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A NEW HVAC SYSTEM BASED ON COGENERATION BY AN I.C. ENGINE

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Abstract—A new HVAC system is proposed and studied. The plant is equipped with a reciprocating i.c. gas engine which produces electric energy with heat recovery from the jacket water heat and the exhaust. The electric energy serves the lighting and emf for the building and individual room reversible air–water heat pumps that mainly satisfy sensible cooling and a fraction of the winter heating load. An open-cycle absorption system, which treats the primary air, meets the summer latent load and year round ventilation requirements. In winter, the system operates as an open-cycle heat pump. The regeneration of the sorbent is obtained by the i.c. engine heat recovery. The proposed system is analysed by comparing its performance with a traditional plant, both in summer and winter mode. Copyright © 1996 Elsevier Science Ltd

Keywords—Cogeneration, HVAC system, open cycle, absorption.

INTRODUCTION

Cogeneration is gaining interest in Italy for large commercial buildings. This technology is promoted by a recent act which allows producers to sell, exchange or transport the electricity via the Electric National Board (ENEL).

A building needs electric energy for the various services (lighting, computer rooms, emf for pumps, fans, elevators, and so on) and thermal energy for winter heating (and service hot water). An important fraction of the installed electric power is due to air-conditioning when the plant is equipped with traditional chillers to drive the compressors. When such a plant is operated in cogeneration, a large fraction of the cogenerated heat must be rejected to the ambient in summer in the absence of heating needs. The utilization of absorption chillers driven by the i.c. engine heat recovery has been suggested. However, the cost of such devices is hardly justified in a seasonal application and a winter heat-pump mode would be desirable.

A new system would be welcome which was able to utilize low-grade thermal energy, say the 90°C available from the cooling of the engine exhaust and jacket to produce summer cooling and winter heating in heat-pump mode. In this way, the cogenerated heat would find utilization in summer, giving rise in winter to a useful heat-pump effect, upgrading heat from an ambient source or from the exhausted air.

Recently, the project of the Building Innovation Center of Padua was presented at the Commission of the European Communities by the Consorzio Padova Ricerche [1]. The building was designed by adopting advanced criteria to reduce energy consumption. Advanced concepts were also requested for the air-conditioning plant.

The heart of the plant is a cogeneration system, composed of 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 unconventional utilization of the recovered heat to drive a sorption dehumidification system employed both in winter and in summer modes, attaining the objectives previously described [2–4].

SYSTEM DESCRIPTION

The air-conditioning plant is based on a primary air distribution system which meets the ventilation requirements and the summer latent loads. Every space is provided with a local adiabatic humidifier and a unitary heat pump which satisfies winter and summer sensible loads. The system was chosen to guarantee a good level of independence to the various spaces as the users are different. In fact, the building should accommodate offices and laboratories of various firms to promote the exchange and development of new technologies.

Basically, one can distinguish between a centralized air-conditioning plant, which treats the air directly, and local air conditioners. The former is an open-cycle absorption system, driven by the i.c. engine heat recovery. In cooling mode the heat is rejected to the outside air in a sort of wet battery. In heat-pump mode the heat is taken from the exhausted 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 it in heat-pump mode.

Summer mode

A schematic view of the proposed system operating in summer mode is shown in Fig. 1. On the left the reciprocating i.c. engine is represented with the heat recovery system. An auxiliary boiler is provided to integrate, if necessary, the recovered heat. A rejection system to the ambient is not represented.

In the packed-bed tower the air can be dehumidified by a sorption solution such as lithium bromide–water. A fraction of the recirculated air is dehumidified and its temperature is raised by contact with the warmer sorbent and due to the heat of absorption. The air is then cooled in a gas–gas heat exchanger (EXCH.3) by adiabatically saturated outside air. The low-humidity air is sent to the room terminals, where it undergoes humidification cooling after being mixed with fresh air. This fresh air is precooled by exhausted air from the room in a gas–gas heat exchanger (EXCH. 4). The air introduced into the rooms satisfies ventilation and latent load needs (its humidity ratio is lower than that of the room). The whole or a fraction of the sensible load is met by a unitary compression air conditioner of the air–water type. The heat sink is a system of vertical-tube ground heat exchangers which assures all year round favourable temperature levels, working both as a sink and a source [5]. Since the compression air conditioner does not have to dehumidify the air, its operating temperature, which must not be lower than the dew point, is significantly higher than usual and the resulting COP can be quite good. The dilute sorbent leaving the packed bed must be regenerated. The solution is preheated in a liquid–liquid heat exchanger (EXCH. 2) before entering the regenerator driven by the recovered heat. The hot regenerated solution must be cooled so it can absorb effectively. The cooling is done in an air–liquid heat exchanger (EXCH. 1) by outside saturated air.

