A Heat Pump Dehumidifier assisted Dryer for Agri-foods

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ABSTRACT

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The motivation of the research presented in this thesis was to investigate the potential of using a commercial 2.3 kW heat pump dehumidifier (HPD) simultaneously as a dryer for high-moisture agricultural products and for other domestic dehumidification/heating applications. A drying system incorporating the HPD was designed and constructed, along with instrumentation to gather data on the properties of process air as well as real-time weight of the material being dried. The HPD was equipped with an external water-cooled condenser that rejected excess heat out of the system. The design of the system allowed for conducting drying with recirculation of air as well as use of electrical heaters. In an open mode, the drying could be carried out simultaneously with room dehumidification and water heating in the secondary condenser.

The drying experiments were conducted with apple, tomato and agar gels. The system was found to be more effective in drying of material with higher amount of free moisture such as tomato. Comparisons were made between HPD assisted drying (partial and complete) and hot air drying (at 45°C and 65°C) in the same system using apple as the test material. Colour changes (L*a*b* values) in the samples were compared between treatments. It was observed that the degree of undesirable colour change was least in case of the HPD assisted system. The HPD dried fruit exhibited better rehydration properties than the hot air dried samples. Water activity of the HPD dried samples was noticeably lower than that of the hot air dried samples at the same water content, indicating that the residual moisture was probably held under higher tension. Histological observation indicated that there was a lesser degree of damage to the cellular structure of apple when dried with the HPD than when dried with hot air alone.

In terms of energy consumption, the process of HPD assisted drying is more

expensive. Much of the energy input is rejected at the secondary condenser as excess heat. Unless this heat is recovered for another purpose, or the system is modified to reuse it for drying, the drying process must carry this loss entirely. The specific moisture extraction rate (SMER) for apple was as low as 0.1 kg per kWh with the HPD assisted system. The SMER values for drying at 45°C was 0.5 kg per kWh and was almost 0.8 kg per kWh at 65°C.

The HPD assisted drying system demonstrated the ability of heat pumps to link different energy related activities viz., drying, space dehumidification and water heating. The energy expenditure is expected to be impressive when considered for all the related applications. The concept of utilizing heat pumps on farms to link up different energy streams for better utilization of the low-grade heat sources is discussed. A possible drying efficiency assessment in the form of exergy-based evaluation is proposed.

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RÉSUMÉ

Séchoir muni d'un système de déshumidification par pompe à chaleur pour le séchage de produits agro-alimentaires

Un séchoir, muni d'un système de déshumidification par pompe à chaleur (DPC), a été conçu, construit et instrumenté afin d'effectuer une étude suivie du séchage, des propriétés de l'air et du poids des matériaux séchés au cours du procédé. L'unité DPC a été équipée d'un condenseur externe, refroidit à l'eau, qui rejette l'excès de chaleur à l'extérieur du système. La conception de ce système permet de sécher en mode de recirculation de l'air, de même qu'avec l'ajout d'éléments chauffants. En circuit ouvert, le séchage peut être effectué avec déshumidification de la cavité, et chauffage de l'eau dans le deuxième condenseur.

Des tests de séchage ont été effectués avec des pommes, des tomates et de gélose agar. Le système s'est avéré être le plus efficace lors du séchage de matériaux à fort taux d'humidité libre comme la tomate. Des tests comparatifs ont été effectués avec des échantillons de pommes qui ont été séchés à l'air chaud à 45°C et 65°C, et en utilisant partiellement l'unité DPC et les éléments chauffants. Les changements de couleur (facteurs L, a et b) des échantillons ont été comparés. Il a été observé que le développement de changement de couleur non désirable a été le moins prononcé dans le cas des échantillons séchés grâce à l'unité DPC. De plus les échantillons séchés par l'unité DPC ont présenté une meilleure capacité de réhydratation face aux échantillons séchés à l'air chaud. Pour un même taux d'humidité, l'activité de l'eau a été nettement réduite pour les échantillons séchés par DPC, indiquant ainsi que le taux d'humidité résiduel est maintenu sous plus forte pression. L'analyse histologique indique une réduction des dommages causés à la structure cellulaire lors du séchage DPC en comparaison avec le séchage à air chaud.

En ce qui à trait à la consommation énergétique le procédé de séchage par déshumidification par pompe à chaleur est plus onéreux. Avec le rejet d'une large fraction de l'énergie apportée au système au niveau du deuxième condenseur, la température de l'air de séchage se maintient à moins de 30°C, ce qui amène un ralentissement du taux de séchage. Le taux spécifique d'extraction de l'humidité des pommes était aussi bas que 0.1 kg par kWh

avec l'unité DPC. Le taux spécifique d'extraction de l'humidité pour le séchage à l'air chaud à 45°C et à 65°C était de 0.5 et de 0.8 kg par kWh respectivement. De nouvelles stratégies visant à utiliser la chaleur perdue au deuxième condenseur sont mises de l'avant.

Le séchoir à DPC a fait la preuve de la capacité d'une pompe à chaleur de faire le lien thermo-énergétique entre le séchage, la déshumidification et le chauffage de l'eau. Le concept de l'utilisation d'une pompe à récupération de chaleur à la ferme est spéculatif quant à son potentiel d'une meilleure utilisation des sources de chaleur. L'absence d'un protocole d'utilisation en séchage est également discuté et une solution est proposée sous la forme d'une évaluation de l'exergie, la partie de l'énergie d'un système thermodynamique qui peut être effectivement transformée en travail.

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CHAPTER 1
INTRODUCTION

The changes brought about in the biosphere during the recent history of mankind, especially in the last century, are alarming. The rapid pace of development in science and technology, coupled with the ever-changing lifestyles of modern societies, have laid immense stress on natural resources as well as on the global ecological balance. The demands of modern civilization have been, and are being, satisfied at the expense of the finite fossil fuel deposits on earth, the disastrous effects of which are essentially intangible on a localized time scale. The nature of scientific research and technological development has evolved in response to the wake-up calls. The focus is now shifting gradually towards seeking changes that are in harmony with the ecological balance and weigh less heavily on the social conscience.

Drying, a major unit operation in agri-food industry, carries a huge environmental cost. The partial or complete removal of water from biological materials is a complex process that requires large amounts of energy. The influencing factors such as the process time, quality of the products, their heat sensitivity etc., produce processing regimes that are often a compromise between altruism and pragmatism. Adoption of eco-friendly drying technologies is slow due to many factors, but short-term or immediate economic profitability is often the underlying reason. Applied research in drying has to focus on this issue and demonstrate the different facets of alternate technologies in order to educate the consumers of the ulterior concerns. The use of heat pumps promises economic and ecological benefits and significant amount of research has been carried out on their use for drying applications, but the efforts have not resulted in their extensive adoption on farms for drying of agricultural products.

In the conventional hot air dryer, air is heated up to the drying temperature (using electrical heaters or heat exchangers consuming fuel) to enhance the heat transfer rate into the drying load. This increases the internal vapour pressure and the moisture diffusion rate in the material towards the surface, from where it diffuses into the process air. In a conventional situation, the absolute humidity of the process air is dependent on the ambient conditions. The use of a heat pump dehumidifier (HPD) in this circuit enables control over the moisture content and the temperature of the process air, as well as the recovery of the latent heat of vaporization of water from the exhaust stream that is otherwise lost as waste heat.

A heat pump is a device that transports energy from a low temperature source to a higher temperature sink; this transfer requires an input of work, which may be supplied mechanically as in a vapour-compression cycle, or as heat, in an absorption cycle (Figure 1.1). The most common type of heat pump operates on the vapour-compression cycle and a basic unit consists of the evaporator, compressor, condenser and the expansion valve (Figure 1.2).

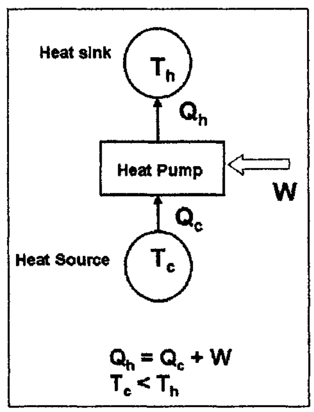

Figure 1.1 Operation of a heat pump

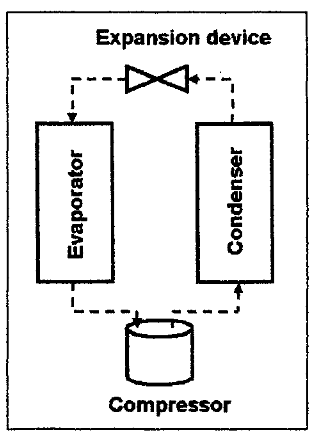

Figure 1.2 Basic components of a vapour compression heat pump.

Heat transport is achieved through phase change of the working fluid (refrigerant). The refrigerant in the evaporator absorbs heat and vaporizes at low pressure and temperature. As the vapour condenses at a higher pressure in the condenser, it rejects heat at higher temperature. When used in a drying system, the HPD cools the process air first to saturation, and then further for condensation of water (dehumidification), thus increasing the drying potential of air. In the process it also recovers low-grade heat (sensible and latent) from the air, which is made available at the condenser as sensible heat of higher quality. Two configurations of the evaporator and the condenser are possible and are shown in Figure 1.3.

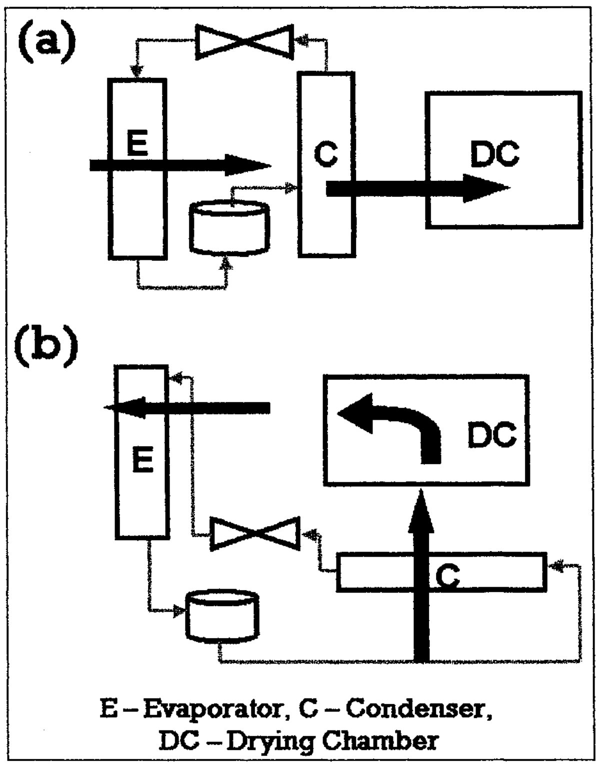

Figure 1.3 Two configurations of the heat exchangers *vis-à-vis* the drying chamber. The dark, thick streams indicate airflow.

In the first case (Figure 1.3a), the HPD operates both as a dehumidifier and a heater for the process air. In the second configuration, the evaporator interjects the humid exhaust stream while fresh air is taken in over the condenser. In this type of placement, the latent heat (along with a quantity of sensible heat) is recovered by dehumidification of the exhaust and is transferred to the process air via the condenser. This configuration is preferred when the ambient air is dry (low relative humidity), but is not very economical during the final stages of drying, as the exhaust stream almost resembles the inlet air. In both configurations, the exhaust from the drying chamber could be returned to the evaporator i.e. the process air could be recycled, completely or partially.

The application of heat pumps in agriculture started out with their use as supplementary devices for heating. Subsequent research and development has resulted in development of drying processes that run solely with a heat pump. Different strategies, such as the use of pressure regulating valves, multiple heat exchangers, airflow controls, variable speed compressors etc., have been worked out to deal with the practical complications of HPD operation. The commercial use of HPD assisted dryers has been reported in many parts of Europe (Norway, France and The Netherlands), Asia and Australia, where the technology has been applied mostly in the marine food-processing sector. Reports indicate that HPD assisted drying processes consume less energy compared to conventional processes.

Despite these demonstrations, large-scale adoption of heat pumps has not materialized in agriculture. One of the main factors for the reluctance to adopt this technology, either to supplement heat or as a main component of the drying process, is the capital cost. Due to their high initial cost and long life spans, existing agricultural dryers occupy a strategic position in the budget of a farm. Hence, their replacement or modification requires careful positioning of the new product offered to the consumers. Heat pump technology has the potential to offer a variety of applications in an agricultural environment and the exploitation of this versatility needs to be investigated and proven. The ability of heat pumps to transport heat from different low-grade sources could be harnessed for efficient energy budgeting by linking up the energy streams on a farm. With this view, an attempt has been made to investigate the feasibility of using a heat pump dehumidifier assisted dryer for agri-food products and the energy aspects of the process.

1.1 Hypothesis

A versatile heat pump dehumidifier can be used effectively in a drying system for agri-food products.

1.2 Objectives

- Design and build a versatile heat pump dehumidifier assisted drying system with instrumentation and devices for operation in different modes.
- Establish the accuracy and resolution of the system for various parameters to be measured.
- Carry out drying experiments in the HPD assisted system with agri-food material and describe the drying characteristics.
- Describe the energy consumption pattern of the system.
- Assess the quality of the material dried with the HPD assisted system.

1.3 Scope of the study

The heat pump dehumidifier used in the study is not a dedicated system (as described by different authors in the literature) and is used for multiple applications simultaneously (described by the term versatile). It was set out to study the feasibility of adopting such a system for practical drying of agri-foods. Different parameters that are of concern for a drying system such as time, energy consumption and product quality were identified for measurement.

CHAPTER 2
REVIEW OF LITERATURE

The use of heat pumps in the power sector, for building heating/cooling and space climate control is well known and has been in practice for a long time. Reay and Macmichael (1988), McMullan and Morgan (1981) and Heap (1979) have dealt with the fundamentals of operation and such applications of heat pumps. The agricultural and agri-food processing applications fall within the perspective of the current work and the related literature was reviewed along with other relevant literature pertaining to fundamentals of drying, design and construction of dryers, psychrometrics and energy aspects of drying.

Some early papers discuss the issues related to adoption of heat pumps for industrial processes and are insightful. Newbert (1982) comments on the usefulness of heat pumps in recovering waste heat as he takes a sober view of their role in industrial processes. According to this article, heat pumps are desirable only under certain conditions such as operation over long periods within a narrow temperature range, and are to be considered only when other energy saving measures are not practical to achieve. Another paper presented at the same meeting (Oliver, 1982) discusses the use of heat pump dehumidifiers (HPD) in process drying (textile, timber and clay products).

2.1 Fundamentals Of Drying And Drying Equipment

The theory and fundamental principles of drying have been discussed by many authors in different books (Vega-Mercado et al., 2001; Mujumdar and Menon, 1995; Pakowski and Mujumdar, 1995; Marinos-Kouris and Maroulis, 1995; Strumillo and Kudra, 1986; Keey, 1972,1978; Williams-Gardner, 1971). In addition to introducing the physico-chemical concepts associated with food dehydration and psychrometrics, Vega-Mercado et al. (2001) classify the different commercial drying systems into four generations. They are –

- Dryers for solids – convective dryers such as cabinet and bed dryers.

- Dryers for slurries and purees such as spray dryers and drum dryers.
- Freeze dryers and osmotic dehydration systems.
- Dryers involving hurdle approach or multiple drying techniques such as fluidization, use of dielectric heating, vacuum etc.

Mujumdar and Menon (1995) also discuss classification, selection and design of dryers. Molnar (1995) presents the different aspects of experimental work related to drying. Accordingly, the general aim of drying experiments are listed as:

- Choice of adequate drying equipment.
- Establishment of the data requirements.
- Investigation of the efficiency and capacity of the existing drying equipment.
- Investigation of the effect of the drying conditions on the final product.
- Study of the mechanism of drying.

Different experimental techniques for determination of the associated parameters such as moisture content, sorption equilibrium characteristics, thermal conductivity, effective diffusivity etc. are discussed. Focus on drying of food material deals with important factors such as the objectives of drying food, residual moisture content for lengthened storage, properties of foods, optimum drying techniques, types of suitable dryers and changes in food associated with drying (Sokhansanj and Jayas, 1995). Raghavan (1995) describes the different drying equipment used on farm for crop drying, essentially drying of grains, and their features. Jayaraman and Dasgupta (1995) take a closer look at drying of fruits and vegetables.

Land (1991) has compiled the necessary information for the selection and design of drying systems for different applications. Similarly, the book by Greensmith (1998) is a good source of information for a "practical dehydrator". Beginning with a brief history of drying as an industry, the text also contains an overview of the commercial drying practices in parts of Europe, Africa and Asia. Different types of dryers, the factors influencing the selection, drying of different fruits and vegetables, the preparatory processes, quality control and the economics of dehydration are discussed.

2.2 Heat Sources On Farm

There are three basic sources of heat in nature that could be utilized with heat pumps - air, water and the earth. The earth represents an inexhaustible supply of heat that is usually exploited for residential space heating/cooling applications using geothermal heat pump systems. Large bodies of water have been shown to be effective in acting as source/sink of heat that could be used effectively with heat pumps (Murphy and Brewer, 1997).

One of the universal sources of low-grade heat, and perhaps the most common for heat pump applications, is the air. Various farm operations add low-grade heat to air in the form of exhaust gases and water vapour. Animal housing provides an appreciable amount of low-grade heat in terms of sensible heat and moisture in the ventilation stream. It is remarked that the barn (cattle) exhaust air contained sufficient latent heat that could, if harnessed, exceed the heating needs of a farmhouse (living quarters) during winter. In this study (Zabliski, 1985) conducted on farms with varying herd sizes (35-190 animals), the sensible heat produced by the animals was estimated to be between 46 and 120 kW, and the latent heat availability was 22-77 kW. According to Bucklin et al. (1992), cattle release 1.9-3 W per kg body weight, swine 2.3-5.9 W, sheep 1.2-1.3 W and poultry, 5.8 W on an average. For fattening pigs of 60 kg body weight, the heat loss is estimated to be 8.3 MJ per day (96 W) and the latent heat released is to the tune of 6.2 MJ per day or 72 W (Woods, 1979). Not only is this heat a waste, but also detrimental to the animals, and hence needs to be removed by costly ventilation arrangements. Heat stress among cattle is a well-documented phenomenon - conditions of high temperature and humidity result in decreased milk production and reproductive performance (Mayer et al., 1999). In one study in the USA (Hahn and Osburn, 1969), losses in milk production related to heat stress during summer was estimated between 25 kg in Maine and almost 450 kg in Texas, for a cow producing 30 kg per day. Greenhouses represent another major source of waste heat; the air is refreshed periodically in an attempt to maintain favourable conditions. The annual energy "penalty" (excess energy) required for operation due to venting off air to maintain the humidity was estimated to be between 144 and 333 MJ per m^2 of greenhouse floor area (Johnstone and Ben-Abdallah, 1989). Climate control in buildings and farm installations is a major operation that results in discharge of large amounts of waste heat.

2.3 Heat Pumps In Agricultural Drying

Heat pumps have been studied for use on farms since the early 1950s and have found applications in sectors such as dairy, grain drying, timber drying etc. Most of the early studies outlined the many benefits of using heat pumps from the energy recovery point of view, but deemed them uneconomical compared to the fuel prices existing at that time. The spiraling fuel costs due to the oil crisis of the early 1970s saw interest revive in the heat pump, and in its ability to dehumidify. Reduced supply of fossil fuels and the need for energy conservation prompted investigation into the use of alternate heat sources for grain drying. Since then, many studies have been carried out in this area, and different processes have been developed for various drying applications such as timber and malt drying in Germany, fish drying in Norway, air conditioning and dehumidification of animal sheds and green houses (Toal et al., 1988a). In Norway, low temperature drying studies have been carried out using a heat pump dryer, and biomaterials were dried at temperatures as low as -25°C (Alves-Filho and Strommen, 1996a,b). Perera and Rahman (1997) and Hesse (1995) provide a general review of heat pump dehumidifier assisted drying. Mason et al. (1994) and Britnell et al. (1994) present the Australian perspective of the field.

Lai and Foster (1977) refer to two unpublished Master's theses (Davis, C.P. Jr., 1949, Purdue Univ., Lafayette, IN and Shove, G.C., 1953, Kansas State Univ., Manhattan, KS) that studied the adaptability of heat pumps for grain drying. The paper by Flikke et al. (1957) is perhaps the earliest of published works in the area of heat pump assisted grain drying. In general, they found the concept to be mechanically feasible but not attractive economically due to low fuel prices prevailing at the time. However, the simultaneous utilization of the heating and cooling capacities of the heat pump was always considered desirable. In their study, they controlled drying air temperature by changing the mass flow of the refrigerant at different air flow rates and showed that at lower air flow rates, the system had a better performance. An experimental grain dryer was constructed and tests were carried out over a wide range of ambient and inlet conditions. The two control variables used in the study were airflow and drying air temperature, the airflow covering a range of 550 to 2000 m^3 per hour, and the temperature of the air between 43 and 54°C. The best specific moisture extraction rates (SMER) were obtained at airflow between 800 and 1000 m^3 per hour. An optimum energy consumption of 2.8 MJ per kg of water removed was reported. After the oil crisis of 1970s, there was a revival of interest and the nature of research also took a different direction. The focus shifted more on the principles and modeling, with limited work on large-scale equipment and proto-types. Hodgett (1976) demonstrated that energy consumption for drying grains could be reduced significantly using a heat pump. Lai and Foster (1977) designed a heat recovery system comprising of a heat pump and a heat pipe, for grain drying. It was reported that energy reduction was possible; however, the cost of the heat recovery equipment exceeded the cost of the dryer.

Hogan et al. (1983) developed a heat pump system for low temperature grain drying. Tested over several seasons, their system was found to be economically desirable when used in open or single pass mode, but they suggested that if exhaust air temperature is taken advantage of through recirculation, the system performance could be improved. Corn was dried from 23% to 14% moisture content, and resistance heaters were used in the control experiments. They reported an energy consumption of about 2.1 MJ per kg water removed compared to 6.44 MJ per kg required in a comparable electrically heated bin dryer.

Cunney and Williams (1984) used an engine-driven heat pump for a novel grain storage/drying system. The heat pump was used as the source of cold air for chilling, as well as hot air from the condenser in the dryer mode. They report that modest energy savings were achieved (specific energy consumption of 4.7-5.1 MJ per kg water removed) with recirculation of the drying air, but about 30-50% reduction over conventional systems was possible by a better design that took into consideration a gamut of factors such as all possible moisture contents in the load, safe storage life of the grain at these moisture levels, use of a more efficient diesel engine, fans that matched the loads and total insulation of the air tight circuit. A detailed analysis of HPD assisted grain drying by Brook (1986) revealed marginal energy savings, but the author opines that the system has economic drawbacks. Overdrying of a large portion of grain, risk of spoilage due to mould growth during the slow drying process and high capital cost for the equipment were cited as some of the reasons. One impressive report from Ting (1987) describes a 22 kW unit with a dual speed compressor (speed control linked to suction pressure) and a dual condenser package with control over refrigerant flow in each heat exchanger. Design details of this unit are not complete in the paper, but it appears to be in-touch with most practical problems related to drying with heat pumps. The design was intended to achieve optimum energy consumption for dehumidification as well as control the temperature of the drying air. The actual performance of the machine was compared with a simulated convective drying process using ambient air. The dehumidifier assisted drying was able to remove 743 kg of water as opposed to 85 kg by ambient air under similar conditions.

Other major literature related to heat pump dehumidifiers in drying during the 1980s deal with the principles and modeling of the dehumidification and drying process. Some modeling work was followed up with experimental verification, but they all restricted the consideration to constant rate period. Besides, the heat pump was assigned a role of dehumidifier and its performance was analyzed ignoring the drying process. However, the papers throw light on the design factors regarding the choice of individual components of the HPD viz., the compressor, heat exchangers and the expansion valve, and their interaction in the system. Tassou et al. (1982) highlighted the problems that have to be encountered in achieving satisfactory control over the operation of the heat pump. Zylla et al. (1982) give a comprehensive description of the dehumidification process as well as the different configurations of the heat exchangers in the system. The focus of the authors was more on the dehumidification aspect and they outline the conditions under which a favourable specific energy consumption (energy per unit mass moisture condensed on the evaporator) can be achieved. A follow-up to this paper is one by Tai et al. (1982a) that describes the experimental setup with a system using R114 and a detailed discussion about the choice of components. The succeeding paper (Tai et al., 1982b) discusses the results of the experimental work. Two other papers in similar vein (Toal et al., 1988a,b) outline the design of low temperature drying equipment using a heat pump dehumidifier. The second report highlights the conflict between the phenomena of drying and dehumidification in the system, and the need to address this issue for successful operation. Two related papers (Jolly et al., 1990 and Jia et al., 1990) describe the development of a detailed model for simulating the performance of a heat pump assisted continuous dryer. The basic heat pump model was made up of individual component models for the evaporator, condenser, compressor and the expansion valve. The discussion by Jia et al. (1990) highlights some of the major practical issues that are associated with heat pump assisted drying as well the compromises that have to be made for a working unit. A seed dryer using a heat pump that operated in the closed air circulation mode was described by Fritz et al. (1990).

Subsequent literature, especially in the second half of 1990s, is dominated by a few groups of researchers from Norway, Thailand and Singapore, with some exceptions (Rossi et al., 1992; Aceves-Saborio, 1993). They describe the efforts aimed at solving the practical operating problems and the success achieved in adopting the heat pump dehumidifier as an integral part of the dryer. Strommen et al. (2001) provide a general overview of the heat pump related research in Norway. Strommen and Kramer (1994) demonstrated the unique potential of heat pump assisted dryers for controlling the quality of the dried products, especially heat sensitive material. Dealing with various products, the paper describes the drying routines used for regulating the physical properties such as sinking velocity of fish feed, rehydration, colour and taste, biological activity of dried bacterial cultures etc.

Other closely related papers (Alves-Filho and Strommen, 1996a,b; Alves-Filho et al., 1998a) describe the drying regime adopted by these researchers to dry biological material such as fruits, vegetables, microbial cultures etc. Frozen material was dried initially with dehumidified air at -25°C to avoid structural collapse. The drying temperature was raised gradually to finish off at 35-50°C. The quality (organoleptic, physical, rehydration etc.) of the dried product was reported to be comparable to that of freeze-dried material. Fluidized beds were used to improve heat and mass transfer characteristics (Strommen and Jonassen, 1996; Alves-Filho et al., 1998a).

The interest in commercial application of heat pump dehumidifiers for drying in the South East Asian region is evident from the literature originating from Thailand and Singapore. Prasertsan and Saen-Saby (1998a,b) provide a good review of the fundamental aspects of heat pump assisted drying, the constraints on operating such a system, current research as well as R&D needs such as refrigerant compatibility with environmental concerns. One of the papers (Prasertsan and Saen-Saby, 1998a) describes drying experiments in a HPD assisted dryer proto-type with wood and banana slices. The SMER (specific moisture extraction rate) ranged from 0.382 kg to 0.543 kg per kWh for wood and 0.260 kg to 0.536 kg per kWh for banana. The equipment handled large loads (80-150 kg) and the trials were conducted in an open mode with varying ambient air conditions. Heat pump dryer simulation models are described by Prasertsan et al. (1996b) with the assumption that the drying takes place at a constant rate. Soponronnarit et al. (1998) describe in detail their experience with a cabinet dryer prototype (100-132 kg loads) that uses a heat pump of one-ton refrigeration capacity. The system was operated in a closed loop air circulation mode (even though the ambient conditions were reported to be high in humidity) with drying temperature of 50°C, but provision was made to bypass the evaporator (63% of air by weight was bypassed during the experiments). The SMER of the system was reported as 0.363 kg per kWh and the quality of the product superior to that obtained from a hot air tunnel dryer.

The research work related to heat pump dryers at the National University of Singapore is highlighted by Chou and Chua (2001). The paper discusses a general classification of heat pump dryers, their industrial application, comparison of different drying methods and the corresponding energy consumption, and in detail, the use of multi-stage vapour compression systems with multiple condensers and evaporators. An earlier related paper (Hawlader et al., 1998a) provides more details regarding the equipment and drying conditions. The system comprised of two internal evaporators operating at different pressures. The one under higher pressure was used to precool the air (to dew point and below), thus reducing the sensible heat load on the second evaporator (which operated at a lower pressure) to achieve higher dehumidification capacity. An external evaporator was used to make up for low latent heat load during finishing stages of drying. One main condenser, two subcoolers and two economizers were used to achieve proper cooling and to recover heat to the drying air. A simulation model to describe the performance of a heat pump assisted dryer is presented by Hawlader et al. (1998b).

2.4 Prediction And Modeling Of HPD Assisted Dryers

Study and analysis of predictive behaviour in drying is necessary for scale-up and optimization. However, a clear physical and mathematical description of the drying process in biological material remains the holy grail of the research and most large-scale applications take a build-and-study approach. Farkas et al. (2000) have published the latest survey of modeling approaches with an extensive bibliography. Most modeling methods adhere to the concept of diffusion of water within the drying material (Sherwood, 1936), an approach that has been debated almost since its conception (Hougen et al., 1940). Tremendous advancement in the field of numerical computing and cheaper computers have armed researchers with tools to handle the partial differential equation systems involved in the models. Lately, there are signs of shifting the basis from concentration gradient to chemical potential gradient (Gekas, 2001). The moisture diffusivity data from different sources have been compiled by Mittal (1999) and Zogas et al (1996).

In the heat pump assisted drying process, the two components viz., the heat pump and the actual drying of the material, are treated separately (Alves-Filho et al., 1998b). An integrated description would be too complex to be of any practical value. Most of the papers devote careful attention to modeling of the heat pump components; the drying is often considered to take place at the constant rate period with a uniform rate of moisture release from the drying load (Toal et al., 1988a; Carrington and Baines, 1988; Prasertsan et al., 1996b).

