FEASIBILITY OF AIR CYCLE SYSTEMS FOR BUILDING AIR CONDITIONING SYSTEMS

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1. SUMMARY

Air cycle systems, based on the Joule-Brayton thermodynamic cycle, are promising for specific applications in refrigeration, like blast freezing. The techno-economic feasibility of an air cycle combined cooling and heat pump system has been investigated, for use in a building air conditioning system. In that case, the air cycle acts as a combined cooling, heating, ventilating, humidifying and dehumidifying unit.
Relatively simple system calculations have been carried out, using realistic annual cooling and heating demands of a building, derived from complex simulation models.
As the main result of the study, air cycle systems for air conditioning applications seem to be able to compete with conventional all-air systems in the field of energy, energy costs and investment costs.

2. INTRODUCTION

Due to concern about the damage that CFCs do to the environment, the search for alternative refrigeration systems has begun. Air is the most natural refrigerant. It is harmless to the environment, human beings, and food, and it is free. It meats nearly all criteria for a refrigerant that is environmentally benign. In a recent study /1/ the air cycle concept for heat pumps, refrigeration systems and air conditioners was further developed in order to recognize applications where an air cycle can be energetically and economically competitive to a vapour compression cycle.
One of the promising applications concerns the heating and cooling of buildings in combination with humidity control. In air conditioning systems the humidity control of the air results in higher heating and cooling demands than strictly needed to achieve the desired room temperature. In air cycles heating and humidification, requested in winter time, go side-by-side. The same goes for cooling and dehumidification, which are required under summer conditions. These and other considerations make it worth while to investigate whether the energetic performance of the air cycle, traditionally the most prevalent disadvantage, turns out to be an additional advantage in this application area. In this paper this investigation is summarized.

3. WORKING PRINCIPLE OF THE AIR CYCLE

In principle the air cycle is a Joule-Brayton cycle with a compression stage, an isobaric heat rejection, an expansion stage and an isobaric heat acceptance. Various configurations are possible /2/. For this study the fully open configuration with two stage compression and intercooling was used. This configuration, also chosen for the test facility at TNO /3/, consists of an electrically driven first stage compressor and a second stage compressor. directly driven by an expander (turbocharger), resulting in a simple and reliable configuration. Moreover, the first stage compressor and the turbocharger are commercially available items, reducing the investment and engineering costs. In the case of air conditioning the air cycle must be suitable to work both as a refrigeration machine and as a heat pump. This leads to the configuration shown in Fig.1. The normal numbers correspond with the cooling mode, the numbers with an apostrophe with the heating mode.

