Characterisation and Optimisation Of Transport Temperature Control Units In Heat Mode

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CHARACTERISATION AND OPTIMISATION OF TRANSPORT TEMPERATURE CONTROL UNITS IN HEAT MODE

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Abstract

The control of air temperature within trailer food-compartment during road transportation of perishable products is critical as it influences the deterioration rate and the quality loss of food more than any other factor. As a result, this thesis focuses on temperature control during the transportation of perishable food products, particularly in cold climates, where sub-zero degree Celsius temperatures exist.
Transport Temperature Control (TTC) units are designed primarily to maintain product temperature, not to reduce or increase it. Therefore, TTC units must be capable of maintaining the set-point temperature of perishable food products by automatically providing cooling and/or heating as necessary. Due to the increased importance of heat mode to the transport temperature control industry, a detailed characterization of TTC units in heating was performed. This project sought to: i) propose a test procedure and test facility capable of accurately measuring TTC unit heating capacity, ii) provide the first complete characterisation of existing standard TTC unit behaviour in heat mode, iii) highlight the problems that are encountered during food transportation in sub 0°C ambient temperature, iv) propose several original design modifications for improved heating performance with general application for all types of TTC units, and v) present the first mathematical model capable of predicting the system behaviour in heating. Based on the improvements implemented the system behaviour was significantly improved. The results show increased heating capacity by 25%, system efficiency by 30%, while the fuel consumption was reduced by 7%. The characterisation of both standard and modified TTC units was obtained through: i) a mathematical model capable of predicting the system performance to an accuracy of ±6% and ii) an experimental study based around the development of a novel test procedure and facility that reduced the measurement uncertainty from ±10% to ±1%.

DECLARATION

The substance of this thesis is the original work of the author and due reference and acknowledgement has been made, when necessary, to the work of the others. No part of this thesis has already been accepted for any degree and is not being concurrently submitted in candidature for any other award.

Signed: Aradu C

Date: 24th of May 2005

Cristina Radulescu (Candidate)

Signed: ______________________

Date: ______________________

Dr. John Lohan (Supervisor)

DEDICATION

To my parents for encouraging me to learn

STATEMENT OF CONFIDENTIALITY

The material contained in this thesis should not be used, sold, assigned or disclosed to any other person, organisation or corporation without the express permission of each of the following:

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PUBLISHED WORK

The following five publications resulted directly from this project:

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NOMENCLATURE

1. Abbreviations

Abbreviation	*Name*	*Unit*
ACIP	Accumulator Inlet Pressure	[Psi]
ACIT	Accumulator Inlet Temperature	[°C]
ACOT	Accumulator Outlet temperature	[°C]
ACOP	Accumulator Outlet pressure	[Psi]
AMB	Ambient Environment Temperature	[°C]
ASHRAE	American Society of Heating, Refrigerating	
ATP	Agreement of the International Carriage of Perishable Foodstuffs	
CAI	Condenser Air Inlet Temperature	[°C]
CLOT	Condenser Outlet temperature	[°C]
CGIT	Condenser inlet gas temperature	[°C]
CGIP	Condenser Inlet Pressure	[Psi]
CGOP	Condenser Gas Outlet Pressure	[Psi]
CLOP	Condenser Outlet Pressure	[Psi]
EAI	Evaporator Air Inlet Temperature	[°C]
EAO	Evaporator Air Outlet Temperature	[°C]
EVOT	Evaporator Outlet Temperature	[°C]
EVIT	Evaporator Inlet Temperature	[°C]
EVOP	Evaporator Outlet Pressure	[Psi]
EVIP	Evaporator Inlet Pressure	[Psi]
EEV	Electronic Expansion Valve	Steps
ETV	Electronic Throttling Valve	Steps
HVIT	Heat Exchanger Inlet temperature	[°C]
HVOT	Heat Exchanger Outlet temperature	[°C]
HVOP	Heat Exchanger Outlet Pressure	[Psi]
H 1	Initial Electric Heat Input	[W]
H 2	Final Electric Heat Input	[W]
PGOT	Compressor Outlet Temperature	[°C]
PVIT	PVIT = Compressor Inlet	[°C]
PVIT Box CS	Compressor suction temp. -Cooling System	[°C]
PGOP	Compressor discharge pressure	[Psi]
PVIP	Compressor Suction Pressure	[Psi]
RLOT	Receiver Tank temperature	[°C]
TTC	Transport Temperature Control Unit	

2. Symbols

Symbol	*Name*	*Unit*
A	Total heat exchange area of the shaped chambers in the scroll compressor	[m^2]
A_d	Compressor discharge area	[m^2]
A_{cd}	Evaporator heat transfer area	[m^2]
C_p	Specific Heat	[KJ/kg°C]
C_{pc}, C_v	Coolant and suction vapor specific heat	[KJ/kg°C]
$C_{p\ liquid}$	Accumulator liquid specific heat	[KJ/kg°C]
$C_{p\ air}$	Specific heat for air	[KJ/ kg°C]
C_{pv}	Specific heat for the saturation temperature of the compressor discharge gas	[KJ/ kg°C]
CF	Correction Factor	
D_h	Hydraulic Diameter	[m^2]
F_1	Function of fan heat input	
F_2	Function of heat losses	
$Fan_{power\ input}$	Fan power input	[W]
K	Global heat exchange coefficient	[W/m^2C]
K_{pipe}	Global heat exchange coefficient for pipe	[W/m^2C]
L	Pipe length	[m]
$Length_{HE}$	Length Heat Exchanger	[m]
M_f	Refrigerant Mass Flow Rate	[Kg/s]
$M_{f\ state\ 8}$	Accumulator outlet refrigerant mass flow	[Kg/s]
$M_{liq\ acc}$	Accumulator Liquid Mass Flow	[Kg/s]
$M_{f\ air}$	Air mass flow rate	[kg/s]
$Pipe_{flow\ area}$	Mass flow rate through the hot-gas line	[Kg/s]
P_n	Pressure at state n	[psi]
Pr	Prandtl Number	
P	Compressor Power Input	[W]
P_1, P_2, P_4	Compressor inlet, outlet pressure and discharge pressure regulator valve outlet pressure	[psi]
Re	Reynolds Number	
R_{aver}	Average radius	[m]
R_1, R_2	Thermal resistance	[m°C/W]
S, S_1	Area for heat losses in the ambient	[m^2]

T_{ao}	Air outlet temperature	[°C]
T_{ai}	Air inlet temperature	[°C]
T_{amb}	Ambient Temperature	[°C]
T_{box}	Temperature inside the box	[°C]
T_c	Coolant temperature	[°C]
T_{ci}, T_{vi}	Coolant and suction vapor inlet temperature in the heat exchanger	[°C]
T_{co}, T_{vo}	Coolant and suction vapor outlet temperature in the heat exchanger	[°C]
T_{liq}	Accumulator liquid temperature	[°C]
T_{fr}	Temperature increase because friction	[°C]
T_{pipe}	Pipe temperature	[°C]
T_{sat}	Saturation temperature	[°C]
T_{scroll}	Scroll temperature	[°C]
T_v	Compressor suction vapor temperature	[°C]
$T_{vi\ comp}$	Vapor temperature at the compressor inlet for modified unit (coolant heat exchanger on compressor suction line)	[°C]
$T_{vi\ com\ in}$	Vapor temperature at the compressor inlet for the standard unit	[°C]
U_{cd}	Overall heat transfer coefficient	[W/m^2°C]
V	Volume Flow Rate	[m^3/h]
V_{air}	Specific volume air	[m^3/kg]
V_i	Volume ratio	
Q	Heat exchange capacity of the coolant heat exchanger	[W]
Q_{amb}	Heat Losses in ambient	[W]
$Q_{TTC\ heat\ cap}$	Measured heating capacity (EHI method)	[W]
Q_{heat}	Heating capacity (ATP method)	[W]
$Q_{in\ 1}$	Heating capability obtained as a result of the H_1 measured electric heat input	[W]
$Q_{in\ 2}$	Heating capability obtained as a result of the H_2 measured electric heat input	
$Q_{box\ loss}$	Heat Losses between box and ambient	[W]
$Q_{difference}$	Heating capacity difference for 700 W	[W]
$W_{compression}$	Compression Work	[W]
$W_{isentropic}$	Isentropic Work	[W]
d_e	External pipe diameter	[m]

d i	Internal pipe diameter	[m]
d/dt	Derivative coefficient, PID control	
$\int dt$	Integral coefficient, PID control	
g	Gravitational acceleration	[m^2/s]
$h_{amb} = h_{air}$	Air (ambient) convective heat transfer coefficient	[W/m^2C]
h_r	Refrigerant convective heat transfer coefficient	[W/m^2C]
h_n	Refrigerant enthalpy for state n	[KJ/kg]
h_c	Coolant convective heat transfer coefficient	[$W/m^2°C$]
h_1, h_2	Compressor inlet – outlet enthalpy	[KJ/kg]
k	Thermal conductivity	[W/m°C]
k_{is}	Isentropic transformation coefficient	
k_{pipe}	Thermal conductivity of the pipes	[W/m°C]
$m_{c;}\ m_v$	Coolant and suction vapor mass flow rate	[Kg/s]
n	Refrigerant State number	
ΔP	Pressure drop characteristic for the discharge regulator valve	[psi]
$\Delta T_{evap,\ air}$	Temperature difference across the evaporator on air side	[°C]
ΔT	Temperature difference across the evaporator	[°C]
ΔT_{max}	Maximum temperature difference on the condenser	[°C]
ΔT_{min}	Minimum temperature difference on the condenser	[°C]
ρ	Refrigerant Density	[Kg/m^3]
ρ_{air}	Air Density	[Kg/m^3]
η_{is}	Isentropic efficiency	
λ	Thermal conductivity	[W/m°C]

GLOSSARY OF TERMS

Air Circulation

The transport temperature control unit fans cause temperature controlled air to circulate around the inside of the food compartment to remove heat which is conducted from outside.

Air Distribution

In order to ensure and even temperature distribution in the trailer body / food compartment it is essential to have air uniformly distributed through out the load. This can be assisted by having the cargo uniformly arranged.

Ambient Air

The ambient air is: i) the air surrounding the trailer body / food compartment served by the TTC unit during field transportation, ii) the air in the Ambient Room surrounding the Calibrated Box during this test method, or iii) the air surrounding the engine–condenser side of a transport temperature control unit during the Isothermal Box test method.

"Box" Heat Losses

Heat transfer through the trailer and calibrated box walls or the front wall of the isothermal box.

Calibrated Box/ Food Compartment/ Trailer

A well-insulated enclosure, with a known heat transfer rate determined through calibration, used for measuring the heat loss or gain between the enclosure's interior temperature and ambient temperature.

Delivery / Outlet Air Temperature

This is the temperature at which the air leaves the temperature control unit's evaporator to be delivered to the interior of the temperature-controlled body trailer / food compartment.

Electric Mode

The TTC unit operates on standby electric motor.

High Speed Diesel Mode

The diesel engine driven TTC unit operates at maximum compressor rotational speed and maximum airflow through the evaporator.

Isothermal Box

An insulated enclosure, with a known heat transfer rate determined through calibration, used for measuring the heat loss or gain between the enclosure's interior temperature and ambient temperature through the front wall. The area surrounding the Isothermal Box is divided into a Surround Area and the Ambient Space.

Low Speed Diesel Mode

The diesel engine driven TTC unit operates at minimum compressor rotational speed.

Relative Humidity

The relative humidity of the air around the food produce. Relative humidity of the air around the product depends on the water activity at the surface of the product, the rate of fresh air ventilation and the relative humidity of the fresh air ventilation.

Return Air Temperature

This is the temperature of the air leaving the temperature-controlled food compartment before entering the TTC unit for heating or cooling. For both containers and road vehicles the air return temperature is generally accepted as the value that represents the average air temperature of the commodity within the carriage space.

Transport Temperature

The temperature of the air in the food compartment/ trailer/ calibrated box during transportation, storage or in-house tests.

CHAPTER 1

INTRODUCTION

CONTENT:

The temperature-controlled transport industry has seen considerable growth since its introduction 120 years ago. Initially, perishable food products transportation was synonymous with cooling and it was predominantly used in refrigerated ships, from a freezing plant to a cold storage facility. However, temperature control now implies the transportation of perishable foods at a controlled set-point temperature from the point of production to the consumer [Morley (1998)]. Attention is directed in this study towards road transportation of fresh food products at sub-zero Celsius ambient temperatures and, in particular, on the mechanical equipment required to maintain above zero-Celsius set-point temperatures using heating mode. An overview of the market trends in the temperature-controlled transportation industry, together with the problems that appear during heat mode operation at low ambient temperatures is presented in this chapter. Finally, the content presented in this thesis is summarised.

1.1 BACKGROUND

The transport sector provides one of the most vital traded services that include road, rail, air and water transportation for both human beings and commodities. The latter can be defined as products that include food, livestock, medicine, fuel and wood. This study is focused on the transportation of perishable food products that need temperature control and, in particular, on the mechanical equipment required to maintain recommended transport temperatures, namely the transport temperature control (TTC) units [Figure 1.1]. Most users refer to these systems as refrigeration units, despite the fact that are designed primarily to maintain product temperature, not to reduce it. However, the transport of perishable foods requires much more elaborate temperature control than that provided by a refrigeration system in cooling. Therefore, rather than use the term "refrigeration", the more general and accurate term of "transport temperature control" is proposed, as it reflects the system's capacity to deliver cooling or heating as required [Lohan *et al.*, (2003)].

In order to assess the current state of the food transport industry the global food production and temperature-controlled transportation market trends are briefly reviewed. Ensuring both food quality and quantity to six billion inhabitants worldwide is one of the most important challenges facing humanity, which is likely to remain as the world population is predicted to increase at the rate of 73 million per year over the next twenty years [Billiard (1999), (2003)]. Billiard (2003) has also identified that the availability of an appropriate use of temperature control to help preserve food quality during both storage and transportation will play a central role in addressing this problem. Global food production reached 5165 million tones in 1997 [FAO (1998)] and the most important food products can be classified into the following groupings: cereals, root crops, vegetables, milk, fruits, oil crops, meat, sugar, seafood, pulses, eggs and miscellaneous (Appendix A). Of the total food production it is estimated that 49.6%, or 2561 million tones, require refrigeration [FAO (1998); Billiard (2003)]. However, in reality, only 350 million tones, or 13.7%, are stored and transported in temperature-controlled environments from the producer to the costumer [Billiard (2003)]. Estimates of total worldwide annual food wastage due to non–use of appropriate temperature control range from 300 million tones, 30% of primary production, or to as much as 40% in the case of fruit and vegetables [Kaminsky (1995)].

As a result of the increased consumption of perishable foods that need temperature control during transportation, the growth in the refrigeration market has also occurred relatively quickly, especially in Central and Northern European markets that have the most significant share of 71% to 80% [Guilpart (2003)]. Market reports show a four-fold increase in the sales percentages of TTC units during the period 1980 and 2002 [Thermo King Market Report (2003)]. Appendix B presents the market segmentation together with a classification and technical details for different existing types of TTC units, while Figure 1.1 shows one of the most popular articulated long single-compartment transport temperature control (TTC) trailer unit.

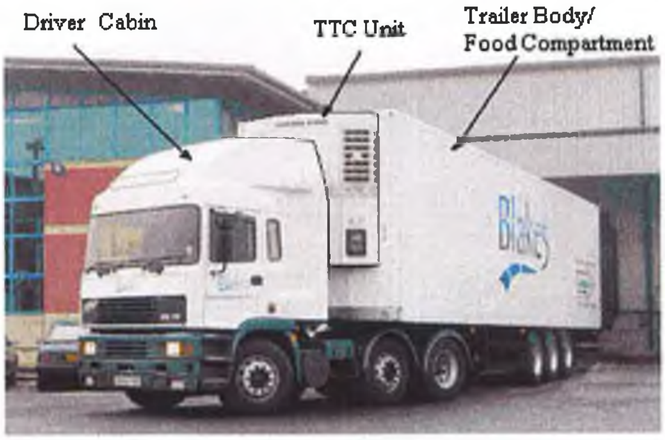


Figure 1.1 Transport Temperature Control (TTC) unit and Trailer Body/ Food Compartment.

The process of transporting perishable commodities is referred to as the "Cold Chain" and includes the procurement, storage, transportation and retailing of food products under controlled temperature. The main purpose of the cold chain is to provide ideal conditions for food products as they journey from the producer to consumer [Figure 1.2]. As a high quality "Cold Chain" is absolutely necessary Guilpart (2003) and Panozzo *et al.*, (2003) presented a comprehensive review of the most important regulations regarding all stages of the cold chain.

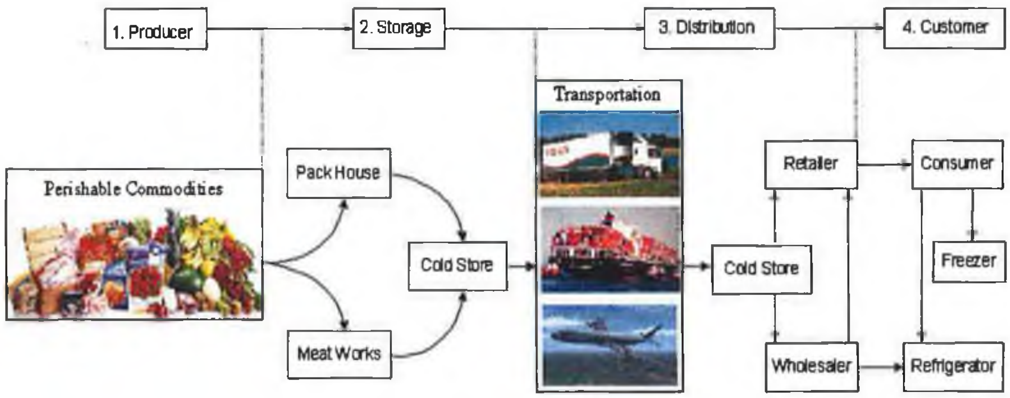


Figure 1.2 Extent of the Food Distribution "Cold Chain".

The most significant factors that influence the quality of perishable commodities as they progress through the cold chain include air temperature and humidity within the trailer body / food compartment as well as air circulation [See Glossary of Terms]. This study is focused on the road distribution sector of the cold chain and, in particular, on the TTC units that have to maintain required transport temperature set-points above 0°C for fresh food products while operating in sub-zero degree Celsius. As indicated in Figure 1.1, the two main components of the transport temperature control facility are the trailer body / food compartment and the transport temperature control (TTC) unit. These are described in the following sections.

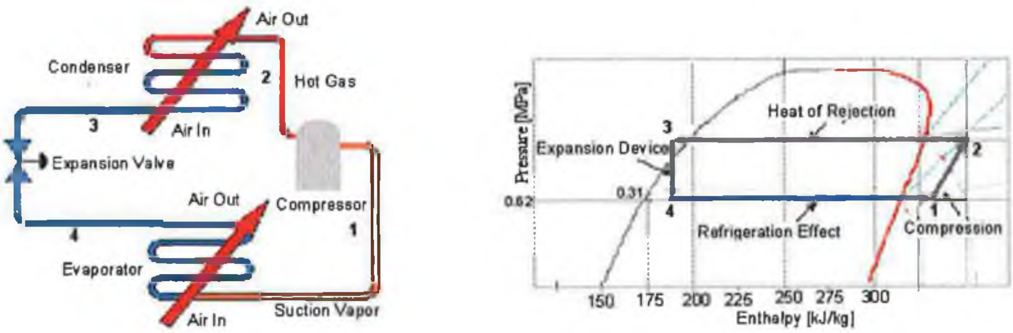
1.1.1 The Trailer Food Compartment

For land transportation, the refrigerated semi–trailer shown in Figure 1.1 is the most popular vehicle type, although rigid-body trucks are also used for local deliveries. The food compartments of these vehicles are typically thermally insulated with polyurethane foam, although PVC, glass and polystyrene foams are also used. The thickness of insulation varies from 100 to 150mm on most road vehicles. Following a change in the construction and use regulations, refrigerated trailer bodies are now being built with sidewalls of 55–60 mm thickness and roof, floor and bulkhead of 100–150mm thickness. The thermal conductivity (K) of these structures is typically lower than 0.6W/m^{2}C [ATP (1970)]. For a 12.2m long semi trailer the refrigeration load at +30°C outside temperature and –20°C inside air temperature would be 2,600 Watts. However, this can be increased significantly due to the intake of hot air during door openings or due to food respiration and a TTC unit with minimum 5,000Watts cooling capacity would be more suitable for this application.

1.1.2 Transport Temperature Control (TTC) Unit

Despite the increased demand from food distributors for TTC units with adequate heating capacity [Lohan *et al.,* (2003)], most previous research has been devoted to improve system efficiency and compressor behaviour when exposed to high ambient temperatures and operating in cool mode [Yun-Hee *et al.*, (2000); Haas *et al.*, (2000); Plastinin *et al.,* (2000)]. While a comprehensive description of the cooling cycle does appear in the literature [Dossat (2003); Stoecker (2003)] this section presents a description of a standard TTC unit in heat mode together with the thermodynamic process that takes place during this cycle [Figure 1.4]. Even if it is considered that a TTC unit in heat mode has constructive similarities with a heat pump it is impossible to apply to the TTC units the existing research available in the literature for heat pumps [MacArthur (1984); Granet *et al.*, (2000), Sakellary *et al.,* (2003)]. The main difference between a TTC unit in heat mode and a typical heat pump is that the latter operates between two external fluids at different temperatures in the condenser and the evaporator sections, whereas a TTC unit depends on heat exchange with only one external fluid, the ambient air flowing through the evaporator, while the second operational temperature of the Carnot Cycle is based on the expansion of liquid in the accumulator and not as a result of the interaction with a second external fluid. Another feature specific to the TTC unit in heat mode is that the heating load capacity can vary considerably, resulting in a wider range of operating conditions. While Figure 1.4 shows the particularities of a TTC unit in heat mode, for a better comparison with the cool mode Figure 1.3 presents schematically the thermodynamic process for this operational mode [Dossat *et al.*, (2003)].

Each TTC unit has four main components that include: evaporator, compressor, condenser and a metering device (expansion valve), which are arranged as shown in Figure 1.3(a). The primary heat exchange elements are the evaporator and the condenser, with the compressor acting as a pump to circulate the refrigerant. The thermodynamic performance can be plotted on a pressure–enthalpy (P-h) chart, similar to that shown in Figure 1.3(b). The heat being extracted by the evaporator (refrigeration effect) between state 4 and 1, as a result of a phase change of the liquid to vapor is facilitated by a cyclic process that consists of: i) isentropic compression (process 1→2), ii) isobaric condensation (process 2→3), iii) adiabatic transformation (process 3→4), and iv) isobaric and isothermal evaporation (process 4→1).

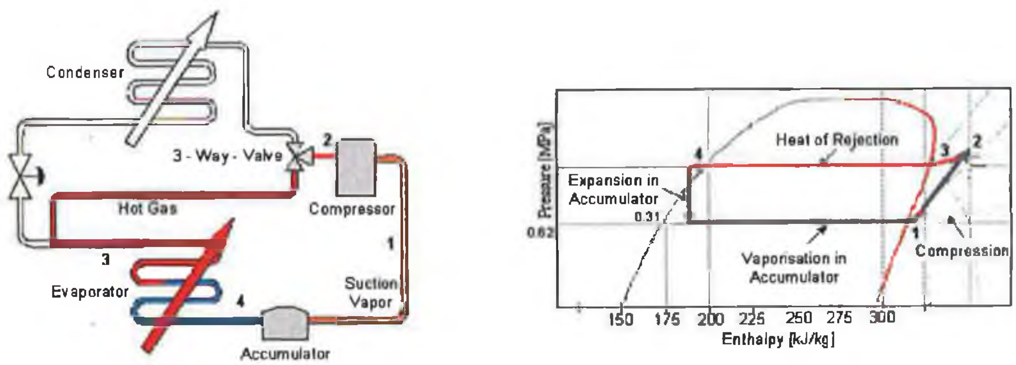


a) Schematic of a classic TTC system.

b) Thermodynamic process.

Figure 1.3 Schematic of a classic TTC system consisting of an evaporator, compressor, condenser and an expansion valve and the Thermodynamic Cycle.

The introduction of the three-way valve shown in Figure 1.4(a) enables a TTC system evaporator to both heat and cool. All Thermo King's units have this type of valve to facilitate this functionality.



a) Schematic of the TTC unit in heat mode.

b) Thermodynamic process.

Figure 1.4 Schematic of the TTC unit operating in heat mode and Thermodynamic process.

When the three-way valve is opened to the condenser, the refrigerant flows exactly as presented in Figure 1.3(a) and the evaporator absorbs heat from the air in the food compartment, thereby providing cooling. If however, the three-way valve is opened to the evaporator, the superheated refrigerant gas from the compressor outlet is directed directly to the evaporator, which now acts as a condenser, by providing heat to the food compartment. The liquid refrigerant is changing the phase into gas (suction vapor) as it travels through the accumulator that it represents the vaporisation stage of the heat cycle.

The thermodynamic performance of the TTC unit in heat mode can be also plotted on a pressure–enthalpy (P-h) chart, as presented in Figure 1.4(b). The main processes of this cycle consist in:

i) Isentropic compression – Process 1→2: The compressor converts low-pressure vapor (State 1) into high-pressure vapor at its outlet (State 2).

ii) Isobaric transformation – Process 2→3: During this process that takes place on the discharge and hot gas lines, a part of the superheat obtained at state 2 is removed while the vapor pressure remains constant.

iii) Isobaric and isothermal condensation – Process 3→4: This process represents the condensation of the superheated inlet vapor (State 3) into saturated liquid (State 4) that takes place at a constant pressure and temperature in the evaporator. The heat rejected to the air in the trailer is the difference between the enthalpy of the refrigerant at points 3 and 4 and it represents the heating effect of the unit.

iv) Adiabatic and isobaric transformation – Process 4→1: When the liquid refrigerant passes through the accumulator it expands at a lower evaporation pressure. During this process, the temperature of the liquid is also reduced from its condensing temperature (State 4) to the evaporating temperature (State 1). The process takes place in the accumulator as a result of an adiabatic transformation. Saturated refrigerant vapor is obtained due to expansion.

1.2 RESEARCH PROBLEMS AND JUSTIFICATION

As defined in section 1.1 the control of air temperature during the transportation of perishable foods is more complex than just running the refrigeration system in cool mode. At present refrigerated transport systems are designed to maintain recommended set-point temperatures for different categories of food product to an accuracy of ±1% or a maximum of ±2°C, using both cooling and heating as necessary [Kennedy (1998)]. The ability of a TTC unit to achieve good temperature control depends on the unit's cooling and heating capacity and on the control algorithm that takes in account the return air temperature (Appendix B). Emphasising the importance of heat mode, a recent study has shown that transport temperature control units used for long distance road transportation of perishable foodstuffs can operate in heat mode for as much as seven months of the year, especially in countries north of 55° latitude [Lohan *et al.*, (2003)]. Despite this clear demand from food distributors situated in cold climates for TTC units with adequate heating capacity, the amount of study conducted on this aspect is limited [Diab *et al.*, (1991)]. As a result, further research is necessary to establish the problems that appear during heat mode operation and to optimise the system efficiency and heating capability.

1.2.1 Cool and Heat Mode Temperature Control

Examples of the typical diesel engine driven TTC unit operation in cool and heat mode are presented for a single-compartment unit in Figure 1.5(a) and (b). The results are extracted from measured experimental data. Note that temperature control is based around alternating between cool and heat modes as the unit operates continuously (Appendix B).

The test presented in Figure 1.5(a) simulates a daily journey behavior of a TTC unit at +20°C ambient temperature during the transportation of frozen products at a set-point temperature of -20°C. The cool mode dominates the operation of the unit. It can be noticed that a pull-down time of 2 hours when the unit operates in high-speed followed by 1 hour for low-speed diesel mode. As the set-point is approached, the controller switches to low speed to facilitate the temperature control, which is maintained in a good range of 3°C using alternating periods of heating and cooling.

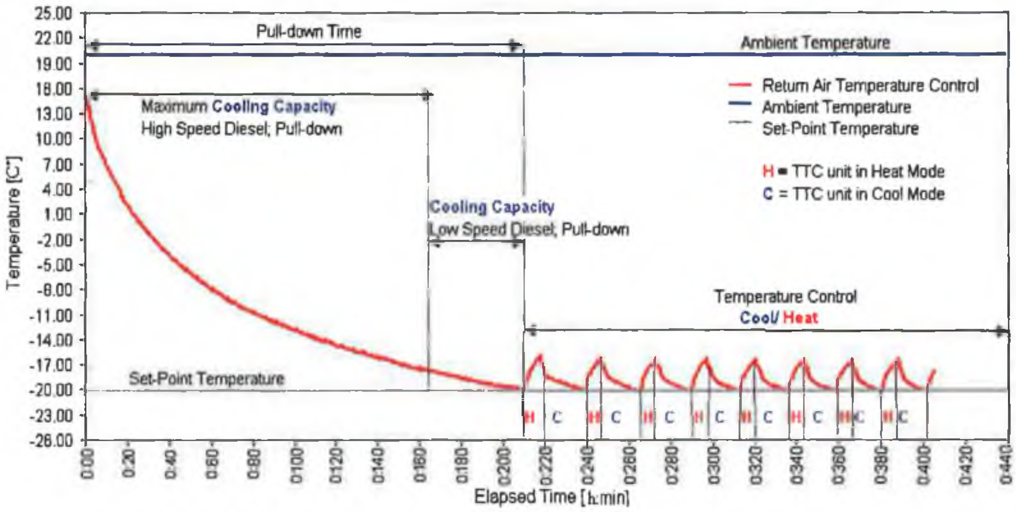


Figure 1.5(a) Traditional Concept of a single-compartment TTC Unit requested to achieve a –20°C set-point in a high temperature external ambient.

Figure 1.5(b) presents the same type of single-compartment TTC unit operating at the same 40°C temperature difference between the trailer and ambient, but in a reversed situation, when it runs predominantly in heat mode, maintaining a +21°C air temperature in the trailer while the external ambient is at –20°C.

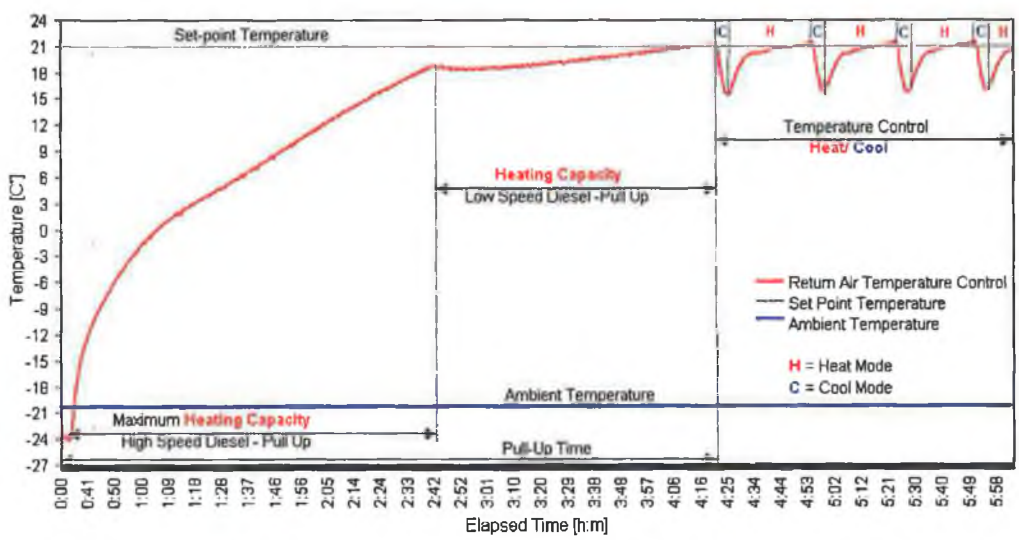


Figure 1.5(b) Traditional Concept of a single-compartment Unit requested to achieve a +21°C set point temperature in low external ambient conditions.

A long "pull-up" time of almost 3 hours is displayed when the unit is running in high-speed, followed by almost 2 hours when the unit is running in low speed diesel mode, and it still struggles to reach the set-point temperature. The TTC unit displays poor air-temperature control within the range of 5°C that is maintained with long 30-minute periods of heat mode operation alternating with shorter time intervals of only 10 minutes for cool mode. Due to low heating capacity the pull-up time for low speed operation mode is up to 2 hours for only 2°C temperature increase. The temperature control range is entirely below the set-point temperature, achieving +15°C in the food compartment when the desired temperature set-point is +21°C.

1.2.2 Field Data – Problems in Heat Mode

The performance data presented in Figure 1.5(a,b) represents the TTC unit behaviour as measured under controlled test environments in the test laboratory. However, field data was also available and is presented in Figure 1.6. Such field data, obtained from a Swedish based food distributor, also highlights the temperature control problems and the long pull-up time that appear during the transportation of fresh products in cold climates [Nilsson (2002)].

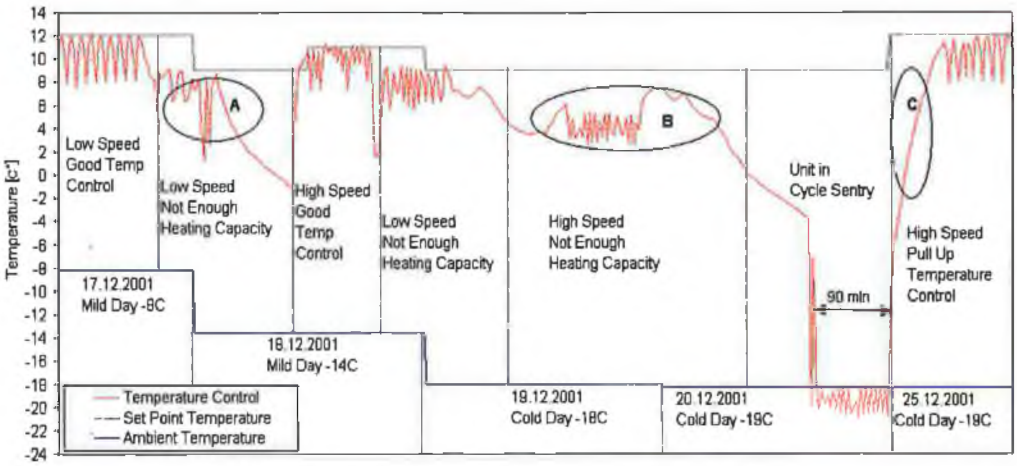


Figure 1.6 Swedish Field Data from a Thermo King dealer that uses the TS 300 TTC unit during December 2001 [Nilsson (2002)].

While only daily average temperatures are shown, it is clear that the TTC unit struggles to maintain set–point temperatures of approximately +10°C, once the external ambient drops below –14°C. The first clear evidence of this lack of heating capacity is exposed during December 17th, when the average temperature of –14°C occurred during the night. The sudden 6°C drop in the external ambient, coupled with the lack of heating capacity caused the compartment temperature to drift by up to 10°C below the +9°C set-point (Region A in Figure 1.6). After 6 hours the unit recovered the set point by switching from Low Speed operation to High Speed operation. Similar problems occurred on December 19th and December 20th. This example highlights problems at three levels when the TTC units operate in cold climates: i) insufficient heating capacity (Region A in Figure 1.6), ii) poor set-point temperature control (Region B in Figure 1.6) and iii) unacceptable long pull-up times (Region C in Figure 1.6).

1.2.3 Heating and Cooling Capacity Measurements

While the TTC unit has sufficient cooling capacity to maintain tight temperature control range during frozen food transportation, Figures 1.5(b) and 1.6 show poor performance of the unit at low ambient temperatures due to insufficient heating capability of the unit. It is acknowledged that the total cooling capacity of the unit is defined by the rate of heat transfer from the air in the food compartment to the refrigerant circulated through the evaporator, while the heating capacity is the rate of heat transfer from the hot-gas refrigerant to the air inside. The existing standard test methods determine with an accuracy of ±3% the cooling capacity of a TTC unit in test facilities similar with the one presented in Figure 1.7 [ATP (1970); Guilpart (2003)]. No such accurate test procedure or dedicated test facility exists for the measurement of heating capacity. The only test method used to determine this last indicator of the unit performance has an accuracy of just ±10% [ATP (1970)]. This explains the differences between the specified figures for heating capacities provided by the manufacturer [Thermo King web site] when compared with the poor behaviour of the unit on field [Figure 1.6]. Further details are presented in Chapter 3.

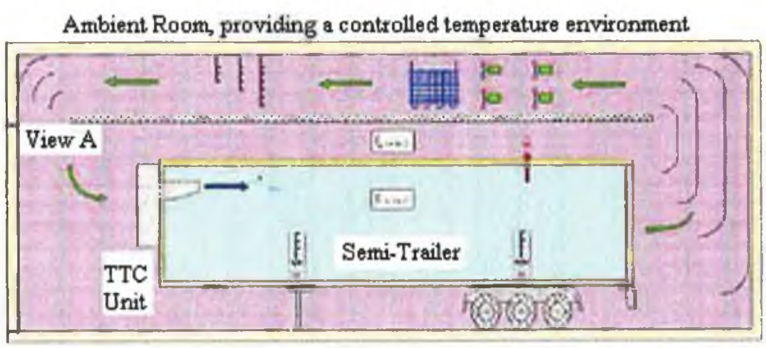


Figure 1.7 Schematic Side View of a typical test facility used for the certification of TTC unit cooling capacity. Test Facility in France, Cemafroid [Guilpart (2003)]. Front elevation from View A shown in Figure 1.1.

1.3 AIMS AND OBJECTIVES

The aim of this study was to perform the first detailed characterization and optimization of TTC units in heat mode. This is achieved by monitoring and controlling the air temperature surrounding perishable foods during road transportation using single-compartment truck and trailer TTC units. Considering the experimental and field data presented in Figures 1.5(b) and 1.6 together with the requirements to maintain tight set-point temperature control within ±2% of recommended transport temperatures [ASHRAE (1997); ATP (1970)], the specific project objectives were to:

i) Establish for the first time the dependence of the transport temperature control industry on heat mode through a complete literature review considering the influence of temperature control on perishable foods quality, recent changes in the market for TTC units and field data from industry food distributors.

ii) Review existing test methods used to measure both cooling and heating capacities and establish a more accurate test method and facility that could be used for accurate product performance rating in heat mode.

iii) Use the more accurate test procedure to conduct a thorough experimental analysis of two existing standard single-compartment TTC units behaviour in heat mode and quantify for the first time the operational problems that occur in low ambient temperatures.

iv) Propose several design changes as the first attempt to optimise the behaviour of the systems in heat mode and to increase the heating capacity of the TTC units selected for study. Present a comprehensive comparison between the standard "off-the-shelf" production units and the impact on the heating performance of the design modifications introduced.

v) Develop the first mathematical model capable of predicting the heating capacity and behaviour of a TTC unit. Extend this mathematical tool with another two simulations used to predict the impact of several design modifications on the heating performance.

The approach structure of this thesis is overviewed in the following section.

1.4 THESIS STRUCTURE

The main objectives identified are addressed in this thesis as follows (Figure 1.8):

i) Chapter 1: This chapter provides the introduction for the thesis. The first section describes a brief overview of the area covered in this research. It then outlines the thesis motivation, after which the thesis objectives and the approach to work is described.

ii) Chapter 2: Addressing Objective I, this chapter presents a literature review that highlights the growing importance of heat mode during temperature controlled road transportation. This marks the first comprehensive analysis undertaken to evaluate the importance of heating in cold climate countries.

iii) Chapter 3: Responding to Objective II, this chapter not only presents a review of existing cooling and heating capacity test methods, but also details a new test method for heating performance measurements that reduces the errors by 70%.

iv) Chapter 4: Presents the first accurate heating capacity test results and quantifies the problems that appear in heat mode for a standard single-compartment SL 400e trailer unit as part of Objective III.

v) Chapter 5: Addressing Objective IV, five different design changes are proposed and accessed to increase heating capacity and system efficiency. The impact of such modifications based on hot-gas injection technique is studied for the first time for a compressor running in low ambient temperature conditions. A complete comparison of the heating behaviour of both standard and modified design of the SL 400e trailer unit is overviewed.

vi) Chapter 6: Presents a characterisation of a standard single-compartment TS 500 truck unit based on field data obtained from a food distributor from Sweden. Three levels of problems are identified and confirmed by an extended in-house research using the new heating capacity test method, as part of Objective III.

vii) Chapter 7: The first mathematical model capable of predicting the heating behaviour of the standard TS 500 unit within ±6% accuracy is described. This chapter also addresses Objective IV and propose two design changes based on the effect of engine coolant used as a primary heat exchange fluid. The impact of these modifications is obtained through two mathematical models.

viii) Chapter 8: The conclusions of the research together with the contribution to body of knowledge in the field are overviewed in this closing chapter.

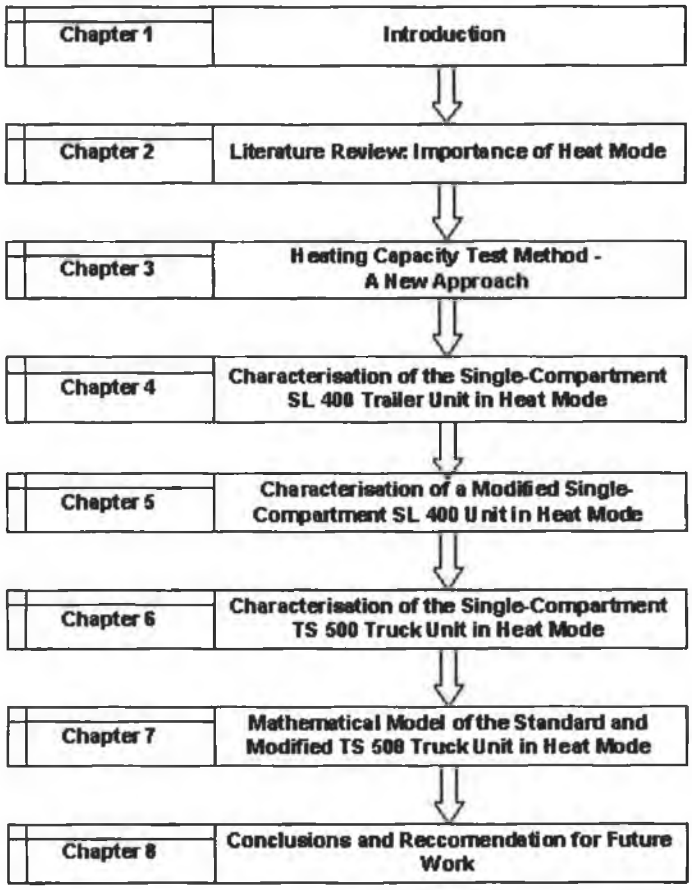


Figure 1.8 Thesis Layout.

1.5 CONTRIBUTION OF THE RESEARCH

The increased demand for heating versus cooling during temperature control transportation of food products is established for the first time [Lohan *et al.*, (2003)]. As a result, further research is undertaken and based on field data three levels of problems are established for TTC units. The analysis was extended through in-house experimental tests based on a new test procedure that measures the heating performance by 70% increased accuracy [Radulescu *et al.*, (2003)]. The need for improved heating performance resulted in several design modifications implemented and tested. Increased heating capacity by an average 30% was obtained while the systems operate with 40% higher efficiency [Radulescu *et al.*, (2005)]. For extended characterisation the first mathematical model capable to predict the heating capability and the behaviour of the unit in heat mode for wide temperature ranges with accuracy within ±6% was proposed and validated [Radulescu *et al.*, (2004)].

CHAPTER 2

IMPORTANCE OF HEAT MODE LITERATURE REVIEW

CONTENT:

While Chapter 1 highlighted problems at three levels due to insufficient heating capacity when the TTC units operate in cold climates, the importance of heat versus cool mode is established in this chapter. The findings of the literature review together with recent market and trade reports from the food transportation industry are presented. This marks the first comprehensive analysis undertaken to evaluate the importance of Heat Mode for temperature control during road transportation of perishable food products [Lohan *et al.*, (2003)].

2.1 INTRODUCTION

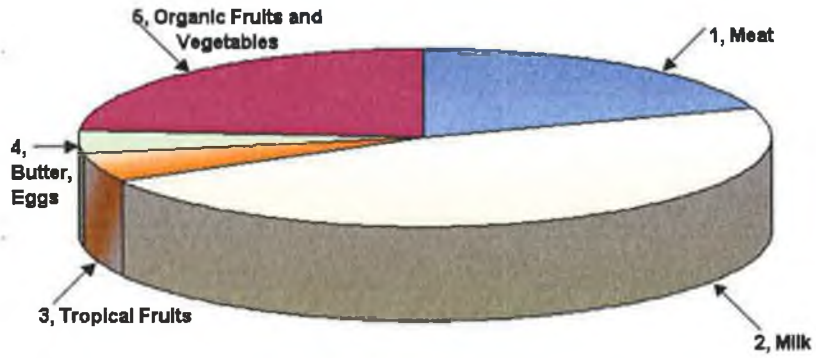
Transport temperature control (TTC) units are designed primarily to maintain product temperature, not to reduce or increase it and as a result both heating and cooling modes are required no matter what trailer set - point or external ambient air temperature exists. Exposing fresh product to even small temperature fluctuations of ±2% [ATP (1970)] can cause significant damage. High temperatures can cause surface pitting and dehydration, while low temperature can spoil loads by creating ice crystals. The term "low temperature" may mislead the reader towards thinking of temperatures at or below 0°C, but bananas for example are very susceptible to chilling injury if the temperature is below +12°C, or below +6°C for citrus fruits [ASHRAE Handbook (1994); ASHRAE Handbook (1998); ATP (1970); Hardenburg *et al.,* (1986)]. Therefore, transport temperature control systems must be capable of maintaining the temperature of commodities by automatically providing cooling and/or heating as necessary and this chapter seeks to establish the importance of heat mode relative to cool mode by reviewing:

- Recent changes in the geographical market for food products and TTC units.
- Recommended transport temperatures for different perishable food categories.
- Weather data reports for different climatic regions.
- Field data for a period of one year from a Swedish food distributor [Nilsson (2002), (2003)].

Combined, the evidence highlights for the first time the growing need for an effective and reliable heating mode for TTC units that can operate in heat mode for as much as seven months of the year, especially in countries north of 55° latitude [Lohan *et al.,* (2003)].

2.2 PERISHABLE FOOD PRODUCTS MARKET

The worldwide production of the most important food products that need temperature control is presented in Figure 2.1, based on the latest market report from the Food and Agriculture Organization of the United Nations, Commodities and Trade Division (2003), and a Guide to Food Transport: fruit and vegetables [Poulsen *et al.*, (1989)]. With 80.3 % classified as fresh products, it can be concluded from Figure 2.1 that the demand for fresh food transportation is higher than frozen food.



1 = 19.7% Meat Production – Frozen
2 = 47.2% Milk Production – Fresh
3 = 5.1% Tropical Fruits Production – Fresh
4 = 4.2% Butter and Eggs Production – Chilled
5 = 23.6% Organic Fruits/ Vegetables – Fresh

Figure 2.1 Market Report presented by the Food and Agriculture Organization of United Nations, [FAO (2003)].

Guilpart (2003) presents recent data for the food consumption in Europe, focusing on products that need temperature control during storage and transportation. While more detailed data is overviewed in Appendix A, it is acknowledged that 10% from the food product consumption consist in frozen products, while 90% consists of fresh foodstuffs that need transportation temperatures above 0°C. This is significant fact since TTC units need to generate heat to achieve these set-points in the food compartment when operate at ambient temperature conditions lower than 0°C.

In order to highlight the difference in the rate of market growth between fresh and frozen products, the following example is provided. As many developed countries currently import large quantities of organic products such as citrus fruits and bananas, the organic fruit and vegetable market is selected. This market has developed quickly over the last ten years with sales value growth of 20% to 30% per year since 1990 [FAO Reports (2003)], while the market for frozen food, meat for instance, has only grown by 5% per year, dropping to 2% during 1999 and 2000 [FAO Reports (1999)]. Even if singular, higher rates were observed for organic fruit and vegetable in the United Kingdom and Italy. For example, in Italy, organic food and vegetable sales have grown at an annual rate of about 85% during the period 1998 to 2000. The biggest exporters of organic food, such as bananas, are the South American countries like Dominican Republic and Argentina who have established a geographically dispersed worldwide market for their produce. For example the destinations by volume of the 53,347 tonnes that represents the total volume of organic products exported from the Dominican Republic in 2002 are presented in Figure 2.2. A representation of the geographically dispersed trade routes for these temperature sensitive products and weather data for January 2002 is defined in Figure 2.3 [FAO (2002); World Meteorological Organization-WMO (2002)].

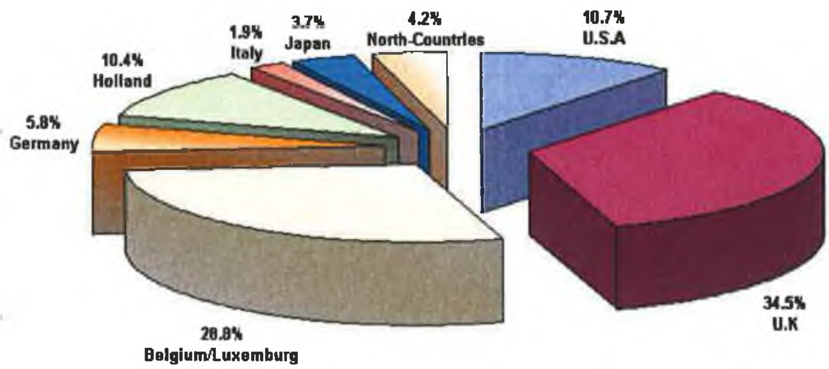


Figure 2. 2 final destinations by volume of organic products from the Dominican Republic in 2002 [FAO Reports (2002)].

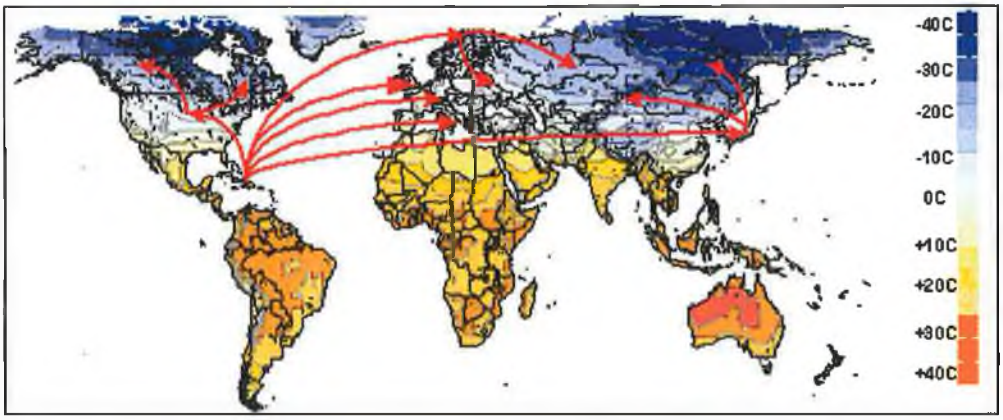


Figure 2.3 Geographical dispersed trade routes, shown in red, for organic products from Dominican Republic in 2002 [FAO (2002); WMO (2002)].

Despite showing a share of just 10.4% of the market as observed in Figure 2.2, Holland is one of the most important market destinations for fruit and vegetables, not only because of its own internal demands, but more so because Dutch companies play an important role in distributing large quantities of food to other European countries, from its Rotterdam port. For example, EOSTA, the largest Dutch organic trader in fruits and vegetables obtain 95% of its total turnover outside the Netherlands, predominantly in Scandinavia, where Holland is the gateway for fresh products [FAO Reports (1998), (2002)].

From the examples presented in Figure 2.2 and 2.3 it is clear that fruits and vegetables can start from Southern Hemisphere countries with an ambient temperature of +35°C and end in Northern Europe where the external air temperatures are often -20°C. Meanwhile, these products have to be maintained at transport temperatures between +5°C and +12°C and a thermal stability of ±1°C. This incredible development in the food transportation market would not be possible without a very reliable refrigeration system capable of maintaining optimum product temperatures by both cooling and heating as necessary.

More specific market reports obtained from Sweden show that the consumption of frozen food and perishables, with a requirement for temperature – controlled distribution, in 1998 amounted to [Swedish Market Report (2000)]:

- 3,400 thousand tonnes of chilled produce and perishables
- 345 thousand tonnes of frozen food
- 55 thousand tonnes of ice–cream with an annual growth estimated at 5%.

This data clearly shows that in 1998, the Swedish consumption of chilled food at 412.5 kg per capita was an order of magnitude higher than that of frozen food, at 41.3 kg per capita [Swedish Market Report (2000)]. From this data it can be concluded that 90% of food that requires temperature control is chilled and only 10% frozen. This is significant, since chilled food is transported at higher temperatures (0°C) than frozen food (-20°C). As already presented in section 1.1.2, this in turn puts greater emphasis on the ability of transport temperature control units to provide heat, especially during winter months.

2.3 TRANSPORT TEMPERATURE CONTROL UNITS MARKET

As a result of the increased consumption of perishable food products that need temperature control during transportation, the growth in the refrigeration market has also occurred relatively quickly as indicated in Figure 2.4. Market reports show a four-fold increase between 1980 and 2000 [Thermo King Market Report (2003)].

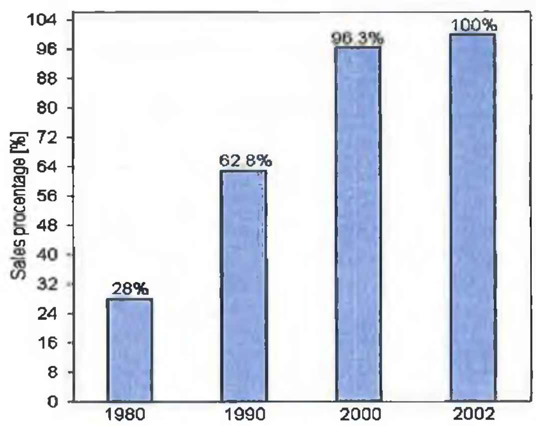


Figure 2.4 Sales Growth as a Percentage of 2002 sales during the period 1980–2002 [Thermo King Market Report (2003)].

The most important and established markets for refrigerated transport systems in Europe are located in countries such as Spain, Germany, France, Netherlands, Sweden, and Belgium. The European refrigeration system market can be sub-divided into the three geographical regions presented in Figure 2.5.

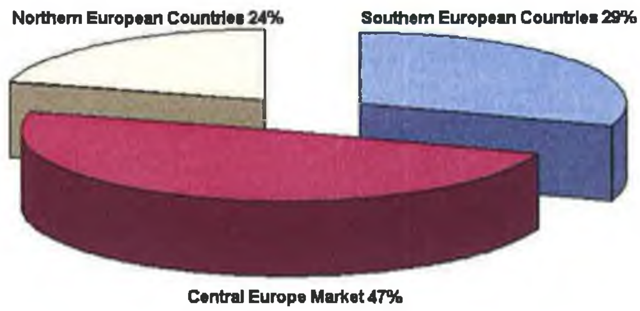


Figure 2.5 European Refrigeration Market [Thermo King Market Report (2003)].

Guilpart (2003) presents an evaluation of the TTC unit's market segmentation in Europe, Africa and Middle East, concluding that the European market has the most important share of 71%, with an average of 58,000 units were sold between 1990 to 2000 of which 26% trailer systems similar to those in Figure 1.1 (Appendix B).

2.4 TRANSPORT TEMPERATURE REQUIREMENTS

In this section, a review containing the following aspects of the food commodities storage and transportation is presented:

- Transport temperature requirements defined by international agreements.
- Classification of perishable food products.
- Economic loss due to food products deterioration as a result of unsuccessful attempts to maintain the narrow ranges for transport temperatures.

2.4.1 Temperature control requirements for food products

The preservation of the food quality is extremely important due to its direct impact on consumer health. Therefore, several regulations have been introduced to ensure that the consumer's best interest was protected.

The most important and comprehensive directive that defines the temperature control requirements for food storage and display are specified by the "Codex Alimentarius", established in 1962 by the Food and Agriculture Organization of the United Nations and the World Health Organization (FAO/WHO). The purpose of the joint was to define a common set of regulations entitled Food Standards Programme used to:

- Protect the health of consumers and to ensure fair practices in the food trade.
- Promote coordination of international organizations.
- Initiate and guide the preparation of draft standards and codes of practice.

In pursuit of common set of rules relating to the production, marketing and distribution of food, the European Council of the United Nations had issued several directives regarding the quality for fresh, chilled and frozen products summarized in Appendix C [EEC 64/433 (1964); EEC 89/108 (1989); EEC 71/118 (1971)].

It is acknowledged that in 1991, a very important regulation regarding the transportation of a new range of hot food products was introduced in the U.K. This regulation was published as a result of increased awareness of the role that temperature plays in preventing food related illness. As a result, in addition to the introduction of the European Council Directive 89/108/EEC for quick frozen food, the U.K Government changed its food hygiene regulations with the Food Hygiene

Amendment (1991), covering the retail, sale and distribution of chilled and hot foods. The regulation specifies three temperatures: +8°C and +5°C for chilled foods and one for hot foods +63°C. This is a significant example as it shows the increased the interest and demand for transport temperature control units to produce heat.

Under the regulations presented, the quality of food can only be maintained by operating very tight temperature control through out the cold chain. The only permitted changes in transport temperature are for fresh and chilled product temperatures of +8°C or +5°C. In this case a maximum temperature increase of just +2°C is permitted, but not for longer than two hours during temporary equipment breakdown, movement of food from one part of the premises to another, market stall or transfer from one vehicle to another, vehicle door openings or some other unavoidable reason [64/433/EEC (1964); 71/118/EEC (1964); Frith (1991)].

This wide palette of regulations regarding the optimum temperatures required to maintain food quality has lead to similar requirements being imposed on the food transportation industry.

2.4.2 Temperature requirements for food transportation

Most developed countries have approved regulations for temperature control and equipment performance evaluation for all stages of the "cold chain", from producer through distribution and onwards to the consumer. The most important requirements defined for the transport temperature control units focus on the TTC unit performance together with the thermal insulation and dimension of the trailer body and the optimum temperature ranges for the carriage of perishable food products. Standards such as the *"Agreement of the international carriage of perishable foodstuffs and on the special equipment to be used to such carriage"* (ATP), the EEC directives presented and the American Standard 1110–2001 and organizations like UNECE, American Society of Heating, Refrigerating and Air–Conditioning Engineers (ASHRAE) and the International Institute of Refrigeration (IIR) try to harmonies the regulations within European and worldwide countries.

The most comprehensive regulation of transport optimum conditions for perishable foodstuffs conforms to the ATP standard established by the United Nations

Economic Commission for Europe [UNECE (1970)]. Under this agreement contracting parties are obliged to take all measures necessary to ensure that equipment and carriage of perishable foodstuffs respect technical standards defined in the agreement. These technical requirements are specified for vehicles transporting temperature–controlled products between the 39 member countries of the agreement, mainly based in Europe.

For temperature controlled transportation, the regulations also require two air temperatures to be continuously measured and recorded, one sensor being placed in the air return and a second positioned in the ceiling of the food compartment at about $^1/_4$ of its length away from the TTC unit, so called "top air delivery" temperature [Appendix C, Figure C.1]. The sensors should be mounted in such a way as to avoid damage during normal use. The sensors should be connected to suitable recording instruments that can be permanently fitted to the trailer or vehicle cabin, which in the case of a trailer is by a detachable connection. Small temperature loggers with built-in sensors can also be used in place of fixed sensors and an automatic recorder. These can be removed after each journey for analysis. An example of a screen shot of the results obtained with such a data logger is presented in Figure 2.7. Multi–compartment vehicles designed to carry food at different temperatures in the different compartments should have sufficient air temperature measurements made for each compartment, so that the performance can be accurately assessed. This may entail the use of multi channel recorders and several small data loggers as an alternative. These rules for sampling product temperatures at the various stages of the cold chain are comprehensively covered in the Department of Health Publication: Guidelines of the Food Hygiene Regulations 1990 [SI (1990) No. 1431].

2.4.3 Classification and Transport Temperatures for food products

In this section a general classification of the main perishable food products together with the transport temperature requirements is presented.

A) Classification

The general classification of food commodities for which transport temperature criteria has been defined [ASHRAE (1994); ASHRAE (1998); ATP Geneva 1970; Hardenburg *et al.*, (1986), Frith (1991)] includes:

i) Fresh and Chilled Produce: Fresh products include perishable foodstuff such as fruits, vegetables and miscellaneous commodities. They are all living organisms and produce heat as they respire. The quantity of heat generated depends on the variety of fruit or vegetable and usually varies with the product temperature. Chilled products include meat, dairy produce, fish, butter, cheese and eggs transported at temperatures above 0°C to avoid freezing.

ii) Frozen Produce: The most important frozen products transported are meat, frozen fish and poultry products like domestic fowl, duck, geese and rabbit. Frozen foods deteriorate very slowly and the lower the transport temperature the lower the rate of deterioration and consequently increased shelf life. Deterioration appears as a loss in quality rather than any dramatic change and is the results of chemical activity such as oxidation and physical changes resulting from evaporation and the growth of ice crystals.

iii) Dried Produce: The dried food category includes products that are dried during manufacture like milk powder and many chemicals. These are best stored in sealed insulated equipment and ventilation. Moist air should be avoided.

B) Optimum Transport Temperatures

Based on the classification presented above, both this section and Appendix C presents the optimum transport temperatures and requirements [ASHRAE (1998); ATP Geneva 1970; Hardenburg *et al.*, (1986), Frith (1991)].

i) Fresh and Chilled Products: Not only does the chilled and fresh food sector occupy 80.3% of the food market, this sector also offers more risk and more temperature control difficulty than the frozen food sector. Difficulties mostly relate to the very narrow temperature ranges imposed, and the difficulty for the consumer or the retailer to evaluate the remaining life span of the product, or even to perceive any risk if damage has been caused during transportation [Commere (1999)]. As a result, the following tight guidelines apply for the main categories of fresh products: fruits, vegetables and miscellaneous commodities. Transport temperatures for fruit fall into two categories:

   - Fruits that are essentially tolerant of low temperatures below +4°C and are transported at temperatures within ±0.5°C of recommended temperatures. The aim is to carry the particular fruit near the freezing point, taking into account control temperature variations and trying to avoid freezing.

- More sensitive fruits are carried at higher temperatures such as +12°C, which is a compromise between the harmful effects of low temperature, which may result in chilling injury, and the benefit from low temperatures such as slow ripening and retarding the rate of decay [Frith (1991)].

Most temperate vegetables are tolerant of low temperatures and are carried close to 0°C. However, higher temperatures are specified for certain vegetables, which would otherwise suffer from chilling injury. These include cucumbers, marrows and most tropical vegetables. Another category of fresh products that need temperature control consists in miscellaneous commodities. The most common products in this category include plants, bulbs and chocolate. Most plants are not tolerant of low temperatures and are carried at temperatures around +21°C.

Chilled foods must be carried at temperatures between about –1.5°C and +5°C and more recently an upper temperature of +8°C is required for some products such as salted pork [Frith (1991)]. Difficulties may arise when some chilled products like eggs and chilled meat are transported with the required return air temperature of between –1°C and 0°C in very low ambient temperatures. In these conditions the heating capacity of the TTC unit has to be enough to maintain strict temperature range around zero, in order to avoid freezing, which can start at temperatures lower than 0°C [Appendix C].

ii) Frozen Products: The most important temperature ranges for frozen food products are below –15°C. The lower the temperature, the lower the rate of deterioration and consequently increased shelf life. Another temperature range of –10°C is used during the transportation of frozen red offal and eggs. The highest temperature at any point in the load during carriage should not be higher than the figures presented below. However, if certain technical operations of the mechanical refrigerated equipment cause a brief rise for a limited extent in any part of the load, a temperature rise by not more than +3°C above the temperatures indicated for each foodstuff is not tolerated.

