conditioning plants with air as working fluid in open cycle have been widely employed and they provide high performances in terms of compactness and low weight. In the on-aboard systems interesting developments have been obtained as well as in the land applications same good improvements can be expected. As a matter of fact the absolute lack of pollution problems and the compactness can make this kind of plant suitable for all that applications in which reliability, maintenance and easy service are fundamental qualifications. The paper analyses the possibility to employ the peripheral machines as compressor and turboexpander in order to improve the compactness and to reduce costs. A complete theoretical analysis of the thermodynamic cycle is followed by the schematic description of a small air cooling experimental prototype.

NOMENCLATURE

COP	Coefficient of performance
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
h	specific hentalpy
k	c_p/c_v
m	operating mass ratio
β	compression ratio
ε	heat exchanger effectiveness
η	efficiency
σ	$(k-1)/k$

Subscript

c	compression/compressor
e	espansion
t	turbine
0	ambient conditions
1	compressor inlet
2	compressor outlet
3	turbine inlet
4	turbine discharge

INTRODUCTION

The scientific and technical communities are looking for replacing the CFC's in the air conditioning plants, in order to solve the well known serious pollution problems. At the same time with the research of new refrigerant fluids, that is providing the first results, the study of alternative cycles or the close re-examination of refrigerating plants until now seldom employed because of their low performances is widely developing. This is the case of the air-cycle refrigeration systems that are more commonly used in the air conditioning of aircrafts than in surface and stationary applications because the lightweight compact equipment typical of air-cycle systems usually offset its inherently low efficiency [1,2,3,4].

The air-cycle refrigerating units have been deeply investigated in the literature with both dry and wet air as working fluid; the thermodynamic analysis pointed out the influence of the compressor, turbine and heat exchanger efficiencies on the global performance, and the influence of the expansion process on the final thermodynamic conditions of the air in the open cycle. An expansion that makes possible to maintain the steam in equilibrium with the dry air allows a lower temperature at the end of transformation expecially with high pressure ratios, affecting improvements in the coefficient of performance; such expansion can be more likely obtained in a turbine rotating at low angular speed.

Air-cycles air conditioning is not economic in many applications because of the high power required and the solution of the pollution problems cannot be the only reason for their larger diffusion. However they are used in specialized applications as the environmental control for remotely located temporary military bases, where efficiency is not the primary factor[3,4]. As a matter of fact in the above mentioned applications the ratio cost/benefit is accettable by virtue of particular conditions connected with the operative system; typical from this point of view is for instance the air conditioning into aircrafts, because the compression of low-pressure air from out-side the airplain to nearly standard atmospheric pressure is in any case operated. All these reasons suggest to look for new ways to reduce the ratio cost/benefit maintaining the compactness and the reliability.

THE PERIPHERAL MACHINES

In the environmental control of small rooms low flow rates are generally required so that the volumetric machines can be regarded as more suitable. However several problems are introduced by their operating charateristics, first of all the internal lubrication, when the working air is used directly to cool the space requiring refrigeration and therefore it cannot be contamined. In the air-cycle refrigeration units until now built centrifugal compressors and radial turbines have found wide applications; the low flow rates involve small dimensions of the impellers that, in turn, require high angular speeds in order to get the necessary value of tip velocity.

Peripheral machines are typical reaction machines with partialized admission and have a field of application limited to extremely low specific speeds so that they are directly competitive with certain types of volumetric machines. The low efficiency of the peripheral machines however causes the lack of interest in them, even though there are a number of construction features, above all in the case of low power, that make them economically convenient. Contrary to total admission turbomachines, in terms of construction, the layout of this type of partialized machine has a non cylindrical flow. The geometry of the machine consists of an open rotor with generally radial, blade-to-blade action, in front of which there is a channel in which the gas flows freely in radial direction; the flow is therefore within a toroidal space, so that the fluid is repeatedly in contact with the blades before being discharged. The case has suction and delivery ports which connect the outside with the channel, divided by dynamically sealed strippers.