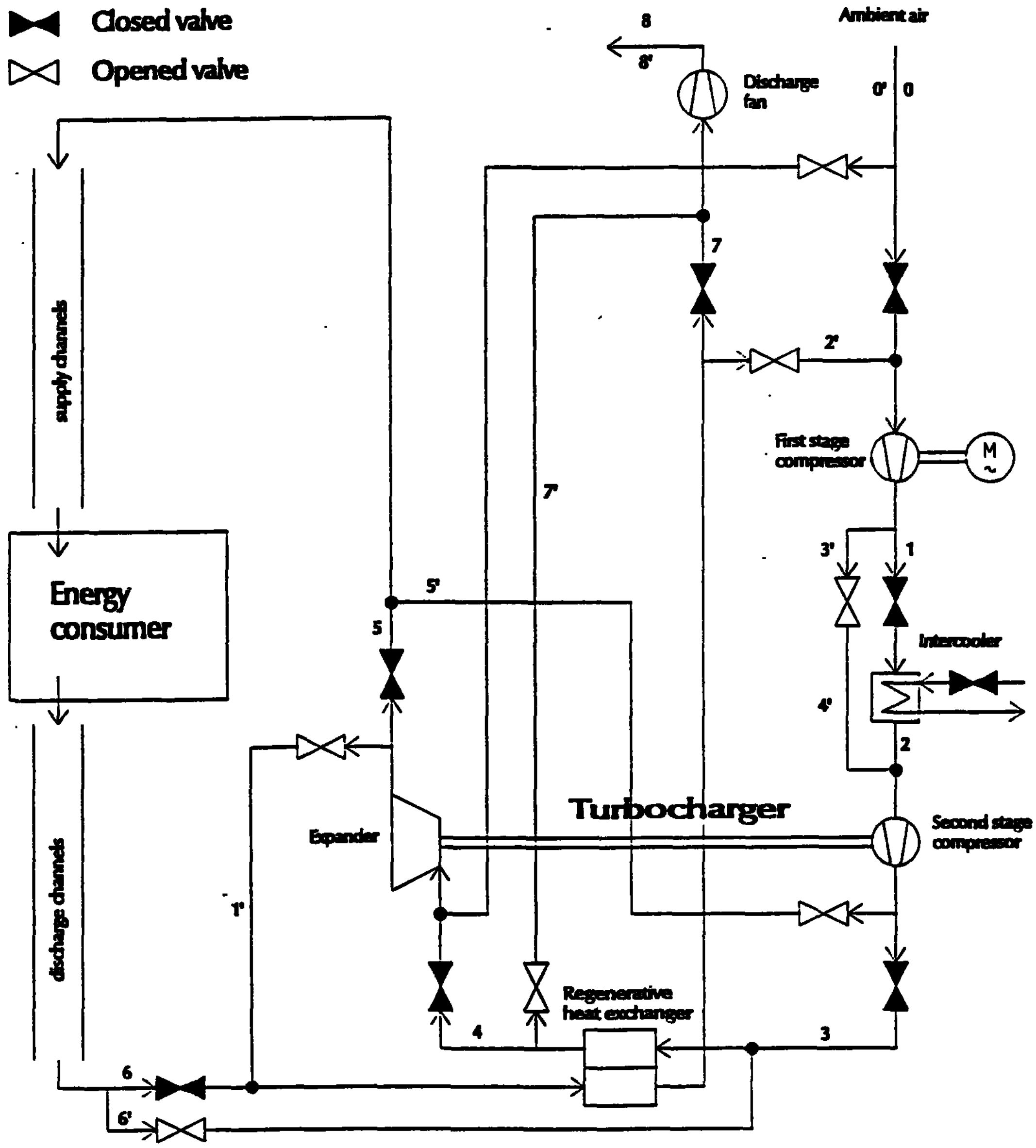


Figure 1 *Possible configuration of reversible air cycle system for air conditioning (operation in heating mode is shown)*

Cooling mode

Ambient air (point 0) is pressurized by an electricity driven compressor. The pressure rise is accompanied by an undesired temperature rise. The air (point 1) is cooled down in the intercooler to point 2. The air is further compressed in the compressor of the turbocharger. The air (point 3) is cooled down in a regenerative heat exchanger to such an extent that water vapour is condensed. In the heat exchanger a bed of spheres absorbs the

heat and moisture. In air conditioning systems. the humidity of ambient air under summer conditions is too high, and dehumidification is applied. With an air cycle this automatically takes place, as just described.
The air (point 4) expands in a turbine to achieve the lowest temperature in the air cycle. Since the air leaves the regenerative heat exchanger at approximately saturation conditions (100% RH=relative humidity), the temperature fall in the turbine results in a further dehumidification (in spite of the pressure fall). The air (point 5) with the desired pressure and temperature, is ready to perform its cooling task. The air returns at a higher temperature (point 6), and absorbs the heat and moisture previously stored in the bed of the regenerative heat exchanger (point 7). A fan releases the air to the environment (point 8).

Heating mode
In the heating mode the ambient air first meets the turbine. The pressure and temperature fall in the turbine enable the air to rise in temperature in the regenerative heat exchanger and the two compressors. In the regenerative heat exchanger the air coming from the turbine, absorbs not only the heat. but also the moisture. For air conditioning applications. ambient air under winter conditions is too dry for reasons of comfortability and humidification is needed. An air cycle takes automatically care of that.

It is interesting to investigate whether these qualitative considerations can be supported quantitatively. A reference climate. a reference building and a reference air conditioning system are the basis for this quantification.

4. REFERENCES FOR AIR CONDITIONING SYSTEM, BUILDING AND INDOOR AND OUTDOOR CLIMATE

4.1 Introduction
In an earlier study /4/ the most frequently found air conditioning systems are described and possibilities for energy savings explored. The energy saving potentials mentioned in /4/ are based on calculations which are generally reported in /5/. The reference building and (indoor and outdoor) climate will be applied in this feasibility study.

4.2 Reference Building
The reference building is shown in Fig.2. It concerns a medium sized, relatively shallow office building with seven floors. The volume is 22,100 m^3. The rooms are orientated north-south, with two distinguishable zones (east and west). Each zone consists of 105 equally sized modules of 3,5 m wide, 5,8 m long and 2,7 m deep.

