Department of Mechanical Engineering

Transcritical CO_2 Air Source Heat Pump for Average UK Domestic Housing with High Temperature Hydronic Heat Distribution System

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Signed: Venkata Satish Duddumpudi Date: 25/12/2010

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Abstract

The main purpose of this project is to develop a platform to investigate the performance of an Air to water residential CO_2 heat pump which reduces the severe environmental problems causing by synthetic refrigerants, for Space heating and domestic hot water, in an average UK domestic house with high temperature hydronic heat distributing system.

For this a numerical simulation model of Air source Transcritical CO_2 heat pump system was developed based on thermodynamic equations using Engineering Equation Solver (EES). Then factors to be considered in components selection in the construction of CO_2 heat pump cycle and numerical simulator and the modifications that improve the performance of the heat pump cycle were studied and analyzed, the values of the model parameters are estimated based on measured data for existing devices for the optimum output from the simulation system. The equations developed to estimate the performance and the behaviour of the components used in the heat pump cycle were based on the thermodynamic relations and laws. And then to validate the model, its performance was compared with Dr Jørn Stene's CO_2 Heat Pump simulation model at similar operating and design conditions.

Finally, the simulation code for CO_2 heat pump was developed in such a way that, the performance of the heat pump can be evaluated when retro-fitted in any domestic housing by running the code with heat demand (space heating and domestic hot water demand) input. In this project heat loads for average UK domestic house and the climatic data were taken from ADEPT developed by ESRU, University of Strathclyde were chosen to find the performance of the CO_2 heat pump when retro-fitted as future research with few modifications to the heating distribution and heat generation system.

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Last but not least **MY FAMILY,** I never forget their help and support in my life and at the times I couldn’t take breath on my own and couldn’t get up from bed. Their help and support was unforgettable.

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1. INTRODUCTION

Now a day's usage of synthetic refrigerants was increasing rapidly through out the world in varied sectors of the refrigeration and air-conditioning, leakage of these refrigerants causes severe environmental problems like stratospheric ozone depletion and global warming.

1.1. NEED FOR ENVIRONMENTAL FRIENDLY REFRIGERANTS

According to The 2007 report on refrigerant usage across the globe and the industry, released by the International Panel for Climate Change (IPCC), the typical average annual leakage rate in these sectors as percentage of total system charge is around 30%, especially leakage rate of 18% in heat pump sector, an example for the scale of carbon foot print of these leakages by Danfoss,

'A supermarket that uses 500 kg of the HFC refrigerant R404a and has an annual leak rate of 20% will have a carbon foot print equal to driving 3,000,000 km in an average family car or 100 cars driving continuously through a full year'

HydroFluoro Carbons (HFC)	HydroChloroFluoro Carbons (HCFC)	ChloroFluoro Carbons (CFC)	Natural Refrigerants
R-404A	*R-22*	*R-12*	*CO_2 (R-744)*
R-507	*R-123*	*R-502*	*Ammonia (R-717)*
R-134a			
R-410a			
R-407C			

Table 1.1.1 Synthetic and Natural Refrigerants

These issues mounting pressure on synthetic refrigerants, pushing the industries and nations towards more environmentally benign alternatives and many conventions are framed to control these synthetic refrigerants threatening the environment, like Montreal Protocol in 1987, Kyoto Protocol 1997 and EU F-gas reg. 2006, forcing the companies to search for environmental acceptable refrigerants by phasing out these synthetic refrigerants to combat with the ozone layer depletion and global warming issues. After an extensive research, the only alternative working fluid or refrigerant found for these synthetic refrigerants are naturally occurring and ecologically safe refrigerants, called as natural refrigerants. The natural refrigerants are not totally new for the market, they were in use long back until CFCs and HCFCs were introduced in market.

1.2. NATURAL REFRIGERANT CARBON DIOXIDE (CO_2, R-744)

Natural refrigerants are naturally occurring, ecologically safe working fluid. There are few different types of refrigerants in this group like Carbon-dioxide (CO_2), Ammonia (NH_3) etc. Out of these natural refrigerants, from an environmental point of view carbon dioxide (R-744) regarded as a best working fluid since it is non-toxic, non-flammable and neither contributes to ozone depletion nor global warming. In 1993 Lorentzen and Pettersen rediscovered Carbon-dioxide (CO_2, R-744) as a working fluid for refrigeration and air-conditioning. Due to its environment friendliness, low price, easy availability, non-flammability, non-toxicity, compatibility with various common materials, compactness due to high operating pressures, excellent transport properties made it best working fluid.

Refrigerant	R-744
Chemical formula	CO_2
Critical Temperature (°C)	31.1
Critical Pressure (kPa)	7384
Critical Density (kg/m³)	466.5
Boiling Point at Atm. Pressure (°C)	-78.4
Flammability	No
Toxicity	No

Table 1.2.1 General Properties of Refrigerant R-744

(source: Properties of CO_2, Union Engineering)

Because of the above characteristics, carbon dioxide systems are best suit in Mobile Air-conditioning and Heat Pumps for heating and cooling, several researches have been carried out in these sectors proved the above statement.

1.3. R-744 VERSUS SYNTHETIC REFRIGERANTS

We already know the natural refrigerants are naturally occurring and they are ecologically friendly so they are having less negative impact on the environment which made this refrigerant group as best working fluid in combating global warming, and synthetic refrigerants are the worst pollutants compared to the natural refrigerants. Following table provide evidence to justify the above statement.

Group	**HFC**		**HCFC**		**CFC**		**Natural**	
Refrigerant	**410A**	**134a**	**22**	**123**	**12**	**502**	**717**	**744**
Chemical formula	—	CH_2FCF_3	$CHClF_2$	$CHCl_2CF_3$	CCl_2F_2	—	NH_3	CO_2
ODP	0	0	0.055	0.02	1	0.25	0	0
GWP	2,100	1,430	1,810	77	10,900	4,700	< 1	1
Flammability	No	No	No	Yes	No	No	Yes	No
Toxicity	No	No	No	No	Yes	No	Yes	No

Table 1.3.1 Comparing characteristics of different refrigerants

(Source: Tiansuwan, J., CO_2 heat pump water heater simulation)

The measure of a materials ability to deplete stratospheric ozone is its Ozone Depletion Potential (ODP), a value relative to that of R-11 which is 1.0. The CFC and HCFC group refrigerants got higher ozone depletion potential (ODP), out of these two CFCs having more ODP, where HFCs and Natural refrigerants having zero potential for ozone depletion.

Halocarbons and non-halocarbons like CFCs, HCFCs, HFCs, hydrocarbons and CO_2 are also Green House Gases (GHG) like other gases warming up the globe. The global warming potential (GWP) of a GHG is an index describing its relative ability to trap radiant energy compared to CO_2 (R-744). Here also CFCs are having higher global warming potential and HFCs and HCFCs comes next compared to the natural refrigerants. It is also important that refrigerant should be safe to use like less in toxicity and non flammable in domestic application.

Environmentally preferred refrigerants should have (1) low or zero ODP, (2) low GWP, (3) appropriate safety properties, (4) provide good system efficiency. By considering taking all these characteristics, R744 is chosen as replacement for synthetic refrigerants in domestic air-conditioning and heat pumps.

1.4. TRANSCRITICAL CO_2 HEAT PUMP CYCLE AND REFERENCE CYCLE

Carbon dioxide based vapour compression refrigeration system was in use long back. Later on carbon dioxide was replaced with synthetic halocarbon refrigerants, due to the problems arising with its low critical temperature and high operating pressure. Later G. Lorentzen (1994) re-introduced carbon-dioxide as working fluid to replace harmful synthetic refrigerants by operating the cycle at transcritical region to overcome the problem of carbon-dioxide lower critical temperature. This led to the development of transcritical carbon-dioxide cycles and further interest in carbon-dioxide based systems in varied applications.

In a transcritical carbon-dioxide cycle, heat is absorbed at a constant temperature at sub-critical pressure and given off at a gliding CO_2 temperature at super-critical pressure. The real transcritical CO_2 heat pump cycle is referred to as the Lorentzen cycle; following pictures illustrate the changes of state in a transcritical CO_2 heat pump cycle.

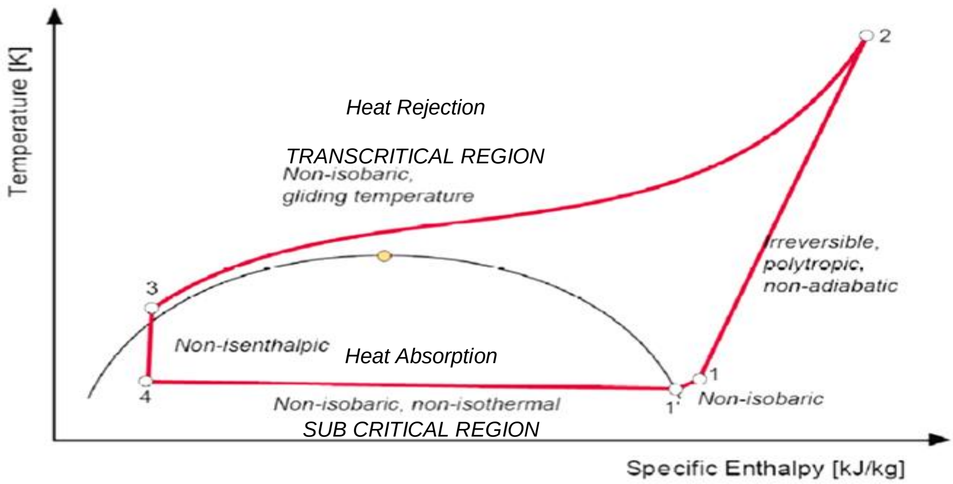


Figure 1.4.1 Transcritical CO_2 heat pump cycle (Lorentzen cycle) T-h diagram (Source: J.Stene, Intergrated CO_2 heat pump for space and hot water in passive houses)

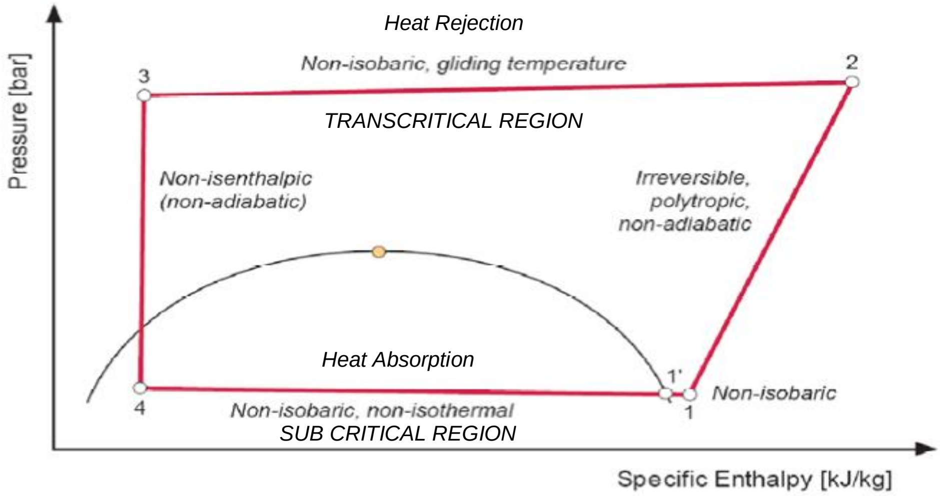


Figure 1.4.2 Transcritical CO_2 heat pump cycle (Lorentzen cycle) P-h diagram (Source: J.Stene, Intergrated CO_2 heat pump for space and hot water in passive houses)

1.5. TRANSCRITICAL CO_2 HEAT PUMP SYSTEM AND COMPARISION WITH CONVENTIONAL SUB-CRITICAL VAPOUR COMPRESSION SYSTEM

A transcritical CO_2 heat pump cycle is quite different from a conventional sub critical vapour compression heat pump cycle, a transcritical cycle operate beyond the critical pressure where a conventional sub-critical vapour compression cycle operates under the critical pressure of the working fluid.

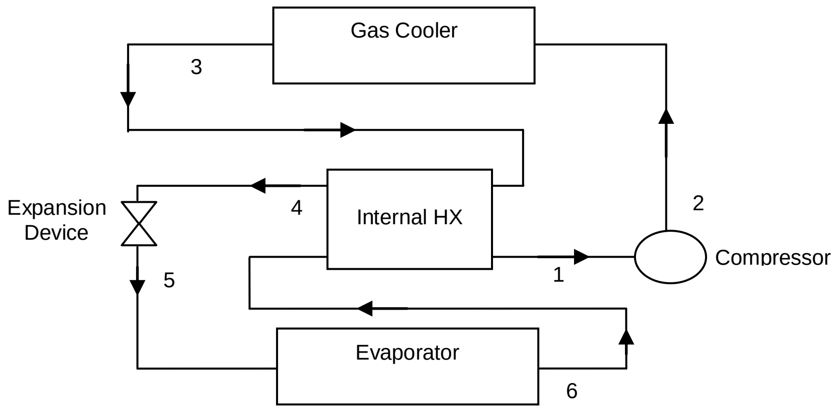


Figure 1.5.1 Layout of transcritical CO_2 system with internal heat exchanger

In a transcritical CO_2 heat pump system the heat from the source was absorbed by refrigerant in evaporator get compressed in a compressor to a desired pressure which is above the critical pressure of the refrigerant, the heat gained in the compression process is given off through the gas-cooler where the heat is given or rejected to the desired space or heating medium at an unusual large temperature glide. Usually a condenser is used in conventional sub-critical vapour compression heat pump system to give off the heat. Cooled refrigerant from the gas cooler is passed through an expansion device to bring the pressure under critical pressure and passes through the evaporator again continues the cycle. Due to the temperature glide at gas-cooler, CO_2 heat pump system offers several unique possibilities such as simultaneous refrigeration and water heating /steam production, simpler control of capacity makes transcritical CO_2 heat pump system more suitable for air-conditioning and heat pumps.

Due to huge expansion losses in transcritical CO_2 heat pump system, coefficient of performance (COP) of the cycle is less compared to conventional heat pump cycles. Several researches were carried to improve the COP of the cycle by modifying the cycle, Lorentzen and Pettersen (1993) improved the cycle COP by introducing a heat exchanger between cooler exit and evaporator exit to further cool the refrigerant before throttling to reduce the throttling loss, there are several other modifications possible to improve the COP of the cycle discussed in later chapters.

2. BACKGROUND OF RESIDENTIAL TRANSCRITICAL CO_2 HEAT PUMP SYSTEMS

Since all residential heat pumps in the market and in use are charged with HFCs, it is relevant to examine whether CO_2 heat pumps can be successfully applied in the residential sector. In recent years a number of universities, research institutions and companies have been evaluating and testing various types of residential CO_2 heat pump systems and its essential components. Some applications include CO_2 heat pump water heaters, CO_2 heat pumps in combination with low-temperature heat distribution system, Integrated CO_2 heat pump systems for space heating and hot water heating in low-energy houses and passive houses etc.

First residential CO_2 heat pump water heaters were commercially introduced into market in Japan by ECO CUTE in collaboration between TEPCO, Denso and CRIEPI in 2001, later on many companies like SANYO, DAIKIN launched residential CO_2 heat pump units. But the performance ratings and performance factors discussed or showed in many product catalogue and research papers are about the newly installations, exactly it is not known how a new CO_2 heat pump unit perform in a retro fitting condition, this work is carried to know the performance of CO_2 heat pump retro fitting in a average domestic housing.

2.1. RESIDENTIAL CO_2 HEAT PUMP WATER HEATERS

One of the most promising applications of the CO_2 heat pump process is production of domestic hot water (DHW), demonstrated by Lorentzen. CO_2 heat pump water heater got high energy efficiency due to the good temperature fit between the working fluid and the water in the counter-flow gas cooler, the excellent heat transfer properties of the CO_2 and the high compressor efficiency. Another advantage of the CO_2 heat pump water heater is the capability of supplying high temperature DHW, eliminates supplementary heating requirement.

