Advanced sequential dual evaporator domestic refrigerator/freezer: System energy optimization

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Article history:
Received 16 August 2013
Received in revised form
14 February 2014
Accepted 2 March 2014
Available online 21 March 2014

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Nomenclature

COP	[–] coefficient of performance
E	[Wh/day] energy consumption
n	[–] run time
p	[pa] pressure
P	[W] electric power
Q	[W] heat
T	[°C] temperature
UA	[W/K] global heat transfer coefficient
DAQ	data acquisition
FC	freezer compartment
FC-CV	freezer evaporator check valve
NTC	negative temperature coefficient thermistor
P/O	pump out
PCM	phase change material
RC	refrigeration compartment
RPM	rotations per minute
SDE	sequential dual evaporator

Subscripts

disch	discharge
cond	condensation
shell	compressor shell
evap	evaporation
filter	filter/drier
suct	suction
ACC	accumulator

1. Introduction

Very common type of domestic refrigerator/freezer in EU is called bottom-mount. It contains two compartments with different temperatures placed over each other where the freezer is the lower one. Freezer compartment (FC) is dedicated to deeply frozen food with corresponding average air temperature bellow −18 °C. Refrigerator compartment (RC) is devoted to fresh food kept at temperatures above freezing point (>0 °C). Common configuration of appliances with two-evaporator is with a refrigeration circuit where two evaporators are connected in series and refrigerant evaporates at the same temperature both in FC and RC compartments. As a consequence, heat is extracted from RC with low temperature and, therefore, low thermodynamic efficiency (Jung and Radermacher, 1991). Circuit with two capillary tubes, single evaporator and air by-pass between the FC and RC compartments was also proposed but negative impact of decreased air humidity in RC was observed (Park et al., 1998). Dual loop appliances, with completely separate two refrigeration circuits, one for each compartment, were also presented by Baskin and Delafield (1999). In spite of high energy efficiency potential this configuration is not widespread due to high cost related to two compressors.

Appliance equipped with sequential dual evaporator (SDE) refrigeration circuit contains one compressor and condenser followed by flow diverting electro-valve and two evaporators connected in parallel and placed in corresponding compartments as it's shown in Fig. 1a. The circuit extracts heat from FC and RC in alternating mode, in other words only one evaporator works at a time. Therefore during RC operation it allows higher evaporation temperature and lower exergy losses as it was discussed in Visek et al. (2012). This leads to higher compressor COP as it's visible from the p–h diagram in Fig. 1b. There is no air exchange between two compartments and as the system requires only one compressor it has a considerable cost advantage versus dual loop appliances. Even though the SDE circuit domestic refrigerators are very rare on the consumer market nowadays.

Sand et al. (1992) experimentally tested SDE circuit charged with R12 and R152a and found out 2.3% and 6% energy saving over single evaporator system charged with R12. They claimed

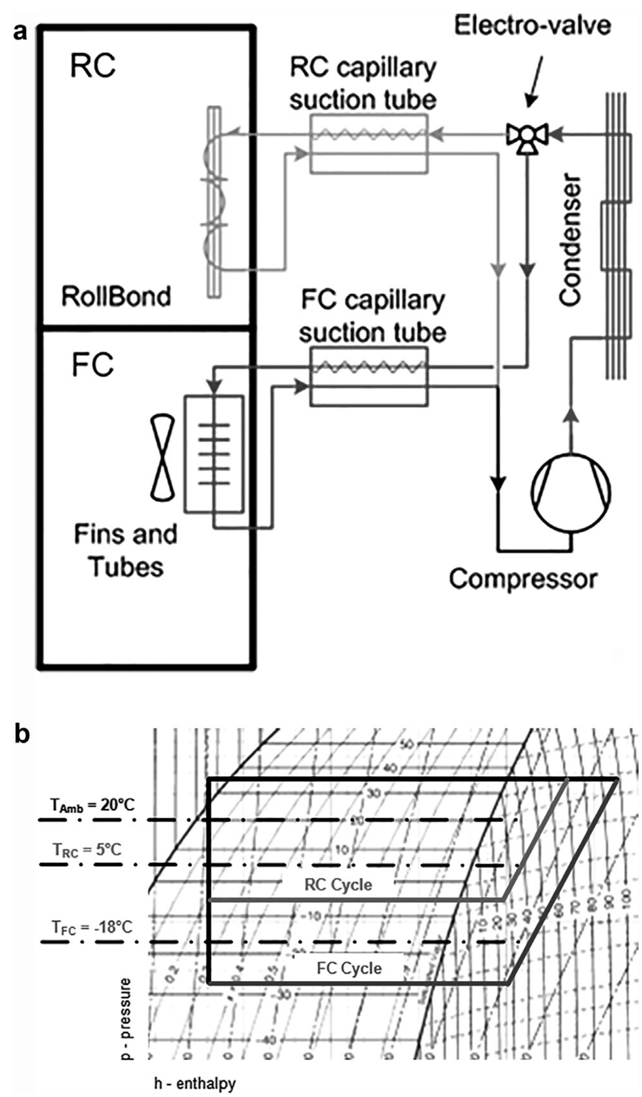

Fig. 1 – a. Traditional sequential dual evaporator refrigeration circuit design. b, Enthalpy-pressure diagram revealing energy efficiency advantage of sequential dual evaporator circuit.

