



NEW IDEAS FOR ENERGY UTILISATION IN COMBINED HEAT AND POWER WITH COOLING: II. APPLICATIONS

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Abstract—Chemical dehumidification can be beneficial in a HVAC plant, both in terms of capital and operating costs. The sorption system can be integrated into a traditional plant, usually satisfying the latent heat load, but it can satisfy even the whole load with particular cycles which also use evaporative cooling. A number of applications are examined here, both integrated and self-sufficient systems, driven by natural gas or waste heat, working with liquid or solid sorption. Different lay-outs, performance and savings suggest new ideas for better energy utilisation in HVAC combined heat and power plants. © 1997 Elsevier Science Ltd. All rights reserved.

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INTRODUCTION

Chemical dehumidification can be useful in the operation of a HVAC plant either as an integrated system, in conjunction with conventional equipment, or as a self-sufficient system. In the first case the advantages are the following: reduction of the electric load; reduction of the capacity of the cooling equipment; higher working temperature of the cooling machines (possibly well in excess of the dew point of the treated air) with a higher COP and cooling capacity (both increase with the evaporator temperature); reduction or elimination of post-heating; purifying and germicide action.

When the system is self-sufficient, the replacement of conventional equipment can be beneficial both in terms of lower initial cost and of operating cost, particularly when waste heat at 80–90°C is available. Even the direct use of natural gas can be beneficial [1, 2].

First of all it is necessary to examine how it is possible to obtain a completely self-sufficient system based on chemical dehumidification even when the sensible load prevails over the latent one [3–9].

Simple processes of evaporative cooling, dehumidification and indirect contact cooling with a water tower are used, as illustrated in the block diagram of Fig. 1. The various processes can be followed easily on a psychrometric chart (Fig. 2).

The return air, mixed with fresh air, is evaporatively cooled so that it can cool the treated air in a heat exchanger. The subsequent following chemical dehumidification appreciably increases its temperature, so that cooling with tower water is possible. Then indirect contact cooling takes place. A final evaporative cooling stage allows the air to be introduced into the space at a low enough temperature (18–20°C) for the air-conditioning needs. The inlet humidity ratio, according to the process, can be lower than the space one, so that the latent load can be satisfied at the same time.

In the following sections some integrated and self-sufficient plants will be described. The equipment which will be encountered in the analysis comprises heat exchangers, evaporative coolers, adsorption/absorption dehumidifiers (already described in a previous paper [10]), desiccant wheels and twin cells.

As is well known, the desiccant wheel, illustrated in Fig. 3, accepts in one sector outside air which can be cooled and dehumidified in the remaining part by exhausted air, whose lower water content allows reactivation.

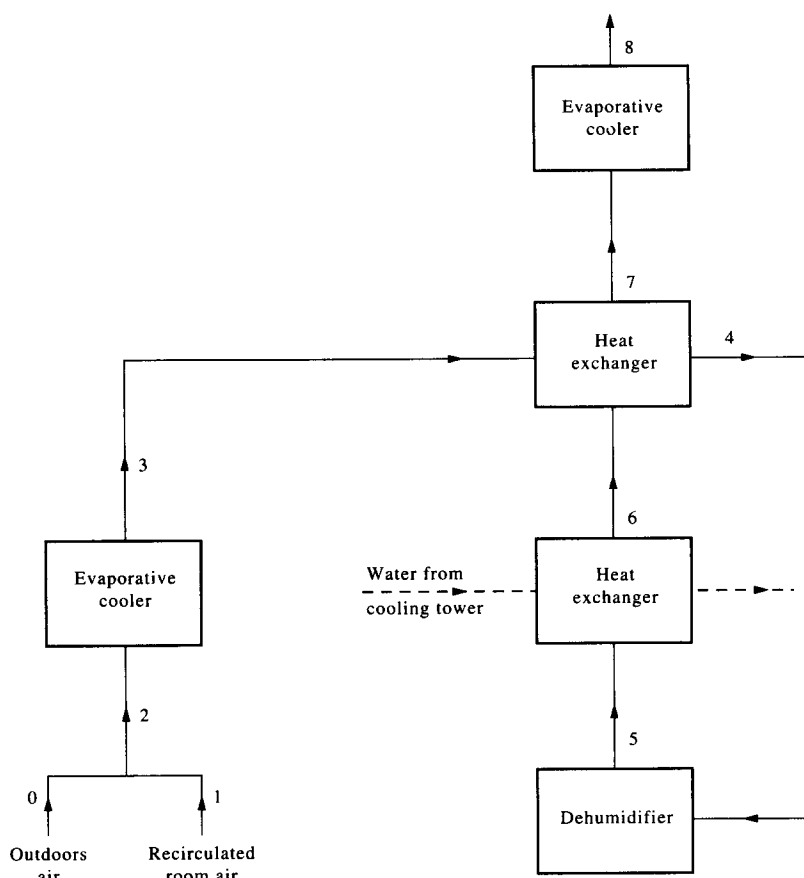


Fig. 1. Block diagram of the d/h (dehumidification/humidification) open cooling cycle.

Twin cells operate in a similar way, they are in effect two coupled spray chambers (Fig. 4). The exhaust air reactivation ability is usually low and the overall efficiency is not particularly high.

SORPTION DEHUMIDIFICATION HVAC SYSTEM OPERATING IN RECIRCULATION AND VENTILATION

Air treatment can be realised using adsorption substances. The device, rather compact, is composed basically of a dehumidifier, a regenerative (rotary) heat exchanger, two humidifiers and a heating coil. A possible lay-out is shown in Fig. 5; in the recirculation mode the air treatment stages are quantified in Fig. 6.

The return air is first dehumidified, then cooled in a regenerative heat exchanger by outside air and finally humidified to meet the required supply condition.

It is useful to follow the outside air path. It is first humidified in order to effectively cool the supply air in the regenerative heat exchanger, then raised in temperature by a coil heated by waste heat. This energy input is sometimes not enough to achieve the desired reactivation in the dehumidifier, therefore the air is sent partly through the first sectors of the desiccant wheel, which have a high moisture content since they have just left the dehumidification process. The reactivation temperature with high moisture content is lower. As the reactivation proceeds the temperature level needed becomes higher; part of the air, preheated by the coil, is then sent to a gas burner which heats it to sufficiently high temperatures. The reactivation air, hot and humid, is finally exhausted.

Some illustrative values for the air conditions in the various parts of the cycle are reported in Fig. 6 and included on the psychrometric chart in Fig. 7.

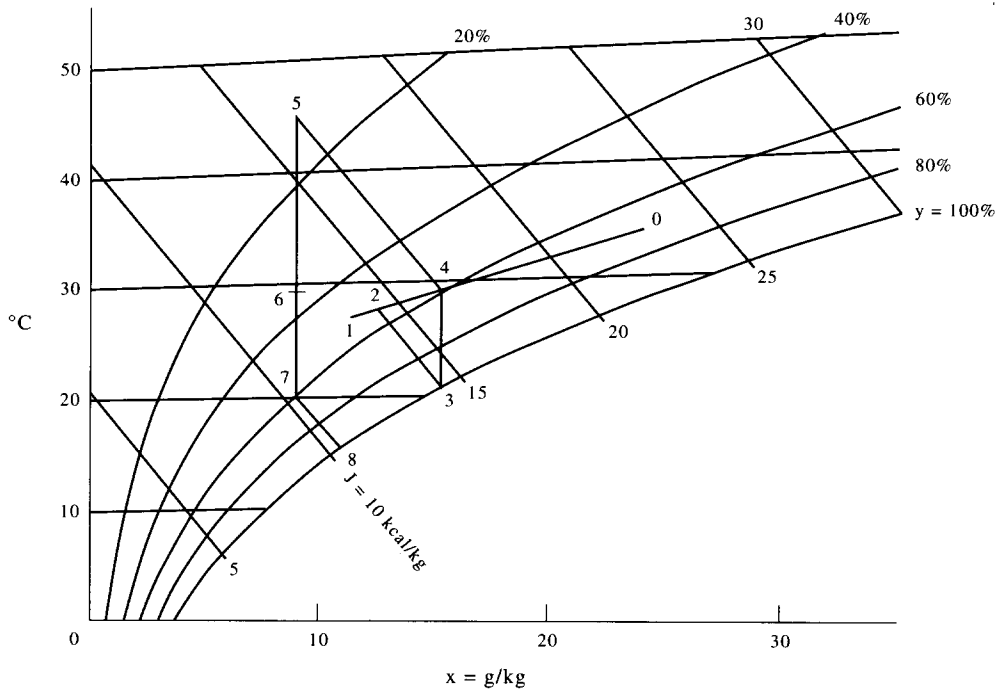


Fig. 2. The d/h cycle on the psychrometric chart.

Consider how the air enthalpy varies between the dehumidifier inlet and outlet owing to sensible heat exchange between the two air flows. Supply and reactivation air flow rates are the same. The reactivation temperature required is rather high.

In order to improve the system performance another mode was studied: ventilation. This time the reactivation air is taken from the conditioned space and all the supply air is fresh. The air taken from the space is first humidified, in order to effectively cool the supply air in the regenerative heat exchanger. At the outlet of this heat exchanger the air is heated by waste heat and then by a gas burner and then admitted to the desiccant wheel to regenerate it. An equal air flow is taken from the outside, dehumidified, cooled in the regenerative heat exchanger and finally humidified before introduction to the conditioned space. Illustrative values of the air conditions in the various processes are reported in Fig. 8 and on the psychrometric chart in Fig. 9.

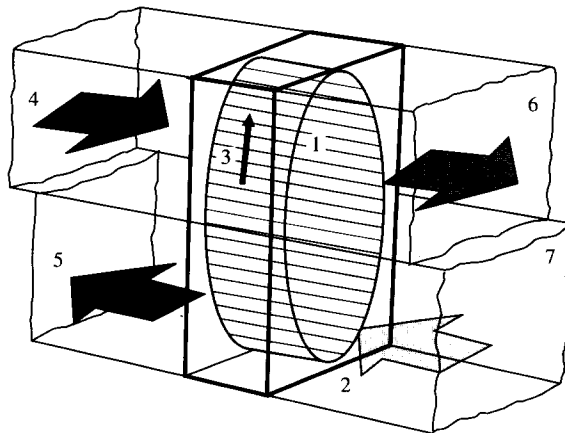


Fig. 3. Schematic view of a desiccant wheel (4: hot humid outside air; 2: cool and dry exhaust air).