Many number of CO_2 heat pump water heater prototypes have been tested by many researchers like Neksa et al, Saikawa and Hashimoto, virtually all installations have been single-stage units using a low-pressure liquid receiver, a suction gas heat exchanger and a counter-flow tube-in-tube gas cooler, similar to the below;

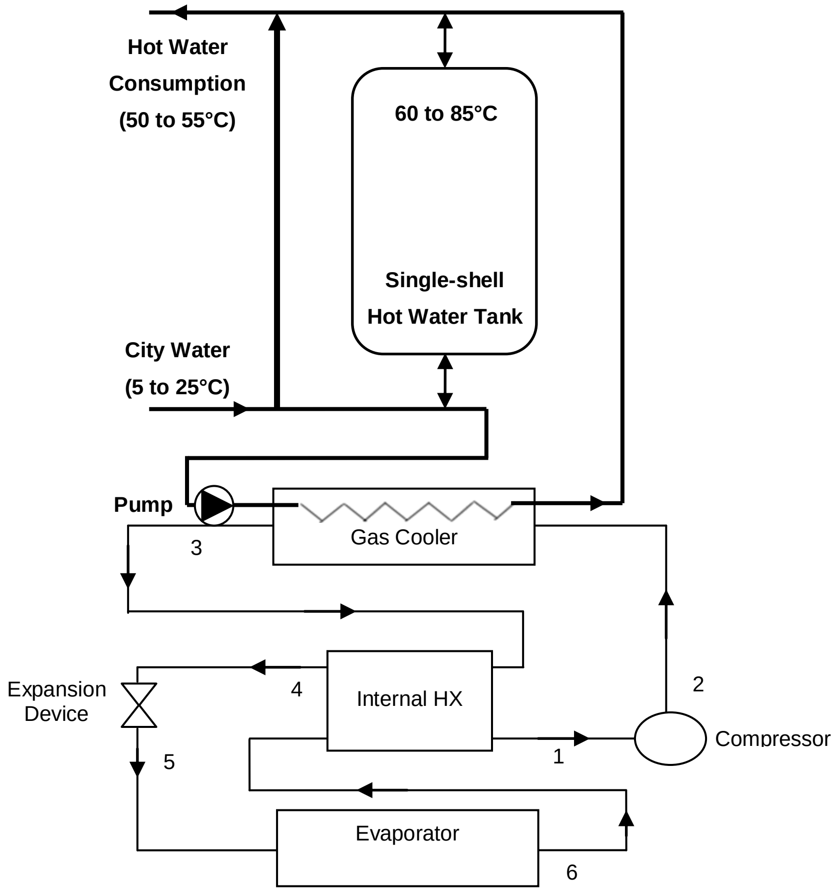


Figure 2.1.1 Simple CO_2 heat pump water heater

Neksa, P. and Rekstad results of 50 kW CO_2 heat pump water heater test rig shows, the COP of the test rig ranged from about 3.0 to 4.3 when the DHW set-point was 60°C and evaporator temperature ranges from -20 to 0°C. The measured COP of the unit was about 3.6 at 80°C DHW temperature and 0°C evaporation temperature. The optimum gas cooler pressure or high-side pressure ranges from about 9 to 11 MPa at DHW temperatures between 60 and 80°C. Measured COPs of prototype investigated by other researchers were found in the same range as the CO_2 heat pump unit which was tested by Neksa, P., Rekstad. In 2001, Denso Corporation Ltd. launched residential air-source CO_2 heat pump water heater in Japan. The 4.5 kW unit delivers 85°C DHW, and the measured SPF in Tokyo climate is reported to be above 3.0.

2.2. RESIDENTIAL CO_2 HEAT PUMP SYSTEM FOR SPACE AND HOT WATER HEATING IN LOW ENERGY AND PASSIVE HOUSES

Jorn. Stene carried a research on residential CO_2 heat pump system for space heating and hot water heating in low energy houses and passive houses, where houses are super insulated and air-tight buildings where the space heating demand is considerably lower than the buildings constructed in accordance with common building codes. Due to the lower space heating demand, the annual heating demand for domestic hot water typically constitutes 50 to 85% of the total heating demand in the residence.

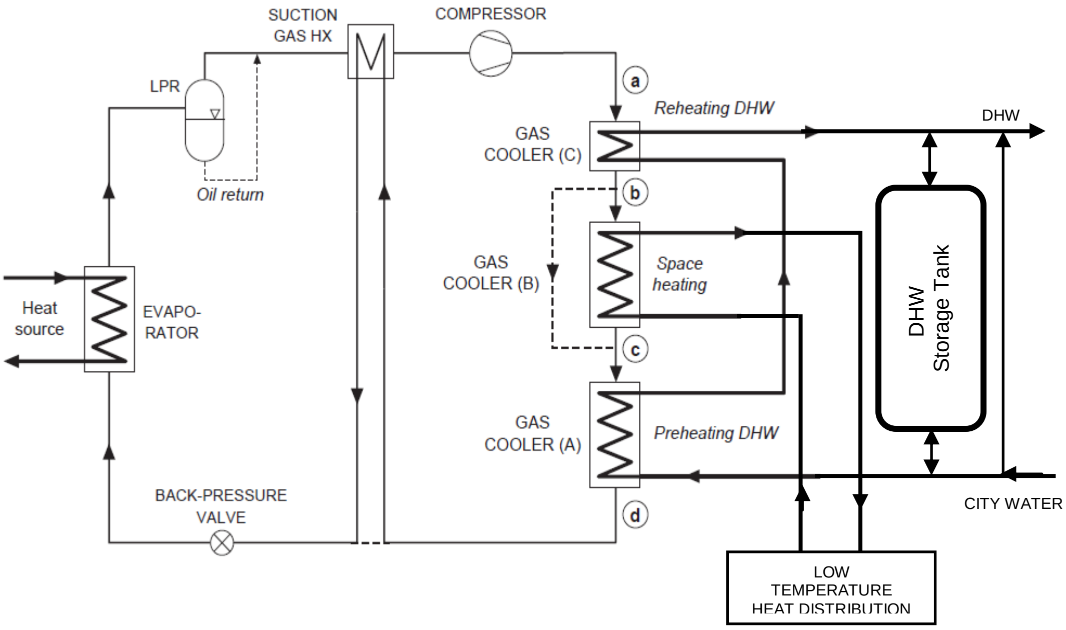


Figure 2.2.1 Principle design of Integrated CO_2 heat pump for space and DHW heating for Passive housing

(Source: J.Stene, Intergrated CO_2 heat pump for space and hot water in passive houses)

In his research different integrated CO_2 heat pump systems have been investigated, focusing on the design of the gas cooler and the DHW system, found that a counter-flow tripartite CO_2 gas cooler in combination with an external single-shell hot water tank and a low-temperature heat distribution system shown in above figure, produce DHW in the required temperature range from 60 to 85°C and contribute to the highest possible COP for the CO_2 heat pump system.

The prototype CO_2 heat pump was equipped with a hermetic rolling piston compressor, a tripartite counter-flow tube-in-tube gas cooler and a counter-flow tube-in-tube suction gas heat exchanger. To control pressure in the gas-cooler a back pressure valve as an expansion valve and a low-pressure liquid receiver were used. Gas cooler units A and C were connected to an un-vented single-shell DHW storage tank and gas-cooler unit B was connected to a low temperature hydronic heat distribution system.

Here, the integrated CO_2 heat pump system was tested in three different modes, (1) simultaneous space heating and DHW heating (combined mode), (2) Hot water heating (DHW mode), (3) space heating (SH mode). During the tapping, hot was delivered at the tapping site, while cold city water entered the bottom of the hot water storage tank. During the charging of the DHW tank in the combined and DHW modes, the cold city water from the bottom of the DHW tank was pumped through gas cooler units A and C, heated to the desired temperature level, and returned to the top of the tank. The CO_2 system was tested at 40/35°C, 35/30°C or 33/28°C supply/return temperatures in the SH system, and 60, 70 or 80°C in the DHW system.

The measure coefficient of performance (COP) at optimum gas cooler pressure, various supply/return temperatures for the space heating systems and 60°C DHW temperature were,

SPACE HEATING SYSTEM Supply/return temp	DHW DEMAND RATIO	COP in Combined mode	COP in DHW mode	COP in SH mode
33/28	49%	3.98	3.8	3.15
35/30	57%	3.89	3.8	3.01
40/35	69%	3.86	3.8	2.78

Table 2.2.1 Measured COP of the CO_2 heat pump system at different modes for different space heating system temperatures and DHW demand ratio

(Source: J.Stene, Intergrated CO_2 heat pump for space and hot water in passive houses)

In a conventional HFC heat pump system for a 35/30°C space heating system in space heating mode COP of the system is 4.8, in DHW mode at 60°C set point temperature and 10°C cold water temperature COP of the system 3.0. So in SH mode a conventional HFC heat pump system performs better than a CO_2 heat pump system, vice versa in DHW mode.

2.3. RESIDENTIAL CO_2 HEAT PUMPS IN COMBINATION WITH LOW-TEMPERATURE HEAT DISTRIBUTION SYSTEM

Kerherve and Clodic compared the performance of a prototype CO_2 heat pump unit with a state of the art R-407C air to water heat pump. During CO_2 heat pump unit testing the supply and return temperature in the hydronic heat distribution system were 32/28°C in floor heating system or 47/43°C in convector system, inlet temperature to the evaporator was varied between -15 to +7°C in each test series and optimum high-side pressure control was used in all experiments.

For ambient air temperatures higher than -10°C COP of the CO_2 heat pump was much lesser than that of R-407C the main reason for the inferior energy efficiency of the CO_2 heat pump unit was the poor temperature fit in the gas cooler and the subsequent high CO_2 temperature before throttling. The only advantage of the CO_2 system was that the heating capacity diminished less rapidly than that of the R-407C system at ambient air temperature below -5°C.

The test data were used to estimate the SPFs for the heat pump units when they were supposed to cover the heating demand of a typical 140m^2 house situated in different climatic regions in France. In the calculations it was presupposed that the heat pumps were operated as monovalent systems. The calculated SPF for the R-407C unit was in average 25% higher than that of the CO_2 unit. Since the CO_2 heat pump had the highest heating capacity at lower ambient air temperatures, the difference in SPF would have been less pronounced if the units had been operated as bivalent heating systems using an external heater to cover the peak load demand.

3. HEATING DEMANDS IN AVERAGE UK DOMESTIC HOUSING

Heating demands both space heating and domestic hot water (DHW) heating in UK domestic housing data, for which the CO_2 heat pump simulator performance investigated is taken from the Advanced Domestic Energy Prediction Tool (ADEPT), developed by Energy Systems Research Unit (ESRU), University of Strathclyde.

3.1. MODELLING DETAILS OF AVERAGE UK DOMESTIC HOUSE

The main objective of Advanced Domestic Energy Prediction Tool (ADEPT) is to produce a controls evaluation methodology based computer modelling of domestic housing and heating systems, for this five different types of houses and five control systems were considered. House type ranges from relatively unimproved turn-of-the-century solid wall building through to modern 2006 regulation complaint construction and controls ranges from a basic system with single room thermostat, through to a two-zone system with two independent thermostats.

3.1.1. DETAILS OF HOUSE TYPE USED FOR THE INVESTIGATION

Out of five different houses models based on BRE specifications, Semi-detached house, EEC stock average house with 90 m^2 floor area and 100mm loft insulation, filled cavity was chosen as house to investigate the performance of CO_2 heat pump.
This house have been modelled with a main living zone, and non-living zone representing the rest of the house. The living zone consists of living and dining room, around 35% of the floor area. Air change rate is fixed at 0.7 for all models. Systems components are in the non-living zone except the living zone thermostat/controller, valve and radiator.

3.1.2. SPACE HEATING SYSTEM AND ITS COMPONENTS

Heating system for this house type consists of two circuits serving the main living, and non-living. The radiators in the zones have been sized according to typical installers practice, for this house type radiators are sized 3.5 kW and 5.4 kW for living and non-living zones respectively. Radiators are sized at -2°C external temperature, with 20% oversize for heat up. Heating set-points are 21°C in the living zone and 18°C in the non-living zone.

Two heating schedules are used, according to Standard Assessment Procedure (SAP);

1. Intermittent operation at weekdays from 07:00 to 09:00 and 16:00 to 23:00
2. Continuous operation at weekends from 07:00 to 23:00

Water flow from the circulation pump is apportioned to each circuit by taking into account the position of on/off (zone) and modulating (TRV) valves. Flows are apportioned according to the relative system sizes to give the correct design temperature drop across radiators. If the pump has to run with no zone calling for heat, the flow goes through a bypass.

LIVING ZONE RADIATOR		**NON-LIVING ZONE RADIATOR**	
Nominal heat emission	3.53 kW	Nominal heat emission	5.42 kW
Nominal supply temp.	80°C	Nominal supply temp.	80°C
Nominal return temp.	70°C	Nominal return temp.	70°C
Nominal environment temp.	21°C	Nominal environment temp.	18°C
Heat transfer exponent	1.21	Heat transfer exponent	1.21

Table 3.1.2.1 Hydronic heat distribution system component description

(Source: Report on ADEPT by ESRU, 2007)

Hot water circulation pump in the heat distribution system send hot water flow to each circuit according to the call by the thermostat in the zones. Description of the circulation pump used here shown in the below table,

Description	**Value**	
Maximum flow rate	0.214	kg/s
Open circuit flow rate	1.0	kg/s
Bypass setting (fraction of maximum pump flow)	0.33	
Total mass	5	kg
Mass weighted average specific heat	2250	J/kg.K
Heat transfer coefficient (to containment)	0.2	W/K
Total absorbed power	150	W

Table 3.1.2.2 description of hot water circulation pump

(Source: Report on ADEPT by ESRU, 2007)

3.1.3. DOMESTIC HOT WATER SYSTEM AND ITS SPECIFICATIONS

Domestic hot water can be serviced in the same way as the space heating circuits, with water diverted to the hot water storage cylinder during the call for DHW. DHW temperature supplied depends on the stored hot water temperature, maintained according to cylinder thermostat and the heating time schedules. Thermostat switches on at 50°C and off at 60°C, water supply temperature enters at 10°C.

The DHW draw-off comprises of 10 draw-offs totalling 122 l/day approximates to the Standard European Pattern. All draw-offs occurs during the heating system on times, the hot water draw off schedules are as below.

Hot water draw-off,

Schedule no.	Time of the day	Duration (sec)	Flow rate (l/s)
1	07:06	36	0.0855
2	07:24	61	0.099
3	07:42	748	0.09
4	08:00	72	0.0855
5	08:24	61	0.099
6	08:54	61	0.099
7	17:00	108	0.0855
8	17:24	61	0.099
9	19:24	61	0.099
10	22:24	61	0.099

Table 3.1.3.1 Domestic Hot Water (DHW) draw-off schedule in the chosen house type and control system (Source: Report on ADEPT by ESRU, 2007)

DHW storage tank used has a capacity of 120 litres, and is insulated to BS1566, description of the hot water storage tank shown in the table below,

Description	Value
Total mass	120 kg
Mass weighted average specific heat	4180 J/kg.K
Heat transfer coefficient	1.03 W/K
Mass of water in coil	15 Kg

Internal heat transfer area	0.5 m^2
Internal heat transfer coefficient	12 $kW/m^2.K$
External heat transfer area	0.55 m^2
External heat transfer coefficient	1.2 $kW/m^2.K$

Table 3.1.3.2 Hot water storage cylinder

(Source: Report on ADEPT by ESRU, 2007)

3.1.4. CONTROL SYSTEM FOR HEATING

The control scheme applied in this house type, Mechanical thermostat for living room heating and TRVs in non living room heating. The following diagram shows different units in the heating control system,

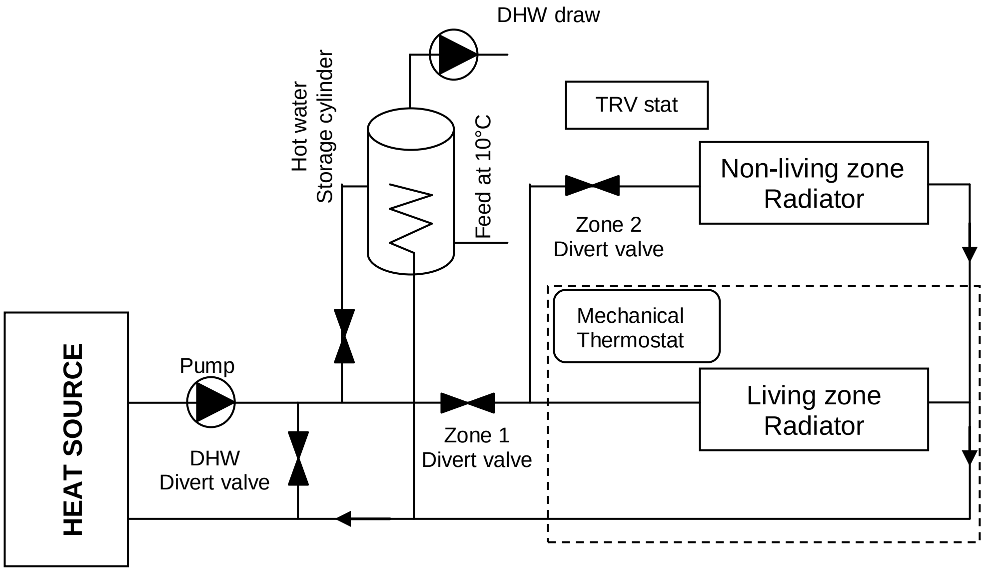


Figure 3.1.4.1 Schematic diagram of control system modelled in chosen house type

(Source: Report on ADEPT by ESRU, 2007)

The current control system uses Mechanical thermostat in living zone with temperature on and off set-points, 20.75°C and 21.25°C respectively, and TRV thermostat in non-living zone with valve fully closed at 20°C and valve fully open at 16°C. Circulating pump pumps water to each circuit according to thermostat on/off position.

On/off room thermostat:

The dynamic aspect of the thermostat model consists of a sensor with radiative convective and selected wall coupling to the zone, with thermal mass, and definable anticipator heating effect. The output of the dynamic model is a sensor temperature which defines the on/off status of the thermostat according to set point and mechanical differential settings. If the thermostat is on then the control output hot water temperature set point is set to the maximum. If the thermostat is off, then the control output is set to the minimum.

