DEHUMIDIFICATION OF GREENHOUSES IN COLD REGIONS

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PERMISSION TO USE

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ABSTRACT

Greenhouses in cold regions often exhibit conditions of high relative humidity (RH). Frequent high RH levels of over 80% result in poor yields and undesirable crop quality, as well as poor working conditions for employees. The objective of this study was to seek an effective and economical method(s) to control RH in greenhouses of cold regions. Three methods were tested and evaluated: finned tubing condensation, air-to-air heat exchangers, and domestic mechanical refrigeration dehumidifiers. Temperature based ventilation and RH based ventilation were used for comparisons.

Finned tubing condensation using chilled water was tested in an environmental chamber. The condensation rates were obtained and statistical models were developed for the selected finned copper tubing with aluminum fins under various room and water conditions. This method was proved highly energy intensive and costly, thus discarded in the field experiment. The field experiment was conducted in a Saskatchewan greenhouse. The air-to-air heat exchangers and domestic mechanical refrigeration dehumidifiers were tested against the control treatment which was the temperature based ventilation control, i.e. relying on infiltration in winter and ventilation using exhaust fans in the other seasons.

The field experimental results proved that dehumidification was needed most of the year with high RH occurring from April to November. The heat exchangers and dehumidifiers controlled RH very well in winter, early morning and night in other seasons, and saved heating cost; however, in mild and warm weather from about 9 am to noon the RH was high before the exhaust fans operated at full capacity. When the ambient air was humid during the warm season, the heat

exchangers were ineffective for RH control. The dehumidifiers controlled RH not as well as the heat exchangers mainly due to the low capacity. Both methods added extra heat to the greenhouse in the warm season, which was desirable in early morning and at night when ambient temperature was low and heating was still needed but undesirable during daytime when cooling was required in the greenhouses.

Comparing energy efficiency, the moisture removal index (MRI) of the dehumidifiers was around -0.629 kW-h/L (produced 0.629 kW-h energy per liter of water removed), the heat exchangers' MRI was 0.916 to 1.020 kW-h/L (consumed 0.916-1.020 kW-h energy to remove 1 L of water), while the RH-based ventilation required 1.099 to 1.373 kW-h/L, and the finned tubing condensation required more than 7.2 kW-h/L. If natural gas was the heat source, the dehumidifier method was the most economical with annual average energy cost of $0.018/L, approximately 60% and 50% of those of the heat exchangers and exhaust fans, respectively. If thermal coal was used as the heat source, the heat exchanger was the most economical with an annual average energy cost of $0.016/L, as compared to $0.019/L and $0.035/L for the dehumidifiers and exhaust fans, respectively.

The mechanical dehumidifier method is energy efficient and effective year-round, and its operating cost is low, thus is recommended for greenhouse dehumidification in cold regions. However, a complete economic analysis that includes the capital cost is needed to evaluate the economic feasibility of the various methods. The heat exchanger is also recommended which can supply CO_2 in winter but it is ineffective in humid weather. Proper sizing of the dehumidification requirement is the key to success.

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LIST OF ABBREVIATIONS

ANOVA	**AN**alysis **O**f **Va**riance
$CaCl_2$	**Ca**lcium **C**h**l**oride
CFD	**C**omputational **F**luid **D**ynamics
CO_2	**C**arbon Di**o**xide
CR	**C**ondensation **R**ate
HE	**H**eat **E**xchanger
HRR	**H**eat **R**ecovery **R**atio
HSD	**H**onestly **S**ignificance **D**ifference
L	**L**iter
MRI	**M**oisture **R**emoval **I**ndex
PRT	**P**latinum **R**esistance **T**emperature detector
RH	**R**elative **H**umidity
RTD	**R**esistance **T**emperature **D**evice
SPSS	**S**tatistical **P**ackage for **S**ocial **S**ciences

LIST OF SYMBOLS

A_g	Floor area of the greenhouse, m^2
C_p	Specific heat of the water, kJ kg^{-1} K^{-1}
C_{pe}	Specific heat of the exhaust air, kJ kg^{-1} K^{-1}
C_{ps}	Specific heat of the supply air, kJ kg^{-1} K^{-1}
CR	Condensation rate of the finned tubing, in g h^{-1}
E_D	Monthly average daytime direct solar radiation (1953-1995), W/m^2
E_N	Net solar insolation into the greenhouse, W
FR	Designed water flow rate for the condensation system, L s^{-1}
FR_a	Measured water flow rate, L s^{-1}
h_0	Enthalpy of the ambient air, kJ kg^{-1}
h_1	Enthalpy of the supply air from the heat exchanger entering the greenhouse, kg m^{-1}
h_2	Enthalpy of the greenhouse inside air, kJ kg^{-1}
h_{e1}	Enthalpy of the outside air, kJ kg^{-1}
h_{e2}	Enthalpy of the exhaust air, kJ kg^{-1}
h_{fg}	Water heat of vaporization, kJ kg^{-1}
M_c	Mass flow rate of the chilled water, kg s^{-1}
M_e	Mass flow rate for the exhaust air of the exhaust fan, kg s^{-1}
$M_{exhaust}$	Mass flow rate of the exhaust air, kg s^{-1}
M_p	Monthly average daytime moisture production rate inside the greenhouse, W
$MRI1$	Modified moisture removal index of the dehumidifier based on the electrical energy consumption, kW-h/L

$MRI2$	Modified moisture removal index of the dehumidifier based on the heat output of the dehumidifier and the latent heat released by the condensate , kW-h/L
MRI_c	Moisture removal index of the condensation system, kW-h/L
MRI_d	Moisture removal index of the dehumidifier, kW-h/L
MRI_e	Moisture removal index of the exhaust fan, kW-h/L
MRI_h	Moisture removal index of the heat exchanger, kW-h/L
M_{rn}	Moisture mass to be removed for day-to-night RH control, kg
M_{supply}	Mass flow rate of the supply air, kg s^{-1}
ΔMRI_c	Overall error for MRI of the finned tubing condensation system, kW-h/L
ΔMRI_d	Overall error for MRI of the dehumidifiers, kW-h/L
ΔMRI_e	Overall error for MRI of the exhaust fan, kW-h/L
ΔMRI_h	Overall error for MRI of the heat exchanger, kW-h/L
m_e	Mass of water removed by the exhaust fans, kg
m_h	Mass of water removed by the heat exchangers, kg
m_w	Mass of the condensed water from the finned tubing, kg
m_{water}	Mass of the condensed water collected by dehumidifiers during last two days of each test cycle, kg
m_{rd}	The required mass ventilation rate for daytime RH, control, kg/s
m_{rn}	The required mass ventilation rate for day-to-night RH, control, kg/s
Q_e	Rate of maximum amount of heat that could be recovered, kW
Q_L	The rate of latent heat gained for the greenhouse, W
Q_r	Rate of total heat recovered by the supply air stream, kW
q_c	Electrical energy consumption of the pump, kW-h
q_d	Electrical energy consumption of dehumidifiers, kW-h
q_e	Electrical energy consumption of the exhaust fans, kW-h

q_h	Electrical energy consumption of the heat exchangers, kW-h
q_l	Latent heat released by condensed water of dehumidifiers, kW-h
q_{lc}	Latent heat released by condensed water, kW-h
q_{le}	Net total heat loss through ventilation of the exhaust fans, kW-h
q_{lh}	Net total heat loss through ventilation of the heat exchangers, kW-h
q_{lossc}	Heat loss (from room) which is absorbed by the chilled water during the heat transfer process, kW-h
q_o	Heat output of the dehumidifiers, kW-h
RH	Designed relative humidity of the chamber, %
RH_0	Ambient relative humidity, %
RH_2	Greenhouse inside relative humidity, %
RH_{id}	Daytime air relative humidity setpoint inside greenhouse, %
RH_{in}	Nighttime air relative humidity setpoint inside greenhouse, %
RH_{od}	Monthly average daytime ambient air relative humidity (1953-1995), %
RH_{on}	Monthly average ambient air relative humidity at sunset (1953-1995), %
S_{1c}	Energy cost of the heat exchanger treatment if thermal coal was the heat source, \$/L
S_{2c}	Energy cost for the exhaust fan (RH based ventilation) if thermal coal was the heat source, \$/L
S_{3c}	Energy cost for the dehumidifier treatment if thermal coal was the heat source, \$/L
S_{4c}	Energy cost for the condensation system if thermal coal was the heat source, \$/L

S_{1g} Energy cost of the heat exchanger treatment if natural gas was the heat source, \$/L

S_{2g} Energy cost for the exhaust fan (RH based ventilation) if natural gas was the heat source, \$/L

S_{3g} Energy cost for the dehumidifier treatment if natural gas was the heat source, \$/L

S_{4g} Energy cost for the condensation system if natural gas was the heat source, \$/L

S_c Electrical energy cost of the condensation system, \$

S_d Electrical energy cost of the dehumidifiers, \$

S_e Electrical energy cost of the exhaust fan (RH based ventilation), \$

S_h Electrical energy cost of the heat exchanger, \$

S_{lc} Thermal coal cost for the latent heat of the condensation system, \$

S_{lcg} Natural gas cost for the latent heat of the condensation system, \$

S_{lec} Thermal coal cost resulting from the heat loss through ventilation of the exhaust fan, \$

S_{leg} Natural gas cost resulting from the heat loss through ventilation of the exhaust fan, \$

S_{lhc} Thermal coal cost resulting from the heat loss through ventilation of the heat exchanger, \$

S_{lhg} Natural gas cost resulting from the heat loss through ventilation of the heat exchanger, \$

S_{lossc} Thermal coal cost resulting from the heat loss of the condensation system, \$

S_{lossg} Natural gas cost resulting from the heat loss of the condensation system, \$

S_{oc}	Thermal coal cost for the heat gained of the dehumidifiers, $
S_{og}	Natural gas cost for the heat gained of the dehumidifiers, $
T_0	Temperature of the supply air entering heat exchanger, K
T_1	Temperature of the supply air from heat exchanger entering
T_2	Temperature of the exhaust air entering the heat exchanger, K
T_3	Temperature of the exhaust air leaving the heat exchanger, K
T_i	Designed air temperature of the chamber, ℃
T_{in}	Measured inlet water temperature of the pipe, ℃
T_{out}	Measured outlet water temperature of the pipe, ℃
T_w	Designed water temperature for the condensation system, ℃
T_{wa}	Measured reserve water temperature, ℃
ΔT	Temperature difference between the inlet and outlet water, K the greenhouse, K
t	Running time of the heat exchanger, h
t_0	Ambient temperature, ℃
t_1	Temperature of the supply air from the heat exchanger entering the greenhouse, ℃
t_2	Greenhouse inside temperature, ℃
t_3	Temperature of the exhaust air leaving the heat exchanger, ℃
t_c	Running time of the condensation system, h
t_e	Running time of the exhaust fans, h
t_{id}	Daytime air temperature setpoint inside greenhouse, ℃
t_{in}	Nighttime air temperature setpoint inside greenhouse, ℃
t_{od}	Monthly average daytime ambient air temperature (1953-1995), ℃
t_{on}	Monthly average ambient air temperature at sunset (1953-1995), ℃
t_{rn}	Required moisture removal time, s
t_w	Temperature of the condensed water, ℃

V_c	Volume of the water condensed by the finned tubing, L
V_d	Volume of the water removed by dehumidifiers, L
V_e	Net volume of water removed by the exhaust fans, L
V_g	Volume of the greenhouse, m^3
V_h	Net volume of water removed by the heat exchangers, L
V_{rd}	The required volumetric ventilation rate for daytime RH control, m^3/s
V_{rn}	The required volumetric ventilation rate for day-to-night RH control, m^3/s
W_0	Humidity ratio of the ambient air, kg kg^{-1}
W_1	Humidity ratio of the supply air from the heat exchanger entering the greenhouse, kg kg^{-1}
W_2	Humidity ratio of the greenhouse inside air, kg kg^{-1}
W_{id}	Daytime air humidity ratio inside greenhouse, kg/kg dry air
W_{in}	Nighttime air humidity ratio inside greenhouse, kg/kg dry air
W_{od}	Monthly average daytime ambient air humidity ratio (1953-1995), kg/kg dry air
W_{on}	Monthly average ambient air humidity ratio at sunset (1953-1995), kg/kg dry air
β_0	Constant of the linear model for the finned tubing
β_1, β_2, β_3 and β_4	Unknown coefficients of the linear model for the finned tubing
$\Delta\mu_{c1}$	Error for MRI of the finned tubing condensation system which was contributed by electrical energy consumption of the pump, kW-h/L
$\Delta\mu_{c2}$	Error for MRI of the finned tubing condensation system which was contributed by heat loss from room to the chilled water, kW-h/L

$\Delta\mu_{c3}$	Error for MRI of the finned tubing condensation system which was contributed by latent heat released by the condensate, kW-h/L
$\Delta\mu_{c4}$	Error for MRI of the finned tubing condensation system which was contributed by net volume of water removed by the finned tubing, kW-h/L
$\Delta\mu_{d1}$	Error for MRI of the dehumidifiers which was contributed by electrical energy consumption of the dehumidifiers, kW-h/L
$\Delta\mu_{d2}$	Error for MRI of the dehumidifiers which was contributed by heat output of the dehumidifiers, kW-h/L
$\Delta\mu_{d3}$	Error for MRI of the dehumidifiers which was contributed by latent heat released by the dehumidifiers, kW-h/L
$\Delta\mu_{d4}$	Error for MRI of the dehumidifiers which was contributed by net volume of water removed by the dehumidifiers, kW-h/L
$\Delta\mu_{e1}$	Error for MRI of the exhaust fan which was contributed by electrical energy consumption of the exhaust fan, kW-h/L
$\Delta\mu_{e2}$	Error for MRI of the exhaust fan which was contributed by net heat loss through ventilation of the exhaust fan, kW-h/L
$\Delta\mu_{e3}$	Error for MRI of the exhaust fan which was contributed by the volume of water removed by the exhaust fan, kW-h/L
$\Delta\mu_{h1}$	Error for MRI of the heat exchanger which was contributed by electrical energy consumption of the heat exchanger, kW-h/L
$\Delta\mu_{h2}$	Error for MRI of the heat exchanger which was contributed by net heat loss through ventilation of the heat exchanger, kW-h/L
$\Delta\mu_{h3}$	Error for MRI of the heat exchanger which was contributed by the volume of water removed by the heat exchanger, kW-h/L
ρ_0	Density of the ambient air, kg m^{-1}
ρ_1	Density of the supply air from the heat exchanger entering the greenhouse, kg m^{-1}

ρ_2	Density of the greenhouse inside air, kg m^{-1}
ρ_{id}	Daytime air density inside greenhouse, kg/kg dry air
ρ_{in}	Nighttime air density inside greenhouse, kg/kg dry air
ρ_{od}	Monthly average daytime ambient air density (1953-1995), kg/m^3
ρ_{on}	Monthly average daytime ambient air density at sunset (1953-1995), kg/m^3
ρ_{water}	Density of water, kg L^{-1}
τ	Solar transmissivity of the double-layer cover of greenhouse

Chapter 1. INTRODUCTION

Climate control is of great importance for greenhouse production in order to achieve high yield in good quality crops that meet the demands of consumers, as well as for economical production (Bakker et al., 1995; De Pascale and Maggio, 2005). Temperature and relative humidity (RH) are two basic climatic parameters usually controlled by heating and ventilation equipment. RH is a common used notation to express the humidity in horticultural practice. It is defined as the ratio of the actual partial water vapor pressure in an air-water mixture to the saturated water vapor pressure at the present air temperature (Albright, 1990). One common method to determine RH in commercial greenhouses is using a psychrometer. The RH can be determined with a psychrometric chart based on the measured dry bulb temperature and wet bulb temperature.

Generally, it is more difficult to control RH than temperature, for RH not only relies on air exchange from the infiltration and ventilation, but also is related to evaporation from growing media and transpiration of the plants. These last two factors in turn depend on various environmental conditions, including air temperature, air pressure, and solar radiation inside the greenhouse (Stanghellini, 1987; Campen, 2009). Relative humidity also fluctuates with changing amounts of condensation on the covering material, which depends on the cover's temperature and the RH level in the greenhouse (Campen and Bot, 2002). For most greenhouse plants, the optimum RH range is from 50% to 80% (Snyder, 2001). However, the RH can easily reach 90% to 100% in an enclosed greenhouse because of the watering and transpiration of the plants. Excessive RH within the greenhouse produces ideal conditions for deleterious fungus, resulting in leaf necrosis and some nutrient deficiency (Hand, 1988; Kranz, 1996) and leads to undesirable changes in crop growth and appearance. High RH may also hinder plant pollination (Bakker, 1991) and shorten the vase life of ornamental plants (De Gelder, 2000).

Modern greenhouses in cold regions tend to be better insulated and sealed with thermal screens and double-layer plastic films to reduce the heat losses. An improved greenhouse cover reduces the total energy consumption (Swinkels et al., 2001); on the other hand however, it brings about less air infiltration and less vapour condensation due to higher temperatures of the inner-surface, making additional dehumidification essential to remove the excessive moisture (Sebesta and Reiersen, 1981; Sonneveld, 1999). For well-insulated greenhouses in cold regions, the incoming vapour mainly produced by crop transpiration and water evaporation remains the same but the outgoing moisture resulted by condensation and infiltration is reduced compared to regular single layer greenhouses, therefore, dehumidification in the greenhouse of a cold region becomes more crucial.

Currently, the most commonly used method for dehumidification of greenhouses is ventilation to exhaust moist air and replace it with drier outside air. There are two major problems for this method. Firstly, it causes a corresponding heat loss during the cold season and thus increases the already high supplemental heating costs. Secondly, it is sometimes ineffective, especially during humid warm weather. Some greenhouse operators take specific measures to reduce moisture production in the greenhouse by improving the method of irrigation and culturing media, but their effectiveness is very limited because plant transpiration is a major source of moisture production. Therefore, it is necessary to develop an effective and economical alternative method for greenhouse dehumidification in cold regions.

Different kinds of dehumidification technologies have been developed and put into the industry practice worldwide, including forced ventilation using fans, air-to-air heat exchangers, condensation by chilled water, condensation by cold air, and chemical or mechanical refrigerant dehumidification methods. However, the above methods are either significantly ineffective or too costly with great capital and energy consumption costs for commercial greenhouse dehumidification in cold regions. Focusing on greenhouse dehumidification, Campen (2009) published his Ph.D. thesis (Wageningen, The Netherlands) in October 2009. He proposed that three potential methods, ventilation, condensation on a cold surface, and absorption using hygroscopic materials, can be applied for greenhouse dehumidification. By studying the three methods, Campen (2009) concluded that from economical, practical, and energetic points of view, using hygroscopic materials was impractical and unfeasible, whereas ventilation with heat recovery proved the most economical, practical, and energy-saving. Based on the suggestions from Campen (2009), ventilation with heat recovery and condensation on a cold surface were recommended as the greenhouse dehumidification options best suited to cold climatic conditions.

The current project was intended to apply a finned tubing condensation system using chilled water, air-to-air heat exchangers, and mechanical refrigeration dehumidifiers in a selected Saskatchewan greenhouse to study their dehumidification performances compared to those of the traditional control method via ventilation based on temperature control. Because there was a lack of design parameters for the finned tubing condensation system applied in a greenhouse, such as condensation rate (the mass of condensate per unit of time), a standard finned tubing condensation system using chilled water was designed and tested in an environmental chamber to obtain the condensation rates of the finned tube. The analyses on energy consumption and cost effectiveness were performed to evaluate its feasibility and economical efficiency for greenhouse dehumidification in cold regions.

Chapter 2. LITERATURE REVIEW

There are various methods available to reduce RH inside a greenhouse (Campen et al., 2003). So far, using natural and forced ventilation is a commonly accepted method of dehumidification in cold regions, but it still has the limitations as illustrated before. Although very few economical, effective, and low-maintenance methods have been accepted by greenhouse producers, a number of previous studies provide useful information on potential ways to reduce RH in greenhouses of northern latitudes. Research on six dehumidification methods is elaborated as follows.

2.1 Reducing Irrigation Evaporation

Reducing water evaporation by improving irrigation and cultural media can help reduce the RH. Using mulch, drip irrigation or filtration irrigation, or inter cultivation, can reduce evaporation from irrigation (Srivastava, 1994). However, these methods are ineffective for controlling moisture production from other sources such as plant transpiration; it is a main source of moisture production in greenhouses (Srivastava, 1994).

2.2 Ventilation Method

Natural ventilation accomplished by opening roof vents or side windows, or forced ventilation using mechanical fans, is a conventional dehumidification method for most greenhouses. Moist greenhouse air can be replaced by relatively dry outside air through ventilation. This is a common practice used to dehumidify greenhouse air and is usually used during summer or in tropical areas. For the long cold winter season at northern latitudes -- for example, in the Prairie Provinces of Canada -- the greenhouses are usually closed without ventilation to prevent heat loss, and CO_2 enrichment is sometimes provided for plant growth, making moisture inside of the greenhouse difficult to remove for RH control. During the heating period (spring, fall, winter, and even summer night when heating is needed in cold regions), an increase of energy consumption for ventilation with cold outside air exists in proportion to its dehumidification needs. Table 2.1 presents the annual transpiration of a tomato crop for two RH setpoints, the corresponding dehumidification requirement, and the energy consumption (Vermeulen, 2008).

Table 2.1 Annual transpiration of a tomato crop, dehumidification by ventilation during heating periods, and the energy consumption per square meter of greenhouse under Dutch climate grown under standard conditions (Vermeulen, 2008) as calculated by KASPRO (De Zwart, 1996)

Conditions	Transpiration $L\ m^{-2}\ y^{-1}$	Dehumidification $L\ m^{-2}\ y^{-1}$	Energy Consumption $MJ\ m^{-2}\ y^{-1}$
Maximum RH 80%	662	158	1459
Maximum RH 85%	640	102	1322

As shown in Table 2.1, the dehumidification requirement decreases by approximately 30% during heating periods and consequently leads to nearly 10% reduction in the energy consumption when the RH setpoint is set from 80% to 85%. Hence, ventilation is not energy friendly for greenhouse dehumidification in cold regions; there is too much heat loss. However, it could be an economically feasible way if it can contribute to energy saving with heat recovery.

De Halleux and Gauthier (1998) evaluated the energy consumption during the ventilation process for greenhouse dehumidification at northern latitudes by using the GX software system. The simulations were made for a tomato crop based on Quebec’s climatic conditions over a year. Results showed that dehumidification with proportional ventilation and on-off ventilation at a rate of one air exchange per hour represented an increase in energy consumption of 18.4% and 12.6%, respectively, compared to no ventilation.

In a more recent study to improve the conventional ventilation method, Campen et al. (2009) designed an air distribution system to mechanically control the ventilation with cold dry outside air for the greenhouse equipped with a closed thermal screen. The thermal screens are popular insulation materials as greenhouse covers because they can be opened during daytime to increase the solar radiation into greenhouses and be closed at night to prevent the heat loss. As proposed in his study, the greenhouse air was exchanged at a low level with the outside air. The outside air was injected with mechanical fans close to the greenhouse floor and distributed by plastic film ducts with holes, thereby forcing the humid air to leak through the cracks on the cover of the greenhouse. The performance of the system was evaluated by a dynamic simulation model and was proved efficient according to the field experiment in a commercial greenhouse in The Netherlands. The system is easy to control and particularly beneficial for the greenhouses using thermal screen cover, thereby avoiding the horizontal temperature difference that results from air exchange by slightly opening the thermal screen. Hence, the thermal screens can be closed for longer periods to produce an energy saving effect.

2.3 Maintaining a High Temperature

Increasing temperature and ventilation rates before closing the greenhouse in the evening is an effective way to lower air RH at night (Wang and Mao, 2005). Supplemental heating may be needed, or an alternative heat insulation method such as hanging up the thermal screens to maintain the temperature, and the mechanical ventilation is applied to reduce the humidity ratio in the air. The RH is likely to be increased during nighttime because inside air temperature is kept lower at night but air humidity ratio is held the same as it is in daytime, so keeping temperatures high at night could maintain the required RH without reduction in the humidity ratio of the air.

Since air temperature should be maintained at optimum level in greenhouses, the application of this method is effective only for reducing the RH before closing down the greenhouse in the evening. It is not useful for normal daytime or nighttime temperature and RH management.

2.4 Chemical Dehumidification

Chemical dehumidification involves using hygroscopic materials to remove water vapour from the air to lower the RH. This method is applicable based on the vapour pressure difference between the absorbing surface and the greenhouse air, and it needs reconditioning equipment to remove the absorbed water (Pritchard and Currie, 1993). The hygroscopic media pumped between the absorbing surface and the reconditioning unit are usually salt-rich solutions (e.g. calcium chloride solution, or a mixture of calcium chloride and lithium chloride, seawater or bromides, etc.) which are costly and may cause serious environmental problems if chemical leakage occurs (Campen and Bot, 2001). The latent heat released as the greenhouse water vapour is absorbed by the hygroscopic system will go back to the greenhouse and assist in heating the greenhouse air. Therefore, hygroscopic dehumidification is preferably used to save heat energy due to the heat released by absorption. However, it is not favourable in greenhouses because of complicated installation and dangerous chemical applications (Campen and Bot, 2003).

Zhang and Zhao (2003) developed a dehumidification and cooling system using chemicals for greenhouses, as shown in Figure 2.1. It is mainly composed of a dehumidification room, a regenerated subsystem, a wet pad, and exhaust fans. When hot and humid outside air comes into a dehumidification room and exchanges heat with the calcium chloride solution sprayed from a sprayer head, the moisture in the air is reduced and then the dried air is evaporatively cooled as it is drawn into the greenhouse through a wet pad using the exhaust fans. In the dehumidification room, the calcium chloride solution is diluted gradually and flows into the solution stock pool, where it will be reutilized through the regenerated subsystem.

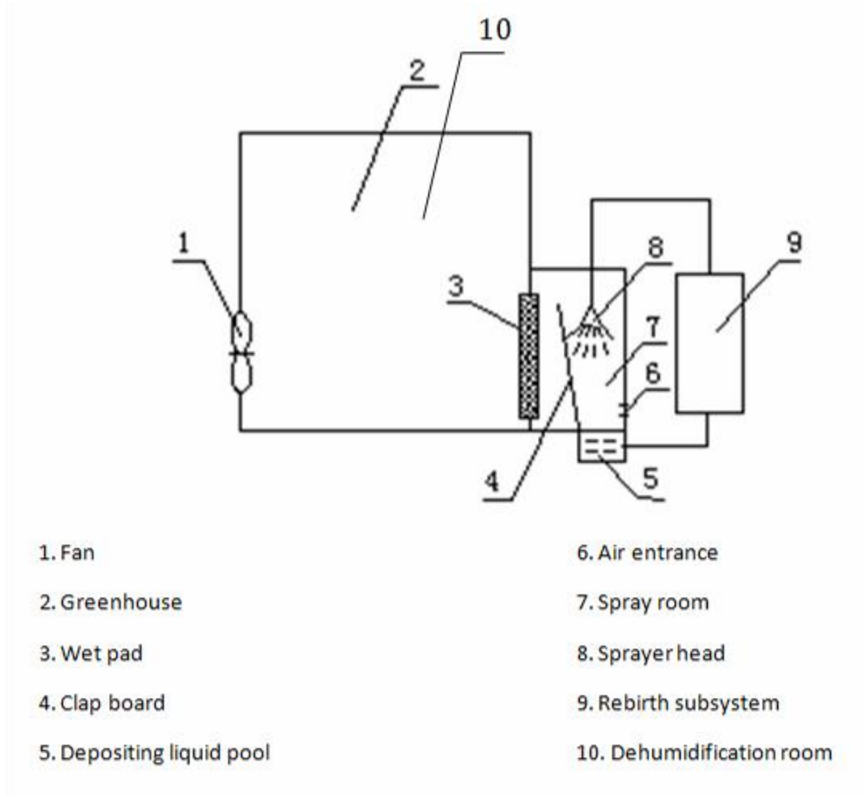

Figure 2.1 Chemical dehumidification system (Zhang and Zhao, 2003, used with permission)

Zhang and Zhao (2003) experimentally tested the above system using 45% calcium chloride solution as the hygroscopic material. When the air speed through the wet pad was around 1.2 m s^{-1}, the mass air flow rate through the dehumidification room was 1.5 to 3.0 kg m^{-2} s^{-1}, and the chemical sprayer mass flow rate was 0.457 kg s^{-1}, the RH of the air could be reduced from 80% to 50%, and the air was cooled by 10 ℃ after it was drawn into the greenhouse.

Zhang and Zhao's (2003) chemical dehumidification system was designed for dehumidifying hot and humid ambient air and concentrated primarily on the cooling purpose for greenhouses. Such weather conditions are uncommon even in summer time at northern latitudes, and the system is not suitable in the heating season. It can be used for greenhouses in cold regions by removing the wed pad of the system, however, the installation of such a system is too complicated and the use of chemical is not desirable in greenhoues.

An experiment using hydroscopic materials ($CaCl_2$) as the dehumidification method in a greenhouse was conducted by Lycoskoufis and Mavrogianopoulos (2008) at the University of Athens. Two cucumber greenhouses were selected: one was equipped with a dehumidification system using the RH setpoint of 80%, and the other was used as a control. The greenhouse air was forced by a fan to pass through the wet pad, which was soaked with $CaCl_2$ solution channelled by a small pump. It was found that the RH in the treatment greenhouse ranged from 81% to 88% during the experiment, and $CaCl_2$ was able to remove 52% of the water vapour on average to keep the RH at 80%. The dehumidification applied in the treatment greenhouse had a significant effect on the water condensate on the internal surface of the greenhouse screen; it was decreased from 90% of the total water losses to 17% compared with the control greenhouse. In this experiment, the $CaCl_2$ solution had to be reconditioned manually by adding the new solution continously, which increased captital and operation costs including the labor and the solution fee. The system was designed for non-heated greenhouse dehumidification in Greece. The two experimental greenhouses were not completely heated, and the temperature setpoint was set to be as low as 8 ℃. Therefore, this method was not applicable to greenhouse dehumidification in cold regions, which need heating most of the time in a year to keep the inside temperature around 20 ℃.

2.5 Chilled Water Condensation Dehumidification

Condensation occurs when water vapour meets an object that has a temperature below its dew point temperature. According to this principle, if one places a low temperature object in a greenhouse to make water vapour condense on it, then collects the condensate, the RH will decrease. Generally, the water vapour could condense on the covering materials inside the greenhouse naturally because of the cold inner-surface, or a designed cold object such as a finned tube cooled by chilled water or air could be set up in the greenhouse to induce condensation of the water vapour by natural convection.

Theoretically, there are four basic modes of condensation: dropwise, filmwise, direct contact, and homogeneous. Filmwise condensation is recognized as the most common mode that takes place in practice (Bell and Mueller, 2001). In filmwise condensation, the condensate wets the cold surface and produces a smooth film through which heat is transferred to continuously condensate the moisture from air. A temperature gradient occurs in the film, and the film actually represents a thermal resistance to heat transfer (Bell and Mueller, 2001). By applying a tube cooled by chilled water flow inside of the tube in an closed environment, the sensible heat transferred from the chilled water to the cold surface is primarily used for two parts: condensing heat load to change the air moisture from gas state to liquid state, and subcooling heat load to cool the condensate film on the surface before it drops under the action of gravity (Holman, 1997). An essential element to determine condensation rate theoretically is the coefficient for heat and mass transfer. The heat and mass transfer coefficients for standard geometries such as a plate or cylinder can be inferred from the literature (Becker, 1986). However, a finned tube is more complicated than a standard geometry and the coefficients for heat and mass transfer are more difficult to determine, which turns out to be a crucial part blocking condensation rate calculation for a finned tube theoretically (Campen, 2009). Therefore, it is necessary to determine the condensation rate experimentally.

2.5.1 Natural Condensation

The natural condensation inside a greenhouse will take a significant amount of water vapour out of the air. The rate of condensation depends on many factors, including inside air temperature and RH, interior surface temperature of the glazing material, and air convection within the greenhouse (Boulard, et al., 1989). The RH in single-glazed greenhouses is usually lower than that in double-glazed greenhouses because higher condensation rates are caused by the lower interior surface temperature of a single-layered glazing material (Reiersen and Sebesta, 1981).

However, the condensation on covering materials reduces the incoming light intensity (Reiersen and Sebesta, 1981). Condensation on plants may cause plant disease, so the issue is an important one; the dripping water can also affect workers' productivity.

2.5.2 Natural Convection Condensation

Campen and Bot (2001) designed a low-energy dehumidifying system for a greenhouse with a cold surface supported by a natural circulation system. The theoretical performance of the system was analyzed and improved by using computational fluid dynamics (CFD) software. A prototype based on the improved design was set up and tested. Its low-energy consumption was realized by natural air circulation driven by the temperature differences between inlet and cold surface, and between cold and hot surface. A heat recovery unit was located between the inlet air and the cooled air to recover sensible heat from the entering warm air to the cooled air. The experimental results showed that the prototype of 1 m length removed 50 and 65 ml h^{-1} when the cold plate temperature was 5.5 ℃ and the hot plate temperature was 50 and 65 ℃, respectively. The condensation calculations based on computational fluid dynamics resulted in a 25-30% higher moisture removal than that of the experiments.

In a later experiment, a natural convection condensation system was designed and tested in the greenhouse by Campen and Bot (2002). The system was mainly composed of finned steel pipes with chilled water; the pipes were installed under gutters in the greenhouse (Figure 2.2). As the air passed the system, the water vapour would condense on the cold surface of pipes cooled by forced flowing cold water, and the inside RH was lowered. The performance of the system was investigated using CFD in terms of its location and dimensions in the greenhouse, and then the system was tested in a two-span Venlo-type greenhouse with cucumber production. For this crop, 20.5 ℃ was set to be the day and night temperature controlled by the heating and ventilation system, and 80% was the RH setpoint controlled by the finned tubing dehumidifying system. The CFD simulation results showed that 1 m of finned pipe with a temperature of 5 ℃ could remove 54 g of water vapour per hour when the inside temperature was 20 ℃ and RH was 80%; this rate is equivalent to 1.3 kg per meter per day. With a total length of 100 m (four pipes with 25 m for each) and a total ground area of 6.4 m by 25 m, the experiment demonstrated that the system removed 40 g of water vapour per hour per square meter of greenhouse floor at the designed environment conditions. The capital and energy cost was not taken into account in the study; thus, there were no data on economic efficiency of the finned tubing system.

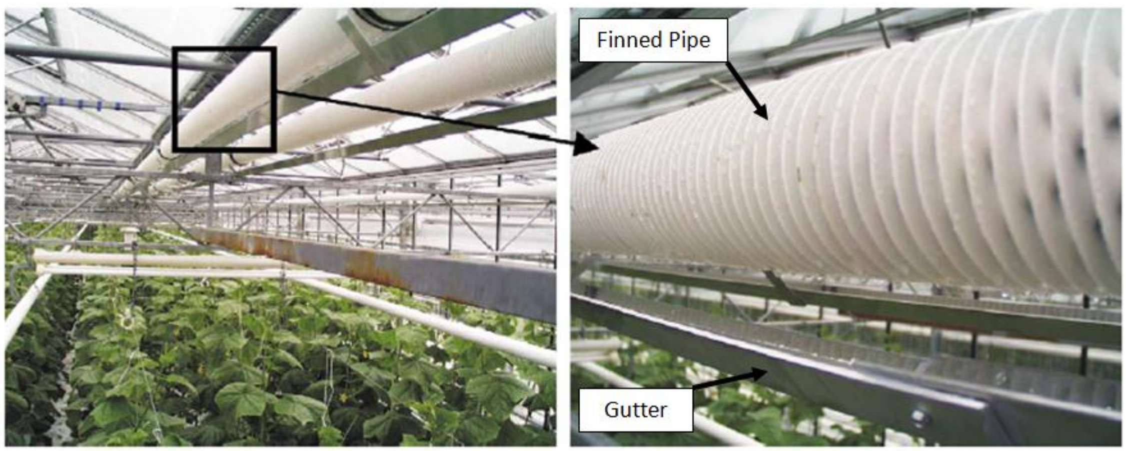

Figure 2.2 Condensation dehumidifying using cold water (Campen and Bot, 2002, used with permission)

2.5.3 Mechanical Refrigeration Condenser

At present, some producers choose to install heat pumps in greenhouses; this installation not only provides heating in the heating season, but also can be used as cooling and dehumidification equipment (with recycled chilled water or other cold source) in the summer time. A typical heat pump system used in a greenhouse is composed of two parts: an outdoor heat pump (contains compressor) and an indoor pipe system. The heat is pulled by the outdoor heat pump out of the air or groundwater and distributed by the indoor pipe system to heat the greenhouse in winter; the process can be reversed to cool the greenhouse in summer. According to their energy source differences, three kinds of heat pumps exist: air, geothermal and water sources (Pro Star, 1988). In summertime, when the chilled water passes through the pipes, the water vapour will condense on the surface; the condensed water will be collected and discharged out of the greenhouse, reducing the RH.

Chasseriaux (1987) attempted to dehumidify a rose production greenhouse with double-layer plastic films of approximately 3000 m^2 using a heat pump. He concluded that the unit could remove nearly 5 L of water per hour with 2.5 kW of electrical power, but could not significantly reduce the RH and improve the greenhouse environment conditions. A dynamic model of water vapour exchange was developed by researchers from France (Boulard et al., 1989) and compared with experimental results using a similar heat pump. It showed that the heat pump did not reduce the inside air humidity significantly, but eliminated the water condensation on the roof almost completely.

Lycoskoufis and Mavrogianopoulos (2008) carried out the experiment using a heat pump to dehumidify a greenhouse covered with polyethylene film at the University of Athens. Two cucumber greenhouses were used: one equipped with a dehumidification system and the other acting as a control. The heat pump was turned on when the RH was above 80% during the night. The results demonstrated that the heat pump was capable of keeping the RH at 80% with a power capacity of 8.78 W m^{-2}. The air temperature in the greenhouse with a heat pump was 2 ℃ higher on average than that in the control. In addition, a hybrid system using both hygroscopic materials ($CaCl_2$) and a heat pump was also evaluated for dehumidification in the same greenhouse. The results showed that the heat pump removed 53% of the total removed water vapour while $CaCl_2$ absorbed the rest. Moreover, the hybrid dehumidification process significantly reduced the amount of condensation on the inner surface of the cover as well as on leaf surfaces in the greenhouse, which was consistent with the conclusions drawn by Boulard et al. (1989).

Previous research on heat pumps applied to greenhouse dehumidification suggests that they were designed primarily for greenhouse heating and cooling purposes rather than for dehumidification; thus the dehumidification capacity levels are greatly insufficient for greenhouses, although there is elimination of water condensation on the covers.

Li (2002) designed a mechanical forced convection system for dehumidification in a solar greenhouse. Composed of a compressor, absorbing tube, fan, heating tube, and capillary tube, it is shown in Figure 2.3.

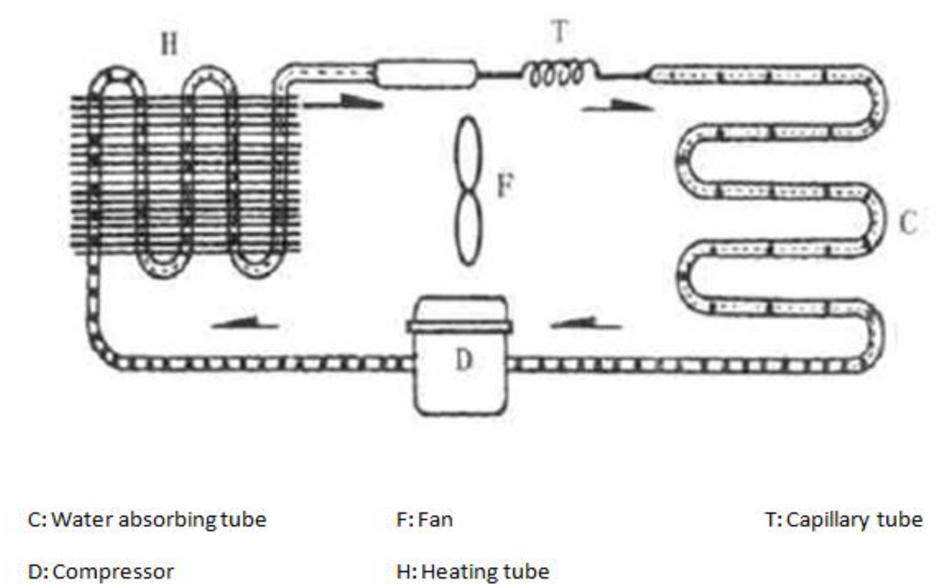

Figure 2.3 Forced convection dehumidifying device (Li, 2002, used with permission)

The system was based on the principle that water vapour in air could be condensed on the cold surface of the tubes due to the cryogen vaporization caused by the pumping function of the compressor. In this process, the heating tube was used to liquefy the cryogen with the function of the capillary tube, and the fan made the released heat spread throughout the greenhouse. As the cryogen was vaporized in the capillary tube while absorbing heat, the temperature of the tube was significantly reduced to below the air's dew point. The moisture of the air was then condensed on the tubes and removed from the air. The cryogen in its gas state returned to the compressor to restart the cycle.

The designed system was tested in a solar greenhouse of Luoyang City in China. It was found that this system was able to control the RH very well. The heat emitted by the whole dehumidification system was much greater than the absorbed heat contributed to the cryogen vaporization. Hence, it added the net heat released to the greenhouse during the dehumidification process, a feature beneficial for the heating season (Li, 2002).

2.6 Air-to-Air Heat Exchanger

This method is actually forced ventilation by using an air-to-air heat exchanger, which is preferred in the cold season. Mechanical ventilation is applied to exchange drier outside air provided by the supply fan with moist greenhouse air channelled by the exhaust fan, exchanging heat between the two air flows in the heat exchanger core. The advantage is that the incoming fresh air can recover some of the heat from the exhaust air by means of heat transfer when the outside temperature is lower than that of the inside air; therefore, it reduces supplemental heating requirements and cost while achieving dehumidification by ventilation.