Recommended transport temperatures described in this section and Appendix C represent optimum values, which can provide good storage and transportation conditions for commodities in order to avoid deterioration. In general, the heat mode is not so important for frozen food transportation and is mainly used to maintain the set-point temperature as indicated in Figure 1.5(a). However, with multi-temperature refrigeration systems, Figure B.1 (Appendix B), the same host refrigeration system might be required to maintain appropriate temperatures defined for fresh, chilled and frozen foods in different compartments. It can be concluded that transport temperatures for frozen foods are between -20°C and -18°C, while for all fresh and chilled products the temperatures are always above 0°C, ranging between 0°C and +15°C. In these conditions the heat mode importance is increased, as the fresh products are often transported in sub zero ambient temperatures.

2.4.4 Influence of Temperature Variations on Food Quality

The stability of the recommended air temperature during the transportation of perishable food products is critical, as it influences the deterioration rate more than any other environmental factor such as relative humidity, mobility, pressure, composition [Lucas (1998), Despre *et al.,* (1998), Nordtvet *et al.*, (1999)]. The US Department of Agriculture carried out studies on a large variety of food products such as frozen products, fruits, vegetables, chickens and turkeys. It was concluded that the deterioration rate is highly dependent upon storage and transport temperature, and the higher the storage temperature the more sensitive are quality losses. Therefore, temperature impacts greatly on the aging process that has to be carefully stopped or slowed down.

Fresh products like fruits, vegetables and flowers produce energy by combustion of sugar in a biological energy generator. The process of energy generation in living organisms is called "respiration", which acts as a generator for an extremely complex and interrelated series of enzymes that function as biological catalysts. In general, the activity of these enzymes is a function of temperature and it doubles for every 10°C rise in temperature [Frith (1991)]. Because respiration is a complex process depending on a large number of enzymes, the rate of respiration also rises, typically between 2 to 4 times for every 10°C rise in temperature. Taking a typical food product, this means that the rate of respiration at +30°C is in the order of 25 times that at 0°C. Since the rate of aging depends on the respiration rate, controlling the temperature is the primary and most important means of reducing aging and quality loss in perishable products [Reid *et al.,* (1999); ASHRAE Handbook (1994); ASHRAE Handbook (1998); ATP Geneva (1970); Hardenburg (1986)]. It has to be noted that while frozen products do not need very tight temperature control, the real challenge from this point of view is to maintain the transport set-point temperature for fresh and frozen foodstuffs that have narrow temperature ranges around the set point within ±2°C [ASHRAE (1998); ATP (1970)]. The key studies that have contributed to our understanding of these issues are summarized below for the most important fresh products that require temperature control: i) fruits, ii) vegetables and iii) chilled meat and fish.

A) Influence of temperature variations on fruits

Hardenburg *et al.,* (1986) showed that the temperature of pears has to be maintained within the range of −1.5°C to −0.5°C. If the temperature falls just 1°C below −1.5°C, then the fruit will freeze. In contrast, if the temperature is above – 0.5°C by as little as 1.5°C for 10 days, the shelf life of the pear is shorted by one week. Another study performed on strawberries after storage at temperatures that fluctuate by ±3°C, showed that the most significant deterioration over 6 days was weight loss [Nunes *et al.,* (1999)]. Over the past 10 years, researchers from New Zealand and Australia have undertaken intensive studies of air and fruit temperatures within transport systems carrying perishable products, predominantly fruits from Australian ports to Asian and European destinations. For example, Amos *et al.*, (2003) quantified the effect of measured temperature differences on the firmness of kiwi fruit transported by container from the initial harvesting in New Zeeland to the final destination in Spain some 3 months later. It was noticed that transport temperatures during the first half of the shipment fluctuated by approximately ±4.4°C about the recommended ±1°C transport temperature range, and this reduced to approximately 3°C by the end of the voyage. The minimum temperatures reached in the container was very close to the freeze limit of −1.2°C. A firmness assessment performed at the destination concluded that due to the high variations in the carriage temperatures, the firmness of the fruits encountered a decrease of 37.5%, from the initial 8kgf measured to the final 5kgf figure. The analysis of the temperature variation for in – package temperatures in the container for the entire voyage indicates that the temperatures were outside the recommended 0°C to +2°C transport set-point for 42% of the time, and this accelerated the rate at which fruit firmness deteriorated. It can therefore be concluded that there is a strong link between changes in fruit firmness and temperature profile. Fruit consistently exposed to temperature fluctuations of ±2°C having significantly different final fruit firmnesses.

B) Influence of temperature variations on vegetables

Important results were obtained for broccoli and cauliflower by both Despre *et al.*, (1998), and McEwan *et al.*, (1990). In a controlled test environment, samples of broccoli and cauliflowers were stored at a controlled optimum ambient temperature and at the same time, similar samples were stored in an environment where the temperature fluctuated by ±3°C during storage. The analyse of pH, treatable acidity, colour and weight performed after 10 storage days showed that there were significant 30 % quality losses for the sample exposed to fluctuating temperatures.

C) Influence of temperature variations on chilled meat and fish

Chilled Meat and Fish: Studies performed on samples of chilled fish stored at different temperatures show that the optimum temperature is 0°C, but an increase of 10°C reduces the storage life in half [Nordtvet *et al.*, (1999); Royvrick (1979)]. Magnuss *et al.*, (1999) did a complete analysis regarding the dependence between fresh fish shelf life and storage temperature. The experiments performed on chilled cod indicate that for 0°C the maximum shelf life (SL) of approximately 14 days is obtained. At temperatures above this limit an exponential fall in the shelf life was found as follows: i) For 4°C; SL falls to 7 to 8 days, ii) For 8°C; SL falls to 5 to 6 days and iii) For 12°C; SL falls to just 3 to 4 days.

The dependence of product quality on temperature control has also lead to numerical simulations and modelling of the transient temperature behaviour of different foods. Gigiel *et al.*, (1998) showed that the optimum design of transport temperature control units can only be obtained when a firm understanding of the interaction between the temperature of the foodstuff and the transport vehicle construction, the air movement within the vehicle, the external environment and the operation of the vehicle is achieved. With so many interacting variables it is difficult to obtain an in - depth understanding of all aspects of system design and operation required to maintain recommended temperatures. In these conditions the necessity and importance of modelling tools becomes obvious. For example, Houska *et al.*, (1998) analysed the thermal history of chilled chickens during storage and transportation, whereas a generic model capable of estimating the time that perishable produce can endure at different ambient temperatures was developed by Tijskens and Polderdijk (1996). A more general model capable of predicting the transient temperature of all categories of chilled food during transportation has also been presented by Tijskens *et al.,* (1998). Due to the sensitivity of fresh products to transport temperature, models were obtained based on TTT (Time-Temperature-Tolerance) and PPP (Product-Process-Packaging) concepts and recommendations were made for practical use [Skeef (1998); Balan (1998)]. An important part of this research has also been invested to improve the quality of perishable foods during transportation [Nunes *et al.,* (1999)], by achieving a better understanding for the weak links in the distribution cold chain [Kennedy *et al.,* (1998)]. However, while these models are useful, this study could not benefit from their use since the transient thermal behaviour of the food was not being investigated.

Transport of perishables, such as fruits and vegetables, to isolated northern communities in parts of Canada has always been a big challenge. Because of the unique conditions (extreme cold, strong winds, snow storms, abundant rain) encountered throughout the distribution process, the cold chain is rarely maintained resulting in enormous losses of perishables upon arrival to the northern communities. Emond *et al.,* (2003), presented an interesting study for this case, having the main objectives to: i) present the situation of the cold chain and ii) identify the weaknesses of the distribution process. A load of fruit and vegetables transported was monitored from both temperature and humidity point during a 60 hours journey using a truck TTC unit, from Montreal throughout northern cities like Quebec and Wabash. Due to the external –33°C ambient conditions, the average temperature recorded inside the compartment was +7.2°C while the desired set point was +10°C. Large temperature variations were observed during transportation. Average temperatures up to +7.9°C were recorded in the front area of the compartment while temperatures as low as –2°C were encountered at the rear. Extremes of –3.9°C were also recorded. While such temperature variations can cause injuries in fruit and vegetables even for a short time of exposure, the effect was very severe for this three days journey. The evaluation of the quality of fruits and vegetables at the destination showed chilling injuries, water losses, bruises, and decay of green peppers; spotting and decay on iceberg lettuce, severe browning on mushrooms, resulting in a massive 50% product loss during transportation due to low heating capacity that has as result the impossibility of maintaining the set point temperatures [Emond *et al.*, (2003)].

EU legislation requires tight tolerances of ±2°C in the recommended temperatures of all food [ASHRAE Handbook (1998)]. The desired temperature control defined is within ±0.5°C of the desired set–point temperatures (Appendix C). However, while this strict temperature control is achieved not only by cooling but also by heating, it has been noted that very few studies have highlighted the important role of Heat Mode in achieving this level of temperature control [Lamothe *et al.*, (1998); Barenboim *et al.*, (1998)].

2.4.5 Economic Loss Analysis

If the consumer health is protected by regulations and inspections regarding the quality of food products, another important aspect that has to be considered is the economic loss that results from damaged products that have to be destroyed as a result of inadequate temperature control. This section presents the economic impact of food quality deterioration.

As fresh fruit and vegetables move through the various steps of the distribution chain the likelihood of spoilage increases. This phenomenon can lead to a change in the state and the value of the goods in a variable space and time context. While losses incurred between harvesting and market are estimated as being as high as 50% worldwide, this is reduced to as little as 10% in countries like the United States where food distribution networks are mature and optimum storage conditions are offered [Harvey (1978)]. These economic losses result in a reduction in the turnover of the typical supermarket fruit and vegetable department, by between 2.5 % and 8.5 % [Laborde (2002)]. Moras (2003) present the impact of deterioration of strawberries and lettuce during transportation on the losses incurred by a typical 18,000m^2 supermarket of which 650 m^2 is devoted to fruit and vegetables. This supermarket is representative of an average store that has an estimated turnover of about 9 million Euros from fruits and vegetables. It was concluded that 60% of the total product losses result from fresh fruits and vegetables stored at room temperature that is 12°C higher than the optimum temperature, which results in a 45% decrease in turnover. In the same study, Moras (2003) also quantifies lettuce sales loss at approximately 1500

Euros per month due to wilting and quality loss, representing a 0.15% drop in turnover, whereas strawberries cost losses due to problems of crushing and over – ripeness are up to 7800 Euros per month, representing 0.20% drop in turnover. Clearly poor temperature control has contributed significantly to these considerable losses of 111600 Euros per year representing a 4.2% yearly drop in turnover only due to lettuce and strawberries sales loss.

Balan (1998) also presents the impact of losses in food quality on economic indices. Loss of product quality depends proportionally on the temperature variations from the recommended transport temperature range. Minimum losses are suffered at the optimum storage temperature. If the temperature is too high, losses increase because of the mass loss and microbiological infection, while too low temperature has results in losses from physiological damage. The quality loss is related with the cost of the product and automatically with the profit of the food suppliers. If the product is 50 % lowered in quality it will be sold for 70% of the initial costs [Balan (1998)].

The review presented in sections 2.2, 2.3 and 2.4 shows increased consumer demand for fresh and chilled food products that represent 80% from the total worldwide food consumption. This sector also offers more risk and more temperature control difficulty due to tighter required temperature ranges within ±2°C. A review of the negative influence of an inappropriate temperature control on the quality and shelf life of the fresh products is also undertaken.

2.5 CLIMATIC TEMPERATURES BY GEOGRAPHICAL LOCATION

By matching the recommended transport temperatures presented in section 2.4.4 and Appendix C with typical climatic temperatures in different geographical regions it was possible to gain a different insight of the importance of heat mode during TTC unit operation. A wide variety of weather data, from locally recorded hourly values to minimum and average air temperatures for different periods of time were available [List of International Weather Data Sources, References]. The average year-round monthly temperatures based on the last ten years data is presented in Figures 2.5 to 2.7 for northern hemisphere countries such as Canada, Sweden and Eastern Europe where temperature controlled food transportation is developed.

2.5.1 Northern Hemisphere

It is estimated that the TTC units market in these countries has increased by a significant rate of +10% per year in the last decade. Based on the data presented in Figure 2.6, in northern regions of above 55° latitude, like Canada, the TTC units run in heat mode for all types of chilled and fresh products during the whole year.

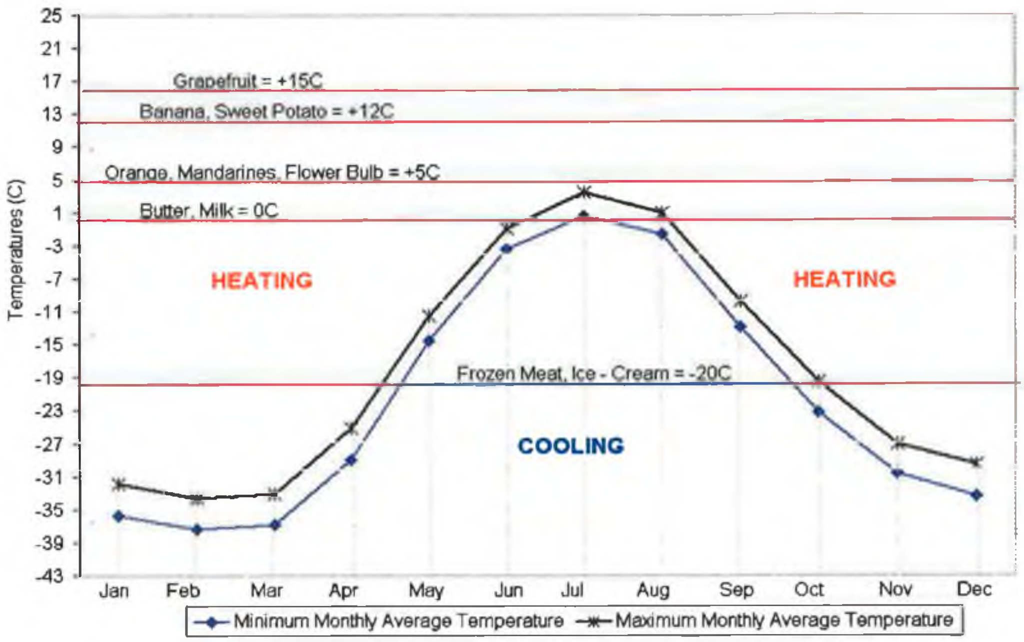


Figure 2.6 Cooling and Heating Modes versus year round monthly average air temperature in Northern Canada based on meteorological data between 1990 – 2000 and recommended set point temperatures for different products.

Even for the most important frozen products like ice – cream and frozen meat with a –20°C recommended transport temperature set-point, the TTC unit operates in Heat Mode for autumn and winter months because of the extremely low external ambient temperatures of sub -30°C for December to March. As a general conclusion it can be said that the TTS unit runs in Heat Mode for 100% of the time for fresh products and 50% of the time for frozen products.

Similar data is presented in Figure 2.7 for Sweden. It can be concluded that for fresh and chilled products with transport temperatures within the 0°C to +5°C range, the TTC unit runs in Heat Mode for six months, during autumn and winter months. For fresh products with transport temperatures within +8°C to +15°C, the TTC unit operates in Heat Mode for a longer period of time, almost 10 months. However, during the summer months of July and August, the unit runs predominantly in cooling. For frozen products the unit operates mostly in Cool Mode for the whole year, while only during the coldest months of January and February the temperature set point demands Heat Mode.

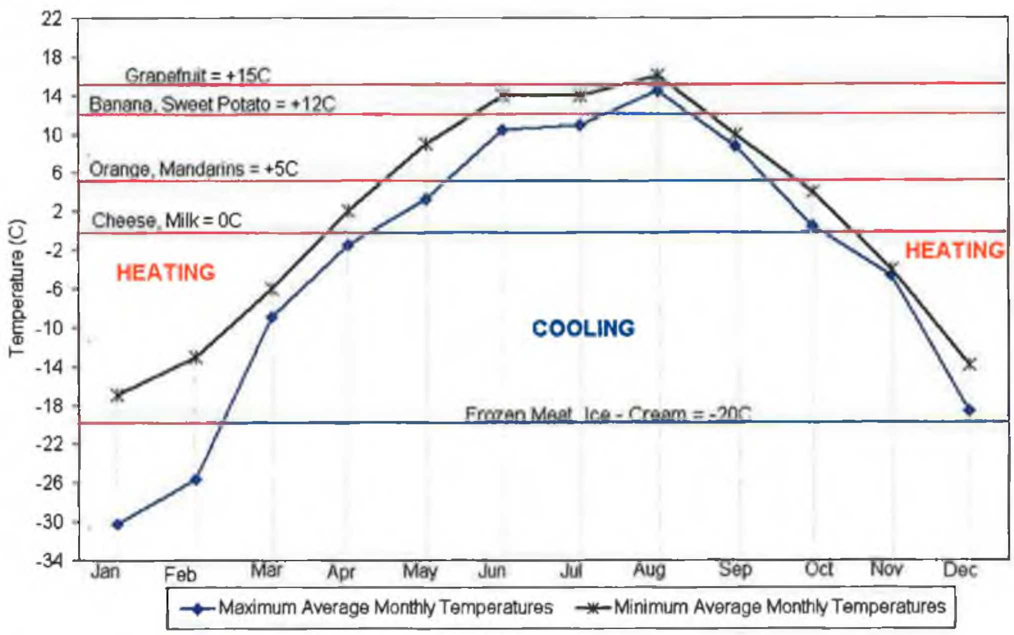


Figure 2.7 Cooling and Heating Modes versus year round monthly average air temperature in Sweden based on meteorological data between 1990–2000 and recommended set point temperatures for different products.

As a general conclusion it can be observed that the TTC unit runs in Heat Mode for 17% of the whole year for frozen products, around 50% for chilled products within 0°C to +5°C temperature range while the TTC unit operates in Heat Mode for 83% of the year for fresh products with transport temperature higher than +8°C.

2.5.2 Central European Countries

Similar climatic data is presented in Figure 2.8 for Central European countries like Poland, Germany, Czech Republic and Romania. For chilled products like butter and milk the TTC unit operate in heat mode for 4 months during winter months. For fresh products with transport temperature higher than +8°C, the TTC unit operates in Heat Mode for a longer period of five months per year. The cool mode is used during the whole year for frozen products. It can be concluded that for frozen products the unit does not need heat mode except for temperature control, for chilled products within

0°C to +5°C around 33% of the time, while for fresh products with transport temperature higher than +8°C the TTC unit runs in Heat Mode for 42% of the year.

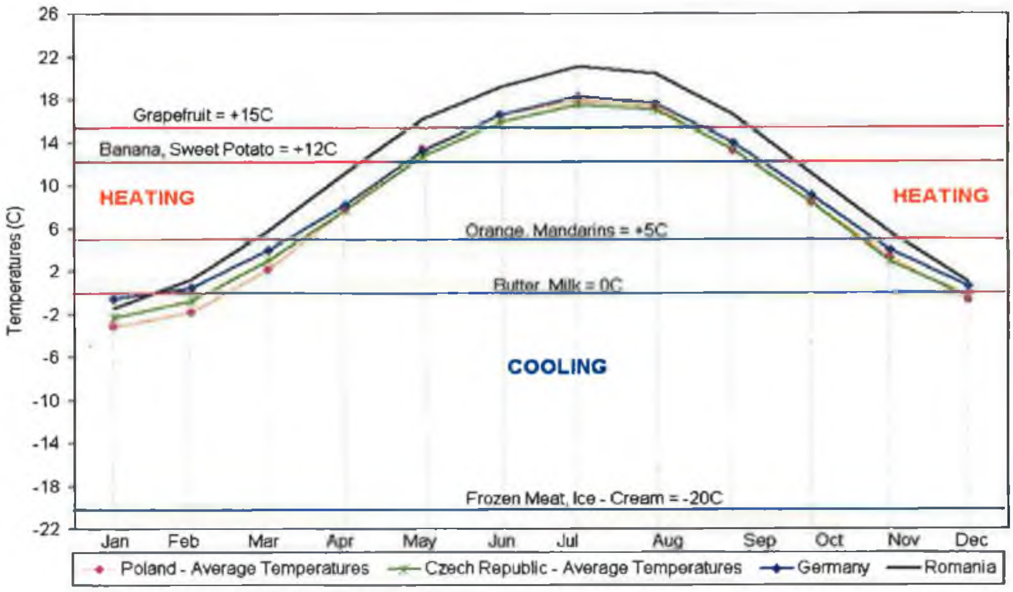


Figure 2.8 Cooling and Heating versus year round monthly average air temperature in Central Europe based on meteorological data between 1990 – 2000 and recommended set point temperatures for different products.

Figures 2.6 through 2.8 show that based on recommended set-point temperatures for the most significant food products and on meteorological data heat mode has to be recognized at equally important with cool mode. The first estimation of the demand for heating versus cooling modes is described in the following section.

2.5.3 Estimating demand for heating versus cooling mode

Analyzing the data presented in Figures 2.6 through 2.8 and taking into account the fact that transport temperatures for all fresh products are above 0°C, it can be easily concluded that in Northern Hemisphere countries such as Canada, Scandinavia and Russia, that the temperature control unit must run in Heat Mode for at least six to seven months per year. However, in Central Europe, America, and Asia the need for a full - time use of heating for fresh foods reduces to approximately two and a half months per year. For frozen products on the other hand, Heat Mode is only required for three or four months per year in the Northern Hemisphere, while in Central Europe, America, and Asia the need for a full time use of heating for frozen products reduces to approximately two months per year. Market reports presented in section 2.2 highlights that there is an increasing consumer demand for chilled and fresh food such as milk, dairy, products, meat, fish, fruits and vegetables. In "Refrigeration and the World Food Industry", it is estimated that from the total 1990 production of food in the world, almost 40% requires temperature control with only 15% being frozen [Kaminski (1995)]. The fact that, in developed countries, consumers purchase approximately ten times more chilled food (in weight) than frozen food [Lucas (1998)], it makes it possible to estimate the requirement for Heat Mode versus Cool Mode more precisely, as: i) 66% Heat Mode requirement for the transportation of food products for the whole year in Northern Countries (Table 2.1) and ii) 40% Heat Mode requirement obtained for the whole year in Central European countries as presented in Table 2.2 [Lohan *et al.*, (2003)].

Table 2.1. Demand for heat versus cool for Northern Hemisphere Countries [Lohan *et al.*, (2003)].

Time of Year	Duration	Fresh Products	Frozen Products
Winter Months December/ February	3 months	100% Heating	100% Heating
Spring and Autumn March/ April September/ November	5 months	90% Heating 10% Cooling	100% Cooling
Summer Months May/ August	4 months	10% Heating 90% Cooling	100% Cooling

Table 2.1 shoes that Heat Mode is required for 66% of the time for fresh foods while just 33% of the time for frozen products. Considering the fact that fresh food products represent 80% from the entire market, for a total 66% of the whole year the TTC unit is required to operate in Heat Mode.

Table 2.2. Demand for heat versus cool modes for Central European Countries [Lohan *et al.*, (2003)].

Time of Year	Duration	Fresh Products	Frozen Products
Winter Months December/ February	3 months	100% Heating	100% Cooling
Spring and Autumn March/ April September/ November	5 months	40% Heating 60% Cooling	100% Cooling
Summer Months May/ August	4 months	100% Cooling	100% Cooling

Table 2.2 shows that Heat Mode is required for 50% of the time for fresh products and 100% of the time for frozen products. An overall 40% demand for Heat Mode of the whole year was obtained for all food products in Central European countries.

2.6 FIELD DATA FROM SWEDEN

As temperature is one of the main factors affecting product quality, the European Community Directive EN 92/1 CEE has made the recording of air temperature during transportation obligatory since 1990. The concept of "traceability" has been proposed and existing tools such as temperature recorders, temperature indicators, and thermometers have been reviewed [Commere (1999)]. An example of the output from such a data acquisition system is presented in Figure 2.9.

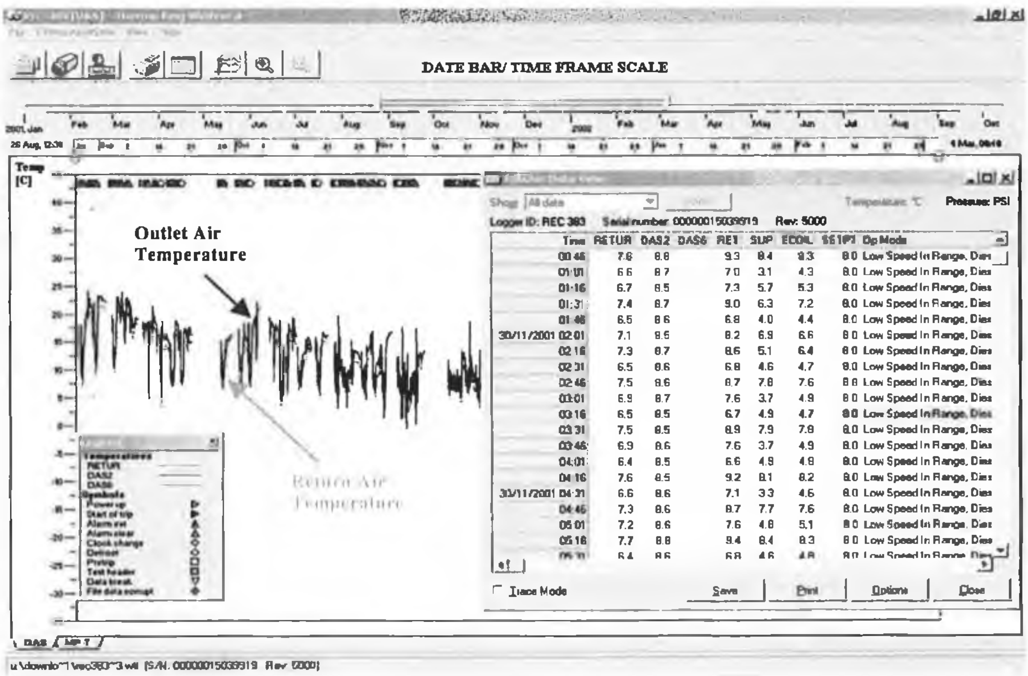


Figure 2.9 Screenshot from the Wintrac data recording and analysis interface.

The main reasons of acquiring data during the cold chain are: i) to confirm correct temperature during transportation, ii) to verify that the conditions during transportation have met the required standard iii) to confirm that the storage temperatures are at the required level, iv) to ensure that the processes are carried out as cost effectively as possible and v) in order to facilitate analyses the data must be accurate, easy to understand and simple to use.

The implementation of these data recording directives enables the analysis performed in the previous section to be supported by field data obtained from a Swedish food distributor, who employs such a data recording system [Nilsson (2003)]. This field data presented in Figures 2.10 and 2.11 highlights both the heating and cooling durations over a one-year period taken during 2001–2002, during the transportation of fresh food products. This field data has been generated as Wintrac 4 software. The program was capable of recording and presenting return air temperature (temperature of air returning to the TTC unit from the compartment), set point temperatures, date and time, alarms and the run mode of the refrigeration unit during transportation in either a tabular mode or in a chart from [Figure 2.9]. Figure 2.10 highlights the temperature profiles experienced during the transportation of fresh fruit and vegetables between October 2001 and September 2002. The recorded set–point temperature and return air temperature to the TTC unit from the compartment are both presented, along with the average external ambient air temperature, which was obtained from the Swedish Institute of Meteorology and Hydrology.

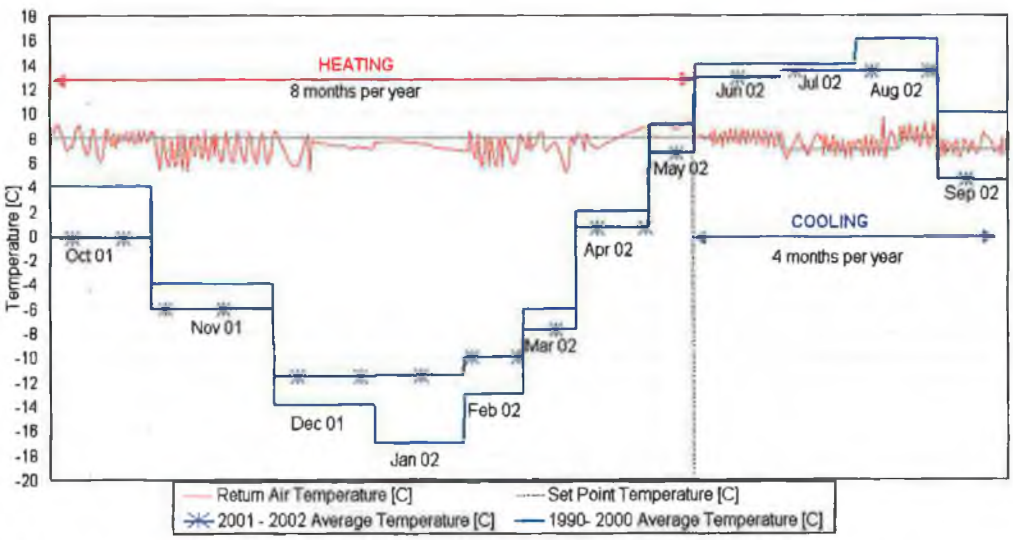


Figure 2.10 Heating versus cooling time for truck unit in Sweden during 2001/ 2002 [Nilsson (2001)]

Note the close agreement with the conclusions drawn from the ten–year long average data in Figures 2.5 through 2.7 and tabulated in Tables 2.1 and 2.2 and that indicated by the actual data for year 2001/2002 in Figure 2.9. Both sets of data indicate that heating is required for eight months of the year. A particular situation is plotted in Figure D.2 (Appendix D), where the unit operates in heat mode for the whole year.

The field data presented in Figure 2.11 highlights the performance of a commercial TTC unit when the ambient temperature drops below –17°C experienced in January. This situation does not reflect the worse case ambient temperatures that can exist. It is common in many countries for the ambient temperature to drop as low as –30°C for extended periods of time as indicated by the data presented in Figure 2.6. While only daily average external temperatures are shown in Figure 2.11 it is clear that the TTC unit struggles to maintain set-point temperature of +10°C, once the external temperature drops below –14°C. The first clear evidence of this lack of heating capacity is exposed during January 23rd and 24th, when the external ambient reached a low value of –28°C the unit can only maintain a +5°C return air temperature while running in low speed diesel mode. As a result, the unit has to run in high-speed mode, where with 60% increased capacity should be capable of maintaining the set point. It can be noticed a forced increase in the set point temperature for maximum heating capacity operation. During December 25th, the extreme low ambient temperature of –30°C, coupled with the lack of heating capacity, caused the compartment temperature to drift by up to 8°C below the +10°C set-point. As a result the unit is forced to operate in high speed diesel mode, but a lack of heating capacity still resulted in a compartment temperature that was +2°C below the desired set - point. During the 26th and 27th of December, for milder ambient temperatures of –22°C and –25°C, the unit maintains a lower set-point temperature of +8°C.

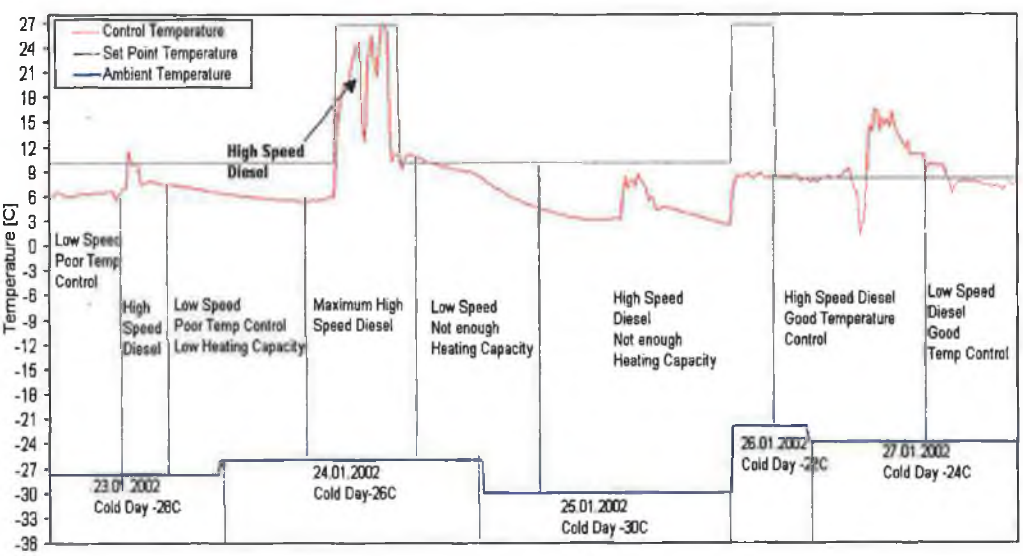


Figure 2.11 Swedish Field Data from a Thermo King dealer – TTC unit TS 500 – January 2002 [Nilsson (2002)].

These practical field data highlights that there is insufficient heating capacity in low speed diesel to maintain internal set-point temperatures higher that 8°C for ambient conditions sub –15°C and the unit is forced to run in high-speed diesel mode for good temperature control. For extreme ambient temperatures below –28°C, the unit can maintain only a maximum +8°C set-point temperature in the compartment even if it is running in high-speed diesel mode.

2.6 SUMMARY

The importance of Heat Mode for transport temperature control has been established for the first time in this Chapter by reviewing:

- The changing geographical market for food products and transport temperature control units.
- The influence of the temperature control during food transportation on the food quality and shelf life.
- The demand for Heat Mode versus Cool Mode based on climatic temperatures by geographical location and transport set-point temperature requirements for the most important food products.
- Problems that appear during heat mode based on field data from Sweden.

Based on the analysis presented the most important conclusions are summarized:

- 80% of the food transported is fresh, as opposed to frozen, and requires to be transported at set-point temperatures above 0°C. Thereby increased pressure on optimum performance in Heat Mode appears.
- Very tight temperature control tolerances of ±1°C are allowed by most important regulations: ATP (1970) and Standard 1110 (2001).
- The demand for Heat Mode over Cool Mode is 66% for Northern Hemisphere countries and 40% in Central European countries.
- The practical field data obtained for TTC units operating in Sweden show insufficient heating capacity to maintain the transport set-point temperature for ambient conditions sub –15°C.

The main contribution of the review presented in this chapter consists not only in a comprehensive analysis of the temperature control requirements during transportation of perishable food products but more important quantifies for the first time the importance of heat mode versus cooling. The field data presented in Figures

1.6 and 2.11 show a discrepancy between the poor heating performance of the TTC units on field and the sufficient heating capacity figures obtained from the manufacturer technical specifications, measured based on the existing standard ATP test procedure that has a poor accuracy of ±10% [ATP (1970)]. As a result, a more detailed characterization of TTC units in Heat Mode is required starting with the development of an accurate heating test procedure and dedicated test facility as proposed in Chapter 3.

CHAPTER 3

MEASUREMENT OF HEATING CAPACITY: A NEW TEST METHOD

CONTENT:

As shown in the previous chapter, transport temperature control (TTC) units used for the transportation of perishable foodstuffs can operate in heat mode for up to 60% of the year in regions north of the 55° latitude [Lohan *et al.*, (2003)]. Despite the clear importance of heat mode, the current test method used to evaluate heating capacity lacks accuracy. While the existing standard method is reviewed in this chapter, details of a new test procedure capable of accurately measuring the heating capacity of TTC units operating in sub zero degree Celsius ambient temperatures are also presented. This more accurate test method is both novel and generic and can be applied to determine the heating capacity of any TTC unit that employs hot gas as a working fluid to an accuracy of ±1% to maximum ±3% [Radulescu *et al.*, (2003)].

3.1 INTRODUCTION

While the performance characteristics of TTC units in cooling are evaluated to an accuracy of ± 1% to a maximum of ± 3% [ATP (1970), Standard 1110 (2001)], no similar accurate test procedure exists for heating capacity. The only test method for performance characterisation of transport refrigeration units in heat mode is based on heat balance calculations on the evaporator for both refrigerant and air side, which has an accuracy of ±10% that obviously compares poorly with the experimental method based on electric heat input for cooling capacity measurement [ATP (1970)]. Based on this fact and on the increased demand of heat mode presented in Chapter 2, it is important to recognise the impact of this uncertainty on the TTC unit heating capacity certification. Taking a typical truck unit with a measured heating capacity of 3500W and based on the fact that this figure has an accuracy of ±10% or 350W, the actual heating capacity could range between 3150W and 3850W. This 700W has the potential to affect the compartment temperature by as much as 9.3°C difference from the set-point at an ambient of –30°C. This situation is assuming a desired +12°C box set-point temperature and that the compartment walls have the same thickness and heat transfer coefficient [Equation 3.1].

$$Q_{difference} = K \times S \times (T_{box} - T_{amb}) \qquad (3.1)$$

However, the new method proposed in section 3.3 provides the first ±3% accurate figures for the heating capacity of a TTC unit.

3.2 EXISTING TEST METHODS

Prior to presenting the details of the new higher accuracy heating capacity test method an overview of existing test procedures for both cooling and heating capacity measurements is first presented in this section.

3.2.1 Measuring Cooling Capacity

The current industrial test procedures used to evaluate the cooling capacity conforms to international standards and agreements for rating the performance of mechanical transport refrigeration units. Contracting parties worldwide have agreed on dedicated test facilities and test procedures required for the evaluation of both cooling and heating capacity of TTC units described in the following regulations:

- "*Agreement on the International Carriage of Perishable Foodstuffs and on the Special Equipment to be Used For Such Carriage*" - ATP, released by the United Nations Economic Commission For Europe, Transport Division in 1970 [ATP (1970)].
- "*Standard for Mechanical Transport Refrigeration Units*" [Standard 1110 (1977), (2001)] released in 1977 in the United States under the supervision of the Air-Conditioning and Refrigeration Institute (IIR).

The experimental test procedures used for the rating of cooling capacity are first reviewed in this section as they are very mature and provide a basis for the discussion of the heating capacity methods. The existing methods are based on either one of the following two methods:

- A heat balance calculation of the cooling capacity, using the air and refrigerant mass flow, enthalpy difference between the evaporator inlet and outlet on refrigerant or airside.
- A test procedure based on electric heat input form electric heaters recorded with a precise wattmeter that can be calibrated yielding an accuracy of ±1% to ±3% [Standard 1110 (1977), (2001)].

The heat balance calculation based test method is less accurate and as a result is not so popular. The later pure experimental method using electric heat input presented below is recommended by organizations such as UNECE and IIR, and is used by transport refrigeration system manufactures and test laboratories all over the world for cooling capacity certification due to its good accuracy of ±3% [ATP (1970)]. The cooling capacity based on electric heat input measurement can be performed using one of two different approaches.

A) Calibrated – Box Method

As indicated in Figure 3.1 and previously in Figure 1.7, the TTC unit's evaporator section is mounted inside a "calibrated box" (2) and the condenser section is exposed to air in the external ambient room (1). The temperature controlled space known as ambient room is capable of simulating different temperature conditions within –30°C and +50°C, using a cooling system with an externally located compressor (4), condenser (5) and internal evaporator (6). The higher ambient temperatures of up to +50°C are obtained using electric heaters (7) situated in front of the evaporator (6).

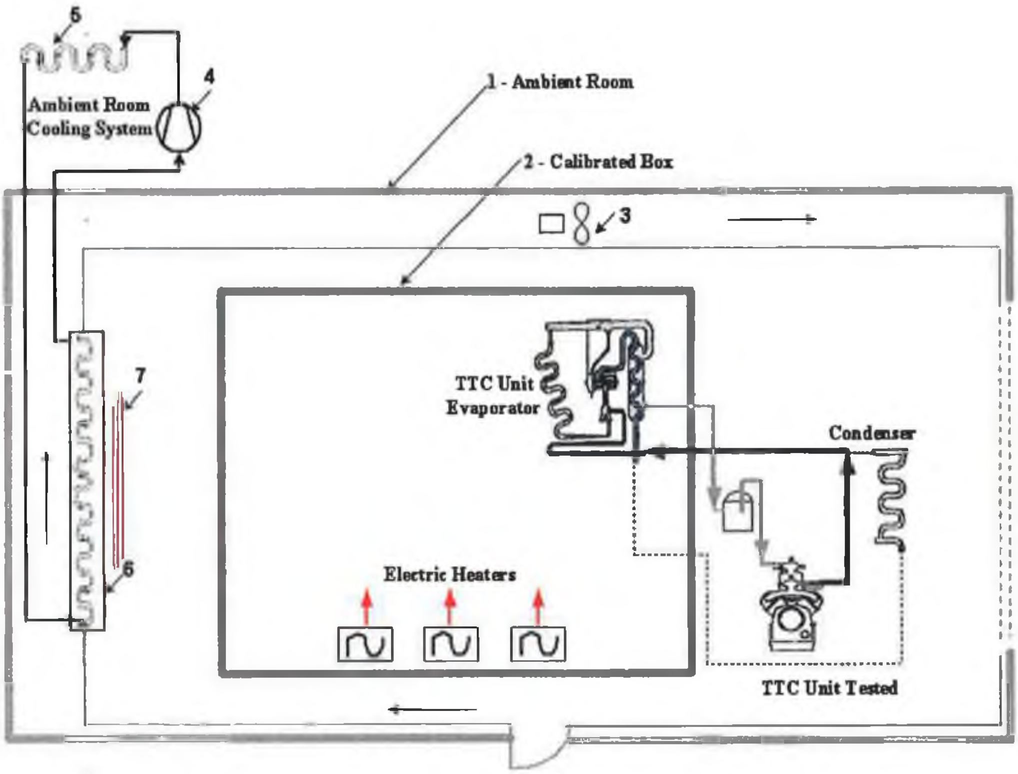


Figure 3.1 Overview of the test facility required to execute the cooling capacity tests using the "calibrated box" test method. View shown in Figure 1.7.

The box is referred to as a "calibrated box" since the heat transfer rates to or from this box are known for a wide range of temperature differences ($\Delta T_{i,o}$) between the inside and outside air temperature. Inside the "calibrated box", electric heaters equipped with fans generate uniformly distributed heat to balance the cooling capacity of the unit until the test set - point temperature is achieved. The cooling capacity of the unit is obtained by adding the heat losses through the box walls for a given temperature difference ($\Delta T_{i,o}$) to the measured electric heat input [Standard 1110 (2001); Radulescu *et al.*, (2003)]. The standard procedure used for heat loss measurements are defined in Appendix E.

B) Isothermal – Box Method

As indicated in Figure 3.2, the TTC unit under test is mounted on the front wall of a box, which is referred to as the Isothermal Box (2). Except for the front wall, the box is completely isolated by the Surround Room (3) that is controlled at the same temperature as the Isothermal Box by a dedicated externally located compressor (7), condenser (8) and inside located evaporator (9).

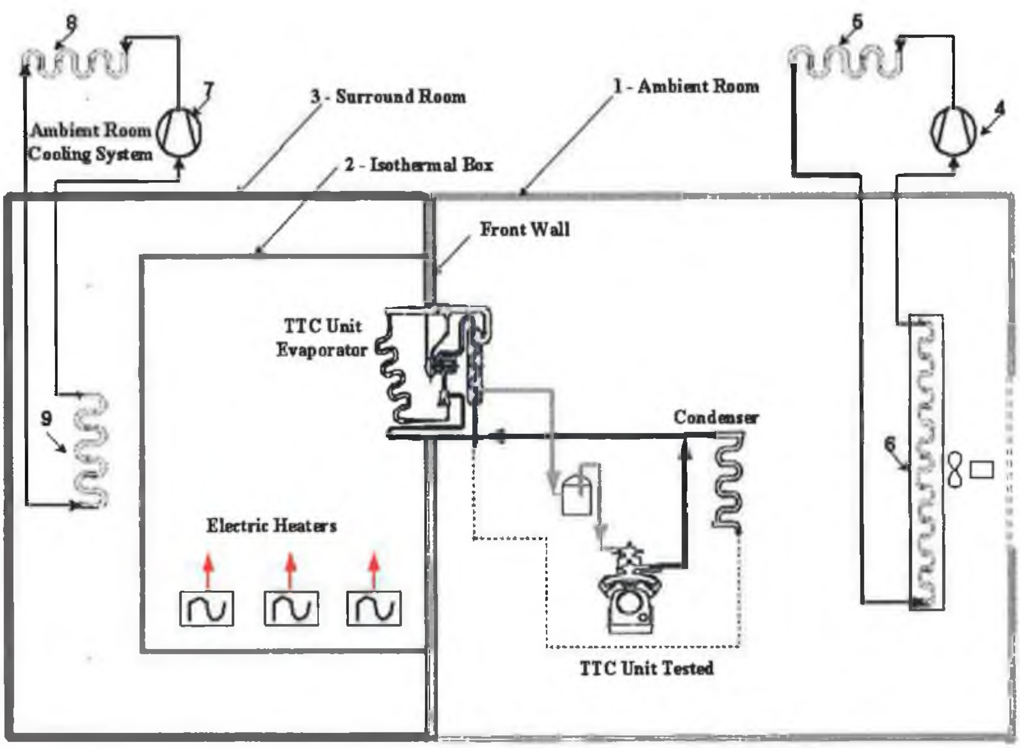


Figure 3.2 Overview of the test facility required to execute the cooling capacity tests using the "isothermal box" test method.

The front wall of the box is accurately calibrated to determine the heat losses for set temperature differences between the air temperature inside the Isothermal Box and the external air temperature in the controlled space, referred to as the Ambient Room (1). This latter chamber is capable of simulating different temperature conditions within the range –30°C and +50°C, using a cooling system with an externally compressor (4) and condenser (5) and an evaporator located inside (6). The unit's cooling capacity is experimentally determined by measuring the amount of energy consumed by the electric heater and any other electrical device inside the isothermal box combination plus the amount of heat dissipated through the front wall to/ from the ambient room space [Standard 1110 (2001); Radulescu *et al.*, (2003)].

The cooling and heating capacities of the TTC unit are recorded only when the steady-state conditions defined below are achieved [Standard 1110, (2001)]

- The average ambient air temperature recorded at a specific measurement point shall be within ±2.2°C of the average ambient air temperature calculated for the total number of measurements occurring for at least one hour.
- The temperature difference between any two ambient air temperature-measuring stations shall not exceed 1.1°C.
- The average box temperature recorded for specific locations [Appendix C] shall be within a range of 0.56°C of the average box for the total number of observations occurring for at least one hour.
- The temperature difference between any two measuring points within the box shall not exceed 1.4°C.
- The temperature difference between any two-temperature stations measuring the return air temperature of the TTC unit evaporator shall not exceed 1.7°C.
- The average inlet air condenser temperature differences shall be within 0.6°C.
- Steady-state heating capacity conditions are achieved when the electric heat input required to maintain a given set-point temperature differs by less than 100W over 20 minutes.

3.2.2 Measuring Heating Capacity

A) Existing Standard test procedure

While the performance characterisation of TTC units in cooling can be evaluated to an accuracy of ±3% [ATP (1970)], no similarly accurate test procedure exists for heating capacity. This reflects the attention that the industry has given to the development of heating capacity in the past. It is fair to say however that this need has only recently emerged due to increased demand for fresh food transportation [Chapter 2] and this thesis reflects one of the first attempts to address this problem.

The only test method for performance characterization of transport refrigeration units in heat mode is based on heat balance calculations for both refrigerant and air side, which has an accuracy of ± 10 % [ATP (1970)]. The heating capacity is evaluated using the average temperature difference across the evaporator coil based on measurements of the air temperature at eight and six locations on the evaporator inlet and outlet respectively. The airflow rate through the evaporator is measured using a wind tunnel with 5436-m^3/h maximum capacity at 2680 rpm and 21°C. Both the airflow and density of air are assumed to vary linearly with rpm respectively with the ratio of absolute temperatures. The heating capacity is evaluated using:

$$Q_{heat} = \rho_{air} \times Cp_{air} \times V_{air} \times \Delta T_{evap,air} \qquad (3.2)$$

B) Alternative Test Method

Recognizing the increased importance of heat mode in maintaining optimum temperature control, a new test procedure to determine heating capacity of TTC units measuring the airflow through different evaporators has recently been proposed [UNECE, Transport Division (2002)]. The working parties accepted that this test procedure is more practical for users but the errors associated with spatial variation in the air temperature and the airflow rated through the evaporator is still not avoided. As this is a potential limitation, the working parties agreed that the heating section should be separated from the cooling performance test section and suggested that a working group might be established to prepare a draft test procedure for heating performance evaluation, which would take account of practice in different countries. To the current authors knowledge, this working group have not yet reported their findings. This recent initiative from Germany further signals the importance of heat mode and the need to establish more accurate test methods to evaluate heating capacity for TTC units.

The following section presents the description of the new heating capacity (EHI) test method that is expected to be adopted as a standard ATP procedure.

3.3 NEW EHI HEATING CAPACITY TEST METHOD

The new test method proposed in this thesis eliminates all the variables that introduce uncertainty in the existing standard method and determines the heating capacity by measuring just one input. Measurement accuracy is improved to ±3%, even ±1% is possible [Radulescu *et al.*, (2003)] and the results have been shown to be repeatable [Figures 4.8 and 6.9]. The accuracy is improved by measuring with high precision just one parameter, the electric heat input, and as a result this novel test method will be referred to as the Electric Heat Input (EHI) method. This new EHI test procedure can be applied to determine heating capacity of any evaporator using hot gas as a working fluid and it has good potential to be adopted as future ATP standard procedure. This section documents the following aspects of the new EHI test method:

- Overview of the heating test facility.
- Description of the novel EHI test procedure.
- Description of the heating test facility capable of performing heating tests based on the EHI method for both single and multi – compartment units.
- Technical details for the key mechanical systems used.
- Description of the control algorithm and capabilities.

3.3.1 Overview of EHI Method Test Facility

While a more detailed description of the test facility, data acquisition and control features is presented in section 3.3.4 and 3.3.5, an overview of the test facility required to execute the EHI method is presented in this section. Figure 3.3 builds on the structure defined in Figure 3.1 and identifies the additional mechanical systems required to conduct the Electric Heat Input (EHI) heating capacity test method. This facility was constructed at Thermo King's Research and Development Center, Galway and makes use of an existing temperature controlled chamber, identified as the ambient room in Figure 3.1. It was possible to generate air temperatures within the range –30°C to +50°C within this room using a dedicated temperature control system that consisted of four externally located high power compressors (4) and condenser (5) and inside located cooling evaporator (6). The calibrated box (2), which simulates the space within the trailer during food transportation, is situated within a temperature controlled ambient room (1) as indicated in Figures 3.1 and 3.3.

This temperature-controlled ambient room has 150 mm thick insulated sidewalls, ceiling and floor and the air circulation is obtained using three axial fans positioned above the test chambers false ceiling. Temperature control within this space is provided by the main cooling system that consists of four compressors rated at 120 kW total power and a 4 kW single condenser situated outside the test facility. Four evaporators with a 64 kW total capacity are situated within the temperature-controlled chamber [Appendix F]. This system is capable of generating environmental air temperature within the range –30°C to 50°C in the temperature controlled chamber. The test temperature within the ambient room can be controlled based on either the condenser air inlet temperature to the unit under test or the average temperature.

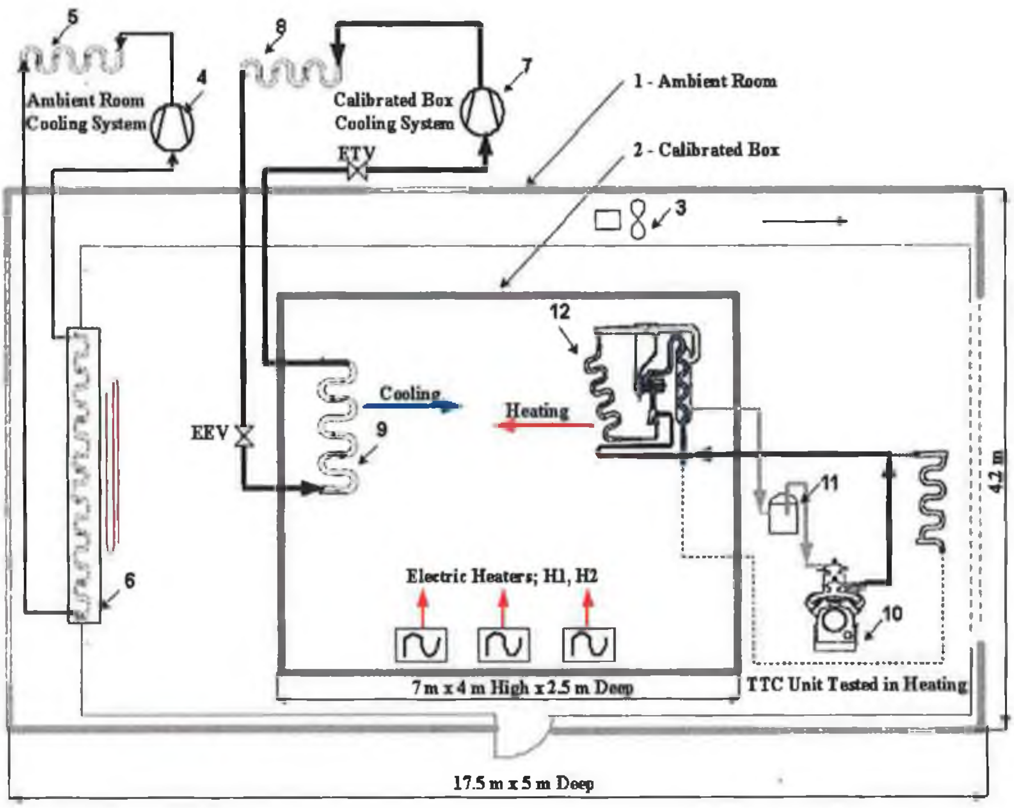


Figure 3.3 Overview of the test facility required to execute the new EHI test method.

The TTC unit under test is installed on the calibrated box (2) with its evaporator exposed internally and condenser section exposed to the air within the ambient room (1). To facilitate the new EHI test method a new cooling system consisting of components 7 through 9 in Figure 3.3 had to be designed, installed and commissioned. This cooling system has a pre-measured cooling capacity and is used to balance the heat input from the TTC unit under test and the heat input from the electric heaters located inside the calibrated box. An electronic expansion valve (EEV) was installed at the evaporator inlet for automatic control of the superheat on the evaporator in order to maintain steady state conditions for the cooling performance of the system.

Based on the notation defined in Figure 3.3, further technical description of the system components is presented in Table 3.1. The new components that had been designed, installed and commissioned as part of the test facility upgrade required to enable the EHI method to be run are identified.

Table 3. 1. Technical specifications of the main mechanical equipments installed and commissioned in the new heating test facility and in the existing temperature controlled chamber – ambient room.

No.	Equipment	Technical Details	Purpose
1	Ambient Room	Length = 17.5m, Height = 4m, Width = 4.5m	Temperature controlled chamber with a range within –30°C to 50°C.
2	Calibrated Box	Length = 7m, Height = 2.5m, Width = 2.2m	Simulates the food compartment. The TTC unit tested is installed on this box.
3	3 Axial fans		Provide air circulation about the Ambient Room
4	4 Reciprocating Compressors	120 kW total cooling power	Used to provide high pressure, high temperature gas for the existing main cooling system.
5	Ambient Room Condenser	4 kW cooling capacity	To dissipate heat absorbed within the Ambient Room.
6	Ambient Room Evaporator	64 kW cooling capacity	Absorbs heat from the Ambient Room.
7*	Compressor Heating test facility	Maximum 30kW cooling capacity, Copeland model 2Z-U2-33-TWD	To provide high pressure, high temperature for the new cooling system.
8*	Condenser	28 kW capacity	Dedicated condenser for the Heating Test Facility. Vapor refrigerant transforms into liquid
9*	Remote Evaporator	28 kW Cooling Capacity	Dedicated remote evaporator for the Heating test facility. Absorbs heat from the calibrated box.
10	TTC unit Compressor	Depends on the TTC unit tested	To provide high pressure, high temperature gas for the TTC unit
11	Accumulator Tank	Depends on the TTC unit tested	The liquid is evaporated in the accumulator
12	TTC unit Evaporator	Depends on the Unit tested	The refrigerant gas is changed into liquid while large amount of latent heat is released in the calibrated box.
13*	Electric Heaters	5x3600W	Used to balance the cooling capacity of the new condensing unit.

Note: * identifies the new components required to enable the EHI method to be run.

The EHI method and the test facility test facility can not only be used to perform tests on single–compartment TTC units using one calibrated box, but also for multi–temperature TTC units consisting of two or more separated calibrated boxes [Figure 3.5]. However, only single–compartment test results are presented in this thesis. The test procedure for the new EHI test method is presented in section 3.3.2 while detailed technical characteristics of the components and of the data acquisition and control program are presented in paragraphs 3.3.4 and 3.3.5 respectively.

3.3.2 Novel EHI Test Procedure for the measurement of Heating Capacity

This new test procedure provides an accurate method of measuring the heating capacity of a TTC unit, reducing the measurement uncertainty by 70% as oppose with high errors of ±10% introduced by the existing ATP method. As a result, a better characterisation of the unit's performance in heat mode can be compared with the existing method for cooling capacity measurement described in section 3.2.2 [Standard 1110 (1977); ATP (1970); ASHRAE (1978); ASHRAE 41.1 (1986)].

Based on present standards [ATP (1970); Standard 1110 (2001)], and weather data [WMO (2002); World Data Center (2002)], heating capacity tests are performed at the temperature conditions presented in Table 3.2.

Table 3. 2. Test Temperatures at which heating capacity tests are currently conducted.

Test No.	Temperature in Ambient Room	Temperature within Calibrated Box
1	- 10°C (14°F)	0°C (32°F); +21 °C (69.8°F); +12 °C (53.6°F); +2°C (34.7°F)
2	-30°C (- 15°F)	0°C (32°F); +21°C (69.8°F); +12°C (53.6°F); +2°C (34.7°F)
3	- 20°C (- 4°F)	0°C (32°F); +21°C (69.8°F); +12°C (53.6°F); +2°C (34.7°F)
4	-30°C (-6.6°F)	0°C (32°F); +21°C (69.8°F); +12°C (53.6°F); +2°C (34.7°F)

Using the component reference numbers introduced in Figure 3.3, the test procedure for the new EHI test method is outlined with the support of Figure 3.4, where the TTC unit operates in a controlled temperature ambient of –30°C and the heating capacity for a +12°C compartment/ box temperature is required. The data presented is based on measured ambient and box temperatures and time periods, during a heating capacity test. Note that CS refers to the Calibrated Box New Cooling System. It can be noticed that this test took 9:30 hours. Once the cooling capacity of the CS system is measured for the test temperature conditions, one heating capacity test can be performed only in four hours.

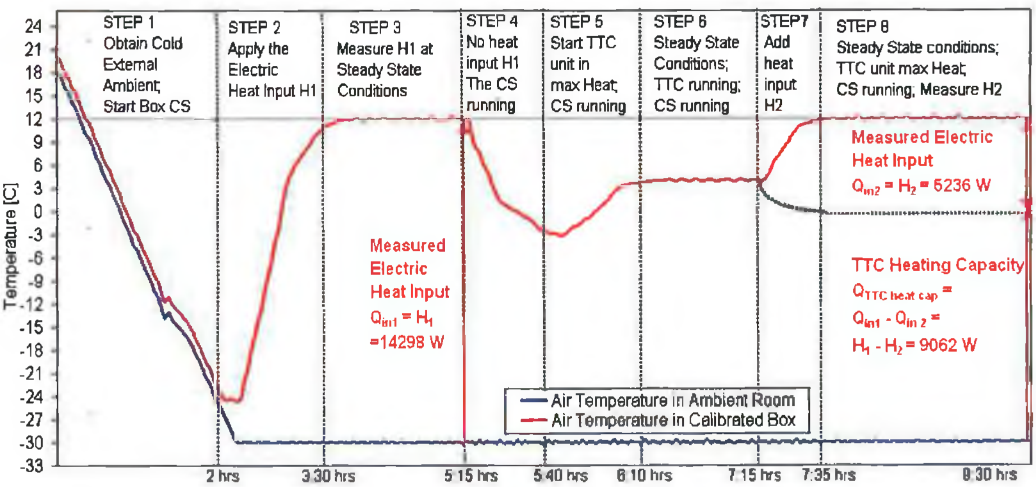


Figure 3.4 Main steps of the EHI test procedure for a heating capacity test at –30°C ambient temperature and +12°C box temperature.

The new EHI heating capacity test procedure is applied as follows:

- Step I: The Ambient Room Cooling System (4, 5, 6) is used to generate the –30°C temperature in the ambient room.
- Step II: With a steady state –30°C established in the ambient room, the cooling capacity of the new Calibrated Box Cooling System (7, 8, 9) is determined in conformity with existing standards [Standard (1110), (2001); ATP (1970)] by measuring the electric heat input H1 [W] required to achieve a temperature that is lower with a few degrees than the set-point within the calibrated box (2). The first electric heat input H1 is recorded after 5:15 hours as shown in Figure 3.4.
- Step III: Once the steady-state thermal conditions are reached, usually within 3 hours, the electric heat input ($Q_{in1} = H_1$) is removed while the new cooling system (7, 8, 9) is maintained in the same conditions, at the same cooling capacity by controlling the evaporator's superheat and the position of the compressor suction electronic throttling valve (ETV) (Figures 3.14 and 3.15).
- Step IV: With a steady state –30°C in the ambient room (1) and steady state conditions at a lower temperature than the set-point in the calibrated box (2), the TTC unit under test (10, 11, 12) is started in maximum heating capacity. The cooling capacity of the cooling system (7, 8, 9) is used to balance the heat input from the unit under test and must be greater than the estimated heating capacity of the TTC unit, in order to obtain an accurate measurement of the

electric heat input. In these conditions, the stabilised final temperature obtained in the calibrated box is lower that the test set point and a second accurate measured heat input can be added until the test set-point temperature is achieved as explained in Steps V and VI.

- Step V: Run both the new cooling system (7, 8, 9) and the TTC (10, 11, 12) unit until steady state conditions are obtained in the calibrated box. Note that a desired lower temperature than the set-point within the calibrated box will most likely have been achieved since the cooling capacity of the calibrated box cooling system (7,8,9) exceeds the heating capacity of the TTC unit.
- Step VI: A controlled auxiliary electric heat input H_2 is added in order to increase the air temperature within the Calibrated Box until the desired set-point temperature of +12°C is obtained.
- Step VII: After the desired steady-state temperature conditions are reached in both ambient room [-30°C] and the calibrated box [+12°C], usually within 2 to 3 hours, the second electric heat input H_2 must be measured.
- Step VIII: The heating capacity of the TTC unit is obtained from the difference between the two measured electric heat inputs, according to equation 3.3 below:

$$\textit{Heating Capacity [W]} = Q_{TTC\ heat\ cap} = Q_{in1} - Q_{in2} = H_1 - H_2 \qquad (3.\ 3)$$

Note that H2 increases during Steps 7 and 8 due to a decrease in TTC heating capacity as the trailer box temperature increases. Steady state conditions required to be established at the end of steps 3, 6 and 8 were defined in section 3.2.1(B). This new EHI heating capacity test method should only be applied following the recommended practice of other existing standard test methods [ATP (1970); Standard 1110 (2001)]:

- Return air temperatures should be measured at a minimum of 8 locations.
- Air–temperature within the calibrated box should be measured at a minimum of 5 locations on the walls of the calibrated box.
- The wattmeter used to measure the electric heat input to the heaters should display a measurement accuracy of at least ± 1%.
- After a minimum of two hours at the same conditions, the air temperatures inside both the calibrated box and the ambient room, together with all TTC

unit parameters, should be recorded for at least one hour in order to indicate that steady state conditions have been maintained during this period.

- To reduce test time, the cooling system (7, 8, 9) used to balance the heating capacity of the TTC 1 unit under test should be capable of being controlled in order to maintain the determined cooling capacity during all tests performed at those specific test temperatures.
- Steady-state heating capacity conditions are achieved when the electric heat input required to maintain a given set-point temperature differs by less than 100W over 20 minutes.

Overall, as indicated in Figure 3.4 it takes between 7 to 11 hours to conduct a heating capacity test for a given set of conditions. It has to be noted that once the cooling capacity is determined, as a result of the control of the calibrated box cooling system, it is possible to maintain the same capacity during all tests and a shorter time can be obtained for all tests. The most important challenge for an accurate heating capacity measurement is the precise control of the calibrated box cooling system (CS), for a constant capacity. The procedure presented above was applied based on a control algorithm that is capable of maintaining the CS unit (7, 8, 9) at the same cooling capacity during the whole duration of the test. As presented more detailed in Figures 3.14, this was achieved by controlling: i) the EEV position for a constant superheat on the evaporator while ii) the ETV position was maintained at the same value for constant compressor suction pressure. The fine-tuning of the test temperature in the calibrated box was obtained through a proportional, integrative and derivative control type of the electric heaters inside the calibrated box (Figure 3.15).