Winter mode

The winter operation of the system is illustrated in Fig. 2. The air extracted from the heated room is 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 exhaust the treated air preheats the outside fresh air in a heat exchanger (EXCH. 3). A further preheating is given by the sorption solution in an air–liquid heat exchanger (EXCH. 1) before the solution enters the packed bed. The preheated fresh air is humidified in the spray chamber and finally takes the heat of condensation of the vapour separated in the regenerator and is sent to the room terminals. Here the mixture of fresh and recirculated air can be admitted into the room. The sorbent cycle is as before.

PERFORMANCE

The Building Innovation Center is a low-energy building with a heated volume of 20,000 m^3 and a heated floor area of 6400 m^2 (Fig. 3). The surface/volume ratio has the favourable value of 0.3 m^2/m^3. The electric load was assumed on the basis of 10 W/m^2 of floor area (50% concurrency). The presence of people is estimated on the basis of 1 person/20 m^2. The ventilation rate is set at

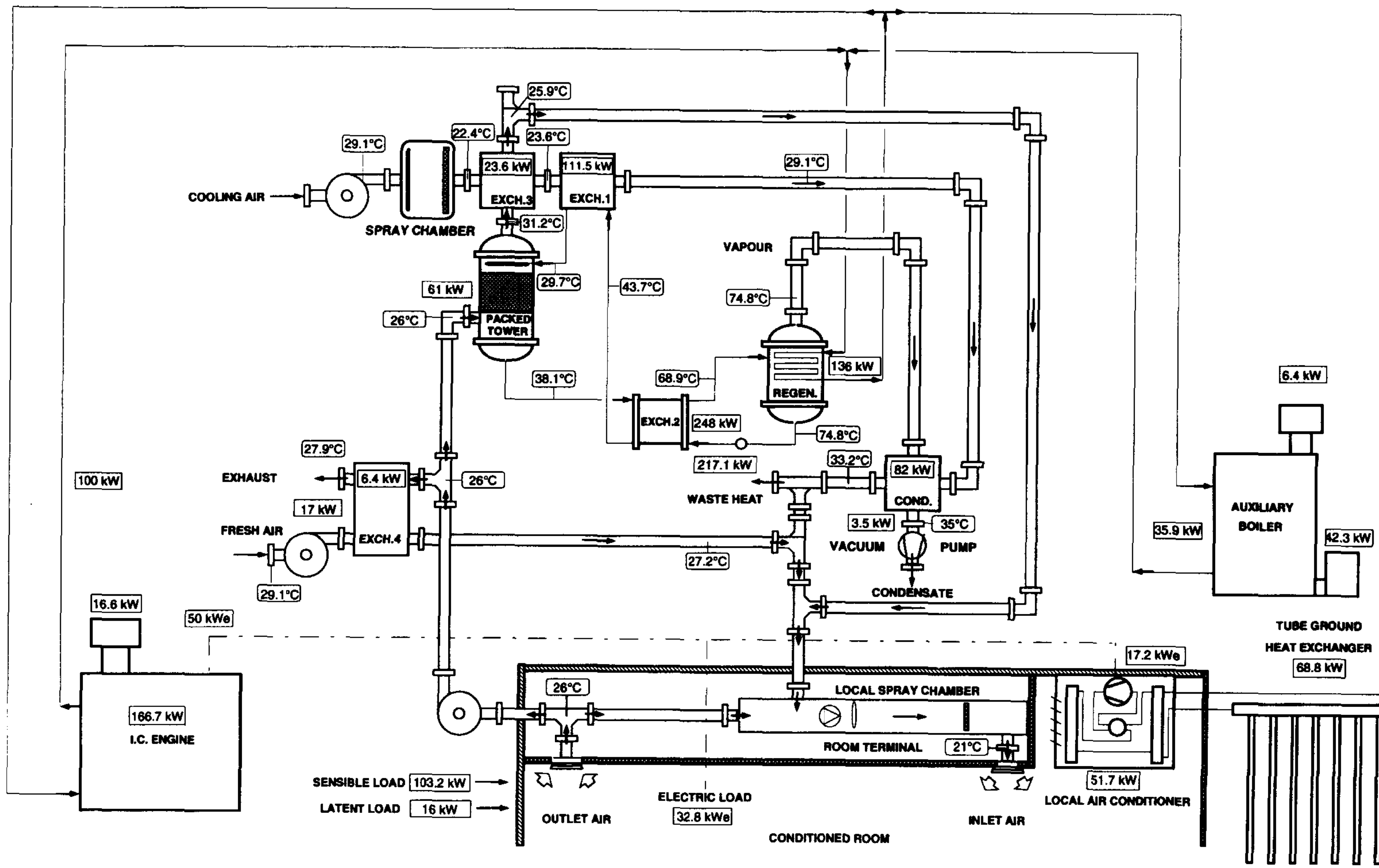

Fig. 1. Block diagram of sorption dehumidification system in summer-mode operation: July conditions.