Albright and Behler (1984) tested an air–liquid–air heat exchanger to control RH in a greenhouse. The results showed that approximately one-third of the enthalpy could be recovered from the exhaust air. De Hallaux and Gauthier (1998) used a simulation method to study this system and concluded that using heat exchangers could lower the energy consumption depending on the efficiency of the heat exchangers.

Heat recovery equipment based on the principle of sensible heat exchange was studied in a greenhouse in Canada by Rousse et al. (2000). They found the efficiency of the heat recovery unit, i.e. the ratio of the actual sensible heat recovered to the maximum heat that could be recovered by the unit, was around 80%, and the air flow rate of 0.9 air-change/h was not sufficient for greenhouse dehumidification. It was also concluded that the coefficient of the performance ranged from 1.4 to 4.8, which was defined as the electrical energy consumption of the fan divided by the recovered heat.

Speetjens (2001) studied a few small heat exchangers which were set up in the gutter of the greenhouse, and the results demonstrated that it was possible to recover 60 to 70% of the sensible heat.

Campen et al. (2003) investigated three dehumidification systems (condensation on cold surface, hygroscopic dehumidifier, and heat exchanger) using a dynamic simulation model under Dutch climates. By comparing these systems based on the energy consumption, capital, and operation cost, the study showed that using heat exchangers was the most promising approach for greenhouse dehumidification; the other two were less competitive, primarily due to the high investment cost.

2.7 Research Gaps

From the above literature review, one can see that very few dehumidification methods are commercially suitable for greenhouse producers in cold regions such as the Canadian Prairie Provinces. The systems have drawbacks such as high energy consumption, maintenance requirement, high capital and operation costs, or undue complexities of installation. Thus, an economical and effective dehumidification technology should be developed. The research gaps are elaborated as follows.

First, the condensation rates of finned tubing at different ambient environmental conditions, defined by mass of water vapour removed per hour, have never been systematically tested. Also, the theoretical equations are not able to predict the condensation rate without the unknown factors that must be drawn from the experiment; the design of such a dehumidification system in a greenhouse concerning the energy efficiency and cost effectiveness has not been undertaken.

Second, most studies found in the literature have focused on dehumidification efficiency in a certain period. However, almost no similar study has been conducted with systematic environmental condition measurements in different seasons of a cold region.

Third, very few studies have evaluated and compared different potential dehumidification methods in greenhouses of cold regions concerning the energy efficiency and cost effectiveness.

Chapter 3. OBJECTIVES

Based on the research gaps, the overall goal of the thesis was to evaluate three dehumidification methods as compared to the traditional temperature ventilation method and a hypothetical method of RH based ventilation; such an evaluation and comparison should provide one or more suitable solutions for greenhouse dehumidification in cold regions.

The first objective was to design and test a chilled water condenser prototype in a phytotron chamber, and measure its condensation rates as affected by four factors: room temperature, room RH, water temperature, and water flow rate. Also, statistical prediction models of condensation rate of the finned tubing as affected by the above four factors would be established. This method was proved the most energy intensive and most costly as compared to the other methods applied in the greenhouse. The total cost of a finned tubing condensation system designed for the greenhouse (including capital and installation costs, operation cost, and maintenance cost) was estimated to be greatly over the budget. Hence, this method was discarded in the field experiment.

The second objective was to experimentally quantify greenhouse dehumidification needs in the Canadian Prairie Provinces by monitoring the diurnal and seasonal environmental conditions inside a tomato production greenhouse in Saskatchewan. Such a dehumidification season determination will help to control the greenhouse environment, and thereby to prevent diseases and reduce pesticide use.

The third objective was to evaluate two dehumidification technologies: air-to-air heat exchangers and mechanical refrigeration dehumidifiers in a commercial greenhouse in Saskatchewan, measuring these technologies against the conventional temperature based ventilation control method. RH based ventilation (a hypothetical method using the data of the exhaust fans of heat exchangers applied) was used for compasions with these two technologies as to determine the dehumidification efficiency, energy efficiency and cost effectiveness based on three criteria: RH control accuracy, moisture removal index, and energy cost.

Chapter 4. MATERIALS AND METHODS

According to the objectives illustrated above, the dehumidification research was composed of two parts: laboratory test and field experiment. The materials and methods for each part are elaborated below.

4.1 Laboratory Experiment for Chilled Water Finned Tube Condensation

The laboratory experiment was conducted in an environment chamber at the University of Saskatchewan from January 12, 2009, to May 30, 2009.

The basic design parameter of the cold surface condensation system with finned tube is the condensation rate, defined by the weight of water condensed by the finned pipe per hour, which was very limited from literatures. Theoretical equations are not able to predict the condensation rates without some essential factors that need to be obtained from experiments (elaborated in Chapter 2.5). Thus it was decided that this rate should be determined experimentally by using a controlled environment chamber and a finned pipe with cold water flowing through it. Under different circumstances (i.e., different room temperatures, RH levels, water flow rates, and water temperatures) the condensation rates of the finned pipe could be measured, providing results to be used for the future field experiment design.

4.1.1 Description of Experimental Chamber

The experiment was conducted in an environment chamber, located in the phytotron area of the Agriculture and Bioresource Building, University of Saskatchewan. The inner chamber is 1.854 m long, 0.800 m wide, and 1.588 m high. The entrance door to the chamber is 0.864 m wide and 1.359 m high. Figure 4.1 shows the configuration of the experimental chamber and its environment control display panel. It is able to control both the temperature and RH inside the chamber. The accuracy of the temperature in the chamber is ±0.5 ℃, while the accuracy of RH is around ±3%. The phytotron chamber has no dehumidification device and is therefore dependent on ventilation to dehumidify its internal air. There is a piece of humidifying equipment to maintain the room RH during the condensation test process, and an air recirculation fan creates a homogeneous climate inside the chamber.

Figure 4.1 Picture of the environment chamber and its environment display panel

4.1.2 Experimental Instrument Set-up and Parameter Measurement

A 1-m copper tubing with a base diameter of 32 mm and with 108 mm by 108 mm aluminum fins (Model ASHDB, Trane Canada Corporation, Saskatoon, SK, Canada) was used in the experiment. This finned tubing is the most commonly used commercial products particularly for heating purposes. The finned tubing was placed on a wood frame inside the chamber at an angle of approximately 30^{o} to the horizontal (the inclined angle allowed the condensate to flow to the bucket from the gutter underneath the finned tubing, and it was assumed to have no impact on the condensation rate), and it was connected to the water inlet and outlet pipes through two perforated holes leading to the outside of the chamber. A submersion pump, connected to the water inlet pipe, was put into an insulated water bath with ice water to provide a circulated flow. The outlet pipe discharged water back to the water bath located outside the chamber. The flow rate through the finned pipe was controlled by a manual ball valve and monitored by a turbine flow meter (Hedland 1100, Division of Racine Federated Inc., Racine, WI, USA). The meter had the measurement range of 18.93 L/min to 189.3 L/min and the accuracy was ±1.0% of the reading. The pipes were all insulated to reduce heat transfer between the pipes and the air inside and outside the chamber.

The temperature of the water bath was controlled by adding ice to the water and was monitored by a RTD sensor (Resistance Temperature Device, Model HEL-775-B-T-0, Honeywell Inc., Morristown, NJ, USA). The sensors were 100-Ohm platinum RTDs with a temperature sensing range of -55 ℃ to 150 ℃. A three-wire half bridge was used to measure each 100-Ohm platinum RTD. The water temperatures at the inlet and outlet of the finned tubing were also monitored by RTDs. The amount of water condensed per hour was weighted by a digital scale (Cole-Parmer Symmetry PR 4200, Cole-Parmer Canada Inc., Montreal, QC, Canada) which had a capacity of 4200 g with 0.01 g readability. The water flow rate and water temperatures (including water bath temperature, water inlet temperature and outlet temperature) were all acquired and recorded by a data logger (CR10X, Campbell Scientific Inc., USA) and a personal computer.

The piping and instrumentation diagram (P & ID) illustrating the experiment set up is demonstrated in Figure 4.2. Table 4.1 gives the basic introductions to the main apparatus and materials used in the condensation system. Figure 4.3 and Figure 4.4 show pictures of equipment installations inside and outside of the chamber.

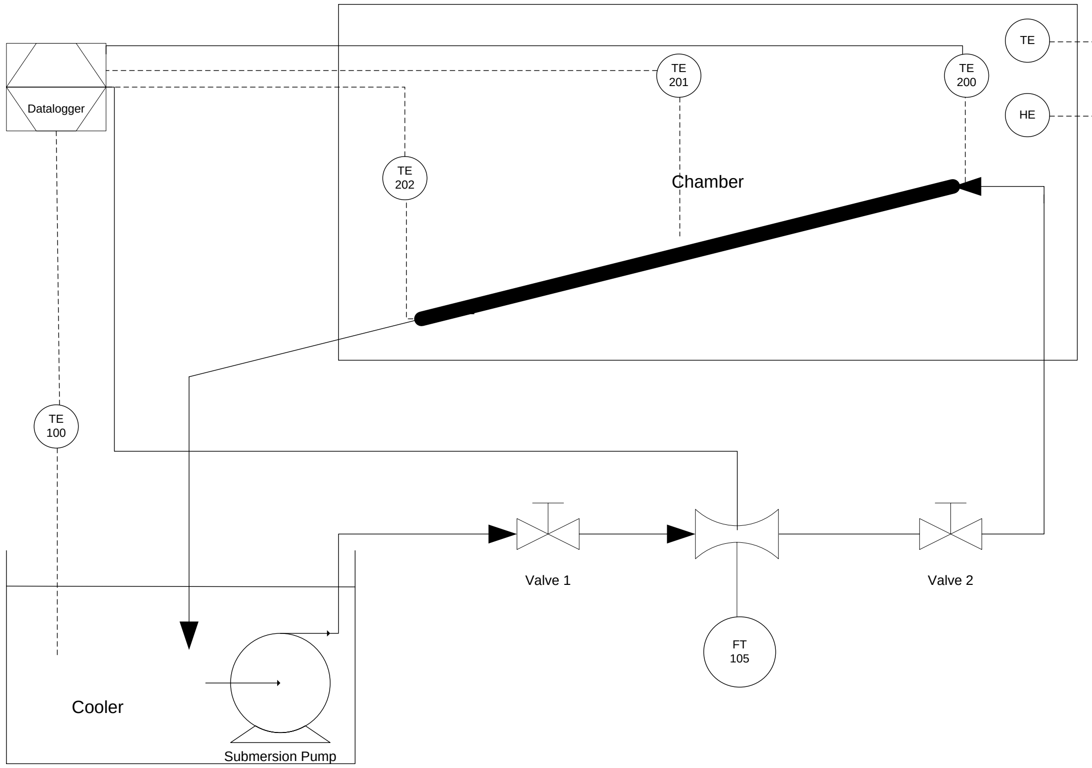

Figure 4.2 P & ID graph for chilled water condensation experiment

Table 4.1 Description of main apparatus and materials used for condensation system

Apparatus & Materials	Function
A phytotron chamber	To provide an enclosed controlled environment
A finned tube	To be used as a cold surface to condense the vapour
A submersion pump	To provide a dynamic water flow circulating through the tube
A turbine flow meter	To measure the water flow rate
RTD Sensors	To measure the water temperature inside of the cooler, and temperatures of water inlet and outlet of the finned tube
A digital scale	To measure mass of the condensate per hour
A cooler	To hold the chilled water
A data logger	To monitor and record the measured parameters

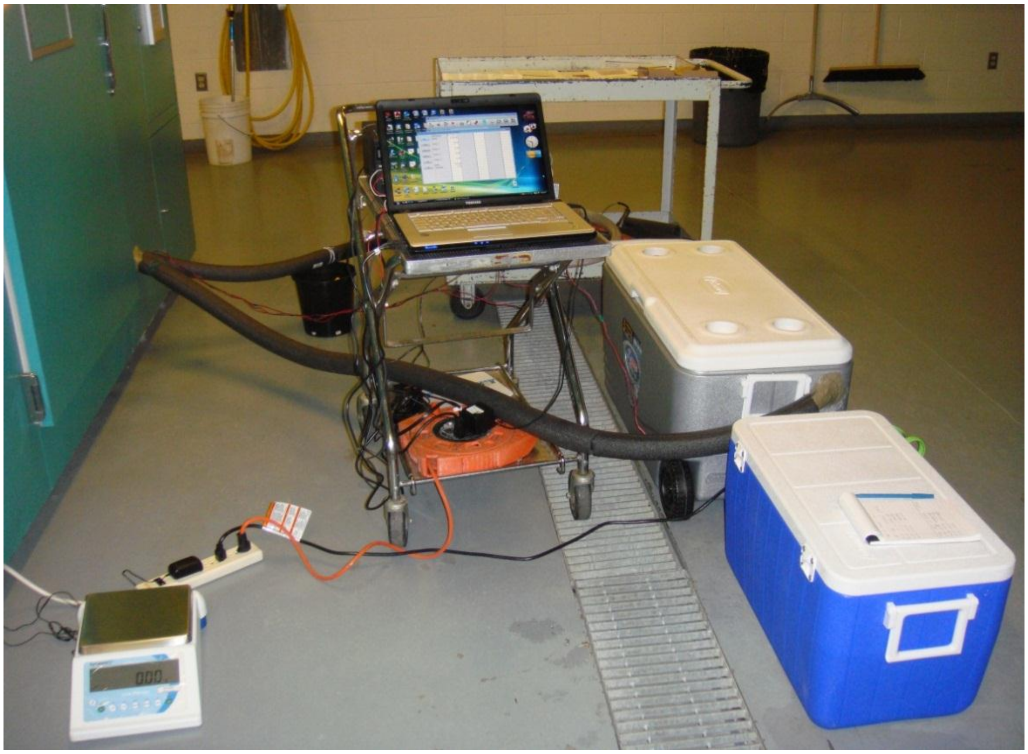

Figure 4.3 Picture of equipment set-up outside the phytotron chamber

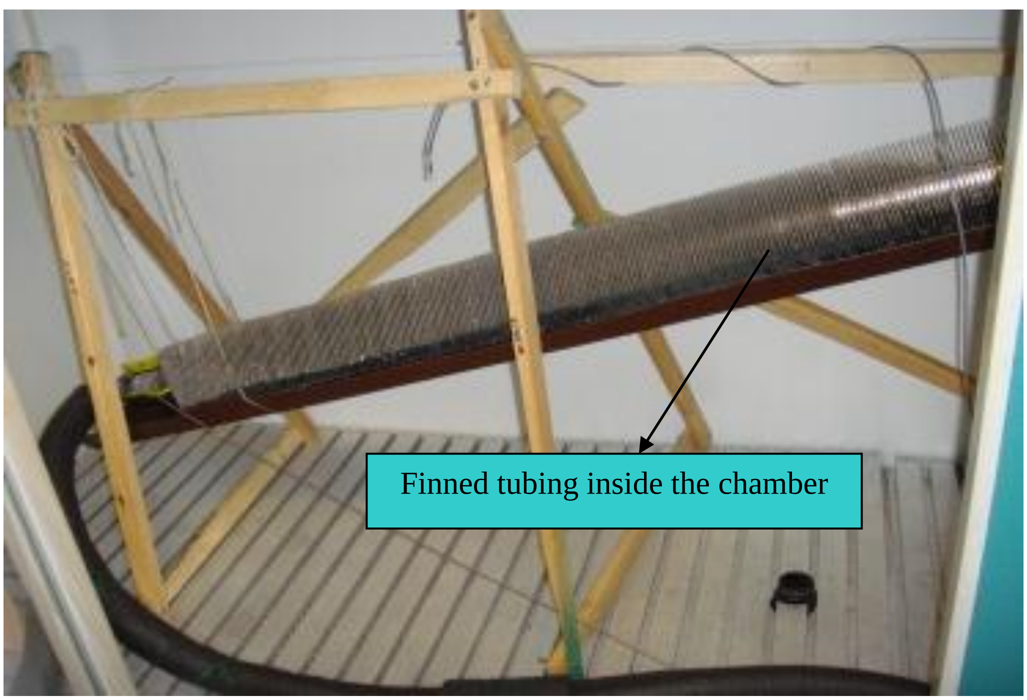

Figure 4.4 Picture of equipment set-up inside the phytotron chamber

A data logger (CR10X, Campbell Scientific Inc., USA) was used to acquire all the monitoring parameters (except the weight of the condensed water). The CR10X was a fully programmable data logger/control with non-volatile memory and a battery backed clock in a small, rugged, sealed module. A personal computer was connected to the data logger to get the real time data readings. The condensed water per hour of each test was weighted and recorded manually.

The data acquisition program was compiled using CR10X programming software. The real time data were displayed every second, and all the data were recorded every minute for each test of an hour. The complete program of the data logger is illustrated in Appendix D.1.

4.1.3 Condensation Rate Measurement Design

Considering that the temperature in a greenhouse usually ranges from 16 ℃ to 24 ℃, and high RH often varies from 70% to 90%, three temperatures, 16 ℃, 20 ℃, and 24 ℃, and three RH levels, 70%, 80%, and 90%, were selected as the room temperature and RH set points. A finned tubing condensation system using chilled water was designed and set up in the chamber (see 4.1.2 for details). A Factorial experiment was designed with the dependent variable being the condensation rate and four independent variables: room temperature, room RH, chilled water temperature, and water flow rate. Two levels for water temperature were designed, 0 ℃ and 5 ℃, and 0.65 L/s, 0.85 L/s, 1.05 L/s, 1.25 L/s, and 1.45 L/s were chosen as five levels for water flow rate. Three replicates were conducted for each treatment.

Considering the difficulties of achieving 5 ℃ water control with ice bath, 0.65 L/s and 0.85 L/s were selected for 5 ℃ water tests first, and the results would be compared with that using 0 ℃ water. If the condensation rates with 5 ℃ water were significantly lower than those with 0 ℃ water, then the other three larger flow rates with 5 ℃ water would be given up. The specific experiment conduction steps are shown in Table 4.2. Five stages correspond to the 5 water flow rates. Each stage had 18 treatments (3 room temperature levels x 3 room RH levels x 2 water temperature levels), except that whether the 5 ℃ water tests for stages 3 to 5 would be conducted was dependent on the results of stages 1 and 2. The results showed that the water temperature had a significant effect on condensation rates; therefore, the tests with three greater water flow rates for 5 ℃ water were eliminated. The tests under 16 ℃ for three larger flow rates were also cancelled because of the extensive work they would demand.

Table 4.2 Condensation rates measurement steps

Stage	Room Temp (℃)			Room RH (%)			Water Temp (℃)		Flow Rate (L/s)
1	24	20	16	70	80	90	0	5	0.65
2	24	20	16	70	80	90	0	5	0.85
3	24	20	N/A	70	80	90	0	N/A	1.05
4	24	20	N/A	70	80	90	0	N/A	1.25
5	24	20	N/A	70	80	90	0	N/A	1.45

Note: N/A means test eliminated.

Before the measurements, two hours "warm-up" time was required for the system to obtain a steady state condensation rate. Each treatment included three repetitions and thus needed more than five hours. There were a total of 108 tests for 0 ℃ water and 54 tests for 5 ℃ water.

4.1.4 Calibration of the Instruments

The temperature sensors, flow meter, and digital scale were all calibrated prior to the instrument installation in the Electronics Laboratory, Department of Agricultural and Bioresources Engineering, University of Saskatchewan. A dry block temperature calibrator (9170 Hart Scientific, Fluke Corporation, American Fork, UT, USA) was used to calibrate the RTDs. For the flow meter, the converter which converted the pulses to the flow rate was calibrated using an empty bucket (20.508 L), data logger (CR10X, Campbell Inc., USA), stopwatch, and tap water. The balance of the digital scale was calibrated using a set of standard weights (1.00 g to

2000.00 g, Rice Lake Weighing Systems Corporation, Rice Lake, WI, USA). The specific calibration procedures and results are presented in Appendix A.

4.1.5 Condensation Experiment Data Analysis

The condensation data analysis was composed of three parts: statistical analysis of the condensation rates as affected by the variables, statistical modeling of the condensation rate, and the energy efficiency analysis of the condensation system.

4.1.5.1 Condensation Rates Statistical Analysis

After the experimental results with two water temperatures (0 ℃ and 5 ℃) and flow rates (0.65 L/s and 0.85 L/s) were obtained under certain environmental conditions with different combinations of temperature (16 ℃, 20 ℃, and 24 ℃) and RH (70%, 80%, and 90%), four-way ANOVA was applied to determine whether each of the four factors had significant effects on the condensation rate. Since the four variables were set and controlled manually, there are no interactions between/among the variables. The data for analysis includes all the measurements with 0.65 L/s and 0.85 L/s water flow rates for 0 ℃ and 5 ℃ water. The model describing the relationship of independent variables and dependent variable can be expressed in Table 4.3.

Table 4.3 Source of variation and degrees of freedom (24, 20, and 16°C room temperature; 90, 80, and 70% RH; 0 and 5°C water; 0.65 and 0.85 L/s)

Source of Variation	Degrees of Freedom
Room temperature (Ti)	2
Room RH (RH)	2
Water temperature (Tw)	1
Water flow rate (FR)	1
Error	101
Total	107

If water temperature was proven to have a significant effect on the condensation rate, the pair-wise comparison was then performed to test the null hypotheses of no difference between the condensation rates with 0 ℃ and 5 ℃ water temperature. The experiments using 5 ℃ water would not be conducted with greater water flow rates (1.05 L/s, 1.25 L/s, and 1.45 L/s) unless the condensation rates with 5 ℃ water were significantly higher than those with 0 ℃ one.

Three-way ANOVA was applied to determine whether the three factors (room temperature, room RH, and water flow rate) have significant effects on the condensation rate when using 0 ℃ water. The data used to analyze the effects include all the measurements with 0 ℃ water for five water flow rates. The model describing the relationship between the independent variables and the dependent variable is shown in Table 4.4.

Table 4.4 Source of variation and degrees of freedom (24, 20, and 16°C room temperature; 90, 80, and 70% RH; 0°C water; 0.65, 0.85, 1.05, 1.25, and 1.45 L/s)

Source of Variation	Degrees of Freedom
Room temperature (Ti)	1
Room RH (RH)	2
Water flow rate (FR)	4
Error	82
Total	89

If the water flow rate had a significant effect on the condensation rate, then Tukey's HSD (Honestly Significant Difference) test would be performed to find which flow rate was different from the others. All the statistical analyses were conducted using Statistical Package for the Social Sciences (SPSS 19.0, SPSS Inc. and IBM Company, Chicago, USA).

4.1.5.2 Statistical Modelling

Given the assumption that the condensation rate of the finned tubing could be mathematically simulated based on room temperature, room RH, water temperature, and water flow rate, two statistical models were developed using SPSS 19.0 to predict the condensation rate as a function of these four predictors, which were applicable to two different conditions: the first model was developed for the room temperature of 16, 20, and 24 ℃, room RH of 70, 80, and 90%, water temperature of 0 and 5 ℃, and water flow rate of 0.65 and 0.85 L/s; the second model was developed for the room temperature of 20 and 24 ℃, room RH of 70, 80, and 90%, water temperature of 0 ℃, and water flow rate of 0.65, 0.85, 1.05, 1.25, and 1.45 L/s. The complete data of each condition were pooled into the stepwise linear regression procedure of SPSS 19.0 to generate a linear model.

To simulate the relationship between measured condensation rates and four predictors, multiple linear regression and second order polynomial regression were both tried in the SPSS regression procedures. However, there was not much improvement in second order polynomial regression models, and the equations were too complicated. Therefore, the multiple linear regression procedure was utilized for model development, as given in Equation 4.1.

$$\mathrm{CR} = \beta_0 + \beta_1 \times \mathrm{T_i} + \beta_2 \times \mathrm{RH} + \beta_3 \times \mathrm{FR} + \beta_4 \times \mathrm{T_w} \tag{4.1}$$

where

CR is the condensation rate of finned tubing system, in g h^{-1} m^{-1},

T_i is room air temperature, in ℃,

RH is the relative humidity of the room air, in %,

FR is water flow rate, in L s^{-1},

T_w is water temperature, in ℃,

β_0 is a constant, and β_1, β_2, β_3, and β_4 are unknown coefficients.

4.1.5.3 Energy Efficiency Analysis of Finned Tubing Condensation System

The third part was the energy efficiency analysis, including moisture removal index and energy cost. Moisture removal index of the finned tubing condensation system was defined as the total energy input (including electrical energy consumption of the pump, heat loss from room air to the water of the finned tubing system, and the latent heat released by the condensate) per liter of the condensate water collected, and was calculated by Equation 4.2.

$$\mathrm{MRI_c} = (q_c + q_{lossc} - q_{lc})/V_c = (q_c + M_c C_p \Delta \mathrm{T} t_c - q_{lc})/V_c \quad (4.2)$$

where

MRI_c is the moisture removal index of the finned tubing condensation system, in kW-h/L,

q_c is the electrical energy consumption of the pump, in kW-h. It is calculated using the power of the pump (provided by the manufacturer) multiplied by its running time,

q_{lossc} is the heat loss (thermal energy loss) from the room, which is absorbed by the chilled water during the heat transfer process, in kW-h,

M_c is the mass flow rate of the chilled water, in kg s^{-1},

C_p is the specific heat of water, in kJ kg^{-1} K^{-1},

ΔT is the temperature difference between the inlet and outlet water, in K,

t_c is the running time of the condensation system, in h,

q_{lc} is the latent heat released by condensed water (thermal energy released), in kW-h; here $q_{lc} = (h_{fg} \times m_w)/(3600 kJ/kW \cdot h)$, where h_{fg} is water heat of vaporization, in kJ kg^{-1}, m_w is the mass of the condensed water, in kg,

V_c is the volume of the water condensed by the finned tubing, in L, here $V_c = m_w/\rho_{water}$, ρ_{water} is the density of water, in kg L^{-1}.

According to the definition given for moisture removal index, lower moisture removal index indicates more energy efficient treatment. The energy cost of the chilled water system was defined as the energy cost per liter of water removed from the air. For the purpose of comparison with other dehumidification methods used in the greenhouse experiment, it was assumed that the energy sources all came from the fuel used for greenhouse heating (natural gas or thermal coal) except electrical energy for chilled water system. Then the energy cost was estimated by the electrical energy usage of the pump, and the fuel consumption resulted from heat loss and the latent heat released.

4.2 Field Experiment

The one-year field experiment was conducted in a tomato production greenhouse in Saskatchewan, from November 18, 2008, to November 17, 2009.

4.2.1 Description of Experimental Greenhouse and Dehumidification Requirement Determination

The greenhouse was located on a farm 20 km northeast of St. Louis, Saskatchewan, near the city of Prince Albert at 53.22^{o} latitude, 105.68^{o} longitude, and 428 m elevation. The building was a vaulted, steel-framed, single-span greenhouse covered with double-layer polythene plastic film on the roof. It was 9.1 m wide, 29.3 m long, and 4.2 m high at the ridge. Tomato plants were growing in the algoid medium in six rows with the intervals of approximately 1.1 m between plants. Detailed structures of the greenhouse are shown in Figures 4.5 and 4.6. There was a north entrance door connected with the header house, which was kept closed most of the time. The north wall used steel siding and 0.025 m polystyrene insulation, and two air inlets were located on the wall. The south wall was made of polycarbonate (PC) panels, and the exhaust fans are placed on the wall. The greenhouse was heated by a hot water heating system; within the greenhouse, the hot air was heated by two water-air heat exchangers which were distributed lengthwise to the greenhouse by two perforated air ducts made of plastic film (Figure 4.7). The boiler, using thermal coal, was located outside of the greenhouse.

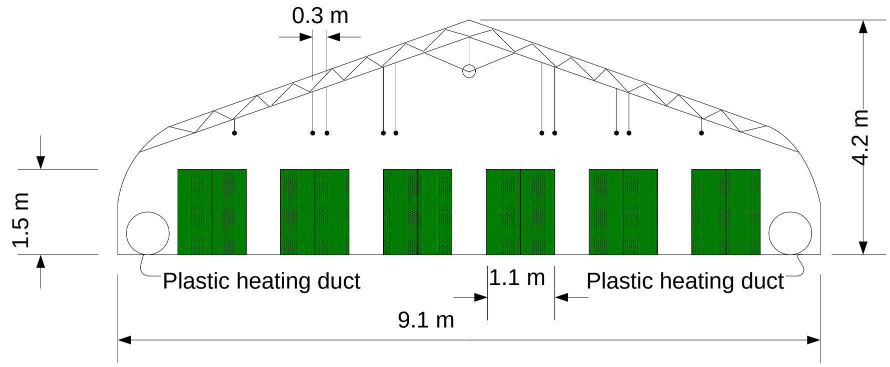

Figure 4.5 Greenhouse cross section

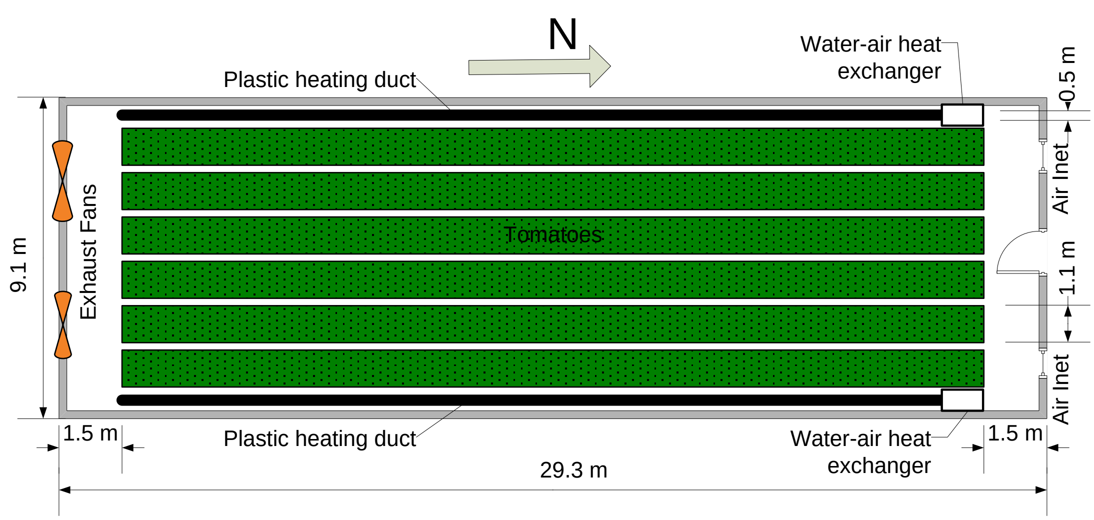

Figure 4.6 Greenhouse plan view

Figure 4.7 Plastic heating duct in the greenhouse (water-air heat exchanger in the front)

In a preliminary field measurement conducted in the spring of 2008, it was found that the maximum RH reached as high as 92.2% and that the average was around 76.7% (Obach, 2008). Excessive RH within the greenhouse produced many problems. One typical example was the water condensed on the inside of the roof dripping onto the plants. The growers complained that the drips hurt the plants and influenced their working efficiency as well.

RH controlled setpoint is a key factor to determine dehumidification requirement of the greenhouse, i.e. the amount of moisture that needs to be removed for adequate dehumidification. Here the RH setpoint was determined based on two concerns: first, the RH setpoint should be in a proper RH range for tomato plants. The optimal RH for greenhouse tomatoes ranges from 65 to 70% (Snyder, 2001). High RH above 80% tends to cause plant disease due to water dripping on the plants from condensate on the interior ceilings/walls of the greenhouse, while low RH under 50% may prevent the nutrition uptake and distribution of the plants; second, the RH setpoint determination should take energy cost and maintenance of dehumidification equipment into account. The RH setpoint of as low as 70% may increase operating time of the equipment, thus increase its power consumption while result in frequent kick on/off of the equipment reducing its longevity. Combined the above two concerns, the RH setpoint was determined to be 75%, although 5% higher but is still in the acceptable range for tomatoes.

Daytime temperature was controlled at 22 ℃ inside the greenhouse and nighttime temperature at 18 ℃. With the RH setpoint of 75% inside the greenhouse, the dehumidification requirement could be estimated for three conditions: daytime RH control, day-to-night RH control, and nighttime RH control. During daytime, moisture production was mainly by crop transpiration and water evaporation from growing media, which was estimated as vaporization of moisture by a quarter of total solar insolation gained by the greenhouse; and moisture removed was primarily dependent on natural or mechanical ventilation of the greenhouse, assuming condensation negligible in the greenhouse. Thus, the dehumidification requirement could be quantified by mass ventilation rate using Equation 4.3.

$$m_{rd} = \frac{M_p}{W_{id} - W_{od}} = \frac{Q_l / h_{fg}}{W_{id} - W_{od}} \tag{4.3}$$

where

m_{rd} is required mass ventilation rate of the greenhouse for daytime RH control, in kg s^{-1},

M_p is daytime moisture production rate inside the greenhouse, in kg s^{-1},

Q_l is the rate of latent heat gain of inside air during daytime, which is obtained from 25% of the net solar insolation into the greenhouse (Albright, 1990), in kW,

h_{fg} is heat vaporization of water, in kJ/kg; it can be calculated by $h_{fg} = 2501 - 2.42t$, where t is the greenhouse inside air temperature, in ℃ (Albright, 1990),

W_{id} is daytime inside air humidity ratio at 22 ℃ and 75% RH, in kg/kg dry air,

W_{od} is daytime ambient air humidity ratio, in kg/kg dry air.

Equation 4.3 neglects condensations and the other moisture sinks, thus the ventilation rate calculated over-estimates the required ventilation rate for adequate dehumidification during daytime.

During evening time around sunset, environmental conditions inside the greenhouse turn from daytime to nighttime control. Given an assumption that moisture produced inside greenhouse around evenings is zero because of little solar insolation at sunset, the dehumidification requirement can be quantified by the moisture mass difference between daytime and nighttime conditions.

During nighttime, inside RH is around the setpoint with little fluctuations after sufficient daytime and day-to-night dehumidification because low moisture production occurs inside the greenhouse at night and condensation and infiltration loss generally can balance the moisture production. Therefore, dehumidifying greenhouse air at night is considered unnecessary unless the dehumidification of daytime and day-to-night is not adequate.

Based on Prince Albert climate normals from 1953 to 1995 (U.S. Department of Energy, 2011), monthly average daytime and day-to-night dehumidification requirements for the experimental greenhouse are shown in Table 4.5. The sample calculations demonstrating the detailed dehumidification requirement calculation procedures are given in Appendix C.1.

Table 4.5 Monthly average dehumidification requirements

Month	Daytime RH control		Day-to-night RH control
	Moisture production rate (kg s^{-1})	Required volumetric ventilation rate (m^3 s^{-1})	Moisture mass to be removed (kg)
Jan.	0.0066	0.41	2.74
Feb.	0.0078	0.51	2.74
Mar.	0.0073	0.50	2.74
Apr.	0.0091	0.77	2.74
May	0.0083	0.84	2.74
Jun.	0.0085	1.26	2.74
Jul.	0.0098	1.71	2.74
Aug.	0.0082	1.53	2.74
Sept.	0.0078	0.89	2.74
Oct.	0.0078	0.72	2.74
Nov.	0.0053	0.38	2.74
Dec.	0.0066	0.43	2.74

As shown in Table 4.5, during daytime, the monthly average moisture production rate ranges from 0.0053 to 0.0098 kg s^{-1}, and the average required volumetric ventilation rate ranges from 0.38 to 1.71 m^3 s^{-1}. The moisture mass need to be removed from day to night is 2.74 kg, and then the moisture removal rate will depend on the intended removal time.

In the spring of 2008, Del-Air Systems from Humboldt, Saskatchewan, agreed to collaborate with the University of Saskatchewan on this project, and offer in-kind support of reduced priced heat exchangers. There are two types of air-to-air heat exchangers from Del-Air, i.e. Model RA400 and RA1000. RA400 has the exhaust ventilation rate of 0.183 m^3 s^{-1} and RA1000 has 0.349 m^3 s^{-1}. The combination of RA400 and RA1000 is able to provide 0.532 m^3 s^{-1} ventilation rate, which can meet the daytime dehumidification requirements during the cold weather period (average of 0.38 to 0.51 m^3 s^{-1}) and those during the mild period (average of 0.72 to 0.89 m^3 s^{-1}) without extra mechanical ventilation, and the warm season with some extra mechanical ventilation using the existing exhaust fans. The summer season dehumidification was originally not included in this project because the greenhouse producers are mainly concerned about "shoulder" season dehumidification, i.e. spring and fall seasons. From day to night, RA400 can meet the ventilation requirement of removing 2.74 kg moisture from the air within one hour (the specific calculations are shown in Appendix C.1.2). These two heat exchangers satisfy the dehumidification needs of the experimental greenhouse during cold and mild weather conditions and thus were selected for the field experiment.

Due to the fact that the capital cost of the commercial-grade dehumidifier which meets the dehumidification requirement was greatly over the budget (around $10,000 per unit plus special wiring and installation cost), it was decided that two units of domestic dehumidifiers ($300 per unit) with a combined capacity of 61.6 L/day were tested in the greenhouse to see the potential of the dehumidifier method for greenhouse dehumidification in a cold region based on the study of energy efficiency. Details of the dehumidifiers are given in the next section. Later in the experiment, it was found that the capacity was insufficient and two more dehumidifiers were added resulting in total capacity of 123.2 L/day.

4.2.2 Greenhouse Dehumidification Instruments Setup and Operational Design

Two Del-Air air-to-air heat exchangers (Model RA400 and RA1000, Del-Air Systems Ltd., Humboldt, SK, Canada) and two identical Danby domestic dehumidifiers (Model DDR6588EE, Danby Products Ltd., Canada) were selected for this greenhouse. Table 4.6 gives the basic specifications of the equipment. The normal ventilation management method of the greenhouse was by temperature controlled ventilation only (without humidity control). Thus, three treatments -- air-to-air heat exchangers, mechanical refrigeration dehumidifiers, and the grower's own control method of temperature based ventilation used as control treatment -- were applied in the greenhouse to test their dehumidification effects.

Table 4.6 Basic technical parameters of the equipment

Equipment		Capacity	Power (W)
Del-Air RA400 heat exchanger	Supply air fan	0.147 m^3/s	235
	Exhaust air fan	0.183 m^3/s	
Del-Air RA1000 heat exchanger	Supply air fan	0.242 m^3/s	368
	Exhaust air fan	0.349 m^3/s	
Danby DDR6588EE dehumidifier		30.8 L/day	668

Both heat exchangers were installed in the south wall of the greenhouse, approximately 2.5 m above the floor, between the two exhaust fans, as shown in Figure 4.8 and Figure 4.9. The identical dehumidifiers were put on wood tables at both ends inside the greenhouse (as shown in Figure 4.10); each was kept around 2 m from the heating pipe to avoid the ineffectiveness of the dehumidifiers because of the dry air close to the heating pipes. The dehumidifier located in the south was labelled as dehumidifier 1, and the north one was dehumidifier 2. A temperature and RH sensor was fitted into a radiation shield installed in the centre of the greenhouse, approximately 1.5 m above the ground. The CO_2 analyzer inlet tubing, pressure transducer inlet tubing for inside air pressure, and the pyranometer were also set up in the central area, nearly 2 m above the floor. The inlet of the tubing for ambient air pressure for the pressure transducer was located outside of the greenhouse 2 m above ground. Two thermocouples (fine gage bare wire and insulated thermocouples, OMEGA Engineering Inc., Quebec, Canada) were fixed near the inlet and outlet of each heat exchanger, respectively. The weather station was installed at flat field without shelters approximately 100 m southwest of the greenhouse (shown in Figure 4.11). The equipment setup, location, measurement point, and layout of the tubing and wires are illustrated in Figure 4.12.