that the circuit would not be readily accepted by refrigerator manufacturers because of small experimental energy saving, increased system complexity and reduced reliability. Opposite became truth when one year later Whirlpool Corporation submitted a patent with several modifications to the SDE circuit (Cur et al., 1995). Group around Radermacher (Lavannis et al., 1998) modified two evaporators in series refrigeration circuit to SDE refrigerator charged with R600a and as RC evaporator used identical evaporator to the FC. The measured reduction of energy consumption was equal to 8.5% but reported were also the issues with improper refrigerant charge in the two loops and charge migration during ON and OFF cycle. Yoon et al. (2011) carried out experimental analysis of a forced convection SDE circuit in a side-by-side domestic refrigerator appliance reporting an energy improvement equal to 7.8% over the system with evaporators in series. Moreover it was stated that by implementing refrigerant recovery sequence (R/S) between FC and RC operation additional energy saving of 1.8% was achievable.

Implementation of phase change material (PCM) based thermal storages to the refrigerators and freezers is a commonly proposed way to improve energy efficiency, reduce temperature fluctuations and improve resistance to electricity blackouts (Azzouz et al., 2008, 2009; Gin et al., 2010; Gin and Farid, 2010; Oro et al., 2012; Li et al., 2013). Subramaniam et al. (2010) showed experimental results on SDE circuit with PCM placed in both FC and RC compartments. They claimed good energy saving potential for RC operation as its evaporation temperature raised by 10 K versus baseline but overall energy benefit was not visible.

The objective of this paper is to present experimental energy consumption results for the SDE refrigerator/freezer prototype including PCM placed in direct contact with visible RC evaporator. In addition, the importance of supplementary components (check valve, block valve and condenser fan) in the SDE circuit is demonstrated.

2. Experimental setup and test procedure

All the experiments were performed on the prototype of bottom-mount built-in refrigerator/freezer appliance with internal volume of 209 L refrigerator and 83 L freezer.

2.1. Prototype description

Cabinet of standard built-in bottom-mount appliance equipped with vacuum insulation panels was taken as a baseline for modifications. Based on the previously performed experimental reverse heat leak tests the heat gains of RC and FC at 20 °C ambient air were calculated to be 16.6 W and 19.7 W with 4.5 °C and −18 °C compartment air temperatures respectively.

Existing refrigeration circuit was adapted to SDE operation. RC and FC suction line capillary tube counter flow heat exchangers were foamed inside the back wall for appropriate insulation from ambient air. Standard foamed-in RC evaporator was changed for natural convection visible aluminum roll-bond type with two active sides in “Test 1” and total heat transfer area of 0.46 m^2. Simple aluminum construction was attached in all other tests to the back side of the roll-bond to create a pocket for the phase change material. PCM, in our case 2.2 L of water, was inserted to the pocket in a plastic bag to avoid leakages. Evaporator with the pocket was clamped to the compartment’s back wall in the way to keep distance from the inner-liner so that again two walls were used for heat transfer as it is shown in Fig. 2b. FC evaporator was not modified versus serial production appliance. It was “no frost” fins and tube type with forced convection fan, mounted on the back wall in FC and covered with shroud. Length of evaporator tube was 5.5 m arranged in 2 rows and 9 tubes per row with 50 aluminum fins and tube diameter 6 mm. Compressor Embraco VEMZ9C with displacement 9.34 cm^3 was used with variable speed inverter adjustable by frequency of 5 V TTL signal in the range from 1600 RPM to 4500 RPM. Condenser was steel natural draft with fins formed out of the metal sheet with heat transfer area 1.3 m^2 attached to the back of the refrigerator. More efficient condenser was experimentally simulated by external DC voltage 5.5 W fan blowing air over existing condenser from bottom up.

Two bi-stable diverter electro-valves were placed on the liquid line after the filter/drier. The first took a functionality of block valve with one of the two outlets completely sealed. Block valve closes the refrigerant flow after the condenser in order to perform refrigerant pump out (P/O) of evaporators. The block valve is open during any compressor OFF period to allow pressure equalization through the capillary tube and smooth compressor restart. Advantage of using bi-stable valve was its pulse operation with almost negligible energy consumption. The second valve operated as a diverter valve was directing refrigerant flow either to RC or FC capillary tube. Ball FC check valve (FC-CV) with manual bypass was mounted to FC suction tube and has a purpose to avoid refrigerant back flow to FC evaporator during the RC operation and OFF period. Both block valve and FC-CV were introduced to the circuit to deal with refrigerant charge migration issues between the two evaporators. The FC loop required less refrigerant charge than RC loop. Therefore FC charge was limiting the maximum charge of the system to protect compressor from liquid refrigerant. Refrigeration circuit was charged by 42 g of hydrocarbon R600a (isobutane) determined experimentally by charging the system until the freezer evaporator was flooded and suction temperature was not bellow the dew point of the ambient.

