THE UNIVERSITY OF ADELAIDE

DEPARTMENT OF MECHANICAL ENGINEERING

AN ALTERNATIVE APPROACH TO THE DESIGN OF ENVIRONMENTAL CONTROL SYSTEMS

by

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This thesis embodies the results of supervised project work making up all of the work for the degree.

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STATEMENT OF ORIGINALITY

The material in this thesis is the original work of the author and contains no material previously published or written by another person, except where due reference is made in the text.

The thesis does not contain what has been accepted for the awarding of a degree or diploma at any university.

SARIT KUMAR GAYEN

23 February 1983

ABSTRACT

A new concept of air-conditioning is described which combines the best features of both refrigeration cycle air-conditioning and "evaporative" cooling. It also incorporates a cold-side chilled-water store. The store is used as a buffer between the instantaneous thermal load and the supply from the chilling system. By introducing a time shift between the "demand" and the "supply", the store allows the refrigeration plant to operate at full load efficiency and at the peak of this efficiency which occurs when ambient conditions on the condenser side are most favourable. When sufficient cooling capacity is in store for anticipated needs, the refrigeration plant is switched off so eliminating part load inefficiencies and reducing maintenance costs.

As cooling capacity can be stored at periods of low demand (usually corresponding to periods of favourable condenser conditions) for use during the relatively short-term demand peaks, the capacities of the components of the refrigeration equipment can be less than would be required in the absence of a cold-side store. This feature reduces both capital and operating costs of the refrigeration plant.

Traditional storage systems have tended to be ruled out of consideration due to their cost and their inherent thermal losses. In the configuration described in this thesis, the store and the heat and moisture exchange surfaces have been integrated in such a way that overall capital cost is reduced and, by shrouding the store with the air to be cooled, store "losses" provide most of the required sensible heat exchange with the air flow. The air can also be brought into contact with an (adjustable) area of the chilled water surface to satisfy the remainder of the sensible heat exchange and provide the required

moisture transfer. In a humid climate, such as Darwin, or where high latent loads are generated within the conditioned space, the chilled water is below the dew-point temperature of the incoming air. The direct contact pass then serves to de-humidify the air. In a dry climate, such as Adelaide, or during the dry season in Darwin, the inverse is true and an evaporative cooling and humidification takes place in the direct contact pass.

The thesis discusses the basic features of the new system and explores aspects of the design process for the prototype units. Both economic and technical aspects are considered. Two prototype systems are described and their realizations and performances documented and discussed.

It is concluded that the system appears to have merit as a realistic alternative to conventional refrigerative air-conditioning systems. It also extends the viable geographic range for evaporative systems. Projected design and development for production units are discussed.

INTRODUCTION

INTRODUCTION

In the best of all possible worlds, the thermal energy supply to, and storage within, a built environment should, at any instant, satisfy the thermal energy "demand" of that environment at the minimum "cost". This implies that the thermodynamic state of the air supplied to the space should be maintained within a set of values prescribed in accordance with the desired conditions within the space, taking account of the properties of the space and its surroundings and the loads contained within it or imposed upon it.

The meaning of "minimum cost" often differs according to viewpoint. For the purposes of this work it is assumed to mean "at minimum owning and operating cost", however it is noted that techniques exist which allow the sizes of components comprising a given system configuration to be "optimised" with respect to any mathematically definable objective function, (see e.g. Tostevin and Luxton (1979)). Despite the power of these optimisation techniques, they require the configuration of the system to be defined before it can be optimised. It is the aim of this thesis to explore a new configuration of an environmental control system which, 'though yet to be optimised, shows considerable potential as a lower cost and more flexible alternative to conventional systems.

The thermal energy demand of a space at any instant is termed the instantaneous load. In a conventional environmental control system, the peak value of the instantaneous load determines:

- the capacity of an energy conversion device which converts primary or secondary forms of energy into thermal energy;

- the capacity of an energy exchange device in which thermal energy is exchanged with selected air;

and

- the capacity of the means of moving the mass of selected air necessary to match the thermal energy supply to the thermal load demand.

The maximum capacity of the energy conversion device can be reduced by increasing the mass of the building walls, roof, floor, by introducing shading devices for the windows and/or by the inclusion of a thermal store as an integral component of a total environmental control system. When a thermal store is incorporated, the additional energy from the store-plus-plant capacity can be used in times of low demand to fill the store which can then satisfy later peaks in the load demand. All buildings store energy within their fabric, and sometimes appropriate thermal storage can be provided by that fabric. More frequenty, however, the incorporation of additional storage within the enivonmental control system can be beneficial. The thermal store is also a necessary component of a real solar powered environmental control system. Solar thermal energy is variable and transient. A thermal store, supported by an auxiliary power source, minimises the fluctuations in thermal energy supply for this type of environmental control system.

A thermal storage has inherent deficiencies. The primary deficiency is the energy loss from the store to the surrounding environment. The magnitude of this loss is dependent upon the temperature differentials between the stored fluid and the surrounding air, the areas of the exposed surfaces and the composition of the membranes which separate the stored fluid from the surrounding air.

To minimise such losses an integrated energy exchange and storage system has been devised. Air tunnels around the storage vessels have been incorporated in which part of the losses can be recovered. This recovered thermal energy can then be utilised for the thermal treatment of the air supplied to the built environment. Moisture transfer can also be accommodated within the integrated system.

Careful design of the geometry and materials of the integrated heat and mass exchanger/store yields other beneficial thermodynamic features;

- the horizontal surfaces of stored fluid can be exposed to the air flow for humidification or dehumidification processes;

- the vertical and the horizontal surfaces of the storage walls can be exposed to the air flow for sensible cooling or sensible heating;

  the actual instantaneous thermodynamic load can be satisfied directly by the store rather than by the thermal energy supply source;

- the thermal energy source has only to respond to the requirements of the store which, having a long time constant, allows the energy conversion to be scheduled when ambient conditions are favourable.

This thesis investigates the effectiveness and the performance of such integrated storage vessels with heat and mass exchangers as used in two prototype systems - one in Adelaide, South Australia, and the other in Darwin, Northern Territory.

The locations were selected to obtain test data for two different ranges of climatic conditions - temperate and tropical.

The thesis is divided into the following sections:

SECTION 1	-	Aims and Significance of the Present Study
SECTION 2	-	Literature Survey and Assessment
SECTION 3	-	System Components, Experimental
SECTION 4	-	Storage Capacity and Its Response to Thermodynamic Demand
SECTION 5	-	Analyses of Thermodynamic State of Moist Supply Air and Parasitic Energy Input Requirements
SECTION 6	-	Utilisation of Storage Vessels/Exchangers in a Solar Powered Environmental Control System
SECTION 7	-	Discussion, Conclusions and Future Work
APPENDIX A	-	Economic Appraisal , B - Acknowledgement
REFERENCES		

DATA VOLUME

A supplementary volume containing all raw data and details of the analysis of that data has been lodged with the Department of Mechanical Engineering of the University of Adelaide for Archival purposes. The essential content of the Data Volume has been incorporated into the present thesis. Specifically, the Data Volume includes:

APPENDIX B Test Data

B.1 Magill House

B.2 Darwin Experimental Station

APPENDIX C Project Information

Mathematical Calculations, Models and Computer Programs, Psychrometric charts.

APPENDIX D Sketches and Drawings

SECTION 1

AIMS & SIGNIFICANCE OF THE PRESENT STUDY

INDEX

1.1. RESEARCH OBJECTIVES

Obviously we face a future not only of depleting sources of traditional forms of energy, but also of spiralling costs of such energy. It is acknowledged that a reduction in dependence on traditional forms of energy must be achieved within a feasible economic framework. As eighty per cent of electricity generated in South Australia is derived from natural gas and virtually all the electricity generated in the Northern Territory is derived from oil, reduction in the use of electricity in these areas is consequentially of great importance.

Under the impact of this realisation, there has been accelerated research and development directed towards means of reducing demand for energy derived from fossil fuels. A major part of this effort is directed towards the design of buildings and of their environmental control systems. The significant parts of this work are reviewed in Section 2.

THE MAIN AIM OF THE PRESENT RESEARCH STUDY IS TO DESIGN AND TO DEVELOP WITHIN A FEASIBLE ECONOMIC NETWORK NOVEL ENVIRONMENTAL CONTROL SYSTEMS WHICH REQUIRE COMPARATIVELY LOW AMOUNTS OF TRADITIONAL FORMS OF ENERGY, SUCH AS ELECTRICITY, FOR THEIR OPERATION.

1.2 SPECIFIC AIMS

The main purpose of these research projects is to design and to develop integrated energy storage vessels with heat and mass exchangers which fulfil the following requirements:

- Reduce fan, pump, condensing unit and boiler power requirements;

- Reduce load imposed by the environmental control systems on electricity generating plant during periods of peak demand;

- Provide heat transfer facilities between the necessary mass of warmer fresh air, or fresh air/return air mixture, and the necessary mass of colder air leaving the storage vessels/exchangers by establishing the transfer surfaces between these two fluid streams whenever sensible heat/total heat ratio is low.

The proposed system also offers simple means by which most suitable low grade energy, such as solar thermal energy, can be utilised.

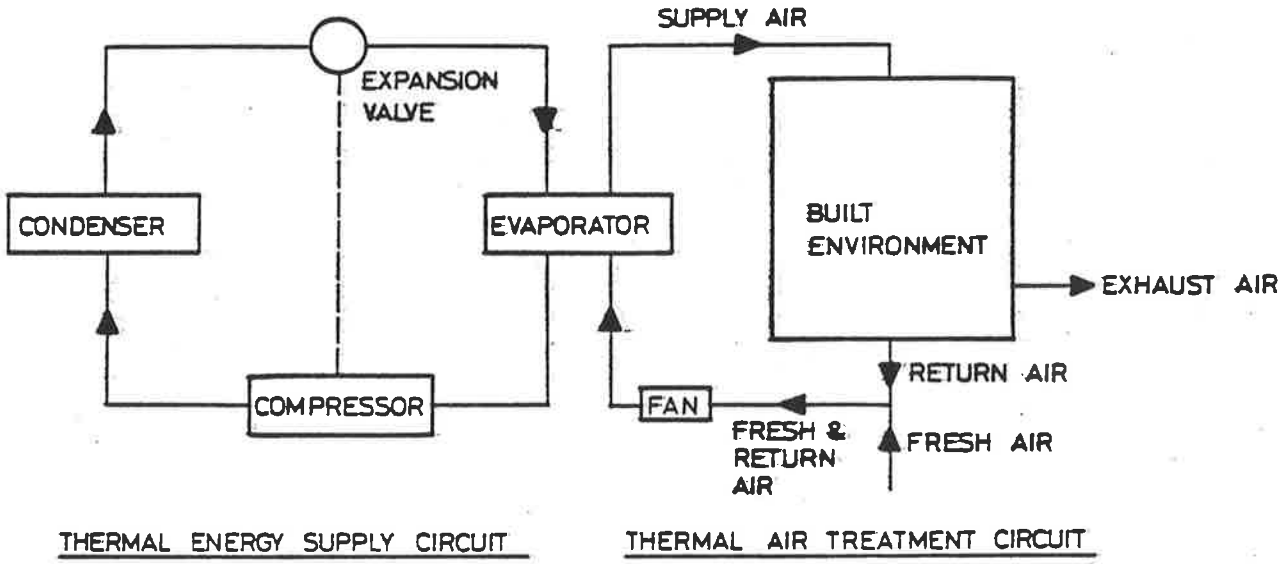
1.1.3 BUILT ENVIRONMENT

1.3.1 General

Any environmental control system involves the automatic control of an atmospheric environment and must operate within prescribed tolerance ranges to provide a comfortable and/or a safe environment for humans, animals and/or plants. Sometimes these systems are required to provide an appropriate environment for high performance industrial processes or for scientific activities.

The purity, movement, temperature and the relative humidity of the air for the built environment must be controlled within the limits proposed by the various Design Authorities.

Fig. 1
A conventional insulated
Environmental Control System



A conventional insulated environmental control system consists of the following major components:

- Thermal air treatment circuit including mass transfer of air into and out of environment; and
- Thermal energy supply circuit.

In the thermal and vapour treatment circuit of air, the mixture of fresh air and return air is supplied with sufficient cooling (or heating) to offset the load of the built environment.

In the heating and cooling supply circuits, energy conversion devices convert the traditional forms of energy into thermal energy and this energy is supplied to the thermal and vapour treatment components to control the thermodynamic state of the moist air supplied to the built environment.

The energy balance in this system requires the rejection of energy by the condenser to the surrounding environment to be equal to the summation of the built environmental load and some of the input of the traditional forms of energy to the heating and cooling supply circuits.

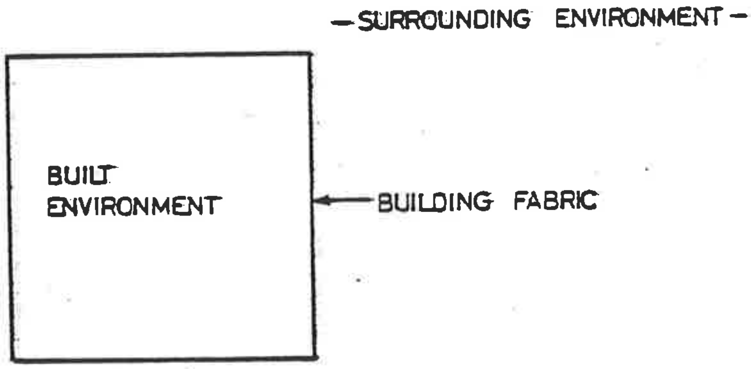
To attain, and to maintain the predetermined energy levels of a built environment, the expenditure of traditional forms of energy is dependent upon:

- Design criteria,
- Thermal energy supply,
- Thermal air treatment systems,
- The storage capacity of the walls, roof and floor.

1.3.2 Design Criteria and Building Fabric Storage Factors

A built environment is isolated from the surrounding environment by the building fabric which act as thermal and vapour barriers. Figure 2 represents such a concept.

Fig.2
Isolation of built environment



The isolation of built environment from the variable and transient surrounding energy levels involves:

- orientation and fenestration techniques applies to the construction of the separating building fabric,

- selection of the building fabric materials,

- surrounding energy levels and their variable and transient characteristics,
- internal load generated by occupants and equipment,
- selection of thermal air treatment systems,
- operating time and subsequent pull-down load,
- capital, operating and maintenance costs.

All these criteria influence the expenditure of traditional forms of energy for maintaining acceptable conditions in a built environment.

Energy flows through the building fabric in the direction of decreasing energy potential. The energy transfer to or from the built environment depends upon energy levels of the surrounding environment. The surrounding environment energy levels are variable and transient and thus an unsteady state of energy movement through the layers of the building fabric results.

The unsteady state of energy movement through the layers of building fabric creates the following major energy parameters:

- Building fabric storage factors

  Storage of energy commences within the thickness of the building fabric before the external energy levels can cause any change in the internal energy levels. The degree of storage is primarily dependent upon the specific heat and the density of the material selected for the building fabric.

. Equivalent temperature differences (Binder 1910), Schmidt (1924))

The equivalent temperature difference means that steady state temperature difference results in the same net energy flow through the building fabric as the unsteady flow caused by the variable and transient external energy levels during a defined time segment.

. Temperature swing

A temperature swing that can be tolerated by the occupants of the built environment is a necessary element which determine the levels of energy expenditure.

The expenditure of traditional forms of energy is not only dependent upon the energy parameters referred to above but also on the following time elements:

. Operating hours

Hours of operation of the input sources of the traditional forms of energy is another necessary element of energy parameters.

. Time Lag

External energy levels require a period of time before it brings any change in internal energy levels. this time lag is caused by the thermal diffusivities of the materials of the building fabric.

The design elements and the construction methods employed for the building fabric influence the magnitude of the built environmental load. The types of construction can be described as light-weight structure, medium-weight structure or heavy-weight structure.

A reduction can be predicted for the energy flow to or from the built environment due to the storage factors, temperature swing and the operating hours. The actual peak value of the load Lp can be expressed by:

$$Lp = L_{EQ} + L_O - L_R \qquad (1.3.2.1)$$

where

L_{EQ} = Energy flow to or from the built environment which is contingent upon the equivalent temperature difference,

L_O = other energy loads,

L_R = a reduction in the energy flow caused by the building fabric storage factor, temperature swing and the operating hours.

1.3.3 **Thermal Energy Supply**

For all-year-round operation, in the majority of the large buildings, an energy conversion device such as a refrigeration compressor operates on a vapour compression cycle or an absorption cycle. Both cycles have in common the evaporation and the condensation of a refrigerant and require an external source of traditional form of energy for operation.

As the condensing pressure against which the compressor has to deliver gaseous refrigerant rises, more mechanical and external energy is expended per kilowatt of refrigeration for the vapour compression cycle. On the other hand, as the suction pressure at the inlet to the cylinders of the compressor rises, less mechanical work needs to be done to secure the same refrigerating effect. Furthermore, with an increase in the suction pressure, the density of the gas entering the cylinders also increases and so, for a given swept volume, the mass of refrigerant handled by the compressor becomes greater. Both the compressor of a vapour compression refrigeration cycle and the absorber/generator of an absorption cycle also attain higher refrigerating effects for a given mechanical and external energy input when the energy is rejected easily by the condenser due to the existence of favourable differences of energy levels between the condenser and the surrounding environment.

For a conventional environmental control system, the energy conversion device must satisfy the peak load Lp. Under this condition, the maximum thermal energy supply Q_{TM} by the energy conversion device can be related to the peak load by the following expression:

$$Lp = Q_{TM} \qquad (1.3.3.1)$$

When the appropriate design parameters are selected, the efficiency of an energy conversion device is highest at full load conditions and this efficiency drops sharply when the actual load declines due to comparative increase in mechanical and friction losses.

1.3.4 **Thermal Air Treatment Systems**

The aim of thermal and vapour treatment systems of moist air is to maintain the thermodynamic state of the moist air supplied to the built environment

The main unit operations of thermal and vapour treatment systems can be categorised as:

1.	Dehumidification	(Mass Transfer)
2.	Humidification	(Mass Transfer)
3.	Sensible Cooling	(Heat Transfer)
4.	Sensible Heating	(Heat Transfer)
5.	Re-heat	(Heat Transfer)

The unit operations involve exchange of energy, momentum and matter within the boundaries of the air saturation profile.

In a conventional environmental control system, including a commercially available solar airconditioning unit, the thermal air treatments are performed as follows:

FUNCTION	SYSTEM COMPONENTS
Dehumidification and sensible cooling	Cooling coil or air washer
Humidification and sensible cooling	Evaporative cooler or air washer
Sensible heating	Heating coil or electric/gas heater
Humidification during winter months	Humidifier
Re-heat after overcooling	Heating coil or electric/gas heater

These treatments are each handled separately as heat and mass transfer processes by components within precisely defined operating limits. The total process of a particular system may involve most of

The load L can be divided into two energy components; sensible and latent.

Sensible energy exchange relates to the heat transfer and the mass transfer causes the latent energy exchange. The total energy exchange is the summation of these two components. Therefore the load L at any particular time can be expressed by:

$$L = q_s + q_L \qquad (1.3.4.1)$$

where

q_s = sensible energy component,
q_L = latent energy component.

The ratio of the sensible energy to the total energy influences the performance of the thermal air treatment system components.

1.4 DESIGN AND DEVELOPMENT

1.4.1 General

In designing the system of integrated storage vessels/heat and mass exchangers, cognizance has been taken of the physical observations as described in this sub-section.

1.4.2 Storage capacities of the building fabric

The arrangement of different layers of building fabric and the construction methods employed determine the following load characteristics:

- Peak value of the load, Lp, (Equation 1.3.2.1)

- A reduction in the energy flow caused by the building fabric storage factor, L_R. (Equation 1.3.2.1)

The peak values of load for different classes of structures enclosing the same built environment are:

$$L_{pl} > L_{pm} > L_{ph} \qquad [1.4.2.1)$$

where

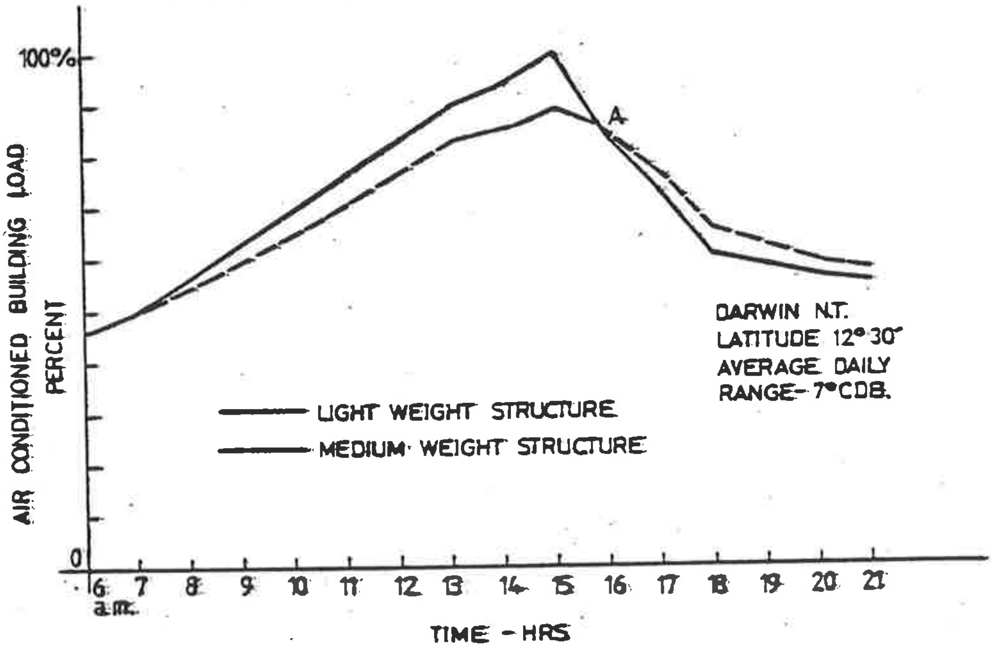
L_{pl} = Peak load value for a light-weight structure

L_{pm} = Peak load value for a medium-weight structure

L_{ph} = Peak load value for a heavy-weight structure.

Figure 3 illustrates the load profiles for two classes of structures - light-weight and medium-weight enclosing the same environment.

Fig. 3
Load profiles, light and medium-weight structures



As can be seen from these experimental data the peak load for the light-weight structure is greater than that of the medium-weight structure.