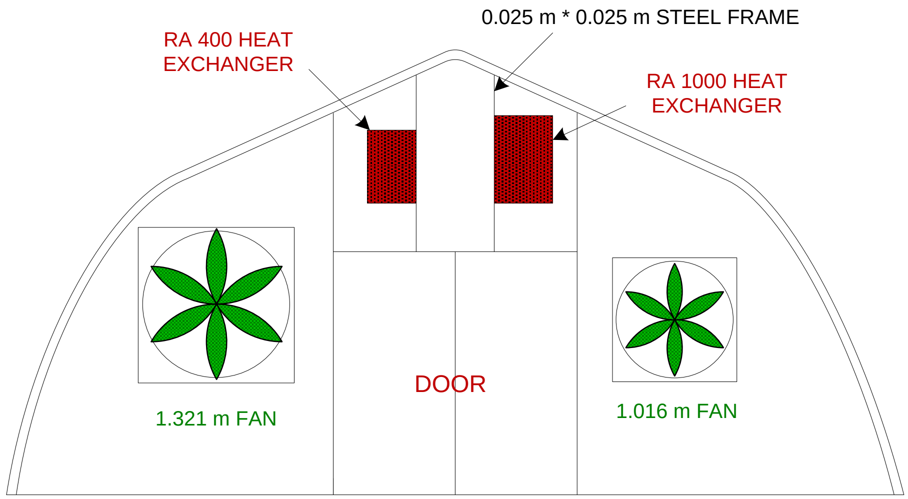

Figure 4.8 Heat exchangers on the south wall

Figure 4.9 Picture of heat exchangers setup

Figure 4.10 Picture of the dehumidifier setup

Figure 4.11 Picture of the weather station setup

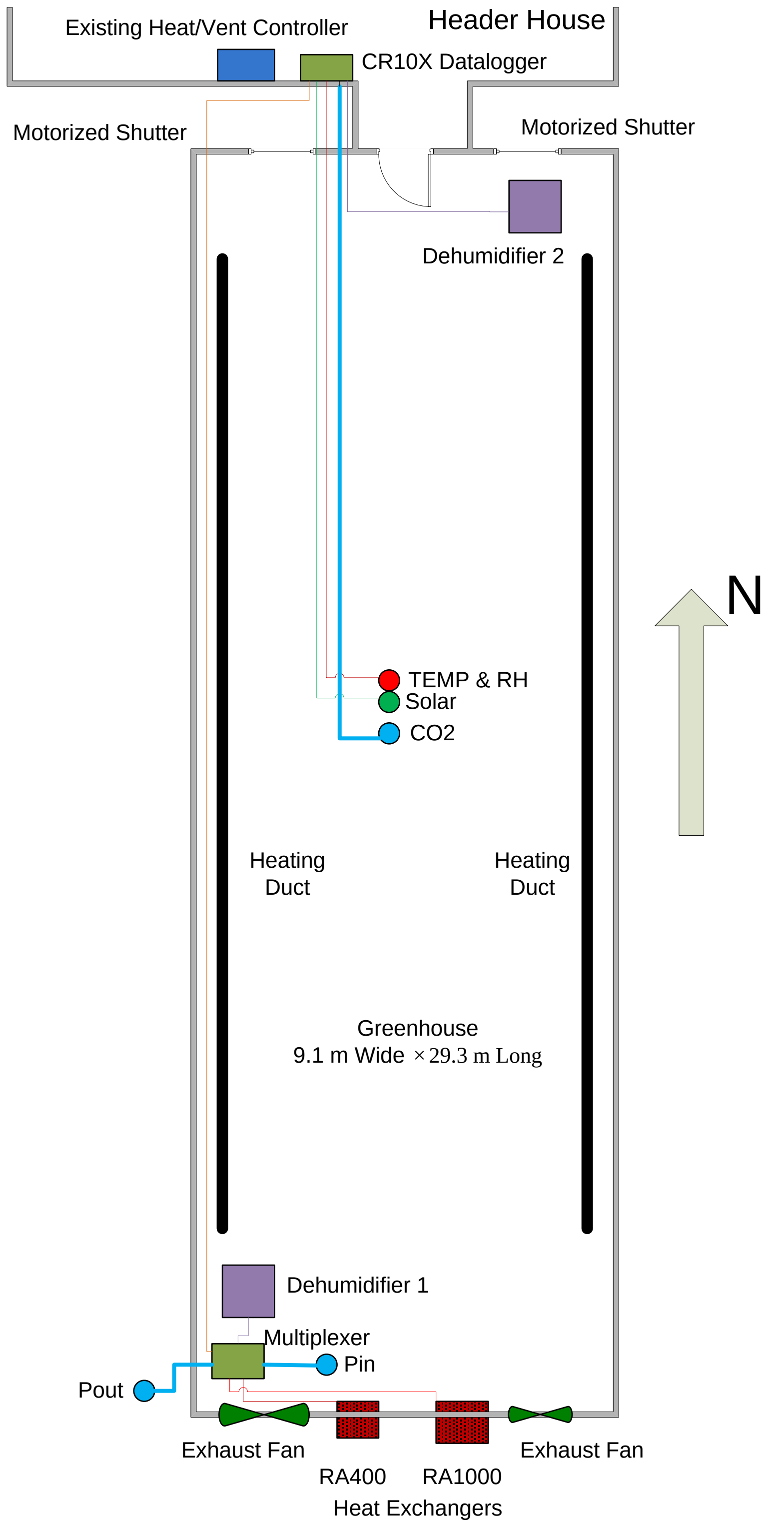

Figure 4.12 Instrumentation layout of the greenhouse experiment

In order to easily compare the dehumidification effects of the three methods, eight days were designed to be one cycle: the first three consecutive days for only heat exchanger treatment, the second three days with only dehumidifier treatment, and the remaining two days for the control method which was the temperature based ventilation control, i.e. relying on air infiltration in winter and ventilation using exhaust fans in the other seasons. The period for the control method was also designed as three days of one cycle at the very beginning, however, after the preliminary experiment with nine days as a cycle during two months before November of 2008, the results showed that the greenhouse inside RH in the control treatment days was particularly high (approximately 60% of the time the RH was above 80%) and the growers frequently complained the high humidity inside the greenhouse. Thus, the period of the control method was reduced to two days in order to reduce the potential damage to the plants and to meet the requirement of the producers as well.

Considering different capacities of two heat exchangers applied (RA400 and RA1000), the heat exchangers were set to be running in two stages: RA400 heat exchanger was switched on when the inside RH reached 75%, and it would be turned off when RH was reduced to 73%; RA1000 was started when the RH exceeded 80% and stopped when RH was lowered to 78%. In the dehumidifiers' running days, both of them were turned on when the RH was above 75% and off at 73%. Figure 4.13, Figure 4.14 and Figure 4.15 demonstrate the main airflows of the greenhouse on the heat exchangers' running days, the dehumidifiers' operation days, and the normal control days, respectively. The cycle was repeated for continuous period from November 18, 2008, to June 17, 2009. Since then, the control treatment was eliminated under the request of the growers because they claimed that the control days resulted in too humid conditions. Therefore, during June 17 to November 17, 2009, a cycle had 6 days, 3 days of heat exchanger treatment and 3 days of dehumidifier treatment.

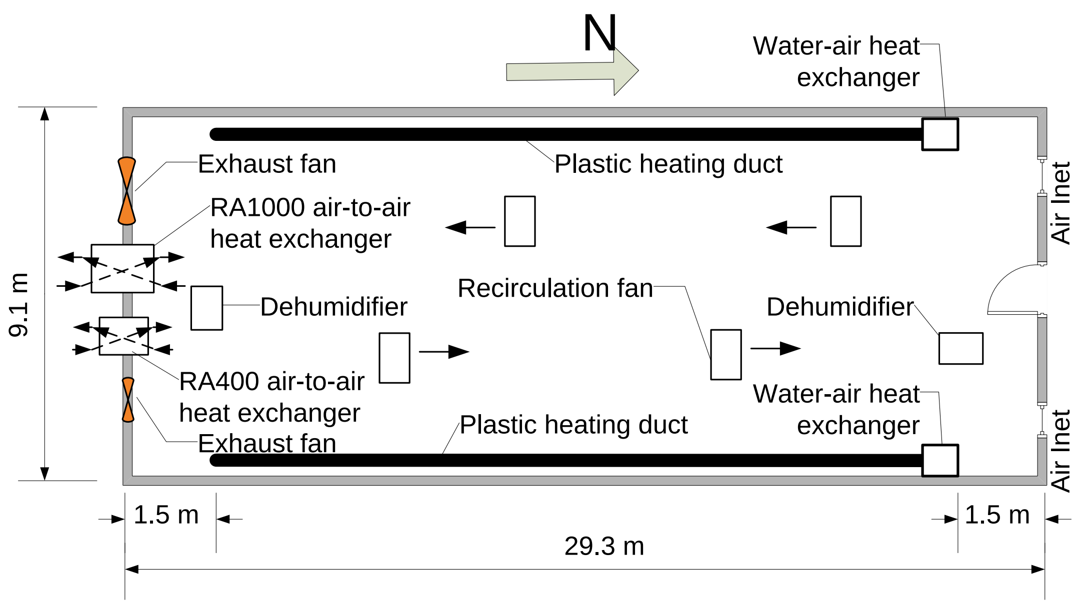

Figure 4.13 Schematic diagram of general airflows on heat exchangers' running days

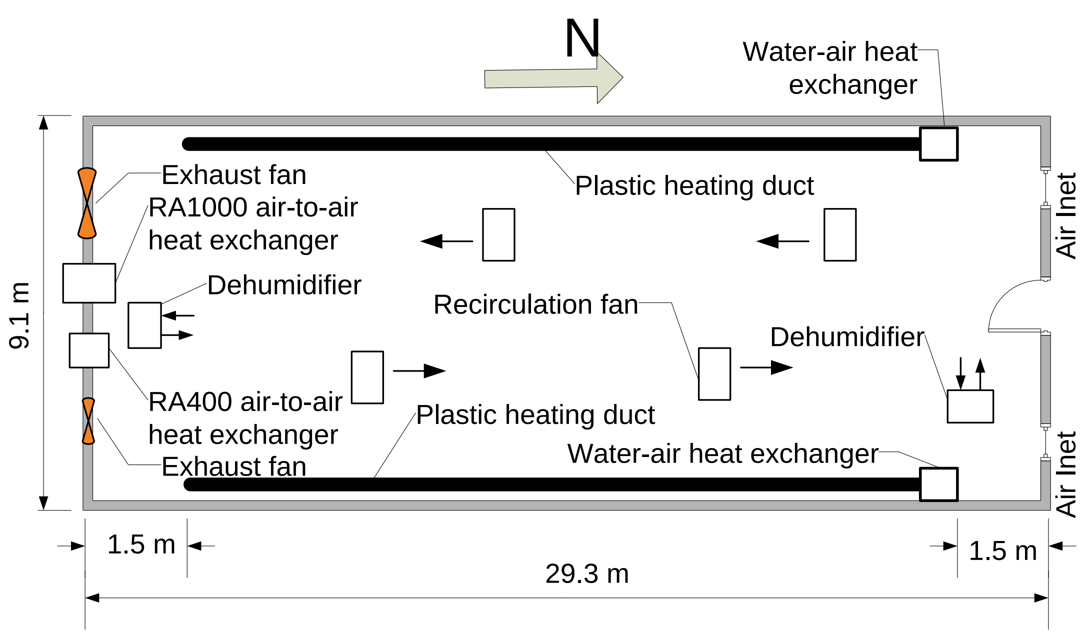

Figure 4.14 Schematic diagram of general airflows on dehumidifiers' running days

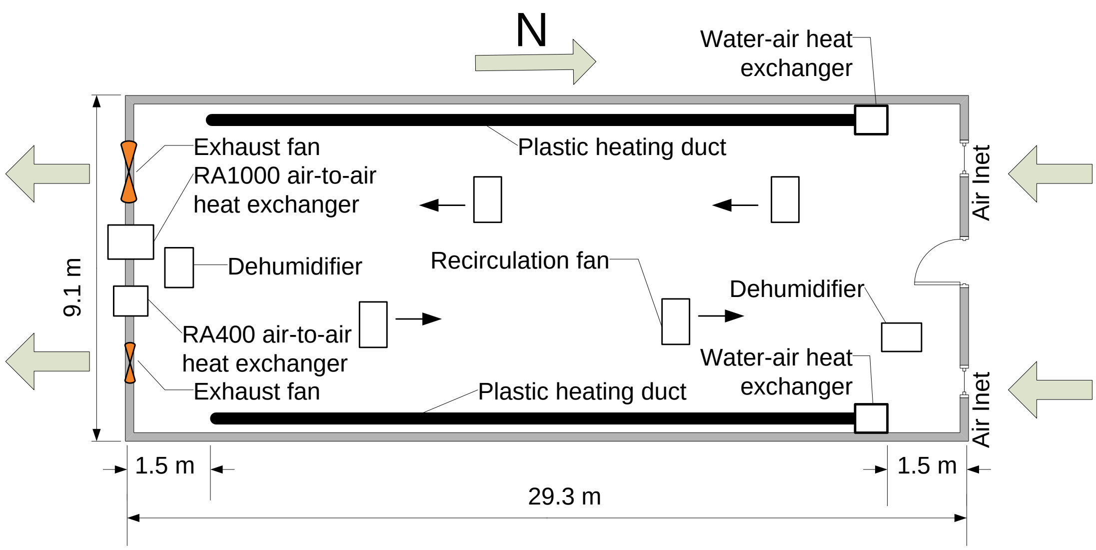

Figure 4.15 Schematic diagram of general airflows on control days

4.2.3 Environment Condition Measurement Design

During the experimental period, the following environmental parameters inside the greenhouse were monitored every minute with the 10 min average recorded from November 18, 2008, to November 17, 2009: air temperature, RH, temperature of supply air through heat exchanger entering greenhouse, temperature of exhaust air leaving heat exchanger, CO_2 concentration, solar radiation, air pressure difference between inside and outside air. The air temperature and RH were measured with a temperature and relative humidity probe (CS500, Campbell Scientific Inc., USA), composed of a platinum resistance temperature detector (PRT) and a Vaisala INTERCAP capacitive RH sensor (CS500 Temperature and Relative Humidity Probe Manual, 2004). The temperature sensor supports a measuring range of -40 to 60 ℃ with accuracy of ±0.2 to ±1.4 ℃ according to the measured temperature. For the greenhouse temperature range of 10 to 30 ℃, its accuracy is normally around ±0.6 ℃. The RH sensor provides a full measurement range of 0 to 100% RH in non-condensing conditions, with accuracy dependent on environmental temperature. At 20 ℃, the accuracy is approximately ±3% in the range of 10 to 90% and ±6% in the range of 90 to 100% RH. The probe was put inside a solar radiation shield to prevent the effects of solar radiation and dripping water. The air temperatures entering and leaving the two heat exchangers (T_1 and T_3) were monitored by thermocouples (fine gage bare wire and insulated thermocouples, OMEGA Engineering Inc., Quebec, Canada). CO_2 concentration was measured using a CO_2 analyzer (Guardian Plus Infrared Gas Monitor, Edinburgh Sensors Ltd., Hingham, MA, USA). The accuracy is ±2.5% of the measuring range (0 to 3000 ppm) and the measurements are not influenced by 0 to 99% RH (Infrared Gas Monitors Manual, 2009). The inlet tubing was cleaned every two months to avoid measurement errors that could be caused by the water vapour condensed inside the tubing. Solar radiation was monitored with a pyranometer sensor (LI-200, LI-COR Inc., Lincoln, Nebraska, USA). Current output in units of watt per square meter is directly proportionate to solar radiation, and the error is usually less than 5% under natural daylight conditions. The pyrometer was set at a horizontal and unshielded location to prevent any obstructions (LI-COR Terrestrial Radiation Sensors Instruction Manual, 2005). The air pressure difference between inside and outside was measured with a pressure transducer (Model 265, Setra System Inc., Boxborough, MA, USA) which was able to convert the differential static pressures into a proportional electric output. The measurement range was from 62.21 to 1244.20 Pa, and the accuracy could be as high as 0.25% FS (Model 265 Pressure Transducer Manual, 2008). The volume of the condensed water (in unit of L) from the dehumidifiers was measured using two metering buckets (0.1 L accuracy). The volume of the water was recorded by the growers every noon around 12:00 pm and every evening around 6:00 pm. The buckets were emptied after the water volume was recorded.

The equipments' operational information was recorded along with the environmental conditions: heat exchangers on/off, dehumidifiers on/off, heaters on/off, and air inlet on/off. The heat exchangers and dehumidifiers were activated when the RH reached the setpoints by using solid state relays (6225AXXSZS-DC3, Magnecraft Corporation, Illinois, USA) with input voltage range of 3 to 32 VDC and output voltage range of 24 to 280 VAC. The heaters and air inlet were controlled by the greenhouse's existing control system -- Wadsworth Step Control System (Wadsworth Control Systems Inc., Arvada, USA) based on temperature setpoints. In addition, the daily maximum and minimum of air temperature, RH, CO_2 concentration, solar radiation, and pressure difference were also recorded.

Weather has a great influence on the greenhouse's indoor climatic conditions. A weather station was installed near the greenhouse to measure the meteorological parameters, including temperature, RH, solar radiation, rainfall, wind speed, and wind direction. The steel support frame for the weather station was approximately 3 m high. Most instruments used to measure the outside air temperature, RH, solar radiation, and rainfall were the same models as the inside ones. The horizontal wind speed and wind direction were measured by Wind Sentry Anemometer and Vane (Model 03002, R. M. Young Company, USA). The anemometer was made up of three cup wheels which rotated as the wind blew to generate an AC sine wave voltage signal, and the produced frequency was proportional to the wind speed. The vane position was conveyed by a precision conductive potentiometer which produced an analog voltage signal proportional to the wind direction angle with a constant excitation voltage input (Model 03002 Wind Sentry Manual, 1999). The wind sentry was mounted on a metal pipe connected to the ground located above any possible obstructions. Each environmental parameter was monitored every minute. The average of twenty minutes was recorded.

4.2.4 Data Acquisition System

Inside the greenhouse, some of the environmental parameters or operational signals, including pressure difference, both heat exchangers on/off, the dehumidifier (located in the south) on/off, were forwarded by a relay multiplexer (AM16/32, Campbell Scientific Inc., USA) to a data logger (CR10X, Campbell Scientific Inc., USA). The others (i.e., air temperature, RH, solar radiation, CO_2 concentration, etc.) were all directly acquired by the same data logger, a fully programmable device with large data storage capacity. Figure 4.16 shows a picture of the acquisition system.

The real time data could be read on a personal computer, and the recorded data were downloaded through 9-pin serial ports communication mode from the data logger.

Figure 4.16 Picture of greenhouse environment acquisition system

In the weather station, all the meteorological parameter signals, including temperature and RH, solar radiation, rainfall, wind speed and wind direction data, were obtained by another data logger (CR1000, Campbell Scientific Inc., USA). It was placed in a closed chest to prevent direct solar radiation and rainfall damage.

The data acquisition program for the greenhouse's inside environmental conditions was compiled using CR10X programming software. Averaged values for every ten minutes were used for evaluation. For meteorological parameters, the data acquisition program was compiled with CR1000 programming software. Averaged values for every twenty minutes were recorded, considering the relatively smaller storage capacity of CR1000 data logger. The complete programmings of the data loggers are given in Appendix D.2 and D.3.

4.2.5 Calibration of the Sensors

Before the instrument installation, all the temperature and RH sensors, CO_2 monitor, pressure transducer, wind speed hall sensor, and wind direction sensor were calibrated in the Electronics Laboratory, Department of Agricultural and Bioresources Engineering, University of Saskatchewan (the CO_2 monitor was calibrated in the laboratory of Prairie Swine Centre, 15 km from Saskatoon, Saskatchewan). Vaisala hand-held humidity and temperature meter (HM70, Vaisala Inc., Woburn, MA, USA), source temperature control system (Model 100, J.C. Schumacher Corporation, Phoenix, AZ, USA), CO_2 calibration gas (1529 ppm and 2295 ppm, Brax Air Corporation, USA), precision pressure indicator/calibrator Druck DPI 605 (GE Industrial Sensing, Fairfield, CT) and manometer (34FB2TM, Meriam Instrument, Div of the Scott & Fetzer Co. Cleveland, Ohio, 44102, USA), 50 MHZ pulse generator (Model 801, Wavetek, CA, USA) , and 1.3 GHZ Frequency Counter (FC130A, Beckman Industrial Co., CA, USA) were used to calibrate these sensors. The calibration procedures and results are presented in Appendix B.

4.2.6 Tomato Crops Management

The experimental greenhouse was operated by the owner. Using the same header room and parallel to the experimental greenhouse, there was a larger multi-span greenhouse to produce cucumbers and peppers. The products were sold to Prince Albert’s grocery stores and farmer’s market.

In the experimental greenhouse, tomatoes were grown in six rows with a total of 612 plants, including three varieties: globe tomatoes, plum tomatoes, and cherry tomatoes. The soilless growing medium was made with peat moss and vermiculite and placed in bags. A drip irrigation system provided the water and nutrition supply for both greenhouses. According to the producers' estimation, the total amount of nutrient solution distributed to the experimental greenhouse was approximately 1200 L/day, nearly 2 L of nutrient solution/plant/day. Mechanical pollination was served by the producers during the flowering period. The tomato production from this greenhouse features pesticide-free organic vegetables; thus, no pesticide was applied in the greenhouse. Plants often experience fungal disease problems because of high RH during spring, fall, and wintertime, which stands as the reason that the producers are willing to cooperate with our research team in order to develop a suitable dehumidification technology for this cold region.

The greenhouse heating system, air inlets, and ventilation fans were all controlled by a Wadsworth control system based on the temperature setpoints. Supplementary lighting and CO_2 enrichment were unavailable for the greenhouse. Much field work is required in the greenhouse every day during its operational period, including pruning, tomato thinning, hand-pollinating, and similar tasks. Thus, a good work environment is necessary. The greenhouse operated from late February until the end of December of each year. During the period of November 18, 2008, to November 17, 2009, the tomato plants were removed and the greenhouse was shut down from January 2 to March 3, 2009.

4.2.7 Greenhouse Dehumidification Data Analysis

4.2.7.1 Greenhouse Dehumidification Seasons Identification in Cold Regions

In order to avoid the environmental effect of the previous treatment, data from the first day of each treatment or control were discarded in the data analysis. Thus for each cycle, two days' data were left for heat exchanger/dehumidifier treatment and one day's data for the control period.

The monthly percentage of the running time of the three treatments (heat exchanger, dehumidifier, and exhaust fan) was calculated using Excel (2010, Microsoft Inc., USA) to identify the dehumidification requirement period for greenhouses in cold regions. The conditions of 75% RH or higher reflected the inability of controlling the RH within the setpoint by the dehumidification methods tested; therefore, the percentage of the time that RH was 75% or above (75% RH exceeding percentage) was calculated for each treatment and the value of greater than 25% can be regarded as the dehumidification season. Monthly and annual exceeding percentages were summarized to conclude the results.

4.2.7.2 Performance Analysis of Heat Exchangers

Prior to the installation of the equipment, both heat exchangers were calibrated by the manufacturer (Del-Air Systems Ltd., Humboldt, SK, Canada) for air flow rates of the supply and exhaust fans. Heat recovery ratio is the most important performance parameter for heat exchangers and thus should be calculated to examine their performances. Heat recovery ratio (HRR) is defined as the fraction of the exhaust sensible heat recovered by the supply air, which was calculated by Equation 4.4.

$$HRR = Q_r/Q_e = \left[M_{supply}C_{ps}(T_1 - T_0)\right]/\left[M_{exhaust}C_{pe}(T_2 - T_0)\right] \qquad (4.4)$$

where

Q_r is the rate of sensible heat recovered by the supply air stream, in kW,

Q_e is the rate of maximum amount of heat that could be recovered, in kW,

M_{supply} is mass flow rate of the supply air, in kg s^{-1},

C_{ps} is specific heat of the supply air, in kJ kg^{-1} K^{-1},

T_1 is the temperature of the supply air from heat exchanger entering the greenhouse, in K,

T_0 is the temperature of the incoming supply air entering the heat exchanger which is the outside temperature, in K,

$M_{exhaust}$ is mass flow rate of the exhaust air, in kg s^{-1},

C_{pe} is specific heat of the exhaust air, in kJ kg^{-1} K^{-1},

T_2 is the temperature of the exhaust air entering the heat exchanger (assume it equal to the inside temperature), in K.

Apparently, the higher the heat recovery ratio, the better is the performance of the heat exchanger. All the temperatures and RH levels when heat exchangers were running were averaged for each cycle using Excel (2010, Microsoft Inc., USA), and psychrometric equations (Albright, 1990) were used to find the specific volume and humidity ratio for air mass flow rate calculation.

4.2.7.3 Greenhouse Dehumidification Effect Evaluation

One of the main objectives of the project is to evaluate the dehumidification effects with three different treatments applied in the greenhouse under different weather conditions in a cold region. Prince Albert's climate information on monthly averages for 30 years from 1971 to 2000 is given in Table 4.7. According to Table 4.7, the weather condition of Prince Albert can be categorized into three groups: cold climate with average temperature below 0 ℃ (January, February, March, November, and December), mild climate with average temperature between 0 to 10.5 ℃ (April, May, September, and October) and warm climate with average temperature above 10.5 ℃ (June, July, and August). However, for the experimental year, September of 2009 fell into the warm period, for the average of the temperature of that month was around 22.5 ℃.

Table 4.7 Prince Albert climate normals 1971-2000 (Environment Canada, 2010)

Month	Temperature (℃)			
	Daily Max	Daily Min	Daily Average	S. D.
Jan.	-13	-25.2	-19.1	4.6
Feb.	-8.3	-20.9	-14.6	4.5
Mar.	-1.3	-13.6	-7.5	3.6
Apr.	9.5	-3.4	3.1	2.5
May	17.7	3.3	10.5	1.9
Jun.	21.8	8.6	15.2	1.5
Jul.	23.9	11.1	17.5	1.2
Aug.	23.1	9.4	16.3	1.9
Sept.	16.6	3.6	10.2	1.7
Oct.	9.3	-2.5	3.4	1.4
Nov.	-2.9	-12.2	-7.6	3.6
Dec.	-10.8	-21.6	-16.2	4.5

Greenhouse dehumidification effects of three treatments were evaluated based on three criteria. The first is RH control accuracy, which is defined as the exceeding percentage when the inside RH was above the setpoint of the treatment, i.e. 75%, during each treatment period. The lower the 75% RH exceeding percentage, the more accurate is the RH control ability of the treatment.

The second criterion is the moisture removal index (MRI), which is defined as the total energy input per liter of water removed from the air of the greenhouse. For heat exchangers, the total energy input was composed of the electrical energy consumption and the total heat loss from the greenhouse through the exhaust air of the heat exchangers. Two-day data of each cycle during one season were used for the MRI calculations of both the heat exchanger and dehumidifier. The MRI of the heat exchangers was calculated by Equation 4.5.

$$\mathrm{MRI_h} = \frac{\mathrm{q_h} + \mathrm{q_{lh}}}{\mathrm{V_h}} = \frac{\mathrm{q_h} + [\mathrm{M_{exhaust}} \mathrm{h_2} - \mathrm{M_{supply}} \mathrm{h_1} - (\mathrm{M_{exhaust}} - \mathrm{M_{supply}}) \mathrm{h_0}] \times \mathrm{t}}{\mathrm{V_h}} \tag{4.5}$$

where

MRI_h is the moisture removal index of the heat exchanger, in kW-h/L,

q_h is the electrical energy consumption of the heat exchangers during the two days, in kW-h. It is calculated using the power of the heat exchanger (provided by the manufacturer) multiplied by its running time,

q_{lh} is the net total heat loss through ventilation of the heat exchangers, in kW-h,

$M_{exhaust}$ is the mass flow rate of the exhaust air, in kg s^{-1},

M_{supply} is the mass flow rate of the supply air, in kg s^{-1},

h_1 is the enthalpy of the supply air as it leaves the heat exchanger prior to mixing with the greenhouse air, in kJ kg^{-1},

h_2 is the enthalpy of the exhaust air entering the heat exchanger, in kJ kg^{-1},

h_0 is the enthalpy of the ambient air, in kJ kg^{-1},

t is the running time of the heat exchanger, in h,

V_h is the net volume of water removed by the heat exchangers, which is the moisture removed by the exhaust fan minus the moisture coming into the greenhouse through the supply air, in L.

For dehumidifiers, the total energy input was the total input in forms of electrical energy consumption, heat output of the dehumidifiers, and the latent heat released by the condensed water. The MRI of the dehumidifiers was calculated by Equation 4.6.

$$MRI_d = \frac{q_d-(q_o+q_l)}{V_d} = \frac{q_d-(q_o+\frac{h_{fg}\times m_{water}}{3600kJ/kW\cdot h})}{V_d} \tag{4.6}$$

where

MRI_d is the moisture removal index of the dehumidifier, in kW-h/L,

q_d is the electrical energy consumption of the dehumidifiers during the two days, in kW-h. It is calculated using the power of the dehumidifier (provided by the manufacturer) multiplied by the running time,

q_o is the heat output of the dehumidifiers, which is assumed to be 90% of its electrical energy consumption (Hosni et.al, 1999), in kW-h,

q_l is the latent heat released by condensed water of dehumidifiers during the two days, in kW-h,

h_{fg} is water heat of vaporization, in kJ kg^{-1}, which is calculated by $h_{fg} = 2501 - 2.42t_w$ (t_w is temperature of the condensed water, ℃, and it is assumed equal to the average room air temperature) (Albright, 1990),

m_{water} is the mass of the condensed water collected during last two days of each test cycle, in kg,

V_d is the volume of the water removed by dehumidifiers, in L, here $V_d = m_{water}/\rho_{water}$, ρ_{water} is the density of water, in kg L^{-1}.

The MRI of the hypothetical treatment RH-based ventilation control was also calculated for the two days of the heat exchanger cycle. The total energy input was composed of electrical energy consumption of the exhaust fans and total heat loss from the greenhouse through the exhaust air. The MRI of the exhaust fan (RH-based ventilation) was calculated by Equation 4.7.

$$MRI_e = \frac{q_e + q_{le}}{V_e} = \frac{q_e + M_e(h_{e2} - h_{e1})t_e}{V_e} \quad (4.7)$$

where

MRI_e is the moisture removal index of the exhaust fan, in kW-h/L,

q_e is the electrical energy consumption of the exhaust fans (assumed the same specifications as the exhaust fans of the heat exchangers) during the two days of each cycle, in kW-h. It is calculated using the power of the exhaust fans multiplied by the running time,

q_{le} is the net total heat loss through ventilation of the exhaust fans during two days, in kW-h,

M_e is the mass flow rate of the exhaust air, in kg s^{-1},

h_{e1} is the enthalpy of the outside air, in kJ kg^{-1},

h_{e2} is the enthalpy of the exhaust air, in kJ kg^{-1},

t_e is the running time of the exhaust fans in two days, in h,

V_e is the net volume of water removed by the exhaust fans, which equals the absolute difference of the moisture contained in the outside air and that in the exhaust air, in L.

The third criterion is energy cost, which is defined as the total energy cost per liter of water removed. For heat exchangers and exhaust fans, energy cost consisted of electrical energy cost and extra heating cost resulting from the heat loss during the ventilation process. The cost of electrical energy could be easily calculated using the latest electricity rate multiplied by the running time of heat exchangers. The extra heating cost was estimated based on the price of the fuel (natural gas or thermal coal) used to heat the greenhouse. For dehumidifiers, energy consumption was mainly from electrical energy used. The heat output of the dehumidifiers and the latent heat released by the condensed water compensated some of the energy used. It was assumed that the energy sources, except the electrical energy, all came from the fuel used in the greenhouse (natural gas or thermal coal). Then the energy cost was calculated by subtracting the fuel cost from the electrical energy cost. The utilization efficiency of natural gas used to heat the greenhouse was estimated as 90%, and the efficiency of the boiler was assumed to be 70%. The transportation heat loss was neglected because of the short distance and well insulated hot water pipe between the boiler room and the greenhouse.

4.2.7.4 Comparison of Greenhouse Dehumidification Treatments

The comparison was made among four different dehumidification methods: (1) heat exchangers; (2) mechanical refrigeration dehumidifiers; (3) ventilation based on RH control (using the exhaust fan of the heat exchanger data); (4) conventional ventilation based on temperature control. Numbers (1) and (2) were the designed dehumidification treatments during the whole year; (4) was the traditional control method, and was normally used in the humid spring and fall seasons; during winter, the air inlets and fans were all sealed. Number (3) was a hypothetical treatment similar to the heat exchanger dehumidification method but with no heat recovery, so the running conditions of the exhaust fans of the two heat exchangers were used for the analysis.

Number (1), (2), and (3) treatments were compared based on RH control accuracy, moisture removal index, and energy cost in different seasons. Since the existing exhaust fans' operating conditions were not monitored during the experiment period, the running time of the exhaust fans was unknown and thus the moisture removed by treatment (4) could not be calculated. Thus, the moisture removal index and the energy cost were not calculated for treatment (4). Instead, the revenue loss from the tomato yield loss caused by the high RH was estimated based on the growers' experience.

Chapter 5. RESULTS AND DISCUSSIONS

5.1 Laboratory Experiment

5.1.1 Finned Tubing Condensation Rate Results

The laboratory experiment was conducted in an environment chamber at the University of Saskatchewan from January 12, 2009, to May 30, 2009. With different designed conditions (room temperature, room RH, water temperature, and water flow rate), the condensation rates of the finned tube together with the other basic parameters such as water temperatures and water flow rates were measured and recorded for each treatment. Three replications were performed for each treatment. Table 5.1 gives the condensation rate results under the different design conditions.

Since the finned tube tested was of a standard size of one meter, the condensation rate was thus the mass of water removed per meter of the finned tube. The measured room temperature, water temperature, and flow rate summarized in Table 5.1 are averages of the three replications for each treatment.

Table 5.1 Summary of the condensation rate measurement results

T_i (℃)	RH (%)	FR (L/s)	T_w (℃)	FR_a (L/s)	T_{wa} (℃)	$CR1$ (g/h)	$CR2$ (g/h)	$CR3$ (g/h)	Mean (S.D.) (g/h)
24	90	0.65	0	0.61	-0.3	311.6	289.5	298.1	299.7 (11.2)
24	80	0.65	0	0.65	-0.41	250.3	245.1	246.1	247.2 (2.8)
24	70	0.65	0	0.61	-0.27	223.9	215.9	216.6	218.8 (4.4)
24	90	0.65	5	0.65	3.95	256.0	248.1	247.1	250.4 (4.9)
24	80	0.65	5	0.65	4.05	189.0	179.1	189.0	185.7 (5.7)
24	70	0.65	5	0.65	4.16	153.2	144.8	149.0	149.0 (4.2)
20	90	0.65	0	0.65	-0.44	207.8	202.6	203.2	204.5 (2.8)
20	80	0.65	0	0.64	-0.44	150.8	144.0	137.4	144.1 (6.7)
20	70	0.65	0	0.61	-0.44	132.1	130.5	132.1	131.6 (0.9)
20	90	0.65	5	0.66	4.18	177.7	172.8	177.5	176.0 (2.8)
20	80	0.65	5	0.65	4.31	125.5	114.6	120.1	120.1 (5.4)
20	70	0.65	5	0.65	4.34	75.7	73.1	74.4	74.4 (1.3)
16	90	0.65	0	0.65	-0.55	139.5	137.1	139.3	138.7 (1.3)
16	80	0.65	0	0.65	-0.55	86.7	84.4	84.4	85.2 (1.4)
16	70	0.65	0	0.56	-0.56	82.4	79.6	79.6	80.5 (1.6)
16	90	0.65	5	0.66	4.26	108.1	102.2	105.6	105.3 (2.9)
16	80	0.65	5	0.65	4.42	46.3	40.2	42.6	43.0 (3.1)
16	70	0.65	5	0.65	4.49	12.7	11.2	10.3	11.4 (1.2)
24	90	0.85	0	0.77	-0.47	320.2	308.3	316.9	315.1 (6.2)
24	80	0.85	0	0.71	-0.24	276.3	258.3	275.0	269.9 (10.0)
24	70	0.85	0	0.81	-0.16	222.4	215.0	218.7	218.7 (3.7)
24	90	0.85	5	0.85	4.03	260.5	258.5	258.9	259.3 (1.1)
24	80	0.85	5	0.84	4.02	197.5	194.6	196.1	196.1 (1.5)
24	70	0.85	5	0.85	4.17	153.1	149.5	153.4	152.0 (2.2)
20	90	0.85	0	0.71	-0.5	225.7	222.0	223.5	223.7 (1.9)
20	80	0.85	0	0.76	-0.51	170.3	161.3	168.9	166.9 (4.9)
20	70	0.85	0	0.69	-0.53	133.6	130.6	132.1	132.1 (1.5)

20	90	0.85	5	0.85	4.25	182.1	172.3	175.8	176.7 (5.0)
20	80	0.85	5	0.85	4.24	138.4	123.0	130.7	130.7 (7.7)
20	70	0.85	5	0.85	4.35	78.6	75.7	77.1	77.1 (1.5)
16	90	0.85	0	0.83	-0.46	141.0	138.5	140.3	139.9 (1.3)
16	80	0.85	0	0.73	-0.61	109.6	100.1	102.2	104.0 (5.0)
16	70	0.85	0	0.83	-0.48	83.3	78.1	80.3	80.6 (2.6)
16	90	0.85	5	0.85	4.45	108.7	104.5	106.4	106.5 (2.1)
16	80	0.85	5	0.84	4.43	61.9	52.1	56.8	56.9 (4.9)
16	70	0.85	5	0.84	4.5	19.8	15.8	17.8	17.8 (2.0)
24	90	1.05	0	1.04	-0.43	336.9	328.3	339.4	334.8 (5.8)
24	80	1.05	0	1.02	-0.79	275.9	274.3	275.1	275.1 (0.8)
24	70	1.05	0	1.05	-0.22	216.8	206.5	216.7	213.3 (5.9)
20	90	1.05	0	1.04	-0.13	220.1	211.9	219.4	217.1 (4.6)
20	80	1.05	0	1.05	-0.05	179.2	172.4	175.7	175.8 (3.4)
20	70	1.05	0	1.06	-0.38	137.5	134.5	136.2	136.1 (1.5)
24	90	1.25	0	1.2	-0.68	344.8	340.9	342.9	342.8 (2.0)
24	80	1.25	0	1.25	-0.62	268.5	263.1	264.5	265.4 (2.8)
24	70	1.25	0	1.26	-0.63	220.9	215.6	214.8	217.1 (3.3)
20	90	1.25	0	1.25	-0.1	217.9	214.0	214.4	215.5 (2.1)
20	80	1.25	0	1.25	-0.53	180.0	176.6	179.0	178.6 (1.7)
20	70	1.25	0	1.24	-0.25	128.5	120.9	123.3	124.2 (3.9)
24	90	1.45	0	1.32	-0.68	343.8	337.5	339.5	340.3 (3.2)
24	80	1.45	0	1.44	-0.44	266.2	261.4	258.5	262.0 (3.9)
24	70	1.45	0	1.43	-0.58	222.6	214.8	217.7	218.3 (3.9)
20	90	1.45	0	1.45	-0.25	226.5	214.0	222.4	220.9 (6.4)
20	80	1.45	0	1.45	-0.37	172.3	159.3	165.8	165.8 (6.5)
20	70	1.45	0	1.46	-0.4	138.3	131.4	135.7	135.1 (3.5)

Note: T_i is designed room temperature, in ℃; RH is designed room relative humidity, in %; FR is designed water flow rate, in L/s; T_w is designed water temperature, in ℃; FR_a is measured water flow rate, in L/s; T_{wa} is measured water temperature, in ℃; CR is condensation rate, in g/h.

5.1.2 Analysis of Water Temperature Effect on the Condensation Rate

The condensation rate results at two water flow rates (0.65 and 0.85 L/s) are shown in Figures 5.1 to 5.3. Each dot in the graphs indicates the average condensation rate of three replicates for each treatment. The standard deviation of three replicates was not displayed in the graphs, for the small range of standard deviation made the graphs unclear. As can be seen in these three Figures, the condensation rates were greater when using 0 ℃ water than when using 5 ℃ water.

As shown in Figure 5.1, at a temperature of 24 ℃, for each of the two water temperature levels, the flow rate of 0.85 L/s led to higher condensation rate than 0.65 L/s when room RH was around 80% and 90%; however, the differences of the condensation rate between using these two flow rates were negligible when RH was 70%.

As demonstrated in Figure 5.2, at a temperature of 20 ℃, for the same water temperature, condensation rates were almost the same for the two water flow rates at 70% RH in the room, however, condensation rates with 0.85 L/s were apparently higher than those with 0.65 L/s at 80% RH. At 90% RH, 0.85 L/s water flow rate at a water temperature of 0 ℃ resulted in much higher condensation rates than that of 0.65 L/s; however, this gap was almost negligible when using 5 ℃ water.

Figure 5.3 also shows that 0.85 L/s led to higher condensation rate than 0.65 L/s at 80% RH of 16 ℃; however, there was not much difference at 90% RH. At 70% RH, no evident difference existed between the condensation rates with two flow rates at a water temperature of 0 ℃, but when using 5 ℃ water, 0.85 L/s did cause a higher condensation rate than 0.65 L/s.