2.5 Performance Evaluation Of HPD Assisted Systems

Early research related to HPD assisted drying had a narrow focus regarding the assessment of performance. The heat pump itself received all the attention, either as a dehumidifier or as a heater, and its effectiveness was often described with disregard of the drying process downstream. Strommen (1986) suggested the term thermal efficiency, which was an indicator of the dehumidification of air.

$$\eta = \frac{r.(x_C - x_B)}{(h_C - h_F)} = \frac{r}{\frac{dh}{dx}} \qquad \text{Equation 2.1}$$

$$SMER = \frac{E}{\frac{dh}{dx}} \qquad \text{Equation 2.2}$$

η: thermal efficiency
r: latent heat of evaporation of water, kJ per kg
x: absolute water content in air, kg water per kg dry air
h: enthalpy of air, kJ per kg dry air
B: dryer inlet
C: dryer outlet
F: point between the evaporator and the condenser
SMER: Specific moisture extraction rate, kg per kW.h
E: coefficient of performance of the heat pump, defined as the ratio of the cooling capacity to the power consumption

Generally, COP is defined in the context of a heat pump either for cooling or heating (when used for space heating). However, it has little relevance in drying. The two common indices used are the specific moisture extraction rate (SMER) and the specific energy consumption (SEC). They are defined as,

$$\mathrm{SMER} = \frac{X}{W} \qquad \text{Equation 2.3}$$

$$\mathrm{SEC} = \frac{E}{X} \qquad \text{Equation 2.4}$$

X: total moisture removed, kg
W: total energy input, kW.h
E: total energy consumed, MJ

Prasertsan et al. (1996a) introduce a term called dryer efficiency, which is defined as a percentage of the difference of absolute humidity of the air passing through the dryer with respect to the difference between the process air and saturated air. The dryer efficiency is indirectly meant to represent the drying rate of the product. Based on this concept, the drying process is divided into different zones described as high, medium and low dryer efficiency.

The use of coefficient of performance (COP) is common in the early papers. The use of specific moisture extraction ratio (SMER) and specific energy consumption (SEC) was adopted during the 1990s, and their scope was expanded to include the entire drying system. However, there is no concurrence yet regarding a common efficiency evaluation protocol for drying of biological material.

2.6 Exergy Analysis

Exergy is understood as the work that is available in a gas, fluid or mass of material, as a result of its nonequilibrium condition relative to some reference condition. Work can be performed only under conditions that are not at rest in the surrounding environment (dead state). More work can be performed when the conditions are farther from equilibrium. The exergy concept is a child of the second law of thermodynamics and its implications are discussed by Haywood (1974a,b), McCauley (1983) and Soma (1983). Jorgensen (2001), Dincer and Cengel (2001), Cornelissen (1997), Brodyansky et al. (1994), Moran and Sciubba (1994), McGovern (1990) and O'Toole and McGovern (1990) provide good discussion of the fundamental concepts of exergy as well as some reviews. At the time of preparation of this manuscript, Wall (2001) maintained a very informative website dedicated to exergy that also has an exhaustive list of related literature.

There is a paucity of common approach to the concepts of exergy; the field of second law analysis is said to be in a state of disarray itself (McGovern, 1990) with a lack of consensus on terms and definitions. Since exergy is the work available from any source, terms can be developed using various phenomena (electric current flow, magnetic field, diffusional flow, chemical potential, momentum, gravity etc., however, gravity and momentum are usually neglected). Kotas et al. (1995) and Kestin (1980) propose a system of nomenclature and symbols for exergy analysis. Dunbar et al. (1992) discuss the rationale for developing explicit equations for exergy and energy that would be useful in dealing with phenomena such as transports and interconversions.

The exergy method of analysis is based on evaluating the work that is available at various points in a system, and hence identifying the losses. Available work is calculated on the basis of a final heat sink reference. The basic procedure for conducting the exergy analysis of a system is to determine the value of the exergy at steady-state points in a system and then the cause of the exergy change for the processes that occur between these states. A general exergy equation can be made up as a sum of all the exergies that contribute to the available work at that point. The exergy components that make up the total exergy will differ from system to system and a general equation would have a large number of terms that would make the exercise cumbersome. The usual approach is to tabulate the common contributors to the work in most systems. Exergy losses are generated throughout the system by the irreversible production of entropy caused by the non-ideal performance inherent to all real systems and components. Cornelissen (1997) lists three ways of formulating exergetic efficiency - simple efficiency, which is the ratio of the total outgoing exergy to that of the incoming exergy flow, Rational efficiency (Kotas, 1995), which is the ratio of the desired exergy output to the utilized exergy and a third form, described as efficiency with transiting exergy (Brodyansky et al., 1994), that discounts untransformed exergy components from the simple efficiency.

2.7 Exergy Analysis And Drying

In the absence of a universal framework for efficiency analysis in drying, exergy analysis seems to be a suitable technique. As Dunbar et al. (1992) point out, "the exergy equations (i) show the desirability for controlling the interconversions, (ii) pinpoint quantitatively the resource expenditures associated with irreversible conversions, and (iii) give insight for discovering prospective means for achieving better utilization of the resources and feedstocks". However, exergy analysis can only indicate the potential or possibilities of improving process performance, but cannot state whether or not the possible improvement is practicable and economically reasonable. Exergy analysis compares real performance to the ideal one in which there is no or little external or supplementary driving force required. When there is an external contribution to the driving force, the process occurs faster, but with greater exergy loss (Feng et al., 1996).

Little application of the exergy principles has been seen in food industry. Related discussions are found in papers by Rotstein (1983) and Larson and Cortez (1995). Carrington and Baines (1988) pointed out that theoretically, the only unavoidable loss in a heat pump system was due to the humidification of air in the drying chamber. This was in stark contrast with conventional dryers where the heating process itself represented a major loss. The unavoidable losses are closely related to the actual thermodynamic losses in a heat pump dryer, and it is remarked that careful design can limit the total losses. Humidity was the controlling factor in recirculating mode and, temperature, in single pass or open mode of airflow. Ying and Canren (1993) break up the exergy change into the physical and chemical components in an analysis related to heat pump assisted drying. The adiabatic saturation process in the drying chamber is associated with a process they call "the exergy change of mixing" and derive relations for the pseudo dead state (enthalpy component) and the true dead state (chemical component). Topic (1995) presents the exergy analysis of a high-temperature forage drying process.

2.8 Drying And Environment

Most of the publications related to drying agree that it is one of the most energy intensive operations on earth. However, very few scientists and researchers related to the field of drying have tried to situate drying operations in a global context. Tamasy-Bano (1998) muses on the responsibilities of drying professionals towards sustainable development in the world. Environmentally appropriate technologies in energy engineering, especially in the drying sector, could provide solutions to reduce process irreversibilities and increase of entropy. Dincer (2000) had similar comments to make in a drying forum and examines the issues related to energy, exergy and the environment from the drying industry perspective.

2.9 Dried And Dehydrated Fruits

According to Strumillo and Adamiec (1996), dried food products are gaining popularity, despite considered to having the lowest quality among processed foods. Increasing demand and higher trading in the market are predicted for dried foods.

The US standards for grades (USDA, 1955) define dried apples as follows - "Dried apples are prepared from sound, properly ripened fruit of the common apple (*Malus pumila*) by washing, sorting, trimming, peeling, coring and cutting into segments. The prepared apple segments are properly dried to remove the greater portion of moisture to produce a semi-dry texture. The moisture content of the finished product shall not be more than 24% by weight. The product may be sulfured sufficiently to retard discoloration".

The US standards (USDA, 1977) define dehydrated apples as follows - "Dehydrated (low moisture) apples, hereinafter referred to as dehydrated apples, are prepared from clean and sound, fresh or previously dried (or evaporated) apples from which the peels and cores have been removed and which have been cut into segments. The dried (or evaporated) apple segments may be cut further into smaller segments in preparation for dehydration whereby practically all of the moisture is removed to produce a very dry texture, and are prepared to assure a clean, sound, wholesome product. The sulfur dioxide content of the finished produce may not exceed 1000 ppm. No other additives may be present". Grade A product should have less than 3% by weight moisture and Grade B, less than 3.5%.

2.10 Characteristics Of Dried Products

The concept of quality is quite complex in food processing sector (Bimbenet and Lebert, 1992). The quality impulsion mostly comes from the consumers via competition and flows upstream to the industry, which translates it into constraints on its own operations, as well as requirements towards its suppliers of raw material and equipment. Banga and Singh (1994) identify this dilemma of a producer. Broadly speaking, the concept of quality is often different for the consumer and the industry. Often in research, quality is associated with the organoleptic characteristics of the product that render it acceptable to the consumers. In literature related to drying, according to Ratti (2001), rehydration is the most studied quality parameter followed by colour deterioration and shrinkage.

The organoleptic properties of dried fruits and vegetables, the factors influencing the changes, control of process parameters for desired quality and the kinetics of property changes are discussed and reviewed by many authors (Ratti, 2001; Krokida et al., 2001; Nijhuis et al., 1998; Krokida et al., 1998a,b; Saravacos, 1993). Colour change is one of the most commonly studied factors along with shrinkage (Sjoholm and Gekas, 1995; Zogzas et al., 1994; Lozano, 1983) and tissue structure (Lewicki, 1998). Processing at higher temperature is associated with deterioration of product quality and consumer acceptance. Some researchers have also developed models to describe the relationship between quality changes and processing parameters (Senadeera, 2000; Krokida and Maroulis, 2000).

Krokida and Maroulis (2000) group quality related properties as -

- Structural properties (density, porosity, pore size, specific volume)

- Optical properties (colour, appearance)
- Textural properties (compression, stress relaxation and tensile test)
- Thermal properties (state of product : glassy, crystalline, rubbery)
- Sensory properties (aroma, taste, flavour)
- Nutritional characteristics (vitamins, proteins)
- Rehydration properties (rehydration rate, rehydration capacity)

Drying related changes of some of these properties are discussed in the paper. Quality characteristics are influenced by many factors during drying, but they are all ultimately related to the temperature of drying and rate of moisture removal.

Added heat and exposure time of the product at elevated temperatures during conventional hot air drying affect the nutrient quality of the food products (Sokhansanj and Jayas, 1995). The major chemical changes that are encountered during drying are browning, lipid oxidation and colour loss. Rehydration, solubility, texture and aroma loss are said to be affected by the drying conditions. Nutritionally, vitamins, proteins and microbial load are dependent on the drying temperature and exposure time. Loss of vitamin C and vitamin A during drying has been the subject of detailed study (Jayaraman and Dasgupta, 1995). Loss of natural pigments such as carotenoids, chlorophyll and xanthophyll are associated with the colour changes in dried fruits and vegetables. Even though colour change is sometimes correlated with undesirable chemical changes in the material, the issue is mainly of consumer acceptance. Preservation of these pigments during dehydration is important mostly to make the product attractive and acceptable to the consumers. Changes in texture and physical structure of the dried material influences behaviour such as reconstitution and rehydration, as well as organoleptic features such as mouth feel. The crisp texture of biological material is directly related to the turgidity of the cells, which in turn is controlled by a complex functioning of the cell wall and the cell membrane of the vegetative cells (Pendlington and Ward, 1965).

Fruits and vegetables are subjected to certain pretreatments with a view to either improve or sustain their characteristics during drying (Jayaraman and Dasgupta, 1995). Dipping in alkaline solutions, sulphiting and blanching are some of the common pretreatments. Sulphur dioxide treatment is commonly used to preserve the colour of the dried material. Sulphur dioxide and sulphites act as inhibitors of enzyme action and prevent enzymatic browning. Dipping of whole fruits in alkaline ethyl oleate hastens the drying process without any undesirable effects on the quality of the product (Venkatachalapathy, 1998; Tulasidas, 1994; Saravacos et al., 1988; Ponting and McBean, 1970).

Storage stability is the most important feature of dried products. It is controlled by a combination of factors such as the packaging, storage conditions, extent of dehydration etc. Villota et al. (1980) discuss the factors that are mainly responsible for storage stability. Their work is a compilation of data on storage stability of several dehydrated products.

2.11 Combined Drying Regimes

Over the last three decades, research has been carried out on the use of multiple drying methods to obtain products with superior quality along with reduced energy consumption due to enhanced heat and mass transfer. Since the heat pumps were traditionally considered to be supplementary devices in drying equipment, these combined drying methods are worth a brief review.

Kompany et al. (1991) describe a drying process comprised of two stages – a pretreatment stage by freezing followed by contact drying (heating by conduction) under vacuum. The product is described to be of very good appearance and its rehydration capacity comparable to that of freeze dried material. The effect of different drying techniques operating simultaneously was studied by Grabowski et al. (1994). Simultaneous osmotic and convection drying of grapes (achieved by immersion in a fluidized bed of sugar and semolina) and pretreatments by dipping in ethyl oleate solution were found to reduce the drying time by a factor of 2. Strumillo and Adamiec (1996) review the aspect of combined drying processes in relation to the energy consumption and quality characteristics.

Use of microwaves as part of the drying system for heating has gained popularity over the years. The selective heating ability of microwaves coupled with the convective mass transport by forced air has been used successfully to dry different type of products (Prabhanjan et al., 1995; Bouraoui et al., 1994; Tulasidas, 1994; Al-Duri and McIntyre, 1991). The quality of such products was further improved by preceding the drying with osmotic partial dehydration (Venkatachalapathy, 1998) or by drying under vacuum (Drouzas and Schubert, 1996; Yongsawatdigul and Gunasekaran, 1996a,b). Similarly, use of microwaves hastened the freeze-drying process of peas as reported by Cohen et al. (1992).

2.12 Economics Of Heat Pump Systems

Sztabert and Kudra (1995) discuss cost estimation methods for drying, with empirical information that could be used to make preliminary cost estimates for setting up drying systems. As for the running costs, they suggest the inclusion of the following costs:

- Depreciation of the equipment and structural facilities
- Labour costs
- Expenses on utilities
- Maintenance costs
- Servicing of credit and other indirect costs

The economic viability of a heat pump is usually assessed by its payback period (Pendyala et al., 1986).

$$PBP = \frac{C_{FC}}{A_{UT} - A_{FC}} \qquad \text{Equation 2.5}$$

$$A_{FC} = C_{FC} f_{AP} \qquad \text{Equation 2.6}$$

$$f_{AP} = \frac{i(1+i)^n}{[(1+i)^n - 1]} \qquad \text{Equation 2.7}$$

PBP: payback period
C_{FC} : additional fixed capital cost for the heat pump
A_{UT} : net annual savings in utilities
A_{FC} : opportunity cost of the fixed capital cost
f_{AP} : annuity present worth factor

i : fractional interest rate
n: life of the equipment

2.13 Other Relevant Literature

Most of the published articles directly related to the central objectives of this thesis and available in the public domain in English language have been reviewed and quoted in this section. However, any other literature that has been referred in relation to the research work is mentioned as and when required in the following chapters.

CHAPTER 3
MATERIALS AND METHODS

3.1 Experimental Setup

The MAM 024 (Dectron Inc., Montreal, Canada) is a compact heat pump dehumidifier unit (Figure 3.1) of 2.3 kW rated power. The unit consists of a Copeland ZR24K3-PFV scroll compressor operating on the refrigerant R-22 and a 485 SCFM capacity centrifugal fan (1/8 hp). There are two heat exchangers (each 0.4064 m high, 0.5588 m wide viz., 16" $\times$ 22", 0.226m^2 surface area with finned tubes) inside the unit in the air path (Figure 3.2) and one water-cooled tubular heat exchanger outside the air circuit (Figure 3.3), that acts as a secondary condenser. The unit is also available with a secondary air-cooled condenser (Figure 3.1) but the water-cooled option was selected for the study, as it was more compatible for instrumentation. The two heat exchanger units in the machine are adjacent to one another and the arrangement ruled out the possibility of establishing a bypass between the two coils. A schematic diagram of the unit is presented in Figure 3.4 and the refrigerant circuit with its components is shown in Figure 3.5.

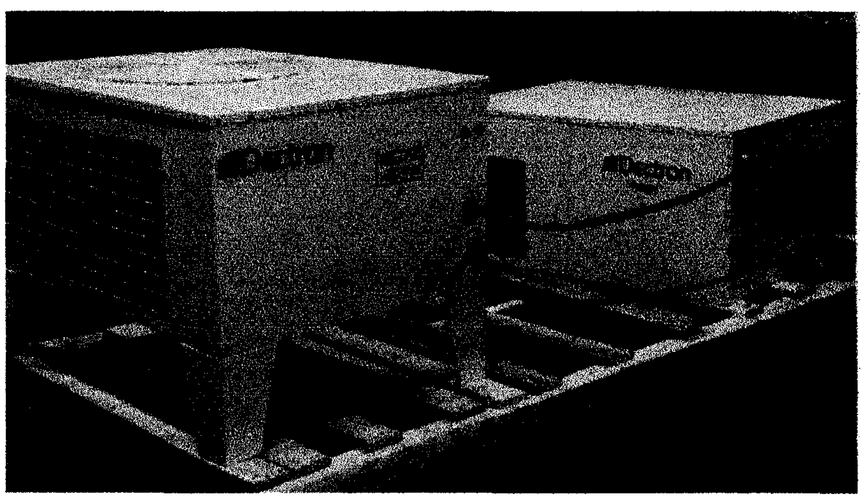

Figure 3.1 The MAM 024 heat pump dehumidifier (on the right, smaller unit) shown with the air-cooled secondary condenser (on the left)

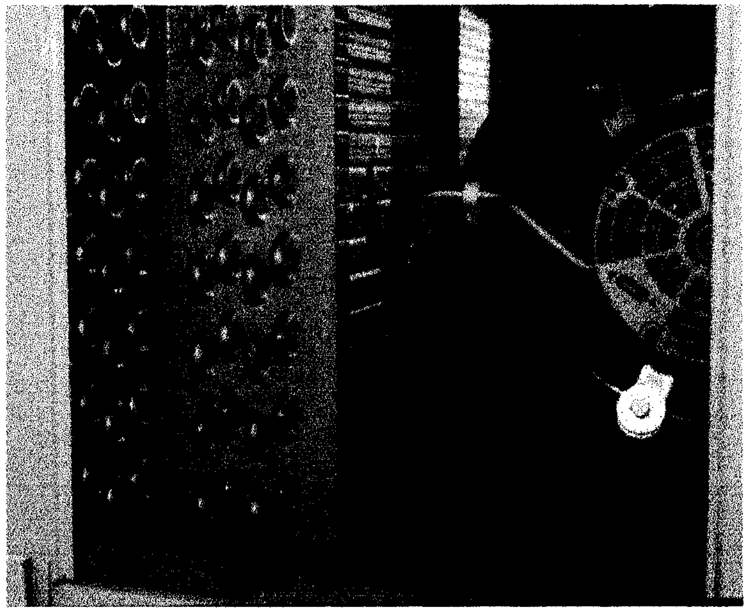

Figure 3.2 Location of the heat exchangers and the fan. The condenser is next to the fan. The drip pan at the bottom catches the condensate from the evaporator.

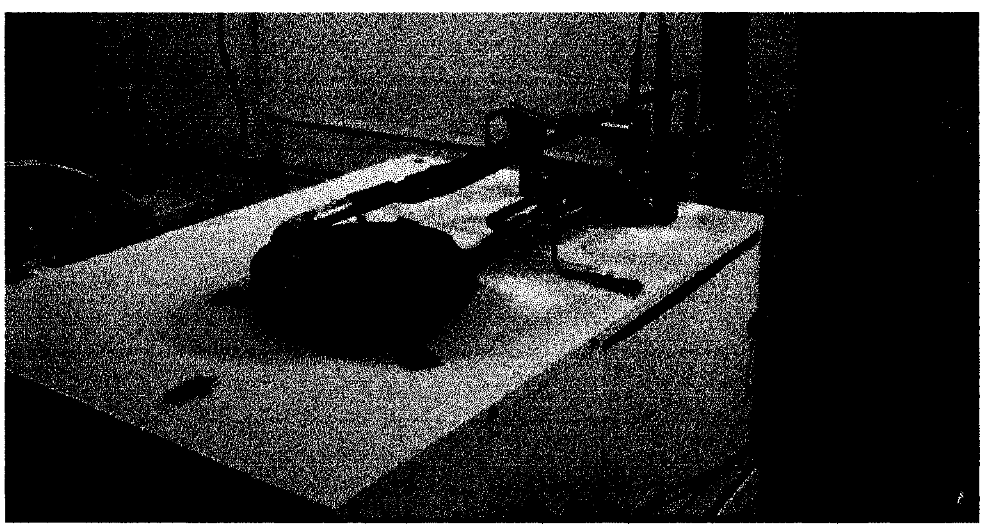

Figure 3.3 The water-cooled condenser that replaced the air-cooled condenser shown in Figure 3.1.

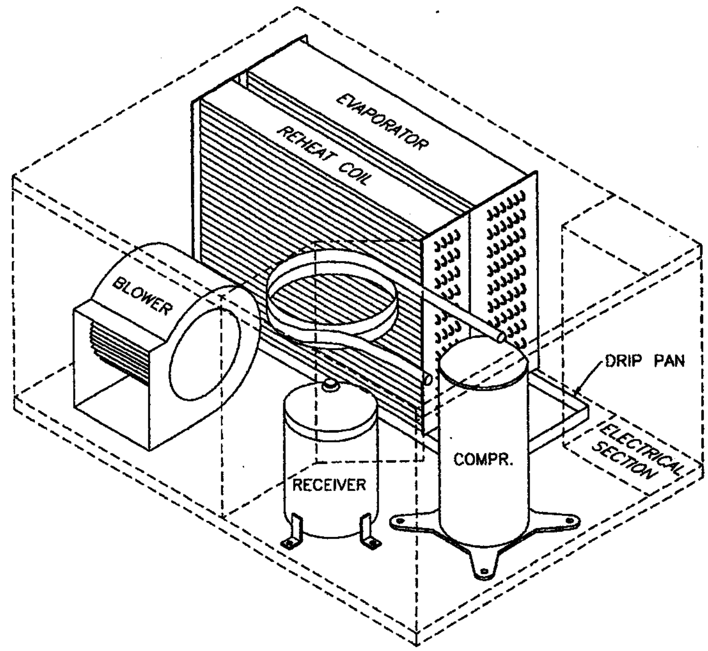

Figure 3.4 Schematic of the HPD. The electrical section, compressor and receiver are insulated from the air path.

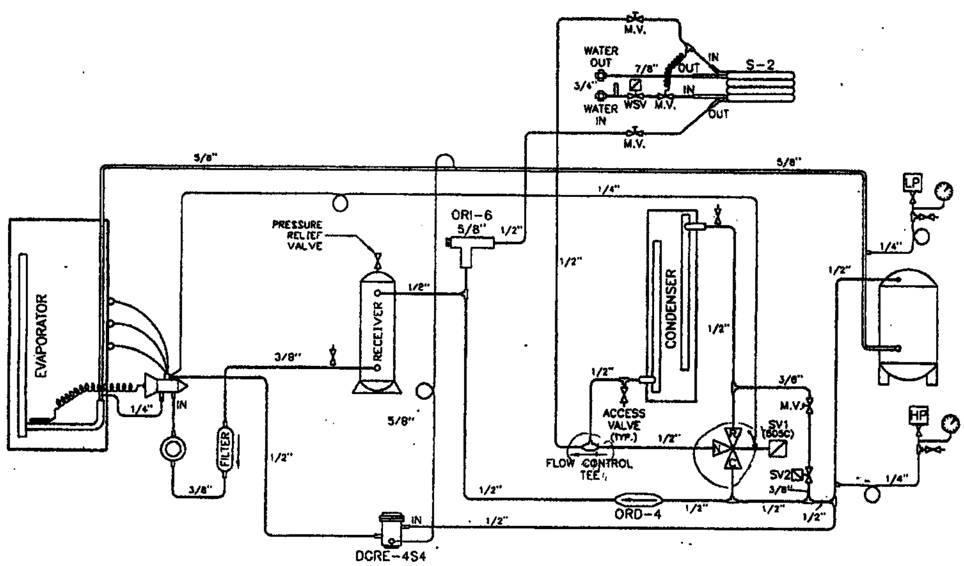

Figure 3.5 Components of the HPD refrigerant circuit.

The thermostatic expansion valve (Sporlan SVE-2) regulates refrigerant flow by maintaining a constant superheat (20°F) at the evaporator outlet. As the superheat (the temperature of the refrigerant over the temperature of the saturated vapour at suction pressure) rises due to increased heat load, the expansion valve increases refrigerant flow until superheat returns to the valve's setting. Conversely, it reduces the flow when superheat decreases. The nominal rating of the Sporlan SVE-2 is 2 tons (7 kJs^{-1}) of refrigeration.

The head pressure control valve (Sporlan ORI-6 in Figure 3.5) regulates the pressure across the expansion valve. The valve's purpose is to hold back enough of the condensed liquid refrigerant so that some of the condenser surface is rendered inactive. This reduction of active surface results in a rise in condensing pressure and sufficient liquid line pressure for normal system operation. The Sporlan ORD-4 (Figure 3.5) is a differential valve that responds to changes in the pressure difference across the valve. It is used with the ORI-6 for head pressure control. As the ORI-6 starts to throttle the flow of liquid refrigerant from the condenser, a pressure differential is created across the ORD. When the differential reaches 20 psi (1.37 bar), the ORD starts to open and bypasses hot gas to the liquid drain line. The operation of the ORI/ORD system is such that a constant receiver pressure is maintained for normal system operation. The ORI is adjustable over a nominal range of 65-225 psig (4.48-15.51 bar) and it is located in the liquid drain line between the condenser and the liquid receiver. The ORD is located in a hot gas line bypassing the condenser. The hot gas flowing through the ORD serves to heat up the cold liquid being passed by the ORI. Thus the liquid reaches the receiver warm and with sufficient pressure to assure proper operation of the expansion valve.

The DGRE-4S4 (Direct-Acting Hot Gas Bypass Regulator) in Figure 3.5 is a pressure-controlling device on the low-pressure side. Its regulator is adjusted to a set point (point at which the regulator starts to open) between 0 and 50 psig (3.44 bar). When the suction pressure decreases below the set point, the regulator opens and allows discharge gas to be bypassed into the evaporator for restoring the suction pressure.

The SV1 shown in the Figure 3.5 is a Sporlan 5D5C three-way heat reclaim solenoid valve that controls the refrigerant distribution between the internal (primary) and the external (secondary) condenser. Directing the flow to the secondary condenser results in cooling and dehumidification of the air, whereas the rejection of heat at the primary condenser allows for dehumidification and re-heating. A hand-operated valve (MV) was installed on a bypass line in order to provide a range of control between the two extremes (full reheat and no reheat) offered by the SV1. This line would ensure supply of refrigerant to both heat exchangers, the proportion depending on the extent of opening of the metering valve. The design is illustrated in Figure 3.6.

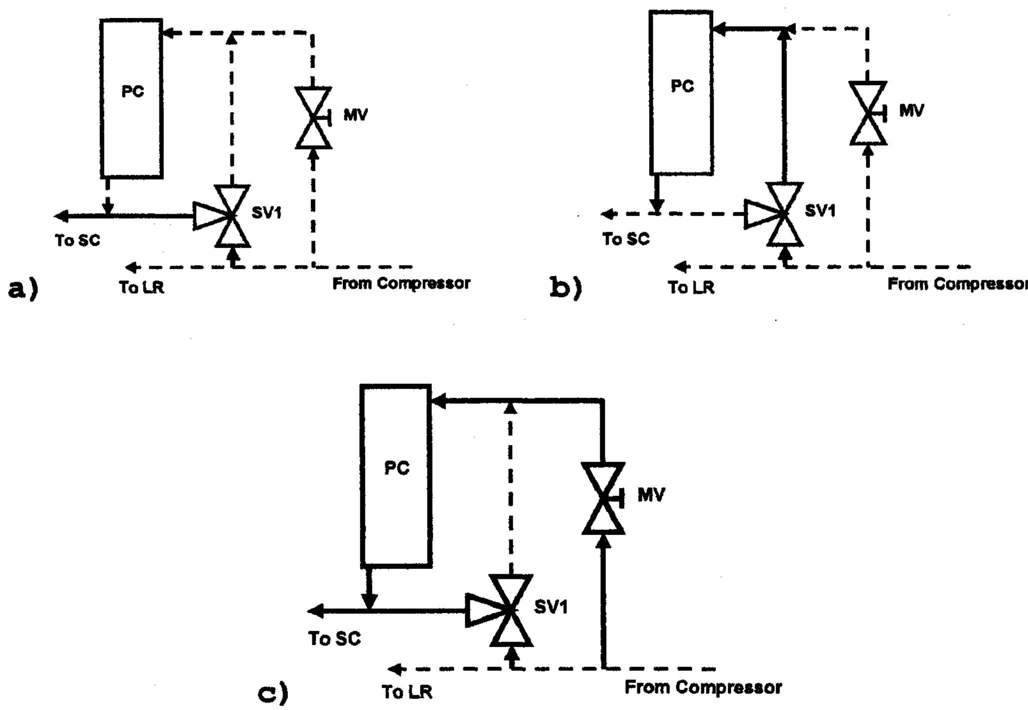

Figure 3.6 Refrigerant circulation controls for complete heat rejection in the secondary condenser (a), complete heat rejection in the primary condenser (b) and partial heat rejection in primary condenser (c). Thick lines indicate the path of refrigerant flow. PC - primary air cooled condenser, SC - secondary water cooled condenser, LR - Liquid receiver, SV1 - three way solenoid valve, MV - hand operated bypass valve.

The term full reheat (situation described in Figure 3.6 b) does not imply that all the heat rejection is achieved at the primary condenser to the air. It only provides for maximum use of the primary condenser; the secondary condenser is used to reject excess enthalpy in the refrigerant for condensation and/or subcooling of the liquid.

The HPD and the drying chamber were connected by rectangular ducts, bolted to flanges fitted to the inlet and the outlet of the HPD. A damper was provided at the air intake to the HPD and another on the discharge side (Figure 3.7). Two 4.5 kW electrical heating coils (Figure 3.8) were placed in the duct between the HPD blower and the drying chamber. The schematic diagram of the system and the photograph are shown in Figure 3.9 and Figure 3.10 respectively. The butterfly valve shown in Figure 3.9 is operated by the airflow. It opens when the two dampers are closed for the recirculation mode and remains closed when the exhaust damper is open.

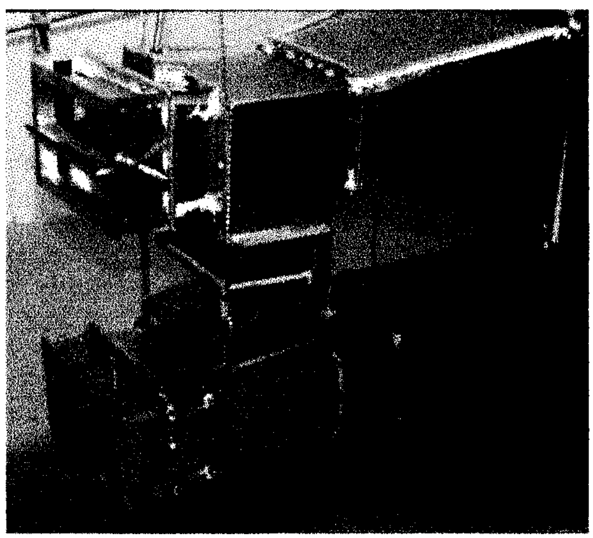

Figure 3.7 The dampers used to remove/make-up air in the system. The damper on top is for suction and the bottom one is on the discharge path.

