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Theoretical analysis of an open-cycle absorption heating and cooling system

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In warm countries, such as Italy and Spain, the increasing demand for space conditioning cannot be met by the current capacity of installed electric power. This has led to a growing interest in cooling with natural gas. An interesting thermal cooling system uses open-cycle absorption. It treats the air directly without heat exchangers and with very low temperature drops. With absorption dehumidification it is possible not only to obtain a cooling system similar in performance to conventional thermal cooling systems, but also, with few modifications, a high-performance heating system fed by a natural gas burner, which combines simplicity with PER typical of a gas driven heat pump. The proposed new system described in the paper is studied both in winter and summer mode, evaluating the influence of the different parameters.
(Keywords: Air conditioning, energy, natural gas, absorption system, open cycle, performance)

Analyse théorique d'un système de chauffage et de refroidissement à cycle ouvert à absorption

Dans des pays chauds, comme l'Italie et l'Espagne, la montante demande de conditionnement d'air ne peut pas être toujours satisfaite par la puissance electrique installée. Pour cela l'intérêt dans le refroidissement par gaz naturel est en train d'augmenter. Un système très interessant est le cycle ouvert à absorption. Il trait l'air directement sans besoin d'exchangeurs de chaleur et avec de très petites différences de temperature. La deshumidification par absorption permet non seulement d'obtenir un système de refroidissement aussi performant que un traditionnel système thermique de refroidissement, mais aussi, avec peu de modifications, un très permormant système de chauffage alimenté par un brûleur à gaz, qui combine simplicité et un PER (Rapport d'Energie Primaire) d'une pompe à chaleur à moteur. Le nouveau système proposé, qui est ici décrit, est étudié dans le fonctionnement en été et en hiver, en évaluant l'influence des différènts paramètres.
(Mots clés: conditionnement d'air, energie, gaz naturel, système à absorption, cycle ouvert, performance)

Thermal cooling is gaining increasing interest in countries such as Italy and Spain. In Italy the Electric National Board is beginning to discourage new electric power contracts for the air conditioning of large commercial buildings, owing to the scarce capacity of installed electric power. At the same time the use of natural gas for winter heating is increasing. In the presence of 'take and pay' contracts, this implies a high surplus of natural gas in summer. The excess gas is stored during the summer in exhausted wells.

Of course an alternative use for gas during the summer is welcome, to balance the consumption throughout the year. Consider *Figure 1*, which shows the monthly consumption in Italy: gas consumption during August is seen to be one-third of that during December. Natural gas companies are now promoting the utilization of gas for summer cooling with either engine-driven compression machines or with absorption systems, allowing economic incentives such as installation grants and favourable gas tariffs, sometimes for gas consumption all year round. The phasing out of CFCs could positively promote the application of absorption systems. At present this is prevented by the high capital cost of the devices (particularly the LiBr machines), and by the summer use only without the possibility of a heat pump mode. The efforts of some manufacturers of ammonia–water

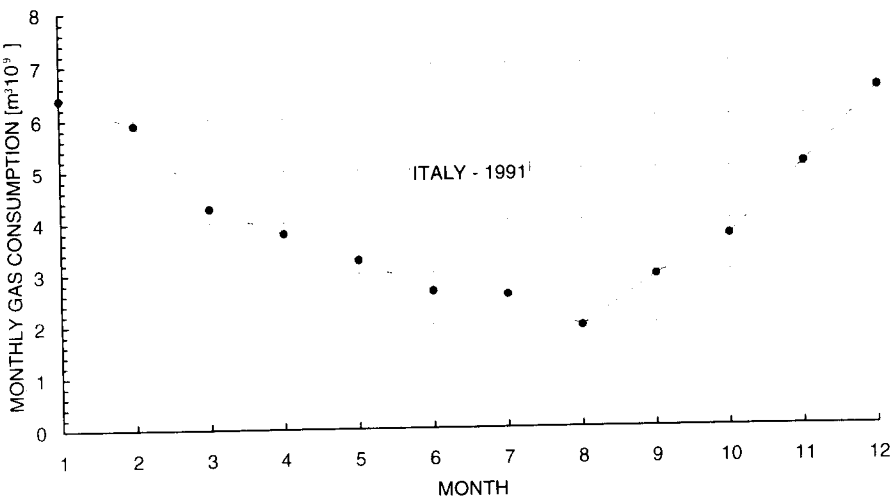

Figure 1 Monthly natural gas consumption in Italy during 1991
Figure 1 *Consommation annuelle de gaz naturel en Italie depuis 1991*

chillers to provide summer–winter appliances are surely well addressed.