The earliest recorded studies on these machines date back to the thirties and started in Germany [5] and in Japan [6] at the same time, after them several theoretical and experimental researches have been presented in literature. Previous papers by the Author [7,8,9] describe the early phases of the research on peripheral machines started by the Dipartimento di Energetica of Ancona University. As discussed in several papers, the low values of specific speed and ratio between wheel speed and spouting velocity, favor the application of the peripheral turbines in several cases; especially where comparatively small amounts of power are to be extracted at low rotative speeds. The low values of speed of revolution in these cases are particularly advantageous since usually gearing between the turbine and load can be eliminated thus offering sizable contructive simplifications and increased reliability [10,11]. Safety also benefits considerably since the runaway speed of peripheral turbines is extremely low, usually within the safe operating limits of bearings and turbine disk, so that a loss of the turbine load will not lead to disastrous failures, even when no overspeed trips are provided.

An other advantage of peripheral turbines looks therefore evident if one compares the areas of the distribution channel cross-sections: the straight cross-section area of the channel should in fact be compared with that of the nozzle of conventional turbines with partialized admission. It is evident that, even at partialized admission, the cascade of a conventional turbine need very close working tolerances, consequently increasing construction problems and costs. It is with regard to costs therefore that the peripheral turbine can have many possibilities of application, because of the more limited technological, production and construction material requirements. Other advantages have been considered in the case of the air-cycle cooling applications: the no internal lubrication requirements and the application of gas-bearings allow to introduce the air directly in the space requiring refrigeration without its contamination; moreover the typical low angular speed permit a slow expansion process with decreasing of the final temperature.

The proposal of employing peripheral machines in the air-cycle refrigeration plants has been realized in the prototype that is now under experimental testing; to choose the best values of the parameters the follow thermodynamic analysis has been developed.

THERMODYNAMIC ANALYSIS

A simple open air cycle system is shown schematically in Fig.1 while its corrispondent thermodynamic cycle is displayed on the temperature-entropy diagram in Fig.2. Refrigeration is obtained by three basic steps. Ambient air is firstly compressed and then the heat produced by the compression is removed by means of the ambient air itself; next air is cooled by extracting work from it as it expands through a turbine. So air leaves the turbine and enters the conditioned space at a low temperature T_4. Neglecting the work done to drive the fan which pulls air over the heat exchanger, the coefficient of performance of the cycle can be computed as follows:

$$COP = \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \qquad (1)$$

Neglecting pressure drops in the heat exchanger and considering dry air, the temperatures of air in the charateristic points of the cycle can be computed from the following equations :

$$T_2 = T_1 \cdot \left[1 + \left(\frac{\beta^{\sigma_c} - 1}{\eta_c}\right)\right] \tag{2}$$

$$T_3 = \varepsilon \cdot T_0 + T_2 \cdot (1 - \varepsilon) \tag{3}$$

$$T_4 = T_3 \cdot \left[1 - \eta_t \cdot \left(\frac{\beta^{\sigma_e} - 1}{\beta^{\sigma_e}}\right)\right] \tag{4}$$

Assuming $\eta_t = \eta_c = 0.6$, since these are the actual efficiencies of the peripheral machines so far designed, and assuming temperature of ambient air $T_0 = T_1 = 308$ K, the Fig.3 and Fig.4 charts can be obtained, where the coefficient of performance of the cycle (Fig.3) and the turbine discharge temperature (Fig.4) are plotted against the heat exchanger effectiveness for different values of the compression ratio. In order to improve the performance of the cycle a refrigeration of the compressor channel has been considered, lowering both the compression work and the heat exchanger burden.

This purpose will be attained by providing the compressor channel with fins and pulling ambient air against it by means of a fan driven by the turbine itself. At this step, to simplify the calculations, the compression phase has been considered isoentropic in every respect, while to compute the work done during an intercooled compression it should be necessary to evaluate, besides the enthalpy variation of the working fluid, the amount of heat removed during the compression.