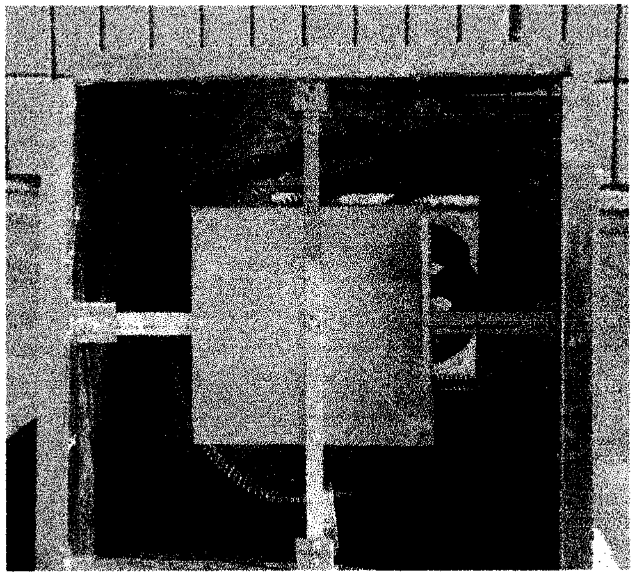

Figure 3.8 Arrangement of the electrical heating elements in the duct. The baffle at the far end is peripheral and the baffle at the near end is at the centre.

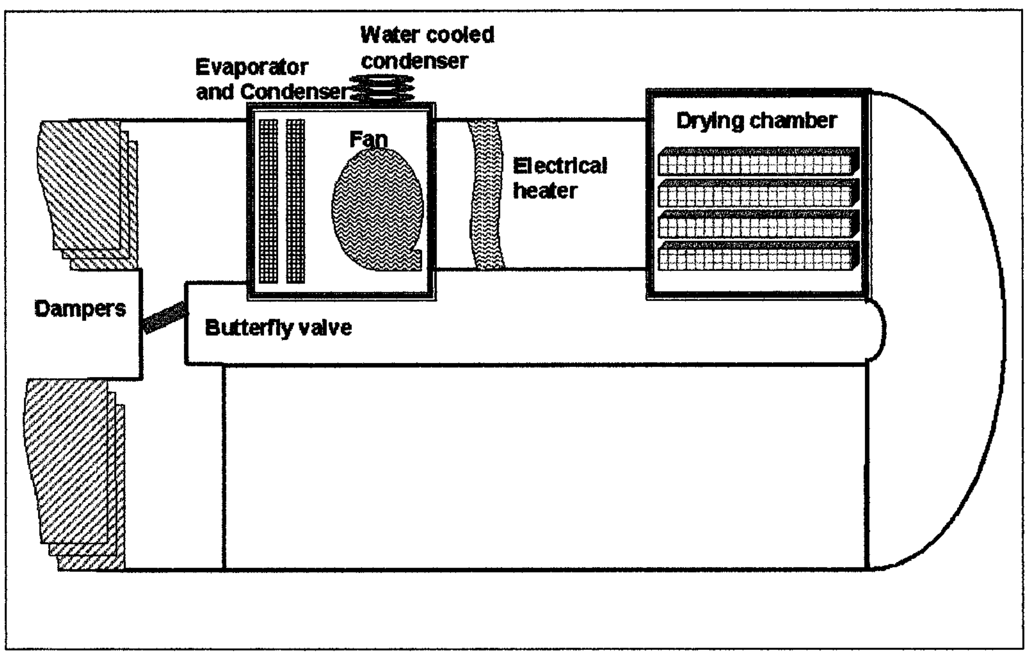

Figure 3.9 Schematic of the complete system.

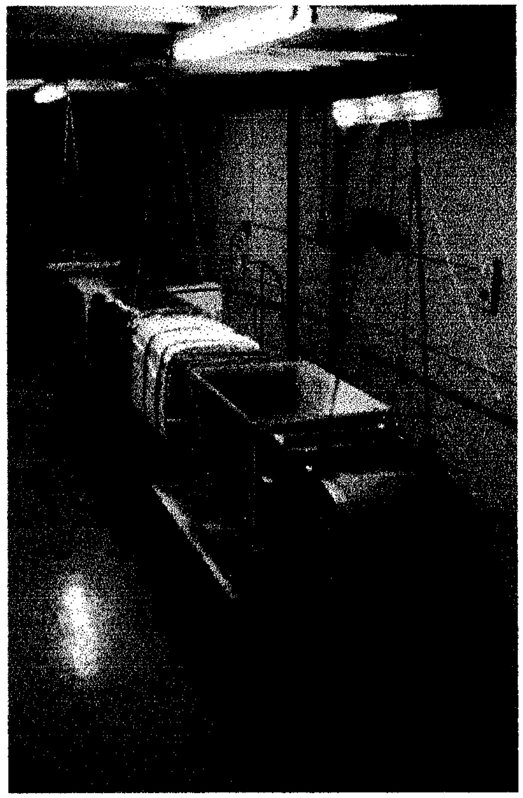

Figure 3.10 The assembled system with the drying chamber and the return duct in the foreground.

The drying chamber (0.175m^3, 0.7m $\times$ 0.5 m $\times$ 0.5 m; l $\times$ h $\times$ w) was constructed using sheets of a polycarbonate material (GE Lexan) of 0.25" (6 mm) thickness. Four adjustable supports were fixed to the base to keep it level. Inside the chamber, the base was fitted with four load cells and an aluminum plate was placed on the load cells to be used as a platform that carried the drying load (Figure 3.11). Holes were drilled on one side of the chamber for insertion of a thermoanaemometer probe.

3.2 Instrumentation And Control

The system was equipped with transducers to measure weight, temperature, electrical power consumption, relative humidity and water flow. Four load cells supported the platform carrying the drying load (Figure 3.11). The output from the loadcells was summed for the response weight. The system was calibrated using known weights before each trial and the coefficients of the linear relation were supplied to the data acquisition program. Similarly, the creep error was estimated with different weights and time periods ranging from 15 minutes to one hour and the behaviour was made available for the program to refine the input. T type thermocouples were used to collect the temperature data. In case of the temperature of refrigerant inside the heat exchangers, the surface temperature was measured and a difference of 5°C was assumed (negative in case of the evaporator). The relative humidity transducers were based on polymer capacitance sensors (± 2% error) and were calibrated using saturated salt solutions (Marsh, 1998). The psychrometric properties of air were derived from the equations prescribed by ASHRAE (1997).

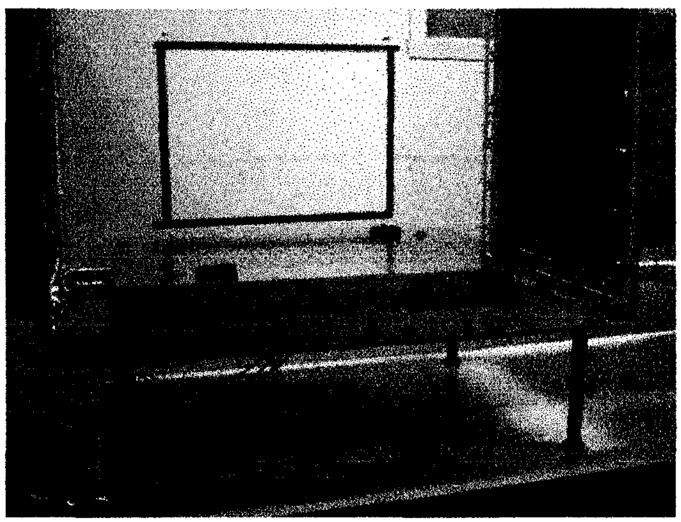

Figure 3.11 Drying chamber with the load cell arrangement at the base.

A water flow transducer (Figure 3.12) was set up on the discharge line of the water-cooled condenser to measure the volume of water used in the cooling. The inlet and outlet temperatures of the water were recorded and used to calculate the heat rejection out of the system. The velocity of air inside the drying chamber was verified using the method suggested by ASHRAE (1997). Six traverse lines were used in adopting the log Tchebycheff rule for rectangular ducts. The system was designed to maintain the velocity at 1.5 ms^{-1} and the measurement confirmed the value ($\pm$ 0.07 ms^{-1}). The uniformity of airflow within the drying chamber was verified visually using smoke (Borozin gun with powder containing zinc stearate, Cole Parmer, USA).

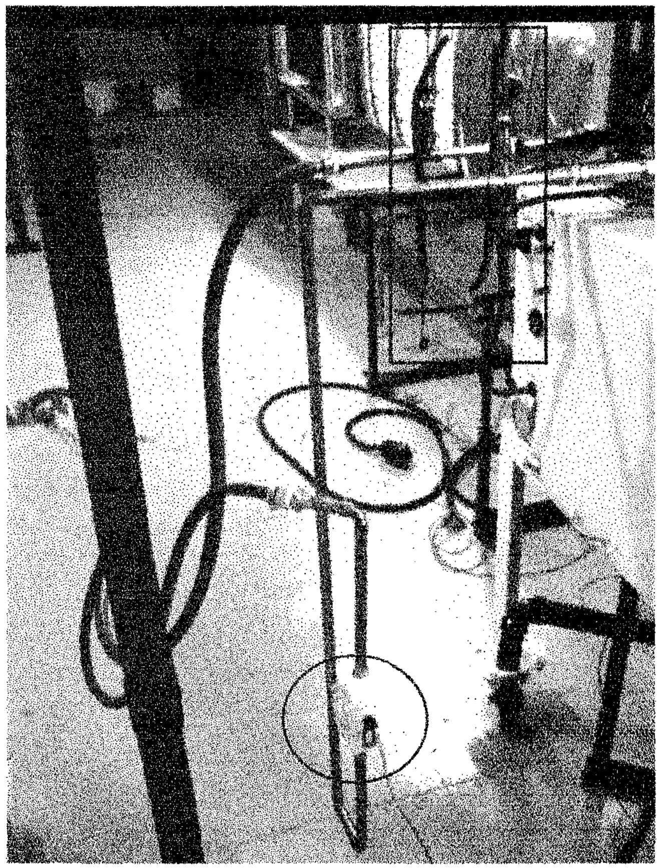

Figure 3.12 Setup to measure the heat rejection from the system. The water flow rate is measured on the discharge side with the flow transducer (circled) and thermocouples are placed in wells both at the inlet and discharge pipes (seen here within the marked rectangle).

A HP34970A (Hewlett Packard) with modules for data acquisition (HP34901A), switching (HP34903A) and multipurpose application (HP34907A) was used for data logging and control in association with a Desktop PC. The switches for the solenoid valve SV1, the SSR controlling the heaters and the compressor cutoff were wired to the switching module for remote control. The power supply to the heater coils (via the SSR) was run through a PID temperature controller (OMEGA CN9000A) whose set point was adjusted to the required temperature. The circuit contained a counter that recorded the time for which the power supply to the heaters was on. The transducers were scanned for input every 10 seconds for the first half hour of an experimental run and every minute thereafter.

3.3 Drying Experiments

Drying experiments were carried out using apple (cultivars Elstar, Gala, Cortland, Empire and Golden Delicious), tomato and agar gels. Apple was purchased from either a local farm (Empire, Elstar, Gala, and Cortland) or a supermarket (Golden Delicious) in boxes of 20kg and tomato was obtained from the supermarket in boxes of 25 lb. The fruit was stored in a refrigerated room (4°C); material for each trial (30-40 fruits) was taken out and kept at room temperature for about 24 hours to equilibrate with the ambient temperature. A minimum of four fruits were drawn from each box and the moisture content was determined (12-16 trials per box of 20 kg lot). The average values were used to represent the moisture content of the entire lot.

Apple was cored and sliced to 1 cm thickness and the rings were spread on steel wire trays evenly. For studies comparing the surface area, the fruit was sliced to obtain cuboids with 1 cm $\times$ 1 cm $\times$ 4 cm sides. The samples were weighed on a bench balance and the trays were stacked in two columns on the platform in the dryer. Tomato was similarly sliced and loaded. Agar gels of 3.75% and 5% solids were prepared in petri dishes and were used for drying trials. The use of these gels was to ensure the homogeneity of the moisture content and distribution in the drying load. In the case of experiments conducted to monitor the moisture distribution in the load, each ring/block was weighed before and after the trial and its position in the chamber noted.

The heat pump dehumidifier was switched on after closing the drying chamber and the two dampers on the system were closed for trials with closed loop drying setups. The dampers were left open in experiments that involved room dehumidification. The hot air drying runs were carried out with the heaters set at the required drying temperatures. The weight indicated by the load cell setup was used to monitor the progress of drying. The unit was switched off upon reaching the desired final weight. Random samples were taken from the finished product to verify the final moisture content.

The surface temperature of the slices during drying was estimated using the images obtained from a LAND Ti35sm infrared Camera. The chamber was opened for a minute during which the IR images were captured.

3.4 Water Activity

The trials conducted for studying the water activity were similar to the regular drying runs. The chamber was opened periodically to draw two random samples. The rings were allowed to cool in the ambient air, weighed and were used for measuring the a_w in a water activity meter (Aqua Lab 3TE from Decagon Devices, Inc., Pullman, WA) that uses the chilled-mirror dewpoint technique. The samples were preserved and their moisture content determined on the basis of their initial weight prior to the water activity measurement.

3.5 Moisture Content

Water content was estimated using a moisture balance equipped with an infrared heating element. A multiple temperature routine (200°C/10 min, 150°C/20 min and 100°C/75 min) that had results comparable with those obtained from the procedure prescribed by AOAC (Official method 920.15.1, AOAC, 1999) was used to estimate the moisture content of the samples in less than 2 hours.

3.6 Colour Measurement

A Minolta CR-300 Chroma meter with an 8-mm dia measuring area was used for colour measurements on apple. The measuring head uses diffuse illumination and a 2°-observer angle. The L*a*b* absolute value measurements were made under CIE (Commission Internationale de l'Eclairage) Illuminant C lighting conditions. The instrument was calibrated to the CR-A43 ($Y=94.4$, $x=0.3141$ and $y=0.3207$) white calibration plate before each set of measurements was made on the apple.

3.7 Rehydration

The extent and rate of rehydration of samples were measured. The dried sample was weighed, placed in a porcelain dish, flooded with distilled water (10°C) and left undisturbed for 10 minutes. It was then removed, placed in a funnel (on Whatman #4 filter paper) that drained into a flask. The flask was evacuated using a vacuum pump and this step removed excess moisture clinging to the surface. The sample was then weighed and returned to the water. The procedure was repeated till three consecutive values of weight agreed within a range of 1g. After the final weighing, the sample was crushed into a puree using a mortar and pestle, transferred into the moisture balance and dried completely to obtain the weight of the solids present.

The rehydration was expressed as the weight of water (g) held by a gram of dried material.

$$\mathbf{Rehydration} = \frac{\mathbf{M_w}}{\mathbf{M_s}} \qquad \text{Equation 3.1}$$

M_w = Mass of water taken up by the sample
M_s = Mass of solids in the sample

The rehydration rate was expressed as the time taken to reach the maximum adsorption of water.

3.8 Organoleptic Observation

The samples were evaluated by a six-member panel for visual appearance, flavour and consistency. No similar commercial product could be found in the market for comparison.

3.9 Storage

The dried material (about 100g) was packed in polythene bags (Ziploc), labeled and placed on shelves at ambient conditions. The bags were opened at 15-day intervals and samples were taken out for measurement of water activity and physical observation. The samples were stored and observed for five months.

3.10 Histological Observation Of The Dried Apple

Anatomical features were examined from prepared sections of apple fruit, cultivars Empire, Gala and Golden Delicious. Radial fruit sections were cut (approx. 0. 5 $\times$ 2.5 $\times$ 2.5 mm; l $\times$ w $\times$ h) from mid-way between the epidermis and core of similar-sized fruits. The pieces were fixed in formalin-acetic acid-alcohol, dehydrated in an ethanol series and embedded in paraffin. These were tangentially sectioned (10 μm), stained and counter-stained in safranin and fast green, and mounted in permount (Sass, 1958). For each treatment, at least 10 fields of view were evaluated, representative micrographs taken and the prints examined. The general shape, size and integrity of the parenchyma tissues were compared for each cultivar/treatment combination (fresh, hot air and heat pump drying).

CHAPTER 4
RESULTS

4.1 Treatments

The HPD system could be operated in different modes depending on the airflow circuit as well as the mode of heat utilization in the unit. The following combinations were tested -

- Closed loop, cooling mode - The inlet and the exhaust dampers were closed and the HPD was used for dehumidification only, with complete heat rejection in the secondary condenser (Figure 3.6a).
- Closed loop, reheat mode (heat reclaim in the primary condenser) - A fixed quantity of air taken in at the beginning was recirculated throughout the drying period (by keeping the dampers closed), the refrigerant was passed through the primary condenser for maximum possible heat recovery within the system (Figure 3.6b).
- Closed loop, reheat mode with heat input for finishing off - The electrical heaters were switched on (with dampers open) to complete the drying process for removal of the final 20-40 points at 45°C and 65°C after having dried partially with the closed loop, reheat mode using the HPD.
- Initial heat input - The heaters were kept on (dampers open) initially (for the first one or two hours) with set points 45°C and 65°C, and the rest of the process was carried out in the closed loop, reheat mode using the HPD.
- Open air circuit with ambient air dehumidification - The HPD was operated in the heat reclaim mode and the dampers were kept open during the drying period.
- Hot air convective drying (control treatment) - Drying was carried out using the heaters (set points 45°C and 65°C). The dampers were kept open; only the fan was operated in the HPD with the compressor switched off.

4.2 Performance Of The System

4.2.1 Closed Loop, Cooling Mode

Two trials were carried out in this mode and the typical conditions of air in the system are shown in Figure 4.1. Apart from the dismal drying rates of less than 0.05 kg per hour, the properties of air underwent severe fluctuations after 10 hours of operation. The low temperatures caused severe frosting on the evaporator surface and the temperature of air leaving the HPD increased gradually. The erratic variations in the system could neither be controlled nor satisfactorily explained and hence, no further tests were carried out in this mode.

4.2.2 Closed Loop, Heat Reclaim Mode

Most of the drying trials were carried out in the air recirculation mode. The typical temperature and RH profile of air in the system for such a run are shown in Figure 4.2. The initial phase is marked by a drop in temperature during the first hour followed by a rapid rise between the second and the sixth hours, after which, the temperature was stable. Based on the measurements of air properties, the energy profiles were calculated (Appendix 1) and are shown in Figure 4.3. The total enthalpy exchanged at the evaporator (H_{ev}) per minute during the run averaged 357 kJ (SD = 16 kJ), which is lower than the 2-ton (422 kJ per min) nominal capacity of the unit. The average heat output at the primary condenser was 363.6 kJ per min (SD = 30 kJ per min). At an average power consumption of 1.5 kW, the COP of the unit works out to 4.

The sensible heat fraction (H_{sens}) predominates in the heat transferred at the evaporator, which is to be expected in batch drying. However, unlike the drying rate of the product, the fraction of latent heat does not peak in the earlier stages of the process. It is rather spread out over the period, tapering off at the finish. Theoretically, the moisture loss in the product should be equal to the increase in the water content of air arriving at the evaporator (indicated by the amount of latent heat in air, H_{lat} in Figure 4.3). But the variation of humidity ratio of the air at the

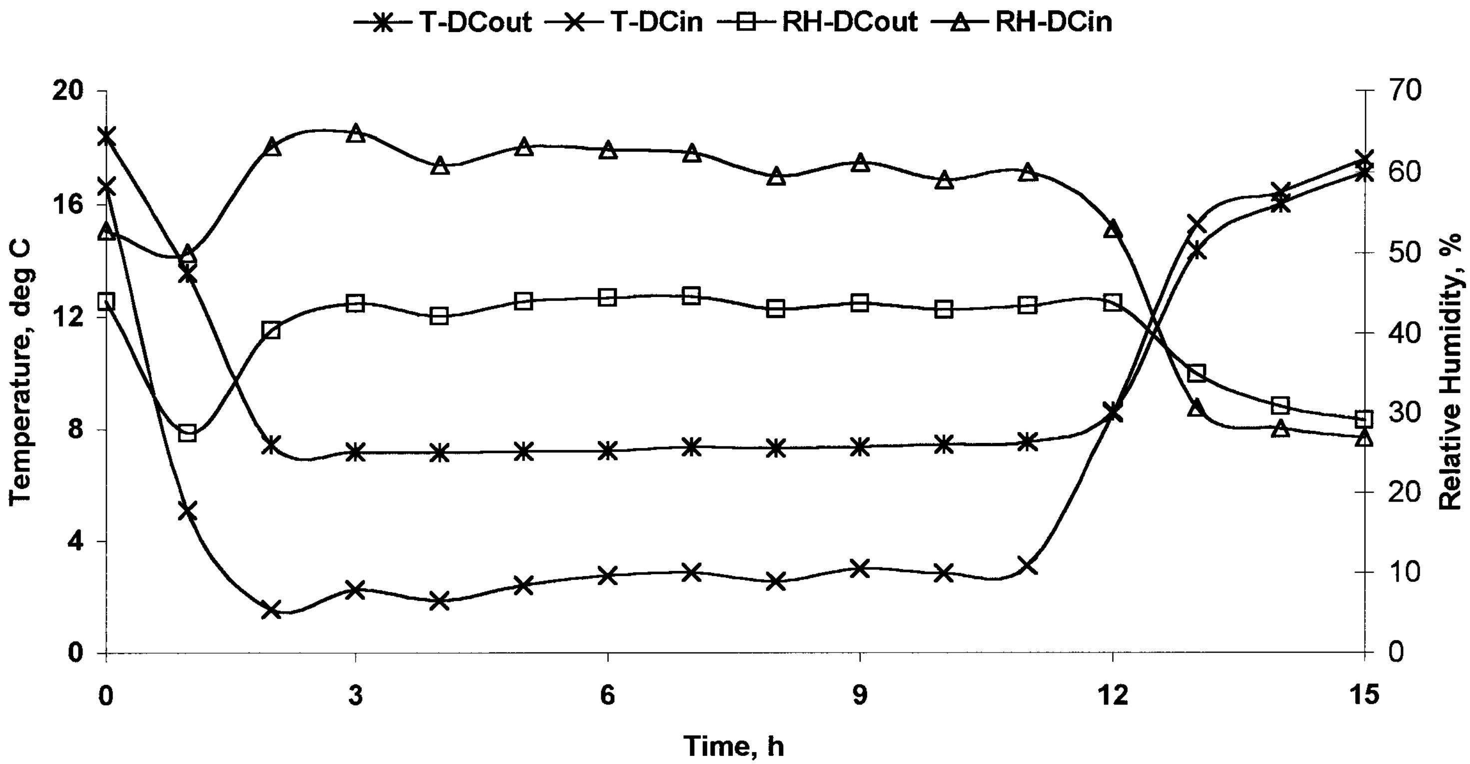

Figure 4.1. Typical conditions of air in the system in the closed loop, cooling mode (DC_{out} - air leaving the drying chamber and arriving at the evaporator, DC_{in} - air leaving the HPD and entering the drying chamber). Data points logged at one-minute intervals.

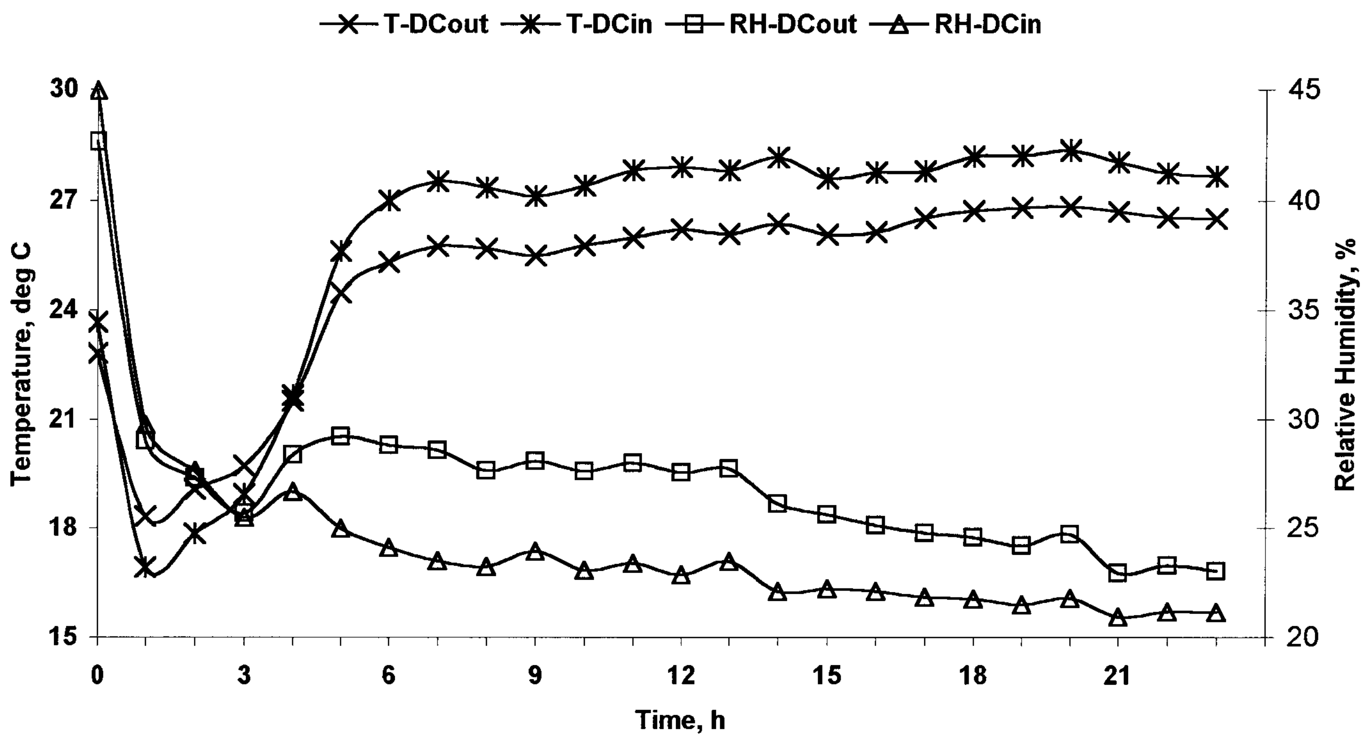

Figure 4.2 Typical conditions of air in the system in closed loop, heat reclaim mode (DC_{out} - air leaving the drying chamber and arriving at the evaporator, DC_{in} - air leaving the HPD and entering the drying chamber). Data points logged at one-minute intervals.

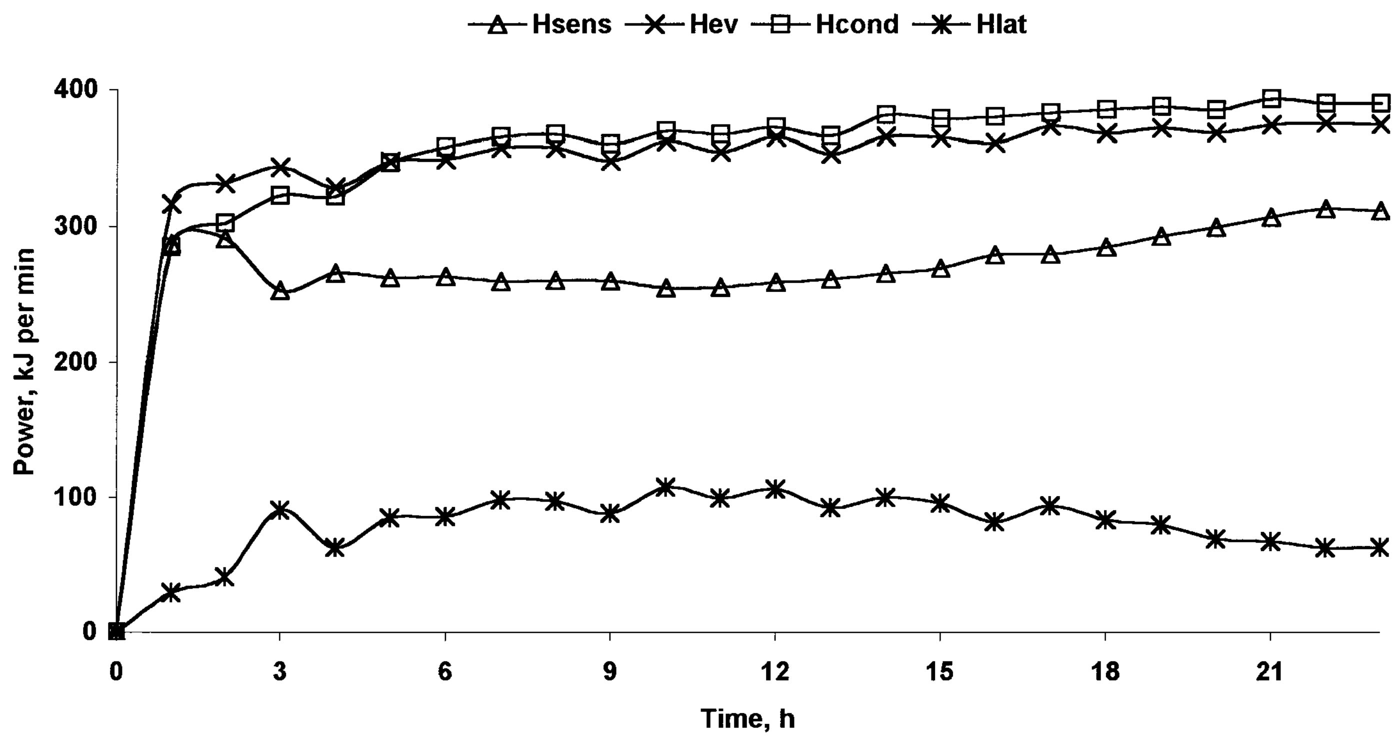

Figure 4.3 Thermal power transferred at the heat exchangers (H_{ev} - enthalpy transfer to evaporator per minute, H_{sens} - sensible heat portion of H_{ev}, H_{lat} - portion of H_{ev} due to latent heat and H_{cond} - enthalpy removed at the primary condenser per minute). Data points logged each minute.

evaporator does not match the drying rate during the process. This could be partly due to the uninsulated portions of the ductwork, which might cool and condense part of the moisture in the air. But as drying progresses and the temperature variations decrease, the residual moisture is brought to the evaporator and condensed.