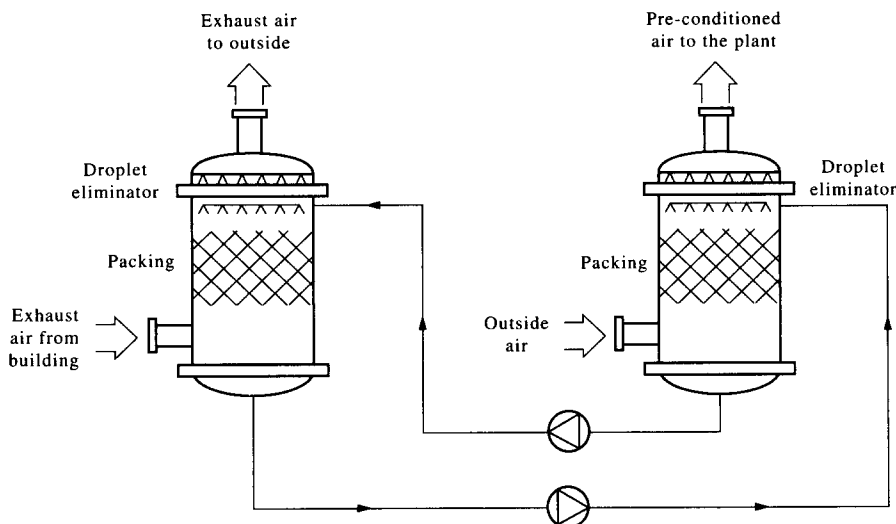


Fig. 4. Twin cell system.

The COP of the proposed systems, i.e. the ratio between the cooling effect and the whole thermal supply, is between 0.4 and 0.5; not particularly high and lower than for the classical single-effect absorption machines. However two advantages must be considered: (i) a direct air treatment without further losses for heat exchanges or for post-heating and (ii) the utilisation, even if partial (about 50%), of waste heat.

TWO-STAGE SORPTION AIR-CONDITIONING SYSTEM

Recently, as it was necessary to achieve a humidity reduction beyond 12 g/kg, a two-stage system was realised for the air-conditioning of a fast food restaurant against outside air conditions of 35°C (dry bulb) and 24°C (wet bulb). The restaurant is located in Brooklyn, New York [11].

The block diagram of the plant is shown in Fig. 10 and the processes are reported on the psychrometric chart in Fig. 11.

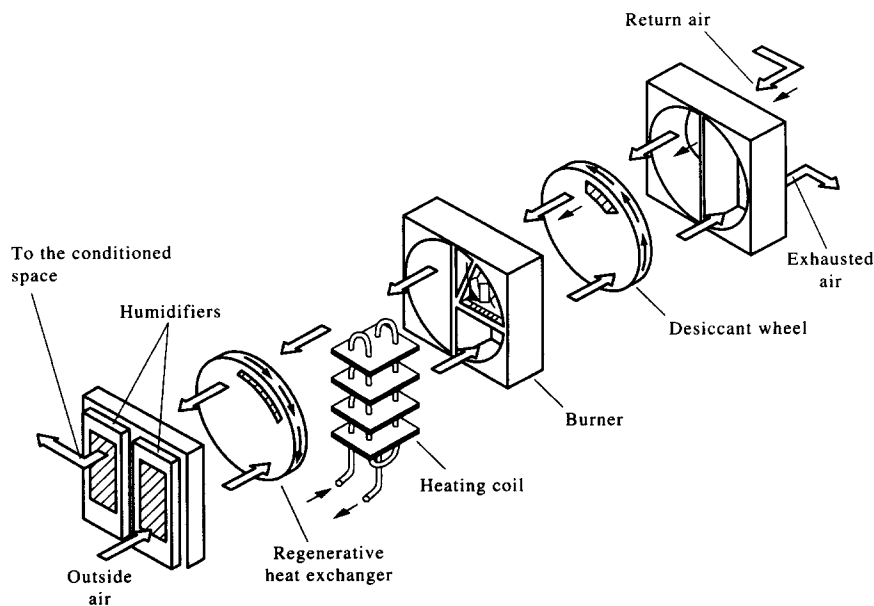


Fig. 5. Basic elements of an open cycle adsorption cooling system.

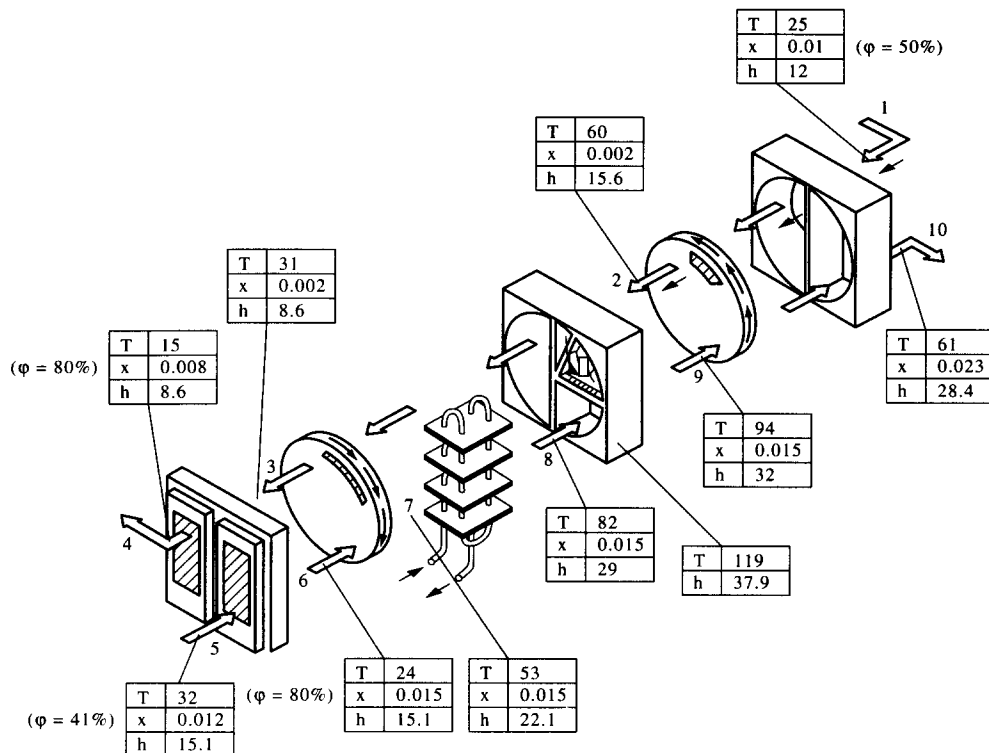


Fig. 6. Indicative values for the air at various points in the open cycle system in recirculation mode.

The outside air, whose humidity ratio is about 17 g/kg, is first dehumidified to a level of 8.6 g/kg (stage 0–1d). Its temperature is then at 58°C. Then sensible cooling is realised in a rotary regenerative heat exchanger with an outside air flow, evaporatively cooled (stage 1d–1x), which in the following is utilised, after sensible heat exchange in a heat pipe heat exchanger and heating by a gas burner, to reactivate the desiccant wheel (stage 1–9).

The treated air, at a constant humidity ratio, but with a temperature as low as 30 °C, undergoes a second dehumidification stage (1x–2d), arriving at less than 2 g/kg. Its temperature increases towards 50°C, and it is cooled in a rotary heat exchanger, as before, to 30°C (stage 2d–2x). The low moisture content allows one to reach a supply temperature of only 13°C with a humidity ratio of less than 9 g/kg after humidification (stage 2x–s). Thus both the sensible and latent loads are satisfied, keeping the space at the requested conditions of 27°C and 50% RH.

The rotary heat exchanger efficiency is claimed to be between 0.85 and 0.89, whereas for the heat pipe heat exchangers the efficiency is considered to lie between 0.66 and 0.70. The COP of this system, driven by a gas burner, is claimed to be 0.85. The highest reactivation temperature is 85°C. In such a case, the system could be driven by heat recovered from a reciprocating i.c. engine.

INTEGRATED SORPTION AIR-CONDITIONING SYSTEM

The plant described in this section has unusual characteristics that can stimulate many ideas for a designer. Firstly, it is a cold air distribution HVAC system; the primary air flows at less than 5°C, as in some plants equipped with ice storage. The advantage is to greatly reduce the air volumes with less room needed by the ducts and less energy for the fans.

In a conventional plant this feature also reduces the COP of the cooling equipment. However it does not happen in the proposed plant [12–15].

The plant is an integrated one, with a conventional vapour compression chiller driven by an i.c. engine and with desiccant wheels. They are reactivated by the waste energy supplied by the engine [16]. The cold primary air satisfies the whole latent heat load and a fraction (between 35

and 50%) of the sensible load. The remaining load is satisfied by the room terminals supplied with water at 10°C, always from the chiller. The plant lay-out is shown in Fig. 12, where the i.c. engine can be recognised at the top, the compression chiller on the left and the recuperative heat exchanger on the right. A gas boiler is provided for integration, both for the desiccant reactivation and for winter heating.

In the centre of Fig. 12 the air treatment unit is shown equipped, in addition to the fans, with a rotary heat exchanger (RHE), a desiccant wheel (RD), two cooling coils supplied with cooled water by the chiller and an adiabatic humidifier. At the bottom of the Figure the room air terminal is shown, where the return air is partly exhausted and partly precooled before mixing with the cold primary air.

