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IEA District Heating and Cooling

Programme of Research, Development and Demonstration on District Heating and Cooling, including the integration of CHP

Absorption Refrigeration with Thermal (Ice) Storage

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Preface

Introduction
The International Energy Agency (IEA) was established in 1974 in order to strengthen the co-operation between member countries. As an element of the International Energy Programme, the participating countries undertake co-operative actions in energy research, development and demonstration.

District Heating offers excellent opportunities for achieving the twin goals of saving energy and reducing environmental pollution. Its is an extremely flexible technology which can make use of any fuel including the utilisation of waste energy, renewables and, most significantly, the application of combined heat and power (CHP). It is by means of these integrated solutions that very substantial progress towards environmental targets, such as those emerging from the Kyoto commitment, can be made.
For more information about this Implementing Agreement please check our Internet site www.iea-dhc.org/

Annex VI

Annex VI started in May 1999.
The countries that participated were:
Canada, Denmark, Finland, Germany, Korea, The Netherlands, Norway, Sweden, United Kingdom, United States of America.

The following projects were carried out in Annex VI:

Title project	**ISBN**	**Registration number**
Simple Models for Operational Optimisation	90 5748 021 2	S1
Optimisation of a DH System by Maximising Building System Temperatures Differences	90 5748 022 0	S2
District Heating Network Operation	90 5748 023 9	S3
Pipe Laying in Combination with Horizontal Drilling Methods	90 5748 024 7	S4
Optimisation of Cool Thermal Storage and Distribution	90 5748 025 5	S5
District Heating and Cooling Building Handbook	90 5748 026 3	S6
Optimised District Heating Systems Using Remote Heat Meter Communication and Control	90 5748 027 1	S7
Absorption Refrigeration with Thermal (ice) Storage	90 5748 028 X	S8
Promotion and Recognition of DHC/CHP benefits in Greenhouse Gas Policy and Trading Programs	90-5748-029-8	S9

Benefits of membership

Membership of this implementing agreement fosters sharing of knowledge and current best practice from many countries including those where:

- DHC is already a mature industry
- DHC is well established but refurbishment is a key issue
- DHC is not well established.

Membership proves invaluable in enhancing the quality of support given under national programmes. The final materials from the research are tangible examples, but other benefits include the cross-fertilisation of ideas which has resulted not only in shared knowledge but also opportunities for further collaboration.

Participant countries benefit through the active participation in the programme of their own consultants and research organizations. Each of the projects is supported by a team of Experts, one from each participant country. The sharing of knowledge is a two-way process, and there are known examples of the expert him/herself learning about new techniques and applying them in their own organization.

Information

General information about the IEA Programme District Heating and Cooling, including the integration of CHP can be obtained from:

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Summary

This report describes and evaluates the results of an experimental study of an innovative absorption cycle refrigeration and thermal (ice) storage system. The report describes the manufacture and testing of a laboratory scale lithium-bromide/water single–effect absorption cycle refrigerator machine and a novel thermal (ice) storage unit.

Experimental data confirming the technical feasibility of the novel system is provided and recommendations for future research and a route to exploitation are also discussed. The investigation included experiments to determine:

(i) Optimum concentration of water-gel used in the construction of the storage elements.
(ii) Optimum layout of the elements within the TIS.
(iii) Entrainment and pressure lift performance of the jet-pump.
(iv) Optimum location of the primary nozzle within the mixing chamber.
(v) Critical back-pressure of the jet-pump.
(vi) Freezing rate of the TIS.
(vii) Coefficient of performance of the cycle.

The results demonstrated that the absorption refrigerator operated satisfactorily, however, there are some minor modifications needed to ensure a more stable operation. Some improvements to the instrumentation are also required.

Before going forward to develop the concept further it will be necessary to define in some detail the capital and life-cycle costs of such a plant compared with a standard absorption machine. This study is required in order to quantify the potential cost and energy savings resulting from the potential downsizing the absorption cycle machine and the cost of the TIS system. Clearly, the anticipated decrease in capital costs would boost the economic feasibility of future applications.

The study has demonstrated the technical feasibility of the novel TIS – absorption cycle concept which is thought to offer the potential to promote the wider use of Combined Cooling Heating and Power (CCHP) schemes. However, to progress further it will be necessary to involve industrial partners.

1. Introduction

The International Energy Agency under their Annex VI funded the research programme described in this report. The report describes and evaluates the results of an experimental study of an innovative absorption cycle refrigeration and thermal (ice) storage system. It describes the manufacture and testing of a laboratory scale lithium-bromide/water single–effect absorption cycle refrigerator machine and a novel thermal (ice) storage unit and goes on to describe and evaluate experiments carried out to confirm the technical feasibility of the novel system. Some recommendations for future research and a route to exploitation are also discussed.

1.1 Description of the invention

Figure 1 shows a schematic layout of the innovative thermal-ice-storage system included with a single-effect absorption cycle refrigerator. The operation of the conventional single-effect vapour absorption refrigerator is well documented in standard text books[1] and therefore, will not be repeated here. The introduction of the thermal ice store is novel and therefore requires explanation. The thermal-ice-store in Figure 1 consists of one or more, vertical cylindrical filled with ice-storage elements. Each storage element contains a polymer paper sachet filled with a quantity of hygroscopic gel. This gel expands on contact with water so that when the elements are charged they contain over 95% water. In practice the physical size of the TIS and the number of elements is determined by the required storage capacity of the system.

1.2 Operation

Charging cycle

In practice the charging of the ice store could be carried out at any other time of the day when waste heat is available and building cooling is not required. However, by charging the store at night-time the operator can take advantage of the lower ambient temperatures and therefore, the potential for greater coefficient of performance values. At the beginning of the charging cycle refrigerant water in the evaporator is diverted to the storage vessel and sprayed over the bed formed by the storage elements. This causes the absorbent gel to become saturated with water. After a short time the storage elements become fully charged with water and at this point the storage vessel is drained of excess water which is returned to the evaporator. The ice-storage elements are then frozen over a period of time by evaporation. The mass of water evaporated from each element during the freezing process is approximately 13%. Therefore, this quantity of water requires to be re-absorbed by the storage elements prior to recharging.

Once the ice-store has been prepared the solution pump, shown in Figure 1, which circulates the $LiBr/H_2O$ solution between the absorber and generator, is activated. Heat is then supplied to the generator and cooling water is fed to the condenser and absorber. At this time the building's air conditioning system chilled water circulation pump is switched off.

For a single-effect machine, once the generator temperature has reached a approximately 100^oC with a concentration of approximately 61% the valve in pipe-line 8, Figure 1, is

[1] A excellent description of absorption cycle refrigerators is provided by Stoecker W.F. and Jones W.J., Refrigeration and Air Conditioning, 2nd Edn. Chap. 17, pp 328 - 347, McGraw - Hill Book Company, 1985, ISBN 0 07 061619 1

opened causing steam at about 16kPa, (2.32 psia), to enter the jet-pump through its primary nozzle. For design purposes it is assumed that the absorber solution will be at 30^{o}C with a concentration of 54%, giving a dew-point temperature in the evaporator of 5^{o}C. At these conditions the jet-pump will draw water vapour from the storage vessel compress it so that its pressure equals that in the absorber and evaporator. To maintain the flow of low pressure steam to the suction side of the jet-pump, water must be evaporated continuously from the saturated absorbent gel within the ice storage elements. This causes the water in elements to freeze. Based on a semi-empirical model it was estimated that under ideal conditions the entrainment ratio of the jet-pump will be close to unity. This is made possible because of the low-pressure lift required between the ice-store and the absorber/evaporator. For the purpose of design, during the store's charging cycle the saturation temperatures within the ice-store and absorber/evaporator are assumed to be of -5^{o}C and +5^{o}C respectively. Detailed calculations have shown that the coefficient of performance of the refrigerator during the charging cycle will be approximately 0.38. In arriving at this result it was assumed that the generator temperature is limited to that required to produce the same water vapour absorption rate at the absorber on a design-day with a 30^{o}C wet-bulb ambient temperature. It should also be noted that the water vapour produced at the generator is equal to the vapour flow from the ice-store at (9) in Figure 1, plus the jet-pump primary flow in pipeline (8). Excess vapour produced by the generator is directed to the condenser via pipeline (4), in Figure 1, where it is condensed and stored. An additional liquid storage volume in this case will equal approximately 13% of original water volume in the ice-store. This is estimated to be approximately 0.4m^3/GJ of thermal (ice) storage, (TIS).

Discharging cycle

The discharge process might be carried out in series or in parallel with the absorption refrigerator, depending upon the demand profile. Parallel operation is technically more interesting. Once the store is charged and the primary flow to the jet-pump is turned off, the pressure within the ice-store will equal the saturation pressure of water-ice at 0^{o}C, (611 Pa). The pressure within the absorber will be controlled by ambient temperature and typically will equal the saturation pressure of water at +5^{o}C, (872 Pa). Therefore, without a flow restriction in line (10), Figure 1, water vapour generated within the evaporator will clearly have a preference to flow into the ice-store vessel before the absorber. Therefore, to balance the cooling effects of the ice-store and refrigerator it may be necessary to fit a damper in the jet-pump suction line. The operation of this damper will regulate to effect of the store on the system's cooling rate. In this way it will unnecessary to have a second heat exchanger, (as required with series operation) because all the cooling of the chilled water will be carried out via coil in the evaporator. Furthermore, it is believed that replenishing the ice-store elements with water as previously described may be unnecessary as this will take place automatically during the discharge process, due to water condensation on the surface of the spheres during discharge. Also, change over from charging to discharging phases will simply require the steam valve in line (8) to the opened. The disadvantage of parallel operation is that the COP of the absorption refrigerator during the discharging phase was theoretically calculated to be not more than 0.65 on a design-day whereas series operation might yield significantly greater values during the discharge process.

2. Rig design and manufacture

Figures 2 and 3 show photographs of the laboratory scale single-effect absorption cycle refrigerator and thermal (ice) store test rig built for this project. The main components shown in these figures include the vapour generator, condenser, evaporator and absorber assembly, jet-pump assembly and the thermal (ice) store. Table list outline design-point specifications for each of the principal components, except the jet-pump, the detail of which are included later. Figures 4 and 5 provide further views of the rig.

Table 1: Specification of system and components

Component	**Design-point Operating conditions**	**Comments**
Overall System	Q_E = 5 kW at design-point	
Condenser	T_{COND}= 35°C (sat),	on-coil condensation, stainless steel vessel,/copper coil, water cooled
Concentrator	T_C = 85°C	nucleate boiling type, glass cylinder/finned copper heater tubing, steam heated
Evaporator	T_E = 5°C (sat)	water spray - flash evaporation, stainless steel cylinder vessel, electrically heated
10 kW Steam Generator	T_G = 102.5°C(sat), 61% LiBr concentration, 16 kPa	stainless steel, maximum test pressure 25 bar
Absorber	T_A = 30°C, 2.54°C (dew-point) 6kW cooling (max)	falling film type, glass vessel/copper coil water cooled

2.1 Vapour Generator

The vapour generator powers the absorption cycle. A schematic view of the generator is given in Figure 6. The generator is powered by four 2.5 kW electrical heater. The generator vessel was manufactured from stainless steel pipe and has a maximum working pressure of 15 bar. The design of the generator included a high-pressure relief system and thermostatic cut-outs. A large capacity liquid-vapour (knockout) separator was also included in the design to ensure that dry-steam only entered the primary nozzle of the jet-pump and to prevent solution from being carried over. This is important to avoid wear and corrosion and also to prevent the primary nozzle spitting solution or liquid water, both of which tend to reduce the performance of the jet-pump.

2.2 Evaporator and absorber assemblies

A 6kW absorber, described in Figure 7, was manufactured from a cylindrical glass vessel, 101 cm tall by 24 cm diameter, fitted with a copper heat exchanger coils and two solution distribution manifolds.