However, if the control of the calibrated box cooling system (7, 8, 9) is based on variations of the capacity instead of maintaining a constant value during the whole test, the EHI procedure can be applied in a different manner. It is estimated that this second version of the test procedure presents the advantage that it takes a shorter time to perform, but very reliable control of the cooling system capacity is required. The steps involved are:

- Instead of running the second and third steps of the previous procedure that requires one start the calibrated box cooling system first and to determine the cooling capacity, in this new version of the test procedure both the TTC (10, 11, 12) unit being tested in heat mode and the new calibrated box cooling system CS (7, 8, 9) can be started in the same time;

- While the TTC unit is running in maximum heating capacity, the calibrated box cooling system (7, 8, 9) is than controlled to balance the heat input until the set - point temperature is achieved (2);
- After steady - state conditions are achieved, the TTC (10, 11, 12) unit can be stopped while the calibrated box cooling system (7, 8, 9) is controlled to maintain the same cooling capacity as specified in the first part of the test;
- To obtain the test set-point temperature, an electric heat input has to be added;

3.3.3 EHI method versus ATP method accuracy

The accuracy of the new EHI test method is within ±1% up to a maximum range of ±3% due to the fact that the electric heat input is measured with high precision using a power wattmeter that has ±0.15% accuracy. Chapters 4 and 6 present the heating capacities obtained with both test methods. While the ATP procedure shows enough heating capacity for all test temperature conditions the figures obtained based on the new EHI method agrees with the field data that shows insufficient capacity at extreme low ambient temperatures, like –30°C. It is well known [ATP (1970)] that the existing standard test method has measurement errors of up to ±10% as it reflects uncertainty in several input measurements together with the theoretical calculation:

- The ATP test method is based on heat balance calculations that take into account the air temperature difference through the evaporator. As a result, high errors are obtained due to the placement of the thermocouples, which is extremely important as the variation in temperature across the evaporator rarely exceeds 5°C. Besides no two thermocouple measurement accuracy which can be very significant if the temperature difference on the evaporator drops to 1°C or 2°C. No two people will place the measurement thermocouples in the exact same position every time a test is performed, which can result in different average temperatures, even when tests are performed under identical conditions. Mixing of return and discharge air also affects measurement accuracy. This may occur as a result of a Venturi affect created near the evaporator outlet that leads to inaccuracies in temperature measurements.

- The airflow through the evaporator is measured based on the wind tunnel testing method. The measurement accuracy can be up to ±3%, which is another important factor that contributes to poor accuracy of the existing standard method [ASHRAE Standard-16 (1988), ASHRAE Standard-25 (1990)].

3.3.4 Heating Test Facility – Multi Compartment Units

As indicated in Figure 3.5, the heating capacity of the TTC unit's remote evaporators in each calibrated box can be balanced by separate cooling evaporators connected to the same condensing unit–compressor and externally located condenser. Figure 3.5 shows a typical setting for a two-compartment TTC, with the host evaporator (12) installed in the "calibrated box 1" and the remote evaporator (14) installed within the "calibrated box 2". When the unit is running in heating for the both compartments/ boxes, another two cooling evaporators are required to balance the heat from the unit (9, 13). The new cooling evaporators are connected on a common suction and discharge line from the externally calibrated box-cooling unit, but are controlled separately through the electronic expansion valves (EEV).

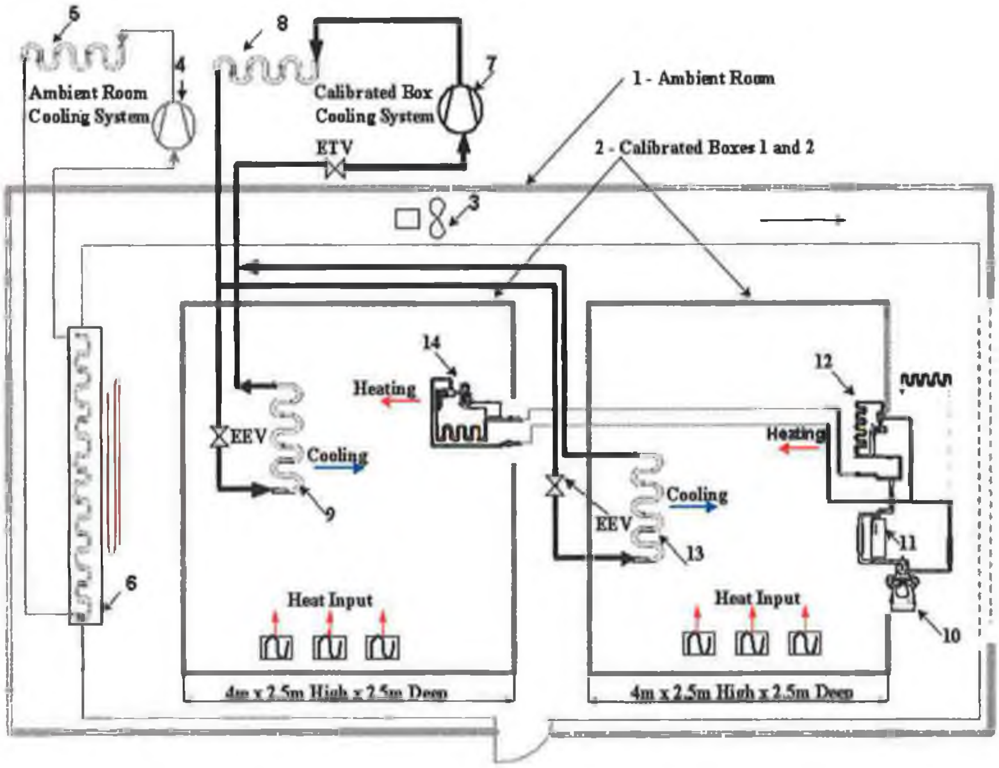


Figure 3.5 Test configuration required to evaluate the heating capacity of a multi – compartment TTC unit [Radulescu *et al.*, (2003)].

3.4 TECHNICAL SPECIFICATION OF MECHANICAL COMPONENTS

The calibrated box cooling system (7, 8, 9) shown in Figures 3.3 and 3.5 is used to balance the TTC unit in heating and consists of:

- Externally located scroll compressor and condenser.
- Remote evaporator installed inside the calibrated box.

The technical specifications of these new components are presented in sections 3.4.1 and 3.4.2 together with a detailed overview in Appendix G.

3.4.1 Scroll Compressor

The externally located Copeland scroll compressor, model ZF–U2–33–TWD is shown in Figure 3.6 with the technical specifications presented in Tables 3.3 and 3.4. Note that the cooling capacity of the condensing unit differs considerably depending on both the evaporating and ambient temperatures. This unit is suited for medium and low temperature applications from –10°C down to –40°C, and features outstanding "pull-down" capabilities starting at 0°C. The unit is equipped with two fans capable of providing 1.79m^3/s airflow each. The condenser has a maximum 20 kW cooling capacity [Appendix G]. The power consumption is also a function of ambient temperature, Table 3.4.

Figure 3. 6 Externally Outside located condensing unit for calibrated box cooling system.

Table 3.3. Refrigeration Capacity of the Copeland model ZF–U2–33–TWD as a function of both ambient temperature and evaporating temperature.

	Evaporating Temperature [°C]					
	-20	-10	-5	0	+5	+7
Ambient [°C]		**Refrigeration**	**Capacity**	**[kW]**		
27	13.79	18.73	21.55	24.83	27.95	29.35
32	12.83	17.30	19.85	22.64	25.87	28.94

Table 3.4. Total Power Consumption of the Copeland model ZF–U2–33–TWD as a function of both ambient temperature and evaporating temperature.

Total Power Consumption kW						
	-20	-10	-5	0	5	7
Ambient [°C]		**Total Power**	**Consumption**	**[kW]**		
27	7.88	9.12	9.84	10.62	11.47	11.83
32	8.50	9.76	10.48	11.25	12.09	12.45

3.4.2 Remote evaporator

The remote evaporator (9) is located inside the calibrated box and shown in Figure 3. 7. It is a finned – tube, cross flow heat exchanger with a maximum cooling capacity of 20 kW designed especially for the requirements of the new test facility. Physical characteristics of this unit are presented in Table 3.5. The design and construction of the new evaporator is based on a known requirement for a 40 kW cooling capacity, required to obtain a minimum 0°C set-point temperature during testing. The evaporator type selected is a refrigerant-air cross-flow coil, with a finned-tube compact heat exchanger construction. The final dimensions were determined based on the overall energy balance and overall heat transfer coefficient presented in the literature for this type of evaporator [Incropera (1990); Carabogdan *et al.*, (1983) and Stefanescu *et al.*, (1983)]. Convection heat transfer coefficient on the airside is calculated using Stanton and Dittus Boetter criteria specified in the literature [Carabogdan e*t al.*, (1983)], including calculations for Reynolds and Prandtl numbers as presented in Appendix G. The constructive details of the cooling remote evaporator are presented in Table 3.5. The axial fans were selected based on the required airflow and on the pressure drop through the evaporator [Incropera, (1990)].

Figure 3.7 Remote Evaporator located inside the calibrated box.

Table 3. 5. Physical Characteristics of the cooling remote evaporator.

Number of rows	8
Number of Circuits	11
Number of Tubes	176
External Diameter of Tubes	0.00952 m
Internal Diameter of Tubes	0.006 m
Primary surface (refrigerant)	7.2 m^2
Secondary Surface (air)	66 m^2
Numbers of fins	189

3.5 DATA ACQUISITION AND CONTROL

A dedicated data acquisition and control system was developed to monitor all parameters for both the test environment and the TTC unit tested and to control the main mechanical equipments of the test facility. An overview of this system, which consists of three key elements, is presented in Figure 3.8 and Appendix G:

i) The measurement sensors embedded within the test facility in the ambient room and calibrated box together with the TTC unit under test.
ii) Data acquisition modules that facilitate the interface between the sensors and the control program.
iii) The control program that monitors and controls both test facility and TTC unit.

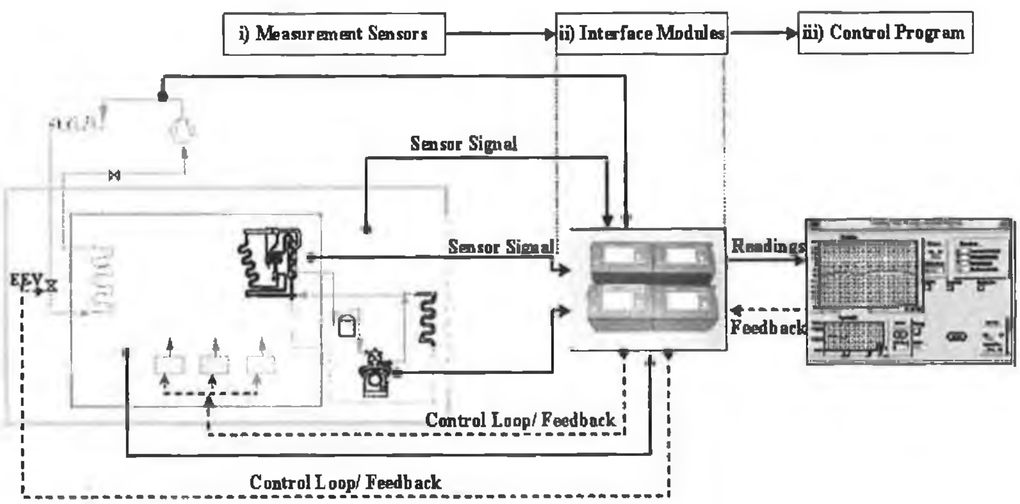


Figure 3.8 Data acquisition and control flow-chart.

The components presented in the flow-chart [Figure 3.8] will be described in the following sections: (3.5.1) Measurement Sensors, (3.5.2) Data Acquisition System and (3.5.3) Control Program.

3.5.1 Measurement Sensors

The location of the transducers used to record both temperatures and pressures of the test facility together with the TTC unit tested in heat mode is presented in Figure 3.9 while a detailed description of the measuring points is presented in Table 3.6. Through the data acquisition system and the control program, the temperatures and pressures are measured at 32 and 8 locations respectively. Each thermocouple was connected to one of four FP–TC–120 modules equipped with cold junction sensor in an isothermal base for accurate cold junction compensation. All thermocouples were calibrated to an accuracy of ±0.5°C using a D55 SE temperature calibrator (from Jofra, Denmark) at 0°C and +10°C (Appendix H).

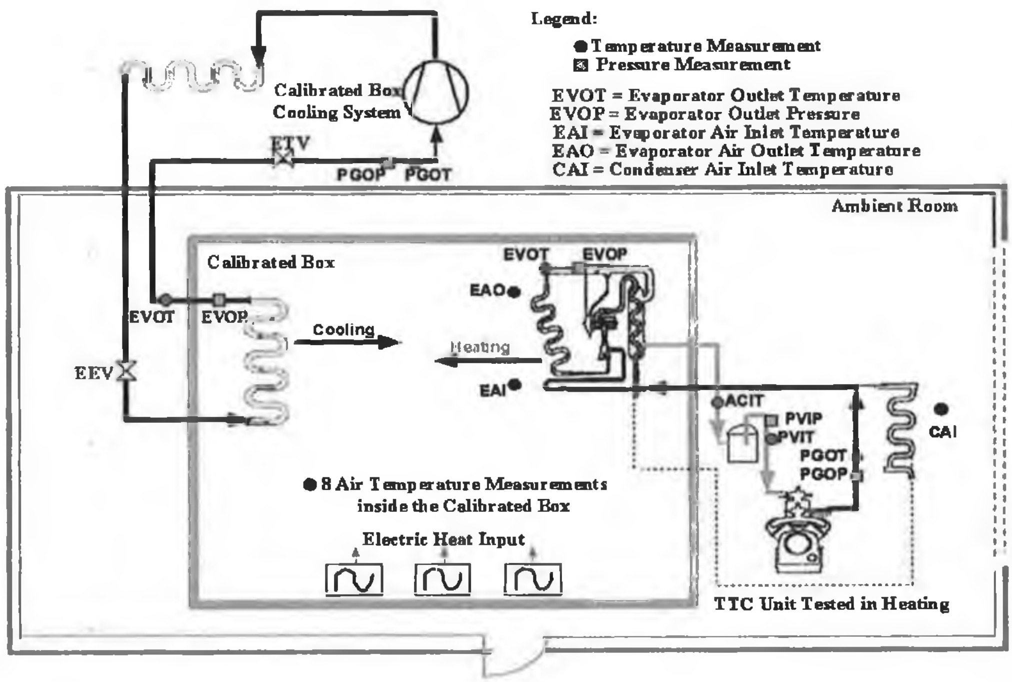


Figure 3.9 Measurement for both temperature and pressure locations about the heating test facility and TTC unit under test.

A total of 8 pressure transducers were used to measure the inlet and outlet pressures for both the test facility and TTC unit compressors and evaporators, together with optional pressure measurements for inlet-outlet of the TTC unit accumulator and condenser as presented in Figure 3.9 and Table 3.6. Two types of pressure transducers were used, a G07 type with a range of 0 – 500 psig and G09 type with a range of 0 – 200 psig, both with an accuracy of ±0.25%. The calibration of the pressure transducers was performed using a Superb pressure calibrator, type PCC3-

H-200-2. One analogue input module, an FP–AI–100 with 12 bits resolution was used to interface these sensors with the control program. Descriptions of these interface modules and the control program is presented in the following sections while the technical details are overviewed in Appendix H. The location and number of the main temperature and pressure measurements together with the control outputs used as feedback process in the control loops are presented in Table 3.6.

Table 3.6. Location and number of temperature and pressure measurements.

No.	Measurement Type	Name	Number
1	Temperature	Ambient Temperature	4
		Box Temperature	8
		CAI = Condenser Air Inlet	6
		EAI – Evaporator Air Inlet	4
		EAO = Evaporator Air Outlet	4
		PGOT – Compressor Outlet	1
		PVIT = Compressor Inlet	1
		EVOT = Evaporator Outlet	2
		EVIT = Evaporator Inlet	2
		ACIT = Accumulator Inlet Temp	1
		Compressor – CS inlet temperature	1
2	Pressure	Compressor discharge pressure	2
		Compressor Suction Pressure	2
		EVOP = Evaporator Outlet Pressure	2
		EVIP = Evaporator Inlet Pressure	1
		ACIP = Accumulator Inlet Pressure	1
3	Heat Input	Electric Heat Input	1
4	Analogue Output Feedback Loop Control	Electronic Expansion Valve	1
		Electronic Throttling Valve	1
		Electric Heat Input Control	1

3.5.2 Data Acquisition System

This system includes a variety of National Instruments analogue input modules defined in Table 3.7 and described in Appendix H.

Table 3.7. Interface modules required to facilitate the Data Acquisition and Control System.

Module	Type	Characteristics	Measurement/ Instrumentation
FP – TC - 120	Temperature Input Module	4 modules 8 channels/ each	32 temperature measurements Type T thermocouple
FP – AI - 100	Analogue Input	1 module 8 channels	8 pressure measurements ± 6 V – voltage input
FP – AO - 200	Analogue Output module	1 module 8 channels	Control electronic step valves and electric heat input Current loop 4 to 20 mA
FP – CTR - 500	Counter Module	8 channels	Counts digital signal of 5 VDC from a wattmeter
FP – RLY - 420	Relay Module	8 channels	Control motors and power circuits

The data acquisition modules allowed both temperature and pressure measurements, together with an analogue output control module for the calibrated box cooling system and electric heaters. The National Instruments modules and the control panels of the test facility are presented in Figure 3.10.

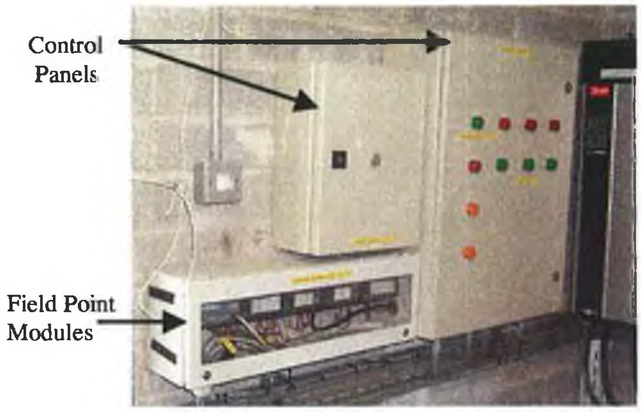


Figure 3.10 Data Acquisition and Control System set-up.

3.5.3 Control Program

Lab View software was selected as the interface for monitoring and controlling the test facility and the unit being tested in heating. The algorithm that defines the control program is presented in this section and in Appendix H.

A) User Interface

This Lab View program developed is not only capable of recording and saving all operating parameters for the TTC unit being tested, the calibrated box, and cooling system, but also to automatically control the cooling capacity and the electric heat input in order to obtain and maintain the test set – point temperature in the calibrated box. The front panel of the user interface program is shown in Figure 3.11 and consists of three sub-windows entitled: i) "Main" Interface Window, ii) "Unit Parameters" Window and iii) Cooling System Control Window.

The main interface to the program consists of two windows presented in Figure 3.11 and Figure 3.12. The most important parameters like evaporator air inlet and outlet temperature, condenser inlet and outlet temperature and calibrated box average temperatures are presented in the "Main" window in both numerical and chart form. Different options exist to save data and set up details of the DAQ sequence.

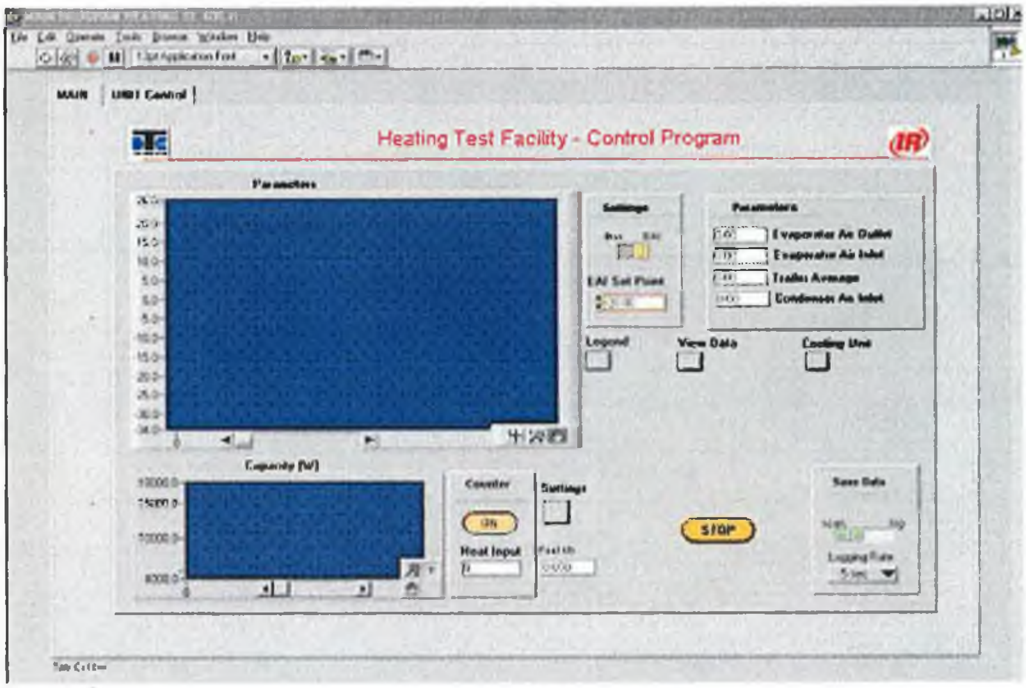


Figure 3. 11 Lab View Control Program – "Main" Interface Window.

The second window of the program presented in Figure 3.12 offers access to all parameters of the TTC unit tested in Heat Mode together with options for controlling the solenoid valves of the unit to enable different operating modes to be activated.

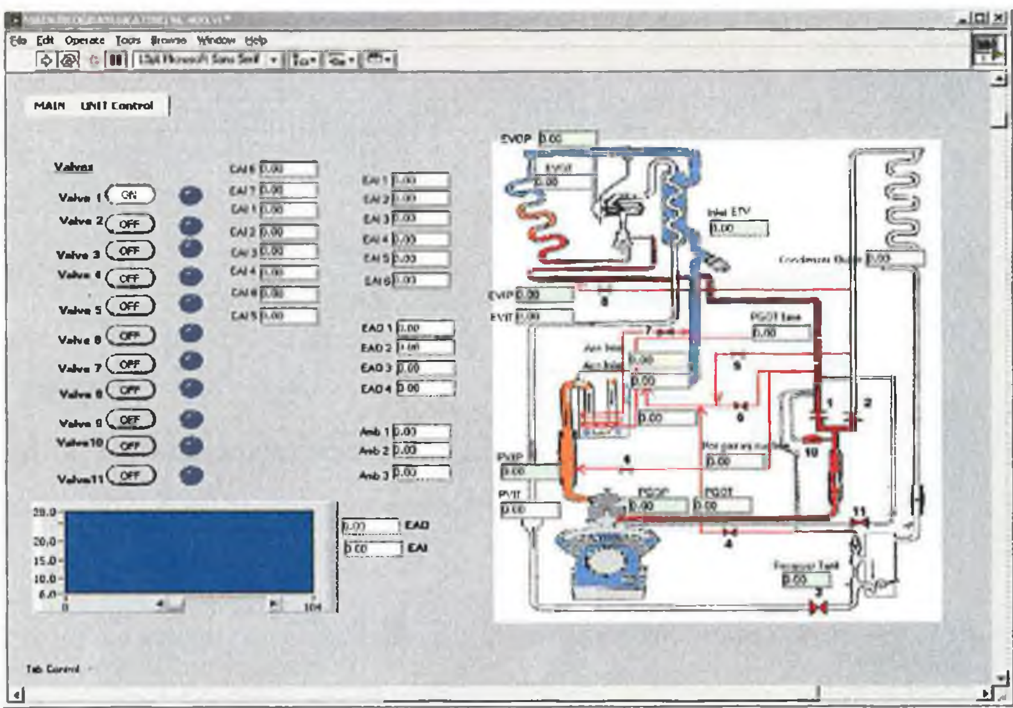


Figure 3.12 "Unit Parameters" window showing TTC unit parameters.

Buttons situated on the "Main" window offer the possibility to view detailed data for the heating test facility and the unit tested. An example of a control panel is presented in Figure 3.13 for the parameters of the calibrated box outside located condensing unit and remote cooling evaporator.

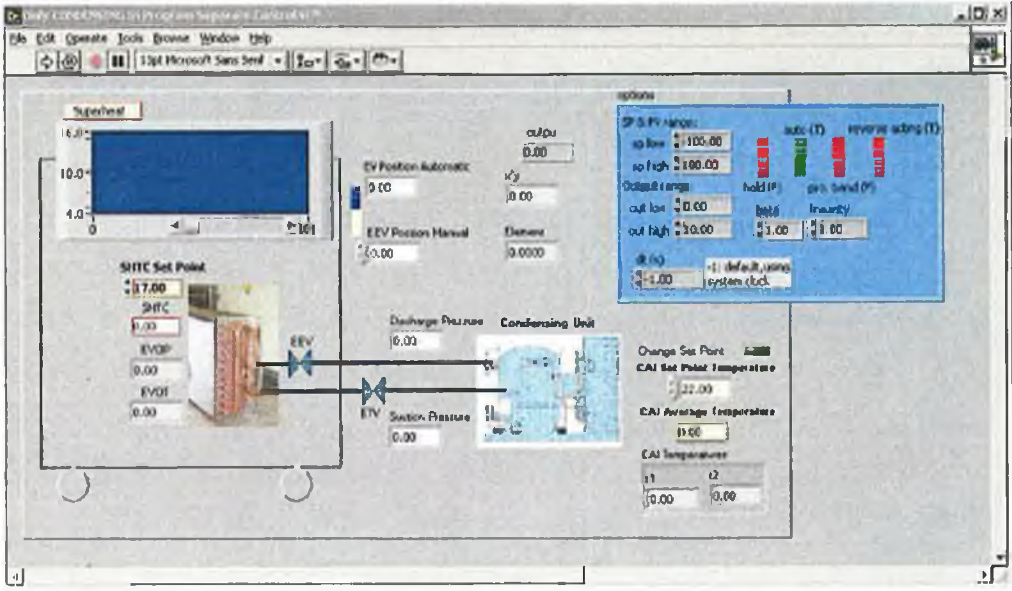


Figure 3.13 Window showing specific data for the condensing unit and the cooling evaporator.

B) Control Program Algorithm

The set-point temperature in the calibrated box is automatically obtained by the new data acquisition program, which is capable of controlling the cooling capacity of the calibrated box-condensing unit and the electric heat input. A schematic of the control algorithm for the calibrated box cooling evaporator is presented in Figure 3.14.

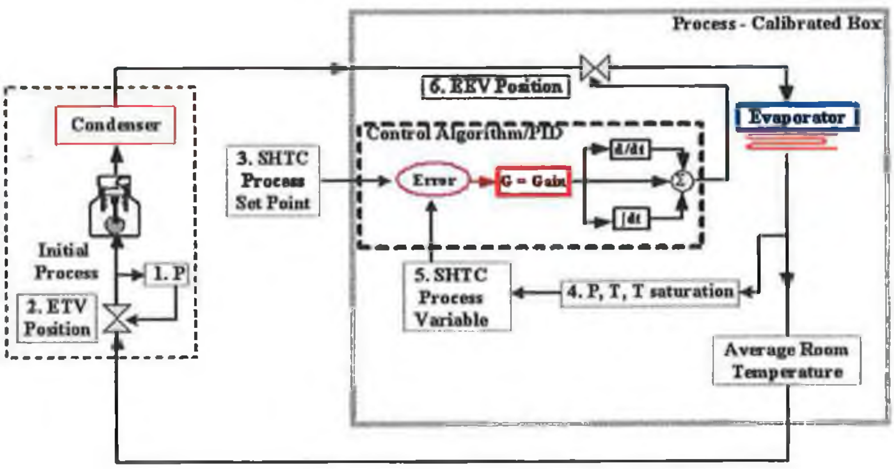


Figure 3.14 Algorithm for controlling the calibrated box-condensing unit cooling capacity.

The cooling capacity control is obtained through two algorithms: the compressor suction pressure through proportional control of the ETV position and the control of evaporator superheat through a Proportional Integral Derivative (PID) closed loop control type. In the initial process the compressor suction pressure (1) was maintained at 50psi set point using the control of the ETV position (2) through step variations. As a result the temperatures and pressures for both condenser and compressor were maintained almost constant. The second control loop based on the evaporator superheat control prevents cooling capacity variations in the calibrated box. The user is able to choose the desired set-point for the superheat (Figure 3.14) and the program automatically reads the evaporator outlet pressure and temperature (4). Based on a calculation algorithm that is using the saturation temperature for that specific pressure the process variable, which is the real superheat (5) is obtained. This latest value is compared with the set-point (3) and the PID control that has fine tuned coefficients is sending the output signal to the EEV (6) for further adjustments of the valve's position in order to eliminate the differences. This closed control loop repeats until a very precise control around ±0.3% of the superheat set-point value is obtained. Based on this control algorithm the capacity of the calibrated box cooling system is maintained at the same value during the test duration. The final set point temperature in the box is obtained through a closed control loop of the electric heaters as presented in Figure 3.15.

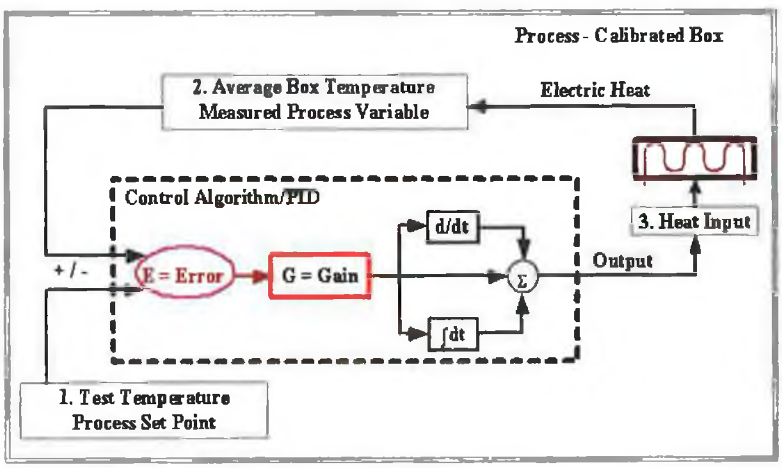


Figure 3.15 Algorithm for controlling the temperature in the Calibrated Box.

Temperature set-point (1) in the calibrated box is obtained using a Proportional Integral Derivative (PID) algorithm, which is capable of comparing the desired temperature value with the existing average box temperature that is the process variable (2) and as a result to control the electric heat input (3) to the calibrated box for all test conditions. The tuning of the PID coefficients was based on the Zeigler Nichols closed loop method to tune a controller [Ramsay (1986); Phillips *et al.,* (1991)]. To use this method the loop was tested with the controller in automatic. The steps followed were to determine the gain at which the loop oscillated with just proportional control, and to determine the integral and derivative coefficients at which the oscillations are suppressed. The automatic control obtained with this algorithm is capable of maintaining the set–point temperatures in each zone to a stability of ±0.5°C in conformity with present standard requirements [ATP (1970)].

3.6 PULL-UP AND TEMPERATURE CONTROL TESTS

Figures 1.6 and 2.10 show not only insufficient heating capacity at ambient temperatures sub –20°C but also long pull-up time and poor temperature control. As a result in-house tests are performed in the heating test facility (Figure 3.3) while the results are presented in Chapters 4, 5 and 6. The test method is applied as follows:

- The ambient room cooling system (4,5,6) is used to generate the test temperatures in the ambient room that are maintained for at least 24 hours until the same temperature is obtained in the calibrated box.
- The TTC unit tested is started in heat mode, at the required test set-point temperature obtained based on the unit's micro controller setting.
- The pull-up time is recorded when the TTC unit is started until the test set-point temperature is reached. The temperature control of the unit is recorded for at least seven hours period of time to simulate the length of a field journey.

3.7 SUMMARY

The need for a more accurate standard test procedure for measuring the heating capacity of TTC units has increased over the last decade. This chapter reviews the existing standard test procedures and much more presents an accurate alternative method. Existing heating test methods use heat balance calculations based on measurement of airside temperature differences and airflow across the evaporator or refrigerant mass flow through the evaporator and can guarantee ±10% accuracy [ATP (1970); Standard 1110 (1977), (2001)]. However, the new test procedure proposed, determines the heating capacity to an accuracy of ±3%, by limiting the measurement input errors when only the electrical power input to electric heaters and fans inside an insulated box are required. Using this Electric Heat Input (EHI) method it is possible to establish the system accuracy of ±1% to maximum ±3%, which is comparable to the accepted accuracy of cooling capacity tests. Besides the improved accuracy, the main advantages of the proposed EHI test procedure are:

- It is a simplified test method that requires only the accurate measurement of electric heat input, while the existing method requires multiple measurements of temperature and either air or refrigerant flow rates.
- The new test method provides a better reflection of overall system heating capacity and is not sensitive to fluctuations introduced by uncertainty associated with fans heat input or airflow distribution through the evaporator, to which other methods are sensitive at.

The fact that the cooling capacity of the Calibrated Box Cooling System has to be determined first may appear a disadvantage, as it requires effort and consumes time. However, such a calibration effort can be a once–off test and with good control the figures obtained can be used for subsequent capacity tests, with recalibration only required at perhaps monthly intervals. The new EHI test procedure proposed responds to the present industrial need for more accurate means of evaluating the TTC units heating capacity.

Heating Capacity measurements were performed on two different single-compartment Thermo King TTC units using both the existing standard method and the proposed EHI method. A detailed analysis of the results obtained from the trailer TTC unit model SL 400e is presented in Chapter 4, and a similar comprehensive analysis is performed on the truck TTC unit model TS 500 in Chapter 6. It is acknowledged that the air mass flow rates through the evaporator were determined by the manufacturer based on the wind tunnel test method and certified by the ATP test facility, Cemograf, France [Figure 1.7]. The air mass flow rates used in the heat balance calculations of the ATP heating capacity test procedure for both SL 400e and TS 500 units were selected from the technical specifications published by the manufacturer [Thermo King, Technical Specifications (2004)]

CHAPTER 4

CHARACTERISATION OF THE SINGLE-COMPARTMENT STANDARD SL 400e TRAILER UNIT IN HEAT MODE

CONTENT:

The importance of Heat Mode and the field problems that occur for different types of TTC units were highlighted in Chapter 2. As a result, increased demand for extended chatacterisation of TTC unit in heat mode through in-house experimental tests was highlighted. Once a new highly accurate test procedure and test facility for heating capacity measurements were proposed and validated, Chapter 4 responds to the need of further tests to analyse the problems that appear in heating. Therefore, the standard, "off-the-shelf", single-compartment SL 400e trailer unit was selected for study. The heating capacity figures together with the problems that appear during heat mode operation are quantified for the first time in this chapter.

4. 1 INTRODUCTION

Despite the clear importance of Heat Mode during the transportation of perishable products there is a marked absence of research regarding the characterization, analysis and optimization of the cycle for both single- and multi-compartment TTC units. It has to be noted that while the manufacturer presents technical specifications of the cooling performance, no figures for the heating capacities are generally provided due to the inability of obtaining accurate measurements based on the existing standard ATP test method. Responding to this need, Chapter 4 investigates the behaviour of a diesel engine driven SL 400e type trailer unit in heat mode [Figure 4.1], based on the novel accurate EHI test procedure. The main objectives are to:

i) Determine the first accurate heating capacities under different test conditions.
ii) Highlight and quantify for the first time the extent of the problems that appear during heat mode operation especially at ambient temperatures below –10°C.

Figure 4. 1 The SL 400e single-compartment trailer unit.

4.1.1 SL Line of Single - Compartment Trailer TTC Units

The most common single-compartment trailer TTC units operating worldwide are the Thermo King SL systems. Four different unit types are available: SL100e, SL200e, SL400e and Spectrum SL and a technical specification for each of them is presented.

Table 4.1. Technical Specifications of Thermo King's SL line of TTC units [Thermo King web site].

SL unit type	Cooling Capacity at 0°C[W]	Engine	Compressor
SL 100e	9200	TK 482; direct injection diesel engine; P = 18.3kW	X 426 Displacement 423 cc
SL 200e	13000	TK 482; direct injection diesel engine; P = 18.3kW	X 426 Displacement 423 cc
SL Spectrum	15000	TK 486; direct injection diesel engine; P = 25.7kW	X 430 RL Displacement 492 cc
SL 400e	17500	TK 486; direct injection diesel engine; P = 25.7kW	X 430 RL Displacement 492 cc

The worldwide sale figures for 2003 presented in Figure 4.2 indicate that the SL 200 type has the highest rate of 57%, while the SL 400e unit analysed in this chapter also reflects a significant 25% of sales [Thermo King Market Report (2004)].

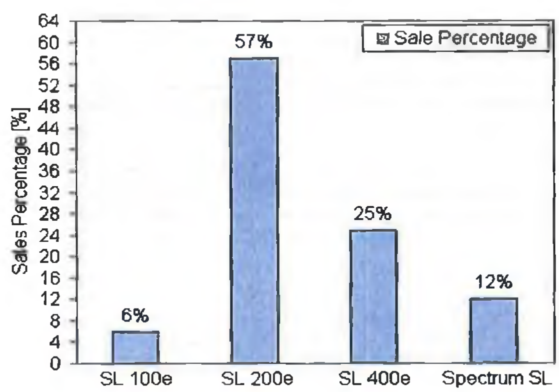


Figure 4.2 Worldwide Sale Percentages % for 2003 of the SL Trailer Units [Thermo King (2004)].

In northern hemisphere countries where heat mode is necessary for long periods of time the demand for SL 400 units is increased up to 40% as a results of the highest heating capacity capability from this product line [Swedish Market Report (2003)]. The SL 400 unit shown in Figure 4.1 is usually employed as a single-compartment trailer unit used for journeys over 300 km, with class B trailer bodies/ food compartments of 13.6 m length, thermally insulated using 150 mm thickness polyurethane foam with 0.3 W/m^2°C thermal conductivity [ATP (1970)].

4.1.2 General Description of the SL 400e Unit

The SL 400e unit shown in Figure 4.1 is designed for rigid body trailers and is usually mounted on the temperature controlled food compartment above the drivers cab [Figure 1.1]. An overview of the main system components together with the principle of operation in both cool and heat modes for a typical TTC unit was provided in Chapter 1, section 1.3.2. However, a more detailed schematic of the SL 400e unit is presented in Figure 4.3. As a particularity, this system has an electronic throttling valve (ETV) installed on the low-pressure side of the system between the evaporator and the accumulator. The ETV (4) is a stepper modulation type valve with maximum 800 steps that is capable of being partially opened and closed. The normal position is open and the valve can be closed incrementally in small steps by varying the supply voltage from a microprocessor controller. During cool mode cycle the ETV protects the system against high compressor discharge pressures when it is operating in ambient temperatures higher than of +35°C, while in heat mode cycle the valve has the role of a suction pressure regulator maintaining a 21-psi nominal value. Other technical specifications for SL 400's main components are:

i) Compressor (1): The unit is equipped with a four–cylinder reciprocating compressor with 492 m^3 displacement Model X 430 RL, Thermo King fabrication.
ii) Evaporator (5): This component is a finned–tube cross flow heat exchanger with one fluid mixed (air) and one unmixed (refrigerant). It has 14 copper circuits and 152 tubes with 0.375 mm external diameter, distributed on 8 rows. The primary heat exchange area is 4.831 m^2, while the secondary surface area is 44.22 m^2. A powerful fan delivering 5500 m^3/h in high speed and 3625 m^3/h in low speed mode draws compartment air through the evaporator coil.
iii) Accumulator Tank (7): This tank, which is 0.7 m height and has 0.3 m internal diameter uses a "U" shaped tube with a large, raised opening at one end that allows unrestricted gas flow to the compressor.

In Figure 4.3 the SL 400e unit is presented schematically during heat mode operation. The compressor (1) receives low-pressure refrigerant vapour through the suction line (8). High-pressure, high temperature vapour travels through the discharge line (2) to the 3-way valve (3), which directs refrigerant gas either to the condenser in cool mode or directly to the evaporator (5) through the hot gas line (4). As it passes through this coil, the latent heat energy of the hot gas refrigerant is released to warm the air in the box/ food compartment, as the refrigerant changes from vapor to liquid. Condensed in the evaporator, high-pressure liquid refrigerant is transferred to the accumulator tank (7). This last component of the heating cycle has a very significant importance because it is designed as an expander in order to provide low pressure and temperature vapour to the compressor suction line while the liquid trapped at the bottom of the accumulator is metered back to the compressor at a slow and safe flow rate.

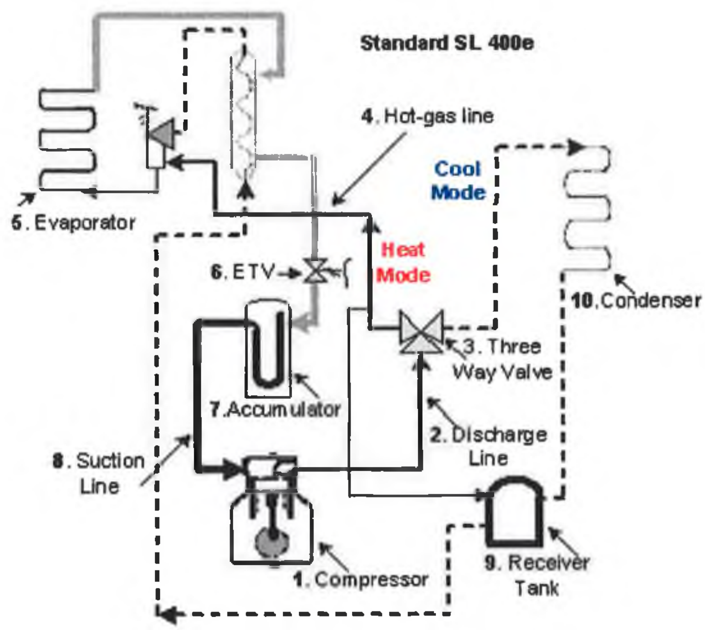


Figure 4.3 Main Components of the single-compartment SL 400e trailer unit in Heat Mode.

While the manufacturer specifies a maximum 60-psi safety limit for the compressor suction pressure, the existing unit design allows an optimum 21-psi to prevent compressor overload. This pressure is automatically maintained by the ETV (6).

4.2 EXPERIMENTAL METHODS AND TEST FACILITY

The SL 400 standard unit was tested using both the new EHI test procedure and the standard ATP method based on heat balance calculations across the evaporator presented in Chapter 3. These tests were performed using the first industrial based heating test facility capable of running tests with ±3% accuracy (Figure 3.3).

4.2.1 Test Facility

Following the schematics presented in Figures 3.1 and 3.3, the heating tests were conducted with the SL 400e trailer unit installed on a calibrated box with the evaporator in heating, while the condenser was positioned within the temperature controlled test chamber's ambient room as presented in Figure 4.4. The test temperature within the ambient room was controlled at the condenser air inlet.

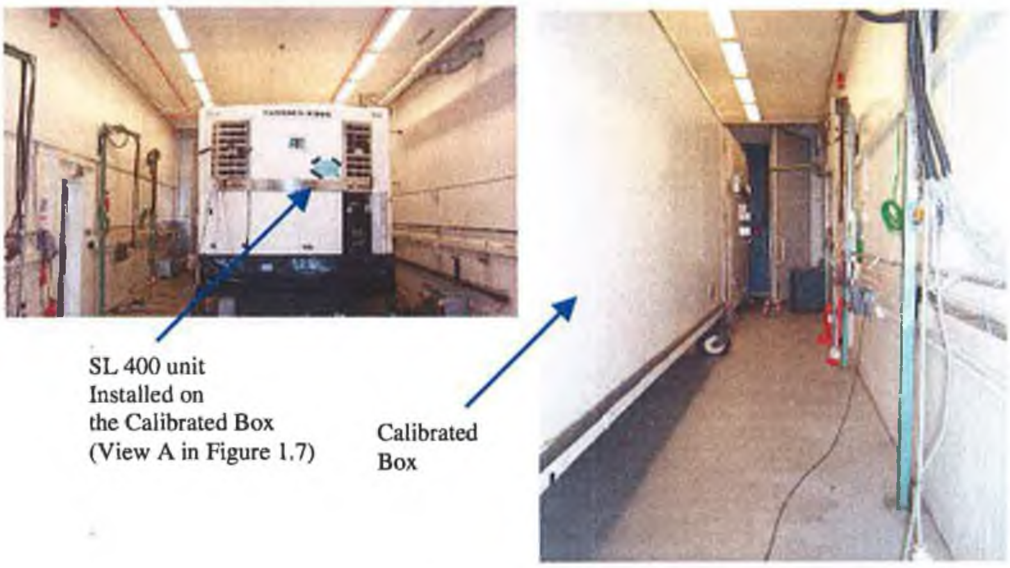


Figure 4. 4. Test configuration for SL 400 Heating Capacity Tests in Ambient Room.

The test ambient temperatures of -10°C, -20°C and –30°C were selected based on average temperatures during autumn and winter months in Europe presented in Chapter 2. The box temperatures for these tests were chosen taking into account the temperature requirements for the most commonly transported fresh products at +2°C for products like orange, mandarins and cheese; +12°C for products like banana, lemon and potato and +21°C selected as a maximum set-point for the unit running in heat mode [ASHRAE (1997)]. The test temperature conditions for both the ambient and box temperatures together with the operational modes are defined in Table 4.2.

Table 4.2. Heating Test Conditions imposed on the SL 400e unit for rigid body trailers.

Test No.	Unit Mode	Ambient Temperature	Box Temperature
1	High Speed Diesel	-30°C	+2°C; +12°C; +21°C
2	High Speed Diesel	-20°C	+2°C; +12°C; +21°C
3	High Speed Diesel	-10°C	+2°C; +12°C; +21°C
4	Electric Mode	-30°C	+2°C; +12°C; +21°C

The heating tests were performed with the SL 400e unit running at maximum compressor rotational speed of 2442 rpm in so called high-speed-diesel mode and with the unit on electric motor standby power, which is entitled electric mode. The standby electric motor has a 9.3 kW maximum power input and is run on 380V, 3 phases, operating at 50Hz.

4.2.2 Test Measurements

The main measurements for the heating test facility and the TTC unit tested were presented in Chapter 3, section 3.5.1. Supplementary temperature and pressure measurements that were recorded for the SL 400e unit for a better understanding of the operation in heating are overviewed in Figure 4.5. Explanations of the temperature and pressure abbreviations together with the logistic of each measurement are described in Table 4.3. The measurements required to monitor and control both the test facility and the SL 400e unit were performed using instrumentation with high measurement accuracy presented in Table 4.4 and Appendix H. All thermocouples were calibrated to an accuracy of ±0.5°C using a D55 SE temperature calibrator, while the calibration of the pressure transducers (±0.4%) was performed using a Superb pressure calibrator, type PCC3-H-200-2.

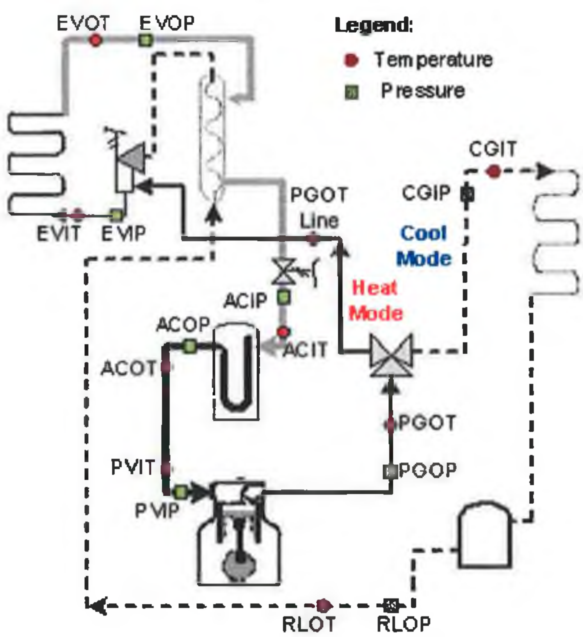


Figure 4.5 Location of Pressure and Temperature measurements on the SL 400e Standard unit. Explanation provided in Table 4.3.

Table 4.3. Defining the pressure and temperature measurements on the SL 400e trailer unit.

Sensor Abbreviation	Explanation	Required to measure
PGOT	Compressor discharge temperature	Compressor superheat.
PGOP	Compressor gas discharge pressure	Compressor superheat.
EVIT	Evaporator inlet temperature	Heating capacity.
EVIP	Evaporator inlet pressure	Evaporator gas inlet enthalpy.
EVOT	Evaporator outlet temperature	Heating capacity.
EVOP	Evaporator outlet pressure	Evaporator liquid outlet enthalpy.
ACIT	Accumulator inlet temperature	Evaporation boiling point
ACIP	Accumulator inlet pressure	Accumulator inlet enthalpy
ACOT	Accumulator outlet temperature	Evaporation temperature
PVIT	Compressor inlet gas temperature	Compressor inlet superheat.
PVIP	Compressor inlet gas pressure	Compressor inlet superheat.
CGIT	Condenser gas inlet temperature	Charge trapped in condenser
CGIP	Condenser gas inlet pressure	Charge trapped in the cycle.
RLOT	Receiver Tank Temperature	Receiver outlet temperature
RLOP	Receiver Tank Outlet Pressure	Charge trapped in receiver tank

Table 4.4 Test Instrumentation used to monitor and control the SL 400e heating capacity tests.

Instrument Description	Instrument Range	Accuracy
Type-T Thermocouples-Special Grade	-200 to 350 °C	± 0.5°C
Watt-hour Transducer; Model DL31KA2	0 to 20 kW	±(0.1% + 1 Watt)
TK Pressure Transducers	0 to 500 psi/ 0 to 200 psi	± 0.4% / ± 0.25%

The fuel consumption was determined using a portable Danfoss fuel mass flow meter, type Mass 3000, with ±0.15% accuracy over a wide measuring range. The mass flow meter is 115/ 230 V a.c. and 24 V d.c. compatible with the technical specifications defined in Table 4.5.

Table 4.5. Fuel Mass Flow Meter Technical Specifications.

Type	Outputs	Measurements
Danfoss	2 analogue outputs	Mass flow rate
Mass 3000	1 frequency – pulse output	Total Mass
	2 relay outputs	Density

4.3 HEATING PERFORMANCE FOR THE STANDARD SL 400e

The heating capacity of the standard SL 400e trailer unit running in both high speed diesel and electric modes is presented in this section based on measured results obtained from both the new EHI test procedure and the standard heat balance method, defined in Chapter 3. These figures for a standard "off-the-shelf" production unit, will be compared with the heating performance resulted from a modified SL 400e trailer unit in Chapter 5.

4.3.1 Heating Capacities – High Speed Diesel Mode

A summary of the results obtained for six different test conditions with the unit operating in high-speed diesel are presented in Table 4.6. The rotational speed of the compressor was 2440 rpm, while the airflow through the evaporator was 5500 m^3/h.

Table 4. 6. Measured heating capacity figures of the SL 400 unit in high-speed diesel mode.

Test No.	Ambient Temp°C	Box Temp°C	Heating Capacity Standard Method [W]	Heating Capacity EHI Method [W]	% Difference
1	-10 °C	21 °C	12,429	11,872	-4.5%
2	-10 °C	12 °C	12,919	11,956	-7.5%
3	-20 °C	2 °C	11,806	11,600	-1.7%
4	-20 °C	12 °C	10,850	10,600	-2.3%
5	-30 °C	2 °C	10,494	10,326	-1.6%
6	0 °C	12 °C	15,497	12,477	-19.48%

The results shown in Table 4.6 are also presented in graphical format in Figure 4.6 together with the reserve heating capacity. This latest indicator of the unit performance is considered to be the difference between the maximum heating capacity measured under specific test conditions and the heat losses between the box/ food compartment air temperature and external ambient temperature through the walls insulation, under the same conditions calculated using equation 4.1. The reserve heating capacity is a measure of the supplementary remained heating capability after the unit compensates the heat losses for the operational conditions.

$$Q_{Box\ Loss} = Heat\ Losses = K \times S \times (T_{box} - T_{amb})\ [W] \qquad (4.1)$$

Figure 4.6 shows that the heating capacities obtained from the EHI method for high-speed mode range between 10,000W and 15,500W depending on test temperature conditions. It can be noted that the heating capacity of the unit is less at lower ambient temperature conditions as a result of increased heat losses through the walls to the external ambient. At the same box temperature of +12°C and different ambient temperatures of -10°C, -20°C and 0°C that correspond with test numbers 2, 4 and 6, it can be concluded that as a reference to the heating capacity obtained for 0°C in ambient room, the figures obtained for the other two ambient conditions are almost 24% lower for the 20°C temperature difference (at ambient –20°C) and 20% for 10°C temperature difference (at ambient –10°C). Based on the data presented it was highlighted that the heating capacities of the unit had increased by a significant 24%

when an ambient of 0°C existed, while smaller differences of +4% were noticed between the heating capacities at lower ambient temperatures like –10°C and –20°C. A more detailed study of the influence of ambient and box temperatures on the heating performance of the TTC unit is presented in Figure 4.9(a,b).

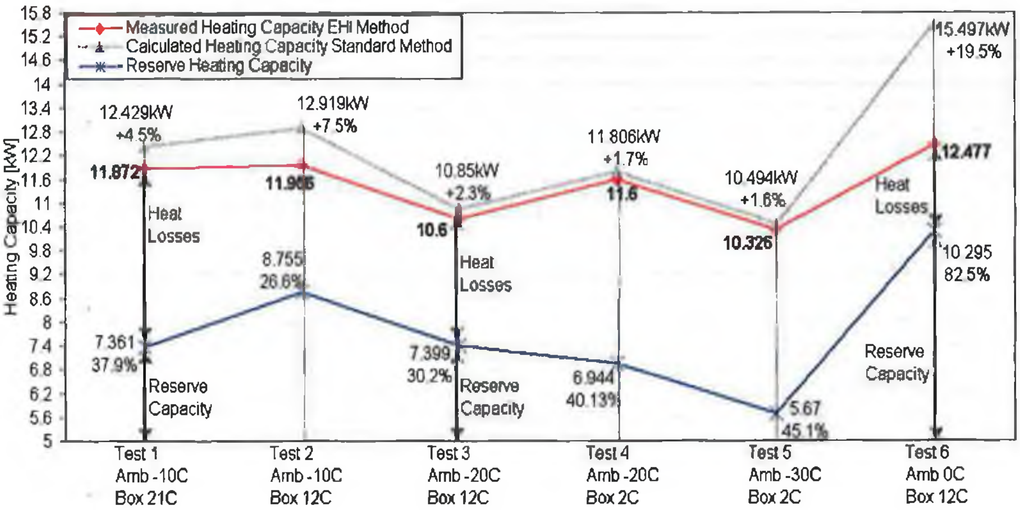


Figure 4.6 Heating Capacities for SL 400e TTC unit in high-speed diesel mode. The percentage difference shown refers to the difference in measured capacity between the two methods as a percentage of the results obtained using the EHI method.

To support the conclusions from Chapter 3 regarding the differences between the heating capacities measured with the EHI and standard ATP test method, it can be noted that the values obtained using the EHI method are between –1.6% and -19.5% lower than the figures obtained with the standard test procedure. These differences can also be explained due to the fact that the calculations for air outlet temperature were performed without taking into account the gains from the air circulating fan motors that influences the results for high ambient temperatures (test 6) or for heat losses from the warm box environment influence to the colder external surroundings that introduce higher errors as the temperature difference between the ambient and the calibrated box/ food compartment increases. These influences were mathematically quantified through an empirical correction factor that improves with 4% the prediction of heat balance calculations on the evaporator included in the first mathematical model that describes the heat mode behaviour of a TTC unit and overviewed in Chapter 7, section 7.4.1. This correction is also proposed for improved accuracy of the heating capacity results obtained with the standard ATP test method as involves the same steps in the heat exchange calculation on the evaporator.

4.3.2 Heating Capacities – Electric Mode

Similar heating capacity test results are presented in Table 4.7 for the unit running in Electric Mode, together with the trends and reserve capacities shown in graphical format in Figure 4.7. The air mass flow rate through the evaporator in these operating mode is 3600 m^3/h.

Table 4. 7. Measured Heating Capacity for the SL400e unit in electric mode.

Test No.	Ambient Temp°C	Box Temp°C	Heating Capacity Standard Method [W]	Heating Capacity EHI Method [W]	% Difference
1	-10°C	21°C	9,512	8,193	-13.8%
2	-10°C	12°C	10,225	8,794	-14%
3	-10°C	2°C	10,334	9.885	-4.4%
4	-20°C	12°C	9,917	7,548	-23.9%
5	-30°C	2°C	8,042	5,992	-25.5%
6	-30°C	12°C	7.886	5,835	-26%
7	0°C	12°C	10.871	9,408	-13.5%

Measured Heating Capacity ranged between 5,800 W and 10,000 W depending on test temperature conditions, which clearly indicates that the electric mode not only produces on average 50% lower heating capacity than the high-speed diesel mode but also is more sensitive to the ambient and box temperature changes due to lower air mass flow rate through the evaporator and box/ food compartment that experiences more rapidly temperature changes. Higher heating capacities are obtained at lower temperature difference between ambient and box set-points due to lower heat losses.

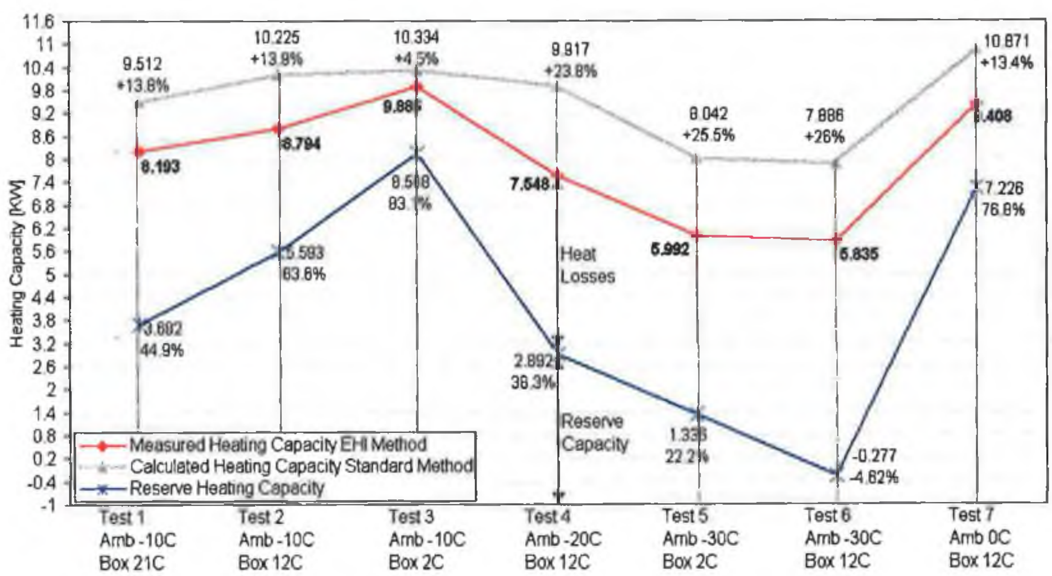


Figure 4.7 Heating Capacities for SL 400 in electric mode. The percentage difference shown refers to the difference in capacity between the two test methods compared with the EHI method results.

Tests 1 and 2 were performed at the same ambient temperature –10°C, but different box temperatures of +21°C and +12°C, yielding a 6.8% difference in the heating capacity for the 9°C increase in box temperature. Tests 2, 4 and 6 were performed at the same box temperature, but different ambient values: -10°C, -20°C and –30°C. Considering that the heating capacity differences were related to the figure obtained for –30°C ambient temperature, it can be noticed that 18.9% difference exists at –20°C ambient temperature and a 21.4% difference for –10°C. It can be concluded that heating capacity drops drastically at ambient temperatures below –20°C. The heating capacities obtained with the new EHI test method are between 4.45% to 27.4% lower than the figures obtained using the standard test method. The difference that appears between the two test procedure results is higher as the temperature difference between the ambient and the box increases. Higher differences are again evident at low ambient temperature conditions as a result of increased heat losses. The influence of condensing and ambient temperatures on the heating capacity is more detailed presented in Figure 4.9.

The good test repeatability was verified based on several tests performed at the same conditions. Maximum 50W capacity variations were obtained between the results of the tests performed in high-speed diesel mode, while maximum 30W differences were obtained for electric mode. As an example, Figure 4.8 shows the changes in the heating capacity figures obtained for three different tests repeated for the same ambient temperature of –30°C and box temperature of +2°C in electric mode. It is noted that to comply with the accuracy requirements of the heating capacity results, the test were repeated at different time periods of three or four days after tests at different ambient and box temperatures were performed.

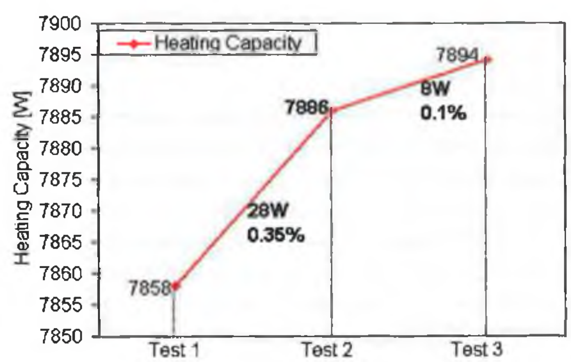
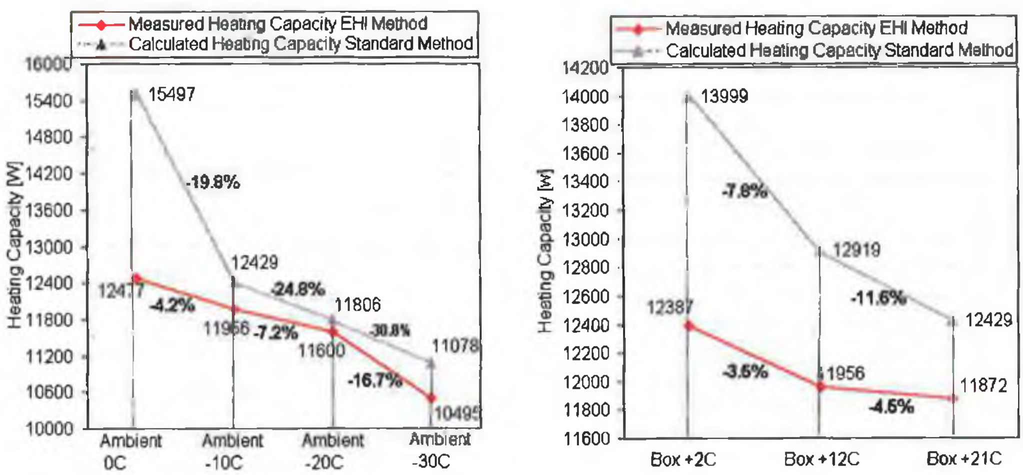


Figure 4.8 Heating capacity tests repeatability at –30°C ambient and +12°C box temperatures.

4.3.3 Influence of Ambient and Box Temperatures on Heating Capacity

A) High Speed Diesel Mode

A better understanding of the ambient temperature influence on the heating capacity of the unit operating in high speed diesel mode can be had by referring to Figure 4.9(a) where differences obtained in the measured heating performance are presented for ambient temperatures of 0°C, –10°C, -20°C, -30°C at the same +12°C set-point. While Figure 4.9(a) shows an overall drop of –16.7% in the heating capacity between 0°C and –30°C, the slope of the graph indicates that the most significant decrease occurs below –20°C, due to increased heat losses through the box walls to the ambient environment as a result of higher temperature difference across the walls. This helps to explain the apparent failure of the system in heating shown by the field data in Figures 1.11 and 2.9 when the ambient temperature dropped below –18°C.