The processes can be followed in Fig. 13 on the psychrometric chart. The fresh air (1), taken from the outside, undergoes a first treatment in a desiccant wheel reactivated by the exhaust air (process 1–2). The fan slightly increases the air temperature (4) ahead of the cooling coil, where the air is cooled to 13°C with slight dehumidification (4–5). The dehumidification is more effective in the desiccant wheel (5–6) with the air heated to 28°C; the second cooling coil again lowers the air temperature to 13°C (6–7). The final temperature decrease, to less than 5°C, is due to the adiabatic humidifier, although the chiller operates at more than 10°C at the evaporator. The fresh air, cooled in this way, arrives at the room terminals, where it is mixed with the return air and pre-cooled to 18°C (9–10).

A study carried out in an office building located in northern New Jersey (a six-storey block with a useful area of about 15 000 m², peak cooling load 1.8 MW, of which 24% is latent),

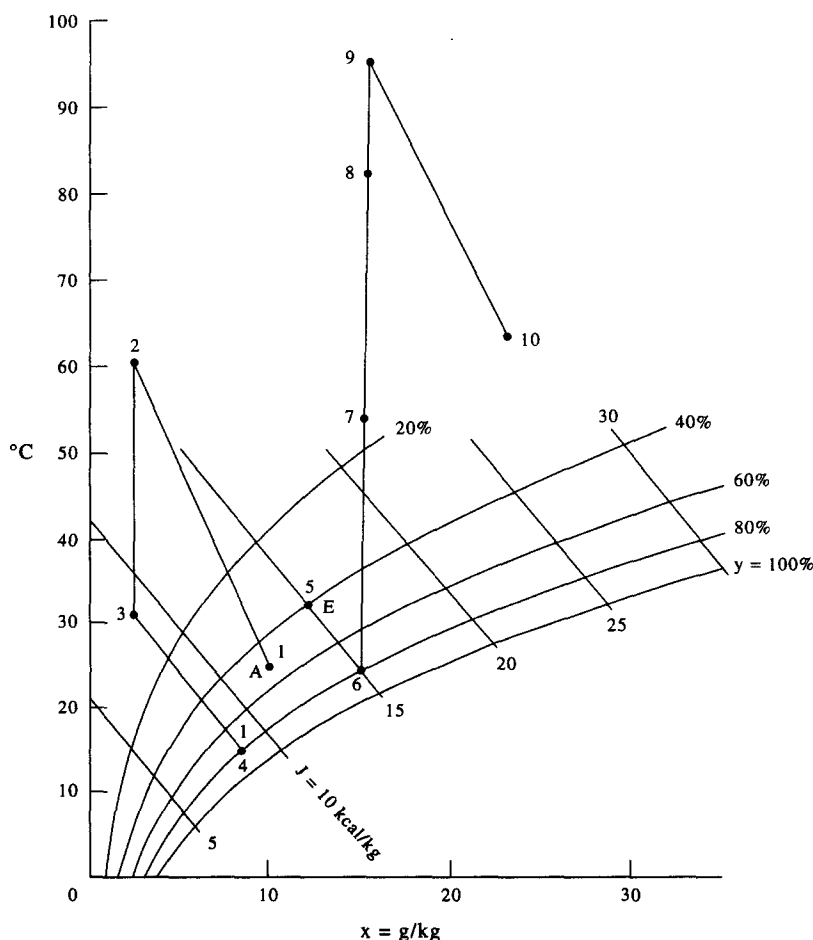


Fig. 7. The open cooling cycle in recirculation mode on the psychrometric chart.

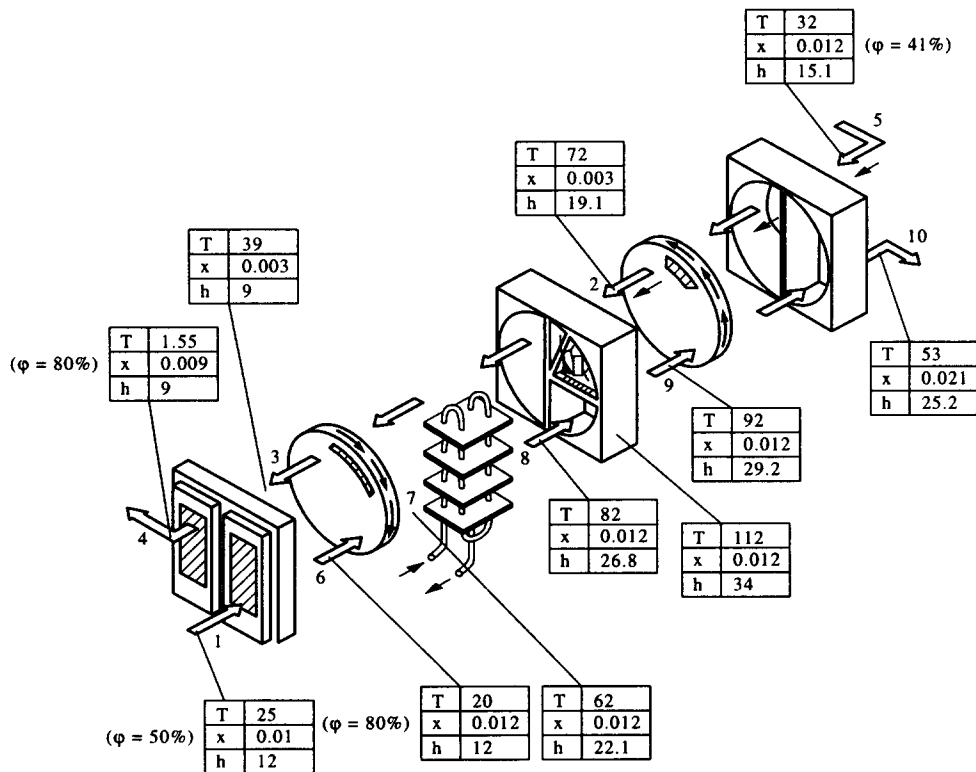


Fig. 8. Indicative values for the air at various points in the open cycle system in ventilation mode.

compared a conventional all-air electrically driven system with an ice partial storage system, always electrically driven, and with the former system driven by an i.c. gas engine.

At first the air distribution costs were examined. They required peak fan powers respectively of 132, 142 and 100 kW. Then the yearly energy costs were considered under different utility rates both for the electricity (on-peak and off-peak) and for the natural gas.

The two alternative systems were always better than the conventional one with savings from 12 to 40% under the various assumptions, concerning the structure and amount of the utility rates. The system described was better than the ice storage one in almost all situations (except in the presence of very low electricity rates in off-peak hours) with advantages even higher than 15%.

The first cost is obviously higher for both the alternative systems, with an overcost estimated between 10 and 15%.

The on-peak electricity rates considered were 0.0756 \$/kWh, whereas the off-peak rates varied from 0.0200 to 0.0597 \$/kWh. The gas rates were from 3.00 to 5.30 \$/MMBtu.

LIQUID SORPTION HVAC

In the examples just examined dehumidification was obtained via adsorption. The plants operating by liquid sorption usually present a different lay-out, frequently with an effective internal heat recovery system. A basic example is shown in Fig. 14 [17].

The return air (23°C and 50% RH) is evaporatively cooled to 17°C, which is a suitable temperature for operating run-around coils. Then it is mixed with fresh air. After filtering the air goes to a spray chamber dehumidifier, where the desiccant is sent after cooling with tower water. The dry air at 30°C is cooled down to 23°C in the run-around coil and finally evaporatively cooled to 14°C before entering the conditioned space.

The desiccant reactivation is obtained by exploiting the low humidity of the exhausted air, separately heating the liquid by waste heat. All reactivation air leaves the regenerator at a relatively

high temperature so that air preheating can be provided before introduction into the regenerator by means of run-around coils. A further heat exchange is provided between weak and strong sorbent.

AN INTEGRATED SYSTEM OPERATING BY LIQUID SORPTION

When operating with an integrated system, it is preferable that the chemical dehumidifier satisfies the latent load. The compression chiller then satisfies the sensible load, working with rather high evaporation temperatures. The desiccant conditioner can supply the room air at the same temperature as the return one, but with a lower humidity ratio.

Such a plant is represented as far as the dehumidification section is concerned in Fig. 15. Only the return air is treated. It is supplied to the rooms at the same temperature, but with a much lower humidity ratio (from 11 to 5 g/kg). The dehumidification is obtained by spraying an aqueous solution of 44% lithium chloride in the return air, passing through a spray chamber cooled by outside air, evaporatively cooled in order to remove the heat of absorption [18].

The solution at the outlet of the spray chamber at 40% concentration is regenerated in a gas-fired boiler; here the brine is heated directly in the boiler at a rather high temperature (149°C). The water is removed directly by boiling and the vapour is exhausted, taking care to avoid desiccant carry-over. The concentrated desiccant brine returns to the spray chamber via a heat exchanger.