The evaporator, also shown in Figure 7, was similarly manufactured from a glass vessel, 51cm tall by 24cm diameter. The evaporator is heated partly by a 2kW electrical heating element positioned at the bottom of the evaporator vessel, and partly by secondary heating circuit, with a maximum capacity of 8kW provided by an immersion heater, not shown. The heater exchanger of this secondary circuit is
however shown in Figure 7. A spray nozzle was fitted at the top of the evaporator vessel through which the heated refrigerant water (at about 5 ^{o}C) was ejected. The system was designed so that under normal operation the electrical heating element would be immersed and refrigerant water would be pumped from the bottom of the vessel, through the secondary heat exchanger, to the spray nozzle through which it re-entered the evaporator as a fine spay of droplets. A fraction of liquid flashes to steam at each pass cooling the droplets as they fall to the bottom of the vessel before being re-heated by the element and the immersion heater. When the refrigerator is cooling, vapour produced in the evaporator flows through a 4 inch (101 mm) diameter plastic pipe between the top of the evaporator vessel and into the absorber.

2.3 Condenser

The construction of the condenser is described in Figure 8. It consists of a stainless steel cylindrical vessel, 80cm in height and 20cm in diameter, manufactured from 16 SWG stainless steel sheet. Bolted flanges were fitted at both ends to allow for access and the fitting of the pipe connections and pressure and temperature transducers. The heat exchanger coil was manufactured from Table X, 12 mm o/d, copper tubing wound into two open coils, one fitted within the other with mean diameters of 16 cm and 12 cm, giving a total heat transfer area of 1.46 m^2. Cooling water was supplied by the laboratories chilled water system at temperatures ranging from 10^{o}C to –10^{o}C.

2.4 Jet-pump design and manufacture

Figure 9 shows the dimensional details of the jet-pump. The jet-pump diffuser and body were manufactured in aluminium whilst the primary nozzle was machined from brass and its supporting spindle assembly was made from stainless steel. Table 2 lists the design-point specification of the jet-pump. Table 3 lists the predicted performance of the jet-pump based on the design-point data in Table 2. Table 4 lists the calculated dimensional data for the jet-pump shown in Figure 9.

Table 2: Jet-pump design-point specification

Driven by the steam from Generator:
Solution temperature 102.5ºC,
Solution concentration 61%,
Dew point temperature of vapour 52ºC,
Dew point pressure of vapour 14 kPa, (2 psia)
Discharged to Absorber:
Solution temperature 30ºC,
Solution concentration 54%
Dew point temperature 2.54ºC
Dew point pressure of vapour 0.783 kPa (0.114 psia)
Suction from the thermal ice storage:
Water temperature 0ºC

Table 3: Jet-pump performance based on conditions listed in Table 2

Power input to the jet-pump
2.6125 kW @ Primary flow rate 0.066 kg/min,
Predicted cooling rate supplied by the jet-pump
3.6711 kW @ Secondary flow rate 0.0881 kg/min

Table 4: Part Sizes of the jet-pump

Primary nozzle
Throat diameter 7.57mm
Exit diameter 16.14mm
Constant pressure mixing chamber
Inlet diameter 68mm
Outlet diameter 33mm
Length 131mm
Constant area mixing chamber
Diameter 33mm
Area ratio 4.35
Diffuser
Inlet diameter 33mm
Exit diameter 73mm
Length 466.9mm

2.5 TIS and Ice storage elements

Figure 10 describes the construction of the TIS vessel. The vessel itself was made from a proprietary glass cylinder, (155cm in height by 35cm o/d and 33cm i/d), fitted top and bottom with bolted flanges and seals. This type of vessel was used because it allows the operator to see what is happening to the ice storage elements during experimentation and also it removes the difficulty of fabricating vessels that will be leak proof at low vacuum pressures. The storage vessel was insulated with 2cm thick refrigeration standard foam (insul) sheet.

Each ice storage element consisted of a 154mm length of stainless steel hollow cylinder, 35mm diameter, perforated along its length with an average of 190 holes, 8mm diameter. The total estimated area for vapour flow, per element, is 9.55·10-3m^2 which represents approximately 51% of the total external surface area of an element (0.0188 m^2). The TIS vessel held 384 ice storage elements fitted in 8 rows. Each row has 48 basket elements, of the type shown in Figures 11 and 12, charged with 124 cc of water absorbent gel with a 98.37% w/w concentration of water. 6 basket elements per row left uncharged to assist the free flow of water vapour between the bottom of the vessel and the jet-pump suction manifold. Figure 13 shows photographs of the upper and lowest layers of elements in the vessel. Tests described later had shown this was necessary to ensure that the charged elements froze at approximately the same rate throughout the vessel.

Each ice storage element was made from perforated stainless steel sheet (0.5 mm thickness) rolled to form an open cylinder (35mm o/d and 150 mm in length) fitted with end caps. An outline drawing of a TIS basket is shown in Figure 3 and a photograph of a sample piece is show in Figure 4. Polymer tubes or bags, which fit inside the TIS baskets to retain the ice-gel, were made from Type 483402/25 gram polymer paper as supplied by J.R. Crompton Ltd of Manchester. This paper is normally used in the manufacture teabags. Recent tests have shown this to be a suitable paper for its present roll in terms of porosity and wet-strength. Tests were also carried out to determine the most suitable gel concentration for use in the elements.

2.6 Other constructional details

It had been found previously that corrosion is particularly prevalent in experimental LiBr/H_2O absorption systems due to there being continually opened to atmosphere between test runs. Therefore, plastic pipe-work and fittings were used as much as possible to connect the separate process vessels. This reduced problems of corrosion in pipe-work normally associated with plain copper tubing and kept costs down. Where temperature or pressure prevented the use of plastic pipe-work stainless steel or heat resistant glass were used in stead. Stainless steel valves were used throughout.

Rotor-dynamic magnetic-drive pumps were used to circulate the refrigerant water within the evaporator and the salt solution within the absorber. Although more expensive than conventional units, the use of magnetic drive couplings overcame air leakage problems found when mechanically sealed pumps had been used in the past.

3. Experimental work

3.1 TIS-gel concentration trials

Aims
The aims of these experiments were to determine (a) a suitable concentration for the TIS water-gel and (b) to confirm the suitability of the polymer paper from which the TIS-gel bags were to be made. The important considerations included the ability of the water-gel to retain liquid water over a period of time and to ensure that the polymer paper had sufficient wet-strength to resist bursting during the vacuum freezing process.

Method
Sample TIS-gel bags were made from Types No 473702-21g/metre and No 483402-25g/metre polymer paper. The sample bags were made with 33mm diameter and 270 mm in length from both paper types. The bags were heat sealed at one end and along their length. The bags were then filled with a mixture of pre-prepared water-gel mixture to produce test specimens over a range of concentrations listed in Table 1.   The consistency of the gel in each case was assessed. The results of this assessment are also listed in Table 1. The gel selected for these experiments was Aqualic CA type H3 manufactured by Nippon Shokubai Co. Ltd. of Tokyo.

Table 5: Mixtures of water/gel tested

No	Mix Ratio by weight	Consistency
1)	50:1	Solid
2)	60:1	Heavy grease
3)	70:1	light grease
4)	75:1	Treacle
5)	80:1	Light oil

Volume of bag = 230cc
Allowance for freezing 15%.
Fill volume = 196 cc mixture

Each bag was filled with 200 cc of gel mixture using a suitable syringe. The bags were then heat sealed at the open end and suspended from one-end and left overnight to allow excess water to drain from them. The quantity of excess water was collected and measured. This measurement was used to determine the water retention ability of the gel in each case. The bags were then frozen using a domestic freezer and defrosted in the suspended condition several times.

Results
The polymers tested were found in all water-gel concentration cases to have sufficient wet-strength to retain the gel over repeated freezing and defrosting cycles. The water retention tests showed that if the concentration of water in the gel solution was greater than 98.37% w/w (or 1 gram of gel holding 60 grams of water) then an excessive amount of water was

lost when left standing. Typically it was found that at 98.59% concentration, a 200 cc gel sample would loose 30cc of water in a 12 hour period whereas a 98.37% concentration would loose only 5cc. This was thought to be an acceptable amount.

Conclusions

From test results it was concluded that the gel-bags should be made using polymer paper type 483402-25g/metre and the concentration of the gel at filling should be 98.27% w/w of water. This means for every gram of gel would contain 60 gram of water

3.2 Small-scale TIS experiments

Before the construction of the full scale TIS was completed it was felt necessary to test charge a small-scale version of the store using an existing jet-pump refrigerator system in order to determine the thermal and mechanical behaviour of the TIS elements.

These preliminary experiments, the construction of the rig and TIS, are described and evaluated in the proposed paper included in Appendix A to this report. The details of the experiment are as follows.

Experimental method and results

A total number of 107 elements were manufactured as previously described for the purposes of these experiments. The weight of each element was recorded before and after each test run. Some of the most important data from the batch of elements are summarised in Table 6.

Table 6: Data on TIS elements used in preliminary tests

Total number of elements	Total mass of the elements (kg)	Total mass of water contained in the elements (kg)	Total mass of gel powder used (g)	Average mass of each element (g)
107	17.341	14.26	237	162.06
Element's mass standard deviation (g)				
5.39				

The elements were distributed in two rows inside the jet-pump rig evaporator, shown in Figure 2 of Appendix A. To measure the temperature variation, thermocouples were inserted into three elements (One at the bottom row, and two diametrically opposite at the top row). The temperature of the water vapour inside the evaporator was measured by a fourth thermocouple. A pressure transducer was used to measure the water vapour pressure variation inside the TIS vessel. The condenser pressure and temperature were also recorded.

Five experimental runs were performed. The first was stopped because, air was released from the gel as the vapour pressure fell, the rate of decrease of the pressure inside of the evaporator was not sufficient to permit the temperature of the gel to reach freezing point. It was concluded that bubbles of air were trapped within the gel and this was being released as the pressure fell. Because of the relative buoyancy between air and water vapour the former could not be removed by the vacuum pump with the suction pipe in its original location. To prevent a reoccurrence of this problem the gel was held under vacuum pressure for 48 hours to allow the trapped air to be released.

During the second experimental test run comparatively high vacuum pressures were achieved. The results of these experiments are listed in Table 7. Two experiments were halted because the condenser pressure did not stabilise. The condenser pressure had a great influence on the process and it was found that any slight increase meant that the vacuum pressure inside the TIS did not drop to a value needed to freeze the elements. This was primarily due to the operation of the test rig and not due to any fundamental problem with the TIS. However, it did demonstrate the importance of maintaining the absorber/evaporator pressure below the design-point value if the jet-pump is to operate effectively. A further problem encountered during these preliminary tests was that resulting from excess water in the system. During the TIS charging process approximately 13% by volume of the water in the gel evaporates. This water needs to be stored. If it is allowed to accumulate in the condenser it can cause the heat exchange performance of this to fall. This may cause the back-pressure produced by the absorber/evaporator (Figure 1) to rise and result in the jet-pump failing to operate if it exceeds the critical (design-point) value.

The experiments were successful and the gel was found to have frozen.

Table 7: Some experimental data from the preliminary TIS trials.

Experiment run on 13/02/01 Minimum Evaporator pressure	Minimum Evaporator Temperature	Minimum Condenser Pressure	Minimum gel temperature	Period of time at freezing temperature
- 1.0238 mbar (Gauge)	3 ^{0}C	-.982 mbar (Gauge)	1 ^{0}C	0 min

Experiment run on 23/02/01

Minimum Evaporator pressure	Minimum Evaporator Temperature	Minimum Condenser Pressure	Minimum gel temperature	Period of time at freezing temperature
- .9959 mbar (Gauge)	- ^{0}C	-.983 mbar (Gauge)	0 ^{0}C	20 min

Experiment run on 26/02/01

Minimum Evaporator pressure	Minimum Evaporator Temperature	Minimum Condenser Pressure	Minimum gel temperature	Period of time at freezing temperature
- .9847 mbar (Gauge)	- ^{0}C	-.983 mbar (Gauge)	0 ^{0}C	45 min

Conclusions

- This set of experiments demonstrated the importance of the following:
- Removal all trapped air from within the gel by holding the storage elements at vacuum pressure for at least 48 hours.
- Ensuring that the TIS system has sufficient water storage volume during the charging cycle.
- The absorber/evaporator pressure are maintained at a value below the critical back-pressure of the jet-pump during normal TIS charging.

