REFRIGERATION CYCLE EFFICIENCY IMPROVEMENT SUPPORTED ON DUAL RESPONSE OPTIMIZATION

Nuno Ricardo COSTA[*,**], **João GARCIA**[*]
[*]Instituto Politécnico Setúbal, Escola Superior de Tecnologia de Setúbal, Campus do IPS, Estefanilha, 2910-761 Setúbal, Portugal joao.garcia@estsetubal.ips.pt
[**]UNIDEMI/DEMI, Faculdade de Ciências e Tecnologia-Universidade Nova de Lisboa, 2829-516 Caparica, Portugal nuno.costa@estsetubal.ips.pt

ABSTRACT

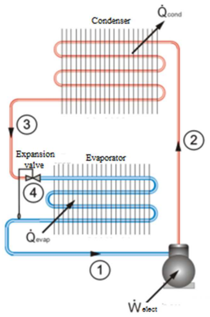
To minimize energy consumption and maximize refrigeration effect of a compression refrigeration cycle, statistically designed experiments were performed and analyzed. A faced-centered cube design was run to support the estimation of regression models fitted to refrigeration and electrical powers. Simultaneous optimization of response models provide a better understanding of how controllable variables impacts on cycle efficiency, yielding optimal variable settings. Results confirm the usefulness of proposed approach for device design and operation purposes.

1. INTRODUCTION

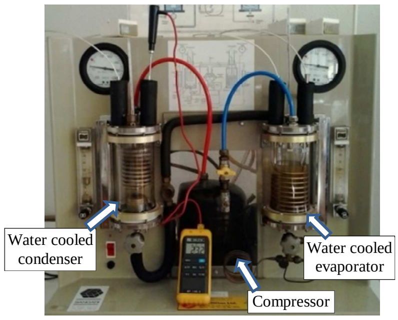
One stage refrigeration compression cycle (hereafter denoted as RC) is a thermodynamic cycle incorporated in a diversity of equipments used, for example, in domestic and public rooms for air conditioning purposes, in food and pharmaceutical industries for refrigeration and conservation as well as in health services to maintain some medicines at low temperature. From theoretical (thermodynamic) point of view the RC has been widely explored (Koelet, 1992; Mackensen et al., 2002; Horbaniuc, 2004; Rasmussen and Shenoy, 2012; Rasmussen, 2012; Anand et al., 2013), namely due to the recent refrigeration fluid restrictions related with environment protection as well as to the need of energy efficiency improvements and energy savings (Palm, 2008; Tassou et al., 2010; Bansal et al., 2012), and its working principle can be summarized as follows (see Figure 1): The work input to the RC drives a compressor which maintains a low pressure in evaporator and a higher pressure in condenser. In the low pressure evaporator the refrigerant fluid evaporates at low temperature, extracting (sensible) heat from the cooled medium and reducing its temperature (section 4-1). The low pressure vapor formed is drawn into the compressor, its pressure is increased, and then is delivered to the condenser (section 1-2) where the high-pressure fluid at higher temperature is condensed and heat is transferred to the cooled medium (section 2-3). When the warm fluid at high-pressure passes through the expansion valve its pressure decreases to evaporator pressure and its temperature falls (section 3-4), returning to the evaporator at a controlled rate.

Efficiency of a system can be defined as the ratio between the energy that we get from the system (output energy) and the energy spent to drive the system (input energy). In the presented refrigeration cycle the output energy is the heat transferred to the cooled medium by unit of time and the input energy is that furnished to drive the system (supplied to the compressor). Thus, cycle efficiency improvement can be achieved by lowering the input energy and increasing the output energy (Dincer, 2003).

Coefficient of Performance (COP) is usually used to assess refrigeration cycle efficiency (Dabas et al., 2011), and is defined as the ratio between refrigeration power (or heat-extraction capacity), denoted by $\dot{Q}_{evap}$, and electric power supplied to the compressor, denoted by $\dot{W}_{elect}$ (Anand et al., 2013). Therefore, simultaneous maximization of $\dot{Q}_{evap}$ and minimization of $\dot{W}_{elect}$ is an appropriate approach to design and improve refrigeration cycles efficiency. For this purpose, statistically designed experiments were performed in a didactic refrigeration cycle installation (see Figure 2), complemented with two auxiliary devices, and second order models fitted to response variables $\dot{Q}_{evap}$ and $\dot{W}_{elect}$ which were then aggregated and this composite function optimized.



