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Air cycle air conditioning: analysis of different configurations.

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ABSTRACT

Air conditioning for passenger transportation needs a global approach. For a railway operator, maintenance costs can overcome energy costs and air conditioning failure can seriously affect customer satisfaction. Sealed refrigeration circuits introduce a failure mode, while air cycle refrigeration systems, although less efficient, can be conceived as open cycles, avoiding the need for a closed circuit.
From the experience accumulated in existing air cycles for terrestrial use, low pressure cycles seem best suited for this application. These systems expand ambient air through a turbine, lowering its temperature. This cold flow draws heat from the passenger compartment air through a heat exchanger and then is brought back to ambient pressure by a compressor. Part of the compression work comes from the turbine.
In this paper, some basic alternative configurations are analyzed. Air humidity is accounted for. Heat recovery is pursued.

1. INTRODUCTION

The use of air compression/expansion to obtain a cooling effect was one of the very first air conditioning techniques (Gladstone 1998). John Gorrie wrote in 1848 "If air were highly compressed, it would heat up by the energy of compression. If this compressed air were run through metal pipes cooled with water, and if this air cooled to the water temperature was expanded down to atmospheric pressure again, very low temperature could be obtained, even low enough to freeze water in pans in a refrigerator box". A patent was issued for his invention in 1851, but he did not manage to commercially exploit his achievement. Soon after, in the early 20th century, the compressed-air ice-making machine was widespread aboard ships.
Nowadays air-cycle refrigeration can be seen as a valid method to replace synthetic refrigerants. However, air has a further significant advantage over any other proposed zero ODP, low GWP fluid. Using air means that the fluid evolving within the circuit is the ambient fluid itself. Hence the option, unique to air cycles, to build an open machine.
The main drawback of air as a refrigerant fluid is that it behaves as a ideal gas. Vapors allow isothermobaric transformations and hence give working cycles resembling (partially at least) Carnot cycles. Isobaric transformations of a perfect gas extend over a wide temperature range, making heat exchange with constant temperature sources highly irreversible. This heavily impacts on the plant COP. Air cycle air conditioning can be competitive only if reduced maintenance costs compensate for increased energy consumption.
On the other hand, the air cycle can be opened on its ambient pressure side, avoiding the irreversibility related to the heat exchanger. Besides, some energy recovery opportunities arise from air humidity. These opportunities and a careful cycle optimization can reduce the COP gap between air and vapor cycles.

2. AIR CYCLE THERMODYNAMIC ANALYSIS

2.1. Ideal cycle

Given two heat sources, the low temperature one being the cold space and the high temperature being the surrounding ambient, the air cooling cycle can be configured in several ways. There are two main options:

- expand the ambient air through a turbine and so obtain a cold, low pressure, stream which can subtract heat from the cooled space. Air is extracted from the low pressure side by a compressor and sent back to the ambient, where it discharges the collected heat plus the work done on it.
- compress the air in a compressor and discharge heat to the ambient. Air is then expanded, producing a cold stream at ambient pressure, which can be directly introduced into the cooled space.

Actually these options may be seen as two versions of a closed cycle with a hot and a cold heat exchanger. The air mass evolving in this closed circuit, and hence the cycle pressure values, can be changed as far as allowed by the circuit components. Cycle temperatures, and hence its efficiency, remain unaffected.