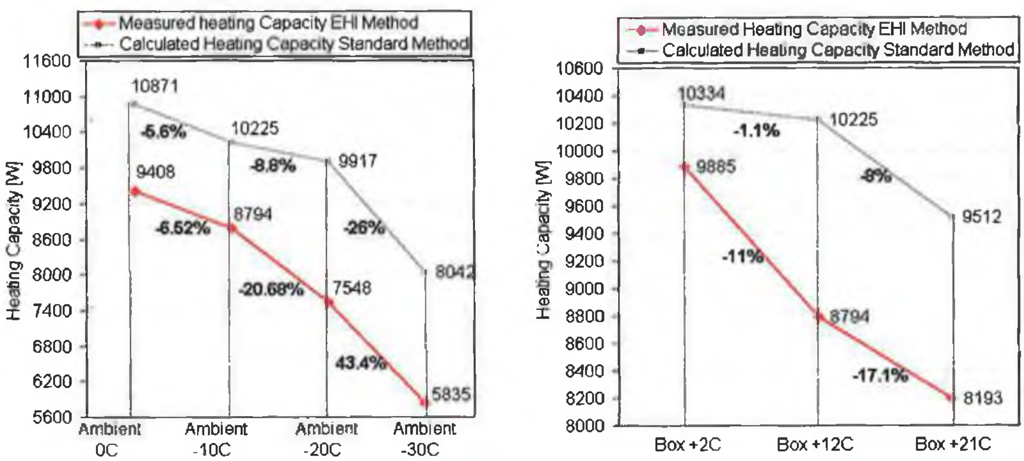
a) Influence of Ambient Temperature. b) Influence of Box Temperature.

Figure 4.9 The influence of condensing and ambient temperatures on the Heating Capacities.

The internal box temperature that is the condensing temperature also influences heating capacity and the performance of the SL 400e unit running in an external ambient temperature of –10°C, but different box temperatures of +2°C, +12°C and +21°C is presented in Figure 4.9(b). It is clear that the heating capacity of the unit decreases with an increase in the internal box or condensing temperature. While the unit delivers a heating capacity of 12,387W at +2°C box temperature, this decreases by almost –5% as the condensing temperature increases to +21°C. The TTC unit is more sensitive at variations of ambient temperatures.

B) Electric Mode

A better understanding of the influence of external ambient temperature on the heating capacity can be had by referring to Figure 4.10(a), where heating capacity of the SL 400e unit running in Electric Mode at the same +12°C box temperature is presented for four different ambient temperatures of 0°C, –10°C, -20°C and –30°C. An overall drop of –43.4% in the heating capacity can be noticed between 0°C and –30°C, while the slope of the graph indicates that the most significant decrease occurs below –20°C due to increased heat losses through the box walls to the ambient environment as a result of higher temperature difference across the walls. This helps to explain the failure of the system operating in heating on field, as presented in Figures 1.6 and 2.11, for similar ambient temperatures below –18°C.



a) Influence of Ambient Temperature b) Influence of Box Temperature.

Figure 4.10 The influence of condensing and ambient temperature on the Heating Capacity.

The internal box temperature that is the condensing temperature of the cycle also influences the heating capacity of the unit. Figure 4.10(b) shows the results obtained for ambient temperature of –10°C and box temperatures of +2°C, +12°C and +21°C. While the unit delivers 9,885W capacity at +2°C, a decrease of –17% is recorded at +21°C box temperature.

C) Discussion

The decrease in the heating capacity of the system with an increasing internal box set-point temperature can be explained by the fact that the theoretical and the actual capacity of the compressor is reduced with an increase in the condensing temperature, which is the test set-point in the calibrated box/ food compartment. Theoretically, the compressor has a displacement volume equal to its swept volume and the condensing temperature does not affect the density of the suction vapour. Therefore, the theoretical mass of refrigerant displaced by the compressor remains constant at all condensing temperatures and the theoretical heating capacity is only a function of the heating effect per unit mass of refrigerant circulated. Based on these assumptions, the difference in the theoretical heating capacity of the compressor at the two different condensing temperatures results entirely from the difference in the heating effect per unit mass. The reduction in actual capacity may be attributed to reductions in the volumetric efficiency and heating effect of the system [Dossat *et al*, (2003)]. Increasing the condensing temperature while the suction temperature remains constant increases the compressor ratio, reducing the volumetric efficiency of the compressor. Consequently, the actual volume flow rate of the vapour displaced by the compressor decreases. Therefore, even though the density of the vapour entering the compressor remains the same at all condensing temperatures, the actual mass flow rate of refrigerant circulated by the compressor decreases. Increasing the condensing temperature also increases the isentropic discharge temperature, which results in higher compression ratio. The loss of compressor efficiency and capacity resulting from an increase in the condensing temperature of the cycle is more serious when the suction temperature of the process is too low. The reduction in the compressor volumetric efficiency is responsible for the greatest portion of the decrease in the actual capacity of the compressor [Stoecker (2003)].

4.4 PROBLEMS THAT APPEAR IN HEAT MODE

To obtain a complete characterisation of the unit behaviour and to emphasise the problems that appear in heat mode, a series of capacity, temperature control and pull up tests were performed on the SL 400 standard unit. Based on the results obtained, five types of problems were highlighted during the heat mode operation at ambient temperatures below –10°C and summarised as follows:

- Low or insufficient heating capacity.
- Capacity variations for the same temperature test conditions.
- Long pull-up time.

- Poor set-point temperature control.
- Low compressor suction and discharge superheat.

The experimental results are overviewed in the following sections.

4.4.1 Insufficient Heating Capacity

Once the first accurate heating capacity tests were performed, it was concluded that while the unit develops adequate heating capacity in high-speed diesel mode, the electric mode lacks the capacity to achieve box temperatures higher than +12°C for sub –20°C ambient temperatures. Figure 4.11 shows the measured heating capacity based on the accurate EHI test method and the reserve capacity of the unit running in electric mode for two tests conducted at a –30°C ambient temperature and +2°C, +12°C box temperatures respectively. It can be concluded that the heating capacity of the SL 400e unit running in electric mode is just capable of maintaining a +2°C set point temperature in the box when the ambient temperature is lower than –25°C, with only 1336 W reserve capacity.

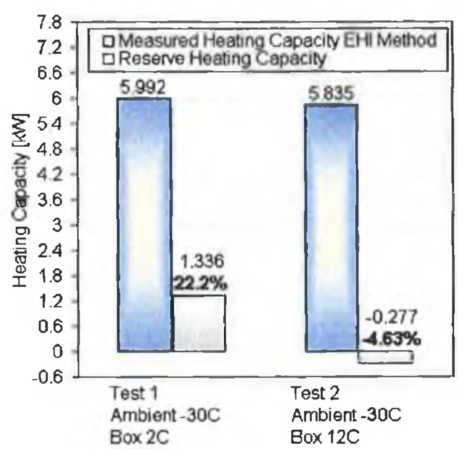


Figure 4.11 Insufficient Heating Capacity for SL 400e unit at ambient –30°C, in electric mode.

Figure 4.11 also shows a deficit of –4.6% in the heating capacity when a +12°C transport temperature is requested in a –30°C ambient temperature. This lack of capacity would be further increased if either the ambient temperature dropped further or a bigger box temperature was requested. It is clear that even if the SL 400e standard unit has the maximum capacity from the Thermo King single-compartment trailer line does not maintain a set-point temperature higher than +10°C while operates in electric mode at ambient temperatures lower than –25°C. Therefore, the lack of heating capability increases for the other types of SL trailer units. The most common unit of this production line, the SL 200 type has an estimated 25% less heating capacity (Table 4.1) that results in the impossibility to achieve box set-point temperatures higher than +2°C, while operates in the same temperature conditions.

4.5.2 Long Pull-Up Time

Figure 4.12 shows the pull-up time required to achieve a box set-point temperature of +25°C for both high-speed diesel and electric modes when exposed to –30°C. It is acknowledged that +25°C represents the maximum set-point temperature that can be selected for the SL type TTC units. Due to low heating capacity in electric mode, the standard SL 400e unit was equipped with 3kW capacity electric heaters installed in front of the evaporator. Even with this maximum enhancement in the heating capacity applied by the manufacturer, the performance of the unit is insufficient, as 8 hours represent a common period of time for the fresh product journey and the unit is able to achieve the set-point temperature after 7 hours of pull-up (Figure 4.12).

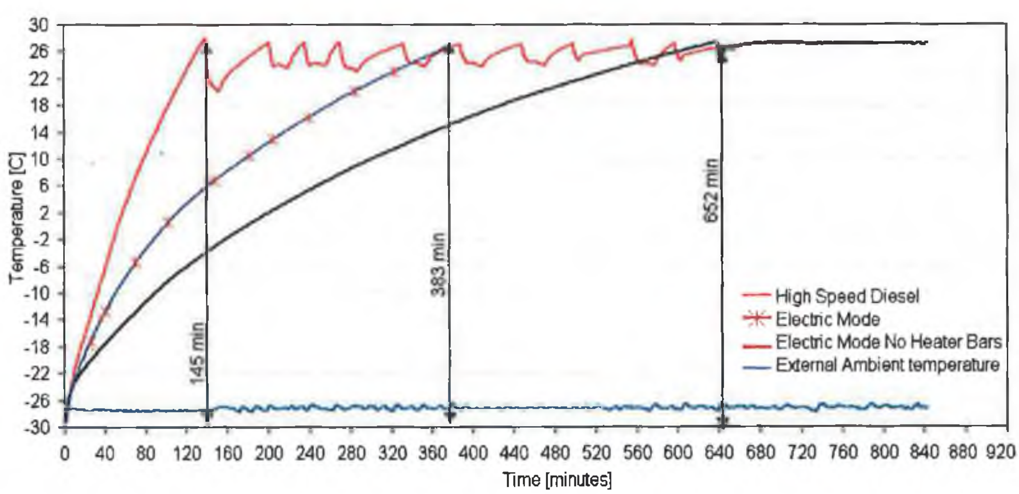


Figure 4.12 Pull-Up time for the SL 400 unit running in high-speed diesel and electric modes to achieve +25°C, while working in –30°C external ambient air.

When the SL 400e unit is operating at ambient temperature conditions lower than –25°C, the pull-up time to obtain the set-point temperature in the trailer can extend to 11 hours when the unit is running in electric mode with the heaters off, based only on refrigerant hot-gas heating capacity and almost 7 hours with the electric heaters on.

For high-speed diesel mode, due to an average of 60% increased heating capacity, the pull-up time is less, of only 2:20 hours.

4.4.3 Poor Set-Point Temperature Control

Figure 4.13 shows the level of set-point temperature control that is achieved at a –30°C ambient temperature for different trailer set-point temperatures of +2°C, +12°C and +21°C, while the unit operates in electric mode.

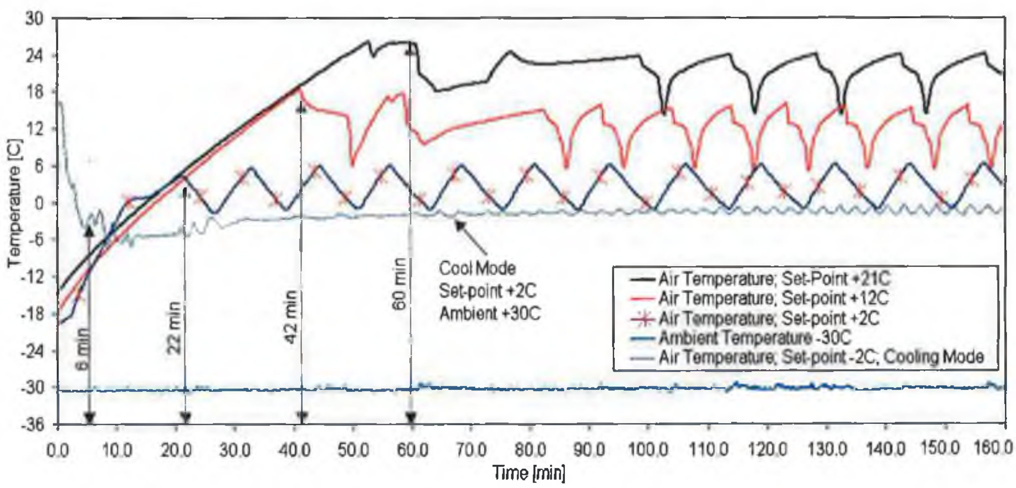


Figure 4.13 SL 400e Standard Unit set-point temperature control in electric mode. Ambient –30°C, Box Temperature +2°C, +12°C and +21°C.

As a result of low heating capacity (Figure 4.11) and "charge migration" (Figure 4.14) in the cooling cycle, the SL 400e unit has poor temperature control within –4°C below and +1°C above the set-point, while operates in low ambient temperature conditions sub –25°C. Again emphasising the heating capacity problem, it can be noted that for +12°C and +21°C calibrated box set-points the unit runs for up to 40 minutes below the desired temperature. For a comparison, the temperature control pattern for cooling at –2°C box temperature and +30°C ambient temperature was also plotted. While the control range is within –1°C and +5°C for the unit operating in heating at the same temperature difference between ambient and box, very good temperature control is obtained for cooling, of ±1°C around set-point as requested by standards [ATP (1970); Standard 1110 (2001)].

4.4.4 Variations in Heating Capacity – Charge Migration

The term "charge migration" was introduced in the previous section and this refers to the significant variations that can result in heating capacity, due to the refrigerant charge that can remain in the condenser and receiver tank after the cooling cycle. Charge migration becomes an issue when the TTC unit-operating mode alternates between cool and heat to achieve the required set-point. When the unit switches to heating, the refrigerant gas from the compressor discharge flows directly into the evaporator (Figure 4.3). As a result, an amount of liquid refrigerant is trapped in the cooling side of the cycle. It is this effect that contributes to the +15% variation in heating capacity shown in Figure 4.14.

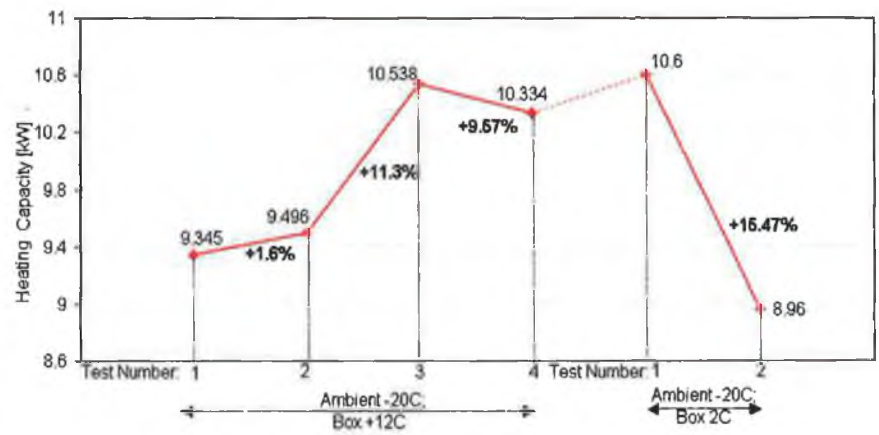


Figure 4.14 Variations in heating capacities because of charge remaining in the system.

While there was no direct means of measuring the amount of charge within the evaporator, following test sequences were run in order to generate different levels of charge in the heat cycle (Table 4.8).

Table 4.8. Unit operation conditions previous to the heating capacity tests.

Test No.	Test Temp Conditions	Unit operation conditions previous to the heating capacity test when the SL 400e was started in maximum heating.
1		Unit was maintaining –20°C temperature set-point in the box, through alternative cooling–heating cycles for 2 hours.
2	Ambient –30°C	Unit was maintaining –20°C temperature set-point in the box, through alternative cooling-heating cycles for 1 hour.
3	Box +12°C	Unit was started directly in heating at initial ambient temperature higher than the box temperature. Ten minutes cool mode operation.
4		Unit was started directly in heating at ambient temperature at the set point and box temperature higher than the box temperature.
1	Ambient –30°C	Unit was started directly in heating at ambient temperature at the set point and box temperature higher than the box temperature.
2	Box +2°C	Unit was maintaining –20°C temperature set-point in the box, through alternative cooling–heating cycles for 2 hours.

For the same ambient and box temperature conditions we can have differences between the heating capacities within 11% to 15.5% that can have as result poor temperature control to maintain the set point and high pull-up time. It can be concluded that the heating capacity of the SL 400e unit varies considerable as a result of charge migration in the cooling cycle, and this variability needs to be eliminated or reduced. The results presented in Figure 4.14 are obtained using the existing standard charge control system that is installed on the SL 400e unit, as described in Chapter 5, Figure 5.1(a).

4.4.5 Compressor Failure Costs

The effect of the unit operation at low suction and discharge superheats is closely related with the failure problems of the compressor components. A general overview of the costs incurred due to compressor failures during 2002, 2003 and 2004 is presented in Figure 4.15 [Thermo King Report (2004)].

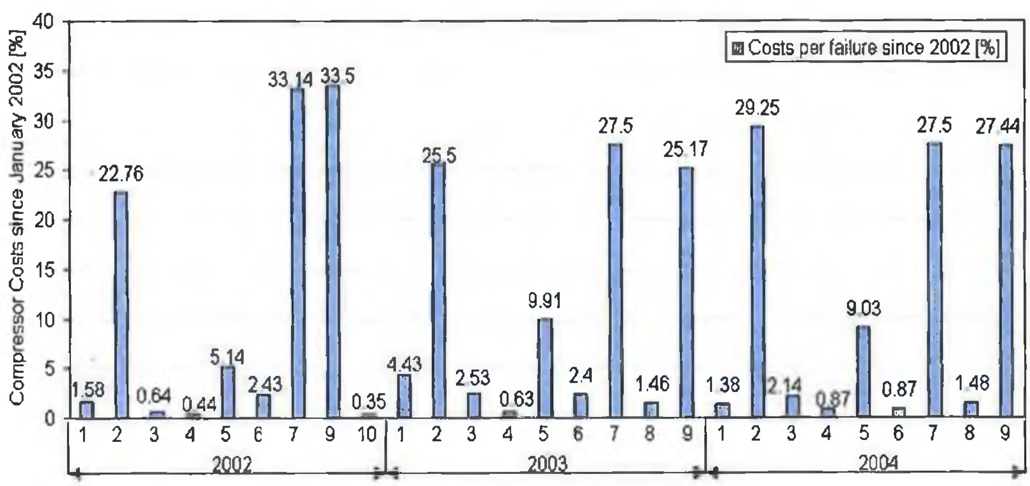


Figure 4.15 Total costs involved in SL 400 compressor claims for 2002, 2003 and 2004.

Table 4.9 shows a classification of the top ten component problems for the SL 400e compressor accounting for 90% compressor returns.

Table 4.9. Top 10 Failures for SL 400e compressor [Thermo King Report (2004)].

Number	Failure	Number	Failure
1	Bearing	6	Gasket
2	Compressor Assy	7	Oil Pump
3	Compressor Body	8	Piston
4	Connecting Rod	9	Shaft Seal
5	Crankshaft	10	Small Failures

Based on the data presented it can be concluded that significant compressor failures are: bearing fretting (5) and the shaft seal failure (9) resulting from long term operation of the compressor at deficient parameters and operating conditions like liquid trapped at the inlet. As a result compressor superheat has an important impact on cost reduction and reliability.

4.4.6 Compressor Suction and Discharge Superheats

At the compressor inlet on the suction line, the refrigerant has to be at least at the saturation limit or superheated. A suction vapor superheat below zero Celsius degrees it means liquid at the compressor inlet. After many hours operating in these conditions, the compressor failures 5 and 9 experiences an increase with high costs involved. Figures 4.16 shows compressor suction superheat when the SL 400 unit is running at –30°C, -10°C ambient temperatures and +2°C, +12°C box set points. It can be concluded that the unit has low suction superheat of an average –6°C and –13°C depending on ambient temperatures. At lower ambient values the compressor suction superheat drops dramatically because of the increased heat exchange with the ambient conditions as a result of lower compressor suction temperature. Figure 4.16 shows that the compressor suction superheat is below 0°C that has as result liquid at the compressor inlet, with negative impact on compressor reliability and efficiency.

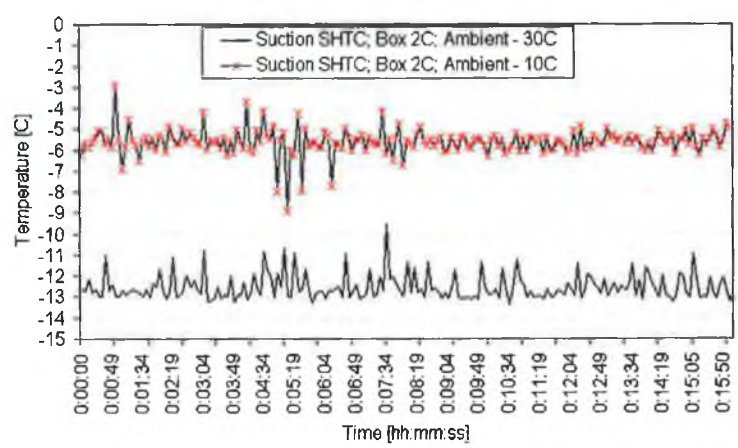


Figure 4.16 Compressor Suction Superheat Box 2°C, Ambient temperature –10°C and –30°C.

Higher compressor discharge superheat offers the guarantee of increased heat released at the evaporator inlet when the vapor temperature drops at the saturation value. While a +20°C superheat obtained at +2°C box temperature is satisfactory, for

+12°C box temperature, a low value of only +6°C is obtained. For the latest situation and considering the heat losses through the pipes the temperature at the evaporator inlet is already close to the saturation value.

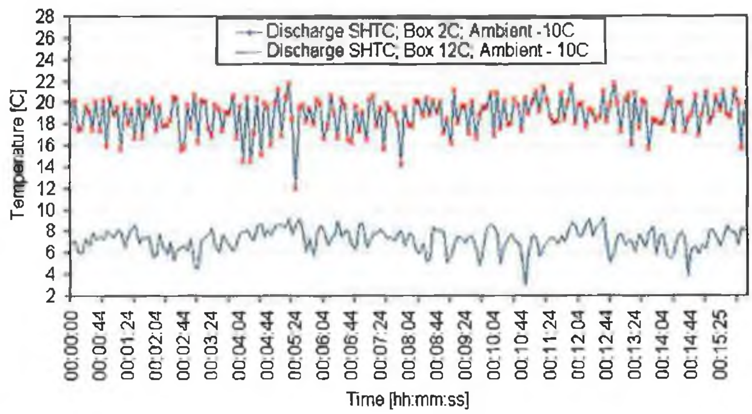


Figure 4.17. Compressor Discharge Superheat at +2°C and +12°C box temperatures and -10° C ambient temperatures.

4.5 SUMMARY

A summary of the most significant heating capacity results together with the first comprehensive analysis of the problems encountered by the single-compartment SL 400e trailer unit during heat mode operation in sub –10°C ambient temperatures is overviewed in this section.

A wide range of tests defined in Table 4.2 was performed on a production SL 400e unit and following summary is provided:

- The heating capacities obtained for electric mode are with 50% lower than the figures obtained for high-speed diesel mode.
- Difference between both test methods, were negligible, less than +4%, when the ΔT between the box and external ambient was less than 15°C. In this instance, heat losses through the box/ trailer walls are small and measured results are comparable. However, when the ΔT across the wall exceeds 20°C, fabric losses are significant and difference of up to 10% can exist between both test methods for high-speed diesel while differences up to 24% were obtained for electric mode.

- Sensitivity of heating capacity to ambient air temperature and condensing temperature was established for the first time in Figures 4.9 and 4.10.

Section 4.5.2 presents the first complete analysis of the problems that appear for SL 400e unit during the operation in heat mode, which can also be extended at all types of TTC units using hot-gas heating cycle as verified in Chapter 6 for a truck TTC unit. It was concluded that the following majors problems are encountered during heat mode operation of the SL 400e trailer unit:

- Low heating capacity at temperatures below –25°C and box temperatures higher than +4°C.
- High pull-up times were recorded at ambient temperatures lower than –25°C for the unit operating in electric mode.
- In the same temperature conditions the unit has poor temperature control within –5°C and +1°C around set-point.
- Variations of up to +15.5% in heating capacity due to charge migration.
- The compressor operates at safety limits with liquid trapped at the inlet as a result of suction superheat below 0°C.

The first comprehensive characterisation and analysis of the problems that appear during heat mode operation are for the first time quantified in this chapter. Considering that the SL 400e unit has the maximum heating capability not only from this production line but also from all types of single-compartment TTC units, it can be concluded that the insufficiencies presented in section 4.4 are amplified for the other systems, which can present insufficient heating capacity even at ambient temperatures of –20°C. This is confirmed by the analysis undertaken for the truck TS 500 unit in Chapter 6, section 6.5 and by the field data presented in Figures 1.6, 2.11 and D.2 (Appendix D). Therefore, the increased need to optimise the TTC unit performance in heat mode has resulted in five design modifications that have a general application on all types of TTC units. These design changes were implemented and tested for SL 440e unit selected for this study and are presented in Chapter 5. A comprehensive comparison between the standard and modified SL 400e units is also overviewed.

CHAPTER 5

CHARACTERISATION OF A MODIFIED SINGLE -COMPARTMENT SL 400e TRAILER UNIT IN HEAT MODE

CONTENT:

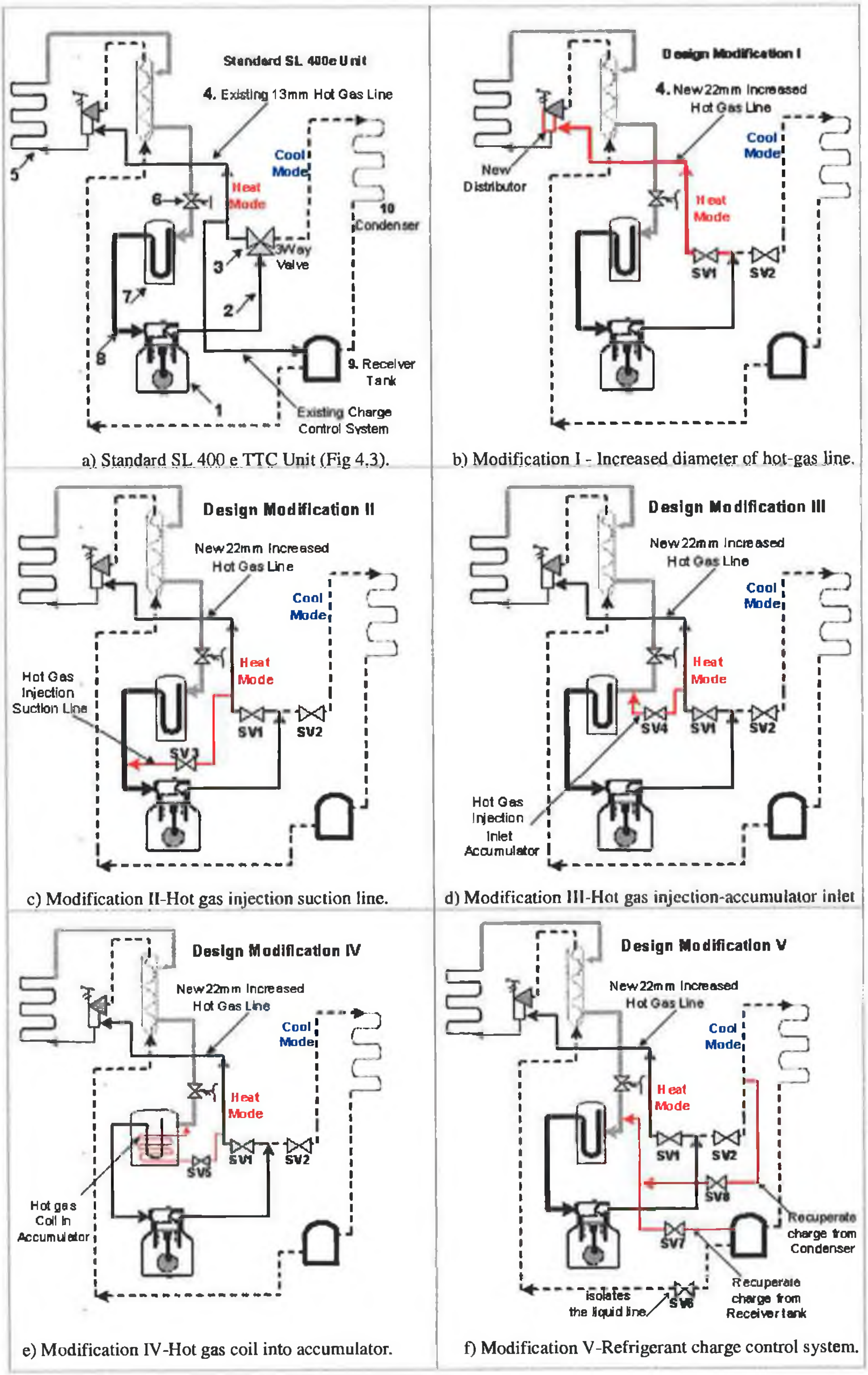
The heating performance of the standard, single-compartment SL 400e production unit was measured and discussed in Chapter 4. The following major problems were established: i) low heating capacity, ii) long pull-up times, iii) poor temperature control, iii) variations in heating capacity due to charge migration and iv) low compressor suction superheat. In light of the system deficiencies identified, five design modifications are defined and accessed in this chapter. These are implemented and tested on a modified SL 400e unit. A complete comparison between the results obtained for the standard versus modified design is overviewed in this chapter [Radulescu *et al.*, (2005)].

5.1 INTRODUCTION

Five design changes were implemented on a standard SL 400e trailer unit presented in Figure 4.3, in an attempt to obtain increased heating capacity and system efficiency in heat mode. These are depicted in Figure 5.1 and include:

- Design Modification I: Increased diameter of the hot-gas line to allow higher hot gas refrigerant mass flow rate through the evaporator.
- Design Modification II: Hot-gas injection at the compressor inlet to increase compressor suction temperature and superheat.
- Design Modification III: Hot-gas injection at the accumulator inlet used to increase both the evaporative temperature and the refrigerant mass flow rate.
- Design Modification IV: Hot-gas coil inserted in the accumulator to increase both the evaporative temperature and the accumulator heat exchange capacity.
- Design Modification V: Charge control system that recuperates the refrigerant liquid trapped in the cooling cycle.

This is the first time that the impact of such modifications has been studied for a compressor running in very low ambient temperature conditions. However, while a complete analysis of the TTC unit's behaviour in cooling mode at high ambient temperatures is presented in the literature [Haas (2000); Plastinin *et al.*, (2000); Dossat *et al.*, (2003)], to the author's knowledge no previous research has been conducted to establish the effect of: i) compressor suction and discharge temperature, ii) evaporating temperature and iii) condensing temperature on the heating behaviour of a TTC unit when it operates at ambient temperatures below –10°C.



a) Standard SL 400 e TTC Unit (Fig 4.3).

b) Modification I - Increased diameter of hot-gas line.

c) Modification II-Hot gas injection suction line.

d) Modification III-Hot gas injection-accumulator inlet

e) Modification IV-Hot gas coil into accumulator.

f) Modification V-Refrigerant charge control system.

Figure 5.1 Summary of the five design modifications introduced to the standard SL 400e production unit to increase the heating capacity and system efficiency in heat mode.

Modification I was made based on an analysis of the optimum pipe sizing of the hot-gas line and the expected optimisation of the compressor operation when different parameters such as compressor suction and discharge temperatures, or evaporating temperature are increased. Another three of the five modifications defined are based on the effect of injecting superheated refrigerant hot-gas from the compressor discharge line into the low side of the heat cycle and these will be grouped and discussed together in section 5.3. A new system used to recuperate the charge trapped in the cool cycle of the unit was also implemented [Figure 5.1(f)] and tested with good results as presented in section 5.4.

5.2 DESIGN MODIFICATION I

In order for any refrigeration system to operate efficiently, the piping connecting its components must be properly sized to assure that the refrigerant flows through the system without creating excessive pressure drops. Such pressure drops are particularly important in refrigeration systems as they can also determine changes in the saturation temperature and specific volume of the refrigerant in a manner that reduces the system's operating efficiency [Dossat *et al.*, (2003)]. Use of the correct pipe diameters can assure the proper pressure drop in a line and can also result in the delivery of an adequate supply of refrigerant to the evaporator, producing a substantial increase in the system efficiency.

5.2.1 Description of Modification I

Modification I corrects a design error inadvertently introduced into the standard TTC unit design as a result of not fully understanding heat mode requirements. After verifying the pipe sizing of the standard unit, attention was concentrated on the hot-gas line, Item 4 in Figure 5.1(a). This circuit is supposed to be sized to minimise pressure drop produced by the flow of hot-gas refrigerant. This pressure drop is added to the saturation pressure of the evaporator in order to determine the discharge pressure of the compressor. When the hot gas line has an undersized diameter these pressures are excessively high, resulting in an increased compression ratio and heat of compression, as well as higher saturation temperature of the condenser. These contribute to reduce the efficiency of the system [Dossat *et al.*, (2003)]. The modification implemented on the standard SL 400e, Figure 5.1(a) involves replacing the existing 13 mm diameter hot-gas line (Item #4) with a 22 mm diameter pipe shown in Figure 5.1(b). To facilitate the introduction of this modified circuit, the existing three–way valve was also replaced by two automatically controlled on/off solenoid valves SV1 and SV 2. When the unit operates in heating valve SV2 is closed, while during cooling operation valve SV1 is maintained completely closed. This replacement was necessary as the solenoid valve offers an increased orifice for the refrigerant mass flow rate, which further reduces the flow resistance. In the future, a new three-way valve design offering lower resistance in heat mode is required to be implemented on the unit. The hot-gas line was sized based on the design capacity of the evaporator and the known requirement of the refrigerant gas velocity through the circuit that has to range between 5 to 10.16 m/s (Appendix I). Equation 5.1 was used to estimate the hot-gas line flow area [Dossat *et al.,* (2003); ASHRAE (1997)].

$$Velocity = C_{pv} \times M_f \times Pipe_{flow\,area} \qquad (5.1)$$

The value of the specific volume in this equation corresponds to the saturation temperature of the discharge gas. The mass flow rate of the refrigerant was calculated by dividing the capacity of the evaporator during full load operation by the change in enthalpy that the refrigerant experiences as it flows through the coil.

5.2.2 Experimental Results

Modification I was tested at the temperature conditions presented in Table 4.2 for both high-speed diesel and electric modes. The following results are analysed:

- The effect of test temperature conditions and unit operating mode on the heating capacity, efficiency and fuel consumption.
- Compressor behaviour including suction and discharge superheat.

The electronic throttling valve – ETV, [Item #6 in Figure 5.1(a)], was controlled automatically during tests between 160 to 400 steps, corresponding to 20% and 50% from the valve maximum operating range, to maintain various compressor suction pressures and as a result certain refrigerant mass flow rates. The compressor pressures are defined in imperial unit [psi] as the company and the manufacturer are based in United States and all technical specifications use the same unit system. Note that 1-psi equals 0.0689 bars. The system efficiency measured in [kW hr/l] was calculated as the ratio of the heating capacity and the fuel consumption of the unit for that specific test condition. It is also acknowledged that [l/hr (litres/hour] is not a standard International System (S.I.) unit for flow rate, but is used in preference to [l/s (litres/second)] in this case, since the fuel flow rate is in the order of $\frac{5}{3,600}$ [l/s].

A) Impact on Heating Capacity, Efficiency and Fuel Consumption

Figure 5.2 shows the results of a series of tests conducted at ambient temperature of –30°C and box temperature of +12°C with the SL 400e trailer unit operating in high-speed diesel mode. The ETV position was controlled from 22% to 38% of the maximum valve capability to determine the optimum unit response in heating.

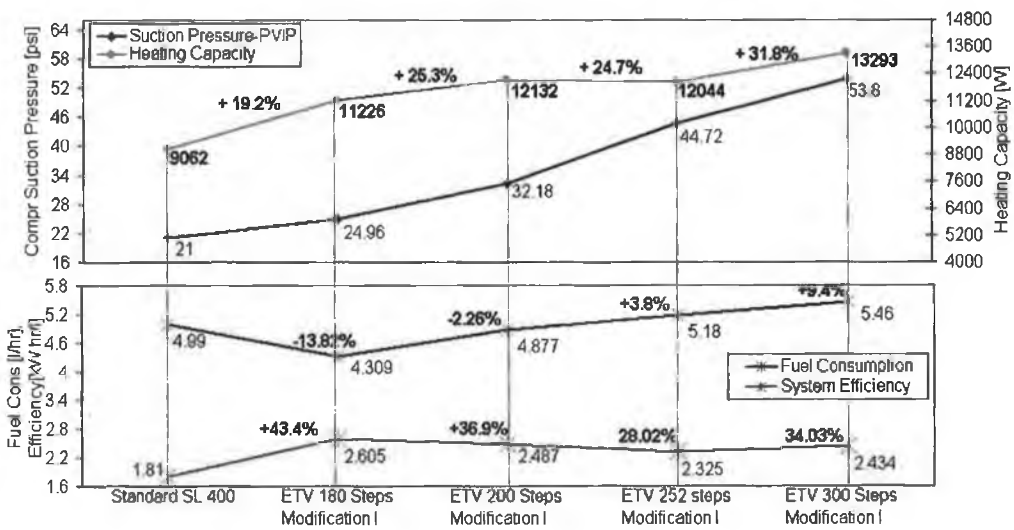
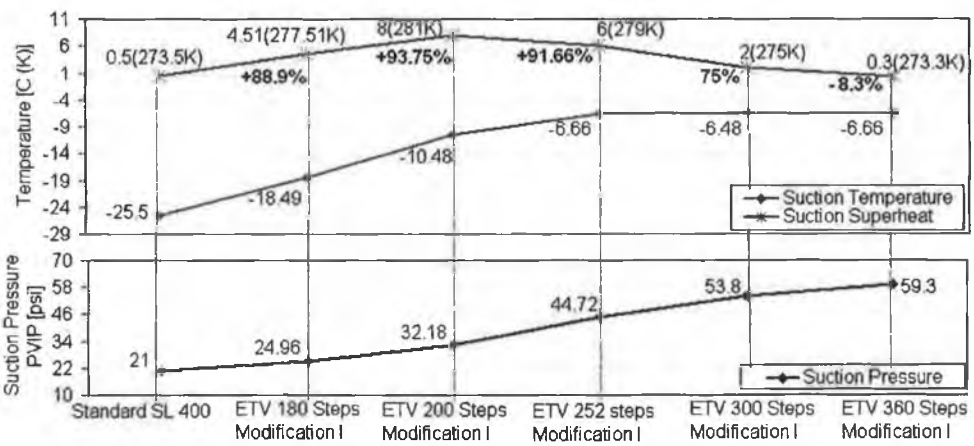


Figure 5.2 Heating Capacity versus fuel consumption and efficiency for Modification I at –30°C ambient and +12°C box temperature versus ETV position at: 180, 200, 252 and 300 steps.

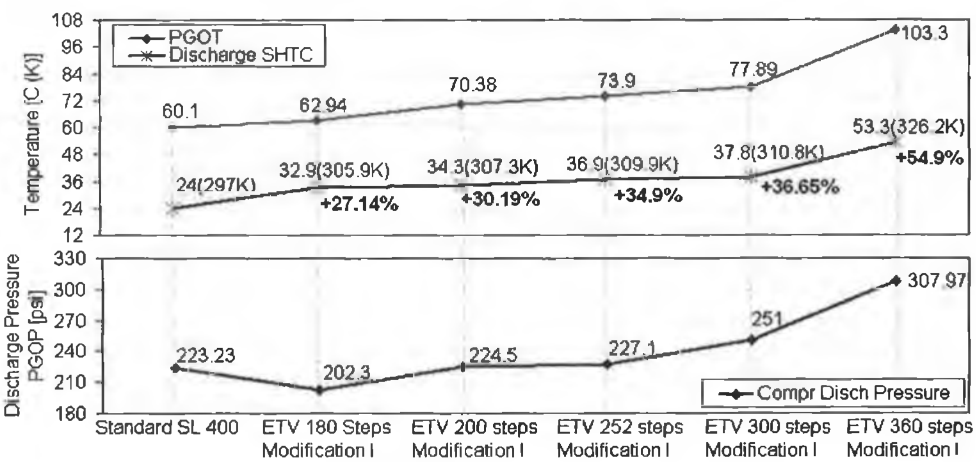
Figure 5.2 shows that the heating capacity is increased by up to +32%, or 4240W for 53.8-psi compressor suction pressure maintaining the ETV at 300 steps. However, the maximum system efficiency that is 43% higher than the standard unit figure is obtained at a lower compressor suction pressure of 24.9-psi, which requires the ETV to be controlled at 180 steps, corresponding with 22% opened. In this latest condition the modified unit operates with 14% lower fuel consumption than the standard SL 400e unit, while a significant 19.2% (2164 W) increased heating capacity is obtained. It is observed an optimum range for both fuel consumption and system efficiency at compressor suction pressure between 24 to 32-psi. For this test conditions the heating capacity of the modified SL 400e unit is increased by between 19% and 25%, corresponding to an additional heating capacity between 2,200W to 3,100W.

B) Compressor Behaviour

As presented in Chapter 4, Figure 4.16, the standard SL 400e unit operates at compressor suction superheat values below zero. As a result, these low superheat temperatures not only that can damage the compressor over time generating the high failure costs depicted in Figure 4.15 but also determine low system efficiency of only 1.8 [kW hr/l] compared with 3.5 [kW hr/l] figure obtained for the same unit when operates in cooling at the same fuel consumption rate [Table 4.1]. A comparison between the standard and modified SL 400e units regarding the compressor suction and discharge parameters is presented in Figure 5.3(a,b).



a) Compressor suction temperature, pressure and superheat.



b) Compressor discharge temperature, pressure and superheat.

Figure 5.3 Compressor behaviour for standard and modified SL 400e unit at ambient temperature –30°C and box temperature +12°C, operating in high-speed diesel mode.

Figure 5.3(a) shows a significant increase of almost 94% in compressor suction superheat for 32-psi suction pressure obtained maintaining the ETV at 200 steps. Due to a more rapid increase of the suction pressure and a slower increase of the suction temperature for ETV position higher than 252 steps, the compressor superheat decreases. It can be concluded that the optimum range is obtained for 23 to 44-psi suction pressure that determines increased compressor superheat by 89% to 94% and suction temperature by 7°C to 19°C.

Compared with the standard SL 400e unit, Figure 5.3(b) indicates that the modified system experiences the largest increase of almost 55% (or 29.3°C) in compressor discharge superheat, while the discharge temperature is higher by 43.2°C, for ETV controlled at 360 steps. Small variations are recorded for compressor discharge superheat due to slow changes of maximum 20-psi for the discharge pressure and of maximum 4°C for discharge temperature at a significant 50 steps difference in ETV position. Therefore, unlike the compressor suction parameters, the discharge temperature and superheat are not significant criteria for selecting the optimum operating range for the modified unit. As a result, based on Figures 5.2 and 5.3 the optimum response in heating of the modified unit tested is obtained at 180 steps ETV position and 24.5-psi compressor suction pressure when an increase of 19.2% in heating capacity is observed, while the fuel consumption is 13.8% lower resulting in an increase of the system efficiency by +43.4%. For a complete characterisation of the benefits obtained, the compressor behaviour is analysed together with the heating capacity, fuel consumption and system efficiency in Table 5.1.

From the total number of 55 tests performed to fully access the impact of Modification I on the unit's performance, Table 5.1 and Appendix J and K show the most representative selection of results for two different ambient temperatures of –30°C and –10°C and two box temperatures of +12°C and +21°C. The optimum condition obtained at ETV position 180 steps (22%) provides an increase in heating capacity of +19.2% and +34.3% respectively, while the fuel consumption is reduced by up to 14% when compared with the standard system. The unit efficiency is increased by 43.4% and 11.8% respectively. At –10°C ambient temperature Modification I has less impact on the heating capacity that experiences a +7.9% increase, while the unit has 2.7% lower fuel consumption and a +14% increase in system efficiency. However, these optimum operating conditions are obtained at a different compressor suction pressure of 48-psi that corresponds to 300 steps ETV position (Appendix J, Figure J.1).

Table 5.1. Impact of Modification I for SL 400e unit operating in high-speed diesel mode at ambient temperatures of –30°C, -10°C and box temperatures of +12°C, +21°C.

Unit	ETV	Heating Cap	Fuel Cons [l/h]	Efficiency [kW/l]	Suc SHTC [°C]/ [K]	Disch SHTC [°C]/ [K]
Standard Unit -30°C/ Box +12°C	195	9062 W	4.99	1.81	0.5 (273.5K)	24
	180	+19.2%	-13.8%	+43.4%	+88.9%	+27.14%
Modification I	200	+25.3%	-2.25%	+36.9%	+93.7%	+30.2%
Ambient –30°C	252	+24.7%	+3.8%	+28.02%	+91.6%	+34.9%
Box +12°C	300	+31.8%	+9.4%	+34.03%	+75%	+38.65%
Standard Unit -30°C/ Box +21°C	180	8510W	6.3	2.08	2.4 (275.4K)	20.74
Modification I	180	+34.3%	-18.09%	+11.86%	+71.3%	+34.6%
Ambient –30°C	200	+39.8%	-9.84%	+9.56%	+78.9%	+46.3%
Box +21°C	252	+43.6%	-17.6%	+24.08%	+60%	+52.6%
Standard Unit -10°C/ Box +12°C	210	12919	4.78	2.5	3.07 (276K)	7.14
Modification I	252	-5%	-6.22%	+8.8%	+53.7%	+83.02%
Ambient –10°C	300	+7.9%	-2.77%	+14%	+42.07%	+82.5%
Box +12°C	340	+19.7%	+0.41%	+16.8%	+6.57%	+85.95%

5.2.3 Discussion of Results – Modification I

The spectacular impact of Modification I on the unit heating capacity is based on the combined effect of several parameters:

- Reduced pressure drop on the hot-gas line. The proper sizing of the hot-gas line reduces the system pressure drop and as a result the compressor discharge pressure decreases. Therefore, a lower compression ratio, heat of compression and saturation temperature of the condenser is obtained, increasing the efficiency of the system.
- Higher refrigerant mass flow rate. The increased diameter of the hot-gas line ensures higher refrigerant flow rate through the evaporator, while the velocity is maintained to entrain oil for all loading conditions. Therefore, the heating capacity experiences a significant increase of +31.8% in ambient temperatures below –25°C. Less spectacular impact of +7.9% on the capacity is obtained at ambient temperatures higher than –10°C. However, it was the lower ambient temperatures that caused problems (Figure 4.11) and this modification targets the most significant problem area and provides a simple, yet effective solution.

- Lower compression ratio. Due to higher suction pressures and lower discharge pressures the modified unit has a lower compression ratio (R) as presented in Figure 5.4. This factor is calculated based on equation 5.2 and is defined as the ratio of the discharge pressure to the suction pressure measured in absolute units [Dossat *et al.*, (2003)].

$$R = \frac{P_{disch}}{P_{suc}} = \frac{PGOP}{PVIP} \qquad (5.2)$$

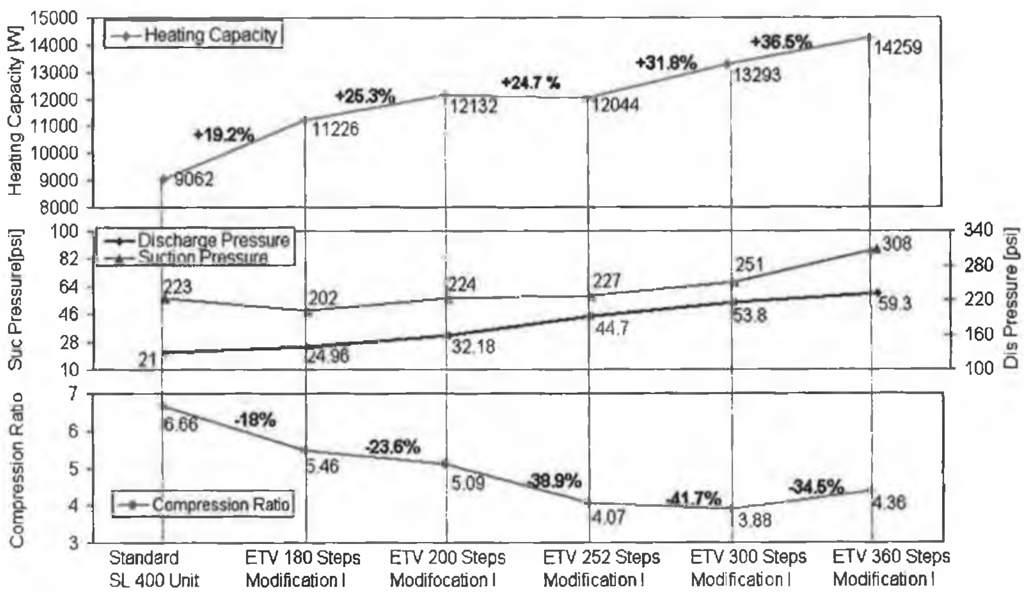


Figure 5.4 Variations in the unit's heating capacity with changes of the compression ratio.

5.3 DESIGN MODIFICATIONS II, III and IV

The effects of modifications II, III, and IV are compared in this section as all use high temperature hot-gas injections. Note that these design changes are integrated to the system that already has the increased diameter of the hot-gas line implemented. While the effect of hot-gas injection on the TTC unit cooling capacity has been established when the compressor operates at high ambient temperatures [Haas (2000); Hironari *et al.* (2000)] together with the influence on the refrigeration capacity of low temperature conditions [Diab *et al.* (1991)], no such research has been conducted on the heating performance at sub –10°C ambient temperature. It is well known that the compressor refrigeration performance and cooling cycle efficiency vary considerably with the system operating conditions [Dossat *et al.* (2003); Stoecker (2003)]. The most important factor governing the capacity of the unit is considered to be the vaporisation and condensing temperatures [Stoecker (2003)]. As the saturation temperature of the liquid in the evaporator increases, the pressure and the density of the suction vapour also increase. Therefore, each unit volume of vapour that is compressed has a greater mass. This characteristic yields a greater mass flow rate of refrigerant circulated by the compressor. However, while these principles are verified for the cooling operation mode [Hironari *et al.* (2000); Haas (2000); Stajic (1999); Yongchan *et al.* (2000)]] no such data is offered for the heating performance. Therefore, hot-gas injections are implemented on the modified SL 400e unit, not only to improve heating capacity but also to generate the necessary data to support a theoretical analysis of the unit at low ambient temperatures. Three different locations in the low-pressure side of the heat cycle were selected:

- Modification II – Hot-gas injection into the compressor suction line.
- Modification III – Hot-gas injection at the accumulator inlet.
- Modification IV – Hot-gas heat exchange coil installed into accumulator tank.

5.3.1 Description of Modifications II, III and IV

With Modification II identified in red, Figure 5.1(c) shows the new hot-gas injection into the compressor inlet. An on/off solenoid valve is also installed on the new 9 mm diameter circuit to isolate the injection from the low-pressure side of the cycle. Figure 5.1(d) shows Modification III that consists of a hot-gas injection at the accumulator tank inlet and Figure 5.1(e) shows a new hot-gas heat exchange coil installed within the accumulator. Both design changes extract the hot gas based on pressure difference between the high and low pressure sides of the system, through 9 mm diameter piping circuits that also have on/off solenoid valves. The main benefits are based on the effect of increased compressor suction temperature, lower compression ratios, increased evaporating temperature and refrigerant mass flow rate that are expected to impact positively on the compressor efficiency and system heating performance while operates in sub-zero ambient temperatures. A comparison between the results is presented in section 5.3.2. It is acknowledged that the hot-gas injections were tested with the unit that had modification I implemented [Figure 5.1].

5.3.2 Experimental Results

The results obtained for Modifications II, III and IV are presented in this section. The modified SL 400e unit was tested at the same temperature conditions from Table 4.2. The following results are analysed:

- Heating capacity, efficiency and fuel consumption.
- Compressor behaviour that includes the suction and discharge superheats.

A) Impact on heating capacity, efficiency and fuel consumption

Figure 5.5 shows the results of the tests conducted at ambient temperature of –30°C and box temperature of +12°C with the unit operating in high-speed diesel mode maintaining the ETV at 180 steps. The heating performance resulting from Modifications II and III is 8.4% and 7.2% higher respectively than the capacity measured after Modification I was included. The system efficiency also shows a further increase of 1% and 4%.

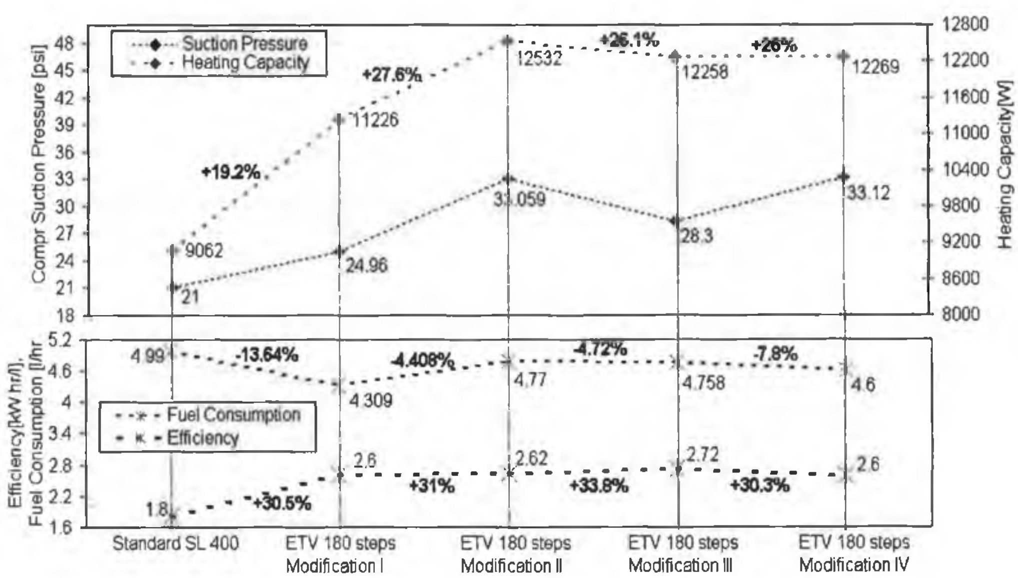
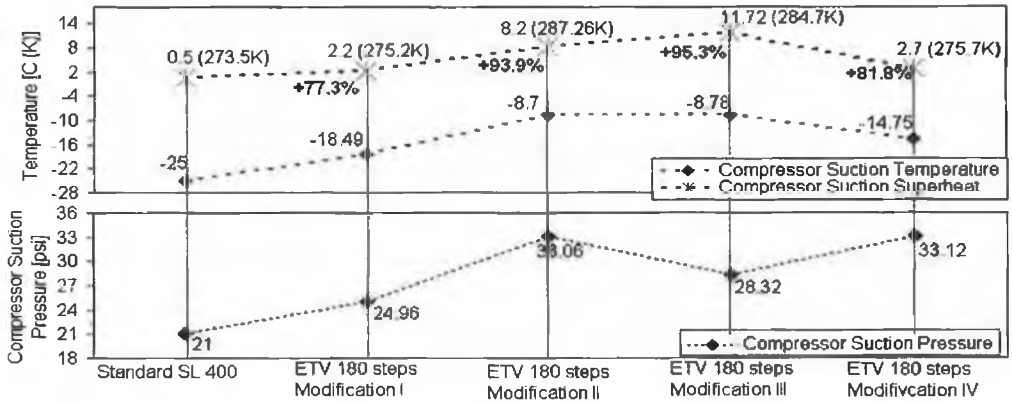


Figure 5.5 Heating Capacity, efficiency and fuel consumption at ambient temperature –30°C and box temperature +12°C, high-speed diesel mode as a result of Modifications I, II, III and IV.

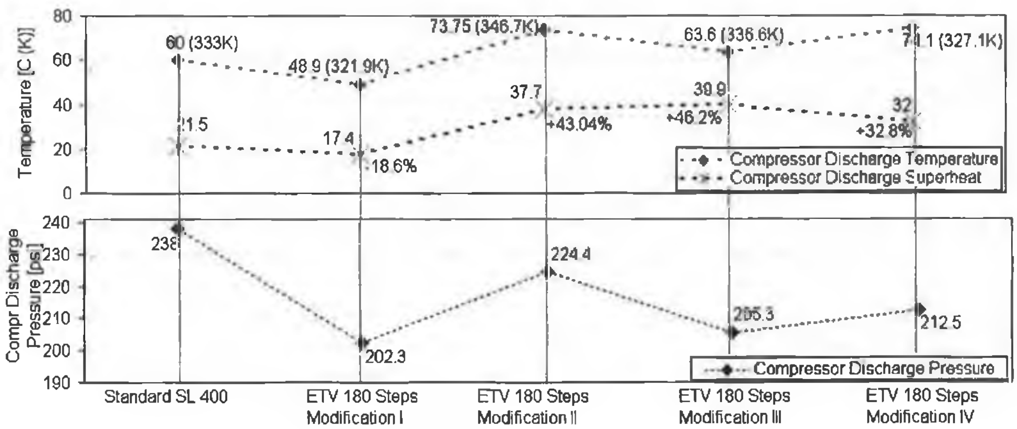
While the hot-gas coil into accumulator has similar impact with Modifications II, III on the heating capacity and 3% lower fuel consumption than the figures obtained for the other two hot-gas injections, no impact on the overall system efficiency is observed, compared with the figures obtained due to Modification I. Considering that at very low ambient temperatures, like –30°C, the reserve heating capacity of the standard SL 400e unit is at the limit (Figure 4.11), Modifications II and III provide a significant increase in the heating capacity of between 7% to 8.4%. These results correspond to an additional capacity of between 1032 W to 1306 W, compared with the unit that has the increased diameter of hot-gas line implemented. This additional heating capability has the potential to affect the food compartment/ calibrated box temperature by as much as another +12°C determined based on equation 3.1. Therefore, as a result of the hot-gas injection, the modified unit has the capability to provide a comfortable +27°C inside the trailer, while the maximum required transport set-point temperature for fresh products is +21°C [ATP (1970); ASHRAE (1997)]. Meanwhile the unit also operates with 4.4% lower fuel consumption and an efficiency that is up to 33% improved. This conclusion is verified for other similar data obtained at different ambient temperature of –20°C and box temperature of +2°C presented in Appendix J. Figure J.2 clearly shows that Modifications I and III offer the optimum behaviour of the modified unit by displaying 5.5% increased capacity and 17.5% higher efficiency obtained with a 12.4% lower fuel consumption than the standard unit. These are significant improvements in all aspects obtained with small changes in the system design and low implementation costs of 10 Euros for each modification, corresponding to only 0.25% increase of the SL 400e unit total market cost [Thermo King (2004)].

B) Compressor Behaviour

Compressor behaviour in the same test temperature conditions is overviewed in Figure 5.6(a,b). While small differences of only 3% are obtained between the suction superheat figures for modifications II and III, it can be observed that an average of +16.5% increase of the suction superheat results from the hot-gas injections versus that generated by Modification I. Figure 5.6(b) shows the impact on the compressor discharge parameters of the hot gas injections compared with the increased hot gas line effect. Modifications II and III provide higher discharge superheat by 25% and 28% respectively, while less impact of only +15% is obtained for modification IV. The necessity of the hot-gas injections during heat mode operation is based not only on increased heating capacity, but also on the need for higher compressor suction superheat. Figures 5.6(a) and Appendix J (Figure J.3) show that while the standard SL 400e unit has +0.5°C superheat, even after the impact of Modification I, the suction superheat is still just +2.2°C at –30°C ambient temperature, and +1.71°C at –20°C ambient. As a result of the hot-gas injection effect an optimum +8.2°C and +11.7°C values for compressor suction superheat are obtained.



a) Compressor suction temperature, pressure and superheat.



b) Compressor discharge temperature, pressure and superheat.

Figure 5.6 Compressor behaviour for the modified SL 400e at –30°C ambient and +12°C box temperature in high-speed diesel mode, for Modifications I, II, III and IV.

Figures 5.5 and 5.6 together with the data presented in Appendix J shows that the optimum effect on the heating behaviour of the modified unit is obtained using Modifications II with I or III with I as a result of direct hot gas injections at the compressor and accumulator inlet. A comparison with the results obtained with those from the standard SL 400e unit show an improved heating response of 26.1% increase in capacity, 33.8% higher efficiency and a reliable 11°C increase in compressor suction superheat for the modified unit operating with 4.7% lower fuel consumption. These results are based on the combined impact of modifications I and III, considered the optimum combination between the effects of higher refrigerant mass flow and hot gas injection.

From the total number of 150 tests conducted for a complete description of the effects of modifications based on hot gas injections, Table 5.2 shows the most representative results compared with the impact of Modification I at the following conditions: i) ambient of –30°C and box of +12°C maintaining the ETV at 200 steps, ii) ambient temperature of –30°C and box of +2°C controlling the ETV at 252 steps and iii) ambient –10°C and box temperature of +12°C maintaining the ETV at 340 steps. Supplementary results are also presented for a different ambient temperature of –20°C in Appendix J. Samples of the primary Excel row data recorded during the tests are presented in Appendix K.

Table 5.2. Comparison between Modifications II, III and IV relative to Modification I, while the unit operates in high-speed diesel mode. The fuel consumption is compared with the standard SL 400e unit. The ETV position was 200, 252, 340 steps for temperature conditions –30°C/+12°C, -30°C/+2°C and –10°C/+12°C respectively.

Modification	Test Conditions Amb/Box	Heating Capacity [%]	Fuel Cons [%]	System Efficiency [KW hr/l]	Suction SHTC [%]	Discharge SHTC [%]
I) Increased Hot Gas Line	-30°C/12°C	12132W/+25%	-2.26%	2.48/+20%	14.3°C/287K	43.39°C
	-10°C/12°C	12674W/+6%	+0.7%	2.6/+0.6%	11.7°C/284K	52.52°C
	-30°C/2°C	10945W/+6%	-25.1%	2.5/+42%	4.34°C/277K	21.3°C
II) Injection Compressor Suction Line	-30°C/12°C	+1.2%	-2.24%	+3%	-61%/5.5°C	-0.02%
	-10°C/12°C	+5%	+3.1%	+4%	-23.9%/+8.9C	-41.9%
	-30°C/2°C	+2.4%	-25%	+5.5%	+4.6%	-8%/+19C
III) Injection Inlet Accumulator	-30°C/12°C	+3.4%	-2.74%	+8%	-28%/10°C	+7.08%
	-10°C/12°C	+4.5%	+2.9%	+4.8%	-49.2%/+5.9C	+15.9%
	-30°C/2°C	+7.4%	-20.4%	+2%	+48%/+8.3C	+30.8%
IV) Hot Gas Coil into Accumulator	-30°C/12°C	+1.5%	-3.1%	+6.3%	-45.8%/7°C	-9.8%
	-10°C/12°C	+4.94%	+1.7%	+3.6%	-34.3%/+7.7C	+26.7%
	-30°C/2°C	+2.6%	-23.2%	+6.2%	+8.4%	+44.5%

While Figures 5.5 and 5.6 show the results obtained while maintaining the ETV at 180 steps, the unit behaviour is slightly different at 200 steps. Even if the hot-gas injections have less impact on the heating capacity of only +3.4% due to modification III, better response is obtained for: i) fuel consumption that decreases by 2.7% below that of Modification I and ii) higher impact on the system efficiency up to +8% than the value obtained for ETV position at 180 steps. This example was chosen to highlight that taking into account the desired heating behaviour of the modified unit the ETV can also be used to generate increased heating capacity or higher system efficiency. Therefore, even if at a different box temperature of +2°C and same ambient of –30°C the optimum balance between the analysed parameters is obtained for 200 steps, another situation is presented when the ETV is controlled at

252 steps. Good impact on the heating capacity of 7.4% is recorded for these conditions but with a small negative effect on the fuel consumption that is 2% higher. The unit displays 8% higher efficiency compared with Modification I. While at –30°C ambient temperature the attention is concentrated on a good heating performance, maintaining the ETV at 252 steps is preferred as the optimum solution corresponding to increased heating capacity and a more efficient unit. At higher ambient temperature like –10°C the effect of the hot gas injections provides an average 5% increase capacity obtained with a negative impact of +3% increase in fuel consumption. A unique situation appears for this higher ambient temperature. While the compressor suction superheat was +11.7°C as a result of Modification I influence, it can be noticed that the hot-gas injections are lowering this parameter at the optimum unit operating superheat of +8°C as concluded in Table 5.4 based on Figure 5.4. A positive impact of +15% and +44% higher compressor discharge superheat is obtained due to the beneficial influence of modifications III and IV.