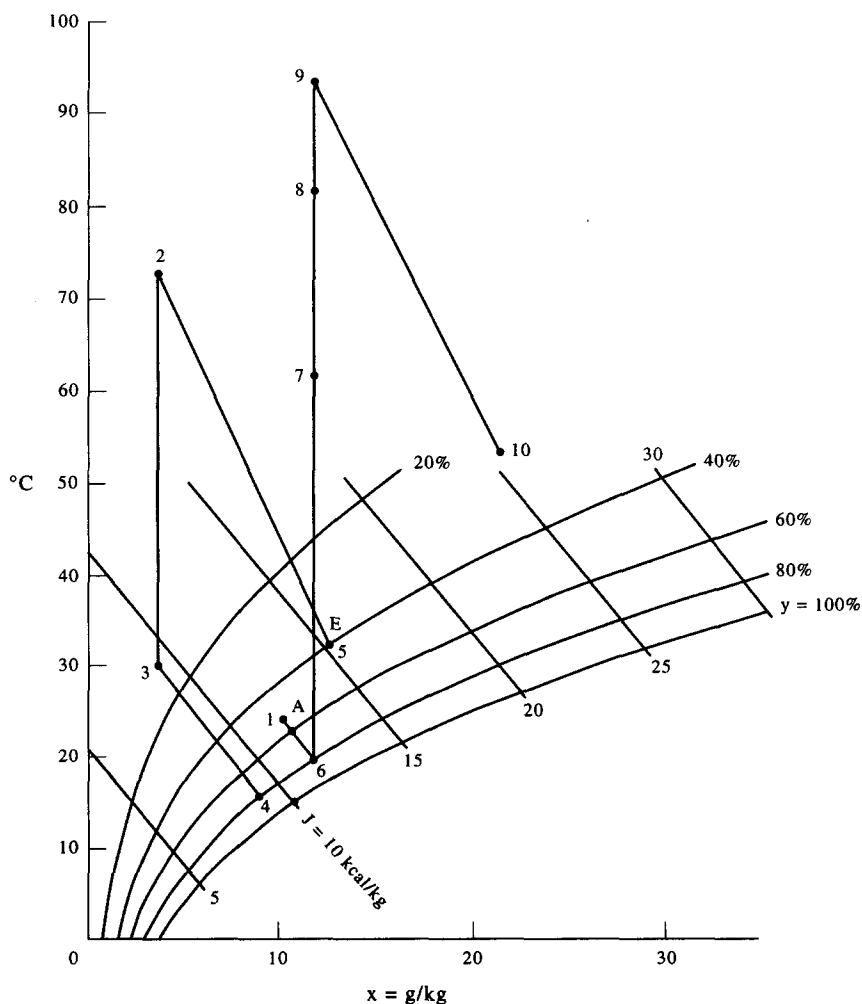


Fig. 9. The open cooling cycle in ventilation mode on the psychrometric chart.

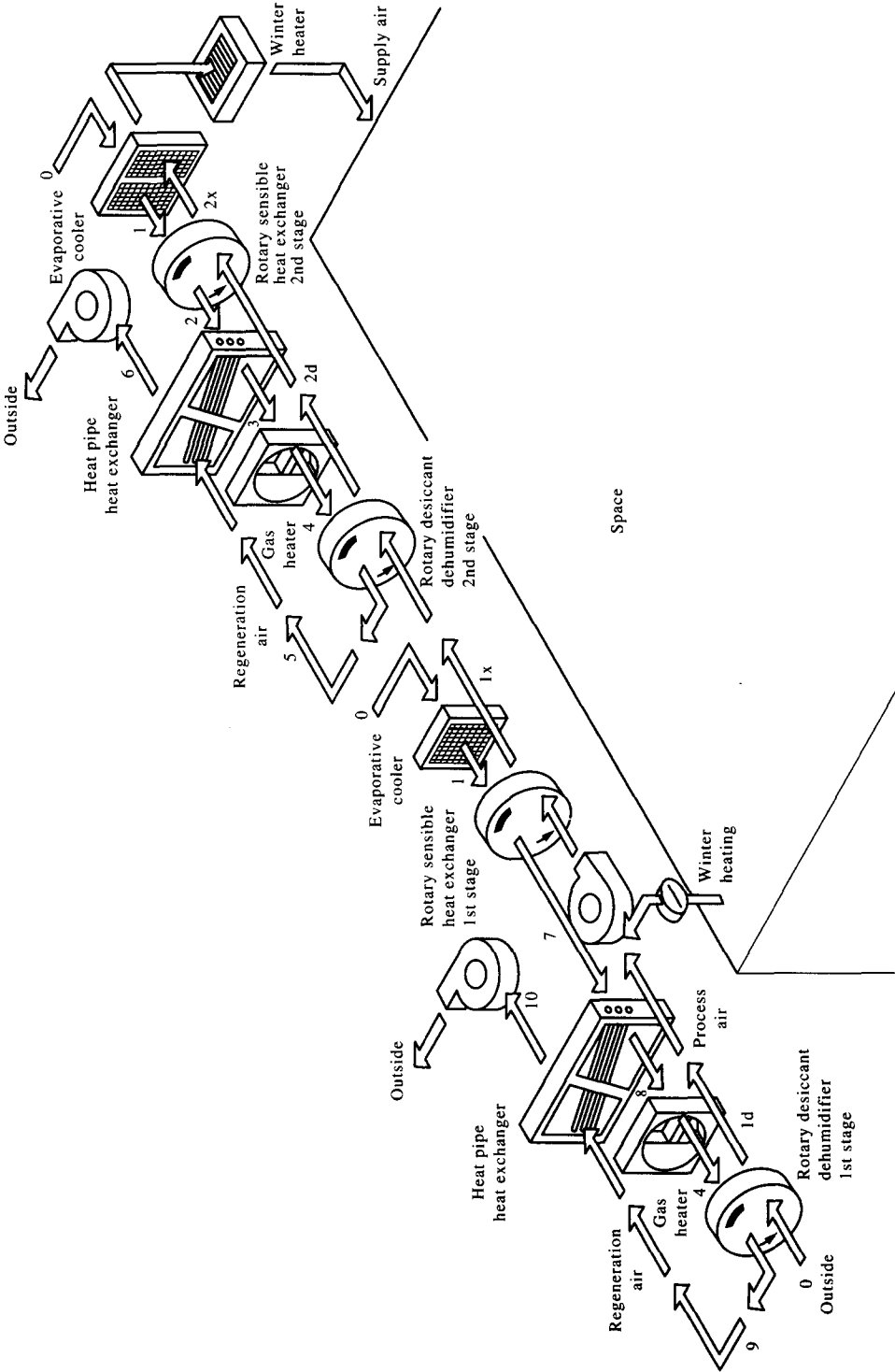


Fig. 10. Block diagram of a two-stage sorption air-conditioning system.

The system is particularly compact in order to favour residential retrofitting. The air treatment unit is represented in Fig. 16. The connections of return and supply air are located on the top. In the middle the cooling system with outside air can be recognised, whereas the regeneration element is the small gas boiler, located on the top right, from where the water vapour is exhausted. On the same side a simple falling film solution heat exchanger can be seen. The regenerated solution film covers a coil where the dilute desiccant brine flows. Its effectiveness is claimed to be 90% and an open vessel is provided in order to exhaust air and trapped gases. The solution pump is a completely hermetic magnetic-coupled type. The compactness of the unit is revealed by the size, about $1.0 \times 1.0 \times 0.50 \text{ m}^3$. The capacity is 1 ton (3.5 kW).

HEATING AND COOLING BY OPEN CYCLE ABSORPTION

Air-conditioning in temperate climates is frequently limited to a few months a year. It would be desirable that the equipment could be employed with profit even in winter heating, just as is the case with heat pumps [19, 20]. Higher initial costs could be justified with a more rational evaluation of the operating costs. In fact when a system finds utilisation for short periods a low initial cost is preferred even in the presence of higher operating costs due to inefficient operation.

Good results can be obtained with an open cycle absorption system and the equipment is in principle simple and cheap. The main elements are packed columns, evaporative coolers, some fans and heat exchangers [21–23].

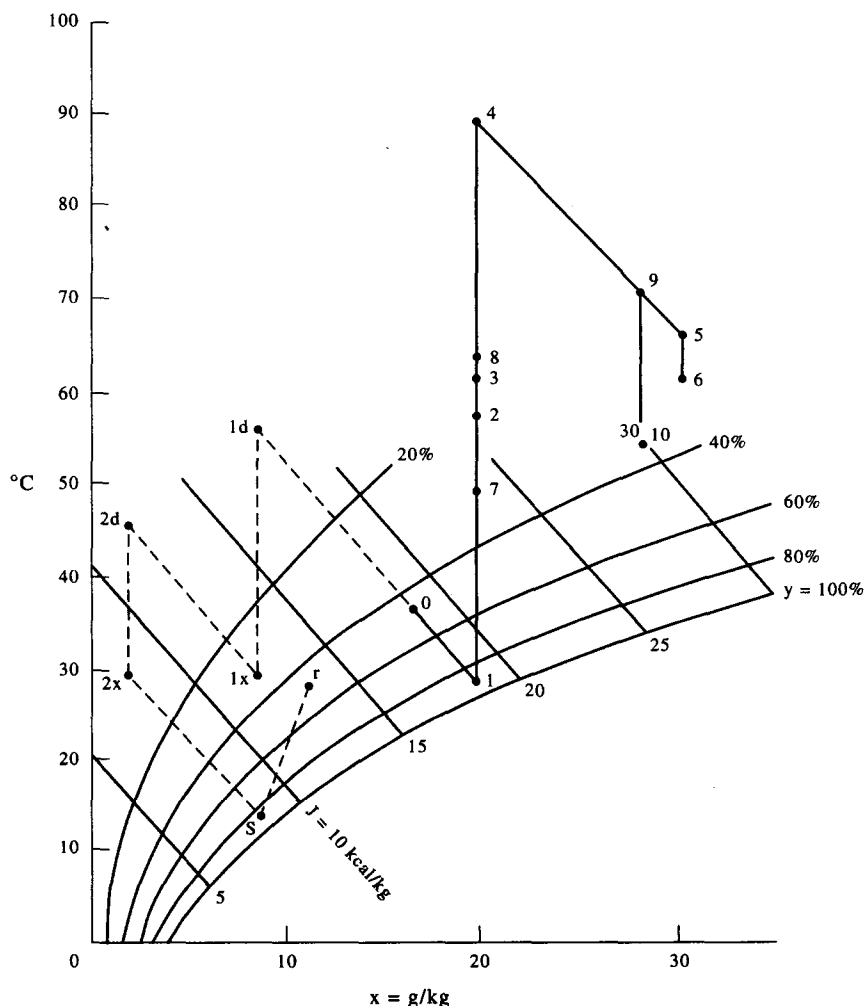


Fig. 11. The two-stage sorption system on the psychrometric chart.