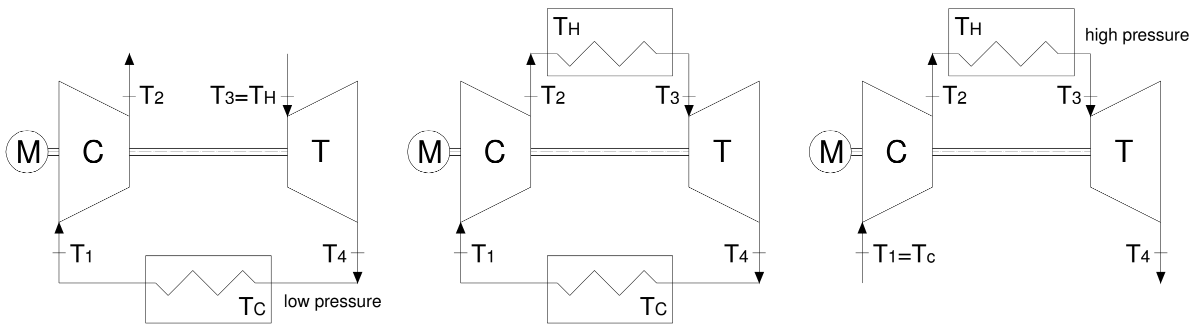

Fig. 1 – Basic cycle options: low pressure (left), closed (center) and high pressure (right)

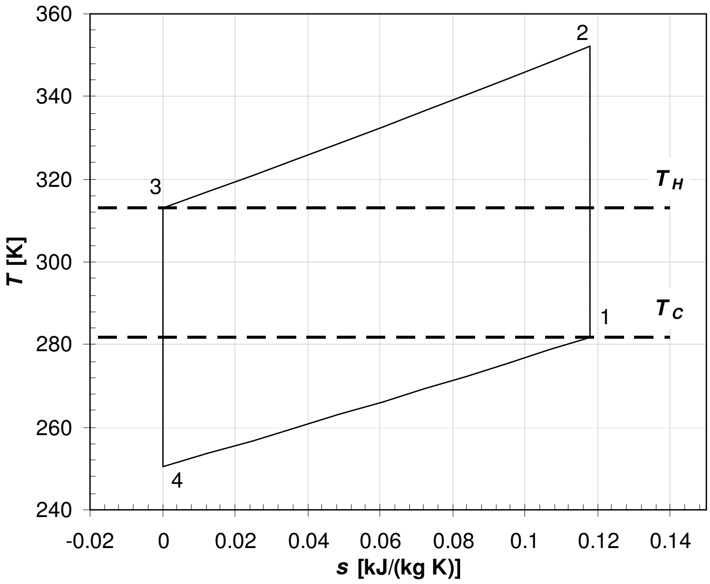

Fig. 2 – Ideal cycle with $\tau = 0.9$ and $q = 0.1$

Detailed thermodynamic analysis of the air cycle can be found for example in (Chen *et al.* 1998). For reversible compression and expansion, the temperature ratios are related to compression ratio $\beta = P_2 / P_1$ by $T_2 / T_1 = T_3 / T_4 = \beta^{\lambda}$, being $\lambda = R/c_p$.
Hence the ideal cycle has:

$$COP = \left(\beta^{\lambda} - 1\right)^{-1} \tag{1}$$

However, in order to effectively subtract heat from the cold space, compression ratio cannot be brought below a minimum value depending on the ambient temperature T_H and cold space temperature T_C. The cooling load Q can be expressed in dimensionless form as:

$$q = \frac{Q}{m \cdot c_p \left(T_H - T_C\right)} = \frac{\tau - \beta^{-\lambda}}{1 - \tau} \tag{2}$$

where τ is the ratio T_C / T_H. and m is the mass flow rate within the cycle. Hence, to have a positive cooling load, β^{λ} must be greater than $1/\tau$. A more significant expression of the cycle efficiency is then:

$$COP = \left[\left(\tau - q + q\tau\right)^{-1} - 1\right]^{-1} \tag{3}$$

This relation is plotted in Figure 3, where it is evident how cooling load decreases with q and τ. Very soon, especially at higher values of τ, the compression ratio needed to obtain high q climbs to unreasonable values.