Based on the results presented it is observed that the hot-gas injections have a very good influence on the compressor behaviour, while a good increase of an average +5% is also obtained for the heating performance compared with Modification I. The optimum unit behaviour in heating is obtained as a result of modifications II or III. The advantage of lower fuel consumption figures resulted from modification IV influence is diminished by less impact on the heating performance than the other two direct hot-gas injection modifications. A significant advantage of the direct hot-gas injections is the flexibility offered during unit operation in heat mode, which results from the possibility to select the desired unit response of either increased capacity or less fuel consumption depending on the temperature conditions. As an example, based on knowledge of insufficient heating capacity from the standard unit at ambient temperatures like –30°C and box temperatures higher than +4°C (Figure 4.9), modifications II and III are highly appropriate as they provide up to 9% increase in the heating capacity, while the unit is still operating with a 4.4% lower fuel consumption than the standard SL 400e. It can be concluded that significant improvements in all aspects were obtained with small changes in the system design and low implementation costs. The optimum response of the modified SL 400e unit is obtained due to the combined impact of Modifications I and III.

5.3.3 Summary of Results – Modifications II, III and IV

A theoretical discussion of the main parameter influences as a result of controlled variations on the heating performance of the modified unit based on the hot-gas injection technique used by modifications II, III and IV is presented.

A) Modification II – Hot gas injection into the compressor suction line

The good behaviour of the unit in heating obtained with this design modification is based on the following effects:

- Increase in compressor suction temperature. As a result of the hot-gas injected directly in the compressor suction line, higher suction temperature is obtained together with increased mass flow rate (Figure 5.6), which has significant impact on the heating capacity effect per unit mass of refrigerant circulated through the evaporator (Figure 5.5). When the suction temperature increases while the condensing temperature remains constant, the compression ratio and the compression work per unit mass decreases (Figure 5.4).
- Lower compression ratio. It is considered that the good behaviour in heating obtained with this design modification is based on increased compressor suction superheat and lower compression ratios. The results presented show that the optimum balance between the compression ratio and compressor suction superheat during heat mode operation is obtained at 5 and 8°C respectively (Figure 5.4).

B) Improvement III – Hot-gas injection into the accumulator inlet circuit

This improvement has the optimum effect on the TTC unit behaviour in heat mode for all test conditions. The hot-gas injection at the accumulator inlet is the only modification that offers positive combined effect of the following parameters:

- Increased evaporative saturation temperature.
- Higher vapor mass flow rate obtained in the accumulator.
- Increased compressor suction temperature.
- Lower compression ratios.

The hot-gas injection at the accumulator inlet has direct effect on the evaporating temperature and on the increase of the liquid refrigerant evaporated in the accumulator. When the saturation temperature of the refrigerant in the accumulator increases higher compressor suction temperatures are obtained and the compression ratio decreases, improving the volumetric efficiency. Figure 5.7 shows the effect of the evaporative temperature changes on the heating capacity.

C) Modification IV – Hot gas coil installed in the accumulator

This design modification has a direct effect on the evaporating temperature in the accumulator. Due to the heat exchange that takes place in this component, the saturation temperature of the refrigerant in the accumulator is higher. The hot-gas coil in the accumulator has less impact on the unit performance due to the fact that the mass flow rate is lower than that obtained with modification III, where the hot-gas is injected directly and a liquid-vapor mixture flows into the accumulator resulting in increased refrigerant vapor mass flow rate. The main advantage of Modification IV consists in lower fuel consumption that also decreases the compressor power requirements compared with the effects of the other two modifications based on hot-gas injections.

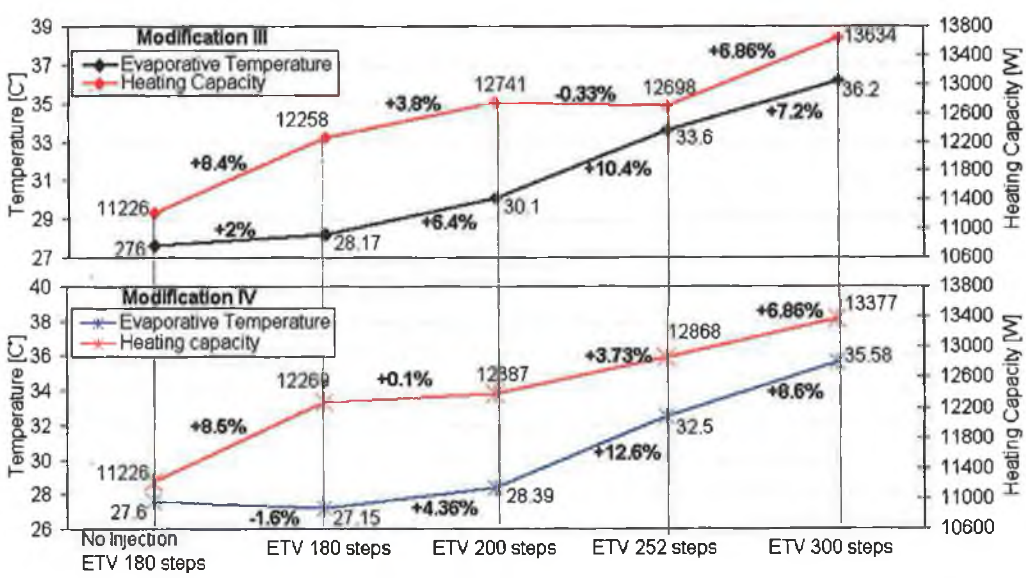


Figure 5.7 Heating Capacity versus Evaporative Temperature for Modifications III and IV.

Based on the experimental data presented in this section and in Appendix J it can be concluded that the combined effect of Modification I and III have the optimum impact on the heating capacity, system efficiency, fuel consumption and on the compressor suction superheat.

5.4 MODIFICATION V

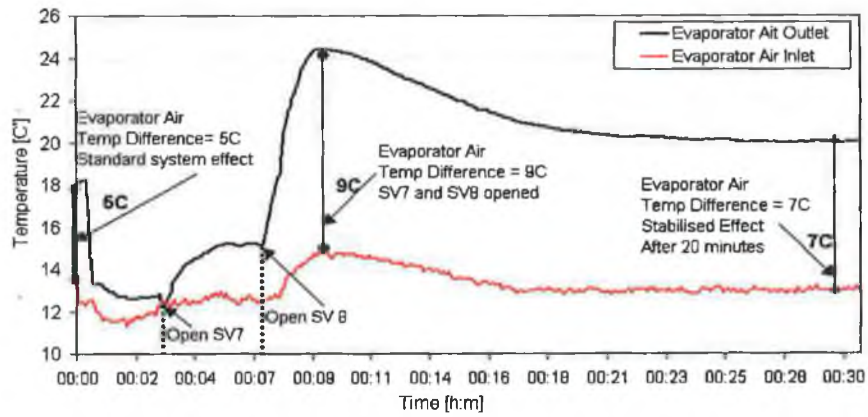
When the SL 400e standard unit operates alternatively in Cool and Heat Modes during temperature control of the air in the calibrated box/ food compartment, the heating capacity of the unit can suffer changes of up to ±15% due to refrigerant charge trapped in the cooling cycle, as presented in Chapter 4, Figure 4.14. To overcome this negative influence on the unit performance and to improve both heating and defrost response the modified SL 400e unit was equipped with a new system capable to recuperate the charge from the receiver tank and condenser into the low-pressure side of the heat cycle. Modification V [Figure 5.1(f)] is described in section 5.4.1, while the results obtained are presented in section 5.4.2.

5.4.1 Description of Modification V

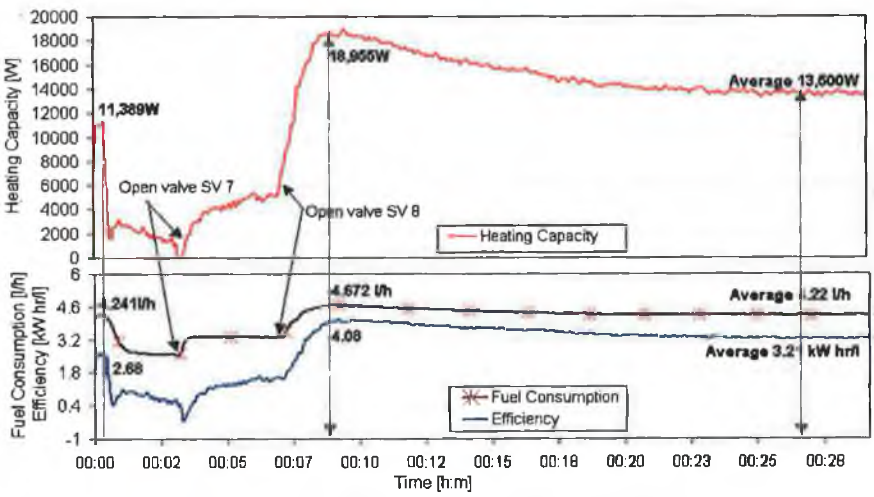
Figure 5.1(a) shows the existing charge control system installed on the standard SL 400e that uses a portion of the high-pressure discharge vapour to pressurise the receiver tank (9). As a result, liquid refrigerant should be pushed from the tank and liquid line through a bleed in the expansion valve, back to the evaporator for increased heating capacity. According to the data presented in Figure 4.12, this system provides poor charge migration control. Modification V introduces a new system capable of completely recovering the refrigerant charge trapped mainly in the receiver tank (9) and condenser (10) as indicated in Figure 5.1(f). This design modification consists of two 9 mm diameter circuits installed from the condenser and receiver tank at the accumulator inlet, with dedicated solenoid on/off valves (SV 7 and SV 8). While the charge trapped has 120-psi, due to pressure difference with the low-pressure side of only 40 psi, the charge migrates to the accumulator inlet, thereby increasing the evaporating temperature and the mass flow rate.

5.4.2 Experimental Results

Figure 5.8(a,b) shows the effect of both the standard (Figure 5.1(a)) and modified (Figure 5.1(f)) control circuits on the evaporator temperature difference, heating capacity, fuel consumption and efficiency. While the standard unit has 5°C air temperature difference, after opening both valves SV 7 and SV 8 the temperature difference increases up to 9°C, which stabilises at 7°C after 20 minutes. Figure 5.8(b) shows the effect of both charge systems on the heating capacity, system efficiency and fuel consumption for the same ambient temperature of –20°C and box temperature of +12°C. If the standard system provided a 5°C difference for the air inlet–outlet temperatures, which correspond to 11,389W heating capacity, after opening both circuits and recuperating the charge from the condenser respectively from the receiver tank, the new air temperature difference on the evaporator is 7°C corresponding to an important increase of 2,111W in the heating capacity. While the final average fuel consumption remains effectively unchanged from that of the standard unit, better results are obtained for the system efficiency, which is increased by 16.5% from 2.68 kW/l to an average of 3.21 kW/l, due to higher heating capacity obtained at the same unit fuel consumption.



a) Temperature difference of air across the evaporator.

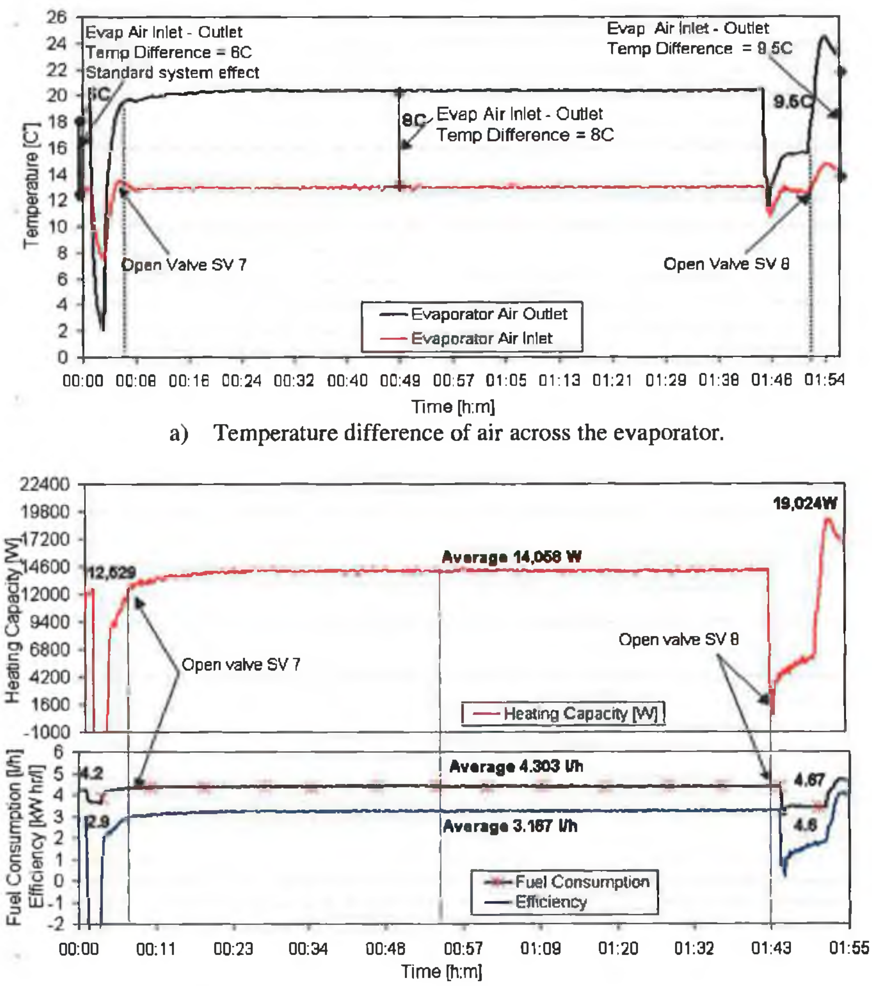


b) Heating Capacity, fuel consumption and efficiency.

Figure 5.8 Results of charge migration control modification on the unit behaviour in heat mode at an ambient temperature of –20°C and box temperature of +12°C.

The tests were repeated four times but from different starting conditions as described in Table 4.6. All test results show the same final heating capacity of 13,500W, small variations of ±30W indicating the excellent repeatability of the new charge control system. Not only that modification V improves the heating performance of the standard unit but Figure 5.8(b) also shows the positive impact of the liquid refrigerant charge recuperated that boosts the heating capacity by additional 7566 W in a short period of time of only 5 minutes after SV 7 and SV 8 valves are opened. As a result, the defrost pull-up time calculated based on the Curve Fitting and Euler methods is estimated to be with 30% shorter [Jobson (1991); Weisstein (2004)].

Figure 5.9(a,b) shows the effect of the liquid refrigerant charge recuperated only from the receiver tank at a different ambient temperature of –10°C.



a) Temperature difference of air across the evaporator.

b) Heating capacity, fuel consumption and efficiency.

Figure 5.9 Results of refrigerant charge recuperated from receiver tank on the unit behaviour in heat mode at –10°C ambient temperature and +12°C box temperature.

Figure 5.9(a) shows the effect of the charge migration in the heating cycle only from the tank (9) by opening valve SV7 while valve SV 8 is maintained closed. While the standard charge control system installed on the standard SL 400e unit (Figure 5.1(a)) was able to provide a 6°C air temperature difference on the evaporator, the new circuit that recuperates the charge from the receiver tank provides increase air inlet-outlet temperature difference of 8°C corresponding with 1,500W higher heating capability. The behaviour of the modified unit due to the additional charge was recorded for what can be considered as a long period of time, almost 2 hours, which indicates the stability of the system. Figure 5.9(b) shows the effect of the charge recuperated from the receiver tank on the heating capacity, fuel consumption and system efficiency. While the standard charge control circuit provided a 12,259W heating capacity, an increase of almost 17.5%, or 1,500W, was obtained after the SV 7 valve was opened and the liquid charge from the tank (9) started to return slowly at the inlet of accumulator, to the pressure difference. While the system efficiency increases with 0.3 kW/l, no significant differences can be observed in the fuel consumption figures for the standard versus the modified SL 400e units.

5.4.1 Discussion of Results – Modification V

Table 5.3 shows the benefits of modification V based on tests conducted at –20°C and –10°C ambient for the same +12°C box temperature. The heating figures for the modified unit are compared with the effect of the charge control system installed on the standard SL 400e unit.

Table 5.3. The effect of the charge recuperated from the cooling cycle on the system performance.

Charge Control Circuit	Test Conditions	Evap air ΔT [C]	Heating Cap [W]	Fuel Cons[l/h]	Efficiency
Existing Circuit on Standard SL 400e Unit	-20°C/+12°C	5°C	11389	4.24	2.68
	-10°C/+12°C	6°C	12529	4.208	2.982
Recuperate charge from condenser and receiver	-20°C/+12°C	7°C	+15.6%	-0.47%	+16.5%
Recuperate charge from receiver tank only	-10°C/+12°C	8°C	+15.67%	+2.2%	+5.84%

Figure 5.8 and Table 5.3 show that the charge recuperated from both receiver tank and condenser provide a significant impact greater impact on the system efficiency than increase by 16.5%, while the heating capacity is higher by 15.6%. While the

standard SL 400e unit experiences variations within 4% to 15% in the capacity due to poor control of the refrigerant charge, Modification V increases both heating capacity and ensures a more stable performance.

5.5 HEATING BEHAVIOUR OF THE MODIFIED SL 400e UNIT

The problems that appear in heat mode for the standard SL 400e unit were overviewed in Chapter 4 based on heating capacity, temperature control and pull-up tests conducted at the temperatures presented in Table 4.2. As a result, five types of insufficiencies for heat mode were emphasised and summarised as follows:

- Low or insufficient heating capacity.
- Capacity variations for the same temperature test conditions.
- Low compressor suction and discharge superheat.
- Long pull-up time.
- Poor set-point temperature control.

As a result of the system deficiencies identified five design modifications were defined and accessed in this chapter not only to investigate unit behaviour in heat mode but also to verify and define for the first time the influence of parameters such as compressor suction temperature, superheat, evaporating and condensing temperatures on system performance in heat mode.

5.5.1 Improved heating behaviour of the modified SL 400e unit

An overview of the problems highlighted for the standard SL 400e unit that are completely eliminated by the modified SL 400e unit is presented in this section.

A) Heating Capacity

In section 4.5.1 it was concluded that while the unit develops adequate heating capacity in high-speed diesel mode, the electric mode lacked the heating capacity to achieve box temperatures higher than +4°C for sub –20°C ambient temperatures. Figure 5.10 summarises the increased heating capacities obtained using modifications I, II, III and IV. The standard SL 400e unit with just 1,336W of reserve capacity was only capable of providing a maximum air temperature of 8.9°C in the food compartment. To confirm this lack of performance, Figure 4.11 shows a deficit of –4.66% in the reserve heating capacity of the standard unit at +12°C box temperature. Modifications I and III provide a solution that boosts heating capacity by 31.3%, corresponding to 2,734W. A higher reserve heating capacity of 67% is also possible, which translates into a food compartment temperature of +19.3°C. As a result, the modified unit has increased its capacity to maintain air temperatures within the trailer from +12°C to +21.3°C, while operating in electric mode.

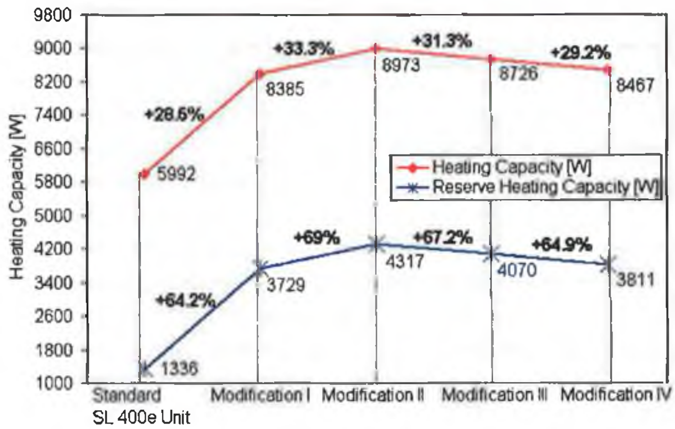


Figure 5.10 Heating and reserve capacities for standard versus modified SL 400e unit operating in Electric Mode at –30°C ambient temperature and +2°C box temperature.

B) Refrigerant Charge Control System

The amount of liquid refrigerant trapped in the cooling side of the cycle can generate heating capacity variations of up to 15%, as indicated in Figure 4.14. Modification V completely recuperates the liquid refrigerant charge from the condenser (10) and receiver tank (9) into the low-pressure side [Figure 5.1(f)]. This not only works to increase heating capacity by +15% but it also has a great impact on stabilising unit performance. Figure 5.8 clearly shows the impact of this modification, which boosts the unit heating performance by 40% before the final heating performance is stabilised. This impact of charge control also takes place in a short period of time of only 10 minutes, which has significant impact on the defrost performance of the unit.

C) Compressor Behaviour

Figure 4.16 shows –6°C and –13°C compressor suction superheat at ambient temperatures of –10°C and –30°C respectively, while the box temperature is maintained at the same +12°C. A suction vapour superheat below zero degrees

Celsius suggests the presence of liquid in the compressor inlet, which is potentially dangerous for compressor life. After many hours operating in these conditions, high losses up to 109 thousand Euros per year that is equivalent with the cost of 37 new units are claimed as a result of compressor components damage due to the liquid trapped as presented in Figure 4.15 [Thermo King (2004)]. However, this deficiency in the standard unit design was totally eliminated due modifications I to IV, which resulted in an increase of up to 90% in the suction superheat.

D) Pull-Up Time

Figure 4.12 highlights a pull-up time of 2.5 hours for high-speed diesel mode and 8 hours in electric mode when operating between a –30°C ambient and a +25°C box temperature. Due to a significant increase of +20% obtained in the heating capacity of the modified unit, the pull-up time is reduced considerably as indicated in Figure 5.11. It can be observed that the modified system reaches the box set-point of +25°C in 1 hour less time for diesel mode while for electric mode the pull-up time is reduced with 3.2 hours, a reduction of approximately 30% on average.

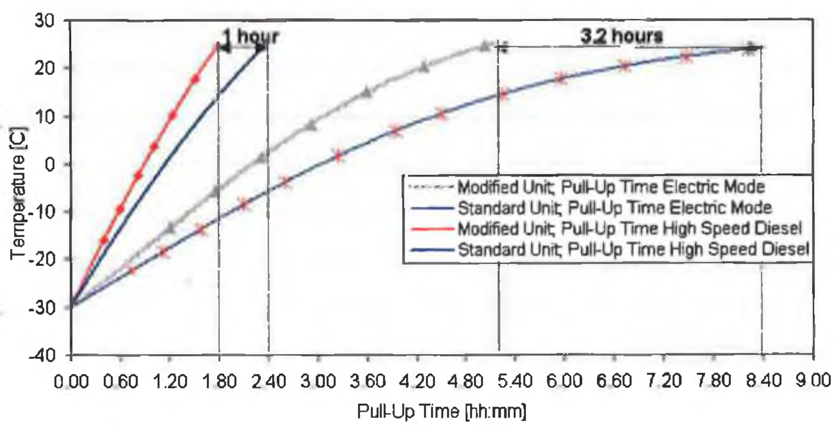


Figure 5.11 Comparison between pull-up times for both standard and modified SL 400e systems. high-speed diesel and electric modes.

5.5.2 Theoretical Considerations for Cool versus Heat Mode

While a considerable body of work has been performed on the effects of different factors such as compressor suction temperature, superheat, evaporating and condensing temperatures on the cooling performance [Dossat *et al.,* (2003); Stajic (1999), Sang-Ho *et al.*, (2000); Xin *et al.*, (2000), Yun-Hee *et al.*, (2000)], the influence of these parameters are studied for the first time in heating. Comparisons of the most important changes that occur in the TTC unit behaviour for cooling mode based on major findings in literature and for heat mode based on the experimental results presented in this work are summarised in Table 5.4.

Table 5.4. Influence of parameter variations on the heating and cooling performance.

Parameter	Measured impact on Heating Capacity	Reported impact on Cooling Mode	
		Major Findings	Authors
1. Increase of compressor suction temperature associated with an increase in the suction superheat	An increase of 0.73% in the heating effect per +1°C [Figure 5.5, Modification I and III]	An increase of +0.31% in the cooling effect per +1°C	Dossat *et al.*, (2003); Sang-Ho *et al.*, (2000)
	An increase of 0.72% in the heating effect per +1°C [Figures 5.5, 5.6 and Appendix H]	Decrease in the cooling capacity by -0.46% per +1°C	Rongchang *et al.*, 2000 Yu-Choung *et al.*, (2000)
2. Increase in compressor suction temperature associated with lower compression ratios	An increase of +3.36% in the heating effect per +1°C [Figure 5.4; 180 steps]	An increase of +0.95% in the cooling effect per +1°C	Dossat *et al.*, (2003); Stoecker (2003)
3. Compression Ratio and compressor suction superheat optimum balance	Optimum results for 5 compression ratio and +8°C superheat [Figure 5.4]	Optimum results for 6 compression ratio and +16.5°C superheat	Stajic (1999)
4. Increase in evaporating temperature	An increase of 4.38% per +1°C [Fig. 5.7; 180steps]	An increase of 2.9% per +1°C	TTC certification (2004) Meurer (2000)
5. Increase in the condensing temperature	An increase of 0.35% per +1°C [Figure 4.7(b)]	An increase of +2% per +1°C	Dossat *et al.*, (2003) Xin *et al.*, (2000)

The selected research for cooling behaviour was conducted for the same type of TTC unit equipped with a similar four-cylinder reciprocating compressor. Generally though, the results presented in Table 5.4 indicate that cooling performance of a TTC unit is greatly influenced by the condensing temperature showing variations of up to +2% per +1°C [Dossat *et al.*, (2003); Xin *et al.*, (2000)]. However, the heating capacity has small variations of only +0.35% per +1°C increases in the condensing temperature [Figure 4.7(b)]. Meanwhile, variations in the evaporating temperature has a significant impact on the cooling capacity that changes by +2.9% per +1°C [TTC unit certification (2004)], the influence on the heating capacity is much greater, changing by +4.38% per +1°C change in evaporating temperature [Figure 5.7]. A great impact on the heating performance increased by +3.36% per +1°C is obtained due to the influence of the compressor suction temperature and superheat associated with higher evaporating temperatures and lower compression ratios. While it is well known that the evaporating and condensing temperatures are the most important parameters that influence the cooling capacity, the test results presented highlight for the first time that the parameters that improve significantly the heating capacity of a TTC unit are increased evaporating temperature together with an optimum compressor suction temperature of +8°C associated with compression ratios of 5.

While the general thermodynamic cycle was overviewed in Figure 1.4, a comparison between the thermodynamic cycles of the standard and modified SL 400 unit based on experimental results is presented in Figure 5.12. It can be noticed that the significant improvement in the Heating capacity is obtained as a result of an increase in the compressor discharge superheat (Figure 5.3). The improved refrigeration effect can be explained due to higher evaporating temperature and pressure combined with higher compressor suction superheat.

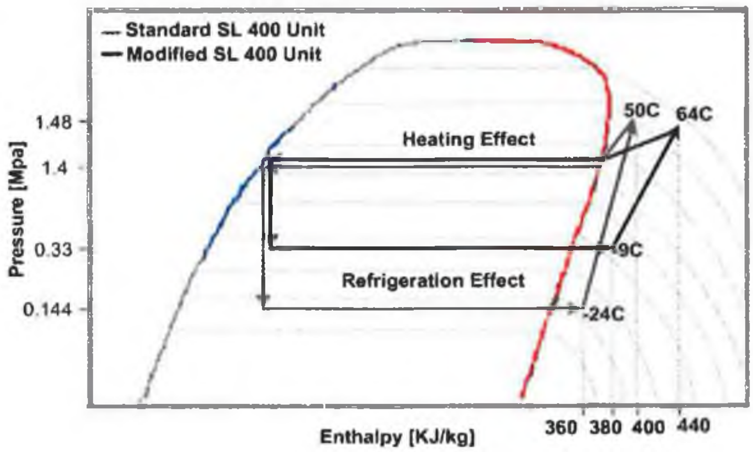


Figure 5.12. Thermodynamic cycle for standard versus modified TTC unit. Amb –30°C, Box +2°C.

5.6 SUMMARY

A comparison of the standard and modified SL 400 units has shown that each of the five design modifications proposed has yielded a significant improvement in the heating performance. The impact of the design changes has been discussed based on the combined effect of:

- Increased evaporative and compressor suction temperatures.
- Lower compression ratios.
- Higher mass flow rates.

Simple yet viable solutions are proposed for the first time to address the performance weaknesses identified in the field data presented in Chapters 1 and 2 and 5.

CHAPTER 6

CHARACTERISATION OF THE SINGLE - COMPARTMENT STANDARD TS 500 TRUCK UNIT IN HEAT MODE

CONTENT:

Chapters 4 and 5 focused on the analysis of a key trailer-mounted TTC unit, the SL 400e system. Chapters 6 and 7 describe a complete system characterisation of the most common single-compartment truck-mounted TTC unit, the TS 500 type, during heat mode operation. This latest type of truck unit has 60% less heating capability than the trailer SL 400e unit. Chapter 6 quantifies for the first time the heating performance together with the problems that appear during heat mode operation for this type of unit through both field and experimental data. The in-house tests heating capacity results are measured based on the EHI method and test facility presented in Chapter 3. The analysis of the unit behaviour is extended through the predictions of the first mathematical model for heat mode operation overviewed in Chapter 7.

6.1 INTRODUCTION

While Chapters 4 and 5 present the first complete characterisation of a single-compartment trailer TTC unit in heat mode together with design modifications implemented to optimise the unit performance, this chapter investigates the behaviour of a standard diesel engine driven single-compartment TS 500 truck unit in heating [Figure 6.1]. The main objectives are to:

- Use the new EHI test method and facility described in Chapter 3 (Figure 3.3) to determine the first accurate measurements of heating capacity under a wide range of test conditions.
- Quantify the problems that appear during heat mode operation at ambient temperatures below –10°C, for this truck TTC unit that has 60% less heating and cooling capacity than the SL 400e trailer unit.

Figure 6.1 The TS 500 single-compartment TTC truck unit.

6.1.1 TS Line Single-Compartment Truck Units

The most common single-compartment truck units that operate worldwide are the Thermo King TS system line. Four different unit types are available on the market: TS 200, TS 300, TS 500 and TS 600 units and a technical specification for each type is presented in Table 6.1. While Thermo King data sheets specify cooling capacity at 0°C and –20°C no such data is available for heating capacity.

Table 6.1. Technical specifications of TS line of TTC truck units [Thermo King web site].

TS Unit type	Cooling Capacity at 0°C	Engine	Compressor
TS 200	5500	TK 3.74 diesel engine, 740cm^3 displacement	TKO 2.9kW, scroll, 65cm^3 displacement
TS 300	7250	TK 3.74 diesel engine, 740cm^3 displacement	TKO 2.9kW, scroll, 65cm^3 displacement
TS 500	8400	TK 3.95 diesel engine, 950 cm^3 displacement	TKO 4.5 kW, scroll, 98cm^3 displacement
TS 600	9700	TK 3.95 diesel engine, 950 cm^3 displacement	TKO 4.5 kW, scroll, 98cm^3 displacement

The worldwide sales of the TS line during 2003 presented in Figure 6.2 indicate that the TS 300 and the TS 500 are the most popular types from this production line, with the highest sales percentage of 28% each [Thermo King Market Report (2004)].

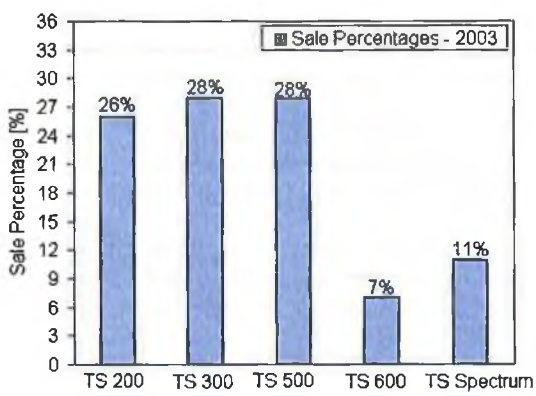


Figure 6.2 Worldwide Sale Percentages of the TS Truck Units in 2003 [Thermo King 2004].

The TS 500 unit was selected for analysis based on its important position in the market and the significant problems highlighted by field data obtained from Sweden (Figure 2.11, 6.4 and 6.5). The TS 500 unit is usually employed as a single-compartment truck unit for journeys of 300 km, with truck bodies of 7m length, thermally insulated using 100mm thickness polyurethane foam [ATP (1970)].

6.1.2 General Description of the TS 500 Unit

The TS 500 unit shown in Figure 6.1 is designed for rigid body trucks and is usually mounted to the temperature controlled food compartment above the drivers cab. An overview of the main system components together with the principle of operation in both cool and heat modes for a typical TTC unit was provided in Chapter 1, section 1.3.2. However, a more detailed schematic of the TS 500 unit is presented in Figure 6.3. This unit has a discharge pressure regulator valve (4) used only in heat mode when the scroll compressor does not provide high discharge pressure. In these conditions, the discharge regulator controls the discharge pressure of the system at 220-psi. Other technical specifications for the main components are:

i) Compressor (1): The unit is equipped with a 950 cm^3 scroll compressor, model TKO 4.5kW, Thermo King fabrication.

ii) Evaporator (5): This component is a finned-tube cross flow heat exchanger with one fluid mixed (air) and one unmixed (refrigerant). It has 11 copper circuits and 77 tubes with 0.375 mm external diameter, distributed on 7 rows. The primary heat exchange area of the copper tubes is 2.64 m^2, while the secondary area of the fins is 24.33 m^2. A fan delivering 3500 m^3/h in high-speed diesel and 1650 m^3/h in low speed diesel draws compartment air through the evaporator coil.

iii) Accumulator Tank (6): This tank, which is 0.5 m height and has 0.3 m internal diameter uses a "U" shaped tube with a large, raised opening at one end that allows unrestricted gas flow to the compressor.

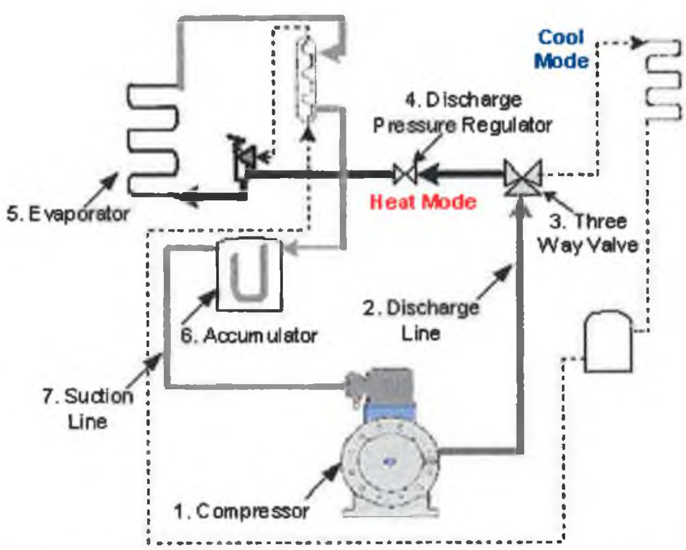


Figure 6.3 Main Components of the TS 500 truck unit in heat mode.

The compressor (1) receives low-pressure refrigerant vapour through the suction line (7). High-pressure, high temperature vapour travels through the discharge line (2) to the 3-way valve (3), which directs hot refrigerant gas either to the evaporator (5) in heat mode or to the condenser in cool mode. When the TTC unit is running in heating, the refrigerant gas is routed by the 3–way valve through a discharge pressure regulator valve (4) to the evaporator-condenser coil (5). The optimum 220-psi discharge pressure for the heat mode is maintained using the discharge pressure regulator valve that opens and closes as required. As the hot-gas refrigerant passes through this coil (5), the latent heat energy is released to warm the air in the food compartment, as the refrigerant changes from vapour to liquid. Condensed, high-pressure liquid refrigerant is transferred from the evaporator to the accumulator tank (6). As described for SL 400e unit in section 4.1.2, this latter step of the heating cycle is designed as an expander in order to provide low pressure and temperature vapour to the compressor suction line while the liquid trapped at the bottom of the accumulator is metered back to the compressor at a safe flow rate.

6.2 FIELD PERFORMANCE OF THE TS 500 UNIT

Based on Swedish field and weather data [Nilsson (2003); Swedish Institute of Meteorology (2003)] the operational problems experienced during one winter day are highlighted in Figures 6.4 and 6.5. The same problems identified in Chapter 5 for SL 400e unit through in-house experimental tests have emerged: i) low heating capacities (region A), ii) long pull-up time (region B) and iii) poor set-point temperature control (region C). Figure 6.4 shows field data for the TS 500 unit during one day in February 2002, with mild ambient temperature conditions of –11°C, while Figure 6.5 shows the unit operation at very low ambient temperatures of –28°C. This detailed field data completes the characterisation of the unit behaviour presented in Figures 2.9 and L.1 (Appendix L) where the behaviour during five days in January 2002 of two different TS 500 units was analysed.

Figure 6.4 shows that after 2 hours in low speed diesel mode operation with good temperature control within ±2°C, the heating capacity of the unit becomes insufficient to maintain the +10°C set-point temperature. As a result, the unit changes the operating mode to high-speed diesel, for increased heating capacity. It has to be noticed that even in these operation conditions it takes up to 2 hours to recover the set-point, after which low speed diesel control is again activated. After another 2 hours of good temperature control in low speed mode, the return air temperature suddenly drops 6°C degrees below the set-point, due to charge trapped in the cooling cycle. As a result, the unit has to reactivate high-speed diesel mode to be able to recover the +10°C set-point truck compartment temperature. During night hours, after 20:00 p.m, when the temperature averages –17°C, the unit runs continuously in high-speed diesel mode. Even in this operation mode that delivers maximum heating capacity, the temperature control is consistently 3°C below the set-point.

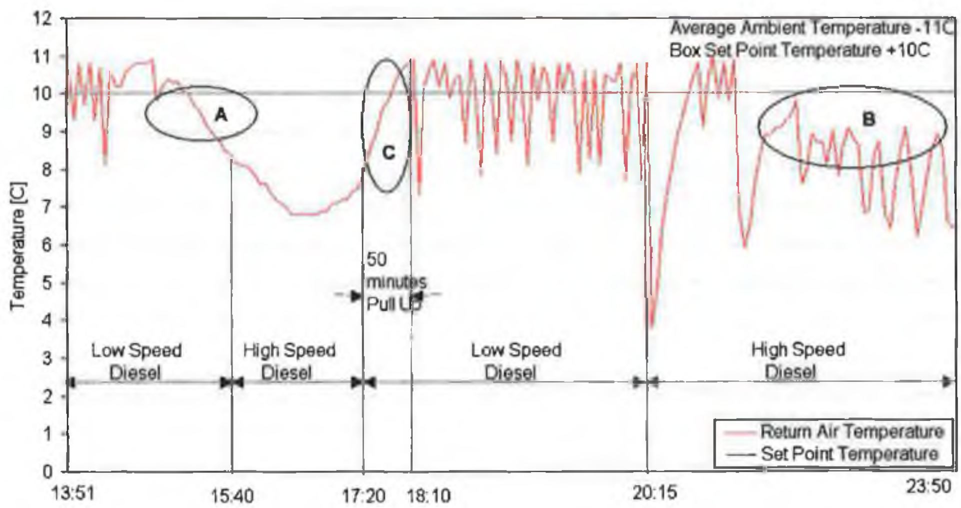


Figure 6.4 Swedish field data for the TS 500 unit operating in mild ambient temperature conditions of –11°C, during one-day 12.01.2002 [Nilsson (2003)].

Based on the field data presented, it can be concluded that the TS 500 unit does not have enough heating capacity to maintain the box set-point temperature in low speed diesel mode even at mild ambient temperature conditions like –11°C. In these conditions the unit is forced to operate in high-speed diesel mode, with 40% increased fuel consumption, a situation that can only deliver a set-point of +7°C during night hours.

Figure 6.5 presents field data for the TS 500 unit during a very cold winter day that recorded a –28°C ambient temperature while it has to maintain a +8°C set-point temperature in the truck food compartment. This data shows the extremely long 7 hours pull-up time from –6°C to +7°C. For a 3-hour period during day the unit is capable of maintaining the set-point due to higher ambient temperature. However, during night hours the TS 500 unit is forced to run in high-speed diesel mode to reach the set point due to very low ambient temperatures. Even in this operating mode the unit struggles to reach the set- point. After almost 7 hours of pull-up while operates in maximum heating capacity the unit regains control at an insufficient temperature range that is with 2°C below the desired set-point.

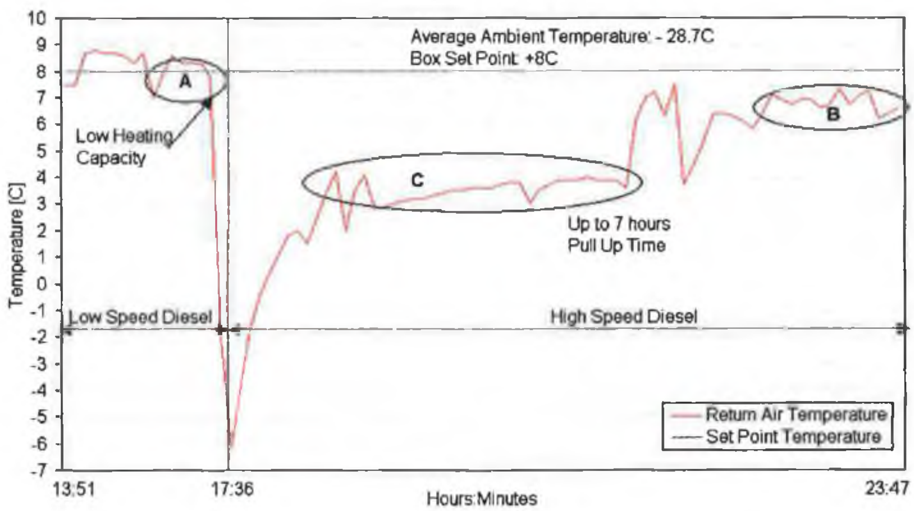


Figure 6.5 Swedish field data for the TS 500 unit operating in severely low ambient temperature conditions of –28°C, during one-day 9.12.2002 [Nilsson (2003)].

It can be concluded that in sub –20°C ambient temperatures the TS 500 unit does not have enough heating capacity even in high-speed diesel mode. A pull-up time of 7 hours is unacceptable during road transportation where this time very often represents the duration of the journey between two destinations. Based on the field data problems highlighted in Figures 2.11, 6.4, 6.5 and L.1 (Appendix L) for TS 500 unit and on the importance of heat mode especially in northern hemisphere countries as presented in Figure 2.10 and Appendix D (Figures D.1 and D.2) in house testing was required to confirm the exact heating capacity as well as the extent of poor temperature control and long pull-up time problems. The novel EHI test method and the heating test facility overviewed in Chapter 3 (Figure 3.3) were used for the in house experimental data. An overview of the test set-up and supplementary pressure and temperature measurements are presented in the following section.

6.3 EXPERIMENTAL METHODS AND TEST FACILITY

The TS 500 standard production unit was tested using both the new EHI test procedure and the standard ATP method based on heat balance calculations across the evaporator as presented in Chapter 3. These tests were performed in the first industrial based heating test facility commissioned in Thermo King, Galway, presented in Figure 3.3 and Appendix G.

6.3.1 Test Facility

Following the schematics presented in Figures 3.1 and 3.3 the heating tests were conducted with the TS 500 unit installed on a calibrated box with the evaporator inside, while the condenser was positioned within the temperature controlled environment of the Ambient Room. The test temperature within this room was controlled by monitoring the condenser air inlet temperature to the TS 500 unit. Figure 6.6 shows the TS 500 unit test set-up.

a) Front View of TS 500 unit.
(View A in Figure 1.7).

b) Side View of TS 500 unit.

Figure 6.6 Test Configuration for TS 500 Heating Capacity Tests in Ambient Room.

Heating tests were performed at three different ambient temperatures of -10°C, -20°C and –30°C that where selected based on average temperatures during autumn and winter months in Europe, presented in Chapter 2. The +12°C box temperature was based on the temperature requirements for the most common transported fresh products like banana, lemon and potato [ASHRAE (1997)]. The test conditions are summarised in table 6.2. It is well known that compressor capacity changes in direct proportion to the compressor's operating speed. As a result, the tests were performed with the TS 500 unit running at maximum compressor rotational speed of 2440 rpm defined as high-speed diesel mode and minimum rotational speed of 1442 rpm during low speed diesel operational mode.

Table 6.2. Heating Test Conditions imposed on the TS 500 TTC unit for rigid body trucks.

Test No.	Unit Mode	Ambient Temperature	Box Temperature
1	Diesel – High Speed	- 10 °C	12 °C
2	Diesel – High Speed	- 20 °C	12 °C
3	Diesel – High Speed	- 30 °C	12 °C
4	Diesel – Low Speed	- 10 °C	12 °C
5	Diesel - Low Speed	- 30 °C	12 °C

6.3.2 Test Measurements

Several measurements were monitored for both TTC unit and heating test facility as indicated in Figure 3.11 and Table 3.7. For a better understanding of the TS 500 system behaviour in Heat Mode additional temperature and pressure sensors were introduced as defined in Figure 6.7 and Table 6.3.

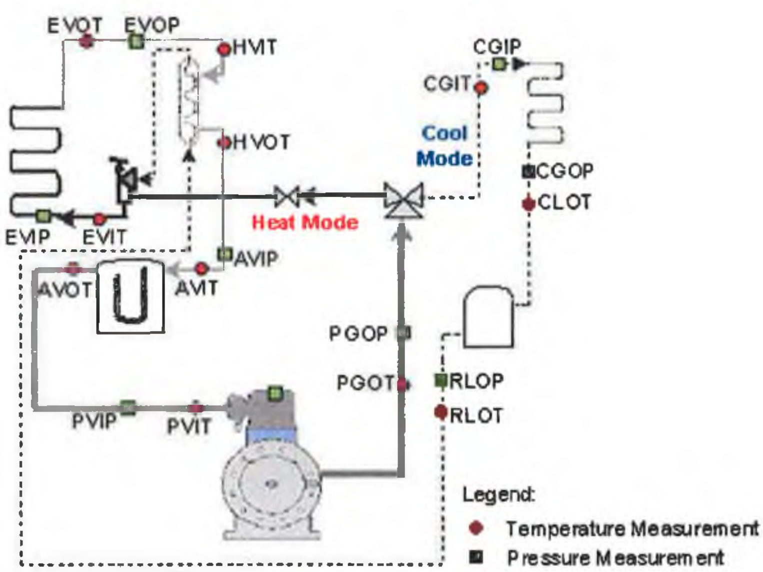


Figure 6.7 Location of Pressure and Temperature Measurements on the TS 500 standard unit tested in heating.

The measurements were performed using test instrumentation with the accuracy presented in Table 6.3 and Appendix H. All thermocouples were calibrated using a D55 SE temperature calibrator while the accuracy of the pressure transducers was obtained using a Superb model pressure calibrator, type PCC3-H-200-2. A description of the abbreviations used to define the temperature and pressure sensors and theirs location is presented in Table 6.3 while the accuracy of the test instrumentation in Table 6.4.

Table 6.3. Defining the pressure and temperature measurements on the TS 500 unit.

Sensor Abbreviation	Explanation	Required to measure
PGOT	Compressor discharge temperature	Compressor superheat.
PGOP	Compressor gas discharge pressure	Compressor superheat.
EVIT	Evaporator inlet temperature	Heating capacity.
EVIP	Evaporator inlet pressure	Evaporator gas inlet enthalpy.
EVOT	Evaporator outlet temperature	Heating capacity.
EVOP	Evaporator outlet pressure	Evaporator outlet enthalpy.
HVIT	Heat Exchanger inlet temperature	Heat exchange
HVOT	Heat Exchanger outlet temperature	Heat exchange.
AVIT	Accumulator inlet temperature	The accumulator evaporation.
AVIP	Accumulator inlet pressure	The accumulator inlet enthalpy
AVOT	Accumulator outlet temperature	The accumulator evaporation.
PVIT	Compressor inlet gas temperature	Compressor inlet superheat.
PVIP	Compressor inlet gas pressure	Compressor inlet superheat.
CGIT	Condenser gas inlet temperature	Charge trapped in the condenser.
CGIP	Condenser gas inlet pressure	Charge trapped in the condenser.
CLOT	Condenser liquid outlet	Charge trapped in the condenser.
CLOP	Condenser outlet pressure	Charge trapped in the condenser.

Table 6.4. Test Instrumentation used to monitor and control the TS 500 heating capacity tests.

Instrument Description	Instrument Range	Accuracy
Type-T Thermocouples-Special Grade	-200 to 350 °C	± 0.5°C
Power Measurements-Digilogic watt-hour Transducer; Model DL31KA2-6270-2-8	0 to 20 kW	±(0.1% + 1 Watt)
TK Pressure Transducers	0 to 500/ 0 to 200 psi	± 0.4% / ± 0.2%

6.4 HEATING PERFORMANCE FOR THE STANDARD TS 500 UNIT

While Chapters 4 and 5 focused on the capacity characterisation of the most powerful trailer-mounted TTC unit, the SL 400e system, this section presents the heating capability of the TS 500 unit that is a small truck-mounted unit with lower designed capacity performance. The heating capability of the standard "off-the-shelf" TS 500 unit is presented in this section for both high and low speed diesel modes. The measured heating capacity figures are obtained from both the new EHI test procedure and the standard ATP heat balance method, while a more detailed analysis through a mathematical model prediction for a wider range of test conditions will be completed in Chapter 7.

A summary of the results obtained for five different temperature test conditions with the unit running in both high and low speed diesel are presented in Table 6.5. The rotational speed of the compressor was either 2440 rpm or 1442 rpm, respectively.

Table 6.5. Measured heating capacities of the TS 500 in high and low speed diesel mode.

Test No.	Mode	Ambient Temp [°C]	Box Temp [°C]	Standard Method Capacity [W]	EHI Method Capacity [W]	% Difference
1	High Speed	-10	+12	5,708	5,525	-3.2%
2	High Speed	-20	+12	5,123	5,026	-1.89%
3	High Speed	-30	+12	4,576	4,033	-11.8%
4	Low Speed	-10	+12	3,836	3,595	-6.28%
5	Low Speed	-30	+12	2,704	2,409	-10.9%

The results presented in Table 6.5 are also shown in graphical format in Figure 6.8. The reserve heating capacity of the TS 500 unit is also presented for both high and low speed diesel modes. The reserve heating capacity of the unit is considered to be the difference between the heating capacity measured in those specific conditions and the estimated heat losses between the box temperature and ambient temperature through the walls insulation, based on equation 4.1.

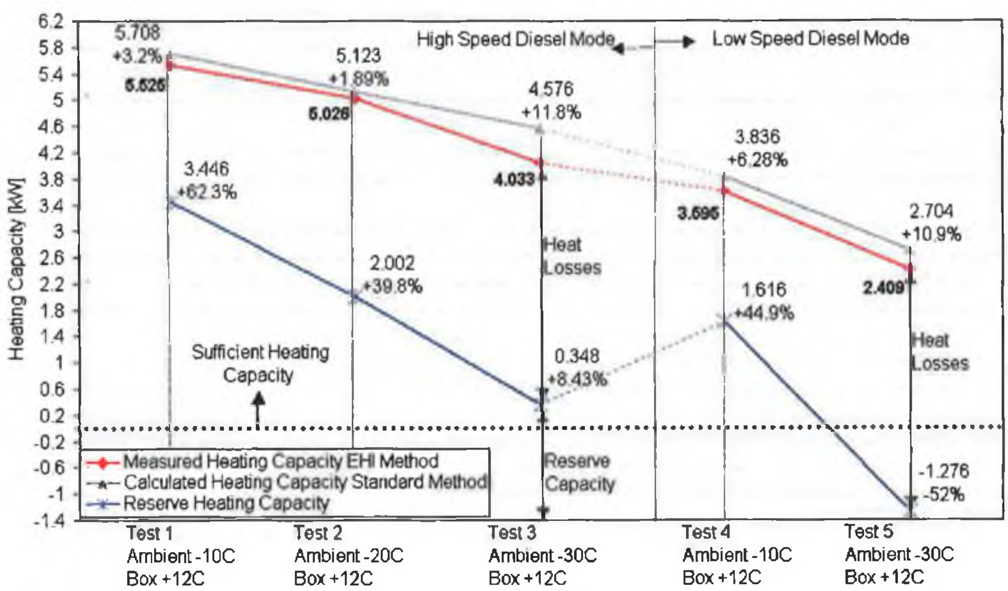


Figure 6.8 Heating Capacities for TS 500 in high and low speed diesel modes (The percentages shows the differences of: i) calculated heating capacities based on standard ATP method reported to the measured EHI method figures and ii) the percentages of the existing reserve capacities from the EHI method measured values for the test conditions).

The heating capacities obtained from the EHI method for TS 500 unit in high-speed diesel mode range between 4000 W and 5800 W, which shows that at the same box temperature the heating capacity drops by 36% when the ambient temperature drops by 18°C. Measured heating capacity for low speed diesel mode range between 2400 and 3600 W, depending on the temperature conditions. Based on these figures it can be concluded that the low speed diesel mode not only produces a 40% lower heating capacity, but also is more sensitive to the operating conditions. Higher capacity figures are obtained at lower temperature difference between the ambient and box set-points because of lower trailer wall (including floor and ceiling) heat losses. Figure 6.8 shows that for the same box temperature of +12°C and two different ambient temperatures that differ by 10°C, the heating capacity decreases by 32.9% due to higher trailer wall heat losses at very low ambient temperatures. It can be noticed that for an ambient temperature of –30°C the unit has insufficient heating capacity in low speed mode, a minimum of 1276W is required for a satisfactory behaviour. For the same test temperatures only 348W reserve capacity that corresponds to 1°C higher heating capability of the box air is recorded for high-speed diesel mode. It can therefore be concluded that the TS 500 unit is not capable of providing sufficient heating capacity to achieve box set-point temperatures higher than +12C for both operational modes. It can also be observed that the capacities measured with the accurate EHI test method are less that the figures obtained with the ATP procedure that confirm the problems from field presented in Figures 6.4 and 6.5. The errors introduced by the latest method based on heat balance calculations are increased for high-speed at –30°C ambient temperature and low-speed mode where 10.9% higher capacity than the existing unit heating capability is recorded.

A better understating of the influence of ambient temperature on the heating capacity can be had by referring to the same Figure 6.8, where heating capacity of TS 500 unit running in high-speed diesel mode at three different ambient temperatures of –10°C, -20°C and –30°C and the same box temperature of +12°C. Figure 6.8 shows an overall drop of -27% in the heating capacity between –10°C and –30°C while the slope of the graph indicates that the most significant decrease occurs below –20°C. This helps to explain the failure of the system in heating identified by the field data in Figures 6.4 and 6.5, when the ambient temperature dropped below –18°C. From the two tests presented in Figure 6.8 for low speed diesel, it can be noticed that an even greater drop of -33% takes place in the heating capacity between -10°C and -30°C ambient temperatures, indicating that the heating capacity of the system is even more sensitive to low ambient temperatures in low speed diesel mode. A more complete analysis of the variations of the heating capacity with ambient and box temperature changes is presented through the mathematical model predictions in Chapter 7, Figure 7.8 and Appendix M, Figure M.1.

Several repeatability tests were performed for the heating capacity tests based on the EHI test method. This indicator shown to be very good, with small differences of just ±50W between test measurements (Figure 6.9). Such repeatability testing was conducted on three different days, while other tests were performed in between. Figure 6.9 shows three different measurements for the heating capacity of TS 500 unit in low speed mode at –30°C ambient temperature and +12°C box temperature. The maximum difference between results is of 43W that corresponds with 1.35%.

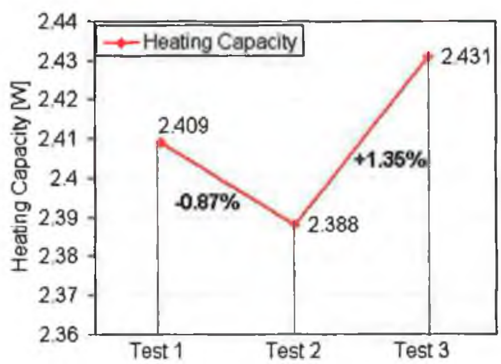


Figure 6.9 Repeatability test results for measured heating capacity using the EHI test method at ambient temperature of –30°C and box temperature of +12°C, while the TS 500 unit was operating in low speed diesel mode.

6.5 PROBLEMS THAT APPEAR IN HEAT MODE

To obtain a complete characterisation of the TS 500 unit in heat mode and to emphasise the problems that appear during the operation at very low ambient temperatures, field data from one of the largest Swedish food distributors was presented in section 6.2 [Nilsson (2003)]. Analysis of these results helped to define the conditions for a series of heating tests in order to investigate the field insufficiencies during heat mode operation as overviewed in this section.

6.5.1 Insufficient Heating Capacity

Figure 6.10 shows insufficient capacity for two test conditions with the same ambient temperature of –30°C and +2°C, +12°C box temperature. It can be concluded that the ability of the TS 500 unit running in high-speed diesel mode to maintain the box temperature above +12°C is limited at ambient temperatures lower than –25°C as it only has a +10% reserve capacity at +12°C temperature. This insufficient heating capacity figure for high-speed diesel mode is in agreement with the field problems that appear at the same ambient and box temperatures, highlighted in Figures 6.5, where the unit requires 7 hours pull-up time to maintain only +8°C box temperature.

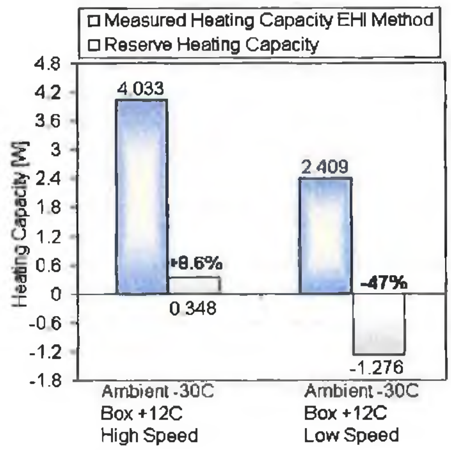


Figure 6.10 Insufficient Heating Capacity for TS 500 unit at ambient temperature of –30°C.

For a better understanding of the insufficient heating capability of the TS 500 unit encountered during field operation in low speed diesel mode (Figure 6.4 and J.3), the in house test quantifies a 47% necessary increase in the heating capacity to maintain +12°C box set-point temperature in an ambient of –30°C.

6.5.2 High Pull-Up Time

Both Table 6.6 and Figure 6.11 present the pull-up time required to achieve a +12°C and +21°C box set-point using low speed diesel mode, when exposed to a –20°C and –26°C ambient temperatures respectively. It can be noticed that the pull-up time is sensitive to both ambient and box temperatures. The unit was tested during pull up from two different ambient temperatures of –20°C and –26°C at the same box set point of +21°C running in low-speed diesel operation mode.

Table 6.6. Pull-up time for the TS 500 unit running in low speed diesel mode.

Box Temperature	Ambient Temperature	Pull Up Time
Box +21°C;	Ambient –26 °C	Pull Up Time 8 hours
Box +21°C;	Ambient –20°C	Pull Up Time 4 hours
Box +12°C	Ambient -26°C	Pull Up Time 2 hours

It can be noticed that the 6°C temperature difference between the ambient conditions, generated a 2 hour and 20 minutes increase in the pull-up time for the same box set-point temperature. Based on the pull-up times obtained for the unit operating in high-speed mode at the same ambient of –26°C but different box temperatures of +12°C and +21°C, a difference of almost 2 hours was obtained with a variation of +9°C for the transport temperature.

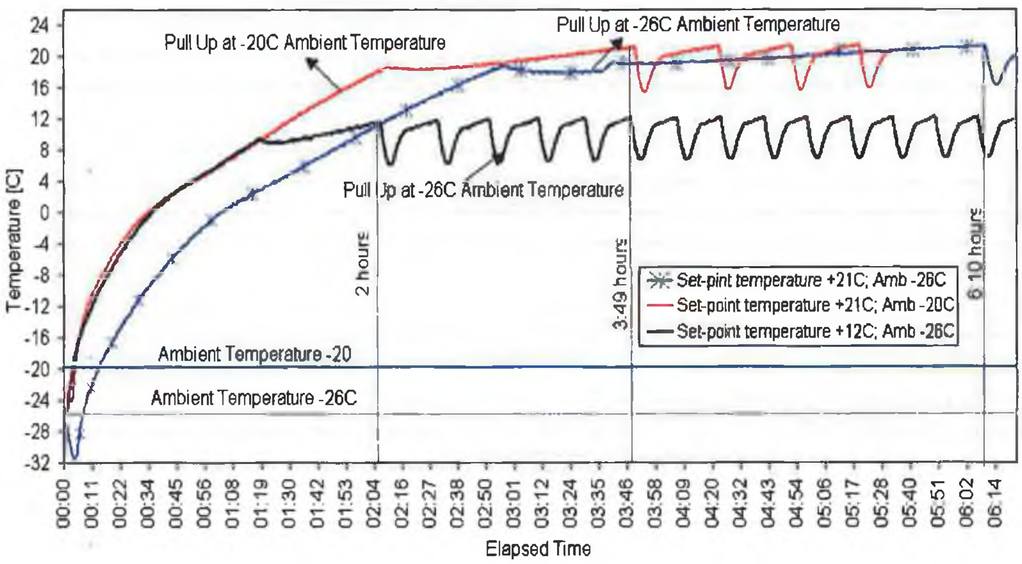


Figure 6.11 Pull - Up Time for the TS 500 unit running in low speed diesel mode to achieve +21°C and +12°C box temperatures in –20°C and –26°C ambient temperatures.

The pull-up time is unacceptably high for ambient temperatures sub –25°C and box temperatures higher than +12°C. From the field behaviour presented in Figures 6.4 and 6.5 it can be noticed that the pull-up time is up to 6 hours for the TS 500 unit due to the fact that the trailer and the unit were operating at very low ambient conditions of –15°C average temperature for almost 1 month prior to February 2002 [Swedish Institute of Meteorology and Hydrology (2002)]. In a test facility is almost impossible to keep very low ambient temperature for such a long time to obtain exact simulation of the field conditions. Note that the in house pull-up tests presented in Figure 6.11 were performed after 5 days of –26°C controlled ambient temperature.

6.5.3 Poor Set-Point Temperature Control

Based on the data presented in the previous sections it can be concluded that the major problems in heat mode operation take place during low speed diesel mode due to insufficient heating capacity. This not only affects the maximum set-point temperature that can be reached but also has a negative impact on the temperature control and pull-up time. Figure 6.12 shows the level of set-point temperature control achieved at +12°C and +21°C when the ambient temperature is -10°C and -20°C.