The compact and commercial nature of the HPD unit presented a serious hurdle in calculation of the energy expenditure and modeling. While the air circuit was furnished with the necessary sensors, the refrigerant side was only equipped with pressure gauges indicating the suction and discharge pressures. The thermostatic expansion valve controlled the refrigerant mass flow to ensure a constant superheat at the evaporator outlet; but the line was not fitted with a flow sensor due to unavoidable (financial and technical) reasons. Furthermore, the presence of the pressure regulating mechanisms (ORI and ORD - explained in the previous chapter in the equipment description) contributed to varying conditions of heat transfers. An attempt was made to calculate the heat balance in the system with the assumption that all the heat that was exchanged at the evaporator was utilized by the refrigerant for evaporation and superheat (Appendix 1). The power distribution in the system during a typical trial was calculated based on the measured heat rejection at the secondary condenser and the results are shown in Fig 4.4. The average total power consumption by the HPD recorded (compressor and the fan) for the period was 1.56 ± 0.1 kW and the average efficiency of the compressor based on the calculated values appears to be very low (46%). However, it should be noted that the low-pressure side was operating at the set point of the DGRE-4S4 regulator (3.3 bar or 48 psi) and the amount of hot gas bypassed could not be measured.

The other practical difficulty in measurement that hindered estimation of the energy balance was the temperature of the condensed water. It is common practice to assume that the water is at the same temperature as the air leaving the evaporator surface, and to calculate its enthalpy for that temperature. However, in the situation described here, the condensed water has significant impact on the heat transfer characteristics of the evaporator. The assumption for temperature holds good only if the water leaves the surface of the heat exchanger quickly (usually by

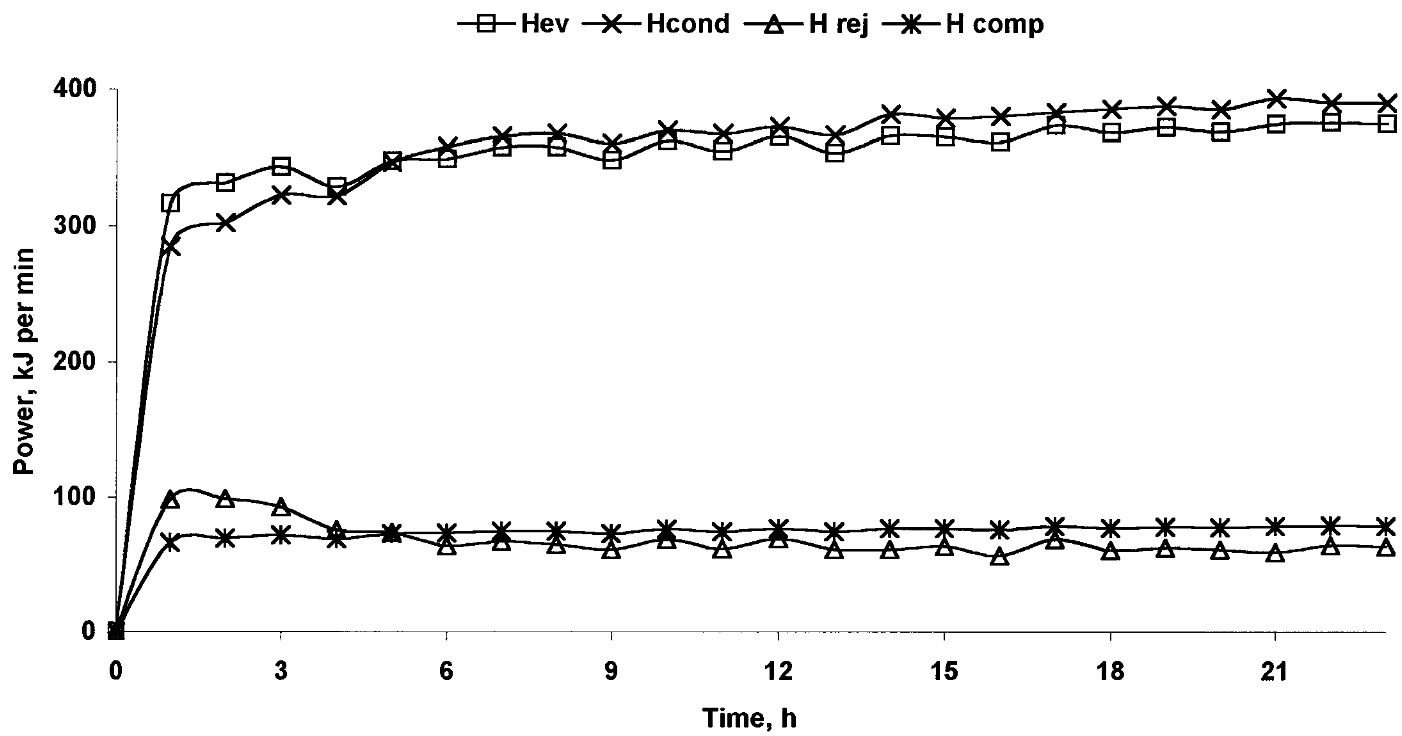

Figure 4.4 Thermal power exchanges across the HPD. (H_{ev} - enthalpy transfer to evaporator per minute, H_{cond} - enthalpy removed each minute at the primary condenser, H_{rej} - enthalpy rejection per minute measured at the secondary condenser and H_{comp} - calculated power input by the compressor to the refrigerant). Based on data points logged each minute.

gravity flow). However, the negative pressure due to airflow will hinder the drain off, especially when the amount of water condensing is low (as opposed to a condition where the air entering the HPD is very humid). The water lingers on and freezes on the surface, contributing to discrepancies in heat transfer as well as enthalpy balance calculations.

4.2.3 Open Circuit For Space Dehumidification

The greatest advantage of a HPD in drying is its ability to harness the low-grade heat in humid air for useful work. Humid, warm air is an impediment to hot air convective drying operations in many parts of the world. But it provides an infinite source of heat for the HPD, and hence an ideal environment for its operation.

It was only possible to conduct two complete trials under circumstances of high ambient humidity during this study. There were not many days during which the suitable conditions prevailed and on two such occasions, the data acquisition system malfunctioned and those trials had to be aborted. The data from one of the successful trials is presented in Figures 4.5 - 4.7. Figure 4.5 indicates the conditions of ambient air and the air entering the drying chamber. Since the operation was in the open mode and the conditions of the air leaving the system were not much different from that of the air entering the drying chamber, the HPD was simultaneously being used for cooling and dehumidification of the ambient air space. It can be noted in Figure 4.6 that the heat exchanged at the evaporator was averaging 382 kJ per min, half of it being the latent heat fraction. It also indicates the heat output at the condenser (196.6 $\pm$ 5.6 kJ per min), which was close to the values observed during the second successful trial (206.85 kJ per min). In both trials, the measured power consumption of the HPD ranged from 1.6 to 1.8 kW and the calculated compressor efficiency was calculated to be 78-86%. With similar assumptions as in the earlier case, the enthalpy balance was worked out and the results are illustrated in Figure 4.7. Large amount of heat is taken out of the system at the secondary condenser (267 kJ per min) and this is a feature of the unit that is primarily aimed at climate control. The HPD is designed to operate in temperature

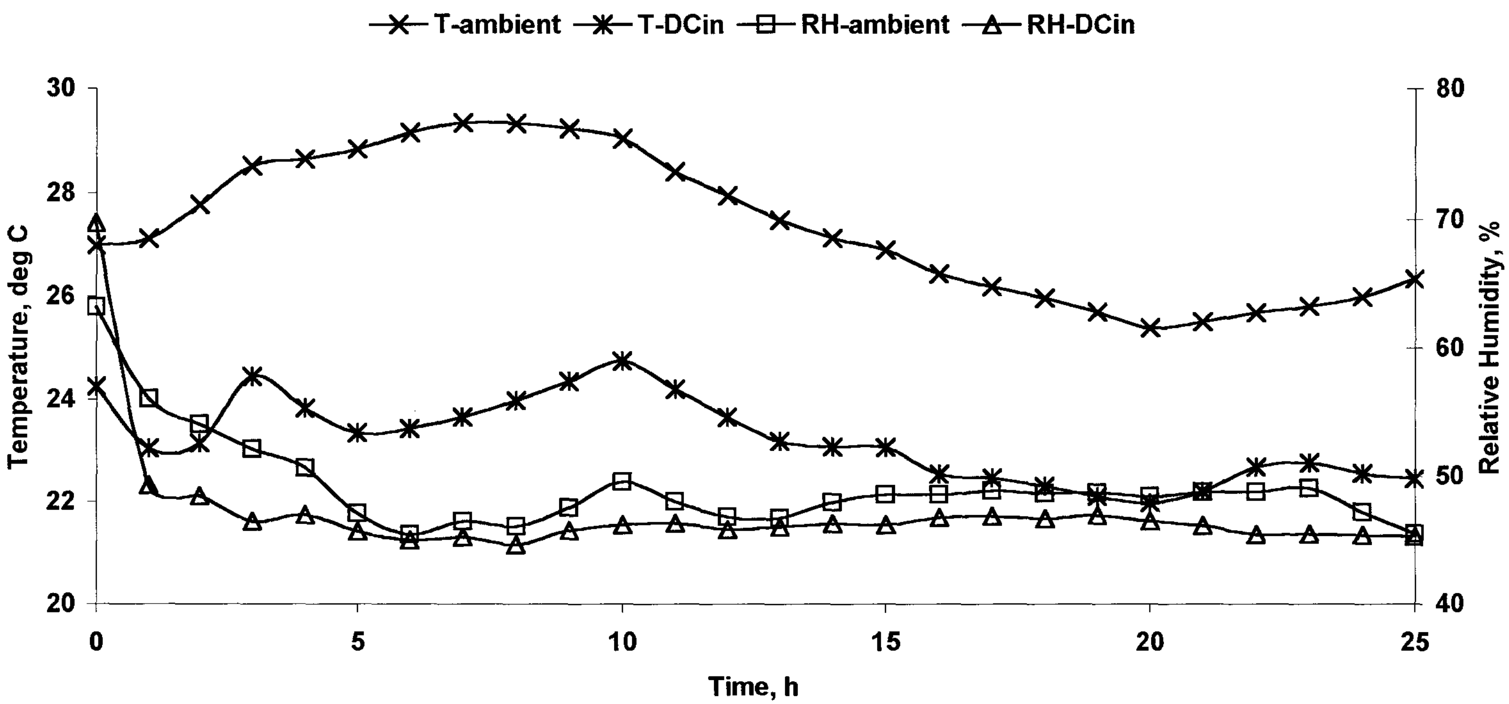

Figure 4.5 Conditions of air in the system in open mode - space dehumidification coupled with drying of apple (DC_{in} - air entering the drying chamber). The properties of air discharged from the system (not shown to avoid cluttering) closely resemble that of the air entering the drying chamber except for the first hour of the process. Data points were logged each minute.

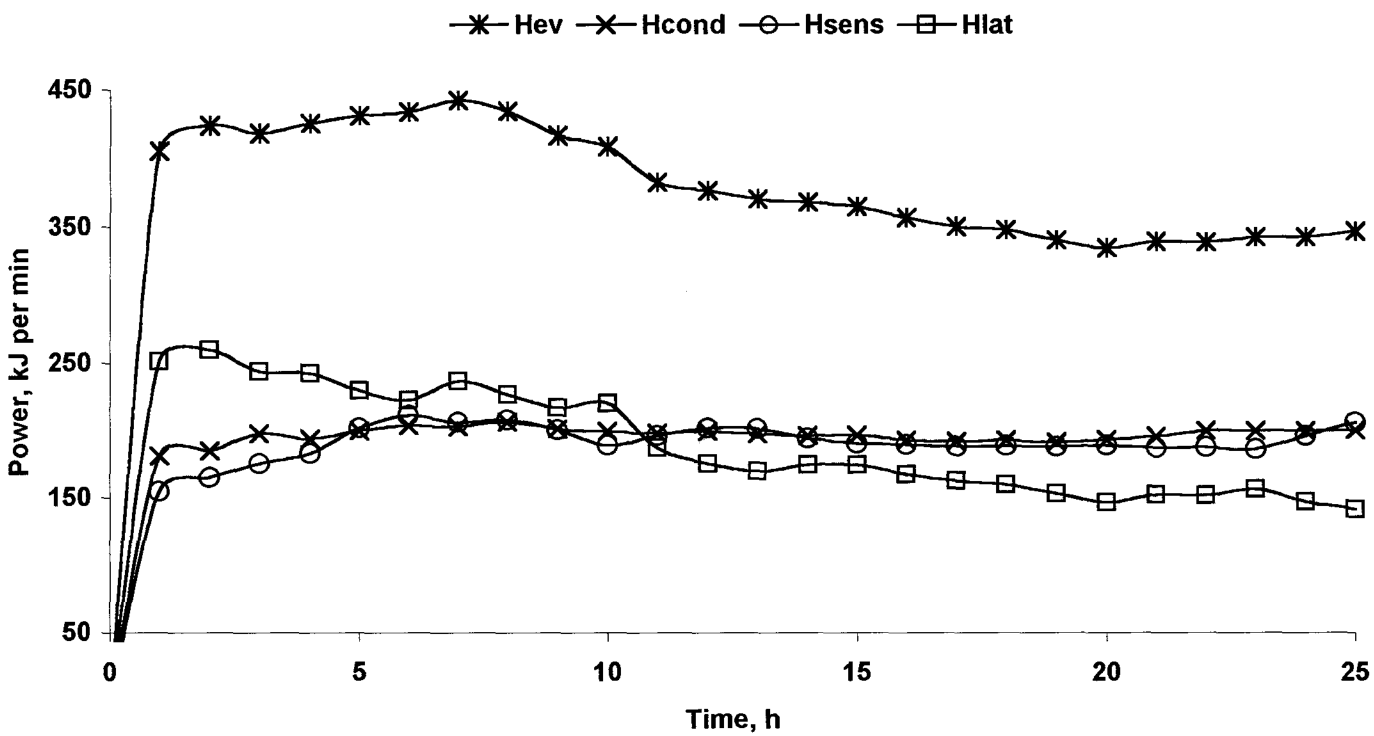

Figure 4.6 Thermal power transfers at the heat exchangers in open mode operation for ambient air dehumidification (H_{ev} - enthalpy transfer per minute to evaporator, H_{sens} - sensible heat portion of H_{ev}, H_{lat} - portion of H_{ev} due to latent heat and H_{cond} - enthalpy removed per minute at the primary condenser). Data was logged each minute.

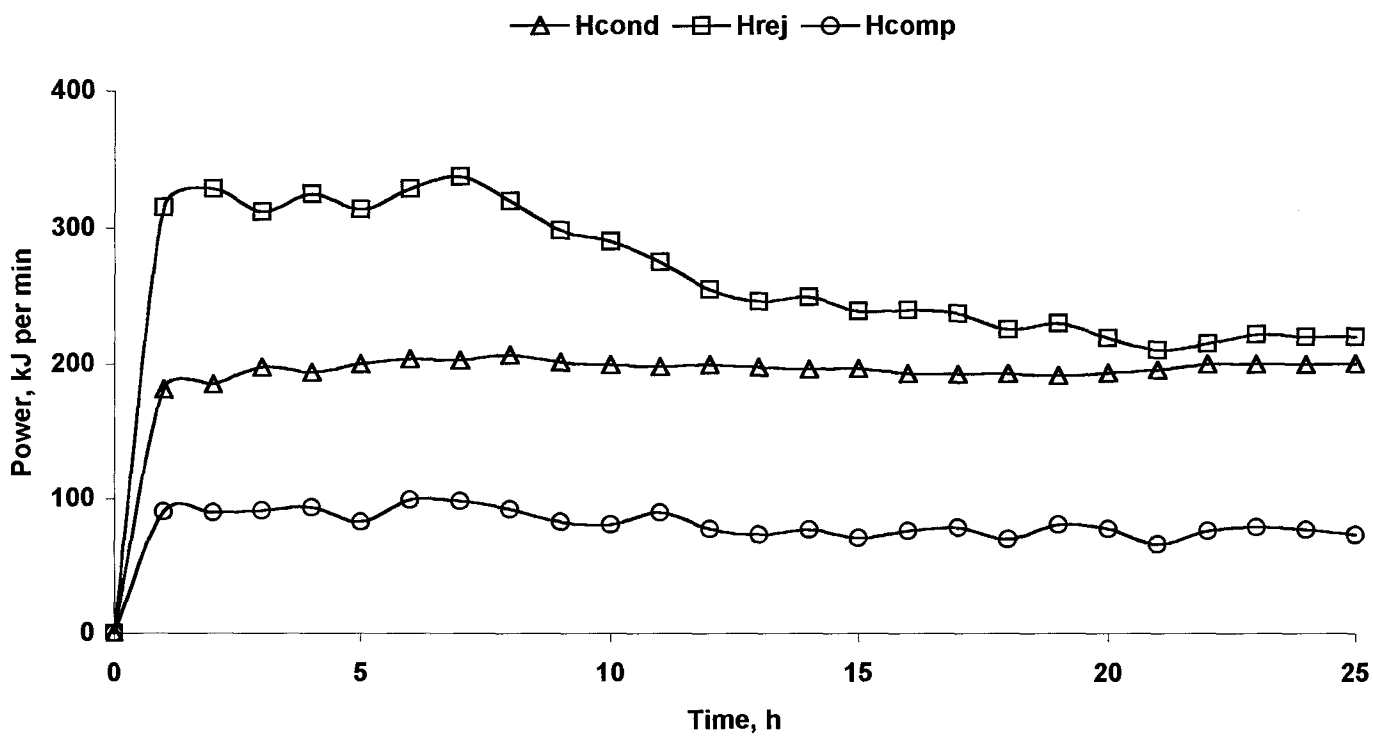

Figure 4.7 Thermal power rejection at the primary (H_{cond}) and secondary (H_{rej}) condensers and the thermal power input by the compressor (H_{comp}) during operation in the open mode for space dehumidification. Based on data logged each minute.

ranges that are considered "comfortable" for domestic living space. Hence, the primary condenser is designed to exchange only limited heat so that the discharge air temperature remains less than 30°C.

These trials clearly indicated the strengths and drawbacks of the machine. The unit performs impeccably in the context of its intended application - space air property control. As a heat pump, its COP (quantity of heat delivered at the condenser per kW input to compressor) is higher than 5. But almost 60% of the energy delivered is taken out of the system at the secondary condenser. A drying system that intends to utilize this stream of energy will have to consider multiple condensers with air circuits that feed collectively into the dryer. The airflow rates, the heat exchanger characteristics and the mixing of the different air streams will have to be matched to make use of the HPD for drying, while maintaining its versatile nature for other applications. In its present form, the secondary condenser could be incorporated in a hot water system and the rejected heat could be utilized.

The main advantage of this mode was that the conditions of the air in the system were constant and did not take long to stabilize. However, the values were dependent on the ambient conditions and the temperatures attained during the two successful trials were lower than those achieved under closed loop drying. It would be necessary to operate under a wider range of ambient air conditions (combinations of temperature and RH) to arrive at a more global evaluation of this operating mode.

4.3 Drying Characteristics

4.3.1 Drying With Hot Air

Trials were conducted in drying of apple at 45°C and 65°C using the heaters (compressor switched off, both dampers open). Typical drying curves for the trials are shown in Figure 4.8. Based on the period of time the heaters were on and the power consumption, the specific energy consumption (SEC) for the process was calculated to be 6.58 MJ (SD = 0.14 MJ) and 4.86 MJ (SD = 0.2 MJ) per kg water

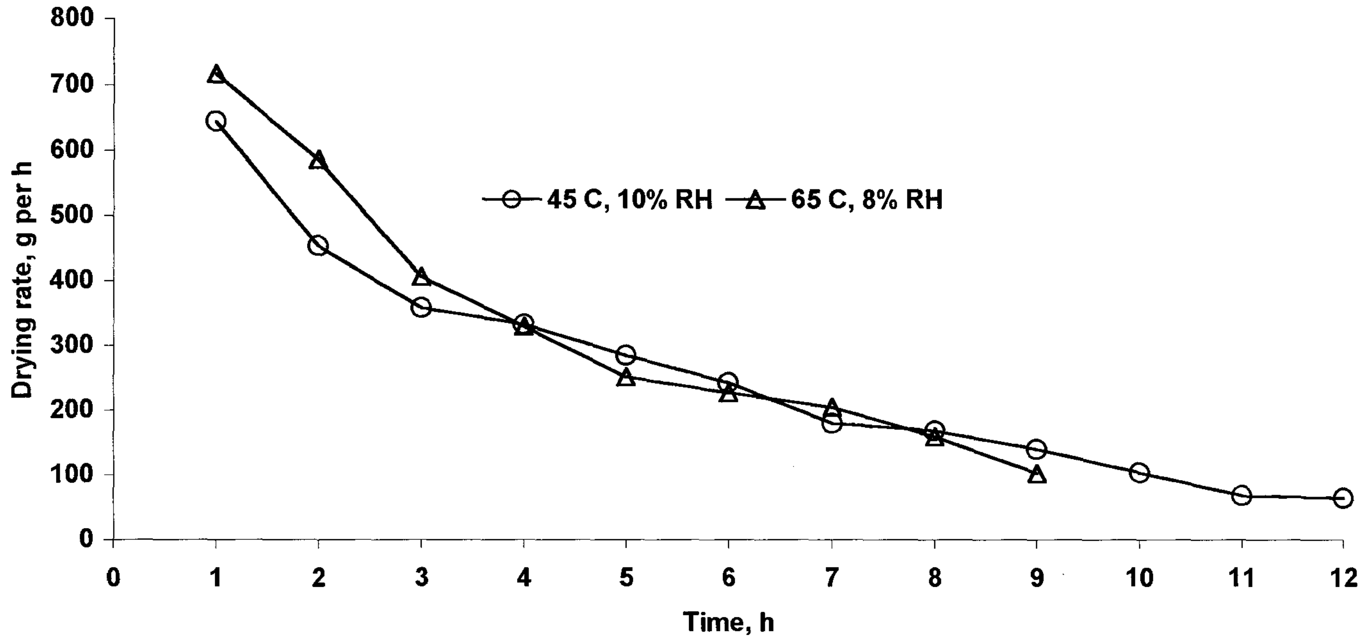

Figure 4.8 Drying curves for apple reduced to 10% final m.c. (3.5 kg initial weight and 86.5% initial m.c. for 65°C and 3.6 kg initial weight and 87.5% m.c. for 45°C trials).

removed for drying at 45°C and 65°C, respectively (specific moisture extraction rate = 0.55 kg and 0.75 kg per kWh, respectively). The ambient conditions were favourable (16-20°C, 45-65% RH) for the process. Not much information regarding the energy consumption for drying of fruits was found in the literature to comment on the efficiency of the setup with respect to the heat transfer characteristics (Chou and Chua, 2001). But the values provided a basis for comparison of the data obtained from HPD assisted drying trials.

4.3.2 Closed Loop Reheat Mode

Typical drying curves of the three materials used for drying experiments are shown in Figure 4.9 and 4.10. Tomato and the agar curves resemble a characteristic drying curve with two zones - a rather distinct constant rate period followed by the falling rate period. However, the constant rate period for apple, if any existed, is not clear. The initial spurt in the drying rate is due to the free water released from the cells during the slicing of the fruit and is merely a clean up of the surfaces. Further drying involves removal of water from the parenchyma tissue. There is more free water available in tomato due to its anatomical features and hence it is reasonable to expect the constant rate period. Agar is a phycocolloid comprising a heterogeneous family of linear galactan polysaccharides (sulfated galactans) that possess the ability to form reversible gels by cooling hot aqueous solutions. The process of agar gelation depends exclusively on the formation of hydrogen bonds. The subsequent formation of aggregates finally produces the macromolecules that constitute agar gels. Agarose, the gelling portion of agar, aggregate to form a three-dimensional framework, which holds the water molecules within the interstices of the framework. Hence agar gels are expected to present a more faithful representation of the "classic" drying curve with clearly defined constant rate and falling rate periods. Drying rates got extremely low after the final states shown in the Figure 4.9. For instance, it took almost 10 hours to bring about a 5-point reduction (25 to 20%, w.b.) in the agar gels.

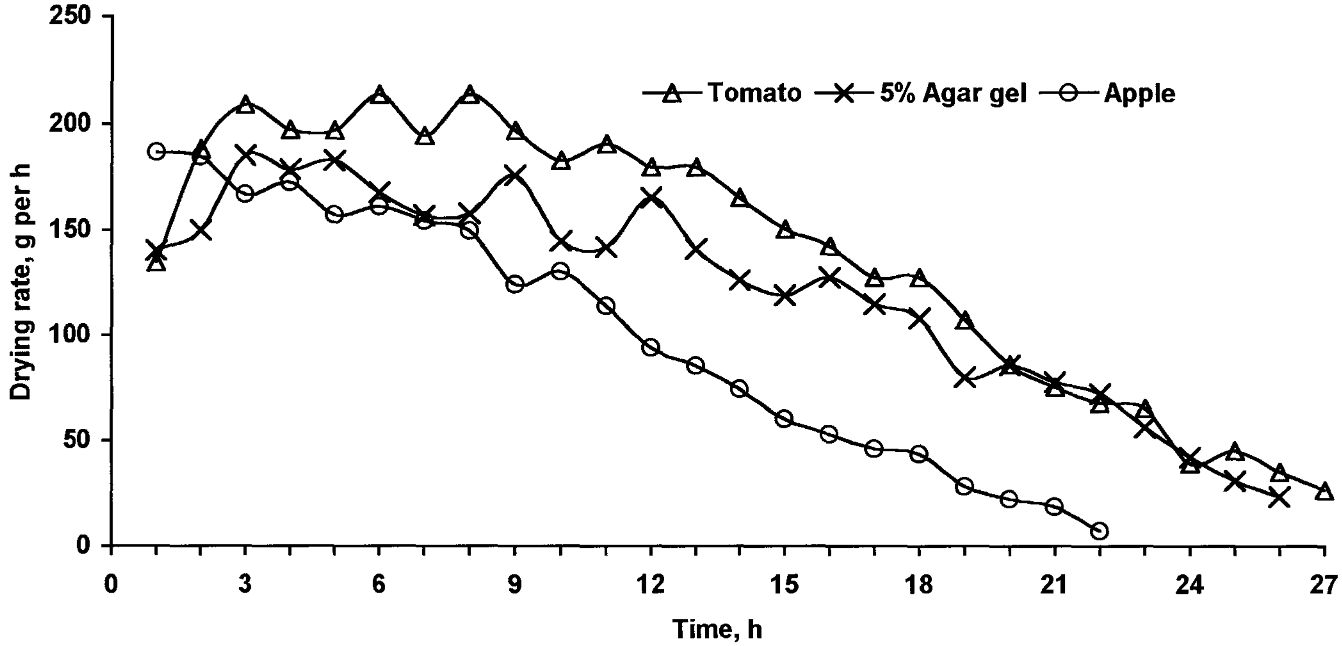

Figure 4.9 Typical drying curves for apple (2.7 kg initial weight, 85% initial m.c. and 12% final m.c.), tomato (4 kg initial weight, 94% initial m.c. and 8% final m.c.) and agar gel (3.38 kg initial weight, 95% initial m.c. and 25% final m.c.) in the closed loop, reheat mode.

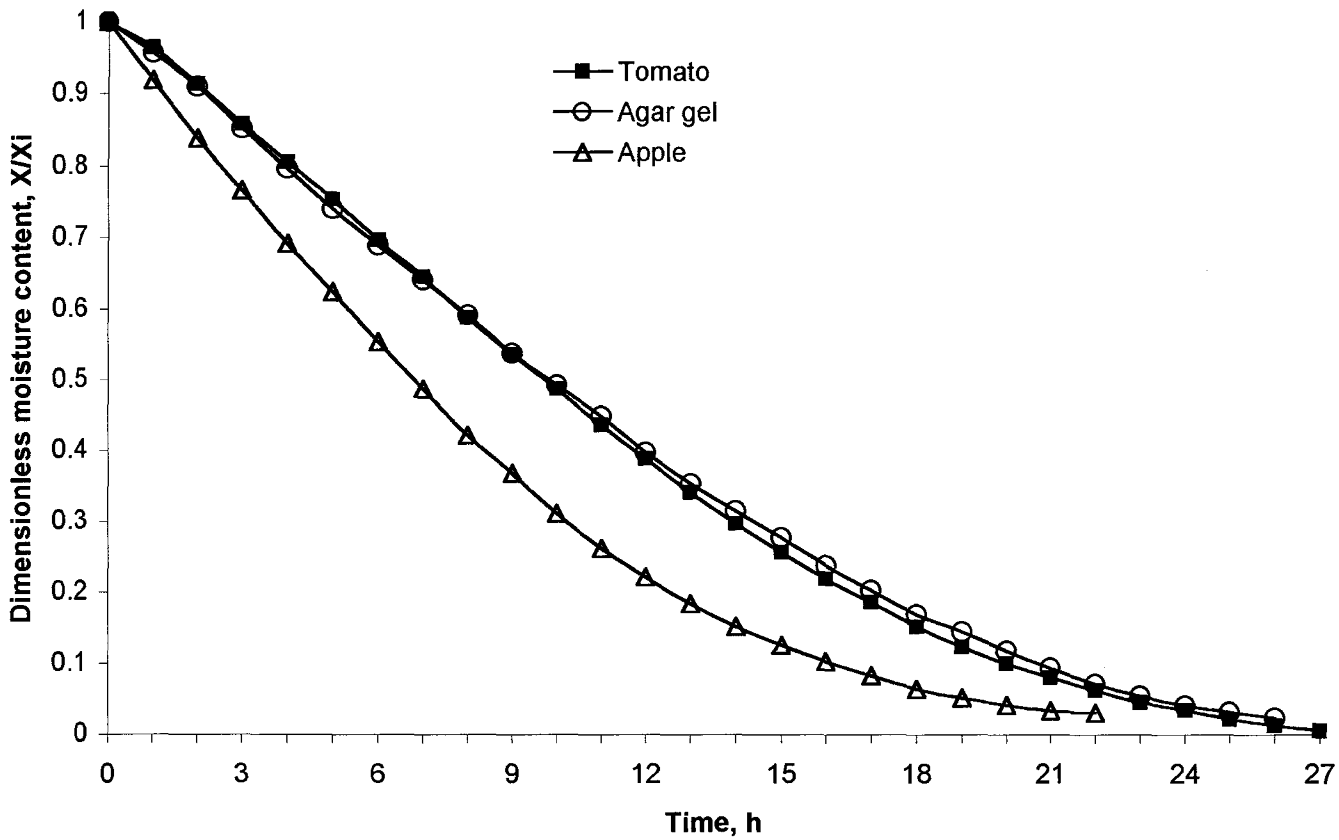

Figure 4.10 Dimensionless moisture ratio (X/X_i - X_i being the initial moisture content) of the material (details in Figure 4.9) during drying in the closed-loop, reheat mode.