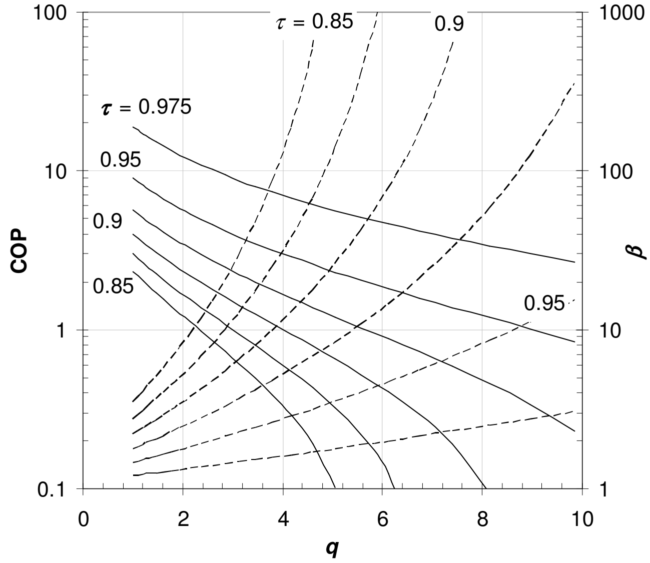

Figure 3 – COP (solid line) and β (dashed line) as a function of temperature ratio τ and cooling load q for T_H = 40°C and λ = 0.2857

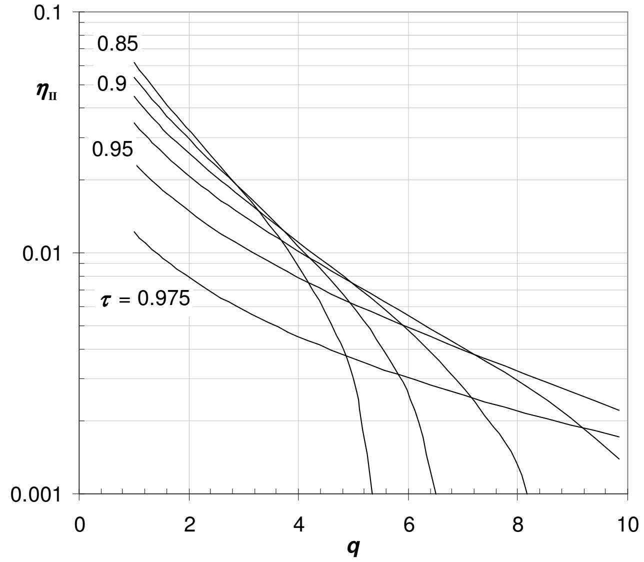

Figure 4 – Second law efficiency as a function of temperature ratio τ and cooling load q for T_H = 40°C and λ = 0.2857

The lower limit for β can be overcome by thermal recuperation, i.e. by introducing a heat exchanger between cold gas at point 1 and hot gas at point 3. In this way, point 1 can be raised above temperature T_C and point 3 can be lowered below temperature T_H. Net work is hence reduced along with β, but cooling load is reduced as well. Efficiency decreases, as one could expect, considering that irreversibility is added by the finite temperature difference across the thermal recuperator.

A second law efficiency can be used to evaluate the air cycle performance with respect to the thermodynamic limit imposed by the heat source temperatures. This efficiency can also be calculated by exergy analysis, as done by Chen and Su (2005).

$$\eta_{II} = \frac{COP}{COP_{Carnot}} = \frac{Ex_{out}}{Ex_{in}} = \frac{\tau - q(1-\tau)}{\tau(1+q)} \tag{4}$$

Alefeld (1987) questioned the use of exergy, due to the need to define an arbitrary environmental state, while the direct use of Second Law is independent of this state.

Second law efficiency is shown in figure 4, where the influence of the temperature ratio τ is reversed. This proves that the intrinsic deficiency of the air cycle tends to disappear as the temperature of the cold space is lowered, making this cycle competitive in the field of refrigeration more than in air conditioning.