2.2. Prototype instrumentation

Appliance prototype was instrumented with 6 standard T-type thermocouples attached to the external part of the refrigeration circuit according to Fig. 2a, further 4 and 3 thermocouples were placed on the RC and FC evaporators respectively following Fig. 2b and c. Additional 4 thermocouples were placed inside of the pocket with phase change material, two in the height of 10 cm and other two in 28 cm from the bottom of the RC evaporator as shown with black dots in Fig. 2b. Besides, 7 weighted thermocouples were measuring air temperature; 1 ambient temperature and 3 evenly spaced in each compartment. Moreover RC and FC had installed separate negative temperature coefficient (NTC) thermistors, guiding control mechanism. Pressure transducers were located at the compressor inlet and outlet. They

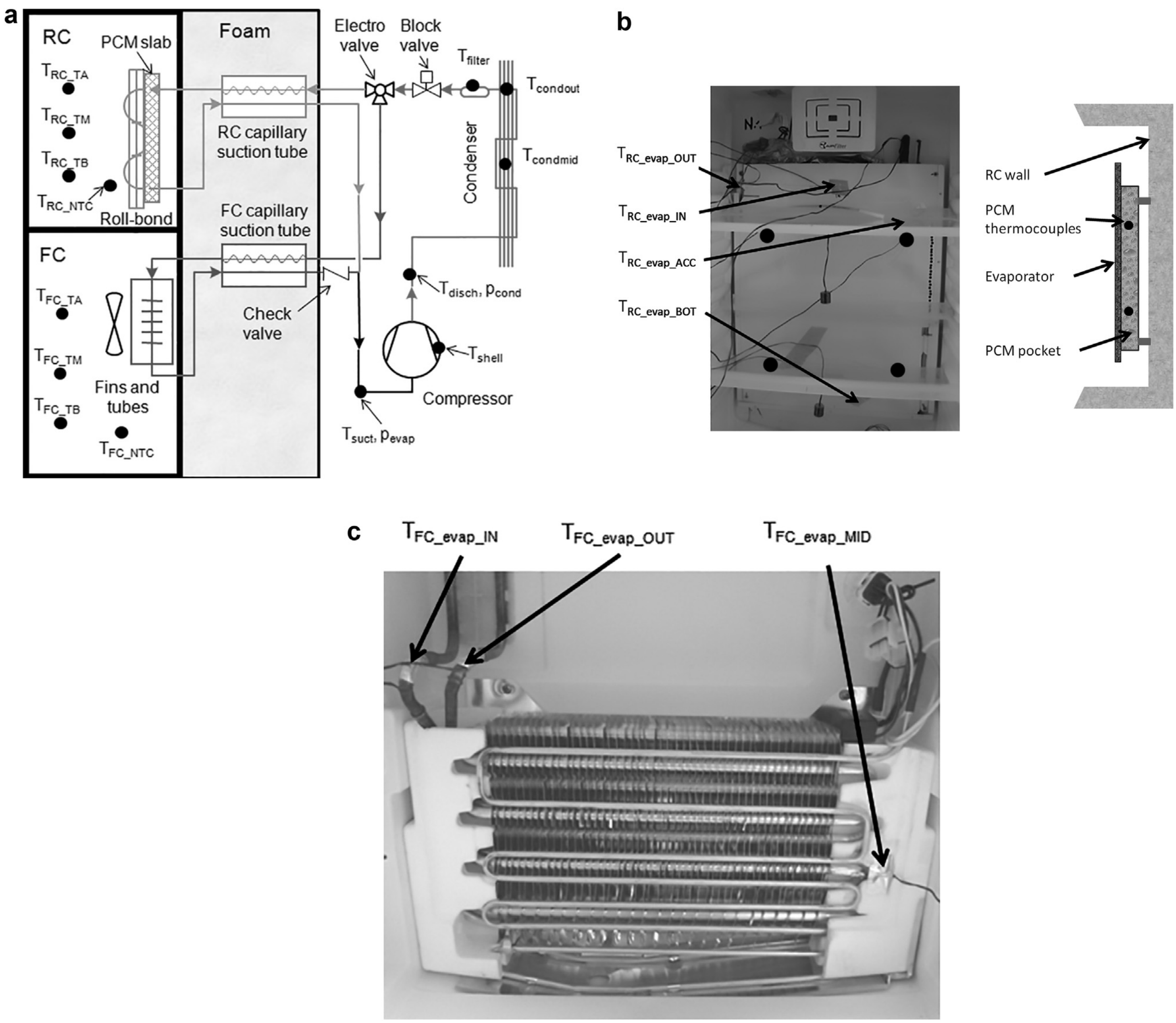

Fig. 2 – a. Instrumentation of cabinet and refrigeration circuit with thermocouples and pressure transducers. b Positions of RC evaporator thermocouples with thermal accumulation pocket in the refrigeration compartment. c Positions of FC evaporator thermocouples.

were manually calibrated before physically added to the circuit. Temperatures and pressures were acquired by the DAQ modules. Electric power and energy consumption was recorder by power transducer. Instruments and their measurement uncertainty are summarized in Table 1.