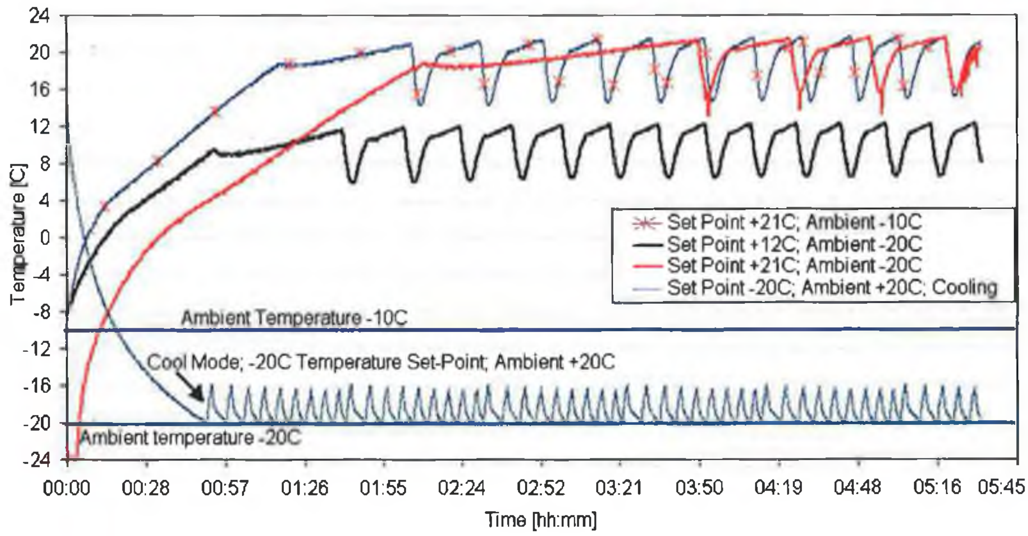


Figure 6.12 TS 500 unit Set-Point Temperature Control in low speed diesel mode, ambient –20°C, -10°C and box temperature +12°C and +21°C. A comparison with –20°C set-point temperature control in cool mode at +20°C ambient temperature is also provided.

Temperature control was shown to be equally poor at both –10°C and –20°C ambient, with the trailer air temperature fluctuation at regular 20 minutes intervals between 2°C above and 7°C below each set-point. This compares poorly against the much tighter set-point temperature control generated for the same 40°C difference between the inside and external ambient air temperatures achieved in cooling mode, where the actual temperature control never drifted above 4°C above the set-point and for much shorter duration. Figure 6.12 also shows very long pull-up time of almost 6 hours in heating mode while the pull-down time in cooling mode for the same temperature difference is just 1 hour. The temperature control ranges summarised for both operational modes in Tables 6.7 and 6.8 show better temperature control results in high-speed diesel mode. The air temperature fluctuates between +1°C above and –4°C below the set-point for the same 40°C temperature.

Table 6.7. Temperature Control achieved by the TS 500 in low speed diesel mode.

Ambient Temp [C]	Trailer Temp [C]	Temperature Control Range
–10°C	+21°C	Temperature Control: Set Point +0.5°C/-6°C
–20°C	+21°C	Temperature Control: Set Point +0.5°C/ -7 °C
-20°C	+12°C	Temperature Control: Set Point +0.5°C/-5 °C

Table 6.8. Temperature Control achieved by the TS 500 in high-speed diesel mode.

Ambient Temp[C]	Trailer Temp [C]	Temperature Control Range
–10°C	+21°C	Temperature Control: Set Point +1°C/-3.5°C
–20°C	+21°C	Temperature Control: Set Point +1°C/ -4°C
-20°C	+12°C	Temperature Control: Set Point +0.5°C/-3°C

The set-point temperature control of the unit is highly dependent on the ambient temperature. This can be clearly seen in Figure 6.12 by comparing the temperature control achieved at +21°C, when the ambient temperature is decreasing from –10°C to –20°C. Not only that it takes twice as long to reach the set-point temperature at -20°C ambient, but the set-point temperature control is significantly worse, varying in a 2°C higher range than the control at –10°C ambient. Such poor set-point temperature control obtained in heat mode versus cool mode is explained as a result of insufficient heating capacity and charge migration in cooling cycle.

6.5.4 Compressor Failure Costs

The top four causes of failures that appear during the operation of scroll compressor during a three year period are: i) low torque on fixed scroll (1), ii) compressor driven in the reverse direction (2), iii) orbiting scroll failures (3) and iv) compressor leaks (4) [Thermo King Report (2004)].

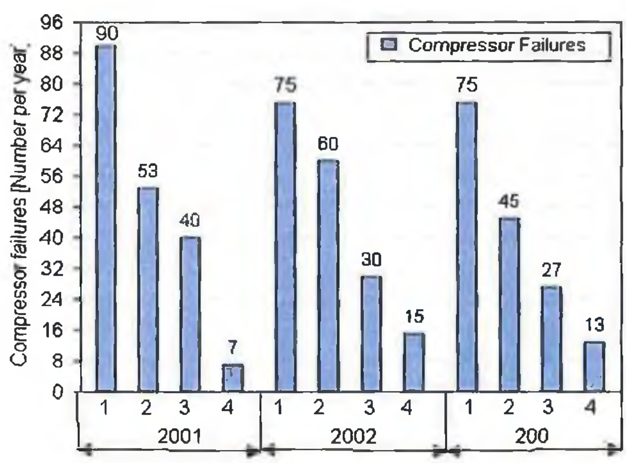


Figure 6.13 TS 500 unit scroll compressor failures during 2001÷2003 [Thermo King (2004)].

Due to a good figure for the compressor suction superheat an average of 23°C; the 4.5 kW scroll compressor has fewer failures due to operating parameters than the other X 430 reciprocating compressor option. The orbiting scroll failure that has 21% is mainly caused by liquid that enters with the refrigerant during compression. Better operating conditions for the compressor running with higher suction superheat values can offer better reliability for the compressor.

6.5.5 Compressor Suction and Discharge Superheats

Figures 6.14 and 6.15 show compressor suction and discharge superheat when the TS 500 unit runs at the test conditions presented in Table 6.2. The suction vapor has to be superheated to guarantee that the compressor is operating with no liquid during the compression process. Figure 6.14 shows good figures for the compressor suction superheat. An important decrease of 56% in the suction superheat is noticed when the ambient temperature drops from –10°C to –30°C while the unit operates in high speed diesel mode. A 29.6% drop is recorded in low speed diesel mode that is less sensitive at ambient temperature variations.

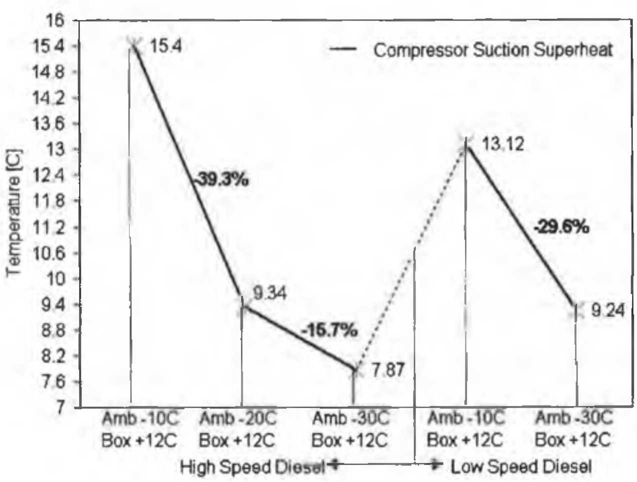


Figure 6.14 TS 500 standard unit – Compressor suction superheat for high and low speed diesel modes, at +12°C box temperature and ambient of –10°C, -21°C and –30°C.

High compressor discharge superheat offers the guarantee of increased heat released at the evaporator inlet when the vapor temperature drops to the saturation value. Figure 6.15 shows acceptable high figures for compressor discharge superheat between +15°C to +23°C. The TS 500 unit operating in low speed mode is more sensitive at ambient temperature variations. A –30% drop was recorded for 20°C temperature difference, while for high-speed diesel mode –24% lower compressor discharge superheat is obtained for the same temperature conditions.

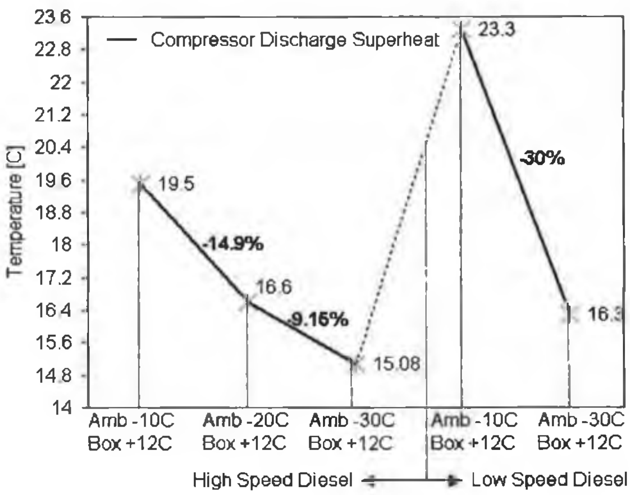


Figure 6.15 TS 500 standard unit – Compressor discharge superheat for high and low speed diesel modes, at +12°C box temperature and ambient of –10°C, -21°C and –30°C.

6.6 SUMMARY

A summary of the most significant results obtained from the heating testing of the single-compartment truck TS 500 unit is presented in this section.

A wide range of tests defined in Table 6.2 was performed on a TS 500 production unit running in both high and low speed diesel modes based on both STP standard method and the new EHI test method. The measured heating capacities together with the problems that appear during heat mode operation are highlighted. While the key results were presented in this chapter, the following comments can be summarised:

- The heating capacities obtained for low speed diesel mode are approximately 50% lower than the figures obtained for high-speed diesel mode.
- Difference between both test methods was negligible, less than +4%, when the temperature differences between the box and external ambient were less than 15°C. In this instance, heat losses through the box/ trailer walls are small and measured results are comparable. However, when the temperature difference across the wall exceeds 20°C, fabric losses are significant and the EHI method

shows 10% lower capacity figures. It can be noticed the higher errors are obtained for the unit operating in low speed diesel mode.

- Sensitivity of heating capacity to ambient air temperature was established for the first time in Figure 6.8, while a more detailed analysis is obtained through the mathematical model prediction in Figures 7.8 and M.1.
- As the TS 500 unit has 60% less heating capacity than the trailer SL 400e unit analysed in Chapter 4, the lack of heating capacity is also increased, while longer pull-up times are also recorded (Figure 6.11). It can be concluded that the main objective is to increase the heating capability of the unit.

Section 6.5 presents the problems that appear for the TS 500 unit during the operation in heat mode. The following problems were encountered:

- Low heating capacity at temperatures below –25°C and box temperatures higher than +2°C for both low and high speed diesel modes.
- Unacceptably long pull-up times up to 7 hours were recorded at ambient temperatures lower than –25°C for the unit operating in low speed diesel.
- Operating under the same temperature conditions the unit displays poor temperature control of –7°C and +0.5°C around set-point for low speed diesel mode and between –4°C to +1°C for high-speed diesel mode. It can be noted that the unit operates for a long time of almost 40 minutes during each cycle heat-cool at a temperature with 6°C below the set-point.

While the positive effect of an average 30% increased capacity obtained for the modified SL 400e unit is also expected for the TS 500 truck unit, the need for greater impact on the heating capability of the truck TS 500 unit resulted in two different proposed types of design changes based on the effect of engine coolant used as a primary heat exchange fluid in two heat exchangers that should be implemented in the cycle. The effect of these modifications on the unit response in heating together with a comparison between the standard and modified TS 500 units is obtained through the first mathematical model for heat mode described in Chapter 7. The changes proposed are as follows:

- Coolant heat exchanger installed on compressor suction line,
- Coolant coil installed in the accumulator.

CHAPTER 7

MATHEMATICAL MODEL OF THE STANDARD AND MODIFIED SINGLE-COMPARTMENT TS 500 UNIT IN HEAT MODE

CONTENT:

This chapter describes the development of the first steady-state mathematical model capable of predicting both system behaviour and heating capacity of the standard single-compartment TS 500 truck unit [Radulescu *et al.,* 2004]. In light of the system deficiencies identified in Chapter 6 and based on a comprehensive analysis of the unit through model predictions as presented in sections 7.1 to 7.5, the design of the standard production TS 500 system is proposed to be modified to make use of the engine coolant. Two new heat exchangers are proposed to be implemented on the compressor suction line and within the accumulator, using coolant as a primary fluid and refrigerant vapor and liquid as a secondary heat-exchange fluid respectively. Another two mathematical models are created and used as design tools to guide the redesign process.

7.1 INTRODUCTION

As in any engineering discipline, the use of validated numerical models offer many advantages. In this instance, numerical models help to meet the need to minimise energy consumption, reduce both test time and costs associated with measuring system performance. Despite the clear importance of heat mode during the transportation of perishable foods there is a marked absence of information related to this aspect of refrigeration system operation [Lohan *et al.,* (2003)]. Most research studies focus on the analysis and modelling of vapour-compression TTC units in cooling and defrost modes [Cleland and Koury (1999)]. Even if a TTC unit in heat mode has similarities with a heat pump it is impossible to apply the existing mathematical models and simulations available for heat pumps [Sakellari *et al.,* (2003); Mac Arthur *et al.,* (1984); Chi *et al.*, (1982)]. The latter operates between two external fluids at different temperatures in the condenser and the evaporator sections, whereas a TTC unit depends on heat exchange with only one external fluid, the ambient air flowing through the condenser, while the evaporation of the liquid refrigerant takes place through expansion into the accumulator tank. As a result of the increased demand for a better understanding of the heat mode behaviour of a TTC unit with all the system particularities, this chapter sought to address this need by presenting the development of the first steady-state mathematical model, capable of predicting both the system behaviour and heating capacity of a standard single-compartment TS 500 TTC truck unit, together with two simulations of a modified system. It is worth noting that while the measured heating performance of the standard TS 500 unit was measured and discussed in Chapter 6 the model introduced in this chapter is capable of extending the analysis of the standard unit to provide supplementary data for a wider range of test conditions (section 7.5). Two design changes based on the effect of the engine coolant are proposed (Figure 7.9). The predicted results obtained from the validated mathematical model accuracy of ±6% also overviewed in this chapter.

The validation of the mathematical models was obtained by comparing the experimental figures from both the new EHI and the standard ATP test procedures and the model predictions. This first steady–state mathematical model for a single-compartment TTC unit in heat mode is based on a modular formulation so that the main components of the system schematically presented in Figure 6.3 and 7.1, including the compressor (1), pressure regulator valve (4), evaporator (5), accumulator (6) are modelled separately. Using existing methods for two separate simulations of compressor [Chen *et al.*, (a,b), (2000)] and evaporator [Willatzen *et al.*, (1998); Hayani *et al.*, (2003)], the proposed model is the first one that for a given set of operating temperature is capable of predicting the heating performance and to couple separate models of the components by their inlet and outlet variables. The equations used for modelling the main unit components are for the first time applied and verified for low ambient temperatures, while the TTC unit operates in heating. An empirical correction factor that accounts for heat gains from the air fan motors and for heat losses from the warm box to the colder external surroundings is presented. The two models introduced for the modified TS 500 unit are based on the algorithm used for mathematical model I of the standard unit. While the modifications proposed are based on a shell-and-tube heat exchanger-modification I and on a new type of accumulator that has a coolant coil installed inside-modification II, the heat exchange capability was modelled and validated within ±7% prediction accuracy after a comparison with experimental data obtained from the manufacturers [Danfoss (2004); Parker (2004)].

7.2 MATHEMATICAL MODEL I – STANDARD TS 500 UNIT

The purpose of the mathematical model presented in this section is to predict heating capacity and system behaviour of the TS 500 standard truck unit. It can be classified as a steady-state, phenomena-oriented model that accounts for all the individual parts presented in Figure 7.1 [Rasmunssen *et al.*, (2000)]. The basic components of the TTC unit in heat mode including the compressor (1), discharge pressure regulator (4), evaporator (5) and accumulator (6) described in section 6.1.3, are modelled and coupled together in terms of mass flow rates, enthalpy, refrigerant temperatures and pressures. Figure 7.1 shows the key refrigerant states:

- Compression Process: States 1→2.
- Pressure Regulator Valve Process: States 3→4.
- Condensation Process in the Evaporator: States 5→6.
- Evaporation Process: States 7→8.

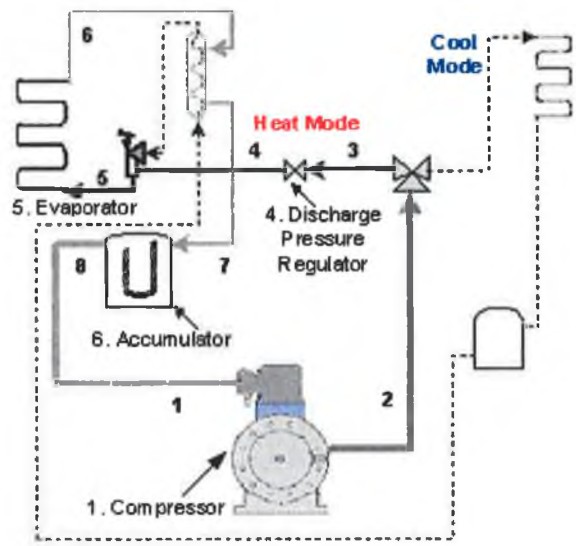


Figure 7.1 General Overview of TS 500-truck unit in Heat Mode including the main refrigerant states.

7.2.1 EES User Interface

The mathematical model has been developed using Engineering Equation Solver (EES) software [EES (2004)], which offers an easy to use interface together with the possibility to generate quality graphical output plots. It has built-in mathematical functions and thermo-physical property databases and is capable of identifying groups of equations that must be solved simultaneously. The model requires the following input parameters: compression volume ratio, isentropic coefficient calculated based on experimental data and test conditions, ambient and box temperatures. The model outputs provide a complete system analysis involving temperature, pressure, enthalpy and mass flow rate for all components in the cycle together with the heating capacity prediction. The main user interface is presented in Figure 7.2. Note that the model has been constructed to mimic the physical system, but overlays the critical thermodynamic states defined in Figure 7.1.

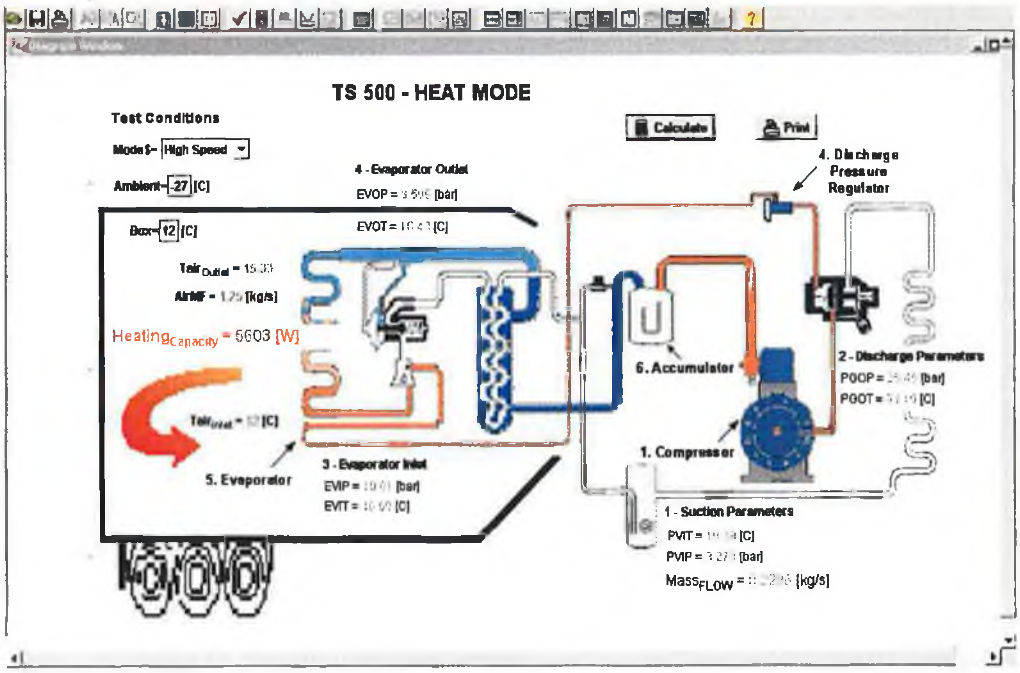


Figure 7.2 EES User Interface to the standard TS 500 steady-state mathematical model I that predicts both system performance and heating capacity.

For a defined set of inputs the calculation is initiated by assuming starting values for the compressor suction state, derived from a comprehensive set of experimental data performed. With these values, the algorithm described in section 7.2.2 and based on the algebraic equations 7.1 to 7.15 is able to determine all parameters for the basic components and to predict the heating performance of the TTC unit.

7.2.2 Model I Algorithm Flow - Chart

A flow-chart of the model algorithm, together with numbers that correspond with key refrigerant states and flow direction is presented in Figure 7.3. The initial inputs of the model are the ambient and box test temperatures. In the model algorithm a chart for the variations of the compressor suction pressure and temperature with the test conditions is introduced. As a result, the user automatically obtains suction pressure and temperature, State 1 in Figure 7.3. The compression process that converts the working fluid from state 1 to 2, provides outputs based on calculations of the compressor discharge parameters and refrigerant mass flow rate (State 2, Figure 7.3). After process 3→4, the pressure and temperature drops are obtained and used as inputs for the condensation process that takes place in the evaporator (5) during process 5→6. The heating capacity of the unit is also obtained. The outlet parameters from the evaporator are used in the verification loop calculation for the accumulator during process 7→8. The refrigerant pressure and temperature at the accumulator outlet (State 8) should be the same as the initial suction compressor parameters as long as no input is changed. All temperatures and pressures are considered as outputs that the user can inspect visualise them on the main EES window (Figure 7.2).

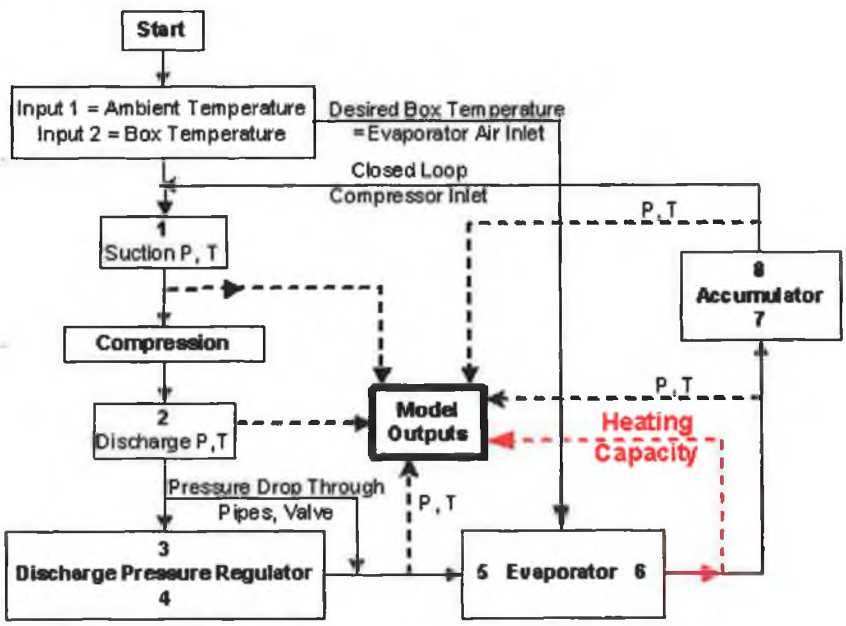


Figure 7.3 Flowchart diagram describing the mathematical model algorithm.

7.2.3 Compression Process State 1→4

In recent years many publications have presented models that simulate scroll compressor behaviour. Some of these models take a very complex construction, oriented to take into account all geometric characteristics together with calculations of refrigerant mass flow and enthalpy for all compressor chambers in transition conditions [Chen et *al.*,(a,b), (2000)]. Others are phenomena-oriented comprehensive models based on heat transfer calculations and the assumption that compression follows an isentropic process of constant index [Chen *et al.*, (a,b), (2000)]. This latter approach was adopted in this model, in order to compute refrigerant mass flow rate, enthalpy and temperature at the compressor outlet. A particularity of the system modelled is that the compression process takes place not only in the scroll compressor, but also as a second stage on the discharge line, mostly because of the dynamic behaviour of the pressure regulator valve [Rasmunssen *et al.*, (2000)]. Compressor modelling has therefore become closely connected to valve modelling for isentropic index (equation 7.1) and refrigerant mass flow rate (equation 7.2) calculation. This assumes assuming that the compression process takes place in one stage, between compressor inlet (State 1) and discharge pressure regulator valve outlet (State 4) Figure 7.1. Using known values for the compressor suction refrigerant pressure P_1, temperature T_1, volume ratio V_i and the pressure drop characteristics of the regulator valve, the discharge and valve outlet pressures P_2, P_4 were calculated using equation 7.3, assuming an isentropic compression process [Chen *et al.*, (2000); Granet *et al.*, (2000)].

$$k_{is} = \frac{log\,\frac{P_2}{P_1}}{log\,V_i} \qquad (7.1)$$

$$M_f = A_d\sqrt{\frac{2\times g\times k_{is}}{k_{is}-1}\times P_4\times\rho\times\left[\left(\frac{P_1}{P_4}\right)^{2+k_{is}}-\left(\frac{P_1}{P_4}\right)^{(k_{is}+1)+k_{is}}\right]} \qquad (7.2)$$

$$P_2 = P_1\times V_i^{k_{is}}; \qquad P_4 = P_2-\Delta P \qquad (7.3)$$

An isentropic process with fixed exponent can describe the net effect of the heat transfer at the process endpoint (State 4), but it is not able to describe the course of the process itself, as a result of heat transfer that arises mainly due to friction between the compressor's internal components and refrigerant. As the refrigerant is transported through the scroll compressor, it absorbs heat from the scrolls and the bottom and top plate. For this model, the convective heat transfer coefficient has been approximated with a correlation for a spiral tube heat exchanger defined using equation 7.4 [Chen *et al.*, (2000); Carabogdan *et al.*, (1983)]. The curved shapes of the spiral plate heat exchanger with its rectangular cross section represent the shaped chambers in the scroll compressor. As a result of the heat transfer calculation based on equation 7.5, the final discharge temperature, T_2, is determined as the sum of the calculated temperature that allows the isentropic transformation of an ideal gas [Granet *et al.*, (2000)] and the temperature rise resulting from heat exchange between the scrolls and the refrigerant (equation 7.6).

$$h_c = 0.23 \times \frac{k}{D_h} \times \mathrm{Re}_S^{\,0.8} \times \mathrm{Pr}^{0.4} \times \left(1.0 + 1.77\frac{D_h}{R_{aver}}\right) \tag{7.4}$$

$$Q = M_f \times C_P \times (T_{fr} - T_1) = A \times h_C \times (T_{scroll} - T_1) \tag{7.5}$$

$$T_2 = T_1 \times \left(\frac{P_2}{P_1}\right)^{\frac{k_{is}-1}{k_{is}}} + T_{fr} \tag{7.6}$$

While the suction and discharge temperature of the refrigerant, T_1 and T_2, the compressor shell temperature due to friction T_{fr}, were measured for different conditions, the scroll temperature (T_{scroll}) during operation was calculated as the average between the compressor's rotor and stator temperatures based upon expected directions for heat transfer and the results of the overall scroll compressor temperature distribution presented as by Chen *et al.*, (2000). As the discharge gas flows at high temperature from the compressor to the pressure regulator valve, the pipe that is not insulated are liable to experience heat losses to the cold external ambient air that can reach even - 30°C. This process is modelled using a heat transfer coefficient for turbulent refrigerant flow over a uniform cross section pipe at constant temperature [Incropera *et al.*, (1990), (1992)]. The heat losses through the discharge pipe up to the valve inlet, at temperature T_3, are calculated using equation 7.7 [Carabogdan *et al.*, (1983), Incropera *et al.*, (1990)]:

$$q = \frac{T_2 - T_{amb}}{\frac{1}{\Pi \times d_e \times h_{amb}} + \frac{1}{2 \times \Pi \times k_{pipe}} \times \ln\frac{d_e}{d_i} + \frac{1}{\Pi \times d_i \times h_r}} \times L = M_f \times C_p \times (T_2 - T_3) \tag{7.7}$$

As the influence of friction through the discharge pressure regulator valve can be neglected, the outlet temperature T_4 can be accurately calculated based on the assumption of the isentropic transformation of an ideal gas [Granet *et al.*, (2000)].

The energetic performance of a compressor can be described by the efficiency, comparing the actual power consumption to the power consumption of a reference process. The adiabatic compression process is used as reference for expressing the isentropic efficiency (η_{is}) of the compressor using equations 7.8 and 7.9 [Chen *et al.*, (2000)]. The actual power consumption and the efficiency depend on heat transfer to the ambient due to the temperature of the compressor surroundings calculated using equations 7.10 and 7.11 [Chen *et al.*, (2000)], while the thermal resistance between the ambient, suction pipe and shell are obtained for turbulent flow heat transfer [Carabogdan *et al.*, (1983); Incropera *et al.*, (1990)].

$$\eta_{is} = \frac{W_{compression}}{P} \tag{7.8}$$

Where: $$W_{compression} = W_{isentropic} = \frac{k_{is}}{k_{is}-1} \times \frac{P_1}{\rho} \times \left[\left(\frac{P_2}{P_1} \right)^{\frac{k_{is}-1}{k_{is}}} - 1 \right] \times M_f \tag{7. 9}$$

$$P = M_f \times (h_2 - h_1) + Q_{ambient} \tag{7. 10}$$

And $$Q_{ambient} = \frac{T_{pipe} - T_{ambient}}{R_1} + \frac{T_{shell} - T_{ambient}}{R_2} \tag{7. 11}$$

7.2.4 Evaporator – Condenser State 5 → 6

The evaporator design is based on a finned-tube compact heat exchanger, used to transfer the heat contained in the hot refrigerant gas to the air as the gas condenses to its liquid state. The condenser capacity is defined as the maximum quantity of heat that the unit can transfer from the condensing medium when operating at its design conditions. To model this behaviour all three processes occurring in the condenser-desuperheating, condensation and liquid sub-cooling, are treated as one zone for enthalpy change occurring at the condensation temperature [Koury *et al.*, (1999); Hayani *et al.*, (2003)]. It is considered that the evaporator-condenser has a constant overall heat transfer coefficient (U_{cd}), based on the overall logarithmic overall temperature difference [Incropera *et al.*, (1990)]. The heat transfer from the condenser is governed by the equations 7.12 to 7.14 presented below [Carabogdan *et al.*, (1983), Incropera *et al.*, (1990)].

$$Q = U_{cd} \times A_{cd} \times \frac{\Delta T_{max} - \Delta T_{min}}{\ln \frac{\Delta T_{max}}{\Delta T_{min}}} \tag{7.12}$$

$$Q = M_f \times C_p (T_5 - T_6) = M_{f\,air} C_{p_{AIR}} \times (T_{ao} - T_{ai}) \tag{7.13}$$

$$T_{ao} = T_{sat} - (T_{sat} - T_{ai}) \times e^{-\frac{A_{cd} \times U_{cd}}{Mf_{air} \times C_{pAIR}}} \tag{7.14}$$

As the evaporator-condenser is part of the heat mode cycle, the refrigerant inlet state is known from the previous component, air inlet temperature is the set-point desired in the box and the air mass flow is determined experimentally using the wind tunnel test procedure [ASHRAE-Standard 25 (1993)]. Therefore, from these equations, three unknowns can be calculated: air and refrigerant outlet temperature and most importantly the heating capacity of the TTC unit. The refrigerant outlet state is completely determined after calculating the refrigerant pressure drop in the coil [Carabogdan *et al.*, (1983)].

7.2.5 Accumulator Tank State 7 → 8

The model for this component of the cycle is used as a verification task for the compressor inlet state (X_1) and mass flow rate (M_f) and as a basis for the next iteration in the model algorithm. Starting form the previously determined State 6, the accumulator inlet pressure and temperature are determined taking into account the heat losses and pressure drops through the circuit from the evaporator. Considering the fact that the accumulator tank is an expander and based on the experimental results that show a 20°C superheating of the outlet vapour, it can be assumed that the temperature decrease is almost zero, while the outlet pressure is obtained from the saturated vapour proprieties for the temperature obtained as a difference between the outlet temperature value and the superheated temperature. Taking into account the determined inlet and outlet state for the refrigerant, the last parameter required to close the model loop is the mass flow rate ($M_{f,\,8}$) at the outlet of the accumulator and this is calculated using equation 7.15 [Carabogdan *et al.*, (1983)].

$$Mf_{state\,8} = Mf \times h_7 + M_{liq\,Acc} \times C_{p\,Liq} \times T_{liq} \tag{7.15}$$

7.3 MODEL VALIDATION

The quoted ±6% prediction accuracy of the model was determined based on a comparison between predictions and the experimental results presented in section 6.3. The initial ±10% prediction accuracy was improved based on refinements introduced to better define the refrigerant mass flow rates and the influence of the heat losses between the internal and external environment of the truck compartment.

7.3.1 Corrections

Comparison between the initial model predictions with experimental data, show differences of just ±10%. While reasonably good, it was possible to increase predictive accuracy by including a better representation of the real process. Because the heating capacity calculated by the initial model was based on a heat balance between the airside and refrigerant side that involves refrigerant mass flow rates and average inlet – outlet temperatures, the main objective was to define the influence of ambient temperature on these parameters. After reviewing existing research regarding the refrigerant mass flow rate, a correction factor was introduced to account for reduced flow rates through pipes [Fox (1992); Chen *et al.,* (2000)], together with a dimensionless factor accounting for difference in the mass flow due to the discharge pressure regulator valve [Driskell (1983)]. On the air side, considering the fact that the air mass flow rates were experimentally determined using a wind tunnel procedure to an accuracy of ±1% and that the specified air inlet-outlet temperature difference predicted by the model was higher than experimental data, it was important to establish what external influence was omitted by the model. It was concluded that the calculations for air outlet temperature were performed assuming ideal conditions, without accounting for heat gains from the air circulating fan motors or for heat losses from the warm box to the colder external surroundings. An empirical correction factor for the air temperature was introduced using equation 7.16 as a variable that depends on two functions determined by basic heat balance calculations that account for heat gain from the fans (F_1) and heat losses through the box walls in the ambient (F_2) described by equations 7.17 and 7.18 respectively. The influence of the correction factors is presented in the model validation section 7.3.2.

$$CF = F_1 + F_2 \tag{7.16}$$

Where:

$$F_1 = \frac{Fan_{Power\ Input}}{M_{f\ Air} \times C_{P_{AIR}}} \tag{7.17}$$

And

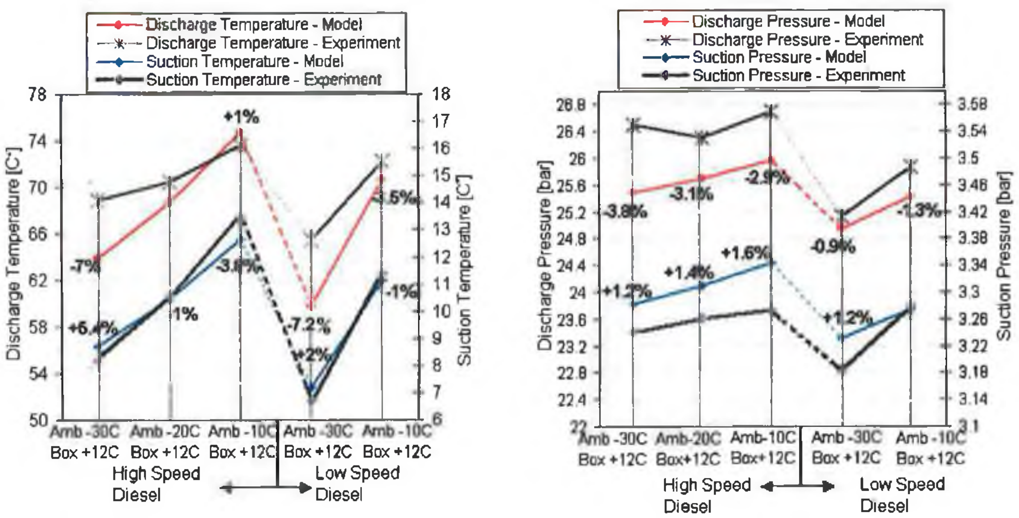
$$F_2 = \frac{k \times S_1 \times (T_{Box} - T_{Amb})}{M_f \times C_{PAIR}} \tag{7.18}$$

7.3.2 Model Validation

Experimental and predicted values for evaporator and compressor inlet-outlet parameters together with the heating performance were compared for five operating conditions presented in Table 6.2. The most significant results that show ±6% accuracy are overviewed in this section in Figures 7.4 through 7.6.

A) Compressor Parameters

Figure 7.4(a,b) shows the comparison between the experimental and predicted values for temperatures and pressures related to the compressor (1). A maximum error of 7% was found for the compressor discharge temperature at very low ambient conditions of –30°C, while the average difference between the measured and predicted values for the compressor suction and discharge parameters is within ±3% for the other test conditions.

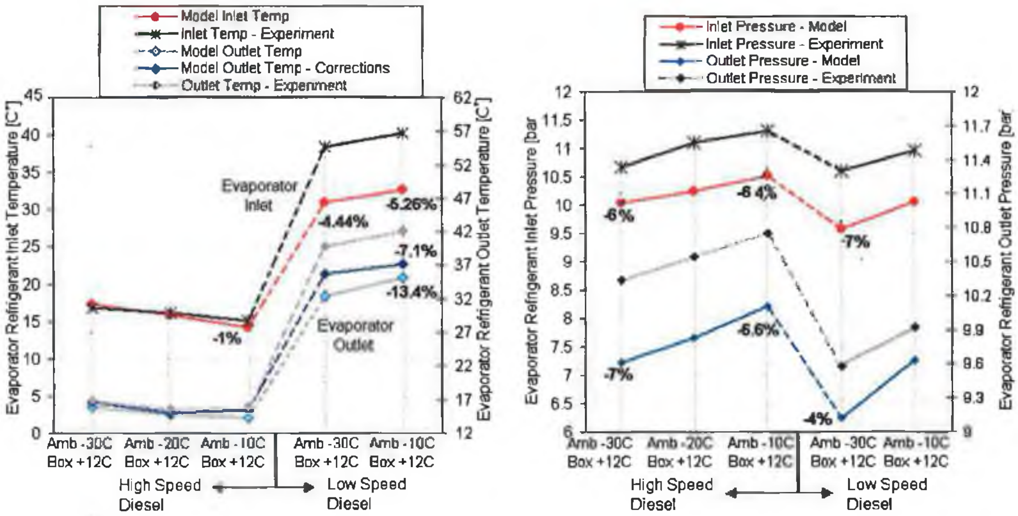


a) Compressor temperatures b) Compressor pressures

Figure 7.4 Compressor Suction and Discharge Parameters – Temperatures, Pressures.

B) Evaporator Parameters

The prediction of the evaporator (5) refrigerant temperatures and pressures are presented in Figure 7.5(a,b). These parameters also show reasonable agreement within ±6% of the experimental data. It must be pointed out that for the high-speed diesel mode the difference between the temperature data just +1%.



a) Evaporator Temperatures

b) Evaporator Pressures

Figure 7.5 Evaporator inlet-outlet parameters; Temperatures and Pressures.

For low speed diesel mode the evaporator inlet temperature differences between the model and the experiment are also good, up to -5.4%, while for the evaporator outlet temperatures increased errors up to –13.4% appear due to the refrigerant mass flow rate influence. The results show a good improvement with predicted figures within just -7.2% differences after the correction factor for refrigerant mass flow rate was applied. For the evaporator inlet-outlet pressures the model predictions are no greater that 6% lower than the experimental values.

C) Heating Capacity Predictions

The most important parameter in the characterisation of the TS 500 unit behaviour in Heat Mode is the capacity of the unit. Figure 7.6 shows a comparison between the measured and predicted results. The measured heating capacities presented in Figure 6.6 are used as the basis for model accuracy in Figure 7.6. The average difference obtained for most of the test conditions and TTC unit operating modes is within 0.5 to 3% with a maximum error of 6.2% for ambient temperatures higher that -12°C.

Figure 7.6 indicates a significant improvement in model prediction that resulted from applying the correction factors. Prediction accuracy improved from a worse case of +11% to a better 8.3% that is very acceptable. It has to be noticed that while the latest difference was obtained at higher ambient temperature of –10°C, for the other conditions the model prediction and experimental results difference is less that 5.4%.

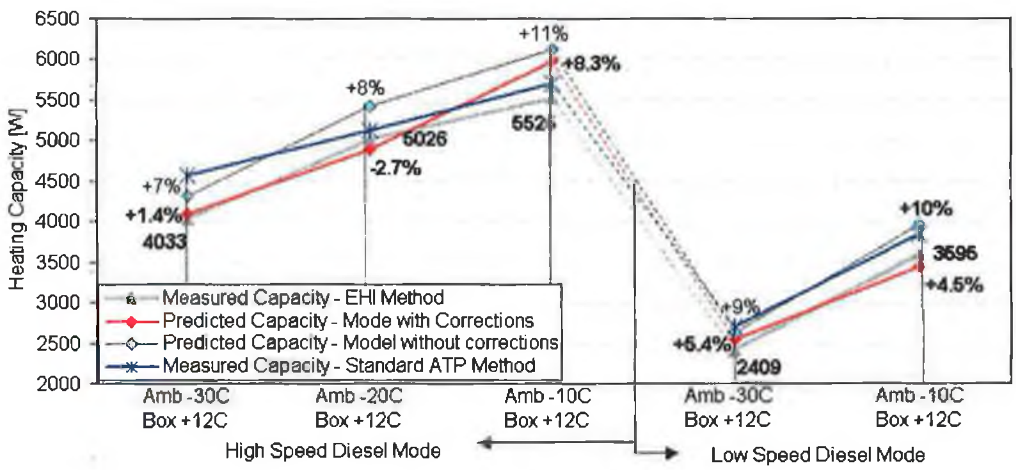


Figure 7.6 Comparison of measured and predicted heating capacity figures.

D) Conclusions – Model Validation

Predictions accuracy was established by comparison with the experimental data:

- Very good agreement within ±3% between the model and experiment for compressor inlet–outlet parameters (Figure 7.4).
- Reasonably prediction accuracy within ±6% for evaporator inlet–outlet pressures and temperatures (Figure 7.5).
- The difference between predicted and measured values of the heating capacity is only up to +5% with an exception for ambient temperature of –10°C where the difference is +8% (Figure 7.6).

7.4 CHARACTERISATION OF STANDARD TS 500 THROUGH MODEL

While the analysis of the standard TS 500 unit in heat mode was restricted in Chapter 6 by the number of experimental tests that it was possible to perform, there is no limit imposed by test conditions to the analysis that can be undertaken using the validated mathematical model.

This latest approach is used in this section to provide a more comprehensive description of the unit behaviour in heat mode.

7.4.1 Efficiency and Coefficient of Performance (COP)

The model offers an opportunity to analyse the system's coefficient of performance (COP), compressor efficiency and the changes in the system response to variations of different inputs like the ambient air and box air temperature, or compressor inlet parameters. While Figure 7.16(a,b) shows both the system efficiency together with the compression work, another example is presented in Figure 7.7 where the optimum performance of the system can be determined for different ambient temperatures using the desired criteria such as: heating capacity, coefficient of performance, compressor efficiency. Clearly, these operating parameters are all a function of ambient air temperature with COP decreasing as temperature increases.

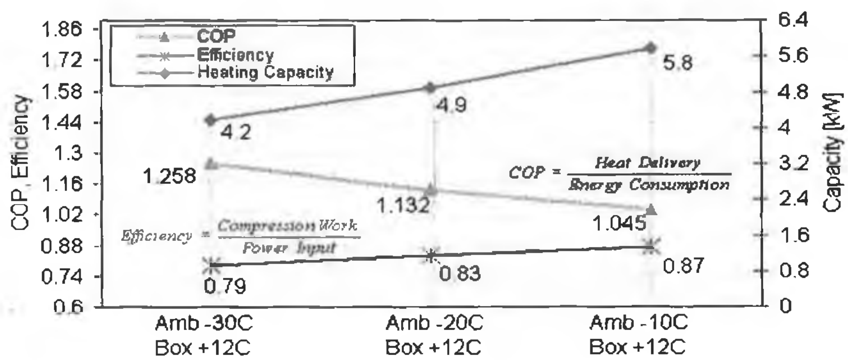
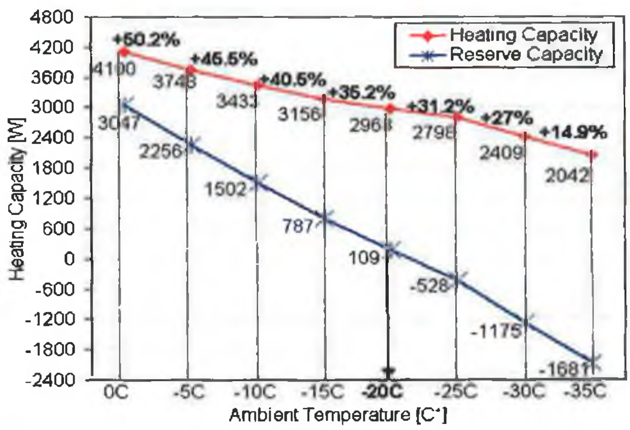


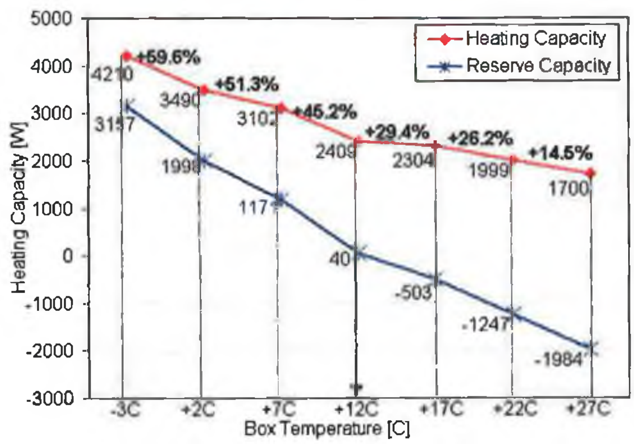
Figure 7.7 Application of the mathematical model to highlight the influence of ambient temperature on the TTC unit's efficiency and COP.

7.4.2 Insufficient Heating Capacity

One of the most important advantages of the mathematical model results from the extended characterisation of the heating performance that it allows. The variation of the heating capacity with different ambient temperatures and different condensing temperatures obtained through prediction is presented in Figure 7.8(a,b) for low speed diesel mode and in Appendix M for high speed diesel mode. It can be noticed that while the decrease in the heating capacity of the unit at ambient temperatures between –10°C and –20°C has an average value of -5%, for ambient temperatures lower than –25°C this difference increases having a dramatic drop of –13.9%. This is happening due to increased heat losses through the walls in the ambient environment as a result of higher temperature difference between the last one and the box air. Figure 7.8(b) shows that the heating capacity of the unit is decreasing with the increase in the condensing temperature. It can be noticed a dramatic decrease in the heating capacity of 13% for condensing temperature higher that +22°C due to maximum heat losses through the trailer walls. Figure 6.10 shows insufficient heating capacity for both high and low speed diesel while the standard TS 500 unit operates at –30°C ambient temperature and +2°C and +12°C box temperature.



a) Box Temperature +12°C.



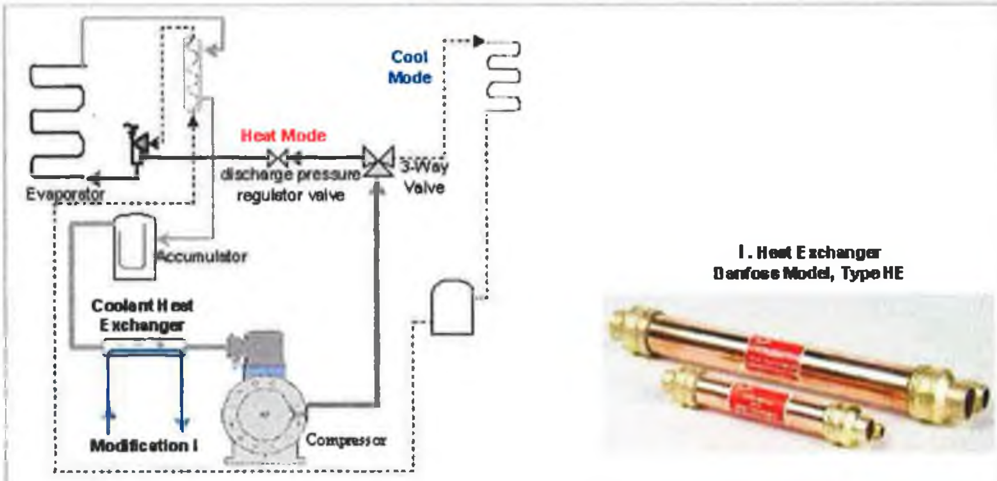
b) Ambient Temperature –30°C.

Figure 7.8 Variation of the heating and reserve capacity with different ambient temperatures and box temperatures – low speed diesel mode.

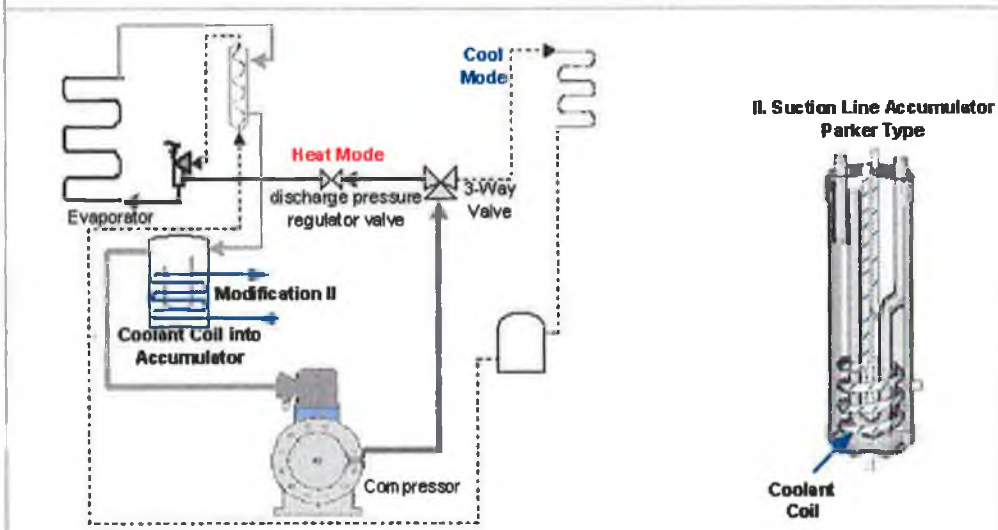
Even if low heating performance was identified as a problem based on the experimental tests, no indication of the ambient and box temperature ranges where the lack in capacity appears was possible. The model predictions are capable of providing supplementary information. Figure 7.8(a) shows that the unit operates at the limit with the reserve heating capacity from ambient temperature lower than -20°C in low speed diesel mode and lower than –25°C for high-speed mode (Figure M.1). Figure 7.8(b) and M.1(b) shows that the unit is running at zero reserve heating capacity from box temperatures higher that +12°C when the ambient is –30°C.

7.5 DESCRIPTION OF THE DESIGN MODIFICATIONS

In light of the system deficiencies identified in section 6.4 and 7.4.2, two design modifications using coolant from the TTC unit engine schematically presented in Figure 7.9 are proposed to optimise the system performance in heat mode.



a) Modification I – Coolant heat exchanger installed on compressor suction line.



b) Modification II – Coolant Coil installed into accumulator.

Figure 7.9 Summary of the two design modifications proposed to be introduces on standard TS 500 production unit to increase the heating capacity and system efficiency in heat mode.

This requires the installation of two new heat exchangers as defined in Figure 7.9(a,b) and described in Appendix M. The proposed modifications are:

i) Modification I: a coolant heat exchanger installed on compressor suction line, which would increase compressor suction temperature.

ii) Modification II: a coolant coil installed into accumulator for increased evaporative and compressor suction temperatures.

The modifications schematically presented in Figure 7.9(a,b) are proposed based on the assumptions that an increase in both the compressor suction and evaporating temperature yields to a positive impact on the unit performance in heat mode (section 5.5.2). It was also based on the fact that the temperature difference of 85°C between the engine coolant outlet temperature and compressor suction temperature has a great potential to facilitate an efficient heat exchange. The predicted impact of these design changes was accessed through mathematical models II and III.

7.5.1 Design Modification I – Description of the Mathematical Model I

A Danfoss type "shell-and-tube" heat exchanger is proposed to be installed on the compressor suction line offering a second stage evaporation to generate an increase in the compressor suction temperature. Based on the experimental results presented in section 5.4 that showed the positive impact of these parameters, the main benefits expected are increased heating capacity and system efficiency. The performance of the modified unit is estimated based on the predictions from mathematical model II that is based on the algorithm of model I as described in the next sections.

A) Mathematical Model II – User Interface

The main interface of this mathematical model has the shell and tube coolant heat exchanger installed on the compressor suction line as presented in Figure 7.10. The user has to introduce the same inputs: ambient and box test temperatures and just press the "Calculate" button for the heat exchanger, which opens a new sub-panel shown in Figure 7.11. The capacity calculations of the new heat exchanger and compressor suction temperature are obtained using this second interface (Figure 7.11). Based on this tool the user can calculate the performance of different types of heat exchangers at different coolant inlet temperatures.

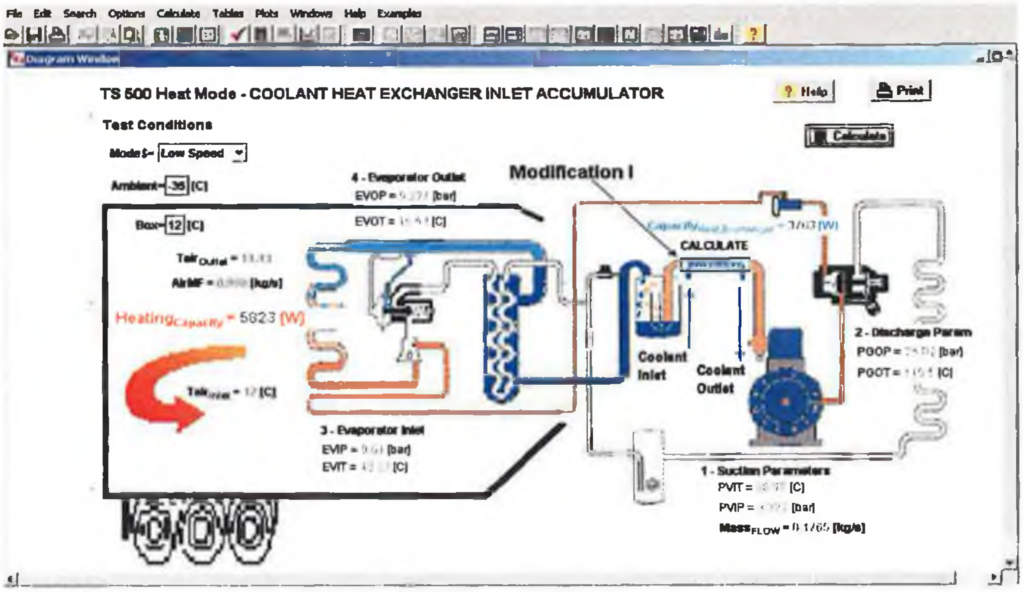


Figure 7.10 Main Interface of the mathematical model predicting the effect of Modification I – Coolant heat exchanger installed on compressor suction line.

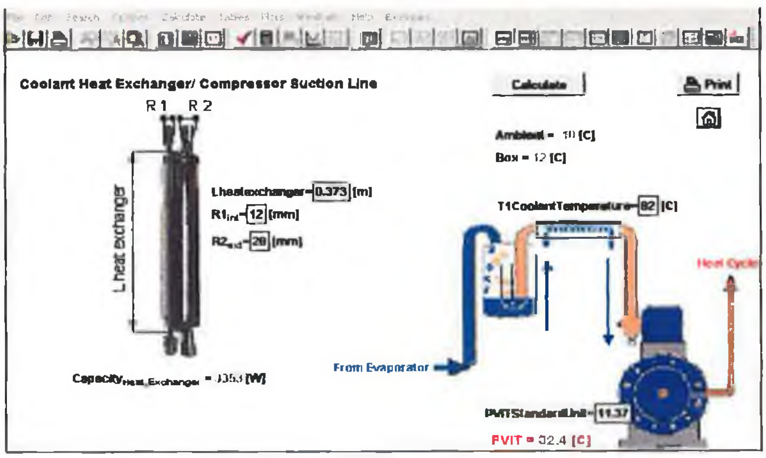


Figure 7.11 Interface of mathematical model II that contain the Coolant Heat Exchanger Calculations.

B) Mathematical Model II - Algorithm

The model created for predicting the effect of the coolant heat exchanger on the heating capacity and system behaviour is based on the simulation presented in Section 7.2 and 7.3. The new elements introduced in the model are heat balance calculations for a "shell-and-tube" heat exchanger, installed on the compressor suction line. The heat transfer calculations for the "shell-and-tube" heat exchanger are presented in equations 7.19 to 7.21 [Carabogdan *et al.*, (1983)]. The heating capacity of the heat exchanger is presented by equation 7.19, while the heat balance for the heat exchanger is described by equation 7.20. The increased compressor suction temperature obtained as a result of the heat exchange with the hot coolant is calculated based on equation 7.21. The EES algorithm is presented in Appendix N.

$$Q = \frac{(T_C - T_V) \times Length_{HE}}{\frac{1}{2 \times 3.14 \times \lambda} \times ln \frac{R_2}{R_1}} \quad [W] \qquad [7.19]$$

$$Q = m_c \times Cpc \times (T_{ci} - T_{co}) = m_v \times Cp_v \times (T_{vo} - T_{vi}) \quad [W] \qquad [7.20]$$

$$T_{v\,icomp} = T_{v\,icomp\,in} + \frac{Q}{m \times C_p} \quad [C] \qquad [7.21]$$

7.5.2 Design Modification II – Description of the Mathematical Model III

Figure 7.9(b) shows the proposed Parker type accumulator. The coolant flows through the 12mm coil into the accumulator to facilitate increased evaporating temperature due the heat transfer to the refrigerant liquid–gas mixture. Based on the experimental results presented in section 5.4 for Modification IV, where the hot-gas was used instead the engine coolant to produce a net positive impact on the unit heating performance and a similar result is expected as the engine coolant has an 85°C higher temperature than the refrigerant gas. The performance of design modification II is predicted using a third mathematical model created as an extension of the steady-state model presented in sections 7.2 and 7.3.

A) Mathematical Model III – User Interface

Figure 7.12 shows the main interface of mathematical model III created for the effect of the coolant coil installed into the accumulator. The user introduces the same inputs: ambient and box temperatures and just pressing on the area designated for the accumulator, a new window shown in Figure 7.13 opens. This latest interface is the user's tool to calculate the heat exchange capacity of the coil installed in the accumulator, introducing two parameters: the coolant temperature and the compressor suction temperature (PVIT) of the standard unit. After presses the "Calculate" button, the user obtains the heat exchange capacity of the accumulator and the new increased compressor suction temperature. Using the main interface the new parameters of the modified unit are obtained.

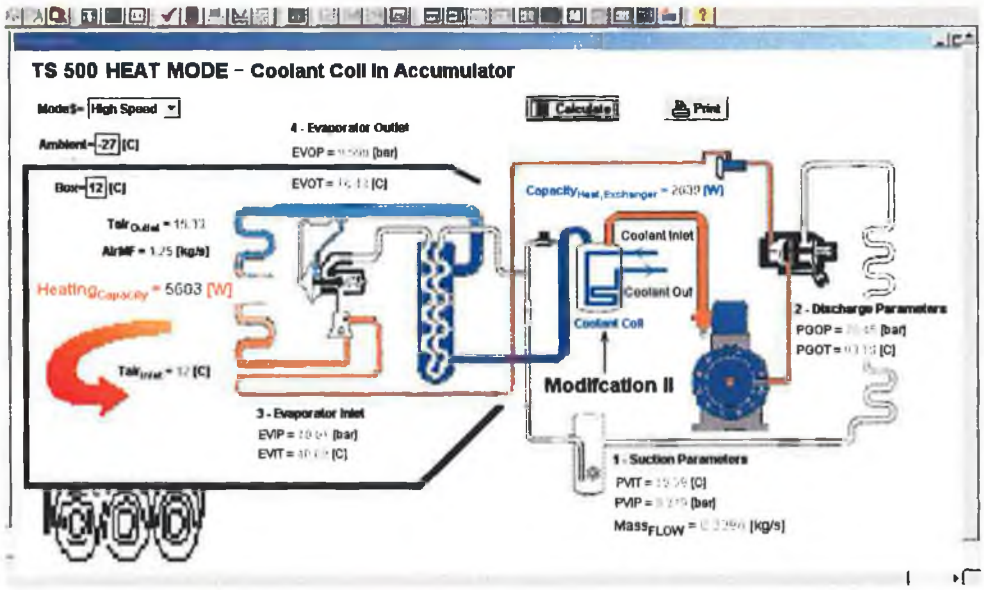


Figure 7.12 Main Interface of mathematical model III for predicting the effect of Modification II – Coolant coil installed within the accumulator.

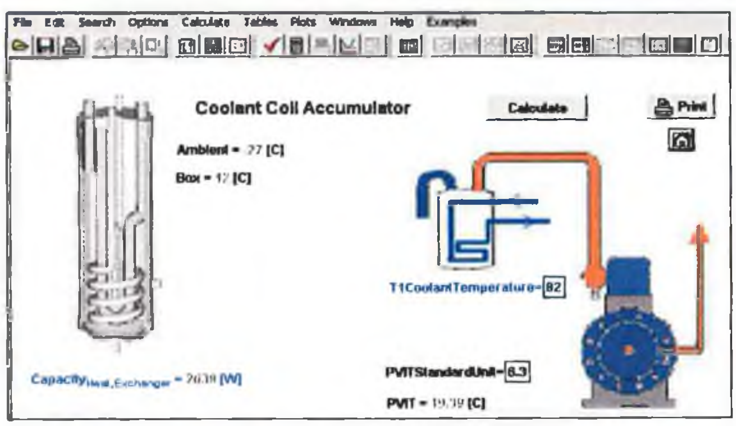


Figure 7.13 Interface that contains the coolant coil heat exchange calculations.

B) Mathematical Model III - Algorithm

The model created for the prediction of the effect of the coolant coil on the heating capacity and system behaviour is based on the simulation presented in Section 7.2 and 7.3. The heat transfer calculations and the compressor suction temperature are based on the same equations 719 to 7.21 [Carabogdan *et al.*, 1983].

7.6 PREDICTED BEHAVIOUR OF THE MODIFIED TS 500 UNIT

A comparison of the measured and predicted heating capacity for both the standard and the modified TS 500 unit is presented in this section. The following results are analysed for the units operation in high and low speed diesel modes:

- Effect of Modification I and II on the heating capacity.
- Impact of Modification I and II on the compressor behaviour.

7.6.1 Heating Capacity

The measured and predicted heating capacities for both the standard and modified TS 500 unit are presented in Figure 7.14. Predicted results for Modification I provide an increased heating capacity of to 46.6% for high-speed diesel and up to 60% for low speed mode. Figure 7.14 also shows the effect of modification II on the heating performance. These results indicate that the introduction of the coolant coil within the accumulator has lower impact on the heating capacity, but still increasing the capacity by 32% in high-speed diesel mode and 47.7% in low speed diesel mode. The impact of both modifications is greater at very low ambient temperatures due to the increased heat exchange that takes place between the coolant and the suction refrigerant vapor. It is observed that the influence of both modifications is greater for low speed mode.

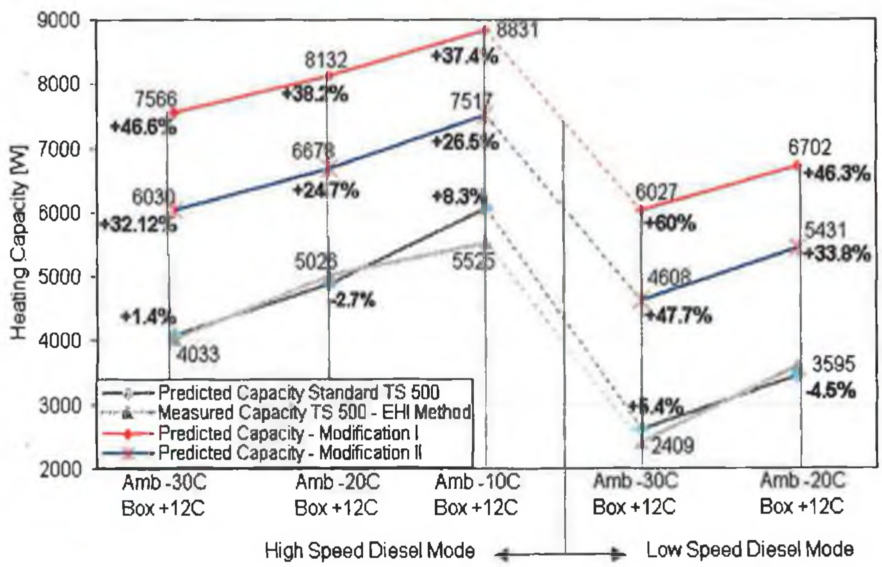


Figure 7.14 Comparison between the impact of modifications I and II of the heating capacities of the standard and modified TS 500 units for high and low speed diesel modes.