After the peak load occurs and when the solar load declines, the load for the medium-weight structure becomes greater than that of the light-weight structure after the point A. These conditions of higher loads for a medium-weight structure occur in Darwin because of its comparatively small average daily temperature range.

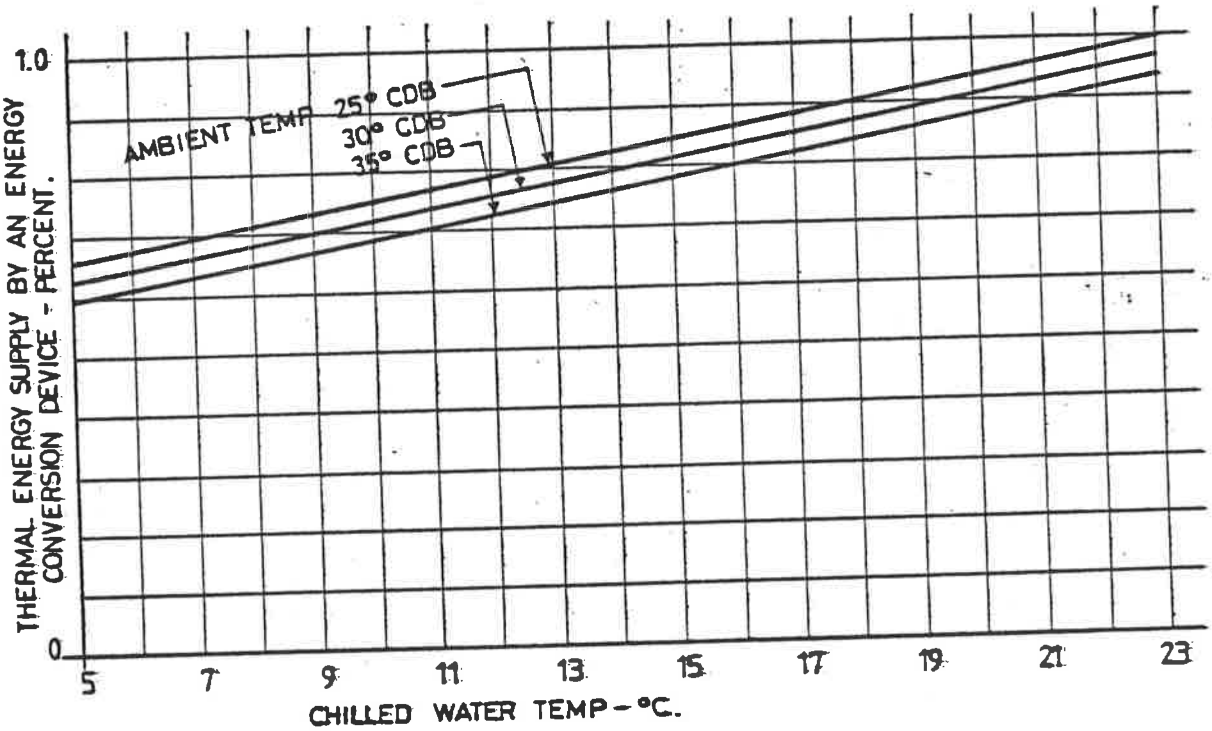
In a conventional environmental control system, the equipment capacities are equal to the peak values, whereas by the introduction of an energy store as an integrated storage vessel/exchanger system the equipment capacities are reduced below those which would be required to satisfy the actual peak load values.

1.4.3 **Favourable Operating Time Segments**

The co-efficient of performance of an energy conversion device such as a refrigeration compressor with air cooled condensers is dependent upon its capacity to reject energy to the surrounding environment. Higher refrigerating effects with comparatively lower input of traditional form of energy such as electricity are attained when there are large favourable energy driving forces available between the condenser and the surrounding environment.

Figure 4 indicates such favourable operating time segments.

Fig. 4
Thermal energy supply by an energy conversion device at different surrounding environment energy levels

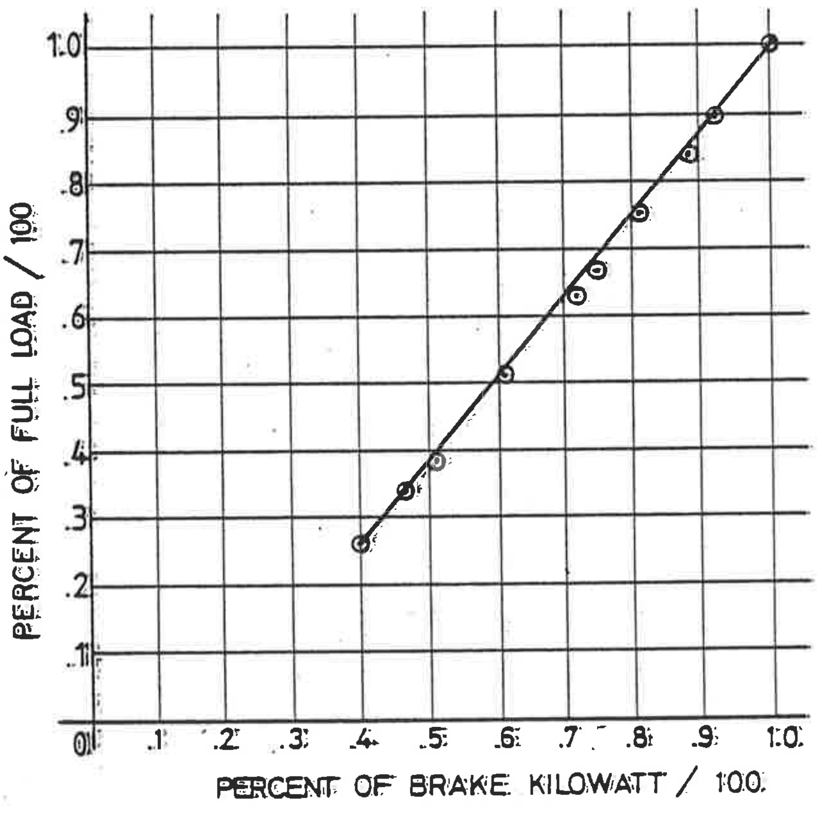


These experimental data illustrate that the co-efficient of performance of an energy conversion device is higher when the ambient energy levels are favourable. Generally, favourable ambient energy levels exist in the morning and in the evening which is before and after the occurrence of the peak load. The thermal energy is collected and this energy is stored by the integrated storage vessels/exchangers when the co-efficients of performance of an energy conversion device are comparatively higher.

1.4.4 Part Load Operations of an Energy Conversion Device

When an energy conversion device is selected to satisfy the peak load, this device operates at part load conditions most of the time as the peak load occurrence is infrequent. At part load conditions, an energy conversion device such as a reciprocating refrigeration compressor unloads its cylinders to adjust its thermal energy supply capacity to the instantaneous load requirement of a built environment. Figure 5 represents the percentage of brake kilowatt against the percentage of full load of such a compressor responding to various load requirements.

Fig. 5
Load - brake kilowatt curve

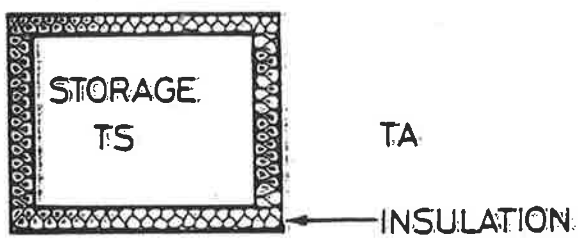


As can be seen from these experimental data the friction and the mechanical losses at part load conditions are comparatively higher than those at the full load conditions. The thermodynamic efficiency of a refrigeration compressor at full load operating conditions is greater than that at part load operating conditions. To reduce energy consumption in the systems described here, a lower capacity energy conversion device has been matched to an integrated storage vessel/exchanger system so that it operates for longer periods at full load than does a higher capacity energy conversion device in a conventional environmental control system. This full load operating time of a lower capacity energy conversion device further allows the use to be made of the time segments when there are large favourable energy driving forces available between the condenser and the surrounding environment.

1.4.5 "Storage Loss"

In a conventional environmental control system where storage becomes a necessary requirement, as in the case of a real solar powered environmental control system, the storage is an insulated vessel.

Fig. 6
Conventional storage

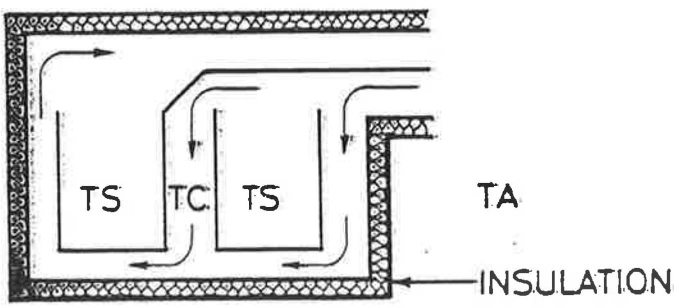


If T_S is the average storage fluid temperature and T_A is the average surrounding ambient air temperature, then the loss from the storage is determined by the mean temperature difference $(T_S - T_A)$, the exposed surface area of the vessel and the thermal and the vapour transmission characteristics of the membranes separating the stored fluid from the surrounding environment.

In a conventional environmental control system, heat and mass transfer equipment such as cooling coils, heating coils, humidifiers and air washers, are also enclosed by an insulated chamber. The loss from these chambers depends upon the corresponding temperature differences, the exposed surface areas of the chambers and the thermal and the vapour transmission characteristics of the insulated chamber casing which separates the heat and mass transfer equipment from the surrounding environment.

In the energy storage vessel/exchanger system, the heat and mass transfer facilities have been incorporated within the storage system. Figure 7 illustrates such facilities.

Fig. 7
Energy storage vessels/exchangers



T_C is the mean temperature of the air leaving the energy storage vessels/exchangers. T_C is always within the range $T_S > T_C > T_A$ for the heating mode and within the range of $T_S < T_C < T_A$ for the cooling mode. The mean temperature difference $(T_S - T_A)$ for an ordinary storage system is always greater than $(T_C - T_A)$ for the energy storage vessels and exchangers.

Though the exposed surface areas of the storage vessels/exchangers have been increased by the introduction of air tunnels around the vessels, the total loss is small because the temperature differential (T_C - T_A) is small. The thermal and the vapour transmission characteristics of the insulated chamber casing for a normal store and an energy storage vessel/exchanger system remain the same.

1.4.6 Momentum, energy and matter transfer

Convection from a solid surface to a surrounding fluid is limited by the area of that surface. In an ordinary cooling coil, the surface area is extended by the addition of fins. Heat transfer is then by conduction along the fin and by convection to the surrounding air from the surface of the fins. Mass transfer is dependent on the temperature of the transfer metal surface with respect to the dewpoint of the surrounding air. The total energy transfer in this coil concept is maximised by increasing the convective transfer co-efficients through increasing the local velocity of fluid over the transfer surface. For a given total mass flow and fixed transfer surface area, this implies an increase in the number of rows of coils. When the number of rows is increased, the effectiveness of downstream rows decreases as the driving force for the transfer process decreases with each successive row.

In these experimental projects, the secondary transfer surfaces in the forms of fins are totally eliminated. This gives a more uniform temperature distribution on the primary transfer surfaces. The primary surface areas have been increased to compensate for the loss of secondary surface. The viscous shear forces for the fluid streams are reduced by minimising the magnitude of local velocities. Residence time is increased by the continued effect of larger primary surface area and reduced flow velocities. The residence time can be defined as the time of contact between the two fluids - air and thermal agent for direct contact and air and separating wall for indirect contact. Thus at the expense of having a less compact heat and mass exchanger, the power consumption necessary to achieve a given exchange can be minimised.

1.4.7 **Re-heat**

Where the moisture removal from the necessary mass of selected air is a significant requirement as in the tropical areas, room sensible heat/total heat ratio lines may not intersect the air saturation curve. In these cases, sensible heat needs to be added to the colder air stream leaving a cooling coil or an air washer.

In a conventional environmental control system, this sensible heat is supplied by heating coils or electric/gas heaters at the expense of additional input of traditional forms of energy.

The dry bulb temperatures of the fresh air or fresh air/return air mixture stream are higher than those of the colder air stream leaving a cooling coil or an air washer.

In energy storage vessels/exchangers, heat transfer surfaces have been incorporated as shown in Figure 7 to transfer heat from the warmer fresh air or fresh air/return air mixture stream to the thermally treated colder air stream.

1.5 **Research, Design and Development**

The design elements referred to above are included in the development of energy storage vessels and heat and mass exchangers to reduce the expenditure of

The other objectives that can be met are:

- Electrical 'J-tariff' can be utilised to store energy during off-peak period, and
- the capacity of the associated electrical sub-station can be reduced.

SECTION 2

LITERATURE SURVEY AND ASSESSMENT

INDEX

SECTION 2: LITERATURE SURVEY AND ASSESSMENT

2.1 INTRODUCTION

Extensive research and development has been directed towards reducing energy used in buildings and many demonstration projects are being monitored to establish the viability of such developments.

These systems and their performances are described in numerous publications. Many reliable computer programmes are available for the prediction of their behaviour and a number of studies have examined their economic viability.

Thermal energy storage either in the form of solid or in the form of fluid forms an essential element of solar powered environmental control systems. Thermal energy storage within the building fabrics which separate the built environment from the external environment also plays a major role in the design of the buildings and of their environments.

2.2 BUILDING FABRIC AND ASSOCIATED ENVIRONMENTAL CONTROL LOADS

Szokolay (1981) indicates that the interior temperature can be kept slightly above the exterior temperature but could never depress it below that of the exterior in a light-weight building with passive cooling system and adequate ventilation. He suggests that these conditions can be met by providing:

- shading of windows;
- use of light coloured (reflective) surfaces for walls and roof;
- insulation of roof;
- shading of walls,
  and
- insulation of walls

for a light-weight building. During summer when the exterior temperatures are well above the recommended comfortable levels, further thermal and vapour treatment becomes necessary for the moist air supplied to the built environment.

Szokolay (1981) indicates that the heavy-weight or mass buildings with long time constant must be socially and economically acceptable. Szokolay points out that at present the technological problem is that the heavy-weight buildings tend to be more expensive than the light-weight ones, particularly where transportation of materials over long distances is a major part of the cost. His suggestion is to explore the possibilities of locally available materials and to commercialise resulting techniques.

The heavy-weight buildings whose peak loads are comparatively lower tend to be more expensive than the light-weight ones particularly in Northern Australia where transportation of materials over long distances is a major part of the cost. Also in these areas daily temperature ranges are comparatively lower. Energy storage for the thermal and vapour treatment of moist air is included in the design presented in this thesis. The purpose of this inclusion is to reduce the size of the energy conversion devices for light-weight buildings.

2.3 THERMAL ENERGY STORAGE

Thermal energy storage forms an essential element of solar powered environmental control systems. In these systems the storage is considered in the light of a solar process system, the major elements of which are: the solar collectors, storage units, conversion devices, load profiles, auxiliary or supplemental energy supplies and the control systems. Duffie and Beckman (1974) point out that three major factors determine the optimum capacity of storage systems for buildings. First is the cost of the storage unit and of the storage medium. Second is the space in which the storage unit is located and third is the cost of operating such a storage unit. Their observation is that the short-time storage to meet loads for shorter periods has most economical advantage in building applications.

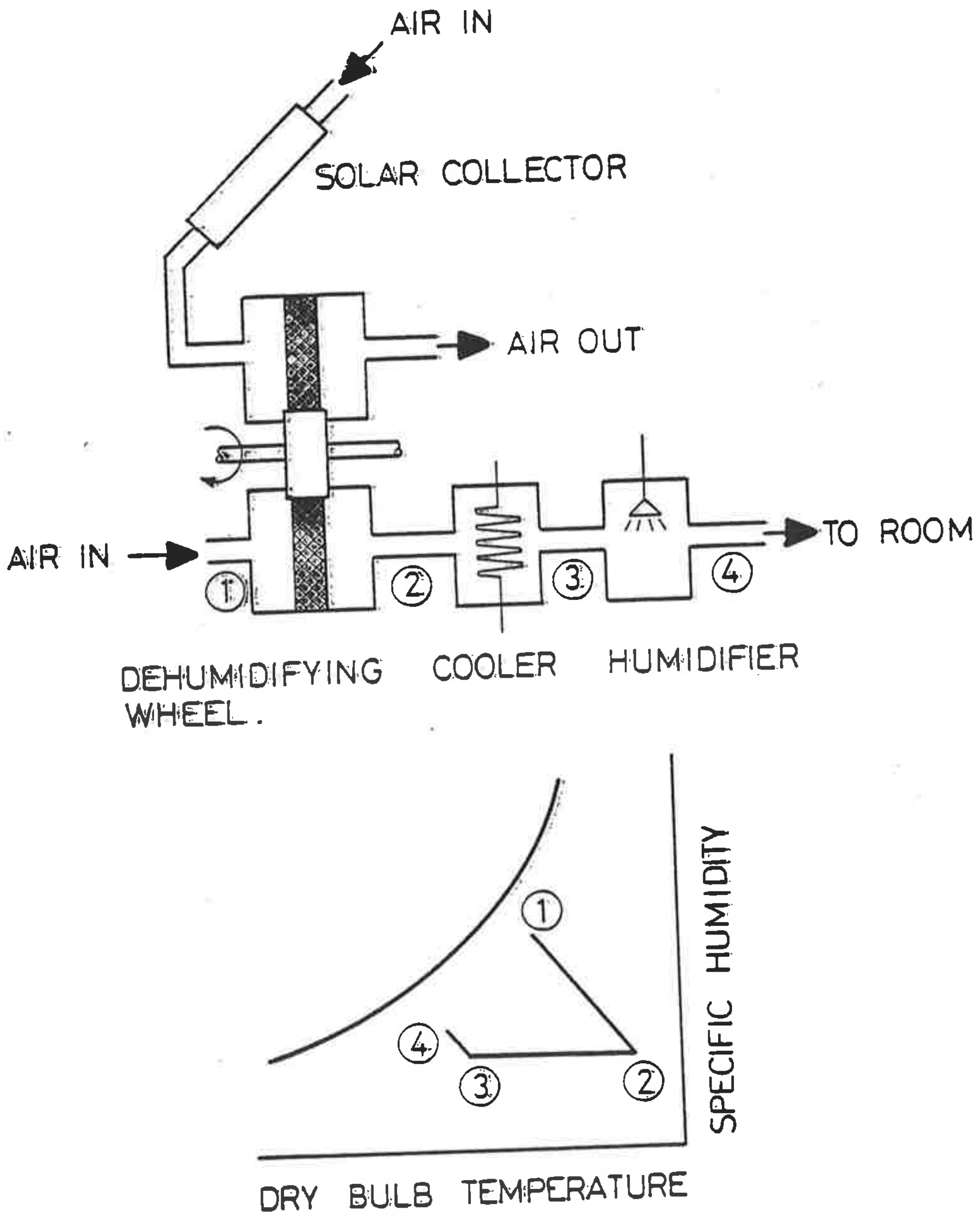
Heat will be transferred through the walls of any storage unit at a rate depending on the temperature difference between the storage media and the surroundings. The total of such energy losses is also a function of the time. If energy storage is to be considered for long periods, thermal losses may become of critical importance. Speyer (1959) who, like others, concluded that for house heating, long-term storage is economically impractical. Duffie and Beckman (1974) suggest that the storage unit may be located within the space to which heat is being added. In this situation, losses are reduced but are an uncontrolled energy transfer from storage to the space to be heated. They indicate that the situation is more critical in house cooling systems if storage losses add to the cooling load.

Energy storage included in the design of vessel/exchanger system is to satisfy the peak load, duration of which is short. Air tunnels are built around these storage vessels to reduce the thermal losses from the store. Storage medium is water as water is an inexpensive, readily available, and useful fluid in which to store sensible heat. The operating power requirements have been kept-to minimum for the vessel/exchanger system by allowing the pump to operate against the height of the fluid within the storage vessels, the friction of the heat exchanger section of a chiller or a boiler and the friction of the minimal pipework connecting the storage vessels to a chiller or a boiler.

2.4 HEAT, MASS AND MOMENTUM TRANSFER FOR THE THERMAL AND VAPOUR TREATMENT OF MOIST AIR

Dehumidification and cooling by absorption together with adsorption systems came into use a long time ago. The earliest practical proposal for a dehumidification process was by Lof (1955). He suggested the use of tetraethylene glycol as a dehydrating agent. The testing of components and studies on adsorption and absorption systems have been done by Hollands, 1963; Dunkle, 1976; Close and Pryer, 1976; Close and Dunkle, 1976; Rush, 1976; Lunde, 1976; Techerner, 1976; Meunior, 1979. Besides tetraethylene glycol, the use of silicagel and zeolite are now being investigated.

Fig. 8
Principle of Open Cycle
Cooling by Chinnappa



The dehumidification process developed by Chinnappa (1981) is illustrated in Fig. 8. Ambient air which is hot and humid is passed through the matrix of the dehumidifying wheel packed with desiccant (molecular sieve or silicagel). The air is heated and then dried along the line 1-2 as it passes through this wheel. It is then cooled along the line 2-3 and finally humidified along the line 3-4 to achieve comfort conditions.

The psychrometric chart illustrates the principles of such a thermal and vapour treatment system for moist air.

Shaw (1980) investigated the influence of Reynolds Number and air velocity on dehumidifier performance. He indicated that in order to select the efficient dehumidifier configuration, it is necessary to analyse the total thermodynamic requirements comprising:

.a sensible heat ratio,
.a fixed temperature difference,
and .a fixed mass flow rate.

These are the factors that determine the correct Reynolds Number to provide a coil condition curve that operates with minimum consumption of energy. He revealed that, at the possible expense of larger surface area of the dehumidifier, the advantages that can be gained, are:

.no overcooling,
.no re-heating,
.reduced fan power,
.smaller chillers,
and .smaller cooling towers.

Another particular example is the evaporative cooling system known as the rock bed regenerative cooler, developed by the CSIRO Division of Mechanical Engineering (1966). In this process, two chambers filled with crushed stones are operated alternatively. One chamber is sprayed and evaporatively cooled by the exhaust air while the other is meanwhile cooling the supply air.

Evaporative cooling systems are economic but their application is limited to the arid areas only. These systems have very little use in humid areas.

Shaw's development by utilising the increased residence time to attain increased dehumidifier performance reduces the energy expenditure for both sources - traditional forms of energy input sources for thermal energy supply, as well as for thermal and vapour treatment systems of moist air.

Chinnappa (1981) predicts that the dehumidification and cooling by absorption and adsorption systems may have some economic merit over the state-of-the-art solar cooling systems but there is an obvious hiatus in available statistics on realiability, viability and performance of habitable buildings cooled by absorption and adsorption systems.