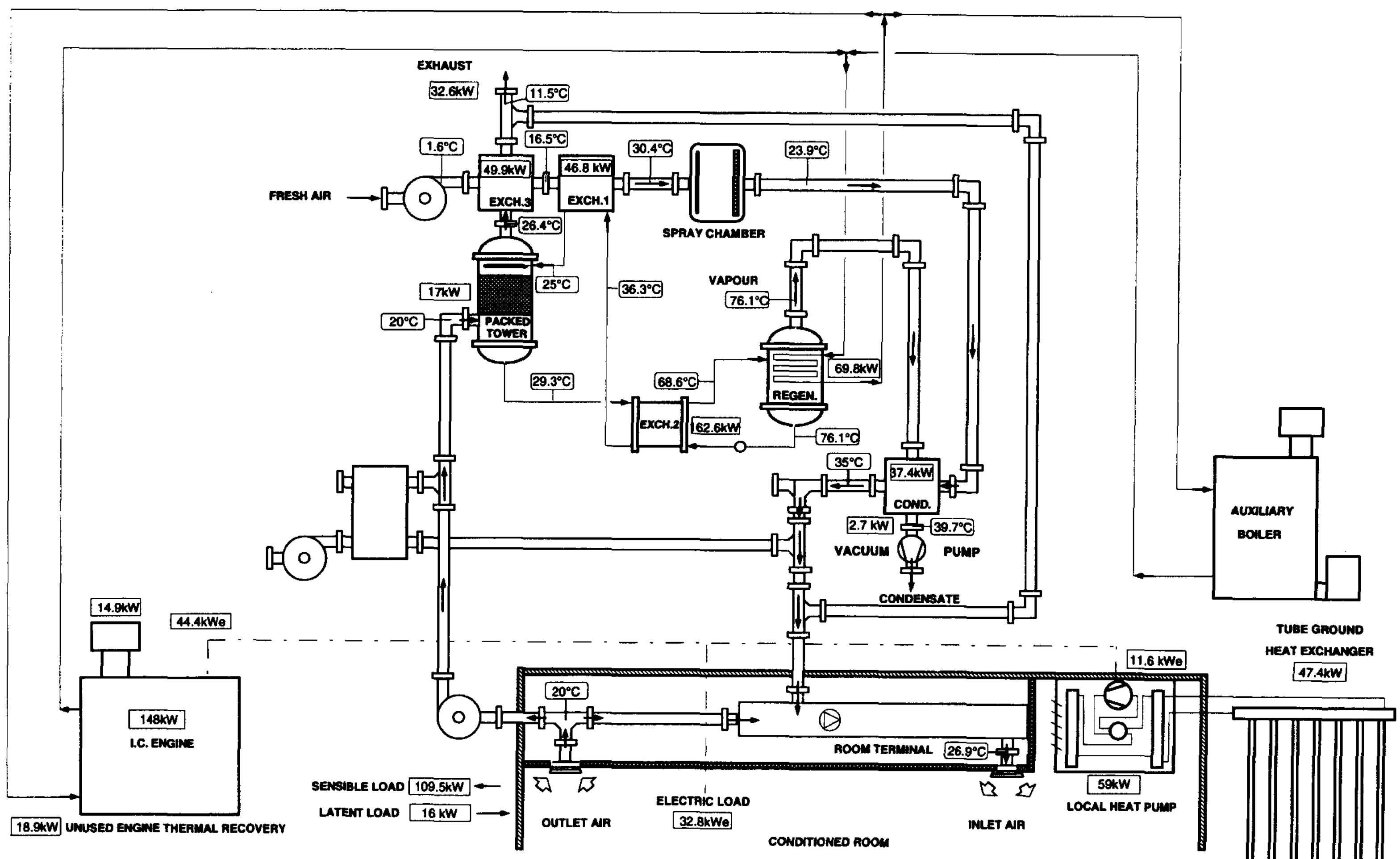

Fig. 2. Block diagram of sorption dehumidification system in winter-mode operation: January conditions.

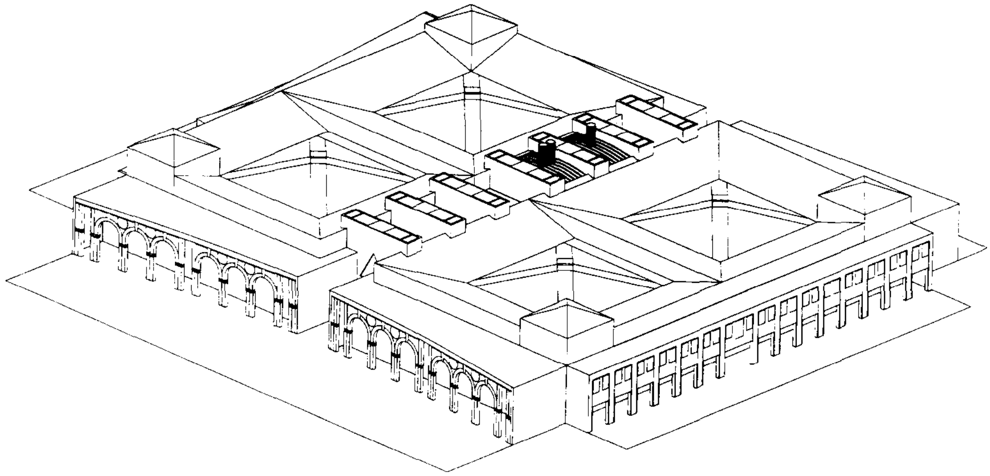

Fig. 3. View of the Innovation Building Center.

0.5 vol/h. Table 1 reports the loads under monthly average conditions during winter and summer for the building.

A reciprocating i.c. gas-driven engine is assumed whose mechanical power is 50 kW, with an efficiency of 30% and a heat recovery of 60% (10% losses). As regards the unitary compression machines, their COP is around 5 in heat pump mode and around 3 as a chiller (40% of the Carnot values). The packed-bed tower is evaluated via a detailed heat and mass transfer balance with different concentrations of LiBr–H_2O mixture [6].

Table 2 reports the assumed efficiencies for the various devices.

The system is compared with a traditional one equipped with an electrically driven compression chiller (COP = 3), a natural-gas boiler (whose seasonal efficiency is 0.85) and a heat exchanger between fresh and exhausted air at an efficiency of 60% (reference plant).

For the evaluations of the primary energy ratio (PER) the usual 0.30 efficiency is assumed for production and distribution of electricity. The traditional system operates the dehumidification via cooling and condensation so that reheating is provided.

Summer performances

The traditional plant demands electric energy for emf, lighting and to drive the compression chiller. The cooling load is increased by reheating. The reheating also has a heating cost, unless it is provided by heat recovery at the condenser of the chiller, where it could, however, lower the COP.