In solar energy researches one of the most extensively analysed processes is solar cooling. The recent improvement of absorption devices is much indebted to solar energy studies. In the context of solar cooling, the open-cycle absorption system was first proposed during the 1970s[1–4]. Several types of potential cycle have been analysed[5–7]. Overall, the dehumidification/humidification cycle has been presented for its ability to treat air directly without heat exchangers and with very low temperature drops. Of course, open-cycle absorption can be easily driven by a natural gas burner, resulting in a cooling system similar in performance to conventional thermal

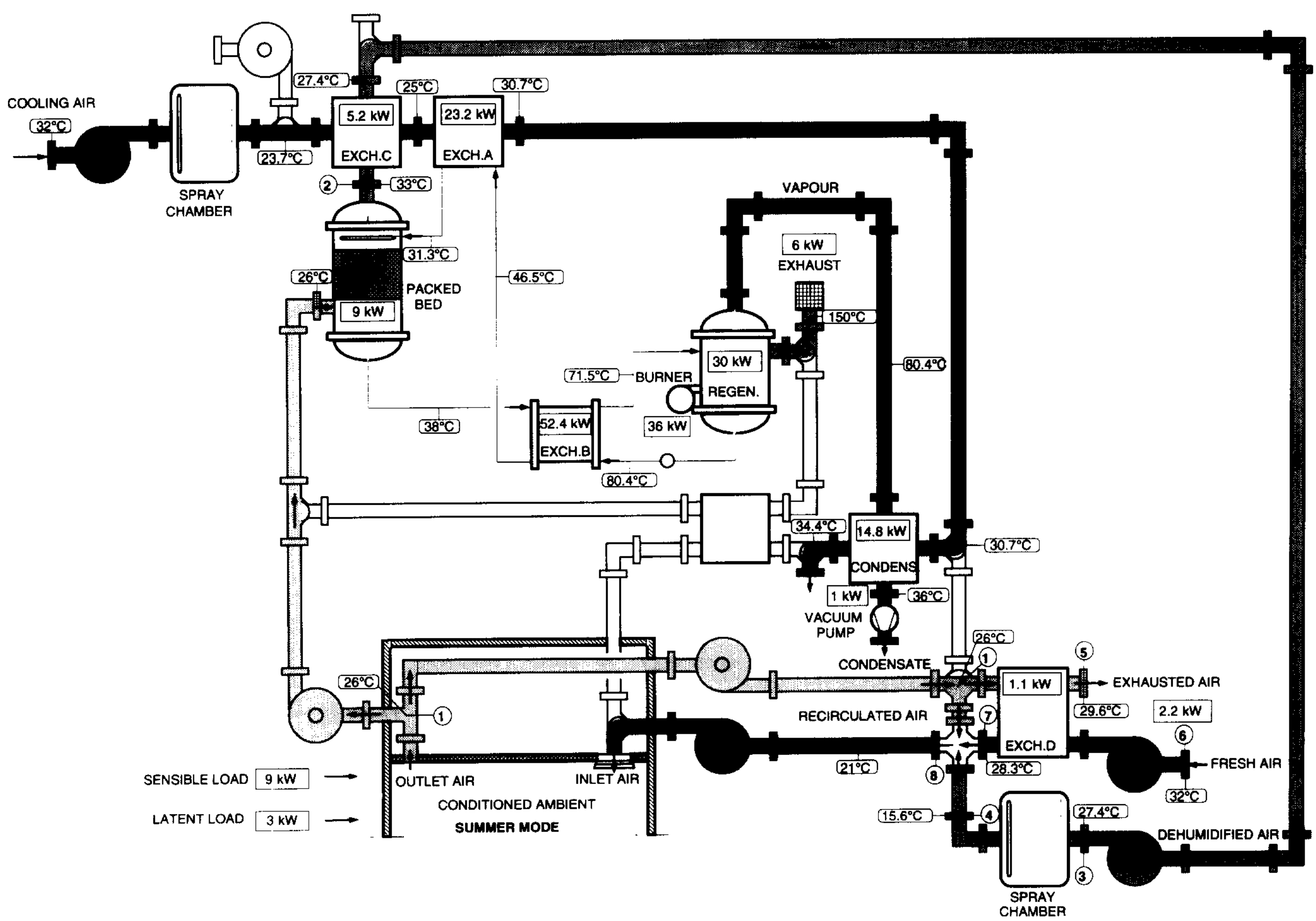

Figure 2 Block diagram of the open-cycle absorption system in summer-mode operation
Figure 2 *Schéma d'un système à absorption à cycle ouvert en fonctionnement estival*

