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A STORAGE HEAT PUMP USING AN 'OZONE-FRIENDLY' REFRIGERANT

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SUMMARY

We describe the integration of a chemical and a vapour-compression heat pump for energy storage applications. The vapour-compression system is designed to operate using the cheap rate "Economy 7" electricity tariff. The system is characterised thermodynamically using various refrigerant/absorbent pairs in the chemical storage circuit and an ozone-friendly refrigerant, R134a, in the vapour-compression circuit. Results indicate that the H_2O/Na_2S pair provides a high energy storage density and is the most suitable for use in this system.

This paper also describes the design features of a domestic-sized version of this heat pump system. Air in the sunspace (conservatory) of a house was used as a heat source for the heat pump.

KEY WORDS - Heat Pump Storage Chemical R134a Sunspace

INTRODUCTION

Continuing concern over the environment and supplies of fossil fuels has highlighted the need for energy conservation. A storage heat pump making full use of electricity's low cost "Economy 7" tariff is clearly attractive as it would have a lower running cost than conventional heating systems. Widespread use of storage heat pumps would allow "levelling" of electric power curves, which currently peak in the mid afternoon and fall off dramatically at night, and thereby improve the efficiency of electricity generation.

At present, electric power plants are divided into three categories according to load. Base-load plants are used to service that part of the demand which continues 24 hrs a day, every day of the year. These plants are designed to operate with the highest efficiency using the least expensive fuel available. Intermediate-load plants are used to service most daily load variations and are generally shut down at night. These plants are usually oil-fired stations. Peak-load plants operate for only a few hours in the day (11 am - 3 pm) and are driven by gas turbines. The cost-efficiency of electricity generation would be improved if 'off-peak' power generated by base-load plants were stored for use during periods of peak demand. This would reduce the need for fuel-burning peaking equipment as well as reducing overall consumption. Furthermore, the use of energy storage systems would assist deceleration of the "Greenhouse Effect" by enabling electric power plants to run at a constant moderate load so decreasing their gaseous emissions.

One of the most important measures of performance of an energy storage system is its energy storage density. Use of a chemical energy store is, without doubt, the best method for achieving a high storage density and if this is combined with a vapour-compression heat pump designed to operate overnight using the cheap rate "Economy 7" electricity tariff, the combined system would have a sufficiently low running cost to make it economically attractive.

The coefficient of performance of a heat pump is largely dependent on the heat source used. If outside air is forced to enter a heat pump evaporator via a sunspace in a house, the combined heat from the ambient air and solar radiation would be an effective heat source for a storage heat pump.

Chlorofluorocarbons (CFCs) are commonly used as refrigerants for vapour compression systems. However, growing awareness of the danger to the earth's protective ozone layer caused by the build-up of CFCs has stimulated industry's efforts to produce "ozone-friendly" alternatives such as R-134a, R152, R-123 and R-141b. This paper describes a chemical compression heat pump employing R134a which is a zero ozone-depletion-potential refrigerant.

DESCRIPTION OF THE HEAT PUMP

The basic refrigerant circuit of the chemical-storage heat pump is shown in Figure 1. The chemical heat pump circuit contains a refrigerant/absorbent pair (e.g. H_2O/Na_2S) and the vapour compression circuit contains R-134a.

The heat pump cycle has two operational phases, namely regeneration and heat pumping. The regeneration phase occurs during the night hours when cheap electricity is available under the "Economy 7" tariff. Initially, the refrigerant/absorbent pair is chemically combined in the generator/absorber. As the compressor starts to operate, the refrigerant vapour separates from the absorbent, latent heat is re-cycled and the refrigerant is stored as a liquid in the accumulator.

The heat pumping phase occurs during the day. Warm exhaust air from the sunspace is chilled in the evaporator and the extracted heat causes the liquid refrigerant in the evaporator to boil. The vapour is then recombined with the absorbent with the emission of heat in the generator/absorber. The heat could be used to warm the interior air of a building and the domestic hot water.

CYCLE ANALYSIS

The pressure- temperature relationship for an absorption process using H_2O/Na_2S is shown in Figure 2. The ideal cycle for the heat pump is shown in Figure 3 and its operational phases are described below:

i) Regeneration Phase

Compressor power input, $P = m\ (h_2 - h_1)$ (1)

The quantity of refrigerant desorbed from the generator/absorber equals that collected in the liquid accumulator and is given by:

$$m_g \quad = \quad m\,(h_2 - h_3)\,/K \tag{2}$$

A small heat input is required at the evaporator to ensure thermal balance. This is given by:

$$Q_e \quad = \quad m\,(h_1 - h_4) - m\,(h_2 - h_3)\,(h_{6(v)} - h_{5(l)})\,/K \tag{3}$$

ii) Heat Pumping Phase

Heat emitted in the generator/absorber is given by:

$$Q_g \quad = \quad m\,(h_2 - h_3)\,[1 - h_{6(v)} - h_{g5})\,/K] \tag{4}$$

Heat required at the evaporator is given by:

$$Q_{ep} \quad = \quad m\,(h_2 - h_3)\,h_{fg5}\,/K \tag{5}$$

The coefficient of performance is given by:

$$\text{C.O.P.} \; = \frac{(h_2 - h_3)\,[1 - (h_{6(v)} - h_{g5})\,/K]}{(h_2 - h_1)} \tag{6}$$

BACKGROUND

CHEMICAL STORAGE SYSTEMS

The successful operation of a chemical storage system depends largely on the refrigerant/absorbent pair employed. Desirable characteristics of refrigerant/absorbent pairs include; low cost, moderate vapour pressure of the refrigerant, low vapour pressure of the absorbent, chemical stability, high latent heat of the refrigerant and good heat and mass transfer properties. The refrigerant/absorbent pair should also be non-toxic, non-corrosive and non-flammable.