Generally, the storage vessels have larger external surface areas and the total lengths of these external surfaces are considerably long. In the design presented in this thesis, these surfaces have been utilised to transfer heat and mass between the stored fluid and the moist air, supplied to the built environment. Residence time is increased by the continued effect of these larger transfer surface areas and reduced flow velocities. At the possible expense of having a less compact heat and mass exchanger, the power consumption necessary to achieve a given exchange for the vessel/exchanger system is minimised.

SECTION 3

SYSTEM COMPONENTS, EXPERIMENTAL

INDEX

SECTION 3: SYSTEM COMPONENTS, EXPERIMENTAL

3.1 SYSTEM AND SYSTEM COMPONENTS

An energy conversion device forms an essential element of the integrated energy storage vessel/heat and mass exchanger system.

The other elements are pump, fan, ductwork, pipework and automatic controls.

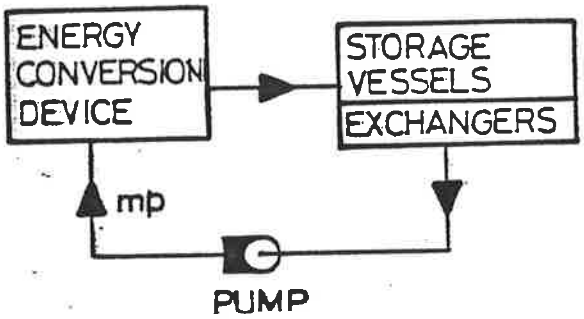
The energy conversion devices are summarised in the following table:

THERMAL AND TREATMENT PROCESSES	ENERGY CONVERSION DEVICES
Sensible cooling and dehumidification	Vapour compression or absorption chiller
Sensible heating and dehumidifiction	Reverse cycle vapour compression chiller or boiler or solar hot water system.

3.2 SYSTEM DESCRIPTION

Storage vessels are an integral part of the system. The stored fluid is either water or another selected non-toxic fluid mixed with the water. The system components for water circulation are as indicated in Figure 9.

Fig. 9
Water circuit



A pump transports thermal energy to the storage vessels/exchangers. When the energy level of the storage arrives at a predetermined value, a water temperature controller de-energises first the energy conversion device, then the pump. When the energy is required for the store, the same water temperature controller energises the pump first, then the energy conversion device.

The air is returned from the controlled environment and then this return air is mixed with the fresh air before the air mixture enters the storage vessels/exchangers as shown in Figure 10. The metal surfaces MSS and the exposed water surfaces WSI and WSO provide the heat and mass transfer facilities.

The temperature of the upper water layer is kept below the dew point of the entering air when the thermal air treatment required is sensible cooling and dehumidification.

The temperature of the upper water layer is kept above the dry bulb temperature of the entering air when the thermal air treatment required is sensible heating and humidification.

The temperature of the upper water layer is kept above the dew point of the entering air but below the dry bulb temperature of that air when the thermal air treatment required is sensible cooling and humidification.

The temperature of the lower water layer is always below the temperature of the upper water layer due to the stratification over the vertical dimension of the storage.

Fig. 10
Directional air flow for
thermal air treatment processes

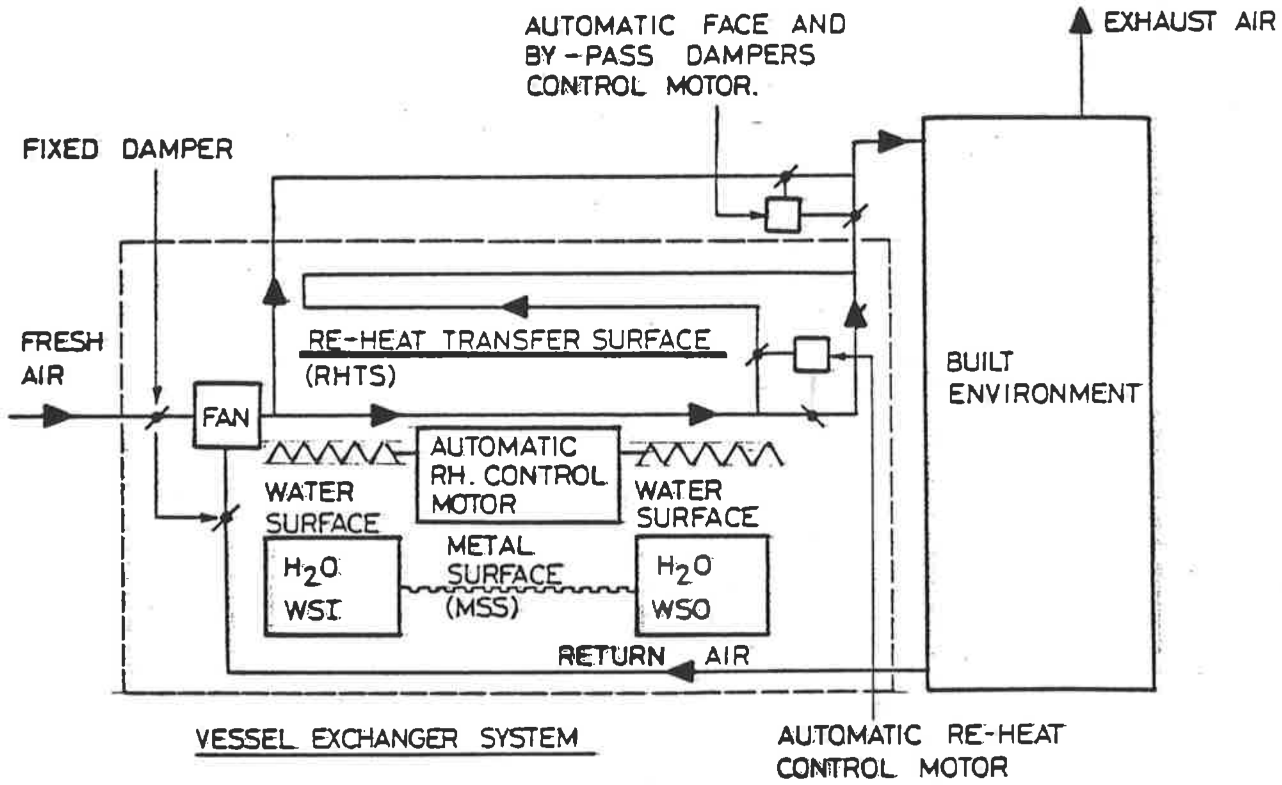
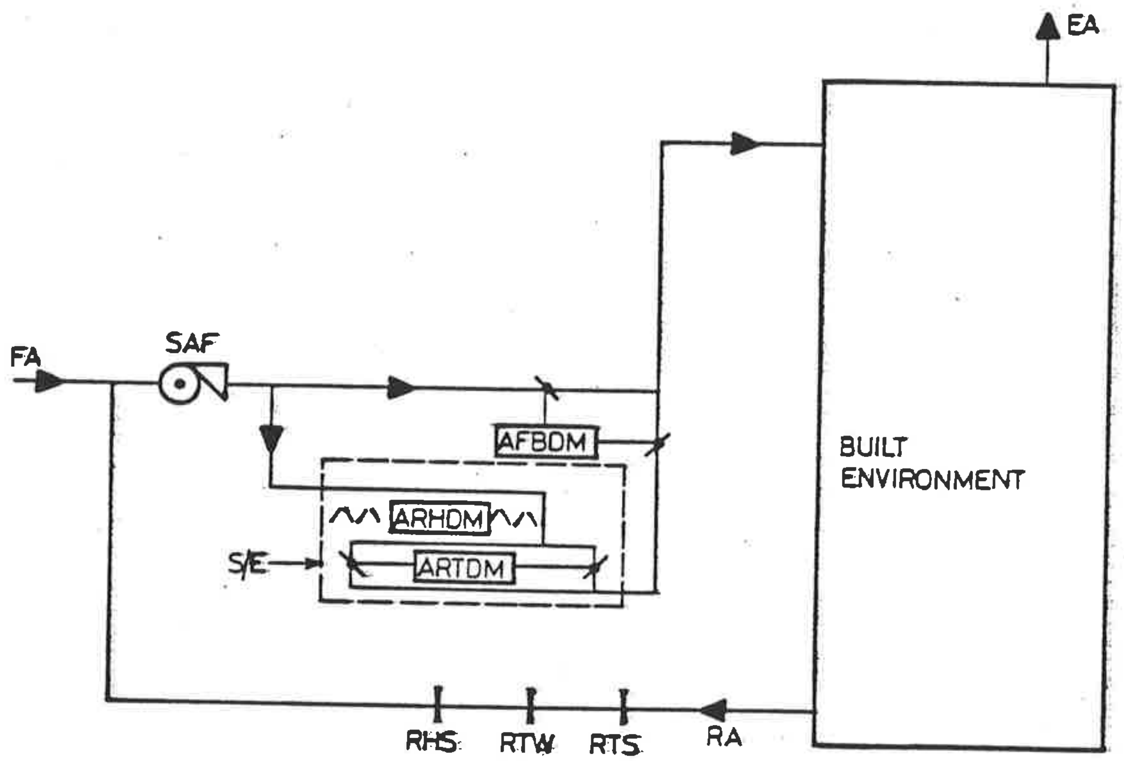


Figure 10 indicates the direction of air flow incorporated within the energy storage vessels/exchangers for the thermal air treatment processes.

Figure 11 indicates the automatic damper systems incorporated within the energy storage vessels/exchangers to maintain the built environmental conditions within the prescribed range of tolerances.

Fig. 11

Automatic damper systems



FA - Fresh air
SA - Supply air
RA - Return air
EA - Exhaust air
SAF - Supply air fan
S/E - Storage vessels/exchangers
RHTS - Heat transfer surfaces for re-heat (Figure 10)
RTS - Return air dry bulb temperature sensor for summer
RTW - Return air dry bulb temperature sensor for winter
RHS - Return air relative humidity sensor
AFBDM - Automatic face and by-pass damper motor
ARTDM - Automatic re-heat damper motor
ARHDM - Automatic relative humidity control damper motor for winter

3.3 AUTOMATIC CONTROL FUNCTIONS

Summer and winter operation

Two return air dry bulb temperature sensors are set to operate in the following manner:

Sensor RTS for summer	25°CDB	$\pm$ 1.5°C
Sensor RTW for winter	20°CDB	$\pm$ 1.5°C

The transfer of mode from heating to cooling or cooling to heating is automatic.

Sensible cooling and dehumidification

When the return air dry bulb temperature sensor for the summer RTS senses the temperature of the air which is above the set point plus the differential, this sensor through a controller modulates face (energy storage vessels /exchangers) damper and bypass (fresh air/return air mixture) damper by the automatic action of the motor AFBDM, proportional to the rise or to the fall of the return air temperature.

The water temperature controller located on the return water line to the energy conversion device, or devices, controls the loading or the unloading stages of that device, or of those devices, directly proportionally to the rise or to the fall of the return water temperature in relation to the set point.

During this mode,

- re-heat damper is fully closed, and
- re-heat bypass damper is fully open, by the automatic action of the motor ARTDM, and
- relative humidity control dampers for winter are fully open by the automatic action of the motor ARHDM.

Sensible cooling, dehumidification and re-heat

During this cooling mode, when the relative humidity sensor RHS located in the return air stream senses the rise above the set point plus the differential, this sensor overrides the action of the air temperature controller and closes the bypass (fresh air/return air mixture) damper and opens the face (energy storage vessels/exchangers) damper by the automatic action of the motor AFBDM.

At the same time this sensor through a controller modulates the re-heat damper and the re-heat bypass damper by the automatic action of the motor ARTDM.

The automatic water temperature controller and the relative humidity controller for this mode operate in the same manner as those of sensible cooling and dehumidification.

Sensible heating and humidification

When the return air dry bulb temperature sensor for the winter RTW senses the drop below the set point minus the differential, this sensor through a controller modulates face (energy storage vessels/exchangers) damper and bypass (fresh air/return air mixture) damper, by the automatic action of the motor AFBDM, directly proportionally to the fall or to the rise of the return air temperature.

During this mode, the re-heat damper is fully closed and the re-heat bypass damper is fully open by the automatic action of the motor ARTDM.

The relative humidity control dampers for the winter are normally closed when the relative humidity of the return air is above the set point plus the differential. When the relative humidity drops below the set point minus the differential, the relative humidity sensor RHS through a controller modulates the relative humidity control dampers by the automatic action of the motor ARHDM directly proportionally to the fall or to the rise of return air relative humidity.

The water temperature controller during the winter controls the loading or the unloading stages of the energy conversion device or devices directly proportionally to the fall or to the rise of the return water temperature in relation to the set point.

Thermodynamic state of the moist air

The automatic damper systems control the mass flow rate of the air through the thermal and the vapour treatment chambers of the energy storage vessels/exchangers.

This treated air when mixed with the proportion of bypassed air achieves the thermodynamic state of the moist air which satisifies the required built environmental load.

The control diagram is included in the section "Sketches and Drawings" of the Appendix.

3.4 TECHNICAL SUMMARY

Magill House

Airconditioned area	103 sq. m
Ventilated area	9 sq. m

Ambient conditions for peak load design:

Summer	36.5°CDB	23°CWB
Winter	3.5°CDB	80% RH

Peak Load (cooling):

Sensible	10,660 watts
Latent	1,168 watts
Total	11,828 watts

Peak Load (heating):

Sensible	11,278 watts
Latent	1,792 watts
Total	13,072 watts

Supply air (design):	680 l/s
Return air (design):	515 l/s
Fresh air (design) :	165 l/s

Electrical energy input sources:

Hermetic refrigeration compressor	2,200 watts
Condenser fan	80 watts
Heating/cooling water pump	100 watts
Control devices (3 off)	72 watts
Supply air fan	460 watts

Storage volume: 2,124 litres

Darwin experimental station

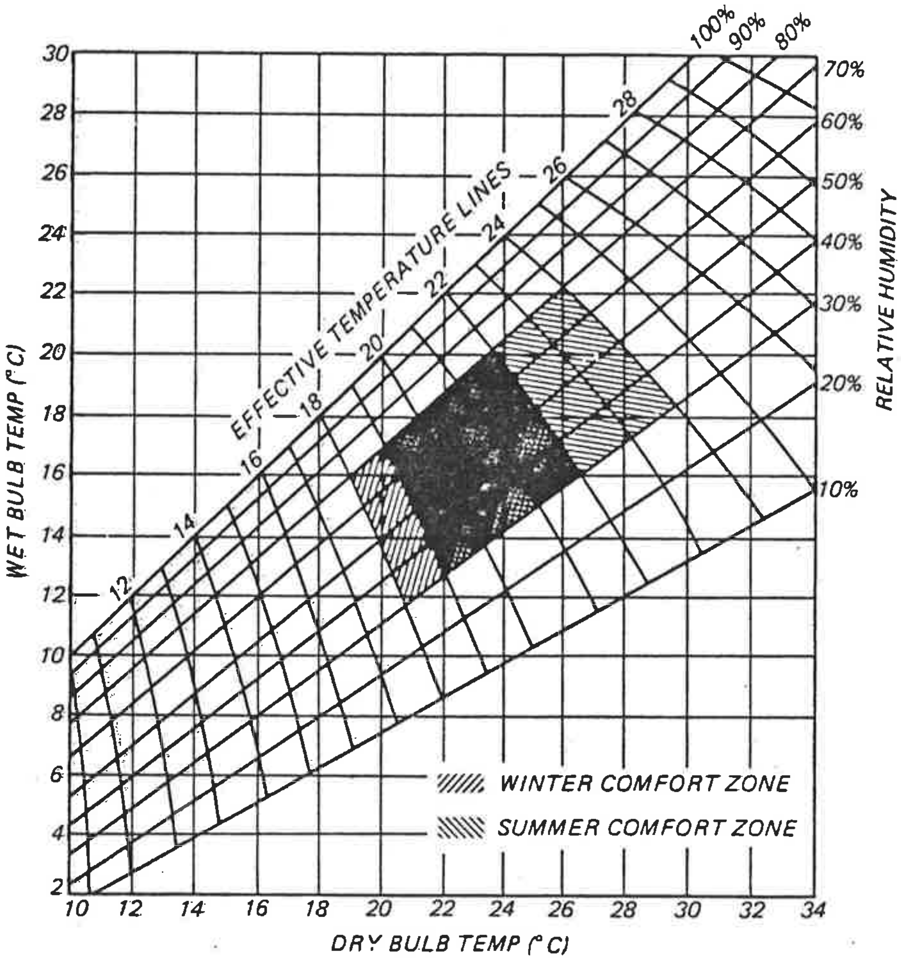
Darwin experimental station is equipped to simulate controlled environmental conditions with the artificial loading provided by the electric heaters and the humidifiers. Electrical energy input sources for the environmental control system are the same as those of Magill House.

Storage volume 1,000 litres

Design indoor conditions:

Relation of effective temperature, to dry and wet bulb temperatures and humidity, with summer and winter comfort zones

Charts for velocities up to 0.1m/s i.e. practically still air.
For an air velocity of 0.4m/s the effective temperature decreases by 1 °C.



As proposed by Porges

SECTION 4

STORAGE CAPACITY

AND

ITS RESPONSE TO THERMODYNAMIC DEMAND

INDEX

SECTION 4: **STORAGE CAPACITY AND ITS RESPONSE TO THERMODYNAMIC DEMAND**

4.1 ENERGY LEVELS

Energy level in the storage is time dependent. The performance of the system is optimised by establishing the following factors:

- the time dependence factor between the energy supply and the energy expenditure;
- the manner in which energy is supplied and the energy is spent;
- the energy losses to the surrounding environment;
- the configuration of vessels to release energy to a load demand;
- an economic analysis based on the above which ultimately determines the magnitude of the storage;
- the degree of reliability needed.

Energy storage is mostly in the form of sensible energy but the contact surface temperatures are maintained within suitable ranges for the required thermal and vapour treatment of moist air. Further by controlling these temperatures latent energy can be added to or can be extracted from the moist air if so required by a particular process.

The storage system operates with significant degrees of stratification over the vertical dimension of the storage, and the temperatures of the stratified fluid layers can be considered as:

Fig. 12
Stratification of Storage System

TSUS, TSUW - average temperature of the upper layers of the storage fluid

TSIS, TSIW - average temperatures of the intermediate layers of the storage fluid

TSLS, TSLW - average temperatures of the lower layers of the storage fluid

Energy can only be removed by the mass air flow from the store when the supply air fan operates. The pump flow rate m_p can be thought of as the actual mass flow rate through an energy conversion device at any time.

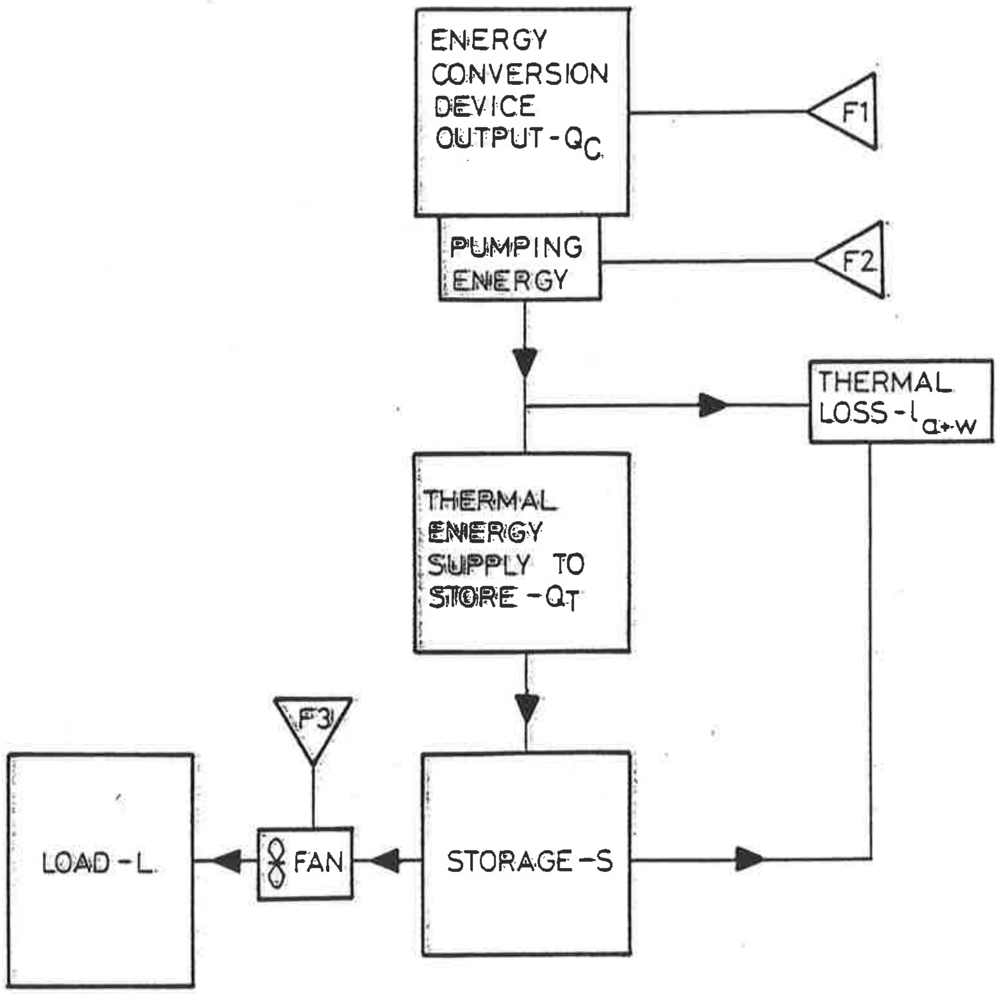
With these interpretations, control functions can be established for the following:

F1 - A control function of an energy conversion device
F2 - A control function of a pump
F3 - A control function of a supply air fan.

Fig. 13 illustrates energy flow diagram for vessel /exchanger system including the control functions referred to above.

Fig. 13

Energy Flow Diagram



F1 = A, B, C, D, E ....
when the energy converstion device operates.

A, B, C, D and E ..... are factors which relate to the actual thermal energy supply by an energy conversion device at any instant with respect to the load demand.

$F1 = 0$ at other times

$F2 = 1$ when the pump operates

$= 0$ at other times

$F3 = 1$ when the supply air fan operates

$= 0$ at other times

4.2 MATHEMATICAL MODELS OF STORED ENERGY

At any given time, the environmental load L lies between two extremities - a peak value and a minimum value.