3.2. HOT WATER TEMPERATURE RQUIREMENT FOR HEATING

In order to investigate the performance of residential CO_2 heat pump system, energy consumption in the chosen house with control system, it is necessary to run the heat pump simulation for seasons of the year. ADEPT got the eight days of data representing typical whether for different seasons of the year including weak days and weak ends. The days selected are shown in the below table

SEASON	DAY TYPE	DAY
Early winter	Weekday	11th Jan
	Weekend	3rd Feb
Spring/Autumn	Weekday	26th Mar
	Weekend	12th Oct
Summer	Weekday	19th May
	Weekend	23rd Oct
Late winter	Weekday	13th Nov
	Weekend	14th Dec

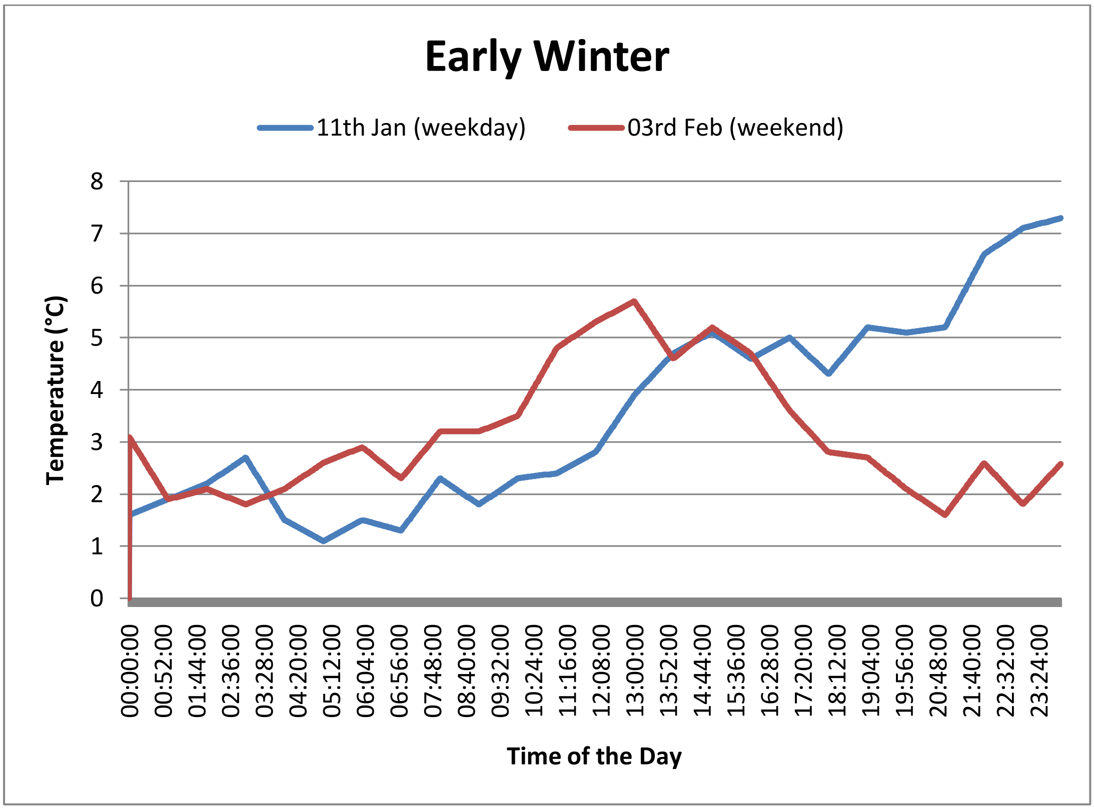
Table 3.2.1 Days chosen for heat pump simulation

Temperature requirements of hot water flowing from a heat source to house heating system to satisfy the heating demand of the house and temperature of the hot water returning to heat source after giving away the heat to the zones in the house and to the hot water stored in the DHW tank, can be taken from the ADEPT.

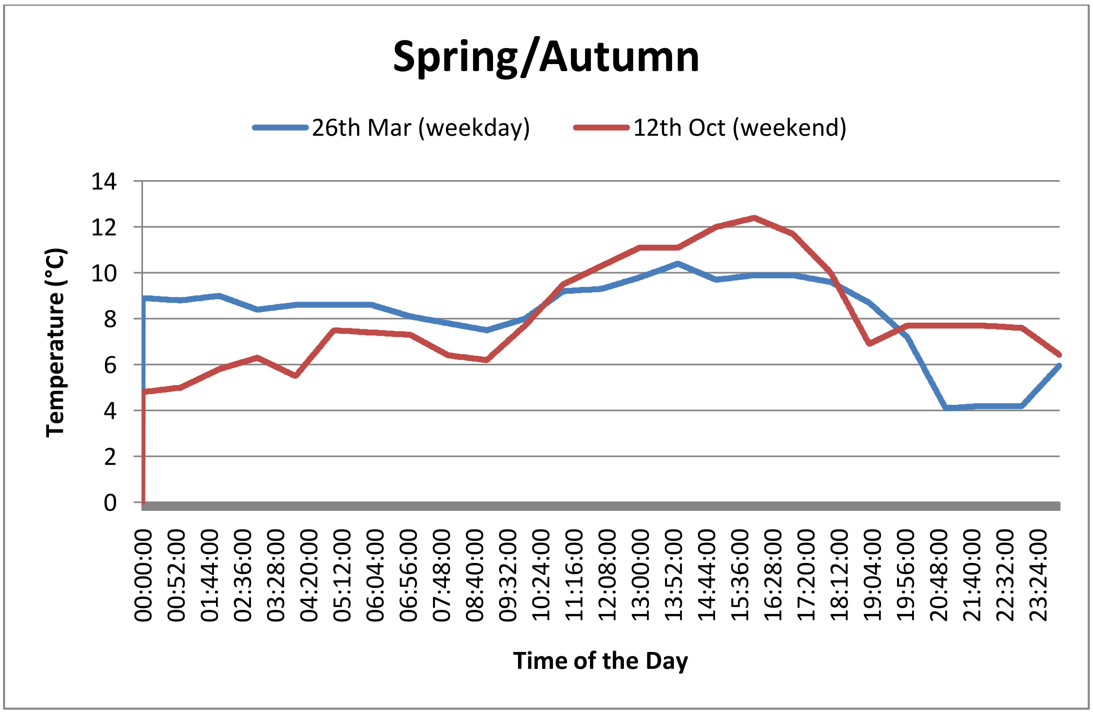
3.2.1. ATMOSPHERIC TEMPERATURES FOR SIMULATION

ADEPT calculate the temperature requirements for space heating and domestic hot water taking in account of local climatic data. Climate data used is Dundee in the 1980, which aligns very closely to the standard UK climate.

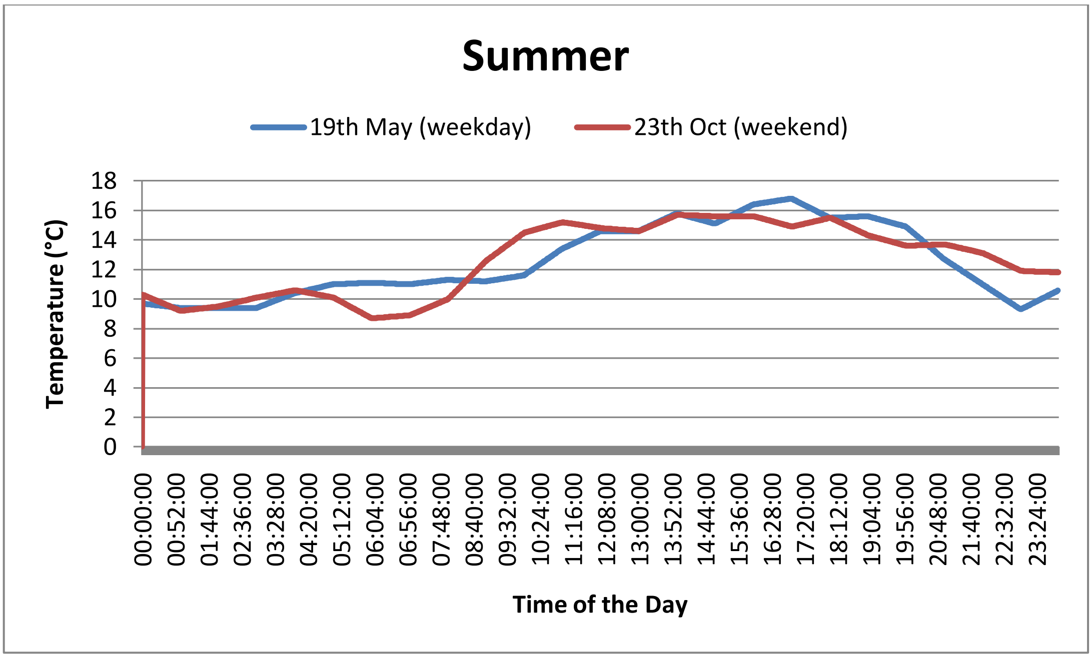
The following graphs show the atmospheric temperature of days in different seasons used for simulation of heat pump to investigate its seasonal performance factor.



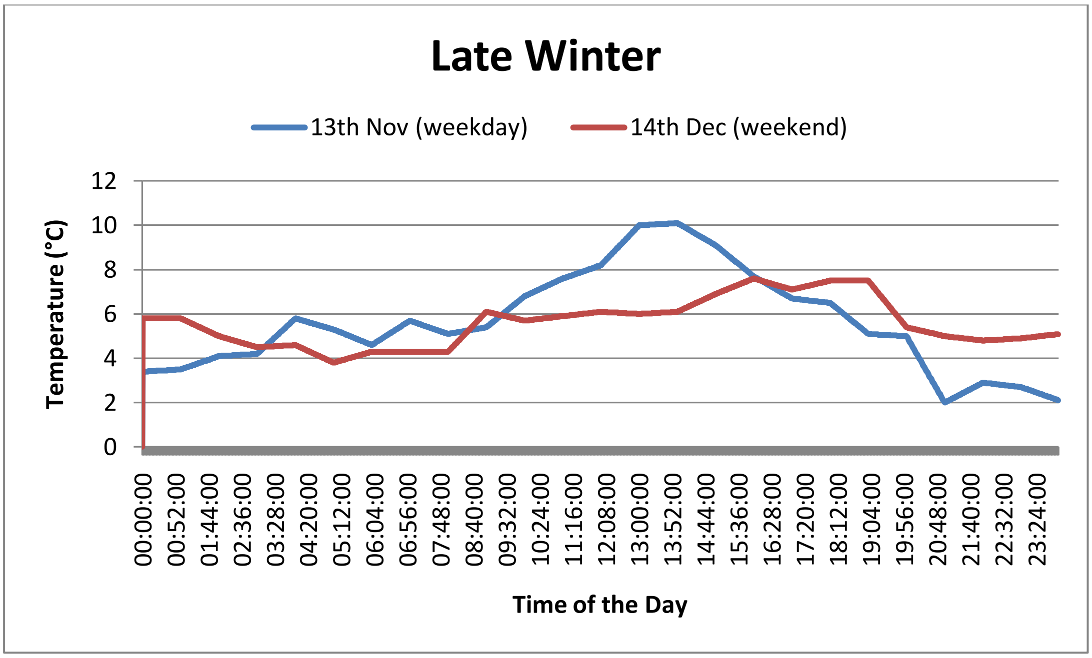
Graph 3.2.1.1 Atmospheric temperature in Early Winter



Graph 3.2.1.2 Atmospheric temperatures in Spring/Autumn season



Graph 3.2.1.3 Atmospheric temperatures in Summer season



Graph 3.2.1.4 Atmospheric temperatures in Late winter season

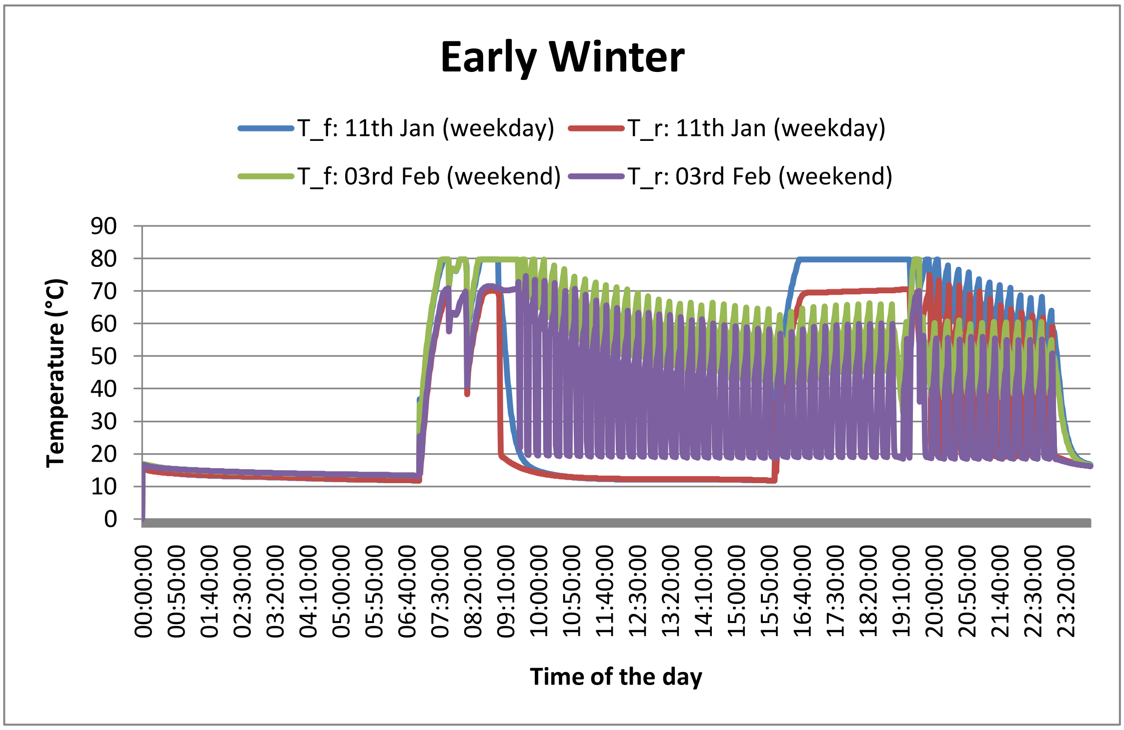
3.2.2. HOT WATER FLOW TEMPERATURES ENTERING AND LEAVING HEAT GENERATOR

To satisfy the heating demands including space and hot water in a house with a hydronic heat distribution system, the heat source has to heat up the water, which is medium of heat transfer from heat source to the sink, circulating in the house heating circuit. The temperature requirements of the water like water flow temperature to the heating system and return water temperature from the heating system need to be known to design the size of the heat generator, a heat pump in this case.

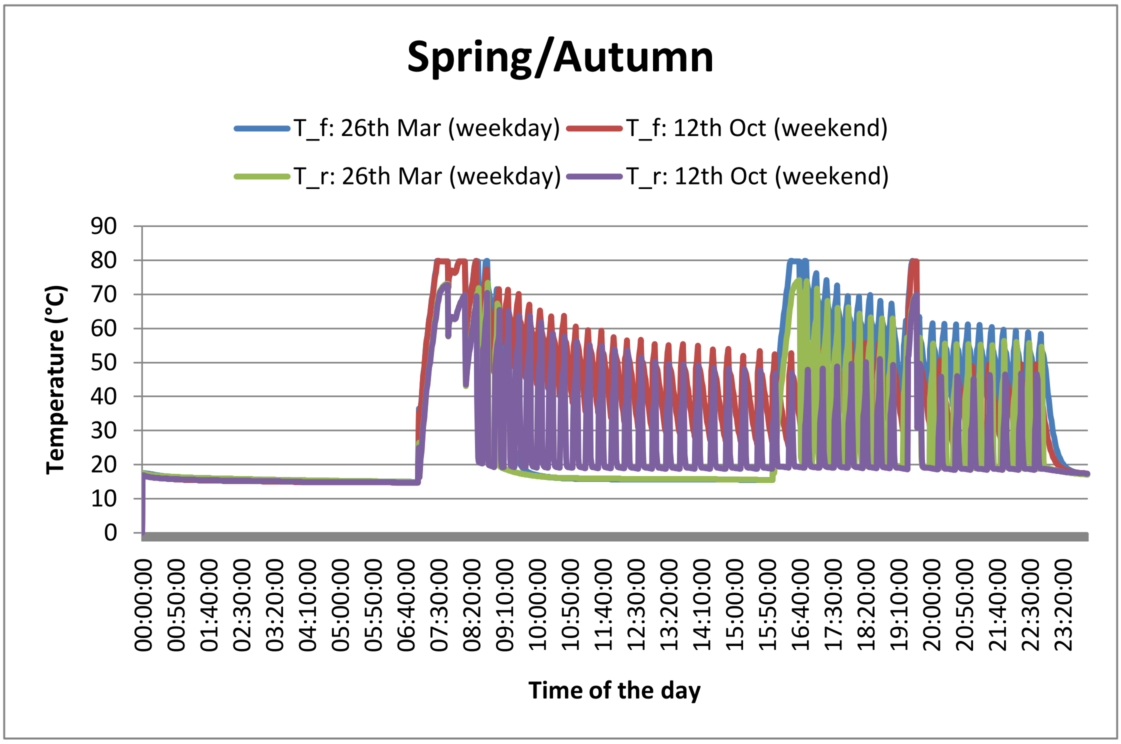
ADEPT can calculate the hot water flow and return temperature from the heating source for different houses with different heating control systems,taking into account building heat loads for external temperatures, building fabric elements and nature of heating system. Hot water flow and return temperatures at heat generator in the current house heating system, for different seasons, from ADEPT are shown in graphical form below,