<Fig. 1 - Refrigeration cycle>



<Fig. 2 – Didactic unit>

2. EXPERIMENTAL INSTALLATION

The didactic installation used in this study is a one stage refrigeration compression cycle produced by P.A. Hilton Ltd (see Figure 2). It includes a compressor (Aspera NEK6214Z), a condenser constructed from a thick-walled glass cylinder with machined brass end plates and a coil of copper tube inside (through which heating water flows), an evaporator constructed from a thick-walled glass cylinder with machined brass end plates and a coil of copper tube inside (through which cooling water flows), and an expansion valve (a float operated needle valve situated in the bottom of the condenser). The refrigeration fluid is R141b and integrated instrumentation enables to measure water flow rates, water temperatures, evaporator and condenser pressures, as well as evaporator and condenser temperatures.

Two auxiliary apparatus were built for heating and cooling water in order to set the temperature in the inlet and outlet of both the evaporator and condenser at planned values. Hot water was produced in a gas burner, stored in a thermo-accumulator tank (SOLCAP- 200 litres) to stabilize the temperature at specified values, and then pumped to the condenser. Cold water was obtained by introducing ice water in a tank where current water was stored, and then pumped to the evaporator at desired temperature. Hot and cold water systems are independent and water mass flow rates controlled. In addition, one thermostat was installed on each system to assure that (hot and cool) water is supplied to the condenser and evaporator at desired temperature.

2.1. Refrigeration and Electric Powers

Refrigeration power is a measure of the heat-extraction capacity of refrigeration equipments that can be calculated by applying the first law of thermodynamics to open stationary systems (where heat, work, and mass can enter and/or leave the system). The first law of thermodynamics is a version of the law of conservation of energy applied to thermodynamic systems and states that the total energy of an isolated system remains constant; energy can be transformed from one form to another, but cannot be created or destroyed. Thus, in RC and under the assumption that heat losses in evaporator are negligible and the process is stationary, the energy received by refrigeration fluid from the water in the evaporator is equal to the energy transferred (released) by water to the refrigeration fluid. Thus, refrigeration power ($\dot{Q}_{evap}$) can be defined as

$$\dot{Q}_{evap} = \dot{m}_{evap} Cp_{water} (Tout_{evap} - T_{evap}) \tag{1}$$

Notation and variable units are as follows:

$\dot{Q}_{evap}$ – Refrigeration Power (W); $\dot{m}_{evap}$ – Water mass flow rate in evaporator (kg/s); Cp_{water} – Specific heat of water at constant pressure (4.18 kJ/kgK); T_{evap} – Inlet water temperature in the evaporator (ºC); $Tout_{evap}$ – Outlet water temperature in the evaporator (ºC).

Electric power is the rate of energy consumption per time supplied to the compressor, expressed in Watts, and in this study was measured with an analyzer Chauvin Arnoux (Qualistar plus CA 8335).

3. DESIGN OF EXPERIMENTS

In industrial and domestic refrigeration systems two variables have a significant impact on cycle efficiency; the called condensation and evaporating pressures. These pressures can be manipulated by varying the inlet temperature and the water mass flow rate in both evaporator and condenser, because the heat transferred in both devices will vary and, consequently, increasing or decreasing the pressure in evaporator and condenser. In particular, decreasing the water inlet temperature and/or the reducing the water flow rate in evaporator will decrease the pressure in this device; the pressure in evaporator will increase by increasing water inlet temperature and/or water flow rate. In the condenser the pressure will decrease by decreasing the water inlet temperature and/or increasing the water flow rate; to increase the water inlet temperature and/or to reduce the water flow rate will increase the pressure in condenser. Moreover, it is known that either decreasing condensation pressure or increasing evaporation pressure will result in less effort in the compressor and, by consequence, to reduce the power consumption of compressor.

In this context, to obtain a COP (cooling) value for the refrigeration cycle under study as high as possible, four parameters (input variables or factors) were considered in the experimental design, namely the inlet water temperature in condenser (T_{cond}), inlet water temperature in evaporator (T_{evap}), water mass flow rate in the evaporator ($\dot{m}_{evap}$), and water mass flow rate in condenser ($\dot{m}_{cond}$).

3.1 Experimental Design

Design and conduct experiments are not trivial tasks, though various authors have presented guidelines to help researchers and practitioners in planning, conducting and analyzing experimental studies (Coleman and Montgomery, 1993; Bisgaard, 1999; Costa et al., 2006; Tanco et al., 2009a; Freeman et al., 2013; Simpson et al., 2013). To select an appropriate experimental design requires some theoretical background, because using an inappropriate experimental design is sure to compromise study conclusions. Tanco et al. (2009b) focused their work on design selection, discussing the key points in this task and illustrated them based on case studies from the literature.