2.2. Ideal cycle with finite size heat exchangers

First of all let us generalize the analysis to finite heat capacity flows at the hot and cold side of the system. The effect of heat exchanger finite size on refrigerators was dealt with for example by Grazzini (1993). *COP* becomes a function of heat exchanger effectiveness ε. This is defined according to Kays and London (1955)

$$\varepsilon_H = \frac{mc_p(T_2 - T_3)}{(mc_p)_{min}(T_2 - T_H)}; \qquad \varepsilon_c = \frac{mc_p(T_1 - T_4)}{(mc_p)_{min}(T_c - T_4)} \tag{5}$$

The mc_p factors can be elided as far as the minimum heat capacity in the exchanger pertains to the cycle fluid, as will be assumed from now onward. The dimensionless cooling load becomes:

$$q = \frac{(\tau - \beta^{-\lambda})(1-\tau)^{-1}}{1/\varepsilon_H + 1/\varepsilon_C - 1} \tag{6}$$

This result is remarkably similar to (2), reduced by a factor depending on heat transfer resistance at hot and cold heat sources. This factor was used for example by Klein (1992), though in a different form. If the cycle is open to the environment on the hot or cold side, heat transfer resistance on that side is eliminated.
Even more straightforward is the result concerning COP, which retains expression (1) unchanged. However, compression ratio changes:

$$\beta^{\lambda} = \left[\tau - (1-\tau)\left(\frac{1}{\varepsilon_H} + \frac{1}{\varepsilon_C} - 1 \right) q \right]^{-1} \tag{7}$$

Hence, the relation between COP and q becomes:

$$COP = \left\{ \left[\tau - (1-\tau)\left(\frac{1}{\varepsilon_H} + \frac{1}{\varepsilon_C} - 1 \right) q \right]^{-1} - 1 \right\}^{-1} \tag{8}$$

The second law efficiency is:

$$\eta_{II} = \left\{ \left[\tau - (1-\tau)\left(\frac{1}{\varepsilon_H} + \frac{1}{\varepsilon_C} - 1 \right) q \right]^{-1} - 1 \right\}^{-1} \left(\tau^{-1} - 1 \right)^{-1} \tag{9}$$

Figures 5 shows the effect of heat exchanger effectiveness on COP and η_{II}, being $\varepsilon_H = \varepsilon_C = \varepsilon$ and $\tau = 0.9$. Noteworthy are the negative values of COP above a limiting q that increases with ε.

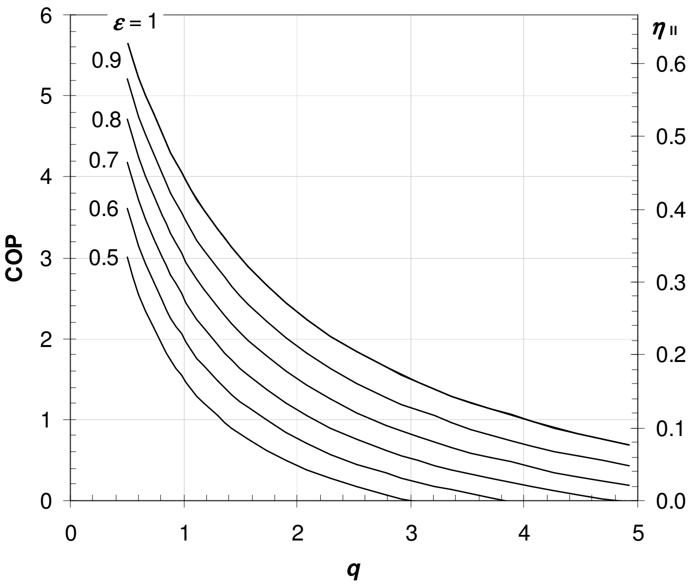

Figure 5 – COP and Second law efficiency as a function of q and ε for $T_H = 40°C$ and $\tau = 0.9$