T_f : FLOW TEMPERATURE

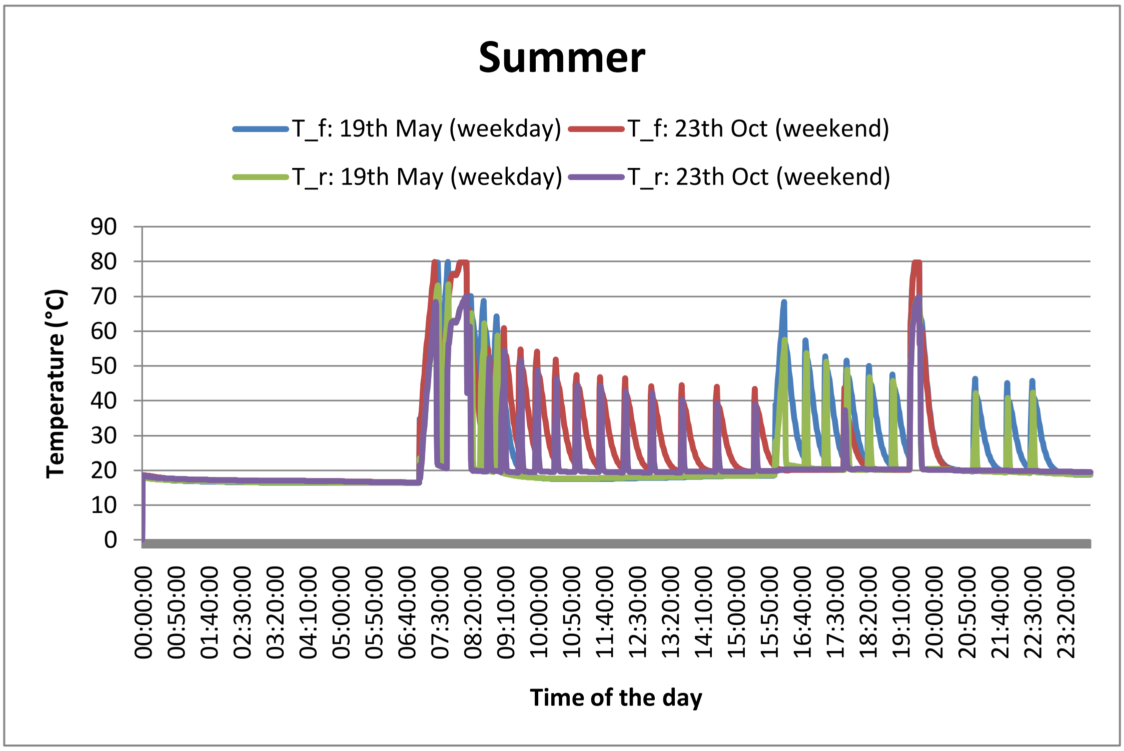
T_r : RETURN TEMPERATURE



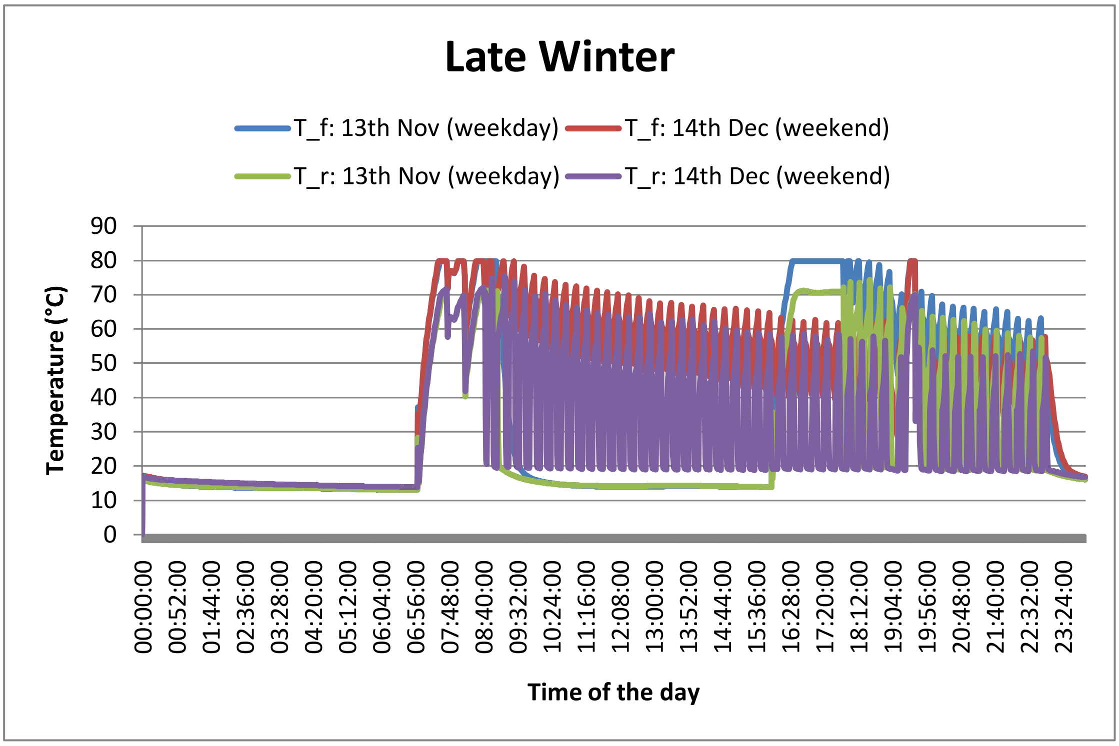
Graph 3.2.2.1 Hot water flow and return temperature in Early winter



Graph 3.2.2.2 Hot water flow and return temperature in Spring/Autumn



Graph 3.2.2.3 Hot water flow and return temperature in Summer



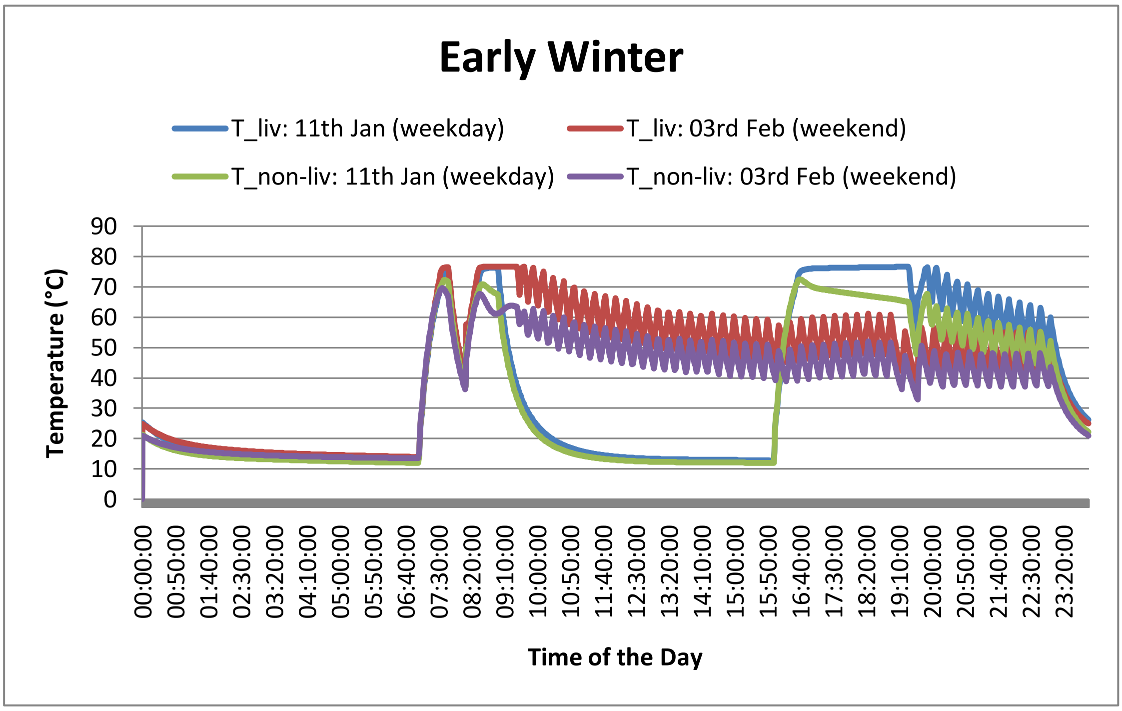
Graph 3.2.2.4 Hot water flow and return temperature in Late winter

3.2.3. SPACE HEATING RADIATORS HOT WATER TEMPERATURE REQUIREMENT

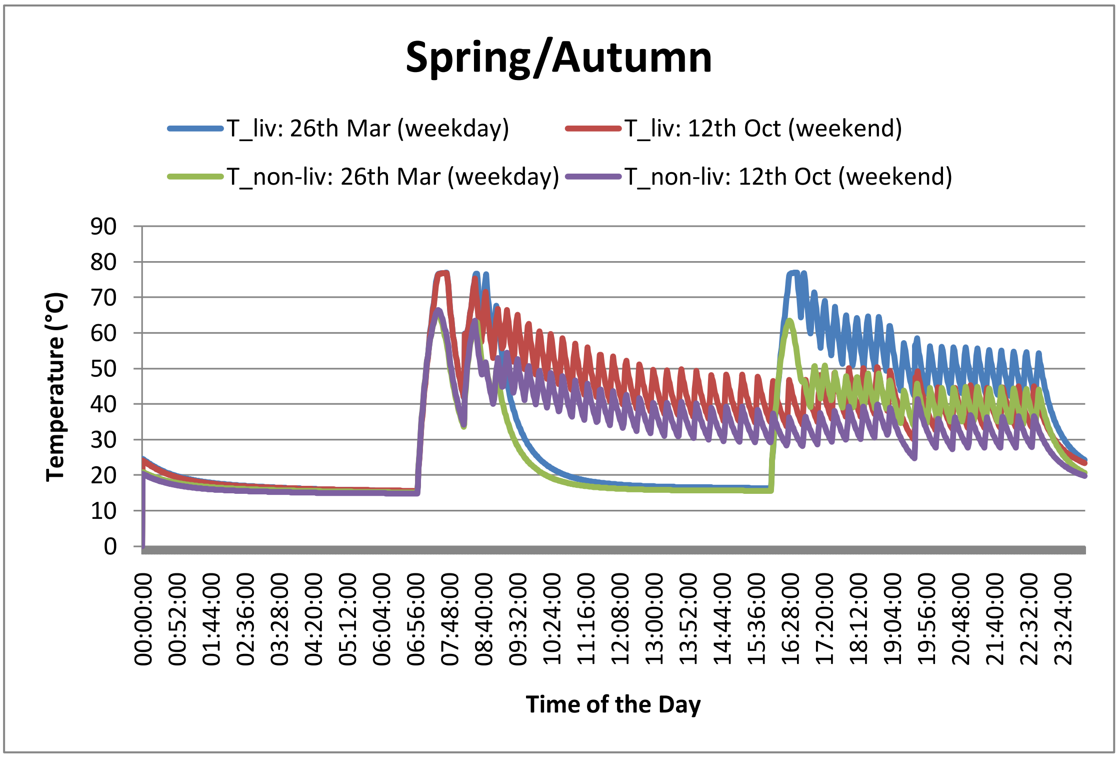
The heating system used in average UK domestic housing high temperature hydronic heating system. This system uses high temperature radiators (80/70°C) as heat distributor to the space. Size of the radiators used in this house 3.5 kW in living zone and 5.4 kW in non living zone. The input temperature to the radiator and the temperature drop across the radiator changes according to the external temperatures to maintain the room set point temperature 21°C in living zone and 19°C in non living zone according to the heating schedules. The temperature requirements for the radiators in different seasons are shown in graphical form below,

T_liv : TEMPERATURE TO THE RADIATOR IN LIVING ZONE

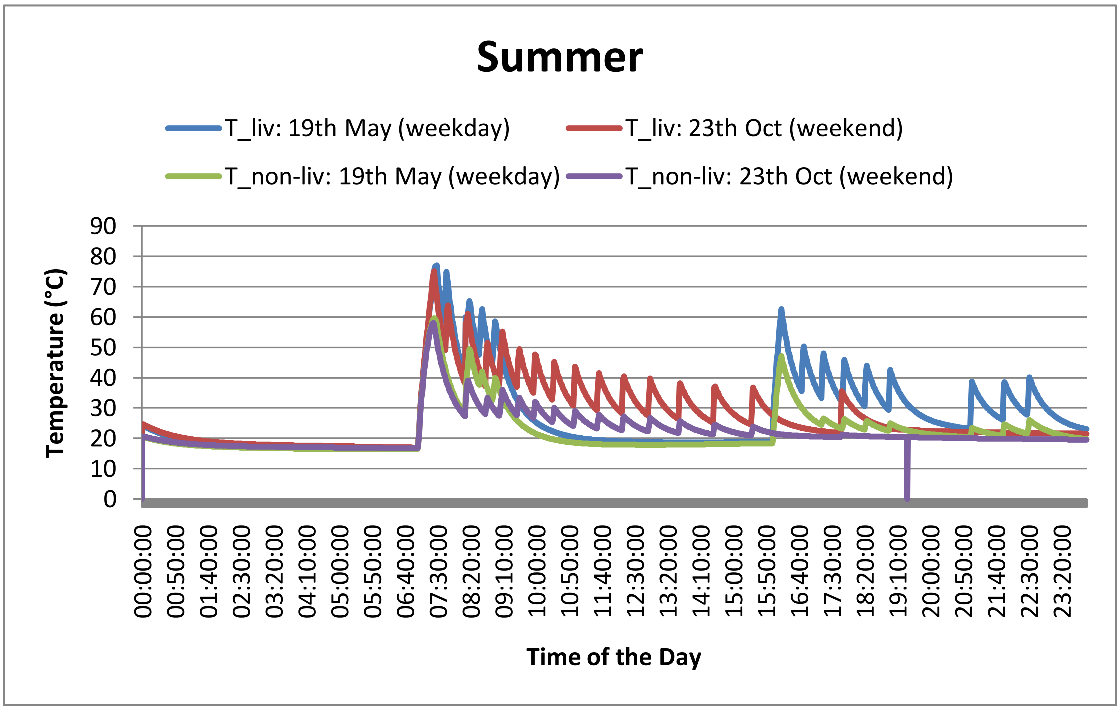
T_non-liv : TEMPERATURE TO THE RADIATOR IN NON-LIVING ZONE



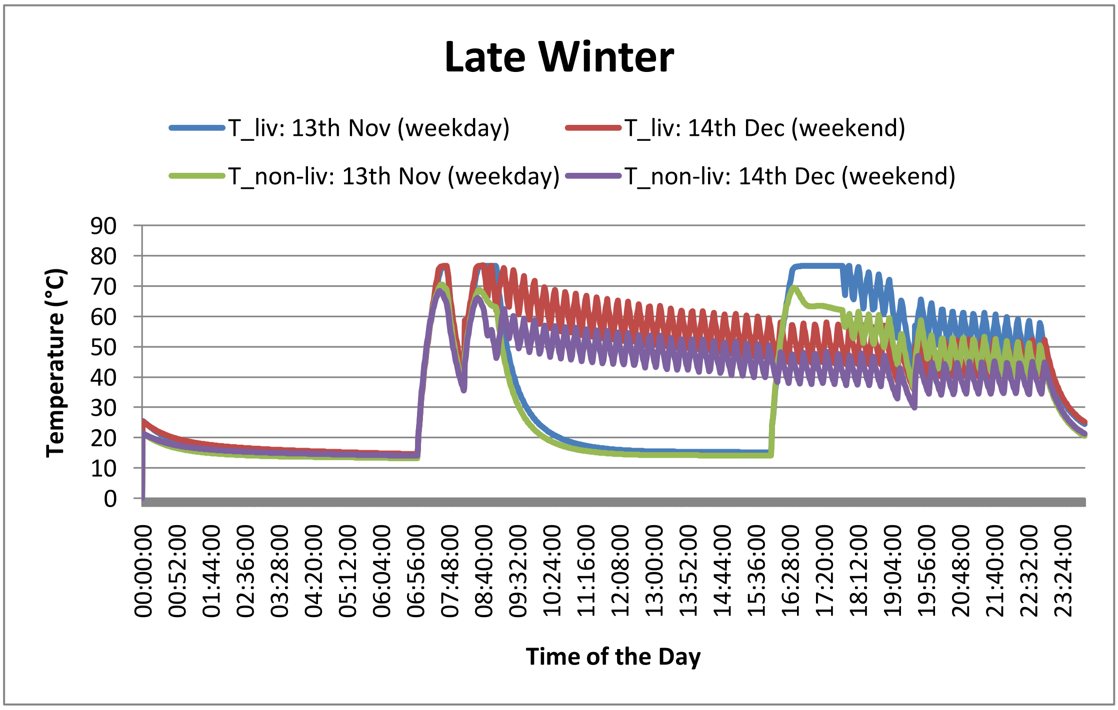
Graph 3.2.3.1 Temperature input to the radiators in living and non-living zones in Early winter



Graph 3.2.3.2 Temperature input to the radiators in living and non-living zones in Spring/Autumn



Graph 3.2.3.3 Temperature input to the radiators in living and non-living zones in Summer