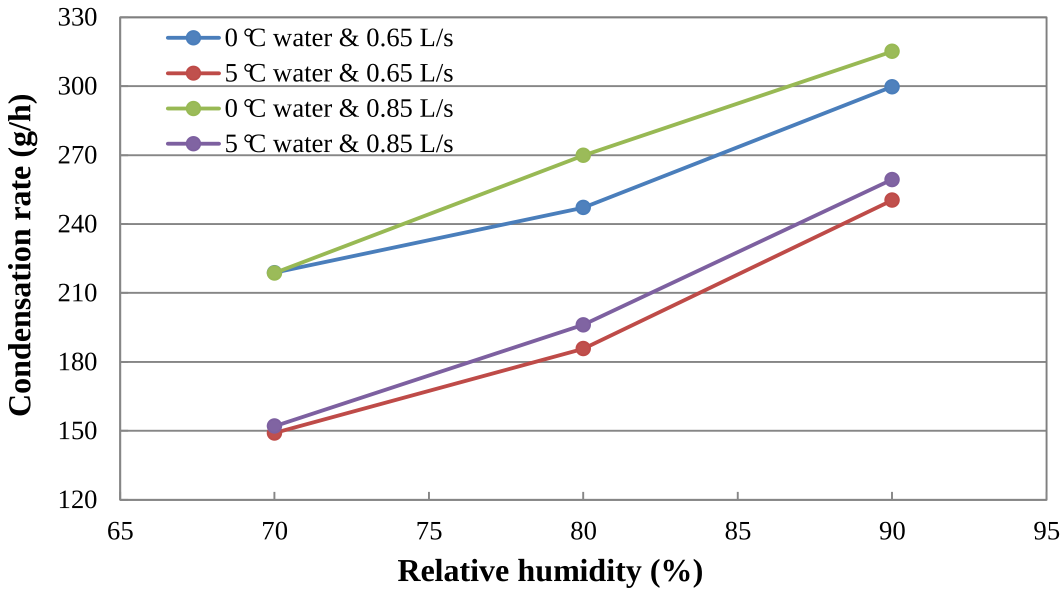

Figure 5.1 Condensation results at 24°C room temperature (90, 80, and 70% RH, 0 and 5°C water, 0.65 and 0.85 L/s)

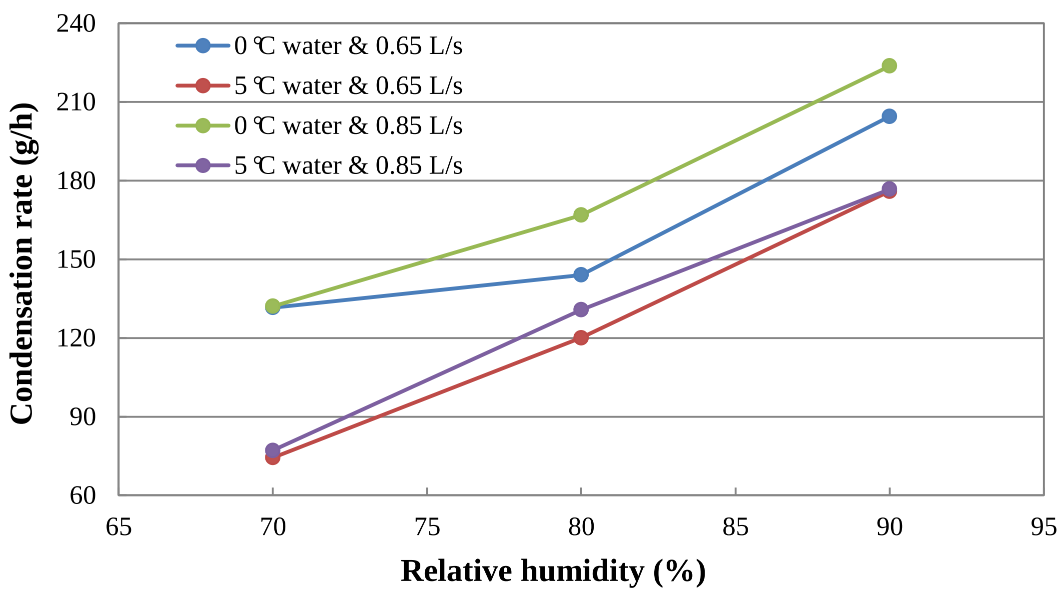

Figure 5.2 Condensation results at 20°C room temperature (90, 80, and 70% RH, 0 and 5°C water, 0.65 and 0.85 L/s)

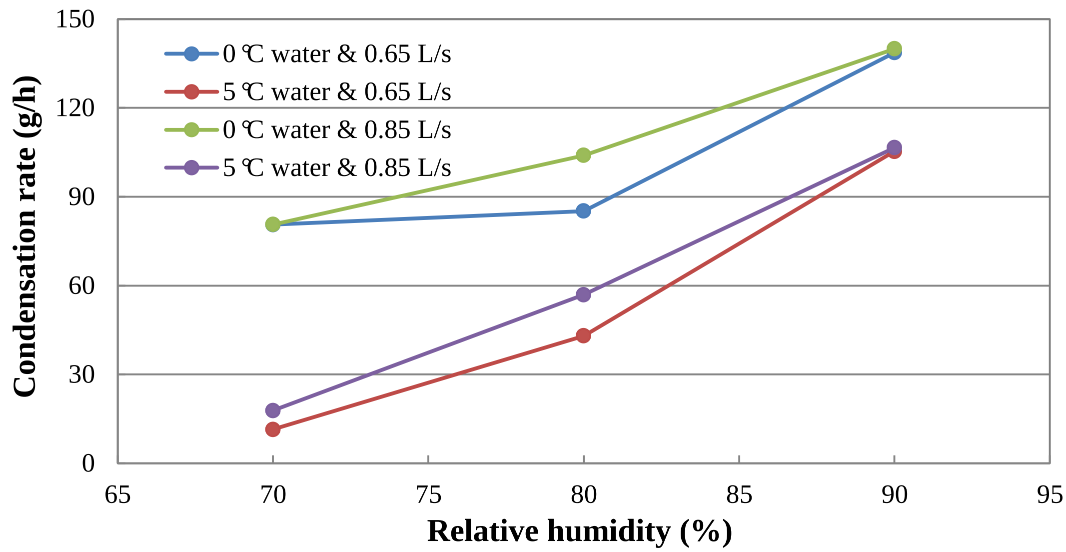

Figure 5.3 Condensation results at 16°C room temperature (90, 80, and 70% RH, 0 and 5°C water, 0.65 and 0.85 L/s)

Four-way ANOVA using SPSS 19.0 was applied to test the following null hypotheses: (1) Ti has no significant effect on CR; (2) RH has no significant effect on CR; (3) Tw has no significant effect on CR; (4) FR has no significant effect on CR. The data used included the results obtained from the following conditions: three levels of room temperature (24, 20, and 16 ℃), three levels of RH (90, 80, and 70%), two levels of water temperature (0 and 5 ℃), and two levels of water flow rate (0.65 and 0.85 L/s). The mean condensation rates with 0 ℃ and 5 ℃ water are shown in Table 5.2. On average, the hourly moisture removed by finned tubing using 0 ℃ water was 50.7 g higher than that achieved by using 5 ℃ water.

Table 5.2 Descriptive statistical results (24, 20 and 16°C room temperature, 90, 80, and 70% RH, 0 and 5°C water, 0.65 and 0.85 L/s)

Tw (℃)	**CR Mean (S.D.)**
0 (I)	177.8 (1.4)
5 (J)	127.1 (1.4)
Difference (I-J)	50.7 (1.9)

Table 5.3 gives the ANOVA table showing the results of the null hypotheses tests. The four factors all have significant effects on the condensation rate ($P < 0.05$ for each factor). Combined with the results drawn from Table 5.2, 0 ℃ water temperature led to significantly higher condensation rate than 5 ℃ within the setting range. Therefore, the experiments using 5 ℃ water were not conducted with three higher larger water flow rates (1.05, 1.25, and 1.45 L/s), for achieving a large condensation rate is one of the main goals of this experiment.

Table 5.3 Four-way ANOVA results (24, 20, and 16°C room temperature, 90, 80, and 70% RH, 0 and 5°C water, 0.65 and 0.85 L/s)

Source of variation	**SS**	**df**	**MS**	**F**	**P**
Ti (℃)	403348.6	2	201674.3	2020.0	0.002
RH (%)	140749.6	2	70374.8	704.9	0.001
FR (L/s)	2093.4	1	2093.4	21.0	0.018
Tw (℃)	69413.1	1	69413.1	695.3	0.001
Error	10083.6	101	99.8		
Total	625688.3	107	5847.6		

Note: SS is sum of squares, df is degrees of freedom and MS is mean squares.

5.1.3 Analysis of Water Flow Rate Effect on the Condensation Rate

The condensation results at five different flow rates when room temperature was 24 ℃ and 20 ℃ were plotted respectively in Figures 5.4 and 5.5. Both figures indicate that flow rate has little effect on the condensation rate when room RH was around 70%. This statement, however, does not hold true at 80% or 90% RH; the condensation rate increased greatly when the water flow rate increased from 0.65 L/s to 0.85 L/s, but not revealed obvious growth when it was higher than 0.85 L/s. At the 24 ℃ condition, the condensation rates were even reduced instead of increased when the flow rates were over 1.05 L/s at 80% RH, and it happened again when the flow rates were increased from 1.25 L/s to 1.45 L/s. These effects may be caused by measurement errors and require future research to confirm the results.

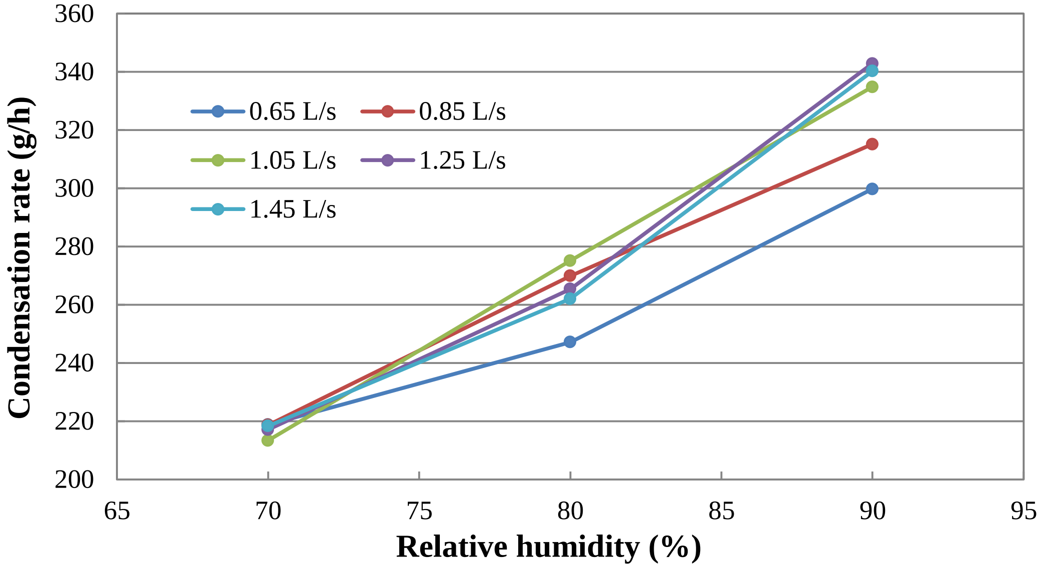

Figure 5.4 Condensation results at 24°C room temperature with 0°C water (90, 80, and 70% RH, 0.65, 0.85, 1.05, 1.25, and 1.45 L/s)

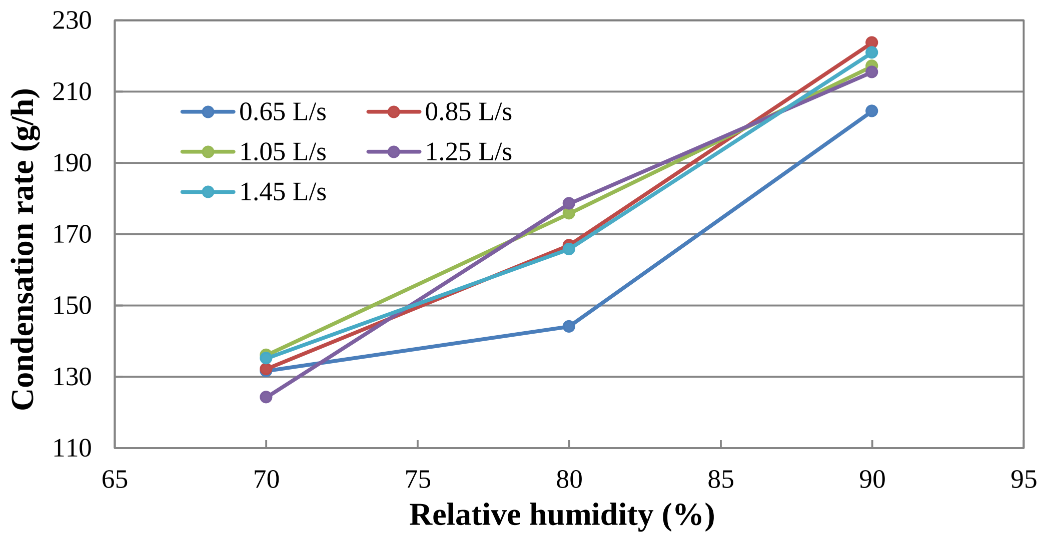

Figure 5.5 Condensation results at 20°C room temperature with 0°C water (90, 80, and 70% RH, 0.65, 0.85, 1.05, 1.25, and 1.45 L/s)

Three-way ANOVA using SPSS 19.0 was applied to test the following null hypotheses: (1) Ti has no significant effect on CR; (2) RH has no significant effect on CR; (3) FR has no significant effect on CR. The data used included the results obtained from the following conditions: two levels of room temperature (24 and 20 ℃), three levels of RH (90, 80, and 70%), one level of water temperature (0 ℃), and five levels of water flow rate (0.65, 0.85, 1.05, 1.25, and 1.45 L/s). Table 5.4 gives the ANOVA table showing the results of the null hypotheses tests. The three variables all have significant effects on the condensation rate when using 0 ℃ water under the tested conditions ($P < 0.05$).

Table 5.4 Three-way ANOVA results (24 and 20°C room temperature, 90, 80, and 70% RH, 0°C water, 0.65, 0.85, 1.05, 1.25, and 1.45 L/s)

Source of variation	**SS**	**df**	**MS**	**F**	**P**
Ti (℃)	215115.0	1	215115.0	2080.5	0.002
RH (%)	142181.5	2	71090.7	687.6	0.001
FR (L/s)	3806.1	4	951.5	9.2	0.006
Error	8478.5	82	103.4		
Total	369581.1	89			

Note: SS is sum of squares, df is degrees of freedom and MS is mean squares.

Tukey's HSD (Honestly Significant Difference) test was performed to find which flow rate was different from the others. Table 5.5 shows the multiple comparison results for five different flow rates, which concluded from the same data as those used for three-way ANOVA analysis. As can be seen in Table 5.5, there was significant increase in the condensation rate between the water flow rate of 0.65 L/s and the others ($P < 0.05$), but no significant increase between 0.85 L/s and the larger ones ($P > 0.05$). When the flow rate was increased from 1.05 to 1.25 L/s or 1.45 L/s, the condensation rate was even reduced by approximately 1.5 g/h, although the difference was not significant ($P > 0.05$). The average difference of the condensation rates between 1.25 and 1.45 L/s was only 0.2 g/h thus can be neglected ($P > 0.05$). In summary, the condensation rates increased significantly when the flow rate increased from 0.65 to 0.85 L/s ($P < 0.05$), but showed no significant change when the flow rate was greater than 0.85 L/s ($P > 0.05$).

Table 5.5 Multiple comparison results (24 and 20°C room temperature, 90, 80, and 70% RH, 0°C water, 0.65, 0.85, 1.05, 1.25, and 1.45 L/s)

(I) Designed FR (L/s)	(J) Designed FR (L/s)	(I-J) Mean Difference (g/h)	P
0.65	0.85	-13.4	.002
	1.05	-17.7	.001
	1.25	-16.3	.001
	1.45	-16.1	.001
0.85	1.05	-4.3	.709
	1.25	-2.9	.916
	1.45	-2.7	.931
1.05	1.25	1.5	.993
	1.45	1.6	.989
1.25	1.45	0.2	1.00

5.1.4 Finned Tubing Condensation Rate Statistical Modeling

The data of the room temperature of 16, 20, and 24 ℃, room RH of 70, 80, and 90%, water temperature of 0 and 5 ℃, and water flow rate of 0.65 and 0.85 L/s, were used to establish the first model (Model 1). The data of the room temperature of 20 and 24 ℃, room RH of 70, 80, and 90%, water temperature of 0 ℃, and water flow rate of 0.65, 0.85, 1.05, 1.25, and 1.45 L/s were used to develop the second model (Model 2). In the statistical models, T_i = "room temperature", RH = "room relatively humidity", FR = "water flow rate", T_w = "water temperature". Table 5.6 provides the SPSS output results for both models.

Table 5.6 SPSS results of condensation rate models

Independent Variable		r^2	Constant	Coefficients (P value)			
				Ti (℃)	RH (%)	Tw (℃)	FR (L/s)
Condensation rate (g/h)	Model 1	0.960	-516.145	19.084 (0.002)	3.851 (0.001)	-12.994 (0.001)	36.202 (0.031)
	Model 2	0.946	-649.966	25.692 (0.001)	4.087 (0.001)	N/A	14.865 (0.006)

Note: N/A means not available. Significant value α = 0.05.

As shown in Table 5.6, the r square for the prediction models are 0.960 and 0.946 for Model 1 and Model 2, respectively. In Model 1, the four factors were all proved to have significant effect on the condensation rate (P < 0.05). The coefficient values of Model 1 indicate that there exist positive correlations between the predictors and the condensation rate (except for water temperature, which is negatively correlated with it). According to the constant and coefficients of Model 1 given in Table 5.6, Model 1 can be written as:

$$\mathrm{CR} = -516.145 + 19.084 \times \mathrm{T_i} + 3.851 \times \mathrm{RH} + 36.203 \times \mathrm{FR} - 12.994 \times \mathrm{T_w}$$

In Model 2, the three factors were also proved to have significant effect on the condensation rate (P < 0.05). The coefficient values of Model 2 indicate that there exist positive correlations between the predictors and the condensation rate. According to the constant and coefficients of Model 2 given in Table 5.6, Model 2 can be written as:

$$CR = -649.966 + 25.692 \times T_i + 4.087 \times RH + 14.865 \times FR$$

Both condensation rate models are only applicable to the same finned tubing as it was tested in the experiment (Model ASHDB, Trane Canada Corporation, Saskatoon, SK, Canada). Model 1 is applicable to the specific conditions: room temperature of 16 to 24 ℃, room RH of 70 to 90%, water temperature of 0 to 5 ℃, and water flow rate of 0.65 to 0.85 L/s. Model 2 is applicable to the following specific conditions: room temperature of 20 to 24 ℃, room RH of 70 to 90%, water temperature of 0 ℃, and water flow rate of 0.65 to 1.45 L/s.

5.1.5 Energy Efficiency Analysis of Finned Tubing Condensation System

The MRI and energy cost calculation methods for the finned tubing condensation system have been introduced in 4.1.5.3, and sample calculations showing the specific MRI and energy cost calculation procedures for the finned tubing are given in Appendix C. The energy cost estimations were based on the assumption that the thermal energy was provided by the fuel used in the greenhouse. Since natural gas and thermal coal are the two commonly used fuels to heat the greenhouses in Saskatchewan, the energy cost were estimated by these two fuels, respectively.

The price of the thermal coal was taken as \$0.06/kg (Natural Resources Canada, 2009). The heating value of the thermal coal was taken as 24 MJ/kg (Fisher, 2003). Because 1 kW-h is 3.6 MJ, then the energy density of thermal coal is 6.67 kW ·h/kg. The efficiency of the boiler was assumed to be 70%, so of the 6.67 kW-h of energy per kilogram of thermal coal, 70% of that, 4.67 kW-h/kg, will be turned into thermal energy. The natural gas price of \$6.81/GJ was used in the calculations (SaskEnergy, 2009), which equals to \$0.028/kW-h considering the utilization efficiency of 90% for natural gas.

The moisture removal index and energy cost were calculated for each treatment, and the top five energy efficient results -- as well as the poorest treatments -- are listed in Table 5.7. The lowest moisture removal index of the finned tubing system was 7.202 kW-h/L, and the corresponding energy cost was \$0.322/L if natural gas was the heat source or \$0.240/L if thermal coal was the heat source. Compared with the other three dehumidification methods (1.007 kW-h/L for heat exchanger, -0.629 kW-h/L for dehumidifiers, and 1.261 kW-h/L for exhaust fan) (Chapter 4), the finned tubing condensation system actually consumed much more energy and thus resulted in much higher energy cost. The highest moisture removal index of finned tubing was 198.348 kW-h/L, and the corresponding energy cost was \$8.736/L (\$6.427/L) if natural gas (thermal coal) was the heat source. The finned tubing condenser is the least energy efficient mainly because the room air lost heat to the chilled water; therefore, this method is highly energy intensive and costly, which is not a suitable method for greenhouse dehumidification during the cold period. In addition, the preliminary capital and maintenance cost estimations showed that the use of this method would be greatly over the budget, thus finned tubing condensation was discarded in the field test.

Table 5.7 Five most energy efficient treatments and five most energy intensive treatments of the finned tubing condensation system

Room T (℃)	Room RH (%)	Flow Rate (L/s)	Water T (℃)	MRI (kW-h/L)	Energy Cost (\$/L)	
					Natural gas	Thermal coal
24	90	0.65	0	7.202	0.322	0.240
24	80	0.65	0	8.375	0.381	0.286
24	90	0.85	0	8.757	0.360	0.253
24	70	0.65	0	8.948	0.416	0.317
24	80	0.85	0	8.970	0.385	0.279
16	70	0.65	5	198.348	8.736	6.427
16	70	0.85	5	157.153	6.436	4.500
16	80	0.65	5	52.598	2.315	1.702
16	80	0.85	5	47.455	1.965	1.386
20	70	1.45	0	44.410	1.797	1.246

5.2 Field Experiment

Since the tomato plants were removed and the greenhouse was shut down during the period of January 2 to March 3, 2009, the data of this period were invalid and thus excluded. The data from December 29, 05:50, 2008, to January 2, 05:50, 2009, June 16, 06:00, to June18, 05:50, June 23, 16:20, to June 24, 05:50, July 3, 06:00 to July 9, 05:50, July 15, 06:00 to July18, 05:50, and August 29, 06:00 to

August 31, 2009 were abnormal due to the data acquisition program updating and were removed. The CO_2 monitor was out of order owing to moisture buildup in the air inlet tubing from March 25 to May 1, 2009, so the invalid CO_2 data during this period was also discarded. Each experiment cycle was composed of eight days: three days for only heat exchangers running, three days for only dehumidifiers running, and the other two days for the producers' ventilation control settings: either air inlets and exhaust fans were shut down and sealed to prevent air infiltration in cold winter, or they were in operation during mild or warm weather. However, the two control days in each experiment cycle were eliminated from the experiment since June 17, 2009 under repeated requests of the producers because they felt it was too humid inside the greenhouse without the heat exchanger or mechanical refrigeration dehumidifier. Therefore, during June 17 to November 17, 2009, there were only two dehumidification treatments: heat exchangers and dehumidifiers.

5.2.1 Greenhouse Dehumidification Season Identification in Cold Regions

One purpose of the diurnal RH monitoring was to measure the high RH occurrence in each month of the year in order to identify the dehumidification requirement in this cold region. The measurement was conducted from November 18, 2008, to November 17, 2009, for a year, and the monthly 75% RH exceeding percentages are shown in Table 5.8.

The monthly 75% RH exceeding percentages from April to October were all above 30%, and from July to October, the percentages were even above 60% although the temperature based ventilation treatment was eliminated. In November of 2008, the percentage was around 25%; however, the percentage in November of 2009 was above 50%. The reason was that more humid outside weather prevailed in November of 2009 than in November of 2008. From the end of December 2008 to the early March of 2009, the tomato plants were removed and the greenhouse was shut down without supplemental heating, resulting in the low RH inside the greenhouse. In early March, the young tomato crops were planted with low water demands and low transpiration of plants, causing low RH. In summary, the inside

RH was kept at a high level from April to November, and this period was thus determined as the dehumidification season for greenhouse tomato production in cold regions.

Table 5.8 also allows to be compared the effectiveness of different treatments during each month. It is obvious that the greenhouse was experiencing a higher RH condition when using their traditional methods than when using heat exchangers and dehumidifiers. Heat exchanger treatment was most effective to keep the RH at a suitable level with the lowest 75% RH exceeding percentage, which was approximately 20% lower than that when applying the conventional control ways and nearly 10% lower than that when using dehumidifiers except in July of 2009. These results occur because in July of 2009, the outside average RH was 70.9%, and the maximum outside RH reached as high as 100%. The heat exchangers were unable to keep the inside RH at the designed level by replacing internal air with humid outside air.

The ineffectiveness of heat exchanger treatment during most of the year, which allowed for above 50% exceeding occurrence of 75% RH, may be caused by two reasons: a) the large heat exchanger's setpoint was 80% instead of 75%, so the using of exceedance of 75% RH to compare the effects of treatments did not reflect the full capacity of the heat exchangers; b) Prince Albert experienced a very humid weather time since June of 2009. The heat exchanger controlled the RH much better in November and December of 2008 with nearly 40% high RH occurrence lower than that of 2009; therefore, humid weather was considered as the main reason resulting in the ineffectiveness of heat exchanger treatment. The dehumidifiers did even worse with the higher RH occurrence, which may be caused by the capacity of the two domestic dehumidifiers being (30.8 L per day for each) insufficient; they were not robust for greenhouse use. A durable commercial dehumidifier with high efficiency and high capacity should be applied and evaluated in the future research.

In addition, as can be seen in Table 5.8, the night 75% RH exceeding percentages were much greater than daytime measurements for the heat exchanger treatment from May to September, approximately 10% to 50% higher. This happened for dehumidifier and the control treatments as well from April to September of 2009. Such an effect may be caused by the lower ambient temperature at night and the low ventilation rate of the greenhouse, resulting in the higher RH. Thus, it is more difficult to dehumidify the greenhouse air at night in the mild and warm period of the cold regions.

Table 5.8 Summary of monthly percentage of RH≧75%

Month			Nov. 08	Dec. 08	Jan. 09	Feb. 09	Mar. 09	Apr. 09	May 09	Jun. 09	Jul. 09	Aug. 09	Sept. 09	Oct. 09	Nov. 09
Percentage that RH≧75% for each treatment (%)	Heat exchanger	Day	45.8	2.1	0	0	4.0	32.4	36.2	45.1	53.5	55.6	56.2	91.2	79.3
		Night	0.4	0	0	0	1.5	14.8	44.5	85.2	100	94.6	62.8	30.4	18.7
		Overall	16.7	0.6	0	0	0.3	25.5	38.9	56.9	68.0	70.5	59.3	57.1	41.2
	Dehumidifier	Day	64.5	12.3	0	0	2.4	30.6	37.3	47.8	48.1	61.6	50.5	93.9	89.5
		Night	7.5	0.9	0	0	0.9	34.7	83.7	93.2	100	100	74.1	59.2	53.6
		Overall	28.7	4.5	0	0	1.6	32.3	53.2	60.7	64.9	76.4	61.5	74.3	66.9
	Control	Day	73.1	28.2	0	0	0	38.0	53.6	52.1	N/A	N/A	N/A	N/A	N/A
		Night	27.7	0	0	0	0	48.0	84.2	98.2					
		Overall	43.9	9.2	0	0	0	42.0	63.7	65.7					
Percentage that RH≧75% for each month (%)			25.5	4.7	0	0	1.8	32.5	50.9	59.8	66.8	73.6	60.4	65.5	54.0

Note: Day/night percentage was calculated by dividing the high RH occurrence time by the total period of daytime/nighttime, and the total percentage for each treatment/month was calculated by dividing the high RH occurrence time by the total treatment period/monthly period.

5.2.2 Greenhouse Dehumidification Effect Evaluation

5.2.2.1 Greenhouse Environment Profiles under Cold Weather Condition

The data of cold weather covered the period of November 18, 2008, to March 28, 2009, and October 30 to November 17, 2009 with the exception of the eliminated data as previously stated. As presented before, from November 18, 2008, to March 28, 2009, each experimental cycle was composed of eight days including two control days; however, the October and November 2009 experiments did not have control treatment because the control treatment was eliminated since June 17, 2009.

A total of 11 cycles of data were obtained and analyzed. The basic outdoor and indoor environmental conditions of each treatment are summarized in Table 5.9.

Table 5.9 Summary of climatic parameters under cold weather

Control option (Date)	Treatment	Ambient environment		CO_2		Inside T		Inside RH		Percentage (Inside RH ≥ 75%) (%)
		Outside T (℃)	Outside RH (%)	Daytime average (ppm)	Min (ppm)	Night average (℃)	Daytime average (℃)	Night average (%)	Daytime average (%)	
Control: Fan/inlet Sealed (Nov.18-Dec.29, 2008, 5 cycles)	Heat exchanger (Std. Dev.)	-13.6 (7.9)	78.1 (9.1)	319.6 (81.7)	188.1	16.7 (1.8)	19.7 (2.7)	61.5 (6.4)	70.3 (5.4)	6.1
	Dehumidifier (Std. Dev.)	-11.9 (9.3)	74.9 (8.0)	322.2 (80.5)	185.3	17.3 (1.3)	20.3 (1.7)	64.0 (6.0)	71.2 (5.9)	10.6
	Control (Std. Dev.)	-11.2 (7.7)	74.3 (9.9)	326.9 (88.9)	162.2	17.0 (1.4)	20.0 (2.0)	64.0 (6.9)	71.8 (5.8)	13.3
Control: Fan/inlet in operation (Mar.4-28, 2009, 3 cycles)	Heat exchanger (Std. Dev.)	-9.0 (8.0)	74.6 (11.4)	311.7 (53.7)	206.1	18.2 (0.7)	22.3 (2.0)	61.9 (9.8)	64.4 (8.3)	5.3
	Dehumidifier (Std. Dev.)	-14.7 (7.9)	61.3 (12.6)	285.8 (59.1)	154.2	17.1 (1.7)	23.3 (3.6)	57.8 (7.4)	60.5 (6.5)	0
	Control (Std. Dev.)	-14.2 (10.3)	65.7 (12.6)	305.4 (40.6)	239.1	17.3 (1.3)	22.6 (3.0)	60.9 (11.7)	60.2 (9.0)	0
Control: Eliminated (Oct.30-Nov.17, 2009, 3 cycles)	Heat exchanger (Std. Dev.)	0.0 (4.9)	70.6 (15.0)	267.0 (55.0)	186.1	18.1 (0.5)	22.2 (2.0)	73.6 (1.9)	77.5 (2.2)	40.9
	Dehumidifier (Std. Dev.)	0.5 (4.6)	66.6 (14.6)	264.0 (76.1)	154.5	18.3 (0.7)	22.9 (2.1)	75.7 (2.1)	78.8 (2.5)	68.2

Note: The data used was collected at 10-minute intervals.

The ambient temperature of cold weather condition ranged from -36.8 ℃ (March 11, 2009) to 14.5 ℃ (November 6, 2009). March 2009 was a cold month and November 2009 was a warm one. The indoor temperature ranged from 8.6 ℃ to 31.4 ℃; the average daytime temperature for each treatment was around 22.0 ℃ (setpoint temperature of daytime was 22 ℃) and the night average was approximately 17.5 ℃ (setpoint temperature of nighttime was 18 ℃). The lowest indoor temperature of 8.6 ℃ occurred on December 14 when the outdoor temperature was -34.5 ℃. The heating system kept a temperature of 14.9 ℃ at the ambient temperature of -36.4 ℃ on December 22, 2008, and 13.5 ℃ at the lowest outdoor temperature of -36.8 ℃, March 11, 2009. These results indicated that a) the heating capacity of the greenhouse was not enough to keep the room temperature at the setpoint of 18 ℃, and b) something was wrong with the heating system on December 14 because the system should be able to maintain a much higher temperature than 8.6 ℃.

During November 18 to December 29, 2008, the heat exchanger treatment reduced the indoor RH with 6.1% exceeding percentage of 75% RH, while the dehumidifier treatment managed 10.6% exceeding percentage, and the control had the highest value of 13.3%. Daytime RH was higher than night RH in the cold season. The daytime average CO_2 concentration was around 320 ppm, but the control treatment had the lowest value of 162 ppm. The heat exchanger treatment had the advantage of supplying CO_2 to the plants, as compared to the dehumidifiers and control when the fans and inlets were sealed. During March 2009, sometimes the weather was extremely cold, but the exhaust fans and air supply inlets were unsealed and in operation if needed. The RH under the dehumidifier and control treatments never exceeded 75%. However, there were a few days during the heat exchanger treatment that the ambient temperature was getting close to zero, which resulted in higher average ambient temperature and relative humidity than the other two treatments. As a result, the indoor RH exceeded 75% for 5.3% of the time, as well as displaying a higher CO_2 concentration from more ventilation by the heat exchangers. Both October and November 2009 were rather mild and should have been analyzed in the mild weather data. However, by season, these months should belong to the cold season; therefore, the records were still kept in this group of data. During this period, the heat exchanger treatment controlled the humidity much better than the dehumidifier treatment (40.9% vs. 68.2% of 75% RH exceeding percentage), and it also maintained a higher CO_2 concentration than the dehumidifier treatment.

In order to easily compare the environmental conditions with different treatments in the cold weather period, the profiles of environmental parameters in a typical cycle from November 26 to December 4, 2008, were plotted as shown in Figure 5.6 to Figure 5.8. Figure 5.6 illustrates the basic environmental parameters profile in heat exchanger running days from 6:00 am, November 26, to 6:00 am, November 29, 2008; Figure 5.7 shows the profile in dehumidifier running days from 6:00 am, November 29, to 6:00 am December 2, 2008, and Figure 5.8 displays the profile in control days from 6:00 am, December 2, to 6:00 am, December 4, 2008. As discussed earlier, the small heat exchanger (RA400) was set to run when the inside RH was above 75% and to stop when RH fell below 73%. The big heat exchanger (RA1000) should have been running when the inside RH was greater than 80% until the RH decreased to 78%. In heat exchanger running days, it can be seen that only the small heat exchanger (RA400) was running when the indoor RH was above 75%. The big heat exchanger (RA1000) was never turned on because the RH never exceeded 80%. In dehumidifiers' running days, both dehumidifiers were set to be turned on when the indoor RH exceeded 75% and turned off when RH reduced to 73%. Figure 5.7 shows that the dehumidifiers were both controlled very well based on the indoor RH setpoint. In control days, the indoor temperature ranged from 18 ℃ to 22 ℃, and the average indoor RH was around 70%.

Overall, the indoor RH was kept at an acceptable level with heat exchangers during the cold period, and the dehumidifiers worked more frequently to maintain a proper RH than did heat exchangers. The inside RH appeared not to become a big problem under cold weather conditions in the graph; however, referring back to Table 5.8, the 75% RH exceeding percentage with control methods in November of

2008 was 43.9%, and it was much greater than that with heat exchangers and dehumidifiers, 16.7% and 28.7%, respectively. This result agrees with the results obtained from Table 5.9 -- that heat exchangers controlled the RH better than the dehumidifiers, and that the temperature based ventilation method was unable to maintain a suitable RH level in the cold period.

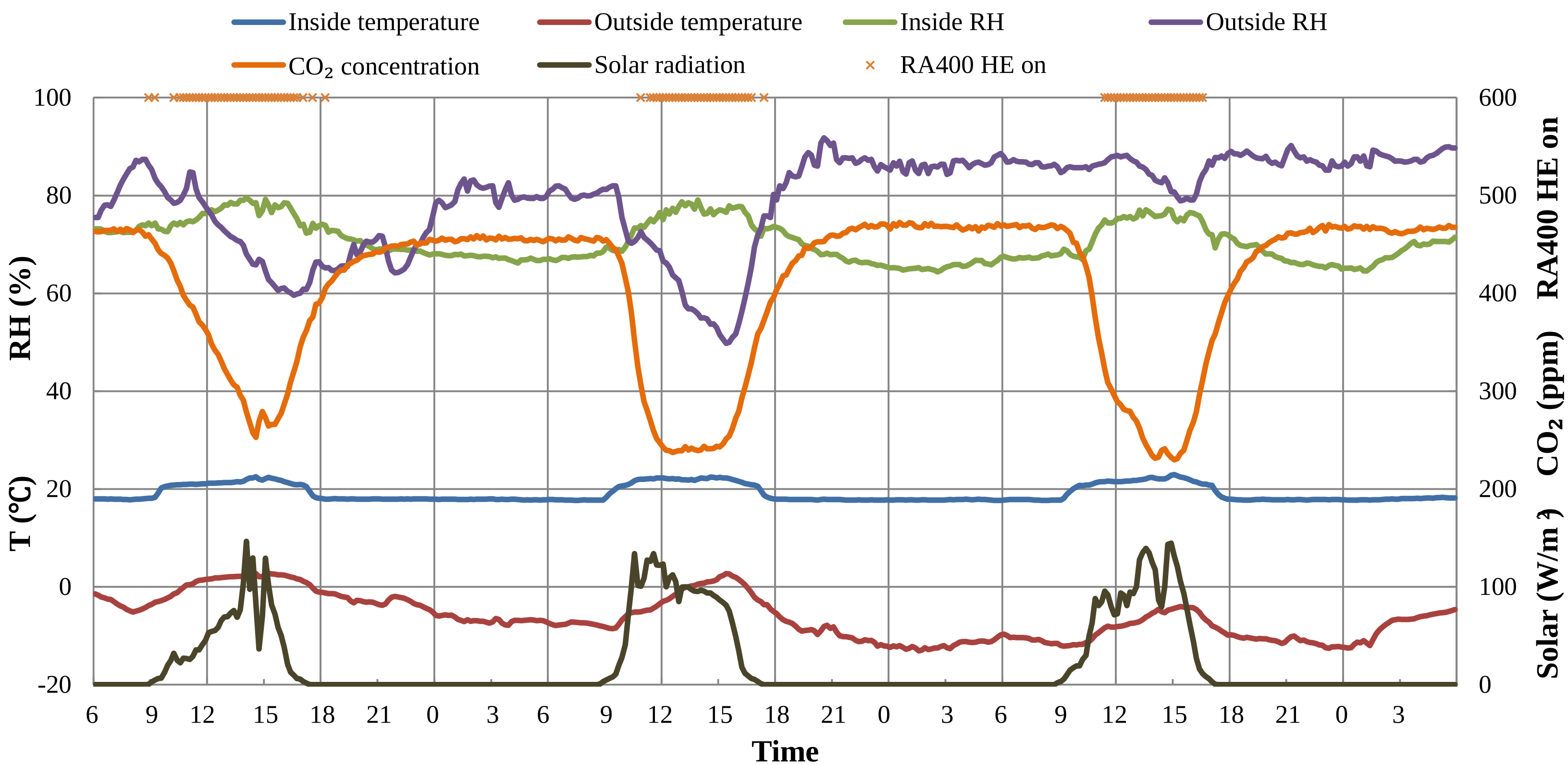

Figure 5.6 Environmental conditions during November 26 to 29, 2008 (heat exchanger treatment)

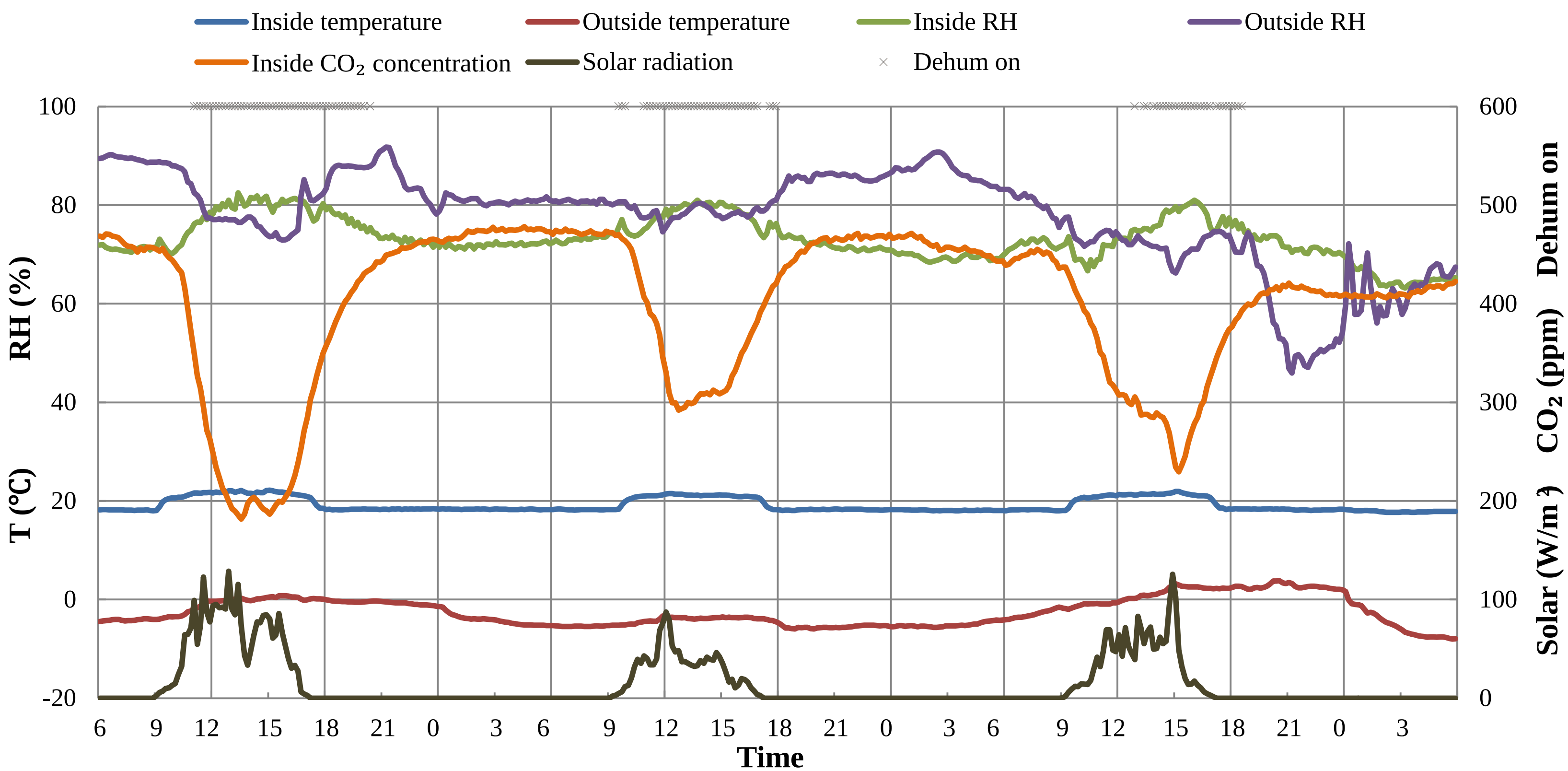

Figure 5.7 Environmental conditions during November 29 to December 2, 2008 (dehumidifier treatment)

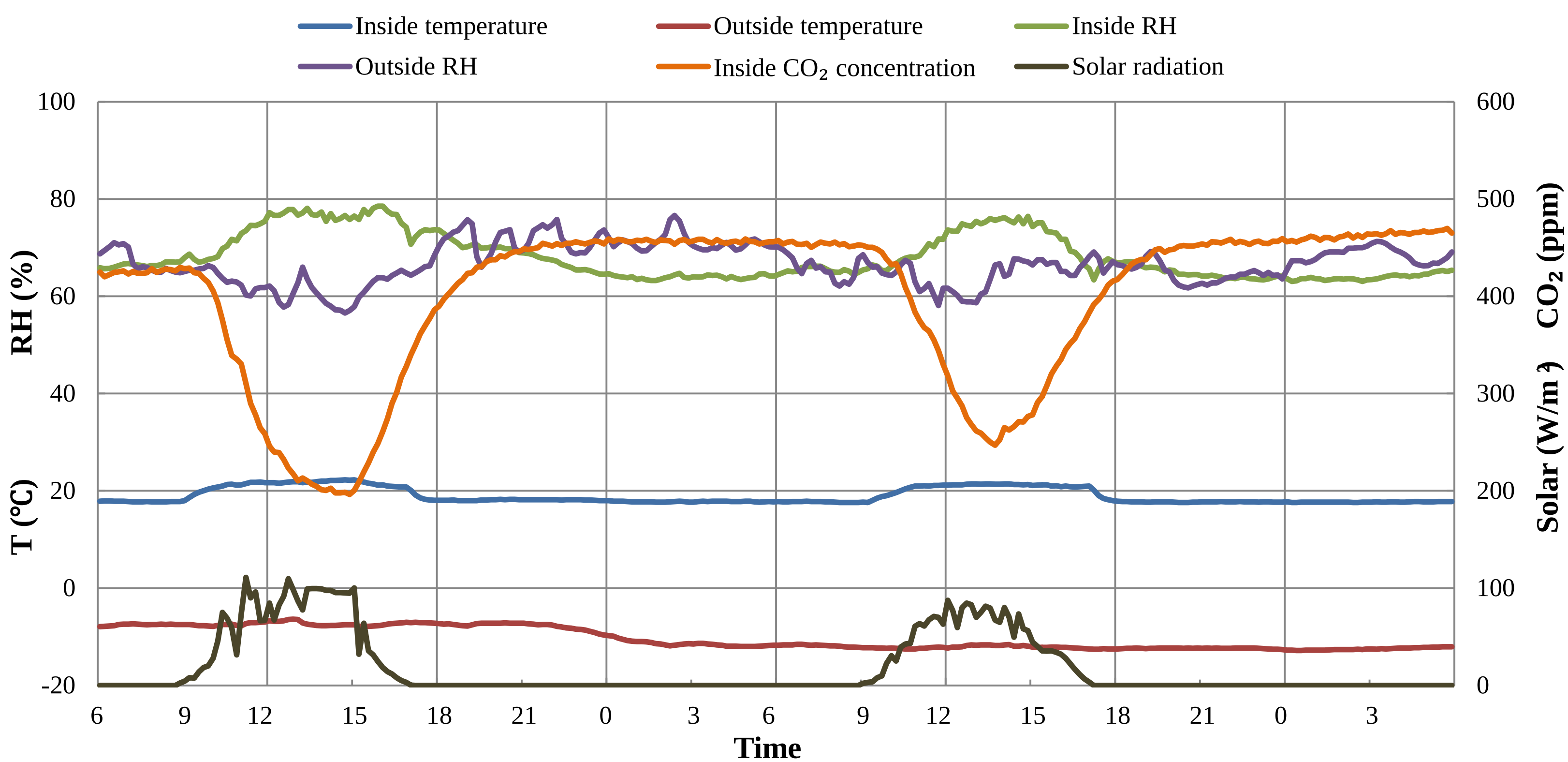

Figure 5.8 Environmental conditions during December 2 to 4, 2008 (control)

5.2.2.2 Greenhouse Environment Profiles under Mild Weather Condition

There are 13 cycles for mild weather data, from March 28 to May 31, 2009, and September 30 to October 30, 2009. The basic environmental conditions of each treatment are summarized in Table 5.10. The ambient temperature of the mild weather periods ranged from -13.4 ℃ (7:00 am, March 29, 2009) to 24.8 ℃ (May 30, 2009). The inside temperature ranged from 12.8 to 29.3 ℃; the average daytime temperature was between 22 and 23.6 ℃, and the night average was between 18.2 and 20 ℃.