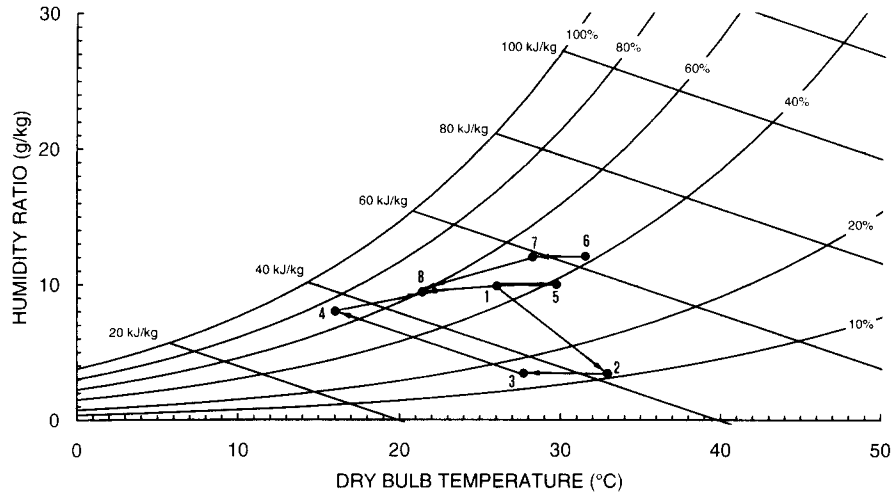

Figure 3 Air-conditioning cycle in summer-mode operation

Figure 3 *Cycle de conditionnement d'air en fonctionnement estival*

systems. With a few modifications to the cycle a heating mode is possible, which combines simplicity with very good thermodynamic performance[8,9]. Thus the system becomes useful in both summer and winter. Moreover, the system can be driven by low-temperature waste heat (say 80–90°C) available, for example, from internal combustion engine cogeneration, which is usually dissipated during summer months. More recently, modifications have been proposed to the cycle, which will provide chilled water (8–10°C)[10]. However, the introduction of indirect heat exchangers lowered the obtainable COP greatly. Moreover, only a cooling mode was presented.

System description

Summer Mode

A schematic view of the proposed open-cycle absorption system, operating in summer mode, is shown in *Figure 2*. The heart of the system is the packed-bed tower, where the air can be dehumidified by a sorption solution such as lithium bromide/water. The other devices (regenerator, condenser, spray chambers, heat exchangers) are separated in the diagram for clarity, but can be easily grouped in a single item of equipment.

The air processes are shown in the psychrometric diagram in *Figure 3*. In process (1)–(2) part of the air extracted from the room (1) is dehumidified, and its temperature is raised by contact with the warmer sorption solution and also from the heat of absorption. At (2) the air is then cooled by adiabatically saturated outside air in heat exchanger C. The low humidity air at (3) is then passed through a water spray chamber, where it undergoes humidification cooling to state point (4) before being mixed with recirculated air at condition (1) and fresh air at condition (7) to achieve state point (8). Fresh air at condition (6) is precooled by exhausted air in heat exchanger EXCH. D to state point (7). The mixture at (8) is introduced into the room to satisfy both the sensible and the latent loads.

The diluted sorption solution leaving the packed bed must be regenerated. Before doing so it is preheated in a liquid–liquid heat exchanger (EXCH. B), before entering the natural-gas-fired regenerator. The hot regenerated solution must now be cooled so that it can absorb effectively. This is carried out in an air-liquid heat exchanger (EXCH. A) by outside saturated air. The water vapour developed in the regenerator is condensed in an air-cooled condenser, operating at a pressure lower than atmospheric (a vacuum pump is provided).

A theoretical analysis was developed for the whole system, with particular regard to the packed bed column, as described later. The analysis gave also the primary energy requirements (the reference is always to the gross calorific value of the natural gas). A pie chart of the primary energy requirements in summer mode operation is shown in *Figure 4*, which reveals that the main fraction is due to the cooling of the system by outside air (EXCH. A, EXCH. C and CONDENSER), while the remaining components are associated with the discharge of the exhaust and the condensate.