The reference building can be more specified in terms of glazing and sun-blind. In this study outer walls with 40% double glazing and inside sun-blind are assumed. This reference building is an existing Dutch office building.

4.3 Reference Outdoor Climate
The reference climate is given by hourly values for sunray, air temperature and air humidity as measured by the Dutch meteorological institute KNMI during the period 1 April 1964 to 31 March 1965. The design conditions of the outdoor climate for the air conditioning system are as follows:

	Summer	Winter
Air temperature (°C)	28	-8
Air humidity (g/kg)	14 (59% RH)	1 (52% RH)

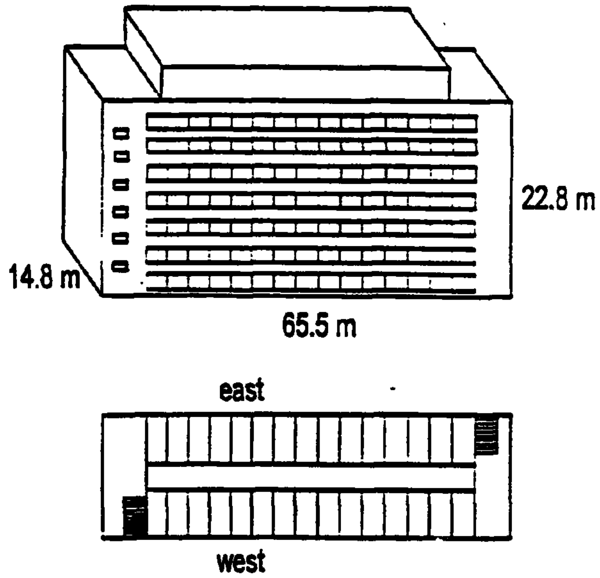


Figure 2 *Reference building*

4.4 Reference Indoor Climate
Besides the outdoor climate the internal heat loads play an important role. Each module is occupied by two persons with each a heat load of 70 W. The heat load of lighting and office equipment (computers etc.) is 30 W/m^2 (ground surface) and it is assumed that 30% of this heat load is carried away with the return air flow from the module. The internal heat load comes to 566 W per module.

The design conditions of the indoor climate are as follows:

	Summer	Winter
Air temperature (°C)	22	22
Air humidity (g/kg)	8.2 (50% RH)	7.4 (45% RH)

4.5 Energy demand of the building
The aforementioned starting points result in energy demands of the building in terms of heating and cooling. These demands are shown in Table I.

TABLE I: Energy demands of the building

	Orientation	
Energy demand per module	East	West
Heating, max. per mudule (W)	1,260	1,260
Heating, annual per module (kWh/yr)	461	449
Cooling, max. per mudule (W)	1,482	1,672
Cooling, annual per module (kWh/yr)	873	1,100

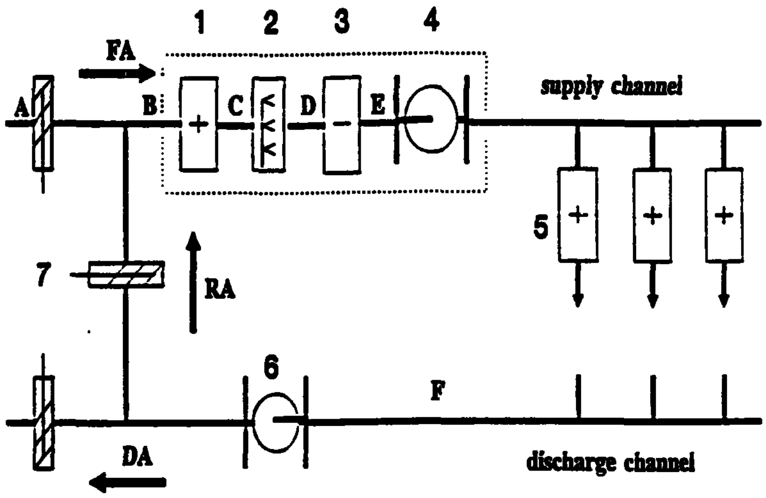
Heating, annual for building (kWh/yr)	95,550
Cooling, annual for building (kWh/yr)	207,165

4.6 Reference Air Conditioning System

At first instance it seems obvious to see the air conditioning system based on the air cycle as an alternative for all-air systems, where air meets the entire cooling demand of the building and most of the heating demand (only additional heat supply in the rooms situated at the outer walls by an additional heating system). One-channel systems with air at a given, season dependent temperature and with certain humidity, are the most frequently applied all-air systems. The desired temperature in the rooms is achieved by controlling the air flow between 30 and 100%. This system is known as the Variable Volume system (VAV).