The energy conversion device supplies the thermal energy QT to the store and

$$Q_C = Q_T + l_{a+w} + P \qquad (4.2.1)$$

where

Q_C = thermal energy supplied by the energy conversion device at any instant

l_{a+w} = piping and storage thermal energy loss

and P = thermal energy added to the store for heating and humidification or thermal energy deducted from the store for cooling and dehumidification by the pump motor.

During part of the time, the thermal energy supply Q_T by the energy conversion device exceeds the load L and at other times it is less. A storage system can be added to store excess of Q_T over L when $Q_T > L$ and expend the energy from the store when $Q_T < L$.

Over a given time period, the differences between the maximum and the minimum of the integral for the load L, the thermal energy supply Q_T and the thermal loss l_{a+w} represent the amount of storage capacity S that would be required to permit the energy supply Q_T to meet all of the loads L and all of the losses l_{a+w}. Hence, the value of

$$[F1 \quad F2 \quad Q_T \; - \; (F_3 L + l_{a+w})] \; dt \qquad (4.2.2.)$$

is a function of time.

The thermal energy stored or expended over a finite period of time can be expressed by the following heat capacity equation:

$$S_{New} = S_{old} + [F1 \quad F2 \quad Q_T \; - \; (F3\ L + l_{a+w})]\ dt \qquad (4.2.3)$$

where the thermal capacity S_{New} of the store at the end of a time interval dt can be calculated from the storage capacity S_{old} at the beginning of that time interval dt.

The thermal energy supply Q_T can be expressed by

$$Q_T = mp \quad Cpw\ (T_e - T_o) \qquad (4.2.4)$$

where

Cpw is the specific heat of the stored fluid, T_e and T_o are the temperatures of the fluid entering and leaving the vessels/exchangers respectively and mp is the mass flow of the fluid through the circuit connecting the energy conversion device to the vessels/exchangers.

Equation (4.2.3) can be re-written as:

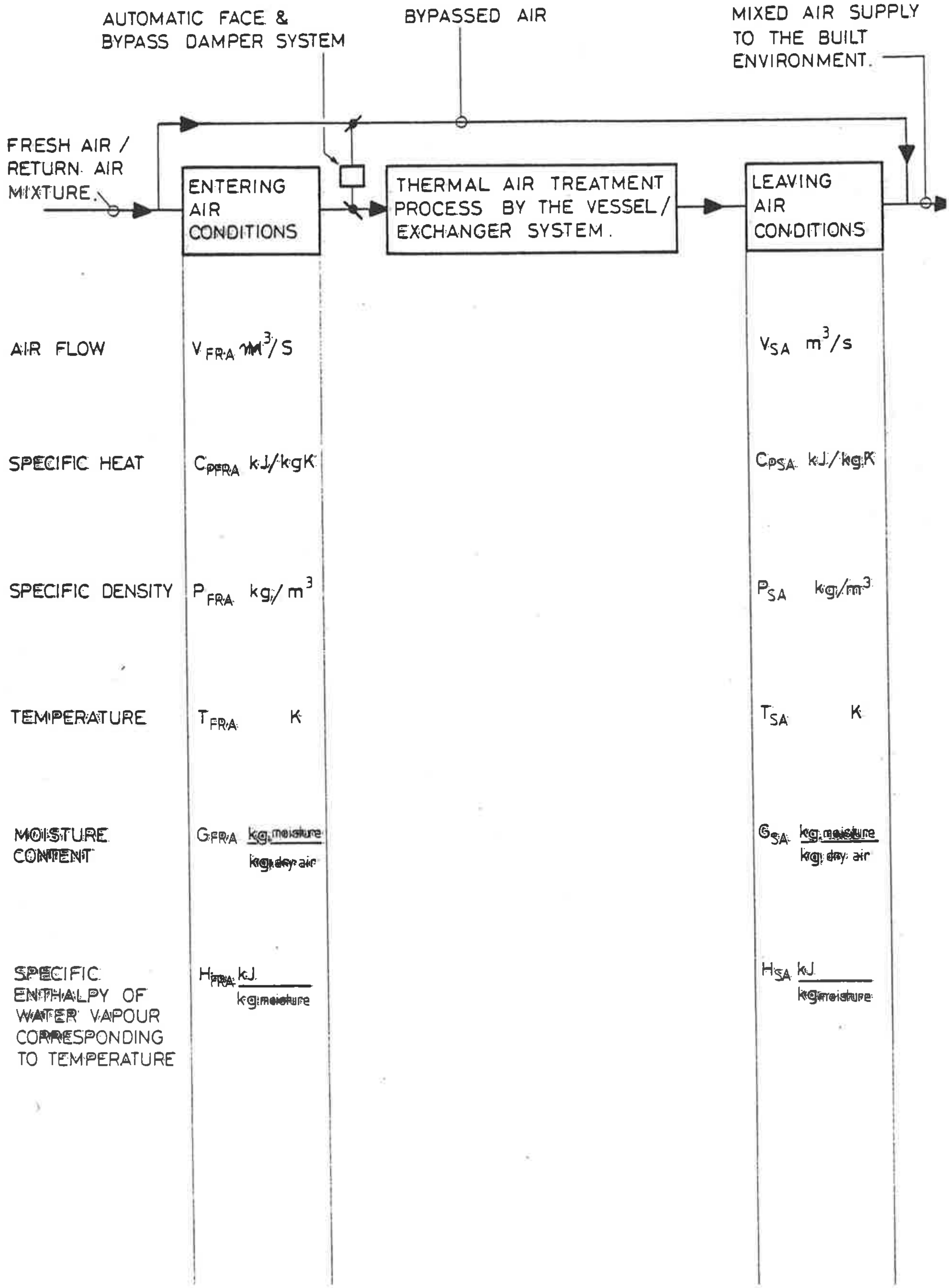
$$S_{New} = S_{old} + [F1 \quad F2 \quad mp \quad Cpw\ (T_e - T_o) - (F3\ L + l_{a+w})]\ dt \qquad (4.2.5)$$

The load L, imposed by the built environment can be assessed from the conditions of the moist air entering and leaving the vessels/exchangers and from the corresponding mass flow of that air. Fig. 14 illustrates such thermal and vapour treatment of moist air by the vessel/exchanger system prior to the moist air being supplied to the built environment.

Fig. 14

Thermal and Vapour

Air Treatment Process



The instantaneous environmental control load L can be established from Fig. 14

$$L = [(V_{FRA}\ C_{pFRA}\ P_{FRA}\ T_{FRA} - V_{SA}\ C_{pSA}\ P_{SA}\ T_{SA}) + (V_{FRA}\ P_{FRA}\ G_{FRA}\ H_{FRA} - V_{SA}\ P_{SA}\ G_{SA}\ H_{SA})] \qquad (4.2.6)$$

The equation (4.2.6) can now be introduced to the equation (4.2.5) to replace L and the final energy balance equation can now be written as:

$$S_{New} = S_{old} + [F1\ F2\ mp\ Cpw\ (T_e - T_o) - [F3\ [(V_{FRA}\ C_{PFRA}\ P_{FRA}\ T_{FRA} - V_{SA}\ C_{PSA}\ P_{SA}\ T_{SA}) + (V_{FRA}\ P_{FRA}\ G_{FRA}\ H_{FRA} - V_{SA}\ P_{SA}\ G_{SA}\ H_{SA})]] + l_{a+w}]dt \qquad (4.2.7)$$

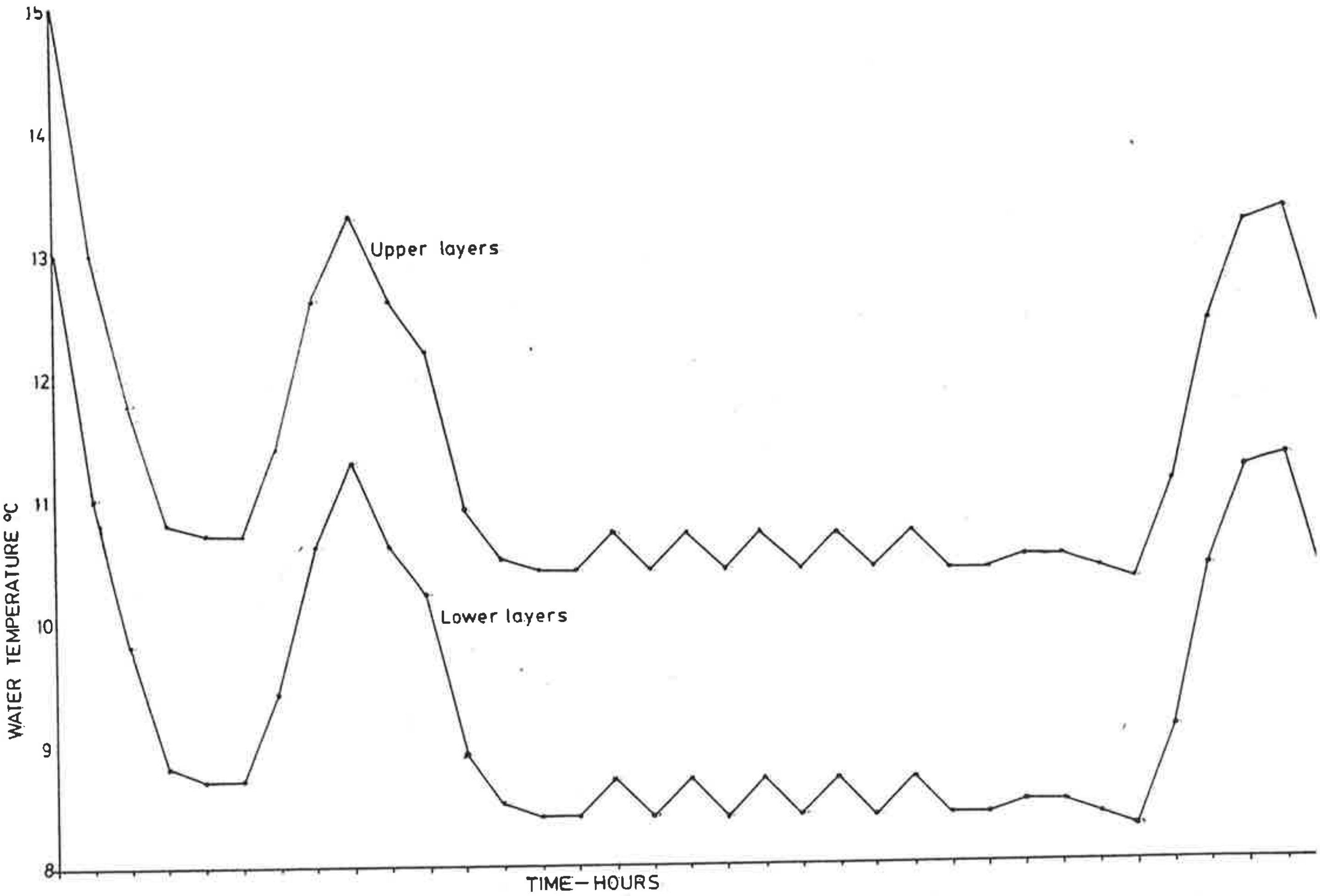
for the vessel/exchanger system.

4.3 STORED FLUID TEMPERATURES

The ultimate processes required for the thermal and the vapour treatment of moist air are dependent upon the conditions of that moist air entering the vessel/exchanger system. The conditions of the moist air leaving the vessel/exchanger system and the mass flow of that moist air through the same system, controlled by the automatic action of face and bypass damper motor, are to satisfy the instantaneous airconditioning load L. At the same time, the effectiveness of thermal and/or vapour treatment of the moist air required for a particular process is dependent upon the temperature levels of the stored fluid. The temperatures of the stored fluid when initially filled are generally below the temperatures required for heating and humidification and those fluid temperatures are generally above the temperatures required for cooling and dehumidification. The temperatures of the upper and the lower layers of the fluid within the store, TSUS and TSLS for summer and TSUW and TSLW for winter, as illustrated in Fig. 12 can be determined from the actual energy added to and/or subtracted from the stored mass over a given period of time dt. The swing in temperatures of the stored mass as illustrated from experimental data in Fig. 15 can be acceptable provided that the environmental control requirements are met within the prescribed range of tolerances.

Fig. 15

Water Temperature Swing during Heat and Mass Transfer Cooling Cycle



The leaving air dry bulb temperature T_{SA} and the leaving air moisture content level G_{SA} of the equation (4.2.7) are dependent upon the temperatures and the energy levels that can be maintained within the store by the selection of appropriate values of T_{SU} and T_{SL}. These storage temperatures need to be variable over a wide range between thermal and vapour air treatment processes of cooling and

The levels of temperatures T_{SU} and T_{SL} at any instant depend upon the mass of the fluid stored Ms, the thermal energy supply Q_T, the environmental load L and the losses l_{a+w}. Considering that the temperatures of the intermediate fluid layers are the averages of the temperatures of the upper and lower layers, the intermediate layer temperatures can be expressed by:

$$S_{New} - S_{old} = Ms\ C_{pw}\ (T_{SINew} - T_{SIold}) \tag{4.3.1}$$

The equation (4.2.7) can now be related to the effective average temperatures of the stored fluid as expressed in the equation (4.3.1).

The equation (4.2.7) can be rewritten as:

$$(S_{New} - S_{old}) = + [F1\ F2\ Mp\ C_{pw}\ (T_e - T_o) - [F3\ [(V_{FRA}\ C_{pFRA}\ P_{FRA}\ T_{FRA} - V_{SA}\ C_{pSA}\ P_{SA}\ T_{SA}) + (V_{FRA}\ P_{FRA}\ G_{FRA}\ H_{FRA} - V_{SA}\ P_{SA}\ G_{SA}\ H_{SA})]] + l_{a+w}]dt \tag{4.3.2}$$

By substituting the values of $(S_{New} - S_{old})$ from the equation (4.3.1) for the equation (4.3.2), we have

$$Ms.\ C_{pw}\ (T_{SINew} - T_{SIold}) = + [F1\ F2\ Mp\ C_{pw}\ (T_e - T_o) - [F3\ [\ (V_{FRA}\ C_{PFRA}\ P_{FRA}\ T_{FRA} - V_{SA}\ C_{PSA}\ P_{SA}\ T_{SA}) + (V_{FRA}\ P_{FRA}\ G_{FRA}\ H_{FRA} - V_{SA}\ P_{SA}\ G_{SA}\ H_{SA})]]+l_{a+w}]dt \tag{4.3.3}$$

At the same time TS_I can be expressed within approximate values in terms of T_{SU} and T_{SL} by the following equations:

$$\frac{T_{SUNew} + T_{SLNew}}{2} = T_{SINew} \qquad (4.3.4)$$

and $$\frac{T_{SUold} + T_{SLold}}{2} = T_{SIold} \qquad (4.3.5)$$

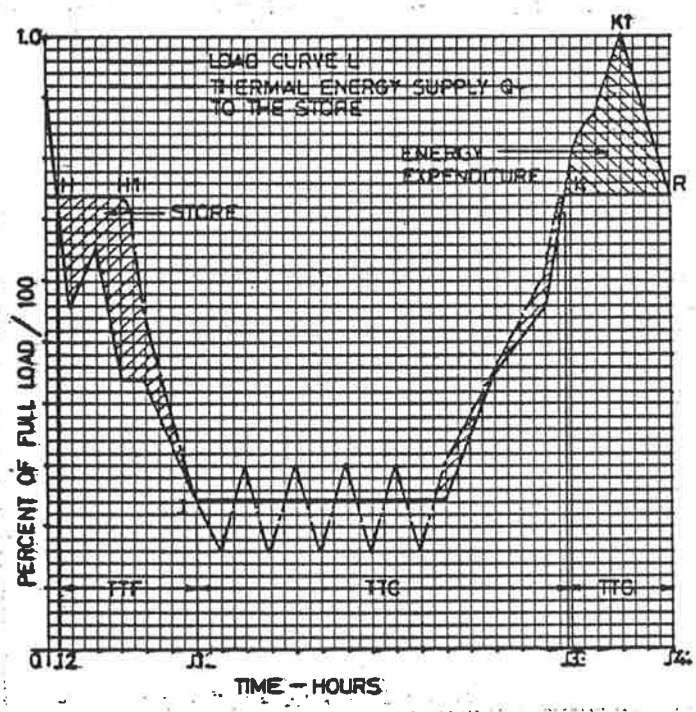
When the temperature of the upper layers including the temperature of the exposed water surfaces is kept below the dry bulb temperature of the entering air i.e., $T_{SU} < T_{FRA}$ and the temperature of the lower layers is approximately equal to the dew point of that entering air i.e., $T_{SL} \approx D.P_{FRA}$, the resulting thermal and vapour air treatment is sensible cooling and humidification.

4.4 STORAGE RESPONSE

An environmental control load is imposed on the vessel/exchanger system when the supply air fan operates. A load curve L has been established for a period of time dt from experimental data, recorded for the means of moving the necessary mass of selected air to match the environmental control load demand. The load curve L indicates the occurrence of the peak load and a least load of approximately 30%. This load curve L remains the same for all environmental control systems - conventional or vessels/exchangers. In a conventional environmental control system, the load must be satisfied instantaneously and therefore the thermal energy supply by an energy conversion device must follow the contour of the load profile L. The thermal energy supply by an energy conversion device for the vessel/exchanger system is to satisfy the requirements of energy for the store. The thermal energy supply curve Q_T developed from experimental data for the vessel/exchanger system over the same period of time dt has been imposed on the load curve L for the analysis of the storage response. Fig. 16 provides the details of the both curves L and Q_T in terms of percentage of peak load.

Fig. 16

Storage Response to Load



The thermal energy supply curve Q_T can be thought of as the actual load curve for the store of the vessel/exchanger system. After attaining a peak, the load L declines. Immediately after the point H, the thermal energy supply Q_T becomes greater than the load L for the vessel/exchanger system and the storage of energy begins.

The thermal energy supply to the store continues until the predetermined energy level of the store is attained at the point J. The time taken to attain this predetermined energy level is TTF. The total thermal energy supplied by the energy conversion device is the area enclosed by the curve J2HH1JJ1. The energy supplied to satisfy the environmental load requirements during that time period is the area enclosed by the curve J2HJJ1. The difference between these two areas enclosed by the curve HH1JH represents the actual energy stored during this time interval and this area is designated by store in Fig. 16. When the water temperature controller senses that the predetermined storage energy level is reached and the loads are of comparatively lower values, the thermal energy supply Q_T cycles around the load curve L for a time interval TCC. During this period of time little energy is either added or subtracted from the store of the vessel/exchanger system until the point K is reached. The areas enclosed by the curves J1JKJ3 are designated by one of broken line and the other of continuous line. The continuous line represents the load curve and the broken line represents the thermal energy supply to the store for the vessel/exchanger system.

Immediately after the point K the load L becomes greater than the thermal energy Q_T supplied by the energy conversion device for the vessel/exchanger system. The energy is expended from the store until the point R is reached during the time interval TTG. After the point R, the thermal energy supply Q_T for the vessel/exchange system exceeds the load L. The energy supplied to satisfy the environmental load requirements during this time period is the area enclosed by the curve J3KK1RJ4. The total thermal energy supply by the energy conversion device for the vessel/exchanger system during this time period is the area enclosed by the curve J3KRJ4. The difference between these two areas KK1RK represents the actual energy expended from the store. This area is designated as Energy Expenditure in Fig. 16.

The area HH1JH must be greater than the area KK1RK to satisfy the environmental control requirements plus the losses from the store. These areas determine the storage capacity for the satisfactory operation of the vessel/exchanger system.

The thermal energy storage has two critical parameters - one is the level of the actual thermal energy supplied to the store and the other is the period of time associated with that supply.

On this basis, the time interval TTC can be stipulated for the effective sizing of the store, as well as the maximum output capacity of the energy conversion device. The height J2H represents the maximum output capacity of the energy conversion device for the vessel/exchanger system. The time interval TTF can be extended and the time interval TTC can be reduced, to optimise the storage capacity level and the maximum thermal capacity output of the energy conversion device.

4.5 ENERGY CONSUMPTION FOR BOTH SYSTEMS - CONVENTIONAL AND VESSEL/EXCHANGER SYSTEMS

For all-year-round operation, in the majority of large buildings, the energy conversion devices are either the reciprocating compressors fitted with the cylinder unloading type capacity controllers or the centrifugal compressors with the guide vane controllers. With this arrangement the suction pressure does not drop below the control setpoint minus the differential since the compressors unload to balance capacity with the instantaneous load.

Capacity control reduction steps for commercially available reciprocating and centrifugal compressors have been established from experimental data which comply with those supplied by the manufacturers.

In Fig. 5 on Section 1, one hundred percent of brake kiloWatt is related to one hundred percent of thermal energy output of such energy conversion devices. The coefficient of performance varies widely between different refrigeration systems, as well as between each system part load conditions. To maintain the uniformity of comparative study between different systems and for ease of empirical comparison, in Fig. 5 the part load is expressed in terms of the fraction of the full load. The brake kilowatt at that part load is expressed in terms of the fraction of the brake kilowatt consumed at the full load. The following table illustrates the relationship between the load and the brake kilowatt.

% OF FULL LOAD CAPACITY	**% OF RELATED BRAKE KILOWATT**
100	100
83.3	87
66.3	74
50	60
33.3	45

For the reduction of the capital cost, as well as maintenance cost, the number of energy conversion devices has been kept to absolute minimum for the experiments. To maintain the uniformity of comparative study, a single energy conversion device has been considered for each system.