Let us consider in Fig. 4 an energy balance of the reference plant in average July conditions. The electric loads are estimated as 117 kWe (33 kWe for the building services and 84 kWe for the compressors). In terms of primary energy this gives 392 kW input to the thermoelectric plant (supplying the electric grid) with 275 kW of losses to the ambient. The thermal gains from the outside and the internal gains (except electricity) add up to 110 kW (about 6 kW can be recovered with the heat exchanger between fresh and exhausted air). The reheating contributes with 119 kW, which means a boiler input of about 140 kW (0.85 efficiency).

Table 1. Operative conditions in winter and summer modes

Month	Nov	Dec	Jan	Feb	Jun	Jul	Aug
Temperature (°C)	7.2	2.9	1.6	4	21	23.6	22.6
Humidity ratio (g/kg)	6	4	3	4	10	12	12
Sensible load (kW)	−59.5	−97.0	−109.5	−87.4	87.5	103.2	106.2
Latent load (kW)	16	16	16	16	16	16	16
Fresh air load (kW)	−55.8	−87.3	−99.8	−83.6	1.7	23.4	19.9
Thermal load (kW)	−99.3	−168.3	−193.3	−155	105.2	142.6	142.1
Electric load (kW)	32.8	32.8	32.8	32.8	32.8	32.8	32.8

Table 2. Sorption dehumidification system specification

Month	Winter	Summer
Solution concentration	56% LiBr	58%
Flow rate ratio L/G	0.6	0.9
Exchanger 1 (liquid/gas) efficiency	70%	70%
Exchanger 2 (liquid/liquid) efficiency	85%	85%
Exchanger 3 (gas/gas) efficiency	60%	60%
Exchanger 4 (gas/gas) efficiency	—	60%
Condenser (air-cooled) efficiency	60%	60%
I.C. engine electric output	30%	30%
I.C. engine thermal output	60%	60%
Local compression heat pump COP	5	—
Local compression chiller COP	—	3

The global primary energy input to the system amounts to 532 kW (392 kW from the electric grid plus 140 kW from the boiler). This is the figure to be compared with the proposed system.

As regards the proposed system, see Fig. 5 for an energy flow balance for the previous considered July average conditions.

The proposed system essentially needs natural gas to drive the i.c. engine. About 167 kW are required to give 50 kWe of electricity, used to meet the building needs and to drive the unitary air conditioners. The requirements of the sorbent regenerator are met by the recovered heat (100 kW) and, when necessary, by an auxiliary boiler (42 kW). Of course both the i.c. engine and the boiler exhibit losses toward the ambient (respectively 17 and 6 kW). The thermal gains are as before (110 kW) with the same heat recovery between fresh and exhausted air (6 kW). The local air conditioners are driven by about 17 kW, with a cooling effect of 51 kW and a rejection to the earth exchanger of 69 kW.

The open-cycle absorption system rejects to the outside air about 217 kW (air–air heat exchanger 3, air–liquid heat exchanger 1 and condenser). The 3.5 kW of condensate losses are the losses associated with the discharge of hot condensate.

The global primary energy input to the system adds up to only 209 kW, i.e. the primary energy saving exceeds 60%. Of course a large fraction of the saving is possible by eliminating the reheating, though a close control of space conditions is still achievable.

Winter performances

The traditional plant demands, like in summer, electric energy for the building needs. Moreover, natural gas is needed to supplement the boiler. Figure 6 reports the energy flow balance for the average January conditions. The primary energy to produce 33 kWe for the electric needs of the