The general observation is that the night RH was higher than the daytime RH except during October 2009 when the average outside temperature fell to 2.4 ℃ or lower, and the daytime ventilation rate reduced caused high RH in the greenhouse. This is opposite to that of the cold season when the daytime RH was higher than nighttime RH due to consistently low ventilation rates most of the time but higher moisture production in the greenhouse by transpiration during the daytime.

From the observations during March 28 to May 31, the heat exchanger treatment resulted in a drier environment with a 75% RH exceeding percentage of 30.1%, comparing with 44.8% for the dehumidifier treatment and 52.9% for the control treatment. During September 30 to October 30, warm and humid ambient air resulted in higher RH in the greenhouse, and heat exchanger treatment also led to a drier environment than that of the dehumidifier treatment. With the high 75% RH exceeding percentage for both treatments, the result indicated that when outside air was warm and humid, neither treatment was able to keep the RH at or below the setpoint.

The main reason is that the capacity of the heat exchangers and the dehumidifiers was not enough to meet the moisture removal requirement of the greenhouse. Heat exchangers were ineffective in removing moisture from the greenhouse when the ambient air was humid; besides, it was undesirable that the heat exchangers still heated up the incoming air when the ambient air temperature was already high. The dehumidifiers were still effective in removing moisture out of the air in the greenhouse; in fact, they were effective all year round. Dehumidifiers also released heat to the greenhouse, which was undesirable during summer. Considering that the dehumidification is mostly needed at nighttime when ambient temperature is low and that heating is most likely required during mild season, both heat exchanger and dehumidifier operation will be desirable for achieving dehumidification and heating purposes.

Table 5.10 Summary of climatic parameters under mild weather

Date	Treatment	Ambient environment		CO_2		Inside T		Inside RH		Percentage (Inside RH ≥ 75%) (%)
		Outside T (℃)	Outside RH (%)	Daytime average (ppm)	Min (ppm)	Night average (℃)	Daytime average (℃)	Night average (%)	Daytime average (%)	
Ventilated with Control Days (Mar.28-May 31, 2009, 8 cycles)	Heat exchanger (Std. Dev.)	4.2 (6.5)	57.7 (18.0)	301.9 (56.2)	161.6	18.2 (1.8)	23.1 (2.8)	74.3 (2.4)	70.8 (10.4)	30.1
	Dehumidifier (Std. Dev.)	4.0 (6.1)	60.9 (21.6)	315.1 (73.0)	158.7	19.3 (1.9)	23.6 (2.6)	75.8 (2.4)	71.8 (11.5)	44.8
	Control (Std. Dev.)	5.4 (7.0)	57.1 (23.0)	318.7 (94.5)	169.7	18.7 (2.0)	23.4 (2.7)	76.4 (3.5)	70.0 (13.0)	52.9
Ventilated without Control Days (Sept.30-Oct.30, 2009, 5 cycles)	Heat exchanger (Std. Dev.)	0.7 (3.7)	80.4 (11.3)	288.8 (51.7)	161.6	18.5 (2.0)	22.0 (2.3)	74.7 (2.0)	78.3 (2.8)	57.4
	Dehumidifier (Std. Dev.)	2.4 (4.0)	81.6 (11.3)	300.5 (64.2)	158.7	20.0 (1.8)	23.0 (1.9)	75.7 (2.4)	79.7 (3.8)	73.1

Note: The data used was collected at 10-minute intervals.

The profiles of environmental parameters in a typical cycle of mild weather are shown in Figure 5.9 to Figure 5.11. Figure 5.9 illustrates the basic climatic parameter profile during heat exchanger running days from 6:00 am, April 13, to 6:00 am, April 16, 2009; Figure 5.10 shows the profile during dehumidifier running days from 6:00 am, April 16, to 6:00 am, April 19, 2009, and Figure 5.11 displays the profile in control days from 6:00 am, April 19, to 6:00 am, April 21, 2009.

For all three treatments, the RH rose and peaked during 9 am to 12 pm during the lower morning temperature and low ventilation rate but high plant transpiration rate, then it reduced because of higher ventilation rates caused by high room temperature. Thus, the highest dehumidification requirement occurred during 9 am to 12 pm. During nighttime, although RH appeared high, the ventilation rate was low and moisture production was low, thus, the dehumidification requirement should be lower than that of the morning period.

Figure 5.9 shows that the inside RH sometimes was greater than 80%, causing the large heat exchanger to be running occasionally. In dehumidifiers' running days, it can be observed that the dehumidifiers both worked well with the setpoint of 75%. In control days, the indoor temperature was between 18 and 22 ℃, and the average indoor RH was sometimes exceeded 80%. By comparison with the RH percentage results of April 2009 from Table 5.8, the heat exchangers still controlled the RH better than the other two treatments; as well, dehumidifiers were less competitive based on the RH control accuracy due to their insufficient moisture removal capacity.

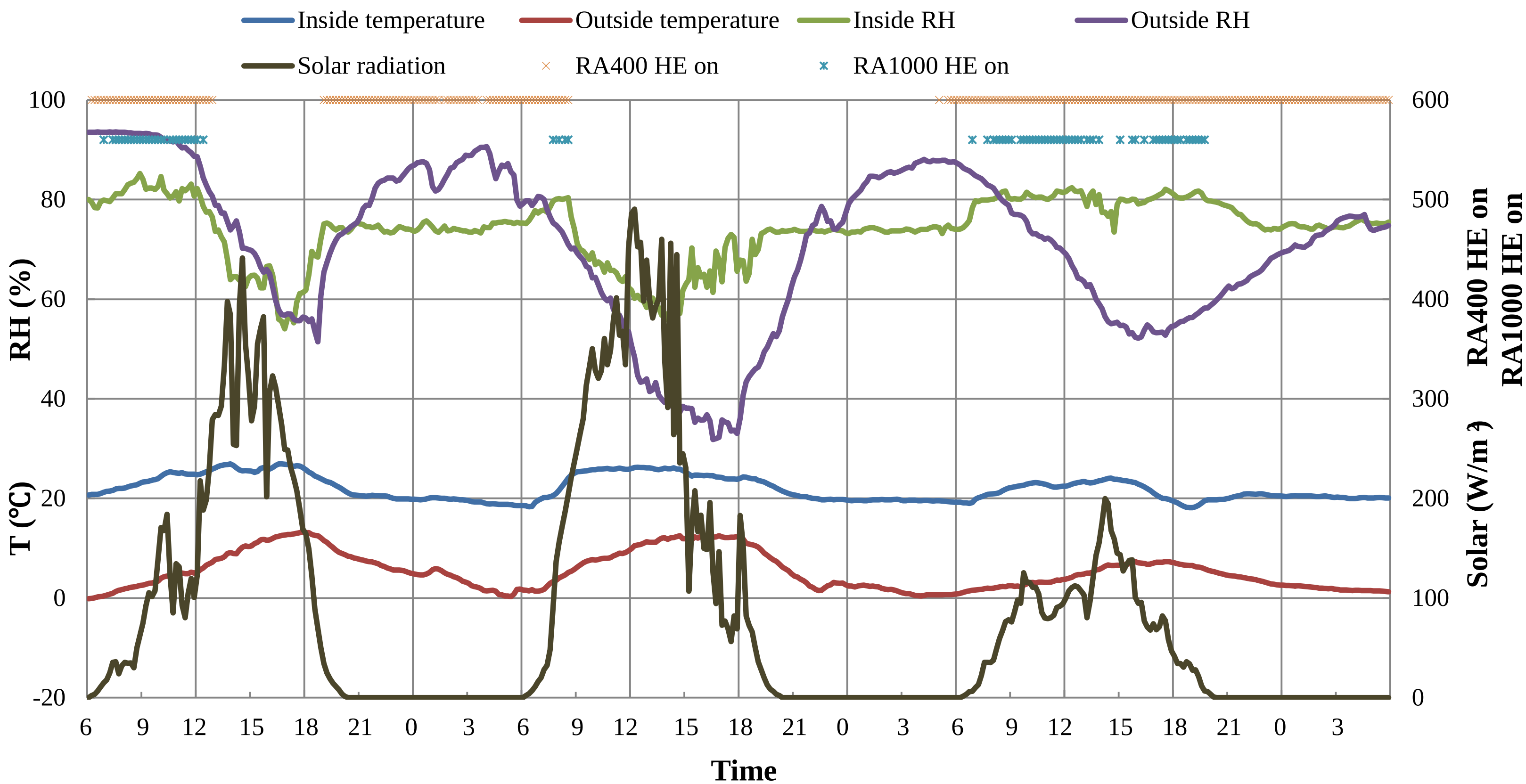

Figure 5.9 Environmental conditions during April 13 to 16, 2009 (heat exchanger treatment)

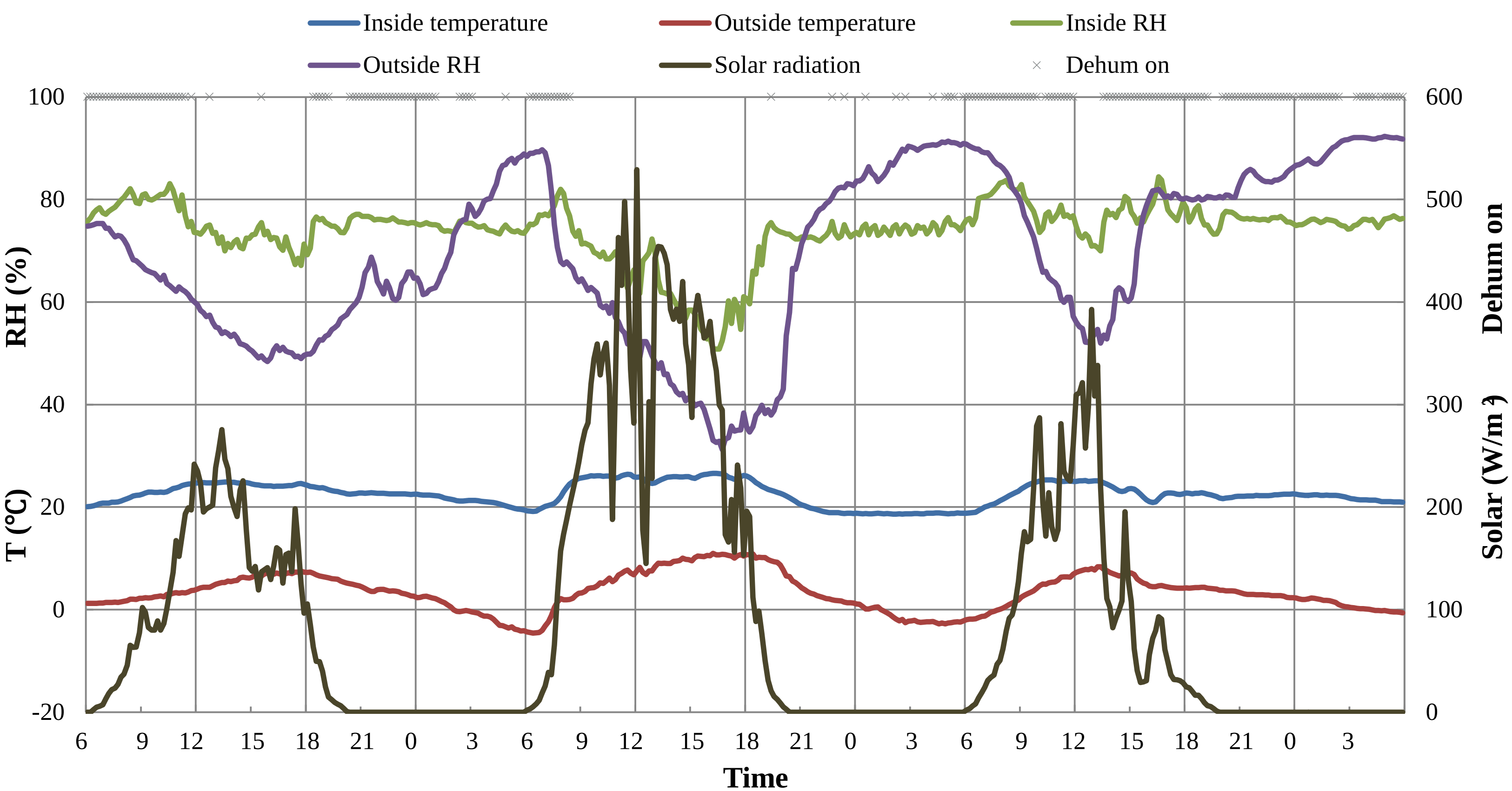

Figure 5.10 Environmental conditions during April 16 to 19, 2009 (dehumidifier treatment)

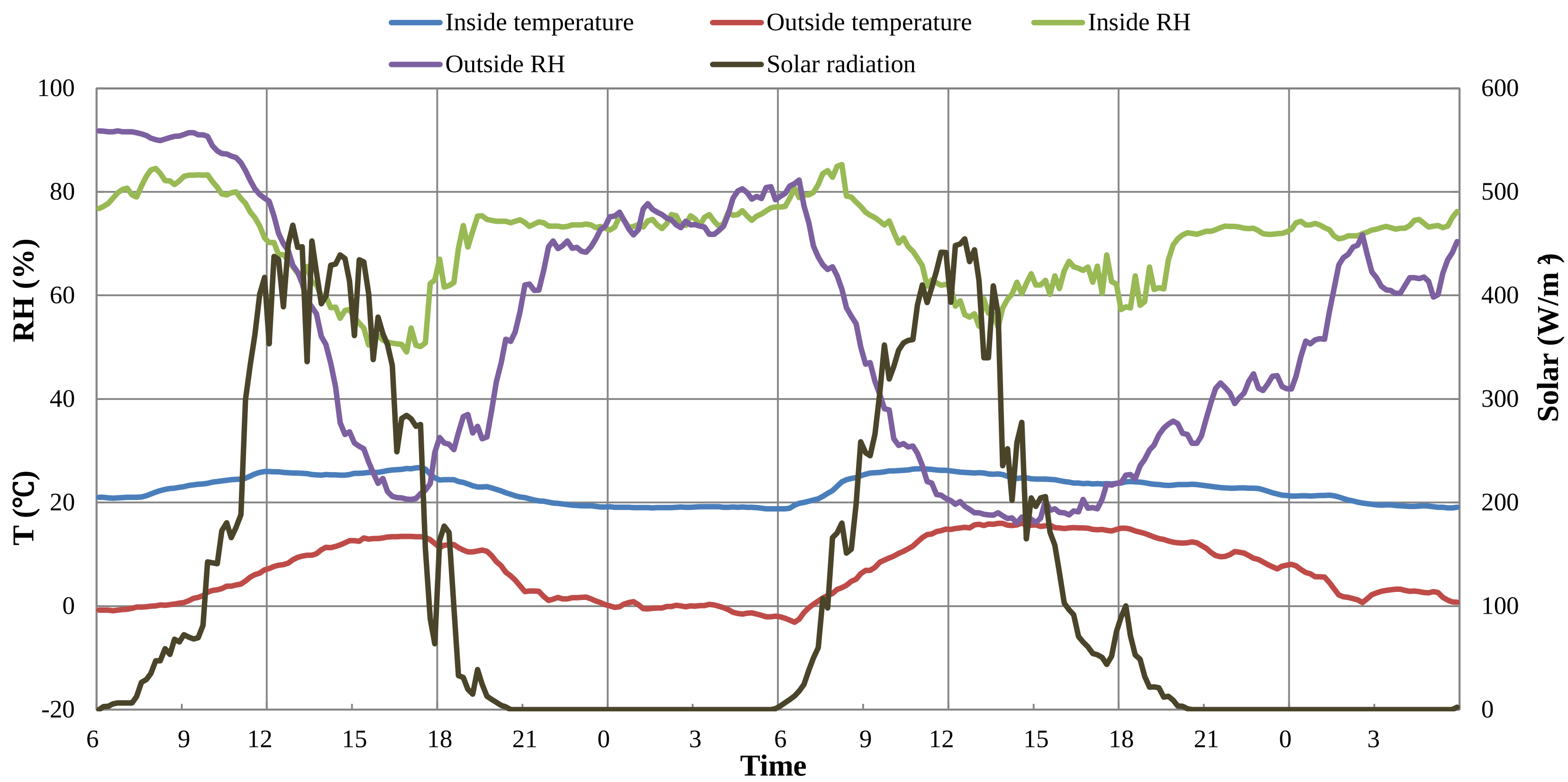

Figure 5.11 Environmental conditions during April 19 to 21, 2009 (control)

5.2.2.3 Greenhouse Environment Profiles under Warm Weather Condition

The data in warm weather covered the period from May 31 to September 30, 2009. The basic environmental conditions with each treatment for warm weather are shown in Table 5.11. The ambient temperature under warm weather condition ranged from -2.4 to 31.6 ℃, and the outside RH could reach 100% because of frequent rains during this period. The inside temperature was between 12 and 31.1 ℃. The average daytime temperature for each treatment was between 23.1 and 24.2 ℃, and the night average was between 17.5 and 18.9 ℃. During the heat exchangers' running days, the inside RH reached as high as 94.7% and the daytime average CO_2 concentration was around 312 ppm. For dehumidifiers' running days, the highest RH was also over 94%, and nearly 92% for control days. The dehumidifier treatment resulted in higher indoor temperatures than the other treatments, which could be caused by the heat output of dehumidifiers and the latent heat released from water vapour condensation as well. Again, the heat exchanger treatment resulted in a drier environment than that of the dehumidifier and control treatments.

Table 5.11 Summary of climatic parameters under warm weather

Date	Treatment	Ambient Environment		CO_2		Inside T		Inside RH		Percentage (Inside RH ≥ 75%) (%)
		Outside T (℃)	Outside RH (%)	Daytime Average (ppm)	Min (ppm)	Night Average (℃)	Daytime Average (℃)	Night Average (%)	Daytime Average (%)	
Ventilated with Control Days (May 31-Jun.16, 2009)	Heat Exchanger (Std. Dev.)	9.8 (5.5)	58.8 (24.3)	316.5 (58.4)	212.7	17.7 (1.0)	23.6 (2.8)	76.0 (2.2)	68.4 (13.4)	48.1
	Dehumidifier (Std. Dev.)	12.3 (7.7)	48.3 (18.5)	326.9 (80.2)	180.2	17.8 (1.1)	24.2 (3.7)	77.7 (2.0)	63.9 (17.6)	50.2
	Control (Std. Dev.)	14.1 (9.3)	64.0 (23.0)	347.6 (93.6)	227.8	18.1 (1.7)	24.1 (3.9)	81.8 (2.4)	68.5 (18.9)	66.7
Ventilated without Control Days (Jun.18-Sept.30, 2009)	Heat Exchanger (Std. Dev.)	16.0 (5.3)	69.3 (19.9)	311.3 (56.6)	175.6	17.5 (3.3)	23.1 (4.4)	83.1 (5.3)	72.7 (15.4)	69.2
	Dehumidifier (Std. Dev.)	15.5 (5.8)	71.8 (18.9)	313.9 (81.5)	129.9	18.9 (2.9)	23.7 (3.6)	83.4 (4.7)	74.1 (14.1)	73.1

Note: The data used was collected at 10-minute intervals.

The profiles of the climatic parameters in a typical cycle during warm weather are shown in Figure 5.12 to Figure 5.14. The inside RH profile was similar to mild weather profiles: high at nighttime, rose and peaked during 9 am to 12 pm, and low during the rest of the daytime as affected by low ventilation rates at night and morning, and high ventilation rates from noon to afternoon. Figure 5.12 illustrates the basic environment parameter profile in heat exchanger running days from 6:00 am, May 31, to 6:00 am, June 3, 2009; Figure 5.13 shows the profile in dehumidifiers' running days from 6:00 am, June 3, to 6:00 am, June 6, 2009, and Figure 5.14 displays the profile in control days from 6:00 am, June 6, to 6:00 am, June 8, 2009. It needs to be pointed out that the outside RH was high because of the rain in the heat exchanger treatment days, making the dehumidification more difficult. Both heat exchangers were running frequently, but the RH was still above 80% most of the time. In summary, the heat exchangers were controlled very well based on the RH setpoints, but they were less effective when the ambient condition was warm and humid. In dehumidifiers' running days, the outside air was extremely humid during the summer, and the dehumidifiers' capacity was not sufficient; so, the RH sometimes exceeded 80% or even 90%. In control days, the indoor RH was also greater than 80% most of the time. By analyzing the 75% RH exceeding percentage results of June 2009 summarized in Table 5.11, it can be seen that heat exchanger treatment still resulted in the lowest 75% RH exceeding percentage conditions in the greenhouse (48.1%). At the same time, the dehumidifiers had a slightly higher 75% RH exceeding percentage (50.2%), and the temperature based ventilation method had the highest exceeding percentage of 75% RH (66.7%).

In summary, the greenhouse without dehumidification measures (i.e., using only temperature based ventilation) has resulted in high RH most of the time during the summer; neither heat exchangers nor dehumidifiers had sufficient capacity to control the RH to the desired level. Heat exchangers are not intended for use in summer when the inside temperature is high because they will add some heat from the exhaust air back to greenhouse. However, considering the cool night and early morning temperatures, heat recovered or released by the heat exchanger or dehumidifier is desirable. Again, sizing the dehumidifier to meet the humidity control requirement is the key for successful humidity control in the greenhouse.

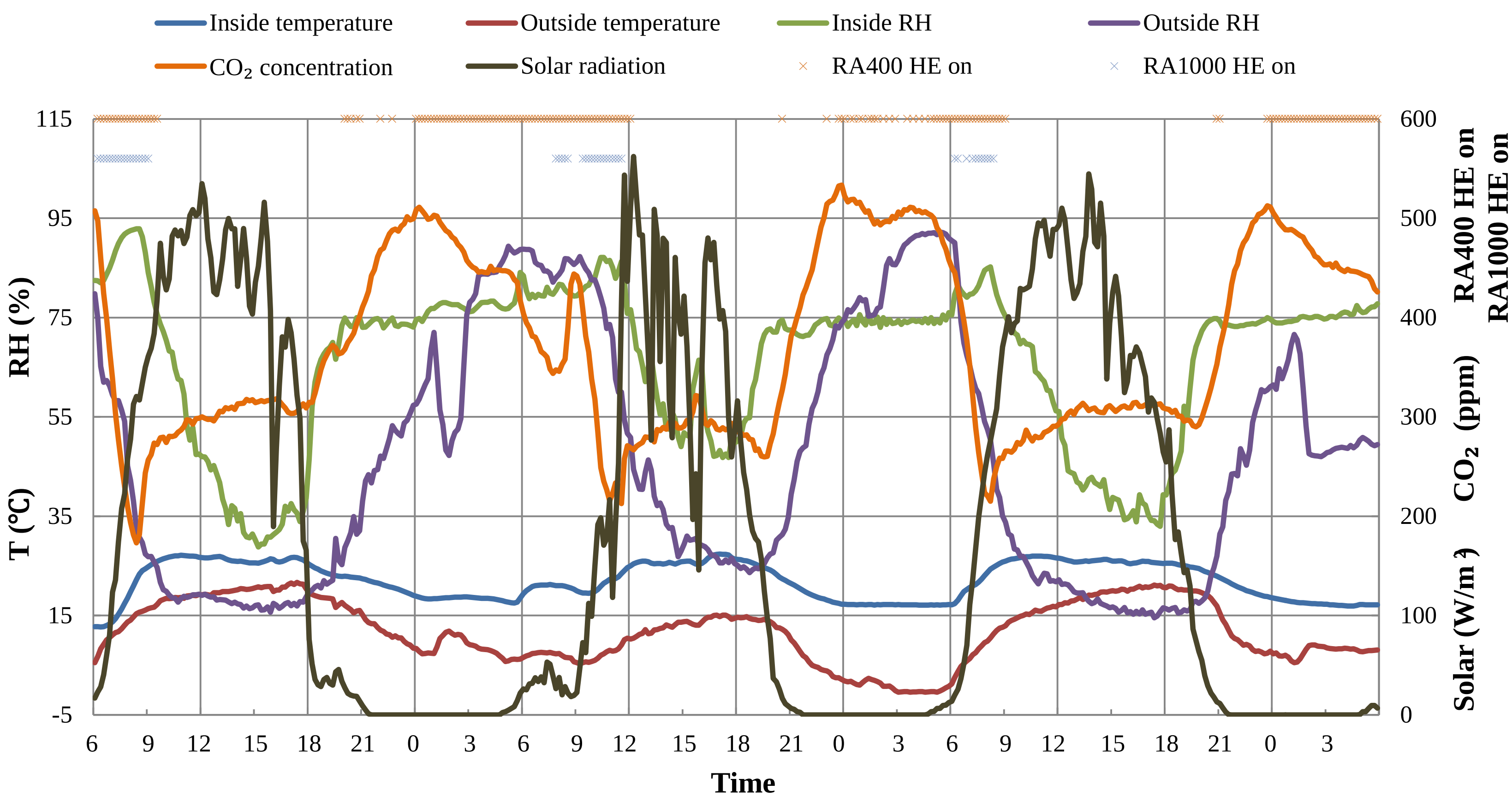

Figure 5.12 Environmental conditions during May 31 to June 3, 2009 (heat exchanger treatment)

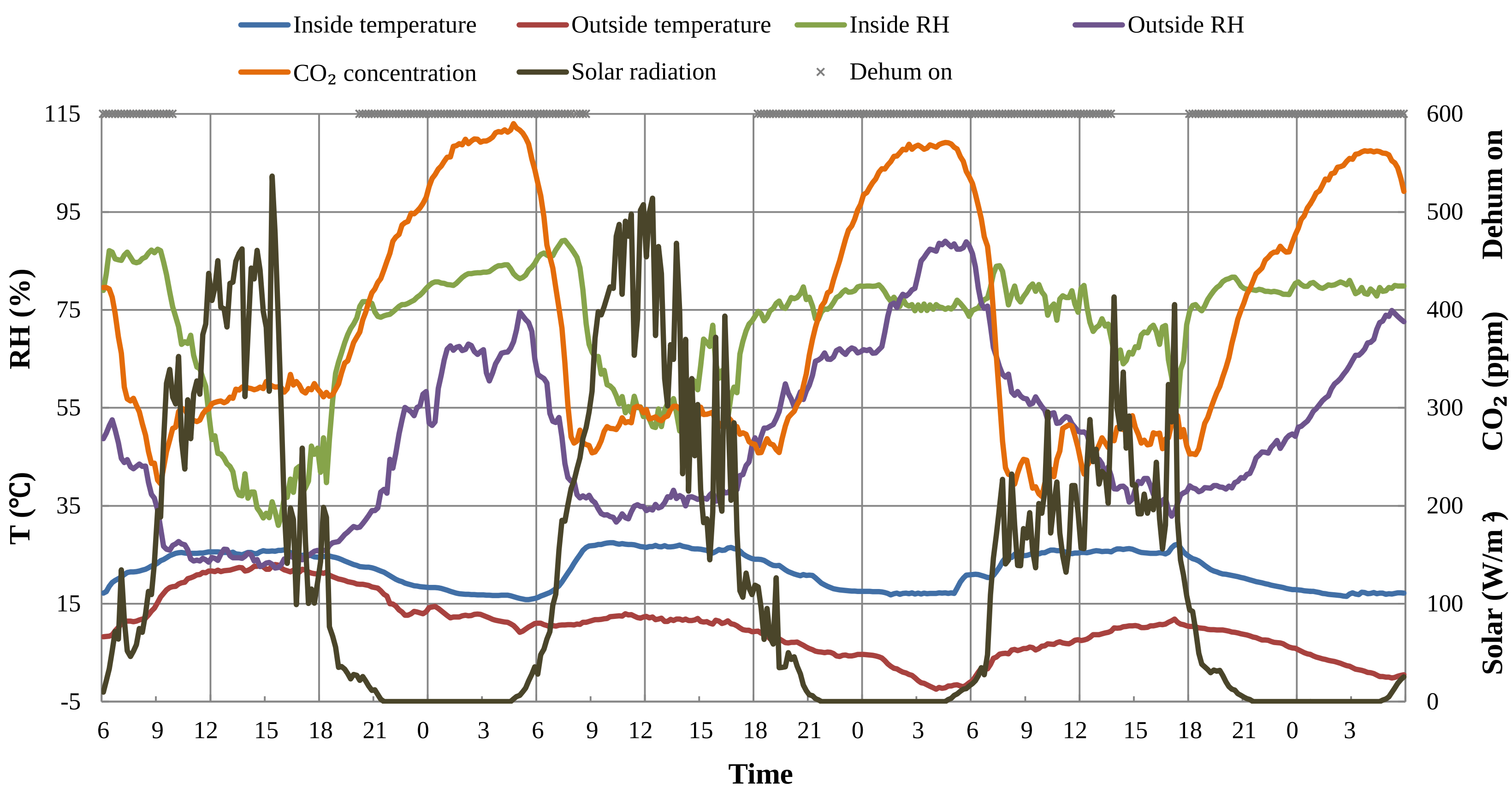

Figure 5.13 Environmental conditions during June 3 to 6, 2009 (dehumidifier treatment)

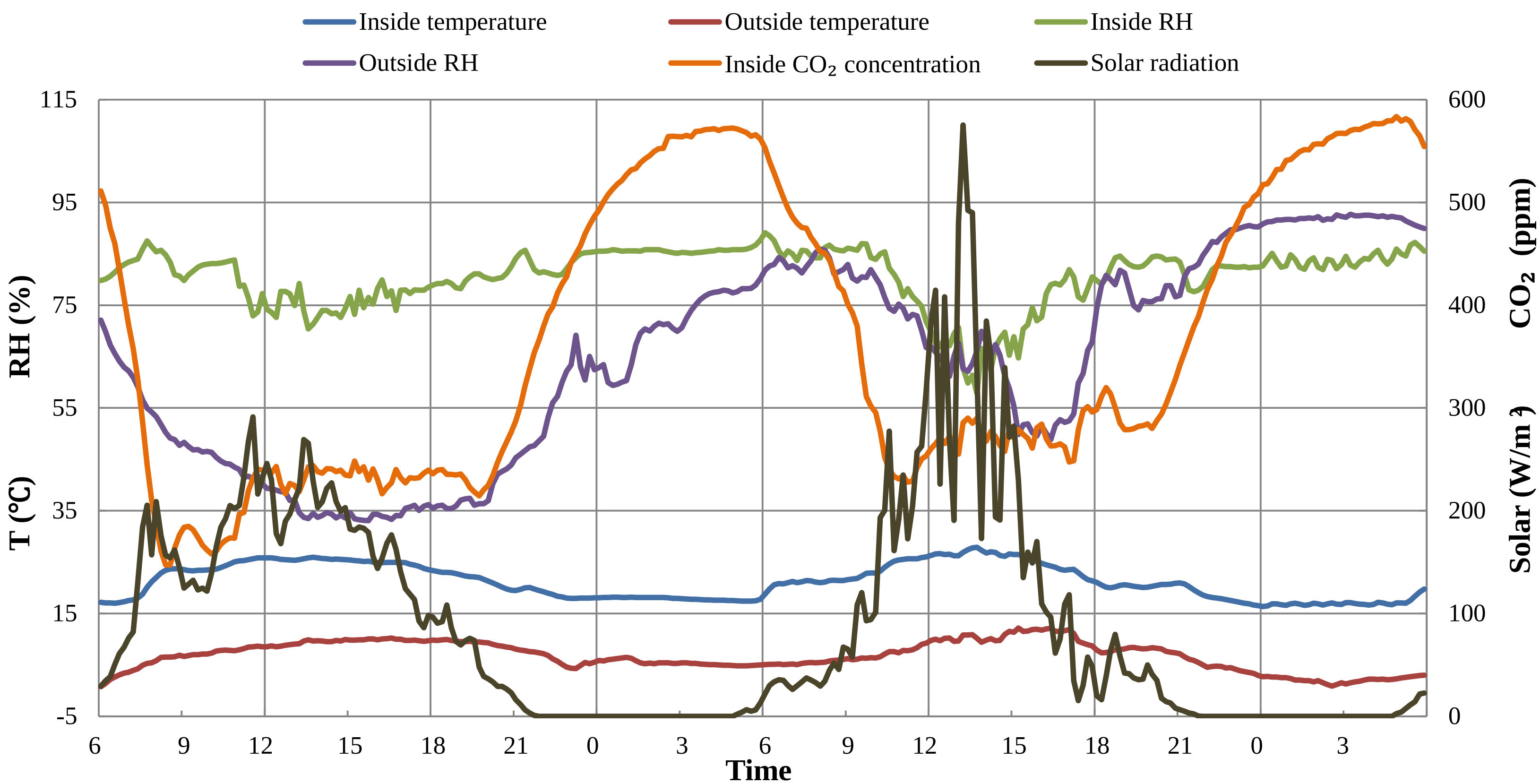

Figure 5.14 Environmental conditions during June 6 to 8, 2009 (control)

During the dehumidification experiment, another problem was discovered: the CO_2 shortage in greenhouses greatly hinders plant growth and yields. However, CO_2 enrichment is rarely used in Saskatchewan and many growers have not realized its importance, although CO_2 enrichment is used quite commonly in commercial greenhouses in most other parts of the world. By increasing CO_2 levels during the daytime from 600 to 1500 ppm depending on the crops, the yield can be increased by 20 to 30%. The experimental greenhouse did not have CO_2 enrichment. The measurement data indicated that the CO_2 level before sunrise was 450 to 500 ppm, but it quickly dropped to as low as 130 ppm, which was close to the CO_2 compensation point (i.e., growth has effectively ceased due to a shortage of CO_2). The daytime average CO_2 concentration was 289 ppm (winter with no active ventilation) to 348 ppm (summer with high ventilation) during the experimental year. This indicates that most of the time in winter, the plants were starving for CO_2. Adding CO_2 in the air in the cold season will result in high CO_2 utilization efficiency in a CO_2 enrichment greenhouse because low ventilation ensures little CO_2 loss by ventilation. However, during mild and warm seasons, higher ventilation rates for humidity and temperature control will result in loss of CO_2 added to the air. This is the main concern of the greenhouse growers for not using the CO_2 enrichment.

5.2.2.4 Dehumidification Effectiveness Statistical Comparison

Greenhouse inside RH conditions with different treatments were compared statistically in order to find whether they are significantly different and to determine which one is most effective based on the statistical results. Considering the control treatment was eliminated after June 17, 2009, the data of each weather condition were divided into two parts: with control treatment or without it. Tables 5.12 and 5.13 summarize the statistical results. One-way ANOVA using SPSS 19.0 was applied to test if the treatment (heat exchanger, dehumidifier, and control) has significant effect on the RH condition. The RH data used for the statistical analysis were the original measurements with 10 minutes average data. Tukey's test was used to find which treatment differed from the others, and the results are demonstrated in

Table 5.12. In the cold season (November 18 to December 29, 2008, and March 20 to 28, 2009), the heat exchanger and dehumidifier both had significantly stronger ability to reduce the RH than the traditional way ($P < 0.05$), with average RH of 1.2% and 1.0% lower than levels obtained using the control method, respectively. However, there was no significant difference between the RH conditions with heat exchanger and with dehumidifiers during the cold period ($P > 0.05$). In mild weather (March 28 to May 31, 2009), there were still significant differences between the RH conditions using heat exchanger/dehumidifier and the control way ($P < 0.05$). The average RH with heat exchanger was 3% lower than that with the control, and the average RH with dehumidifier was 1.7% lower. The heat exchanger was significantly more powerful to dehumidify the greenhouse air than the dehumidifier in mild season ($P < 0.05$) with average RH of 1.3% lower. During the warm period (May 31 to June 17, 2009), there were significant differences between the RH conditions using dehumidifier and heat exchanger or the control method ($P < 0.05$). Dehumidifiers had the significantly strongest ability to remove moisture during the warm period with average RH of 4.4% or 2.7% lower than that applying control or heat exchanger treatment. Heat exchanger was significantly more powerful to remove the moisture than the control, with average RH of 1.7% lower than that of the control ($P < 0.05$).

Table 5.12 Dehumidification effectiveness statistical analysis results for three treatments (heat exchanger, dehumidifier and control)

Weather condition	RH Mean (S.D.) (%)			(I) Treatment	(J) Treatment	RH Mean difference (I-J) (%)	P
	Heat exchanger	Dehumidifier	Control				
Cold (Nov.18-Dec.29, 2008, Mar.20-28, 2009)	66.0 (7.5)	66.2 (6.5)	67.2 (7.1)	Heat exchanger	Dehumidifier	-0.2	0.562
					Control	-1.2	0.014
				Dehumidifier	Control	-1.0	0.021
Mild (Mar.28-May 31, 2009)	69.8 (9.2)	71.1 (10.2)	72.8 (10.5)	Heat exchanger	Dehumidifier	-1.3	0.014
					Control	-3.0	0.005
				Dehumidifier	Control	-1.7	0.012
Warm (May 31-Jun. 17, 2009)	70.7 (11.8)	68.0 (16.1)	72.4 (17.1)	Heat exchanger	Dehumidifier	2.7	0.006
					Control	-1.7	0.010
				Dehumidifier	Control	-4.4	0.002

Note: The data of the first day of each cycle were eliminated for the statistical analysis. S.D. is the standard deviation. Confidence level was set to be 0.05.

Table 5.13 Dehumidification effectiveness statistical analysis results for two treatments (heat exchanger and dehumidifier)

Weather condition	RH Mean (S.D.) (%)		(I-J) RH Mean difference (%)	P
	(I) Heat exchanger	(J) Dehumidifier		
Cold (Oct.30-Nov.17, 2009)	75.1 (2.8)	76.9 (2.7)	-1.8	0.010
Mild (Sept.30-Oct.30, 2009)	76.3 (3.0)	77.4 (3.7)	-1.1	0.019
Warm (Jun.17-Sept.30, 2009)	76.6 (14.0)	77.7 (12.3)	-1.1	0.020

Note: The data of the first day of each cycle were eliminated for the statistical analysis. S.D. is the standard deviation. Confidence level was set to be 0.05.

For the data with no control method, two independent samples T test was used to test if the RH condition using heat exchanger was the same as that using dehumidifier. Table 5.13 demonstrates that the RH conditions between using these two treatments were significantly different during the test period ($P < 0.05$). In cold season (October 30 to November 17, 2009), the heat exchanger kept the average RH at 1.8% lower than dehumidifier treatment. In mild and warm periods (June 17 to October 30, 2009), the average RH with heat exchanger was approximately 1.1% lower than that with dehumidifier. Therefore, the heat exchanger had a significantly stronger ability to lower the RH than the dehumidifier.

In summary, the dehumidification treatments applied had significant effects on the greenhouse RH conditions all year round. The heat exchanger was the most effective way to lower RH in cold and mild seasons, whereas the control method was significantly less capable of dehumidifying the moist air than the other two treatments. The dehumidifier treatment had significantly stronger ability to remove moisture than the control did all year round, although it was less powerful than the heat exchanger treatment.