Taking into account the pressure drop in the heat exchanger, as a function of its effectiveness, the coefficient of performance and the turbine discharge temperature have been obtained and they have been displayed versus compression ratio in Figs 5 and 6. The COP improvements are remarkable, compared with the previous system, expecially for those compression ratios that maximize the COP. Furthermore it must be remembered that in the first case the pressure drops have been neglected. As one can see from the diagrams, the maximum values of the COP can be attained at low compression ratios and this turns to a machines design advantage. Nevertheless the turbine-discharge temperature is still too high for conditioning purposes, particularly at low heat exchanger effectivenesses, so that the use of a so called "regenerative system" could be required.

In the regenerative system, some of the air discharged from the turbine passes back to cool the air coming out from the heat exchanger before it enters the turbine. In this way the temperature of air entering the turbine can be lower than would be possible by cooling with ambient air only. The regenerative system and its thermodynamic cycle, displayed on a T-s diagram, are shown in Fig.7 and Fig.8. The coefficient of performance of such a cycle can be computed as:

$$COP = \frac{m \cdot (h_1 - h_4)}{(h_{2'} - h_1) - (h_3 - h_4)} \tag{5}$$

where "m" is the operating mass ratio, that is the mass flow ratio between air entering conditioned space and air compressed.

Assuming $T_1 = T_0 = 308$ K and $\varepsilon = 0.7$, curves of Fig.9 and Fig.10 can be derived. They represent the coefficient of performance and the turbine discharge temperature as function of "m" for some values of the compression ratio. Taking m=1 one can read the COP and T_4 for an air cycle without regeneration. As shown in Fig.9 the COP reaches a maximum for a particular value of m which, for the compression ratios considered, ranges from 0,6 to 0,7. Fig.11 shows the maximum COP reachable adopting pressure ratios from 1.1 to 3, for some values of the heat exchangers effectiveness.

The experimental prototype is schematically showed in Fig.12; the two cross sections show both the air-cooled and the ambient air ways and the compact arrangement of the unit.

CONCLUSIONS

The proposal of the peripheral machines to compress and to expand air in an open refrigerating air-cycle appears trustworthy even if the COP values have not been improved as regards those obtainable with radial turbomachines. The compactness and the low speed of revolution allow to reduce the size problems for the system allocation and to rely on low level sound emissions that instead can be troublesome adopting the traditional machines. The first phase of the experimental program will provide interesting data in this field; instead the economic evaluation of the construction costs has been already developed in comparison with the traditional air conditioning plants at the same refrigeration power. The proposed group is characterized by the considerable reduction in the manufacturing cost and by very low maintenance costs which the high running cost consequent up on low value of the COP is in contrast with.

Waiting for the experimental results, above all as for the hypothesis of the heat exchange during the compression process, the high reliability and the no pollution property at low cost can be considered interesting peculiarities for some particular line of market.

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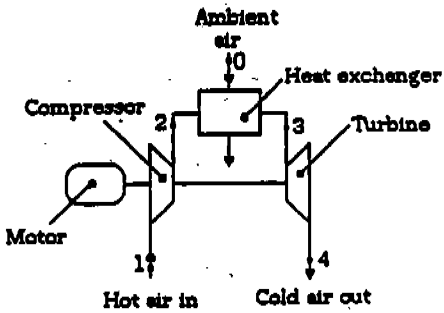