Various refrigerant/absorbent pairs have been used, or suggested for use, in absorption machines. Hainsworth[1] considered 180 refrigerant/absorbent combinations as possible working fluids for absorption systems. Other qualitative studies of refrigerant/absorbent pairs have been carried out by Buffington[2] and more recently by Raldow[3].

The NH_3/H_20 refrigerant/absorbent pair is used for refrigeration applications, e.g. "Electrolux" domestic gas-refrigerator. As water has a high volatility other chemicals such as NaSCN have also been considered for use with ammonia. Studies carried out at the University of Wisconsin demonstrated the superiority of NH_3/NaSCN over NH_3/H_2O with respect to stability. The phase diagram of the NH_3/NaSCN system has been documented in detail by Blytas was published first in thesis form[4] and later in the literature[5]. However the high temperature lift provided by the NH_3/NaSCN pair restricted its use to small number of industrial applications. Other refrigerant/absorbent pairs which have been tested include $NH_3/LiNO_3$, NH_3/KSCN, $NH_3/NaBH_4$, NH_3/NaBr and $NH_3/BaCl_2$.

Experimental work has also been carried out to test mixtures of different absorbents[6]. The $NaSCN/H_2O/NH_3$ and $CaCl_2/H_2O/NH_3$ systems have been found to have of good heat and mass transfer properties and high energy storage densities.

Other workers[7, 8] used insoluble refrigerant/absorbent pairs, such as $NH_3/CaCl_2$ and $NH_3/SrCl_2$ but found that they had poor heat and mass transfer characteristics. Taube[9] Wentworth[10] and Riffat[11] have carried out work with the aim of improving heat and mass transfer in solid absorbent beds. $NH_3/CaCl_2$ and $NH_3/SrCl_2$ were found to form mobile slurries in inert liquids such as n-heptanol and kerosene but energy storage densities were poor owing to the small solid content of these slurries.

As ammonia is toxic, the $H_2O/LiBr$ system has been used as an alternative to the NH_3/H_2O pair, especially in large scale commercial chillers. Other workers[12] investigated the use of the dimethyl-ether-tetra- ethylene-glycol (DME-TEG) absorbent/R21 pair but results showed it to be unsatisfactory.

Investigations have also been carried out on other systems such as $CH_3NH_2/LiSCN$ and $CH_3OH/LiBr$. These systems are most suited to use in continuous absorption machines while the $CH_3OH/CaCl_2$, H_2O/H_2SO_4 and H_2O/Na_2S pairs have been used in chemical heat pump application[13-15]. The N_2O/Ha_2S system has a high energy storage density and has the additional advantage of low cost as both water and sodium sulphide are inexpensive.

A summary of the various methods available for chemical energy storage is shown in Table 1.

Table 1. Comparison of various energy storage systems.

System	Temperature lift (°C)	Heat of absorption (kJ/kg)	Energy storage density (kWh/m3)	Method of storage
$CaCl_2 . 2CH_3OH$	63	1622	259	Solid bed
$CaCl_2 . 4\text{-}8NH_3$	60	2354	350	Solid bed
$SrCl_2 . 2\text{-}8NH_2$	62	2378	231	Solid bed
$Na_2S . 5H_2O$	54	3120	673	Solid bed
$CaCl_2 . 2H_2O\ 0\text{-}6NH_3$	70	2348	56	Solid bed
H_2SO_4/H_2O	38-324	1400	100	Solution
NH_3/H_2O	50-80	-	58	Solution
NH_3/NaSCN	78	1281	65	Solution
H_2O (Sensible heat)	45	-	51	Water

It is clear from Table 1 that the H_2O/Na_2S system provides the highest energy storage density. Use of solid absorbent beds generally leads to higher energy storage densities than obtained with solutions or sensible-storage media.

HEAT SOURCES

The thermal and economic performance of a heat pump are largely dependent on the type of heat source used. Various types of heat sources including air, water, soil and solar energy have been examined[16]. More recently the trend towards increased airtightness of house envelopes has encouraged researchers to investigate the use of exhaust air from mechanical ventilation systems as a heat source[17].

A chemical-compression heat pump requires use of a heat source during the day and we have employed the solar heat gain from the sunspace of a house as a heat source.

DOMESTIC-SIZED HEAT PUMP

We describe here the design features of a domestic-sized heat pump which employs a sunspace as a heat source.