The VAV-system is shown schematically in Fig.3. The air is brought to a certain temperature (e.g. 12 °C) and to the desired humidity in a centrally placed system. If needed the air can be heated further in a decentrally placed heater. Normally one heater supplies a number of modules (a so-called zone).

Recirculation is not applied if the ambient temperature lies between 12 and 24 °C.



1. Preheater
2. Humidifier
3. Air cooler
4. Supply fan
5. Afterheater (by zone)
6. Discharge fan
7. Valve register

FA Fresh air
DA Discharged air
RA Recirculated air

Figure 3 *Schematic representation of VAV-system*

The operation of the central system under design conditions is shown in Fig.4. The state points in the diagram correspond with the points in Fig.3.

<u>Operation in summer time</u>

Ambient air (point A) is mixed with a part of the return air (point F) to reach point B. The air is cooled down below the dew point and point E (12 °C and 90% RH) is reached. The air is transported to the modules and returns with conditions according to point F.

<u>Operation in winter time</u>

Ambient air (point A') is mixed with a part of the return air (point F') to reach point B'. The air is heated to point C'. However, the humidity is too low. In the humidifier the humidity content is increased with undesired consequences for the temperature (adiabatic cooling to point D'). Behind the humidifier the conditions are 12 °C and 80% RH. Further heat absorption takes place in the afterheaters, and after heat rejection and moisture absorption in the modules finally point F' is reached.