Fig.1- Symple open air cycle diagram

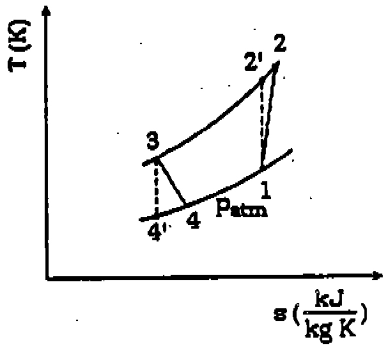


Fig.2- Thermodynamic air cycle T-s diagram

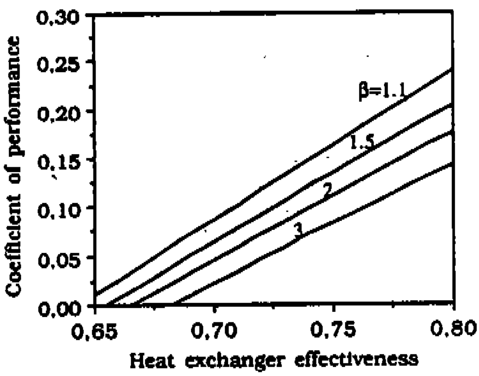


Fig.3 - Coefficient of performance vs heat exchanger effectiveness

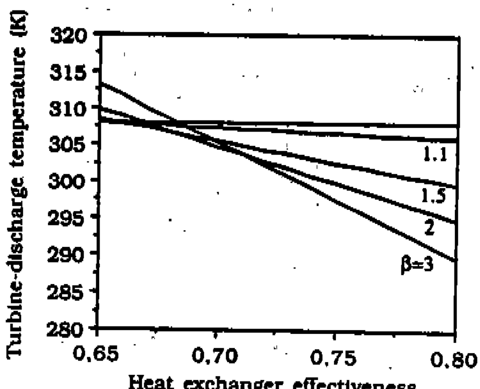


Fig.4 - Turbine discharge temperature vs heat exchanger effectiveness

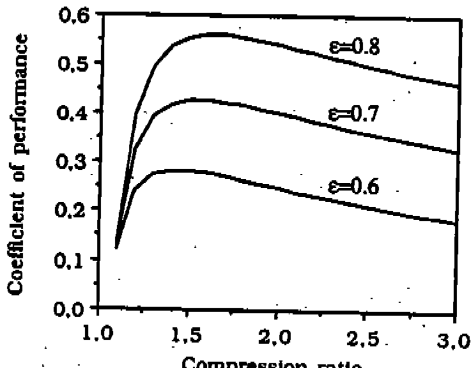


Fig.5 - Coefficient of performance vs compression ratio

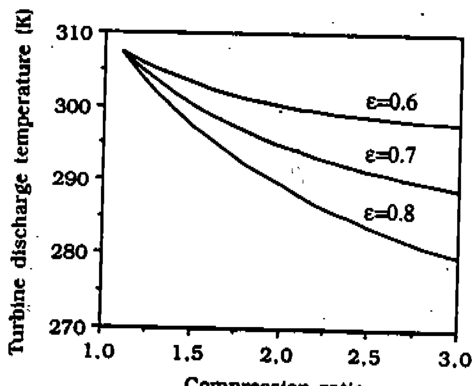


Fig.6 - Turbine discharge temperature vs compression ratio

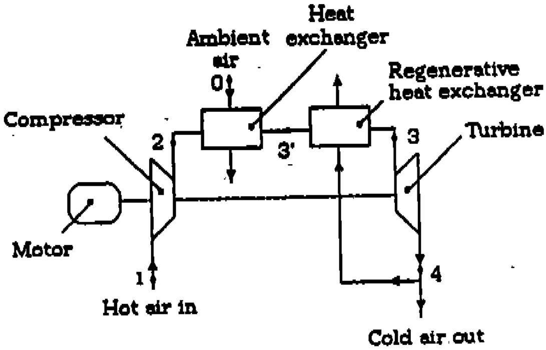


Fig.7- Regenerative air cycle diagram

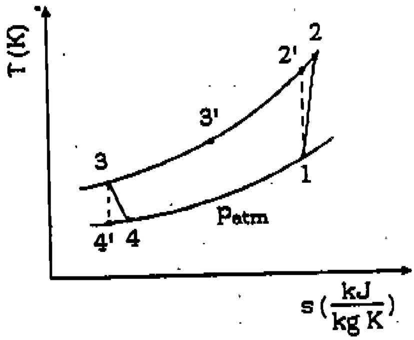


Fig.8 - Regenerative air cycle T-s diagram