In all the drying trials, fruits of similar size and weight were selected to fill the drying chamber to its capacity volume; the mass of the entire batch was monitored and the representative moisture content was used to calculate the desired final weight. But in reality, the contents of the drying chamber lose moisture at different rates spatially and the final designated moisture content is the mean of the entire load. While it is unrealistic to expect a dryer to achieve completely uniform drying, a good one enables the production of individual samples whose final moisture content huddle close around the mean. In order to study this performance, trials were conducted with apple and agar gels. The averaged results from five such trials using apple are shown in Figure 4.11. A distribution curve would have a kurtosis value of 1.43 (indicating a close cluster around the mean) and would be positively skewed indicating a higher proportion of products with final moisture content equal to or lower than the mean moisture content value.

Four trials were conducted with agar gels (5% solids) wherein the load matrix was weighed at 3-hour intervals to observe the spatial distribution of drying rate. Other than the indication that the slices closer to the air inlet dried at a slightly higher rate, the pattern was random and could not provide any specific trends.

4.3.3 Closed Loop Heat Reclaim Mode With Heater

The drying time could be decreased by 25% with the use of heaters at the beginning of, or to finish the process. The temperature of the air was raised to 45°C and 65°C (with the compressor switched off and dampers kept open) after different initial periods of operation that reduced the moisture content of the load to 50, 40 and 30% (wet basis). The treatment, as expected, hastened the completion of the process; its implications on the quality of the dried material will be discussed later. Similarly, the use of heaters initially for a period of one or two hours with temperature controls set at 45°C and 65°C reduced the drying period.

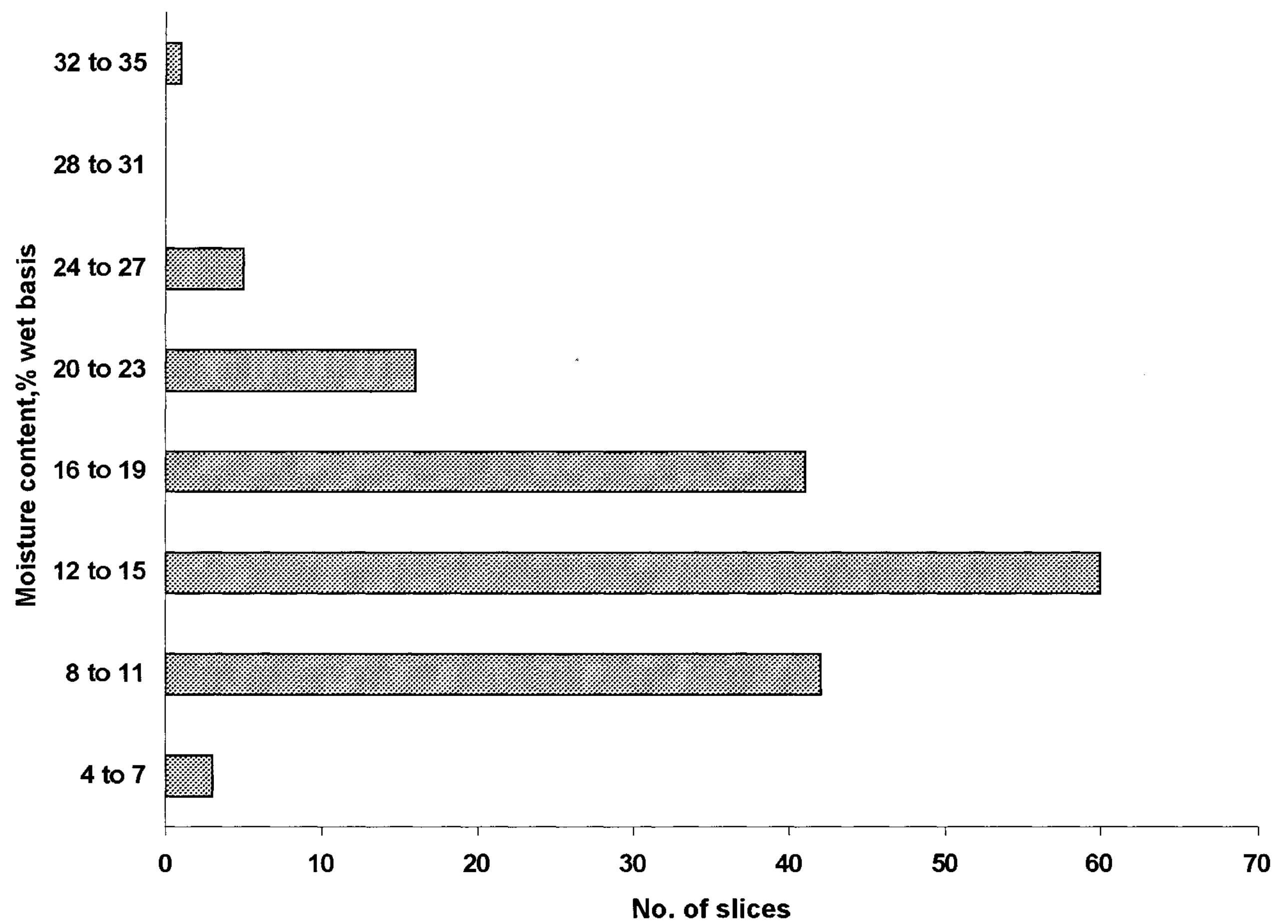

Figure 4.11 Distribution of final wet basis moisture content values for apple dried in the HPD assisted system (3.5 kg initial weight, 86.5% initial m.c. and 15% final m.c.) in closed-loop, reheat mode.

4.4 Energy Comparison

The specific energy consumption (SEC) and the specific moisture extraction rates (SMER) calculated for the different treatments are shown in Figure 4.12. Based on the report of Chou and Chua (2001), the values for hot air treatments might be considered reasonable.

The values related to the HPD assisted drying trials are lower than those reported in literature (Chou and Chua, 2001; Prasertsan and Saen-Saby, 1998a; Soponronnarit et al., 1998; Perera and Rahman, 1997; Mason et al., 1994). However, it should be noted that the system was not designed to operate solely as a dryer; it is capable of performing two or more simultaneous functions (e.g., space dehumidification). The energy appraisal should therefore discount the portion of the energy input that benefits the parallel process. The HPD transfers 60-90 kJ per minute to a water line at the secondary condenser during the drying of apple (Figure 4.4). If this portion of energy is discounted from the total energy input to the HPD, the SEC for apple reduces to well below 20 MJ per kg water removed (SMER of 0.18 kg per kWh input). The energy calculations involving HPD assisted systems should consider the "opportunity cost" of the low-grade heat that is rejected by conventional dryers while performing similar drying functions.

The results underline some of the main problems associated with the adoption of HPD for drying applications. The above mentioned literature have reports of successful adoption of the heat pump technology using units designed specifically for the drying purpose. They involve the use of variable speed compressors, multiple heat exchangers and pressure regulation valves that enable the system to respond effectively to the drying phases. In such cases, the HPD is an integral part of the dryer and is usually assisted by supplementary heat input. The HPD unit (used for multiple applications) has more potential if it is used only for dehumidification of the process air during drying process. In such a case, the machine has to be positioned either between the ambient air and the heat input or in

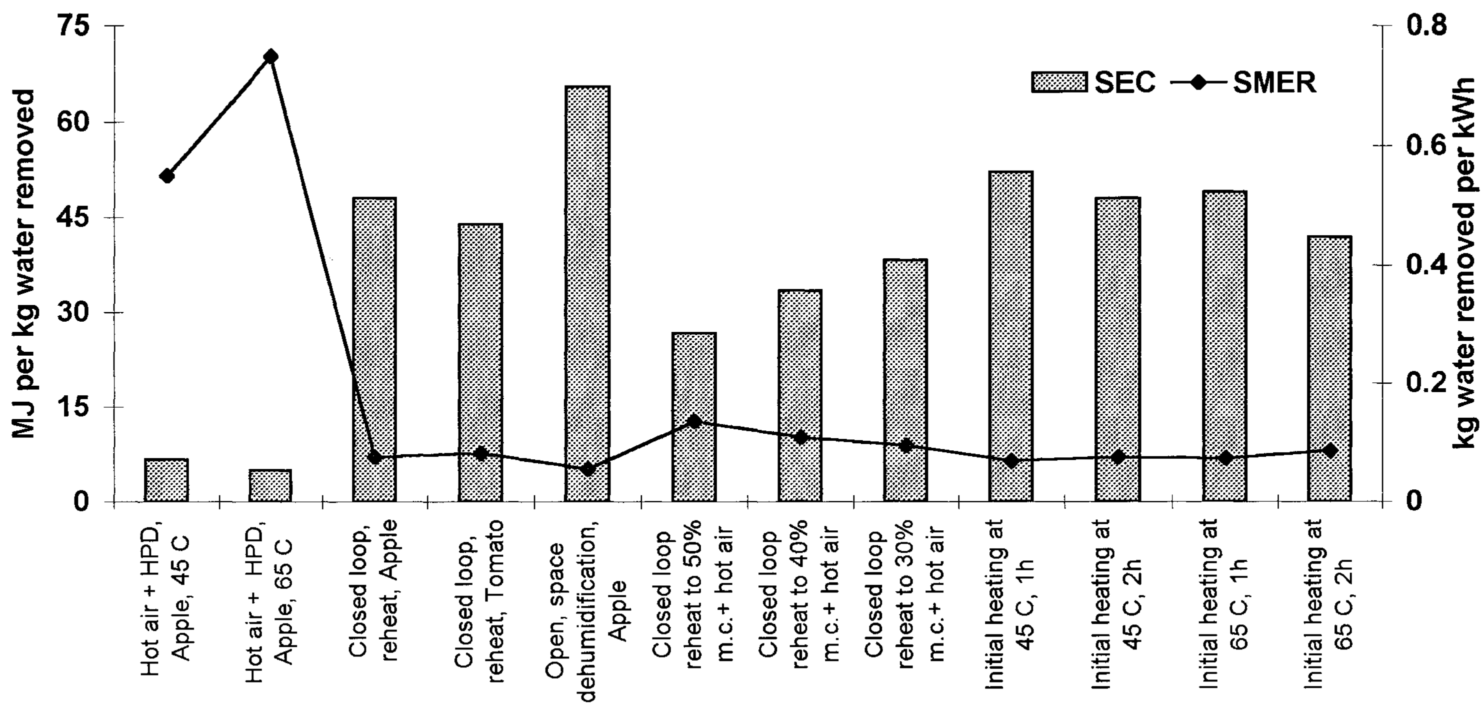

Figure 4.12 Comparison of energy consumption among the different modes of drying.

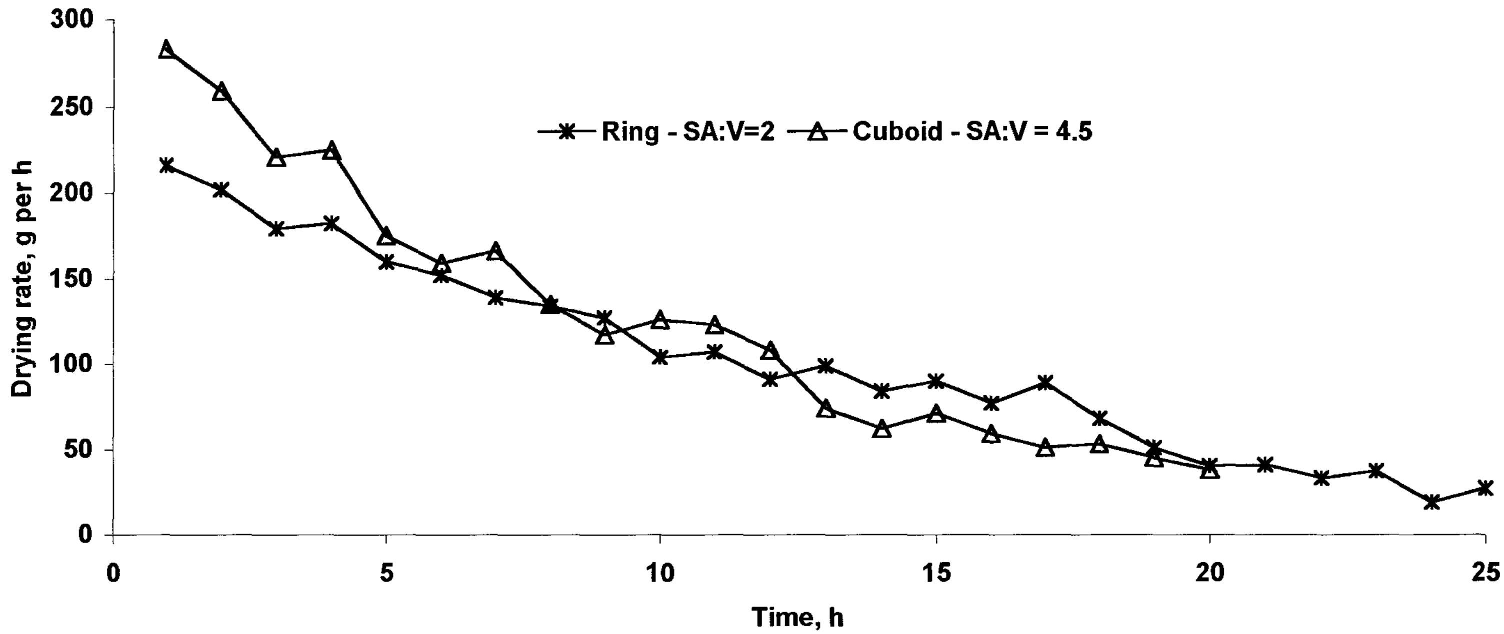

Figure 4.13 Effect of material geometry on the drying rate of Apple (3 kg initial weight, 86.5% initial moisture content and 11% final moisture content) in closed, reheat mode of operation, SA - surface area and V - volume.

a split design, the evaporator is placed at the exit of the dryer while the condenser is located between the ambient air and the heater.

4.5 Effect Of Material Geometry

Modification of particle size was one of the options available for enhancement of heat and mass transfer in this setup. Size reduction not only makes a larger surface area available for drying, but also increases the free water content by releasing more water from the fruit tissue.

The typical drying curves for apple with two different surface area-to-volume ratios dried under the closed loop, reheat mode are shown in Figure 4.13. In the case of apple rings, the peel was kept on; however, in the cuboid samples, only pieces with no peel were selected. The latter show a better drying rate overall; the drying rates for the first hour ranged from 230-280 g as opposed to 196-227 g for the rings.

4.6 Modeling Of The Dryer Operation

As an integral part of the drying system, the HPD provides a unique drying situation where drying takes place only by the difference in the water potential between the product and its environment. The energy consumption for such a drying process is dependent on the efficient matching of the capacity of the HPD with the drying rate of the material. But simultaneously, the drying rate is dependent on the conditions of the air entering the drying chamber.

In the current situation, the speed of the compressor is fixed and the mass flow rate of the refrigerant is a variable that could not be measured during the study. On the other hand, the drying behaviour of the material under the varying temperature and humidity conditions presented at the drying chamber is difficult to predict based on theoretical models. The exercise would also require the measurement of parameters such as the sorption isotherm (in the temperature range for desorption), material porosity, shrinkage behaviour etc. and these were beyond the scope of the study. It was decided to approach the problem armed with

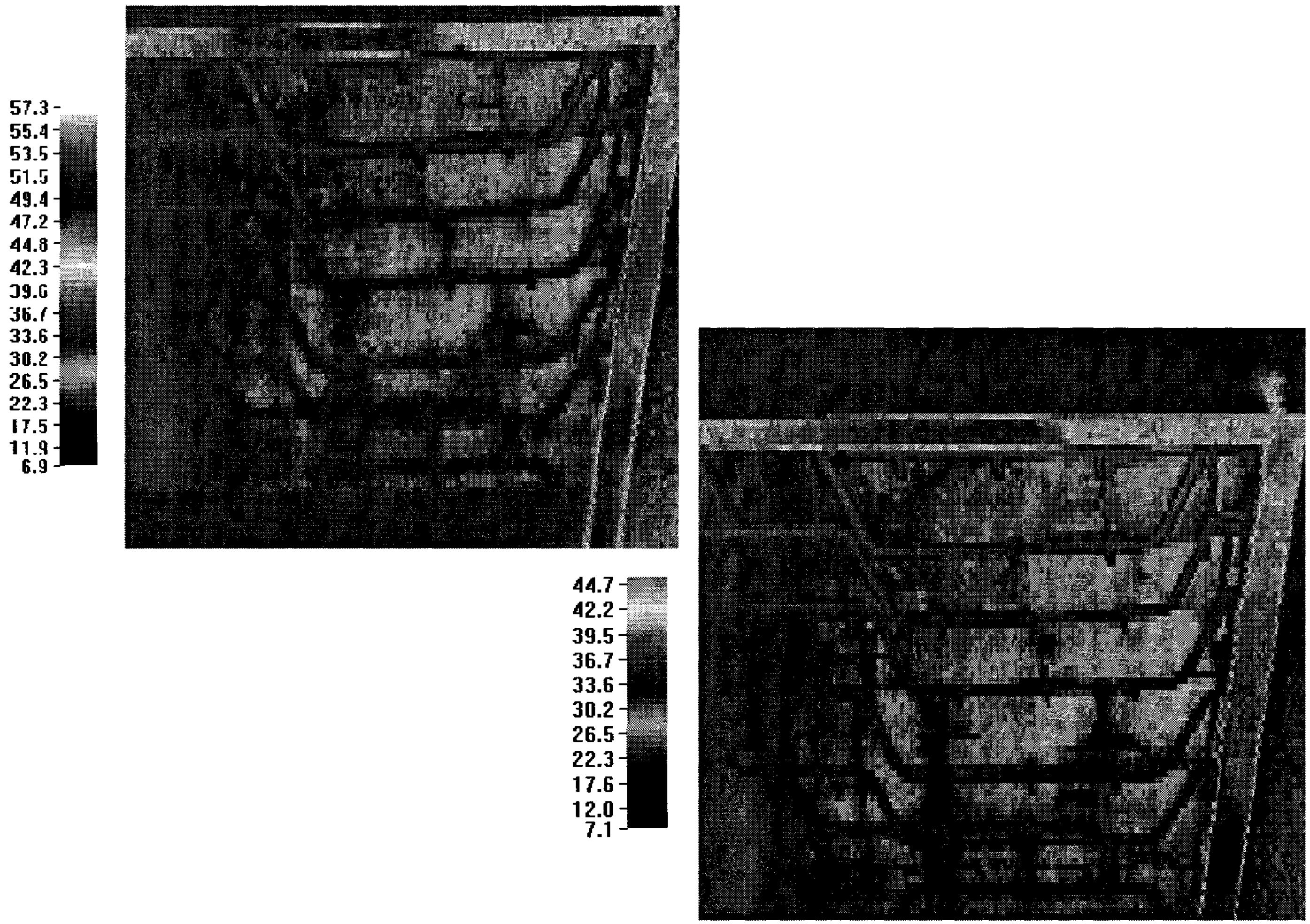

Figure 4.14 Two sample thermal images indicating the surface temperature of the apple slices in the dryer during drying in closed, reheat mode of HPD operation. Corresponding temperature of air in the drying chamber was between 25 and 27°C.

the substantial amount of data made available with the experimental runs. The following factors were considered to arrive at a predictive model of the behaviour of the machine.

- The drying takes place at temperatures close to the temperature of the drying air. Thermal images with an infrared imager were captured during the drying trials at regular intervals (two such images are shown in Figure 4.14 when the temperature of the air was around 25–27°C) that indicate that the surface temperature of the apple slices was almost equal to the temperature of the air. The rapid dynamic equilibrium of temperature between the material and air is assumed to occur at the beginning of the time segment. Amount of heat transferred to apple was calculated based on its cp value (Appendix 2).
- The conditions of the air entering the drying chamber vary continuously over the drying period. However, small time slices (10-minute segments) could be considered during which the average temperature and RH of the air are determined and assumed to represent constant conditions.
- The process in the drying chamber is assumed to be adiabatic saturation of air.
- It is assumed that there are no heat or mass losses in the system. Similarly, the water condensing at the evaporator is assumed to leave the system and its temperature is assumed to be the same as that of the air leaving the evaporator surface.
- Psychrometric calculations were based on the relations prescribed by ASHRAE (1997).

4.6.1 Physical Description Of The Process

1. Air enters the drying chamber (point A on the psychrometric chart in Figure 4.15) at the ambient dry bulb temperature and RH (T_1 and R_1). The air properties T_{wb1} (wet bulb temperature), h_1 (specific enthalpy) and w_1 (humidity ratio) can be calculated for this point. The pressure in the system (P) and mass flow rate

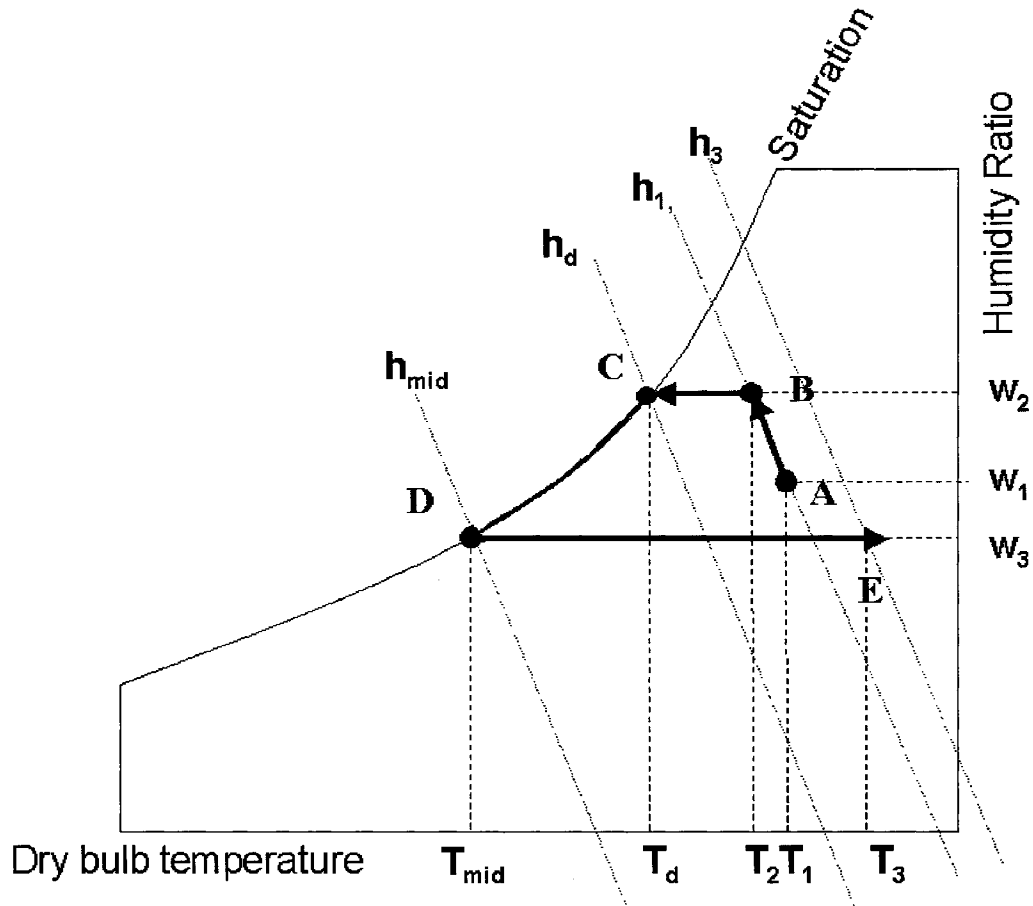

Figure 4.15. Psychrometric illustration of the process in the HPD assisted drying.

of dry air (M_a) are measured. The total values of the properties are designated by upper case letters H and W.

$$H = h.M_a \qquad \text{Equation 4.1}$$

$$W = w.M_a \qquad \text{Equation 4.2}$$

2. The process in the drying chamber is adiabatic saturation and proceeds along the constant wet bulb line on the psychrometric chart (line AB). The drying rate for this period will provide the amount of moisture added to the air in the drying chamber. The statistical method devised to calculate the drying rate will be described shortly. The conditions of the air leaving the drying chamber can now be calculated knowing the T_{wb} (=T_{wb1}), P and w_1. The properties are now T_2, R_2, h_2 and w_2. It is also important to calculate the T_d (dew point) for the air at this point as well as the enthalpy at saturation (h_d).

3. At the evaporator, the air is cooled and the temperature moves from B towards C (Figure 4.15). The quantity of heat transferred to the evaporator in this step is the sensible heat.

$$H_s = H_2 - H_d \qquad \text{Equation 4.3}$$

The total heat lost by the air (and gained by the evaporator, H_e) in a dehumidifying process is the sum of the sensible and the latent heat (H_l) fractions. If the heat transfer characteristics of the evaporator are known, it is possible to calculate the quantity of heat extracted by the refrigerant under given conditions. But the heat transfer coefficient of the evaporator was not available. To overcome this handicap, the observed data was used to predict the heat removal behaviour of the evaporator. The quantity of heat transferred to the evaporator was predicted based on the total enthalpy in the air at the evaporator and the latent heat fraction.

4. On the surface of the evaporator, air is saturated after the loss of sensible heat fraction, H_s. The quantity H_l is used to calculate the total amount of water condensed at the evaporator (M_w). The humidity ratio of the air is now reduced to w_2 after the evaporator and the air is saturated (R_{mid}=100%). These inputs are used to estimate the temperature T_{mid} between the evaporator and the condenser. The

enthalpy of the water leaving the system (H_W) can be calculated by knowing M_w and assuming that it leaves the system at T_{mid}. The enthalpy H_{mid} of the air is calculated as

$$H_{mid} = H_2 - (H_e + H_w) \qquad \text{Equation 4.4}$$

At the end of dehumidification, the air is at point D.

5. Since the heat transferred at the condenser is only sensible heat, the total enthalpy exchanged can be calculated by knowing the surface area of the heat exchanger, the heat transfer coefficient and the T_{mid}. The heat transfer coefficient calculated during the experiments was found to vary with the T_{mid}. Hence, the data points observed during the trials were used to obtain a regression equation that predicted the heat transfer coefficient for a given T_{mid}. The enthalpy transferred to the air is termed H_c (with specific enthalpy h_c) and the air is marked at point E. The temperature T_3 at E can be calculated using w_2, h_3 and P.

6. Air enters the drying chamber at conditions T_3, R_3, h_3 and w_3 to begin the next segment of iteration. The iteration is continued until the desired final moisture content is attained in the drying load. The iterations for the first hour are done for one-minute steps and 10-minute steps thereafter. Ambient air conditions of 20°C and 50% RH were considered for initial guesses.

4.6.2 Statistical Models - Enthalpy Exchange At The Evaporator

The heat exchange at the evaporator involves both sensible and latent heat. Over 3000 data sets covering the possible air properties encountered during the experimental trials were analyzed to predict the enthalpy transfer. A strong correlation was observed between the total enthalpy transfer to the evaporator (H_e) and the two indicators - total enthalpy of the air arriving at the evaporator (H_2) as well as the ratio of the latent heat to the total enthalpy in the air (H_{l2}/H_2; H_{l2} is calculated from W_2). It was also observed that there were two ranges of H_2 over which the correlation differed.

For H_2 values over 800 kJ (during a minute), the correlation between H_e and H_2 was positive. However, for values below 750 kJ, the H_e values showed a slight negative correlation. Similarly, for latent heat fractions below 0.37, the correlation was negative. This behaviour could be explained based on the observations recorded while operating the dryer in that range. The lower values occur mostly during the late stages of drying or when the dryer is presented with smaller moisture loads. Under these conditions, the evaporator surface is severely frosted, altering the heat transfer characteristics. Based on this observation, the data was divided into two groups separated at 800 kJ values for H_2 and analyzed separately. A multiple regression equation shown in Equation 4.5 was obtained for each group and the parameters are given in Table 4.1.

$$H_e = b_0 + b_1H_2 + b_2H_r + b_{11}H_2^2 + b_{22}H_r^2 + b_{12}H_2H_r \qquad \text{Equation 4.5}$$

H_e = quantity of heat absorbed at the evaporator (per minute), kJ
H_2 = total enthalpy of the air arriving at the evaporator (per minute), kJ
H_r = fractional ratio of latent heat to total heat content of the air at the evaporator

Table 4.1 Regression coefficients for estimation of heat exchange at the evaporator

Range of input variables	Parameter estimates
H_2 = 450 - 800 kJ H_l/H_2 = 0.3 - 0.4	b_0 = 1287.46 b_1 = 0.206 b_2 = -3985.83 b_{11} = -0.00211 b_{12} = 7.716 b_{22} = -3180
H_2 = 800 - 1050 kJ H_l/H_2 = 0.4-0.5	b_0 = 1878.65 b_1 = 3.54 b_2 = -14206 b_{11} = 0.001816 b_{12} = -13.6 b_{22} = 28374

4.6.3 Statistical Models - Drying Rate

The selection of the parameters for arriving at a response surface was based on the observed behaviour of drying rate as well as theoretical considerations. Temperature and relative humidity of the air were obvious choices. The moisture ratio or the dry basis moisture content (mass of water per unit mass of solids) was also selected as a criterion that would represent the extent of "free water" in the material. However, it is a ratio that would not provide any indication of the actual mass of material sample in the dryer. It was noted that the initial mass of water in the load had an influence on the drying rate; hence the mass of water (in grams) present in the initial sample was introduced as one of the indicator variables.

A preliminary linear multiple regression routine run with the data set showed a discrepancy. In that model, temperature was negatively correlated with the drying rate. The data sets were obtained from trials where the variables were not controlled. The samples were loaded into the chamber, air was taken in for a minute and the dampers were closed. The temperature profile for rest of the process establishes itself and is not interfered with. Due to this situation, the highest drying rates are observed during the first hour when the temperatures are the lowest. As the process continues, the system warms up in 5-6 hours, but the drying rate either sustains or continues to fall. The reasons for the lag period could not be clearly identified; but it can be expected to be minimized in a well-insulated system.

In order to rectify this aberration, short trials (periods of 6-7 hours) were conducted. In these trials, the initial mass of samples was varied and the initial temperature was varied between 20 and 30°C with the use of heaters, to provide a broader data set. This was not only expected to account for the possibility of rapid stabilization in the system, but also to simulate situations where the ambient temperatures could be higher (most trials were conducted during winter and spring when the ambient conditions were mild and dry) during the beginning of the process.