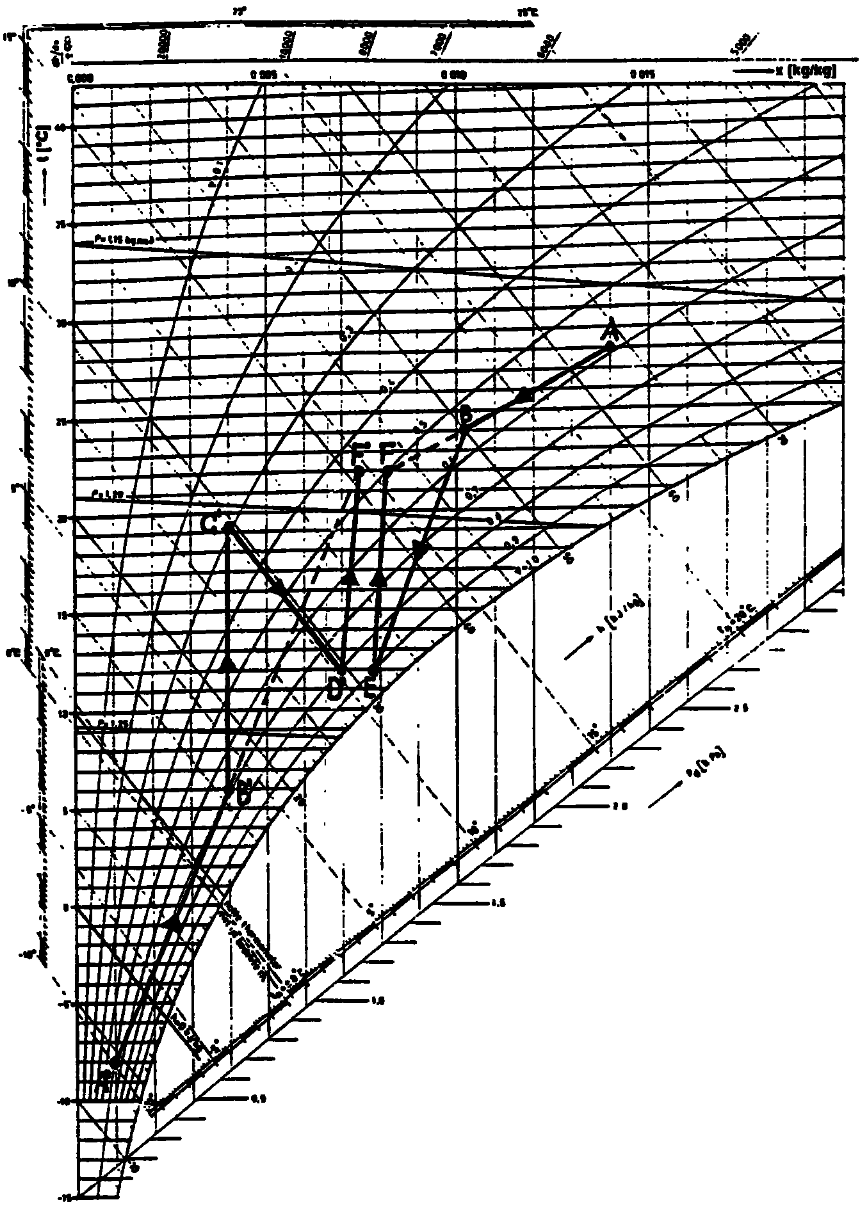


Figure 4 *Conditioning of the air in reference system (design conditions)*

Control
The supply of heat or cold can be controlled by controlling the air flow to the module in a range from 30 to 100%. If with 30% of the air flow the heat supply is still larger than the heating demand, the afterheater is controlled down (by example in steps of 2 °C). If the heat supply then turns out to be too low, the air flow (still at 30%) can be increased.

The investment costs of the complete system (boilers, refrigeration machines, air conditioning, control) is in the order of magnitude 2.5 million NLG (1.1 million ECU) /5/.

5. RESULTS REFERENCE AIR CONDITIONING SYSTEM

The calculation results of the reference case are shown in Table II. The electricity consumption of the pumps of the boilers and the afterheaters are ascribed to heat production. The electricity consumption of the pumps of the cooling towers and the refrigeration machines are ascribed to the production of cold. Finally, the electricity consumption of the pumps of the preheaters, humidifiers and dehumidifiers (air coolers) as well as the elec-

tricity consumption of the fans in the supply channels, discharge channels and the cooling towers are ascribed to the transport function.
For the heat production a higher heating value (HHV) of 35 MJ/m^3, which is valid for Slochteren natural gas, is used.
The efficiency of the boiler is assumed to be 85% (HHV) and constant. The efficiency (COP) of the refrigeration machine follows a curve characteristic for speed-controlled machines.

Comparing Tables I and II shows that even with high-efficiency boilers the efficiency of the heat production (based on HHV) is only 0.17 (95.55/555). This is caused by the fact that during most of the time the boilers operate at part load and that the humidification of the air results in extra heating demands.

The efficiency of the cold production (Coefficient of Performance for cooling, COP_c = ratio between produced cold and consumed electricity) appears to be 2.2 (207.165/95.4).

Assuming energy prices of 0.44 NLG/m^3 (0.2 ECU/m^3) (natural gas) and 0.11 NLG/kWh (0.1 ECU/kWh) (electricity), the annual energy costs come to 51,000 NLG (37,000 ECU).

Considering that in the Netherlands the nationwide efficiency of electricity generation is 0.37 (HHV), the consumption of primary energy equals 555 + 235.3/0.37 = 1191 MWh/yr (based on HHV).

TABLE II: Energy figures for the reference case

Energy consumption reference case	Gas (MWh/yr (HHV))	Electricity (MWh/yr)
Heat production		
- boilers	555.0	2.1
- pumps		9.0
Cold production		
- refrigeration machines		95.4
- pumps		24.2
Transport		
- pumps		11.3
- fans		93.4
Total	555.0	235.3

The relatively simple model used in /1/ to calculate the thermodynamic states in the air cycle as a refrigeration machine is used in this study to get insight into the performance of the air cycle under different operating conditions. A similar model was developed to investigate this performance when operating in the heating mode (heat pump operation). An annual COP can be estimated by using the formula for the so-called Integrated Part Load Value (IPLV) /6/. The IPLV is a weighted average of the COPs at 25, 50, 75 and 100% part load:

$$IPLV = 0.11*COP_{25\%} + 0.33*COP_{50\%} + 0.39*COP_{75\%} + 0.17*COP_{100\%}$$

Although the value of the weight factors are valid for the USA, it can be expected that similar values can be determined for Europe.