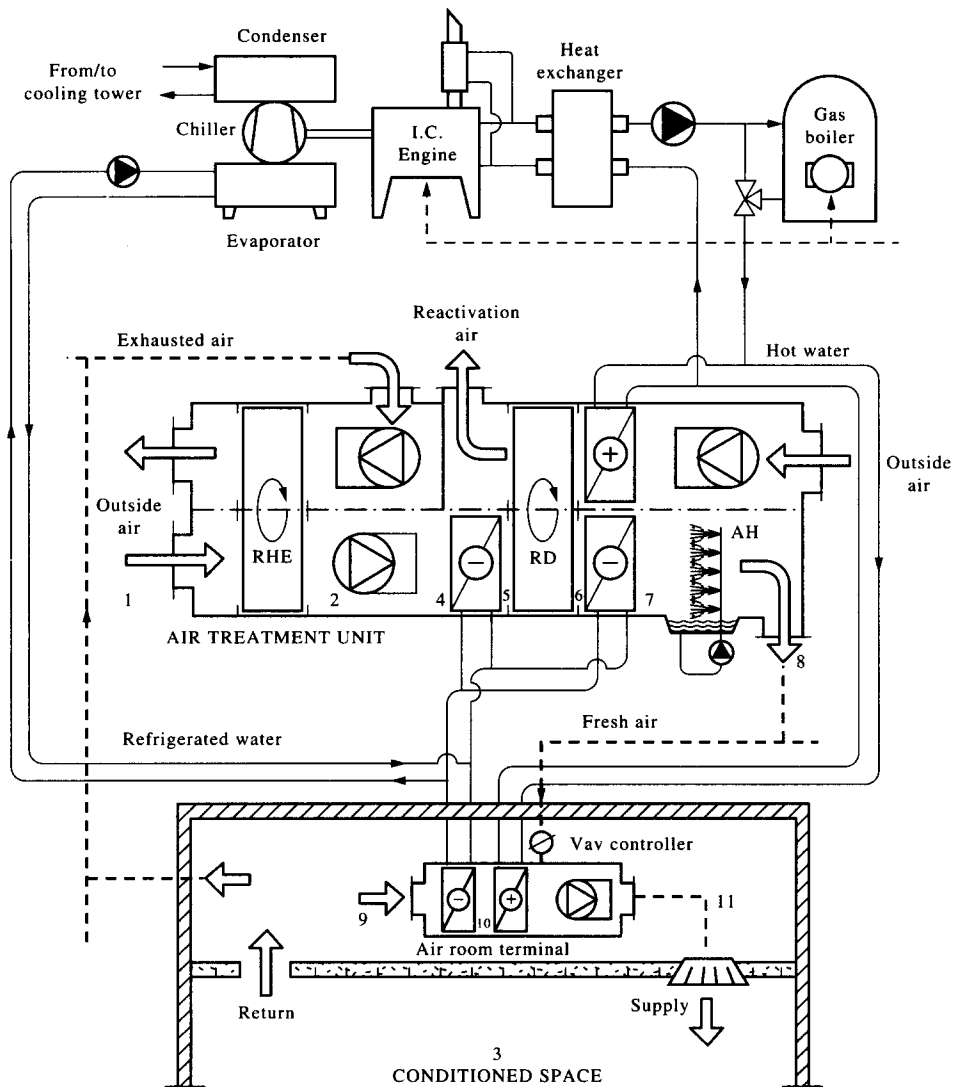


Fig. 12. Plant lay-out for an integrated sorption air-conditioning system.

A possible scheme of the system is shown in Fig. 17. The air, taken from the conditioned rooms, is partly exhausted and partly recirculated. The exhausted air, as usual, precools the fresh air. The return air is dehumidified in the packed column, leaving at a lower humidity but at a higher temperature owing to the sorption. Cooling is then provided by outside evaporatively cooled air. The return air, treated in this way, can be supplied to the conditioned ambient, after evaporative cooling which lowers the temperature to that of the supply conditions. The processes can be followed on the psychrometric chart of Fig. 18.

The sorbent from the packed bed is dilute and must be regenerated. The reactivation is realised, after preheating in a heat exchanger, in a packed column heated by a gas burner. The sorbent brine comes out hot and regenerated, so that the aforementioned preheating of the dilute brine is advisable. At the outlet of the heat exchanger the solution is still too hot to absorb effectively. Further cooling is effected in exchanger 1 using the air that has just cooled the recirculated air.

A conservative evaluation of system performances, assuming a reasonable effectiveness for the various thermal exchanges (liquid-liquid 0.9, liquid-gas 0.7, gas-gas 0.5 effectiveness), gives a 0.5 COP (ratio between the satisfied cooling load and the GCV of the fuel). The value is appreciable considering both the simplicity of the system and because the COP refers to a cooling effect in the conditioned room.

The described system can find utilisation for winter heating as an open cycle absorption heat pump. The possible operation as a heat pump is illustrated in Fig. 19. The air, exhausted from the conditioned rooms, is dehumidified after mixing with the combustion flue gases of the regeneration and integration system. In fact the flue gases are still hot (150°C) with a high relevant moisture content (more than 100 g/kg). Dehumidification removes most of the latent enthalpy content. The sensible content, instead, increases slightly. Then effective preheating of the fresh air can be provided (exchanger 3).

The desiccant brine leaving the packed bed must be regenerated as before, but brine cooling can be used for further fresh air preheating. The fresh air, preheated to more than 30°C , is mixed with the recirculated air.

During reactivation the dilute desiccant brine generates steam, and the steam condenser is cooled by the supply air. A conservative evaluation gives a COP higher than 1.3, always with reference to the fuel GCV.

THE NEW HVAC SYSTEM PROPOSED FOR THE PADUA BUILDING INNOVATION CENTRE

Recently the project of the Padua Building Innovation Center (BIC) was presented to the Commission of the European Communities by the Consorzio Padova Ricerche [24].

The building was designed adopting advanced criteria to reduce energy consumption. The advanced HVAC system allows very effective heat recovery from the exhaust air. The heart of the plant is a cogeneration system, comprising a reciprocating i.c. engine with heat recovery both on the exhaust and on the engine jacket. The electric energy satisfies the building needs (lighting, emf) and drives room unitary heat pumps for building heating and cooling. The novelty of the plant is an unconventional utilisation of the recovered heat to drive a sorption dehumidification system employed both in winter and summer mode.

The main purpose of the design is to achieve high energy savings, retaining good quality of the conditioned ambient. At the same time a high degree of independence of the various spaces has to be guaranteed, as the users are different (the building should accommodate offices and

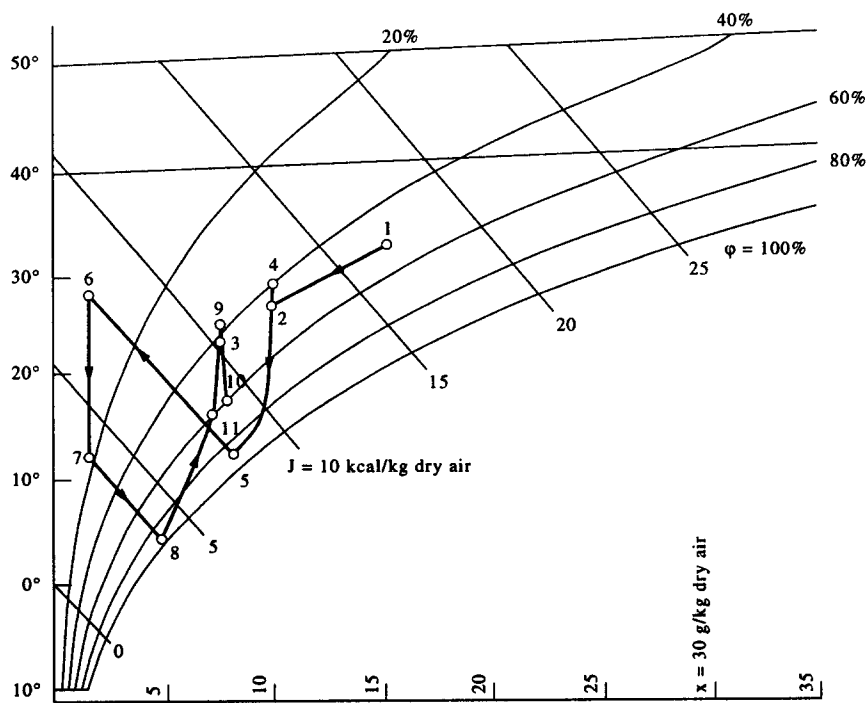


Fig. 13. The processes of the plant of Fig. 12 on the psychrometric chart.