2.3. Compressor and turbine irreversibility

In order to account for compressor and turbine irreversibility, isentropic efficiency may be used. However, this efficiency depends on compression ratio, which is used as a variable parameter in the present analysis. Therefore we prefer to use polytropic efficiency, which measures thermodynamic performance of turbomachinery filtering the effect of compression ratio. From the measure of inlet and exit temperature of a compressor/turbine of given compression ratio, the exponent of the equivalent polytropic transformation can be evaluated from:

$$\lambda_c = \frac{ln(T_2/T_1)}{ln(\beta)} > \lambda \quad \text{for compression,}$$
$$\lambda_e = \frac{ln(T_3/T_4)}{ln(\beta)} < \lambda \quad \text{for expansion.} \tag{10}$$

The exponents are related to the polytropic efficiency as follows:

$$\eta_{pol,c} = \lambda / \lambda_c \quad \text{for compression and} \quad \eta_{pol,e} = \lambda_e / \lambda \quad \text{for expansion.} \tag{11}$$

Radial compressor and turbines commonly used for supercharging internal combustion engines have reached isentropic efficiency $\eta_{is} = 0.8$ for compression ratio around $\beta = 3$. Therefore, for a compression starting at ambient temperature T_1 = 300 K the exit temperature would be T_2 = 438 K and the polytropic exponent would be $\lambda_c = 0.345$, giving a polytropic efficiency $\eta_{pol,c} = 0.828$.

Assuming the same polytropic efficiency for expansion, we have $\lambda_e = 0.237$.

The cycle can be calculated using these exponents and the reduction in q, COP and η_{II} can be evaluated. Two noticeable features emerge from Figure 6, showing COP v/s q at $\eta_{pol,c} = 0.828$ and $\varepsilon_H = \varepsilon_C = 0.7$:

- COP has now a well defined maximum around $q = 0.6$, corresponding to $\beta = 3.9$
- q does not reach a value of 2. Actually, the last points of this curve refer to unreasonably high values of β and were reported for sake of completeness. At higher β, q starts to decrease.
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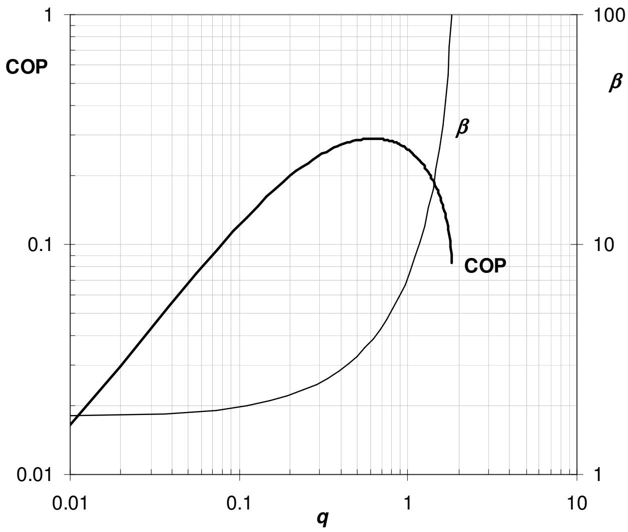

Figure 6 – COP v/s q at $\eta_{pol,c} = 0.828$; $\varepsilon_H = \varepsilon_C = 0.7$

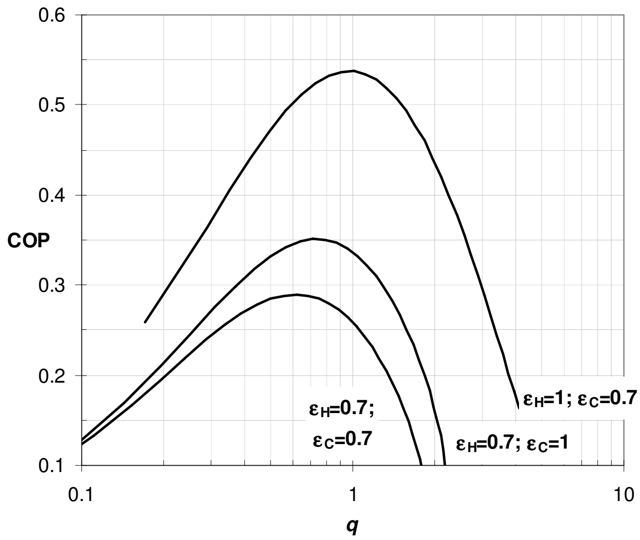

Figure 7 – COP v/s q with different values of heat exchanger effectiveness.