We have carried out computer modelling of the heating load of a house using the Designer/Modeller package. The building used for the model was built to current UK standards. The floor area was $100m^2$ and three people were assumed to be in occupation. The set-point temperatures for the living room and bedrooms were assumed to be 21^oC and 18^oC, respectively. The combined area of the windows in the house was $12m^2$. The U-values for the walls and roof were assumed to be 0.6 and 0.35 $W/m^{2o}C$ respectively, in accordance with British Standards.

External air was assumed to be drawn into the sunspace of the house. Figures 4 and 5 show schematic diagrams of the house using different methods of supplying heat to the evaporator of the heat pump. The first method (Figure 4) is suitable for use with a wet-heating system where the air from the sunspace can be used directly to supply heat to the evaporator. This arrangement allows the heat produced by solar radiation to be extracted at the evaporator. The second method (Figure 5) is suitable for use with a warm-air heating system and uses a high-level evaporator which is located in the roofspace of the house. In this case the heat input consists of solar heat gain from the sunspace/roofspace, stray heat gain from the house and exhaust-air heat input from the ventilation system.

The infiltration rate was assumed to 0.5 ac/h for all rooms and the ventilation rate was assumed to be 2 ac/h for the sunspace. The model showed that the heat gain from solar radiation should provide sufficient enthalpy for efficient operation of the heat pump, even in cloudy climates.

The annual energy requirement for domestic hot water and space heating of the house used for the model was estimated to be approximately 85 GJ/y if the heat pump were not employed. The computer model indicated that this figure would be reduced to approximately 34 GJ/y if a chemical compression heat pump using the sunspace as a heat source were employed.

This would result in an energy saving of approximately 50GJ/y (13,889kW/y) per house. Assuming a market penetration of 2% over 10 years (based on previous heat pump sales[18]), the total energy savings would be approximately 1.2 x 10^{10} TOE (Assumption: 500,000 houses; 2% of these using this type of heat pump, energy saving over a 10 year period, energy saving 50GJ/y/house).

ANALYSIS OF THE HEAT PUMP SYSTEM

The heat pump system is analysed thermodynamically for operation using $CH_3OH/CaCl_2$, $NH_3/CaCl_2$, and H_2O/Na_2S pairs. The analysis is based on a heat pump system which is designed to provide about 106 kWh/day (7.11 kW heating output) for a detached house of $100m^2$.

Calculations are based on the ideal cycle shown in Figure 2 where compression is assumed to be an isentropic and the terminal temperature difference is assumed to be 10^oC for all heat transfer processes. Subcooling and superheating of the refrigerant were assumed to be negligible. The cycle was analysed for various temperature differences between the condenser and evaporator. Table 2 shows the design values for the domestic-sized heat pump for $CH_3OH/CaCl_2$, $NH_3/CaCl_2$, and H_2O/Na_2S systems. It is clear from this table that the H_2O/Na_2S system has the highest energy-storage density and coefficient of performance. The high energy storage density minimises the required volumes of the generator/absorber and the liquid accumulator used in the heat pump system. For example the volumes of the generator/absorber and liquid accumulator would be 0.092 m^3 and 0.066 m^3, respectively for the H_2O/Na_2S system compared with 0.253 m^3 and 0.159 m^3, respectively for the $CH_3OH/CaCl_2$ system. Table 2 also shows that a small amount of heat input Q_e, is required during the regeneration phase to achieve thermal balance. If both superheating and subcooling of refrigerant are taken into account, as is the case for a practical cycle, the amount of heat required during the regeneration phase would be extremely small. In addition, the coefficient of performance for the system using H_2O/Na_2S could reach as high as 3.57.

Figure 6 shows the variation of $C.O.P._H$ with temperature lift for the H_2O/Na_2S system using refrigerants R134a or R12 in the vapour-compression circuit. For R134a, the $C.O.P._H$ was found to vary between 3.15 and 2.6 depending on the temperature lift between the generator/absorber and liquid accumulator. For R12 the $C.O.P._H$ was found to be higher than that of R134a by about 5 to 11% for temperature lifts of 52.4 and 55.3^oC, respectively. These results indicate that a drop in the coefficient of performance would result if R134a refrigerant were employed.

CONCLUSIONS

A chemical storage/vapour compression heat pump has been analysed thermodynamically using different refrigerant/absorbent pairs. The work indicated that the H_2O/Na_2S system is the most promising for use in our heat pump as it provides a high energy storage density and a temperature lift suitable for domestic applications. The use of R134a refrigerant resulted in a decrease in the coefficient of performance of 5-11% compared with R12 refrigerant. The use of combined heat from ambient air and solar radiation in the sunspace of a house was found to be effective heat source for the storage heat pump.

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NOMENCLATURE

$C.O.P._H$	Coefficient of performance (dimensionless)
h_x	Enthalpy per unit mass at state x (kJ/kg)
h_g	Enthalpy of saturated vapour (kJ/kg)
h_{fg}	Latent heat of evaporation (kJ/kg)
h_v	Enthalpy of superheated vapour (kJ/kg)
K	Heat of absorption (kJ/kg)
m_g	Rate of desorption of refrigerant in the chemical heat pump (kg/s)
P	Compressor power input (kW)
p_1	Pressure in evaporator (bar)
p_2	Pressure in condenser (bar)
Q_e	Heat input at the evaporator during the regeneration phase (kW)
Q_{ep}	Heat input at the evaporator during the heat pumping phase (kW)
ΔT	Temperature lift [$\Delta T = T_6 - T_5$] (^oC)
T_5	Temperature of liquid accumulator (^oC)
T_6	Temperature of generator/absorber (^oC)

Table 2. Design values for the domestic-sized heat pump.