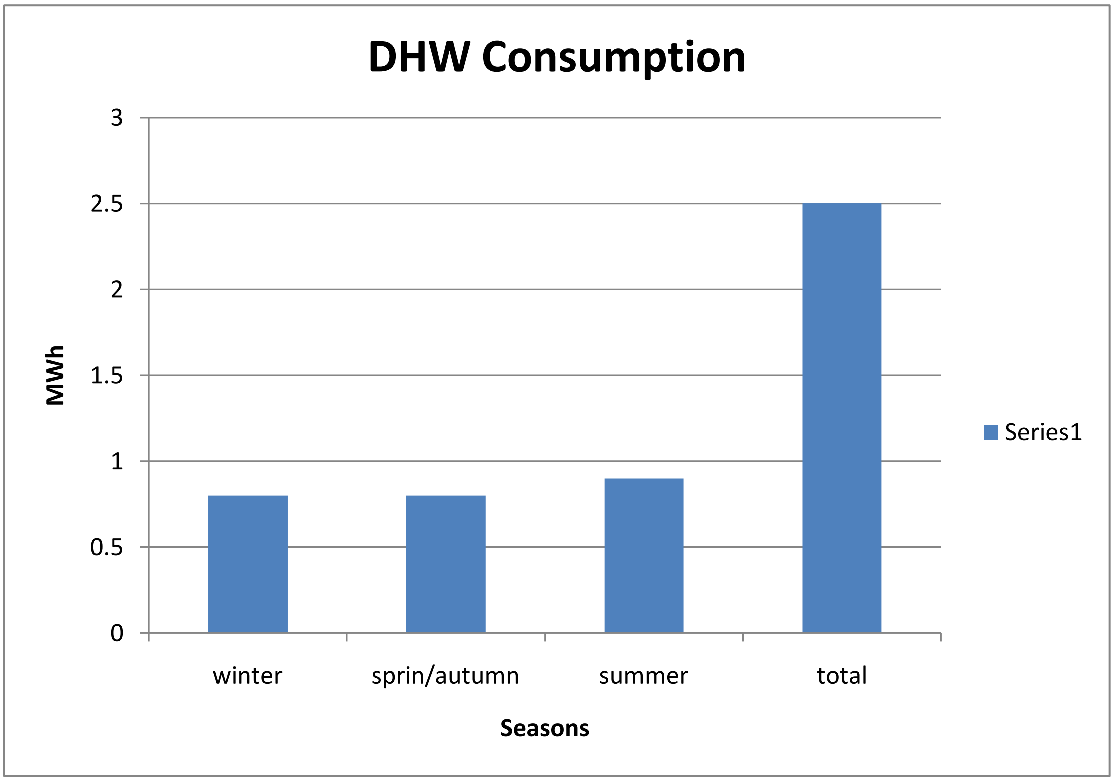


Graph 3.2.3.4 Temperature input to the radiators in living and non-living zones in Late winter

3.2.4. DOMESTIC HOT WATER CONSUMPTION IN DIFFERENT SEASONS FOR AVERAGE UK DOMESTIC HOUSING

In the UK domestic housing hot water consumption is more or less same in all seasons. Following is a graph showing domestic hot water consumption in average UK domestic housing.

in average UK domestic housing, demand for space heating is much higher than demand for domestic hot water on an average in year. So, it is important matter to consider in designing heating source for a domestic house.



Graph 3.2.4.1 Seasonal and annual domestic hot water consumption in average UK domestic house

4. NUMERICAL SIMULATION OF RESIDENTIAL CO_2 HEAT PUMP SYSTEM FOR HEATING LOADS

The main objective of this project is to investigate the performance of the CO_2 heat pump system, retrofitting in a high temperature hydronic heating system. To do this a numerical CO_2 heat pump simulator based on thermodynamic equations was developed and its performance was analysed for heating loads using EES software and EXCEL.

Factors influenced in construction of CO_2 heat pump cycle and a numerical simulator and modifications made to improve the performance of the heat pump cycle and choosing components in the heat pump system were discussed in detail in following sections,

4.1. MODELLING OF TRANSCRITICAL CO_2 HEAT PUMP CYCLE

The real transcritical CO_2 heat pump cycle, often referred to as the Lorentzen cycle, is characterized by the following changes of state:

- 1 – 2 Irreversible polytropic non-adiabatic compression
- 2 – 3 Non-isobaric supercritical heat rejection (gliding temperature)
- 3 – 4 Non-isenthalpic (non-adiabatic) expansion
- 4 – 5 Non-isobaric (non-isothermal) heat absorption

The above change of state represents a basic transcritical CO_2 heat pump cycle whose performance is lesser than conventional heat pump cycle. The following section explains the factors influencing the performance of the cycle and possible improvements or modifications to the cycle to improve its performance.

4.1.1. COEFFICIENT OF PERFORMANCE (COP) OF THE TRANS - CRITICAL CO_2 HEAT PUMP CYCLE

Coefficient of performance is measure for the performance of any heat pump. In a transcritical CO_2 heat pump cycle the coefficient of performance (COP) is calculated as the ratio of heat rejected in gas cooler to the work done by the compressor.

The major disadvantage of a CO_2 cycle is its lower COP due to the huge expansion loss compared to conventional heat pump cycles. Several researches have been carried on cycle modification of CO_2 cycle to enhance the COP of cycle and several other things to overcome the set backs due to significant expansion losses.

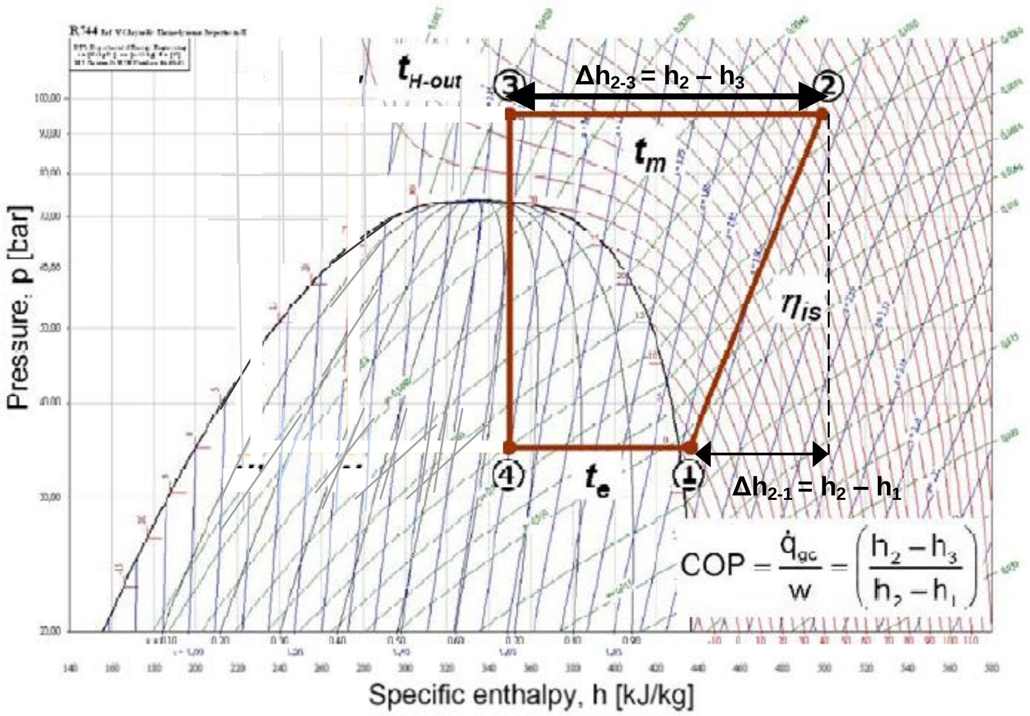


Figure 4.1.1.1 P-h diagram for COP of simple transcritical CO_2 heat pump cycle

4.1.2. MODIFICATIONS OF TRANSCRITICAL CO_2 HEAT PUMP CYCLE FOR OPTIMUM PERFORMANCE

A single –stage transcritical CO_2 heat pump cycle needs modification to improve COP of the cycle, capacity enhancement for a given system and component sizes, adaptation of the heat rejection temperature profile to given requirements and keeping the pressure ratio and discharge temperature of the compressor within the limit.

In principle a large number of possible modifications are possible, including staging of compression and expansion, splitting of flows, use of internal heat exchanger, and work generating expansion instead of throttling. Practically it is not possible to investigate all the modification to the current cycle, and many researcher include *Lorentzen* pointed an internal heat exchanger is the best modification to improve the COP of the cycle. So in this work investigation of performance of CO_2 heat pump retrofitting in domestic housing carried out with an internal heat exchanger in the cycle.

4.1.2.1. TRANSCRITICAL CO_2 HEAT PUMP CYCLE WITH INTERNAL HEAT EXCHANGER

The figure below shows transcritical CO_2 heat pump cycle with internal heat exchanger. In this configuration, high temperature gas leaving the gas-cooler is sub-cooled prior to the throttling or expansion to the evaporator pressure by the rather low temperature refrigerant vapour leaving the evaporator. Thus, the internal heat exchanger is used to lower the refrigerant temperature before entering the expansion device by the low temperature vapour coming out from the evaporator gains the heat and gets superheated before entering the compressor for compression.

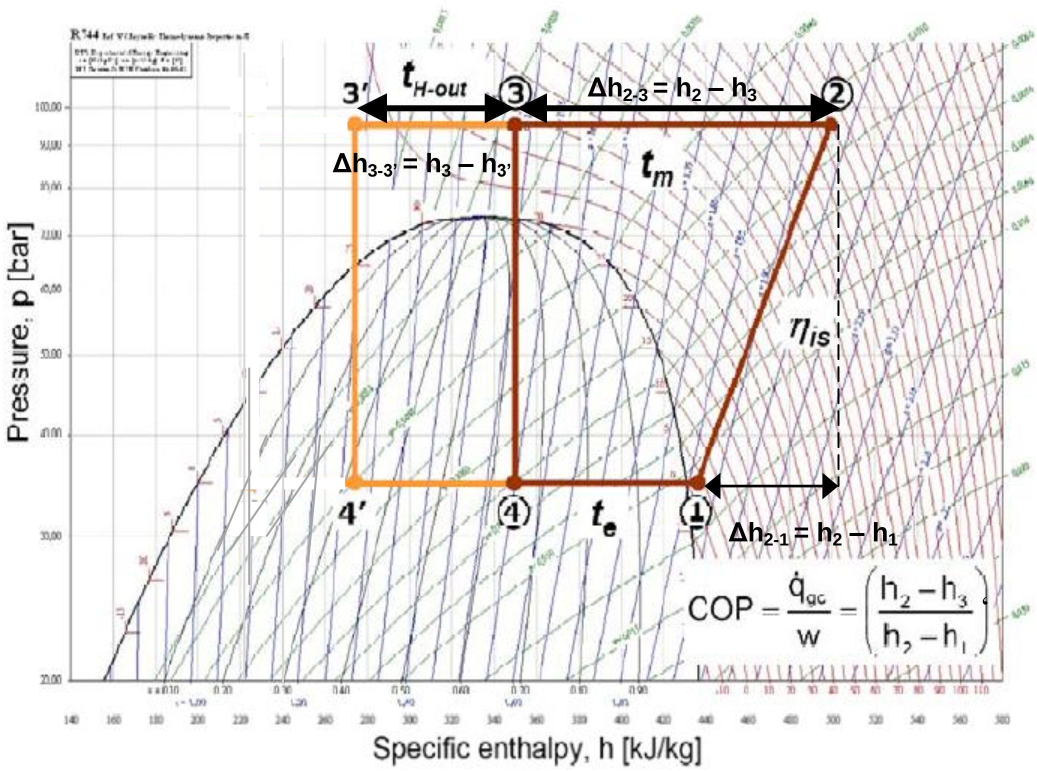


Figure 4.1.2.1.1 P-h diagram for COP of a transcritical CO_2 heat pump cycle with internal heat exchanger

Many researchers have done extensive research on the performance of transcritical CO_2 heat pump with a internal heat exchanger. Lorentzen and Pettersen (1993) represented that an improvement in COP by the internal heat exchanger is mainly due to the reduction of throttling loss resulting from cooling of the refrigerant before throttling. However, another effect of the increased average temperature in the gas cooler is less desirable. Bullock represented the cycle COP is raised by less than 5% in case that the amount of evaporator exit superheat developed by the internal heat exchanger is increased from 0 to 20°C. Boewe showed that an internal heat exchange had a significant influence on system performance by increasing its efficiency up to 25%.

4.1.3. FACTORS INFLUENCING COP OF THE CYCLE

The coefficient of performance (COP) for a single stage transcritical CO_2 heat pump is influenced by many factors in the cycle. In order to achieve a high COP for the CO_2 heat pump, it is essential that useful heat is rejected over a large temperature range resulting in a high enthalpy difference for the CO_2 in the gas-cooler and a relatively low CO_2 temperature before throttling. The CO_2 outlet temperature in gas-cooler is mainly determined by,

- The characteristics of the fluids to be heated
    1. The inlet temperatures
    2. The CP values
- The design and configuration of the gas coolers
- The high side pressure
- The compressor discharge temperature
    1. The suction pressure and temperature
    2. The high-side pressure
    3. The isentropic efficiency and heat loss for the compressor
- The CO_2 mass flow rate
    1. The suction pressure and temperature
    2. The compressor swept volume and rotational speed
    3. The volumetric efficiency of the compressor

4.1.3.1. INLET TEMPERATURE OF FLUIDS TO BE HEATED AT GAS-COOLER

To achieve a higher COP, it is essential that useful heat is rejected over a large temperature range in other words higher temperature drop across the gas-cooler. For this, return temperature of the fluid in the house heat distribution system should be as low as possible. For the houses with high temperature radiators for space heating will have less temperature drops, so the fluids return to the heat pump for heating got high temperatures. So, heat rejected by gas-cooler is low i.e. low temperature drop across the gas-cooler resulting lower COP of the cycle.

In high temperature heat distribution system, higher return temperatures can be minimised by connecting heating systems in the house like radiators, ventilation heater batteries etc in series may obtain relatively low return temperatures. Preheating and reheating of domestic hot water will lead to further decrease in return temperature.

The return temperature in the heat distribution system is determined by the heating effect and the temperature requirement for the different heating demands. The ratio of the heating effects for space heating and heating of ventilation air depends on, for example, the insulation standard and air tightness of the building, air flow rates and type of ventilation systems, period of use of the ventilation system during the day/week and the hydronic heat distribution systems.

4.1.3.2. DESIGN AND CONFIGURATION OF THE GAS-COOLERS

In a transcritical CO_2 heat pump cycle heat rejection process is of particular interest when compared with a conventional heat pump cycle. In a transcritical CO_2 heat pump cycle heat rejection occurs along an isobar that is close to the critical isobar. The heat exchanger in which the heat rejection process occurs is called a *gas-cooler,* instead of a condenser used in a conventional heat pump cycle, because the heat rejection occurs at a supercritical pressure and a phase change does not takes place but the thermo-physical properties of the refrigerant changes drastically during heat rejection process. A tube-in-tube counter flow gas-cooler is used in the retrofitting of transcritical CO_2 heat pump system in average UK domestic housing.

4.1.3.3. TRANSCRITICAL CO_2 HEAT PUMP CYCLE HIGH-SIDE PRESSURE

The heating capacity and the coefficient of performance (COP) of a transcritical CO_2 heat pump cycle are affected by the high-side pressure i.e. the pressure at which heat rejected in gas-cooler according to *Lorentzen and Pettersen*. Following figure illustrate a transcritical cycle in a temperature-enthalpy diagram taken from *Jorn Stene* work, high-pressure ranges from 8 to 11 MPa. The evaporator temperature is -5°C, the superheat is 5 K; the isentropic compressor efficiency is 60%. And the CO_2 outlet temperature from the gas-cooler is kept constant at 35°C.

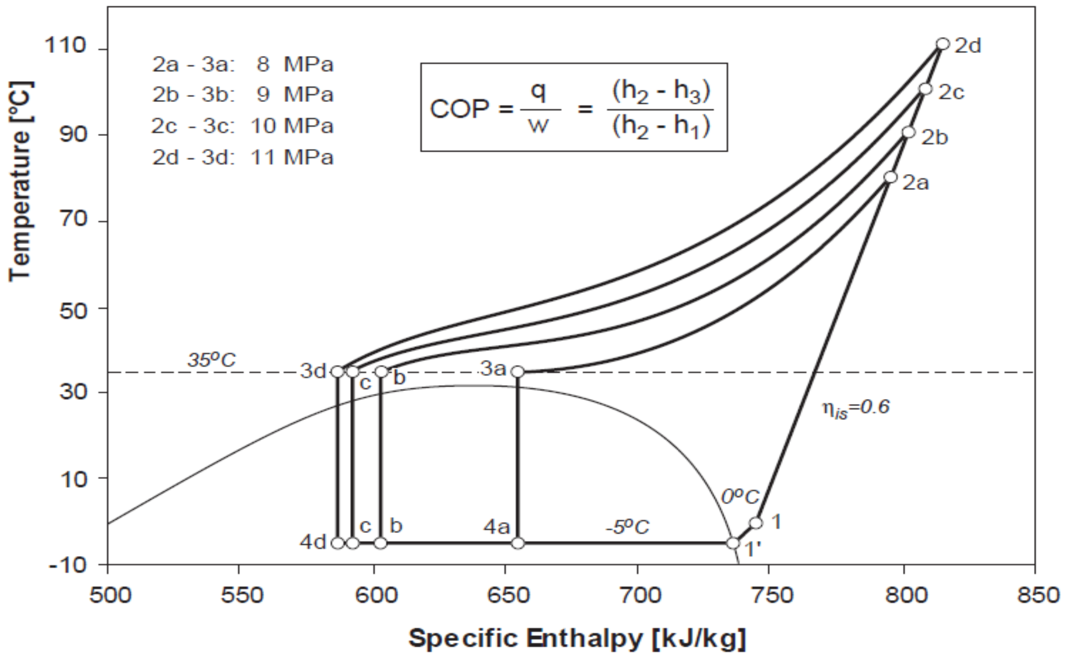


Figure 4.1.3.3.1 the transcritical CO_2 heat pump cycle operated at four different high-side pressures and gas-cooler outlet temperature assumed to be constant at 35°C (Source:CO_2 heat pump for combined SH and DHW)

The inlet enthalpy to the gas-cooler increased and the outlet enthalpy decreased when the high-side pressure is raised. Due to the great variation in the specific heat capacity at pressures and the temperature above and near the critical point, the slope $(\partial T/\partial h)_P$ is not constant and the isobars are not parallel. As a consequence, the change in the enthalpy difference in the gas cooler is not proportional to the change in the specific compressor work, and for each fixed outlet temperature from the gas cooler there will therefore be an optimum high-side pressure leading to a maximum COP.