3.3 Layout of elements within the TIS vessel

Aim

To investigate the effect of the storage element layout within the ice store on freezing and melting performance.

Description of the apparatus and experimental method

In order to determine the performance of the experimental TIS device, a batch of elements was tested using the steam jet-pump rig described in Appendix A.

The experimental TIS device consisted of a glass vessel with 8 rows of elements, giving a total of 352 units placed in concentric circles. The elements have a total heat and mass transfer area of 6.62 m^2 representing 72 times the area of free water surface inside the storage device. The elements have a plain stainless steel lid at both ends. The cylinders are filled with sachets made from a permeable material that easily allows the diffusion of water. The sachets in their turn are filled with gel made of water mixed with a super absorbent polymer. Each element contained on average 125g of water and 2.2g of polymer. When fully charged the batch of 352 elements could retain up to 43 kg of water.

Results and Conclusions

The most important findings resultant from these tests were:

(1) Performance of the TIS is directly related with the elements layout inside the storage device/evaporator and the released vapour flow dynamics;

(2) This part of the investigation showed that an open arrangement of the storage elements was necessary to permit the flow of water vapour from elements at the bottom of the store, and so the freezing rate, to be similar throughout.

(3) Agglomerating of the polymer has a negligible influence on the physical properties of water or effected mass transport occur during the charging period of the TIS;

(4) Heat contained within the elements can be easily withdrawn and the storage medium reaches freezing temperatures provided that there are no significant constraints to the vapour flow.

3.4 Setting up the jet-pump and the optimum positioning of the primary nozzle

Aim

The aim of this set of experiments was to determine the optimum axial position of the primary nozzle within the mixing chamber of the jet-pump The optimum position is determined in terms of secondary pressure ratio and entrainment ratio. Both these parameters are defined below. It is known that the optimum NXP is sensitive to both primary and secondary pressures. Also, small variations in NXP can make large differences to both secondary pressure ratio and entrainment ratio.

Method

The following methodology was used to determine the entrainment ratio over a range of NXP values:

- All the water in the generator was drained into the condenser where the water level was recorded and the total water mass in the condenser (m_{C1}) was calculated.

- The water was then pumped back into the generator vessel until the level in the condenser had reached the predetermined reference level at which the mass of water in the condenser was calculated to be m_{C0}. Therefore, the mass of water in the generator was by,

$$\mathbf{m_{Gi} = m_{C1} - m_{C0}} \qquad (1)$$

- The NXP was then adjusted to the desired value.

- The generator heater was switched on and the generator pressure control set at 14 kPa (absolute); the design-point value.

- The rig was run with the condenser and evaporator pressures set at their design-point values.

- After a measured time, (t-seconds), of about 60 minutes, the rig was shut down and the generator heater switched off.

- The water level in the condenser was then recorded and the total water mass **,**m_{C2}**,** calculated.

- The total mass of water condensed during the experiment equalled the sum of the primary nozzle flow (m_P) and secondary flow (m_S) from the TIS. This was given by,

$$\mathbf{m_{P} + m_{S} = m_{C2} - m_{C0}} \qquad (2)$$

- The water in the generator was again drained back to the condenser and the level recorded and the mass, (m_{C3}), calculated.

- The mass of water remaining in the generator after the experiment, (mgf) was given by,

$$\mathbf{m_{Gf} = m_{C3} - m_{C2}} \qquad (3)$$

- The steam flow through the primary nozzle during the experiment was then determined from,

$$\mathbf{m_P = m_{Gi}\text{-}m_{Gf} \quad (kg)} \qquad (4)$$

- The time averaged primary flow was given by,

$$\mathbf{\dot{m}_P = \frac{m_P}{t} \quad (kgs^{-1})} \qquad (5)$$

- The secondary flow was determined by substituting Equation (4) into Equation (2):

$$\mathbf{m_S = (m_{C2} - m_{C0}) - m_P} \qquad (6)$$

- The entrainment ratio is determined from $\mathbf{R_m = m_S / m_P}$ (7)

Results and conclusions
The primary flows and secondary flows where measured over a range of NXP values ranging from –60 mm to +20 mm from the NXP zero-datum. The primary nozzle is shown at its zero-NXP in Figure 9. Figure 14 shows the measured variation in both primary and secondary flows with NXP. Figure 15 shows the measured variation in entrainment ratio with NXP with the jet-pump operating at its design inlet and discharge pressures. From these results it was clear that the greatest entrainment was provided with the nozzle exit at +10 mm. However, it was felt that this result would not give the best overall part-load performance and because of this it was decided to set the NXP = -20mm for the remainder of the trial which was thought to provide the best compromise.

3.5 Determination of the jet-pump's critical back-pressure.

Background and aims
In the present refrigerator the pressure within the absorber/evaporator assemblies produces a back-pressure against which the jet-pump must operate. This back-pressure rises with ambient air temperature. At a certain value the critical back-pressure is reached and the jet-pump will cease to operate. The critical-back pressure of a jet-pump, therefore, limits its pressure lift and entrainment performance. Above this pressure flow instability occurs, causing the performance of the pump to fall and with further increase, flow reversal occurs in the diffuser causing the primary flow to enter the

TIS vessel. The maximum back-pressure that a jet-pump can operate against depends on the total pressures of the primary and secondary flows and the geometry of the jet-pump. The particular jet-pump used in these experiments was designed to operate against a back-pressure of 0.783 kPa, which was equal to a design-point absorber dew-pont temperature of 2.54 ^{o}C. The aim of these experiments was to determine whether the jet-pump design had been successful in this respect.

Method and Results
The TIS - absorption refrigerator was operated in the charging mode with the total pressure at the inlet to the primary nozzle set at approximately 14 kPa, the secondary flow at 650 Pa

in the jet pump suction manifold and approximately 1.2 kPa in the absorber. The latter is about twice the design-point value and is equivalent to a solution temperature of 35^{o}C and a LiBr concentration of 54%. The system was operated at this design condition for approximately 65 minutes. After this time the cooling water to the absorber was partially closed causing the pressure in the absorber vessel to rise, as shown in Figure 16. At 74 minute point the temperature of the steam in the suction manifold began to rise indicating that the direction of the primary flow had begun to reverse in the diffuser and the critical back-pressure had been exceeded. The absorber pressure at which flow reversal just began was approximately 1.4 kPa.

Conclusions.
The results of these experiments indicated that the critical back-pressure of the jet pump at its design operating conditions is 1.4 kPa. This is approximately 620 Pa greater than the required normal operating value. It was also estimated that the entrainment ratio (R_m) was about 0.45 under these conditions. If the diffuser throat diameter were to be increased from 33 mm to 49mm it is estimated that the entrainment ratio would rise to unity whilst the critical back-pressure would fall to approximately 1 kPa but the entrainment ratio would rise to nearer the unity design-point value.

3.6 TIS Freezing Experiments

Aims
The aims of these experiments were,

- to compare the freezing rates of the ice storage elements at three layers within the TIS; upper or 1st layer, middle or 5th layer, lower or 8th layer
- to estimate the cycle COP during the freezing process

Method
The elements were placed inside the TIS in 8 layers with approximately 48 elements per layer, as shown in Figure 13. Thermocouple probes were inserted in 7 elements placed in different layers. The elements were chosen amongst the closest to the layers central core. From previous experiments it could be assumed that the elements would freeze from the outer surface towards the centre and if the elements were partially frozen then the thermocouples readings taken at the core of each element would record a value greater than 0^{0}C.

The system was cycled several times with the generator, absorber and TIS store at its design condition. The temperature at the core of a centrally located element at the 1st, 5th and 8th layers was measured and recorded along with the absorber temperature. Figure 17 show some typical results during the freezing process.

Results and Conclusions
It can be seen from these that the all layer tended to freeze at about the same rate. There was also evidence of super-cooling, at about the 35 minute mark, before freezing began. This process lasted a further 165 minutes. During this time approximately 4.825 kg of water vapour was sublimated from the original 41 kg of water in the store, which left 36.17 kg of ice. The jet-pump entrainment ratio (R_m) during this experiment was estimated to be 0.443. The heat input at the generator was estimated to 2.6 kW whilst that removed with the steam from the TIS was estimated to be 1.22 kW. This gives and overall time-averaged coefficient of performance of 0.469.

3.7 Full cycle testing

Aim
To determine the operating characteristics of the absorption refrigerator, in terms of stability and control, over an extended period.

Method
The generator was filled with solution and heated to drive out water, raising its concentration to approximately 55%. The cooling water to the condenser and absorber were switched on and the evaporator heater, were activated. The absorption refrigeration cycle was operated for several hours at design temperature conditions.

Results and Conclusions
With the generator temperature of 80^0C, a condenser temperature of 14^0C, the temperature in the absorber was approximately 30^0C. At the time the precise cooling rate of the machine could not be determined because the heat meter was found to be inoperative at the beginning of these preliminary tests. However, the cooling rate at the evaporator was thought to at least equal the design-point requirement.

Several operating problems were identified that are now being rectified. The generator pressure was found to fluctuated and on occasions suddenly. This was thought to be caused by variations in the water level in the condenser, which as it rose covered the cooling coils and so reduced the condensation rate forcing the pressure to rise. To overcome this minor problem it was necessary to reduce the restriction in the expansion valve between the condenser and evaporator, which caused the water to drain back to the evaporator more easily under reduced pressure difference.

A further weakness of the experiments was a lack of information on the concentration entering and leaving the absorber. To overcome this problem a densiometer is to be fitted in the pipeline between the absorber and generator. This will be fitted in such a way as to allow the density of both the flow and return solutions to be measured alternately.

Full rig tests are now proceeding.

4. Discussion

4.1 Progress to date

The manufacture of the rig has been completed and all sub-systems have been tested. Experiments have been carried out to determine the optimum concentration of water-gel used in the construction of the storage elements. Preliminary testing of a small-scale TIS with a jet-pump refrigerator (described in Appendix A) revealed the importance of providing a clear flow passage for steam from the lower regions of the TIS. Without a clear flow path it was found that the rate at which elements in the upper part of the TIS froze was significantly greater than those at the bottom of the vessel. Leaving 5 elements uncharged at each level solved this problem.

The entrainment and pressure lift performance of the jet-pump have also been measured. These results were used to determine the optimum location of the primary nozzle within the mixing chamber was determined. The critical back-pressure of the jet-pump was also measured and found to be 1.4 kPa when the system was operating at its design pressures. The freezing rate of the TIS was measured and the heat removal rate was estimated to be 1.22 kW with a generator heat input of 2.6 kW. The coefficient of performance in this case was 0. 469. Although a full rig test has been completed and the results so far did demonstrated that the absorption refrigerator operated satisfactorily with the TIS, there were some problems with the instrumentation, particularly the measurement of the salt concentration in the absorber and generator. Some improvements to the instrumentation are now in-hand and the testing programme will be continued.

4.2 Future research and development

We have secured funds to continue the project for a further 18 months. During this time the experimental programme will continue. We will also begin the process of defining the capital and life-cycle costs of a plant compared with those of a standard absorption machine. The purpose of this exercise is to quantify the potential capital cost and energy savings resulting from the potential downsizing the absorption cycle machine and the cost of the TIS system. A search for an industrial partner will continue, using the services of the School’s Business Development Officer, to fund further development.

4.3 Pathway to Market

The novel TIS – absorption cycle concept has a potential to promote the use of Combined Cooling Heating and Power (CCHP) schemes in a wide range of potential commercial and industrial applications ranging from a tens of kilowatt to megawatt electricity generator capacity. These applications include buildings that require daytime comfort conditioning but also require nighttime electricity and when waste heat can be used to generate cooling for a TIS. Such applications might include supermarkets, office complexes and hospitals. Other applications, such as found in the food and chemical process industries and the dairy industry, that use batch production in which cooling is only required during only part of the day might also take advantage of the technology.