	System		
	$CaCl_2 \cdot 2CH_3OH$	$CaCl_2 \cdot 4\text{-}8NH_3$	$Na_2S \cdot 5H_2O$
Regeneration Phase			
Evaporator, bar (°C)	2.432 (-5)	1.064 (-25)	3.492 (5)
Condenser, bar (°C)	26.157 (79.9)	15.572 (57)	25.990 (79.6)
Accumulator, bar (°C)	0.055 (5)	2.360 (-15)	0.017 (15)
Absorbent, bar (°C)	0.055 (69.9)	2.360 (47)	0.017 (69.9)
Temperature lift, (°C)	64.9	62	54.6
h_1, (kJ/kg)	291.772	280.115	297.598
h_2, (kJ/kg)	339.840	334.434	337.919
h_3, (kJ/kg)	220.960	181.495	220.389
h_4, (kJ/kg)	220.960	181.495	220.389
$h_{5(l)}$, (kJ/kg)	11.53	112.50	62.9
$h_{6(v)}$, (kJ/kg)	1299.60	1570.38	2632.0
K, (kJ/kg absorbent)	1622	2354	3120
m, (kg/s)	0.0632	0.0495	0.0625
m_g, (kg/s)	0.0046	0.0032	0.0024
Q_e, (kW)	-1.491	0.189	-1.223
P, (kW)	3.039	2.690	2.519
Heat Pumping Phase			
Accumulator, bar (°C)	0.023 (-7.5)	3.550 (-5)	0.007 (2.3)
Absorbent, bar (°C)	0.023 (55)	3.550 (55)	0.007 (55)
Temperature lift, (°C)	62.5	60	52.7
$h_{5(v)}$, (kJ/kg)	1191.0	1438.80	2504.9
$h_{6(v)}$, (kJ/kg)	1278.4	1582.80	2604.0
h_{fg}, (kJ/kg)	1209.0	1280.55	2495.2
Q_g, (kW)	7.11	7.11	7.11
Q_{ep}, (kW)	5.601	4.120	5.873
C.O.P.	2.340	2.643	2.822
Mass of absorbent, (kg)	202.21	66.09	51.39
%age filling of absorber	80	80	80
Volume of gen / abs, (m^3)	0.253	0.155	0.092
Mass of refrigerant, (kg)	116.76	81.14	59.31
%age filling of accumulator	90	90	90
Volume of accumulator, (m^3)	0.159	0.150	0.066
Energy stored, (kWh)	106.65	106.65	106.65
Energy storage density,(kWh/m^3)	258.65	349.67	673.53

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FIGURES

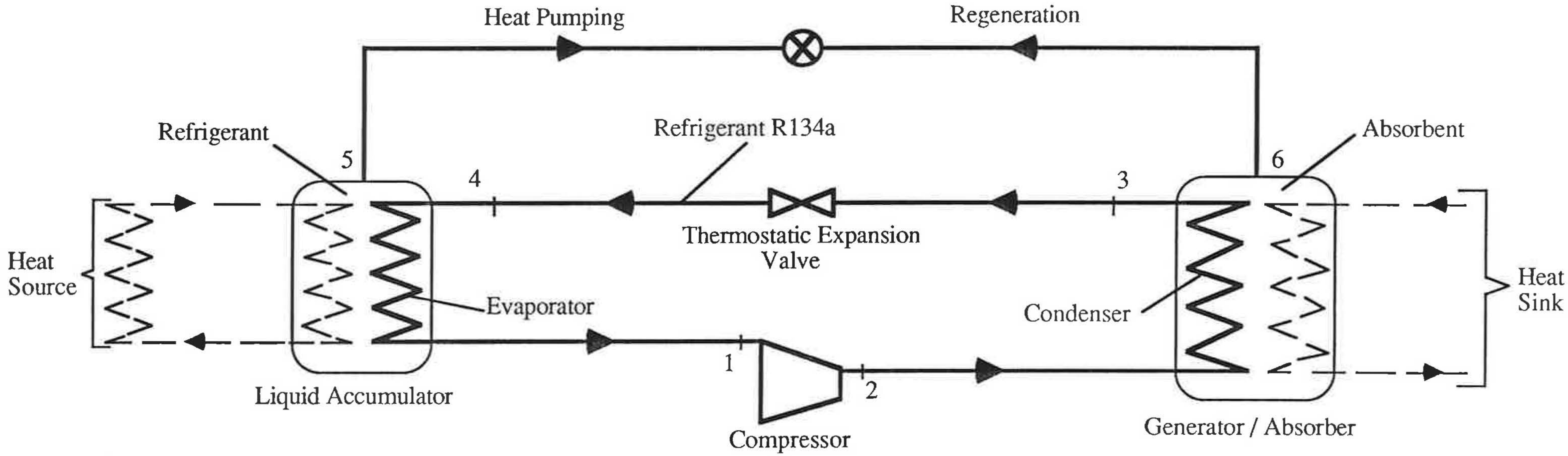

Heat Pumping
Regeneration
Refrigerant
Refrigerant R134a
Absorbent
5
4
3
6
Heat
Source
Thermostatic Expansion
Valve
Heat
Sink
Evaporator
Condenser
1
2
Liquid Accumulator
Compressor
Generator / Absorber