2.3. *Control mechanism*

Basic idea was that FC had a priority to be cooled in our SDE circuit. Therefore if RC evaporator was running and FC asked to be cooled in the same time, circuit switched to the FC mode. After FC was completed electro-valve returned to RC position and continued RC cooling. This control strategy is very simple and sufficient for determination of appliance's energy consumption. Control mechanism was managing following actuators: compressor alimentation and RPM, bi-stable electro-valves, FC evaporator fan and auxiliary condenser fan. RPM of compressor during RC and FC mode was constant and fixed to 54 Hz (1620 RPM) and 100 Hz (3000 RPM) respectively. The speed choice was based our prior experience with this circuit, in order to minimize energy consumption of RC and to preserve temperature pull down capability in FC (Visek et al., 2012).

Table 1 – Measurement instruments and their accuracy.

Instrument	Pressure transducer	Wattmeter, energy meter	Thermocouple
Manufacturer	Gems-sensors	Ohio Semitronics	Tewire
Type	6200FF0C0B	GH-002DT52/250K	P-26-TT
Operating range	0–10 Bar	0–1000 W	−40 to 350 °C
Accuracy	0.15% F.S. typical	0.2% Rdg., ±0.05% F.S.	±0.5 °C

Table 2 – Appliance prototype configuration during the energy consumption tests.

Test#	Control	Refrigerator components			
		SDE	PCM	FC-CV	Cond fan
1	NO P/O	X			
2		X	X		
3		X	X		
4	P/O before RC	X	X	X	
5		X	X	X	X

Further on, pump out (P/O) phase before each RC cycle was introduced in the control strategy to deal with refrigerant migration issues. During the P/O phase electrical energy is spent and negligible cooling effect is obtained thus it is wise to keep the length and number of P/O phases as low as possible. According to Visek (2013) performing P/O phase before RC in SDE-PCM appliance is more effective because only few RC cycles are required per day. Parameters of P/O phase such as P/O length 180 s, compressor RPM during P/O 1620 RPM and FC fan ON were also taken from the earlier study.

2.4. *Energy consumption tests procedure*

Five energy consumption tests were performed with varying control and appliance setup according to Table 2. All the energy consumption tests were performed during the cyclic steady condition of the prototype appliance in the environmental room. Ambient temperature 20 °C instead of typically required 25 °C was selected for all the tests to further reduce heat load of the RC and assure complete melting of the PCM. Air temperature in RC was kept between 3 °C and 6 °C and in FC between −18 °C and −21 °C with both compartments empty. Acquisition length for energy consumption tests differentiated for non PCM and PCM refrigerator setup; 24 h for non PCM and at least two full RC periods for PCM prototype. One RC period in PCM prototype lasted at least 14 h, thus 2 periods represent more than 28 h. The energy consumption results were scaled to 24 h of operation with measurement unit Wh day^{-1}.

3. Results and discussion

Average compartments' air temperatures were obtained as an average of 3 weighted thermocouples placed in each compartment and further averaged over the time. The averages differed between tests. In general measured FC average temperatures at different tests were very close to each other with standard deviation of only 0.1 K. Measurements of RC temperature had larger standard deviation 0.7 K between the tests. For valid energy comparison temperature correction procedure as described in the Appendix was applied and maximum temperature correction was 0.91 K. The measured energy consumption results were corrected to reflect unique temperatures RC = 4.5 °C and FC = −18 °C. Basic assumption of the procedure was that COP remains constant and only the compressor's run time changes with small correction of compartment air temperature especially for PCM refrigerator. Corrected energy consumption results and calculated COP's are summed up in Table 3. Detailed explanation of the correction procedure is available in the Appendix, followed by Table 4 showing measured experimental data.

RC average temperature was typically higher when PCM was present in the compartment because during compressor OFF cycle temperature was rising very modestly once it exceeded 4 °C as it is visible in Fig. 3. The reason for that is close match between heat absorbed by PCM pocket and heat gained by RC from the ambient.

$$UA_{PCM}(T_{RC} - T_{PCM}) = UA_{RC}(T_{AMB} - T_{RC}) \quad (1)$$

The equality allowed proper freezing and melting of the PCM. T_{PCM} was taken as an average between four thermocouples immersed in the PCM and measurements showed that at the beginning of charging sequence the temperature was above 0 °C thus PCM was in the liquid state. Then phase changing freezing is visible in Fig. 3 at 0 °C and at the end of the charging process certain sub-cooling of the ice was reached. Importance of suitable design of UA_{PCM} and UA_{RC} for completing PCM melting process is evident from the Equation (1). If T_{PCM} is considered constant it is clearly understood that ratio between UA values has to be adjustable to actual ambient temperature and RC thermostat setting. Thus appliance production design would most probably include RC fan and/or dumpers for air flow distribution to deal with changing thermal load.