The parameter estimates of the response surface generated (Equation 4.6) using the data are given below.

$$DR = b_0 + b_1 T + b_2 R + b_3 M + b_4 X + b_{11} T^2 + b_{44} X^2$$
$$+ b_{12} T.R + b_{13} T.M + b_{14} T.X + b_{23} R.M + b_{34} M.X$$

Equation 4.6

DR - drying rate, g per h
T - temperature, $^{\circ}C$
R - relative humidity, %
M - total weight of the water present in the sample (mass of sample X percent moisture content), g
X - moisture content, d.b., dimensionless (mass of water/mass of solids)

$b_0 = -461.72$
$b_1 = 44.8$
$b_2 = 14$
$b_3 = -0.27$
$b_4 = -44.41$
$b_{11} = -0.86$
$b_{12} = -0.7$
$b_{13} = 0.0082$
$b_{14} = 1.946$
$b_{23} = 0.0035$
$b_{34} = 0.00927$
$b_{44} = 2.73$

Response surface regression models are used mostly for optimization (Draper, 1981; Ratkowsky, 1990) and cannot substitute for a drying theory. The model adopted for this study was only intended to assist the attempt to comment on the potential of the HPD for handling larger samples and the energy expenditure in the process. The response surface extrapolates the sample size and predicts the drying times, which indicate the total energy consumed. Results of the successful simulation runs are shown in Table 4.2. The sample is dried from the initial moisture content of 85% to the final moisture content of 12% w.b. The sample is assumed to be apple since the data for the regression equations were obtained during drying of apple.

Table 4.2 Predicted performance of the drying system handling larger loads.

Sample Size (kg)	Predicted drying time (hours)	SEC (MJ per kg water removed)	SMER (kg water removed per kWh)
50	40	10.2	0.350
75	47	10.08	0.356
100	56	10.76	0.334
125	69	10.73	0.335
150	73	9.9	0.363

Each data point was verified for validity during iterations before passing it to the next step. Since it was not possible to measure the mass flow rate of the refrigerant or the temperature of the air between the two heat exchangers, the values of enthalpy transfer at the evaporator were worked out based on the observation that the air was dehumidified. However, this assumption does not hold good at all times during the simulation. In some instances, the values predicted for H_e were not sufficient to reduce the temperature of the air to the dew point. In such cases, only sensible heat is transferred and the predicted value for H_e is not valid. The iteration was aborted in such instances and the guess for the initial RH value of air taken in was increased till a compatible H_e value was obtained.

Scale-up and optimization are some of the main reasons for modeling dryer processes. Often, experiments are carried out on a small scale with extensive instrumentation and the results are used to build a model that predicts the performance of a system that is a magnified version of the lab setup. In this case, it was an unfortunate mix of the two extreme situations. The HPD has a capacity high enough to qualify as a pilot scale unit, but due to unavoidable reasons, comprehensive instrumentation of the unit on the refrigerant side was not possible. Besides, the unit was equipped with pressure regulating mechanisms that kept the operating conditions constant, which was not helpful in elaboration of the process, as the changes could not be measured. In order to acquire experimental data, the size of the drying chamber had to be small enough to allow real-time mass monitoring. While the system did provide conditions suitable to achieve drying, it was slightly handicapped for calculation of the energy expenditure.

4.7 Water Activity

The water activity of the apple being dried was measured by drawing out samples at different times during the drying trials. The results for the samples that were dried at 65°C and in the HPD assisted system are shown in Figures 4.16 to 4.18. The region of interest for comparison lies below the moisture content values of 0.5 (corresponding to 33% wet basis) that are observed during the final stages of drying. The scatter of the a_w values measured for the samples dried in the experimental treatment lie distinctly to the left of the scatter for the hot air drying (Figure 4.18), indicating lower water activity in the same range of moisture content. The monolayer moisture content of the widely used BET equation (Brunauer et al., 1938) calculated based on the observed values worked out to 5.4% d.b. (5.12% wet basis) for the hot air drying and 7% d.b. (6.54% wet basis) for the HPD assisted drying. The previously reported value of the monolayer moisture content for apple during desorption is 4.2% (3.84% wet basis) at 19.5°C in the exhaustive list compiled by Iglesias and Chirife (1982).

4.8 Rehydration

The rehydration behaviour of the dried samples, expressed as grams of water taken up per gram of solid content is shown in Table 4.3. The extent of rehydration of the HPD dried samples was significantly different (α = 0.01) from results for the hot air treatments (45°C and 65°C). However, when the material was dried initially in the HPD and the drying finished with heaters at 45°C, the difference in the rehydration capacity was not significantly different in both cases (reduction to 30 and 50% wet basis moisture content). The differences were significant when higher temperature (65°C) was used.

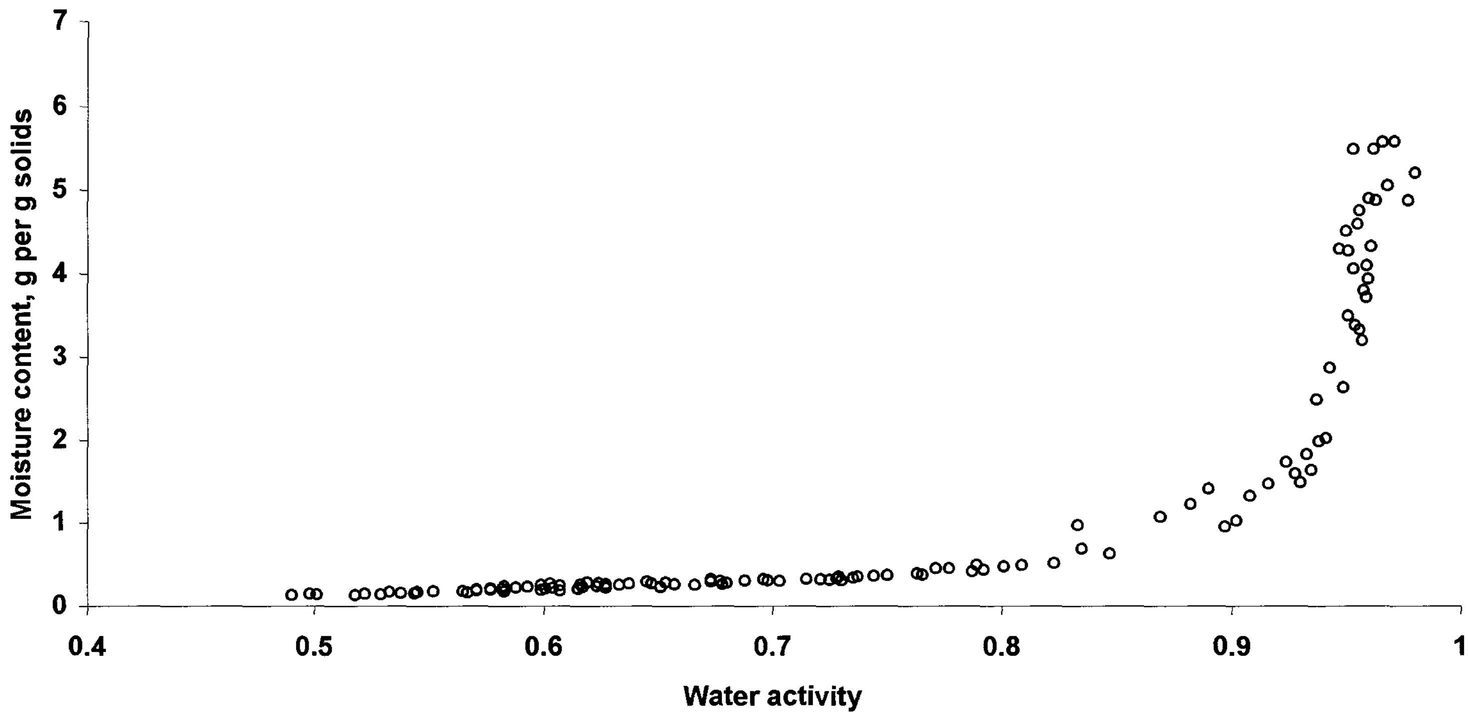

Figure 4.16 Water activity (measured at 25°C) of apple at different moisture contents during drying at 65°C.

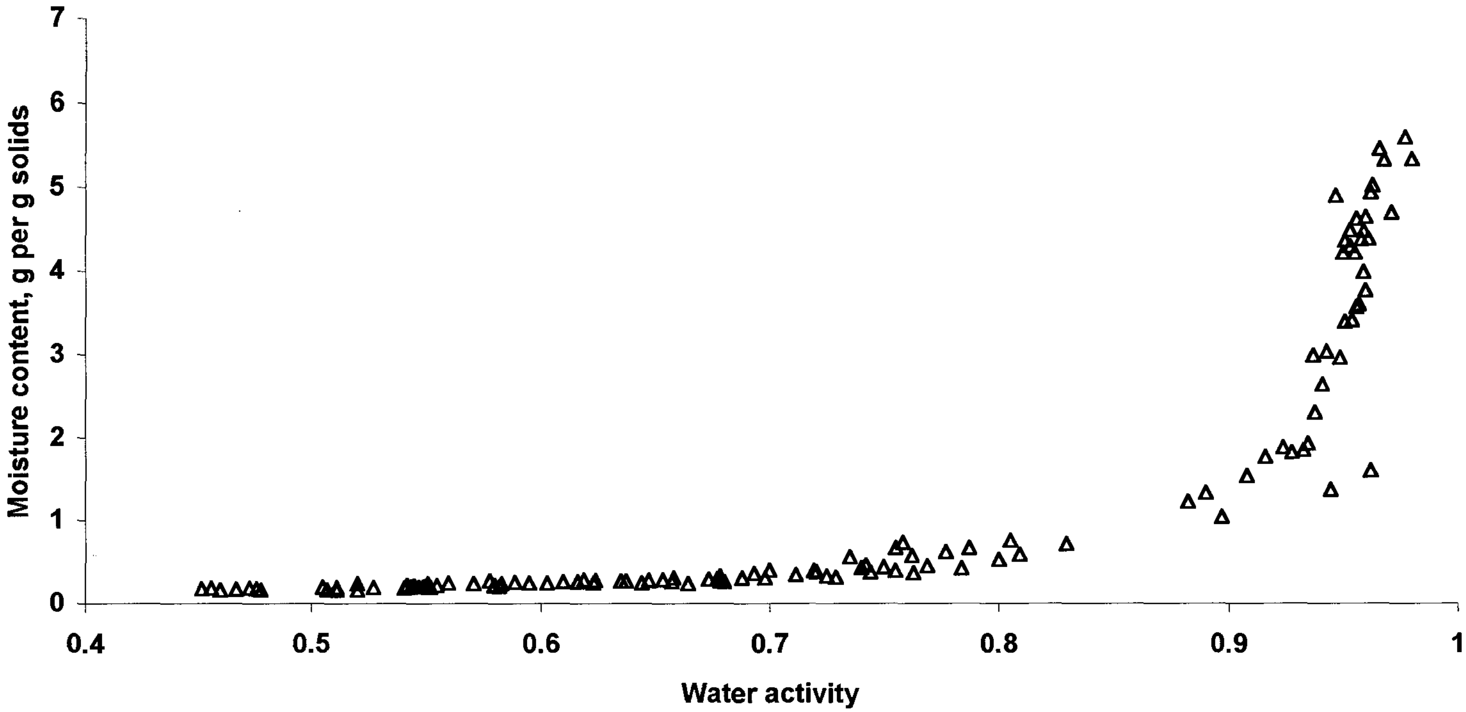

Figure 4.17 Water activity (measured at 25°C) of apple during the HPD assisted drying (closed, reheat mode, maximum drying temperature 28°C).

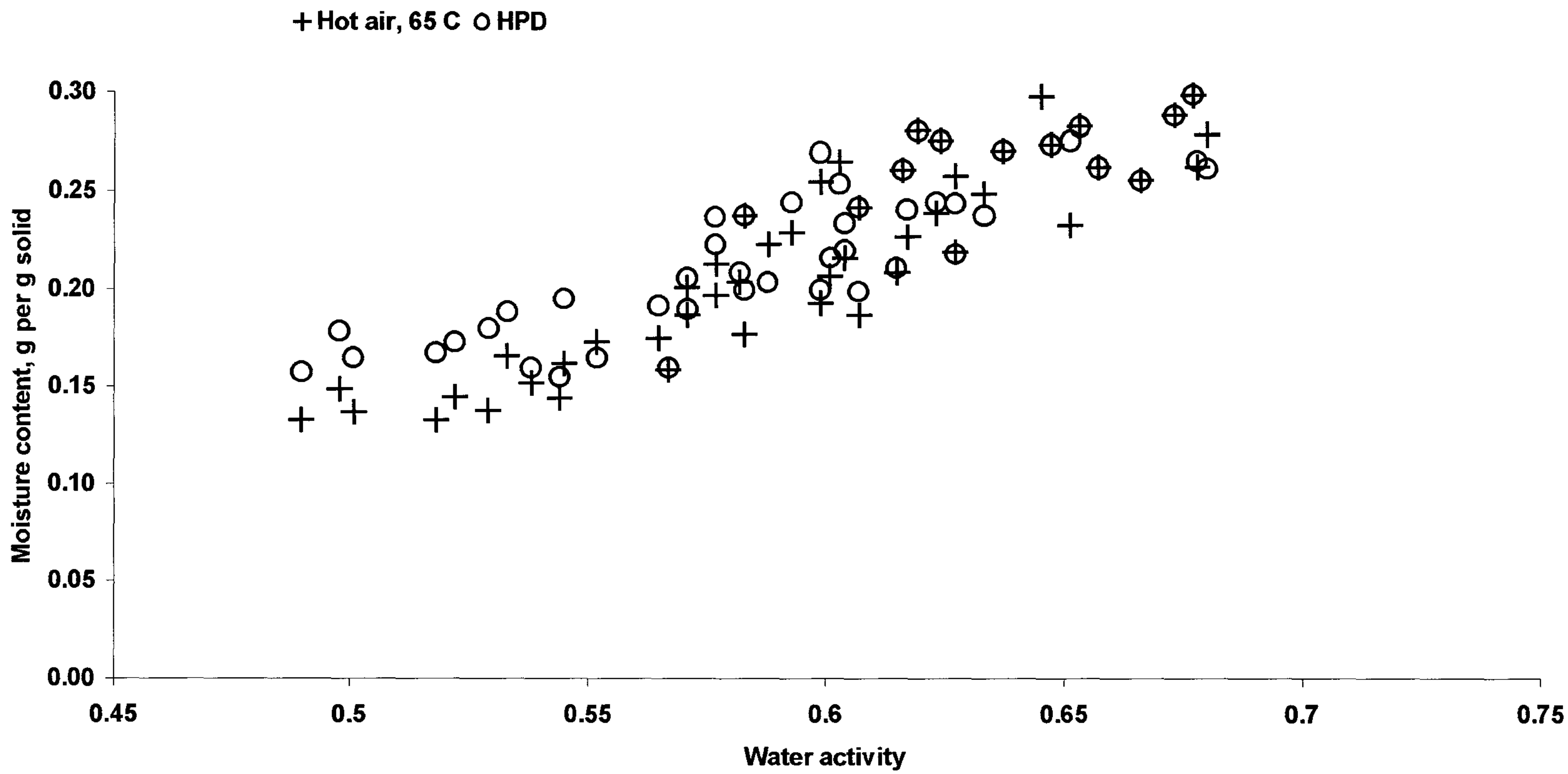

Figure 4.18. Section of the curves in Figure 4.16 and 4.17 showing the values at lower moisture content levels.

Table 4.3. Rehydration characteristics of the dried apple

Treatment	Rehydration capacity, g water per g solids	Rehydration time, minutes
HPD assisted drying	5.121 $\pm$ 0.103	50 $\pm$ 12
Drying at 45°C	4.895 $\pm$ 0.086	44 $\pm$ 9
Drying at 65°C	3.350 $\pm$ 0.205	31 $\pm$ 4
HPD to 50% m.c. + hot air, 45°C	4.982 $\pm$ 0.071	53 $\pm$10
HPD to 50% m.c. + hot air, 65°C	4.65 $\pm$ 0.121	49 $\pm$ 15
HPD to 30% m.c. + hot air, 45°C	5.028 $\pm$ 0.078	55 $\pm$ 16
HPD to 30% m.c. + hot air, 65°C	4.613 $\pm$ 0.069	51 $\pm$ 10

4.9 Colour Studies

The changes in the colour of the apple rings due to drying were compared based on the measured CIE 1976 L*a*b* values. Preliminary tests using four varieties of apples (Figure 4.19) revealed that the variety Gala was more susceptible to colour changes as perceived visually, and hence it was used to compare different treatments. It was also observed that the colour underwent change during the process of sample preparation for drying (slicing, arranging on trays and loading into the chamber). Colour measurements were made after preparing the samples and prior to loading, and the treatment is termed "Cut". The results are shown in Figure 4.20 and 4.21, and Table 4.4.

The difference in colour is estimated by measuring the change between a standard colour and the target colour (Appendix 3). In Table 4.4, the first treatment of the pairs indicates the treatment used as a reference when comparing the colour difference. It was found that the change in colour due to any drying process was significant when compared with the change that occurs due to sample preparation i.e. the drying process contributes significantly to the change in colour. However, upon visual observation, it was noticed that the change was of undesirable nature

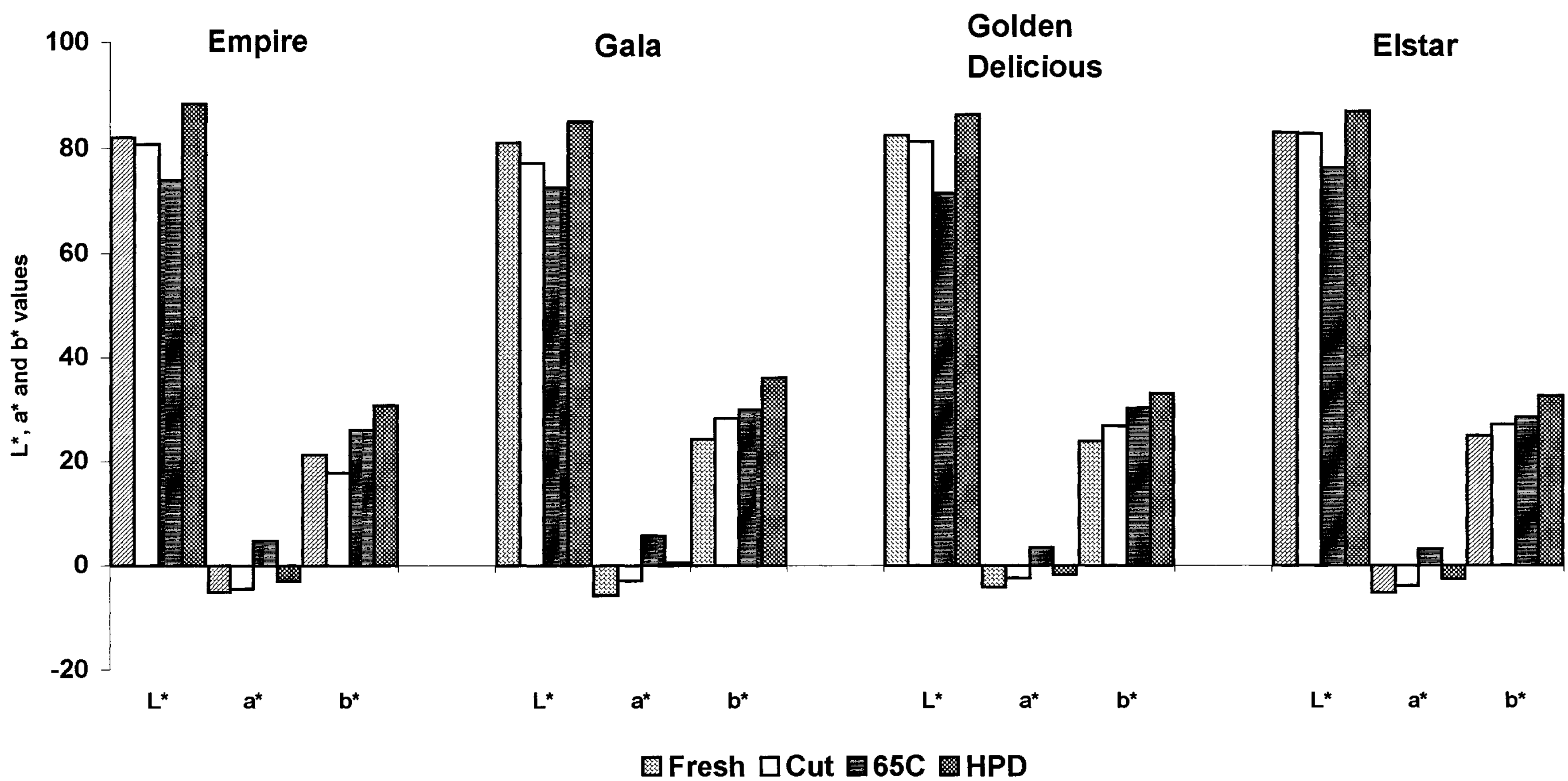

Figure 4.19 Comparison of colour changes in apple cultivars during drying.

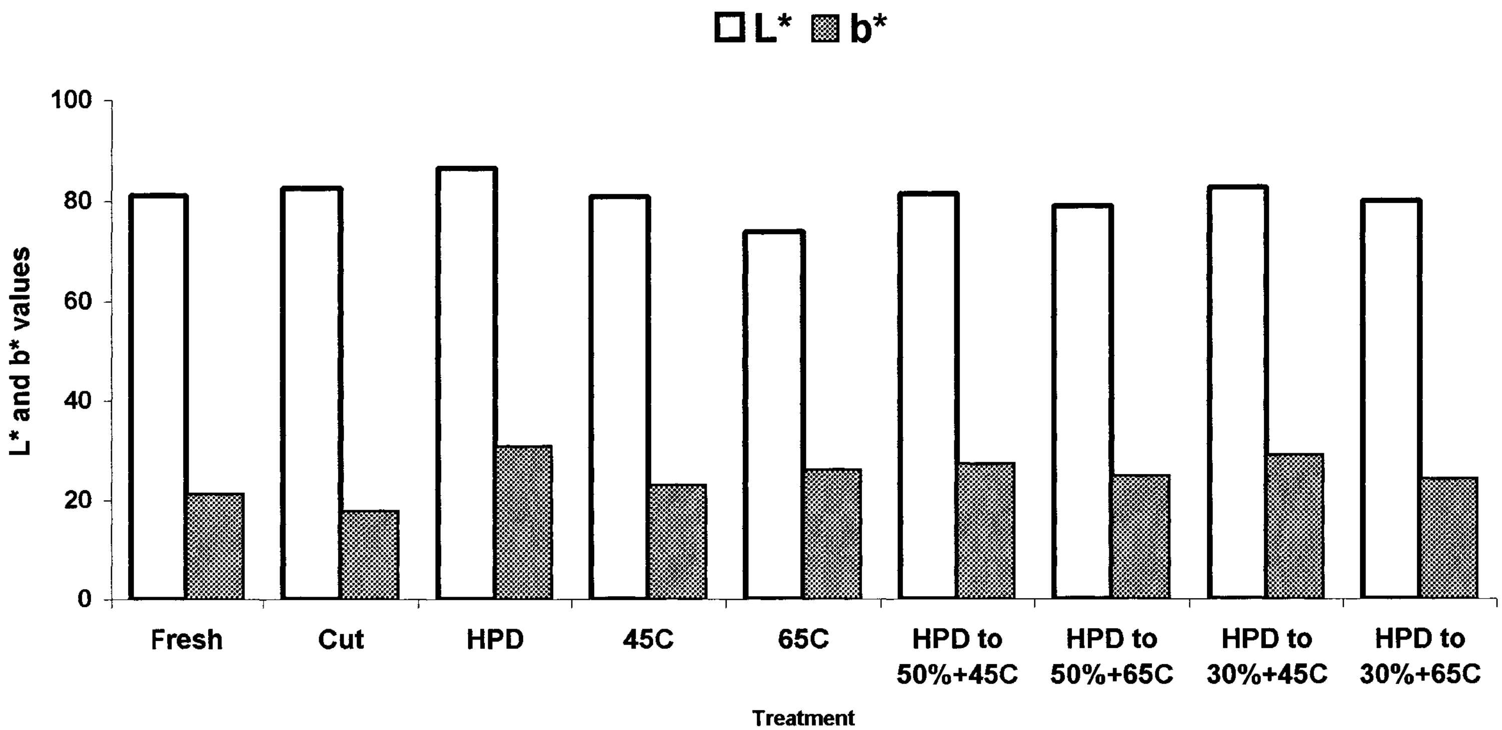

Figure 4.20 Comparison of colour changes (L* and b* values) under different drying regimes in apple, cultivar Gala.

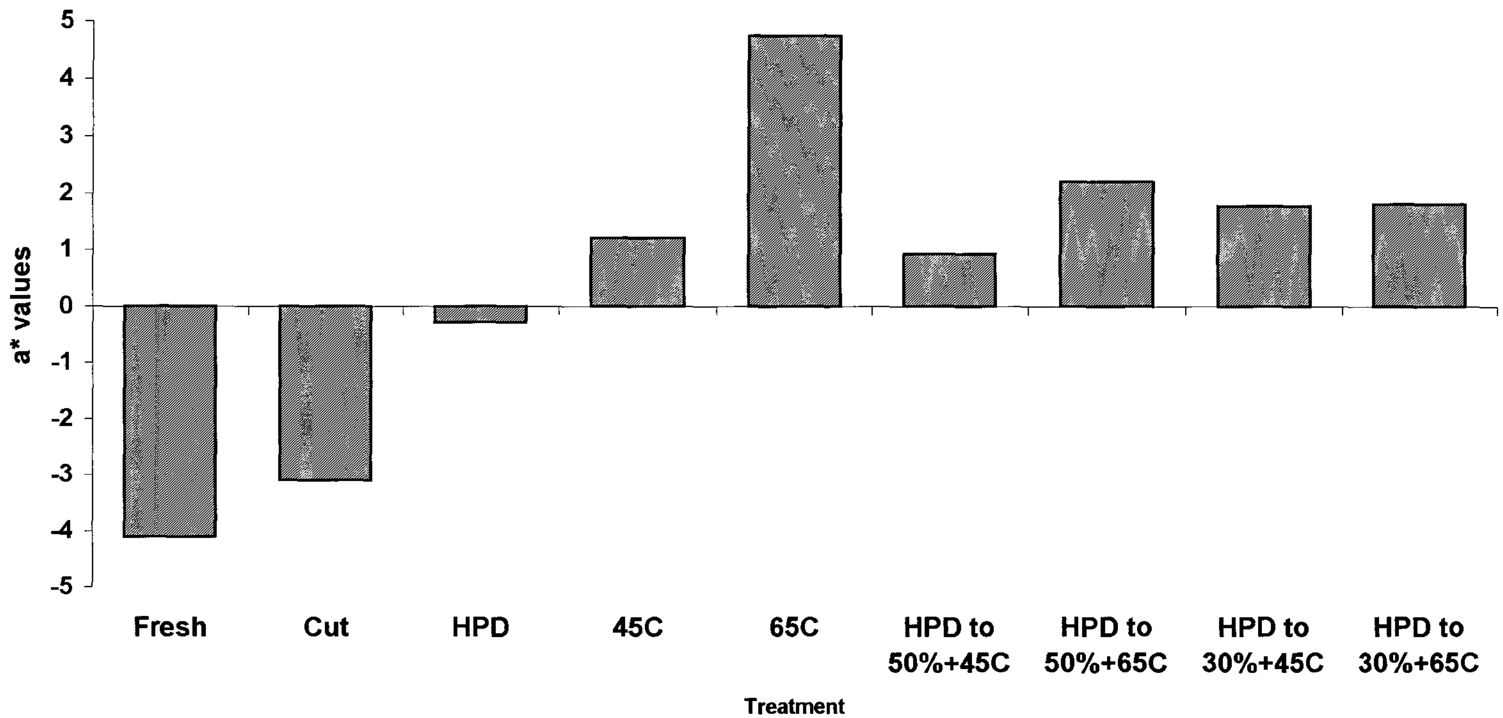

Figure 4.21 Comparison of colour changes (a* values) under different drying regimes in apple, cultivar Gala.

during drying at 65°C, whereas the changes calculated in case of the HPD assisted drying as well as at 45°C were actually favourable.

Table 4.4 Colour difference for different drying treatments

Comparison	ΔE^* value	SD
Fresh – Cut	4.936	1.071
Fresh – HPD	16.387	3.596
Fresh - 45°C	8.640	2.750
Fresh - 65°C	17.045	3.851
HPD - 45°C	10.689	3.625
HPD - 65°C	15.927	5.933
Cut – HPD	14.210	2.753
Cut - 45°C	16.475	4.007
Cut - 65°C	19.503	3.855
HPD - HPD to 50% m.c. + 45°C	7.493	3.627
HPD - HPD to 50% m.c. + 65°C	11.683	2.860
HPD - HPD to 30% m.c. + 45°C	3.712	2.609
HPD - HPD to 30% m.c. + 65°C	7.141	3.066

The colour change (ΔE^*) in the CIE $L^*a^*b^*$ colour space is the distance of a straight line between the points defined by the two sets of $L^*a^*b^*$ co-ordinates of the colours being compared (Appendix 3). The values shown in Figure 4.20 indicate that the L^* value for HPD dried samples increases whereas it shows a decreasing trend in the hot air dried samples. But the a^* (Figure 4.21) value clearly increases in all treatments, but to a higher degree at higher temperatures. Higher L^* values would make the samples appear brighter, whereas the increasing a^* values are associated with a shift towards red colour (perceived as browning).

Even though the difference was statistically significant in all treatments, the visual appearance of the HPD dried samples resembled that of fresh samples and was rated as desirable due to the brighter surface. There was no perceivable difference between the HPD dried samples and the samples dried in combination drying at 45°C. Higher temperature during drying caused a darker surface with distinct discolouration that was considered undesirable.

4.10 Organoleptic Observations

The responses and comments that were collected from the panel are presented in Table 4.5.