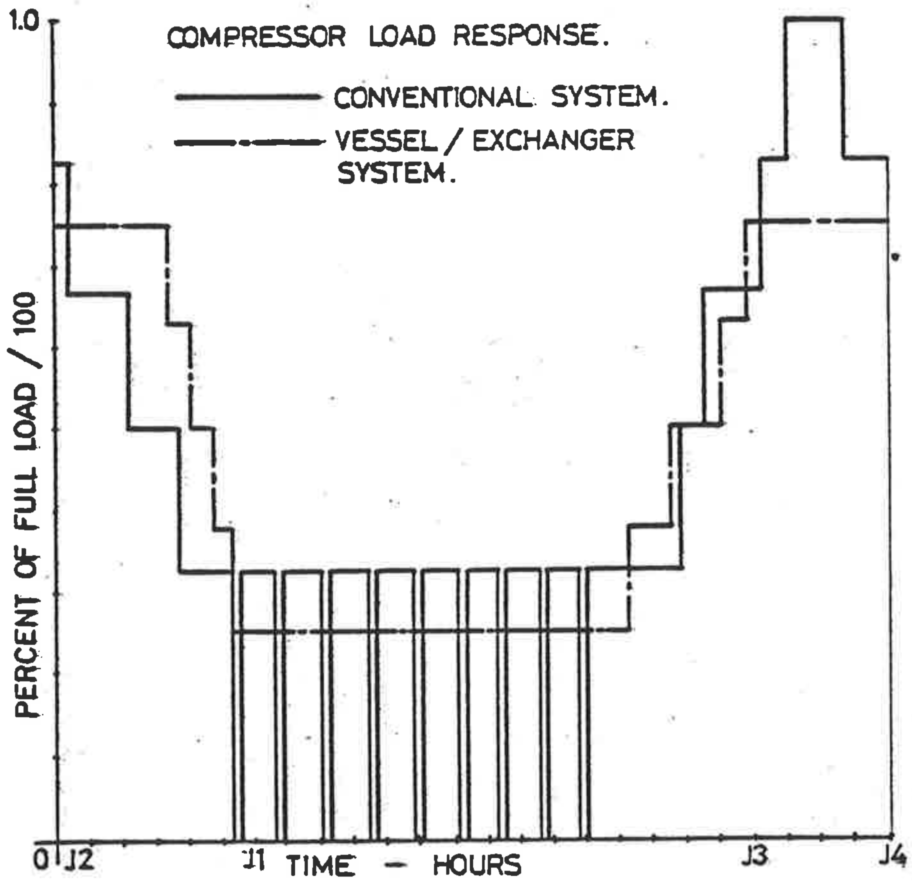
The comparative energy consumption for a given environmental load curve over a time period has been established for two systems.

Energy conversion device for a conventional system which is sized to offset the peak load is designated by System X in Figures 17, 18 and 19.

Energy conversion device for vessel/exchanger system which is sized to offset the major part of the peak load is designated by system Y in Figures 17, 18 and 19.

Fig. 17 provides the details of loading and unloading of cylinders of reciprocating compressors for both systems.

Fig. 17
Loading & Unloading of
Cylinders for both Systems



For the conventional system the cylinders load and they unload with respect to the load curve L of Fig. 16. For the vessel/exchanger system, the cylinders load and they unload with respect to the thermal energy supply Q_T to the store of Fig. 16.

The analysis time period for which both systems are analysed is the summation of the time segments TTF, TTC and TTG of Fig. 16. In Fig. 17, time segments taken for different steps of loading and unloading of cylinders for reciprocating compressors are clearly defined. The brake kilowatt curves for both systems are then developed in Fig. 18 on the basis of these time segments for reciprocating compressors.

Fig. 18
Brake Kilowatt Consumption
Reciprocating Compressors

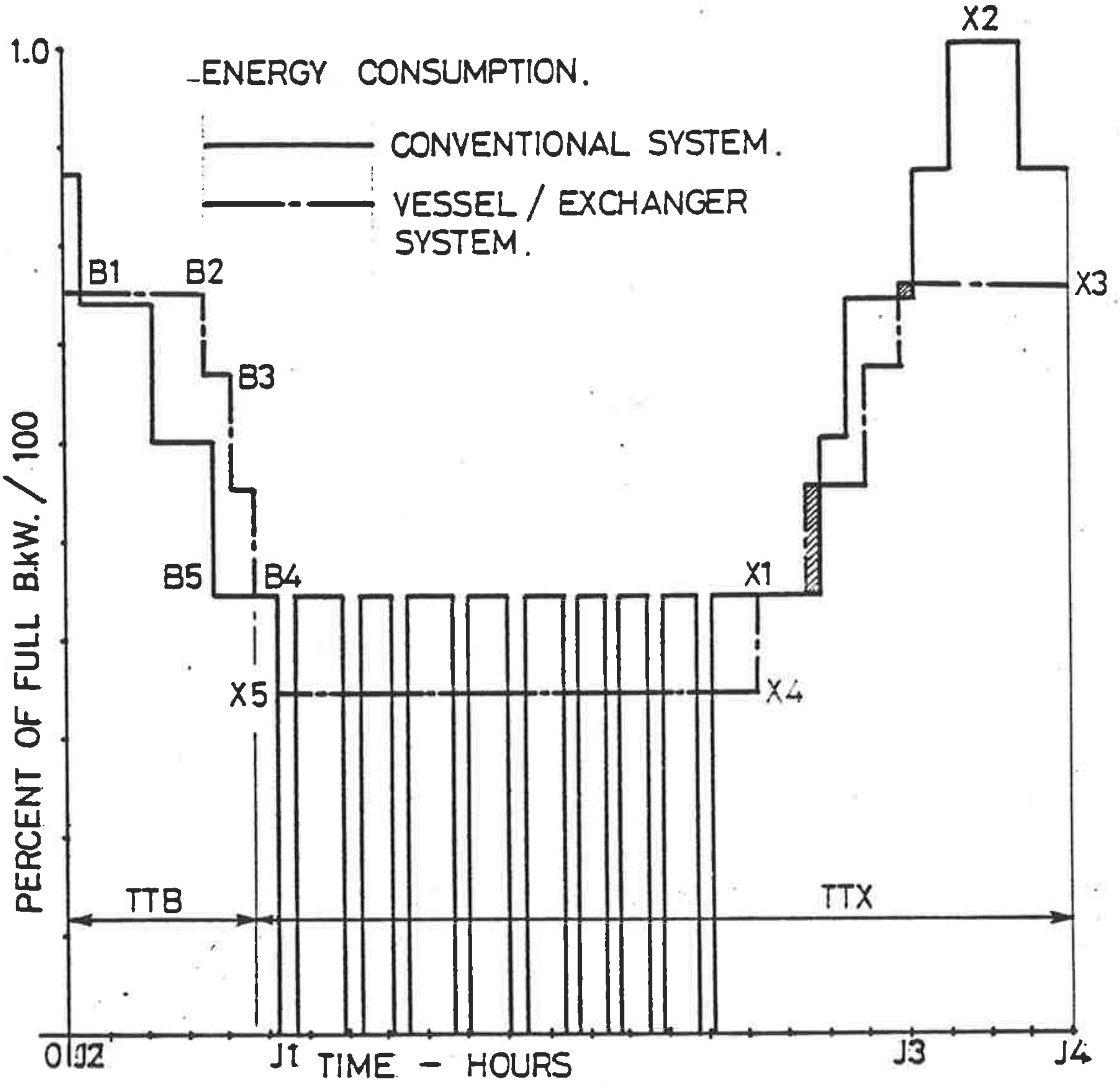
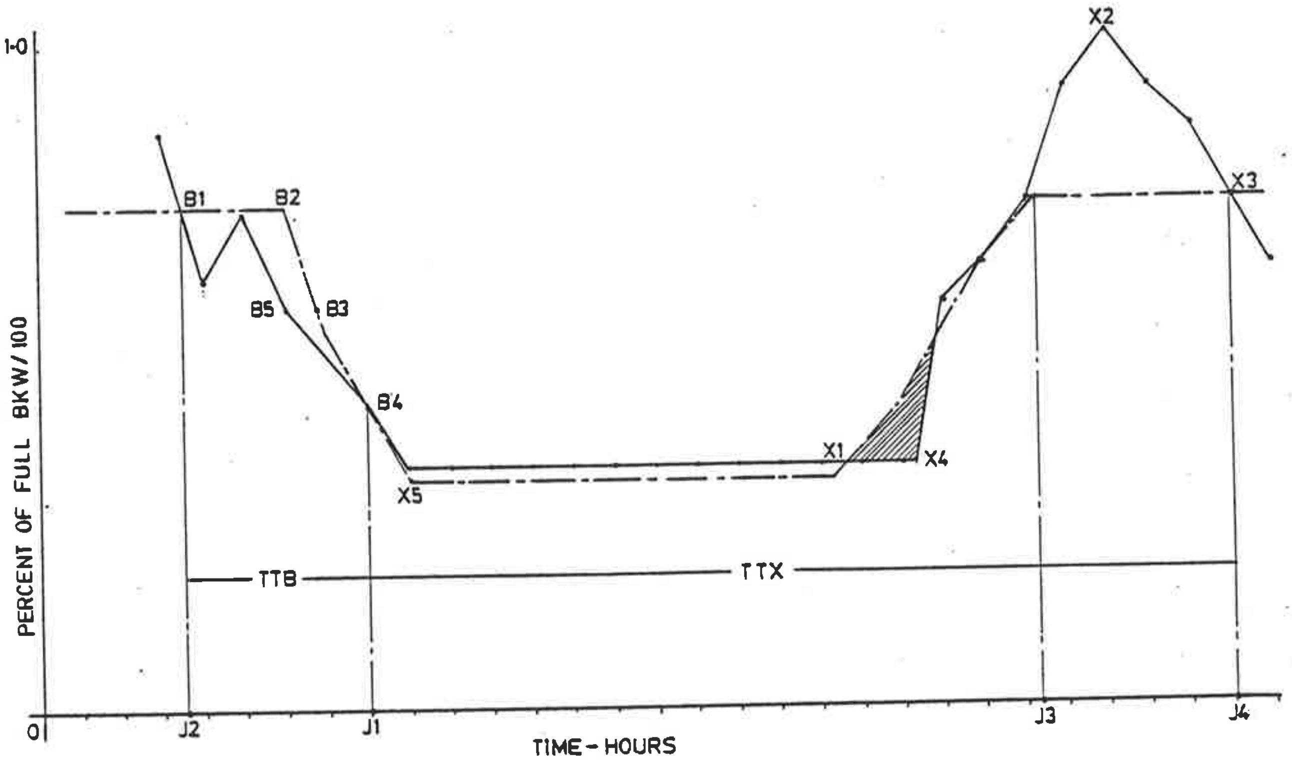


Fig. 19 provides the details of the brake kilowatt curves for both systems when centrifugal type refrigeration compressors are used instead of reciprocating compressors.

Fig. 19
Brake Kilowatt Consumption
Centrifugal Compressors



Motor and transmission losses are not considered as these losses are common to the both systems.

The maximum leaving condenser water temperature recorded was 35°C.

The brake kilowatt consumption for both systems varies with the time. The brake kilowatt, then, can be expressed in the following terms:

Conventional system, system X

BKW = F(t)

Vessel/exchanger system, system Y

BKW = g(t)

In Figures 18 and 19 the area B1B2B3B4B5B1 represents the additional brake kilowatt hour consumed by the vessel/exchanger system over that of the conventional system during the time period TTB. The area B4X1X2X3X4X5B4 minus the shaded areas represents the additional brake kilowatt hour consumed by the conventional system over that of the vessel/exchanger system during the time period TTX. For the reciprocating compressor the one parameter of the shaded areas related to B4X1X4X5 represents the time when the compressor for the conventional system is off and the other parameter related to this area is the brake kilowatt which is not consumed during this period. For the same compressor the shaded areas enclosed within the boundaries of X1X2X3X4X1 represents the additional brake kilowatt hour consumed by the vessel/exchanger system over that of the conventional system. The additional brake kilowatt hour consumed by the conventional system over that of the vessel/exchanger system at least load conditions is due to the fact that the compressor for the conventional system is operating at a much lower proportional load band than that of the vessel/exchanger system.

The area B1B2B3B4B5B1 can be expressed as

$$\int_{t_{o1}}^{t_{n1}} [g_1(t) - f_1(t)]dt$$

and the area B4X1X2X3X4X5B4 minus the shaded areas can be expressed as

$$\int_{t_{n1}}^{t_{n2}} [f_2(t) - g_2(t)]dt$$

The additional brake kilowatt hour consumed by the conventional system is greater than the additional brake kilowatt hour consumed by the vessel/exchanger system.

The storage of energy for the vessel/exchanger system begins when both the load L and the energy levels of the surrounding environment are of comparatively lower values. The increase in the thermal energy supply Q_T for lower brake kilowatts occurs when these conditions exist.

SECTION 5

ANALYSES OF THERMODYNAMIC STATE OF

MOIST MIXED AIR

AND

PARASITIC ENERGY INPUT REQUIREMENTS

INDEX

SECTION 5: **ANALYSES OF INPUT REQUIREMENTS**

5.1 GENERAL

The thermodynamic processes of the integrated storage vessels/exchangers involving the transfer of heat, mass and momentum have been studied for these experiments.

During the cooling and dehumidification cycle, the heat and moisture transfer are effected at both the metal surfaces MSS and the exposed water surfaces WSI and WSO of Figure 7.

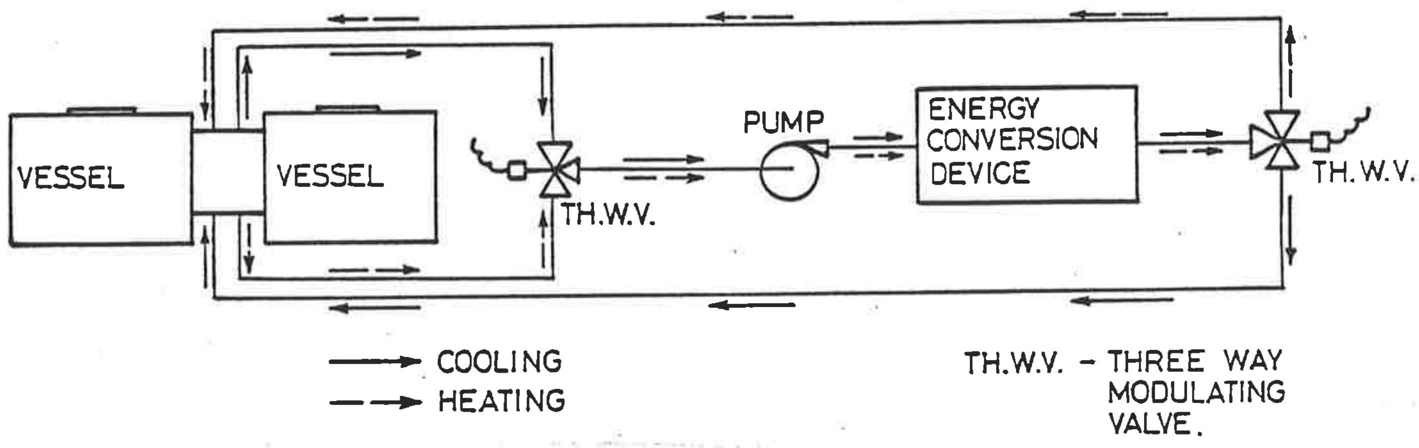
During the heating and humidification cycle, sensible heat is added via the metal surfaces MSS and both heat and moisture is added at the exposed water surfaces.

During the summer, when the cooling and/or dehumidification requirements are predominant, the warmer water is drawn from the top layers by the energy conversion device and is then returned to the bottom layers.

During the winter, when the heating and/or humidification requirements are predominant, the colder water is drawn from the bottom layers, is heated by the energy conversion device and is then returned to the top layers.

The transfer mode between heating to cooling or cooling to heating is automatic and is achieved by automatic three-way valves as shown in Figure 20.

Fig. 20
Transfer Mode



The return air temperature sensor selects the mode of operation and operates the three-way valves as required.

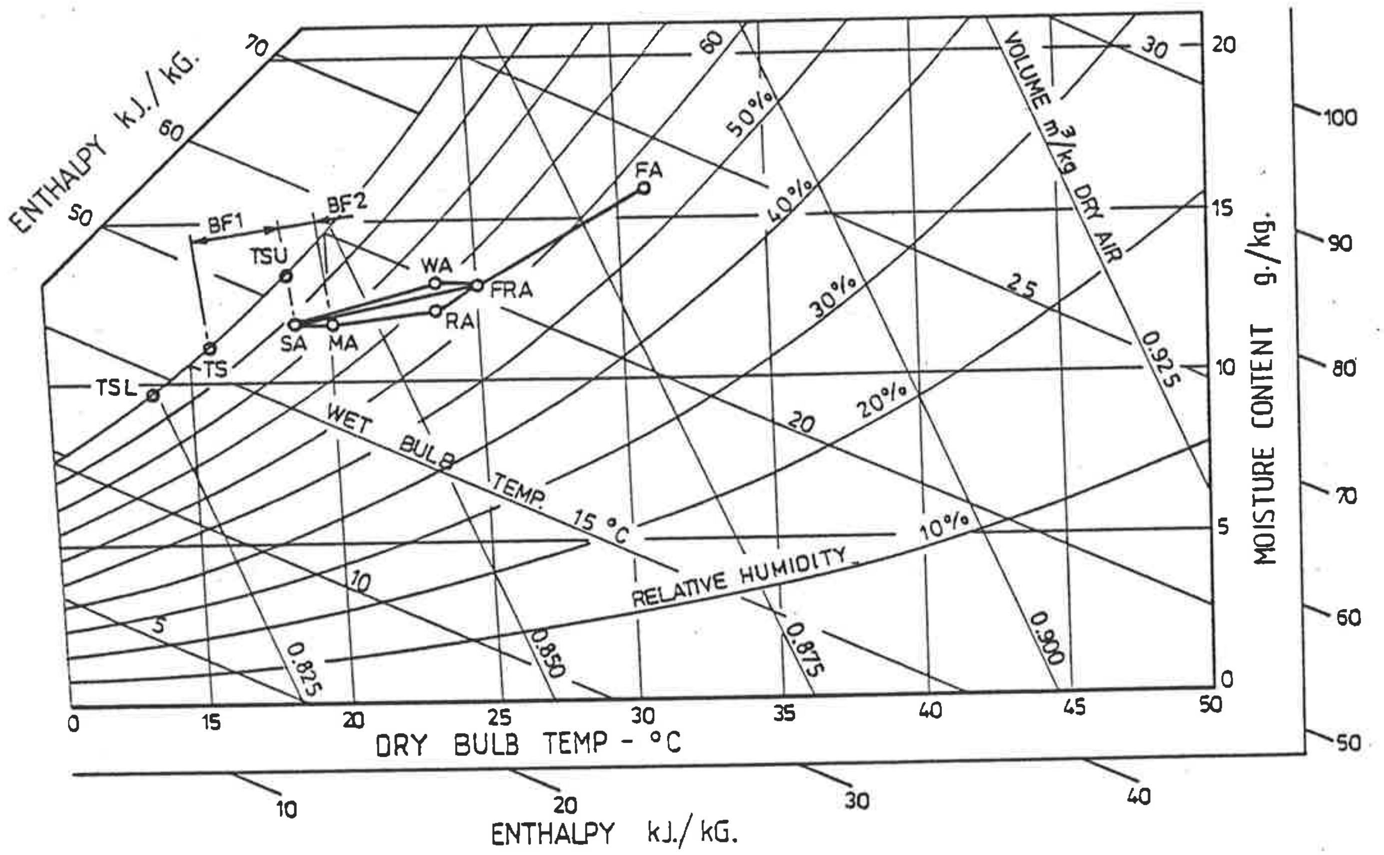
When a centrifugal type air fan is used, the condition of the air is turbulent at the discharge. This turbulent energy is normally completely lost but in the present systems it is utilised to augment the transfer of heat and moisture.

The experimental units are fitted with:

- an automatic face and bypass damper system,
- an automatic re-heat damper system, and
- an automatic relative humidity damper system.

Psychrometric charts based on experimental data are included in this section to indicate the thermodynamic state of the moist air at different locations within the system during the thermal and vapour treatment process.

Fig. 21
Cooling cycle, Magill House



The notations for psychrometric charts are:

FA - Fresh air
FRA - Fresh/return air mixture (entering air)
RA - Room or return air
WA - Air leaving water/metal combined horizontal surface
SA - Air leaving the integrated storage/exchangers
MA - Mixed air. A mixture of FRA and SA
T_{SU} - Upper water layer temperature
T_{SL} Lower water layer temperature
T_S - A hypothetical saturated air temperature.

The fresh air FA mixes with the return air RA and the mixture FRA represents the conditions of the air entering the storage/exchangers. Some portion of this air bypasses the storage exchangers through the bypass duct. WA, SA and MA represent the air

- leaving the water/metal combined horizontal surface,
- leaving the storage/exchangers,

and the mixed air respectively.

The mixed air MA then is supplied to the controlled environment and MARA is the room sensible heat factor.

5.2 BYPASS FACTORS

The automatic face and bypass damper system allows the fresh air/return air mixture to bypass the thermal exchange chambers if so required by the process. A return air sensor modulates this automatic face and bypass damper motor. Also some portion of the fresh air/return air mixture remains unaltered (unmixed) when it passes through the thermal and vapour exchange chambers. Thus the system effectively has two bypass factors.

$$\text{Bypass factor BF1} = \frac{\text{Distance between the two points FRA \& SA}}{\text{Distance between the two points FRA \& } T_S}$$

in the figure 21 established the portion of the fresh air/return air mixture which remains unaltered when passes through the thermal exchange chambers.

$$\text{Bypass factor BF2} = \frac{\text{Distance between the two points FRA \& MA}}{\text{Distance between the two points FRA \& SA}}$$

in the figure 21 establishes the fresh air/return air mixture which has by-passed the thermal and vapour exchange chambers through the automatic bypass duct.

5.3 THERMAL AND VAPOUR TREATMENT SYSTEM FOR MOIST AIR

The application of this system is complicated by the following:

- when the temperatures of the upper and lower layers of the stored fluid are above the wet bulb temperature of the fresh air/return air mixture FRA then both the dry bulb and wet bulb temperature of the supply air SA rise.

- when the temperatures of the upper and lower layers of the stored fluid are below the wet bulb temperature of the fresh air/return air mixture FRA, then a lowering of both the dry bulb and wet bulb temperatures of the supply air SA occurs.

- humidification and sensible cooling processes occur when the upper layer water temperature is above the dew point of the fresh air/return air mixture FRA but at or below the wet bulb temperature of that mixture.

- dehumidification and sensible cooling processes occur when the upper layer water temperature is below the dewpoint of the fresh air/return air mixture FRA.