To explore the relationship between the four independent factors (T_{cond} ,T_{evap} , $\dot{m}_{evap}$, $\dot{m}_{cond}$) and each one of the two dependent variables (responses) considered in this study, namely the $\dot{Q}_{evap}$ and $\dot{W}_{elect}$, a faced-centered design (FCD) was selected. The experimental design consists of a two level full factorial design (2^4 = 16 experiments), 8 star points and 4 center points, allowing to estimate second order and other non-linear components of the relationship between factors and response. The 4 center points can produce the required design variance stability, because the region delimited by the factors range represents the regions of interest and operability. Factor levels are listed in Table 1 and experimental design (matrix of experiments) is displayed in Table 2. Further information about FCD, and other designs, namely designs evaluation and comparison can be found in Anderson-Cook et al. (2013), Dejaegher and Heyden (2011), and in classical books about Response Surface Methodology like that by Myers et al. (2009).

Table 1- Variable settings

Level	Coded value	T_{cond} (ºC)	$\dot{m}_{cond}$ (g/s)	T_{evap} (ºC)	$\dot{m}_{evap}$ (g/s)
Maximum	1	35	30	24	30
Center point	0	30	20	17	20
Minimum	-1	25	10	9	10

4. RESPONSES MODELLING

The designed experiments were run in the thermodynamic laboratory of Setubal Polytechnic Institute – ESTSetubal, and the response results are presented in Table 2. The data were analysed using the statistical software STATISTICA® and second order models fitted to refrigeration power ($\dot{Q}_{evap}$) and electric power supplied to the compressor ($\dot{W}_{elect}$) based on analysis of variance (ANOVA) results.

The estimated regression coefficients for $\dot{Q}_{evap}$ response are given in Table 3, and the model fitted to this response, after sent to the ANOVA error term some variables/interactions, is as follows:

$$\dot{Q}_{evap} = 165.2534 + 8.5287\,x_1 + 52.7144\,x_3 + 16.4878\,x_4 + 26.1552\,x_1^2 - 24.7834\,x_3^2 + 8.5802\,x_1x_3$$

where x_i ($1 \leq x_i \leq 1$ for i =1,…,4) denotes the coded label of the i-th independent variable.

This model explains 91.2% of the variation in the data (Adjusted R-sqr = 0.912), and graphical residual analysis did not show violations of the ANOVA assumptions (residuals Normality, Independence, and Homoscedasticity).

Table 2 – Matrix of Experiments and Results

Standard order of runs		T_{cond} (ºC)	$\dot{m}_{cond}$ (g/s)	T_{evap} (ºC)	$\dot{m}_{evap}$ (g/s)	$\dot{Q}_{evap}$ (W)	$\dot{W}_{elect}$ (W)
1	Full factorial design	25	10	9	10	100.32	183
2		25	10	9	30	125.40	185
3		25	10	24	10	167.20	185
4		25	10	24	30	250.80	187
5		25	30	9	10	104.50	186
6		25	30	9	30	125.40	187
7		25	30	24	10	175.56	184
8		25	30	24	30	225.72	186
9		35	10	9	10	104.50	191
10		35	10	9	30	137.94	192
11		35	10	24	10	229.90	204
12		35	10	24	30	238.26	215
13		35	30	9	10	91.96	191
14		35	30	9	30	125.40	194
15		35	30	24	10	229.90	208
16		35	30	24	30	263.34	212
17	Star points	25	20	17	20	175.56	187
18		35	20	17	20	183.92	198
19		30	10	17	20	158.84	184
20		30	30	17	20	183.92	182
21		30	20	9	20	83.60	181
22		30	20	24	20	167.20	188
23		30	20	17	10	167.20	193
24		30	20	17	30	175.56	195
25	Center points	30	20	17	20	175.56	191
26		30	20	17	20	167.20	183
27		30	20	17	20	167.20	182
28		30	20	17	20	183.92	189