Table 4.5 Responses and comments of the panel for organoleptic examination

Treatment	**Characteristics**
Drying at 45°C	Satisfactory visual appearance Good flavour and taste Good consistency (slightly rubbery)
Drying at 65°C	Unsatisfactory visual appearance (dark colour) Satisfactory flavour and taste Unsatisfactory consistency (rubbery)
HPD assisted drying	Good visual appearance Good flavour and taste Very good consistency (chewy)
HPD drying to 50% m.c. + 45°C	Satisfactory visual appearance Good flavour and taste Satisfactory consistency (rubbery)
HPD drying to 50% m.c. + 65°C	Unsatisfactory visual appearance Good flavour and taste Satisfactory consistency
HPD drying to 30% m.c. + 45°C	Good visual appearance Good flavour and taste Very good consistency
HPD drying to 30% m.c. + 65°C	Satisfactory visual appearance Good flavour and taste Satisfactory consistency

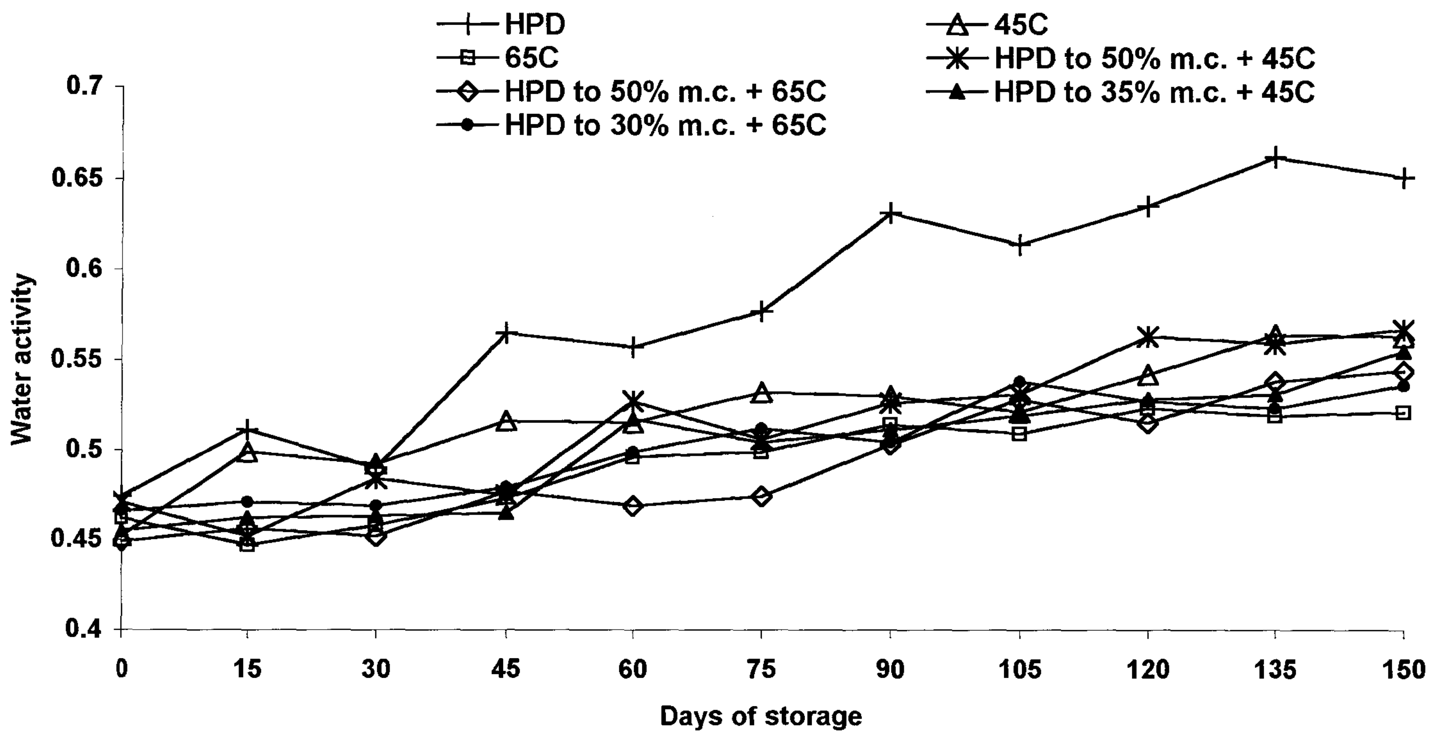

Figure 4.22 Water activity of dried apple samples during storage.

4.11 Storage Studies

The water activity of the stored samples tested at 15-day intervals is shown in Figure 4.22. Between 4 and 7 samples were tested for each data point; the standard deviations ranged from 0.015 to 0.039. It is clear that the water activity of the HPD dried samples was less stable when compared with other treatments where heating was involved.

The physical quality of the samples also showed different characteristics during storage. Both the hot air dried samples (45°C and 65°C) as well as the samples finished with hot air drying were very stable in their appearance and physical characteristics. No noticeable changes occurred on the surface of the samples. The appearance of the HPD dried samples, however, deteriorated after six weeks of storage. The samples had a soggy body with marked surface discolouration (browning or darkening).

4.12 Histological Observation

The microphotographs in Figures 4.23 to 4.25 show the parenchyma cells in the control (fresh) and dried samples for the cultivars Empire, Golden Delicious and Gala. The protocol for slide preparation involved rehydration and hence the observations are an indirect indication of the state of the cells upon drying. The dried tissues have more residual safranin in the cell walls, suggesting increased wall in-folding. Drying artifacts were most exaggerated in the hot air treatment compared with the HPD assisted samples and there were clear differences in cultivar response to heat treatment. Overall tissue integrity was most affected in Gala.

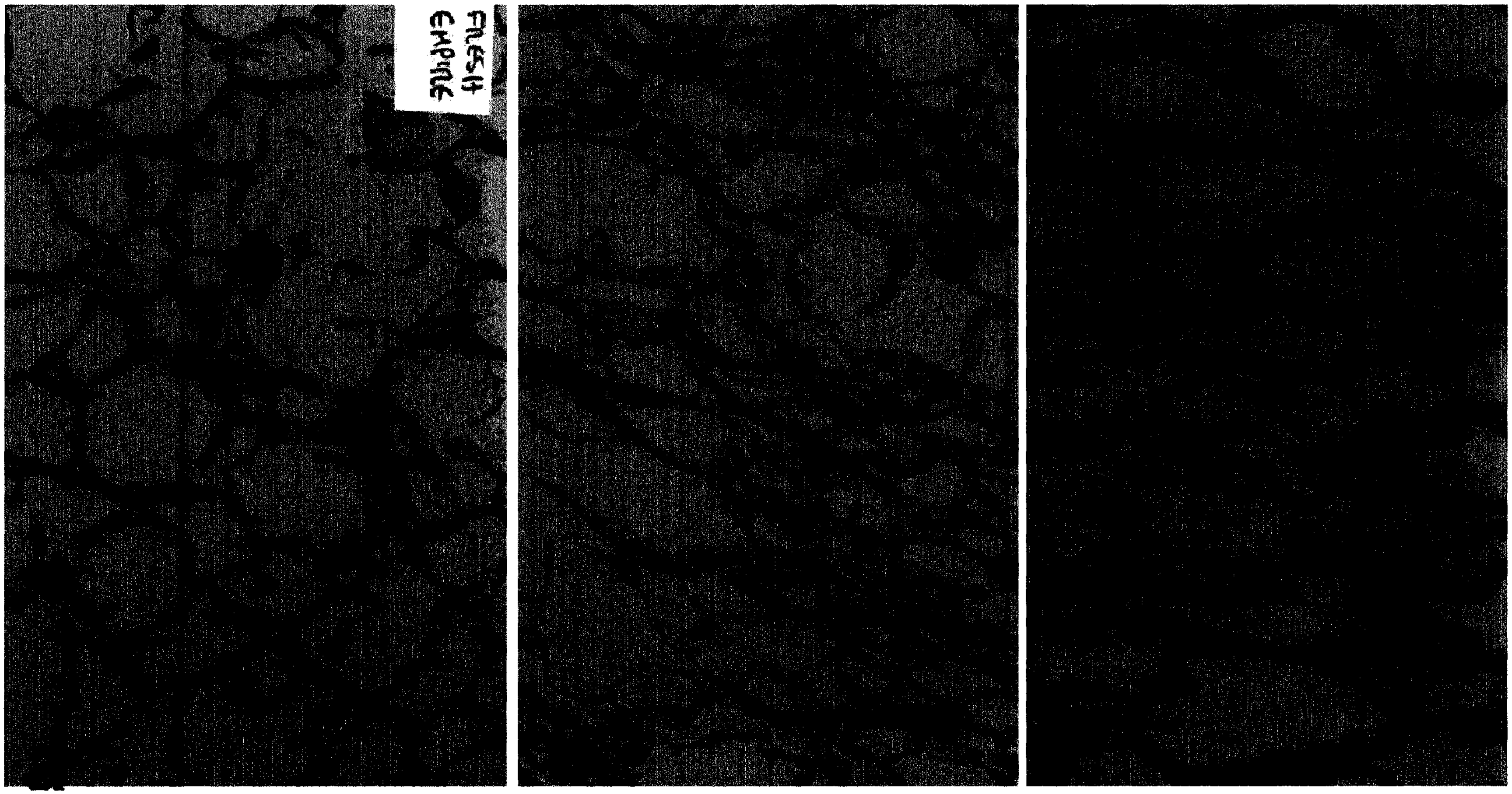

Figure 4.23 Histological comparison of apple, cultivar Empire (a - fresh sample, b - sample dried at 65°C and c - dried in HPD assisted system).

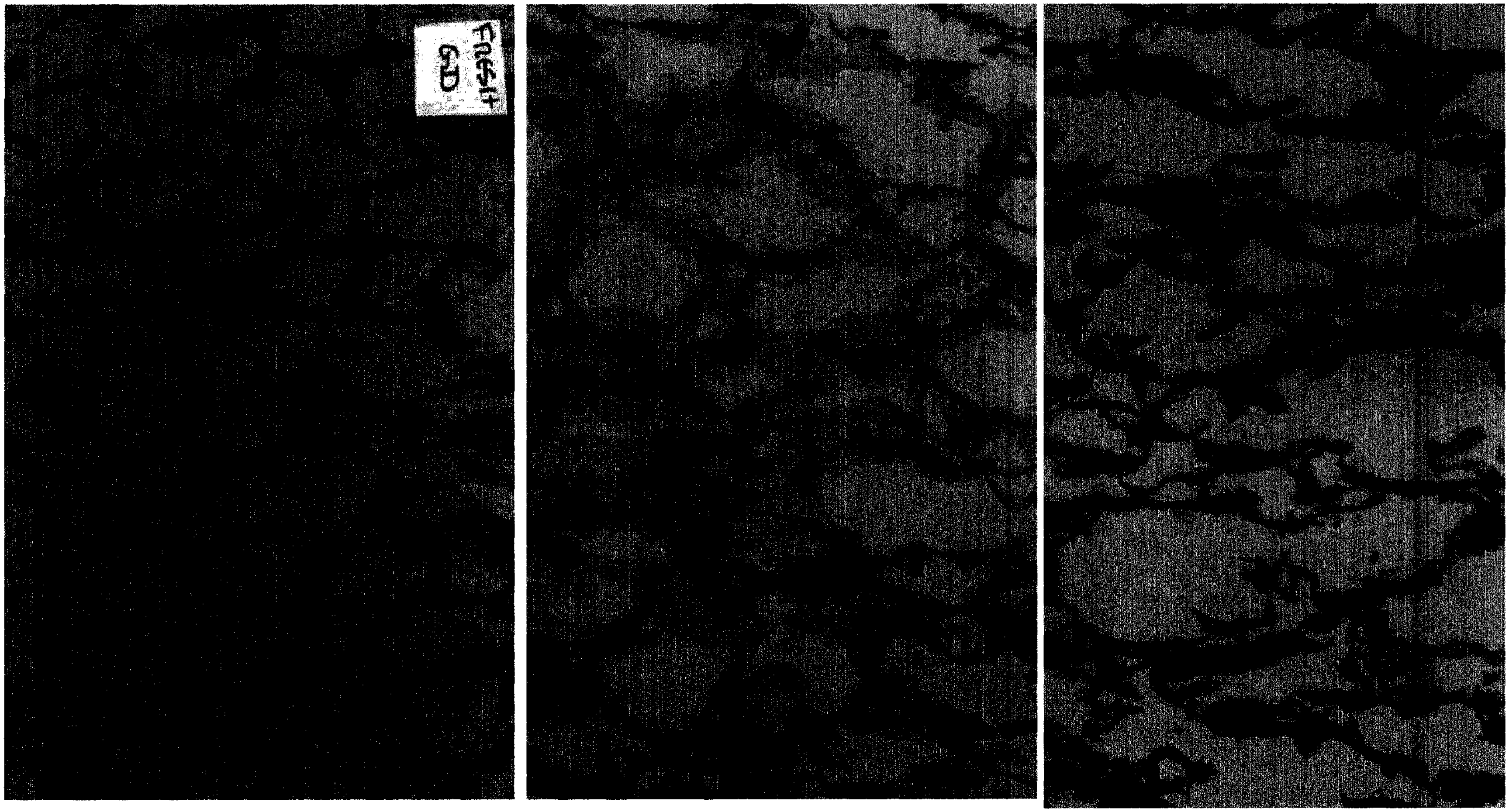

Figure 4.24 Histological comparison of apple, cultivar Golden Delicious (a - fresh sample, b - sample dried at 65°C and c - dried in HPD assisted system).

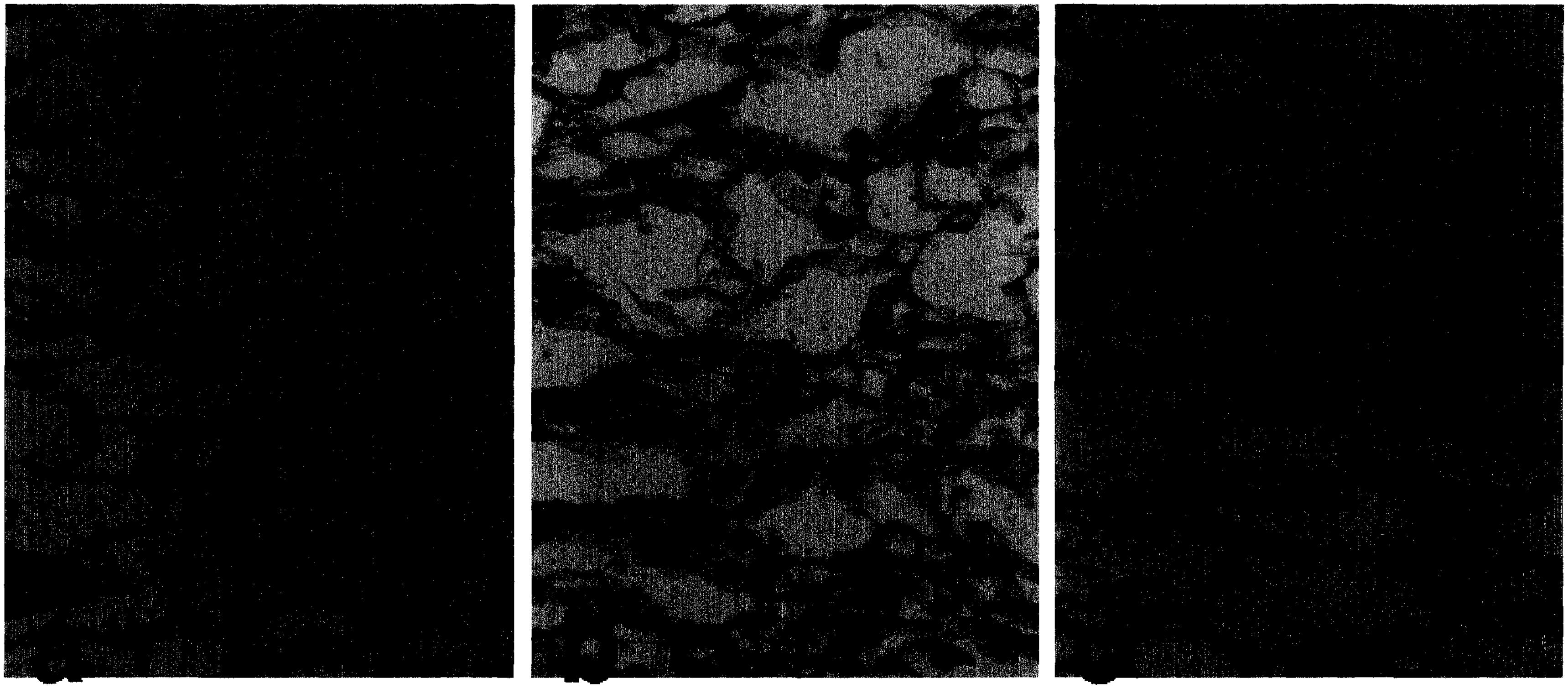

Figure 4.25 Histological comparison of apple, cultivar Gala (a - fresh sample, b - sample dried at 65°C and c - dried in HPD assisted system).

CHAPTER 5
DISCUSSION

A review of the exhaustive body of literature related to drying uncovers some of the fundamental problems that face researchers, producers and consumers associated with the field. Drying of homogeneous material that will not undergo significant structural and chemical changes is well understood and can be described quite satisfactorily by mathematical models. However, the drying of biological material such as agricultural products has proven to be a far more complex phenomenon. The drying of agricultural products has provided several challenges to research and development, as primarily motivated by the need to reduce the energy inputs required to stabilize such products within the time and financial constraints of marketing and distribution. The evaluation of the success of new or modified drying procedures for biomaterials must not only deal with the efficiency of the process in its broad sense (viz. energy, time and cost), but also with a multidimensional factor, quality of the final product.

The criteria for illustration of quality are primarily organoleptic, although nutritional characteristics and salubrity are becoming more of an issue when the products are destined for consumption in the "industrialized" world. While organoleptic characteristics on their own do not avail themselves easily to standardization due to the subjective preferences of consumers, the added factors of safety and nutrition may complicate matters since their rates of change during the drying process are not necessarily the same or even in the same direction as those of the organoleptic characteristics (flavour, texture, colour etc).

In case of the efficiency of a dryer or drying process, there is no globally-accepted rational definition. Researchers generally compare numerical indicators such as specific moisture extraction ratio (SMER) and specific energy consumption (SEC), which are related to tangible economic performance. However, these cannot be considered comprehensive because they ignore the exergy costs of the energy sources on which they are based, as well as the impact of the process waste streams on the surroundings, both of which are quite different depending on the type of energy input used in a given process. Simultaneously, the concept of quality fogs the issue as the end products may or may not be similar or comparable.

In this context, the main objective of this research project, which was to conduct a cross-disciplinary evaluation of a prototype drying system based on a commercial heat pump dehumidifier (HPD), becomes a challenging affair, as will be described in the following sections.

5.1 Operation Of The HPD Assisted Drying System

It is important to outline the original purpose of the HPD in order to understand its limitations as the main element of a dryer. The unit was designed to stabilize the humidity and temperature of the air in an enclosed space (such as an indoor swimming pool). The unit was made compact by placing the two heat exchangers inside the unit, the evaporator and the condenser, right next to one another. A secondary condenser is fundamental to the successful operation of a HPD (to remove the constant heat input from the compressor work) and in this case, a water-cooled condenser outside the system serves the purpose. In application to indoor swimming pools, the heat removed during cooling in the secondary condenser is recycled to the water in the pool, thereby contributing towards maintaining its temperature. In its original design and in practice, the dehumidification capacity of the unit matches the rate of evaporation from the surface of the pool. Hence, the amount of heat transferred to the air, determined by the capacity of the primary condenser, is limited.

The rationale behind adopting this particular unit for drying is economical in nature. Recent literature on drying with heat pumps describe the theory, construction and operation of HPD assisted systems (Chou and Chua, 2001; Strommen et al., 2001). The systems described are designed with the sole purpose of drying, and involve careful matching of the system components to the drying process with extensive controls and feedback mechanisms for continuous, safe and optimal operation. However, a system that has multi-purpose nature is more attractive economically since its capital cost (and in some modes of operation, some of the operating cost) would be shared between the drying operation and the other application(s). A versatile unit that could be used for drying and climate control in agricultural structures should have a better potential. In order to maintain its versatility, the changes that are done to the unit should be either easily reversible or kept at a minimum to suit the different needs. Accordingly, the Dectron MAM 024 was modified minimally to maintain its original status and integrate with the drying system.

In the unit, complete rejection of heat in the secondary condenser (Figure 3.6a) achieves only dehumidification of the process air. In a well-insulated system, this is unsuitable for drying because the air is completely saturated at the end of the dehumidification process (state D in Figure 4.15). However, the temperature of the air leaving the evaporator is low and it warms up passing over the metallic parts and the ducts. Trials were run in this mode to observe the drying at very low temperatures (0.5 - 4°C). Apart from the very low drying rates achieved, the starving of the evaporator leads to severe frosting on the surface, causing fluctuations in the temperature of the air in the system. The repeated thawing and refreezing of the evaporator surface caused unpredictable changes in drying conditions. It was therefore decided that this mode was unsuitable for drying operations.

Alves-Filho and Strommen (1996a,b) and Alves-Filho et al. (1998a) describe a drying technique with a heat pump assisted dryer that carries out partial drying at temperatures below the freezing point of fruits such as apple and strawberry. The fruit slices are frozen (in a freezer outside the dryer), are then introduced into the drying chamber. The heat pump is operated so as to achieve a drying temperature of –25°C, at which much of the moisture is removed from the product. The operating temperature is then raised and drying completed. Removal of moisture at sub-zero temperatures is said to maintain the open porous structure of the product and consequently enhances the rehydration characteristics. However, the papers do not discuss the practical problems of frosting or the amount of time taken for the initial drying step. In fact, none of the papers related to HPD assisted drying discuss the problem of frosting. Removal of water by sublimation, as in freeze-drying, is known to result in products with lesser collapse, better retention of colour and flavour as well as good rehydration characteristics (Ratti, 2001). Microbial spoilage is also prevented even though the drying periods tend to be long. However, maintaining constant low temperature conditions using the heat pump dehumidifier in a continuous, closed loop operation is extremely difficult, if not impossible. The long drying periods also automatically increase the energy consumption and cost of the process. On the other hand, the problem of frosting might not be a serious problem in an open or single pass mode under conditions of high humidity. During controlled-environment tests done with the MAM 024 HPD (prior to incorporation into the drying system), the frosting of the evaporator was minimum and the condensed water flowed freely out of the drip pan when operated under room conditions of 27°C and 98% RH.

The system operated satisfactorily in the closed loop, reheat mode where the air was recirculated continuously and the primary condenser was used for heat rejection (Figure 3.6b). In this mode, the heat recovered at the evaporator arrives at the primary condenser where it is transferred to the recirculating air stream. The maximum amount of heat rejected at the condenser is limited by the design; however, the amount of heat rejected is also dependent on the initial state of air arriving at the evaporator (which dictates the state of air that passes over the condenser). The additional heat rejection (including subcooling of the refrigerant) takes place in the secondary condenser.

It is clear that safe and continuous operation of an HPD involves rejecting a component of heat from the system. In a perfectly matched system at steady state, this component is equal to the quantity of heat supplied in the form of compressor operation. But in this study, the unit is limited in the heat rejection at the primary condenser and the maximum temperature at the discharge is kept below 30°C. The excess heat is lost in the secondary condenser. The recovery and use of this heat in the drying air involves a design that makes use of multiple condensers (Strommen and Jonassen, 1996; Ting, 1987).

One of the possible designs that would be compatible with the objective of maintaining the integrity of the unit would be the use of multiple condensers that substitute for the secondary condenser, with bypass control. An additional heat exchanger in the air path with control valves to selectively distribute the refrigerant flow through it could be considered (Figure 5.1). However, higher temperature in the system presents a problem at the evaporator during the later stages of drying.

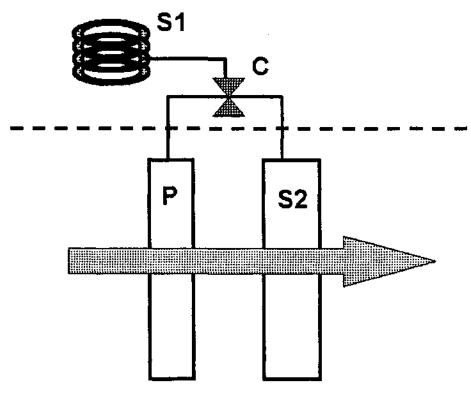

Figure 5.1 Suggested modification of secondary condenser arrangement. P - primary condenser, S1 - water-cooled secondary condenser, S2 - air cooled secondary condenser and C - three-way variable control valve for divergent refrigerant flow.

Even though higher temperatures promote faster drying, the decreasing humidity of the air leaving the drying chamber leads to reduction of the latent heat recovery at the evaporator. In the dedicated systems, this problem is dealt with the use of variable speed compressors that control the suction pressure (Strommen and Jonassen, 1996) or venting off part of the air or partial recirculation (Soponronnarit et al., 1998; Prasertsan et al., 1996a). While these solutions are not compatible with the current purpose, a slight increase in the temperature lift is beneficial to the drying process. An attractive mode of operation is coupling space dehumidification with the drying process (Figures 4.5 - 4.7). Ambient air could be used for drying after removal of moisture and addition of heat. The process is feasible, but the drying rate is dictated by the ambient conditions. An improvement over the process is to use dehumidification along with additional heating equipment. In such a design, the HPD would supplement the dryer to achieve quicker drying (with dehumidified, warm air) as well as enable drying even at conditions of high ambient humidity.

The condensate at the evaporator poses a serious problem in estimation of the energy transfer profiles at the evaporator. Engineering calculations for cooling and partial condensation on a fin-tube heat exchanger assume that a condensate film is formed on the surface, with the water draining away from the heat exchanger (Martin, 1992). The rate of condensation of water is too low and will not lead to a flowing film, especially on the unclean, uneven surface of the heat exchanger. The resident moisture is frozen quickly leading to frosting of the evaporator. This influences the heat transfer characteristics of the evaporator and makes it difficult to perform the energy calculations. The enthalpy of the condensed water cannot be measured or estimated correctly in that situation. The evaporator pressure and the hot gas bypass valve (DGRE-4S4 in Figure 3.5) setting were selected based on preliminary trials. At pressures lower than 3.3 bar (48 psi), the frost formation on the evaporator surface was severe and higher pressures were not useful for dehumidification as the dew point of the air leaving the drying chamber was often very low.

The temperature profile shown in Figure 4.2 indicates that the temperature lift on the discharge side of the HPD occurs after an initial lag. It was assumed that the effect was due to location of the unit (in a large, uninsulated but heated hall, where the ambient temperature averaged 18-20°C) and uninsulated parts of the duct and other metal parts that were present in the air path. The temperature increases after a steady state is achieved in the system and reaches the maximum halfway through the experiment. Trials were conducted to study the variation in the drying process if the drying load was introduced into the system after steady state had been achieved. The system was run until the air temperature reached 24-25°C and the trays carrying apple rings were placed in the drying chamber to continue the drying process. The actual drying time was reduced by 2-3 hours; however, the machine had to be run at least 5 hours to reach steady state and hence the total drying time and power consumption increased overall. Complete insulation of the system, including the drying chamber (which was constructed of transparent polycarbonate sheets to facilitate visual examination), could lead to faster establishment of steady conditions, ultimately leading to reduction in drying time and energy consumption.

5.2 Drying Characteristics

Prior to integrating the HPD with the drying system, the unit was tested in a facility where the ambient conditions could be varied with a steam generator. Under conditions of high humidity, the unit demonstrated a capability to condense 13.5 kg water per hour at the evaporator, recover a maximum of 13.4 kJ s^{-1} (air in at 27°C, 98% RH) and reject 3.75 kJ s^{-1} at the primary condenser. It is not realistic to expect such conditions during batch drying of biomaterials. The maximum heat recovery at the evaporator occurs during the initial stages of drying where moisture loss from the material occurs at a higher rate. Heat recovery then drops progressively as the drying rate decreases. The amount of water that could be condensed reveal the potential of the machine and the load size of the batches that could possibly be handled using the unit. The predicted performance indicators of the system based on the observed experimental results (Table 4.2) support that inference.

The experimental setup was a quasi-prototype drying system in which the capacity of the HPD outmatched the maximum sample size that would fit in the drying chamber. This mismatch of the two components was unavoidable due to the experimental nature of the study. The weight of the material being dried had to be monitored in order to obtain the drying curve. It is a regular practice in drying experiments to remove the sample periodically and replace it after quick measurements. However, it is neither advisable nor practical to do this when running a closed loop operation with high volume loads. On the other hand, very large mass samples are difficult to weigh online. A compromise was arrived at and a drying chamber equipped with a real-time weighing system was designed to handle a maximum of 5 kg sample. The experimental samples weighed between 2.5 and 4.5 kg.

The most noticeable conclusion from the results is that the problem lies in the amount of moisture released from the material during drying. The highest drying rates observed are during the first few hours and that period amounts to almost 50% of the moisture removal. The tests with tomato and agar gels demonstrated that the system was efficient in removing moisture from the surface of the product. Comparisons with the hot air drying trials demonstrated that temperature is the main factor for acceleration of the process. Hence, it is the limitation to the temperature lift in the HPD system that leads to longer drying periods. The iterative model based on the regression models developed from the experimental data indicates the possibility of drying larger loads (Table 4.2) with enhanced SMER (specific moisture extraction rate) values. Any improvement in the performance beyond these values cannot be achieved without harnessing the heat rejection at the secondary condenser.

5.3 Losses and Irreversibilities

One of the design problems was related to the air path and the duct system. Minimum distances between the heat exchanger/fan discharge and duct transitions were necessary for optimum performance of the HPD unit. Maximum airflow over the evaporator/condenser HE surfaces was assured by extending the duct lengths past the bends and transitions. Due to this, the entire system measured over 20 feet in length with the ducts accounting for 60% of it. The ducts connecting the HPD discharge and the drying chamber, which also contained the heater elements, were well insulated, but the return duct was not.

The loss of energy in the duct system was calculated based on the information obtained from ASHRAE (1997). The lack of insulation was important only for the closed loop reheat system of operation, and in these trials, the loss amounted to 457 kJ per hour (0.127 kW). Insulation of the lower duct would reduce the losses to about 252 kJ per hour (0.07 kW). The system was checked meticulously for leaks and sealed; hence it was assumed to be airtight and leakage losses can be neglected. The most important and largest loss occurs in the secondary condenser. Even though the hot water stream leaving the coil has potential to be put to use in a shared application, it is considered a loss for the drying process. Almost 40-50% of the heat equivalent of the total input to the system is discharged at the secondary condenser.

The energy losses due to lack of insulation or leakage fall under the category of first law losses. These losses are usually avoidable or could be minimized with suitable improvements to the equipment (Thompson et al., 1981). A more critical category is that of second law losses, those that result due to process irreversibilities (Carrington and Baines, 1988). These cannot be avoided without undertaking major process improvements. Irreversibilities in the process lead to loss as well as destruction of available work (exergy).