4.2. MODELLING OF NUMERICAL CO_2 HEAT PUMP SIMULATOR

To simulate residential CO_2 heat pump system for space heating and domestic hot water, a heat pump cycle model was developed based on energy balance of individual components of the cycle.

4.2.1. NUMERICAL SIMULATION MODEL

The cycle consists of a compressor, gas-cooler, internal heat exchanger, evaporator and an expansion device. Following are general assumptions have been made in this simulation,

1. Only the steady state solution is being analyzed.
2. Heat losses from the heat exchangers and expansion device are neglected.
3. Pressure drop in Heat exchangers, connection pipelines, bends and water side are assumed negligible.
4. Changes in kinetic and potential energies are negligible.
5. *UA* values of heat exchanger were taken from manufacturer data

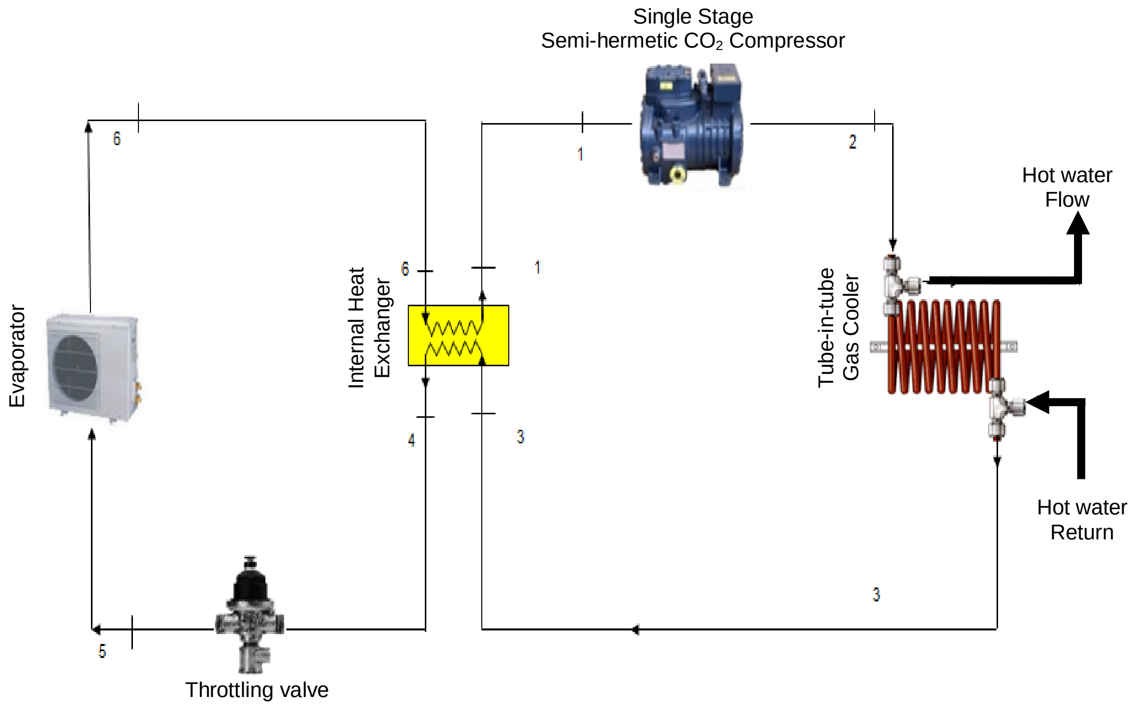


Figure 4.2.1.1 Transcritical CO_2 Heat pump simulator

4.2.1.1. COMPRESSOR MODEL

For the simulation, a single-stage, semi-hermetic compressor model manufactured by DORIN have been used.

The refrigerant mass flow rate through the compressor can be calculated by the following equation

$$m_r = \rho \, .\eta_{vol} \, . \, V_s . \, (N/60)$$

The following correlations have been used for volumetric and isentropic efficiencies respectively for the semi-hermetic compressor, which have taken from the simulation model developed by J. Sarkar.

Vol. Efficiency (η_{vol}.) = $1.1636 - 0.2188 \times (P_{ratio}) + 0.0163 \times (P_{ratio})^2$

Isen. Efficiency ($\eta_{is,c}$) = $0.61 + 0.0356 \times (P_{ratio}) - 0.0257 \times (P_{ratio})^2 + 0.0022 \times (P_{ratio})^3$

$P_{ratio} = (P_{dis} / P_{suc})$

4.2.1.2. GAS-COOLER MODEL

Gas-cooler was modelled in this simulation as tube-in-tube counter flow heat exchanger unit, where the CO_2 was flowing in the inner tube and water in the annulus or outer tube.

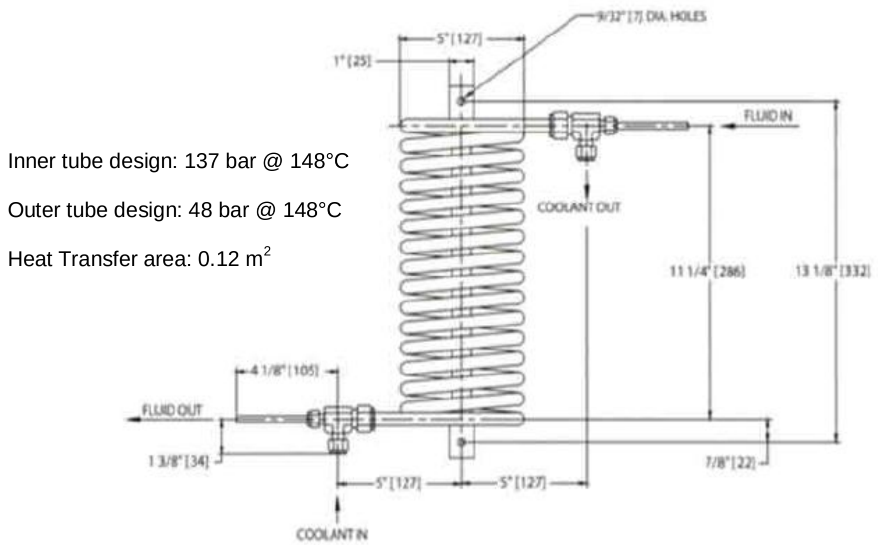


Figure 4.2.1.2.1 Sentry-equip made tube-in-tube counter flow gas-cooler (source: Sentry-equip, Model no. DTC-CUA/CUA-4-1-1)

To derive the equations for outlet temperature from gas-cooler, each gas-cooler unit was divided into fixed number of sub-sections, and the thermo physical properties of the fluids were treated as constant within each sub-section. The following picture sketches the principle of the inlet and outlet conditions for one gas cooler sub-section.

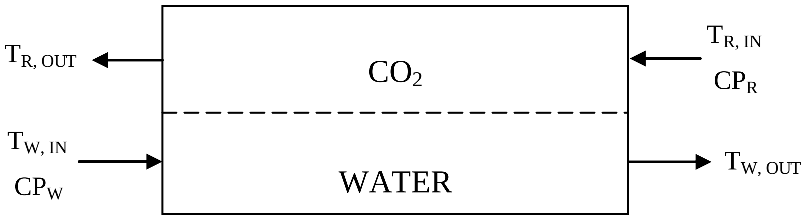


Figure 4.2.1.2.2 Principle of a gas-cooler sub-section

Since the specific heat capacity of the fluids within each gas cooler sub-section was considered as constant, the UA-LMTD method can be used for calculating the energy equations with that outlet temperature of CO_2 and water from each subsection when the inlet CO_2 temperature and the inlet water temperature were known, can be estimated as developed by Stoecker (1989).

$$T_{R,OUT} = T_{R,IN} - \left(T_{R,IN} - T_{W,IN}\right) \times \left(\frac{1 - e^{X_R}}{\frac{CP_R}{CP_W} - e^{X_R}} \right)$$

$$T_{W,OUT} = T_{W,IN} - \left(T_{W,IN} - T_{R,IN}\right) \times \left(\frac{1 - e^{X_W}}{\frac{CP_W}{CP_R} - e^{X_R}} \right)$$

Where,

$$X_R = U.A. \left[\left(\frac{1}{\dot{m}\,.C_P} \right)_R - \left(\frac{1}{\dot{m}\,.C_P} \right)_W \right]$$

$$X_W = U.A. \left[\left(\frac{1}{\dot{m}\,.C_P} \right)_W - \left(\frac{1}{\dot{m}\,.C_P} \right)_R \right]$$

4.2.1.3. INTERNAL HEAT EXCHANGER MODEL

The internal heat exchanger is modelled in same way as the gas-cooler modelled, counter flow heat exchanger. Discretization and energy balance in the internal heat exchanger have been carried out the same way as that in the gas-cooler and correlation have been developed for the exit temperatures.

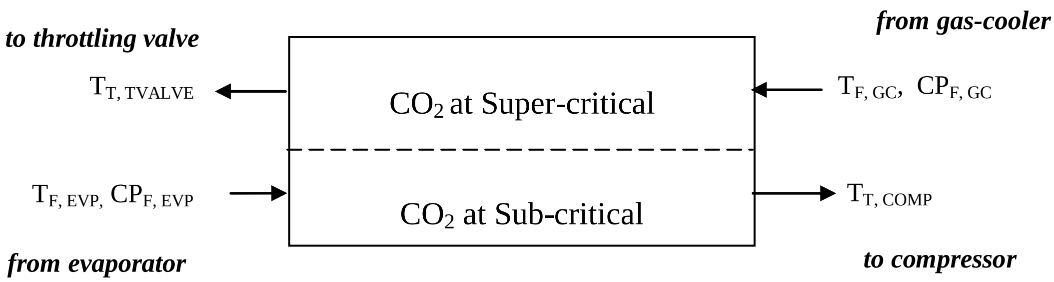


Figure 4.2.1.3.1 Principle of internal heat exchanger sub-section

Outlet temperatures for internal heat exchanger,

$$T_{T,TVALVE} = T_{F,GC} - \left(T_{F,GC} - T_{F,EVP}\right) \times \left(\frac{1 - e^{X_{SUPER}}}{\frac{CP_{SUPER}}{CP_{SUB}} - e^{X_{SUPER}}} \right)$$

$$T_{T,COMP} = T_{F,EVP} - \left(T_{F,EVP} - T_{F,GC}\right) \times \left(\frac{1 - e^{X_{SUB}}}{\frac{CP_{SUB}}{CP_{SUPER}} - e^{X_{SUB}}} \right)$$

Where,

$$X_{SUPER} = U.A.\left[\left(\frac{1}{\dot{m}.C_P}\right)_{SUPER} - \left(\frac{1}{\dot{m}.C_P}\right)_{SUB}\right]$$

$$X_{SUB} = U.A.\left[\left(\frac{1}{\dot{m}.C_P}\right)_{SUB} - \left(\frac{1}{\dot{m}.C_P}\right)_{SUPER}\right]$$

4.2.1.4. EXPANSION DEVICE MODEL

The expansion process is considered to be isenthalpic under the assumption that the heat exchange with its surroundings is negligible, yielding:

$$h_4 = h_5$$

4.2.1.5. EVAPORATOR MODEL

Evaporator modelled as perfect heat exchanger, exchanges refrigerant heat with heat in atmospheric air. So temperature of refrigerant leaving evaporator has exact temperature of the out-side air.

4.2.2. NUMERICAL SIMULATION PROCEDURE

A computer code in Engineering Equation Solver (EES) has been developed to simulate the transcritical carbon-dioxide heat pump system, retrofitting in UK domestic house with high temperature heat distribution system. EES provides many built-in mathematical and thermo-physical property functions useful for engineering calculations. Steam tables are implemented such that any thermodynamic and transport properties for most organic refrigerants, air tables are built-in, as are psychrometric functions and JANAF table data for many common gases.

4.2.2.1. INPUT DATA AND ASSUMPTIONS IN SIMULATION PROGRAM

Water inlet temperatures and mass flow rate for gas-cooler, compressor data, gas-cooler dimensions, compressor suction pressure and discharge pressure are the input data for the simulation. Pressure drop in heat exchangers, pressure drop and heat loss in connecting lines are neglected, therefore the outlet state of one component becomes the inlet state of the next component, and evaporator temperature is equal to atmospheric temperature.

The flow chart shown below illustrates the simulation process in step wise manner. In the simulation, by giving the constant input parameters values like UA for heat exchangers taken from manufacturer data, mass flow rate of refrigerant and hot water, optimum discharge pressure and variable input parameters which changes by time in the day and seasons like atmospheric temperatures and hot water return temperature. Refrigeration conditions at evaporator inlet and at gas cooler outlet as well as internal heat exchanger were calculated based on mathematical model of gas-cooler and internal heat exchanger, finally COP can be calculated.

The variable parameters input values changes with time in the day and seasons. COP of the system can be calculated for any time step and change of COP can be noted for different time steps in the day by changing variable input values which can be found in ADEPT. Seasonal performance factor (*SPF*) of the system can be found as above process.

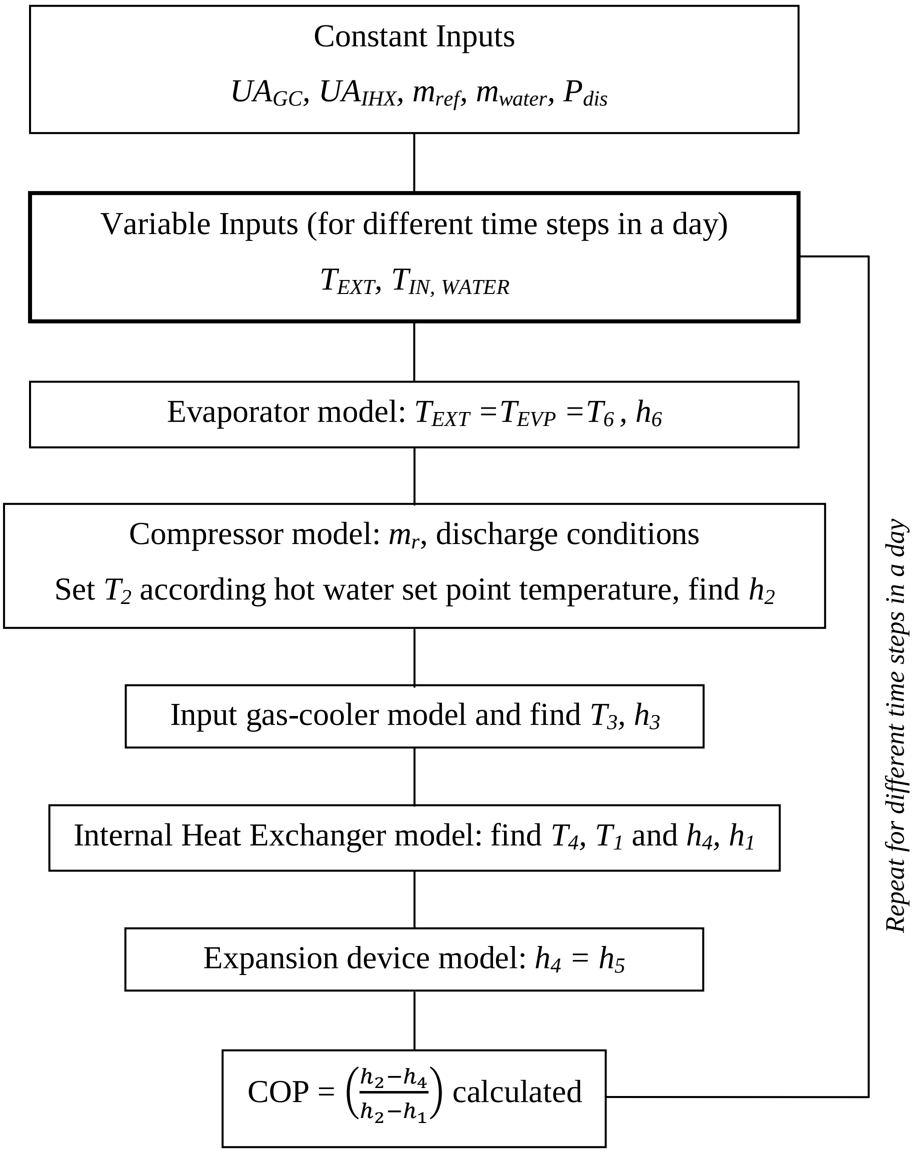


Figure 4.2.2.1.1 Flow chart for the numerical simulation

4.2.2.2. COMPUTER CODE FOR SIMULATION IN EES

$UAgc = 4$

$UAix = 0.3$

$\dot{m}_{CO2} = 0.5$ mass flow rate of refrigerant

EVAPORATOR MODEL

$T_{atm} = 2.5$ Atmospheric temperature

$T_6 = T_{atm}$ Refrigerant temperature at evaporator

$X_6 = 1$ Quality of refrigerant a saturated vapour exit

$P_6 = \mathbf{P}\ (\text{'R744'}, T = T_6, x = X_6)$ Saturated pressure of ref. at evaporator exit