5.2.3 Performance Analysis of Heat Exchangers

Heat recovery ratio (HRR) was used to evaluate the sensible heat recovery performance of the heat exchangers in winter and mild weather periods. The HRR calculation method has been introduced in 4.2.7.2, and a sample calculation showing the specific calculation procedures is given in Appendix C.2.

Under the warm weather period, the ambient temperature was relatively high and sometimes higher than the inside setpoint, making the heat recovery function useless or undesirable because of the extra heat added to the greenhouse; thus, the heat recovery ratio was calculated only for cold and mild weather conditions. Summarized in Table 5.14 are the calculation results, which were the averages of the cold or mild periods. The heat recovery ratios of RA400 and RA1000 heat exchangers were approximately 55.7% and 46.8%, respectively. The results were both higher than the claimed energy efficiency values by the manufacturer, which were 41% and 34%, respectively. The main reasons were, first, HRR values only took the sensible heat into account whereas the energy efficiency used the total heat including the latent heat in the calculations; second, these two heat exchangers were modified by the manufacturer to meet our requirement for air flow rates. Based on the heat recovery ratio results, both heat exchangers were capable of recovering sensible heat from the warm exhaust air, thus saving the supplemental heating costs in heating season and also during night and early morning in the mild and warm seasons.

Table 5.14 Heat recovery ratio of the heat exchanger

Weather condition		HRR (%)	
		RA400 heat exchanger	RA1000 heat exchanger
Cold	Average (S. D.)	56.5 (2.3)	46.3 (1.8)
Mild	Average (S. D.)	54.8 (1.3)	47.3 (1.4)
	Average	55.7	46.8

5.2.4 Moisture Removal Index Analysis

The MRI calculation methods for each dehumidification treatment have been introduced in 4.2.7.3, and sample calculations showing the specific MRI calculation procedures for each treatment are given in Appendix C. The average moisture removal index for each dehumidification treatment are given in Table 5.15.

Table 5.15 Average moisture removal index for each dehumidification treatment

Weather Condition	1	2	3	4	5			6
	MRI (kW-h/L)	MRI (kW-h/L)	MRI (kW-h/L)	MRI (kW-h/L)	MRI (kW-h/L)	MRI1 (kW-h/L)	MRI2 (kW-h/L)	MRI (kW-h/L)
Cold	1.020 (0.031)*	0.916 (0.029)*	1.373 (0.086)*	1.219 (0.050)*	-0.630 (0.001)*	0.514	-1.144	7.202 (24 ℃, 90% , 0.65 L/s, 0 ℃)
Mild	1.007 (0.035)*	0.945 (0.041)*	1.311 (0.078)*	1.230 (0.070)*	-0.629 (0.001)*	0.514	-1.143	8.375 (24 ℃, 80%, 0.65 L/s, 0 ℃)
Warm	0.960 (0.052)*	0.992 (0.068)*	1.099 (0.099)*	1.111 (0.107)*	-0.629 (0.001)*	0.514	-1.143	8.757 (24 ℃, 90%, 0.85 L/s, 0 ℃)
Average	0.996	0.951	1.261	1.187	-0.629	0.514	-1.143	

Note: 1 = “RA400 heat exchanger”, 2 = “RA1000 heat exchanger”, 3 = “Exhaust fan (using exhaust fan’s parameters of RA400 heat exchanger)”, 4 = “Exhaust fan (using exhaust fan’s parameters of RA1000 heat exchanger)”, 5 = “Dehumidifiers”, and 6 = “Finned tubing condensation system using chilled water”. *values in the brackets are standard deviations.

As shown in Table 5.15, the moisture removal index of the dehumidifiers was around -0.629 kW-h/L, which means that they could release 0.629 kW-h energy per liter of water removed. The energy obtained from heat output of the dehumidifiers and the latent heat released by the condensed water was greater than the electrical energy consumption, making the MRI negative. In order to better understand this value and make it easily comparable with the other treatments, the modified moisture removal index values were also calculated for dehumidifiers as given in Table 5.15. First, MRI1 was calculated by dividing the electrical energy consumption of the dehumidifiers with the total volume of water removed, and the result is 0.514 kW-h/L, which means the dehumidifier consumed 0.514 kW-h electrical energy per liter of water removed. Second, MRI2 was calculated by dividing the sum of the heat output of the dehumidifiers and latent heat released of the condensed water by the total condensed water volume. It is around -1.143 kW-h/L, which means the dehumidifier released 1.143 kW-h thermal energy by removing one liter of water vapour. Referring back to Table 5.8 demonstrates that dehumidification in the nights of mild and warm seasons could be achieved by the existing capacities of heat exchangers or dehumidifiers, but it was more difficult to do so during morning or daytime when ambient humidity was high. The dehumidifier can supplement some heat at night while dehumidifying the air, which is a great advantage for greenhouse dehumidification. Therefore, as compared with two heat exchangers, dehumidifiers are more energy efficient based on the moisture removal index results. In addition, they are more advantageous during heating season and even during night and early morning in mild and warm seasons because of the heat released into greenhouses, but undesirable during warm season because of the extra cooling load added to the greenhouses. Compared to the solar heat gain, the amount of the heat released should be negligible.

Table 5.15 also lists the MRI values for the hypothetical dehumidification treatment -- RH based ventilation using the exhaust fans (the data of RA400 and RA1000 heat exchanger's exhaust fans were used for MRI calculations), and the top three smallest MRI values for the finned tubing condensation system. Compared to the RH based ventilation method, RA400 heat exchanger has approximately 25.7% and 23.2% lower MRI, and RA1000 heat exchanger has 24.9% and 23.2% lower MRI in cold and mild seasons, respectively. In the warm season, the MRI of RA400 heat exchanger is 12.6% lower and MRI of RA1000 heat exchanger is 10.7% lower than that of the ventilation based on RH control. So even in the warm season of the cold regions, the heat exchanger could still recover some heat and thus save energy particularly during the early morning and mid-night when the ambient temperature is very low. The MRI of the dehumidifier is approximately 53.0% of the levels for the RH based ventilation (RA1000 exhaust fan), whereas the MRI of the finned tubing condensation system is nearly 7 times. Undoubtedly, the dehumidifier is the most energy efficient based on the moisture index results, for it can save energy when removing moisture. The finned tubing condensation system is the most energy intensive, thus becoming an impractical method for greenhouse dehumidification in cold regions.

The MRI calculations were based on the experiment that can never avoid errors including systematic errors (non-random-imperfections mainly caused by accuracy of instruments) and random errors in measurements and observations. The overall error estimation results of MRI are provided for each treatment in Table 5.16 to demonstrate MRI uncertainties. The detailed error estimation procedures are given in Appendix C.8.

As shown in Table 5.16, the absolute errors of MRI for each treatment were less than 0.06 kW-h/L except for the finned tubing condensation system which were around 0.4 kW-h/L. Taking the RA400 heat exchanger in cold weather conditions as an example, the error for MRI value was around 0.043 kW-h/L, which was approximately 4.2% of the MRI value for RA400 heat exchanger in cold weather conditions that provided in Table 5.15. Therefore, the MRI difference within the error range may not be caused by different treatment applied, but resulted from the errors that existed in the experiment and calculation procedures.

Table 5.16 Overall error estimation results for each dehumidification treatment

Weather Condition	1	2	3	4	5	6
	MRI error (kW-h/L)	MRI error (kW-h/L)	MRI error (kW-h/L)	MRI error (kW-h/L)	MRI error (kW-h/L)	MRI error (kW-h/L)
Cold	0.043	0.038	0.058	0.051	0.026	0.307 (24 ℃, 90% , 0.65 L/s, 0 ℃)
Mild	0.042	0.040	0.055	0.052	0.025	0.348 (24 ℃, 80%, 0.65 L/s, 0 ℃)
Warm	0.039	0. 040	0.046	0. 046	0.025	0.389 (24 ℃, 90%, 0.85 L/s, 0 ℃)

Note: 1 = “RA400 heat exchanger”, 2 = “RA1000 heat exchanger”, 3 = “Exhaust fan (using exhaust fan’s parameters of RA400 heat exchanger)”, 4 = “Exhaust fan (using exhaust fan’s parameters of RA1000 heat exchanger)”, 5 = “Dehumidifiers”, and 6 = “Finned tubing condensation system using chilled water”.

Statistical analysis using one-way ANOVA was performed to test if the moisture removal index among different treatments was significantly different. The treatments tested include: (1) RA400 heat exchanger; (2) RH based ventilation (using the data of RA400 heat exchanger's exhaust fan); (3) RA1000 heat exchanger; (4) RH based ventilation (using the data of RA1000 heat exchanger's exhaust fan); and (5) mechanical refrigeration dehumidifiers. Tukey's test was used to find the difference between each of the two treatments, and the results are shown in Table 5.16.

As can be seen in Table 5.17, MRI of the heat exchanger treatment is significantly different from that of the RH based ventilation method in cold and mild weather conditions ($P < 0.05$). In cold and mild weather conditions, the average MRI of RA400 heat exchanger is around 0.325 kW-h/L lower than that of the corresponding exhaust fan, and the average MRI of RA1000 heat exchanger is nearly 0.300 kW-h/L lower than that of the corresponding exhaust fan. In warm season, the mean difference between the MRI of the heat exchanger and the MRI of the corresponding exhaust fan is much smaller compared to the other seasons (0.139 kW-h/L and 0.119 kW-h/L for RA400 and RA1000 heat exchanger, respectively). The result shows no significant difference between MRI of the heat exchanger and MRI of the corresponding exhaust fan treatment in warm season ($P > 0.05$).

Thus, the heat exchanger can save more energy than the exhaust fan in cold and mild weather periods, but in warm season, no significant difference of energy consumption exists between these two treatments. The MRI of the dehumidifier is significantly lower than that of the other two treatments ($P < 0.05$). In cold and mild seasons, it is approximately 1.640 kW-h/L and 1.972 kW-h/L lower than those of RA400 heat exchanger and the corresponding exhaust fan, respectively; in the warm period, it is around 1.589 kW-h/L and 1.728 kW-h/L lower. Therefore, the dehumidifier is undoubtedly the most energy efficient dehumidification method among these treatments according to the MRI statistical comparison results.

Table 5.17 MRI statistical comparison results for five treatments

(I) Treatment	(J) Treatment	(I-J) MRI Mean difference (kW-h/L) (P)		
		Cold	Mild	Warm
1	3	-0.353 (0.018)	-0.304 (0.020)	-0.139 (0.188)
	5	1.650 (0.005)	1.636 (0.007)	1.589 (0.008)
2	4	-0.303 (0.021)	-0.285 (0.026)	-0.119 (0.207)
	5	1.546 (0.008)	1.574 (0.008)	1.621 (0.007)
5	3	-2.003 (0.002)	-1.940 (0.003)	-1.728 (0.006)
	4	-1.849 (0.003)	-1.859 (0.003)	-1.740 (0.006)

Note: The MRI data of each cycle during different weather conditions were used for the statistical analysis. 1 = "RA400 heat exchanger", 2 = "RA1000 heat exchanger", 3 = "Exhaust fan (using exhaust fan's parameters of RA400 heat exchanger)", 4 = "Exhaust fan (using exhaust fan's parameters of RA1000 heat exchanger)", 5 = "Dehumidifiers". Confidence level was set to be 0.05.

5.2.5 Energy Cost Analysis

The energy cost calculation methods for each dehumidification treatment have been introduced in Section 4.2.7.3, and sample calculations showing the specific energy cost calculation procedures for each treatment are given in Appendix C.7. The price of natural gas (thermal coal) used in the calculations was the same as that introduced in Chapter 5.1.5. Table 5.18 provides the cost estimation results for each dehumidification treatment applying natural gas as the thermal energy source, and Table 5.19 gives the cost estimation results using thermal coal as the thermal energy source.

As shown in Table 5.18, if natural gas was used as the heat source, the top three lowest energy costs for the finned tubing condensation system were \$0.322/L, \$0.360/L, and \$0.381/L, respectively. The lowest one -- \$0.322/L -- was approximately 18 times the energy cost using dehumidifier treatment, 11 times that using heat exchanger treatment, and 9 times that applying RH based ventilation by exhaust fans. If thermal coal was used as the heat source (shown in Table 5.19), the top three lowest energy costs for the finned tubing condensation system were \$0.240/L, \$0.253/L, and \$0.286/L, respectively. The lowest one -- \$0.240/L -- was approximately 7 times the energy cost using dehumidifier treatment, 15 times that using heat exchanger treatment, and 13 times that applying RH based ventilation by exhaust fans. Therefore, the finned tubing condensation was the most costly in terms of the energy cost no matter which fuel (natural gas or thermal coal) was used to heat the greenhouse.

As seen in Table 5.18, the dehumidifier treatment was the most economical if natural gas was used as the greenhouse heating source. The average energy cost of the dehumidifiers was \$0.018/L, nearly 60% of the heat exchanger's cost, and around 45% of the exhaust fan's cost. In cold and mild weather conditions, the energy cost of RA400 heat exchanger decreased by 20.5% compared to that of the RH based ventilation using the exhaust fan of RA400 heat exchanger, and the energy cost of RA1000 heat exchanger decreased by 22.5% compared to that of the RH based ventilation using the exhaust fan of RA1000 heat exchanger. During the warm season, the energy costs for the heat exchanger treatment and RH based ventilation treatment were very close. The annual energy cost of the heat exchangers was approximately 1.8 times that of the dehumidifiers. Cost1 and Cost2 shown in Table 5.18 indicate the electrical energy cost of the dehumidifiers and the rewards from the heat output of the dehumidifiers and the latent heat released by the condensate, respectively. Therefore, the energy cost of \$0.018/L for the dehumidifiers was composed of two parts: \$0.050/L used for the electrical energy consumption, and \$0.032/L rewarded from the heat output and the latent heat released.

However, referring to Table 5.19, the heat exchanger was the most economical if thermal coal was used to heat the greenhouse. In cold and mild weather conditions, RA400 heat exchanger had 15.8% lower energy cost than RH based ventilation using the exhaust fan of RA400 heat exchanger, and RA1000 heat exchanger had 17.6% lower energy cost compared to RH based ventilation using the exhaust fan of RA1000 heat exchanger. During the warm season, the energy costs for the heat exchanger treatment were approximately 11.1% higher than that for the RH based ventilation treatment. The reason was that during summer of 2009, the temperature difference and humidity ratio difference between inside and outside air were small. Thus, the heat loss between these two treatments were close whereas the electrical energy consumptions of the heat exchangers were greater than the exhaust fans, causing the MRI of the heat exchangers was $0.002/L higher. The annual energy cost of the dehumidifiers was around 2 times that of the heat exchangers. Cost1 and Cost2 shown in Table 5.19 have the same indications as those shown in Table 5.18. Hence, the energy cost of $0.035/L for the dehumidifiers was also composed of two parts: $0.050/L used for the electrical energy consumption, and $0.015/L rewarded from the heat output and the latent heat released.

Combined with the moisture removal index results shown in Table 5.15, the finned tubing condensation system was undoubtly the most energy intensive, and the most costly method. The dehumidifier treatment was the most energy efficient according to the MRI results, and also the most economical if natural gas was employed as the heat source. However, it was more costly than the heat exchanger treatment and RH based ventilation using exhaust fans if the heat source was thermal coal. The heat exchanger was the most economical dehumidification method if therrmal coal was used to heat the greenhouse, however, it was more energy intensive than the dehumidifiers although more energy efficient than the RH based ventilation using exhaust fans.

Table 5.18 Energy cost estimation of four treatments (use natural gas as thermal energy source)

Weather Condition	1	2	3	4	5			6
	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost1 ($/L)	Cost2 ($/L)	Cost ($/L)
Cold	0.031	0.027	0.040	0.035	0.018	0.050	-0.032	0.322 (24 ℃, 90% , 0.65 L/s, 0 ℃)
Mild	0.031	0.028	0.038	0.036	0.018	0.050	-0.032	0.360 (24 ℃, 90%, 0.85 L/s, 0 ℃)
Warm	0.033	0.033	0.034	0.034	0.018	0.050	-0.032	0.381 (24 ℃, 80%, 0.65 L/s, 0 ℃)

Note: 1 = “RA400 heat exchanger”, 2 = “RA1000 heat exchanger”, 3 = “Exhaust fan (using exhaust fan’s parameters of RA400 heat exchanger)”, 4 = “Exhaust fan (using exhaust fan’s parameters of RA1000 heat exchanger)”, 5 = “Dehumidifiers”, and 6 = “Finned tubing condensation system using chilled water”. Here “Cost” indicates energy cost. “Cost1” indicates the electrical energy cost of the dehumidifiers; “Cost2” indicates the thermal energy cost of the dehumidifier treatment.

Table 5.19 Energy cost estimation of four treatments (use thermal coal as thermal energy source)

Weather Condition	1	2	3	4	5			6
	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost ($/L)	Cost1 ($/L)	Cost2 ($/L)	Cost ($/L)
Cold	0.016	0.014	0.019	0.017	0.035	0.050	-0.015	0.240 (24 ℃, 90% , 0.65 L/s, 0 ℃)
Mild	0.016	0.014	0.019	0.017	0.035	0.050	-0.015	0.253 (24 ℃, 90%, 0.85 L/s, 0 ℃)
Warm	0.020	0.020	0.018	0.018	0.035	0.050	-0.015	0.286 (24 ℃, 80%, 0.65 L/s, 0 ℃)

Note: 1 = “RA400 heat exchanger”, 2 = “RA1000 heat exchanger”, 3 = “Exhaust fan (using exhaust fan’s parameters of RA400 heat exchanger)”, 4 = “Exhaust fan (using exhaust fan’s parameters of RA1000 heat exchanger)”, 5 = “Dehumidifiers”, and 6 = “Finned tubing condensation system using chilled water”. Here “Cost” indicates energy cost. “Cost1” indicates the electrical energy cost of the dehumidifiers; “Cost2” indicates the thermal energy cost of the dehumidifier treatment.

In summary, from an energy conservation point of view, the dehumidifier treatment represented the best way for greenhouse dehumidification all year round because it was the most energy efficient; from an economic benefit point of view, the dehumidifier treatment saved more energy cost than the heat exchanger treatment if natural gas was used to heat the greenhouse, however, the heat exchanger was more energy cost effective than the dehumidifier treatment if thermal coal was used as the heat source. Considering the special climate in Saskatchewan, where the heating season is long and even summer nights are cool (average daily low temperature in Prince Albert is 9 to 11 ℃ from June to August) and which needs heating in greenhouses, the energy efficiency of the mechanical refrigeration dehumidifier is desirable. Its drawback is adding extra heat to the greenhouse during the warm daytimes of summer, but compared with the solar heat gain, the amount of heat added by dehumidifiers during daytime is negligible. Also, natural gas is the most commonly used fuel to heat greenhouse in Saskatchewan. Therefore, mechanical refrigeration dehumidifier shows the greatest potential for greenhouse dehumidification, for it was proved both energy efficient and cost effective.

However, in the experiments conducted during 2008 to 2009, the capacity of the two domestic dehumidifiers was insufficient, and they were not robust for use in the greenhouse. This circumstance resulted in excessively high RH occurring from April to November (up to 73% of the time RH was at or over 75%). Both of these units failed within a year and were replaced with new ones. A high efficiency, high capacity, and durable commercial grade dehumidifier should be used and evaluated. Another problem that needs to be solved concerns how to precisely quantify the dehumidification requirement of a greenhouse in order to size and select a dehumidifier for a given greenhouse situation. Numerous factors affect the amount of water vapour in the air: soil evaporation rate, plant transpiration rate, inside and outside temperature, relative humidity, condensation on the interior surfaces of the greenhouse, ventilation rates, and water input to the plants, to name a few. It is necessary to determine the moisture balance in the greenhouse air in order to calculate the dehumidification requirement and to select the dehumidifier accordingly.

Based on the grower's experience in the last five years, every year approximately 20% yield loss is caused by fungus disease from the high RH. In 2010 (with no control treatment), the total tomato production was around 23,868 lb from April to September and 7,956 lb from October to December. The average price of the tomato in wintertime is approximately $3.8/lb, and this price is reduced by one dollar in summertime. Thus, the total tomato revenue of 2010 was approximately $97,063 and the net revenue was estimated to be $67,944. Therefore, 20% yield loss in the past year means around $13,588 net revenue loss.

Chapter 6. CONCLUSIONS

The laboratory experiment showed that the condensation rate of the finned tubing was significantly higher when using 0 ℃ water instead of 5 ℃ water, and it increased significantly as the flow rate increased from 0.65 L/s to 0.85 L/s ($P < 0.05$), but no significant changes were observed when the flow rate exceeded 0.85 L/s ($P > 0.05$). The four variables of room temperature, room RH, water temperature, and water flow rate, were all proven to have significant effects on the condensation rate ($P < 0.05$). Two linear regression models were developed to simulate the condensation rate using these predictors. There existed highly positive correlations between the variables and the condensation rate except for water temperature, which was negatively correlated with condensation rate. Future research is needed for the model validations.

Comparing energy consumption with the other methods tested in the field experiment, the finned tubing condensation system was least energy efficient (even the smallest MRI of the condenser can reach as high as 7 times that of the RH based ventilation treatment). Comparing energy cost, the finned tubing condensation system was most costly. The lowest energy cost of the condenser was also 7 times the energy cost of the dehumidifiers no matter which fuel (thermal coal or natural gas) was applied for calculations. Not only the great energy consumption and high energy cost, but also the high capital and installation costs stopped the test of the finned tubing condensation system in the field experiment.

One year of experimental results from the field experiment indicated that dehumidification was needed for most of the year in Saskatchewan greenhouses during April to November. The high RH occurred less during winter than in spring/fall and summer. The monthly 75% RH exceeding percentage ranged from

32.5 to 73.6% from April to November and less than 10% from December to March. During winter, high RH occurred mostly during daytime but remained low at night; during mild and warm seasons, high RH occurred mostly at night and early morning, but stayed low from late morning to late afternoon. The peak RH occurred during the period of 9 am to nearly 12 pm when the ambient temperature was still not high, and the ventilation rate was still low, so the moisture produced in the greenhouse was not removed even by its maximum ventilation rate.

The heat exchangers controlled indoor RH very well under cold and mild weather conditions. However, when the ambient air was humid during the spring/fall and summer, this method worked less effectively. The fact that it added extra heat to the greenhouse by recovering some heat from the exhaust air during mild and warm seasons did not appear to be a problem. Rather, it was quite desirable most of the time because the dehumidification was needed often during night and early morning when the temperature was low in Saskatchewan, making supplemental heating a need during those periods of time. During sunny days when high temperature occurred in the greenhouse, the heat released by the heat exchangers was negligible compared with the solar heat gain. Therefore, the heat exchangers can be applied all year round for greenhouse dehumidification in Saskatchewan except during the humid weather conditions.

The dehumidifiers were less competitive than the heat exchangers based on RH control accuracy; however, this may be caused by insufficient capacity of the domestic units used. If high efficient commercial grade dehumidifiers were installed, the performance would improve. The required capacity of the dehumidifiers should be re-evaluated and increased to meet the cold and mild seasons' dehumidification requirements. This method was more energy efficient than the others during heating seasons because almost all the electrical energy consumed was converted to heat, and the latent heat from the water vapour condensation was released to the greenhouse, too. Consequently, there was almost no wasted energy. The dehumidifier method was effective for heating seasons and also desirable for early morning and nights during mild and warm seasons when heating was still needed, but not desirable for cooling season (particularly for daytimes in summer, because of adding heat to the greenhouse). However, the added heat was negligible as compared with the high solar energy gain of the greenhouse.

The RH based ventilation method had the equal energy consumption in warm season as that using the heat exchangers, however, it was less energy efficient and less economical than heat exchangers in cold and mild seasons. These times represent a critical long period for greenhouse dehumidification in cold regions.

The traditional method of opening vents and exhaust fans based on temperature control did not result in prolonged high RH under cold weather, although its 75% RH exceeding percentage was higher than values for the heat exchanger and dehumidifier treatments. It also resulted in a higher heating requirement during cold period. Under mild and warm weather conditions, this method was ineffective to keep the RH at an acceptable range: 52.9 to 62.7% of the time the RH was at or above 75%.

Comparing energy consumption, the energy requirement per liter of water removed from the greenhouse using RA400 heat exchanger was 25.7% and 23.2% lower, and that using RA1000 heat exchanger was 24.9% and 23.2% lower than the energy demands of RH based ventilation method in cold and mild seasons, respectively; in the warm season, RA400 heat exchanger had 12.6% lower energy need per liter of water and RA1000 heat exchanger had 10.7% lower energy demands than the RH based ventilation treatment. The energy requirement per liter of water for the dehumidifier treatment was the least of all the treatments, which was the only method that could release approximately 0.629 kW-h energy rather than consume the energy.

Comparing energy cost, the dehumidifier method was the most economical with annual average energy cost of $0.018/L if natural gas was the heat source, approximately 60% and 50% of those of the heat exchaners and exhaust fans, respectively. If thermal coal was used as the heat source, the heat exchanger method was the most economical with annual average energy cost of \$0.016/L, as compared to \$0.019/L and \$0.035/L for the dehumidifiers and exhaust fans, respectively. Thus, the energy cost of each treatment not only relied on the energy consumption, but also depended on which fuel would be used to heat the greenhouse.

In summary, dehumidifier shows the greatest potential for greenhouse dehumidification of cold regions based on the following reasons: first of all, the use of dehumidifiers will allow air-tight greenhouses because ventilation for humidity control is eliminated, as a result, heat loss by ventilation for humidity control will be minimized. The commercial grade dehumidifier with high capacity and high efficiency will be effective for most time of the year, including cold and mild seasons, and some of the warm seasons when supplental heating is still required (low temperature nights and cloudy daytimes, etc.); in addition, dehumidifier has the advantage of offering supplemental heat while dehumidifying the air, and it is proved to be the most energy efficient method of all the potential treatments tested; furthermore, it is also the most economical method if natural gas (most commonly used fuel for Saskatchewan greenhouses) is used as the greenhouse heating source; last but not least, since CO_2 shortage in the greenhouse was discovered most of the time, the application of dehumidifiers can bring an air-tight greenhouse environment, which will promote CO_2 utilization efficiency by plants if CO_2 enrichment is added.

A heat exchanger is also a good choice for greenhouse dehumidification of cold regions primarily according to the following reasons: first, it can be easily set up for use all year round although it is ineffective during humid weather which is uncommon in northern cold regions; second, it has the advantage of supplying CO_2 when the greenhouse ventilation is low; third, it can reduce the heat loss through ventilation by recovering some heat from the exhaust air, which is quite welcomed for energy saving when the ambient tempature is low; last but maybe the most important, it is the most economical method if thermal coal (also a popular fuel used for greenhouses) was employed as the heating source.

Chapter 7. RECOMMENDATIONS

The followings are suggested for future study in greenhouse dehumidification.

1) Although the theoretical equations are not able to predict the condensation rates of the finned tube without experiments, the data obtained from this experiment could be used to verify the equations and related parameters, and compare the statistical models with the theoretical models in the future.

2) The mechanical refrigeration dehumidifier is the recommended method for greenhouse dehumidification; however, the performances of the domestic grade dehumidifiers were not satisfactory, and breakdown occurred to the dehumidifiers during the test. It is recommended that commercial grade dehumidifiers with high capacity and high efficiency should be evaluated in a greenhouse; economic analysis of cost and benefit should be performed.

3) This study found that the incapacity of the heat exchangers and dehumidifiers to control RH is caused by the insufficient capacity of the units. Therefore, determining the requirement of dehumidification is the key to successful control of RH in the greenhouse. It is suggested that the moisture balance in the airspace of the greenhouse and the water balance of the greenhouse be modelled in order to determine precise dehumidification requirements.

4) Since there appears to be a low CO_2 concentration in the greenhouse all year round, it is suggested that we should have CO_2 enrichment. Researchers could then evaluate the efficiency of CO_2 enrichment in the greenhouses with dehumidifiers, and then proceed to conduct economic analyses on costs and benefits. As elaborated in the conclusions, the use of dehumidifiers will allow airtight greenhouses, consequently, CO_2 enrichment will take place with high utilization

efficiency by plants (close to 100% due to the air-tight greenhouse) most of the year (spring, fall, and winter). Even in summer, in the morning when the greenhouse temperature is not high and ventilation rate is low, CO_2 enrichment can still be used effectively. The yield increase can be significant (20 to 30%).

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APPENDIX A. CALIBRATION OF THE SENSORS FOR LABORATORY EXPERIMENT

A.1 Calibration of the Temperature Sensors

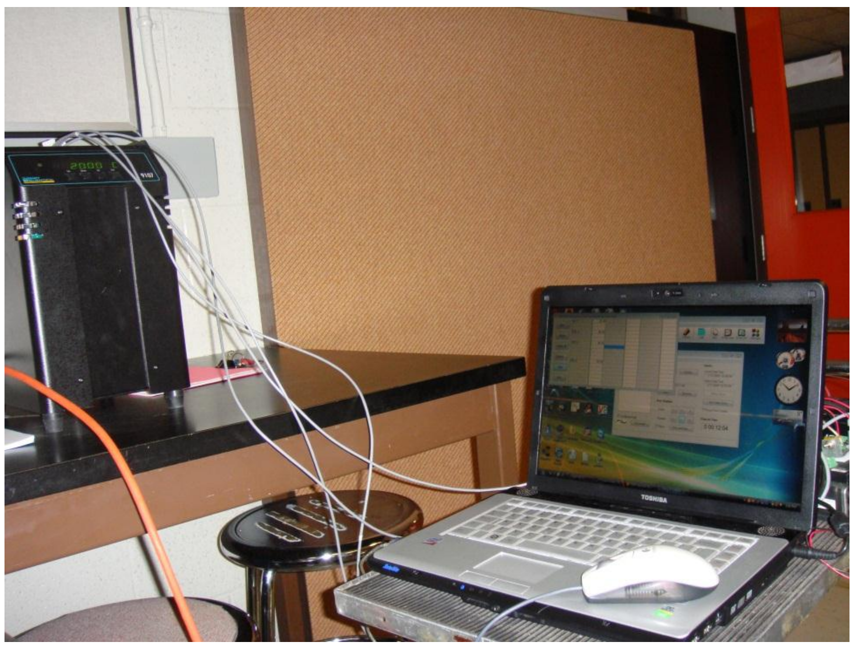

Figure A.1 Calibration of temperature sensors (RTDs)

The temperature sensors (Resistance Temperature Device, Model HEL-775-B-T-0, Honeywell Inc., Morristown, NJ, USA) were 100-Ohm platinum RTDs with temperature sensing range of -55 to 150 ℃. Three-wire half bridge was applied to measure each 100-Ohm platinum RTD. The temperature calibration procedure was conducted in the laboratory in the Department of Mechanical Engineering, University of Saskatchewan. All the four sensors were put in the climatic control chamber (Dry block temperature calibrator, 9170 Hart Scientific, Fluke Corporation, American Fork, UT, USA); adjusting the set point temperature from -15 to 35 ℃ by step of 5 ℃. All the data were recorded by CR10X data logger and displayed on the laptop. The

calibration results between the temperature measured by calibrator and RTDs were listed in Table A.1.

Table A.1 Calibration results of RTDs

Temperature sensors	Calibration equation	r^2
RTD 1	Tc = 1.0718Tr-0.7801	0.9993
RTD 2	Tc = 1.057Tr-0.1089	0.9994
RTD 3	Tc = 1.0391Tr-0.6281	0.9995
RTD 4	Tc = 1.0568Tr-0.4918	0.9987

Note: Tc is the temperature measured by the calibrator, ℃; Tr is the temperature measured by the RTDs, ℃.

A.2 Calibration of the Flow Meter

Figure A.2 Hedland model 1100 turbine flow meter

The turbine flow meter (Hedland 1100, Division of Racine Federated Inc., Racine, WI, USA) has a measurement range of 18.93 to 189.3 L/min and the accuracy is ±1.0% of the reading. Calibration is needed in order to know the exact relationship between the flow rate and the pulses. The calibration procedure was conducted in Hardy Lab in the Department of Chemical and Biological Engineering, University of Saskatchewan.

The procedure is elaborated as follows. After all the instruments were set up (shown in Figure A.3), the power was turned on to get the system running for about 2 minutes to obtain a steady state. An empty bucket (20.5 L) was then used to catch the water while clicking the stopwatch to record the time and the datalogger was switched on to record the pulses. When the bucket was full of water, the stopwatch was clicked again and the datalogger was turned off. Then, the converter was the total of pulses over the volume of the bucket (20.5 L). Repeat the process for three times,; the average calculated converter 3.224 was put into the program.

Figure A.3 Calibration of the flow meter

A.3 Calibration of the Digital Scale

The digital scale (Cole-Parmer Symmetry PR 4200, Cole-Parmer Canada Inc., Montreal, QC, Canada) has a capacity of 4200 g with 0.01 g readability.

Figure A.4 Calibration of the digital scale

The digital scale has the self-calibration function which can be calibrated using an external mass. The calibration procedure was conducted in Room 1A91 in Department of Chemical and Biological Engineering, University of Saskatchewan. The calibrator weights (S/N: 3GWH, Rice Lake Weighing Systems, Rice Lake, WI, USA) of 1 g, 2 g, 5 g, 10 g, 20 g, 50 g, 100 g, 500 g and 2000 g were used for the calibration. Figure A.4 shows the calibration with a 2000 g weight. The results showed that it had $\pm$0.20 g accuracy.

APPENDIX B. CALIBRATION OF THE SENSORS FOR FIELD EXPERIMENT

B.1 Calibration of the Sensors of Weather Station

Figure B.1 Calibration of temperature and relative humidity sensors and pyranometers

The sensors of the weather station were all calibrated in the Hardy Lab in the Department of Chemical and Biological Engineering, University of Saskatchewan.

The CS500 Temperature and RH sensor of the weather station was calibrated against a humidity generator. The result showed that the multiplier was 0.1517 and the offset was 2.1921.

The LICOR Pyranometer was calibrated against a new LICOR and the multiplier was 96.4 with a 100-ohm resistor which was 72.31.

The Young 00330 Wind Direction faced 0 degrees which was North (multiplier = 0.142).

The results of the wind speed sensor showed that the muliplier was 0.02666 provided 60 seconds interval or 1.5997 for one second, the offset was 0.4470.

B.2 Calibration of the CO_2 Monitor

The CO_2 monitor was calibrated using the calibration gas with 1529 ppm and 2295 ppm gas at the Prairie Swine Center, Saskatoon, Canada. The camera was not allowed into the barn because of the biosecurity purpose; thus no pictures of CO_2 calibration equipment set-up are shown here. The potentiometer inside of the monitor was adjusted to get the accurate readings. The result is C=1.8653V-694.14, where V is the voltage output, in mV, C is the CO_2 concentration, in ppm.

B.3 Calibration of the Pressure Transducer

Figure B.2 Calibration of the pressure transducer

The pressure transducer was calibrated in the Wind Tunnel Laboratory of the Department of Mechanical Engineering, University of Saskatchewan. The pressure was adjusted from 0 to 0.25 inch of water by increasing 0.05 inch of water step. The result is $P = 0.001V-0.0573$, where P is the pressure, in inch of water, and V is the voltage output, in mV.

APPENDIX C. SAMPLE CALCULATIONS

C.1 Dehumidification Requirement Determinations for the Greenhouse

The average daytime solar radiation of each month was used to calculate the average moisture production rate during daytime, and the corresponding average ambient air temperature and RH were used to determine the ambient air humidity ratio. Tables C.1, C.2 and C.3 provide the average hourly statistics for direct solar radiation, dry bulb temperatures, and relative humidity for Prince Albert, respectively (U.S. Department of Energy, 2011).

As shown in Table C.1, the data filled in the green grids indicate the daytime hourly direct solar radiation for each month, while the data filled in the orange grids indicate the hourly solar radiation around sunset time. In Table C.2 and C.3, the data in the green grids show the corresponding daytime hourly temperatures and RH, while the data in the orange grids demonstrate the corresponding hourly temperatures and RH around sunset time.

As elaborated in Section 4.1.2, dehumidification is mostly needed during daytime and from day to night. Nighttime dehumidification is not required because of little moisture produced at night unless daytime and day-to-night dehumidification is not adequate. Therefore, the dehumidification requirements were only quantified for daytime RH control and day-to-night RH control.

Table C.1 Average hourly direct normal solar radiation of Prince Albert 1953-1995 (U.S. Department of Energy, 2011)

Unit: W/m^2	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0:01- 1:00	0	0	0	0	0	0	0	0	0	0	0	0
1:01- 2:00	0	0	0	0	0	0	0	0	0	0	0	0
2:01- 3:00	0	0	0	0	0	0	0	0	0	0	0	0
3:01- 4:00	0	0	0	0	0	0	0	0	0	0	0	0
4:01- 5:00	0	0	0	0	0	0	0	0	0	0	0	0
5:01- 6:00	0	0	0	0	81	257	137	0	0	0	0	0
6:01- 7:00	0	0	0	160	366	396	376	180	7	0	0	0
7:01- 8:00	0	0	95	401	472	428	534	387	319	18	0	0
8:01- 9:00	0	51	324	497	552	459	562	455	423	361	48	0
9:01-10:00	146	348	373	543	597	523	603	534	476	484	279	100
10:01-11:00	393	465	501	563	574	534	578	559	481	526	325	366
11:01-12:00	433	511	538	577	542	509	589	555	518	562	344	438
12:01-13:00	448	548	541	531	511	499	611	563	521	562	369	454
13:01-14:00	471	600	535	522	501	462	622	528	545	561	394	457
14:01-15:00	467	592	503	537	459	467	582	511	523	558	371	451
15:01-16:00	422	578	454	546	417	460	548	469	484	534	365	387
16:01-17:00	327	493	472	524	428	437	549	460	490	467	269	122
17:01-18:00	20	308	390	520	431	468	558	471	517	275	4	0
18:01-19:00	0	1	255	470	482	461	531	438	352	6	0	0
19:01-20:00	0	0	2	259	403	445	510	317	46	0	0	0
20:01-21:00	0	0	0	0	137	330	345	34	0	0	0	0
21:01-22:00	0	0	0	0	0	0	0	0	0	0	0	0
22:01-23:00	0	0	0	0	0	0	0	0	0	0	0	0
23:01-24:00	0	0	0	0	0	0	0	0	0	0	0	0

Table C.2 Average hourly dry bulb temperatures of Prince Albert 1953-1995 (U.S. Department of Energy, 2011)

Unit: ℃	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0:01- 1:00	-15.9	-14.2	-9.2	0.6	6.7	13	14.8	13.6	7.2	2.6	-7.3	-14.5
1:01- 2:00	-16.2	-14.3	-9.3	0.2	5.8	12.3	14.2	13.2	6.8	2.3	-7.4	-14.8
2:01- 3:00	-16.6	-14.6	-9.8	0	5.3	11.7	13.6	12.6	6.3	1.9	-7.6	-15
3:01- 4:00	-16.5	-14.7	-9.9	-0.4	4.8	11.3	13.1	12.4	5.8	1.7	-7.7	-15.5
4:01- 5:00	-16.5	-14.9	-10.1	-0.8	4.8	10.9	12.4	12.1	5.7	1.4	-7.8	-15.5
5:01- 6:00	-16.5	-15.1	-10.5	-0.2	6.3	11.1	12.3	11.7	5.5	1.2	-8	-15.5
6:01- 7:00	-16.3	-15	-10.8	0.8	8.2	12.4	14.3	12.3	6.6	0.9	-8.1	-15.3
7:01- 8:00	-16.5	-14.9	-10.8	2.3	10.7	13.8	16.3	13.6	8.6	0.9	-7.7	-15.5
8:01- 9:00	-16.9	-14.5	-10.1	3.9	12.9	15.2	18.1	15.4	10.9	1.8	-6.6	-15.4
9:01-10:00	-16.9	-13.3	-9.3	5.4	14.4	16.7	20	16.9	12.6	3.7	-5.4	-15.1
10:01-11:00	-16.5	-12.1	-8	6.5	15.3	17.9	21.3	18.6	13.9	5.6	-4.2	-13.9
11:01-12:00	-15.9	-10.7	-6.7	7.5	16.3	18.8	22.2	19.5	14.9	7.2	-2.8	-12.9
12:01-13:00	-15	-10.1	-5.8	7.9	17.2	19.8	23.1	20.1	15.6	8.7	-2.1	-11.6
13:01-14:00	-14.1	-9.5	-5.1	8.4	17.8	20.5	23.6	20.7	15.9	9.9	-1.8	-11
14:01-15:00	-13.7	-9.5	-4.6	8.6	17.9	20.6	24	20.7	16.1	10.5	-2	-10.7
15:01-16:00	-13.7	-9.9	-4.4	8.6	17.8	20.8	24.2	21	15.8	10.7	-3.2	-11.1
16:01-17:00	-14	-10.5	-4.6	8	17.6	20.7	24.3	20.9	15.3	10.1	-4.5	-11.9
17:01-18:00	-14.3	-11.4	-5	7.1	16.9	20.3	24	20.6	13.5	8.7	-5.6	-12.5
18:01-19:00	-14.8	-11.9	-5.9	5.3	15.5	19.8	23	19.7	11.6	6.9	-6.4	-13
19:01-20:00	-15	-12.5	-6.6	3.7	13.3	18.9	21.9	18.2	10.3	5.6	-6.8	-13.4
20:01-21:00	-15.2	-12.7	-7.1	2.8	11.3	17.2	20	16.8	9.2	5	-7.1	-13.6
21:01-22:00	-15.4	-13.1	-7.4	2.4	10.1	15.8	17.9	15.7	8.5	4.3	-7.1	-13.9
22:01-23:00	-15.5	-13.4	-7.9	2	9.1	14.5	16.7	14.9	7.9	3.4	-7.2	-14.1
23:01-24:00	-15.5	-13.6	-8.1	1.3	7.9	13.7	15.9	14.3	7.4	2.9	-7.6	-14.4

Table C.3 Average hourly RH of Prince Albert 1953-1995 (U.S. Department of Energy, 2011)

Unit: %	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0:01- 1:00	72	81	74	69	66	68	67	76	72	76	80	75
1:01- 2:00	73	82	74	72	69	71	70	77	73	76	80	76
2:01- 3:00	70	82	75	72	71	73	71	79	74	77	80	76
3:01- 4:00	71	81	74	73	72	75	73	80	76	77	81	76
4:01- 5:00	71	81	74	75	73	76	75	81	76	78	82	75
5:01- 6:00	71	81	74	72	69	77	75	83	76	78	82	74
6:01- 7:00	71	80	74	70	63	74	72	82	75	77	81	74
7:01- 8:00	71	79	73	65	55	70	67	78	72	78	79	75
8:01- 9:00	71	78	73	59	47	65	63	73	65	77	77	76
9:01-10:00	71	77	73	52	43	60	56	68	61	73	75	76
10:01-11:00	70	75	72	50	40	56	50	61	56	68	74	75
11:01-12:00	70	75	69	48	38	52	46	57	52	63	69	74
12:01-13:00	69	75	67	47	36	48	43	55	49	58	67	74
13:01-14:00	70	74	67	45	34	45	41	53	48	54	68	71
14:01-15:00	70	75	67	45	35	44	39	53	47	52	67	70
15:01-16:00	69	76	67	45	34	43	38	51	48	51	70	70
16:01-17:00	69	76	67	46	34	43	38	52	50	52	73	72
17:01-18:00	70	77	68	50	35	44	38	52	55	56	75	73
18:01-19:00	72	78	69	57	39	46	41	55	60	60	77	75
19:01-20:00	72	78	71	63	45	48	44	60	64	65	79	76
20:01-21:00	72	79	72	66	50	54	50	64	67	66	79	76
21:01-22:00	71	80	72	67	55	58	56	69	68	70	80	76
22:01-23:00	72	81	72	68	58	64	60	72	70	72	80	76
23:01-24:00	72	81	73	70	61	67	63	73	71	73	82	76

C.1.1 Daytime Dehumidification Requirement

The daytime dehumidification requirement calculation methods for each month are the same. The detailed calculation procedures for October were given as an example.