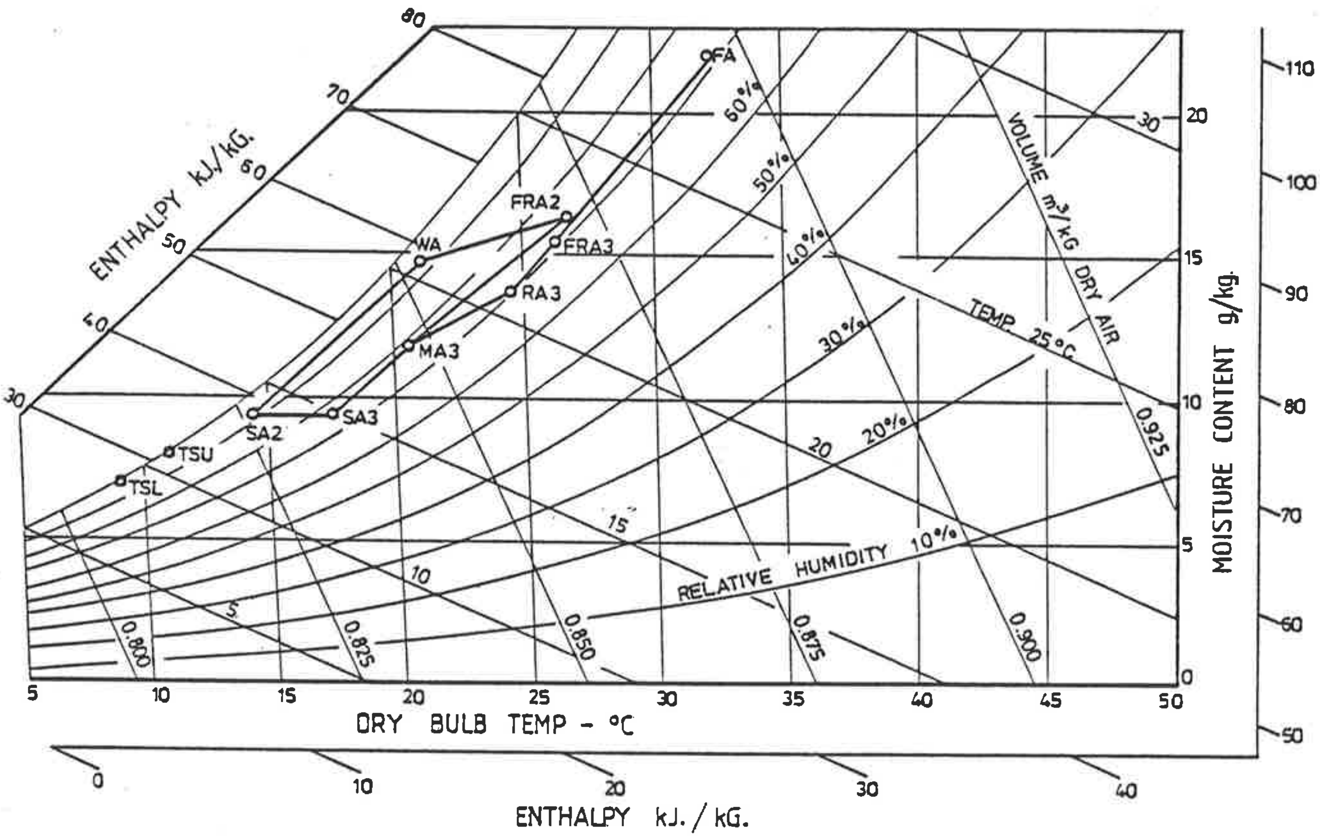
- sensible heat is transferred from the fresh air/return air mixture FRA to the supply air SA for re-heat purposes.

- the thermodynamic state of the moist mixed air MA must be maintained in such a condition that the pre-set values of the controlled environment are met.

5.4 SENSIBLE COOLING, DEHUMIDIFICATION AND RE-HEAT

5.4.1 Re-heat

Fig. 22
Re-heat, Darwin station



Heat transfer surfaces have been incorporated within the units to transfer heat from the warmer fresh air/return air mixture stream to the thermally treated colder air stream. During the time segment taken by the incoming fresh air/return air mixture to travel through the inlet chamber, heat and mass transfer take place between the exposed water surface and the said air mixture stream and simultaneously heat transfer takes place between the colder air stream leaving the units and the said air mixture stream. The changes in the thermodynamic state of the moist mixed air in the inlet chamber are from FRA2 to WA and the changes in the thermodynamic state of the moist colder air leaving the units are from SA2 to SA3. FRA2 represents the fresh air/return air mixture prior to the re-heat treatment and FRA3 represents the fresh air/return air mixture after the re-heat treatment.

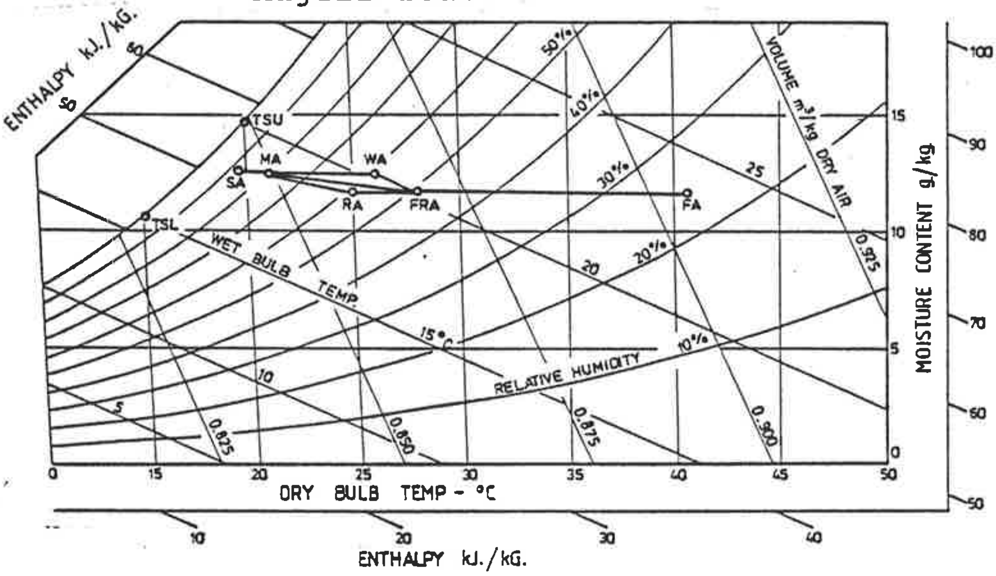
5.4.2 Sensible cooling and dehumidification

Figures 21 and 22 illustrate the cooling and dehumidification processes of the experimental units. The fresh air/return air mixture FRA has higher moisture content for the conditions presented in Figure 22 than that for the conditions presented in Figure 21. The temperatures of the upper and lower layers of storage fluid for the Figure 22 are higher than those of the Figure 21. The moisture content of the supply air SA needs to be lowered when the return air relative humidity rises above the set point plus the differential. The lowering of moisture content of supply air SA results in the lowering of the dry bulb temperature of supply air SA. Sensible heat ratio factor MARA for Figure 22 is lower than that for Figure 21. When the room sensible heat factor MARA declines as in the case of Figure 22 the sensible heat SA1SA2 is added to the supply air SA to maintain the pre-set values of the controlled environment.

5.5 EVAPORATIVE COOLING EFFECT AT THE WATER SURFACE

In an evaporative cooling system, the thermodynamic state of the moist supply air results from the exchange of sensible energy for latent energy. When the operating mode of the present system changes from heating to cooling or from cooling to heating, the water temperatures pass through a band in which advantage can be taken of the evaporative cooling effect.

Fig.23
Evaporative cooling effect,
Magill House



T_{SU} and T_{SL} are the temperatures of the upper and lower layers of the storage. The dew point of the fresh air/return air mixture FRA is less than the surface water temperature T_{SU} and this dew point falls within the temperature range of T_{SU} and T_{SL}. Some portion of the mixture FRA passes over the exposed water surface of the experimental units. The conditions of the air leaving the water surface are represented by WA. Evaporative cooling takes place between the points FRA and WA. The exposed water surface area can be adjusted to shift WA close to T_{SU} if so required for a particular process.

5.6 HEATING AND HUMIDIFICATION

Fig.24
Heating and humidification,
Magill House

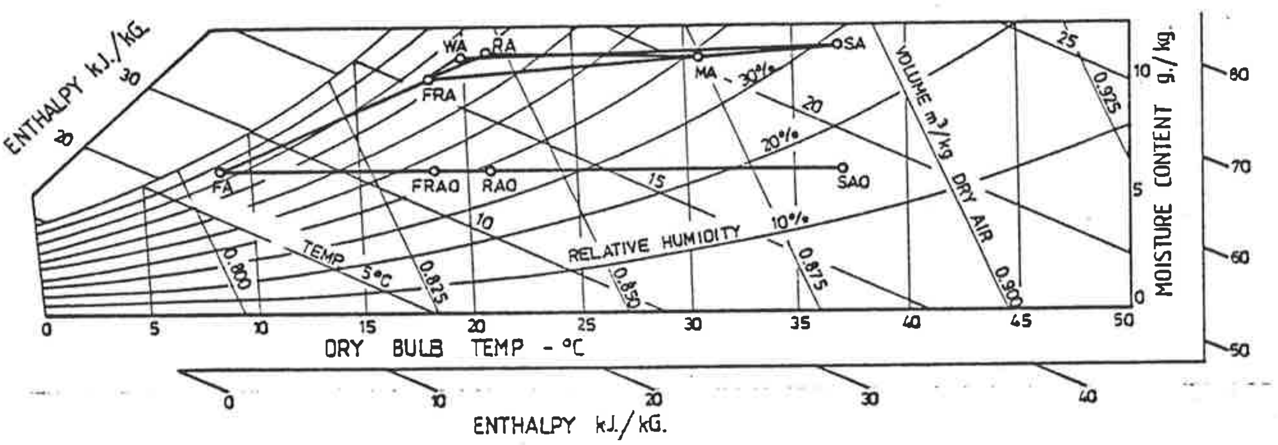


Figure 24 illustrates the heating and humidification process for the experimental units. The moisture added to the portion of the air passing through the experimental units is the difference in moisture content levels between the conditions FRA and SA.

When the heating mode is provided by an ordinary heater or a water heating coil, the fresh air/return air mixture is represented by FRAO and the return air conditions are at RAO. The moisture must then be added to this system to maintain necessary comfort conditions.

5.7 ENERGY TRANSFER

The energy is drawn from the energy storage for the thermal treatment of the moist supply air. The parameters affecting the performance of the experimental units can be classified as constant quantities, variable quantities and adjustable quantities.

Constant quantities

The constants introduced as design elements are:

- length of vertical metal air tunnels,
- vertical contact surfaces, air and water,
- mass flow of transport agent (water),
- maximum mass flow of air.

Variable quantities

the variables involved at any particular instant of operation are:

- air mass flow and air velocity through the thermal and vapour exchange chambers of the experimental units,

- conditions of fresh air/return air mixture entering the thermal and vapour exchange chambers,

- upper and lower layer temperatures of the stored fluid,

- total energy level of the stored fluid.

Adjustable quantities

The adjustable variables are exposed horizontal water surface area and horizontal metal surface area at the exposed water surface.

The driving potential behind the transfer of heat and moisture for these experiments cannot be kept constant, as their potential values vary with the fluctuating load.

5.8 MOIST MIXED AIR

The moist supply air SA leaving the thermal treatment chambers mixes with the fresh air/return air mixture FRA and the thermodynamic state of the mixed air MA must satisfy all load values between the peak value and the minimum value.

In Figures 21, 22, 23 and 24 the fresh air/return air mixture FRA enters at the horizontal combined water/metal surfaces. The leaving air conditions are represented by WA on the psychrometric charts.

The heat exchange can be represented by

$$V_{FRA}\ C_{pFRA}\ \ PFRA\ T_{FRA} - V_{WA}\ C_{pWA}\ \ PWA\ T_{WA} = A_{WM}\ LMTD_{WM}\ h_{WM}$$

where

$$LMTD_{WM} = \frac{T_{FRA} - T_{WA}}{\ln \dfrac{T_{FRA} - T_{WM}}{T_{WA} - T_{WM}}}$$

T_{WM}	combined water/metal surface temperature, ^{o}C
h_{WM}	total heat transfer coefficient, $W/m^2\ ^0C$
A_{WM}	combined water/metal surface area, sq.m
FRA	fresh air/return air mixture conditions
WA	conditions of air leaving the combined surface
V	air flow, m^3/s
Cp	specific heat, Kj/KG of DA. ^0C
P	density, $\frac{\text{kg of DA}}{m^3}$
T	temperature, ^{o}CDB

$$\text{so } h_{WM} = [V_{FRA} \cdot C_{pFRA} \cdot\ \ PFRA \cdot T_{FRA} - V_{WA}\ C_{pWA}\ \ P_{WA}\ TWA] / [LMTD_{WM}\ \ A_{WM}] \qquad (5.6.1)$$

Similarly the mass transfer rate per unit can be established by

$$M_{WM} = [V_{FRA} \quad p_{FRA} \, G_{FRA} - V_{WA} \, p_{WA} \, G_{WA}] \; / \; A_{WM} \qquad (5.6.2)$$

where

G = gm of moisture/kg of D.A.,

M_{WM} = mass transfer rate per unit area, $\frac{\text{gm of moisture}}{\text{sq. m. s}}$

for combined water/metal surface.

The air leaving the horizontal water/metal surface WA enters the metal tunnels. Further heat and moisture transfer process takes place - this time between the air and the vertical metal surfaces. The air leaving the metal tunnels is represented by the conditions SA. The heat and mass transfer for the metal surfaces can be represented by

$$h_M = (V_{WA} \quad C_{pWA} \quad p_{WA} \quad T_{WA} - V_{SA} \, C_{pSA} \quad P_{SA} \, T_{SA}) / (LMTD_M . A_M) \qquad (5.6.3)$$

where SA - conditions of air leaving the vertical thermal air treatment chambers,

$$LMTD_M = \frac{T_{WA} - T_{SA}}{\ln \dfrac{(T_{WA} - T_M)}{(T_{SA} - T_M)}}$$

T_M = mean metal surface temperature, ^{o}C

A_M = vertical metal surface area, sq.m

h_M = heat transfer coefficient of vertical metal surface $W/m^2 . ^{o}C$

and

$$M_M = [V_{WA}\ P_{WA}\ G_{WA} - V_{SA}\ \ P_{SA}\ G_{SA}] \ / \ A_M \tag{5.6.4}$$

where

M_M - mass transfer rate per unit area, $\frac{\text{gm of moisture}}{\text{sq.m, s}}$ for vertical metal surface

Figures 25-31 illustrate the heat transfer coefficients and the mass transfer rates for the experimental units.

Fig. 25

Combined surface, heat transfer coefficients, heating and humidification

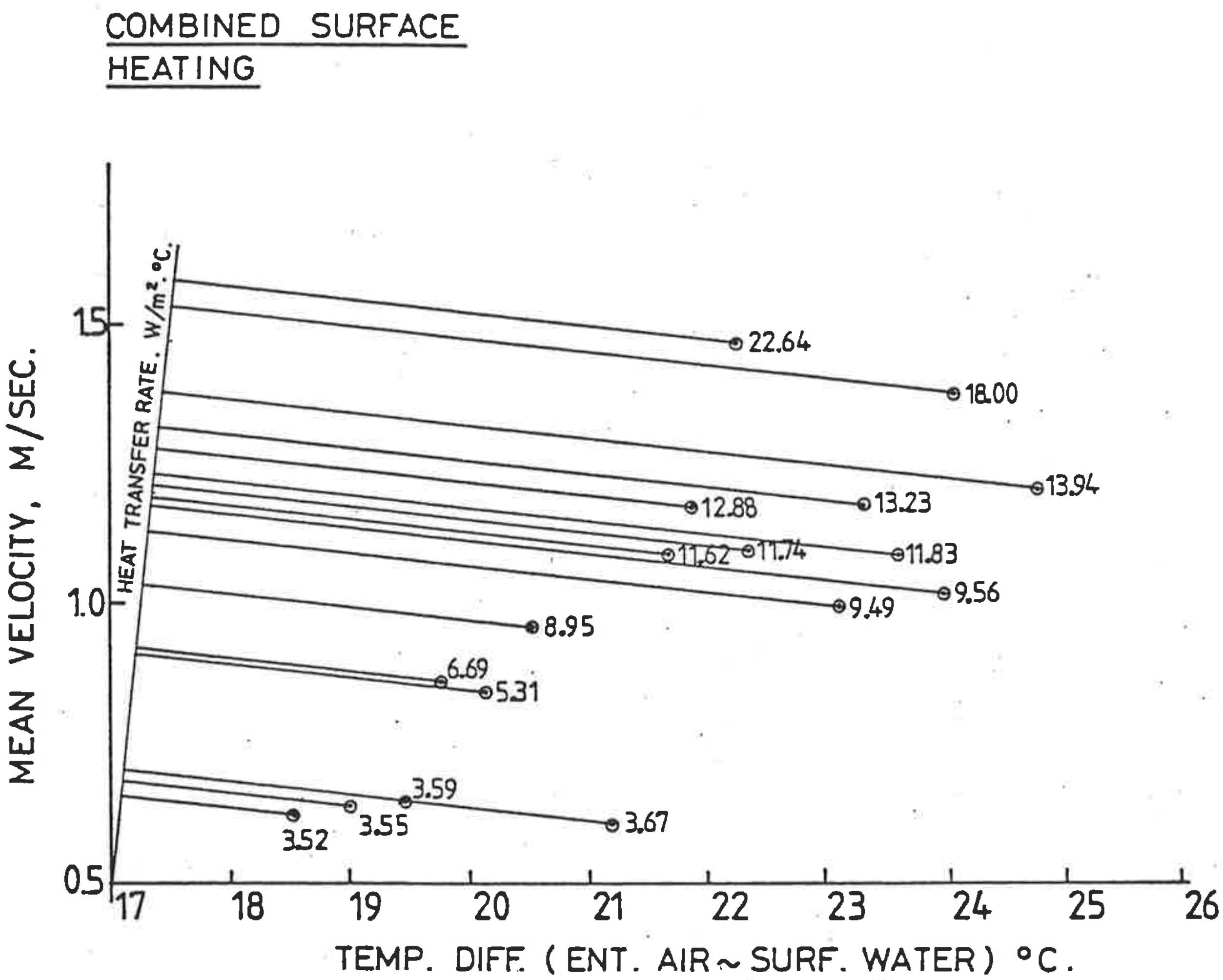


Fig. 26
Combined surface mass transfer rates per unit area, heating and humidification