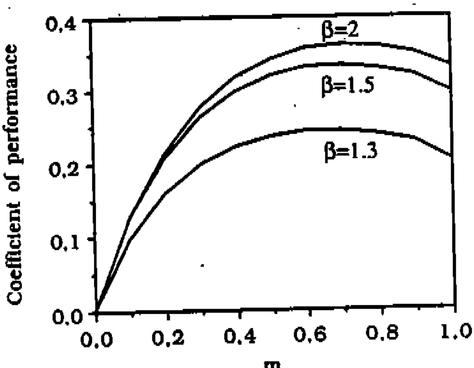


Fig.9 - Coefficient of performance vs operating mass ratio for different values of the pressure ratio, in the regenerative cycle

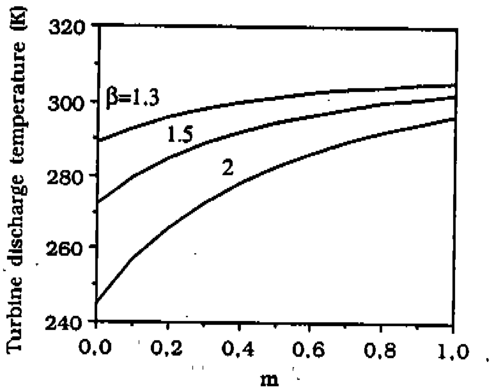


Fig.10 - Turbine-discharge temperature "T4" vs operating mass ratio for different values of the pressure ratio, in the regenerative cycle

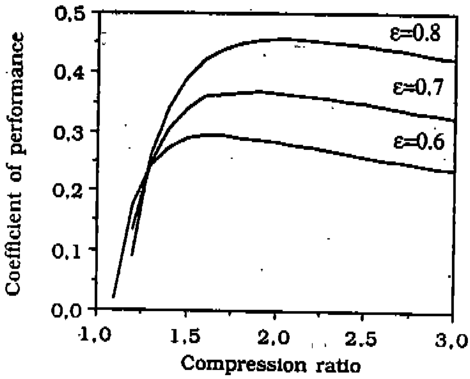
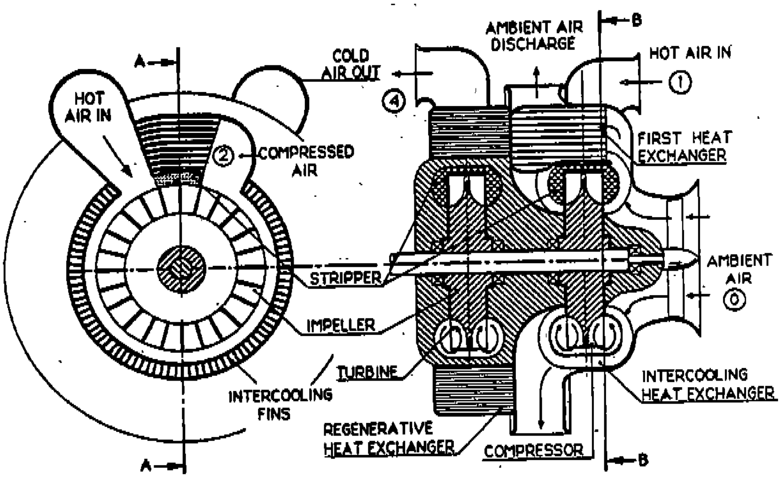


Fig.11- Maximum coefficient of performance vs pressure ratio for different values of the heat exchanger effectiveness, in the regenerative cycle.



CROSS SECTION B-B *CROSS SECTION A-A*

Fig. 12- Scheme of experimental prototype of air-cooling regenerative unit
The circled numbers refer to the diagram of Fig. 7.