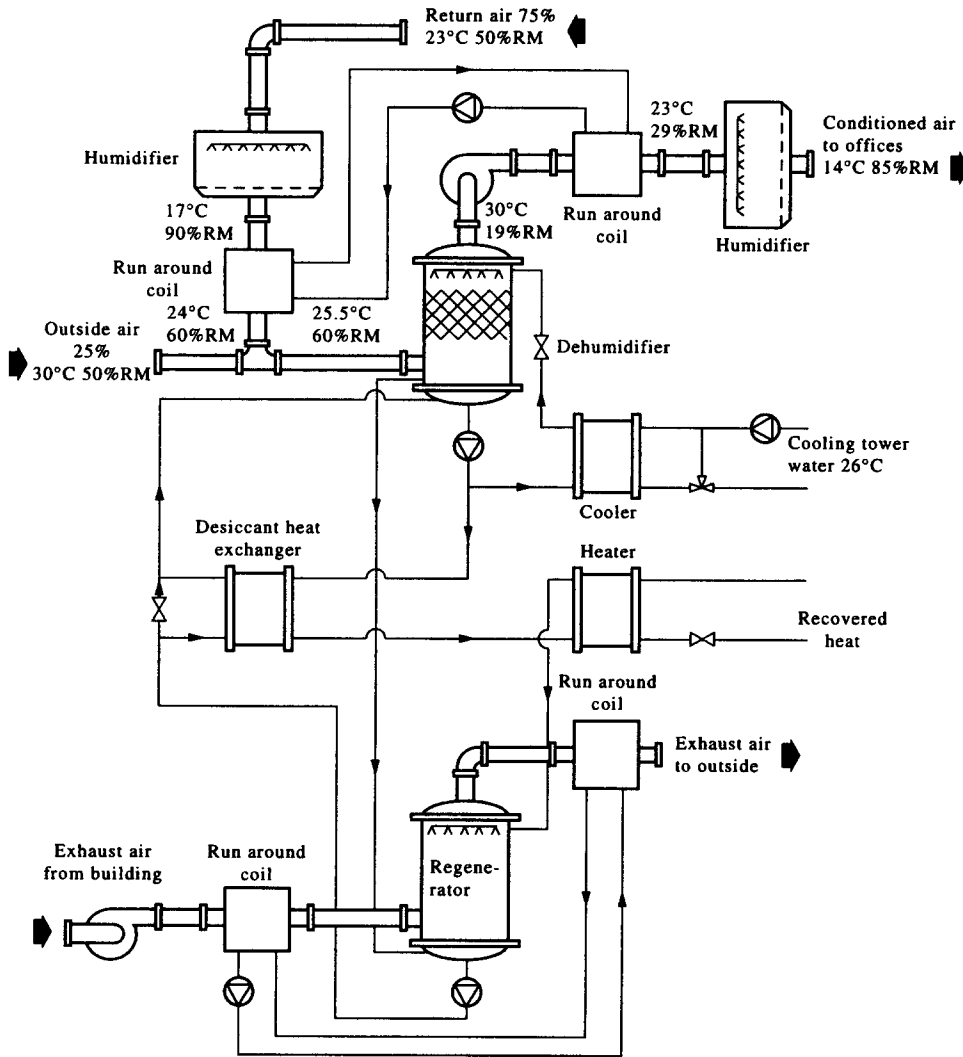


Fig. 14. Scheme of a liquid sorption HVAC system.

laboratories of various firms to promote the exchange and development of new technologies). Basically one can distinguish a centralised air-conditioning plant, which directly treats the air, and local air-conditioners. The former is an open cycle absorption system, driven by the i.c. engine heat recovery. In the cooling mode the heat is rejected to the outside air in a pad evaporative cooler. In the heat pump mode the heat is extracted from the exhaust air. The local air-conditioners are driven by the electricity produced by the cogeneration group, rejecting the heat in cooling mode to a vertical tube earth exchanger, taking heat from the ground in the heat pump mode.

Winter mode (Fig. 20)

Starting from the conditioned room, one notes a reversible unitary heat pump and the room terminal for air supply and withdrawal. The fresh air is supplied to the room suitably treated in the centralised HVAC plant; the air is mixed with the recirculated air and can be humidified on site before supply. Thus very fine control both of temperature and humidity is possible for every room. The exhaust air is sent to a packed bed sorption dehumidifier, fed by liquid desiccant. The air is dehumidified and then preheats the fresh air in a gas-gas heat exchanger. Finally it is exhausted at a very low enthalpy (peak winter conditions, dry bulb temperature 6.5°C, humidity ratio 2.4 g/kg, enthalpy 12.5 kJ/kg). The dilute desiccant leaving the packed tower must be regenerated. The reactivation is implemented in a vessel heated by hot water, at a sub-atmospheric

pressure, where water vapour is separated. The hot concentrated desiccant brine preheats the dilute solution in a liquid–liquid exchanger, then it preheats the fresh air in a liquid–gas exchanger. Thus the fresh air is preheated twice, by the exhaust air and by the concentrated solution, arriving in peak winter conditions at 27°C. The fresh air is further heated in the condenser. A large fraction of the heat of reactivation is recovered here.

The fresh air can be sent to the conditioned rooms, where it is mixed with the recirculated air and humidified according to the needs.

The building electric demand is mainly satisfied by a reciprocating i.c. gas engine which drives an electric alternator. Of course connection with the utility is provided and electricity purchases or sales can be negotiated with the National Electric Board according to Act 9/91, mentioned in a previous paper [10].

Heat recovery is operated on the engine exhaust and jacket. In principle the engine flue gases could be treated together with the exhaust air, further improving system performance. The recovered heat drives the sorbent regenerator.

The unitary electric heat pump cold source is a water circuit connected to a vertical tube earth exchanger, so that the temperature is favourable and rather uniform all year round. Heat pumps directly heat the ambient air without distribution losses. The seasonal COP can be as high as 4–5 due to the favourable temperature levels at the condenser and evaporator.

Summer mode (Fig. 21)

The scheme is similar to the above one. Here heat pumps operate in the cooling mode (four-way valve). A further gas–gas heat exchanger pre-cools the fresh air. The gas–gas heat exchanger, that was employed in winter to preheat the fresh air, is now used to cool a fraction of the recirculated air with outside evaporatively cooled air.

Starting from the room air withdrawal, a fraction is exhausted after precooling of the fresh air. The remaining air is, instead, treated in the packed bed. The air is dehumidified and purified by the desiccant. The design conditions are 10.5 g/kg for the humidity ratio at the inlet of the dehumidifier, 3 g/kg at the outlet, a temperature of 26°C at the inlet and more than 30°C at the outlet. The treated air is then cooled in the gas–gas heat exchanger by evaporatively cooled outside

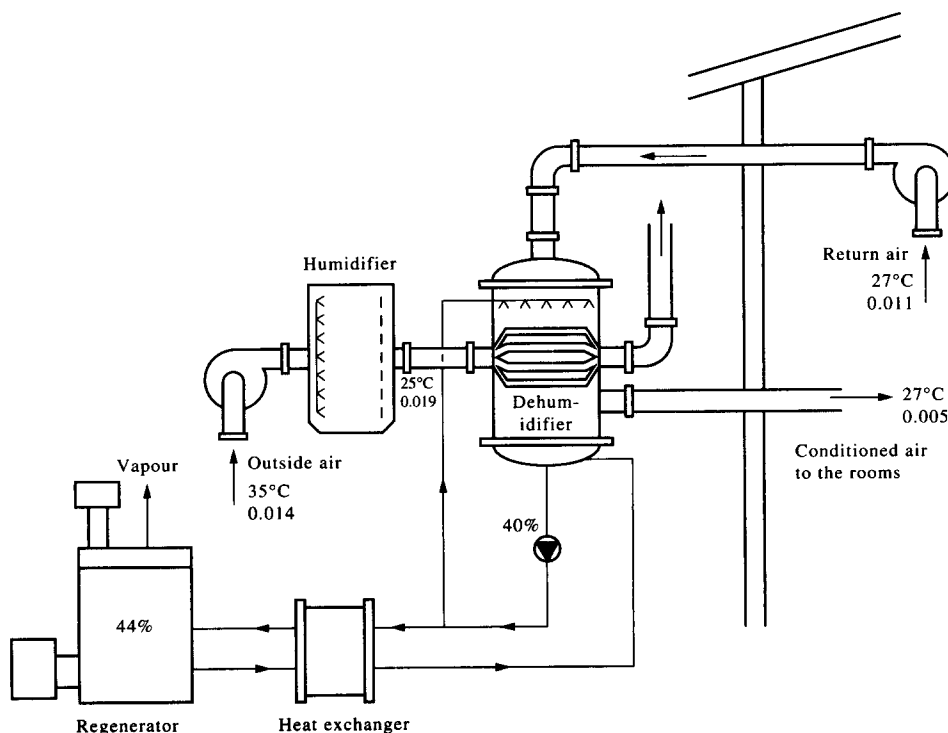


Fig. 15. Block diagram of an integrated liquid desiccant HVAC system.

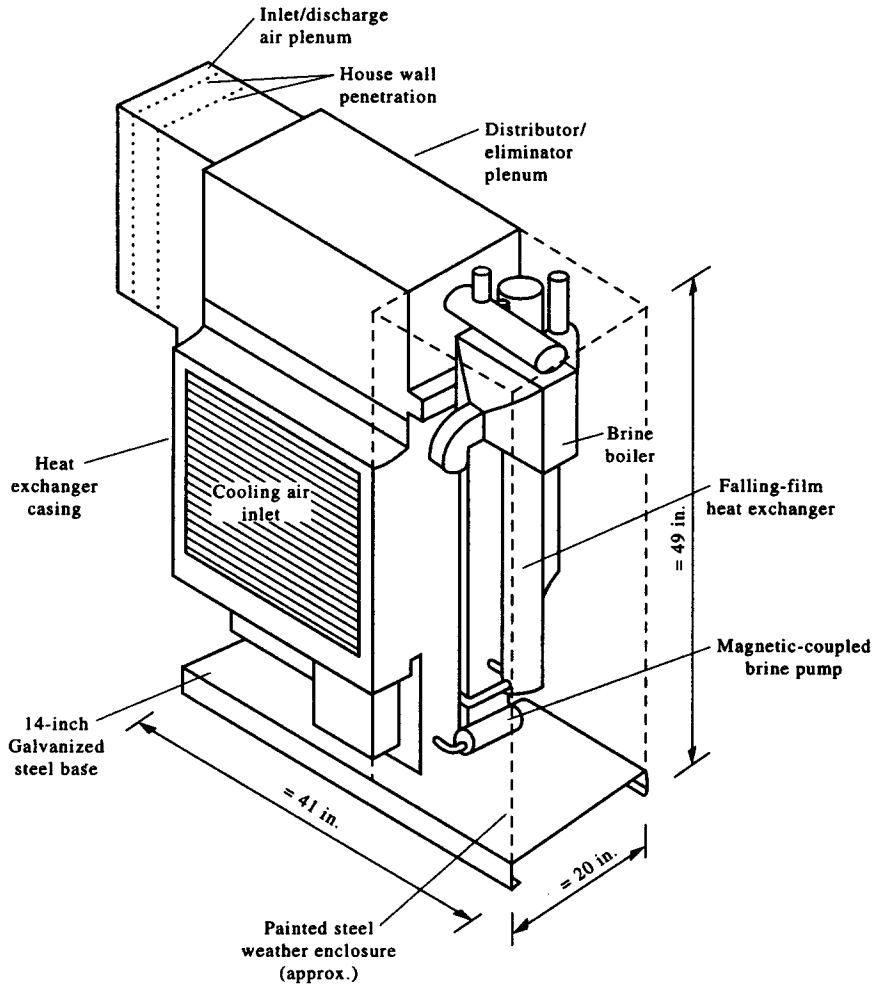


Fig. 16. Air treatment unit for the system of Fig. 15.

air. This air then finds utilisation in cooling the sorbent brine before introduction in the packed column, and finally in the condenser cooling.