We will continue to search for an industrial/commercial partners to fund the development of the machine. These might include existing manufacturers of absorption cycle cooling machines and end users. Funding is required to support not only the further research and development already described but also the manufacture of pre-production unit and the cost of its installation in a suitable application.

5. Conclusions

The manufacture and testing of a laboratory scale lithium-bromide/water single–effect absorption cycle refrigerator machine and a novel thermal (ice) storage system was described. Recommendations for future research and a route to exploitation were also discussed. The main findings of the research programme to date are as follows:

- Experimental data confirmed the technical feasibility of the novel system

- Optimum concentration of water-gel used in the construction of the storage elements was measured at approximately 98%.

- Optimum layout of the elements within the TIS, including the positioning of the 'empty' storage elements were determined, Figure 13.

- Optimum location of the primary nozzle within the mixing chamber was determined to 20 mm upstream from the zero datum position at the inlet plane to the conical mixing section.

- Critical back-pressure of the jet-pump was found to be 1400 Pa at the design-point generator pressure.

- Entrainment ratio and secondary pressure lift ratio of the jet-pump were experimentally measured to be 0.443 and 2.15. The required secondary pressure lift ratio was approximately 1.3. This means that the diffuser throat diameter of the jet-pump may be increased to permit more secondary mass flow and thereby increase the entrainment ratio closer to the anticipated design value of about unit.

- With only the jet-pump cycle operating, that is with only water in the absorber, the machine was able to produce approximately 36 kg of ice in 165 minutes which is equivalent to approximately 1.22 kW. When a LiBr-water solution was added to the system the charging rate was significantly greater.

- Coefficient of performance of the cycle during the jet-pump only charging cycle was estimated to be 0.469.

- The results demonstrated that the absorption refrigerator operated satisfactorily, however, there are some minor modifications needed to improvement the instrumentation.

- Before going forward to develop the concept further it will be necessary to define in some detail the capital and life-cycle costs of a commercial plant compared with a standard absorption machine. This study is required in order to quantify the potential cost and energy savings resulting from the potential downsizing the absorption cycle machine and the cost of the TIS system.

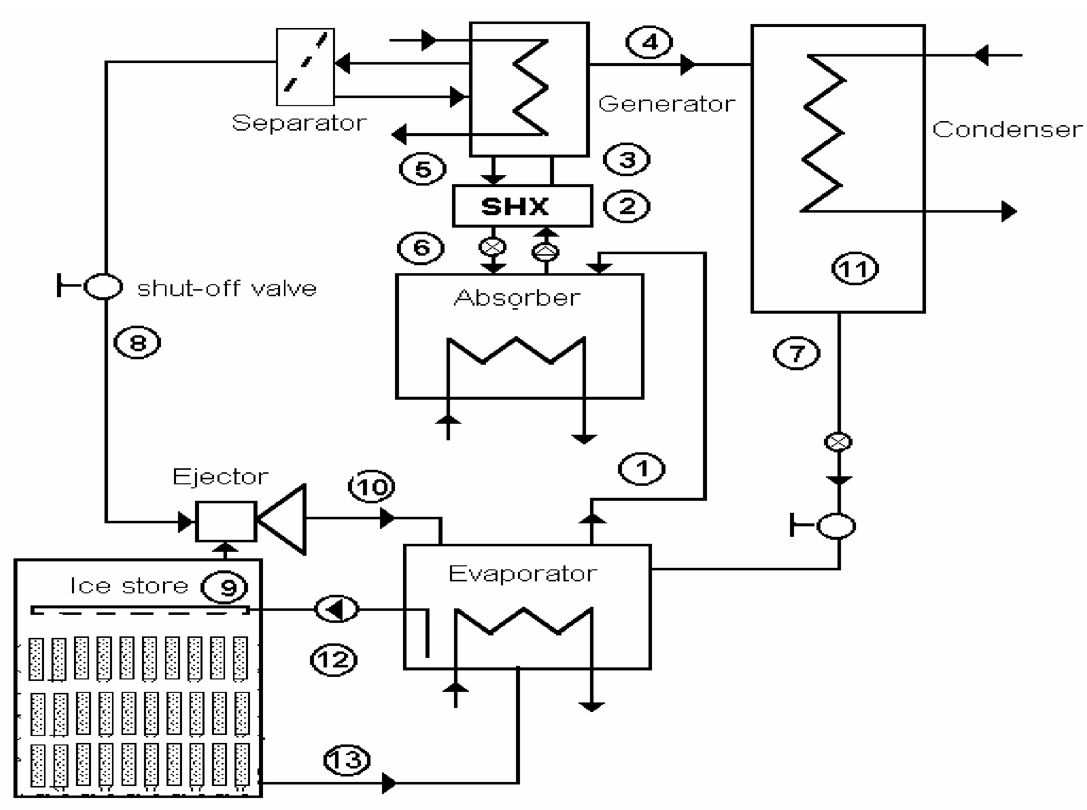


Figure 1: ***Absorption Cycle Thermal (ice) Store System***

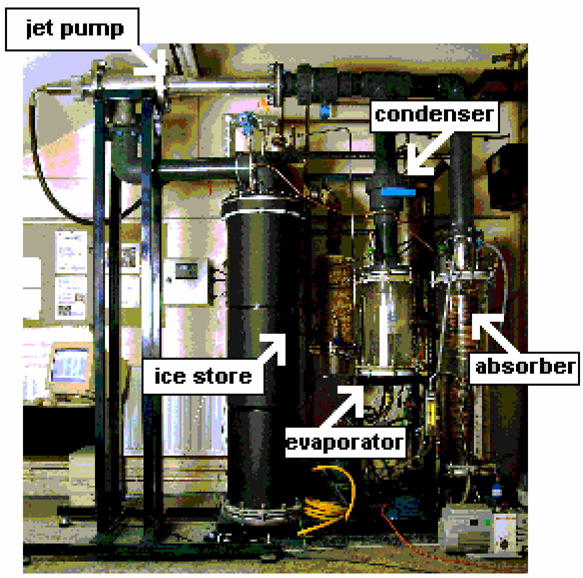


Figure 2: *Photograph of experimental rig showing main components*

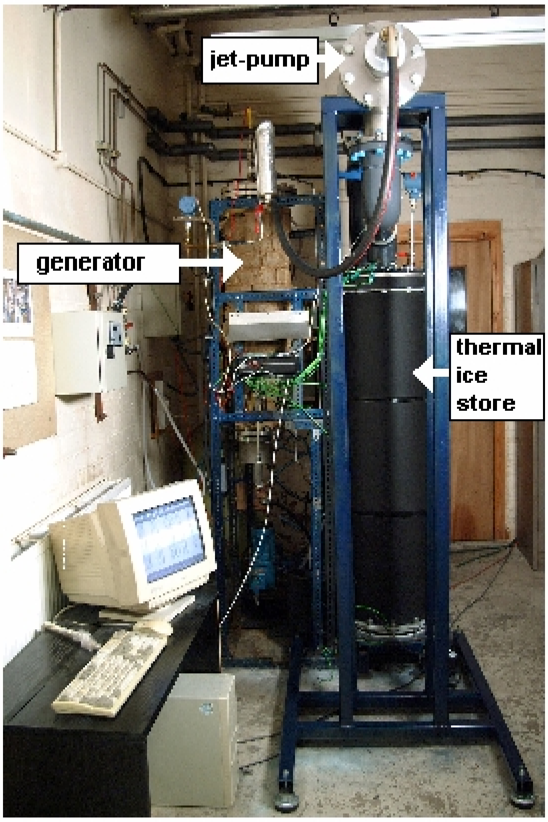


Figure 3: *Side-view of test rig*

Figure 4: *A further view of the test rig*

Figure 5: *A close-up view of test rig showing jet-pump connected to plastic suction and discharge pipes fitted with 4 inch nominal diameter (101.6 mm) plastic ball-valves.*