Analytic results are rather complicated. For a given heat source temperature ratio, the temperature of cycle point 1 can be evaluated as a function of compression ratio, polytropic compression and expansion exponents and heat exchanger effectiveness as follows:

$$\frac{T_1}{T_H} = \frac{\beta^{\lambda_e}\varepsilon_c \tau + \varepsilon_H (1-\varepsilon_c)}{\beta^{\lambda_e} - (1-\varepsilon_c)(1-\varepsilon_H)\beta^{\lambda_c}} \tag{12}$$

Clearly, for $\varepsilon_H = \varepsilon_C = 1$ and $\lambda_c = \lambda_e = \lambda$, it is $T_1 / T_H = \tau$. The other cycle temperatures can be found from T_1:

$$T_2 = \beta^{\lambda_c} T_1 \; ; \qquad T_3 = \varepsilon_H T_H + (1-\varepsilon_H) T_2 \; ; \qquad T_4 = \beta^{-\lambda_e} T_3 \tag{13}$$

Hence, the dimensionless cooling load and COP are:

$$q = \frac{T_1 - T_4}{T_H - T_C} \; ; \quad COP = \left(\frac{T_2 - T_3}{T_1 - T_4} - 1 \right)^{-1} \; ; \tag{14}$$

Going back to Figure 1, one may ask if eliminating one of the two heat exchangers can improve this result and, if so, which one gives the greatest improvement. Figure 7 shows a comparison between the two options. The advantage of eliminating the high temperature heat exchanger is evident. Elimination of the low temperature heat exchanger gives a significantly lower gain. This circumstance can be explained considering

that heat release through the high temperature heat exchanger is greater, being equal to cooling load plus work, plus the effect of irreversibility.
A drawback of the cycle without hot exchanger is that the cold exchanger works at low pressure and, hence, density. Heat exchanger flow area must be wide enough to accommodate for this low density flow, otherwise pressure loss will become excessive. Anyway, given the thermodynamic superiority of the "low pressure" cycle, the analysis that follows will focus on this option.

2.4. Effect of air humidity

One of the major drawbacks of an open air cycle is humidity, which is inevitably drawn inside the cycle with external air. If anywhere in the cycle the temperature falls below the dew temperature, water condensation may occur. This can cause mechanical problems, mainly in the turbine, and can modify the cycle temperature as soon as the latent heat of condensation is released within the air flow. When this happens, air can no longer be modeled as an ideal gas. Water condensation in air turbines is analyzed in (Roumeliotis, 2006).
In principle, water condensation could be accounted for using water-steam thermodynamic tables or routines. Expanding air from T = 40°C and relative humidity φ = 0.5, along a polytropic with exponent $\lambda_e = 0.237$, condensation should start at 23.8°C, corresponding to an expansion ratio β = 1.25.
However these data refer to equilibrium properties, while expansion in a turbine is typically very fast. Most likely, vapor will not condensate within the turbine, unless expansion goes very deep inside the saturated water area. Water vapor will exhibit a metastable behavior. Unfortunately, data about this behavior are still quite scarce. Furthermore, existing computer codes do not incorporate metastable data.
In what follows we assume that condensation does not occur within the turbine. Fluid properties are constant across the expansion. Downstream of turbine exit, equilibrium is attained and water condenses. This water is difficult to evacuate from the circuit, being at low pressure. As soon as the temperature rises within the exchanger, water will evaporate again. Phase change occurs within the heat exchanger and the definition of its effectiveness should be modified accounting for the presence of a possible pinch point.
In order to avoid icing, air should not be expanded below 0°C.