Winter mode

The winter operation of the system is illustrated in

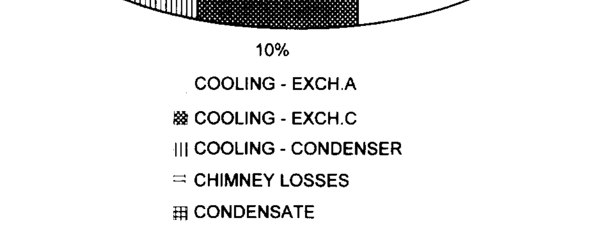

Figure 4 Distribution of primary energy losses in summer-mode operation

Figure 4 *Distribution des pertes en énergie primaire en fonctionnement estival*

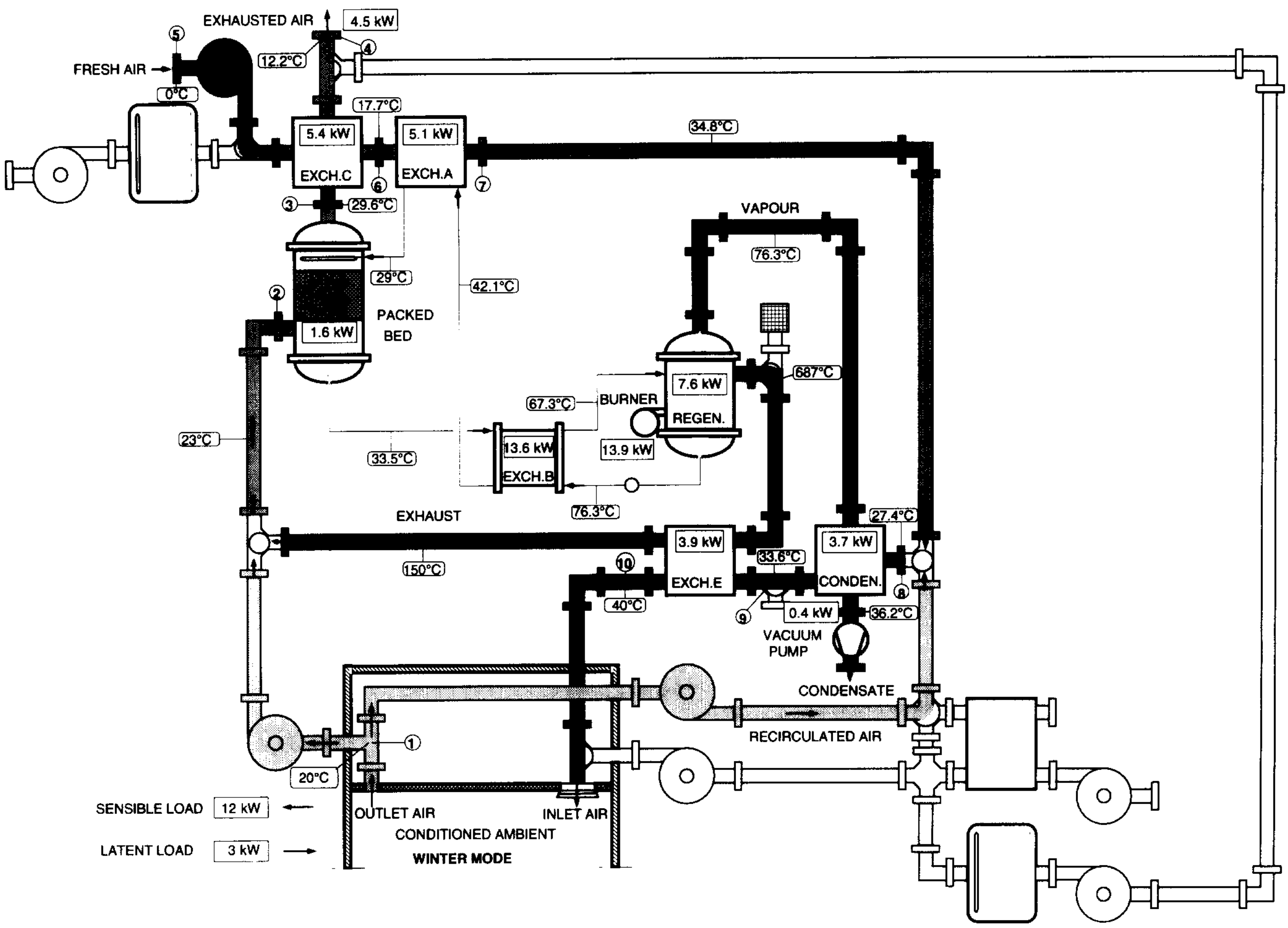

Figure 5 Block diagram of the open-cycle absorption system in winter-mode operation
Figure 5 *Schéma d'un système à absorption à cycle ouvert en fonctionnement hivernal*

the schematic diagram shown in *Figure 5* and on the psychrometric chart in *Figure 6*. Air extracted from the heated room at (1) is mixed with the exhaust from the regenerative burner (at 150°C with a humidity ratio $100\,g\,kg^{-1}$). The air-exhaust mixture (2) is then passed through the packed-bed tower, where it is dehumidified so that much of its latent energy content is taken up by the sorption solution. The sensible heat increases slightly. Before being exhausted to atmosphere the exhaust/air mixture at (3) is used to preheat the outside fresh air entering at (5) in heat exchanger (EXCH. C) till (6). Further preheating of fresh air now at (6) is given till state point (7) by the sorption solution in air-liquid exchanger (EXCH. A) before it enters the packed-bed.