$h_6 = \mathbf{h}\ (\text{'R744'}, T = T_6, x = X_6)$ enthalpy of the refrigerant at evaporator exit

$s_6 = \mathbf{s}\ (\text{'R744'}, T = T_6, x = X_6)$

$Cp_{co2,6} = 2$

$Cp_{co2,3} = \mathbf{Cp}\ (\text{'R744'}, T = T_3, P = P_3)$

INTERNAL HEAT EXCHANGER

Sub-critical side

$P_1 = P_6$ Neglecting pressure drop across the heat exchanger

$$T_1 = T_6 - \left[(T_6 - T_3) \cdot \left(\frac{1 - \mathbf{exp}\left[\frac{UAix}{\dot{m}_{CO2}} \cdot \left(\frac{1}{Cp_{co2,6}} - \frac{1}{Cp_{co2,3}} \right) \right]}{\frac{Cp_{co2,6}}{Cp_{co2,3}} - \mathbf{exp}\left[\frac{UAix}{\dot{m}_{CO2}} \cdot \left(\frac{1}{Cp_{co2,6}} - \frac{1}{Cp_{co2,3}} \right) \right]} \right) \right]$$

$h_1 = \mathbf{h}\ (\text{'R744'}, T = T_1, P = P_1)$

$s_1 = \mathbf{s}\ (\text{'R744'}, T = T_1, P = P_1)$

COMPRESSOR MODEL

$$P_r = \frac{P_2}{P_1}$$

$$Eff_{Isen} = 0.63 + 0.0356 \cdot P_r - 0.0257 \cdot P_r^2 + 0.0022 \cdot P_r^3$$ Isentropic efficiency of the compressor

$$P_2 = 9.5$$

$$Ts_2 = T_{w,out} - 10$$

$$Ss_2 = s_1$$

$$hs_2 = \mathbf{h}\,('R744', T=Ts_2, P=P_2)$$

$$h_2 = h_1 + \frac{hs_2 - h_1}{Eff_{Isen}}$$

$$T_2 = \mathbf{T}\,('R744', P=P_2, h=h_2)$$

$$s_2 = \mathbf{s}\,('R744', T=T_2, P=P_2)$$

$$Cp_{co2,2} = \mathbf{Cp}\,('R744', T=T_2, P=P_2)$$

GAS-COOLER MODEL

$$T_{w,in} = 30$$ Temperature of the hot water return

$$T_{w,out} = 80$$ Temperature of the hot water flow

$$\delta T_{water} = T_{w,out} - T_{w,in}$$

$$Cp_{water} = 4.2$$

$$\dot{m}_{water} = 0.213$$

$$T_3 = T_2 - \left[(T_2 - T_{w,in}) \cdot \left(\frac{1 - \mathbf{exp}\left[UAgc \cdot \left(\frac{1}{\dot{m}_{CO2} \cdot Cp_{co2,2}} - \frac{1}{\dot{m}_{water} \cdot Cp_{water}} \right) \right]}{\frac{\dot{m}_{CO2} \cdot Cp_{co2,2}}{\dot{m}_{water} \cdot Cp_{water}} - \mathbf{exp}\left[UAgc \cdot \left(\frac{1}{\dot{m}_{CO2} \cdot Cp_{co2,2}} - \frac{1}{\dot{m}_{water} \cdot Cp_{water}} \right) \right]} \right) \right]$$

$$h_3 = \mathbf{h}('R744', T=T_3, P=P_3)$$

$$P_4 = P_2$$

$$P_5 = P_6$$

$$\delta T_{gc} = T_2 - T_3$$

INTERNAL HEAT EXCHANGER

Supercritical side

$$T_4 = T_3 - \left[(T_3 - T_6) \cdot \left(\frac{1 - \exp\left[\frac{UAix}{\dot{m}_{CO2}} \cdot \left(\frac{1}{Cp_{co2,3}} - \frac{1}{Cp_{co2,6}}\right)\right]}{\frac{Cp_{co2,3}}{Cp_{co2,6}} - \exp\left[\frac{UAix}{\dot{m}_{CO2}} \cdot \left(\frac{1}{Cp_{co2,3}} - \frac{1}{Cp_{co2,6}}\right)\right]} \right) \right]$$

$$h_4 = \mathbf{h}('R744', T=T_4, P=P_4)$$

$$s_4 = \mathbf{s}('R744', T=T_4, P=P_4)$$

EXPANSION DEVICE MODEL

$$h_5 = h_4$$

$$T_5 = T_6$$

$$\delta T_{hx} = T_3 - T_4$$

$$Q_{2,3} = \dot{m}_{CO2} \cdot Cp_{co2,2} \cdot (T_2 - T_3)$$

$$Q_{3,4} = \dot{m}_{CO2} \cdot Cp_{co2,3} \cdot (T_3 - T_4)$$

$$Q_{6,1} = \dot{m}_{CO2} \cdot Cp_{co2,6} \cdot (-T_6 + T_1)$$

PERFORMANCE OF THE HEAT PUMP

$$COP = \left| \frac{h_2 - h_4}{h_2 - h_1} \right|$$

4.2.3. NUMERICAL SIMULATION RESULTS AND ANALYSIS

Following are the results for the simulation carried with assumptions, cycle model, and description of components in the cycle discussed in the previous section, using Engineering Equation Solver (EES).

As we discussed before the coefficient of performance (COP) for a single stage transcritical CO_2 heat pump is influenced by many parameters of the cycle. To achieve a high COP for the CO_2 heat pump cycle, it is essential that useful heat is rejected over a large temperature range resulting in a high enthalpy difference for the CO_2 in the gas-cooler and a relatively low CO_2 temperature before throttling. The CO_2 outlet temperature in gas-cooler is mainly determined by,

- The temperatures of the fluids to be heated
- The high side pressure
- The temperature characteristics of gas-cooler
- The Evaporator temperature

4.2.3.1. ANALYSIS OF CYCLE OPTIMUM HIGH SIDE PRESSURE

Simulation was carried to find optimum high side pressure in the cycle. Cycle coefficient of performance was calculated for different discharge pressures of compressor at constant evaporator temperature at 0°C, water side temperatures in the gas-cooler as 80/30 °C i.e., water heated in the gas-cooler to 80 °C set point temperature and water entering at 30 °C, the isentropic efficiency of the compressor is 70% and the input temperature to the gas-cooler or output temperature from the compressor as constant.

For this analysis the high-side pressure ranging from 8 to 12 MPa was taken, it was found that COP was tend to rise from 8 to 8.5 MPa of high-side pressure, then the COP tends to drop. From this, it was found that the maximum COP of the cycle is achieved at a high-side pressure of 8.5 MPa which is taken as the optimum high-side pressure of the cycle.

Throughout the residential CO_2 heat pump cycle simulation, high-side pressure in the cycle kept constant at 8.5 MPa for calculations of all other parameters influencing performance of the cycle

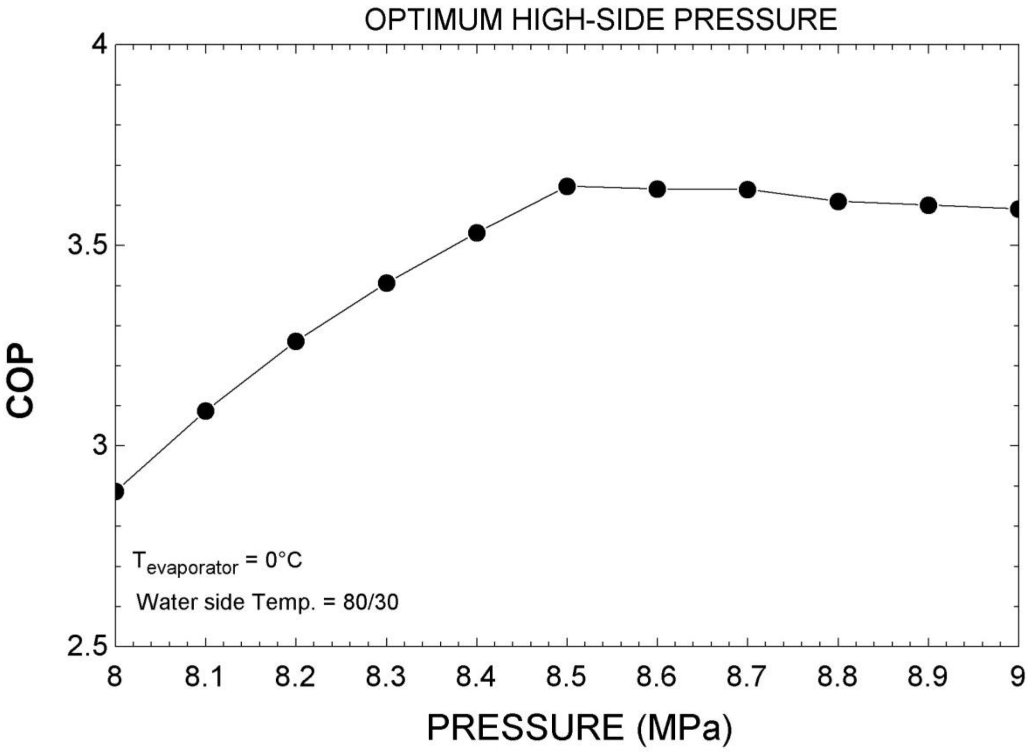


Figure 4.2.3.1.1 Graph for COP vs. PRESSURE to find Optimum high-side pressure of the transcritical CO_2 heat pump cycle, developed using EES

4.2.3.2. ANALYSIS OF GAS-COOLER TEMPERATURE PROFILE

Gas-cooler plays a prominent role in a CO_2 heat pump cycle. The cycle performance mainly depends on the gas-cooler temperature profile both on refrigerant side and water side. The amount of heat rejected by the refrigerant is gained by the heat carrying fluid to the sink, water in this case. In turn the amount of heat to be dissipate by the refrigerant depends on the heat requirement of the water, higher the heat requirement of water, higher the temperature drop in the refrigerant in other words higher the enthalpy difference, higher COP of the cycle.

4.2.3.2.1. WATER SIDE OF THE GAS-COOLER

Heat requirement of the water in the gas-cooler set the amount of heat to be rejected in the gas-cooler by refrigerant. Higher the heat requirement higher the heat rejected by refrigerant, larger temperature glide by gas-cooler. Following is a graph produced in EES, for analysis of hot water return temperature influence on cycle COP, at some fixed parameters. For this analysis, the evaporator temperature kept constant at 0°C, high-side pressure of 8.5 MPa, hot water exit temperature at gas-cooler 80°C constant.

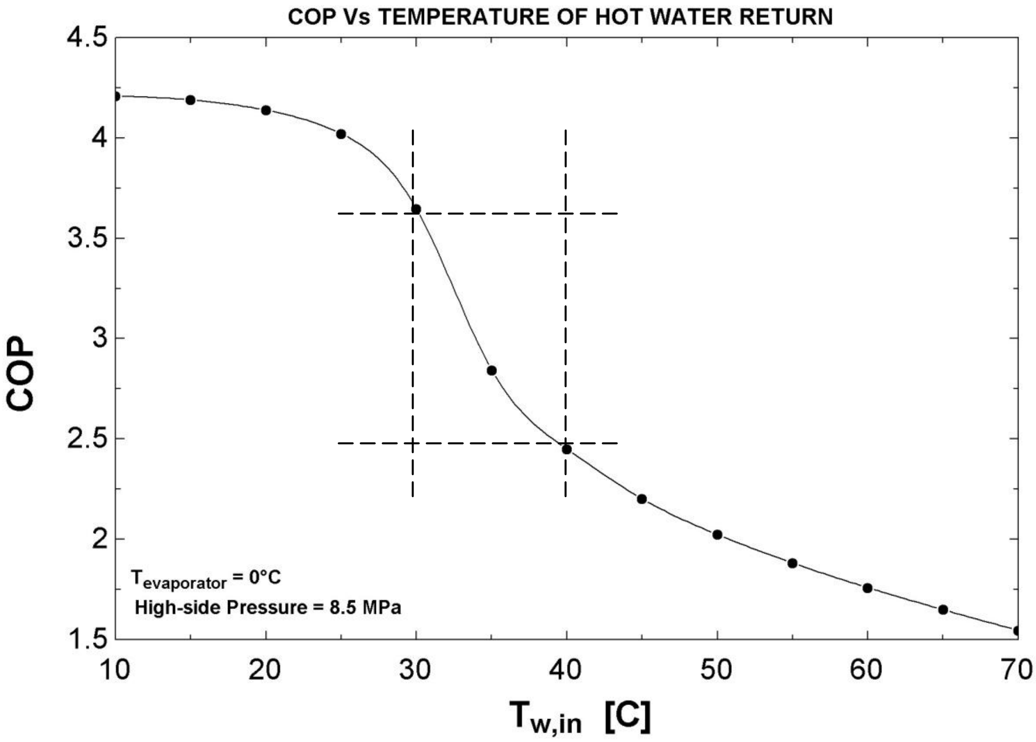


Figure 4.2.3.2.1 Graph for COP Vs Hot water return temperature, developed using EES

At a certain hot water return temperature range, the influence on cycle COP is much higher. By observing the graph we can notice, a range of water returns temperature to gas-cooler from heating distribution system from 40 – 30°C the COP of cycle rise from around 2.5 to 3.5 i.e., for 10°C drop of hot water return temperature there is a rise of COP by 1, can clearly seen in the above figure. So to get *a decent COP of the system it is important to keep the water return temperature below 50°C*.

4.2.3.2.2. REFRIGERANT SIDE OF GAS-COOLER

As discussed in previous sections, the amount of heat dissipated in gas-cooler by refrigerant depends on the heat requirement of hot water. Following are the graph for different temperature requirements of the hot water. These graphs show the temperature glide in gas-cooler also included temperature drop in internal heat exchanger for different temperature requirements of hot water in gas-cooler.

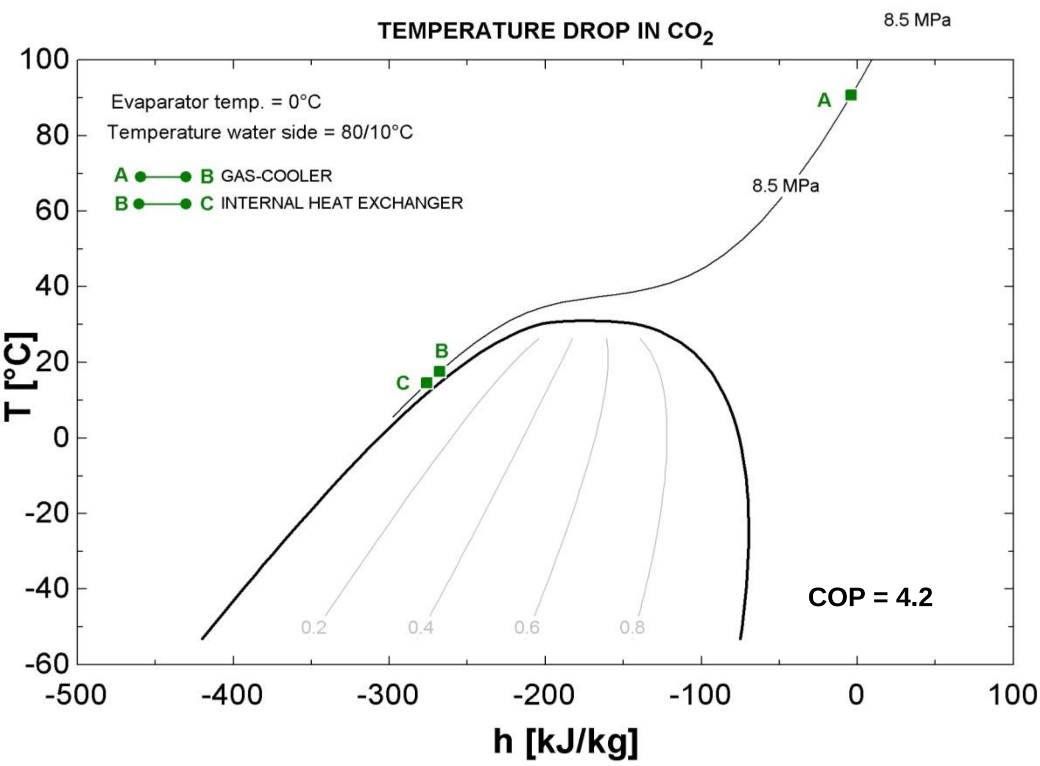


Figure 4.2.3.2.2 Temperature glide in gas-cooler, for water side temperature of 80/10°C at a high-side pressure of 8.5 MPa. Developed using EES

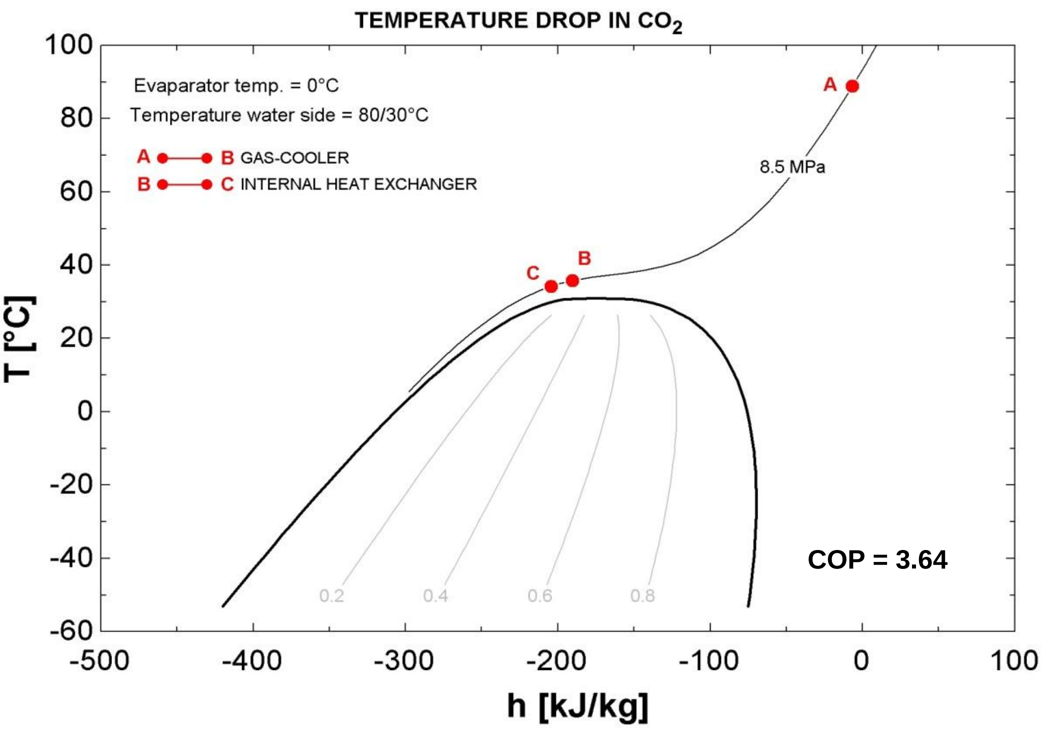


Figure 4.2.3.2.3 Temperature glide in gas-cooler, for water side temperature of 80/30°C at a high-side pressure of 8.5 MPa. Developed using EES

4.2.3.2.3. GAS-COOLER APPROACH TEMPERATURE

Approach temperature of a gas-cooler ΔT_A is the difference between the CO_2 outlet temperature and the inlet temperature of the return hot water from sink. A small temperature approach leads to a large heating capacity to the compressor. In this simulation the gas-cooler approach temperature increases with decrease in hot water return temperature.