6. AIR CONDITIONING SYSTEM BASED ON AIR CYCLE

For the calculations the following assumptions were the most important:

- first compressor: the isentropic efficiency was taken 75% for the 50 and 75% part loads, and 60% for the 25 and 100% part loads. These values are quite conservative and higher values are possible, but not yet available for this type of applications.
- turbocharger: the isentropic efficiency was taken 75% for the 100% part load, 70% at 75% part load, 65% at 50% part load and 60% at 25% part load.
- pressure losses /5/: 1000 Pa in the supply channels and 200 Pa in the discharge channels at 100% part load, and decreasing quadraticly with the volume flow.
- supply temperatures: 10 °C in the case of cooling demand, and 34 °C in the case of heating demand. The latter one is rather high compared to the conventional all-air air conditioning systems, but lower values result in very low COPs.
- discharge conditions: as mentioned in 4.4.

When operating in the cooling mode, the COPs of the air cycle vary between 0.35 and 0.55, with an IPLV of about 0.5. In the heating mode, the COPs for heating (COP_h = ratio between produced heat and consumed electricity) are in the range 1-1.15 with an IPLV of 1.1.
With these values the annual energy consumption becomes 95.550/0.5 + 207.165/1.1 = 379.3 MWh/yr. The consumption of primary energy will be 379.3/0.37 = 1025 MWh/yr, which is 14% less than with the reference air conditioning system.

The annual energy costs are 379,300*0.11 = 42,000 NLG/yr (19,000 ECU/yr), which is almost 20% less than for the reference system. This means that the allowable investment costs of the air cycle alternative are (slightly) higher than the costs for the reference system.

With respect to the operating conditions of the air cycle alternative the following remarks can be made.

- In conventional systems recirculation of the discharge air is often applied to recover heat (in the winter) or cold (in the summer). With an air cycle in the cooling mode, recirculation is not necessary. The discharge air flow is used in the regenerator to cool the supply air sufficiently. This means that the discharge air flow is heated to a temperature far above the ambient temperature, which makes recirculation unattractive. However, recirculation in the heating mode is possible and profitable (higher COP and less heat to be transferred in the regenerator).
- The calculations showed that an intercooler is not useful in the cooling mode because of the low pressure ratios of the first compressor.
- In the heating mode, if the supply temperature is but a few degrees above the desired room temperature, or if the isentropic efficiencies of the machinery are very low, the COP may be less than 1. This means that the discharge temperature is above the ambient temperature (or mixed temperature in case of recirculation), and that the heat source is heated in stead of cooled down. Both the heating of the rooms and the heating of the heat source has to be accomplished by the energy input in the compressors. This situation can be avoided by choosing the supply temperature sufficiently higher than the room temperature and by selecting turbomachinery with good isentropic efficiencies.
- In the heating mode condensation and even ice formation may occur in the expander under certain conditions.
- In the heating mode, on balance, the air is humidified because of the moisture absorption in the rooms and the condensation of water vapour if the heat source is cooled below the dew point. If the moisture content of the supply air is too low, water injec-

tion between the two compressors, or at the inlet of one of both, may be applied. However, water injection implies a cooling effect which will have to be neutralised by the compressors and should therefore be avoided if possible.
- In the air cycle alternative separate afterheaters are not incorporated. In the reference building there are four, decentrally placed, afterheaters to bring the air to the temperature which is desired by the accompanying zone. In this sense the air cycle is less flexible.

7. CONCLUSIONS

Air conditioning systems based on the air cycle principle are technically attractive. In the heating mode heating, humidification and transport of air are integrated in the cycle. The same applies for cooling, dehumidification and transport in the cooling mode. Although not investigated in this study, attention has to be paid to the hygienic quality of the supplied air, especially possible bacteriological problems in the regenerative heat exchangers.

Tentative calculations show good possibilities for the air cycle alternative to have a better energetic performance (in terms of primary energy consumption) than conventional all-air (VAV) systems. Whether this advantage is also found in the energy costs, depends on the ratio between electricity and gas consumption for the reference system, and the prevailing tariff structure. The results of the study give enough rise to a more detailed investigation based on hourly values for outdoor climate conditions during one year.

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RÉSUMÉ

On a examiné un système frigorifique et une pompe à chaleur combinée à base d'une cycle thermodynamique 'Joule-Brayton' à l' air, pour climatiser (chauffer, refroidir, humecter, dehumecter) un bâtiment. Plusieurs calculations a été fait, utilisé les programmes à l'ordinateur simples pour systèmes, bâtiments et climats (internel et externel), déduits des modèles complexes. Les calculations donnent un système attractif, à cause de l'integration des fonctions (chauffer, refroidir, transport de l'air, humecter et dehumecter). L'examination résulte au système à l'air semblent compatible par rapport à l'énergy, la charge de l'énergy et les investissements.