The preheated fresh air at condition (7), after mixing with recirculated air at condition (1), achieves state

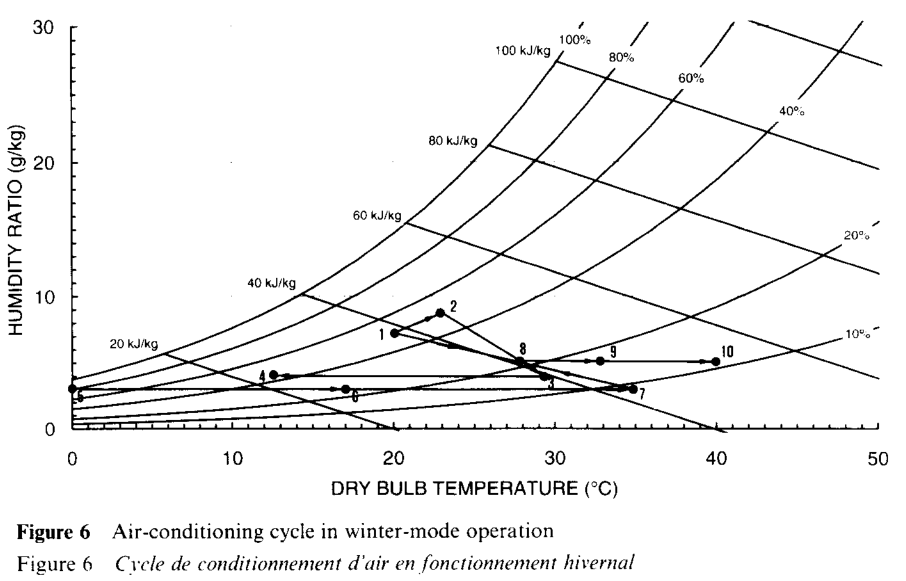

Figure 6 Air-conditioning cycle in winter-mode operation
Figure 6 *Cycle de conditionnement d'air en fonctionnement hivernal*

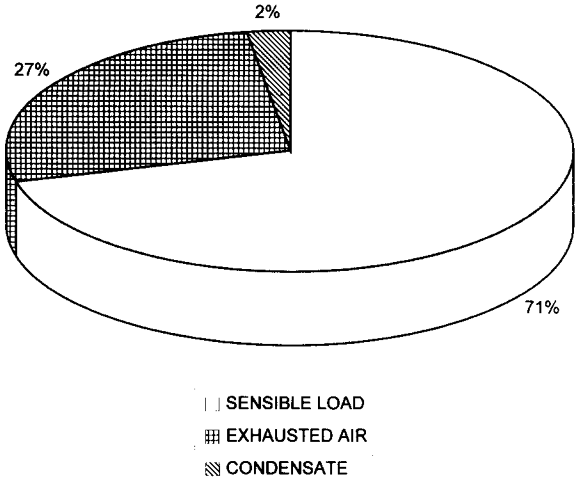

Figure 7 Distribution of primary energy losses in winter-mode operation

point (8) and then passes through the condenser and takes on the heat of condensation of the vapour separated in the regenerator (9). Final heating is provided by the hot exhaust gases from the regenerator in heat exchanger EXCH. E and the air reaches the condition (10) to be supplied to the room. The sorption solution cycle is the same as in the summer cooling mode.

Figure 7 shows a pie chart of the primary energy requirements in winter mode operation. The main energy fraction in this case is due to the sensible heating load, while the discharge of the exhausted air and the condensate represent the other energy components.

Performance

Air conditioning and heating of a small building was simulated to predict the performance of the proposed system, and to evaluate the significance of heat exchanger efficiency. The packed-bed tower was described by a heat and mass transfer model, choosing for the different conditions the most effective solution concentrations and flowrate ratios (L/G is the ratio between solution and air flowrate in the tower, i.e. the mass rate of solution per one air unit mass)[11–15]. Three types of heat exchanger were investigated, with possible effectiveness range of:

1. gas–liquid heat exchanger (A) with effectiveness 0.6–0.8 (typically 0.7);
2. liquid–liquid heat exchanger (B) with effectiveness 0.7–0.9 (typically 0.8);
3. gas–gas exchanger (C, D and E) with effectiveness 0.5–0.7 (typically 0.6).

Condenser effectiveness was assumed to be 0.7 and the adiabatic saturator efficiency was assumed to be 0.8. During summer cooling mode the outside air, adiabatically saturated, was assumed to be the heat sink.