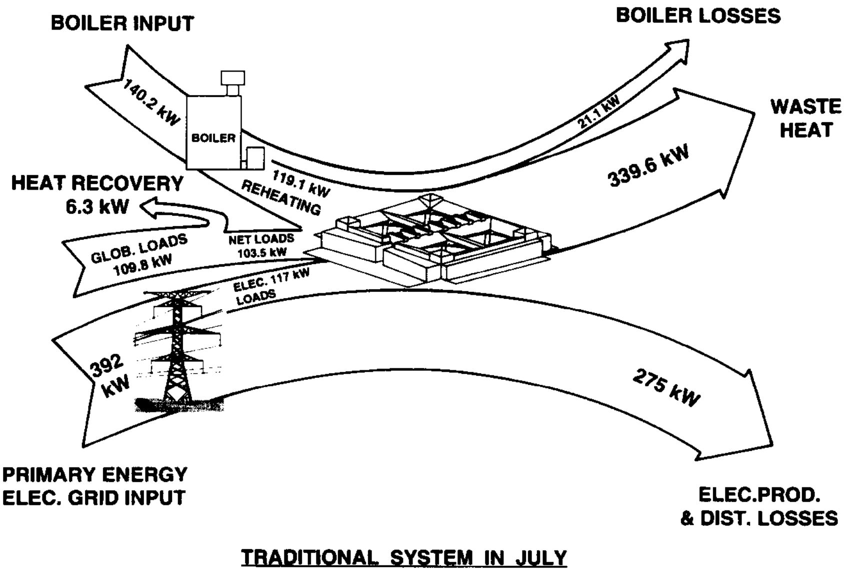

Fig. 4. Thermal balance for the traditional system in summer-mode operation.

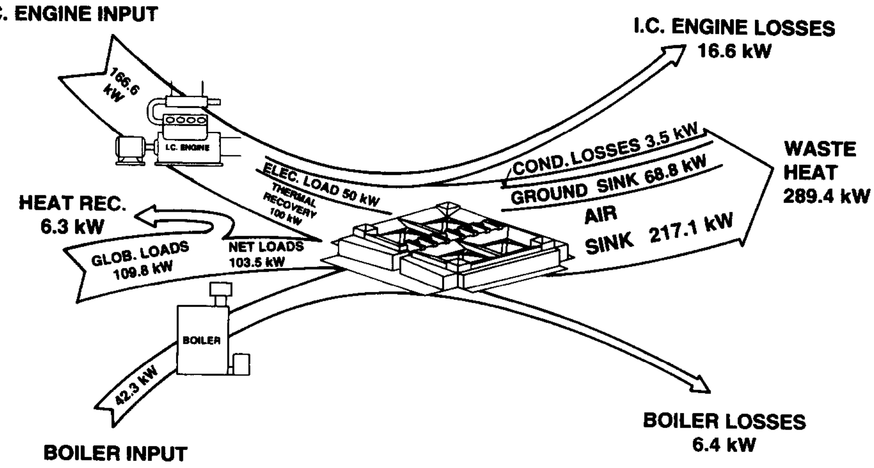

Fig. 5. Thermal balance for the proposed system in summer-mode operation.

building amounts to 109 kW. The transmission losses are 126 kW, whereas ventilation and infiltration give 100 kW. A heat recovery of 37 kW is allowed by the heat exchanger between fresh and exhausted air. The electric loads can be considered as internal gains for 33 kW to alleviate the thermal loads. On the whole, heating of 156 kW must be supplied, which means a boiler input of 184 kW. Thus the global primary energy input is about 293 kW (109 kW from the electric grid and 184 kW from the boiler).

These performances can be compared with the proposed system by means of the flow diagram of Fig. 7. Again, the proposed system mainly needs natural gas to drive the i.c. engine. The engine produces electricity for the building needs and to drive the unitary heat pumps (respectively, 33 kWe and 12 kWe). The sorbent regenerator is energized by the recovered heat (70 kW) and, if necessary (but the average January conditions do not require it), by an auxiliary boiler.

The primary energy input to the i.c. engine is 148 kW and this satisfies the various building needs. The unitary heat pumps give 59 kW, taking 47 kW from the earth source. The open-cycle heat