The recirculated air, now dehumidified and cooled to a temperature of about 26–28°C in the gas–gas heat exchanger, is sent to the conditioned rooms after mixing with fresh air. Further cooling of the recirculated air before its supply can be obtained in an evaporative cooler, as the withdrawn water in the packed column is higher than the latent load (in mass) due to internal loads and to ventilation. A fraction of the sensible load is satisfied by the unitary heat pump whose heat sink is a vertical tube earth exchanger. The unitary heat pumps are driven by the electricity produced by the i.c. engine, which also covers all the building electric loads in summer. The heat recovery on the engine drives the sorbent regenerator so that the recovered heat is useful even in summer.

Since the latent loads are taken by the primary air, the unitary heat pumps can operate at evaporator temperatures higher than usual. A very high COP is expected, with a seasonal value of about 3–4.

Energy needs

A seasonal comparison was developed in terms of primary energy between the described system and a conventional one, where the heating is obtained by a gas boiler (seasonal efficiency 0.85), with a gas–gas heat exchanger between exhaust and fresh air at 60% efficiency. In summer the conventional plant is supplied by a compression chiller whose seasonal COP is set at 3. The

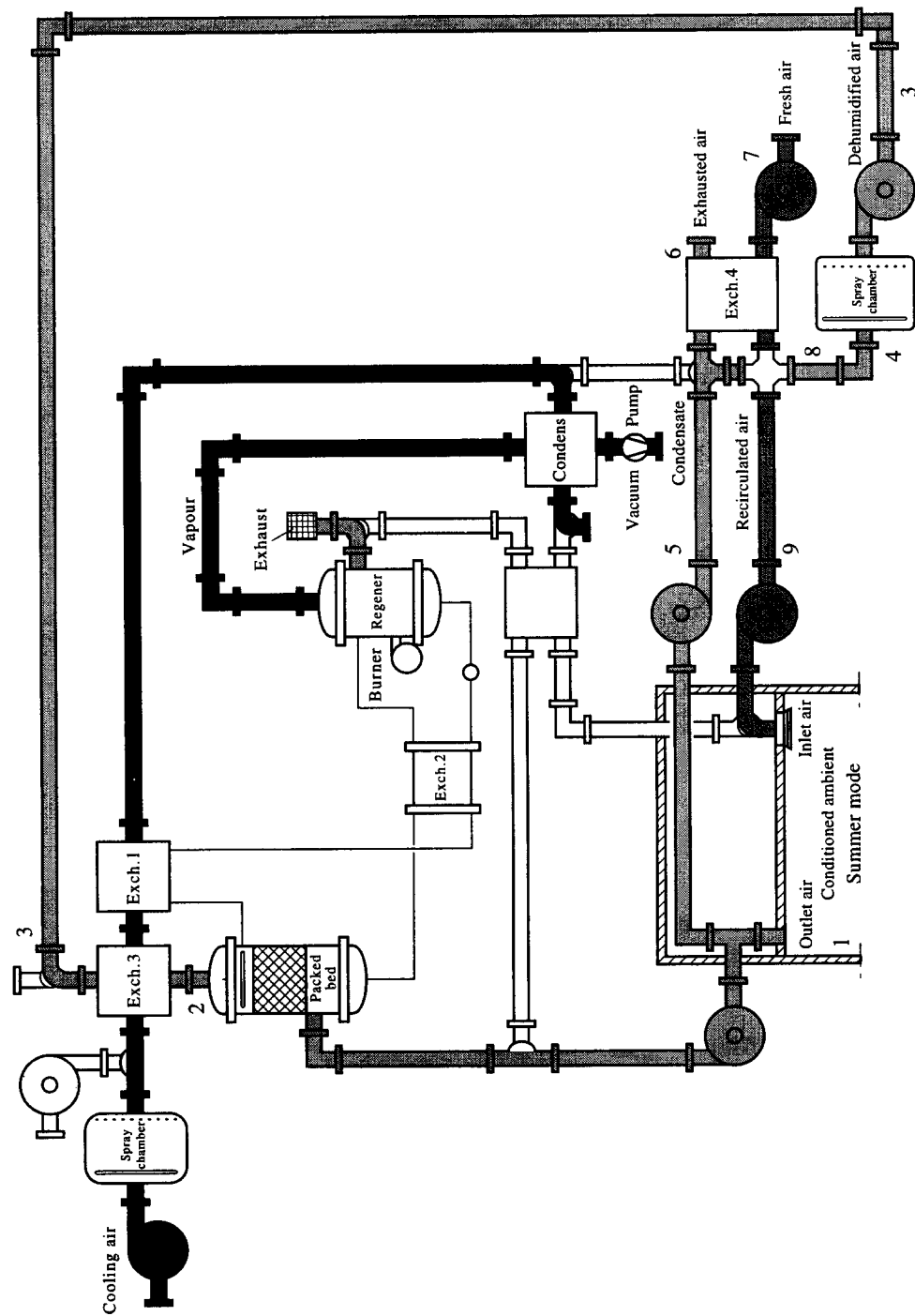


Fig. 17. Scheme of the open cycle absorption system in cooling mode.

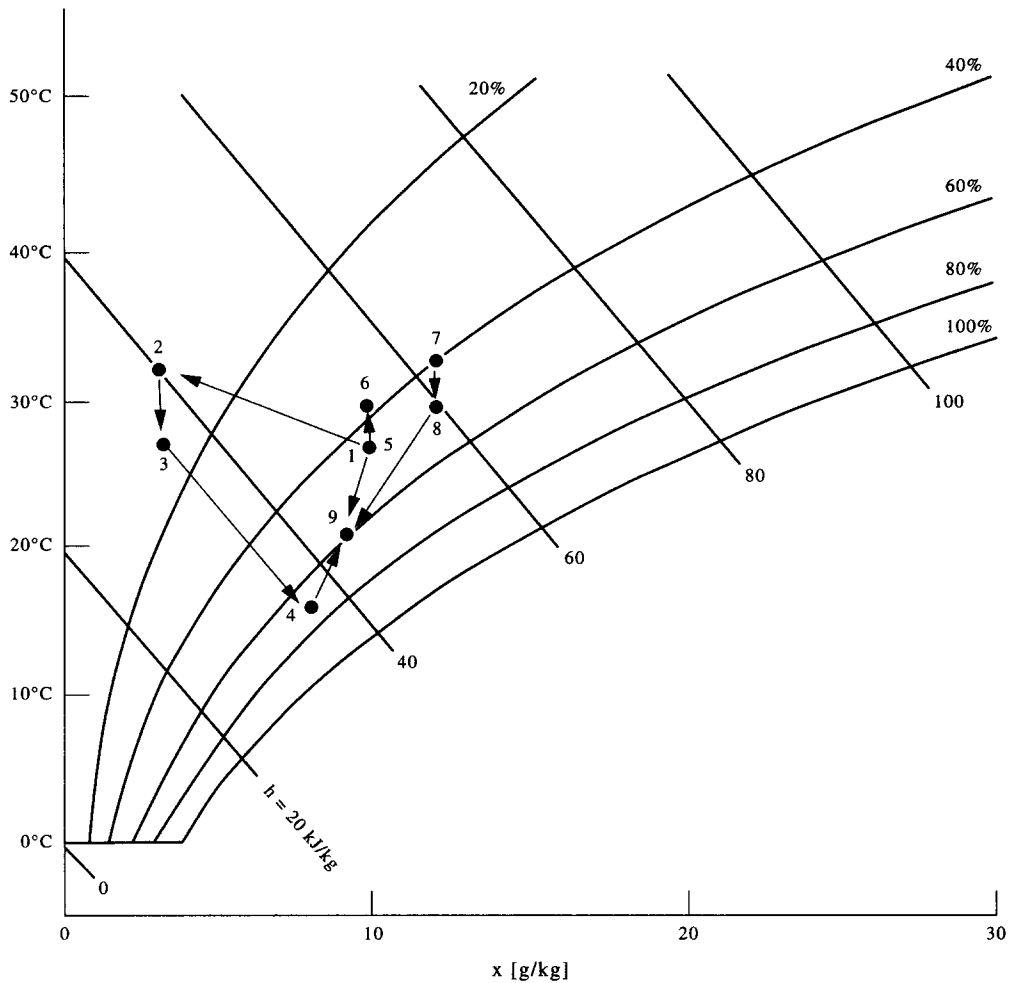


Fig. 18. The open cycle absorption system in cooling mode on the psychrometric chart.

conventional plant is an all-air plant with postheating so that the supply temperature is at 18°C [25, 26].

The CHP group in the considered plant has an electricity production efficiency of 33%, whereas the thermal recovery is 60%. The unitary heat pumps have a seasonal COP of 4.5. As illustrated, the cogenerator must supply electricity for lighting and emf for the building and to drive the unitary heat pumps. When the cogenerator is working the system is practically self-sufficient for energy. In fact the engine heat recovery drives the sorbent regenerator. Then only the fuel to drive the gas engine must be accounted for. In summer the unitary heat pumps show a seasonal COP of 4. Again the engine heat recovery is enough to drive the regenerator and no other energy input need be supplied except for the natural gas to operate the engine.

The usual 0.33 efficiency is assumed for production and distribution of electricity at the utility. The percentage energy savings in the various months are reported in Fig. 22, where one can appreciate the large energy savings both in winter and in summer. Even in the intermediate seasons, although the energy needs are similar, as an equal production efficiency was assumed for the self-generated and the utility electricity, good economical savings can result from the favourable cost of self-generation in on-peak hours.

As far as a yearly evaluation is concerned, the overall primary energy savings can be higher than 50%.