It was imposed in the choice of the parameters that the regeneration temperature would not be higher than 80°C, so that eventual utilization of waste heat energy was possible.

Cooling mode

Outside conditions of 32°C air temperature and 12 g kg^{-1} humidity ratio were assumed. The room air temperature was set at 26°C with a humidity ratio of 10 g kg^{-1}. The internal load was assumed to be 9 kW sensible and 3 kW latent cooling. With a fresh air flowrate of 1080 kg h^{-1} (0.3 kg s^{-1} the ventilation loads were then 1.8 kW latent and 1.5 kW sensible. The performance of the system is given by the primary energy ratio (PER), i.e. the ratio between the total cooling demand and the gross calorific value (GCV) times the gas consumption. The additional power for fans and pumps was assumed to be negligible, and was therefore, neglected.

Figure 8 illustrates the effect on PER of the gas–liquid heat exchanger effectiveness for different liquid–liquid efficiencies. It is clearly seen from *Figure 8* that influence of the latter on PER is greater than that of the former. In effect the recovery of the sensible heat from the regenerated solution is of fundamental importance in reducing gas consumption. It was assumed

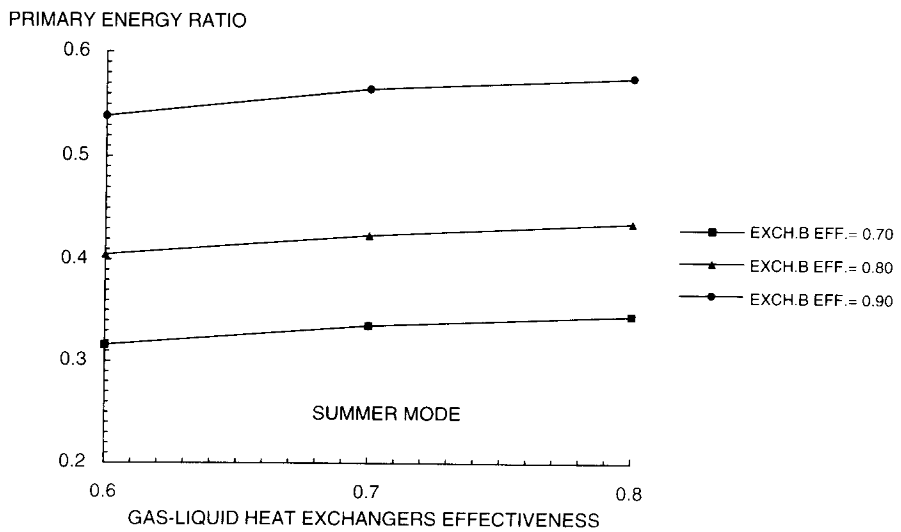

Figure 8 Primary energy ratio in summer mode as a function of gas–liquid heat exchanger effectiveness for different liquid–liquid exchanger effectiveness

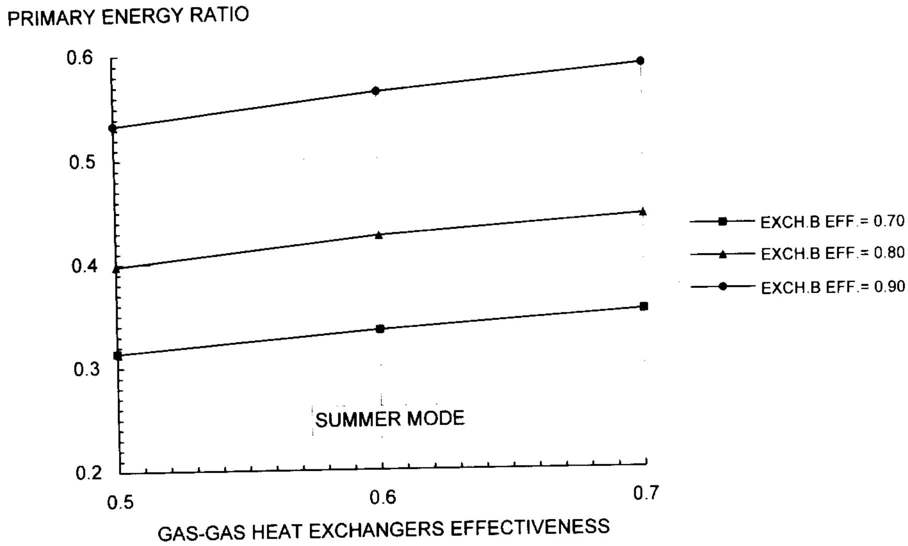

Figure 9 Primary energy ratio in summer mode as a function of gas–gas heat exchanger effectiveness for different liquid–liquid exchanger effectiveness

that the flowrate ratio (liquid–gas) equalled 0.9, i.e. the solution flowrate is at 2900 kg h^{-1} (the recirculated air is at 3300 kg h^{-1}). The same conclusion can be drawn as regards the gas–gas heat exchanger (*Figure 9*) and the adiabatic saturator efficiency (*Figure 10*). Even with the lowest considered efficiencies for the various exchangers, provided the liquid–liquid is at 0.9, a PER in excess of 0.5 is achievable. Although this is not particularly high, it is of the same order as that of most ammonia–water absorption chillers, and it must be appreciated that in this system air is treated directly without the additional temperature differences imposed by fan-coil units or extra energy consumption of reheaters.