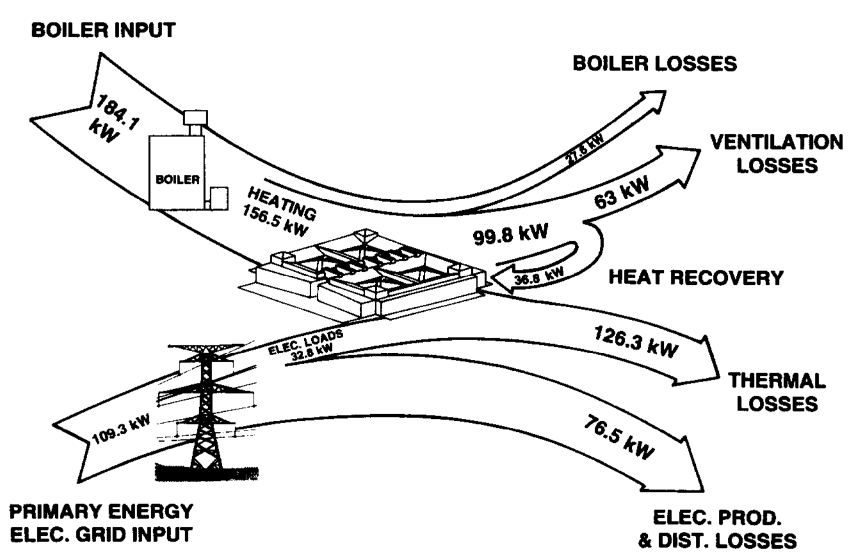

Fig. 6. Thermal balance for the traditional system in winter-mode operation.

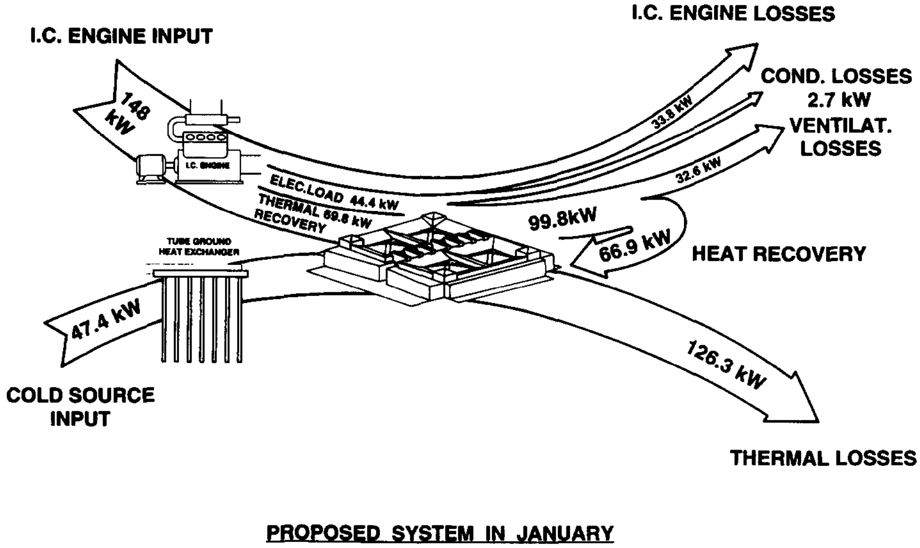

Fig. 7. Thermal balance for the proposed system in summer-mode operation.

pump allows one to increase the heat recovery from the exhausted air from 37 kW for the traditional system (60% efficiency air–air heat exchanger) to 67 kW, so that the ventilation losses are now only 33 kW instead of 63 kW. Of course the i.c. engine losses must be accounted for (34 kW, higher than in summer, as the requirements of the sorbent regenerator are lower than the possible recovery). As before, a small loss (3 kW) is due to the condensate discharge.

The energy saving is still high, around 50%, since the sorption treatment of the air not only recovers heat from the i.c. engine, but it operates as an open-cycle heat pump whose source is the exhausted air [7]. The PER of this heat pumps reaches values as high as 1.4, typical of engine-driven compression heat pumps.

CONCLUSIONS

A new HVAC system provided with cogeneration is described. It comprises an open-cycle absorption system, driven by the engine heat recovery, which satisfies the summer latent loads and year-round ventilation requirements. In winter it operates as an open-cycle heat pump whose source is the exhausted air. Winter and summer sensible loads are met by a unitary heat pump of the air–water type connected to a tube ground heat exchanger. The performances are analysed both in summer and in winter: the comparison with the best conventional system in terms of primary energy is quite favourable, exceeding a 60% saving in summer and 50% in winter.

A thorough analysis via the program TRNSYS is being carried out to account for instantaneous losses and gains, as well as for variable electric loads, evaluating the annual savings both from the primary energy point of view and economically taking into account even the timetable structure of the Electric National Board rates.

An experimental analysis is in progress to survey the real ability of desiccants in absorbing and releasing water vapour in a test rig [8] equipped with a packed tower and a regenerator connected to a condenser operating under vacuum.

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