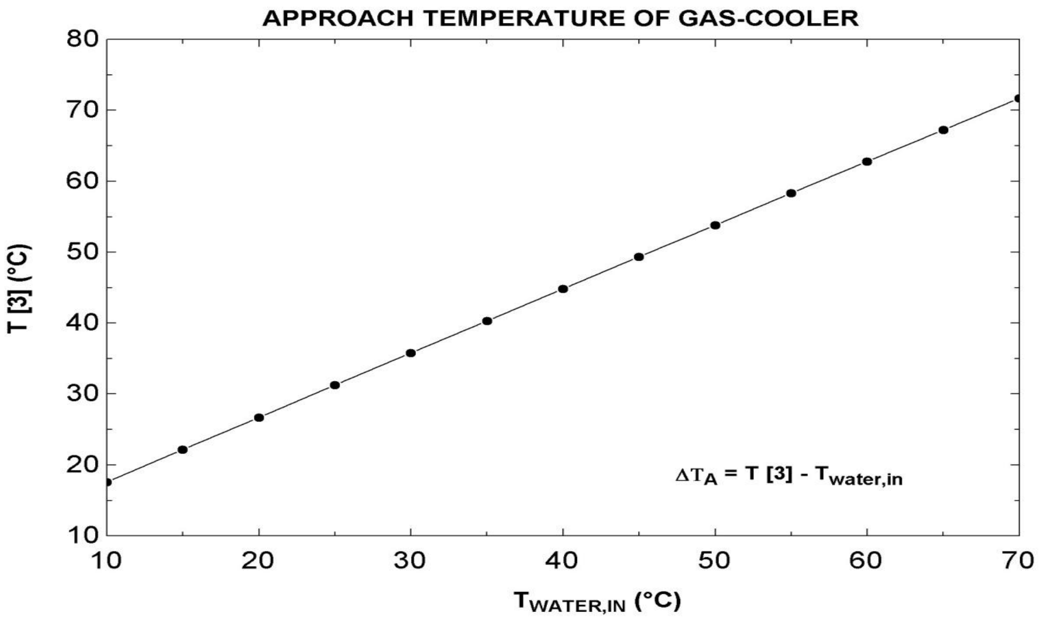


Figure 4.2.3.2.4 Graph showing the effect of hot water return temp. on refrigerant exit temperature in gas-cooler, their difference is denoted as ΔT_A. developed using EES

At a 10°C hot water return temperature from the sink, the gas-cooler approach temperature ΔT_A is 5 K. with increase in return water temperature the approach temperature of gas-cooler decreases. At a 70°C of return temperature the gas-cooler temperature approach ΔT_A is around 0.2 K.

4.2.3.3. ANALYSIS OF EVAPORATOR TEMPERATURE

In this simulation the evaporator temperature is taken as the temperature of the atmospheric air, because it is an air source heat pump. So the atmospheric or surrounding air temperature of the heat pump is the source temperature, hence evaporator temperature had a great influence on heat pump. Following is a graph drawn COP of the cycle and hot water return temperature to the gas-cooler for three different atmospheric temperature i.e. evaporator temperature at constant high-side pressure of 8.5 MPa, set-point 80°C temperature of hot water flow to the sink, constant isentropic efficiency of the compressor 70%.

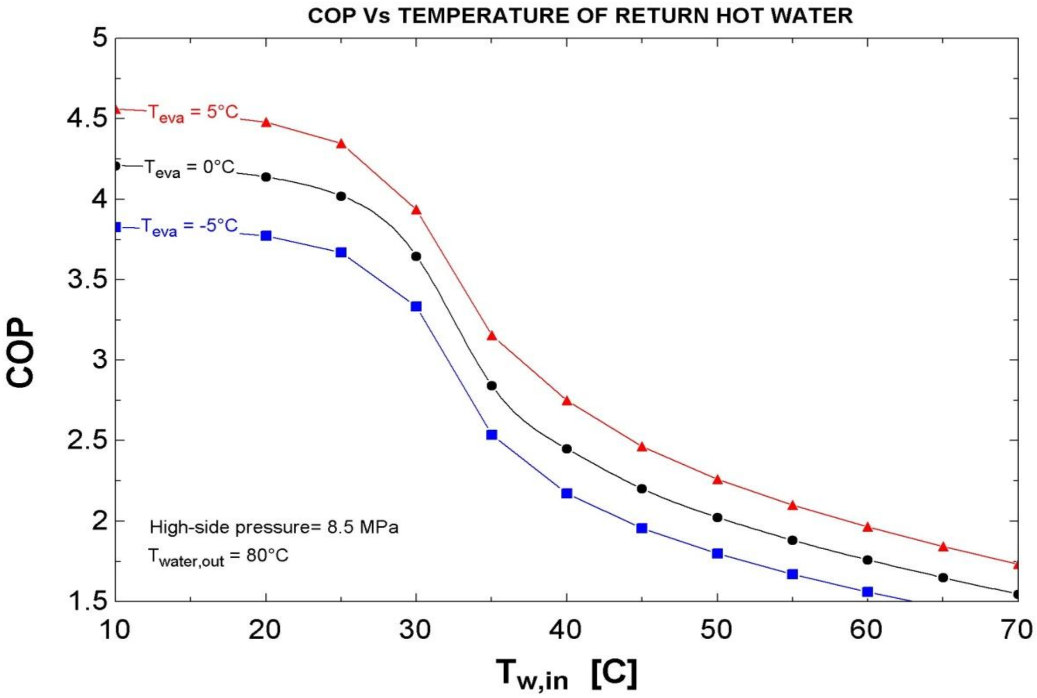


Figure 4.2.3.2.5 Graph for COP at different hot water return temperatures for different evaporator temperatures. Developed using EES.

For this analysis the COP of the system for different hot water return temperatures tested at three different evaporator temperatures -5°C, 0°C and +5°C. from this analysis it is found that the COP of the system changes when evaporator temperature changes, at an instant when the return temperature of the hot water is 30°C the COP of the system increases by 0.3 when evaporator temperature changes from -5 to 0°C, similarly there is an increase in COP by 0.3 when evaporator temperature increases from 0 to +5°C. This trend was continuing more or less same at different hot water return temperatures.

The equation relating COP and return water temperature $T_{w,in}$ at fixed high-side pressure, temperature of water leaving gas-cooler $T_{w,out}$ heated to set-point temperature generally fixed,

At outside air T_{eva} = 0 ° C

$$\boxed{COP = 3.516 - 0.065 \times T_{w,\,in} + 0.0223 \times T_{w,\,in}^{2} - 0.0015 \times T_{w,\,in}^{3} + 0.037\,E\text{-}3 \times T_{w,\,in}^{4}}$$

4.2.3.4. VARIABLE PARAMETERS OF CO_2 HEAT PUMP SIMULATOR

Following is the P-h diagram for residential CO_2 heat pump system for pre-defined parameters some are variable and some are fixed. Variable parameters generally can be changed to test this heat pump system at different heating loads and the fixed parameters cannot be changed even testing for different heating loads.

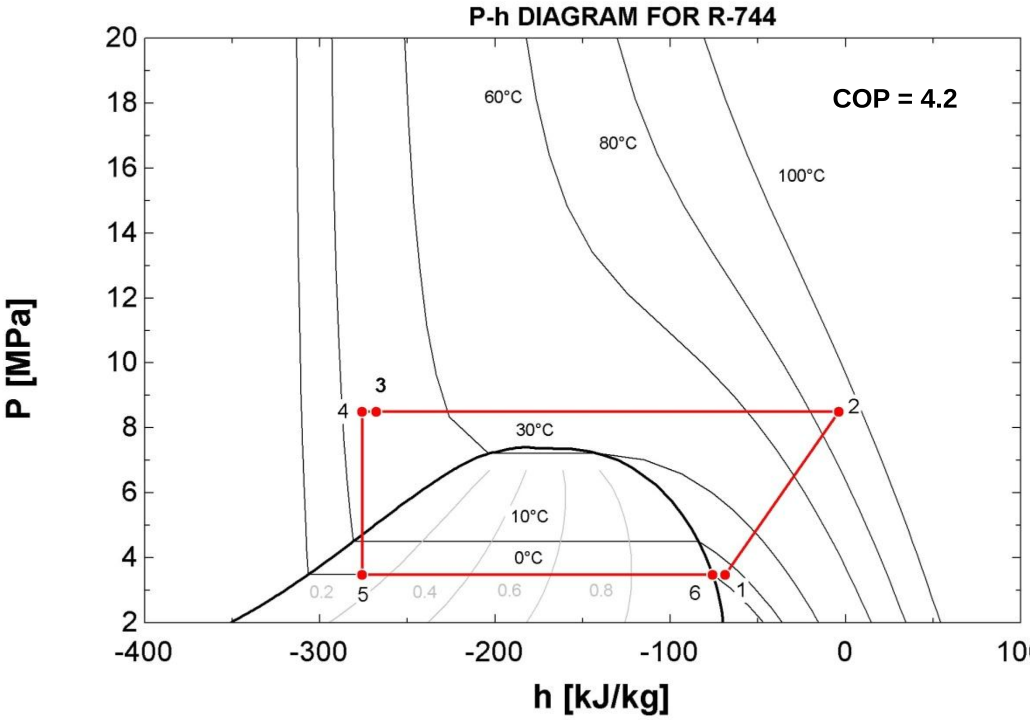


Figure 4.2.3.2.6 P-h diagram of a CO_2 heat pump system, when hot water temperature profile 80/10°C, high-side pressure of 8.5 MPa. Developed with EES

In the above system the parameters that can be changed or variable parameters are,

- Evaporator temperature (depends on surrounding air, changes very often)
- Hot water flow temperature (set-point, wont change often)
- Hot water return temperature (depends on heating system and heat loads in the heat sink i.e., house)
- Compressor output temperature (depends on hot water flow temperature which is an set-point)
- High-side pressure (depends on the above parameters, may change often)

4.2.3.5. CONCLUSIONS

Finally the simulation leads to development of a CO_2 heat pump system platform (simulation code developed in EES Software) that can be used to test performance of CO_2 heat pump for different heating loads in different seasons in a house.

There main parameters influence the performance of CO_2 heat pump simulator discussed in previous sections, most of them are kept fixed in the analysis except temperature of the outside air i.e. the air which acts as heat source, and the temperature of the return water entering gas-cooler from heat distribution system, where it get heated to set-point temperature, which changes with the time steps in the analysis.

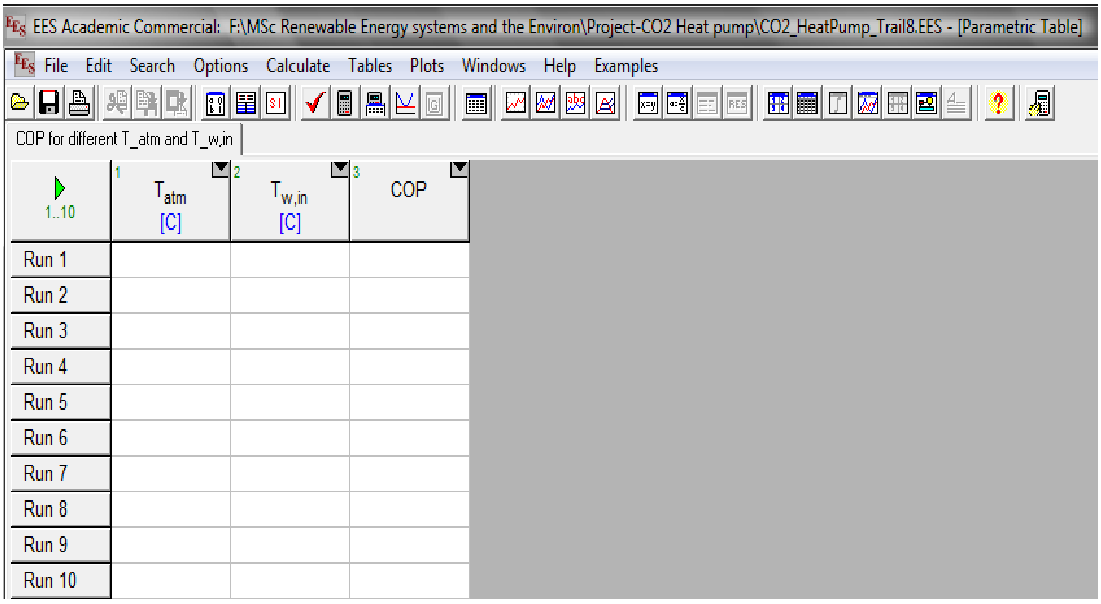


Figure 4.2.3.5.1 Parametric table in EES to calculate performance of numerical CO_2 heat pump simulator for the variable input values

The above parametric table is used to calculate the performance of the CO_2 heat pump simulator with the input parameters atmospheric air temperature and return water temperature from house heat distribution system which changes with time step based on the heat loads and demand pattern.

By providing those values in the above parametric table, part of the code developed for the CO_2 heat pump numerical simulator in the EES Software, the performance of the heat pump will be calculated based on that input.

5. FUTURE RESEARCH

5.1. TESTING CO_2 HEAT PUMP PERFORMANCE FOR DOMESTIC HEATING

The performance (COP) of the numerical CO_2 heat pump simulator for any domestic heating loads can be calculated using the code developed in EES software.

By use the heating demands and heating pattern for different seasons in a average UK domestic housing, shown in the previous sections, provided by the ADEPT software developed by ESRU, the performance of the CO_2 heat pump can be calculated.

The heating systems and heat loading parameters taken from ADEPT are generally designed for condensing or non condensing boiler. Generally these boilers provide continuous heat supply to the heat distribution system. These heat loading parameters and the heating systems may not support a heat pump, since a heat pump cannot provide continuous heating like a gas fired or oil fired boilers. Hence the heat distribution system and heating parameters has to be modified to support a CO_2 heat pump system.

By making the needed modifications the CO_2 heat pump can be tested in the case of retro-fitting in the place of conventional boilers, using our numerical CO_2 heat pump simulator.

5.2. SCOPE FOR FURTHUR RESEARCH IN NUMERICAL SIMULATION MODEL

The current numerical simulator was modelled with many assumptions, to reduce the complexity in modelling. Following are the assumptions made,

1. Heat losses from the heat exchangers and expansion device are neglected.
2. Pressure drop in Heat exchangers, connection pipelines, bends and water side are assumed negligible.
3. Changes in kinetic and potential energies are negligible.
4. *UA* values of heat exchanger were taken from manufacturer data

Future work in this area is to make the simulation model more realistic by modelling the simulator with these assumptions as design parameters; hence the final result will be more accurate.

5.3. SCOPE FOR FURTHUR RESEARCH IN MODELLING OF CO_2 HEAT PUMP CYCLE

In the current simulation, the CO_2 heat pump cycle was modelled with only few modifications for the cycle to improve the performance example introducing an internal heat exchanger. A full length investigation has to be carried out on cycle modifications to improve its performance like,

1. Investigate performance of cycle with multi-stage compressor in place of single-stage
2. Study on expander which reduces the expansion losses which have a potential to improve cycle performance.

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