A significant operating cost of the proposed system is possible due to the movement of large quantities of air through the system (about 15 000 kg h^{-1}) necessary for cooling the treated air after the packed bed and the regenerated solution. This task could be achieved via a cooling tower with the possible advantage of a free-- cooling during the mid-seasons[16]. The behaviour of the proposed system is, however, particularly favourable in heating mode.

Heating mode

For the purpose of analysis, outside air conditions of 0°C and 3 g kg^{-1} humidity ratio were assumed. Room air temperature was set at 20°C with a humidity ratio of 8 g kg^{-1}. An internal heating load was assumed to be 9 kW. The losses toward the outside were assumed to be 12 kW sensible load and the internal vapour development due to people was assumed to be 3 kW of latent gain. Fresh air flowrate was again assumed to be 1080 kg h^{-1}, which gives rise to a sensible ventilation heating load of 6 kW.

Figure 11 gives the PER as a function of the gas–liquid heat exchanger efficiency for different liquid–liquid exchanger efficiencies. Again the influence of the latter is significant, but now it is of the same order as the former. Note that the PER is appreciably higher than unity: the system is really an open-cycle heat pump. Its

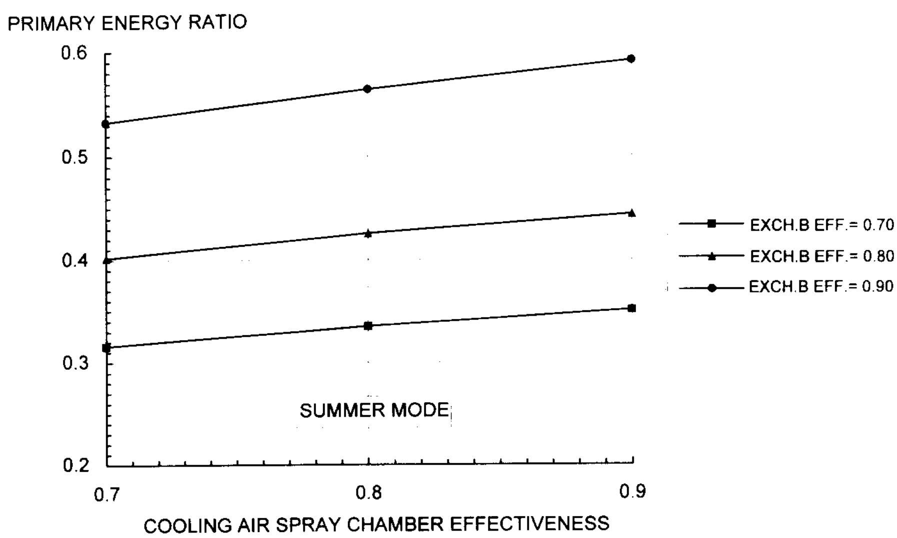

Figure 10 Primary energy rate in summer mode as a function of spray chamber 1 efficiency for different liquid–liquid exchanger effectiveness

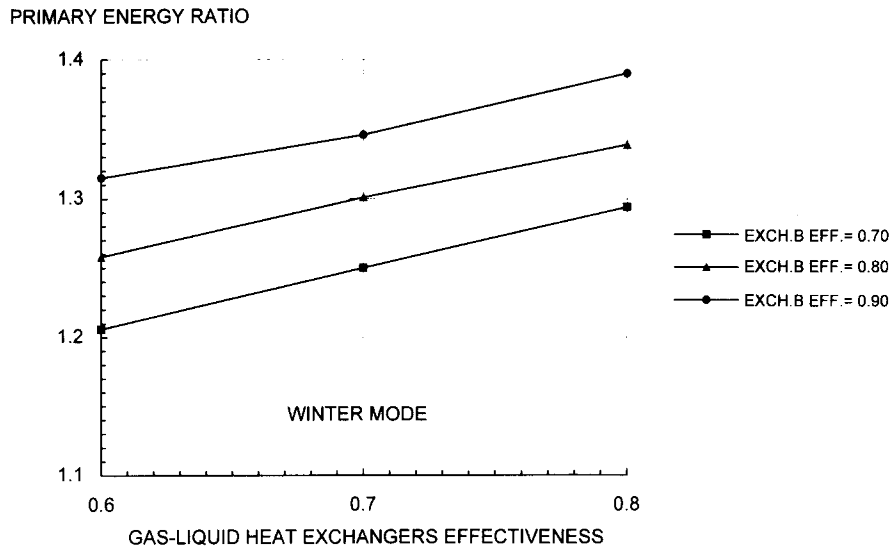

Figure 11 Primary energy ratio in winter mode as a function of gas–liquid heat exchanger effectiveness for different liquid–liquid exchanger effectiveness