At the same time the air quality can be better due to the fine room control both in terms of temperature and humidity, with good independence of the various spaces, achieved by turning off the unitary heat pumps.

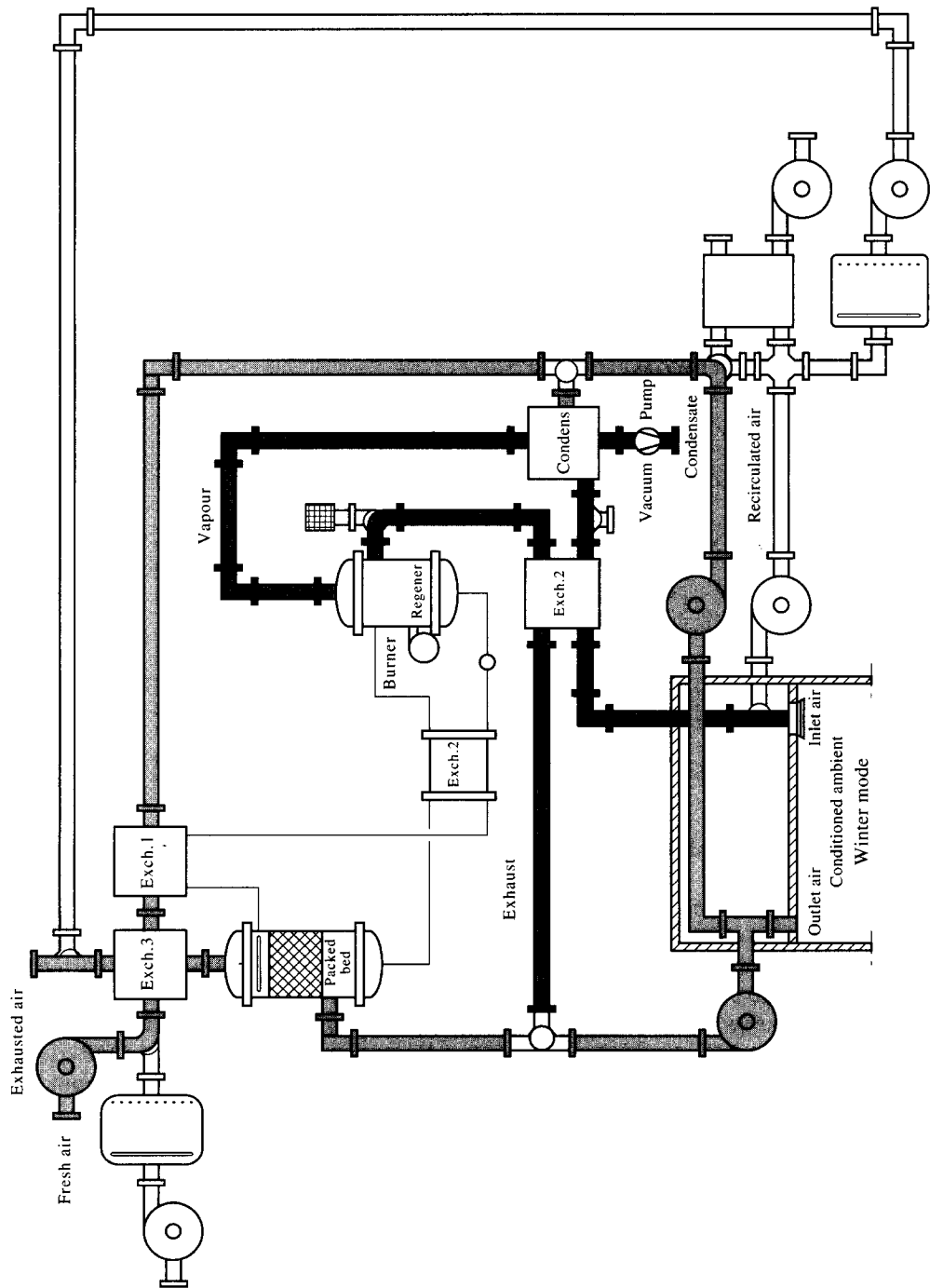


Fig. 19. Scheme of the open cycle absorption system in heating mode.

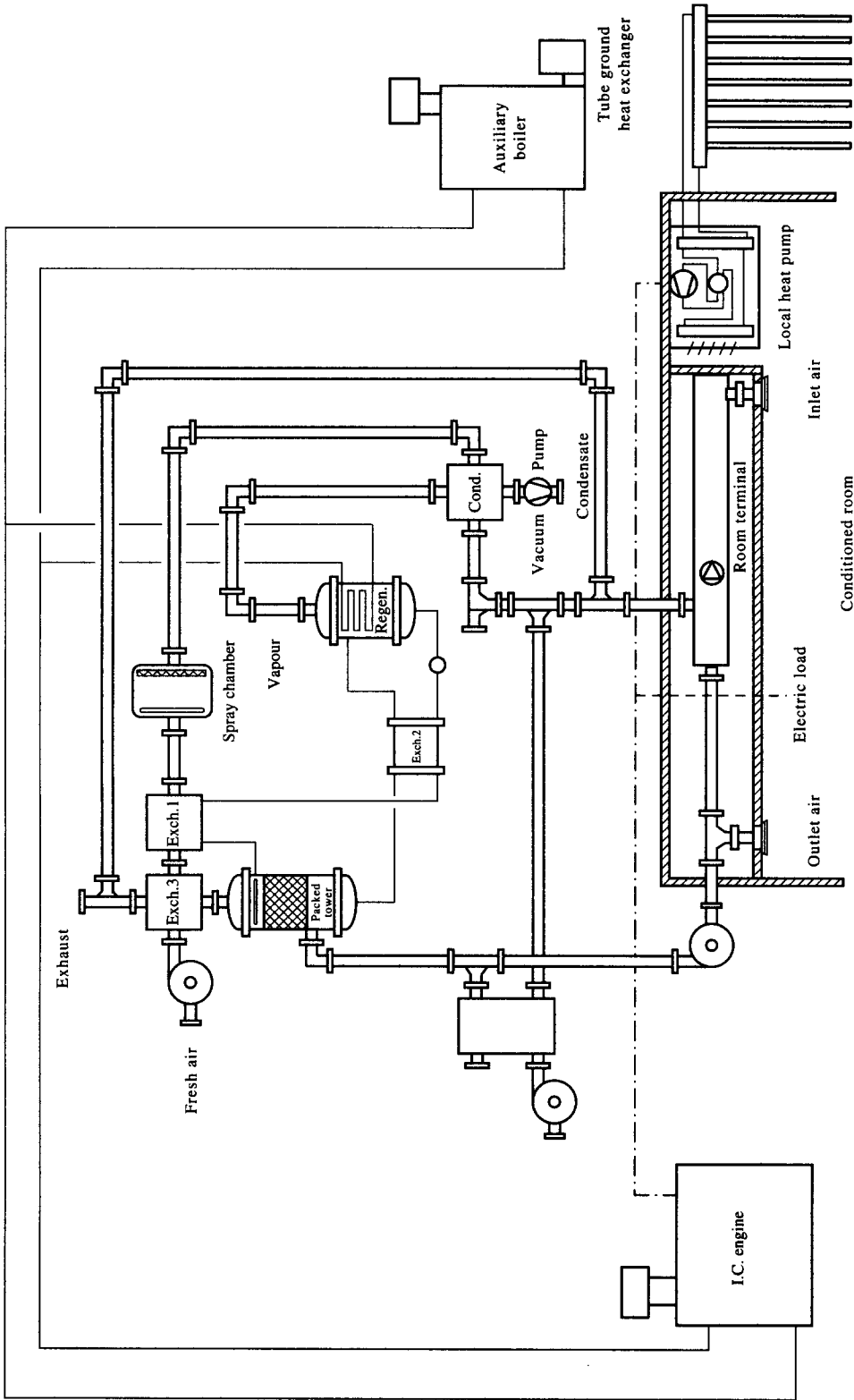


Fig. 20. Scheme of the plant proposed for the Padua Building Innovation Centre in heating mode.

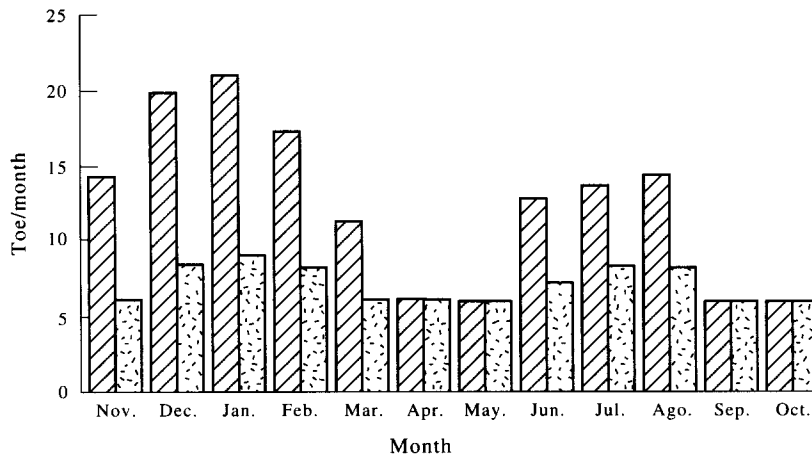


Fig. 22. Comparison of the monthly primary energy needs for a conventional plant and the plant proposed for the Padua Building Innovation Centre.

CONCLUSIONS

The many proposed applications discussed in this paper reveal the interesting possibilities offered by alternative HVAC techniques. The various solutions can be combined with storage, control and management systems which should be evaluated, considering also the utility rates, the availability of heat recovery, and the seasonal and peak demands, so that the most suitable solution is selected. Of course attention must be paid at the same time to the air quality and comfort.

The designer is thus obliged to innovate in choosing the technical solutions as a function of the building use, the climate, the utility rates, etc. Standardised HVAC projects for every situation are no longer acceptable; the work of the engineer must become creative, while accounting where possible for the calculated risk of the innovation.

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