Fig. 1

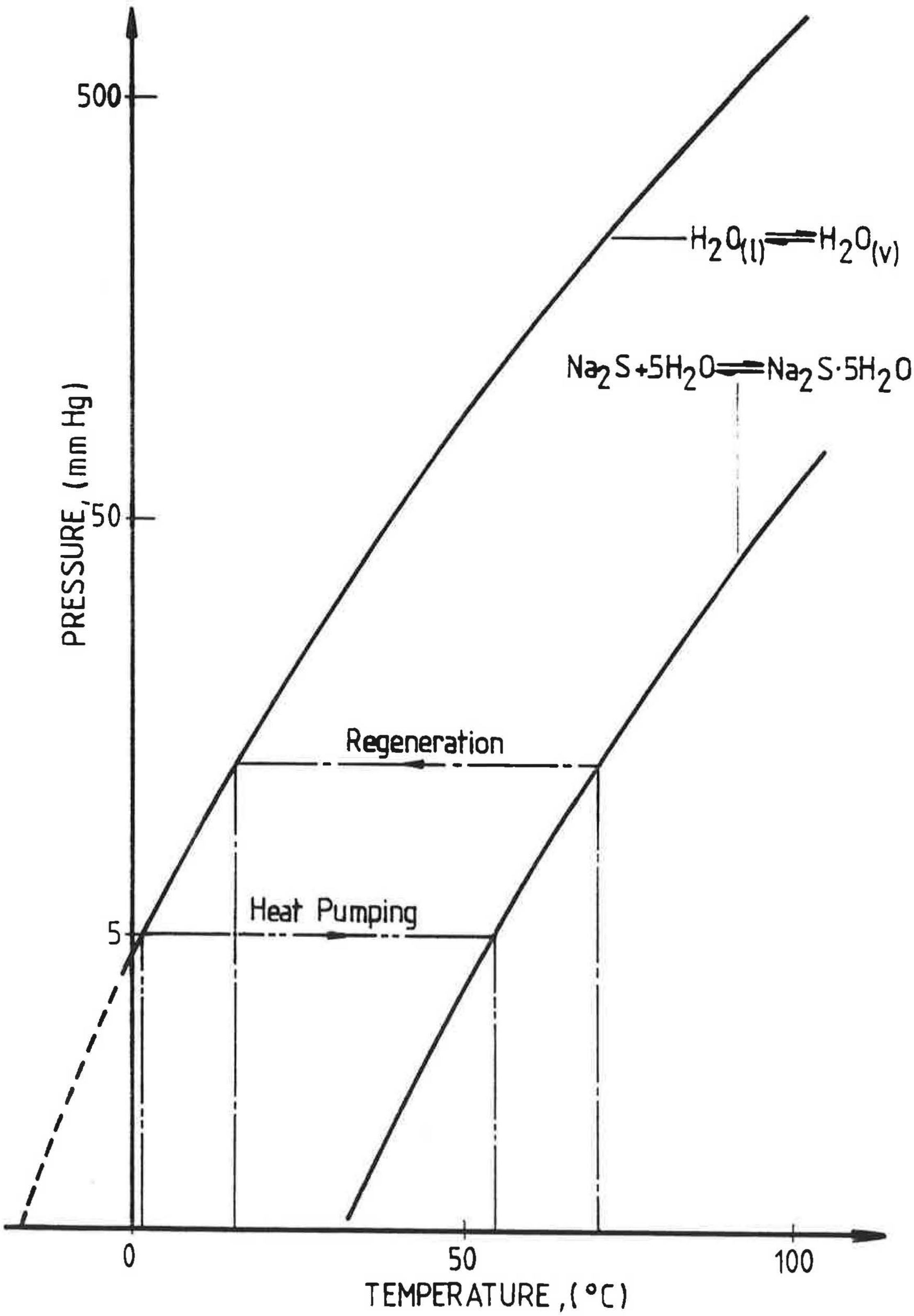
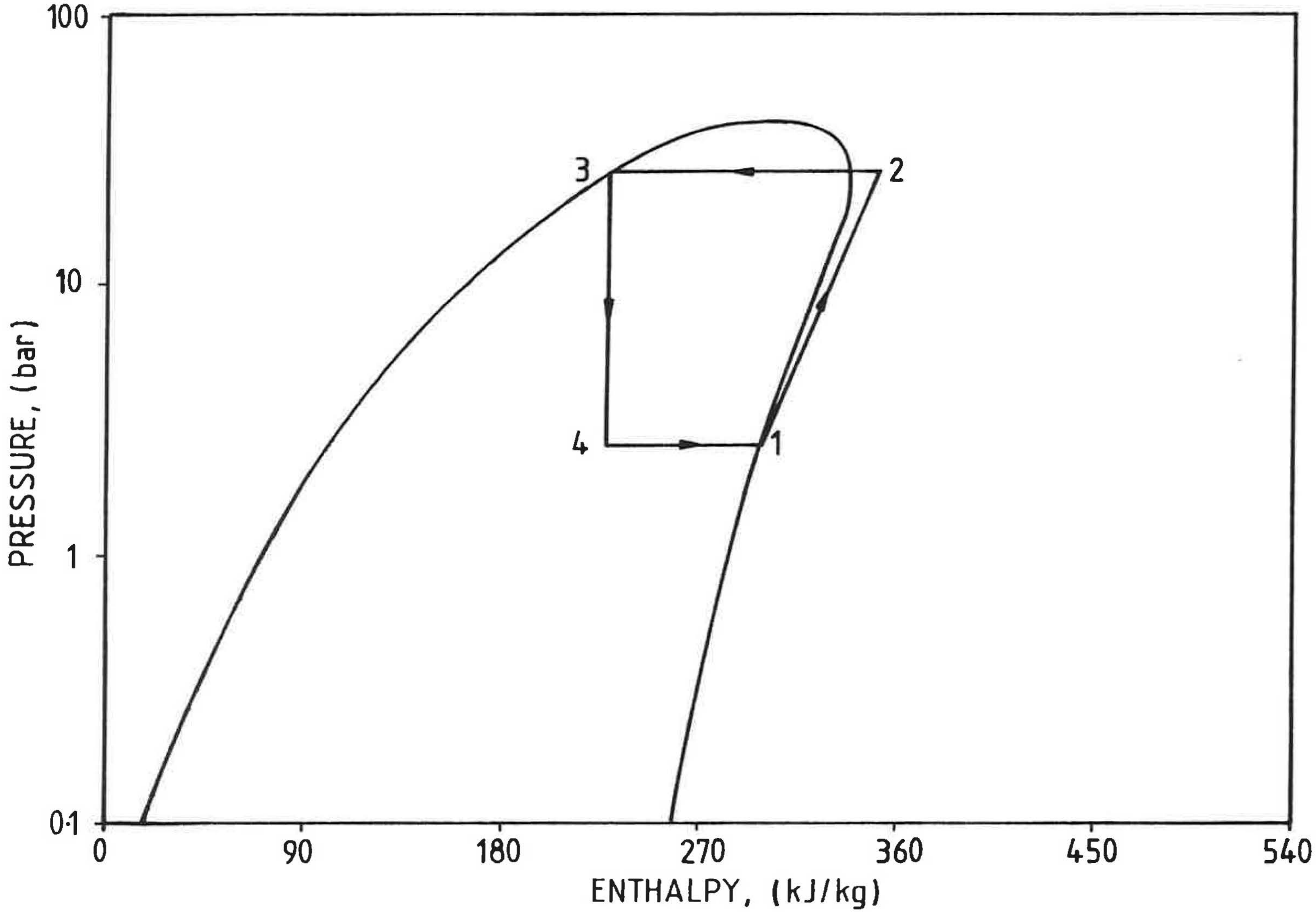