The throttling of the high-pressure liquid refrigerant, pressure drops in the refrigerant line and heat exchangers and compression are inherent sources of exergy loss/destruction in the HPD. Due to lack of instrumentation on the refrigeration side, accurate measurements of the refrigerant flow values could not be made in this study. However, the equipment manufacturer and parts' suppliers (Sporlan Valve Co., Washington, MO, Copeland Corp., Sidney, OH and Alco Controls, Emerson Electric Co., St. Louis, MO) provided information based on which, it was possible to estimate the pressure drop across different components. Depending on the operating conditions, a refrigerant flow rate of 108.86 kg per hour (240 lbph) was suggested and was used for calculation of exergy values for the R-22 refrigerant. The refrigerant properties (Kamei and Beyerlein, 1992) were considered in the I-P system for convenience (as most of the manufacturer's data was also for the I-P system) and exergy differences were calculated based on -40°F dead state (Please see Appendix 4 for exergy calculations). The compression process destroys about 31% of the exergy supplied to the HPD. The throttling process causes an irreversible loss of 15-16%. Due to the compact nature of the unit, the exergy destruction due to pressure drops is comparatively lower; 3.42-3.6% in the line between the compressor and the liquid receiver and 1.1-1.15% in the low pressure line. The process of bypassing hot gas to the low-pressure line to maintain the superheat results in exergy destruction. However, the magnitude of this loss could not be estimated, as the quantity of bypassed gas was not measurable in the setup.

5.4 Comparison Of Dryer Performance

This thesis promotes the philosophy that the measurement of efficiency for drying of biological material is interwoven with the quality of the product. Hence the discussion needs to consider the two aspects together.

5.4.1 Quality Of The Product

The HPD dried material in this study showed interesting results regarding quality. Even at 10% moisture content (wet basis), the samples appeared fresh, pliable and succulent, as opposed to the darkened, hard, crispy nature of the samples dried at higher temperatures. This radical difference in the physical appearance prompted the microscopic studies. The microphotographs clearly demonstrate that the damage to the cellular structure of the apple is least in the HPD assisted drying compared to the hot air treatments. Similarly, the development of undesirable colour was also minimum in this treatment and the rehydration characteristics were better than the other drying methods. Organoleptic characteristics were also rated high for the product.

In apple, the fruit flesh is composed of irregularly shaped parenchyma cells, vascular tissue and, interstitial spaces which constitute 20-27% of the fruit volume (Reeve, 1953; Lapsley, 1992). Almost 90% of the water in the fruit is held within the osmotically active central vacuoles of the parenchyma cells. The vacuoles are bound by the tonoplast, and the cells by plasmalemma, which are both selectively permeable membranes. The structural stability to the flesh is imparted by the rigid cell walls, which also limit the water intake of the cells due to osmosis. Thus, drying involves water movement across two membrane structures against a gradient (due to osmotic potential of the cytoplasm) and through the interstitial spaces/vascular tissue to the surface. The situation is simplified in hot air drying where the membranes and most of the cell wall complexes are destroyed by heat, releasing the water into the matrix (Lewicki and Drzewucka-Bujak, 1998). However, the assumption that the food material is a porous saturated medium or a bundle of capillaries is not valid for drying at mild temperatures.

NMR studies of changes in subcellular water compartmentation in apple parenchyma during drying in mild air conditions (22°C) showed that the water loss occurred mostly from the vacuolar compartment and was associated with overall shrinkage of the cells (Hills and Remigereau, 1997; Hills and Le Floc'h, 1994). It was reported that the dried tissue appeared and felt fresh, suggesting that membrane integrity had been maintained and the cell walls had not collapsed. The authors also reported that although freeze drying removes more water, including that from the cell wall and the cytoplasm, it usually results in membrane damage, cell wall collapse and loss of turgor. A similar opinion is expressed by Yano et al. (1981) who promote the idea of reducing the water content of fresh leafy vegetables (with subsequent rehydration) for transportation and storage. The authors found that the extent of reversible moisture reduction depended on the mode of drying and milder conditions were more favourable (for almost 50% reduction in total weight).

The HPD used in this study is operated to maintain the temperature of the air below 30°C with low relative humidity. Most of the drying takes place during the first 8-10 hours at temperatures less than 25°C. It can be safely assumed that the cellular integrity of the apple parenchyma is maintained during the drying process. The membranes are expected to collapse eventually due to the stresses developed upon removal of water and shrinkage. However, the collapse is due to mechanical reasons and is not accompanied by adverse changes as in a hot air drying process. The values for rehydration were higher for the HPD dried material than those obtained with other treatments. The rehydrated product was very succulent and had a crispy texture, which could be attributed to the return of cell walls to their original condition. The significant increase in the water activity of the HPD dried product during storage is also an indirect indication of the reversibility of the drying process. The polythene bags used for storage were not evacuated and were also permeable to water vapour, thus rehydrating the product during storage.

The drying conditions also influence the appearance of the product. The browning developed during low temperature drying process is mostly enzymatic. The oxidation of dihydroxypolyphenols by polyphenoloxidase is responsible for such discoloration (Mathew and Parpia, 1971). In an intact fruit, the enzymes and the substrates are kept separated in the fruit cells without the occurrence of chemical changes. As drying progresses, the removal of water and its reduced availability further retard the browning reactions.

It is clearly established that higher temperatures during drying lead to destruction of nutritional quality. Ascorbic acid, for instance, is a characteristic reductone that enters into the non-enzymatic Maillard browning reactions that occur at higher temperatures. The irreversible hydrolysis of dehydroascorbic acid into biologically inactive compounds is also facilitated by heat. Drying at low temperature (25-30°C) has been shown to ensure retention of ascorbic acid (Chua, 2000).

5.4.2 Measurement of Dryer Performance

A standard procedure to determine dryer efficiency has not yet been established due to the differences in drying processes (Strumillo et al., 1995). Dryer construction is a specialized business based on experience and trials, due to the complexity of the processes involved. A universal definition of the concept of work and efficiency in the field of drying is hard to envisage. One of the reasons for this perplexity is the quality factor. Mere removal of water cannot be defined as the intended work, as different drying regimes achieve this objective with varying final product quality. The retention or development of desired quality factors such as colour, structure, porosity, rehydration characteristics, nutritional value etc. complicate the definition of the final product.

Dryer performance is usually evaluated by indices based primarily on energy consumption. The HPD assisted dryer performance is usually associated with the coefficient of performance (COP), the specific moisture extraction rate (SMER) and the specific energy consumption (SEC). The coefficient of performance of the heat pump is defined by the ratio of heat rejected at the condenser to the work input to the compressor. The COP is a useful indicator in applications such as space heating, where heat is the expected output, but has little significance in drying. The SMER (mass of water removed per unit energy input) could be applied to the HPD alone (water removed from air viz., dehumidification performance) or to the whole HPD assisted system (water removed from the drying material viz., drying process). The SEC indicates the total energy required for the removal of a unit mass of water and can be assessed for either the HPD or the drying system. The latter two indices are more commonly used in heat pump dryers and are convenient when comparing similar end products. In light of the previous discussion on quality, it could be stated that the end product of the drying process in the HPD assisted system is different from that of a higher temperature process. The use of SMER or SEC to compare the two products may not be very appropriate in such situation.

Pakowski and Mujumdar (1995) list different performance factors, some even termed efficiency, to evaluate dryers. One of them is called energy efficiency and is given as the fraction of the energy consumed for evaporation of water. In its simplest form, the energy required for evaporation of water is obtained by multiplying the total water content and the latent heat of evaporation of water. Cenkowski et al. (1992), Palani Muthu and Chattopadhyay (1993) and Lengyel et al. (1998) propose a more pragmatic approach to calculating the latent heat of material based on the sorption isotherm data; Kiranoudis et al. (1993) have compiled the heat of desorption data for some common vegetables. However, the use of such efficiency terms is probably valid only when higher temperatures are used and in products where the moisture is "bound" due to adsorption.

Developments in the field of plant science have intensified the focus on chemical potential gradient as the basis for mass transfer (Gekas, 2001). The removal of water from plant tissue without thermal disruption of the structure requires reduction of the water potential of the environment and the apoplast below that of the protoplast. But the state of knowledge in the area of water transport within the tissue is not sufficient to satisfactorily explain the kinetics of drying and account for the energy requirement. The water potential for cells is determined by several components that originate from the effects of solute, pressure, porous matrices, gravity etc. Solutes and matrices with wettable surfaces lower the chemical potential of water, whereas pressure and gravity increase or decrease it depending on their values.

In plant tissue, water moves readily into and out of cells according to the water potential differences between the protoplast and the apoplast compartments. In an intact fruit, the water is in dynamic equilibrium due to the balancing influences of the solute in the protoplast and the rigidity (pressure) of the cell wall. But there exists a strong tendency for water to move into the cell as proposed by Ray (1960) and other supporting studies (Robbins and Mauro, 1960; Mauro, 1965). Negative pressures are created in the pores of the plasmalemma that are transmitted to other parts of the tissue in the apoplast. Pure water is held under tension in the pores and water movement into the cell is depicted to be primarily by pressure-driven bulk flow. The estimation of these forces would be the first step in establishing the minimum energy requirements for drying.

One approach that would probably provide a uniform basis for comparison between different drying methods would be that of exergy analysis. A framework for such an analysis is shown in the Figure 5.2. The broken line represents the control volume over which the exergy balance could be calculated. The sources of irreversible losses within the HPD unit have been discussed earlier. The exergy rejected by the refrigerant in the secondary condenser should be considered as exergy associated with heat flow and hence a loss with respect to the drying process. Similarly, the exergy of condensed water is a loss in the material flow out of the system.

But the critical issue in this case is the exergy of the material being dried. Biological tissues, despite representing a significant part of the global economy, have not received enough attention regarding their exergy values. It is the practice of some specialists to use energy values (calorific value), based on the quantity of heat liberated at oxidation, as exergy values. However, this evaluation of highly organized biological structures and compounds as simple fuels is not considered appropriate (Brodyansky, 1998). The concept of Maxwell's demon and Information Theory (Leff and Rex, 1990; Loewenstein, 1999) might as well be extended to drying to achieve a sense of purpose and establish the fundamental basis for comparison. The nutritive and/or pharmaceutical value of food, colour and flavour, represented by the biologically active molecules could be used to establish the exergy values.

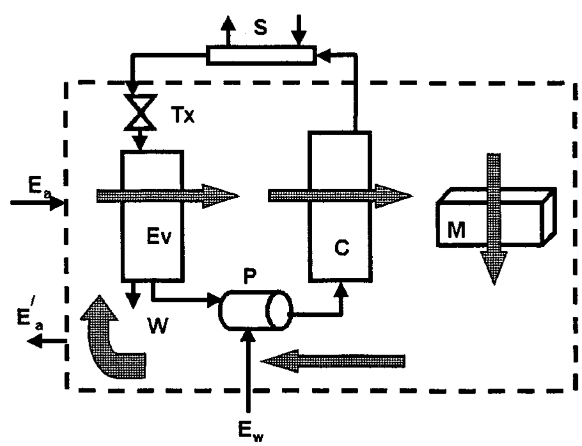

Figure 5.2 Components of the HPD assisted drying system for exergy analysis. E_a - Exergy of the intake air, E'_a - Exergy of the exhaust air at the end of the process, Ew - exergy of work (power consumption of the HPD), W - Exergy of the water condensed, M - Material exergy, C - Condenser, Ev - Evaporator, Tx - Thermostatic expansion valve, P - Compressor, S - Secondary Condenser.

The quest for quality in terms of texture and rehydration draws attention to the structural quality of the food material. Loss of these characteristics represents an increase in the entropy of the food material and their retention on the other hand, extracts a price in terms of additional exergy consumption during processing. Arriving at comprehensive reference standards that include all aspects of the material (physical and chemical) is a very complex task to achieve. A system of assigning exergy reference values to the major chemical components would be a good start. The calculation of the substance's chemical exergy involves two components - choice of a reference substance and the calculation of the chemical exergy relative to the chosen reference substance and the environment in which the reference substance exists (Brodyansky et al., 1994). The enormity of this task requires a mix of careful deliberation, inter-disciplinary interaction and database management.

5.4.3 Drying With The HPD Assisted System – A Deliberation

Storage, and to a lesser extent, transportation, are the main reasons for drying of food material. Removal of water reduces the bulk and the weight of the material, and could be intended for transportation purposes depending on the cost-benefit ratio. Preservation, enhanced shelf life and value addition are the more common motivating factors for industrial drying of biological material. The type of drying method is undoubtedly influenced by the purpose and the final product characteristics. The reduction of water content to prevent microbial spoilage is aimed at decreasing the availability of water or the water activity. The trend over the last decade has been shifting towards minimally processed foods with intermediate-moisture content (with a_w ranging between 0.65 and 0.95).

The water activity of apple measured at 25°C for hot air drying and the HPD assisted process (Figures 4.16 – 4.18) provides circumstantial evidence regarding the possible distribution of water in the product upon dehydration. For the same water content, lower water activity values were measured for the HPD dried material than the hot air dried samples (Figure 4.18). Due to the lack of sufficient number of data points and the practical difficulty in controlling the variables (the moisture content of dried apple cannot be controlled accurately enough to have similar values for the two treatments), the statistical significance of the differences could not be determined. However, linear regression equations for the two sets of data indicate a lower intercept and a steeper slope for the hot air dried samples. It might be argued that the difference in the thermodynamic state of water could be due to the physical structure of the products. The apple tissue is expected to retain the integrity of its cell walls much better in the HPD dried material, which in turn exert tension on the water in the system, thus reducing the water activity. More work is necessary to confirm this phenomenon, but it offers a promising possibility that the HPD assisted drying could be used for production of stable products without exposure to thermal damage. However, storage results indicate that the product dried with the HPD assisted system is very unstable in ambient environments (or packaging permeable to moisture) and need special, airtight packaging.

The advantages of drying at low temperatures using a heat pump dehumidifier are difficult to situate in a competitive market. In a narrow analysis, a rapid drying process at higher temperatures would be preferred by most consumers. The benefits of using low-grade heat are not transparent enough to appreciate for consumers who are not sympathetic to the deteriorating environmental conditions. However, the use of the HPD in an interconnected network of applications would reduce the share of capital cost for the drying segment. Simultaneous operation of the unit for drying, heating water (via the secondary condenser) and dehumidification of structures would split the operating cost over three heads. The reduced dependence on fossil fuels as a result has benefits much greater than the immediate economic gain.

5.5 A Role for Heat Pumps on Farm

The use of heat pumps for drying had its beginnings during the second half of 1900s as an accessory. By the turn of the century however, they had been positioned differently. The heat pump is now being projected as an integral part of the dryer. A great deal of research including an enormous number of trials has gone into setting up control and feedback mechanisms for these systems. However, the potential for heat pumps is much greater to be used in a restricted application. It was demonstrated in this work that the heat pump dehumidifier could be used for drying of agri-food products and could be used simultaneously for space dehumidification and water heating (Figures 4.5 – 4.7).

This thesis suggests a different role for the heat pumps on a farm, where it acts as a link between various energy streams to utilize the low-grade heat rejected to the atmosphere. The primary advantage of a heat pump is its ability to make low-grade heat available for useful operations. It is more beneficial in the long run to identify the sources of such waste heat (in form of exhaust gases, humid air, warm liquid drains etc.) and attempt to connect them using heat pumps with energy consuming operations (for example, Figure 5.3). Having conceived such a design, the focus of the future research would then attempt to tackle the practical issues such as the odour removal, filters, flow control and modulation, switchovers between operations etc. The link-up of such proportions would provide for a comprehensive energy budgeting on the farm and a more optimum utilization of the energy input.

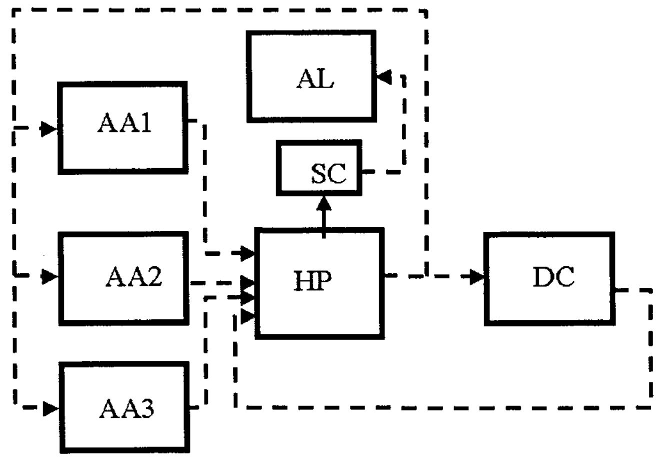

Figure 5.3 Interconnecting applications with a heat pump. HP - heat pump, DC - drying application, SC - secondary water cooled condensers, AL - application involving a liquid stream e.g., a hot water tank, AA1 - AA3 - applications involving air as a waste heat source or sink (e.g., barn, farm house and a sugar shack).

5.6 Relevance Of The Research And Contribution To Knowledge

The study successfully completed the design and fabrication of an HPD assisted drying system that could be used for drying of biological material. A multi-purpose role for the HPD was proposed and its role in drying has been described. The operation of the unit revealed the strengths and drawbacks of the system with respect to the product quality, energy consumption and the operating parameters. Based on this experience, a few conclusions could be drawn that would assign a role of such multi-purpose HPD systems in agri-food processing.

It can be safely stated that the system is unsuitable for use as a stand-alone application i.e. exclusively for drying. The utilization of the power input is not economically favourable for a single application. The rectification of this drawback would change the versatile nature of the HPD unit. But the system could be integrated with a parallel application without major modifications for better energy economy. For instance, the unit could be used to reduce the humidity in a sugar shack (maple syrup processing in open pans) and transfer the low-grade heat to a hot water system or used to pre-heat the fuel. As demonstrated under controlled humid conditions, almost 50 MJ per hour (COP 5.75) can be rejected to the water stream. At a slightly higher capital cost, providing the water-cooled secondary condenser with an option to switch with an air-cooled one and using two different air streams, the system could be very effectively used as a preheater for high temperature grain drying under humid conditions.

As a low temperature drying process, it was demonstrated that products of superior quality could be produced with this system. Good retention of colour, flavour, appearance and structural components was observed in a dried product that also exhibited lower water activity at the same moisture content than did the product dried in hot air.

The experience gained during the study leads to contemplation of the exercise in relation to the state of knowledge in the field of drying. The study proves the feasibility of successfully drying biological material in an environment of reduced vapour pressure and mild temperature conditions. It also underlines the importance of product quality as a parameter in arriving at an agreement for defining work or efficiency in the field of drying. The need for a broad-based definition of the actual objective in drying is fundamental to comparisons between different techniques and treatments. The need for intense research into the interactions of water with biological tissue is also essential for a better description of the drying process. Finally, the thesis promotes the opinion that future energy related applied research should strongly consider the linking of compatible processes to achieve energy and exergy efficiency.

CHAPTER 6
SUMMARY AND CONCLUSIONS

Finite energy resources on earth need judicious management to be utilized for sustenance of human civilization. To advance the cause of humankind in harmony with rest of the biosphere calls for a responsible course of action. As a major consumer of energy in agriculture, the unit operation of drying has enormous scope for improvement; however, the changes and adjustments required are often not transparent enough to reveal the benefits directly to the consumers. It is the responsibility of the technology sector to coax the end-user to be more discerning in the use of energy resources. Easy availability, replication and economic profitability are the features that need to be addressed to achieve that goal.

This thesis is based on the conviction that heat pumps can act as links in interconnected chains of energy dependent farm applications. The main objective in this research project was to study the cross-disciplinary application of a commercial heat pump dehumidifier (HPD) for drying of agri-food products. A compact domestic unit used for building climate control was utilized with minimal modifications and its performance was evaluated. The study was an investigative exercise where ultimately more questions were formed than the answers obtained by probing the system. It also throws light on issues that need to be addressed for defining a common ground on which different drying processes can be compared.

It was demonstrated that a heat pump dehumidifier (HPD) could be successfully used for drying of agri-food material and simultaneously used in parallel applications such as space dehumidification and water heating. The feasibility of operating under different modes (single pass or open mode and recirculation mode) was established; the process could easily be adapted to different environmental conditions. By identifying the sources of waste heat and the energy dependent processes on a farm, heat pump linked systems could be envisaged that enable optimum use of the high-quality energy input.

Products dried with the HPD assisted system were of high organoleptic quality. Undesirable colour changes were minimal, and rehydration and flavour retention superior compared to conventionally dried products. A difference in the nature of water retention within the product was also observed. Microscopic examination revealed a less disturbed physical structure compared to the control.

The experimental work with drying of fruits and vegetables, and the subsequent analysis of the product characteristics, point towards the diversity of final product output in terms of quality. The structural and perhaps, chemical changes that accompany the dehydration, imply differing moisture transfer patterns under various drying regimes. Higher temperatures hasten drying process by breaking down the structural barriers to mass transfer such as cell walls and cell membranes, leading to irreversible chemical and physical changes. The heat pump dehumidifier assisted drying process proceeds at a slower pace due to the lower temperatures prevailing in the system, but results in products of higher quality with much lesser damage. Considering the absence of an efficiency expression in drying of biological material, this observation assumes significance. Affixing standard exergy values to biological components could provide a basis for arriving at a universally accepted exergy-based evaluation protocol. That would enable the professionals to assess the performance of different systems, having chosen the final quality of the product. This would probably enable drying to become more of a science, than the art that it is now.

6.1 Contributions To Knowledge

The specific contributions of the thesis are –

- Creation of a niche for heat pumps in agricultural environment and demonstration of an instance as applied to drying of agri-food material.
- Proposal of an exergy based efficiency analysis approach in drying.
- A call for the adoption of a universal quality-based energy expense evaluation in drying of biological material.

- Demonstration of low temperature drying of agri-food material for high quality products and estimation of the energy consumption.

6.2 Suggestions for Related Research Work in Future

The following are some of the recommendations for research that would follow-up on the work described in this thesis.

- Experimental work under different environmental conditions or simulated conditions of temperature and relative humidity – Practical application of the idea of linking energy streams need a viable protocol for control and modulation of the network components. Extensive experimental work is necessary to generate the data and predict the behaviour of the components towards different environmental conditions. Such work is essential before proto-types could be set up and studied with real-life situations.
- Energy budgeting on farm – A comprehensive survey of energy related activities on farms to record the input and output of all forms of energy in such activities is necessary for providing the framework on which the energy network could be established. It will help identify the different energy streams that could be linked with heat pumps, as well as selection and location of the equipment.
- Strategies to address the issue of biomaterial exergy standards – A comprehensive review of the subject is necessary to determine the approach that would lead to creation of a database of universal standard exergy values.
- Economic analysis – Detailed energy budgeting should be followed up by the economic analysis that is based both on money and exergy.
- Physico-chemical behaviour of biological structures and their importance to drying – Significant amount of research work has been carried out on the qualitative description of biological tissue. In

addition to qualitative features, drying requires evaluation of quantitative characteristics such as the water binding properties, surface area, temperature related changes, etc. The importance of such information for description of low-temperature drying kinetics cannot be overstated.

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Appendix 1
Cooling, dehumidification and heating of air (ASHRAE, 1997)

Figure A1 shows a schematic setup for cooling and dehumidification of air. It is assumed that condensed water is cooled to the final air temperature before it leaves the coil.

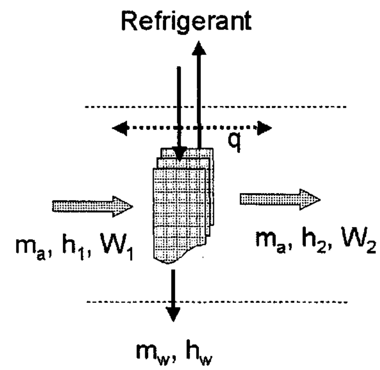

Figure A1. Schematic for heating/cooling of air

$$m_a h_1 = m_a h_2 + q + m_w h_w \qquad \text{Equation 1}$$

$$m_a W_1 = m_a W_2 + m_w \qquad \text{Equation 2}$$

$$m_w = m_a (W_1 - W_2) \qquad \text{Equation 3}$$

$$q = m_a [(h_1 - h_2) - (W_1 - W_2) h_w] \qquad \text{Equation 4}$$

If we assume the heat exchanger in Figure A1 to be a heater instead, it will lead to heating of the air flowing across.

$$q = m_a (h_2 - h_1) \qquad \text{Equation 5}$$

Where,
m_a = mass flow rate of air (kg s^{-1} or kg min^{-1} or kg h^{-1})

m_w = mass of water condensed (kg)

$h_{1,2}$ = specific enthalpy of moist air (J or kJ per kg dry air)

h_w = enthalpy of water (J or kJ per kg)

W = humidity ratio of moist air (mass of water per unit mass of dry air, $kgkg^{-1}$ or gkg^{-1})

q = quantity of heat, J or kJ (negative for evaporator).

To calculate the quantity of refrigerant that corresponds to the quantity of heat absorbed at the evaporator, the following equation is used –

$$q = m_r (h_{lat} + c_p \Delta T) \qquad \text{Equation 6}$$

m_r = Mass flow rate of refrigerant, kg

h_{lat} = latent heat of evaporation of the refrigerant at the suction pressure, Jkg^{-1} or $kJkg^{-1}$

c_p = specific heat of refrigerant vapour at the suction pressure and evaporation temperature, $Jkg^{-1}{}^{\circ}C^{-1}$ or $kJkg^{-1}{}^{\circ}C^{-1}$

ΔT = superheat, $^{\circ}C$

Appendix 2
Specific heat of apple

The specific heat of biological material could be estimated by the following relation (Choi and Okos, 1983).

$$C_p = (4.18 \times W) + (1.711 \times P) + (1.928 \times F) + (1.547 \times C) + (0.908 \times A) \qquad \text{Equation 7}$$

Where W, P, F, C and A are fractional percentages of the water, protein, fat, carbohydrates and ash contents. The representative values for the constituents of apple (*Malus sylvestris*) was obtained from USDA (2001) and proportional adjustments were made with reference to the water content which was determined in the experimental samples.

Appendix 3
Colour Measurement

The L*a*b* colour system is one of the uniform colour spaces recommended by CIE in 1976 as a way of more closely representing perceived colour and colour difference (Kuehni, 1997). It is meant to closely represent human sensitivity to colour. L* is the lightness variable; a* and b* are chromaticity coordinates. They are defined as,

$$L^* = 116\left(\frac{Y}{Y_0}\right)^{1/3} - 16 \qquad \text{Equation 8}$$

$$a^* = 500\left[\left(\frac{X}{X_0}\right)^{1/3} - \left(\frac{Y}{Y_0}\right)^{1/3}\right] \qquad \text{Equation 9}$$

$$b^* = 200\left[\left(\frac{Y}{Y_0}\right)^{1/3} - \left(\frac{Z}{Z_0}\right)^{1/3}\right] \qquad \text{Equation 10}$$

X, Y and Z are the tristimulus values based on the colour-matching functions of the CIE 2° Standard Observer. X_0, Y_0 and Z_0 are tristimulus values of the illuminant. For Standard Illuminant C, the values are 98.072, 100 and 118.225 respectively, and for Standard Illuminant D_{65}, the values are 95.045, 100 and 108.892 respectively.

The above relations hold good when X/X_0, Y/Y_0 and Z/Z_0 are greater than 0.0088526. When these ratios are lesser, the following substitutions are used,

$$\left(\frac{X}{X_0}\right)^{1/3} \text{ is replaced by } 7.787\left(\frac{X}{X_0}\right) + \frac{16}{116} \qquad \text{Equation 11}$$

$$\left(\frac{Y}{Y_0}\right)^{1/3} \text{ is replaced by } 7.787\left(\frac{Y}{Y_0}\right) + \frac{16}{116} \qquad \text{Equation 12}$$

$$\left(\frac{Z}{Z_0}\right)^{1/3} \text{ is replaced by } 7.787\left(\frac{Z}{Z_0}\right) + \frac{16}{116} \qquad \text{Equation 13}$$

The colour difference values ΔL^*, Δa^* and Δb^* are calculated as follows

$$\Delta L^* = L^* - L^*_t \qquad \text{Equation 14}$$

$$\Delta a^* = a^* - a^*_t \qquad \text{Equation 15}$$

$$\Delta b^* = b^* - b^*_t \qquad \text{Equation 16}$$

L*, a* and b* are measured values of the specimen. The subscript t indicates the values measured for the target colour point.

The total colour difference ΔE^* between two colour co-ordinates is calculated as

$$\Delta E^* = \sqrt{(\Delta L^*)^2 + (\Delta a^*)^2 + (\Delta b^*)^2} \qquad \text{Equation 17}$$

Appendix 4
Exergy calculations

Exergy of matter flow can be calculated from the relation (Brodyansky et al., 1994)

$$E = h - h_0 - T_0(s - s_0) = h - T_0 s + c \qquad \text{Equation 18}$$

where h, s, h_0 and s_0 are specific enthalpy and entropies of a flow in the states defined by p, T and by p_0, T_0 respectively. The pressure factor is necessary only for calculating the absolute values of E. It has no influence in calculations connected with the differences in exergy.

The magnitude of exergy is usually calculated from expressions that contain only thermal parameters of a state.

$$dh = c_p dT - [T(\frac{\partial v}{\partial T})_p - v]dp \qquad \text{Equation 19}$$

$$ds = \frac{c_v}{T}(\frac{\partial T}{\partial p})_v dp + \frac{c_p}{T}(\frac{\partial T}{\partial v})_p dv \qquad \text{Equation 20}$$

Also,

$$dE = dh - T_0 dS \qquad \text{Equation 21}$$

and hence,

$$dE = (\frac{T - T_0}{T})[c_p(\frac{\partial T}{\partial v})_p + c_v(\frac{\partial T}{\partial p})_v dp] + vdp \qquad \text{Equation 22}$$

substituting for c_v,

$$dE = (\frac{T - T_0}{T})[c_p dT - T(\frac{\partial v}{\partial T})_p dp] + vdp \qquad \text{Equation 23}$$

In a constant pressure process, the last two terms are removed. The solutions are found if the equation of state and the dependence of c_p on T is known.

The simplified approach while calculating the exergy difference at constant pressure between two temperatures T_1 and T_2 and with reference to a dead state T_0 is also given by McCauley (1983).

$$\Delta E = \Delta H - T_0 \Delta S \qquad \text{Equation 24}$$

and

$$\Delta S = Mc_p \ln(\frac{T_1}{T_2}) \qquad \text{Equation 25}$$

S = entropy, kJ per kg per K
H = enthalpy, kJ per kg
T = temperature, K
c_p = specific heat at constant pressure, kJ per kg per K
M = Mass flow rate, kg per h

Properties of air were determined using the relations prescribed by Hyland and Wexler (1983).