Impact of PCM in RC, P/O phase, FC check valve and condenser fan on the appliance energy consumption was analyzed. Fig. 4 shows energy consumption of the appliance and only of the RC, in addition condensing and evaporating temperatures during RC operation are reported for each test. Actual energy consumption decomposition between FC, RC, P/O and condenser fan is presented in Fig. 5 for the same set of tests.

Table 3 – Summary of energy consumption tests results after temperature correction procedure.

Test#	Corrected energy consumption						
	RC			FC			Total
	Energy [Wh day^{-1}]	Run time	COP	Energy [Wh day^{-1}]	Run time	COP	Energy [Wh day^{-1}]
1	197	18.9%	2.02	336	20.2%	1.49	533
2	186	15.6%	2.14	335	19.4%	1.48	521
3	180	12.5%	2.39	349	19.2%	1.42	529
4	179	11.7%	2.39	346	19.0%	1.44	525
5	167	11.6%	2.76	336	19.5%	1.48	503

Table 4 – Summary of measured test results.												
Test#	Measured											
	Energy [Wh day^{-1}]						Temperature [°C]			Run time		
	Appliance	RC	FC	FC fan	Condenser fan	P/O	Ambient	RC	FC	RC	FC	P/O
1	554	211	325	17.8			20.5	3.9	−18.3	20.2%	20.6%	
2	514	178	320	16.9			19.9	5.2	−18.3	14.9%	19.5%	
3	522	160	333	16.7		13.0	20.0	5.3	−18.1	10.7%	19.3%	1.3%
4	514	157	329	16.4		12.1	19.9	5.4	−18.1	9.7%	18.9%	1.4%
5	499	138	321	17.0	13.4	8.9	20.0	5.3	−18.3	10.1%	19.6%	1.0%

3.1. *Baseline circuit*

The baseline appliance energy consumption is represented by results in the “Test 1”. Measurement was performed on the SDE circuit with simple visible roll-bond evaporator in RC without any PCM pocket. Simple control strategy without a P/O phase was implemented. Appliance energy consumption was 533 Wh day^{-1} out of which 197 Wh day^{-1} belonged to RC. RC evaporation temperature −22.9 °C was already 6.1 °C higher than FC which brought theoretically 19% compressor COP advantage in RC mode. However there was still an enormous temperature lift 26.9 K between RC evaporation temperature and RC air temperature which offered high potential for further improvement. Compressor run time was 18.9% and 20.2% with average period duration 67 and 41 min for RC and FC respectively.

Fig. 6 shows measured temperatures during refrigerator cycling for “Tests 1–5”. T_{evap} and T_{cond} are evaporation and condensation temperatures calculated by Refprop from acquired pressures. It would be expected that T_{evap} and thermocouple $T_{RC\_evap\_IN}$ should provide similar reading. No cover or insulation was applied on the evaporator thermocouples and therefore the temperature reading is significantly higher. Readings $T_{RC\_evap\_IN}$ and $T_{RC\_evap\_OUT}$ were mainly used to identify filling of the evaporator with refrigerant.

3.2. *PCM in contact with RC evaporator*

Basic idea of PCM in the contact with RC evaporator is to increase evaporation temperature by improving evaporator heat transfer coefficient and to store excess cooling capacity in the latent heat of PCM. During “Test 2” only PCM was placed in the pocket and no additional changes were made to the control and circuit versus baseline. Results showed a very modest increase in evaporation temperature during RC operation by only 2.3 K from −22.9 °C to −20.6 °C. It led to improvement of 5.7% in energy consumption during RC operation and 2.3% overall as shown in Fig. 4. Such small evaporation temperature raise was related to insufficient refrigerant charge available for RC operation. Refrigerant tends to naturally migrate to FC evaporator as it’s the coldest place of the refrigeration circuit. Thus when electro-valve switched to RC mode, refrigerant only very slowly started to move by help of the compressor from FC loop to running RC loop. It is visible from Fig. 6 “Test 2” as gradually growing evaporation temperature. On the contrary to “Test 1” even after more than 30 min the RC evaporation temperature didn’t reach maximum and neither exceeded FC air temperature. Hence, even at the end of RC operation there was still some liquid refrigerant reminding in the FC evaporator. Even though RC compressor run time decreased from 18.9% to 15.6% thus compressor was delivering higher cooling capacity. One complete RC period had duration of more than 18 h, which created extremely high resistance of RC to electricity blackouts.