According to Table C.1, the average daytime direct solar radiation of October (E_D) is calculated to be 409.5 W/m^2 during 7:00-19:00, and the corresponding average daytime ambient air temperature (t_{od}) is 7.1 ℃ (calculated from Table C.2) and RH (RH_{od}) is 61.8% (calculated from Table C.3). The ambient air humidity ratio (W_{od}) and the ambient air density (ρ_{od}) can be found using the Program PLUS (Albright, 1990). The greenhouse inside temperature (t_{id}) was kept at 22 ℃ during daytime and the RH setpoint (RH_{id}) was determined to be 75%. The inside air humidity ratio (W_{id}) and the inside air density (ρ_{id}) can also be found using PLUS (Albright, 1990). The results are: $W_{od} = 0.004\ kg/kg$, $\rho_{od} = 1.19\ kg/m^3$, $W_{id} = 0.013\ kg/kg$, and $\rho_{id} = 1.13\ kg/m^3$.

The calculation procedures are elaborated as follows.

(1) The transmissivity of the greenhouse double-layer cover (τ) is taken to be 0.70 (ASABE, 2008). The floor area of the greenhouse (A_g) is 266.63 m^2. Therefore, the net solar insolation into the greenhouse (E_N) during the daytime of October (7:00-19:00) is:

$$E_N = E_D \times \tau \times A = 409.5\ W/m^2 \times 0.70 \times 266.63\ m^2 = 76429\ W.$$

(2) 25% of the net solar insolation is added to the greenhouse air as latent heat which is used for moisture production (Albright, 1990). So the rate of the latent heat gained during this period (Q_L) is:

$$Q_L = 25\% \times E_N = 25\% \times 76429\ W = 19.11\ kW.$$

(3) The heat vaporization of water (h_{fg}) inside the greenhouse during daytime is: $h_{fg} = 2501 - 2.42t_{id} = (2501 - 2.42 \times 22)kJ/kg = 2447.76\ kJ/kg$. The average daytime moisture production rate of October is:

$$M_p = Q_L/h_{fg} = 19.11\ kW/2447.76\ kJ/kg = 0.008\ kg/s.$$

(4) The required mass ventilation rate (m_{rd}) for daytime RH control is: $m_{rd} = \frac{M_p}{W_{id}-W_{od}} = \frac{0.008\ kg/s}{(0.013-0.004)\ kg/kg} = 0.860\ kg/s$, and the corresponding volumetric ventilation rate (V_{rd}) is: $V_{rd} = \frac{m_{rd}}{\rho_{od}} = \frac{0.860\ kg/s}{1.19\ kg/m^3} = 0.720\ m^3/s$.

C.1.2 Day-to-night Dehumidification Requirement

Day-to-night dehumidification usually occurs around sunset when the solar radiation is close to zero. The moisture production rate during this period is very low such that can be neglected. Therefore, the moisture mass difference between the air of daytime and nighttime together with the intended moisture removal time determine day-to-night dehumidification requirement for each month.

Still taking October as an example, the sunset time is around 19:00 as shown in Table C.1. The corresponding ambient air temperature (t_{on}) is 5.6 ℃ (given in Table C.2) and RH (RH_{on}) is 65% (shown in Table C.3). The greenhouse inside temperature (t_{in}) is kept at 18 ℃ at night and the RH setpoint (RH_{in}) at night is determined to be 75%. Using PLUS (Albright, 1990), the night inside and outside air humidity ratio and density can be found: $W_{on} = 0.004\ kg/kg$, $\rho_{on} = 1.20\ kg/m^3$, $W_{in} = 0.010\ kg/kg$, and $\rho_{in} = 1.150\ kg/m^3$.

The calculation procedures are elaborated as follows.

(1) The volume of the greenhouse (V_g) is calculated to be 880.93 m^3, then the moisture mass need to be removed for day-to-night RH control (M_{rn}) is:

$M_{rn} = V_g \times \rho_{id} \times W_{id} - V_g \times \rho_{in} \times W_{in} = 880.93\ m^3 \times 1.13\ kg/m^3 \times 0.013\ kg/kg - 880.93\ m^3 \times 1.15\ kg/m^3 \times 0.010\ kg/kg = 2.740\ kg.$

(2) If the required moisture removal time (t_{rn}) was taken as 1 h (3600 s), the corresponding required mass ventilation rate for day-to-night RH control (m_{rn}) would be: $m_{rn} = \frac{M_{rn}/t_{rn}}{W_{in} - W_{on}} = \frac{2.740\ kg/3600\ s}{(0.010 - 0.004)\ kg/kg} = 0.120\ kg/s$, and the corresponding required volumetric ventilation rate (V_{rn}) would be: $V_{rn} = \frac{m_{rn}}{\rho_{on}} = \frac{0.120\ kg/s}{1.190\ kg/m^3} = 0.10\ m^3/s$.

C.2 Heat Recovery Ratio (HRR) of Heat Exchangers

The HRR calculation methods for both heat exchangers are the same. The measurement data table and sample calculations for HRR of the RA400 heat exchanger were shown as an example. The period of 6:00 am to 9:00 am, October 13, 2009, were taken for example, the measurement data used for HRR calculation of RA400 heat exchanger are shown in Table C.4.

As shown in Table C.4, t_0 is the ambient temperature, in ℃; RH_0 is the outside RH, in %; t_2 is the inside temperature, in ℃; RH_2 is the inside RH, in %; t_1 is the temperature of the supply air from the heat exchanger entering the greenhouse, in ℃; t_3 is the temperature of the exhaust air leaving the heat exchanger, in ℃; "Index" indicates the treatment applied for the greenhouse ("0" means heat exchanger treatment, "1" indicates dehumidifier treatment, and "2" is control); "Day count" counts the number of the days for the applied treatment ("1" indicates the first day of the treatment, "2" indicates the second day, and "3" is the third day); "RA400 on/off" indicates the operation of RA400 heat exchanger ("0" means off, and "1" means on).

Table C.4 Basic measurements for HRR sample calculations of RA400 heat exchanger (6:00 am-9:00 am, October 13, 2009)

Year	Julian day	Hour	t_0 (℃)	RH_0 (%)	t_2 (℃)	RH_2 (%)	t_1 (℃)	t_3 (℃)	Index	Day count	RA400 on/off
2009	286	600	-8.3	90.4	17.2	74.4	7.4	7.4	0	2	1
2009	286	610			16.8	74.6	7.3	7.4	0	2	1
2009	286	620	-9.0	89.7	16.6	74.5	7.3	8.0	0	2	1
2009	286	630			17.0	75.1	7.8	8.3	0	2	1
2009	286	640	-9.4	89.4	16.9	74.4	7.5	7.8	0	2	1
2009	286	650			16.6	73.9	7.2	7.4	0	2	1
2009	286	700	-9.8	88.8	17.0	74	7.6	7.9	0	2	0
2009	286	710			16.8	74.3	15.3	8.8	0	2	1
2009	286	720	-10.2	88.1	16.7	74.8	7.3	7.7	0	2	1
2009	286	730			16.8	73.5	7.6	7.2	0	2	0
2009	286	740	-10.7	87.7	17.3	75	12.3	8.6	0	2	1
2009	286	750			18.4	74.4	8.9	9.2	0	2	1
2009	286	800	-10.9	87.6	19.1	73.5	12.0	9.5	0	2	0
2009	286	810			19.7	74.6	21.2	15.1	0	2	0
2009	286	820	-10.6	87.6	20.2	74.3	13.7	12.3	0	2	1
2009	286	830			20.3	73.4	9.8	10.4	0	2	1
2009	286	840	-9.8	87.1	20.6	74.2	15.4	11.1	0	2	1
2009	286	850			20.9	74.3	10.4	10.8	0	2	1
2009	286	900	-9.0	86.8	21.3	74.5	10.9	11.3	0	2	1

The HRR calculation procedures are elaborated as follows:

(1) t_0, t_2, and t_1 were averaged respectively during the running period of RA400 heat exchanger. In this example, when “RA400 on/off” equals to 1, the average t_0, t_2, and t_1 are -9.6 ℃, 18.2 ℃, and 9.9 ℃, respectively.

(2) According to Del-Air manufacturer’s calibration results, the mass flow rate of the supply air for RA400 heat exchanger (M_{supply}) is 0.173 kg s^{-1}, which equals to 10.357 kg min^{-1}, or 621.430 kg h^{-1}. The mass flow rate of the exhaust air for RA400 heat exchanger ($M_{exhaust}$) is 0.209 kg s^{-1}, which equals to 12.512 kg min^{-1}, or 750.713 kg h^{-1}.

(3) The HRR of RA400 heat exchanger of this period (6:00 am-9:00 am, October 13, 2009) can be calculated according to Equation 4.4:

$$HRR = \frac{M_{supply}C_{ps}(T_1 - T_0)}{M_{exhaust}C_{pe}(T_2 - T_0)} = \frac{0.173\ kg/s \times [(9.9+273.16)\ K - (-9.6+273.16)K]}{0.209\ kg/s \times [(18.2+273.16)\ K - (-9.6+273.16)K]} = 0.580.$$

The supply air specific heat (C_{ps}) was assumed to be equal to the exhaust air specific heat (C_{pe}), for the difference is extremely small and the ratio of close to 1 can be neglected in this calculation.

C.3 Moisture Removal Index of Heat Exchangers

The MRI calculation methods for both heat exchangers are the same. The measurement data table and sample calculations for MRI of RA400 heat exchanger were shown as an example. The period of 6:00 am to 9:00 am, October 13, 2009, was taken for example, the measurement data used for MRI calculation of RA400 heat exchanger are shown in Table C.5.

Table C.5 Basic measurements for MRI sample calculations of RA400 heat exchanger (6:00 am-9:00 am, October 13, 2009)

Year	Julian day	Hour	t_0 (℃)	RH_0 (%)	t_2 (℃)	RH_2 (%)	t_1 (℃)	t_3 (℃)	Index	Day count	RA400 on/off	RA400 Running Time (min)
2009	286	600	-8.3	90.4	17.2	74.4	7.4	7.4	0	2	1	0
2009	286	610			16.8	74.6	7.3	7.4	0	2	1	10
2009	286	620	-9.0	89.7	16.6	74.5	7.3	8.0	0	2	1	10
2009	286	630			17.0	75.1	7.8	8.3	0	2	1	10
2009	286	640	-9.4	89.4	16.9	74.4	7.5	7.8	0	2	1	10
2009	286	650			16.6	73.9	7.2	7.4	0	2	1	10
2009	286	700	-9.8	88.8	17.0	74	7.6	7.9	0	2	0	0
2009	286	710			16.8	74.3	15.3	8.8	0	2	1	0
2009	286	720	-10.2	88.1	16.7	74.8	7.3	7.7	0	2	1	10
2009	286	730			16.8	73.5	7.6	7.2	0	2	0	0
2009	286	740	-10.7	87.7	17.3	75	12.3	8.6	0	2	1	0
2009	286	750			18.4	74.4	8.9	9.2	0	2	1	10
2009	286	800	-10.9	87.6	19.1	73.5	12.0	9.5	0	2	0	0
2009	286	810			19.7	74.6	21.2	15.1	0	2	0	0
2009	286	820	-10.6	87.6	20.2	74.3	13.7	12.3	0	2	1	0
2009	286	830			20.3	73.4	9.8	10.4	0	2	1	10
2009	286	840	-9.8	87.1	20.6	74.2	15.4	11.1	0	2	1	10
2009	286	850			20.9	74.3	10.4	10.8	0	2	1	10
2009	286	900	-9.0	86.8	21.3	74.5	10.9	11.3	0	2	1	10

As shown in Table C.5, t_0 is the outside temperature, in ℃; RH_0 is the outside RH, in %; t_2 is the inside temperature, in ℃; RH_2 is the inside RH, in %; t_1 is the temperature of the supply air from the heat exchanger entering the greenhouse, in ℃; t_3 is the temperature of the exhaust air leaving the heat exchanger, in ℃; "Index" indicates the treatment applied for the greenhouse ("0" means heat exchanger treatment, "1" indicates dehumidifier treatment, and "2" is control); "Day count" counts the number of the days for the applied treatment ("1" indicates the first day of the treatment, "2" indicates the second day, and "3" is the third day); "RA400 on/off" indicates the operation of RA400 heat exchanger ("0" means off, and "1" means on). "RA400 running time" were calculated based on the columns of "Hour" and "RA400 on/off", and the sum of the values in the column will show the running time of RA400 heat exchanger during this period (in unit of minute).

The MRI calculation procedures are elaborated as follows.

(1) t_0, RH_0, t_2, RH_2, t_1, and t_3 were averaged respectively during the running period of RA400 heat exchanger. Here when "RA400 on/off" equals to 1, the average t_0, RH_0, t_2, RH_2, t_1, and t_3 are -9.6 ℃, 88.4%, 18.2 ℃, 74.4%, 9.9 ℃, and 9.1 ℃, respectively.

(2) The average temperature and RH of the outside air and the inside air were obtained from the first step. The outside air density (ρ_0), humidity ratio of the outside air (W_0), enthalpy of the outside air (h_0), inside air density (ρ_2), humidity ratio of the inside air (W_2), and enthalpy of the inside air (h_2) were found by applying the program PLUS (Albright, 1990), which is the psychrometric look-up substitute. In this example, $\rho_0 = 1.270\ kg/m^3$, $W_0 = 0.002\ kg/kg$, $h_0 = -5.9\ kJ/kg$, $\rho_2 = 1.140\ kg/m^3$, $W_2 = 0.010\ kg/kg$, and $h_2 = 44.3\ kJ/kg$.

(3) The humidity ratio of the supply air from the heat exchanger entering the greenhouse (W_1) should be equal to that of the outside air, for there is only sensible heat exchange between the incoming supply air and the exhaust air at the core of the heat exchanger. So $W_1 = W_0 = 0.002\ kg/kg$. Combined with the temperature of the

supply air from the heat exchanger entering the greenhouse ($t_1 = 9.89$℃), its density (ρ_1) and enthalpy (h_1) were also found using program PLUS. Here $\rho_1 = 1.180\ kg/m^3$, and $h_1 = 13.8\ kJ/kg$.

(4) “RA400 running time” were calculated based on the columns of “Hour” and “RA400 on/off”. The running time (t) was obtained by summing the values in the column of “RA400 running time”. The result shows $t = 110\ min = 1.83\ h$.

(5) The power of RA400 heat exchanger is 0.235 kW based on the manual. So the electrical energy consumption (q_h) is: $q_h = 0.235\ kW \times 1.83\ h = 0.431\ kW \cdot h$.

(6) The net total heat loss through ventilation of the heat exchanger (q_{lh}) was calculated as follows:

$$q_{lh} = \left[M_{exhaust}h_2 - M_{supply}h_1 - \left(M_{exhaust} - M_{supply}\right)h_0\right] \times t = [0.209\ kg/s \times 44.3\ kJ/kg - 0.173\ kg/s \times 13.8\ kJ/kg - (0.209 - 0.173)kg/s \times (-5.9)kJ/kg] \times 1.83\ h = 12.963\ kW \cdot h.$$

(7) The mass of water removed by RA400 heat exchanger (m_h) during this period was calculated as the moisture removed by the exhaust fan minus the moisture coming into the greenhouse through supply air and infiltration, in kg. The calculation is shown as follows.

$$m_h = \left[M_{exhaust}W_2 - M_{supply}W_1 - (M_{exhaust} - M_{supply})W_0\right] \times t = [750.713\ kg/h \times 0.010\ kg/kg - 621.430\ kg/h \times 0.002\ kg/kg - (750.713 - 621.430)\ kg/h \times 0.002\ kg/kg] \times 1.83\ h = 11.953\ kg.$$

So the net volume of the water removed by the heat exchanger (V_h) is: $V_h = \frac{m_h}{\rho_{water}} = \frac{11.953\ kg}{1\ kg/L} = 11.95\ L$.

(8) The MRI of RA400 heat exchanger of this period (6:00 am-9:00 am, October 13, 2009) can be calculated according to Equation 4.5:

$$MRI_h = \frac{q_h + q_{lh}}{V_h} = \frac{0.431\ kW \cdot h + 12.963\ kW \cdot h}{11.95\ L} = 1.12\ kW \cdot h/L.$$

C.4 Moisture Removal Index of the Exhaust Fans

The MRI calculation methods for both exhaust fans (use the data of the exhaust fans of RA400 heat exchanger and RA1000 heat exchanger) are also the same. The sample calculations for MRI of RH-based ventilation using the data of RA400 heat exchanger's exhaust fan were shown as an example. The period of 6:00 am to 9:00 am, October 13, 2009 was taken as an example. The measurement data shown in Table C.5 were used for MRI calculation of the exhaust fan except the columns of t_1 and t_3.

The MRI calculation procedures are elaborated as follows.

(1) The first step is the same as that used for MRI of RA400 heat exchanger calculation. When the exhaust fan was turned on, the average t_0, RH_0, t_2, and RH_2 are -9.6 ℃, 88.4%, 18.2 ℃, 74.4%, respectively.

(2) The second step is also the same as that used for MRI of RA400 heat exchanger calculation. The results are: $\rho_0 = 1.270\ kg/m^3$, $W_0 = 0.002\ kg/kg$, $h_0 = -5.9\ kJ/kg$, $\rho_2 = 1.140\ kg/m^3$, $W_2 = 0.010\ kg/kg$, and $h_2 = 44.3\ kJ/kg$. Thus $h_{e1} = -5.9\ kJ/kg$, and $h_{e2} = 44.3\ kJ/kg$.

(3) The running time of the exhaust fan is the same as that of RA400 heat exchanger, for the exhaust fan was assumed to be controlled based on the same RH setpoints as that for RA400 heat exchanger. Hence, the running time of the exhaust fan (t_e) is: $t_e = 1.83\ h$.

(4) The power of RA400 heat exchanger's exhaust fan is 0.130 kW based on the manual. So the electrical energy consumption of the exhaust fan (q_e) is: $q_e = 0.130\ kW \times 1.83\ h = 0.238\ kW \cdot h$.

(5) The net total heat loss through ventilation of the exhaust fans (q_{le}) during this period can be calculated as follows: $q_{le} = M_e(h_{e2} - h_{e1})t_e = 0.209\ kg/s \times [44.3\ kJ/kg - (-5.9\ kJ/kg)] \times 1.83\ h = 19.200\ kW \cdot$

(6) The mass of the water removed by RA400 heat exchanger's exhaust fan (m_e) during this period is: $m_e = M_{exhaust} \times (W_2 - W_0) \times t = 750.713\ kg/h \times (0.010\ kg/kg - 0.002\ kg/kg) \times 1.83\ h = 11.953\ kg$. So the net volume of the water removed is: $V_e = \frac{m_e}{\rho_{water}} = \frac{11.953\ kg}{1\ kg/L} = 11.953\ L$.

(7) The MRI of the exhaust fan during this period (6:00 am to 9:00 am, October 13, 2009) can be calculated according to Equation 4.7:

$$MRI_e = \frac{q_e + q_{le}}{V_e} = \frac{0.238\ kW \cdot h + 19.200\ kW \cdot h}{11.953\ L} = 1.626\ kW \cdot h/L.$$

C.5 Moisture Removal Index of Dehumidifiers

The MRI for both dehumidifiers can be calculated simultaneously, for both dehumidifiers are of the same model. The measurement data table and sample calculations for MRI of both dehumidifiers were given in this section. The period of 6:00 am to 9:00 am, October 16, 2009 was taken for example, the measurement data used for MRI calculation of the dehumidifiers are shown in Table C.6.

As shown in Table C.6, t_2 is the inside temperature, in ℃; RH_2 is the inside air relative humidity, in %; "Index" indicates the treatment applied for the greenhouse ("0" means heat exchanger treatment, "1" is dehumidifier treatment, and "2" is control); "Day count" counts the number of the days for the applied treatment ("1" indicates the first day of the treatment, "2" indicates the second day, and "3" is the third day); "Dehum on/off" indicates the operation of both dehumidifiers ("0" means off, and "1" means on). "Dehum running time" was calculated based on the columns of "Hour" and "Dehum on/off", and the sum of the values in the column will show the running time of the dehumidifiers during this period (in unit of minute).

Table C.6 Basic measurements for MRI sample calculations of both dehumidifiers (6:00 am-9:00 am, October 16, 2009)

Year	Julian day	Hour	t_2 (℃)	RH_2 (%)	Index	Day count	Dehum on/off	Dehum Running Time
2009	289	600	21.7	75.3	1	2	1	0
2009	289	610	21.8	75.2	1	2	1	10
2009	289	620	21.8	75.3	1	2	1	10
2009	289	630	21.8	75.3	1	2	1	10
2009	289	640	21.8	75.5	1	2	1	10
2009	289	650	21.9	75.5	1	2	1	10
2009	289	700	21.8	75.3	1	2	1	10
2009	289	710	21.8	75.4	1	2	1	10
2009	289	720	21.8	75.7	1	2	1	10
2009	289	730	21.8	76	1	2	1	10
2009	289	740	21.7	76.2	1	2	1	10
2009	289	750	21.7	76.4	1	2	1	10
2009	289	800	21.7	77	1	2	1	10
2009	289	810	21.8	77.3	1	2	1	10
2009	289	820	21.8	77.6	1	2	1	10
2009	289	830	21.9	78.3	1	2	1	10
2009	289	840	22.0	78.6	1	2	1	10
2009	289	850	22.0	78.8	1	2	1	10
2009	289	900	21.5	78.5	1	2	1	10

The MRI calculation procedures for the dehumidifiers are elaborated as follows.

(1) t_2 was averaged during the running period of the dehumidifiers. In this example, when "Dehum on/off" equals to 1, the average t_2 is 21.8 ℃.

(2) The temperature of the condensed water t_w was assumed to be equal to the average room temperature. So $t_w = t_2 = 21.8℃$. So the water heat of vaporization (Albright, 1990) can be calculated: $h_{fg} = 2501 - 2.42t_w = 2501\ kJ/kg - 2.42\ kJ/kg \cdot ℃ \times 21.8℃ = 2448\ kJ/kg$.

(3) "Dehum running time" were calculated based on the columns of "Hour" and "Dehum on/off". The running time of both dehumidifiers (t_d) was obtained by summing the values in the column of "Dehum running time". The result shows $t_d = 180\ min = 3\ h$. The mass of water removed by the dehumidifiers during this period was measured to be 7.8 kg, so the volume of the condensate was: $V_d = \frac{m_{water}}{\rho_{water}} = \frac{7.8\ kg}{1\ kg/L} = 7.8\ L$.

(4) The power of each dehumidifier is 0.668 kW from the manual. So the electrical energy consumption of both dehumidifiers during this period is: $q_d = 0.668\ kW \times 3\ h \times 2 = 4.008\ kW \cdot h$.

(5) The heat output of the dehumidifiers was assumed to be 90% of the electrical energy consumption. Thus, the heat output of both dehumidifiers is: $q_e = 90\% \times q_d = 90\% \times 4.008\ kW \cdot h = 3.607\ kW \cdot h$.

(6) The latent heat (thermal energy) released by the water condensed by both dehumidifiers is: $q_l = \frac{h_{fg} \times m_{water}}{3600} = \frac{2448\ kJ/kg \times 7.8\ kg}{3600\ s/h} = 5.304\ kW \cdot h$.

(7) The MRI of the dehumidifiers of this period (6:00 am-9:00 am, October 16, 2009) can be calculated according to Equation 4.6:

$$MRI_d = \frac{q_d-(q_o+q_l)}{V_d} = \frac{4.008\ kW\cdot h-(3.607\ kW\cdot h+5.304\ kW\cdot h))}{7.8\ L} = -0.629\ kW \cdot h/L.$$

(8) In order to better understand this value and make it easily comparable with the other treatments, the modified moisture removal index values were also calculated for the dehumidifiers. First, MRI1 was calculated by dividing the electrical energy consumption of the dehumidifiers with the total volume of water removed. For this example, $MRI1 = \frac{q_d}{V_d} = \frac{4.008\ kW\cdot h}{7.8\ L} = 0.514\ kW \cdot h/L$.

The result is 0.514 kW-h/L, which means the dehumidifiers consumed 0.514 kW-h electrical energy for 1 L water removed.

Second, MRI2 was calculated by dividing the sum of the heat output of the dehumidifiers and latent heat released by the condensed water by the total condensed water volume. In this example, $MRI2 = \frac{-(q_o+q_l)}{V_d} = \frac{-(3.607\ kW\cdot h+5.304\ kW\cdot h)}{7.8\ L} = -1.142\ kW \cdot h/L$.

The result is is -1.142 kW-h/L, which means the dehumidifiers released 1.142 kW-h thermal energy by removing 1 L of water vapour.

C.6 Moisture Removal Index of the Finned Tubing

The period of 12:21 pm to 13:21 pm, April 22, 2009 was taken for example, the measurement data used for MRI calculation of the finned tubing are shown in Table C.7. Each parameter shown in Table C.7 was monitored every second and recorded every minute. The condensed water (m_w) during this period was weighted to be 176.73 g.

Table C.7 Basic measurements for MRI sample calculations of finned tubing (12:21 pm-13:21 pm, April 22, 2009)

Year	Julian day	Hour	T_{ia} (℃)	RH (%)	T_{wa} (℃)	T_{out} (℃)	T_{in} (℃)	FR_a (L/s)
2009	112	1221	19.3	90	3.9	5.4	4.7	0.844
2009	112	1222	19.2	90	4.1	5.6	4.9	0.847
2009	112	1223	19.2	90	4.4	5.8	5.1	0.847
2009	112	1224	19.1	90	4.5	6.0	5.2	0.847
2009	112	1225	19.2	90	4.4	5.8	5.2	0.844
2009	112	1226	19.2	90	4.1	5.5	4.8	0.847
2009	112	1227	19.1	90	4.2	5.7	5.0	0.847
2009	112	1228	19.2	90	4.4	5.8	5.1	0.847
2009	112	1229	19.2	90	4.5	6.0	5.3	0.850
2009	112	1230	19.2	90	3.8	5.5	4.8	0.850
2009	112	1231	19.2	90	3.8	5.4	4.6	0.847
2009	112	1232	19.3	90	4.0	5.6	4.8	0.847
2009	112	1233	19.4	90	4.2	5.7	5.0	0.847
2009	112	1234	19.4	90	4.4	5.9	5.2	0.847
2009	112	1235	19.4	90	4.6	6.1	5.3	0.847
2009	112	1236	19.4	90	4.2	5.8	5.1	0.847
2009	112	1237	19.4	90	4.1	5.6	4.9	0.847
2009	112	1238	19.4	90	4.2	5.7	5.0	0.847
2009	112	1239	19.4	90	4.3	5.8	5.1	0.847
2009	112	1240	19.4	90	4.4	5.9	5.2	0.850
2009	112	1241	19.4	90	4.2	5.6	5.0	0.844
2009	112	1242	19.4	90	3.9	5.4	4.7	0.847
2009	112	1243	19.4	90	4.0	5.5	4.8	0.847
2009	112	1244	19.4	90	4.1	5.6	4.9	0.850
2009	112	1245	19.4	90	4.4	5.8	5.1	0.85
2009	112	1246	19.4	90	4.5	6.0	5.3	0.847
2009	112	1247	19.4	90	4.4	5.9	5.3	0.850
2009	112	1248	19.4	90	4.1	5.6	4.9	0.847
2009	112	1249	19.4	90	4.2	5.6	4.9	0.850
2009	112	1250	19.3	90	4.3	5.8	5.0	0.847
2009	112	1251	19.2	90	4.5	6.0	5.3	0.853
2009	112	1252	19.1	90	4.4	5.9	5.3	0.847
2009	112	1253	19.1	90	4.1	5.5	4.8	0.847
2009	112	1254	19.1	90	4.1	5.5	4.9	0.847
2009	112	1255	19.0	90	4.4	5.8	5.1	0.850
2009	112	1256	19.2	90	4.4	5.9	5.2	0.847

2009	112	1257	19.2	90	4.6	6.0	5.4	0.850
2009	112	1258	19.3	90	4.2	5.7	5.0	0.850
2009	112	1259	19.2	90	4.0	5.5	4.8	0.850
2009	112	1300	19.3	90	4.1	5.6	4.9	0.850
2009	112	1301	19.2	90	4.4	5.8	5.1	0.847
2009	112	1302	19.3	90	4.4	5.9	5.2	0.847
2009	112	1303	19.3	90	4.7	6.1	5.4	0.847
2009	112	1304	19.4	90	4.4	6.0	5.4	0.850
2009	112	1305	19.3	90	4.1	5.5	4.9	0.850
2009	112	1306	19.3	90	4.1	5.6	4.9	0.850
2009	112	1307	19.3	90	4.3	5.8	5.1	0.847
2009	112	1308	19.3	90	4.6	5.9	5.3	0.847
2009	112	1309	19.2	90	4.6	6.1	5.4	0.850
2009	112	1310	19.3	90	4.1	5.6	4.9	0.847
2009	112	1311	19.2	90	4.1	5.6	4.9	0.850
2009	112	1312	19.2	90	4.2	5.7	5.0	0.847
2009	112	1313	19.2	90	4.4	5.8	5.1	0.850
2009	112	1314	19.3	90	4.6	6.0	5.3	0.850
2009	112	1315	19.3	90	4.0	5.5	4.8	0.847
2009	112	1316	19.3	90	4.0	5.5	4.8	0.850
2009	112	1317	19.2	90	4.1	5.6	4.9	0.847
2009	112	1318	19.3	90	4.4	5.8	5.1	0.850
2009	112	1319	19.3	90	4.5	5.9	5.3	0.850
2009	112	1320	19.3	90	4.6	6.1	5.4	0.850
2009	112	1321	19.2	90	4.7	6.1	5.4	0.847

Note: T_{ia} is the measured room temperature, in ℃; RH is the designed room RH, in %; T_{wa} is the measured reserve water temperature, in ℃; T_{out} is the measured outlet water temperature, in ℃; T_{in} is the measured inlet water temperature, in ℃; FR_a is measured water flow rate, in L/s.

The MRI calculation procedures for the finned tubing are elaborated as follows.

(1) T_{ia}, T_{wa}, T_{out}, T_{in}, and FR_a was averaged during the period of 12:21 pm to 13:21 pm, April 22, 2009, respectively. The results are: $T_{ia} = 19.3$℃, $T_{wa} =$ 19.3℃, $T_{out} = 5.8$℃, $T_{in} = 5.1$℃, and $FR_a = 0.848\ L/s$.

(2) The electrical energy consumption of the pump (q_c) was calculated using the power of the pump (0.523 kW according to the manual) multiplied by the

running time (1 hour). The result for this example is: $q_c = 0.523\ kW \times 1\ h = 0.523\ kW \cdot h$.

(3) Since $FR_a = 0.848\ L/s$, the mass flow rate of the chilled water (M) is: $M = 0.848\ L/s \times 1\ kg/L = 0.848\ kg/s$. The specific heat of the water C_p is 4.187 kJ kg^{-1} K^{-1}. The temperature difference between the inlet and outlet water (ΔT) is: $\Delta T = T_{out} - T_{in} = (5.8 - 5.1)\ K = 0.7\ K$. The running time of the pumpt_c is 1 h. The heat loss from the room, which was absorbed by the chilled water during the heat transfer process (q_{lossc}) was calculated as follows: $q_{lossc} = MC_p\Delta T t_c = 0.848\ kg/s \times 4.187\ kJ/kg \cdot K \times 0.7\ K \times 1\ h = 2.485\ kW \cdot h$.

(4) The temperature of the condensed water T_w was assumed equal to the average room temperature. So $T_w = T_{ia} = 19.3$℃. The mass of the water condensed by the finned tubing during this priod is: $m_w = 176.73\ g = 0.177\ kg$. So the water heat of vaporization (Albright, 1990) can be calculated as follows:

$$h_{fg} = 2501 - 2.42 t_w = 2501\ kJ/kg - 2.42\ kJ/kg \cdot ℃ \times 19.3℃ = 2454\ kJ/kg.$$

The latent heat (thermal energy) released by the water condensed by the finned tubing is: $q_{lc} = \frac{h_{fg} \times m_w}{3600} = \frac{2454\ kJ/kg \times 0.177\ kg}{3600\ s/h} = 0.120\ kW \cdot h$.

(5) The volume of the condensate (V_c) is: $V_c = \frac{m_w}{\rho_{water}} = \frac{0.177\ kg}{1\ kg/L} = 0.177\ L$. The MRI of the finned tubing condensation system (MRI_c) during this period (12:21 pm-13:21 pm, April 22, 2009) can be calculated according to Equation 4.2.

$$MRI_c = \frac{q_c + q_{lossc} - q_{lc}}{V_c} = \frac{0.523\ kW \cdot h + 2.485\ kW \cdot h - 0.120\ kW \cdot h}{0.177\ L} = 16.316\ kW \cdot h/L.$$

C.7 Energy Cost Estimation

As stated in Chapter 4, energy cost is defined as the total energy cost per liter of water removed (in unit of $/L). It was assumed that the energy sources, except the electrical energy, were from the fuel (natural gas or thermal coal) that used to heat the greenhouse. The efficiency of the boiler for the greenhouse was assumed to be 70%, and the transportation heat loss was neglected because of the short distance and well insulated hot water pipe between the boiler room and the greenhouse.

The electrical energy rate for the farm was found to be \$0.09722/ kW-h on SaskPower website (SaskPower, 2009). The price of natural gas was taken as \$6.81/GJ, which equals to \$0.025/kW-h (SaskEnergy, 2009). The heating efficiency of natural gas was estimated to be 90%, so the price of natural gas was \$0.028/kW-h considering the combustion efficiency. The price of the thermal coal was given as \$0.06/kg on the website of Natural Resources Canada (Natural Resources Canada, 2009). The heating value of the thermal coal is roughly 24 MJ/kg (Fisher, 2003). Because 1 kW-h is 3.6 MJ, then the energy density of thermal coal is 6.67 kW ħ/kg. The efficiency of the boiler was assumed to be 70%, so of the 6.67 kW-h of energy per kilogram of thermal coal, 70% of that -- 4.67 kW-h/kg -- can successfully be turned into thermal energy.

C.7.1 Energy Cost Estimation for the Heat Exchangers

The energy cost of the heat exchangers consisted of electrical energy cost and extra heating cost resulting from the heat loss during the ventilation process. The cost of electrical energy was calculated using the latest electricity rate multiplied by the running time of heat exchangers. The extra heating cost was estimated based on the fule (natural gas or thermal coal) price. The calculation for RA400 heat exchanger during the period of 6:00 am to 9:00 am, October 13, 2009 was taken for example.

Based on the results given in Appendix C.3, the net volume of the water removed by RA400 heat exchanger (V_h) is: $V_h = 11.953\ L$; the electrical energy consumption during this period (q_h) is: $q_h = 0.431\ kW \cdot h$; the net total heat loss through ventilation of the heat exchanger (q_{lh}) is: $q_{lh} = 12.963\ kW \cdot h$.

So the electrical energy cost of RA400 heat exchanger during this period (S_h) is: $S_h = 0.431\ kW \cdot h \times \$0.097/kW \cdot h = \$0.042$. The natural gas cost resulted from the heat loss (S_{lhg}) is: $S_{lhg} = 12.963\ kW \cdot h \times \$0.028/kW \cdot h = \$0.363$. The thermal coal cost resulted from the heat loss (S_{lhc}) is: $S_{lhc} = \frac{12.963\ kW \cdot h}{4.67\ kW \cdot h/kg} \times \$0.06/kg = \$0.167$.

Therefore, if natural gas was used as the heat source, the energy cost for RA400 heat exchanger during the period of 6:00 am to 9:00 am, October 13, 2009 (S_{lg}) is: $S_{lg} = \frac{S_h + S_{lhg}}{V_h} = \frac{\$0.042 + \$0.363}{11.953\ L} = \$0.034/L$; if thermal coal was used as the heat source, the energy cost for RA400 heat exchanger during the period of 6:00 am to 9:00 am, October 13, 2009 (S_{lc}) is: $S_{lc} = \frac{S_h + S_{lhc}}{V_h} = \frac{\$0.042 + \$0.167}{11.953\ L} = \$0.017/L$.

C.7.2 Energy Cost Estimation for the Exhaust Fans

The energy cost of the exhaust fans consisted of electrical energy cost and extra heating cost resulting from the heat loss during the ventilation process. The cost of electrical energy was calculated using the latest electricity rate multiplied by the running time of heat exchangers. The extra heating cost was estimated based on the thermal coal price. The calculations of the exhaust fan (use the data of RA400 heat exchanger's exhaust fan) during the period of 6:00 am to 9:00 am, October 13, 2009 were shown as an example.

Based on the results given in Appendix C.4, the net volume of the water removed by the exhaust fan (V_e) is: $V_e = 11.953\ L$; the electrical energy consumption during this period (q_e) is: $q_e = 0.238\ kW \cdot h$; the net total heat loss through ventilation of the exhaust fan (q_{le}) is: $q_{le} = 19.200\ kW \cdot h$.

So the electrical energy cost of the exhaust fan during this period (S_e) is: $S_e = 0.238\ kW \cdot h \times \$0.097/kW \cdot h = \$0.023$. The natural gas cost resulted from the heat loss through the ventilation of the exhaust fan (S_{leg}) is: $S_{leg} = 19.200\ kW \cdot h \times \$0.028/kW \cdot h = \$0.538$. The thermal coal cost resulted from the heat loss through the ventilation of the exhaust fan (S_{lec}) is: $S_{lec} = \frac{19.200\ kW \cdot h}{4.67\ kW \cdot h/kg} \times \$0.06/kg = \$0.247$.

Therefore, if natural gas was used as the heat source, the energy cost of the exhaust fan during the period of 6:00 am to 9:00 am, October 13, 2009, (S_{2g}) is: $S_{2g} = \frac{S_e + S_{leg}}{V_e} = \frac{\$0.023 + \$0.538}{11.953\ L} = \$0.047/L$; if thermal coal was used as the heat source, the energy cost of the exhaust fan during the period of 6:00 am to 9:00 am, October 13, 2009, (S_{2c}) is: $S_{2c} = \frac{S_e + S_{lec}}{V_e} = \frac{\$0.023 + \$0.247}{11.953\ L} = \$0.023/L$.

C.7.3 Energy Cost Estimation for the Dehumidifiers

For dehumidifiers, energy consumption was mainly from electrical energy used. The heat output of the dehumidifiers and the latent heat released by the condensed water compensated some of the energy used. Since it was assumed that the energy sources, except the electrical energy, were from thermal coal, then the energy cost of the dehumidifiers was calculated by subtracting the thermal coal cost from the electrical energy cost. The calculations during the period of 6:00 am to 9:00 am, October 16, 2009, were taken as an example.

Based on the results given in Appendix C.5, the net volume of the water removed by the dehumidifiers (V_d) is: $V_d = 7.8\ L$; the electrical energy consumption during this period (q_d) is: $q_d = 4.008\ kW \cdot h$; the heat output of the dehumidifiers (q_o) is: $q_o = 3.607\ kW \cdot h$; the latent heat released by the condensate from the dehumidifiers is: $q_l = 5.304\ kW \cdot h$.

So the electrical energy cost of the dehumidifiers during this period (S_d) is: $S_d = 4.008\ kW \cdot h \times \$0.097/kW \cdot h = \$0.390$. The natural gas cost of the heat gained (S_{og}) is: $S_{og} = -(3.607\ kW \cdot h + 5.304\ kW \cdot h) \times \$0.028/kW \cdot h = -\$0.250$. The thermal coal cost of the heat gained (S_{oc}) is: $S_{oc} = \frac{-(3.607\ kW \cdot h + 5.304\ kW \cdot h)}{4.67\ kW \cdot h/kg} \times \frac{\$0.06}{kg} = -\$0.114$.