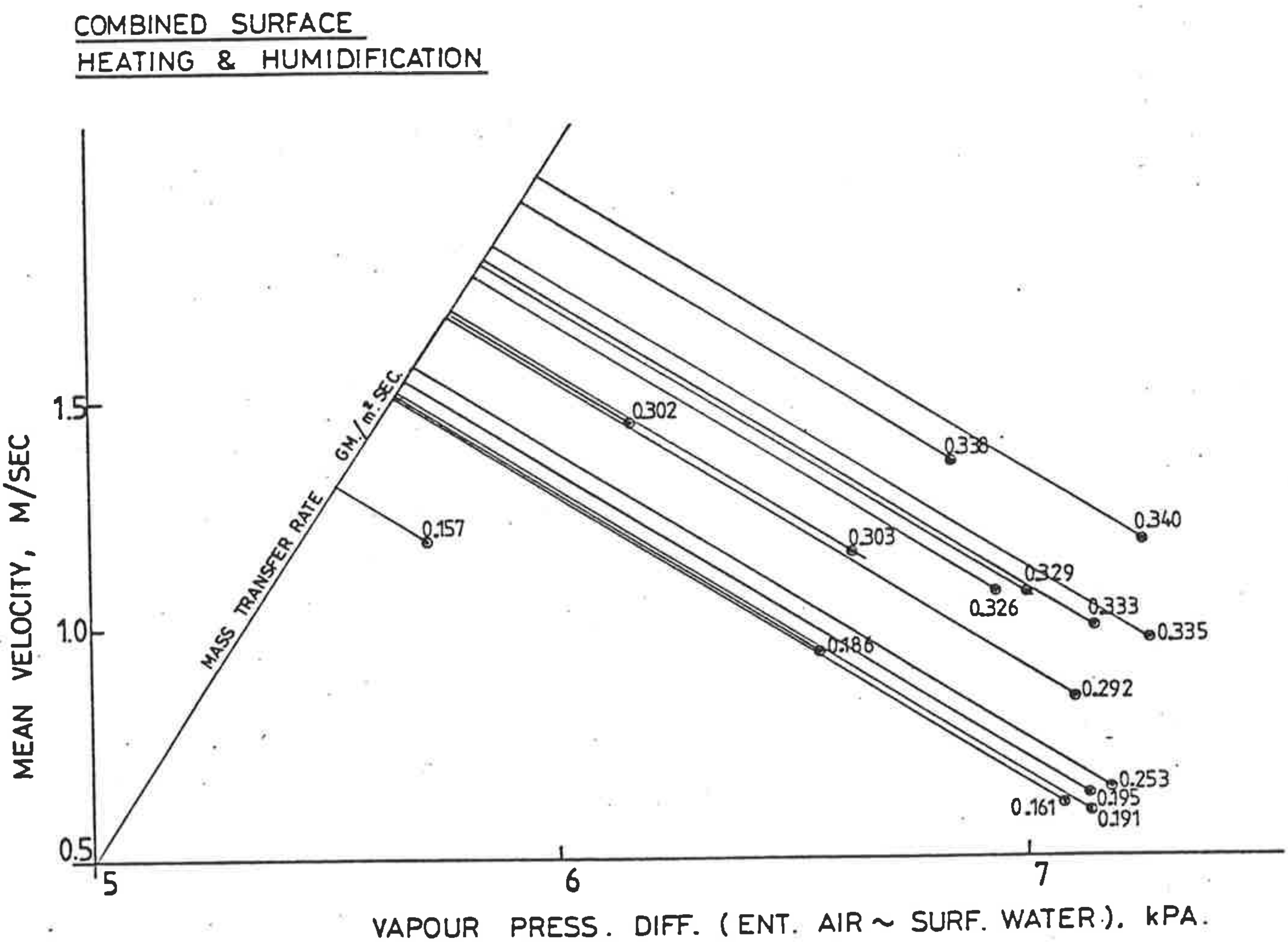


Fig. 27
Metal surface, heat transfer co-efficients, heating and humidification

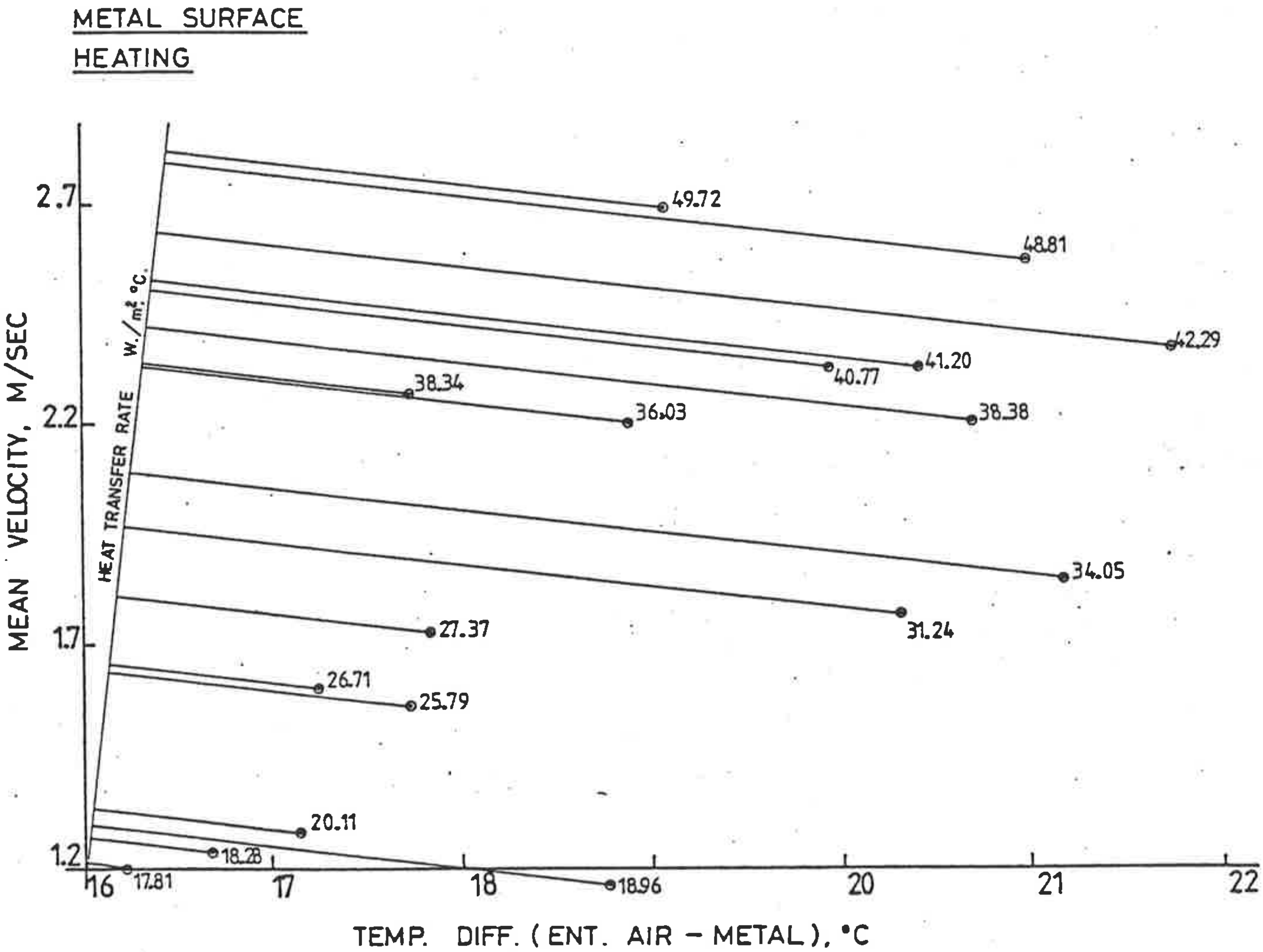


Fig. 28

Combined surface heat transfer coefficients, cooling and dehumidification

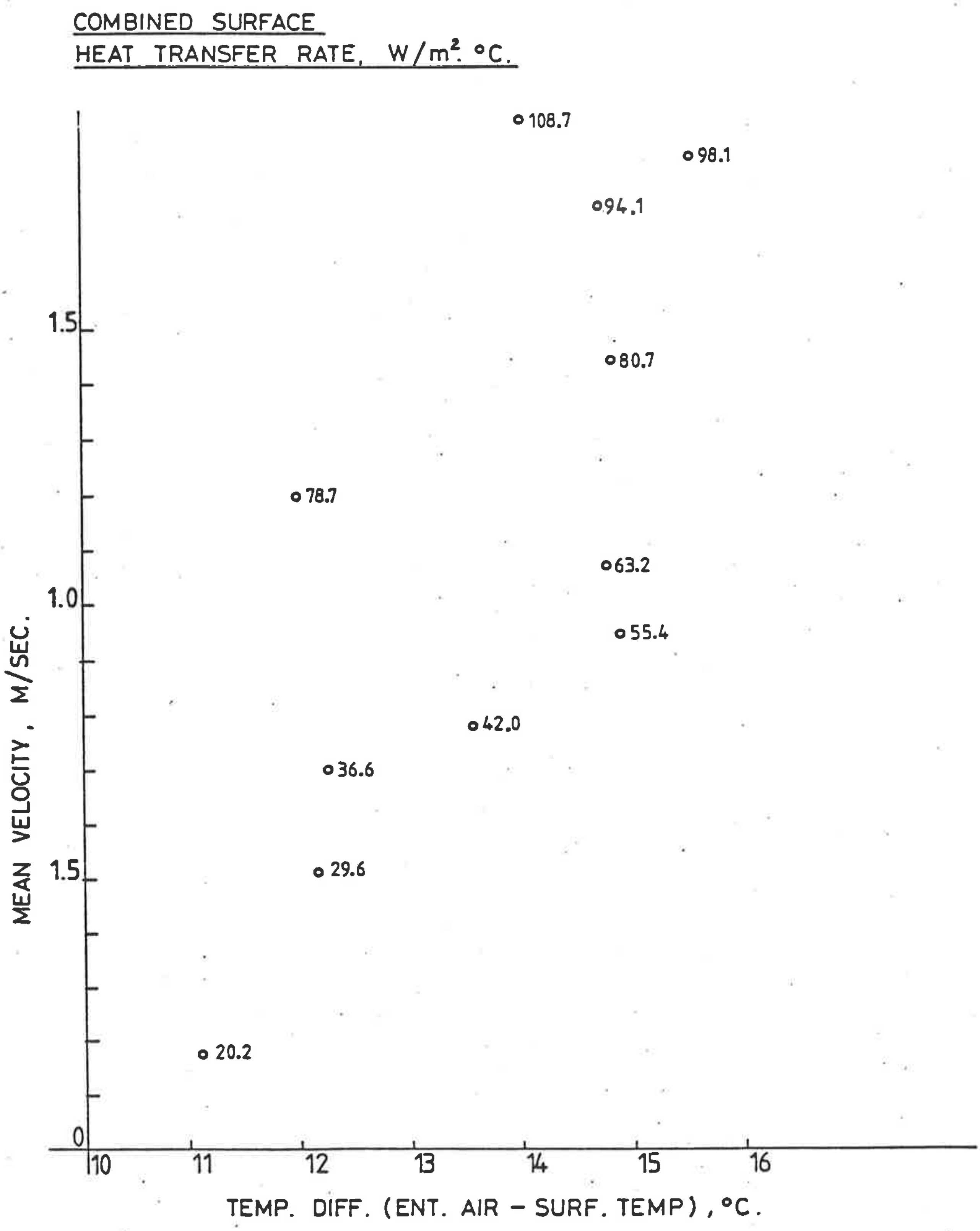


Fig. 29
Combined surface, mass transfer rates per unit area, cooling and dehumidification

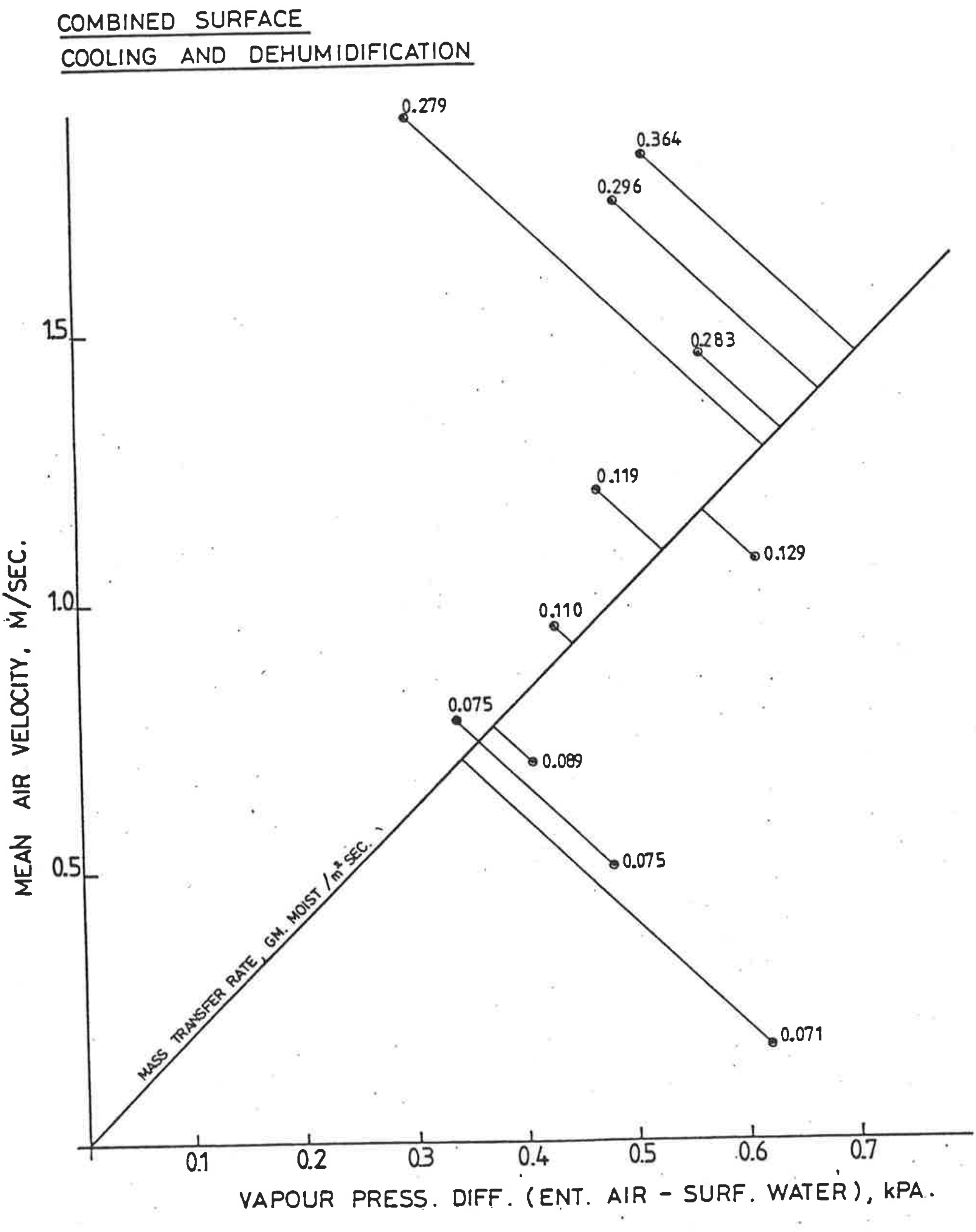


Fig. 30

Metal surface, heat transfer coefficients, cooling and dehumidification

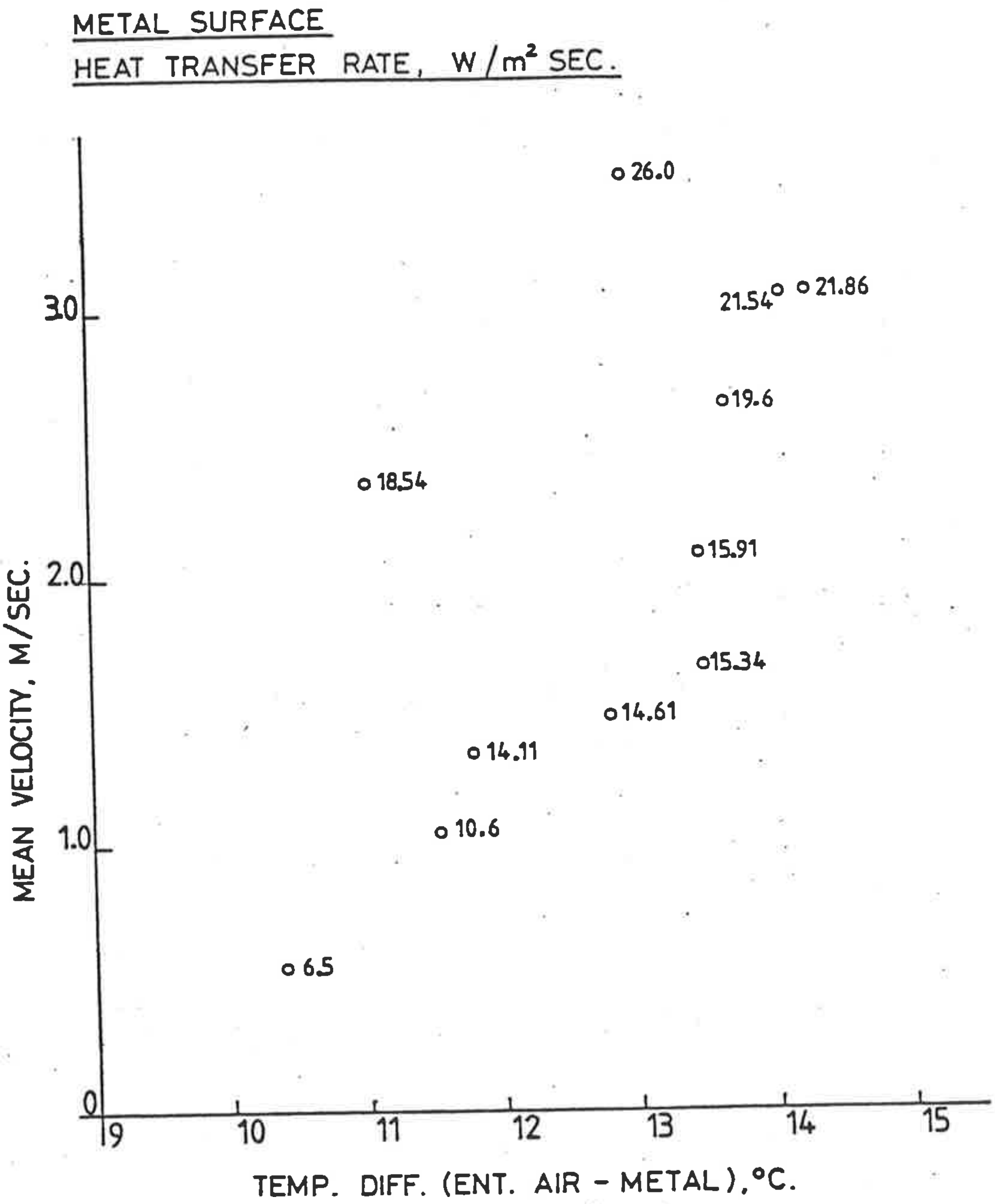
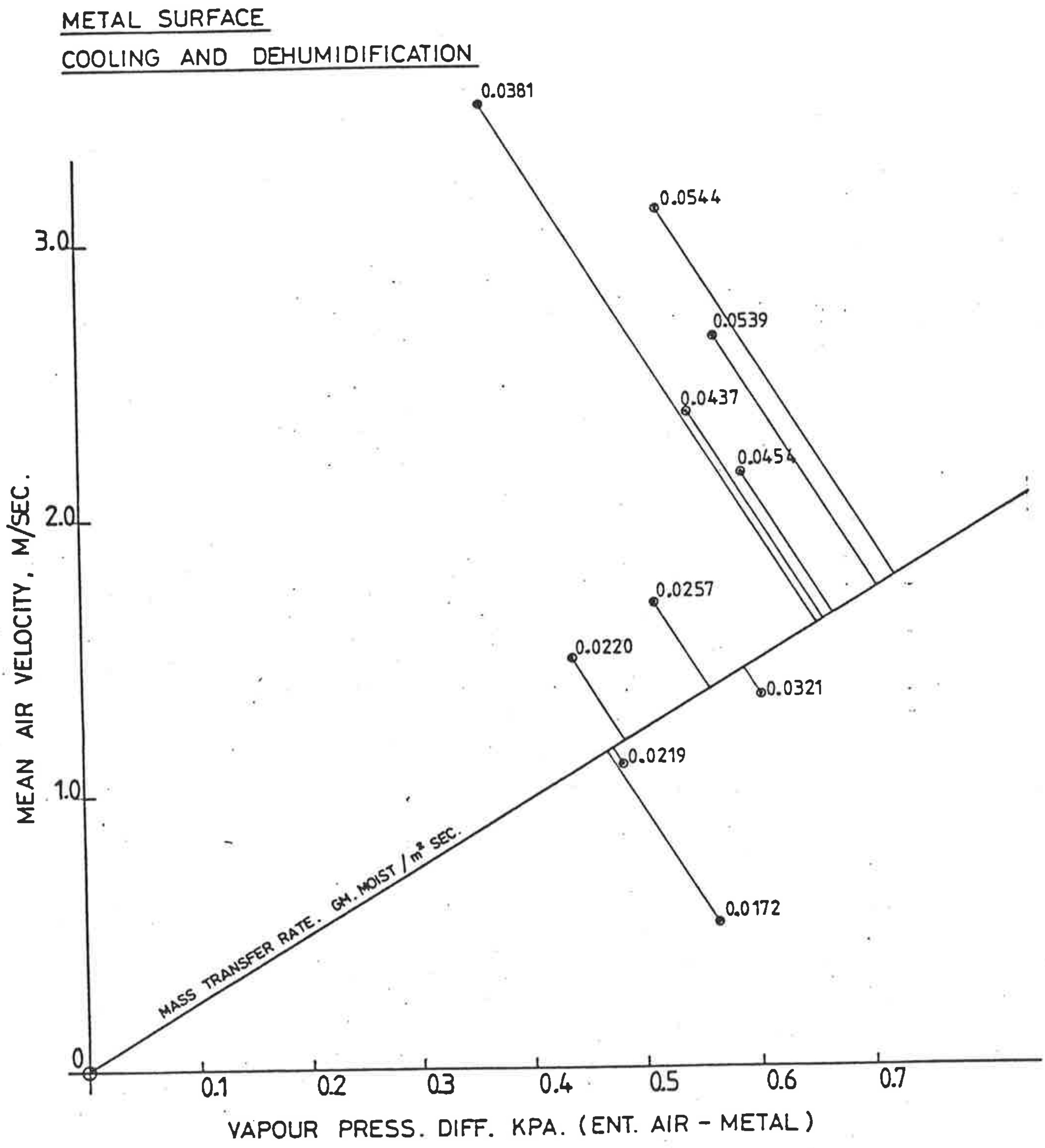


Fig. 31

Metal surface, mass transfer rates per unit area, cooling and dehumidification



5.9 CONCLUSION

The characteristics which dominate the heat and moisture transfer processes in these experiments are:

- the temperature differentials between:
  - the entering air and the mean surface water for the combined metal/water surface,
  - the entering air and the mean metal for the metal surface.

- the vapour pressure differentials between:
  - the entering air and the mean surface water for the combined metal/water surface,
  - the entering air and the mean metal for the metal surface.

- the mean air velocities,
- the configuration of the air tunnels,
- the roughness of the surfaces,
- the profile of the air flow,
- the characteristic length of contact,
- the wave forms of the exposed water surfaces.

Analyses of all these criteria are beyond the scope of the present study; the intention of this research is to assess the overall performance of the experimental units. The performances are recorded for essential characteristics, such as:

- the conditions of fresh air/return air mixture entering the experimental units,
- the conditions of air leaving the combined water/metal surface,
- the condtions of air leaving the metal surfaces,
- the mass air flow,
- the surface water temperature,
- the temperature of water at the bottom layers of the storage,

- the thermal energy supply by the energy conversion device,
- the primary energy input.

At any instant, the air velocity through the metal tunnels is higher than that through the combined metal/water surface chamber. The rates of heat and mass transfer for the combined horizontal surface are greater than those of the vertical metal surfaces for the same mean air velocity and the same temperature differential. The length and area of the metal/water surface at the same time are far less than those of the exposed metal surfaces.

The experimental data conform to the established thermodynamic principles, including the following:

- heat and moisture transfer rates increase with the increase in velocity and temperature differential,

- heat and moisture transfer rates decrease with the decrease in velocity and temperature differential,

- low mass transfer rates are the results of low Reynolds number,

- the portion of the fresh air/return air mixture FRA which remains unchanged when it passes through the thermal exchange chambers increases with the increase in velocity.

PRESSURE DIFFERENTIALS

The pressure differentials for the heat and moisture transfer have been kept at least values for these experiments. The static pressure differentials for the air treatment and the pressure differential for the circulation of water through the storage vessels have been measured.

A four row water cooling coil whose thermodynamic characteristics and air mass flow are similar to those of the experimental units was selected and the pressure differentials for both the fluids were measured.

Experimental units

Installation	Air quantity l/s	Static pressure differential Pa
Magill, SA	504	72
Darwin, NT	455	79

	Water quantity Kg/s	Total pressure differential kPa
Magill, SA	0.38	10.8
Darwin, NT	0.32	7.8

4-row Cooling Coil

Face Velocity m/s	Fins per metre	Air quantity l/s	Static pressure differential Pa
2.5	395	500	163

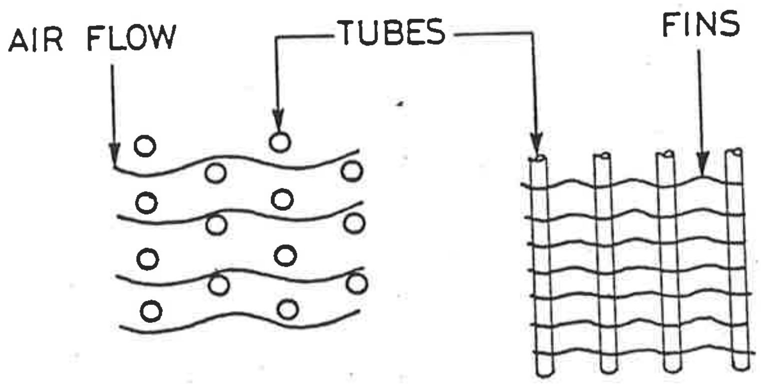
Water quantity l/s	Total Pressure differential kPa
0.59 l/s	30.3

The metal surfaces in the Magill installation are of galvanised sheet steel and those in Darwin installation are of sand-blasted aluminium sheet.

The surface areas and the lengths of contact with the transport fluid for an ordinary cooling coil are less than those in these experiments. The surface areas and the lengths of contact with the air for an ordinary cooling coil are nearly equal to those in these experiments.

Fig. 32

A typical cooling coil



In an ordinary cooling coil, as illustrated in Figure 32, the tubes carrying the transport fluid have been staggered for an intimate and prolonged contact between the air and the cooling surface. The rippled corrugated fin construction augments air turbulence for higher heat transfer, but at the cost of increased pressure drop.