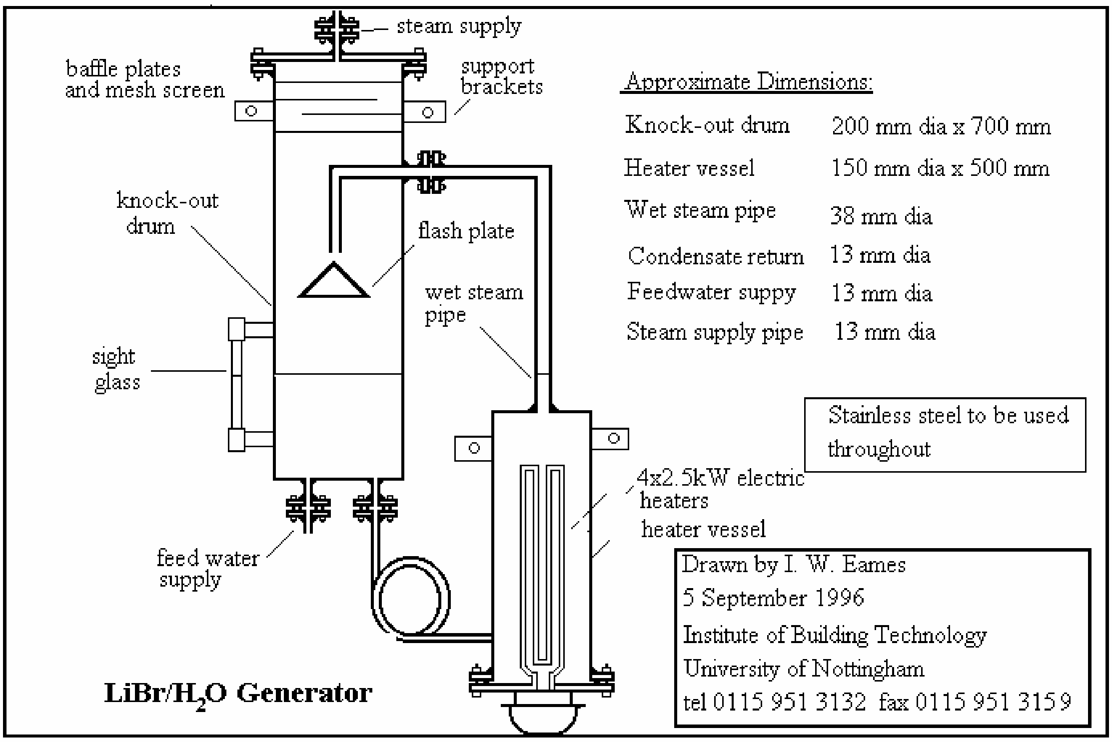


Figure 6: ***LiBr/H2O Vapour Generator***

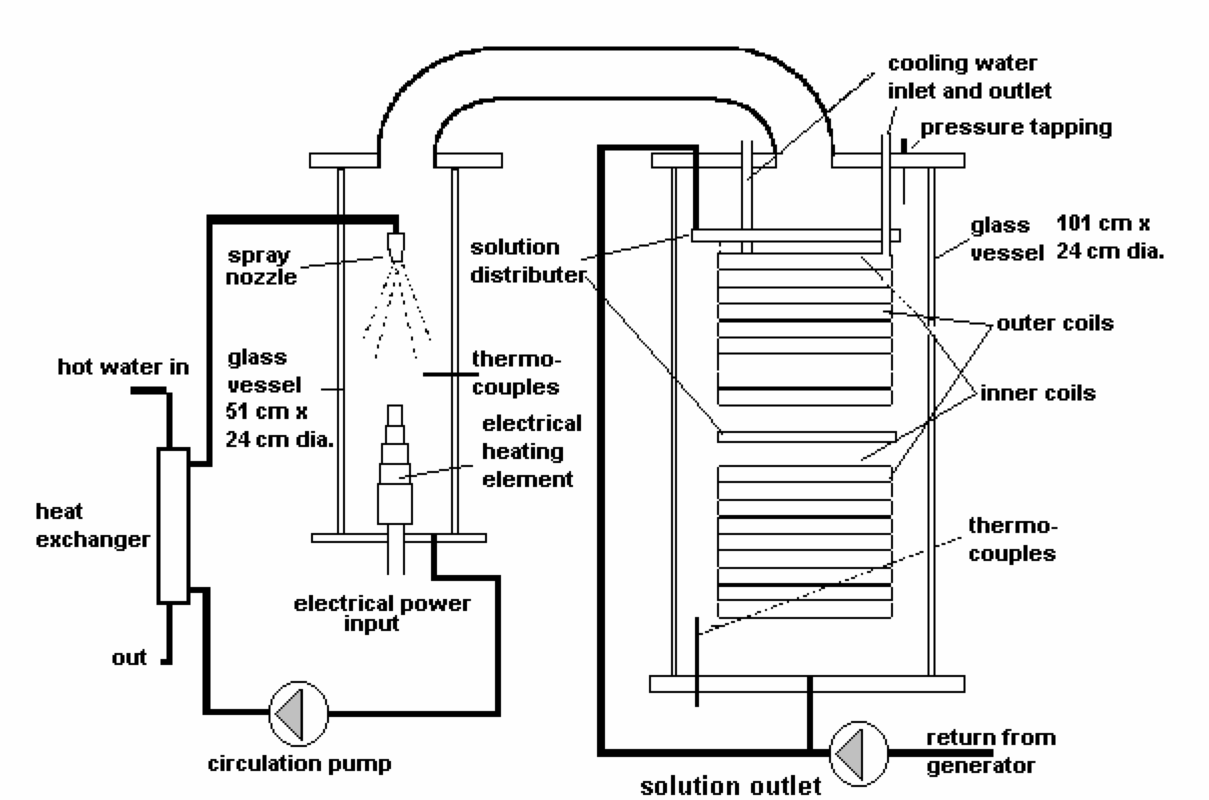


Figure 7: *Evaporator – absorber assembly*

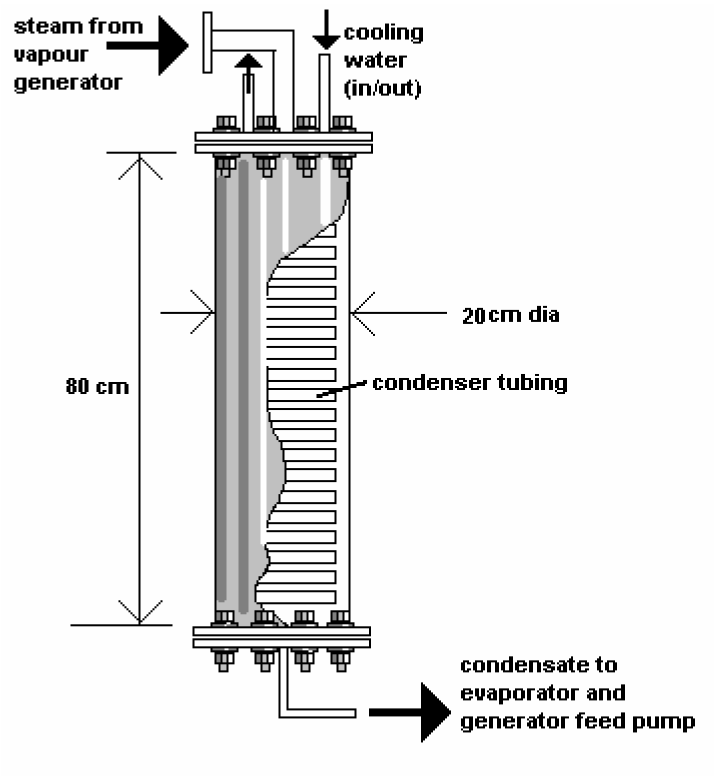


Figure 8: *Condenser construction*

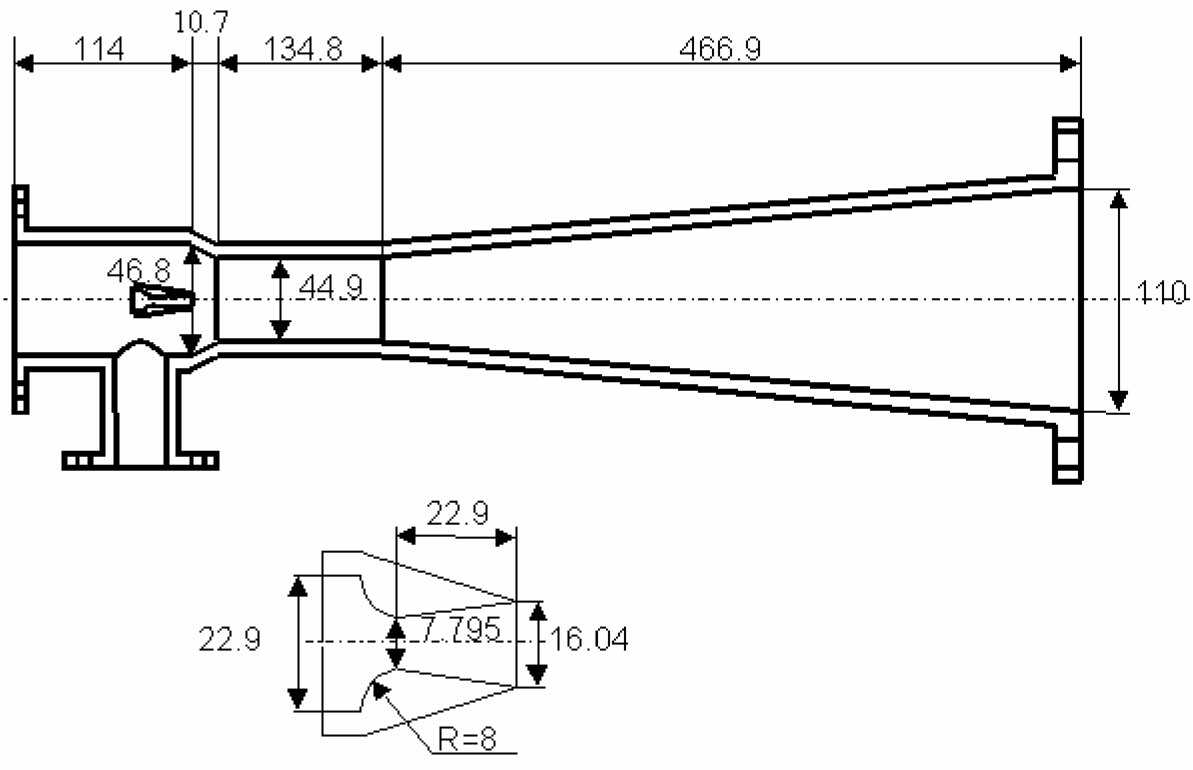


Figure 9: *Steam ejector design details*

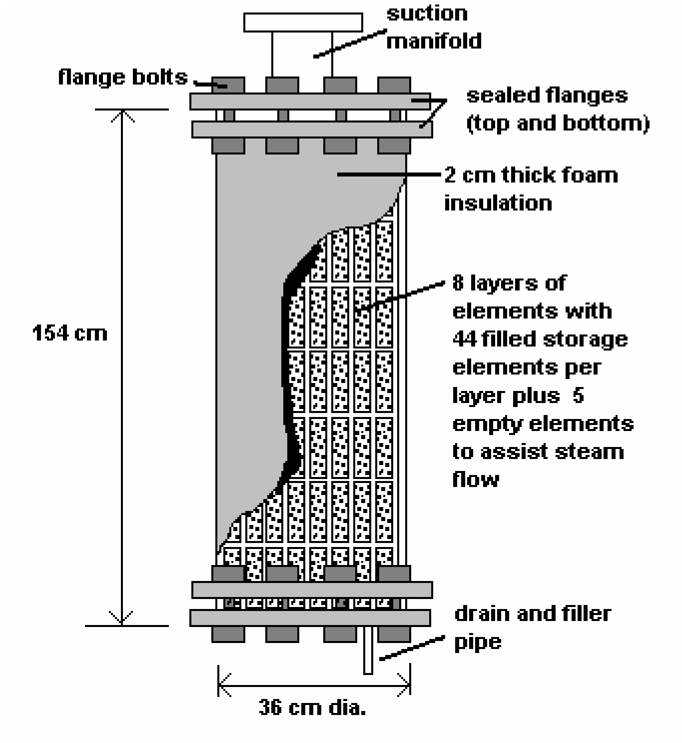


Figure 10: *Thermal Ice Storage vessel*

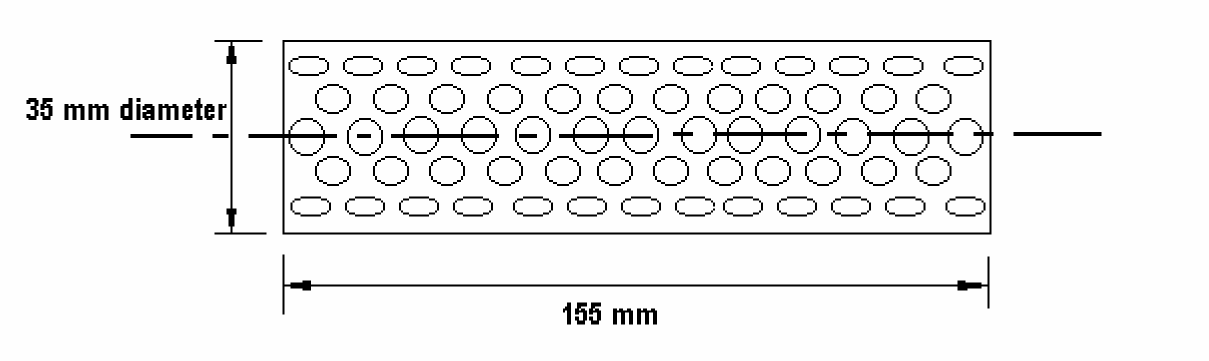


Figure 11: *Gel-ice Storage basket*

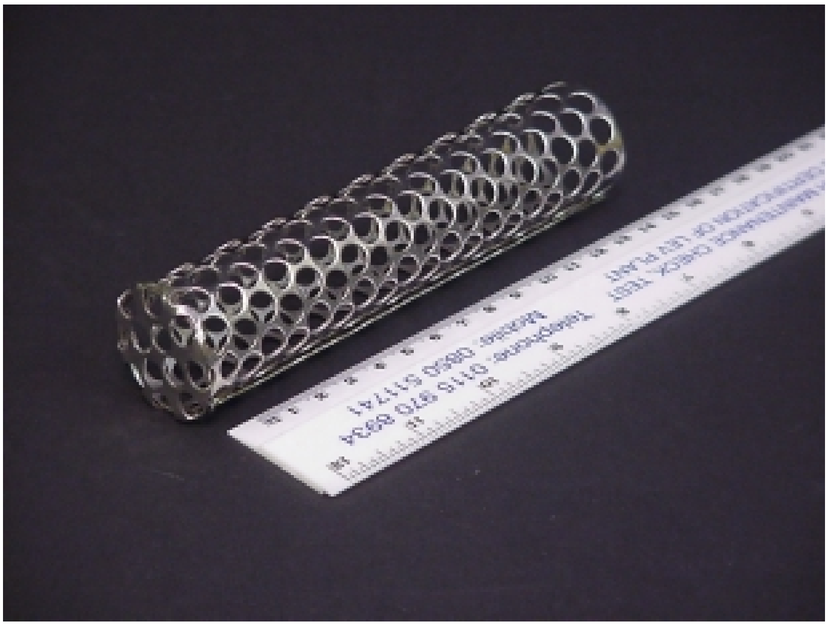


Figure 12: *Sample gel-ice storage basket from manufacturer*

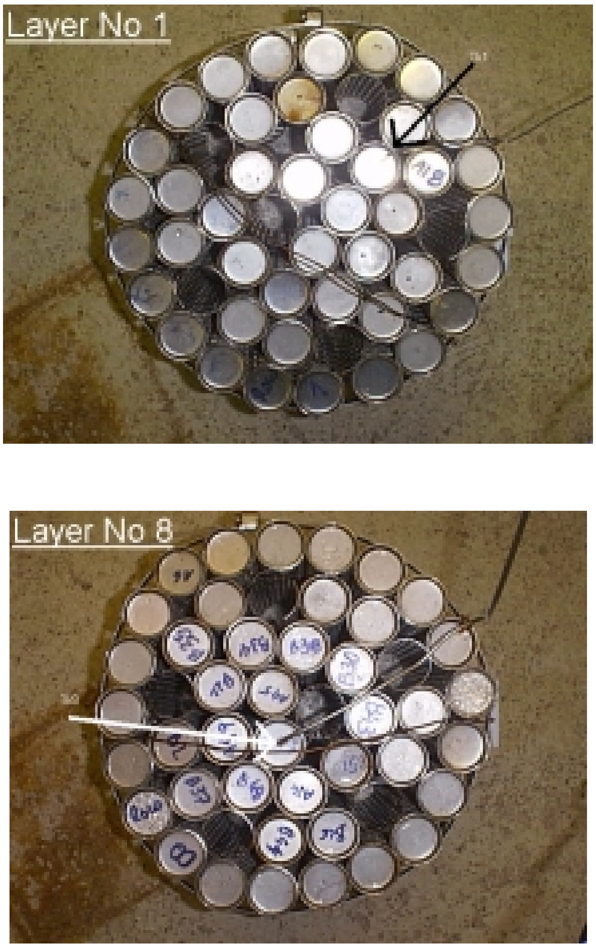


Figure 13: *Showing the distribution of storage elements in the upper lay (No 1) and the lowest layer (No 8) within the TIS vessel.*