500
50
5
PRESSURE, (mm Hg)
$H_2O_{(l)} \rightleftharpoons H_2O_{(v)}$
$Na_2S + 5H_2O \rightleftharpoons Na_2S \cdot 5H_2O$
Regeneration
Heat Pumping
0
50
100
TEMPERATURE ,(°C)

Fig.2


100
10
1
0·1
PRESSURE, (bar)
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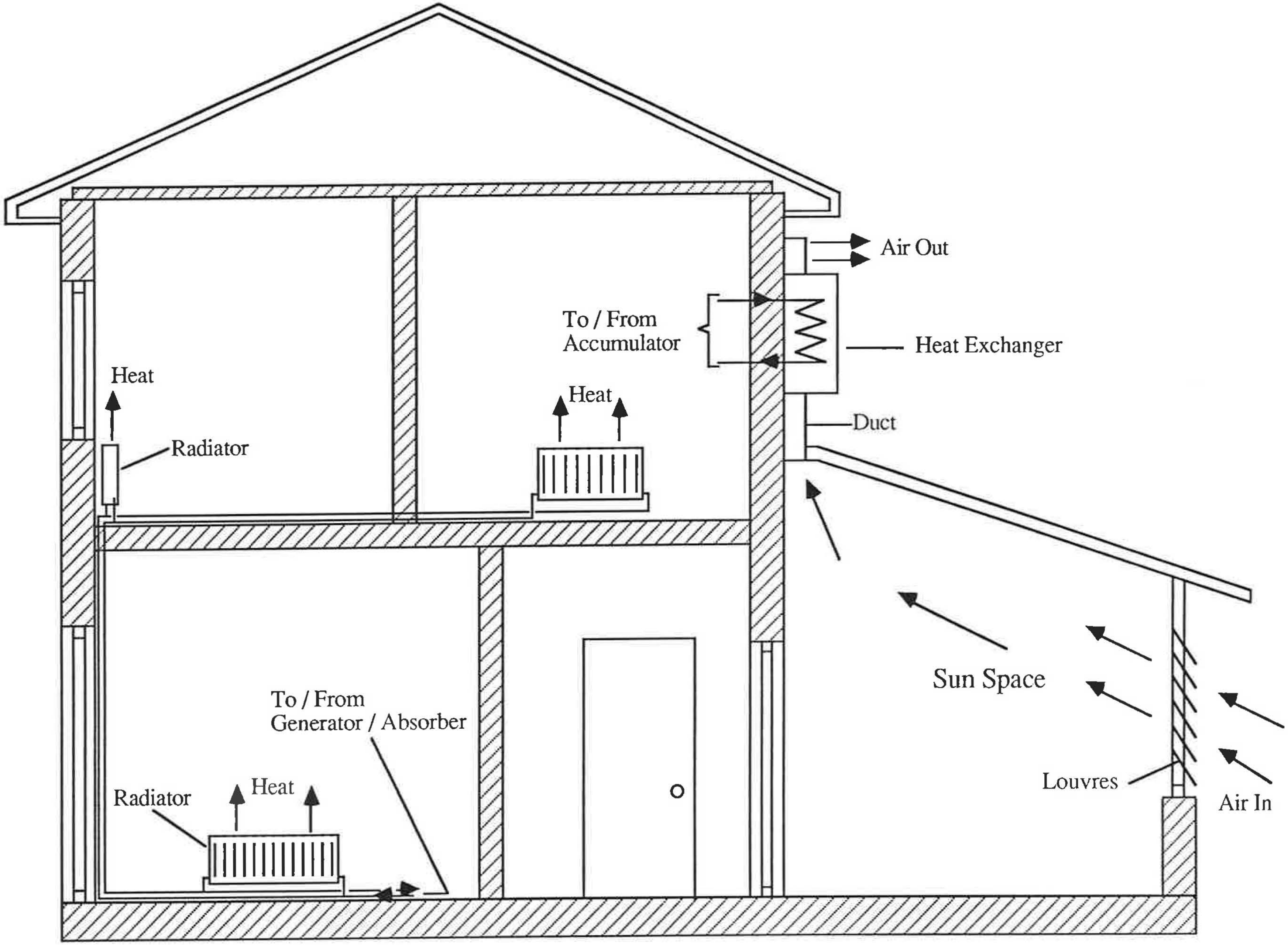
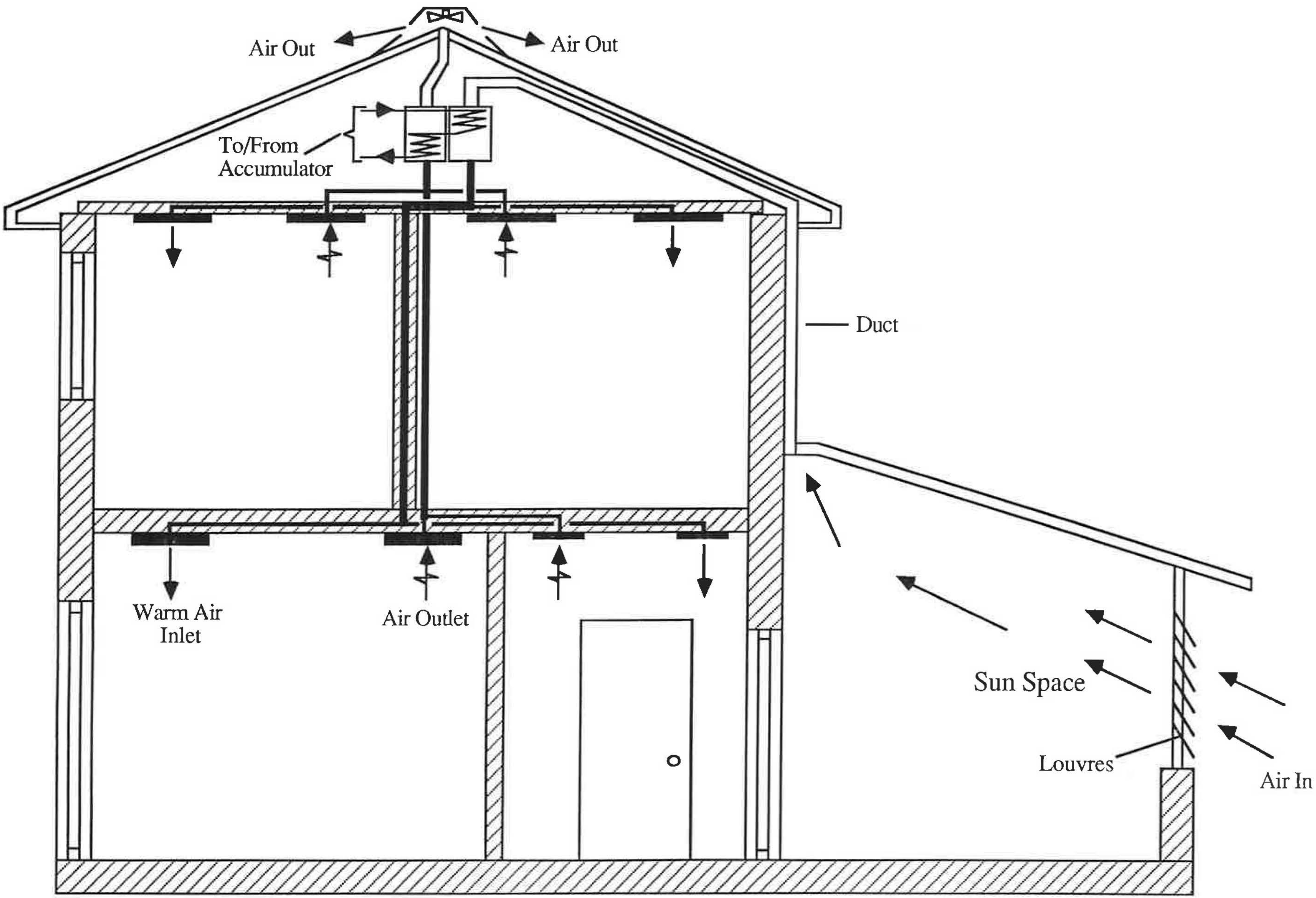
90
180
270
360
450
540
ENTHALPY, (kJ/kg)
1
2
3
4