In these experiments the fluid (water) is circulated through the vessels. The rate of change of mass of fluid within the vessels with respect to the total mass is kept to an absolute minimum value to optimise the stratification. The viscous shear forces on the separating membranes are reduced by minimising the magnitudes of the local velocities and the heat, mass and momentum transfer are optimised by increasing the contact time between the fluid and the transfer surfaces.

The design objectives as stated under Momentum, Energy and Matter Transfer of Section 1.1.4.6 are satisfied by reducing the magnitudes of the viscous shear forces for the fluid streams and by increasing the residence time.

SECTION 6

UTILISATION OF INTEGRATED STORAGE VESSELS/EXCHANGERS FOR SOLAR POWERED ENVIRONMENTAL CONTROL SYSTEMS

INDEX

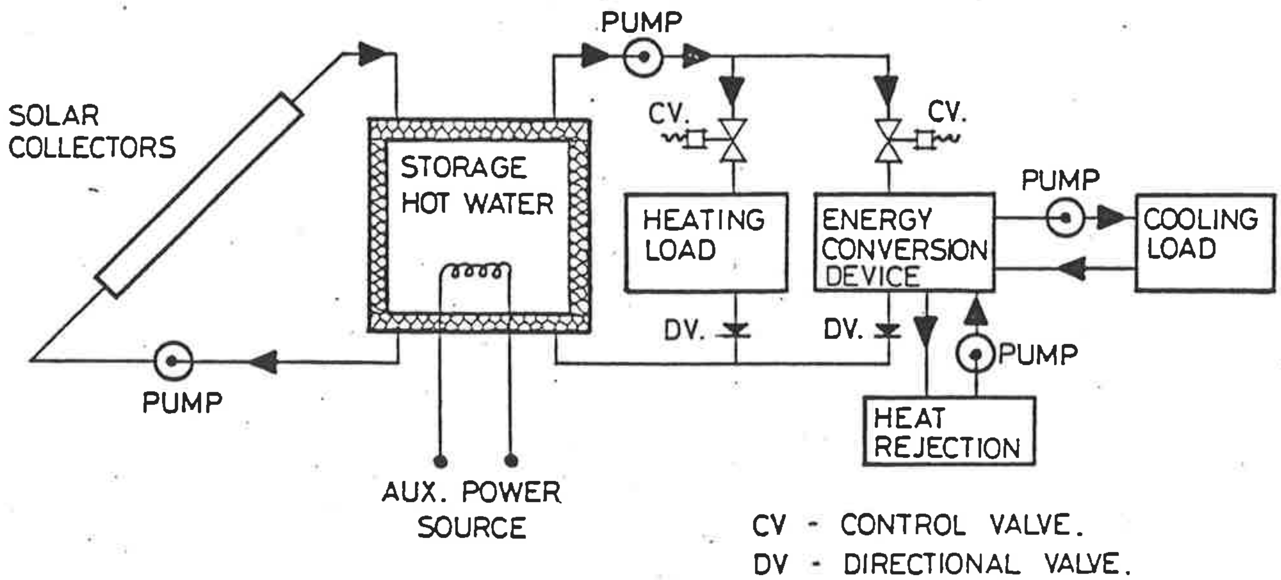
SECTION 6: UTILISATION OF INTEGRATED STORAGE VESSELS/EXCHANGERS FOR SOLAR POWERED ENVIRONMENTAL CONTROL SYSTEMS

6.1 GENERAL

Availability of solar energy is time-dependent. The energy requirements for an environmental control system are also time-dependent. The supply of thermal energy has to meet all of the requirements of the environmental control load.

The major elements of a solar powered environmental control system are: solar collectors, storage, conversion devices, loads, auxiliary energy supplies and the control system. The characteristics and performance of each of those elements is related to that of the others. A hot water store supported by an auxiliary power source forms a steady source of thermal energy supply. The energy flow for an ordinary solar powered environmental control system is illustrated below.

Fig. 33
Schematic Diagram of an Ordinary
Solar Powered Environmental Control System



Where sensible cooling and dehumidification is a predominant requirement, the sensible energy stored in the form of hot water needs to be converted into cold water by an energy conversion device to satisfy the cooling load.

When heating is required, the hot water is directly supplied to meet the heating load.

6.2 SENSIBLE COOLING AND DEHUMIDIFICATION

The temperature of the hot water required for the satisfactory operation of the Lithium Bromide - water absorption chiller is in the range of $82^{\circ}C$ - $88^{\circ}C$. considering an average ambient temperature of $34^{\circ}C$ DB, as is typical in Darwin, the temperature differential across the insulated wall of the hot water store is

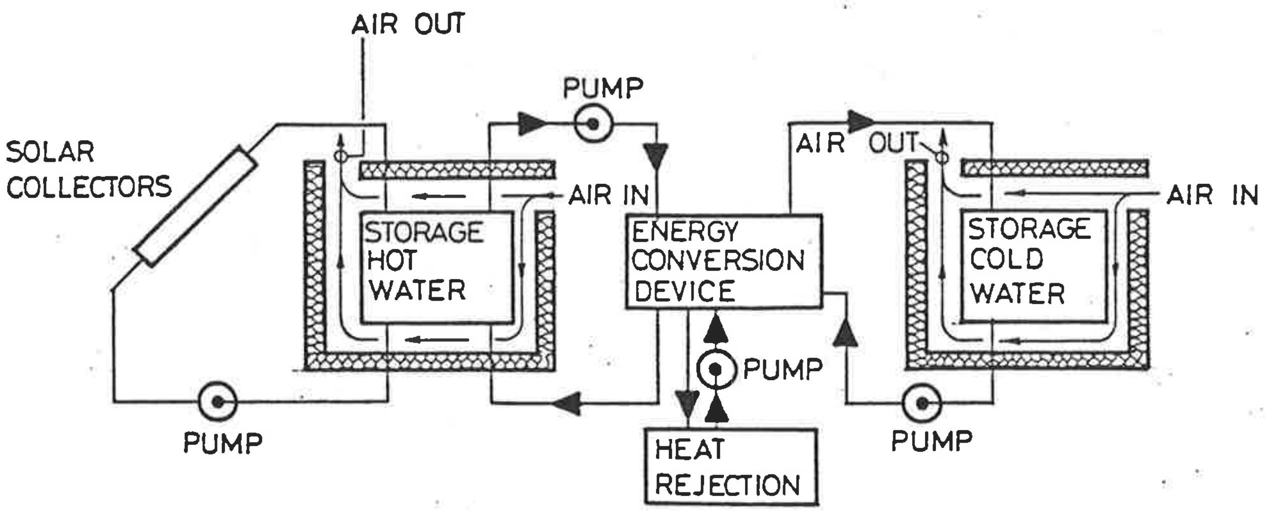
$$\frac{88 + 82}{2} - 34 = 51^{\circ}C \qquad (6.1)$$

assuming the mid range of allowable temperature. The rate of energy loss from the hot water storage is dependent upon the temperature differential, heat transfer rate and the areas of the insulated walls.

The present integrated storage vessels/exchangers have several storage vessels. Both hot and cold water can be stored within these vessels.

For a solar powered environmental control system, the storage vessels/exchangers can be incorporated as detailed in the following diagram.

Fig. 34
A Typical Solar Powered Environmental Control System with Energy Storage Vessels/Exchangers



A cold water storage can be introduced on the downstream side of the energy conversion device as shown in the diagram.

For sensible cooling and dehumidification process, when sufficient energy is collected and it is stored in the form of hot water, this energy can then be easily converted into the cold water by an absorption compressor. The cold water temperature range, that is required for the sensible cooling is $9^{\circ}C$ to $14^{\circ}C$. The temperature differential across the insulated storage wall then is

$$34 - \frac{14 + 9}{2} = 34 - 11.5 = 22.5^{\circ}C. \qquad (6.2)$$

this temperature differential is for an ordinary insulated cold water storage vessel.

For integrated storage vessels/exchangers, the average air temperature within the tunnels is approximately 17^{o}CDB and therefore the temperature differential across the insulated wall is

$$34 - 17 = 17^{o}C \qquad (6.3)$$

Where sensible cooling and dehumidification is a predominant requirement as in Northern part of Australia, the major part of the energy store can be in the form of cold water. Besides, by utilising energy storage vessels/exchangers, the advantage that can be obtained is the reduced rate of energy loss by minimising the temperature differential across the insulated wall by

$$51 - 17 = 34^{o}C \qquad (6.4)$$

The percentage by which the temperature differential can be reduced is approximately

$$\frac{(273 + 51) - (273 + 17)}{(273 + 51)} = \frac{34}{324} = 10.5\%$$

6.3 STORAGE VOLUME

The integrated energy storage vessel/exchanger system provide the facilities for both water storages - hot and cold.

When one litre volume of hot water is stored for the sensible cooling and dehumidification process, the cooling energy that can be available from this volume is

$$0.9684 \times (88-82) \times 4.197 \times 0.6 = 14.6 \qquad (6.6)$$

where

0.9684 = Density, kg/m^3
88 - 82 = 6 = usable temperature change of hot water assuming LiBr H_2O system, ^oC
4.197 = specific heat, kj/kg ^oC
0.6 = Absorption compressor efficiency
14.6 = energy content

When one litre volume of cold water is stored for the sensible cooling and dehumidification process, the cooling energy that can be available from this volume is

$$0.9989 \times (14 - 9) \times 4.189 = 20.92 \qquad (6.7)$$

where
0.9989 = Density, kg/m^3
14-9 = 5 = usable temperature change for cold water, ^oC
4.189 = specific heat, kj/kg ^oC
20.92 = Energy content, kj/litre

Where the sensible cooling and dehumidification process is a predominant requirement, further advantage can be taken of the cold water storage. The advantages of this is the reduction in storage volume for the same duty of the system. This reduction can be established from the unit litre energy storage capacity, i.e. $\frac{20.92 - 14.6}{14.6} = 0.432$ i.e, 43.2%

6.4 CONCLUSION

The integrated energy storage vessels/exchangers may have many storage vessels, the numbers of which are determined from the requirements of the storage capacity and the surface areas, which are determined from the heat and mass transfer requirements. The hot water store and the cold water store as shown in the Fig. 34 can be the integral parts of the energy storage vessels/exchangers.

Where the sensible cooling and dehumidification process is predominant, reduction of the rate of energy loss and in the storage volume can be achieved by the incroporation of energy storage vessels/exchangers of the type studied in this thesis.

SECTION 7

DISCUSSIONS, CONCLUSIONS AND FUTURE WORK

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7.1 DISCUSSIONS AND CONCLUSIONS

7.1.1. Buildings

Environmental loads when rising from least values to peak values over a period of time are always higher for a light-weight building than for a medium-weight or a heavy-weight building.

Where higher average daily temperature ranges exist, the ambient temperatures and other external energy levels move at a faster rate over a period of time between low and high extremities. In these areas, external energy levels require a longer period of time before they can bring any change in the internal energy levels of a medium-weight or a heavy-weight building. For a light-weight building, these changes occur within a comparatively shorter period. These time lags are caused by the thermal diffusivities of building materials. The ambient energy levels between least values and peak values pass through an acceptable band of energy levels. The duration period of these acceptable bands varies widely. When the duration period is lengthy, the building can be ventilated by mechanical means with ambient air to reduce the expenditure of traditional forms of energy.

Where lower average daily temperature ranges exist, the ambient temperatures and other external energy levels move at a slower rate over a period of time between low and high extremities. Again in these areas, the values of lower extremities are in close proximity of the internal energy levels. In these areas, the instantaneous environmental control load of a light-weight building is greater than that of a medium-weight or a heavy-weight building when the load is in the rising mode. But after the peak load occurence when the solar load declines, the instantaneous load of a light-weight building is less than that of a medium-weight building or a heavy-weight building. These higher loads for medium to heavy-weight mass buildings are caused by the higher building fabric storage factors.

Medium-weight or heavy-weight buildings have advantages with regard to the decrease of the peak load values but they tend to be more expensive, particularly where transportation of materials over long distances is a major part of the cost. The suggestion of Szokolay (1981) was to explore the possibilities of locally available materials and to commercialise these resulting techniques. On the other hand, an economic compromise can be made by the acceptance of a vessel/exchanger system for a light-weight building where energy can be drawn from the artificial store of a vessel/exchanger system to satisfy peak load of a light-weight building.

7.1.2 Environmental Control Systems

Both the compressor of a vapour compression refrigeration cycle and the absorber/generator of an absorption cycle attain a higher refrigerating effect for a given mechanical and external energy input when there are large favourable energy driving forces available between the condenser and the surrounding environment. During this favourable time period, the actual load is only fraction of the peak load and compressors and absorbers/generators are only partly loaded for a conventional environmental control system. Instead of matching the thermal energy supply to the instantaneous environmental control load, equipment supplies thermal energy directly to the store for a vessel/exchanger system. The equipment is mostly fully loaded during this higher refrigerating effect period for a vessel/exchanger system and this collected energy satisfies later peaks in the load demand.

The thermodynamic efficiency of the condensing units at full load operating conditions is greater than that at part load operating conditions. For the vessel/exchanger system, the size of the condensing units are lower than those of the conventional environmental control system. The electrical and other external energy consumption is further reduced for the vessel/exchanger system by operating the lower capacity condensing units at higher load ratio levels at part load conditions than operating higher capacity condensing units at lower load ratio levels. Besides, the full load operating time of lower capacity condensing units have been made longer than that of higher capacity condensing units.

In an ordinary cooling coil, the total energy transfer is maximised by increasing the convective transfer coefficients through increasing the local velocities of fluid over the transfer surfaces. For a given mass flow and fixed transfer area this implies an increase in the number of rows of coils. When the number of rows is increased, the effectiveness of downstream rows decreases as the driving force for the transfer process decreases with each successive row.

For the vessel/exchanger system, the secondary transfer surfaces in the form of fins are totally eliminated. The primary surface areas have been increased to compensate for the loss of the secondary surface. The viscous shear forces for the fluid streams are reduced by minimising the magnitude of the local velocities. Residence time is increased by the continued effect of larger primary transfer area and reduced flow velocities. Thus at the possible expense of having a less compact heat and mass exchanger and larger primary transfer surface areas, the power consumption necessary to achieve a given exchange is reduced.

In tropical areas, where dehumidification is a significant requirement, sometimes overcooling is required to remove the excess moisture present in the air. To maintain the built environmental conditions within a range of prescribed values, necessary sensible heat is then added after the overcooling process.

In a conventional environmental control system, this sensible heat is supplied from an another external source at the expense of additional traditional forms of energy.

The traditional forms of energy expended to achieve this re-heat requirement for a conventional environmental control system is totally eliminated for the vessel/exchanger system by the incorporation of heat transfer facilities between the warmer fresh air or fresh air/return air mixture stream and the colder air stream leaving the thermal and vapour treatment chambers of the vessel/exchanger unit.

7.1.3 Energy Storage

The capacities of the energy conversion devices such as refrigeration compressors and absorbers/generators have been reduced by the addition of thermal stores for the vessel/exchanger systems. By the incorporation of artificial thermal store, the additional energy plus plant capacities which fill the store in times of low demand satisfy the later peaks of high demand. The duration of these peaks is relatively short and thus the size of the store can be kept reasonably small.

Thermal losses are of critical importance. When the storage is made large enough to satisfy the loads for a considerable period, it becomes economically impractical. To recover some of the thermal losses from short-time storage, air tunnels have been incorporated around the storage vessels of the vessel/exchanger systems. This reclaimed energy is then utilised for the thermal and vapour treatment of moist air supplied to the built environment.

7.2 FUTURE WORK

Analyses of all physical and thermodynamic criteria are beyond the scope of present experiments. Further work is needed to investigate.

- the shape of the storage vessels

  The shape of the storage vessels dictates the configuration of air tunnels. The separating membranes which separate the stored fluid from the air stream are responsible for thermodynamic exchange.

- the waves formed on the exposed water surfaces

  When the moist air passes over the exposed water surfaces, the waves are formed on those surfaces. The shapes of the waves differ with different air and water velocities.

- the roughness of the metal surfaces

  The metal surfaces separate the stored fluid from the stream of moist air. The roughness of these surfaces has greater impact on the thermodynamic exchange.

- <u>optimisation of the sizes of system components</u>

  The trade-off between capital and operating cost will lead to different distribution of the sizes of components in the system for different climatic regions and building types. A general optimisation method is highly desirable.

APPENDIX A1

ECONOMIC EVALUATION

INDEX

SECTION 8: ECONOMIC EVALUATION

8.1 GENERAL

Two systems have been considered for economic evaluation.

A conventional airconditioning unit, designated by System X and Energy storage vessels/exchangers designated by System Y.

The capital costs for the common components such as ductwork and air filters are deleted from both systems for the comparative evaluation of costs and energy.

The economic factors considered for evaluation are:

- capital recovery in terms of amortisation and depreciation,
- capital, operating and maintenance costs,
- useful life span of the plant,
- expected rate of increase of price for traditional forms of energy,
- expected rate of inflation.

8.2 ECONOMIC APPROACH

Assuming

- extra capital cost of vessels/exchangers Co($/kW of refrigeration)
- current electrical and other changes Fo($/Unit of energy)
- annual energy savings by the vessels/exchangers over a conventional airconditioning system E(units of energy)
- expected rate of increase of electrical and other charges Y_e(percent per annum)
- expected interest (discount) rate i_c(percent per annum)
- expected inflation rate J(percent per annum)
- life cycle n(years)

effective rate of increase in electrical and other charges becomes

$$Y = \frac{Y_c - J}{1 + J} \qquad (8.1)$$

and

$$i = \frac{i_c - J}{1 + J} \qquad (8.2)$$

Then the present worth of savings S in year n can be represented by

$$S = EF_o \frac{[1+Y]^n}{[1+i]^n} \qquad (8.3)$$

The extra capital cost Co for the vessel/exchanger system is also subject to the interest rate ic and the inflation rate J. Therefore the effective interest rate is applied to Co. Thus compound interest Coi calculated for each consecutive year is applied to the extra capital cost Co for the vessel/exchanger system.

The actual pay-back period of 'p' years can be calculated by deducting the compound interest Coi for any year from the present worth of savings S for that year.

For ease of comparison, the unit cost is based on each kilowatt of refrigeration and the operational time for both systems has been considered as 24 Hrs/day for 365 days per annum.

8.3 CAPITAL COSTS

Table I is drawn to provide the details of capital costs for the installations located in Darwin. The conventional airconditioning unit is one of the batch production, whereas the vessel/exchanger system is one-off production.

TABLE VIII.I

ITEMS	CONVENTIONAL UNIT (SYSTEM X) $/kW OF REFRIGERATION	VESSELS/EXCHANGERS (SYSTEM Y) $/kW OF REFRIGERATION
Refrigeration condensing set and controls	149.30	-
Evaporation coil, TX valve and refrigeration piping	35.20	-
Casing	38.70	-
Supply air fan	41.20	36.80
Controls	25.80	33.70
Refrigeration chiller and controls	-	183.90
Energy storage vessels exchangers	-	181.40
TOTAL:	290.20	435.80

8.4 OPERATING COSTS

Table II is drawn to provide the details of annual energy consumption for two systems - conventional and vessels/exchangers

TABLE VIII.II

QUARTER	CONVENTIONAL UNIT SYSTEM X KWH/KW OF REFRIGERATION	VESSELS/EXCHANGERS SYSTEM Y KWH/KW OF REFRIGERATION
First	757	616
Second	404	345
Third	410	351
Fourth	521	438
TOTAL:	2,092	1,750

The electrical tariff for Darwin is as follows:

First	300	Kwh	units	-	20.64 cents/each
Next	1200	"	"	-	15.01 cents/each
Next	4500	"	"	-	11.92 cents/each
Next	24000	"	"	-	10.01 cents/each
Next	60000	"	"	-	7.48 cents/each
Remainder		"	"	-	7.04 cents/each

Considering an average tariff of 14.4 cents per unit of Kilowatthour and 41 cents per Kilolitre of water, the operating costs for the two systems are:

TABLE VIII.III

ITEM	CONVENTIONAL UNIT (SYSTEM X) $/kW OF REFRIG- ERATION/ANNUM	VESSELS/EXCHANGERS (SYSTEM Y) $/kW OF REFRIG- ERATION/ANNUM
Energy	301.20	252.00
Water	-	1.10
TOTAL:	301.20	253.10

8.5 MAINTENANCE COSTS

Table IV is drawn to provide the details of annual maintenance costs for two systems - conventional unit and vessels/exchangers. Actual costs incurred are divided by the total kilowatts of refrigeration.

TABLE VIII. IV

	CONVENTIONAL UNIT (SYSTEM X) $/kW OF REFRIGERATION	VESSELS/EXCHANGERS (SYSTEM Y) $/kW OF REFRIGERATION
Maintenance cost	8.30	4.60

PRESENT WORTH OF SAVINGS

Considering

$n = 15$ years

$ic = 13\%$

$J = 8\%$

$Yc = 10\%$

We have

$$Y = \frac{.1 + .08}{1 + .08} = 0.0185$$

$$i = \frac{.13 - .08}{1 + .08} = 0.0463$$

The factor $E.F_o$ of equation (8.3) can be represented by the summation of the differences of operating and maintenance costs from the tables VIII.II and VIII.IV

$$EF_o = (301.20 - 253.10) + (8.30 - 4.60)$$
$$= 48.1 + 3.7 = 51.8$$

Table V is drawn to provide the details of PRESENT WORTH OF SAVINGS for the expected life span of the installations.

The factor Co is determined from the difference of capital costs given in Table VIII.I.

TABLE VIII.V

NUMBER OF YEARS	FACTOR E.Fo $	PRESENT WORTH SAVINGS S $	EXTRA CAPITAL COST FOR VESSELS/ EXCHANGERS	COMPOUND INTEREST ON Co Coi $	SIMPLE PAY-BACK $ (S-Coi)
1	51.80	50.42	145.6	6.74	43.68
2	51.80	49.08	145.6	7.06	42.02
3	51.80	47.78	145.6	7.40	40.38
4	51.80	46.50	145.6	7.70	38.80
5	51.80	44.06	145.6		
6	51.80	44.06	145.6		
7	51.80	42.89	145.6		
8	51.80	41.75	145.6		
9	51.80	40.64	145.6		
10	51.80	39.56	145.6		
11	51.80	38.51	145.6		
12	51.80	37.48	145.6		
13	51.80	36.49	145.6		
14	51.80	35.51	145.6		
15	51.80	34.57	145.6		
					164.88

The extra capital cost for the vessel/exchanger system is $145.60. By the present worth of savings method, the simple pay-back period for the vessel/exchanger system is approximately four years.

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