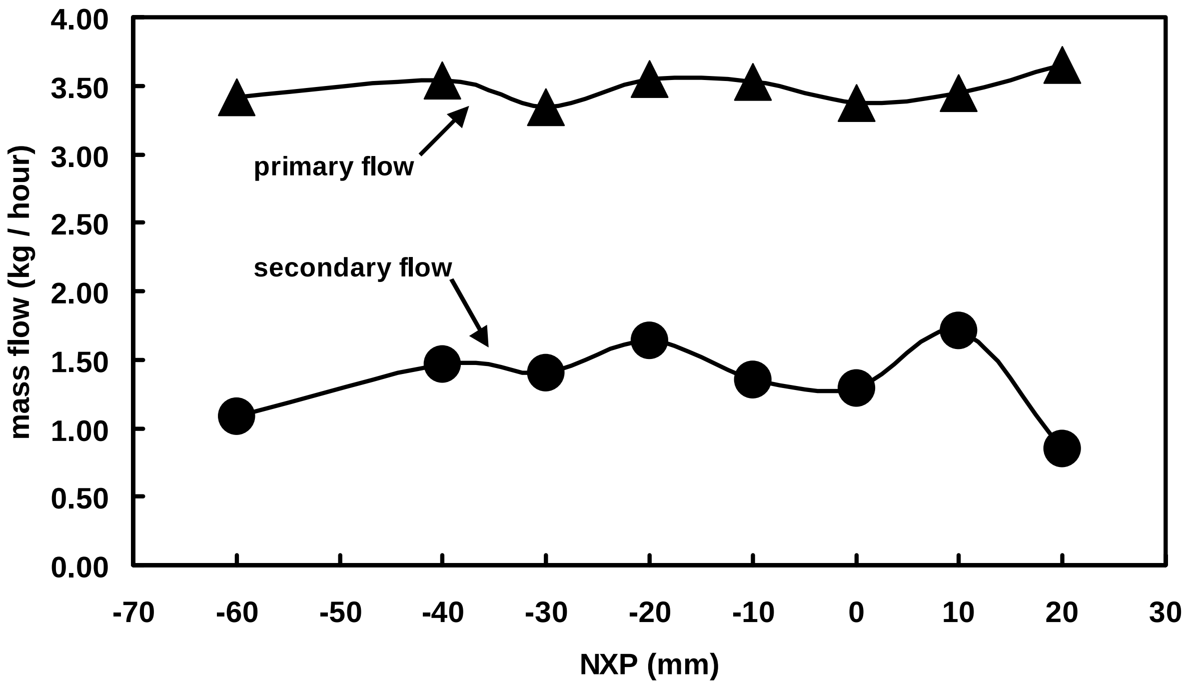


Figure 14: *Variation in jet-pump primary and secondary mass flows with nozzle exit position (NXP)*

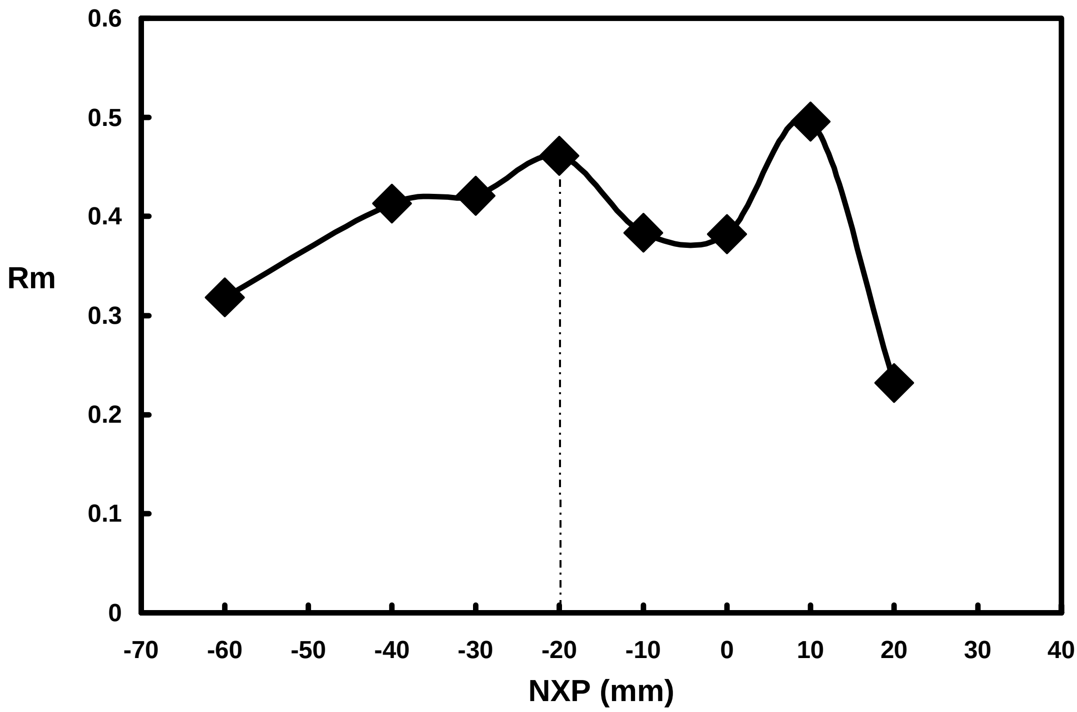


Figure 15: *Variation in jet-pump entrainment ratio with NXP*

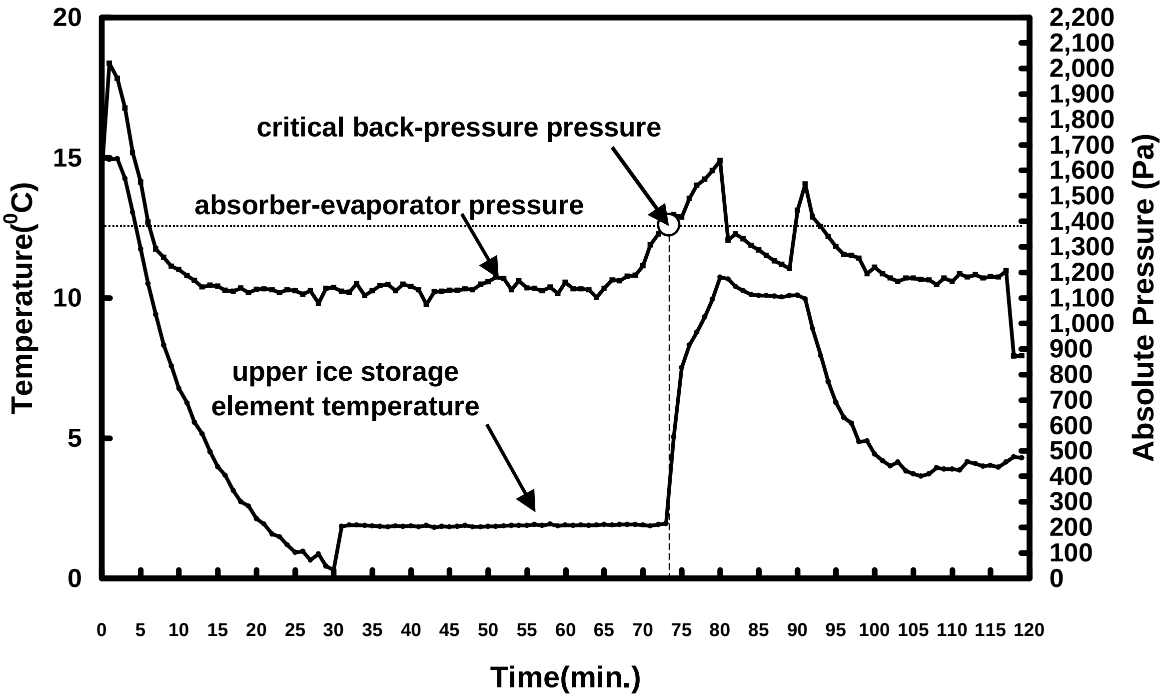


Figure 16*: Variation in TIS element temperature with back-pressure showing the critical value to be 1.4 kPa*

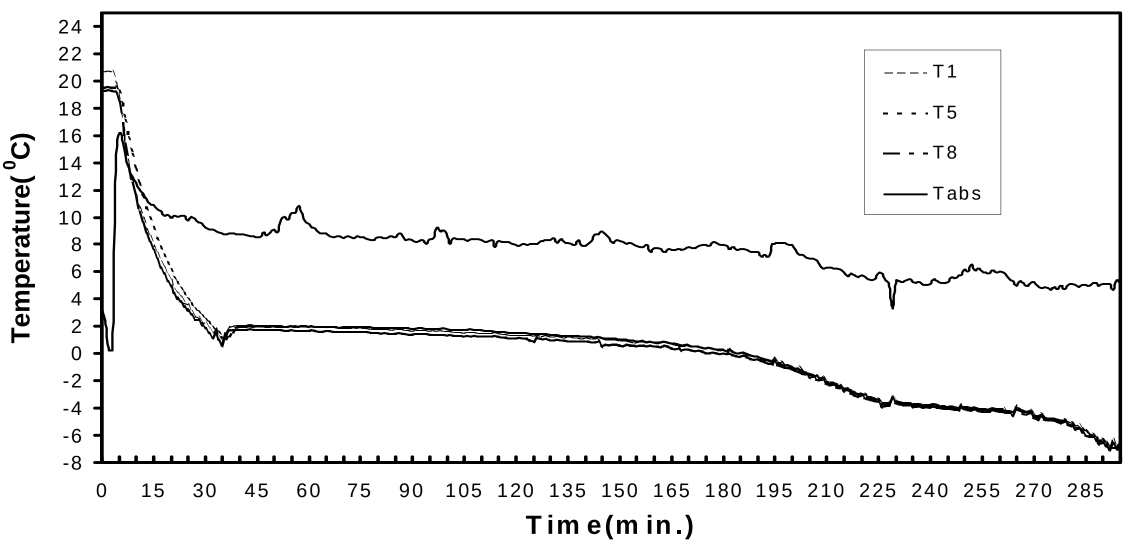


Figure 17: *Showing the time variation in absorber temperature and element temperatures at three levels with the TIS.*

Appendix A

Vacuum freezing ice storage as an alternative to conventional methods

Ian W. Eames, Jorge A. J. Caeiro

Abstract

This paper describes the process of thermal (ice) storage using a vacuum freezing process. Conversely to other ice storage methods, the vacuum freezing process relies heavily on coupled heat and mass transfer of the storage medium. In a vacuum freezing process the storage medium is used at the same time as the refrigerant. In the evaporation process a loss of storage capacity occurs. To predict the amount of water to be evaporated a semi-empirical expression was derived. The results show that in the range of usual working temperatures 12% to 16% of the initial mass of water has to be evaporated in order to freeze the remaining.