Table 3 - Estimated Regression Coefficients: $\dot{Q}_{evap}$

R-sqr=0.95202; R-Adj:0.90034

Term		Coeff.	Std. Error	$t_{(13)}$	*p*
	Mean/Interc.	162.9683	5.36152	30.39590	**0.000000**
$\mathbf{x_1}$	$\boldsymbol{T_{cond}}$	8.5287	3.646447	2.33890	**0.035962**
$\mathbf{x_1^2}$	$\boldsymbol{T_{cond} \times T_{cond}}$	18.1588	9.632302	1.88520	**0.081951**
x_2	$\dot{m}_{cond}$	0.6824	3.646447	0.18714	0.854440
x_2^2	$\dot{m}_{cond} \times \dot{m}_{cond}$	9.7988	9.632302	1.01728	0.327579
$\mathbf{x_3}$	$\boldsymbol{T_{evap}}$	52.7144	3.646334	14.45683	**0.000000**
$\mathbf{x_3^2}$	$\boldsymbol{T_{evap} \times T_{evap}}$	-32.8128	9.678384	-3.39032	**0.004831**
$\mathbf{x_4}$	$\boldsymbol{\dot{m}_{evap}}$	16.4595	3.646447	4.51385	**0.000582**
x_4^2	$\dot{m}_{evap} \times \dot{m}_{evap}$	9.7988	9.632302	1.01728	0.327579
x_1x_2	$T_{cond} \times \dot{m}_{cond}$	0.7838	3.867522	0.20265	0.842546
$\mathbf{x_1x_3}$	$\boldsymbol{T_{cond} \times T_{evap}}$	8.5802	3.866567	2.21909	**0.044894**
x_1x_4	$T_{cond} \times \dot{m}_{evap}$	-4.4413	3.867522	-1.14835	0.271518
x_2x_3	$\dot{m}_{cond} \times T_{evap}$	1.9265	3.866567	0.49824	0.626635
x_2x_4	$\dot{m}_{cond} \times \dot{m}_{evap}$	-0.7838	3.867522	-0.20265	0.842546
x_3x_4	$T_{evap} \times \dot{m}_{evap}$	3.8143	3.866567	0.98648	0.341902

A similar analysis was done with the electric power values, and the model fitted to $\dot{W}_{elect}$ is as follows:

$$\dot{W}_{elect} = 186.1514 + 7.4677\,x_1 + 4.4087\,x_3 + 1.5556\,x_4 + 5.0338\,x_1^2 - 4.4662\,x_2^2 + 6.5338\,x_4^2 + 4.3562\,x_1x_3$$

The Adjusted R-sqr = 0.919 and violations of ANOVA assumptions were not identified.

5. OPTIMIZATION

To simultaneously optimize the two estimated models ($\dot{Q}_{evap}$ and $\dot{W}_{elect}$), it was used the Excel-Solver tool and an easy-to-implement, yet effective, aggregate function introduced by Costa (2010). This function was extensively evaluated by Costa and Lourenço (2014; 2011), and is defined as

$$Minimize \quad \sum_{i=1}^{n}\left(\frac{\left|\hat{y}_i-\theta_i\right|}{U_i-L_i}\right)^{p_i} \tag{2}$$

where p_i is a preference parameter (shape factor; $p_i>0$), θ_i is the target value for the i-th estimated response $\hat{y}_i$ ($\dot{Q}_{evap}$, $\dot{W}_{elect}$), and U_i and L_i are the upper and the lower response specification limits, respectively. Compromise solution was selected based on cumulative value of the responses deviation from target, such as recommended by Costa et al. (2011). This metric is defined as

$$B_{cum} = \sum_{i=1}^{n} W_i\left|\hat{y}_i^* - \theta_i\right| \tag{3}$$

where $\hat{y}_i^*$ represents the i-th estimated response ($\dot{Q}_{evap}, \dot{W}_{elect}$) value at “optimal” variables setting and W_i is a parameter that takes into account the response type, dimension and scale (Costa et al., 2011). This parameter is defined as $W=1/(U-L)$ for responses that must be either higher than a minimum value, such as it is the case of $\dot{Q}_{evap}$, or lower than a maximum value, such as it is the case of $\dot{W}_{elect}$.

Specification limits for $\dot{Q}_{evap}$ are $L_1=70$ and $U_1=230$, with target θ_1=230; for $\dot{W}_{elect}$ the specification limits are $L_2=170$ and $U_2=230$, with target θ_2=170. The compromise solution that yields the lowest B_{cum} value, as well as the higher COP (cooling) value ($\dot{Q}_{evap}/\dot{W}_{elect}$), is denoted by S1 in Table 4. To show that reducing (improving) $\dot{W}_{elect}$ value leads to degradation of $\dot{Q}_{evap}$ value, namely in terms of B_{cum} and COP values, solution denoted by S2 is also included in Table 4.