Fig.3


Air Out
To / From
Accumulator
Heat Exchanger
Heat
Heat
Radiator
Duct
Sun Space
To / From
Generator / Absorber
Heat
Radiator
Louvres
Air In

18

Fig. 4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Air Out
To/From
Accumulator
Duct
Warm Air
Inlet
Air Outlet
Sun Space
Louvres
Air In

Fig. 5

20

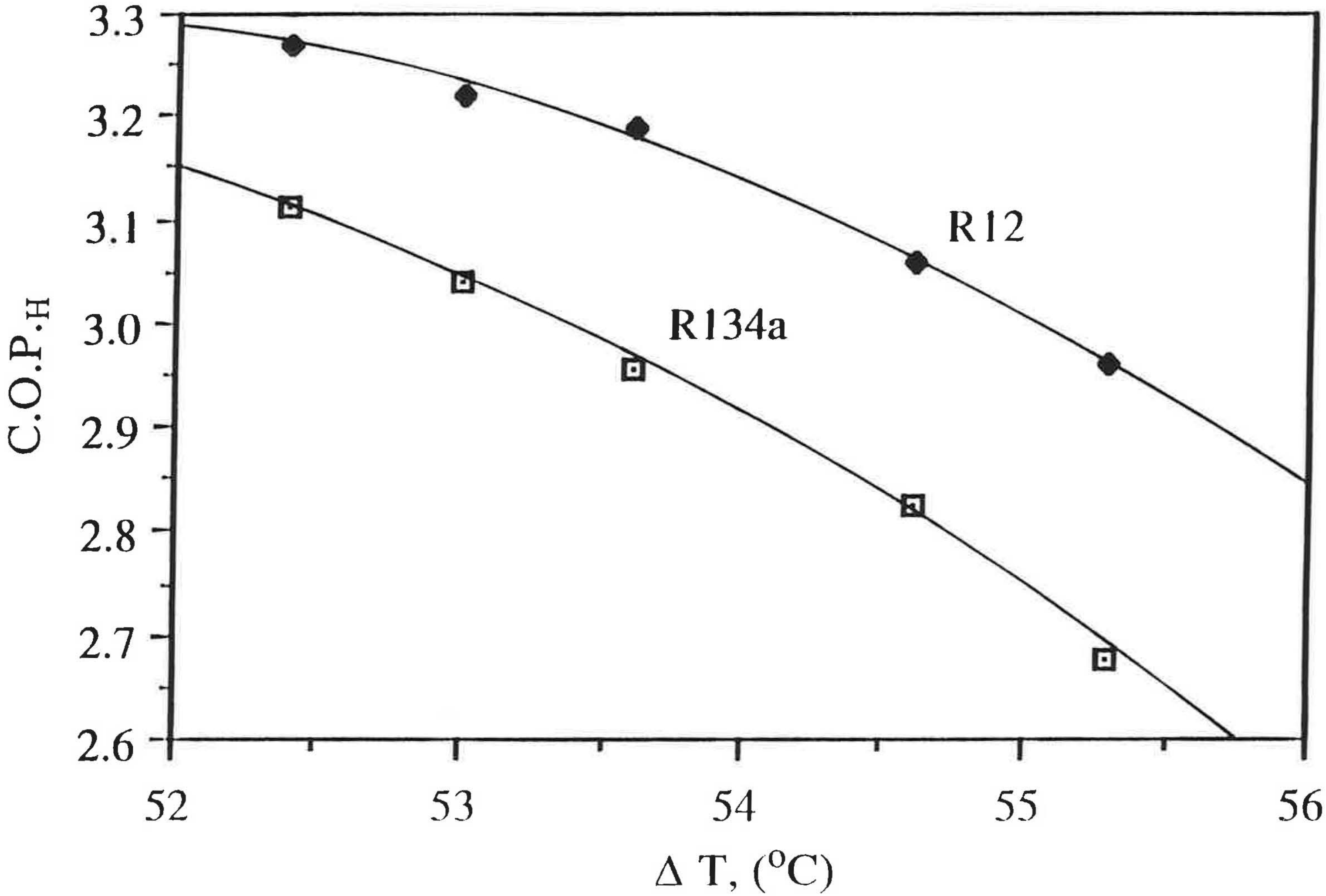

Fig. 6