Keywords: Thermal energy storage; Thermal ice storage; Heat powered refrigeration cycles; Vacuum freezing, Ejector refrigeration cycles

1.Introduction

Over the years, demand for electricity in the UK has gone up averaging 2 per cent a year **[1]**. The rapid depletion of fossil fuels due to an increasing energy demand requires a more efficient use of this finite natural resource.

Environmental concerns due to CO_2 emissions, namely the greenhouse effect, are overtaking simplistic economic considerations because of the unpredictable macro scale costs and losses that climate change can eventually produce.

Demand for electricity consumption in the service sector has been growing sharply since the early 1980's. This increase in electricity demand is due mainly to an ever-increasing reliance on electrical equipment (information technology, medical and leisure equipment) and the growing use of air conditioning.

Air conditioning systems are traditionally based on electrically powered vapour compression cycles. Heat powered refrigeration cycles have comparatively lower Coefficients of Performance (COP), ranging from 0.2 to 0.8, than vapour compression systems -typically 2-4. The COP is a measure of the performance of the cycle on its own. If several cycles of the same kind are compared with each other, the cycle with the larger COP is considered the most efficient. From an overall energy efficiency point of view however the COP is not an appropriate measure because it does not take into account the quality of the energy being used. The decisive value for the energetic efficiency evaluation must be the Primary Energy Rate (PER) defined as the ratio of primary energy demand to the required output. The system with the lowest value of PER is considered the best with regard to energy consumption as is shown in Ref. **[2]**.

Heat powered refrigeration systems can use low-quality energy which makes them particularly suitable for sites where exists an available source of waste heat or a renewable source of energy. When heat powered refrigeration systems are combined with Thermal (Ice) Storage (TIS) systems a more efficient and effective utilisation of the equipment can be made. In addition to these advantages further benefits are reduced equipment

Nomenclature

A	area of heat and mass transfer (m^2)	m_{sub}	mass of ice sublimated
dw/dt	rate of evaporation in the constant rate period (g/m^2s)	m_s	total mass of water evaporated (kg)
h	heat transfer coefficient from the wet surface to the low pressure vapour (W/m^2 K)	P_0	vapour pressure inside the evaporator (Pa)
h_{fg}	latent heat of vaporisation (kJ/kg)	P_w	water vapour pressure (Pa)
h_f	enthalpy of water at the beginning of the process (kJ/kg)	Q_f	total heat to be removed from ice store
h_i	enthalpy of ice at 0^0C (kJ/kg)	$Q_m = \Sigma\ m\ c\Delta T$ = heat to be removed from containing materials (kJ)	
h_{if}	latent heat of freezing (kJ/kg)	Q_W	heat to be removed from water (kJ)
h_{sub}	latent heat of sublimation at 0^0C(kJ/kg)	***Greek***	
k_G	mass transfer coefficient for diffusion from the wet surface through the vapour film (g/m^2s Pa)	ΔHv	enthalpy of water vaporisation (kJ/kg)
m_0	mass of water at the beginning of the process (kg)	ΔHw	enthalpy of wetting the solid (kJ/kg)
m_f	mass of water at the end of the process (kg)	ΔT	temperature difference between the water vapour and the surface (K)
m_{int}	remaining water at the beginning of ice sublimation (kg)		

size, decreased capital and life cycle costs. Decreasing first costs is particularly important in sorption refrigeration systems because of their higher capital costs when compared with other commercially available systems.

2. Description of the vacuum freezing ice storage process

In most Ice Storage systems, ice is still made at night using conventional refrigeration systems that take advantage of the low electricity rates in effect during the night. However, in recent years, it has become standard practice to consider combined heat and power systems (CHP) early in the design stage of commercial buildings. Distributed power generation makes available waste heat that can be used in heat powered refrigeration cycles.

Four main types of Ice Storage systems are commonly used in commercial and industrial applications: ice-on-coil-type systems, static ice tank systems, ice harvesting systems and encapsulated ice store systems. More detailed information on the different systems can be found elsewhere, e.g. Hasnain **[3]**.

An alternative method of ice production is by vacuum freezing. A steam ejector is needed to drive out the large volume flow rate of vapour generated inside the TIS and sustain the pressure below the triple point of water. Although this process is commonly used in heat sensitive applications like food and pharmaceutical processing industries it has seldom been used for air conditioning purposes because of the low COP.

Over the last years however, the working performance of steam ejectors refrigerators has been thoroughly investigated and improved **[4-10]**. Although steam ejectors have comparatively low efficiencies they are mechanically simple devices with no moving parts meaning little wear and maintenance. They can also be driven by low grade heat which decreases significantly their running costs.

Figure 1 shows a schematic diagram of an ejector refrigeration cycle with TIS.

In the steam generator is produced the primary –motive- fluid (m_p). The primary flow enters the ejector nozzle, where it expands to produce a supersonic flow that creates a low-pressure region within the mixing chamber. This region of low pressure draws the secondary flow (m_s) from the TIS producing the cooling effect. Conversely to other storage processes, vacuum freezing relies on coupled heat and mass transfer of water being the cooling effect a function of the amount of vapour released by the storage medium. Water is used simultaneously as the storage medium and as the refrigerant. The TIS is the system's evaporator.

To increase the area of mass transfer the medium is placed in perforated capsules (storage elements) that allow water to freely flash into the evaporator. This particularity avoids the need for some of the expensive heat exchangers usually required in conventional systems. An agglomerating material is added to water in order to retain it inside the capsules and form a semi-solid gel.

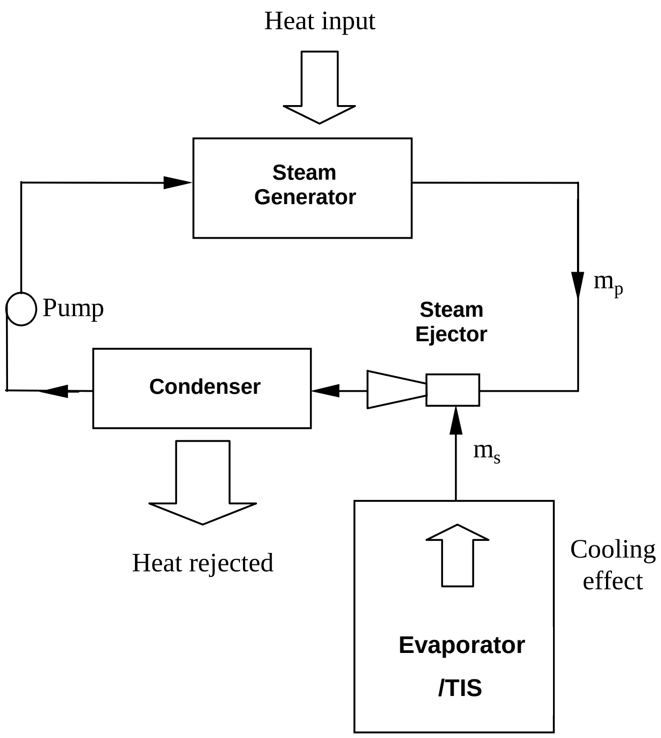


Figure 1 – Steam ejector refrigeration cycle with TIS

2.1. Some of the polymer physical properties

The material used to agglomerate water is a superabsorbent polymer (partially neutralised poly acrylic acid). Superabsorbent polymers are one special group of polymers because of their ability to absorb many hundreds of times their own mass of water. In their dry crystalline form they are specially formulated to swell very rapidly in water but they do not dissolve. Their swelling behaviour is characterised mainly by the amount of water they absorb and the rate of absorption. The physical mechanisms behind the process of polymer swelling can be found more in detail elsewhere **[11-13]**.2.2. The loss of moisture of the elements and its influence on the vacuum freezing process

Gibbs free energy for water in the liquid state and its vapour in an equilibrium state must be the same

$$g_f = g_g$$

By decreasing the vapour pressure inside the evaporator this equilibrium is broken. In an adiabatic process the thermodynamic equilibrium is restored at the expenses of the fluid own internal energy. The evaporation process leads eventually to freeze the water. There are two well defined periods of moisture release from a drying solid. A first period where the rate of drying is constant and a second where there is a steady fall in the rate of drying as the moisture content is reduced. The moisture content at the end of the constant rate period is known as the *critical moisture content.* During the constant rate period it is assumed that moisture migrates from the saturated surface of the material by diffusion of the water vapour through a stationary gas film into the surrounding environment. For normal engineering design work equation 1 is commonly used.

$$\frac{dw}{dt} = k_G A(P_w - P_0) = \frac{hA\Delta T}{h_{fg}} \qquad (1)$$

The rate of moisture release is thus determined by the values of h, ΔT, *and* A, and is not influenced by the conditions inside the solid.

$(P_w - P_0)$ is the driving force to remove moisture from the wet surface into the surrounding environment.

In the ice storage elements moisture may be merely trapped in spaces between the solid particles and no significant lowering of the vapour pressure occurs, or maybe bound to the solid and so no longer is the full vapour pressure exerted.

Three moisture–retention regimes are observable in a sodden non-hygroscopic condition:

1. Moisture is unbound, held on the surface.

2. Moisture bridges particles in the solid skeleton.

3. Moisture is held in the finest capillaries.

Free moisture that falls in the first regime can be easily released, with relatively low energy expenditure. Equation 2 shows that for lower p_w more work has to be done to remove moisture from the elements affecting the COP of the freezing process.

$$W = -\Delta G = -RTln\frac{p_0}{p_w} \qquad (2)$$

The work W (kJ/kmol) in driving off moisture can be expressed in terms of the relative vapour pressure p_0 / p_w for an isothermal, reversible process without change of composition.

Work has to be done not only to promote the evaporation of water but also to remove the moisture retained in the gel. The total entalphy for moisture evaporation is equal to (ΔH_v + ΔH_w). The enthalpy of wetting increases greatly when the moisture content decreases.

According to *Keey* **[14]** however even with markedly hygroscopic materials, the total enthalpy for moisture evaporation differs significantly from the latent heat of vaporisation for free moisture ΔH_w only at moisture contents of order 0.1 kg/kg or less.

The moisture lost by the elements during the charging period has to be restored periodically before they become too dry falling in a moist hygroscopic condition and loosing excessive storage capacity.

Circulating water through the TIS during the discharging period can for example restore the initial moisture content.

3-The fundamental equation of a vacuum freezing process

Two distinct regimes occur during the vacuum freezing process: a sensible and a latent cooling regimes. During the first regime, sensible cooling occurs and the elements temperature drops down to temperatures below 0^0C. After a short period of time the outer layer temperature reverts to 0^0C, indicating that nucleation and macroscopic ice crystal growth has begun. During the second regime the crust of ice sublimates and the rest of the water is progressively frozen towards the centre.

To derive this equation it is initially assumed that the storage medium and containing materials are all at the same initial temperature and that the system is adiabatic. It is also assumed that no sub cooling of the ice occurs after the freezing process is completed.

The heat $\mathbf{Q_f}$ that has to be removed from the control volume in order to freeze the water is given by equation 3

$$Q_f = Q_w + Q_m \qquad (3)$$

Equation 3 can also be expressed in terms of mass and state properties

$$\begin{aligned} & m_0 h_f - m_f h_i + \Delta T \sum_{i=1}^{n} m_i c_i = \\ & = (m_0 - m_{int}) \cdot h_{fg} + m_{sub} h_{sub} \qquad (4) \end{aligned}$$

The heat withdrawn during sensible cooling is expressed by $(m_0\text{-}m_{int})\cdot hfg$, where h_{fg} represents the mean value between the latent heat of vaporisation at the initial temperature of the medium and at the lowest temperature reached in the sensible cooling regime. This simplification is introduced in order to overcome some cumbersome calculations to determine Q_{sens}.