3.3. *P/O before RC*

In the “Test 3”, pump out (P/O) phase was introduced to the control mechanism and activated before RC operation to deal with insufficient charge during RC operation. Refrigerant was pumped from FC evaporator to the condenser by mean of running compressor and closed block valve. After P/O was concluded the refrigerator switched directly to RC mode. Experimental results showed significant grow by 6.5 K

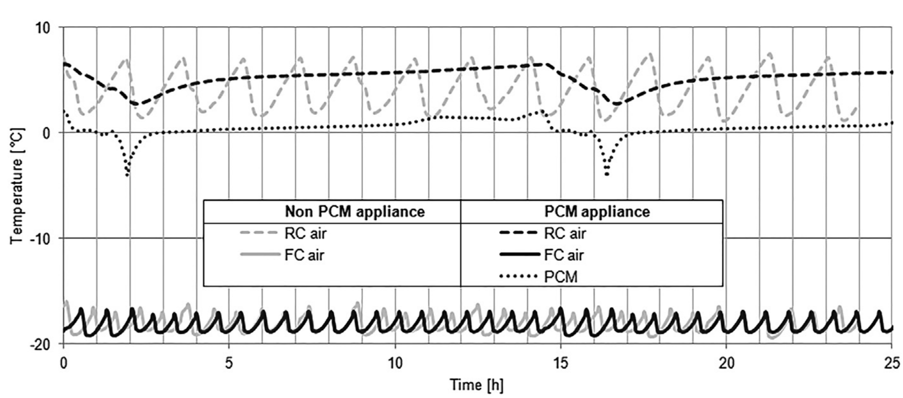

Fig. 3 – Comparison of cycling behavior of compartments air temperatures in non-PCM and PCM equipped appliance.

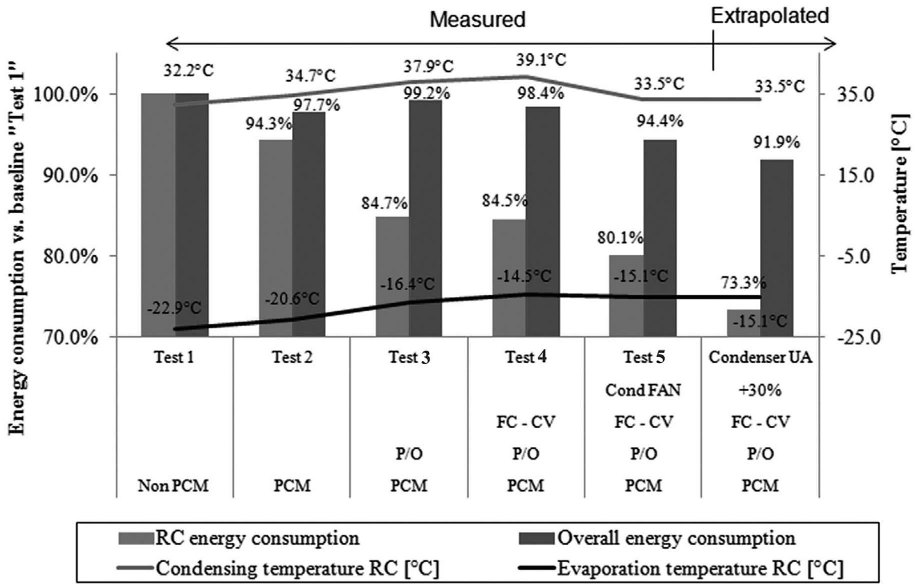

Fig. 4 – Impact of circuit and control modifications on the evaporation and condensation temperatures during RC and energy consumption of RC and overall appliance.

in the RC evaporation temperature from −22.9 °C to −16.4 °C shown in Fig. 6 "Test 3". It brought also noteworthy improvement of 15.3% in energy consumption of RC without taking into account P/O phase. But negligible energy improvement of less than 1% was observed overall as reported in Fig. 4.

3.4. *FC check valve*

In addition to P/O also the FC-CV was activated in the circuit during the "Test 4" to avoid back flow of refrigerant to the FC evaporator during RC operation and OFF cycles. Evaporation temperature rose by 8.4 K from −22.9 °C to −14.5 °C. Together with evaporation temperature also the condensing temperature was rising significantly from 32.2 °C to 39.1 °C as it is visible in Fig. 6 "Test 3 and 4". This means that higher heat capacity was absorbed by RC evaporator and also higher amount of heat needed to be dissipated by the condenser. RC energy consumption decreased by 9% from 197 to 167 + 12 Wh day^{-1} considering P/O phase being part of RC operation, see Fig. 5. But overall appliance energy consumption decreased only by negligible 1.7%. Thus FC-CV allowed increase of RC evaporation temperature by extra 2 K and overall saved approximately 1% of energy consumption.