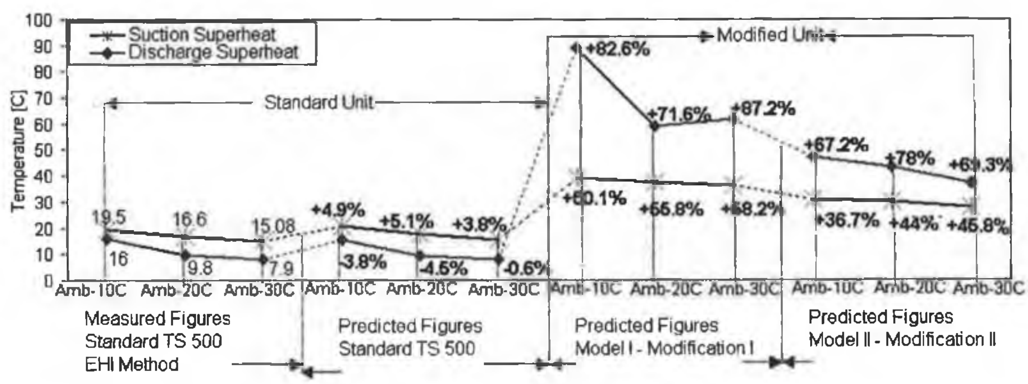
Modification I has an average +10% greater impact on the heating performance than modification II in high speed mode, while in low speed diesel mode has an overall 20% greater impact than modification II, which still delivers a 34% improvement.

7.6.2 Compressor Behaviour

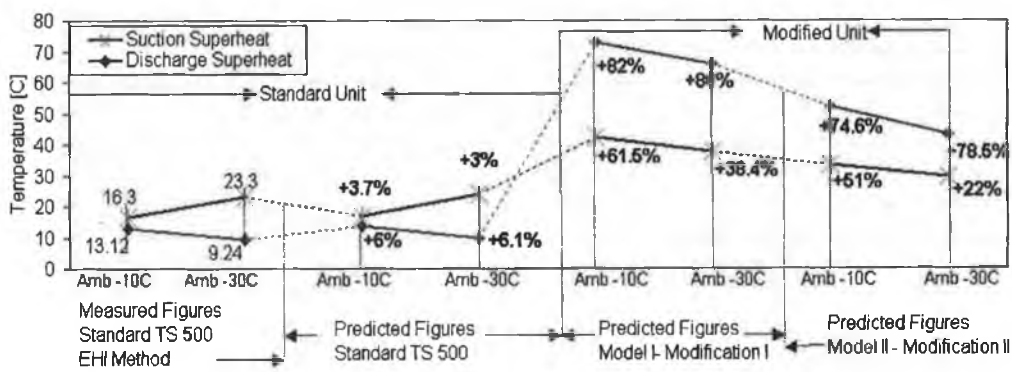
The compressor behaviour is characterised through: i) compressor suction and discharge superheat and ii) compression work together with the compressor efficiency. This latest indicator is calculated as the ratio between the compression work [W] and the power input [W] as defined in Figure 7.7.

A) Compressor suction and discharge superheat

Figure 7.15(a,b) shows the compressor suction and discharge superheat for the standard versus modified TS 500 unit operating in high and low speed diesel modes. The comparison is based on test results and mathematical model predictions.



a) Compressor suction and discharge superheat – high-speed diesel mode.



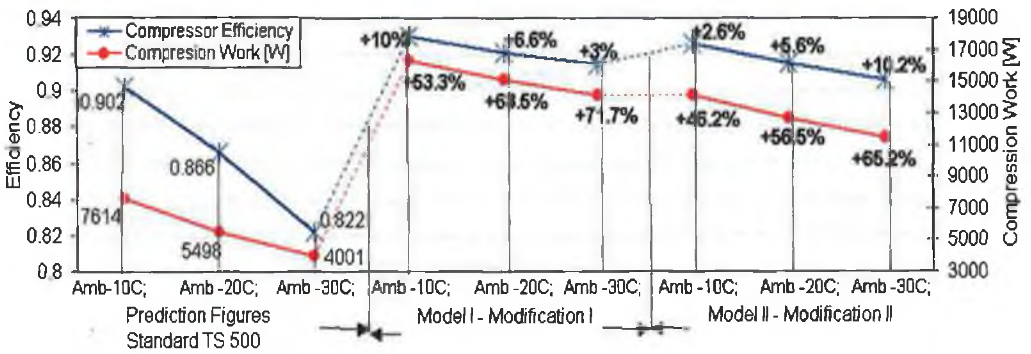
b) Compressor Suction and Discharge superheat – low speed diesel mode.

Figure 7.15 Comparison between the impact of Modifications I and II on the compressor suction and discharge superheat for the standard and modified TS 500 unit in high and low speed diesel modes.

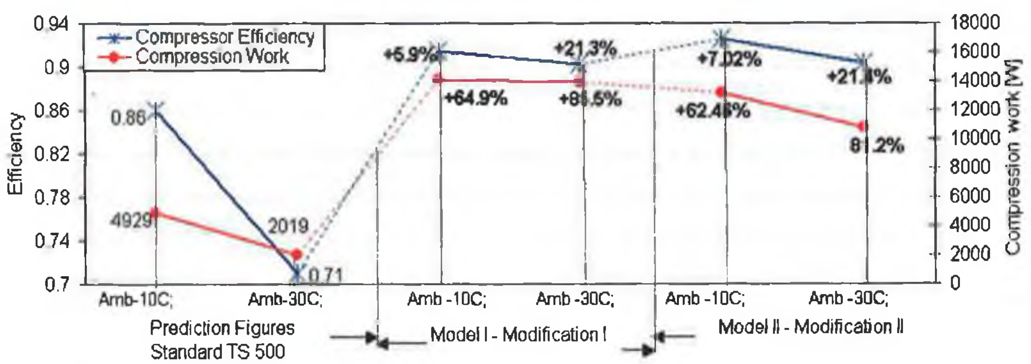
An increase of maximum 87% in the compressor suction superheat and a maximum of 39% increase in the discharge superheat due to the impact of modification I, while the unit operates in high-speed diesel mode. Figure 7.15(b) shows 86% and 61% increase in the compressor suction and discharge superheat respectively due to modification I. The coolant coil installed on the compressor suction line has increased effect on compressor superheat figures above modification II by an average of 9% and 12% for both operation modes. The differences between the predicted and measured figures are very small, with a maximum value of only 6%. The impact of the heat exchanger installed on the suction line has up to 15% increased effect on compressor discharge superheat figures while a smaller difference of 9% in compressor suction superheat is evident between both modifications in high speed diesel mode. Figure 7.15(b) presents the effect of both modifications on the same parameters while unit operates in low speed diesel. The impact of modification I has up to 7.16% increased effect on compressor discharge superheat than modification II, while less difference up to 9.6% is obtained for compressor suction superheat.

B) Compressor work and efficiency

Figures 7.16(a,b) shows the compression work and efficiency changes for the same temperature conditions. The compressor efficiency for the modified unit are between 2.9% and 10.3% higher than the figures obtained for the standard TS 500 unit in high-speed diesel, while in low speed it is between 5.9% and 21.4% higher. However, Figure 7.16 also shows a significant increase in compression work for the modified unit over the standard unit of between 53% and 71% in high-speed diesel mode and between 64% and 85% in low speed. Modification I has a greater impact not only on the efficiency, but also on the compression work. Both modifications have a greater impact on both indicators at low ambient temperatures. When the unit operates at –30°C ambient it is observed that the efficiency is 18.3% higher in low than in high-speed mode. This can be explained by higher compressor power consumption due to increased mass flow rate and compression ratios in high-speed mode. Both coolant heat exchanger and coolant coil in accumulator have the same effect on compressor efficiency with small differences up to 0.7%. The coolant heat exchanger installed on compressor suction line has increased impact on compression work up to 6.6%, while the unit operates in high-speed mode. For low speed, modification I has greater impact up to 5% for compression work while the differences between compressor efficiency are small, at just 1.2%.



a) Compression work and efficiency – high-speed diesel mode.



b) Compression Work and Efficiency – low-speed diesel mode.

Figure 7.16 Comparison between the impact of Modifications I and II on the compressor efficiency and compression work for the standard and modified TS 500 unit in high and low speed diesel.

7.6.3 Discussion of Results – Benefits of Modifications I and II

A discussion of the main parameters that influence the heating performance of the modified TS 500 unit together with a summary of the results obtained based on modification I and modification II is overviewed.

A) Summary of results:

- For high-speed diesel mode the impact of the coolant heat exchanger installed on the compressor suction line has an overall 14% greater impact on the heating performance of the system than the coolant coil installed in the accumulator, which still delivers a 32% increased heating capacity at –30°C ambient and +12°C box temperature conditions.

- The impact of the heat exchanger installed on the suction line has up to a 15% increased effect on compressor discharge superheat figures while there is less difference for compressor suction superheat up to 9% between modifications.
- Both the coolant heat exchanger and coolant coil in the accumulator tank have the same effect on compressor efficiency with small differences up to 0.7% for high and low speed modes.
- The coolant heat exchanger installed on compressor suction line has increased impact on compression work up to 6.6% for high-speed diesel mode while only 1.6% can be noted for low speed diesel.

B) Theoretical Considerations

The good behaviour of the unit in heating obtained with both design modifications is based on the same effects overviewed in Chapter 5, section 5.3 except that the hot gas is replaced with the engine coolant. The main parameter changes that improve the unit behaviour in heating are summarised as follows:

- Increased evaporative saturation temperature.
- Higher vapor mass flow rate obtained in the accumulator.
- Increased compressor suction temperature.

7.7 IMPROVED PERFORMANCE OF THE MODIFIED TS 500 UNIT

The problems that appear in heat mode for the standard TS 500 unit are overviewed in Chapter 6 based on experimental results and field data. Three main types of problems for heat mode behaviour were emphasised and are summarised as follows:

- Insufficient heating capacity.
- Long pull-up time.
- Poor set-point temperature control.

A series of design modifications were proposed and accessed in this chapter using a mathematical model not only for improved unit behaviour in heat mode but to verify and define for the first time the influence of parameters such as the compressor suction temperature, superheat, evaporating and condensing temperatures on heat mode performance. It was shown that the compressor suction temperature and superheat was the most influential parameter. A comparison between the standard and modified units regarding the heating capability and the compressor behaviour is accessed in this section.

A) Heating Capacity

In sections 6.4.2, 7.5.2 and Appendix I it was concluded that the TS 500 unit lacks the heating capacity to achieve box temperatures higher than +12°C for sub –20°C ambient temperature. Figure 7.14 shows the predicted increased heating performance of the TS 500 unit with modifications I and II. While the new heating capability of the modified unit versus the TS 500 was presented (Figure 7.6), the new reserve heating capacity obtained through coolant modifications appears in Figure 7.17.

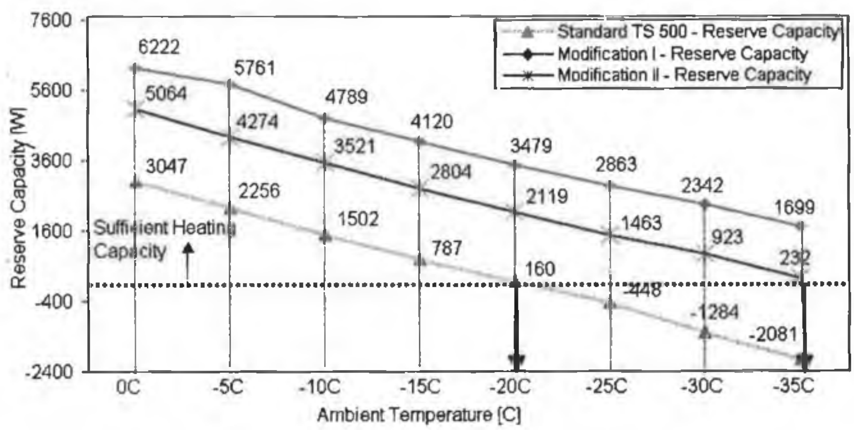


Figure 7.17 Heating and reserve capacities for standard versus modified TS 500 unit operating in low speed diesel mode at +12°C box temperature.

While the reserve capacity of the standard unit is lower than zero for ambient temperatures sub –20°C due to insufficient heating capability, modification II provides a better behaviour, heating at +12°C food compartment temperature set-point for a minimum –35°C ambient temperature. Based on the weather data for Northern Canada presented in Figure 1.7 where the temperatures during winter are –40°C, it can be concluded that this set-point cannot be maintained even with this modification installed on the unit. As a result, Modification I is proposed as the most effective solution that provides sufficient heating capacity even at such extreme ambient temperature as –40°C. The improved behaviour of the modified TS 500 unit also eliminates the field problems presented in Figures 6.8 and 6.9 and the long pull-up time and poor temperature control presented in Figures 6.11 and 6.12.

B) Compressor Behaviour

Figure 6.14 highlights good compressor suction superheat for ambient temperatures higher than –25°C, while at –30°C the superheat is very low at just 7.8°C. Due to the effect of the coolant on the suction temperature, Figure 7.15 shows a 58% increase in the superheat, assuring an improved response from the unit.

7.8 SUMMARY

The steady-state model proposed in this chapter represents the first precise tool for predicting the heating capacity and the system behaviour in heat mode [Radulescu *et al.*, 2004]. The model algorithm has been developed in a modular structure so that further modifications can easily be introduced. This chapter outlines not only the model algorithm but also validates its prediction accuracy. The indicators show that the model has the average predictive accuracy of ±6%, better than the typical accepted accuracy general discrepancy of ±10% produced by existing models of either TTC units in cool mode or heat pumps [Koury *et al.,* (1999); Sakellari *et. al.*, (2003)]. This improved accuracy was achieved by introducing a novel correction factor to allow for the influence of ambient temperature and the heat from the fans inside the food compartment (calibrated box). As a result of a complete analysis of the standard TS 500 unit performed through both experimental tests (section 6.4) and predictions (section 7.5) several problems of the unit in heat mode were identified, all resulting from insufficient heating capacity. In light of the lack in capacity, two design modifications based on the engine coolant effect are defined and accessed using two dedicated mathematical models created as extensions of the first one. The best predictions for the modified system show a spectacular increase of 46% using the coolant "shell-and-tube" type heat exchanger proposed to be installed on the compressor suction line. It has to be noted that the modifications proposed and tested for the SL 400e unit can have similar positive effect on the TS 500 truck unit heating capacity that was estimated to increase by 30%. However, the engine coolant used in a heat exchanger installed on the compressor suction line is recommended as it has higher impact of 46% increase in the heating capability that is more necessary than for a trailer TTC unit, with still low costs of 80 Euros per unit, corresponding to 2% increase of the total market cost of a TS 500 system [Thermo King (2004)].

CHAPTER 8

CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

CONTENT:

8.1 CONCLUSIONS

This study addressed the challenge faced by designers of Transport Temperature Control (TTC) units to produce systems capable of achieving and maintaining the desired air temperature during the transportation of perishable food products, particularly in cold climates where sub-zero degree Celsius ambient temperatures exist for seven months per year. When perishable fresh food products with recommended set-point temperatures above 0°C are transported at ambient temperatures below 0°C, the TTC unit must deliver heat. This research sought to: i) establish the importance of heat mode, ii) develop a novel test procedure and facility for accurate heating capacity measurements, iii) characterise the TTC unit behaviour in heat mode and quantify the technical problems that appear at sub 0°C ambient temperatures, iv) propose design changes for two types of TTC units and v) determine the impact of these modifications on the heating capacity and system efficiency through both experimental methods and predictions obtained from the first mathematical model. The main conclusions of this research are listed below.

- A review of the perishable food products transportation industry and TTC unit markets has shown that:
  i) 80.3% of all food products being distributed can be classified as fresh and chilled, demanding temperatures above 0°C.
  ii) 24% of the TTC system market is in Northern Hemisphere countries above 55° latitude where the climate and food types being distributed demand the use of heat mode for 66% of the year. Meanwhile this drops to 40% of the year in Central European countries that consume 47% of the TTC unit market.

  Based on the increased demand for further analysis of the TTC unit behaviour in heating, field data from a Swedish food distributor was used [Nilsson (2002), (2003)]. It was concluded that the main dysfunctions during heat mode operation were: i) insufficient heating capacity, ii) poor temperature control, iii) long pull-up time and iv) low compressor suction superheat. Considering the contradiction between the existing manufacturer specifications for the heating capacities that show enough reserve capacity at very low ambient temperatures and the fact that the main cause of the problems mentioned above is a lack of sufficient heating capacity, further research on the existing test procedures and test facilities currently being used to evaluate the TTC units heating capacity was required in order to establish the measurement precision.

- While the existing test methods for measuring cooling capacity deliver results accurate to ±3%, no similarly precise test method exists for heating capacity. The only test method for performance characterisation of a TTC unit in heat mode has an accuracy of ±10% [ATP (1970)] and, obviously, compares poorly with the experimental cooling capacity measurement method [±3%]. As a result, a new Electric Heat Input (EHI) test procedure was proposed, as it provides results accurate to ±3% [Radulescu *et al.*, (2003)]. The EHI method described in Chapter 3 is based on the measurement of power input from electric heaters and can provide results that account for additional heat input from the evaporator's fans and non-uniform airflow through the evaporator, both of which are limitations of the existing method. Based on the test results obtained for the SL 400e unit presented in Figures 4.6 and 4.7 it can be noted that the heating capacity measured using the new EHI method are up to 19% lower than that figures obtained using the standard test method. These differences can be explained due to the fact that the calculations for air outlet temperature were performed without accounting for the heat inputs from the air circulating fan motors that influences the results for high ambient temperatures, or for heat losses from the warm box to the colder external surroundings that introduce errors for very low ambient temperatures.
- Responding to the need to improve system performance in heat mode, the first full characterisation for two different types of single-compartment TTC units was performed using the new facility specially commissioned to be able to conduct the new accurate EHI test method [Radulescu *et al.*, (2003)]. To verify the problems highlighted by the field data, heating capacity, pull-up and temperature control tests were performed. As a result, the key problems in heat mode were highlighted for both units: i) low heating capacity as presented in Figures 4.11 and 6.10, ii) long pull-up time (Figures 4.12 and 6.11), iii) poor set-point temperature control (Figures 4.13 and 6.12), iv) poor reliability in heat mode due to the charge migration in the cooling cycle

(Figure 4.14) and v) low compressor suction superheat, which results in the compressor failures identified in Figures 4.15 and 4.16.

- In light of the deficiencies identified, a series of design modifications are defined and accessed in Chapters 5 and 7. Five design changes were implemented on the standard SL 400e trailer unit in an attempt to obtain increased heating capacity, system performance and higher efficiency in heat mode. These modifications are depicted in Figure 5.1 and include: i) Modification I – Increased diameter of the hot-gas line, ii) Modification II – hot-gas injection into the compressor suction line, iii) Modification III – hot-gas injection into the accumulator inlet, iv) Modification IV - hot-gas coil into accumulator together with v) Modification V – a new charge control system. A comparison between the standard and modified SL 400e units shows an average 25% increased heating capacity, 30% higher system efficiency obtained with 7% lower fuel consumption as summarised in Tables 5.1 and 5.2. The impact of the design changes has been discussed based on the combined effect of: i) increased evaporative and compressor suction temperatures, ii) lower compression ratios, and iii) higher mass flow rates. The influence of these parameters on the heat mode performance of a TTC unit operating at ambient temperatures below –10°C is identified for the first time in this work and simple, yet viable solutions are proposed to address the performance weaknesses identified [Radulescu *et al.*, (2005)].
- While mathematical models of these systems in cooling mode exist, no such model existed for heating mode. Chapter 7 presents the first mathematical model capable of predicting the heat mode behaviour to an accuracy of ±6% [Radulescu *et al.*, (2004)]. In light of the system deficiencies identified for the standard TS 500 unit through experimental and field data together with more comprehensive predictions of the simulation, two design modifications based on the effect of introducing engine coolant used as a primary heat exchange fluid are analysed through two modified mathematical models. It can be concluded that for high-speed diesel mode the impact of the coolant heat exchanger installed on the compressor suction line has an overall 20% greater impact on the heating performance of the system than the coolant coil installed in the accumulator, which still delivers a 34% improvement. In low speed diesel mode it can be noted that a 14% higher heating capacity can be

delivered with the heat exchanger installed on compressor suction line. The impact of the heat exchanger installed on the suction line has up to a 15% increased effect on compressor discharge superheat while there is less difference for compressor suction superheat up to 9% between improvements (Figure 7.14 to 7.17). These steady-state models described in Chapter 7 represent the first precise mathematical tool for predicting the heating capacity and characteristics of a TTC system in heat mode with an accuracy of ±6%.

8.2 RECOMMENDATIONS FOR FUTURE WORK

This work may be continued in two directions: the first involves a further experimental investigation of the hot-gas injection and the hot-gas coil included in the accumulator on the temperature control of the unit and relates to system performance in defrost mode. During temperature control at the set-point temperature, the unit runs simultaneously in heat and cool mode. Due to the system inertia it takes a long time until the whole charge from the cooling cycle is recuperated in the heat cycle. Using the hot-gas injections at the compressor and accumulator inlet, instead of redirecting the refrigerant during the cool cycle, generates higher suction temperature when the unit operates completely in cooling. A trial system should be tested in the field during Winter 2005/2006 and compared with the performance already recorded.

The defrost mode is identical with the heat mode except that, during defrost operation, the damper door is closed. To improve the defrost response of the unit, the research focuses on two aspects: i) on the defrost control algorithm and the need to determine the optimum defrost time intervals and criteria for the defrost initiation and ii) the need to operate with very short defrost durations, for a better temperature control in the trailer [Fahlen (1999); Payne (1992)]. The increased refrigerant mass flow rate through the evaporator can have an important effect in the latest problem, which might lead to 30% shorter defrost time (Figure 5.9).

Figure 5.12 shows the effect of the modifications implemented on the SL 400 unit on the cooling capacity. A significant increase in the refrigeration effect is obtained as a result of higher evaporative conditions together with the effect of higher compressor suction superheat. Further research regarding the impact of these modifications on the unit performance in cooling will be conducted.

CHAPTER 9

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LIST OF INTERNATIONAL WEATHER DATA SOURCES

Finland – The Finnish Meteorological Institute (www.fmi.fi)

France – Meteo France (www.meteo.fr/e_index.html)

Hungary – Hungarian Meteorological Service (www.hnms.gr)

International Research Institute for Climatic Prediction (http://iri.columbia.edu/)

Romania – National Institute of Meteorology and Hydrology (www.inmh.ro)

World Meteorological Organization (www.wmo.ch)

World Data Center for Meteorology (www.ncdc.noaa.gov/wdcamet.html)

World Radiation Center (wrdc-mgo.nrel.gov)

http://www.climateriskunit.com/forecast.html

http://www.met-office.

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http://lwf.ncdc.noaa.gov/oa/climate/research/anomalies/anoma

http://lwf.ncdc.noaa.gov/oa/climate/research/anomalies/anomalies.html

APPENDICIES

CONTENT:

APPENDIX A: PERISHABLE FOODS TRANSPORTATION MARKET

The total food production in 1997, together with all categories of food products recognised by the Food and Agriculture Organisation of the United Nations in the latest statistic published (1998), is presented in Figure A.1.

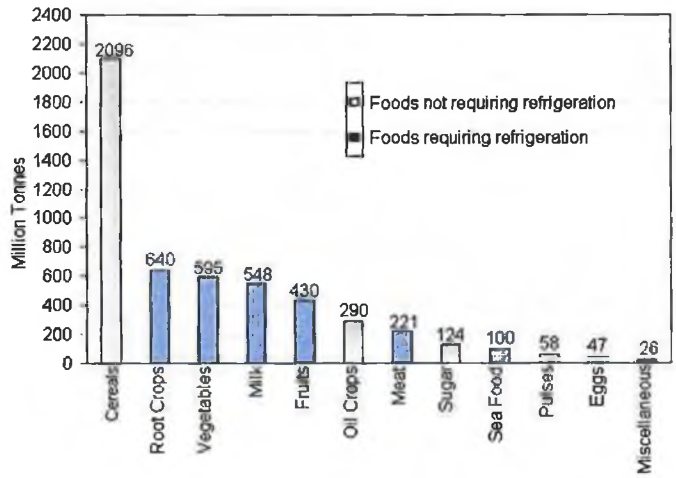


Figure A.1 Global Food Production 1997 [FAO (1998)].

Guilpart (2003) presents more recent data for the food consumption in Europe, focusing only on products that need temperature control during storage and transportation [Figure A.2]. It can be concluded that 10% from the food product consumption consists of frozen perishable, while 90% consists of fresh food commodities that need transportation temperatures above 0°C.

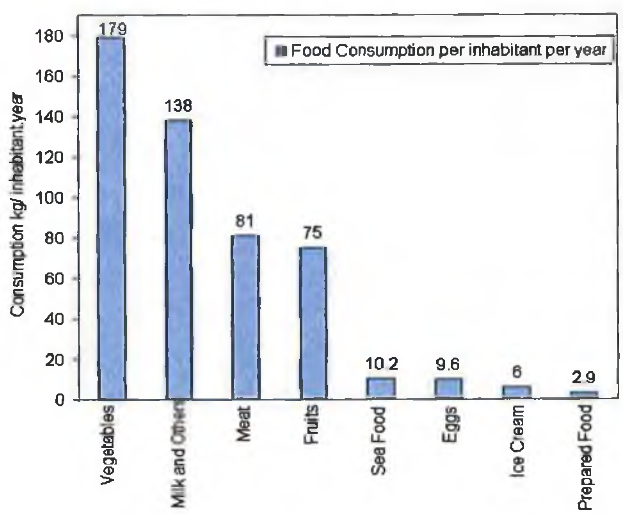


Figure A.2 European food consumption per inhabitant per year 2002 [Guilpart (2003)].

APPENDIX B: TEMPERATURE CONTROL ALGORITHM AND TECHNICAL SPECIFICATIONS FOR TTC UNITS

While section 1.1.2 provides general considerations regarding the TTC unit components and thermodynamic process, Appendix B presents more detailed technical specifications, together with the typical temperature control algorithm based on alternative cool and heat mode operations.

B.1 Technical Specifications of TTC units

On refrigerated semi-trailers the compressor is usually driven by a two-speed diesel engine, which also drives the condenser and evaporator fans. Some units also include an electric motor drive as a standby for use where noise regulations are imposed. Guilpart 2003 presents a comprehensive review of the number and technical characteristics of existing road transportation TTC units, based on the most recent United Nations Economic Committee (UNECE) report (1998). The transport temperature control units are constructed within different sized vehicles, depending on the food being distributed and the distance travelled. These ranged from car-sized vans to 20 m long semi-trailers, and Table B.1, defines the three classes that exist, along with typical cooling capacity.

Table B.1. TTC Units – Classification and Technical Data [Guilpart (2003)].

Refrigeration System Type	Capacity [W]	Charge [kg]
Direct drive vans	500 to 3000	2
Trucks	2500 to 5500	5
Trailer and semi Trailer	6000 to 10000	10

The cooling capacities presented are achieved using different refrigerant as working fluids, defined in Table B.2. The total refrigerant charge that corresponds for the total number of TTC units employed is also identified.

Table B.2. Total refrigerant charge evaluation [Guilpart (2003)].

TTC System Type	Refrigerant	Total Refrigerant [Tonnes]
Direct Drive Vans	CFC (R12 and R502)	50
Trucks	HCFC (R22 and blends)	800
Trailer and semi Trailer	HFC (R134a and R404a)	1300

The insulation level of the refrigerant compartment is classified in Table B.3, and the values presented conform to the specifications introduced by the Agreement on the International Carriage of Perishable Foodstuffs and on the Special Equipment to be used for such Carriage [ATP (1970)].

Table B.3. Insulation of the bodies [Guilpart (2003)].

Insulation Type	Thermal Conductivity k [W/m^2K]	Thickness [mm]
Heavily insulated	< 0.4 W/m2K	70 - 80
Normally insulated	0.4 < k < 0.7	45 - 50

B.2 Temperature Control Algorithm

On many diesel driven TTC units, the compressor, condenser fan and evaporator fan are connected to a common drive train, consequently the evaporator fan speed reduces when the compressor goes to low speed and the reduced airflow allows the temperature gradient across the load to increase. Control cycles of this type can cause chilling and freezing injury of sensitive fruit and vegetables. The main problem is the practice of controlling the return air temperature combined with relatively wide control swings. Where parts of load are at several degrees above the set-point, the thermostat may cause the compressor to run on full cool and thus freeze other parts of the load [Emond *et al.*, (2003)]. The temperature control vary from simple on/off, which is used on many road vehicles at all temperatures and for frozen control on marine containers, to sophisticated capacity regulation using electronic control of chill temperatures on TTC units (Figure 4.3, Item 6). Table B.4 presents a typical temperature control algorithm when the unit operates continuously, in so-called "Continuous Mode". [Frith (1991)].

Table B.4 Temperature Control Algorithm [Frith (1991)].

Temperature Control Reference	Temperature [°C]	Unit Mode
Return Air	Set Point + 2°C	High Speed Cool
Return Air	Set Point + 1°C	Low Sped Cool
Return Air	Set Point – 1°C	Low Speed Heat
Return Air	Set Point – 2°C	High Speed Heat

B.3 Temperature Control of a Multi-Compartment TTC unit

While in Chapter 1, Figure 1.5(a,b) the typical temperature control was described for a single-compartment TTC unit, Figure B.1 shows the temperature control pattern for a multi-compartment truck TTC unit that consists of two thermally insulated temperature-controlled zones. This is achieved using one host evaporator and one remote evaporator, both of them operating in cool mode but attempting to achieve different temperature conditions: 0°C for the "back" compartment and –20°C in the "front", while the ambient temperature is +30°C.

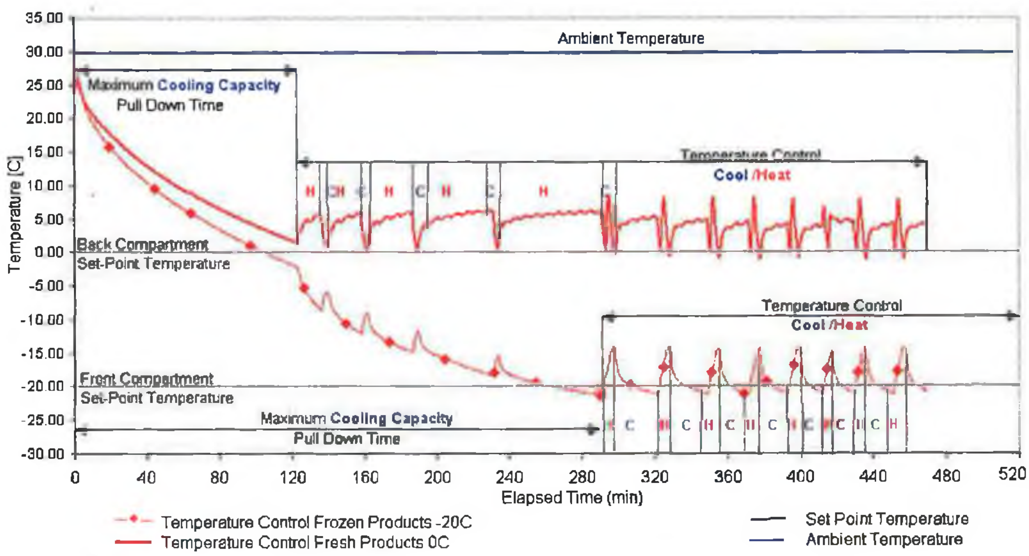


Figure B.1 Traditional Concept of a Multi-Compartment TTC Unit requested to achieve 0°C and –20°C set-point temperatures in high external ambient temperature conditions.

It can be noticed that for the back compartment that is transporting chilled products at 0°C the unit has a good pull-down time of 2 hours, but a poor temperature control within the 5°C temperature range. For the back compartment where the unit is maintaining –20°C for frozen products, it can be concluded that the pull-down time is very long, almost 5 hours. The temperature control is again poor, ranging between –21°C and –16°C.

APPENDIX C: TRANSPORT TEMPERATURE REQUIREMENTS

C.1 Food transport temperature regulations

In pursuit of common set of rules relating to the production, marketing and distribution of food, the European Council of the United Nations had issued several directives regarding the quality for fresh, chilled and frozen products summarized in

i) Chilled meat directive (64/433/EEC): Issued in 1964, this regulation deals with veterinary health inspection, marking and certification. Meat must be chilled to an internal temperature of +7°C for carcasses and cuts and +3°C for offal and it should be kept constantly at that temperature during transportation and storage.

ii) Quick Frozen Food Directive (89/108/EEC): Issued in 1989, the main articles of this directive describe the storage and transportation temperature conditions for frozen product. It has to be mentioned that the temperature of the quick – frozen foodstuffs must be maintained at all points within the product at –18°C or lower. However tolerances in the product temperature, in accordance with good storage and distribution practice shall be permitted to rise up to –15°C during local distribution and up to –12°C in retail display cabinets.

iii) The Poultry Meat Directive (71/118/EEC) has existed since 1971 and covers rules governing the health and storage requirements. Fresh poultry should be cooled and transported at +4°C.

C.2 Transport Temperature Set-Points

The following transport temperatures are presented: i) vegetables in Table C.1, ii) miscellaneous commodities in Table C.2, iii) chilled products in Table C.3, and iv) frozen products in Table C.4.

Table C.1. Recommended Transport Temperatures for Fruits and Vegetables [ASHRAE Handbook (1994); ASHRAE Handbook (1998); ATP Geneva (1970); Hardenburg *et al.*, (1986); Frith (1991)].

Commodity	Transport Temp [°C]	Lowest Safe Temp [°C]	Character of injury
Apples	0	-1	Freezing injury
Banana	+13 to +14	+12	Chilling injury
Lemon	+10	+10	Pitting and chilling injury
Orange, Mandarins	+4.5	+3	Cold storage injury – no flavor
Beans, Carrots	+7 to +10	+7	Decay, pitting, discoloration
Sweet Potato	+13 to +15	+13	Decay, pitting, discoloration
Tomatoes	+7 to +10	+7	Water soaking and softening, decay

Table C.2. Transport Temperatures for Miscellaneous Commodities [ASHRAE Handbook, (1994); ASHRAE Handbook (1998); ATP Geneva (1970); Hardenburg *et al.*, (1986); Frith (1991)].

Commodities	Transport Temperatures (°C)
Daffodil, Gladiolus, Bulbs	+21
Tulip Bulbs	+10
Chocolate	+7

Table C.3. Transport Temperatures for Chilled Meat Diary Produce and Fish [ASHRAE Handbook (1994); ASHRAE Handbook (1998); ATP Geneva (1970); Hardenburg et al., (1986); Frith (1991)].

Commodities	Transport Temperatures (°C)
Bacon	-1
Butter	0
Cheese	+2
Lamb and Pork	-1.5
Meat Products, Fish	-0.5

Table C.4. Recommended transport temperatures for Frozen Food [ASHRAE Handbook (1994); ASHRAE Handbook (1998); ATP Geneva (1970); Hardenburg et al., (1986); Frith (1991)].

Commodities	Transport temperatures (°C)
Ice – Cream	-20
Meat	-18
Frozen Fish	-20 or lower 4
Juice Concentrates	-20

C.3 Location of temperature measurements

Both return air and top air delivery from the TTC unit have to be measured in specific locations that conform to the standard requirements [ATP (1970); Standard 1110 (2001)]. The air temperature is measured and recorded not only during road transportation but also during capacity tests as presented in Chapter 3, section 3.2.1.

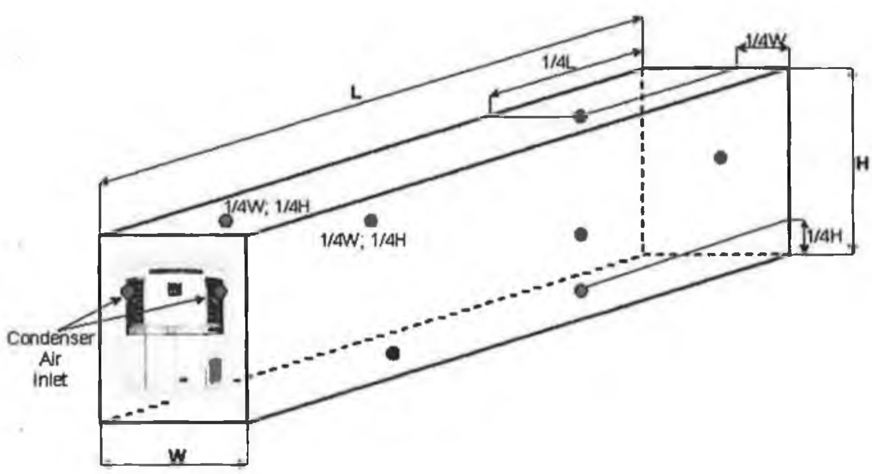


Figure C.1 Location of the main air temperature measurements in the food compartment and both calibrated and isothermal box.

The standard locations for the temperature measurements in the food compartment and during tests in a calibrated or an isothermal box are presented in Figure C.1. The following recommendations were established for the instrumentation, accuracy and calibration: i) the accuracy of the temperature measuring instruments should be within ±0.5%, ii) the accuracy of the pressure measuring instruments shall be within ±2 $^{1/2}$ percent of the absolute pressure, except not to exceed 2-psi, and iii) the accuracy of electrical measuring instruments should be within ±1% of the quantity measured.

APPENDIX D: HEATING VERSUS COOLING DURATIONS

Based on field data, Figure 2.10 shows that heating is required for eight months of the year while Figure D.1 confirms this observation. A particular example when a TTC unit is required to operate in heating during the whole year is presented in Figure D.2.

Field data is presented for another two different TS 500 truck units: i) record RHF 174 – Serial Number 1099S68308 and ii) record RYG 239 – Serial Number 0809S63215. The recorded set–point temperature and return air temperature to the TTC unit from the compartment are both presented, along with the 2001–2002 and 1990–2000 average external ambient air temperature, which was obtained from the Swedish Institute of Meteorology and Hydrology.

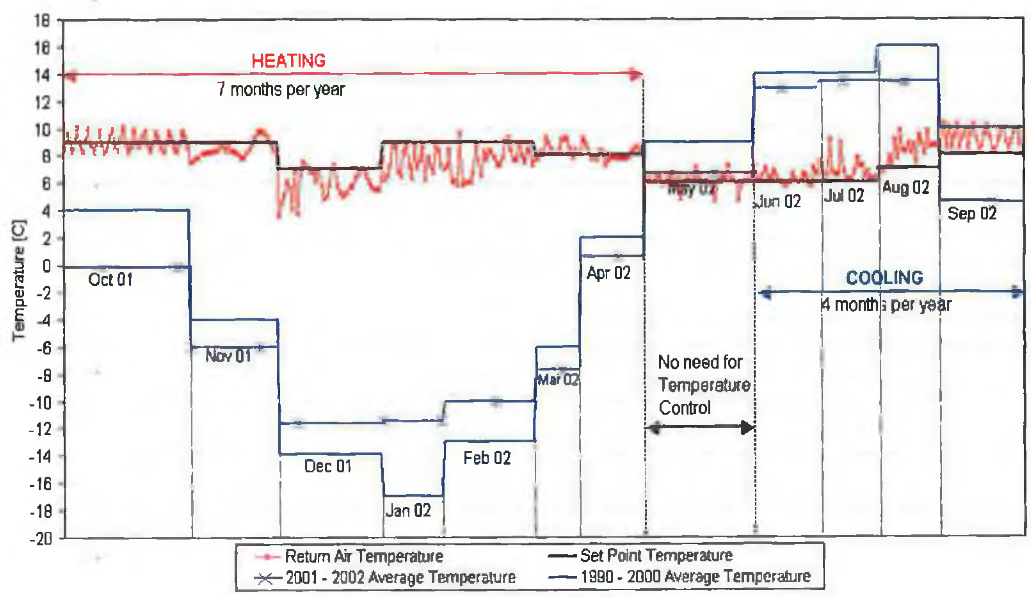


Figure D.1 Heating versus Cooling Time for TS 500 truck unit in Sweden during 2001/2002 [Nilsson (2002)].

Based on the data presented in Figure D.1 it can be concluded that the TTC truck unit is running in heat mode for 7 months per year when the ambient temperature is lower than the box set point and 4 months in cool mode for ambient temperature higher than the required transportation set point.

In Figure D.2 the same temperatures are presented for another TS 500 unit, record RYG 239 – Serial Number 0809S63215. During October, November 2001 this unit was used for the transportation of citrus fruits, from December until May for vegetables until May and for flowers during summer months. Based on the data presented in Figure D.2 the TTC unit was running in heat mode for the whole year. During summer months the transportation temperature set-point was +21°C for flowers, which is higher than the monthly average temperatures.

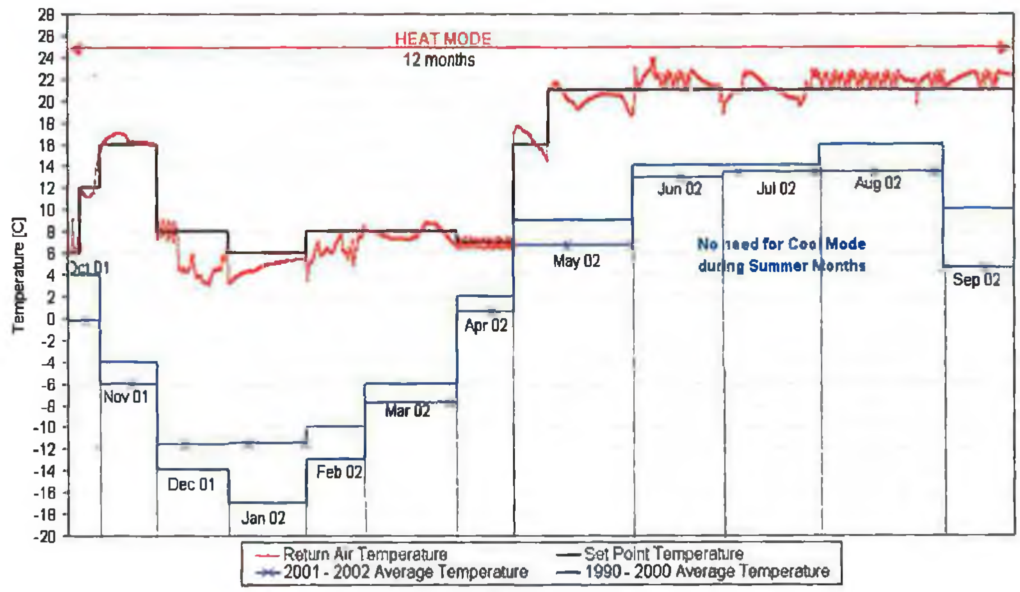


Figure D.2 Heating versus Cooling Time for TS 500 truck unit in Sweden during 2001/2002 [Nilsson (2002)].

APPENDIX E: HEAT LOSSES FOR CALIBRATED AND ISOTHERMAL BOX METHODS

Appendix E explains important concepts used for the calibration of the boxes before capacity tests performed with the test methods overviewed in Chapter 3, section 3.2.

E.1 Calibrated Box Method – Heat Losses

The following procedure should be used to calibrate the calibrated box prior to conducting a test: i) measure the temperatures within the calibrated box, ii) establish and maintain the average ambient temperature at a value not less than 23.9°C below the average temperature in the calibrated box, while operating the electric heaters until steady state conditions are reached, iii) after a minimum of five hours heating, readings of the temperature at 15 minutes intervals should be taken until nine consecutive sets (two – hour period) indicate the steady state conditions have been achieved within tolerances of ±1.1°C in the ambient and ±0.56°C in the box and iv) electrical heat input to the heaters, fans, motors and other electrical devices within the box have to be measured to establish the heat losses through the walls.

E.2 Isothermal Box – Heat Losses

The calibration of the isothermal – box consists of obtaining the heat transfer for the common wall between the isothermal and the condenser air enclosure (Figure 3.2). The heat losses are calculated as the heat transfer from the box through the front wall in the ambient room, while maintaining higher temperature in the isothermal box. The same minimum 23.9°C average temperature difference should be maintained between the enclosures. The surround room should have the same temperature like the air in the isothermal box. Electrical input to the heaters, fans, motors, lights should be measured and all additional data recorded as necessary to establish the leakage through the common wall between the ambient room and the isothermal box.

APPENDIX F: TECHNICAL DETAILS OF THE EXISTING CALIBRATED BOX COOLING TEST FACILITY

In Chapter 3, sections 3.2.1 and 3.3.1 a description of the existing standard test methods for measuring the TTC unit cooling capacity together with the existing mechanical equipments from the temperature controlled chamber – ambient room (Figure 3.3) were presented. The detailed technical specifications of the existent equipment of the calibrated box test facility, which is used as a basis for the implementation of the new heating test facility are overviewed in this Appendix.

F.1 Technical Details of the existing equipment

As presented in Chapter 3, the new heating test facility implemented in Thermo King, Galway is using a calibrated box within an existing Ambient Room. The description of the existing test facility capable of providing the –30°C to +50°C temperature range in the Ambient Room is provided in this section as extra information added at the technical specifications for the existing equipments presented in Table 3.1. Based on the schematic presented in Figure 3.3 the technical specifications for the main existing equipments are overviewed.

F.1.1 Existing Main Cooling System

Figure F.1 shows the ambient room main compressors (Item 4 in Figure 3.3).

Figure F.1 Ambient Room main compressors (Item 4, Figure 3.3).

The Ambient Room Cooling System is a Copeland type hermetic refrigeration unit that consists of 4 compressors model D8DT450X with a total refrigeration capacity of 106kW at –35°C evaporating temperature and +38°C condensing temperature and

84.57 kW total power consumption. The constructive specifications of the main cooling system are as follows: i) weight 3 tonnes, ii) Length 3.8 m, Height 2.2 m.

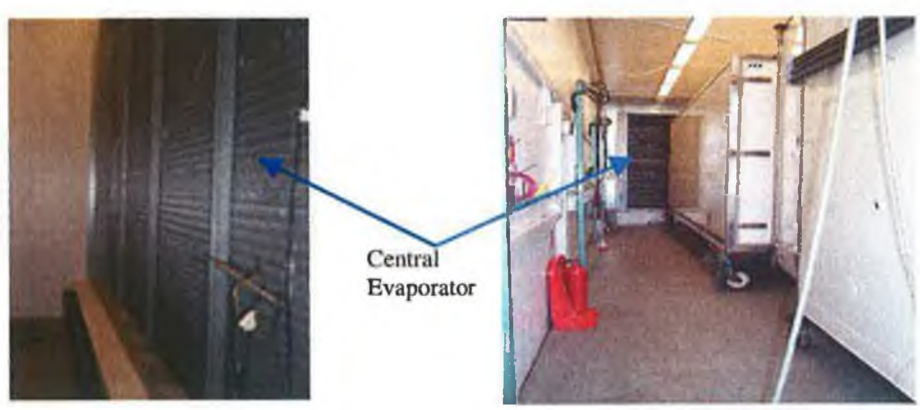
F.1.2 Air Cooled Condenser Pack

The condenser pack (Item 5 in Figure 3.3) is a Searle type model MDG 205-12D (Figure F.2), with the following technical specifications: i) total heat of rejection 207 kW at ambient temperature +28°C and condensing temperature +37.8°C, ii) Heat Exchange Surface Area 643.8 m^2 equipped with 8 fans with a total air volume of 25.6 m3/s. The dimensional characteristics are: I) overall length 5.108m, ii) overall width 2.301 m, and overall height 1.225 m.

Figure F.2 Air Cooled Condenser Pack.

F.1.3 Ambient Room Evaporator pack

The main evaporator situated in the Ambient Room (Item 6 in Figure 3.3) is a Searle model 5HF 5808 M12–500 and it has 4 separate circuits (Figure F.3).



a) Evaporator - Back View b) Evaporator – Front View

Figure F.3 Ambient Room Cooling Evaporator (#6 in Figure 3.3).

The total cooling capacity is 100 kW at –35°C evaporating temperature and +38°C condensing temperature with an optimum 6°C superheat. The air in the Ambient Room is diverted via a false ceiling and returned to the evaporator face using 5 axial fans with a total airflow of 153 000 m^3/hr.

F.1.4 Dehumidifier

To maintain dry air in the Ambient Room especially during tests performed at low temperatures, a Munters MTX 2100 type Dehumidifier is used with 2100 m^3/h airflow at 15.3 kW reactivation heaters (Figure F.4).

Figure F.4 Dehumidifier used to maintain dry air in the ambient room.

F.1.5 Control System

The automatic control of the compressor pack is obtained through a Danfoss Adap–Kool Control System that is maintaining the compressor suction pressure and compressor On/Off sequence against room temperature. The control of the heaters in the calibrated box is obtained through a Eurotherm TE 200 type thyristor, with a three-wire configuration from the analog out put module FP–AO–200 to achieve stable box temperature during the test.

APPENDIX G: TECHNICAL SPECIFICATIONS OF THE NEW HEATING CAPACITY TEST FACILITY

While the main mechanical equipments implemented in the first heating test facility were overviewed in Chapter 3, section 3.4, Appendix G presents supplementary technical details and design characteristics.

G.1 Heating Capacity Test Facility

Figure G.1 shows supplementary technical details regarding the design of the heating test facility described schematically in Figure 3.3. The interconnecting pipe work dimensions and its various fittings in a three-dimension arrangement are presented.

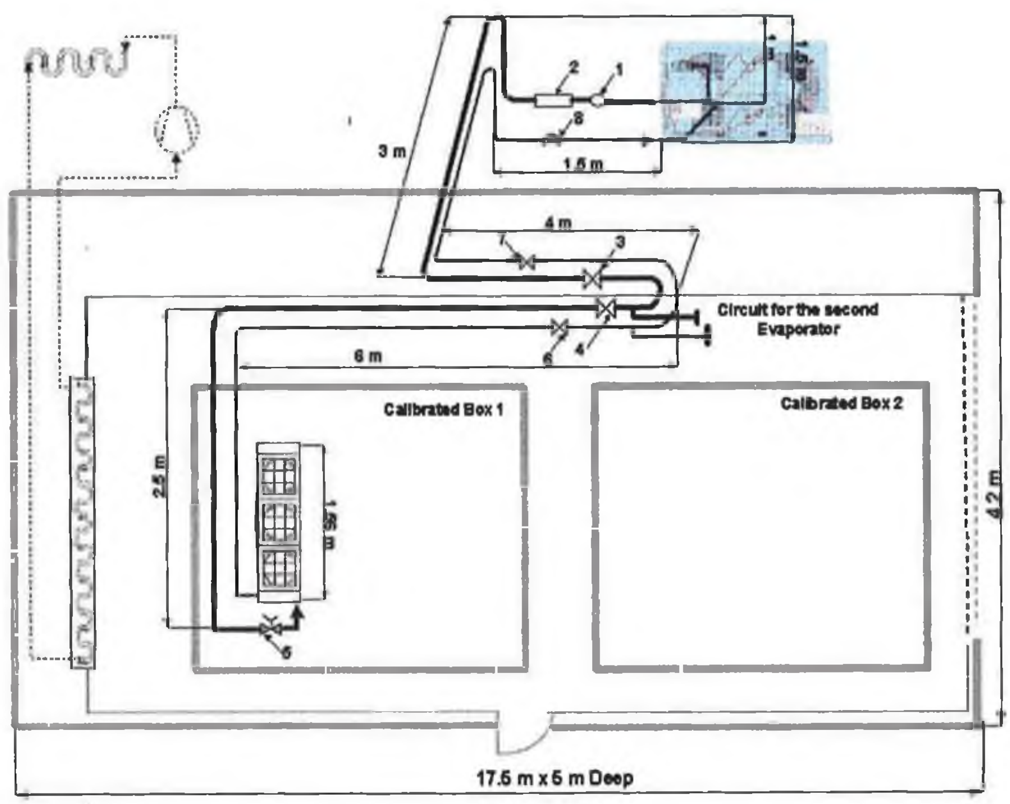


Figure G.1 Three-dimensional representation of the interconnecting pipe circuits and fittings of the new heating capacity test facility.

The valves and fittings have the following purpose and characteristics:

- Sight Glass: used to indicate the condition of the refrigerant in the plant liquid line and the moisture content in the refrigerant (type SGN, Danfoss).

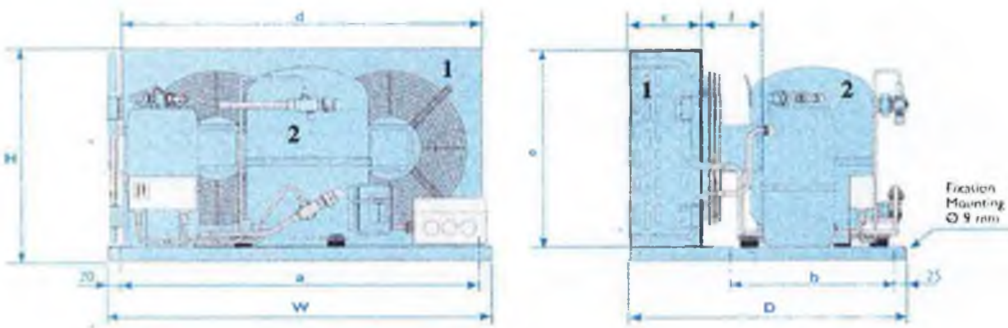
- Liquid Line filter drier: used to protect the refrigeration system from moisture, acids and solid particles. The Danfoss drier DML type was selected, with a solid core of 100% molecular sieves, optimised for the HFC refrigerants and with high drying capacity minimizing the risk of acid formation.
- Manual Valves: Manual ball valves (3, 4, 6, 7) used for isolating the liquid and suction lines. The Danfoss type GBC ball valves selected, are manual ball valves suitable for bi–directional flow. These valves offer maximum tightness both externally as well as internally across the seat. The technical characteristics are: for the valves 3 and 4 situated on the liquid line, a 22mm solder ODF connection was used with 27.5 m^3/h K_v value; for the valves 6, 7 installed on the discharge line a 35 mm solder ODF connection was used with 73.3 m^3/h K_v value (K_v = Valve Characteristic).
- Electronic Expansion Valve (EEV); a Sporlan type SEI–3.5–8 valve was used. The SEI is an electronically operated step motor flow control valve, used for the precise control of the liquid refrigerant flow. This valve was selected in order to maintain the superheat on the evaporator as explained ion Chapter 3.
- Electronic Throttling Valve (ETV)–Thermo King stepper motor valve for pressure and refrigerant mass flow control.

For the optimum placement of the Copeland condensing unit, remote evaporator and pipe connections the following design consideration were considered:

- The condensing unit was located in a well-ventilated area, with no restrictions of the airflow.
- On the horizontal sections of the suction line the circuits were slopped downwards towards the compressor, due to the fact that the evaporator was installed above the compressor.
- The suction line was insulated to avoid condensation and abnormal superheat.
- The suction risers were fitted with a U-trap at the bottom and a P trap at the top and they were never higher than 4 m.

G.2 Copeland Condensing Unit

The cooling capacity and power consumption in relation with the evaporating temperature for the hermetic Copeland condensing unit model ZF–U2–33–TWD were presented in Chapter 3 Table 3.3 and 3.4 (Item 7 in Figure 3.3). This unit is appropriate for medium and low temperature applications from –10°C down to –40°C, and features good pull down capabilities starting at 0°C. The unit is equipped with two fans capable of providing 1.79m^3/s airflow each. The condenser has a maximum 20 kW cooling capacity. Information regarding the constructive dimensions is presented based on Figure G.2.



D = 820 mm
H = 633 mm
W = 1130
1 = Condenser

C = 25 mm F = 20 mm
Base Mounting =1095x475mm
2 = scroll compressor

Figure G.2 Copeland Condensing Unit – Compressor, condenser and receiver tank.

G.3 Remote Evaporator

The evaporator (Item 9 in Figure 3.3) was designed especially for the condensing unit (Item 7, 8) as a compact finned heat exchanger with refrigerant as a primary fluid and air as a secondary heat exchange fluid. Convection heat transfer coefficients were calculated using the following criteria: Nusselt, Prandtl, Reynolds, Dittus Boetter and Miheev [Carabogdan *et al.*, (1983)]. The following design criteria were applied for the design calculations:

- The total heat transfer between the cold and hot fluids [Incropera, 1999]:

$$Q = m \times c\,p \times \Delta T \ [W] \qquad (G.1)$$

- Mean logarithmic temperature difference [Incropera, 1999]:

$$\Delta T_{lm} = (\Delta T_M - \Delta T_m) / \ln(\Delta T_M / \Delta T_m) \ [°C] \qquad (G.2)$$

Where: ΔT_M = Th_I–Tc_0; T_{hI} = air inlet temperature; T_{c0} = refrigerant outlet temperature; ΔT_m = T_{ho}–Tc_I; Tc_I = refrigerant inlet temperature; T_{ho}= air outlet temperature.

- Overall heat transfer coefficient:

$$1/U_h = \frac{1}{h\,c\,\frac{Ac}{Ah}} + \frac{R\,f\,c''}{\frac{Ac}{Ah}} + Rw\,x\,Ah + \frac{R\,f\,h''}{\eta\,o\,h} + 1/(\eta_{oh}\;x\;h_h) \qquad (G.3)$$

Where: U_h = overall heat transfer coefficient for the hot fluid; C and h refer to the cold and hot fluid; η = overall efficiency or temperature effectiveness; Ac = the total area for the cold fluid; Ah = the total area for the hot refrigerant; Rfc" = 0.0002 m^2 K/W= fouling factors = additional thermal resistance which shows the effect of impurities on the heat exchange surfaces; Rfh" = 0.001 m^2 K/W.

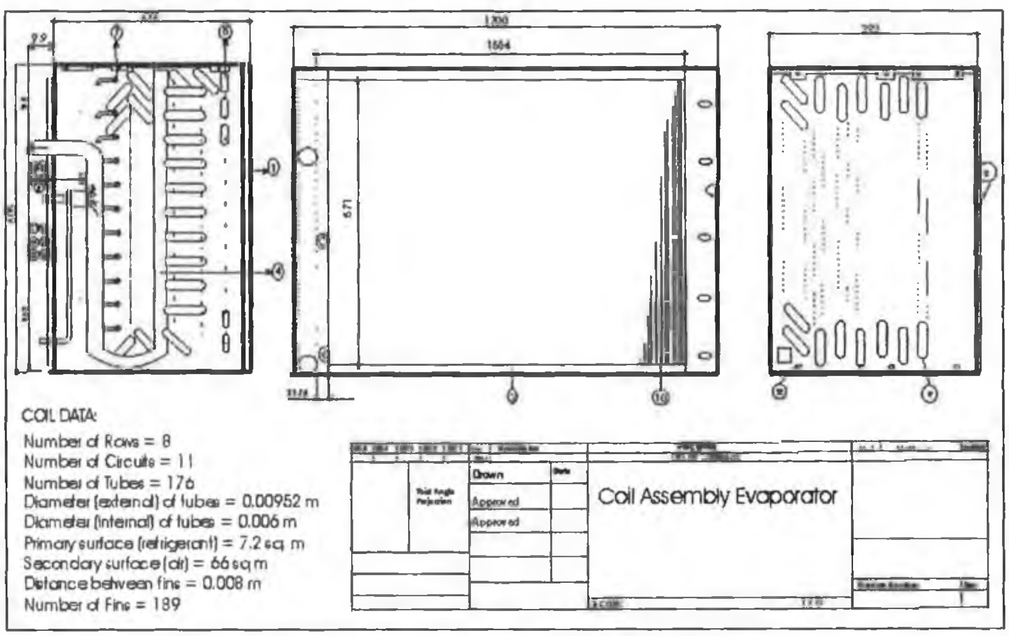


Figure G.3 General Assembly of the Evaporator.

G.4 Axial Fans

A total 6000-m^3/h airflow for one evaporator is necessary to obtain the desired capacity, based on the calculations presented in Section G.2. Three EBM axial fans type 4E. 420 were selected, capable of delivering 2000m^3/h at 120Pa pressure drop.

APPENDIX H: DATA ACQUISITION AND CONTROL PROGRAM FOR THE HEATING CAPACITY TEST FACILITY

In Chapter 3, sections 3.5.2 and 3.5.3 an overview of the data acquisition system and Lab View control program for the new heating capacity test facility was undertaken. Further details are presented in Appendix H.

H.1 General Overview of the data acquisition system

In recent years personal computers have evolved into powerful and cost effective computing platforms and are being increasingly used in research laboratories and industry for data acquisition applications. A general overview of a standard data acquisition system is presented in this section together with a description of the basic concepts [Figure 3.8].

A data acquisition and control system typically consists of the following:

- Sensors, which measure physical variables such as temperature, pressure and flow. In order to sense and measure the physical variables mentioned, different types of transducers (sensors) are used to convert physical variables into electrical signals and to transmit these signals either to a signal-conditioning device or directly to the data acquisition board.
- Signal conditioning device, which converts the sensor outputs into signals readable by the A/D board in the PC. This device amplifies and filters the sensor signal, then outputs a voltage, which is easy to capture with an analog input board.
- Data acquisition and analysis hardware, which helps measure information presented by both digital and analog signals. Digital signals can come from a variety of sources such as switch closures, relay contacts or TTL compatible interfaces. Analog signals come from instruments, sensors, or transducers that convert phenomena like pressure, position or temperature into voltage or current. The most common boards are: analog input, analog output and digital input–output.
- A computer system with the appropriate application software to acquire and log your data to disk, to process, analyse, and/or provide a graphical display of the data.

Some important theoretical aspects of the analog input, analog output, timing I/O signal modules are explained below:

- Analog Inputs: Common data acquisition board specifications are the number of channels, sampling rate, resolution, range, accuracy, acceptable noise levels and non-linearity, all of which affect the quality of the digitised signal.
- Sampling rate: Determines how often conversions can take place. To properly digitise a signal, the Nyquist Sampling Theorem requires that the sampling rate be more than twice than the rate of the maximum frequency component. However, signal produced by temperature transducers do not require high sampling rates because temperature does not change quickly.
- Counter/Timers are useful for many applications, including counting the occurrences of events, digital pulse timing, and generating square waves and pulses. These applications can be implemented using three counter/timer signals- gate, source and output. The gate is a digital input that enables or disables the function of the counter. The source is a digital input that causes the counter to increment each time it toggles, and therefore provides the time base for the operation of the counter. Finally, the output generates digital square waves and pulses at the output line. The most significant specifications for operation of a counter/timer are the counter size and clock frequency. The counter size is the number of bits that counter uses. A larger size means that the counter can count higher and longer. The clock frequency determines how fast you can toggle the digital source input. With higher frequency the counter can measure and generate higher frequency signals and provide higher resolution for measuring time.

As presented in section 3.5.2, a dedicated data acquisition system based on the National Instruments Ethernet Field Point Modules and Lab View software were used for the heating capacity test facility.

H.2 Transducers and DAQ – Heating Test Facility

As shown Figure 3.10 and Table 3.7 four types of measurements were performed: i) analog input type for temperatures and pressures, ii) analog output type for the control of electric heat input and the electronic expansion valves and iii) digital input

–output type for switching the valves installed on the unit and different mechanical equipments and iv) counter for the power input calculation.

H.2.1 Transducers

A) Temperature Measurements

In this application 32 temperatures are measured using thermocouples to convert temperature into an analog voltage signal. The thermocouple is one of the simplest of all sensors. It consists of two wires of dissimilar metals joined near the measurement point. The output is a small voltage measured between the two wires.