Table 4 - Compromise solutions

	Shape factors	x_i	$\dot{Q}_{evap}$	$\dot{W}_{elect}$	B_{cum}	COP
S1	(1.00. 4.00)	(0.91, 1.0, 0.71, 0.31)	230.00	199.62	29.62	1.15
S2	(6.00. 1.00)	(-0.76, 1.0, -0.11, -0.09)	167.12	178.71	71.59	0.94

6. RESULTS DISCUSSION

From model fitted to refrigeration power ($\dot{Q}_{evap}$) one can see that linear term x_3 (T_{evap}) is the most important for response maximization. Its coefficient is almost 6.2 times greater than that of x_1 (T_{cond}) and x_1x_3 (T_{cond} x T_{evap}), 3.2 times greater than that of x_4 ($\dot{m}_{evap}$), and 2 times greater than that of x_1^2 and x_3^2. In absolute value, the effects of x_1^2 and x_3^2 have a similar impact on $\dot{Q}_{evap}$, though they are opposite; x_3^2 has a negative effect on $\dot{Q}_{evap}$, whereas x_1^2, like all the other terms, has a positive effect. In practice, the greater the value of these later terms are, the higher refrigeration effect will be, such it is desired because higher refrigeration power values lead to higher cycle efficiency (COP cooling value).

Electric power model ($\dot{W}_{elect}$) includes linear and quadratic terms as well as an interaction term statistically significant. The linear term x_1 (T_{cond}) and x_4^2 ($\dot{m}_{evap}$ x $\dot{m}_{evap}$) are the highest in magnitude. Slightly slower values are those of x_3 (T_{evap}), (T_{cond} x T_{evap}), (T_{cond} x T_{cond}). In contrast to all the other terms, the greater x_2^2 ($\dot{m}_{cond}$ x $\dot{m}_{cond}$) value is, the smaller $\dot{W}_{elect}$ value will be, which is desirable, because electric power value must be as low as possible in order to maximize the cycle efficiency and to increase the compressor life-cycle. However, the magnitude of x_2^2 is much smaller than the sum of all the other terms, which are also terms of the $\dot{Q}_{evap}$ model. This means that increasing $\dot{Q}_{evap}$ value will lead to an increase in the $\dot{W}_{elect}$ value, which is not favourable. To increase $\dot{Q}_{evap}$ and reduce the $\dot{W}_{elect}$ values would be more desirable. However, this is difficult to put in practice. In this context, compromise solution denoted by S1 in Table 4 is the best alternative found. To validate this solution, two confirmatory experiments with variables at optimal values were run. The results of these two experimental runs are in close agreement with that of S1 solution.

COP (cooling) value yielded by S1 is, theoretically, low (slightly higher than one), which is not unexpected taking into account the installation used in this study (Bjork, 2012). This does not mean that experimental methodology and study results are of no interest or unhelpful, because one must be aware that both evaporator and condenser are didactic components made in glass, with very low heat transmission capacity, the expansion valve is a float operated needle valve, and compressor technical specifications are not the most favourable, which impacts on COP values significantly. In fact, old-time and small didactic units like the one used in this study are not designed or built with efficiency purposes. They are a valuable teaching aid for students in a wide range of courses from craft and technician training to Polytechnic and University levels, and are used to help them in visualizing and understanding the events within the various components. The performance of current refrigeration systems is, indeed, higher because they integrate components (compressor, evaporator, condenser, refrigerant fluid, etc.) of higher quality (with better technical characteristics).

7. CONCLUSIONS

Statistically designed experiments were performed to maximize refrigeration effect and minimize energy consumption of a compression refrigeration cycle, using a small didactic installation. Second order models were fitted to refrigeration and electric powers, providing a better understanding of how the considered input variables affect both responses, which will be very useful for refrigeration cycle design and operation purposes. Optimal variable settings for the inlet water temperature and water mass flow in both condenser and evaporator were suggested and validated by confirmatory experiments.

Results provide evidence that illustrated experimental approach is appropriate for refrigeration cycle design and operation improvement purposes so, as future research, a plan to apply the methodology in domestic and industrial equipments as well as to test other refrigeration fluids and compressor types is been scheduled.

ACKOWLEDGEMNTS

The authors thanks to IPS-ESTSetubal for providing the didactic refrigeration cycle and all the other instrumentation and to the Eng. Vitor Pinho for his contribution in the experimental part of this research.