Single RC period duration increased to 14 h 20 min with compressor run time only 10.3% and 1.4% P/O run time. Long PCM charging period had to be interrupted by FC cooling periods several times and FC consumed 11 Wh day^{-1} more energy during these FC cooling periods and penalized the overall energy consumption. The first reason for increased FC energy consumption during RC charging was already warm compressor and condenser captured in Fig. 6 which absorbed

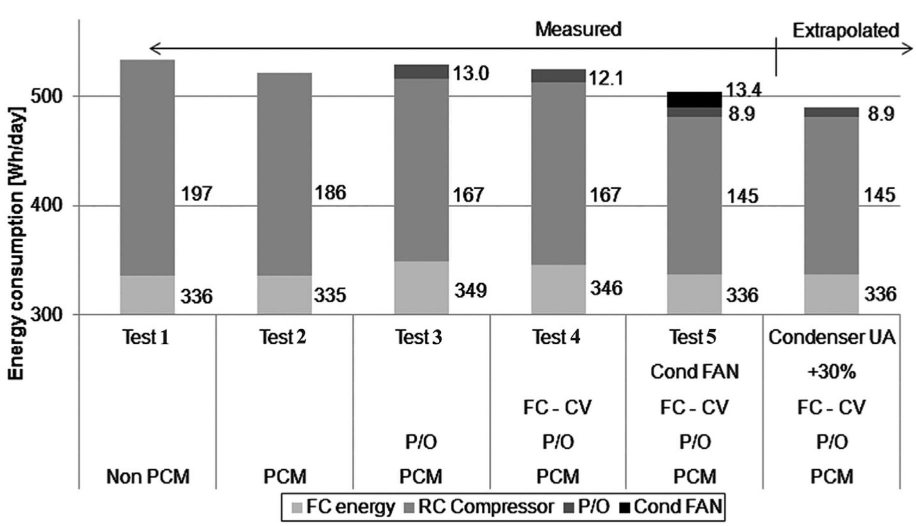

Fig. 5 – Actual energy consumption decomposition between FC, RC, P/O and condenser fan.

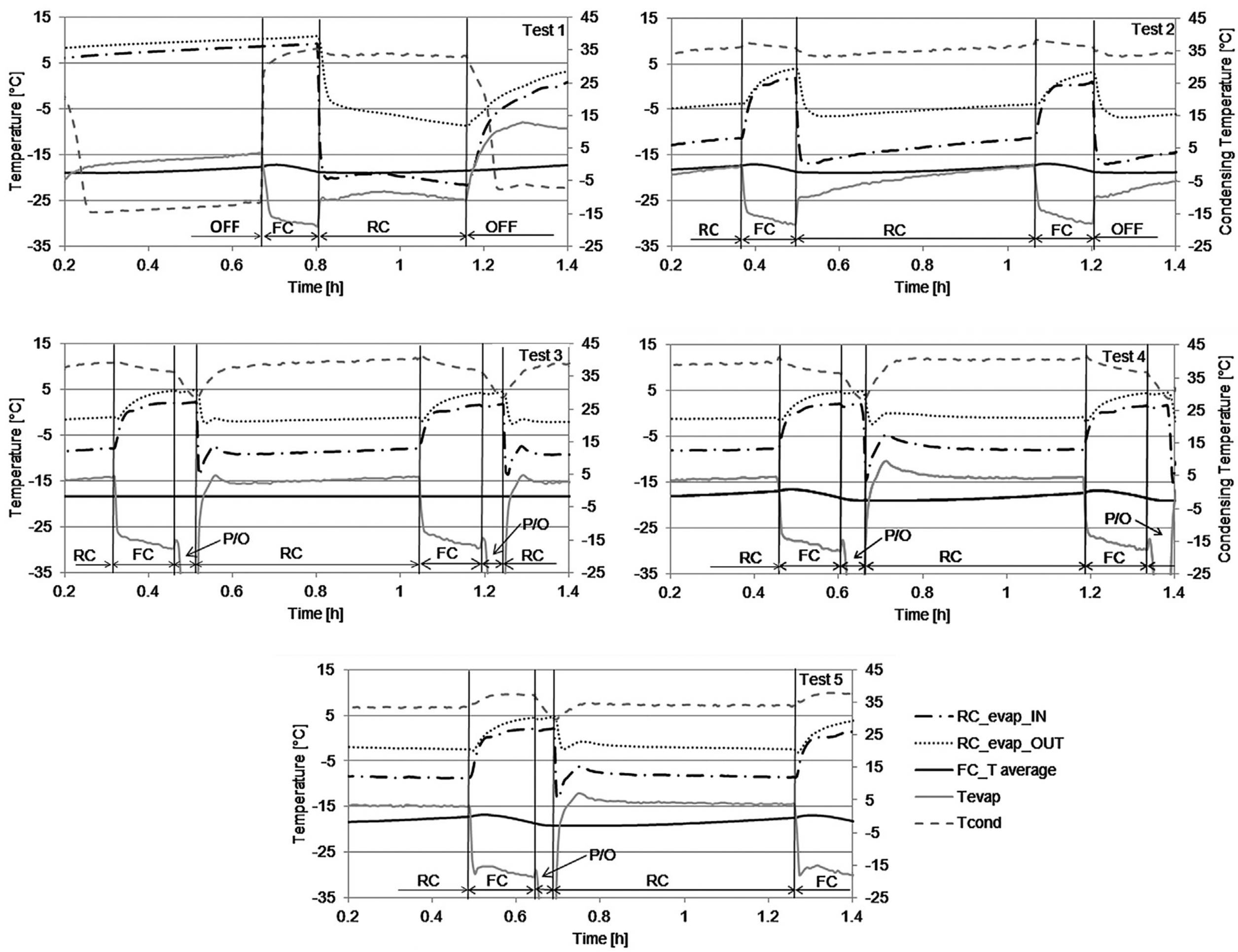

Fig. 6 – Measured RC evaporator, FC air and condensing temperature curves during "Tests 1–5".

less heat from the refrigerant fluid. The second reason might be related to the necessity for refrigerant fluid migration from RC to FC evaporator.