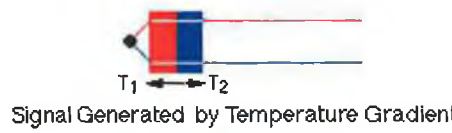
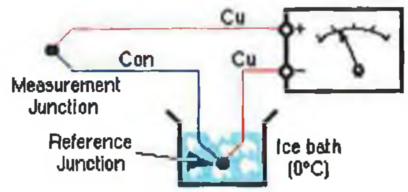


Figure H.2 The thermocouple theory.

A thermocouple circuit has at least two junctions: the measurement junction and a reference junction. Typically, the reference junction is created where the two wires connect to the measuring device. The latest one consists in a junction for each of two wires, but because they are assumed to be at the same temperature (isothermal) they are considered as one thermal junction. It is the point where the metals change - from the thermocouple metals to whatever metals are used in the measuring device - typically copper. The output voltage is related to the temperature difference between the measurement and the reference junctions. This phenomenon is known as the Seebeck effect. In this application type T thermocouple was used, with Cu as positive material and Cu, 45%Ni as negative material. This type of thermocouple has class two-type accuracy of 0.75% or 1°C and –60°C to +100°C the temperature range. All thermocouples were calibrated with an accuracy of ±0.5°C using D55SE temperature calibrator (Jofra, Denmark) at 0°C and +10°C. The D55SE calibrator has the following characteristics: accuracy ±0.5°C, temperature range within –60°C to 253°C, 24 VDC power output [Figure H.3(a)]. Based on the fact that the voltage generated by a thermocouple is a function of the temperature difference between the measurement and reference junctions, a traditional thermocouple measurement using the ice bath was also performed [Figure H.3(b)].

a) D55SE calibrator.



b) Traditional ice bath measurement.

Figure H.3 Thermocouple measurement calibration.

The measurement correction was done mathematically in the software, based on the fact that the type T thermocouple can be linearized using a first order polynomial expression ($y = a + bx$).

B) Pressure Measurements

Eight pressure measurements were performed using Thermo King pressure transducers (Figure H.4).

Figure H.4 Voltage output pressure transducers.

This sensor is a transducer that converts pressure into an analog electrical signal. Although there are various types of pressure transducers, one of the most common is the strain-gage base transducer. The conversion of pressure into an electrical signal is achieved by the physical deformation of strain gages, which are bonded into the diaphragm of the pressure transducer and wired into a Wheatstone bridge configuration. Pressure applied to the pressure transducer produces a deflection of the diaphragm that introduces strain to the gages. The strain will produce an electrical resistance change proportional to the pressure. The pressure transducers used are a voltage output based type; they include integral signal conditioning which provide a much higher output than a millivolt transducer. The output is normally 0-

5Vdc or 0-10Vdc. Because they have a higher level output these transducers are not as susceptible to electrical noise as millivolt transducers and can therefore be used in much more industrial environments. Two types of pressure transducers were used: G07 type with the range of 0 – 500 psig and G09 type with the range 0 – 200 psig, both with an accuracy of ±0.25%. The calibration of the pressure transducers was performed using a pressure calibrator, type PCC3-H-200-2.

C) Wattmeter

The Watt–hour transducer was used to measure the total power input to heaters and fans in the calibrated box via analogue output of instantaneous power through the panel and digital pulse output of watt – hour energy as consumed through the panel. A Scientific Columbus wattmeter type was used, model number DL 31K5A2–6270–2–8–12, serial number 37069. It has the following specifications: 2 elements with 3 phase, 1 mA dc, one count per Wh, 500 Watts per element, 120 Volts, 5 Amperes at 60Hz, frequency range 58 to 62 Hz. The accuracy of the wattmeter is ±0.5%. The digital output was set for maximum 20 000 counts per hour or each count representing one watt–hour.

D) Mass Flow Meter

The fuel consumption was measured using a portable Danfoss fuel mass flow meter, type Mass 3000, with 0.15% accuracy over a wide measuring range.

Table H.1. Technical Specifications – Fuel Mass Flow meter.

Type	Outputs	Measurements
Danfoss	2 analogue outputs	Mass flow rate
Mass 3000	1 frequency – pulse output	Total Mass
	2 relay outputs	Density

H.2.2 DAQ - Field Point Modules

The data acquisition system used in this application is the National Instruments product: Field Point Modular Distributed I/O [Figure H.5]. These types of solutions are used when the measurement and control functionality have to be distributed out into the field. Therefore the network cable replaces the signal wires. Distributed I/O and data acquisition systems are working with the following networks: Ethernet TCP/IP, RS 232 and RS 485, and Wireless. In the application the Ethernet TCP/IP network connection was selected. The main advantage of this platform is that the Ethernet to network I/O devices can provide a high speed between 10 to 100 Mb/s and great flexibility.

Figure H.5 Filed- Point Modules, Ethernet TCP/IP network connection.

Some of the most important advantages and characteristics of this type of data acquisition modules are presented:

- The modular architecture of Field Point modules (Figure H.5) allows matching the best combination of I/O modules, terminal base style, and network interface. A wide temperature range (from +40°C to 70°C), isolation, watchdog timers, programmable power-up states, embedded diagnostics are the mechanical features for Field Point modules.
- Terminal Bases – The I/O modules are installed on terminal bases that provide terminals for field wiring connections, as well as module power and communications. Therefore the I/O modules can be plugged or unplugged without disconnecting the field wiring.
- Network Modules provide connectivity to open industrial networks. The network modules communicate with the local I/O modules via the high-speed local bus formed by linked terminal bases.
- The Field Point system is designed to operate in the harsh environments for industrial applications. Field Point components can operate over a wide temperature range of –40°C to +70°C and are specified for a high degree of vibration and shock.

In Table 3.7 the Field Point modules used were presented. For a complete description supplementary details are provided as follows:

- *FP–TC–120.* These devices are temperature input modules with 8 channels that are used to measure thermocouples. The FP–TC–120 modules include eight differential inputs for thermocouples. It also provides cold junction compensation using a thermistor embedded in the terminal base or connector block.
- *FP–AI–100.* These devices are analogue input modules for Field Point that can be used to measure voltages ranging from the milivolt level to the 120 V high voltage level in applications such as measurement from the pressure transducers. These modules can also measure 0 to 20 or 4 to 20 mA current loops.
- *FP-A –200.* The National Instruments FP–AO–200 device is a Field Point analogue output module with eight 0–20/4–20 mA current loop outputs.
- *FP–RLY–420.* The National Instruments FP–RLY–420 devices are versatile relay modules that are used to control digital signals ranging from low voltages to 125 VDC and to 250 VAC. These modules are commonly used to control indicator lights, motors and power circuits.
- *FP–CTR–500.* The National Instruments FP–CTR–500 devices are high – speed digital counter input modules for Field Point that can be used to count digital signals ranging from 5 to 30 VDC and to measure frequency. The counter module features low pass filters to eliminate high – frequency noise.

H.3 Software Capabilities

Twp types of software were used: Field Point Explorer that contains software tools for the Field Point modules configuration and Lab View 6.1, a graphical programming type software, used as a tool for the control program.

A) Field Point Explorer

The Field Point system includes a full suite of software tools that simplifies integration of Field Point into a variety of software environments. Field Point Explorer is a Windows based program that can be used to configure the Ethernet Field Point systems and to verify the operation. Based on a graphical windows, the user can interactively set every configurable parameter of Field Point modules, including ranges and power-up states, as well as network parameters (Figure H.6).

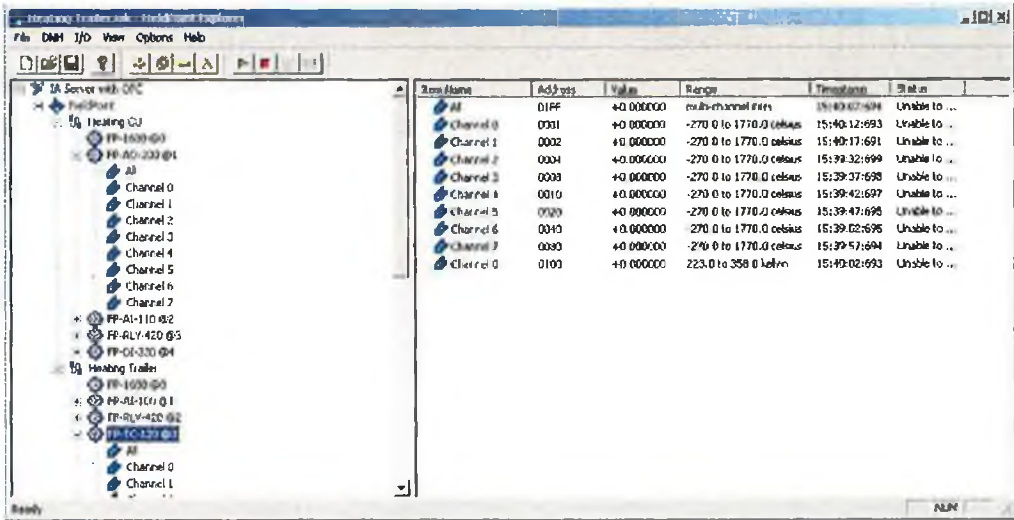


Figure H.6 Field Point Explorer – Modules Configuration.

B) Lab View Software (Version 6.1)

The general purpose of the software is to transform the PC and data acquisition hardware onto a complete data acquisition, analysis and display system. Using Lab View 6.1 it is possible to create application programs called Virtual Instruments (vi). Each VI has three main parts: the front panel, the block diagram and the icon/ connector. The block diagram is the VI's source code, which is created using a graphical programming language called G, and is presented in the form of a series of interconnected icons. Each icon is a representation of a VI. The front panel provides an interface with the VI and allows one to interact with it. The connector acts as a port through which data passes from diagram to VI. The icon/ connector together represent the VI in a manner analogous to a subroutine call statement when the VI is used as a sub VI in another VI's block diagram. The front panels of the program were presented in Chapter 3, Figures 3.11 to 3.13. For a more detailed description the main diagrams of the program are presented in this section. Figure H.7 shows the sub VI entitled "Inputs". In this sub module (sub VI) readings from the analog input modules are collected in an array using specific functions that are programmed to read from the desired modules.

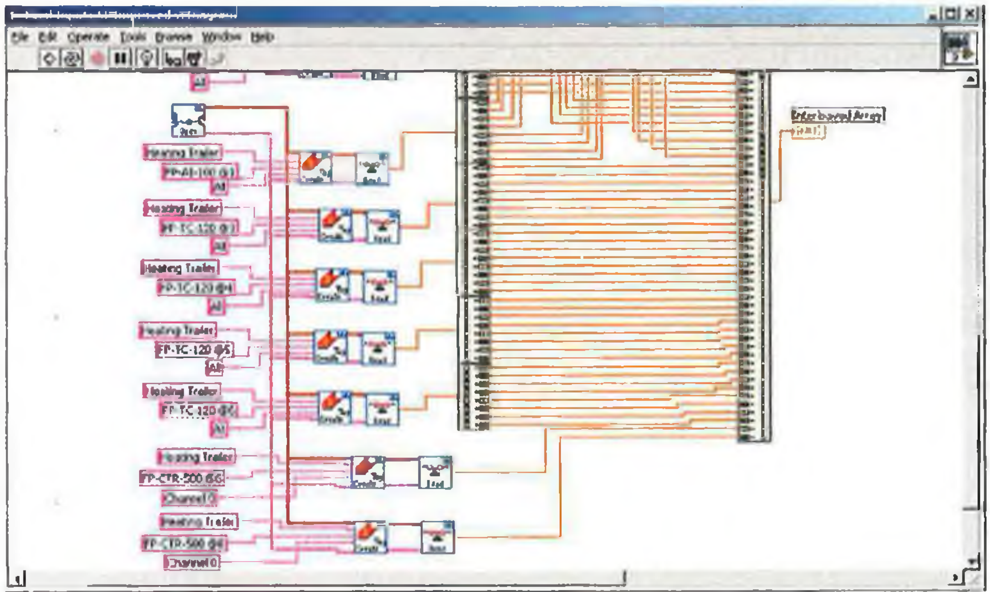


Figure H.7 Diagram of the DAQ connections.

Figure H.8 shows the program module used for the control of the calibrated box heaters. The algorithm described in Figure 3.15 is created in Lab View based on existing PID functions. The signal is sent to the analogue output module that controls the heat input in the calibrated box through a dedicated function named "Write".

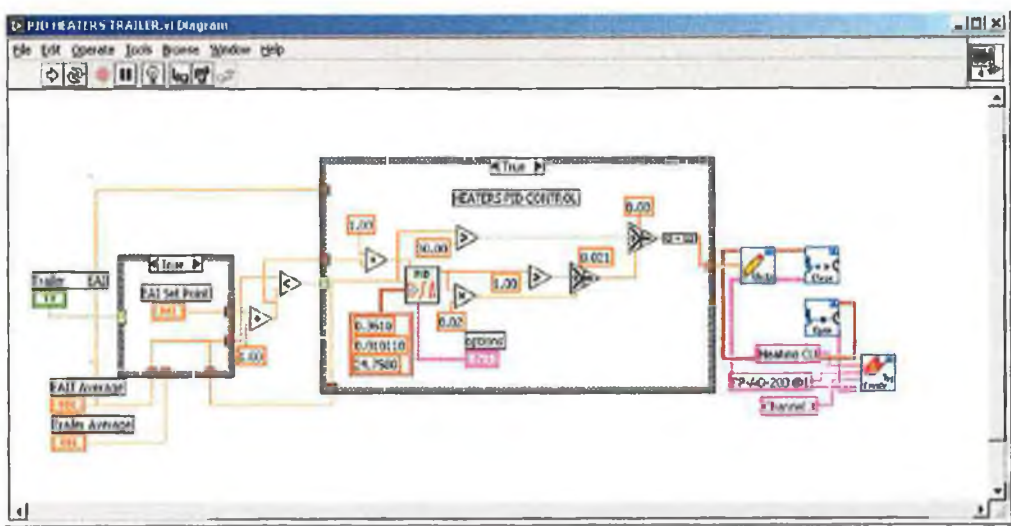


Figure H.8. Calibrated Box heaters sub VI.

In Figure H.9 the sub VI that is reading the counts of the counter module and is calculating the watt-hour consumption and automatically the capacity of the unit is presented. The watt-hour result is obtained dividing the difference between two counts at the precise time measured between them.

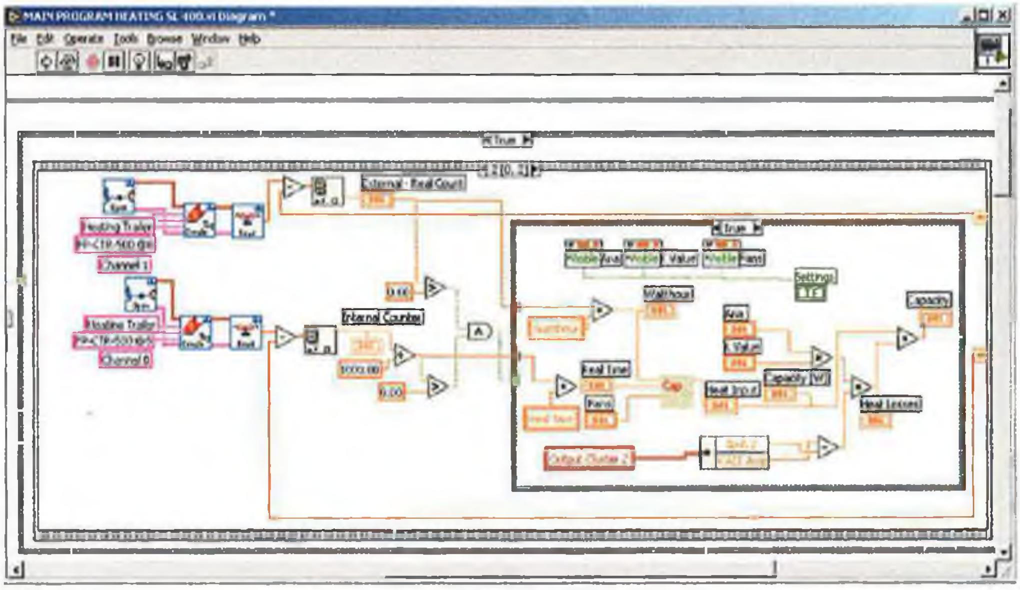


Figure H.9 Counter and Capacity calculation sub VI.

Figure H.10 is presenting the sub VI used for saving the data from the program. This section converts the array data collected from the Lab View program into a text string and writes it in an Excel file.

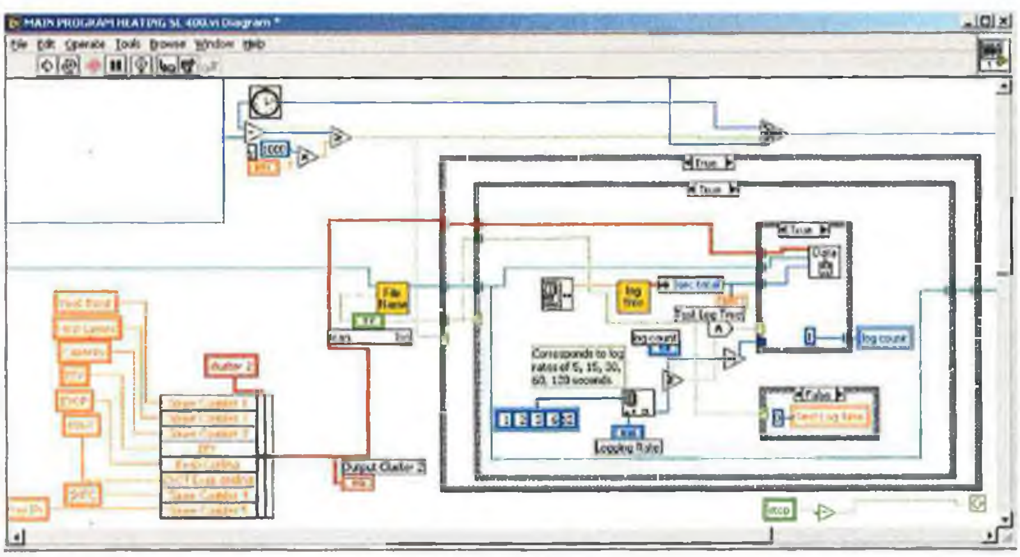


Figure H.10. Save data sub VI.

APPENDIX I: HOT-GAS LINE SIZING AND THREE-WAY VALVE CHARACTERISTICS

I.1 Three-Way Valve Connection Size

Chapter 5, section 5.2 presents modification I that consist in a new hot-gas line with increased diameter implemented on the modified unit. The 22 mm size was obtained after calculations overviewed in Appendix I. The design characteristics of the new three-way valve that has to be installed on the unit are also presented.

I.1.1 Three-way valve outlet connection on the hot-gas line

A new connection size of 22.2mm (7/8in) outside diameter and 1.143mm(0.045in) wall thickness should be used for the increased hot gas line outlet. Based on the sizing calculations presented below a range within 18mm to 23mm was obtained for the hot gas line and the new 3-way valve outlet. The standard dimensions for Copper tubes within this range are: 22.2mm (7/8in) and 19mm (3/4in). The 22.2mm (7/8in) copper tube was selected for the new line due to increased gas mass flow rate while it maintains the velocity in the required limits at a 6.2m/s value.

I.1.2 Sizing criteria for the valve outlet and the new hot gas line.

Known technical characteristics of the system together with theoretical assumptions regarding the velocity of the refrigerant that were used in the calculations are presented below:

- Capacity of the evaporator during full load operation.
- The change in enthalpy that the refrigerant experiences through the coil: 160KJ/kgK.
- The velocity range of the hot gas flowing through the defrost pipes was considered to be between 305m/min and 610m/min.
- The following equation was used:

$$Velocity = v \times M_f \times Area$$

Where: v = specific volume at the saturation temperature of the discharge gas,

M_f = mass flow rate,

Area = pipe flow area

I.1.3 Pressure Drop through the three-way valve.

Pressure drop through the valve was obtained between 10psi and 18psi range. It was considered that the valve pressure drop has to be 33% from the total pressure drop between the compressor discharge pressure and evaporator inlet pressure. Several practical test results were performed to determine the optimum compressor discharge and evaporator inlet pressures.

I.1.4 Maximum flow rate.

Maximum flow rate requirement is: 0.1752kg/sec. This flow rate is considered to be the maximum required value for a 1psi pressure drop. The same value was obtained for the refrigerant mass flow rate (the maximum capacity in tonnes) through the new 22.2mm hot gas line. The maximum compressor discharge line capacity in tonnes was obtained as 0.22kg/sec. An increase of almost 50% of the flow through the 3way valve restriction for the heat mode cycle is expected for these new conditions.

I.2 Three-Way Valve New Specifications

The new 3-way-valve design has to meet the specifications presented in table I.1.

Table I.1. New Three-Way Valve technical specifications.

No.	Parameter	New Value	Comparison with the old valve
1	Connection size for the hot gas line outlet	22.2mm (7/8in); wall thickness 1.143mm (0.045in)	58% increased dimension
2	3 way valve pressure drop	Within 10 to 18-psi	Within 35 to 55-psi
3	Maximum mass flow rate at 1-psi pressure drop	0.1752 kg/sec	0.09 kg/sec
4	Mass flow rate through the valve restriction	Depends on the Valve characteristic Cv	A 55% increase is expected

APPENDIX J: EXPERIMENTAL RESULTS OF THE MODIFIED SL 400e TRAILER UNIT

J.1 Modification I

Chapter 5, section 5.2.2 shows the experimental results obtained for modification I – increased diameter of the hot-gas line at ambient temperature of –30°C and box temperature of +12°C. Supplementary data for different test conditions are presented in Figure J.1. A comparison between the standard and modified SL 400e unit is undertaken for –20°C ambient temperature and +12°C box temperature, while the unit operates in high-speed diesel mode.

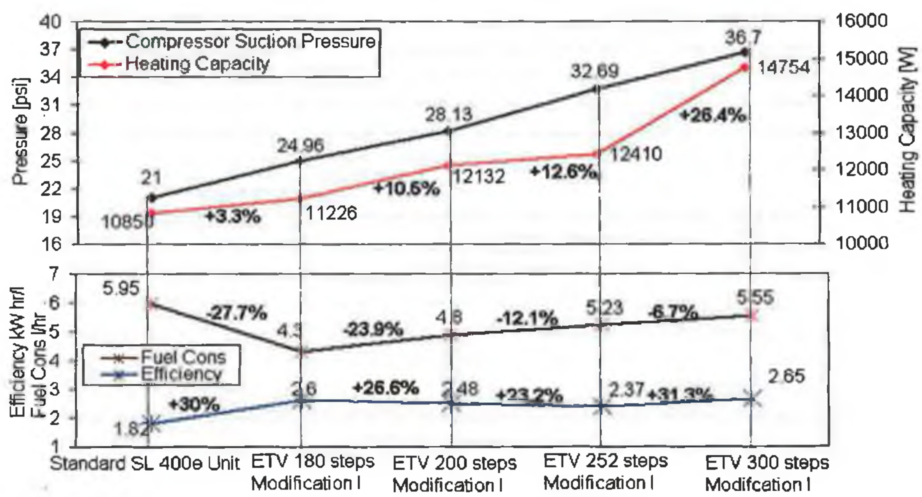


Figure J.1 Heating Capacity, fuel consumption and efficiency at –20°C ambient temperature and +12°C box temperature.

Increased heating capacity up to +26.4% corresponding with supplementary 3904W is obtained for 36.7-psi compressor suction pressure maintaining the ETV at 300 steps. For the same condition the efficiency is up to 31.3% higher, while the unit has 6.7% less fuel consumption than the standard SL 400e unit. However, the minimum fuel consumption by 27.7% lower is obtained controlling the ETV at 180 steps. In these conditions the impact on the heating capacity is only 3.3% higher while the system efficiency also experiences a significant improvement of +30%. The flexibility of the unit operation optimum condition that depends on the desired result: higher capacity or very low fuel consumption highlighted in Chapter 5 is confirmed.

J.2 Modifications II, III and IV

Chapter 5, section 5.3 shows a comparison between the modifications II, III and IV based on hot-gas injections into the low-pressure side of the heating cycle. The same type of results but for different ambient temperature of –20°C and box temperature of +2°C are overviewed in this Appendix.

J.2.1 Heating Capacity

For ambient –20°C and box +2°C test temperatures, modification I has low impact on the heating capacity that is by only 0.9% higher. While –20°C is a low ambient temperature and the need for increased heating capacity was highlighted (Figure 1.11), the beneficial effect of the hot-gas injections is necessary. An average of +5.5% corresponding to 655 W is obtained due to the influence of Modifications II, III and IV. Optimum response of -12.4% less fuel consumption and +17.5% increased system efficiency is also recorded due to the impact of modification III – hot gas injection at the accumulator inlet.

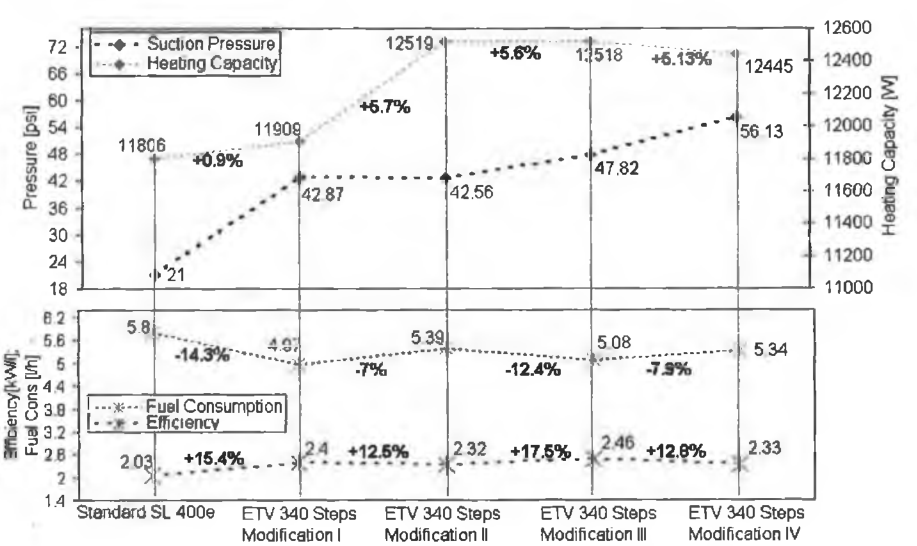
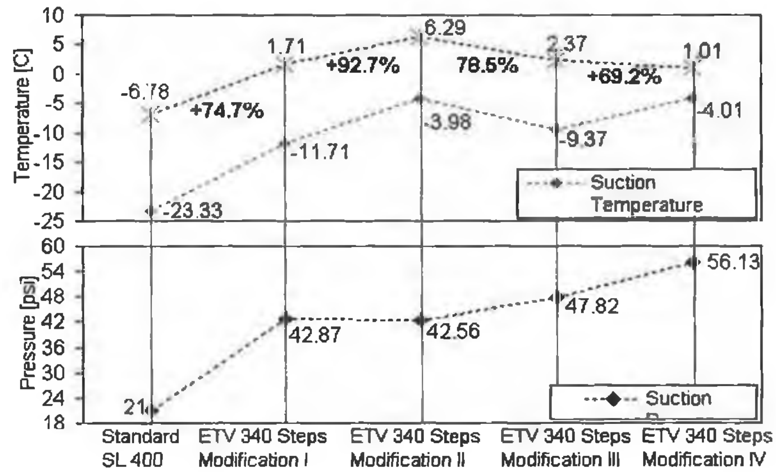


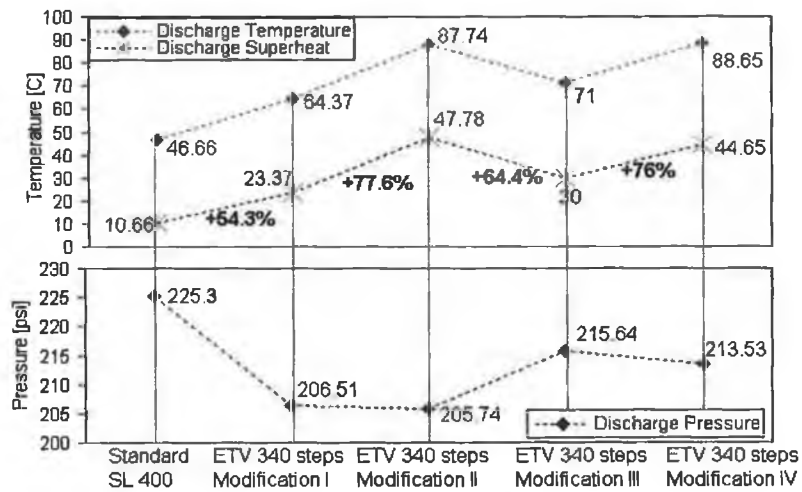
Figure J.2 Heating Capacity, efficiency and fuel consumption at ambient temperature –20°C and box temperature +2°C.

J.2.2 Compressor behaviour

The compressor behaviour at the same temperatures is overviewed and presented in figure J.3(a,b). Optimum compressor behaviour is obtained for improvement II. It has to be noted that while the standard unit operates with a negative suction superheat of –6.8°C, as a result of modification I this value is increased but at a still low +1.71°C temperature. As a result, the hot gas injection effect is absolutely necessary to improve the operational conditions of the compressor, with an increased superheat of 6.3°C obtained with modification II.



a) Suction temperature and superheat



b) Discharge temperature and superheat.

Figure J.3 Compressor behaviour at ambient temperature of –20°C and box temperature of +2°C.

APPENDIX K: TEST DATA FOR SL 400e STANDARD AND MODIFIED UNIT

While Chapters 4 and 5 present an overview of the heating capacity results for both standard and modified SL 400e trailer unit, Appendix K shows the primary test data for the following test conditions:

K.1 Measure of the power heat input H 1 for the calibrated box cooling system at –30°C ambient temperature and +2°C box temperature, while the SL 400e unit is off (Table K.1).

K.2 Heating capacity figure of the standard SL 400e unit as the difference between the electric heat inputs H1–H2, at –30°C and +2°C ambient and box temperatures, for high-speed diesel mode (Table K.2).

K.3 Measure of the power heat input H 1 for the calibrated box cooling system at –30°C ambient temperature and +2°C box temperature (Table K.3), while the modified SL 400e unit is off.

K.4 Main parameters together with the final heating capacity of the modified SL 400e unit (modifications I and III) as the difference between first and second electric heat inputs (H1–H2), at –30°C and +2°C ambient and box temperatures, while the unit operates in high-speed diesel mode, maintaining the ETV at 252 steps (Table K.4).

Table K.1. Test Data for the cooling system at –30° ambient and +2°C box temperatures.

	A	B	C	D	E	F	G	H	I	J
1	date & time	test time	EAO Avg	Trailer Avg	EAI Avg	Ambient Avg	Amb 1	Amb 2	Amb 3	H 1
2	11/15/2003 19:02	0	1.988	-3.883	1.994	-30.685	-28.039	-27.136	-31.37	14258.69
3	11/15/2003 19:02	20	2.187	-3.977	1.913	-30.716	-28.444	-27.167	-31.494	14271.07
4	11/15/2003 19:02	45	2.205	-3.837	1.963	-30.809	-28.35	-27.292	-31.494	14270.58
5	11/15/2003 19:03	65	2.162	-3.764	2.031	-30.623	-28.475	-27.074	-31.494	14260.99
6	11/15/2003 19:03	91	2.193	-3.795	2.081	-30.778	-28.35	-27.167	-31.183	14253.94
7	11/15/2003 19:04	120	2.454	-3.951	2.006	-30.965	-28.257	-27.198	-31.868	14248.59
8	11/15/2003 19:04	140	2.118	-3.837	1.994	-31.027	-28.63	-27.541	-31.65	14248.52
9	11/15/2003 19:04	165	2.261	-3.826	1.988	-30.84	-29.097	-27.572	-31.712	14245.73
10	11/15/2003 19:05	185	1.969	-3.888	1.969	-31.058	-28.288	-27.385	-31.837	14247.06
11	11/15/2003 19:05	210	2.205	-3.665	2.044	-30.84	-28.226	-27.292	-31.868	14263.37
12	11/15/2003 19:06	230	2.38	-3.837	2.1	-30.934	-28.444	-27.447	-31.712	14257.15
13	11/15/2003 19:06	255	1.969	-3.914	1.938	-30.84	-28.475	-27.572	-31.37	14259.72
14	11/15/2003 19:06	275	2.249	-3.863	2.075	-30.716	-28.475	-27.634	-31.152	14259.03
15	11/15/2003 19:07	305	2.554	-3.769	2.131	-30.498	-27.914	-27.292	-30.934	14245.3
16	11/15/2003 19:07	330	2.299	-3.748	2.093	-30.467	-27.914	-27.292	-31.121	14237.94
17	11/15/2003 19:08	351	2.106	-3.8	1.969	-30.623	-28.63	-27.51	-31.307	14222.77
472	11/15/2003 22:04	10935	2.162	-3.847	2.093	-30.903	-29.409	-28.693	-31.432	14224.12
473	11/15/2003 22:04	10956	2.168	-3.743	1.994	-30.623	-29.222	-28.693	-30.965	14224.24
474	11/15/2003 22:05	10980	2.187	-3.785	1.981	-30.623	-29.191	-28.63	-31.058	14225.82
475	11/15/2003 22:05	11000	2.143	-3.655	2.081	-30.498	-28.848	-28.35	-30.84	14226.22
476	11/15/2003 22:05	11026	2.181	-3.743	1.969	-30.809	-29.346	-28.506	-31.307	14226.97
477	11/15/2003 22:06	11055	1.994	-3.925	1.95	-30.716	-29.284	-28.568	-31.121	14227.52
478	11/15/2003 22:06	11080	2.237	-3.759	2.081	-30.467	-29.191	-28.381	-30.965	14229.51
479	11/15/2003 22:07	11101	2.355	-3.613	2.149	-30.591	-29.097	-28.599	-30.934	14228.75
480	11/15/2003 22:07	11121	2.318	-3.743	2.112	-30.591	-29.066	-28.568	-30.84	14225.93
481	11/15/2003 22:07	11145	2.143	-3.837	1.975	-30.809	-29.16	-28.257	-31.401	14225.59
482	11/15/2003 22:08	11175	2.012	-3.894	1.925	-30.84	-29.409	-28.693	-31.307	14228.65
483	11/15/2003 22:08	11200	2.274	-3.795	2.1	-30.685	-29.284	-28.693	-31.027	14227.31
484					2.022861		-28.8725			14227 H 1

Table K.2. Test Data for the Heating Capacity of the standard SL 400e unit at –30°C ambient and +2°C box temperatures.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
1	date & time	test time	EAO Avg	Trailer Avg	EAI Avg	PGOP	PVIP	EVOP	EVIP	EVIT	PGOT	PVIT	AVOT	EVOT	H 2	
2	11/15/2003 15:07	00:00:00	8.008	-1.974	2.921	243.157	31.127	111.679	108.824	5.611	54.856	-24.55	-18.79	3.152	3653.852	
3	11/15/2003 15:07	00:00:28	8.076	-1.969	2.953	227.266	30.783	108.767	109.635	5.704	54.918	-24.9	-19.07	3.245	3661.044	
4	11/15/2003 15:07	00:00:43	8.089	-1.912	3.015	224.997	31.127	109.9	109.635	5.767	54.825	-25.05	-19.29	3.339	3642.214	
5	11/15/2003 15:08	00:00:58	7.995	-2.021	2.847	232.127	30.754	110.223	108.824	5.642	54.825	-24.8	-19.01	3.183	3681.798	
6	11/15/2003 15:08	00:01:13	8.02	-1.948	2.94	238.296	30.783	109.253	109.149	5.673	54.794	-24.93	-19.17	3.183	3741.244	
7	11/15/2003 15:09	00:01:28	8.008	-1.99	2.928	229.534	30.525	109.414	108.824	5.673	54.763	-24.9	-19.07	3.307	3757.414	
8	11/15/2003 15:09	00:01:43	8.051	-1.875	2.965	233.1	30.439	108.929	108.824	5.704	54.794	-25.02	-19.2	3.339	3767.169	
9	11/15/2003 15:09	00:01:58	8.051	-1.938	3.009	240.564	32.187	109.253	109.311	5.767	54.732	-24.99	-19.14	3.307	3754.956	
10	11/15/2003 15:10	00:02:13	8.082	-1.933	3.002	245.426	30.697	110.547	109.311	5.798	54.825	-24.99	-19.2	3.245	3743.205	
11	11/15/2003 15:10	00:02:28	8.07	-1.974	2.959	233.424	31.041	109.738	109.311	5.767	54.918	-24.86	-19.04	3.058	3731.47	
12	11/15/2003 15:11	00:02:43	8.058	-1.964	2.928	224.835	30.564	108.282	109.311	5.704	54.856	-24.86	-19.07	3.245	3756.047	
13	11/15/2003 15:11	00:02:58	8.02	-2.01	2.977	241.375	30.639	111.193	108.986	5.642	54.794	-24.8	-19.01	3.339	3764.457	
14	11/15/2003 15:11	00:03:13	8.033	-1.927	2.903	230.669	30.783	108.929	108.986	5.673	54.856	-24.86	-19.04	3.245	3795.752	
15	11/15/2003 15:12	00:03:28	8.051	-2.01	2.915	226.942	30.754	109.253	108.986	5.673	54.856	-24.68	-18.89	3.245	3794.026	
16	11/15/2003 15:12	00:03:43	8.095	-1.86	2.984	227.59	30.697	108.444	109.149	5.767	54.825	-24.93	-19.14	3.214	3796.997	
17	11/15/2003 15:13	00:03:58	8.07	-1.964	2.99	227.104	30.783	108.606	109.635	5.767	54.856	-24.93	-19.17	3.245	3758.518	
232	11/15/2003 16:36	00:57:43	8.082	-2.021	2.94	224.835	31.184	108.929	108.824	5.673	55.323	-24.93	-19.17	3.214	3884.798	
233	11/15/2003 16:37	00:57:58	8.045	-2.073	2.921	227.752	30.84	109.414	108.662	5.642	55.323	-24.55	-18.79	3.183	3887.297	
234	11/15/2003 16:37	00:58:13	8.082	-2.031	2.965	227.266	30.439	108.767	109.149	5.704	55.323	-24.77	-19.01	3.183	3888.961	
235	11/15/2003 16:38	00:58:28	8.045	-1.99	2.94	227.914	30.725	112.164	108.662	5.642	55.292	-24.68	-18.89	3.37	3889.94	
236	11/15/2003 16:38	00:58:43	8.082	-1.995	2.971	231.155	31.413	110.061	109.149	5.704	55.354	-24.62	-18.83	3.37	3890.934	
237	11/15/2003 16:38	00:58:58	8.12	-2.01	2.99	243.643	30.926	111.679	109.149	5.704	55.354	-24.52	-18.67	3.245	3888.387	
238	11/15/2003 16:39	00:59:13	8.051	-2.01	2.896	240.402	31.212	111.355	108.5	5.642	55.416	-24.18	-18.42	3.089	3888.665	
239	11/15/2003 16:39	00:59:28	8.064	-2.088	2.959	241.213	31.041	111.679	108.986	5.642	55.479	-24.43	-18.67	3.183	3892.767	
240	11/15/2003 16:40	00:59:43	8.051	-2.047	2.896	250.936	33.046	112.002	108.824	5.642	55.51	-24.18	-18.45	3.183	3896.604	
241	11/15/2003 16:40	00:59:58	8.107	-2.062	2.959	238.123	31.041	111.193	109.149	5.704	55.51	-24.49	-18.76	3.245	3899.691	
242	11/15/2003 16:40	01:00:13	8.095	-2.026	2.959	237.961	32.903	112.164	108.824	5.642	55.479	-24.37	-18.58	3.245	3900.66	
243			8.061539	-1.99783	2.95745	235.42	31.014				55.0575	-24.73			3900	H 2
244															14227	H 1
245											Heating Capacity = H1 - H2 =				10327	
246																
247																
248																

Table K3. Test Data for the cooling system at –30°C ambient and +2°C box temperatures/ Modified SL 400e Unit.

	A	B	C	D	E	F	G	H	I	J	K
1	date & time	test time	EAO Avg	Trailer Avg	EAI Avg	Amb Avg	Amb 1	Amb 2	Amb 3	H 1	
2	2/13/2004 19:06	0	2.444	3.038	2.61	-31.603	-30.591	-30.342	-30.125	14309.4	
3	2/13/2004 19:06	0	2.56	3.152	2.71	-30.003	-31.027	-31.183	-31.401	14312.55	
4	2/13/2004 19:06	10	2.553	3.079	2.685	-31.708	-30.809	-30.467	-30.685	14294.7	
5	2/13/2004 19:06	15	2.584	3.1	2.679	-31.774	-31.307	-30.591	-31.37	14284.36	
6	2/13/2004 19:06	25	2.615	3.11	2.747	-30.07	-31.868	-31.245	-31.65	14263.3	
7	2/13/2004 19:06	30	2.381	3.038	2.548	-31.774	-31.152	-30.591	-30.56	14266.76	
8	2/13/2004 19:06	35	2.428	3.079	2.554	-31.505	-30.716	-30.093	-30.934	14269.77	
9	2/13/2004 19:07	46	2.568	3.131	2.822	-30.117	-31.712	-31.245	-31.525	14273.49	
10	2/13/2004 19:07	55	2.475	3.048	2.585	-30.179	-31.619	-31.401	-31.743	14269.61	
11	2/13/2004 19:07	60	2.56	3.1	2.679	-30.179	-31.743	-31.401	-31.432	14273.1	
12	2/13/2004 19:07	70	2.467	3.089	2.616	-30.117	-31.152	-31.276	-31.401	14279.18	
13	2/13/2004 19:07	80	2.467	3.069	2.647	-31.681	-31.058	-30.342	-31.058	14285.42	
14	2/13/2004 19:07	90	2.444	3.048	2.679	-30.101	-32.086	-31.183	-32.179	14262.58	
15	2/13/2004 19:07	95	2.467	3.017	2.573	-30.003	-31.774	-31.027	-32.521	14266.24	
16	2/13/2004 19:08	105	2.529	3.069	2.685	-31.899	-31.65	-30.778	-31.743	14262.33	
17	2/13/2004 19:08	111	2.56	3.058	2.672	-30.008	-31.837	-30.934	-31.868	14262.33	
114	2/13/2004 19:21	915	3.058	3.515	3.083	-30.117	-31.992	-31.774	-32.335	14489.18	
115	2/13/2004 19:21	920	3.004	3.484	2.984	-31.743	-31.401	-31.027	-31.556	14490.75	
116	2/13/2004 19:21	930	3.012	3.473	3.021	-31.85	-31.37	-31.245	-31.556	14487.88	
117	2/13/2004 19:21	935	3.089	3.619	3.027	-31.977	-31.743	-31.494	-32.117	14488.87	
118	2/13/2004 19:22	945	3.191	3.619	3.127	-30.054	-32.086	-31.65	-32.241	14485.97	
119	2/13/2004 19:22	950	3.113	3.556	3.065	-30.179	-32.241	-31.93	-32.335	14482.08	
120	2/13/2004 19:22	961	3.066	3.525	3.002	-30.708	-31.65	-31.121	-32.086	14483.17	
121	2/13/2004 19:22	970	2.747	3.442	2.784	-30.899	-31.774	-31.494	-31.837	14494.24	
122	2/13/2004 19:22	980	2.833	3.536	2.872	-30.85	-31.619	-31.401	-31.93	14504.95	
123	2/13/2004 19:22	990	2.903	3.556	2.884	-30.996	-31.121	-29.689	-31.837	14520.48	
124	2/13/2004 19:22	995	3.035	3.556	3.058	-30.57	-31.712	-30.84	-31.93	14525.9	
125					2.678463	30.18095				14525	H 1

Amb-30 Box 2 Unit Off-0

Table K.4. Test Data for the Heating Capacity of the modified SL 400e unit at –30°C ambient and +2°C box temperatures.

	A	B	C	D	E	F	G	H	I	J	K	L	M
1	date & time	test time	EAO Avg	Trailer Avg	EAI Avg	CAIT Unit	EVOP	EVIP	PVIP	PGOP	PVIT	H 2	
2	4/22/2004 15:06	0	9.642	4.719	3.027	-29.471	117.077	181.484	45.32	208.896	-11.696	2275.035	
3	4/22/2004 15:06	10	9.665	4.729	3.058	-29.206	116.915	180.998	45.512	197.546	-11.696	2275.035	
4	4/22/2004 15:06	20	9.681	4.76	3.127	-28.755	117.402	182.457	45.485	203.546	-11.696	2275.035	
5	4/22/2004 15:06	25	9.673	4.77	3.096	-29.082	118.213	180.673	45.43	199.006	-11.696	2275.035	
6	4/22/2004 15:06	35	9.673	4.76	3.158	-29.673	118.051	182.943	45.375	212.463	-11.696	2275.035	
7	4/22/2004 15:07	45	9.704	4.739	3.114	-29.704	117.239	180.511	45.595	197.222	-11.634	2275.035	
8	4/22/2004 15:07	55	9.704	4.802	3.139	-29.191	116.752	181.808	45.073	205.491	-11.634	2275.035	
9	4/22/2004 15:07	65	9.728	4.667	3.177	-28.802	117.726	181.484	45.512	203.059	-11.696	2275.035	
10	4/22/2004 15:07	75	9.704	4.698	3.046	-29.3	117.402	182.457	45.348	200.789	-11.634	2275.035	
11	4/22/2004 15:07	85	9.704	4.698	3.027	-29.518	117.077	180.836	45.457	194.455	-11.634	2275.035	
12	4/22/2004 15:07	94	9.689	4.708	2.959	-29.689	117.239	182.295	45.457	198.033	-11.696	2152.671	
13	4/22/2004 15:08	105	9.665	4.687	3.077	-29.471	117.402	181.322	45.32	204.356	-11.696	2368.317	
14	4/22/2004 15:08	115	9.658	4.708	3.052	-29.346	117.402	180.511	45.54	197.06	-11.696	2368.317	
15	4/22/2004 15:08	125	9.681	4.698	3.033	-29.704	116.59	181.322	45.265	203.546	-11.696	2368.317	
16	4/22/2004 15:08	134	9.65	4.719	3.009	-29.735	117.402	181.808	45.43	197.06	-11.696	2597.627	
17	4/22/2004 15:08	144	9.658	4.667	3.052	-29.611	118.375	180.511	45.238	200.465	-11.696	2252.361	
324	4/22/2004 15:54	2889	9.681	4.729	3.033	-29.455	117.564	180.187	45.567	192.023	-11.634	2831.456	
325	4/22/2004 15:54	2895	9.658	4.667	3.065	-29.268	117.402	179.376	45.485	199.33	-11.634	2831.456	
326	4/22/2004 15:54	2904	9.65	4.687	3.052	-29.097	117.077	179.052	45.485	199.33	-11.603	2837.413	
327	4/22/2004 15:54	2910	9.658	4.625	3.052	-29.549	117.564	179.538	45.512	194.618	-11.634	2837.413	
328	4/22/2004 15:54	2919	9.658	4.698	2.984	-29.595	118.051	179.214	45.485	209.383	-11.634	2856.87	
329	4/22/2004 15:55	2924	9.658	4.698	2.984	-29.237	117.077	180.025	45.128	197.708	-11.634	2843.258	
330	4/22/2004 15:55	2935	9.696	4.687	3.108	-29.595	116.915	179.701	45.76	201.438	-11.634	2832.96	
331	4/22/2004 15:55	2940	9.689	4.739	3.04	-29.58	117.564	179.538	45.457	197.708	-11.634	2832.96	
332	4/22/2004 15:55	2950	9.673	4.75	3.002	-29.907	117.077	179.863	44.935	189.753	-11.634	2832.96	
333	4/22/2004 15:55	2959	9.65	4.708	3.083	-29.673	117.077	180.673	45.348	197.06	-11.696	2815.149	
334	4/22/2004 15:55	2965	9.658	4.677	3.04	-29.782	117.077	180.187	45.595	207.275	-11.634	2815.149	
335			**9.691102**		**3.048369**	**-29.67**			**45.31803**		**-11.6506**	**2815**	**H 2**
336												**14525**	**H 1**
337								Heating Capacity = H1 - H2 =				11710	
338								**Modifications I and III provide 11.8% increased capacity**					
339													
340													

Amb -30 Box 2 252 steps inj inl

APPENDIX L: FIELD DATA FOR TS 500 UNIT OPERATING IN LOW AMBIENT TEMPERATURES

A complete analysis performed on the TS 500 truck unit operating in low ambient temperature conditions is described in Chapter 6. Appendix L presents supplementary field data that confirms the problems during heat mode highlighted for this type of unit. While Figures 6.8 and 6.9 present the detailed heat mode operation for one day, for a complete picture of the TS 500 unit behaviour, in Figure L.1 the operation of the standard unit on field is presented for three days, during January 2002, with different ambient temperature ranges: i) a relatively warm day with temperatures within –4°C to –2°C, ii) a mild day with temperature of –14°C and iii) a very cold day with –29°C ambient temperature.

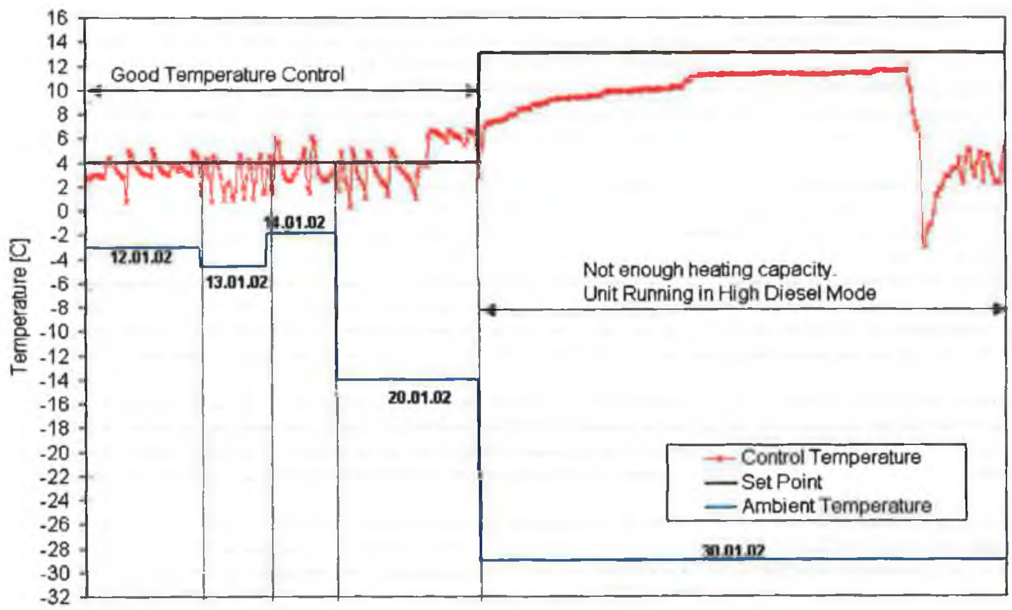


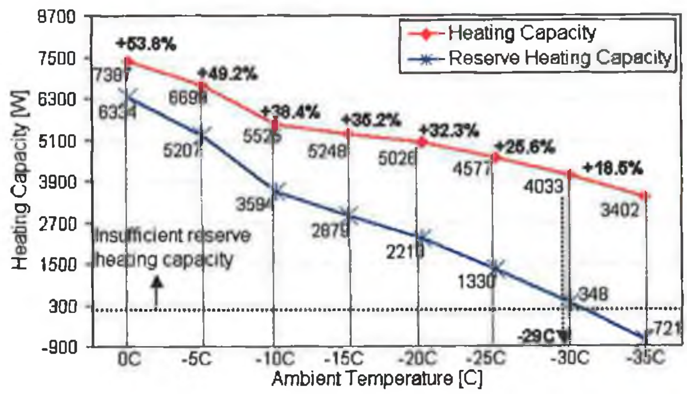
Figure L.1. Swedish Field Data for the TS 500 TTC unit – January 2002 [Nilsson (2002)].

Figure L.1 shows that for higher ambient temperatures like –2°C or –5°C, the temperature control of the unit while running predominantly in heat mode is satisfactory. For a mild ambient temperature like –14°C and a low temperature set point in the box of 4°C, the unit response is good. It can be noticed that the problems appear when the TTC unit is running at very low ambient temperature of –29°C and higher set point in the box +12°C. In these conditions, even if the unit is running in high-speed mode, a lack of heating capacity is noticed.

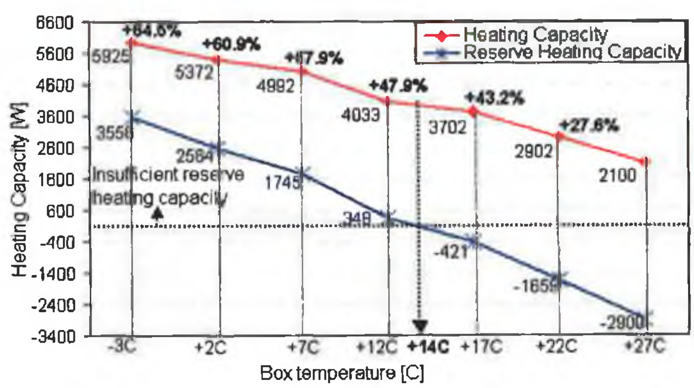
APPENDIX M: STANDARD VERSUS MODIFIEDTS 500 UNITS CHARACTERISATION THROUGH MATHEMATICAL MODELS

M.1 Standard TS 500 Unit Characterisation

One of the most important advantages that the mathematical model can offer is an extended characterisation of the TS 500 unit in heat mode. While Chapter 7 shows the variation of the heating capacity with different ambient and condensing temperatures for low speed diesel mode, Figure M.1(a,b) shows the same type of variations but in high-speed diesel mode.



a) Box temperature +12°C



b) Ambient Temperature –30°C.

Figure M.1 Variation of the heating and reserve capacity with different ambient temperatures and box temperatures for high-speed diesel mode.

It can be noticed that while the decrease in the heating capacity of the unit at ambient temperatures between –10°C and –20°C has an average value of -5%, for lower ambient temperatures this difference increases having a dramatic drop of –11.8% at ambient temperatures lower than –25°C. This is happening due to increased heat losses through the walls in the ambient environment as a result of higher temperature difference between the last one and the box air. Figure M.1(b) shows that the heating capacity of the unit is decreasing with the increase in the condensing temperature. It can be noticed a dramatic decrease in the heating capacity of 27.6% at an increase of the condensing temperature from +22°C to +27°C due to maximum heat losses through the trailer walls. Based on supplementary data obtained through model prediction, it can be observed that the standard TS 500 unit operates at the limit with the reserve heating capacity from ambient temperature lower than –29°C and from box temperatures higher that +14°C in high-speed diesel mode.

M.2 Modified TS 500 unit – Engine Coolant

A typical TTC unit engine is using a closed, circulating type, and pressurised cooling system. A centrifugal pump is circulating the coolant through the cylinder block of the engine to a dedicated engine radiator located behind the condenser. Coolant temperature is controlled and maintained within a specified range by condenser fans and a thermostat located on the engine water outlet. In Chapter 7 two different design modifications are presented based on the effect of the engine coolant used as a primary heat exchange fluid (Figure 7.9): i) a heating coolant coil situated inside the existing accumulator of the heat cycle, and iii) coolant refrigerant heat exchanger located on the compressor suction line. These modifications involve the implementation an on/off solenoid valve installed on the cooling circuit of the TTC unit that is energised in the “Open” position to allow the coolant flow through the new heat exchangers only during the heat cycle.

APPENDIX N: MATHEMATICAL MODEL I - EES ALGORITHM

N.1 Equation Window

While in Chapter 7, section 7.2.1 the main user interface of the EES mathematical model I for the standard TS 500 unit was overviewed in Figure 7.2, in Appendix N a sample of the main equations and the EES program algorithm is presented. Figure N.1 shows the equation window of the Engineering Equation Solver (EES) software. The basic capability provided by EES is the solution of a set of non-linear algebraic equations. The main variables and the equations used for the calculation algorithm are introduced in the equation window, while the "Options" menu provides commands for setting the guess values and bounds of variables, the unit system, default information, and program preferences.

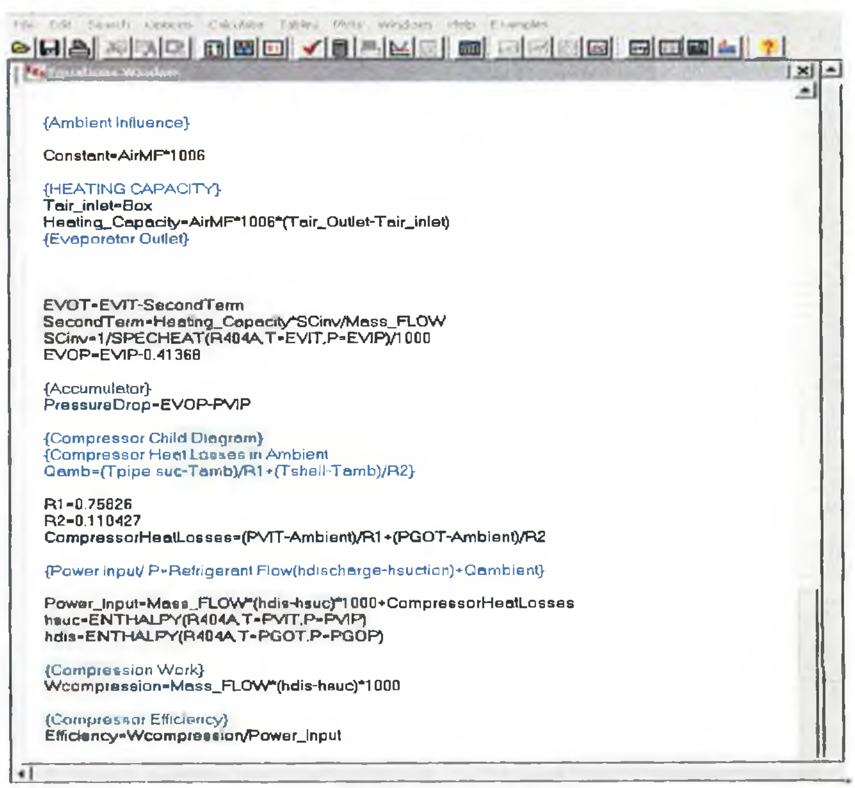


Figure N.1 Sample of the equation window of the EES mathematical model I algorithm of the standard TS 500 truck unit.

APPENDIX O: PUBLISHED WORK-ABSTRACTS

O.1 Paper 1

Title: **"Importance of Refrigeration System Heating Mode to Maintain Specified Temperatures During Food Transportation"**

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Conference: Eurotherm Seminar Thermodynamics Heat and Mass Transfer of Refrigeration Machines and Heat Pumps, Valencia, Spain, April 2003.

Abstract

While refrigeration systems are synonymous with cooling, this paper reviews the importance of the heat mode in maintaining accurate temperature control during the transportation of perishable food products, especially over winter months in cold climates. Such refrigeration systems are capable of maintaining good temperature control by automatically providing heating or cooling as required and this paper seeks to establish the relative importance of the heat and cooling modes. This is achieved by documented information from four sources: i) recent market trends for transport temperature control systems; ii) recommended transport temperatures for perishable food products and deterioration rates of both perishable and frozen food products as a function of temperature; iii) weather data reports for different climatic regions, and iv) actual field data over one year period for refrigeration system operated by a Swedish food distributor. It is concluded that a more detailed and rigorous characterisation of temperature control systems is required for heating mode, and a call is made to develop industry wide test facilities and test procedures for establishing system-heating capacity so that performance could be assured in colder climates.

O.2 Paper 2

Title: **"Performance Characterisation of Transport Temperature Control Units in Heat Mode – A New Test Method"**

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Conference: 21st International Congress of Refrigeration, IIR, Washington D.C., U.S, August 2003.

Abstract

A recent study has shown that transport temperature control (TTC) units used for the transport of perishable foodstuffs, can operate in heat mode for up to 60% of the year in regions north of the 55° latitude [1]. Despite the clear importance of heat mode during the transportation of perishable foods, current test methods used to evaluate heating capacity lack accuracy. As a result, this paper presents a new test method for accurately measuring the heating capacity of TTC units operating in low ambient temperatures, below -10 °C. Existing test methods allow heating capacity to be evaluated to an accepted accuracy of ± 10%, based on either measurement of airflow rate through the evaporator and air inlet outlet temperature difference across the evaporator, or refrigerant flow rate and refrigerant temperature difference across the evaporator. However, the new method proposed in this paper is based on electrical heat input that can be measured to an accuracy of ± 1%. The new test method was performed on a truck TTC unit, positioned within a test facility capable of generating controlled low ambient air temperatures and the results are compared with those obtained using existing methods. In contrast with the existing test methods, this new heating capacity test procedure is based on the measurement of electrical power input from electric heaters and can provide results that account for additional heat influence from the evaporator's fans and non-uniform airflow through the evaporator. The test procedure described is generic and can be applied to determine heating capacity of any TTC unit that employs hot gas as a working fluid.

O.3 Paper 3

Title: **"Cooling and Heating Capacity Evaluation for Single and Multi Compartment Transport Temperature Control Units".**

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Conference: International Conference of Energy and Environment CIEM, Bucharest, Romania, October 2003.

Abstract

This paper describes the test procedures and test facilities used to evaluate cooling and heating capacities of both single and multi compartment transport temperature control (TTC) units. The purpose of the test facilities is to simulate typical ambient operating conditions that TTC unit are exposed to and produce standard performance indicators or characteristics at these controlled environments. The paper focuses on: i) the implementation of an improved test facility for cooling evaluation of single-compartment TTC units, including the control and data acquisition systems, and ii) construction of the first industry test facility capable of performing heating capacity tests based on a highly accurate new test procedure for single- and multi compartment TTC units.

O.4 Paper 4

Title: **"Predicting the Performance of a Single Compartment Transport Temperature Control Unit in Heat Mode".**

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Conference: 4th Thermal Science Conference, Birmingham, U.K, April 2004.

Abstract

Despite the clear importance of heat mode during the transportation of perishable foods [1, 2, 3], most research has focused on the experimental and predictive analysis of transport temperature control (TTC) units operating in cooling and defrost modes [4, 5, 6]. As a result, this paper describes the development of the first steady - state mathematical model, capable of predicting both the system behaviour and capacity in heat mode of a single – compartment TTC unit used for the transportation of perishable food. The relative precision and the robustness of the mathematical model is determined by comparing the model predictions with experimental results obtained using two test methods applied for measuring the TTC unit heating capacity: i) the existing standard test method based on air flow and temperature difference measurements across the evaporator, upon which a heat balance calculation establishes performance to an accuracy of ± 10% and, ii) a new test method proposed by the current authors, based on the measurement of electric heat input to the higher accuracy of ± 3 %. An empirical correction factor was introduced to account for the influence of the ambient air temperature and heat loss from the air circulating fans on the evaporator air inlet – outlet temperatures is proposed so that predictive accuracy could be improved. As results are typically within ± 6%, this model is proposed for the prediction of heating performance for single-compartment TTC units.

O.5 Paper 5

Title: **"Impact of hot-gas injection on the Heating Capacity of a Transport Temperature Control unit Operating in Low Ambient Temperatures".**

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Conference: Future International IIR Conference on Latest Developments in Refrigerated Storage, Transportation and Display of Food Products, IIR, Amman, Jordan, March 2005.

Abstract

Despite the clear demand from food distributors situated in cold climates for transport temperature control (TTC) units with adequate heating capacity [1], the amount of research conducted on this aspect of unit behavior is very limited [2, 3]. As a result, this paper sought to address this weakness in the literature by presenting the benefits of using: i) hot-gas injection technique and ii) a refrigerant charge control system on the heating behavior of a TTC trailer unit used for long distance road transportation of perishable food products, while operating in sub-zero Celsius degrees temperature conditions. The results show a significant increase of the heating capacity by +8.4% and +15% respectively, while the unit operates with lower consumption and higher efficiency. Noticeable improvement in the compressor behavior and pull-up time was also obtained.