8. REFERENCES

Anand, S., Gupta, A., Tyagi, S. (2013). Simulation studies of refrigeration cycles: A review. *Renewable Sustainable Energy Rev.* 17(1): 260-277.
Anderson-Cook C, Borror C, Montgomery D. Response surface design evaluation and comparison. *Journal of Statistical Planning and Inference* 2009; 139: 629-641.
Bansal P, Vinyard E, Abdelaziz O. Status of not-in-kind refrigeration technologies for household space conditioning, water heating and food refrigeration. *International Journal of Sustainable Built Environment* 2012; 1: 85-101.
Bisgaard S. Proposals: a mechanism for achieving better experiments. *Quality Engineering* 1999; 11: 645-649.
Bjork, E. 2012. Energy efficiency improvements in household refrigeration cooling systems (PhD thesis). Royal Institute of Technology, UK.

Coleman D, Montgomery D. A systematic approach to planning for a designed industrial experiment. *Technometrics* 1993; 35: 1-12.
Costa, N., 2010. Simultaneous Optimization of Mean and Standard Deviation. *Qual. Eng.* 22, 140-149.
Costa, N., Lourenço, J., 2014. "A Comparative Study of Multiresponse Optimization Criteria Working Ability", Chemometrics and Intelligent Laboratory Systems, 138: 171 - 177.
Costa, N., Lourenço, J., 2011. "Multiresponse Optimization and Pareto Frontiers", Quality and Reliability Engineering International 28, 7: 701 - 712.
Costa, N., Pereira, Z., Tanco, M., 2011. Assessing Response's Bias, Quality of Predictions, and Robustness in Multiresponse Problems, in Sio-Iong, A., Len, G. (Eds.), *Electrical Engineering and Applied Computing*. Dordrecht: Springer Netherlands, pp. 445 - 457.
Costa N, Pires A, Ribeiro C. Guidelines to help practitioners of design of experiments. *TQM Magazine* 2006; 18: 386-399.
Dabas, J., Dodeja, A., Kumar, S., Kasana, S., 2011. Performance Characteristics of Vapour Compression Refrigeration System Under Real Transient Conditions. *Int. J. Advancements Technol.* 2, 584-593.
Dejaegher B, Heyden Y, Experimental designs and their recent advances in set-up, data interpretation, and analytical applications. *Journal of Pharmaceutical and Biomedical Analysis* 2011; 56 141-158.
Dincer, I., 2003. *Refrigeration Systems and Application.* Wiley & Sons Ltd, England.
Freeman L, Ryan A, Kensler J, Dickinson R, Vining G. A Tutorial on the Planning of Experiments. *Quality Engineering* 2013; 25: 315-332.
Horbaniuc, B., 2004. Refrigeration and Air-Conditioning. *Reference Module in Earth Systems and Environmental Sciences*, from *Encyclopedia of Energy*, 5, 261-289.
Koelet P, *Industrial Refrigeration - Principles, Design and Applications*. The MacMillan Press. 1992.
Mackensen, A., Klein, S., Reindl, D., 2002. Characterization of Refrigeration System Compressor Performance, in *International Refrigeration and Air Conditioning Conference*, Paper 567 (http://docs.lib.purdue.edu/iracc/567).
Myers, R., Montgomery, D., Anderson-Cook, C., 2009. *Response Surface Methodology: process and product optimization using designed experiments* (3rd ed.). John Wiley & Sons, New Jersey.
Palm B. Hydrocarbons as refrigerants in small heat pump and refrigeration systems – A review. *International Journal of Refrigeration* 2008; 31: 552-563.
Rasmussen B. Dynamic modeling for vapor compression systems—Part I: Literature review, *HVAC&R Research* 2012; 18 934-955.
Rasmussen B, Shenoy B. Dynamic modeling for vapor compression systems—Part II: Simulation tutorial, *HVAC&R Research* 2012; 18: 956-973.
Simpson J, Listak C, Hutto G. Guidelines for Planning and Evidence for Assessing a Well-Designed Experiment. *Quality Engineering* 2013; 25: 333-355.
Tanco M, Costa N, Viles E. Experimental design selection: guidelines for practitioners. *International Journal of Productivity and Quality Management* 2009b; 4: 283-302.
Tanco M, Viles E, Ilzarbe L, Alvarez M. Implementation of design of experiments projects in industry. *Applied Stochastic Models & Data Anal*ysis 2009a; 25: 478-505.
Tassou S, Lewis J, Ge Y, Hadaway A, Chaer I. A review of emerging technologies for food refrigeration applications. *Applied Thermal Engineering* 2010; 30: 263-276.