The error eventually introduced is not significant as h_{fg} varies linearly and very slightly in the usual range of working temperatures.

When the ice starts forming, the existing mass of water inside the control volume is given by equation 5.

$$m_{int} = m_f + m_{sub} \qquad (5)$$

Under adiabatic sublimation freezing, the cool stored, as ice is equal to the heat released due to sublimation

$$m_f h_{if} = m_{sub} h_{sub} \qquad (6)$$

With these three expressions, a system of equations can be set and solved to determine the three unknowns $\boldsymbol{m_f}$, $\boldsymbol{m_{int}}$ ***and*** $\mathbf{m_{sub}}$,.

After solving the system of equations, expression 7 was found

$$m_f = \frac{m_0 \cdot (h_f - h_{fg}) + Q_m}{\left(h_i - h_{fg} - \dfrac{h_{if} \cdot h_{fg}}{h_{sub}} + h_{if} \right)} \qquad (7)$$

By giving values from the steam tables to $\boldsymbol{h_i}$, $\boldsymbol{h_{if}}$ and $\boldsymbol{h_{sub}}$ and assuming that $|\boldsymbol{h_i}| \approx |\boldsymbol{h_{if}}|$, equation (7) can be further simplified into expression (8)

$$m_f = \frac{m_0 \cdot (h_{fg} - h_f) - Q_m}{1.118 \cdot h_{fg}} \qquad (8)$$

The former expression may be considered the fundamental equation of a vacuum freezing process as it allows calculating the total water evaporated during the freezing process and its loss of storage capacity as a function of the initial conditions of the medium.The total mass of water to be evaporated can be promptly determined from

$$m_s = m_0 - m_f \qquad (9)$$

4. Comparison of experimental results with theoretical

In order to prove the validity of the above equation, experimental data obtained by *Worral* **[15]** during his research were compared with the calculated theoretical results. The following sub chapters describe how his experiment was conducted and the experimental data obtained.

4.1.Experiment description

4.1.1.TIS testing rig

The test facility used by Worral is represented schematically in figure 2.Its main components are a 7kW electrically powered steam generator, an ejector assembly, a 8 kW water cooled copper coil condenser and the flash evaporator.

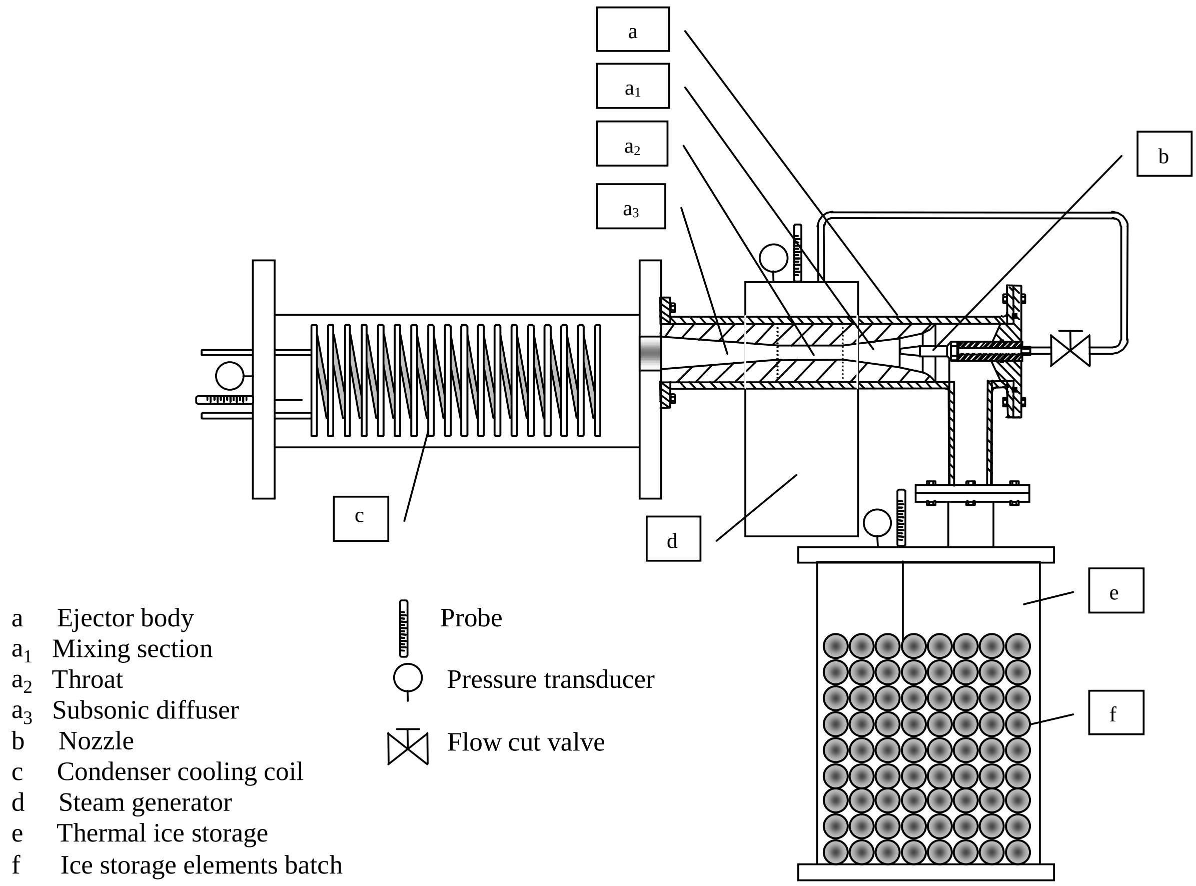


Figure 2. – TIS testing rig

The condenser and the evaporator were column shaped glass vessels. At the ends of the glass vessel were two stainless steel plates. The top end plate was welded to the secondary flow inlet tube. The evaporator was partially filled with copper spheres. The spheres (storage elements) were filled with sachets made from a permeable material that easily allows the diffusion of water vapour. The sachets on their turn were filled with a water saturated polymer (gel). When the refrigeration process was being undertaken the evaporator was covered with a ½ inch sheet of insulating foam to prevent heat gains either by conduction or radiation. A probe was inserted into one of the elements to record its temperature variation throughout the experiment. The amount of water desorbed by the elements was determined by measuring the amount of water condensed and subtracting the primary flow. The primary flow was determined directly from the generator variation of water volume.

4.1.2.Comparison of computed with experimental results

There was a loss of mass during sensible cooling due to evaporation and by latent cooling due to sublimation. The initial mass of water was estimated to be **8.268 kg**. The mass and specific heat of materials in evaporator vessel are as in Table 1.These values can be found in most heat transfer textbooks varying however slightly from author to author.

At the start of the freezing process the elements were at an initial temperature of **19^{0}C**. The minimum temperature reached in the sensible cooling period was approximately **–2 ^{0}C**.

Table 1- Mass and specific heat of materials in evaporator vessel

Mass of copper m_c = 24.853 kg	c_c = 0.385 kJ.kg/K
Mass of stainless steel m_{ss} = 19.075 kg	c_{ss} = 0.480 kJ.kg/K
Mass of glass m_g = 15.220 kg	c_g = 0.835 kJ.kg/K

By using the former values the heat that had to be removed from the materials in order to drop their temperature from the initial temperature of 19^{0}C to -2^{0}C was calculated to be **Q_m= 660.10 kJ**.

From the steam tables, the values of h_{fg} and h_f corresponding to 19^{0}C are equal to **2,455.85kJ/kg** and **79.71 kJ/kg** respectively, and at –2^{0}C the latent heat of vaporisation is equal to **2,834.72 kJ/kg**. The mean value of h_{fg} was found to be **2,645.3 kJ/kg**.

According to equation 8 the remaining mass of water at the end of the freezing process should be

$$m_f = \frac{8.268 \cdot (2,645.3 - 79.71) - 660.1}{1.118 \cdot 2,645.3} = 6.949 \ \ kg$$

Hence the total mass of water to be evaporated should be equal to **m_s=1.319 kg.**

From equation 6 it was determined that the mass of ice to be sublimated should be equal to

$$m_{sub} = 333.39 \times \frac{6.949}{2,834.2} = 0.817 \ \ kg$$

The mass of water to be evaporated before the ice started forming should be equal to

$$m_{sen} = m_0 - m_f - m_{sub} = 8.268 - 6.949 - 0.817 = 0.502 \ kg$$

Table 2 summarises the theoretical and experimentally based results.

Table 2- Comparison between theoretical results and experimental data

Results	Theoretical	Experimental
m_s (kg)	1.319	1.305
m_{sub} (kg)	0.817	0.819
m_{sen} (kg)	0.502	0.486

4.1.3.Results and discussion

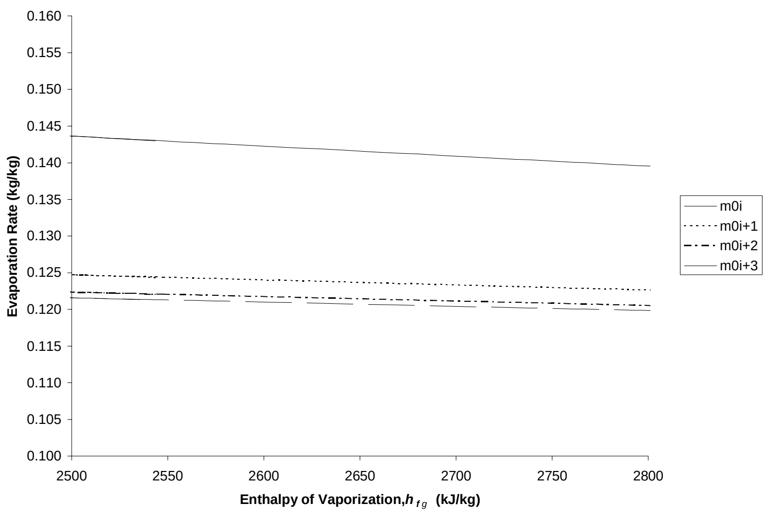
The difference in percentage of the calculated value of m_s relatively to the experimental data is about **1.06%** higher. For m_{sub} and m_{sen} the discrepancy is of about **–0.245%** and **3.18%**, respectively.

The discrepancy of the computed value of m_s when compared with the experimental may be considered negligible for most practical applications. In view of this excellent agreement this relation is believed to be an adequate representation of the physical phenomena.

The rate of water evaporation, defined as $\boldsymbol{\varepsilon = m_s/m_0}$ can be directly determined from (10)

$$\varepsilon = 0.106 + 0.894 \cdot \frac{h_f}{h_{fg}} + 0.894 \cdot \frac{Q_m}{h_{fg} \cdot m_0} \qquad (10)$$

Expression 10 results were plotted as a function of h_{fg} and m_0. All the other factors were kept constant, as their influence on the final results was found to be negligible. For illustrative purposes a range of m_0 from 10 kg to 150



kg, was arbitrarily chosen, with increasing steps of 50 kg ($m_{0i+n} > m_{0I}$)

The results clearly show that for different values of m_0 the variation of $\boldsymbol{\varepsilon}$ is very little influenced by a variation in h_{fg}.

Conversely, the ratio Q_m / m_0 is the factor that more influences the rate of water to be evaporated. Lower values of this ratio mean lower storage capacity loss . It can be seen from the graphic that with an increase of m_0 the rate of evaporation decreases progressively. Eventually cost/benefit factors will determine what is the highest value of m_0 to be used in a particular situation.

5.Conclusions

The main aim of this paper is to introduce vacuum freezing as an alternative to commercially available ice storage systems and to underline its main differences. During the process of vacuum freezing a loss of energy storage capacity occurs. The amount of storage capacity lost depends on the initial conditions of the medium and the physical properties of the building materials.

As a rule of thumb it can be assumed, that about 12 to 16 % of the initial mass of water has to be evaporated in order to freeze the remaining. This percentage is low enough to assume that the moisture released is free from the polymer influence and that no extra expenditure of energy is needed for this purpose.

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