3.5. *Condenser fan*

Increased requirement on the condenser heat dissipation when PCM is present in RC brought penalty to the compressor efficiency in terms of higher condensing temperature. External fan was mounted under the existing condenser to increase air heat transfer coefficient. Power consumption of the fan was 5.5 W. During the "Test 5" the control was modified to run this fan only during the RC operation. The condensing temperature was pushed down to 33.5 °C shown in Fig 6 "Test 5" and it showed improvement of 19.9% in RC energy consumption including condenser fan electricity. Overall appliance energy consumption dropped by 5.6% versus the baseline. Air flow performance of fan accounted for approximately 30% increase in the UA value of the condenser. Last column in Fig. 4 assumed that 30% increase in UA value was done by modifications to the condenser design, thus fan's 13 Wh day^{-1} could be removed. RC energy usage decreased by 26.7% and appliance overall 8.1% energy saving resulted. More efficient condenser could positively impact also FC efficiency but this was not considered in the analysis.

4. Conclusions and future work

Bottom mount built-in domestic refrigerator/freezer appliance prototype with advanced SDE circuit was designed, built and tested. Several technologies such as block valve, check valve, P/O phase, condenser fan and PCM in contact with RC evaporator were implemented to increase RC evaporation temperature by maximum 8.4 K and to provide 5.6% overall appliance energy saving. The condenser fan is a well known technology but natural convection condensers are preferred option mainly because of noise reduction, reliability and cost. The potential to reduce the energy consumption by 8.1% can be achieved by implementing natural convection condenser with approximately 30% higher UA value.

Properly performed P/O and activated FC-CV were crucial to reach high evaporation temperatures during RC operation. Block and diverter functionality performed by two separate valves can be included in the future to one stepper three way electro-valve and provide additional functionality of opening both capillary tubes in the same time. Moreover condenser fan was used to improve global heat transfer coefficient by 30% and in the future this improvement should be reached by design modifications such as larger condenser heat transfer area and two rows of tubes.

Almost 15 K temperature lift between PCM and refrigerant fluid temperature is still present and offers extra potential for improvement. To further reduce heat transfer restrictions between PCM and refrigerant two steps should be performed. First, study of charge distribution in the sequential system similar to work of Bjork (2005) and Bjork and Palm (2006) should be carried out, followed by determination of optimal internal volume of the evaporators to assure sufficient refrigerant filling of both evaporators in the sequential system. In the second step it is necessary to improve contact thermal conductivity and heat transfer area between PCM and evaporator external surface.

Acknowledgment

We would like to express appreciation to Whirlpool R&D for making this research possible by providing prototypes and laboratory apparatus for measurements.

Appendix A

Temperature and energy consumption correction was performed in 4 simple steps

1. Based on measured ambient and internal temperatures actual compartment heat gain was calculated. Global heat transfer coefficients of RC and FC were determined beforehand from measured reverse heat leakage tests ($UA_{RC} = 1.073$ and $UA_{FC} = 0.518\ W\ K^{-1}$)

$$Q_{xC} = UA_{xC}\left(\overline{T}_{amb} - \overline{T}_{xC}\right) \tag{2}$$

2. RC and FC COP were resolved. E_{xC} is electrical energy consumed solely by compressor during xC operation without P/O, FC fan and condenser fan.

$$COP_{xC} = \frac{Q_{xC} 24\ h}{E_{xC}} \tag{3}$$

3. Correction of RC and FC energy consumption was performed for internal air temperatures 4.5 °C and −18 °C respectively. $Q_{RC\ 4.5\ ^\circ C}$ was a steady state heat gain of RC at 4.5 °C and ambient temperature 20 °C. Equivalent is valid for $Q_{FC-18\ ^\circ C}$.

$$E_{RC\ cor} = \frac{Q_{RC4.5^\circ C} 24\ h}{COP_{RC}} + E_{P/O} + 24\ n_{RC} P_{cond\ fan} \tag{4}$$

$$E_{FC\ cor} = \frac{Q_{FC-18^\circ C} 24\ h}{COP_{FC}} + 24\ n_{FC} P_{FC\ fan} \tag{5}$$

4. RC and FC compressor run times were recalculated by following equations.

$$n_{RC\ cor} = \frac{n_{RC} E_{RC\ cor}}{E_{RC} + E_{P/O} + 24\ n_{RC} P_{cond\ fan}} \tag{6}$$

$$n_{FC\ cor} = \frac{n_{FC} E_{FC\ cor}}{E_{FC} + E_{P/O} + 24\ n_{FC} P_{FC\ fan}} \tag{7}$$

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