3. APPLICATION TO AIR CONDITIONING

On the other side of the heat exchanger there is a finite air flow, at ambient pressure, which must be delivered to a livable space. Depending on system configuration, ambient air will be mixed with recycled air upstream or downstream of the heat exchanger.
A unitary air mass flow rate for conditioned ambient is assumed. The need to dehumidify air down to a specified title $x = m_{H2O} / m_{air}$ is a further constraint for the system. Once the dew point is reached within the heat exchanger, vapor partial pressure P_{H2O} will be equal to saturation pressure at the local temperature $P_s(T)$. Given x, vapor partial pressure P_{H2O} is:

$$P_{H2O} = \frac{xP}{0.622 + x} = P_s(T) \tag{15}$$

Saturation pressure can be calculated with relation (3) chapter 6 of ASHRAE Fundamentals (2005). This relation has been inverted to give $T_s(P_{H2O})$ yielding:

$$T_s(P_{H2O}) = 0.00183807\,P_{H2O}^{\;4} + 0.0627662\,P_{H2O}^{\;3} + 0.966395\,P_{H2O}^{\;2} + 14.0773\,P_{H2O} + 7.32097 \tag{16}$$

Assuming, for example, a required condition in the passenger compartment T_{in} = 27°C ; φ_{in} = 0.5, the title is x=0,0111. If vapor is produced within the passenger compartment and must be withdrawn, cool air at admission must have a lower title. In a typical situation (bus or train) we can assume x=0,01 that gives a temperature T_S = 14°C. If this value is too low for the specific application, re-heating can be easily performed recovering some energy from the hot air at compressor exit.
At heat exchanger entrance, ambient air is mixed with cooler recycled air from the passenger compartment. The ratio of the two flows can be specified through a recirculation ratio $\mu = m_{\mathrm{rec}} / m_{\mathrm{amb}}$ and depends on the air quality requirements. As μ varies from 0 to 1, the air temperature T_{mix} at heat exchanger entrance varies from ambient temperature T_{amb} to passenger compartment temperature T_{in}. Ambient conditions T_{amb} = 40°C and φ_{amb} = 0.5 are assumed.

Polytropic efficiency of compressor and turbine is $\eta_{pol,c} = 0.828$. Instead of heat exchanger effectiveness, a minimum temperature difference of 10°C is imposed between the flows. Therefore, $T_1 = T_{mix}$ -10. On the other side of the exchanger, the condition $T_4 \le T_s$ -10 poses a lower limit to compression ratio. Turbine exit temperature T_4 is calculated accounting for vapor condensation. At point 4 the fluid is an equilibrium mixture of air, vapor and liquid water.
The heat subtracted by the cooling cycle, per unit mass of dry air delivered, is:

$$q^* = \mu J_{in} + (1-\mu) J_{amb} - J_s \tag{17}$$

where J is the enthalpy per unit mass of dry air. Once the enthalpies in 4 and 1 are calculated, the cycle mass flow rate can be evaluated by heat exchanger energy balance. Increasing the compression ratio above its lower limit decreases the cycle mass flow rate and hence the cross section of the circuit.
Cool air exhausted from the passenger compartment can be used in the cooling cycle, lowering the temperature and the title at turbine entrance (Figure 8). As recirculation ratio increases, the available exhaust air decreases and the turbine inlet temperature increases. Figure 9 shows COP as a function of β and μ.