Figure 11 *Rapport d'énergie primaire, en fonctionnement hivernal, en fonction de l'efficacité de l'échangeur de chaleur gaz–liquide, pour différentes efficacités de l'échangeur liquide–liquide*

cold source is the exhausted mixture of air and combustion gases. The absorption in the packed bed turns some latent heat into sensible heat, producing an effective preheating of the ventilation air and a temperature rise in the sorbent. In an absorption heat pump heat is given up at the both absorber and condenser, just as in the proposed system. In both cases the generator has the same function. In the proposed cycle, however, the evaporator is absent, as the vapour is already produced inside the room and is contained in the combustion exhaust.

Figure 12 evaluates the effects of the efficiency of the gas–gas heat exchanger which influences the performance similarly to the liquid–liquid heat exchanger.

Overall the predictions indicate that the PER varies from a minimum of 1.2 to more than 1.4 with respect to the gross calorific value of natural gas. PER values as high as these are difficult to obtain with more complex and probably much more expensive closed-cycle absorption or vapour compression cycle engine-driven heat pumps.

An experimental analysis is in progress to survey the real ability of desiccants in absorbing and releasing water vapour in a test rig[17] equipped with a packed tower and a regenerator connected to a water-cooled condenser. The first results indicate a slight overprediction of the humidification for the theoretical model (within 20%), which does not substantially modify the possibilities of applications of the system.

Conclusions

Open-circle absorption systems can provide both heating and cooling. Cooling is thermally driven and CFC-free. Heating is high performance, as the operation is that of

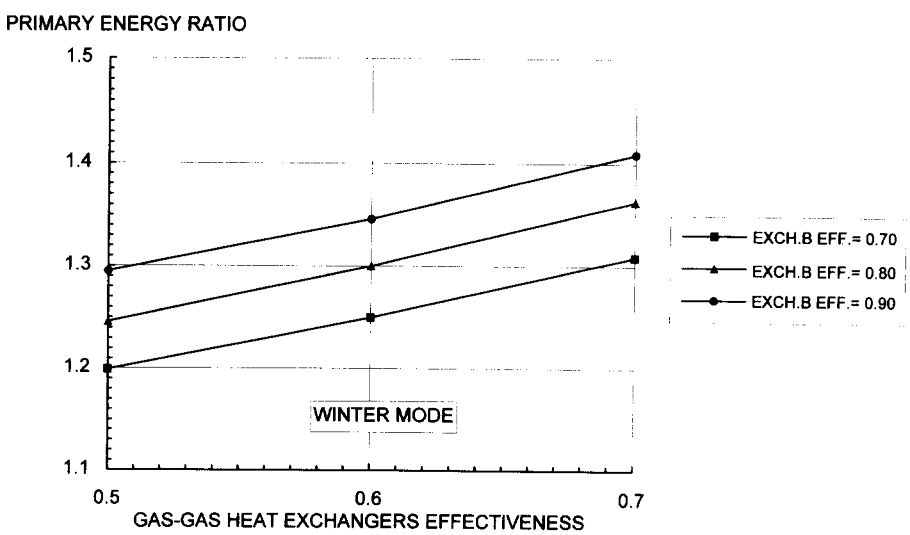

Figure 12 Primary energy ratio in winter mode as a function of gas–gas heat exchanger effectiveness for different liquid–liquid exchanger effectiveness

Figure 12 *Rapport d'énergie primaire, en fonctionnement hivernal, en fonction de l'efficacité de l'échangeur de chaleur gaz–gaz, pour différentes efficacités de l'échangeur liquide–liquide* an open-cycle heat pump. This allows a performance that is comparable to the possible performance of a closed-cycle absorption heat pump or even of an engine-driven heat pump. The devices are, however, much simpler and probably less expensive.

The heart of the system is the packed-bed tower, where air is dehumidified. Experimental verifications of the behaviour of the tower are in progress. The first results, obtained with the mixture water-lithium bromide, appear to be in satisfactory agreement with the theoretical predictions.

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