Therefore, if natural gas was used as the heat source, the energy cost of the dehumidifiers during the period of 6:00 am to 9:00 am, October 16, 2009, (S_{3g}) is: $S_{3g} = \frac{S_d + S_{og}}{V_d} = \frac{\$0.390 - \$0.250}{7.8\,L} = \$0.018/L$; if thermal coal was used as the heat source, the energy cost of the dehumidifiers during the period of 6:00 am to 9:00 am, October 16, 2009, (S_{3c}) is: $S_{3c} = \frac{S_d + S_{oc}}{V_d} = \frac{\$0.390 - \$0.114}{7.8\,L} = \$0.035/L$.

C.7.4 Energy Cost Estimation for the Finned Tubing Condensation System

The energy cost of the chilled water system was estimated by the electrical energy usage of the pump, and the thermal coal consumption resulted from the heat loss and the latent heat released. The calculations during the period of 12:21 pm to 13:21 pm, April 22, 2009, were taken as an example.

Based on the results given in Appendix C.6, the net volume of the water removed by the finned tubing (V_c) is: $V_c = 0.177\,L$; the electrical energy consumption of the pump during this period (q_c) is: $q_c = 0.523\ kW \cdot h$; the heat loss from the room, which was absorbed by the chilled water during the heat transfer process (q_{lossc}) is: $q_{lossc} = 2.485\ kW \cdot h$; the latent heat released by the condensate from the finned tubing (q_{lc}) is: $q_{lc} = 0.120\ kW \cdot h$.

So the electrical energy cost of the condensation system during this period (S_c) is: $S_c = 0.523\ kW \cdot h \times \$0.097/kW \cdot h = \$0.051$. The natural gas cost resulted from the heat loss (S_{lossg}) is: $S_{lossg} = 2.485\ kW \cdot h \times \$0.028 = \$0.070$. The natural gas cost of the latent heat gained (S_{lcg}) is: $S_{lcg} = -0.120\ kW \cdot h \times \$0.028 = -\$0.003$. The thermal coal cost resulted from the heat loss (S_{lossc}) is: $S_{lossc} = \frac{2.485\ kW \cdot h}{4.67\ kW \cdot h/kg} \times \$0.06/kg = \$0.032$. The coal cost of the latent heat gained (S_{lc}) is: $S_{lc} = \frac{-0.120\ kW \cdot h}{4.67\ kW \cdot h/kg} \times \$0.06/kg = -\$0.002$.

Therefore, if natural gas was used as the heat source, the energy cost of the finned tubing condensation system using chilled water during the period of 12:21 pm to 13:21 pm, April 22, 2009, (S_{4g}) is: $S_{4g} = \frac{S_c+S_{lossg}+S_{lcg}}{V_c} = \frac{\$0.051+\$0.070-\$0.003}{0.177\ L} = \$0.667/L$; if thermal coal was used as the heat source, the energy cost of the finned tubing condensation system using chilled water during the period of 12:21 pm to 13:21 pm, April 22, 2009, (S_{4c}) is: $S_{4c} = \frac{S_c+S_{lossc}+S_{lc}}{V_c} = \frac{\$0.051+\$0.032-\$0.002}{0.177\ L} = \$0.457/L$.

C.8 Error Estimation for Moisture Removal Index

The calculations based on the experiment can never avoid errors including systematic errors (non-random-imperfections mainly caused by poor adjustment of instruments) and random errors in measurements and observations. Since these errors inevitably exist, rough estimations for errors of MRI should be given based on instrument measurement accuracy and common sense, in order to provide a proper range to MRI that make it more reliable and convinced. Table C.8 provides rough error estimations for each component of MRI.

Table C.8 Individual errors for each component of the treatment

ΔMRI_h	ΔMRI_e	ΔMRI_d	ΔMRI_c
$\Delta q_h = \pm 2\% q_h$	$\Delta q_e = \pm 2\% q_e$	$\Delta q_d = \pm 2\% q_d$	$\Delta q_c = \pm 1\% q_c$
$\Delta q_{lh} = \pm 3\% q_{lh}$	$\Delta q_{le} = \pm 3\% q_{le}$	$\Delta q_o = \pm 5\% q_o$	$\Delta q_{lossc} = \pm 5\% q_{lossc}$
$\Delta V_h = \pm 3\% V_h$	$\Delta V_e = \pm 3\% V_e$	$\Delta q_l = \pm 0.07\ kW \cdot h$	$\Delta q_{lc} = 0$
		$\Delta V_d = \pm 0.1\ L$	$\Delta V_c = 0$

C.8.1 Error Estimation for MRI of Heat Exchangers

The calculations for RA400 heat exchanger during the period of 6:00 am to 9:00 am, October 13, 2009, were taken for example. Based on the results given in Appendix C.3, the net volume of the water removed by RA400 heat exchanger (V_h) is: $V_h = 11.953\ L$; the electrical energy consumption during this period (q_h) is: $q_h = 0.431\ kW \cdot h$; the net total heat loss through ventilation of the heat exchanger (q_{lh}) is: $q_{lh} = 12.963\ kW \cdot h$.

The individual errors were calculated as follows:

(1) Electrical energy consumption of the heat exchanger ($\Delta\mu_{h1}$):

$$\Delta\mu_{h1} = \frac{\partial MRI_h}{\partial q_h} \times \Delta q_h = \frac{1}{V_h} \times \Delta q_h = \frac{1}{11.953\ L} \times 2\% \times 0.431\ kW \cdot h = 0.001\ kW \cdot h/L.$$

(2) Net heat loss through ventilation of the heat exchanger ($\Delta\mu_{h2}$):

$$\Delta\mu_{h2} = \frac{\partial MRI_h}{\partial q_{lh}} \times \Delta q_{lh} = \frac{1}{V_h} \times \Delta q_{lh} = \frac{1}{11.953\ L} \times 3\% \times 12.963\ kW \cdot h = 0.033\ kW \cdot h/L.$$

(3) Net volume of water removed by the heat exchanger ($\Delta\mu_{h3}$):

$$\Delta\mu_{h3} = \frac{\partial MRI_h}{\partial V_h} \times \Delta V_h = -\frac{1}{{V_h}^2} \times (q_h + q_{lh}) \times \Delta V_h = -\frac{1}{11.953\ L \times 11.953\ L} \times (0.431 + 12.963) kW \cdot h \times 3\% \times 11.953\ L = -0.034\ kW \cdot h/L.$$

So the overall error for the MRI of RA400 heat exchanger (ΔMRI_h) during the period of 6:00 am to 9:00 am, October 13, 2009, is:

$$\Delta MRI_h = \sqrt{{\Delta\mu_{h1}}^2 + {\Delta\mu_{h2}}^2 + {\Delta\mu_{h3}}^2} = \sqrt{(0.001)^2 + (0.033)^2 + (-0.034)^2}\ kW \cdot h = 0.047\ kW \cdot h.$$

Percent error is: $\frac{\Delta MRI_h}{MRI_h} \times 100\% = 4.2\%$.

C.8.2 Error Estimation for MRI of Exhaust Fans

The calculations for exhaust fan (use the data of RA400 heat exchanger's exhaust fan) during the period of 6:00 am to 9:00 am, October 13, 2009, were taken for example. Based on the results given in Appendix C.4, the net volume of the water removed by the exhaust fan (V_e) is: $V_e = 11.953\ L$; the electrical energy consumption of the exhaust fan during this period (q_e) is: $q_e = 0.238\ kW \cdot h$; the net total heat loss through ventilation of the exhaust fan (q_{le}) is: $q_{le} = 19.200\ kW \cdot h$.

The individual errors were calculated as follows:

(1) Electrical energy consumption of the exhaust fan ($\Delta\mu_{e1}$):

$$\Delta\mu_{e1} = \frac{\partial MRI_e}{\partial q_e} \times \Delta q_e = \frac{1}{V_e} \times \Delta q_e = \frac{1}{11.953\ L} \times 2\% \times 0.238\ kW \cdot h = 0.0004\ kW \cdot h/L.$$

(2) Net heat loss through ventilation of the exhaust fan ($\Delta\mu_{e2}$):

$$\Delta\mu_{e2} = \frac{\partial MRI_e}{\partial q_{le}} \times \Delta q_{le} = \frac{1}{V_e} \times \Delta q_{le} = \frac{1}{11.953\ L} \times 3\% \times 19.200\ kW \cdot h = 0.048\ kW \cdot h/L.$$

(3) Net volume of water removed by the exhaust fan ($\Delta\mu_{e3}$):

$$\Delta\mu_{e3} = \frac{\partial MRI_e}{\partial V_e} \times \Delta V_e = -\frac{1}{{V_e}^2} \times (q_e + q_{le}) \times \Delta V_e = -\frac{1}{11.953\ L \times 11.953\ L} \times (0.238 + 19.200)\ kW \cdot h \times 3\% \times 11.953\ L = -0.049\ kW \cdot h/L.$$

Then the overall error for the MRI of the exhaust fan (ΔMRI_e) during the period of 6:00 am to 9:00 am, October 13, 2009, is:

$$\Delta MRI_e = \sqrt{{\Delta\mu_{e1}}^2 + {\Delta\mu_{e2}}^2 + {\Delta\mu_{e3}}^2} = \sqrt{(0.004)^2 + (0.048)^2 + (-0.049)^2}\ kW \cdot h = 0.069\ kW \cdot h.$$

Percent error is: $\frac{\Delta \mathrm{MRI_e}}{\mathrm{MRI_e}} \times 100\% = \frac{0.069\ \mathrm{kW{\cdot}h}}{1.626\ \mathrm{kW{\cdot}h}} \times 100\% = 4.2\%$.

C.8.3 Error Estimation for MRI of Dehumidifiers

The error calculations for the dehumidifiers during the period of 6:00 am to 9:00 am, October 16, 2009, were shown as an example. Based on the results given in Appendix C.5, the net volume of the water removed by the dehumidifiers (V_d) is: $V_d = 7.8\ L$; the electrical energy consumption during this period (q_d) is: $q_d = 4.008\ kW \cdot h$; the heat output of the dehumidifiers (q_o) is: $q_o = 3.607\ kW \cdot h$; the latent heat released by the condensate from the dehumidifiers is: $q_l = 5.304\ kW \cdot h$.

The individual errors were calculated as follows:

(1) Electrical energy consumption of the dehumidifiers ($\Delta\mu_{d1}$):

$$\Delta\mu_{d1} = \frac{\partial MRI_d}{\partial q_d} \times \Delta q_d = \frac{1}{V_d} \times \Delta q_d = \frac{1}{7.8\ L} \times 2\% \times 4.008\ kW \cdot h = 0.010\ kW \cdot h/L.$$

(2) Heat output of the dehumidifiers ($\Delta\mu_{d2}$):

$$\Delta\mu_{d2} = \frac{\partial MRI_d}{\partial q_o} \times \Delta q_o = -\frac{1}{V_d} \times \Delta q_o = -\frac{1}{7.8\ L} \times 5\% \times 3.607\ kW \cdot h = -0.023\ kW \cdot h/L.$$

(3) Latent heat released by the condensate ($\Delta\mu_{d3}$):

$$\Delta\mu_{d3} = \frac{\partial MRI_d}{\partial q_l} \times \Delta q_l = -\frac{1}{V_d} \times \Delta q_l = -\frac{1}{7.8\ L} \times 0.07\ kW \cdot h = -0.009\ kW \cdot h/L.$$

(4) Net volume of water removed by the dehumidifiers ($\Delta\mu_{d4}$):

$$\Delta\mu_{d4} = \frac{\partial MRI_d}{\partial V_d} \times \Delta V_d = -\frac{1}{V_d{}^2} \times (q_d - q_o - q_l) \times \Delta V_d = -\frac{1}{7.8\ L \times 7.8\ L} \times (4.008 - 3.607 - 5.304) kW \cdot h \times 0.01 L = -0.0008\ kW \cdot h/L.$$

Then the overall error for the MRI of the dehumidifiers (ΔMRI_d) during the period of 6:00 am to 9:00 am, October 16, 2009, is:

$$\Delta MRI_d = \sqrt{\Delta\mu_{d1}{}^2 + \Delta\mu_{d2}{}^2 + \Delta\mu_{d3}{}^2 + \Delta\mu_{d4}{}^2} = \sqrt{(0.010)^2 + (-0.023)^2 + (-0.009)^2 + (-0.0008)^2}\ kW \cdot h = 0.027\ kW \cdot h.$$

Percent error is: $\frac{\Delta \mathrm{MRI_d}}{\mathrm{MRI_d}} \times 100\% = \frac{0.027\ \mathrm{kW \cdot h}}{0.629\ \mathrm{kW \cdot h}} \times 100\% = 4.3\%.$

C.8.4 Error Estimation for MRI of Finned Tubing Condensation System

The error calculations for the MRI of the finned tubing during the period of 12:21 pm to 13:21 pm, April 22, 2009, were given as an example. Based on the results given in Appendix C.6, the net volume of the water removed by the finned tubing (V_c) is: $V_c = 0.177\ L$; the electrical energy consumption of the pump during this period (q_c) is: $q_c = 0.523\ kW \cdot h$; the heat loss from the room, which was absorbed by the chilled water during the heat transfer process (q_{lossc}) is: $q_{lossc} = 2.485\ kW \cdot h$; the latent heat released by the condensate from the finned tubing (q_{lc}) is: $q_{lc} = 0.120\ kW \cdot h$.

The individual errors were calculated as follows:

(1) Electrical energy consumption of the pump ($\Delta\mu_{c1}$):

$$\Delta\mu_{c1} = \frac{\partial MRI_c}{\partial q_c} \times \Delta q_c = \frac{1}{V_c} \times \Delta q_c = \frac{1}{0.177\ L} \times 1\% \times 0.523\ kW \cdot h = 0.030\ kW \cdot h/L.$$

(2) Heat loss from room to the chilled water ($\Delta\mu_{c2}$):

$$\Delta\mu_{c2} = \frac{\partial MRI_c}{\partial q_{lossc}} \times \Delta q_{lossc} = \frac{1}{V_c} \times \Delta q_{lossc} = \frac{1}{0.177\ L} \times 5\% \times 2.485\ kW \cdot h = 0.702\ kW \cdot h/L.$$

(3) Latent heat released by the condensate ($\Delta\mu_{c3}$):

$$\Delta\mu_{c3} = \frac{\partial MRI_c}{\partial q_{lc}} \times \Delta q_{lc} = -\frac{1}{V_c} \times \Delta q_{lc} = -\frac{1}{7.8\ L} \times 0 = 0\ kW \cdot h.$$

(4) Net volume of water removed by the finned tubing ($\Delta\mu_{c4}$):

$$\Delta\mu_{c4} = \frac{\partial MRI_c}{\partial V_c} \times \Delta V_c = -\frac{1}{{V_c}^2} \times (q_c + q_{lossc} - q_{lc}) \times \Delta V_c = -\frac{1}{0.177\ L \times 0.177\ L} \times (0.523 + 2.485 - 0.120)\ kW \cdot h \times 0\ L = 0\ kW \cdot h/L.$$

Then the overall error for the MRI of the dehumidifiers (ΔMRI_c) during the period of 12:21 pm to 13:21 pm, April 22, 2009, is:

$$\Delta MRI_d = \sqrt{{\Delta\mu_{c1}}^2 + {\Delta\mu_{c2}}^2 + {\Delta\mu_{c3}}^2 + {\Delta\mu_{c4}}^2} = \sqrt{(0.030)^2 + (0.702)^2 + (0)^2 + (0)^2}\ kW \cdot h = 0.703\ kW \cdot h.$$

Percent error is: $\frac{\Delta \mathrm{MRI_c}}{\mathrm{MRI_c}} \times 100\% = \frac{0.703\ \mathrm{kW \cdot h}}{16.316\ \mathrm{kW \cdot h}} \times 100\% = 4.3\%$

APPENDIX D. CR10X DATA LOGGER PROGRAMMING

D.1 Program for Condensation Experiment

```
;{CR10X}
*Table 1 Program
01: 1        Execution Interval (seconds)
;Compare Switch to store data
1:  Excite-Delay (SE) (P4)
 1: 1        Reps
 2: 5        2500 mV Slow Range
 3: 12       SE Channel
 4: 3        Excite all reps w/Exchan 3
 5: 0000     Delay (0.01 sec units)
 6: 2500     mV Excitation
 7: 1        Loc [Switch]
 8: 1.0      Multiplier
 9: 0.0      Offset
2:  If (X<=>F) (P89)
 1: 1        X Loc [Switch]
 2: 3        >=
 3: 2000     F
 4: 10       Set Output Flag High (Flag 0)
;Measure four RTD's
3:  3W Half Bridge (P7)
 1: 1        Reps
 2: 23       25 mV 60 Hz Rejection Range
 3: 1        SE Channel
 4: 1        Excite all reps w/Exchan 1
 5: 2100     mV Excitation
 6: 2        Loc [RTD_1]
 7: 98.50    Multiplier
 8: 0.0      Offset
4:  Temperature RTD (P16)
 1: 1        Reps
 2: 2        R/R0 Loc [RTD_1]
 3: 6        Loc [Temp_1]
 4: 1.0718   Multiplier
 5: -0.7801  Offset
5:  3W Half Bridge (P7)
 1: 1        Reps
 2: 23       25 mV 60 Hz Rejection Range
 3: 3        SE Channel
 4: 1        Excite all reps w/Exchan 1
 5: 2100     mV Excitation
 6: 3        Loc [RTD_2]
 7: 98.47    Multiplier
 8: 0.0      Offset
6:  Temperature RTD (P16)
 1: 1        Reps
```

2: 3 R/R0 Loc [RTD_2]
3: 7 Loc [Temp_2]
4: 1.057 Multiplier
5: -0.1089 Offset
7: 3W Half Bridge (P7)
1: 1 Reps
2: 23 25 mV 60 Hz Rejection Range
3: 5 SE Channel
4: 2 Excite all reps w/Exchan 2
5: 2100 mV Excitation
6: 4 Loc [RTD_3]
7: 98.91 Multiplier
8: 0.0 Offset
8: Temperature RTD (P16)
1: 1 Reps
2: 4 R/R0 Loc [RTD_3]
3: 8 Loc [Temp_3]
4: 1.0391 Multiplier
5: -0.6281 Offset
9: 3W Half Bridge (P7)
1: 1 Reps
2: 23 25 mV 60 Hz Rejection Range
3: 7 SE Channel
4: 2 Excite all reps w/Exchan 2
5: 2100 mV Excitation
6: 5 Loc [RTD_4]
7: 98.6 Multiplier
8: 0.0 Offset
10: Temperature RTD (P16)
1: 1 Reps
2: 5 R/R0 Loc [RTD_4]
3: 9 Loc [Temp_4]
4: 1.0568 Multiplier
5: -0.4918 Offset
;Hedland Flow meter
11: Pulse (P3)
1: 1 Reps
2: 1 Pulse Channel 1
3: 0 High Frequency, All Counts
4: 10 Loc [pulse]
5: 1.0 Multiplier
6: 0.0 Offset
12: Z=F x 10^n (P30)
1: 3.224 F
2: 2 n, Exponent of 10
3: 11 Z Loc [Convers]
13: Z=X/Y (P38)
1: 10 X Loc [pulse]
2: 11 Y Loc [Convers]
3: 12 Z Loc [FlowRate]
;Save to final storage with date and time
14: Real Time (P77)^13823
1:1111 Year,Day,Hour/Minute,Seconds
(midnight = 0000)
15: Sample (P70)^2423
1: 1 Reps
2: 1 Loc [Switch]
16: Sample (P70)^16145
1: 1 Reps
2: 6 Loc [Temp_1]

17: Sample (P70)^16470
1: 1 Reps
2: 7 Loc [Temp_2]
18: Sample (P70)^5915
1: 1 Reps
2: 8 Loc [Temp_3]
19: Sample (P70)^7838
1: 1 Reps
2: 9 Loc [Temp_4]
20: Sample (P70)^28643
1: 1 Reps
2: 10 Loc [pulse]
21: Sample (P70)^12888
1: 1 Reps
2: 12 Loc [FlowRate]
*Table 2 Program
02: 1 Execution Interval (seconds)
*Table 3 Subroutines
End Program

D.2 Program for Greenhouse Measurements

;{CR10X}
*Table 1 Program
01: 60 Execution Interval (seconds)
1: Batt Voltage (P10)
1: 1 Loc [Batt_Volt]
2: If time is (P92)
1: 360 Minutes (Seconds)
2: 1440 Interval (same units as above)
3: 30 Then Do
3: Signature (P19)
1: 2 Loc [Prog_Sig]
4: End (P95)
;Measure the CO_2 monitor 4-20mA
out with 100 ohm resistor
5: Volt (Diff) (P2)
1: 1 Reps
2: 25 2500 mV 60 Hz Rejection Range
3: 1 DIFF Channel
4: 3 Loc [CO_2]
5: 1.8653 Multiplier
6: -694.14 Offset
;Measure the CS500 Temperature
6: Volt (SE) (P1)
1: 1 Reps
2: 25 2500 mV 60 Hz Rejection Range
3: 3 SE Channel
4: 4 Loc [AirTC]
5: 0.1 Multiplier
6: -40.0 Offset
;Measure the CS500 relative humidity
7: Volt (SE) (P1)
1: 1 Reps
2: 25 2500 mV 60 Hz Rejection Range
3: 4 SE Channel
4: 5 Loc [RH]
5: 0.1 Multiplier

6: 0 Offset

;Limit the maximum relative humidity to 100%

8: If (X<=>F) (P89)

1: 5 X Loc [RH]

2: 3 >=

3: 100 F

4: 30 Then Do

9: Z=F x 10^n (P30)

1: 100 F

2: 0 n, Exponent of 10

3: 5 Z Loc [RH]

10: End (P95)

;Turn on MUX1

11: Do (P86)

1: 41 Set Port 1 High

;One 107 reference probe

12: Beginning of Loop (P87)

1: 0000 Delay

2: 1 Loop Count

;Switch between channels on MUX1

13: Do (P86)

1: 72 Pulse Port 2

;Temperature reference for thermocouples

14: Temp (107) (P11)

1: 1 Reps

2: 5 SE Channel

3: 21 Excite all reps w/E1, 60Hz, 10ms delay

4: 6 Loc [T107_C]

5: 1 Multiplier

6: 0 Offset

;Four thermocouples

15: Do (P86)

1: 72 Pulse Port 2

16: Excitation with Delay (P22)

1: 1 Ex Channel

2: 0 Delay W/Ex (0.01 sec units)

3: 1 Delay After Ex (0.01 sec units)

4: 0 mV Excitation

17: Thermocouple Temp (DIFF) (P14)

1: 1 Reps

2: 3 25 mV Slow Range

3: 3 DIFF Channel

4: 1 Type T (Copper-Constantan)

5: 6 Ref Temp (Deg. C) Loc [T107_C]

6: 7 Loc [HE1inlet]

7: 1.0 Multiplier

8: 0.0 Offset

18: Do (P86)

1: 72 Pulse Port 2

19: Excitation with Delay (P22)

1: 1 Ex Channel

2: 0 Delay W/Ex (0.01 sec units)

3: 1 Delay After Ex (0.01 sec units)

4: 0 mV Excitation

20: Thermocouple Temp (DIFF) (P14)

1: 1 Reps

2: 3 25 mV Slow Range

3: 3 DIFF Channel

4: 1 Type T (Copper-Constantan)
5: 6 Ref Temp (Deg. C) Loc [T107_C]
6: 8 Loc [HE1out]
7: 1.0 Multiplier
8: 0.0 Offset
21: Do (P86)
1: 72 Pulse Port 2
22: Excitation with Delay (P22)
1: 1 Ex Channel
2: 0 Delay W/Ex (0.01 sec units)
3: 1 Delay After Ex (0.01 sec units)
4: 0 mV Excitation
23: Thermocouple Temp (DIFF) (P14)
1: 1 Reps
2: 3 25 mV Slow Range
3: 3 DIFF Channel
4: 1 Type T (Copper-Constantan)
5: 6 Ref Temp (Deg. C) Loc [T107_C]
6: 9 Loc [HE2inlet]
7: 1.0 Multiplier
8: 0.0 Offset
24: Do (P86)
1: 72 Pulse Port 2
25: Excitation with Delay (P22)
1: 1 Ex Channel
2: 0 Delay W/Ex (0.01 sec units)
3: 1 Delay After Ex (0.01 sec units)
4: 0 mV Excitation
26: Thermocouple Temp (DIFF) (P14)
1: 1 Reps
2: 3 25 mV Slow Range
3: 3 DIFF Channel
4: 1 Type T (Copper-Constantan)
5: 6 Ref Temp (Deg. C) Loc [T107_C]
6: 10 Loc [HE2out]
7: 1.0 Multiplier
8: 0.0 Offset
;Two pressure transducers
;Switch between channels on MUX1
27: Do (P86)
1: 72 Pulse Port 2
;Measure pressure transducers 4-20mA
with 100 ohm resistor
28: Excitation with Delay (P22)
1: 1 Ex Channel
2: 0 Delay W/Ex (0.01 sec units)
3: 1 Delay After Ex (0.01 sec units)
4: 0 mV Excitation
29: Volt (Diff) (P2)
1: 1 Reps
2: 25 2500 mV 60 Hz Rejection Range
3: 3 DIFF Channel
4: 11 Loc [Press_1]
5: .0001 Multiplier
6: -.0573 Offset
30: Do (P86)
1: 72 Pulse Port 2
;Measure pressure transducers 4-20mA
with 100 ohm resistor
31: Excitation with Delay (P22)

```
 1: 1        Ex Channel
 2: 0        Delay W/Ex (0.01 sec units)
 3: 1        Delay After Ex (0.01 sec units)
 4: 0        mV Excitation
32:  Volt (Diff) (P2)
 1: 1        Reps
 2: 25       2500 mV 60 Hz Rejection Range
 3: 3        DIFF Channel
 4: 12        Loc [Press_2]
 5: .0002    Multiplier
 6: -.0646   Offset
33:  End (P95)
;Turn off  MUX1
34:  Do (P86)
 1: 51       Set Port 1 Low
;Measure the LICOR Pyranometer
for solar radiation
35:  Volt (Diff) (P2)
 1: 1        Reps
 2: 4        250 mV Slow Range
 3: 4        DIFF Channel
 4: 15       Loc [Solar]
 5: 96       Multiplier
 6: 0.0      Offset
;Measure from Step50A 1.5-13.5 voltage
stepped down to 278mV-2457mV set heater
and fan indicators
36:  Volt (Diff) (P2)
 1: 1        Reps
 2: 25       2500 mV 60 Hz Rejection Range
 3: 5        DIFF Channel
 4: 25       Loc [ Step50A ]
 5: 1.0      Mult
 6: 0.0      Offset
37:  If (X<=>F) (P89)
 1: 25       X Loc [Step50A]
 2: 2        < >
 3: 1481     F
 4: 30       Then Do
38:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 19       Z Loc [FanVent]
39:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 18       Z Loc [ Fan1 ]
40:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 17       Z Loc [ Vent ]
41:  Else (P94)
42:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 19       Z Loc [Fan2]
43:  If (X<=>F) (P89)
 1: 25       X Loc [ Step50A]
 2: 2        < >
 3: 1296     F
```

```
 4: 30       Then Do
44:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 18       Z Loc [Fan1]
45:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 17       Z Loc [Vent]
46:  Else (P94)
47:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 18       Z Loc [Fan1]
48:  End (P95)
49:  End (P95)
50:  If (X<=>F) (P89)
 1: 25       X Loc [Step50A]
 2: 2        < >
 3: 1111     F
 4: 30       Then Do
51:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 17       Z Loc [Vent]
52:  Else (P94)
53:  If (X<=>F) (P89)
 1: 25       X Loc [Step50A]
 2: 3        >=
 3: 925      F
 4: 30       Then Do
54:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 16       Z Loc [Heat1]
55:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 17       Z Loc [Vent]
56:  Else (P94)
57:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 16       Z Loc [Heat1]
58:  End (P95)
59:  End (P95)
;Measure Tipping Bucket 1 from
 Dehumidifier 1
60:  Pulse (P3)
 1: 1        Reps
 2: 1        Pulse Channel 1
 3: 2        Switch Closure, All Counts
 4: 13       Loc [DeHumML_1]
 5: 1        Multiplier
 6: 0        Offset
;Measure Tipping Bucket 2 from
 Dehumidifier 2
61:  Pulse (P3)
 1: 1        Reps
 2: 2        Pulse Channel 2
```

```
 3: 2        Switch Closure, All Counts
 4: 14        Loc [DeHumML_2]
 5: 1        Multiplier
 6: 0        Offset
;62:  If (X<=>F) (P89)
; 1: 20     X Loc [Index]
 2: 1        =
 3: 2        F
 4: 3        Call Subroutine 3
62:  If (X<=>F) (P89)
 1: 20        X Loc [Index]
 2: 1        =
 3: 1        F
 4: 2        Call Subroutine 2
63:  If (X<=>F) (P89)
 1: 20        X Loc [Index]
 2: 1        =
 3: 0.0      F
 4: 1        Call Subroutine 1
64:  Do (P86)
 1: 99        Call Subroutine 99
*Table 2 Program
01: 10.0000   Execution Interval (seconds)
1:  Serial Out (P96)
 1: 71        Storage Module
*Table 3 Subroutines
;****************************
1:  Beginning of Subroutine (P85)
 1: 1        Subroutine 1
;if time is 6am
2:  If time is (P92)
 1: 360    Minutes (Seconds)
 2: 1440  Interval
             (same units as above)
 3: 30       Then Do
;add one to day count
3:  Z=Z+1 (P32)
 1: 21       Z Loc [DayCount]
;if day count is day 4 then switch to
 next set of experiments
4:  If (X<=>F) (P89)
 1: 21      X Loc [DayCount]
 2: 3        >=
 3: 4        F
 4: 30       Then Do
5:  Z=Z+1 (P32)
 1: 20       Z Loc [ Index]
6:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 21       Z Loc [DayCount]
7:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 22       Z Loc [Exchan_1]
8:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 23       Z Loc [Exchan_2_]
;Saves and ends program
```

```
9:  Do (P86)
 1: 99       Call Subroutine 99
10:  End (P95)
11:  End (P95)
12:  If (X<=>F) (P89)
 1: 5        X Loc [RH]
 2: 3        >=
 3: 80       F
 4: 30       Then Do
13:  Set Port(s) (P20)
 1: 0011     C8..C5 = low/low/high/high
 2: 9999     C4..C1 = nc/nc/nc/nc
14:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 22       Z Loc [Exchan_1]
15:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 23       Z Loc [Exchan_2]
16:  Else (P94)
17:  If (X<=>F) (P89)
 1: 5        X Loc [RH]
 2: 3        >=
 3: 75       F
 4: 30       Then Do
18:  Set Port(s) (P20)
 1: 0001     C8..C5 = low/low/low/high
 2: 9999     C4..C1 = nc/nc/nc/nc
19:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 22       Z Loc [Exchan_1]
20:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 23       Z Loc [Exchan_2]
21:  End (P95)
22:  End (P95)
23:  If (X<=>F) (P89)
 1: 5        X Loc [RH]
 2: 4        <
 3: 73       F
 4: 30       Then Do
24:  Set Port(s) (P20)
 1: 0000 C8..C5 = low/low/low/low
 2: 9999     C4..C1 = nc/nc/nc/nc
25:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 22       Z Loc [Exchan_1]
26:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 23       Z Loc [Exchan_2]
27:  End (P95)
28:  End (P95)
;*************************
29:  Beginning of Subroutine (P85)
 1: 2        Subroutine 2
```

```
;if time is 6am
30:  If time is (P92)
 1: 360      Minutes (Seconds)
 2: 1440  Interval (same units as above)
 3: 30       Then Do
;add one to day count
31:  Z=Z+1 (P32)
 1: 21       Z Loc [DayCount]
;if day count is day 4 then switch to
next set of experiments
32:  If (X<=>F) (P89)
 1: 21       X Loc [DayCount]
 2: 3        >=
 3: 4        F
 4: 30       Then Do
33:  Z=Z+1 (P32)
 1: 20       Z Loc [Index]
34:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 21       Z Loc [DayCount]
35:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 24       Z Loc [Dehum]
;Saves and ends program
36:  Do (P86)
 1: 99       Call Subroutine 99
37:  End (P95)
38:  End (P95)
39:  If (X<=>F) (P89)
 1: 5        X Loc [RH]
 2: 3        >=
 3: 75       F
 4: 30       Then Do
40:  Set Port(s) (P20)
 1: 1100 C8..C5 = high/high/low/low
 2: 9999     C4..C1 = nc/nc/nc/nc
41:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 13       Z Loc [DeHumML_1]
42:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 14       Z Loc [DeHumML_2]
43:  End (P95)
44:  If (X<=>F) (P89)
 1: 5        X Loc [RH]
 2: 4        <
 3: 73       F
 4: 30       Then Do
45:  Set Port(s) (P20)
 1: 0000 C8..C5 = low/low/low/low
 2: 9999  C4..C1 = nc/nc/nc/nc
46:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 13       Z Loc [DeHumML_1]
47:  Z=F (P30)
```

```
 1: 0.0      F
 2: 00       Exponent of 10
 3: 14       Z Loc [DeHumML_2]
48:  End (P95)
49:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 24       Z Loc [Dehum]
50:  End (P95)
;****************************
51:  Beginning of Subroutine (P85)
 1: 3        Subroutine 3
;if time is 6am
52:  If time is (P92)
 1: 360      Minutes (Seconds)
 2: 1440     Interval (same units as above)
 3: 30       Then Do
;add one to day count
53:  Z=Z+1 (P32)
 1: 21       Z Loc [DayCount]
;if day count is day 4 then switch to
next set of experiments
54:  If (X<=>F) (P89)
 1: 21       X Loc [DayCount]
 2: 3        >=
 3: 3        F
 4: 30       Then Do
55:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 20       Z Loc [Index]
56:  Z=F (P30)
 1: 1        F
 2: 00       Exponent of 10
 3: 21       Z Loc [DayCount]
;Saves and ends program
57:  Do (P86)
 1: 99       Call Subroutine 99
58:  End (P95)
59:  End (P95)
60:  Set Port(s) (P20)
 1: 0000     C8,C7,C6,C5 Options
 2: 9999     C4..C1 = nc/nc/nc/nc
61:  End (P95)
;**************************
62:  Beginning of Subroutine (P85)
 1: 99       Subroutine 99
;saves and ends program
;Store to final storage
63:  If time is (P92)
 1: 0        Minutes (Seconds)
 2: 10 Interval (same units as above)
 3: 10 Set Output Flag High (Flag 0)
64:  Set Active Storage Area
     (P80)^15139
 1: 1        Final Storage Area 1
 2: 101      Array ID
;Mark time in file
65:  Real Time (P77)^11899
 1: 1220  Year,Day,Hour/Minute
```

(midnight = 2400)

66: Average (P71)^12872

1: 1 Reps

2: 1 Loc [Batt_Volt]

67: Average (P71)^29934

1: 3 Reps

2: 3 Loc [CO_2]

68: Average (P71)^25450

1: 6 Reps

2: 7 Loc [HE1inlet]

3: 4 Loc [AirTC]

69: Average (P71)^20594

1: 1 Reps

2: 15 Loc [Solar]

70: Totalize (P72)^336

1: 1 Reps

2: 13 Loc [DeHumML_1]

71: Totalize (P72)^32647

1: 1 Reps

2: 14 Loc [DeHumML_2]

72: Sample (P70)^28789

1: 9 Reps

2: 16 Loc [Heat1]

73: If time is (P92)

1: 360 Minutes (Seconds)

2: 1440 Interval (same units as above)

3: 10 Set Output Flag High (Flag 0)

74: Real Time (P77)^4178

1: 1220 Year,Day,Hour/Minute

(midnight = 2400)

75: Sample (P70)^6618

1: 1 Reps

2: 2 Loc [Prog_Sig]

76: Maximum (P73)^6007

1: 1 Reps

2: 10 Value with Hr-Min

3: 3 Loc [CO_2]

77: Minimum (P74)^22339

1: 1 Reps

2: 00 Time Option

3: 3 Loc [CO_2]

78: Maximum (P73)^3675

1: 1 Reps

2: 00 Time Option

3: 4 Loc [AirTC]

79: Minimum (P74)^3588

1: 1 Reps

2: 00 Time Option

3: 4 Loc [AirTC]

80: Maximum (P73)^11153

1: 1 Reps

2: 00 Time Option

3: 5 Loc [RH]

81: Minimum (P74)^1810

1: 1 Reps

2: 00 Time Option

3: 5 Loc [RH]

82: Maximum (P73)^12351

1: 1 Reps

2: 00 Time Option

3: 7 Loc [HE1inlet]
83: Minimum (P74)^22939
1: 1 Reps
2: 00 Time Option
3: 7 Loc [HE1inlet]
84: Maximum (P73)^914
1: 1 Reps
2: 00 Time Option
3: 8 Loc [HE1out]
85: Minimum (P74)^5666
1: 1 Reps
2: 00 Time Option
3: 8 Loc [HE1out]
86: Maximum (P73)^13109
1: 1 Reps
2: 00 Time Option
3: 9 Loc [HE2inlet]
87: Minimum (P74)^5259
1: 1 Reps
2: 00 Time Option
3: 9 Loc [HE2inlet]
88: Maximum (P73)^29760
1: 1 Reps
2: 00 Time Option
3: 10 Loc [HE2out]
89: Minimum (P74)^24177
1: 1 Reps
2: 00 Time Option
3: 10 Loc [HE2out]
90: Maximum (P73)^13232
1: 1 Reps
2: 10 Value with Hr-Min
3: 15 Loc [Solar]
91: Minimum (P74)^29028
1: 1 Reps
2: 10 Value with Hr-Min
3: 15 Loc [Solar]
92: Totalize (P72)^4151
1: 2 Reps
2: 13 Loc [DeHumML_1]
93: Do (P86)
1: 0 Go to end of Program Table
94: End (P95)
End Program

D.3 Program for Weather Station

```
;{CR10}
*Table 1 Program
01: 60        Execution Interval (seconds)
;Measure the CS500 Temperature
1:  Volt (SE) (P1)
 1: 1        Reps
 2: 5        2500 mV Slow Range
 3: 1        SE Channel
 4: 1        Loc [T_C]
 5: .1       Mult
 6: -40      Offset
;Measure the CS500 relative humidity
2:  Volt (SE) (P1)
 1: 1        Reps
 2: 5        2500 mV Slow Range
 3: 2        SE Channel
 4: 2        Loc [RH_pct]
 5: .1       Mult
 6: 0.0      Offset
;Limit the maximum relative humidity to 100%
3:  If (X<=>F) (P89)
 1: 2        X Loc [RH_pct]
 2: 3        >=
 3: 100      F
 4: 30       Then Do
4:  Z=F (P30)
 1: 100      F
 2: 0        Exponent of 10
 3: 2        Z Loc [RH_pct]
5:  Z=X*F (P37)
 1: 2        X Loc [RH_pct]
 2: 0.0517   F
 3: 7        Z Loc [RHfinal]
6:  Z=X+F (P34)
 1: 7        X Loc [RHfinal]
 2: 2.1921   F
 3: 7        Z Loc [RHfinal]
7:  End (P95)
;Measure the LICOR Pyranometer
for solar radiation
8:  Volt (Diff) (P2)
 1: 1        Reps
 2: 4        250 mV Slow Range
 3: 2        DIFF Channel
 4: 3        Loc [solar]
 5: 96.4     Mult
 6: 0.0      Offset
;Set negative values to zero
9:  If (X<=>F) (P89)
 1: 1        X Loc [T_C]
 2: 4        <
 3: 0        F
 4: 30       Then Do
10:  Z=F (P30)
 1: 0        F
 2: 0        Exponent of 10
 3: 3        Z Loc [solar]
11:  End (P95)
```

```
;Measure wind direction Young Vane 00330
12:  Excite-Delay (SE) (P4)
 1: 1        Reps
 2: 5        2500 mV Slow Range
 3: 5        SE Channel
 4: 1 Excite all reps w/Exchan 1
 5: 2        Delay (units 0.01 sec)
 6: 2500     mV Excitation
 7: 4        Loc [wdirect]
 8: 0.142    Mult
 9: 0.0      Offset
13:  If (X<=>F) (P89)
 1: 4        X Loc [wdirect]
 2: 3        >=
 3: 360      F
 4: 30       Then Do
14:  Z=F (P30)
 1: 0.0      F
 2: 00       Exponent of 10
 3: 4        Z Loc [wdirect]
15:  End (P95)
;Measure wind speed Met One 013A-C
16:  Pulse (P3)
 1: 1        Reps
 2: 1        Pulse Input Channel
 3: 2        Switch Closure, All Counts
 4: 5        Loc [wspeed]
 5: .02666   Mult
 6: 0.4470   Offset
17:  Batt Voltage (P10)
 1: 6      Loc [BATTVOLT]
;Detects program changes
 or ROM failure
18:  Signature (P19)
 1: 8       Loc [ProgSig]
;Store to final storage
 every ten minutes
19:  If time is (P92)
 1: 0000     Minutes (Seconds)
 2: 20 Interval
          (same units as above)
 3: 10       Set Output Flag High
20:  Real Time (P77)
 1: 1220  Year,Day,Hour/Minute
          (midnight = 2400)
21:  Average (P71)
 1: 7        Reps
 2: 1        Loc [T_C]
;At 6pm daily take some total stats
22:  If time is (P92)
 1: 1080     Minutes (Seconds)
 2: 1440     Interval
             (same units as above)
 3: 10       Set Output Flag High
23:  Maximum (P73)
 1: 7        Reps
 2: 10       Value with Hr-Min
 3: 1        Loc [T_C]
24:  Minimum (P74)
 1: 7        Reps
```

2: 10 Value with Hr-Min

3: 1 Loc [T_C]

25: Sample (P70)

1: 1 Reps

2: 8 Loc [ProgSig]

*Table 2 Program

02: 0.0000 Execution Interval (seconds)

*Table 3 Subroutines

End Program