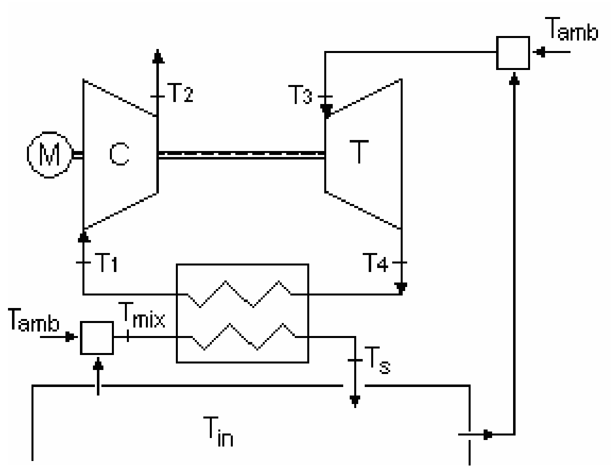

Figure 8 – Scheme of plant with recuperation of exhaust air

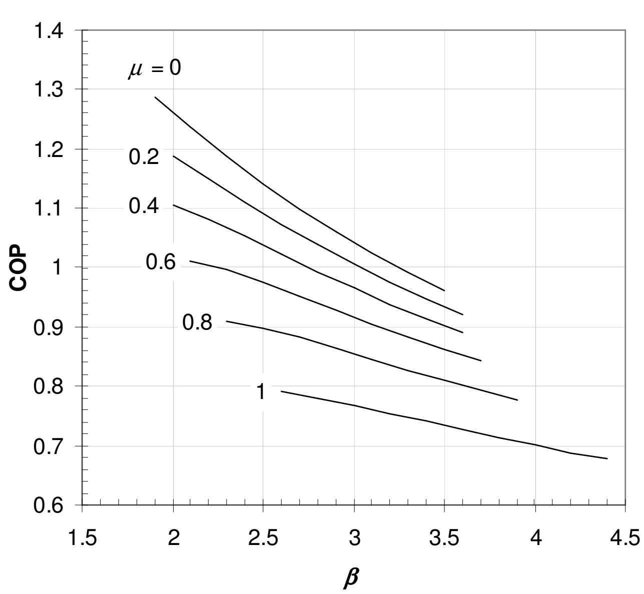

Figure 9 – COP v/s β as a function of recirculation ratio

The effect of thermal recuperation from the exhaust cold flow is evident. At low values of μ, temperatures T_1 and T_3 become very close, i.e. the system temperature ratio τ approaches unity. This explains the fairly high values of COP. On the other hand, as the recirculation ratio decreases, the cooling load increases. Therefore the power consumed by the system is actually higher at lower μ.
For a given recirculation ratio, in the permitted range of compression ratio, the COP is monotonically decreasing. Therefore, for what concerns energy efficiency, the minimum allowable β should be selected. However, this choice gives the biggest physical dimensions of the system.

CONCLUSIONS

A thermodynamic analysis of a air cycle refrigerator has been proposed. This analysis is fairly general and has shown that an open cycle has a higher COP, especially if the cycle is open on the hot side. Subsequently, the analysis was applied to an air conditioning system. Even if, in principle, the air cycle is better suited to low temperature applications, the air conditioning system has an acceptable COP and offers a very simple and effective way of recovering energy from the cold exhaust air. Lower performances in terms of energy efficiency are overcome by the elimination of fluid leakage and related maintenance expenditures. Further energy recovery options can be pursued from the condensed water.

NOMENCLATURE

c_p	Constant pressure specific heat	λ	Polytropic exponent
COP	Coefficient of performance	μ	recirculation ratio
Ex	Exergy	τ	Temperature ratio T_C / T_H
J	Enthalpy per kg of dry air	φ	Relative humidity
m	Mass flow rate		
P	Pressure		Subscripts
Q	Heat	amb	Ambient
q	Dimensionless heat	C	Low temperature thermal source
q^*	Subtracted heat per kg of dry air	c	Compression
R	Ideal gas constant	e	Expansion
s	Entropy	H	High temperature thermal source
T	Temperature	II	2^{nd} Law of Thermodynamics
		is	Isoentropic
	Greek letters	mix	Mixture
β	Compression ratio	pol	Polytropic
ε	Heat exchanger effectiveness	s	Saturation
η	Efficiency		

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